CASSELL'S

ENGINEERS' HANDBOOK

COMPRISING FACTS AND FORMULÆ, PRINCIPLES AND PRACTICE, IN ALL BRANCHES OF ENGINEERING

BY

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PREFACE.

THIS work has, by gradual accretions, grown out of the author's private note-book kept during the twelve years he was in charge of outdoor contracts for the well-known firm of Sir W. G. Armstrong & Co. Under a slightly different title it has already passed through four previous editions but has now been thoroughly revised and partly re-written to bring it up to date. The Fahrenheit degrees have been retained as being more familiar to English workmen, but the equivalent Centigrade degrees have been added in each case to suit the modern scientific engineer. It is not a mere formula book nor an ordinary student's text-book, but rather an *aide-mémoire* for those who have passed through their elementary training, and are now in practice. A reviewer has called it "as indispensable as 'Molesworth,' and more readable," and that has been the author's aim in matter and form, keeping in view also the formation of a dictionary of technical terms both current and obsolete. Special care has been taken to make the index as complete as possible.

HENRY ADAMS.

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Section 1.—*FUNDAMENTAL PRINCIPLES.*

1. THE ENGINEERING PROFESSION.

CIVIL Engineering became a recognised profession, as distinguished from Military Engineering, in 1828, when the Institution of Civil Engineers was founded under the Royal Charter of King George IV. The functions of a civil engineer as defined by the charter are to "adapt, convert, and apply the great sources of power in Nature to the use and convenience of man." All branches of engineering more or less overlap. Among the chief sub-divisions of Civil Engineering in its more limited application—that is, as distinguished from Mechanical Engineering—are railway engineering, dock and harbour engineering, water supply engineering, and sanitary engineering. Mechanical Engineering is the other main branch; among its sub-divisions are marine engineering, locomotive engineering, tool manufacture, structural engineering, electrical engineering, refrigeration, etc. Municipal Engineering enters partly into several of the sub-divisions, and is chiefly concerned with the maintenance of ways and works. The Institution of Mechanical Engineers was founded in 1847, the Society of Engineers in 1854, the Civil and Mechanical Engineers' Society in 1859, the Institution of Gas Engineers in 1863, the Iron and Steel Institute in 1869, the Institution of Electrical Engineers in 1871, the Incorporated Association of Municipal and County Engineers in 1873, the Junior Institution of Engineers in 1884, the Institute of Marine Engineers in 1889, the Institute of Sanitary Engineers in 1895, and the British Association of Waterworks Engineers in 1896.

It is reported that Prince Albert, the Consort of Queen Victoria, used to say that if he wished to *talk* about a thing he would send for an architect, but if he wanted to *do* a thing he would send for an engineer. An engineer is pre-eminently a man of action.

2. PHYSICAL SCIENCE.

Physical science is concerned with the relations of the fundamental things, time, space, and matter. Each of these is said to be *sui generis*—neither can be explained nor defined in terms of the others. Although, however, it is not possible to say what they are in themselves, it is possible to investigate the relations which subsist between them.—W. M. Hicks.

3. INDUCTION AND DEDUCTION.

The process of *induction* is a logical system of forming conclusions *from the special to the general*, by which we advance from many individual experiences to a general law; *deduction*, on the other hand, draws a conclusion *from the general to the special*, from a general law of nature to an individual case.—Haeckel.

The *inductive method* is called working *a posteriori*, the *deductive method* is called working *a priori*.

4. FORCE, MATTER, AND MOTION.

Motion is change of place. Intensity of motion is called velocity. Time is measured by motion (as the rotation of the earth). Without motion we should have no idea of time, and the only conception we have of motion is relative—i.e., the change of position of a body relatively to some other body.

Velocity is motion considered in relation to time. The velocity of a moving point is the rate of change of its position.

Force is that which produces or destroys motion, or which tends to produce or destroy it; or which alters or tends to alter its direction. Force is measured by the quantity of motion it can produce in a given mass.

Matter is that which is the subject of motion or a tendency to motion. It is the element of resistance in the sensible world. Matter always occupies space, and a given piece of matter, occupying a definite space, is called a *body*.

Pressure is a force balanced by a resistance.

"Force and matter are correlates, inconceivable apart; they necessarily involve acceptance of space and time."—Stallo's "Concepts."

"Force is not a fact at all, but an idea embodying what is approximately the fact."—W. K. Clifford.

"By the term force we are to understand muscular exertion and whatever else is capable of producing the same effects. Muscular action impeded gives us our primitive idea of force; our sense of muscular exertion is a primary one, for which we have special nerves, and it is not resolved into anything simpler. When any inanimate agent produces an effect on bodies exactly similar to that which would be produced by muscular exertion on the part of an animal, it also is said to exert force."—DR. OLIVER LODGE.

If one force alone acts upon a body, motion must ensue. Forces in equilibrium are called pressures or reactions. Pressures and resistances are the active and passive states of force; in whatever direction they are exerted they may be measured in lbs., and when exerted through any given space may be measured in foot-lbs. Force may be measured by the pressure it produces upon some obstacle, compared with gravity, or by the motion which it produces in a body in a given time. Motion may be uniform or variable: uniform motion is when a body continues to pass over equal spaces in equal times; variable motion may be uniformly accelerated, uniformly retarded, or irregular.

Philosophically, Matter and Force (Substance and Accident) are known as Mass and Motion, and the ultimate inseparables of Mass, Form and Motion together constitute Matter or the Reality of Things.

5. INDESTRUCTIBILITY OF MATTER, OR CONSERVATION OF MASS.

Matter is indestructible. The atoms composing it may enter into new combinations, or may be subjected to new conditions, but no variation can be made in the absolute quantity of matter in the universe. This was first clearly enunciated by Lavoisier, but recent hypotheses founded upon the action of radium place the constitution of matter in a new light which will be dealt with later under the subject of electricity.

6. PARTICLES, MOLECULES, AND ATOMS.

Particles are the smallest visible or tangible portions of the mass.

Molecules are the smallest physical portions of matter retaining the properties of the bulk.

Atoms are the ultimate indivisible portions of matter, probably spherical and less than the one-hundred-millionth of an inch in diameter.

Dalton enunciated the atomic theory in 1807, but the fundamental principle was propounded by Lucretius, B.C. 500, and Democritus, B.C. 470.

Sir William Thomson (Lord Kelvin) calculated that if a single drop of water were magnified to the size of the earth (8,000 miles diameter), a single molecule would appear somewhat larger than a shot and smaller than a cricket ball.

The researches of Prof. J. J. Thomson seem to show that atoms are made up of "corpuscles," or particles of matter having only one-thousandth of the mass of an atom of hydrogen. Some physicists believe these particles are not matter at all in any ordinary acceptation of the term, but are, so to speak, merely "disembodied charges of electricity."

Cauchy defined atoms as "material points without extension."

Dr. Paul Carus suggests that the world-substance consists of minute units (possessing a continuity which places them in constant relation to each other), and in its simplest form identical with what physicists call ether; two or more ether monads forming what we call an atom, various combinations of ether-monads forming the various elementary atoms.

"Whatever may be the nature of the external universe, it is our senses alone which give to it all its apparent realities."—KANT.

Polarity is believed to be inherent in every atom of the universe, material or immaterial.

7. CHEMISTRY.

Chemistry is the science which treats of the composition of matter, and of the changes produced by the action and reaction of different kinds of matter upon each other when brought into contact.

8. DALTON'S ATOMIC THEORY. (1804.)

Matter is capable of division up to a certain point only, the ultimate particles being called atoms. In the case of the same element the atoms are all alike, but in the case of different substances the atoms differ in weight and chemical properties. When chemical combination takes place between two substances the combination actually takes place between the atoms.

9. ANALYSIS AND SYNTHESIS.

Analysis is the separation of a compound body into its chemical elements.

Synthesis is the building up of a compound body by bringing its chemical constituents into juxtaposition and causing them to combine.

10. CHEMICAL ELEMENTS.

Molecules are the ultimate products of the physical division of matter, atoms the ultimate products of its chemical decomposition. The attraction between atoms is called chemical affinity, that between molecules cohesion, and that between masses gravitation.

Substances which, after subjection to all methods of analysis at present known, are not separable into two or more components, are known as chemical elements (Boyle, 1689). There are at the present time about seventy acknowledged elements, and two or three doubtful ones. According to Lavoisier an element is "the actual term whereat analysis has arrived," leaving open the possibility of further simplification.

11. MENDELÉEFF'S LAW OF PERIODICITY.

It was pointed out by Mendeléeff (1871) that if the elements are grouped in the order of their atomic weights, it will be found that nearly the same properties recur periodically throughout the entire series; the law is stated thus:—"The properties of an element are a periodic function of its atomic weight."

"By this we are compelled to regard the elements as not elementary, but as expressions of one element."—DR. SALEEBY.

12. CHEMICAL COMPOUNDS.

H_2O, as the chemical symbol for water, means that it consists of two volumes of the gas hydrogen combined with one volume (at the same temperature and pressure) of the gas oxygen. By weight, however, the proportion is 2 of hydrogen to 16 of oxygen, their atomic weights being $H = 1$, $O = 16$, or in other words, oxygen weighs sixteen times as much as an equal bulk of hydrogen.

The percentage composition of pure water is therefore by weight $H = 11 \cdot 11$, $O = 88 \cdot 89$, and, by volume, $H = 33 \cdot 33$, $O = 66 \cdot 67$.

In chemically compound substances a molecule must consist of atoms of all the component elements of the substance, in their proper relative proportions. In chemically simple substances the atoms probably exist in combination as molecules, various combinations producing the phenomena of allotropism, isomorphism, isomerism, etc.

Allotropic substances are those which exist under more than one form;

the most striking example is carbon, which occurs as diamond, graphite, charcoal and lamp-black.

Isomeric substances are composed of the same elements in the same proportions, but exhibit different properties.

Isomorphous substances are those having the same crystalline form and analogous chemical composition.

Dimorphism, as when a substance crystallises in either of two different orders of crystals, is a form of allotropy.

Amorphous substances are those which have an indefinite form, as pitch, gutta-percha, etc.

Isotropic substances are those which have the same properties in all directions, as mild steel contrasted with wrought iron.

13. MODES OF CHEMICAL ACTION.

All known instances of chemical action can be referred to one of three modes in which the re-arrangement of the atoms can take place :—

(a) By the direct combination of two molecules to form a more complex molecule, as iron Fe + sulphur S = ferrous sulphide FeS.

(b) By a mutual exchange of atoms in two molecules, as cupric sulphide 2 Cu_2S + ferric chloride Fe_2Cl_6 = ferrous chloride $FeCl_2$ + cuprous chloride 2 Cu_2Cl_2 + free sulphur 2 S.

(c) By the re-arrangement of the atoms in the molecule, as ammonium cyanate (CN) O (NH_4) + heat = urea $(NH_2)_2CO$ or CH_4N_2O.

14. ATOMICITY OR QUANTIVALENCE.

Atomicity is the measure of the number of atoms of other elements with which one atom of each element can combine. Elements whose atomicity is one are called monads; whose atomicity is two, dyads; three, triads; four, tetrads; five, pentads; and six, hexads.

Among the more common elements are :—

Monads—Bromine, chlorine, fluorine, hydrogen, iodine, lithium, silver.

Dyads—Barium, cadmium, calcium, copper, magnesium, oxygen, strontium, zinc.

Triads—Boron, gold.

Tetrads—Aluminium, carbon, cobalt, iridium, lead, nickel, platinum, silicon, tin.

Pentads—Antimony, arsenic, bismuth, nitrogen, phosphorus.

Hexads—Chromium, iron, manganese, selenium, sulphur, tungsten, uranium.

A knowledge of atomicity enables graphic formulæ of chemical compounds and combinations to be built up, and gives a clearer view of the interaction of chemical processes.

15. ACID, BASE, ALKALI, AND SALT.

An *acid* may be defined as a body containing hydrogen, which hydrogen may be replaced by a metal (or group of elements equivalent to a metal) when presented to it in the form of an oxide or hydrate.

A *base* is a compound, usually an oxide, or hydrate, of a metal (or group of elements equivalent to a metal), which metal (or group of elements) is capable of replacing the hydrogen of an acid when the two are placed in contact.

An *alkali* is a base of a specially active character, soluble in water, to which it imparts a soapy taste and touch.

A *salt* is the body produced when an acid and base react upon each other so that the hydrogen of the acid is replaced by the metal of the base.

Example :—

Acid.		*Alkaline base.*		*Salt.*		*Water.*
H_2SO_4	+	CaO	=	$CaSO_4$	+	H_2O
Sulphuric acid		Calcium oxide (Quick lime)		Calcium sulphate		Water.

—W. Jago.

16. PHYSICAL PROPERTIES OF MATTER.

Essential properties :—

Impenetrability, two bodies cannot occupy the same space at the same time.

Extension, the fact of occupying space, expressed by the three dimensions of length, breadth, and thickness.

Figure or form, presenting a definite shape at a given instant.

Accessory properties :—

Divisibility, one grain of iodide of potassium dissolved in 480,000 grains of water, when mixed with a little starch, will tint every drop of the fluid blue on the addition of a solution of chlorine.

Flexibility, the property of permitting change of shape without disintegration or fracture, as indiarubber.

Tenacity or toughness, a steel wire may bear a direct load of 100 tons per sq. in. without failure.

Brittleness depends largely upon molecular arrangement, steel when heated and suddenly cooled in water may be very brittle, but this condition may be removed by again heating and cooling slowly.

Elasticity is the property of recovering its shape after distortion, a bent spring tends to regain its original form immediately the distorting pressure is removed. In a perfectly elastic body the force of restitution would be equal to the impressed force, but no body is perfectly elastic.

Malleability is the property of allowing the shape to be changed by forging or hammering without severance of the constituent molecules. Soft charcoal iron, gold and lead are typical of malleable metals.

Ductility is the capability of being drawn into wire. Platinum, silver, iron, and copper are particularly ductile metals.

Hardness may be associated with brittleness, but not necessarily so. Hard wrought-iron is more tenacious than soft iron, but breaks more readily under a sudden blow. The diamond is the hardest substance known.

Conditions:—

Solid, viscous, liquid, gaseous, depending upon the comparative intensity of the molecular forces of attraction and repulsion, and influenced largely by temperature.

17. SOLIDS, LIQUIDS, AND GASES.

In solids the molecules are relatively fixed, in liquids they are coherent but not fixed, in gases they are repellent to each other. Hence solids press downwards only, liquids press downwards and sideways, gases press in all directions.

The pressures are the effects of gravity only, when the substances are unconfined.

18. MOLAR AND MOLECULAR MOTION.

Molar motion is the motion of masses in contradistinction to the motion of molecules. It expresses the motion of a body as a whole.

Molecular motion. The molecules of all bodies are in a state of continual agitation. The hotter a body is the more violently are its molecules agitated.

In solids the path described by a given molecule is limited and confined to a very small space.

In liquids a molecule is unlimited in its motion and may penetrate to any part of the space occupied by the liquid.

In gases the molecules move with great velocity in straight lines, and in all directions. They therefore diffuse rapidly, the lighter gases more so than the heavier.—CLERK MAXWELL.

19. ATTRACTION OF COHESION AND ADHESION.

Attraction of cohesion is the molecular attraction between the particles of the same body, or *cohesion* is the property of mutual attraction by which the particles or molecules of the same body maintain their relative positions, called also tenacity.

Attraction of adhesion is the physical attraction of the particles of dissimilar bodies in opposition to the force of cohesion, or *adhesion* is that property by which one body remains in contact with another, either by the exclusion of air from closely-fitting surfaces, or by the cementing properties of other substances interposed.

Capillary attraction is a form of adhesion, and the term capillarity includes all effects depending upon the adhesion or repulsion between fluids and solids. Capillary attraction determines the ascent of a liquid in capillary (or hair-like) tubes, and the absorption of liquid by porous bodies.

20. LAWS OF CAPILLARY ATTRACTION.

(1) The elevations and depressions are different with different liquids.

(2) With the same liquid the elevation or depression varies inversely as the diameter of the tube.

Note.—Water and liquids that wet the tube are raised, mercury is depressed.

With a tube $\frac{1}{50}$ in. diameter water will rise about $2\frac{1}{2}$ in.; with a tube $\frac{1}{100}$ in. diameter water will rise about 5 in.; with a tube $\frac{1}{200}$ in. diameter water will rise about 10 in.

21. NEWTON'S LAW OF UNIVERSAL GRAVITATION.

All bodies whatever attract each other with a force proportional directly to their masses and inversely to the squares of the distances between them. "The reason of these properties of gravity," he said, "I have not as yet been able to deduce; and I frame no hypotheses."

By some writers this statement is divided into two laws of gravitation, thus—

First law. Every body or substance in the universe attracts every other body with a force proportionate to its mass.

Second law. Bodies attract each other inversely as the square of the distance between them.

22. FORCE OF GRAVITY.

Gravity, or the attraction one body has for another, being proportional to the mass of the body, that of the earth practically overwhelms all others. The direction of attraction is towards the centre of the mass, hence, under the action of gravity, all bodies tend to fall towards the centre of the earth.

Accelerating Force of Gravity, or *Acceleratrix of Gravity*, is the velocity in one second imparted to bodies falling near the surface of the earth.

For example	lat. 45° 0′	= 32·1695 ft. per sec.
Paris	lat. 48° 50′	= 32·1819 „
Greenwich	lat. 51° 29′	= 32·1912 „

$\therefore$ say $g = 32{\cdot}2$.

23. DENSITY, MASS, AND WEIGHT.

Density is the degree of condensation of matter in a body; or it is the quantity of matter, or units of mass, in a unit of volume; $\therefore$ density = mass ÷ volume.

Mass is the quantity of matter in a body of any volume or temperature, and is constant at all heights and in all latitudes = density × magnitude.

Masses of different substance are equal when the same force acting upon them for the same time produces the same velocity.

Weight is the mass × force of gravity, which is only constant at the same level and same latitude, $\therefore$ $W \propto m\,g$, and $m = \dfrac{W}{g}$.

"$\dfrac{W}{g}$ is the *inertia* of a body and usually, but unwisely, called its *mass*." —Perry.

"The mass of a certain lump of platinum marked 'P.S. 1844, 1 lb.' deposited in the Standards Department of the Board of Trade at Westminster is the English *unit of mass*, and is called the pound avoirdupois."—Glazebrook.

Weight means the tendency towards the earth's centre possessed by all bodies, therefore the weight of the standard lb.-mass is indefinite unless the place where it is to be weighed is stated, as at sea level at Greenwich. —DUNCAN.

The weight is proportional to the mass, but varies inversely as the square of the distance from the earth's centre. The weight of a body is the resultant of its gravity towards all other bodies of the universe compounded with its centrifugal force.

Mass is independent of weight. A body carried from the equator to a pole remains unchanged as to mass, but gains $\frac{1}{2}$ per cent. as to weight.

A body weighs at sea-level $\frac{1}{295}$ less at the equator than in London; due partly to centrifugal force, and partly to difference in distance from centre of gravity of the earth—i.e., its centre.

A body weighing 200 lbs. at equator weighs 201 lbs. at poles; $\frac{1}{3}$ of the increase is due to shorter radius to centre of earth, and remainder due to absence of centrifugal force.

The effect of centrifugal force at the equator is $\frac{1}{289}$ of the attraction of gravity.

The French use the word *poids* as meaning the quantity weighed out in a balance, and *pesanteur* as the force of the attraction of the earth on the quantity. The English use the word weight for both.

24. PLUMB LINE.

A plumb line is supposed to hang vertically—i.e., to point to the earth's centre, but owing to the centrifugal force caused by the earth's rotation a plumb line in the latitude of London deviates southward by $\frac{1}{594}$ of its length. —TOMLINSON'S "MECHANICS."

25. SPECIFIC VOLUME AND DENSITY.

Specific volume is the number of cub. feet to the lb. weight.

Specific density is the number of lbs. in a cubic foot. By other writers, this would be merely density, specific density being taken as the density of a substance compared with that of a standard, water being taken as the standard for solids and liquids.

26. SPECIFIC GRAVITY.

The specific gravity of a body is the ratio of its density to that of some standard substance, generally water or air.

The standard for solids is pure distilled water at 60° F. (15·5° C), weighing 1,000 oz. per cub. foot = say $62\frac{1}{2}$ lbs.

The standard for gases and vapours is atmospheric air at 60° F., 30 inches bar., weighing 31 grains per 100 cub. inches = ·07 lbs. per cub. foot. $\frac{\text{wt. cub. ft. water}}{\text{wt. cub. ft. air}} = \frac{62{\cdot}32}{0{\cdot}07} = 890{\cdot}3$, Sp. gr. water as unit × 890·3 = sp.gr. air as unit.

Let W = weight of substance in air.
W_1 = " " water.
V = volume.
S = specific gravity of substance.
w = weight of unit of standard.
w_1 = " " substance.

Then for laboratory purposes the specific gravity of a substance is the ratio of its weight to the weight of water it displaces when immersed therein,

$$S = \frac{W}{W - W_1}.$$

For practical purposes the specific gravity of a body is generally considered as taking account of the body as a whole, and not only of the material particles of which the body is composed—e.g., coke, cement, brickwork, etc. Specific gravity is then the ratio of the weight of the material to that of an equal bulk of water.

$$W = VSw, \quad S = \frac{W}{Vw}, \quad S = \frac{w_1}{w}.$$

Upon this view the definition becomes "specific gravity is the measure of the weight of a substance in contradistinction to density, which is the measure of its mass."—F. Wood.

By the former rule the specific gravity of Portland cement in fine powder would be about 3·1, and by the latter about 1·3.

27. SPECIFIC GRAVITY OF VARIOUS SUBSTANCES.

Water being 1.

Cork	0·240	Elm wood	0·671
Poplar wood	0·383	Walnut "	0·671
Fir "	0·500	Alder "	0·800
Cedar "	0·561	Ash "	0·845
Willow "	0·585	Beech "	0·852
Lime "	0·604	Pumice stone	0·915

27. SPECIFIC GRAVITY OF VARIOUS SUBSTANCES
(continued).

Water being 1.

Oak	0·925	Cast iron	7·207
Lignum vitæ	1·330	Tin	7·285
Chalk	1·386	Wrought iron	7·788
Ivory	1·826	Steel	7·816
Stock brick	2·000	Brass, cast	8·396
York stone	2·416	„ wire	8·544
Portland stone	2·496	Copper, cast	8·788
Aluminium	2·615	„ wire	8·879
Granite	2·654	Nickel	8·800
Slate	2·672	Bismuth	9·882
Marble	2·717	Silver	10·474
Crown glass	2·488	Lead	11·445
Flint „	3·329	Mercury	13·596
Antimony	6·860	Gold	19·358
Zinc	6·862	Platinum, forged	21·837

Approximately :—Wt. lbs. per cub. ft. = sp. gr. × 62·32.

Note.—The values will vary somewhat with different samples.

28. UNITS OF FORCE.

A force of P pounds means a force equal to the weight of a mass containing P pounds.—GLAZEBROOK.

A poundal, absolute unit of force, British kinetic unit, or *Gaussian unit,* is that force which acting for unit time would impart unit velocity to unit mass. If 1 lb. = unit mass, 1 second = unit time, 1 foot per second = unit velocity, then force in poundals = pressure of $\frac{1}{g}$ lb.

The *metrical absolute unit of force* is the force that, acting on the mass of one cubic centimetre of water (mass of gramme) at maximum density, 4° C. (39·1° F) for one second, generates in it a velocity of one centimetre per second. This is also called the *dyne*, or *metrical kinetic unit.*

British gravitation units of force, or lbs. ÷ 32 = British absolute units of force, or poundals.

The F.P.S. or foot-pound-second system of gravitation units is used by engineers; the C.G.S. or centimetre-gramme-second, or metric system, by physicists and mathematicians.

29. FUNDAMENTAL UNITS.

	(s) Space	(m) Mass	(t) Time	(g) Gravity
Metric system	Centimetre	Gramme	Second	981
British system	Foot	Pound	Second	32·2

Velocity $= \frac{\text{space}}{\text{time}} = \frac{s}{t}$. Acceleration $= \frac{\text{velocity}}{\text{time}} = \frac{s}{t^2}$.

Momentum $=$ mass $\times$ velocity $= \frac{m \times s}{t}$.

Force:

$=$ Momentum generated in unit time $= \frac{ms}{t} \times \frac{1}{t} = \frac{ms}{t^2}$.

$=$ Acceleration produced in unit mass $= \frac{s}{t^2} \times m = \frac{ms}{t^2}$.

Energy (work):

$=$ Force $\times$ space $= \frac{ms}{t^2} \times s = \frac{ms^2}{t^2}$.

30. WORK AND ENERGY.

Work may be defined as a continued motion accompanied by a continuous pressure, = weight × space passed through vertically; or pressure exerted × space passed through in any direction. Briefly, *Work* is done when *Resistance* is overcome.

A unit of work, U, is the power expended when a pressure of 1 lb. is exerted through a space of 1 foot = 1 foot-lb. The amount of work performed in overcoming a given resistance through a given space is independent of the time occupied.

Power is the time-rate of expending energy in doing work. Unit power is unit work done in unit time, or 1 foot-lb. per second.

A horse-power (Jas. Watt) is the exertion of 33,000 units of work or foot-lbs. in the period of 1 minute.

Energy (Dr. Young) in mechanics means capacity for moving against a resistance or performing work, and is measured in foot-lbs.

Potential energy (Rankine), *Statical energy* (Thomson), *Sum of the tensions* (Helmholtz), *Energy of position* or *Positional energy*, is the product of the effort or pressure into the distance through which it is capable of acting.

Actual energy (Rankine), *Kinetic energy* (Thomson and Tait), *Dynamic* *energy* (Tyndall), or *Accumulated work* of a moving body, is the product of the mass of the body into half the square of its velocity, or the weight of the body into the height from which it must fall to acquire its actual velocity.

$$U = \tfrac{1}{2} m v^2 = \frac{W v^2}{2 g}.$$

Work done in raising a body of materials (as in building a house) = work done in raising whole weight to height of centre of gravity.

31. VIS VIVA AND INERTIA.

Vis viva (Leibnitz), or *Energy of motion* of a moving body, is the product of the mass of the body into the square of its velocity, or double the actual energy $= m v^2 = \frac{W v^2}{g}$, the units of work being $= \frac{W v^2}{2 g}$;

The vis viva of a body measures the whole effect which will be produced before the velocity is destroyed, thus the penetration of bullets will vary as $m v^2$. *Work* depends upon the principle of vis viva, but to compare with other units the unit of work is made $\frac{1}{2} m v^2$.

Inertia, sometimes called *vis inertiæ* or force of inactivity, implies the absolute passiveness of matter, or a perfect indifference to rest or motion—i.e., any change of state must arise from the action of external force.

32. CONVERTIBILITY OF ENERGY.

All forms of energy (as light, heat, and mechanical work) are mutually convertible. They are "modes of motion," and consist of *waves*, the direction of displacement of each vibrating particle varying in each case. Actual energy of any form being once existent cannot be annihilated; it can only be transferred into some other form, or to some other matter.

Energy waves of all kinds move with the same speed, viz. 300,000,000 metres per second, or say 186,400 miles per second.—Sir W. H. Preece.

33. CONSERVATION OF ENERGY.

The total energy of any body or system of bodies is a quantity which can neither be increased nor diminished by any mutual action of these bodies, though it may be transformed into any of the forms of which energy is susceptible.—Clerk Maxwell.

The expression *persistence of force* (Herbert Spencer, 1861) means the same as conservation of energy.

The sum total of energy in the universe is constant. Descartes' doctrine that "the quantity of motion conserved in the world is always the same," took account of one constituent only of energy, and was therefore imperfect. —F. MOHR and J. R. MAYER.

The "conservation of energy" may also be taken to mean its preservation in such a form as to be available for use whenever desired; (a) due to position, as head of water; (b) to pressure, as a coiled spring; (c) to temperature, as of steam; (d) to electrical conditions, as in a battery, etc.

34. INERTIA AND MOMENTUM.

As understood by practical engineers, *Inertia* is resistance to communication of motion, *Momentum* is resistance to extinction of motion. They are equal to each other, and of opposite character.

They are compared with *Work* by ascertaining h necessary to create the v under action of g, and considering W as moved through h, giving result in foot-lbs.

$$= \frac{W v^2}{2g}, \text{ or } W h, \text{ since } h = \frac{v^2}{2g}.$$

In calculating the power exerted in moving a load, as a truck on a railway, we have the inertia overcome in reaching the velocity attained $\left(\frac{W v^2}{2g}\right)$ added to work done transporting the load through the space passed over ($W \mu s$).

In coming to rest the inertia is given up again as momentum. The value of the momentum is irrespective of the distance in which the velocity was acquired; its effect depends entirely upon the distance in which it is expended.

35. GALILEO'S LAW OF INERTIA.

A material point, when once set in motion, free from the action of an extraneous force and left wholly to itself, continues to move in a straight line so as to describe equal spaces in equal times. This is also Newton's "First Law of Motion."

36. D'ALEMBERT'S PRINCIPLE, OR THEOREM.

"In whatever manner several bodies change their actual motions, if we conceive that the motion which each body would have in the succeeding instant, if it were quite free, is decomposed into two others, of which one is the motion which it really takes in consequence of their mutual actions, the second must be such that if each body were impelled by this force alone (that is, by the force which would produce this second motion), all the bodies would remain in equilibrio."

This is evident; for if these second constituent forces are not such as would put the system in equilibrio, the other constituent motions could not be those which the bodies really take in consequence of their mutual action, but would be changed by the first.—GREGORY'S "MECHANICS."

37. MOMENTUM.

Pressure (f) applied to a body of given mass (m) free to move, and continued for some definite time (t), causes motion at a certain velocity (v).

$$v \propto f\,t, \qquad f\,t = m\,v, \qquad f\,t = \frac{W\,v}{g}, \qquad f = \frac{W\,v}{g\,t},$$

or the effect varies inversely as the time occupied, and directly as the mass or weight moved and the velocity of movement.

When the body is already moving with the velocity (v) and it is increased to (v_1), then

$$f\,t = m\,v_1 - m\,v. \qquad s = \tfrac{1}{2}\,(v + v_1)\,t.$$

Momentum or *Quantity of motion* (Descartes, Newton) = mass × velocity, and represents the constant force which acting for one second would stop a moving body = $m\,v$. A mass in motion, having momentum = $m\,v$, will, after impact with mass m^1 at rest, have a resulting velocity of $v^1 = \dfrac{m\,v}{m + m^1}$ or $m\,v = (m + m^1)v^1$.

Moving force, or the *Moving quantity of a force*, is the momentum generated in one second.

The term momentum has been applied indifferently to express the quantity of motion existing in a body and its striking force or power of overcoming resistance, but the latter is more properly denoted by *vis viva*.

Momentum varies as the velocity, and is the measure of a given force *during a given time of action*.

Vis viva varies as the square of the velocity, and is the measure of the force *acting through a given distance*.

$$\text{Energy} = \tfrac{1}{2}\,m\,v^2 = f\,s. \qquad \text{Impulse} = m\,v = f\,t.$$

$$\text{Average force} = \frac{\frac{1}{2}\,m\,v^2}{s} = \frac{m\,v}{t}\,;$$

In its technical (workshop) use the term momentum signifies the same as actual energy or accumulated work, and is independent of time.

In old books on mechanics "the duplicate ratio of the velocity" means v^2.

Two unequal balls with velocities inversely as their masses will have equal momenta, and the same power to overturn an obstacle, but the swifter ball will penetrate a soft body further than the other, or do more *work*. They will both overcome the *same resistance in the same time*, but to have equal piercing effects their masses must be inversely as the squares of their velocities, so that their *momentum* × *velocity* may be equal.

If two bodies be acted upon by the same or equal forces for the same time the velocity and also the energy imparted to each will be inversely as its mass—e.g., rifle and ball fired from it by explosive.

38. MODERN NOTATION IN DYNAMICS.

Velocity is *time-rate of displacement*. The SECOND is taken as the unit interval [of time] and the FOOT as the unit distance. Velocity is measured in feet displacement per second, the unit of which is a displacement of 1 foot in 1 second, and this unit velocity is called a VELO. Every velocity requires an interval of time in which to produce a finite displacement however small.

When velocity is uniformly increasing, acceleration is measured by the increased velocity in feet-per-second per second, the unit acceleration is an increased velocity of 1 foot per second in 1 second, or 1 velo per second; this unit is called a CELO.

The quantity of matter in any body is called its mass, the unit mass is a pound or 1 lb. Force applied to mass produces acceleration in the direction of the force, the unit force is that force, which acting upon 1 lb. produces 1 celo, and is called a POUNDAL. The force which produces a celos in m lbs. is $m\ a$ poundals, and a mass of m lbs. with a celos has a MASS-ACCELERATION of $m\ a$ POUND-CELOS. The acceleration of any mass due to gravity (g) is 32·2 pound-celos, hence a weight of 1 lb. = 32·2 poundals, or a weight of m lbs. is a force of $m\ g$ poundals.

The MASS-VELOCITY or MOMENTUM of a body is the product of the number of lbs. in the body by the number of velos it has. A body of 1 lb. has unit mass-velocity when it has one velo; it is then said to have a POUND-VELO.

A force acting for a definite interval produces mass-velocity and is called an IMPULSE; the unit impulse is that which acting on 1 lb. produces in it

1 velo, and is called a PULSE. It has the same effect as 1 poundal acting for 1 second in producing 1 pound-velo. A pulse might be called a poundal-second.

In units of work 1 foot-pound = g foot-poundals.

A committee of the British Association proposed to call a velocity of 1 centimetre per second a KINE, and a momentum of 1 gramme moving with a velocity of 1 centimetre per second a BOLE.

39. LAWS OF MOTION.

Generally known as "Newton's Laws of Motion" (1686).

Summary:
- I. Change of state is due to external force.
- II. Every force produces its own result.
- III. Action and reaction are equal.

FIRST LAW OF MOTION (Kepler, also ascribed to Galileo). All motion is naturally rectilinear and uniform. A body at rest will continue at rest, and if in motion will continue to move in a straight line with uniform velocity, unless acted upon by some external force.

SECOND LAW OF MOTION (Galileo). If a body be acted upon by two or more forces for a given time, the effect will be the same as if the forces acted independently for the same length of time. This is the foundation of the parallelogram of forces.

Gwilt's rendering of this law is: "Every motion or change of motion in a body is proportional to the force exerted to produce it, and is in the direction of the right line in which such force acts."

THIRD LAW OF MOTION (Newton). Action and reaction are always equal and contrary in direction. When a body receives motion from another, the second body loses a quantity of motion equal to that which the first receives. When a pressure produces motion, the quantity of motion, or momentum generated in a given time, is proportional to the pressure.

40. EQUILIBRIUM

may be stable, unstable, indifferent, or mixed.

When a body is resting on another, in such a position that its centre of gravity is the lowest possible, it is in stable equilibrium—e.g., when vertically under the point upon which it is supported. When the highest possible, it is in unstable equilibrium—e.g., when vertically over point of support. When constant for any position, the equilibrium is indifferent or neutral—e.g., a sphere. When stable with regard to movement in one direction, and unstable or indifferent with regard to another direction, it is said to be in a position of mixed equilibrium—e.g., a cylinder lying on its side.

41. CENTRE OF GRAVITY

is that point in a body through which the resultant of the gravities (or weights) of its parts passes, in every position the body can assume.

The centre of gravity of two weights, or areas, A, B, placed l distance apart, will be x distance from A when

$$x = \frac{B}{A + B} l.$$

The centre of gravity x of a number of bodies in a straight line with regard to any point A at one end of line, W being the weight, and y the distance of W from A,

$$A x = \frac{W y + W_1 y_1 + W_2 y_2 + \&c.}{W + W_1 + W_2 + \&c.}$$

Bodies in same plane but not in same line must be referred to co-ordinate axes. Bodies not in same plane must be referred to co-ordinate planes.

If the centre of gravity be supported the whole body will be supported in equilibrium, but the centre of gravity is not necessarily situated in the solid portion of a body or enclosed by its surfaces; it is simply the mean central point of the mass, and in such cases the statement of support will not hold good unless the c.g. point be connected with the body.

42. CENTROID, OR CENTRE OF GRAVITY OF FORM.

Triangle. Bisect two sides, draw to opposite angles, intersection = c.g. at ⅓ height.

Trapezium. Divide into two triangles, find c.g. of each and join them. Divide into two triangles in the other direction, find c.g. of each and join them. Intersection of c.g. lines = mean c.g.

Tapered Girder Web.

t = thickness top,
b = ,, bottom,
h = height.

$$\text{Height of c.g.} = \tfrac{1}{3} h \left(1 + \frac{t}{t+b}\right).$$

Retaining Wall, vertical back.

$$\text{Height of c.g.}' = \tfrac{1}{3} h \left(1 + \frac{t}{t+b}\right).$$

$$\text{Distance of line through c.g. from face at foot} = \frac{2b}{3} - \frac{t^2}{3(t+b)}.$$

Tee-Iron ⊥. a = area lower part (flange).
A = „ upper „ (web).
d = total depth.
t = thickness.

$$\text{Height of c.g. from lower edge} = \tfrac{1}{2}\left(d + t - \frac{a d}{A + a}\right).$$

Parabola c.g. at $\frac{2}{5}$ height.

Semi-parabola c.g. at $\frac{3}{5}$ of height from apex and $\frac{3}{8}$ of base from axis.

Circular arc (as wire) Dist. from centre $= \dfrac{\text{chord} \times \text{rad}}{\text{length arc}}$

$$\text{When arc is semicircle} = \frac{2r}{\pi} = \cdot 63662\, r$$

$$\textit{Segment of circle} = \frac{\text{chord}^3}{12 \times \text{area segment}}$$

$$\text{When segment is semicircle} = \frac{4r}{3\pi} = \cdot 42441\, r = \text{approx. } \tfrac{3}{7} r$$

43. CENTRE OF GRAVITY OF REGULAR SOLIDS.

Prism or cylinder	=	$\frac{1}{2}$ height from base
Wedge or paraboloid ...	=	$\frac{1}{3}$ „ „
Pyramid or cone	=	$\frac{1}{4}$ „ „
Hemisphere or hemispheroid	=	$\frac{3}{8}$ „ „
Semicylinder or semicircle..	=	·4244 of its radius from axis $= \frac{1}{24}\pi^2 r$
Semi-annular lamina ..	=	$\cdot 4244 \dfrac{R^3 - r^3}{R^2 - r^2}$
Segment of disc or of cylinder	=	$\dfrac{\text{chord}^3}{12 \text{ area}}$ = distance from axis
Sector of disc	=	$\frac{2}{3} \dfrac{\text{chord} \times \text{radius}}{\text{arc}}$ = do.

Segment of sphere (from vertex) $= \dfrac{8 \text{ rad} - 3 \text{ versin}}{12 \text{ rad} - 4 \text{ versin}} \times$ versin

Sector of sphere (from vertex) $= \dfrac{2 \text{ rad} + 3 \text{ versin}}{8}$

Frustum of cone .. .. $= \frac{3}{4} h \dfrac{3 R^2 + 2 R r + r^2}{R^2 + R r + r^2}$

44. CENTROBARYC THEOREM (TOMLINSON).

The volume of a "solid of revolution" is equal to the area of its generating plane × the circumference described by the centroid of this plane during revolution. In other words,

a = area of semi-section parallel with axis;

r = radius or distance of c.g. of semi-section from axis;

then contents $= 2 \pi r a$.

This may be used in finding the weight of iron vases, caps and bases of columns, oval counterweights, etc., when great accuracy is desired.

45. CENTRIFUGAL AND CENTRIPETAL FORCE.

A body in motion resists any force tending to make it deviate from a straight line. When the body is rotating, the particles are constrained to move in circular paths. The inertia of the mass resists this constraint and produces tension acting outwards from the centre of rotation. The inertia is in this case called centrifugal force, and the tension centripetal force.

46. CENTRIFUGAL FORCE.

Centrifugal force is the amount required to restrain a body, or part of a body, travelling in a circle, from flying off at a tangent, and is perpendicular to the curve or tangent at each point.

The centrifugal force varies as the square of the angular velocity divided by the radius of the centre of gravity of the section on one side of axis.

Centrifugal force in absolute units $= m v^2/r$, in gravitation units $=$ W $v^2/g r$.

Centrifugal force from the earth's rotation acts in opposition to gravity at the equator, and diminishes towards the poles, where it is entirely absent.

47. CENTRE OF GYRATION

is that point in a revolving body at which, if the whole mass were collected, the accumulated work per revolution would remain the same. It is also such that the same angular velocity would be generated in the same time by a given force at any place as would be generated by the same force acting similarly on the body itself. It is measured from the centre of revolution and gives the "radius of gyration."

Circular wheel, uniform thickness ..	$= r\sqrt{\frac{1}{2}} = \cdot 7071\, r.$
Rod revolving about its extremity ..	$= l\sqrt{\frac{1}{3}} = \cdot 57735\, l$
,, ,, centre	$= l\sqrt{\frac{1}{12}} = \cdot 288675\, l.$
Flywheel rim..	$= \sqrt{\frac{R^2 + r_2}{2}}.$
Solid sphere, revolving about an axis through centre	$= r\sqrt{\frac{2}{5}} = \cdot 6325\, r.$
Wire ring, revolving about a diameter ..	$= r\sqrt{\frac{1}{2}} = \cdot 7071\, r.$
Thin circular plate ,, ,, ...	$= \cdot 5\, r.$
Thin hollow globe	$= r\sqrt{\frac{2}{3}} = \cdot 8165\, r.$
Solid sphere revolving round an external axis at c distance from centre of sphere	$= \sqrt{c^2 + \frac{2}{5} r^2}.$
Cylinder round its axis	$= r\sqrt{\frac{1}{2}} = \cdot 7071\, r.$
,, ,, parallel external axis ..	$= \sqrt{c^2 + \frac{1}{2} r^2}.$
Rectangular plate	$= \frac{d}{\sqrt{12}}.$

48. MOMENTS OF INERTIA OF SIMPLE MASSES.

About axes through the centre of gravity (m = mass).

Straight line, length $2a$, about axis perpendicular to the line	$= m\frac{a^2}{3}$
Circular wire, radius r, axis perpendicular to plane of wire	$= m\, r^2$
Ditto, about a diameter	$= m\frac{r^2}{2}$
Rectangular lamina, sides $2a$, $2b$, axis perpendicular to plane	$= m\frac{a^2 + b^2}{3}$

Rectangular lamina, parallel to side a .. $= m\frac{b^2}{3}$

Triangular lamina, sides a b c.
Same as that of three equal particles at the mid points of the sides

Ditto, axis perpendicular to plane .. .. $= m\frac{a^2 + b^2 + c^2}{36}$

Circular disc, radius r, axis perpendicular to plane $= m\frac{r^2}{2}$

Ditto, about diameter .. .. $= m\frac{r^2}{4}$

Spherical shell, radius r, about diameter .. $= m\frac{r^2}{3}$

Rectangular parallelopiped, sides $2a$, $2b$, $2c$, about axis perpendicular to b c .. .. .. $= m\frac{b^2 + c^2}{3}$

Right prism, length $2\,l$ and k k' radii of gyration of a section, about axis .. .. .. $= m\,k^2$

Ditto, perpendicular to axis .. .. .. $= m\,(k'^2 + \frac{1}{3}\,l^2)$

Circular cylinder, radius r, length $2\,l$, about axis $= m\,\frac{r^2}{2}$

Ditto, perpendicular to axis .. .. .. $= m\left(\frac{r^2}{4} + \frac{l^2}{3}\right)$

Sphere, radius r, about a diameter .. .. $= m\,\frac{2\,r^2}{5}$

Right cone, altitude a, radius of base r, about axis $= m\,\frac{3\,r^2}{10}$

Ditto, perpendicular to axis .. .. .. $= m\,\frac{3}{20}\left(r^2 + \frac{a^2}{4}\right)$

—W. M. Hicks.

49. VIBRATION AND OSCILLATION.

The vibration of a pendulum is the movement from one extreme of its position to the other. The angle formed by the extreme positions is called the *amplitude* of the vibration. The duration of a vibration is the time occupied in passing through this angle. The beat of a pendulum corresponds to one vibration.

An oscillation is a completed cycle, or two vibrations, permitting a return to the starting point. This to-and-fro movement is called by some modern writers a "swing-swang."

Length of a pendulum in London, in inches, to give any required number (n) of vibrations per minute

$$= \frac{140{,}901 \cdot 48}{n^2},$$

50. CENTRE OF OSCILLATION

is that point in a vibrating body in which, if the whole mass were collected, the body would continue to vibrate through the same angle; and such that any force applied there would generate the same angular velocity in a given time as the same force at the centre of gravity, the parts of the body or system revolving in their respective places. The distance from the point of suspension is equal to the length of a simple pendulum vibrating in the same time.

The time of vibration in seconds of a simple pendulum $= \pi \times \sqrt{\frac{\text{length in ft.}}{g}}$ $\therefore$ vibration varies as $\sqrt{l}$.

r = radius to centre of gravity.
R = " " gyration.
R_1 = " " oscillation.

$$R_1 = \frac{R^2}{r}.$$

The centre of oscillation is interchangeable with the centre or point of suspension, which then becomes the centre of oscillation.

51. PENDULUM.

The formula for a clock pendulum is derived from that for a conical or revolving pendulum, as when the amplitude is small the bob will take the same time to make one revolution as to go straight across the circle and back again. The revolving pendulum forms a cone in space, the sloping side being length of pendulum, say l, the height, say h, and the radius of base r. There will be three forces acting on the bob—the weight acting downwards, the centrifugal force acting horizontally, and the tension of the rod in the direction of the slope of cone. These forces will be in proportion to the three sides of the triangle h, r and l; hence denoting the centrifugal force by C, we get $h : r :: W : C$. Now the centrifugal force will be

$\frac{v^2}{r} \times$ mass, or $\frac{W\,v^2}{g\,r}$, where v is the velocity in feet per second; hence $\frac{h}{r} = \frac{g\,r}{v^2}$, or $h = \frac{g\,r^2}{v^2}$. Now suppose the bob makes one revolution in t seconds, then in one revolution it goes a distance $2\,\pi\,r$; then, as $\frac{\text{distance}}{\text{time}} =$ velocity, we get $v = \frac{2\,\pi\,r}{t}$; substitute this value of v in the equation for h, and we get, after cancelling the r's, $h = \frac{g\,t^2}{4\,\pi^2}$, from which $t = 2\,\pi\sqrt{\frac{h}{g}}$. When the amplitude is small, h and l may be taken as equal, so that for a clock pendulum each double beat will be done in $2\,\pi\sqrt{\frac{l}{g}}$ seconds.—M.I.C.E., BATH.

52. CONICAL PENDULUM.

h = height in feet from plane of rotation to point of suspension.
t = time of revolution in seconds.

$$\frac{\pi^2}{8} \times h = t^2$$

whence $t = 1{\cdot}11072 \sqrt{h}$, and $h = {\cdot}81057\, t^2$.

53. LONDON SECONDS PENDULUM.

The length of a London seconds pendulum to give 60 oscillations (double swings) per minute is

$$l \text{ in ft.} = \frac{g}{4\,\pi^2} = 9{\cdot}7848 \text{ ins.}$$

The length of a London seconds pendulum to give 60 vibrations (single swings) per minute is

$$\text{in ft.} = \frac{g}{\pi^2} = 39{\cdot}1393 \text{ ins.}$$

54. CENTRE OF PERCUSSION

is that point in a body revolving about an axis, at which, if it struck an immovable obstacle, all its motion would be destroyed, or it would not incline either way: it is that point with which, if the body strike against any obstacle, no shock will be felt at the point of suspension: it is the same point in a body as the centre of oscillation.

55. CENTRE OF SPONTANEOUS ROTATION.

or spontaneous gyration, is that point which remains at rest when a body is struck, or about which it begins to revolve.

56. TRANSMISSIBILITY OF FORCE.

Any force acting in a plane may be considered as acting at *any point in its line of direction.* This is called "the principle of the transmissibility of force."

57. PARALLELOGRAM OF FORCES.

If three forces act in a plane upon a free point which remains at rest, they may be represented in direction and magnitude by three lines; two of which form adjacent sides of a parallelogram and the third is equal and opposite to the diagonal.

58. EQUILIBRIUM OF FORCES.

Forces acting upon a body at rest, but free to move, are said to be in equilibrium.

59. LAMÉ'S THEOREM.

If three forces acting at a point are in equilibrium, each force is proportional to the sine of the angle between the other two.

60. SENSE OF FORCES.

The word *sense* is used to assist the word *direction* in dealing with forces; direction may be looked upon as relating to the *position of the line* and sence as relating to the *position of the arrow-head* with regard to the line, or which way the arrow faces.

61. TRIANGLE OF FORCES.

The three lines described under "parallelogram of forces" will also form a triangle, the arrow-heads pointing all the same way round—i.e., running circuitally or concurrently.

62. POLYGON OF FORCES.

When more than three forces in one plane acting upon a point are in equilibrium they may be represented in magnitude and direction by lines forming a closed polygon. More fully defined in next paragraph.

63. FORCE POLYGON.

When forces acting upon a point are represented by concurrent lines to form a polygon, open or closed, part of which may overlap other parts, it is called the *force polygon*, and when unclosed requires a closing line, representing a new force, known as the *equilibrant*, to balance the remainder. The *resultant* of a number of forces is equal and opposite to their equilibrant. The resultant of any number of forces does not depend upon the order in which they are drawn as the sides of the polygon, provided their *senses* are concurrent or circuital.

64. LINK POLYGON.

When forces act together in a system but not through one point, their leverage or turning moment through a point or pole is found by means of the *link polygon* or *funicular polygon* of the forces, which gives the position of the resultant force, otherwise unattainable.

The link polygon is obtained by drawing the force polygon and selecting any point (internal or external) for a pole, drawing lines from the pole to the junctions of the sides of the force polygon, and constructing a new polygon with sides parallel to these lines, commencing at any point on one of the force lines in its original position, and each side terminating upon meeting the direction of the next force, at which point the next side will commence. The resultant force passes through the last intersection, the direction, sense and magnitude being taken from the force polygon.

65. RESULTANT OF PARALLEL FORCES.

The resultant of two parallel forces is parallel to them, in the same plane, and (*a*) if they are in the same direction is equal to their sum, and acts in the same direction at a distance from the greater of

$$\frac{\text{distance between the forces} \times \text{the smaller force}}{\text{sum of the two forces}},$$

and (*b*) if they act in opposite directions is equal to their difference, and acts in the same direction as the greater force at a distance beyond the greater force of $\frac{\text{smaller force} \times \text{distance between forces}}{\text{greater force}}$.

66. COMPOSITION AND RESOLUTION OF FORCES.

Composition of forces takes place when a *resultant* is substituted for two or more component forces.

Resolution of forces takes place when a single force is replaced by two or more forces equivalent to it, and is the reverse of the former case.

67. MOMENTS.

The *moment* of any physical agency is the numerical measure of its importance.—THOMSON *and* TAIT.

In mechanics, a moment is generally the product of a force into a leverage.

68. MOMENT OF A FORCE.

The product of a force and the perpendicular distance of its direction from any given point, is termed the moment of the force about that point. The moment of a resultant about any point is equal to the sum of the moments of the components about that point.

The term pound-feet is preferable for moments in leverage to avoid confusion with foot-pounds of energy, both being feet × lbs. but not acting alike. Pound-feet then belongs to statics and foot-pounds to dynamics.

69. PRINCIPLE OF THE EQUALITY OF MOMENTS.

When a body is in equilibrium the sum of the moments of any number of forces that tend to turn the body in one direction is equal to the sum of the moments of any number of forces that tend to turn the body in the opposite direction.

When a body is kept in equilibrium by any number of co-planar forces, the algebraical sum of all the forces about any point in their plane is zero.

70. PRINCIPLE OF LEAST RESISTANCE.

Moseley (1833). "If there be a system of forces in equilibrium, among which are a given number of resistances, then is each of these a minimum, subject to the conditions imposed by the equilibrium of the whole."

Slightly amplified, it becomes: "If the forces which balance each other in or upon a given body or structure be distinguished into two systems, called respectively *active* and *passive*, which stand to each other in the relation of cause and effect, then will the passive forces be the least which are capable of balancing the active forces, consistently with the physical condition of the body or structure."

71. PRINCIPLE OF LEAST WORK.

"The work stored in an elastic system in stable equilibrium is always the smallest possible."—H. M. MARTIN.

72. COUPLES.

Two equal and oppositely directed parallel forces whose lines of action do not coincide, acting upon a body tend to cause rotation, and are called "a couple." The arm of a couple is the perpendicular distance between the two equal forces. The moment of a couple is one of the forces multiplied by the distance between their lines of action, or the two forces × radius to centre on which they would rotate (i.e., half the distance between their lines of action). A couple can only be equilibrated by another couple tending to cause rotation in the opposite direction and having an equal moment. The moment of a couple about any point in its plane is constant. A couple tending to turn a body in the same direction as the hands of a watch is called a positive couple, if in the opposite direction a negative couple.

The term "torque" is generally used by electrical engineers instead of "moment of a couple," "statical moment," or "turning moment."— C. H. W. BIGGS.

73. INTERCEPTS.

When forces in a plane do not meet in one point their position is defined by taking a point of origin (O) on a line (OX) cutting their directions and giving (1) the "intercept," or distance of intersection, from the point O along OX to each force, together with (2) the angle of the force contra-clockwise up to 360 degrees from OX to the direction of the force away from the intersection, and (3) the magnitude of the force to a given scale. If the intercept be behind O it will be a minus quantity.

74. CLASSIFICATION OF THE SCIENCE OF MECHANICS.

Old classification :—

MECHANICS	STATICS, treating of the equilibrium of particles and bodies.
	DYNAMICS, treating of the motion of particles and bodies.
	HYDROSTATICS, treating of the equilibrium of fluids — i.e., liquids and gases.
	HYDRODYNAMICS, treating of the motion of fluids.

—POISSON.

Modern classification :—

DYNAMICS	STATICS, as above.
	KINETICS, treating of the motion of particles and bodies with reference to the forces producing it.
	HYDROSTATICS, as above.
	HYDROKINETICS, the same as hydrodynamics above.
KINEMATICS	treats of motion without reference to the forces producing it, and is a branch of pure mathematics. Kinematics has also been called the geometry of the path of a moving point.

Statics is the science of forces in equilibrium, or pressures.

Dynamics or kinetics is the science of forces not in equilibrium—i.e., those producing motion.

Kinematics deals with the comparison of motions with each other, without reference to their causes.

These are the three great divisions of pure, abstract, or general mechanics. —RANKINE.

Applied Mechanics is that branch of applied science which explains the principles upon which machines and structures are made, how they act, and how their strength and efficiency may be tested and calculated.—JAMIESON.

75. THEORY OF MACHINES.

Machines are mechanical arrangements for transmitting force and utilising it in a convenient manner. Power is a constant sum consisting of pressure and movement, or force and velocity, either of which may be increased with a corresponding reduction of the other. The common phrase, "what is gained in power is lost in speed," would be less liable to misapprehension if the word *pressure* were substituted for *power*.

"A machine is an appliance by means of which energy is transferred from one point to another."

76. MECHANICAL POWERS.

A *Mechanical Power* is any simple arrangement by which a small force can overcome a greater, and *Mechanical Advantage** is the ratio of the greater force to the less. The Mechanical powers are more properly called *Mechanical Elements, or Simple Machines.* They are commonly described as seven, but all referable to two of the number, thus :—

Lever :—

Wheel and axle	Modifications of the lever.
Toothed wheels	
Pulley	

Inclined plane :—

Wedge	Modifications of the inclined plane.
Screw	

77. THE LEVER, WHEEL AND AXLE, AND TOOTHED GEARING.

The Lever.—Three orders; fulcrum, weight and power, alternately between the other two, principle identical.

$$P\,x = W\,y \therefore P = \frac{W\,y}{x}, W = \frac{P\,x}{y}, x = \frac{W\,y}{P}, y = \frac{P\,x}{W},$$

or, taking weight of lever into account,

$$P\,x = W\,y + W'\,y'.$$

In bent levers the length is measured from the fulcrum on a perpendicular to the direction of the force.

Wheel and Axle.—Same principle, taking radius as leverage.

Toothed Gearing.—Ditto.

The *Stanhope lever*, as applied in the Stanhope nand printing press, for pressing the platen carrying the sheet of paper on to the inked type, consists of an arrangement of levers by which a long arm moves a short arm through

* For definition of *Mechanical Efficiency*, see later.

an arc approaching the extremity of its reach, thus moving it through a considerable space compared with the motion given to the platten and consequently magnifying the pressure.

A *toggle joint*, *genou* (knee), or *knuckle joint*, is similar in principle, and is used in one form of copying press where a screw turned by a wheel presses on to the joint between two levers making an angle with each other approaching 180 degrees; in another form one of the two levers is bent to form a long handle.

Compound levers consist of a series of two or more levers, the end of one giving pressure on the next.

78. THE PULLEY.

n = number of cords shortened by raising the weight.

$$\frac{W}{n} = P, \text{ or motion of } W : \text{motion of } P :: P : W.$$

Pulleys are sometimes divided into three systems as follows:—

First System.—Each cord has one end fixed and the other passed round a running sheave. The last cord passes over a fixed sheave.

Second System.—Sheaves contained in a pair of blocks, cord passing from strop of upper block round sheave in upper and lower block alternately.

Third System.—All cords connected at one end to load, the other end of the first passes over a fixed sheave to strop of a running sheave, second cord passes over this running sheave to strop of next, and so on. Last cord passes over running sheave to the hand. Similar to first system, but inverted.

Tension in rope over frictionless pulley put in motion by difference of two weights $= \frac{2\,W\,w}{W + w}$.

79. BLOCK-AND-TACKLE, OR PULLEY GEAR.

A *block* is the frame in which the wheels, pulleys, or *sheaves* are secured by means of the pivot, axle, or *sheave pin*. The rope or chain passing over the sheaves, or *reeved* through the blocks, is called a *fall*. A combination of blocks, sheaves and fall is called a *tackle*, the upper block is generally called the *fixed block*, and the lower one the *runner* or *running block*. A tackle containing more than one rope is called a *Spanish barton*. *Snatch blocks* are blocks containing one sheave and a movable side permitting the bend or *bight* of a rope to be inserted to change its direction or *lead*.

80. THE INCLINED PLANE, WEDGE, AND SCREW.

Inclined Plane.

$$L : H :: W : P \quad \therefore \quad P = \frac{H}{L} W.$$

Wedge.—L : t : : W : P (P being direct pressure without friction).

Percussion and friction must be considered in any practical calculation.

Screw.—R = radius of lever, p = pitch of screw.

$$2\pi R : p :: W : P, \text{ or } \frac{W}{P} = \frac{2\pi R}{p}.$$

Differential Screw.

$$2\pi R : p' - p :: W : P.$$

Hunter's differential screw obtains an extremely slow movement without employing too fine a thread, by means of the difference in pitch between two threads on the same cylinder; it is used for micrometer screws.

In any practical case the friction of screws will largely reduce their efficiency for the transmission of power, which may be as low as ·1 to ·25.

Endless Screw or Worm.

N = number of teeth in wheel.
n = ,, threads in worm.
R = radius of handle or power.
r = ,, axle or weight.

$$RN : rn :: W : P.$$

An endless screw is a coarse thread of short length formed upon an axle, and geared tangentially into a toothed wheel called a "worm wheel." When the endless screw consists of one thread, each revolution moves the wheel one tooth, and a double thread moves the wheel two teeth for one revolution of the screw.

31. STEELYARDS AND WEIGHING MACHINES.

Roman Statera.—Lever of first order, balance-weight movable.

Common Steelyard.—Similar, but with two fulcra on opposite sides of beam, and two corresponding sets of divisions.

Danish Balance.—Fixed balance weight at one end, fulcrum movable.

Common Balance or Scales.—Arms equal, substance counterpoised by equivalent loose weights.

Bent Lever Balance.—Fulcrum fixed, counterbalance constant, virtual length of arms altered by movement due to weight of substance.

Spring Balance.—Weight indicated by amount of tension or compression upon a spring.

Pooley's Weighing Machine.—System of compound levers on principle of Roman Statera.

Armstrong Crane Steelyard.—Lifting chain passing over pulley on short arm of steelyard, small weights hung to end of long arm, fractional weights coupled together with loose joints so that balance is automatically obtained when sufficient number are lifted.

Duckham's Weighing Machine.—Weight indicated by increase of pressure upon liquid enclosed in cylinder hung on lifting chain, weight being hung from piston rod.

Shapton's Hydrostatic Weighing Machine.—Similar in principle to Duckham's, but pressure induced by lifting chain passing over sheave on piston rod, instead of load being hung direct.

82. USEFUL WORK OF MEN IN FOOT-POUNDS PER MINUTE.

Working for	10 *hrs. per day.*	8 *hrs. per day.*	6 *hrs. per day.*
Raising own body ..	..	4250	..
Working treadmill ..	..	3900	..
Drawing or pushing horizontally ..	..	3120	..
,, ,, vertically ..	..	2380	..
Turning handle ..	..	2600	..
Arms and legs, as rowing ..	..	4000	..
Wheeling material on ramp ..	720	..	..
Throwing earth up 5 feet ..	470	..	..
Raising material with pulley ..	..	..	1560
,, ,, hands ..	..	..	1470
Carrying ditto on back, returning empty ..	..	..	1126

83. COMPARISON OF ANIMAL POWER

A horse, travelling at the rate of 2 miles per hour, can exert a continuous pull of 125 lbs. $= \frac{5280 \times 2 \times 125}{60} =$ 22,000 ft.-lbs. per minute.

Horse	.. ..	22,000	ft.-lbs. per minute.
Mule = $\frac{2}{3}$ horse	.. .. ..	14,667	ft.-lbs. per minute.
Ass = $\frac{1}{5}$,,	.. .. ..	4,400	,, ,,
Man = $\frac{1}{10}$,,	.. .. ..	2,200	,, ,,

84. MEASUREMENT OF VELOCITY.

$$\text{Uniform velocity} = \frac{\text{displacement}}{\text{time}}, \text{ or } v = \frac{s}{t}.$$

Variable velocity is measured by the velocity at given instants.

$$\text{Acceleration} = \frac{\text{change of velocity}}{\text{time required}}, \text{ or } A = \frac{V - v}{t}.$$

85. FORMULÆ FOR FALLING BODIES.

h = Height of fall in feet.
v = Velocity in feet per second.
g = Force of gravity or acceleratrix of gravity in feet = 32·2.
t = Time of fall in seconds.
H = Highest point reached in feet.
T = Time to reach ditto.
V = Velocity imparted otherwise than by gravity.

Falling from Rest.

$$h = \frac{g t^2}{2} = \tfrac{1}{2} v t = \frac{v^2}{2 g},$$

$$v = g t = \frac{2 h}{t} = \sqrt{2 g h},$$

$$t = \frac{v}{g} = \frac{2 h}{v} = \sqrt{\frac{2 h}{g}},$$

Thrown Downward.

$$h = V t + \tfrac{1}{2} g t^2,$$

$$v = V + \sqrt{2 g h} = V + g t,$$

$$t = \frac{2 h g + V^2 - V v}{g v}.$$

Thrown Upward.

$$h = V t - \tfrac{1}{2} g t^2 = \frac{V^2}{2 g}, \qquad H = \frac{V^2}{2 g},$$

$$v = V - \sqrt{2 g h} = V - g t, \qquad V = \sqrt{2 g H},$$

$$t = \frac{2 h g + V^2 + V v}{v g}, \qquad T = \frac{V}{g}.$$

Thrown Horizontally.

$$h = \tfrac{1}{2} g t^2 = \frac{v^2}{2 g}, \qquad d = V t,$$

$$v = g\,t = \frac{2\,h}{t} = \sqrt{2\,g\,h}, \qquad I = \sqrt{V^2 + v^2},$$

$$t = \frac{v}{g} = \frac{2\,h}{v} = \sqrt{\frac{2\,h}{g}}, \qquad \tan a = \frac{v}{V}.$$

d = horizontal distance reached.

I = velocity of impact.

a = angle of impact from horizontal.

Every uniformly accelerated motion acting freely is subject to similar laws; but it must be understood that these are theoretical formulæ—i.e., only true for bodies falling in vacuo. For precise calculations of bodies falling in the air, the weight of the body must be taken into account, the diminution of the weight due to the upward pressure of the air, and the resistance offered by the air to the passage of the body.

If W = weight of body,
w = weight of equal bulk of air,
k = retardation due to air resistance (varying approximately as v^2),

then g will become $= \dfrac{W - w}{w}\,g - k$.

"The resistance which a body suffers from the fluid medium through which it is impelled depends upon the velocity, form, and magnitude of the body, and on the inertia and tenacity of the fluid." . . . "We must carefully distinguish between *resistance* and *retardation*. Resistance is the quantity of motion, retardation the quantity of velocity, which is lost; therefore, the retardations are as the resistances applied to the quantities of matter; and in the same body the resistance and retardation are proportional."—Gregory's "Mechanics."

Retardation due to falling through a medium—"Under such circumstances the body is not effectively urged with the whole force of gravity, but it is equal to the force of gravity minus gravity $\times \dfrac{\text{spec. grav. medium}}{\text{spec. grav. body}}$."—Warr's "Statics and Dynamics."

Retardation due to atmospheric resistance appears to be about

$$c\,a \times \frac{\text{wt. equal vol. air}}{\text{wt. body}} \times \frac{v^2}{2\,g}.$$

a = area of surface presented to air.

c = coefficient due to shape of surface.

86. ATTWOOD'S MACHINE.

Attwood's machine was devised in 1784 for ascertaining the accelerating force of gravity by means of a partially balanced load falling at a reduced speed.

W = weight of each of the permanent loads.

w = plus weight put on descending load, in same units.

s = space in feet passed over by system in 1 second (synchronised with pendulum).

a = acceleration produced in feet per second per second.

g = force of gravity in feet per second per second.

$$a = \frac{w}{2\,W + w} \times g, \qquad s = \tfrac{1}{2}\,a$$

$$g = \frac{2\,s\,(2\,W + w)}{w}.$$

Example :— $s = \cdot 5$ ft., $W = 62$ oz., $w = 4$ oz.

$$g = \frac{2 \times \cdot 5\,(2 \times 62 + 4)}{4} = \frac{128}{4} = 32.$$

87. VELOCITY UNDER CONSTANT ACCELERATION.

v = velocity ft. per sec. at commencement.

V = ,, ,, finish.

v' = mean velocity ft. per sec. throughout given period.

t = time in seconds.

s = space passed over in feet.

a = acceleration in ft. per sec. per sec.

Final velocity $V = v + a\,t$.

Mean velocity $v' = v + \frac{1}{2}\,a\,t$.

Space passed over in given time.

$$s = v'\,t = (v + \tfrac{1}{2}\,a\,t)\,t = v\,t + \tfrac{1}{2}\,a\,t^2.$$

88. GRAVITATIONAL MEASUREMENT OF DEPTHS.

t = time in seconds actually occupied by falling body.

T = time noted between dropping stone and hearing sound of it reaching bottom.

d = depth of fall in feet.

v = velocity of sound in air, ft. per sec.

g = acceleratrix of gravity.

$d = \frac{1}{2} g t^2,$

but $T—t$ = time the sound takes to rise,

$\therefore d = v (T—t),$

hence $\frac{1}{2} g t^2 = v (T—t),$

whence $t = \sqrt{\frac{2 v T}{g} + \left(\frac{v}{g}\right)^2} - \frac{v}{g}$

and $d = v \left(t - \sqrt{\frac{2 v t}{g} + \left(\frac{v}{g}\right)^2} + \frac{v}{g}\right).$

Approximately, $d = 16 T^2$.

89. CARTESIAN CO-ORDINATES

are measurements in perpendicular directions from points at given distances on two lines at right angles to each other. The distance on each line being taken as the length of the perpendicular on the other, the intersection gives a point which may be one of a series found in the same way, and producing a straight line or curve either regular or irregular.

Cartesian co-ordinates have been used by the author for testing all his experiments and tables of proportions since 1866, but they did not come into general use by engineers for this purpose until about twenty years later, and are even now (1907) not nearly so often applied as their usefulness dictates. The adoption of "squared paper" greatly facilitates the work of plotting the curves. The plotting of curves from their Cartesian co-ordinates is called "curve tracing." The space being divided by vertical and horizontal centre lines, measurements (x) on right of vertical line are +, on left —; measurements (y) above horizontal line are +, below —. Abscissæ and ordinates in geometrical drawing and curve tracing are equivalent to distances and offsets in land surveying.

In the United States diagrams of plotted curves, under the name of charts, are much used in designing all kinds of work.

90. OBLIQUE CO-ORDINATES.

When co-ordinates are not at right angles they are called *oblique*, and the angle between their two positive directions is denoted by the Greek letter ω. The position of any point P is then given as x, y, ω.

If a point O be taken as a *pole* or *origin* and OX as an *initial line*, any point P is fixed by the length of its *radius vector* r from the pole, and the angle of its direction from OX, or *vectorial angle* θ, its position being denoted by its *polar co-ordinates* r, θ.

91. GRAPHIC NOTATION AND VECTORS.

In graphic notation, $A = A_\alpha$ means line or vector A has a magnitude A and a direction making an angle α from OX, O being the point of origin. If $A = 41'_{208}{}^\circ$ the line is 41 feet long, and makes an angle of 208°, anti-clockwise from OX drawn horizontally from left to right—i.e. eastward.

The word *vector* is a general term for all directed quantities like displacements, velocities, forces, etc., which are compounded by, or subject to, the law of the parallelogram.

92. LOCUS OF A POINT.

When a point moves so as always to satisfy a given condition, or conditions, the path it traces out is called its *locus* under these conditions.

$x + y = 1$ is equation to a straight line
$x^2 + y^2 = 4$ „ „ circle.
$y^2 = 4x$ „ „ parabola.

The *equation to a curve* is the relation which exists between the co-ordinates of any point on the curve, and which holds for no other points except those lying on the curve.

An equation of the first degree contains no products, squares or higher powers of x and y, and its locus is always a straight line.

An equation of the second degree contains products, squares, or higher powers, and the locus is in general a curved line.

If three perpendiculars be drawn from the vertices of a triangle upon the opposite sides they will meet in a point called the *orthocentre* of the triangle.

93. HODOGRAPH OF MOVING POINT.

If a particle be moving in a curved path, and any fixed point O be taken from which are drawn lines parallel to tangents to the curve at any points, and of a length equal to the velocity of the particle at the respective points, the free ends of the lines will lie on a curve which is called the *hodograph* of the path. The acceleration of a point along the path is represented in magnitude and direction by the velocity of the corresponding point in the hodograph.—W. M. HICKS.

94. ORTHOGONAL PROJECTION.

The *orthogonal projection* of any line A B inclined at angle θ to X Y is A B cos θ.

95. SIMPLE HARMONIC MOTION.

If a point move in a straight line, so that its acceleration is always directed towards, and varies as its distance from, a fixed point in the straight line, the point is said to move with simple harmonic motion.

96. PARABOLA.

A *parabola* is the path traced out by a point which moves so that its distance from a fixed point called the *focus* is the same as its perpendicular distance from a fixed straight line, called the *directrix*. A line through the focus perpendicular to the directrix is called the *axis*, and the point where the curve meets the axis is called the *vertex*. The double ordinate through the focus, perpendicular to the axis, is called the *latus rectum*.

97. VARIETIES OF CURVES.

Circle, curve struck from one centre or formed by any horizontal section of cone or cylinder. A portion of a circular curve is called a circular arc. Whether the circumference or the enclosed area is intended by the term circle must be judged from the context. Equation $y = \sqrt{a x^2 - x^2}$.

Ellipse, appearance produced by circle seen at an angle between 0° and 90°, or any diagonal section of a cylinder or cone. Equation $y = \sqrt{\frac{p}{a}(a x - x^2)}$.

Hyperbola, any vertical section of a cone except a central section; the curve formed in chamfering nuts; basis of expansion curves. Equation $y = \sqrt{\frac{p}{a}(a x + x^2)}$.

Parabola, any section of a cone parallel with one side; the line of thrust for an arch under a load uniformly distributed over the span. Equation $y = \sqrt{p x}$.

Catenary, the curve in which a perfectly flexible cord would hang in vacuo;

practically the curve in which a chain hangs when supported at the two ends and carrying its own weight only; the curve in which the chains of a suspension bridge hang; an inverted catenary is the line of thrust for an arch without superincumbent weight. The usual line of thrust is a modified catenary. Equation of catenary $y = p\,(\text{hyp. log. } \frac{p + x + \sqrt{2px + x^2}}{p})$.

Helix, a spiral produced by an inclined plane wrapped round a cylinder; the outline of a screw thread or spiral staircase.

Archimedean spiral (Archimedes). Curve proceeding outwards from centre at a uniform rate for equal angles of rotation. A false but useful spiral is produced when compass curves are struck consecutively from the corners of a square.

Logarithmic spiral (Descartes). Curve proceeding outwards from the centre and making a constant angle with radius vector; called also equiangular spiral.

Cycloid (Descartes, 1615), formed by point in circumference of cylinder rolling on plane, as carriage-wheel on straight road, and terminating in cusps. A body falling down an inverted cycloid will fall through any height to the lowest point in the same time. A pendulum vibrating between cycloidal arcs will make all vibrations of whatever amplitude in the same time. Toothed racks may have the teeth cycloidal.

Epicycloid, is a cycloid traced by one wheel rolling on the outside of the circumference of another.

Hypocycloid, is a cycloid traced by one wheel rolling on the inside of the circumference of another. The teeth of wheels may be epicycloidal outside the pitch line, and hypocycloidal within it.

Curtate cycloid, is formed when the tracing point is beyond the circumference of the generating circle of a cycloid, and terminates in loops or nodes.

Prolate cycloid, formed when tracing point is within the circumference of generating circle, and terminal points are rounded.

Cissoid (Diocles) somewhat like hyperbolic expansion curve in appearance but differently generated. Equation $y = \sqrt{\frac{x^3}{a - x}}$.

Conchoid (Nicomedes) has various shapes according to the conditions of its generation.

Witch of Agnesi (Agnesi, 1750), called also the curve of pursuit; outline of antifriction pivot.

Cardioide, shape of heart wheel cam.

Lemniscate, or figure 8 curve. Equation $y = \frac{x\sqrt{a^2 - x^2}}{a}$.

Quadratrix, formed by the intersection of radii at equal angles in a quadrant, with the same number of equidistant lines perpendicular to one of the outer radii.

Given the law of motion in a curve the Differential Calculus enables the curve to be found, and given certain points in a curve the Integral Calculus enables the law of motion to be found.

98. ROLLING ON INCLINED PLANES.

In rolling a body down an inclined plane, the final velocity, omitting friction, is dependent solely on the height passed through, and will be the same as if falling freely. The average velocities will therefore be the same in descending all planes of equal height, and the times of descending will be proportional to the length.

If the diameter of a circle be perpendicular to the horizon, and chords be drawn from either extremity, the times of descent down all the chords will be equal, and each equal to the time of free descent through the vertical diameter.

99. BRACHYSTOCHRONE, OR CURVE OF QUICKEST DESCENT.

An inverted semi-cycloid with its base passing through the starting point, and its vertex passing through the terminal point, is the curve of quickest descent; a circular arc with its centre on a vertical line through the terminal point is next; and a straight line joining the extremities is the slowest although the shortest route. From whatever part of the cycloid the body commences its descent it will always occupy the same time in reaching the bottom.

100. VELOCITY OF SOUND.

Sound is a function of three independent variables, *acuteness or pitch, intensity* and *timbre.*

All sounds travel at the same velocity, being about

1125 feet per second in air at 62° F. ($16\frac{2}{3}$° C.)
1090 „ „ 32° F. (0° C.)

and 17 times faster in iron, 17 to 11 times in wood, and $4\frac{1}{2}$ times in water.

V = velocity ft. per sec.
D = density of medium.
E = elasticity of medium.

$$V = \sqrt{\frac{E}{D}} \text{ (Newton).}$$

When the temperature of air is constant the density and elasticity vary in the same proportion.

P = pressure of air in lbs. per sq. in.

$$V = \sqrt{\frac{P}{D}} \text{ (Laplace).}$$

T = temperature in degrees C.

$$T = \frac{V - 1090}{2}, \quad V = 1090 + 2T.$$

k = specific heat of air at constant pressure,
c = " " " volume,

$$\frac{k}{c} = \frac{\cdot 238}{\cdot 169} = 1\cdot 408 = \gamma$$

H = height of homogeneous atmosphere at 32° F. = 26,214 ft.
g = 32·2

Velocity of sound in air at 32° F. in ft. per sec. $= \sqrt{g \gamma H}$.

The velocity of sound is increased by a rise of temperature approximately 1 foot per second for 1° F.

101. VIBRATING STRINGS.

In a vibrating string the junction between the depression and crest of a wave, or point of no vibration, is called a *node*, the vibrating parts are called *ventral segments*. A wave consists of two ventral segments, and a *wave-length* is therefore twice the distance between two consecutive nodes.

The number of transverse vibrations per second is proportional to the $\sqrt{\text{tension}}$, inversely proportional to the length of the string, to its thickness and to $\sqrt{\text{density}}$.

N = number of vibrations
l = length of string
t = tension "
d = density "
k = thickness "

$$N = \frac{1}{kl}\sqrt{\frac{t}{d}}.$$

Length of vertical vibrating rod fixed at one end = wave length ÷ 4.

102. SOUND WAVES.

L = length of sound-wave in feet,
n = number of vibrations per second,
v = velocity of sound in feet per second,

$$v = Ln, \quad L = \frac{v}{n}.$$

103. INTENSITY OF SOUND.

I = intensity of sound,
A = amplitude of vibration or wave,
D = distance of sounding body,

$$I = A^2 = \frac{1}{D^2}.$$

About 100 ft. is the shortest distance that will give an echo, but reflection of sound may be observed as close as 10 ft., as when passing a notice board.

104. MUSICAL SOUNDS.

Musical sounds depend upon uniform and rapid vibrations, the greater the rapidity the higher the *pitch* of the note. "The middle C" on the pianoforte corresponds to 264 vibrations per sec. The relative rates of vibration of the various notes are C 8, D 9, E 10, F $10\frac{2}{3}$, G 12, A $13\frac{1}{3}$, B 15, C 16. The quality or *timbre* of a note is produced by the mixture of overtones with the fundamental note.

105. VACUUM.

Vacuum is the name given to the condition of a closed space from which the air or other gaseous fluid has been more or less completely removed. When a sensible quantity of gas or vapour remains it is called a *partial vacuum.*

An ordinary suction pump or lift pump acts by creating a partial vacuum which permits the pressure of the atmosphere to force the water through the valve into the body of the pump. It cannot raise water higher than 34 ft., as the weight of the atmosphere is insufficient to do more, but 26 ft. is, as a rule, as much as can be practically relied upon.

An air pump or vacuum pump is used to create or maintain vacuum for many purposes.

a = volume of receiver and pipe,
b = volume of pump,
n = number of strokes,
e = degree of exhaustion measured by ratio of air left in receiver to original volume,

$$e = \left(\frac{a}{a+b}\right)^n$$

106. RELATIVE VELOCITIES.

Snail crawling	0·005 ft. per sec.
Falling body	32 ,,
Race-horse	50 ,,
Fast train	90 ,,
Cannon-ball	1,700 ,,
Gun-cotton (flame)	15,000 ,,
Earth in orbit	95,000 ,,
Meteorite	250,000 ,,
Light	1,100,000,000 ,,

—PROF. DEWAR.

107. THE FASTEST MILE.

Man swimming	26 min.	52 sec.
,, walking	6 ,,	23 ,,
,, in snow shoes	5 ,,	39¾ ,,
,, rowing singly in boat	5 ,,	1 ,,
,, running	4 ,,	12½ ,,
,, on tricycle	2 ,,	49⅖ ,,
,, on bicycle	2 ,,	29⅘ ,,
,, skating	2 ,,	12⅗ ,,
Horse running	1 ,,	25 ,,
Railway train	0 ,,	40¼ ,,

—"PRACTICAL ENGINEER," 1891.

Walking record	6 min.	$19\frac{3}{5}$	sec.
(By A. T. Yeomans, Bath, August 18th, 1906)			
Bicycle record (flying start)	1 „	41	„
(By J. Platt Betts, Catford, July 20th, 1896.)			
Motor Car record	0 „	$28\frac{1}{5}$	„
(By Fred. Marriott on Florida beach, January, 1906.)			

108. ASTRONOMICAL UNIT OF DISTANCE.

The astronomical unit of linear measurement is the distance light travels in a year at the rate of 186,000 miles per second, and is equal to 5,796,000,000,000 miles. The nearest "fixed star" has a distance of 23 million million miles, or say 4 units nearly, while the next, 61 Cygni, has a distance of about 10 units, and 85 Pegasi 60 units. A parallax of 0·02 seconds represents the angle subtended by one-sixteenth of an inch at a distance of 10 miles, and yet the vast majority of so-called fixed stars are too distant to give any parallax. —J. Logan Lobley.

Parallax is the apparent change of position of an object when viewed from different places.

Section II.

VARIETIES AND PROPERTIES OF MATERIALS.

109. VARIETIES OF IRON.

Wrought Iron.—Fibrous—Tough—Soft—Ductile at high temperatures, but not fluid—Pressed in moulds at 1500° to 2000° F. (875° to 1090° C)—Welded at 2500° to 2800° F. (1390° to 1540° C)—Easily oxidised—Forged, hammered, or rolled to various shapes—Contains very little carbon.

Steel.—Fibrous to crystalline—Containing small amount of carbon may be welded, and with more carbon may be cast—Can be forged—Very tough and strong—May be tempered—Special properties due to some extent to silicon, but more to the mode of manufacture—That with much carbon used chiefly for tools, and with less carbon for boilers, bridge plates, and structural work.

Cast Iron.—Crystalline—Brittle—Fluid at high temperatures—Takes complicated shapes by casting in a mould—Contains much carbon—The various qualities known as Nos. 1, 2, 3.

110. TO DISTINGUISH WROUGHT IRON, STEEL AND CAST IRON.

If made red hot and hammered, cast iron or malleable cast iron will fly to pieces. If plunged in water while red hot, steel will harden, while wrought iron will remain soft. They are also distinguished by the grain of the fractured surface. A drop of nitric acid on bright steel will produce a black spot, while wrought iron remains bright; the darker the spot the harder the steel, because of the greater amount of carbon exposed.

These tests are for workshop use, and only indicate the broad features of difference. Some steel may contain no more carbon than some wrought iron, and even a chemical analysis would then fail to discriminate between the two materials. A bending test would probably indicate the wrought iron by its easy fracture when bent backwards and forwards, while the steel might resist fracture when repeatedly bent.

111. EFFECT OF CARBON IN IRON.

No.	*Name.*	*Percentage of Carbon.*	*Properties.*
1	Malleable iron	0·25	Is not sensibly hardened by sudden cooling.
2	Steely iron .	0·35	Can be slightly hardened by quenching.
3	Steel . .	0·50	Gives sparks with a flint when hardened.
4	,, . .	1·00 to 1·50	Limits for steel of maximum hardness and tenacity.
5	,, . .	1·75	Superior limit of welding steel.
6	,, . .	1·80	Very hard cast steel, forging with great difficulty.
7	,, . .	1·90	Not malleable hot.
8	Cast iron .	2·00	Lower limits of cast iron, cannot be hammered.
9	,, .	6·00	Highest carburetted compound obtainable.

—BAUERMAN.

112. COMMON ORES OF IRON.

Oxides :—

Magnetic Oxide, or Magnetite—from Sweden, Norway, North America, etc.

Red Hæmatite, or Kidney Ore—from Whitehaven and Ulverston.

Specular Iron Ore—is same composition, but composed of crystallised masses; found in Russia, Spain, Elba, etc.

Brown Hæmatite—differs from Red Hæmatite in having water in its composition; from Forest of Dean, Alston Moor, Northamptonshire, etc.

Carbonates :—

Spathose Iron Ore, Spathic Ore, or Iron Glance—from Northumberland and Durham.

Argillaceous :—

Clay Ironstone or Clay Band—from South Wales, Dudley, North Staffordshire, Yorkshire, etc.

Black Band Ironstone—from Ayrshire and Lanark, containing coaly impurities.

113. SOURCES OF ORE SUPPLY.

The following table gives the source of the ore supply of each district, showing how some districts are much dependent upon others for their ore supply :—

District.	Ore Supply.
Scotch Pig	Scotland—largely the blackband and clayband ores, which are very similar in character to those formerly, and to a small extent still, found in Staffordshire.
Cleveland or Yorkshire Pig	Cleveland Hills of North-East Yorkshire. These are known in market reports as "G.M.B.'s."
Lincolnshire Pig	Lincolnshire—chiefly in the Frodingham district. Several *West* Yorkshire brands of pig come under this head.
Derbyshire Pig	Formerly Derbyshire ore, but as the deposits have nearly given out, ores from Northampton, Leicester, Nottingham, and Rutland are now largely used, as well as part Lincolnshire.
Nottingham Pig	Many of these are classed with Derbyshire irons, and draw their ore supply from similar sources—*i.e.*, chiefly the South, Midlands.
Northampton Pig	Rutland, Northampton, and Oxfordshire.
Staffordshire Pig (hot blast)	Hot blast irons were formerly made from local ores, but as few of these ores are now available, most of the ore supply is now obtained from Leicester, Northampton, and Oxfordshire.

Staffordshire Pig (cold blast)	Cold blast irons are largely made from local ores, but hæmatite ores low in phosphorus are to some extent used.
Yorkshire Cold Blast Pig	There is practically no Yorkshire cold blast pig on the market, as those firms who make it use nearly all they can produce for best Yorkshire wrought iron. The ore supply is local.
Refined Pig	These are not "virgin" irons—that is, made direct from ore—but are selected pig-irons which undergo special treatment.

The above described irons come under the heading of foundry and forge pigs, but in addition to these there are hæmatites which, whilst chiefly used for the production of steel, enter more or less into the manufacture of many classes of the better qualities of iron castings. This iron can be divided into three classes: (1) made from foreign ores; (2) made from local ores; and (3) from part local and part foreign.—E. ADAMSON (*Sheffield*).

114. SCALE OF HARDNESS OF MINERALS.

1. Talc.
2. Rock-salt.
3. Calcite.
4. Fluor spar.
5. Apatite.
6. Felspar.
7. Quartz.
8. Topaz.
9. Corundum.
10. Diamond.

Each mineral in the above list can scratch those preceding and may itself be scratched by those succeeding.

In Von Mohs' scale of hardness No. 9 is sapphire.

115. ROASTING AND SMELTING.

Ore broken into pieces, mixed with coal in large heaps, and allowed to burn slowly to drive off water, carbonic acid gas and sulphur. Or roasted in a common kiln something like a lime kiln, or in a special form known as Gjers roasting kiln. The operation is called calcining or roasting.

Roasted ore, with earthy matters to form a flux, and fresh fuel to maintain heat, are smelted together in a *blast furnace,* 50 to 100 feet high, to obtain the metal from the ore. Charge consists of, say, 5 cwt. ore, 2 cwt. limestone, 5 cwt. coke, repeated every half hour, furnace being kept full. Molten metal run off every 12 hours into channels in sand, long lines called *sows*, branches three or four feet long called *pigs*. Furnace not blown out for six or seven years, unless under special circumstances.

116. CHEMICAL ACTION OF BLAST FURNACE.

The silica, alumina and lime in the ore and flux combine by the aid of heat to form a glassy slag, which floats on the molten metal and runs off near the bottom of the furnace. A small portion of the carbon combines with the iron and keeps it fluid until drawn off at the tap-hole. The remainder of the carbon of the fuel combines with the oxygen in the ore and the blast to form carbonic oxide (burning with blue flame) and carbonic acid (uninflammable) which pass out at the top.

Carbonic oxide (carbon monoxide) will withdraw oxygen from ironstone at temperatures over 500° F. (260° C.).

Carbonic acid (carbon dioxide) will take up more carbon to form carbonic oxide at temperatures exceeding 1000° F. (540° C.).

117. UTILISATION OF BLAST-FURNACE GASES.

With an effluent blast-furnace gas of 120 British thermal units in value per cub. ft., 600 to 700 cub. ft. are requisite for developing 1 h.p. for one hour by combustion under boilers, but employed in a gas-engine the same volume will develop from 5 to 7 i.h.p., the working pressure being 60 to 70 lbs. per square inch.

The air blast is heated to from 1,200° F. to 1,400° F. (650° to 760° C.), and supplied under a pressure of from 5 to 10 lbs. per square inch above the atmosphere. Where coke is the fuel employed 170,000 to 180,000 cub. ft. of gases measured at 60° F. (15·55° C.) are given off for each ton of coke. With raw coal as fuel the gases fall to about 130,000 cub. ft., but the proportion of combustible gas is higher. The percentage of carbonic oxide by volume is from 25 to 35 per cent., hydrogen about 2 per cent., and marsh gas from ½ to 2 per cent.

118. PIG IRON.

Hot-blast and Cold-blast.—Named from the temperature of the blast used in smelting the ores. Hot-blast (Neilson, 1829) generally quicker and more economical, requiring only 30 cwt. of coke per ton of metal instead of 40 cwt., but the metal is not considered to be so strong. Difficult to distinguish the two varieties, but, other circumstances being equal, hot-blast iron has rather a finer grain, duller fracture, with sometimes patches of coarse grains, and usually more impurities. Increasing the blast or reducing the supply of fuel makes the iron whiter, harder and less suitable for re-melting, but better for conversion into wrought iron or steel. Temperature of the blast usually from 600° F. to 1000° F. (315 to 540° C.), but higher temperatures have been attained in the Cleveland district. The pig-iron made by the Farnley Iron Co. is cold blast only.

119. ANALYSES OF PIG IRONS.

Description.	*C*	*Mn*	*Si*	*S*	*P*
Foundry—Glengarnock . .	3·677	1·777	2·40	·602	1·010
Bessemer—Workington . .	4·941	·065	1·572	·038	·007
Swedish—Lily . . .	4·603	1·276	·070	·006	·015

SPIEGELS.

Ebbw Vale . . .	3·734	8·958	·215	·064	·088
J. Brown & Co.'s. .	4·675	25·12	·445	·002	·056
Ferro-manganese . .	6·588	65·13	·187	·081	·059

120. CLASSIFICATION OF PIG IRON.

Bessemer Iron.—A variety of pig iron made from hæmatite ores for conversion into steel; very free from impurities.

Foundry Iron.—All pig iron having grey fracture and large proportion of uncombined carbon; produced under high temperature and full supply of fuel.

Forge Iron.—White pig iron, almost free from uncombined carbon, suitable for conversion into wrought iron; produced with low temperature or insufficient fuel, frequently run from blast furnace into iron moulds, rendering it brittle for ease in breaking up.

121. REFINING.

Refining is a combination of chemical and mechanical processes by which pig iron is deprived of its impurities previously to its conversion into wrought iron.

It consists simply of melting the pig iron with coke or charcoal in an open hearth or "refinery furnace," supplied with an air blast so as to impinge on the melted metal and furnish an oxidising atmosphere. This carries off a portion of the carbon, and at the same time removes a portion of the impurities, particularly silicon, in the form of slag. The melted metal, having lost some of its carbon, is then poured into a cast-iron trough lined with loam, kept cold by water circulating below, and the sudden chilling has the effect of converting soft grey iron into hard silvery-white metal, the carbon which formerly existed in the shape of graphite entering into perfect chemical combination. By this change the fluidity of the iron is reduced, and the subsequent puddling process facilitated.

For common wrought iron the pig metal goes direct to the puddling furnace without undergoing the intermediate refining.

The loss of weight in refining crude iron averages 10 per cent., and the weekly production of a refinery furnace is from 80 to 160 tons.

122. PUDDLING.

Dry Puddling (Cort, 1784) is the process of obtaining wrought iron by burning the carbon out of refined cast iron in a reverberatory furnace. The oxygen of the air, at the high temperature employed, combines with the carbon to form carbonic oxide gas, which escapes; and with the silicon to form silica, which runs off as slag. In hand-puddling the mass is stirred about until it is of sufficient tenacity to be lifted out of the furnace in balls or blooms of 60 to 80 lbs. each; a 5 cwt. charge takes about two hours to work off. In Danks' rotary furnace the revolution of the furnace effects mechanically the same purpose as the hand labour.

If the operation be stopped before the carbon is all removed, puddled steel is obtained.

Wet Puddling or *Pig-boiling* is the more modern process, in which grey unrefined pig iron is converted direct. The bed of reverberatory furnace is lined with broken slag, cinder, scale, etc., fused together, and over these a fettling of soft red hæmatite or "puddlers' mine" is placed. The stages of the puddling process are—(1) graphitic carbon converted into combined carbon, and silicon partly oxidised by roasting and melting; (2) metal drawn from sides, and mixed with that in centre; (3) metal "boiled" for twenty minutes, impurities being oxidised by agitation of the mass; (4) pasty metal "balled" and re-balled, ready for shingling.

123. SQUEEZING, SHINGLING AND ROLLING.

After removal from the puddling furnace, at a welding heat, the "ball" is put under a heavy trip hammer, a rotary squeezer, or a hydraulic press, to remove the slag and impurities from the spongy mass, and to solidify the metal forming a "bloom." It is then passed through chilled rolls, and drawn down to a convenient size for handling, and is then called a "puddled bar."

124. MOLECULAR CONDITION OF IRON.

Mr. Jeremiah Head, during a discussion that took place after the reading of a paper "On Reversing Rolling Mills," at the Cleveland Institution of Engineers, said: "I am inclined to think that the term fibrous, when applied to the structure of wrought iron, is really inappropriate and misleading. A truly fibrous material, such as wood, resembles wrought iron only in the appearance of the fracture. But the fibres of wood are not at all ductile, and therefore its appearance, when broken, arises from the broken fibres, of which it is built up, becoming apparent. But the similar appearance of a fractured piece of wrought iron arises from the ductility of the molecules of iron, the apparent fibres having been made for the first time in the act of bending. If we could see into the iron before bending, we should probably find it quite innocent of any fibres, however ductile the quality."

Corroded bolts, however, frequently look like a bundle of fibres, perhaps from the removal of the scale which was rolled into a reedy form with the metal.

125. CRUDE WROUGHT IRON.

Puddled Bar is the material after passing a bloom through the first series of rolls.

Merchant Bar is made by cropping, piling, re-heating, welding and rolling puddled bar.

Single, double and treble best signifies the number of times the material is again put through these processes.

126. QUALITIES OF WROUGHT IRON.

(*a*) Iron easily worked hot, and hard and strong when cold, used for rails.*

(*b*) Common iron, used for ships, bridges, and sometimes for shafting.

(*c*) Single, double, and treble best iron, from Staffordshire and other parts where similar qualities are made. The single or double best is used for boilers. Double and treble best are used for forging.

(*d*) Yorkshire iron, from Lowmoor, Bowling, or other forges where only fine qualities are made. The best Yorkshire iron is very reliable, and uniform in quality. It is used for tyres*, for difficult forgings, for furnace plates exposed to great heat, for boiler plates which require flanging, etc.

(*e*) Charcoal iron, very ductile and of best quality.—UNWIN'S "MACHINE DESIGN."

127. SINGLE AND DOUBLE SHEET IRON.

Iron sheets, up to No. 20 B.W.G. inclusive, are called *singles;* Nos. 21 to 24, *doubles;* Nos. 25 to 28, *lattens;* and above No. 28, *extra lattens*. Singles are less than $\frac{3}{16}$ inch in thickness, and when the sheets are less than about $\frac{1}{30}$ inch they are too thin to be rolled separately, therefore two are placed together.

Terne plates are thin sheet iron used for making tin plates.

128. IRON ROLLING MILLS.

Weight of piles to produce boiler plates, allowing for waste in the furnace and waste in shearing :—Add for every $\frac{1}{16}$ inch in thickness 1·06 lb. to every square foot of plate over and above the finished weight.

* Rails and tyres for railway rolling stock are now usually made of a moderately hard mild steel; the Bowling Iron Co. ceased on 13th March, 1896, to manufacture rolled wrought iron.

To make boiler plates from slabs, allow one-third more than the weight of finished plate ; and for re-heating and doubling, 5 lbs. to every 100 lbs. more than one-third must be allowed.

For plates narrower than 20 inches an allowance of 10 lbs. extra to every 100 lbs. must be made for greater waste from shearing.

To make sheets from piles varying from 11 to 30 wire gauge, add one-half more than the finished weight, which is sufficient for waste and shearing upon both bar and sheet.

For merchant bars of all kinds, which are rolled from the pile in one heat, one-fifth more than the finished weight is sufficient to allow for waste and cropping.

As regards wages, the ironworker is paid per ton long weight. What is termed long weight is 2,400 lbs. to the ton.—"MECHANICAL PROGRESS."

129. DEFECTS IN WROUGHT IRON.

Cold-shortness is produced by the presence of a small quantity of phosphorus as an impurity. The iron is brittle when cold, but of ordinary character when heated. It cracks if bent cold, but may be forged and welded at high temperatures.

Red-shortness is generally produced by the presence of sulphur, sometimes by arsenic, copper, and other impurities. The iron is tough when cold, but cannot be welded, and is difficult to forge at high temperatures.

130. COLD ROLLED IRON.

The following general conclusions are obtained from Prof. R. H. Thurston's report :—

1. The process of cold-rolling produces a very marked change in the physical properties of the iron thus treated.

(*a*) It increases the tenacity from 25 to 40 per cent., and the resistance to transverse stress from 50 to 80 per cent.

(*b*) It elevates the elastic limits under both tensile and transverse stresses, from 80 to 125 per cent.

(*c*) The modulus of elastic resilience is elevated from 300 to 400 per cent. The elastic resilience to transverse stress is augmented from 150 to 425 per cent.

2. Cold-rolling also improves the metal in other respects.

(*a*) It gives the iron a smooth, bright surface, absolutely free from the scale of black oxide unavoidably left when hot-rolled.

(*b*) It is made exactly to gauge, and for many purposes requires no further preparation.

(*c*) In working the metal, the wear and tear of the tools are less than with hot-rolled iron, thus saving labour and expense in fitting.

(*d*) The cold-rolled iron resists stresses much more uniformly than does the untreated metal. Irregularities of resistance exhibited by the latter do not appear in the former; this is more particularly true for transverse stress, as is shown by the smoothness of the strain-diagrams produced by the cold-rolled bars.

(*e*) This treatment of iron produces a very important improvement in uniformity of structure, the cold-rolled iron excelling common iron in its uniformity of density from surface to centre, as well as in its uniformity of strength from outside to the middle of the bar.

3. This great increase of strength, stiffness, elasticity, and resilience is obtained at the expense of some ductility, which diminishes as the tenacity increases. The modulus of ultimate resilience of the cold-rolled iron is, however, above 50 per cent. of that of the untreated iron.

Cold-rolled iron thus greatly excels common iron in all cases where the metal is to sustain maximum loads without permanent set or distortion. —JONES *and* LAUGHLINS.

131. CASE-HARDENING.

When polished wrought iron is heated to a cherry red and placed in contact with broken prussiate of potash (K_4FeCy_6), scraps of leather, etc., the surface is converted into steel by absorption of carbon, and is then hardened by quenching in water. The nitrogen in the mixture is supposed to play an important part.

In locomotive engine factories a mixture of wood charcoal, soda ash, and a little lime is used, and in some other factories saltpetre and leather scrap are used in the proportion of 1 lb. of the former to 8 lbs. of the latter.

Other nitrogenous matters, such as bone-dust, horn, hoof, and hide clippings, are often used. If heated with the mixture in a close box the effect is greater. The case-hardening may extend to a depth of about $\frac{1}{16}$ inch for ordinary work, and $\frac{1}{8}$ inch for special cases. The surface shows a mottled appearance before re-polishing.

This method of hardening is used largely for motion blocks, links, pins, and eyes, and generally for small articles or portions of them which have to stand much friction. It is cheaper than using steel, but the tendency of the articles to crack and twist is an objection.

132. CASTING WROUGHT IRON.

In the "Mitis" process (Nordenfelt's) a small amount of aluminium, say $\frac{1}{2000}$ to $\frac{1}{700}$ by weight, is added to Swedish wrought iron, which causes it to melt and flow at a temperature insufficient to cause the occlusion of gases, say 2,200° F. (1,200° C.), and sound tough castings are obtained, having all the properties of the best forged iron, except that they are perfectly homogeneous and free from stratification. Mitis metal will weld and harden.

An addition of 0·05 to 0·1 per cent. of aluminium to mild steel (say 1 lb. per ton) produces sound and fine grained ingots free from blow holes, giving increased weight and solidity without reducing the strength.

133. DEFINITION OF STEEL.

Steel may be made by the addition of carbon to wrought iron, or the abstraction of carbon from cast iron; both methods are in use commercially; but the old classification, by which the percentage of carbon alone determined the designation, is now nearly discarded, and the better definition would seem to include "all those malleable forms of commercial iron containing iron and carbon produced from a state of fusion into a malleable ingot." When the carbon contained is less than 0·5 per cent. the result is "mild" steel. Structural steel should contain not more than 0·3 per cent. of carbon or phosphorus.

"In Germany it is becoming usual to describe iron and steel produced by puddling as *weld metal*, and to distinguish the product of processes where fusion takes place as *ingot metal*."—W. N. TWELVETREES.

Structural steel may be divided into three grades; *mild or soft*, below 0·15 per cent. carbon, suitable for boiler plates and similar uses; *medium*, 0·15 to 0·30 per cent. carbon, for joists and general structural purposes; *hard*, above 0·30 per cent. carbon, for axles and shafts, etc., where wearing surfaces are desired.—PENCOYD.

134. METALLOGRAPHY OR MICRO-METALLURGY.*

The use of the microscope in its application to metallurgy was first mentioned by Réaumur in 1722 in connection with the structure of a chilled casting. In 1833 François followed with the microscope the successive steps in the reduction of iron from its ores. In 1864 Dr. Sorby, studying polished and etched specimens of iron and steel, said: "Steel must be regarded as an artificial crystallised rock, and, to get a complete knowledge of it, must be regarded as such." From 1885 onwards many workers have been investigating the same subject, extending their labours to the micro-structure of various alloys.

Two steels of identical chemical composition may differ greatly in character; Prof. Arnold, after microscopical analysis, says: "Steel is an igneous rock made up of crystals of pure iron and of carbide of iron with inter-crystalline spaces filled with the compounds of the constituents of steel." He found that the hardness of quenched steel was due to a sub-carbide, corresponding to $Fe_{24}C$, and that the saturation point was reached when iron contained about 0·9 per cent. of carbon.

135. COMPOSITION OF IRON AND STEEL.

Steel may be defined as any variety of iron cast into a malleable ingot, and the two parallel series, the irons and the steels, may be classified as follows:—

Percentage of Carbon.			
0·0 to 0·2	0·2 to 0·35	0·35 to 0·55	0·55 to 1·5 or more.
Series of the Irons.			
Ordinary irons.	Granular irons.	Steely irons and soft puddled steels.	Hard puddled steels. Cemented Steel. Styrian steel.
Series of the Steels.			
Very soft steels.	Soft steels.	Half soft steels.	Hard steels.

—W. Hackney.

* The author recommended a study of the crystallisation of metals to an inquirer for subjects for original microscopical research in the "English Mechanic" about 1870, but the subject was not taken up practically until about 25 years later, when several investigators entered the field.

136. PARALLEL SERIES OF IRONS AND STEELS.

Carbon percentage.	*Wrought iron series.*	*Steel series.*
0·0 to 0·15	Ordinary iron.	Extra soft steel.
0·15 „ 0·45	Granular iron.	Soft steel.
0·45 „ 0·55	Puddled steel.	Medium steel.
0·55 „ 1·50	Cemented steel.	Hard steel.

—M. GREINER.

137. BLISTER STEEL AND SPRING STEEL.

Blister Steel is produced by a process called cementation. Bars of purest wrought iron are placed in a furnace between layers of charcoal powder, and kept at a high temperature, say 1400° F. (760° C.) for from five to fourteen days. The bars are now brittle, crystalline and more or less covered with blisters. Small regular blisters and fine grain denote good quality. Used for facing hammers, etc., but not for edge tools ; used largely for conversion into other kinds of steel.

Spring Steel is blister steel heated to an orange-red colour, and rolled or hammered.

138. CLASSIFICATION OF BLISTER STEEL.

No. 1. Spring heat	$\frac{1}{2}$	per cent.	of carbon.
„ 2. Country heat	$\frac{5}{8}$	„	„
„ 3. Single-shear heat	$\frac{3}{4}$	„	„
„ 4. Double-shear heat	1	„	„
„ 5. Steel-through heat	$1\frac{1}{4}$	„	„
„ 6. Melting heat	$1\frac{1}{2}$	„	„

—SEEBOHM.

139. SHEAR STEEL AND CRUCIBLE STEEL.

Shear Steel (sometimes called *tilted steel*) is blister steel cut into short lengths, piled into faggots, sprinkled with sand and borax, and placed at welding heat under a tilt hammer. "Single" and "double" shear steel denotes the number of times this process is repeated. Fibrous character now restored. Used for large knives, scythes, plane irons, shears, etc., frequently in conjunction with iron.

Crucible Cast Steel.—Originally made by melting fragments of blister steel in covered fireclay crucibles, and running into iron moulds. Now generally made direct from Swedish bars cut up and placed in crucibles, with a small quantity of charcoal, with subsequent addition of spiegeleisen or oxide of manganese. Variations on this process are known as Heath's and Mushet's, also Tungsten steel, Chrome steel, etc. Forged at low heat, unweldable, fracture grey, crystals very minute.

140. BESSEMER STEEL.

Bessemer Steel (1856).—Made from grey pig iron containing a large proportion of free carbon, small quantity of silicon and manganese, free from sulphur and phosphorus. Iron melted in cupola, and run into a converter lined with firebrick and suspended on hollow trunnions. Air blown through the metal about twenty minutes, removing all carbon; 5 to 10 per cent. spiegeleisen then added, and blowing resumed long enough to incorporate the two metals, say 30 minutes in all. Steel then run out into ladle and moulds. Ingots being porous are reheated and put under steam hammer, then rolled or worked as required. Used for rails, tyres, axles, common cutlery and tools, roofs, bridges, etc.

141. VARIETIES OF SIEMENS STEEL.

Siemens Steel.—Pig iron melted on hearth of Siemens regenerative furnace (1856); good ore and limestone are then added and heat kept up, process resulting in carbonic acid gas, slag, and steel. Open hearth process is more under control and gives more time to adjust the composition than the Bessemer process does. Open hearth steel is used for shipbuilding, boiler making, and constructive work generally.

All varieties are known generally as "mild steel."

Siemens-Martin Steel.—Pig iron melted in furnace, three or four times its weight of heated wrought-iron scrap or steel added, together with spiegeleisen or ferro-manganese, until required proportion of carbon, etc., is obtained, to give steel of requisite hardness; then run into ingot moulds.

The leading British works are now nearly all converted to *basic* processes of steel making. Formerly the open hearth *acid* process was generally used, and harder qualities of steel were favoured as being cheaper, the position is now reversed. If too hard a quality is used it is difficult to obtain clean, well-rolled sections of full dimensions.—SKELTON, 1906.

Dorman, Long and Co. manufacture their mild steel by the "Basic Open Hearth Process," and their standard tests are 28 to 32 tons per square inch, tensile stress, 20 per cent. elongation in 8 inches, 40 per cent. reduction of area at point of fracture. Their sections are stocked in lengths of every foot from 10 to 40 feet. The trade margin in rolling is 2½ per cent. over or under list dimensions and weights, with 1 inch margin of length over or under. An extra is charged for cutting within ⅛ inch of exact length, and for machining square.

Landore Siemens Steel.—Iron ore is treated in a rotatory furnace with carbonaceous material, and converted into balls of malleable iron, which are transferred direct to steel-melting furnace. Spiegeleisen, etc., then added. The result is steel of very ductile quality, dense and uniform in texture, and particularly suitable for replacing wrought iron where increased strength is required, in addition to all the best properties of wrought iron.

142. OTHER VARIETIES OF STEEL.

Galy-Calazat Steel.—Superheated steam is forced through the molten metal, thus oxidising the carbon, and also removing the sulphur and phosphorus as sulphuretted and phosphoretted hydrogen. Used in France.

Heaton Steel.—The melted metal is acted on by certain salts, such as nitrate of soda, etc., by which the carbon is oxidised out. Henderson employed fluorides, and Bell, oxide of iron.

Gilchrist-Thomas, or *Basic Steel* (1879).—Similar to Bessemer, but difference in the lining of the converter, which is basic or non-siliceous, made from burnt dolomite or magnesian limestone, enabling impure phosphoric pig-irons to be treated. Phosphorus eliminated quickly and cheaply by combining with the lime; the resulting slag containing 10 to 18 per cent. of phosphorus is used as manure when pulverised. Siliceous or acid linings are used with hæmatite pig.

Compressed Steel is made by the application of pressure to the fluid metal, and the resulting steel is of superior density and tenacity, besides being free from blow holes. It was used by Sir Joseph Whitworth for making hollow marine crank and propeller shafts, where the increase of strength and reduction of weight enabled a smaller external diameter to be adopted, with a consequent saving of frictional losses.

143. DANNEMORA CAST STEEL.

Carbon.	*Temper.*	*Tools suited for*	*Remarks.*
per cent. 1½	Razor	Turning and planing, drills, etc.	Great skill required in forging, spoilt if overheated.
1¼	Turning tool	Turning, planing, and slotting tools, drills, small cutters and taps.	Not weldable.
1⅛	Punch	Mill picks, circular cutters, taps, rimers, small shear-blades, large turning tools and drills, punches, and screwing dies.	May be welded with great care.
1	Chisel	Cold chisels, hot setts, medium-size shear-blades, large punches, large taps, miners' drills for granite.	Will weld with care.
⅞	Sett	Cold setts, minting dies, large shear-blades, miners' drills; smiths' tools, as sett hammers, swages, flatteners, fullers, etc.	Will weld without difficulty.
¾	Die	Boiler-cups, snaps, hammers, stamping and pressing dies, welding steel for plane-irons, etc.	Will weld like iron.

144. HIGH-SPEED TOOL STEEL.

No. 1 temper, containing about 1·30 per cent. carbon, is suitable for small turning and planing tools, drills, and small cutters; also for tools requiring the sharpest edges, razors, and surgical instruments. Great care must be taken not to overheat this quality.

No. 2 temper, containing about 1·15 per cent. carbon, is suitable for heavier turning, planing, and slotting tools, drills, cutters, reamers, and engraving tools.

No. 3 temper, containing about 0·90 per cent. carbon, is suitable for large circular cutters, reamers, taps, and screwing dies, heavy turning tools, large drills, and taps.

No. 4 temper, containing about 0·80 per cent. carbon, is suitable for cold chisels, hot setts, small shear blades, and large taps.

No. 5 temper, containing about 0·75 per cent. carbon, is suitable for screwing dies, cold setts, hammers, swages, minting dies, miners' drills, smiths' tools, punches, and shear blades.

No. 6 temper, containing about 0·65 per cent. carbon, is suitable for snaps, dies, cup drifts, hammers, and stamping dies.

"Vita" Brand is a very superior water-hardening steel specially adapted for finishing brass and steel at increased speeds in turret and capstan lathe work, and for turning chilled rolls and very hard materials.

"Dura" Brand is a special water-hardening steel suitable for ordinary drills, twist-drills, small mining tools, brass finishing tools, reamer blades, and wood-working tools.

It is very important to the producer that users of steel should state the purpose for which they require it, for it is obvious from the above grades that using, say, a No. 5 temper where a No. 1 is required, must be unsatisfactory alike to both parties, and often, in consequence, the steel is alleged to be bad and not unnaturally condemned; whereas, the fact is really that it has been applied to a purpose it was not made for.—J. M. GLEDHILL.

145. "EIDSFOS STÖBESTAAL" CAST STEEL.

Quality.	*Percentage Carbon. Prof. Eggertz's method.*
For turning and planing tools for metals . . .	1·55 to 2·00
,, slotting and boring tools	1·45 ,, 1·55
,, cold chisels, etc.	1·25 ,, 1·45
,, edge tools, joiners' tools, etc.	1·10 ,, 1·25
,, mining tools, fine springs, twist drills, and for tools requiring toughness	0·90 ,, 1·10
,, buffer springs, axles, shafts, tools requiring great toughness, etc.	0·75 ,, 0·90
,, gun barrels, and for tools requiring the greatest degree of toughness	0·40 ,, 0·75

146. ELECTRIC STEEL.

Sybry, Searls and Co., Ltd., Sheffield, state that their steel is produced from the purest Swedish raw metal by means of induced electricity in an electric furnace. They claim that the use of electricity for heating reduces the cost of production, and enables a uniform and homogeneous tool steel to be obtained with perfect freedom from impurities, the unhardened steel having great ductility and softness, and welding freely. It is produced in seven qualities of temper or hardness.

147. NICKEL STEEL.

The addition of 0·7 per cent. of nickel to steel suddenly converts it from one having fair tensile strength and ductility into an exceedingly hard and strong metal with only 2 per cent. elongation, which breaks suddenly like a chisel.

148. ANNEALING STEEL CASTINGS.

Steel castings should be annealed for twenty-four hours at a temperature of 1700° F. (930° C.). Annealing reduces tensile strength but increases ductility and removes internal stress.

149. PRECAUTION IN WORKING MILD STEEL.

When any bending or flanging requires the steel to be heated care must be taken that no work is done upon it at a " blue heat," or from 400 to 600° F. (205 to 315° C.), as within that range of temperature the steel is very brittle, and may be irretrievably damaged; for this reason it is frequently necessary to suspend the work and reheat the material. Hydraulic flanging is a particularly suitable method owing to the amount of work that can be done at one time with little variation of temperature.

150. OVERHEATED AND BURNT STEEL.

" Steel which has been exposed to a very high temperature is known as 'burnt.' It is cold-short and brittle, can be forged and welded only with care, and has a low tensile strength. Its fracture is coarse and even flaky, crystalline, with brilliant facets. Steel known as 'overheated' has a coarse structure, which may be removed more or less completely by reheating or careful forging. Excessively long or strong overheating produces the structure known as 'burnt,' and the coarseness and brittleness due to burning are removed with greater difficulty and much less completely than those due to overheating, yet in the same manner and by the same expedients."—PROF. HOWE.

151. SOLIDIFICATION AND RECALESCENCE.

A steel ingot freshly poured loses its heat regularly except at two points, the point of solidification, about 2700° to 2600° F. (1480° to 1430° C.), where it falls more slowly, and the point of recalescence, between 1300° and 1200° F. (700° and 650° C.), depending on the amount of carbon, where it either remains stationary, or rises again for a short time. At each of these points a change of structure takes place.

"The fluid steel begins to crystallise at the point of solidification, and the slower the rate of cooling from there down the larger the crystals will be when the ingot is cold. At the point of recalescence, however, it would seem as if the crystallisation, so to say, locks itself, for after the ingot has become cold, if we reheat it to a temperature below this point, on again becoming cold we shall find that the crystallisation is not affected; but if we reheat it a little above the recalescent point, when it is again cold the crystallisation will be found to be much smaller than before. In fact, it is known that if steel is heated slightly above the recalescent point, all previous crystallisation is destroyed, and a fine, amorphous condition is produced at that temperature. As soon as cooling begins again, crystallisation sets in and continues until the ingot is cold. As, however, the time of cooling from the recalescent point is comparatively short, the resultant crystallisation is correspondingly small. It can be readily understood that when heat treatment can completely change the internal condition of steel, it should bear an important part in the manufacture of forgings made of that metal."—H. F. P. Porter.

152. DEFECTS IN STEEL BOILER PLATES.

Laminations are usually caused by blow holes in the ingots when cast; when the plate is rolled they extend in the same manner as the plate. If the plate is of great length in proportion to its width, this lamination will be of similar form. Thus, in shell plates, while laminations from blow holes rarely extend more than a few inches across the end, they may be several feet in length. Blow holes sometimes form in the centre of the ingot, the material shrinking away from the centre towards the outside, and the impurities in the steel to some extent collecting there. This class of defect is called "piping," and when it occurs a large lamination with defective material adjoining is formed, and is serious. Another class of defect is that caused by small cracks in the sides of the ingots when cast. These fractures are flattened down in the plate, and are usually detected by a faint irregular line running across or along the plate. When this is chipped, it is found to extend from ⅛ in. to half through the plate. Another class of lamination is caused by the rolls folding over the end of the bloom when rolling it into a plate. When finishing the plate this folded portion is rolled closely together, and sometimes is not entirely sheared off. When laminations of this kind occur at the end of shell plates they will generally run across the full width of it, but not extend to any great length.

Another class of defect is caused by the oxide formed while the steel is hot being rolled into it, and causing pittings. These defects, if not disclosed previously, generally show themselves by the scale cracking off when the plates are being bent to form. Foreign material is also sometimes rolled into the plate, causing deep, sharp pittings or grit embedded in the plate. —E. G. Hiller.

153. MAXIMUM DIMENSIONS TO WHICH STEEL PLATES ARE ROLLED.

Thickness in inches.	*Length in feet.*	*Width in inches.*	*Area in feet.*	*Thickness in inches.*	*Length in feet.*	*Width in inches.*	*Area in feet.*
$\frac{1}{8}$	14	48	48	$\frac{3}{4}$	44	81	220
$\frac{3}{16}$	20	60	80	$\frac{13}{16}$	44	81	200
$\frac{1}{4}$	26	72	100	$\frac{7}{8}$	44	81	200
$\frac{5}{16}$	32	75	150	$\frac{15}{16}$	44	81	185
$\frac{3}{8}$	36	81	200	1	44	81	175
$\frac{7}{16}$	44	81	200	$1\frac{1}{16}$	44	81	165
$\frac{1}{2}$	44	81	200	$1\frac{1}{8}$	44	81	155
$\frac{9}{16}$	44	81	235	$1\frac{1}{4}$	44	81	135
$\frac{5}{8}$	44	81	235	$1\frac{3}{8}$	44	81	125
$\frac{11}{16}$	44	.81	230	$1\frac{1}{2}$	44	81	115

⅛ in. plates are rolled full to the gauge and about 6 lbs. per square foot.

Can roll diameter plates ⅜ in. thick and over up to 7 feet wide.

The steel plates may be cold rolled up to 2 ft. 6 in. wide and ¾ in. thick, 3 ft. wide and ⅝ in. thick, 6 ft. wide and ½ in. thick.—Consett Iron Co., Ltd.

154. RELATIVE PIG-IRON PRODUCTION OF DIFFERENT COUNTRIES.

—	1886.	1905.
	tons.	*tons.*
Great Britain	7,750,657	9,592,737
United States	4,044,526	22,992,380
Germany and Luxemburg	3,751,775	10,987,623
France	1,628,941	3,076,550
Austria and Hungary	760,000	1,514,840
Belgium	714,677	1,310,290
Russia	498,000	2,765,000
Sweden	430,504	527,300
Spain	126,269	385,000
Italy	24,778	140,825
All other countries	150,000	700,000

—"Iron and Steel Trades' Journal."

Pig-iron Production of the World:

Year		*Tons*	*Year*		*Tons*
1870	about	12,500,000	1900	about	40,000,000
1880	,,	18,500,000	1904	,,	45,000,000
1890	,,	24,500,000	1906	estimated	55,000,000

—"Society of Arts Journal."

155. NOTES ON CAST IRON.

Stronger in compression than wrought iron, but much weaker in tension. Not so safe as wrought iron when subjected to impact or suddenly-applied loads, owing to its brittleness.

Used for complex parts of machines, because easier to mould in casting than wrought iron in forging. Principally for wheels, bed-plates, and framings.

If thickness of different parts varies much, the castings will be strained in cooling. All edges should be well rounded and hollows filleted.

Expands at moment of solidification in casting, but contracts in cooling. Contraction varies with size and thickness of casting, and quality of metal.

Heat escapes perpendicular to surfaces and crystals arrange themselves in the same direction, hence square angles give weak junctions.

156. QUALITIES OF CAST IRON.

No. 1. *Grey.*—Soft. Deficient in strength. Used for ordinary castings. Very fluid when melted. 0·6 to 1·5 per cent. carbon chemically combined, 2·9 to 3·7 per cent. mechanically combined.

No. 2. *Mottled.*—Variable hardness. Stronger than No. 1. Used for larger castings. More carbon chemically combined, and less mechanically.

No. 3. *White.*—Hard. Fusible. Strong. Used for conversion into wrought iron. 3 to 5 per cent. of carbon all chemically combined.

These varieties are mixed in various proportions for special purposes.—UNWIN'S "MACHINE DESIGN."

157. CHILLED AND MALLEABLE CAST IRON.

Chilled Cast Iron is ordinary cast iron rapidly cooled during solidification, by using a mould of white or hard cast iron for the part requiring to be chilled, protected by a wash of loam, causing a chemical combination of the molten iron and carbon. Very hard. Fracture silvery. Direction of crystallisation strongly marked.

Malleable Cast Iron is made by heating ordinary castings, preferably of white cast iron, from two to forty hours, according to size, in contact with oxide of iron or powdered red hæmatite, causing partial conversion into wrought iron by abstraction of carbon.

158. TOUGHENED CAST IRON.

Toughened cast iron is produced by adding to the cast iron, and melting amongst it, from one-fourth to one-seventh of its weight of wrought-iron scrap, which removes some of the carbon from the cast iron, and causes an approximation to steel.—"NOTES ON BUILDING CONSTRUCTION," iii., 252.

159. COPPER.

Very malleable, and hence specially suited for hammering into thin hemispherical pans, rolling into sheets, etc., also ductile to a less degree. Rendered brittle by absorption of carbon, refined and toughened during manufacture, but may be spoilt again by careless manipulation. May be cast. Can be forged cold, or at red heat, but rapidly scales when hot. Addition of 2 to 4 per cent. of phosphorus improves its fluidity and tenacity. Used for fire-boxes, etc., because it is a good conductor of heat, but loses tenacity in proportion to its temperature. Separate pieces are joined by brazing or riveting according to circumstances.

A mixture of copper containing $2\frac{1}{2}$ per cent. of silicon may be readily drawn into wire or rolled into sheets, and excellent spring metal has been made from it.

Copper is much used in forming alloys.

160. ALUMINIUM.

Aluminium, by the Deville-Castner process, produced by electrolysis from alumina contained in the mineral bauxite, is made at a tenth of its former price, and for many of the lighter parts of mechanism or delicate machinery may become a substitute for the more common metals, as it does not tarnish even when exposed to damp and impure air. It is, however, not a strong metal. For stamping, bending, or working cold it is necessary that it be free from silicon, as a very small percentage renders it brittle. It is the lightest of the commercial metals, and is largely used for the manufacture of domestic articles.

Aluminium is called by the makers Alium, for brevity. Since 1889 the production of aluminium has increased largely. In that year the total output was only 85 tons, in 1905 it was roughly 8,000 tons. As the production has increased the price has fallen. From 10s. 6d. per pound it has dwindled to 1s. 3d., and if a cheaper raw material than refined alumina could be used further reduction in price would be possible. There are at the present time nine works operating either the Hall or the Héroult methods of aluminium production, and between 40,000 and 50,000 horse-power are employed in the industry. Of these works only one is in the United Kingdom. Three are in America, two in France, one each in Germany, Switzerland, and Austria. A works is in course of erection in the Valley of Pescara, Italy. The demand for the metal is growing in connection with motor car and railway carriage work, the latest example of its use being for the inside of cars for one of the London underground tube lines.

161. DEFINITION OF ALLOY.

A substance possessing the general physical properties of a metal, but consisting of two or more metals, or of metals with non-metallic bodies in intimate mixture, solution, or combustion with one another, forming, when melted, a homogeneous* fluid.—COMMITTEE IRON AND STEEL INSTITUTE.

"The structure of alloys has been most satisfactorily explained by considering that different metals are soluble in each other in different proportions under different states of concentration and at different temperatures; that of steel has been especially thoroughly worked out in the same way, and it has been shown that it consists of a solid solution of carbon in pure iron, while that of cast iron is explained by the fact that the amount of carbon soluble in the molten iron is so great that a portion separates out, as graphite, on cooling."

162. ALLOYS.

Bronze is a mixture of (say) 10 copper, 1 tin.

Brass is a mixture of (say) 2 copper, 1 zinc.

The terms "higher" and "lower" applied to brass express the greater or less quantity of zinc in the composition. High brass consists of 2 copper to 1 zinc. Low brass 4 copper to 1 zinc.

Alloys of tin sometimes called "ternary" alloys.

Gun Metal is a mixture of copper, tin, and zinc in various proportions, according to the hardness or toughness required: say 16 copper, 2 tin, 1 zinc. May be also called bronze.

Muntz Metal is a mixture of 3 copper, 2 zinc, and is therefore a brass.

Alloys generally fuse at a lower temperature than the average of the component metals.

Willans and Robinson, Ltd., Derby, have a patented method of treating any of the commercial white or yellow alloys for bearings, which they claim increases the crushing strength and reduces the coefficient of friction to a very considerable extent.

163. EFFECT OF ALLOYING WITH COPPER.

Tin increases the hardness, and whitens the colour through various shades of red, yellow, and grey.

Zinc in small quantity increases fusibility without reducing the hardness, in greater quantity increases malleability when cold, but entirely prevents

* This must not be taken as inferring homogeneity in the solid mass which, it is well known, does not exist.

forging when hot; 1 to 2 per cent. of zinc enables sounder castings to be made.

Lead increases the ductility of brass, and makes alloy more suitable for turning, filing, etc.; in large quantity causes brittleness.

Phosphorus increases the fluidity and tenacity, reduces the effect of the atmosphere, and allows of tempering. It also produces sounder castings.

164. DECAY OF METALS AND ALLOYS.

In presence of sea-water zinc tends to disappear from alloys by lapse of time, as much as $\frac{5}{6}$ of the original quantity vanishing. Cast iron becomes altered so that it may be cut by a knife as if plumbago. Bronze and brass become pitted. Gun metal propellers do not decay as much as those having zinc in their composition.

The following summary is given by Messrs. Milton and Larke, Min.Proc. Inst.C.E. CLIV.:—

1. "Decay" is more frequent in metals which have a duplex or more complex structure than in those which are comparatively homogeneous.

2. "Decay" is due to a slower or less energetic action than that causing "corrosion"; moreover, it requires an action which removes part only of the constituents of the metal, whereas "corrosion" removes all the metal attacked.

3. Both "decay" and "corrosion" may result from chemical action alone, or from chemical and electrolytic action combined.

4. "Pitting," or intense local corrosion, is probably often due to local segregation of impurities in the metal; but it may also in some places be due to favourable conditions furnished by local irregularities of surface or structure producing local irregularities in the distribution of galvanic currents.

5. For brass exposed to sea-water, tin is distinctly preservative, while lead and iron are both injurious, rendering the alloy more readily corrodible. The percentage of the two latter metals should therefore be kept as low as possible in all brass intended for purposes where contact with sea-water is inevitable.

6. With a view to obtain a minimum of corrosion, the internal surfaces of condenser-tubes should be as smooth and uniform as possible; and in order

to ensure this condition the cast pipe from which they are drawn should be smoothly bored inside, either before the drawing is commenced, or in an early stage of the process, as is done in the manufacture of brass boiler-tubes.

7. The experiments with an applied electric current show that electrolytic action alone, even where exceedingly minute currents are employed, may result in severe corrosion or decay. Every effort, therefore, should be made to prevent such action, by careful insulation of all electric cables. Where galvanic action is inevitable, through the proximity of different metals exposed to the same electrolyte, the currents resulting should be neutralised by the application of zinc plates in the circuit, so arranged that they will be negative to both of the other metals.

Blocks of cast zinc and also of thick rolled zinc have been fixed at several points inside marine boilers below water line, to counteract the electrolytic action which took place, and caused the pitting of the steel plates. In some cases they have been quite effective, but in others they were of no apparent use, as the pitting continued unchecked. The reason for the difference was not discovered.

165. BRONZE ALLOYS.

Name.	*Copper.*	*Tin.*	*Zinc.*
Soft gun metal	16	1	..
Mathematical instruments	12	1	..
Pumps (very tough)	32	3	1
Pump plungers	14	1	1
Small toothed wheels	10	1	..
Locomotive bearings	64	7	1
Engine bearings	112	13	¼
Locomotive straps and glands	130	16	1
Admiralty mixture for valves and mountings	90	10	2½
Best ordinary	86	10	4
Hard gun metal for bearings	8	1	..
Baily's metal	32	5	2
G.M. for heavy bearings	32	5	1
Maximum hardness for bearings	5	1	..
Hydraulic valve faces	4	1	..
Tam-tam (Chinese gongs)	4	1	..
Bell metal	3 or 5	1	..
Speculum metal	2	1	..

166. BRASS ALLOYS.

Name.	*Copper.*	*Zinc.*	*Tin.*	*Lead.*
Tough for engine work . .	100	15	15	..
For turning and fitting . .	3	1	..	$\frac{1}{12}$
Soft for hammering and for cartridge cases	7	3	..	..
Yellow brass	2	1	..	..
Stop-cocks and valves . .	88	10	2	..
,, ,, ,, . .	73	8	7	12
Rolling-stock bearings . .	77	..	8	15
Flanges for brazing . . .	32	1	..	1
Brass for soldering . . .	8	3	..	..
Brass, various	60–92	8–40	$\frac{1}{4}$–3	$\frac{1}{2}$–3
Muntz metal sheathing. . .	3	2	..	..
,, locomotive tubes .	66	33	..	1
Condenser tubes (best) . .	78	21	1	..
,, ,, (common) .	68	32	..	..
Admiralty tubes (common) .	70	29	1	..
Nails for sheathing . . .	87	4	9	..
Statuary bronze . . .	90	5	2	..
Red brass (Tombak) . . .	8–10	1	..	..
Red sheet brass (German) .	11	2	..	..
Bronze for lamps . . .	27	6	1	1

167. ANTIMONY ALLOYS.

Name.	*Copper.*	*Tin.*	*Lead.*	*Antimony.*	*Bismuth.*
Babbitt's metal . .	1	10	..	1	..
,, lining do. .	1	24	..	2	..
Antifriction do. (hard) .	1	50	..	5	..
,, ,, (soft) .	..	..	81–88	12–19	..
White bronze . .	5·5	83·5	..	11	..
Expanding alloy . .	..	..	..	2	1
Pewter . . .	..	100	..	17	..
Type metal . .	..	..	3–7	1	..
Stereotype metal . .	..	..	77	15	8
White brass . .	1	..	7	7	..
,, ,, . . .	3	90	..	7	..
Alloy contracting when heated . . .	..	1	1	..	2

168. NICKEL ALLOYS.

Name.	*Copper.*	*Zinc.*	*Nickel.*	*Iron.*
Common German Silver . .	60	25	15	..
Better ,, ,, . .	50	25	25	..
Chinese Packfong . . .	55	17	23	3
Argentan, for hammering or rolling	40·4	25·4	31·5	2·6
,, for plating . . .	62	19	13	4–5
,, hard . . .	57·4	25	13	9
Electro	8	3·5	4	..
Solder for German silver (coarsely powdered)	8	7·5	4	..

169. VARIOUS ALLOYS.

Name.	*Copper.*	*Tin.*	*Zinc.*	*Various.*
Silver-bell metal .	80	10	6	4 lead.
Pot or cock metal .	5	..	..	2 lead.
Ship nails . . .	10	..	8	1 iron.
Cowper's metal . .	..	2	..	1 bismuth.
Aluminium bronze .	90	..	..	10 aluminium.
Lining metal for heavy bearings . .	..	25	25	50 ,,
Sterro-metal . .	60	2	35	3 wrought iron.
Gedge's metal . .	60	..	38·2	1·8 ,,
Delta metal . .	55½	¼	41½	1 lead, 1 iron, ¾ manganese.
Phosphor bronze .	82	10	..	7½ lead, ⅛ iron, ⅛ nickel, ¼ phosphorus.
Common pewter .	..	83	..	17 lead.
Japanese bell metal (according to size) .	60	24–18	9–6	0 to 12 lead, 3 iron.
Forgeable alloy . .	100	..	75	1 iron.
Manganese bronze for polishing . .	67	..	13	1¼ aluminium, 18¾ manganese.
Do. for strength .	88	10	..	2 manganese.
British coinage :—				
Bronze . . .	95	4	1	..
Silver . . .	7½	..	..	92½ silver.
Gold . . .	..	..	..	91⅔ gold.

170. FUSIBLE ALLOYS.

Melting Point.		Lead.	Tin.	Bismuth.	Zinc.	Corresponding Absolute Steam Pressure.*
deg. F.	deg. C.					lbs.
212	100	1	3	5	..	14·7
246	119	1	4	5	..	28
286	141	..	1	1	..	54
310	154	3	3	1	..	78
320	160	4	4	1	..	90
334	168	..	2	1	..	110
336	169	2	3	..	..	112
340	171	1	2	..	..	118
356	180	1	3	..	..	146
365	185	1	4	..	..	162
378	192	1	5	..	..	191
381	194	1	6	..	..	200
392	200	..	8	1	..	230
442	228	..	1	..	..	384
472	244	..	..	1	..	..
612	322	1	..	..	..	..
648	342	..	..	..	1	..

171. ALLOYS FUSIBLE BELOW 212° F. (100° C).

Melting Point.		Lead.	Tin.	Bismuth.	Mercury.	Cadmium.
deg. F.	deg. C.					
212	100	5	3	8	..	..
210	99	4	3	8	..	..
203	95	31	19	50	..	..
200	93	1	1	4	..	..
149	65	28·5	17	45·5	9	..
138	59	8	4	15	..	3
122	50	3	5	3	3	..

* This column is added to make the table useful for adjusting the composition of fusible safety plugs for boilers.

172. SOLDERS.

Name.	*Tin.*	*Lead.*	*Copper.*	*Zinc.*	*Silver.*	*Various.*
Aluminium solder .	92	..	..	8	..	..
Do. (best) „ .	20	..	..	11	..	1 aluminium and 1 of 10 per cent. phosphor tin.
Plumbers' fine „ .	1	1	..	..	..	..
Do. coarse „ .	1	3	..	..	..	..
Tinmen's fine „ .	3	1	..	..	..	..
Do. coarse „ .	2	1	..	..	..	..
Spelter, hard „ .	..	..	3	2	..	..
Do., soft „ .	..	..	1	1	..	..
Pewterers' „ .	1	1	..	..	..	3 bismuth.
Silver, hardest „ .	..	..	1	..	4	..
Do., hard „ .	..	..	1	..	3	..
Do., soft „ .	..	1 brass wire		..	2	..

173. MELTING POINTS, OR TEMPERATURES OF FUSION, OF VARIOUS METALS, ETC.

	deg. F.	deg. C.
Temperature of electric arc . . .	about 10,000	5,500
Platinum	„ 8,500	4,700
Wrought iron	3,250 to 4,300	1,800 to 2,400
Steel	3,250 to 4,100	1,800 to 2,250
Cast iron	2,200 to 2,750	1,200 to 1,500
Copper.	2,000	1,090
Gun metal	1,900	1,040
Yellow brass	1,850	1,010
Silver	1,830	999
Aluminium	1,200	650
Antimony	810	430
Zinc	750	400
Lead	620	330
Cadmium	608	320

	deg. F.	deg. C.
Bismuth	480	250
Tin	450	232
Sulphur	234	112
Beeswax	150	65
Tallow	100	38
Water	32	0
Mercury	— 38	— 38·5

When a substance $\left\{\begin{matrix}\text{expands}\\ \text{contracts}\end{matrix}\right\}$ in the act of fusion, the solid parts will $\left\{\begin{matrix}\text{sink}\\ \text{rise}\end{matrix}\right\}$ in the liquid. Such substances have their temperature of fusion $\left\{\begin{matrix}\text{raised}\\ \text{lowered}\end{matrix}\right\}$ while under pressure. Example $\left\{\begin{matrix}\text{cast iron}\\ \text{water}\end{matrix}\right\}$.

The density of ice = 0·9167, therefore floating ice has $\frac{8}{9}$ of its depth below the surface of the water.

Bismuth expands on cooling and solidifying, similar to water, while india-rubber contracts on being heated and expands on cooling.

174. COEFFICIENTS OF EXPANSION.

The coefficient of linear expansion is the fraction of its length which a substance increases when its temperature is raised 1°.

a = coefficient of linear expansion
l = length at t degrees
L = length at T degrees

$$L = l + l\,a\,(T - t)$$

The coefficient of superficial expansion is the fraction of its area which a substance expands when its temperature is raised 1°, and is equal to twice the linear coefficient.

Coefficient of superficial expansion $= 2\,a + a^2$, but a^2, being very minute, may generally be omitted.

The coefficient of cubical expansion is the ratio between the increase of volume for a rise of 1° and the original volume, and is three times the linear coefficient.

Coefficient of cubical expansion $= 3\,a + 3\,a^2 + a^3$, but $3\,a^2 + a^3$, being very minute, may generally be omitted.

175. EXPANSION OF METALS BY HEAT.

In fractions of each dimension for 1° C.

Wrought iron	·00001235	Steel	·00001145
Cast iron	·00001127	Brass	·00001894
Copper	·00001717	Platinum	·00000884
Lead	·00002818	Glass	·00000861

—PERRY.

Ordinary gun metal consisting of 88 copper, 10 tin, 2 zinc has nearly the same coefficient of expansion and contraction as cast iron.—"THE ENGINEER."

Water expands $\frac{1}{22}$ of its bulk from 32° F. to 212° F. (0° C. to 100° C.).

From 32° F. to 572° F. (0° C. to 300° C.) iron expands $\frac{1}{227}$, copper $\frac{1}{177}$.

For a rise of temperature of 10° F. (5·5° C.)—

Iron expands about $\frac{1}{15000}$.
Steel ,, ,, $\frac{1}{17000}$.
Copper ,, ,, $\frac{1}{10500}$.
Brass ,, ,, $\frac{1}{9500}$.

176. WEIGHT OF VARIOUS METALS IN POUNDS.

Name.	*Cubic Inch.*	*Cubic Foot.*	*Sq. ft.* 1 *in. thick.*
			say
Gold	·70	1203	100
Lead	·41	710	60
Copper	·32	550	46
Gun metal	·31	530	44
Brass	·30	525	44
Muntz metal	·29	510	43
Steel	·28	490	41
Wrought iron	·28	480	40
Tin	·26	460	38
Cast iron	·26	450	37½
Zinc	·25	435	36
Aluminium, cast	·0936	161·25	13½
,, rolled	·0986	170·6	14

177. MULTIPLIERS TO REDUCE CUBIC FEET TO TONS.

Wrought iron	·2143
Steel	·2175
Cast iron	·2009

178. USE OF WOOD IN ENGINEERING.

Pattern-making.—American yellow pine, New Zealand pine, mahogany, alder, sycamore.

Bearings.—Lignum vitæ (end grain).

Brake Blocks.—Willow, poplar.

Pulley Sheaves.—Lignum vitæ, box.

Buffer Beams.—Oak.

Cylinder Lagging.—Teak, mahogany, oak.

Floats for Paddle-wheels.—Willow, American elm, English elm.

Sluice Paddles.—Oak, greenheart.

Wheel Teeth.—Hornbeam, beech, holly, apple, oak if in damp place.

Joiners' Tools.—Beech, box.

Hammer Shafts.—Ash (cleft).

Tool Handles.—Ash, beech.

Shafts and Springs.—Ash, hickory, lancewood.

Ordinary framing, piling, etc.—Yellow deal, Memel, Riga, or Dantzic (creosoted for outdoor work).

Carriage-building.—Teak.

Fender and Rubbing pieces.—American elm.

Scaffold poles.—Spruce fir.

Earth Waggons and Barrows.—Elm.

Rough Gangways.—White deals.

Piles.—Greenheart, oak, larch. Creosoted Memel. Elm and alder only when wholly immersed.

Warehouse and Factory Floors.—Maple, pitch pine. (In narrow batten widths' splay rebated and secret nailed.)

Elm and larch bear the driving of nails and bolts better than any other timber.—H. Law.

179. FIR, DEAL AND PINE.

Fir is a general term for wood used in the rough as distinguished from *Deal,* a general term for wood wrought and used by the joiner.

4*

Pine is another general term used for even grained stuff suitable for panels. Also for pitch pine.

Yellow deal and red deal are botanically classed as pine.

White deal and spruce deal are botanically classed as fir.

Deal is not a botanical term.

Planks, deals, and battens, and narrow battens are trade terms for boards of certain widths—viz., planks 11 inches, deals 9 inches, battens 7 inches, narrow battens 4½ inches.

180. STANDARD OF DEALS.

In Liverpool and London the St. Petersburg Standard Hundred is adopted as the basis of timber measurement (written "Petg-Std." or "P.S.H."). It consists of 120 boards 12 feet long, 11 inches wide and 1½ inch thick, or the *equivalent quantity* of any other size = 165 cub. ft.

181. PRESERVING IRONWORK.

Painting.—For ordinary work red lead paints are on the whole most suitable, with a little white lead in the first two coats to permit of the paint being worked well into the corners; good raw linseed oil only should be used to mix them for use. Iron oxide paints are cheaper than lead. Coal tar may be used for rough ironwork, underground pipes, etc.; the tar being heated and ½ lb. to 1 lb. finely sifted slaked lime added per gallon of tar, with sufficient naphtha to thin it for laying on. It must be used hot, but not kept on the fire too long. The lime kills the acid in the tar.

Painting steel work with Portland cement wash is desirable in fireproof floors and wherever the material is cased in brickwork or concrete.

For ironwork exposed to damp atmosphere various graphite paints or bitumastic solutions are most suitable, such as Dixon's silica-graphite, Siderosthen, etc.

Bower-Barff Process.—This is specially suited to small pieces exposed to the weather, but not to blows—e.g. rain-water gutters, sanitary fittings and pipes. The articles are raised to a red heat (say 1200° F.) and subjected for some hours (say 6 to 12) to the action of superheated steam, which causes the deposit of a coating of black oxide of iron. It will not stand riveting.

Dr. Angus Smith's Composition.—Is a pitch composition rendered plastic by tar and coal oil, used hot and chiefly for pipes. (See later for full particulars.)

Tallow paint for bright work is now superseded by a composition of crude petroleum prepared by the Ragosine Company.

In reinforced concrete, or ferro-concrete construction, it is generally acknowledged that the iron or steel should be slightly rusted all over before encasing, and that no protective coating should be applied. The concrete should be well tamped round the metal so as to be in close contact, and a chemical compound is then formed which resists further corrosion.

182. COMPARATIVE LIABILITY TO OXIDATION IN MOIST AIR.

Cast iron	=	100
Wrought iron	=	129
Steel	=	133

—MALLET.

Note.—The author's impression, from observation only, not measurement, is that wrought iron oxidises more rapidly than cast iron or steel, that the rust is in thicker flakes, and that the powdery oxide produced on cast iron and steel more readily drops off, or rubs off, so that less moisture is held in contact with the metal.

183. CORROSION OF IRON AND STEEL.

The following table was compiled by Mr. B. H. Thwaites :—

C = coefficient of corrosion during one year's exposure in pounds avoirdupois per square foot. (For values of C, see table.)

W = weight in pounds of 1 ft. length of the section exposed.

L = length in feet of the perimeter exposed. If both the inside and outside perimeters are exposed to the corrosive action, they must both be included.

Y = number of years' life of the metal $= \dfrac{W}{C\ L}$.

Y is based on the assumption that the metal is tolerably uniform in thickness, otherwise the thin portion will have a shorter life than the average of the section.

If painted once a year, multiply the result by 2·0.

If painted once in two years, multiply the result by 1·8.

If painted once in three years, multiply the result by 1·6.

—	Corroding agents.					
	Foul sea water.	*Clear sea water.*	*Foul river water.*	*Pure air or clear river water.*	*Air of city or manufacturing district or sea air.*	*Sea water of average foulness.*
	lb.	lb.	lb.	lb.	lb.	lb.
Cast iron . . .	·0656	·0635	·0381	·0113	·0476	..
Wrought iron . . .	·1956	·1285	·1440	·0123	·1254	..
Steel	·1944	·0970	·1133	·0125	·1252	..
Cast iron, skin removed by planing . . .	·2301	·0888	·0728	·0109	·0884	..
Cast iron, surface protected by galvanising . .	·0895	·0359	·0371	·0048	·0199	..
Cast iron in contact with brass	..	..	..	..	..	·1908
Cast iron in contact with copper	..	..	..	..	..	·2003
Cast iron in contact with gun metal . . .	..	..	..	..	..	·3493
Best wrought iron in contact with brass . .	..	..	..	..	..	·2779
Best wrought iron in contact with copper . .	..	..	..	..	..	·4012
Best wrought iron in contact with gun metal .	..	..	..	..	..	·4537

184. LIMEWHITING.

Limewhiting for walls, or any other purpose, is freshly burnt chalk lime mixed with water, with say 1 lb. alum to each bucketful to keep it white and adhesive. It is not suitable for iron work as it holds moisture too readily.

185. CLAY PUDDLE.

The best material for clay puddle is a stiff loamy clay. A pure clay requires about one-third soil worked up with it to prevent cracking in dry seasons.

Section III.

STRENGTH OF MATERIALS AND STRUCTURES.

186. CLASSIFICATION OF STRAINS*.

Tension	Stretching or pulling.
Compression	Crushing or pushing.
Transverse Strain	Cross strain or bending.
Torsion	Twisting or wrenching.
Shearing	Cutting, or when acting along the grain of timber, detrusion.

187. DEFINITIONS OF STRAIN AND STRESS.

A system of two equal and opposite forces is called a *stress*, when they act from one another they produce *tension*, and when towards one another they produce *compression*. The deformation produced by tension or compression is called a *strain*.—W. M. HICKS.

Strain.—Every load which acts on a structure produces a change of form, which is termed the strain due to the load. The strain may be temporary or permanent, the former disappearing when the load is removed, the latter remaining as permanent set.

Stress (Rankine, 1855).—The molecular forces, or forces acting within the material of a structure, which are called into play by external forces, and which resist its deformation, are termed stresses.—UNWIN'S "MACHINE DESIGN."

Thus the *strength* of a piece in a given position may be such that a *load* of so many *lbs.* produces a *stress* of so many *lbs. per square inch*, the result being a *strain*, or change of form of a certain amount, whether temporary or

* See the author's "Strains in Ironwork," Spon, 5s.

permanent, and, when large enough, appearing as stretching, shortening, bending, crumpling, or twisting.

Intensity of stress is the pressure per unit of surface, or stress per unit of sectional area.

Breaking stress and *ultimate strength* are used as synonymous terms.

188. PROOF STRENGTH.

It was formerly supposed that the proof strength of any material was the utmost strength consistent with perfect elasticity; that is, the utmost stress which does not produce a *permanent set*. Mr. Hodgkinson, however, has proved that a set is produced in many cases by a stress perfectly consistent with safety. The determination of proof strength by experiment is now, therefore, a matter of some obscurity; but it may be considered that the best test known is, *the not producing an increasing set* by repeated application.—RANKINE'S "APPLIED MECHANICS."

189. FACTOR OF SAFETY

is an amount fixed by practical experience, varying with the material used, and the manner of using. It is the ratio of the greatest safe stress to the ultimate resistance of the material, such as $\frac{1}{4}$, $\frac{1}{10}$, etc.; and the calculated resistance of any section, multiplied by the factor of safety suitable to the circumstances, will give the safe working load.

If structures never deteriorated they might be loaded to one-third of their breaking weight with perfect safety, but to guard against ordinary contingencies one-fourth of the breaking weight is the maximum permanent load allowable under any circumstances.

The factor of safety is usually given in its reciprocal form as 4 or 4 to 1, etc., meaning that the ultimate calculated resistance is four times the working load, thus

$$\text{Factor of safety} = \frac{\text{Breaking load}}{\text{Working load}}.$$

Material.	*Factors of Safety.* Dead load.	*Live load.*
Wrought iron and steel girders	3–4	5–6
" columns and struts	4–5	6–7
Cast iron girders	5–6	8–9
" columns	6–7	8–10

As the elastic limit is the highest stress that can be put upon any piece without causing actual damage, the factor of safety should be a ratio determined by the elastic limit rather than by the ultimate strength. Upon this basis the working stress should not exceed one-half and the proof stress two-thirds of the elastic limit.

190. DEFINITION OF ELASTICITY.

Elasticity is that property of matter by virtue of which it tends to return to its original shape and dimensions when the applied forces are removed.

191. HOOKE'S LAW OF ELASTICITY.

Hooke's law was "Ut tensio sic vis," which may be freely translated, "As the pull, so the stretch," or, in other words, the elongation or compression is proportional to the stress.

192. LIMIT OF ELASTICITY.

The limit of elasticity is the maximum stress the material will bear continuously applied or incessantly repeated for an indefinitely long time without continually increasing strain.—R. H. Smith.

The old definition was, "The maximum stress per square inch sectional area, which any material can undergo without receiving a visible permanent set, is called its limit of elasticity, or elastic limit, or elastic strength." This is now obsolete, as more delicate methods of measurement show permanent set with all stresses.

The average limits of elasticity are—Wrought iron, 10 tons; cast iron, 2 tons; steel, 15 tons; but that of cast iron is not clearly defined. Time is an important element in questions of permanent set.

The average elongations under a stress of 1 ton per square inch are—

Wrought iron $\frac{1}{10000}$. Cast iron $\frac{1}{7500}$. Steel $\frac{1}{13000}$.—Anderson.

Wrought iron $\frac{1}{12000}$. Cast iron $\frac{1}{6000}$. Steel —.—Kennedy.

The elastic limit averages—

For wrought iron bars	·5	of ultimate strength
" " " plates	·6	" "
" mild steel	·45	" "
" hard "	·8	" "

193. YIELD POINT.

In testing mild steel the strain is proportional to the stress up to the elastic limit; but shortly after this is passed the deformation is suddenly augmented without any increase of the load, and the stress at which this occurs is known as the *yield point.* Beyond this the strain or deformation increases in a higher ratio over the stress than before, and when a certain stage is reached the deformation continues while the stress is actually reduced.

194. FATIGUE OF METALS.

When repeatedly strained beyond their elastic limit, wrought iron and steel take an increasing permanent set, and ultimately break with less than their original maximum load; but if periodically annealed before rupture takes place, their elasticity may be renewed. This loss of strength, *being recoverable*, may be termed *fatigue.*

When the elastic limit is not overpassed it is still possible under certain conditions to fracture wrought iron and steel by repeating the stress a sufficient number of times. Wöhler showed that it was the *range of stress*, and not the maximum stress, that determined the number of applications before rupture.

195. FAILURES DUE TO FATIGUE.

Material.	*Millions of revns.*	*Range of stress in lbs. per sq. in.*
Wrought iron crank shaft . .	113 ..	7,500
Mild steel „ „ . .	214 ..	10,700
Compressed „ „ . .	270 ..	8,200
Cross head pin	191 ..	15,000
„ „	24 ..	22,000

—M. Longridge.

196. DEFINITION OF MODULUS.

The term "Modulus" simply means a constant, coefficient or multiplier, by means of which one series or system of quantities can be reduced to another similar series or system.

197. MODULUS OF ELASTICITY.

A bar in tension or compression is elongated or shortened by an amount proportionate to the stress within certain limits. The weight in lbs. per square inch sectional area of the bar, to produce this result, is the modulus of elasticity (E). The amount depends upon the kind and quality of the material employed, and may vary 50 per cent.

$$E = \frac{\text{stress per unit of section}}{\text{strain per unit of length}}.$$

198. DEFINITIONS OF MODULUS OF ELASTICITY.

When expressed in feet the modulus of elasticity gives the height to which a body would have to be piled in order that any small addition to its top of its own substance might compress the rest to an extent equal to the bulk of that added quantity.—DR. YOUNG.

It is the weight in lbs. that would stretch or compress a bar, having a sectional area of one square inch, by an amount equal to its own length, called Hooke's law.—CARGILL'S "STRAINS."

The modulus of direct elasticity of a material is the ratio of the stress per unit of section of a bar, to the elongation or compression per unit of length, produced by the stress.—UNWIN'S "MACHINE DESIGN."

When a bar is subjected to a force within the elastic limit of the material which tends to *lengthen* or *shorten* it, it is said to be in *tension* or *compression*. The force in lbs. divided by the area of the cross section of the bar in square inches is termed the *stress*. The ratio of the increase or decrease of length when under stress to the original length of bar is called the *strain*. The ratio of the stress to the strain is called the *modulus of elasticity*.

$$\text{Briefly } \left.\begin{matrix}\text{pull}\\ \text{push}\end{matrix}\right\} = \left\{\begin{matrix}\text{tension}\\ \text{compression}\end{matrix}\right. ; \quad \frac{\text{tension or compression}}{\text{area}} = \text{stress};$$

$$\frac{\text{elongation}}{\text{length}} = \text{strain}; \quad \frac{\text{stress}}{\text{strain}} = E.$$

199. YOUNG'S MODULUS.

Young's modulus of elasticity was originally expressed in feet, and may be obtained from the common table of moduli in lbs. per sq. in. as follows :—

$$\frac{\text{E in lbs. per sq. in.}}{\text{wt. of cub. in. in lbs.} \times 12} = \text{E in feet (Young's Modulus).}$$

The modulus is, however, not now used in this form.

200. FORMULA FOR ELONGATION BY ELASTICITY.

E = Modulus of direct elasticity (see table).
l = Length in inches.
w = Load per sq. inch sectional area in lbs.
e = Elongation in inches.

$$e = \frac{w \times l}{\text{E}}.$$

A bar 1 sq. inch in sectional area put under tension of W lbs. will extend $\frac{\text{W}}{\text{E}}$ of its original length.

Approximately :—

$$\frac{\text{W in tons} \times \text{L in ft.}}{\text{sq. ins. area} \times 1000} = e \text{ in inches for wrought iron.}$$

201. MODULI OF ELASTICITY.

	lbs. per sq. inch.
Cast steel, tempered	40,000,000
Steel, mild	29,000,000
Wrought-iron bar	26,000,000
Ditto plate	25,000,000
Cast iron	18,000,000
Copper	16,000,000
Phosphor bronze	14,000,000
Zinc	13,000,000
Gun metal	10,000,000
Brass	9,000,000
Tin	5,000,000
Lead	720,000
Oak and teak	2,000,000
Fir timber	1,500,000

The above is generally called "Young's Modulus," out of compliment to the originator; it would be more correctly called "Hooke's Modulus," but is really the "Stress-Strain Ratio," or "Coefficient of Elasticity."

202. MODULUS OF ELASTICITY OF BULK.

The pressure in lbs. per square inch upon the exterior of any substance, or the external stress, produces a diminution of bulk per cubic inch, called the *cubical strain* of the substance. The strain is proportional to the stress, and is equal to the stress divided by a certain number called the *modulus of elasticity of bulk*, or modulus of *cubic compressibility*, and represented by K.

K =	Water	300,000
	Cast iron	14,000,000
	Wrought iron	20,000,000
	Steel	24,000,000
	Copper	30,000,000

203. NEUTRAL AXIS.

That layer or plane of fibres in a beam, the length of which is unaltered when the beam is bent by the action of a load, is called the neutral surface, and the line in which this layer cuts any cross section of the beam is called the neutral axis of the section.

204. MOMENT OF INERTIA.

The moment of inertia of a section is the summation of the areas of all its individual parts, multiplied by the squares of their distances from the neutral axis.

$$\Sigma\, a\, y^2 = \text{I}.$$

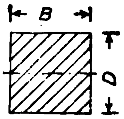
—UNWIN.

In such structures as beams, etc., the moment of inertia of a section is determined from the arrangement of the parts or by the radius of gyration measured from the neutral axis. It is equal to the area of the section multiplied by the square of the radius of gyration. $\text{I} = \text{A}\, r^2$.

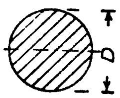
If the moment of inertia (I) of any area (A) be given about an axis through the centre of gravity, its value about any parallel axis, such as the neutral axis, at a distance (d) will be $= \text{I} + \text{A}\, d^2$.

By some modern writers the moment of inertia is called the second moment, second moment of area, moment of the second degree for an area, and geometrical moment of inertia.

205. MOMENTS OF INERTIA.

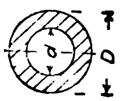


$$A = B\,D. \qquad I = \frac{B\,D^3}{12}. \qquad Z = \frac{B\,D^2}{6}.$$



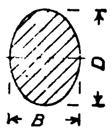
$$A = \frac{\pi}{4} D^2 = \cdot 7854\ D^2. \qquad I = \frac{\pi}{64} D^4 = \cdot 0491\ D^4.$$

$$Z = \frac{\pi}{32} D^3 = \cdot 0982\ D^3.$$



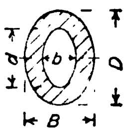
$$A = \frac{\pi}{4} (D^2 - d^2). \quad I = \frac{\pi}{64} (D^4 - d^4).$$

$$Z = \frac{\pi}{32} \left(\frac{D^4 - d^4}{D}\right).$$



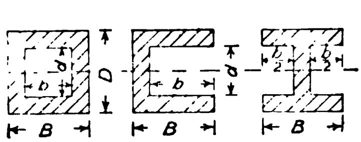
$$A = \frac{\pi}{4} B\,D = \cdot 7854\,B\,D. \quad I = \frac{\pi}{64} B\,D^3 = \cdot 0491\ B\,D^3.$$

$$Z = \frac{\pi}{32} B\,D^2 = \cdot 0982\ B\,D^2.$$



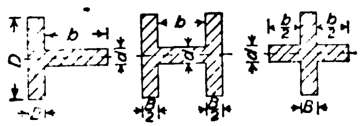
$$A = \frac{\pi}{4} (B\,D - b\,d). \qquad I = \frac{\pi}{64} (B\,D^3 - b\,d^3).$$

$$Z = \frac{\pi}{32} \left(\frac{BD^3 - b\,d^3}{D}\right).$$



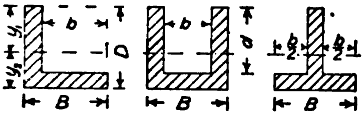
$$A = B\,D - b\,d. \quad I = \tfrac{1}{12} (B\,D^3 - b\,d^3).$$

$$Z = \frac{B\,D^3 - b\,d^3}{6\,D}.$$



$$A = B\,D + b\,d. \quad I = \tfrac{1}{12} (B\,D^3 + b\,d^3).$$

$$Z = \frac{B\,D^3 + b\,d^3}{6\,D}.$$



$$A = BD - bd. \qquad y_1 = \frac{BD^2 - bd^2}{2(BD - bd)}.$$

$$y_2 = \frac{BD^2 - 2bdD + bd^2}{2(BD - bd)}:$$

$$I = \frac{(BD^2 - bd^2)^2 - 4BDbd(D-d)^2}{12(BD - bd)}.$$

$$Z_1 = \frac{I}{y_1} = \frac{(BD^2 - bd^2)^2 - 4BDbd(D-d)^2}{(6BD^2 - bd^2)}.$$

$$Z_2 = \frac{I}{y_2} = \frac{(BD^2 - bd^2)^2 - 4BDbd(D-d)^2}{6(BD^2 - 2bdD + bd^2)}.$$

206. FIRST AND SECOND MOMENTS.

The product of an area into its distance from any axis is called its *simple moment*, or *first moment*, about such axis ; while the product of an area into the square of its distance from such axis is called the *second moment*, or the *moment of the second degree.* In the latter case the area must be very small compared with the distance from the axis.

First moment.—The product of a force, mass area, or volume by the length of its lever arm is called the first moment of the force, mass, area, or volume, or simply the moment.

Moments of leverage are expressed in lb.-ft., ton-ins., etc., to avoid confusion with units of work as ft.-lbs., inch-tons, etc.

Second moment.—The product of a force, mass, area, or volume by the square of the length of its arm is called the second moment of the force, mass, area, or volume. The second moment is sometimes called the moment of inertia, but strictly this expression should only be used in dealing with questions involving the inertia of bodies.

Second polar moment is the moment of inertia round an axis at right angles to the plane of the figure.—Prof. Goodman.

207. SECOND MOMENT AND RADIUS OF GYRATION OF GEOMETRICAL FIGURES.

	Second Moment.	*Radius of Gyration.*
Rectangle about one edge . .	$\frac{bh^3}{3}$	$\frac{h}{\sqrt{3}}$

Rectangle about central axis . .	$\frac{b h^3}{12}$	$\frac{h}{\sqrt{12}}$
„ „ an axis distant R from c.g. . . .	$b h \left(\frac{h^2}{12} + R^2\right)$	$\sqrt{\frac{h^2}{12} + R^2}$
Triangle about axis through apex par. to base	$\frac{b h^3}{4}$	$\frac{h}{\sqrt{2}}$
Do. about c.g.	$\frac{b h^3}{36}$	$\frac{h}{\sqrt{18}}$
Do. at base	$\frac{b h^3}{12}$	$\frac{h}{\sqrt{6}}$
Square about diagonal . . .	$\frac{s^4}{12}$	$\frac{s}{\sqrt{12}}$
Circle about diameter . . .	$\frac{\pi d^4}{64}$	$\frac{d}{4}$
Annulus about diameter . .	$\frac{\pi}{64}(D^4 - d^4)$	$\frac{\sqrt{D^2 + d^2}}{4}$
Ellipse about minor axis . .	$\frac{\pi D^3 d}{64}$	$\frac{D}{4}$
„ „ major axis . . .	$\frac{\pi d^3 D}{64}$	$\frac{d}{4}$

—PROF. GOODMAN.

208. RADIUS OF GYRATION.

The moment of inertia (I) divided by the area of the section (A) gives the square of the radius of gyration (r)

$$r^2 = \frac{I}{A}, \text{ whence } r = \sqrt{\frac{I}{A}}$$

This is used in ascertaining the strength of struts and columns.

For solid rectangle . . .	$r^2 = \frac{d^2}{12}$
Thin hollow square . . .	„ $= \frac{d^2}{6}$
Thin hollow rectangle . .	„ $= \frac{d^2(d + 3b)}{12(d + b)}$
Angle, tee or cross-section, equal sided	„ $= \frac{d^2}{24}$

Angle or tee section, unequal sided ,, $= \frac{d^2 b^2}{12 (d^2 + b^2)}$

Channel vertical section— .

A = area of flanges }
B = area of web } . . ,, $= \frac{A}{12 (A + B)} + \frac{A B}{4 (A + B)^2}$

Rolled joist section as pillar (w = width flange) . . . ,, $= \frac{w^2}{12} \times \frac{A}{A + B}$

Solid cylinder . . . . ,, $= \frac{d^2}{16}$

Thin hollow cylinder . . . ,, $= \frac{d^2}{8}$

Solid rectangular strut . . r = ·289 side.

Solid cylindrical strut . . r = ·25 diam.

Hollow cylindrical column . . d = $12t$, $10t$, $8t$, $6t$, $4t$

r = ·$325d$, ·$32d$, ·$313d$, ·$301d$, ·$279d$

Equal sided angle bar, approx. . r = ·2 side.

209. ELLIPSE OF INERTIA.

If the greatest and least radii of gyration are known the radius in any other direction can be ascertained graphically by describing an ellipse of which the major and minor axes are equal to twice the greatest and least radii of gyration respectively. This is the *ellipse of inertia*, or *ellipse of stress.*

210. MODULUS OF SECTION.

The modulus of a section is a function of the dimensions proportional to the moment of inertia of the section. It is the moment of inertia divided by the distance from the neutral axis to the furthest part on the extended or compressed side.

$$Z = \frac{I}{y_t}, \quad Z_c = \frac{I}{y_c}.$$

Maximum stress tension or compression × modulus of section is the Moment of Resistance and is = Bending Moment.

$$M = f_t\, z_t = f\, z_c.$$

—UNWIN.

Z for solid rectangular section $= \frac{b h^2}{6}$.

Z for solid circular section $= \frac{\pi r^3}{4}$.

211. BENDING MOMENT, OR MOMENT OF FLEXURE

and called by some writers "Moment of Rupture," is the moment of the external forces on one side of a transverse section estimated relatively to the section.

In beams and girders it varies according to the position of load and mode of support—e.g. a beam supported at the ends and loaded in the centre.

$$M_c = \frac{W l}{4};$$

if load be uniformly distributed

$$M_c = \frac{W l}{8}, \text{ or } \frac{w l^2}{8};$$

and if load be distributed over a length z in centre of span

$$M_c = \frac{W z}{8} + \frac{W (l-z)}{4}.$$

The bending moment M at a section is equal to the stress at one inch from the centre of gravity of the section multiplied by the moment of inertia I of the section.

$\frac{M}{I}$ = stress at one inch from neutral axis.—UNWIN.

In a flanged beam $\frac{\text{bending moment}}{\text{mean depth}}$ = stress in flange.

In a beam of any section, the stability depends upon the equation—

Bending Moment = Moment of Resistance, or M = R.

$M = \frac{E I}{\rho}$ expresses the relation between the bending moment and the curvature of a bar under transverse strain.—UNWIN.

212. MOMENT OF RESISTANCE.

The moment of resistance of a section is the moment of inertia multiplied by the modulus of rupture and divided by the distance of the neutral axis from the furthest edge of the section.

$$R = \frac{C\,I}{y} = \frac{I}{y} \times C = Z\,C$$

—HUMBER.

The moment of resistance of a beam at any section is the sum of all the products obtained by multiplying the actual longitudinal stress taken at each square inch of the section by its distance from the neutral axis. The moment of resistance in a flanged girder is the longitudinal strength of the weakest flange multiplied by the mean depth of the girder.—PERRY.

" The cross-section is the shape of the figure and the area that any material, such as a beam, would show, should it be cut into two pieces by a plane perpendicular to its length, and its *resistance to rupture* at this plane or section is the number of *inch-pounds* that its fibres will offer to forces tending to *cross-break* the beam or material of which it is a section : this is called the *Moment of Resistance of the cross-section*."—R. H. COUSINS.

213. WORKING LOAD FOR GIVEN MOMENT OF RESISTANCE.

f = greatest safe intensity of stress (assumed equivalent to C above).

Z = modulus of section.

For distributed load let $M = \frac{W\,L}{8} = f\,Z$, then $W = \frac{8\,f\,Z}{L}$.

For concentrated load let $M = \frac{W\,L}{4} = f\,Z$, then $W = \frac{4\,f\,Z}{L}$.

214. MODULUS OF TRANSVERSE RUPTURE (RANKINE).

$$f = \frac{M\,h}{I}.$$

The modulus of rupture does not coincide either with the tensile strength or with the compressive strength, and its value varies with the form of the cross-section of the material. Several theories have been propounded in explanation of this peculiarity, though none of them are entirely satisfactory. Rankine assumes one cause to be the fact that the resistance of a material to direct stress is increased by preventing or diminishing the alteration of its transverse dimensions. He also suggests that when a bar is directly torn asunder, the strength indicated is that of the centre part, which is the weakest, whilst when it is broken transversely the strength indicated is that of the outer part, which is the strongest. In any event, the difference really exists, and the modulus of rupture can only be determined experimentally. The unit value of the coefficient f is expressed in tons, cwts., or lbs. per sq. in.—e.g. :—

Cast iron rectangular bars 1 inch wide . . .	=	20 tons
Wrought iron I beams	=	27 „
„ T „	=	24 „
„ ⊥ „	=	23 „
Mild steel I beams	=	32 „

"The modulus of rupture is a fictitious stress value of the maximum stress at the instant of rupture, and is obtained by making the false assumption that $R = \frac{f\,b\,d^2}{6}$ beyond the elastic limit. But after the elastic limit of any wood is reached, the subject becomes very complicated. It has never been fully dealt with, and for practical purposes it is necessary to accept the modulus of rupture as calculated and deduced from experiment."—W. H. BETAMBEAU.

The theoretical value of this modulus is the resistance of the material to direct compression or tension, but it is found from experiments on cross breaking that this value is, from various causes, not sufficiently high, and Professor Rankine has adopted a modulus which is 18 times the load required to break a bar of 1 square inch section, supported on two points 1 foot apart, and loaded in the middle between the supports.

C =	lbs.		lbs.
Cast iron . . .	40,000	Fir .	5,000 to 10,000
Wrought iron . .	42,000	Oak .	10,000 to 13,600

C varies also to some extent with the form and area of the cross section. —HUMBER.

C in tons =

Cast iron . . .	$17\frac{3}{4}$	Oak . .	$5\frac{1}{4}$
Wrought iron . . .	$18\frac{1}{2}$	Fir . .	$3\frac{1}{4}$
Mild steel . . .	20		

C is also known as f, but f is better retained for the direct tensile or compressive strength.

The modulus of rupture for transverse strength is C in the formula

$$M = \frac{W\,l}{4} = Z\,C \text{ (W being in lbs. and } l \text{ in ins.).}$$

The coefficient of transverse strength is c in the formula

$$W = \frac{c\,b\,d^2}{L} \text{ (W and } c \text{ being cwts., } b \text{ and } d \text{ in ins., and L in ft.).}$$

In general $C = 18\,c \times 112$, C is the so-called extreme fibre stress = K of Molesworth, k of Tredgold, and f of other writers, and c is the weight in cwts. in centre required to fracture a bar 1 inch square and 1 foot long.

A further comparison may be made in the case of cast iron.

Average tensile strength per square inch .	=	16,500 lbs.
Average compressive strength . . .	=	99,000 lbs.
Average resistance to shearing . . .	=	27,700 lbs.
Average modulus of rupture (C) . . .	=	44,000 lbs.
Coefficient of transverse strength (c) . .	=	2,445 lbs.

(Note that C varies from the average tensile strength in flanged beams to 2¼ or 2½ times that in rectangular beams.)

215. SCANTLINGS FOR LOAD PER FOOT SUPER.

In designing floors the scantling of fir beams or joists to carry a uniformly distributed load per foot super. will be found as follows :—

$b\,d$ = breadth and depth in inches.
L = span in feet.
D = distance centre to centre of beams in feet.
W = safe load in cwts. per foot super. on floor allowing a factor of safety of 7.

$$b\,d^2 = W\,D\,L^2, \quad W\,L\,D = \frac{b\,d^2}{L}, \quad W = \frac{b\,d^2}{D\,L^2}.$$

216. FENWICK'S SOLID OF EQUAL RESISTANCE.

"Instead of a beam with a constant section, let us suppose that we have a solid whose cross-section varies progressively throughout its entire length.

"Then if M be the moment of the exterior forces which act on a section of the solid, I the moment of inertia of this section about the neutral axis of the section, and x the distance of a fibre of the section from the neutral axis, the tension S per unit of surface of the section is $S = \frac{M\,x}{I}$.

"Now from one point to another of *this section*, S varies with x and attains its greatest value at the same time that x attains its greatest value. From one section to another of the solid this maximum value of S may in general vary with M, I, and the maximum value of x. When S *is constant on all sections*, the body is said to be a *solid of equal resistance*.

"Amongst all bodies of equal tenacity, the one of equal resistance has the least quantity of matter; the solid of equal resistance, therefore, is the most suitable for practical constructions."—FENWICK'S "MECHANICS OF CONSTRUCTION, 1861."

With uniform breadth a solid cantilever loaded at the end will have straight top and semi-parabola for bottom, vertex under load. With uniform depth the outline of plan will be a triangle, and so on for other conditions of load and support.

217. COMBINATION OF LONGITUDINAL AND TRANSVERSE STRESS.

$$S = \frac{M c}{I \pm \frac{a F l^2}{b E}} \qquad \left(\begin{array}{l} + \text{ for tension} \\ - \text{ for comp.} \end{array} \right)$$

S = increase in extreme fibre stress due to transverse load lbs. per square inch.

M = bending moment due to ditto in lb.-ins.

c = distance from neutral axis to extreme fibres in inches.

I = moment of inertia of section of beam.

F = direct force of tension or compression lbs.

l = length of beam in inches.

E = modulus of elasticity in lbs.

A = area of section in square inches.

$\frac{F}{A}$ = fibre stress due to direct force tension or compression.

$S + \frac{F}{A}$ = total fibre stress under combined loading.

a and b are constants depending on method of supporting beam and nature of load.

Supported ends loaded centre $\frac{a}{b} = \frac{1}{12}$

" " load distributed $\frac{a}{b} = \frac{1}{9 \cdot 6}$

—PENCOYD.

218. POISSON'S RATIO.

Poisson's ratio is the ratio of the lateral swelling to the longitudinal shortening, or of the lateral contraction to the longitudinal lengthening, according as the stress is compressive or tensile.

For building stones . . .	= 0·091 to 0·345
„ mild steel	= 0·286
„ lead	= 0·4282
„ indiarubber	= 0·5

—W. DUNN.

Poisson's ratio (σ) is the ratio of sectional contraction to elongation—

$$\sigma = \frac{E}{2\,N} - 1$$

219. MODULUS OF RIGIDITY, OR SHEAR MODULUS,

is the ratio between the shear stress, in lbs. per square inch, and the shear strain, or movement of a particle, in inches at one inch from the fixed end.

$$N = \frac{\text{Stress}}{\text{Strain}}$$

approximately for wrought iron = 5,000 tons per square inch, and for steel = 5,400 tons per square inch.

Shear stress on a beam is measured by the angle through which the side of a square is distorted by the stress.

$$N = \frac{\text{Shear stress applied}}{\theta \text{ radians.}}$$

The torsional resistance of any material is proportional to the modulus of rigidity.

Shearing stress on a shaft is measured by the turning force × twice the radius = the torque (T).

$$T = \frac{f\,\pi\,r^3}{2} \text{ for round shaft,}$$

where f = maximum shear stress lbs. per square inch.

r = radius of shaft in inches.

The angle of twist is proportional to the torque applied, directly proportional to the length, and inversely proportional to the diameter.

220. RESILIENCE.

Resilience or *Spring* is the quantity of mechanical work required to produce the proof stress on a given piece of material, and is equal to the product of the proof strain or alteration of figure, into the mean load which acts during the production of that strain: that is to say, in general, very nearly one-half of the proof load. In other words, it is the work done in stretching or compressing a body up to its elastic limit.

The *Resilience* or *Spring* of a *Beam* is the work performed in bending it to the proof deflection:—in other words, the energy of the greatest shock which the beam can bear without injury: such energy being expressed by the product of a weight into the height from which it must fall to produce the shock in question. This, if the load be concentrated at or near one point, is the product of half the proof load into the proof deflection. —RANKINE.

The resistance of beams to transverse impact, or a suddenly applied load, is termed their resilience. It is simply proportional to the mass or weight of the beam, irrespective of the length or the proportion between the depth and breadth. Thus, if a given beam break with a certain steady load, a similar beam of twice the length will break with half the load applied in the same way; but if the short beam be deflected or broken by a certain falling load, the long beam will require double the load dropped from the same height or the load dropped from twice the height, to produce the same effect. —ANDERSON'S "STRENGTH OF MATERIALS."

The work done in deforming a bar up to the elastic limit is termed the resilience of the bar.—UNWIN.

The resilience of a bar $= \dfrac{f^2}{2\,E}$ in inch-tons per cubic inch of material; f being elastic limit in tons per square inch, and E modulus of elasticity.

The resilience of a bar of good wrought iron 1 square inch sectional area and 10 inches long is about 0·08 inch-tons, and the energy expended to produce rupture is about 55 inch-tons.—J. DUNCAN.

The resilience of a bar, or other body of uniform section, is the mean stress in lbs. per square inch × alteration of length in inches.

Approximately mean stress, lbs. per square inch

$$= \frac{\text{ultimate stress, lbs.} \times \text{sectional area, square inches}}{2}$$

Alteration of length in inches

$$= \frac{\text{ultimate stress, lbs.} \times \text{length inches}}{\text{modulus of elasticity (E)}}$$

$$\text{Resilience} = \frac{\text{stress} \times \text{area}}{2} \times \frac{\text{stress} \times \text{length}}{\text{E}}$$

$$= \frac{\text{stress}^2 \times a \times l}{2\,\text{E}}$$

but $a \times l =$ the volume of bar

$$\therefore \text{ Resilience} = \tfrac{1}{2} \text{ vol. of bar} \times \frac{\text{stress}^2}{\text{E}}.$$

221. MODULUS OF RESILIENCE.

The modulus of resilience is the coefficient $\frac{f^2}{\text{E}}$ which is multiplied by half the volume of a bar to obtain the *resilience* or *work done* in stretching it to the proof strain. The coefficient is obtained as follows:—Elongation of bar under proof load is $a\,n = \frac{f\,x}{\text{E}}$; force acting through this space varies from 0 to f S, with a mean value of $\frac{f\,\text{S}}{2}$; the work done is therefore $\frac{f\,x}{\text{E}} \times \frac{f\,\text{S}}{2} = \frac{f^2}{\text{E}} \times \frac{\text{S}\,x}{2}$, where $f =$ force applied in lbs. per square in., $x =$ length of bar in inches, E = modulus of elasticity in lbs., S = sectional area of bar in square inches.

The modulus of resilience can be computed in inch-lbs. per cubic inch by dividing the square of the proof tenacity by the modulus of elasticity—e.g.—

$$\text{Mild steel} \begin{cases} \text{Proof tenacity} \quad . = \quad 30{,}000 \text{ lbs.} \\ \text{Mod. elasticity} \quad . = \quad 29{,}000{,}000 \text{ lbs.} \end{cases}$$

$$\text{Modulus of resilience} = \frac{30{,}000^2}{29{,}000{,}000} = 31\cdot03.$$

R = Modulus of resilience in inch lbs.
W = load in lbs. per square inch sectional area;
E = Modulus of direct elasticity.
e = elongation in inches.
l = length in inches.

$$\text{R} = \text{W}\,\frac{e}{l}.$$

222. EFFECT OF IMPACT OF LOAD ON BEAM.

D = dynamic deflection due to fall of load P on centre of beam supported at ends.
d = static deflection due to load P.
T = extreme fibre stress due to fall of load P.
S = extreme fibre stress due to static load P.
W = weight of beam.
P = weight of load.
h = height of fall.

$$D = d + \sqrt{2mhd + d^2}$$

$$T = S\left(1 + \sqrt{\frac{2mh}{d} + 1}\right)$$

$$m = \frac{35P}{35P + 17W}.$$

—Pencoyd.

223. TESTING WROUGHT IRON.*

The strength of a bar should be measured by the *work* done in producing rupture—i.e. the product of the elongation into the mean stress. A convenient approximation to relative toughness is obtained by observing the maximum stress and the elongation in a given length. The length usually taken is 8 inches, but 6¼ inches is now sometimes adopted, so that the increase of length in sixteenths of an inch will represent the elongation per cent. The elongation being principally local, the percentage specified for a length of 8 inches$\times\frac{128}{100}$ or 1·28, will give the proper percentage for a length of 6¼ inches.

224. LENGTH OF TEST PIECE.

The longer the piece, the less will be the apparent elongation per cent. The comparative percentage for similar elongations will be approximately—

Length	10″	8″	6¼″
	100	125	160
	80	100	128
	62	78	100

* See leaflet by the author on "The Behaviour of Materials under Strain."

The following formula for comparative elongation is based upon experiments, but may be only approximate, as the localisation of the extension will vary according to the hardness and other properties of the material.

Let p = percentage elongation at 8 inches.
l = any other length between gauge points.
c = corresponding percentage elongation.

$$\text{Then approximately } c = \frac{p}{6{\cdot}3}\left(\frac{52{\cdot}5}{l + 0{\cdot}35}\right).$$

225. MECHANICAL WORK TO PRODUCE FRACTURE IN TENSION.

40 to 45 ton Bessemer steel	40 to 45 inch tons
3 ton mild steel	55 „ 63 „
First rate Yorkshire bar iron	50 „ 55 „
Ordinary wrought iron plate	12 „ 25 „

—PROF. KENNEDY.

"Dr. (Sir William) Fairbairn regarded the mechanical work done in breaking a bar under tensile stress as being of the highest importance in the consideration of the quality of metal best suited to resist a strain analogous to that of impact."—E. RICHARDS.

The work done in testing may also be stated "per cubic inch" of the specimen, measured between the gauge points, being about 12 inch-tons for best Yorkshire iron and 14 inch-tons for mild steel.

226. TENSILE STRENGTH OF WROUGHT IRON.

Result of 587 experiments by Kirkaldy:—

Number of experiments.	*Breaking weight per sq. in. of original area.*		
	Highest.	*Lowest.*	*Mean.*
188 rolled bars . . .	30·7 tons	19·9 tons	25·7 tons
72 angles and straps . .	28·5 „	16·9 „	24·4 „
167 plates, lengthwise . .	27·9 „	16·7 „	22·6 „
160 plates, crosswise . .	27·1 „	14·5 „	20·6 „

227. USUAL ALLOWANCE FOR DEAD LOAD PER SQUARE INCH SECTIONAL AREA.*

Material.	*Breaking Strain.*	*Safe Load.*
WROUGHT IRON—		
Tension	22 tons	5 tons
Compression	18 "	4 "
Shearing	20 "	4 "
Bearing	—	7½ "
MILD STEEL—		
Tension	30 "	7½ "
Compression	26 "	6 "
Shearing	24 "	5 "
Bearing	—	10 "
CAST STEEL—		
Tension	35 "	8 "
Compression	50 "	12 "
Shearing	—	7½ "
Bearing	—	15 "
CAST IRON—		
Tension	7 "	1½ "
Compression	42 "	7½ "
Shearing	14 "	2½ "
Bearing	—	8 "

The compression and shearing values assume that the parts are unable to bend, as in the case of short specimens.

The ordinary figures given for strength of materials are for dead, quiescent, or gradually applied loads. A live load may be considered to distress a beam more than a dead load in the ratio $\frac{3}{2}$ or $\frac{8}{5}$ = 1·5 to 1·6. A quickly-applied load may distress the beam more than a dead load in a ratio varying from 1·5 to 2 according to the speed of application. A suddenly applied load causes double the stress. A load dropped on or applied with impact causes a greatly increased stress according to its velocity. When the stress varies

* See paper by the author on "Strength of Iron and Steel." Demy 8vo, 16 pp. and folding plates. Spon, 6d.

in character or intensity due to periodic changes in the load, special allowances have to be made, as in the following table.

228. MAXIMUM WORKING STRENGTH IN TONS PER SQUARE INCH (WÖHLER'S EXPERIMENTS, 1860-70).*

Material.	*Constant Load.*	*Variable Load.*	
Wrought Iron for Machinery.	Tension only 5. Compressn. only 4.	Tension only 3. Compressn. only $2\frac{1}{2}$.	Alternate Tension and Compression $1\frac{1}{2}$.
Mild Steel for Machinery.	Tension only $7\frac{1}{2}$. Compressn. only 6.	Tension only $4\frac{1}{2}$. Compressn. only $3\frac{3}{4}$.	Alternate Tension and Compression $2\frac{1}{4}$.
Cast Iron for Machinery.	Tension only $1\frac{1}{4}$. Compressn. only 6.	Tension only $\frac{3}{4}$. Compressn. only $4\frac{1}{2}$.	Alternate Tension and Compression $\frac{3}{8}$.

229. WÖHLER'S LAW.

The fracture of any piece of material may be produced either by a single application of a load or by repeated application of a much smaller load. In either case the work necessary to cause fracture is the same. Kirkaldy found that under a sudden application of stress the ultimate strength of wrought iron was reduced about 18 per cent.

230. GERBER'S PARABOLIC EQUATION.

Analysing Wöhler's experiments, Gerber constructed a formula to agree with them, showing that with the same range of stress the maximum safe

* See also Min. Proc. Inst. C. E., for April, 1906, on "Reversals of Direct Stress."

stress under an infinite number of repetitions was in the proportion of 3 for statical constant load, 2 for variable load of the same character, tension or compression, 1 for alternating load of tension or compression. As the range is increased so the maximum stress must be reduced to retain the same limit of safety.

231. ULTIMATE STRENGTH OF VARIOUS METALS AND ALLOYS.

Name.	*Tension. Tons per sq. inch.*	*Compression. Tons per sq. inch.*
Mitis iron (cast)	27	..
Aluminium	8 to 18	5
Phosphor bronze	25	40
Delta metal	23	..
Muntz metal	20	..
Malleable cast iron	15	45
Copper (wire)	25	..
Copper (sheet and bolt)	15	40
Copper (cast)	10	..
Gun metal	12	48
Brass	10	25
Cast iron for machinery (strongest)	9	45
Zinc	3	15
Tin	2	6
Cast lead	$1\frac{1}{2}$	3

232. TESTING CAST IRON.

"The best and most certain test of the quality of a piece of cast iron is to try any of its edges with a hammer; if the blow of the hammer make a slight impression, denoting some degree of malleability, the iron is of good quality, provided it be uniform; if fragments fly off and no sensible indentation be made, the iron will be hard and brittle. The utmost care should be employed to render the iron in each casting of an uniform quality, because in iron of different qualities the shrinking is different, which causes an unequal tension among the parts of the metal, impairs its strength, and renders it liable to sudden and unexpected failures. When the texture is not uniform, the surface of the casting is usually uneven where it ought to have been even.

This unevenness, or the irregular swells and hollows on the surface of a casting, is caused by the unequal shrinkage of the iron of different qualities."
—TREDGOLD.

233. SPECIFICATION TESTS OF CAST IRON.

Three bars, each 3 feet 6 inches long, 2 inches deep and 1 inch wide, to be cast on edge in dry mould from each melting at which any of the specified work is cast. These bars to be tested separately as follows:—The lower side, or thin edge, of the casting to be placed downwards* upon rigid bearings, with 3 feet clear span, each bar to deflect not less than $\frac{3}{10}$ inch with a load of 25 cwt. in centre having a bearing not more than 1 inch wide upon the bar, to break with a minimum load of 28 cwt. and an average upon the three bars of not less than 30 cwt.

Samples prepared in lathe to bear $2\frac{1}{2}$ tons per square inch tensile strain before loss of elasticity, and to break with not less than 7 tons per square inch, or an average on three samples of $7\frac{1}{2}$ tons.

Test bars are sometimes cast as projections from an important casting and broken off for testing, but this is a bad method, and gives 10 to 20 per cent. lower results.

234. TESTS OF CAST IRON FOR PIPE-MAKING.

" A bar of metal 40 inches long, 2 inches deep and 1 inch wide, the weight of which must not exceed 21 lbs., shall, when supported on edge at points 36 inches apart, sustain a load of 3,000 lbs. supported at the middle of its bearing for one hour, and shall under this load deflect at least $\frac{3}{8}$ inch at the middle; and a bar 8 inches long and 1 inch square in section shall sustain a load of 8 tons tensile stress for one hour."

Note.—The test bar should deflect $\frac{1}{10}$ inch with 10 cwt., and recover its position when the load is removed. See also under Hydraulic Machinery.

235. COMPARISON OF TEST BAR AND CASTING.

" When test bars are run separately, they are almost invariably cast hotter than the castings they are to represent, and this has a material effect on the strength of the test bar. This is in favour of the test bar as compared

* Placed the other way up a reduction of 15 to 20 per cent. in the apparent strength may occur.

with the casting, which is usually cast dull. The reason foundrymen cast heavy castings with iron below the hottest and most fluid condition, is because they desire to avoid the searching action of hot iron by which the carbonaceous material of the mould would be burned out, and the iron still have enough fluidity left to take the place of the burnt-out carbon. The result would be a rough casting. One other reason is, a widespread belief that hot iron readily causes scabbing of the mould ; and another reason is, that where feeding a casting is required, casting dull shortens the period required for feeding. Foundrymen get very skilful in judging the temperature and fluidity of the iron so as to fill the mould perfectly and yet avoid the dangers mentioned."—R. BUCHANAN.

236. STRENGTH OF CAST IRON.

Mr. W. J. Keep, of Detroit, Mich., investigated the result of thirty-one sets of tests (Proc. Inst. Mech. Eng., June, 1904), and gave the following summary :—

A variation of size of a casting causes a great variation in strength, because of the change in the rate of cooling.

A variation of shape of castings which have the same area of cross section causes a great variation in strength.

It is very difficult to calculate the strength of one form or size of test-bar from the measured strength of another size.

A test-bar should be cast horizontally in the ordinary way and in ordinary sand, the same as other castings.

The average strength of at least two test-bars cast together should be taken.

The distribution of metal in a square test-bar gives a stronger casting than in a round bar of the same area of cross section, and more nearly represents the ordinary shape of castings.

A test-bar 1 inch square is the size and shape in general use.

Transverse or tensile strength is reckoned as so much per square inch.

237. SHEARING ANGLE OF CAST IRON.

Shearing angle of cast iron under compression ranges from 32° to 42° from direction of crushing force (HODGKINSON). The greatest intensity of shearing stress is on a plane making an angle of 45°, the deviation of the shearing plane from this shows that the resistance is not pure shear but partly due to a force analogous to friction (RANKINE). The ratio of length to diameter of test specimens for compression should not be less than 3 to 2 to allow free play for shearing action (HODGKINSON).

238. STRENGTH OF MALLEABLE CAST IRON.

Ultimate tensile strength per square inch = 14 tons.
Elongation on 4 inches = $1\frac{1}{2}$ per cent.
Elastic limit = 7 tons.
Ultimate compressive strength per square inch = 50 tons.
Elastic limit = 5 tons.

239. SHEARING STRENGTH COMPARED WITH TENSILE STRENGTH.

Is variable, but averages for—

Wrought iron . .	85 per cent.	Mild steel . .	81 per cent.
Cast iron	40 ,,	Hard steel .	64 to 70 ,,

—PLATT *and* HAYWARD.

240. APPROXIMATE STRENGTH OF GIRDERS.

Safe load in tons distributed when supported at both ends and loaded uniformly.

For cast iron = Sectional area of bottom flange in square inches.
For wrought iron = Gross sectional area of bottom flange plates × $2\frac{2}{3}$.
For rolled iron joist = Area one flange × 4 × depth inches ÷ span feet.
For steel joist = Area one flange × 5 × depth inches ÷ span feet.

241. GIRDER LOADED OUT OF CENTRE.

Load in centre = W. equivalent load out of centre = W_1, with span divided into x and y.

$$\frac{W_1}{W} = \frac{\left(\frac{x+y}{2}\right)^2}{x \times y}$$

242. BRIDGES AND GIRDERS.*

$A' =$ area of one flange in square inches at centre.
$a =$ „ „ at x feet from one end.
$D =$ depth in feet at centre.
$d =$ „ „ at intermediate points.
$S =$ span in feet.
$W =$ load in tons concentrated in centre.
$c =$ constant $=$ stress per square inch allowed in flange:

$$W = \frac{A D c}{\frac{1}{4} S}, \quad A = \frac{\frac{1}{4} W S}{D c}, \quad D = \frac{\frac{1}{4} W S}{A c}.$$

To find section required at any given distance from one end $= a$,

$$a = \frac{A x (S - x)}{(\frac{1}{2} S)^2}.$$

—W. G. A. & Co., Elswick.

243. SPECIFICATION TESTS OF WROUGHT IRON (BRIDGE AND GIRDER WORK).

Class.	*Tensile Strength, tons per square inch.*	*Elongation per cent. in 8 inches.*	*Contraction per cent. at point of fracture.*
Rivet iron . .	25	10	30
Rod and bar iron . .	24	$7\frac{1}{2}$	20
Angle and tee iron . .	22	6	15
Plates, with grain . .	21	$4\frac{1}{2}$	10
Plates, across grain . .	18	2	5

244. ALLOWANCE IN BRIDGES FOR CHANGES OF TEMPERATURE.

Variation of 15° F. alters length of wrought iron as much as strain of 1 ton per square inch.

In exposed situations an allowance of $\frac{7}{16}$ of an inch movement, per 100

* For general designing, see the author's "Practical Designing of Structural Ironwork." Demy 8vo, cloth, 200 pp., with 14 folding plates, containing 180 diagrams. (Spon, 8s. 6d.)

feet length, is necessary for the purpose of eliminating the strains due to change of temperature.—GRAHAM SMITH.

245. SPECIFICATION TESTS—COMMON WROUGHT IRON.

Class.	*Tensile Strength, tons per square inch.*	*Contraction per cent. at point of fracture.*
Rivet iron	22	20
Rods, bars, and angles	21	12½
Plates	20	10

—TIMMINS.

246. STEEL AND IRON SHIPBUILDING.

Lloyd's Regulations allow a reduction of 20 per cent. in the scantlings of a steel ship as compared with iron, but the total weight of material used is only about 14 per cent. less. The cost is about the same in steel or iron.

247. SPECIFICATION TESTS OF WROUGHT IRON AND STEEL (SHIPBUILDING).

Class.	*Tensile Strength, tons per square inch.*	*Elongation* per cent. on fracture.*	*Toughness.†*
Rivet iron . . .	26	25	650
Rod and bar iron . .	24	15	360
Angle and tee iron . .	22	12½	275
Iron plates, with grain .	20	7½	150
,, ,, across grain .	19	6	114
Steel plates (both directions)	28	20	560
,, ,, bars and angles	30	25	750

* In a length of 6¼ inches.

† Should the actual elongation in sixteenths of an inch, multiplied by the stress in tons per square inch, upon rupture, be more than 10 per cent. under the amounts given in the last column, the material will be rejected.

Wrought Iron.—Cold bending in vice—½-inch plate 35°, ⅜-inch plate 55°, $\frac{5}{16}$-inch plate 63°, ¼-inch plate 70°, rivet iron to double close, without cracking.

Steel.—Steel plates should be capable of bending to an inside radius of 1½ times their thickness when heated to a low cherry red and cooled in water of a temperature of 58° C. = 82° F. For Admiralty tests, see "Molesworth."

Iron drawn into bars, rods and wire, has, other things being equal, a higher proportionate strength as the diameter is reduced, owing to the compression and hardening which take place, but if the rods, etc., be annealed after drawing down this increased strength disappears. Tables of strength of steel published before 1880, and some later, give the strength of steel much too high for present use, as the material has greatly altered with the mode of manufacture.

248. LIMITING ALLOWANCE FOR STRENGTH OF STEEL.

Admiralty rule—26 to 30 tons per square inch, with elongation of 20 per cent. in 8 inches.

Lloyd's rule—27 to 31 tons per square inch.

Liverpool Underwriters' Registry—28 to 32 tons per square inch.—FIDLER.

249. ADMIRALTY SPECIFICATIONS.

B.B., or first-class iron.

	Tensile stress tons per sq. in.		*Elongation per cent. in 8 ins.*
Rivet and bolt iron, and bars square, round, or flat	24	..	15
Angle, tee, channel, and flats of 12 in. width and under	22	..	10
Plates, lengthways	22	..	8
Plates, crossways	18	..	3

B., or second-class iron.

Bars, square, round, or flat	22	..	10
Angle, tee, channel, and flats of 12 in. width and under	21	..	8
Plates, lengthways	20	..	7
Plates, crossways	17	..	2½

250. COMPARATIVE STRENGTH OF IRON AND STEEL PLATES.

Quality.	*Ultimate Tensile Strength in tons per sq. inch.*		*Elongation per cent. in length of 8 inches.*	
	With Grain.	*Across Grain.*	*With Grain.*	*Across Grain.*
Mild steel . . .	28—30	28—30	20	20
Best Yorkshire iron.	24	22	12	7½
B.B. Staffordshire „ .	22	19	9	5
B. „ „ .	20	18	6	2½

The present tendency (1906) is to reduce the tensile strength of mild steel for structural work to 26–28 tons with the same elongation. The heaviest rolled sections are sometimes capable of 25 tons per square inch only.

The Farnley brand of best Yorkshire iron is more than double the price of good ordinary Staffordshire, but the quality is greatly superior, and it is less expensive to forge.

251. STRENGTH OF BUCKLED PLATES.

Usually of wrought iron or mild steel, 3 feet square and ¼ to ½ inch thick with transverse joint supported by ⊥ or other section.

D = total concentrated load in lbs.
g = uniform load in lbs. per square foot.
h = depth of buckle in inches, say 2 to 3 inches.
l = length of buckle in inches.
t = thickness in inches.
k = permissible stress in lbs. per square inch, say 6,000.
P = total uniformly distributed load in lbs. buckled plate can carry.

$$D = \frac{100\,k\,h\,t - 0{\cdot}175\,g\,l^2}{6\,h + 15\,t} \times t$$

$$P = 4\,k\,h\,t.$$

—PENCOYD.

252. TEST OF BOILER PLATES AND QUALITY OF MATERIAL.

The following specification and tests are recommended for land boilers by the National Boiler and General Insurance Co., Ltd., Manchester.

All the plates, angles, and rivets to be of ductile steel, made by Siemens-Martin process, free from laminations, pittings, or any other defects, with a tensile strength of not less than 26 tons and not more than 30 tons per square inch, except the furnace and flue tube plates, which must have 24 to 28 tons tensile strength, the elongation of all in a length of 10 in. to be not less than 20 per cent., and to be, together with the tests thereof, entirely to the satisfaction of the chief Engineer of the N. B. and G. I. Co., Ltd.

Tests as follow to be made by the plate-maker :—In the case of shell plates, one test strip shall be cut across from each end of the plate and tested for tensile strength, and shall comply with the limits specified in the foregoing. One strip shall be cut across each end of the plate, and may be machined on the edges if the plate-maker desires, and that from one end shall be tested by bending when in its normal condition, and that from the other end be tested by bending after heating to dull cherry-red heat and slaking in water at about 82° F. These strips shall stand bending without cracking until the sides are parallel, the radius in the corner being not greater than one and a half times the thickness of the plate.

In the case of the furnace plates, one test strip to satisfy the above tensile tests shall be cut from each plate in the part which is most convenient, and in addition two test pieces for bending tests shall be cut from each plate, and shall satisfy the same test as specified for the shell plates.

A list of the plates to be furnished by the maker, giving the consecutive or index number of each, with its breaking stress and elongation, as obtained by the testing strips cut therefrom.

These tests of the plates to be witnessed by one of the N. B. and G. I. Co.'s inspectors, when those plates found satisfactory by him will be stamped with the Company's stamp against the brand. The boiler-maker must satisfy himself that each plate is so stamped before using it in the construction of the boiler.

The number and the maker's name or trade-mark to be stamped on each plate, and to be encircled by a line of white paint, so that they may be easily found. The brand of steel to be used to be submitted for approval.

253. COMPARISON OF THE PHYSICAL AND CHEMICAL PROPERTIES SPECIFIED FOR STRUCTURAL STEEL (AMERICAN PRACTICE).

	Asso. Am. Steel Mfrs. Feb. 6, 1903. Structural Steel.	*Am. Soc. for Test Mat. Aug. 10, 1901. Steel for Buildings.*	*Am. Soc. for Test Mat. Aug. 10, 1901. Steel or Bridges and Ships.*	*Am. Ry. Eng. & Maint. Way Asso. March, 1903. For Steel Structures.*	*Am. Bridge Co.; now in force. Steel for Buildings.*
PHYSICAL PROPERTIES.					
Rivet Steel—					
Ultimate Strength . .	48,000 to 58,000	50,000 to 60,000	50,000 to 60,000	45,000 to 55,000	48,000 to 58,000
Elastic limit . . .	½ ult. strength	½ ult. strength	½ ult. strength	...	½ ult. strength
Elongation in 8 ins. .	1,400,000 ÷ ult. str.	26 per cent.	26 per cent.	1,500,000 ÷ ult. str.	26 per cent.
Bending test . . .	180 deg. flat	180 deg. flat	180 deg. flat	180 deg. flat	180 deg. flat
Steel for Railway Bridges—					
Ultimate strength . .	55,000 to 65,000	...	52,000 to 62,000	55,000 to 65,000	...
Elastic limit . . .	½ ult. strength	...	½ ult. strength	...	...
Elongation in 8 ins. .	1,400,000 ÷ ult. str.	...	25 per cent.	1,500,000 ÷ ult. str.	...
Bending test . . .	180 deg. over diam. = thickness	...	180 deg. flat	180 deg. flat	...
Medium Steel—					
Ultimate strength . .	60,000 to 70,000	60,000 to 70,000	60,000 to 70,000	...	55,000 to 65,000
Elastic limit . . .	½ ult. strength	½ ult. strength	½ ult. strength	...	½ ult. strength
Elongation in 8 ins. .	1,400,000 ÷ ult. str.	22 per cent.	22 per cent.	...	24 per cent.
Bending test . . .	180 deg. over diam. = thickness	180 deg. over diam. = thickness	180 deg. over diam. = thickness	...	180 deg. over diam. = thickness
CHEMICAL PROPERTIES.					
Steel for buildings, train sheds, highway bridges.	Not over 0·10 per cent. phos.	Not over 0·10 per cent. phos.	...	...	If acid not over 0·08 per cent., if basic 0·06 per cent. phos.
Steel for railway bridges .	Not over 0·08 per cent. phos.	...	If acid not over 0·08 per cent., if basic 0·06 per cent. phos.; not over 0·06 per cent. sulph.	If acid not over 0·08 per cent., if basic 0·04 per cent. phos.; not over 0·05 per cent. sulph.	...
Rivet steel	Not over 0·08 per cent. phos.	...	If acid not over 0·08 per cent., if basic 0·06 per cent. phos.; not over 0·06 per cent. sulph.	Not over 0·04 per cent. phos. or 0·04 per cent. sulph.	Same

—ENGINEERING RECORD.

254. COMPARISON OF THE PHYSICAL AND CHEMICAL PROPERTIES SPECIFIED FOR OPEN HEARTH PLATE AND RIVET STEEL (AMERICAN PRACTICE).

—	*Assoc. Am. Steel Mfrs., Feb.* 6, 1903.	*Am. Soc. Test. Mat. Aug.* 10, 1901.
PHYSICAL PROPERTIES.		
Extra Soft Steel—		
Ultimate strength .	45,000 to 50,000	45,000 to 55,000
Elastic limit . .	½ ultimate strength	½ ultimate strength
Elongation in 8 ins. .	28 per cent.	28 per cent.
Cold and quench bends	180° flat	180° flat
Firebox Steel—		
Ultimate strength .	52,000 to 62,000	52,000 to 62,000
Elastic limit . . .	½ ultimate strength	½ ultimate strength
Elongation in 8 ins. .	26 per cent.	26 per cent.
Cold and quench bends	180° flat	180° flat Also a homogeneity test.
Flange or Boiler Steel—		
Ultimate strength .	55,000 to 65,000	55,000 to 65,000
Elastic limit . .	½ ultimate strength	½ ultimate strength
Elongation in 8 ins. .	25 per cent.	25 per cent.
Cold and quench bends	180° flat	180° flat
Boiler rivet steel . .	Use "Extra soft steel"	Use "Extra soft steel"
CHEMICAL PROPERTIES.		
Extra soft steel . .	Not over ·04 % phos. and ·04 % sulphur.	Not over ·04 % phos. and ·04 % sul.; 0·30 to 0·50 % mangan.
Firebox steel . . .	Not over ·04 % phos. and ·04 % sulphur.	If acid not over ·04 %, if basic ·03 % phos.; not over ·04 % sul.; 0·30 to 0·50 % mang.
Flange or boiler steel .	Not over ·06 % phos. and ·04 % sulphur.	If acid not over ·06 %, if basic ·04 % phos.; not over ·05 % sul.; 0·30 to 0·60 % mang.

Form of test specimens.—Rivet rounds to be tested of full size as rolled. Other material to be parallel section and not less than ½ inch square section.

Material to be free from injurious seams, flaws, cracks, or defective edges, and have a clean, smooth finish. If the finished material is to be annealed the test specimen must receive the same treatment before testing.

Variation in Weight of more than 2½ per cent. from that specified may be cause for rejection.—"ENGINEERING RECORD."

Specification weights are usually calculated from dimensions, wrought iron 5 lb. per foot super. per ⅛ inch thick, mild steel 2 per cent. additional.

255. TESTS OF IRON AND STEEL.

PHYSICAL.

Brand.	*Point of Permanent Set in tons per square inch.*	*Tension in tons per square inch.*	*Elongation per cent.*
Lowmoor iron	..	25·50	42·15
Staffordshire iron . . .	16·82	25·57	27·50
Mild steel	17·92	28·86	45·00
Medium steel . . .	20·87	33·25	35·92
Hard steel	25·60	39·84	30·50
Tool steel	..	57·68	14·40
Very hard steel . . .	..	68·67	7·00

CHEMICAL.

Brand.	*C.*	*Mn.*	*Si.*	*P.*	*S.*
Parkhead common iron .	·09	trace	·020	·316	·027
Leeds forge best iron . .	·14	·03	·110	·085	·028
Bowling best iron . .	·11	trace	·10	·101	trace
Farnley best iron . .	·11	·01	·090	·096	·012
Lowmoor best iron . .	·10	·01	·120	·142	·022
Landore mild steel . .	·18	·64	·013	·077	·074
Mild steel	·22	·399	·062	·043	·042
Medium steel	·34	·536	·024	·052	·019
Tool steel	·97	·148	·074	·034	·059

256. ANKARSRUMS (SWEDISH) CAST IRON.

Guaranteed tensile strength	=	17·8 tons per square inch.
Average " "	=	19·5 " "
Extension on 4 inches .	=	0·28 per cent.

—WESTMAN.

257. TESTS OF STEEL FOR BRIDGE BUILDING.

For compression members :—

Sample bars $\frac{3}{4}$ inch diameter to bend cold 180° round same diameter. Elastic limit 50,000 lbs. per square inch. Ultimate strength 80,000 lbs. per square inch. Elongation 15 per cent. in 8 inches. Reduction of area at point of fracture 35 per cent. Chemical tests, carbon ·34 to ·42 per cent., phosphorus ·1 per cent.

For tension members :—

Sample bars $\frac{3}{4}$ inch diameter to bend cold 180° round same diameter, and set back again by hammer without flaw. Elastic limit 40,000 lbs. per square inch. Ultimate strength 70,000 lbs. per square inch. Elongation 18 per cent. in 8 inches. Reduction of area at point of fracture 42 per cent. Chemical tests, carbon ·25 per cent., phosphorus ·1 per cent.—LOUISVILLE AND NEW ALBANY BRIDGE, 1886.

258. TESTS OF MILD STEEL.

Chemical composition, carbon ·192, silicon trace, sulphur ·040, phosphorus ·048, copper ·021, manganese ·430, iron 99·269—total 100.

Tensile test, turned specimen 1 square inch sectional area.

Elastic limit or yield point 17·41 tons per square in.

Maximum stress 28·35 tons, elongation at this load 18·75 per cent. in 8 inches, and sectional area reduced 18 per cent., making cohesive strength at this point 34·32 tons per square inch.

Stress at time of fracture 25·05 tons, ultimate extension 27·5 per cent., sectional area being then reduced 44·59 per cent., making the cohesive strength 45·21 tons per square inch.

If the maximum stress be calculated on the final sectional area the cohesive strength would appear as 51·16 tons per square inch.

The mechanical work done in breaking this sample = 55·85 inch tons.

Fracture silky.—E. RICHARDS.

259. COMPOSITION OF STEEL FOR ELECTRIC TRAM RAILS.

Specified by British Standards Committee.

Carbon	from 0·40 to 0·55	per cent.
Manganese	„ 0·70 „ 1·00	„ „
Silicon	not to exceed 0·10	„ „
Phosphorus	„ 0·08	„ „
Sulphur	„ 0·08	„ „

Carbon makes the steel hard, and increases its resistance to wear; 0·45 should be the lowest permitted. Manganese reduces the electrical conductivity of steel, and 0·90 per cent. ought to be the maximum allowed. Phosphorus makes the steel brittle, and it should be kept down to 0·07 per cent.

260. STEEL FOR RAILWAY RAILS

Should contain per cent.—

	Min.		*Max.*
Carbon	0·35	to	0·5
Silicon	0·05	„	0·10
Sulphur	0·04	„	0·08
Phosphorus	0·00	„	0·08
Manganese	0·75	„	1·00
Iron	remainder.		

261. TESTING RAILWAY MATERIAL.*

Certificate of Inspection given with each delivery. 5 per cent. of the material to be tested for weight, $2\frac{1}{2}$ per cent. variation allowed. No error of shape or position to exceed $\frac{1}{32}$ inch; $\frac{1}{4}$ inch clearance allowed between ends of rails.

Rails.—Charge or blow number to be stamped on every web 18 inches from ends, $\frac{1}{2}$ per cent. to be subject to tensile and drop tests. Tensile test from rail heads 35 to 40 tons per square inch, elongation 12 per cent. in 8 inches. Drop test to vary with rail section, supports 3 feet apart, moment of first blow to be about (wt. lbs. per yard) $^2 \times 6 =$ ft.-lbs., and remaining blows lighter until deflection $= \dfrac{20}{\text{height of rail ins.}}$

Fish plates.—Tensile test 28 to 30 tons.

* See Bodmer's "Inspection of Railway Materials." (Whittaker & Co., 5s.)

262. TESTING INDIARUBBER.

Indiarubber should not give the slightest sign of superficial cracks on being bent to an angle of 180 deg. after five hours' exposure in a closed air bath to a temperature of 125° C. (= 257° F.). The test pieces should be about 6 centimetres (= $2\frac{3}{8}$ inches) thick.

Rubber containing not more than 50 per cent. by weight of metallic oxides should stretch to five times its length without breaking.

Pure caoutchouc free from all foreign matter, except the sulphur necessary for its vulcanisation, should stretch seven times its length without breaking.

The extension measured immediately after rupture should not exceed 12 per cent. of the original length of the test-piece. The test pieces should be from 3 to 12 millimetres (= $\frac{1}{8}$ in. to $\frac{1}{2}$ in.) wide, and not more than 6 millimetres (= $\frac{1}{4}$ in.) thick and 3 centimetres (= $1\frac{3}{16}$ in.) long.

The percentage of ash gives a certain indication of the degree of softness, and may form a basis for the choice between different qualities for certain purposes.—LIEUT. VLADIMIROFF.

263. SUGGESTED MECHANICAL TESTS FOR INDIARUBBER.

Three test pieces shall be cut, each 2 inches × 2 inches × 1 inch.

1. This piece shall float in fresh water and be retained for future reference with the date and name of maker written on it in ink.

2. This piece shall be screwed up between two plates to a thickness of $\frac{5}{8}$ inch and left in a temperature between 50° and 70° F. (10° and 21° C.) for 24 hours, the permanent set or less of thickness one hour after release shall not exceed $12\frac{1}{2}$ per cent. = $\frac{1}{8}$ inch.

3. This piece shall suffer 10,000 machine blows at the rate of 120 per minute, each compressing it to $\frac{3}{4}$ inch thickness, with a permanent set not exceeding $12\frac{1}{2}$ per cent. and without cracking at the edges.

264. STRENGTH OF STRUCTURES.

The strength of structures varies as the square of the linear dimensions of similar parts, excluding the effect of weight; but the weight varies as the cube of the linear dimensions. The strength of a structure of any kind is not therefore to be determined by that of its model, which will always be much stronger in proportion to its size. All works, natural and artificial, have limits of magnitude which, while their materials remain the same, they cannot surpass.—LARDNER.

265. SAFE LOAD ON STRUCTURES.

Cast-iron columns	= $\frac{1}{4}$ breaking weight.
Cast-iron girders for tanks	= $\frac{1}{4}$ breaking weight.
Wrought-iron structures	= $\frac{1}{4}$ breaking weight.
Cast iron for bridges and floors	= $\frac{1}{6}$,,
Stone and bricks	= $\frac{1}{8}$,,
Timber, live loads	= $\frac{1}{10}$,,
Do., dead loads and temporary structures	= $\frac{1}{6}$,,

—MOLESWORTH.

266. SAFE LOAD ON FLOORS.

Churches and public buildings	$1\frac{1}{2}$	cwt.	per sq. foot.
Warehouses	$2\frac{1}{2}$	,,	,,
Dwelling houses	1	,,	,,

exclusive of weight of floor itself.

267. WEIGHT OF MEN IN CROWDS.

Mr. Cowper found by experiment that a number of men averaged 140 lbs. per square foot.

Mr. Parsey considers that men packed closely would weigh at least 112 lbs. per square foot, but that in ordinary crowds 80 lbs. might be taken as sufficient.

On the Continent it is not usual to estimate so high. Belgians weigh about 140 lbs. each, Frenchmen 136 lbs., while Englishmen weigh 150 lbs.

Mr. F. Young states 80 lbs. per square foot is quite safe in practice.

Mr. Thomas Page packed picked men on a weighbridge with a result of 84 lbs. per foot super.

Mr. George Gordon Page says that for troops on march $35\frac{1}{4}$ lbs. per square foot is sufficient.

The usual practice is to assume the live load as 100 lbs. per square foot. —A. T. WALMISLEY.

	lbs. per sq. foot.
French practice (quoted by Stoney and Trautwine) .	41
Hatfield in "Transverse Strains," for soldiers . .	70
Nash, architect of Buckingham Palace (quoted by Tredgold)	120
W. K. Kernot, Working Men's College, Melbourne .	126
Prof. W. C. Kernot, Melbourne University . . .	143·1
B. B. Stoney, in "Stresses"	147·4

—PROF. KERNOT.

100,000 people standing cover 6 acres = $2\frac{1}{2}$ sq. ft. each. A cart-horse weighs 18 cwt.

268. MAXIMUM LOAD DUE TO A CROWD.

A careful attempt to determine exactly how great a load of people may be crowded within any given space was carried out at Harvard University, U.S.A., by Prof. L. J. Johnson.

In obtaining these data, a box 6 feet square, provided with a gate at one side, was built, and a certain number of men, whose separate weights had been carefully taken, were placed within it. By dividing the aggregate weight of the men by the number of square feet within the box, the load per square foot was determined. In the first case the box was occupied by eleven men, whose average weight was 154·6 lbs. This gave a load per square foot of 47·2 lb., which is 2·2 lbs. more than the loading that has been assumed in the designing of some floors, platforms, and bridges. That this loading does not represent the weight of an average crowd is proved by the fact that, when the men were lined up side by side around the walls of the box, they covered only three sides of it. Twenty-eight men of an average weight of 167·7 lbs. were found to be equivalent to a load of 130·4 lbs. per square foot. With the men all facing one way, a result of 176·4 lbs. per square foot was obtained, and with forty heavy men in the box, all tall and carefully selected, the maximum reached was 181·3 lbs. per square foot.

269. FLAT CAST-IRON FLOOR PLATES.

$$\text{Thickness ins.} = \frac{\sqrt{\text{load lbs. sq. ft.}} \times \text{span ins.}}{380}$$

Another formula :—

$$\text{Thickness ins.} = \sqrt{\frac{\text{Distrib. load} \times \text{shorter span ins.}}{8000 \times \text{greater span ins.}}}$$

270. WROUGHT IRON FLOOR PLATES.

Edges not fixed, $t = \sqrt{\frac{\cdot 00009\ W\,l}{b}}$

Riveted two edges, $t = \sqrt{\frac{\cdot 000044\ W\,l}{b}}$

271. WEIGHT OF MATERIALS FOR ESTIMATING.

Wrought iron	480	lbs. per	cub. ft.
Cast iron	450	,,	,,
Gun-metal and brass	530	,,	,,
Cast steel	504	,,	,,
Mild steel	490	,,	,,
Lead	700	,,	,,
Copper	550	,,	,,
Zinc	450	,,	,,
Greenheart	60	,,	,,
Oak	50	,,	,,
Fir	40	,,	,,
Granite	160	,,	,,
Bramley Fall and Hard York	140	,,	,,

272. THEOREM OF THREE MOMENTS.

If A B C be three consecutive supports of a continuous girder of any number of spans, whether equal or unequal, and l_1 l_2 the consecutive spans; then let p_1 p_2 = the loads per unit of span on l_1 l_2 respectively; and M_1 M_2 M_3 = the bending moments on A B and C respectively. The relation between M_1 M_2 and M_3 is always expressed by the equation

$$M_1\,l_1 + 2\,M_2\,(l_1 + l_2) + M_3\,l_2 = \tfrac{1}{4}\,(p_1\,l_1^3 + p_2\,l_2^3).$$

273. LOAD ON THE SUPPORTS OF CONTINUOUS GIRDERS

of equal spans uniformly loaded, the load on each span being unity, and the supports perfectly level and rigid. The piers are counted from either end when loaded symmetrically.

No. of Spans.	*Abutment.*	*1st Pier.*	*2nd Pier.*	*3rd Pier.*	*4th Pier.*	*5th Pier.*	*6th Pier.*	*7th Pier.*
2	·375	1·25	..	..	..	..	..	..
3	·4	1·1	..	..	..	..	..	..
4	·393	1·143	·93	..	..	..	..	..
5	·394	1·131	·989	..	..	..	..	..
Infinite	·3943	1·134	·9641	1·0096	·9974	1·0007	·9998	1·00

When the number of spans exceeds five, the loads on the supports are nearly the same as when the number is infinite.

The division of total load with two spans will be $\frac{3}{16}$, $\frac{5}{8}$, $\frac{3}{16}$, and with three spans $\frac{4}{30}$, $\frac{11}{30}$, $\frac{11}{30}$, $\frac{4}{30}$.

274. APPROXIMATE SAFE LOAD ON COLUMNS AND PIERS.

Cast-iron column or stanchion with metal ¾ inch thick or upwards.

Up to 10	diameters long	5	tons per sq. inch.
10 to 15	,,	4	,,
15 to 20	,,	3	,,
20 to 25	,,	2	,,
25 to 30	,,	1½	,,
30 to 35	,,	¾	,,

If less than ¾ inch thick take ¼ ton per sq. inch less for each ⅛ inch less in thickness.

Hard York or Portland stone piers 12 tons per foot super.

Stock brick in cement, if covered with stone template . .	6	,,	,,
Do. without do. .	4	,,	,,
Cement concrete (5 to 1) . .	4	,,	,,

General thickness of concrete in foundations = ¼ width. For walls and piers width of concrete = 3 times thickness of wall, depth ¾ thickness of wall. Or projection 6 inches beyond bottom course of footings, thickness = 1½ times projection.

Safe tensile strength lime mortar 5 lbs. per square inch, cement mortar 20 lbs. per square in.

Lbs. per square inch × ·0643 = tons per square foot.
Tons per square foot × 15·55 = lbs. per square inch.
Brickwork weighs 112 to 120 lbs. per cubic foot.
Cement concrete weighs 130 lbs. per cubic ft.
450 stock bricks = 1 ton.
1,000 new bricks stacked dry = 56 cubic feet.

275. SAFE LOAD ON BRICK PIERS.

W = tons per square foot as given above.
r = ratio of height to least thickness.
S = safe tons per square foot.

$$S = W\left(\frac{24-r}{18}\right)$$

276. STRENGTH OF PRISMS IN COMPRESSION.

Prof. Bauschinger has expressed the compressive strength of prisms of different heights, but same sectional area as follows :—

$$s = a + b\frac{\sqrt{f}}{l}$$

s = compressive strength.
f = sectional area.
l = height of prism.
a and b = constants to be determined by experiment.

For dissimilar cross-sections he proposed the following—

$$s = \left(a + b\frac{\sqrt{f}}{l}\right)\sqrt{\frac{4\sqrt{f}}{u}}$$

where u = the circumference of the cross section, and other letters as before.

277. EFFECT OF LOAD NOT BEING AXIAL.

When the centre of pressure, or resultant of the forces acting on a cross section, does not coincide with the centre of gravity of the section the strength is reduced and the maximum stress increased as follows :—

W = total load tons.
d = distance of centre of pressure from neutral axis of section (i.e. line through centre of gravity).
A = area of section.
s = mean stress in tons.
S = maximum ,,
D = distance of point of maximum stress from neutral axis.
I = moment of inertia of the section.

$$s = \frac{W}{A}, \quad S = s\left(1 + d\,\frac{D\,A}{I}\right).$$

The same units, feet or inches, must be used throughout.

278. COMBINED THRUST AND BENDING MOMENT.

P = stress at edge of section, + for compression edge and — for tension edge, in tons per square inch.
W = total load in tons.
A = area of section in square inches.
M = bending moment in inch-tons.
Z = modulus of section in square inch units.

$$P = \frac{W}{A} \pm \frac{M}{Z}.$$

279. DEVIATION OF CENTRE OF PRESSURE,

or position of resultant, to reduce stress on one edge to zero, as in the case of chimney shafts, walls, etc.

A = area of bed joint.
y = distance from c.g. of figure of joint to edge farthest from centre of pressure.
h = total breadth of joint in same direction.

I = moment of inertia of the figure relative to neutral axis through c.g. at right angles to direction of deviation of centre of pressure.

d = deviation of centre of pressure when pressure becomes zero at opposite edge.

$$d = \frac{I}{A\,y}.$$

—RANKINE.

Under these circumstances the mean pressure will be doubled at edge nearest centre of pressure and reduced to nil at opposite edge.

280. CENTRAL ELLIPSE AND CORE OF A SECTION.

If from the centre of gravity of a horizontal solid rectangular section be set off a series of distances in every direction equal to the radius of gyration for those directions the curve drawn through the points so found will give the *central ellipse of the section.*

The *core of the section* is obtained by taking a distance from the centre of gravity in each direction $= \dfrac{\text{distance to ellipse}^2}{\text{distance to edge of section}}$.

In order that there should be no tension on the section when the resultant force acts out of the centre the resultant must not fall outside the core.

281. THRUST ON RETAINING WALL.

P = horizontal thrust at one-third height in lbs.

w = weight of supported material per cubic foot in lbs.

ϕ = angle of natural slope of supported material.

h = height of retaining wall in feet.

h_1 = height of surcharge in feet.

P_1 = total horizontal thrust in lbs. at one-third height, including that produced by surcharge.

$$P = \tfrac{1}{2}\,w\,h^2 \left(\frac{1 - \sin\phi}{1 + \sin\phi}\right)$$

$$= \tfrac{1}{2}\,w\,h^2 \tan^2 \left(\frac{90 -}{2}\right)$$

$$P_1 = \frac{P\,h + 8\,P\,h_1}{h + 2\,h_1}$$

282. RETAINING WALLS AND WALLS UNDER WIND PRESSURE.

K = maximum pressure outer edge tons per square foot.
R = resultant of thrust in tons when beyond middle third.
W = direct load or vertical component of resultant.
d = distance of resultant from outer edge in feet.

All for one foot run of wall.

$$K = \frac{2}{3} \times \frac{W}{d} \text{ (when no tension is allowed).}$$

283. WROUGHT-IRON STRUTS.

Angle, tee, or cross section, ends fixed.

l = length, inches.
d = least width, inches.
f = factor of safety = 5 to 8.

$$\text{Safe load lbs. per sq. inch, sect. area} = \frac{42000}{f} - 120\frac{l}{d}.$$

$$\text{,, tons ,, ,,} = \frac{20}{f} - \cdot 05\frac{l}{d}.$$

284. MILD STEEL STANCHIONS AND STRUTS.

Approximate safe load lbs. per square inch = $8000\left(\frac{100 - R}{100}\right)$, where $R = \frac{l}{d}$, between 10 and 80 diameters long, ends fixed.

Another rule :—

Approximate maximum safe load lbs. per square inch = $15000 - 40 \frac{\text{length ins.}}{\text{rad. gyr.}}$.

The Differdange broad flange beams are specially suitable for stanchions owing to the least radius of gyration being greater than with narrow flange beams of equal sectional area. Instructions for their use with notes prepared by the author will be found in the book of sections issued by Messrs. H. J. Skelton and Co.

285. GORDON'S FORMULA.

This is really Tredgold's formula, but called Gordon's because he fixed the value of the constants from Hodgkinson's experiments.

l = length in inches.

d = effective diameter or least cross-width in inches.

f = greatest intensity of stress in tons per square inch due to thrust and flexure when on the point of buckling,

for wrought iron = 18
,, mild steel = 26

p = average thrust or compressive force in tons per square inch on section of strut, which will be the crippling stress per square inch when f is taken as above.

a = constant varying according to different authorities. It seems in theory to be made up as follows:—$a = \frac{m}{n\,q}$ where

m = a fixing modulus.

Say 1 for both ends fixed
4 ,, ,, rounded
2·5 for one fixed and one rounded
16 for one fixed the other free.

n = a shape modulus, say

1½ for hollow cylindrical section
1 ,, solid rectangular ,,
¾ ,, ,, cylindrical ,,
½ ,, + L H T ,,
⅓ to ¼ ,, built-up beams

q = a strength modulus, say tons per square inch tensile working strength × 500.

Material.	*Stress.*	q =
Wood	½ ton	250
Cast iron	1½ ,,	750
Wrought iron	5 ,,	2500
Mild steel	7½ ,,	3750

$$p = \frac{f}{1 + \frac{m}{p\,q}\left(\frac{l}{d}\right)^2}$$

$$\text{Factor of safety} = 4 + \cdot 05\,\frac{l}{d}$$

286. RANKINE-GORDON FORMULA.

f = about $\frac{2}{3}$ compressive strength of the material in lbs. per sq. inch.
26,000 lbs. for wrought iron.
40,000 lbs. for mild steel.
60,000 lbs. for cast iron.

A = area of cross section in square inches.

c = constant $\frac{1}{18000}$ for column with round ends.
$\frac{1}{24000}$ for column with one end rounded.
$\frac{1}{36000}$ for column with fixed ends.

l = length of column in inches.

r = least radius of gyration of cross section.

P = ultimate load in lbs.

$$P = \frac{f\,A}{1 + c\left(\frac{l}{r}\right)^2}$$

Factor of safety, 4 for dead load, 5 for live load.

287. EULER'S FORMULA FOR LONG COLUMNS.

E = modulus of direct elasticity, lbs. per square inch; wrought iron, 26,000,000; mild steel, 29,000,000; cast iron, 14,000,000; fir, 1,500,000.

I = moment of inertia of section.

P = greatest load consistent with stability up to stage of incipient failure = resilience of column.

l = length of strut or column in inches.

m = constant depending upon mode in which ends are fixed = $\frac{1}{4}$ for one end free the other fixed, = 1 for both ends free but guided in the direction of the load, = 2 for one end fixed, the other free, and guided in the direction of the load, = 4 for both ends fixed.

n = factor of safety for working load = $\frac{1}{5}$ wrought iron and steel, $\frac{1}{6}$ cast iron, $\frac{1}{10}$ wood.

W = safe working load = P × n.

$$P = m\,\pi^2\,\frac{E\,I}{l^2}.$$

This formula was derived by Euler from the theory of elasticity, and the proof of it is given in Rankine's Applied Mechanics. It assumes ideal conditions which do not occur in practice.

288. FIDLER'S PRACTICAL FORMULA FOR STRENGTH OF COLUMNS.

p = load in lbs. per square inch to produce stress f.
f = ultimate compressive stress in lbs. per square inch.
R = resilient force of ideal column in lbs. per square inch.
L = length of column in inches.
l = for fixed ends $\frac{6}{10}$ L.
r = radius of gyration in incnes measured in plane of easiest flexure.
E = modulus of direct elasticity of material.

26,000,000 lbs. for wrought iron.
14,000,000 ,, for cast iron.
29,000,000 ,, for mild steel.

f = wrought iron 36,000 lbs. per sq. inch.
cast iron 80,000 ,, ,, ,,
hard steel 70,000 ,, ,, ,, } 60,000 average.
mild steel 48,000 ,, ,, ,, }

Maximum p or B.W. of ideal column $= \text{R} = \text{E}\,\pi^2 \times \left(\frac{r}{l}\right)^2$.

Minimum p or B.W. of practical column $= \dfrac{f + \text{R} - \sqrt{(f + \text{R})^2 - 2{\cdot}4\,f\,\text{R}}}{1{\cdot}2}$

Factor of safety $4 + \dfrac{\text{L}}{70\,r}$.

289. TIE RODS FOR BRICK ARCHES.

L = distance between tie rods in feet.
A = sectional area of beam in square inches.
F = width of flange in inches.
w = lateral pressure in lbs. per foot run.
c = constant = 1300 for I beams.
1650 ,, channels.
2650 ,, angles.

$$\text{L} = \sqrt{\frac{c\,\text{A}\,\text{F}}{w}}.$$

W = load in lbs. per square foot.
S = span of arch in feet.
R = rise of arch in inches.

$$w = \frac{1 \cdot 5\, W\, S^2}{R}.$$

—PENCOYD.

290. STRENGTH OF CAST-IRON COLUMNS.

Cast-iron hollow columns :—

d = external diameter inches ($\frac{1}{10}$ to $\frac{1}{30}$ length).
t = thickness in inches (not to exceed $\frac{1}{8}$ d).
L = length in feet (ends flat and fixed).

$$\text{Safe load tons per sq. inch} = (t + 1)\,\frac{2\,d}{L}.$$

Cast-iron solid columns :—

W = breaking weight tons per sq. inch.
r = ratio of length to least diameter.

$$W = \frac{42}{1 + \cdot 003\, r^2}.$$

—PLANAT.

d = diameter inches, l = length feet.

$$\text{Safe load tons} = \frac{4\,d^4}{4\,d^2 + \cdot 18\,l^2}.$$

Safe load hollow column = difference of solid columns of internal and external diameters. —BOURNE.

When the length is 26·4 times the diameter, pillars, columns, or vertical struts are of equal strength whether of wrought or cast iron; when shorter, cast iron is stronger; when longer, wrought iron is stronger.—GORDON.

Cast-iron columns under 5 diameters long fail entirely by crushing; from 5 to 20 diameters, partly by crushing partly by bending; over 20 diameters entirely by bending.

291. APPROXIMATE SAFE LOADS ON POSTS.

Fir post, 10 diameters long, $\frac{2}{10}$ ton per sq. inch.
Oak „ „ $\frac{3}{10}$ „

Approximate safe permanent load in tons on square timber posts of fir, d and l in inches—

$$= 50 \frac{d^4}{l^2}.$$

—REULEAUX.

Another rule for fir posts, flat ends :-

$$\text{Working load lbs. per sq. inch} = 1000 - 10\frac{l}{d}.$$

—STANWOOD.

Another rule: Approx. safe load on fir post :—

$$\frac{b \text{ ins.} \times d \text{ ins.} \left(60 - \frac{l \text{ ins.}}{b \text{ ins.}}\right)}{250} = \text{safe load, tons.}$$

For oak posts :—

b = breadth of side in inches.

L = length in feet.

$$\text{Safe load in lbs.} = \frac{b^3 \times 3960\,L}{4\,b^2 + \frac{1}{2}\,L^2}.$$

—BOURNE.

292. PILLARS AND STRUTS OF WOOD.

d = diameter or width narrowest side, inches.

F = crushing force, short specimen, tons per sq. inch,

l = length in inches.

S = sectional area, sq. inches.

W = breaking weight in tons.

$$W = \frac{F\,S}{1 + \frac{l^2}{250\,d^2}}. \qquad F = \begin{cases} \text{Oak } 3{\cdot}2 \\ \text{Fir } 2{\cdot}5 \end{cases}$$

—RANKINE.

$$W = \frac{F\,S}{1{\cdot}1 + \frac{l^2}{418\,d^2}}$$

—LOVE.

W = safe load tons total.

a = sectional area, sq. inches.

d = least diameter or width side, inches.

L = length, feet.

$$W = 1{\cdot}0752\,a\,\frac{d^2}{L^2} \text{ or } W = \begin{cases} {\cdot}45\,a \text{ for oak.} \\ {\cdot}27\,a \text{ for fir.} \end{cases}$$

The lesser of these two values to be taken. If unseasoned, the safe load will only be one-half above.

P = breaking load in lbs.

r = least radius of gyration $= \frac{d^2}{12}$ for rectangular section.

f = crushing force, short specimen, lbs. per sq. inch = 7200 for timber.

c = constant = 3000 for timber.

S = sectional area, square inches.

l = length in inches.

$$P = \frac{fS}{1 + \frac{l^2}{c\,r^2}}.$$

—Rankine.

293. ULTIMATE STRENGTH OF WOOD POSTS.

12	diameters	long	= $\frac{5}{6}$	crushing	load on	short	specimens.
24	,,	,,	= $\frac{1}{2}$	,,	,,	,,	
36	,,	,,	= $\frac{1}{3}$	,,	,,	,,	
48	,,	,,	= $\frac{1}{6}$	,,	,,	,,	
60	,,	,,	= $\frac{1}{12}$	,,	,,	,,	
72	,,	,,	= $\frac{1}{24}$	,,	,,	,,	

294. FERRO-CONCRETE BEAMS.

1 cement, 2 sand, 4 broken brick $\frac{1}{8}$ in. to $\frac{3}{4}$ in. gauge.

W = safe load at one month, in cwts. distributed, including weight of beam.

b = breadth of concrete in inches.

d = depth of concrete in inches to centre of reinforcement.

p = percentage of reinforcement (0 to $2\frac{1}{2}$) covered by at least $1\frac{1}{2}$ inch of concrete.

L = clear span in feet.

$$W = (\cdot 37p + \cdot 214)\,\frac{bd^2}{L}.$$

If a ferro-concrete floor slab is continuous over several spans, the outer spans should be about three-fourths the width of the others, and if proper provision is made for the reversal of stress over the beams, and for shear, the safe load per foot super. may be taken as $1\frac{1}{2}$ times what it would be on a portion of floor taken as a simple beam.

295. ULTIMATE STRENGTH OF TIMBER IN SHORT SPECIMENS.

Name.	*Tension per sq. inch.*	*Compression per sq. inch.*
Ash	7½ tons	4 tons
Beech	5 ,,	4 ,,
Elm	6 ,,	4 ,,
Memel and Riga fir	5 ,,	2½ ,,
Larch	5 ,,	1½ ,,
Honduras mahogany	4½ ,,	3½ ,,
English oak	6 ,,	4 ,,
Dantzic ,,	5½ ,,	3½ ,,
Quebec ,,	5½ ,,	3 ,,
Teak	7 ,,	5 ,,
Pitch pine	4½ ,,	3 ,,
Hornbeam	4 ,,	3½ ,,
Hard wood	12,000 lbs.	8,000 lbs.
Soft wood	10,000 ,,	6,000 ,,

296. MAXIMUM SAFE LOAD ON TIMBER.

Fir and deal :—

With the grain = 450 lbs. per sq. inch.
Across ,, = 250 ,, ,,

and half these amounts are sufficient for ordinary working loads.

Maximum working load direct tension or compression 2½ cwt. per square inch, or if material be specially selected and compression members be calculated by Gordon's formula, working load may be increased to 4 cwt. direct tension and 7 cwt. direct compression per square inch.

297. COMPARATIVE STRENGTH AND STIFFNESS OF RECTANGULAR BEAMS.

Strength is proportional to the breadth, to the square of the depth, and inversely proportional to the length.

Stiffness is proportional to the breadth, to the cube of the depth, and inversely proportional to the cube of the length.

Strength is measured by the load which the beam will carry, stiffness by the reciprocal of the deflection of the beam under a given load.

Comparative strength and stiffness with the same load will be as follows :—

	Transverse strength.	*Stiffness.*
Cantilever, loaded at end	1	1
,, load distributed	2	$\frac{8}{3}$
Beam, supported at end, load in centre . .	4	16
,, ,, ,, load distributed .	8	$\frac{128}{5}$
,, fixed both ends, load in centre . .	8	64
,, ,, ,, load distributed . .	12	128

298. FORMULA FOR STRENGTH OF TIMBER BEAMS.

L = span feet, b = breadth inches, d = depth inches, W = safe load cwts. distributed.

$$W = \frac{b\,d^2}{L}.$$

This requires no constant, and allows a factor of safety of 7.

When load is concentrated in the centre the safe load will be $W' = \frac{1}{2}\,\frac{b\,d^2}{L}$.

When load is not central, dividing span into x and y

$$W'' = W' \times \frac{\frac{1}{2}L \times \frac{1}{2}L}{x \times y} = \frac{L\,b\,d^2}{4\,x\,y}.$$

The maximum bending moment in cwt.-feet from working load, multiplied by $8 = b\,d^2$ for fir. This applies however the load may be distributed or the beam supported.

Safe deflection = $\frac{1}{40}$ inch per foot span.

In calculating the scantling of timber for practical use under tension or transverse stress, an addition should be made to each dimension to allow for the contingency of a knot occurring in the piece.

When loaded on top and supported at the ends, the soundest side of a square beam should always be placed downwards, and if rectangular then the soundest of the narrow sides should be downwards.

A common rule for fir joists is $d = \frac{1}{2}L + 2$, $b = \frac{1}{3}d$, where L is span in feet, d depth in inches, b breadth in inches.

299. CONSTANTS FOR STRENGTH OF RECTANGULAR BEAMS.

In the formula $W = \frac{c\,b\,d^2}{L}$,

W = breaking weight in cwts. in centre.
c = constant depending upon nature of material.
b = breadth of beam in inches.
d = depth of beam in inches.
L = clear span in feet.

c = weight in cwts. in centre required to fracture a bar 1 inch square and 1 foot long.

Wrought iron	22	English oak	5
Cast iron	18	Quebec and Baltic oak	4·5
Brass	10	Memel, Dantzic and Riga fir	4
Greenheart	8	Spruce fir and larch	3·5
Teak	6	English elm	3

300. EXPERIMENTS ON RECTANGULAR BEAMS OF SELECTED PINE.

B.w. lbs. centre $= 6080 \frac{b\,d^2}{l}$ (all inches); or if L in feet then $= 506\frac{2}{3} \frac{b\,d^2}{L}$.

In a series of experiments on American yellow pine (*Pinus strobus*) c was found to vary from 5,112 lbs. to 7,704 lbs. for l in inches, and 426 to 642 lbs. for L in feet.

If a given rectangular beam be under a given stress by a given load in a given position which divides the span in the proportions x and y, then to obtain the same maximum stress when the load divides the span in the proportions m and n, the depth d will be altered to $d^1 = d \times \sqrt{\frac{m\,n}{x\,y}}$.

301. PROPORTIONS OF BEAMS FOR STRENGTH AND STIFFNESS, WITH MINIMUM AMOUNT OF MATERIAL.

Strongest.	*Stiffest.*
$d : b :: \surd 2 : 1$	$d : b :: \surd 3 : 1$

Approximately for strength, d to b as 1 to ·7; and for stiffness as 1 to

·58; but 1 to ·5 is often used for beams, where the ends can be fixed sideways, because two can be cut out of a square log, and 1 to ·33 or three out of a square log when intermediate staying can be applied, as in joists.

Out of a round log of diameter d the strongest beam that can be cut is $\cdot 816\, d \times \cdot 577\, d$, and the stiffest $\cdot 866\, d \times \cdot 5\, d$.

302. APPROXIMATE PROPORTIONS OF BEAMS.

Strength.	*Stiffness.*	*Convenience.*
inches.	*inches.*	*inches.*
12 × 8½	12 × 7	12 × 9 or 12 × 6
10 × 7	10 × 6	10 × 5
9 × 6½	9 × 5¼	9 × 6 or 9 × 4½
8 × 5½	8 × 4¾	8 × 6 or 8 × 4
7 × 5	7 × 4	7 × 4½ or 7 × 2
6 × 4½	6 × 3½	6 × 4
5 × 3½	5 × 3	5 × 3
4 × 3	4 × 2½	4 × 3 or 4 × 2½
3 × 2	3 × 1¾	3 × 2

303. STRENGTH OF RECTANGULAR BEAMS.

W = load in cwts. in centre of beam supported at ends.
b = breadth in inches.
d = depth in inches.
L = length of span in feet.
a = constant limiting deflection to $\frac{1}{480}$ span.
c = constant for breaking weight.

Name of Timber.	*Constant a.*	*Constant c.*	*Name of Timber.*	*Constant a.*	*Constant c.*
Greenheart . .	0·62	8	Pitch pine . .	0·63	5
Ash . . .	0·85	6	Memel and Dantzic	1·00	4
American elm. .	..	5·5	Riga and spruce .	0·77	3·5
Teak . . .	1·00	5	Larch . .	0·46	3·5
English oak . .	0·80	5	English elm .	0·42	3

For stiffness (working load) $W = \frac{a\,b\,d^3}{L}$.

For strength (ultimate load) $W = \frac{c\,b\,d^3}{L}$.

If concentrated load divides span into x and y, $W^1 = \frac{W\,L^2}{4\,x\,y}$.

Distributed load $= 2$ W.

Factor of safety if calculated for strength, temporary work or dead load $= 5$, permanent and live load 10.

304. APPROXIMATE WEIGHT OF TIMBER.

	lbs. per cub. ft.		*lbs. per cub. ft.*
White deal and spruce	30	American elm	45
Riga	32	Beech	43
Memel	35	Ash	45
Dantzic	40	American oak	50
Christiana yellow	42	English oak	55
Larch	35	Teak	50
English elm	35		

305. STRENGTH AND STIFFNESS OF TIMBER.

Name.	*Stiffness.*	*Strength.*	*Resilience.*
Ash	89	119	160
Beech	77	103	138
Riga fir	98	80	64
Memel fir	114	80	56
Larch	79	103	134
Honduras mahogany	93	96	99
English oak	100	100	100
Dantzic ,,	117	107	99
Quebec ,,	114	86	64
Teak	126	109	94
Pitch pine	73	82	92

Oak being taken for comparison as = 100.

306. TIMBER TREES.

Name.	*Mean Diameter of Trunk.*	*Average Length of Trunk.*
	inches.	*feet.*
Ash	23	38
Beech	27	44
Chestnut	37	44
Elm	32	44
Riga fir	20	75
Larch	33	45
Mahogany	72	40
Norway spruce	15	60
Canadian oak	34	53
English oak	32	42
Sycamore	29	32

—LAW.

307. SIZES OF FIR TIMBER IN BALK.

Stettin	18 to 20 in. square.	—
Dantzic	14 „ 16 „	40 to 50 ft. long.
Memel	13 „	35 „
Riga	12 „	40 „
Swedish and Norwegian	8 „ 12 „	—

308. NOTES ON PILE-DRIVING.*

Gauge, guide, or main piles are whole timbers 9 to 15 inches square, driven about 10 feet apart.

Waling-pieces, or walings, are horizontal timbers formed of half balks secured to the guide piles in pairs, one pair near the top and another pair near low-water mark. These serve as guides in driving the intermediate piles.

Sheet piling is formed of piles 9 inches by 4½ inches or 12 inches by 6 inches, the bottom end chisel-shaped and raking so as to be drawn towards the piles already driven.

* See paper by the author on "Timber Piling in Foundations and other Works," 2nd ed., 24 pp. and folded plate (Spon, 1s.).

Intermediate piles may be whole timbers or sheet piling according to circumstances.

Puddle is well punned clay filled in between the walls of a cofferdam to prevent passage of water.

All piles should be shod with iron; if unprotected the wooden points would break and cause the piles to drive out of line. Shoes for main piles weigh from 10 to 15 lbs. each, and for sheeting piles from 5 to 8 lbs. each.

The heads of all piles should be hooped, ringed, or rung to prevent them from splitting under the blows of the ram or monkey.

When piles have to be scarfed to obtain sufficient length, the scarfs should break joint at 6 feet intervals in adjacent piles.

It should be noted that ferro-concrete piles are now coming largely into use.

309. FORMULÆ FOR PILE-DRIVING.

P = ultimate supporting power in tons = $W f$.

W = safe working load in tons.

w = weight of ram in lbs. = not less than $\frac{L s}{4}$.

H = height of fall in feet.

d = set, or distance driven by last blow, in inches.

L = length of pile in feet.

s = mean sectional area of pile in square inches.

f = factor of safety = say from 2 to 3.

x = energy of last blow in foot-tons = $\frac{w H}{2240}$.

c = constant = $\frac{125 s}{L}$.

$$P = \sqrt{c x + \left(\frac{c d}{24}\right)^2} - \frac{c d}{24}.$$

$$x = \frac{P d}{12} + \frac{P^2}{c}.$$

$$d = 12 \left(\frac{x}{P} - \frac{P}{c}\right).$$

$$H = \frac{2240 x}{w}.$$

310. TIMBER ROOFS.

Thickness of truss in inches = $\frac{1}{5}$ span ft. for king post truss or $\frac{1}{7}$ span ft. for queen post truss.

Depth of tie-beam in inches = 0·3 span in feet + 3.

King or queen post square in middle, width of ends = twice thickness.

King post truss up to 30 feet span, queen post truss for larger spans.

These proportions allow about 28 lbs. per ft. super. for truss and covering and factor of safety of 6.

$\frac{1}{4}$ pitch = rise of $\frac{1}{4}$ span	=	$26\frac{1}{2}$ degrees.
$\frac{1}{3}$ " = " $\frac{1}{3}$ "	=	$33\frac{2}{3}$ "
Common slope for slates	=	30 "
Common slope for tiles	=	45 and 60 degrees.
Slope for heaviest slates	=	$22\frac{1}{2}$ degrees.
Slope for lead or zinc flat	=	$1\frac{1}{2}$ inches in 10 ft.

311. WIND PRESSURES.*

36 lbs. per sq. foot steady wind pressure, and 56 lbs. per sq. foot for gusts in exposed situations, is sufficient to provide for in roofs, bridges, etc., for ordinary cases.

312. FORCE OF WIND.

Miles per hour × 88 = feet per minute.

" $\times \frac{22}{15} =$ " second.

$$p = \frac{v^2 \text{ ft. per sec.}}{500}.\text{—Hutton.}$$

p = ·144 velocity miles per hour.—Crosby.

p = pressure in lbs. per sq. foot against a plane surface normal to direction of wind.

a = area of maximum section in sq. feet perpendicular to direction of wind.

θ = angle of exposed surface with plane of section.

c = coefficient according to shape of surface presented.

P = total resistance of surface in direction of wind.

* See papers by the author on "Wind Pressure on Roofs," 2nd ed., demy 8vo, 12 pp., with folded plate (6d.), and "The Force of the Wind," 8 pp. (3d.).

$$P = c \cdot a \cdot p.$$

Coefficients, c =

Disc or rectangular plane	=	1
Cylinder	=	$\frac{\pi}{4}$
Sphere	=	$\frac{\pi}{8}$
Wedge, edgeways	=	$\sin\theta$

—ADAMS.

An inclined plane of area a will at small angles present a resistance approximately varying as $\sin^2\theta\ a\ p$, and at large angles as $\sin\theta\ a\ p$. A formula agreeing closely with Hutton's experiments is

$$P = a\,p\,\sin\theta^{1\cdot 84\cos\theta},$$

a being full area of surface.

In Hutton's experiments a = 32 sq. inches, which was too small to make the experiments of much practical value.

For wind pressure on roofs between 10° and 60° pitch, $p_n = p\sin^2(1\cdot 2\,\theta + 18)$.—E. F. ETCHELLS.

313. CLASSIFICATION OF WIND FORCE.

The classification by different writers of the force of the wind varies considerably, the following is a fair average.

Description.	*Velocity in miles per hour.*	*Approximate corresponding pressure lbs. per sq. ft*
Barely perceptible wind	2½	$\frac{1}{32}$
Light breeze	5	⅛
Pleasant breeze	7½	¼
Good breeze	10	½
Strong breeze	15	1⅛
High wind	20	2
Half gale	30	4½
Strong gale	40	8
Whole gale	50	12½
Great storm	60	18
Hurricane	80	32
Violent hurricane	100	50

314. COMPARISON OF VELOCITY AND PRESSURE OF WIND.

P = pressure in lbs. per sq. foot.
V = velocity in miles per hour.

$$P = a V^2$$

a =	·00254	Carus-Wilson.
	·003	Aspinall.
	·005	Smeaton.
	·00537	Hawksley.
	·00585	D. K. Clark.

The theoretical pressure of the wind due to its velocity is given by Professor C. A. Carus-Wilson as

$$P = \frac{G}{2g} v^2$$

where p = pressure in lbs. per sq. foot
v = velocity in feet per second
g = accelerating force of gravity
G = weight of air in lbs. per cub. feet
whence $p = 0{\cdot}00254\, V^2$, V being velocity in miles per hour.

Mr. J. A. F. Aspinall, M.Inst.C.E., gives the demonstration as follows :—

Let v = velocity of wind in feet per second.
W = weight of air delivered per sq. foot per second in lbs.
P = pressure of wind lbs. per sq. foot on exposed area.
m = mass of air delivered per sq. foot per second.

$$\text{Then } P = m v, \text{ therefore } P = \frac{W v}{g},$$

but W = weight of 1 cub. foot air (w) × velocity of the air in feet per second (v), therefore $W = w v$, and $P = \dfrac{w v^2}{g}$.

Or, expressed in miles per hour (V), and taking the weight of 1 cub. foot of air as 0·0807 lb.,

$$P = \frac{0{\cdot}0807 \times (1{\cdot}466\, V)^2}{32} = 0{\cdot}0054\, V^2.$$

The velocity of the wind is usually taken by a cup anemometer, but there is reason to believe that the result obtained is only approximate; it is, however, sufficiently accurate for general purposes.

The pressure of the wind has been taken by the resistance against small planes on a whirling table, by hinged boards supported by springs, by the difference of level produced on water in U tubes, and other means. By experiments upon large areas it is found that considerable variation exists in the pressure at different points at the same time, and the larger the area the lower the average pressure.

315. RESISTANCE OF PLANE SURFACES.

Normal to direction of movement, in lbs. per sq. foot when surface moving, v being velocity in feet per second.

$$= v^2 \times \cdot 0017 \text{ for air}$$
$$= v^2 \times \cdot 976 \text{ for water.}$$

Resistance is greater when air or water is moving against surface at rest. —DU BUAT.

316. WINDMILL.

a = total sail area in sq. feet.
V = velocity of wind in feet per second.
H P = horse power.

$$\text{H P} = \frac{a \text{V}^3}{10{,}800{,}000}.$$

317. VARIATION OF WIND PRESSURE ACCORDING TO HEIGHT AND AREA EXPOSED.

A formula based on practical requirements and recorded experiments is the empirical one,

$$\log p = 1\cdot 125 + 0\cdot 32 \log h - 0\cdot 12 \log w$$

where p = ultimate wind pressure in lbs. per sq. foot necessary to be allowed for against a plane surface normal to the wind.

h = height of centre of gravity of surface considered, above ground level in feet.

w = width in feet of part to be taken as one surface;

and when the surface is inclined at θ degrees to the direction of the wind, the ultimate pressure normal to the surface may be taken as $p \sin \theta$, or its effect in the same direction as the wind $= p \sin^2 \theta$. —ADAMS.

This would give allowances as in the following table.

318. WIND PRESSURE LBS. PER SQUARE FOOT ON PLANE SURFACE.

(For very exposed positions 25 per cent. may be added.)

Height in feet.	*Width in feet.*						
	5	10	20	50	100	200	500
150	54·6	50·3	46·3	41·4	38·1	35·1	31·4
100	48·0	44·2	40·7	36·4	33·5	30·8	27·6
50	38·4	35·4	32·5	29·1	26·8	24·7	22·1
20	28·7	26·4	24·3	21·7	20·0	18·4	16·5
10	23·0	21·1	19·5	17·4	16·0	14·8	13·2
5	18·4	17·0	15·6	13·9	12·8	11·8	10·6

Multipliers for angle.

θ	10	20	30	40	50	60	70	80	90
sin	·174	·342	·500	·643	·766	·866	·940	·985	1
$\sin^2$	·0303	·117	·250	·413	·587	·750	·884	·970	1

—ADAMS.

In the case of a boundary wall the length to consider would be a portion say equal to 1½ times the height. In the case of a roof the distance from centre to centre of the trusses, unless the stability of a whole building be under consideration when the full length should be taken. In the case of a lattice girder bridge add 50 per cent. to the actual area and take both girders.

319. APPROXIMATE WEIGHT OF TIMBER ROOFS.

King or queen truss, span in feet²	=	lbs. per truss.
Common rafter and purlins	=	7 lbs. per ft. super.
¾-inch slate boarding	=	2½ ,, ,,
Slate battens	=	1¼ ,, ,,
Roofing felt	=	½ ,, ,,

Slates and nails (general) . . .	= 9 lbs.	per	ft. super.
Ceiling (complete).	= 12	„	„
Snow	= 7½	„	„
Wind (horizontally)	= 56	„	„

The combined effect in vertical load with trusses usual distance apart may be taken at 60 lbs. per foot super.

320. GALVANISED CORRUGATED SHEET IRON.

B.G. No.	*Thickness inches.*	*Weight lbs. per 100 ft. sup.*	*Width flutes.*	*Uses.*
16	·0625	380	5 in.	Where great strength is required.
17	·0556	320	„	For first class work generally.
18	·0495	280	„	
19	·0440	252	„	
20	·0392	224	3 in.	For ordinary work.
21	·0349	205	„	
22	·03125	185	„	
23	·02782	165	„	Shipped abroad.
24	·02476	150	„	
26	·01961	112	„	

Sheets 6 or 8 feet long, 3 feet 2 inches wide before corrugation, 2 feet 6 inches wide with 5-inch corrugations, depth of corrugation ¼ width.

To be laid with 6-inch laps when on slope, 3-inch when vertical, riveted in each flute, and same pitch at sides.

321. CORRUGATED IRON.

Sheets 30 inches wide, 20 to 26 gauge in thickness, before corrugating becomes 27 inches wide, ⅝ inch deep, and 2½ inches centre to centre of corrugations after rolling.

W = crippling load in lbs. per sq. foot.
t = thickness of metal in inches.
b = centre to centre of corrugations in inches.
d = depth of corrugation in inches.
l = length of span in inches.

$$W = \frac{98000\, t\, b\, d}{l}.$$

—PENCOYD.

Safe load say ⅔ of crippling load.

322. WEIGHT OF CORRUGATED IRON.

Weight of galvanised corrugated iron sheets per square of 100 feet super. in lbs.

Gauge.	6*ft.* × 4 *ft.* or 8 *ft.* × 3 *ft.*	7 *ft.* × 4 *ft.*	6 *ft.* × 5 *ft.* or 10 *ft.* × 3 *ft.*	8 *ft.* × 4 *ft.*
14	430·5	426·5	410	421·5
15	—	—	385	—
16	340	336·5	326	332
17	—	—	295	—
18	264·5	262·5	253	259
19	—	—	228	—
20	210	208·25	202	205
21	—	—	180	—
22	175·5	—	171	—
24	145	—	140	—
26	124	—	120	—

The weights given include 6 inch end laps and one corrugation side lap. 7 lbs. of rivets and hook bolts, or spikes, are required to fix one square of sheeting.

323. SHEET COPPER.

Sheets 4 feet by 2 feet.

B.W.G. 20	=	1·625 lb.	per foot super.
22	=	1·25	,, ,,
24	=	1·0	,, ,,
26	=	·75	,, ,,
28	=	·5	,, ,,
30	=	·375	,, ,,

No. 20 or 22 gauge used for gutters and flats.

No. 24 or 26 gauge used for flushings, dormer sides, etc.

They can also be obtained for roofing 5, 6, 7, and 8 feet long × 3 feet wide in gauges 18 to 25.

1 sq. foot, 1 inch thick, weighs 46 lbs.

324. SHEET LEAD.

Cast sheets, 6 feet wide × 16 to 18 feet long.

Milled sheets, 7 feet wide × about 25 feet long.

Made 3 to 10 lbs. per foot super.

4 or 5 lbs. lead used for aprons, flashings, and soakers.

5 to 7 lbs. lead for drips and ridges.

6 to 8 lbs. lead for gutters and flats, and wherever it is liable to be walked upon.

7 to 8 lbs. lead for soil pipes.

Lbs. per foot × ·017 = thickness in decimals of an inch.

1 sq. foot, 1 inch thick, weighs 60 lbs.

325. SHEET ZINC.

Sheets 2 feet 8 inches and 3 feet wide, 7 feet and 8 feet long, rolled at 200° to 300° F. (93° to 149° C.).

Brittle when cold and again at 400° F. (205° C.), malleable at 212° F. (100° C.).

Gauge.		*Oz. per ft. sup.*	*Corresponding to old B.W.G.*	*Thickness inches.*
(Z.G.) No.10	=	11½	25	·019
12	=	15⅛	23	·025
14	=	18¾	21	·031
16	=	24¾	19	·041
18	=	30¾	17	·058

1 sq. foot, 1 inch thick, weighs 37½ lbs.

No. 12 or 13 used for flashings, No. 13 or 14 for dormers, No. 14 or 15 for flats, No. 15 or 16 for gutters.

326. SHEET ALUMINIUM.

1 sq. foot, 1 inch thick, weighs 14·2 lbs.

327. HANDY NUMBERS FOR WEIGHT OF IRON.

Wrought iron :—

Sectional area sq. inches × 3⅓ = lbs. per foot run.

Cub. inches × ·282 = lb.

Round iron, $d^2 \times 2{\cdot}62$ = lbs. per foot run.

Sq. feet per $\frac{1}{8}$ inch thick $\times$ 5 = lbs.

For weight of rivets in plate girders, take 5 per cent. of weight of plates and angle irons, and in lattice or box girders $2\frac{1}{2}$ per cent.

Cast iron :—

Sectional area, sq. inches $\times$ 3·2 $\times$ length in feet = lbs.

Weight of wrought — 5 per cent. = weight of cast.

23 cub. inches = 6 lbs.

40 lb. per sq. foot, 1 inch thick, is sometimes taken to allow for inaccurate casting.

Thickness in $\frac{1}{8}$ inch $\times$ width in $\frac{1}{4}$ inches $\times$ length in feet = lbs. weight of castings.

Cub. inches $\times$ ·263 = weight in lbs.

Mild Steel :—

Weight of wrought iron + 2 per cent. = weight of mild steel.

Some designers add $\frac{1}{48}$ and some $\frac{1}{50}$ to weight in wrought iron.

The American ton, or short ton, is 2.000 lbs.

328. MARKET SIZES OF PLATES.

In a well-assorted specification for a fair quantity of material, Staffordshire plates may now be obtained at a minimum price up to 10 cwts. each, 30 feet long, and 5 feet 6 inches wide, and Cleveland plates up to 15 cwts. each, 30 feet long and 5 feet wide.—WALMISLEY, 1888.

For ordinary prices mild steel plates may be obtained in one piece up to 20 cwts., 30 feet long, 6 feet 6 inches wide, $1\frac{1}{2}$ inch thick, or 60 feet super., and to double these limits for a moderate addition to the price.

329. LIMITS OF ORDINARY PRICES, STAFFORDSHIRE DISTRICT.

Plates.—Weight 8 cwts., length 20 feet, width 4 feet 6 inches, 40 feet super., shape regular.

Angle and Tee Irons.—Length 40 feet, size $2\frac{1}{2}$ inches by $2\frac{1}{2}$ by $\frac{1}{4}$ up to 8 united inches.

Bars.—(Round and square), diameter $\frac{1}{2}$ inch to 3 inches, length 25 feet.

Bars.—(Flat), size 1 inch by $\frac{1}{4}$ inch up to 6 inches by 1 inch, length 25 feet.

330. EXTRACT FROM THE CLEVELAND LIST OF LIMITS AND EXTRAS.

Weight, to 10 cwts. Beyond, 10s. per ton for every cwt. or portion thereof.

Length, to 20 feet. Beyond, 2s. 6d. per ton per foot or part thereof.

Width, 12 inches to 54 inches. For $\frac{3}{16}$ inch and $\frac{1}{8}$ inch thick, 12 inches to 48 inches. Beyond or under, 5s. per ton per inch or part thereof.

Area $\begin{cases} \text{60 sq. feet for thicknesses from } \frac{1}{4} \text{ inch to 1 inch inclusive.} \\ \text{48 ,, } \frac{3}{16} \text{ inch thick.} \\ \text{36 ,, } \frac{1}{8} \text{ ,,} \end{cases}$

Beyond (if sellers undertake them at all), 1s. per ton per sq. foot.

Boiler plates, except B B B boiler, 48 sq. feet.

,, B B B boiler, 36 sq. feet.

Beyond (if undertaken), 2s. 6d. per ton per sq. foot.

Thickness, $\frac{1}{4}$ inch to 1 inch. $\frac{3}{16}$ inch 10s. per ton, and $\frac{1}{8}$ inch 30s. per ton extra.

Sketches, 20s. per ton. Curved sketches, 40s. per ton. 4 inch taper allowed before counting sketch.

Guarantee.—In case of serious defect, or error in dimensions, a plate will be replaced, and on receipt of the rejected one the amount originally charged will be credited. Dimensions will be worked to as nearly as practicable, but absolute exactness must not be expected. No further liability is undertaken by sellers except by special contract.

Stoppage of Works.—Should the works of the makers or buyers be stopped by a strike, or by accident to machinery or buildings, current contracts to be suspended during such interruption, but not to be thereby cancelled.—Fox, Head and Co.

331. SIZE OF STEEL PLATES.

The maximum dimensions to which the Consett Iron Co., Ltd., supply steel plates are as follows :—

Thickness in inches.	*Length in feet.*	*Width in inches.*	*Area in sq. ft.*	*Thickness in inches.*	*Length in feet.*	*Width in inches.*	*Area in sq. ft.*
$\frac{1}{8}$	14	48	48	$\frac{15}{16}$	44	81	185
$\frac{3}{16}$	20	60	80	1	44	81	175
$\frac{1}{4}$	26	72	100	$1\frac{1}{16}$	44	81	165

Thickness in inches.	*Length in feet.*	*Width in inches.*	*Area in sq. ft.*	*Thickness in inches.*	*Length in feet.*	*Width in inches.*	*Area in sq. ft.*
$\frac{5}{16}$	32	75	150	$1\frac{1}{8}$	44	81	155
$\frac{3}{8}$	36	81	235	$1\frac{1}{4}$	44	81	138
$\frac{7}{16}$ to $\frac{3}{4}$	44	81	235	$1\frac{3}{8}$	44	81	125
$\frac{13}{16}$	44	81	220	$1\frac{1}{2}$	44	81	115
$\frac{7}{8}$	44	81	200				

Rolling margin $2\frac{1}{2}$ per cent. and length 1 inch over or under.

332. COMMON PROPORTIONS FOR ANGLES.

$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{3}{16}$	$3 \times 3 \times \frac{3}{8}$	$4\frac{1}{2} \times 4\frac{1}{2} \times \frac{5}{8}$
$2 \times 2 \times \frac{1}{4}$	$3\frac{1}{2} \times 3\frac{1}{2} \times \frac{1}{2}$	$4 \times 3 \times \frac{1}{2}$
$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{5}{16}$	$4 \times 4 \times \frac{9}{16}$	$6 \times 4 \times \frac{5}{8}$

but either size may vary in thickness from the preceding to the succeeding one.

333. DEFLECTION AND CAMBER.

Deflection is the displacement of any point in a loaded beam from its position when the beam is unloaded.

Camber is an upward curvature, similar and equal to the maximum calculated deflection given to a beam or girder or some line in it in order to ensure its horizontality when fully loaded.

334. RADIUS OF CURVATURE

is the radius of the circle coinciding most nearly with a curved line or portion of one.

Curvature is the reciprocal of this radius. Thus, if radius be 100 feet, curvature is $\frac{1}{100}$. If radius alters further on to 120 feet, the change of curvature will be $\frac{1}{100} - \frac{1}{120} = \frac{1}{600}$.

The curvature of a circle is inversely proportional to its radius, and is measured by the fraction $\frac{1}{\text{radius}}$.—GOODEVE.

Radius of curvature of neutral axis of a beam at any section when under transverse stress $= \rho = \frac{E\,I}{M}$.—UNWIN.

The following equality should also be noted :—

$$\frac{M}{I} = \frac{f}{y} = \frac{E}{\rho}$$

335. DEFLECTION OF SOLID BEAMS.

Δ = deflection in inches.
L = length in feet.
b = breadth in inches.
d = depth in inches.
W = load in cwts. in centre.

c = constant =

Cast steel . . .	650	Quebec oak . . .	40
Wrought iron . .	550	Fir and deal . . .	33
Cast iron . . .	330	Dantzic oak . . .	27
Teak	50	Pitch pine . . .	25

Rectangular beam, $\Delta = \frac{W L^3}{c\,b\,d^3}$.

Square beam, side $= \sqrt[4]{\frac{W L^3}{\Delta\, c}}$;

Cylindrical beam, diameter $= \sqrt[4]{\frac{W L^3}{\Delta\, c}} \times 1{\cdot}7$.

Safe deflection in timber $= \frac{1}{480}$ length, or $\frac{1}{40}$ inch per foot span.

336. COEFFICIENTS FOR DEFLECTION—RECTANGULAR BEAMS.

c =		δ =	
Wrought iron . .	·000002	Fixed one end, loaded the other	128
Cast iron . . .	·000003	„ load distributed	48
Steel	·0000016	Supported ends, load central .	8
Oak	·0000375	„ load distributed	5
Ash	..	Fixed both ends, load central .	2
Fir	..	„ load distributed	1

$$\text{Deflection} = \frac{W \text{ lbs.} \times l^3 \text{ feet} \times \delta \times c}{b \text{ inches} \times d^3 \text{ inches}}.$$

General formula for beams of uniform section, fixed one end, loaded the other, deflection $\Delta = \frac{W\,l^3}{3\,E\,I}$.

337. COMPARATIVE DEFLECTION.

(Same total load throughout.)

		Box.	*Unwin.*
Fixed one end, loaded the other	K =	32	16
„ „ load distributed	=	12	6
Supported both ends, load central	=	1	1
„ „ load distributed	=	$\frac{5}{8}$	$\frac{5}{8}$
Fixed both ends, load central	=	$\frac{2}{3}$	..
„ „ load distributed	=	$\frac{5}{12}$	..

338. LOAD AND DEFLECTION.

Comparative safe loads on beams of similar section and span, and deflection of beams under those loads.

	Load.	*Def.*
Fixed one end, loaded the other	$\frac{1}{8}$	$3\frac{1}{5}$
„ „ „ load distributed	$\frac{1}{4}$	$2\frac{2}{5}$
Supported both ends, load central	$\frac{1}{2}$	$\frac{8}{10}$
„ „ „ load distributed	1	1
Fixed both ends, load central	1	$\frac{4}{10}$
„ „ „ load distributed	$1\frac{1}{2}$	$\frac{3}{10}$

339. APPROXIMATE DEFLECTION OF WROUGHT-IRON FLANGED GIRDERS

of uniform strength, supported at both ends, and carrying uniformly distributed load. Stress allowed = 5 tons per sq. inch tension, 4 tons per sq. inch compression.

s = span in feet.

d = mean depth in inches.

D = deflection in inches in centre.

$$D = \frac{\cdot 0144\, s}{d}.$$

If depth = $\frac{1}{10}$ span, D = ·012 s ; $\frac{1}{12}$ = ·0144 s ; $\frac{1}{15}$ = ·018 s.

In girders with parallel flanges of uniform strength, the deflection produces a circular curve, the amount of deflection varies directly as the load × the sum of the areas of both flanges × the cube of the length, and inversely as the area of top flange × area of bottom flange × depth of web squared, or

$$\Delta = \frac{W \times (a_t + a_b) \times L^3}{a_t \times a_b \times d^2} \times c.$$

$c =$	Wrought iron.		Cast iron.
Load centre	·016	..	·025
Load distributed	·01	..	·018

340. GENERAL RULES FOR DEFLECTION.

l = span in inches.
W = load in tons distributed, ends supported.
I = moment of inertia.
E = modulus of elasticity in lbs. per sq. inch.
S = stress allowed in tons per sq. inch.
δ = deflection in inches.

For girder of uniform section :—

$$\delta = \frac{5\,W\,l^3}{384\,E\,I}.$$

For girder of uniform strength :—

$$\delta = \frac{S\,L}{4\,E\,D}.$$

Common Rule.—Girders to be constructed with a camber of ¼ to ½ inch per 10 feet of span, to allow for deflection when loaded.

Feet span × ·005 to ·0075 = safe deflection in inches under ordinary loads.

Feet span × ·02 to ·03 = safe deflection in inches under special loads.

Deflection with safe concentrated load in centre = $\frac{8}{10}$ of deflection corresponding to safe distributed load which would be double the central load.

American practice. Feet span × ·01 = safe deflection in inches after permanent set.

Board of Trade allows ¾ inch per 100 feet span (= $\frac{1}{1600}$ = ·0075) for deflection caused by maximum rolling load beyond the deflection due to maximum dead load.

The actual deflection in any given case depends to some extent upon the workmanship. Loose rivets cause an apparently excessive permanent set.

341. DEFLECTION TESTS.

Two main girders 60 feet span erected in yard with cross girders and bearing for railway viaduct. Weight complete, one span with temporary timber, 22 tons.

Deflection in centre with		30 tons distributed	= ·32	inches.
"	"	60 "	= ·685	"
"	"	90 "	= 1·085	"
"	"	121·5 "	= 1·53	"
"	"	do. and 10 tons centre	= 1·77	"
"	"	loads removed	= ·47	"

342. LOAD ON BRIDGES.

Assuming deflection to vary directly as load, the WORK done by gradually applied load = load lbs. × ½ deflection feet, but with suddenly applied load = load lb. × deflection feet, because it drops through the whole distance, and the deflection being double that due to the same load gradually applied, the WORK will be quadrupled. A rolling load on a girder is not quite a suddenly applied load, but somewhere between that and a dead load. The stress in a given beam varies as the deflection.

Load.	*Ratio of Deflection or Maximum Stress.*
Dead, gradually applied	5
Live, rolling on	8
Live suddenly applied	10

343. DEFLECTION OF BEAMS UNDER IMPACT.

P = weight of load falling upon centre of beam.
h = vertical height of fall to surface of unstrained beam.
d = static deflection due to P.
D = actual dynamic deflection due to impact of falling load.
W = weight of beam.
m = constant depending upon ratio of P to W.

$$m = \frac{35\,P}{35\,P + 17\,W}.$$

$$D = d + \sqrt{2\,m\,h\,d + d^2}$$

In the case of a suddenly applied load, $h = 0$ and $D = 2\,d$.
—MERRIMAN'S "MECHANICS OF MATERIALS."

344. STRENGTH OF FLAT CARRIAGE SPRINGS.

For spiral springs, see under BOILERS.

E = modulus of elasticity for spring steel = 16,000 tons.
e = ultimate extension of fibre, say ·0025.
S = ultimate stress, tons per sq. inch = $E\,e$ = 40.
L = half length of spring from buckle in inches.
b = breadth of plate in inches.
n = number of plates.
W = total load on spring in tons.
d = length of offset $= \dfrac{L}{n}$.
v = deflection of spring in inches per ton of load.
V = working deflection = $v\,W$.
r = radius of curve of camber $= \dfrac{E\,t}{2\,S}$ = approx. 200 t.

Safe working load of spring in tons $= \dfrac{S\,b\,t^2\,n}{3\,L}$.

$$v = \frac{4\,L^3}{E\,b\,t^3\,n}, \qquad n = \frac{L\,W}{13 \cdot 3\,b\,t^2}.$$

Half span of spring from buckle $= \sqrt{(2\,r - V)\,v}$.
The deflection varies directly as the load.

Another rule :—

d = deflection in sixteenths of an inch per ton of load.
s = span in inches.
b = breadth in inches.
t = thickness of leaves in sixteenths of an inch.
n = number of leaves.

$$d = 1 \cdot 64 \times \frac{s^3}{b\,(n.t^3)}.$$

345. NOTES ON TORSION AND SHAFTING.

Torsion is measured by the load acting at 1 foot radius which is required to fracture a specimen 1 inch diameter.

Strength varies as $\frac{d^3}{r}$, stiffness as $\frac{d^4}{l}$.

Long shafts are not designed in strict accordance with rule, as they would then be tapered from driving end, involving extra assortment of driving pulleys.

Every alteration in diameter of a shaft, unless made at a coupling, must be made gradually by means of a curve at the junction of the two diameters, or a long taper.

Factor of safety, long shafts less than 4½ inches diameter = $\frac{1}{10}$; short shafts and all over 4½ inches diameter = $\frac{1}{8}$.

346. SUPPORTS FOR SHAFTING.

Distance apart of supports in feet = $5\sqrt[3]{d^2}$. Common rule :—Centre distance of bearings in feet = twice diam. ins. × 4.

Another rule, distance = $\sqrt{32\,d}$.

347. PERMISSIBLE TWIST IN SHAFTING.

Per foot of length :—

0·10° for easy service without fluctuation of load.

0·075° for fluctuating loads suddenly applied.

0·050° for loads suddenly reversed.—JAS. CHRISTIE.

To run smoothly, long shafting must not twist more than 1° in 10 feet under maximum load.—REULEAUX.

348. APPROXIMATE STRENGTH OF SHAFTING.

The safe load on wrought-iron shaft 1 inch diameter at 1 foot radius is 100 lbs.

$$\therefore \ W = 100\frac{d^3 \text{ ins.}}{\text{lev. ft.}}, \qquad d = \sqrt[3]{\frac{W \times \text{lev.}}{100}}, \qquad \text{lev.} = \frac{100\,d^3}{W}.$$

349. ULTIMATE TORSIONAL STRENGTH OF VARIOUS METALS.

Round bars 1 inch diameter, load applied at 1 foot radius.

Cast steel	. . . .	average 1500 lbs.
Mild steel	. . . .	„ 1200 „

Wrought iron	average	800 lbs.
Cast iron	„	700 „
Wrought copper	„	400 „

These, although average test loads, are rather higher than are usually adopted in practical calculations. See section on calculation of engine shafts.

350. TORSIONAL MODULUS OF ELASTICITY.

The torsional modulus of elasticity is about 46 per cent. of the modulus in tension, and nearly constant for all classes of material substances.—PLATT *and* HAYWARD.

351. TRANSMISSION OF POWER BY SHAFTING.

Strength of shaft to transmit power depends upon velocity; thus, shaft able to transmit 20 horse-power at 60 revolutions is sufficient for 60 horse-power at 180 revolutions. The explanation is, that the actual strain is the same in each case, the increase in horse-power being due to the increase in speed only. Power consists of pressure and velocity, and varies directly as the amount of each.

352. COMPARISON OF TORSIONAL AND BENDING RESISTANCE OF SHAFTING.

Torsional resistance $= \frac{\pi}{16} d^3 f$

where d = diameter in inches,

and f = greatest shearing stress lbs. per sq. inch.

Bending resistance $= \frac{\pi}{32} d^3 c$

where d = diameter in inches,

c = constant for strength of material.

Thus, assuming $c = f$, the moment of resistance of a circular shaft to twisting is double that of its resistance to bending.

353. FORMULA FOR STRENGTH OF SHAFTING.

W = B.W. in lbs. at 1 foot radius of shaft 1 inch diameter.

c = coefficient in safety = $\frac{1}{6}$ to $\frac{1}{10}$.

d = diameter of wrought-iron shaft in inches.
l = leverage in feet.
s = strain in lbs. at circumference of wheel.

$$d = \sqrt[3]{\frac{s\,l}{W\,c}}. \qquad s = \frac{W\,d^3}{l} \times c.$$

354. MOLESWORTH'S FORMULA FOR WROUGHT-IRON SHAFTING.

D = diameter of shaft in inches.

$$K = \begin{cases} 320 \text{ for crank shafts and prime movers.} \\ 200 \text{ for second motion shafts.} \\ 100 \text{ for ordinary shafting (but never less than 80).} \end{cases}$$

H = actual horse-power to be transmitted.
n = number of revolutions per minute.
l = leverage in feet.
f = force applied in lbs. at circumference of wheel.

$$H = \frac{2\pi l n f}{33000}. \qquad H = \frac{D^3 n}{K}. \qquad f = \frac{D^3}{2\pi l} \times K.$$

$$f = \frac{33000\,H}{2\pi l n}. \qquad D = \sqrt[3]{\frac{H}{n} \times K}.$$

$$D = \sqrt{\frac{2\pi l f}{33000} \times K}.$$

355. DIAMETER OF COUPLING BOLTS IN SCREW SHAFTS.

D = diameter of shaft.
d = ,, bolts.
n = number of bolts.
r = radius of pitch circle for bolts.

All in inches.

$$d = \cdot577 \times \sqrt{\frac{D^3}{n\,r}}.$$

356. COMBINED BENDING AND TWISTING MOMENTS.

Ordinary formula :—

$$M = \tfrac{1}{2}\,m + \tfrac{1}{2}\sqrt{m^2 + t^2}$$

Improved formula :—

$$M = \sqrt{m^2 + t^2}.$$—" THE ENGINEER."

357. PROPORTIONS OF SOLID WROUGHT-IRON FLANGE COUPLING ON SCREW SHAFT.

Let d = diameter of shaft. Then there should be eight bolts, each $\frac{1}{4}\,d$ in diameter, the diameter of circle passing through the centres being $1\frac{1}{2}\,d$. The flanges should be $2\,d$ in diameter and $\frac{1}{4}\,d$ thick.—UNWIN.

Note.—Six bolts are commonly used, up to 6 inches diameter of shaft.

For marine crank shaft, web of throw $= \frac{3}{4}\,d$ thick, pin $= d$ diameter, area of bolts (total) = area of shaft.

358. TRANSVERSE STRENGTH OF SHAFTS.

Load distributed on wrought-iron crank pin or overhanging journal in lbs., $c = 1200$.

Ditto, concentrated on shaft supported at ends, $c = 2400$.

Ditto, distributed ,, ,, $c = 4800$.

$$\text{Safe load} = \frac{c\,d^3}{l}. \qquad d = \sqrt{\frac{W\,l}{c}}.$$

Forces may be taken to act at the centres of journals in cases where supports are not contiguous to journals.

359. PROPORTIONS OF BOLTS, NUTS AND WASHERS IN CARPENTRY.

Thickness of nut	=	1	diameter of bolt.
,, head	=	$\frac{3}{4}$	,,
Diameter of head or nut over sides .	=	$1\frac{5}{8}$	,,
Side of square washer for fir . .	=	$3\frac{1}{2}$	,,
,, ,, ,, oak . .	=	$2\frac{1}{2}$	,,
Thickness of washer	=	$\frac{1}{4}$	,,

When the nuts are let in flush in fir, the washers should be the same size as for oak.

The length of a bolt is usually taken as the distance from the underside of the head to the point.

The wood measure, or "grip," of a bolt is usually taken as the distance between the washers when a full thread is in the nut.

The safe working load on bolts in tension may be taken as—

1 in. bolt for $1\frac{1}{2}$ tons.		$1\frac{3}{8}$ in. bolt for 3 tons.	
$1\frac{1}{8}$,, ,, 2 ,,		$1\frac{1}{2}$,, ,, $3\frac{1}{2}$,,	
$1\frac{1}{4}$,, ,, $2\frac{1}{2}$,,			

In bolting woodwork, the crushing of the material in the hole by a shear stress on the bolts is a matter that requires particular attention.

360. SIZE OF BOLTS IN TIMBER.

Thickness.	*Diameter.*	*Thickness.*	*Diameter.*	*Thickness.*	*Diameter.*
3 in. ..	$\frac{1}{2}$ in.	6 in. ..	$\frac{5}{8}$ in.	12 in. ..	$\frac{7}{8}$ in.
4 in. ..	$\frac{9}{16}$ in.	9 in. ..	$\frac{3}{4}$ in.	14 in. ..	1 in.

No bolts should be less than $\frac{1}{2}$ inch diameter in structural work. The thickness of the timber is measured across the bolt. The bolts may be single or double, according to circumstances, but should not be in the same line of grain except in deep narrow timber.

361. STRENGTH OF BOLTS.

Bolts in machinery subject to varying loads should not be strained to more than 2 tons per sq. inch of minimum section. A bolt 1 inch diameter, being ·84 diameter, or ·55 area at bottom of thread, will take not more than (say) 2000 lbs., including initial strain in screwing up.

Let d = outside diameter of thread in inches; $2000\ d^2$ = safe load in lbs. for 1 inch bolts and upwards; $2000\ d^3$ = safe load in lbs. for 1 inch bolts and under.

The ordinary force used in screwing up bolts is liable to break a $\frac{3}{8}$ inch bolt and seriously injure a $\frac{1}{2}$ inch bolt; hence bolts for joints requiring to be tightly screwed up should not be less than $\frac{3}{4}$ inch in diameter.

Spanners are proportioned in length according to the diameter of the bolt so that a man of ordinary strength can put on sufficient force. Increasing the leverage by a piece of gas barrel, or otherwise, may seriously damage the bolt.

The approximate area of Whitworth bolts at bottom of thread = diameter of bolt in $\frac{1}{8}$ths inch $\times$ (diameter in $\frac{1}{8}$ths inch — 1) $\div$ 100.

For proportions of Whitworth's Standard, see tables further on.

According to Prof. Unwin:—

	Per sq. inch net area.
(*a*) Bolts not requiring to be tightened before load is applied, also (*c*) when cylinder exceeds 60 inches diameter	Safe load = 6000 lbs.
(*b*) Bolts accurately fitted and requiring to be tightened moderately, also (*c*) when cylinder exceeds 20 inches diameter	„ = 4000 „

(c) Bolts used to draw joints steam-tight and resist the pressure in addition . . . . . — *Per sq. inch net area.* Safe load = 2000 lbs.

362. FLANGE STUDS OF STEAM CYLINDERS.

For small cylinders allow 2700 lbs. per sq. inch of net section (minimum diameter of bolts ⅝ inch).

For large cylinders (over 18 inches diameter) allow 3000 lbs. ditto ditto.

363. TO SECURE CHECK OR LOCK NUTS.

Put on check nut (½ diameter of bolt in thickness), screw up as tight against flange or work as an ordinary nut would be screwed under the circumstances, then put on ordinary thick nut (1 diameter thick), screw it up with the same force and hold on to it with the spanner. Then with a thin spanner reverse the check nut against the other as far as it will go with about the same pressure as before. The check nut has then only the screwing up force to resist, while the thick nut has in addition the strain which may be brought upon it by load or vibration.

364. CHECK NUTS.

. . . . This loosening of a nut can be prevented by adding another nut, which must be screwed hard down upon the first to increase the pressure upon the thread.—WILLIS' "MECHANISM."

Note.—As described here, the second nut would only be equivalent to thickening the first nut, and would be useless as a check, unless tightened up to the limits of abrasion.

365. PRESSURE ON BEARING AREA IN HOLES.

The pressure of a pin in an eye, or a bolt in a hole, or a rivet in a plate, resisting a side pull or shearing stress, should be limited to the safe pressure on bearing surface. The maximum pressure (P) per sq. inch, assuming the bearing surface to be ¼th of the circumference, will be $= P/\cdot 7854\, d\, t$, where P = total pressure, d = diameter, t = thickness.

Example.—1½-inch pin, load 3 tons, thickness ¾ inch, $P = 3/\cdot 7854 \times 1\cdot 5 \times \cdot 75 = 3\cdot 4$ tons per sq. inch. Or if required to limit pressure on bearing area to say 2 tons per sq. inch, then 1½-inch pin with 3 tons load will require thickness in eye of $t = 3/\cdot 7854 \times 1\cdot 5 \times 2 = 1\cdot 28$ inches.

Section IV.

PATTERN-MAKING, MOULDING AND FOUNDING.

366. PATTERN-MAKING.

SMALL patterns made of mahogany or New Zealand pine. Larger patterns made of white or yellow pine. Metal patterns used where a great number of similar castings are required. Wood patterns coated with varnish, to prevent distortion from damp sand, black for general body, red for ends of prints or cores, and yellow for machined faces. Some are one colour only.

Patterns should have rounded edges, and filleted angles wherever possible. The thickness of metal throughout a casting should be as uniform as possible, sudden changes of direction being avoided. Sharp angles in a casting are always weak; the crystals while cooling arrange themselves perpendicularly to the surface, and hence at a sharp turn there is an awkward junction, which becomes a source of weakness. Sufficient taper, say ⅛ inch per foot, must be given to draw out of the sand, and allowance made for knocking to loosen in mould.

Holes for bolts, etc., may be "cast in," or "cored out"; when cast in, sufficient taper must be given to draw the pattern, and small side of hole must be large enough for bolt; when cored out a print must be put on one or both ends to form support for core. Prints should project from ½ inch to 3 inches, according to weight of core to be carried. Heel cores are made when the print is at any distance from the parting.

367. BLACK VARNISH FOR PATTERNS.

Lampblack 1 part, shellac 5 parts, methylated finish 16 parts, all by weight. First coat rubbed over with glass paper when dry and second coat then laid on.

368. WEIGHT OF CASTING FROM PATTERN.

Multiply weight of deal pattern by—17 for cast iron, 18 for brass, 19 for copper, 25 for lead.—Hurst.

369. ALLOWANCE FOR MACHINING.

Average on iron castings = $\frac{1}{8}$ inch, brass $\frac{1}{16}$ inch. Castings likely to twist in cooling require more, very small castings require less. In small cylinders $\frac{1}{4}$ inch in the diameter is sufficient, cylinders over 4 feet diameter say $\frac{3}{8}$ inch.

370. MOULDING IN FOUNDRY.

Green-sand Moulding.—Used for light iron castings, fire-bars, rough machine castings, etc. The ordinary damp sand of the foundry is used in iron boxes or " flasks " for receiving impression from " patterns," the hollow parts being formed of baked sand " cores." Long cores are supported by " chaplets," small and complicated cores are made of " loam." The top box is called the " cope " and the bottom one the " drag."

Dry-sand Moulding.—Used for ornamental ironwork, important machine castings, and for casting in brass. The sand consists of fresh sand mixed with loam which has been used, or of fresh sand only. When finished, the moulds are dried for several hours. " Blackening " prevents sand melting.

Loam Moulding.—Used for steam cylinders, bent pipes and complicated work. The mould is often built up without patterns, and consists of brickwork coated with loam and " swept " to required shape by a " loam board." Long straight cores are formed of iron pipe with haybands twisted on to hold the loam, and other cores of loam strengthened by bent " core-irons." The loam is common brick-clay mixed with horse-dung, cow-hair, sand, etc. " Runners " and " gates " are openings in the sand to let the metal into the mould; " vents " are openings to let the gases out, formed by pricking the sand.

371. SAND FOR MOULDING.

Moulding Sand consists of 93 to 96 per cent. of sharp sand and 3 to 6 per cent. of clay. Quality varies for different castings; the smaller the castings, the more clay the sand may contain; heavy castings require poorer and coarser sand. Coal and coke are used to make the sand more porous; this makes the castings rougher, but by giving free vent to the gases makes them sounder. Moulding sand after use is " screened " and wetted before being used again.

Parting Sand is the burnt sand scraped off castings, and is used to facilitate the division of the upper and lower boxes in moulding.

Core Sand consists of 90 per cent. sharp sand and 10 per cent of clay, and should be used fresh.

372. FOUNDRY DRYING STOVE.

Brick chamber of three sides with arched top shut with close iron doors on fourth side. Size about 10 feet × 10 feet × 7 feet high. Fire-place on one side, flue near ground on opposite side to spread the heat and carry off the moisture, fire fed through a door on outside. Iron shelves on walls for drying small cores and boxes. Rails run from crane into drying stove so that large moulds may be wheeled in. Stoves of various sizes in large foundry, the larger ones only used when required for very large moulds.

373. NOTES ON MOULDING AND CASTING.

Keep most important side of casting at the bottom to ensure density in the metal, as tension flange of girder, etc. Make ample provision for escape of gases by pricking the mould, providing vents, etc. Support long cores and stiffen with core irons to prevent displacement by molten metal. Knock pattern slightly before drawing from mould to enable it to be lifted without breaking the sand. Provide sufficient number of gates to ensure the mould being completely filled with metal. Allow ample head on important castings to cut off all "sullage" or porous and honeycombed portion. The molten metal should be stirred through the gates with an iron rod, called a "feeding rod," to agitate it and cause it to fill angles and corners, more metal being added if required. Directly the metal is run into the mould the gases should be fired to prevent explosion. Metal usually run in afternoon, allowing all night for castings to cool. 2½ cwt. of coke required to melt 1 ton of iron.

374. CLEANING CASTINGS.

Moulds taken apart and sand removed as soon as castings have set, castings taken out with tongs and left to cool, time varying according to weight and mass. Gates, or "gits," and partings, or "fins," broken off, and heavy or hard cores removed in foundry before casting is cold. Projections removed in cleaning or fettling shop with chisel, sharp hammer, or worn-out file, and casting well brushed with steel wire brush. Grindstones or emery wheels used in some shops instead of chisel and file, for removing projections and a sand-blast for cleaning the surfaces. Blow-holes (if any) stopped with black putty, cement, or lead, and castings painted with black wash. Badly honey-combed castings thrown on the scrap heap. The scrap averages 25 per cent. of the castings, less on large work.

375. CLASSIFICATION OF IRON ORES.

Mr. Truran classifies the ores of Great Britain into four great divisions, thus :—

A. The argillaceous ores of the coal formations, having clay, but sometimes silica, as the chief impurity.

B. The carbonaceous ores of the coal formations, distinguished by their large percentage of carbon.

C. The calcareous or spathic ores, or the sparry carbonates of iron, having lime as their chief earthy admixture.

D. The siliceous ores, having silica as their predominating earth. This class is subdivided into the red and brown hæmatites, the ores of the oolitic formation, the white carbonates, and the magnetic oxides.

376. CHARGES EMPLOYED AT DOWLAIS FOR DIFFERENT KINDS OF PIG IRON.

——	*Foundry Pig.*	*White Forge Pig.*	*Common Forge Pig.*
	cwt.	*cwt.*	*cwt.*
Calcined " mine " (fresh ore)	48	28	..
Red hæmatite ore . .	..	10	16
Forge and refinery cinder .	..	10	25
Limestone . . .	17	14	16
Coal	50	42	36
Weekly make . .	130 tons	170 tons	190 tons

377. ANALYSES OF PIG IRON.

Carbon, partly combined and partly in a graphitic form	2·3	to	5·5	per cent.
Silicon	0·13	„	5·7	„
Manganese	0·0	„	7·6	„
Sulphur	0·0	„	0·87	„
Phosphorus	0·0	„	1·66	„

378. FOUNDRY IRON.

Scotch iron is considered the best for foundry purposes; it is strong and uniform in quality, and will mix well with other brands. Cleveland iron is harder than Scotch iron, and not so strong. It is much used in the Cleveland district, a mixture of Nos. 1 and 3 being suitable for large work. Lincolnshire iron is very similar to Cleveland. Cumberland "hæmatite" iron is generally used for steel-making, but it is also employed to improve other irons for foundry use; it is very tough and strong, but does not run well when melted by itself; a little mixed with good Scotch irons makes a strong mixture. The best known cold-blast iron is the Blaenavon; it shows great strength and fineness of grain, and is used to close the grain and increase the strength of other irons.—F. CAMPIN.

379. FOUNDRY PIG.

No. 1 Pig is chiefly used in the foundry. Colour dark grey, crystals large and leafy, carbon in form of graphite. Very soft, melts very fluid, but being coarse-grained, will not give a sharp impression. Cools slowly. For fine castings the presence of a little phosphorus is advantageous: the grain is finer, the iron a lighter colour, and the impressions sharper. Used for small castings, hollow ware, small machinery, etc.

No. 2 Pig, grey and mottled in colour. Carbon partly combined. Used for large castings in dry sand or loam. Melts fluid, is tough, close texture, fills the mould well, more free from impurities than No. 1. Heavy machine castings made from No. 2, or various mixtures of 1, 2, and 3.

No. 3 Pig, hard and white, used for mixing. Carbon all chemically combined.

380. MIXTURES OF PIG IRON.

Mixture recommended for girders, etc., where rigidity and strength are required:—

Lowmoor, Yorkshire, No. 3	30	per cent.
Blaina or Yorkshire No. 2	25	,,
Shropshire or Derbyshire No. 3	25	,,
Good old cast scrap	20	,,
	100	

—FAIRBAIRN.

	Mixture for steam cylinders, strong and close grained.	For the same where greater hardness is required.
No. 5 charcoal pig	8 parts.	2 parts.
Scotch pig	10 ,,	4 ,,
Good cast scrap	10 ,,	30 ,,

Piston rings should be of softer metal than the cylinders.—RIGG'S "STEAM ENGINE."

381. SPECIFICATIONS FOR FOUNDRY CASTINGS.

Specifications for foundry castings, adopted very largely at the present time (1899), give chemical requirements as follows:—

Quality of iron.	*Hard.*	*Medium.*	*Soft.*
	Per cent.	*Per cent.*	*Per cent.*
Silicon	1·20 to 1·60	1·40 to 2·0	2·20 to 2·80
Sulphur not to exceed	0·095	0·085	0·085
Phosphorus less than	0·70	0·70	0·70
Manganese not above	0·70	0·70	0·70

The physical tests for which are:—

Transverse strength of a 1 in. square bar, on supports 12 in. apart . . lbs.	*Per cent.* 2,400	*Per cent.* 2,200	*Per cent.* 2,000
Deflection not less than	0·08	0·09	0·10
Shrinkage on a length of 1 ft. not under	0·161 in.	0·151 in.	0·141 in.
Chill not over	0·25 in.	0·15 in.	0·05 in.

Mr. G. Henderson quotes a mixture which has given satisfactory results for cylinder castings, the cupola charge being made up as follows:—

Limestone	70 lbs.	Pig iron	800 lbs.
Steel scrap	800 ,,	Cast scrap	2,400 ,,

The castings on analysis showing—

Combined carbon	0·62 per cent.	Phosphorus	0·65 per cent.
Graphite	2·45 ,,	Sulphur	0·68 ,,
Silicon	1·51 ,,	Manganese	0·33 ,,

—"THE PRACTICAL ENGINEER."

382. CONDITION OF CARBON IN CAST IRON.

Grey pig iron, total carbon 3 to 4 per cent. When molten all combined but according to slowness of cooling will be the amount separated as graphitic carbon, leaving 0 to 0·5 per cent. combined. The carbon itself is inert as regards taking the graphitic form, but is influenced in that direction by the silicon. The "rate of cooling" is another way of expressing the period of time, be it less or more, through which the silicon is allowed to act upon the carbon by causing it to separate out as graphite, before the temperature of the metal drops too low for such action to be effective. To some degree silicon may be neutralised by sulphur, but in grey cast iron such a neutralising can only be very partial. Sulphur undoubtedly has the power of causing the combined carbon to retain that form and so cause hardness, but any hardening of the iron so caused is exceeded by the direct hardening of the sulphur itself. If castings are heavy and so take a considerable time to cool, the silicon under such conditions will gradually reduce the percentage of combined carbon, notwithstanding the presence of sulphur. If combined carbon be present in very minute quantities, then such castings are soft and weak from excess of graphitic carbon. Greatest general strength is obtained when combined carbon is present from 0·5 to 0·6 per cent., other constituents being normal. Seeing that a fairly definite percentage of combined carbon is necessary in the best and strongest irons we can produce, it follows that we must vary the percentages of silicon as the castings are light or heavy. Light castings cool quickly, and so the silicon is kept high; In a heavy casting, cooling through as many hours, perhaps, as the other does minutes, the silicon is kept low. For greatest general strength each should finish cooling with somewhat similar percentages of combined carbon.—R. BUCHANAN.

383. MECHANICAL BLOWERS.

Blacksmiths' fires require 70 cub. feet per minute for each tuyere, and safety valve on pipe.

Foundry cupolas require 30 000 cub. feet of air per hour per ton of iron to be melted. Or, according to size, 1·6 cub. foot of air per minute per sq. inch area of cupola at melting level.

C = cub. feet delivered per minute.

p = pressure lbs. per sq. inch at blower.

$$IHP = \cdot 0002\, C p.$$

384. MELTING METAL FOR CASTINGS.

Crucibles are sometimes used for melting iron for trinkets and small goods. The best castings, whether iron, bronze, or other metal, for machine frames, bells, statues, etc., are made from a *reverberatory furnace*, run directly from the furnace in dry sand ditches to the mould. The *cupola* has the advantage of melting iron cheaper than any other furnace ; where strength is unimportant, it is the best method.

385. CONTRACTION OF METALS IN COOLING.

Metal.	*In Fractions of Linear Dimensions.*	*In Parts of an Inch per Foot of Linear Dimensions.*
Cast iron	$\frac{1}{96}$	$\frac{1}{8}$
Gun metal	$\frac{1}{72}$	$\frac{1}{8}$
Yellow brass	$\frac{1}{64}$	$\frac{3}{16}$
Copper	$\frac{1}{60}$	$\frac{1}{5}$
Zinc and tin	$\frac{1}{48}$	$\frac{1}{4}$
Lead	$\frac{1}{39}$	$\frac{5}{16}$

386. CONTRACTION OF CASTINGS.

Heavy pipes	=	$\frac{1}{8}$	inch per foot.
Girders, beams, etc.	=	$\frac{1}{8}$	,, in 14 inches.
Engine beams / Connecting rods	=	$\frac{1}{8}$	,, in 16 inches.
Large cylinders, say 70 inches diameter × 10 feet stroke, the contraction of diam.	=	$\frac{3}{8}$	,, at top.
	=	$\frac{1}{2}$	,, at bottom.
Ditto in length	=	$\frac{1}{8}$	,, in 16 inches.
Small narrow wheels, about	=	$\frac{1}{25}$	,, per foot diam.
Large heavy wheels	=	$\frac{1}{10}$	,, or more ,,
Thin brass	=	$\frac{1}{8}$	,, in 9 inches.
Thick brass	=	$\frac{1}{8}$	,, in 10 inches.
Gun-metal rods	=	$\frac{1}{8}$	,, in 9 inches.
Zinc	=	$\frac{5}{16}$	,, per foot.
Copper	=	$\frac{3}{16}$	,, ,,
Bismuth	=	$\frac{5}{32}$	,, ,,
Tin and lead, each	=	$\frac{1}{4}$	,, ,,
Aluminium	=	$\frac{17}{64}$	,, ,,

Pattern-makers commonly allow for iron castings $\frac{1}{8}$ inch per foot, and for brass castings $\frac{3}{16}$ inch per foot. The apparent contraction varies considerably according to the amount of "rapping" the pattern receives.

387. EXPANSION OF CASTINGS.

Some castings, owing to their form, expand in one direction while contracting in another. This is known to pattern-makers as "compression." It is usually a contraction of surface area and expansion in thickness, the expansion taking place in the direction in which the heat most readily radiates, and being chiefly noticeable in tram plates and such like forms.

Cast iron bars repeatedly heated red hot and cooled gain in length and cross section but decrease in specific gravity. The increase in their linear dimensions may reach 7 per cent. with a reduction of specific gravity from 7·13 to 6·01.—A. E. OUTERBRIDGE.

388. CAST ZINC.

Zinc is troublesome to cast, especially in small thin pieces. Lining the mould with whiting and water, which must be allowed to dry thoroughly, will often cause the metal to fill the mould well. Burning of the zinc (oxidising) may be prevented by covering the metal while in the crucible or ladle with a layer of common salt, or, better, with a layer of charcoal.—J. DONALDSON.

389. BRONZE AND BRASS CASTINGS.

Melted in crucibles, wasting prevented by covering surface with mixture of potash, soda, and charcoal powder. Copper melted first, then tin, zinc, or antimony, then covering applied. Zinc is best added in form of brass, calculating the copper contained. Large strong castings require the metal exposed to fire in fluid state 8 or 10 hours, proof taken by small ladle and broken when cool, judged by crystallisation, and copper or tin added as required. Before casting, bronze is well stirred with heated iron rods. Brass made by melting together copper scraps, crude zinc or spelter, and charcoal powder, remelted for casting. About 7 lbs. per cwt. is allowed for waste.

Pin-holes, or blow-holes, in brass castings are produced by overheating the metal; by allowing the metal to *soak* while in the fire—i.e., to remain in the fire too long after it has become liquid; or by melting very finely divided metal such as chips, grindings, or similar forms.—"ENGINEERING AND MINING JOURNAL."

Section V.

FORGING, WELDING, RIVETING, ETC.

390. DUCTILITY AND MALLEABILITY.

Ductility is the capability of being drawn into wire. Order of ductility—gold, silver, platinum, iron, copper, zinc, tin, lead, nickel.

Malleability is the capability of being hammered into shape. Order of malleability—gold, silver, copper, tin, platinum, lead, zinc, iron, nickel.

391. FORGING.

Wrought iron at a red heat may be hammered into various shapes, called "forging." When a piece is drawn down smaller it is called "swaging"; if jumped up thicker, it is called "upsetting." Common iron is not suitable for forging, as the scale or slag in it causes cracks. Double and treble best Staffordshire and ordinary Yorkshire are suitable. The best Yorkshire is used for flanging and difficult forgings; charcoal iron for light and complicated work.

The chisels used by the smith for cutting iron, like sharp hammers with ash handles, are called *cold* and *hot setts*. The shaping is done by *fullers* or *fullering tools*, which are blunt ended tools with a wire rod handle or twisted hazel rods. Flat surfaces are worked with a *flatter* or *sett hammer*. Rounded parts like bolts are worked with *top* and *bottom swages*. *Tongs* of various kinds are used for holding the material, with a ring or *coupler* driven on to hold them close. Anvil tools are inserted in the square hole of the anvil to be used in conjunction with the sets and swages.

Steel may be forged gradually at a low heat. The greater the proportion of carbon contained, the greater the difficulty of forging. All forging should proceed by easy stages, and care be taken not to burn the iron or steel. Large pieces have a rod or "porter" welded to them for convenience in handling by a crane.

Drop-forging, called also drop-stamping, consists of heating a mass of wrought iron or mild steel to forging temperature and placing it between dies under a steam-hammer or hydraulic press.

392. WELDING

is the process of joining two pieces of wrought iron or steel by heating, and hammering them together. To weld iron the pieces must be brought to a white heat, and the scale swept off before they are put together. Steel requires a much lower heat, and the surfaces should be sprinkled with sand, borax, or silicate of soda, to aid the surface fusion. Borate of soda similarly aids the surface fusion of spelter in hard soldering. The welding temperature depends upon the amount of carbon contained: hence the extra difficulty of welding two pieces of different composition. Mild steel approaches wrought iron in its welding qualities. Steel faces may with care be welded on to iron tools; shear steel is generally used for this purpose. Average loss of strength in weld is 15 to 20 per cent.

393. ELECTRIC WELDING.

In electric welding, a current is passed through the abutting edges which are pressed together, surface-fusion is almost immediately produced, and the junction commences at the centre, proceeding uniformly to the outside. This weld is said to be of equal strength with the solid material, but the loss may reach 10 per cent.

394. GAS WELDING.

The flame of the oxyhydrogen blow-pipe has been successfully adopted by Frederick Braby and Co., Ltd., for welding sheet iron and steel in lighter gauges than can be done by any other process.

395. THERMIT WELDING.

Thermit is a mixture of aluminium and oxide of iron in fine grain and in chemical proportion. It can be ignited by a powder formed of finely divided peroxide of barium and aluminium, which may be lighted by a match. The thermit when fused has a temperature of 3000° C. (5432° F.), and a little poured upon rail ends to be welded, confined in a suitable mould or case, quickly raises them to welding heat, and simple pressure completes the joint. This means of welding tramway rails has been largely used by the Leeds Corporation, who were the first to adopt it in this country upon a large scale. It may be used for prompt repairs of general machinery and in breakdowns at sea.

Aluminium and magnesium surpass all other metals in the production of heat on combining with oxygen. 18 kilos. (=40 lbs.) aluminium combining with 16 kilos. (=35½ lbs.) of oxygen set free 130,500 large calcries (C.) of heat (=517,866 B.Th.U.) at a temperature of about 3000° C. (5432° F.).

396. FACE-WELDING OR VENEERING OF METALS.

Welding, equivalent to veneering, may also be accomplished by the hot rolling of clean metal ingots, of similar or different kinds, having a thin sheet of aluminium interposed. Thus a compound sheet of iron and copper may be produced in this way with any relative thickness of the two external metals.

397. TEMPERING.

Steel when heated to a cherry red, and suddenly cooled in water or oil, is rendered very hard. Some suppose that the carbon is caused to take the crystalline or diamond form. For tempering the hardened steel a portion is brightened with a piece of broken grindstone, and then reheated until the film of oxide formed on the surface shows the requisite temperature; it is then quenched in water, and the hardness is found to be "let down" to the "temper" required. Springs are tempered by cooling in oil, then blazing off the oil in the fire and again cooling. Small drills are sometimes tempered by thrusting them into a tallow candle after heating them in a red-hot piece of iron pipe. Tempering was formerly considered to be the only true test of steel, but some steel is now made with so little carbon that it cannot be tempered.

As used by steel manufacturers, the term temper is almost synonymous with the degree of carbonization, "high-temper" steel contains much carbon, which makes it hard, "low temper" contains less carbon, and makes a softer steel. The classification of temper is often by the amount of carbon suitable for various tools, as in the case of Dannemora cast steel, where they give razor temper, chisel temper, etc.

398. COLOURS CORRESPONDING TO TEMPERATURE.

	Deg. F.	*Deg. C.*		*Deg. F.*	*Deg. C.*
Lowest red heat visible in the dark	635	335	Orange	2010	1100
Faint red	960	516	Bright orange	2190	1200
Dull red	1290	700	White heat	2370	1300
Brilliant red	1470	800	Bright white heat	2550	1400
Cherry red	1650	900	Dazzling white heat	2730	1500
Bright cherry red	1830	1000	Welding or scintillating heat	2800	1550

—BECQUEREL, POUILLET, ETC.

	Deg. F.	*Deg. C.*		*Deg. F.*	*Deg. C.*
Dark blood red, black red	990	532	Salmon, orange, free scaling heat	1650	900
Dark red, blood red, low red	1050	566	Light salmon, light orange	1725	941
Dark cherry red	1175	635	Yellow	1825	996
Medium cherry red	1250	677	Light yellow	1975	1080
Cherry, full red	1375	746	White	2200	1205
Light cherry, bright cherry, scaling heat,* light red	1550	843	Taken by a Le Chatelier pyrometer.		

—M. WHITE *and* F. W. TAYLOR, U.S.A.

From experiments made by Wedgwood, there is reason to believe that all bodies susceptible of the requisite temperature become red hot at exactly the same point. Wood and most liquids are dissipated before their temperature can be sufficiently raised to be luminous. Gases do not become luminous even at a much higher temperature than suffices for solids.

399. TEMPERING STEEL.

Colours produced at various Temperatures, and Alloys Fusible at same.

Colour of Film.	*F°*	*C°*	*Nature of Tool.*	*Lead*	*Tin*
None	400	205	. . . .	22	16
Very pale yellow straw	430	221	Lancets and turning-tools for cast steel	30	16
A shade of darker yellow	450	232	Razors and ditto	34	16
Golden yellow	470	243	Penknives, turning-tools for iron	42	16
Orange yellow	490	254	Cold chisels, drills, screw taps, wood tools	56	16
Brownish yellow	500	260	Hatchets, plane-irons, chipping-chisels, saws for iron, tools for working granite, turning-tools for brass	66	16
Brown tinged with purple	520	271		100	16
Light purple	530	277	Swords, ordinary springs, tools for cutting sandstone	120	16
Full purple	550	288		192	16
Full blue	570	300	Small saws, watch-springs, augers	150	8
Grey blue	600	316	Large saws, pit and hand saws	50	2
Pale blue with tinge of green	620	327	Too soft for steel instruments	All	0
Grey	750	400			

* Heat at which scale forms and adheres, i.e., does not fall away from the piece when allowed to cool in air.

400. BLUE SHORTNESS.

A critical temperature in the working of both iron and steel is at the so-called "blue heat," which is from 450 to 600° F. (232 to 316° C.). At this temperature iron and steel are much more brittle than when cold or at redness, and if the piece be worked in this range of temperature it will retain the brittleness after cooling and show a great loss of ductility, as measured by the bending test. The danger to steel is more pronounced than in iron, but it exists to a greater or less extent in all grades of iron, especially in the poorer qualities. The character of the metal may be restored by annealing or reheating to redness. Blue working without subsequent annealing is prohibited in boiler work.—PROF. H. E. SMITH.

401. NOTES ON RIVETED JOINTS.

A rivet consists of head, shank, and tail, the latter being formed by the riveting up.

Hard wrought iron is weakened from 15 to 30 per cent. by punching. In punched plates the small sides of the holes should come together. Drilled holes should have the edges chamfered.

The tension in a rivet may be estimated at 21,000 lbs. per sq. inch of its section. Friction due to this tension would be about 7000 lbs. per sq. inch of rivet section.

The usual diameter of rivets in hand riveting varies from $\frac{1}{2}$ inch to $\frac{7}{8}$ inch. In machine riveting they may be used up to $1\frac{1}{4}$ inch diameter.

Maximum efficiency of single riveted joint = $\frac{2}{3}$ strength of plate. Ordinary efficiency = $\frac{9}{16}$. Maximum efficiency of double riveted joint = $\frac{4}{5}$ strength of plate. Ordinary efficiency = $\frac{3}{4}$.

Pitch of rivets (for equal area of plate and rivet) =

$$\frac{\text{Sect. area of rivet} \times \text{effective No. of rows}}{\text{Thickness of plate}} + \text{diam. of rivet.}$$

$$\left.\begin{array}{r}\text{Chain}\\ \text{riveting}\end{array}\right\} = \begin{array}{cccc}\times & \times & \times & \times\\ \times & \times & \times & \times\end{array}. \qquad \left.\begin{array}{r}\text{Zigzag, reeled,}\\ \text{or staggered}\end{array}\right\} = \begin{array}{cccccc}\times & & \times & & \times & \\ & \times & & \times & & \times\end{array}.$$

To rivet by hand requires a minimum of 1 diameter, and by machine $1\frac{1}{8}$ diameter of rivet to form head. Length of rivet for good head = thickness of plates passed through + $1\frac{1}{2}$ diameter + $\frac{1}{16}$ inch for each joint. Rivets 6 to 8 diameters long often draw off their heads. Rivets are usually $\frac{1}{16}$ inch smaller than hole, generally $\frac{3}{4}$ inch iron in $\frac{13}{16}$ inch hole, but may be $\frac{11}{16}$ inch iron in $\frac{3}{4}$ inch hole. Countersunk rivets 60°, countersunk $\frac{2}{3}$ diameter of rivet.

A rivet hole cannot be punched with its edge nearer the edge of the plate than its own diameter without risk of its bursting through. To this it is safe to add $\frac{1}{8}$ inch to $\frac{1}{4}$ inch on the plate as the size of rivet and thickness of plate increase. The edges of two holes cannot be nearer than 1 to $1\frac{1}{2}$ diameter without risk of the second hole distorting the first, or the two holes punching into one.

The efficiency of the bearing surface of rivets = 5 tons per square inch; thus a $\frac{7}{8}$-inch rivet in a $\frac{3}{4}$-inch plate $= \frac{7}{8} \times \frac{3}{4} \times 5 = 3{\cdot}3$ tons nearly.

18 rivets go to the "yard" for piece work, irrespective of the pitch.

402. PRESSURE TO CLOSE RIVETS.

Experiment:—Cold riveting, $\frac{3}{8}$-inch rivets.

At 10,000 lbs. rivet swelled and filled hole without forming head.

At 20,000 lbs. head formed and plates slightly pinched.

At 30,000 lbs. rivet well made.

At 40,000 lbs. metal in plates round rivet began to stretch.

Therefore, approximately, d in $\frac{1}{8}$ths$^2 \times 2 =$ tons pressure required for cold riveting per sq. inch of rivet section, and d in $\frac{1}{8}$ths$^2 =$ tons pressure for hot riveting per sq. inch rivet section.

A $\frac{3}{4}$-inch rivet in cooling puts a friction of about 4 tons on the plates, but the shearing value is thereby reduced.

403. MACHINE RIVETING FOR BOILERS.

With $\frac{3}{4}$-inch rivets the closing pressure in riveting $\frac{3}{8}$-inch plates is 38 tons, $\frac{1}{2}$-inch plates 40 tons, and steel plates 45 tons. The cup must be left on until the rivet is black.

In hydraulic riveting the pressure on the cup head = 12,000 to 16,000 lbs. per sq. inch of surface.

404. PROPORTION OF RIVET DIAMETER TO THICKNESS OF PLATE.

In punching, the resistance of steel in section of punch = 100 per sq. cm., and that of iron in cut surface of hole = 30 per sq. cm.

$$\text{Punch} = 100 \times \frac{\pi d^2}{4}, \text{ plate} = 30\,\pi d \times e.$$

$$\therefore d > \frac{30\,\pi\,d\,e}{100\,\dfrac{\pi\,d}{4}} = 1{\cdot}2\,e \ (e = \text{thickness});$$

but d must be $< 3\,e$, or crushing by the pressure of the rivet on edge of plate will occur, hence the usual proportion of $d = 2\,e$.—PLANAT.

405. RIVETING.

Heads $\cdot 66\,d \times 1{\cdot}66\,d$ with radius of $\cdot 86\,d$. Length to make this $= 1{\cdot}16\,d$, N = tension, ω = section, t = temperature of heated rivet when closed, E = coefficient of elasticity, then

$$\frac{N}{\omega} = \frac{7\,E\,t}{11 \times 81{,}500},$$

N being the tension capable of producing a stretch equal to that produced by temperature t.

The tension is independent of the length, and varies solely as the closing temperature, which should not exceed 100° C. (212° F.). Adhesion due to this temperature $= 9{\cdot}4$ per sq. mm. (= 6 tons per sq. inch), and at 150° C. (302° F.) = 14 (= 9 tons per sq. inch).—PLANAT.

406. RIVETING IN BOILER OR TANK WORK.

t = thickness of plates in inches.
d = diameter of rivets ,,
p = pitch ,, ,,
l = lap of plates ,,
L = length of rivet ,,
ll = lap of plates for double riveting.

$$d = t + \tfrac{5}{16}$$
$$p = 1{\cdot}6\,t + 1\tfrac{1}{4}$$
$$l = 3\,t + 1\tfrac{1}{8}$$

Another rule :—

$$d = 1\tfrac{1}{8}\,t + \tfrac{1}{4}$$
$$L = 4\tfrac{1}{8}\,t + \tfrac{1}{8}$$
$$p = 4\,d$$
$$l = p + \tfrac{1}{4}$$
$$ll = 1\tfrac{3}{8}\,l$$

407. RIVETS IN TIE BARS AND DIAGONAL RIVETING GENERALLY.

Prof. Kennedy, in his "Abstract of Results of Experiments on Riveted Joints," as made by the Research Committee of the Inst. Mech. Eng., says: "It has been found that the net metal measured zigzag should be from 30 to 35 per cent. in excess of that measured straight across in order to ensure a straight fracture. This corresponds to a diagonal pitch of $\frac{2}{3}p + \frac{d}{3}$, if p = the straight pitch and d the diameter of the rivet hole."

408. NOTES ON CAULKING.

Caulking consists of burring up the inner edge of the plates in a joint by means of a tool like a flat-ended chisel, to prevent leakage in boilers, tanks, etc.

Plates with rough sheared edges should be chipped even, to a slight bevel, before caulking.

Joints appearing at all open should be closed by a flogging hammer before caulking.

When the caulking is done on one side only, it should be upon the same side as the riveting. In best work the joints are caulked inside and out.

When the lap exceeds three times diameter of rivet the caulking is apt to open the joint, unless done very lightly.

409. CAULKING TOOLS.

The caulking tool should be flat-ended and slightly bevelled, from $\frac{1}{8}$ inch to $\frac{3}{16}$ inch thick × 1 inch to $1\frac{1}{4}$ inch wide, with one edge square, and the other rounded to prevent cutting into the plate.

The rounded edge should be held next to the plate the first time of going along the joint, called "splitting the lap," and afterwards reversed.

The finished caulking should appear like a parallel groove about $\frac{1}{32}$ inch deep × $\frac{1}{8}$ inch wide in a $\frac{3}{8}$-inch plate.

Section VI.

WORKSHOP TOOLS AND GENERAL MACHINERY.

410. OBJECT OF MACHINES.

THE object of machines is to change the direction of motion, or to regulate the distribution of power. They transmit energy and modify it in direction, intensity, or velocity, but they can neither create nor increase power. An *engine* transmitting energy from *natural forces* is called *a prime mover*, but is otherwise a machine.

The POWER of a machine is measured by the WORK which can be done in a given TIME.

Machines are used for—

1. Accumulating force upon a given point or object.
2. Increasing or decreasing velocity of motion.
3. Prolonging the action of a power.
4. Changing the direction of motion.
5. Reducing the time of labour.
6. Producing accuracy in work.

The parts may be divided into—

1. Receivers. 2. Communicators. 3. Operators.

Motive Power may be derived from—

1. Man and animals.
2. Fall of water.
3. Force of wind.
4. Descent of weights.
5. Action of springs.
6. Expansion of elastic fluids.
7. Electricity and magnetism.
8. Chemical reactions.

A *machine* is a combination of resistant bodies, whose relative motions are completely restrained, and by means of which the natural energies at our disposal may be transformed into any special form of work.—PROF. KENNEDY.

A *mechanism* consists of a combination of simple links, arranged so as to give the same relative motions as the machine, but not necessarily possessing the resistant qualities of the machine parts; thus a mechanism may be regarded as a skeleton form of a machine.—PROF. GOODMAN.

411. CONSTRAINED AND FREE MOTION.

Motion may be constrained or free. A body which is free to move in any direction relatively to another body is said to have *free* motion, but a body which is constrained to move in a definite path is said to have *constrained* motion. In both cases the body moves in the direction of the resultant of all the forces acting upon it; but in the latter case, if any of the forces do not act in the direction of the desired path, they automatically bring into play constraining forces in the shape of stresses in the machine parts.—PROF. GOODMAN.

412. MACHINERY IN MOTION.

In engines or machines in motion, when the power exceeds the work the speed will be accelerated, unless prevented, until the resistance + the useful work = the power. When the resistance + the useful work exceeds the power, the speed will be retarded until a balance is again obtained. In the former case the inertia of the parts will absorb some of the power, and in the latter this power will be again given out as momentum.

Motion may be rectilinear or curvilinear—direct or reciprocating—uniform or variable (uniformly accelerated, uniformly retarded, or irregular).

413. USEFUL WORK AND EFFICIENCY.

Useful work of a machine is that performed in producing the effect for which the machine is designed.

Lost work is that performed in producing other effects, as overcoming friction, loss by leakage, etc.

The *power* of a machine is the energy exerted, and the *effect* the useful work performed, in some interval of time of definite length.

The *efficiency* [or *mechanical efficiency*] of a machine is a fraction expressing the ratio of the useful work to the whole work performed or energy expended. This ratio is also called the *modulus* or *coefficient* of the machine.

The *counter-efficiency* is the reciprocal of the efficiency, and is the ratio in which the energy expended is greater than the useful work.—RANKINE'S "APPLIED MECHANICS."

Most machines when tested carefully are found to have a coefficient of the form $P = x + y\frac{W}{r}$, P being the power applied, W the load moved, r the velocity ratio, or multiplying power, or mechanical advantage, x and y constants.

414. ECONOMICAL WORKING OF MACHINES.

In every machine a certain rate of work develops the maximum efficiency. A medium load with a fair velocity produces more units of work than a heavier load with a less velocity, or a lighter load with a greater velocity.

415. MECHANICAL EFFICIENCY OF VARIOUS MACHINES.

From experiments in all cases with more than quarter full load.

	Per cent.
Weston pulley block (½ ton)	20 to 25
Epicycloidal pulley block	40 ,, 45
One ton steam hoists or windlasses	50 ,, 70
Hydraulic windlass	60 ,, 80
,, jack	80 ,, 90
Cranes (steam)	60 ,, 70
Travelling overhead cranes	30 ,, 50
Locomotives $\frac{\text{draw bar H.P}}{\text{I.H.P.}}$	65 ,, 75

Lancashire Cotton Mills (*see* Proceedings of the Manchester Association of Engineers, 1892)		
	About 1,000 h.-p. engines, spur-gearing and engine friction	74
	Rope drives	70
	Belt ,,	71
	Direct (400 h.-p. engines)	76

—PROF. GOODMAN.

416. LOSS OF USEFUL ENERGY.

"As a common example, and one which gives a fair idea of the magnitudes of some of these losses, take the following case of electrical power production. Suppose 100 units of energy to be liberated from some coal in the boiler furnace. About 75 of these will enter the steam and the remaining 25 will be lost by the passage of the smoke and heated gases up the chimney, or by radiation and other causes. Of the 75 units of energy reaching the engine in the steam, about 6 will be converted into mechanical energy and the remaining 69 will be lost. The 6 units of mechanical energy given to the dynamo will produce about 5 units of electrical energy, 1 unit being lost. If these 5 units be reconverted into mechanical energy by an electrical motor, about 4·5 units will be produced. We utilise in this way about 4·5 per cent. of the original energy and lose 95·5 per cent."—J. DUNCAN.

"Out of an average of 13,500 British Thermal Units which are contained in a pound of good coal, not more than the mechanical equivalent of 1,200 units are delivered to the belt from the steam engine. If these 1,200 units are further transformed into electricity for lighting purposes, less than 1,000 of them will be delivered to the lamp; if the electricity is re-converted into brake-power in a motor, less than 900 will ultimately be available for driving purposes."—"POWER."

417. PIECEWORK RATES, ROWAN SYSTEM.

a = ordinary time-wage rate of pay.
T = standard time allowed for completion of the job.
t = actual time taken.

$$\text{Piece work rate per hour} = a + a\left(\frac{T-t}{T}\right).$$

—JAMES ROWAN, Glasgow.

418. VELOCITY RATIO.

The *velocity ratio* in any machine is the proportion between the movement of the power and the movement of the resistance, in the same interval of time; for example, in a punching press it may be 100 to 1 = $\frac{100}{1}$, and in a hydraulic crane 1 to 8 = $\frac{1}{8}$. These proportions also express the amount of the resistance (including friction), compared with the power or pressure applied. (See also the definitions of *virtual velocity*.)

The term *purchase* of a machine is applied either to the motion or pressure of the resistance compared with the power; in above examples, the purchase of the punching press would be 100, that of the hydraulic crane 8, but the term is generally restricted to the gaining of pressure by the sacrifice of speed, as in the first case.

By the *mechanical advantage* of any machine is meant the ratio of the weight (or resistance) to the power, when in equilibrium, friction being supposed to be absent. It is then inversely as the velocity ratio. This is sometimes improperly called the mechanical advantage, and the term *real or actual mechanical advantage* is also used occasionally, but improperly, to express the mechanical efficiency.

419. HODOGRAPH.

"When a point is moving along any kind of curve let us suppose that through some other point, which is kept fixed, a line is always drawn which represents the velocity of the moving point both in magnitude and direction. Since the velocity of the moving point will in general change, the line will also change both in size and direction, and the end of it will trace out some sort of curve. This curve, described by the end of the line which represents velocity at any instant, may be regarded as a map of the motion, and was for that reason called by Hamilton the *Hodograph*. If we know the path of the moving point and also the hodograph of the motion we can find the velocity of the moving point at any particular position in its path. All we have to do is to draw through the centre of reference of the hodograph a line parallel to the tangent to the path at the given position; the length of this line will give the rate of motion, or the velocity of the point as it passes through that position in its path. The great use of the hodograph is to give us a clear conception of the rate of change of the velocity; this rate of change is called the *acceleration*."—W. K. CLIFFORD.

420. INSTANTANEOUS VELOCITY.

If there is a certain velocity to which the mean velocity during the interval succeeding a given instant can be made to approach as near as we like by taking the interval small enough, then that velocity is called the instantaneous velocity of the body at the given instant.—W. K. CLIFFORD.

421. INSTANTANEOUS CENTRE.

The plane motion of a body is completely known when we know the motion of any two points in the body. If the paths of the points be circular and concentric, then the centre of rotation will be the same for all positions of the body. Such a centre is termed a *permanent* or *fixed* centre; but when the centre shifts as the body shifts, its centre at any given instant is termed its *instantaneous* or *virtual* centre.

Virtual radii are the distances from the virtual centre to any points in the body.

The *point-path* of the virtual centres in any motion is known as the *centrode* or *axode*.—PROF. GOODMAN.

422. INSTANTANEOUS CENTRE OF CONNECTING ROD.

The instantaneous centre of a connecting rod at any part of its travel is found by drawing a perpendicular at the crosshead from the guide bars and producing the centre line of the crank to meet it. Let v = velocity of crank pin, A I = radius from crosshead to instantaneous centre, B I = radius from crank pin to instantaneous centre, V = velocity of crosshead, then $V = v \times \frac{A\,I}{B\,I}$. If G = any other point in the connecting rod and G I = radius from G to I, then velocity of G $= V' = v \times \frac{G\,I}{B\,I}$.—J. Duncan.

423. PRINCIPLE OF VIRTUAL VELOCITIES (GALILEO).

If any machine without friction be in equilibrium and the whole be put in motion, the initial pressure P will be to the final pressure p as the final velocity V is to the initial velocity v, or $P : p :: V : v$, or $p\,V = P\,v$.

Instead of velocity (V and v) we may take "space moved over" (S and s).

In practice, as all machines have friction, p will depend upon the friction, but V will be in accordance with the calculation of the leverage or gearing.

Let e = the final pressure by experiment, then $p - e$ = friction, and the coefficient or modulus of machine $M = \frac{e}{p}$.

424. DEFINITIONS OF THE PRINCIPLE OF VIRTUAL VELOCITIES.

Rankine's.—The effort and resistance are to each other inversely as the velocities, along their lines of action, of the points where they are applied.

Twisden's.—If a system of pressures, in equilibrium, act on any machine which receives any small displacement, consistent with the connection of the parts of the machine, the algebraical sum of the virtual moments of the pressure will equal zero.

425. WORK, IN TERMS OF ANGULAR MOTION.

r = radius = leaverage.
$2\pi r$ = circumference.
p = pressure at circumference.
$r\,p$ = moment of pressure.
n = number of revolutions.

$2\pi n$ = angular motion.
$r\,p \times 2\pi n$ = foot-lbs. work performed.
rate of work = work performed in a unit of time as 1 second or 1 minute.

—Rankine.

426. ANGULAR VELOCITY.

The angular velocity of a wheel, or velocity of spin, is the speed of a point in the circumference of an imaginary wheel with unity as radius, and making the same number of revolutions per minute as the given wheel.

Velocity is taken in feet per second.

Revolutions are taken at per minute.

$$\text{Circumferential velocity} = \frac{2\pi r n}{60} = \frac{\pi r n}{30} = \cdot 10472\, r\, n.$$

$$\text{Angular velocity} = \frac{2\pi (r) n}{60} = \frac{\pi n}{30} = \cdot 10472\, n.$$

$$\frac{\text{Velocity of any point in wheel}}{\text{radius of ditto in feet}} = \text{angular velocity in radians per second.}$$

427. ANGULAR MEASUREMENT OF FORCES.

The *circular measure* of an angle is $\frac{\pi \times \text{angle}}{180}$.

A *radian*, or unit of angular rotation, is an arc of a length equal to radius; it contains 57·2958 degrees $= \frac{180^\circ}{\pi}$. A right angle therefore contains 1·5708 radians, two right angles 3·1416 radians, and four right angles 6·2832 radians; or one revolution $= 2\pi$ radians and

$$n \text{ revolutions per minute} = \frac{2\pi n}{60} \text{ radians per second.}$$

$$\frac{\text{Degrees in an angle}}{57\cdot 2958} = \text{No. of radians.}$$

$$\text{Radius} \times \text{No. of radians} = \text{length of arc.}$$

The angular velocity of a wheel may be measured in radians per second.

The *chord of an angle* $= \frac{\text{chord}}{\text{radius}}$.

A *round* is the angular space traversed in one revolution. A round contains 6·2832 radians. The linear velocity of a point in a wheel is equal to the angular velocity $\times$ the distance in feet of the point from the axis. All points in a revolving wheel have the same angular velocity.

A *torque* (Jas. Thomson) is a system of forces, not meeting in one point, which, acting upon a body, may be parallel to and proportional to the sides of a closed polygon, but whose turning moments do not balance about any axis. It is equivalent to a "couple." In machinery it means turning moment, or turning force × distance from centre of shaft.

In Ayrton and Perry's dynamometer coupling, or transmission dynamometer, the total amount of the forces of the springs in pound-feet, or the "torque," × angular velocity per minute ÷ 33,000 = the horse-power, thus:

$$\text{H.P.} = \frac{\text{torque} \times \text{angular } v. \text{ per minute}}{33{,}000}.$$

428. ANGLE OF TWIST.

A straight line drawn along a shaft not transmitting power becomes a spiral while power is being transmitted. The angle between the spiral line at any point and the original direction divided by the radius of the shaft is called the angle of twist.—Perry.

429. WORKSHOP TOOLS

are divided into two classes, hand tools and machine tools. In the former are included hammers, chisels, files, ratchet braces, spanners, etc.; and in the latter, lathes, planing, shaping, drilling, and slotting machines, etc., in the fitting shop; and punching and shearing machines, bending rolls, steam hammers, etc., in the smiths' shop.

The machine tools are now mostly driven by steam power through shafting connected by belts, or by electric motors.

A workshop should be so arranged that the raw material coming in at one end would be received at the various tools in the order of the work to be done upon it, and be removed in a finished state at the other end.

430. HAMMERS.

Name.	*Weight. lbs.*	*Length of Shaft. ins.*
Sledge	28, 24, 18, and 14	40
Flogging . . .	7 and 5	30
Riveting . . .	4 and 3	24
Hand	2	20
Fitting	$1\frac{3}{4}$	16
Bench	$1\frac{1}{2}$	14
,, ...	1	12

431. WORK OF HAMMER.*

Hammer 2 lbs., velocity 20 feet per second, drives nail $\frac{1}{2}$ inch into hard wood; required the equivalent dead pressure. (v. after striking = 20 to 0, mean = 10; therefore t in driving $\frac{1}{2}$ inch = $\frac{1}{240}$ of a second.)

1st by $\quad F = \frac{W v}{g t}, \quad \frac{2}{32 \cdot 2} \times \frac{20}{\frac{1}{240}} = 298$ lbs.

2nd by $\quad \frac{W v^2}{2 g}, \quad \frac{2}{32 \cdot 2} \times \frac{20^2}{2} = 12 \cdot 42$ ft.-lbs., and $\frac{12 \cdot 42}{\frac{1}{24}}$

= 298 lbs. as before,

but this will only be the *mean* pressure. From experiments it appears that the maximum pressure required is about $1\frac{3}{4}$ times mean pressure, so that the actual dead pressure required to force same nail same depth would be 298 $\times$ 1·75 = 521·5 lbs., and the force required to extract it, being about $\frac{4}{5}$ of pressure to insert it, would be $521 \cdot 5 \times \frac{4}{5} = 417$ lbs. Where the resistance varies simply as the depth driven, the maximum pressure is double the mean. The same principles apply to pile driving.

432. IMPACT OF MOVING BODIES.

In these formulæ mass may be substituted for weight without affecting the result.

W =	weight of	body	A	giving blow	
V =	velocity	,,	A	,,	
V^1 =	,,	,,	A	,,	after impact.
w =	weight	,,	B	receiving blow	
v =	velocity	,,	B	,,	
v^1 =	,,	,,	B	,,	,,

Bodies Perfectly Soft or Inelastic.	Bodies Perfectly Elastic.

(1) Both moving in same direction.

Inelastic:

$$V^1 = v^1 = \frac{W V + w v}{W + w}.$$

Elastic:

$$V^1 = 2 \frac{W V + w v}{W + w} - V.$$

$$v^1 = 2 \frac{W V + w v}{W + w} - v.$$

* See papers by the author on "The Force of Hammers; or, Percussion *v.* Pressure," and "Timber Piling in Foundations and other Works."

(2) A moving, B at rest.

$$V^1 = v^1 = \frac{W\,V}{W + w}.$$

$$V^1 = 2\frac{W\,V}{W + w} - V.$$

$$v^1 = 2\frac{W\,V}{W + w}.$$

(3) Both moving in opposite directions.

$$V^1 = v^1 = \frac{W\,V - w\,v}{W + w}.$$

$$V^1 = 2\frac{W\,V - w\,v}{W + w} - V.$$

$$v^1 = 2\frac{W\,V - w\,v}{W + w} - v.$$

Resulting motion towards A if result —.

" " B " +.

Or, putting R = mutual action between two bodies moving in opposite directions,

$$R = \frac{W\,w\,(V + v)}{W + w}.$$

$$V^1 = \frac{w\,v - R}{w}.$$

$$v^1 = \frac{W\,V - R}{W}.$$

$$V^1 = \frac{w\,v - 2\,R}{w}.$$

$$v^1 = \frac{W\,V - 2\,R}{W}.$$

For intermediate condition of matter, between perfectly soft and perfectly elastic, use coefficient e R.

Example of Case 2.

Body A weighing W = 10 lbs., moving at velocity V = 20 feet per second, strikes body B weighing w = 30 lbs. at rest. When perfectly soft or inelastic $\frac{W\,V}{W + w} = \frac{10 \times 20}{10 + 30} = 5$ feet per second as the resulting velocity of A and B moving together. But, by formula for kinetic energy, if the units of work existing in A remain in the combined masses after striking, $\frac{W\,V^2}{2\,g} = \frac{(W + w)\,V_1{}^2}{2\,g}$, the resulting velocity would appear to be

$$V_1 = V\sqrt{\frac{W}{W + w}} = 20\sqrt{\frac{10}{10 + 30}} = 10 \text{ feet per second.}$$

The explanation is that the total *momentum* is always the same, but the *energy* is only constant when the bodies are perfectly elastic—i.e., when the restitution is complete. When the elasticity is imperfect, part of the *work* is used in compressing the particles, and the lost velocity is transformed into *heat*.

If the same bodies were perfectly elastic the resulting velocity of A would be

$$V^1 = 2\frac{W V}{W + w} - V = 2\left(\frac{10 \times 20}{10 + 30}\right) - 20 = -10 \text{ ft. per second,}$$

i.e., it would rebound at half the striking velocity, and the resulting velocity of B would be

$$v^1 = 2\frac{W V}{W + w} = 2\left(\frac{10 \times 20}{10 + 30}\right) = 10 \text{ feet per second in forward direction.}$$

The energy before and after would be

$$\frac{W V^2}{2g} = \frac{W V_1{}^2}{2g} + \frac{w v_1{}^2}{2g},$$

$10 \times 20^2 = (10 \times -10^2) + (30 \times 10^2)$, or $4000 = 1000 + 3000$.—Q.E.D.

433. COEFFICIENT OF RESTITUTION.

Newton found that "when two bodies collide, the velocity of separation after impact is in a constant ratio to the velocity of approach before impact." This ratio depends on the nature of the colliding bodies, and is sometimes called the *coefficient of restitution* (e) or of rebound. The law may also be expressed as "the relative velocity after impact is $-e$ times the relative velocity before."

Let m_1 m_2 be the two masses, and u_1 u_2 their respective velocities before and v_1 v_2 after impact. Then since the momentum is unaltered,

$$m_1 v_1 + m_2 v_2 = m_1 u_1 + m_2 u_2$$

Since the relative velocities after and before are in the ratio $-e$.

$$v_1 - v_2 = -e\,(u_1 - u_2).$$

From these two equations v_1 v_2 can be found. To do this multiply the latter throughout by m_2, then

$$m_2 v_1 - m_2 v_2 = m_2 e\,(u_1 - u_2)$$

and this to the first, and

$$m_1 v_1 + m_2 v_1 = m_1 u_1 - m_2 e u_1 + m_2 (l + e) u_2$$

$$(m_1 + m_2) v_1 = (m_1 + m_2) u_1 - m_2 (l + e) u_1 + m_2 (l + e) u_2$$

$$v_1 = u_1 - (l + e)\frac{m_2}{m_1 + m_2}(u_1 - u_2)$$

and similarly

$$v_2 = u_2 - (l + e)\frac{m_1}{m_1 + m_2}(u_2 - u_1)$$

which equations completely give the motion.—W. M. HICKS.

434. SHOP SHAFTING.

Average velocity of shop shafting 100 revolutions per minute.

Average h.p. transmitted by shafting $= d^3 \times$ revns. per minute $\div$ 75.

Average size of shop shafting 2-inch to 4-inch diameter.

435. ENGINEERS' FILES.

The various degrees of fineness are classified as follows, both single and double cut:—

Rough, middle, bastard, second cut, smooth, dead smooth.

436. HACK SAWS

are made from $12 \times \frac{5}{8} \times 21^s$ gauge with 12 points per inch to $17 \times 1 \times 16^s$ gauge with 8 points per inch. The distance centre to centre of holes is half inch less than full length.

437. ENGINE FITTERS' VICE.

Top of vice jaws from floor = 40 inches to 44 inches, say average of 42 inches, or level with the elbow.

438. HOLTZAPFFEL'S CLASSIFICATION OF CUTTING TOOLS.

Shearing tools act by dividing the material operated on into two parts, which separate from each other by sliding at the surface of separation.

Paring tools cut a thin layer or strip called a shaving from the surface of the work, and thus produce a new surface.

Scraping tools scrape away small particles from the surface of the work, thus correcting the small irregularities which may have been left by the paring tool.

439. ANGLES OF TOOLS.

	Angle of Tool.
For wood	30° to 40°
,, wrought iron	60°
,, cast iron	70°
,, brass	80°

Angle of relief for all tools, 3° to 10°

440. GRINDSTONES.

$$\text{Weight of grindstone in lbs.} = \frac{d \text{ ins.}^2 \times t \text{ ins.}}{16}$$

441. CUTTING SPEED OF MACHINE TOOLS.

	Ft. per Min.
Cast steel	10 to 12
Mild	12 ,, 15
Cast iron	15 ,, 20
Wrought iron	15 ,, 25
Gun metal	20 ,, 40
Yellow brass	40 ,, 60
Wood (when material revolves)	500 ,, 2000
Wood (when tool revolves)	3000 ,, 5000
Grindstone	800
Milling wrought iron	80 ,, 100
,, cast steel	25 ,, 30

Average for wrought or cast iron in lathe, shaping, slotting, etc., 20 feet per minute.

Generally the cutting speed should be as fast as possible without the tool overheating and losing its temper.

442. AVERAGE CUTTING SPEEDS AND FEEDS.

Material.	*Roughing.*		*Finishing.*	
	Speed.	*Feed.*	*Speed.*	*Feed.*
	ft. per min.	*cuts per in.*	*ft. per min.*	*cuts per in.*
Wrought iron	25	20	25	25
Steel	18	25	15	30
Cast iron	25	16	25	6

443. SPEED OF MACHINE TOOLS.

Speed of cut { Wrought iron, 20 feet per minute.
Cast ,, 16 ,, ,,
Cuts per inch, 16 to 80.

For flat work:

Speed in inches per second × 5 = speed in feet per minute.

For small diameters:

Diameter in inches × revolutions in 16 seconds = speed in feet per min.

For large diameters:

$$\frac{\text{Diam. in inches} \times 16}{\text{Seconds for 1 revolution}} = \text{speed in feet per minute.}$$

$$\frac{\text{Cutting speed in feet per min.} \times 5}{\text{Cuts per inch.}} = \text{sq. feet tooled per hour.}$$

—"ENGINEERING."

444. CUTTING SPEEDS.

Shearing and punching	2 ft. per minute.
Turning malleable cast iron	3 ,,
Screwing	6 ,,
Turning steel	10 ,,
,, cast iron.	16 ,,
,, wrought iron	21 ,,
,, bronze	30 ,,

—"ENGLISH MECHANIC."

445. SPEED IN CUTTING METALS.

Turning chilled rolls	3	to	4	ft. per minute.
Screw-cutting steel in lathe			$7\frac{1}{2}$	,,
Turning and planing steel			10	,,
Boring cast-iron cylinders			12	,,
Turning, planing, and shaping cast iron	15	,,	20	,,
Do. do. wrought iron and very soft cast iron	20	,,	40	,,
Do. do. steel	24	,,	30	,,
Do. do. brass	36	,,	100	,,
Screw-cutting gun-metal	30			,,
Turning copper	30			,,
Band-saws for hot iron and steel	200	,,	300	,,
Circular saws for do. do.	12,000	to	27,600	,,

—KEERAYEFF.

Circular saw, consisting of soft iron disc running at circumferential speed of 12,000 feet per minute, is used for cutting ends of steel rails, with jet of water playing on circumference of saw.

Armour plate and bars of high-speed tool steel are cut by soft steel discs 7 ft. diameter with a peripheral speed of about 4 miles per minute, having a roughened edge but no teeth. A jet of water is played upon the cut. The harder the metal the easier it is cut.—"THE ENGINEER."

446. PLANER FORMULÆ.

The old practice of judging the comparative values of planing machines by comparing their speeds on cut and return has been found very misleading. This is because of the momentary stoppage of the table at each end of the stroke and the time lost before full speed is attained after reversal. In some machines these losses are very considerable, and materially reduce the productiveness of the tool.

The only accurate means of ascertaining the earning capacity of a planer is to take the cycle times as indicated below :—

Time of 1 cycle = time of 1 cut + time of 1 return.

L = length of stroke in feet.

T = time of N cycles in seconds.

N = number of cycles.

$$\text{Average (or earning) speed} = \frac{2\,L \times N \times 60}{T}.$$

Example :—A 42-inch by 14 feet electrically driven machine, having a mean forward speed of 48 feet per minute and return of 147 feet per minute, completes 10 cycles in 3 minutes 56 seconds (236 seconds) when on a 14-foot stroke. Therefore the average speed is $\frac{14 \times 2 \times 10 \times 60}{236} = 71$ feet per minute.—Bateman's Machine Tool Co., Ltd.

447. PERIPHERAL SPEED OF DRILLS.

Brass	25 to 40 ft. per minute.
Wrought iron	20 „ 25 „
Cast iron	15 „ 17 „
Steel	14 „ 20 „
Feeds up to ½ inch	200 revns. to 1 in feed.
Feeds ½ inch to 1½ inch . . .	150 „ „
Feeds above 1½ inch	100 „ „

—J. W. E. Littledale.

448. RESISTANCES IN MACHINE TOOLS.

Twist Drill.

Pressure on head of twist drill in lbs. requisite to produce proper cut = diameter of drill in inches and decimals × 1500.

LATHE.

Material.	*Width of Cut.*	*Depth of Cut.*	*Speed of Cut.*	*Resistance to Traverse of Tool.*
	inches.	*inches.*	*ft. per min.*	*lbs.*
Steel . . .	$\frac{1}{20}$	$\frac{1}{20}$	5	600
Wrought iron . .	$\frac{1}{20}$	$\frac{1}{10}$	10	700
Cast iron . .	$\frac{1}{20}$	$\frac{1}{16}$	15	325

PLANING MACHINE.

Cast iron, width of cut $\frac{1}{16}$ inch, speed of cut 11 feet per minute. With depth of cut = $\frac{1}{32}$ inch pressure against tool varied from 356 to 396 lbs., averaging 373 lbs., or 4065 ft.-lbs. work per minute. With depth = $\frac{1}{16}$ inch: pressure varied from 340 to 559 lbs., averaging 458 lbs., or 5000 foot-lbs. work per minute.

449. TOOLS FROM BETHLEHEM STEEL CO., U.S.A.

Lathe cuts $\frac{3}{16}$ inch deep $\frac{1}{16}$ inch feed. Feet tooled per minute—soft, 150; medium hard, 60; very hard, 15.

This steel retains its temper and edge when heated to a visible red heat.

Average 137·3 lbs. of metal removed per hour per tool with cut 0·3 inch deep, 0·087 inch feed, 25 feet per minute.

450. SPEED OF MILLING CUTTERS.

	Ft. per min.		*In. per min.*
For brass . . .	120	Feed	2·66
,, cast iron . . .	60	,, ...	1·66
,, wrought iron . .	48	,, ...	1·00
,, steel	36	,, ...	0·50

Angle of teeth 70°, clearance angle 10°.

4 inches diameter = 35 teeth, 6 inches diameter = 43 teeth, 8 inches diameter = 51 teeth.—ADDY.

Material.	*Roughing cut.*		*Finishing cut.*	
	ft. per min.	*rev. per min.*	*ft. per min.*	*rev. per min.*
Steel . . .	30	$57/r$	40	$76/r$
Wrought iron . .	40	$96/r$	55	$105/r$
Cast iron . .	60	$114/r$	75	$143/r$
Brass . . .	100	$190/r$	120	$228/r$

r = radius of cutter in inches. —J. W. E. LITTLEDALE.

MILLING CUTTERS MADE FROM "BÖHLER" STEEL.

Diameter of cutter . .	36 mm.	Feed	5 mm.
Revolutions per minute .	110	Length of cut . .	80 „
Travel per minute . .	32 mm.	Weight of steel cut per hour	6 kilos.

The above results have been obtained by work on pieces of middling hard steel—that is to say, a steel equal to a resistance of 55 to 60 kilos. per mm.; the average time occupied has been 5 hours, without showing any deterioration to the tool.

451. POWER REQUIRED TO DRIVE LATHE.

HP_L = Horse-power absorbed running light.
HP_W = „ „ in work.
HP = Total „ „ $= HP_L + HP_W$.

N = number of revolutions per minute.
C = constant depending on material and class of tool, average 0·026 cast iron, 0·030 wrought iron, 0·044 steel.
W = weight of chips removed per hour in lbs. $= 18{\cdot}7\ S\,d\,f$.
S = cutting speed in feet per minute.
d = depth of cut in inches.
f = feed per revolution in inches.

Small Lathe (under 20-inch swing).

Back gear thrown out $HP_L = 0{\cdot}095 + 0{\cdot}0012\ N$.
„ „ in „ $= 0{\cdot}10 + 0{\cdot}006\ N$.
„ „ „ $HP_W = C\,W$. —FLATHERS.

Under ordinary conditions the same horse-power would remove 6 lbs. of cast iron, 5 lbs. of wrought iron, and 3½ lbs. of steel chips per hour.—"MECHANICAL WORLD."

In the lathe tool dynamometer experiments by Prof. J. T. Nicholson, D.Sc., Manchester, it was found that the maximum endurance in cutting cast iron was given by a tool having a clearance angle of 6° and included angle of 75°, or total cutting angle from vertical of 81°, and a plan angle of 45°.

In cutting mild steel a clearance angle of 5°, included angle 65°, total angle 70°, and plan angle 45°, gave the best results.

The power required per lb. of material removed was about 2 H.P. for both cast iron and steel.

452. SCREW CUTTING.

Set of change wheels numbers 22; increasing by 5 teeth from 20 to 120, two being alike, generally 80 or 90. When 25 in a set, the extra wheels are 130, 140, and 150.

Wheels of 10 and 15 teeth are supplied when the screw-cutting gear works the slide rest.

Leading screw has usually 2, 3, or 4 threads per inch.

Double train must always be used when $\dfrac{\text{leading screw}}{\text{screw required}}$ is less than $\frac{1}{6}$, generally when less than $\frac{1}{4}$.

When the number of threads per inch required to be cut can be divided without remainder by the number of threads per inch in the leading screw, the clamping nut under the saddle will drop into gear with the leading screw without chalking.

Always retain the mandrel wheel for a screw-cutting train when possible.

The "pitch" of a thread is the distance along the axis from one thread to a similar point on the next.

Generally, "threads per inch" means projections per inch, so that a double thread screw with 4 threads to the inch would have a true "pitch" of $\frac{1}{2}$ inch.

To Find the Wheels for any Pitch.

Single train—

$$\frac{\text{Threads per inch in leading screw}}{\text{,, \quad ,, \quad screw to be cut}} = \frac{\text{driver}}{\text{follower}}.$$

Double train—

$$\frac{\text{Threads leading screw}}{\text{Threads screw required}} = \frac{\text{driver}}{\text{follower}} \times \frac{\text{driver}}{\text{follower}}.$$

Examples of Change Wheels.

Single trains—

Leading screw, 4 threads
Required ,, 7 ,,

$$\frac{4}{7} \times \frac{5}{5} = \frac{20}{35}, \text{ or } \times \frac{15}{15} = \frac{60}{105}$$

Leading ,, 4 ,,
Required ,, $2\frac{3}{4}$,,

$$\frac{4}{2\frac{3}{4}} = \frac{16}{11} \times \frac{10}{10} = \frac{160}{110} + \frac{2}{2} = \frac{80}{55}$$

Leading ,, 4 ,,
Required ,, ·75 pitch

$$\frac{\cdot 75 \times 100 \times 4}{100} = \frac{300}{100} = \frac{30}{10} \times \frac{4}{4} = \frac{120}{40}$$

Double trains—

Leading screw, 4 threads
Required „ $\frac{5}{8}$ pitch
$$\frac{5 \times 4}{8} = \frac{5 \times 4}{2 \times 4} = \frac{50 \times 40}{20 \times 40} = \frac{\mathbf{50 \times 80}}{\mathbf{20 \times 80}}$$

Leading „ 4 threads
Required „ 100 „
$$\frac{4}{100} = \frac{2 \times 2}{5 \times 20} = \frac{20 \times 20}{50 \times 200} = \frac{\mathbf{20 \times 10}}{\mathbf{50 \times 100}}$$

Leading „ 4 „
Required „ ·08 pitch
$$\frac{4 \times \cdot 08 \times 100}{100} = \frac{4 \times 8}{10 \times 10} = \frac{40 \times 80}{100 \times 100} = \frac{\mathbf{20 \times 80}}{\mathbf{50 \times 100}}$$

Trains to be used are shown in broad-faced type.

453. SCREW FOR WORM WHEEL.

To find change wheels to cut screw,

- D = Diametrical pitch of worm wheel.
- d = diameter of worm wheel at pitch circle.
- n = number of threads per inch in leading screw.
- t = number of teeth in wheel.

$$\frac{\pi n}{D} = \frac{22 n}{7 D} = \frac{Driver}{Follower},$$

$$\frac{\pi d n}{t} = \frac{22 d n}{7 t} = \frac{Driver}{Follower}.$$

To cut double, treble, or more threads or worms :—

Find the smallest set of wheels that will cut the required pitch single thread, then multiply the driver by the number of threads required.

454. VELOCITY OF WOOD-WORKING MACHINERY.

Saw frame (several saws)	8 ft.	per second.
„ (one saw)	10 to 15	„
Band saw	40 to 50	„
Turning wood	15 to 40	„
Revolving cutters	60 to 100	„
Circular saw (across grain) . .	80 to 100	„
„ (with grain) . .	100 to 130	„

—Keerayeff.

To saw green oak lengthways requires 29,000 ft.-lbs. work per foot super.

Planer and tenoner . . .	4500 to 5000	ft. per minute.
14-in. rip-saw	2500 „ 2800	revns. per minute.
Vertical spindle moulder . . .	6500 „ 7000	„ „

—" Woodcraft."

9*

Machine.	*Cutting Speed.*	*Approx. h.p. req.*
Circular rip saw	7,000 to 9,000 ft. per minute	$\frac{\text{diam.}^2}{40}$
Band saw . . .	3,500 ,,	$\frac{\text{diam. wheel}^2}{300}$
Planer .	.3,500 to 10,000 ,,	$\frac{\text{width cut}}{4}$
Boring machine .	375 to 750 revns. per minute	$\frac{\text{diam. hole}}{2}$
Pattern-makers' lathe	900 ,, ,,	$\frac{\text{diam. stuff}}{6}$
Moulder . . .	5,000 ,, ,,	width + depth of cut

—PROF. DODGE.

CIRCULAR SAWS FOR WOOD.

Provide 1 B.H.P. for every 1 inch depth of cut required, and diameter of saw 3 times depth of cut.

455. SPEED OF POLISHING AND GRINDING.

Tool grindstone.	400 to 900 ft. per minute.
Polishing by emery and oil . .	750 ,,
,, by grindstone . . .	2000 ,,
,, by dry emery wheel . .	3000 to 4000 ,,

—KEERAYEFF.

456. ROLLING MILL SPEEDS.

VELOCITY OF ROLLS IN FEET PER SECOND.

Squeezing	3
Plates	4 to $6\frac{1}{2}$
Rails, angles, and tees	$5\frac{1}{2}$
Rods and bars	6 to 8
Fly-wheels for mill	80 to 100
Wire-drawing rollers	1 to $3\frac{1}{2}$
Cold rolling	$\frac{1}{3}$
Plate bending	$.\frac{1}{16}$

—KEERAYEFF.

457. SHEARING AND PUNCHING.

Resistance to shearing of wrought iron averages 50,000 lbs. per sq. inch area of surface cut. This will be the pressure required on the material at the commencement of the stroke.

The mechanical work in punching or shearing is estimated by Weisbach as this pressure exerted through one-sixth the thickness of the plate, and the coefficient or modulus of the machine as ·66, the friction being taken at 33 per cent. of the gross pressure.

For rectangular bars the pressure may be taken as exerted through one-fourth the thickness, and for round bars one-third the diameter.

The energy stored in the flywheel should be double that required for one full-power stroke.

Formula for calculating power required :—

t = Thickness of plate or bar.
l = Length or circumference of cut.
f = Resistance of material to shearing.
M = Modulus of machine, say ·66.
P = Gross pressure in lbs.

$$P = \frac{t\,l\,f}{M}.$$

Pressure required to punch wrought-iron plates (from experiments).

	d		t		P		c
To punch	$\frac{1}{8}$ hole	in	$\frac{1}{8}$ plate	requires	$2\frac{1}{4}$	tons =	144
,,	$\frac{1}{4}$	,,	$\frac{1}{4}$	,,	$6\frac{1}{2}$	,,	104
,,	$\frac{3}{8}$	,,	$\frac{3}{8}$	,,	13	,,	92
,,	$\frac{1}{2}$	,,	$\frac{1}{2}$	,,	22	,,	88
,,	$\frac{5}{8}$	,,	$\frac{5}{8}$	,,	$33\frac{1}{2}$	,,	86
,,	$\frac{3}{4}$	,,	$\frac{3}{4}$	,,	$47\frac{1}{4}$	,,	84
,,	$\frac{7}{8}$	,,	$\frac{7}{8}$	,,	$62\frac{3}{4}$	,,	82
,,	1	,,	1	,,	80	,,	80

$$P = d \times t \times c.$$

Approximately diameter $\times$ thickness $\times$ 88 = pressure in tons; or, area of cut surface $\times$ 28 = pressure in tons.

Diameter of die = diameter of punch $\times$ $1\frac{1}{16}$.

Point of punch coned 5° with hollow curve.

Shearing machine: falling blade bevelled 3° to 8° in elevation, and 15° in section; fixed blade horizontal and square.

458. FLY PRESS.

Shearing resistances may be taken as follows :—

Cold rolled steel	60,000 lbs. per sq. inch.
Mild steel	50,000 ,, ,,
Wrought iron	45,000 ,, ,,
Soft brass	30,000 ,, ,,

—HODGSON.

459. STEAM HAMMERS.

Weight of hammer in lbs. for shaft forging = 80 × diameter shaft inches^2: Weight of anvil = 10 times weight of hammer.

460. STEEL FORGING PRESSES.

Pressure required = 16,000 lbs. per sq. inch on the die.

461. OBSERVED H.P. REQUIRED TO DRIVE SHOP TOOLS.

Small screw-cutting lathe, 12-inch swing	0·33
Screw-cutting lathe, 20-inch swing	0·47
Large facing lathe, 68-inch swing	0·91
Small shaper, 9½-inch stroke	0·24
Shaper, 15-inch stroke	0·63
Large shaper, 29-inch stroke	1·14
Planer, 36 inches × 36 inches × 11 feet	0·84
Large planer, 76 inches × 76 inches × 57 feet	1·47
Small drill press	0·62
Large drill press	1·24
Radial drill, 6-foot swing	0·53
Small slotter, 8-inch stroke	0·28
Medium slotter, 9½-inch stroke	0·44
Large slotter, 15-inch stroke	0·95
Universal milling machine	0·28
Milling machine, 13-inch cutter head, 12 cutters	0·66

Small punch and shear combined, 7½ inches × 1½ inches .	0·79
Large plate shears, knives 28 inches × 3-inch stroke .	7·12
Large punch press, 3-inch stroke through 1½ inches thick .	4·41
Plate bending rolls, 9½ feet × 13 inches	2·70
Wood planer, 28-inch rotary knives	5·00
Fret saw, cutting 1-inch plank	0·3 to 0·5
Circular saw for wood, 23 inches	3·23
Circular saw for wood, 35 inches	5·64
Circular saw, 36 inch, sawing 10-inch teak. . .	14 to 16
Band-saw for wood, 34-inch wheel	0·96
Band-saw, 20 inch sawing 4-inch plank . . .	3 to 4
Tenon and mortising machine	2·73
Wood moulding machine, 7½ inches × 2½ inches . .	2·45
Grindstone for tools, 31 inches × 6 inches, 680 feet per minute	1·55
Grindstone for stock, 42 inches × 12 inches, 1,680 feet per min.	3·11
Emery wheel saw-grinder, 11½ inches × ½ inch . .	0·56

—FLATHERS.

462. FITTING OF TURNED PINS, ETC.

D = nominal diameter in inches.

d = difference of diameter in thousandths of an inch.

Shaking fit	$d = -\ 10\,D$
Easy fit	$d = -\ 5\,D$
Hand fit	$d = -\ 2\,D$
Driving fit	$d = \pm\ 0\,D$
Hydraulic fit	$d = +\ 1\,D$

463. DEPRECIATION OF MACHINERY.

The average depreciation of machinery is considered to be 15 per cent. per annum.

For screw tugs the allowance is 25 per cent. for depreciation, adding the cost of renewals during the year, but not ordinary repairs.

Section VII.

POWER TRANSMISSION BY BELTS, ROPES, CHAINS AND GEARING.

464. TRANSMISSION OF MOTION.

By *rolling contact*, as spur wheels and pinions, crown wheel and pinion, face wheel and lantern, bevel wheels, cones, rack and pinion, etc.

By *sliding contact*, as inclined plane, wedge, cams, swash plate, crown wheel escapement, screw, etc.

By *wrapping contact*, as cords and pulleys, belts and pulleys or riggers, speed pulleys, capstan, fusee of watch, etc.

By *link work*, as levers, cranks, treadle of lathe, etc.—Tomkins' "Machine Construction."

465. NOTES ON BELT GEARING.

Coefficient of friction between ordinary leather belting and cast-iron pulleys or drums = 0·423. Ultimate strength of ordinary leather belting = 3,086 lbs. per sq. inch. Belts vary from $\frac{3}{16}$ inch to $\frac{1}{4}$ inch thick, average $\frac{7}{32}$ inch. The strongest part is one-third of the thickness on the flesh side.

	Breaking Strain.	*Safe Working Strain.*
Through solid part	675 lbs.	225 lbs. per inch wide.
Through riveting	382 lbs.	127 ,, ,,
Through lacing	210 lbs.	70 ,, ,,

The working strength of the belt must be taken as that of its weakest part, which is the lacing. Lacing should be done from the centre towards the edges, and the laces should not be crossed on the side next the pulley.

The lower side of a belt should be made the driving side when possible, so that the arc of contact may be increased by the sagging of the following side. In some good shops in America the hair side is run next the pulley.

To increase the capability for transmission of power, the diameters of the pulleys may be increased, retaining the same ratio, the increase of power being obtained by the increased velocity alone.

The velocity ratio of pulleys driven by belts is not exactly proportional to the diameters, on account of slip, which ranges from 1 to 2 per cent.

Wide belts are less effective per unit of sectional area than narrow belts. Where a belt would exceed 18 inches wide it is better to use two belts. Long belts are more effective than short belts. All belts should hang slack when not in use. The sag of a belt when running should at least equal its width.

Vertical belts are not so effective as horizontal. The maximum angle with the horizontal should not exceed 45° if possible for main belts, and the distance centre to centre of pulley shafts in feet need not exceed eight times $\sqrt{\text{width belt inches}}$.

When the arc of contact = 180°, the force able to be transmitted may be taken as 50 lbs. per inch wide for single belt and 70 lbs. for double belt. If more or less than half circumference be embraced by belt, the force transmitted may be increased or reduced by about 2·8 lbs. for every 10° difference from 180°.

The sum of the tensions, or cross strain on shafting, may be taken as 90 lbs. per inch wide. In tightening, a belt may be shortened 1 inch for every 10 feet of its length.

The tension of the driving side, which must not exceed the safe working strength of the belt, = force transmitted + mean normal tension. The force transmitted = the difference between the tension of the driving side and the tension of the following side.

Convexity of pulleys to receive belt = $\frac{1}{8}$ to $\frac{1}{2}$ inch per foot wide, depending on the speed. Width of pulley = $\frac{1}{4}$ more than belt. Thickness of rim = 0·7 t of belt + ·005 d of pulley. The proportion between the diameters of two pulleys working together should not exceed 6 to 1. The revolutions per minute of two pulleys embraced by the same belt will be inversely proportional to their diameters.

The adhesion of a belt is lessened by the layer of air shut in between the belt and pulley while running, hence the anomaly of the greater adhesion of an open link belt from which the air can escape.

A smooth pulley gives the best adhesion, and a greasy belt has only three-eighths of the adhesion of a clean one. Resin makes a belt hard and brittle, petroleum rots it, neat's foot oil causes it to stretch.

The grease in a belt that has become saturated with oil may be removed by applications of fuller's earth, prepared chalk, or similar material, spread on $\frac{1}{8}$ inch thick when the belt is idle, and scraped off before putting it to work again; about three applications will generally be sufficient.

Ordinary shop shafting 100 revolutions per minute; belting say 1,000 to 2,500 feet per minute. The velocity of lathe belts should be from 25 to 50 feet per second = 1,500 to 3,000 feet per minute.

One driving pulley may be used to give motion to several belts placed one over the other, the driven pulleys being placed at varying distances away in the same line.

Pulleys from 2 to 3 feet diameter transmit approximately 1 H.P. per inch width of belt at ordinary velocities, say 700 ft. per minute; or sq. inches belt in contact with pulley × velocity feet per minute ÷ 72,000 = H.P.

Loss of power in shafting and belts averages 25 to 50 per cent. of total given out by engine (W. Geipel).

466. STRENGTH OF LEATHER BELTS.

H.P. = Effective H.P. transmitted.

v = velocity of belt in feet per minute.

w = width in inches of single belt.

$$\text{H.P.} = \frac{w\,v}{470}. \qquad w = \frac{470\ \text{H.P.}}{v}.$$

For double belts multiply H.P. × 1·5 or $w \times \frac{2}{3}$.—Bagshaw and Sons.

Another rule :—

R = revolutions per minute.

D = diameter pulley feet.

c = 25 for single belts.
17 for double belts.

d = diameter shaft inches, but if overhung, increase by $\frac{1}{4}\,d$.

$$\text{H.P.} = \frac{3\,w\,\text{D}\,\text{R}}{25\,c}. \qquad d = \sqrt[3]{\text{H.P.}}$$

Another rule (A. Towler) :—

d = diameter smaller pulley inches.

a = ratio of arc covered by belt to circumference.

$$\text{Single belts H.P.} = \frac{d\,w\,a\,\text{R}}{2000}.$$

$$\text{Double ,, H.P.} = \text{do.} \times 1{\cdot}75.$$

467. LARGE DOUBLE BELTS.

w = width of double belt in inches.
v = velocity feet per second.
l = length inches of arc of contact on lesser pulley.
H.P. = horse-power transmitted.

$$w = \frac{66000 \times \text{H.P.}}{l \times v}.$$

—EVAN LEIGH.

Double belts should not be used over pulleys less than 3 feet 6 inches diameter.

Leather link belting is the most suitable for transmitting great power and running at a high velocity.

Compound indiarubber and canvas belting should be used when exposed to the weather.

468. TO FIND LENGTH OF BELT EMBRACING PULLEYS.

R = radius of larger pulley to centre of belt.
r = ,, smaller ,, ,,
E = ,, equal pulleys.
d = distance between centres of pulleys.
n = number of degrees between radii from tangent points.
t = length of each of the tangent portions.
C = length of part embracing circumference of larger pulley.
c = length of part embracing circumference of smaller pulley.
L = total length exclusive of laps.

$$t = \sqrt{d^2 - (\text{R} - r)^2}.$$

n = tabular degrees corresponding to cosine having value of $\frac{\text{R} - r}{d}$.

$$\text{C} = \frac{360 - 2n}{360} \times 2\pi\text{R}.$$

$$c = \frac{2n}{360} \times 2\pi r.$$

$$\text{L} = 2t + \text{C} + c.$$

$$\text{E} = \frac{\text{L} - 2d}{2\pi}.$$

With a crossed belt on stepped speed cones R + r = constant, but with open belts this is not quite the case.

469. HALF-CROSS OR TWIST-BELTS.

For half-cross or twist-belts, where the shafts meet at right angles, the distance apart of the shafts should be at least four times the diameter of the pulleys and twenty times the width of belt if full power is to be transmitted. Then, with a velocity of 30 feet per second, a stress of 50 lbs. per inch width of belt may be transmitted. The pulleys should be flat faced, and the angle of faces should be plumb, not the centres.—JAS. LEE AND SONS.

470. AVERAGE WEIGHT OF LEATHER BELTING.

	Per 100 *ft. run.*
Single	11 to 13 lbs. per inch of width.
Light double . . .	17 ,, 22 ,, ,,
Heavy double . . .	20 ,, 24 ,, ,,

The wider belts are the heavier in each case.

471. LENGTH OF ROLLED BELT.

To ascertain the length of a roll of leather belting without unrolling it,

Let n = number of complete turns in roll.

D = outside diameter of roll in inches.

d = inside diameter of roll in inches.

π = ratio of circumference to diameter = 3·1416.

L = length of belt in feet when unrolled.

Then

$$L = \frac{\pi n}{24}(D + d).$$

When the last turn is not complete n will consist of a whole number for the full turns and a fraction for the partial turn.

472. AVERAGE STRENGTH OF LEATHER BELT JOINTS.

The percentage strength as compared with the average strength of leather belts, taken as 4,132 lbs. per sq. inch. From tests made by A. H. Barendt at the Liverpool School of Science.

Kind of Joint.	Average Strength in lbs. per sq. inch.	Per cent. Strength.
Cemented (only)	2,254	52·12
Ordinary white thong, without rivets . .	2,404	59·4
Waxed thread, without rivets . . .	3,240	78·4
Wire sewn or clinched, without rivets . .	3,272	79·18
White thong, with two rivets . . .	2,262	54·74
Waxed thread, with two rivets . . .	2,802	67·83
Copper wire sewn, with two rivets . .	3,050	75·81
White thong, with one rivet . . .	3,226	78·06

—PROF. JAMIESON.

473. VARIETIES OF LEATHER BELTING.

Link chain belting, made by Messrs. Tullis and Son, of Glasgow, is composed of a series of short leather links bound together by steel pins and washers. These belts possess considerable flexibility, are easily shortened and jointed, and have considerable adhesion, as no air cushion is caused between the belt and pulley face.

Vee leather belts are now largely used in grooved pulleys with electric motors.

Laminated belting made by James Hendry, of Glasgow, consists of strips of leather on edge, laid together, spliced at intervals to break joint with other strips and sewn from side to side. The equivalents to ordinary belting are: No. 1 equal to heavy single, No. 2 double, No. 3 heavy double, No. 4 3-ply. It is practically a belt of uniform section throughout, of any width or thickness, and without projections as in ordinary lapped joints.

474. NOTES ON HEMP ROPES.

Ropes are usually measured by their circumference; hence a 6-inch rope is one 6 inches in circumference, or about 1⅞ inch diameter.

A "plain-laid" rope consists of three twisted strands twisted together. A "hawser-laid" rope is made by twisting three plain-laid ropes together, so that a section would show nine strands.

Round ropes are better than flat for all purposes. Flat ropes are made up of 4 or 8 round ropes.

Italian hemp ropes are stronger than Russian hemp.

New white ropes are stronger and more pliable than tarred ropes, but the latter retain their strength for a longer period, owing to the protection afforded against atmospheric influences. The quantity of tar found most suitable is about 15 per cent. of the weight of the rope.

Tarred ropes are stiffer than white by about one-sixth, and in cold weather somewhat more.

Ropes which have been some time in use are more flexible than new ones, the stiffness of ropes increases after a little rest.

Wet ropes, if small, are a little more flexible than dry; if large, a little less flexible. Ropes shorten and swell when wetted. A wet rope, or one saturated with grease, loses half its strength.

All ropes should be kept dry and free from lime.

There is considerable loss of strength from strain, and exposure after use, although a rope may appear perfectly sound.

Ultimate strength of new white ropes is about 6,000 lbs. per sq. inch sectional area, but good ropes may stand 10,000 lbs. per sq. inch.

Small ropes are slightly stronger, in proportion to their sectional area, than large ones.

Double rope slings are not twice the strength of single rope, owing to inequality of strain; but in a rope fall with sheaves in good order, each fold of the rope may be counted for the strength.

The work absorbed in bending a rope fall over a sheave varies with the quality of the rope, directly as the tension, as the diameter^2, and inversely as diameter of sheave, and is irrespective of velocity.

Include weight of running block in calculating load on fall, and both blocks together with the rope, in weight on strop. Snatch block makes practically no difference in lifting power, if it has a good lead.

In rope tackle it is usual to allow for the friction in bending round sheaves, &c. = $\frac{1}{3}$ of the load to be lifted.

When blocks are hung from sheave legs or derrick poles the stress is that due to the whole load and not merely to the pull on the rope.

475. COULOMB'S FORMULA FOR STIFFNESS OF HEMP ROPES.

D = diameter of pulley in inches.

d = „ rope „

P = weight applied in lbs.

W = weight absorbed by stiffness for each 180 degrees of bending.

$$W = d^{2} \left(\frac{140 + 3\,P}{10\,D}\right) \text{ for new rope.}$$

$$W = d^{\frac{3}{2}} \left(\frac{140 + 3\,P}{10\,D}\right) \text{ for old rope.}$$

476. STRENGTH OF MANILA ROPES.

Manila rope varies from 10,000 lbs. per sq. inch net section ultimate strength for a 2-inch diameter rope to 12,000 lbs. per sq. inch for a ½-inch diameter rope.

Net sectional area = 0·81 of area of circumscribing circle.

d = diameter in inches of circumscribing circle.

S = breaking weight in lbs.

$$S = 100\,d^{2}\,(83 - 10\,d).$$

—Prof. J. J. Flather.

477. FORMULÆ FOR STRENGTH OF HEMP ROPES.

Breaking weight new rope, cwts.	=	circumference ^{2} × 4 to 5
Safe load on „ „	=	wt. lbs. per fath. × 3.
B.W. new stretched rope „	=	(diameter in ⅛ths)^2.
Safe load „ „	=	wt. lbs. per fath. × 4.
„ on new rope fall „	=	circumference ^{2}.
„ good „ „	=	⅔ „
„ sound old „ „	=	½ „
Weight of clean dry rope per fathom, in lbs. . .	=	¼ „
Do. do. tarred rope . .	=	⅓ „
Minimum diameter of sheave in inches	=	circf. rope + 2 in.

Flat ropes, width about 4 times thickness.

„ wt. lbs. per fath. approx. = circf. × 2.

„ B.W. tons = wt. lbs. per fathom.

478. HIDE ROPES.

Made by G. Pitts and Sons, Kirkdale, Liverpool, for hand-power delivery cranes, at 1s. 10d. per lb.

Dipped in Stockholm tar to prevent destruction by rats.

$$\frac{\text{Circumference}^2}{5} = \text{weight lbs. per fathom.}$$

479. FLY ROPES.

When power is transmitted over considerable distances by an endless rope running at a high velocity, the rope is called a fly rope. Used in engineering shops for driving travelling cranes, carrying heavy pieces of machinery. A three-ply manila rope, or cotton rope, with beeswax well rubbed in together with a little blacklead, is best. Run 3,000 to 5,000 feet per minute in cast-iron pulleys with V-grooves, angle 30° to 45°, latter for dry rope, former if lubricated. Working strain transmitted about 50 lbs. per circular inch area. Rope tightened by jockey pulley giving 250 to 300 lbs. per circular inch stress. Total stress must not exceed one-twentieth ultimate strength. Supported every 10 or 12 feet by flat plates of chilled cast iron. Friction of pulleys is inversely as their diameter, they should not be less than 30 times diameter of rope, nor have a higher peripheral speed than one mile per minute. By experiment, a new rope ½ inch diameter stretched 1 inch per foot per cwt.

Breaking weight in lbs. averages 720 × circumference², but ropes above 1 inch diameter are comparatively weaker, and below that size stronger. Mechanical efficiency of fly ropes = 0·6.

480. TESTS OF ROPES.

—	*Ultimate Tension.*	*Elongation.*
	tons per sq. in.	*per cent.*
White hemp	4·75	18
Tarred hemp	3·5	16
White manila	4·5	15
White aloes	2·5	..
Esparto and cocoa fibre	1·0	..
Flat ropes, hemp or manila tarred	3·5	5

Round ropes, with moderate attention, may be worked at a stress equal to one-third breaking stress, and flat ropes at one-fourth.

Portable machines for testing ropes are made by W. & T. Avery, Ltd., Birmingham, by which long ropes may be strained at any part of their length without severance, or short specimens may be tested separately.

481. ROPE DRIVING.

Introduced by James Combe, of Belfast (1856). Loss of power in transmission say 30 per cent., about same as gearing. Three-strand ropes best, manila for large, cotton for small. Speed 3,300 to 3,500 feet per minute. Long splicing absolutely necessary. Diameter of pulley in feet = diameter of rope in $\frac{1}{4}$ inches — 2, but they may be used up to 40 times diameter of rope. Angle of groove 45°. At 100 revolutions per minute approximate I.H.P. transmitted = 4 (d rope $\frac{1}{4}$ in. — 4).

a = sectional area of rope in sq. inches.

s = speed in feet per minute.

n = number of ropes.

H.P. = effective horse-power transmitted.

c = constant = hemp 100.

$$\text{H.P.} = \frac{c\,a\,n\,s}{33{,}000}; \qquad a = \frac{33{,}000 \text{ H.P.}}{c\,n\,s}.$$

—J. Bagshaw and Sons, Batley.

Leather rope, 8 narrow strips secured together and properly jointed, forming $1\frac{1}{2}$ inch square, running in V- groove, angle 90°, weighs 1 lb. per foot run, and will transmit 320 lbs. per sq. inch of section.

Cotton rope, $1\frac{3}{4}$ inch circumference, weighing 1 lb. per foot run, will transmit 50 I.H.P. at velocity of 5,000 feet per minute (600 lbs. less 60 for tension = 540 lbs. working strain). At a speed of 2,500 feet per minute the ropes will last longer and transmit H.P. = $c^2 \times 12$.

Steel rope, $\frac{1}{2}$ inch diameter, weighing $\frac{1}{3}$ lb. per foot run, will bear working stress of 405 lbs.

Hemp ropes, although stronger than cotton, do not stand so well. Approximate H.P. of hemp rope = circumference in inches × diameter of driving pulley in feet × revolutions per minute ÷ 200. Another rule: H.P. = circumference² × velocity in feet per minute × one less than number of ropes ÷ 5,000.

482. ADVANTAGES OF ROPE DRIVING.

The advantages of rope driving are :—

1. Any amount of power may be transmitted from a prime mover without noise or slip by increasing the number of ropes.

2. The power may be run any distance and in any direction, as in cable tramways.

3. The economy in first cost and maintenance is superior to many other methods.

4. With large smooth pulleys, the rope protected from damp and dust, and the strain moderate, the ropes will have a longer life.

5. Rope drives may be substituted for heavy leather belts and trained to any angle.

483. PULLEYS FOR ROPE DRIVING.

Manila rope	1¼	inch diameter	on	3-foot	pulley	will	transmit	5	I.H.P.
,,	1½	,,	,,	4	,,	,,	,,	8	,,
,,	1¾	,,	,,	5	,,	,,	,,	11	,,
,,	2	,,	,,	6	,,	,,	,,	15	,,

—James Combe.

By another rule the pulleys should not be less in diameter than circumference of rope² × 3.

For rope driving the sides of groove make an angle of 10° with each other down to the semi-diameter, and 45° below, the bottom of groove being curved with a radius of ⅜ diameter of rope, with a clearance of ¼ diameter.

484. AVERAGE TENSILE STRENGTH OF ROPES.

Specimens 13 feet long, ends wound on grooved pulleys.

	lbs. per sq. inch.
White hemp	10,500 to 11,200
Tarred hemp	7,700 ,, 8,400
White manila	9,800 ,, 10,600
White aloes	5,600 ,, 7,000
Flat, tarred hemp, or manila	7,800 ,, 8,400
Unannealed wire rope	55,000
(elongation 6 to 8 per cent.)	
Annealed wire rope	45,000
(elongation 12 to 15 per cent.)	

Factor of safety 3 to 4. —A. Duboul.

Wire Ropes for Lifting, Hoisting and Hauling.

Diameter of pulley in inches = circumference of rope (Lang's lay) in sixteenths of an inch.

Safe working load including weight of rope = $\frac{1}{10}$ breaking weight for high speed lifts, $\frac{1}{8}$ for ordinary lifts, and $\frac{1}{6}$ for hauling on incline.

$1\frac{3}{4}$ inch steel ropes are now much used for suspending lifts, used four together, safe working load on each is considered to be 15 cwts.

Gross tractive force in lbs. required on a gradient of 1 in x to haul a total load W lbs. including weight of rope $= \frac{W}{x} + \frac{W}{100}$.

485. CALCULATION OF LOAD ON WIRE ROPE FOR HAULING.

Example :—Plane 800 yards. Load 20 tons = 400 cwts. Maximum inclination 7° or 1 in 8·14.

Gravity of load	$400 \div 8{\cdot}14$	=	49·140
Friction of load	$400 \div 100$	=	4·000
Gravity of rope,	$\frac{800 \times 2 \text{ lbs.}}{8{\cdot}14}$	=	1·744
Friction of rope	$\frac{800 \times 2 \text{ lbs.}}{112 \times 100}$	=	0·143
Net hauling power required in cwts.			55·027

—Dixon and Corbitt *and* R. S. Newall and Co., Ltd.

486. USE OF WIRE ROPES.

Wire ropes should be uncoiled on a reel or turntable to avoid kinks.

Those for hauling and winding should be kept lubricated with a heavy-bodied hydro-carbon oil well rubbed in. An unlubricated rope failed after 16,000 bends, whilst the same rope lubricated stood 38,700 bends before fracture.

Round wire rope may be had in all qualities, from "charcoal iron" at 35 tons per square inch tensile strain, up to "extra plough steel" at 130 tons.

For vertical winding the working load should not be more than $\frac{1}{8}$ to $\frac{1}{9}$ of breaking strain, but for hauling it may be increased to $\frac{1}{6}$.

487. EXPERIMENTS ON WIRE ROPE AT FORTH BRIDGE.

Crucible cast steel wire rope was used. With a diameter of sheave = 6 times *circumference* of rope, rope bent over sheave 5,000 times before failure commenced, 15,000 before final destruction. With a diameter = 8 times circumference, 10,000 times and 36,000 respectively.

488. LANG'S PATENT WIRE ROPES.

	Bessemer Steel.	*Crucible Steel.*	*Patent Steel.*	*Plough Steel.*
Strength of material in tons per sq. in.	45	56	75	111
Round rope, 6 strands of 6 wires each, up to	4 in. circf.	3 in.	$3\frac{1}{2}$ in.	4 in.
	9 wires in each strand above these sizes.			
Approx. B.W. in tons . . . =	$c^2 \times 1{\cdot}5$	2	2·5	3·5

Working load . . . . . = $\frac{1}{10}$ breaking weight.

Weight of round wire ropes in lbs. per fathom = circf.^{2} × $\frac{7}{8}$.

—J. Bagshaw and Sons, Batley.

In "Lang's lay" the strands and rope itself are all with right-hand twist and the wires take a diagonal position. In "ordinary lay" the rope has a right-hand twist, but the strands have a left-hand twist so that the wires take a longitudinal position. Lang's lay is better for hauling or winding, but is not suitable when a load is to be lifted from a free end.

489. HOUGHTON'S FLEXIBLE STEEL WIRE ROPES.

For cranes, cargo and purchase falls, elevators, well-boring, etc.

Description.	*Quality.*	*Wt. per fath. lbs.* = c^2 ×	*B.W. tons.* = c^2 ×	*Min. dia. barrel* = c ×
Improved patent steel wire, galvanised	A. Flexible	·678	2·1	6
Do. do. . .	B. Sp. flex.	·9	2·79	5·4
Do. do. . .	C. Ext. sp. flex.	·83	2·58	4·5
Improved plough steel wire, plain	D. Flexible	·96	4·42	7·15
Do. do. . .	E. Sp. flex.	·96	4·42	5·85
Do. do. . .	F. Ext. sp. flex.	·96	4·42	4·87

—W. D. Houghton, Warrington.

490. AMERICAN STEEL WIRE ROPES

are made of 7, 12, and 19 wires.

D = proper diameter of sheave in inches.

d = diameter of wire rope in sixteenths of an inch.

c = constant = 4·8 for 7-wire rope.
= 3·5 ,, 12 ,, ,,
= 3 ,, 19 ,, ,,

$$D = c \cdot d \text{ for steel.}$$
$$= 2cd \text{ ,, iron.}$$

—W. Hewitt, New Jersey.

Two kinds of wire rope are manufactured. The most pliable variety contains 19 wires in the strand, and is generally used for hoisting and running rope.

For safe working load, allow $\frac{1}{5}$ or $\frac{1}{7}$ of the ultimate strength, according to speed, so as to get good wear from the rope. Wire rope is as pliable as new hemp rope of the same strength; but the greater the diameter of the sheaves the longer wire rope will last.

Experience has proved that the wear increases with the speed. It is, therefore, better to increase the load than the speed. Wire rope must not be coiled or uncoiled like hemp or manila, all untwisting or kinking must be avoided.

In no case should galvanised rope be used for running. One day's use scrapes off the zinc coating.

491. R. S. NEWALL & CO.'S IRON WIRE ROPES.

Round—

Weight in lbs. per fathom = $C^2 \times \frac{7}{8}$.

B.W. tons = weight in lbs. per fathom × 2.

Safe load cwts. = ,, ,, × 6.

Flat— Width = $4\frac{1}{2}$ to $5\frac{1}{2}$ times thickness.

Sectional area × 10 = weight in lbs. per fathom.

Weight in lbs. per fath. × $\frac{9}{5}$ = B.W. tons.

B.W. tons × $\frac{20}{9}$ = safe working load cwts.

Drum for wire rope = 2 feet 6 inches diameter for every $\frac{1}{8}$ inch diameter of rope, speed 30 to 50 miles per hour. For slow speeds drum 80 times diameter of rope.

492. CURVE OF ROPE.

A rope or chain when deflected by its own weight hangs in a catenary curve. It approximates to a parabola, and is indistinguishable from one when the deflection is not more than one-tenth of the span. The deflection increases with the weight of rope and decreases as the tension is increased.

493. TELEDYNAMIC TRANSMISSION OF POWER.

In transmitting power by means of wire rope the wires are subjected to two tensions, the working tension and the tension due to bending about the pulleys. The sum of these tensions should not exceed the limit of elasticity of the wires. The proper ratio between the diameter of rope and pulleys is that which will permit the maximum working tension to be obtained without overstraining the wires in bending. For rope of 7-wire strands the ratio is about 1 : 150, for 12-strands 1 : 115, and for 19-wire strands 1 : 90.

The horse-power transmitted approximately equals $3\frac{1}{10}$ times the square of the diameter in inches multiplied by the velocity in feet per second.

d = diameter of rope in inches.

v = velocity in feet per second.

W = weight of rope, intermediate and terminal pulleys and axles in lbs. (i.e., the whole moving weight).

$$\text{H.P. required} = 4{\cdot}75\, d^2. - {\cdot}000{,}006\, W^2.$$

The proper deflection to give the rope to secure the necessary tension may be determined by the formula

$$h = {\cdot}0000695\, s^2$$

where h = deflection of rope at rest, s = span, both in feet.

When the deflection at rest is less than 3 inches the transmission cannot be effected with satisfaction; this corresponds to a span of about 60 feet. It is customary to make the underside of the rope the driving side, as the greater deflection of the driving side occurs when the rope is at rest. The maximum limit of span is determined by the maximum deflection that may be given to the upper side of the rope when in motion. Assuming that the clearance between the upper and lower sides should not be less than 2 feet and that the pulleys are at least 10 feet in diameter, this corresponds to a span of about 340 feet on level ground. Much greater spans than this, however, are practicable in cases where the upper side of the rope may be made the driver, as in crossing valleys, where transmission up to 1,700 feet in a single span has been effected.

Splices are easiest made in six-strand ropes, they should be from 20 to 50 feet long, according to the size and working conditions.—Dixon and Corbitt *and* R. S. Newall and Co.

494. STRENGTH OF CHAINS.

d = Diameter of iron in $\frac{1}{8}$ths of an inch. Or diameter of smallest part, if worn.			*Example $\frac{3}{4}$ Chain.* tons	cwts.
B.W. in tons, B.B. short-link crane chain . .	=	$\frac{1}{2}d^2$	18	0
,, ,, ordinary chain	=	$\frac{2}{5}d^2$	14	8
,, ,, short-link (Board of Trade)* . .	=	$\frac{3}{8}d^2$	13	10
Elswick test in tons, 10 per cent. above Admiralty proof	=	$\frac{33}{160}d^2$	7	8½
Admiralty and Board of Trade proof strain in tons	=	$\frac{3}{16}d^2$	6	15
Safe load in tons (Molesworth, 11th ed.) . .	=	$\frac{1}{8}d^2$	4	10
,, ,, at 5 tons per sq. inch sectional area	=	..	4	8½
,, ,, in tons (Molesworth, 21st ed.) . .	=	$\frac{1}{9}d^2$	4	0
,, ,, in tons, common rule	=	$\frac{1}{10}d^2$	3	12
Maximum temporary load on good annealed chain in cwts.	=	$2d^2$	3	12
Safe load, ordinary chain (Anderson), in tons .	=	$\frac{3}{32}d^2$	3	7½
,, ,, for ordinary cranes, in cwts. . .	=	$1\frac{1}{2}d^2$	2	14
,, ,, at 3 tons per sq. inch sectional area .	=	..	2	13
,, ,, coal cranes, in cwts.	=	$1\frac{1}{4}d^2$	2	5
,, ,, old chain, quality and condition unknown in cwts.	=	d^2	1	16
Weight in lbs. per fathom, short-link crane chain .	=	d^2	36	
,, ,, ,, ordinary ,, .	=	$\cdot 88d^2$	31½	

Safe load (5 tons cranes and upwards) in tons = $\frac{1}{8}d^2$ when made of good iron, but large chains are frequently of common quality.

Size of links for crane chains . . . = $3\frac{1}{3}d \times 4\frac{2}{3}d$.

Admiralty proof strain on rings, in tons . . = d in $\frac{1}{8}$ths^2 ÷ 16.

,, ,, stud chain ,, . . = d in $\frac{1}{8}$ths^2 × ·281

Common chain cables B.W. lbs. . . . = 1,000,000 $(\frac{1}{4}d)^2$.

Minimum diameter of chain shears is 20 times diameter of chain iron, but 30 times is desirable.

* This is known as the "Statute breaking stress."

495. REMARKS ON CRANE CHAINS.*

$\frac{9}{16}$ inch B.B. tested short link crane chain (Crown S.C.) should break with a load of 13 tons, if the iron bar from which it is made break with 26 tons per sq. inch ultimate stress; but a test-piece of the chain 4 feet long breaks usually with a load of 9 to 10 tons, generally opening at the welds. Each chain is tested before use with a maximum load of 4½ tons, examined link by link, and used on hydraulic coal cranes to lift maximum gross load of 1¼ tons, examined again at frequent intervals and annealed; any links reduced by wear to ½ an inch at ends are condemned as worn out; worn links cut out and remainder used down to same limit. A good chain, properly looked after, will make from 100,000 to 150,000 lifts before it is entirely worn out. These chains occasionally fail in use, although the factor of safety adopted allows so great a margin.

496. EXAMINATION OF CHAINS AT THE DOCKS IN LONDON.

All chains are taken down, annealed and examined as follows, viz.:—

Hydraulic crane, lead, lift, etc., chains, every six months.

Hand and steam crane, traveller, dockgate and chain gear, every twelve months.

The chain gear comprises chain runners, chain necklaces, sweeping and guy chains, chain slings, cattle slings, shackles, dogs and lead hooks.

497. CIRCULAR RINGS FOR MOORING AND SLING CHAINS.

Circular rings in connection with mooring chains are made of a diameter proportionate to the size of chain, fixed by each maker, but generally four to six times the diameter of the iron. The Admiralty test for rings depends upon the diameter of the iron alone, and is independent of the diameter of the ring. It is

$$\text{Test load cwts.} = 1\tfrac{1}{4}\ (d \text{ in } \tfrac{1}{8}\text{ths})^2.$$

To find proper diameter of circular ring in mooring and sling chains:

d = diameter of iron of chain in inches suitable for lifting given load.

D = diameter of iron of ring in inches.

R = mean radius of ring in inches.

$$D = \sqrt[3]{R\,d^2}.$$

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* See paper on "Use and Care of Chains for Lifting and Hauling," read by the author before the Civil and Mechanical Engineers' Society. 2nd edition (Spon, 1s.)

498. TOOTHED GEARING.

A *spur wheel* has the teeth projecting radially on the circumference.

A *pinion* is the name given to the smaller of two wheels working in gear together. Hence spur wheel and spur pinion, bevel wheel and bevel pinion.

A *bevel wheel* has the teeth projecting on a rim which is inclined to the plane of the circumference at an angle usually between 30° and 60°.

Mitre wheels are bevel wheels of equal size, geared together at an angle of 90°.

A *crown wheel* has the teeth projecting at right angles to the plane of the circumference.

A *lantern wheel* has round pins to act as teeth, fixed between two discs, near the circumference.

A *hunting-cog* is an additional tooth on a wheel making the teeth of the wheel and pinion *prime* to each other and equalising the wear. Numbers prime to each other are those which have no divisor in common.

A *mortice wheel*, or shell wheel, is a cast-iron wheel from ordinary patterns, but with hard wood teeth secured in mortices cast in the rim.

A *rag wheel* is a wheel with strong projections upon it which enter the spaces of a special chain called a pitched chain, or link chain, for transmitting power.

An *intermediate wheel*, or idle wheel, on a screw-cutting lathe is used to connect two wheels on different spindles without altering their velocity ratio.

A *Marlborough wheel* is one of double breadth, gearing at the same side into two wheels on different shafts, whose axes are so nearly in the same line as to prevent the use of ordinary spur gear. In effect it is the same as an intermediate wheel.

An *epicyclic train* consists of two or more wheels gearing into one another, the axis of one of them being fixed in space and the others rotating round it; or the wheels are attached to a rotating frame or bar in such a manner that they can derive motion from the rotation of the frame or bar. The mechanism obtains the name of epicyclic because the curves described by points in the moving wheels are epicycloidal.

A *Geneva stop* consists of a disc provided with one tooth, and another disc with five or more spaces, all the parts between the spaces (except one) being hollowed to fit the first disc, so that at each revolution of the first disc the tooth carries the second one through a portion of a revolution, until further rotation is prevented by the part which is not hollowed out coming into contact with the shoulder at side of single tooth. It is a device to prevent overwinding.

A *fusee* is a small conical drum upon which a chain is wound to equalise the effect of a coiled spring, by giving a varying leverage, as in a watch, clock, or other mechanism.

A *snail drum* is a form of fusee used as a counterweight drum for lifting loads which become less as they rise—e.g., a bascule bridge.

Backlash is the noise made by toothed gearing when working, and is owing to the small slip that occurs when the load passes from one pair of teeth to the next pair.

Chamber Gears.—This term has been applied to such mechanisms as Roots's blowers and various forms of rotary pumps, where the contact is constant between the wheels and their casings, so that fluid cannot pass through in a direct course, but must occupy the spaces swept through by the rotating wings or teeth.

499. NOTES ON TOOTHED GEARING.

Pinions, wheels, and racks are made of cast iron, cast steel, and malleable cast iron ; the latter is strong, but liable to twist or warp. Pinions are sometimes made of wrought iron ; small gearing is frequently made of gun-metal. Raw hide (Angus and Co.) has also been used for noiseless spur and bevel gearing.

If moulded from patterns, wheels should be geared so that the taper ends of teeth are on opposite sides. Gearing is increased in strength about one-third by shrouding or flanging up to pitch line.

The pitch line or pitch circle is the mean circumference of the teeth, or the circumference of a plain wheel without teeth, which would produce the same velocity ratio of slipping were prevented. The teeth are only to prevent slipping.

The pitch of the teeth is the distance from a point on one tooth to a similar point on an adjacent tooth measured along the arc of the pitch circle.

The comparative wear of gearing is inversely proportional to the number of teeth ; hence, pinions wear quicker than wheels.

In toothed gearing there should never be fewer than two pairs of consecutive teeth in action at any time, and the maximum obliquity of action should never exceed 30°. Therefore the least number of teeth (or pins) in a lantern wheel will be 6, in ordinary gear 9, and in cycloidal gear 12.

The *power* capable of being transmitted by gearing depends, within reasonable limits, entirely upon the *speed;* the possible *pressure* (at pitch line) depends upon the *pitch.*

The speed should not exceed 1,800 feet per minute circumferential velocity for ordinary cast-iron wheels, or 2,400 for mortise wheels.

The velocities of geared wheels are in the inverse ratio of their diameters.

The transmission of the power strains the teeth as cantilevers, or $s = c\,\frac{b\,d^2}{l}$, c for cast iron safe load = 600, for cast steel = 2,000.

The working load should not exceed $\frac{1}{10}$ of the breaking weight.

The dimensions of the teeth are proportional to the pitch; hence, in ordinary proportions the strength is represented by p^2c, c for cast iron being 1,000 as a maximum.

The breadth of tooth on face beyond a certain amount, say twice the pitch, cannot be reckoned upon for strength, owing to irregularities in the teeth, and probability of unequal bearing, but the extra breadth is useful up to $2\frac{1}{2}$ times the pitch.

500. WHEEL GEARING, CIRCULAR PITCH.

P = pitch in inches = (tooth + space) on pitch circle.
D = diameter of pitch circle in inches.
N = number of teeth in wheel.
π = constant = 3·1416.

$$P \times N = D \times \pi$$

$$N = \frac{\pi}{P} D, \quad D = \frac{P}{\pi} N, \quad \frac{D}{N} = \frac{P}{\pi}.$$

P has finite values, generally $\frac{1}{4}$ inch to 2 inches, advancing by $\frac{1}{8}$ inch, 2 inches, to $3\frac{1}{2}$ inches advancing by $\frac{1}{4}$ inch, etc. N must always be a whole number.

$$\text{Approx. diam. pitch circle} = \frac{\text{over-all diam.} \times \text{no. teeth}}{\text{no. teeth} + 2}.$$

501. STRENGTH AND WEIGHT OF TOOTHED GEARING.

Safe pressure in lbs. at pitch line on wheel teeth of average proportions :—

Cast iron, little shock	. . . .	= 625 × pitch².
„ moderate shock	. .	= 400 × pitch².
„ excessive shock	. .	= 277 × pitch².

The latter case also applies to the iron teeth of mortise wheels, which are made thinner than ordinary teeth of same pitch.

J. B. Francis' rule for pitch $= \cdot 044 \sqrt{\text{lbs. pressure}}$.

Breadth of teeth = 2 to $2\frac{1}{2}$ times pitch.

The weight of toothed gearing in lbs. approximately, is for spur wheels $\cdot 38\ n\,b\,p^2$, bevel wheels $\cdot 325\ n\,b\,p^2$, where n is number of teeth, b breadth on face, and p pitch.

502. FORMULÆ FOR STRENGTH OF GEARING.

s = strain in lbs. to be transmitted, calculated at pitch circle.

p = pitch in inches.

c = constant, when teeth of ordinary proportion =

Material	*Plain.*	*Shrouded.*
Cast steel	4000	6000
Wrought iron	3000	4500
Malleable cast iron	2000	3000
Gun metal	1500	2000
Cast iron	1000	1500

$$s = p^2 c. \qquad p = \sqrt{\frac{s}{c}}$$

For slow speeds and uniform pressure c may be increased one-fourth.

503. WHEEL GEARING, MANCHESTER PITCH.

Diametrical pitch (Manchester pitch) usually chiefly for small gearing = number of teeth in wheel to each inch of pitch diameter.

$$\text{Diametrical pitch} = \frac{\text{No. of teeth}}{\text{diameter of pitch circle in inches}}.$$

$$\text{Circular pitch} = \frac{\pi}{\text{diametrical pitch}}.$$

No. of teeth in wheel = pitch diameter × diametrical pitch.

$$\text{Diameter of wheel} = \frac{\text{No. of teeth}}{\text{diametrical pitch}}.$$

Addition to diameter for increased No. of teeth

$$= \frac{\text{No. to be added}}{\text{diametrical pitch}}.$$

Outside diameter of wheel

$$= \frac{2}{\text{diametrical pitch}} + \text{diameter pitch circle.}$$

For example :—A 10-pitch wheel (Manchester or diametrical pitch) 7·5 inches diameter will have $10 \times 7 \cdot 5 = 75$ teeth; another in the same set 4 inches in diameter would have $10 \times 4 = 40$ teeth, and their true pitch would be

$$\frac{3 \cdot 1416 \times 7 \cdot 5}{75} = \frac{3 \cdot 1416 \times 4}{40} = 0 \cdot 31416 \text{ inches,}$$

or generally, with n-pitch wheels, true pitch $\times \frac{\pi}{n}$ inches.

For Manchester pitch the number of teeth per inch diameter is taken as the standard, usually 3, 4, 5, 6, 7, 8, 9, 10, 12, 14, 16, and 20, and this leads to a fractional circular pitch, as in the example of 10-pitch wheels above.

As a rule it will be found that Manchester pitch is more convenient for wheels of less than 1 in. circular pitch, and the ordinary reckoning for wheels of greater pitch.

504. MILL GEARING.

H = H.P. actual.
b = breadth on face inches.
D = pitch diameter in feet.

p = pitch inches.
R = revolutions per minute.
n = No. of teeth.

$$H = \frac{p^2 b D R}{306}$$

$$D = p \operatorname{cosec} \frac{180^\circ}{n}$$

$$p = \sqrt{\frac{306 H}{b D R}}$$

$$p = \frac{D}{\operatorname{cosec} \frac{180^\circ}{n}}$$

Another formula:

$$\text{N.H.P.} = \sqrt{(D R)} \times p^2 b \times \begin{cases} \cdot 05 \text{ wood} \\ \cdot 043 \text{ cast iron} \\ \cdot 15 \text{ cast steel} \end{cases}$$

Gudgeons (*Tredgold*): (cast iron ?)

$$\text{Diameter inches} = \frac{\sqrt{w \text{ lbs.} \times l \text{ inches}}}{9}.$$

505. SPEED OF MILL GEARING.

Maximum safe speeds under favourable conditions for toothed gearing:

Ordinary cast-iron wheels	1800 ft. per min.
Helical „ „	2400 „
Mortice „ „	2400 „

Ordinary cast-steel wheels	2600 ft. per min.
Helical „ „	3000 „
Special cast-iron machine-cut wheels . . .	3000 „

—A. TOWLER.

Machine-moulded cast-iron wheels . . .	2000 ft. per min.
Machine-moulded cast-steel wheels . . .	2500 „

—W. H. THORNBERY.

506. DETERMINING THE PROPORTIONS OF GEARING.

In toothed gearing exact ratios should be sacrificed to obtain numbers prime to each other. When the wheels are to be equal, one of them should have an additional tooth called a "hunting-cog"; then each tooth of the one will encounter each tooth of the other, equally often, and equalise the wear.

Numbers are prime to each other when they have no common measure—i.e., cannot both be divided without remainder by any number except 1.

For wheels to gear properly the number of teeth in each must be proportionate to their diameters—in other words, their pitch must be equal.

To find the number of revolutions of a pair of toothed wheels before the same teeth come into gear again, let N = number of teeth in larger wheel, n = number of teeth in smaller wheel, R = revolutions of larger wheel, r = revolutions of smaller wheel; then divide the ratio $\frac{N}{n}$ by the highest common factor, or, in other words, bring it down to its lowest terms, and the result will be $\frac{r}{R}$. Example 1: two wheels of 35 and 49 teeth respectively, $\frac{N}{n} = \frac{49}{35}$, divide both numerator and denominator by 7, $\frac{49 \div 7}{35 \div 7} = \frac{7}{5} = \frac{r}{R}$, or, after every 7 revolutions of the smaller wheel and every 5 of the larger wheel the same teeth come into gear. Example 2: two wheels of 31 and 49 teeth respectively, $\frac{N}{n} = \frac{49}{31}$, which is in its lowest terms as the numbers are prime to each other, therefore $\frac{49}{31} = \frac{r}{R}$, or the same teeth will come into gear after every 49 revolutions of the smaller wheel and every 31 revolutions of the larger wheel. It makes no difference whether the wheels are driving or driven.

507. VARIETIES OF TOOTHED GEARING.

Gee's toothed gearing is sometimes used when wheels have to run in one direction only. The teeth are on the principle of the "buttress thread" as used in Armstrong breech-loading guns. The working faces are of the ordinary shape, but the backs are shaped to give a narrow point and thick root, and are said to be 35 per cent. stronger than ordinary gear teeth.

Hooke's stepped gearing is like an ordinary spur wheel split up into several thin wheels rigidly fixed side by side, but arranged so that the teeth in one are a short distance behind the teeth in the next, the distance being such that the teeth in the first wheel are at the same distance behind the teeth in the last wheel, as the teeth in each of the others is behind those of the preceding one. If n be the number of steps then $2n$ will be the number of teeth virtually in action at the same time so that backlash is reduced.

Helical gearing is upon the same principle as Hooke's, but carried to the extreme, so that the distance between the teeth of successive wheels is infinitely small, and the number of wheels infinitely great. This is obtained by using a single set of teeth in each wheel inclined at an angle of about 35° to the axis of rotation.

Double helical gearing, or vee teeth, are upon the same principle, but the half of each tooth being in the reverse direction from the other half, the tendency of the wheels to separate axially is counteracted. They are about 30 per cent. stronger than single helical teeth, but one of the wheels must be capable of a small lateral motion, so that the pressure may be equally distributed on each half.

In the *Wüst patent double helical gear*, the half teeth do not meet, but one side is set forward half pitch beyond the other side, and backlash is reduced to a minimum.

The *Hindley spiral gear* has concave faces on both wheel and pinion, and drives at right angles.

508. RAWHIDE NOISELESS GEAR WHEELS.

Best selected ox-hide is prepared in such a way as to retain the gelatinous substance and fibrous tissue of the green hide unimpaired. The circular blanks are stored for twelve months or longer to season them, they are then cemented together, and compressed between gun-metal discs, or shroud plates, connected by strongly countersunk rivets. Above 9 inch diameter cast-iron shroud plates are used with a bush cast on one of the plates. The pinion teeth are then machine-cut through the shroud plates and the leather. The plates are $\frac{1}{8}$ inch thick for very small pinions with narrow face, $\frac{3}{16}$ inch for all average sizes up to 5 inches diameter, $\frac{1}{4}$ inch to $\frac{5}{16}$ inch for pinions and wheels of a large diameter and having a wide face, $\frac{3}{8}$ inch to $\frac{1}{2}$ inch plates are used in exceptional cases. The pinions are made sufficiently wide for the shroud plates to stand beyond the teeth of the spur wheel to prevent noise.

Raw hide gearing is lubricated with a little black lead or French chalk when new, and occasionally after thorough cleansing a very small quantity of raw linseed oil may be put on the teeth with a brush. They should otherwise be kept free from oil and grease.

509. POWER TRANSMITTED BY RAWHIDE GEARING.

d = diameter of pitch circle in inches (for bevel wheels mean of pitch circles at both ends must be taken).

p = circular pitch in inches. Pinions 15 teeth minimum, 18 teeth preferable, 20 teeth if full power is to be transmitted.

b = working breadth of tooth on face in inches, which should not be less than 4 times the pitch, and more than this when velocity at pitch circle exceeds 1,200 feet per minute.

n = number of revolutions per minute.

H.P. = horse-power capable of being transmitted.

$$\text{H.P.} = \frac{d\,p\,b\,n}{1000}$$ —D. Brown and Sons.

510. EFFICIENCY OF WORM GEARING.

μ = coefficient of friction, say ·05.

p = axial pitch of worm.

r = pitch-radius of worm.

$$\text{Efficiency} = \frac{1 - \mu \dfrac{p}{2\,r\,\pi}}{1 + \mu \dfrac{2\,r\,\pi}{p}}$$

Best result obtained with triple-thread worm $4\frac{9}{16}$ inch pitch-diameter $1\frac{5}{8}$-inch pitch of teeth, $4\frac{7}{8}$-inch axial pitch, driving a 30-tooth wheel, giving a reduction of 10 to 1. Worm of steel and wheel of phosphor bronze, running in oil-bath, ball thrust-bearings, efficiency 86 per cent. Allowing 4 per cent. for losses in bearings, μ = 0·033 for screw-friction.—C. W. Hill.

511. GLOBOID WORM GEAR.

In "globoid" worm gearing the worm-thread is cut by means of a tool swinging on a centre at a radius equal to that of the wheel. The pitch-line of the worm, instead of being a straight line parallel to the worm-shaft, thus becomes an arc of a circle coinciding with the pitch-circle of the wheel. By this means a longer line of contact is obtained, the oil has freer play between the teeth, and the coefficient of friction is reduced.—C. W. HILL.

512. PROPORTIONS OF WHEEL TEETH.

	Parts.		*Per cent.*	*Other*	*Authorities.*	
Pitch	15	or	100	100	100	100
Whole length of tooth	12	,,	80	75	75	60
Pitch line to point	5½	,,	36·6	35	33	25
,, ,, root	6½	,,	43·3	40	42	35
Thickness at pitch line	7	,,	46·6	45	48½	48
Width of space at ditto	8	,,	53·3	55	51½	52

The right-hand column gives the later practice.

Curve	radius = pitch, or cycloidal.
Breadth of tooth on face	250 per cent.
Thickness of rim	44 to 50
Projecting ribs inside ditto	44 to 50
Thickness of arms	44 to 50
Breadth of arms at rim	175
,, of taper increasing to boss	½ inch per foot.
Thickness of rib on arms	25
,, metal in boss	75 to 80

Diameter of rolling circle for cycloidal teeth = pitch × 2·5.

513. ORDINARY PROPORTIONS OF KEYS.

$$\text{Width of key} = \begin{cases} \frac{1}{4} \text{ diam. of shaft up to 4 inches.} \\ \frac{1}{5} \quad ,, \quad ,, \quad \text{4 inches to 8 inches.} \\ \frac{1}{6} \quad ,, \quad ,, \quad 8 \quad ,, \quad 12 \quad ,, \end{cases}$$

Key square at thick end. Taper ⅛ inch per foot.

One-third of thickness let in shaft, remainder in wheel.

Another rule for size of sunk keys :—

Width of key $= \frac{1}{4}$ diam. of shaft $+ \frac{1}{8}$.
Middle thickness $= \frac{1}{8}$ diam. of shaft $+ \frac{1}{8}$.
For key on flat $\frac{3}{4}$ of above thickness.

514. PROPORTIONS OF COTTERS THROUGH BARS.

b = Breadth of cotter.
t = Thickness of cotter.
d = Diameter of bar.

Through round bars,

$$b = 1 \cdot 4635\, d. \qquad t = \frac{d}{5}.$$

Through square bars,

$$b = 1 \cdot 5 \text{ side of bar.} \qquad t = \frac{\text{side of bar}}{4}.$$

515. JOURNALS FOR SHAFTS AND AXLES.

Length of brass = 0·9 to 1·0 length of journal. Less liable to score in wearing, if slight end play can be given.

Thickness and projection of collar and radius of curves

$$= \frac{d}{8} + \frac{1}{8} \text{ in.} \quad \text{to} \quad \frac{d}{10} + \frac{1}{8} \text{ in.}$$

Vertical shafts should be shaped to a hollow-sided cone or "antifriction" pivot at the foot; the sides being shaped to the "curve of pursuit" or "Witch of Agnesi," which see.

Thrust shafts should have collars bearing against a series of loose horse-shoe plates between rigid abutments, to divide the pressure over several bearing surfaces. Roller end-bearings may also be adopted to reduce the friction.

516. POWER OF CRANEMAN, &c.

Radius of handle	1 ft. 3 in. to 1 ft. 6 in.
Height to centre of axle . . .	2 „ 6 „ 3 „
Height from ground to path of handle .	1 „ 6 „ 1 „ 9 „
Revolutions of handle per minute	28 to 23

Speed at circumference of handle for continuous work while lifting . .	220 feet per minute
Do. do., when lifting and lowering .	330 ,, ,,
Force of ordinary labourer on handle .	12 lbs. + friction
,, ,, craneman ,, . .	15 ,, ,,
Maximum ditto, for short time, say 5 minutes, at 440 feet per minute . .	30 ,, ,,

At 8 hours per day, on long lifts, the effective work averages 2380 to 2420 foot-lbs. per minute per man.

One man can raise 1 ton with a multiplying power of 150, the friction being about $6\frac{1}{2}$ lbs., and the effective pressure 15 lbs., making the gross pressure on the handle $21\frac{1}{2}$ lbs., or coefficient = ·7.

Speed of lifting with hand-power crane = 2 feet per second.

In raising weights with a pulley a man can maintain a downward pull of 40 lbs. permanently, and equal to his own weight temporarily.

517. HAND POWER CRANE.

W = load in lbs.
P = power required in lbs. to overcome load.
F = friction of gearing of crane without load.
f = friction of gearing due to load.
M = multiplying power of gearing.
E = efficiency of crane under various loads.

$$P = F + f + \frac{W}{M}.$$

$$E = \frac{W}{MP} = \cdot 5 \text{ to } \cdot 75.$$

By experiment with 10 cwts. crane—

$$M = 40, \quad F = 4 \cdot 21 \text{ lbs.}, \quad f = \cdot 0179\, W.$$

—R. S. Ball.

1 ton crane, 4 men at handles, 25 lbs. each man, multiplying power 24 to 1.

Delivery cranes, short lift, lowering by brake, allow 25 lbs. for each man, handle 16-inch radius, 30 revolutions per minute, coefficient ·75.

Landing cranes, long lift, allow 15 lbs. for each man.

518. CRAB WINCHES.

R = radius of handle.
r = radius of barrel to centre of rope.
A = radius, diameter, or number of teeth in pinion.
B = ,, ,, ,, ,, wheel.
W = load lifted in lbs.
P = power applied in lbs.
M = modulus of efficiency, or coefficient, say ·75.

Single purchase crab,

$$W = P \times \frac{R}{r} \times \frac{B}{A} \times M.$$

Double purchase crab,

$$W = P \times \frac{R}{r} \times \frac{B}{A} \times \frac{B_1}{A_1} \times M.$$

The nominal power of a crab winch assumes the use of 2 and 3 sheave pulley blocks. Lifting direct from the barrel the actual load will only be one-third.

519. ROPE TACKLE FOR LIFTING.

Diameter of sheave in inches	3	3½	4	4½	5	6
Circumference of rope in inches	1	1½	2	2½	3	4
Average strain on rope in cwts. for full load	1	2¼	3¾	6	8½	15
Number of men required for full load	1	3	6	10	crab	crab
Maximum power in cwts.—						
2 and 1 sheave	2¼	4½	9	..	..	..
2 ,, 2 ,,	3	7	12	..	..	..
3 ,, 2 ,,	..	8¾	15	25	35	60
3 ,, 3 ,,	..	10½	17½	30	42	72
4 ,, 3 ,,	..	..	20	35	49	84

With equal sheaves the fast end must be on top block ; unequal on bottom. Snatch block makes practically no difference if the rope has a good lead. Larger blocks than 6 inches should have chain fall. Blocks 4 inches to 6 inches may have rope or chain.

520. SAFE LOAD ON SHEAR LEGS AND DERRICK POLES.

D = inches diameter at bottom.
d = „ „ top.
L = length in feet.
R = rake or overhang in feet.
W = safe load in tons per pole.

$$\text{Approximate } W = \frac{3\,D\,d}{L + R}.$$

521. DIFFERENTIAL PULLEY CALCULATIONS.

D = diameter of larger pulley. d = diameter of smaller pulley.

$$D : \frac{D - d}{2} : : W : P \quad \therefore \; P = \frac{W \times (D - d)}{2\,D}.$$

M = modulus or efficiency of machine, then W × M = actual load lifted. Load will not lower by itself when M is less than ·5.

$$\text{Velocity ratio} = \frac{2\,D}{D - d} \text{ to } 1.$$

By experiment with various differential pulleys—

Max. Load.	*Multiplying Power.*	*Coefficient.*
5 cwt.	16 to 1	·4
10 „	30 „ 1	·33
30 „	53 „ 1	·25

Approximately, m = multiplying power, coefficient $= \dfrac{100 - m}{200}$.

522. SCREW JACK.

Efficiency of common screw jack = ·25.
Do. do. prepared for experiment = ·28.
Screw jacks are now made with roller bearings in head to reduce the friction and increase the efficiency.

Section VIII.

FRICTION AND LUBRICATION.

523. FRICTION.

Friction is the resistance to relative motion of bodies in contact, caused by the pressure between their surfaces.

Friction is not a force; being passive, it can only act as a resistance.

524. LAWS OF FRICTION.

The friction between two surfaces at rest is greater than when they are in relative motion, but when in motion the friction is practically independent of the velocity so long as the surfaces are kept cool. It has been proposed to call the friction of rest *sticktion*, to distinguish it from the friction of motion, which would be called *friction*. The common term for the former is *statical friction*, and the latter is sometimes distinguished as *kinetic friction*. Statical friction varies with the length of time the surfaces are in contact.

The friction between two surfaces, dry or only slightly greasy, is in direct proportion to the force with which they are pressed together (within the limits of abrasion), and is independent of the area of the surfaces in contact. With ample lubrication the friction is reduced, but the heavier the pressure per unit of surface the greater must be the consistency of the lubricant, to prevent it from being squeezed out.

The friction under motion, when the surfaces are dry, averages 20 to 40 per cent. less than the statical friction, and when they are lubricated the reduction averages 80 per cent. for low pressures to 97½ per cent. for high pressures.

A pair of true "surface plates" resting on each other may be moved with the minimum of friction, as if the film of air formed a continuous roller bearing, but if the air be squeezed out by attempting to slide one on to the other in actual contact the surfaces are liable to seize and become abraded.

The laws of friction (Coulomb) are sometimes stated as follows :—

First Law of Friction.—The friction is proportional to the pressure when the surfaces are the same.

Second Law of Friction.—Friction is independent of the area of the surfaces in contact.

Third Law of Friction.—Friction is independent of the velocity when the surfaces are in motion.

The ordinary laws of friction hold good with pressures from 1 to 100 lbs. per sq. inch for velocities under 10 feet per second. With dry surfaces the coefficient increases with the pressure when it exceeds 100 lbs. per sq. inch, and increases also when reduced below $\frac{3}{4}$ lb. per sq. inch. With velocities exceeding 10 feet per second the coefficient diminishes as the velocity increases.

With lubrication the coefficient depends upon the consistency of the lubricant compared with the intensity of the pressure.

Prof. Fleeming Jenkin (1877), experimenting at very low velocities, found a continuous rather than a sudden change in the value of the coefficient between the conditions of rest and motion.

Beauchamp Tower (1883), experimenting upon lubricated journals, found that the absolute friction was nearly constant under all loads within ordinary working limits.

Hirn (1885), in experimenting on lubricated journals, found $\mu = c\sqrt{\frac{v}{p}}$, where c is a constant found by experiment, v = velocity of rubbing, p = intensity of pressure. Also that friction diminished as the temperature increased.

Prof. Thurston (1885) found that the coefficient first decreases, but after a certain point increases, with the velocity; the point of change varying with the pressure and temperature.

"The ordinary theory of solid friction is that it varies in direct proportion to the load; that it is independent of the extent of surface; and that it tends to diminish with an increase of velocity beyond a certain limit. The theory of liquid friction, on the other hand, is that it is independent of the pressure per unit of surface, is directly dependent on the extent of surface, and increases as the square of the velocity. The results of these experiments seem to show that the friction of a perfectly lubricated journal follows the laws of liquid friction much more closely than those of solid friction. They show that under these circumstances the friction is nearly independent of the pressure per sq. inch, and that it increases with the velocity, though at a rate not nearly so rapid as the square of the velocity."—PROF. A. JAMIESON.

525. FRICTIONAL RESISTANCES.

The frictional resistances of dry and lubricated surfaces are contrasted in the following synopsis by Prof. Goodman :—

Dry Surfaces.	*Lubricated Surfaces.*
1. The frictional resistance is nearly proportional to the normal pressure between the two surfaces.	1. The frictional resistance is almost independent of the pressure with bath lubrication, and approaches the behaviour of dry surfaces as the lubrication becomes more meagre.
2. The frictional resistance is nearly independent of the speed for low pressures. For high pressures it tends to decrease as the speed increases.	2. The frictional resistance varies directly as the speed for low pressures. But for high pressures the friction is very great at low velocities, becoming a minimum at about 100 feet per minute, and afterwards increases approximately as the square root of the speed. N.B.—The friction of liquids varies as the *square*, not as the *square root*, of the speed ; hence the friction of a well-lubricated bearing is not merely that of the lubricant.
3. The frictional resistance is not greatly affected by the temperature.	3. The frictional resistance depends more upon the temperature than on any other condition—partly due to the variation in the viscosity of the oil, and partly to the fact that the diameter of the bearing increases with a rise of temperature more

Dry Surfaces.	*Lubricated Surfaces.*
	rapidly than the diameter of the shaft, and thereby relieves the bearing of side pressure.
4. The frictional resistance depends largely upon the nature of the material of which the rubbing surfaces are composed.	4. The frictional resistance with a flooded bearing depends but slightly upon the nature of the material of which the surfaces are composed, but as the lubrication becomes meagre, the friction follows much the same laws as in the case of dry surfaces.
5. The friction of rest is slightly greater than the friction of motion.	5. The friction of rest is enormously greater than the friction of motion, especially if thin lubricants be used, probably due to their being squeezed out when standing.
6. When the pressures between the surfaces become excessive, seizing occurs.	6. When the pressures between the surfaces become excessive, which is at a much higher pressure than with dry surfaces, the lubricant is squeezed out, and seizing occurs. The pressure at which this occurs depends upon the viscosity of the lubricant.
7. The frictional resistance is greatest at first, and rapidly decreases with the time after the two surfaces are brought together, probably due to the polishing of the surfaces.	7. The frictional resistance is least at first, and rapidly increases with the time after the two surfaces are brought together, probably due to the partial squeezing out of the lubricant.
8. The frictional resistance is always greater immediately after reversal of direction of sliding.	8. Same as in the case of dry surfaces.

Tests of lubricating oils, by the number of revolutions that can be made at a given speed without rise of temperature, are inadequate as usually conducted. The viscosity compared with the pressure on the bearings is an important element.

526. DEFINITIONS OF FRICTION.

The *angle of repose* is the angle (ϕ) made by a flat surface with the horizontal when a weight just ceases to move down it by gravity. The corresponding coefficient of friction ($\tan \phi$) is the fraction of the weight required as pressure just insufficient to produce motion on a horizontal plane.

Angle.		*Coeff.*	*Angle.*		*Coeff.*	*Angle.*		*Coeff.*
1¾°	=	·03	13°	=	·23	19½°	=	·35
2	=	·04	13½	=	·24	20	=	·36
3	=	·05	14	=	·25	26½	=	·50
4	=	·07	14½	=	·26	28	=	·53
4½	=	·08	15	=	·27	29½	=	·57
8½	=	·15	16½	=	·30	31	=	·60
11½	=	·20	18½	=	·33	35	=	·70

The Limiting Angle of Resistance ϕ is the angle through which any surface requires to be lifted from the horizontal to cause a body to be on the point of sliding (friction of rest) or to continue sliding (friction of motion). Its magnitude is fixed by the physical nature of the surfaces in contact. It is also the angle from the vertical made by the resultant of the force or forces acting upon a body when sliding is just about to take place or is taking place.

The Coefficient of Friction μ is the ratio of the pressure P required to overcome the friction of a body on any given horizontal surface, to the whole load W of and on the body $\left(\mu = \frac{P}{W}\right)$. Trigonometrically it is equal to the tangent of the limiting angle of resistance ($\mu = \tan \phi$).

527. MORIN'S EXPERIMENTS ON FRICTION OF MOTION.

Dry:			
Wrought iron on brass	·172	Brass on wrought iron	·161
Cast „ „	·147	„ cast „	·217
Greasy:			
Wrought iron on brass	·160	Brass on wrought iron	·166
Cast „ „	·132	„ cast „	·107
Lubricated with olive oil:			
Wrought iron on brass	·078	Brass on wrought iron	·072
Cast „ „	·078	„ cast „	·077

Oak upon elm dry = $\frac{5}{9}$ of friction of elm upon oak dry (!).

Note.—These results reduced from General Morin's experiments (1831–34) appear to be very questionable, and indicate the necessity for further investigation.

528. COEFFICIENTS OF FRICTION.

Wood on wood	0·25 to 0·50
Metal on wood	0·20 „ 0·60
Metal on metal	0·15 „ 0·30
Leather on wood	0·25 „ 0·50
Leather on metal	0·30 „ 0·60
Stone on stone	0·40 „ 0·65

The friction of rest is approximately about 25 per cent. greater than the friction of motion, but it depends largely upon the length of time the bodies have been in contact.—PROF. GOODMAN.

Wood on wood or metal—dry, ·4 to ·6; greasy, ·2 to ·4; lubricated, ·1 to ·2.

Metal on metal—wet, ·3; dry, ·2; greasy, ·15; lubricated, ·1 standing or ·08 moving.

Leather on metal—wet, ·25; dry, ·5.

Friction of motion = friction of repose × ·7.

Friction varies with the nature of the surfaces, the lubricant, and the temperature.

Unguents should be thick for heavy pressures, that they may resist being forced out; and thin for light pressures, that their viscidity may not add to the resistance.—RANKINE.

In estimating the power to overcome friction, the friction of rest must be taken; but in estimating the effect of friction as a power to resist motion, say a brake strap, the friction of motion must be taken.

529. SAFE WORKING PRESSURE ON MOVING SURFACES.

v = velocity feet per second.

p = pressure lbs. per sq. inch.

$$p = \frac{2240}{3v + 1},$$

but p must not in any case exceed 1200.—RANKINE.

530. EXPERIMENTS ON FRICTION.

Pine upon pine, grain crossed, slide 9 inches × 9 inches, load 14 to 112 lbs. in motion.

$$\mu W = 1 \cdot 44 + \cdot 252 W.$$

—PROF. BALL.

In an experiment with a hand brake on the tender of a locomotive on the Northern Railway of France, it was found that 82·3 per cent. of the whole power applied was absorbed by friction before reaching the brake block.

531. FRICTION AND HEAT.

Friction of any kind, however produced, results in the conversion of mechanical work into heat. One horse-power or 33,000 foot-lbs. of work per minute expended in friction produces $\frac{33,000}{772} = 43$ British thermal units per minute.

532. FRICTION OF MACHINES.

The friction of a machine reduces its efficiency, hence the coefficient of a machine is the fraction representing the ratio of the actual advantage to the theoretical advantage.

533. FRICTION OF JOURNALS.

Coefficient of friction (μ), average ·08; but under favourable conditions may be as low as ·01.

Work expended in friction in foot-lbs. per minute =

$$\mu W \frac{\pi}{12} d R = \cdot 021 \; W \, d \, R.$$

Heat units to be dissipated per minute $= \frac{U}{J}$ (J = 772).

Length of journal depends upon the load and speed, length being increased for high speeds.

$$l = \frac{W \text{ (50 + velocity in feet per minute)}}{70,000 \, d.} \text{ (BOURNE)},$$

or

$$l = d \, (\cdot 004 \, R + 1) \text{ (UNWIN)}.$$

$$l = \frac{W R}{250,000, \text{ to } 300,000},$$

or

$$l = \cdot 4 \text{ to } \cdot 33 \frac{\text{I.H.P.}}{\text{rad. crank inches}}.$$

Increasing diameter increases friction, because the rubbing surface has further to travel in one revolution.

Increasing length reduces the friction per sq. inch, but does not affect the total friction, because for a given space passed through, with a constant load, the friction is independent of surfaces in contact.

The "bearing area" is taken to be the length × diameter.

When an overhanging journal is increased in length the diameter must also be increased slightly, to give same strength as before, $D = d\sqrt[(3)]{\frac{L}{l}}$. Pressure on bearings in lbs. per sq. inch longitudinal section may be

$$= \frac{70{,}000}{50 + \text{velocity in feet per minute}},$$

but must never exceed 1000, maximum say 800 in slow-running engines, down to 400 lbs. in quick-speed engines.

A small endlong play in a parallel journal increases the life by preventing ridges from forming.

534. SHOP SHAFT BEARINGS.

Allowing 1 sq. inch per thermal unit per minute.

P = load in lbs.
μ = frictional coefficient (say ·02).
S = surface speed feet per minute.
J = Joule's equivalent = 772.
T = thermal units evolved per minute.
d = diameter of shaft in inches.
l = length of bearing in inches.

$$T = \frac{P\mu S}{J}, \qquad l = \frac{T}{d}.$$

—PROF. GOODMAN.

Taper of friction cone = $8\frac{1}{2}°$.

Friction of ordinary shop shafting is about 1 horse-power per 100 feet.

Kynoch's roller bearings.—Coefficient of friction of repose = ·0115, of motion = ·0082, or one-tenth that of ordinary bearings. Centre distance of roller bearings in feet = twice diameter shaft inches × 4.

535. LUBRICATION.

The object of lubrication is to reduce the friction between surfaces sliding, revolving, or rolling upon one another, and to reduce the wear of surfaces in motion when in contact with one another.

536. METHODS OF LUBRICATION.

For steam engines there are three methods of lubrication in general use, viz. :—

1. The ordinary gravity system of lubrication, in which the oil is simply allowed to run or drop upon the bearing or surface to be lubricated, being fed by some suitable lubricator or pipe led from a cistern or reservoir.

2. The splash system of lubrication. When this system of lubrication is adopted all the motion work of the engines is enclosed in a case forming the engine frame and base. This system is only used for engines of the high speed type. The base of the engine is filled with oil and water to such a level that the cranks dip as they revolve, and so throw the lubricant on all parts requiring lubrication.

3. Forced lubrication engines. In this system all bearings and slides are supplied with oil under pressure, that is, all parts are coupled up by pipes to a common pump, which forces the oil between the surfaces to be lubricated.

Bearings are frequently lubricated with grease instead of oil, but in this case each bearing is usually supplied with its own pump or lubricator, which forces in the grease under pressure.

Systems 1 and 3 are used in connection with all classes of machinery and gearing, but 2 is only used for high-speed engines of the single-acting type similar to those made by Messrs. Willans and Robinson.—DAVIDSON.

537. LUBRICANTS FOR VARIOUS CASES.

Under very great pressure with slow speed :—Graphite, soapstone, tallow and other greases.

Under heavy pressure and high speed :—Sperm oil, castor oil and heavy mineral oils.

Under light pressures and high speed :—Sperm oil, refined petroleum, olive, rape and cotton-seed oil.

Ordinary machines :—Lard oil, heavy mineral and other vegetable oils.

Steam cylinders :—Heavy mineral oils.—RALLINGS

For lubricating the dies in a screwing machine cheap mineral oil is generally used.

To lubricate turning and milling tools, working on wrought iron or mild steel, a mixture of oil, soft soap, and water is used; or a special lubricant such as "Lardol" (Cowens and Co., Newcastle-on-Tyne) diluted with 15 to 20 parts of water.

In some tests of lubricants for turning tools, with a cutting speed of 131 ft. per minute and a depth of cut of 0·26 in., the following results were obtained :—

	Dry Tool.	*Water.*	*Soft soap* 1, *Water* 20.	*Beckett's Oil* 1 *Water* 40.
H.P. expended at the Cutting Edge.	14·2	14·1	14·26	13·8
Duration of Cutting Edge in seconds	69	174	148	256

538. CYLINDER LUBRICATION.

Animal and vegetable fats and oils are glycerides which are decomposed by high pressure steam into stearic acid, oleic acid, and glycerine, the acids having a corroding effect on metal.

Cylinder oils, in common with other lubricants, should be absolutely neutral in their natural condition and incapable of developing acid in use, free from mucilaginous or gumming properties, and from mechanical impurities. Cylinder oils are distinguished from other lubricants, even of the same class, by (1) their great body or viscosity, particularly at high temperatures; (2) their high flashing point, which in the best oils ought not to be under 500° F.; (3) their high vaporising point, and consequently (4) low percentage of loss by evaporation at working temperatures.

A good cylinder oil should have a specific gravity of about ·9 at 60° F., flash point (open test) 550° F. Taking tallow at 180° F. as the standard, the viscosity of cylinder oil should be about 3½ times as great at that temperature, about equal at 280° F., and about $\frac{7}{10}$ at 360° F.

Cylinder oils may be rapidly tested for gumming properties by exposing them for four hours in thin films to the action of steady currents of hot air at say 300° F.

For gas engines and high-pressure steam engines worked with superheated steam a pure hydrocarbon oil of very high flash point should be used.

539. TESTS FOR ENGINE LUBRICATING OIL.

	Temperature of Steam.	
	Up to 400° F. (204° C.)	400° *to* 500° *F.* (204° *to* 260° *C.*)
Flash point, close test . .	500° F. (260° C.)	600° F. (315° C.)
Fire test.	600° F. (315° C.)	700° F. (371° C.)
Viscosity at 250° F. (121° C.) (Rape oil at 60° F. = 100) .	15	20

and to be free from all grit and fatty acids.—DAVIDSON.

540. SPECIFIC GRAVITY AND VISCOSITY OF VARIOUS STANDARD OILS.

Sperm Oil as standard for Viscosity at 70° F. = 100.	*Specific Gravity at* 60° *F.*	*Viscosity.*	
		At 70° *F.*	*At* 120° *F.*
Sperm oil	875/880	100	45
Whale oil	920/925	190	70
Neatsfoot oil	915/918	247	80
Lard oil	915/918	225	78
Olive oil	915/918	213	75
Rape seed oil	914/917	250	88
Lancastrine Textile oil (Price & Co.)	865/870	173	..
Neutraline (Do.)	880/885	320	..
Albaline (Do.)	880/885	200	55
Climax axle oil (Do.) . . .	880/885	600	120
Sherwood valve oil, A.F. (Do.) .	885/890	Too thick	550
Belmont cylinder oil (Do.) .	895/900	to test.	700

—J. VEITCH WILSON.

541. FLASHING POINT OF OIL.

The flashing point, or flash point, of oil is the temperature at which it gives off inflammable vapour. The flashing point of petroleum and other volatile oils is fixed by the Petroleum Act, 1879, at 73° F. (22·8° C.) in the "close test" by an "Abel" apparatus, which is equal to 100° F. (37·8° C.) by the open test. The test is started with the water in the bath at a uniform temperature of 130° F. (54·4° C.). No external heat is applied during the course of a test, the lamp being only used when, for successive tests, it is necessary to raise the water to the initial temperature. Access to the interior of the oil vessel is obtained through a small opening in the cover of the cup which is closed by a small sliding shutter. When the temperature of 66° F. (18·9° C.) in the oil has been reached, the shutter is slowly pushed aside by the operator, and by the same action the flame, 0·15 of an inch in diameter, from a small oil lamp, or from a gas jet, which is suspended immediately above the opening, is momentarily tilted into it and withdrawn by the return of the shutter to its normal position. This operation is repeated at intervals corresponding with every rise of 1° F. in the temperature of the oil till a flash is observed in the oil vessel, the temperature then indicated by the thermometer in the oil being noted as the flashing point of the latter.

The flashing point of heavy oils is obtained by Gray's apparatus, and the evaporation at high temperatures by Archbutt's vaporimeter. In the latter the oil is contained in a platinum tray, 3 inches long by ½ inch wide and ¼ inch deep. The quantity of oil used for a test is 0·5 grammes, which forms a thick layer on the bottom of the tray, and is exposed for one hour to an air current of two litres per minute by means of a simple regulator. For cylinders working at 160 lbs. pressure per sq. inch the oil is tested at 370° F. (187·8° C.), and should not lose more than from 0·5 to 1 per cent. in weight in the time specified.—J. VEITCH WILSON.

542. OXIDATION OF OILS.

Different oils display, to a widely divergent extent, an affinity for oxygen. In the finer qualities of mineral oil the tendency is entirely absent, justifying their classification as hydrocarbons. Some of the lower grades of mineral oil exhibit a tendency to discolour bright metallic surfaces and to produce on these thin varnish-like films, which are doubtless due to the presence of the products of oxidation.

The animal oils exhibit this tendency to a greater extent than the mineral oils, but in the case of sperm, neatsfoot, lard, and tallow oils, the oxidising tendency is not sufficient to interfere with their use as lubricants. Vegetable oils exhibit the tendency in a higher degree. In the case of olive and rape oils the oxidising tendency is not sufficient to interfere with their use as lubricants, but in the case of cotton seed and linseed oils the tendency is so high as to entirely preclude them from use for this purpose.

The effects of oxidation exhibit themselves in various ways :—

1. In producing the sticky deposits with which most engineers are familiar.

2. In developing acidity in the oil, which corrodes bearings and other metallic objects with which the oil comes into contact.

3. In the generation of heat with the danger of spontaneous combustion.

Oxidation proceeds slowly at normal atmospheric temperatures and more rapidly at higher temperatures, and it is customary to test samples under both conditions.—J. VEITCH WILSON.

543. ACTION OF OILS ON METALS.

The results of twelve months' experiments, by Prof. Redwood, show that—

Iron is least affected by seal oil, very little by rape oil, and most by tallow oil.

Brass is not affected by rape oil, least by seal oil, and most by olive oil.

Tin is not affected by rape oil or whale oil, least by olive oil, and most by cotton-seed oil.

Lead is least affected by olive oil, and most by whale oil; but whale, lard and sperm oils all act to very nearly the same extent on lead.

Zinc is not acted on by mineral lubricating oil, least by lard oil, and most by sperm oil.

Copper is not affected by mineral lubricating oil, least by sperm oil, and most by tallow oil.

Mineral Lubricating Oil has no action on zinc and copper, acts least on brass, and most on lead.

Olive Oil acts least on tin and most on copper.

Rape Oil has no action on brass and tin, acts least on iron, and most on copper.

Tallow Oil acts least on tin and most on copper.

Lard Oil acts least on zinc and most on copper.

Cotton-seed Oil acts least on lead and most on tin.

Sperm Oil acts least on brass and most on zinc.

Whale Oil has no action on tin, acts least on brass, and most on lead.

Seal Oil acts least on brass and most on copper.

From the foregoing results it will be seen that mineral lubricating oil has, on the whole, the least action on the metals experimented with, and sperm oil the most.

For lubricating the journals of heavy machinery, either rape or sperm oil is the best oil to use in admixture with mineral oil, as they have the least effect on brass and iron, which two metals generally constitute the bearing surfaces of an engine. Tallow oil should be used as little as possible, as it has considerable action on iron.

544. ROLLING FRICTION

is directly as the pressure, and inversely as the diameter of the rolling bodies.

545. TRACTION, OR FRICTION ON ROADS.

Cart on common road	=	$\frac{1}{30}$	load.
Carriage on plank road	=	$\frac{1}{100}$	,,
,, on railroad	=	$\frac{1}{300}$	,,

For traction on railways, see later.

546. LAWS OF FRICTION FOR LIQUIDS.

The resistance is proportional to the extent of the surface wetted by the liquid.

The resistance is proportional to the viscosity of the liquid.

The resistance is independent of the material of which the boundary is made and of its surface, provided it is not too rough.

The resistance is independent of the pressure to which the liquid is subjected.

The resistance is diminished by a rise of temperature of the liquid.

The resistance is very small at slow speeds, and below a certain critical speed depending upon the liquid used and its temperature the resistance is proportional to the speed; at speeds above this, the resistance is proportional to some power, approximately the square of the speed.

The practical friction of water and other liquids, passing through valves and pipes, will be dealt with later on under the head of hydraulics.

Section IX.

THERMODYNAMICS, AND STEAM.

547. IMPONDERABLES.

LIGHT, heat, electricity and magnetism were formerly supposed to be material substances without weight, and were known as "imponderables"; they are now considered as modes of motion. Sir Humphry Davy held that heat was merely "a vibratory motion"; while his contemporaries believed it to be a substance known as "phlogiston," or "caloric."

548. UNIVERSAL ETHER.

Sound waves require air for their transmission through space; heat and light are independent of air in their passage, and may be transmitted across a vacuum. It is therefore supposed that there is a medium, more rarefied than air, pervading all space, which transmits waves of heat and light as air does sound.

Aristotle had an idea of such a medium, for he said: "In a void there could be no difference of up and down; for as in nothing there are no differences, so there are none in a privation or negation." Balfour Stewart says: "Is there, after all, a very great difference between this argument and that of modern physicists in favour of a plenum, who tell us that matter cannot act where it is not?"

549. RANKINE'S DYNAMICAL THEORY OF HEAT.

Each atom of matter consists of a nucleus or central physical point enveloped in an elastic atmosphere, which is retained in its position by forces attracted towards the nucleus or centre.

The elasticity due to heat arises from the centrifugal force of revolutions or oscillations among the particles of the atomic atmospheres; so that quantity of heat is the *vis viva* of those revolutions or oscillations.

The medium which transmits light and radiant heat consists of the nuclei of the atoms vibrating independently, or almost independently, of their atmospheres. So that the absorption of light and radiant heat is the transference of motion from the nuclei to their atmospheres, and the emission of light and radiant heat the transference of motion from the atmospheres to their nuclei.

550. SOURCES OF HEAT.

Friction, Percussion, Mechanical stress, Chemical action, Electrical action.

551. HEAT OF EARTH AND SUN.

The temperature of the earth increases 1° F. for every 80 feet of depth below the surface. The temperature of the sun as a whole is estimated at 6,000° F. (3315° C). The temperature of the electric arc is about 10,000° F. (5538° C).

552. SENSIBLE HEAT.

Heat and cold are comparative terms only, what is called cold has merely a lesser degree of heat. The sensation of $\left\{\begin{matrix}\text{warmth}\\ \text{cold}\end{matrix}\right\}$ is produced by temperature $\left\{\begin{matrix}\text{higher}\\ \text{lower}\end{matrix}\right\}$ than that of the body. When a substance feels cold it is gaining heat from the hand, and when it feels hot it is losing heat, heat passing from one to the other until they are of equal temperature.

The *temperature* of a body is its thermal state considered with reference to its power of communicating heat to other bodies.—CLERK MAXWELL.

This is commonly called its sensible heat.

Temperature is a condition of bodies that determines which of two bodies when placed in contact will part with heat to the other.—D. E. JONES.

Temperature bears the same relation to the quantity of heat in a body as the height of water bears to the quantity of water in a given vessel.—J. SPENCER.

A *thermometer* measures the intensity of heat as a pressure gauge measures the intensity of pressure in a gas or liquid.

For purposes of measurement some definite effect produced by heat must be selected—e.g., the alteration in length or volume of a substance which expands and contracts uniformly when heated or cooled.

At all ordinary temperatures the ratio of increment in volume to increment in absolute temperature is practically constant in the case of mercury; it is, moreover, a liquid at such temperatures, and easily measured; hence the *Mercurial Thermometer* is that most commonly used for determining the temperature of a body.

Mercury freezes at—40° F. (—40° C.), boils at 590° F. (310° C.) in vacuum and 660° F. (349° C.) in open air. Another writer gives the freezing point as —38·5° C., and boiling point 357° C.

Air thermometers under constant pressure are more correct than mercurial thermometers because—

1. Thermometers which measure temperature by the expansion of the permanent gases, although they differ from liquid thermometers, agree amongst themselves.

2. If equal quantities of heat are given to a permanent gas they cause equal increments in volume.

553. COMPARISON OF THERMOMETERS.

—	*No. of Degrees between Freezing and Boiling Point of Water.*	*Absolute Zero of Temperature**	*Freezing Point of Water.*	*Point of Maximum Density of Water.*	*Boiling Point of Water.*
Great Britain and America:					
Fahrenheit (1709) = F.	180	—461·2†	32	39·1§	212
Sweden, France, etc.:					
Centrigrade or Celsius = C.	100	—274‡	0	4	100
Russia and Spain:					
Réaumur (1730) = R.	80	—219·2	0	3·2	80
De Lisle = D	150	—411	0	6	150

$\therefore 9° F. = 5° C. = 4° R = 7\tfrac{1}{2}° D.$

To convert from one scale to another:

$$F° = \tfrac{9}{5}\,C° + 32,\quad C° = \tfrac{5}{9}\,(F° - 32),\quad R° = \tfrac{4}{9}\,(F° - 32),$$
$$F° = \tfrac{9}{4}\,R° + 32,\quad C° = \tfrac{5}{4}\,R°,\quad R° = \tfrac{4}{5}\,C°.$$

The greatest cold produced by Prof. Dewar was — 258° C.

The "absolute zero" is the zero of a scale of temperature, the adoption of which enables us to reduce the equation expressing the physical properties of gases to the simplest possible form.—Wormell.

* Or point of absolute negation of heat. † Box - 458·4, Goodeve - 459·13. ‡ Hicks - 273°. § Variously given as 39·1 and 39·2.

554. EFFECT OF CHANGE OF TEMPERATURE.

All bodies expand by heat and contract by cold—i.e., expand by addition of heat and contract by loss of heat; more precisely—change of temperature alters the relation between the attractive and repulsive forces of the atoms of a solid body, and therefore alters the distance at which they would remain in equilibrium, neither attracting nor repelling each other. In the case of gases, the atoms repel each other at all temperatures, and the effect of a change of temperature is to alter the amount of the repulsive force and pressure upon the containing vessel, increasing them with increase of temperature and *vice versâ*.

555. TRANSFER OF HEAT.

Radiation of heat is the transfer which takes place between bodies at all distances apart, in the same manner and according to the same laws as the radiation of light.

The intensity of radiant heat diminishes as the square of the distance from the radiating body.

Conduction is the transfer of heat between two bodies, or parts of a body, which touch each other.

Convection, or carrying of heat, means the transfer and diffusion of the state of heat in a fluid mass by means of the motion of the particles of that mass.

556. PREVOST'S THEORY OF EXCHANGES.

All bodies are continually radiating heat at a rate depending upon their temperatures and upon the nature of their surfaces. If any body radiates more heat than it receives, its temperature falls; if it absorbs more than it radiates, the temperature rises; if the temperature remains uniform, it radiates just as much as it absorbs.

557. FOURIER'S THEORY OF CONDUCTION.

When two contiguous portions of matter are at different temperatures, heat is transferred from the warmer to the colder. This process is called conduction of heat.

"Experiment.—Consider a slab of homogeneous solid bounded by two parallel planes. Let the substance be kept at two different temperatures over these parallel planes by suitable sources of heat and cold. Whatever particular plans of heater and refrigerator be adopted, care must be taken that the temperature be kept uniform over the whole, or over a sufficiently large area of each side of the slab, to render the isothermal surfaces sensibly parallel planes through the whole of the slab intercepted between the two calorimetric areas, and that the temperature at each side is prevented from varying with time. It will be found that heat must continually be applied at one side and removed from the other, to keep the circumstances in the constant condition thus defined."

When heat is unequally distributed among the different parts of a solid mass it tends to attain equilibrium, and passes slowly from the parts which are more heated to those which are less ; and at the same time it is dissipated at the surface, and lost in the medium or in the void. The tendency to uniform distribution and the spontaneous emission which acts at the surface of bodies, change continually the temperature at their different points. The problem of the propagation of heat consists in determining what is the temperature at each point of a body at a given instant, supposing that the initial temperatures are known.

Different materials conduct heat at different rates, but no reason has been assigned for the order in which they stand. It does not depend upon their atomic weight or specific gravity, but agrees fairly with their electrical conductivity.

558. RELATIVE CONDUCTIVITY OF METALS FOR HEAT.

Silver	100·0	Iron	11·9
Copper	73·6	Lead	8·5
Gold	53·2	Platinum	8·4
Aluminium	31·3	German silver	6·0
Zinc	28·1	Antimony	4·0
Brass	24·0	Bismuth	1·8
Tin	15·2	Mercury	1·3

559. THERMAL UNITS TRANSMITTED BY CONDUCTION

per hour per 1° F. difference of temperature per inch thickness.

Copper	555	Glass	6·6
Iron	233	Sand	2·16
Lead	113	Fir	·748

560. DIATHERMANCY.

Diathermanous bodies are those which transmit radiant heat most readily. Luminous heat rays, as from the sun, pass more readily than dark heat rays, as from a heated body.

Dry air transmits	. . . . . .	100 per cent.
Rock-salt ,,	. . . . . .	92 ,,
Fluor spar ,,	. . . . . .	72 ,,
Iceland spar ,,	. . . . . .	39 ,,
Plate glass ,,	. . . . . .	39 ,,
Distilled water ,,	. . . . . .	10 ,,
Alum ,,	. . . . . .	9 ,,
Ice ,,	. . . . . .	6 ,,

—J. SPENCER.

The *diathermancy* of a substance is the ratio of the heat transmitted through unit thickness to the heat entering the front face. The latter quantity is equal to the incident heat minus the amount reflected.

Alum in solution is transparent to light but opaque to heat radiation, water behaves similarly, but to a lesser degree. Glass is opaque to the radiation of heat from a source at a comparatively low temperature, but transmits a considerable proportion of the radiation from a source at a high temperature. —D. E. JONES.

561. PHOSPHORESCENCE, ETC.

Phosphorescence is the property which many substances and organic beings possess of emitting light under certain conditions. Becquerel traces five causes of phosphorescence (1) Spontaneous action; (2) Elevation of temperature; (3) Mechanical action, as friction, percussion, or cleavage; (4) Electricity; (5) Insolation or exposure to the heat and light of the sun.

Fluorescence (Stokes, 1852) is that quality which exists in the rays of light by which, in certain circumstances, they undergo a change of refrangibility. Hence certain solutions which, when viewed by transmitted light are colourless, become bluish under reflected light.

Opalescence is a play of colour formed by the action of light on opals, labrador spar, and mother-o'-pearl.

Caloresence (Tyndall) is the exhibition of light by a body intensely heated by the concentration upon it of invisible heat rays.

Combustion is a rapid form of chemical action, resulting in the combination of two or more bodies and the formation of new compounds, accompanied by the evolution of light and heat. Usually it is a case of oxidation, and combustion is complete only when the whole of the combustible matter is oxidised to its highest state of oxidation.

562. MECHANICAL EQUIVALENT OF HEAT.

British Thermal Unit (B.Th.U.), or unit of heat, is the quantity of heat required to raise 1 lb. of pure water, at its point of maximum density (= 39·1° F.), through 1° F. This unit is sometimes expressed by the symbol Θ, the Greek capital letter *theta*.

Joule's Equivalent (J), or the *dynamical equivalent of heat*, is the mechanical effect resident in one thermal unit = 772 foot-lbs., or 424 kilogrammetres. By Micalesco's experiments, with modern appliances, a closer value would seem to be 772·3 foot-lbs. Prof. Osborne Reynolds and Mr. W. H. Moorby obtained the value of 776·94 from 32° F. to 212° F., which is equivalent to about 773 at 39° F. By Chase's value of γ (= the ratio of specific heats of gases = 1·405,285), Prof. Thurston makes J = 778·12 foot-lbs., or 427 kilogrammetres per calorie.

When the centigrade scale is used, the point of maximum density of water will be 4° C., the thermal unit the quantity of heat required to raise 1 lb. water through 1° C., and its mechanical equivalent 1389·8 say 1390 foot-lbs. The metric unit of heat is the amount required to raise 1 gramme of water through 1° C., and its mechanical equivalent 3·065 foot-lbs.

The Quantity of Heat involved in any operation may be expressed directly by its mechanical equivalent in foot-lbs. The conversion of heat into work consists of the transfer of kinetic energy from particles in molecular motion to particles in molar motion.

Carnot showed that no work could be got out of heat unless there was a flow of heat from a higher temperature level to a lower, just as no work could be got out of water unless it was flowing from a higher level or pressure to a lower.

563. CALORIE OR FRENCH UNIT OF HEAT.

A calorie represents the heat required to raise 1 kilogramme of pure water 1° C. from its point of maximum density 4° C. A calorie is equal to 3·96832 British heat units = C. This is called a "*grand calorie*" or *kilo-calorie*, and is used for engineering purposes.

By other writers, especially on modern physics, a calorie is said to be the amount of heat required to raise 1 gramme of pure water 1° C. from its point of maximum density 4° C. This is made = *c*, and is known as the "*petite calorie*," *cent. gramme caloric* (*S.C.C.*), *gramme-calorie*, *gramme-degree*, *water-*

gramme-degree, thermic unit, or *therm*; and by one writer it is called the *gramme-water-centigrade-degree heat unit.*

A *therm* is the heat equivalent of an *erg* on the C.G.S. system. The centigrade heat unit (C.H.U.), used by a few writers, is the heat required to raise 1 lb. water 1° C.

1 C = 3·96832 B.Th.U.	1 B.Th.U. = 0·251996 C.
1 *c* = 0·003968 B.Th.U.	1 B.Th.U. = 252 *c*.
1 C.H.U. = 1·8 B.Th.U.	1 B.Th.U. = 0·5 C.H.U.

564. COMPARISON OF HEAT UNITS.

Grand Calorie (C)	3066·	foot-lbs.
British Thermal Unit F° (J)	772·	„
British Thermal Unit C°	1390·	„
Petite Calorie (*c*)	3·066	„
Joule (electrical)	0·7373	„ per second.

565. MAYER'S EXPERIMENT.

Dr. Mayer, of Heilbronn, found that 1 cub. foot of air at 32° F., 14·7 lbs. pressure, heated to 525·2° F., expansion being prevented, requires 6·73 units of heat. Heated to same temperature with expansion under constant pressure requires 6·73 + 2·746 = 9·476 units, and volume will be doubled (as the temperature is raised from 32 + 461·2 = 493·2 to 525·2 + 461·2 = 986·4 and 986·4 ÷ 493·2 = 2). The pressure of 1 atmosphere or 2116·3 lbs. on sq. foot is moved through 1 foot, or 2116·3 foot-lbs. of work has been done, and 2116·3 being divided by 2·746, the units of heat which have disappeared, we obtain 770·7 foot-lbs. as the mechanical equivalent of 1 unit of heat. Although the 2·746 units of heat cease to exist as sensible heat, they cannot be called latent as they are transformed into work.

566. ENTROPY.

Entropy (Clausius, 1848) *Thermodynamic Function* (Rankine) is such a quantity as, multiplied by absolute temperature, will give the capacity which heat has theoretically of performing mechanical work.

Mechanical work, electricity, and heat are different forms of energy. Mechanical work is measured by foot-lbs., being the product of the force in lbs. into the space in feet through which it acts. Electrical energy is measured by the watt, being the product of the intensity of current in volts into the quantity of current in amperes. The same requirement applies to heat energy; the intensity is measured by temperature and the quantity by entropy.

"Increase of entropy is a quantity which, when multiplied by the lowest available temperature, gives the incurred waste. In other words, the increase of entropy multiplied by the lowest temperature available gives the energy that either has been already irrevocably degraded into heat during the change in question, or must, at least, be degraded into heat in bringing the working substance back to the standard state. We can only discuss increase or decrease of entropy of a body, we cannot evaluate the whole entropy. The entropy of a body is therefore measured by comparison with a standard state, as, for instance, water at θ 32° F. is taken as the standard for water and steam; the entropy of steam is really the difference between its entropy and that of the same weight of water at 32° F."—J. SWINBURNE.

"If a body takes in or gives out heat at any temperature, the quantity of heat taken in or given out divided by the temperature at which it is taken in or given out measures the amount by which the body changes its entropy." —PROF. EWING.

Entropy has been described by Prof. Dwelshauver-Dery as a heat scale which varies with the absolute temperature, as gravity varies on the surface of the earth with the distance from the centre.

567. CAPACITY OF BODIES FOR HEAT.

Capacity for heat (Irvine), or *thermal capacity*, of a body is the number of units of heat required to raise one pound weight of the body one degree in temperature.

568. SPECIFIC HEAT.

Different bodies require different amounts of heat for changing their temperature.

The *Specific Heat* (Gadolin) of a body is its capacity for heat compared with that of an equal weight of water. It is the quantity of heat requisite to change its temperature any stated number of degrees ($= a$) compared with that which would produce the same effect on water at 60° F. and 30 inch barometer ($= b$), and it is therefore expressed by the fraction $\frac{a}{b}$, which may be made referable to weight or volume.

The quantity of heat needed to effect a given change of temperature is different for different substances.

If a unit mass of a substance absorbs a quantity of heat q in passing from a temperature T, to a temperature T + t, then the ratio q/t is termed the *mean specific heat for t° from the temperature* T.

The limit of the ratio q/t, as t is diminished, is termed the *true specific heat at the temperature* T.

The specific heat of all bodies increases slightly with the temperature, but in the case of air and gases this increase is very slight. For example: Messrs. Holborn and Austen (*vide* "Physical Review," 1905) found the specific heat of CO_2 between 20° C. and 800° C. was about $0{\cdot}2028 + 0{\cdot}0001384\,t - 0{\cdot}000{,}000{,}05\,t^2$.

The specific heat of a gas at constant pressure under which it expands is greater than at constant volume.

569. DULONG AND PETIT'S LAW.

Dulong and Petit's Law (1819).—The specific heats of the chemical elements are inversely proportional to their atomic weights, so that their product is in all cases constant. It is generally expressed as, "the atoms of all elementary bodies have the same specific heat."

Neumann, Regnault and Kopp have shown that this law applies to compounds as well as elements, the specific heat of a compound being the sum of the specific heats of its component elements.

570. SPECIFIC HEATS OF VARIOUS BODIES.

Specific heat of water at 39·1° F. = 1.

Magnesium	·2500	Tin	·0562
Aluminium	·2185	Silver	·0557
Glass	·2000	Antimony	·0508
Grey cast iron	·1268	Platinum	·0330
Steel	·1175	Gold	·0320
Wrought iron	·1138	Mercury	·0320
Nickel	·1086	Lead	·0314
Copper	·0944	Bismuth	·0308
Zinc	·0940	Alcohol	·6150
Brass (70 Cu. 30 Zn.)	·0939	Turpentine	·4250

Specific heat of	air at constant pressure	=	·238
" "	air at constant volume	=	·169
" "	steam gas at constant pressure	=	·475
" "	steam gas at constant volume	=	·364

Prof. Carpenter (*Sibley Journ., Eng.*) found the specific heat of superheated steam = 0·462 + 0·001525 abs. press.

Prof. Dieterici gives the specific heat of superheated steam at constant volume = 0·5 when the volume much exceeds saturation volume; and as the volume diminishes to saturation volume the specific heat increases to 0·7; specific heat at constant pressure varies from 0·6 to 0·8.

The specific heat of superheated steam at constant pressure increases with increasing datum pressure, and diminishes with increasing temperature. It varies approximately between 0·5 and 0·6.

The specific heat of saturated steam (= ·305 water being unity, or ·1281 air being unity) is the quantity by which the total heat of steam is increased for each degree increase of temperature. The specific heat at constant pressure is 1·324 times the specific heat at constant volume.

The specific heat of hydrogen is 3·4, of other gases 0·25.

Thermal units required to raise any body $t°$ in temperature = weight × specific heat × $t°$.

571. RESULTING TEMPERATURE OF MIXTURES.

W = weight of substance in lbs.
S = specific heat of substance in °C.
T = temperature of substance in °C.
W′ = weight of water lbs.
T′ = temperature of water in °C.
x = resulting temperature in °C.

$$x = \frac{S\,W\,T + W'T'}{W' + S\,W}.$$

572. LATENT HEAT.

Latent heat (Black, 1757) is the heat absorbed or disengaged by a body without alteration of temperature, upon a change of state or alteration in the aggregation of its molecules. Approximately the latent heat of steam = 1115 — ·7 times sensible heat F°.

Ice in melting absorbs as much heat as would raise the same weight of water at 32° F. to 174·65° F.

Water in evaporating from 212° F. absorbs as much heat as would raise 966 times the quantity 1° F., or six times the quantity from 51° F. to 212° F.

On the metric system the latent heat of water (liquefaction) is 80° C., and the latent heat of steam (vaporisation) 537° C.—BALFOUR STEWART.

"It is only within about a century that proofs have been gradually arrived at that sensible or thermometric heat consists of motion; while the so-called latent heat of Black may possibly not be heat at all, but may consist of position."—"THE ENGINEER," 19 FEB., 1864.

573. LATENT HEAT OF FUSION.

	Calories per kilo.	*Heat units per lb.*		*Calories per kilo.*	*Heat units per lb.*
Mercury	2·83	5·1	Silver	21·07	38·0
Lead	5·37	9·7	Zinc	28·13	50·7
Bismuth	12·64	22·8	Aluminium	28·50	51·4
Tin	14·25	25·7	Water	79·00	142·2

574. TOTAL HEAT

is the sum of the latent heat and sensible heat in a body at a given time. The total heat in steam at atmospheric pressure is the number of units of heat required to raise 1 lb. of water from 32° F. to the temperature of evaporation and afterwards to convert it into steam.

Dr. Black's theory of the latent and sensible heat of steam was that the sum of the two was constant at all temperatures. Regnault's experiments showed that the total heat was not constant, but increased slowly with increase of temperature, and was equal in F°. to

·305 (Sensible temperature in F°. — 32) + 1123·7.

Approximately the total heat of steam = 1115 + ·3 times sensible heat F°.

GASES AND VAPOURS.

The term gas was adopted by Van Helmont (1640) to distinguish other elastic fluids from common atmospheric air.

Permanent gases are constant elastic fluids which cannot be liquefied.

The temperature being constant, the volume of a gas is inversely as its pressure.

The product of the volume and pressure of any gas is proportional to the absolute temperature.

$$\left.\begin{array}{l} v = \text{volume of a perfect gas} \\ t = \text{absolute temperature} \\ p = \quad ,, \quad \text{pressure} \end{array}\right\} \frac{v\,p}{t} = \text{constant.}$$

In raising the temperature of a gas under constant pressure, mechanical work is done in providing the necessary space for its expansion.

When a gas is heated, the expansion is about $\frac{1}{273}$ of its volume at 0° C. for each degree C. increase of temperature, or permanent gases expand about $\frac{1}{490}$ of volume for each degree increase of temperature from 32° F. under a constant pressure.

Coefficient of expansion of gases = ·003665 per ° C.
,, ,, ,, = ·00203 per ° F.

575. GASES AND VAPOURS.

Ordinary gases are those which do not liquefy at ordinary temperatures or pressures, and the farther they are removed from their point of liquefaction the nearer they approach the character of permanent gases.

Vapours are gases near their point of liquefaction. Ordinary high or low pressure steam is a vapour, superheated steam is a gas.

Vapour of water is absorbed by the air at all temperatures, the higher the temperature of the air the more water it is capable of holding in solution.

A *saturated vapour* is a vapour which is in contact with an excess of its own liquid. Its pressure depends only upon the nature of the liquid and the temperature. At the boiling point of a liquid the vapour-pressure becomes equal to the pressure of the atmosphere.

Absolute humidity is the amount of water vapour present in a given volume of air, or the statement of its *hygrometric condition.*

Relative humidity is the ratio between this and the amount required to saturate the air without alteration of temperature.

Dew-point is the temperature at which air in cooling begins to deposit its moisture in the form of dew.

576. EXPANSION OF AIR.

v = volume of air at t° F.
V = ,, ,, T° F.

$$V = v \times \frac{458 + T}{458 + t}.$$

1 volume air at 60° F. (15·5° C.) becomes

2	volumes at	578° F.	(303° C.)
3	,,	1096° F.	(591° C.)
4	,,	1614° F.	(879° C.)
5	,,	2132° F.	(1166° C.)
6	,,	2650° F.	(1454° C.)

1 volume at 32° expands $\frac{1}{490}$ for each ° F. increase.

,, 60° ,, $\frac{1}{518}$,, ° F. ,,

,, $x°$,, $\frac{1}{490 + x}$,, ° F. ,,

577. KINETIC THEORY OF GASES.

A gaseous body consists of a swarm of innumerable solid particles incessantly moving about with different velocities in rectilinear paths of all conceivable directions, the velocities and directions being changed by mutual encounters at intervals which are short in comparison with ordinary standards of duration, but indefinitely long as compared with the duration of the encounters.

"Gases consist of atoms which behave like solid, perfectly elastic spheres moving with definite velocities in void space."—KROENIG.

A gas consists of a number of molecules, flying in straight lines, and impinging like little projectiles not only on one another, but also on the sides of the vessel holding the gas. Gases of every kind will diffuse into each other. It is thought that the velocity of a molecule of hydrogen at 32° F. and at the atmospheric pressure is 6097 feet per second.—GOODEVE.

578. DEVELOPMENT OF THE KINETIC THEORY OF GASES.

1. The molecules of the same gas must be alike, but those of different gases must differ in properties or structure. They must be separated by intervals which are very great, compared with the size of the molecules.

2. The molecules of a gas move in straight lines.

3. When the molecules come into contact they impinge so that their directions of motion change.

4. All the molecules of the same gas have the same mass, and when they impinge they always rebound.

5. In the same gas, or mixture of gases, the mean energy for each particle is the same.

6. The pressure of a gas per unit of area is proportional to the number of molecules in a unit of volume, and the average energy with which each strikes this area.

7. The pressure per unit of area is proportional to the density of the gas and the average square of the velocity.

8. The temperature of a gas is proportional to the average energy of the molecules.

9. If two equal vessels contain two different gases at the same pressure and temperature, each contains the same number of molecules.

10. To calculate the mean velocity of the molecules of any gas at a given temperature, let M be the mass, for the pressure per unit area, and v the volume at the given temperature. Let V^2 be the average square of the velocity, then—

$$p r = \frac{1}{3} M V^2, \text{ whence } V^2 = \frac{3 p v}{M}.$$

579. LAWS OF GASES.

Boyle's Law (1662), also enunciated by *Marriotte* (1676). The volume of a gas varies inversely as the pressure, or the pressure of a gas is proportional to its density. The modern statement of it is, "The pressure on a given mass of gas at constant temperature is inversely proportional to its volume."

It has been found that Boyle's law is not true in every case. While being nearly accurate for the so-called permanent gases, such as oxygen, hydrogen, etc., it varies for gases capable of condensation to a liquid at ordinary temperatures by the application of pressure; such a gas is carbonic acid. The law is most nearly fulfilled when the temperature of the gas is farthest removed from its point of condensation.

Charles' Law (1787). All gases expand equally, and the volume varies directly as the absolute temperature.

Dalton (1801). A gas at any temperature increases in volume for a rise of 1° by a constant fraction of its volume at that temperature.

Gay-Lussac (1802). The augmentation of volume which a gas receives when the temperature increases 1° is a certain fixed proportion of its initial volume at 0° C.

Under a constant pressure all gases expand uniformly with equal additions of heat, and with a constant volume all gases increase equally in pressure for equal increments of heat.

Avogadro's Law (1811), also attributed to *Ampère* and *Gay-Lussac.* Equal volumes of all substances, when in the gaseous state and under like conditions of pressure and temperature, contain the same number of molecules.

Boyle's is sometimes called the first law of gases, and Charles' the second law.

Graham's Law ( ). The diffusion of gases is inversely as the square root of their densities.

Poisson's Law ( ). If air is suddenly compressed it rises proportionally in temperature, and if suddenly allowed to expand, it falls in temperature.

580. DENSITY OF A GAS.

The density of a gas is the weight of any volume compared with that of the same volume of hydrogen, measured at the same temperature and pressure and taken as unity.

One litre of hydrogen at 0° C. and 760 mm. pressure weighs 0·0896 grammes. and 11·2 litres weigh 1 gramme. The weight of a litre of hydrogen is called a *crith.*

A litre contains 61·027 cub. inches. A gramme is the weight of 1 cub. centimetre ($\frac{1}{1000}$ of a litre) of pure water at its point of maximum density (4° C.). Therefore a cub. foot of hydrogen at 39° F. (4° C.) weighs about ·02637 lbs.

The volume of all gases is directly proportional to their absolute temperature and inversely proportional to the pressure to which they are subjected. The standard temperature is 0° C. (32° F.) and 750 mm. (29·527 inches), say 30 inches of mercury.

581. SPECIFIC GRAVITY OF GASES (30″ Bar., 60° F.).

Atmospheric air	1·0000	Carbonic oxide	0·9720
Hydrogen	0·0694	Carbonic acid	1·5196
Oxygen	1·1111	Coal gas	0·4
Nitrogen	0·9691		

582. DALTON'S LAW OF PRESSURES.

When two or more gases of different kinds are mixed the total pressure exerted by the mixture is equal to the sum of the pressures exerted by the different gases. In other words, each gas exerts its own pressure as though no other gas were present.

583. VOLUME OF A GAS AT GIVEN PRESSURE AND TEMPERATURE.

V = volume of gas at T° and P lbs.

v = " " t° " p lbs.

$$v = V \times \frac{458 \cdot 4 + t}{458 \cdot 4 + T} \times \frac{P}{p}.$$

—Box, on "Heat."

v = volume of elastic fluid given weight and pressure at 32° F.

V = volume it will occupy at same pressure at t° F.

$$V = v + \cdot 00202\, v\,(t - 32).$$ —Gay Lussac.

The volume of a gas under constant pressure expands 1·3665 times when raised from 32° to 212° F.

The initials N.T.P. are used for "normal temperature and pressure."

584. PRESSURE AND TEMPERATURE OF STEAM.

p = lbs. per sq. inch absolute pressure.

t = temperature F°.

1 to 24 atmospheres :

$$p = (\cdot 2697 + \cdot 006803\, t)^5.$$

$$t = 147 \sqrt[5]{p} - 39 \cdot 644.$$

—Arago *and* Dulong.

1 to 4 atmospheres :

$$p = \left(\frac{103 + t}{201 \cdot 18}\right)^6.$$

$$t = 201 \cdot 18 \sqrt[6]{p} - 103.$$

Up to 90 lbs. per sq. inch :

$$p = \tfrac{1}{2}\left(\frac{t + 100}{177}\right)^6.$$

$$t = 177\,(\sqrt[6]{2p}) - 100.$$

—Tredgold.

1 to 4 atmospheres :

$$p = \left(\frac{98 \cdot 8 + t}{198 \cdot 56}\right)^6.$$

$$t = 198 \cdot 56 \sqrt[6]{p} - 98 \cdot 8.$$

—De Pambour.

Up to 1000 lbs. per sq. inch :

$$t = \frac{2938 \cdot 16}{6 \cdot 1993544 - \log p} - 371 \cdot 85.$$

—D. K. Clark.

585. RELATION OF PRESSURE TO TEMPERATURE.

p = original pressure (absolute).

t = " temperature (F°.).

p_1 = new pressure.

t_1 = new temperature.

$$p_1 = p\left(\frac{t_1}{t}\right)^{4\cdot4}.$$

$$t_1 = t\sqrt[4\cdot4]{\frac{p_1}{p}}.$$

586. PRESSURE AND VOLUME OF STEAM BY BOYLE AND MARRIOTTE'S LAW.

P = original pressure.

p = new pressure.

V = original volume.

v = new volume.

$$p = \frac{\text{P V}}{v}, \qquad v = \frac{\text{P V}}{p}.$$

587. RELATIVE VOLUME OF STEAM.

The ratio of the volume of steam to that of the water from which it is produced is called the relative volume.

p = total pressure in lbs. per sq. inch.

t = temperature in F°.

V = relative volume for pressures between 6 and 60 lbs.

$$\text{V} = 37\,\frac{460 + t}{p}, \qquad t = 147\sqrt[5]{p} - 40.$$

Rel. vol. steam at 1 atmos. = 1691·4.

By *Boyle* and *Marriotte's law.*

With constant temperature the volume varies inversely as the pressure ∴ volume × pressure = constant.

$$\text{V approximately} = \frac{25{,}000}{p}.$$

Rel. vol. steam at 1 atmos. = 1700.

By *Navier's modification.*

The temperatures not being constant, in ordinary cases for varying pressures.

$$\text{V up to 26 lbs. per sq. inch} = \frac{27{,}000}{p + 1}.$$

$$\text{V above ,, ,, ,,} = \frac{30{,}000}{p + 4}.$$

Rel. vol. steam at 1 atmos. = 1719.

By *Pole's formula.*

$$V = \frac{24{,}250}{p} + 65, \qquad p = \frac{24{,}250}{v - 65}.$$

Rel. vol. steam at 1 atmos. = 1711·2.

By *Hann's formula.*

$$V = \frac{17{,}149 + 37\,t}{p}.$$

Rel. vol. steam at 1 atmos. = 1700.

Under a pressure of 1 atmosphere 1 cub. foot of alcohol expands into 493·5 cub. feet vapour, ether 212·18, and turpentine 192·15.

588. SATURATED STEAM.

Vapour in a closed space in contact with the generating liquid is said to be *saturated*—i.e., some portion will liquefy upon the smallest increase of pressure or reduction of temperature.

When saturated steam expands in doing work, a portion of it liquefies. —HIRN.

589. PROPERTIES OF SATURATED STEAM.

Absolute Pressure.	*Gauge Pressure.*	*Sensible Temperature.*	*Latent Heat.*	*Total Heat.*	*Weight of Cub. Foot.*	*Relative Volume.*
lbs.	*lbs.*	*deg. F.*			*lbs.*	
14·7	0	212·0	966·1	1178·1	·0380	1644
65	50	298·0	906·3	1204·3	·1538	405
70	55	302·9	902·9	1205·8	·1648	378
75	60	307·5	899·7	1207·2	·1759	353
90	75	320·2	890·9	1211·1	·2089	298
115	100	338·0	878·5	1216·5	·2628	237
135	120	350·1	870·1	1220·2	·3060	203
165	150	366·0	858·9	1224·9	·3695	169
215	200	388·0	843·7	1231·7	·4707	132

590. ATMOSPHERIC PRESSURE.

The pressure of the atmosphere at 60° F. (15·55° C.) and 30 inches (76 cm.) height of barometer is 14·6757 lbs. per sq. inch.

Number of atmospheres × ·006557 = tons per sq. inch.

Absolute pressure is the pressure from zero, or the pressure of the atmosphere added to the indication of the pressure gauge, say gauge pressure + 14·7 lbs. All questions of expansion and compression of steam must be worked from absolute pressure or perfect vacuum line of indicator diagram.

1 lb. of air at 32° F. and 30 inch bar. = 12·384 cub. feet.

1 lb. of steam at 212° F. and 30 inch bar. = 26·37 cub. feet.

To find work done by steam in repelling air during formation:

1 lb. water occupies ·016 cub. foot.

·016 × relative volume = space occupied by steam.

Steam space — ·016 = augmentation of volume.

$$\frac{\text{lbs. per sq. in. abs. press.} \times 144 \times \text{aug. vol.}}{772} = \text{units heat.}$$

Torricellian vacuum is the name given to the space above the column of mercury in a barometer, being the nearest approach possible to a perfect vacuum.

591. LEVELLING WITH BAROMETER.

H = height of barometer at lower station.

h = " " " upper "

T = temperature at lower station.

t = " upper "

Approximate difference of level in feet =

$$\frac{H - h}{H + h} \times 55761 \begin{cases} + 117 \text{ for each degree } \frac{T + t}{2} \text{ exceeds 60.} \\ - 117 \quad " \quad " \quad " \quad \text{falls short of 60.} \end{cases}$$

592. EXPANSION CURVES.

An *Isothermal line* is a curve showing the relations between pressure and volume in a fluid while a constant temperature is maintained. For a perfect gas the isothermal line is a rectangular hyperbola in accordance with Boyle's Law, then $p\,v$ = a constant.

On account of its simplicity this formula and its corresponding curve are frequently used in rough estimations of the practical expansion of steam.

When the temperature is variable, then by Dalton's Law $p\,v = R\,t$, R being a constant and t the absolute temperature.

An *Adiabatic line* is a curve showing the relations between pressure and volume in a fluid while the quantity of heat it contains is maintained constantly uniform; then

$$p\,v^{\gamma} = \text{a constant.}$$

γ = for air 1·4, steam gas 1·3, saturated steam 1·0646, for adiabatic expansion curve 1·324.

When saturated steam expands in a non-conducting cylinder, and during its expansion performs mechanical work, its pressure falls—(1) on account of increase of volume, (2) because of liquefaction. The performance of work by the steam causes an equivalent loss of heat, and the amount of heat transformed into work is sufficient not only to lower the temperature of the steam to that corresponding to its reduced pressure, but also to cause liquefaction of a portion of it. When a small portion liquefies, it liberates its latent heat and keeps the remainder at the temperature of saturation. Rankine gave the following approximate formula for steam expanding under these conditions $p\,v^{\frac{10}{9}} =$ constant, which is nearly an adiabatic curve.—JAMIESON.

Saturation Curve.—When dry saturated steam expands, doing external work, if heat be supplied to it in sufficient quantity just to keep it up to the point of saturation, its pressure is maintained above that given by the adiabatic expansion curve (since there is no condensation), but falls below the isothermal or hyperbolic curve, since its temperature does not remain constant, but drops to the temperature corresponding to the reduced pressure. The formula given by Rankine for the pressure and volume of steam expanding in this way is $p\,v^{\frac{17}{16}} =$ constant, and by Fairbairn and Tate $(v - \cdot 41)\,(p + \cdot 35) =$ constant.—JAMIESON.

As applied to a steam cylinder the saturation curve occupies a position midway between the isothermal and adiabatic curves.

593. LAWS OF EBULLITION.

1. When a liquid is heated it begins to boil at a certain temperature (called its boiling point), and further heating does not raise the temperature of the liquid, but simply converts it into vapour.

2. This temperature is constant for a given liquid as long as the pressure is constant.

3. When the pressure increases the boiling point rises, and when the pressure decreases the boiling point becomes lower.

4. A definite amount of heat (called the latent heat of vaporisation) is absorbed in converting unit mass of liquid at the boiling point into vapour at the same temperature.—D. E. JONES.

594. BOILING POINT.

The boiling point of water or other liquid is that temperature at which the tension or elastic force of its vapour is exactly equal to the pressure of the atmosphere.

Ether boils at		96° F.	(35·5° C.).
Alcohol	„	173° F.	(74·4° C.).
Water	„	212° F.	(100° C.).
Mercury	„	680° F.	(360° C.).

595. TEMPERATURE OF BOILING WATER AND STEAM.

The temperature of boiling water varies with its density, purity, pressure and nature of containing vessel. The temperature of steam from the same or other water will always be uniform for a given pressure.

After water reaches temperature due to pressure, additional heat goes entirely to convert a portion of the water into steam.

596. HEAT REQUIRED FOR EVAPORATION.

Supposing that a certain quantity of water is raised from 32° F. to 212° F. by 1,000 units of heat, then it will require 5,359 additional units to evaporate this quantity, which is given up again upon the condensation of the steam.

To evaporate 1 lb. water at 212° F. under atmospheric pressure = 966 thermal units.

597. SOLUTION AND EVAPORATION OF STEAM.

1 oz. steam passed into 6·35 oz. water at 60° F. raises temperature to 212° F. Poured into shallow pan and allowed to cool to 60° F., the evaporation will reduce the water exactly to its original weight of 6·35 oz. This experiment has been held to show that the apparent increase in temperature of water upon application of heat is due to dissolved steam only.—C. Wye Williams.

598. FIRST LAW OF THERMODYNAMICS.

Heat and mechanical energy are mutually convertible; and heat requires for its production, and produces by its disappearance, mechanical energy in the proportion of 772 foot-lbs. for each British unit of heat.—Rankine.

When work is transformed into heat, or heat into work, the quantity of work is mechanically equivalent to the quantity of heat.—Clerk Maxwell.

Heat and work are mutually convertible, and Joule's equivalent is the rate of exchange.—Jamieson.

When equal quantities of mechanical effect are produced by any means whatever from purely thermal sources, or spent in producing purely thermal effects, equal quantities of heat are put out of existence or are generated.—Wormell.

599. SECOND LAW OF THERMODYNAMICS.

If the total actual heat of a homogeneous and uniformly hot substance be conceived to be divided into any number of equal parts, the effects of these parts in causing work to be performed are equal.—RANKINE.

It is impossible, by the unaided action of natural processes, to transform any part of the heat of a body into mechanical work, except by allowing heat to pass from that body into another at a lower temperature.—CLERK MAXWELL.

It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.—CLAUSIUS.

It is impossible, by means of inanimate material agency, to derive mechanical effect from any portion of matter by cooling it below the temperature of the coldest of the surrounding objects.—SIR W. THOMSON.

Under existing conditions it is impossible to convert the whole of any given quantity of heat into work, and the proportion which can be converted into work follows a certain ratio determined by the absolute temperature of the source of heat and the lowest surrounding temperatures.—" PRACTICAL ENGINEER."

If an engine be such that, when it is worked backwards, the physical and mechanical agencies in every part of its motions are all reversed, it produces as much mechanical effect as can be produced by any thermodynamic engine, with the same temperature of source and refrigerator, from a given quantity of heat.—SIR W. THOMSON.

Heat cannot be made to pass from a cold body to a hot one without the expenditure of work.—WORMELL.

600. CARNOT'S AXIOM.

If a body, after having experienced any number of transformations, be brought identically to its primitive physical state as to density, temperature, and molecular constitution, it must contain the same quantity of heat as it initially possessed.

601. CARNOT'S LAW OR FUNCTION (1824).

The ratio of the maximum mechanical effect to the whole heat expended in an expansive engine is a function solely of the two temperatures at which the heat is respectively received and emitted, and is independent of the nature of the working substance.

602. SIR W. THOMSON'S MODIFICATION OF CARNOT'S LAW (1851).

The efficiency of a perfect heat engine is expressed by the ratio of the difference of the absolute temperatures of the source and condenser, to the absolute temperature of the source; absolute temperature being measured according to a scale so graduated that the temperature of a homogeneous body shall vary in simple proportion to the quantity of energy it possesses in the form of sensible or thermometric heat.

603. LAW OF EFFICIENCY OF THERMODYNAMIC ENGINES.

The heat transformed into mechanical work is to the whole heat received as the range of temperature is to the absolute temperature at which it is received.

$$\frac{\tau_1 - \tau_2}{\tau_1} = \frac{T_1 - T_2}{T_1 + 461} \text{ (Fahr.)} = \frac{T_1 - T_2}{T_1 + 274} \text{ (Cent.)}.$$

Example.—What is the efficiency of a perfect steam engine working at an absolute initial pressure of 100 lbs. per sq. inch, corresponding to about 328° F., the temperature of the condenser being 104° F.?

$$\text{Efficiency} = \frac{328 - 104}{328 + 461} = \cdot 283.$$

Same example with the steam superheated to 600° F.:

$$\text{Efficiency} = \frac{600 - 104}{328 + 461} = \cdot 45.$$

—HY. DYER.

604. PROF. THOMSON'S FORMULA FOR A PERFECT THERMODYNAMIC ENGINE.

S = temperature of source of heat in C°.
T = " " refrigerator in C°.
H = total heat, thermal units C°., entering engine in a given time.
J = Joule's equivalent of 1,390 foot-lbs. per 1° C.
W = work performed or power produced in foot-lbs.

$$W = J\,H\,\frac{S - T}{S + 274}.$$

605. GENERAL VIEW OF HEAT ENGINE.

Heat supplied = work done + heat rejected.

$$\text{Efficiency} = \frac{\text{work done}}{\text{heat supplied}}.$$

As heat engines, the best marine engines only utilise from $\frac{1}{7}$ to $\frac{1}{8}$ of the theoretical value of the steam, ordinary engines about $\frac{1}{10}$.—SEATON.

1 lb. coal contains about 10 million foot-lbs. of potential work.

606. SUPERHEATED STEAM.

Superheated, *surcharged* (Hann and Gener), or *anhydrous* (Dr. Haycraft) steam or *stame* (Frost), is common steam heated away from contact with water. Theoretically it is more economical in use than common steam, as expansion takes place with less condensation, but, owing to its dryness and heat, the packing of the glands and the rods themselves are rapidly destroyed. The difficulties of lubrication are minimised when the superheat is maintained with regularity.

Dr. Siemens found that in isolated steam generated at 212° F. (100° C.), superheated and maintained at atmospheric pressure, expansion proceeded rapidly until the temperature rose to 220° F. (104·4° C.) and less rapidly up to 230° F. (110° C.). Above that it behaved as a permanent gas; below that it augmented in volume five times as fast as air.

607. ADVANTAGES OF SUPERHEATING STEAM.

	Per cent.		
Expenditure of furnace heat in superheating .	5	10	15
Net gain in work done for heat supplied . .	12	38	70

This is due to removal of the loss by cylinder condensation.

Temperature on entering engine should be 300° F. above the temperature normal to the pressure.—PROF. RIPPER.

The improvement in economy from the use of superheated steam is not due, except in very small measure, to the mere fact of increase in temperature in the steam fed to the engine, or to the influence of such increase in temperature commonly known as the thermodynamic or ideal efficiency of the engine. In reciprocating engines the gain comes from the suppression in varying degree of the loss due to cylinder condensation. Superheating is more effective in saving with simple than with compound or multiple-stage engines, an addition of 40° to 50° F. (22° to 28° C.) is usually necessary before any marked economy can take place.

Superheated steam loses temperature rapidly during passage through pipes. In well-lapped pipes Dr. Berner, of Berlin, found a loss of ·3° to 1° F. per foot run. The drop varies inversely as the diameter and steam velocity.

608. DRYNESS OF STEAM.

The dryness of steam may be tested by the M'Innes Separating and Wire-drawing Calorimeter. Saturated steam, or current steam, is steam in contact with the water from which it has been generated and may be technically "dry"—i.e., only containing the water of constitution, or may contain more or less "priming water," or particles in suspension. The "dryness fraction" is the ratio $\frac{W_1}{W_1 + W_2}$ where W_1 = weight of condensed steam which represents weight of dry steam, W_2 = weight of water in separating chambers which represents weight of moisture, then $W_1 + W_2$ = total weight of steam used.

609. CONDENSATION OF STEAM.

Steam may be condensed in

1. The vessel where its power is exerted . .	Savery, 1698. Newcomen, 1705.
2. A separate vessel	Watt, 1769.

Steam may be condensed by

1. Projecting a cold fluid against the vessel containing it	Savery.
2. Injecting a cold fluid amongst it . . .	Newcomen.
3. Exposing it to large surfaces of cold fluids or solids	Watt. Cartwright.
4. The pressure of cold fluids against the vessel containing it	Perkins.
5. By the combination of two or more of these methods	"

—TREDGOLD.

610. VELOCITY OF FLUIDS FLOWING FROM ATMOSPHERE INTO VACUUM.

W = weight per cub. foot of the fluid in lbs.

p = atmospheric pressure in lbs. per sq. foot;

g = force of gravity = 32·2.

v = velocity in feet per second.

$$v = \sqrt{\frac{2\,g\,p}{W}}.$$

Usually $p = 2116 \cdot 4$, then $2 \times 32 \cdot 2 \times 2116 \cdot 4 = 136{,}296 \cdot 16$, and approximately

$$v = \sqrt{\frac{136{,}300}{W}},$$

or for water $v = 46 \cdot 5$, and air $= 1338$.

In all cases allowance must be made for friction, say

$$\text{approx. } v = \sqrt{\frac{100{,}000}{W}}.$$

Velocity from one medium to another of given pressures P and p.

$$v = \sqrt{\frac{2\,g\,(P - p)}{W}}.$$

Steam of all pressures will rush into a perfect vacuum with a velocity of about 2,000 feet per second, no allowance being made for friction.

Steam of 60 lbs. pressure will rush into atmosphere about 1,800 feet per second.

611. DISCHARGE OF STEAM THROUGH PIPES.

The velocity of discharge in pipes is in all cases proportional to the sectional area divided by the circumference; in round pipes this equals one-fourth of the diameter, thus:

$$\frac{d^2 \frac{\pi}{4}}{\pi\, d} = \frac{d}{4},$$

and quantity discharged therefore varies as the diameter^3.

The pressure lost in discharging a fixed volume of steam varies inversely as the 4th power of the diameter of the orifice.

The steam pipe for an engine must be calculated as if constantly passing steam of the maximum velocity required to supply any part of the stroke. With single cylinder engines the maximum velocity may be taken as $1 \cdot 57$ times the mean velocity; and with double cylinder engine, cranks at right angles, maximum $= 1 \cdot 11$ times mean.

Single cylinder:

$$\begin{array}{ll} \text{Max.} & = \pi\, s\, R. \\ \text{Mean} & = 2\, s\, R. \end{array} \quad \therefore \text{ Max. exceeds mean by } \frac{\pi}{2} = 1 \cdot 57.$$

—Box.

612. DIAMETER OF STEAM PIPES.

A = area of piston in sq. inches.

S = piston speed feet per minute.

$$\text{Area steam pipe sq. inches} = \frac{A\,S}{4800}.$$

Another rule (approximate):

$$\text{Diameter steam pipe in inches} = \sqrt{\frac{\text{I.H.P.}}{6}}.$$

Winton's rules:

Main steam pipe for 2 cylinders = $\frac{2}{3}$ area of branch pipes combined. Steam pipe to each valve casing = $\frac{1}{20}$ piston area.

Another rule:

Diameter steam pipe = $\frac{1}{5}$ diameter piston.

613. VELOCITY OF STEAM IN PIPES.

100 feet per second.—UNWIN.

Through main steam pipe	. .	130 feet per second.
„ stop and throttle valves	.	90 „ „
„ steam ports	. . .	80 „ „

—"PRACTICAL ENGINEER."

$1\frac{1}{2}$ miles per minute.—KEENAN AND CO.

Several hydraulic pumping stations gave an average of 90 feet per second in main steam pipe with all engines at full speed.

Velocity of exhaust steam through pipes = 66 feet per second, or exhaust pipe $\frac{1}{4}$ diameter larger than steam pipes.

614. THICKNESS OF STEAM PIPES.

Cast-iron steam pressure pipes between 2 inches and 12 inches diameter, and up to 70 lbs. boiler pressure.

$$d + 4 = t \text{ in } \tfrac{1}{16}\text{ths of an inch.}$$

For exhaust steam, suction and ordinary low-pressure pipes of cast iron,

$$d + 10 = t \text{ in } \tfrac{1}{32}\text{nds of an inch.}$$

Large copper steam pipes (length = 5 diameters).

d = inside diameter inches.

t = thickness inches.

p = working pressure (factor of safety 6).

$$\text{By experiment } t = \frac{p\,d}{4560}.$$

615. EXPANSION OF STEAM PIPES.

Steam pipes expand and contract about 1 inch in 50 feet, or ·02 inches per foot; hence the necessity for inserting expansion pipes between each rigid connection.

616. LOSS OF HEAT BY PIPES AND BOILERS.

A 4-inch steam pipe covered in hair felt and canvas loses about 120 units of heat per foot run per hour at 60 lbs. per sq. inch pressure; bright copper pipe 350 units, rough black pipe 700 units.—Box.

Boilers exposed to open air burn 20 to 25 per cent. less fuel when covered up with Eagle non-conducting cement.

Pipe 1,000 feet long, transmitting 100 H.P., lost by radiation about 11 per cent. when covered by wood and hair together.—"PRACTICAL ENGINEER."

Loss by condensation in uncovered steam pipe, continually under steam and exposed to the atmosphere, equals ½ ton coal per sq. foot of uncovered pipe surface per annum, and covered with best lagging one-sixth of a ton. —W. GEIPEL.

One horse-power is lost by radiation in every 152 feet of 2-inch pipe left uncovered, protected by "Argus" covering the same pipe may be carried nearly 800 feet without greater loss of power.—BELL'S ASBESTOS CO., LTD.

617. COLD STORAGE.

Insulating materials are dry air, sawdust, granulated cork, hair felt, slag wool, etc. Granulated cork weighs 5 lbs. per cub. foot, and costs £10 per ton. A double thickness of matchboard with Willesden paper between for the sides of a hollow wall with 3-inch space filled with granulated cork makes a good insulation.

The effect of the velocity of the air in cooling substances placed in its path is such that the loss of heat is proportional to the square root of the speed when over 3½ feet per second, below this the loss through supports must be taken into account.

618. EXPERIMENTS WITH INSULATING MATERIALS.

The materials employed were (1) granulated cork, (2) silicate cotton, (3) charcoal.

The thickness of material used was the same in all cases.

The following table gives the results of the trials:—

Fall.	*Cork.*		*Silicate cotton.*		*Charcoal.*	
Temperature of starting	100° C.	212° F.	100° C.	212° F.	100° C.	212° F.
Temp. after 12 hours .	37	98·6	30	86	22	71·8
,, 24 ,, .	29	84·2	22	71·8	18	64·4
,, 36 ,, .	23	73·4	17	62·6	15	59

Rise.	*Cork.*		*Silicate cotton.*		*Charcoal.*	
Temperature at starting	0° C.	32° F.	0° C.	32° F.	0° C.	32° F.
Temp. after 12 hours .	10	50	13	55·4	15	59

In these experiments the tests were carried on until the water within one apparatus had attained the temperature of the open air, 15° C. (59° F.).—TATLOCK *and* THOMSON.

619. COMPARATIVE TRANSMISSION OF HEAT.

1. Through various materials in mass.

Poultry feathers	6·2	Plaster of Paris	36·2
Hair felt	11·4	Asbestos powder	47·9
Cork powder	13·6	Fossil meal	52·1
Sawdust	14·2	Fine sand	56·3

2. Through various materials prepared as non-conducting coverings.

Slag wool, hair and clay paste	10·0	Coal ashes and clay paste, wrapped with straw . .	29·9
Fossil meal and hair paste .	10·4	Clay, dung, and vegetable fibre paste . . .	39·6
Paper pulp alone	14·7	Paper pulp, clay and vegetable fibre . . .	40·6
Asbestos fibre wrapped tightly	17·9		
Fossil meal and asbestos-powder	26·3		

Another account gives the relative conducting power as follows :—

Slag wool or mineral wool .	100	Charcoal	140
Hair felt	117	Sawdust	163
Cotton wool	122	Gas works breeze . . .	230
Sheep's wool	136	Wood and air space . .	280
Infusorial earth or fossil meal	136		

C. E. Emery gives the following as the relative efficiency of non-conducting material :—

Hair felt	1·00	Paste of fossil meal and asbestos	0·47
Slag wool	0·83	Fibrous asbestos	0·36
Fossil meal	0·66		

Note.—Other considerations, such as cost and durability, must receive attention in any practical application.

Peclet gave the relative heat conductivity of various substances as follows:

Brick-dust	10	Brick	50
Sand	20	Limestone	100

620. NON-CONDUCTING DRY HAIR FELT.

Maker's Number.	*Approx. Weight per sheet* 34″ × 20″. *oz.*	*Approx. Thickness uncompressed. inches.*
0	12	$\frac{3}{16}$ to $\frac{1}{4}$
1	16	$\frac{1}{3}$ to $\frac{3}{8}$
2	24	$\frac{1}{2}$
3	32	$\frac{5}{8}$ to $\frac{2}{3}$
4	40	$\frac{3}{4}$ to $\frac{5}{8}$
5	48	1

621. WEIGHT PER CUBIC FOOT OF NON-CONDUCTING MATERIALS.

Granulated cork	6·1 lbs.
Hair felt	7·9 „
Slag wool (silicate cotton)	10·0 „
Sawdust	13·1 „
Charcoal	14·5 „
Kieselguhr	15·0 „

622. SILICATE COTTON OR SLAG WOOL

is made by passing molten slag through a blast of steam at high pressure; the best is made from Cleveland slag, which, consisting largely of silica and alumina with little free lime, is more fibrous and tough than other sorts. It is practically fire-proof and indestructible, a good non-conductor of heat and cold, and deadener of sound. It weighs about 12 lbs. per cub. foot when properly packed, and costs about 8d,

623. COMPARATIVE RADIATION.

Approximate units of heat emitted per sq. foot per hour by pipes per 1° F. difference of temperature by radiation and air contact combined.

Dull tinned or galvanised surface .	·62 + ·005	Diff. in F°.
Black iron	·9 + ·005	,,
Rusted iron, wrought or cast .	1·04 + ·005	,,

Or, say for any system of pipes 2 to 4 inches diameter 1·5 units, and for $\frac{3}{4}$-inch thin brass pipes 2·25 units, per 1° F. difference of temperature.

Approximately 1 H.P. is lost by radiation in every 150 feet of 2-inch pipe left uncovered, and so on in direct proportion to diameter.

624. HEATING BY STEAM.

When the external temperature is 10° F. below freezing point, in order to maintain a temperature of 60° F. there will be required, with steam at 212° F.:—

(*a*) One sq. foot of pipe surface for each 6 sq. feet of window glass.

(*b*) One sq. foot of ditto for each 6 cub. feet per minute of air escaping for ventilation.

(*c*) One sq. foot of ditto for each 120 sq. feet of roof, wall, or ceiling.

(*d*) One sq. foot of ditto for each 80 cub. feet of space.

Approximately 1 cub. foot boiler space is sufficient for 2,000 cub. feet space in rooms. Each foot-run of 4-inch pipe will heat 200 cub. feet air 1° F. per minute. Each H.P. of boiler will warm 40,000 cub. feet of space. —Dr. Neil Arnott.

Copper coil pipe transmits 312 units of heat per sq. foot of surface per 1° F. difference of temperature per hour.

The fan system of heating by driving air over steam pipe coils requires only $\frac{1}{3}$ to $\frac{1}{5}$ as much surface as direct radiation, with a velocity of air from 1,000 to 1,800 feet per minute.

625. HEATING BY HOT WATER.

Grate surface, 50 sq. inches per 100 feet run of 4-inch pipe.

Boiler surface exposed to fire, 2 sq. feet per 100 feet of 4-inch pipe.

Ordinary coal will give off 10,000 British thermal units per lb.

Feet-run for 1 foot super.	4-inch pipe =	·85
,, ,, ,,	3 ,, ,, =	1·08
,, ,, ,,	2 ,, ,, =	1·7

With a hot-water temperature of 180° F. (82° C.) and external air temperature of 40° F. (4½° C.), the loss of heat per sq. foot heating surface is 0·18 British thermal units per hour.—T. W. ALDWINCKLE.

Fuel required, 2½ to 5 lbs. per hour per 100 feet run of 4-inch pipe.

Pipes to be laid on rollers to allow for expansion and contraction, which equals 1½ inches in 100 feet. From freezing to boiling point, inches expansion = ·0133 ft. length.

Air-cocks to be provided at highest points of pipe and wherever air is likely to lodge.

Stop-cocks may be half size of pipe, say 4-inch pipe = 2-inch cock.

Supply cistern = $\frac{1}{30}$ contents of boiler and pipes, and connected to return pipe, or at upper side of flow pipe where it joins the return. Where radiators are in use the expansion tank may = $\frac{1}{20}$ total capacity of radiators, piping and boiler.

Rust joint cement, 1 lb. sal-ammoniac, 1 lb. flour of sulphur, 1 cwt. cast-iron borings, made to a dry paste with water and caulked into sockets. The materials should be thoroughly mixed dry in small quantities and the water sprayed on while the mixing is continued. Increasing the sal-ammoniac makes the cement rotten and porous; increasing the sulphur quickens the setting but ultimately bursts the joint.

Special joints with india-rubber rings are now generally used for cast-iron hot-water pipes. Sometimes common socket pipes have round indiarubber rings forced in and the space filled up with Portland cement.

Rainwater store tank for greenhouse, 1 gallon for each sq. foot area of roof.

626. LIVE STEAM HEATING.

Approximate requirements for each 1,000 cub. feet of space in factories and workshops 5 to 7 sq. feet of heating surface; in dwelling rooms, offices, etc., 7 to 9 sq. feet; in drying rooms with a temperature of about 120° F. and a moderate air current passing through, 100 to 150 sq. feet.

Where exhaust steam is employed for heating about one-half more surface must be allowed, and with hot-water heating about twice as much surface of pipes as is requisite for live steam.—T. LEDWARD AND CO.

627. APPROXIMATE RULE FOR HEATING BY HOT WATER.

$$\text{Feet-run of 4-inch pipe} = \frac{\text{cub. contents}}{c},$$

where c = 200 for churches
150 „ dwelling houses
30 „ conservatories
20 „ forcing houses.

Rosser and Russell's rule :

Multiply the cub. contents of the room by the number of degrees to which the air is to be raised, and the product divided by 190, and by the time within which the effect is to be obtained, will give the answer.

Another rule, feet run of 4-inch pipe per 1,000 cub. feet.

For factories and churches . . . . . . 5 to 6
For waiting-rooms, work-rooms, schools, etc. . 7 to 8

Hood's rules :

——	*Feet-run of 4-inch pipe to every 1,000 cub. feet*	*Will give a temperature of*	*Remarks.*
Public rooms	5	55° F.	In cold weather.
Dwelling houses . . .	12	65	. .
Do.	14	70	. .
Halls, shops, etc. . .	10	55	. .
Do.	12	60	. .
Workrooms, etc. . .	6	50 to 55	. .
Do.	8	60	. .
Schools, etc.	7	55 to 58	. .
Linen, etc., drying rooms .	150 to 180	120	When empty.
Do.	150 to 180	80	When full of wet linen.
Bacon, etc., drying rooms .	20	70	. .
Greenhouses	35	56	In coldest weather.
Graperies and storehouses .	45	65 to 70	. .
Do.	50	70 to 75	. .
Pineries, hothouses, and cucumber pits . . .	55	80	. .

628. HEATING FOR GREENHOUSES AND HOT-HOUSES.

G = glass surface in sq. feet.
W = wall surface in sq. feet.
D = degrees Fahrenheit required above external air.
R = required pipe surface in sq. feet.
A = fire grate area in sq. feet.

For low pressure steam:

$$R = (G + \tfrac{1}{5} W) \times \cdot 004 (D - 20)$$

$$A = \frac{\cdot 75 R}{100}$$

For hot water:

$$R = (G + \tfrac{1}{5} W) \times \cdot 006 (D - 20)$$

$$A = \frac{R}{250}$$

Common rule, F — 20 = feet run of 4-inch pipe per 1,000 cub. feet, F being required temperature.

629. HEATING BY WARM AIR.

The loss of heat by transmission through glass is 12 heat units per 10° F. difference of temperature per sq. foot per hour, increased by 25 per cent. for north or east exposure and 15 per cent. for west. Through ordinary walls one-quarter as much as through glass, and ceiling one-twentieth. Add 1¼ heat units for every cub. foot of air escaping. The heat units lost per hour divided by 1·1 gives the volume of heated air required. A heating furnace may be assumed to burn 5 lbs. coal per sq. foot per hour, and give out 8,000 effective heat units per lb. of coal, and approximately 1 heat unit will raise 50 cub. feet of air 1° F. Furnaces have generally 15 to 20 sq. feet of heating surface per sq. foot of grate surface. Each sq. foot of heating surface may be reckoned to give off 2,000 to 2,500 heat units per hour.

630. HEATING AIR FOR TRANSMISSION.

Heating by steam, or hot water radiators, the warm air ducts should have 1½ to 2 sq. inches area per sq. foot heating surface of radiator + 20 sq. inches. Cold air ducts ¾ area warm air ducts. Area of registers one-fourth more than warm air ducts.

631. CHARRING AND IGNITION OF FIR.

Charring temperature for fir 450° F. (232° C.).
Ignition „ „ „ 600° F. (316° C.).

This is of importance in connection with questions of wood fittings in engine and boiler rooms or near bakers' ovens, heating furnaces, etc., but where brickwork intervenes there is always the danger of an unfilled joint permitting direct access of fire by ignited soot, or otherwise. Under the London Building Act no wood may be built in a wall nearer than 12 in. to the inside of any flue, nor may it be applied within 2 in. of the face of any wall containing a flue unless rendered.

632. NEWTON'S LAW OF COOLING.

See Tait's "Heat" (ed. 1884, p. 279).

θ = excess of temperature of cooling body over constant temperature of enclosure.

a = constant = 0·0038 for a body cooling in vacuo.

$$\text{Rate of cooling} = \theta\,(1 + a\,\theta).$$

633. NATURAL VENTILATION.

v = velocity of air in feet per second.
H = height of exit aperture from the ground in feet.
h = „ entrance „ „ „ „
T = temperature of inside air in F°.
t = „ outside „ „
r = ·002, or $\frac{1}{491}$ (the ratio of expansion of air for every degree F. is $\frac{1}{491}$ of its volume).
g = acceleratrix of gravity = 32·2.

Montgolfier's formula.

$$v = \sqrt{2\,g\,h} \quad (h = \text{height corresponding to pressure due to difference of temperature}).$$

De Chaumont's formula.

$$v = \sqrt{2\,g\,r\,(\mathrm{H} - h)\,(\mathrm{T} - t)}$$
$$= \sqrt{\cdot 13\,(\mathrm{H} - h)\,(\mathrm{T} - t)}.$$

Section X.

STEAM BOILERS.

634. VARIETIES OF BOILERS.

NOTE.—A boiler explosion must be reported to the **Board of Trade within** twenty-four hours.

Early Forms.—Spherical and cylindro-spherical, of cast iron, afterwards all wrought iron.

Haystack or Balloon Boiler.—Used formerly in Staffordshire: conical sides, dome top, small flat or hollow bottom.

Wagon Boiler.—Used formerly in Lancashire: flat ends, cylindrical top, hollow curved sides and bottom, held by stays.

Egg-ended Boiler—or cylindro-spherical, set horizontally, with "flash" flues, afterwards made with internal flue, furnace always external.

Rastrick Boiler.—Same as last, but set vertically, one or more horizontal flues leading to main flue through boiler.

Cornish Boiler (Trevithick).—Cylindrical, flat ended, with one flue tube containing furnace. London form shorter than original Cornish.

Lancashire Boiler (Fairbairn, 1844).—Similar to last, but with two flue tubes side by side containing furnaces.

Breeches-flued Boiler.—Similar to last, but with flue tubes uniting into one at back of bridges.

Butterley Boiler.—Similar to Cornish, but with flue tube enlarged at front end, and made elliptical to take wide furnace.

Galloway Boiler—of Cornish or Lancashire type, but with taper water tubes placed diagonally across flue tubes.

The Thompson Boiler is a Cornish or Lancashire boiler with the ends dished to avoid the use of stays.

French, or Elephant Boiler.—Formed of three horizontal cylindrical parts connected to each other by necks, two of these (heaters or bouilleurs) surrounded by brick flues.

Fairbairn Boiler.—Same type as last, but with flue tube through each heater.

Marine Boilers.—Formerly made flat-sided, or any shape to fit ship, stayed where required. Now made cylindrical, short, large diameter, one, two, or three furnace tubes, combustion chamber at back end in one or more divisions. 50 to 250 small tubes from combustion chamber to smoke box at front end. Called *Scotch* boilers.

Locomotive Boilers.—Square furnace box at one end, water jacketed, connected with cylindrical boiler shell containing 200 to 300 small tubes for passage of gases to chimney.

Field Boiler.—Vertical cylindrical, with furnace contained in inner cylinder, top of latter below water line and holding suspended in flame 50 to 60 small double tubes for circulation of water.

Babcock and Wilcox Boiler.—Sectional water-tube boiler with inclined wrought-iron tubes expanded front and back into vertical sinuous headers which are connected by vertical tubes to cross-boxes on a horizontal steam and water drum. No stayed surfaces, and all joints metal to metal. All parts being of small diameter, the boiler is exceptionally safe from disastrous explosions.

Yarrow Water-tube Boiler.—Straight inclined tubes from side semicylinders at bottom to horizontal central drum at top. Furnace in triangular space between sides. Tubes being straight are readily cleaned. Cost low and efficiency easily maintained.

Thornycroft Water Tube Boilers (1884).—Bent tubes connecting bottom cylinders at sides of firegrate with horizontal drum at top. Inner and outer rows of tubes form smoke-tight firebox and flues; the latter containing remaining tubes. Positive circulation; all descending water passing through special tubes. Great working range, with minimum variation of efficiency. Very dry steam. Three principal types.

635. TUBULAR AND WATER-TUBE BOILERS.

In tubular boilers, as the Cornish, Lancashire, marine, and locomotive types, the hot furnace gases pass through tubes which are surrounded by water.

In water-tube boilers, as the Roots, Babcock and Wilcox, and Belleville types, the water circulates through the tubes and the hot gases are in contact with the exterior.

The chief advantages of water-tube boilers are:

1. A large amount of heating surface can be obtained in a comparatively small space.

2. The circulation of the water inside the tubes prevents the formation of scale and facilitates the heating of the water.

3. The parts being all of small diameter are of less thickness for a given pressure and the whole weight is reduced.

4. The risk of damage by explosion is small, as a tube might burst without affecting the remainder, but each tube can easily be made of excessive strength.

636. PRODUCTION OF STEAM IN CORNISH AND LANCASHIRE BOILERS.

An old rule was—Approximately 1 sq. foot of grate surface, 1 sq. yard of heating surface, 1 cub. yard of boiler space, will evaporate 1 cub. foot of water in 1 hour, producing 1 N.H.P., each cub. inch of water forming 1 cub. foot of steam at atmospheric pressure.

637. HORSE POWER OF BOILERS.

Nominal H.P. of Boiler = cub. feet of water evaporated from 60° F. at any pressure in one hour = say 70,000 heat units.

Heat H.P. of Boiler is the amount of heat expressed in foot-lbs. transferred from the fuel to the water and steam per minute ÷ 33,000.

Mechanical H.P. of Boiler is the mechanical work done per minute by the water as it evaporates and expands into steam ÷ 33,000.

If P be absolute steam pressure in lbs. per sq. foot,

V = No. of cubic feet of steam produced per minute.

Then $\frac{P\,V}{33{,}000}$ = mechanical horse-power of boiler.

In America a commonly accepted unit of horse-power for steam boilers is the evaporation of 30 lbs. water per hour from and at 212° F.

The actual horse-power developed by the steam from a boiler depends upon the engine in which it is utilised. In an average modern engine 1 cub. foot of water evaporated per hour will develop 4 horse-power.

638. HORSE-POWER OF BOILERS FROM DIMENSIONS.

S = heating surface in sq. yards.

g = grate surface in sq. feet.

$$\text{H.P.} = (S + g) \times \begin{cases} 1 \text{ for ordinary coal.} \\ 1\tfrac{1}{2} \text{ for good steam coal.} \\ 2 \text{ for best coal only.} \end{cases}$$

—R. ARMSTRONG.

a = area in sq. feet of water surface in boiler + horizontal sectional area of furnace tube in Cornish or Lancashire boiler.

	H.P. =			
Plain cylindrical boiler . .	$\frac{a}{6}$	..	..	$\sqrt{S\,g}$
Cornish or Lancashire boiler .	$\frac{a}{6 \text{ to } 8}$	..	$\frac{3}{5}$ S	..
Galloway boiler . . .	$\frac{a}{4 \cdot 5}$	..	..	..
Multitubular boiler . . .	..	$\frac{g}{\cdot 5 \text{ to } \cdot 8}$	$\frac{1}{2}$ to $\frac{1}{3}$ S	$1 \cdot 8 \sqrt{S\,g}$
Marine boiler (I.H.P. = 5 N.H.P.)	..	..	..	$\cdot 7 \sqrt{S\,g}$

Another rule :—Nom. H.P. = $\frac{1}{6}$ length boiler in feet × diameter in feet.

Some makers consider 15 sq. feet of heating surface equivalent to 1 Nom. H.P.

639. COCHRAN VERTICAL BOILERS.

Allow 4 to 5 sq. feet heating surface per I.H.P., 5 to 6 feet per B.H.P., and 10 to 12 feet per N.H.P. Each sq. foot of heating surface will evaporate about 5 lbs. of water per hour, or 7 to 8 lbs. water per lb. of good steam coal, or 1 cub. foot water per 12 sq. feet heating surface at a working pressure of 100 lbs. per sq. inch, and about 20 lbs. of fuel can be consumed per sq. foot fire grate per hour.

640. TUBULAR BOILERS.

Each nominal horse-power requires :—

8 cub. feet total boiler capacity.

2 ,, steam-room.

10 sq. feet of heating surface—the whole tube surface being taken as effective.

$\frac{1}{2}$ sq. foot of fire-grate surface.

1 cub. foot of water per hour.

10 sq. inches sectional area of tubes.
13 ,, flue area.
6 ,, chimney area.

Diameter of tubes $\frac{1}{30}$th of their length.

—JONES *and* LAUGHLINS.

641. COST OF BOILER POWER.

Total cost of boiler power per horse-power per annum, including interest on capital, fuel, attendance and renewals, say £5 10s. for Lancashire and Cornish boilers, and £8 for locomotive type.

642. EFFICIENCY OF BOILERS.

The *Efficiency*, *Evaporative efficiency*, or *Economic efficiency* of a steam boiler is measured by the proportional quantity of the whole heat of combustion of a given fuel, which is absorbed into the boiler and applied to the conversion of water into steam, and is expressed by the weight of water evaporated from and at 212° F. by 1 lb. of the fuel.

The *Evaporative power* of a boiler is expressed by the total quantity of water evaporated per hour, or per sq. foot of grate area per hour, or per sq. foot or sq. yard of heating surface per hour.—D. K. CLARK.

643. CALCULATING EFFICIENCY OF BOILERS.

E = Efficiency of boiler in per cent.
W = Actual observed evaporation per lb. of fuel.
C = Actual calorific value in British thermal units of 1 lb. fuel used.
H = Total heat of 1 lb. steam in B.Th.U. from water at 32° F.
h = Heat in 1 lb. of feed water above 32°.
h_1 = Heat due to superheating 1 lb. steam, equals superheat × specific heat of steam. (For practical purposes the specific heat of superheated steam = 0·55.)
S = Superheat = $T_1 - T$ in F°.
T_1 = Temperature of superheated steam in F°.
T = Temperature of saturated steam in F°.

The value for W, H, h, h_1, and C may be expressed in English or metrical measurements—that is, lbs., and B.Th.U., or kilos and calories.—BABCOCK *and* WILCOX.

644. SPACE OCCUPIED BY COAL.

Solid coal, say 40 cub. feet per ton.

Coal stores contain 45 cub. feet per ton, or weight say 50 lbs. per cub. foot.

Navy allowance for bunkers, 48 cub. feet per ton.

Coals will run down shoot at slope of 6 inches in 1 foot, or 26°; and down screen bars at 36°. Natural slope of heap = 40°.

Pressure on side of bunker at any point say 12 lbs. per foot head, but should be calculated from natural slope, as a retaining wall.

Average capacity of coal buckets (skips, tanks, corves, tubs, trucks, trollies, etc.), with ordinary topping up, for house coal 2·19 cub. feet per cwt., for gas coal 1·94 cub. foot per cwt., and for Welsh coal 1·52 cub. foot per cwt. Nominally 2 cub. feet per cwt. or 40 cub. feet per ton, but when any particular class of coal is to be dealt with the actual weight should be taken into account.

645. CLASSIFICATION OF COAL.

Lignites	Bituminous wood, containing 15 to 20 per cent. water. Brown coal, ditto.
Bituminous coal	Non-caking, rich in oxygen. Caking. Non-caking, rich in carbon.
Anthracite	Containing about 90 per cent. of carbon.

—Dr. Percy.

646. ELEMENTARY COMPOSITION OF THE COAL SERIES.

Variety.	*Carbon.*	*Hydrogen.*	*Oxygen.*
Wood	100	12·18	83·07
Peat	100	9·85	55·67
Lignite	100	8·37	42·42
Coal	100	6·12	21·23
Anthracite	100	2·84	1·74
Graphite	100	—	—

—F. Roth, U.S.A.

647. COMPOSITION OF COAL.

Variety.	*Composition.*			*Coke yield per cent.*	*Sp. gr.*
	Carbon.	*Hydrogen.*	*O + N.*		
1. Dry coal burning with long flame .	75–80	4·5–5·5	15–19·5	50–60	1·25
2. Caking coal burning with long flame (gas coals) . .	80–85	5–5·8	10–14·2	60–68	1·28–1·3
3. Caking coal proper (furnace coal) .	85–89	5–5·5	5·5–11	68–74	1·30
4. Caking coal burning with short flame (coking coal) .	88–91	4·5–5·5	5·5–6·5	74–82	1·3–1·35
5. Lean or sandy coal burning with short flame . . .	90–93	4–4·5	3–5·5	82–90	1·35–1·4
6. Anthracitic . .	93–95	2–4	3	over 90	1·6

—GRUNER.

648. PETROLEUM AS FUEL.

Petroleum is largely used as fuel, being cheap and developing great heat on combustion. Liquid hydrocarbon or petroleum, say C_7H_{16} = 84 per cent. carbon by weight, 16 per cent. hydrogen by weight, requires 15¼ lbs. net of air per lb. fuel for complete combustion, and gives out about 20,000 units of heat, good solid coal giving 15,000 and average coal 13,000. It occupies less space than coal, a ton of coal occupying 40½ cub. feet, oil 33 cub. feet. It also presents the advantages of greater efficiency of evaporation per unit measure of heating surface, more equable generation of steam, greater cleanliness and freedom from ash, avoidance of heat caused by frequent opening of furnace doors, and instantaneous extinction of furnace fires. It should be heated and burnt through an atomising jet with an air pressure of 20 lbs. per sq. inch giving a balloon-shaped spray. The boiler plates must be protected by fire brick, which helps to diffuse and equalise the heat. Smaller flues are required than with coal owing to the reduced quantity of air passing through.—B. H. BROUGH.

649. CALORIFIC VALUE OF FUELS.

The *calorific power* (Dr. Percy) of a substance is the number of units of heat produced by the combustion of a unit weight of the substance.

Coals of lowest calorific capacity are those which burn with a long flame, their heat of combustion varying from 7,840 to 8,570 calories per kilogramme, after which come the gas coals, varying from 8,400 to 8,770 calories. The most advantageous coals appear to be generally the bituminous and semi-bituminous varieties, which show from 8,570 to 8,870 calories. Some anthracitous coals possess considerable calorific value, while the true anthracites approach, by their heat of combustion, the ordinary flaming coals, giving 8,700 to 8,100 calories. Petroleum gives 11,000 calories.—MAHLER.

Calories per kilog. × 1·8 = British heat units per lb. fuel.

Although Welsh coal has a higher calorific value than butuminous coal, the cost is greater in proportion, so that bituminous coal is more economical, but it requires the furnace to be specially constructed to consume the smoke.

"The method of giving refractory linings to furnaces so that the flames are not cooled before proper and perfect combustion has taken place is the true remedy; it is the right way to prevent smoke, it does not require excess of air, and it does not cause dilution losses; on the contrary, where-ever this system has been there is not only smoke prevention but the highest economical results are found."—COL. CROMPTON.

When fuel is damp much heat is lost in evaporating the moisture, but coals stored in water are said not to deteriorate so much as in air.

650. CHEMICAL COMPOSITION OF FUELS.

——	*Coal (mean 97 kinds).*	*Coke.*	*Wood (ord. state).*	*Peat (ord. state).*
Carbon	·8040	·850	·408	·464
Hydrogen	·0519	..	·042	·048
Oxygen	·0787	..	·334	·248
Nitrogen and sulphur	·0246	..	..	..
Water	..	..	·200	·200
Ashes	·0408	·150	·016	·040
	Totals 1·0000			

651. THEORETICAL UNITS OF HEAT PER LB. OF FUEL.

		A
Coal (mean of 97)	13,006	294
Coke	10,970	269
Wood (dry)	6,582	161

Wood (ordinary)	5,265	129
Charcoal	12,000	294
Peat (dried)	8,736	202

Column A gives cub. feet of air at 62° F. required per lb. of fuel.—Box on "Heat."

652. THEORETICAL HEAT UNITS OF COMBUSTION.

	Units per lb.
Hydrogen burned in oxygen at 0° C. ($2H_2 + 0_2 = 2H_2O$)	62,032
Hydrogen burned in oxygen at 100° C. ($2H_2 + O_2 = 2H_2O$)	51,717
Carbon burned to carbon monoxide ($C_2 + O_2 = 2CO$)	4,451
Carbon burned to carbon dioxide ($C_2 + 2O_2 = 2CO_2$)	14,544
Carbon monoxide burned to carbon dioxide ($CO + O = CO_2$)	4,326
Marsh gas burned to carbon dioxide and water ($CH_4 + 2O_2 = CO_2 + 2H_2O$)	23,513
Olefiant gas burned to carbon dioxide and water ($C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$)	21,344

Units of heat in fuel, C, H, and O being taken at their weight per cent.

$$= 14{,}500 \left\{C + 4{\cdot}265 \left(H - \frac{O}{8}\right)\right\}$$

For complete combustion:

1 lb. hydrogen requires about 36 lbs. air.

1 lb. carbon burnt to CO requires about 6 lbs. air.

1 lb. carbon burnt to CO_2 requires about 12 lbs. air.

Babcock and Wilcox give the formulæ as follows:—

Heat value in B.Th.U. per lb. dry fuel $= 14{,}544\ C + 62{,}032\left(H - \frac{O}{8}\right) + 4{,}032\ S$.

Heat value in calories per kilo $= 8{,}080\ C + 34{,}462\left(H - \frac{O}{8}\right) + 2{,}240\ S$.

C H O and S being the percentage of carbon, hydrogen, oxygen, and sulphur in the fuel.

Dulong's formula:—

Heating value in B.Th.U. $= \frac{1}{100}\left[14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}050\ S\right]$.

Heating values in calories $= \frac{1}{100}\left[8{,}140\ C + 34{,}400\left(H - \frac{O}{8}\right) + 2{,}250\ S\right]$.

Mahler's formula :—

Heating value, calories $= \frac{1}{100}$ [8,140 C + 34,500 H — 3,000 (O + N)].

In the above, C = carbon, H = hydrogen, O = oxygen, N = nitrogen, S = sulphur.

653. ABSOLUTE HEATING POWER OF FUEL.

p = absolute heating power of fuel in "calories."
C = percentage of carbon in fuel.
H = percentage of hydrogen in fuel.
W = percentage of chemically combined or hygroscopic water.

$$p = 80\cdot8\ C + 296\cdot3\ H - 6\cdot4\ W.$$

—"INDUSTRIES."

654. UNITS OF HEAT PER LB. OF FUEL DEVELOPED IN PRACTICE.

Hydrogen burning to water	50,000
Carbon burning to carbonic oxide	3,500
Carbon burning to carbonic acid	14,000
Carbonic oxide burning to carbonic acid	4,000
Welsh coal	8,500
Newcastle coal	8,000
Lancashire coal	7,500
Derbyshire coal	7,000
Wood (ordinary state)	5,000
Liquid hydrocarbons yield about	20,000
Good coal yields about	14,000

$2\frac{1}{2}$ lbs. dry wood = 1 lb. coal.

Note on above :—

(*a*) 1 lb. carbon burning to carbon monoxide develops 3,500 British thermal units. (*b*) 1 lb. carbon burning to carbon dioxide develops 14,000 B.Th.U. (*c*) 1 lb. carbon monoxide burning to carbon dioxide develops 4,000 B.Th.U., so that 6,500 units are apparently lost by doing the work in two stages. The explanation is that in (*c*) only 1 lb. carbon monoxide is taken instead of the whole of that produced by the combustion of 1 lb. carbon, thus, 1 lb. C burning to CO forms 31·6 cub. feet, but 13·55 cub. feet weigh 1 lb., therefore the weight produced will be $\frac{31\cdot6}{13\cdot55} = 2\cdot333$ lbs., and as 1 lb.

develops 4,000 B.Th.U., 2·333 lb. will develop 9,332 B.Th.U., making with the 3,500 developed in forming the CO a total of 12,832 B.Th.U., against 14,000 B.Th.U. developed when the complete combustion is effected in one operation.

655. HEATING BY CONTACT OF GASES.

When difference of temperature is doubled, the rate of transmission is increased 2·35 times.

656. RATE OF TRANSMISSION OF HEAT.

In locomotive boilers the rate of transmission per sq. foot of heating surface is 11 thermal units per hour per degree Fahrenheit of difference in temperature.—J. A. LONGRIDGE.

In the boiler of s.s. *Meteor*, tested by Prof. Kennedy, 4,769 thermal units per sq. foot heating surface per hour were transmitted, or only 3 thermal units per hour per degree Fahrenheit of difference in temperature.—D. HALPIN.

The average number of units of heat transmitted through boiler plates per sq. foot of surface per hour and per degree difference of temperature varies from 5 to 6 B.Th.U. (British thermal units).

Experiments on the conductivity of metals have shown that an iron plate 1 foot square and 1 inch thick, whose opposite surfaces are kept at a uniform difference of temperature of 1° F., will transmit in an hour 473 British thermal units.—GANOT'S "PHYSICS."

On the metric system the quantity of heat which passes per second through a plate 1 cm. in thickness when one face is maintained at a temperature of $t°$ C. and the other at $(t-1)°$ C. is called the thermal conductivity of the material of the plate. The latent heat of vaporisation of water = 536.

M = Total heat transmitted in given time in F units.

$(t - t_1)$ = the difference between the temperatures on the two sides of the iron in degrees F.

C = the quantity of heat transmitted per hour per degree of difference of temperature through unit thickness.

E = thickness of metal.

$$M = (t - t_1)\frac{C}{E}. \qquad \text{—PECLET.}$$

657. CONDITION OF BOILER AFFECTING TRANSMISSION OF HEAT.

Heat units absorbed per sq. foot per hour per degree F. of difference in temperature.

Condition of Boiler.	*1 sq. ft. Heating Surface per lb. Coal Burned per hour.*	*4 sq. ft. Heating Surface per lb. Coal Burned per hour.*
Very clean boiler . . .	6·5	4·6
Fairly clean boiler . . .	6·0	4·3
Rather dirty boiler . . .	5·5	4·1

—M. LONGRIDGE.

658. LOSS OF STRENGTH IN COPPER PLATES WHEN HEATED.

At boiling point, 60 lbs. pressure, 307·5° F. (153° C.) = 10 per cent.
At 500° F. (260° C.) . . . . 50 ,,
At faint red heat, 1000° F. (538° C.) . . . . 75 ,,
At dull red heat, 1300° F. (704° C.). . . . . 100 ,,

659. LOSS OF STRENGTH IN IRON PLATES WHEN HEATED.

At boiling point, 60 lbs. pressure, 307·5° F. (153° C.) = *nil.*
If anything, the strength increases up to 320° F. (160° C.).
At about 550° F. (288° C.) decrease begins to be perceptible.
At faint red heat, say 1000° F. (538° C.) . . 25 per cent.
At dull red heat, say 1300° F. (704° C.) . . 50 ,,

660. COMPARATIVE VALUE OF HEATING SURFACES.

Area of shell exposed to flame . . . . . . = 1
Horizontal area above flame . . . . . . = 1
Surface inclined towards flame . . . . . . = $\frac{3}{4}$
Vertical surface. . . . . = $\frac{1}{2}$
Surface inclined from flame . . . . . . = 0
Horizontal surface below flame . . . . . . = 0
Internal cylindrical flues . . . . = $\frac{1}{2}$ circumference.
Small tubes . . . . = $\frac{2}{3}$,,
Shell of Cornish or Lancashire boiler . = $\frac{2}{3}$ to $\frac{3}{4}$ of lower half.

All measurements should be taken on the surface next the hot gases.

661. HEATING SURFACE OF BOILERS.

Class.	*Proportion of Heating Surface to Grate Surface.*	*Heating Surface to evaporate 1 cub. foot per hour.*
Plain cylindrical	10–16 to 1	18 sq. feet.
Cornish and Lancashire .	15–25 to 1	14 ,,
Multitubular	30–40 to 1	9 ,,
Locomotive	60–90 to 1	6 ,,
Vertical	..	10–16 ,,
Marine	22–35 to 1	8 ,,

In portable boilers tried by Bramwell for the Royal Agricultural Society, with a heating surface varying from 16 to 37 sq. feet per cub. foot evaporated per hour, the total heat effect varied from ·651 to ·776, and the temperature of escaping gases from 775° F. (413° C.) to 500° F. (260° C.). He recommended in the ordinary way 12 to 15 sq. feet heating surface per cub. foot evaporated per hour, and a grate surface of $\frac{1}{24}$ to $\frac{1}{30}$ of this.

Babcock and Wilcox water-tube boiler, 11½ sq. feet heating surface to each N.H.P.

A Lancashire boiler will evaporate 5 lbs. water per sq. foot total heating surface per hour, or 2 cub. feet per sq. foot fire-grate surface per hour, without pushing.

662. PRODUCTS OF COMBUSTION.

100 lbs. coal	80 lbs. carbon . .	293⅓ lbs. carbonic acid gas, say 2520 cub. feet.
	5 lbs. hydrogen .	45 lbs. water, say 1086 cub. feet steam.
	15 lbs. sundry . .	15 lbs. ash.
960 lbs. air	746⅔ lbs. nitrogen .	886⅔ lbs. nitrogen, say 12,000 cub. feet.
	213⅓ lbs. oxygen .	
180 lbs. air	140 lbs. nitrogen .	
	40 lbs. oxygen .	

These figures assume perfect combustion and no losses.

663. AIR REQUIRED TO BURN FUEL.

For the complete combustion of 1 lb. of fuel, the lbs. air theoretically necessary = ·117 times the percentage of carbon + ·35 times the percentage of free hydrogen.

e.g. carbon, 70 per cent.; hydrogen, 3 per cent.

$\cdot 117 \times 70 + \cdot 35 \times 3 = 9{\cdot}24$ lbs. air per lb. fuel.

—" INDUSTRIES."

Good Lancashire coal requires theoretically 10 lbs. weight of air per lb. of coal for perfect combustion, but should be allowed 15 to 16 lbs. in practice. —M. LONGRIDGE.

Practically, we may say, 13 cub. feet of air at 60° F., 30" bar., weigh 1 lb., and 12 lbs. air are required to combine with constituents of 1 lb. coal for perfect combustion, but to allow for working conditions 24 lbs. may be necessary; or 312 cub. feet = 700,000 cub. feet of air per ton of coal.

100 cub. inches atmospheric air at 60° F. and 30" bar. = 31 grains; ∴ 1 cub. foot = ·093 lbs.; 12 cub. feet oxygen weigh 1 lb., and to obtain 1 lb. oxygen, 5 lbs. air must pass through fire = 65 cub. feet.

2 to 3 lbs. oxygen required to burn 1 lb. of coal, or, assuming only two-thirds effective, 185 to 290 cub. feet of air will be required, according to the class of coal.

In general, the quantity of air provided should be double the minimum theoretical quantity.

Air and smoke together equal about 2,000 cub. feet per cub. foot of water evaporated, temperature say 800° F. (432° C.).

Air spaces in fire door = 3 sq. inches per sq. foot of fire-grate.

If the escaping gases contain less than 10 per cent. CO_2, the air supply, either above or below the incandescent fuel, is in excess of practical requirements, and fuel is being wasted to heat it.

664. WEIGHT OF AIR AND GASES.

Taken at 62° F., and 30 inch bar.

	Cub. feet per lb.	*Lbs. per cub. foot.*
Air	13·14	·0761
Nitrogen (N)	13·50	·0741
Oxygen (O)	11·88	·0842

	Cub. feet per lb.	*Lbs. per cub. foot.*
Hydrogen (H)	189·70	·0053
Carbon monoxide (CO)	13·55	·0738
„ dioxide (CO_2)	8·60	·1163
Marsh gas (CH_4)	23·32	·0429
Olefiant gas (C_2H_4)	13·46	·0743

The specific gravity of steam at atmospheric pressure is 0·493 times that of air at 62° F (16·7° C.), and 0·0006 that of water at same temperature.

26·4 cub. feet of steam at atmospheric pressure weigh 1 lb.

Approximately to weigh 1 lb. it will require—

13 cub. feet air.	30 cub. feet coal gas.
8½ cub. feet carbonic acid.	190 cub. feet hydrogen.

665. PRODUCTION OF DRAUGHT IN BOILER FURNACE.

Natural draught, as by a tall chimney, acts above and through the fuel, requiring a thin fire and causing a liability to burn into holes.

Induced draught, as by a suction fan (Sturtevant system) at base of short chimney, acts above and through the fuel in a similar manner.

Forced draught, as by a fan or steam jet in ashpit, acts below the fuel and through it, enabling a thicker fire to be maintained.

Maximum economical draught for boilers = pressure due to ½ inch head of water, causing consumption of 36 lbs. coal per hour per sq. foot of fire-grate, and requiring 24 lbs. air per lb. coal.

In *Howden's system* applied to marine boilers the air is drawn through pipes in the smoke box and delivered into the closed ashpit. In *Meldrum's system* an induced current is produced by steam jets carrying the air into a closed ashpit.

666. ADVANTAGES OF FORCED DRAUGHT.

Under a good system of forced draught several advantages will follow.

1. The working of the furnace will be entirely independent of both wind and weather.

2. There will be practically no loss through leaky brickwork.

3. The blast can be so regulated that no cold air need be drawn through the fire doors when they are opened for cleaning, or otherwise attending to the fire, so that the heating surfaces may be kept at a more uniform temperature.

4. The fire-grate may be reduced in area, and the fire thereby concentrated, by which means the large excess of air over the theoretical quantity required for combustion may be very materially reduced, with a resultant higher and more even temperature, and a consequent greater proportionate evaporation.

5. The heat of combustion will be available down to a much lower temperature than could be the case under natural draught. By the use of feed-water heaters, otherwise known as economisers, by increasing the number of economiser pipes and by partially closing the dampers, the heated gases may be kept in contact with the boiler and its flues for a longer period, and consequently enter the chimney much reduced in temperature.

6. The lower grade fuels, that can only be burned with difficulty under chimney draught, can be efficiently burned under forced draught.

7. Refuse fuels, such as coke dust, coal dust, pan breeze, spent tan, sawdust, ashpit refuse from puddling and reheating furnaces, muffles, etc., which it would be impossible to burn under chimney draught, may be burned with ease by forced draught, the higher rate of combustion compensating for the lower evaporating value of these fuels.

8. The evaporation under forced draught can be increased according to other conditions from 20 per cent. to 50 per cent. without loss of fuel efficiency. —R. B. Hodgson.

667. MELDRUM SYSTEM OF FORCED DRAUGHT.

The Meldrum superheated steam-jet is most economical, using only 3 per cent. of the steam made. The fire door is solid with air-tight joint. The ash-pit is closed by a casting carrying a pair of blower tubes, about 4 inches internal diameter at front end, where a steam nozzle is fixed through which a jet of superheated steam issues, carrying with it a current of air below the fire-bars. The fire-bars are very narrow, with spaces between of $\frac{1}{16}$ inch to $\frac{1}{8}$ inch, and are kept cool by the steam and air. An air valve in the dead-plate is provided for use when occasion arises for the admission of air over the fuel. Less total air supply is needed with forced draught, as it is more thoroughly mixed with the fuel gases.—R. B. Hodgson.

668. DISCHARGE OF AIR UNDER PRESSURE.

P = pressure in inches of water.
V = velocity in feet per second.
C = cub. feet discharged per minute.
d = diameter of orifice in inches.

$$V = 66 \cdot 1 \sqrt{P}.$$
$$C = 21 \cdot 64\, d^2 \sqrt{P}.$$ —Box.

Air discharged under pressure expands, and the heat contained in it, having to occupy a much larger space, is reduced in intensity; any moisture in the air is deposited as snow, and heat is absorbed from surrounding bodies.

669. HEAT IN FLUE.

With Cornish boiler, temperature of escaping gases at base of flue may be as low as 500° F. (260° C.).

With short multitubular boiler, as high as 1,200° F. (650° C.).

A pyrometer indicating up to 1,000° F. (538° C.) was placed in the flue at end of a multitubular boiler of locomotive type, containing tubes 7 feet × 2¾ inches, when the pointer went beyond the range of the instrument = say 1,100° F. (593° C.).

The temperature is generally ascertained by hanging strips of metal foil on an iron wire, across the flue, and noting which are melted by the heat—viz. copper 2,000° F. (1093° C.), aluminium 1,800° F. (982° C.), zinc 750° F. (399° C.), lead 630° F. (332° C.), tin 440° F. (227° C.).

Temperature of boiler furnace, say 2,400° F. (1321° C.).

670. BRICK CHIMNEY SHAFTS, STALKS, OR STACKS.

The bond usually adopted is one course of headers to four of stretchers. Hoop iron may be inserted at intervals of 5 feet. Lias lime mortar is considered superior to Portland cement. Up to 120 feet high, or 5 feet diameter, the top length is generally one brick thick; above that height or diameter, top length 1½ brick thick. Height of any length of uniform section should not exceed 30 feet, and should be less in thin sections. In very high chimneys the distance between each set-off may be more than 30 feet, when special calculations of stability are made—e.g., the Townshend chimney has steps of 40, 80, 80, 80, 80, 94 feet from bottom to top, and St. Rollox 50, 60, 96, 140, and 85 feet. A chimney shaft should not be built up too quickly; 6 feet per day is quite sufficient; the mortar takes some time to set, and a gale occurring might cause compression and a consequent leaning over.

The fire-brick lining should have space at back, 1½ inches or 2 inches, to allow of expansion. It should extend $\frac{1}{3}$ height of shaft + 10 feet, and the air space should be covered at top by a sailing course projecting inwards from the outer shell to prevent choking by dust. No air-bricks should be inserted.

The fire-brick lining need not be increased $4\frac{1}{2}$ inches every 20 feet in height, like the exterior, because it is not subject to the pressure of the wind. An increase of $1\frac{1}{2}$ inches every 20 feet will be sufficient when a clear space is left between the lining and the outer shell. One objection to a fire-brick lining stopping short of the full height is that the difference of temperature of the outer shell just above and below the top of the lining is liable to cause fracture.

An ordinary height of chimney for two steam boilers is 45 feet, but in some towns, as Manchester and Leeds, the minimum height allowed is 90 feet.

A minimum wind pressure of 55 lbs. per sq. foot is generally allowed for in calculating stability—for convenience say $\frac{1}{2}$ cwt.

Round chimneys should not exceed 25 times internal diameter in height.

671. LONDON COUNTY COUNCIL RULES FOR FURNACE CHIMNEY SHAFTS.

The width of a shaft at the base, if square on plan, must be at least one-tenth, and if circular on plan at least one-twelfth of the total height.

A shaft must have a batter of not less than $2\frac{1}{2}$ inches in every 10 feet of height = 1 in 48.

The brickwork must be at least $8\frac{1}{2}$ inches thick at the top of the shaft and for 20 feet below, and must be increased $4\frac{1}{2}$ inches in thickness for every 20 feet of additional height, measured downwards.

No portion of the enclosures of a shaft is permitted to be constructed of fire-brick, and any fire-brick lining to be used must be in addition to the thickness of, and independent of, the brickwork.

No cornice or other projection is allowed to project more than the thickness of the brickwork at the top of the shaft.

672. SIZE OF FACTORY CHIMNEY FOR BOILERS.

W = weight of coal burnt in lbs. per hour.
A = area of chimney in sq. feet at top.
H = height of chimney in feet.
c = cub. feet water evaporated per hour.

$$A = \frac{W}{14\sqrt{H}}, \qquad W = 14\,A\sqrt{H}, \qquad H = \left(\frac{W}{14\,A}\right)^2$$

or,

$$A = \frac{\frac{2}{3}c}{\sqrt{H}}.$$

Chimney for single boiler, area $= \frac{1}{8}$ fire-grate.

Do. under 150 feet high for more than one, area . . $= \frac{1}{10}$ „

Do. over 150 feet high do., area $= \frac{1}{15}$ „

Area of chimney in sq. inches $= \dfrac{\text{lbs. coal per hour} \times 12}{\sqrt{\text{height feet}}}$.

—BOURNE.

Area of chimney usually $\frac{1}{10}$ area of fire-grate and 40 feet high.—SCOTT RUSSELL.

20 sq. inches area per N.H.P. of engine.

Height of about 20 times internal diameter.

Flues $\frac{1}{5}$ area of fire-grate, diminishing to $\frac{1}{10}$ at chimney.

Height of chimney $= 45$ feet.

Area of chimney $= \dfrac{\text{area fire-grate}}{\sqrt{\text{height}} \times 1{\cdot}58}$.—ELSWICK.

Do. $= 1\frac{1}{2}$ sq. inches per lb. of coal per hour.—MURRAY.

Area chimney sq. inches $= \dfrac{120 \times \text{grate surface sq. feet}}{\sqrt{\text{height feet}}}$.

—J. T. HENTHORN.

Area sq. feet $= \dfrac{\frac{3}{10}\,\text{boiler H.P.} + 10}{\sqrt{\text{height feet}}}$.—BERG.

Area chimney top sq. inches $= \dfrac{15 \times \text{lbs. coal per hour}}{\sqrt{\text{height feet}}}$.

Do. do. $= \dfrac{150 \times \text{I.H.P. of engine}}{\sqrt{\text{height feet}}}$.—JONES *and* LAUGHLINS.

Boiler horse-power of chimney $= 3\frac{1}{3}$ (area sq. feet $-$ $0{\cdot}6\sqrt{\text{area}}$) $\sqrt{\text{height feet}}$.—W. KENT.

H.P. $= 2\frac{1}{2} \times d\ \text{feet}^2 \times \sqrt{h\ \text{feet}}$.—W. C. COFFIN.

Chimney H.P. $= \dfrac{d\ \text{inches}^2 \times \sqrt{h\ \text{feet}}}{70}$.—ADAMS.

Approximate horse-power of round chimney $= \dfrac{d\ \text{inches}^3}{300}$.

Funnel for marine engine $= 3$ to 5 sq. inches per indicated horse-power.

Effective area of chimney = 2 inches less all round than actual area. Every 2 feet horizontal flue beyond boiler requires about 1 foot additional height.

A = area of chimney at top in sq. feet.
H = height of chimney above fire bars in feet.
F = consumption of fuel in lbs. per hour.
C = constant, ·1 for coal, ·09 for coke, ·06 for wood.

$$H = \frac{F}{20} + 65 \text{ when F is under 500 lbs. per hour.}$$

$$H = \frac{F}{20} + 100 \text{ when F is above 500 lbs. per hour.}$$

$$A = C\,\frac{F}{20}.$$

—H. Oliver.

W = lbs. of coal consumed per hour.
A = area at top of chimney in sq. feet.
H = height of chimney in feet.
H.P. = horse-power of chimney.
w = inches water balanced by draught.
V = velocity of escaping gases feet per second.

$$W = \frac{A\sqrt{H}}{\cdot 07} \qquad H.P. = \frac{A\sqrt{H}}{0\cdot 75}$$

$$A = \frac{\cdot 07\,W}{\sqrt{H}} \qquad A = \frac{0\cdot 75\,H.P.}{\sqrt{H}}$$

$$H = \left(\frac{\cdot 07\,W}{A}\right)^2 \qquad w = \cdot 0073\,H$$

$$\text{A per H.P., 1 or 2 brs.} = \frac{0\cdot 8}{\sqrt{H}} \qquad \text{A per H.P., several brs.} = \frac{0\cdot 5}{\sqrt{H}}$$

$$\text{Area of flue} = \frac{1\cdot 0}{\sqrt{H}} \qquad \text{V escaping gases} = 2\cdot 4\,\sqrt{H}.$$

—R. Wilson.

They are sometimes proportioned for height according to the coal burnt per week of 56 hours, thus—

4 tons per week	75 feet high.
13 „ „	100 „
26 „ „	120 „
50 „ „	150 „
100 „ „	180 „
150 „ and over	200 „

Another rule is to make the height of chimney three times length of boiler + twice distance of furthest boiler to chimney. This allows 1 foot height for every foot the gases have to travel round the boiler and 2 feet height for every foot of external flue.

Chimneys not over 3 feet 6 inches diameter at the top cannot usefully be increased in height beyond 300 feet.

A chimney need only be large enough for the boilers in work at one time, present or future, and not for the whole number put down, as where there are several boilers some will be laid off for cleaning. In a chimney of unnecessarily large sectional area the gases travel so slowly that they cool down and lower the efficiency, and the draught is then improved when more boilers are put to work.

673. PRINCIPLES OF CHIMNEY DRAUGHT.

D = density or weight of a cub. foot of air at 62° F. = 0·0761.

d = density of chimney gases.

t = temperature of ditto in deg. F.

$$d = 0{\cdot}0761 \left(\frac{523}{461 + t}\right).$$

H = height of chimney.

P = motive force or difference of pressure between column of gas and column of outer air.

$$P = H\,(D - d).$$

h = equivalent head of cold air to produce P,

$$h = \frac{P}{D}.$$

T = temperature of the atmosphere F°.

a = coeff. of expansion of air at 0° F. for each F°. increase of temperature = 0·00217.

$$h = a\left(\frac{t - T}{1 + a\,t}\right) H.$$

v = theoretical velocity in feet per second of air entering furnace.

$$v = \sqrt{2\,g\,h}.$$

V = theoretical velocity of hot gases in chimney in feet per second.

$$V = v \times \frac{D}{d}.$$

Q = volume of gas discharged per second by chimney.
A = area of chimney in sq. feet.

$$Q = V A. \qquad \text{—R. Wilson.}$$

674. CHIMNEY DRAUGHT.

D = density or weight of a cub. foot of air in lbs. at 62° F. = ·0761.
d = average density of gases in chimney shaft.
t = average temperature of ditto, F°.
c = degrees F. required to double the volume of air from 62° F.
a = absolute zero F°.

$$d = D\left(\frac{c}{a + t}\right) = \cdot 0761\left(\frac{523}{461 + t}\right) = \frac{39\cdot 8}{461 + t};$$ for t 600, d = ·0375.

H = height of chimney in feet.
P = difference of pressure per sq. foot = H (D — d).

Example: say for chimney 100 feet high.

$$P = 100\,(\cdot 0761 - \cdot 0375) = 3\cdot 86 \text{ lbs. per sq. foot.}$$

h = head in feet of cold air.

$$= \frac{P}{D} = \frac{3\cdot 86}{\cdot 0761} = 50\cdot 723 \text{ feet.}$$

$$w = \text{inches water} = \frac{h \times 12}{62\cdot 5} = \frac{50\cdot 723 \times 12}{62\cdot 5} = 9\cdot 75 \text{ inches.}$$

v = theoretical velocity of entering cold air.

$$= \sqrt{2\,g\,h} = 8\sqrt{50\cdot 723} = \text{say 60 feet per second;}$$

and of the hot gases

$$= 8\sqrt{h \times \frac{D}{d}} = 8\sqrt{50\cdot 723 \times \frac{\cdot 0761}{\cdot 0375}} = \text{say 80 feet per second}$$

theoretically, but the practical velocity allowing for losses is about = $2\cdot 4\sqrt{H} = 2\cdot 4\sqrt{100}$ = 24 feet per second.

The allowance of air for good chimney draught averages 21 lbs. air per lb. of fuel = 275 cub. feet at 60° F. (15½° C.), and the volume of hot gases resulting = 558 cub. feet at 600° F. (315½° C.). Air at 62° F. is increased in bulk by $\frac{1}{523}$ for each F°. increase of temperature.

½ inch of water is the minimum force of draught for economical firing. The direction of the wind has a very decided effect upon the draught.

675. VELOCITY OF GASES IN CHIMNEY.

Velocity of gases in feet per second $= 8\sqrt{\text{motive height}} = 8\sqrt{h\left(\frac{T-t}{459+T}\right)}$.

Do. practically $= 6\sqrt{\frac{h}{1+\frac{T-t}{500}}}$.

—TREDGOLD.

Ordinary velocity of gases in chimney shaft $= 2\cdot4\sqrt{H}$.—MORIN.

Most economical temperature of escaping gases = 600° F. on leaving boiler; at this temperature the volume of air entering furnace is doubled on exit.

A cub. foot of water requires 10 lbs. coal to evaporate it; 10 lbs. coal require 210 lbs. air for complete combustion, = say 2,750 cub. feet.

The force of the draught in a chimney stack is the deficiency of weight of the column of rarefied air in the chimney compared with a similar column of the external air.

676. HYDRANT OF CHIMNEY DRAUGHT.

The "hydrant" of chimney draught equals inches head of water equivalent to pressure or suction. It will be about

$$= 0\cdot0013\,\frac{H\ (T-t)}{T+459}$$

H being height of chimney in inches, T temperature of gases, t temperature of external air.

Another rule :—

d = draught in inches of water—i.e., vacuum with natural draught or pressure with forced draught.

H = height of chimney in feet.

t = temperature of the atmosphere F°.

T = the absolute temperature of flue gases.

$$d = 7\cdot6\ H\,\frac{T-t}{T\times t}.$$

A factory chimney erected by Boulton and Watt, 80 feet high, 400 sq. inches area, coal consumption 300 lbs. per hour, had a suction in chimney = 1 inch of water.

Another chimney 134 feet high gave average draught of 1 inch of water.

Average in practice = ¾ inch.

Different fuels require different draughts; e.g., round steam coal $\frac{1}{2}$ inch to $\frac{3}{4}$ inch. For slack add 25 per cent.; for coke breeze add 50 per cent.; for round anthracite add 100 per cent. Town refuse requires 2 inches.

677. TO FIND HEIGHT OF CHIMNEY FOR GIVEN HYDRANT.

h = hydrant or required head in inches of water.

t = difference of temperature F°. between outside air and chimney gases.

H = height of chimney in feet.

$$H = \frac{h\ (523 + t)}{0 \cdot 0146\, t}.$$

For ordinary cases $t = 538$, then $H = 135\ h$.—R. WILSON.

678. COST OF TALL BRICK CHIMNEY.

Average cost of chimney say 1s. to 1s. 6d. per cub. foot or about $\frac{H^2}{40} = £$.

679. PHENOMENA OF COMBUSTION.

The changes in its physical and chemical properties undergone by a charge of coal after it is thrown on the fire are as follows:—

(*a*) Previous to putting on a charge of coal the temperature of the bed of coals is from dull red heat (700° C. or 1,292° F.) up to a bright white heat (1,400° C. or 2,552° F.), or even higher.

(*b*) The coal, when fired, is about 15° C. or 60° F. (temperature of the room). As soon as it reaches the firebed it begins to heat by conduction from the hot coals beneath. The hot gases—products of combustion of the coal beneath—also heat the new charge of coal.

(*c*) The heating of the coal causes the volatile matter to distil off. The amount distilled at any given temperature is unknown, but it is certain that traces of volatile combustible matters are given off as low as 110° C. (220° F.).

(*d*) At about 400° C. or 750° F. the coal reaches the temperature of ignition and burns to carbon dioxide.

(*e*) At about 600° C. or 1,100° F. most of the gases given off by coal (hydrogen, marsh gas, and other volatile hydrocarbons) will ignite if oxygen be present.

(*f*) At 800° C. (1·470° F.) the carbon dioxide, as soon as formed from the coal, will give up one atom of its oxygen to burn more coal, thus $CO^2 + C = 2CO$. This carbonic oxide will burn back to carbon dioxide if mixed with oxygen at the necessary temperature, which is between 650 and 730° C. (1,200 and 1,350° F.).

(*g*) At about 1,000° C. or 1,832° F. the H_2O formed by the burning of the hydrogen in the volatile matter in the coal begins to dissociate.

(*h*) At about 1,000° C. or 1,832° F. any carbon dioxide not previously burned to carbonic oxide begins to dissociate to carbonic oxide and oxygen.

(*i*) The various hydrocarbons which begin to be distilled at 110° C. (230° F.), and possibly lower, undergo many changes, dissociations, and breakings-up at the various temperatures they pass through. So many of these are unknown that it is useless to state the few we do know.

Above 700° C. (1,300° F.) both the hydrocarbons and the carbonic oxide will unite with oxygen if the latter be present and intimately mixed with them. If they do not burn, the tendency is always to break up into simpler and more volatile compounds as the temperature rises.

Observations show that about a minute, more or less, after firing, the percentage of carbon dioxide rises sharply and suddenly to a maximum, and then begins to fall, in general continuing to fall until the next firing.—"Mechanical World."

680. CO_2 RECORDING APPARATUS.

The efficiency of the combustion may be ascertained from the percentage of carbon dioxide (CO_2) in the escaping gases. Sanders, Rehders & Co., London, make the "Ados" Automatic CO_2 Recording Apparatus, by which this may not only be tested but automatically registered. Its working is based upon the well-known fact that a solution of caustic potash absorbs carbonic acid gas. The theoretical amount of 21 per cent. of CO_2 in the flue gases cannot be attained in practice, 15 per cent. being about the maximum under best conditions.

15	per cent.	CO_2	=	12	per cent.	fuel	wasted.
12	,,	,,	=	15	,,	,,	,,
9	,,	,,	=	20	,,	,,	,,
5	,,	,,	=	36	,,	,,	,,
4	,,	,,	=	45	,,	,,	,,

681. RATE OF COMBUSTION.

In lbs. of coal burnt per sq. foot of fire-grate per hour.

Cornish boilers	$3\frac{1}{2}$
Old land boilers	10
Recent land boilers	13–14
Modern marine boilers	16–24
Locomotive boilers	80–120

Another account:—

Cornish boilers for pumping engines	4–10
,, and others for factory uses	10–15
Marine boilers, ordinary rates	15–20
Boilers with strong chimney draught	20–30
Locomotives	60–120

A boiler may be made to do 70 per cent. more work if the consumption of fuel can be doubled, but the life of the boiler will be considerably shortened.

Generally on land boilers 10 to 12 lbs. of hard coal, or 18 to 20 lbs. of soft coal, can be burned per hour with natural draught, and nearly double these quantities with forced draught.

682. BOILER FURNACES.

With bituminous fuel the layer in the furnace should be about 4 to 6 inches thick, and should never exceed 12 inches. Thin firing is more economical, but requires more careful stoking. Fresh fuel should be put in front of the fire and the red-hot fuel pushed back, or should be spread thinly over the surface after the hollows are filled up. With coke or hard coal the fire may be thicker, especially if a blast be used. For locomotive boilers the fire may be 18 inches thick. Coal for furnaces should not exceed 1-lb. lumps, say 3-inch cubes.

For land boilers with hand firing, fuel should be added about every half-hour, and more air admitted for the next ten minutes.

Small coal, or slack, has about half the evaporative power of coal or coke.

The clear sectional area over bridge should be $\frac{1}{6}$ of fire-grate area.

North-country coal has practically the same calorific value as Welsh coal, but there is more difficulty in burning it in boiler furnaces, and much smoke is generated. "It is the smokelessness, and not the calorific power, which gives to Welsh coal its great market value."—Prof. Unwin.

683. HEAT IN BOILER FURNACES.

1. Temperature of furnace, say about 2,500° F. (1371° C.).
2. ,, escaping gases, say 600° to 1,200° F. (316° to 649° C.).
3. ,, steam and water in boiler, say 300° F. (149° C.).
4. ,, water in condenser, say 100° F. (38° C.).

Difference between (1) and (2) is absorbed by the water in raising its temperature, by the steam as latent heat, and by the air entering furnace in excess of quantity required for combustion.

Difference between (2) and (3) is utilised in creating draught; 600° F. is the most economical temperature of escaping gases, as it allows sufficient difference of temperature for rapid passage of heat to water, and the density is sufficiently reduced to give rapid ascending current in chimney shaft.

Difference between (3) and (4) is utilised in the engine.

The difference of temperature or quantity of sensible heat does not by itself represent the comparative efficiency.

684. LOSS OF HEAT IN BOILERS.

Assuming that it requires 10 lbs. of coal to evaporate 1 cub. foot of water from 60° into steam at 60 lbs. per sq. inch gauge pressure, the loss of heat may be shown, as follows, to be nearly 50 per cent.:—

Total heat of combustion in 1 lb. of coal in British thermal units = say 13,000.
13,000 units per lb. × 10 lbs. coal . . . . = 130,000

Steam at 60 lbs. pressure has a total heat of 1,207 units. 1,207 — 60° temperature of feed-water = 1,147 units per lb. of water. 1 cub. foot water = 62·5 lbs. 62·5 × 1,147 . = 71,687

Loss in chimney, 24 lbs. air, required to burn 1 lb. coal. 24 × 10 = 240 lbs. to burn 10 lbs. coal. Specific heat of air = ·2374. Temperature of escaping gases = 600°.
240 × ·2374 × 600 . . . . = 34,185

Loss in hot ashes, fuel dropped through, etc., say 7 per cent. of total heat	=	9,100
Loss by radiation and conduction, say 7 per cent.	=	9,100
Loss by imperfect combustion, say 4½ per cent.	=	5,850
		129,922

In ordinary cases large boilers utilise about 8,000 units of heat per lb. of coal.

Another summary shows:—

	B.Th.U.
Lost in ashes	135
„ radiation from boiler	675
Carried off in gases	2,970
„ „ auxiliary exhaust	190
Lost in radiation and leakage, main pipes	210
„ „ „ „ small pipes	30
„ „ „ „ in engine	280
Rejected to condenser	7,737
	12,227
Converted to power supplied to engine	1,273
Value of coal	13,500

With natural-draught furnaces from 15 to 25 per cent. of the heat is expended in producing chimney draught, from 15 to 20 per cent. is lost in heating the air unnecessarily admitted to the furnace, from 5 to 10 per cent. is lost from minor causes, such as radiation, absorption in seatings, leaky brickwork in flues, and unburnt fuel dropped through widely-spaced firebars. It follows, therefore, that out of 100 units of heat generated in the furnace from 35 to 55 per cent. are dispersed in other ways than steam raising.—R. B. Hodgson.

685. BOILER LAGGING.

To prevent radiation, the exposed parts of steam boilers should always be covered with non-conducting composition. About 2 inches thick should be first applied, then wire-netting of large mesh should be closely tied round and a final ½ inch of composition applied. When quite dry a coat of coal tar should be applied.

686. HAND-FIRING AND MECHANICAL STOKING.

Summary of trials of Vicars' mechanical stokers at the City of London Electric Lighting Station, Bankside, S.E., against hand-firing.

—	*Babcock and Wilcox Boilers.*	
	Hand-Firing.	*Vicars' Patent System.*
Date of trial	26th April, 1894	4th June, 1894
Duration of trial	9 hours	6 hours
Description of fuel used	Nixon's Navigation	Bituminous rough small
Price of fuel used	16s. per ton	10s. per ton
Fuel consumed	13,664 lbs.	9,184 lbs.
Ashes and clinker	532 lbs.	1,022 lbs.
Weight of combustible	13,132 lbs.	8,162 lbs.
Per cent. of ash	3·9 per cent.	11·1 per cent.
Draught in flue (av.)	·48″ water	·55″ water
Temperature in flue (av.)	210·8° C.	453° F.
Water evaporated	118,535 lbs.	82,100 lbs.
Temperature of feed water	54·3° F.	62° F.
Average steam pressure	145·9 lbs.	157·5 lbs.
Water evaporated per hour	13,170 lbs.	13,683 lbs.
Water evaporated per lb. of fuel under actual conditions	8·67 lbs.	8·94 lbs.
Water evaporated per lb. of combustible	9·02 lbs.	10·05 lbs.
Water evaporated per lb. of fuel from and at 212° F.	10·50 lbs.	10·78 lbs.
Water evaporated per lb. of combustible from and at 212° F.	10·93 lbs.	12·12 lbs.
Cost of evaporating 500 gallons	40·8d.	24·8d.

687. STEAM RAISING TEST,

with Babcock and Wilcox boiler, at Belfast, fitted with the Bennis machine stoker and self-cleaning compressed air furnace.

Date of trial	March 7th, 1902.
Duration of trial	$5\frac{1}{4}$ hours.
Coal used	"Mynydd Newydd."
Calorific value of coal by calorimeter . .	14,049 B.Th.U. per lb.
B.Th.U.'s required to evaporate 1 lb. of water, including superheat	1,234·5 B.Th.U.

Heating surface of boiler	4,228 sq. feet.
Grate surface	50 sq. feet.
Ratio of heating to grate surface	84·56 to 1.
Steam pressure by gauge (average)	161·4 lbs. per sq. inch,
Average temperature of superheated steam leaving boiler	525° F.
No. of degrees F. of superheat	154.
Feed water temperature (average)	66·5° F.
B.Th.U.'s required to evaporate 1 lb. of water, omitting superheat	1,160·6 B.Th.U.
Factor of equivalent evaporation, as from and at 212° F.	1·2018.
Total fuel burnt	9,243 lbs.
Fuel burnt per hour	1,760·6 lbs.
Fuel burnt per sq. foot of grate per hour	35·2 lbs.
Percentage of carbonic acid in flue gases leaving boiler (average)	14·5 per cent.

	As from actual conditions.	*As from and at* 212° F.
Total water evaporated	92,000 lbs.	110,565·6 lbs.
Water evaporated per hour	17,523·8 lbs.	21,060·1 lbs.
Water evaporated per sq. foot of boiler heating surface per hour	4·14 lbs.	4·98 lbs.
Water evaporated per lb. of coal	9·95 lbs.	11·96 lbs.
Thermal efficiency of boilers, with Bennis stoker and furnace	82·23 per cent.	
Thermal efficiency of boilers with Bennis, including superheat	87·46 per cent.	

688. TEST OF LANCASHIRE BOILERS WITH FORCED DRAUGHT.

Boiler 30 feet × 8 feet, grate area 35·75 sq. feet with the "Koker" stoker, and "Meldrum" steam jets.

Date of test	October 1st, 1900.
Duration of test	8 hours.
Kind of fuel used	Durham Rough Slack (sea-borne).

Total fuel burnt	12,544 lbs.
Fuel burnt per hour	1,568 lbs.
„ per sq. foot of grate, per hour .	43·86 lbs.
Average steam pressure	97 lbs.
Temperature of feed water (through economiser)	213° F.
Total water evaporated	89,000 lbs.
Water evaporated per hour	11,125 lbs.
„ „ per lb. of coal, actual .	7·11 lbs.
Total weight of clinker and ash . . .	1,890 lbs.
Percentage „ „ „ . . .	15 per cent.

Boiler 26 feet × 7 feet 6 inches, grate area 36 sq. feet with M. and C. "Sprinkler" stoker, and "Meldrum" steam jets.

Dates of tests	October 1st, 2nd, 3rd, and 4th, 1901.
Economiser	144 pipes.
Fuel	Silkstone slack.
Duration of each day's test	8 hours.
Heating surface of boiler	825 sq. feet.
„ „ economiser . . .	1,440 sq. feet.
Steam pressure, average by gauge . .	78·3 lbs.
Temperature of outside air	75·0° F.
„ gases in downtake . .	1,273·0° F.
„ gases entering economiser .	883·0° F.
„ gases leaving economiser .	434·0° F.
„ feed water entering economiser	94·0° F.
„ feed water leaving economiser	256·0° F.
Water evaporated per hour	12,390 lbs.
Water evaporated per hour, per sq. foot of heating surface of boiler	15·01 lbs.
Water evaporated per lb. of fuel, actual . .	9·38 lbs.
Water evaporated per lb. of fuel, from and at 212° F., boiler only	9·33 lbs.
Water evaporated per lb. of fuel, from and at 212° F., boiler and economiser. . .	10·86 lbs.
Fuel burned per hour	11 c. 3 q. 6 lbs. = 1,322 lbs.
Fuel burned per hour per sq. foot of grate .	36·7 lbs.
Total weight of clinker	430 lbs.

Total weight of ash . . . . . . 82 lbs.
Percentage of clinker . . . . . . 4·32 per cent.
Percentage of ash . . . . . . ·77 per cent.
Calorific value of 1 lb. of dry coal, B. Th. U. 13,534.
Percentage of total evaporation used by steam jets . . . . 3·54 per cent.

Analysis of flue gases :—

Percentage of carbonic acid, CO_2 . . . 15·7 per cent.
" " free oxygen, O . . . 3·26 per cent.
" " carbonic oxide, CO . . Nil.
" " nitrogen, N . . . . . 81·1 per cent.

—R. B. Hodgson.

689. EXPERIMENTS ON EVAPORATION IN BOILERS.

Class.	*Size.*	*Lbs. Water per lb. Coal.*	*Lbs. Coal per cub. foot Water.*
Cornish	20 H.P.	6·764	9·212
Lancashire	25 "	7·547	8·256
Galloway	35 "	9·5	6·579
Field	..	10·9	5·734

In a case where an engine was allowed to get into bad condition, with considerable leakage past valves and pistons, it appeared as if 3 lbs. of water were evaporated by 1 lb. of coal, or 21 lbs. of coal were required to evaporate 1 cub. foot of water. This was on the assumption that the engine required only the normal amount of steam.

In a clean boiler half a ton of coal will evaporate 1,000 gallons of water; if scale be allowed to collect, the evaporation will be reduced by 20 to 25 per cent.

Generally, 1 lb. best coal may be reckoned to evaporate about 10 lbs. water from and at 212° F., or say 1 lb. coal will evaporate 1 gallon of water.

690. EFFECT OF SUPERVISION OF STOKERS.

Men being aware the work was measured: 100 hours, evaporation (average from and at 212° F.) = 9·7 lbs. water per lb. fuel.

Men not being aware the work was measured: 220 hours, evaporation (average from and at 220° F.) = 9·3 lbs. water per lb. fuel.

Difference = 4½ per cent. in favour of supervision.—E. Bennis.

691. CONSUMPTION OF STEAM IN ENGINES.

Non-condensing . .	30 lbs. steam	per I.H.P.	per hour.
Compound condensing .	20 ,,	,,	,,
Triple compound . .	15 ,,	,,	,,
Willans' triple expansion	13 ,,	,,	,,
Sulzer engine with superheated steam . .	9·4 ,,	,,	,,

A well-known pump maker on being asked what steam the pumps tendered for by him would consume, said, "About 18 lbs. per I.H.P. *with the best quality Welsh coal.*" He either believed the coal made a difference, or was discounting a possible test of the pump, but steam is steam however it is generated, and the quality of the coal could make no difference in the quantity of steam required.

Some years ago Mr. Bryan Donkin found that 100 different works engines taken at random consumed an average of 150 lbs. steam per I.H.P. per hour. Sir T. Richardson found the average consumption of 31 different engines in his own works, exclusive of losses in steam pipes, to be 51 lbs. steam per I.H.P. per hour.

692. APPROXIMATE STEAM REQUIRED TO DRIVE ENGINE.

A = area of cylinder in sq. inches.

S = stroke in inches before cut off.

(Usually 50 to 75 per cent. of whole stroke.)

N = Number of revolutions per minute.

W = weight of steam in lbs. per cub. foot at working pressure (approx. = ·003 gauge pressure).

$$\text{Lbs. steam per hour} = \cdot 07\, S\, A\, W\, N.$$

693. DUTY OF ENGINES.

"Duty" dates from Lean's "Reporter," published in 1811.

s = standard of comparison in lbs. =

Cwt. any coal	112 lbs.
Bushel Welsh coal	94 ,,
,, Newcastle coal	84 ,,

w = lbs. weight coal burnt per I.H.P. per hour.

n = No. of cwts. or bushels burnt per hour.

$$\text{Duty in foot-lbs. per standard} = \frac{\text{I.H.P.} \times 33{,}000 \times 60}{n}.$$

$$\text{Duty in foot-lbs. per standard} = \frac{33{,}000 \times 60 \times s}{w}.$$

$$\text{Duty in million foot-lbs. per cwt.} = \frac{221 \cdot 76}{w}.$$

Cornish duty (prior to 1855):

g = gallons of water pumped per hour.

f = feet lift of water pumped.

n = bushels of 94 lbs. coal.

$$\text{Duty} = \frac{10\,g\,f}{n}.$$

Since 1855 Cornish duty has been reckoned upon the cwt. of 112 lbs.

694. PROGRESS IN DUTY OF ENGINES.

A.D.		
1700	Savery	5 million foot-lbs. per 100 lbs. fuel.
1770	Newcomen . . .	12 ,, ,, ,, ,,
1780	Watt	27 ,, ,, ,, ,,
1830	Cornish engine . .	87 ,, ,, ,, ,,
1890	Multiple cylinder . .	120 ,, ,, ,, ,,

But even in the last case less than one-eighth of the theoretical value of the fuel is obtained.—Prof. Thurston.

695. DUTY OF ENGINES COMPARED WITH COAL USED.

C = consumption of coal per I.H.P. in lbs.

D = duty in million lbs. raised 1 foot high by 1 cwt. of coals.

C.	D.	C.	D.
1	221·760	5	44·352
1·5	147·840	6	36·960
2	110·880	7	31·680
2·5	88·704	8	27·720
3	73·920	9	24·640
4	55·440	10	22·176

696. MODERN DUTY.

The duty of a steam engine alone is measured by the amount of steam used per hour per I.H.P.

The duty of an engine and boiler combined is measured by the coal consumed per hour per I.H.P.

The E.H.P or Brake H.P. ought, however, to be taken in preference to the I.H.P.

697. EVAPORATIVE VALUE AT DIFFERENT TEMPERATURES.

In stating the evaporative power of a boiler, it is usual to express it in terms of feed-water evaporated from 212°.

t = actual temperature of feed-water.
T = total heat of steam under given pressure.
c = cub. feet of water evaporated from t°.
C = ,, ,, ,, from 212° by same quantity of heat.
T — t = heat imparted.

$$C = c\,\frac{T - t}{966 \cdot 1}.$$

698. FEED-WATER REQUIRED FOR BOILERS.

Gallons feed-water required per hour = say nom. H.P. of boiler × 10 to allow for losses, or I.H.P. of engine × 5 for ordinary work, or × 6 for maximum work.

Boilers supplying engines pumping water against accumulator pressure and working intermittently require about 2½ gallons per working hour per effective H.P. on the average of the year.

Water tube boilers require great care in management, and particularly to keep grease out of the feed water.

699. ADVANTAGE OF HEATING FEED-WATER.

1 lb. water requires 160 units of heat to raise it from 52° F. to 212° F., and 1,000 units of heat to evaporate it from and at 212° F. If, therefore, the feed-water be raised to 212° F. by means of exhaust steam, the 160 units will be saved, and the resulting economy will be

$$\frac{160 \times 100}{1000 + 160} = 13 \cdot 8 \text{ per cent. gain.}$$

Or, putting it another way, the percentage of gain at any pressure by increasing the temperature of feed-water may be found by the following formula :

H = total heat of steam at boiler pressure.
T = temperature of feed after heating.
t = " " before "

$$\text{Gain per cent.} = \frac{100\,(T - t)}{H - t}.$$

Example:—A boiler working at 100 lbs. pressure is supplied with water at 100° F. from a condensing engine. When passed through a Green's economiser, the temperature of the feed is raised to 250° F. What is the gain per cent.?

$$H = 1216\cdot5. \qquad T = 250. \qquad t = 100.$$

Then

$$\frac{100\,(T - t)}{H - t} = \frac{100\,(250 - 100)}{1216\cdot5 - 100} = 13\cdot4 \text{ per cent. gain.}$$

Approximately there is a saving of 1 per cent. in fuel burned for every 11° F. that feed-water is heated.

700. COMPARISON OF FUEL CONSUMPTION PER E.H.P. HOUR.

Steam engine $1\frac{3}{4}$ lb. good coal.
Gas engine $1\frac{1}{3}$ lb. anthracite or 17 cub. feet illuminating gas;
Oil engine 0·82 lb. oil.
Hot air engine 2 to 8 lbs. coal.

701. FUEL ECONOMISERS.

Those by Green and Son, Limited, of Manchester and Wakefield, are best known. They act by heating the feed-water and removing lime, and consist of a series of 4-inch cast-iron pipes, about 9 feet long, in four or more rows, placed vertically in main flue and connected by top and bottom boxes. The feed-water passes through these on its way to the boiler, and the products of combustion pass on the outside in the opposite direction. An economiser receiving the gases at 650° F., and reducing their temperature to 350° F., may raise the feed-water from 100° F. to 250° or 300° F., and produce an economy of from 10 to 15 per cent. in the fuel consumption. The number of economiser pipes per boiler is four pipes per ton of coal consumed per week, or say, one pipe to every 3 I.H.P., or to every 50 lbs. water evaporated per hour.

702. CONSUMPTION OF FUEL.

The consumption of coal per I.H.P. depends upon the boiler as well as the engine, say

Non-condensing	engine	3 lbs. coal	per I.H.P.	per hour.
Simple condensing	„	2	„	„
Compound „	„	1·75	„	„
Triple compound	„	1·5	„	„
Quadruple „	„	1·25	„	„

703. POSSIBLE ECONOMY IN COAL CONSUMPTION.

Total units heat (B.Th.U.) in 1 lb. coal say 12,000, representing 12,000 × 772 foot-lbs. = 9,264,000 foot-lbs., or if all utilised a consumption of $\frac{33,000 \times 60}{9,264,000} = \cdot 213$ lb. coal per H.P. per hour, while the average consumption is actually about ten times that amount. But ·213 lbs. is less than a perfect engine would consume owing to the unavoidable loss in the condenser —e.g., let 1 lb. coal supply steam without any loss, at 300 lbs. pressure = 417° F., the condenser being 100° F., then $\frac{417 - 100}{417 + 460} = \cdot 36$ which is the ratio of the available units to the total units of heat, and $\frac{\cdot 213}{\cdot 36} = \cdot 6$ lb. coal per H.P. per hour as the maximum possible efficiency with a perfect engine under the conditions stated, or from $\frac{1}{3}$ to $\frac{1}{4}$ of what is now usually obtained.

704. REFUSE DESTRUCTORS.

1 lb. to 2 lbs. water evaporated per 1 lb. refuse.

Refuse say from $\frac{1}{10}$ to $\frac{1}{5}$ the calorific value of coal.

705. H.P. PER TON WEIGHT.

Maximum I.H.P. of engines per ton of boiler weight, including fittings, mountings and water.

Torpedo boats, Thornycroft boiler	77·8
Locomotive engines	55·4
Small high-pressure marine	12·0
Do. compound „	16·0
Do. triple compound „	20·0

In Maxim's flying machine the total weight of engines and boiler complete = 8 lbs. per E.H.P. = 280 H.P. per ton.

706. TO CALCULATE SIZE OF BOILER.

Say Cornish boiler for high-pressure engine:

d = diameter of cylinder in feet.

s = stroke in feet.

R = revolutions per minute.

r = ratio of cut-off.

p = boiler pressure, lbs. per sq. inch by gauge.

n = number of cylinders.

S = cub. feet steam required per hour, allowing 25 per cent. for contingencies.

$$S = 1\cdot 25\, d^2 \frac{\pi}{4}\, s\, r\, 2\, n\, R\, 60 = \text{say } 120\, d^2\, s\, r\, n\, R.$$

v = relative volume of steam at p pressure.

W = weight of water to be evaporated in lbs. per hour.

$$W = \frac{62\cdot 5\, S}{v}.$$

c = combustion of coal in lbs. per sq. foot fire-grate per hour, say for Cornish boiler = 12 lbs.

e = evaporation in lbs. of water from 60° F. per lb. of coal, say for Cornish boiler = 7 lbs.

$c \times e$ = lbs. water evaporated per sq. foot fire-grate per hour.

A = area of fire-grate in sq. feet.

$$A = \frac{W}{c\, e}.$$

l = length of fire-grate in feet, say 4·5 to 5·5, but must not exceed twice the width.

w = width of fire-grate in feet.

$$w = \frac{A}{l} + \cdot 166.$$

D = diameter of boiler shell = $1\cdot 75\, w$.

L = length of boiler shell = 4 D.

When w exceeds 3·25, make two Cornish boilers or one Lancashire.
For latter, D = $2\frac{1}{4}\, w$ (w being width of *one* furnace).

707. CORNISH BOILER.

Approximate heating surface in sq. yards of unit value,

$$= \frac{L\frac{\pi}{2}(d + \frac{2}{3} D)}{9} - \text{space occupied by seatings.}$$

$$= \text{say } \cdot 17\, L\, (d + \tfrac{2}{3} D).$$

Total capacity in cub. feet

$$= L \frac{\pi}{4} (D^2 - d^2) = \cdot 8 L (D^2 - d^2).$$

Approximate space for steam, remainder water,

$$= \frac{L D^2 \frac{\pi}{4}}{6} = \cdot 13 L D^2.$$

Minimum steam space = quantity required for 10 revolutions of engine. Water space 5 to 10 cub. feet per N.H.P., and 5 feet super. water surface per N.H.P.

708. COMPARISON OF CORNISH BOILER WITH I.H.P. OF ENGINE.

I.H.P. of engine = effective heating surface of boiler in sq. yards of unit value.

$\frac{1}{2}$ I.H.P. of engine = cub. yards total capacity of boiler, of which $\frac{1}{4}$ = steam, $\frac{3}{4}$ = water.

$\frac{3}{8}$ I.H.P. of engine = area of fire-grate in sq. feet.

Steam receiver may be attached to boiler, maximum size equal in diameter to flue tube, length = twice diameter.

Steam dome to allow supply pipe to take dry steam, may equal $\frac{3}{4}$ diameter of flue tube, with height = diameter of flue.

709. PROPORTIONS OF BOILERS.

Cornish Boilers (One Flue).

N.H.P.	*Length.*		*Diam.*		*Furnace.*				*Flue Diam.*	
					Diam.		*Length.*			
	ft.	*in.*	*ft.*	*in.*	*ft.*	*in.*	*ft.*	*in.*	*ft.*	*in.*
15	16	6	4	9	2	9	4	0	2	9
20	21	6	5	0	3	0	5	0	2	6
30	26	0	5	9	3	3	6	6	2	6

Lancashire Boilers (Two Flues).

	ft.	*ft.*	*in.*	*ft.*	*in.*	*ft.*	*in.*	*ft.*	*in.*
20	20	6	0	2	0	4	0	1	9
25	25	6	0	2	0	4	6	1	9
30	28	6	6	2	3	5	0	2	0
35	30	7	0	2	6	5	6	2	3

For more than 35 H.P. two or more boilers are required.

—Sir W. G. Armstrong and Co.

CORNISH BOILERS.

Horse Power	5	6	8	10	12	14	16	18	20	25	30
	ft. in.	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*
Length . .	8 6	10 0	12 0	14 0	15 0	16 0	17 6	18 0	21 0	22 6	24 0
Diameter. .	3 6	4 0	4 3	4 6	4 8	4 9	4 9	5 0	5 4	5 6	6 0
„ of flue	1 9	2 0	2 2	2 2	2 4	2 6	2 6	2 8	2 8	2 9	3 0
Height of dome	1 6	1 6	1 9	2 0	2 0	2 0	2 0	2 6	2 6	2 9	2 9
Diam. of dome	1 3	1 6	1 6	1 9	1 9	2 0	2 0	2 0	2 3	2 6	2 9

LANCASHIRE BOILERS.

Horse Power .	20	25	30	35	40	45	50	60
	ft. in.	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*	*ft. in.*
Length . . .	17 0	20 0	22 0	24 0	26 0	27 6	28 0	30 0
Diameter . .	6 0	6 0	6 3	6 6	6 6	6 9	7 0	7 0
„ of flues .	2 3	2 3	2 4	2 5	2 5	2 6	2 8	2 8
Approx. Weight in Cwts. . .	160	180	205	225	240	270	290	315

—GRANTHAM CRANK AND IRON CO.

Another rule :—

Cornish : 4 to 6 feet diameter, 1 furnace diameter = $\frac{1}{2}$ D + 3 inches, length = 2 width — 6 inches.

Lancashire : $6\frac{1}{2}$ to $8\frac{1}{2}$ feet diameter, 2 furnaces diameter = $\frac{1}{3}$ D + 6 inches, length = 2 width — 4 inches.

All boilers 4 diameters long, side flues $\frac{1}{5}$ fire-grate area, main flue $\frac{1}{8}$. Boiler measurements are taken as follows : length, inside end plates ; diameter, inside outer rings ; flue diameter, outside ring plate.

Goodeve's proportions for Lancashire boilers are :

d = diameter of one tube. Shell = $2\frac{1}{2}\,d$. Space between tubes = $\cdot 15\,d$. Space between tubes and shell = $\cdot 12\,d$. Width bottom flue = $1\frac{1}{4}\,d$.

710. STEAM BOILERS FIRED BY GAS.

Boilers to be fired by coke oven or blast furnace gases are made by Galloways, Ltd., Manchester, 10 feet diameter by 30 feet long with five flues, each 30 inches diameter.

711. FIRE-BARS.

Ordinary furnaces should not exceed 6 feet in length for Welsh coal, the bars being in two lengths; for Newcastle or other flaming coal, say 4 feet 6 inches long, with bars in one length. Dead-plate should be 9 to 15 inches wide. Fire-bars say 3 feet long, 3 inches deep in middle, $\frac{3}{4}$ inch thick at top, tapered to $\frac{3}{8}$ inch thick at bottom; bevelled one end to rest on dead-plate, to allow for expansion, and notched at other to rest on wrought-iron bearer: if notched both ends, there should be not less than 1 inch play. Chipping faces or distance pieces on bars should be made at both ends and middle. Air spaces between bars $\frac{3}{8}$ inch to $\frac{5}{8}$ inch, usually $\frac{1}{2}$ inch. The fire-grate should incline downwards towards the back, $\frac{3}{4}$ inch to 1 inch per foot. Sometimes the front set is closer pitched than the back set. Passage above bridge = one-sixth area of grate. Perforations in furnace door, $\frac{3}{8}$ inch to $\frac{1}{2}$ inch diameter; total area, from 2 to 5 sq. inches per sq. foot fire-grate.

712. BOILER SEATINGS.

With old form of wheel draught the boiler was set on a mid-feather: this is a bad arrangement. Should be set on fireclay blocks forming side walls, the resting surfaces not wider than one-thirtieth diameter of boiler, or $\frac{3}{8}$ inch per foot diameter.

Flues should be large enough for a man to pass entirely round, area should be kept as uniform as possible, corners rounded, and angles filled up.

Plain cylindrical boilers should be hung by wrought-iron brackets at intervals, riveted on, and supported on stone but not bolted down, and should be set with flash flues—i.e., the gases pass directly from furnace, over the bridge, and along bottom of boiler, to chimney. Boilers should be set with a fall of about 1 in 200, or $\frac{1}{16}$ inch per foot, towards front.

713. SIZE OF MANHOLES IN BOILERS.

12 × 8	inch	can be	entered by	small boy.
13 × 9	,,	,,	,,	lad.
14 × 10	,,	,,	,,	average man.
15 × 12	,,	,,	,,	stout man.

Manhole openings should be strengthened by welded rings or solid castings to compensate for the sectional area cut away. Oval manholes should lie across the centre line of boiler, as less material is removed from line of greatest stress.

714. BOILER TUBES.

Class of Boiler.	*Ratio, Length to Diameter.*	*Ratio, Tube Area to Grate Area.*
Multitubular boilers, with chimney draught	24 to 1	1 to 7
Locomotive boilers	120 to 1	1 to 4
Small marine boilers, with high pressure engines	33 to 1	1 to 6
Large marine boilers, with condensing engines	20 to 1	1 to 3

1 sq.-foot of fire-box is equal to 3 sq. feet tube surface ; ½ diameter should be left between the tubes for circulation and escape of steam.

Heating surface of small tubes = ⅘ of circumference, of furnace tubes = ⅔ circumference.

In multitubular boilers the stay-tubes should be spaced so as to support the whole plate, irrespective of support from other tubes.

715. FUSIBLE SAFETY PLUGS.

Boilers should have an Adams' or other fusible safety plug in the furnace crown to quench the fire before shortness of water has damaged the crown plates.

A boiler explosion must be reported to the Board of Trade within 24 hours.

716. BOILER FITTINGS AND MOUNTINGS.

These vary according to circumstances, but approximately :—

The *fittings* consist of fire bars and bearers, fire door and frame, man and mudhole doors with bridges and bolts, ashpit door and pricker bar bearer.

The *mountings* consist of safety valve, stop valve, water-gauge frames and test cocks, pressure gauge with pipe and cock, feed-check valve, blow-off cock, fusible safety plug.

717. WATER-GAUGE GLASS.

Lowest sight-level of water-gauge glass should be 3 to 4 inches above furnace crown or highest point of boiler exposed to flame.

Water heated from its point of maximum density (39° F.) to boiling point (212° F.) expands about one twenty-third of its volume.

718. TAPER OF PLUGS FOR BOILER-COCKS

For pressures up to 30 lbs. per sq. inch, a taper of 1 in 8 on each side is found to work well; but for pressures of about 100 lbs., a taper of 1 in 12 is necessary to insure tightness; say 1 in 10 minimum for pressure of 60 lbs.

719. BLOW AND SCUM.

The sediment in a boiler, and the floating impurities, should be blown out after a short period of rest, say during meal times.

When laying off for cleaning, the water should not be all blown out, or the scale will harden excessively and be more difficult to remove.

720. BLOWING-OFF TO PREVENT INCRUSTATION.

A 20-H.P. boiler working at a pressure of 70 lbs. per sq. inch (316° F.) will blow off 120 cub. feet of water in a day of 12 hours. To replace the water thus blown off, 120 cub. feet of cold water at 60° have to be introduced; and to bring it to 316°, 1,904,640 heat units, otherwise 272 lbs. of coal, are required. The remedy for this is to soften and heat the feed-water before its introduction to the boiler.

721. BOILER SCALE.

Increased quantity of fuel required to evaporate water:—

Scale	$\frac{1}{16}$	inch thick	= 15	per cent.
,,	$\frac{1}{4}$	,,	= 60	per cent.
,,	$\frac{1}{2}$	,,	=150	,,

—Prof. J. G. Rogers.

Scale	$\frac{1}{6}$	inch thick	= 16	per cent.
,,	$\frac{1}{4}$	,,	= 50	,,
,,	$\frac{1}{2}$	,,	= 150	,,

—Prof. Lewis.

Scale	$\frac{1}{16}$	inch thick	causes	25	per cent.	waste	of fuel.
,,	$\frac{1}{8}$	,,	,,	38	,,	,,	,,
,,	$\frac{1}{4}$	,,	,,	60	,,	,,	,,

—Bell's Asbestos Co.

Order of deposition of impurities as water becomes concentrated:—

1. Carbonate of lime.
2. Sulphate of lime.
3. Salts of iron, as bases or oxides, and some of these of magnesia.

4. The silica or alumina, usually with more or less of organic matter.
5. Common salt.—M. COUSTÉ.

The Bez process for de-incrustation consists in the use of "William's liquid," composed of tannin and mucilaginous materials from the tan-yard with the addition of barium chloride. It appears to be very efficient.—"PUBLIC WORKS," VOL. V., P. 127.

Soda (carbonate) is a common de-incrustant.

722. HARDNESS OF WATER.

When a water contains in solution one part by weight of lime, or other equivalent hardening salt, in 100,000 parts of water, it is said to possess 1° of hardness.

Water of less than 6° of hardness softens lead to an extent dangerous to health if used for domestic purposes.

Parts carbonate lime per 100,000 water $\times \frac{7}{10}$ = grains per imperial gallon, or degrees of hardness on Clark's scale. Carbonate of lime produces temporary hardness in water, sulphate of lime produces permanent hardness.

Water is said to be hard when it contains more than 7 grains of dissolved mineral matter per gallon. The water of London holds about 16 grains, that of Kent 24 grains. All well waters are more or less hard.

723. WELL WATER.

Proximate saline constituents in deep well water at Royal Mint, London.

	Grains per gallon.
Chloride of sodium	10·53
Sulphate of soda	13·14
Carbonate of soda	8·63
" lime	3·5
" magnesia	1·5
Silica	0·5
Organic matter	trace
Phosphoric acid	"
Iron	"
	37·8

$\frac{\text{grains per gallon}}{768}$ = percentage of impurities, but some tables, nominally percentage, are based upon parts per 1,000.

Water from the well supplying Trafalgar Square fountains contains 66 to 79 grains mineral matter per gallon, but of this 60 to 72 grains are common salt and soda, leaving only about 5½ grains of hardness due to salts of lime and magnesia.

724. THAMES WATER.

Thames water contains about ·03 per cent. impurities, or say 23 grains per gallon.

725. INCRUSTATION IN BOILERS.

If water contains 20 grains of mineral impurities per gallon, ½ cwt. of scale is precipitated and left by the water boiled away in a week of 60 hours, at the rate of 350 gallons evaporated per hour. If allowed to accumulate, this gives a coating of $\frac{3}{16}$ inch in 3 months over 250 sq. feet of plate.

If the feed-water contains 30 grains of solid matter per gallon, a 20-H.P. boiler will deposit half a pound per hour.

726. SEA-WATER.

		Parts per 1,000.
Proportion of salt in water of	open sea	32 to 38
,, ,, ,,	Red Sea	43
,, ,, ,,	Mediterranean	38
,, ,, ,,	British Channel	35·5
,, ,, ,,	Arctic Ocean	28·5
,, ,, ,,	Black Sea	21
,, ,, ,,	Baltic	6·6

—URE.

Average specific gravity of sea-water at 60° F., pure distilled water being 1:—

Faraday	1·027	Marcett	1·0277
Mallett	1·0278	Fitzroy	1·027

Salts contained in sea-water :—	*Parts per* 1,000.
Chloride of sodium	25
Muriate of magnesia	3
Sulphate of magnesia	2
Sulphate of lime	1
Others	1
Total	32

—FARADAY.

Weight of 1 cub. foot about 64·14 lbs.

727. CONSTITUENTS OF SEA-WATER BY WEIGHT.

Place	Brighton	Mediterranean.
Analyst	Schweitzer	Laurens.
Specific gravity	1·0274	1·0293
Water	964·74	959·06
Sodium chloride	27·06	27·22
Magnesium chloride	3·67	6·14
Potassium chloride	0·76	—
Magnesium bromide	0·03	—
Sulphate magnesia	2·30	7·02
Sulphate lime	1·41	0·15
Carbonate lime	0·03	0·09
Carbonate magnesia	—	0·11
Carbonic acid	—	0·20
Potash	—	0·01
	1000·00	1000·00

728. SEA-WATER BRINES IN MARINE BOILERS.

Sp. gr. at 60° F.	*Density or ratio of salt to normal quantity.*	*Boiling temp. F.°*	*Salinometer indication at 90° F.*
1·0138	·466	212·6	5·0
1·0180	·628	212·88	6·5
1·0220	·778	213·06	7·9
1·0240	·850	213·2	8·7
1·0277	1·000	213·4	10·0
1·0360	1·325	213·84	13·0
1·0500	1·822	214·6	18·0
1·0554	2·099	214·86	20·0*
1·0720	2·780	215·73	26·0
1·0830	3·234	216·3	30·0

* This is a safe point.

729. BOILERS FED WITH SALT WATER.

E = evaporation in cub. feet per hour.

D = density allowed in boiler, normal sea-water being 1·000.

d = density of feed-water.

C = cub. feet of brine to be blown out per hour.

$$C = \frac{E\,d}{D - d}.$$

To find D when $\frac{1}{3}$ feed blown off:

$$\frac{1}{3} = \frac{\frac{2}{3}\,d}{D - d}. \qquad \therefore\ D = 3\,d.$$

730. CAUSES OF PRIMING.

Changes in density of feed-water, as fresh to salt, or *vice versâ*.

Rapid extraction of steam after perfect rest; sometimes sudden starting of engines.

Feeding with muddy water, or water contaminated with sewage.

Steam-space too limited.

Defective circulation in boiler.

Hard firing.

731. CORROSION OF BOILERS.

When rivet heads between high and low water line are attacked, the corrosion has been reduced in land boilers by painting the inside of boiler between those levels with a mixture of "drippings from shafts, boiled oil and blacklead" every time the boilers are cleaned out. This was adopted at Woolwich Arsenal.

732. GREASE IN BOILER,

when carried over with the steam, is very detrimental to life and efficiency of boiler. The thin coating of grease deposited only during a ship's trial trips has been found to reduce the efficiency of the tubes as heating surfaces from 8 to 15 per cent., the mean result of many experiments being 11 per cent.

733. SAFETY VALVES FOR BOILERS

should always be in duplicate.

Area in sq. inches for each = ·004 to ·006 area of fire-grate surface, usually ·025 sq. inches per sq. foot heating surface, or ·5 sq. inches per sq. foot grate surface, irrespective of working pressure.

Actual lift of valve $= \frac{2\,d}{p}$ or $\frac{d}{36}$, but freedom must be allowed for a lift of $\frac{1}{4}d$. The lift required is less for large valves and heavy pressures than for small valves and light pressures.

Valves should be flat faced to prevent sticking, face $\frac{1}{8}$ inch to $\frac{1}{12}$ inch wide, but some locomotive engineers still prefer a conical seat with a bearing of $\frac{1}{16}$ inch.

In estimating the blow-off pressure, add $\frac{1}{16}$ inch to the actual diameter inside face of seat.

When diameter would exceed 4 inches, two or more valves must be provided.

A = effective area of heating surface, sq. feet.

H = boiler H.P. (1 cub. foot per hour evaporated from 60°).

G = grate surface, sq. feet.

$$A = 8\,(H + 2\cdot 5\,\sqrt{H}), \qquad G = \frac{H + 2\cdot 5\,\sqrt{H}}{2}, \qquad G = \frac{A}{16}.$$

$$\text{Diameter of safety valve, inches} = \sqrt{\frac{A}{27}}.$$

—Box.

$$\text{Diameter of safety valve, inches} = \sqrt{\frac{\text{grate surface, sq. feet.}}{\text{gauge pressure, lbs.}}}$$

—Tredgold.

Twin safety valves, each—

$$\text{Area} = 18\ \frac{\text{grate surface, sq. feet}}{\text{absolute pressure, lbs. sq. inch}};$$

$$\text{or area} = \frac{0\cdot 6\ \text{heating surface, sq. feet}}{\text{absolute pressure, lbs. sq. inch}}.$$

Or one as above fitted as an easing valve, and one as follows loaded to 1 lb. per sq. inch less—

$$\text{Area} = 4\ \frac{\text{grate surface, sq. feet}}{\text{absolute pressure, lbs. sq. inch}} + \text{area of guides of valve};$$

$$\text{or area} = \frac{0\cdot 133\ \text{heating surface, sq. feet}}{\text{absolute pressure, lbs. sq. inch}} + \text{ditto}.$$

If the heating surface exceeds 30 sq. feet per sq. foot of fire-grate, safety valve must be determined from heating surface.—Inst. Eng. Scot.

Heating surface in sq. feet ÷ 25 = area valve disc sq. inches.—U.S. Board of Supervisors.

·005 × lbs. water evaporated per hour = area valve disc sq. inches.—Committee of U.S. Board of Supervising Inspectors.

Area sq. inches = $\frac{4}{5}$ grate area sq. feet.—MOLESWORTH.

Orifice of safety valve (flat faced) = circf. × lift.

" " (mitred) = circf. × lift ÷ 1·414.—SOMERSCALES.

Area sq. inches up to 100 lbs. pressure = ·006 weight of water evaporated in lbs. per hour.—HASWELL.

a = effective area of opening.

d = diameter, l = lift.

Flat-faced, $a = l\,d\,\pi$.

Mitred, $a = 2\frac{1}{4}\,l\,d + 1\frac{1}{8}\,l$.

734. BOARD OF TRADE RULES FOR SAFETY VALVES.

P = boiler pressure in lbs. per sq. inch.

A = area of safety valve in sq. inches per sq. foot of fire-grate.

P	A	P	A	P	A
60	·500	110	·300	160	·214
65	·468	115	·288	165	·208
70	·441	120	·277	170	·202
75	·416	125	·267	175	·197
80	·394	130	·258	180	·192
85	·375	135	·249	185	·187
90	·357	140	·241	190	·182
95	·340	145	·234	195	·178
100	·326	150	·227	200	·174
105	·312	155	·220	205	·170

735. TO CALCULATE SAFETY-VALVE LEVERAGE.

a = area of valve in sq. inches.

p = gauge pressure in lbs. per sq. inch.

W = weight on end of lever in lbs.

w = weight of lever in lbs.

w' = weight of valve and stud in lbs.

L = distance between weight and fulcrum in inches.

g = " centre of gravity of lever and do.

l = " valve centre and do.

$$W = \left[p\,a - \left(w' + \frac{w\,g}{l}\right)\right]\frac{l}{L}. \qquad L = \left[p\,a - \left(w' + \frac{w\,g}{l}\right)\right]\frac{l}{W}.$$

$$p = \frac{\frac{wg + WL}{l} + w'}{a}. \qquad a = \frac{\frac{wg + WL}{l} + w'}{p}.$$

The lever safety valve was invented by Papin.

736. NOTES ON SPIRAL SPRINGS.

Effective number of coils = generally two less than apparent number, owing to flattening at ends for bases.

Stroke = effective number of coils × compression or extension of each coil.

Minimum pitch of spiral = diameter of steel in inches + twice compression of one coil under full load, but coils may lie close when spring is for tension only.

Diameter of coil = say 8 times diameter of steel.

Working load may stretch each coil = ½ diameter of steel composing spring.

To increase stroke, add to the number of coils.

Spring in tension is more accurate for exact work than one in compression.

Best form of section is circular, but square form is stronger, as 10 to 7.

Two or more springs may be used, one within the other.

737. SPIRAL SPRINGS.

Formula for Strength and Deflection.

E = Compression or extension of one coil in inches.

D = Diameter of coil in inches from centre to centre.

d = Diameter, or side of square, of steel composing spring in $\frac{1}{16}$ths of an inch.

W = Weight applied in lbs.

c = a constant found by experiment, which may be taken as 22 for round steel and 30 for square steel.

$$E = \frac{W D^3}{c\, d^4}.$$

738. SPIRAL SPRINGS, RANKINE'S FORMULA.

d = diameter of wire in inches.

c = coefficient of transverse elasticity of wire, say 10,500,000 to 12,000,000 for charcoal, iron, wire, and steel.

r = radius to centre of wire in coil.
n = effective number of coils.
f = greatest safe shearing stress, say 30,000.
W = any load not exceeding greatest safe load.
v = corresponding extension or compression.
W′ = greatest safe steady load.
v' = greatest safe steady extension or compression.

$$\frac{W'}{2} = \text{greatest safe sudden load.}$$

$$\frac{W}{v} = \frac{c\,d^4}{64\,n\,r^3}. \qquad W' = \frac{0\cdot196\,f\,d^3}{r}. \qquad v' = \frac{12\cdot566\,n\,f\,r^2}{c\,d}.$$

Ratio $\frac{W}{v}$ should be ascertained by direct experiment. — RANKINE'S "MACHINERY AND MILLWORK."

In two series of experiments analysed by the author, the ratio W to v was greater by 12 and 30 per cent. respectively than given by the formula, the former in tension, the latter in compression.

739. SPIRAL SPRINGS FOR SAFETY VALVES.

a = area of valve in sq. inches.
c = 11,000 for sq. steel = 8,000 for round steel.
D = diameter of spring, inches centre to centre of coil.
E = compression or extension of one coil, inches.
p = pressure lbs. per sq. inch on valve.
d = diameter of steel or side of square in inches.
d_1 = " " " in sixteenths.

$$d = \sqrt[2]{\frac{a\,p\,D}{c}}.$$

Let

$$E = \frac{D^3\,a\,p}{30\,d_1{}^4} \text{ for square steel;}$$

then

$$E = \frac{D^3\,a\,p}{22\cdot8\,d_1{}^4} \text{ for round steel.}$$

—"PRACTICAL ENGINEER."

740. INITIAL COMPRESSION OF SPRINGS FOR SAFETY VALVES

may be 40 times the lift of the valve, and assuming the lift of all sizes to be $\frac{1}{10}$ inch, the initial compression will then be 4 inches.

Or may be 1·11 diameter of valve in inches.

Or, by another rule :

$$\text{Initial compression} = \frac{80 \times d' \text{ of valve inches}}{p \text{ lbs. sq. inch.}}.$$

If lever is used, then movement of lever must be taken in calculating spring, instead of lift of valve.

741. SPRING-BALANCE SAFETY VALVES.

The levers are generally proportioned so that 1 lb. pressure per sq. inch on the valve gives 1 lb. pull on the spring, but the spring is tightened up to the blowing-off pressure, so that the actual indication is only shown when the blowing-off pressure is exceeded. The length of lever from centre of valve to fulcrum is made equal to diameter of valve, and the length from fulcrum to centre of attachment of spring is made equal to the diameter of valve multiplied by its area, all inches. The total length may be increased if the same proportion of its subdivisions be retained. Means must be provided, by ferule or otherwise, to prevent overloading.

742. TO CALCULATE SPRINGS FOR SAFETY VALVES.

Given boiler pressure and grate surface, find—

1. Diameter of valve.
2. Load required.
3. Lift of valve.
4. Initial compression of spring.
5. Assume diameter of coils.
6. Find diameter of steel.
7. Compression of each coil.
8. Effective number of coils.
9. Pitch of spiral.
10. Effective length of spring.
11. Total length.

743. FACTOR OF SAFETY, STEAM BOILERS.

Test pressure = $\frac{1}{3}$ ultimate strength.

Working pressure, if under periodical inspection, = $\frac{1}{5}$ do.

Working pressure, if not under independent inspection, = $\frac{1}{6}$ do.

In estimating ultimate strength, ample allowance to be made for defects in design or workmanship.

744. TESTING BOILERS.

Government Yards.—New boilers to be tested by hydraulic pressure to three times their working pressure. Boilers in use not to be worked more than 300 hours without being laid off for examination. To be tested periodically to twice their working pressure.

Best Private Practice.—New boilers to be tested to twice their working pressure. Boilers in use not to be worked more than 1,000 hours without being laid off for examination. To be tested after repairs to 1½ times their working pressure. If working with impure water, to be examined after 500 hours.

Locomotive Boilers.—Usually tested by hydraulic pressure to not more than 10 per cent. above working pressure, say 160 lbs. working pressure = 175 lbs. test pressure.

745. RIVETING FOR BOILERS.

In iron :—

Ring seams to be single riveted, longitudinal seams double riveted.

For equal area of plate and rivet, the linear pitch in single riveted joints and diagonal pitch in double riveted joints should be

$$= \frac{\text{sectional area of one rivet}}{\text{thickness of plate}} + \text{diameter of rivet.}$$

For same conditions, the linear pitch in double riveted joints should be

$$= \frac{2 \text{ sectional area of one rivet}}{\text{thickness of plate}} + \text{diameter of rivet,}$$

but is generally made about one-sixth less than this, to avoid straining in caulking. Double riveting should always be zigzag.

For rivets in double shear, take 1·75 times above areas.

Fairbairn estimated strength of solid plate at 50,000 lbs. per sq. inch, double riveted joint as worth 70 per cent., and single riveted joint 56 per cent. He recommended double riveted longitudinal joints 2½ inches linear pitch, 2 inches diagonal pitch [say for ¾ inch rivets and ⅜ inch plates]. For best work all edges of all plates should be planed.

746. BOILER RIVETING.

	Cup heads.	Cone heads.
¾ inch rivet	$1\frac{1}{4} \times \frac{3}{8}$	$1\frac{1}{4} \times \frac{9}{16}$
⅞ ,, ,,	$1\frac{1}{2} \times \frac{7}{16}$	$1\frac{1}{2} \times \frac{21}{32}$

	Minimum pitch.	
Finished diameter of rivet.	*Single riveting.*	*Double riveting.*
$\frac{3}{4}$ inch	$1\frac{3}{4}$ inch	2 inch.
$\frac{13}{16}$ „	2 „	$2\frac{1}{2}$ „
$\frac{7}{8}$ „	$2\frac{1}{8}$ „	$2\frac{3}{4}$ „
$\frac{15}{16}$ „	$2\frac{1}{4}$ „	3 „

Shearing stress on rivets 15,000 lbs. per sq. inch.

For $\frac{3}{8}$ inch plate the rivets should be $\frac{3}{4}$ inch diameter;
„ $\frac{7}{16}$ to $\frac{5}{8}$ „ „ „ $\frac{13}{16}$ „
„ $\frac{11}{16}$ to $\frac{3}{4}$ „ „ „ $\frac{7}{8}$ „
„ $\frac{13}{16}$ to $\frac{15}{16}$ „ „ „ $\frac{15}{16}$ „

—E. G. Hiller.

747. SMALL SCREWED STAYS OR WATER-SPACE STAYS.

p = working pressure in lbs. per sq. inch.
P = pitch of stays in inches.
a = net sectional area of stay.
s = safe stress in lbs. per sq. inch = 4,000 copper, 5,000 wrought iron, 6,000 steel.

$$a = \frac{P^2 p}{s}, \qquad P = \sqrt{\frac{s a}{p}}, \qquad p = \frac{a s}{P^2},$$

say 4 to $4\frac{1}{2}$ inches pitch for locomotive work, 6 to 8 inches for marine work. Diameter $\frac{3}{4}$ to 1 inch, generally double the thickness of plates.

748. LONG STAY BOLTS

should be strong enough to support the area assigned to them, with a factor of safety of $\frac{1}{6}$, assuming no support from the thickness of plate.

749. DIAGONAL OR GUSSET STAYS.

p = tension on longitudinal stay in lbs.
P = equivalent tension on diagonal stay in lbs.
l = length of stay in inches.
d = distance from end plate to centre of attachment to shell in inches.

$$P = \frac{p\,l}{d}.$$

750. PLATES FOR CORNISH AND LANCASHIRE BOILERS.

No plate less than $\frac{3}{8}$ inch thick, end plates $\frac{1}{2}$ inch to $\frac{11}{16}$ inch.

Each ring in one plate with longitudinal seam in steam space above flue covering.

Thickness of end plate in sixteenths + 1 = inches clear width of "breathing space"— i.e., distance between stiffeners or gussets.

For much useful information see Hiller's Notes on Land Boilers (National Boiler and General Insurance Co., Ltd., Manchester, 1s.).

751. STRENGTH OF FLAT PLATES OF WROUGHT IRON SUPPORTED BY STAYS (LLOYD'S RULES).

p = working pressure lbs. per sq. inch.
P = greatest pitch of stays in inches.
t = thickness of plate in sixteenths of an inch.
c = constant =

- 90 for plates up to $\frac{7}{16}$ inch thick held by screw stays with riveted heads.
- 100 for plates above $\frac{7}{16}$ inch do. do.
- 110 for plates up to $\frac{7}{16}$ inch thick held by screw stays and nuts.
- 120 for plates above $\frac{7}{16}$ inch do. do.
- 140 for plates held by plain stays with double nuts.
- 160 for do. do. and washers at least half thickness of plates and diameter of $\frac{2}{5}$ pitch, riveted to the plates.

In the case of front plates of boilers in steam space and exposed to direct action of heat, reduce these numbers by 20 per cent.

$$p = \frac{c\,t^2}{P^2}.$$

752. STRENGTH OF FLAT PLATES.

t = thickness in inches.
r = radius in inches if circular.
l = length in inches if rectangular.
b = breadth in inches if rectangular.
p = pressure in lbs. per sq. inch.
f = maximum stress on material lbs. per sq. inch.
a = distance centre to centre of stays in inches.
s = side of square plate in inches.

Circular plate supported at edge,

$$f = \frac{5}{6} \cdot \frac{r^2}{t^2} \cdot p.$$

Circular plate encastré,

$$f = \frac{2}{3} \cdot \frac{r^2}{t^2} \cdot p.$$

Square plate stayed,

$$f = \frac{2}{9} \cdot \frac{a^2}{t^2} \cdot p.$$

Square plate encastré,

$$f = \frac{1}{4} \cdot \frac{s^2}{t^2} \cdot p.$$

Rectangular plate encastré,

$$f = \frac{1}{2} \cdot \frac{l^4\, b^2}{l^4 + b^4\, t^2} \cdot p.$$

—UNWIN.

753. STRENGTH OF BUCKLED PLATES.

Mild steel buckled plates (dished both ways) riveted all round.

r = rise or depth of dishing.
t = thickness in inches.
s = width of side (square).
W = total safe distributed load in lbs. including own weight.

W = 24,000 $t\,r$ for any size.

For permissible concentrated load deduct distributed structural load (say 120 lbs. per foot super. for bridge floors), and take half remainder.

754. CAST IRON RECTANGULAR DOORS.

p = maximum pressure lbs. per sq. inch.
t = thickness in inches.
b = breadth inside box in inches.
l = length „ „ „

then $t = \sqrt{\frac{K\, b^2\ (15\, p)}{70{,}000}} + 0{\cdot}5$, where $K = \frac{l^4}{l^4 + b^4}$.

To be stiffened by ribs $2{\cdot}5\, t \times t$ at a pitch P found as follows :—

t = thickness in $\frac{1}{16}$ths inch.
P = pitch in inches.

$$P = \sqrt{\frac{50\, t^2}{p}}.$$

—SEATON *and* ROUNTHWAITE.

755. STRENGTH OF FLAT ENCASTRÉ CIRCULAR WROUGHT-IRON PLATES.

p = working pressure lbs. per sq. inch.
P = test " " "
P = ultimate " " "
P = bulging " " " to elastic limit.
t = thickness in inches.
T = thickness in sixteenths of an inch.
d = diameter of plate in inches.
r = radius of plate in inches.
f = maximum stress on material in lbs. per sq. inch.

$p = \frac{440\ (T + 1)}{d^2 - 12}$; —"Pract. Eng. Pocket-Book."

$P = 60{,}000\ \frac{t^2}{r^2}$ (safe load = $\frac{1}{2}$ test pressure).—H. Cherry.

P $= 1000\ \frac{t}{d}$ (12 tons elastic strength per sq. inch of material; $\frac{1}{3}$ = safe load).—D. K. Clark.

P $= \frac{f\ 3\ t^2}{2\ r^2}$ (f = 44,800 lbs.; $\frac{1}{4}$ P = safe load).—Unwin.

P $= \frac{f\ t^2}{r^2}$ (safe load $\frac{1}{4}$).—Rankine.

756. ULTIMATE STRENGTH OF BOILER-SHELL.

p = bursting pressure lbs. per sq. inch.
d = diameter of boiler in inches.
l = length of boiler in inches.
t = thickness of plates in inches.
c = ultimate strength of material, lbs. per sq. inch.

Longitudinal strength:

$$p\,d\,l = 2\,t\,l\,c \quad \therefore \quad p\,d = 2\,t\,c,$$

$$p = \frac{2\,t\,c}{d} \qquad\qquad t = \frac{p\,d}{2\,c}.$$

Transverse strength:

$$p\ \frac{d^2\,\pi}{4} = \pi\ (t + d)\ t\,c,$$

divide by $\pi\ d$, then

$$p\,\frac{d}{4} = \left(\frac{t}{d} + 1\right) t\,c;$$

but $\frac{t}{d}$ will rarely exceed ·01, and may therefore be omitted.

$$\therefore \quad p \frac{d}{4} = t\,c, \qquad p = \frac{4\,t\,c}{d}, \qquad t = \frac{p\,d}{4\,c},$$

or the transverse strength is double the longitudinal. In other words, the stress on the ring seams is only half that on the longitudinal seams.

757. HELICAL JOINTS FOR BOILERS.

Ratio of strength to longitudinal joint

$$= \frac{2}{\sqrt{(3 \cos^2 \phi + 1)}},$$

ϕ = angle of inclination from longitudinal direction.

Diagonal seams at 45° in a cylindrical boiler are 11·8 per cent. stronger than similar seams parallel with the axis.

758. COLLAPSING PRESSURE OF BOILER TUBES.

Length not exceeding 15 diameters.

Cylindrical:

$$p = 33 \cdot 61 \times \frac{(100\,k)^{2 \cdot 19}}{\mathrm{L}\,d};$$

—FAIRBAIRN.

or

$$\log p = 1 \cdot 5265 + 2 \cdot 19 \log 100\,k - \log \mathrm{L}\,d;$$

or, approximately,

$$p = \frac{800{,}000\,t^2}{\mathrm{L}\,d}.$$

Elliptical:

$$p = \frac{800{,}000\,t^2}{\mathrm{L}\,(2\,r)}, \qquad r = \text{radius of flatter curve,}$$

$$p = \frac{800{,}000\,t^2}{\mathrm{L}} \times \frac{2\,\mathrm{D}^2}{d}.$$

D d are the two diameters in inches, L the length in feet.

759. BOILERS.—COMPARISON BETWEEN BURSTING AND COLLAPSING PRESSURES.

P = internal or bursting pressure in lbs. per sq. inch.

p = external or collapsing pressure in lbs. per sq. inch.

c = ultimate strength of single riveted joint = say 30,000 lbs.

L = length of unsupported cylindrical tube in feet.

D = diameter of boiler in inches.

d = diameter of tube in inches.
T = thickness of shell plate in inches.
t = thickness of tube plate in inches.
R = ratio of tube diameter to shell diameter $= \frac{d}{D}$.

$$P = \frac{2\,T\,c}{D} = \frac{60{,}000\,T}{D}.$$

$$p = \frac{800{,}000 \cdot t^2}{L\,d}.$$

$$\frac{P}{p} = \frac{60{,}000\ T\ L\,d}{800{,}000\ t^2\ D} = \frac{T\,l\,R}{13 \cdot 3\,t^2}.$$

$$\therefore \text{ When } P = p, \text{ then } L = \frac{13 \cdot 3\,t^2}{R\,T}.$$

760. COLLAPSING PRESSURES OF FLUES.

$L' \times D''$	$\frac{1}{4}''$	$\frac{5}{16}''$	$\frac{3}{8}''$	$\frac{7}{16}''$	$\frac{1}{2}''$	
400	97	158	235	329	441	lbs. per sq. in.
500	77	126	188	263	353	,,
600	65	105	157	219	294	,,
700	55	90	134	188	252	,,
800	48	79	117	164	229	,,
900	43	70	104	146	196	,,
1000	38	63	94	131	176	,,

—MUNRO.

Length 7 diameters or over.
t = thickness in sixteenths of an inch.
D = diameter in feet.

$$p = \frac{16\,t^2}{D^2}.$$

Where length is less than 7 diameters the strength is inversely as the length, or the collapsing pressure

$$p = \frac{7\,D}{L}.$$

—W. I. ELLIS.

l = length flue inches.
t = thickness flue inches.
S = maximum stress on plate at collapse in tons per sq. inch.

$$S = 3 + \frac{169{,}000}{13{,}000 + \left(\frac{l}{t}\right)^2}.$$

—STROMEYER.

761. STIFFENING OF FURNACE TUBES.

Plain tubes with lap joints were found to be very liable to collapse, and various methods have been devised to stiffen them.

Angle iron rings were riveted on upper half with distance pieces, or thick washers of small diameter, between tube and angle iron to reduce mass of metal immediately in contact with fire. If neglected the space was apt to choke up with deposited scale. They are now principally used in repairs.

Tee iron rings were used to join the various lengths in butt joints, but the increased quantity of metal collected at one point allowed overheating and burning.

Bowling hoops, while stiffening the joint without adding thickness enough to cause overheating, were intended to render the connection more flexible to prevent grooving the end plates, but it is doubtful whether it really had any such effect.

The Adamson flanged seam (1851), with a caulking strip between the flanges, stiffened the flue, and permitted efficient caulking while not adding any thickness to the metal exposed to the fire.

Various forms of corrugation of the plates themselves have since been introduced—Fox's, Purves', etc.—and have been largely adopted.

762. FOX'S CORRUGATED FLUES.

t = thickness in inches.
D = mean diameter inches
p = working pressure in lbs. per sq. inch.

$$p = \frac{14{,}000\,t}{D}.$$ —BOARD OF TRADE.

t = thickness in sixteenths.
D = maximum diameter inches.

$$p = \frac{1234\,(t - 2)}{D}.$$ —LLOYD'S REGISTRY.

763. LOCOMOTIVE BOILER.

Pressure 130 to 150 lbs. per sq. inch.
Feet of heating surface = inches diameter piston^2 $\times$ 4.
Heating surface of firebox = $\frac{1}{12}$ to $\frac{1}{10}$ of total.
Sq. feet area fire-grate = inches diameter piston — 1.

Tubes 10 to 12 feet long, $1\frac{5}{8}$ inch to $1\frac{3}{4}$ inch internal diameter, 11 to 13 W.G., $\frac{5}{8}$ inch clearance between.

Shell plates $\frac{3}{8}$ inch to $\frac{5}{8}$ inch, $t = \frac{p\,d}{960}$ when t = thickness sixteenths, p = pressure lbs. sq. inch, d = diameter inches.

Diameter of shell = diameter piston × 3.

Smoke-box tube plate = $1\frac{1}{2}\,t$.

Side plates, outer casing, fire-box = $t + \frac{1}{16}$ inch.

Throat plate and back plate = $t + \frac{1}{8}$ inch.

Firedoor = 18 inches × 12 inches. Inner casing of fire-box, copper.

Inner tube plate, upper part = $\frac{1}{4}$ inch thicker than lower part.

Holes in tube plates = $\frac{1}{8}$ inch smaller at fire-box end and $\frac{1}{8}$ larger at smoke-box end than mean outside diameter of tubes.

Stay bolts, 4 inch pitch, $\frac{3}{4}$ inch over thread with $\frac{3}{8}$ inch plate, $\frac{7}{8}$ inch with $\frac{1}{2}$ inch plate, $\frac{15}{16}$ inch with $\frac{5}{8}$ inch plate.

Girder stays (8) in two plates 5 inches × $\frac{5}{8}$ inch to 6 inches × $\frac{3}{4}$ inch, 2 inches clearance above crown, secured by stay bolts same size as in sides of fire-box.

Steam dome = $\frac{1}{2}$ diameter of barrel, height = diameter, thickness same as shell plates, top $\frac{3}{4}$ inch to $\frac{7}{8}$ inch thick, and $7\frac{1}{2}$ inches high.

Manholes, 16 inches diameter.

Twin safety valves, each with clear passage of area = $\frac{1}{12000}$ of heating surface $\therefore$ diameter in inches = ·08 $\sqrt{\text{heating surface, square feet}}$. Conical seats, bearing $\frac{1}{16}$ inch wide.

Chimney, 13 feet 3 inches from rail level to top, smallest diameter in inches = 4 $\sqrt{\text{grate area, sq. feet}}$.

Steam pipe = $\frac{1}{16}$ area of piston.

Air space through bars = $\frac{1}{5}$ of grate area.

Fire-bars, centre depth = $\frac{1}{8}$ length; thickness, top = $\frac{1}{50}$ length; thickness, bottom = $\frac{1}{100}$ length; end depth = $\frac{3}{5}$ middle depth.

—"RAILWAY PRESS."

Tubes used in locomotive boilers vary in size from $1\frac{1}{2}$ to $2\frac{1}{2}$ inches outside diameter, and in length from about 8 to 15 feet. The thickness in general use ranges from No. 13 to No. 10 Birmingham wire gauge, or expressed decimally from 0·095 to 0·134.—F. J. COLE.

Locomotives average a consumption of 3,000 gallons of water per 100 miles run.

Section XI.

THE STEAM ENGINE.

764. EARLY ENGINES.

Savery's Engine.—A receiver was filled with steam from a boiler, the communication closed and water applied externally; condensation allowed water to rise through a bottom clack; steam again admitted above drove the water up to a higher level through an upper clack.

Newcomen's Engine.—Open-topped cylinder had a loosely fitting piston attached by rod and chain to one end of a beam, the beam pivoted at its centre and attached by chain to pump rods at other end. Steam admitted under piston at atmospheric pressure allowing weight of pump rods to lift piston and force water up from the pump. Jet of water admitted into the cylinder then caused condensation, and pressure of atmosphere forced piston down while lifting pump rods.

Watt's First Engine was a Newcomen engine with the cylinder closed on top and steam admitted instead of air, and with a separate condenser. In the old atmospheric engine increased power required increased diameter and stroke of piston; in Watt's engine increased power was obtainable by increasing the steam pressure only.

The general proportions of beam engines were :—Depth of beam = diameter of cylinder. Stroke of piston = twice diameter. Length of beam = three times the stroke. Area of beam flanges at centre = 3 sq. inches per 2,000 lbs. on piston. Indoor stroke when piston going into cylinder, outdoor stroke reverse.

765. ECONOMY OF HIGH-PRESSURE STEAM.

The pressure of steam increases in a greater ratio than its density, whence it follows that the higher the pressure to which the steam is raised, the less *proportionate* quantity of water it contains, and therefore the less fuel is consumed, since a given quantity of fuel will evaporate nearly the same weight of water at all temperatures.—Pole.

The expenditure of heat, to produce a given weight of steam at a pressure of 10 atmospheres, is only 4 per cent. more than that required at a pressure of 1 atmosphere.

Doubling the pressure in the boiler, with one-third more coal, doubles the power obtained from the engine. Thus the power obtained is greater in proportion than the extra amount of coal used to increase the pressure of steam in the boiler.

766. ADVANTAGE OF EXPANDING STEAM.

When steam is cut off at $\frac{2}{3}$ of the stroke, the power of an engine is only diminished by 7 per cent., while the consumption of steam is diminished by 33 per cent. Cut off at half stroke the power is reduced 16 per cent., and the consumption of steam 50 per cent.

767. ECONOMY OF COMPOUND ENGINES.

The economy of compound engines consists mainly in the higher pressure of steam employed permitting greater expansion, and in the subdivision of the work over two or more cylinders, limiting the range of temperature in each, and therefore the loss from condensation.

768. PROGRESS OF COMPOUND ENGINES.

1781. Hornblower patented the use of two cylinders where the steam first operated in one and then by expansion in the other also, and applied them to a single-acting pumping engine.

1782. Watt patented cutting off steam before end of stroke, but had previously adopted it.

1804. Woolf patented small and large cylinders of same stroke, pistons moving in same direction and parallel. Steam used first in small cylinder, then in large. Applied to double-acting engines with separate condenser.

1805. Earle patented the use of large and small cylinders superposed, with two pistons mounted on the same rod.

1820. Aitken and Steel built engines with three cylinders, two small and one large.

1834. Wolff patented a compound engine with two cylinders and intermediate reservoir to regulate the pressures. Also the conversion of simple engines to compound by the addition of a high or low-pressure cylinder to a low or high-pressure engine.

1839. Whitman patented the trunk piston, the steam first acting in the annular space of the cylinder, then expanding into the other end.

1841. Sims patented Earle's arrangement, with the exception that the bottom of the smaller piston was in constant communication with the top of the larger and with the condenser.

1842. Zander patented the use of a small cylinder to receive the steam, passing after slight expansion into two larger cylinders all connected with same crank shaft. The low-pressure cylinders were steam jacketed.

1844. Smith patented high and low-pressure oscillating cylinders working same crank. Perkins adopted very high pressures.

1845. McNaught patented addition of a high-pressure cylinder between the main centre and crank of beam engine.

1854. First successful application of compound cylinders to marine engines by Randolph, Elder and Co.

769. ECONOMY OF SUPERHEATING.

The thermal efficiency of an engine is increased theoretically and actually by increasing the range of temperature of the working fluid. This may be done by increasing the boiler pressure, by compounding, by superheating, and by lowering the back pressure. By increasing the amount of superheat up to the point necessary to obtain dryness at the point cut off, the thermal efficiency also rapidly increases. Beyond this point, increase in superheat is attended by increase in thermal efficiency, but at a somewhat reduced rate. Now in order to obtain dry steam at *cut off only*, a superheat of from 150° F. to 250° F. is required, depending chiefly upon the number of expansions employed. A further superheat of at least 100° F. is necessary to secure dryness at release. In compound engines it is practically impossible to sufficiently superheat the steam before entry to the high pressure cylinder to enable it to continue dry up to the point of cut off in the second cylinder. To obtain the advantages of superheating in the second and later cylinders, the practice of re-heating is sometimes resorted to, but is not economical. —W. A. F. Crawford.

With 100° to 150° F. of superheat an economy of 10 to 15 per cent. can be gained in a well designed type of engine ; with an uneconomical type the saving will be much greater. Metallic packing should be used in the high pressure cylinder and mineral oil lubricants only.

Some laboratory-tests showed that the first 40° or 50° F. of superheat account for an enormous reduction in steam consumption where the valves and pistons of the engine are at all leaky. In some of the tests, 40° F. of superheat at the pressure of 60 lbs. per sq. inch reduced the amount of leakage by 45 per cent., which was not materially further reduced by any increase in superheat up to 100° F. The beneficial effect of superheated steam in any engine may, therefore, be divided into two parts : first the effect of reducing the amount of steam passing through leaky valves or pistons, and second, the more economical behaviour of the superheated steam in the engine, irrespective of the amount of leakage. In the case of a high speed set of 200 H.P., the electrical output went up exactly 100 per cent. in relation to the weight of coal consumed, when the steam at the engine was superheated only from 20° to 30° F.—H. LEA.

770. HORSE-POWER.

Actual H.P. = 33,000 foot-lbs. per minute in all calculations, but the actual work of a horse is about 22,000 foot-lbs. per minute. One H.P. of 33,000 foot-lbs. per minute = approximately, 15 foot-tons per minute.

771. NOMINAL HORSE-POWER.

Watt's nominal H.P. for low-pressure engine (pressure 7 lbs. per sq. inch* above atmosphere).

$= \text{area sq. inches} \times 7 \times 128 \times \sqrt[3]{\text{stroke feet}} \div 33{,}000.$

$= d \text{ in inches}^2 \times \sqrt{\text{stroke feet}} \div 47.$

Boulton and Watt's N.H.P. for high-pressure engines,

$= d^2 \div 14$ ($\therefore$ 11 sq. inches piston per N.H.P.).

Do. do. for condensing engines,

$= d^2 \div 28$ ($\therefore$ 22 sq. inches piston per N.H.P.).

Bourne's N.H.P. three times that of Watt, viz., for a pressure of 21 lbs. above atmosphere.

Royal Agricultural Society's old rule was 10 circular inches of piston area and 50 lbs. pressure represent a nominal horse-power.

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* In all machinery actuated by fluid pressure, the sq. inch, which is the standard unit, introduces a needless complication. James Watt lost a good opportunity in not establishing the circular inch as the standard.

Circ. inches × ·7854 = sq. inches.
Sq. inches × 1·27324 = circ. inches.

At the present time N.H.P. is a useless commercial term, generally depending upon size of cylinder, and irrespective of pressure or speed.

Sometimes N.H.P. for non-condensing engines was $d^2 \times \sqrt[3]{\text{stroke feet}} \div 20$; for simple condensing engines $d^2 \div 30$; and for compound engines $(D^2 + d^2) \div 33$ or 30.

Admiralty N.H.P. was formerly used in classifying the power of marine engines,

$= \text{area sq. inches} \times \text{speed feet per minute} \times 7 \div 33{,}000.$

$= d \text{ in.} \times \text{speed feet per minute} \div 6{,}000.$

= about one-sixth of the indicated H.P.

Seaton's estimated H.P.

$= D^2 \text{ l.p. cylr.} \times \sqrt{p} \times \text{revolutions per minute} \times \text{stroke feet} \div 8{,}500.$

Lloyd's Committee N.H.P. (1872).

$$= \frac{1}{2}\left(\frac{D^2 \times \text{stroke feet}}{630} + F\right),$$

where F = total width of fire-grate in inches.

N. E. C. Inst. Eng. and Shp. (1877) *Normal I. H. P.*

$= \text{for screw engines } \frac{1}{100}(D^2 \times \sqrt[3]{\text{stroke feet}} + 3\,B)\sqrt[3]{p}.$

$= \text{for paddle engines } \frac{1}{160}(D^2 \times \sqrt[3]{\text{stroke feet}} + 5\,B)\sqrt[3]{p}.$

where B = the heating surface of the boilers in sq. feet, and if there are two low pressure cylinders D^2 = sum of squares of diameters.

772. INDICATED HORSE-POWER.

Indicated H.P. = mean p lbs. sq. inch from indicator diagram × area of piston (+ same for other pistons) × speed feet per minute ÷ 33,000

$$= \frac{p\,.\,a\,.\,s}{33{,}000} \quad \text{or} \quad \frac{p\,.\,l\,.\,a\,.\,n}{33{,}000},$$

p being mean pressure lbs. per square inch, l length of stroke in feet, a area of piston in sq. inches, n number of strokes per minute, s piston speed feet per minute.

When a table of areas is not available a convenient equivalent formula

$$\text{I.H.P.} = \frac{238\,p\,.\,l\,.\,d^2.\,n}{10{,}000{,}000}.$$

Rough estimate of I.H.P. of engine $= \left(\frac{\text{diam.}}{2}\right)^2$, which is correct for a mean effective pressure of 42 lbs. per sq. inch and piston velocity of 500 feet per minute.

773. NUMERICAL EXPRESSIONS OF ONE HORSE-POWER.

——	*In terms of*	*Per hour.*	*Per minute.*	*Per second.*
Elementary	Mile-tons . . .	$0{\cdot}1674 = \frac{1}{6}$	0·002 79	$46{\cdot}5 \div 10^6$
	Mile-pounds . .	$375 = \frac{3}{8} \times 10^3$	$6{\cdot}25 = \frac{100}{16}$	$0{\cdot}104 = \frac{1}{10}$
	Foot-tons . . .	884	14·73	$0{\cdot}2455 = \frac{1}{4}$
	Foot-lbs. . . .	$1{\cdot}98 \times 10^6$	33×10^3	550
	Inch-tons . . .	$10{\cdot}61 \times 10^3$	176·8	2·946
	Inch-pounds . .	$23{\cdot}76 \times 10^6$	$0{\cdot}396 \times 10^6$	0·0066
	Kilogram-mètres .	270×10^3	4500	75
Hydraulic	Water cub. feet × feet fall . . .	31 731	528·8	8·813
	Fluid gallons flow × lbs. per sq. in. pressure	85 800	1430	23·83
	Fluid cub. feet flow × lbs. per sq. in. pressure	13 750	$229\frac{1}{6}$	3·82
	Fluid cub. feet flow × inches water gauge .	$0{\cdot}38 \times 10^6$	6346	$105\frac{3}{4}$
Thermal .	Lb., F°. heat units (each 778 ft.-lbs.) .	2545	42·42	0·707
	Kilogram, C°. heat units (metric h.p.) .	632·55	10·54	0·1757
	Steam, lbs. evaporated from and at 212° F.	2·63	0·0438	$0{\cdot}73 \div 10^3$
	Fuel burnt, lbs. at 14,000 heat units per lb. .	0·1818	$3{\cdot}03 \div 10^3$	$50{\cdot}5 \div 10^6$
	Fuel burnt, lbs. at 12,725 heat units per lb. .	0·2	$3{\cdot}3 \div 10^3$	$5{\cdot}5 \div 10^6$
	Lighting gas burnt, cub. feet at 636 heat units per cub. foot	4	$\frac{1}{15}$	$\frac{1}{900}$
	Producer gas burnt, cub. feet at 127 heat units per cub. foot . . .	20	$\frac{1}{3}$	$\frac{1}{180}$
Electrical .	Joules ≡ Watt-secs ≡ 10^7 Ergs . . .	$2{\cdot}68 \times 10^6$	44 760	$746 = \frac{3}{4}$ kilo-watt-sec.

—"THE ENGINEER."

774. EQUIVALENT VALUES OF 1 H P

550 ft.-lbs. per second.
33,000 ft.-lbs. per minute.
1,980,000 ft.-lbs. per hour.
23,750,000 inch-lbs. per hour.
$\frac{1}{4}$ ft.-ton per second.
$\frac{7}{10}$ British heat-unit per second (B.Th.U.).
42$\frac{3}{4}$ British heat-units per minute (B.Th.U.).
746 watts or volt-ampères.
0·746 Board of Trade unit (B.T.U.).

775. EFFECTIVE OR BRAKE HORSE-POWER.

Effective H.P. = actual H.P. of work done, or useful effect given out from engine, either estimated, or found by friction brake, or by measurement of work performed. It is the net work done by the engine after deducting friction and loss. The effective H.P. of any engine, compared with the steam used, is the measure of its efficiency or economy.

Brake H.P. = the power given off from the crank shaft, through the fly-wheel, or a pulley, to an absorption or transmission dynamometer. It may be conveniently ascertained by the electrical output of a dynamo of known efficiency.

To ascertain Brake H.P. by friction dynamometer W = weight lbs., L = leverage feet, R = revolutions per minute.

$$\text{B.H.P.} = \frac{W\,L\,2\,\pi\,R}{33{,}000}.$$

An "output of 100 k.w." = 100 kilowatts per hour = 100 × 1·34 H.P. = 134 E.H.P. or B.H.P.

Drawbar H.P. is the effective H.P. of a locomotive engine.

776. FRENCH HORSE-POWER.

French H.P. (*Force de cheval* or *Cheval-vapeur*).

1 kilogrammètre = 1 kilogramme (2·205 lbs.) raised 1 mètre (3·281 feet) = 7·2346 foot-lbs.

1 kilogrammètre per second = 434 foot-lbs. per minute.

75 kilogrammètres per second, of 4,500 kilogrammètres per minute = 32,550 foot-lbs. per minute, or about $\frac{1}{70}$ less than a British H.P., hence "*Chevaux de* 75 *kilogs*," or metric horse-power.

French H.P. × ·9863 = British H.P.

British H.P. × 1·014 = French H.P.

French I.H.P. = Cheval indiqué.

777. FRENCH DYNAMIC UNIT.

1 cub. mètre of water (2,208 lbs.) raised 1 mètre (3·281 feet) high = 7244·45 foot-lbs.

778. MODULUS OF STEAM ENGINE.

The modulus of a steam engine, or coefficient of mechanical efficiency, or simply the mechanical efficiency, is found by dividing the effective or brake H.P. by the indicated H.P. It varies generally with the size of engine from ·75 to ·85, the larger engines having the higher efficiency.

$$\text{Steam efficiency} = \frac{\text{mean pressure}}{\text{terminal pressure}}.$$

779. DYNAMOMETERS

used for ascertaining Brake H.P. are of two chief varieties.

(*a*) Absorption dynamometers—e.g., belt, cord, Prony brake, etc.

(*b*) Transmission dynamometers—e.g., Ayrton and Perry, Alteneck or Siemens, Froude or Thornycroft.

Where the load varies from time to time it is generally economical to put down small direct-driving engines so that one or more may be used as required. Any permanent increase of load should be met by adding more engines of identical pattern. The size adopted in the first instance would depend upon the circumstances, but no future variation should be permitted in the pattern except for very substantial reasons.

780. DE PAMBOUR'S PRINCIPLES.

1. When the engine has attained a uniform motion, the work done by the steam in the cylinder is equal to the work which is due to the total resistance.

2. The steam which is generated in the boiler is equal to that expended in the cylinder.

781. STEAM WORKED EXPANSIVELY.

p = absolute initial pressure.

s = stroke.

m = mean pressure.

n = ratio of whole stroke to stroke before cut-off.

When cut off at any part of stroke, as $\frac{1}{n}$; then its

Efficiency $= 1 + \text{hyp. log } n.$

Mean pressure $= \frac{1}{n} p \,(1 + \text{hyp. log } n).$

Initial pressure $= \dfrac{m\,n}{1 + \text{hyp. log } n}.$

Pressure at any point in the expansion curve at x distance from commencement of stroke $= \dfrac{\frac{1}{n} s}{x} p.$

Advantage of working expansively $= 1 + \text{hyp. log } n$ to 1, or $100 \times$ hyp. log n per cent. gain.

Distance travelled to attain maximum velocity

$$= \frac{p\,s}{m\,n} \text{ or } \frac{s}{1 + \text{hyp. log } n}.$$

Cut-off for maximum efficiency (Pole)

$$= \frac{\dfrac{24{,}250}{p} + 65}{\dfrac{24{,}250}{\text{useless resistances}} + 65}.$$

Terminal pressure $= \frac{p}{n}$, or $\frac{1}{n}$th of p, usually from 8 to 12 lbs. in condensing engines and 18 to 22 in non-condensing engines.

Units of work per sq. inch of piston in one stroke

$$= p \frac{s}{n} (1 + \text{hyp. log } n).$$

All pressures are measured from perfect vacuum, the atmospheric line is a variable element.

Above formulæ assume theoretically perfect indicator diagrams and expansion according to Boyle and Marriotte's law.

In ordinary land engines the mean pressure found above must be multiplied by 0·8 to give the mean pressure from an indicator diagram.

Clearance spaces each end $= \frac{1}{15}$ to $\frac{1}{20}$ cylinder capacity, but often more in the high pressure cylinder of compound engines.

782. TABLE OF HYPERBOLIC LOGARITHMS.

Cut-off.	*No.*	*Hyp. Log.*	*Cut-off.*	*No.*	*Hyp. Log.*
$\frac{5}{6}$	1·2	·1823215	$\frac{1}{3}$	3·0	1·0986124
$\frac{4}{5}$	1·25	·2231435	$\frac{1}{4}$	4·0	1·3862943
$\frac{3}{4}$	1·33′	·2851788	$\frac{1}{5}$	5·0	1·6094379
$\frac{2}{3}$	1·5	·4054652	$\frac{1}{6}$	6·0	1·7917595
$\frac{3}{5}$	1·66′	·5068176	$\frac{1}{7}$	7·0	1·9459100
$\frac{1}{2}$	2·0	·6931472	$\frac{1}{8}$	8·0	2·0794414
$\frac{2}{5}$	2·5	·9162907	$\frac{1}{9}$	9·0	2·1972245
$\frac{3}{8}$	2·66′	·9783260	$\frac{1}{10}$	10·0	2·3025851

Com. log × 2·3025851 = Hyp. log.

Hyp. log × 0·434294819 = Com. log.

783. ADVANTAGE OF CONDENSATION IN STEAM ENGINES.

(1) At various points of cut-off.

Absolute steam pressure 120 lbs. Back pressure in condenser 2 lbs.

Cut-off.	*Wt. of steam used in cylr. of capacity* 1.	*Useful effect non-condensing.*	*Useful effect condensing.*	*Percentage of gain.*
1	120	105	118	12·5
$\frac{1}{2}$	60	86½	99½	15
$\frac{1}{4}$	30	56½	69½	23
$\frac{1}{8}$	15	31	44	42

(2) At various pressures.

Absolute steam pressure lbs. per sq. in.	*Pressure of steam above atmosphere.*	*Net work obtained without condensation.*	*Net work obtained with condensation.*	*Percentage of gain.*
120	105	105	118	12·5
60	45	45	58	28·8
30	15	15	28	86·6
15	3	3	16	433·0

—BRAMWELL.

784. MEAN PRESSURE WITHOUT LOGARITHMS OR SCALES.

To find mean pressure of theoretical indicator diagram (say at Exam.) without logarithms or scales. Example :—Cut-off at $\frac{6}{10}$ of stroke, then intermediate pressures will be as follows :—

	By Inside Rectangles.	By Outside Rectangles.
At 1st tenth . . .	1·	1·
2nd ,, . . .	1·	1·
3rd ,, . . .	1·	1·
4th ,, . . .	1·	1·
5th ,, . .	1·	1·
6th ,, . . .	1·	1·
7th ,, $= \frac{6}{7}$ . .	·857	1·
8th ,, $= \frac{6}{8}$ . .	·75	·857
9th ,, $= \frac{6}{9}$ . .	·667	·75
10th ,, $= \frac{6}{10}$ . .	·6	·667
	8·874	9·274

$$\begin{array}{r} 8\cdot874 \\ 9\cdot274 \\ 2)\overline{18\cdot148} \\ 10)\overline{9\cdot074} \\ \overline{\cdot9074} \end{array} \quad \therefore \text{ mean} = 0\cdot9074 \text{ of boiler pressure.}$$

Checking this by hyp. log the multiplier = 0·9041, so that the error is less than $\frac{1}{2}$ of 1 per cent.

785. DIAGRAM FACTOR

is the ratio of the actual diagram area to that of a theoretical diagram, it varies usuallv from 0·7 to 0·9.

786. ORDINATES TO HYPERBOLIC EXPANSION CURVES.

Initial pressure $= p$.

Cut-off at $\frac{2}{10}$, then ordinate at $\frac{3}{10} = \frac{2}{3}\,p$.

,, ,, $\frac{4}{10} = \frac{2}{4}\,p$.

,, ,, $\frac{10}{10} = \frac{2}{10}\,p$, and so on.

Cut-off at $\frac{1}{4}$ $(= \frac{5}{20})$, then ordinate at $\frac{3}{10}$ $(= \frac{6}{20}) = \frac{5}{6}\,p$.

,, ,, $\frac{4}{10}$ $(= \frac{8}{20}) = \frac{5}{8}\,p$.

,, ,, $\frac{5}{10}$ $(= \frac{10}{20}) = \frac{5}{10}\,p$, and so on.

787. SIMPSON'S RULE.

For area of any irregular figure.

Divide area into any even number of parts by odd number of lines or ordinates. Take the sum of the extreme ordinates, four times the sum of the even ordinates, and twice the sum of the odd ordinates (omitting the first and the last ordinates), multiply the total by one-third of the distance between ordinates, this equals the area.

For indicator diagram, divide length into ten equal parts by eleven lines, measure effective length of each, and number them. Then

(1st + 11th) + 4 (2nd + 4th + 6th + 8th + 10th) + 2 (3rd + 5th + 7th + 9th) ÷ 30 = mean pressure.

788. RESISTANCE IN STEAM ENGINES.

1. The load or useful work.
2. The friction of the unloaded engine = 1 to 3 lbs. per sq. inch.
3. Additional friction due to the load = say $\frac{1}{7}$ of mean pressure.
4. Back pressure = 4 to 5 lbs. absolute—i.e., above perfect vacuum for condensing engines, or 15 to 18 lbs. absolute for non-condensing engines.

The coefficient or modulus will then be ·6 to ·75.

Generally the friction may be taken as 10 per cent. of the H.P. in a non-condensing engine, and 18 per cent. of the H.P. in a condensing engine.

The average friction of a stationary engine with shafting is considered to be = 3 lbs. per sq. inch of the boiler pressure; and of a marine engine, $1\frac{1}{2}$ lbs. per sq. inch.

Water in cylinder of coupled engines, cranks 90°, is liable to cause fracture owing to the slide shutting in the water and the other piston acting as a Stanhope lever.

789. MEAN EFFECTIVE PRESSURE, COMPOUND ENGINE.

m = mean effective pressure, supposing all work done in low pressure cylinder.

p = boiler pressure by gauge.

$$m = \sqrt{6p}.$$

—J. MACFARLANE GRAY.

790. PISTON CONSTANT FOR INDICATOR DIAGRAMS.

When several are to be worked out the "piston constant" will be found useful, thus:

piston constant $= \dfrac{\text{area sq. inches} \times \text{feet stroke}}{33{,}000}$, multiplied by 2 if same diagram is to answer for both ends of stroke, and by 2 again if to answer for 2 cylinders. Then,

I.H.P. = constant × mean pressure lbs. per sq. inch × revolutions per min.

In finding the effective pressure on the piston at any part of stroke, take steam pressure $\times$ area one side — back pressure $\times$ area other side. In an ordinary indicator diagram from one end of cylinder, the steam line and exhaust line belong to the same side of piston, and would therefore only give the effective pressure approximately.

791. CRANK AND PISTON NOTES.

a = Length of connecting rod.
b = Length of crank.
x = Distance of piston from end of stroke furthest from crank, when point of maximum leverage is reached.
x' = Distance as before, when crank has made quarter revolution from dead centre.

$x = (a + b) - \sqrt{a^2 + b^2}$. $\qquad x' = (a + b) - \sqrt{a^2 - b^2}$.

These values divided respectively by $2\,b$ will give the proportion of stroke where these points occur.

All the distances are measured from the end of stroke furthest from crank.

p = pressure on piston (total).
p_1 = thrust in connecting rod.
θ = angle of connecting rod with horizontal.
p_2 = pressure on guide bar.
p_3 = turning effort on crank.
ϕ = angle of crank with horizontal, then*

$$\sin\phi = \sin\theta\,\frac{a}{b}$$

$$\sin\theta = \sin\phi\,\frac{b}{a}$$

$$p_1 = p \times \sec\theta$$
$$p_2 = p \times \tan\theta$$

Between tangential points in 1st and 4th quadrants,

$$p_3 = p \times \cos\theta \times \sin(\phi + \theta).$$

Beyond tangential points through 2nd and 3rd quadrants,

$$p_3 = p \times \sec\theta \times \sin(\phi - \theta).$$

* For a simple introduction to Trigonometry, see the author's "Practical Trigonometry," 2nd edition (Whittaker & Co., 2s. 6d. net).

d = distance as before for any angle of crank

$$= b + a - (b \cos \phi + \sqrt{a^2 - b^2 \sin^2 \phi}.$$

$$\cos \theta = \sqrt{1 - \frac{b^2}{a^2} \sin^2 \phi}.$$

$$\frac{\text{Velocity of piston}}{\text{Velocity of crank pin}} = \frac{\sin(\phi + \theta)}{\cos \theta}.$$

Approximately the maximum pressure on guide bar may be found thus:

Length con. rod. : length crank : : pressure on piston : pressure on guide bar.

Pressure on guide bar should be limited to from 100 to 400 lbs. per sq. inch.

Piston speed = twice stroke feet × revolutions per minute.

In an engine running one way round only one half of the crank pin is in the direct line of thrust, so that there is an arc of about 80° of the crank pin surface which is never used. A worn crank pin could therefore be turned through half a revolution to bring practically a new surface into use.

792. INSTANTANEOUS CENTRE.

The instantaneous centre of a connecting rod is found by producing the centre line of crank through end of connecting rod to meet a line drawn through crosshead pin at right angles to axis of piston rod.

The velocity of crank pin is to velocity of piston as radius from instantaneous centre to crank pin is to radius from instantaneous centre to crosshead pin.

793. RAMSEY CRANK MECHANISM.

Ordinary reciprocating engines have the axis of crank shaft at the same level as the axis of cylinder, and single acting engines have the connecting rod usually five cranks long. The Ramsey crank mechanism, applicable to all single acting engines, and particularly internal combustion engines, has the crank shaft set off from the axis of cylinder a distance equal to the length of crank, and the connecting rod approximately $3\frac{3}{8}$ cranks long. The direction of action of the piston during the power stroke is tangent to the circle of the crank motion—full power acting on full leverage. The proportion of the crank circle during which effective pressure is applied by the piston to the crank is increased to 192°, and the piston stroke is increased by $5\frac{1}{4}$ per cent. over the crank circle. The return stroke begins slowly, and allows the burnt gases to exhaust with less back pressure, the remainder of the return stroke being very rapid.

794. PARALLEL MOTIONS.

The figure of 8 produced by the completed movement of a simple Watt parallel motion (1784) is called a lemniscate curve, half the total height is utilised for guiding the piston rod, and being approximately straight is called "lemniscate straight line motion." The later adaptations were of the form of a jointed parallelogram on the principle of the pantograph, the three points of pin at end of piston rod, pin at end of air pump, and main centre of beam being in one straight line. The Peaucellier exact straight line motion (1864) is a combination of seven bars, four equal bars forming a diamond-shaped cell, two long bars from the sides of cell joining at a pole, and a short bar joining end of cell to another pole. The two poles are fixed centres, and when the combination is moved on these centres the extremity of the cell traces a true straight line.

795. PISTON SPEEDS FOR VARIOUS ENGINES.

Ordinary commercial, horizontal type	250 to 450	feet per minute.
High-class Corliss mill engines .	up to 700	„
Single acting high speed . . .	350 to 700	„
Slow pumping engines . . .	125	„
Beam pumping engines . . .	230 to 360	„
High speed electric light engine .	800 to 1200	„
Marine engines	600 to 1000	„
Torpedo boat engines . . .	up to 1200	„
Locomotives frequently over . .	1000	„

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796. MODERN ENGINE SPEED.

	Revns. per min.	*Piston speed. ft. per min.*
Torpedo boat destroyer	400	1200
First class cruiser	125	1000
Ocean going mail boat	86	950
Cross Channel mail boat . . .	160	880
First-class cargo boat	84	750
Cargo boat	66	530

—W. A. F. CRAWFORD.

797. SLIDE VALVES, EXPLANATION OF TERMS.

Travel of slide valve is twice throw of eccentric where connected direct, and should not be less than 2 (outside lap + steam port).

Lap, outside lap, steam lap, or *cover* is the distance the slide reaches beyond outer edge of steam port when in centre of travel.

Inside lap, or *exhaust lap,* is the distance from inside edge of steam port to edge of slide port, or space in slide, when in centre of its travel.

Negative, or *minus, inside lap* occurs when both steam ports are open to the exhaust in the central position, sometimes found in quick-running engines.

Lead is the amount of opening of steam port by slide when piston is at commencement of stroke, due to eccentric being set in advance of crank. Generally equal each end and varying from $\frac{1}{32}$ inch to $\frac{1}{4}$ inch. Inverted engines usually set with more lead at bottom than top, to allow for dropping of slide due to wear, and to equalise the diagrams.

Angular advance of eccentric is lap + lead, set off from centre, along centre line of crank, and transferred perpendicularly on to circumference of throw circle.

Width of face of slide valve = width of steam port + steam lap + exhaust lap.

Amount of opening of port, steam or exhaust, = half travel — lap.

Cushioning of steam is not merely the compression of the exhaust steam after the port is closed; it is the balance of this compression over the pressure on the other side of the piston.

798. AREA OF STEAM PORTS.

A = area of piston.

a = area of steam port.

b = area of exhaust port.

v = velocity of piston feet per minute.

$$a = \frac{v\,A}{4800}.$$

$$b = \frac{v\,A}{6000} + \text{part covered.}$$

Baker (Weale's Series, "Steam Engine," p. 71): Watt's condensing engines, a = 1 sq. inch per N.H.P.

Bourne ("Hbk. Steam Engine," p. 313): a = 1 sq. inch per N.H.P. or $\frac{1}{25}$ A. Also a = diam. cylr.^{2} × v × ·032 ÷ 140.

Burgh ("Slide Valve," p. 10): high press. engine, a = H.P. × ·6 or ·5. Low press. engine, a = H.P. × 1 · to ·75.

Rankine ("Steam Engine," p. 414): $a = \frac{1}{12}$ A for v 200 to 240.

Sir W. G. A. & Co.: $a = \frac{1}{24}$ A, b = 2 a, v = 200.

Winton, $a = \frac{1}{19}$ A, $b = \frac{12}{8}$ A.

Shapton, a = diam. cylr.$^2 \times \cdot 038$.

Adams, $a = \frac{1}{15}$ to $\frac{1}{12}$ A, $b = 1\frac{3}{4}\, a$, v = 350 to 500.

Rigg, $a = \text{A} \times v \div 6000$.

Barr, $a = \text{A} \times v \div 5500$, $v = 600$.

Thickness of bar between ports = ·5 steam port, minimum 1 inch, or = thickness of metal in cylinder.

Length of steam port should be in proportion to diameter of cylinder, say ·6 to ·8 cylinder diameter. To shorten travel, increase length of port.

Locomotives and other fast running engines should have the lap a little over $\frac{1}{5}$ of the travel, and lead $\frac{1}{14}$ travel. (BOURNE.)

Ordinary engines with simple D slide, lap = $\frac{1}{4}$ travel and cut-off at $\frac{2}{3}$ stroke.

799. SLIDE VALVE NOTES.

r = ratio of cut-off in cylinder.
T = travel of slide.
L = lap ,,
l = lead ,,
w = width of steam port.
x = fraction of stroke to be completed when cut-off occurs.
s = stroke of piston.
P = steam port opening.

$$\text{T} = 2\,(w + \text{L}).$$

$$\text{L} = (\tfrac{1}{2}\,\text{T}\,\sqrt{1 - r}) - \tfrac{1}{2}\,l.$$

$$r = 1 - \left(\frac{2\,\text{L} + l}{\text{T}}\right)^2.$$

If $\dfrac{\sqrt{x}}{1 - \sqrt{x}} = \text{C}$ then $\text{L} = \text{C}\left(\text{P} - \dfrac{l}{2}\right) - \dfrac{l}{2}$.

Effect of obliquity of connecting rod is to make cut-off later on the outdoor stroke and earlier on the indoor stroke, or, in other words, to draw all points of an indicator diagram nearer the crank or stuffing-box end of a cylinder.

800. POINT OF CUT-OFF

when slide is set with equal lead.

A = distance travelled by piston before cut-off.
B = remainder of stroke.
C = distance centre crosshead to centre crank shaft at point of cut-off $= a + b -$ A.

Cut-off on outdoor stroke = A.

Do. indoor „ $= A - \frac{A \times B}{C}$.

To equalise cut-off, shift slide.

801. NUMBER OF EXPANSIONS.

The steam is usually expanded in

Simple condensing engines	from	3	to	5	times.
Two cylinder compounds	„	7	„	9	„
Triple compounds	„	12	„	15	„
Quadruple compounds	„	16	„	18	„

The total expansion is found by dividing the capacity ($d^2\, l$) of the low-pressure cylinder up to point of release by the capacity ($d^2\, l$) of high pressure cylinder up to point of cut-off. Intermediate cylinders do not affect the ratio. Thus the power of any compound or multiple expansion engine is the same as if all the work were done in the low pressure cylinder at the same total rate of expansion.

802. TRIPLE EXPANSION ENGINES.

Theoretical terminal pressure should be about 12½ lbs. absolute.

Cut-off in H.P. (high pressure) cylinder should be about half stroke.

Ratio of capacity $\frac{\text{L.P.}}{\text{H.P.}}$ = abs. br. press. × ·04 = say 6½.

Do. $\frac{\text{I.P.}}{\text{H.P.}}$ = say 2½.

Do. $\frac{\text{L.P.}}{\text{I.P.}}$ = not less than 2½.

Approximately, in a triple expansion engine 13 lbs. water per hour converted into steam of 175 to 200 lbs. pressure will give 1 horse-power.

803. CYLINDER RATIOS.

Two-cylinder compounds :—

p = initial pressure in cylinder, say 5 lbs. less than boiler.

n = number of tenths up to cut-off in high-pressure cylinder.

$$r = \frac{p\, n}{105}.$$

Common rule :—Diam. low press. cylr. = 2 ins. less than twice diam. high press. cylr.

Another rule :—

$$\frac{\text{Vol. L.P. cylr.}}{\text{Vol. H.P. cylr.}} = \frac{4\,p + 40}{100}.$$

Triple compounds :—

p = gauge pressure, from say 125 to 175.

r for small = 1

„ intermediate = ·015 p + ·3

„ large = ·075 p — 4·75.

Another rule :—

At p = 120 lbs. ratios are H.P. 1, I.P. 2·5, L.P. 5

„ = 180 „ „ „ 1, „ 3, „ 8

Quadruple compounds :—

p = gauge pressure, from say 175 to 250.

Multiple cylinder compounds :—

n = no. of cylinders.

r = ratio of successive cylinders.

E = total no. of expansions.

$$r = \sqrt[n]{E}$$

Find size of largest cylinder as a single engine, then

$$\text{3rd} = \frac{\text{4th}}{r},\ \text{2nd} = \frac{\text{3rd}}{r},\ \text{1st} = \frac{\text{2nd}}{r}.$$

804. AVERAGE RESULTS OF MARINE ENGINES, 1872—1901.

Boilers, Engines, and Coal.	*Average Results.*			
	1872.	1881.	1891.	1901.
Boiler pressure . . . lb. per sq. in.	52·4	77·4	158·5	197
Heating surface per sq. foot of grate . . . . . sq. ft.	—	30·4	31	38 & 43*
Heating surface per I.H.P.. . sq. ft.	4·41	3·917	3·275	3·0
Coal per sq. ft. of grate . . lb.	—	13·8	15	18 & 28*
Revolutions per minute . . revs.	55·67	59·76	63·75	87
Piston Speed . . feet per minute	376	467	529	654
Coal per I.H.P. per hour . . lb.	2·11	1·83	1·52	1·48
Average consumption on prolonged sea voyage . . . . lb.	—	2	1·75	1·55

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* Natural and forced draught respectively.

805. WORKING STEAM PRESSURES

usually adopted in different types of engines.

Type of engine.	*Steam pressure lbs. per sq. in.*	
	Non-condensing.	*Condensing.*
Simple	80—90	60—80
Compound	100—130	80—110
Triple expansion	160—180	140—160
Quadruple expansion . . .	200—250	200
Semi-portable compound . .	140	—
Locomotive simple . . .	150	—
Locomotive compound . . .	180	—

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The progressive increase of working pressures for marine engines may be shown as follows:—

Prior to	1850, simple engine . . .	5	lbs. per sq. in.
About	1850, „ „	20	„
„	1866, compound engine . . .	60	„
„	1872, triple expansion engine .	150	„
„	1882, quadruple „ „ .	200	„
„	1896, quintuple „ „ .	250	„

806. ADVANTAGE OF STEAM JACKETING.

Triple expansion engine indicating 175 H.P. at 55 revolutions per minute with 120 lbs. boiler pressure.

High-pressure	jacket alone gave		1¼	per cent. gain.
Intermediate	„	„	2¼	„
Low-pressure	„	„	6½	„

Best result with boiler steam in all three jackets was 14·1 lbs. steam per I.H.P. per hour (including jacket steam) with 146 lbs. boiler pressure and 61 revolutions per minute.—B. DONKIN.

Steam jacketing prevents much of the condensation in the cylinder, which takes place in the jacket instead. Its usefulness is greater as the expansion is greater, owing to the increased range of temperature.

807. LINK MOTIONS.

Stephenson's.—Link curved, concave side towards eccentrics, shifted to vary position of motion block, block moving in direct line with slide rod, lead increasing towards midgear with open rods and decreasing with crossed rods. Length of link three times travel of valve.

Gooch's.—Link curved, concave side towards spindle, maintained in central position by rod swinging on a stud, motion block shifted in link by radius rod connected to valve spindle, lead constant.

Allan's.—Link straight, link and motion block moved in opposite directions by rocking shaft, lead increasing towards mid-gear with open rods, and decreasing with crossed rods.

Joy's.—Link curved, moving on a fixed pivot, concave side towards valve, no eccentrics; pendulum rod attached to centre of connecting rod at one end and to radius bar at other end; another bar pivoted on motion block, one end connected to valve rod and other end to pendulum rod; link moved on centre to alter valve; distribution of steam symmetrical.

808. RADIAL VALVE GEARS.

A radial valve gear has been defined as one in which the motion is taken from some point of a vibrating link, a second point of which moves in a straight line or open curve, but the characteristic feature is rather in the mode of reversing. The link motions reverse by moving the free end of the valve radius rod, with its sliding piece, to one side or the other of the centre of an oscillating link. The radial gears reverse by altering the path of the fulcrum point in the valve lever or vibrating link.

809. CORLISS VALVE GEAR.

This consists of separate steam and exhaust valves for each end of cylinder. They are of cylindrical form, turning through small arcs by the partial rotation of a disc worked by a rod from an eccentric on the crank shaft. The steam valves are closed by a spring after being released by trip gear, all the other motions are due to the eccentric.

810. WATT'S GOVERNOR,

commonly called a pendulum governor, usually makes 30 revolutions per minute; then $h = 39{\cdot}1$ inches = length of London seconds pendulum. Whole arm 3, upper portion 2, link 2, variation of velocity 10 per cent. Weight of ball $= 3{\cdot}174 \times$ resistance of throttle valve connections. Generally:

w = weight required to open throttle valve in lbs.

W = weight of one ball of governor in lbs. + ½ weight of arm.

L = whole length centre of suspension to centre of ball in inches.

l = length from centre of suspension to centre of attachment of link in inches.

h = height from centre line of balls when rotating at given speed to centre of suspension in inches.

R = revolutions per minute of governor.

$$h = \frac{35225}{R^2}, \qquad W = \frac{100\, l\, w}{10\, L}, \qquad R = \frac{187 \cdot 68}{\sqrt{h}}.$$

Weight of cast-iron ball $= \frac{d^3}{7 \cdot 27}$. Diam. $= \sqrt[3]{7 \cdot 27\, W}$. —HANN.

811. EFFICIENCY OF GOVERNOR.

In the Watt governor, the virtual point of suspension does not coincide with the actual points, owing to the pendulum arms being pivoted to projecting lugs at top of spindle. For accurate work the height is measured from the plane of rotation of the balls to the intersection of the direction of the arms and therefore, in the ordinary construction, it reduces as the balls fly out, whence the efficiency also reduces. The sensitiveness is increased by increasing the length of the arms or reducing the apex angle.

In Head's governor the arms are continued through the spindle and pivoted beyond, so that as the balls fly out the height is slightly increased and also the efficiency.

The position of the balls of a Watt governor, or the height at which they will run, or "floating height," depends entirely upon the number of revolutions per minute that it makes, and is independent of the length of arms or weight of balls.

812. CENTRE-WEIGHT GOVERNORS.

In governors with a sliding centre-weight or counterpoise, of which the Porter governor is the type, much greater sensitiveness is obtained.

C = weight of centre-weight in lbs. = 20 to 30 times W (the greater the difference the greater the number of revolutions required, and the more sensitive the governor).

$$h = \frac{35225}{R} \times \frac{C + w + W}{W} \qquad R = \sqrt{\frac{35225}{h} \times \frac{C + w + W}{W}}$$

$$C = W\left(\frac{R^2\, h}{35225} - 1\right) \qquad W = \frac{35225\, C}{R^2\, h - 35225}.$$

813. PORTER GOVERNOR.

The Porter governor has a sliding load on the central spindle which is raised when the balls fly out, the balls are placed at the junctions of two links whose other ends are pivoted at the head and sleeve. When the links are equal in length the sliding load rises twice as fast as the balls.

$$H = \frac{35225}{R^2}\left(1 + \frac{W}{w}\right)$$

when H = height in inches from plane of balls to point of suspension,

R = revolutions per minute,

w = weight of one ball in lbs.,

W = weight of sliding load in lbs.

Or with given length of links :—

$$R = \sqrt{\frac{35225}{H}\left(1 + \frac{W}{w}\right)}$$

When revolutions vary :—

R = greater number of revolutions per minute.

r = lesser " " " "

E = working effort in lbs.

$$E = (w + W)\,\frac{R^2 - r^2}{r^2}$$

h = rise or fall of balls in inches due to change of velocity,

$$h = 2936\,\frac{w + W}{w}\left(\frac{R^2 - r^2}{R^2 \times r^2}\right)$$

Travel of sleeve due to change of velocity when arms are equal $= 2\,h$.

814. POWER OF GOVERNORS.

The "power" of a governor is its capacity for overcoming external resistances. The greater the power, the greater the external resistance it will overcome with a given alteration in speed. In a simple Watt governor, crossed arm, and others of a similar type the energy in foot-lbs. = weight of both balls in lbs. $\times \frac{1}{12}$ vertical rise of balls in inches. For Porter and other loaded governors the energy in foot-lbs. = weight of both balls in lbs. $\times \frac{1}{12}$ vertical rise of balls in inches + weight of central weight in lbs. $\times \frac{1}{12}$ its vertical rise in inches.

The energy required is from 0·75 to 1 foot-lb. per inch diameter of cylinder, for working fairly well balanced throttle valves.—PROF. GOODMAN.

Very sensitive governors are fitted with a dash-pot, to prevent them from flying suddenly in or out, and thus causing the engine to "hunt."

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815. FLUCTUATION OF SPEED WITH GOVERNOR.

"What happens under ordinary conditions with an engine running on no load and controlled by a governor? It always hunts. Every governor hunts. There is no such thing in the world as a governor which does not hunt, and there will not be until we get one which has no friction, because before the governors can act, the change in centrifugal force must be enough to overcome the friction of the governor. That friction is a definite amount, and therefore the change of speed must be a definite amount. That process is repeated up and down continually, and when we say the governor does not hunt, we mean the changes are made so rapidly that we are unable to see the effect upon the speed by the eye, but the hunting is there all the same."—J. S. Raworth.

816. QUICK REVOLUTION ENGINES.

The essential requisites for successful quick revolution engines are:—

1. High efficiency, thermal and mechanical, in view of the rapid rate in which they transform energy, i.e., spend and earn money.
2. Absence of vibration in themselves and their surroundings.
3. Minimum wear internally and cost of upkeep.
4. Thorough and automatic lubrication.
5. Efficient governing to ensure only small variations of speed, even between the extreme limits of no load to full load.
6. Simplicity of design. —W. A. F. Crawford.

817. FLY-WHEELS, NOTES AND FORMULÆ.

F = total centrifugal force in lbs. radially

$$= \frac{W\,v^2}{r\,g} = \cdot 00034\; r\; W\; R^2$$

= total tension in arms to be divided by number of arms for tension in each. (The tension in the arms is sometimes spoken of as the central force or centripetal force.)

d = mean diameter and r = mean radius of rim in feet.

W = weight of rim in lbs.

R = revolutions per minute.

v = circumferential velocity in feet per second.

Tension at any cross section of rim in lbs. per sq. inch

$$= \frac{F}{2\,\pi} \text{ (safe limit = 2,500 lbs.);}$$

also $= \frac{v^2}{10}$ approx. $\left\{ \begin{array}{l} \text{(where } v = \text{linear velocity feet per sec.; max.} = 80 \text{ to} \\ \text{100 feet per sec. for cast iron, 200 for cast steel).} \end{array} \right.$

"Cast steel fly-wheels, with a tensile strength of 27 tons per sq. inch, will stand a speed 90 per cent. greater than cast iron fly-wheels."

M = foot-lbs. momentum at 1 revolution per minute.

U = units of work (foot-lbs.) accumulated in fly-wheel at any velocity.

$$M = \frac{W\,d^2}{23000}. \qquad U = M\,R^2.$$

E = excess of demand or supply in any given time in foot-lbs.

R max. R min. = greatest variation allowed in speed, i.e., revolutions per minute.

$$M = \frac{E}{R \text{ max.} - R \text{ min.}}$$

The diameters of fly-wheels will be as $\sqrt[5]{M}$, the dimensions of rim being proportional to diameter.—PERRY.

n = number of revolutions per second.

r = effective radius of gyration in feet.

U = units of work stored in wheel.

$$U = \frac{W\,(2\,\pi\,r\,n)^2}{2\,g}.$$

Variation from mean velocity not to exceed $\frac{1}{m}$, usually $\frac{1}{20}$ to $\frac{1}{60}$

a = area of section of rim in sq. inches.

$$a = \frac{\text{H.P.} \times 1803 \times m}{r^3 \times R^3}.$$

$$\text{Weight in tons} = \frac{\text{H.P.} \times 2275 \times m}{r^2 \times R^3}.$$

$$\text{Mean radius} = \frac{12 \cdot 17}{R} \times \sqrt[3]{\frac{\text{H.P.} \times m}{a}}.$$

—MORIN.

R = revolutions per minute.

A = sectional area rim sq. feet.

r = radius feet to inside of rim.

H = I.H.P. of engine.

n = ratio of mean velocity to difference between mean and either extreme (say 10 for a difference of 10 per cent.).

$$r = \frac{12}{R} \times \sqrt[3]{\frac{n\,H}{A}}.$$

—O. BYRNE.

Ultimate velocity at centre of gravity of rim to produce bursting = 19,235 feet per minute; $\therefore$ safe velocity, say 5,000 feet per minute.—C. E. EMERY.

U foot-lbs. energy required to be given out in t seconds or revolutions, when s seconds or revolutions can be used to restore it, will require an average of E foot-lbs. in fly-wheel.

$$E = \frac{s}{s+t} \times U.$$

And if the variation of velocity must not exceed c per cent. on either side of v, mean velocity in feet per second when running at n revolutions per minute,

$$v = \frac{2\pi r n}{60},$$

$$v_{max} = \frac{v\,(100 + c)}{100}, \qquad v_{min} = \frac{v\,(100 - c)}{100},$$

and the necessary weight W of fly-wheel rim will be

$$W = \frac{E\,2g}{v^2_{max} - v^2_{min}}.$$

—ALLEN.

Fly-wheels may be designed for an accumulated energy equal to the foot-lbs. given out by the engine in three revolutions.—JAS. HENDRY.

P = total average pressure on piston in lbs.
S = stroke in feet.
D = mean diameter of rim in feet.
W = weight of rim in cwts.
A = sectional area of rim in sq. inches.

$$W = \frac{P S}{45 D}, \qquad A = \frac{1 \cdot 42\,W}{D}.$$

Multiply by 1·5 when cut off earlier than ½ stroke. Diameter usually 3½ to 4 times stroke of engine.—"PRACTICAL ENGINEER POCKET BOOK."

818. INVESTIGATION OF FLY-WHEELS.

$\frac{W}{g}$ = mass (m), $\frac{v^2}{2g}$ = height (h), kinetic energy = $\frac{1}{2} m v^2$, potential energy = W h, accumulated work = $\frac{W v^2}{2g}$,

then

$$\frac{W v^2}{2g} = W h = \tfrac{1}{2} m v = M R^2$$

but

$$v^2 = \left(\frac{2\pi r R}{60}\right)^2 = R^2\left(\frac{2\pi r}{60}\right)^2 \therefore \frac{W}{2g}\left(\frac{2\pi r}{60}^2\right) = M,$$

or

$$W r^2 \frac{4 \times 9 \cdot 87}{3600 \times 64 \cdot 4} = \frac{W r^2}{5871} = M.$$

$$\therefore \text{ Energy of fly-wheel } = \frac{W r^2}{5871} \times R^2.$$

Energy stored up in any rotating body $= \frac{1}{2} I a^2$, where I = moment of inertia about the axis $= \Sigma m y^2$, a = any velocity in radians per second.

$$a = \frac{2 \pi R}{60} \qquad \therefore \text{ Energy } = \frac{1}{2} I \left(\frac{2 \pi R}{60}\right)^2 = I \frac{\pi^2 R^2}{1800}.$$

$$\therefore \text{ M of fly-wheel } = \frac{I \pi^2}{1800} = \cdot 00548 \text{ I}.$$

But

$$\text{M of fly-wheel } = \frac{W r^2}{5871}. \qquad \therefore \cdot 00548 \text{ I} = \frac{W r^2}{5871},$$

and

$$I = \frac{W r^2}{5871 \times \cdot 00548} = \frac{W r^2}{32 \cdot 17}.$$

819. STRENGTH OF CRANK PIN.

p = uniformly distributed load in lbs.
l = length of journal in inches.
d = diameter of journal in inches.
f = greatest safe stress per sq. inch:

Say, wrought iron . . .	6,000 to 9,000
steel	9,000 to 13,500
cast iron	3,000 to 4,500

$\frac{p l}{2}$ = greatest bending moment at fixed end of journal.

$$M = \frac{\pi}{32} d^3 = \cdot 0982 \, d^3 = \text{modulus of circular sec.} = \frac{2 I}{d}.$$

$$I = M \frac{d}{2} = \frac{\pi}{32} d^3 \times \frac{d}{2} = \frac{\pi}{64} d^4 = \cdot 0491 \, d^4 = \text{moment of inertia of circular section.}$$

$$p = \cdot 0982 \, d^3 f \frac{2}{l} = \frac{\cdot 1964 \, d^3 f}{l} = \frac{d^3 f}{5 \cdot 1 \, l}.$$

$$d = \sqrt[3]{\frac{p l}{\cdot 1964 f}} = \sqrt[3]{\frac{5 \cdot 1 \, p l}{f}}.$$

820. FLY-WHEEL SHAFT FOR ROLLING MILL.

d = diameter steel shaft, inches.
W = weight fly-wheel, tons.
S = span between bearings in feet.

$$d = 3\sqrt[3]{\frac{W\,S}{2}}.$$

821. CALCULATION OF ENGINE SHAFTS.

By law of virtual velocities, mean pressure on crank pin

$$= d^2\frac{\pi}{4} \times m \times \frac{2\,s}{\pi\,s} = \frac{d^2\,m}{2} = \frac{a\,m}{1\cdot 57};$$

but the force being irregular, the maximum must be taken for the crank and fly-wheel shaft; say full pressure on piston acting at radius of crank,

$$= \frac{d^2\,\pi\,p}{4} \text{ at radius } \frac{s}{2}.$$

Beyond the fly-wheel $\frac{d^2\,m}{2}$ may be substituted for $\frac{d^2\,\pi\,p}{4}$, as the strain will there be practically uniform.

p = maximum boiler pressure, lbs. per sq. inch.
m = mean pressure in cylinder „ „
s = stroke of piston in feet.
d = diameter of piston in inches.
a = area of piston in sq. inches.
f = factor of safety.

	Steam engine.	*Hydc. eng. and steam winches.*
Wrought iron and steel . . .	$\frac{1}{8}$	$\frac{1}{10}$
Cast iron	$\frac{1}{10}$	$\frac{1}{15}$

k = ultimate strength, 1-inch bar, 1 foot radius.

	Cast steel.	*Mild steel.*	*Wrought iron.*	*Cast iron.*
	1250	1000	750	600
c = constant or safe load = $f\,k$.				
Steam engine . . .	200	175	125	60
Hydraulic engine, etc. .	125	100	75	40

D = diameter of shaft in inches.

For crank shaft:

$$D = \sqrt[3]{\frac{d^2 \times \pi \times p \times s}{4 \times 2 \times f \times k}} = \sqrt[3]{\frac{d^2\,p\,s}{2\cdot 5\,c}}.$$

And beyond the fly-wheel:

$$D = \sqrt[3]{\frac{d^2 \times m \times s}{2 \times 2 \times f \times k}} = \sqrt[3]{\frac{d^2\, m\, s}{4\, c}}.$$

For two cylinders, let diameter $= D + \cdot 15\, D$.
For three cylinders „ $= D + \cdot 3\, D$.

Hollow shafts:

Let d = diameter required when solid, and x the fraction of external diameter required for the hole, as $\frac{1}{2}$, $\frac{1}{4}$, etc. Then the requisite external diameter of hollow shaft $= \sqrt[3]{\left\{\frac{d^2}{1 - x^4}\right\}}$.

822. STEAM ENGINE DIMENSIONS.

t = thickness in inches.
d = diameter in inches.
p = boiler pressure, lbs. per sq. inch.
A = area of piston in sq. inches.
S = length of stroke in inches.
D = diameter of piston in inches.

Cylinder walls, $t = \frac{p\, D}{4000} + \cdot 5$, or $\cdot 0003\, p\, D + \frac{\sqrt{D}}{8}$.

Cylinder covers, $t = \frac{1}{8}$ inch thicker than cylinder, and stiffened as required, or $\cdot 01\, l\, \sqrt{p}$ (l = unsupported length ins.), or $\frac{p\, D^2}{12000}$.

Cylinder covers, curved $t = \frac{p\, D^2}{14400 \text{ rise}}$.

Cover studs, d = maximum stress on net section, 2000 lbs. per sq. inch, minimum diameter $= \frac{5}{8}$ inch, or $d = \frac{D}{75} \times \sqrt{\frac{p}{n}}$

Cylinder flange, t = diameter studs $\times 1\frac{1}{2}$.

Cylinder liner, $t = \frac{p\, d}{3500} + \frac{1}{2}$ for cast iron $= \frac{p\, d}{3000}$ for steel.

Pitch of studs $\sqrt{\frac{100\, T}{p}}$, or 4 to 5 diameters apart.

(T being thickness of cover in sixteenths).

Piston rod, $d = \cdot 0167\, D\, \sqrt{p}$ if iron,
$= \cdot 0144\, D\, \sqrt{p}$ if steel, and net area of screwed part not less than $\frac{A\, p}{5000}$.

Piston, depth = $\frac{1}{3}$ D in small to $\frac{1}{4}$ D in larger,
Gland, diameter = diameter rod + $\frac{1}{16}$ in.
Stuffing box, diameter = $1\frac{1}{2}$ diameter, rod.
Stuffing box, depth = ·75 to $1\frac{1}{2}$ diameter rod.
Ramsbottom rings, side = $\frac{1}{30}$ D.

Crank shaft, if well supported, $d = \sqrt[3]{\frac{A\,p\,S}{3600}}$.

Connecting rod, length = 3 S, diameter same as crosshead pin.
Crosshead pin, diameter = d piston rod × 1·25.
Crosshead pin, length = do. × 1·4.

Crank pin, diameter = $\sqrt[3]{\frac{A\,p\,l}{1800}}$ (l = length = $1\frac{1}{2}$ diam.).

Cap bolts, diameter = $\frac{D}{100} \sqrt{p}$ for iron, $\frac{D}{112} \sqrt{p}$ for steel.

Pressure on bearing surfaces = 400 lbs. per sq. inch, except crossheads and slide blocks which should not exceed 100 lbs.

A practical rule for thickness of steam cylinder for small engines = $\frac{1}{6}$ $\sqrt{d}$ + thickness for re-boring, with a minimum of $\frac{5}{8}$ inch.

Safe load on piston rod

$$\text{W lbs.} = \frac{4000 \text{ A in sq. ins.}}{1 + \frac{16\,l^2}{9 \times 2250 \times d^2}} = \frac{3141 \cdot 6\, d^2}{1 + \cdot 000079 \left(\frac{l}{d}\right)^2}$$

823. FITTING PISTON RINGS.

Say plain cast iron piston ring 3″ × $\frac{3}{4}$″ in section.

d = diameter of cylinder in inches.
d + ·0243 d = turned diameter of ring before cutting.
·0555 d = amount cut out of circumference.
d — ·002 d = diameter of ring after final turning.

824. CONDENSERS.

In the old jet or spray condenser the air pump had to remove at each stroke the water used by the engine as steam and also the condensing water. In the surface condenser only the former has to be removed by the air pump, the circulating pump dealing with the water producing condensation. In the multitubular surface condensers the water producing condensation passes through the tubes while the steam is in contact with the outside, or *vice versâ*.

In the jet condenser the loss of heat from blowing off the brine averaged

12 to 15 per cent., the maximum density allowed being three times the density of sea water.

In the surface condenser with cooling water at 60° F., 25 lbs. of steam should be condensed per sq. foot of the tube surface per hour.

Condensers should be tested to 30 lbs. per sq. inch.

825. AIR PUMP.

Circulating water say 60° F., hot-well 120° F., net capacity of air pump = 36 to 40 times volume of water condensed from steam. Efficiency of single acting air pump say 50 per cent.

826. CIRCULATING PUMP.

The quantity of water provided by the circulating pump must be such that its velocity through the condenser tubes is sufficient to abstract the heat from the steam and convert it to water in the hot-well at a suitable temperature for re-transfer to the boiler by the air pump. The heat gained by the circulating water is equal to the heat lost by the steam.

827. CIRCULATING WATER FOR CONDENSATION.

The ordinary surface condenser requires 30 lbs. circulating water to condense 1 lb. steam. The Körting self-acting condenser without regulation requires 25 lbs. water, and with regulation 18 lbs. water, to condense 1 lb. steam.

The steam turbine is increased 4 to 5 per cent. in efficiency by an allowance of 50 times the full-load steam consumption for condensing water instead of 30 times.

828. COOLING PONDS.

Cooling ponds for circulating water from condensers should have 2·5 sq. feet surface area and 12·5 cub. feet contents for each lb. of steam condensed per hour.—Henry Davey.

829. EVAPORATIVE CONDENSERS

consist of a nest of tubes through which exhaust from engine passes. Water allowed to trickle over them outside. Fresh air supplied by fan to carry off heated vapour and promote further evaporation. Made by D. Stewart and Co., Glasgow. Minimum cooling surface = 1 sq. foot per 10 lbs. steam condensed per hour. Circulating water = 20 times weight of steam. Water lost by evaporation = 1 to 1½ lbs. per lb. of steam.—Henry Davey.

"In a well designed condenser of this kind a quantity of water less than the feed supply to the boilers (sometimes half this quantity) will suffice to ensure a good vacuum."—"THE PRACTICAL ENGINEER POCKET BOOK."

See detailed report of a trial of a Edward's Evaporative Condenser in THE PRACTICAL ENGINEER, 28th October, 1892.

830. AIR CONDENSERS.

Steam passed through thin brass tubes, air circulated by fan outside. With $\frac{3}{4}$ inch inside diameter of tubes, 5-feet run = 1 sq. foot cooling surface. Weight of condenser, say 1 ton per 800 sq. feet cooling surface. Difference of temperature of air at entrance and exit say 80° F. May be used for producing blast after leaving condenser. Loss of heat by $\frac{3}{4}$-inch pipe for air contact only = 2·25 units of heat per sq. foot per hour per 1° F. difference of inside and outside. At 212° F. 1 lb. of steam contains 966 units of latent heat which are given up on condensation.

Example.—Steam 212°, air mean 100°, difference 112°, $112 \times 2 \cdot 25 = 252$ units passing per sq. foot per hour, $\frac{252}{966} = \cdot 26$, say $\frac{1}{4}$ lb. steam may be condensed per sq. foot per hour. Experiments on single pipes give much higher efficiency, but this is probably on account of radiation playing a more important part. By an article in "ENGINEERING" (1869) $\frac{1}{3}$ lb. may be taken.

At a consumption of 30 lbs. steam per H.P. per hour, and an estimate of $\frac{1}{4}$ lb. condensed per sq. foot per hour, cooling surface = 120 sq. feet per H.P. And for 80° F. difference of air temperatures we have weight of air required per lb. of steam =

$$\frac{\text{Units of heat to be absorbed by air}}{\left.\begin{matrix}\text{Difference of entrance and}\\ \text{exit, F}^\circ\end{matrix}\right\} \times \left\{\begin{matrix}\text{specific heat of air at}\\ \text{constant pressure}\end{matrix}\right.}$$

$$= \frac{U}{(T - t) \cdot 238} = \frac{966}{80 \times \cdot 238} = 50 \text{ lbs., and}$$

50×30 lbs. steam per H.P. = 1,500 lbs. air per H.P. At $\frac{1}{10}$ lb. as the weight of a cub. foot of air we require $1{,}500 \times 10 = 15{,}000$ cub. feet air per H.P. per hour.

If boiler evaporates 5 lbs. per sq. foot heating surface per hour, then cooling surface must equal 20 times heating surface.

When temperature of air = 59° F., copper will condense about 0·28 and cast iron 0·36 lbs. steam per sq. foot per hour.

831. COMPARISON OF STEAM ENGINES OF SAME TYPE AND DIFFERENT SIZE.

Engines same type. Boiler pressure same. Cut-off same. Multiplier for proportionate linear dimensions equals

$$\sqrt{\frac{\text{required H.P.}}{\text{original H.P.}}} = \sqrt{r},$$

and the revolutions per minute will be

$$\frac{\text{original revolutions}}{\sqrt{r}},$$

without allowing for difference in proportion of friction. Friction varies approximately as $\sqrt[3]{r}$.

Marine engineers' rule for engines of varying power, but same type:

Required cylinder diameter =

$$\sqrt{\frac{\text{required H.P.}}{\text{original H.P.}} \times \frac{\text{original piston speed}}{\text{required piston speed}} \times \left(\frac{\text{original cylinder}}{\text{diameter}}\right)^2}.$$

When the cylinder diameter alone is altered the revolutions will vary directly as the diameter.

Examples.

I.H.P., 6,000.	I.H.P., 2,750.
Cylinder 43, cut-off $8\frac{1}{2}/10$.	Cylinder 30, cut-off $8\frac{1}{2}/10$.
„ 62 „ $8\frac{1}{2}/10$.	„ 44 „ $8\frac{1}{2}/10$.
„ 96 „ 8/10.	„ 68 „ 8/10.
Piston rods same size.	——
Stroke, 4 feet 3 inches.	Stroke, 3 feet.
Boiler pressure, 135.	Boiler pressure, 135.
Revolutions, 95.	Revolutions, 130.
Piston speed, 807·5.	Piston speed, 780.

832. WEIGHT OF ENGINES PER I.H.P.

High speed steam engines, including fly-wheel .	100 to 135 lbs.
Low speed do. do.	135 to 240 lbs.
Marine engines average	5 cwt.

833. EARLY STEAM TURBINES.

Steam turbines are of two principal kinds, (*a*) reaction and (*b*) impulse, the progenitors being (*a*) the Hero steam engine, where a tube having the ends bent at right angles rotated upon its centre by the pressure due to the reaction of the escaping steam, and (*b*) the Branca steam engine, where the impulse of escaping steam impinging upon the vanes of a wheel caused its rotation.

834. MODERN STEAM TURBINES.

In the *action* or *impulse* class, as the De Laval, Rateau, Curtis, and Riedler-Stumpf turbines, the steam is completely expanded, and acquires its full velocity in specially formed nozzles. It is then passed into the turbine proper, the velocity being reduced by the vanes, and its energy reappearing as work on the turbine shaft.

In the *reaction* class, of which the Parsons turbine is the most practical, the generation and absorption of the kinetic energy in the steam proceed simultaneously, so that the velocity of the steam is never very great compared with what it is in turbines of the other class. It follows that since the velocity is being generated whilst the steam is passing through the turbine, the passages through the guide and wheel vanes must be carefully designed, each passage being, in fact, part of a special kind of steam nozzle.—F. FOSTER.

Steam turbines, like hydraulic turbines, are usually divided into classes :—

1.—Impulse turbines, in which the pressure of the steam on both sides of the turbine wheel is equal, and

2.—Reaction turbines, in which the pressure is greater at the entrance to the blades of the wheel than at the exit. The two classes may be again divided into "axial" and "radial" flow turbines; in the first case the direction of the flow of the steam is parallel to the axis of the machine, and in the second the steam is admitted at the centre and flows radially to the periphery of the wheel.

These classes are sometimes again divided into machines with "full peripheral admission" and "partial peripheral admission," depending on whether the steam is admitted to the entire circumference of the first rotating wheel or only partly. Practically all the commercial machines at present on the market are of the axial flow type, with either full or partial peripheral admission. As examples of the various classes may be given the following :—

1.—Single-stage impulse turbine—Laval. Multi-stage impulse turbine—Curtis and Rateau.

2.—Reaction turbine, multi-stage—Parsons.—G. D'A. MEYNELL.

835. STEAMSHIPS.

Rankine's Rules :—

s = speed of ship in knots per hour.

la = length of after body in feet.

lf = length of fore body in feet.

$la = \frac{4}{9}s^2$, $\quad lf = 1$ to $1\frac{1}{2}\,la$, $\quad s = \sqrt{2\frac{1}{4}\,la}$, Augmented surface = mean immersed girth × length × (1 + 4).

Probable resistance of ship in lbs. = augmented surface × s^2 × c

c = average of ·01.

Approximate resistance = $s^2 (\sqrt[3]{D})^2$.

Scott Russell's Rules:—

Greatest speed in knots = $\sqrt{2\frac{1}{2} \text{ times length of after body}}$.

At moderate speeds, resistance in lbs. = speed knots ^{2} × $(\sqrt[3]{\text{displacement tons}})^2 \times \cdot 8$ to $1 \cdot 5$.

$$\text{Effective H.P.} = \frac{\text{resistance lbs.} \times \text{speed knots}}{326};$$

$$\text{Indicated H.P. average} = \frac{\text{resistance lbs.} \times \text{speed knots}}{200}.$$

836. SPEED IN KNOTS.

Admiralty knots = nautical miles of 6,080 feet per hour.

Speed in knots × 1·15 = miles per hour.

5280 feet = 1 ordinary mile.

Feet per minute ÷ 88 = miles per hour.

Feet per minute ÷ $101\frac{1}{3}$ = knots per hour.

Speed of ship in knots (per hour)

$$= \sqrt[3]{\frac{\text{I.H.P.} \times \text{sectional coeff. of performance, say } 600}{\text{area immersed midship section, sq. feet}}},$$

or

$$= \sqrt[3]{\frac{\text{I.H.P.} \times \text{displacement coeff. of performance, say } 240}{\text{cube root of square of displacement in tons}}}$$

∴ the power required to propel ships varies as speed^3.

837. DEFINITIONS RELATING TO SCREW PROPELLERS.

Length = $A^1 B^1$ measured along the axis of the shaft.

Angle = P O H, which is a plane triangle when developed.

Pitch = the distance traversed on $A^1 B^1$ for one complete revolution of A^1 P.

Slip = the difference between the theoretical forward motion, calculated from the pitch of the screw, and the actual progress of the ship.

Area = A^1 P O B, surface of blade in sq. feet.

Thread or Helix = Outer edge of blade, O P.

Diameter = Diameter of cylinder circumscribing the thread of screw. A^1 P = radius.

838. NOTES ON SCREW PROPELLERS.

In the common form of propeller the screw surface is generated by a line perpendicular to the axis of the shaft revolving round the shaft and progressing uniformly along it.

Screw surfaces are also generated by a line at right angles to a conical surface; in some cases the vertex of the cone points aft, and in others forward. In some the surface is traced out by a line perpendicular to a sphere; the object in such cases being to diminish, if possible, centrifugal action of the water.

Screws of same pitch have different angles if their diameters differ; angle reducing as diameter increases.

The screws are either right or left-handed, and may have two, three, or four blades.

839. RELATIVE EFFICIENCY OF LARGE AND SMALL SCREWS.

"As regarded the relative efficiency of large and small screws, if consideration were confined to the propellers alone, apart from the vessels they were designed to propel and the services they were intended to perform, efficiency was independent, within certain limits, of the absolute size of the screw. According to Mr. Froude, the screw for a vessel of 500 I.H.P., and 10 knots speed per hour, might be 10 feet in diameter, 0·8 pitch-ratio, and run at 138 revolutions per minute; or it might be 15½ feet in diameter, 2·5 pitch-ratio, and run at 33½ revolutions. Both screws would be credited with an efficiency of 69 per cent.; but the large screw was at a disadvantage when placed in a following stream, on account of the greater difference in the velocity of wake currents which it experienced, and also because of its greater liability to emerge from the water. To maintain a high speed against head winds and sea, a relatively large screw was desirable; the case was analogous to that of a tug. For such a purpose increased diameter should not be associated with increased pitch-ratio."

840. SLIP OF SCREW PROPELLER.

Slip is less when pitch is small and speed great, but more danger from heated bearings. When pitch is small, the propeller is less liable to break from a blow.

The slip is diminished, *cæteris paribus*, by

1. Decreasing the angle of the screw.
2. Increasing the diameter of the screw.
3. Increasing the length of the screw.

But the friction increases rapidly with the surface of the blade.

The indicated horse-power varies as the square of the speed of the ship × number of revolutions of screw × pitch.

The most economical speed is when the vessel steams half as fast again as the opposing current, or half as fast again as a vessel it desires to overtake.

841. NEGATIVE SLIP.

Negative slip in screw propellers is caused either by the skin friction of the ship giving a forward velocity to the water in which the screw works, depending upon the lines of the ship, and the position and size of screw, being greater with full lines aft; or it is caused by an increase of pitch due to the straining of a weak propeller by the pressure of the water; or it is due to the pitch of the propeller being incorrectly estimated. It may vary from nil to 20 per cent.

842. PITCH OF SCREW PROPELLER.

Ordinary propellers have the pitch uniform throughout each blade, the angle varying with the distance from the axis, originally known as Smith's propeller.

Screws of increasing pitch are sometimes used, and known as Woodcroft's propeller.

Propellers with two blades are common in large ships, but those with three or four blades are better when the draught is small or in a rough sea.

Feathering-screws have the blades pivoted so that the angle, and thereby the pitch, may be altered.

The pitch of a screw varies with the ratio of the circle described by the screw to the immersed midship section.

A common proportion for pitch is 1 to 1·8 times diameter.

843. RELATION OF PITCH TO DIAMETER.

There does not appear to be any advantage in adhering to a fixed relation of pitch to diameter. Dimensions rather than form regulate the efficiency. In Thornycroft's experiments it was found that—

1.—The disc-area was proportional to the I.H.P. and inversely proportional to the cube of the speed.

2.—The revolutions per minute were proportional to the speed, and inversely proportional to the diameter.

3.—The constants were of the form

$$C_A = \text{disc area} \times \frac{V^3}{\text{H.P.}},$$

$$C_R = \text{revolutions} \times \frac{D}{V}.$$

when V = speed of screw through the water.
D = diameter of screw in feet.
H.P. = effective H.P. in screw shaft.

844. FORMULA FOR PITCH OF PROPELLER.

C = constant. = 737 ordinary mercantile marine.
= 600 cargo ships with full run.
R = revolutions per minute.
D = diameter propeller in feet.

$$\text{Pitch in feet} = \frac{C}{R}\sqrt[3]{\frac{\text{I.H.P.}}{D^2}}.$$

The pitch should never exceed $2\frac{1}{2}$ times diameter.
Another rule :—
Blade surface = 35 per cent. of disc area.
Breadth of blade = $\frac{1}{3}$ pitch.
Ratio pitch to diameter, average $1\frac{1}{3}$ to 1.
Coarse pitch requires more surface than fine pitch.

—" MECHANICAL WORLD."

Area of propeller circle =

$$\frac{\text{area immersed midship section}}{2\cdot75}.$$

845. ALTERATION OF PITCH.

With same mean pressure on piston, for small alterations of pitch

$$\text{pitch} \times \text{knots}^2 = \text{constant};$$

$$\text{pitch}^3 \times \text{revolutions}^2 = \text{constant}.$$

∴ increasing pitch reduces revolutions and speed.—SOMERSCALES.
P = pitch of propeller.
N = number of revolutions per minute.

$$P^3 \propto \frac{1}{N^2}.$$

846. INDICATED H.P. REQUIRED FOR SCREW PROPELLER.

R = revolutions per minute.
D = diameter of propeller in feet.
L = length of propeller in feet.
P = pitch of propeller in feet.
s = slip of propeller in fraction of unity (as $\frac{1}{4}$).
θ = angle of blade at periphery.

$$\text{I.H.P.} = \frac{D^3\,R^3}{480{,}000}\left(L\,s\,\cos\theta + \frac{1}{9}\right).$$

$$\text{Knots (per hour)} = \frac{3\,P\,R}{304}\,(1 - s).$$

847. BUILT-UP CRANK SHAFTS.

City of Rome s.s., Whitworth compressed steel. Difference in diameter of fitting parts allowed for shrinkage = $\frac{1}{1000}$ diameter.

848. TWIN SCREWS.

$$\text{Twin screws, dimensions} = \frac{\text{single screw}}{\sqrt{2}}.$$

$$\text{,, \quad revolutions} = \text{single screw} \times \sqrt{2}.$$

849. PADDLE WHEELS.

k = speed of vessel in knots.
N = revolutions of engine per minute.
r = radius of rolling circle in feet, or circle with circumferential velocity equal to ship's motion.

$$\frac{6080\,k}{60} = 2\,\pi\,r\,N. \quad \therefore \quad r = \frac{6080\,k}{60\,\pi\,2\,N} = \frac{16\,k}{N}.$$

R = radius outside wheel in feet.
b = breadth radially of float-board or paddle in feet.
m = mean radius, to centre of gyration, of float-boards.

$$m = r - b + \sqrt[3]{\frac{(R - r + b)^4}{4\,b}}.$$

v = circumferential velocity of centre of pressure of float-boards in feet per second.

$$v = \frac{2\,m\,\pi\,N}{60} = \cdot 10472\,m\,N.$$

a = area of float-boards in sq. feet.
p = pressure in lbs. on vertical float-board.

$$p = \frac{62\cdot 5\,a}{2\,g} \times \left(v - \frac{6080\,k}{3600}\right)^2 = a\,(v - 1\cdot 7\,k)^2.$$

n = number of paddle wheels.

$$\text{Effective H.P. required} = \frac{v\,n\,p}{33{,}000}.$$—HANN *and* GENER.

850. EFFICIENCY OF PADDLE WHEELS.

Common, light draught	=	·666
,, deep ,, ...	=	·553
Feathering (Morgan's patent) all depths . .	=	·666

851. EQUILIBRIUM OF FLOATING BODIES, AS SHIPS.

When a floating body is in equilibrium, the centre of gravity of the body and the c.g. of the displaced fluid, called the centre of buoyancy, are in the same vertical line. When the floating body is moved through a small angle, the intersection of the originally vertical line through c.g. of body, with vertical line through c.g. of now displaced fluid, is called the *metacentre* (Bouguer). The floating body will return to its original position so long as the meta centre remains above the c.g. of body. The equilibrium is stable, unstable, or indifferent, respectively, as the metacentre falls above, below, or coincides with the c.g. of the body.

When a body floats on a fluid it displaces a quantity equal in weight to itself, and when it sinks it displaces a quantity equal in bulk to itself.

852. DISPLACEMENT OF SHIPS.

Length × breadth × draught × coefficient of fineness = displacement, e.g., H.M. *Blake*.

$$375 \times 65 \times 25{\cdot}75 \times {\cdot}520 = 9{,}000 \text{ tons.}$$

Admiralty displacement formula, $c = \frac{D^{\frac{2}{3}} V^3}{\epsilon}$, is not correct, it should be

$$\log \epsilon = \log \frac{D^{0{\cdot}6} V}{c} + a V.$$

—R. MANUEL.

853. DIFFERENCE OF DRAUGHT IN SALT AND FRESH WATER.

S = draught in feet in salt water.

F = ,, ,, fresh ,,

$$F = \frac{36}{35} S, \qquad S = \frac{35}{36} F.$$

854. COEFFICIENTS OF STEAMSHIPS.

Sectional coefficient of performance = 500 to 700.

$$= \frac{\text{knots per hour}^3 \times \text{area immersed mid. section}}{\text{I.H.P.}}.$$

Displacement coefficient of performance = 180 to 260.

Both coefficients increasing with increase of length and fineness, and decreasing with speed.

Coefficient of fineness $= \cdot 55$ to $\cdot 70$.

$$= \frac{\text{tons displacement} \times 35}{\text{length between perps.} \times \text{beam} \times \text{mean draught less depth keel}}.$$

Coefficient of water lines $= \cdot 63$ to $\cdot 76$.

$$= \frac{\text{tons displacement} \times 35}{\text{area immersed mid. section} \times \text{length between perps.}},$$

finer lines giving smaller figures in the last two cases.

855. MEASUREMENT OF SHIPS.

Tonnage O. B. M. (Old Builders' Measurement)

$$= \frac{(\text{length between perps.} - \frac{3}{5}\ \text{breadth}) \times \text{breadth}^2}{2 \times 94}$$

At, say, 13 knots, $\frac{\text{tonnage}}{4 \cdot 5} =$ H.P. required.

Tonnage ratios for merchant vessels $= \cdot 55$ to $\cdot 75$ of parallelopiped dimensions.

856. FROUDE'S LAW OF COMPARISON.

If the linear dimensions of a ship are λ times those of its model, and if at the velocities $v_1, v_2, v_3, \ldots$ of the model in water, the resistances are $r_1, r_2, r_3, \ldots$ then the resistances $R_1, R_2, R_3, \ldots$ of the ship, at the velocities $V_1, V_2, V_3, \ldots$ (which are respectively equal to $v_1 \sqrt{\lambda}$; $v_2 \sqrt{\lambda}$; $v_3 \sqrt{\lambda} \ldots$) will be $R_1 = \lambda^3 r_1$; $R_2 = \lambda^3 r_2$; $R_3 = \lambda^3 r_3$.

—Dr. Robert Caird.

857. PROPORTIONS OF HULLS AND MACHINERY.

——	*Hull.*	*Cylr.*	*Br.*	——
Trawler.	108½.21½.12¾	$\frac{10.16\frac{1}{4}.27}{24}$	9 *d* 9½ *l* 200 lbs.	190 gr. tng. 300 I.H.P. 9¾ knots.
Steam yacht.	193.25¼.15¼	$\frac{14.23.38}{24}$	175 lbs.	500 gr. tng.
Steamer	386.52¼.28	$\frac{25.4.68}{48}$	—	—
Steamer	351.49.25½	$\frac{23\frac{1}{2}.39.66}{45}$	180 lbs.	—

858. TUGS FOR RIVER TOWING.

Approximate cost of engines, boiler, shafting, propeller, etc., complete.

Compound condensing	£12 per I.H.P.
Water per annum	0·75 tons "
Coal "	3 " "
High pressure non-condensing	£10 per "
Water per annum	40 tons "
Coal "	5 " "

859. SIGNALS FOR TUG ENGINEER.

1 bell = easy ahead, 1 bell = stop;
2 bells = go astern, 1 bell = stop;
more than 2 bells = full speed ahead;
1 bell = easy, 1 bell = stop.

Captains vary slightly, some ring 3 or more for going astern instead of ahead.—W. Cory and Son, 1890.

860. SHIPS' BELL TIME.

Bells.	*Time* A.M. *and* P.M.		
1	12.30	4.30	8.30
2	1.0	5.0	9.0
3	1.30	5.30	9.30
4	2.0	6.0	10.0
5	2.30	6.30	10.30
6	3.0	7.0	11.0
7	3.30	7.30	11.30
8	4.0	8.0	12.0

The bells are given two at the time in quick succession and odd one by itself.

A ship's watch is 4 hours, 8 bells to 8 bells, except 4 p.m. to 6 p.m. and 6 p.m. to 8 p.m., which are "dog-watches," half the length of the others.

861. THE MORSE CODE.

A ·—	F ··—·	K —·—	P ·——·	U ··—
B —···	G ——·	L ·—··	Q ——·—	V ···—
C —·—·	H ····	M ——	R ·—·	W ·——
D —··	I ··	N —·	S ···	X —··—
E ·	J ·———	O ———	T —	Y —·——
				Z ——··

Example :—

862. TRACTIVE FORCE OF LOCOMOTIVES.

d = diameter of piston in inches.
a = area of piston in sq. inches.
l = length of stroke in feet.
n = number of cylinders.
D = diameter driving wheel in feet.

Then the tractive force at circumference of driving wheels for each lb. per sq. inch mean effective pressure on piston

$$= \frac{2\,a\,n\,l}{\pi\,D}, \quad \text{or} \quad = \frac{d^2\,l}{D}.$$

Also let μ = adhesion of wheels to rails (say ·2)
W = weight on driving wheels,
then $W\mu$ = maximum possible tractive force.

The greatest mean effective pressure on piston is commonly assumed to be 85 per cent. of boiler pressure, but this will be different for each design of valve gear, other things being equal. The ordinary mean effective pressure on piston would probably not exceed 50 per cent. of boiler pressure. The tractive power of a locomotive decreases as the speed increases.

T = traction in lbs. for two cylinders.
p = boiler pressure lbs. per sq. inch.
d = diameter of piston in inches.
l = stroke in inches.
D = diameter driving wheel in inches.
K = coefficient = ·65 for cut-off at $\frac{3}{4}$ths.

$$T = K\,\frac{p\,d^2\,l}{D}.$$

—De Pambour.

H.P. of locomotive =

$$\text{tractive force lbs.} \times \text{V miles per hour} \times \frac{5280}{60 \times 33{,}000} = \frac{88\,T\,V}{33{,}000}$$

A six-wheeled coupled tank engine on level railway 17 × 24 inch cylinders, 5 feet 6 inch wheels, at 20 miles per hour, exerts tractive force of 8,500 lbs. —A. Ross.

T = Tractive force at drivers.
t = draw-bar pull.
H = sq. feet of heating surface.
S = speed in miles per hour.
d = diameter of piston in inches.
L = stroke in feet.
D = diameter of drivers in feet.
W = weight of engine and tender, less weight on drivers.

$$T = 161 \frac{H}{S}.$$

$$t = 161 \frac{H}{S} - 3{\cdot}8 \frac{d^2 L}{D} - W(2 + \tfrac{1}{8} S) - 0{\cdot}11 S^2.$$

—PROF. GOSS.

For purposes of comparison in questions of traction, railway vehicles are sometimes reduced to the standard of 15 tons = 1 coach.

863. ADHESION OF LOCOMOTIVE WHEELS.

Locomotive driving wheels will commence to slip if the force at circumference equals about

$\frac{1}{5}$ of the load	=	448	lbs. per ton.
Westinghouse and Galton . . .	=	246·4	,,
Poirée	=	465·9	,,
Pennsylvania Railroad . . .	=	550	,,
Northern Pacific Railroad . . .	=	670	,,

—"ENGLISH MECHANIC."

Adhesion depends principally upon the state of the weather, and varies from a maximum of $\frac{1}{4}$ load to a minimum of $\frac{1}{10}$ load, average say $\frac{1}{8}$ load.

864. TRAIN RESISTANCE ON RAILWAYS.

Straight and level railway, in good condition, resistance (R) in lbs. per ton of total load (W).

$$= \frac{\text{speed miles per hour}^2}{171} + 8$$

$$\text{Do. on incline of 1 in } m = R + \left(\frac{1}{m} W \times 2240\right).$$

but speed² is believed to be too high a ratio.

-D. K. CLARK.

R = resistance in lbs. per ton drawn (bogie coaches with oil lubrication).
V = velocity of train in miles per hour.
L = length of train in feet over coach bodies.
W = weight of train in tons.

$$R = 2 \cdot 5 + \frac{V^{\frac{5}{3}}}{50 \cdot 8 + 0 \cdot 0278\, L}.$$

—J. A. F. ASPINALL.

$$R = 2 \cdot 5\, W + \left\{ 2 + 0 \cdot 0035\, L - \frac{200}{100 + W} \right\} V^{\frac{5}{3}}.$$

—PROF. R. H. SMITH.

The starting resistance of trains is about 17 lbs. per ton, and the engine absorbs in itself about 35 per cent. of the H.P. developed.—ASPINALL.

A. Ross (G.N.R.) takes $R = \frac{1}{224} = 10$ lbs. per ton average, made up as follows :—

Passenger carriages	5	lbs. per ton.
Goods engines	7	,,
Passenger engines	14	,,
Tenders	10	,,

and adds 5 lbs. per ton for curves of 600 feet radius diminishing to $\frac{3}{4}$ lb. for curves 5,000 feet radius.

On Prussian railways, R is taken at $\frac{1}{100}$ W = 22·4 lbs. per ton.

By experiment in railway goods stations, R = 30 lbs. per ton moving slowly.

A train of 300 tons total can be hauled 40 miles per hour on a level with 600 I.H.P. (F. W. DEAN). This gives a resistance of 18·75 lbs. per ton, assuming E.H.P. = I.H.P.

A 300-ton train can be hauled on a level road at 60 miles an hour with a consumption of about 40 lbs. of coal per mile, say 2,400 lbs. per hour, or 3 lbs. per I.H.P. per hour on 800 I.H.P.—"THE ENGINEER."

Approximate maximum speed of locomotive in miles per hour = diameter of driving wheel in inches

$$\frac{336}{\text{diameter wheel inches}} = \text{revolutions per mile.}$$

865. FRICTION OF WHEELS AND AXLES OF RAILWAY ROLLING STOCK.

Axles lubricated with oil or grease.

D = diameter of wheel.

d = diameter of axle.

μ = coefficient of rubbing friction = ·018.

μ' = „ rolling „ = ·001.

T = tractive force required to overcome friction in lbs. per ton on straight level.

$$T = 2240 \times \frac{\mu' D + \mu d}{D}.$$

Example :—D = 37, d = $4\frac{1}{4}$

$$T = 2240 \times \frac{\cdot 001 \times 37 + \cdot 018 \times 4\cdot 25}{37} = 6\cdot 87 \text{ lbs. per ton.}$$

—Adams and Co.

866. LOCOMOTIVE EXPRESS ENGINES.

Inside cylinder engine, 17 inches diameter, 24 inches stroke, firegrate 15 sq. feet area, heating surface, firebox 89 sq. feet, tubes 1,013 sq. feet. Load on axles, 9·45 tons leading, 11 tons driving, 8·75 tons trailing. Total wheel base 15 feet 8 inches. Can draw 293 tons on a level, at 45 miles per hour, with 120 lbs. per sq. inch. Leading wheels 3 feet $7\frac{1}{2}$ inches diameter, driving and trailing coupled 6 feet $7\frac{1}{2}$ inches diameter. Coal 26·3 lbs. per mile, with 10 coaches.—L & N. W. Railway.

867. EFFECT OF SPEEDS AND GRADIENTS.

An engine of uniform power will pull 40 vehicles at 20 miles per hour, 30 at 30 miles, 21 at 40 miles, 15 at 50 miles, 11 at 60 miles; and running at 15 miles per hour will pull 42 vehicles on a level, 34 up 1 in 600, 27 up 1 in 300, 20 up 1 in 150, 15 up 1 in 100, 12 up 1 in 75, 9 up 1 in 50.

—Du Bosquet.

868. LOCOMOTIVE TYRES.

D = diameter of wheel without tyre.

d = diameter inside tyre before putting on.

$$d = D - \frac{D}{1000}.$$

869. POWER TO ARREST A MOVING TRAIN.

W = load in tons.

v = initial velocity in feet per second.

f = distance in feet travelled during stoppage.

t = time occupied in seconds.
g = acceleratrix of gravity = 32·2.
F = force required in tons uniformly exerted.

$$F = \frac{W v^2}{2 g f} = \frac{W v}{g t}. \qquad t = \frac{2 f}{v}.$$

Miles per hour × $1\frac{1}{2}$ = approximate feet per second.

Example.—A train weighing 100 tons and travelling at the rate of 20 miles per hour is brought to rest in 100 yards, what will be the average resistance, and how many seconds will be occupied? Assume the ordinary friction as $\frac{1}{250}$ of the load.

$$F = \frac{W v^2}{2 g f} = \frac{100 \times 30^2}{2 \times 32 \times 300} = 4{\cdot}7 \text{ tons required to be exerted.}$$

$$\tfrac{1}{250} \times 100 = {\cdot}4 \text{ tons, } 4{\cdot}7 - {\cdot}4 = 4{\cdot}3 \text{ tons resistance.}$$

$$t = \frac{2 f}{v} = \frac{2 \times 300}{30} = 20 \text{ seconds.}$$

870. RAILWAY CURVES.

W = maximum rigid wheel base of rolling stock in feet.
G = gauge of railway—i.e., inside measurement between rails in feet.
R = minimum radius of curve in feet.

$$R = 9 W G.$$

Examples from practice:—

Coal wagon, wheel base 8 feet 6 inches, gauge 4 feet $8\frac{1}{2}$ inches, wheels 3 feet diameter, radius of curve = 360 feet = $5\frac{1}{2}$ chains radius.

Four-wheel-coupled tank locomotive, wheel base 3 feet 6 inches, gauge 2 feet, wheels 1 foot 6 inches diameter, radius of curve = say 60 feet radius.

Coal store trollies, wheels loose, $10\frac{1}{2}$ inches diameter, $\frac{3}{4}$ inch thick, solid, 2 feet 5 inches over all, 2-feet $1\frac{1}{2}$-inch centres, on 3-inch × 2-inch angle iron as rails, 2 feet 7 inches between flanges, can be worked round a radius of 8 feet 6 inches to centre line, say 4 times wheel base, but require 12-feet 9-inch radius at 2-feet $6\frac{1}{2}$-inch gauge for fairly easy running, say 6 times wheel base.

Gas works trucks, weight $7\frac{1}{2}$ cwts., contents 16 to 18 cwts., wheels 1 foot diameter, 1-foot 6-inch centres, bridge rails 1-foot 8-inch gauge with points and curves 12 feet radius.

2 feet chord has rise of $\frac{1}{n}$ foot, then radius of curve $= \frac{1}{2}\left(n + \frac{1}{n}\right)$ feet.

Tram-cars, rail gauge 3 feet 6 inches, wheel gauge 3 feet $5\frac{3}{4}$ inches, wheel diameter 2 feet 6 inches, wheel base 6 feet, width of groove $1\frac{1}{8}$ inches, except on curves less than 60 feet radius, where it is $1\frac{1}{4}$ inches. The sharpest workable curve is 33 feet radius to inner rail, but 35 feet is the least allowed.

871. ROLLING STOCK AND RAILS.

Diameter of wheels, gauge of rails, load per wheel, and width of rail heads, should be proportioned, so that the load in tons on each wheel per foot of wheel diameter should not exceed $\frac{2}{3}\sqrt{\text{rail gauge in feet}}$, and the load per inch of wheel diameter per inch width of rail head (at $\frac{1}{16}$ in. from highest point) should not exceed 150 lbs.

Example 1.—4-feet $8\frac{1}{2}$-inch gauge, 33-inch wheels, rail head 2 inches wide. $4{\cdot}8\frac{1}{2} = 4{\cdot}7$, $\sqrt{4{\cdot}7} = 2{\cdot}168$, $\frac{2}{3} \times 2{\cdot}168 \times \frac{33}{12} = 3{\cdot}97$, say 4 tons per wheel maximum load. Load on rail per inch wheel diameter per inch rail head $= \dfrac{4 \times 2240}{33 \times 2} = 135{\cdot}7$ lbs., which is well under 150. Or allowing 150 lbs., the wheel diameter will be $= \dfrac{4 \times 2240}{2 \times 150} = 29{\cdot}86$, say 30 inches minimum.

Example 2.—2-feet gauge, 12-inch wheel, rail head $1\frac{1}{2}$ inch wide. $\sqrt{2} = 1{\cdot}414$, $\frac{2}{3} \times 1{\cdot}414 \times \frac{12}{12} = {\cdot}943$, say 1 ton per wheel maximum load. Load on rail per inch wheel diameter per inch rail head $= \dfrac{1 \times 2240}{12 \times 1{\cdot}5} = 124{\cdot}4$, say 125 lbs., which is well below 150. Or allowing 150 lbs., the wheel diameter will be $= \dfrac{1 \times 2240}{1{\cdot}5 \times 150} = 9{\cdot}95$, say 10 inches minimum.

872. CENTRIFUGAL FORCE AND SUPERELEVATION OF OUTER RAIL.

P = centrifugal force in lbs.
g = force of gravity feet per second.
W = weight in lbs.
v = velocity in feet per second.
r = radius in feet.

$$P : W :: \frac{v^2}{g\,r} : 1 \quad \therefore \quad P = \frac{W\,v^2}{g\,r}.$$

S = superelevation of outer rail in inches.
G = gauge of rails in inches.
V = velocity in miles per hour.

$$\frac{v^2}{g\,r} : 1 :: s : G$$

$$\therefore s = \frac{G\,v^2}{g\,r} = G\,\frac{2{\cdot}15\,V^2}{g\,r} = \text{(approx.) } G \times \frac{V^2}{15\,r}\,;$$

—A. Ross.

873. ROAD TRACTION.

The load (in addition to cart) one horse can safely draw on a level each working day is said to be

On sand road	228 lbs.
„ ordinary earth	456 „
„ ordinary cobble stones	730 „
„ worn stone blocks	1,137 „
„ hard earth	1,193 „
„ hard gravel	1,279 „
„ hard macadam	1,391 „
„ ordinary stone blocks	1,828 „
„ best stone blocks	3,006 „
„ asphalte	6,095 „

—Morin, Gordon, etc.

Section XII.

WATER SUPPLY, SEWERAGE AND HYDRAULIC MACHINERY.*

874. WEIGHT AND BULK OF WATER.

A STANDARD or imperial gallon of water was formerly 277·274 cub. inches, is now 10 lbs. avoirdupois at 62° F. and 30 in. bar. = 277·123 cub. inches, or ·160,372 cub. feet.—CAPTAIN E. M. SHAW.

A cub. foot of pure water at its point of maximum density, 39° F. [39·1° F., or 4° C.], weighs 998·8 ounces = 62·425 lbs.—TWISDEN.

Standard weight of water = 62·321 lbs. per cub. foot.—SALE OF GAS ACT, 1859.

The experiments of the Standards Office of the Board of Trade show that a cub. inch of water weighs 252·286 grains instead of 252·458 grains, of which 5,760 go to the lb. Troy, and 7,000 to the lb. Avoirdupois, therefore a gallon of water now equals 277·463 cub. inches.—"THE ENGINEER," 1889.

U.S. standard gallon weighs $8\frac{1}{3}$ lbs., and contains 231 cub. inches.

A cub. foot of average sea water weighs 64 lbs.

A cub. foot of ice at 32° F. is 5 lbs. lighter than a cub. foot of water at same temperature.

Water in freezing expands $\frac{1}{9}$ of its bulk.

Weight per Cub. Foot.	*lbs.*
Ice	58·078
Water. maximum density $39\frac{1}{3}$° F.	62·4491
,, ,, 60° F.	62·39
,, ,, 212° F.	59·745
,, average, say	62·4

—SPON'S DICTIONARY.

* See lecture by the author on "Hydraulic Machinery, Past and Present," demy 8vo, 42 pp., and folding plate of illustrations. (Spon, 1s.)

875. USEFUL NUMBERS IN CONNECTION WITH WATER

Cub. feet × 6·232, say $6\frac{1}{4}$ = gallons.

Cub. feet per minute × 9,000 = gallons per 24 hours.

Head in feet × ·434 = lbs. per sq. inch.

Lbs. per sq. inch × 2·3 = feet head.

Tons × 224 = gallons.

Tons × 36 = cub. feet.

Diameter inches^2 ÷ 10 = gallons per yard.

Weight of sea water = 1·027 weight of fresh water.

168 gallons = 21 bushels = 27 cub. feet = 1 cub. yard.

Inches rain per hour × 60·5 = cub. feet per minute per acre.

876. COMPOSITION OF WATER (CAVENDISH, 1781).

Pure water (H_2O) is a chemical combination of the two gases Oxygen (O) and Hydrogen (H) in the proportion of two volumes of hydrogen to one volume of oxygen, or by weight one part of hydrogen to eight parts of oxygen.

877. SUMMARY OF HYDRAULICS.

The quantities discharged from different apertures of similar character vary directly as the areas, and as $\sqrt{\text{head}}$.

On account of friction, a small orifice discharges proportionally less water; and of several orifices having the same area, that with the smallest perimeter discharges most: hence a circular orifice is most advantageous.

Water issuing from a sharp-edged circular aperture is contracted at distance of ½ diameter from orifice, from 1 to . . . $\left\{\begin{array}{ll}\text{Bossut} & \cdot 666\\ \text{Venturi} & \cdot 631\\ \text{Eytelwein} & \cdot 64\end{array}\right.$ in area, called "vena contracta." Vein contracts more with greater head, therefore discharge slightly diminished below theoretical discharge due to altitude or head. When the orifice is not sharp-edged, the contraction is partially suppressed and the flow increased.

Water flowing from pipe of sectional area A into one of less sectional area a, will have a coefficient of contraction

$$= \frac{1}{\sqrt{\left(2\cdot 618 - 1\cdot 618\ \frac{a^2}{A^2}\right)}} = \cdot 618 \text{ when A is infinite, say a large tank.}$$

—RANKINE.

The discharge through a tube of diameter = length is the same as through simple orifice of equal diameter. The discharge increases up to a length of 4 diameters.

The discharges through horizontal conduit pipes are directly as the $\sqrt{\text{head}}$ and inversely as $\sqrt{\text{length}}$. To have perceptible and continuous discharge, head must not be less than $\frac{\text{length}}{1300}$. Vertical bends discharge less water than horizontal, and horizontal bends less than straight pipes.

Right angle bends 1 foot radius, with a flow of 32 feet per second, lose approximately 1 foot head, or for any other flow, say $\cdot 001\ v^2$.

The discharge through pipes varies approximately as diam^2.

In prismatic vessels twice as much is discharged from the same orifice if the vessel be kept full, during the time it would take to empty itself.

Water issuing from hole in thin plate,

Coefficient of contraction	0·64
" velocity	0·97
" discharge $d = c\,v$	0·62

878. TORRICELLI'S THEOREM.

Particles of fluid escaping from an orifice possess the same velocity as if they had fallen freely *in vacuo* from a height equal to that of the fluid surface above the centre of the orifice.

879. BERNOULLI'S THEOREM.

When a liquid is flowing in a pipe or channel, it possesses kinetic energy in virtue of its motion in addition to the potential energy due to its position and pressure, and the total energy is the sum of these three.

880. PRESSURE OF WATER.

Water transmits pressure equally in all directions (Pascal), and its own weight acts as additional pressure in proportion to the depth from surface. Pressure is perpendicular to containing surface. Water is only compressible to a very small extent. Pressure per unit of area is affected solely by depth, and is entirely independent of extent of surface.

Area of any portion of containing surface in square feet × distance of its centre of gravity in feet below surface of liquid × weight of liquid per cubic foot = pressure upon that portion of containing surface.

The pressure of the air is not able to sustain a column of water more than

34 feet high, hence water cannot by any possibility be raised by direct suction from a greater depth—the exact amount varies with the barometric pressure and the method employed.

If pressure be applied to a liquid entirely filling a closed vessel, that pressure will be transmitted equally to all parts of the liquid.

881. CENTRE OF PRESSURE

is the point of application of the resultant of the infinite number of parallel forces caused by the pressure of a liquid upon a containing surface at right angles to it, and increasing from the surface of the liquid downwards.

Shape of containing area.	*Distance of centre of pressure from surface.*
Square or rectangle	$\frac{2}{3}$ height.
Triangle, apex upwards	$\frac{3}{4}$,,
Triangle, apex downwards	$\frac{1}{2}$,,
Trapezium—	
Side a at surface, side b at base and parallel	$\frac{a + 3b}{a + 2b} \times \frac{h}{2}$
Circle	$\frac{5}{8}$ diameter.

Note.—When the area considered is below the surface of the liquid the centre of pressure will be nearer the centre of gravity of the area, and at infinite depth will coincide with it.

General expression for depth of centre of pressure of any plane area

$$= \frac{\Sigma\, w z \alpha \cdot z}{\Sigma\, w z \alpha} = \frac{\Sigma\,(z^2 \alpha)}{\Sigma\,(z \alpha)}$$

where α = area of any small portion between horizontal lines, and z its depth below the surface. The pressure on the whole area $= w\,\Sigma\,(z \alpha)$.

882. THE HYDROSTATIC ARCH.

The hydrostatic arch is a linear arch suited for sustaining normal pressure at each point proportional, like that of a liquid in repose, to the depth below a given horizontal plane; and is sometimes called "the arch of Yvon-Villarceaux," from the name of the mathematician who first thoroughly investigated the properties of its figure by the aid of elliptic functions. It is found to present some resemblance to a trochoid, with which, however, it is by no means identical.—RANKINE.

883. BUOYANCY AND FLOTATION POWER OF WATER.

Buoyancy is the upward resultant pressure of the water against a floating body. The centre of buoyancy is the centre of gravity of the displaced water.

When a solid body floats on a liquid, the weight of the liquid displaced is equal to the weight of the body.

When a heavy body is immersed in water, it displaces an equal bulk. and loses weight equal to the weight of water displaced.—ARCHIMEDES.

$$\text{Specific gravity} = \frac{\text{weight of body in air}}{\text{weight of equal bulk of water}},$$

or

$$= \frac{\text{weight of body in air}}{\text{weight in air} - \text{weight in water}}.$$

Solid cast iron loses 14½ per cent. of its weight when immersed in water.

884. HYDROSTATIC PARADOX.

"Any quantity of fluid, however small, may be made to balance and support any quantity or weight, however great."

Thus the water in a 3-inch pipe from a tank on the top of a building may support a load of many cwts. in the cradle of a hoist.

885. PRINCIPLE OF ARCHIMEDES.

When a body is immersed in water it loses weight, and the loss of weight is equal to the weight of the water displaced by the body.

The same title is also given to the following theorem:

If a body be immersed in a liquid, every portion of the surface of the body is subjected to a pressure acting at right angles to it, and varying proportionally to the depth.

886. PASCAL'S PRINCIPLE.

"If a vessel full of water, closed on all sides, has two openings, the one a hundred times as large as the other, and if each be supplied with a piston which fits exactly, a man pushing the small piston will exert a force which will equilibrate that of a hundred men pushing the piston which is a hundred times as large, and will overcome that of ninety-nine. And whatever may be the proportion of these openings, if the forces applied to the pistons are to each other as the openings, they will be in equilibrium."—BLAISE PASCAL'S "EQUILIBRIUM OF LIQUIDS."

887. COMPRESSIBILITY OF WATER.

Water is popularly supposed to be incompressible, but "If the water of the ocean were to suddenly cease being compressible, the result would be that 4 per cent. of the habitable land on the globe would be submerged, because the mean depth of water would be raised by 116 feet."—TAIT.

By another account Professor Tait calculated that with a depth of 6 miles the water level was 620 feet lower than it would be without compression. Assuming the compression to be proportional to the pressure this would make the compression of water in a cylinder 1 foot long with a pressure on the ram of 1 ton per sq. inch approximately ·0005 of an inch.

It is also said that water is compressed $\frac{1}{20000}$ of its bulk by 1 atmosphere of pressure = 14·7 lbs. per sq. inch, and that if all the air is expelled from the water the compression is only $\frac{1}{46000000}$ of its bulk with the same pressure.

Water compresses $\frac{1}{70}$th of its volume under a pressure of 2 tons per sq. inch.

888. COMPARISON OF DISCHARGE THROUGH VARIOUS APERTURES.

Theoretical velocity in feet per second

$$= \sqrt{\text{head in feet} \times 2g}.$$

Theoretical discharge being 1.

Short tube projecting into reservoir = ·5.

Orifice in thin plate, 1 inch diameter = ·62.

Tube 2 diameters long = ·82.

Conical tube approaching form of contracted vein = ·92.

" " " edges rounded off = ·98.

Or, say theoretical velocity feet per second = 8·02 $\sqrt{\text{head}}$ feet.

Effective velocity through orifices of the form of vena contracta, well-placed sluices, large bridge openings, etc. . . . . = 7·5 $\sqrt{h}$.

" large vertical pipes and narrow bridge openings = 6·75 $\sqrt{h}$.

" sluices without side walls, dock gates, and mill stream sluices . . . . . . = 5 $\sqrt{h}$.

889. PRACTICAL DISCHARGE OF WATER.

h = head in feet.

c = discharge in cub. feet per minute.

a = area in sq. feet.

$$k = \text{constant} = \begin{cases} 450 \text{ for bridges, etc.} \\ 400 \text{ for short pipes, etc.} \\ 300 \text{ for ordinary sluices.} \end{cases}$$

$$c = k\,a\,\sqrt{h}, \quad h = \left(\frac{c}{k\,a}\right)^2, \quad a = \frac{c}{k\,\sqrt{h}}.$$

—BEARDMORE.

When the outlet is "drowned" the head will be the difference in level between water over inlet and outlet.

The practical discharge from different sized pipes having equal length and equal fall is found to be as $\frac{D^2 \times \sqrt{D}}{d^2 \times \sqrt{d}}$ to 1.

890. RAINFALL.

Upon a 20 years' average, along the western shores of England the usual rainfall is 40 to 50 inches; in many districts 50 to 75 inches; that of the southern coast 30 to 40 inches, while in the eastern counties it is less than 25 inches. In some exceptional positions, such as Seathwaite, in Cumberland, the average rainfall is 140 inches. In others, such as Hunstanton, in Norfolk, and in some parts of Lincolnshire, it is little more than 20 inches.

In any part of this country—

The wettest year's rainfall is 50 per cent. above the mean.

The driest year is 33⅓ per cent. less than the mean.

The driest two consecutive years will each have a rainfall 25 per cent. less than the mean.

The driest three consecutive years will be 20 per cent. less than mean.

—G. J. SYMONS.

891. PERCOLATION AND EVAPORATION.

Percolation and evaporation each average about 25 per cent. of the rainfall. Reservoirs in a wet locality should hold 150 days' supply, in a dry locality 200 days'.—G. H. HUGHES.

Mean evaporation of water from open surface in London—large body of water 21 inches per annum, small body 50 inches; rainfall during same period 25 inches.

892. SAND FILTERS.

Washed sand	12 in. to 1 ft. 6 in. thick.
Fine gravel ⅛-inch mesh	6 inches thick.
Small gravel ¼-inch mesh	6 ,,

Coarse gravel ½-inch mesh . . .	6 inches thick.
Large gravel 1-inch mesh . . .	6 „
Stones 2-inch mesh	6 „
	3 ft. 6 in. to 4 ft.

With a depth of water on top of 1 foot to 1 foot 6 inches the supply of filtered water is 500 gallons per yard super. per 24 hours.—G. H. Hughes.

893. STANDARD OVAL SEWER.

Diameter = D. Height = 1½ D. Radius of crown = ½ D.
Radius of sides = 1½ D. Radius of invert = ¼ D.

Condition.	*Sectional area of flow.*	*Wetted perimeter.*	*Hydraulic mean depth.*
Running full	$1 \cdot 149\ D^2$	3·96 D	·29 D
Filled to ⅔ height	$\cdot 756\ D^2$	2·39 D	·3163 D
„ ½ „	$\cdot 509\ D^2$	1·90 D	·2680 D
„ ⅓ „	$\cdot 285\ D^2$	1·38 D	·2065 D

⅔ capacity occurs at ·65 height.
½ „ „ ·55 „
⅓ „ „ ·44 „

The mean velocity of flow in an oval sewer running two-thirds full is greater than the mean velocity when running full owing to the excessive proportion of wetted perimeter in the latter case.

The carrying capacity or bulk of flow varies as sectional area of liquid × square root hydraulic mean depth.

894. LARGE TOWN SEWERS.

For sewers over 18 inches diameter :—

$$D = \frac{3 \log A + \log N + 8}{10}$$

D = diameter of sewer in inches.

A = area drained in acres.

N = distance in feet in which sewer falls 1 foot.

For sewers up to 18 inches diameter :—

$$D = \frac{3 \log A + \log N + 6 \cdot 8}{10}$$

—H. R. Asserson, Brooklyn. U.S.A.

895. DISCHARGE OF CIRCULAR DRAIN.

The discharge or "carrying capacity" of a circular pipe under no initial head varies as sectional area of flow × √hydraulic mean depth. The maximum discharge is found to occur when the centre angle to surface of flow is 78½°. Assuming pipe 1 foot diameter, P = wetted perimeter, A = sectional area of stream, H = hydraulic mean depth, C = carrying capacity. Then running full

$$P = D = 3\cdot 1416 \times 1 = 3\cdot 1416, \quad A = \frac{\pi}{4} D^2 = \frac{3\cdot 1416}{4} \times 1^2 = \cdot 7854,$$

$$H = \frac{A}{P} = \frac{\cdot 7854}{3\cdot 1416} = \tfrac{1}{4}, \quad C = A \times \sqrt{H} = \cdot 7854 \times \tfrac{1}{2} = \cdot 3927.$$

Filled to height shown by angle 78½°.

$$P = 3\cdot 1416 - 3\cdot 1416 \times \frac{78\cdot 5}{360}, = 3\cdot 1416 - 0\cdot 685 = 2\cdot 4566$$

$$A = \cdot 7854 - \cdot 05 = \cdot 7354, \quad H = \frac{\cdot 7354}{2\cdot 4566} = \cdot 3$$

$$C = \cdot 7354 \times \sqrt{\frac{\cdot 7354}{2\cdot 4566}} = \cdot 4044$$

$$\text{Ratio of discharge} = \frac{\cdot 4044}{\cdot 3927} = 1 \text{ to } 1\cdot 0298.$$

A circular drain therefore discharges 2·98 per cent. more when filled to ·88 of the height than when quite full, providing there is no initial head above the diameter.

Example.—With a pipe 1 foot diameter having a fall of 10 feet per mile the velocity when full will be 2·044 feet per second and the quantity carried will be 96·58 cub. feet per minute. With the same fall and the surface of flow at ·88 of height the velocity will be 2·246 feet per second and the quantity carried 99·1 cub. feet per minute.

896. TESTS OF STONEWARE DRAIN PIPES.

Diameter.	*Thickness.*	*Breaking weight per foot run.*	*Breaking weight per sq. ft. area occupied.*
4 in.	0·6 in.	3320 lbs.	2371 lbs.
6 in.	0·7 in.	1820 „	937 „
9 in.	0·9 in.	2336 „	834 „

These pipes were above ground. In another series of experiments on pipes about 1 foot below ground 6 inches to 24 inches diameter and $\frac{7}{10}$ to 1½ inches thick, the pipes cracked with a fairly uniform pressure of 2,800 lbs. per foot run applied on the surface.—A. Bowes.

897. VELOCITIES OF STREAMS.

s = surface velocity centre of stream in inches per second.
b = bottom ,, ,, ,, ,, ,,
m = mean velocity of whole stream ,, ,, ,,

$$b = (\sqrt{s} - 1)^2, \quad m = \cdot 8 \frac{s + b}{2}, \quad \text{or } m = \cdot 8\,(s - \sqrt{s} + \cdot 5).$$

—Du Buat.

$$b = (s + 1) - 2\sqrt{s}. \quad m = (s + 0{\cdot}5) - \sqrt{s}.$$ —Molesworth.

$$m = \cdot 705\, s + \cdot 001\, s^2.$$ —von Wagner.

$$m = s + 2{\cdot}5 - \sqrt{5\, s}.$$ —Beardmore.

$$m = \cdot 835\, s.$$ —Neville.

$$m = \tfrac{2}{3}\, s.$$ —Adams.

$$m = \cdot 653\, s.$$ —Baumgarten.

$$m = \frac{s\,(s + 7{\cdot}783)}{s + 10{\cdot}345}.$$ —Prony.

Velocities may be taken in feet per minute, or inches per second, but when a formula is complex it will only suit the unit for which it is designed.

Inches per second × 5 = feet per minute.

Mean velocity of running stream = velocity found by bottle sunk to level of cork and floating down midstream — constant. Constant = 15 per cent. for smooth wooden trough, 17 per cent. for brick channel, 29 per cent. for earth channel, 36 per cent. for rough mountain stream.—Alliance Electrical Co., Ltd.

898. DISCHARGE OVER WEIR OR TUMBLING BAY.

h = true head from sill to still surface in feet.
c = discharge in cub. feet per minute per foot width.

$$c = 214\,\sqrt{h^3}.$$

When the water passes the point where the constant head begins to deflect, with an appreciable initial velocity = v feet per second,

$$c = 214\,\sqrt{h^3 + \cdot 035\, v^2\, h^2}.$$

Santo Crimp's formula has the constant 195 instead of 214 in each case.

For small weirs:

l = length of weir or notch in inches.
g = gallons discharged per minute.
d = depth of head in inches.

$$g = 2l\sqrt{d^3}.$$

Width of notch should not exceed say 10 times depth of flow, and should be bevelled on down stream side.

Approximate rule:

h = height of flow on edge of rule over square notch or edge of horizontal weir.
c = cub. feet per minute per foot width.

h = 1 inch,	then	c =	5·10
$1\frac{1}{4}$	,,	,,	7·14
$1\frac{1}{2}$	,,	,,	9·23
$1\frac{3}{4}$	,,	,,	11·78
2	,,	,,	14·43

—HAWKSLEY.

Q = cub. feet per minute per foot width.
d = true head over weir in inches.
v = mean velocity of approach feet per second.

$$Q = 1 \cdot 06 \left\{(3d + v^2)^{\frac{3}{2}} - v^3\right\}$$

or approximately

$$Q = (3d + v^2)^{\frac{3}{2}}.$$ —W. POLE.

$$Q = 5 \cdot 5 \sqrt{d^3 + \cdot 8\, d^2 v^2}$$

—GALTON *and* SIMPSON.

$$Q = 4 \cdot 8 \sqrt{d^3 + \cdot 1875\, d^2 v^2}$$

(*Note.*—This gives 12 to 27 per cent. below true value).—BIDDER, HAWKSLEY, *and* BAZALGETTE.

Q = cub. feet per second.
b = breadth of notch in feet.
h = height of surface of still water above bottom of notch in feet.
c = a coefficient of discharge.
B = breadth of weir in feet.

$$Q = \tfrac{2}{3}\, c\,.\,b\,h\,.\,\sqrt{2\,g\,h} = 5 \cdot 35\, c\,b\,h\,\sqrt{h}.$$

If $b = \frac{1}{4}$ width of weir (the minimum advisable), $c = \cdot 595$
,, ,, whole width of weir . . . . $c = \cdot 667$

For any intermediate proportions $c = \cdot 57 + \dfrac{b}{10\,B}$.

—COTTERILL'S "APP. MECH."

L = length of notch in feet.

n = number of end contractions, being 0, 1, or 2, according to form of approach.

$$Q = 3 \cdot 33 \, (L - \tfrac{1}{10} \, n \, h) \, h^{\frac{3}{2}}.$$

—PROF. JAS. THOMSON.

899. FLOW OF WATER THROUGH TRIANGULAR NOTCH.

Prof. James Thomson, of Dublin, proposed the use of triangular notches for measuring the discharge over weirs, because the periphery always bears the same ratio to the area of the stream flowing over it, which is not the case with any other form.

$$Q = \frac{8}{15} \, c \, . \, \frac{b \, h}{2} \, . \, \sqrt{2 \, g \, h}.$$

When $b = 2 \, h$, $c = \cdot 595$, $Q = 2 \cdot 54 \, h^{\frac{5}{2}}$.

„ $b = 4 \, h$, $c = \cdot 620$, $Q = 5 \cdot 3 \, h^{\frac{5}{2}}$,

—COTTERILL'S "APP. MECH."

900. RIVERS, SEWERS, DRAINS, &c.

D = hydraulic mean depth in feet

of streams or pipes partly full $= \dfrac{\text{sectional area}}{\text{wetted perimeter}}$,

of pipes running full or half full only $= \dfrac{\text{diameter}}{4}$.

f = fall in feet per mile.

M = mean velocity in feet per minute.

d = diameter of pipe in feet.

l = length in feet.

h = head or fall in feet.

a = area of flow in sq. feet.

c = cub. feet per minute.

I = mean hydraulic inclination $= \dfrac{l}{h}$.

$$M = \sqrt{D \times 2 f} \times 55, \quad c = a \, M.$$

—BEARDMORE *and* EYTELWEIN.

$$M = \frac{6000 \sqrt{D}}{\sqrt{I}}.$$

—LESLIE.

$$c = \frac{2356 \sqrt{d^5}}{\sqrt{I}}.$$

—EYTELWEIN.

$$M = 92 \cdot 26 \sqrt{I \, D}.$$

901. EFFICIENCY OF HYDRAULIC WATER-RAISING MACHINES.

Hydraulic ram	·75
Turbines and pumps	·60
Overshot waterwheel and pumps	·55
Poncelet ,, ,,	·45
Breast ,, ,,	·44
Undershot ,, ,,	·28
	—G. H. Hughes.
Inclined chain pump	·38
Upright chain pump	·53
Chinese wheel	·58
Bucket wheel	·60
Pumps for draining mines	·66
Archimedean screw	·70
	—Morin.
Roots' Rotary Pump	·74

902. WATERWHEELS.

Undershot Wheel.—Float boards radial, or inclined 20° towards current when not used in tidal stream. Breadth may equal or exceed diameter. Maximum efficiency when velocity of wheel equals half velocity of stream.

Breast Wheel.—Floats shrouded or covered at the sides and curved to form buckets. Breastwork of masonry built up round wheel as high as centre line. Stream led down a masonry slope to act on wheel by momentum and gravity. Suited for moderate supply of water and fall of 6 or 8 feet.

Overshot Wheel.—Floats formed into buckets. Water led in trough to top of wheel. Ratio of width to diameter usually small. Requires less water to drive it than the other forms, and is more than twice the power of an undershot wheel of same size. Fall must not be less than diameter of wheel. Smeaton found that in ordinary wheels the velocity of circumference should not exceed 3 feet per second.

Poncelet Waterwheel.—Undershot, floats curved to meet stream, maximum effect when velocity of stream = $2\frac{1}{2}$ times velocity of wheel. Modulus = ·7.

Pelton Wheel.—Cast iron wheel with buckets or cups, of gun metal in small sizes and chilled cast-iron in large sizes, receiving high pressure jet of water. Each cup divided into two parts by curved septum to split jet and release it without velocity. Modulus = ·7.

The theoretical velocity at centre of cups should be half the velocity of jet and efficiency then about 80 per cent., but in an experiment upon a small model wheel the maximum efficiency of 56 per cent. occurred with velocity of cups ·38 velocity of jet. With so small a wheel the friction would naturally absorb a large proportion of the power.

A Pelton wheel (P. Pitman, Stanley Road, Halifax) having an efficiency of 75 per cent., using 1,000 gallons of water at 700 lbs. pressure, will yield about 16,000,000 foot-lbs., or approximately 6 kilowatt-hours. So that, compared with an electric motor with electricity at 4d. a unit and water at 2s. per 1,000 gallons, the two motors would be on the same basis as regards cost.

903. TURBINES.

Fourneyron's (1827).—Water admitted in centre of wheel, passing along curved guides, and discharged at circumference against guides curved in opposite direction.

Thomson's Vortex Wheel.—Water admitted at circumference and discharged at centre, can be fixed above tail-race up to 30 feet, power being obtained by suction.

Fontaine's and Jonval's Parallel Flow Wheels.—Water admitted above through fixed inclined vanes and discharged below, axis vertical, inclined vanes on wheel with angle reversed.

Horizontal incased turbine.

H = head in feet or height of fall.

Q = quantity of water available in cub. feet per minute.

B = brake horse power, allowing 75 per cent. efficiency.

$$B = \frac{H\,Q}{706}, \quad Q = \frac{706\,B}{H}, \quad H = \frac{706\,B}{Q}.$$

904. HYDRAULIC RAM.

Where a fall of water of not less than 2 feet can be obtained through an inclined supply pipe, the hydraulic ram may be used for raising water to a considerable height, say 150 feet, without the intervention of other machinery. The action is as follows:—The water passing through the drive pipe (also known as the flow, fall, supply, or injection pipe) gradually increases in velocity until it suddenly closes the pulse valve through which it is escaping, when a small quantity is forced by the momentum of the bulk through the delivery valve into the air vessel, and thence into the delivery pipe or rising main. The escape valve being then relieved, the supply water again flows through until its velocity is sufficient to close the valve. This alternate motion is repeated as long as the conditions remain unaltered. The pulsations vary from 30 to 100 per minute, according to the fall of water.

Average proportions and results are:

D = diameter of fall pipe in inches.
f = fall in feet.
d = diameter of rising main in inches.
h = head of ditto in feet.
G = gallons per minute to work ram.
g = gallons raised in 24 hours.

h does not generally exceed 50 f.

$$d = \tfrac{1}{2} D. \qquad G = 3\ D^2.$$

$$g = 3000\ D^2 \frac{f}{h}.$$

A difference of level between the drive pipe and escape pipe of not less than 8 feet is desirable, length of drive pipe $2\frac{1}{2}$ to 3 times difference of level, and water raised 20 to 30 times difference of level.

By another rule:—

For small rams the length of drive pipe should be equal to head of water in delivery pipe, and as the water in drive pipe is only flowing about one-third of the time the pipe should be calculated for 3 times the required quantity.

g = gallons per minute to be lifted.
l = length of delivery pipe in feet.
h = height of lift in feet.
f = extra head in feet to be added for friction.
d = diameter of delivery pipe in inches.
G = gallons per minute required in drive pipe.
H = head of water in do. in feet.
L = length of do. in feet.
D = diameter of drive pipe in inches.
C = constant by d'Aubuisson's formula.

$\frac{h+f}{H}$	C	$\frac{h+f}{H}$	C
4	·86	10	·53
5	·79	11	·49
6	·73	12	·45
7	·68	13	·41
8	·63	14	·37
9	·58		

$$f = \frac{g^2 l}{720\, d^5}, \qquad L = h$$

C from table opposite value of $\frac{h + f}{H}$

$$G = \frac{g\,(h + f)}{H \times C}, \qquad D = \sqrt[5]{\frac{(3\,G)^2 \times \frac{1}{3} L}{H}},$$

$$g = \frac{G \times H}{h + f} \times C, \qquad d = \sqrt[5]{\frac{g^2 \times \frac{1}{3} l}{f}}.$$

When d and f are both unknown f must be assumed for trial.

905. CENTRIFUGAL PUMPS.

Power required to drive them varies as the 1·5 power of the lift.

a = gallons lifted per minute.

H = lift in feet.

$$\text{I.H.P.} = \frac{10\, a + H^{1 \cdot 5}}{2 \times 33{,}000}.$$

Speed at periphery = $8\,\sqrt{H}$ = feet per second.—A. Hanssen.

Note.—$H^{1 \cdot 5} = H^{\frac{3}{2}} = \sqrt{H^3}$.

W = weight of water passing per second in lbs.

v = velocity in feet per second of blade tips.

r = extreme radius of blades in feet.

n = number of revolutions per minute.

h = height of elevation in feet.

$\frac{W v^2}{2 g}$ = work developed per second in foot-lbs.

k = coefficient of efficiency = ·65.

$$k\,\frac{W\,v^2}{2\,g} = Wh \quad \therefore \quad h = \frac{1}{2\,g}\,k\,v^2 = \cdot 0155\; v^2.$$

$$v = \frac{2\,\pi\,r\,n}{60} = \cdot 10472\; r\,n, \quad h = \cdot 00011\, r\, n$$

V = velocity feet per *minute* of blade tips.

$$V = 550 + 550\,\sqrt{h}.\text{—Appold.}$$

D = diameter of fan in feet.

H = head of water in feet from water level to water level + head corresponding with friction of pipes, etc.

S = speed of periphery of fan in feet per second.

Q = quantity of water lifted in cub. feet per minute.

S = $8\,\sqrt{H}$ in small fans, = $9 \cdot 5\,\sqrt{H}$ in large fans.

$$H = \frac{S^2}{64} \text{ in small fans, } = \frac{S^2}{90 \cdot 25} \text{ in large fans.}$$

$$D = c\sqrt{\frac{Q}{\sqrt{H}}} \text{ (} c \text{ varying from } \cdot 12 \text{ to } \cdot 18\text{).—Unwin.}$$

Radial vanes = ·46 efficiency.
Curved „ = ·68 „

There are generally six vanes. An 18-inch pump will work well with 20-ft. lift, and a 36-inch pump with 30-ft. lift. Twice the diameter of fan will discharge four times the quantity of water. Approximately the discharge in gallons per minute = diameter of fan in inches squared ÷ 0·12.

906. A MINERS' INCH OF WATER.

A miners' inch of water in California and elsewhere is the flow through an orifice in a vertical plane, of 1 sq. inch in area, under an average head of 6½ inches (6 inches above top of orifice) = 1½ cub. feet per minute.

In another account it was stated that approximately 500 miners' inches at 400 feet pressure = 500 horse-power, sufficient to drive 250 heads of stamps.

In another account a miners' inch was said to equal 2½ H.P.; it appears, therefore, that where water is delivered under pressure a miners' inch may have any head and the term relates to area of orifice only.

Then $\frac{100}{48}\sqrt{H}$ feet = cub. feet per minute per sq. inch area of outlet, for maximum flow without work.

907. OPEN WATER-COURSES.

An artificial water-course is known as a canal, aqueduct, flume, or leet according to its construction.

908. TREDGOLD'S HYDRODYNAMIC UNIT

was 1 cub. foot of water raised 1 foot high in 1 second = approximately 3,750 foot-lbs. per minute, and was given as the effective work of a man continuously employed. It does not appear to have come into general use.

909. PUMP HORSE POWER.

Q = quantity of water lifted per minute in cub. feet.
H = mean height of lift in feet.
G = gallons lifted per minute.

$$\text{H.P.} = \cdot 00189\, Q\, H = \cdot 0003\, G\, H.$$

910. AIR-LIFT SYSTEM OF WATER-RAISING.

Invented by Pohle of Arizona, introduced by Murray, London, 1894.

Compressed air forced down small pipe inserted in larger one or into boring, and in rising to surface brings water with it. An improved arrangement consists of taking air down separate pipe outside rising main, as flow of water is not impeded by sockets on air pipe. The method is often convenient for quantities less than 15,000 gallons per hour, but although initial cost of apparatus is small the method is not economical. High lifts involve a greater loss of efficiency. No definite rules are yet fixed for the proportions of the parts.

Approximate proportions :—

Air pressure lb. per sq. inch $= \frac{7}{10}$ depth to inlet.

Cub. feet air per minute per 1,000 gals. per hour $= \frac{1}{5}$ air pressure in lb.

$$\text{Diam. delivery pipe} = \sqrt[3]{\frac{\text{gals. per hour}}{100}}$$

$$\text{Diam. air pipe} = \sqrt{\frac{\text{gals. per hour}}{75}}.$$

911. EXAMPLE OF AIR-LIFT PUMPING.

A great advantage of this method is the fact that an ordinary borehole may be utilised and the machinery may be placed 500 yards away if necessary, as at Tunbridge Wells by W. H. Maxwell (*see* "Engineering," November 13th, 1903). In comparison with other methods the cost, including interest on capital, may be taken as follows for 100-ft. lift :—

	Per 1,000 *gals.*
Compressed air plant with air lift	2·233d.
Steam engine and borehole pumps	2·9d.
Cheap oil engine, plant, and do.	1·78d.

Approximate rules for proportion drawn from above source would be as follows :—

H = height of lift required from mean surface of water in feet.

I = immersion of air tube below level of water in feet.

$$I = 3\,H.$$

A = cub. feet of free air required per minute.

G = gallons of water required per minute.

$$A = \frac{G\,H}{125}.$$

a = diameter of air pipe in inches down centre of water pipe (to be sufficient to supply air under pressure at a velocity of 20 feet per minute).

b = diameter of borehole.

w = diameter of water pipe (rising main) in inches.

$$a = \tfrac{1}{6} b, \qquad w = \tfrac{1}{2} b.$$

Air compressors to be two-stage with intercoolers from 60 to 300 lbs. per sq. inch, and three-stage from 300 to 1,000 lbs. per sq. inch.

p = pressure of air per sq. inch.

H.P. = horse-power required to compress 1 cub. foot of air per minute to pressure p.

$$\text{approx. H.P.} = \cdot 08 + \frac{p}{750}.$$

Mixed air and water must not travel at less velocity than 6 ft. per sec.

With larger air compressors borehole could have been used as rising main.

912. HYDRAULIC MEAN GRADIENT.

The hydraulic mean gradient of a line of pipes depends upon their diameter, fall, and internal condition. If a series of vertical pipes, open at the top, were inserted at intervals along the line, the h.m.g. would be given by the various levels at which the water would stand.

To draw the h.m.g. for a series of pipes assume any given quantity of water to be passed through, say 100 gallons per minute, then find the head necessary for this quantity for each size of pipe according to its diameter and length. Then as the total head for assumed quantity is to total available head, so is head for assumed quantity in each length of pipe respectively to the proportion of total head for that length, the head giving the h.m.g.

913. DISCHARGE THROUGH PIPES FROM NATURAL HEAD.

——	*d.*	*c.*	*d.*	*c.*
H = head of water in feet	1	4·71	7	612·32
L = length of pipe in feet	$1\frac{1}{4}$	8·48	8	854·99
d = diameter of pipe in inches . . .	$1\frac{1}{2}$	13·02	9	1147·61
c = constant (see table)	2	26·69	10	1493·47
W = cub. feet discharged per minute . .	$2\frac{1}{2}$	46·67	12	2356·00
	3	73·50	15	4115·93
$W = \dfrac{c}{\sqrt{\dfrac{L}{H}}}.$	4	151·02	18	6493·14
	5	263·87	24	13328·0
	6	416·54	30	23282·0

—BEARDMORE.

$$W = 4 \cdot 71 \sqrt{\frac{d^5 H}{L}}, \quad d = \cdot 538 \sqrt[5]{\frac{L W^2}{H}}.$$

—EYTELWEIN.

G = gallons delivered per minute.

L = length of pipe in yards.

$$d = \tfrac{1}{3} \sqrt[5]{\frac{G^2 L}{H}}, \quad G = \sqrt{\frac{(3\,d)^5 H}{L}}.$$

—HAWKSLEY.

r = hydraulic mean depth in feet.

s = sine of inclination = $\dfrac{\text{total fall}}{\text{total length}}$:

v = velocity feet per second.

$v = 140 \sqrt{r\,s} - 11 \sqrt[3]{r\,s}, \quad W = 47 \cdot 124\, d^2 v.$—NEVILLE.

Q = discharge cub. feet per second.

R = hydraulic radius in feet.

A = area of pipe in sq. feet.

S = hydraulic gradient in terms of the line of slope.

N = coefficient of roughness of internal surface of pipe and other irregularities, say ·013.

$$Q = \frac{V R}{N} \left\{ \frac{M + 1 \cdot 811}{M + V R} \right\} A \sqrt{R S}.$$

$$M = N \left\{ 41 \cdot 6 + \frac{\cdot 00281}{5} \right\}.$$

—KUTTER.

This is for the calculation of mains as ordinarily laid.

H = head in feet (between surfaces of water at each extremity).

h = actual length of pipe in feet.

v = velocity in feet per second.

d = diameter of pipe in feet (1 to 4 feet).

Riveted wrought iron pipes fairly smooth.

$$\frac{H}{h} = \frac{0 \cdot 0004 \times v^{1 \cdot 87}}{d^{1 \cdot 4}}$$

$$\text{whence } \log v = \frac{\log H + 1 \cdot 4 (\log d) - (\log h + \log 0 \cdot 0004)}{1 \cdot 87}$$

For very rough pipes

$$\log v = \frac{\log H + 1 \cdot 1 (\log d) - (\log h + \log 0 \cdot 0007)}{2}.$$

—PROF. UNWIN.

914. FRICTION OF WATER IN PIPES.

h = head in feet, d = diameter in inches,

l = length in feet, v = velocity in feet per second,

$$\text{Effective head} = \frac{4}{5} \frac{h}{\frac{l}{50} + d},$$

or allow $\frac{1}{3}$ to $\frac{1}{4}$ more diameter than is theoretically required for the quantity.

—BIRD *and* BROOKE.

10 to 12 feet head is absorbed in friction per mile of pipe.—BATEMAN.

L = length in miles.

V = velocity in feet per second.

D = diameter of pipe in feet.

H = loss of head in feet by friction.

$$H = \frac{2 \cdot 25\, L\, V^2}{D}.$$

—BOULTON *and* WATT.

H = head in feet required to overcome friction.

L = length of pipe in feet.

d = diameter of pipe in inches.

V = real velocity of water in feet per second.

$$H = \left(0 \cdot 0144 + \frac{0 \cdot 01716}{\sqrt{V}}\right) \frac{L\, V^2}{5 \cdot 367\, d}.$$

—WEISBACH.

$$H = \frac{L}{d} + \frac{4\, V^2 + 5\, V - 2}{1200}.$$

—W. COX.

By experiment at the London Docks the friction in the accumulator mains at 700 lbs. per sq. inch is equivalent to a loss of pressure of 1 lb. per 100 feet.

—ROBERT CARR.

915. FRICTION OF BENDS.

h = head in feet to overcome resistance due to bend or curve.

r = inside radius of pipe in feet or inches.

R = radius of axis of curve in same unit.

v = mean velocity of flow in pipe feet per second.

g = acceleration of gravity = 32·2 feet per second.

Square right angle bend, $h = \frac{0 \cdot 98\, v^2}{2\, g}$.

Rounded right angle bend, $h = [0 \cdot 131 + 1 \cdot 847\, (r^7 \div R^2)]\, v^2 \div 2\, g^2$.

—WEISBACH.

D = internal diameter of pipe.

L = length of straight pipe offering same resistance as the extra amount due to bend.

l = length of curved pipe.

R = radius to centre line of bend (not less than 2½ times diameter of pipe).

$$L = 12 \cdot 85\, l \left(\frac{D}{2\,R}\right)^{\cdot 88}$$

For bend of 2½ diameters, $L = 3 \cdot 378\, l$.—C. W. L. Alexander.

916. WATER SUPPLY.

The approximate proportions for pipes and head, to include allowance both for velocity and friction, are given by the following formulæ:

$$v = 50\sqrt{\frac{d\,H}{L + 50\,d}}, \qquad H = \frac{\left(\frac{v}{50}\right)^2 (L + 50\,d)}{d}$$

where

$$\left.\begin{array}{l} H = \text{head or fall} \\ d = \text{diameter of pipe} \\ L = \text{length of pipe} \\ v = \text{velocity per second} \end{array}\right\} \text{all in feet.}$$

To find accurate supply by given head through given lengths of various size pipes, assume a probable flow, then find the head necessary to produce the velocity in the smallest pipe, thus $H = \frac{G^2}{215\,d^4}$: Then add the head necessary to overcome friction in each length of pipe separately by the formula $H = \frac{G^2\,L}{240\,d^5}$, where H = head in feet, G = gallons per minute, L = length yards, d = diameter inches. If the total found thus does not agree with the given head the true discharge will be the assumed discharge × square root of true head ÷ square root of head found above. This is without allowing for bends. If delivering at a jet, the diameter of jet must be taken in first formula.

Water companies mains are generally calculated for a velocity of 2 to 3 feet per second, with a minimum diameter of 3 inches.

917. HYDRAULIC JETS.

d = diameter in eighths of an inch.

h = height of jet in feet.

H = head in feet after deducting that due to friction;

G = gallons discharged per minute.

$$h = H - \cdot 0125 \frac{H}{d}. \qquad G = \cdot 24\, d^2 \sqrt{H}.$$

The probable height of a jet

$$= \text{head to produce velocity} - \frac{(\text{head to produce velocity})^2}{80 \text{ times diameter of jet in } \frac{1}{8}\text{ths.}}.$$

918. FIRE HOSE AND FITTINGS

Diameter of leather hose $2\frac{1}{2}$ inches, reduced by "branch" pipe to $1\frac{1}{2}$ inches and by nozzle to $\frac{7}{8}$ inch with a length of $\frac{3}{4}$ inch parallel. Pressure at engine 150 lbs. per sq. inch, at nozzle 120 lbs. per sq. inch. Maximum height of jet 135 feet delivering 180 gallons per minute.

D = diameter of nozzle in eighths of an inch.
G = gallons discharged per minute.
H = actual pressure at nozzle in feet head.
h = maximum height of jet in feet.

$$D = \sqrt{\frac{G}{\cdot 24 \sqrt{H}}} \qquad G = \cdot 24\, D^2 \sqrt{H}.$$

$$H = \left(\frac{G}{\cdot 24\, D^2}\right)^2 \qquad h = H - \frac{\cdot 0125\, H^2}{D}.$$

The maximum pressure that can be used with advantage is 40 feet head for each $\frac{1}{8}$th inch diameter of jet. The quantity thrown varies as diam.^{2} of nozzle.

Sprinkler systems for fire protection require a large quantity of water, approximately 1 gallon per inch run of perforated pipe per minute.

919. TESTS OF METAL FOR PIPES.

The American Waterworks Association recommends test bars for proving the quality of metal for pipes to be 26 inches long, 2 inches wide, 1 inch thick, loaded in centre between supports 24 inches apart, to have a deflection of not less than ·25 inches before breaking, a transverse strength of 1,900 lbs., and tensile strength of 20,500 lbs. The pipes to be sounded with a 3-lb. hammer while under test pressure of 300 lbs. per sq. inch.

Socket-pipes should be cast at an angle of not less than 5° from the horizontal for every inch diameter, and always socket end downwards. When specified to be cast on end, or cast vertically, an angle of 30° is sometimes claimed as sufficient.

920. FREEZING OF WATER.

Water at 39·1° F. is at its point of maximum density. The specific gravity of ice (pure distilled water at 39·1° F. being 1) is ·865, therefore, on conversion to ice, water expands $\frac{1\cdot000}{\cdot865}$ of its bulk = 1·156 times = 15½ per cent. In pipes containing freezing water the expansion of the ice during solidification may be taken as transverse; and, if no leakage (or compression of locked-in air) takes place, the diameter must increase by $\sqrt{1\cdot156} = \frac{75}{1000}$. This would be roughly equivalent to a stretch 75 times greater than that at the elastic limit, so that no cylindrical pipe could withstand the effect of freezing when completely filled with water, but possibly an oval section might enable it to do so temporarily.

In England a depth of 2 feet 6 inches from the surface of the ground to top of pipe is usually sufficient to prevent freezing. Where the water may remain quiescent for 24 hours, 3 feet is necessary.

921. DIMINUTION OF BULK UNDER PRESSURE.

Water is said to diminish in bulk by ·000003 per lb. per sq. inch pressure, but the curve of pressure would probably rise more rapidly than the compression.

922. LONDON HYDRAULIC POWER COMPANY.

Minimum charge for water at a pressure of 700 lbs. per sq. inch £1 5s. per machine per quarter—i.e., for 3,000 gallons or less.

Gallons.	*Per* 1,000.
3,000 to 4,000	8s. 0d.
4,000 ,, 5,000	7s. 0d.
5,000 ,, 10,000	6s. 0d.
10,000 ,, 30,000	5s. 0d.
30,000 ,, 50,000	4s. 0d.
50,000 ,, 100,000	3s. 0d.
100,000 ,, 300,000	2s. 3d.
above 300,000	2s. 0d.

For minimum supply of 500,000 gallons per quarter, £50; and for excess, 1s. 6d. per 1,000.

For minimum supply of 2,500,000 gallons per quarter, 1s. 6d. per 1,000.

For minimum supply of 3,750,000 gallons per quarter, 1s. 4d. per 1,000.

Consumers taking their supply in more than one building will be charged the same rates *en bloc*.

For continuously running motors not less than ½ horse-power, with a guaranteed minimum per brake horse-power of

Gals. per. qr.	*Per* 1,000.
30,000	2s. 0d.
60,000	1s. 8d.
100,000	1s. 6d.

180 gallons per hour is supplied for each brake horse-power required.

The lowest rate of 1s. 6d. per 1,000 gallons will give 555 brake horse-power hours per quarter, or an average of about 7½ hours per day for 75 days per quarter divided over a number of machines.

For Fire Hydrants :—

Minimum charge 10s. per hydrant per quarter.

For first three hours' use	£5
For each hour after	£2
Breaking seal and testing	10s.

923. HYDRAULIC PRESSURE ACCUMULATOR,

invented by Lord Armstrong in 1850, consists of vertical cylinder and ram, to the crosshead of which a load of 20 to 120 tons is hung to create the pressure necessary for working the machinery, obviating the use of a high tower giving a natural head of water.

The load is usually contained in a cylindrical casing. Clean washed heavy Thames ballast, weighing 27 cwts. per cub. yard, is the cheapest and best procurable in London. Where convenient, railway ballast may be used. Iron slag is sometimes used ; it has the advantage of weight, and therefore occupies less space, but is expensive and very awkward to handle. Copper ore slag is not suitable, owing to the galvanic action set up. Water has been used for ballast where the pressure is required to be varied occasionally. Clay has also been used in its natural state, but is better when burnt. Iron kentledge, brickwork, cast-iron blocks and direct steam pressure have also been used by various manufacturers for producing the load.

The accumulator is a limited reservoir of power enabling the steam engine to work at the average speed requisite to supply machinery working intermittently. The capacity is equal to the possible excess of water required by the machinery over that supplied by the engine, in a given time.

924. PROPORTIONS FOR HYDRAULIC ACCUMULATORS.

d = diameter of ram in inches.
l = length of stroke in feet.
D = diameter of casing in feet.
L = length of casing in feet.
W = load on foundations in tons.
c = contents of cylinder in cub. feet.
g = contents of cylinder in gallons.

$$l = 1\tfrac{1}{4}\,d, \qquad L = l + 3\tfrac{1}{2} \text{ in.},$$
$$D = \sqrt{5\,d}, \qquad W = 0{\cdot}3\,d^2,$$
$$c = {\cdot}0068\,d^3, \qquad g = {\cdot}0425\,d^3$$

Gallons capacity.

10-inch	=	$42\frac{1}{2}$ gallons.	16-inch	=	174 gallons.
12 ,,	=	$73\frac{1}{2}$,,	18 ,,	=	248 ,,
14 ,,	=	$116\frac{1}{2}$,,	20 ,,	=	340 ,,

925. FRICTION OF ACCUMULATORS.

P = pressure in lbs. per sq. inch taken at half stroke, accumulator rising slowly.
p = pressure in lbs. per sq. inch, accumulator falling slowly.
d = diameter in inches.
f = friction of ram in lbs. per sq. inch.

$$f = \frac{P - p}{2}.$$

At the Marseilles Docks the friction of a 17-inch accumulator amounted to 7·355 lbs. per sq. inch, or not quite 1 per cent. of the gross load.—Hawthorn.

At Scottish Wharf, London, the friction of a 17-inch accumulator was 10 lbs. per sq. inch.

From experiments at Liverpool and Birkenhead, the difference of pressure with the accumulator rising or falling was about 30 lbs. per sq. inch, of 15 lbs. for the single friction.

Generally, $$f = \frac{170}{d}.$$

926. FRICTION OF CUP LEATHERS.

Depth of leather, or length of ram exposed to pressure sideways, makes no difference. Friction increases as the pressure increases. Friction of leathers, under same pressure, of different diameters, increases in direct proportion with the diameters, or with the square root of the respective gross loads.

d = diameter. p = lbs. per sq. inch.

Gross friction in lbs. = $\cdot 0315\, d\, p$.

Friction per cent. of total pressure = $\dfrac{4}{d}$.

—JOHN HICK.

927. AIR ACCUMULATORS.

W = working capacity in cub. feet of water.

C = mean capacity for air in cub. feet.

a = cub. feet air required at atmospheric pressure to charge accumulator.

p = mean pressure in lbs. per sq. inch.

P = maximum " " "

P′ = minimum " " "

$$P = \frac{p}{1 - \dfrac{W}{2\,C}}. \qquad P' = \frac{p}{1 + \dfrac{W}{2\,C}}.$$

$$C = \frac{P'\,W}{2\,(p - P')}. \qquad a = C\,\frac{p}{15}.$$

May be proportioned as follows:

D = inside diameter in feet.

L = inside length in feet.

$$D = \sqrt{\cdot 4244\,W}. \qquad L = 11\,D. \qquad C = 3\,W.$$

Total capacity divided thus:

Air under maximum pressure . . . . = $\frac{15}{22}$

Water " " . . . . = $\frac{6}{22}$

Margin from level of outlet to lowest water-level . . . . = $\frac{1}{22}$

If p = 700, then P = 840, and P′ = 600.

928. DELIVERY OF WATER IN PIPES.

v = velocity in feet per second through pipe.

a = area of pipe in sq. inches.

d = diameter of pipe in inches.

W = discharge in cub. feet per minute.

$$W = \frac{v\,a}{2\cdot 4}, \qquad v = \frac{2\cdot 4\,W}{a}, \qquad a = \frac{2\cdot 4\,W}{v}.$$

Approximately:

$$W = \frac{v\,d^2}{3}. \qquad v = \frac{3\,W}{d_2}. \qquad d = \sqrt{\frac{3\,W}{v}}.$$

929. VELOCITY OF WATER THROUGH PIPES AND VALVES.

With an accumulator pressure of 700 lbs. per sq. inch, the natural velocity (theoretical) is 322·32 feet per second. It is found in practice that not more than $\frac{1}{10}$th of this can be obtained through the pipes and $\frac{1}{3}$rd through the valves, in order to maintain the proper speed for the machinery. The loss from friction in the pipes is about 1 lb. per sq. inch per 100 feet length, after they have been laid some time; 1 lb. additional for each bend, and 10 lbs. each branch.

In order to allow for the furring-up of the small pipes, it is not safe to reckon upon more than three times the diameter of pipe in inches as the velocity obtainable in feet per second. It is also usual to calculate the velocity through the valves at not more than 98 feet per second.

930. PRESSURE IN PIPE-MAINS.

Working pressure averages 700 lbs. per sq. inch when given by accumulator, but may be from 350 to 1,000 lbs.

700 lbs. per sq. inch = 549·78 lbs. per circular inch, equivalent to 1613·2 feet head.

All pipes subject to the accumulator pressure to be tested to 2,500 lbs. per sq. inch before leaving the works, and to 2,000 lbs. per sq. inch after being laid.

Water companies' pipes to be tested with a pressure equal to 500 feet head, and while under pressure to be sounded from end to end with a 5-lb. hammer.

Pressure in water companies' mains is at maximum between 2 and 3 A.M., minimum 6 A.M. to 6 P.M., variation, say from 10 to 60 lbs. per sq. inch.

931. VARIATION OF ACCUMULATOR PRESSURE DUE TO WORKING OF MACHINERY.

Normal pressure, say 700 lbs. per sq. inch. Average variation from 50 lbs. below to 100 lbs. above the normal pressure. Maximum variation 250 lbs. above and below, but this only occurs on a long line of pipe where the accumulator is at some distance from the machine.

932. MECHANICAL VALUE OF FLUIDS UNDER PRESSURE.

U = units of useful work in foot-lbs.

p = pressure in lbs. per sq. inch.

Q = quantity used in cub. feet.

M = modulus of machine, or coefficient of effect found by experiment, and varying with class of machine or arrangement.

$$U = 144\,p\,Q\,M.$$

933. MECHANICAL VALUE OF WATER UNDER ACCUMULATOR PRESSURE.

Theoretically the mechanical value of water under accumulator pressure of 700 lbs. per sq. inch (549·78, say 550 lbs. per circular inch) is 100,800 foot-lbs., or 45 foot-tons per cub. foot of water, irrespective of the time in which it is consumed; or 3·0545 H.P. per cub. foot per minute; or 1 H.P. requires ·32738 cub. feet per minute.

Approximately this equals 1 H.P. from 2 gallons of water per minute; but practically, allowing for all losses, about 3½ gallons are required; or 4 cub. feet (= 25 gallons) will give out 100 foot-tons in work.

Theoretically 1 gallon = 7·2 foot-tons of work, practically 10 gallons will lift 1 ton to a height of 40 feet at a cost of one penny. If cradle, cage, bucket or tub be used, the weight must be included and the efficiency will be thereby reduced.

934. COST PER H.P. PER HOUR OF PRESSURE WATER.

p = pressure in lbs. per sq. inch.

d = charge in pence per 1,000 gallons.

C = cost per gross H.P. per hour in pence.

$$C = \frac{85\cdot 69\,d}{p} = \text{say } \frac{86\,d}{p}.$$

935. COST OF PUMPING WATER UNDER ACCUMULATOR PRESSURE.

The average cost on the L. & N. W. R. of pumping water at 700 lbs. per sq. inch is found to be 1s. 1d. per 1,000 gallons = 6s. 9d. per 1,000 cub. feet. This includes fuel, stores, wages, and materials, etc., for maintenance and repairs of engines, boilers, and accumulator, interest and depreciation on buildings and machinery. The quantity of water is arrived at by counting the revolutions of engine and estimating the full capacity of the pumps.

—W. ADAMSON.

Millwall Docks. . . .	3s. per 1,000 gallons.
St. Katherine's Dock . .	10s. per 1,000 cub. feet, or 1·49d. per 100 foot-tons.
London Docks	10s. per 1,000 cub. feet, or 1·21d. per 100 foot-tons.
Cotton's Wharf . . .	1·89d. per 100 foot-tons.
Great Western Railway . .	1·23d. ,, ,,
Albert Dock, Hull . . .	1·25d. ,, ,,
Hull Power Co. . . .	4s. per 1,000 gallons.
Hull Dock Co. pay . . .	4s. do. + £15 per machine.

936. ATMOSPHERIC PRESSURE.

The average barometric pressure at sea level = 30 inches mercury = 34 feet water; so that pumps cannot draw by suction deeper than 34 feet, whatever be the arrangement of the mechanism; 27 or 28 feet is a usual maximum.

937. EFFICIENCY OF PUMPS AND ACCUMULATOR.

R = any number of revolutions of engine.
r = rise of accumulator in inches for same number of revolutions.
D = diameter of accumulator ram in inches.
d = diameter of pump in inches (piston if double-acting, ram if single-acting).
s = stroke of pump in inches.
n = number of pumps.

$$\text{Efficiency} = \frac{D^2 r}{d^2 s n R}.$$

$$\left.\begin{array}{l}\text{Loss per cent. of working capacity}\\ \text{of pumps}\end{array}\right\} = \frac{100\,\{(d^2 s n R) - (D^2 - r)\}}{d^2 s n R}.$$

With all parts in good order the loss in the pumps averages 5 per cent.

938. COEFFICIENT OF STEAM PUMPING ENGINE.

Example.—Horizontal high-pressure direct-acting pumping engine working against accumulator pressure of 700 lbs. per sq. inch, specified as 84 H.P. at 60 revolutions per minute. Two steam cylinders each 16 inches diameter × 20 inches stroke; 2 double-acting force pumps with piston, each 5·1 inches diameter = $\frac{1}{10}$ area of steam piston, and ram 3·6 inches diameter = $\frac{1}{2}$ area of pump. Boiler pressure 60 lbs. per sq. inch by gauge. Cut-off $\frac{5}{8}$ stroke.

Mean pressure by calculation = 56 lbs. per sq. inch, by indicator diagram 45 lbs. per sq. inch.

16 inches diameter = 201·06 sq. inches area, 5·1 inches diameter = 20·43 sq. inches area, 3·6 inches diameter = 10·18 sq. inches area.

$$\text{Power} = \frac{201\cdot06 \times 45 \times 2 \times 120 \times 1\frac{2}{3}}{33{,}000} = 109\cdot67 \text{ I.H.P.}$$

$$\text{Effect} = \frac{20\cdot43 \times 700 \times 2 \times 60 \times 1\frac{2}{3}}{33{,}000} - 5 \text{ per cent. loss} = 82\cdot33$$

E.H.P.

$$\text{Coefficient} = \frac{82\cdot33}{109\cdot67} = \cdot75,$$

or 75 per cent. on the indicated horse-power.

In connection with the above engine the following particulars may be useful :—Fly-wheel 9 feet diameter; two wrought-iron Lancashire boilers, 6 feet diameter × 20 feet long; two flues, each 28 inches diameter, with five Galloway tubes. Double-acting lift pump. Tank, 1,500 gallons, for return water. 18-inch accumulator, 23 feet stroke.

In another case the coefficient was found to be as high as ·82, but ·75 is more usual.

939. PACKING FOR FORCE PUMPS.

Cup-leathers (invented by Bramah) may be single, double, or treble. If single, the open end should be turned towards the delivery end of the pump. If double, they may be back to back, or both turned towards delivery end of pump. If treble, two should be back to back, and the third put as a duplicate to the one turned towards delivery end. In all cases the back of the leather should be closely supported by a washer curved to the shape of the leather. Double leathers back to back are generally used, and last from two days to four months, average say one month. Only the middle of the back of best oil-dressed hide is used.

Spun-yarn is sometimes used, the same as for glands of hydraulic machinery generally. It is plaited and formed into rings by splicing, soaked in tallow, and screwed up in a mould to form solid rings of exact size to fit pump.

Rope is sometimes used in the same way, being selected of the exact diameter required. The two latter methods are said to last from four to six months, but there is probably more leakage than with leathers.

940. SPEED OF PUMPING

depends entirely upon circumstances and provision made to resist shocks. Ordinary direct-acting pumping engines will run against accumulator pressure of 700 lbs. per sq. inch at a piston speed of 200 feet per minute without knocking. Large pumping engines lifting from wells run slower, and small pumps quicker, the extreme range is 150 feet to 450 feet per minute.

941. CLACK VALVES OF HYDRAULIC PUMPING ENGINE.

To ensure the valves closing rapidly the spindles of the pump valves should have indiarubber rings with alternate metal washers, of sufficient total thickness to undergo a small compression when the clack is shut, viz. :—

Suction clack, $\frac{1}{16}$ inch compression for each 5 feet head of water.

Middle clack, $\frac{1}{16}$ inch compression.

Delivery clack, $\frac{1}{8}$ inch compression.

When a valve is leaking the beat is heavier than usual.

942. INDIARUBBER.

Indiarubber washers are used to form a closing spring on mushroom clack valve spindles. By experiment they compress one-third of their thickness in ordinary work and the same amount of compression is obtained by a direct load of $1\frac{1}{2}$ cwt. per sq. inch on the indiarubber.

Say a 4-inch clack has 12 washers each 3 inch diam. $\times$ $\frac{3}{8}$ inch thick with $1\frac{1}{2}$ inch hole leaving an area of 5·3 sq. inches. These will be compressed from $4\frac{1}{2}$ inches to 3 inches while working, equivalent to a load of $8\frac{1}{4}$ cwt.

943. COST OF BOILERS FOR HYDRAULIC PLANT.

A pair of Cornish boilers in mild steel, each 17 feet 6 inches long by 5 feet diameter, with 3 feet furnace tube tapering to 2 feet 3 inches and 3 Galloway tubes, having also horizontal steam drum 6 feet 6 inches long by 3 feet diameter with communicating neck and bracket, will cost £275 delivered, and weigh $9\frac{1}{4}$ tons. The brick setting will weigh 45 tons, costing £50 for material and £25 for labour. They will evaporate about 1,250 lbs. of water each per hour. The drums are to give extra steam space owing to the intermittent working of the engine. The boilers should be covered with 2 inches thickness of non-conducting composition.

944. POWER REQUIRED TO WORK HYDRAULIC MACHINERY.

In hotels, wharves, etc., with several machines, allowance must be made for $\frac{2}{3}$ of the machinery working to half the full height every $1\frac{1}{2}$ minutes.

$\therefore$ power per minute $= \frac{2}{9}$ total capacity of machinery.

At wharf with several cranes, $\frac{2}{3}$ machinery full lift, every $1\frac{1}{2}$ minute.

$\therefore$ power per minute $= \frac{4}{9}$ capacity of machinery.

At railway goods stations, docks, etc., where many machines are idle at one time, say $\frac{1}{8}$ machinery full height every $1\frac{1}{2}$ minute.

$\therefore$ power $= \frac{1}{12}$ capacity of machinery.

At small wharves where cranes are rapidly worked, all machinery, full height every $1\frac{1}{2}$ minute.

$\therefore$ power $= \frac{2}{3}$ capacity of machinery.

945. APPROXIMATE CONSUMPTION OF WATER FOR LIFTING BY HYDRAULIC POWER.

Low pressure machinery :—

$$\frac{\text{Max. load tons} \times \text{lift in feet}}{\text{feet head}} \times 500 = \text{gallons per lift, allowing } \cdot 45$$

effective.

Accumulative pressure machinery :—

$$\frac{\text{Max. load tons} \times \text{lift in feet}}{4} = \text{gallons per lift, allowing } \cdot 56$$

effective.

946. HORSE POWER REQUIRED FOR CRANES.

$$\frac{\text{Tons lifted per hour} \times \text{feet lift}}{884} = \text{Actual H.P. of work done.}$$

Indicated H.P. required in engine averages 4 times H.P. of work done.

Hydraulic cranes while working consume about 1 H.P. per cwt. lifted. In a large plant the engine power should be about 1 H.P. to each ton of lifting power of cranes, etc.

947. PROPORTIONS OF HYDRAULIC PIPES.

For accumulator pressure of 700 lbs. per sq. inch :—inside diameter (d) in inches $+ 2 =$ thickness of metal in $\frac{1}{8}$ths. Filling pipes made by local firms, $\frac{1}{16}$ inch thicker.

Flanges oval, $2{\cdot}85\, d \times 1{\cdot}55\, d$ and $\frac{1}{2}\, d$ thick, with two square necked bolts each $\frac{1}{4}\, d$ in diameter for 5-inch pipes and upwards, or $d + 5 =$ diameter in $\frac{1}{8}$ths for 5-inch pipes and under.

Another rule for bolts is $2\,(d + 2) =$ diameter in $\frac{1}{8}$ths.

Gutta percha cord for joint rings is plastic at 151° F.

948. THICKNESS OF PIPES FOR HYDRAULIC ACCUMULATOR MAINS.

For 700 lbs. per sq. inch :—

Armstrong . . . . $t = \frac{d}{8} + \cdot 25.$

Brown . . . . $t = \frac{d}{6}.$

949. THICKNESS OF PIPES FOR WATER CO.'S MAINS.

H = head of water in feet.
d = inside diameter of pipe in inches.
t = thickness of metal in inches.
x = 0·37 for pipes less than 12 inches diameter.
0·5 „ from 12 to 30 „ „
0·6 „ „ 30 to 50 „ „
p = working pressure in lbs. per sq. inch.
r = inside radius of pipe in inches.
c = working strength of metal in lbs. per sq. inch.
= 3360 for cast iron, 500 for lead.

For 200 feet head :—

Hawksley . . . . $t = \cdot 18 \sqrt{d}.$

Unwin . . . . . $t = \cdot 11 \sqrt{d} + \cdot 1.$

Box . . . . . $t = \left(\frac{\sqrt{d}}{10} + \cdot 15\right) + \left(\frac{d\,\mathrm{H}}{25000}\right),$

or say $= \frac{\sqrt{d}}{10} + \cdot 15 + \frac{d}{125}.$

Molesworth . . . . $t = \cdot 000054\,\mathrm{H}\,d + x,$

or say $= \cdot 0108\, d + \frac{\sqrt{d}}{10}.$

Hurst (with coeff. corrected) $t = \frac{d\,p}{c}.$

Burnell . . . . . $t = \frac{p\,r}{c - r}.$

B. Latham . . . . $t = \cdot 2 \sqrt{d}.$

Rankine . . . . . $t = \frac{\mathrm{H}\,d}{12000}$, or $\sqrt{\frac{d}{48}}$, whichever is greater, with a minimum of $\frac{3}{8}$ inch.

Trautwine gives as the usual American practice,

$$t = \left\{\left(\frac{\frac{p}{m}}{f} \div 2\right) + 1\right\} \frac{\frac{p}{m}}{f} \times \frac{d}{2},$$

but suggests as an improvement,

$$t = \left\{\left(\frac{\frac{p}{m}}{8} \div 2\right) + 1\right\} \frac{\frac{p}{m}}{8} \times \frac{d}{2} + \cdot 3,$$

where m = cohesion of metal in lbs. per sq. inch, say 16,800.

f = factor of safety, say 6.

p = internal pressure in lbs. per sq. inch.

A common rule is $t = \frac{1}{5}\sqrt{d}$.

950. GENERAL RULES FOR THICKNESS OF CAST-IRON PIPES.

Unwin . . . $t = \cdot 5\, d \left(\sqrt{\dfrac{2775 + p}{2775 - p}} - 1\right)$.

Barlow . . . $t = \dfrac{\cdot 5\, d}{\dfrac{16000}{p} - 1} \times 5$ for safety.

H. Law . . . $t = \cdot 000104\, d\, \mathrm{H} + 0 \cdot 313$.

Campin . . . $t = \dfrac{p\, d}{6000} + \cdot 66$.

Adams . . . $t = \dfrac{d\, p}{6000} + \dfrac{\sqrt{p}}{100} + \dfrac{\sqrt{d}}{10}$ (+ ·125 for steam).

951. CAST IRON GAS AND WATER PIPES: ADMIRALTY (WORKS DEPARTMENT) RULES.

Thickness $\frac{5}{32}\sqrt{d}$ up to 50 lbs. per sq. inch static working pressure. From 50 to 150 lbs. per sq. inch $\frac{3}{16}\sqrt{d}$ with a minimum of $\frac{3}{8}$ inch.

Test pressures for a thickness of $\frac{5}{32}\sqrt{d}$ = 150 lbs. per sq. inch. For a thickness of $\frac{3}{16}\sqrt{d} = \left(150 + \dfrac{140}{\sqrt{d}}\right)$ lbs. per sq. inch.

952. NOTES ON PIPES.

Iron, composition, and lead pipes are measured by their inside diameter, brass and copper pipes by their outside diameter.

Wrought-iron pipes are bent by filling with sand and making red-hot, keeping the joint on the side of the bend.

A string of reels shaped like small barrels is used in bending lead pipe, to prevent flattening.

Copper and brass pipes when small may be filled with resin before bending.

Pipe bending machines are now sometimes used.

Red lead for pipe joints, etc., is made of red oxide of lead in powder mixed with boiled linseed oil, and sometimes with a little white lead. The graphite pipe joint compound (1s. per lb.), made by the Joseph Dixon Crucible Co., Jersey City, N.J., U.S.A., and 28, Victoria Street, London, S.W., makes as tight a joint and takes apart when old more readily.

953. LEAD AND YARN FOR SOCKET PIPES.

Diam. of pipe in ins.	*Weight of Lead.*	*Length of Yarn.*	*Diam. of pipe in ins.*	*Weight of Lead.*	*Length of Yarn.*
	lbs.	*ft.*		*lbs.*	*ft.*
3	2½	4½	7	7¾	8½
4	4	5	9	10½	10½
5	5	6½	12	19	15
6	6½	7½			

—J. WRIGHT CLARKE.

954. DR. ANGUS SMITH'S COMPOSITION FOR COATING PIPES.

Original recipe was 30 gallons coal tar, 30 lbs. fresh slaked lime, 6 lbs. tallow, 3 lbs. lampblack, 1½ lb. resin; to be well mixed, boiled 20 minutes and put on hot.

The modern practice varies, but a good mixture is 3½ barrels coal tar, ½ barrel coal oil, ½ barrel pitch, with 6 tons gas coke for heating pipes. Made and used as follows:—Into a wrought-iron tank long enough to take a 9-foot pipe, sufficient coal tar to half cover a pipe is put, then pitch beaten to a powder, and sprinkled on the tar, and coal oil poured on the pitch. After being thoroughly cleansed the pipes are heated to 180° to 200° F., or as hot as the hand can bear, put into the liquid separately, and turned over and over for 2 or 3 minutes, then placed at an angle to drain, with the lower end clear of the liquid. The above quantities will do about 1,000 pieces (bends, branches, and straight pipes), or say ¾ barrel coal tar to 100 9-foot lengths of 4-inch pipes. This method avoids risk from the liquid catching fire. The cost is about 5s. per ton.

For better work use linseed oil instead of coal oil, increase the temperature of the pipes to 500° to 700° F., or until plumbers' solder will melt when pressed against them, and leave them in the liquid for 10 minutes after turning them over.

955. NICKEL TUBING.

Nickel tubing is now made by the Benedict Burnham Manufacturing Company, of Waterbury, Conn., U.S.A., which presents a perpetually brilliant silvery surface, is not readily tarnished, and can be kept polished without being scraped away as occurs with nickel plated brass tube. It has 50 per cent. greater tensile strength than brass, and hydraulic tests show that 50 per cent. greater internal pressure is required to burst it. It is homogeneous and free from imperfections liable to produce fracture. It is ductile and can be coiled, bent, or flanged either hot or cold. These properties render it superior in every way for high-class exposed plumbing work, and wherever metal railing is subject to excessive and continual wear and rubbing, as for stores, banks, cafés, entrances to public buildings, clubs, theatres, etc., and for all places where a railing would be subjected to constant and severe use.

The tubes can be supplied in all sizes from ⅛ inch to 8 inches in diameter in a variety of gauges, as well as the usual fittings such as tees, ells, elbows, etc. Billets of the same alloy can also be supplied for special castings made to take the tube.

956. CISTERNS AND TANKS.*

Square Tanks.—Thickness of plates for cast iron, no plate less than ½ inch thick, and when exceeding 6 feet deep none less than ⅝ inch thick. Bottom and lower side plates by the formula:

$$\textit{Square tank, } t = \frac{d\sqrt{D}}{192}, \qquad \textit{Circular tank, } t = 0{\cdot}0015\,D\,r,$$

where t = thickness of plate in inches.
d = depth (width) of plate in inches.
D = depth of lower edge from surface of water in feet.
r = radius in feet.

Intermediate plates to be graduated between these thicknesses.

Bolts diameter = t, but not less than ¾ inch.

Flanges project 3 times diameter of bolts.

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* For full information on designing cisterns and tanks, see the author's "Designing Ironwork." 2nd Series, Part III. (E. & F. N. Spon, Ltd., 2s.).

957. HYDRAULIC PRESS WITH HAND-PUMP.

P = pressure in lbs. on handle of pump.

d = diameter of pump in inches.

l = effective leverage = $\frac{\text{power leverage}}{\text{resistance leverage}}$.

D = diameter of press in inches.

M = modulus or coefficient of press, say = ·8.

W = total load in lbs., or maximum effort of press.

$$W = P\,l\,\frac{D^2}{d^2}\,M.$$

Moseley ("Illustrations of Mechanics," p. 197) says, "The discovery of it [the hydraulic press] is usually attributed to Pascal; it belongs, however, to the celebrated Stevin, mathematician to the Prince of Nassau, the inventor of decimals."

958. HYDRAULIC FORGING PRESSES.

Hydraulic forging presses capable of exerting a pressure of 1,000 to 10,000 tons are used in large works for converting steel ingots into large forgings, etc. They are actuated by a pressure of between 2 and 3 tons per sq. inch to keep down the size of the rams. Smaller presses, exerting say from 25 to 250 tons, are worked at a pressure of 100 atmospheres, or 1,500 lbs. per sq. inch.

Ordinary presses are worked from the accumulator pressure of 700 lbs. per sq. inch, sometimes with the addition of an intensifier to give the final squeeze.

959. HYDRAULIC PRESS CYLINDERS.

d = diameter of ram in inches.

c = clearance between ram and cylinder.

t = thickness of cylinder in inches.

p = working pressure in lbs. or tons per sq. inch.

S = maximum tensile strength per sq. inch of material in same units.

$$t = \frac{2\,d\,p}{S}. \qquad c = \frac{d}{12}.$$

Bottom hemispherical inside and out, except flat part outside to stand on = $\frac{1}{2}\,d$ in diameter, and joined with easy radius.

Another rule :

P = bursting pressure in tons per sq. inch.

D = outside diameter in inches.

d = inside ,, ,,

S = tensile strength tons per sq. inch of material.

$$P = S\,\frac{D^2 - d^2}{D^2 + d^2}.$$

Another rule :

p = internal bursting pressure, lbs. per sq. inch.

r = inside radius of cylinder in inches.

s = ultimate tensile strength of metal per sq. inch.

say cast iron 18,000 lbs.

,, gun-metal 36,000 lbs.

t = thickness of metal in inches.

$$p = \frac{s\,t}{r + t}. \qquad t = \frac{p\,r}{s - p}. \qquad s = \frac{p\,(r + t)}{t}.$$

—P. Barlow, 1836.

p = internal working pressure lbs. per sq. inch.

s = ultimate tensile stress lbs. per sq. inch.

k = factor of safety, cast iron = 3.

d = internal diameter of cylinder in inches.

t = thickness of metal in inches.

ϵ = base of Napierian logarithms = 2·71828.

$$t = \frac{d\,p\,k}{2\,s}\left(1 + \frac{p\,k}{2\,s}\right). \qquad \text{—Reuleaux.}$$

$$t = \frac{d}{2}\left(\frac{\frac{p\,k}{s}}{\epsilon - 1}\right). \qquad \text{—Clark.}$$

$$t = \frac{d}{2}\left(-1 + \sqrt{\frac{3\,s + 2\,p\,k}{3\,s - 4\,p\,k}}\right). \qquad \text{—Grashof.}$$

Common rule :

Thickness of cylinder = radius of bore. This is supposed to be safe at 3 tons per sq. inch working pressure, but is really not safe at more than 2 tons per sq. inch with ordinary metal.

Permanent safe working pressure = ⅓ bursting pressure. Maximum working pressure allowable = ½ bursting pressure.

Note.—A press that worked occasionally up to ⅗ of its bursting pressure burst after 4½ years' use.

960. CRANES OVER FOOTWAYS.

In the City of London the regulations of the Streets Committee of the Public Health Department (revised to March 24th, 1904), so far as cranes are concerned, provide that—

(1) No . . . crane shall be erected until notice has been sent to the Town Clerk and a licence from the Corporation has been obtained.

(2) Every application for a . . . crane shall be accompanied by drawings, showing the dimensions, mode of fixing, total projection from the frontage line, and the height from the pavement to the underside of such . . . crane.

(9) Every crane shall be above the first floor level, shall not project more than 3 inches when flat against the building, and shall be used only during such hours as the Corporation may prescribe.

(12) All easements over the public ways shall remain only during the pleasure of the Corporation.

Note.—As it is practically impossible and unnecessary to make any outside crane to come within a space of 3 inches from the building, a liberal interpretation is placed upon these regulations, and it is understood that the bottom footstep alone is subjected to criticism, but public bodies ought to consult experts and not frame such impossible regulations.

961. HEIGHT OF LIFT FOR CRANES.

Wool warehouse cranes.—Height of lift = net height bottom floor to top floor + 6 feet. Underside of jib head sheave 10 feet 6 inches above level of top floor.

Coal cranes.—Minimum height of lift on floating wharf = 40 feet. Height of lift at fixed jetty 50 to 60 feet, depending upon the rise and fall of the tide and other circumstances. Height of lift to riverside hoppers 60 to 80 feet. An extra lift of say 10 feet is sometimes required to reach the coal tubs in corners of hold.

962. EFFECTIVE PRESSURE FOR HYDRAULIC CRANES AND HOISTS.

p = accumulator pressure in lbs. per sq. inch.

m = ratio of multiplying power.

E = effective pressure in lbs. per sq. inch, including all allowances for friction, but not for weight of moving parts.

$$E = p\,(\cdot 84 - \cdot 02\,m).$$

963. SPEED OF LIFTING WITH HYDRAULIC POWER.

Warehouse cranes and jiggers 6 feet per second.

Platform cranes and small luggage lifts, 4 feet per second.

Waggon hoists, 2 feet per second.

Hotel and office passenger lifts, 2 to 4 feet per second.

Large passenger hoists, over 50 feet lift, first and last 10 feet average 4 feet per second, intermediate height 8 feet per second. At the Blackpool Tower the hoists were designed to lift 325 feet in rather less than 1 minute, say average $5\frac{1}{2}$ feet per second.

Maximum desirable speed under any circumstances, 10 feet per second.

General formula for warehouse cranes:

W = load in tons.
h = height of lift in feet.
v = velocity in feet per second.

$$v = \frac{h}{W + 10}.$$

964. DIAPHRAGM REGULATOR FOR HYDRAULIC MACHINERY.

When a hydraulic crane or hoist works too quickly, and it is desired to reduce the speed to a safe limit, it is usual to partially close the stop valve; but when there is a risk of this being interfered with, a brass diaphragm, $\frac{1}{8}$th diameter thick and about $\frac{1}{8}$ inch at edge, is placed in a pipe joint near the working valves. The hole in the diaphragm should be tapered, the small side being next to the machine.

To find size:

A = area of lifting ram, sq. inches.
m = ratio of multiplying power.
s = speed of lifting with full load, feet per second.
p = accumulator pressure, lbs. per sq. inch.
a = area of small side of hole (large side = twice diameter of small side).

$$a = \frac{A\,s}{6\,m\,\sqrt{1 \cdot 932\,p - \cdot 046\,m}}.$$

For direct-acting passenger lifts a diaphragm is always required next to the cylinder.

Umney's rule :

For 700 lbs. per sq. inch.

D = diameter of lifting ram in inches.

d = diameter of hole in inches.

$$d^2 = \frac{D^2 s}{220\, m}.$$

For other pressures,

$$d^2 = \frac{D^2 s}{8 \cdot 34\, m \sqrt{p}}.$$

965. DIRECT-ACTING LOW PRESSURE HOISTS.

Friction in exhaust pipes, say 10 per cent.

„ of guides, glands, etc., say 6 per cent.

„ of counterweight, say 6 per cent.

966. COAL-WEIGHING CRANES.

To find maximum section of wrought-iron weigh-beam, rectangular bar single or double.

T = lbs. diagonal thrust on sheave-pin per cwt. of load.

L = effective leverage of thrust in inches, measured perpendicularly to main knife edge.

C = cwts. maximum gross load.

b = breadth, or combined breadth, of weigh-beam in inches.

d = depth of weigh-beam in inches at centre of motion.

$$b\, d^2 = \frac{T L C}{1000}$$

for a statical factor of safety of 7½ to 1.

Knife edges and beds should be of square cast steel, sharpened to 60° to 90° and tempered to a straw yellow. As the beam lifts, the rolling of worn or rounded knife edges increases the ratio of the leverage and makes the weights too heavy, but this is partly counteracted by the wear and tear of the weights.

Knife edges for heavy testing machines are made 1 inch long for each 5 tons load ; for coal-weighing cranes where the work is very rough 1 inch per ton is a more suitable proportion.

Cheese-weights should not exceed 5 inches diameter. To adjust them :— To reduce, drill out on underside ; to increase, drill out double the amount, undercut, add lead and caulk flush.

A cataract cylinder, or dash-pot, is useful at bottom of weight spindle to prevent excessive movement when at work.

967. BALANCES AND STEELYARDS.

A good balance must have the following properties :—

1.—The line joining the points of suspension of the scale pans must be perpendicular to the line joining the point of support and the mean centre of gravity of the beam and the scale pans (supposed collected at their respective points of suspension).

2.—The arms must be equal and the scale pans must have equal weight.

3.—The arms must be horizontal when the masses in the pans are equal.

4.—The balance must be sensitive.

5.—The balance must recover itself quickly—i.e., it must be stable.

In one sentence, it must be *true*, *sensitive*, and *stable*.

The conditions for truth are given above. The stability is increased by increasing the distance from the point of suspension to the centre of gravity. The sensitiveness is increased by lengthening the arms, but the longer the arms the greater the inertia and the slower the movement.

The conditions are not similar for the weigh-beams or steelyards of coal-weighing cranes. In these the two knife edges, and centre of sheave pin, must be in the same line, so that the relative leverage remains constant. With an inclined beam, when fulcrum is $\begin{Bmatrix}\text{higher}\\ \text{lower}\end{Bmatrix}$ the weights as designed to suit the leverage of the beam on the bench are too $\begin{Bmatrix}\text{light,}\\ \text{heavy,}\end{Bmatrix}$ and, if adjusted on the crane, when the beam commences to lift they become too $\begin{Bmatrix}\text{heavy}\\ \text{light}\end{Bmatrix}$ as the beam lifts, but are not required to be so $\begin{Bmatrix}\text{light}\\ \text{heavy}\end{Bmatrix}$ as first made. The difference of leverage $\begin{Bmatrix}\text{increases}\\ \text{reduces}\end{Bmatrix}$ as beam lifts, and the brasses would require to be $\begin{Bmatrix}\text{lighter}\\ \text{heavier}\end{Bmatrix}$ towards the bottom, but the cheese weights being all lifted at one time could not be correctly adjusted. With a horizontal beam —i.e., the centre of sheave pin and knife edge at tail end horizontal, and the fulcrum $\begin{Bmatrix}\text{higher}\\ \text{lower}\end{Bmatrix}$, the leverage on the crane will be the same as on the bench at starting, but as the beam lifts the difference of leverage $\begin{Bmatrix}\text{increases}\\ \text{reduces}\end{Bmatrix}$ with the same tendency in results as above.

968. LIFTING RAMS FOR HYDRAULIC CRANES.

W = load to be lifted in lbs.

w = weight of ram, crosshead, sheaves, and chain.

l = height of lift in feet.

m = multiplying power.

c = coefficient of effect = $\cdot 84 - \cdot 02\,m$.

a = area of ram in sq. inches.

s = stroke of ram in inches.

p = accumulator pressure in lbs. per sq. inch.

C = capacity of cylinder in cub. feet.

For horizontal cylinders:

$$a = \frac{W\,m}{p\,c}. \qquad C = \frac{W\,l}{144\,p\,c}.$$

For vertical cylinders:

$$a = \frac{W\,m + w}{p\,c}. \qquad C = \frac{W\,l + w\,s}{144\,p\,c}.$$

For inverted cylinders:

$$a = \frac{W\,m - w}{p\,c}. \qquad C = \frac{W\,l - w\,s}{144\,p\,c}.$$

969. TURNING RAMS FOR HYDRAULIC CRANES.

W = load in tons.

R = radius of crane in feet.

l = length between bearings in feet.

d = diameter of turning drum in feet.

p = accumulator pressure, lbs. per sq. inch.

m = multiplying power of turning cylinder (usually 2 to 1).

a = area of turning ram in sq. inches.

Alternative formulæ :—

$$a = \frac{120\,W\,R^2\,m}{l\,d\,p}. \qquad a = \frac{3000\,W\,R\,m}{l\,d\,p}.$$

$$a = \left(5906\,\frac{W\,R\,m}{l\,d\,p}\right) - 3\cdot 3.$$

970. HYDRAULIC CYLINDER COVERS.

Circular cover, sound grey cast iron, 19½ inches diameter inside cylinder, 25¾ inches over all, 2 inches thick, fractured all over by hydraulic pressure in cylinder of 700 lbs. per sq. inch. Radius of bolt holes 11 inches, 22 bolts.—A. H. TYLER.

From this, say, ultimate $p = \frac{3500\,t^2}{d}$, or safe $p = \frac{700\,t^2}{d}$, or for 700 lbs. per sq. inch $t = \sqrt{d}$.

971. AREAS OF VALVES FOR MACHINERY UNDER ACCUMULATOR PRESSURE.

A = area of lifting ram.
m = ratio of multiplying power.
v = velocity of load in feet per second.
V = velocity of water through valve, feet per second.
W = weight of ram, crosshead, sheaves, chain, etc., in lbs.
a = area of lifting valve (mitred spindle).
a_1 = area of lowering valve (mitred spindle).

$$a = \frac{A\,v}{m\,V}. \qquad a_1 = \frac{A\,v}{m\sqrt{13 \cdot 8\,\frac{W}{A}}}.$$

When cylinder is horizontal, then $\frac{W}{700}$ = area of returning ram.

972. AREAS OF PORTS IN SLIDE VALVES.

v = velocity of load in feet per second.
m = ratio of multiplying power.
A = area of ram in sq. inches.

Area of pressure port $= \frac{A\,v}{98\,m}$ (opening side, V-shaped).

Area of exhaust port $= \frac{1 \cdot 5\,A\,v}{98\,m}$.

The dimensions of the slide should be such that the unbalanced pressure does not exceed 1,000 lbs. per sq. inch on the net working surface of metal.

973. HYDRAULIC SLIDE VALVES.

In designing hydraulic slide valves the size of the pressure port must be first determined, the assumed velocity of the water being 98 feet per second. Then the bar $\frac{1}{2}$ to $\frac{3}{8}$ inch wide, then the exhaust port say $1\frac{1}{2}$ times area pressure port (examples vary from 1 to 2 times). The pressure port is often made triangular or with outer end triangular to reduce the shock of admission, the exhaust port is always made rectangular. The port in the slide is then made to include the two ports and bar in valve face. The outside of slide should have not less than 3 times the area of the interior, but where the slide is small it may with advantage be made 5 times the area of the interior to reduce the intensity of the pressure on the working face. The pressure will then be limited to 1,050 lbs. per sq. inch for 3 times area, or 875 lbs. for 5 times area. The valve face should be same width as slide and long enough to allow full stroke less $\frac{1}{32}$ inch each end to avoid leaving a ridge in working. The life of a slide depends largely upon the intensity of pressure. From actual observations with an intensity of 1,290 lbs. per sq. inch the slide seized in 11 hours, 1,096 lbs. required frequent facing up, 1,059 lbs. required oiling every other day, 1,035 lbs. seized in 67½ hours, 794 lbs. never require attention until worn out. The probable life before repair is given by the following formula :

p = intensity of pressure in lbs. per sq. inch.
h = hours work before repair needed.

$$\log h = 25{\cdot}9374683 - 8 \log p.$$

It should be remembered that while increasing the area of surface reduces the intensity of pressure it also increases the total pressure, and therefore the labour required to work the lever, or the leverage being increased the range of movement must also be increased to obtain sufficient stroke.

The ratio of leverage in the working lever is usually 5 times the area of slide valve in sq. inches, but not less than 20 to 1.

974. COUNTERWEIGHTS FOR CRANE CHAINS.

The overhauling weights should be oval—i.e., egg-shaped, with small end on top to avoid catching under beams, etc. Hole for chain should be $\frac{1}{8}$ inch larger than cross section of links, and interior should be cored out to $\frac{1}{4}$ inch clear all round. The approximate weight of counterbalance required is $\frac{1}{20}$th of the load.

975. STRESS ALLOWED ON WROUGHT IRON IN HYDRAULIC CRANES.

	Tons per sq. inch.	
	Tension.	*Compression.*
Ballast and coaling cranes	2½	1
Warehouse and other cranes lifting from 1 to 5 tons	3	2
Cranes lifting more than 5 tons . .	3½	3

976. LOCK GATES.

The span of a pair of gates should form the diagonal of a square, the curve of the centre line of gates being struck from the opposite corner of the square, radius = length of side = ·707 span, giving angle of 136°, or rise of $\frac{1}{5}$ span.

The pressure of water per square foot varies at different depths, being 62·5 × difference of head on the two sides at the point considered.

The hauling strain on gate chains averages 336 lbs. per foot width of entrance, but in practice hydraulic and other machines are calculated for an effective strain on the chain of 375 lbs. per foot width of entrance. The total weight of a pair of gates averages 2½ tons per foot width of entrance.

977. POWER AND SPEED OF HYDRAULIC HAULING MACHINES.

	Strain on Rope.	*Hauling speed ft. per min.*
Railway capstans	2000 lbs.	180
	2240 „	200
Barge „	1½ tons	120
Ship „	2½ to 5 tons	80
Railway traversers	75 lbs. per ton of load.	
Lock gate machines	375 lbs. per foot width of entrance.	

978. SHIP CAPSTANS.

A simple hydraulic ship capstan fixed at a dock entrance has a capstan body 2 feet 3 inches in diameter (= 7·068 feet circumference) on a 6-inch shaft, with a height of 3 feet 4½ inches to centre of capstan bars. Under the capstan body is a bevel wheel 7 feet diameter with 116 teeth 2¼ inch pitch, and geared into this on an intermediate shaft 4 inches diameter a bevel pinion 1 foot $0\frac{5}{16}$ inch diameter with 17 teeth 2¼ inch pitch. Upon the same shaft is a spur wheel 2 feet $7\frac{1}{16}$ inches diameter with 56 teeth 1¾ inch pitch gearing into a spur pinion on the engine crank shaft 1 foot $6\frac{1}{16}$ inches diameter with 32 teeth 1¾ inch pitch. The hydraulic engine has three oscillating cylinders with plain rams 2⅝ inches diameter (= 5·4 sq. inches area) and 12-inch stroke. Cost £295 fixed, exclusive of foundations. This capstan is estimated to exert a hauling strain of 5 tons at 40 feet per minute with an accumulator pressure of 700 lbs. per sq. inch = 13½ E.H.P. Assuming that the engine makes 68 revolutions per minute in full work, the calculation for speed is $\frac{17 \times 32 \times 68}{116 \times 56} = 5{\cdot}7$, $5{\cdot}7 \times 7{\cdot}068$

= 40·3 feet per minute on hawser, and assuming t
by an effective pressure of ·6 accumulator pressu
lation for power is $\frac{116 \times 56 \times 6 \times 5{\cdot}4 \times {\cdot}6 \times}{17 \times 32 \times 13{\cdot}5 \times 2240}$

The true coefficient or modulus of the mach
inch with no losses is given by the formula:

$$\frac{\text{E.H.P.} \times {\cdot}32738}{\text{cub. ft. water used per min.}} = \frac{13{\cdot}5}{5{\cdot}4 \times 12}$$

or allowing for the calculated power and speed ins
E.H.P. = 14·69 and the coefficient = ·628.

This represents the fair average practice and met
by Sir W. G. Armstrong and Co. at the time the
in England were fitted with hydraulic machinery.

The hand capstans displaced by hydraulic w
body 1 foot 9½ inches diameter (= 5 feet 8 inches
8 feet 9 inches extreme radius, 3 feet 3 inches from
It was assumed that 20 men at a mean radius of 7
able force (allowing for friction) of 28 lbs. each, t
= 4375 lbs. on hawser or about 2 tons at a speed
feet per minute.

979. MANUAL LABOUR AT CAP

Ordinary force of each man 30 to 33 lbs. at a
per minute = 3,600 to 4,950 foot-lbs. per minu
about 100 lbs. and absolute velocity about 4 mile
minute. According to Dr. Olinthus Gregory the m
when the working velocity = ⅓ absolute velocit
absolute force. By note book, 40 men at a mean
a pull of 4 tons at 40 feet per minute.

980. HORSE WHIM.

A horse moving with a constant velocity of 174
whim will exert a constant force of 95 lbs. at end
per minute (Warrington Smyth). A strong hor
lbs. at 220 feet per minute = 33,000 foot-lbs. per

average horse will exert a pull of 125 lbs. at 2 miles per hour = 176 feet per minute = 22,000 foot-lbs. per minute.

981. WINDMILLS.

Smeaton's proportions for windmill sails :—Total radius of arm or "whip" = r, length of sail = $\frac{5}{6}$ r, breadth at end nearest axis = $\frac{1}{5}$ r, breadth at tip = $\frac{1}{3}$ r. Angles made by surface of sail with plane of revolution, end nearest axis 18°, tip 7°, efficiency 0·29.

982. ANNULAR SAIL WINDMILLS.

Allowing for 8 hours' work per day, and a breeze of 14 miles per hour giving a pressure of 1 lb. per sq. foot of sail area :—

Diameter of sail.	*Horse-power.*
10 feet	0·25
12 ,,	0·50
16 ,,	1·00
20 ,,	1·75
25 ,,	2·50
30 ,,	4·00
35 ,,	6·00
40 ,,	8·00

The following figures apply to four-arm mills :—

Diameter of arms, sails tip to tip,	44 feet	6 H.P.
,, ,, ,, ,,	54 ,,	8 ,,
,, ,, ,, ,,	56 ,,	10 ,,
,, ,, ,, ,,	64 ,,	12 ,,
,, ,, ,, ,,	70 ,,	14 ,,
,, ,, ,, ,,	72 ,,	20 ,,

—G. H. HUGHES.

983. AMERICAN WINDMILL PUMPS

require a wind velocity of 8 miles an hour. Observations in the Midland counties extending over three years showed that—

Absolute calm existed for 4 days 3 hours.

The velocity exceeded 10 miles an hour during 63·8 per cent. of the time.

The velocity exceeded 5 miles per hour during 90 per cent. of the time.

There were never more than five consecutive days when the wind did not rise to more than 10 miles an hour.

It appeared, therefore, that practically a mill could have been worked daily for at least 8 hours a day.

984. WINDMILL MOTORS.

Wheel 8 feet diameter, 3-inch force and lift pump, 300 gallons per hour raised 50 feet with fair wind.

Wheel 12 feet diameter, same quantity lifted to a height of 100 feet.

For drainage purposes an 8-foot mill will raise 500 to 600 gallons per hour, and a 12-foot mill 1,200 to 1,400 gallons.—MACLELLAN, GLASGOW.

985. COMMON HAND PUMP.

Nominal diameter of pump is inside diameter of working barrel, advancing from 2 to 6 inches. Stroke usually 10 inches, lever or handle, short arm 6 inches long arm 36 inches, giving leverage 6 to 1. Average work 30 strokes per minute. Loss say 10 per cent. Power absorbed in friction say 20 per cent.

PROPORTIONATE SIZES OF PUMPS.

Height of Water to be raised.	*Diameter of Pump Barrel.*	*Water delivered per hour with 30 strokes per minute.*	*Size of the Suction Pipe.*
Feet.	*Inches.*	*Gallons.*	*Inches.*
14	6	1640	4
20	5	1140	4
30	4	732	2½
40	3½	555	2½
50	3	412	2
75	2½	260	2

Section XIII.

ELECTRICAL ENGINEERING, INCLUDING GAS AND OIL ENGINES.

986. THE HYPOTHESIS OF A UNIVERSAL ETHER.

ALL space, interatomic as well as interstellar, is filled with a continuous, elastic, perfectly fluid, vibrating medium, in which are propagated light, radiant heat, and electricity, as sound is in air. This fluid cannot, however, consist of ordinary matter, as it possesses some of the properties of a solid.

987. UNDULATORY THEORY OF LIGHT.

The undulatory theory of light, replacing the old emissive corpuscular theory accepted by Newton, was formulated by Young in 1801. Energy waves of all kinds move with the same speed as light—viz. 300,000,000 metres per second, or say 186,400 miles per second.—W. H. PREECE.

Huyghens conceived the undulatory theory of light with sufficient distinctness to account for double refraction, and after him Young, Fresnel, and their followers have developed the theory to account for other phenomena. —BALFOUR STEWART.

988. VELOCITY OF LIGHT.

Light is propagated through interstellar space at the rate of about 191,515 or say 192,000 miles per second (OLAF ROEMER, 1676), and through terrestrial space at about 185,000 miles per second (FIZEAU and FOUCAULT). The undulations of ether (i.e., rays of light) are said by Periere to strike the eye the following number of times per second :—

Violet ray	699 billions.
Indigo „	658 „
Blue „	622 „
Green „	577 „

Yellow ray	535 billions.
Orange „	506 „
Red „	477 „
Mean of white light—mixture of all the rays .	541 „

while a stretched string, giving the musical note known as middle C, vibrates only 522 times in a second. It has been suggested that if the string were bisected 40 times, it would, supposing it were possible to make it vibrate, evolve not a sound, but a yellowish green light; the vibration of a cord increasing in rapidity in proportion to the diminution of its length. In a prismatic spectrum the red rays are the least refracted, and the violet the most refracted—i.e., deviate most from original direction or bent most towards the base of prism.

989. LAWS OF REFLECTION.

1. When light is reflected from a surface, the incident ray of light, the normal to the surface, and the reflected ray of light, are all in the same plane.

2. The angle between the incident ray and the normal is the same as the angle between the reflected ray and the normal, or the angle of incidence is equal to the angle of reflection.—Dr. Saleeby.

990. FERMAT'S LAW, OR LAW OF LEAST TIME.

If a ray pass from one point to another, after any number of reflections at fixed surfaces, the length of its whole path from one point to the other is the least possible—subject to the condition that it shall *meet* each of the reflecting surfaces.—Dr. Saleeby.

991. HEAT AND ELECTRICITY.

Heat is a form of energy, from the action of which a molecule of any substance may be conceived as a small solar system, the addition of heat causing increased velocity of revolution, and therefore of centrifugal force, and producing ultimately the disruption of cohesion and of chemical affinity.—Balfour Stewart.

Electricity is another form of energy due either to the permanent rotation of the atoms about an axis giving a polarised effect, or to the existence of a distinct electrical atom of some imaginary quasi-matter, *sui generis*, to which the name of *electron* has been given, possessing similar rotatory properties of opposite sense.—Sir W. H. Preece.

The real mystery of radium is the constant emission of 90 calories per gramme per hour, and as this heat could not possibly come from the store of energy lost during the time interval of the heat production, Lord Kelvin suggested an absorption of ethereal waves by the radium.—SIR O. LODGE.

992. ELECTRONS OR RADIANT MATTER.

Radiant matter (CROOKES, 1879), electrons (DR. JOHNSTONE STONEY), satellites (LORD KELVIN), corpuscles or particles (J. J. THOMSON), represent the fourth state of matter, or matter in an ultra-gaseous state which is, with good warrant, supposed to constitute the physical basis of the universe. Prof. Crookes says, "We have seen that in some of its properties radiant matter is as material as this table, whilst in other properties it almost assumes the character of radiant energy. We have actually touched the borderland where matter and force seem to merge into one another."

A chemical *ion* consists of a material nucleus or atom of matter, constituting by far the larger portion of the mass, and a few *electrons* or atoms of electricity.

It is likely that eventually it will be known that matter is built entirely of electrons, and that all atoms are of the same nature; that matter in its ultimate constitution is the same thing, and contains blank spaces of relatively immense size, and that from this hypothesis of the electrons we shall eventually be able to explain all such phenomena as gravitation and cohesion.

The electrons in an atom are probably as far apart relatively as the planets in the solar system.—SIR OLIVER LODGE.

Lord Kelvin suggested (Brit.Assoc., 1903) a mechanism for explaining the constitution of matter, consisting of a rigid hollow sphere containing a smaller sphere possessing mass and connected with the enveloping shell by springs. The outer spheres he calls atoms, and the inner spheres electric charges, or electrons. The negative electron occupies the centre, and the positive electrions are uniformly distributed on the inside of the shell, so that the whole sphere is neutral as regards the outside. The whole system oscillates, and it may occur that the negative electrons are sufficiently disturbed to be thrown outside the shell.—"ENGINEERING."

993. RADIO-ACTIVE PROCESSES.

The discovery of radium has introduced new conceptions of matter, which are summarised in the following extract:—

"The so-called atom consists of a system of two kinds (*a*) a distribution of positive electricity over a surface the size of which is comparable with that of the ordinary hydrogen atom; (*b*) a collection of negatively charged corpuscles in the neighbourhood of the positive particle and in electric equilibrium with it. The corpuscles may be within or without the positive 'shell,' and differ from it in being extremely minute, while travelling with a velocity comparable with that of light, perhaps one hundred thousand miles a second. Many a well-known phenomenon can be described from this starting-point. When a discharge of electricity is passed through a tube containing a rarefied gas, a separation of positive and negative constituents takes place, the latter forming the well-known cathode stream. This flow of rapidly-moving particles, if its course be deflected by an obstacle, say a metal screen, sets up pulses which, spreading in all directions, form the familiar Röntgen radiation, itself travelling with a like high velocity. Then, again, coming to the still more recent case of radium, the so-called *a* rays, those which travel slowly and are easily absorbed by a thin sheet of lead foil, are nothing more than the positively charged shells, while the β rays, so rapid in their motion, and so capable of penetration as to be able to pass through considerable thicknesses of lead, are the negative corpuscles, ready to set up on their own account fresh radiation similar to waves of ordinary light."—"THE STANDARD," MAY 7TH, 1904.

There are three types of rays from radio-active substances: (1) the positively electrified a rays (or emissions), which are stopped by most, even gaseous, screens; (2) the more penetrative, negatively electrified β rays; and (3) the most penetrative γ rays. The a rays might, in accordance with the doctrine of Aepinus, be atoms (molecules) of matter of radium itself or of radium bromide; the β rays atoms of resinous electricity or electrons; the γ rays simply vapour of radium.—LORD KELVIN.

The Beta rays given off by radium are the ultimate units of matter. They are probably identical in all their properties wherever found. Each either *carries* or *is* a charge of negative electricity. If the latter be true, then the ultimate units of the atoms of matter are atoms of electricity.—PROF. J. J. THOMSON.

Other rays are the X rays (Röntgen), the Becquerel rays (Becquerel, 1869), and the N rays (Blondlot) or Odyllic flames (von Reichenbach).

994. EVOLUTION OF MATTER.

"By various converging lines of experiment it has been proved that the simplest of all atoms—namely, that of hydrogen—consists of about 800 separate parts; while the number of parts in the atom of the denser metals must be counted by tens of thousands. These separate parts of the atom have been called corpuscles or electrons, and may be described as particles of negative electricity. . . . The existence in the atoms of this community of negative corpuscles is certain, and we know further that they are moving with a speed which may in some cases be comparable to the velocity of light—namely, 200,000 miles a second. But the mechanism by which they are held together in a group is hypothetical. . . . Thomson suggested (1904) as representing the atom, a mechanical or electrical model whose properties could be accurately examined by mathematical methods. . . . Thomson's atom consists of a globe charged with positive electricity, inside which there are some thousand or thousands of corpuscles of negative electricity, revolving in regular orbits with great velocities. Since two electrical charges repel one another if they are of the same kind, and attract one another if they are of opposite kinds, the corpuscles mutually repel one another; but all are attracted by the globe containing them. . . . It appears that there are definite arrangements of the orbits in which the corpuscles must revolve, if they are to be persistent or stable in their motions. . . . Definite numbers of corpuscles are capable of association in stable communities of definite types. An infinite number of communities are possible, possessing greater or lesser degrees of stability. . . . We are thus led to conjecture that the several chemical elements represent those different kinds of communities of corpuscles which have proved by their stability to be successful in the struggle for life. If this is so, it is almost impossible to believe that the successful species have existed for all time. . . But if the elements were not eternal in the past, we must ask whether there is reason to believe that they will be eternal in the future . . . analogy with other moving systems seems to suggest that the elements are not eternal."—PROF. G. H. DARWIN, BRITISH ASSOCIATION ADDRESS, 1905.

995. APPROXIMATE DATES OF PUBLIC USE IN GREAT BRITAIN.

Electric telegraph	1840	Electricity supply	1883
Submarine cable	1854	Electric motors	1886
Electric bell	1855	Electric tramcars	1883
Telephone	1878	Electric railways	1892
Electric light	1878	Wireless telegraphy	1899

—PROF. S. P. THOMPSON.

996. ANALOGUE OF ELECTRICITY.

Electric transmission by continuous current may be illustrated by its analogy to hydraulics. The dynamo is essentially a rotary pump, pumping electricity instead of water. In the following sentences the analogous electrical terms are bracketed.

The pump (dynamo) forces the water (current) at a certain number of pounds pressure (volts), as indicated by a pressure gauge (volt-meter) to overcome the friction (resistance) of the pipes (wire) in order that the water (current) may flow at the rate of so many gallons (ampères) per minute, as recorded by the water meter (ammeter). The larger the pipe (wire) the more water (current) can be carried, and the less will be the friction (resistance). Manifestly the pipe (wire) might be so small that the friction (resistance) would absorb a very large proportion of the power of the pump (dynamo), leaving but little remaining for useful effect. If the pipe (wire) be too large, it will cost too much; if it be too small, the loss will be too great. The pipes (wire) require valves (switches) to regulate and direct the water (current), with fittings (contacts) sufficient to convey the water (current) without leak (drop), and safety relief valves (fusible strips) must be provided to prevent damage from over-pressure (over-voltage).

The continuous current in electricity is similar to a pump drawing from a reservoir and forcing water through a system of pipes upon which machines may be connected to do work, and the water returned to the reservoir.

The alternating current is similar to a plunger pump forcing water through the pipes on the outstroke and drawing it back again on the instroke, the motion being repeated with great rapidity.

997. ELECTRICAL TERMS.

An electric current flows in a battery from the *Positive* (or +) *plate* to the *Negative* (or —) *plate*, and outside the battery from the *Positive pole* (connected to the — plate) through *a conductor* to the *Negative pole* (connected to the + plate). If the *Electromotive force*, E.M.F. or *Potential difference* = 1 *Volt*, and the resistance through which the current flows = 1 *Ohm*, the strength of the current = 1 *Ampère*, the quantity of electricity flowing per second = 1 *Coulomb*, and the work per second = 1 *Joule*. If it requires 1 coulomb of electricity to charge a condenser to a potential of 1 volt, the capacity of the condenser = 1 *Farad*. If the mean force of attraction between two opposite charges of electricity = 1 *dyne*, the work done per centimetre displacement = 1 *Erg*. If electricity flows through any measuring instrument, the terminals at which it enters and leaves are *electrodes*; that at which the current enters = *anode*, that at which it leaves = *cathode*. A fluid decomposable by electricity is an *electrolyte*, the products of the decomposition are *ions*.

A volt is about 7 per cent. less than E.M.F. of Standard Daniell cell. An ohm is the resistance of a column of mercury 106·2 cm. long × 1 sq. mm. section, at 0° C. It is about the resistance of a pure copper wire $\frac{1}{20}$ inch diameter and 250 feet long. The legal ohm = ·9975 true ohm; B.A. (British Association) ohm = ·9889 legal ohm = ·9863 true ohm. Siemens' ohm is ·9434 of the legal ohm. One ampère deposits 1·118 milligrammes of silver per second. The capacity of a knot (6,080 feet) of submarine cable is about $\frac{1}{3}$ of a microfarad.

The prefix *meg* multiplies the unit by one million, *micro* divides it by one million, *milli* divides it by one thousand.—C. E. GROVE.

998. ELECTRICAL UNITS.

The VOLT is the practical unit of electromotive force, or difference of potential or electrical pressure.

The OHM is the practical unit of resistance, which varies directly as the length and inversely as the area of section of a conductor.

The AMPÈRE, formerly called the Weber, is the practical unit of strength of current, or velocity. It is sometimes described as the measure of current density.

The COULOMB is the practical unit of quantity, and represents the amount of electricity given by one ampère in one second. (The term " coulomb " is becoming obsolete.)

The FARAD is the unit of capacity of an electrical receiver; one-millionth of this, or the MICROFARAD, is taken as the practical unit. A condenser of one farad capacity would be raised to the potential of one volt by the charge of one coulomb of electricity.

The WATT is the practical unit of work, and is the amount of work required to force one ampère through one ohm during one second.

The JOULE is the unit of heating, and represents the heating effect caused by one ampère of current passing through a resistance of one ohm for one second.

The VOLT may be understood as a measure of pressure, the ampère of quantity, the watt of power; thus a current of 10 ampères at 10 volts = 100 watts.

The Board of Trade COMMERCIAL UNIT (B.T.U.) = 3,600 joules = 1,000 watt-hours = 1 kilowatt per hour = 1·34 H.P. working for one hour, and can be produced at a central station for say 2d. It equals a current of 1,000 ampères at an E.M.F. of 1 volt flowing for 1 hour. For private lighting it may be taken as 10 A × 100 V, and for public lighting 5 A × 200 V. As 16 c.p. requires 60 watts, 1 unit is practically equal to sixteen 16-c.p. lamps for 1 hour.

999. MEASURE OF ELECTRICAL WORK.

A = strength of current in ampères.
V = electromotive force in volts.
O = resistance in ohms.
C = quantity of electricity in coulombs.
t = time in seconds.
H.P. = actual horse-power.
W = units of work or watts (1 unit = 10 million ergs absolute C.G.S. measurement).

$$A = \frac{V}{O} = \frac{C}{t}, \qquad C = A\,t.$$

$$H.P. = \frac{A\,V}{746} = \frac{A^2\,O}{746} = \frac{W}{746}.$$

$$W = A\,V = A^2\,O.$$

1 watt = $\frac{1}{746}$ of a H.P. = 1 volt-ampère = 10^7 ergs per second.
1 kilowatt = 1,000 watts = 10^{10} ergs per second.
1 kilowatt-hour = 1·34 H.P. acting for 1 hour = say $2\frac{2}{3}$ million ft.-lbs.

1,000. OHM'S LAW.

The strength of a current (ampères or amps) varies directly as the electromotive force (volts), and inversely as the resistance (ohms), or current = $\frac{\text{pressure}}{\text{resistance}}$, or ampères = $\frac{\text{volts}}{\text{ohms}}$.

1,001. ELECTRICAL EQUATIONS.

Ampères × volts = Watts.
Joules × seconds = Watts.
Coulombs per second = Ampères.
Watts ÷ 746 = Effective H.P.

Coulombs ÷ volts = Farads.
·7373 foot-lb. per second = 1 Joule.
Volts × coulombs = Joules.
Watts × 44·236 = foot-lb. per minute.
Kilowatts × 1·34 = H.P.

An erg is the work done by 1 dyne acting through 1 centimetre. A dyne is the $\frac{1}{981}$th of a gramme. A gramme is the weight of a cub. centimetre of pure water at 4° C. = ·00220462 lb. A centimetre is ·032809 feet. 1 erg = ·00000007373229 foot-lb., and 1 million ergs = ·07 etc. foot-lb. 10^7 ergs = 10 million ergs = ·7 etc. foot-lb. = 1 Joule.

1,002. USEFUL FIGURES.

To convert—

Mils to millimetres . . .	multiply by	·0253994
Inches " . . .	"	25·3994
Sq. inches to sq. mm. . . .	"	645·137
Cub. inches to cub. mm. . .	"	16,386·18
Yards to metres	"	·914383
Miles to kilometres . . .	"	1·6093
Pounds to kilogrammes . .	"	·45359
Millimetres to mils . . .	"	39·3708
Millimetres to inches . . .	"	·0373708
Sq. millimetres to sq. ins. . .	"	·00155006
Cub. mm. to cub. inches . .	"	·000061027
Metres to yards . . .	"	1·09363
Kilometres to miles . . .	"	·62138
Kilogrammes to lb.	"	·204621

1 kilometre = 1093·6 yards.
1 mile = 1·6093 kilometres.
1 kilo = 2·2046 lbs.

Pure copper weighs 555 lbs. per cub. foot.

—Conrady and Co.

1.003. VITREOUS AND RESINOUS ELECTRICITY.

The old terms vitreous and resinous electricity have now given place to the terms positive and negative, but these must be considered as merely expressing the fact of polarity.

1,004. FRICTIONAL ELECTRICITY.

A Wimshurst inductive electrical machine gives a pressure of 10,000 to 50,000 volts. A single cell of a battery cannot give more than 2 volts, but by adding to the number of cells any required pressure can be obtained.

1,005. GALVANIC BATTERIES

will produce an electric current sufficient for telegraphic or telephonic purposes and electric bells, but not sufficient for lighting. When coupled up in *series*—i.e., copper of first cell to zinc of second, and so on—the electromotive force or " pressure " is increased in proportion to the number of cells. When coupled up *parallel*—i.e., all the coppers to one wire and all the zincs to another—the E.M.F. is the same as from one cell, but the strength of current or " volume " is greater because the internal resistance is reduced. Coupling up in series gives intensity, and parallel gives quantity.

In a charged copper-zinc cell, when the poles are connected the current flows within the cell from the zinc, which is the positive element to the copper which is the negative element, and outside the cell through the conducting wire from the positive pole attached to the copper plate to the negative pole attached to the zinc plate, the positive being always the part from which the current flows and the negative that to which it flows. If the ends of two wires in a circuit be dipped in salt water, bubbles of hydrogen gas will be given off from the wire connected to the negative pole.

The principal forms are :—

Daniell's.—Amalgamated zinc rod in porous earthenware pot containing dilute sulphuric acid, inserted in outer copper case containing saturated solution of copper sulphate.

Grove's.—(*a*) Thin platinum plate in a wood frame, insulated from two amalgamated zinc plates on sides of frame, is inserted in stoneware cell containing dilute nitric acid.

Or (*b*) platinum plate in a porous cell charged with 3 parts nitric acid to 1 sulphuric acid placed in an outer cell containing diluted sulphuric acid (1 to 7) and a zinc plate.

Bichromate.—Zinc for positive element and two plates of carbon for negative element attached to the wooden top of a glass jar containing bichromate of potash dissolved in water and acidulated with $\frac{1}{12}$ of its bulk of sulphuric acid.

Bunsen.—Similar to Grove's, but carbon plate substituted for platinum.

Leclanché.—Used very largely for electric bell work, not constant, but recovers itself when work is intermittent. Carbon plate surrounded by crushed binoxide of manganese in a porous pot, placed in a glass jar containing solution of salammoniac (2 oz. salt to 1 pint water) in which zinc rod is inserted. E.M.F. of 1 pint cell = 1·6 volts, resistance = 1·13 ohms. No. 18 or 20 S.W.G. copper wire used for bells, and 1 cell for every 2 pushes. Two or more cells in series are commonly used, as an electric bell requires from 3 to 10 volts.

1,006. ELECTROLYSIS.

Electrolysis is the separation of water into its components oxygen and hydrogen by means of a galvanic battery, oxygen being given off from the positive pole and hydrogen from the negative. Similar electrolytic action may take place with other liquids. The decomposable fluid in a battery is called an electrolyte.

1,007. PRIMARY BATTERIES.

Primary batteries (galvanic batteries) are used almost exclusively for telegraphy and telephony. The most simple forms are very inconstant, the chemical action diminishes, the current rapidly decreases, and shortly practically ceases, a result mainly due to polarisation from bubbles of hydrogen collecting on the negative element.

Lbs. zinc consumed per H.P. hour in each cell $= \dfrac{1 \cdot 995}{E}$ (where E = electromotive force of whole battery), and in no case of a primary battery is the cost less than 30 times that of producing a "unit" by steam engine and dynamo.—T. E. Gatehouse.

1,008. COMPARATIVE VALUE OF BATTERY OF CELLS VARIOUSLY ARRANGED.

In parallel or multiple arc, thus—

$$\begin{array}{cccccc} Cu & Cu & Cu & Cu & Cu & Cu \\ Zn & Zn & Zn & Zn & Zn & Zn \end{array} = \left\{\begin{array}{l} 6\text{ A} \\ 1\text{ V} \end{array}\right.$$

In multiple series, thus—

$$-\left[\begin{array}{l} Cu\,Zn - Cu\,Zn - Cu\,Zn \\ Cu\,Zn - Cu\,Zn - Cu\,Zn \end{array}\right]- = \left\{\begin{array}{l} 2\text{ A} \\ 3\text{ V} \end{array}\right.$$

In multiple series, thus—

Cu	Cu	Cu	
Zn	Zn	Zn	$= \begin{cases} 3\ \text{A} \\ 2\ \text{V} \end{cases}$
Cu	Cu	Cu	
Zn	Zn	Zn	

In series, thus—

—Cu Zn — Cu Zn — Cu Zn — Cu Zn — Cu Zn — Cu Zn —

$$= \begin{cases} 1\ \text{A} \\ 6\ \text{V} \end{cases}$$

1,009. SECONDARY BATTERIES, OR ACCUMULATORS.

A secondary battery, or storage battery, or electric accumulator (Planté, 1859) has two chief forms.

The Faure, or pasted battery, consists of lead plates formed of cast grids of lead in pairs, into the holes of which oxide of lead has been forced. Upon a current being passed through them the red-lead is reduced in one plate to metallic lead in a spongy condition, and in the other plate is converted to peroxide. In this condition a reverse current can be set up and utilised. A pressure of 2½ volts per cell is necessary for charging the battery which then gives out a current of 2 volts, so that to supply 100-volt lamps a battery of 50 cells would require to be joined in series, and the dynamo for charging would have to work at 125 volts.

The modern Planté battery has the surface of the lead plates increased to four or five times the area by fine corrugations or laminations. The plates can be "formed" by electrolysis—e.g., that attached to the positive pole becomes coated with peroxide of lead and the one attached to the negative pole becomes coated with finely divided pure lead; but the preliminary forming is usually done chemically. When put in a circuit with a dynamo machine and charged, they become in a condition to transform their chemical energy into electrical energy, and can therefore be used for producing a current. It is heavier than the Faure battery, but costs less for maintenance and repairs.

Storage batteries can only be used with direct or continuous currents.

The "Chloride Electric Storage Batteries" combine the advantages which are derived from the use of both these principles, and are said to be free from the disadvantages of either. The plates possess great durability, have a larger capacity and higher rate of discharge, weight for weight, than any other battery at present in the market. Different types are installed for different kinds of work. The electrolyte employed in the cells is the usual dilute sulphuric acid at a specific gravity of 1·215.

Planté formed his plates simply by means of the electric current in sulphuric acid, and required several months for their formation, the tendency of all large accumulator works has been to shorten this process as much as possible by the use of chlorine, saltpetre, etc., by which means a quick process of formation is obtained, but the slightest trace of these destructive substances is fatal to the life of the plates. The British Accumulator Company, Ltd., have succeeded in making use of a process of formation by which the use of any destructive substances is entirely discarded, and by which, therefore, the life of their plates is guaranteed.

"Considering the advantages accumulators give in a central station, in providing for emergencies, reducing staff, and keeping the pressure even engineers would be wise in recommending their corporations or their boards to instal accumulators instead of extra plant when the plant is a continuous one. How far this should proceed is a moot point, dependent on circumstances. Possibly somewhere about one-third the maximum full load capacity would be about right now; but with a load-factor of 10 to 15 it might be found desirable to instal accumulators for as much as two-thirds, or even three-fourths of the full load, which would place electric distribution nearly on a par with gas distribution in respect to storage."—G. L. Addenbrooke.

1,010. CHARGING STORAGE BATTERIES.

For charging the battery it is necessary that the source be on a direct-current system. Several methods may be employed; the simplest and cheapest—especially if the battery is of moderate capacity (say up to 80 ampère hours)—is from the incandescent light circuit, which is always available; sometimes, however, it is necessary to draw from either the 500-volt trolley circuit or an arc light circuit. In the case of the incandescent light and the trolley circuits resistance will be necessary to reduce the potential to the proper amount; it is generally most convenient to use lamps for this purpose, the charging current being adjusted by varying the number in circuit. The lamps are in series with the battery to be charged, and the combination is connected across the circuit furnishing the current. The number of lamps in series is determined by the voltage of the circuit and the number in parallel by the charging rate. For example, if the charging source is a 100 to 120 volt circuit, and the rate required is 5 ampères, then ten 16 c.p. or five 32 c.p. lamps in parallel, with the battery connected in series with them, would be all that is necessary; but if the circuit is of 500 volts pressure, then five sets of lamps will have to be in series, making 50 in all if those of 16 c.p. are used, or 25 if of 32 c.p. In case it is more convenient to charge at a lower rate, say 3 ampères, then a proportionately fewer number of lamps will be needed, but the length of time required to complete the charge will be proportionately increased. If but one case of cells is being charged, the lamps will burn almost as brightly as if on the circuit alone, thus showing that the proper amount of current is passing; but if several cases are in at once, being connected in series with each other and the lamps, then the latter will be somewhat dimmed in proportion to the number of cases, and the time required for charging will be increased; as, for instance, if it takes 10 hours to charge one case, 12½ would be required for three—provided, of course, the same number of lamps are in circuit in each case. Instead of lamps a rheostat is sometimes used; its resistance should be such as to produce, when carrying the normal charging current, a drop in volts equal to the difference between the pressure of the charging source and that of the battery to be charged; thus, if a case of three cells, giving six volts, is to be charged from a 110-volt circuit at a 5 ampère rate, the resistance would be according to Ohm's law:—$\frac{110 - 6 \text{ volts}}{5 \text{ ampères}} = 20 \cdot 8$ ohms. The carrying capacity should be slightly in excess of the current required for charging the battery.—CHLORIDE ELECTRICAL STORAGE CO., LTD.

In a battery working in parallel with a shunt-wound generator at such a speed as to give about the mean output so that the battery neither gains nor loses very much, the number of cells should be determined by the formula, Number of cells $= \frac{\text{line volts}}{2 \cdot 08}$. When in the best condition for working in this way a battery capable of giving 450 ampères for one hour will discharge 200 ampères with a fall of pressure of 25 volts, and will charge 200 ampères with a rise of pressure of 25 volts, these limits being the greatest permissible on a 500-volt circuit.

With compound-wound generators the pressure rises with the load, so that the battery working in parallel with the generator will do very little work.—J. S. HIGHFIELD.

1,011. DYNAMOS.

Gramme dynamo (1873).—A dynamo consists of two essential parts, the field and the armature. The field consists of iron cores upon which layers of insulated wire are wound, causing the cores to become electro-magnets while the current is passing round them. These magnets are securely connected to the iron frame, and the armature, consisting of an iron shaft carrying a number of coils of insulated wire, rotates between the poles. The current generated in the armature is collected by means of bundles of wires or copper plates called brushes for transmission to the main cables.

The rotating armature has little mechanical friction, but meets with very considerable electrical resistance, as is always the case when a conductor is passed through a magnetic field. It is this resistance that causes the consumption of power, which is at the same time converted from mechanical work into electric current.

The gramme or ring armature is not suitable for large machines for which a drum or cylinder armature is generally used.

The proper construction of a dynamo depends upon the work it has to do.

In the *continuous current dynamo* the current always flows in the same direction, a commutator being arranged on the armature from which the current is taken off by the brushes. A continuous current dynamo, being reversible, may be used as a motor, giving out mechanical energy from the conversion of an electric current applied to the armature.

"The principle of the electric motor was discovered by Faraday in 1822." —PROF. S. P. THOMPSON.

In the *alternating current dynamo* the current flows at rapid intervals in opposite directions, having a mean rate of 50 cycles per second. The brushes rest on two metal rings. In these dynamos an "exciter" is used to generate a continuous current for exciting the magnets, as the field-wire coil has no connection with the brushes. These dynamos were at one time used for arc lighting only, but are now used for long distance incandescent lighting by means of transformers and converters. The complications of two- and three-phase alternating current systems outweigh, in some cases, any advantage they have over single-phase.

A *transformer* (GAULARD and GIBBS, 1882) in its simplest form consists of a wrought-iron core round which the coils of wire are wound, one of many turns of fine wire through which the high pressure current from the dynamo passes, and the other of fewer turns of thicker wire which forms part of the lamp circuit. By varying the proportions of these two coils any desired transformation of current can be secured.

By a transformer having twenty times as many coils on its primary core as on its secondary core a current of 1 ampère at 10,000 volts is converted into a current of 20 ampères at 500 volts. A frequent transformation is from a main current at 2,000 volts to a working current at 100 volts, but Ferranti transforms in two operations, first to 2,000 and then to 100 volts.

The *series-wound dynamo* has the circuit continuous from one brush round the arms of the field magnets, and by the external wiring to the other brush. Being connected in series—i.e., one behind the other—the current generated in the armature passes in equal strength through the field magnets and lamp circuit. These dynamos are mostly used for arc lighting, and are not suitable for electro plating or for charging accumulators, as they occasionally reverse their poles.

The *shunt-wound dynamo* has the current divided between two circuits. The coils of wire round the field magnets are connected in shunt, or parallel, to the brushes. The greater part of the current is conducted from the brushes to the lamp circuit, only a small portion being required for the excitation of the field coils. A resistance box, rheostat, or regulator, is connected in the shunt winding of the field, so that by varying the resistance the E.M.F. can be increased or decreased. They may be used for either incandescent or arc lighting.

The *compound-wound dynamo* combines the winding of both the series and the shunt dynamo. The magnets are wound with thick wire, which is in series with the armature and lamp circuit; they also have a winding of fine wire, which is in shunt with the brushes. These dynamos are the best for incandescent lighting, as they give a more uniform pressure, and also for combined power and lighting installations.

"It is important for dynamo construction to have iron absolutely free from manganese, and copper from arsenic; minute quantities of these, which could hardly be detected by chemical analysis, having the effect of largely altering the permeability of the iron or the conductivity of the copper. Minute quantities of certain metals, such as copper and arsenic, have an equally prejudicial effect in the lead used for accumulators; and the discovery of this effect has had a most important effect on the manufacture and cost of upkeep."—COLONEL CROMPTON.

1,012. CHIEF SYSTEMS OF ELECTRICAL TRANSMISSION.

1. Alternating current—where the current flows in different directions—or the high tension system.

2. The continuous current, or low pressure, or storage system.

3. Pulsating current, continuous in direction, but varying in strength, usually of high tension.

For brevity, initials or marks may be used as follows:—a.c. or ∽, or ⌣ ⌢, c.c. or =, p.c. or = = =.

1,013. BOARD OF TRADE DIVISION OF SYSTEMS.

Low Tension System, if working below 300 volts with continuous currents, or 150 volts with alternating currents.

High Tension System, if working above these limits.

A pressure of 250 volts is allowed by the Board of Trade in consumers' premises.

1,014. CHOOSING AN ELECTRICAL SYSTEM.

"For nearly all practical requirements of town lighting the direct supply system will prove simpler and less costly as well as more advantageous than any system, whether direct or alternating, in which transformers are used; in fact, the necessity for this class of apparatus, except for special cases, seems to vanish. I think, therefore, that when the facts are fairly faced, there does not seem much doubt that those engineers who have advised their clients to erect c.c. direct supply stations in the past have done wisely, and will be found to have served their clients' best interests, and that for the future a.c. stations, even for country towns and scattered districts, should only be erected after the most careful weighing of possibilities.

"The use of 220 volt lamps on a three-wire system does away with the necessity for complications. All that is needed is a continuous-current motor transformer of the "booster" type. In this, one armature circuit is across the mains and wound with a high resistance and many turns, while a current to the accumulators passes through the second circuit direct, the second circuit being wound with a few turns, so that the voltage of the main current in passing through the armature is raised, say, 30 per cent. The motor-transformer then needs only to be about one-third the capacity for the charging circuit, and the cells can be charged at any time while the circuits are supplying lamps, so that continuity of supply without complication is ensured."—G. L. ADDENBROOKE.

"It is clearly most desirable that the electrical engineer under present circumstances, when a.c. distributions are so widely prevalent, should have a thoroughly practical means of tapping that a.c., and commutate a portion of it into c.c. for electrolytic and metallurgical purposes. Not until something of this kind is done shall we be entirely satisfied to see the whole of the electrical supply carried on by means of a.c., which otherwise have so many advantages." —PROF. S. P. THOMPSON.

"In America a frequency has generally been adopted of 133 per second for lighting purposes. This was done with the object of reducing the cost of transformers, which were supplied to each separate house, and consequently were always of small size and expensive; also because American engineers working with a.c. have generally put forward the view that parallel working is not desirable, and the fact that par. working is more difficult at high frequency, which is one of the principal objections to high frequency, has not been seriously considered in America. In Europe the usual frequencies are from 70 to 100, except Ganz & Co., of Buda Pesth, who have adopted 42, because it is the lowest frequency available with arc-lights, so as not to produce serious flickering, and their desire was to lower the frequency as far as possible in order to ensure parallel working."—PROF. G. FORBES.

"If the installation requires chiefly light and some power, say in the proportion of two-thirds to one-third, then it should be single-phase; if light and power are about equal, it should be two-phase; and if it is chiefly power (viz., for fixed motors, such as those for workshops, lifts, and cranes) and some light, say in the proportion of two-thirds to one-third, it should be three-phase. Having regard to the high degree of efficiency and perfection, and ease of starting recently attained in a.c. motors, it may be confidently predicted that, as in lighting installations and power transmissions for industrial purposes, so also in electric traction a.c. is destined to supplant c.c. A.c., whether single or multiphase, will not only admit of electric traction being applied over much longer distances than is economically possible with c.c., but it will ensure a saving of something like 30 per cent. in the weight of dynamos and motors, irrespective of the saving in copper of feed and contact wires, and will thus considerably simplify and cheapen electrical installation and equipment for tractive purposes."—DR. PRELLER.

"I do not think engineers realise, as yet, that three-phase distribution with a neutral conductor does for the alternating current precisely what the three-wire system did for the continuous current as regards the amount of copper to be used in the distribution. It does no more and no less than this. To be sure, as Mr. Field points out, it is possible to adopt a three-wire system for single phase alternating currents, and in fact this has frequently been done; but when, with about the same expenditure on mains, one can, with three-phase system, get all the advantages which are to be derived from the use of polyphase currents, it is difficult to conceive how anyone would adopt the older system of single-phase, even with a three-wire system.

"Of course, here we are heavily handicapped by the existence of old single-phase stations. These are being changed over gradually to a lower frequency, and at the same time the polyphase system is being introduced. Two phases are adopted because it is easier, having regard to the existing system of mains, to effect the change for two phases than for three. But I think it is safe to assert that the last single-phase station has been erected in Great Britain some time since. There will be no more of them. The new stations, provided they are not influenced by already existing conditions, will either generate polyphase currents of three-phases, or simple continuous currents." —W. B. Esson.

"Some few years ago I attended the demonstration at Frankfort, where power was transmitted from Lauffen, and I brought over to this country the first set of drehstrom apparatus, which was supplied by the Allgemeine Company, of Berlin. This was seen by a great many people at my office in Westminster, among others, by the late Mr. Greathead, who was very much struck and charmed with the idea of being able to combine the advantages of high-tension currents in connection with stationary transformers working multiphase motors generating direct currents, and he told me he believed that, when the work in this country was expanded, this would probably be the best method of working."—H. Edmunds.

"We have learned from the trip to Germany that on the Continent more multiphase are used than continuous-current machines, and if only everybody in this country would understand that a multiphase machine is a good deal simpler than a continuous-current, far more multiphase work would be done in this country. As a matter of fact, multiphase machines are looked upon by the large majority of the people as somewhat mysterious, and as so many engineers have not had any experience with these machines, they are not so popular nor used as much as they should be."—C. Leven.

"I do not think it is possible to lay down a hard and fast rule as to the selection, beyond stating that the two most suitable for a works are the two-wire continuous-current system and the three-phase system, using Star winding, so that lamps may be connected between the neutral wire and the phase wires. There is little or nothing to choose between the two systems on the score of efficiency, so far as concerns the percentage of power lost in the transmission. The continuous current is probably more suitable for small works, where the number of motors is few, and the commutators of less importance. For larger works there are advantages in the use of the three-phase system which should certainly be carefully considered. In the first place, the generator itself is more mechanical, the current passes direct from the winding of the armature, through bound terminals, to the conducting cables. There is no commutator required beyond that on the exciter, which has to deal with less than 5 per cent. of the total current. The heavy armature bars and connections are stationary, and there is not, therefore, the same difficulty in effectually insulating them as is found in the case of continuous-current generators."—W. GEIPEL.

1,015. ELECTRIC GENERATING STATIONS.

Alternating current suitable for large areas where consumers are scattered, enabling generating works to be established by the riverside, or where land is cheap and coal easily unloaded.

High pressure direct current with rotary transformers to reduce pressure, applicable to scattered or isolated compact areas.

Low pressure direct current produces and distributes electricity at the same pressure at which it is supplied to consumers.

The direct current systems permit the use of storage batteries, which cannot be used with the alternating current; greater efficiency in distribution and greater adaptability to motive power.

The works being liable to sudden demands through fogs, the quick steaming qualities of the water tube type of boiler are of great advantage. Engines of the marine type are preferred for large outputs, the high-speed engine not being used for larger powers than 750 H.P. Some engineers find engines of 350 H.P. sufficiently large and the most convenient unit to adopt, with the dynamos always connected direct to the engines. High speed engines must have three cranks to be free from appreciable vibration.

Storage by secondary batteries is expensive, but is used by a few works in London for maintaining the supply after midnight and in the daytime in summer.

High-pressure current is distributed in heavily insulated cables in iron pipes, low-pressure current in insulated cable in stoneware conduits or in cables heavily armoured and laid direct on the ground. Rubber is now little used, paper and jute impregnated with insulating compounds being extensively adopted.—A. H. PREECE.

Lancashire boilers in conjunction with water-tube boilers make the best arrangement for a central station. The Lancashire boilers are economical and almost everlasting, and may be used to provide for the mean load. Water-tube boilers are more expensive to maintain, but are rapid steam raisers and may be used for emergencies.

1,016. LOAD-FACTOR AND DIVERSITY-FACTOR.

Load factor (Colonel Crompton) is the ratio between the mean load over a given period of time, generally a year, and the maximum load which occurred during the time. In other words, if the load-curve is plotted in the usual way, it is the ratio of the area under the curve to that of the rectangle surrounding it.

Diversity factor is the ratio between the maximum demands of the consumers and the maximum load on the station, taken over a year. The greater the diversity in the purposes for which the current is employed the greater will be the ratio.

1,017. COST OF GENERATING ELECTRICITY.

The actual cost of generating electricity is now about 1½d. per unit, management, etc., 1d., total 2½d. to 3d.; the average charge made is 5½d. per unit. The direct current is produced at ½d. to 1d. per unit, or say 20 per cent. cheaper than the alternating current.—A. H. PREECE, 1898.

The cost of producing a unit of electricity reaches its minimum with a plant capable of serving not less than 12,000 16-c.p. lamps, and as not more than ⅔ of the number installed will ever be in use at one time, the above plant would allow of 18,000 lamps being fixed.—PROF. ROBINSON.

In Bradford the cost of a B.T.U. is 1.22d., but the general cost may be taken at 2d.—W. H. PREECE, 1901.

"Mr. Swinburne has stated that electrical energy, whenever the demand is continuous, such as for certain electrolytic or electro-metallurgical processes, can be generated and distributed to short distances at a works-and-maintenance cost of ½d. per unit. I believe Mr. Swinburne has good grounds for this estimate, if cheap fuel be available. . . . We can hardly hope for many years to come to improve the load-factors of wires much above 20. . . . At a load-factor of 20, maintenance charges work out nearly five times as great as they would be if the output was continuous."—COLONEL CROMPTON.

The price charged by local authorities for electricity for motors varies from 1d. to 2d. per B.T.U.

The Hackney Corporation are supplying (1905) electrical energy for power at ½d. per unit + £1 per quarter per kilowatt of maximum demand.

1,018. COST AT CENTRAL STATIONS FOR ELECTRIC LIGHTING.

Fuel	·75d. per unit.
Oil, waste, water, stores, etc.	·16d. „
Wages	·68d. „
Repairs and maintenance	·5d. „
Rates	·4d. „
Management	·8d. „
Total	3·29d.

For great economy to be attained the output must be at least 700,000 units.

At Manchester (Prof. Kennedy, 1905) six compound engines of 2,500 H.P. with cooling tower condensers, generating three-phase current at 6,000 volts, cost per unit:—

Coal at 10s. per ton	0·17d.
Stores	0·02d.
Wages and salaries	0·05d.
Repairs and maintenance	0·04d.
	0·28d.

"A central station complete with mains, distribution, and everything, costs somewhere about £100 per k.w. If a k.w. capacity of plant will light about fifteen 16-c.p. lamps at once, and, taking 8-c.p. lamps at 40 watts, about twenty-five 8-c.p. lamps, say an average of about twenty lamps as taken together, 8's and 16's. On a central station, roughly, about double the number of lamps are installed which are ever required at once. Therefore, if we have a certain k.w. capacity in the station, we shall have about twice that on the lamps, and consequently on this basis we shall have about forty lamps wired for each k.w. capacity in the station. If we put these lamps down at an average of £1 apiece for wiring, that comes to £40, which shows us that while we are spending £100 in the central station, we are spending £40 in wiring the houses."—G. L. ADDENBROOKE.

1,019. COMPARISON OF CAPITAL COST OF ELECTRIC LIGHTING.

$$\frac{\text{Actual capital cost}}{\text{Kilowatts} \times \text{street mileage of mains}} = \text{comparative capital cost.}$$

1,020. REDEMPTION OF CAPITAL.

Capital borrowed under the authority of the Local Government Board must be redeemed as follows :—

For land . .	in 60 years.	For machinery .	in 15 years.
,, buildings . .	,, 30 ,,	,, electric mains .	,, 30 ,,

" Mr. Snell (Sunderland) gave details of the views of the Local Government Board in the matter of differentiating between various plant, thus the period for cables had been reduced from 25 years to 20 years, motor transformers would have 15 years, meters 5, arc lamps 7, and the lamp-posts 10 years. The council suggested as reasonable the following :—Land 100 years, buildings 60, boilers 20, engines, etc., 25, sub-station equipment 20, switchboards 25, stoneware casings, etc., 60 years, cables in conduit 15, and laid solid 25 years, batteries 10, meters and accessories 15, arc lamp-posts and street work 25, and arc lamps 15 years. Mr. Midgeley (York) thought that the period for armoured cable depended on the soil it was laid in."—" ELECTRICAL TIMES," JULY 20TH, 1905.

1,021. REPAYMENT OF LOAN AND INTEREST.

To find annual payment to pay loan and interest by equal instalments for given period :—

A = annual sum to be paid.
D = debt or amount of loan.
n = number of years in which D is to be repaid.
p = interest on £1 for one year.

$$A = \frac{D\,p\,(1 + p)}{(1 + p) \; - 1}.$$

1,022. ELECTRIC TRACTION.

Lancashire boilers for railway work, Babcock and Wilcox for tramways. Pressure 150 to 200 lbs. per sq. inch. Coal stored overhead with gravity feed to mechanical stoker. Feed water heated by economisers or surface condensers.

Engines connected direct to electric generators working at 200 revolutions per minute, regulated by Porter type governor or throttle governor on crank shaft, maximum variation of speed ½ per cent. Engines three-crank compound or triple type, generator on one or both ends of crank shaft.

Generators for direct current are multipolar, the field coils set in a cast frame of homogeneous iron, divided into halves horizontally. Armature built up of thin stampings carried on a "spider." Slots cut on surface and coils placed in them. Commutator carried on extension of spider, consisting of copper segments insulated by sheets of mica. Brushes made of carbon.

For economy in power distribution some systems use three-phase alternating current with rotary converters in sub-districts. On the Central London Railway current is supplied to the converters at 310 volts by static transformers which reduce the main feeder pressure of 5,000 volts to that amount.

Railway rolling stock with foreign traffic consists of separate locomotives with trailing carriages, but with own traffic only the trains may be made up wholly or partially of motor carriages to distribute the dead-weight and facilitate shunting.

In English tramway practice the single-truck car with motor on each axle and the double-bogie car with motor on each truck are both in general use.—B. Pontifex.

The new (1902) alternate current system of Messrs. Ganz, of Budapest, promises great economy in traction by electrical power.—W. H. Preece.

All tramway poles and supports are placed 40 yards apart.

1,023. LOSSES IN ELECTRIC TRACTION.

Electrical units absorbed per car mile:

Overhead system	0·68 to 0·85
Conduit system	1·0
Accumulator system	1·95
Combined overhead and accumulator system	1·37 to 1·5

1,024. COMPARISON OF ELECTRICITY WITH OTHER POWERS.

Electricity should not be compared to steam or gas, both of which generate and exert a force or power of themselves; but it would be proper to compare electricity with hydraulic, or belt, or rope transmission of power.—Sandwell.

1,025. GENERAL TRANSMISSION OF POWER.

There is no best system of transmitting power; the prevailing conditions can alone decide which method is most suited to the needs of the case. As to the merits of the three principal systems in large towns, Sir Frederick Bramwell has put them forcibly and clearly. They are, in effect: If gas costs 2s. 9d. per 1,000 feet, it is possible to get a brake horse-power for a little over ¾d. per hour; if a Board of Trade electrical unit costs 4d., a brake horse-power will cost $3\frac{7}{10}$d.; while if high-pressure water costs 3s. per 1,000 feet, it means that a brake horse-power will cost 4½d.—"Mechanical World."

Approximately pence per B.T. unit of electricity are equivalent to shillings per 1,000 cub. feet of gas. A rate of 2d. per unit = say £18 15s. per E.H.P. for a year of 300 working days at 10 hours per day.

1,026. COMPARATIVE COST OF TRANSMITTING POWER.

Method.	1,093 *Yards.*		5,465 *Yards.*	
	10 *E.H.P.*	50 *E.H.P.*	10 *E.H.P.*	50 *E.H.P.*
By cables . .	1·77	1·35	4·69	2·65
„ electricity . .	2·21	1·87	2·64	2·37
„ hydraulics . .	2·90	2·07	5·29	3·02
„ compressed air .	2·98	2·29	4·66	2·99

—"REVUE UNIVERSELLE DES MINES."

"The electric motor is the cheapest of all systems of driving machinery, when the load-factor is at its lowest. . . . As you increase the hours during which these machines are worked the electric motor will lose its advantages, and a point will be reached at which first the gas-engine and eventually the steam-engine will be found to compete favourably as regards cost."—R. CROMPTON.

1,027. COMPARATIVE COST OF POWER.

To produce 50 B.H.P. for 54 hours per week, 50 weeks in the year:—

Electricity, current at 1d. per B.T.U. £375

Steam, non-condensing. Coal, 10 lbs. per B.H.P. hour at 8s. per ton, £241. Water, 4 gallons per B.H.P. hour at 1s. per 1,000 gallons, £27. Labour, one man at 25s. per week, £65 . . . . . . £333

Oil, at 6d. per gallon. Consumption (with Kynoch Forward Oil Engine), 0·625 lb. per B.H.P. hour, £264. Labour, one man (part time) at 15s. per week, £37 10s. . . . . £301 10s.

Town gas, at 1s. 6d. per 1,000 cub. feet. Consumption (with Kynoch Forward Gas Engine) 16·5 cub. feet per B.H.P. hour, £167 10s. Labour, one man (part time) at 10s. per week, £25 . . . . . £192 10s.

Suction gas. Coal, 1 lb. per B.H.P. hour at 22s. 6d. per ton, £68. Water, 2½ gallons per B.H.P. hour at 1s. 1,000 gallons, £17. Labour, one man (part time) at 15s. per week, £37 10s. . . . . . . £122 10s.

"It is commonly stated that an electric motor with electricity at one penny a unit costs one penny per B.H.P. to run for an hour. One farthing is the average price which we have received during the last year for the current we have sold for motors."—J. A. JECKELL.

Steam engine and boiler.—Coal consumption, 7 lbs. per B.H.P. = 10 lbs. per k.w., at 15s. per ton = 0·8d. per B.T.U.

Gas engine.—Consumption 15 to 25 cub. feet per B.H.P. or 22½ to 37½ cub. feet per k.w., at 2s. 6d. per 1,000 = 1·12d. per B.T.U.

Oil engine.—Consumption $\frac{1}{10}$ gallon per B.H.P. = ·15 gallons per k.w., at 4d. per gallon = ·6d. per B.T.U.—W. N. TWELVETREES.

1,028. STATOR AND ROTOR.

"POLYPHASE MOTORS. . . . These considerations raise the question whether either part, the primary or the secondary mass, can be truly called an armature or a field-magnet. . . .

"Hence we may regard the primary part of the polyphase motor which receives the current as corresponding to the armature, while the secondary part, in which the magnetism is nearly fixed in direction with respect to the metal masses, corresponds to the field-magnet. The rotor or secondary is, in fact, a field-magnet which is not magnetised by any separate currents or by any commuted part of the current, but is magnetised by the closed currents which are induced in it.

"However, since the workman has got the notion that the revolving part must be called an armature, it is quite common to find the rotating part of polyphase motors so described. Yet in reality in almost all polyphase motors the true armature is the part that stands still and surrounds the rotating part.

"To avoid all confusion on this head we shall generally avoid the use of the terms armature and field-magnet in describing the parts of polyphase motors, and shall speak of the *stator and rotor* as defined above. In fact, these terms are fast becoming universal."—PROF. SILVANUS THOMPSON.

1,029. EFFICIENCY OF ELECTRIC MOTORS.

The loss in electric motors, stated as a percentage of full power, varies from about 8 to 25 per cent. The efficiency may be roughly stated as follows :—

100 H.P. motors	. 92 % efficiency.	2 H.P. motors	. 82 % efficiency.
50 ,,	. 90 ,, ,,	1 ,,	. 79 ,, ,,
25 ,,	. 88 ,, ,,	½ ,,	. 75 ,, ,,
5 ,,	. 85 ,, ,,		

—W. GEIPEL.

1,030. LOSSES IN ELECTRIC DRIVING.

Friction in steam engine	10°/₀,	resulting in	90°/₀	of I.H.P.
Loss in generators	8	„	82·8	„
„ conductors	2	„	80·5	„
„ motors	15	„	68·5	„

Approximately $\frac{2}{3}$ of the I.H.P. is delivered at the shaft of the motor. —W. GEIPEL.

In comparing the electric driving of shop tools with the same work done by shafting, the losses under the two systems will vary according to local conditions, the engine being the same in each case, and no general rule can be laid down.

The counter shafting will often be common to both systems with a loss of say 15 per cent. The loss in generators 8 per cent., conductors 2 per cent., and motors 15 per cent., total 25 per cent., may then be put against an average of 20 per cent. loss in main shafting displaced, showing an apparent net loss of 5 per cent. of power by changing from mechanical to electrical, but the loss in shafting is going on continuously, and in the motors only when they are running.

1,031. ADVANTAGES OF ELECTRIC DRIVING.

1. Absence of overhead shafting, which requires special construction of shops, and causes vibration, dust, and dirt.
2. Absence of belts, dust arising therefrom, repairs necessary thereto and shadows.
3. Clear head room for the use of electric cranes and hoists.
4. Better light and cleanliness.
5. Regularity of speed of machinery, and saving of wear and tear thereof.
6. Placement of machinery to facilitate handling of work.
7. Easy application of motors for special tools, such as key seating, cylinder boring, air compressing.
8. Facility of running one or two machines without the rest for working overtime.
9. Special suitability for working cranes and lifting apparatus.
10. Ease of extension.
11. Utility for electric lighting and other purposes.
12. General flexibility of the system.
13. Increased output.

In driving machine tools by electric motors the heavy tools should each have its own motor, but the lighter tools should be arranged in groups by means of a limited extent of shafting, so that they may be operated by one large motor which is more economical than a number of smaller ones.

—W. GEIPEL.

1,032. COMPARISON OF HYDRAULIC AND ELECTRIC MOTORS.

1,000 gallons water per hour at 700 lbs. per sq. inch pressure = $8\frac{1}{3}$ H.P. 1 Board of Trade unit = $1\frac{1}{3}$ H.P. hours, ∴ 1,000 gallons per hour $= \dfrac{8\frac{1}{3}}{1\frac{1}{3}} = 6\frac{1}{4}$ B.T. units.

A 1-ton crane, multiplying 6 to 1, lifting at 6 feet per second, requires water at the rate of 80 gallons per minute at 700 lbs. pressure. 1 gallon per minute = $\frac{1}{2}$ H.P., ∴ 80 gallons per minute = 40 H.P. while working. The advantage of hydraulic power is that the energy, being stored up in the water, can be consumed in bulk when required. 1 ton lifted at 6 feet per second = $24\frac{1}{2}$ H.P. net, ∴ efficiency of hydraulic crane $= \dfrac{24\frac{1}{2}}{40} = \cdot 6125 = 61\frac{1}{4}$ per cent. To work the same crane by electric motor would require either a very powerful motor or a considerable reduction in the speed. 1 ton lifted at 2 feet per second = $8\frac{1}{4}$ H.P. net, and a 10-H.P. electric motor would probably be considered capable of replacing the hydraulic motor, but working at only one third of the speed and thus showing a large apparent economy when the speed is not taken into account. Electric motors are more economical than hydraulic when the loads to be lifted are much less than the capacity of the machine. The cage and half the load being balanced by a counterweight, the motor lifts either cage or counterweight according to the load, and average loads therefore cost very little.

1,033. COMPARISON OF VARIOUS SYSTEMS FOR CAGE LIFTS.

First cost:

Hydraulic high pressure	75·0 per cent.
„ low „	87·5 „
Electric	100·0 „

Efficiency with full loads:

Hydraulic high pressure	45·22 „
„ low „	45·22 „
Electric	50·47 „

Cost of running 40 lifts, 7-cwt. loads, 42 feet 8 inches travel:

Hydraulic high pressure at 2s. 10d. per 1,000 gals.	.	4·35d.
„ low „ at 6d. „ „	.	15·36d.
Electric at 2½d. per unit	. . . . .	2·5d.

—SMITH AND STEVENS.

Comparative cost of running in ordinary work:

Hydraulic high pressure	4 per cent.
„ low „	10 „
Electric	1 „

Repairs equal, say, £6 per annum. —EASTON AND CO., LTD.

Passenger lift for four persons and attendant, say 7½ cwts., height of lift 50 feet, speed 150 feet per minute, 100 lifts per day.

Hydraulic (high pressure). First cost, £200:

Interest and depreciation at 10 per cent.	£20	
Working cost	40	
	—	60

Electric (direct current). First cost, £320:

Interest and depreciation at 10 per cent.	£32	
Working cost at 2d. per unit . . .	10	
	—	42
Balance in favour of electric . .		£18

—R. WAYGOOD AND CO.

Passenger lift, 5 persons. Electric:

7-cwt. load, 42 feet 8 inches travel.

42 complete round trips, up and down, with full load, for expenditure of 1 unit of current = 2½d.

1 unit = 1 kilowatt = 1,000 Watt hours = 1·34 H.P. hours.

33,000 × 60 × 1·34 = 2,653,200 ft.-lbs.

7 cwts. = 784 lbs., 784 × 42⅔ × 40 = 1,338,027 ft.-lbs.

$$\frac{1{,}338{,}027}{2{,}653{,}200} = 50\cdot 47 \text{ per cent. efficiency.}$$

Passenger lift, same capacity and travel. High pressure hydraulic, 1,000 *lbs.:*

Consumption per round trip = 3·2 gallons.

40 lifts at 3·2 gallons per lift and 2s. 10d. per 1,000 gallons = 128 gallons = 4·35d.

$$1 \text{ lift} = 33{,}450 \text{ ft.-lbs. } M = \frac{U}{144\,p\,Q} = \frac{33{,}450\cdot 7 \times 6\cdot 232}{144 \times 1{,}000 \times 3\cdot 2} = 45\cdot 22$$

per cent. efficiency.

Ditto ditto low pressure:

50 lbs. sq. inch at 6d. per 1,000 gallons, say 64 gallons per lift.

40 lifts = 2,560 gallons at 6d. per 1,000 = 15·36d.

1,034. ELECTRIC LIFTS.

Direct-coupled reversing motors worked by a continuous or direct current are the most costly in original outlay, but are quiet and very economical in working, as no current is consumed when the lift is at rest. They are the only suitable arrangement for hotels, offices, residential flats, hospitals, and private houses. For factories alternating current motors may be used, but must be kept always running, and use from 2 to 5 times the power of the direct coupled machine.—SMITH AND STEVENS.

An alternating current motor cannot be started against a load, and is therefore not economical for intermittent work.

1,035. ELECTRIC LIFT CONTROLLING DEVICES

may be divided into two main classes, one in which the rate of switching is controlled solely by time, and the other in which the switching is effected through relays or automatic electric switches cutting out or short-circuiting successive portions of the resistance. The first class is somewhat simple in construction, and gives satisfactory results when everything is in good order. The second class is much more complicated, is somewhat delicate in constitution, and has with it the risk which is always present where a large number of independent pieces of apparatus have to be kept in perfect condition, and where the duty of each part of the apparatus is not perfectly obvious to the somewhat unskilled attendant usually in charge of it. Messrs Monté-Callow and Co. have designed an apparatus which, it is claimed, is simple, easily understood, and meets the difficulties and risks in a satisfactory manner. A lower shaft carries the usual rope wheel receiving the control rope from the lift car; by means of the position of this rope wheel, starting, stopping, and reversing of the motor is effected. Above this is a three-arm contact arrangement, the contact fingers moving over the triple set of contact blocks connected to the resistances. Above is a second shaft connected by means of a chain or other gear to the motor itself, and this shaft has a crank in it which operates the three-arm arrangement cutting out the resistance.

In the Penrose system of electric lifts, the lift has six speeds, the starting speed being 170 feet per minute, rising in steps of 50 feet per minute till the maximum of 500 feet per minute has been reached. The latter speed is reached within 8 feet of travel, and when within a similar distance from the end of its journey the lift begins automatically to reduce its speed. This system is specially suited for high lifts.

The London Hydraulic Power Co. issue a pamphlet giving the chief points of difference in the question of hydraulic power *versus* electric power.

1,036. POWER REQUIRED FOR ELECTRIC LIGHTING.

Under good conditions the engine power required equals—

Arc lights	1 I.H.P.	per	1,000 c.p.
Incandescent lights	1	„	200 to 250 c.p.
Semi-incandescent lights . . .	1	„	500 to 800 c.p.

Ordinarily 1 I.H.P. will supply sixteen 8-candle incandescent lamps. 30 watts per 8 c.p. lamp is considered very economical, the Nernst lamp is said to give 40 c.p. for 60 watts.

When a number of arc lamps have to be maintained in one circuit a current of high electromotive force is required, whilst for a number of incandescent lamps a current of less electromotive force and greater quantity is needed.

Hospital Installation by W. N. Twelvetrees, 155 16 c.p. lamps, 10 k.w. belt-driven bipolar dynamo E.C.C. type = 100 ampères at 100 volts. Storage battery 216 ampère hours, discharging 36 ampères per hour in 6 hours = 8,000 B.T.U. per annum.

In London one is not far wrong in assuming that the average householder uses energy sufficient to keep his lamps alight for one hour a day all the year round. For shops the consumption is about 1½, and for clubs about three hours daily.—PROF. KENNEDY.

There is no doubt that a certain amount of work is done when keeping a cable charged, as one can usually tell whether a cable is alive or dead by grasping it tightly over the insulation when, if alive, a distinct vibration will be felt.—MR. BOOT.

1,037. RULES FOR ELECTRIC LIGHTING.

The Metropolitan Companies require all installations to be in accordance with the Phœnix Fire Office Rules. In wiring the following are some of the points to be observed. All conductors to be of tinned copper properly insulated with rubber and hemp. No conductor to be of less area than 20 B.W.G., and those larger than 16 B.W.G. to be stranded. Conductors never to be fixed with staples. No lead-covered wires to be used. Never to be in metal tubes for a greater length than 6 feet. Casings to be of well seasoned wood, and not less than $1\frac{3}{4}$ inch across and $\frac{1}{2}$ inch thick, the lids to be firmly screwed on, not nailed. All switches and fuses to be approved. Gas fittings not to be wired unless insulated from supply pipes. Every circuit to be controlled by separate switch, and protected by separate double pole fuse. The cost, approximately, of wiring town houses, not including fittings, is about 20s. per point. The cost of the light supplied from a Company's main is approximately about $\frac{1}{2}$d. per hour per 16 c.p. lamp, and $\frac{1}{4}$d. per hour per 8 c.p. lamp.

1,038. ELECTRIC WIRING.

Table showing legal standard wire gauge, with the equivalent in millimetres. The number of ampères required to fuse it. Safe carrying current in ampères. Number of 100 volt 16 c.p. lamps that it can supply with a drop of 5 per cent. in voltage per 1,000 yards. Gauge of tin fuse wire required to protect it.

S.W.G.	*Millimetres.*	*Safe Current in Ampères.*	*Ampères that will Fuse it.*	*No. of 100 Volt Lamps 16 C.P.*	*S.W.G. of Tin Wire for Safety Fuse.*
22	0·71	1	45	0	30
21	0·81	2	50	0	27
20	0·91	3	60	1	22
19	1·02	4	75	2	20
18	1·22	5	95	4	20
17	1·42	6	125	5	19
16	1·63	8	170	6	18
15	1·83	10	200	8	17
14	2·03	13	250	9	16
13	2·34	15	300	11	15
12	2·64	20	360	13	14
11	2·95	25	430	16	13
10	3·25	30	500	20	12
9	3·66	35	580	26	11
8	4·06	40	670	33	10
7	4·47	45	790	41	9
6	4·87	50	900	50	8
5	5·38	60	1100	60	7
4	5·89	70	1400	75	6
3	6·40	80	1600	90	5
2	7·01	100	2900	110	4
1	7·62	120	3300	140	3

In running wires, wherever a small wire is branched from a larger one, insert a fuse to protect the smaller wire. Fuses should be on porcelain or slate with screwed covers.—Conrady and Co.

To test direction of current in electric wire put pocket compass underneath lying N and S, if current flowing S to N needle deflects to W, if N to S needle deflects to E. Current flows always from + to —.

Three-wire system, middle wire earthed. Tumler switches fixed on live side of circuit. The positive lead should be coloured differently from the negative.

Gas and water pipes should be kept at a distance from electrical conductors.

1,039. CONDUCTORS.

In a series-circuit with incandescent lamps if one went wrong all the others went out, as the circuit was broken. The parallel-circuit was an improvement, as breaking the circuit in one lamp did not affect the others, but Hopkinson's three-wire system represents the present practice. By this system smaller conductors may be used, and the third or neutral wire acts as an equaliser to the other two when the loads vary. Usually the middle wire is half the sectional area of the others, but on the Continent one-fourth is the rule.

Every engineer knows the enormous difference in conductivity between pure copper and copper containing small percentages of impurity. The conductivity of copper produced by ordinary smelting process may be only 50 per cent. of that of electrolytically-refined copper. Hence every engineer specifies that conductors shall be electrolytic copper of 100 per cent. Mathieson's standard, but in few cases is this tested, especially with smaller ones bought commercially, and there are copper conductors in the market of not more than 80 per cent. (Mathieson's standard) conductivity, and these are accepted and used for house wiring and other purposes without discovery.—L. B. Atkinson and C. J. Beaver.

1,040. COPPER FOR ELECTRICAL PURPOSES.

Tensile strength of soft copper wire rods and bars used as electrical conductors :—

English requirement	14 tons per sq. in.
Continental requirement	22 kilogs. per sq. mm.
American requirement	32,000 lbs. per sq. in.

Copper is used for wiring as it is not easily oxidised, is pliable, easily jointed, and is fairly cheap; while its conductivity is only 4 per cent. less than that of silver and 600 per cent. greater than that of iron. The resistance of copper increases by ·002 per cent. for each 1° F. rise of temperature.

Not less than $\frac{1}{1000}$ sq. inch of sectional area per ampère should be allowed.

1,041. RELATIVE ELECTRICAL CONDUCTIVITY OF PURE METALS.

Silver	100·00	Platinum	14·45
Copper	97·61	Tin	14·39
Gold	76·61	Nickel	12·89
Aluminium	55·44	Steel, Siemens	12·00
Magnesium	39·44	Lead	8·42
Zinc	29·57	Mercury	1·75
Iron, Swedish	16·00	Bismuth	1·40

"The American Underwriters' National Electrical Association reported recently that insulated aluminium wire has a safe carrying capacity of 84 per cent. that of similarly insulated copper wire of the same size, whilst bare aluminium has only 77 per cent. the capacity of the corresponding size of bare copper wire. I do not remember seeing aluminium mentioned in any fire office rules, though there is no reason why it should not be used under certain conditions."—S. H. Gowdy.

1,042. ELECTRIC CABLES.

Approximate Rules for finding Cable Sections. (Power factor taken as unity.)

Continuous current—

$$\text{Single conductor area in sq. ins.} = \frac{\text{ampères} \times \text{lead and return in yards}}{\text{drop of volts} \times 40{,}000}.$$

Alternating current, single phase—

Practically as above for small concentric cable, at low periodicities, in short lengths.

Two phase, four wire—

$$\text{Amps. in each conductor} = \frac{\text{apparent watts delivered}}{\text{volts} \times 2}.$$

Two phase, three wire—

$$\text{Amps. in each outer} = \frac{\text{apparent watts delivered}}{\text{volts} \times 2};$$ and amps. in neutral $= \text{amps. in either outer} \times 1{\cdot}41$.

Three phase, mesh—

$$\text{Amps. in each conductor} = \frac{\text{apparent watts delivered}}{\text{volts per phase} \times 1{\cdot}73}.$$

Three phase, Y with neutral—

$$\text{Amps. in each main} = \frac{\text{apparent watts delivered}}{\text{volts between main and neutral} \times 3}.$$

And volts between main and neutral = volts between mains × ·58.

Capacity.

$$\text{Amps.} = \frac{\text{volts} \times \text{periodicity} \times \text{microfarads} \times 6{\cdot}28}{1{,}000{,}000}.$$

C.M.A. standard conductors nominal area.	*Number and diameter of strands.*	*Nearest S.W.G. sizes.*	*Their equivalent areas.*
·05	19/·058	19/17	·046
·1	19/·082	19/14	·095
·125	19/·092	19/13	·125
·15	37/·072	37/15	·149
·2	37/·082	37/14	·184
·25	37/·092	37/13	·25
·3	37/·101	61/14	·303
·35	37/·110	37/11	·38
·4	61/·092	61/13	·4
·5	61/·101	61/12	·512
·6	91/·092	91/13	·598
·7	91/·098	61/11	·645
·75	91/·101	91/12	·763
·8	91/·104		
·9	91/·110	91/11	·950
·0	91/·118		

Impurity in rubber insulation was formerly detected by prolonged immersion in water, the insulation resistance then steadily falls, and the specific capacity rises. Cables are now tested under pressure in hours instead of days.

1,043. FLEXIBLE CORD FOR ELECTRIC LIGHTS.

The cord should be insulated with pure and vulcanised indiarubber, with a minimum insulation resistance of 600 megohms per mile at 60° F. after twenty-four hours' immersion in water and one minute's electrification, and tested with 1,000 volts alternating current for fifteen minutes.

1,044. FUSING FACTORS.

The current required to fuse a wire of circular cross-section of given material is dependent upon the following variables :—

1. Diameter.
2. Time of current flow.
3. Length of fuse wire employed.
4. Environment and position of fuse.
5. Previous history of fuse wire.

It will be seen from the foregoing that there may be an infinite number of fusing currents for any particular wire. The "normal fusing current" for a given wire, the length and environment of which are constant, may be defined as—the minimum current required to fuse the wire in such a time interval as shall be necessary for the wire to have attained its maximum steady temperature. The "normal carrying capacity" or "rating" of a fuse wire may be defined as—the maximum current which the fuse is capable of carrying continuously without deterioration or undue heating. It should be taken as one-half of the normal fusing current for tinned copper wires.—A. Schwartz *and* W. H. N. James.

The function of a fuse is to protect the whole of the circuit, including the conductors and the appliances connected to them. Many of these appliances require special consideration owing to their varying capacity for overload. Generally speaking, the capacity of the circuit for overload is considerably in excess of the overload capacity of the fuse which protects it; there are cases, however, which require special treatment. Arc lamps, for instance, are difficult to deal with, as they require a fuse which will stand the large current consequent on starting and the similar current rushes due to the sticking of the mechanism and other causes. On the other hand, if the fuses are set to blow with a hundred per cent. excess over the normal working current, the lamps may be burned out with a far less overload than this if it be continued for some time. It is difficult to see how the fuse can be set for a lower limit than stated above without giving trouble, due to frequent blowing. The remedy seems to lie in careful and frequent inspection of the lamps. With regard to motors, the matter is somewhat simpler; the fuse should be set for 100 per cent. excess over the normal working full-load current, it will then be able to stand the variations in current likely to occur and to protect the leads from overheating. For the protection of the motor from severe overload, the overload release on the starter must be relied upon. It would be an advantage if these releases could be adjusted and then closed or secured, so as to be safe from the attentions of unauthorised persons.—Schwartz *and* James.

Experiments were made with a tinned copper wire, No. 22 s.w.g., diameter 0·074 cms., with a normal fusing current of 49·8 ampères. The wires were arranged in parallel, horizontally ½ inch apart. The results are given in the following table, from which it will be seen that with three wires and upwards the current per wire is practically constant:—

TIN COPPER WIRES IN PARALLEL.

Number of Wires.	*Fusing Current Ampères.*	*Fusing Current per Wire Ampères.*
1	49·8	49·8
2	90	45
3	130	43·3
4	170·5	42·6
5	218	43·6
6	259	43·1

—SCHWARTZ *and* JAMES.

1,045. CONVERSION OF ELECTRICITY INTO HEAT.

Heat effect = current² × resistance.

Thus a current of C ampères through a resistance of R ohms produces 0·0315 C^2 R Joules per minute.

Rooms may be conveniently heated by electric radiation, known as the "luminous lamp system," consisting of incandescent lamps in round ended ground glass cylinders with back reflectors. Electricity is also adapted for cooking, and domestic heating of all kinds.

The Prometheus method of electric cooking and heating apparatus supplied by Rashleigh Phipps and Co. is based upon a surface system which has many advantages.

1,046. LIGHTNING CONDUCTORS.

Lightning conductors properly fixed protect the space below them which would be contained in a cone, of which the upper terminal forms the apex and the base at any level has a radius equal to the height.

The upper end should terminate with a sharp point, and one foot below the point there should be attached by screws and solder a copper ring, bearing three or four copper needles 6 inches long, tapering from ¼ inch to a fine point. For a factory chimney a coronal or copper band with stout copper points 12 inches long at intervals of 3 or 4 feet round the circumference will be most suitable. All points should be platinised, gilded, or nickel plated to prevent corrosion.

The upper terminal should be securely fixed in place and connected to the copper tape by Cutting's Registered coupling or other safe connection.

The main conductor should be of solid rolled copper tape proportioned in sectional area to the height of the building, say sectional area in sq. inches = about ·01 $\sqrt{h}$ feet. It should touch the building all the way down without insulation, held by clips every 4 to 6 feet, and be electrically connected to all masses of metal in its neighbourhood. The usual tape sections are $\frac{3}{4} \times \frac{1}{8}$, $1 \times \frac{1}{8}$, $1\frac{1}{2} \times \frac{1}{8}$ for conductors, and $\frac{1}{2} \times \frac{1}{8}$ for connections to adjacent metals. Some authorities say it should be kept 2 inches away from the brickwork by means of projecting holdfasts.

The lower end should be carried down 2 feet under ground and 10 feet away from the building, riveted and soldered to a sheet of copper 3 square feet area for each 100 feet in height, $\frac{1}{16}$ to $\frac{1}{8}$ inch thick and embedded in moist earth. Coke breeze is often used, but has a deleterious effect on the copper. Killingworth Hedges finds that a small tube well forms the best earth terminal, the conductor being threaded down it and the entry joint soldered up. A branch pipe led to a rainwater pipe ensures the bottom being kept moist.

The cost fixed complete varies from about £3 10s. for a villa to £50 for a steeple or chimney shaft 300 feet high exclusive of scaffolding.

Sir W. Snow Harris (1861) considered $\frac{3}{4}$ inch diameter solid copper rod, or other section of equal area, sufficient under any circumstances for a lightning conductor. Faraday considered $\frac{1}{2}$ inch diameter rod sufficient.

The sectional areas in sq. inches of the various sizes named above are as follows, $\frac{1}{2} \times \frac{1}{8} = \cdot 0625$, $\frac{3}{4} \times \frac{1}{8} = \cdot 09375$, $1 \times \frac{1}{8} = \cdot 125$, $1\frac{1}{2} \times \frac{1}{8} = \cdot 1875$, $\frac{1}{2}$ inch diameter $= \cdot 1963$, $\frac{3}{4}$ inch diameter $= \cdot 4417$.

1,047. ELECTRIC LAMPS.

Arc lamps consist essentially of a metal framework for holding the carbons, together with automatic arrangements for separating and feeding the carbons and for focussing the light. The lamps of Gaiffe, Serrin, Gülcher, Siemens, Brush, and Crompton are typical forms.

Electric candles consist of two parallel rods of carbon, at the top of which the arc forms. They require alternating currents. Typical forms are the Jablochkoff and the Rapieff.

Semi-incandescent lamps are those in which a carbon rod is made to press against the negative terminal, which may be of carbon (Regnier), copper (Werdermann), or iron (Joel). A number of minute arcs form about the point of contact. In the Sun lamp of this class the arc plays upon a block of heated lime.

Incandescent lamps consist of a resisting filament enclosed in a glass globe, which may be either vacuous or filled with hydro-carbon vapour. The filament may be carbonised cotton thread (Swan), bamboo cane (Edison), cellulose (Brooks), root of a grass (Lane-Fox), compound of carbon and platinum (Gatehouse).

The Nernst lamp works with a heated filament, and requires an external steadying resistance. It is very economical.

1,048. ARC LAMPS.

Arc lamps are more economical than incandescent where much light is required, giving about six times the light for the same consumption of electricity.

A suitable voltage for arc lamps in parallel is 65 volts, for 2 in series 110 volts, for 4 in series 200 volts.

For indoor use 7 ampère continuous or 11 ampère alternating arc lamps, and for outdoor use not less than 10 ampères continuous or 15 ampères alternating current.—Moody Bros.

Arc lamps are best burnt "in series," requiring 9 or 10 ampères at 40 to 50 volts per lamp.

An arc lamp requiring a current of 10 ampères at an E.M.F. of 50 volts will give a light of from 1,000 to 1,500 candles, and will require approximately 1 H.P. to supply it.—A. B. Holmes.

Arc lamps are most economical for street lighting; the smallest practical size requires 6 ampères at a pressure of 50 volts, giving an average of 300 c.p. and using 3 B.T.U. per hour, at a total cost, including carbons, of 1·7d. per lamp her hour or ·006 per c.p. per hour. To provide for at least double the light of gas these lamps should be say 30 feet high and 240 feet apart.—Prof. H. Robinson.

Arc lamps burn badly with alternating current and last as a rule only 8 hours. A better result is obtained with oval carbons 25 × 10 × 310 mm. long, standard current 10 ampères, energy expended 400 to 500 watts. These lamps burn very steadily, and give a light of about 800 c.p. at angle of maximum intensity.—Sir W. H. Preece.

The height of an arc lamp to give uniform effect has, theoretically, nothing whatever to do with the candle power. The shape of the photometric curves of a 5-ampère arc and of a 15-ampère arc are practically the same. The distance apart should regulate the height. The maximum light in all open arcs is thrown out at 45 degrees, and lamps spaced 40 yards apart should be approximately 20 feet high, while those spaced 60 yards apart should be nearly 30 feet high.—W. W. Lackie.

1,049. CARBONS IN ARC LAMPS.

Upper carbon positive, fed downwards by solenoid mechanism as fast as consumed, hollow crater at end produced by action of arc. Lower carbon negative and pointed. With direct current the upper carbon burns twice as fast as the lower, if $\frac{1}{2}$ inch diameter it burns at the rate of about 2 inches per hour.

Enclosed arc lamps from which the air is practically excluded have the carbons further apart and give a long arc, they burn away much slower, say $\frac{1}{8}$ inch per hour, and therefore require less labour in trimming, but require double the energy for the given amount of light.

The Gilbert enclosed arc lamps have a current consumption of six ampères at 100 volts = 600 watts, equivalent to $\frac{7}{10}$ unit per 1,000 c.p. hours. They burn with a clear and steady light for at least 120 hours without re-carboning, trimming, or any attention.—S. L. PEARCE, MANCHESTER.

1,050. COMPARISON OF C.C. AND A.C. ARC LAMPS.

C.C. lamps—all light thrown downwards, upper carbon double section or double length of lower, voltage 40 to 45, if shunt wound the mechanism adjusts arc for voltage only, if differential wound adjusts both voltage and current.

A.C. lamps—light thrown up as much as down, both carbons same size, voltage 28 to 33, always differential wound.

In practice the supply voltage is much higher and is reduced by burning two or more lamps in series, the number being determined as follows, $\frac{\text{supply voltage}}{\text{50 for C.C. or 38 for A.C.}}$ = number of lamps in series. In order to burn steadily, each lamp or series of lamps requires a steadying resistance in ohms $= \frac{\text{line voltage} - \text{sum of terminal voltages of lamp}}{\text{current in ampères}}$. Wire resistances are used with C.C. lamps, and choking coils or transformers with A.C. lamps. —THE ELECTRICAL CO., LTD.

1,051. INCANDESCENT LIGHTING.

The incandescent or "glow lamp" was brought out by Edison and Swan in 1881. Glow lamps may be made for a low voltage 50 to 120 volts, or, at a slight extra cost, for a high voltage 150 to 200 volts. Their efficiency is about 3·5 watts per c.p. for the former and 3·75 watts per c.p. for the latter in the smaller sizes up to 50 c.p., and 2 to 2·5 watts per c.p. in the larger sizes up to 300 c.p. A lamp of high efficiency, say 1·5 watts per c.p., would have a short life, say 100 hours, whereas a lamp of 3·5 to 3·7 watts per c.p. may have a life of 3,000 hours, and not drop more than 30 per cent. in c.p. for the first 1,000 hours instead of the first 40 hours, as in the former case. When the c.p. drops to one-third it is generally cheaper to break the lamp and have a new one, as only the same power is required to run either, and lamps are cheaper than current. Electric lamps of foreign manufacture are generally rated by the "Hefner standard" of candle power, which is only 88 per cent. of the English standard; the rating of lamps "made in Germany" should therefore be received with caution. Well-made incandescent lamps will last from 1,000 to 1,500 working hours.

An incandescent circuit should not consist of more than eight 16-c.p. or twelve 8-c.p. lamps. For three 16-c.p. lamps and under use 3/22 s.w.g., and for from three to eight 16-c.p. lamps use 7/21½ s.w.g.

Incandescent lamps are best burnt "in parallel," requiring a constant potential of 50 to 100 volts with about ½ ampère of current per lamp. At 200 volts an 8-c.p. lamp requires ⅛ ampère. Flexible cord fittings should have a switch behind the cord. Switchboards are best of ebonite, but may be of marble or slate. Wood casings should be prepared with cyanite to render them uninflammable. Lamps are made for a pressure of from 50 to 250 volts, and must correspond in voltage with the pressure on the mains.

"From investigations which have been made in connection with the energy consumed by an incandescent lamp, we find that of the total heat units in the coal practically the whole are dissipated, and only a remainder of ½ per cent. is converted into the light which it has been our object to produce."—H. A. EARLE.

"Incandescent lamps as sold show a wide variation both in actual candle power emitted and energy consumed for the same nominal amount of light. Say a 16-c.p. lamp requiring 60 watts has a life of 1,000 hours before the c.p. drops 60 per cent. At 6d. per B.T.U. cost $= \frac{1000 \times 60}{1000} \times$ 6d. = 30s. + 1s. lamp = 31s. A second lamp of 16 c.p. requiring 50 watts and having a life of 600 hours with 20 per cent. drop cost $= \frac{1000 \times 50}{1000} \times$ 6d. = 25s. + $\frac{1000}{600} \times$ 1s. = 1s. 8d. for renewals, = in all 26s. 8d. against 31s., or a saving of 4s. 4d. per lamp."—E. C. DE SEGUNDO.

The Nernst lamp has twice the efficiency of the ordinary glow lamp, being 1·7 watts per candle on low voltage and 1·4 watts per candle on high voltage circuits. For 32 c.p. the consumption of ordinary lamps is 120 watts, and Nernst lamps 52½ watts (¼ ampère 210 volts). Allowing for first cost the saving would be nearly 50 per cent. in 1,200 hours (about a year). In a life test of 18 direct current Nernst lamps ¼ ampère 230 volts, the lowest duration was one failing at 411 hours, five lasted from 476 to 824 hours, two from 1,494 to 1,675 hours, and the remaining ten were still burning after 1,802 hours. Assuming those still burning as effete, the average life would be 1,375 hours.

The Nernst lamp has an incombustible filament, and can therefore be used without exhausting the globe. The filament when cold is a low conductor, but when heated by a coil round it becomes a better conductor, and is then made vividly incandescent. It is necessary to run it with a resistance to prevent the light becoming unstable. The efficiency is very high, being ·95 watts per c.p. on a 100-volt circuit, and ·88 watts on high voltage circuits. A 1-ampère lamp gives 105 c.p. on a 100-volt circuit, and 250 c.p. on a 220-volt circuit, being practically three times the amount of light given by an ordinary incandescent lamp using the same power.

Blondel's incandescent vapour lamp (1906) consumes only one-sixth of a watt per c.p.—COLONEL CROMPTON.

Linolite is a patented system of electric illumination by tubes made in lengths of 1 foot to 8 feet 6 inches, with an overall width of 2¼ inches and depth of 1 inch, costing 1d. per yard per hour for energy at 6d. per B.T.U., and the aluminium pattern weighs 6 ounces per foot-run. It is particularly suitable for cornices and for the borders of large shop windows.

1,052. PRIVATE INCANDESCENT INSTALLATIONS.

In a private installation there will, on the average, be about three 8-c.p. lamps to one 16 c.p. The average cost for steam engine and boiler, dynamo, switchboard, wiring, storage battery to supply half the number of lamps for 8 hours, and 50 yards of main cable will be about £200 + £25 per 100 c.p. up to 5,000 c.p. exclusive of housing. Up to 25 H.P., or say 6,000 c.p., an oil engine is quite suitable, and the first cost is about the same as with steam engine. A coal-gas engine plant costs a trifle less. The first cost of a suction-gas engine plant is about 30 per cent. more than a steam engine plant, but the cost of working is much less. If ⅔ of the lamps are in use at a time the brake horse-power required will be about $\frac{1}{240}$ of the total c.p. The same plant may be used for pumping, heating, cooking, chaff-cutting, ventilating, etc., by electric motors.

"To make the matter quite clear, let a practical illustration be taken. Let it be supposed that a house has to be lighted by 100 incandescent lamps, each requiring a current of ·75 of an ampère urged by an electromotive force of 100 volts. The rate at which energy is expended in each lamp, expressed in volt-ampères or watts, of which 746 are equal to a horse-power, will be ·75 × 100, that is 75. The energy expended in the 100 lamps will be at the rate of 7,500 watts, which are equal to 10·05 H.P. But this, it must be remembered, is the actual rate at which energy is expended in the lamps. The energy that has to be developed by the engine is greater, for no dynamo-electric machine is perfectly efficient, no dynamo machine gives out as electrical energy the exact equivalent of the mechanical energy expended upon it. Let it be supposed that the machine used in our installation has a 'commercial efficiency' of 80 per cent., that is, that 80 per cent. of the mechanical energy put into the machine reappears in the external or lamp circuit as electrical energy, the balance being wasted in heating the armature coils, and the friction of axles, slipping of belts, and other mechanical sources of loss, then the rate at which energy is generated by the steam engine must be 10·05 × 1·25, that is 12·55 H.P. This mechanical energy is to be produced by the combustion of coal, and if all the heat liberated in the combustion of coal could be collected and utilised, the supply of coal required to generate energy at the rate of 12·55 H.P. would be very small; but, unfortunately, steam engines even of the best make have but low efficiency, and a horse-power-hour of energy requires in practice somewhere about 4½ lbs. of coal for its production; 12·55 horse-power-hours will therefore require about 56½ lbs. of coal—say, roughly, half a hundredweight, the cost of which is not more than 6d. Assuming that the lamps were required to burn for 1,800 hours a year—that is, on an average, nearly 5 hours a day—the annual cost for coal would be £45. The prime cost of a suitable dynamo machine and engine (with boiler) would be, say, £300, the interest on which at 4 per cent. would be £12, and the annual depreciation, at 10 per cent., £30; the cost of attendance would be about £60; so that the prime cost would be £300, and the total annual cost £147, or £1 9s. 5d. per lamp."—PROBERT.

A unit of electricity will supply sixteen 16-c.p. lamps for one hour, and at 8d. per unit each lamp will cost ½d. per hour. This is equivalent to gas at 7s. 6d. per 1,000 cub. feet, or, roughly, pence per unit = shillings per 1,000 cub. feet. In domestic lighting there are generally not more than 25 per cent. 16-c.p. lamps to 75 per cent. 8-c.p. lamps.

1,053. ELECTRIC LIGHTING ESTIMATE TO REPLACE GAS.

£6,000 for gas at 3s. 2d. per m. = say 4,000 8-c.p. lamps = 250 H.P. average at £3 per lamp first cost. Power station = £12,000, main and sundry = £8,000; total £20,000.

The usual standard of gas lighting is represented by the amount of light falling on a unit area of pavement 50 feet away from a 12-c.p. gas lamp 9 feet high.

Average c.p. of gas per cub. foot per hour = 3. Average cost per hour per candle power = ·01d. Average hours per light per annum, private houses 550, shops 1,100. With electric light average cost per c.p. per hour = ·02d., or ·25d. including renewals.—PROF. H. ROBINSON.

The public do not understand anything about watts; they do not wish to buy watts, they buy candle-power, and they only think of the amount of light they get as compared with the bills they are called upon to pay. What engineers have to consider in attempting to popularise electric lighting is to bring it more nearly to the price of gas. The present efficiency of boilers, engines, and dynamos leaves comparatively little room for improvement, whereas there is a practically unlimited scope for invention and improvement in the distributing system, and, above all, in incandescent lamps. Now, if the incandescent lamp could be so improved as to give 1 c.p. for 2½ watts, with the same percentage loss of c.p. and the same life as with 4 watts, it would be equivalent to bringing down the price of the electric light to the consumer from 6d. to 3¾d. per unit.—E. DE SEGUNDO.

1,054. COMPARISON OF GAS AND ELECTRICITY.

For equal lighting electricity at 8d. per unit is practically the same cost as coal gas at 3s. per 1,000 feet, but allowing for the ordinary increase of light required when electricity is adopted the cost per unit should not exceed 4½d. for equal expenditure.

Incandescent gas burners, although giving four to five times the light for the same consumption of gas used in the old burners, fall off in efficiency 25 per cent. in 100 hours, and 33 per cent. in 300 hours.—W. N. TWELVETREES.

High pressure gas lighting, as by the Keith intensified incandescent gas burner, is the most economical, being about half the cost for gas of an ordinary incandescent or one-tenth the cost of an ordinary fish-tail burner.

1,055. ELECTRIC LIGHTING FOR COAL WHARVES.

Steam engine 3 B.H.P. per crane.

Boiler 75 lbs. working pressure tested to 100 lbs.

Lamps (incandescent) per crane :—

2—80 c.p. for ship's hold.
1—50 c.p. for land-side.
1—50 c.p. for overside.
2—16 c.p. for office and works.
1— 8 c.p. for meter.

= 300 c.p. total per crane.

For steamships incandescent lamps are alone suitable, placed in groups if required, with reflectors for cargo handling.

1,056. LIGHTING FOR FACTORIES, ETC.

For lighting a factory, $\frac{1}{4}$ floor area in sq. feet = c.p. required.

The exhibition held by the Royal Sanitary Institute at Bristol, 1906, had a floor area of $3\frac{3}{4}$ sq. feet per c.p. and was efficiently lighted.

1,057. PERCENTAGE OF ENERGY TRANSFORMED INTO HEAT AND LIGHT.

	Heat.	*Light.*
Candles	98	2
Oil lamps	98	2
Coal gas—		
Flat flame and Argand	98	2
Regenerative	93	6
Incandescent	88	12
Electric light—		
Geissler tubes	97	3
Incandescent	95	5
Arc	90	10
Magnesium light	85	15
Sunlight	70	30
Glow worms, fireflies, and luminous beetles	1	99

—PROF. V. B. LEWES.

1,058. COMPOSITION OF COAL GAS.

Average composition of 16 candle coal gas, free from enrichment—

	Per cent.
Hydrogen	54·0
Methane (marsh gas) and hydrocarbons of the same character	34·0
Ethylene and hydrocarbons of the same character	3·0
Benzene	1·0
Carbon monoxide	6·0
Nitrogen, etc.	2·0
	100·0

All of these, with the exception of the nitrogen, contribute their quota to the heating value of the gas, and their relative value in this direction may be stated as follows :—

Gas.	*Gross.*	*Nett.*	*Illuminating Value per 5 cub. feet.*	*B.Th.U.'s per Candle.*
Hydrogen . .	325	272	—	—
Methane . . .	1,024	919	5·2	312·5
Ethylene . .	1,603	1,510	70·0	114·5
Benzene (vapour) .	3,718	3,574	820·0	22·6
Carbon monoxide .	330	330	—	—

Applying these thermal values to the coal gas, we obtain :—

Hydrogen	54	×	325	=	17,550
Methane	34	×	1,024	=	34,816
Ethylene.	3	×	1,603	=	4,809
Benzene	1	×	3,718	=	3,718
Carbon monoxide . . .	6	×	330	=	1,980
					62,873

or 628·73 B.Th.U.'s gross for a cub. foot, which is about the value given by the calorimeter.—Prof. V. B. Lewes.

1,059. AVERAGE ANALYSES OF GASES.

Average volumetric analysis per cent., weight, and energy per 1,000 cub. feet, of the four types of gases used for heating and illuminating purposes.

—	*Natural Gas.*	*Coal Gas.*	*Water Gas.*	*Producer Gas.*	
				Anthracite.	*Bituminous*
CO . .	0·50	6·0	45·0	27·0	27·0
H . . .	2·18	46·0	45·0	12·0	12·0
CH_4 . .	92·6	40·0	2·0	1·2	2·5
C_2H_4 . .	0·31	4·0	—	—	0·4
CO_2 . .	0·26	0·5	4·0	2·5	2·5
N . . .	3·61	1·5	2·0	57·0	55·3
O . . .	0·34	0·5	0·5	0·3	0·3
Vapour . .	—	1·5	1·5	—	—
Lbs. weight .	45·6	32·0	45·6	65·6	65·9
B.Th.U. . .	1,100,000	735,000	322,000	137,455	156,917

—Mason's Gas Power Co.

1,060. COAL GAS.

William Murdoch (1792) began investigations into the gases given off by different materials, which eventuated in his lighting (1798) part of Boulton and Watt's establishment by coal gas. It was first used for public lighting in London by Clegg at Westminster in 1813, and by 1816 it had become fairly common.

1 ton of coal gives about 10,000 cub. feet of illuminating gas and 12 cwts. of coke.

The pressure of gas at consumers' premises averages 1 inch head of water. 1 cub. foot at pressure of 3 inches = 214 grains.

One volume of coal gas mixed with 8 vols. air forms the most explosive mixture, but with from 4 to 14 vols. of air explosion may take place.

On combustion through a burner for lighting purposes coal gas yields about half its volume of carbon dioxide and 1½ times its volume of watery vapour. On combustion through an atmospheric burner for heating purposes it yields a considerable proportion of carbon monoxide, hence the poisonous character of the fumes, as distinguished from the mere suffocation produced by carbon dioxide.

1,061. COAL AND COKE.

1 ton of coal produces 1 chaldron of coke = 12½ to 15 cwts., say average 13⅓ cwts., or 1½ chaldrons = 1 ton.

Coke quenched with water weighs 20 to 25 per cent. more than when dry. Gas coke weighs about 35 lbs. per cub. foot. Hard coke from coking ovens about 50 lbs. per cub. foot.

1 ton of coal (loose) averages 45 cub. feet, 1 chaldron of coke averages 50 cub. feet, 1 ton of coke averages 75 cub. feet. A sack contains about 4½ cub. feet.

1,062. DIMENSIONS OF COAL GAS APPARATUS

per 1,000 cub. feet per day of 24 hours.

Condensers 4 to 6 feet super.

Purifiers 1 sq. yard of sieve for dry lime, 2½ inches deep in 5 sieves to each purifier.

Connecting pipes. Diameter in inches = $\sqrt{\text{area of sieves in feet}}$.

Scrubber, 1 cub. foot wet coke.

Gasholders, 1,000 cub. feet.

Joints made with iron gauze wire covered with red lead mixed with boiled linseed oil.

Discharge of gas through pipes $\propto \frac{\sqrt{p}}{\sqrt{l}}$.

Average cost of gas works £1 per head of population, more in large towns, less in small. Velocity of gas through mains = $1\frac{1}{2}$ miles per hour.

1,063. WATER GAS.

In "straight" water gas, a deep bed of coke, enclosed in a suitable brick-lined generator, is subjected to a blast of air from beneath until the internal heat has been raised to a high point. The reaction of C oxidising to CO_2 liberates heat. Steam is then blown into the coke bed, and is immediately dissociated into its constituent elements, the hydrogen escaping in free form and the oxygen serving to further oxidise some of the excess carbon present to form combustible monoxide, which likewise forms a stable constituent of the resultant gas. As this operation of dissociation of steam absorbs heat, or is an endothermic reaction, the internal heat in the generator is reduced. The operation of water gas making is, therefore, an intermittent process, coke being first "blasted" with air to increase the temperature and then "blasted" with steam until the temperature has again fallen to its lowest limit permissible. Usually the blast gas is wasted, after first being passed through a regenerator or "superheater," consisting of an enclosed checker work of brick, which absorbs from the passing gases a large percentage of their sensible heat. This reclaimed heat is then used in "carburetting" the water gas subsequently generated. As the gas passes through the checker work, crude oil is admitted, which is itself dissociated into oil gas upon striking the hot checker work, thus enriching the lean water gas with the desired quantity of illuminants.—J. R. Bibbins.

1,064. ACETYLENE GAS.

Acetylene gas is produced by bringing calcium carbide (CaC_2) into contact with water, when the gas acetylene (C_2H_2) is immediately evolved, leaving as a residue slaked lime (CaO_2HO), thus $CaC_2 + 2H_2O = CaOH_2O + C_2H_2$. Calcium carbide is formed by the intimate combination of coke and lime in an electric furnace. The actinic value of the acetylene light is greater than that of any other artificial light, and forms a steady flame, with great diffusive power. It may be transmitted by pipes to the usual gas fittings, is less poisonous than coal gas, and does not blacken ceilings. The cost may be taken as one-tenth of a penny per 15 c.p. burner per hour. A suitable apparatus for domestic lighting is made by the Forbes and British Pure Acetylene Gas Co., Ltd., 15, Victoria Street, Westminster.

1,065. DE LAITE GAS.

De Laite gas is produced from petrol, 1 gallon making about 1,000 cub. feet; an automatic apparatus is made by the De Laite Gas Machine Syndicate, Ltd., 117, Middlesex Street, London, E.

1,066. LIGHT-TESTING, OR PHOTOMETRY.

The *standard candle* is a sperm candle, six to the pound, $\frac{7}{8}$ inch in diameter, and burning at the rate of 120 grains of spermaceti per hour.

The "*One-candle Pentane Air-gas Flame*" invented by Dr. A. Vernon Harcourt represents the true average value of the light of the legal standard sperm candle, but it is actually equivalent to 1·001 sperm candles. A similarly constructed 10-candle lamp is now (1898) permissible in the official testing of London gas.

The *standard gas burner* used for testing the illuminating power of all qualities of gas is the Sugg "London" (No. 1) argand burner, consuming gas at the rate of 5 cub. feet per hour, with a chimney 6 inches long and $1\frac{7}{8}$ inch internal diameter.

In France the standard illuminant is the *Carcel lamp*, burning 648 grains of pure oil per hour and equal to about $7\frac{1}{2}$ candles.

The German (Hefner) standard candle-power is only 88 per cent. of the English standard; German lamps are therefore rated too high compared with English. The Hefner lamp was adopted in 1896 as the practical international unit measure of light.

The intensity of light from these three sources is in the proportion of Hefner 1, Carcel 10·8, Pentane 11.

The candle-power of an arc light depends upon the angle at which it is measured. It is about four times as great at an angle of 40 degrees below the horizontal as it is when measured horizontally. On this account arc lights should be placed high, when they may be considered to illuminate a radius of eight times the height, making the economical distance apart sixteen times the height.

1,067. INTENSITY OF LIGHT.

The quantity of light received on a unit of surface is called its intensity. The intensity diminishes as the square of the distance.

Rumford's Shadow Photometer.—A rod is placed near a white screen, shadows are cast by a standard candle and the light to be tested. When the shadows are of equal intensity the distance of each light from the screen is measured and the illuminating power is the ratio of these distances squared.

Bunsen's Photometer.—A grease spot is made on a screen of white paper, the standard candle and the light to be tested are placed on opposite sides. The screen is moved until the grease spot becomes invisible, owing to equal illumination. The squares of the distances of the lights from the screen give the ratio of the illuminating power.

1,068. VISIBILITY OF LIGHT AT A DISTANCE.

1 candle-power visible at	. . . .	1 nautical mile.
3 " "	. . . .	2 "
10 " " (with opera glass)	. .	4 "
19 " " " "	. .	5 "

—AMER. INT. MAR. CONG., 1889.

1,069. THE BRODHUN GLOBE PHOTOMETER.

The general theory is as follows: When a source of light is placed inside a spherical shell having a matt surface, the light received by any part of the interior surface can be divided into two parts—(*a*) the light received directly from the lamp, and (*b*) the light received from the remainder of the interior surface of the sphere, after one or more reflections. The quantity (*a*) is that which is measured in the usual photometers which determine the intensity of the light emitted in one direction, and is not considered at all in the Globe photometer. By the theory of the Globe photometer the quantity (*b*) is constant all over the surface of the shell, and is proportional to the total amount of light emitted by the lamp, quite independently of its position in the shell.

In other words, if a lamp be hung inside a spherical shell with a matt lining, the illumination of any part of the shell which is screened from the direct light of the lamp is the same as of any other part, and measures the total amount of light generated by the lamp.

The strict application of the above argument requires a perfectly matt surface, and that the fittings or mechanism of the lamp shall not interfere with the repeated reflections by the surface of its own light; neither of which conditions can be entirely met in practice. The first condition is met approximately by coating the surface with lithopone (barium sulphate), and the second by making the shell large compared with the lamp; and experiments show that the errors do not prevent useful results being obtained.—ELECTROTECHNISCHE ZEITSCHRIFT, 1905.

1,070. ILLUMINATION UNITS.

The intensity of illumination is now generally given in "candle-feet," or, in other words, the particular illuminating effect at any point is expressed in terms of the equivalent number of candles at a distance of 1 foot away from that point. Ordinary street lighting illumination varies from 0·03 to 1·5 candle-feet, anything over 1 is very good, the readings being taken at 4 feet 6 inches above the ground.

M.C.F. = mean candle-feet.

1,071. COLOUR OF ARTIFICIAL LIGHTS.

Recorded observations of the spectrum of various artificial lights as tabulated by Munsterberg give the following results, taking as unity the monochromatic light of each of the five main colours of the sun's spectrum:—

Colour in Spectrum.	*Electricity.*		*Coal Gas.*		*Acetylene.*		*Sun-light.*
	Arc.	*Incan-descent.*	*Self lumin-ous.*	*Incan-descent.*	*Alone.*	*With 3 per cent. Air.*	
Red	2·09	1·48	4·07	0·37	1·83	1·03	1
Yellow	1·00	1·00	1·00	0·90	1·02	1·02	1
Green	0·99	0·62	0·47	4·30	0·76	0·71	1
Blue	0·87	0·91	1·27	0·74	1·94	1·46	1
Violet	1·03	0·17	0·15	0·83	1·07	1·07	1
Ultra violet	1·21	—	—	—	—	—	—

—W. Kennedy.

1,072. EFFECT OF VARIATION IN PRESSURE ON COAL GAS.

No. 4, Union Jet Burner.

Pressure in inches.	*Consumption in cub. feet.*	*Unit efficiency in Candles.*	*Total c.p.*
0·5	3·9	3·0	11·70
1·0	5·6	2·4	13·44
1·5	7·13	1·9	13·55
2·0	8·5	1·5	12·75
2·5	9·6	1·22	11·71
3·0	10·5	1·11	11·66

Incandescent gas mantles at ordinary pressure give 20 c.p. per cub. foot per hour.

A Block burner with a single large mantle gives a light of 300 c.p. with a consumption of 6 cub. feet per hour.

A 1,000 c.p. millennium light requires only ·0344 cub. feet per c.p. per hour, and 1,000 c.p. costs 1·103d. per hour.

The most economical artificial light known at present is the Keith light, where the normal pressure of ordinary coal gas is increased by a compressor to 8-inch water gauge, and burnt in specially designed incandescent burners costs 1d. per 1,000 c.p. per hour.

1,073. ABSORPTION OF LIGHT BY GLASS.

Light is diminished in transmission through glass, as follows :—

British polished plate, ¼ inch thick	13 per cent.
32-ounce sheet glass	22 ,,
Rough cast plate, ¼ inch thick . . .	30 ,,
Rough rolled plate, ¼ inch thick . . .	53 ,,

1,074. LOSS OF LIGHT BY GLASS SHADES.

Clear glass	10·57 per cent.
Ground glass	29·48 ,,
Smooth opal	52·83 ,,
Ground opal	55·85 ,,
Ground opal with painted figures . . .	73·98 ,,

—W. KING.

Plain glass globes 10 per cent., frosted glass, 25 per cent., opal or coloured, over 50 per cent.—A. B. HOLMES.

1075. EFFICIENCY OF REFLECTORS.

Polished aluminium	81 per cent.
White enamelled	79 ,,
Nickelled copper	68 ,,

—J. S. DOW.

1,076. SIZE OF GAS PIPES.

For lighting houses and factories.

Diameter ⅛ths × 20 = maximum length feet.

$(d\ \tfrac{1}{8}\text{ths})^2 \times 3$ = capacity of supply in feet per hour.

The distribution of the light in numerous jets increases the convenience but decreases the total quantity of light, e.g. :—

5 cub. feet per hour burnt in 1 jet gives a light = 28 candles.
,, ,, ,, 2 jets, each 2½ ft., = 21·16 ,,
,, ,, ,, 5 jets, each 1 foot, = 15 ,,

Minimum size of gas tubing usually required for :—

3 lights . .	¼ inch	50 lights . .	1 inch
6 ,, . .	⅜ ,,	75 ,, . .	1¼ ,,
12 ,, . .	½ ,,	110 ,, . .	1½ ,,
18 ,, . .	⅝ ,,	150 ,, . .	1¾ ,,
27 ,, . .	¾ ,,	200 ,, . .	2 ,,

Gas stoves should have a separate pipe from the meter; a gas fire for warming requires not less than ½-inch iron pipe; a gas cooker requires from ½ inch to ¾-inch pipe. according to its size and the number of jets kept burning at one time.

1,077. MILD STEEL CYLINDERS FOR STORING HIGH PRESSURE GASES.

Standard pressure for gas = 120 atmospheres.

	Solid Drawn.	*Lap Welded.*
Ultimate tensile strength lbs. per sq. inch sectional area of metal	66,000	54,000
Standard sizes:	*Thickness.*	
External diameter, 4 inches	$\frac{5}{32}$ inch	$\frac{6}{32}$ inch
,, ,, 5½ ,,	$\frac{7}{32}$,,	$\frac{8}{32}$,,
,, ,, 7 ,,	$\frac{9}{32}$,,	not made
Bursting pressure per sq. inch	2½ tons	2¼ tons
Limit of elasticity per sq. inch of metal	45,000 lbs.	35,000 lbs.
Actual hydraulic test pressure per sq. inch	1½ tons	1¼ tons

Prof. Goodman's formula:—

P = internal bursting pressure tons per sq. inch.
d = internal diameter of cylinder in inches.
t = thickness of sides of cylinder in inches.
x = percentage of extension on the material at the stress f (say 20 per cent. on 10 inches).
f = maximum stress on the material in tons per sq. inch (say 32 tons).

$$P = \frac{2ft}{d\left(1 + \frac{x}{100}\right)}.$$

1,078. CALORIMETER TEST OF GAS.

The calorimeter consists of a metal cylinder with a water inlet and thermometer at bottom and a water outlet and thermometer at top. The gas passing through a meter is burnt from a Bunsen burner under a cone projecting into bottom of cylinder. The apparatus is worked until the conditions are constant, the gas consumption is then noted, while a given quantity of the hot water is run off and the temperature at inlet and outlet are noted. The water condensed from the gas is collected during the same time.

Example :—

Temperature of water at inlet = 10° C.

Temperature of water at outlet = 28° C.

Gas consumption for 2 litres of water = $\frac{14}{60}$ cub. feet.

Water condensed = 22·5 c.c.

$$\text{Then heat} = \frac{\text{water} \times \text{temperature}}{\text{gas}}.$$

$$\text{Calories (C)} = \frac{2 \text{ litres water} \times (28 - 10)\ ^\circ \text{C. rise of temperature}}{\frac{14}{60} \text{ cub. feet gas}} =$$

154·3 gross.

Subtract for condensed water :—

$$22{\cdot}5 \text{ cub. centimetres} \times {\cdot}6 \text{ constant} = \frac{13{\cdot}5}{140{\cdot}8} \text{ net.}$$

$$140{\cdot}8 \times 3{\cdot}9684 \text{ constant} = 558{\cdot}75 \text{ B.Th.U.}$$

1,079. INTERNAL COMBUSTION ENGINES.

A hot-air engine (e.g., Stirling's) is in theory more economical of fuel than a steam engine, but no appreciable advance towards practical realisation was made for fifty years, the great difficulty to be overcome being the slow transfer of heat through metal to air. The principle of internal combustion, however, not only removes this difficulty, but the sources of failure in the old air engine become the chief elements of success in the new. An internal combustion engine is a heat engine in which the working fluid is atmospheric air, and the fuel is an inflammable gas or vapour, and it differs from a hot air or steam engine in one important point—that is, the heat to supply the motive power is given directly to the working fluid by combustion within the motor cylinder.—DUGALD CLERK.

1,080. DEVELOPMENT OF THE GAS ENGINE.

Lenoir gas engine (1860). Mixture of gas and air drawn into cylinder and fired at half stroke, exhaust made on return stroke. Engine double acting, explosion occurring on each side of piston at every stroke. Cost of working excessive, as 100 cub. feet coal gas were required per I.H.P. per hour.

Otto and Langen gas engine (1867). Free piston driven up by explosions, continued movement causing a vacuum, return stroke made by atmospheric pressure. Economy greater than the Lenoir, but obtained at expense of practical convenience, the engine being noisy, irregular, and uncertain in action.

Otto cycle (1876) applied to gas engines by various makers up to 1903.

Chiefly developed by Messrs. Crossley Brothers, Halifax; The British Westinghouse Company, The Premier Gas Engine Company, of Sandiacre; Messrs. Fielding and Platt, of Oldham; and The Société Cockerill, of Seraing.

The Dugald Clerk gas engine (1876), improved by Messrs Körting (1903) and introduced to this country by Messrs. Mather and Platt, is described in "Engineering," December 18th, 1903, p. 845, and is now being made to develop 2,000 B.H.P. in twin cylinders.

The latest type of engine is the Diesel, and this promises to give very good results, but is still in course of development.—D. S. CAPPER.

1,081. GAS ENGINES.

The Otto cycle of working :—

First revolution	Outstroke draws in air and gas. Instroke compresses charge.
Second revolution	Outstroke caused by the explosion. Instroke discharges burnt products.

In a single cylinder 2-cycle gas engine the number of power strokes = number of revolutions.

In a single cylinder 4-cycle gas engine the number of power strokes = half number of revolutions.

Mean effective pressure averages 80 lbs. per sq. inch.

The Körting type is most suitable for gas engines of large power where each side of the piston receives an impulse at every revolution, or four times as many impulses as the Otto in the same number of revolutions. Mather and Platt build these engines up to 1,000 B.H.P. in one cylinder, or double this in two cylinders.

1 B.H.P. from a gas engine requires about 15 cub. feet of gas per hour, but this will depend upon the thermal units in the gas, which range from about 500 to 700.

Producer-gas (invented by Dr. Ludwig Mond) is the most economical for working gas engines and for heating purposes.

1,082. HEATING VALUE OF GASES.

Blast furnace gas . . .	90	B.Th.U. per cub. foot.
Water gas	90	,, ,, ,,
Producer gas	105	,, ,, ,,
Carbon monoxide CO . .	320	,, ,, ,,
Hydrogen H_2	320	,, ,, ,,
Marsh gas CH_4 . . .	1000	,, ,, ,,
Illuminating gas C_2H_4 . .	1600	,, ,, ,,

Another account:—

Coal gas (mixed) 14 c.p. . .	500	B.Th.U.	per	cub.	foot.
Coal gas (enriched) 18 c.p. .	650	,,	,,	,,	
Mond gas . . .	140 to 148	,,	,,	,,	
Blue water gas . . .	328	,,	,,	,,	

1,083. EFFICIENCY OF POWER GASES.

$$\text{General efficiency of manufacture} = \frac{\text{calorific value of gas} \times \text{yield per lb. of fuel}}{\text{caloric value of fuel used}}$$

Coal gas alone	efficiency	25 per cent.
Coal gas, including coke	,,	60 ,,
Water gas	,,	60–75 ,,
Producer gas	,,	70–85 ,,

1,084. THEORETICAL EFFICIENCY OF POWER GAS PRODUCTION.

1 lb. anthracite coal	13,495 B.Th.U.
Gas produced from 1 lb. coal	11,573 ,,
Efficiency of conversion 86 per cent.	
1 lb. bituminous coal	14,375 ,,
Gas produced from 1 lb. coal	12,995 ,,
Efficiency of conversion 90 per cent.	

But coal burnt under a steam boiler may only realise $\frac{8}{13}$ of its energy = 61 per cent. while the gas carries its whole energy into the gas engine.

1,085. POWER GAS PLANTS.

Town gas.—Suitable only for small engines say up to 60 H.P., very handy for rapid starting, but not so economical as producer gas.

Pressure producer gas.—Dowson's system requires anthracite coal or coke, and a steam boiler to produce the steam necessary in forming the gas. Economy falls off rapidly as full load drops, and plant gives off unpleasant smell. Average of 1½ lb. coal required per B.H.P. per hour.

Körting, Wilson, Duff, Mond, etc., use refuse bituminous coal. A steam boiler, generator, regenerator, blower, scrubbers, and gasholder are required. Economical for large powers, i.e., over 150 H.P., using 1 to 1¼ lb. of slack per B.H.P. per hour.

Suction producer gas.—The principle of the plant is that the gas engine, by drawing air and water vapour through an anthracite or coke fire, makes its own gas as it wants it. A generator, vaporiser, and one or two scrubbers are required, with a fan for starting; the apparatus is very compact, and suitable for 10 to 150 H.P. The consumption averages 1 lb. coal per B.H.P. per hour.

1,086. SUCTION GAS PRODUCER.

The Suction Gas plant is the best type of power generator for gas engines from 10 to 100 H.P. It is the simplest and cheapest method of obtaining power, requires no boiler or chimney, occupies very little space, requires no gasholder, is ready for work in a few minutes, can be tended by an ordinary labourer, does not require constant attention, is perfectly safe and automatic in action, produces no smoke, soot, tar, or smell when working, may be fixed in cellars or basements of buildings, and comprises :—

Generator formed of riveted steel casing lined with firebrick, having a closed ashpit, the usual grate bars, cleaning doors, poker holes and gas outlet at the top.

Vaporiser in the form of a tortoise boiler which takes up the heat of the gases passing from the generator, the steam thereby raised being drawn through the generator and enriching the gas.

Automatic Water Feed arranged for the vaporiser, and the steam is automatically drawn through the fire.

Feed Hopper on the top of generator with small platform for feeding same, with a suitable box shape scoop.

Washer and Cooler, formed of steel casing with suitable trays, taps, pipes to connect the same to generator supplied with water spray pipe, siphon overflow, cleaning doors, gas outlet pipe ready for supply, and draw-off pipes to be connected.

Sawdust box arranged with screens, and gas storage box fixed above the scrubber or washer.

Hand Fan for starting up the plant, which can be re-started in a few minutes after standing over-night. This is provided with suitable blow-off pipe, arranged to be carried to the atmosphere at the nearest point.

Cost of Production.—One penny per 1,000 cub. feet.—MASON'S GAS POWER Co.

1,087. GAS PRODUCER REGULATIONS

have been issued by the National Board of Fire Underwriters, of New York. Pressure systems must be located in independent buildings, but suction producers up to 250 H.P. may be placed in a separate, enclosed, well-ventilated room in any building where the natural light is good. While the plant is not in operation the connection between the generator and scrubber must be closed, and the connection between the producer and vent pipe opened, so that the products of combustion can be carried into the open air. This must be accomplished by means of a mechanical arrangement which will prevent one operation without the other. If illuminating or other pressure gas is used as an alternative supply the connections must be so arranged as to make the mixing of the two gases or the use of both at the same time impossible. The opening for admitting fuel must be provided with some charging device so that no considerable quantity of air can be admitted while charging.

1,088. ADVANTAGES OF GAS ENGINES.

"In construction, an ordinary gas plant saves the cost of boilers and chimney-shaft, and occupies less space; thereby effecting an economy in cost of building and cost of land. In maintenance, there ought to be a saving in labour; no stoking is required when working off coal gas, and as a rule the attendance upon engines and dynamos is shared by the electrical attendants—engine drivers and stokers are non-existent. With Dowson's plant more space is occupied than if coal gas only is relied on; and a man is also required for stoking and attendance upon generators. It remains to be seen if sufficient economy attaches to the employment of Dowson gas to compensate for the additional cost of plant, space, buildings, and, as regard maintenance, repairs and labour."—W. LANGDON.

1,089. BRAKE HORSE-POWER OF GAS ENGINE.

A = area of piston in sq. inches.
S = stroke in feet.
N = number of revolutions per minute.
C = 600 for 2-cycle engine.
400 ,, 4-cycle ,,

$$\text{B.H.P.} = \frac{A\,S\,N}{C}.$$

1,090. COMPARISON OF GAS AND OIL ENGINES.

In a gas engine, either coal gas, or some kind of specially prepared gas, such as Dowson gas, is drawn into a cylinder, compressed, exploded, and expanded in successive strokes of the engine. Precisely the same thing takes place in an oil engine, except that, instead of a permanent gas being employed, drawn from the public main or special generator, the inflammable vapour from some kind of oil is used, prepared within the engine itself and requiring no storage room beyond that which is sufficient to contain the oil to work the engine, and the tank for holding the jacket cooling water which is common to both kinds of engines. It is more economical than the gas engine, but requires more skilled attention in working.—W. C. POPPLEWELL.

1,091. OIL ENGINES, TEMPERATURE OF FUEL.

(*a*) Fuel vaporises at ordinary temperatures—e.g., benzine, as used in Daimler motors.

(*b*) Fuel heated to 75°—150° F. before vaporisation—e.g., heavy oils as used in Priestman, Fielding and Platt, Tangye, or Hornsby-Ackroyd engines.

The fuel being vaporised the engine becomes virtually a gas engine.

1,092. OIL ENGINES. CLASSIFICATION BY VAPORISERS.

1. Oil is injected, together with the whole of the air supply required for its combustion, into a large vaporising chamber, separate from the cylinder, as in Priestman's engine.

2. Oil is injected into a small vaporising chamber together with some air, but the greater volume of air enters the cylinder through a separate valve, as in Fielding and Platt's engine.

3. Oil is injected into the cylinder through a vaporiser, together with all the air, as in Tangye's engine.

4. Oil is injected into the combustion chamber, and is vaporised therein. The air for its combustion is drawn into the cylinder through a separate valve, and is compressed into the chamber to mingle with oil vapour and form the explosive mixture, as in the Hornsby-Ackroyd engine.—F. Grover.

1,093. OIL ENGINES.

Consumption of oil about 1 lb. per B.H.P. per hour.

Priestman Oil Engine.—General appearance similar to gas engine. Eccentric on crank shaft works small air pump alongside cylinder by means of which air is compressed and pumped into oil supply tank which is placed in engine bed. The oil under pressure, and the compressed air as well, are admitted to the "spray maker" in which the jet of oil is broken up into fine spray, which mixes with more air and is admitted to the "vaporiser," a closed chamber round which the hot exhaust gases from the cylinder circulate and keep it at a high temperature. The vapour thus formed from the spray and air are admitted to the cylinder, compressed, ignited, and expanded as in a gas engine working on the Otto cycle. Ignition is effected by an electric spark sent across two terminals in the cylinder at the proper moment. The best oil for use in this engine is a fairly heavy petroleum or "paraffin," sp. gr. 0·80 to 0·82, and flashing point 100° to 150° F.

Hornsby-Akroyd Oil Engine differs from the Priestman engine in having vaporising chamber adjacent to and communicating with cylinder instead of separate vaporiser. Works on the Otto four-stroke cycle. During suction stroke air alone is drawn into cylinder. Clearance space connected with further chamber at back of cylinder, and connected with it by small bottle neck opening. During the compression stroke the heated compressed air is forced into this "combustion chamber," which is kept at a red heat by the explosion and combustion of successive charges of oil vapour and air, at the same time a few drops of oil are forced into combustion chamber by small pump. As the oil comes in contact with the red-hot interior, which is fitted with internal radiating ribs, it is at once converted into a gaseous state. This oil vapour, with the compressed air, forms the explosive charge. Ignition is effected spontaneously, the air being raised in temperature by the compression, and the heat of the combustion chamber, are sufficient to ignite the charge and cause explosion. The oil in this engine is generally 0·85 sp. gr. and 150° F. flashing point, but slightly heavier oil may be used.

Consumption, ¾ pint of oil (at ½d. per pint) per B.H.P. per hour at full load. Oil, 8½ lbs. per gallon, 96° flashing point. Vaporiser heated by lamp for starting.

Otto Oil Engine.—The Otto cycle is employed. During the suction stroke air is drawn past a nozzle through which oil is allowed to flow from a tank, and a spray of oil is formed. This spray passes through a hot chamber on its way to the cylinder, where it is mixed with more air, and thus forms the explosive charge to be compressed and fired. The charge is exploded by means of a tube igniter. The oil used is 0·82 sp. gr. and 86° F. flashing point, but heavier oils may be used.

Zephyr Oil Engine.—Similar in principle to condensing steam engine. Light oil spirit evaporated in a boiler, same as water in a steam boiler, but pressure rises more quickly for given rise of temperature; work done on piston, and vapour afterwards condensed and returned to boiler.

—W. C. Popplewell.

The *Diesel Oil Engine* (Mirrlees Watson Co., Ltd., Glasgow) is single acting, with four-stroke cycle—viz. (1) on first down stroke of piston, air drawn direct from atmosphere to cylinder; (2) on first upstroke air compressed and temperature rises sufficiently to cause ignition of fuel when admitted; (3) at first portion of next downstroke when compression at maximum, fuel oil blown into cylinder by compressed air, amount of oil being determined by governor; (4) on next upstroke products of combustion expelled to atmosphere. Advantages: no sudden rise of pressure on explosion, no possibility of back-firing as air only is compressed, no carburettor or vaporiser or separate ignition device required, engine arranged vertically, almost any class of oil can be used, valves and other parts well arranged for accessibility and automatic lubrication.

An installation consisting of an 80 B.H.P. Diesel Engine coupled direct to a continuous current dynamo, working an average of nine hours per day, will show a cost for current including interest on outlay, depreciation, labour, stores, and fuel of about two-thirds of a penny per unit (0·67d.) as compared with 2d. to 4d. per unit charged for the supply from central stations. The consumption of fuel is less than ½ lb. of crude oil per B.H.P. hour, costing from 2d. to 3d. per gallon. Cost of running about ⅛d. per B.H.P. hour.

In the Diesel oil engine the governor acts directly upon the fuel supply, and the momentary increase or loss of speed with a 25 per cent. change of load does not exceed 2 to 3 per cent. At half-load the fuel consumption per B.H.P. shows practically no increase over that at full load, as it varies in direct proportion to the work done and is under ½ lb. crude oil per B.H.P. hour. The combustion is perfect, and the exhaust always clear and free from smell and unburnt gases. The engine is made in all sizes from 20 to 1,000 B.H.P. Can be started as easily as an engine under steam, and has no stand-by losses.

1,094. COMPARISON OF MINERAL OILS.

Constituent Oils.			*Percentage by Volume.*		
Name.	*Sp. gr.*	*Flashing point ° F.*	*Scotch shale oil.*	*American petroleum.*	*Russian petroleum.*
Benzine-naphtha	·68 to ·75	15 to 32	5	16	4
Kerosene (lamp oil)	·80 to ·82	77 to 122	35	50	27
Intermediate	·85 to ·86	220	2	—	12
Lubricating	·88 to ·89	230 to 490	18	15	32
Paraffin wax	—	—	12	2	1
Residue	—	—	28	17	24

—ROBINSON.

Section XIV.

SUNDRY NOTES AND TABLES.

1,095. MATHEMATICAL CONCEPTS.

In ARITHMETIC we deal with *number*, and by inference with *magnitude* or *quantity*. In GEOMETRY we add the ideas of *space* and *direction*. In STATICS we add to the foregoing the idea of *pressure*, and in DYNAMICS we add *force* and *motion*.

1,096. GEOMETRICAL DEFINITIONS.

A *definition* is a strict and complete description.

An *axiom* is a truth admitted without demonstration.

A *postulate* is something to be done of which the possibility is admitted.

A *theorem* is a geometrical truth capable of demonstration by reasoning from known truths.

A *problem* is a geometrical construction to be effected by the aid of certain instruments.

A *corollary* is a geometrical truth easily deducible from a theorem.

Q.E.D. stands for *quod erat demonstrandum*, and is placed at the end of a *theorem* to mark that the truth of the theorem has been proved.

Q.E.F. stands at the end of a *problem*, and means *quod erat faciendum*, to show that the problem has been done.

A *reflex* angle is that which is greater than two right angles.

A *scalene* triangle has three unequal sides.

The opposite angles made by two straight lines which intersect are called *vertically opposite angles*.

The following signs may be used in writing out propositions :—$\therefore$ Therefore, $\because$ because, $\parallel$ parallel, $\parallel$m. parallelogram, $=$ equal to, $\angle$ angle, $\llcorner$ right angle, $\odot$ circle, sq. square, sqq. squares, $\triangle$ triangle, $\triangle$s. triangles.

—D. WALKER.

1,097. TERMS USED IN MODERN GEOMETRY.

Most of the circumlocutory phrases of the ancient geometry are replaced by single names—e.g., a line which crosses *any* system of lines is a *transversal.* A line which bisects an angle or another line is a *bisector.* A system of lines co-intersecting in a point are *concurrent,* and the sum total of all the lines that can lie in a point is a *range* or *pencil.* Lines, and sides of triangles upon the same base, that terminate in the same point, are *conterminous.* A system or *range* of points through which a line can be drawn are *collinear.* A system of not less than four points, through which a circle can be drawn, are *concyclic.* The centre of the circle inscribed in a triangle is the *incentre;* of the escribed circle is the *excentre;* and of the circumscribing circle is the *circumcentre.* The intersection of the *altitudes* or perpendiculars drawn from the vertices to the opposite sides is the *orthocentre,* and the intersection of the *medians,* or lines drawn from the vertices to the middle points of the opposite sides, is the *centroid* (= "*centre of gravity*") of the triangle.

Notwithstanding the prolific multiplication of special names, names of wide connotation are preferred to those of narrow extension—e.g., figures which are symmetrical or regular are defined as such without special names, the "isosceles" triangle of Euclid being now known as the *symmetrical* triangle, for no other triangle is symmetrical; and the *equilateral* triangle as the *regular* triangle. The term *conjugate* defines two angles, which together make *an angle of continuation,* two arcs or sectors which together make a circle, and several other pairs of related elements. In *congruent* figures—i.e., figures which are identically equal—the name *corresponding* is given to all pairs of points which coincide when the two figures are made to coincide. The intersection of a number of lines, planes, etc., is in every case a *join.* Many names are such as to denote special properties, quite independent of the relation which the figure bears to others. Such, for example, are the *nine point* circle, the *triplicate ratio* circle, the *cosine* circle. To this class also probably belong such names as *Kite,* denoting any quadrilateral which is symmetrical about one of its diagonals, no matter what its shape; *pedal* triangle, denoting the triangle formed by joining the "feet" of the "altitudes" of an exterior triangle, etc. Lastly, it is a growing but perhaps a reprehensible practice, to denote elements having special properties by the names of their discoverers. Thus we have *Brocard* points, *Brocard* circle and ellipse, *Taylor* circles, *Tucker* circles, the *Lemoine* point, *Tarry's* point, *Simson's* line, etc.—C. E. GROVE.

1,098. MATHEMATICAL SIGNS.

$+$ Plus, or add.	$\backsim$ Difference of.
$-$ Minus, or subtract.	$<$ Less than.
$\times$ Multiply by.	$>$ Greater than.
$\div$ Divide by.	$\leqq$ Equal to or less than.
$=$ Is equal to.	$\geqq$ Equal to or greater than.
$\therefore$ Therefore.	$\propto$ Varies as.
$\because$ Since, because.	∞ Infinity.
(Single bracket.	$\parallel$ Parallel with.
{ Double bracket.	$\perp$ Perpendicular to.
[Square bracket.	$\overline{}$ Vinculum or bar.

$\pm$ Plus minus—i.e., either plus or minus, according to circumstances.

Σ *Sigma*, the sum, or "summation of the products of."

π *Pi*, the ratio of circumference to diameter.

θ *Theta*, angle of incidence.

μ *Mu*, coefficient of friction, also $\frac{1}{25000}$ inch in bacterial microscopy.

ϕ *Phi*, angle of repose.

$\odot$ Circle, or station point. $\triangle$ Triangle, or Trig. station.

$\sqrt{}$ Square root. $\sqrt[3]{}$ Cube root. $\sqrt[4]{}$ Fourth root.

$$2\,\overline{a\,b}|^2 = 2\,(a\,b)^2.$$

$\underline{|5}$ means continued product up to $5 = 1 \times 2 \times 3 \times 4 \times 5$.

a b c used for known quantities; x y z for unknown.

$\sin^{-1} a =$ the angle whose sine is a.

$\int_o^x$ is sign of integration between limits o and x.

The symbol $\int_a^b y\,.\,d\,x$ means "the area of the curve whose ordinate is y from $x = a$ to $x = b$."

$\Sigma \sin^3 \theta \int_{0°}^{90°}$ means the summation of the cubes of the sines of the angles from 0 to 90.

Note.—The sign $+$ or $-$ has a higher separating power in a formula than $\times$, therefore parts connected by $\times$ may be multiplied out before passing beyond the other signs. Remember that plus times plus $= +$, plus times minus $= -$, minus times minus $= +$.

1,099. GREEK ALPHABET AND USE OF LETTERS.

Α α	Β β	Γ γ	Δ δ	Ε ε	Ζ ζ	Η η	Θ θ	Ι ι	Κ κ
Alpha,	bēta,	gamma,	delta,	epsilon,	zēta,	hēta,	thēta,	iōta,	kappa,
Λ λ	Μ μ	Ν ν	Ξ ξ	Ο ο	Π π	Ρ ρ	Σ σ ς	Τ τ	Υ υ
lambda,	mu,	nu,	xi,	ŏmīcron,	pi,	rho,	sigma,	tau,	ūpsilon,
Φ φ	Χ χ	Ψ ψ	Ω ω.						
phi,	kai,	psi,	ōmĕga.						

Some of the above Greek letters are commonly used by engineering writers —e.g., α for angle of intersection of tangents in setting out railway curves, β for centre angle of railway curve, γ for angle of deflection of chord of railway curve, Δ or δ for deflection of beam, θ for angle between sloping roof and direction of wind, λ for a length or angle, μ for coefficient of friction, π for ratio of circumference of circle to diameter, ρ for radius of curve, Σ for sign of summation, or total of a series of quantities, τ for temperature. ω for a length or angle.

1,100. LINEAL MEASURE.

7·92 inches	= 1 link.
12 inches	= 1 foot.
3 feet	= 1 yard.
6 feet	= 1 fathom.
25 links, or 5½ yards	= 1 rod or pole.
100 links, or 66 feet, or 4 poles	= 1 chain (Gunter's).
40 poles, 10 chains, or 660 feet	= 1 furlong.
320 poles, or 80 chains, 8 furlongs, 5,280 feet, or 1,760 yards	= 1 mile.
120 fathoms	= 1 cable length.
6080·27 feet	= 1 nautical mile.
6080 feet per hour	= 1 Admiralty knot.

1,101. SQUARE MEASURE.

144 sq. inches	= 1 sq. foot.
9 sq. feet	= 1 sq. yard.
625 sq. links, or 30¼ sq. yards	= 1 perch.
40 perches, or 2½ sq. chains	= 1 rood.
100,000 sq. links, 160 perches, 10 sq. chains, or 4 roods.	= 1 acre.
640 acres	= 1 sq. mile.
43,560 sq. feet, or 4,840 sq. yards	= 1 acre.
640 acres.	= 1 sq. mile.

1,102. CUBE MEASURE.

1728 cub. inches	= 1 cub. foot.
1·2832 cub. feet	= 1 bushel.
27 cub. feet or 21 bushels	= 1 cub. yard.

1,103. MEASURES OF WEIGHT.

1 lb. troy	= 5,760 grains.
1 lb. avoir.	= 7,000 „
175 lbs. troy	= 144 lbs. avoir.
Lbs. troy × 1·2153.	= lbs. avoir.

Lbs. avoir. × ·82286	= lbs. troy.
1 cental	= 100 lbs.
1 short ton, or American net ton . .	= 2,000 lbs.
1 metric ton = 1,000 kilos . . .	= 2,204·6 lbs.
1 ton = 20 cwts.	= 2,240 lbs.

When no special mention is made pounds (lbs.) are understood to be avoirdupois.

1,104. ENGLISH APOTHECARIES MEASURE.

ↄ = ½ grain, o = 1 grain, ↄjs. = ½ scruple, ℈j = 1 scruple, ʒjs. = ½ dram, ℈ij = 2 scruples, ʒj = 1 dram, ʒij = 2 drams, ℥j = 1 ounce, 1 bj = 1 pound.

20 grs. = 1 scr., 3 scr. = 1 dr., 8 drs. = 1 oz., 12 oz. = 1 lb.

For typing prescriptions the following symbols are used:—Pint = Oj, dram = 3j, minim = mj, grain = grj, per cent. = %.

1,105. ARITHMETICAL TERMS.

Item	Minuend	Multiplicand		Factor
Item	Subtrahend	Multiplier	or	Factor
Sum	Difference	Product		Product

Divisor) Dividend
Quotient

$$\text{Fraction} = \frac{\text{Numerator}}{\text{Denominator}}.$$

1,106. NOMENCLATURE OF LARGE NUMBERS.

The 0's show the number of places occupied by each group.

In England—

Billions.	*Millions.*	*Thousands.*	*Units.*
000,000	000,000	000	000

In France and United States—

Quadrillions.	*Trillions.*	*Billions.*	*Millions.*	*Thousands.*	*Units.*
000	000	000	000	000	000

But in France a thousand millions is colloquially called a "milliard" instead of a billion.

Scientific method of notation 29,000,000 = 29×10^6.

1,107. DUODECIMALS.

	ft.	ins.	
	ft.	ins.	
(ft. × ft.)	(ft. × ins.)		
	(ins. × ft.)	(ins. × ins.)	
sq. ft.	twelfths	sq. ins.	

Commonly called feet, inches, and parts.

1,108. MULTIPLICATION AND DIVISION.

Say 12·345 × 6·789.

Ordinary form :

```
    12·345
     6·789
   -------
    111105
    98760
   86415
  74070
 ---------
 83·810205
```

Contracted form :

```
          12·345
          987·6
         -------
           74070
            8642
             987
             111
Ans. 83·81  83·810
```

Say $\frac{5623}{1547}$

```
1547 ) 5623 ( 3·6347 etc.
       4641
       ·9820
        9282
        ·5380
         4641
         ·7390
          6188
          12020
          10829
          ·1191
```

```
1547 ) 5623 ( 3·635
       4641
       .982
        928
        ·54
         46
         ·8
          8
Ans.  3·635
```

For the contracted multiplication reverse the multiplier.

In 2nd line	7 × 5 = 35	∴	carry	4	to	(7 × 4).
„ 3rd „	8 × 4 = 32	∴	„	3	„	(8 × 3).
„ 4th „	9 × 3 = 27	∴	„	3	„	(9 × 2).

For the contracted division.

In 2nd line	6 × 7 = 42	∴	carry	4	to	(6 × 4).
„ 3rd „	3 × 4 = 12	∴	„	1	„	(3 × 5).
„ 4th „	5 × 5 = 25	∴	„	3	„	(5 × 1).

For modern examinations the contracted form is essential.

1,109. PRIME AND IRRATIONAL NUMBERS.

Prime numbers are those which have no divisor without remainder, as 3, 5, 7, 11, 13, 17, 19, 23, 29, 31, 37, 41, 43, 47, 53, 59, 61, 67, etc.

Numbers *prime to each other* are those which have no divisor in common.

Irrational numbers, or *surds*, or *incommensurables* are those for which square roots cannot be expressed, as 2, 3, 5, 6, 7, 8, 10, etc. The ratio π is an incommensurable quantity, its value has been found to 750 places of decimals without reaching finality.

1,110. ARITHMETICAL AND GEOMETRICAL MEANS.

Arithmetical mean of a and b or simply the mean of a and b $\Big\} = \frac{a+b}{2}$

Geometrical mean of a and b or mean proportional of a and b $\Big\} = \sqrt{a\,b}$

If A G H be the arithmetic, geometric, and harmonic means between a and b,

$$A = \frac{a+b}{2}, \quad G = \sqrt{a\,b}, \quad H = \frac{2\,a\,b}{a+b};$$

$$\therefore\ A\,H = \frac{a+b}{2} \times \frac{2\,a\,b}{a+b} = a\,b = G^2.$$

1,111. ARITHMETICAL AND GEOMETRICAL SERIES.

Arithmetical Series: The *following* number is produced by a constant addition to the *preceding* number, as

1, 2, 3, 4, 5, 6, 7, 8,

or 1, 3, 5, 7, 9, 11, 13, 15.

Geometrical Series: The *following* number is produced from the *preceding* number by a constant multiplier, as

1, 2, 4, 8, 16, 32, 64, 128,

or 1, 3, 9, 27, 81, 243, 729, 2187.

1,112. ARITHMETICAL, GEOMETRICAL AND HARMONIC PROGRESSION.

Arithmetical :—

s = sum of series.
n = number of terms.
a = first term.
l = last term.
d = common difference.

$$s = \frac{n}{2}(a + l)$$

$$l = \frac{s}{\frac{1}{2}\,n} - a, \qquad l = a + (n - 1)\,d.$$

Geometrical :—

r = common ratio or constant multiplier.

$$l = a\,r^{n-1}$$

Harmonical :—

A series of terms is said to be in harmonical progression when the reciprocals of the terms are in arithmetical progression.

1,113. TYPES OF VULGAR FRACTIONS.

$$\tfrac{3}{16} \times 9 = \frac{3 \times 9}{16}, \qquad \tfrac{3}{4} \times \tfrac{5}{7} = \frac{3 \times 5}{4 \times 7}, \qquad \tfrac{2}{3} \div \tfrac{7}{11} = \frac{2 \times 11}{3 \times 7}.$$

$$\tfrac{5}{6} \div 3 = \frac{\tfrac{5}{6}}{3} = \frac{5}{6 \times 3}, \qquad 3 \div \tfrac{5}{6} = \frac{3}{\tfrac{5}{6}} = \frac{3 \times 6}{5},$$

$$\frac{\tfrac{3}{4}}{\tfrac{5}{8}} = \frac{3 \times 8}{4 \times 5}, \qquad \tfrac{3}{8} \text{ of } 16 = \frac{16 \times 3}{8}, \qquad \frac{4 \times \tfrac{1}{3}}{7} = \frac{4}{3 \times 7},$$

$$\tfrac{4}{5} \text{ of } \tfrac{5}{7} = \frac{4 \times 5}{5 \times 7}, \qquad \frac{23\tfrac{4}{5}}{7} = \frac{(5 \times 23) + 4}{7 \times 5} = 3\tfrac{14}{35},$$

$$\tfrac{3}{5} + \tfrac{4}{7} = \left.\begin{array}{l} 3 \times 7 = 21 \\ 5 \times 4 = 20 \\ 5 \times 7 = 35 \end{array}\right\} 41 = \frac{41}{35}, \qquad \tfrac{3}{5} - \tfrac{4}{7} = \left.\begin{array}{l} 3 \times 7 = 21 \\ 5 \times 4 = 20 \\ 5 \times 7 = 35 \end{array}\right\} 1 = \frac{1}{35}.$$

1,114. RECIPROCALS.

If the product of two quantities be equal to unity each is the reciprocal of the other, thus a and b are reciprocals when $a\,b = 1$, because $a = \frac{1}{b}$ and $b = \frac{1}{a}$. Similarly *sec* and *cos* of an angle are reciprocals because sec $\times$ cos $= 1$, also sec $= \frac{1}{\cos}$ and cos $= \frac{1}{\sec}$.

1,115. ALGEBRAIC FRACTIONS.

A fraction will not be altered in value if the sign of every term in numerator and denominator be changed.

If the signs be changed only in the numerator or only in the denominator the sign of the whole fraction must be changed.

$$\frac{-a}{-b} = \frac{a}{b}, \qquad \frac{-a}{b} = -\frac{a}{b}, \qquad \frac{a}{-b} = -\frac{a}{b}.$$

1,116. RATIO AND PROPORTION.

The ratio of 1 to 2 is $\frac{1}{2}$ or $\cdot 5$; the ratio of a to b is the fraction $\frac{a}{b}$; or in other words, the ratio between two quantities is the proportion the first bears to the second, and is represented by the first divided by the second, thus 1 is the $\frac{1}{2}$ of 2, and a is the $\frac{a}{b}$ of b. The ratio of a to b is variously written as $a : b$, $\frac{a}{b}$, $a \div b$, $^{a}/_{b}$, $a\,b$.

$a \times b$, $a \,.\, b$, $a\,b$ are three modes of indicating a multiplied by b.

$a \div b$, $\frac{a}{b}$, $a\,b$ are three modes of indicating a divided by b.

1,117. REDUCTION OF FRACTION TO LOWEST TERMS.

Example :—Reduce $\frac{913}{1079}$ to its lowest terms.

Thus—

$$\begin{array}{l} 913)1079(1 \\ \quad\;\; 913 \\ \quad\;\; 166)913(5 \\ \qquad\quad\;\; 830 \\ \qquad\quad\;\; \cdot 83)166(2 \\ \qquad\qquad\quad 166 \\ \qquad\qquad\quad \ldots \end{array}$$

Then, 83 being the least divisor without remainder is the ***highest common factor*** (H.C.F.), or the ***greatest common measure*** (G.C.M.), and

$$\frac{913 \div 83}{1079 \div 83} = \frac{11}{13}. \textit{ Ans. } \text{required.}$$

The ***least common multiple*** (L.C.M.) of a series of quantities is the smallest quantity which can be divided by each item of the series without remainder.

L.C.M. of 4, 8, and 12 = 12
of 5, 7, and 8 = 280
of $\frac{2}{3}$, $\frac{3}{4}$, and $\frac{4}{5}$ = $\frac{24}{60}$

The ***highest common factor*** (H.C.F.) of two or more algebraical expressions is the ***greatest common measure*** (G.C.M.) of the numerical coefficients prefixed to the highest power of each letter which divides every one of the given expressions.

The ***least common multiple*** (L.C.M.) of two or more algebraical expressions is the ***least common multiple*** (L.C.M.) of the numerical coefficients prefixed to the lowest power of each letter which is divisible by every power of that letter in the given expressions.

1,118. CURIOUS PROPERTIES OF RECURRING DECIMALS.

The same figure repeated after a decimal point gives the number of ninths, as $\cdot 3$ recurring $= \frac{3}{9}$, $\cdot\dot{7} = \frac{7}{9}$, etc. The vulgar fractions of $\frac{1}{11}$, $\frac{2}{11}$, etc., are represented by $\cdot\dot{0}\dot{9}$, $\cdot\dot{1}\dot{8}$, etc., the first figure going up 1 and the second figure down 1 with each rise of numerator. For sevenths there is a repeater of six decimals, keeping a constant order but commencing at a different point each time. The repeater is $\cdot\dot{1}4285\dot{7}$ for $\frac{1}{7}$, and if these figures are placed clockwise round the circumference of a circle the point of commencement is shown by the following table :—

Number of sevenths	1	3	2	6	4	5
Commencing figure	1	4	2	8	5	7

Common recurring decimals are $\cdot 833 = \frac{5}{6}$, $\cdot 6\dot{6} = \frac{2}{3}$.

Rule for finding value of recurring decimals: Subtract the non-recurring decimals from the whole sum, the remainder is the numerator of an equivalent vulgar fraction; the denominator consists of as many nines as there are recurring decimals and noughts for the non-recurring decimals—e.g., for $\cdot 8\dot{4}\dot{2}$, $842 - 8 = 834$, then $\cdot 8\dot{4}\dot{2} = \frac{834}{990}$, also $2{\cdot}25 \times 3{\cdot}\dot{5} = 2{\cdot}25 \times 3\frac{5}{9} = \frac{2{\cdot}25 \times 32}{9} = 8.$

1,119. FULL VALUE OF RECURRING DECIMALS.

Examples showing how full value may be obtained in calculations:—

Addition:

$\cdot\dot{3} + \cdot\dot{4} = \cdot\dot{7}$

$\cdot 3 + \cdot\dot{4} = \cdot 7\dot{4}$

$\cdot\dot{9} + \cdot 1 = 1{\cdot}0\dot{9}$

$\cdot\dot{3} + \cdot 4 = \cdot 7\dot{3}$

$\cdot\dot{7} + \cdot\dot{6} = 1{\cdot}\dot{4}$

$\cdot\dot{9} + \cdot\dot{3} = 1{\cdot}\dot{3}$

Subtraction:

$1 - \cdot\dot{3} = \cdot\dot{6}$

$1 - \cdot 6\dot{7} = \cdot 3\dot{2}$

$\cdot 75 - \cdot\dot{5}\dot{6} = \cdot 18\dot{4}\dot{3}$

$1 - \cdot\dot{4}\dot{5} = \cdot\dot{5}\dot{4}$

$\cdot\dot{4}\dot{5} - \cdot\dot{3}0\dot{3} = \cdot\dot{1}5124\dot{2}$

$\cdot 57 - \cdot 3\dot{2} = \cdot 24\dot{7}$

Multiplication:

$\cdot\dot{6} \times \cdot 8 = 5{\cdot}\dot{3}$

$\cdot\dot{3} \times \cdot\dot{4} = \cdot 147\dot{9}$

$\cdot\dot{7}\dot{2} \times \cdot\dot{6}\dot{3} = \cdot 46\dot{2}763$

$\cdot\dot{7} \times \cdot 5 = 3{\cdot}\dot{8}$

$\cdot 6\dot{9} \times \cdot 5\dot{9} = \cdot 42$

$\cdot 5\dot{5} \times \cdot\dot{7} = \cdot\dot{4}3209876\dot{5}$

Division:

$1 \div \cdot\dot{3} = \frac{1}{\cdot 3} \times \frac{3}{3} = \frac{3}{1} = 3$

$1 \div \cdot\dot{5} = 1 \times \frac{9}{5} = \frac{9}{5} = 1{\cdot}8$

$1 \div \cdot 6\dot{9} = 1 \times \frac{90}{63} = \frac{10}{7} = 1{\cdot}\dot{4}2857\dot{1}$

$1 \div \cdot\dot{6} = \frac{1}{\cdot 6} \times \frac{3}{3} = \frac{3}{2} = 1{\cdot}5$

$\cdot\dot{7}\dot{2} \div \cdot\dot{6}\dot{3} = \frac{72}{99} \times \frac{99}{63} = \frac{8}{7} = 1{\cdot}\dot{1}4285\dot{7}$

$\cdot 6\dot{9} \div \cdot 5\dot{9} = \frac{63}{90} \times \frac{90}{54} = \frac{7}{6} = 1{\cdot}1\dot{6}$

1,120. PERCENTAGES AND AVERAGES.

A percentage having been added to a quantity and a certain result produced, if it be afterwards desired to deduct the same, or to reduce the percentage allowed, it must not simply be taken from the last result, because, for example—

100 + 50 per cent. = 150, 150 — 50 per cent. = 75.

Let $a + m$ per cent. $= b$, then $a = \dfrac{b}{1 + \dfrac{m}{100}}$, and applying this to the above figures the original amount will be obtained thus—

$$150 \text{ less the added percentage } = \frac{150}{1 + \dfrac{50}{100}} = \frac{150}{1\frac{1}{2}} = 100.$$

Again, taking 60 at 20 per cent. + 30 at 40 per cent. does not give an average of $\dfrac{60 + 30}{2} = 45$ at $\dfrac{40 + 20}{2} = 30$ per cent., but $60 + 30 = 90$ at an average of $\dfrac{60 \times 20 + 30 \times 40}{60 + 30} = 26\frac{2}{3}$ per cent.

In the same way 20 workmen making an average of 40s. in one week, and 25 workmen making an average of 32s. in another week, will not give an average of $\dfrac{20 + 25}{2} = 22\frac{1}{2}$ at $\dfrac{40 + 32}{2} = 36$s., but $22\frac{1}{2}$ at $\dfrac{20 \times 40 + 25 \times 32}{20 + 25}$ $= 35\frac{5}{9}$s.

1,121. POWERS AND ROOTS.

$$a \times a = a^2, \quad (a^2)^3 = a^6, \quad a^{\frac{1}{2}} = \sqrt{a}, \quad a^{\frac{2}{3}} = (\sqrt[3]{a})^2 = \sqrt[3]{a^2},$$

$$a^{3\cdot 5} = a^{\frac{7}{2}} = (\sqrt{a})^7,$$

$$a^{1\cdot 5} = a \times \sqrt{a}, \quad a^{-2} = \frac{1}{a^2}, \quad a^{\frac{-1}{n}} = \frac{1}{\sqrt[n]{a}},$$

$$a^{\frac{-2}{3}} = \frac{1}{\sqrt[3]{(a^2)}}, \quad a^{\frac{3}{4}} = [\sqrt{(\sqrt{a})}]^3,$$

$$H\sqrt[3]{H} = \sqrt{H^4 H} = H^{\frac{4}{3}}.$$

An *even* power of a negative quantity is *positive*, an *odd* power *negative*, thus— $\quad -2^2 = +4, \quad -2^3 = -8$

$$\sqrt{\cdot 75} = \sqrt{75} \div 10, \qquad \sqrt{1\cdot 43} = \sqrt{143} \div 10,$$

$$\sqrt{\cdot 043} \doteq \sqrt{430} \div 100.$$

$$\sqrt{0\cdot 4} = \sqrt{40} \div 10, \quad \sqrt[3]{\cdot 035} = \sqrt[3]{35} \div 10.$$

To square a number ending in $\frac{1}{2}$, multiply the whole number by the next higher and add $\frac{1}{4}$, e.g.

$$17\tfrac{1}{2}^2 = (17 \times 18) + \tfrac{1}{4} = 306\tfrac{1}{4}.$$

It is very useful to remember the following results, $1\frac{1}{2} \times 1\frac{1}{2} = 2\frac{1}{4}$, or $(1\cdot 5)^2 = 2\cdot 25$, or $15 \times 15 = 225$, or $150 \times 150 = 22{,}500$. Also $2\frac{1}{2} \times 2\frac{1}{2} = 6\frac{1}{4}$, $3\frac{1}{2} \times 3\frac{1}{2} = 12\frac{1}{4}$, $4\frac{1}{2} \times 4\frac{1}{2} = 20\frac{1}{4}$.

Note.—When a formula contains l and d to different powers the coefficient is made to suit either feet or inches, as the case may be, but it cannot apply to either feet or inches indiscriminately.

1,122. SOLVING ROOTS BY FACTORS.

Bear in mind $\sqrt{2} = 1\cdot4142$, $\sqrt{3} = 1\cdot732$, $\sqrt{5} = 2\cdot236$.

Examples :

$$\tfrac{1}{5}(\sqrt{48}) = \tfrac{1}{5}(\sqrt{4 \times 4 \times 3}) = \tfrac{4}{5}(\sqrt{3}) = \cdot8 \times 1\cdot732 = 1\cdot3856.$$

$$\sqrt{\cdot75} = \sqrt{\cdot5 \times \cdot5 \times 3} = \cdot5 \times \sqrt{3} = \cdot866.$$

$$\sqrt{\frac{3}{16}} = \frac{\sqrt{3}}{4} = \frac{1\cdot732}{4} = \cdot433.$$

$$\frac{1}{\sqrt{20}} = \frac{\sqrt{20}}{20} = \frac{\sqrt{4 \times 5}}{2 \times 10} = \frac{\sqrt{5}}{10} = \frac{2\cdot236}{10} = \cdot2236.$$

$$\sqrt{\tfrac{1}{2}} = \frac{1}{\sqrt{2}},\ \sqrt{\frac{3}{16}} = \frac{\sqrt{3}}{4},\ \sqrt{\frac{1}{12}} = \sqrt{\frac{\frac{1}{4}}{3}} = \frac{\frac{1}{2}}{\sqrt{3}} = \frac{1}{2\sqrt{3}},$$

$$\sqrt{4} = \sqrt{2} \times \sqrt{2} = 2,\quad \sqrt{8} = 2\sqrt{2},\quad \sqrt{36} = 3\sqrt{4},\quad \frac{\sqrt{20}}{6} = \frac{\sqrt{5}}{3},$$

$$\sqrt[4]{x} = \sqrt{(\sqrt{x})},\quad \sqrt{12} = \sqrt{3 \times 2 \times 2} = 2\sqrt{3},$$

$$(a + b)^2 = a^2 + 2ab + b^2,\quad (a - b)^2 = a^2 - 2ab + b^2,$$

$$(a + b)(a - b) = a^2 - b^2,\qquad (a \pm b)^2 = a^2 \pm 2ab + b^2.$$

$x^n + y^n$ is divisible by $x + y$ when n is odd.

$x^n - y^n$ „ $x + y$ when n is even.

$x^n - y^n$ „ $x - y$ when n is either odd or even.

1,123. QUADRATIC EQUATIONS.

To find x when $x^2 + ax = m$, add to both sides the square of half the coefficient of x to complete the square of left hand portion, thus $x^2 + ax + \left(\frac{a}{2}\right)^2 = m + \left(\frac{a}{2}\right)^2$, extract the square root, making $x + \frac{a}{2} = \sqrt{m + \left(\frac{a}{2}\right)^2}$, then $x = \mp\sqrt{m + \left(\frac{a}{2}\right)^2} - \frac{a}{2}$.

Bear in mind that $\left.\begin{array}{r} -2^2 = +4 \\ +2^2 = +4 \end{array}\right\} \therefore \sqrt{4} = \pm 2,$

also that $\sqrt{a^2 - 2ab + b^2}$ may be either $a - b$ or $b - a$.

Example :—

Given $\begin{cases} x^2 + y = 18 & (1) \\ y^2 + x = 8 & (2) \end{cases}$

from (2) $x = 8 - y^2$ and $\therefore x^2 = 64 - 16y^2 + y^4$ (3)

from (2) $y^2 = 8 - x$ and $\therefore y^4 = 64 - 16x + x^2$ (4)

Taking (4) from (3)—

$$x^2 - y^4 = 16x - 16y^2 + y^4 - x^2$$

$$2x^2 - 16x = 2y^4 - 16y^2$$

$$x^2 - 8x = y^4 - 8y^2$$

Adding 16 to each side (being the square of ½ coeff. of x)—

$$x^2 - 8x + 16 = y^4 - 8y^2 + 16$$

Taking square root—

$$x - 4 = y^2 - 4$$

$$\therefore x = y^2$$

Substituting for y^2 in (2)

$$\text{we have } x + x = 8$$

$$\therefore x = 4$$

$$\text{and } y = 2$$

In simplifying algebraic expressions remember that

(1) when the index is odd $a^n + x^n$ is divisible by $a + x$, and $a^n - x^n$ is divisible by $a - x$.

(2) when the index is even $a^n + x^n$ is not divisible by either $a + x$ or $a - x$, but $a^n - x^n$ is divisible by both $a + x$ and $a - x$.

De Moivre's theorem is used in finding higher roots. For explanation see W. K. Clifford's Common Sense of the Exact Sciences, p. 192.

1,124. PONCELET'S THEOREMS.

Let $\lambda = \sqrt{\mu^2 + \nu^2}$, then if the values of μ and ν are unlimited, so that μ may be infinitely great or small as compared with ν, $\lambda = 0{\cdot}8284\ \mu + 0{\cdot}8284\ \nu$ nearly, the maximum error being about 17 per cent. If ν is essentially less than μ, $\lambda = 0{\cdot}9605\ \mu + 0{\cdot}3978\ \nu$, and the maximum error is about 4 per cent.

Let $\lambda = \sqrt{\mu^2 - \nu^2}$, then if the minimum possible value of μ is $1{\cdot}1\ \nu$, $\lambda = 1{\cdot}1319\ \mu - 0{\cdot}7264\ \nu$ with a possible error of 13 per cent. If the minimum value of μ is 2ν, $\lambda = 1{\cdot}0186\ \mu - 0{\cdot}2729\ \nu$, with a possible error less than 2 per cent.—UNWIN.

1,125. EXTRACTION OF THE SQUARE ROOT.

To extract the square root is to find a number which multiplied by itself will produce the given number.

Rule.—Mark a point over the unit figure, and over every alternate figure which will divide the line into periods of two figures each. Place the root whose square most nearly approximates to the first period in the quotient. Subtract the square of it from the first period. Bring down the next period to the remainder for a fresh dividend. Double the figure in the quotient for a divisor, and find how many times it is contained in the *tens* of the dividend. Place the figure representing the number of times in the quotient, and also as the unit figure in the divisor. Multiply the divisor by the last figure in the quotient. Subtract the product; bring down another period, and proceed thus till the whole are brought down. The quotient is the root required.

Note.—If there be decimals, the first point is still to be on the unit figure of the *whole* numbers, and the alternate decimals will be marked from it.

The value of a remainder is found by continuing to add periods of noughts.

Example.—What is the square root of 105846·16? or what is the length of the side of a square if it contains 105846·16 sq. inches on its surface?

```
      105846·16(325·34  Ans.
       9
   62) 158
       124
   645)3446
       3225
   6503)22116
        19509
   65064)260700
         260256
```

1,126. TO EXTRACT THE CUBE ROOT.

Rule.—Point the given cube into periods of three figures, and so that the unit figure be the last in its period; then from the first period subtract the greatest cube it contains; put the root as a quotient, and to the remainder bring down the next period for a dividend.

Find a divisor by multiplying the square of the root by 300; see how often it is contained in the dividend; and the answer gives the next figure in the root.

Multiply the divisor by the last figure in the root. Multiply all the figures in the root by 30, except the last; and that product by the square of the last. Cube the last figure in the root; add these three last found numbers together, and subtract this sum from the dividend; to the remainder bring down the next period for a new dividend, and proceed as before.

Example.—Required the cube root of 444194947.

```
                               444194947(763
                               343
 7 ×  7 × 300 =   14700)101194
                         95976
76 × 76 × 300 = 1732800)5218947
                        5218947
```

1. Divisor	14700	2. Divisor	1732800
	6		3
	88200		5198400
$7 \times 30 \times 36 =$	7560	$76 \times 30 \times 9 =$	20520
$6 \times 6 \times 6 =$	216	$3 \times 3 \times 3 =$	27
	95976		5218947

But generally logarithms would be used to find the cube root.

1,127. LOGARITHMS.

$a^2 = m$, then 2 is the logarithm of m to the base a.

$a = n$,, 3 ,, ,, n ,,

Tables of logarithms facilitate calculation, addition and subtraction being substituted for multiplication and division, and multiplication and division for involution and evolution.*

$\text{Log}_{10}\, x$ = log of x to base 10,—i.e., common log of x.

$\text{Log}_e\, x$ = log of x to base 2·71823—i.e., Naperian or hyperbolic log of x.

$$10^{-\cdot 57708} = 10^{-1+(1-\cdot 57708)} = 10^{\bar{1}\cdot 42292} = \cdot 2648$$

1,128. INVERSE NOTATION.

Let $\sin\theta = a$, divide both sides by sin, then $\theta = \dfrac{a}{\sin}$, but $\dfrac{a}{\sin} = \dfrac{1}{\sin}\, a$, and $\dfrac{1}{\sin} = \sin^{-1}$, $\therefore\ \theta = \sin^{-1} a$.

$\text{Sin}^{-1}\, a$ is used in formulæ to express "the angle whose sine is a."

1,129. FUNCTIONS.

Whenever two quantities are related in some such way, so that one of them being given the other can be calculated or found, each is said to be a function of the other.—W. K. Clifford.

1,130. BINOMIAL THEOREM.

An expression containing two terms as $a + b$ is a *binomial*. The name Binomial Theorem is an abbreviation for "Theorem concerning any power of a binomial expression" (Newton), the formula for which is

$$(a + b)^n = \Sigma \frac{\pi n}{\pi p . \pi q} a^p b^q$$

where π = product of $1.2.3 \ldots n$, and $p + q = n$, p taking all values from n down to 0, and q all values from 0 up to n.

* See the author's "Practical Trigonometry for Engineers, Architects and Surveyors," Whittaker & Co., 2nd edition (2s. 6d. net).

It may be very simply explained as follows :—

$$(a + b)^2 = a^2 + 2\,a\,b + b^2$$
$$(a - b)^2 = a^2 - 2\,a\,b + b^2$$
$$(a + b)\,(a - b) = a^2 - b^2$$

The number of terms is always one greater than the power required.

If both terms of the binomial are positive all the quantities remain positive, but a residual binomial $(a - b)$ has its odd terms positive and its even terms negative.

The exponent of the first or *leading* quantity (a) is always the index of the required power, and the powers of this quantity decrease by one in each successive term. The index of the second term of the quantity to be involved (b), called the *following* quantity, begins with 1 in the second term, and increases by 1 in each successive term until it equals the required power.

The coefficient of the first and last terms is always 1. The coefficient of any term multiplied by the index of the first letter (leading quantity) in that term, and divided by the index of the second letter (following quantity) increased by 1, gives the coefficient of the next succeeding term. The coefficient of the second and penultimate terms is the index of the required power.

The algebraic formation of the theorem is

$$(a + b)^n = a^n + n\,a^{n-1}\,b + n.\,\frac{n-1}{2}\,a^{n-2}\,b^2$$
$$+ n.\,\frac{n-1}{2}\cdot\frac{n-2}{3}\,a^{n-3}\,b^3, \text{ etc.}$$

By the substitution of the letters A B C for the coefficients n, $n.\,\frac{n-1}{2}$, $n.\,\frac{n-1}{2}\cdot\frac{n-2}{3}$, etc., the formula is much simplified, and becomes

$$(a + b)^n = a^n + \mathrm{A}\,a^{n-1}\,b + \mathrm{B}\,a^{n-2}\,b^2 + \mathrm{C}\,a^{n-3}\,b^3, \text{ etc.}$$

Hence, for example,

$$(a + b)^6 = a^6 + 6\,a^5\,b + 15\,a^4\,b^2 + 20\,a^3\,b^3 + a^2\,b^4 + 6\,a\,b^5 + b^6.$$

1,131. METHOD OF FLUXIONS AND DIFFERENTIAL CALCULUS.

The particular process of deriving one algebraical rule from another was first investigated by Newton. He was accustomed to describe a varying quantity as a *fluent*, and its rate of change he called the *fluxion* of the quantity. On account of these names the entire method of solving these problems by means of the process of deriving one algebraical rule from another was termed the *Method of Fluxions*.

Because two differences are used in the argument which establishes the process for changing the one rule into the other this process was called, first in other countries and then also in England, the Differential Calculus. The name is an unfortunate one, because the rate of change which is therein calculated has nothing to do with differences, the only connection with differences being that they are mentioned in the argument which is used to establish the process. However this may be, the object of the differential calculus or of the method of fluxions (whichever name we choose to give it) is to find a rule for calculating the rate of change of a quantity when we have a rule for calculating the quantity itself.—W. K. Clifford.

1,132. USEFUL NUMBERS.

$\pi = 3\cdot1416$	$\sqrt{3} = 1\cdot732$	$\dfrac{\pi}{12} = \cdot2618$
$\sqrt{\pi} = 1\cdot772$	$\dfrac{1}{\sqrt{2}} = \cdot7071$	$\sqrt[3]{2} = 1\cdot26$
$\sqrt[3]{\pi} = 1\cdot465$	$\pi_2 = 9\cdot8696$	$\sqrt[3]{3} = 1\cdot44$
$\dfrac{\pi}{6} = \cdot5236$	$\dfrac{1}{\pi} = \cdot3183$	$\dfrac{\sqrt{3}}{2} = \cdot866$
$\sqrt{2} = 1\cdot414$	$\dfrac{1}{\sqrt{\pi}} = \cdot564$	

Lbs. per sq. inch × ·0643 = tons per sq. foot.
Tons per sq. foot × 15·5 = lbs. per sq. inch.

1,133. NOTES ON CIRCLES.

$\dfrac{c^2}{4\pi}$	Area of circle to circumference unity	$= 0\cdot07958$
$d^2\dfrac{\pi}{4}$	,, ,, diameter ,,	$= 0\cdot7854$
πr^2	,, ,, radius ,,	$= 3\cdot1416$
πd	Circumference of circle to diameter	$= 3\cdot1416$
$2\pi r$	,, ,, ,, radius ,,	$= 6\cdot2832$
$2\sqrt{\pi a}$	,, ,, ,, area ,,	$= 3\cdot5449$
$2r$	Diameter ,, ,, radius ,,	$= 2\cdot0000$
$2\sqrt{\dfrac{a}{\pi}}$	,, ,, ,, area ,,	$= 1\cdot2837$
$\dfrac{c}{\pi}$	,, ,, circumference,,	$= 0\cdot3183$

Ordinates to circular curve. Let v = versin, r = radius, c = chord, then

$$v = r - \sqrt{\left(r^2 - \frac{c^2}{4}\right)}$$

and any ordinate y at distance x from centre line

$$= \sqrt{r^2 - x^2} - (r - v).$$

1,134. EPITOME OF MENSURATION.

a = area.	h = height.
b = base.	n° = number of degrees.
p = perpendicular.	c = circumference.
r = radius.	s = span or chord.
d = diameter.	v = versin or rise.

Square, $a = \text{side}^2$, side $= \sqrt{a}$, diagonal $= \sqrt{\text{side}^2 \times 2}$.

Rectangle or parallelogram, $a = b\,p$.

a = product of two sides and sine of included angle.

Trapezoid (2 sides parallel), a = mean length parallel, sides $\times$ distance between them.

Any triangle, $a = \frac{1}{2}\,b\,p$.

a = product of two sides and $\frac{1}{2}$ sine of included angle.

Equilateral triangle, $a = \frac{\sqrt{3}}{4}\,\text{side}^2 = 0{\cdot}433\,\text{side}^2$.

Area of triangle from length of sides a, b, c,

$$\text{half sum of sides} = \frac{a+b+c}{2} = S,$$

$$\text{then area} = \sqrt{S\,(S-a)\,(S-b)\,(S-c)}.$$

Trapezium—

Divide into two triangles by one diagonal, then $a = \frac{1}{2}$ sum of perps. $\times$ diagonal.

Irregular four-sided figure—

Diagonals D d, angle of intersection a.

$$a = \tfrac{1}{2}\,D\,d \sin a.$$

Polygons—

$$a = \tfrac{1}{2}\ \text{sum of sides} \times \text{perp. from centre.}$$

Irregular figure, a = weight of template $\div$ weight of sq. inch similar material.

Circle—

$$a = \pi r^2 = d^2 \frac{\pi}{4} = {\cdot}7854\,d^2 = {\cdot}5\,c\,r = {\cdot}07958\,c^2 = \frac{c^2}{4\,\pi}.$$

$$= \text{approximately } d^2 \times \tfrac{11}{14}.$$

$$c = 2\,\pi\,r = d\,\pi = 3{\cdot}1416\,d = 3{\cdot}54\sqrt{a} = \text{approx. } \tfrac{22}{7}\,d.$$

$$d = \sqrt{\frac{4a}{\pi}} = \sqrt{\frac{a}{{\cdot}7854}} = {\cdot}3183\ c.$$

Side of equal square $= {\cdot}8862\,d$

Side of inscribed square $= {\cdot}7071\,d.$

A circle has the maximum area for a given perimeter.

Annulus of circle—

$$a = (D + d)(D - d)\frac{\pi}{4} = (D^2 - d^2)\frac{\pi}{4} = \pi(R + r)(R - r).$$

Segment of circle—

$$\text{Length arc} = \frac{\pi n^\circ r}{180} = \cdot 0174533\, n^\circ r.$$

$$\text{,,} \quad = \tfrac{1}{3}\left(8\sqrt{\frac{s^2}{4} + v^2} - s\right).$$

,, (approx. up to 90°) = $\frac{1}{3}$ (8 times chord of $\frac{1}{2}$ arc — chord whole arc). This is known as Huygen's approximation.

$$d = \frac{(\frac{1}{2}s)^2}{v} + v.$$

$$r = \frac{s^2}{8v} + \frac{v}{2} = \frac{(\frac{1}{2}s)^2 + v^2}{2v}.$$

$$\text{chord of half arc} = \sqrt{v(2r - v)}.$$

$$s = \sqrt{4v(d - v)} = \sqrt{8vr - 4v^2}.$$

$$s = \frac{r \sin\theta}{\frac{1}{2}(180 - \sin\theta)} \left(\theta \text{ being centre angle} = 360\,\frac{\text{length arc}}{2\pi r}\right).$$

$$v = \pm\sqrt{(r + \tfrac{1}{2}s)(r - \tfrac{1}{2}s)} + r = r - \sqrt{r^2 - (\tfrac{1}{2}s)^2}.$$

Area of segment = roughly $\frac{2}{3}\,s\,v$, or approx. $= \frac{2}{3}\,s\,v + \frac{v^3}{2s}$,

$$\text{or } \tfrac{2}{3}\,s\,v + \frac{8\,v^3}{15\,s}.$$

$$\text{Do.} \quad = \left\{\left(\frac{s^2}{16\,v} + \frac{v}{4}\right) \times \frac{\left(8\sqrt{\frac{s^2}{4} + v^2}\right) - s}{3}\right\} - \frac{s}{2}\left(\frac{s^2}{8\,v} - \frac{v}{2}\right).$$

$$\text{Do. (Escott)} \quad = \tfrac{2}{3}\,s\,v + \frac{56\,v^3 s}{15\,(7\,s^2 + 4\,v^2)}.$$

$$\text{Do. (Molesworth)} = \frac{4\,v}{3}\sqrt{(0\cdot 625\,v)^2 + (\tfrac{1}{2}s)^2}.$$

Do. $= \frac{1}{2}r^2(\theta - \sin\theta)$ where θ = circular measure of centre angle.

Do. = sector — triangle $= \frac{r}{2}(l - p)$,

where l = length of arc and p = length of line drawn from one end of arc perp. to direction of radius from other end.

Sector of circle—

$$a = \cdot 5\,r \times \text{length arc}.$$

$$a = n^\circ \times \text{area circle} \div 360.$$

Ellipse—

$$a = \frac{\pi}{4} D\,d, \qquad c \text{ approx.} = \sqrt{\frac{D^2 + d^2}{2}} \times \pi$$

$$a = \pi R\,r. \qquad c \quad ,, \quad = \pi \frac{D + d}{2}.$$

Parabola, $a = \frac{2}{3}\,b\,h$.

Cone or pyramid—

$$\text{Surface} = \frac{\text{circf. base} \times \text{slant height}}{2} + \text{base}.$$

$$\text{Contents} = \text{area base} \times \tfrac{1}{3} \text{ vertical height} = \frac{\pi}{12} D^2 H.$$

Wedge—

Contents = (length of edge + 2 length of back or base). $\times$ height of wedge $\times$ breadth of base.

Frustum of cone or pyramid—

Surface $= (C + c) \times \frac{1}{2}$ slant height + ends.

Contents $= \frac{1}{3} h\,(A + a + \sqrt{A \times a})$.

For frustum of cone only—

$$\text{Contents} = \frac{\pi}{12} h\,(D^2 + d^2 + D\,d).$$

$$= \frac{\pi h}{3}\,(R^2 + R\,r + r^2).$$

Sphere—

$$\text{Surface} = d^2 \pi = 4 \pi r^2. \quad \text{Contents} = d^3 \frac{\pi}{6} = \frac{4}{3} \pi r^3.$$

Segment of sphere—

$$r = \text{rad. of base, contents} = \frac{\pi}{6} h\,(3 r^2 + h^2) = \pi h \left(\frac{r^2}{2} + \frac{h^2}{6}\right).$$

$$R = \;,, \quad \text{sphere,} \;,, \quad = \frac{\pi}{3} h^2 (3 R - h).$$

Frustum of sphere, major (with centre) or minor (without).

$$\text{Contents} = \pi h \left(\frac{r^2 + R^2}{2} + \frac{h^2}{6}\right) = \frac{\pi}{2} h\,(\tfrac{1}{3} h^2 + R^2 + r^2).$$

Hemisphere or any slice of sphere wholly on one side of diameter.

h = height from top of slice to top of sphere.
H = ,, bottom ,, ,, ,,
R = radius of sphere.

$$\text{Contents} = \frac{\pi}{3} \{3 R (H^2 - h^2) - H^3 + h^3\}.$$

Surface of convex part of segment or zone of sphere—

$= \pi d$ (of sph.) $h = 2 \pi R h = \pi (r^2 + h^2)$.

Volume of hollow sphere $= \frac{\pi}{6}(D^3 - d^3)$.

Solid of revolution (Centrobaryc theorem).

Contents = area of generating plane × circumference described by centroid of this plane during revolution.

Surface of ditto (method of Pappus or Guldinus)—

G = girth of surface parallel to axis.

ρ = radius from axis to c.g. of section through surface taken as fine wire.

$$a = 2\,\pi\,\rho\,G.$$

Spheroid—

Contents = revolving axis^2 × fixed axis × $\frac{\pi}{6}$, called oblate when long axis revolves and prolate when short axis revolves.

Ellipsoid—

An ellipsoid has 3 diameters at right angles to each other, all unequal.

Contents $= \frac{4}{3}\,\pi\,a\,b\,c$, where a, b, c are the semi-axes.

Elliptical Dome—

$$\text{Contents} = \frac{\pi}{6}\,(3\,d - 2\,h)\,\frac{D\,h^2}{d}.$$

Paraboloid—

$$\text{Contents} = \frac{\ }{8}\,D^2\,H.$$

Prismoidal formula—

$$\text{Contents} = \frac{\text{end areas} + 4 \text{ times middle area}}{6} \times \text{length}.$$

Anchor ring, (r = rad. of section, R = mean rad. of ring)—

Contents $= 2\,\pi^2 R\,r^2$. (d = diam. section, D = mean diam. ring).

$$\text{Contents} = \frac{\pi}{4}\,d^2\,(\pi\,D) = 2\cdot 47\,d^2\,D.$$

Contents of cask (all dimensions in inches)—

Approx. $\frac{\pi}{12}\,h\,(d^2 + 2\,D^2)$.

Gallons $= (39\,D^2 \times 25\,d^2 \times 26\,D\,d)\ \cdot 00003147\,l$.

$$\text{Ullage lying, } G = \frac{\text{depth liquid}}{D} \begin{cases} \text{if} < \cdot 5 \text{ take } - \tfrac{1}{2} \text{ diff.} \\ \text{if} > \cdot 5 \quad ,, \quad + \tfrac{1}{2} \quad ,, \end{cases} \times \text{whole capacity.}$$

$$,,\ \text{ on end, } G = \frac{\text{depth liquid}}{l} \begin{cases} \text{if} < \cdot 5 \text{ take } - \tfrac{1}{10} \text{ diff.} \\ \text{if} > \cdot 5 \quad ,, \quad + \tfrac{1}{10} \quad ,, \end{cases} \times \text{whole capacity.}$$

Simpson's first rule (Prismoidal formula)—

Mean Section $= \frac{1}{6}\,(S_0 + 4\,S_1 + S_2)$.

Simpson's second rule—

Mean section $= \frac{1}{8}\,(S_0 + 3\,S_1 + 3\,S_2 + S_3)$.

The "two-thirds" rule for area of trapezoid with curved boundary opposite base, i.e., offset piece with curved boundary.

Area = base × (height on centre line to chord of curve ± ⅔ of continuation to tangent line of curve), viz., + for outward curve, —for inward curve.

For railway cuttings, etc., see Lodge's Mensuration for Senior Students, p. 164 *et seq*.

Simpson's first rule (for area of plane figures), commonly called "Simpson's Rule." To be used with an odd number of ordinates, except 7.

S = common interval.
A = sum of extreme ordinates.
B = sum of even ordinates.
C = sum of odd ordinates, omitting first and last.

$$\text{Area} = \frac{S}{3}(A + 4B + 2C).$$

$$\text{Mean ordinate} = \frac{\text{area}}{\text{length}}.$$

1,135. TABLE OF MULTIPLIERS FOR REGULAR POLYGONS.

No. of Sides.	*Name of the Polygon.*	*Radius of the circumscribing Circle, that of the inscribed being 1.*	*Side of the Polygon, radius of the circumscribing Circle being 1.*	*Radius of the circumscribing Circle, the side being 1.*	*Area of the Polygon, = side ² ×*
3	Trigon . . .	2·0000000	1·7320508	·5773503	·4330127
4	Tetragon . .	1·4142136	1·4142136	·7071068	1·0000000
5	Pentagon . .	1·2360680	1·1755706	·8506508	1·7204774
6	Hexagon . .	1·1547005	1·0000000	1·0000000	2·5980762
7	Heptagon . .	1·1099163	·8677676	1·1523825	3·6339126
8	Octagon . .	1·0823922	·7653668	1·3065630	4·8284272
9	Nonagon . .	1·0641778	·6840402	1·4619022	6·1818242
10	Decagon . .	1·0514622	·6180340	1·6180340	7·6942088
11	Undecagon .	1·0422171	·5634652	1·7747331	9·3656415
12	Duodecagon .	1·0352762	·5176380	1·9318517	11·1961524
13	Tridecagon .	1·0299279	·4786312	2·0892913	13·1857718
14	Tetradecagon .	1·0257169	·4450418	2·2469806	15·3345084
15	Pentadecagon .	1·0223406	·4158234	2·4048672	17·6423629
16	Hexadecagon .	1·0195912	·3901806	2·5629155	20·1093580
17	Heptadecagon .	1·0173219	·3674990	2·7210970	22·7355038
18	Octadecagon .	1·0154266	·3472964	2·8793853	25·5207681
19	Nonadecagon .	1·0138273	·3291892	3·0377692	28·4652110
20	Eicosagon . .	1·0124651	·3128690	3·1962266	31·5687575

The linear dimensions of different sized polygons will be in the above proportions, but the areas will be as the squares of the linear dimensions.

Example.—The side of a regular octagon being 1·5, the radius of the circumscribing circle will be $1·5 \times 1·3065630$, and the area will be $(1·5)^2 \times 4·8284272$.

1,136. SLIDE RULE.

The slide rule is an instrument for calculating mechanically by logarithmic computation. A 20-inch Gravêt rule of seasoned mahogany with celluloid scales is the best for office use, but a 10-inch rule is sufficient for ordinary work. There are four scales, A and D on the rule, B and C on the slide. The left index figure of the C and D scale may represent any value that is a multiple or submultiple of 10, as 1, 10, 100, or 0·1, 0·01, 0·001, etc., and this fixes the ratio of value of the whole scale. The same remarks apply to the A and B scales, but the second half of each will have a tenfold value, e.g., left 10, centre 100, next figure 200. The C and D scales are used for multiplication, division, and proportion, e.g.—

Multiplication	set 1 on C	under b on C
$a \times b = x$	over a on D	read x on D
Division		
$\frac{a}{b} = x$	set b on C	under 1 on C
	over a on D	read x on D
Proportion	set a on C	under c on C
$a : b :: c : x$	over b on D	read x on D

For full explanation see Pickworth's Manual of the Slide Rule (Emmott & Co., Ltd., 56 and 57, Fleet Street, 2s.).

1,137. SECTIONAL LINES IN MECHANICAL DRAWING.

The sectional shading to indicate the materials should be as follows:—

Cast iron . .	thin lines at an angle of 45°.
Wrought iron . .	alternate thick and thin do.
Brass or brickwork .	alternate thin and broken lines do.
Steel or stone . .	all broken or dotted lines do.
Lead . . .	thin lines at angle 60° in both directions.
Wood . . .	rings and rays in imitation of grain.

This sectional shading was first proposed by Prof. Unwin, and is generally adopted in class work. The two pairs with similar shading are, from the circumstances of their use, never likely to be mistaken one for the other.

1,138. COLOURS USED IN ARCHITECTURAL AND MECHANICAL DRAWING.

Materials.	*Elevation.*	*Section.*
Wrought iron . .	Prussian blue, very pale	Prussian blue, dark.
Cast iron . . .	Payne's grey „	Payne's grey „
Yellow brass . .	Gamboge „	Gamboge „
Gun metal . . .	Indian yellow „	Indian yellow „
Steel	Violet carmine, very pale	Violet carmine „
Lead	Indigo, very pale .	Indigo „
Zinc	French blue, very pale	French blue „
Leather . . .	Burnt umber „	Burnt umber „
Chain	Prussian blue, dot and stroke.	
Rope	Burnt sienna, dot and stroke.	
Copper . . .	Crimson lake and burnt sienna.	Crimson lake and burnt sienna, dark.
Fir and deal (wrot.) .	Burnt sienna, pale .	Burnt sienna, dark.
„ „ (rough) .	Raw sienna „ .	Edged „ „
Oak	Burnt umber „ .	Burnt umber „
Brickwork . . .	Roman ochre . .	Crimson lake „
Red bricks . . .	Light red . . .	Light red „
York and Bath stone .	Sepia, very pale .	Sepia „
Granite and Portland stone	Indigo „ . .	Indigo „
Concrete . . .	. . . . . .	Payne's grey and sepia.
Earth	Ink stippling . .	Sepia, light and dark.
Plaster and cement .	Indian ink, pale .	Indian ink, dark.
Slate	Payne's grey . .	Payne's grey „
Line of section . .	Vermilion, stroke and dot	
Mahogany . . .	Light red and burnt sienna.	Light red and burnt sienna, dark.
Greenheart . . .	Indigo and gamboge .	Indigo and gamboge dark.
Old brickwork . .	Indian ink, pale .	Indian ink, dark.
Blue bricks . . .	Indigo and indian ink	Indigo „
Stone dressings . .	French blue, very pale	French blue „
Windows inside . .	Ditto, washed, pale .	Hooker's green, No. 2, dark.
„ outside . .	Payne's grey, washed, dark.	Hooker's green, No. 2, dark.
Rain-water pipe . .	Prussian blue, outline.	
Soil pipe . . .	Burnt sienna „	
Water . . .	Prussian blue, washed	Prussian blue, lines.
Existing timber . .	Indian ink, pale .	Indian ink, etched.

1,139. COMPOSITION OF COLOURS FOR DRAWINGS.

In the equivalent mixtures named below the first colour is required in the largest proportion.

Usual Colour.	*Equivalent Mixtures.*
Payne's grey . .	Indigo, indian ink and crimson lake.
Burnt umber . .	Vandyke brown or sepia, and burnt sienna.
Roman ochre . .	Yellow ochre, or pale burnt sienna and sepia.
Indian red . .	Light red.
Indian yellow . .	Gamboge and pale burnt sienna.
Violet carmine . .	Prussian blue and crimson lake.
Hooker's green . .	Prussian blue and gamboge.

1,140. BASIS OF FRENCH MEASUREMENTS.

The mètre professes to be the one-ten-millionth part of the quadrant of the meridian passing through France from Dunquerque to Formentera, but is actually the length, when at the temperature of melting ice, of a platinum rod made by Borda. The exact length is doubtful, viz.:

French Academy	39·3827 inches.
Capt. Kater	39·37079 ,,
Mr. Hassler (U.S.)	39·3802 ,,
Ordnance Department, Great Britain .	39·37043196 ,,

A gramme is the weight of a cubic centimètre of pure water at 4° C.

A litre is a kilogramme of pure water at 4° C.

A cubic mètre of water contains 1,000 litres and weighs 1,000 kilos, or one (metric) ton.

In French weights and measures the submultiple prefixes are—

$$\text{deci} = \tfrac{1}{10},\ \text{centi} = \tfrac{1}{100},\ \text{milli} = \tfrac{1}{1000},$$

and the multiple prefixes are—

$$\text{deca} = 10,\ \text{hecta} = 100,\ \text{kilo} = 1{,}000,\ \text{myria} = 10{,}000.$$

In abbreviations decamètre, etc., is marked Dm., etc., and decimètre, etc., is marked dm., etc.

1,141. EQUIVALENTS OF METRIC SYSTEM.

	1 millimètre		·03937 inches.
10 mm. =	1 centimètre	=	·3937 ,,
10 cm. =	1 decimètre	=	3·937 ,,
10 dm. =	1 mètre	=	39·37 ,,

1 kilomètre = 3280·9 feet = 1093·6 yards = ·6213 mile = approx. ⅝ mile.

1 inch = 25·4 mm.
1 foot = 304·8 mm.
1 arc = 1076·4 sq. ft.
1 litre = 61·028 cub. ins
1 stère = 35·31716 cub. ft.
1 sq. inch = 6·45137 sq. cm.
1 sq. ft. = 928·997 sq. cm.
1 cub. in. = 16·3866 cub. cm.
1 cub. ft. = 28·3153 litres.
1 cub. yd. = ·764534 cub. mètre.

1 gramme = ·002205 lb. avoir.
1 kilo = 2·2046 lbs.
1 tonne, tonneau, or Millier = ·98421 ton
1000 kilos = 1 ton nearly.
1 quintal (100 kilos) = 1·97 cwt.
1 sq. cm. = ·155006 sq. in.
1 sq. mètre = 10·7643 sq. ft.
= 1550 sq. ins.
1 cub. cm. = 0·061 cub. in.
1 cub. mètre = 1·308 cub. yd.
1 cub. mètre = 61025·38 cub. ins.

Approximately, inches $\times \frac{254}{10}$ = millimètres, yards $\times \frac{35}{32}$ = mètres.

22 yards = 20 mètres, 22 lbs. = 10 kilos, 22 gallons = 100 litres.

1,142. FRENCH MEASURES.

1 mètre or 1 m. = 3·281 feet, say 3 feet $3\frac{3}{8}$ inches.
1 decimètre or 1 dm. (very seldom used) = $3\frac{15}{16}$ inches, or nearly 4 inches.
1 centimètre or 1 cm. or 1 c/m. = $\frac{3}{8}$ $\frac{1}{64}$ inch., or say $\frac{3}{8}$ inch full.
1 millimètre or 1 mm. or 1 m/m. = $\frac{3}{64}$ inch, or about $\frac{1}{25}$ inch.
Millimètres par mètre × ·012 = inches to 1 foot.
Weight in lb. × ·45 = weight in kilos.
Mètres per second × 3·281 = feet per second.
,, × 196·85 = feet per minute.
Centimètres carrés × ·155 = sq. inches.
Kilos. per centimètre carré × 14·22 = lbs. per sq. inch.
Approx. kilos. per c m. car. × ·9 = tons per sq. foot.
Tonneaux-mètres = foot-tons × 3·23.
1·575 kilogs. per sq. mm. = 1 ton per sq. inch.
Kilos per mètre × 2 = lbs. per yard.
Lbs. per sq. inch × ·17874 = kilos per c.m carré.
Echelle = scale. Fraction thus $\frac{1}{250}$ gives proportion of drawing to real size.

For useful tables of comparisons see Brook's "French Measures and English Equivalents" (Spon, 1s.).

1,143. UNITS IN METRIC SYSTEM,

Or centimètre-gramme-second (C.G.S.) system.

Unit length = centimètre = 0·0328 .. feet.
,, mass = gramme = 0·0022046 .. lbs.
,, interval = second.

Unit power = 1 gramme raised 1 centimètre in 1 second.
„ velocity = vélo = 1 cm. per second = 0·0328 vélos.
„ acceleration = célo = 1 vélo per second = 0·0328 célos.
„ force = dyne = 1 gramme × 1 célo.
„ energy = erg = 1 centimètre × 1 dyne.
The force of g in London = ·981 celos.
∴ the weight of a gramme in London = 981 dynes.
A poundal = 1·3825 .. × 10^4 dynes.
A foot-poundal = 4·214 .. × 10^5 centimètre-dynes (ergs).
A foot-lb. weight = 1·356 .. × 10^7 ergs.
A foot-lb. = 1·3825 .. × 10^4 gramme-centimètres.
An erg (or centimètre-dyne) = 7·37 .. × 10^{-8} ft.-lb.-wts.

—Lock's "Mechanics."

Foot-lbs. × 1·3565 = joules.
„ × 0·3262 = calories (therms).
„ × 54404 = ergs.
„ × 13825 = g. ergs.
„ × g = foot-poundals.
„ × 0·13825 = kilogram-metres.

Joule's equivalent = 42,000,000 ergs.

If P poundals (or dynes) acting as w lbs. (or grammes) produces acceleration f celos (or spouds) $P = wf$;

but if P is the force of a pound (or gramme) in the old-fashioned gravitation unit employed by engineers, then $P = \frac{w}{g}f$.

1,144. UNITS IN FOOT-POUND-SECOND SYSTEM.

Foot = unit of length.
Second = unit interval of time.
Vélo = unit velocity = 1 foot per second.
Célo = unit acceleration = 1 foot per second.
Pound or lb. = unit mass.
Poundal = unit force, or force which acting upon 1 lb. produces 1 célo.
1 foot-pound = g foot-poundals.

1,145. UNITS EMPLOYED IN ENGINEERING CALCULATIONS.

Dimensions in inches. Loads or forces in lbs. Stresses in lbs. per sq. inch. Fluid pressure in lbs. per sq. inch. Velocities and accelerations in feet per second. Mechanical work in foot-lbs.

Speeds of rotation in revns. per min., or in angular velocity per second.

Statical moments (as bending and twisting moments) in inch-lbs.—Unwin's "Machine Design."

1,146. WHITWORTH STANDARD BOLTS AND NUTS.

Vee threads 55°, ⅙ depth rounded off top and bottom, depth = ·64 pitch, and ·475 pitch for square threads. Thickness of nut = diameter of bolt. Weight of head and nut = 1·07 d^3 for hexagon, or 1·35 d^3 for square.

Diam. Bolt. ins.	*Threads per inch.*	*Diam. Bottom Thread.*	*Area Bottom Thread.*	*Thick-ness. Head.*	*Diam. over Flats.*	*Diam. over Angles.*	*Diam. of Tapping. Hole.*
½	12	·3932	·1215	·4375	·9191	1·0612	$\frac{13}{32}$
⅝	11	·5085	·2030	·5468	1·1010	1·2713	½ + $\frac{1}{32}$
¾	10	·6219	·3037	·6562	1·3012	1·5024	⅝ + $\frac{1}{32}$
⅞	9	·7327	·4216	·7656	1·4788	1·7075	¾
1	8	·8399	·5540	·8750	1·6701	1·9291	⅞
1⅛	7	·9420	·6969	·9843	1·8605	2·1483	$\frac{15}{16}$ + $\frac{1}{64}$
1¼	7	1·0670	·8941	1·0937	2·0483	2·3651	1$\frac{5}{64}$
1½	6	1·2865	1·2998	1·3125	2·4134	2·7867	1$\frac{5}{16}$
1¾	5	1·4938	1·7525	1·5312	2·7578	3·1844	1½ + $\frac{1}{32}$
2	4½	1·7154	2·3110	1·7500	3·1491	3·6362	1¾
2½	4	2·1798	3·7300	2·1875	3·8940	4·5000	2$\frac{3}{16}$
3	3½	2·6340	5·4510	2·6250	4·5310	5·2320	2⅝

Bright nuts approx. = 1½ d over sides, 1¾ d over angles. Number of square threads = ½ number V threads. Approximate diameter washer = 2 diameters bolt, or ¼ inch more than diameter over angles. (Diameter of bolt in ⅛ths × 3) + 1 = diameter of tapping hole in $\frac{1}{32}$nds approximately. Approximate diameter bottom of thread ½ inch bolt = ·40, and add ·11 for every ⅛ inch increase of diameter; or (D in ⅛ths × ·11) − ·04.

The net sectional area at bottom of thread is approximately (11 D in ⅛ths − 4)^2 × ·0007854.

1,147. WEIGHT OF BOLT "HEAD AND NUT" IN LBS.

Diam.	½	⅝	¾	⅞	1	1¼	1½
Hexagon .	·128	·267	·43	·73	1·10	2·14	3·78
Square .	·164	·320	·55	·88	1·31	2·56	4·42

A pair of round washers weigh about ⅙ of a head and nut.

A pair of square washers for wood weigh about 1½ times a head and nut.

Approximately the weight of a hexagon head and square nut equals a length of 5 diameters of bolt.

The weight of a wrought iron rod in lbs. per foot run = 2·65 d^2 or per inch run = ·218 d^2.

1,148. TABLE OF WEIGHTS, IN POUNDS, OF BLACK BOLTS AND NUTS.

(Hexagon head and nut and round neck.)

Length of Bolt in inches from under head to point.	*Diameter in inches.*														
	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2
$1\frac{1}{2}$	.038	.068	.108	.161	.228	.312	.414	.677	1.033	1.489	..	..	..	..	..	..
2	.046	.078	.124	.183	.257	.349	.458	.742	1.121	1.604	2.214	2.951	3.851	..	..	..
$2\frac{1}{2}$	.053	.090	.140	.205	.286	.385	.504	.806	1.209	1.718	2.359	3.131	4.067	5.161	7.904	11.473
3	.060	.101	.156	.227	.315	.422	.549	.871	1.296	1.833	2.505	3.309	4.235	5.420	8.255	11.932
$3\frac{1}{2}$	.067	.112	.173	.248	.343	.458	.592	.935	1.384	1.928	2.650	3.489	4.502	5.678	8.606	12.391
4	.074	.123	.189	.270	.372	.494	.638	1.000	1.473	2.062	2.795	3.667	4.718	5.936	8.957	12.849
$4\frac{1}{2}$	.081	.135	.205	.292	.401	.531	.683	1.064	1.561	2.177	2.941	3.846	4.936	6.195	9.309	13.308
5	.089	.145	.221	.315	.429	.567	.727	1.129	1.649	2.291	3.085	4.025	5.152	6.452	9.661	13.767
$5\frac{1}{2}$	.096	.157	.237	.337	.458	.603	.772	1.193	1.736	2.406	3.230	4.204	5.369	6.710	10.011	14.226
6	.103	.169	.254	.359	.487	.640	.817	1.258	1.824	2.521	3.376	4.383	5.587	6.968	10.363	14.685
$6\frac{1}{2}$	.110	.179	.269	.381	.515	.676	.863	1.323	1.913	2.635	3.521	4.562	5.804	7.227	10.714	15.144
7	.117	.191	.285	.403	.544	.712	.907	1.388	2.001	2.749	3.666	4.741	6.020	7.485	11.065	15.603
$7\frac{1}{2}$	.124	.202	.302	.425	.573	.749	.952	1.452	2.088	2.865	3.812	4.920	6.238	7.743	11.416	16.061
8	.129	.213	.319	.447	.601	.785	.997	1.517	2.176	2.979	3.957	5.098	6.455	8.001	11.768	16.520
$8\frac{1}{2}$	.136	.223	.334	.469	.631	.822	1.041	1.581	2.264	3.094	4.102	5.278	6.671	8.259	12.120	16.980
9	.143	.233	.346	.484	.659	.857	1.086	1.646	2.353	3.208	4.248	5.456	6.888	8.517	12.470	17.439
$9\frac{1}{2}$	.150	.244	.362	.505	.687	.876	1.131	1.710	2.441	3.323	4.393	5.636	7.106	8.775	12.822	17.898
10	.156	.254	.377	.526	.703	.896	1.177	1.774	2.528	3.437	4.531	5.814	7.322	9.033	13.162	18.357
$10\frac{1}{2}$	.163	.265	.393	.546	.730	.953	1.221	1.839	2.616	3.553	4.684	5.993	7.529	9.292	13.535	18.816
11	.170	.276	.407	.567	.758	.988	1.237	1.904	2.704	3.667	4.828	6.172	7.757	9.550	13.876	19.274
$11\frac{1}{2}$	.177	.287	.423	.588	.785	1.02	1.281	1.934	2.793	3.782	4.973	6.351	7.973	9.808	14.237	19.733
12	.183	.298	.438	.608	.813	1.06	1.324	2.000	2.887	3.896	5.119	6.531	8.191	10.066	14.579	20.192
For each additional Foot in length add	.164	.256	.369	.502	.656	.831	1.025	1.476	2.011	2.62	3.32	4.09	4.96	5.90	8.03	10.49

1,149. DIMENSIONS, WEIGHTS, AND SAFE DISTRIBUTED

Ref. No.	*Size of beam.*		*Wght. per ft. in lbs.*	*Clear span in*							
				2	4	6	8	10	12	14	16
B.S.B.											
30	24 ×	7½	100	..	..	..	138.2	110.6	92.2	79.0	69.1
29	20	7½	89	..	..	..	104.4	83.6	69.6	59.7	52.2
28	18	7	75	..	..	..	79.8	63.9	53.2	45.6	39.4
27	16	6	62	..	..	..	56.7	45.4	37.8	32.4	28.4
26	15	6	59	..	..	..	52.4	41.9	34.9	30.0	26.2
25	15	5	42	..	..	..	35.7	28.5	23.8	20.4	17.8
24	14	6	57	..	..	..	47.6	38.0	31.7	27.2	23.8
23	14	6	46	..	..	..	39.3	31.5	26.2	22.5	19.7
22	12	6	54	..	..	52.2	39.1	31.3	26.1	22.4	19.6
21	12	6	44	..	..	43.8	32.9	26.3	21.9	18.8	16.4
20	12	5	32	..	..	30.6	22.9	18.3	15.3	13.1	11.5
19	10	8	70	..	..	57.6	43.1	34.5	28.8	24.6	21.6
18	10	6	42	..	..	35.3	26.5	21.2	17.6	15.1	13.2
17	10	5	30	..	..	24.3	18.2	14.6	12.1	10.4	9.1
16	9	7	58	..	..	42.5	31.9	25.5	21.3	18.2	16.0
15	9	4	21	..	22.5	15.0	11.3	9.0	7.5	6.4	5.6
14	8	6	35	..	34.5	23.0	17.3	13.8	11.5	9.9	8.6
13	8	5	28	..	28.0	18.6	14.0	11.2	9.3	8.0	7.0
12	8	4	18	..	17.4	11.6	8.7	7.0	5.8	5.0	4.4
11	7	4	16	..	14.0	9.3	7.0	5.6	4.7	4.0	3.5
10	6	5	25	36.3	18.2	12.1	9.1	7.3	6.0	5.2	4.5
9	6	4½	20	28.8	14.4	9.6	7.2	5.8	4.8	4.1	3.6
8	6	3	12	16.8	8.4	5.6	4.2	3.4	2.8	2.4	2.1
7	5	4½	18	22.7	11.4	7.6	5.7	4.5	3.8	3.2	2.8
6	5	3	11	13.6	6.8	4.5	3.4	2.7	2.3	1.9	1.7
5	4¾	1¾	6.5	7.1	3.6	2.4	1.8	1.4	1.2	1.0	
4	4	3	9.5	9.4	4.7	3.1	2.3	1.9	1.5	1.3	
3	4	1¾	5.0	4.6	2.3	1.5	1.1	.92	.76	.65	
2	3	3	8.5	6.3	3.2	2.1	1.6	1.26	1.05		
1	3	1½	4.0	2.76	1.38	.92	.69	.55	.46		

LOADS IN TONS OF BRITISH STANDARD STEEL BEAMS.

feet—i.e., *distance between Supports.*

18	20	22	24	26	28	30	32	34	36	38	40
61.4	55.3	50.2	46.1	42.5	39.5	36.9	34.6	32.5	30.7	29.1	27.6
46.4	41.8	38.0	34.8	32.1	29.8	27.8	26.1	24.6	23.2	22.0	20.9
35.5	31.9	29.0	26.6	24.6	22.8	21.3	20.0	18.8	17.7	16.8	
25.2	22.7	20.6	18.9	17.4	16.2	15.1	14.2	13.3			
23.3	21.0	19.1	17.5	16.1	15.0	14.0	13.1	12.3			
15.9	14.3	13.0	11.9	11.0	10.2	9.5	8.9	8.4			
21.2	19.0	17.3	15.9	14.6	13.6	12.7					
17.5	15.7	14.3	13.1	12.1	11.2	10.5					
17.4	15.6	14.2	13.0	12.0	11.2						
14.6	13.1	11.9	10.9	10.1	9.4						
10.2	9.2	8.3	7.6	7.1							
19.2	17.3	15.7	14.4								
11.8	10.6	9.6	8.8								
8.1	7.3	6.6	6.1								
14.2	12.8	11.6									
5.0	4.5	4.1									
7.7	6.9										
6.2	5.6										
3.9	3.5										
3.1											
4.0											
3.2											
1.9											

NOTES.

(1) The loads in the table are calculated for a maximum stress of 7½ tons per sq. inch.

(2) The zig-zag line across the Table denotes the limit of Span taken at 20 times the depth of Beam.

(3) The Moment of Resistance of any section can be found on referring to the figures given in Table at 8-feet Span, the figures being read as foot tons = for example, required the Moment of Resistance of an 18 by 7 beam = 79·8 feet tons.

(4) The Moment of Resistance is equal to the greatest allowable bending moment, providing for a limit of stress of 7½ tons per sq. inch.

(5) Safe load for Beams supported both ends and load distributed, see figures in the Table.

" " " " " " " " at centre, divide the tabular figures by 2.

" " " fixed one end and load distributed, divide the tabular figures by 4.

" " " " " " load at other end, divide the tabular figures by 8.

—Fulham Steel Works Co., Ltd.

1,150. WHITWORTH GAS THREADS.

As adopted by the tube trade generally.

Diameter of pipe. (Inside.)	*Threads per inch.*	*Diameter at bottom of thread.*	*Diameter of tapping hole to nearest $\frac{1}{64}$.*	*Diameter of pipe. (Inside.)*	*Threads per inch.*	*Diameter at bottom of thread.*	*Diameter of tapping hole to nearest $\frac{1}{64}$.*
$\frac{1}{8}$	28	·344	$\frac{11}{32}$	$1\frac{1}{8}$	11	1·375	$1\frac{3}{8}$
$\frac{1}{4}$	19	·451	$\frac{29}{64}$	$1\frac{1}{4}$	11	1·533	$1\frac{17}{32}$
$\frac{3}{8}$	19	·589	$\frac{19}{32}$	$1\frac{3}{8}$	11	1·628	$1\frac{5}{8}$
$\frac{1}{2}$	14	·734	$\frac{47}{64}$	$1\frac{1}{2}$	11	1·705	$1\frac{45}{64}$
$\frac{5}{8}$	14	·811	$\frac{13}{16}$	$1\frac{5}{8}$	11	1·965	$1\frac{31}{32}$
$\frac{3}{4}$	14	·949	$\frac{61}{64}$	$1\frac{3}{4}$	11	2·042	$2\frac{3}{64}$
$\frac{7}{8}$	14	1·097	$1\frac{3}{32}$	$1\frac{7}{8}$	11	2·128	$2\frac{1}{8}$
1	11	1·192	$1\frac{3}{16}$	2	11	2·23	$2\frac{15}{64}$

1,151. SIZES FOR COPPER PIPES.

To facilitate uniformity in the sizes of Copper Pipes, it has been decided to establish the following as a standard list of thicknesses which will be made for each diameter :—

COPPER PIPES, TO STANDARD IRON TUBE SIZES.

Diameter		*Imperial W.G.*	*Diameter.*		*Imperial W.G.*
Internal.	*External.*		*Internal.*	*External.*	
Inches.	*Inches.*		*Inches.*	*Inches.*	
$\frac{1}{8}$	$\frac{3}{8}$	10	1	$1\frac{5}{16}$	8
$\frac{1}{4}$	$\frac{1}{2}$	10	$1\frac{1}{4}$	$1\frac{5}{8}$	6
$\frac{3}{8}$	$\frac{5}{8}$	8	$1\frac{1}{2}$	$1\frac{7}{8}$	6
$\frac{1}{2}$	$\frac{13}{16}$	8	2	$2\frac{3}{8}$	6
$\frac{3}{4}$	$1\frac{1}{16}$	8	$2\frac{1}{2}$	3	3

COPPER PIPES, STANDARD SIZES.

Diameter. Internal.	*Imperial W.G.*	*Diameter. Internal.*	*Imperial W.G.*	*Diameter Internal.*	*Imperial W.G.*
Inches.		*Inches.*		*Inches.*	
$\frac{1}{8}$	18	$1\frac{1}{8}$	13	$2\frac{1}{4}$	10
$\frac{1}{4}$	16	$1\frac{1}{4}$	12	$2\frac{1}{2}$	9
$\frac{3}{8}$	16	$1\frac{3}{8}$	12	$2\frac{3}{4}$	9
$\frac{1}{2}$	16	$1\frac{1}{2}$	11	3	9
$\frac{5}{8}$	15	$1\frac{5}{8}$	11	$3\frac{1}{4}$	8
$\frac{3}{4}$	14	$1\frac{3}{4}$	10	$3\frac{1}{2}$	8
$\frac{7}{8}$	14	$1\frac{7}{8}$	10	$3\frac{3}{4}$	8
1	13	2	10	4	7

The above in lengths of 12 to 16 feet. If cut to dead lengths, — per lb. extra.

1,152. SIZES OF WOOD SCREWS.

Lengths from $\frac{1}{4}$ inch to 6 inches by $\frac{1}{8}$ inch up to 1 inch, by $\frac{1}{4}$ inch from 1 to 3 inches, and by $\frac{1}{2}$ inch from 3 to 6 inches.

Diameters varying generally by $\frac{1}{64}$ inch. *Example :—*

Number	1	5	10	14	18	22	27	32	40
Diameter	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$
Length (average)	$\frac{1}{2}$	1	$1\frac{1}{2}$	2	3	4	$4\frac{1}{2}$	5	6

1,153. BRITISH ASSOCIATION (B.A.) GAUGE FOR APPARATUS SCREWS.

This is adopted as the Standard Screw Gauge by the Post Office Telegraphs Department and most large electrical firms, as modified by the Screw Gauge Committee in 1900.

No.	*Nominal Dimensions in Thousandths of an inch = mils.*			*Absolute Dimensions in millimètres.*	
	Diameter.	*Pitch.*	*Threads per inch.*	*Diameter.*	*Pitch.*
25	10	2.8	353	0.25	0.072
24	11	3.1	317	0.29	0.080
23	13	3.5	285	0.33	0.089
22	15	3.9	259	0.37	0.098
21	17	4.3	231	0.42	0.11
20	19	4.7	212	0.48	0.12
19	21	5.5	181	0.54	0.14
18	24	5.9	169	0.62	0.15
17	27	6.7	149	0.70	0.17
16	31	7.5	134	0.79	0·19
15	35	8.3	121	0.90	0.21
14	39	9.1	110	1.0	0.23
13	44	9.8	101	1.2	0.25
12	51	11.0	90.7	1.3	0.28
11	59.1	12.2	81.9	1.5	0.31
10	66.9	13.8	72.6	1.7	0.35
9	74.8	15.4	65.1	1.9	0.39
8	86.6	16.9	59.1	2.2	0.43
7	98.4	18.9	52.9	2.5	0.48
6	110.2	20.9	47.9	2.8	0.53
5	126.0	23.2	43.0	3.2	0.59
4	141.7	26.0	38.5	3.6	0.66
3	161.4	28.7	34.8	4.1	0.73
2	185.0	31.9	31.4	4.7	0.81
1	208.7	35.4	28.2	5.3	0.90
0	236.2	39.4	25.4	6.0	1.00

For optical instruments various threads are used—viz., 37 per inch, called the "Societies" thread for micro-objectives; 38 used by French makers, 22 for telescope work, 28 for cells. These threads are sharp-angled, not rounded.—W. J. LANCASTER.

1,154. BIRMINGHAM WIRE GAUGE (B.W.G.).

Obsolete since 1st March, 1884, having been replaced by the gauges on the two following pages, but still occasionally specified.

No.	*Parts of an inch.*	*No.*	*Parts of an inch.*	*No.*	*Parts of an inch.*
5 0	= 0·500	9	= 0·148	22	= 0·028
4 0	454	10	135	23	025
3 0	425	11	120	24	0220
2 0	380	12	109	25	0200
0	340	13	095	26	0180
1	300	14	083	27	0160
2	284	15	072	28	0140
3	260	16	065	29	0130
4	238	17	058	30	0120
5	220	18	050	31	0100
6	203	19	041	32	0090
7	180	20	035	33	0080
8	165	21	032		

In a similar manner Stubs' American Wire Gauge has given place to the American (or Brown and Sharpe's) Standard adopted by the American Brass Manufacturers, January, 1858.

The Lancashire pinion wire gauge was a very full list of sizes—e.g., Z to A = 0·413 to 0·234, 1 to 50 = 0·227 to 0·069, 51 to 80 = 0·066 to 0·013.

"Pocket wire gauges" are small plates of various shapes with round holes perforated near the edges, and the intervening space cut through and finished to the exact size of the gauge indicated by a number stamped or engraved against it. In consequence of the change of gauge, care should be taken that the implement used is in accordance with the revised tables.

"Pocket micrometer gauges or calipers" are very accurate, highly-finished instruments for measuring small thicknesses, say thousandths of an inch. This is effected by the partial revolution of a fine-pitched screw with a vernier reading against a fixed portion.

"The Whitworth Millionth measuring machine" is a more precise instrument of the same character by which small lengths or thicknesses may be gauged to the millionth of an inch.

1,155. STANDARD SHEET AND HOOP-IRON GAUGE (B.G., S.G., OR S.S.G.).

(From March 1st, 1884.)

Issued by the South Staffordshire Ironmasters' Association, February 28th, 1884.

No. of Gauge.	Thickness in			Approximate Weight per superficial foot of Sheet Iron in pounds.	No. of Gauge.	Thickness in			Approximate Weight per superficial foot of Sheet Iron in pounds.
	Ordinary Fractions of an inch.	Decimals of an inch.	Millimètres.			Ordinary Fractions of an inch.	Decimals of an inch.	Millimètres.	
3°	½	.500	12.700	20.	20	..	.0392	.996	1.568
2°	..	.4452	11.288	17.808	21	..	.0349	.886	1.396
1°	..	.3064	10.068	15.856	22	$\frac{1}{32}$	.03125	.794	1.25
1	..	.3532	8.971	14.128	23	..	.02782	.707	1.1128
2	...	.3147	7.993	12.588	24	..	.02476	.629	.9904
3	..	.2804	7.122	11.216	25	..	.02204	.560	.8816
4	¼	.250	6.350	10.	26	..	.01961	.498	.7844
5	..	.2225	5.651	8.90	27	..	.01745	.4432	.698
6	..	.1981	5.032	7.924	28	$\frac{1}{64}$	.015625	.3969	.625
7	..	.1764	4.480	7.056	29	..	.0139	.3531	.556
8	..	.1570	3.988	6.28	30	..	.0123	.3124	.492
9	..	.1398	3.551	5.592	31	..	.0110	.2794	.440
10	⅛	.1250	3.175	5.	32	..	.0098	.2489	.392
11	..	.1113	2.827	4.452	33	..	.0087	.2210	.348
12	..	.0991	2.517	3.964	34	..	.0077	.1956	.300
13	..	.0882	2.240	3.528	35	..	.0069	.1753	.276
14	..	.0785	1.994	3.14	36	..	.0061	.1549	.244
15	..	.0699	1.775	2.796	37	..	.0054	.1371	.216
16	$\frac{1}{16}$	.0625	1.587	2.50	38	..	.0048	.1219	.192
17	..	.0556	1.412	2.224	39	..	.0043	.1092	.172
18	..	.0495	1.257	1.98	40	..	.00386	.0980	.1544
19	..	.0440	1.118	1.76					

1,156. IMPERIAL STANDARD WIRE GAUGE (S.W.G.).

Table of sizes, weights, lengths, and breaking strains of iron wire under the Imperial Standard Wire Gauge issued by the Iron and Steel Wire Manufacturers' Association. (In force from March 1st, 1884.)

Size on Wire Gauge.	*Diameter.*		*Sectional Area in sq. inches.*	*Weight of*		*Length of cwt.*	*Breaking Strain.*	
	Inch.	*Millimètres.*		*100 yards.*	*Mile.*		*Annealed.*	*Bright.*
				lb.	*lb.*	*yards.*	*lb.*	*lb.*
7/0	0.500	12.7	0.1963	193.4	3,404	58	10,470	15,700
6/0	0.464	11.8	0.1691	166.5	2,930	67	9,017	13,525
5/0	0.432	11	0.1466	144.4	2,541	78	7.814	11,725
4/0	0.400	10.2	0.1257	123.8	2,179	91	6,702	10,052
3/0	0.372	9.4	0.1087	107.1	1,885	105	5,796	8,694
2/0	0.348	8.8	0.0951	93.7	1,649	120	5,072	7,608
1/0	0.324	8.2	0.0824	81.2	1,429	138	4,397	6,595
1	0.300	7.6	0.0707	69.6	1,225	161	3,770	5,655
2	0.276	7	0.0598	58.9	1,037	190	3,190	4,785
3	0.252	6.4	0.0499	49.1	864	228	2,660	3,990
4	0.232	5.9	0.0423	41.6	732	269	2,254	3,381
5	0.212	5.4	0.0365	34.8	612	322	1,883	2,824
6	0.192	4.9	0.0290	28.5	502	393	1,544	2,316
7	0.176	4.5	0.0243	24	422	467	1,298	1,946
8	0.160	4.1	0.0201	19.8	348	566	1,072	1,608
9	0.144	3.7	0.0163	16	282	700	869	1,303
10	0.128	3.3	0.0129	12.7	223	882	687	1,030
11	0.116	3	0.0106	10.4	183	1,077	564	845
12	0.104	2.6	0.0085	8.4	148	1,333	454	680
13	0.092	2.3	0.0066	6.5	114	1,723	355	532
14	0.080	2	0.0050	5	88	2,240	268	402
15	0.072	1.8	0.0041	4	70	2,800	218	326
16	0.064	1.6	0.0032	3.2	56	3,500	172	257
17	0.056	1.4	0.0025	2.4	42	4,667	131	197
18	0.048	1.2	0.0018	1.8	32	6,222	97	145
19	0.040	1	0.0013	1.2	21	9,333	67	100
20	0.036	0.9	0.0010	1	18	11,200	55	82

1 mil = $\frac{1}{1000}$ inch. 1 micron = 0·01 millimètre.

In electric wiring 3/22 S.W.G means a strand composed of 3 wires each of 22 gauge. 7/21½ S.W.G. would be a strand of 7 wires each of a size between 21 and 22 gauge.

The Whitworth Standard Gauge gives the thickness in mils—e.g., No. 36 $= \frac{36}{1000} =$ ·036 inch.

1,157. DECIMAL EQUIVALENTS TO FRACTIONS OF AN INCH.

.96875 =	$\frac{7}{8} + \frac{3}{32}$	.625 =	$\frac{5}{8}$	.28125 =	$\frac{1}{4} + \frac{1}{32}$
.9375	$\frac{7}{8} + \frac{1}{16}$	.59375	$\frac{1}{2} + \frac{3}{32}$	.25	$\frac{1}{4}$
.90625	$\frac{7}{8} + \frac{1}{32}$	.5625	$\frac{1}{2} + \frac{1}{16}$	.21875	$\frac{1}{8} + \frac{3}{32}$
.875	$\frac{7}{8}$	.53125	$\frac{1}{2} + \frac{1}{32}$	.1875	$\frac{1}{8} + \frac{1}{16}$
.84375	$\frac{3}{4} + \frac{3}{32}$	.5	$\frac{1}{2}$	.15625	$\frac{1}{8} + \frac{1}{32}$
.8125	$\frac{3}{4} + \frac{1}{16}$	.46875	$\frac{3}{8} + \frac{3}{32}$	.125	$\frac{1}{8}$
.78125	$\frac{3}{4} + \frac{1}{32}$	.4375	$\frac{3}{8} + \frac{1}{16}$	.09375	$\frac{3}{32}$
.75	$\frac{3}{4}$	.40625	$\frac{3}{8} + \frac{1}{32}$	.0625	$\frac{1}{16}$
.71875	$\frac{5}{8} + \frac{3}{32}$	.375	$\frac{3}{8}$	.03125	$\frac{1}{32}$
.6875	$\frac{5}{8} + \frac{1}{16}$	.34375	$\frac{1}{4} + \frac{3}{32}$	.015625	$\frac{1}{64}$
.65625	$\frac{5}{8} + \frac{1}{32}$	.3125	$\frac{1}{4} + \frac{1}{16}$	.0078125	$\frac{1}{128}$

8ths.

$\frac{1}{8}$ =	.125	$\frac{5}{8}$ =	.625
$\frac{1}{4}$	.250	$\frac{3}{4}$	.750
$\frac{3}{8}$	.375	$\frac{7}{8}$	.875
$\frac{1}{2}$	.500		

16ths.

$\frac{1}{16}$ =	.0625	$\frac{7}{16}$ =	.4375	$\frac{13}{16}$ =	.8125
$\frac{3}{16}$	.1875	$\frac{9}{16}$	.5625	$\frac{15}{16}$	.9375
$\frac{5}{16}$	.3125	$\frac{11}{16}$	.6875		

32nds.

$\frac{1}{32}$ =	.03125	$\frac{13}{32}$ =	.40625	$\frac{23}{32}$ =	.71875
$\frac{3}{32}$	.09375	$\frac{15}{32}$	.46875	$\frac{25}{32}$	.78125
$\frac{5}{32}$	.15625	$\frac{17}{32}$	.53125	$\frac{27}{32}$	.84375
$\frac{7}{32}$	.21875	$\frac{19}{32}$	.59375	$\frac{29}{32}$	.90625
$\frac{9}{32}$	.28125	$\frac{21}{32}$	.65625	$\frac{31}{32}$	.96875
$\frac{11}{32}$	.34375				

64ths.

$\frac{1}{64}$ =	.015625	$\frac{23}{64}$ =	.359375	$\frac{45}{64}$ =	.703125
$\frac{3}{64}$	.046875	$\frac{25}{64}$	.390625	$\frac{47}{64}$	.734375
$\frac{5}{64}$	.078125	$\frac{27}{64}$	.421875	$\frac{49}{64}$	.765625
$\frac{7}{64}$	.109375	$\frac{29}{64}$	.453125	$\frac{51}{64}$	.796875
$\frac{9}{64}$	.140625	$\frac{31}{64}$	.484375	$\frac{53}{64}$	.828125
$\frac{11}{64}$	.171875	$\frac{33}{64}$	.515625	$\frac{55}{64}$	.859375
$\frac{13}{64}$	.203125	$\frac{35}{64}$	.546875	$\frac{57}{64}$	.890625
$\frac{15}{64}$	.234375	$\frac{37}{64}$	.578125	$\frac{59}{64}$	.921875
$\frac{17}{64}$	.265625	$\frac{39}{64}$	.609375	$\frac{61}{64}$	.953125
$\frac{19}{64}$	.296875	$\frac{41}{64}$	.640625	$\frac{63}{64}$	.984375
$\frac{21}{64}$	.328125	$\frac{43}{64}$	.671875		

1,158. DECIMAL EQUIVALENTS TO FRACTIONS OF A FOOT.

	Inches :—	0	1	2	3	4	5	6	7	8	9	10	11
Decimals of unity, or an inch.	*Fractions of an inch (parts)*	*Foot.*	*Foot.*	*Foot.*	*Foot.*	*Foot.*	*Foot.*	*Foot.*	*Foot.*	*Foot.*	*Foot.*	*Foot.*	*Foot.*
			.083	.166	.250	.333	.416	.500	.583	.666	.750	.833	.916
.0625	$\frac{1}{16}$	.005208	.088	.172	.255	.338	.422	.505	.588	.671	.755	.838	.922
.1250	$\frac{1}{8}$	.010416	.094	.177	.260	.344	.427	.510	.594	.677	.760	.844	.927
.1875	$\frac{3}{16}$	.015625	.099	.182	.265	.349	.432	.515	.599	.682	.765	.849	.932
.25	$\frac{1}{4}$	.020833	.104	.187	.271	.354	.437	.520	.604	.687	.771	.854	.937
.3125	$\frac{5}{16}$	.026041	.109	.193	.276	.359	.443	.526	.609	.692	.776	.859	.943
.3750	$\frac{3}{8}$	.03125	.114	.198	.281	.364	.448	.531	.614	.697	.781	.864	.948
.4375	$\frac{7}{16}$	.036458	.120	.203	.286	.371	.453	.536	.620	.703	.786	.870	.953
.5	$\frac{1}{2}$	.04166	.125	.208	.291	.375	.458	.541	.625	.708	.791	.875	.958
.5625	$\frac{9}{16}$	.046875	.130	.213	.297	.380	.463	.547	.630	.713	.797	.880	.963
.6250	$\frac{5}{8}$	.052083	.135	.219	.302	.385	.469	.552	.635	.718	.802	.885	.969
.6875	$\frac{11}{19}$	.057291	.140	.224	.307	.390	.474	.557	.640	.723	.807	.890	.974
.75	$\frac{3}{4}$	.06250	.146	.229	.312	.396	.479	.562	.646	.729	.812	.895	.979
.8125	$\frac{13}{16}$	.067708	.151	.234	.318	.401	.484	.568	.651	.734	.818	.901	.984
.8750	$\frac{7}{8}$	.072916	.156	.239	.323	.406	.489	.573	.656	.739	.823	.906	.989
.9375	$\frac{15}{16}$	.078125	.161	.245	.328	.411	.494	.578	.661	.744	.828	.911	.995

Note.—The first column gives the decimals corresponding to the fractional parts of units in the next column; thus—$\frac{3}{16}$ of an inch is ·1875 of an inch. The remaining columns give the decimals of a foot corresponding to inches and parts; thus—$5\frac{3}{8}$ inches is ·448 of a foot, and $1\frac{1}{8}$ inches is ·094 of a foot.—EDWARD W. TURNER (LEEDS).

1159. AREAS OF CIRCLES, ADVANCING BY EIGHTHS.

Diam.	Areas.							
	0	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
0	.0	.0122	.0490	.1104	.1963	.3068	.4417	.6013
1	.7854	.9940	1.227	1.485	1.767	2.074	2.405	2.761
2	3.142	3.546	3.976	4.430	4.909	5.412	5.939	6.492
3	7.069	7.670	8.296	8.946	9.621	10.32	11.04	11.79
4	12.57	13.36	14.19	15.03	15.90	16.80	17.72	18.66
5	19.63	20.63	21.65	22.69	23.76	24.85	25.97	27.11
6	28.27	29.46	30.68	31.92	33.18	34.47	35.78	37.12
7	38.48	39.87	41.28	42.72	44.18	45.66	47.17	48.71
8	50.26	51.85	53.46	55.09	56.74	58.43	60.13	61.86
9	63.62	65.40	67.20	69.03	70.88	72.76	74.66	76.59
10	78.54	80.52	82.52	84.54	86.59	88.66	90.76	92.89
11	95.03	97.21	99.40	101.6	103.9	106.1	108.4	110.8
12	113.1	115.5	117.9	120.3	122.7	125.2	127.7	130.2
13	132.7	135.3	137.9	140.5	143.1	145.8	148.5	151.2
14	153.9	156.7	159.5	162.3	165.1	168.0	170.9	173.8
15	176.7	179.7	182.7	185.7	188.7	191.7	194.8	197.9
16	201.1	204.2	207.4	210.6	213.8	217.1	220.4	223.7
17	227.0	230.3	233.7	237.1	240.5	244.0	247.4	250.9
18	254.5	258.0	261.6	265.2	268.8	272.4	276.1	279.8
19	283.5	287.3	291.0	294.8	298.6	302.5	306.4	310.2
20	314.2	318.1	322.1	326.1	330.1	334.1	338.2	342.3
21	346.4	350.5	354.7	358.8	363.1	367.3	371.5	375.8
22	380.1	384.5	388.8	393.2	397.6	402.0	406.5	411.0
23	415.5	420.0	424.6	429.1	433.7	438.4	443.0	447.7
24	452.4	457.1	461.9	466.6	471.4	476.3	481.1	486.0
25	490.9	495.8	500.7	505.7	510.7	515.7	520.8	525.8
26	530.9	536.0	541.2	546.4	551.5	556.8	562.0	567.3
27	572.6	577.9	583.2	588.6	594.0	599.4	604.8	610.3
28	615.8	621.3	626.8	632.4	637.9	643.6	649.2	654.8
29	660.5	666.2	672.0	677.7	683.5	689.3	695.1	701.0
30	706.9	712.8	718.7	724.6	730.6	736.6	742.6	748.7
31	754.8	760.9	767.0	773.1	779.3	785.5	791.7	798.0
32	804.2	810.5	816.9	823.2	829.6	836.0	842.4	848.8
33	855.3	861.8	868.3	874.8	881.4	888.0	894.6	901.3
34	907.9	914.6	921.3	928.1	934.8	941.6	948.4	955.3
35	962.1	969.0	975.9	982.8	989.8	996.8	1003.8	1010.8
36	1017.9	1025.0	1032.1	1039.2	1046.4	1053.5	1060.7	1068.0
37	1075.2	1082.5	1089.8	1097.1	1104.5	1111.8	1119.2	1126.7
38	1134.1	1141.6	1149.1	1156.6	1164.2	1171.7	1179.3	1186.9
39	1194.6	1202.3	1210.0	1217.7	1225.4	1233.2	1241.0	1248.8
40	1256.6	1264.5	1272.4	1280.3	1288.3	1296.2	1304.2	1312.2

1160. A TABLE OF RECIPROCALS FOR OBTAINING DECIMAL EQUIVALENTS.

No.	*Recip.*	*No.*	*Recip.*	*No.*	*Recip.*	*No.*	*Recip.*	*No.*	*Recip.*
1	1.000000	51	.019608	101	.009901	151	.006623	201	.004975
2	.500000	52	.019231	102	.009804	152	.006579	202	.004950
3	.333333	53	.018868	103	.009709	153	.006536	203	.004926
4	.250000	54	.018519	104	.009615	154	.006494	204	.004902
5	.200000	55	.018182	105	.009524	155	.006452	205	.004878
6	.166667	56	.017857	106	.009434	156	.006410	206	.004854
7	.142857	57	.017544	107	.009346	157	.006369	207	.004831
8	.125000	58	.017241	108	.009259	158	.006329	208	.004808
9	.111111	59	.016949	109	.009174	159	.006289	209	.004785
10	.100000	60	.016667	110	.009091	160	.006250	210	.004762
11	.090910	61	.016393	111	.009009	161	.006211	211	.004739
12	.083333	62	.016129	112	.008929	162	.006173	212	.004717
13	.076923	63	.015873	113	.008850	163	.006135	213	.004695
14	.071429	64	.015625	114	.008772	164	.006098	214	.004673
15	.066667	65	.015385	115	.008696	165	.006061	215	.004651
16	.062500	66	.015152	116	.008621	166	.006024	216	.004630
17	.058824	67	.014925	117	.008547	167	.005988	217	.004608
18	.055556	68	.014706	118	.008475	168	.005952	218	.004587
19	.052632	69	.014493	119	.008403	169	.005917	219	.004566
20	.050000	70	.014286	120	.008333	170	.005882	220	.004545
21	.047619	71	.014085	121	.008264	171	.005848	221	.004525
22	.045455	72	.013889	122	.008197	172	.005814	222	.004505
23	.043478	73	.013699	123	.008130	173	.005780	223	.004484
24	.041667	74	.013514	124	.008065	174	.005747	224	.004464
25	.040000	75	.013333	125	.008000	175	.005714	225	.004444
26	.038462	76	.013158	126	.007937	176	.005682	226	.004425
27	.037037	77	.012987	127	.007874	177	.005650	227	.004405
28	.035714	78	.012821	128	.007813	178	.005618	228	.004386
29	.034483	79	.012658	129	.007752	179	.005587	229	.004367
30	.033333	80	.012500	130	.007692	180	.005556	230	.004348
31	.032258	81	.012346	131	.007634	181	.005525	231	.004329
32	.031250	82	.012195	132	.007576	182	.005495	232	.004310
33	.030303	83	.012048	133	.007519	183	.005464	233	.004292
34	.029412	84	.011905	134	.007463	184	.005435	234	.004274
35	.028571	85	.011765	135	.007407	185	.005405	235	.004255
36	.027778	86	.011628	136	.007353	186	.005376	236	.004237
37	.027027	87	.011494	137	.007299	187	.005348	237	.004219
38	.026316	88	.011364	138	.007246	188	.005319	238	.004202
39	.025641	89	.011236	139	.007194	189	.005291	239	.004184
40	.025000	90	.011111	140	.007143	190	.005263	240	.004167
41	.024390	91	.010989	141	.007092	191	.005236	241	.004149
42	.023810	92	.010870	142	.007042	192	.005208	242	.004132
43	.023256	93	.010753	143	.006993	193	.005181	243	.004115
44	.022727	94	.010638	144	.006944	194	.005155	244	.004098
45	.022222	95	.010526	145	.006897	195	.005128	245	.004082
46	.021739	96	.010417	146	.006849	196	.005102	246	.004065
47	.021277	97	.010309	147	.006803	197	.005076	247	.004049
48	.020833	98	.010204	148	.006757	198	.005051	248	.004032
49	.020408	99	.010101	149	.006711	199	.005025	249	.004016
50	.020000	100	.010000	150	.006667	200	.005000	250	.004000

The numbers in the table are the denominators of the fraction; hence, multiply the reciprocal of the denominator by the numerator of the fraction, and the product is the decimal equivalent.

Thus, suppose the decimal equivalent of 7-16ths be required:—

Reciprocal of 16 = ·0625 x 7 = ·4375 its decimal equivalent.

1161. DECIMAL APPROXIMATIONS FOR RAPID CALCULATIONS.

Feet . . .	×	·00019	=	miles.
„ . . .	×	1·5	=	links.
Yards . .	×	·0006	=	miles.
Links. . .	×	·22	=	yards.
„ . . .	×	·66	=	feet.
Chains . .	×	·0125	=	miles.
Sq· inches . .	×	·007	=	sq. feet.
Sq. feet . .	×	·111	=	sq. yards.
Sq. yards . .	×	·00021	=	acres.
Acres . .	×	4840	=	sq. yards.
Circular inches .	×	·0055	=	sq. feet.
„ „ .	×	·7854	=	sq. inches.
Cylindrical inches	×	·0005	=	cub. feet.
„ „	×	·0028	=	gallons.
„ feet	×	·0291	=	cub. yards.
„ „	×	4·895	=	gallons.
Cub. inches .	×	·00058	=	cub. feet.
„ feet . .	×	·04	=	cub. yards.
„ „ .	×	6·232	=	gallons.
„ „ .	×	·779	=	bushels.
„ inches .	×	·00045	=	„
„ „ .	×	·263	=	lbs. cast iron.
„ „ .	×	·282	=	lbs. wrought iron.
„ „ .	×	284	=	lbs. steel.
„ „ .	×	·304	=	lbs. brass.
Bushels . .	×	1·284	=	cub. feet.
Gallons . .	×	·1605	=	„ „
Lbs. . . .	×	·009	=	cwts.
„ . . .	×	·00045	=	tons.
Tons . . .	×	1000	=	kilos.
Miles per hour .	×	1·47	=	ft. per second.
„ „ .	×	·8684	=	knots.
Inches rainfall .	×	3630	=	cub. ft. per acre.
„ „ .	×	101·283	=	tons per acre.

3 pennies = 1 ounce, 1 halfpenny = 1 inch.

See also pages 81, 397, 454, 523.

1,162. SQUARES, CUBES, SQUARE ROOTS, AND CUBE ROOTS OF INCHES AND FRACTIONS.

No.	*Square.*	*Cube.*	*Square Root.*	*Cube Root.*	*No.*	*Square.*	*Cube.*	*Square Root.*	*Cube Root.*
⅛	.015	.0019	.353	.5	4¼	18.062	76.765	2.061	1.61
¼	.062	.0156	.500	.629	4½	20.250	91.125	2.121	1.65
⅜	.140	.0527	.612	.721	4¾	22.562	107.171	2.179	1.68
½	.250	.1250	.707	.793	5	25	125	2.236	1.71
⅝	.390	.244	.790	.855	5¼	27.562	144.703	2.291	1.73
¾	.562	.421	.866	.908	5½	30.250	166.375	2.345	1.76
⅞	.765	.669	.935	.956	5¾	33.062	190.109	2.397	1.79
1	1	1	1.000	1	6	36	216	2.449	1.81
1⅛	1.265	1.423	1.060	1.04	6¼	39.062	244.140	2.500	1.84
1¼	1.562	1.953	1.118	1.07	6½	42.250	274.625	2.549	1.86
1⅜	1.890	2.599	1.172	1.11	6¾	45.562	307.546	2.598	1.88
1½	2.250	3.375	1.224	1.14	7	49	343	2.645	1.91
1⅝	2.610	4.291	1.274	1.17	7¼	52.562	381.078	2.692	1.93
1¾	3.062	5.359	1.322	1.20	7½	56.250	421.875	2.738	1.95
1⅞	3.515	6.591	1.369	1.23	7¾	60.062	465.484	2.783	1.97
2	4	8	1.414	1.25	8	64	512	2.828	2
2⅛	4.515	9.595	1.457	1.28	8¼	68.062	561.515	2.872	2.02
2¼	5.062	11.390	1.500	1.30	8½	72.250	614.125	2.915	2.04
2⅜	5.640	13.396	1.541	1.33	8¾	76.562	669.921	2.958	2.06
2½	6.250	15.625	1.581	1.35	9	81	729	3	2.08
2⅝	6.890	18.087	1.620	1.37	9¼	85.562	791.453	3.041	2.09
2¾	7.562	20.796	1.658	1.40	9½	90.25	857.375	3.082	2.11
2⅞	8.265	23.763	1.695	1.42	9¾	95.062	926.859	3.122	2.13
3	9	27	1.732	1.44	10	100	1000	3.162	2.15
3⅛	9.765	30.517	1.767	1.46	10¼	105.062	1076.89	3.201	2.17
3¼	10.562	34.328	1.802	1.48	10½	110.250	1157.625	3.240	2.18
3⅜	11.390	38.443	1.837	1.50	10¾	115.562	1242.296	3.278	2.20
3½	12.250	42.875	1.870	1.51	11	121	1331	3.316	2.22
3⅝	13.140	47.634	1.903	1.53	11¼	126.562	1423.828	3.354	2.24
3¾	14.062	52.734	1.936	1.55	11½	132.250	1520.875	3.391	2.25
3⅞	15.015	58.185	1.968	1.57	11¾	138.062	1622.234	3.427	2.27
4	16	64	2	1.58	12	144	1728	3.464	2.28

1,163. SQUARE ROOTS AND CUBE ROOTS.

No.	Square Roots.	Cube Roots.	No.	Square Roots.	Cube Roots.	No.	Square Roots.	Cube Roots.
1	1.0000	1.0000	36	6.0000	3.3019	71	8.4261	4.1408
2	1.4142	1.2599	37	6.0827	3.3322	72	8.4852	4.1601
3	1.7320	1.4422	38	6.1644	3.3619	73	8.5440	4.1793
4	2.0000	1.5874	39	6.2449	3.3912	74	8.6023	4.1983
5	2.2360	1.7099	40	6.3245	3.4199	75	8.6602	4.2171
6	2.4494	1.8171	41	6.4031	3.4482	76	8.7177	4.2358
7	2.6457	1.9129	42	6.4807	3.4760	77	8.7749	4.2543
8	2.8284	2.0000	43	6.5574	3.5033	78	8.8317	4.2726
9	3.0000	2.0800	44	6.6332	3.5303	79	8.8881	4.2908
10	3.1622	2.1544	45	6.7082	3.5568	80	8.9442	4.3088
11	3.3166	2.2239	46	6.7823	3.5830	81	9.0000	4.3267
12	3.4641	2.2894	47	6.8556	3.6088	82	9.0553	4.3444
13	3.6055	2.3513	48	6.9282	3.6342	83	9.1104	4.3620
14	3.7416	2.4101	49	7.0000	3.6593	84	9.1651	4.3795
15	3.8729	2.4662	50	7.0710	3.6840	85	9.2195	4.3968
16	4.0000	2.5198	51	7.1414	3.7084	86	9.2736	4.4140
17	4.1231	2.5712	52	7.2111	3.7325	87	9.3273	4.4310
18	4.2426	2.6207	53	7.2801	3.7562	88	9.3808	4.4479
19	4.3588	2.6684	54	7.3484	3.7797	89	9.4339	4.4647
20	4.4721	2.7144	55	7.4161	3.8029	90	9.4868	4.4814
21	4.5825	2.7589	56	7.4833	3.8258	91	9.5393	4.4979
22	4.6904	2.8020	57	7.5498	3.8485	92	9.5916	4.5143
23	4.7958	2.8438	58	7.6157	3.8708	93	9.6436	4.5306
24	4.8989	2.8844	59	7.6811	3.8929	94	9.6953	4.5468
25	5.0000	2.9240	60	7.7459	3.9148	95	9.7467	4.5629
26	5.0990	2.9624	61	7.8102	3.9364	96	9.7979	4.5788
27	5.1961	3.0000	62	7.8740	3.9578	97	9.8488	4.5947
28	5.2915	3.0365	63	7.9372	3.9790	98	9.8994	4.6104
29	5.3851	3.0723	64	8.0000	4.0000	99	9.9498	4.6260
30	5.4772	3.1072	65	8.0622	4.0207	100	10.0000	4.6415
31	5.5677	3.1413	66	8.1240	4.0412	101	10.0498	4.6570
32	5.6568	3.1748	67	8.1853	4.0615	102	10.0995	4.6723
33	5.7445	3.2075	68	8.2462	4.0816	103	10.1488	4.6875
34	5.8309	3.2396	69	8.3066	4.1015	104	10.1980	4.7026
35	5.9160	3.2710	70	8.3666	4.1212	105	10.2469	4.7176

1,163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	*Square Roots.*	*Cube Roots.*	No.	*Square Roots.*	*Cube Roots.*	No.	*Square Roots.*	*Cube Roots.*
106	10.2956	4.7326	141	11.8743	5.2048	176	13.2664	5.6040
107	10.3440	4.7474	142	11.9163	5.2171	177	13.3041	5.6146
108	10.3923	4.7622	143	11.9582	5.2293	178	13.3416	5.6252
109	10.4403	4.7768	144	12.0000	5.2414	179	13.3790	5.6357
110	10.4880	4.7914	145	12.0415	5.2535	180	13.4164	5.6462
111	10.5356	4.8058	146	12.0830	5.2656	181	13.4536	5.6566
112	10.5830	4.8202	147	12.1243	5.2776	182	13.4907	5.6670
113	10.6301	4.8345	148	12.1655	5.2895	183	13.5277	5.6774
114	10.6770	4.8488	149	12.2065	5.3014	184	13.5646	5.6877
115	10.7238	4.8629	150	12.2474	5.3132	185	13.6014	5.6980
116	10.7703	4.8769	151	12.2882	5.3250	186	13.6381	5.7082
117	10.8166	4.8909	152	12.3288	5.3368	187	13.6747	5.7184
118	10.8627	4.9048	153	12.3693	5.3484	188	13.7113	5.7286
119	10.9087	4.9186	154	12.4096	5.3601	189	13.7477	5.7387
120	10.9544	4.9324	155	12.4498	5.3716	190	13.7840	5.7488
121	11.0000	4.9460	156	12.4899	5.3832	191	13.8202	5.7589
122	11.0453	4.9596	157	12.5299	5.3946	192	13.8564	5.7689
123	11.0905	4.9731	158	12.5698	5.4061	193	13.8924	5.7789
124	11.1355	4.9866	159	12.6095	5.4175	194	13.9283	5.7889
125	11.1803	5.0000	160	12.6491	5.4288	195	13.9642	5.7988
126	11.2249	5.0132	161	12.6885	5.4401	196	14.0000	5.8087
127	11.2694	5.0265	162	12.7279	5.4513	197	14.0356	5.8186
128	11.3137	5.0396	163	12.7671	5.4625	198	14.0712	5.8284
129	11.3578	5.0527	164	12.8062	5.4737	199	14.1067	5.8382
130	11.4017	5.0657	165	12.8452	5.4848	200	14.1421	5.8480
131	11.4455	5.0787	166	12.8840	5.4958	201	14.1774	5.8577
132	11.4891	5.0916	167	12.9228	5.5068	202	14.2126	5.8674
133	11.5325	5.1044	168	12.9614	5.5178	203	14.2478	5.8771
134	11.5758	5.1172	169	13.0000	5.5287	204	14.2828	5.8867
135	11.6189	5.1299	170	13.0384	5.5396	205	14.3178	5.8963
136	11.6619	5.1425	171	13.0766	5.5504	206	14.3527	5.9059
137	11.7046	5.1551	172	13.1148	5.5612	207	14.3874	5.9154
138	11.7473	5.1676	173	13.1529	5.5720	208	14.4222	5.9249
139	11.7898	5.1801	174	13.1909	5.5827	209	14.4568	5.9344
140	11.8321	5.1924	175	13.2287	5.5934	210	14.4913	5.9439

1,163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	Square Roots.	Cube Roots.	No.	Square Roots.	Cube Roots.	No.	Square Roots.	Cube Roots.
211	14.5258	5.9533	246	15.6843	6.2658	281	16.7630	6.5499
212	14.5602	5.9627	247	15.7162	6.2743	282	16.7928	6.5576
213	14.5945	5.9720	248	15.7480	6.2827	283	16.8226	6.5654
214	14.6287	5.9814	249	15.7797	6.2911	284	16.8522	6.5731
215	14.6628	5.9907	250	15.8113	6.2996	285	16.8819	6.5808
216	14.6969	6.0000	251	15.8429	6.3079	286	16.9115	6.5885
217	14.7309	6.0092	252	15.8745	6.3163	287	16.9410	6.5962
218	14.7648	6.0184	253	15.9059	6.3247	288	16.9705	6.6038
219	14.7986	6.0276	254	15.9373	6.3330	289	17.0000	6.6114
220	14.8323	6.0368	255	15.9687	6.3413	290	17.0293	6.6191
221	14.8660	6.0459	256	16.0000	6.3496	291	17.0587	6.6267
222	14.8996	6.0550	257	16.0312	6.3578	292	17.0880	6.6342
223	14.9331	6.0641	258	16.0623	6.3660	293	17.1172	6.6418
224	14.9666	6.0731	259	16.0934	6.3743	294	17.1464	6.6493
225	15.0000	6.0822	260	16.1245	6.3825	295	17.1755	6.6569
226	15.0332	6.0911	261	16.1554	6.3906	296	17.2046	6.6644
227	15.0665	6.1001	262	16.1864	6.3988	297	17.2336	6.6719
228	15.0996	6.1091	263	16.2172	6.4069	298	17.2626	6.6794
229	15.1327	6.1180	264	16.2480	6.4150	299	17.2916	6.6868
230	15.1657	6.1269	265	16.2788	6.4231	300	17.3205	6.6943
231	15.1986	6.1357	266	16.3095	6.4312	301	17.3493	6.7017
232	15.2315	6.1446	267	16.3401	6.4392	302	17.3781	6.7091
233	15.2643	6.1534	268	16.3707	6.4473	303	17.4068	6.7165
234	15.2970	6.1622	269	16.4012	6.4553	304	17.4355	6.7239
235	15.3297	6.1710	270	16.4316	6.4633	305	17.4642	6.7313
236	15.3622	6.1797	271	16.4620	6.4712	306	17.4928	6.7386
237	15.3948	6.1884	272	16.4924	6.4792	307	17.5214	6.7459
238	15.4272	6.1971	273	16.5227	6.4871	308	17.5499	6.7533
239	15.4596	6.2058	274	16.5529	6.4950	309	17.5783	6.7606
240	15.4919	6.2144	275	16.5831	6.5029	310	17.6068	6.7678
241	15.5241	6.2230	276	16.6132	6.5108	311	17.6351	6.7751
242	15.5563	6.2316	277	16.6433	6.5186	312	17.6635	6.7824
243	15.5884	6.2402	278	16.6733	6.5265	313	17.6918	6.7896
244	15.6204	6.2487	279	16.7032	6.5343	314	17.7200	6.7968
245	15.6524	6.2573	280	16.7332	6.5421	315	17.7482	6.8040

1163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*
316	17.7763	6.8112	351	18.7349	7.0540	386	19.6468	7.2810
317	17.8044	6.8184	352	18.7616	7.0606	387	19.6723	7.2873
318	17.8325	6.8256	353	18.7882	7.0673	388	19.6977	7.2936
319	17.8605	6.8327	354	18.8148	7.0740	389	19.7230	7.2998
320	17.8885	6.8399	355	18.8414	7.0806	390	19.7484	7.3061
321	17.9164	6.8470	356	18.8679	7.0873	391	19.7737	7.3123
322	17.9443	6.8541	357	18.8944	7.0939	392	19.7989	7.3186
323	17.9722	6.8612	358	18.9208	7.1005	393	19.8242	7.3248
324	18.0000	6.8682	359	18.9472	7.1071	394	19.8494	7.3310
325	18.0277	6.8753	360	18.9736	7.1137	395	19.8746	7.3372
326	18.0554	6.8823	361	19.0000	7.1203	396	19.8997	7.3434
327	18.0831	6.8894	362	19.0262	7.1269	397	19.9248	7.3495
328	18.1107	6.8964	363	19.0525	7.1334	398	19.9499	7.3557
329	18.1383	6.9034	364	19.0787	7.1400	399	19.9749	7.3619
330	18.1659	6.9104	365	19.1049	7.1465	400	20.0000	7.3680
331	18.1934	6.9173	366	19.1311	7.1530	401	20.0249	7.3741
332	18.2208	6.9243	367	19.1572	7.1595	402	20.0499	7.3803
333	18.2482	6.9313	368	19.1833	7.1660	403	20.0748	7.3864
334	18.2756	6.9383	369	19.2093	7.1725	404	20.0997	7.3925
335	18.3030	6.9451	370	19.2353	7.1790	405	20.1246	7.3986
336	18.3303	6.9520	371	19.2613	7.1855	406	20.1494	7.4047
337	18.3575	6.9589	372	19.2873	7.1919	407	20.1742	7.4107
338	18.3847	6.9658	373	19.3132	7.1984	408	20.1990	7.4168
339	18.4119	6.9726	374	19.3390	7.2048	409	20.2237	7.4229
340	18.4390	6.9795	375	19.3649	7.2112	410	20.2484	7.4289
341	18.4661	6.9863	376	19.3907	7.2176	411	20.2731	7.4349
342	18.4932	6.9931	377	19.4164	7.2240	412	20.2977	7.4410
343	18.5202	7.0000	378	19.4422	7.2304	413	20.3224	7.4470
344	18.5472	7.0067	379	19.4679	7.2367	414	20.3469	7.4530
345	18.5741	7.0135	380	19.4935	7.2431	415	20.3715	7.4590
346	18.6010	7.0203	381	19.5192	7.2495	416	20.3960	7.4650
347	18.6279	7.0271	382	19.5448	7.2558	417	20.4205	7.4709
348	18.6547	7.0338	383	19.5703	7.2621	418	20.4450	7.4769
349	18.6815	7.0405	384	19.5959	7.2684	419	20.4694	7.4829
350	18.7082	7.0472	385	19.6214	7.2747	420	20.4939	7.4888

1163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*
421	20.5182	7.4948	456	21.3541	7.6970	491	22.1585	7.8890
422	20.5426	7.5007	457	21.3775	7.7026	492	22.1810	7.8944
423	20.5669	7.5066	458	21.4009	7.7082	493	22.2036	7.8997
424	20.5912	7.5125	459	21.4242	7.7138	494	22.2261	7.9051
425	20.6155	7.5184	460	21.4476	7.7194	495	22.2485	7.9104
426	20.6397	7.5243	461	21.4709	7.7250	496	22.2710	7.9157
427	20.6639	7.5302	462	21.4941	7.7306	497	22.2934	7.9210
428	20.6881	7.5361	463	21.5174	7.7361	498	22.3159	7.9264
429	20.7123	7.5419	464	21.5406	7.7417	499	22.3383	7.9317
430	20.7364	7.5478	465	21.5638	7.7473	500	22.3606	7.9370
431	20.7605	7.5536	466	21.5870	7.7528	501	22.3830	7.9422
432	20.7846	7.5595	467	21.6101	7.7584	502	22.4053	7.9475
433	20.8086	7.5653	468	21.6333	7.7639	503	22.4276	7.9528
434	20.8326	7.5711	469	21.6564	7.7694	504	22.4499	7.9581
435	20.8566	7.5769	470	21.6794	7.7749	505	22.4722	7.9633
436	20.8806	7.5827	471	21.7025	7.7804	506	22.4944	7.9686
437	20.9045	7.5885	472	21.7255	7.7859	507	22.5166	7.9738
438	20.9284	7.5943	473	21.7485	7.7914	508	22.5388	7.9791
439	20.9523	7.6001	474	21.7715	7.7969	509	22.5610	7.9843
440	20.9761	7.6059	475	21.7944	7.8024	510	22.5831	7.9895
441	21.0000	7.6116	476	21.8174	7.8079	511	22.6053	7.9947
442	21.0237	7.6174	477	21.8403	7.8133	512	22.6274	8.0000
443	21.0457	7.6231	478	21.8632	7.8188	513	22.6495	8.0052
444	21.0713	7.6288	479	21.8860	7.8242	514	22.6715	8.0104
445	21.0950	7.6346	480	21.9089	7.8297	515	22.6936	8.0155
446	21.1187	7.6403	481	21.9317	7.8351	516	22.7156	8.0207
447	21.1423	7.6460	482	21.9544	7.8405	517	22.7376	8.0259
448	21.1660	7.6517	483	21.9772	7.8460	518	22.7596	8.0311
449	21.1896	7.6574	484	22.0000	7.8514	519	22.7815	8.0362
450	21.2132	7.6630	485	22.0227	7.8568	520	22.8035	8.0414
451	21.2367	7.6687	486	22.0454	7.8622	521	22.8254	8.0466
452	21.2602	7.6744	487	22.0680	7.8676	522	22.8473	8.0517
453	21.2837	7.6800	488	22.0907	7.8729	523	22.8691	8.0568
454	21.3072	7.6857	489	22.1133	7.8783	524	22.8910	8.0620
455	21.3307	7.6913	490	22.1359	7.8837	525	22.9128	8.0671

1163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*
526	22.9346	8.0722	561	23.6854	8.2474	596	24.4131	8.4155
527	22.9564	8.0773	562	23.7065	8.2523	597	24.4335	8.4202
528	22.9782	8.0824	563	23.7276	8.2572	598	24.4540	8.4249
529	23.0000	8.0875	564	23.7486	8.2621	599	24.4744	8.4296
530	23.0217	8.0926	565	23.7697	8.2670	600	24.4948	8.4343
531	23.0434	8.0977	566	23.7907	8.2719	601	24.5153	8.4390
532	23.0651	8.1028	567	23.8117	8.2767	602	24.5356	8.4436
533	23.0867	8.1079	568	23.8327	8.2816	603	24.5560	8.4483
534	23.1084	8.1129	569	23.8537	8.2864	604	24.5764	8.4530
535	23.1300	8.1180	570	23.8746	8.2913	605	24.5967	8.4576
536	23.1516	8.1230	571	23.8956	8.2961	606	24.6170	8.4623
537	23.1732	8.1281	572.	23.9165	8.3010	607	24.6373	8.4670
538	23.1948	8.1331	573	23.9374	8.3058	608	24.6576	8.4716
539	23.2163	8.1382	574	23.9582	8.3106	609	24.6779	8.4762
540	23.2379	8.1432	575	23.9791	8.3155	610	24.6981	8.4809
541	23.2594	8.1482	576	24.0000	8.3203	611	24.7184	8.4855
542	23.2808	8.1532	577	24.0208	8.3251	612	24.7386	8.4901
543	23.3023	8.1583	578	24.0416	8.3299	613	24.7588	8.4948
544	23.3238	8.1633	579	24.0624	8.3347	614	24.7790	8.4994
545	23.3452	8.1683	580	24.0831	8.3395	615	24.7991	8.5040
546	23.3666	8.1733	581	24.1039	8.3443	616	24.8193	8.5086
547	23.3880	8.1782	582	24.1246	8.3491	617	24.8394	8.5132
548	23.4093	8.1832	583	24.1453	8.3539	618	24.8596	8.5178
549	23.4307	8.1882	584	24.1660	8.3586	619	24.8797	8.5224
550	23.4520	8.1932	585	24.1867	8.3634	620	24.8997	8.5270
551	23.4733	8.1981	586	24.2074	8.3682	621	24.9198	8.5316
552	23.4946	8.2031	587	24.2280	8.3729	622	24.9399	8.5361
553	23.5159	8.2080	588	24.2487	8.3777	623	24.9599	8.5407
554	23.5372	8.2130	589	24.2693	8.3824	624	24.9799	8.5453
555	23.5584	8.2179	590	24.2899	8.3872	625	25.0000	8.5498
556	23.5796	8.2228	591	24.3104	8.3919	626	25.0199	8.5544
557	23.6008	8.2278	592	24.3310	8.3966	627	25.0399	8.5589
558	23.6220	8.2327	593	24.3515	8.4013	628	25.0599	8.5635
559	23.6431	8.2376	594	24.3721	8.4061	629	25.0798	8.5680
560	23.6643	8.2425	595	24.3926	8.4108	630	25.0998	8.5726

1163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*
631	25.1197	8.5771	666	25.8069	8.7328	701	26.4764	8.8832
632	25.1396	8.5816	667	25.8263	8.7372	702	26.4952	8.8874
633	25.1594	8.5862	668	25.8456	8.7416	703	26.5141	8.8917
634	25.1793	8.5907	669	25.8650	8.7459	704	26.5329	8.8959
635	25.1992	8.5952	670	25.8843	8.7503	705	26.5518	8.9001
636	25.2190	8.5997	671	25.9036	8.7546	706	26.5706	8.9043
637	25.2388	8.6042	672	25.9229	8.7590	707	26.5894	8.9085
638	25.2586	8.6087	673	25.9422	8.7633	708	26.6082	8.9127
639	25.2784	8.6132	674	25.9615	8.7677	709	26.6270	8.9169
640	25.2982	8.6177	675	25.9807	8.7720	710	26.6458	8.9211
641	25.3179	8.6222	676	26.0000	8.7763	711	26.6645	8.9253
642	25.3377	8.6267	677	26.0192	8.7807	712	26.6833	8.9294
643	25.3574	8.6311	678	26.0384	8.7850	713	26.7020	8.9336
644	25.3771	8.6356	679	26.0576	8.7893	714	26.7207	8.9378
645	25.3968	8.6401	680	26.0768	8.7936	715	26.7394	8.9420
646	25.4165	8.6445	681	26.0959	8.7979	716	26.7581	8.9461
647	25.4361	8.6490	682	26.1151	8.8022	717	26.7768	8.9503
648	25.4558	8.6534	683	26.1342	8.8065	718	26.7955	8.9545
649	25.4754	8.6579	684	26.1533	8.8108	719	26.8141	8.9586
650	25.4950	8.6623	685	26.1725	8.8151	720	26.8328	8.9628
651	25.5147	8.6668	686	26.1916	8.8194	721	26.8514	8.9669
652	25.5342	8.6712	687	26.2106	8.8237	722	26.8700	8.9711
653	25.5538	8.6756	688	26.2297	8.8280	723	26.8886	8.9752
654	25.5734	8.6801	689	26.2488	8.8322	724	26.9072	8.9793
655	25.5929	8.6845	690	26.2678	8.8365	725	26.9258	8.9835
656	25.6124	8.6889	691	26.2868	8.8408	726	26.9443	8.9876
657	25.6320	8.6933	692	26.3058	8.8450	727	26.9629	8.9917
658	25.6515	8.6977	693	26.3248	8.8493	728	26.9814	8.9958
659	25.6709	8.7021	694	26.3438	8.8535	729	27.0000	9.0000
660	25.6904	8.7065	695	26.3628	8.8578	730	27.0185	9.0041
661	25.7099	8.7109	696	26.3818	8.8620	731	27.0370	9.0082
662	25.7293	8.7153	697	26.4007	8.8663	732	27.0554	9.0123
663	25.7487	8.7197	698	26.4196	8.8705	733	27.0739	9.0164
664	25.7681	8.7241	699	26.4386	8.8748	734	27.0924	9.0205
665	25.7875	8.7285	700	26.4575	8.8790	735	27.1108	9.0246

1163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*
736	27.1293	9.0287	771	27.7668	9.1696	806	28.3901	9.3063
737	27.1477	9.0328	772	27.7848	9.1735	807	28.4077	9.3101
738	27.1661	9.0368	773	27.8028	9.1775	808	28.4253	9.3140
739	27.1845	9.0409	774	27.8208	9.1815	809	28.4429	9.3178
740	27.2029	9.0450	775	27.8388	9.1854	810	28.4604	9.3216
741	27.2213	9.0491	776	27.8567	9.1894	811	28.4780	9.3255
742	27.2396	9.0531	777	27.8747	9.1933	812	28.4956	9.3293
743	27.2580	9.0572	778	27.8926	9.1972	813	28.5131	9.3331
744	27.2763	9.0613	779	27.9105	9.2012	814	28.5306	9.3370
745	27.2946	9.0653	780	27.9284	9.2051	815	28.5482	9.3408
746	27.3130	9.0694	781	27.9463	9.2090	816	28.5657	9.3446
747	27.3313	9.0734	782	27.9642	9.2130	817	28.5832	9.3484
748	27.3495	9.0775	783	27.9821	9.2169	818	28.6006	9.3522
749	27.3678	9.0815	784	28.0000	9.2208	819	28.6181	9.3560
750	27.3861	9.0856	785	28.0178	9.2247	820	28.6356	9.3599
751	27.4043	9.0896	786	28.0356	9.2287	821	28.6530	9.3637
752	27.4226	9.0936	787	28.0535	9.2326	822	28.6705	9.3675
753	27.4408	9.0977	788	28.0713	9.2365	823	28.6879	9.3713
754	27.4590	9.1017	789	28.0891	9.2404	824	28.7054	9.3750
755	27.4772	9.1057	790	28.1069	9.2443	825	28.7228	9.3788
756	27.4954	9.1097	791	28.1247	9.2482	826	28.7402	9.3826
757	27.5136	9.1137	792	28.1424	9.2521	827	28.7576	9.3864
758	27.5317	9.1177	793	28.1602	9.2560	828	28.7749	9.3902
759	27.5499	9.1218	794	28.1780	9.2599	829	28.7923	9.3940
760	27.5680	9.1258	795	28.1957	9.2637	830	28.8097	9.3977
761	27.5862	9.1298	796	28.2134	9.2676	831	28.8270	9.4015
762	27.6043	9.1338	797	28.2311	9.2715	832	28.8444	9.4053
763	27.6224	9.1377	798	28.2488	9.2754	833	28.8617	9.4091
764	27.6405	9.1417	799	28.2665	9.2793	834	28.8790	9.4128
765	27.6586	9.1457	800	28.2842	9.2831	835	28.8963	9.4166
766	27.6767	9.1497	801	28.3019	9.2870	836	28.9136	9.4203
767	27.6947	9.1537	802	28.3196	9.2909	837	28.9309	9.4241
768	27.7128	9.1577	803	28.3372	9.2947	838	28.9482	9.4278
769	27.7308	9.1616	804	28.3548	9.2986	839	28.9654	9.4316
770	27.7488	9.1656	805	28.3725	9.3024	840	28.9827	9.4353

1,163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*
841	29.0000	9.4391	876	29.5972	9.5682	911	30.1827	9.6940
842	29.0172	9.4428	877	29.6141	9.5719	912	30.1993	9.6976
843	29.0344	9.4466	878	29.6310	9.5755	913	30.2158	9.7011
844	29.0516	9.4503	879	29.6479	9.5792	914	30.2324	9.7046
845	29.0688	9.4540	880	29.6647	9.5828	915	30.2489	9.7082
846	29.0860	9.4577	881	29.6816	9.5864	916	30.2654	9.7117
847	29.1032	9.4615	882	29.6984	9.5900	917	30.2820	9.7153
848	29.1204	9.4652	883	29.7153	9.5937	918	30.2985	9.7188
849	29.1376	9.4689	884	29.7321	9.5973	919	30.3150	9.7223
850	29.1547	9.4726	885	29.7489	9.6009	920	30.3315	9.7258
851	29.1719	9.4761	886	29.7657	9.6045	921	30.3479	9.7294
852	29.1890	9.4801	887	29.7825	9.6081	922	30.3644	9.7329
853	29.2061	9.4838	888	29.7993	9.6117	923	30.3809	9.7364
854	29.2232	9.4875	889	29.8161	9.6153	924	30.3973	9.7399
855	29.2403	9.4912	890	29.8328	9.6190	925	30.4138	9.7434
856	29.2574	9.4949	891	29.8496	9.6226	926	30.4302	9.7469
857	29.2745	9.4986	892	29.8663	9.6262	927	30.4466	9.7504
858	29.2916	9.5023	893	29.8831	9.6297	928	30.4630	9.7539
859	29.3087	9.5059	894	29.8998	9.6333	929	30.4795	9.7575
860	29.3257	9.5096	895	29.9165	9.6369	930	30.4959	9.7610
861	29.3428	9.5133	896	29.9332	9.6405	931	30.5122	9.7644
862	29.3598	9.5170	897	29.9499	9.6441	932	30.5286	9.7679
863	29.3768	9.5207	898	29.9666	9.6477	933	30.5450	9.7714
864	29.3938	9.5244	899	29.9833	9.6513	934	30.5614	9.7749
865	29.4108	9.5280	900	30.0000	9.6548	935	30.5777	9.7784
866	29.4278	9.5317	901	30.0166	9.6584	936	30.5941	9.7829
867	29.4448	9.5354	902	30.0333	9.6620	937	30.6104	9.7854
868	29.4618	9.5390	903	30.0499	9.6656	938	30.6267	9.7889
869	29.4788	9.5427	904	30.0665	9.6691	939	30.6431	9.7923
870	29.4957	9.5464	905	30.0832	9.6727	940	30.6594	9.7958
871	29.5127	9.5500	906	30.0998	9.6763	941	30.6757	9.7993
872	29.5296	9.5537	907	30.1164	9.6798	942	30.6920	9.8028
873	29.5465	9.5573	908	30.1330	9.6834	943	30.7083	9.8062
874	29.5634	9.5610	909	30.1496	9.6869	944	30.7245	9.8097
875	29.5803	9.5646	910	30.1662	9.6905	945	30.7408	9.8131

1,163. SQUARE ROOTS AND CUBE ROOTS (*continued*).

No.	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*	*No.*	*Square Roots.*	*Cube Roots.*
946	30.7571	9.8166	966	31.0805	9.8853	986	31.4006	9.9531
947	30.7733	9.8201	967	31.0966	9.8887	987	31.4165	9.9564
948	30.7896	9.8235	968	31.1126	9.8921	988	31.4324	9.9598
949	30.8058	9.8270	969	31.1287	9.8955	989	31.4483	9.9631
950	30.8220	9.8304	970	31.1448	9.8989	990	31.4642	9.9665
951	30.8382	9.8339	971	31.1608	9.9023	991	31.4801	9.9699
952	30.8544	9.8373	972	31.1769	9.9057	992	31.4960	9.9732
953	30.8706	9.8408	973	31.1929	9.9091	993	31.5119	9.9766
954	30.8868	9.8442	974	31.2089	9.9125	994	31.5277	9.9799
955	30.9030	9.8476	975	31.2249	9.9159	995	31.5436	9.9833
956	30.9192	9.8511	976	31.2409	9.9193	996	31.5594	9.9866
957	30.9354	9.8545	977	31.2569	9.9227	997	31.5753	9.9899
958	30.9515	9.8579	978	31.2729	9.9261	998	31.5911	9.9933
959	30.9677	9.8614	979	31.2889	9.9295	999	31.6069	9.9966
960	30.9838	9.8648	980	31.3049	9.9328	1000	31.6227	10.0000
961	31.0000	9.8682	981	31.3209	9.9362			
962	31.0161	9.8716	982	31.3368	9.9396			
963	31.0322	9.8751	983	31.3528	9.9430			
964	31.0483	9.8785	984	31.3687	9.9463			
965	31.0644	9.8819	985	31.3847	9.9497			

The use of the above tables may be extended by observing the following relationships:—

$$\sqrt{0{\cdot}75} = \sqrt{75} \div 10 = \frac{8{\cdot}66}{10} = {\cdot}866,$$

$$\sqrt{0{\cdot}4} = \sqrt{40} \div 10 = \frac{6{\cdot}3245}{10} = 0{\cdot}63245,$$

$$\sqrt{{\cdot}043} = \sqrt{430} \div 100 = \frac{20{\cdot}736}{100} = {\cdot}20736,$$

$$\sqrt{1{\cdot}43} = \sqrt{143} \div 10 = \frac{11{\cdot}958}{10} = 1{\cdot}1958,$$

$$\sqrt{\frac{3}{16}} = \frac{\sqrt{3}}{4} = \frac{1{\cdot}732}{4} = 0{\cdot}433,$$

$$\sqrt[3]{{\cdot}035} = \sqrt[3]{35} \div 10 = \frac{3{\cdot}27}{10} = 0{\cdot}327.$$

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