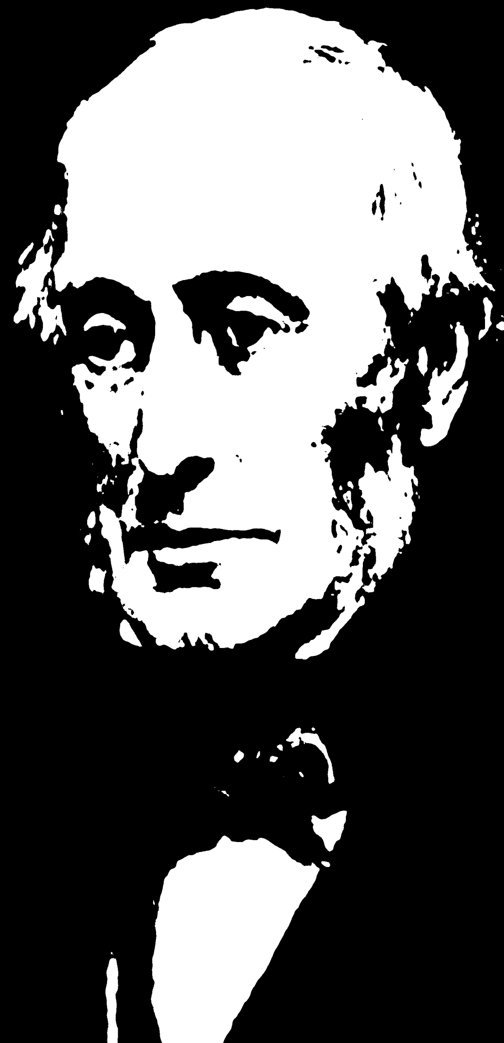

Hydraulic machinery

Robert Gordon Blaine

1. Machinery, Hydraulic

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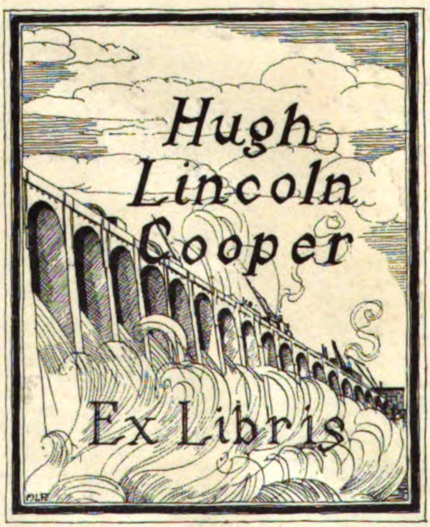

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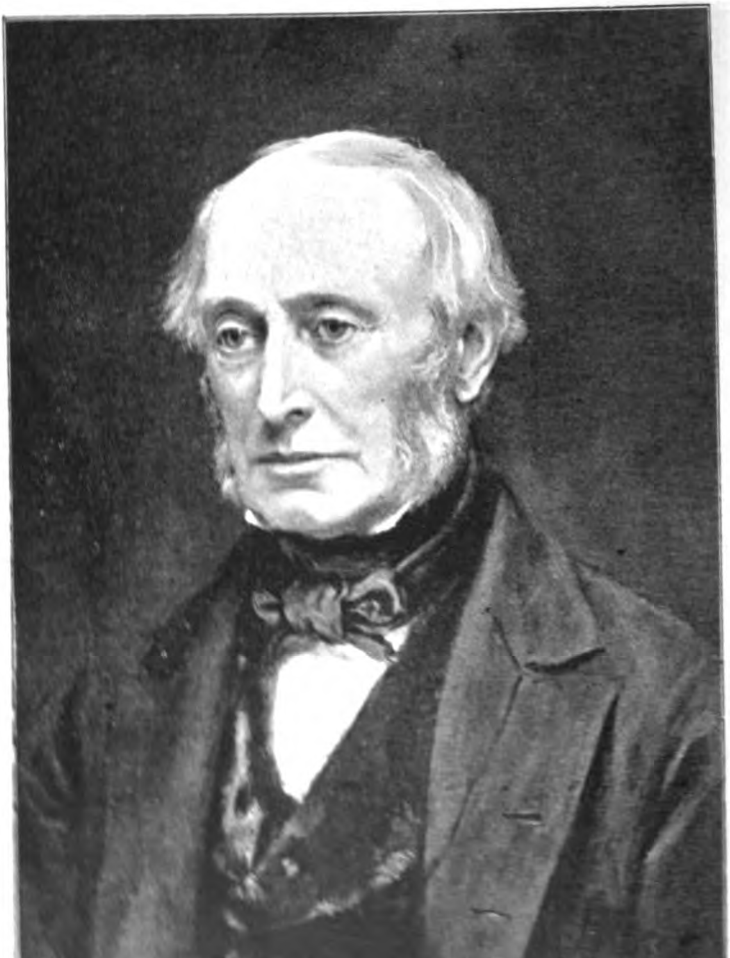

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HYDRAULIC
MACHINERY

WITH AN INTRODUCTION TO HYDRAULICS

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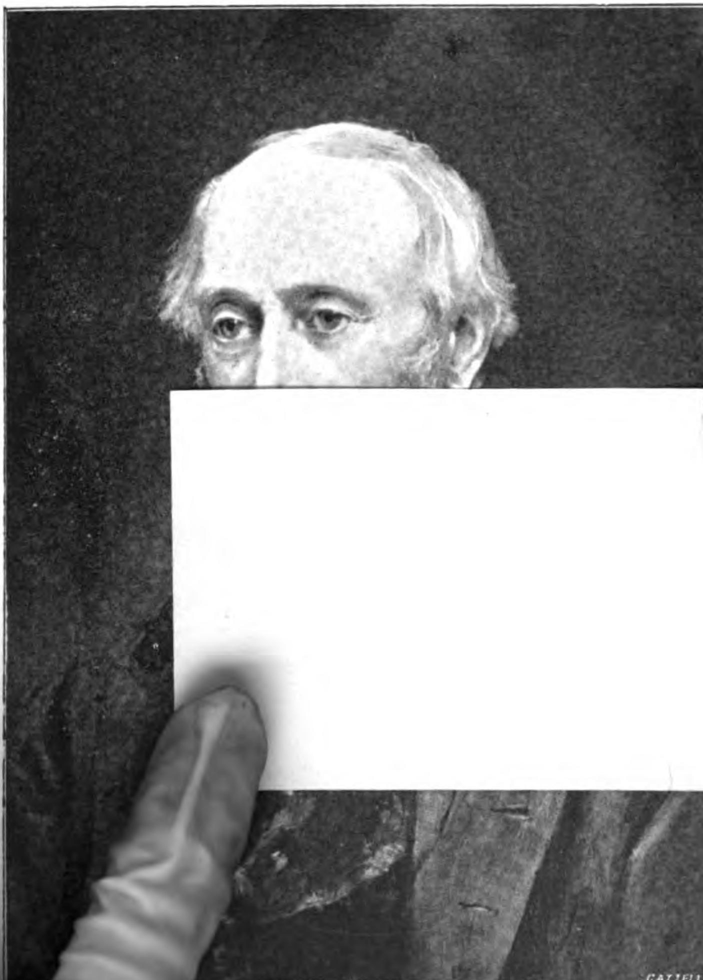

Armstrong

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HYDRAULIC MACHINERY

WITH AN INTRODUCTION TO HYDRAULICS

BY

[redacted]BERT GORDON BLAINE, M.E.

ASSOC. M. INST. C.E., ETC.

[redacted]TRATOR AND LECTURER IN THE MECHANICAL ENGINEERING DEPARTMENT
OF THE CITY AND GUILDS OF LONDON INSTITUTE'S
TECHNICAL COLLEGE, FINSBURY

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PREFACE

TO

FIRST EDITION.

I VENTURE to believe that there is still room for a book on the subject of Hydraulic Machinery; this belief being confirmed by the fact that students frequently ask me to name a work on this subject suited to their wants. The difficulty in naming a work, obtainable at a moderate price, and which contains really sound information, couched in language that ordinary students and readers can understand, has led me to produce the present volume.

In books of this class an attempt is usually made to avoid using the calculus, or to disguise its use in the language of so-called elementary mathematics; this course is not altogether free from objections, the proofs given being usually long, difficult, and not too exact.

The present work is the result of a suggestion by Professor Perry, F.R.S., whose treatment of the theoretical portions of the subject I have followed; and I venture to think that, although in some cases it has seemed necessary to make use of elementary applications of the calculus, the proofs are simple, easy, and satisfactory. The student who does not possess the small amount of knowledge necessary to follow the reasoning, had better accept the results without proof than attempt to master those often given.

My many years' connection with Professor Perry as his chief assistant, precludes any idea on my part of putting forward a claim to originality in a subject which the Professor always invests with a peculiar interest. I therefore take this opportunity of acknowledging my indebtedness, and returning my thanks, not only to Professor Perry for his readily given and generous help, but to all others who have assisted me. I would mention the name of my colleague, Mr. Robert Johnston, Whitworth Scholar, as one to whom I am specially indebted for valuable practical suggestions, and assistance in the preparation of drawings for the illustrations.

It is impossible to refer by name to all who have, beyond my hope even, assisted me; but I would state that to the Council of the Institution of Civil Engineers, and that of the Institution of Mechanical Engineers, as well as to the proprietors and editors of 'The Engineer,' of 'Engineering' and of 'Cassier's Magazine' I am under great obligations for permission to reproduce illustrations which have appeared in their respective journals. The last-mentioned have enhanced the value of their permission by the loan of some valuable blocks.

To the heads of engineering firms and private friends I am also much indebted. Among the former the directors of Sir W. G. Armstrong & Co, and the directors of Tweddell's System, Limited, have my special thanks for generous help. I hope that all others who have assisted me—and whose names I have, as far as possible, mentioned throughout the work—will accept this method of publicly returning them my hearty thanks.

I have tried to produce a work containing sound information, not only in regard to the elements of the subject, but also in respect of good modern examples of hydraulic machinery of almost every class.

I trust that not only ordinary readers, and students of engineering, but also those of higher practical attainments may find that the book will repay their perusal.

ROBERT GORDON BLAINE.

CITY GUILDS' TECHNICAL INSTITUTE,
FINSBURY, LONDON, E.C.

PREFACE

TO

THE SECOND EDITION.

THE exhaustion of a very large first edition, made specially large in order to admit of the price being reduced, has given an opportunity for a complete revision of the work, for the re-writing of a considerable portion, and for the introduction of much new material.

Of the latter, sections dealing with "The Stability and Resistance of Ships," the "Efficiency of Centrifugal Pumps," "Graphic Methods," in connection with turbine design, "The Largest Turbine yet Constructed," "Turbines for Low Falls," "Hydraulic Governors," including a description of the powerful governors now being erected to control the Ontario Power Company's 10,000 horse-power units at Niagara, "Hydraulic Foundry and Steelworks Cranes, and a Hydraulic Gantry," and "Hydraulic Gas-Stoking Machinery," all fully illustrated, will no doubt be of interest to the student, whether he be a novice, or one more happily situated as regards experience.

The section on "Hydraulics" has been completely re-written and greatly extended. New tables have been inserted, recent experiments and deductions introduced, and an interesting, simplified, graphic solution of Ganguillet and Kutter's difficult equation, giving the value of c in the well-known formula $v = c\sqrt{m\,i}$, has been given, so that the reader may for himself construct a drawing rendering him independent of tables, and embracing the exact conditions with which he has to deal.

The chapter on "Hydraulic Rams" has also been re-written and extended to more than twice its original length, most of the typical forms are described and illustrated, and curves of efficiency are supplied. Extensive use has, in fact, been made throughout of curves or "graphs," as it is often of more importance that a student shall be able to form a good idea of the law of variation of two related quantities, than merely to obtain their numerical values under given conditions.

The desire to keep the book of a moderate size and price has led to the condensation of some sections, but the references to original papers and authorities will enable the reader to prosecute the study of details in which he may be specially interested. The first edition, whilst fairly successful here, circulated more widely in the United States and other countries; and that in its present form the work may be still more useful and suggestive is the earnest wish of

THE AUTHOR.

LONDON: *May* 1905.

CONTENTS.

HYDRAULIC MACHINERY.

INTRODUCTION.

MACHINERY actuated by water is termed "hydraulic machinery," and writers often include under this title machines, such as pumps, which act *on* water. Hydraulic appliances were known and used from a very early date. Many of these, mainly for raising water, were employed long before the beginning of the Christian era.

The use of water as a natural source of power has not been as much resorted to in this country as in many others, owing to our large supplies of coal, and the fact that a water supply with sufficient fall is not often available where the power is required. The perfection attained in the construction of turbine water wheels, together with the decline of our coal supply and the perfecting of electrical methods of transmission, render this source of power one of increasing importance.

Without referring at length to the history of the development of hydraulic machinery, it may be mentioned that the invention of the force pump by Ctesibius about 200 B.C., of the double-acting pump by La Hire in 1718, the hydraulic ram by Whitehurst in 1772, and the hydraulic press by Joseph Bramah in 1802, are important epochs.

The suitability of water as a medium for the *transmission* of power has been fully recognised in recent years, thanks mainly to the late Lord Armstrong, to whose inventive genius we are indebted for the initiation of our modern central station hydraulic systems.

The provision of an efficient, moderate-speed, self-governing, high-pressure water motor for variable powers—now occupying the attention of inventors—will, no doubt, greatly extend the use of hydraulic power. The following pages are written with the hope of assisting the student to obtain a fairly thorough groundwork of knowledge in connection with this subject.

I.

COMPRESSIBILITY OF WATER.

A FLUID is "something which flows," and may range in consistence from the very viscous pitch which breaks with a glossy fracture, but which, if left heaped up in a bucket, gradually settles down and "flows" over the edge of the bucket in festoons, to a very volatile and highly compressible fluid such as a gas. Fluids of that class, which are only very slightly compressible, offering very little resistance to change of shape but great resistance to change of volume, are called "liquids." Water is a good representative of this class, and we shall confine our attention mainly to it.

Water is *not* incompressible, though the old Florentine philosophers thought it was. They devised an experiment which they thought would settle the matter. They knew that a sphere contains a larger volume than any other figure of the same surface area; hence they took a hollow spherical globe of gold, filled it with water and sealed it hermetically. The globe was then beaten so as to make its shape no longer spherical, when small drops of water made their appearance on the surface of the globe, having oozed through the gold rather than submit to a diminution in bulk. The philosophers then decided that water was incompressible, which was not proved by the experiment: all that was proved being the fact that water *resists* compression very much.

A cast-iron shell filled with water, and fitted with a small screw which could be screwed into the shell, gave a similar result, water finding its way to the outer surface of the shell in the form of fine spray when the pressure became very great, the shell shortly afterwards falling gently to pieces. This non-dangerous method of fracture produced by water pressure renders it a favourite medium for the testing of boilers, etc.

Water is compressible, but only to a very slight extent. Hooke's famous law, "Stress is proportional to strain," enables us to find the actual compressibility of water.

The law is:—

$$\left.\begin{array}{l}\text{Change of hydrostatic pressure}\\ \text{all over the body's surface} \;..\end{array}\right\} = \left\{\begin{array}{l}\text{K} \times \text{the fractional change}\\ \text{of volume,}\end{array}\right.$$

where K is the *modulus of cubic compressibility*. Stated algebraically it is,

$$\delta p = -K \frac{\delta v}{V},$$

the negative sign indicating that the volume *diminishes* as the pressure increases. For water K = 300,000 lbs. per square inch, and if we take a change of pressure = one atmosphere (14·7 lbs. per square inch) and an original volume (v) of 1 cubic inch,

$$14{\cdot}7 = 300{,}000 \frac{\delta v}{1},$$

or

$$\delta v = \frac{14{\cdot}7}{300{,}000} = \frac{1}{20{,}410} = \frac{1}{20{,}000} \text{ nearly.}$$

We see, therefore, that the fractional change of volume corresponding to a change of pressure of one atmosphere is about $\frac{1}{20{,}000}$th.

It will not be very far wrong, therefore, to assume that water is incompressible; if, as in many problems, *the pressure is no longer changing*, the volume, of course, remains constant, and, in any case, the change of volume is very small.

II.

FLUIDS AND FLUID PRESSURE.

Fluidity.

Before studying other hydraulic machines, it may be well, in order to understand their action fully, to consider some elementary laws regarding the pressure and flow of fluids.

It is well known that when a substance is kept subjected to stresses for a long time the strains or deformations produced in the substance usually increase with time.

This increase is, however, of importance only in the case of certain substances which have been called *plastic*. Mud, mortar, etc., have high degrees of plasticity, but the solids, wax and pitch, also exhibit this property.

It is probable that if the stresses in the case of sealing wax are only small enough, the wax will behave like steel, but with even such stresses as are produced by its own weight, it bends more and more from day to day, nearly the whole of its deformation being a permanent set. If any substance is subjected to sufficiently high stresses it exhibits *plasticity*. Thus steel can be drawn through a die to form pianoforte wire, and the plasticity of lead, copper and other metals is well known.

Perfect Fluids.

A perfect fluid is incapable of resisting—except by its inertia—a change of shape; that is, it is impossible for it to exert distorting or tangential stresses. Such a substance does not actually exist, for all fluids have *viscosity* or internal friction, which is defined as a resistance to change of shape depending on the rate at which the change is effected. The fluids with which engineers have to deal are water and vapours or gases, and it simplifies some of our calculations to assume that they have no internal friction.

Hydrostatics.

Hydrostatics deals with perfect fluids at rest, and the laws of hydrostatics are found practically to be applicable to water, air, gas, etc., when at rest, or moving slowly as in the hydraulic press. The laws would be applicable to even much more viscous fluids if the motion were only slow enough. The study of the behaviour of fluids in motion is not at all simple, and our knowledge of the laws relating to, say, water in motion is of an elementary kind. Since the laws of hydrostatics, referring primarily to water at rest, may be applied in many calculations connected with hydraulic machines, it may be well to refer briefly to some of the more important of them.

The Nature of Fluid Pressure.

An ordinary fluid at rest, or a perfect fluid under any circumstances, cannot exert tangential forces; hence the pressure on any surface—whether it be the boundary of a solid body or an imaginary interface between two contiguous portions of the fluid—is at every point *perpendicular to the surface.*

The average intensity of pressure on a small surface is measured by dividing the total force distributed over the surface by the area of the surface. As the area becomes smaller and smaller round a point the quotient approaches more and more nearly a limit which is the true value of the intensity of pressure at the point. In a fluid at rest any portion of it is kept at rest by forces acting on its boundary; we may therefore regard this portion as a rigid body.

Two Important Propositions.

At any point in a fluid at rest, the intensity of pressure is the same on any interface, whatever its direction may be; and if no external forces, like gravity, act on the mass of the fluid, the pressure is the same at every point in the fluid. These two propositions may be proved as follows:—

The resultant of the fluid pressures on any portion of a spherical surface must, like its components, pass through the centre of the sphere. Hence, if we imagine a portion of the fluid—of the shape of a plano-convex lens (as in Fig. 1) – solidified, the resultant pressure on the plane side must pass through the centre of the sphere; and therefore, being perpendicular to the plane, must pass through the centre of the plane area. If we take two concentric circles of nearly the same radius, the resultant of the pressures on each must pass through the common centre, from which it follows that the pressure is uniformly distributed over the narrow annulus. Now take the intersecting circles A B E and D E F (Fig. 2), the intensity of pressure at B is the same as at A, since the points are equidistant from the centre O, and the intensity of pressure at D is the same as at A, for they are equidistant from P; hence the intensity of pressure at D on the lune A B E D is the same as at B, and so on for other points.

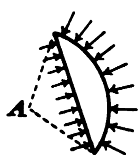


Fig. 1.

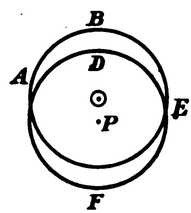


Fig. 2.

Hence the pressure on any plane area is uniformly distributed over the area, and the resultant pressure must therefore pass through the centre of the area or "centre of gravity" of the area.

Next imagine a triangular prism of the fluid, with ends perpendicular to the axis, to become solidified.

Let the areas of its ends be a_1 and a_2, and of its sides a_3, a_4 and a_5 respectively. The forces on the two ends are the only forces in the direction of the axis of the prism; these forces must be equal.

$$\therefore \quad p_1 a_1 = p_2 a_2;$$

but

$$a_1 = a_2;$$

$$\therefore \quad p_1 = p_2;$$

or the intensity of pressure on any two parallel planes is the same. We have seen that the resultant of uniformly distributed pressure over an area acts at the centre of the area.

Now on our prism the forces at right angles to the axis balance, therefore they are parallel to the sides of a triangle (since the ends of the prism are parallel, the resultant forces act in one plane), which

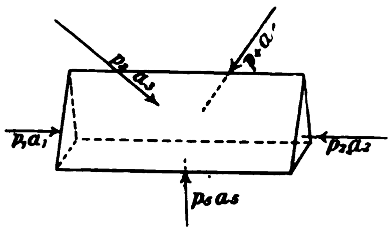


FIG. 3.

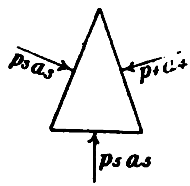


FIG. 4.

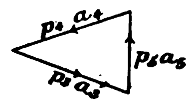


FIG. 5.

triangle has its sides perpendicular respectively to the sides of a right section of the prism; therefore these figures are similar (Figs. 3, 4 and 5): hence

$$p_5 a_5 : a_5 :: p_4 a_4 : a_4,$$

since the sides of the triangle, Fig. 5, are proportional to the areas a_3, a_4 and a_5 respectively, or $p_5 = p_4$.

Similarly,

$$p_3 = p_4 = p_5,$$

hence the pressure at a point in a fluid is the same in all directions.

PRESSURE DUE TO GRAVITY.

In the foregoing, volumetric forces like gravity were not taken into account. Consider a liquid acted on *only* by gravity. In Figs. 6 and 7 are seen a side and front view of a plane area immersed in the liquid. Let the intensity of pressure at depth y be p (variable); then the pressure on a very small area δa is $p\,\delta a$, and the whole pressure $= \Sigma p\,\delta a$ for the whole area. But from the rule for finding $\bar{y}$, $\Sigma y\,\delta a = \bar{y}\,A$, where A is the whole area, $\bar{y}$ the depth of the "centre of gravity" of the area,

$$\therefore\ w\,\Sigma y\,\delta a = w\,A\,\bar{y},$$

or

$$\Sigma w\,\delta a\,y = w\,A\,\bar{y}.$$

But $w\,\delta a\,y = p\,\delta a$, for $w\,\delta a\,y$ = pressure on area δa.

$$\therefore \quad \Sigma p\,\delta a = w\,A\bar{y},$$

or the pressure on the whole area is found by multiplying *the weight of unit volume of the liquid by the area and by the depth of its centre of gravity below the surface.*

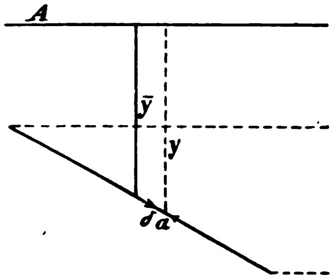


FIG. 6.

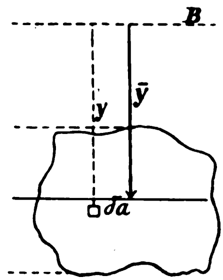


FIG. 7.

Let this total pressure be denoted by R; then $p_0\,A = R$, if p_0 is the average intensity of pressure, i.e. $p_0\,A = w\,A\bar{y}$; or $p_0 = w\bar{y}$, and $w\bar{y}$ is the pressure on unit area at depth $\bar{y}$. Hence the *average intensity of pressure is the intensity of pressure at the "centre of gravity" or centre of the area.*

These rules also hold for areas which are not plane.

Centre of Pressure.

To find the *centre of pressure,* or point of application of the resultant of all the pressures on a plane area, consider gravity alone acting on an incompressible fluid.

We saw that $R = w\,\Sigma h\,a$. Now, if we take two axes of reference in the plane of the area in question, the axis of y being the line in which this plane meets the water surface, the axis of x a line in the plane at right angles to the first axis; then, since the sum of the moments of all the elementary forces about either of these axes must be equal to the moment of their resultant, if the element of area δa has co-ordinates x and y, and if the centre of pressure has co-ordinates $\bar{\bar{x}}$ and $\bar{\bar{y}}$, and if the inclination of the plane area to the vertical is θ, the whole pressure on δa is $w\,\delta a\,x\cos\theta$, and the moments of this about the two axes are

$$\delta a\,w\,x\cos\theta\,.\,x \quad \text{and} \quad \delta a\,w\,x\cos\theta\,.\,y,$$

so that

$$R \bar{\bar{x}} = w \cos \theta \Sigma \delta a x^2,$$
$$R \bar{\bar{y}} = w \cos \theta \Sigma \delta a x y.$$

The expression $\Sigma \delta a x^2$ is the moment of inertia of the area A about the axis of y, and may be denoted by I; the expression $\Sigma \delta a x y$ is sometimes called the *product of inertia* about the axes of x and y.

If, then $\bar{x}$ and $\bar{y}$ are the co-ordinates of the centre of area, as $\bar{h} = \bar{x} \cos \theta$, we have

$$\bar{\bar{x}} = \frac{w \cos \theta \, I}{w \bar{x} \cos \theta \, A},$$

or

$$\bar{\bar{x}} = \frac{I}{\bar{x} A},$$

and

$$\bar{\bar{y}} = \frac{\Sigma \delta a x y}{\bar{x} A}.$$

We see, then, that the position of the centre of pressure is independent of θ—the inclination of the area to the vertical.

Example.—A rectangle inclined at the angle θ to the vertical has one side, a feet, just on the surface, its inclined sides being each b feet long. Find the position of a single force which will balance all the pressure on the rectangle.

Here $\bar{x} = \frac{b}{2}$, and we find by calculation or from a table that I of a rectangle about side a is $\frac{a b^3}{3}$, so that

$$\bar{\bar{x}} = \frac{\frac{a b^3}{3}}{\frac{b}{2} \times a b} = \frac{2}{3} b;$$

and

$$R = w \bar{x} \,.\, \cos \theta \,.\, A$$
$$= w \frac{b}{2} \cos \theta \, a b$$
$$= w \cos \theta \frac{a b^2}{2}.$$

R is the force, and it is at right angles to the rectangle at a point two-thirds of the way downwards, along a central line parallel to the side b. It is evident that the centre of pressure is in this line from symmetry, and it is at a point two-thirds of the way down, whether the plane of the rectangle be inclined or vertical.

If I_0 is the moment of inertia of the area in question about a horizontal line through its centre, we know that

$$I = I_0 + A(\bar{x})^2,$$

and

$$I_0 = A K^2$$

where K is the radius of gyration of the area about this axis.

$$\therefore \quad \bar{\bar{x}} = \frac{A K^2 + A(\bar{x})^2}{A \bar{x}} = \frac{K^2}{\bar{x}} + \bar{x}.$$

Hence the distance measured parallel to the axis of x of the centre of pressure from the centre of area is $\frac{K^2}{\bar{x}}$, or, if h is the depth of the centre of area, this distance is $K^2 \cos\theta \div h$. This distance is zero when the area is horizontal, and is negligible when $\bar{x}$ is great compared with K^2.

Example.—Find the centre of pressure of, and the total pressure on, a triangular area immersed in water, base 6 feet, height 10 feet, base in the surface and its plane inclined at 60° to the horizontal. The moment of inertia of a triangle of height h about its base is $A\frac{h^2}{6}$ where A is the area of the triangle.

$$\bar{\bar{x}} = \frac{I}{\bar{x} A} = \frac{A h^2}{6 \times \frac{h \times A}{3}} = \frac{h}{2}.$$

In this case the centre of pressure is 5 feet from the base of the triangle. The total pressure is

$$62 \cdot 4 \times 30 \times \frac{10}{3} \times \sin 60^\circ = 62 \cdot 4 \times 10 \times 10 \times \frac{\sqrt{3}}{2}$$
$$= 62 \cdot 4 \times 50 \sqrt{3} = 5403 \cdot 8 \text{ lbs.}$$

Another law of hydrostatics of importance in studying hydraulic machinery, known as the principle of Archimedes—capable of easy experimental demonstation—that *a body loses in weight by immersion in a liquid an amount equal to the weight of the liquid displaced*, may be proved as follows:—

Imagine a portion of the liquid mass to become solidified without change of weight or volume; this portion is at rest under the action of the surface pressures and its own weight, hence the upward resultant of the surface pressures must be equal and opposite to its weight, and must act through its centre of gravity. If this mass be replaced by one exactly the same in size and shape, but of, say, a heavier material, the surface pressures are the same as before, hence it, too, is subjected to an ***upward pressure equal to the weight of that portion of the liquid displaced by it.***

Stability of Floating Bodies.

The term, "centre of buoyancy," is given to that point which is the centre of gravity of the displaced liquid, and it is through this point that the resultant of the upward or buoyancy forces acts.

When a body floats either wholly or partially immersed, it is necessary for equilibrium that—(1) the weight of the body shall be equal to the weight of the liquid displaced by it; (2) that the centre of gravity of the whole floating body shall be in the same vertical line as the centre of buoyancy; and (3) in the case of a body wholly immersed, the centre of gravity must be below, and in the case of a body partially immersed it must be above, the centre of buoyancy.

Suppose Fig. 8 to represent the cross-section of a ship, G being her centre of gravity, O the centre of buoyancy when in the vertical or position of equilibrium. If the ship heels though a small angle θ, A B being the old and E F the new water-line, the new centre of buoyancy being D, then if a vertical line be drawn through D to meet in M the line O G, the righting couple or moment of stability is $D \times GK = D \times GM \sin \theta$, where D is the weight or displacement of the ship. The point M is called the ***metacentre***, and the comparative stability of the ship is proportional to G M the ***metacentric height.*** By stability is here meant transverse stability, which is the minimum stability of the ship, and is therefore that of most importance.

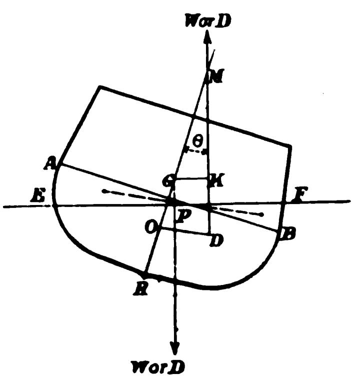


Fig. 8.

Technically a ***stiff*** ship—as regards stability—is one in which the righting couple or moment of statical stability is fairly large; a ***crank*** ship is one which opposes little resistance to inclination or heeling; and a ***steady*** ship is one which, when exposed to the action of waves in a seaway, keeps nearly upright. It frequently happens that a stiff ship is least steady, whilst crank ships are most steady in a seaway. Looking at the matter from the metacentric point of view, the ship is like a pendulum with its point of suspension at M and its weight all concentrated at G. If the pendulum be held aside, through an angle θ from its mean position, its weight D, acting downwards, has a tendency to bring the pendulum to its mean position ; this tendency may be expressed as a moment or couple

$$= D \times G M \sin \theta,$$

but the comparison, except so far as regards time of oscillation, fails unless the pendulum and ship are at rest.

Changes in the height of M above G, produce corresponding changes in the stiffness of the ship. For every position of a ship in which she can float between lightest load and heaviest, the position of M, the metacentre, can be found, and its position will be varied by different distributions of load, or changes in the form of the ship, and in all probability the lines O G and D K will not intersect in the same point if the inclinations be much above 15° for ordinary ships. The following rules are evident :--

(1) If the centre of gravity of a ship lies *below* the metacentre, she tends to return to the upright position when displaced, and the equilibrium is *stable*.

(2) If it lies *above* the metacentre, she tends to move away from the upright position, and the equilibrium is *unstable*. And

(3) If it coincides with the metacentre, she tends to move neither way, and the equilibrium is *indifferent*.

When the position of the metacentre can be found, it gives an easy means of determining the line of action of the buoyancy force for moderate angles of inclination in ships of ordinary form, and avoids the necessity of finding the exact location of the centre of buoyancy in an inclined position. In practice, the position of the metacentre is fixed with reference to the centre of buoyancy in the vertical position in the following way.

The intersection of the two water-lines being P, the deviation of the centre of buoyancy is O D, which is the same as the movement of the centre of gravity of the mass of water displaced if the wedge A P E were moved into the position F P B. Let this distance be l, then if O D be drawn parallel to the line joining these centres, $O D = \frac{l s}{V D}$, where s is the volume of the wedge referred to and V D is the volume of displacement of the ship. This gives D the new centre of buoyancy. The angle which O D makes with the horizontal may be taken as $\frac{\theta}{2}$. $\therefore$ O D = 2 O M sin $\frac{\theta}{2}$ approximately. The volume s is usually about proportional to 2 sin $\frac{\theta}{2}$, so that if c be a constant depending on the form of the water-line section $s = c \times 2 \sin\frac{\theta}{2}$ nearly. Hence the height O M is given by the formula

$$OM = OD \div 2 \sin\frac{\theta}{2} = \frac{lc}{VD}.$$

The product $ls = lc \times 2 \sin\frac{\theta}{2}$, and is double of the statical moment of one of the wedges relatively to a fore and aft medial line through P if the density be unity.

Let distances measured lengthwise on this line be denoted by x, and let y denote the distance of a point from this line on a plane bisecting the angle A P E, and let the thickness of the wedge at a point xy be $= y \times 2 \sin\frac{\theta}{2}$. Then

$$s = 2 \sin\frac{\theta}{2} \int\int y \,.\, dy\, dx$$

and

$$c = \int\int y \,.\, dy\, dx;$$

also

$$ls = 4 \sin\frac{\theta}{2} \int\int y^2 \,.\, dy\, dx,$$

whence $lc = 2\int\int y^2 \,.\, dy\, dx$, which is the moment of inertia of the water-line section about the axis through P.

Hence

$$OM = \frac{I}{V.D}.$$

It may be taken that $I = k \times L \times B^3$, where k is a coefficient which has been determined for particular types of ships, L being the length, and B the extreme breadth of the ship, at load-line.

In ships with fine load-line form to others with moderately fine form	k = 0·04 to 0·055
In ships of full form of load-line .	k = 0·06 to 0·065
And for a rectangle . . .	k = 0·0833*

* The student should consult Sir W. H. White's "Manual of Naval [Architec]ture," in which will be found the principal data here given.

It has been found that roughly the centre of buoyancy is from $\frac{2}{5}$ to $\frac{9}{20}$ of the mean draught of the ship below the water-line in ordinary ships, though in yachts it is only from 27 to 30 per cent. of the mean draught. In using these coefficients, care must be taken that the length and extreme breadth are those *at the load-line.*

Example.—In a ship the I of the water-line section $= 0 \cdot 05 \times 286 \times (53)^3$, the volume of displacement being 180,000 (one foot being unit of length); find the distance of the centre of buoyancy in the vertical position from the metacentre, and if the centre of gravity be 4 feet above the centre of buoyancy, find the metacentric height, also the height of the metacentre above the water-line, the draught being 18 feet.

Answer.—11·83 feet. 7·83 feet. 4·63 feet.

The position of the centre of gravity is difficult to find by calculation for all conditions of loading, and after a ship is completed, experiments are often made by moving a given weight of ballast or deck-load from side to side and noting the inclination, and thus the true position of the centre of gravity for a given condition is determined. Also the displacing moment being known, by equating it to the righting couple the metacentric height may be roughly found, since a plummet line drawn on a thwart-ship partition before and after displacement will give the angle θ.

Example.—A ship of 5000 tons displacement lies in still water, and the moving of a weight of 4 tons from the centre to the side of the deck, a distance of 28 feet, causes an apparent deviation of a plummet (7 feet long) of 1·5 inch. Find (approximately) the metacentric height. *Answer.*—1·25 foot.

Tanks for water-ballast should be completely filled before a vessel goes to sea, else the motion of the water in them may give rise to a serious reduction of stability.

The following values of metacentric heights for actual ships are interesting:

	feet.
In warships with moderate freeboard	$3\frac{1}{2}$ to $4\frac{1}{2}$
,, ditto with central citadel turrets	$5\frac{1}{2}$ to 8
,, troopships and storeships	2 to 3
,, tugs and small non-seagoing vessels . . .	1 to 2
,, torpedo boats (later types)	0·8 to 2
,, transatlantic mail steamers of new type with cellular double-bottom large deck houses, and light rig, about	-1 to $+1$.

In submerged ships or submarines, as regards stability they are like cigar-shaped ships, the inclination producing no change in the form of displacement, hence for all inclinations the buoyancy force acts upwards through the same point. Stable equilibrium, as already stated, is only possible when the centre of gravity lies below the centre of buoyancy. In case of these wholly submerged vessels, the centre of buoyancy takes the place of the metacentre.

Resistance of Ships.

The total resistance to the passage of a ship through the water is due to frictional resistance (similar to that referred to in page 26), wave-making resistance depending on the shape and dimensions of the vessel, and eddy-making resistance due mainly to the bluntness of the stern. The first is the most important, and may amount to from 50 to 90 per cent. of the total resistance. Experiments at Haslar showed that for battleships and cruisers going at full speed only 55 per cent. of the whole resistance was due to skin friction. At a speed of 10 knots the percentages were 79 and 84 respectively. The eddy-making resistance is not usually more than 10 per cent. of the whole, and is often much less.

Froude's law connecting power, velocity and displacement may be stated somewhat as follows :

Let L_1 and L_2 = lengths of ship and its model respectively.
D_1 and D_2 = the displacements of ship and model.
R_1 and R_2 = the resistances " "

It has been found that the resistances are proportional to the displacements, which again are proportional to the cubes of the lengths.

Now if v_1 and v_2 are the velocities of ship and model respectively,

$$\frac{v_1^2}{v_2^2} = \frac{L_1}{L_2}, \quad \text{also} \quad \frac{D_1}{D_2} = \frac{L_1^3}{L_2^3}$$

and

$$\frac{R_1}{R_2} = \frac{D_1}{D_2} = \left(\frac{v_1}{v_2}\right)^6.$$

The horse-power and therefore the coal consumption per hour being proportional to $R\,v$, or in other words, to $D^{\frac{7}{6}}$ or $L^{\frac{7}{2}}$ or v^7, the coal consumption per mile is proportional to D or L^3 or v^6.

Now since R is proportional to L^3, i.e. is proportional to $L \times L^2$, it is proportional to $v^2 \times D^{\frac{2}{3}}$,

or

$$R = c\,v^2\,D^{\frac{2}{3}}.$$

Hence $R\,v$ (horse-power) $= c\,v^3\,D^{\frac{2}{3}}$

or

$$\frac{v^3 D^{\frac{2}{3}}}{H\,P} = \text{a constant for ships of the same class.}$$

This constant is for many ships about 240, if v is expressed in knots and D in tons, the *indicated* horse-power being taken.

Example.—If a ship of 1800 tons displacement moves at 10 knots when the indicated horse-power is 660, find the horse-power necessary to propel the same ship at 16 knots, the displacement being unaltered.

Answer.—$2703\frac{1}{3}$.

MAXIMUM POWER AND SPEED.

In the case of deep oceans the depth of the water has not to be taken into account in deducing a connection between maximum speed and horse-power necessary to propel a ship at that speed, but in the case of ships which make their voyages mainly in shallow seas, this depth is of importance. The following rule has recently been deduced from speed trials of vessels of different types belonging to the Danish and Russian navies:

$$\frac{h}{H} = \left(\frac{v}{V}\right)^p$$

where

$$p = 2 + i + C\,(1 + i)\,(i)^2\,;$$

i being the ratio $\frac{v}{V}$ and $C = K\,(10)^{\mu - 0.2}$ where $\mu = \frac{t}{T}$ and $K = \frac{L}{B}$.

In this law V is the maximum speed in knots, in water of mean depth T, h the indicated horse-power at speed v (less than V) in water of the same mean depth. L = length of ship measured on line of flotation, and B = greatest breadth at same plane, t being the mean draught of the ship. T, t, L and B are measured in metres. The value of p approximates towards 2 or 3 according as the ratio $\frac{v}{V}$ decreases or increases, and reaches its maximum value for a ratio

$$\frac{v}{V} = \frac{1}{3} + \sqrt{\frac{1}{9} + \frac{1}{3\,C}}.$$

A curve giving the connection between $\frac{v}{V}$ and $\frac{h}{H}$ will show the maximum value of V for given values of the other quantities.*

* Bulletin de l'Association Technique Maritime, 1903.

III.

LINES OF FORCE AND EQUIPOTENTIAL SURFACES.

Lines of force in a fluid are such that the direction of any one of them shows the direction of resultant force on a particle of the fluid there.

If a fluid were acted on by gravity only, the lines of force would be radial to the centre of the earth, and a series of curves cutting these lines orthogonally would generate by revolution a series of *equipotential surfaces* or "level" surfaces. Equipotential or level surfaces are, therefore, in the case of gravity, nearly spherical surfaces. Small portions of the lines of force may be taken as parallel, and the surfaces appear as plane surfaces. To prove that

EQUIPOTENTIAL SURFACES ARE SURFACES OF EQUAL PRESSURE AND EQUAL DENSITY.

Since we are most concerned with that class of fluids called liquids, consider a prism of a liquid at rest relative to the rest of the liquid. Let *a* be the area of either end (Fig. 9). The end pressures are the same, the side forces producing no effect endwise, hence the resultants of the side pressures are at right angles to the axis of the prism, i.e. a line of force (same in direction as one of the resultant forces on the sides of the prism) is perpendicular to an equipotential surface. In an equipotential surface, therefore, there is no force tending to move a particle in the direction of, or along, the surface.

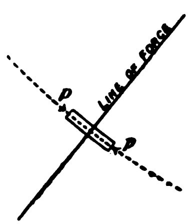


FIG. 9.

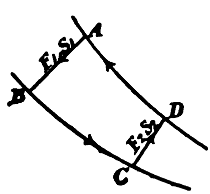


FIG. 10.

Assume no friction.

Let A B and C D (Fig. 10) be lines of force, B C and A D sections of equipotential surfaces. If a particle falls along A B it stores energy in itself equal to $F_1 S_1$; F_1 being the force acting on it, and S_1 the distance A B.

It passes from B to C without effort, passes up from C to D, expending an amount of energy $F_2 S_2$ in doing so, passing from D to A without effort.

On the whole no work is done, the particle arriving where it started from. Hence

$$F_1 S_1 = F_2 S_2.$$

As $F_1 S_1$ is the work stored up in the body in falling from the one equipotential surface to the other, $F_1 S_1$ is the difference of potential of the body in the two positions = the work done on the body in moving it from the first to the second position.

The potential energy of 1 lb. of matter is called "potential," denoted by the letter V.

Let V = the potential energy of 1 lb. of the stuff in the lower level surface (Fig. 11).

$V + \delta V$ = the potential energy of 1 lb. of the stuff in the higher level surface.

δV = the work done in lifting the 1 lb. from lower to higher level surface along a line of force.

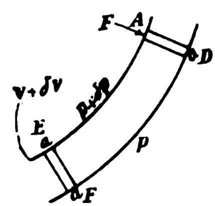


FIG. 11.

If gravity alone acts,

$$\delta V = \delta h.$$

Take little prism of base area a, height A D; its volume is a . A D. If F = force on 1 lb. of stuff, w = weight of unit volume (say number of lbs. in 1 cubic foot).

$$a \,.\, AD \,.\, w \,.\, F = \text{total force on prism},$$
$$a\,(p + \delta p) = \text{force on one end},$$
$$a p = \text{force on the other};$$

hence $a \,.\, \delta p$ must balance the effect of F.

$$\therefore \quad a \,.\, \delta p = - a \,.\, AD \,.\, w \,.\, F,$$

or

$$-\delta p = F \,.\, AD \,.\, w.$$

But

$$F \,.\, AD = \delta V,$$

or

$$F = \frac{\delta V}{AD}.$$

$$\therefore \quad -\delta p = \frac{\delta V}{AD} \,.\, AD \,.\, w.$$

Hence

$$-\delta p = w\,\delta V$$

a most important result, to be carefully remembered. It shows that the change in pressure is proportional to change in potential.

Here w is constant, showing that the density of a liquid between two surface levels is always the same.

It follows, then, that a level surface is

$$\left.\begin{array}{l}\text{an equipotential}\\ \text{an equal pressure}\\ \text{an equal density}\end{array}\right\}\text{surface.}$$

If gravity alone acted,

$$\delta h = \delta V,$$

or

$$-\delta P' = w\,\delta h,$$

i.e. change of pressure is proportional to change of depth, it being assumed that w is constant. It is usual to assume w constant for water, but this is not absolutely accurate.

$$-\delta P = w\,\delta h, \text{ whence } -P = w \times h + \text{a constant.}$$

Let $h = -H$, and let H represent depth in feet.

$-P = -w H$ (together with a constant, which may be negative).

$$\therefore \quad P = w \times H + c.$$

Let $P = P_0$ when $H = 0$.

Then $P = w H + P_0$, or $P - P_0 = w H$, a well-known result.

If $P_0 = 0$ when $H = 0$, i.e. if we neglect atmospheric pressure, $P = w H$. For water $w = 62{\cdot}4$ lbs. H being in feet, P is the pressure per square foot, which $= 62{\cdot}4\,H$, or the pressure per square inch (p) due to a depth $H = \dfrac{62{\cdot}4}{144} H, = \dfrac{H}{2{\cdot}3}$.

The actual law, taking the change in the density of the water into account, is

$$p = 43{\cdot}2 \times 10^6 \left\{ \epsilon^{\frac{H}{592{,}800}} - 1 \right\}.*$$

Liquid whirling about an Axis.

Consider 1 lb. of liquid at P a distance of r feet from the axis (Fig. 12).

Let a be the angular velocity in radians per second. The centrifugal force on the 1 lb. is $\dfrac{r a^2}{g}$ $\left(\text{since mass} = \dfrac{1}{g}\right)$. The force of

* See the author's "Numerical Examples in Practical Mechanics," p. 19.

gravity is 1 lb. Therefore the resultant force on the 1 lb. of liquid is

$$\sqrt{\frac{r^2 \alpha^4}{g^2} + 1}, \qquad \tan\theta = \frac{r\alpha^2}{g}.$$

The slope of the line of force is here negative.

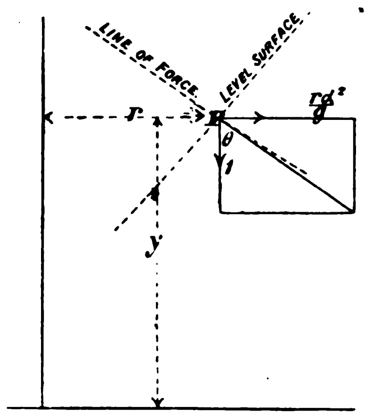


FIG. 12.

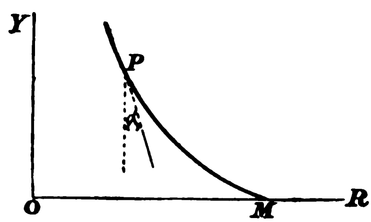


FIG. 13.

$$\therefore \quad -\tan\theta = \frac{r\alpha^2}{g} = \frac{-dr}{dy}.$$

$$\therefore \quad y = \frac{-g}{\alpha^2}\int \frac{dr}{r},$$

or $$y = \frac{-g\log r}{\alpha^2} + c.$$

Hence $$\frac{-\alpha^2}{g}(y - c) = \log r,$$

or the *lines of force are logarithmic curves.*

The student will find it a useful exercise to draw some of these curves. Suppose, for instance, we wish to draw the line of force, which cuts the horizontal axis O R in M (Fig. 13).

Let $y = 0$ when $r = \text{O M}$, and we find for c the value $\frac{g}{\alpha^2}\log_e \text{O M}$. In fact our equation

$$(2) \quad y = \frac{-g}{\alpha^2}\log r + c$$

becomes for this line

$$(3) \quad y = \frac{+g}{\alpha^2}\log_e \frac{\text{O M}}{r}.$$

The following instructive example has been worked out by Professor Perry. A mass of water makes half a revolution per second about a vertical axis. Draw the line of force which passes through a point 4 feet from the axis.

Here $\alpha = \pi$, $g = 32 \cdot 2$, O M = 4, all dimensions being in feet. He has taken the following values of r and calculated the corresponding values of y from (3).

r	4	3·5	3	2·5	2	1·5	1	0·5	0·3	0·2	0·1	0
y	0	0·4357	0·9386	1·5335	2·2614	3·2001	4·5228	6·7840	8·4552	9·7740	12·036	∞

The student should remember that the ordinary logarithm of a number must be multiplied by 2·3026 to get the Napierian logarithm

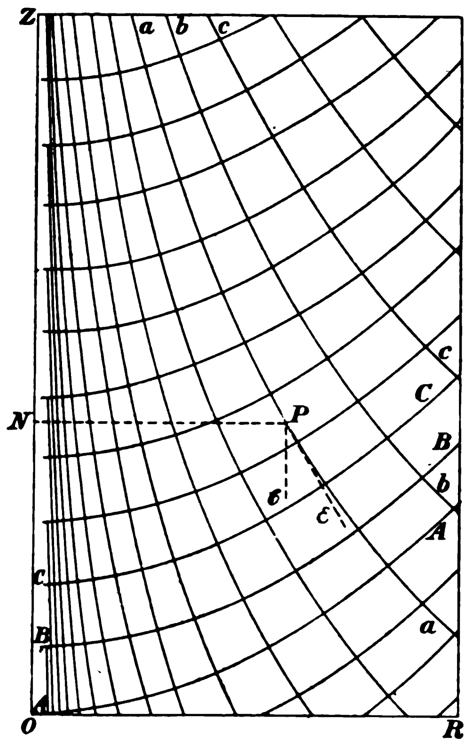


Fig. 14.

(or logarithm to base ε) used in the above equation. The matter may be put thus

$$\log_\epsilon \frac{OM}{r} = 2\cdot3026 \log_{10} \frac{OM}{r}.$$

Hence the equation really employed to calculate these numbers was

$$y = \frac{32\cdot2}{(3\cdot1416)^2} \times 2\cdot3026 \log \frac{OM}{r},$$

using an ordinary book of logarithms.

The numbers in this table when plotted as the co-ordinates of points on squared paper, and the points thus found being joined (Fig. 14), the curve *a a* is found to be the line of force required; the upper and lower parts of it being omitted in the figure.

By displacing this line vertically we get all the other lines of force shown in the diagram. One such curve being drawn and a template cut from it in cardboard, the whole series may readily be drawn by displacing the template vertically.

Equipotential Surfaces.

We have seen that the surfaces of equal pressure are everywhere at right angles to lines of force. If then r and y are the co-ordinates of a point on the line in which an equal pressure (equipotential) surface cuts a vertical plane through the axis, the tangent of the inclination of this line must be equal to minus the co-tangent of the inclination of the line of force at the point, and hence

$$\frac{dr}{dy} = \frac{g}{a^2 r};$$

or

$$\frac{a^2}{g} r.dr = dy.$$

The integral of this is

$$(4) \quad y = \frac{a^2 r^2}{2g} + C,$$

where C is some constant.

This equation belongs to a parabola, and the surfaces of equal pressure are paraboloids of revolution with their vertices downwards.

In Fig. 14, A A and B B show the sections of these surfaces of equal pressure calculated from (4) on the assumption of a speed of half a revolution per second. The parabola A A is drawn by making $y = 0$ when $r = 0$, hence $C = 0$, and giving to r the values below.

r	4	3	2	1	0·5	0
y	2·452	1·38	0·613	0·153	0·038	0

Vertical Level Surfaces.

Since centrifugal force acts radially, the equipotential surfaces for it will be concentric cylinders with the axis of rotation as axis (Figs. 15 and 16). The centrifugal force acting on 1 lb. of the stuff is $\frac{a^2 r}{g}$ where a is the angular velocity. Force × distance = work done, hence

$$\frac{a^2 r}{g} \times \delta r = \delta V$$

$$- \delta P = w \, \delta V = \frac{w r a^2}{g} . dr.$$

We may assume w constant for water, hence integrating

$$(a) \quad P = \frac{a^2 r^2}{2g} w + \text{a constant.}$$

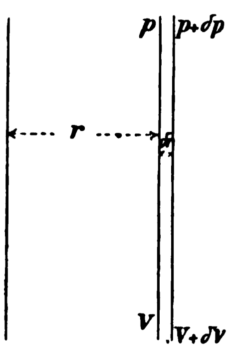


FIG. 15.

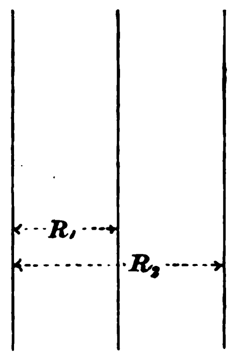


FIG. 16.

If r or a is large enough we may neglect gravity.
Let $P = P_1$ when $r = R_1$ (Fig. 16)

$$\therefore \quad P_1 = \frac{w a^2 R_1^2}{2g} + \text{a constant.}$$

$$\therefore \quad \text{the constant} = P_1 - \frac{w a^2 R_1^2}{2g},$$

and equation (a) becomes

$$(\beta) \quad P = \frac{w a^2 r^2}{2g} + P_1 - \frac{w a^2 R_1^2}{2g},$$

or

$$P - P_1 = \frac{w a^2}{2g} (r^2 - R_1^2).$$

Similarly

$$P - P_2 = \frac{w a^2}{2g} (r^2 - R_2^2)$$

$$(\gamma) \quad \therefore P_2 - P_1 = \frac{w\,a^2}{2\,g}(R_2^2 - R_1^2).$$

When at the axis $R_1 = 0$, $P_1 = 0$, and from equation (β)

$$\therefore \quad P = \frac{w\,a^2\,r^2}{2\,g} = w \times \text{head},$$

since

$$a^2\,r^2 = v^2 \text{ and } \frac{v^2}{2\,g} = \text{head } h, \text{ due to velocity } v.$$

Inside a centrifugal pump a mass of water is made to rotate in the above way, and if we neglect the fact that the water is really moving radially, or if we suppose that the pump is merely used to create as great a pressure as possible without any water flowing, and if we neglect frictional resistances, we can calculate from this rule what is the difference of pressure at the inner and outer circumferences of the revolving part of the pump.

We shall see afterwards that this is not the total difference of pressure available in a centrifugal pump, because there is always a space outside the inner wheel in which the rotation is not of the above kind, but in which there is a further gain of pressure.

The importance of having a space outside the inner wheel was first shown by the late Professor James Thomson, and the enlargement of this space constitutes the basis of his patent.

IV.

MOTIONS OF FLUIDS.

1. In the moving fluids we may have *streams* which are moving masses of water, completely or incompletely bounded by solids. When the solid boundary is complete, the water moves in a pipe; if the upper part of the boundary is incomplete, the water moves in a channel or canal.

2. If the stream of fluid considered is bounded laterally by a differently moving fluid of the same kind, the portion considered is called a *current*.

3. A *jet* is a stream bounded all round by a fluid of a different kind.

4. If the particles of the mass of fluid considered move spirally or circularly we have *vortex* or *eddy* motion.

5. In a stream we may consider the particles as moving along definite paths in space. A chain of particles following each other along a definite path in space may be called a fluid filament or *elementary stream.*

The actual motions of the particles of water in any given case are usually very complex. Simpler modes of motion are usually assumed in order to simplify our calculations, but the result in many cases does not agree very well with experiment.

Thus we can study the motion as belonging to one or more of the following classes, viz. :—

Plane layer motion, which is one of the simplest, in which particles in a plane are supposed to move so as to remain in a plane during motion, though the plane may not remain parallel to its initial position.

Laminar motion is also comparatively simple. We imagine the fluid divided into thin laminæ which slide on each other, as in the case where the velocity is not all the same across a section such as that of a river.

Stream line motion.—In the laminar motion all the particles in one lamina are supposed to have the same velocity. But the cross-section of a stream may be supposed divided into indefinitely small areas, each being the cross-section of a *fluid filament* or *stream line.*

If the motion is steady these stream lines have fixed positions in space. Like the lines of force used in magnetic and electric theory, they are imaginary, but very convenient, lines for defining and assigning the motions of fluids. A number of these lines, enclosing a mass of moving fluid, form a *stream tube.*

The actual velocity in a river, say, at any point is not constant, but the average velocity for 5 or 10 minutes may be (usually is) nearly constant, and may be used in calculations instead of the actual velocity, which is *variable.*

The fundamental law is

$$Q = A\,V \text{ (V being the average velocity),}$$

or

$$Q = \int v\,d\mathrm{A}$$

if we take the actual velocity at a given point; also if the flow is continuous

$$\frac{V_1}{V_2} = \frac{A_2}{A_1},$$

or

$$V_1 A_1 = V_2 A_2 = Q,$$

where

Q = the volume passing in unit time
A = the area of the section normal to direction of velocity.

Meaning of the Term "Head" in Hydraulics.

Head is an old millwright's term, meaning the vertical height through which a mass of water descended in actuating a hydraulic machine.

If we have an orifice of area a, covered by a lid or valve, the intensity of pressure there being P, then we know that $P \propto h$; or $P = 62 \cdot 4\, h$ in the case of water, P being the pressure per square foot and h the head in feet. This may be written, $p = \frac{h}{2 \cdot 3}$ as proved at page 18. Now if $\frac{P}{62 \cdot 4} = h$, h may here be termed the *pressure head* at the orifice.

Similarly, in the case of water issuing freely from an orifice,

$$v^2 = 2\, g\, h \text{ very nearly}$$

or

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

and h here may be called the "*velocity head*," meaning the head necessary to give, in a freely issuing fluid jet, the velocity v. v is not really $= \sqrt{2\, g\, h}$, but is about 0·97 of it in most cases.

Then again, part of the energy of a given mass of water is usually wasted in passing along a given length of pipe or channel. This waste may be expressed in feet of water and called "*friction head*" or "*head wasted by friction.*" Thus if 1 lb. of water loses 4 ft.-lbs. of energy in passing along a given length of pipe, then it loses energy equal to that of the 1 lb. descending through 4 feet, and hence loses 4 feet of "head" by friction. Hence the rule:—"*The loss of energy of* 1 *lb.* (*expressed in ft.-lbs.*) *is the loss of head in feet of water.*"

Friction of Water at Different Velocities.

A perfect fluid cannot exert tangential force or stress. Actual fluids, with which we have to deal, *do* exert tangential forces; for instance, water flowing through a pipe tends to drag the pipe along with it, on account of friction. In all actual fluids there is viscosity or internal friction, but if the relative motion is only slow enough it makes little difference whether the fluid is viscous or not.

Ordinary fluids will change in shape under the action of a force, however small, if you only give time enough for the change to take place, and the rate of change of shape under a given force is a measure of the viscosity.

When a fluid flows between two infinite parallel plane surfaces, it is not known with certainty whether the particles very near the surfaces move or not; probably the velocity is infinitely small at an infinitely short distance from the surface. For instance, the velocity at different points in the section of a river has been ascertained with some degree of accuracy.

A Commission of the United States Government found from very exhaustive experiments, that in a longitudinal vertical section of a river the velocities, if represented by horizontal lines, formed the abscissæ of a parabola with its axis parallel to the surface, and passing through the point of maximum velocity, which is situated

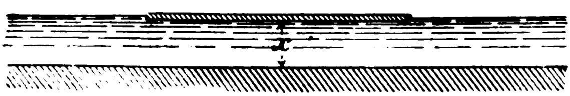


FIG. 17.

at about 0·3 of the depth below the surface. An up-stream wind increases, and a down-stream wind diminishes the depth of this point.

The velocities in a horizontal section also follow a parabolic law, the vertex of the curve being, as before, at the point of greatest velocity.

This and most other things in hydraulics can only be settled by experiment; the student will do well to distrust all laws or formulæ which have not received experimental verification.

The above assumption in regard to the particles touching the solid surface being at rest, involves that of a shear strain of the fluid.

Thus, if a plane surface be moved through a liquid like water, as in Fig. 17, neglecting the effect produced by the ends, if the wetted area be, A the force necessary to keep up a low velocity v is proportional to $\frac{A\,v}{x}$.

Mr. William Froude, Colonel Beaufoy, and others made many experiments with plates having sharpened edges, which were drawn through water in a long tank at different velocities, the force necessary to thus move them being observed. Eliminating, as far as possible, the end effects, the force R was connected with V, etc., by a law of the kind, $R = \mu AV^n$. This law should, however, be used with caution, as it is discontinuous.

Using Froude's results, we find that μ has the value 0·0032 for clean varnished surfaces, and 0·00456 for medium sand-paper, A being in square feet, V in feet per second. R is in pounds, and n is 1·85 for smooth surfaces, but 1·9 to 2·1 for rough surfaces. It might be thought that a result nearly correct would be obtained by taking $n = 2$, since the actual values are so close to that number, but a trial will show the student that using $n = 2$ for smooth surfaces makes the coefficient μ *double* of its actual value.

The law is usually put in the following form (assuming velocity and roughness sufficient to give index 2):

$$R = f w A \frac{v^2}{2g},$$

where

R is the total frictional resistance
w is the specific weight of the fluid
and f is the (so-called) coefficient of friction.

The work absorbed by frictional resistance

$$= Rv = f w A \frac{v^3}{2g}.$$

VALUES OF f (COEFFICIENT OF FRICTION) FOR LARGE SURFACES MOVING IN A VERY LARGE MASS OF WATER.

Nature of Surface.	f
New well-painted iron plate	0·00489
Planed and painted plank	0·0035
Surface of iron ships	0·00362
Varnished surface	0·00258
Fine sand surface	0·00418
Coarse sand surface	0·00503

Professor Unwin carried out very important experiments, by causing discs of different kinds to rotate in water, and measuring the tendency of the containing vessel to follow the disc. He obtained in this way results very similar to those of Froude. Professor Perry, for a similar purpose, used the apparatus shown in Fig. 18, where a hollow cylinder F, supported by a wire and capable of moving with a motion of rotation round the wire as axis, dips into water or other liquid contained in the annular space between D and E. The vessel D D E E was rotated at different speeds, and the amount of torsion of the suspending wire, showing the moment necessary to balance the tendency of the suspended cylinder to rotate, was observed in each case. For very low speeds this moment (or F) seemed to be proportional to the velocity, whilst for higher speeds it was nearly proportional to the square of velocity, there being a want of continuity in the law. Many experiments with oils at various temperatures were also made. Values of log F and log V being plotted on squared paper, gave the lines shown in Fig. 19; the first, being inclined to the axis at 45°, shows $F \propto V$; the second nearly agrees with $F \propto V^2$.

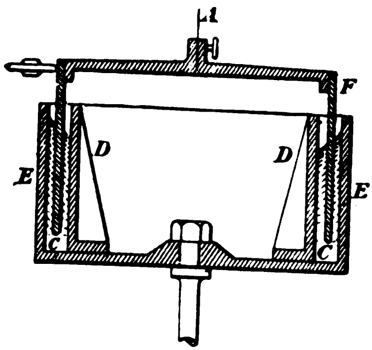


FIG. 18.

Professor Osborne Reynolds has made probably the most careful experiments on this point yet completed. He caused water to flow through glass tubes at different velocities. The tubes were about 4½ feet long and fitted with bell-shaped mouthpieces *m*, *m*, *m* (Fig. 20). Water flowed through the tubes from a tank, the head being varied at will.

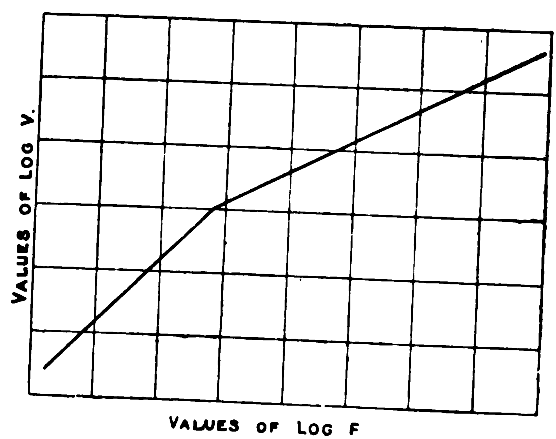


FIG. 19.

A little aniline dye was introduced into each by a pipette *s*. It was found that up to a certain velocity the coloured band extended uniformly along the tube, as at (*a*), but as the velocity was increased, at a certain velocity the band of colour became disturbed, as sho-

(*b*). When examined by an electric spark, the colour band was found to have become broken up into eddies, as at (*c*). The sudden change in the law is very clearly shown by plotting log F and log V as already explained, this method being due to Professor Reynolds.

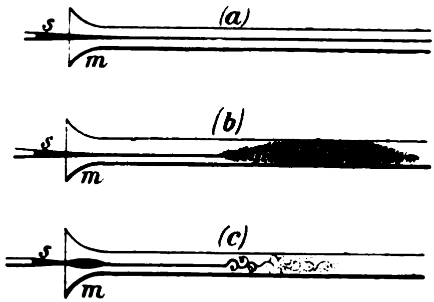


FIG. 20.

The results seem to point to the dissipation of energy in the formation of eddies at this "critical" velocity, when a change occurs in the *law* of flow, just as in the case in which a sudden change in the *direction* of flow is produced.

Professor Reynolds found that the critical velocity at which this sudden change in the index of V takes place depends upon the temperature of the liquid, being lower for higher temperatures.* His results give the following law:

$$A \frac{D^3}{P^2} i = \left(B \frac{D}{P} v\right)^n$$

or

$$(a) \quad i = \frac{B^n \times P^{2-n} \times D^{n-3} \times v^n}{A},$$

where A and B are coefficients, D is the diameter of the pipe, i the resistance per unit length of pipe. If metres and degrees Centigrade are employed,

$$A = 67 \cdot 7 \times 10^6, \qquad B = 396,$$

and

$$P = 1 \div (1 + 0 \cdot 0336\, t + 0 \cdot 000221\, t^2),$$

t being the temperature.

Also, the critical velocity v_c is given by the rule

$$(\beta) \quad v_c = \frac{1}{278} \times \frac{P}{D}.$$

* Dr. Coker's experiments on the whole confirm this, and also show that increase of pressure adds to the stability of flow, increasing the critical velocity.

The index n is 1 up to the critical velocity, afterwards it is 1·722 for lead pipes, 1·7 for the smoothest pipes, reaching 2 when the pipes were roughest.

Professor Reynolds' Rule in English Units.

Professor Osborne Reynolds' results are expressed by him in the formula (in metres and degrees Centigrade already given);

$$i = \frac{B^n}{A} \times P^{2-n} \times \frac{v^n}{D^{3-n}},$$

i being the resistance per unit length of pipe expressed in (weight of) cubic units of water. i is therefore the slope $= \frac{h}{L}$, and independent of the units chosen.

To change to British units.

Let $q = 3{\cdot}2809$, the number of feet in 1 metre.

$$d \text{ in feet is given by } D = \frac{d}{q}.$$

$$v = \frac{V}{q},$$

V being in feet per second,

$$A = 67{\cdot}7 \times 10^6$$

$$B = 396.$$

$$\therefore \quad \frac{h}{L} = \frac{B^n P^{2-n}}{A} \times \frac{1}{\left(\frac{d}{q}\right)^{3-n}} \left(\frac{V}{q}\right)^n$$

$$h = \left(\frac{B^n \times P^{2-n}}{A} \times q^{3-2n}\right) \frac{V^n}{d^{3-n}} \times L$$

$$h = 0{\cdot}000706 \frac{V^2}{d} \times L$$

when $n = 2$; h being the head, in feet, lost in L feet of pipe.

Compare this with D'Arcy's rule,

$$h = \frac{4fL}{d} \cdot \frac{V^2}{2g}.$$

$$f = 0{\cdot}005\left(1 + \frac{1}{12d}\right).$$

Some of Professor Reynolds' pipes were about 1 inch in diameter;

$$\therefore \quad f = 0{\cdot}005\left(1 + \frac{1}{12 \times \frac{1}{12}}\right)$$

$$= 0{\cdot}005 \times 2 = 0{\cdot}01.$$

$$\frac{4f}{2g} = \frac{4 \times 0{\cdot}01}{64{\cdot}4} = 0{\cdot}000621.$$

Hence D'Arcy's rule gives

$$h = 0{\cdot}000621 \times \frac{V^2}{d} \times L.$$

There is then a close agreement when we remember that D'Arcy's experiments were conducted only with pipes of larger diameter, and hence his coefficient may not be correct for small pipes such as Professor Reynolds used. Also Professor Reynolds' index is in most cases less than 2; hence his coefficient must be greater than D'Arcy's to give the same result.

Note that when $n = 2$, P^{2-n} becomes 1, or temperature may be neglected.

Mr. Mair has made experiments with a 1½-inch brass pipe, giving results agreeing with the following formula:

$$\frac{h}{l} = 0{\cdot}00031\ (1 - 0{\cdot}00215\ t)\ \frac{v^n}{d^{3-n}}.$$

These and other experiments have been carried out at comparatively low pressures. No complete set of results for the friction of water at high pressures has been obtained, but it is generally assumed that the friction is independent of pressure, and that the ordinary rules for low pressures are applicable even for such pressures as we have in hydraulic power mains. Observations of the pressures in mains at different points seem to confirm this assumption.

V.

FLOW OF WATER THROUGH ORIFICES.

Coefficients of Discharge.

When water flows from an orifice, say in the vertical side of a vessel which is of large dimensions compared with the size of the orifice: if the level of the water be kept at a given height—h' feet—above the centre of the jet, the velocity of the issuing water, if there were absolutely no physical resistance to efflux, would be that of a stone which has fallen freely through the height h', or

$$v = \sqrt{2gh'} = 8 \cdot 02 \sqrt{h'},$$

g being 32·2.

By experiments with a jet directed vertically upwards, it has been found that the actual velocity is not quite so great as this, varying from 0·959 of this for 0·66 feet head to 0·994 of it for 55 feet head, the average velocity being only about 0·97 of it for well-formed orifices.

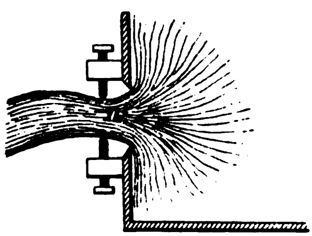


Fig. 21.

Now the discharge Q should follow the rule $Q = Av$, A being the area of the orifice; but by gauging the actual flow it is found to be not much over half this in many cases, on account of the contraction of the jet at a point such as n, Figs. 21 and 23, at which the stream lines are most nearly parallel. It is at a point such as this only that our rule $Q = Av$ should be applied. The ratio of the area of the jet at this place to that of the orifice, is about 0·64 for small sharp-edged orifices; hence for such cases

$$Q = 0 \cdot 97 \times 0 \cdot 64 \, A \sqrt{2gh'} = 0 \cdot 62 \, A \sqrt{2gh'}.$$

The general rule is, $Q = c \times A \sqrt{2gh}$ where c is the coefficient of discharge.

Many experiments have been made to determine this coefficient for particular shapes of orifices, and at different velocities of flow.

Divergent Mouthpiece.

In the case of the divergent mouthpiece shown in Fig. 22, there will be a certain limiting velocity. As the velocity at section P Q is greater than that at R S, the pressure at the latter is less than at the former, and when the pressure at R S becomes less than atmospheric, the stream disengages itself from the mouthpiece, and the latter no longer runs full.

Let a, v_1 and P_1 be the section velocity and pressure at P Q.

Let A, v_2, P_a be the same quantities at section R S.

$$\frac{P_1}{w} = \frac{P_a}{w} - \frac{(v_1 - v_2)^2}{2g}$$ (see page 67).

Suppose $\frac{A}{a} = m,$

then $v_1 = v_2 m.$

$$\frac{P_1}{w} = \frac{P_a}{w} - \frac{v_1^2}{2g}(m^2 - 1)$$

$$= \frac{P_a}{w} - (m^2 - 1) h.$$

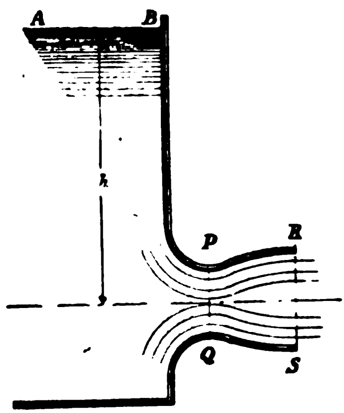


Fig. 22.

Hence $\frac{P_1}{w}$ will be zero or negative respectively, if

$$\frac{A}{a} \text{ is greater than or equal to } \sqrt{\frac{h + \frac{P_a}{w}}{h}} = \sqrt{1 + \frac{P_a}{wh}}.$$

If $\frac{P_a}{w}$ (the pressure head at R S) be put = 34 feet,

the conditions are that if

$$\frac{A}{a} \text{ is greater than or equal to } \sqrt{\frac{h + 34}{h}}$$

$\frac{P_1}{w}$ will be zero or negative.

In practice there will be an interruption of the full-bore flow when the ratio $\frac{A}{a}$ is a little less than that given by putting the sign of equality into the above equation, owing to the disengagement of air from the water. Taking, however, the theoretical limit as true, the maximum discharge from a mouthpiece of this kind is

$$Q = a\sqrt{2g\left(h + \frac{P_a}{w}\right)};$$

pressures being in lbs. per square foot, areas in square feet, and $w =$ weight of a cubic foot of the liquid, Q will be in cubic feet per second.

In the case of the re-entrant mouthpiece of Borda, shown at B, Fig. 23, Mr. Froude has shown experimentally that the coefficient of contraction, as found from theoretic considerations, is correct. The

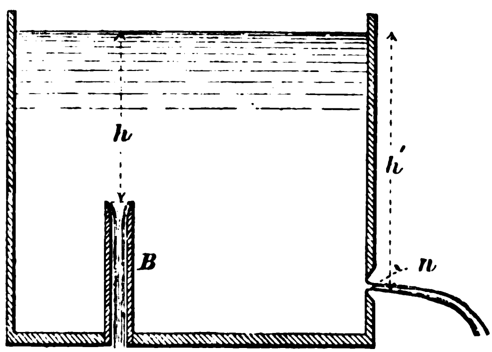


FIG. 23.

force on a valve closing the orifice is $w\,h\,A$. This should be equal to the momentum generated per second in the issuing water, which is $m\,v$, or $\frac{Q}{g}\,w\,v$. But $Q = a\,v$ where a is the area of the contracted jet. The momentum is therefore

$$\frac{a\,v^2\,w}{g} = 2\,a\,h\,w,$$

assuming $v^2 = 2\,g\,h$. Hence

$$w\,h\,A = 2\,w\,h\,a,$$

or

$$A = 2\,a,$$

the coefficient of contraction being $\frac{1}{2}$, and the coefficient of discharge is also often taken as about $\frac{1}{2}$.

In sharp-edged orifices it diminishes slightly with increase of head, and also with increase of area of orifice, being more nearly independent of *h* in the case of large orifices.

For circular orifices it varies from 0·64 to 0·59 (*h* 1 foot to 100 feet, diameter of orifice 0·02 to 1 foot), square orifices giving almost the same values, and rectangular orifices 0·63 to 0·6.

For well-shaped rounded orifices the value varies from 0·64 to 1, depending on the closeness of approximation to the natural shape of the stream.

Variation of the Coefficient of Discharge.

The way in which the coefficient varies is best shown by curves. Those in Fig. 24 have been plotted from the experimental results of Mr. J. Hamilton Smith and others. Curve A shows the values of the

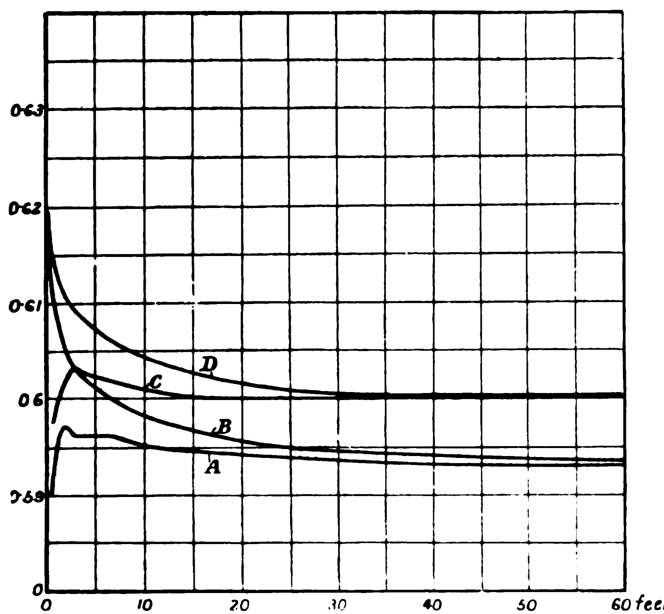


Fig. 24.

coefficient for a sharp-edged circular orifice 1 foot in diameter, whilst B gives it for a similar orifice 0·1 foot in diameter. It will be noticed that in these two cases the variation is in opposite directions for the smaller heads up to 10 feet. Curve C is for a sharp-edged square orifice of 1 foot side, whilst D gives the coefficient for a square orifice of 0·1 foot side. It will be noticed that the coefficient varies somewhat rapidly with heads of less than 5 feet, also that on the whole it decreases as the head increases, and increases (in the case of circular orifices) as the size of the orifice is taken smaller and smaller, but is practically constant for a particular orifice for heads of over 60 feet. For a submerged orifice, if the effective head be taken as "head" in plotting, it will be found that the coefficient is smaller than for the same orifice with free discharge, being about 0·599 for a circular

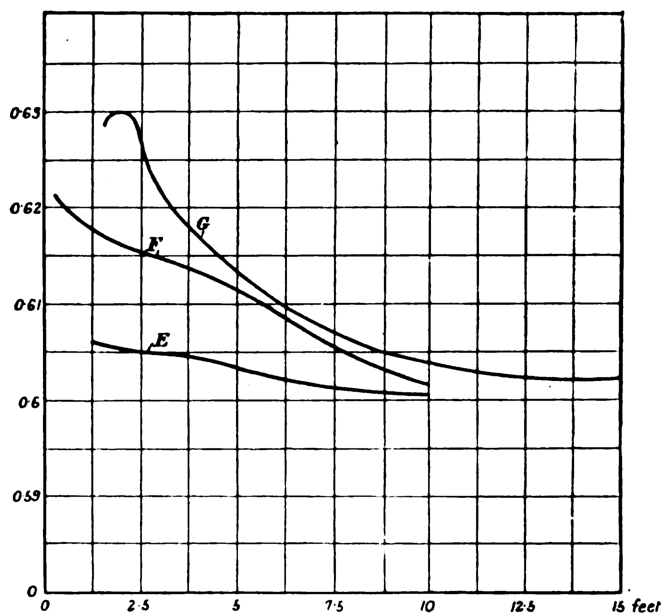


FIG. 25.

orifice 0·1 foot in diameter. With the same effective head depth of submersion does not appear to affect the result, but this has not been very fully tried.

In Fig. 25 the values of the coefficient for various rectangular orifices are shown. Curve E is for a rectangular orifice 1 foot wide and 1 foot deep, in other words for a square orifice, and is given here for purposes of comparison. F is for a rectangular orifice 0·5 foot wide and 1 foot deep, and G is for a rectangular orifice 1 foot wide and 2 feet deep. The coefficient is greater as the orifice departs more and more from the square in shape, and is (contrary to that in the case of circular orifices) greater as the area of the orifice is taken greater and greater, the ratio of two adjacent sides remaining the same, but the coefficient becomes nearly constant for heads of over 20 feet.

It will be noticed that the coefficient is least for a large circular orifice; then a small circular orifice comes next in order of value, the square next, and the rectangle greatest of all. Hexagonal and octagonal orifices approximate closely to the circular as regards value of the coefficient. The triangular shape of orifice has been tried; its value varies from about 0·631 for a head of 1 foot to 0·605 for a head of 20 feet, the triangle being equilateral.

This orifice is, however, of little practical importance.

The curves in Figs. 24 and 25 show the variation in c for various sharp-edged orifices in a thin vertical wall when h is kept constant during the determination of each value of c. In accurate calculations, c should not be assumed constant for ratios of $\frac{h}{d}$ less than 10, where d is the diameter of the orifice. Experiments with different liquids have shown that c for thick oil is 0·72, for water 0·628, and for mercury 0·595, with a head of 3 feet and an orifice 0·02 foot diameter. In fact, the more viscous the liquid, the greater is the value of c. The value, however, becomes more nearly the same for all liquids as the head becomes greater, and at heads of, say, 100 feet, with an orifice such as that referred to above, all liquids would probably have practically the same coefficient of discharge.

Discharge from Tanks and Reservoirs.

Bearing in mind the limitations necessary in using c where h is *variable* or where $\frac{h}{d}$ is small, one may obtain, approximately, the time taken to empty a given tank or reservoir through an orifice, or to equalise the water-level in two adjoining chambers.

Let the horizontal section of the tank be constant and equal to A square feet; then if h remained constant, owing to the tank having a constant supply, the time taken to discharge a volume *equal* to that of the tank down to the centre of the orifice would be

$$\frac{A\,h}{Q} = \frac{A\,h}{c\,a\,v}$$

$$= \frac{A\,h}{8\cdot02\,c\,a\,h^{\frac{1}{2}}} \quad (\text{since } Q = c\,a\,v \text{ and } v = 8\cdot02\,\sqrt{h})$$

$$\therefore \quad Q = \frac{A\,h^{\frac{1}{2}}}{8 \cdot 02\, c\, a}$$

where Q is the volume, in cubic feet, discharged per second and *a* is the area of the orifice in square feet. Applying this reasoning to the case where no water enters, if the time taken to discharge an elementary layer *d h* thick be *d t*, we have

$$d\,t = \frac{A\,d\,h}{8 \cdot 02\, c\, a\, h^{\frac{1}{2}}},$$

or

$$t = \frac{A}{8 \cdot 02\, c\, a} \int_0^{h} h^{-\frac{1}{2}}\, d\,h = \frac{A\,h^{\frac{1}{2}}}{4 \cdot 01\, c\, a}.$$

This is evidently just twice the time taken to discharge the same volume with a *constant head h*. If we take $c = 0 \cdot 6$, the time with constant head is $\frac{0 \cdot 208\, A\, h^{\frac{1}{2}}}{a}$ seconds, and if no water enters the time is twice as long.

If the reservoir be not vertical sided, the top area must be multiplied by a constant to obtain the area at any given depth. Thus for a wedge-shaped reservoir the area $= A\,\frac{h}{h'}$, and for a pyramidal reservoir $A\left(\frac{h}{h'}\right)^2$, at a section *h* from apex where the orifice is supposed to be situated, *h'* being the head at start. Thus for the former

$$d\,t = \frac{A\,\frac{h}{h'}\,d\,h}{8 \cdot 02 \times c \times a \times h^{\frac{1}{2}}} = \frac{A\,h^{\frac{1}{2}}\,d\,h}{8 \cdot 02 \times c \times a \times h'};$$

whence

$$t = \frac{A}{8 \cdot 02\, c\, a\, h'} \int_0^{h'} h^{\frac{1}{2}}\, d\,h = \frac{A\,(h')^{\frac{1}{2}}}{12 \cdot 03 \times c \times a}.$$

The time, therefore, required to empty such a reservoir when no water enters it is $\frac{4}{3}$ of the time taken to discharge the same volume if the head be kept constant. For a pyramidal reservoir the time if no water enters is $\frac{6}{5}$ of that taken to pass the same volume with constant head.

Tanks or Chambers Communicating.

The time taken to *equalise the water-levels* in two adjoining tanks or reservoirs, which can be placed in communication through an orifice, may be obtained in the same way, supposing the rising water in the receiving tank to act with a back-pressure in retarding the flow. If the tanks are alike in size and shape, the time taken to equalise levels is found by Rankine's rule, which is as follows: "The time taken to equalise the level of water in two adjoining basins with vertical sides—such as lock chambers or canals—when a communication is opened between them under water, is the same as that required to empty a vertical-sided reservoir of a volume equal to the volume of water *transferred* between the chambers and of a depth equal to their greatest difference of level." If, for instance, the chambers are equal, the time required to equalise levels is that necessary to discharge a volume equal to the full of one of them *with a constant head.* It must be borne in mind that for a submerged orifice, c is less than with free discharge, but the difference between its value and the normal value diminishes as the head increases, and for large orifices of say 1 foot square, the difference is inappreciable except for very small heads.

Examples.

(1) Taking $c = 0 \cdot 62$, find time taken to fill a tank holding $\frac{1}{2}$ ton of water through a 1-inch sharp-edged circular orifice, the head over the orifice being maintained at 6 feet. *Ans.* 4·5 minutes.

(2) A rectangular sharp-edged orifice, 2 inches deep and 1 inch wide, in the side of a tank is 10 feet below the surface of the water in the tank, the level of water being being kept constant. Find the rate of discharge in gallons per hour through the orifice. $c = 0 \cdot 61$.

Ans. 4831 gallons per hour.

(3) A rectangular chamber, 120 feet square in section and with vertical sides, is filled with water to a depth of 15 feet. If this water is allowed to flow out through a rectangular orifice 2 feet by 1 foot, the centre of the orifice being on a level with the floor of the chamber; find how long it will take to empty the chamber, the coefficient of discharge being taken as 0·6. Also if the orifice discharges into a second chamber similar to the first, find how long it will take to equalize the levels of the water surface in the two chambers, the coefficient being taken as 0·59 in the latter case.

Ans. 3 hrs. 13 min. 19 sec.; 1 hr. 38 min. 13 sec.

VI.

FLOW OF WATER IN PIPES AND CHANNELS.

The Hydraulic Gradient.

One of the first things to decide in questions relating to the flow of water in pipes, is the "hydraulic gradient." An example or two will best explain this.

Suppose the water in a reservoir to stand at a constant height *h*, shown in Fig. 26, then if a horizontal pipe were fitted, as shown, with vertical tubes, when the pipe is closed at *a* the water in these tubes would stand to the height *h* if the tubes were long enough. When

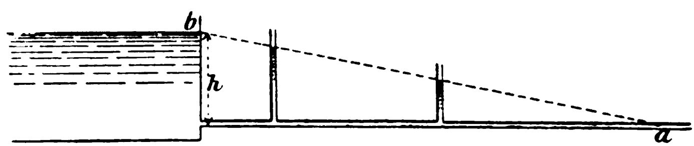


Fig. 26.

the pipe is opened at *a* and water flows through it with a steady velocity, the height of the free surface of the water in the tubes will be that of the dotted line *a b*. This line is called the "hydraulic grade line," and its slope the "hydraulic gradient" or virtual slope. The height of the water in each tube shows the hydraulic or

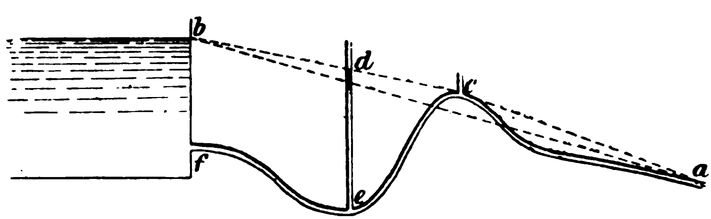


Fig. 27.

"pressure" head, by American writers called the "piezometric" height, to distinguish it from the "velocity" head or from the hydrostatic head shown when there is no flow.

Many interesting cases might be taken up. For instance, if a point *c* in the pipe *f c a* (Fig. 27) rises above the straight line *a b*, the water in *d e* will now stand to the height *d*, the flow at *c* must be calculated from the hydraulic gradient *b c*, and the pipe *c a* having a steeper hydraulic gradient than *b c* will, if of the same diameter and roughness as the rest, not have sufficient flow in it to keep it full, but will act as a trough.

We might say that if the pipe were air-tight the pressure at *c* would be less than atmospheric by an amount represented by the height from *c* to *a b* measured downwards; but in practice air will accumulate and spoil the siphon action.

Hence no point in a pipe should rise above the hydraulic grade line, if it is to run full.

The water in a tube at *c* will not stand above the pipe, and if a branch pipe be taken off here, a valve closing its end will sustain about atmospheric pressure.

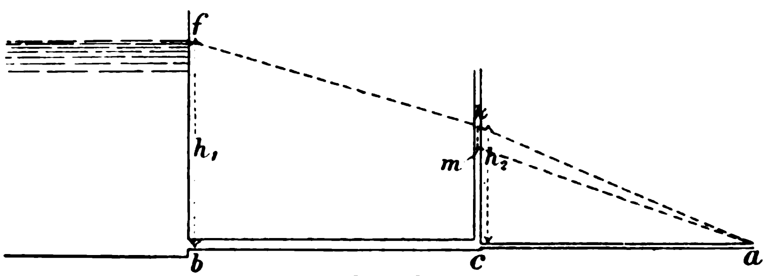


FIG. 28.

If a pipe varies in section the hydraulic gradient will be correspondingly affected. This will be seen in Fig. 28, where a pipe of large diameter joins one of smaller diameter.

It is evident that the gradient must be steeper for the small pipe than for the large one, the discharge of both being the same.

HYDRAULIC GRADIENT FOR PIPES OF VARYING DIAMETERS.

The consideration which gives us the height required here (Fig. 28) is that the flow through all portions of the compound pipe must be the same.

One example will show how the matter may be taken up. In Fig. 28, let $h_1 = 50$ feet, $bc = 500$ feet, $ca = 500$ feet, diameter of $bc = 1$ foot, that of $ca = \frac{1}{2}$ foot; find the hydraulic gradient *f k a*. Neglecting the head wasted at *c*, D'Arcy's rules (see page 43) give $h = \lambda \frac{v^2 L}{d}$, and $Q = \frac{\pi}{4} d^2 v$ whence, eliminating v,

$$Q^2 = \frac{d^5 h \times 0 \cdot 616}{L \times \lambda},$$

and Q, hence Q^2 is the same for both

$$\lambda \text{ for 6-inch pipe} = 0\cdot00036,$$
$$\lambda \text{ for 1-foot pipe} = 0\cdot000336.$$

Hence

$$\frac{1^5 \times (50 - h_2) \times 0\cdot616}{500 \times 0\cdot000336} = \frac{(0\cdot5)^5 \times h_2 \times 0\cdot616}{500 \times 0\cdot00036},$$

from which $h_2 = 48\cdot57$ feet.

Example 1.—At a point in this pipe where the grade line is 20·289 feet above the pipe, a horizontal branch pipe 3 inches in diameter and 1000 feet long is inserted. Neglecting any change thereby produced in the hydraulic gradient, find the population that this pipe will supply at the rate of 20 gallons per head per day.

$$Q = \sqrt{\frac{(0\cdot25)^5 \times 20\cdot289 \times 0\cdot616}{1000 \times 0\cdot000414}} = 0\cdot1716 \text{ cubic feet per second,}$$

or, since there are $6\frac{1}{4}$ gallons in one cubic foot, the discharge is 1·07 gallons per second, that is, 92,851·2 gallons in twenty-four hours, which, at 20 gallons per head, will supply 4642 persons.

In these examples the total head is supposed to be utilised in overcoming friction, but the head necessary to give the assumed velocity is neglected. For even 2 feet per second it is only $\frac{1}{16}$ foot. Taking friction at *c* into account, the gradient will really assume some such shape as *f k m a*; the distance *k m* can readily be found from the data given on page 58.

Example 2.—Calculate the proper diameter for a pipe to supply 100,000 inhabitants at the same rate, the distance being 5 miles and the slope of the hydraulic gradient $1\frac{1}{2}°$. (Here a likely value of λ must first be assumed, say that for 6-inch pipe.) *Ans.* 0·5001 foot.

In these examples the coefficient for smooth pipes has been taken, but if the flow is to be maintained when the pipe becomes encrusted, it is better to use that for rough pipes. The maximum velocity of flow in town mains should be from 2 to 7 feet per second.

It will be well to give now a few of the best authenticated rules usually adopted in making calculations relating to flow in pipes.

Rules for Flow of Water in Pipes.

In pipes the flow is often calculated on the assumption that the resistance due to friction is proportional to v^2.

D'Arcy's experiments are perhaps the most complete guide we have, and his formula based on the above assumption, with a properly varying coefficient, may be applied to a considerable range of velocities.

There is no very satisfactory formula for rough pipes, as the flow depends very much on the degree of roughness of the surface, but on the whole Tutton's deductions seem most consistent with theory and experiment.

The rules deduced by D'Arcy from a very complete and exhaustive series of experiments carried out at the Paris Waterworks, may be put simply thus.

The head wasted by friction is proportional to the length of pipe to the square of the velocity of flow, and is inversely proportional to the diameter of the pipe, or as a formula,

$$h \propto \frac{v^2 L}{d},$$

or

$$h = \lambda \frac{v^2 L}{d} \qquad . \quad . \quad . \quad . \qquad (a)$$

The rule is given by D'Arcy in a somewhat different form. Thus he found that the loss of energy of the water per pound (or loss of head) was $f \times \frac{4 L}{d}$ times its kinetic energy $\left(\frac{v^2}{2g}\right)$, where f is a coefficient sometimes called D'Arcy's coefficient of friction $= 0 \cdot 005\left(1 + \frac{1}{12 d}\right)$ for clean pipes, and $= 0 \cdot 01\left(1 + \frac{1}{12 d}\right)$ for encrusted pipes. This can be put in the simpler shape given in formula (a). Values of λ are given below calculated from D'Arcy's rules.

It must be borne in mind that D'Arcy's coefficient has been determined mainly from experiments on *smooth* pipes of somewhat small diameters. The formula for *rough* pipes is, however, a good deal used, and it seems reasonable to suppose that pipes of intermediate roughness should have a coefficient intermediate between $0 \cdot 005\left(1 + \frac{1}{12 d}\right)$ and $0 \cdot 01\left(1 + \frac{1}{12 d}\right)$, and thus we may adopt the the value of $f = 0 \cdot 0075\left(1 + \frac{1}{12 d}\right)$, though there is no experimental confirmation of this.

The following are some values of

$$\lambda = \left\{0 \cdot 005\left(1 + \frac{1}{12 d}\right)\frac{4}{64 \cdot 4}\right\}; \text{ etc.}$$

Cast-Iron Pipes (D'Arcy Formula).

Diameter of Pipe in feet = d in formula.	Diameter of Pipe, in inches.	Values of λ. Smooth Pipes.	Values of λ. Pipes of Medium Roughness.	Values of λ. Rough Pipes.
0·25	3	0·0004141	0·0006212	0·0008282
0·333	4	0·0003884	0·0005820	0·0007768
0·4166	5	0·00037402	0·00056103	0·00074804
0·5	6	0·0003623	0·0005434	0·0007246
0·5833	7	0·0003549	0·0005323	0·0007098
0·666	8	0·0003493	0·0005239	0·0006986
0·75	9	0·00034506	0·00051759	0·00069012
0·833	10	0·0003416	0·0005124	0·0006832
0·9166	11	0·0003391	0·0005086	0·0006782
1·0	12	0·0003363	0·0005044	0·0006726
1·25	15	0·0003312	0·0004968	0·0006624
1·5	18	0·0003278	0·0004917	0·0006556
1·75	21	0·0003253	0·0004879	0·0006506
2·0	24	0·0003234	0·0004851	0·0006468
3·0	36	0·0003192	0·0004788	0·0006384
4·0	48	0·0003171	0·0004756	0·0006342

Thus to find the frictional loss of head (in feet of water) in a pipe L feet long and d feet in diameter, the velocity of flow being v feet per second, *multiply the proper value of λ, found from the above table, by the length* L, *by the square of the velocity* v, *and divide by the diameter of the pipe d.*

To obtain D'Arcy's coefficient f (if it be required) multiply the corresponding value of λ by 16·1.

Example.—Find the loss of head due to friction in a pipe of medium roughness 1 mile long and 15 inches in diameter, the velocity of flow being 3·5 feet per second.

Here $\lambda = 0\cdot0004968$, $L = 5280$, $d = 1\cdot25$ and $v^2 = 12\cdot25$.

$$\therefore \text{Head wasted by friction} = \frac{0\cdot0004968 \times 5280 \times 12\cdot25}{1\cdot25} = 25\cdot7 \text{ feet of water.}$$

Many other formulæ are given, but D'Arcy's is the simplest and has been much used.

Weisbach's law, which is, in English measures (unit of length 1 foot),

$$h = \left(0{\cdot}0144 + \frac{0{\cdot}01716}{\sqrt{v}}\right) \frac{L}{d} \cdot \frac{v^2}{2g},$$

has also been a good deal used, and tables have been compiled from it by the late Professor James Thomson and Professor Fuller for pipes from 3 to 70 inches in diameter, and velocities from 2 to 7 feet per second.

Robinson and Thrupp's Formula.

A useful formula for small pipes is that deduced by Professor Robinson and Mr. Thrupp.

$$Q = \frac{d^{2{\cdot}6116}}{c\sqrt[n]{s}}$$

where

Q = discharge in cubic feet per second

d = diameter of pipe *in inches*

s = cosecant of inclination $= \frac{\text{length}}{\text{head}}$.

n and c are coefficients depending on the material and state of the pipe.

For new cast-iron .	$n = 2{\cdot}00$, $c = 13{\cdot}66$ (lowest)
or	$n = 2{\cdot}00$, $c = 15$ (average)
Well-cleaned old cast-iron . . .	$n = 1{\cdot}75$, $c = 11{\cdot}74$
Old encrusted iron .	$n = 2{\cdot}00$, $c = 22{\cdot}5$
Wrought-iron gas-piping	$n = 1{\cdot}90$, $c = 11{\cdot}5$
to	$n = 1{\cdot}85$, $c = 16{\cdot}6$
Wrought-iron pipes generally . . .	$n = 1{\cdot}85$, $c = 10{\cdot}7$ to $14{\cdot}5$
Lead pipes. . .	$n = 1{\cdot}74$ (average), $c = 10$ (average)
The formula is not applicable if . .	$s = 10{,}000$ and Q is less than $0{\cdot}006$
	$s = 1{,}000$,, ,, $0{\cdot}0018$
	$s = 100$,, ,, $0{\cdot}001$

TUTTON'S FORMULÆ.

Mr. Tutton (1896)* following the rational method of Professor Osborne Reynolds described at page 28, brought into neat and consistent shape the results of numerous experimental data available.

Reynolds' law may be put in the form

$$v = c\,m^x\,i^y,$$

where m is the "hydraulic mean depth," or radius (= cross-sectional area ÷ wetted perimeter), i the inclination $= \frac{h}{L}$.

Adopting the method described at page 28:

$$\text{since } v = c\,(m)^x\,(i)^y,$$

$$\log v = \log c + x \log (m) + y \log (i):$$

hence, if values of log v are plotted as ordinates and values of log i, as abscissæ for various values of m, a series of parallel straight lines is obtained making with the horizontal axis an angle $\tan^{-1} y$. Also by plotting log v and log m in the same way another series of parallel straight lines is obtained, making with the horizontal axis $\tan^{-1} x$, and the value of log v corresponding to log $m = 0$ is the log of c. Thus the constants c, x and y can be obtained. It has been found that $x + y = 1 \cdot 17$ nearly, in all cases, hence the rule may be put in the form $v = c\,(m)^x\,(i)^{1 \cdot 17 - x}$. In this way, using the results obtained by many observers, Mr. Tutton deduced the formulæ given below for pipes of various materials.

Wooden pipes	$v = 129\,(m)^{\cdot 66}\,(i)^{\cdot 51}$
New wrought-iron pipes . .	$v = 127$ to $165\,(m)^{\cdot 62}\,(i)^{\cdot 55}$
Ditto, asphalted	$v = 139$ to $188\,(m)^{\cdot 62}\,(i)^{\cdot 55}$
New cast-iron, and cement-lined pipes	$v = 126$ to $158\,(m)^{\cdot 66}\,(i)^{\cdot 51}$
Rusted or mud-coated iron pipes:	
(with *light* tuberculations) .	$v = 87$ to $132\,(m)^{\cdot 66}\,(i)^{\cdot 51}$
(„ *heavy* „) .	$v = 31$ to $80\,(m)^{\cdot 66}\,(i)^{\cdot 51}$
Glass pipes	$v = 141$ to $169\,(m)^{\cdot 61}\,(i)^{\cdot 56}$

The following are some values of c and x for pipes fulfilling the given conditions.

* The student will find this matter fully discussed, and the curves given, in Dr. Bovey's "Hydraulics," 2nd edition (1902).

Material.	c	x
Tin	189	0·59
Lead	168	0·59
Brass	165	0·61
Wrought iron	160	0·62
Wood (stave)	125	0·66
New cast iron	130	0·66
Lap-riveted wrought iron pipe .	100 to 115	0·66
Wrought iron, asphalted . .	170	0·62
Ordinary service-pipe . . .	104	0·66
Encrusted pipe	30 to 80	0·66
Brick conduit	91	0·65
Large ditto	110	0·65

The comparative constancy of x shows the accuracy of the method of deduction.

Flow of Water in Large Pipes.

In D'Arcy's experiments only comparatively small pipes were used, none being over 1 foot in diameter. It is therefore very doubtful whether his rules are applicable to large pipes; his assumption that the first power of the diameter only is to be used seems wrong. For rough pipes it is exceedingly difficult to obtain a formula which will give even approximately accurate results for varying degrees of roughness.

Hagan in 1854 suggested an empirical formula, $\frac{h}{l} = \frac{m\,v^r}{d^x}$, in which the three quantities m, r and x, representing the effects of the three principal causes of variation of resistance—viz. roughness, velocity, and diameter—were to be determined experimentally.

In a series of articles in 'Industries' for 1886, Professor Unwin gave, in curves, the results of a great number of experiments previously made by many observers, and deduced the values of the constants referred to. His method has the great merit of showing what variation in resistance is really due to each of the three factors, and the formula he gives is—

$$\frac{h}{l} = \frac{0·0004\,v^{1·87}}{d^{1·4}}$$

1 foot being the unit of length.

Mr. H. D. Pearsall has shown that this formula may be regarded as giving the resistance for ***all pipes of large diameter and in good condition***, rather than the more restricted application to riveted wrought-iron pipes which Professor Unwin suggests. The pipes varied from 0·9 foot to 4 feet in diameter. Many of the experiments from which the rule is deduced are described in detail by Mr. Hamilton Smith in his 'Hydraulics.'

The formula also agrees closely with results of subsequent experiments at Seville and Hoboken on pipes 20 inches and 21 inches in diameter.

This formula is only approximately true if the pipe be very smooth. Sufficient data for accurately determining the flow of water in large pipes are not yet available, but it is best to allow a margin for excessive friction, and to guard against repeating such an expensive mistake as that made at Newark (East Jersey Water Co.), U.S.A., where about 21 miles of riveted steel mains had to be duplicated, the flow being not more than 70 per cent. of that expected. The projecting lap joints and rivet heads caused considerable hydraulic resistance—probably nearly equal to that of rough pipes.

For very rough pipes the index of v is about 2, and that of d 1·1, the coefficient being 0·0007 ; but these numbers vary with every different class of pipe. The index of d is, however, always greater than 1, showing that an increase in diameter is of more importance in reducing frictional loss of head than D'Arcy's rules seem to indicate.

As a useful example, the diameter of pipe required in the above case may be calculated, the slope being about 2 feet per 1000, and flow necessary 77·37 cubic feet per second (50,000,000 gallons per twenty-four hours).

First, by Unwin's formula for rough pipes,

$$\frac{2}{1000} = \frac{0 \cdot 0007\, v^2}{d^{1 \cdot 1}};$$

also

$$Q = 0 \cdot 7854\, d^2 v,$$

or

$$v = \frac{77 \cdot 37}{0 \cdot 7854\, d^2};$$

whence

$$d^{5 \cdot 1} = 0 \cdot 35 \left(\frac{77 \cdot 37}{0 \cdot 7854}\right)^2,$$

or

$$d = 4 \cdot 923 \text{ feet.}$$

Second, by D'Arcy's formula for rough pipes,

$$h = \frac{\lambda\, v^2\, L}{d}.$$

λ for 4-feet pipe = 0·0006.

$$d^5 = 0·0006\left(\frac{77·37}{0·7854}\right)^2 \times 500.$$

$$d = 4·93 \text{ feet.}$$

By Tutton's formula,

$$v = 100\,(m)^{·66}\,(i)^{·51}$$

$$m = \pi\frac{d^2}{4} \div \pi\,d = \frac{d}{4}$$

$$i = \frac{2}{1000} \quad \text{and} \quad v = \frac{77·37}{0·7854\,d^2}.$$

$$\therefore \quad \frac{77·37}{0·7854\,d^2} = 100\left(\frac{d}{4}\right)^{·66} \times \left(\frac{2}{1000}\right)^{·51}$$

or

$$\frac{77·37}{0·7854} \times \frac{(1000)^{·51}}{2^{·51}} \times \frac{4^{·66}}{100} = d^{2+·66}$$

whence d = 4·617 feet.

By the simple formula proposed by Mr. E. Sherman Gould,

$$d = \sqrt[5]{\frac{Q^2}{H}},$$

where H is fall in 1000 feet of pipe.

$$d = \sqrt[5]{\frac{77·37^2}{2}}.$$

$$d = 4·96 \text{ feet.}$$

The actual diameter adopted was 4 feet, and the flow 34,000,000 gallons per twenty-four hours. If the diameter is proportional to $\sqrt[5]{Q^2}$, find the proper diameter. *Ans.* $4\sqrt[5]{\frac{50^2}{34^2}} = 4·668$ feet.

Flow of Water in Channels.

Accurate experimental data are not so plentiful in the case of channels as for pipes, but it has been found by experiment that the square of the mean velocity of flow in channels varies approximately as what is called the hydraulic mean depth m, i.e. ***the cross-sectional area of the stream divided by the wetted perimeter*** of the section; and that it is proportional to the slope (i) or sine of the angle of inclination of the water surface. Thus

$$v = c\sqrt{m\,i},$$

c being a coefficient which depends on the nature of the surface and also on the value of m.

D'Arcy and Bazin give the rule, $c = \left(\frac{m}{a\,m + a\,b}\right)^{\frac{1}{2}}$, where a and b are constants depending on the nature of the surface.

D'Arcy's values of c for $m = 1, 2, 4, 6$ and 8 are :—

—	$m =$				
	1	2	4	6	8
(1) For very smooth surface of cement or wood .	141	144	146	147	147
(2) Smooth ashlar, brickwork or planks . .	119	125	128	129	130
(3) Channels, such as rubble masonry . .	87	98	106	110	111
(4) Channels in earth	48	62	76	84	88

The coefficient c may be obtained more accurately from Ganguillet and Kutter's formula,

$$c = \frac{41\cdot6 + \frac{1\cdot811}{n} + \frac{0\cdot00281}{i}}{1 + \left(41\cdot6 + \frac{0\cdot00281}{i}\right)\frac{n}{\sqrt{m}}}$$

where n, the coefficient of friction $= 0\cdot0098$ for wooden stave pipe, $0\cdot011$ for cement and sand when set, or for iron pipe, and $0\cdot013$ for ashlar or brickwork.

With irregular sections n has the following values :—

In very firm gravel, $n = \cdot02$.

Canals and rivers tolerably uniform, and free from stones and weeds, $n = \cdot025$.

Where stones and weeds are more plentiful, $n = \cdot03$.

In channels with surface in bad order, $n = \cdot035$ to $\cdot04$.

SOME VALUES OF *c* (TRAUTWINE) FOR A SLOPE OF 1 IN 1000.

Hydraulic Mean Depth *m*.	*n* = ·01	·02	·025	·03	·035	·04
0·2	113	45	34	27	22	18
0·4	131	56	43	34	28	24
0·6	142	63	48	39	32	27
0·8	150	68	52	42	35	30
1	155	71	56	45	38	33
1·5	165	78	62	50	43	37
2	171	83	66	54	46	40
3	179	89	71	59	51	45
4	184	93	75	63	54	48
6	190	99	81	68	59	52
10	197	105	87	74	65	58
20	205	113	94	81	72	65
50	212	120	101	89	79	72
100	216	124	105	94	85	77

CONSTRUCTION TO FIND *c*.

The following construction, modified and simplified from that given by Ganguillet and Kutter, is interesting, and when once made for any given slopes and values of *n* enables *c* to be found at once by simply laying a straight-edge across the diagram.

Draw X X′ horizontal and A Y vertical (Fig. 29). On A Y lay off a scale to suit the values of *c* for which the diagram is to be used. On A X′ lay off a scale showing square roots of values of *m* to be considered in using the diagram. (The author prefers to dispense with this scale, which can readily be done by putting a scale of values of *m* up right hand side and drawing a curve whose abscissæ represent $\sqrt{m}$.)

Determine the flattest slope for which the diagram is to be employed.

Then let

$$k = 41 \cdot 6 + \frac{0 \cdot 0028}{\text{flattest slope}} \quad . \quad . \quad (1)$$

For instance, if we take this flattest slope = 0·00005 (1 in 20,000), and take $n = \cdot 01,\ \cdot 02,\ \cdot 03,\ \cdot 04$, then for each value of n for which the diagram is to be used, we may take it that

$$y - k = \frac{1 \cdot 811}{n} \quad \text{or} \quad y = \frac{1 \cdot 811}{n} + k \quad . \qquad (2)$$

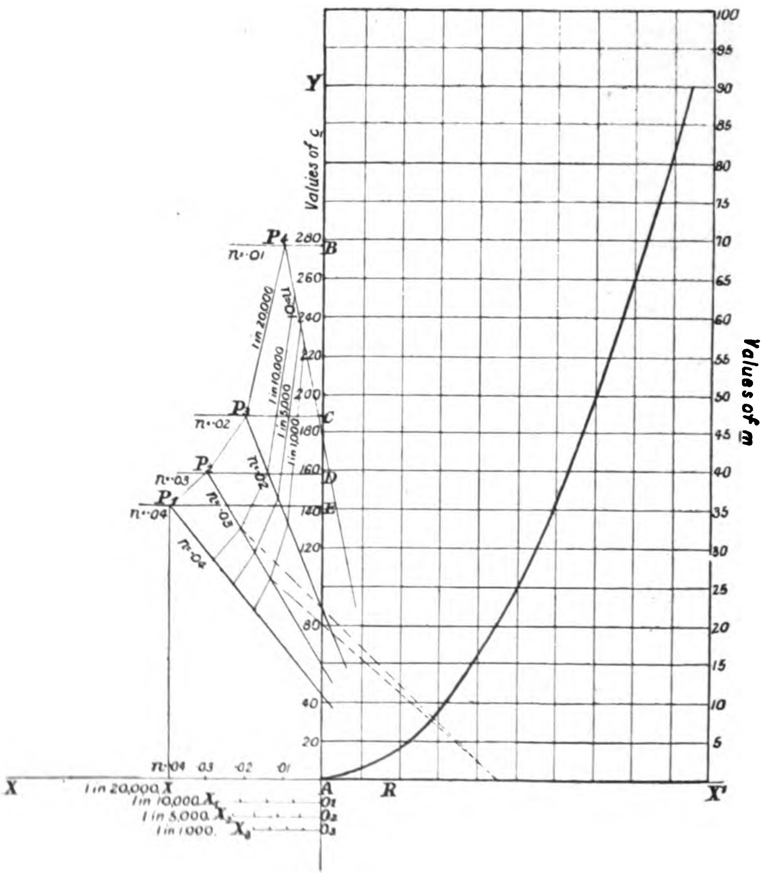


FIG. 29.

For $n = \cdot 01$ the value of

$$k = 41 \cdot 6 + \frac{0 \cdot 0028}{0 \cdot 00005} = 41 \cdot 6 + 56 = 97 \cdot 6,$$

whence

$$y = \frac{1 \cdot 811}{\cdot 01} + 97 \cdot 6 = 278 \cdot 7.$$

Similarly, if

$n = \cdot 02 \qquad y = 188 \cdot 15$
$n = \cdot 03 \qquad y = 158$
$n = \cdot 04 \qquad y = 143.$

(Intermediate values should also be taken.)

Set these values up at A B, A C, A D, A E, on scale already determined for *c*. Draw horizontal lines to the left through the points B, C, D, and E.

Let $x = k \times$ greatest value of n,
$k = 97 \cdot 6$, greatest $n = \cdot 04$.
$\therefore \quad x = 97 \cdot 6 \times \cdot 04 = 3 \cdot 9.$

Draw from A to the left A x = x, the distance A x being measured on the *square root* scale. This can be done by squaring $3 \cdot 9 = 15 \cdot 21$. Set $15 \cdot 21$ on scale of m, run along horizontal line to curve, then under point where this horizontal cuts curve is extremity of the distance required to be set off at A x. Divide A x into as many equal parts as there are values of n taken, and erect a perpendicular at each of these points to meet horizontal from A Y bearing corresponding value of n. These points of intersection, P_1, P_2, P_3, P_4, etc., are on a hyperbolic curve, but may be joined by straight lines P_1, P_2, etc. Look up one metre ($3 \cdot 28$ feet), on right-hand scale and get its square root abscissa; this gives the point R. Draw the radial lines P_1 R, P_2 R, P_3 R, P_4 R, etc., and mark these with proper values of n. For other slopes draw separate horizontal lines $O_1 x_1$, $O_2 x_2$, $O_3 x_3$, to represent on the scale of square roots of m, the values of x_1, x_2, x_3, where

$$x_1 = \left(41 \cdot 6 + \frac{0 \cdot 0028}{\text{slope}}\right) \text{greatest } n \quad . \quad . \qquad (3)$$

For instance, take slopes say

$0 \cdot 0001$ (1 in 10,000), $\therefore \quad x_1 = 2 \cdot 78$
$0 \cdot 0002$ (1 in 5,000), $\qquad x_2 = 2 \cdot 22$
$0 \cdot 001$ (1 in 1,000), $\qquad x_3 = 1 \cdot 72.$

Divide each of the lines $O_1 x_1$, etc., into the same number of equal parts as there are values of n. Each of the portions represents the value of n as in A x. From these dividing points erect perpendiculars to meet the radial lines P_1 R, P_2 R, etc., noting that corresponding values of n are taken. The intersections of these perpendiculars and the corresponding radial lines are to be joined to give the link-polygon-like lines for the slopes and values of n chosen. The diagram is now complete.

Example.—Given m, slope, and n, to find c.

Suppose $m = 20$ (1 foot being unit of length).
slope = 1 in 1000, $n = \cdot 03$.

From the intersection of the proper slope link-curve, and proper radial line for value of n ($\cdot 03$), draw a line to the point on A X′ which marks the $\sqrt{\ }$ of the proper value of m ($\sqrt{20}$ in this case). This radial line cuts A Y in the point showing the value of c required.

The dotted line shows the radial line required in this case, and it cuts A Y at the point $c = 80$, the drawing from which this illustration was prepared being a small one measuring only 8 inches by 7 inches. The actual value given by the tables is 81.

If the slope is 1 in 10,000, m being 20, and $n = \cdot 03$ as before, the second dotted line gives the value $c = 88$; the tables give 89.

If done to a large scale the result is wonderfully accurate.

It is evident that if slope, n, and c are given, m can be found, or if slope, m, and c be given, n can be found readily from the diagram. The solutions for these by algebraic methods are very troublesome.

In the above construction, 1 foot is taken as the unit of length. If 1 metre be the unit,

Equation (1) should be $k = 23 + \dfrac{0 \cdot 00155}{\text{flattest slope}}$.

" (2) " $y = \dfrac{1}{n} + k$.

" (3) " $x_1 = \left(23 + \dfrac{0 \cdot 00155}{\text{slope}}\right)$ greatest n.

Tutton (1893) deduced the law of flow for channels

$$v = \frac{1 \cdot 54}{n} m^{\frac{2}{3}} (i)^{\frac{1}{2}},$$

and the rule of Messrs. Santo Crimp, and Bruges

$$v = 124\, m^{\frac{2}{3}} (i)^{\frac{1}{2}}$$

is recommended by high authorities as the best general formula for sewer work.

Many practical men use the rule $v = 1 \cdot 23 \sqrt{m\text{H}}$, where H is the fall in feet per mile; but this cannot be at all accurate under different circumstances.

VII.

COEFFICIENTS OF HYDRAULIC RESISTANCE.

Sudden Enlargement of Pipe.

If the cross-section of a pipe suddenly changes, as shown in Fig. 30, the direction of flow being from the narrower to the wider section, there is a corresponding ultimate change in the velocity of flow, since velocity × area must be constant if the pipe runs full. This change of velocity is accompanied by a loss of energy or "head," due to the creation of eddies. The rule usually employed to calculate this loss is not capable of very satisfactory mathematical proof, but the following is probably as good as any:—

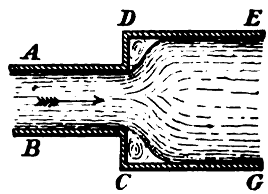


Fig. 30.

Let a_1 and a_2 be the areas of the cross sections at A B and E G respectively, V and v the corresponding velocities. Let the pipe be horizontal, so that h may be neglected. Let P_1 be the intensity of pressure at A B, P_2 that at E G.

Then the energy per lb. at A B is $\frac{P_1}{w} + \frac{V^2}{2g}$; that at E G $\frac{P_2}{w} + \frac{v^2}{2g}$ (see p. 67). Hence the loss of energy per lb., or loss of head,

$$h = \frac{V^2 - v^2}{2g} - \left(\frac{P_2}{w} - \frac{P_1}{w}\right)$$

$$(a) \quad . \quad . \quad . = \frac{V^2 - v^2}{2g} - (h_2 - h),$$

where h_2 and h_1 are the "pressure heads" corresponding to P_2 and P_1.

Now the pressure which acts in the direction opposite to that of the flow is $a_2(P_2 - P_1)$. This force may be considered as that which causes the velocity to diminish from V to v.

But if a force acts for a very short time, the force, or impulse, is measured by the whole change of momentum produced by it. Thus the force necessary to change the velocity of W pounds from V to v is $\frac{W}{g}(V - v)$. Hence

$$a_2(P_2 - P_1) = \frac{W}{g}(V - v) = \frac{w a_2 v}{g}(V - v),$$

where w is the weight in lbs. of one cubic unit of water. Hence

$$\frac{P_2}{w} - \frac{P_1}{w} = \frac{v}{g}(V - v) = h_2 - h_1,$$

or

$$h_2 - h_1 = \frac{v}{g}(V - v).$$

Putting this value of $h_2 - h_1$ into equation (α), we get

$$(\beta)\ h = \frac{V^2 - v^2}{2g} - \frac{2v}{2g}(V - v) = \frac{(V - v)^2}{2g}.$$

The head wasted in such a case is therefore the *height due to the change of velocity.*

Since $V = \frac{a_2}{a_1} v$, we may write equation (β) thus,

$$h = \left(\frac{a_2}{a_1} - 1\right)^2 \frac{v^2}{2g},$$

$\left(\frac{a_2}{a_1} - 1\right)^2$ being called the coefficient of hydraulic resistance.

It is also easy to see how this loss may be expressed in terms of the higher velocity V.

$$F' \frac{V^2}{2g} = \frac{(V - v)^2}{2g},$$

or

$$F' = \frac{(V - v)^2}{2g} \times \frac{2g}{V^2} = \frac{V^2 - 2Vv + v^2}{V^2} = \left(1 - \frac{v}{V}\right)^2.$$

But

$$Va = vA,$$

or

$$\frac{v}{V} = \frac{a}{A};$$

$$\therefore\ F' = \left(1 - \frac{a}{A}\right);$$

and if $A = ra$, A being the larger and a the smaller area,

$$F' = \left(1 - \frac{1}{r}\right)^2.$$

As shown above, F in terms of v is

$$F = (r - 1)^2.$$

Similarly, at all sorts of obstacles in a pipe, the head wasted is expressed by the product of a coefficient—called the coefficient of hydraulic resistance—and $\frac{v^2}{2g}$.

SUDDEN CONTRACTION, OF AREA.

At a suddenly contracted section a similar, but usually smaller, loss is experienced.

Let A, α, and a (Fig. 31) be the larger pipe, the contracted vein and the smaller pipe areas, v, v' and V being the corresponding velocities, then the loss of head due to the expansion of the stream from α to a is

$$h_1 = \left(\frac{a}{\alpha} - 1\right)^2 \frac{v'^2}{2g} = \left(\frac{1}{c'} - 1\right)^2 \frac{V^2}{2g},$$

where c' represents the ratio $\frac{\alpha}{a}$.

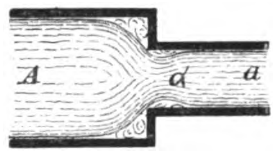


FIG. 31.

It is usual to neglect the very small loss of head due to the contraction of the stream from A to α, hence we get a rule similar to that for a like enlargement, the loss of head being equal to $f \frac{V^2}{2g}$, where f is obtained as before.

If we take the above small loss into account, we may assume it to be of the form $f' \frac{v^2}{2g}$; hence the total loss of head is

$$f' \frac{v^2}{2g} + f \frac{V^2}{2g}.$$

But

$$A v = a V,$$

$$\therefore \quad v = \frac{a}{A} V;$$

or the loss is

$$\left\{ f' \left(\frac{a}{A} V\right)^2 + f V^2 \right\} \frac{1}{2g} = \frac{V^2}{2g} \left\{ f' \frac{a^2}{A^2} + f \right\} = F \frac{V^2}{2g},$$

where F may be obtained by experiment.*

In a channel, the head necessary to give the required velocity

* The author does not know whether these laws for loss of head due to the sudden change of area have been authenticated by any complete and reliable experiments. If so, they are worthy of that respect which a study of the usual proofs given of them does not inspire. There does not, for instance, seem to be any good reason for assuming anything of the nature of impact. Energy is wasted in eddies set up by internal friction, yet we deduce a law *independent* of viscosity, and seeming to indicate that with a given pipe and flow there would be the same waste whether the fluid were tar or water, which is at least very doubtful.

forms a considerable portion of the total head h, $h - \frac{v^2}{2g}$ being the head available for overcoming frictional resistances.

$$h - \frac{v^2}{2g} = F \times \frac{v^2}{2g},$$

or

$$F = h \times \frac{2g}{v^2} - 1;$$

also

$$v^2 = c^2 m \frac{h}{L}.$$

$$\therefore \quad F = \frac{2gL}{c^2 m} - 1.$$

The following table of coefficients of hydraulic resistance includes most of the values required in practice; some usually given, but of doubtful accuracy, are omitted.

TABLE OF COEFFICIENTS OF HYDRAULIC RESISTANCE.

Resistance due to	Coefficient of Resistance F.	Explanation of Symbols, etc.
Square-edged entrance to pipe	0·5	
Well-shaped bell-mouthed entrance to pipe	0·05	
Sharp-edged orifice in thin plate	0·06	
Surface friction of clean pipe	$4f \cdot \frac{L}{D}$	$f = 0{\cdot}005\left(1 + \frac{1}{12D}\right)$
Surface friction of rough pipe	$4f' \frac{L}{D}$	$f' = 0{\cdot}01\left(1 + \frac{1}{12D}\right)$ (D'Arcy's rules)
Surface friction of channel of uniform section	$\frac{2gL}{c^2 m} - 1$	m = hydraulic mean depth; c, a coefficient (given at p. 51).
Sudden enlargement of pipe areas a to A as 1 to r	$\left(1 - \frac{1}{r}\right)^2$	Referred to higher velocity
" "	$(r - 1)^2$	Referred to lower velocity
Sudden contraction of pipe from area A to area a, a being c_1 times the area of vena contracta.	$(K - 1)^2$	K given by Rankine's rule $K = c_1 \sqrt{2{\cdot}618 - 1{\cdot}618 \frac{a^2}{A^2}}$, referred to higher velocity.

TABLE OF COEFFICIENTS—*continued.*

Resistance due to	Coefficient of Resistance F.	Explanation of Symbols, etc.
Curved bend in pipe . .	$(0{\cdot}0128+0{\cdot}0186\,R)\frac{L}{R}$	L = length of bend measured along centre of pipe in feet; R = radius of bend measured as above (Navier's formula)
	or	
" "	$c_b \times \frac{\phi}{180^\circ}$ where $c_b = 0{\cdot}131+1{\cdot}847\left(\frac{d}{2R}\right)^{\frac{7}{2}}$	Weisbach's formula, d = diameter of pipe, R = radius of centre line of bend (see B, Fig. 32). *
Sharp bend or elbow in pipe (see A, Fig. 32).	$0{\cdot}9457\sin^2\frac{\phi}{2}$	(Weisbach). Satisfactory experimental data wanting.
	$+\ 2{\cdot}047\sin^4\frac{\phi}{2}$	See page 61.
If $\phi = 20^\circ$ . .	0·046	
30° . .	0·0725	
40° . .	0·139	
45° . .	0·1824	
60° . .	0·364	
80° . .	0·740	
90° . .	0·984	
100° . .	1·260	
110° . .	1·556	
120° . .	1·861	
130° . .	2·158	
Diaphragm in pipe—Central orifice a in section, pipe A in section. .	$\left(\frac{A}{c_1 a} - 1\right)^2$	Values of c_1 for various values of $\frac{a}{A}$ are given on page 60.
Sluices, valves, etc. .	.. ..	See page 61.

* See Appendix.

LOSS OF HEAD DUE TO OBSTRUCTIONS IN PIPES.

Diaphragm in pipe: central orifice a in section; pipe A in section.

Contracted area of water $= c_1 a$.

$$\text{Loss of head} = \frac{1}{2g}\left(\frac{vA}{c_1 a} - v\right)^2$$

$$= \frac{v^2}{2g}\left(\frac{A}{c_1 a} - 1\right)^2$$

$$= F\left(\frac{v^2}{2g}\right), \quad \text{where} \quad F = \left(\frac{A}{c_1 a} - 1\right)^2.$$

VALUES DETERMINED BY WEISBACH.

F	$\frac{a}{A}$	c_1	F	$\frac{a}{A}$	c_1
225·9	0·1	0·624	1·79	0·6	0·712
47·77	0·2	0·632	0·79	0·7	0·755
30·83	0·3	0·643	0·29	0·8	0·813
7·8	0·4	0·659	0·06	0·9	0·892
3·75	0·5	0·681			

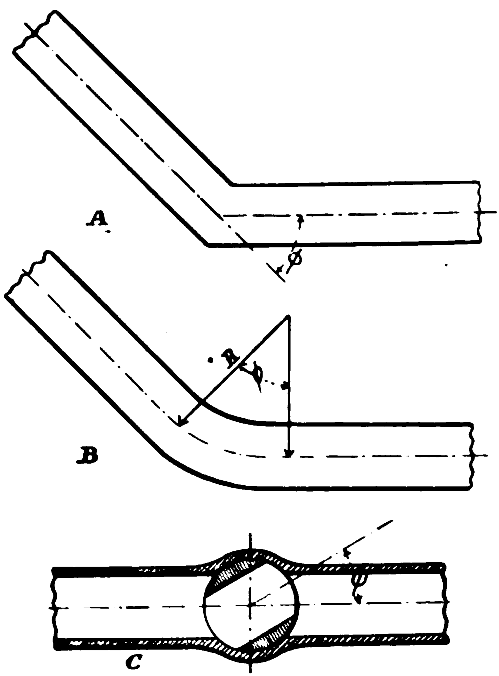


FIG. 32.

COEFFICIENTS OF HYDRAULIC RESISTANCE.
Sluices, valves, etc. (*Weisbach.*)

Sluice in pipe of rectangular section:

a = area of pipe;
s = area of sluice opening

$$\text{head lost} = F \frac{v^2}{2g}.$$

F	$\frac{s}{a}$	F	$\frac{s}{a}$
0·00	1	4·02	0·5
0·09	0·9	8·12	0·4
0·39	0·8	17·8	0·3
0·95	0·7	44·5	0·2
2·08	0·6	193	0·1

Sluice in cylindrical pipe:
r = ratio of height of opening to diameter of pipe.

F	r	F	r
0·00	1	2·06	0·5
0·07	0·875	5·52	0·375
0·26	0·75	17·00	0·25
0·81	0·625	97·8	0·125

Cock in cylindrical pipe (C, Fig. 32):
r = ratio of cross-section of opening to that of pipe.

F	r	ϕ	F	r	ϕ
0·05	0·926	5°	17·3	0·385	40°
0·29	0·85	10°	31·2	0·315	45°
0·75	0·772	15°	52·6	0·25	50°
1·56	0·692	20°	106·0	0·19	55°
3·1	0·613	25°	216·0	0·137	60°
5·47	0·535	30°	486·0	0·091	65°
9·68	0·458	35°			

Experiments on Waste of Energy at Bends.

The formula of Weisbach seems doubtful, and as there is much to be learned about the hydraulic resistance due to obstacles of various kinds in pipes, the following reference to a simple apparatus used by the author in the hydraulic laboratory of the Technical College, Finsbury, and to some of the preliminary results obtained, may be interesting.

The apparatus consisted essentially of a mercury U tube connected at its two ends to two small side tubes inserted in the pipe to be tested at points 3 feet apart. A straight pipe was first employed, and the loss of head in the 3 feet of pipe determined for different velocities of flow, the velocity being determined by weighing the water passing in say five minutes, from this determining Q and hence v, since $v = Q \div \frac{\pi}{4} d^2$, d being the inside diameter of the pipe. Then similar pipes but with sharp bends or knees with angles of 30°, 45°, 60°, 90°, and 130°, etc. were used in the same way, and deducting the friction due to the internal surface of the pipe as if straight, the waste of energy at the bend was in each case determined. The whole of the results need not be given, but it may be stated that the waste of energy at a bend of this kind is not independent of the roughness of the pipe. For all the pipes tried, the coefficient is higher for all angles than that given by Weisbach's formula.

Angle of Bend (ϕ).	Value of F (by experiment).		Value of F (Weisbach).
	Smooth Brass Pipe.	Rough Iron Gas-pipe.	
30	0·367 (?)	0·935	0·0725
60	1·169	1·736	0·364
90	2·02	3·098	0·984
130	3·46	4·598	2·158

Observations made on glass pipes, into which dark liquid was injected during the flow of water through them, showed that in all probability undue importance has been attached to the supposed formation of a *vena-contracta*. It is more likely that the eddy friction follows much the same kind of law as ordinary skin friction, and that therefore the waste of head at sharp bends, whilst depending on the angle ϕ, is also about proportional to $\frac{v^n}{d^p}$, where n is about 1·7 for the brass pipes and 2 for the rough iron pipes, and may be obtained in each case by plotting values of log h and log v from observations on a straight pipe of the given kind: p being taken at, say, 1·04 for brass and 1·32 for rough iron, as per Tutton's rules. The brass pipes in these experiments were a little over, and the iron pipes a little under, half the diameter of those used by Weisbach.

EXAMPLES.

1. In a water main in which water flows at a steady velocity, find the pressure at a point 100 feet below the hydraulic grade line.

Ans. 43·5 lbs. per square inch.

2. A pipe, 1 foot in diameter, discharges into one 2 feet in diameter; if the velocity in the larger is 2 feet per second, find the loss of head at the junction. *Ans.* $\frac{9}{(16 \cdot 1)}$ feet.

3 A pipe, 6 inches in diameter, discharges into one 9 inches in diameter, the flow being 80,000 gallons per hour. Find the head wasted at the junction ($6\frac{1}{4}$ gallons = 1 cubic foot). *Ans.* 3·35 feet.

4. Find the horse-power necessary to pump 1,000,000 gallons per day to a height of 200 feet, and through a 6-inch straight pipe for a distance of 1 mile. The coefficient of resistance at entrance is 0·5, and that for pump-valves, etc., 4.

$$1{,}000{,}000 \text{ gallons per 24 hours} = \frac{1{,}000{,}000}{6 \cdot 25 \times 24 \times 60 \times 60}$$

$$= 1 \cdot 85 \text{ cubic feet per second;}$$

$$\therefore \text{ work done by pumps per second} = 62 \cdot 4 \times 1 \cdot 85 \times 200$$

$$+ 62 \cdot 4 \times 1 \cdot 85 \left(4 + \frac{4fL}{d} + 0 \cdot 5\right) \frac{v^2}{2g}.$$

$$Q = av; \quad \therefore \quad 1 \cdot 85 = 0 \cdot 7854 \left(\tfrac{1}{2}\right)^2 v,$$

or

$$v = 9 \cdot 423 \text{ feet per second.}$$

$$f = 0 \cdot 0058$$
$$L = 5280$$
$$d = \tfrac{1}{2}$$

$$\therefore \quad \text{work per sec.} = 62 \cdot 4 \times 1 \cdot 85 \left\{200 + (4 + 245 + 0 \cdot 5) \frac{9 \cdot 423^2}{64 \cdot 4}\right\}$$

and

$$HP = \frac{\text{work done per sec.}}{550} = \frac{62 \cdot 4 \times 1 \cdot 85 \times 544 \cdot 1}{550} = 114 \cdot 2.$$

5. A pipe 1 foot in diameter suddenly contracts to 6 inches in diameter. If the flow is 20,000 gallons per hour, find the head wasted at the junction ($c_1 = 1 \cdot 3$). *Ans.* 0·347 foot.

6. In the last case, if the flow were doubled find the waste of head at the junction. *Ans.* 1·388 foot.

7. A clean horizontal pipe is 1 mile long and 6 inches in diameter.

It has three bends in it, each including an angle of 120° and with 5 feet radius; also six bends, each of 90°, and a radius equal to twice the diameter of the pipe. Find the head wasted at entrance, at the bends, and in the 1 mile of pipe. Velocity of flow 1·965 feet per second.

Ans. { At entrance 0·03 foot; 3 bends 0·054 „; 6 „ 0·360 „ }

In 1 mile of pipe, 14·69 feet. Total 15·144 feet.

8. In a semicircular channel, 4 feet diameter, running full, find the head necessary to give a velocity of 2 feet per second in 1 mile of channel ($c = 100$). *Ans.* 2·1 feet.

9. Find the coefficient of resistance in the last example. *Ans.* 33.

10. In a clean 6-inch pipe, 1000 feet long, there are four sharp bends or knees, one including an angle of 60°, two an angle of 90°, and one an angle of 120°. If the flow is 140 gallons per minute, find the total head wasted at the square-edged entrance, at the four bends, and in the straight part of the pipe. *Ans.* 3·172 feet.

11. A uniform channel has the following section: flat bottom, 8 feet wide; two sloping sides, each making an angle of 30° with the horizontal, the water being 4 feet deep. Find the hydraulic mean depth and the flow, if the fall per mile is 3 feet. $c = 126$.

Ans. $m = 2 \cdot 488$.
Flow = 283 cubic feet per second.

Steady Flow.

The reader who has followed the foregoing work carefully will readily understand some of the practical results of Bernouilli's great theorem now to be referred to.

A practical illustration of this theorem is due to Mr. Froude, who brought the matter before the British Association in 1875.

Fig. 33 shows the illustration adopted by Mr. Froude.

The pipe C B is of varying section, and as the velocity in it must vary so that $Q = 0 \cdot 7854\, d^2 \times v$ shall always be the same, where d is the diameter of the pipe at the given place and v the velocity of the water there, it is evident that at a wider section the velocity is less, and Froude's experiment showed that the pressure is greater, than at a smaller section if the pipe be level. The pressure inserted in the pipe show water levels corresponding to the *pressures* at the respective sections, each height in feet in the pressure tube being $\frac{f}{w}$, where f is the pressure of the water in lbs. per square foot at that section, and w is the weight of 1 cubic foot of the water.

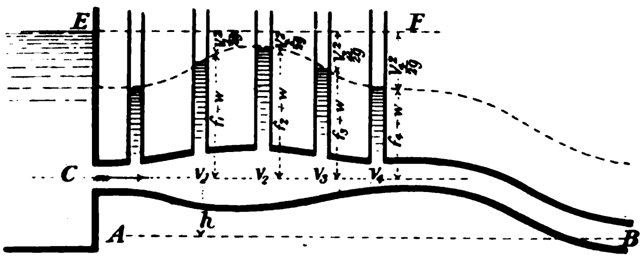


FIG. 33.

It will be seen that the head *lost* is (neglecting friction) in each case the kinetic energy of 1 lb. of the water, whilst the remaining potential energy of the 1 lb. is h, if A B represent the datum line.

VIII.

DISTRIBUTION OF ENERGY ALONG, AND AT RIGHT ANGLES TO, STREAM LINES.

Bernouilli's law is as follows :—

$$h + \frac{v^2}{2g} + \frac{f}{w} \text{ is constant for each 1 lb. of water ;}$$

this constant being in the figure represented by the vertical distance between the lines E F and A B.

The illustration shows very well how the total "head" or energy is distributed. Neglecting change in level of the pipe, which, for pipes conveying pressure water to hydraulic machines is usually permissible, we see that wherever the water flows slowly the pressure increases, and where it flows faster the pressure diminishes.

This fact has a very important application in the case of the jet pump of the late Professor James Thomson, referred to at page 69.

PROOF OF LAW OF CONSTANT ENERGY.

Imagine a very small mass of water flowing along stream lines, as shown in Fig. 34. Imagine it to be a frictionless fluid acted on only by gravity. Let a be the cross-sectional area of the little

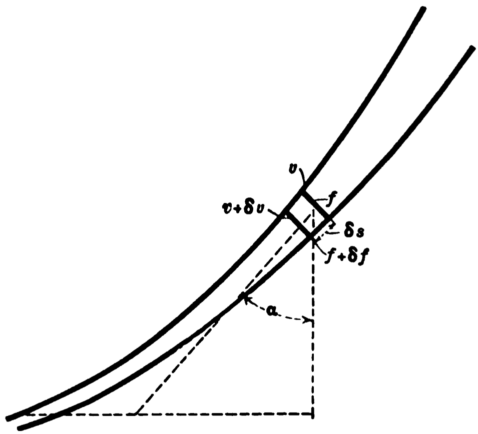


FIG. 34.

column, its length being δs, the velocity and pressure being v and f at one end, $v + \delta v$ and $f + \delta f$ at the other. Since force = mass × acceleration, the resultant force in the direction of the stream tube is $\frac{w}{g} a \, \delta s \frac{\delta v}{\delta t}$.

Since the force of gravity resolved along the stream tube, together with the resultant of the pressures on the ends of the column, is equal to the acting force, it must equal that represented by the above expression.

The resolved part of gravity is $w a \, \delta s \cos \alpha$, the resultant impressed force in the same direction is $f a - (f + \delta f) a$, hence—

$$\frac{w}{g} a \, \delta s \frac{\delta v}{\delta t} = w a \, \delta s \cos \alpha + f a - (f + \delta f) a.$$

Dividing across by a we get

$$\frac{w}{g} \delta s \frac{\delta v}{\delta t} = w \, \delta s \cos \alpha - \delta f.$$

We have taken δs any small element of length; take it such that $\frac{\delta s}{t} = v$, also let $\delta s \cos a = -\delta h$; then our equation becomes

$$\frac{w}{g} v \, \delta v = - w \, \delta h - \delta f,$$

or

$$\frac{1}{g} v \, \delta v + \delta h + \frac{\delta f}{w} = 0.$$

Letting the quantities becomes indefinitely small, and integrating, we get

$$\frac{v^2}{2g} + h + \int \frac{df}{w} = \text{constant} \quad . \quad . \quad . \quad . \quad (1)$$

Or, in the case of water,

$$\frac{v^2}{2g} + h + \frac{f}{w} = \text{constant} \quad . \quad . \quad . \quad . \quad (2)$$

These terms may be called respectively the kinetic, the potential, and the pressure energy of the 1 lb. of water. $2 \cdot 3 \, p$ may be written for $\frac{f}{w}$ where p is the pressure in lbs. per square inch, f being the pressure in lbs. per square foot, and w the weight in lbs. of 1 cubic foot.

The term "pressure energy" has been objected to, and the nature of an objection which is urged will be gathered from the following illustration :—

Suppose that a strong box is filled with water, and that by screwing a small screw into it we produce a great pressure p in the water. Are we justified in regarding every pound of the water as being possessed of $2 \cdot 3 \, p$ ft.-lbs. of pressure energy, or energy due to the pressure p? No. For if we open a valve and allow the water to escape, though there may have been a great pressure p just for a moment, the pressure almost instantly dies away, and the water flows out quietly and almost without energy. Evidently each pound of water possessed very little energy. It is the question whether or not the *state of pressure will continue*, and a steady flow at that pressure be assured, that determines our right to call this kind of energy "pressure energy."

Suppose a man has a certain income, say from a sum invested in British Consols, and suppose you are perfectly certain that this income is constant, this certainty constitutes the income a store of wealth and a saleable commodity. To say that a man makes a sovereign a day is not of much importance—anyone may do that once in a while, but if he has ***a regular income*** of one pound a day, that makes him an important member of society.

For a like reason, if we establish in communicating pipes, by means of pumps or other mechanism, a working difference of pressure at two points A and B, and if we know that this difference is likely to be maintained and is a thing we can depend upon, then we know that the flow from the place of higher to that of lower pressure will, in a given pipe, be the same at all times, and the same amount of work can be got out of the water every minute.

Leaving out of account for the moment the question of how the difference of pressure is produced, the certainty of that difference of pressure being maintained and a steady flow available, constitutes our right to regard each cubic foot or each pound of the water as possessed of energy which, like any other kind of energy, has a commercial value. Under these circumstances, therefore, the term "pressure energy" is a convenient one.

Thus each pound of water at the pressure of the atmosphere possesses $14 \cdot 7 \times 2 \cdot 3 = 33 \cdot 8$ ft.-lbs. of pressure energy. It would have the same store of energy if at zero pressure and a height of $33 \cdot 8$ feet. One cubic foot of water, at a pressure of 700 lbs. per square inch, possesses $62 \cdot 4 \times 2 \cdot 3 \times 700 = 100{,}464$ ft.-lbs. of pressure energy.

If, then, a person receives per minute from a hydraulic power company 100 gallons of water at a pressure of 700 lbs. per square inch, he receives every minute $1000 \times 2 \cdot 3 \times 700 = 1{,}610{,}000$ ft.-lbs. of energy in the shape of *pressure* energy (since 1 gallon of water weighs 10 lbs.), which is equivalent to $\frac{1{,}610{,}000}{33{,}000} = 48 \cdot 8$ horse-power. He may also receive a little energy in the shapes of potential and kinetic energy, but this amount is so small compared with the enormous store of the energy due to pressure that it may be neglected. Thus, take 1 lb. of the water, imagine it to be moving, when received, at a velocity of 2 feet per second, and that it is 40 feet above the datum level. It possesses 40 ft.-lbs. of potential energy, $\frac{2^2}{64 \cdot 4} = \frac{4}{64 \cdot 4} = \frac{1}{16}$ ft.-lb. of kinetic energy, and $2 \cdot 3 \times 700$, or 1610 ft.-lbs. of pressure energy. Evidently the pressure store is the only one of much importance.

Professor James Thomson's Jet Pump.

This apparatus affords a practical example of the foregoing law.

The water whose flow supplies the energy required, enters at F. (Fig. 35.) Near E the stream is contracted, and hence flows with greater velocity and smaller pressure, water being drawn in at R, which forms the suction pipe of the pump. At E the streams

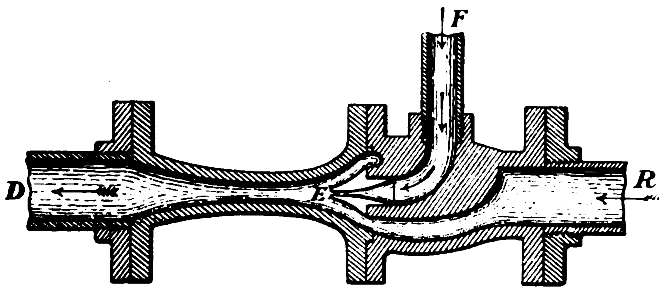


Fig. 35.

unite, and they are discharged together at D. Evidently this arrangement only gives a certain—not very great—diminution of pressure at E as compared with that at F and D; hence, if we wish the pump to draw water, say from a well or marsh, the pressure at D should be atmospheric, because if the pressure at D is high, it will be impossible to reduce it at E below that due to the atmosphere.

The Korting Water-Jet Elevator.

This appliance seems to act somewhat in the same way as the Thomson jet-pump. High pressure water from a tank or reservoir at a considerable height, passing through a narrow neck in a pipe, draws in water which has accumulated at the lower level, discharging it, together with that taken from a higher level, at an intermediate height. Even a comparatively low head can be utilised in this way to cause considerable suction; the appliance being useful in tunnelling or where it is required to raise water from the deep sump of a mine to a pump at some intermediate height.

Example.

A horizontal pipe conveys 6 gallons of water per second; at a point where the diameter is 4 inches, the pressure is 50 lbs. per square inch. Find the velocity of flow and pressure at a point where the diameter is $2\frac{1}{2}$ inches, allowing 10 per cent. loss of head by friction between the two points.

Ans. 40·34 lbs. per square inch.

CHANGE OF ENERGY AT RIGHT ANGLES TO STREAM LINES. AVERAGE "ROTATION."

The law for change of pressure *along* a stream line is given at page 67, the total energy of unit weight of the water being always the same. Let us now inquire what is the law for change of pressure as we go at right angles to the direction of flow.

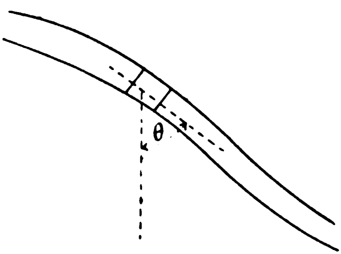


FIG. 36.

Consider a small prism of the fluid in a stream tube, its ends at right angles to the stream lines, as shown in Figs. 36 and 37. Let it be unit depth at right angles to the paper, and thickness (or breadth) $2\,\delta r$.

Along stream lines the pressures have been considered (page 66). At right angles to the stream lines the forces acting on the prism are due to—

(1) Inside pressure $p - \delta p$.

(2) Outside pressure $p + \delta p$.

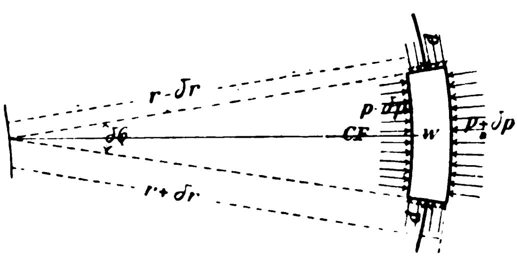
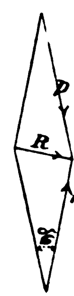


FIG. 37.



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(3) End pressures (their outward component = seen from Fig. 38).

(4) Centrifugal force.

(5) Weight of block (inward compon

The mass of the prism is $\frac{r\,\delta\phi\,.\,2\,\delta r\,.\,w}{g}$, its velocity being v, the centrifugal force (4) is $\frac{v^2}{r}\,.\,\frac{r\,\delta\phi\,.\,2\,\delta r\,.\,w}{g}$.

(1) is $(p - \delta p)\,(r - \delta r)\,\delta\phi$.
(2) is $(p + \delta p)\,(r + \delta r)\,\delta\phi$.
(3) is $p\,.\,2\,\delta r\,.\,\delta\phi$.
(4) (as above) $\frac{v^2}{r}\,.\,\frac{r\,\delta\phi\,.\,2\,\delta r\,.\,w}{g}$.
(5) is $r\,\delta\phi\,2\,\delta r\,w\sin\theta$. (Fig. 36.)

There is no motion at right angles to a stream line.

$\therefore$ The sum of the above forces = 0.

$$\therefore\quad \frac{v^2}{r}\,.\,\frac{r\,\delta\phi\,.\,2\,\delta r\,.\,w}{g} + (p - \delta p)\,(r - \delta r)\,\delta\phi + p\,\delta\phi\,2\,\delta r$$
$$= (p + \delta p)\,(r + \delta r)\,\delta\phi + r\,\delta\phi\,.\,2\,\delta r\,w\sin\theta,$$

or

$$\frac{w\,v^2\,.\,2\,\delta r}{g} + pr - p\,\delta r - r\,\delta p + \delta p\,.\,\delta r + 2p\,\delta r$$
$$= pr + p\,\delta r + r\,\delta p + \delta p\,.\,\delta r + 2\,r\,.\,w\,.\,\delta r\sin\theta,$$

whence, cancelling, we get

$$\frac{w\,v^2\,\delta r}{r\,.\,g} - \delta p = \delta r\,.\,w\,.\,\sin\theta,$$

or

$$\frac{\delta p}{\delta r} = \frac{w\,v^2}{gr} - w\sin\theta \quad . \quad . \quad . \quad . \quad . \quad (a)$$

Also

$$(\beta)\quad \delta h + \frac{\delta p}{w} + \frac{v\,.\,\delta v}{g} = 0$$

(by differentiating law for total energy constant, given at p. 67), and

$$\frac{\delta h}{\delta s} = \cos\theta \text{ (Fig. 39)},$$

whence, multiplying (β) by $\frac{w}{\delta s}$, it becomes

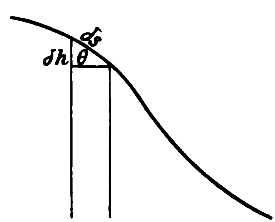


FIG. 39.

$$w\cos\theta + \frac{\delta p}{\delta s} + \frac{w\,.\,v\,.\,\delta v}{g\,.\,\delta s} = 0.$$

$$(\gamma)\quad \therefore\quad \frac{\delta p}{\delta s} = \frac{-\,w\,v}{g}\,.\,\frac{\delta v}{\delta s} - w\cos\theta.$$

This also holds if the increments are made smaller and smaller without limit.

Now

$$h + \frac{p}{w} + \frac{v^2}{2g} = E$$

(the total store of energy of 1 lb. of incompressible fluid moving along stream line).

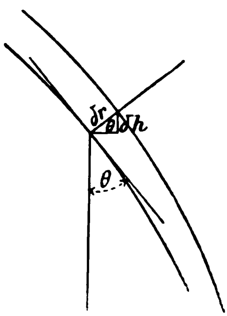


Fig. 40.

Differentiating, we get

$$\therefore \quad \frac{dh}{dr} + \frac{1}{w} \cdot \frac{dp}{dr} + \frac{v}{g}\frac{dv}{dr} = \frac{dE}{dr},$$

the law of change of store of energy as you cross stream lines. But

$$\frac{dh}{dr} = \sin\theta \text{ (Fig. 40)},$$

$$\frac{dp}{dr} = \frac{wv^2}{gr} - w\sin\theta, \left[\text{from } (a)\right].$$

$$(\delta) \quad \therefore \quad \frac{dE}{dr} = \sin\theta + \frac{1}{w}\left(\frac{wv^2}{gr} - w\sin\theta\right) + \frac{v\,dv}{g\,dr}.$$

$$(\epsilon) \quad \frac{v^2}{gr} + \frac{v\,dv}{g\,dr} = \frac{dE}{dr} = \frac{v}{g}\left\{\frac{v}{r} + \frac{dv}{dr}\right\}.$$

If the block is moving, say, along a tube of decreasing diameter, the fluid at the top of the block has a smaller velocity than that underneath; the block is in a state of shear.

Lines in the block are being turned through an angle owing to this shear. We can now get the average angular velocity of these lines. This is called by Professor Cotterill the "molecular rotation." It is not "molecular"; it is simply the average rotation of every line in the block, and better called the "rotation." It is equal to

$$\tfrac{1}{2}\left(\frac{v}{r} + \frac{dv}{dr}\right).$$

We see from this that if the cross-section of a tube of flow is constant, v is constant; hence the "rotation" is the same at every point of the stream if r is constant.

We see also that if the total energy of a particle of unit weight is the same in two stream lines, it always remains the same, [illegible] instance, if all stream lines come from rest in the same [illegible] of water, there cannot exist any "rotation" in any of [illegible]

"Irrotational" motion means

$$\frac{dv}{dr} + \frac{v}{r} = o.$$

If the radius is infinite, $\frac{v}{r} = o$, hence $\frac{dv}{dr} = o$; and there is no "rotation."

This is one instance of irrotational motion.

If $\frac{v}{r}$ is constant (i.e. the velocity always proportional to r), $\frac{dv}{dr} = o$, and the "rotation" is constant.

If straight lines join into, say, circular stream lines, $\frac{v}{r}$ is no longer zero. Hence, $\frac{dv}{dr} + \frac{v}{r}$ cannot $= o$, and by similar reasoning for other points where the curvature suddenly changes we see that *there must be a change in the "rotation" wherever there is discontinuity of curvature*, and along neighbouring radii a unit has a different rate of change of total store of energy.

IX.

THE MEASUREMENT OF FLOWING WATER.

In order that the efficiency of a water-power installation may be tested, or the amount of power available at any point in a stream or river determined, the rate of flow,* i.e. the number of cubic feet or gallons of water passing a given point per unit time, must be measured. It is not an easy thing to do this with anything like accuracy. There are several methods which may be employed, which will now briefly be referred to. The

"Q = AV" Method

consists in obtaining the area of the cross-section of the stream at the place selected, and multiplying this by the average velocity of the

* Rate of flow here does not mean rate of *motion*, but refers to quantity, as explained.

water. If the first is in square feet and the latter in feet per second, the product gives the rate of flow in cubic feet per second.

To obtain the section of the stream, a cord or rope may be stretched across it at right angles to the direction of the stream, numbered marks being placed on this at regular, and not too distant, intervals; soundings are then taken at these points, the depth of the water at each number being entered in a notebook. The section is then plotted to scale, and the area of the figure obtained by a good planimeter or any of the methods usually employed for finding such an area. The scale of the drawing being known, the area of the section in square feet, say, is thus found approximately.

The second step is to obtain the mean velocity of the water. This is sometimes done by finding the surface velocity near the centre of the stream by floats; thus two observers are stationed at a convenient distance apart, about half the distance being on each side of the selected section. By the firing of a pistol or shouting, the first man indicates when the float passes him; the time till it reaches the second observer is shown by his watch. Thus the surface velocity can be roughly found. This, however, is a very unsatisfactory method, as the float will *not* go down stream in anything like the required way. It may be found with a fair amount of accuracy by the Pitot tube, a vertical glass tube with a right-angled bend in it, the horizontal portion being placed so as to face up-stream when immersed.

The water rises in the vertical portion, above the surrounding water, by an amount $= v^2 \div 2g$ nearly, hence one reading is sufficient to enable the velocity at that point to be calculated. D'Arcy improved the apparatus by providing two glass tubes in the vertical portion with a means of closing the same, so that the difference of level in them can be read after the apparatus is lifted out of the water.

The surface velocity being found, the mean velocity varies from 0·62 to 0·85 of it, depending on the nature of the channel. Recent experiments give 0·65 as probably the best number. It may be found approximately by a formula like that of Basin $v_m = v_s - 25 \cdot 4 \sqrt{i\,m}$. v_s being the surface and v_m the mean velocity, m and i having the meanings already assigned to them. Prony's formula,

NOTE.—A miner's inch of water is a rate of flow equivalent to 12 U.S. gallons per miuute. 1 U S. gallon of fresh water weighs 8·33 lbs., containing 231 cubic inches; there being therefore 7·48 such gallons to 1 cubic foot. The imperial (English) gallon weighs 10 lbs., containing 277·27 cubic inches; therefore 6·23 gallons (usually taken as 6¼ gallons) = 1 cubic foot.

$$V = \frac{v(v + 7{\cdot}77)}{v + 10{\cdot}33},$$

V being the mean and v the surface velocity, has been a good deal used.

Current Meters.

Another and a much better way of obtaining the average velocity of the water is by means of current meters. A modern instrument of this class is shown in Fig. 41. It is an instrument furnished with vanes like a screw propeller, which when immersed in flowing water revolve, their speed being a measure of the velocity of the water.

The figure will readily be understood. A pair of differential wheels B have a worm wheel engaging with them, this worm being

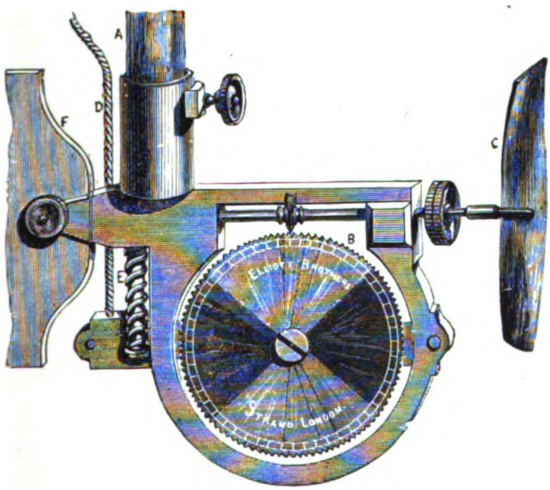


Fig. 41.

on the shaft of the propeller C. The apparatus is clamped firmly on a rod A, and is inserted to the required depth in the water. At a given instant the propeller is thrown into gear by means of the check line D, and at the end of the required interval it is again thrown out of gear. The reading on the counter gives the number of revolutions, or, if suitably calibrated, the speed of the current.

Thus the velocity at a great many points in the section can be found, and hence the mean velocity determined much more accurately than by floats. As the velocity close to the bed and sides of the channel falls off considerably, probably the flow is a little less than this method indicates; hence a turbine tested by this method combined with a dynamometer will probably show a lower efficiency than if the more accurate method by weir gauges is employed.

This may to some extent account for the fact that Continental tests of turbines give a lower efficiency than that usually found for American wheels, where the latter method of measurement of flow is followed.

The Measurement of Flow by Weir-Gauges.

By far the most accurate method of measuring the flow of water in streams or rivers is by means of the weir-gauge, or gauge-notch, as it is sometimes called. This usually consists of a plate of wood or suitable material with a notch cut in it of a given form, the water to be measured passing through this notch. There are two kinds of notches in general use for this purpose—the V-shaped notch of the late Professor James Thomson (brother of Lord Kelvin), and the rectangular notch, associated with the excellent experiments of Mr. Francis, carried out at Lowell, Massachusetts, in the United States. The former is most accurate for variable flows, the latter most convenient for considerable flows. The splendid work of the late Professor James Thomson in connection with this part of hydraulics is known to most engineers and students, his great generalisation in connection with the law of flow from similar orifices being a most remarkable and useful one. Professor Thomson's investigations will be found recorded in the 'Proceedings' of the British Association for 1858 and 1876.

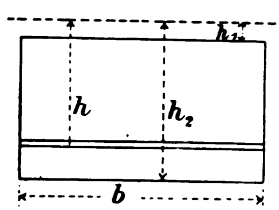


Fig. 42.

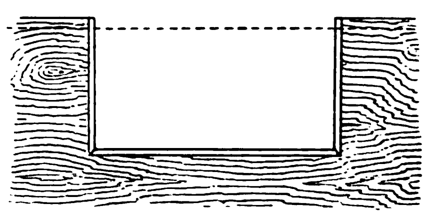


Fig. 43.

Space does not admit of a full record, but his reasoning may be said to follow somewhat the following lines, though it is too complete to admit of being well given in abstract. He shows that the method adopted by many writers of finding, or attempting to find, the flow through a rectangular notch by the methods of the integral calculus is incorrect.

The usual method is to take a small rectangular portion (Fig. 42) of the rectangular orifice, find the flow though it, and integrate all such flows to get the total flow through the notch (Fig. 43).

Let Q = the volume passing per unit time (usually the number of cubic feet per second).

$$d\,Q = \sqrt{2gh} \times b\,d\,h.$$

whence

$$Q = \int_{h_1}^{h_2} b\,.\,d\,h\,\sqrt{2gh} = b\,\sqrt{2g}\int_{h_1}^{h_2} h^{\frac{1}{2}}\,d\,h,$$

or

$$Q = \tfrac{2}{3}\,b\,\sqrt{2g}\,(h_2^{\frac{3}{2}} - h_1^{\frac{3}{2}}).$$

This is called the "theoretic" flow, and it is multiplied by a coefficient called the coefficient of discharge to get the true flow, giving

$$Q = \tfrac{2}{3}\,c\,b\,\sqrt{2g}\,h_2^{\frac{3}{2}}$$

if h_1 is zero.

This method is wrong for the following reasons :—

First. The velocity is *not* the same all along the little band as here assumed.

Second. In any little element of area of the orifice it is *not* the velocity of the water at it which, multiplied by the area, will give the flow, but the *component of the velocity normal to the plane of the element.*

Third. We have no right to assume that at any element of the area the velocity is found by the rule $v = \sqrt{2gh}$, for, except at the boundary of the jet, the water is under *more* than atmospheric pressure, and hence, by Bernouilli's law, it must have less than the velocity given by the rule above.

These and other objections, pointed out by Professor Thomson, show that the usual method is altogether misleading and wrong.

Professor Thomson goes on to show that if there are two similar vessels with exactly similar orifices, the dimensions of the larger orifice and vessel being n times that of the smaller, then the lines of flow from the two vessels will be similar, and the velocities will be as 1 : $\sqrt{n}$.

...e the orifices filled with similar stream tubes : the water will ...strainedly in the one as in the other, and it can be shown ...ide tubes have really no duty to perform, and the total homologous pressures on similarly situated small areas at u and u_1 are as 1 to n^3.

From the similarity of the forms of the two similar imaginary tubes (Fig. 44) we have in each

$$\frac{\text{area at E}}{\text{area at } u} = \frac{\text{area at E}_1}{\text{area at } u_1}.$$

Hence the

$$\frac{\text{velocity at E}}{\text{velocity at } u} = \frac{\text{velocity at E}_1}{\text{velocity at } u_1};$$

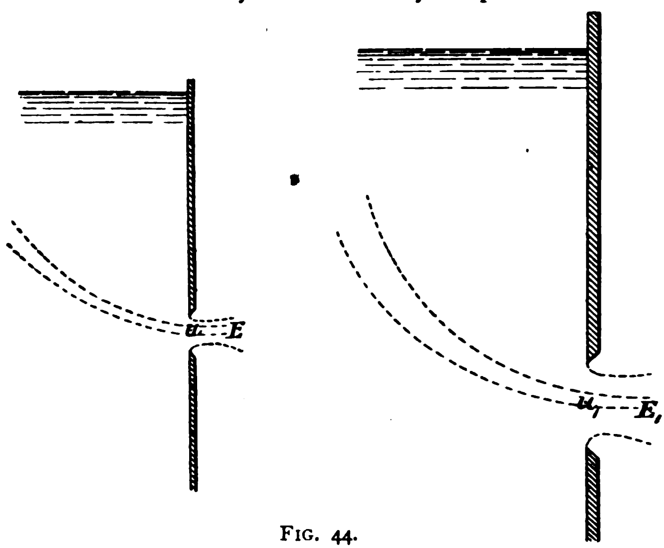


FIG. 44.

and from falls of free levels it follows that

$$\frac{v_E}{v} = \frac{v_E \sqrt{n}}{v_1}.$$

$$\therefore \quad v_1 = v\sqrt{n};$$

this rule applying to any or all homologous points in the two regions of flow.

Applying the rule to Professor Thomson's V-shaped notch, where the notch consists of an isosceles right-angled triangle, the apex (or lowest corner of the notch) being a right angle (Fig. 45), it is evident here that if the depth of the angle of the notch below the level of still water in one notch be to that in another as 1 to n, so all homologous linear dimensions in the two flows will be as 1 to n, the similar areas of little filaments similarly situated being as 1 to n^2, and the velocity of flow as 1 to $\sqrt{n}$, therefore the volume of water flowing per unit time, varying jointly as the area and velocity, will be as 1 to $n^2 \sqrt{n}$. Since this reasoning holds for every pair of similar streams throughout the two flows, the quantity flowing per unit time, Q, $\propto n^{\frac{5}{2}}$.

Instead of considering two separate notches with different streams, we may take the same notch with different depths of water flowing over it; then, if we denote the depth of the vertex of the notch below still-water level by h,

$$Q = c\,h^{\frac{5}{2}}.$$

This notch has the great advantage that the water section is always the same shape, whatever the depth of flow may be.

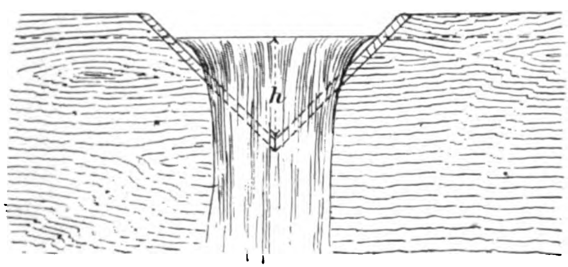


FIG. 45.

Professor Thomson has determined the constant by a large number of accurate experiments, and found that in cubic feet per minute it is (if the notch be sharp-edged) $Q = 0{\cdot}317\,h^{\frac{5}{2}}$, h being in inches; or $Q_1 = 2{\cdot}635\,h_1^{\frac{5}{2}}$ * where h_1 is measured in feet, Q_1 in cubic feet per second.

EXPERIMENT WITH THOMSON WEIR.

The following example shows how the law of flow may be deduced from given experimental data. The values of h_1, the head over the weir, having been obtained by a hook gauge, the corresponding values of Q are noted by weighing the water passing over the weir in a given number of seconds and dividing that amount of water (in

* Rankine and other writers give the coefficient as 2·54, but Professor Thomson's own number, here given, is not likely to be inaccurate.

cubic feet) by the number of seconds taken. Such data are given below.

Values of Q.	Values of h_1.	Log Q	Log h_1.
84·33	4	1·926	0·602
60·88	3·517	1·784	0·550
35·48	2·830	1·550	0·452
12·59	1·84	1·10	0·268
5·861	1·38	0·768	0·140

The values in the third and fourth columns when plotted give the straight line shown in Fig. 46.

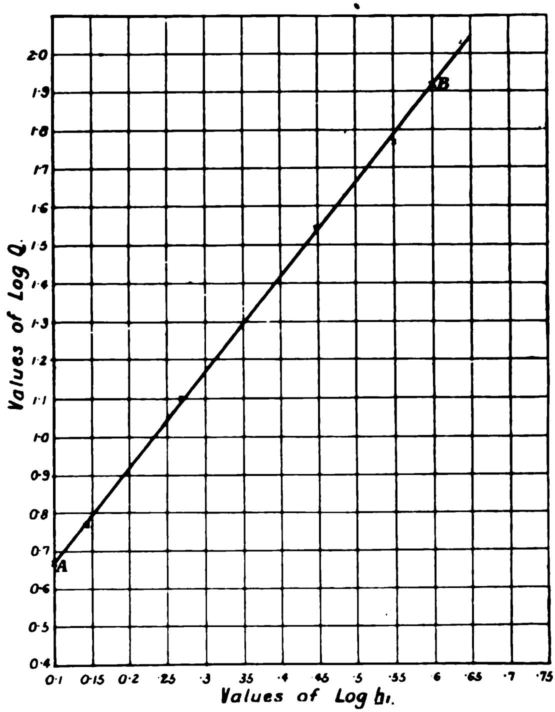


FIG. 46.

The law of this line is $\log Q = a \log h_1 + c$, where a and c are constants.

At A, $\log Q = 0{\cdot}67$, $\log h_1 = 0{\cdot}1$,
and at B $\log Q = 1{\cdot}92$, $\log h_1 = 0{\cdot}6$. Substituting these values in the equation or law, we have

$$0{\cdot}67 = a \times 0{\cdot}1 + c$$
$$1{\cdot}92 = a \times 0{\cdot}6 + c.$$

Subtracting, we have $1{\cdot}25 = a \times 0{\cdot}5$

or $a = 2{\cdot}5$, also $c = 0{\cdot}42$, and the law of the line is as follows:

$$\log Q = 2{\cdot}5 \log h_1 + 0{\cdot}42\,;$$

from which it is evident that

$$Q = 2{\cdot}63\, h_1^{\frac{5}{2}},$$

since antilog $0{\cdot}42 = 2{\cdot}63$.

THE HOOK GAUGE.

For the accurate measurement of head in an experiment such as the foregoing, a hook gauge is necessary. Such an apparatus is depicted in Fig. 47. A brass rod R has a square end F which fits tightly into a socket in a casting which can be fastened to the upper edge of the side of a tank or trough. Several of these sockets may be placed in proper positions on different pieces of apparatus, and thus the same gauge can be applied to each in turn. The rod R is made a good tight sliding fit inside a brass tube bearing a rack B into which works a pinion turned by the milled head A. The pinion and A are on a sliding piece C which is moved up and down on the tube by turning the pinion, and fits the tube very exactly. The arm E, forming part of C, bears the hook H which can be lowered under the water and gradually raised till the sharpened point of the hook just "dimples" the surface of the water.

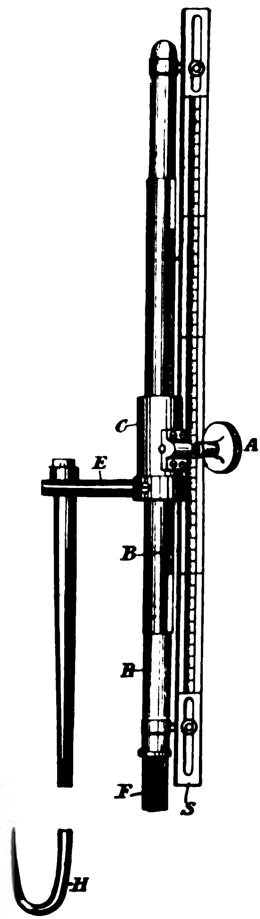


FIG. 47.

In using the a[illegible] F is placed in its socket, the tube carrying B is [illegible] to [illegible] eight such that the rack B shall include the full range of head to be measured. When the water is at its lowest level the hook is lowered, and then raised until its point is in the surface of the water. The scale S is then moved on its two set screws (which work in longitudinal slots), till the zero of the vernier attached to C (not shown) is on one of the principal divisions of the scale. The experiment is proceeded with, and the hook is raised till its point is in the surface of the water when the latter is at its highest level. A reading of the vernier is then taken, and by noting the number of principal divisions on the scale over which the zero of the vernier has passed, the difference of height of the two surface levels is easily read off. This is the head required. In the case of an experiment such as that described on the last page, the head is, of course, taken in still water at a point some distance back from the weir. It is necessary that the part C and all sliding surfaces shall fit well and not shake or move laterally. If the gauge is intended to be permanently fixed in one place the socket is replaced by a strap or stirrup of the same width internally as the side of the tank, this stirrup being screwed to the side of the tank; and such a stirrup with set screws is perhaps better in any case than the socket. The hook H should be capable of being clamped to E at different points if required, so as to allow the hook to be moved further out or in, but this may, to some extent, be accomplished by turning C and S round through an angle. The brass tube may be fitted with a clamping ring and tightening screw if necessary.

Rectangular Weir. Francis' Formula.

In the case of a rectangular notch, Professor Thomson has shown that the formula of Mr. Francis is a rational one. A notch may be made so long relatively to the depth of water on it, that for any increase of length the increase of flow will be proportional to the increase of length. Let $m\,h$ be such a length. In Fig. 48 two portions, each $= \frac{1}{2}\,m\,h$, have been supposed taken off, then over the central part of length $l = L - m\,h$, the flow is proportional to l, if l be varied whilst h remains constant.

The flow through this portion may be regarded as bounded by two vertical planes, and suppose the two remaining parts of the notch to be brought together as in the lower portion of the figure, we can now study the flows separately. In the lower figure the width of the notch bears a constant ratio to h, and $= m\,h$; then by similar reasoning to that employed for the V-shaped notch, we fin[illegible] [illegible]ries with the depth of the water: and if Q_1 represe[illegible] the lower portion, it is easy to show, as before,

$$Q_1 = a\,h^2 \sqrt{h}, \text{ where } a \text{ is a constant coefficient.}$$

Next to find Q_2, the quantity flowing over the central portion. Consider a portion for convenience of length $= h$.

The flow over this portion will be $b\,h^2 \sqrt{h}$, where b is a constant. This is for the length h, hence the flow over unit length is $= b\,h \sqrt{h}$, and for length $l = b\,h\,l \sqrt{h}$. In other words,

$$Q_2 = b\,(L - m\,h)\,h \sqrt{h}.$$

Adding Q_1 and Q_2 to get the flow through the whole notch, we have

$$Q = b\,(L - m\,h)\,h \sqrt{h} + a\,h^2 \sqrt{h}$$

$$= b\,L\,h \sqrt{h} - (b\,m - a)\,h^2 \sqrt{h},$$

or

$$Q = b\left(L - \frac{b\,m - a}{b}\,h\right) h^{\frac{3}{2}}$$

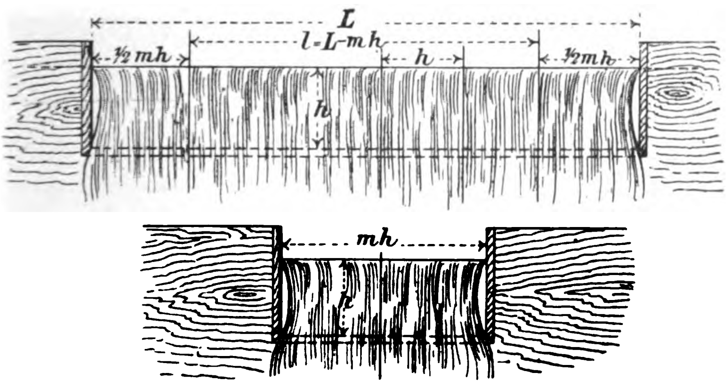


FIG. 48.

$\frac{b\,m - a}{b}$ is a coefficient which evidently depends on b and m, and will be different if, for instance, the stream is contracted at one end of the notch only instead of at both.

It is evident that this rule is similar in form to that deduced by Mr. Francis, which is

$$Q = 3{\cdot}33\left(L - \frac{n}{10}\,h\right)h^{\frac{3}{2}}$$

where n is the number of end contractions.

The variation in the value of n will be understood from the plans of the notch shown in Fig. 49.

Mr. Francis states that his formula is not applicable to cases where the height h exceeds one-third of L, nor is it applicable if h is very small. He is of opinion that it is correct for depths varying from 6 inches to 2 feet. Probably it may be applied to greater depths than 2 feet if the notch be properly proportioned. In a triangular weir, such as Professor James Thomson's, the coefficient is more nearly constant than in weirs of any other section.

Trapezoidal weirs have been tried, being like rectangular weirs in which the sides have been forced outwards so that they slope to the vertical at a slope which, according to Crippoletti, should be 1 in 4.

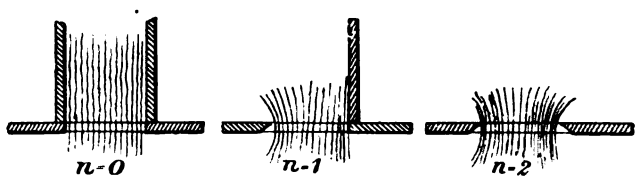


FIG. 49.

Thus the surface of the water section is $\frac{1}{4}h$ longer (neglecting contraction) at each end than the sill of the weir. It is supposed that in this case the flow through the end sections will balance the loss due to contraction.

Hence $Q = 3 \cdot 33\, l h^{\frac{3}{2}}$ (Francis) may be used, but Crippoletti gives the coefficient as 3·367.

If there is velocity of approach, replace h by $h + 1 \cdot 4\, h_1$ where h_1 is the head required to give the velocity of approach. (See below.)

No doubt a similar correction may be necessary in the case of the Thomson weir.

VELOCITY OF APPROACH.

In usual experimental weirs this is of little importance, but in the case of large weirs in rivers it may be important. The theoretic discharge over rectangular weirs is $Q = \frac{2}{3}\sqrt{2g} \cdot b \cdot H^{\frac{3}{2}}$, the velocity of approach being neglected. But if this velocity be considerable, then calculate the head h to give this velocity. The theoretic discharge is $Q = \frac{2}{3}\sqrt{2g} \cdot b(H + h)^{\frac{3}{2}}$.

Mr. Hamilton Smith proposed to modify this, because the velocity is *not* constant across the stream, being greater near the surface than near the bottom, hence he proposed the modification

$Q = \frac{2}{3}\sqrt{2g} \, . \, b\,(H + n\,h)^{\frac{3}{2}}$, where n is a coefficient between 1 and 1·5 (often taken as 1·4).

Francis' method of correcting for velocity of approach is different. His formula (for weir without end contractions) is $Q = 3·33\,l\,h^{\frac{3}{2}}$.

For two end contractions it is $Q = 3·33\,(l - 0·2\,h)\,h^{\frac{3}{2}}$.

If h' = head necessary to give velocity of approach, the formula becomes

$$Q = 3·33\,l\,[(h + h')^{\frac{3}{2}} - h'^{\frac{3}{2}}]$$

for weirs without end contractions, and

$$Q = 3·33\,(l - 0·2\,h)\,[(h + h')^{\frac{3}{2}} - h'^{\frac{3}{2}}]$$

where there are two end contractions.

Submerged Weirs.

For a submerged weir, i.e. one in which the level of the water on the downstream side rises higher than the crest of the weir, Herschel has proposed the formula

$$Q = 3·33\,l\,(n\,h)^{\frac{3}{2}},$$

where n is a coefficient depending on ratio $\frac{h'}{h}$, where h' is head on downstream side of weir and h is that on upstream side. Values of n for different values of this ratio are given below.

n	Ratio $\frac{h'}{h}$
1·00	0·18
0·9725	0·25
0·959	0·50
0·892	0·75
0·866	1·00

Numerical Examples.

(1) In a rectangular weir-gauge, the length of the notch being 5 feet, depth of water 2 feet, find the flow if there is only *one* end contraction (i.e. if only *one* end of the notch has a sharp edge), and compare this with the flow if there are two end contractions.

The formula is

$$Q = 3\cdot 33\left(L - \frac{1}{10}\,n\,h\right)h^{\frac{3}{2}},$$

where n is the number of end contractions.

$$Q = 3\cdot 33\left(L - \frac{1}{10}\,h\right)h^{\frac{3}{2}} \text{ if } n = 1.$$

$$Q' = 3\cdot 33\left(L - \frac{2}{10}\,h\right)h^{\frac{3}{2}} \text{ if } n = 2.$$

$$\frac{Q'}{Q} = \frac{L - 0\cdot 2\,h}{L - 0\cdot 1\,h} = \frac{5 - 0\cdot 2 \times 2}{5 - 0\cdot 1 \times 2} = \frac{5 - 0\cdot 4}{5 - 0\cdot 2} = \frac{4\cdot 6}{4\cdot 8} = \frac{46}{48},$$

or flow with one end contraction is to flow with two as 48 to 46, or 1·0435 times as great.

(2) In a rectangular gauge with two end contractions, the minimum flow being 50 cubic feet per second, find the dimensions of the notch, h being $\frac{1}{3}$ L, for this flow. *Ans.* L = 5·87 feet.
h = 1·957 feet.

(3) A rectangular weir-gauge is employed to measure the flow in a stream. It has sharp edges. The length of the notch is 5 feet and the depth of water 2 feet; find the flow.

Ans. 43·32 cubic feet per second.

(4) A V-shaped Thomson weir-gauge is used to measure the flow in a stream, h being 4 feet; find the flow.

Ans. 84·33 cubic feet per second.

(5) If the water passing through both these notches, with a fall of 25 feet, drive turbines of 0·7 efficiency; and if the dynamos, etc., driven by the turbines have an efficiency of 80 per cent., find the number of kilowatts given out by the dynamos.

If the dynamos light arc and glow lamps, the number of the latter being three times that of the former, find the number of each.

The arc lights take 12 amperes of current at a pressure of 50 volts, and the glow lamps 65 watts each.

Ans. 151·26 kilowatts, 571 glow lamps, 190 arc lamps.

(6) Near a certain town is a river with a fall of 20 feet. The Town Council wish to light a promenade with 25 arc lamps, like the above, and to supply 2500 60-watt glow lamps. Taking the efficiencies as before, what height of water will be required, if the flow be measured by a V-shaped Thomson gauge? *Ans.* 5·34 feet.

Maximum Power from a Given Waterfall.

The power obtainable from a waterfall varies as Q × H, where H is the effective fall; but Q and H are not usually greatest at the same time, owing to the difficulty the tail-water finds in getting away in time of flood. Observations of the height h of the water over a measuring-weir, and the corresponding effective head H should be made to determine the law connecting the two variables Q and H, when the problem can easily be solved mathematically. In a recent turbine installation at Newry, the engineer, Mr. Ball, found that the law was approximately, $H = 6{\cdot}6 - 2{\cdot}4h$. Neglecting the effect of end contractions in the weir, which if the length of the weir be considerable is comparatively small, we have (by Francis' formula) the flow over the weir per second $= 3{\cdot}33\, l h^{\frac{3}{2}}$; or the flow per foot of weir per second, $Q = 3{\cdot}33\, h^{\frac{3}{2}}$.

But the horse-power per foot of weir

$$= \frac{Q \times H \times 62{\cdot}4}{550};$$

$$= \frac{3{\cdot}33 \times 62{\cdot}4}{550}(6{\cdot}6 - 2{\cdot}4h)\, h^{\frac{3}{2}};$$

$$= 0{\cdot}377 \times 6{\cdot}6\, h^{\frac{3}{2}} - 0{\cdot}377 \times 2{\cdot}4\, h^{\frac{5}{2}};$$

$$= 2{\cdot}488\, h^{\frac{3}{2}} - 0{\cdot}9048\, h^{\frac{5}{2}}.$$

Differentiating and equating to zero,

$$\frac{d\,(\text{H P})}{d\,h} = \frac{3}{2} \times 2{\cdot}488\, h^{\frac{1}{2}} - \frac{5}{2} \times 0{\cdot}9048\, h^{\frac{3}{2}} = 0;$$

or

$$3{\cdot}732\, h^{\frac{1}{2}} - 2{\cdot}262\, h^{\frac{3}{2}} = 0.$$

Dividing across by $h^{\frac{1}{2}}$

$$\frac{3{\cdot}732}{2{\cdot}262} = h, \text{ or } h = 1{\cdot}65 \text{ feet nearly.}$$

When, therefore, the flow is such as to give a constant height of 1·65 feet of water over the measuring-weir, the water-level is that which will give the greatest horse-power obtainable from the fall.

The case taken is one in which the fall is low, as it is in such cases that the solution is of greatest importance.

WATER-METERS.

Small or moderate supplies of water are most easily measured by passing the fluid through a water-meter. The first meters invented were for "fluids," including gases, that of W. Pontifex of London (1824) being probably the earliest. Hanson's patent for a meter for gas, water, etc., bears date 1840, and is one of the earliest in which the use of a piston-valve is described.

The Siemens meter is of the "inferential" type, acting on the principle of Barker's Mill. Siemens has also another meter of the direct impact paddle-wheel type. The inferential type of meter, whilst very simple and useful for quick flows, is not suitable for small flows, especially after standing idle for a time, or when the fluid contains dirt.

The Tylor meter is another inferential meter, discharging radially. In these the water acts on paddles or floats. Positive meters are more accurate over greater ranges of flow.

The Kennedy Meter

is a well-known specimen of this class, consisting of a vertical cylinder with a piston moving in it watertight. The piston is nearly as long as the stroke, and it is packed by an india-rubber ring which rolls on the piston, being prevented from coming off by flanges on the piston, which fit the cylinder fairly well. The counting gear is in a separate chamber where it is not under water. The valve is a four-way cock operated by a tumbler which is moved by the piston-rod.

When the piston moves up or down to the end of its stroke this tumbler falls over, reversing the valve and admitting the water to the other end of the cylinder, at the same time opening the end now filled with water to discharge. The *travel* of the piston, in any given interval of time—*not* the number of strokes—is represented on the counting mechanism, a most ingenious system of pawls and ratchets operated by a pinion working into a rack on the piston-rod effecting this result. Comparatively great accuracy at different speeds is thus obtained. The meter, however, is somewhat bulky, and not silent in working.

Schonheyder Water-Meter.

This meter is really a water motor with three cylinders, the pistons of which actuate a counter and so show how often the cylinders have been filled and emptied. The feature of the meter is its rolling cup-shaped valve, which works on the top of a spherical central seat.

The valve has three ports cut right through it and also a central cavity. There is also a central exhaust passage communicating with the discharge pipe. Corresponding to the three ports cut in the valve are three other ports in the fixed valve seat, whilst a central exhaust port corresponds to the cavity of the valve. The valve contains on its periphery three extensions, forming cups in which the ball-shaped heads of the connecting-rods rest. As each piston descends it drags

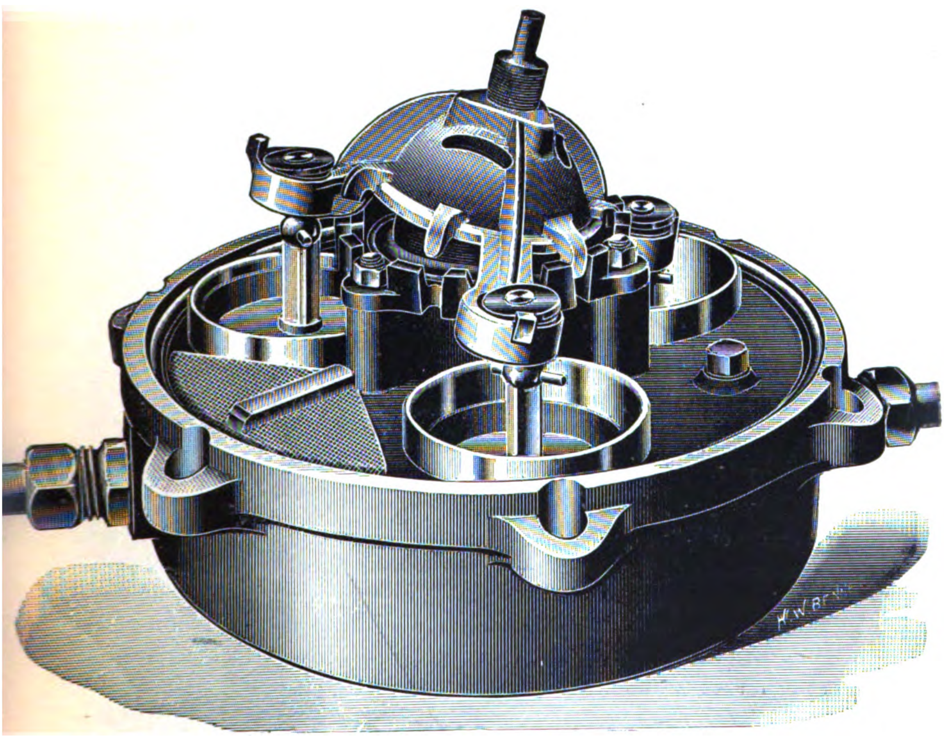


FIG. 50.

down the valve with it until the valve flange comes into contact with the flange round the central pillar. The valve has a rolling or nutatory motion on its seat, but does not rotate, the rocking of the central valve actuating the counting gear. Figs. 50, 51 and 52 show the arrangement.

C^1 C^2 C^3 (Figs. 51 and 52) are the cylinders fixed to the casing, with pistons D^1 D^2 D^3 depending by rods d^1 d^2 d^3 with spherical heads from the valve E. E rests on a similar surface F fixed to the casing by the pillar A^1.

The space G below each cylinder communicates by a lateral passage G with one of the passages H^1 H^2 H^3, leading through the pillar to F, where they terminate in ports h^1 h^2 h^3.

I is the central exhaust passage communicating by I^1 with the discharge pipe B^3; *e* is the central cavity referred to above, which allows any port h^1 to communicate with the exhaust I. If the valve is in the central position all the ports are covered, but a slight inclination of E—which is always assumed in practice—is sufficient

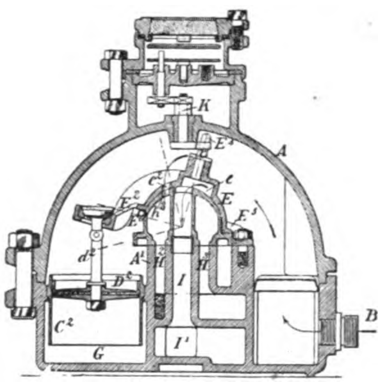


FIG. 51.

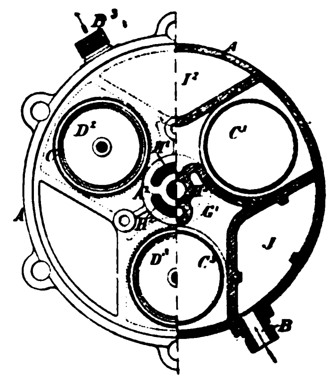


FIG. 52

to uncover a port, and by its cavity *e* allow water to pass. It will thus be seen that the rocking of the valve allows the cylinders to be filled and emptied in succession. The way in which the counting gear is moved will be seen at E^4 and K. The motion is noiseless, and the meter in many cases is accurate to within 1 per cent. both at high speeds and at such very low speeds as, say, half a gallon per hour, maintaining its accuracy for long periods, owing mainly to the peculiar motion of the valve, which causes it to become even more closely fitting by wear. The seat is of vulcanite, and the valve of gun-metal or similar alloy.

The "Frost" or Manchester meter is a well-known positive meter of the packed-piston type.

The Kent "Absolute" Meter

is also a good type of this class. It has two cylinders with pistons and valves, the piston of one cylinder actuating the valve of the other somewhat as in a duplex pump. The way in which the piston is packed will be seen in Fig. 53. The water being admitted through the holes in the piston cover at L and M presses the leathers L and

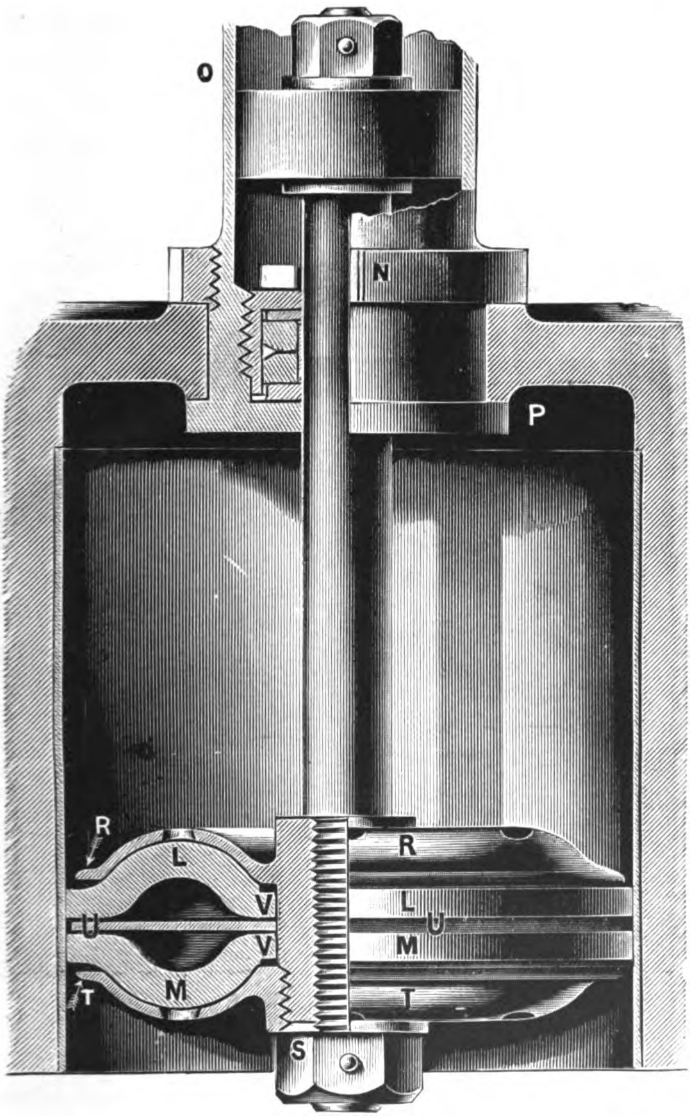


FIG. 53.

M inwards, thus causing the edges of the leathers to fit the cylinder more closely at R and T. Each piston moves a rod which actuates a pawl, advancing the counting ratchet one tooth per stroke.

These meters are tested to within 1 per cent. + or −, at such low flows as one gallon per hour; as well as at the highest flow for which they are designed.

The Venturi Meter.

For very large flows this meter is probably the only one which can be employed without causing an appreciable obstruction. Its use has been developed by the experiments of Mr. Clements Herschel, which established its reliability. It consists (Fig 54) of a double cone

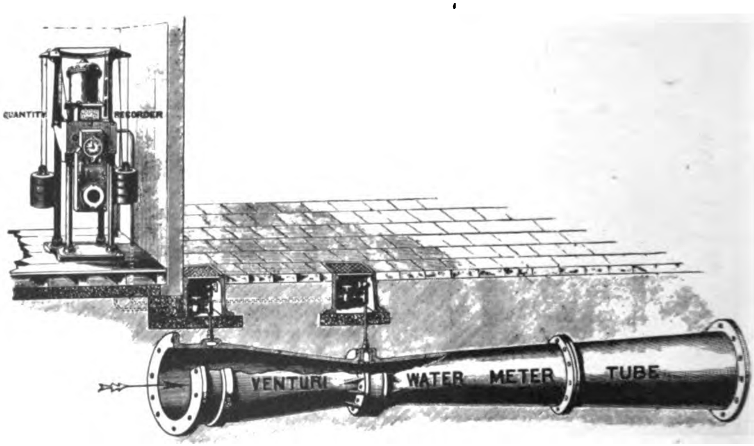


Fig. 54.

which can be inserted in the main, the flow of which is to be measured. The water flowing through the contracted neck of the cones, flows with greater velocity than in the main, and hence under less pressure by the law referred to so often in these pages.

Herschel's Formula for Venturi Water-Meter.

The formula is as follows:

$$Q = c \frac{a_1 a_2}{\sqrt{a_1^2 - a_2^2}} \sqrt{2g(H_1 - H_2)}.$$

Where $H_1 - H_2$ is the difference of head shown by piezometers at up-stream and contracted sections of meter respectively, whose areas are a_1 and a_2 respectively.

The coefficient c = 0·94 to 1·04 (best values 0·96 to 1·01, usually 0·96 to 0·99). When the pressure at a_2 is postive water stands in the central piezometer to the height H_2. When this pressure is negative, air is rarefied and a column of water = h_2 would be raised by syphons placed there. If E is the height of top of section a_2 above datum, then when the pressure is negative use $E - h_2$ instead of H_2 in the formula. For a given meter with given pipes, as long as H_2 remains positive, the flow is proportional to $\sqrt{H_1 - H_2}$, except in so far as c may be affected by varying the velocity of flow. If the proportionality holds it is evidently possible to record values of $\sqrt{H_1 - H_2}$ as indicated in the illustration, and hence of the flow by a recording mechanism. A simple record of difference of pressure, however, will not give Q directly, as Q is proportional to the *square root* of this difference.

American Meters.

A class of meter used almost exclusively in America, and to some extent in this country, has the merit of simplicity, possessing only one moving part (exclusive of the counting gear), this part doing duty both as a valve and piston, a point on it moving usually in a circular, or nearly circular, path.

A typical example is the "Hersey" meter, shown in outline in Fig. 55. The piston A B moving in the casing S, acts also as a valve. It revolves about a centre C, in the cylinder S, which has internal projections with spaces into which the teeth or lobes A of the piston work. In this case the number of teeth is the same as the number of spaces in the casing, hence each tooth works in one space.

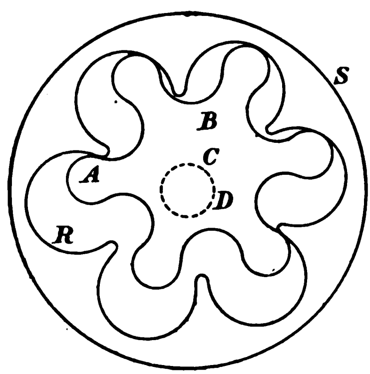
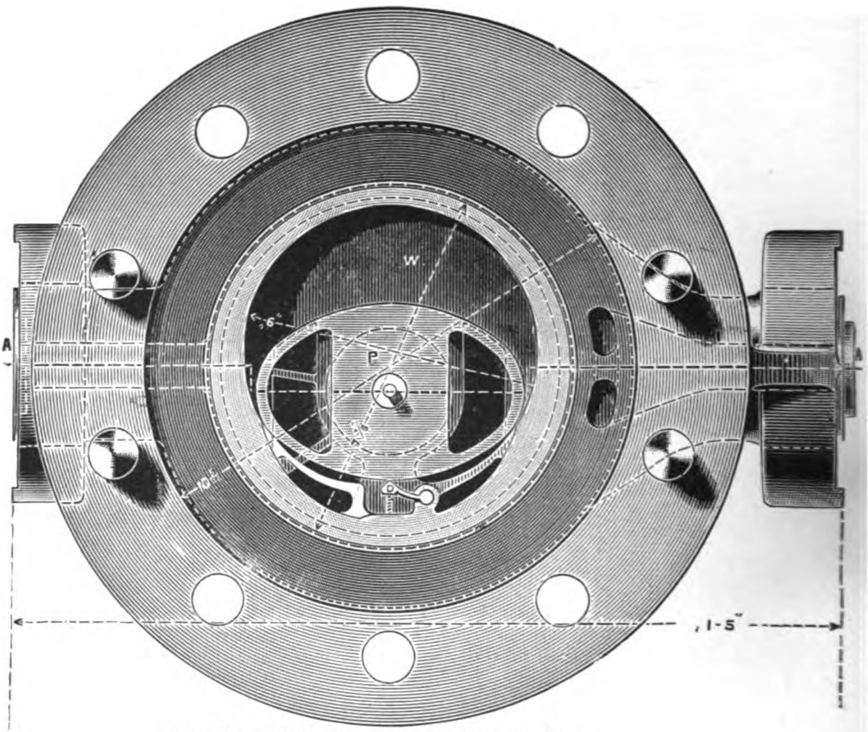


Fig. 55.

The spindle C describes a circle, shown dotted in the figure, the spindle transmitting motion to the counting gear, each revolution of the piston allowing a certain quantity [illegible] luid to pass through the eccentric spaces, this quantity being r[illegible]

These are, therefore, positive [illegible] ese [illegible] the piston or valve has no packing, and yet must be sufficiently free to move, even when the water is charged with small impurities, they are often defective in measuring small flows, whilst the presence of coarser impurities often causes the meter to "stick fast."

In America, where the supply of water allowed per head per day is often as much as 100 gallons, a small inaccuracy is of little con-



PLAN WITH TOP COVER REMOVED

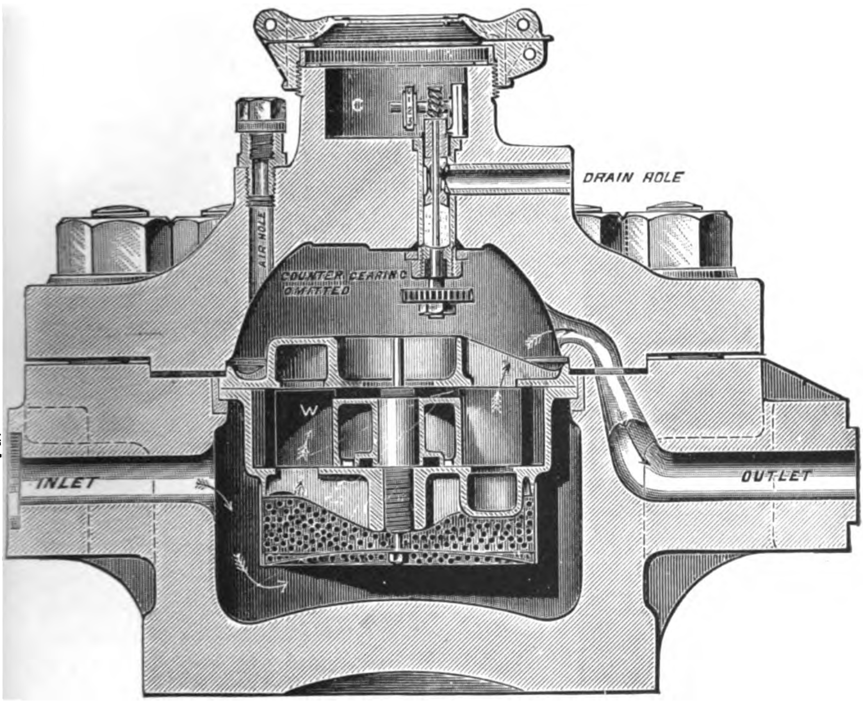
FIG. 56.

sequence, but in this country, where one-fifth of this amount only is frequently allowed, greater accuracy is often necessary.

For measuring pressure water into hydraulic power mains, or to a consumer's plant, great accuracy is necessary, as the water often costs as much as *2s.* or *2s. 6d.* per 1000 gallons, whereas in London, taking 20 gallons per head per day as the ordinary domestic supply, the cost is usually not more than half that amount.

A good meter of the above type is the "Uniform" meter of

Mr. Kent, shown in sectional plan and elevation in Figs. 56 and 57. The hollow piston P is elliptical in plan and revolves about a central cylindrical stud fitting the inner shorter axis of the ellipse exactly. The piston is of vulcanite, working in a gun-metal cylinder, the water entering through the orifices shown under the left end of the piston and helped by water which enters at D, causes the piston to revolve. until it escapes by the outlet at the right-hand end. The pressure of



SECTION THROUGH A.A.

FIG. 57.

water on the little door at D, which has a triangular glass corner abutting against the piston, assists in preserving the fit of the piston under varying conditions. An eccentric pin actuates the counting gear, the position of which is indicated in Fig. 57.

This meter has been a good deal used for ordinary mains, and also for measuring the flow to consumers from hydraulic pressure mains. The illustrations are of a meter for the latter purpose, tested to resist a fluid pressure of 2000 lbs. per square inch.

X.

JET PROPULSION.

Introductory Experiment.

If a jet of water be allowed to issue from a vessel supplied with the liquid from some outside source, as long as the discharge continues there is a force urging the vessel in the opposite direction to that of the flow. This is said to be due to the "reaction" of the jet.

An experimental illustration of this is easily arranged, as shown in Fig. 58, where a suspended vessel, having a hole fitted with a converging mouthpiece N, is supplied with water by the pipe P. If the position of the centre of gravity o of the vessel and water when the orifice is closed be obtained, then, when the orifice is opened, the vessel assumes the position shown by the dotted lines, c being the new position of the centre of gravity. If l be the height from o to the point of suspension, it will be seen from the triangle of forces that W being the weight of vessel and liquid, and F the reactive force of the jet on the vessel,

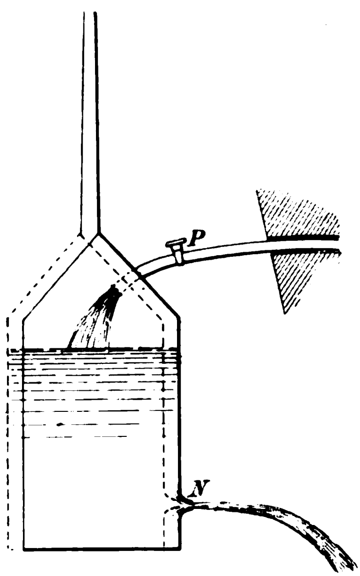


Fig. 58.

$$\frac{F}{W} = \frac{oc}{l}, \quad \text{or} \quad F = W \times \frac{oc}{l}.$$

Now, to obtain F from theoretical considerations, it is only necessary to find the momentum of the water leaving the vessel per second —the supply, being from an outside source, need not be taken into account here, though in some cases, where the water is drawn in by machinery inside the vessel itself, it is most important to consider *how* the water is drawn in.

In this experiment the orifice is, for convenience, so shaped that its area may be taken as equal to the area of the jet, but it will be understood that where these are not equal, it is the area of the *jet* where the stream lines are parallel that must be introduced in the calculation.

If the experiment be carefully carried out, the result agrees closely with the calculated value of the propelling force F.

If the water remains at a constant height of h feet inside the vessel, Q being the rate of flow in cubic feet per second and A the area of the cross-section of the jet in square feet, then

$$Q \times \frac{62 \cdot 4}{32 \cdot 2} \times v$$

is the momentum of the water leaving the vessel per second, and hence the momentum lost per second, which is equal to the propelling force in pounds.

But $v = \sqrt{2gh}$, neglecting the coefficient of velocity, which is about 0·97. Hence,

$$\text{propelling force } \mathbf{F} = \frac{62 \cdot 4}{32 \cdot 2} \times A \times v^2$$

$$= \frac{62 \cdot 4}{32 \cdot 2} \times A \times 2 \times 32 \cdot 2 \times h,$$

or

$$F = 2 \times 62 \cdot 4 \times A \times h;$$

a force equal to the *weight of a column of water whose base is the area of the jet, and whose height is twice that due to the velocity of the jet.*

If the vessel moves under the action of the jet—as in the case where the vessel floats in water—with an average velocity of V feet per second, the work done on the vessel per second is F × V, and the horse-power of the jet, neglecting friction, is

$$\frac{F \times V}{550} = 0 \cdot 227 \times A \times h \times V.$$

Reaction-Wheel or Barker's Mill.

This is a good illustration of the practical application of the principle referred to, the propelling force being due to a constant head of water inside.

The wheel, which was formerly a good deal used in the North, is shown in elevation and plan in Fig. 59; the water being brought by a trough drops into the wheel, which consists of a conical part with radiating pipes, each pipe being bent at its outer end in a backward direction, relatively to the direction of motion of the wheel. If each pipe is bell-mouthed—as it should be—at its inner end, and if we neglect the energy wasted at the bends, etc., then the velocity of the

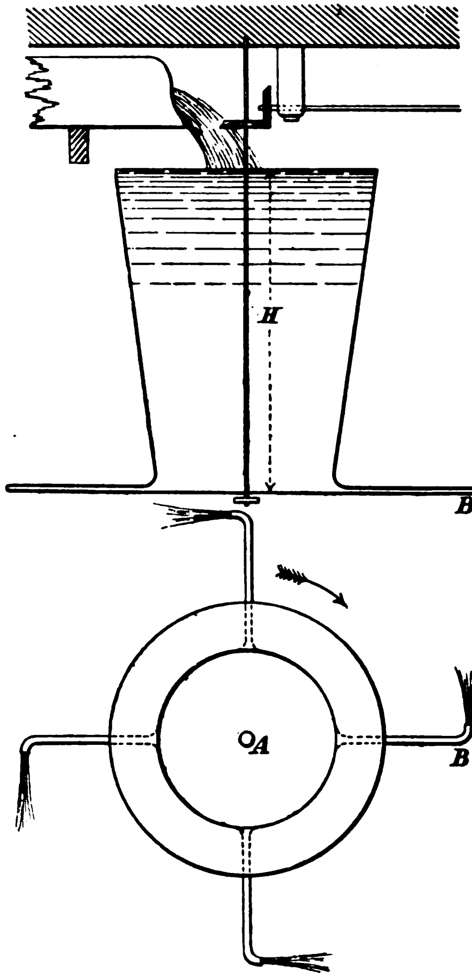


FIG. 59.

issuing water, when the wheel is at rest, will be $v = \sqrt{2gH}$, nearly, the quantity issuing per second (Q) = A v. We may in this case assume the area A to be the combined area of the pipes.

Let u be the velocity of the horizontal pipe at the orifice, the distance of the centre of which from the vertical axis is r. The total head

$$H_1 = H + \frac{u^2}{2g}.$$

Neglecting friction and inertia,

$$v = \sqrt{2gH + u^2},$$

which may be written in the more convenient form,

$$v = 8\sqrt{H + 0{\cdot}614\frac{r^2}{t^2}},$$

where t is the time of one revolution in seconds. The absolute velocity of the issuing water—relative to the earth—is $v - u$, and the momentum generated in it per second is $\frac{wQ}{g}(v - u)$, which by the law of the equality of action and re-action is equal to the propelling force at this point.

The work done per second $= \frac{wQ}{g}(v - u)\,u$, whilst the potential energy lost $= wQH$. The efficiency, on the assumption made above, is

$$= \frac{(v - u)\,u}{gH}.$$

Friction and inertia oppose centrifugal force, and it is a matter which experiment alone can decide, what the actual value of v will be in a given case, and what value of u will give the highest efficiency.

The rule for turbines $u = 0{\cdot}66\sqrt{2gH}$ is sometimes taken though $u = \sqrt{2gH}$ gives a higher efficiency. To apply these rules to an example, let the speed be 10 revolutions per minute ($t = 6$), $H = 10$, $r = 5$, whence $v = 25{\cdot}8$, and from the rule for turbines, $u = 15{\cdot}8$, the efficiency being 49 per cent. If the speed be 20 revolutions per minute ($v = 27{\cdot}38$), the efficiency is 55 per cent.; and at 40 revolutions per minute ($v = 31{\cdot}2$) the efficiency works out at 75 per cent. In fact the efficiency increases as u increases, but, if friction and inertia are taken into account, this does not hold true. The highest practical efficiency is about 60 per cent.; often not much more than half this is obtained.

The cross-section of the vertical pipe should be designed from the ratio

$$\frac{A_1}{A} = \frac{\sqrt{H}}{\sqrt{d}},$$

where A_1 is the area of a section at depth d, A being the area of that at depth H.

JET-PROPELLED BOATS.

In the foregoing examples the velocity of the jet was supposed to be due to a constant head of water maintained by an outside supply. Consider a vessel propelled by a jet, the water for which is drawn into the vessel by pumps or other mechanism inside the vessel itself. It is now all-important to consider *how* the water supplying the jet is drawn in, because the water outside has a certain momentum relative to the moving vessel which may or may not be partially utilised.

If the velocity of the vessel be u feet per second relative to the sea, a mass m has momentum $m\,u$ relative to the vessel before entry. Suppose we could scoop up this water at the bows by a gradually sloping pipe without any loss, then if the jet issues backward with a velocity v the momentum in this direction is $m\,v$. Evidently, if $m\,v = m\,u$ there is no propelling force, $m\,v - m\,u$ is the momentum given by the pumps to the water, and if there is no loss this is also the momentum given by the jet to the ship, since the pumps have not to draw the water in. Let Q be the volume of water leaving the ship per second, then the propelling force

$$F = (v - u)\,\frac{Q\,w}{g}.$$

The backward kinetic energy with which the water again reaches the sea is (relative to the sea)

$$\frac{(v - u)^2}{2}\,\frac{Q\,w}{g},$$

which is the work lost per second.

The work utilised per second is

$$(\text{Force} \times \text{distance}) = (v - u)\,\frac{Q\,w}{g} \times u,$$

and the efficiency of the jet

$$= \frac{\text{work utilised}}{\text{work utilised} + \text{work lost}}$$

$$= \frac{(v - u)\frac{Q\,w}{g}u}{(v - u)\frac{Q\,w}{g}u + \frac{(v - u)^2}{2\,g}Q\,w};$$

or

$$\text{Efficiency} = \frac{u}{u + \frac{v - u}{2}} = \frac{2\,u}{u + v}.$$

This expression must not be used when $u = v$, as the efficiency then $= 1$, but there is no propelling force. This may be regarded as the case in which the water simply passes through the boat without friction from the bows to the stern. If v and u are nearly equal the efficiency is high, but the propelling force is small. We can never in practice realise the ideal here assumed of utilising *all* the momentum of the feed water.

The case in which the water is drawn in vertically might next be considered. None of the relative momentum is, in this case, utilised. The water may even be drawn in from the stern, when it will be necessary to give to it a forward velocity a little greater than u in order that it may reach the pumps. If it be discharged backward with the velocity u there is no propelling force, though work has been done by the pumps. The three methods have been here referred to in the order of their efficiencies, the first method being clearly the best.

This is very evident from Rankine's formula for the efficiency of a jet, which is

$$\text{Efficiency} = \frac{\frac{w\,v\,s}{g}}{\frac{w\,v\,s}{g} + \frac{w\,s^2}{2\,g} + f\frac{w\,v^2}{2\,g}}$$

Where w represents the weight, in pounds, of water discharged per second ;

v represents the speed of the vessel in feet per second ;

s represents the slip, or acceleration, or additional velocity imparted by the pumps ;

f represents a coefficient depending on the completeness with which the velocity of feed is lost.

(If water is well taken in, as in Case 1, f may be as low as $0{\cdot}033$. If the velocity of feed is all lost, $f = 1$.)

$$\text{If } f = 1, \text{ the efficiency} = \frac{2vs}{(v+s)^2}.^*$$

and is a maximum when $s = v$.

The following actual experimental numbers, taken from Mr. Barnby's paper on 'Hydraulic Propulsion,' † will illustrate the use of the formula.

Discharge 1 ton per second ($w = 2240$).
Velocity of discharge 37·25 feet per second ($v + s = 37{\cdot}25$).
Velocity of vessel 21·4 feet per sec. ($v = 21{\cdot}4 \therefore s = 15{\cdot}85$).

$$f = \frac{\text{loss of velocity of feed}}{\text{actual velocity of feed}} = 0{\cdot}0374.$$

These data substituted in the formula give the jet efficiency as 0·7; the efficiency of the pumps was 0·46, and the mechanical efficiency of the engine 0·76.

It will be seen that whilst the jet itself is as efficient as a screw propeller, the low pump efficiency reduces the resultant efficiency much below that of propeller machinery.

The reader must be cautioned against falling into the common mistake of supposing that it matters whether the jet is sent out above or below the sea-level. As a matter of fact, it makes no difference to the reactive force due to the jet whether the discharge orifice is above or below water. This is evident from the preceding calculations, as we are concerned merely with the resultant momentum lost per second, not at all with the effect of the jet on the sea or air against which it impinges.

Hydraulically Propelled Lifeboat.

Jet propulsion has, however, special advantages which make it peculiarly suitable for lifeboats where heavy seas have to be encountered, and where the consequent racing of the engines or grounding of the boat may cause failure of engines, propeller, or paddles. Lifeboats have recently been built with jet propulsion, and they seem to answer very well.

Figs. 60 and 61 give two views of the *City of Glasgow* lifeboat, recently built for the Harwich Station by Messrs. R. and H. Green, of Blackwall, London. The boat is 53 feet long, 16 feet beam, and 5½ feet deep, with a mean draught, on trial, of 3 feet 3 inches, and a displacement of 30 tons.

* Rankine's *Scientific Papers*. † *Proceedings Inst. C.E.*, 1884.

In regard to the machinery, referring to the figures—which show the situation of the machinery and orifices—it will be seen that

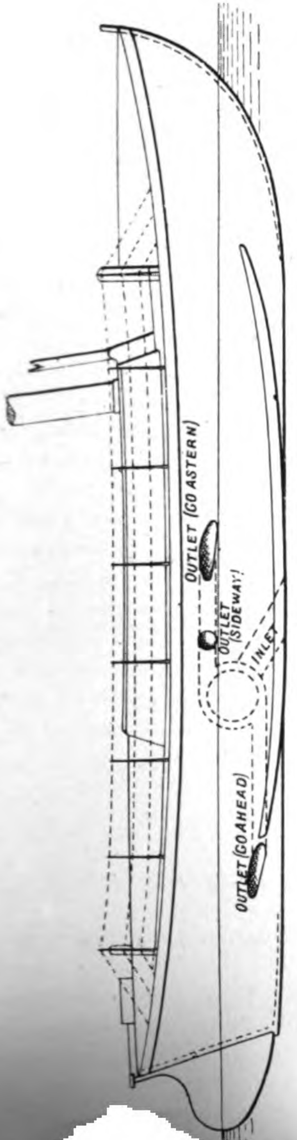


FIG. 60.

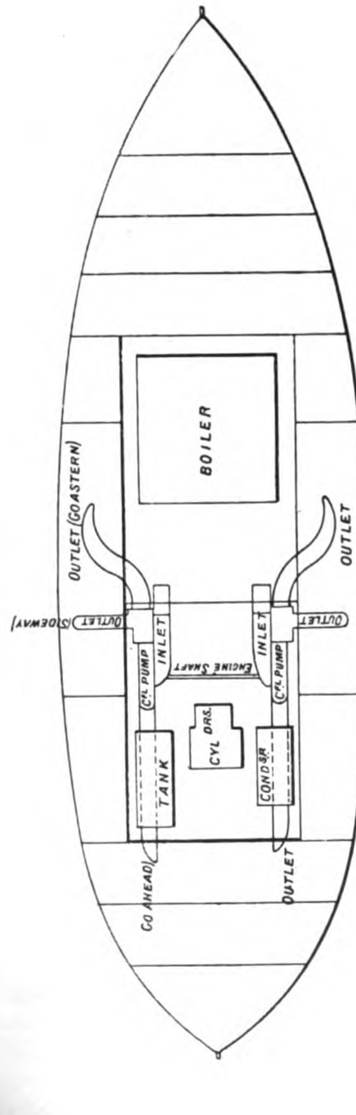


FIG. 61.

each end of the shaft of the compound engine is connected to a centrifugal pump 2 feet 6 inches in diameter, which draws in water through an inlet orifice in the bottom of the boat, each passage from the pump to the orifice sloping forwards and downwards, so that some of the momentum of the water may be utilised. Each pump forces the water out in a jet through a 12-inch hole in the sternwards direction, and *under water*. This is probably more convenient when leaving another vessel or a landing stage, than if discharged above water. Similar jets—in this case discharging above water—propel the boat astern when necessary, and an orifice on each side is provided for lateral propulsion, should that be required, to prevent the boat from bumping against a wreck, or to facilitate her getting away. The boiler is of the water-tube type, which, with the machinery, was constructed by Messrs. Penn. On trial, the boat proved in every way satisfactory, both as regards speed and manœuvring capabilities, the speed on the measured mile being 8½ knots, engines 360 revolutions per minute, steam pressure 115 lbs. per square inch, indicated horse-power 200. Another boat of this kind, the *Duke of Northumberland*, had previously been constructed by the same firm.

No information is, so far as we know, yet available as to the effect of the rolling of the boat on the bearings of the pumps. If the pumps were driven by belting or gearing, the rolling of the boat, forcing the rapidly revolving pump shaft to assume different angles, would probably cause wearing of the bearings in a direction at right angles to the plane in which the shaft oscillates, from the well-known resistance which a revolving body offers to a change of the direction of its axis. This resistance is proportional to the moment of inertia and to the angular velocity of the body. The method of coupling the pumps direct to the engine shaft, in all probability, gets over the difficulty.

Pressure of a Jet against a Surface.

In the foregoing the reader's attention has been directed to the propulsive effect of a jet on the vessel from which it issues. It is sometimes necessary to find the *pressure* of a jet on a surface against which it strikes. Without going fully into the matter, which is beyond the scope of this work, the graphic solution shown in Fig. 64 will give all that is usually required.

The jet is deflected through an angle β. Construct an isosceles triangle A C B, each side representing v, the vertical angle being β. The change in direction of motion in one second is represented by

A B; or, if the scale be suitably altered, A B will represent the *change of momentum per second.* It is easy to see that

$$A\,B = 2\,v \sin\frac{\beta}{2},$$

and the change of momentum per second

$$F = 2\,w\frac{Q\,v}{g}\sin\frac{\beta}{2} = (\text{since } Q = A\,v) \text{ to } 4\,w\,A\,H \times \sin\frac{\beta}{2}.$$

This pressure may be resolved into two components, F_p parallel and F_n normal to the original direction of the jet. These are found from the rectangle A D B E as shown. Evidently

$$F_p = w\frac{Q\,v}{g}(1 - \cos\beta),$$

and

$$F_n = w\frac{Q\,v}{g}\sin\beta.$$

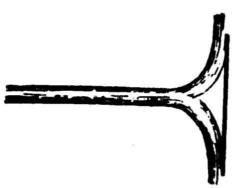


FIG. 62.

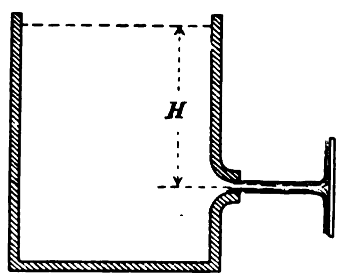


FIG. 63.

If $\beta = 90°$, as in Figs. 62 and 63,

$$F_p = \frac{w\,Q\,v}{g}.$$

If $\beta = 180°$, as in Fig. 65, $F_p = 2\frac{w\,Q\,v}{g}$, since $\cos\beta = -1$.

In Fig. 63, if the plate be placed as shown, the pressure on it is

$$w\frac{Q\,v}{g} = 2\,w\,A\,H, \text{ since } Q = A\,v.$$

Now let the plate move up to the vessel till it closes the orifice, the pressure on it is only w A H, or half what it was before. This is sometimes regarded as a paradox, but the forces compared are obtained in totally different ways.

If the plate in Fig. 64 moves in the direction of A C, with a velocity of translation u, v being the velocity of the jet,

$$F_p = \frac{w\,Q}{g}(v - u)(1 - \cos\beta),$$

and the energy transmitted per second to the plate is (neglecting losses)

$$F_p u = \frac{w\,Q\,u}{g}(v - u)(1 - \cos\beta).$$

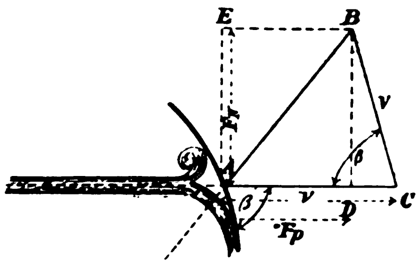


FIG. 64.

FIG. 65.

This is greatest when $u = \frac{v}{2}$, and is then $= \frac{w\,Q\,v^2}{2\,g}$ when $\beta =$ 180° as in the Pelton wheel.

Experiments show that the pressure of a jet is usually less than that given by the above theory—probably owing to eddy-losses—but it approximates most closely to it in the case of the Pelton wheel.

PRACTICAL APPLICATIONS OF ABOVE RULES.

An undershot water-wheel gives a good example. Formerly these wheels had floats consisting of flat boards fixed radially, the efficiency of the wheel being about 0·3.

Poncelet improved them by introducing curved floats, so curved that the water enters without shock and leaves without energy in the direction of the wheel's motion. With this arrangement an efficiency of 0·6 is possible. Fig. 66 shows how the first condition is fulfilled. v is the velocity of the jet, v_t the tangential velocity of the wheel, by completing the parallelogram of velocities as in the figure, v_r the velocity of the jet relative to the wheel is obtained. The float or vane at its outer edge must be tangential to the side representing v_r.

To fulfil the second condition, the fall of the water relative to a horizontal line from its highest point on the moving vane, should be the same as its rise to that point. Draw the vane in the position of entry and exit of the water, both, as assumed above, touching at

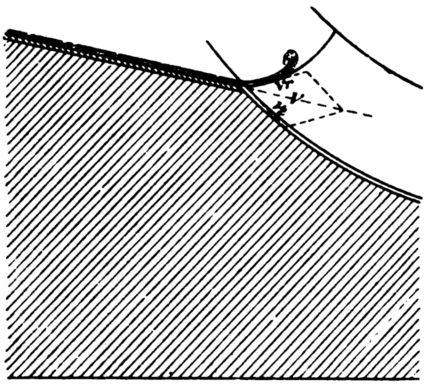


FIG. 66.

the outer edge a horizontal line. Bisect the angle contained by radii drawn from the vane points to the centre of the wheel. Then it is easy to show that (neglecting friction) the vane should make with the circumference an angle = this half angle. This gives a usual vane angle of about 15°.

An application of the laws for the pressure of a jet on a moving surface may be found in the

PELTON WHEEL.

This consists of a wheel with a series of cups fastened at equal intervals round its circumference, into which water from a jet is directed; the cups being so shaped internally that the jet is returned practically parallel to its original direction.

Fig. 67 shows a perspective view of the usual type of wheel, whilst Fig. 68 shows a form with which more than one jet can be used, thus increasing the power of the wheel two or threefold. Pelton wheels have come very much into use in America. They are very efficient for high "heads," but are not recommended for falls of less than 30 feet, whilst a fall of 2000 feet is often employed. By changing the nozzle the power obtainable from a given wheel may be varied. Their efficiency varies from 80 to 86 per cent. under favourable conditions.

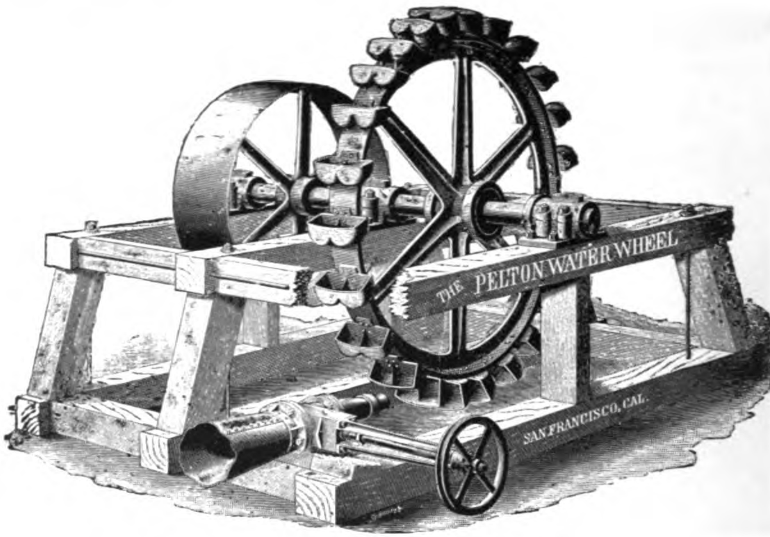


FIG. 67.

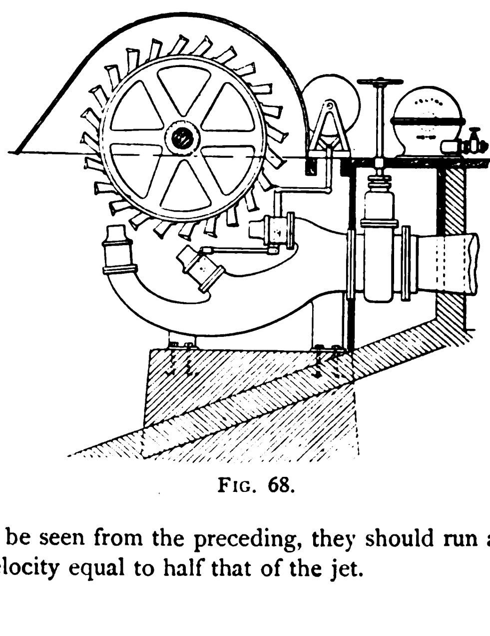


FIG. 68.

As will be seen from the preceding, they should run at a circumferential velocity equal to half that of the jet.

Applying the rule already given, and assuming $v = \sqrt{2gH}$, where H is the "head" of the jet, the student can work out the following examples. Remember work of jet per second $\div$ 550 is water horse-power, or the horse-power actually lost by the water.

Numerical Examples.

1. A jet 1 inch in diameter and with a head of 10 feet, impinges on a plane surface at right angles to it. If the velocity of the jet is 0·97 of that due to the head, find the pressure of the jet on the surface.

Ans. 6·4 lbs.

2. If a jet of the same area, and with the same velocity as the last, impinges on a surface making an angle of 60° with its direction, find the amount and direction of the resultant pressure due to the jet on the surface.

Ans. 6·4 lbs. Direction 60° with original axis of jet.

3. Find the pressure, in its own direction, of the same jet acting on the concave surface of a hemispherical cup, symmetrically situated with respect to the axis of the jet. *Ans.* 12·8 lbs.

4. Pelton wheel, 2 feet diameter, speed 821 revolutions per minute, pressure of water 200 lbs. per square inch, 100 cubic feet per minute being utilised; find the water horse-power. If the actual horse-power is 70·3, find the efficiency.

Ans. 86·8 horse-power. Efficiency 81 per cent.

5. Wheel 12 inches diameter, head 170 feet, speed 997 revolutions per minute, flow 11·39 cubic feet per minute, actual horse-power 2·92; find the water horse-power and efficiency.

Ans. 3·69 horse-power. Efficiency 80 per cent.

XI.

NOZZLES AND JETS.

A nozzle somewhat of the shape shown in Fig. 69 is often used for fire-hoses.

The coefficient of discharge is high; probably that for a plain cone would be as high.

To find the velocity with which the water leaves the nozzle, let the level of the orifice be datum. Then the energy in ft.-lbs. of each pound of water at A B or C D is $2 \cdot 3 p_1 + \frac{v^2}{2g}$, p_1 being the pressure in lbs. per square inch obtained by a pressure gauge at A B, when the water is flowing, and v the velocity there in feet per second.

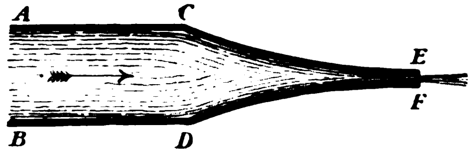


FIG. 69.

Assume the pressure to be zero, at or just outside the orifice, then the total head, i.e. the velocity head + the pressure head at A B is available to give a velocity V at the orifice. Then

$$V = c_1 \sqrt{2g\left(2 \cdot 3 p_1 + \frac{v^2}{2g}\right)}.$$

Also $D^2 v = d^2 V$, where D and d are the diameters at C D and E F respectively. Eliminating v, we have

$$V^2 \left\{ \frac{1}{c_1^2} - \left(\frac{d}{D}\right)^4 \right\} = 4 \cdot 6 g p_1,$$

whence

$$V = c_1 \sqrt{\left(\frac{4 \cdot 6 g p_1}{1 - c_1^2 \left(\frac{d}{D}\right)^4}\right)} = \frac{c_1 \times 12 \cdot 16 \sqrt{p_1}}{\sqrt{1 - c_1^2 \left(\frac{d}{D}\right)^4}},$$

c_1 being the coefficient of discharge.

c_1 for a well-shaped nozzle of the kind shown may be taken as about $0 \cdot 97$, which gives $V^2 = \dfrac{139 \cdot 2 p_1}{1 + 0 \cdot 94 \left(\frac{d}{D}\right)^4}$.

In the case of a jet driving a Pelton wheel, if the pipe conveying the water be straight, L feet long and D feet in diameter, without bends, valves, or other obstructions, the skin friction is $\frac{\lambda v^2 L}{D}$ as given at page 43. If H be the total head available at entrance

$$H = \frac{\lambda v^2 L}{D} + \frac{V^2}{2g}; \qquad \text{or} \qquad \text{since } v = \frac{d^2}{D^2} V$$

$$H = V^2 \left\{ \frac{\lambda L d^4}{D^5} + \frac{1}{2g} \right\}.$$

From this $$V = \sqrt{\left(\frac{H}{\frac{\lambda L d^4}{D^5} + \frac{1}{2g}}\right)},$$

a convenient rule for the velocity of the jet when H, d and D are known.

Example 1.—If D = 1 inch, $d = \frac{3}{8}$ inch, $p_1 = 60$, find the velocity of efflux and the height to which the jet will rise, neglecting resistance of air, etc. *Ans.* 90·57 feet per second. Height 127·4 feet.

Example 2.—Neglecting obstructions, what will be the velocity of a $1\frac{1}{2}$-inch jet driving a Pelton wheel, the pipe conveying the water being smooth, 6 inches in internal diameter, and half-a-mile long, the head available being 630 feet? *Ans.* 198·2 feet per second.

Nozzles for Fire-Hoses (Experimental Data).

The two forms of nozzle most in use are (1) plane conical with a diaphragm at end having a small circular hole through which the water issues; and (2) hyperboloidal or conoidal surface of revolution like that shown in Fig. 69.

From the Ellis experiments, the following are the heights to which a given nozzle with the stated head will throw the jet. The figures (1) and (2) denote nozzles of kinds described above after (1) and (2) respectively.

Pressure p_1, in lbs. per square inch	Head in feet.	Height of Jet, Nozzle 1 inch diam.		Height of Jet, Nozzle $1\frac{1}{8}$ inch diam.		Height of Jet. Nozzle $1\frac{1}{4}$ inch diam.	
		(1)	(2)	(1)	(2)	(1)	(2)
10	23	22	22	22	22	23	22
20	46	43	42	43	43	43	43
30	69	62	61	63	62	63	63
40	92	79	78	81	79	82	80
50	115	94	92	97	94	99	95
60	138	108	104	112	108	115	110
70	161	121	115	125	121	129	123
80	184	131	124	137	131	142	135
90	207	140	132	148	141	154	146
100	230	148	136	157	149	164	155

Coefficients of discharge for smooth conical nozzles without diaphragms :

In. diam.		In. diam.	
¾ . .	= 0·983	1⅛ . .	= 0·976
⅞ . .	= 0·982	1¼ . .	= 0·971
1 . .	= 0·972		

For square ring nozzles the coefficient is about 0·74.

The Ball Nozzle.

This nozzle, shown in Fig. 70, promises to be of great use for many purposes. The ordinary nozzle emits a jet of great velocity, which can, therefore, be directed to a considerable height, but it covers a very small area, and in case of a fire which is in an easily accessible position, does not answer well. The ball nozzle, which consists of a nozzle terminating in a cup in which a ball is loosely seated, gives an umbrella-shaped spray of great value for quenching flame and smoke.

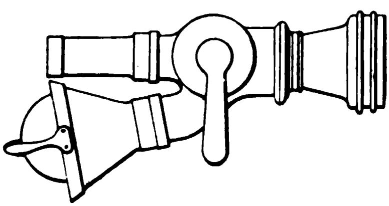


Fig. 70.

It might at first sight be supposed that the pressure of the jet on the ball would tend to drive it away from the nozzle, but such is not the case. Fig. 71 will explain the reason of this.

If a jet issues from an orifice *a*, and impinges upon a flat plate P, we know how to calculate the force F, necessary to keep the plate in position *when it is some distance from the orifice.*

Now let P be brought nearer and nearer : when it reaches some such position as that shown, F diminishes rapidly till as the plate nears the orifice it is finally sucked in towards *a*, stopping the flow. As soon as the flow is stopped, the plate experiences the hydrostatic pressure due to the head of water in the vessel, which forces it away from the orifice, and the action is repeated as before. An intermittent spray is thus produced, but if the plate does not fit the surface perfectly a continuous spray may be obtained. The explanation of the phenomenon is easy. Since the corners are rounded, little energy is lost from *a* to *b* or *c*. Thus, neglecting *h*, for every pound of water $\frac{v^2}{2g} + 2\cdot3\,p$ is constant. But the area round *b c* is much greater than that at *e d*, hence the velocity at the latter section must be much greater than that at the former; therefore, if the pressure at *c b* be atmospheric, that at *d e* will be less than atmospheric, and the pressure of the air on the outside of the plate will force it up to the orifice.

In the case of the ball nozzle a similar action takes place: the ball not being able to completely close the orifice, spreads the issuing jet into an umbrella-shaped cascade.

The nozzle shown in the illustration has two branches, one consisting of the ordinary straight nozzle, which may be used for projecting a jet to some distance, whilst the ball nozzle on the other branch may be used for purposes which require a spread or sprayed jet. It is used with low pressures.

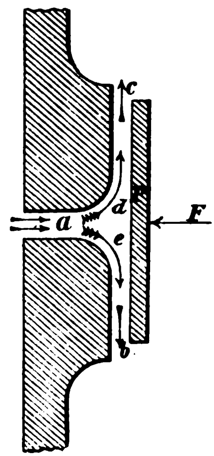


FIG. 71.

TENDER APPARATUS FOR PICKING UP WATER.

In the foregoing cases the nozzle is at rest and the water moves. Consider a case in which the nozzle moves relatively to water at rest.

Fig. 72* gives two views of the apparatus provided on some express locomotives for picking up water without stopping.

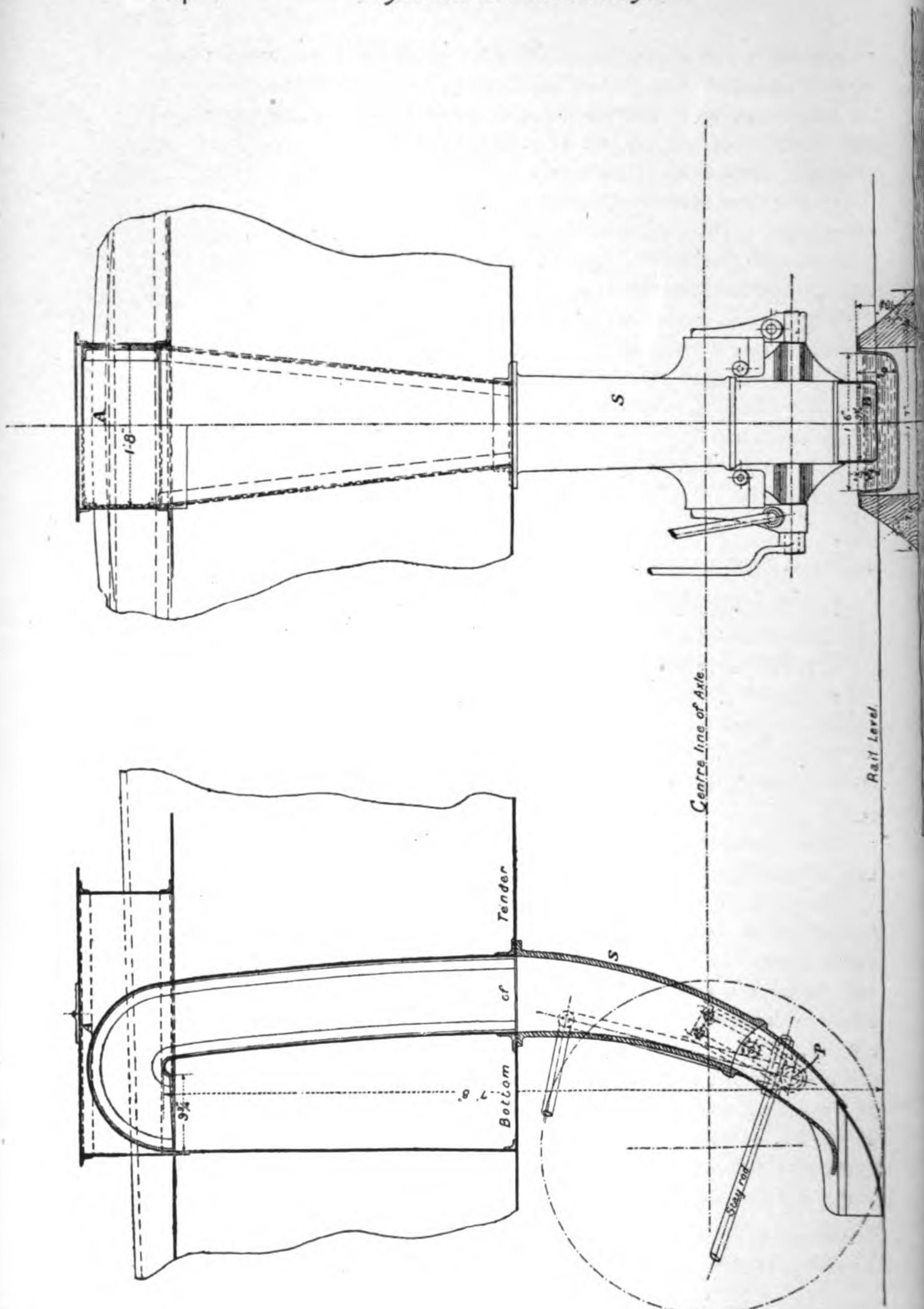
A long shallow tank, shown in section in the right-hand figure, is fixed between the rails, and is kept filled to the requisite height with water.

The trough has no ends, but the rails and trough are slightly raised near the terminations of the trough so as to retain the water.

A scoop S, curved so as to point in the direction of motion, projects downwards from the tender, this scoop being furnished with a mouthpiece which can be turned about P so as to lift it out of the way when not wanted. If the speed of the engine be sufficient, the water which enters the mouthpiece finds its way up the pipe into the tender.

Suppose the height of A above B, the surface of the water, to be H feet, then every pound of water, when it reaches A, has gained H ft.-lbs. of potential energy. Let the level of B be datum; then since the pressure of the water is atmospheric, the kinetic energy, imparted to each pound of it at entrance minus the energy necessary

* Inserted by the courtesy of Mr. F. W. Webb, formerly Chief Mechanical Engineer, London and North Western Railway.


A
S
B
Centre line of Axle
Rail Level
Bottom of Tender
Stay rod
P
S

to overcome resistances is equal to the energy—kinetic and potential—it has at A.

Let V be the velocity of the scoop relative to the water in the tank, and v the velocity of the water at A, F being the coefficient of hydraulic resistances. Then

$$F \times \frac{V^2}{2g} = H + \frac{v^2}{2g}.$$

Suppose F to be 0·5, the speed of the locomotive 30 miles an hour and H 8 feet, find the velocity at A. Find also the least speed sufficient to raise the water.

30 miles an hour is 44 feet per second.

$$\therefore \quad 0 \cdot 5 \times \frac{44^2}{64 \cdot 4} - 8 = \frac{v^2}{64 \cdot 4},$$

whence $v = 21 \cdot 15$ feet per second.

If v be zero we get the limiting speed. In this case

$$0 \cdot 5 \times \frac{V^2}{64 \cdot 4} - 8 = 0,$$

which gives V = 32·1 feet per second, or 21·9 miles an hour.

The speed of the locomotive must be in excess of this in order to fill the tank. Knowing the quantity to be supplied, the time available, and the area of the discharge pipe, the necessary velocity for the water in it can be found. Some allowance must also be made for friction.

Injector Hydrant.

In this apparatus, due to Mr. Greathead, a high-pressure jet is used to intensify the pressure of water from ordinary mains, so as to give a jet of sufficient pressure to reach the tops of the highest houses.

The jet taken from the high-pressure pipes is a small one, the main volume coming from the ordinary mains.

The way in which the high-pressure jet is used to intensify the pressure of the larger supply will be understood from an inspection of Fig. 73, which represents a section of the injector used at the docks of the Manchester Ship Canal. The pressure supply from the hydraulic mains not only raises the water from the docks, but gives it sufficient pressure to enable it to be used in large quantity if required at a height well above the roofs of the highest buildings on the dock quays; some of the buildings being of seven stories, are

71 feet above the ground level. Since the hydraulic mains and accumulators are always charged with pressure supply, the appliance is most valuable in securing prompt suppression of fire, which is of special importance where cotton and other inflammable goods are dealt with. Mr. Ellington has invented an *automatic* injector hydrant for use in conjunction with automatic sprinklers; the water for the latter, being drawn mainly from the ordinary street mains, a tank on the premises, or other handy supply, is intensified in pressure and delivered at the sprinklers. The injector is intended to supersede steam fire pumps in public buildings, hotels, warehouses, etc.

The apparatus consists of one or more injectors which deliver into a discharge pipe leading to the sprinklers, also a loaded accumulator in communication with that pipe; this accumulator by its rise and fall operates a valve which controls the supply of high-pressure water from the hydraulic mains to the injectors. The accumulator is kept slowly moving up and down within a certain range, a certain

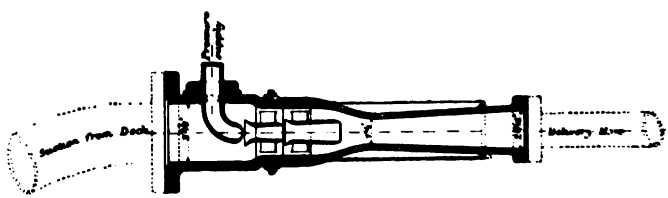


FIG. 73.

leakage being permitted for this purpose. This ensures the apparatus being always ready for action. In the event of fire, a considerable draught being made on the discharge main by the sprinklers coming into action, the accumulator falls quickly, opening the valve freely, and allowing a free passage of water from the hydraulic mains to the injectors. If the flow through the injectors be greater than that required, the pressure in the discharge main is increased, the accumulator rises and partially cuts off the high-pressure supply. Should the discharge cease altogether, the accumulator rises rapidly, closes the valve, and immediately resumes its slow up-and-down motion under the control of the automatic gear. Experimental data (by Professor Robinson) show that with such appliances there is a considerable waste of energy in some cases.

Thus, with a low-pressure supply at a pressure of 20 lbs. per square inch, 32·4 gallons per minute at a pressure of 700 lbs. per square inch are required to intensify the pressure of the delivery of 150 gallons per minute to a pressure corresponding to 100 feet head, i.e. to $\frac{100}{2 \cdot 3}$ or 43½ lbs. per square inch; but with a low pressure supply under a head of 138 feet, only 3·7 gallons per minute are required from the high-pressure mains.

The advantage of diminishing the difference of pressure of the two supplies is clearly shown in the following table, compiled from the experimental data already referred to.

Quantity from Low-pressure main.	Pressure Head of this Supply.	Velocity due to this Head.	Energy from Low-pressure Main.	Quantity from High-pressure Main.	Pressure Head of this Supply.	Velocity due to this Head.	Energy from High-pressure Main.	Energy wasted in overcoming Friction of Hose.	Energy represented by combined Jet. 100 feet Head at Back Nozzle.	Energy unaccounted for.
galls.	feet.	ft. per sec.	ft.-lbs.	galls.	feet.	ft. per sec.	ft.-lbs.	ft.-lbs.	ft.-lbs.	ft.-lbs.
117·6	46	54·43	54,096	32·4	1,610	322	521,640	75,000	150,000	350,736
131·9	92	76·9	121,348	18·1	1,610	322	291,410	75,000	150,000	187,758
146·3	138	94·3	201,894	3·7	1,610	322	59,570	75,000	150,000	36,464

These numbers show that, as one would expect, there is great waste of energy when a stream of water moving with a high velocity is forced to combine with one moving at a low velocity. It appears

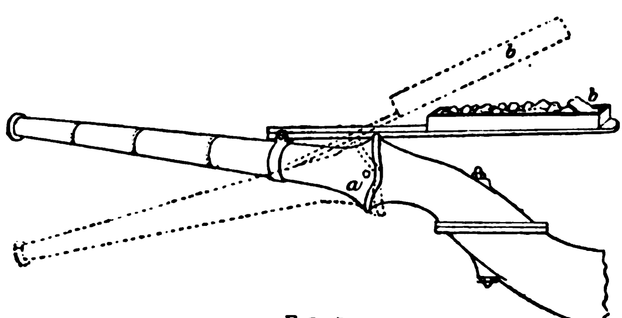


FIG. 74.

that to give even a moderate efficiency the low-pressure head should not be less than $\frac{1}{16}$ of that of the high-pressure supply.

It should be borne in mind, however, that for fire extinction the question of efficiency must be subordinated to that of promptitude in the saving of life and property.

"Hydraulicising."

Jets of water at high pressure are often used for gold mining. In that case a nozzle is employed which can readily be turned in various directions without moving the pipe conveying the pressure water. Fig. 74 shows one of these nozzles—the "little giant." It is said to be very efficient. It can be rotated completely horizontally, and moved vertically on a knuckle joint *a*, which is kept in position by the counterpoise *b*. The packings are of leather, and the nozzle is fitted inside with three rifle-plates, which prevent the jet from assuming that rotary motion which is usual with high velocities, and which impairs the effectiveness of the jet.

XII.

HYDRAULIC GENERATION OF POWER. WATER-WHEELS.

Strictly speaking, this term would include the various types of wheel propelled by water, from the old water-raising apparatus to the modern turbine. It is here used in a limited sense, including wheels rotating about horizontal axes and of the following kinds.

Overshot Wheels.

In these the water acts mainly by its weight, though a small portion of its kinetic energy also is utilised.

The water passes over the summit of the wheel, as shown in Fig. 75, and falls against and into the buckets. This type of wheel is used for falls varying from 10 to 70 feet, with head-water level varying not more than 2 feet. Its efficiency would be greater were it not for the loss of water owing to the horizontal velocity of the latter, and to the fact that the tail-water does not readily leave the wheel-pit, being projected from the wheel in the opposite direction to that of tail-race flow.

The efficiency is given by Fairbairn * as about 60 per cent., but is generally more. Unwin gives 75 per cent.

* Fairbairn's 'Mills and Millwork,' Part I. p. 123.

Taking 70 per cent. efficiency, the useful horse-power is $0 \cdot 70 \times \frac{62 \cdot 4 \times Q H}{550} = 0 \cdot 08 \, Q H$ nearly, where H is available head in feet.

The water should have a greater velocity than the circumference of the wheel, the latter being about 6 feet per second; the former should be about 10 feet per second. This velocity is acquired by falling through a height $10^2 = 2 g h$ or $h = \frac{100}{64 \cdot 4} = 1 \cdot 55$ feet, or the water should enter the wheel at a point that distance below the level of the surface of head water.

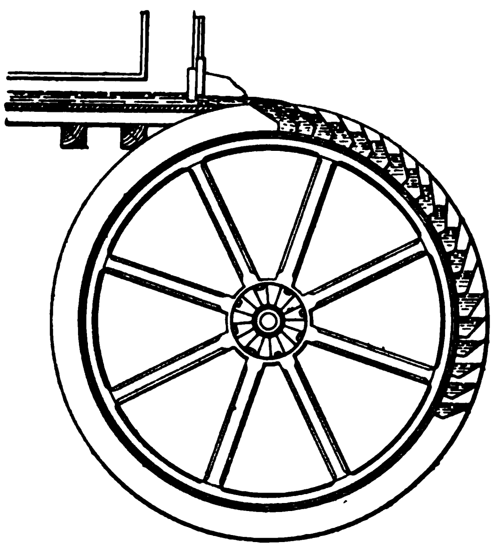


Fig. 75.

The construction of the wheel is shown in the figure. The depth of the shrouding s (Fig. 76) is from 10 to 18 inches. The diameter of the wheel is from $H - 1 \cdot 3$ to $H - 2 \cdot 5$. The number of buckets n being $= \frac{\text{circumference}}{s}$.

If b is the inside breadth of the wheel, neglecting thickness of buckets, the capacity of that portion of the wheel which passes the sluice in one second is $v \, b \, s$, v being the velocity of the wheel. If the water supply is more than one-third of this, there is great loss by spilling of the water.

Breast Wheels.

This consideration, and the fact that these wheels do not readily clear themselves of tail-water, nor work well if immersed more than 1 foot in it, led Fairbairn to devise, or improve, the breast wheel.

This type of wheel, shown in Fig. 76, has been much used. The water here acts by its weight alone, dropping into the buckets nearly vertically through the apertures in the end of the pen-trough P, which is shaped to fit the circumference of the wheel. The breast B, of masonry, serves to some extent to prevent spilling of the water; but in high-breast wheels over 20 feet in diameter no breast is required. The earliest form of the high breast wheel was called a pitch-back wheel, which was a modification of the overshot wheel introduced by

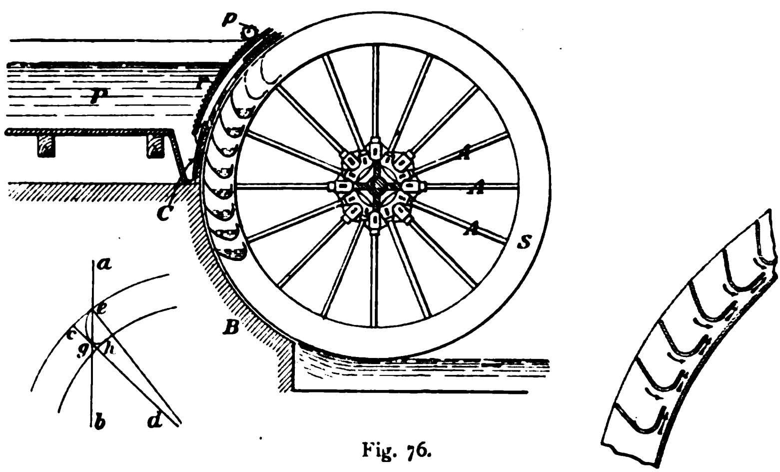


Fig. 76.

the millwright in cases where the support of the trough over the summit was difficult, and where the tail-water had not a free flow. The water was dropped into the wheel at a point varying from 25° to 30° from the summit. High-breast wheels take the water within wider limits, but in all cases above half diameter.

Fairbairn's improvements consisted mainly in (1) putting in sheet-iron buckets instead of wooden ones; (2) only partially buckets and ventilating them, as shown at right-hand side (3) making a close breast to prevent undue waste of generally using iron instead of wood in the construc- parts where possible, and providing [illegible] ns of ver.

The older wheels usually drove from the axle, a spur-wheel on the latter gearing with a pinion, which in turn drove other gearing, and hence the machinery.

Fairbairn usually employed a segmental spur-wheel—fastened to the arms (A A, etc., Fig. 76) and shrouding near the inner circumference of the latter—which drove a pinion. This method had two advantages: it relieved the arms of the wheel of bending stresses, and it gave a greater speed to the pinion, often allowing intermediate gearing to be dispensed with.

Construction for Curve of Buckets.

The following is Fairbairn's construction. Refer to sketch on lower left-hand corner of Fig. 76.

Let *a b* be a line cutting the outside circumference of the wheel where the water is to enter, and in the same direction. Measure *e c* = the distance apart of the buckets (5 to 8 inches for high-breast, and 9 to 12 inches for low-breast wheels). From point *c* draw a radius of the wheel *c d*. Then *g h* is the flat part of the bucket, and *e g* the sloping part if the buckets are of wood. If of iron, draw the curve at discretion, as shown, making due allowance for the speed of the wheel.

The construction of the wheel is readily seen from the illustration. The axle is of cruciform section with sockets keyed on it, into which the arms are fixed by cotters or bolts. The shrouding has little guides of angle-iron fastened to its inner side, to which the sheet-iron buckets are bolted, the soling being also of sheet iron. The water is admitted through the apertures shown in the pen-trough P, and drops into the buckets, the supply being cut off by the curved plate C, which is drawn over the inlet orifices by the rack *r*, actuated by the pinion *p*, which may be moved by hand or governor, the lower orifices being closed first to preserve the efficiency with partial supply by increasing the average head. Fig. 117, p. 179, shows the type of governor used by Fairbairn.

Large apertures for the passage of the water are necessary, and in practice the ordinary vertical sluice is often employed, instead of that shown.

The efficiency, under favourable circumstances, is from 70 to 75 per cent.

Low-breast wheels were used by Fairbairn for as low falls as 5 to 8 feet, the diameter of the wheel being about 16 feet, and an efficiency of about 50 per cent. was obtained.

UNDERSHOT WHEELS.

These belong to the oldest type of water-wheel, as a reference to Ewbank's description of ancient water-raising appliances will show. The old types were very inefficient. Smeaton improved them, but Poncelet brought them to a high stage of efficiency. The theory of his construction has been referred to.

The wheel is shown in Fig. 77. It is used for falls up to 6 feet. It acts on the same principle as the impulse turbine, the momentum of the water being utilised. The water enters the vanes with a velocity nearly $= \sqrt{2gh}$. It glides up the float, comes to rest,

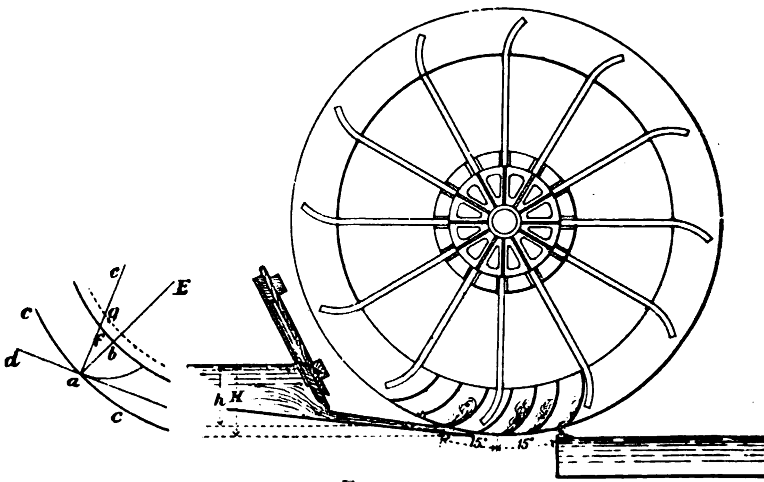


FIG. 77.

and then leaves the wheel with very little horizontal velocity relative to the earth, hence with little horizontal kinetic energy.

The best circumferential velocity of the wheel, v_2, is from $0 \cdot 5$ to $0 \cdot 6 \sqrt{2gH}$, speed of wheel v_2 is $\pi d n$, where d is the diameter and n the number of revolutions per second. Thickness of water stream entering wheel should not exceed about 10 inches.

$$b = \frac{6Q}{H\sqrt{2gH}},$$

b being the width of the stream, and the wheel is made about 4 inches wider than this. The efficiency is often 60 per cent., but may be as high as 68 per cent.

Construction for Curve of Vanes.

The following is given by Fairbairn :—Draw *c c* the external circumference (lower left-hand corner of Fig. 77) *a* E the radius of the wheel. Take $a\,b = \frac{1}{3}$ to $\frac{1}{4}$ of the fall.

Draw the inner circumference of shrouding through *b*. Suppose water to strike bucket at *a* and in direction *a d*; draw *a e* perpendicular to *a d*, so that the angle *e a* E is from 24° to 28°. Take on *a e*, $f g = \frac{1}{6}\,a f$, and from centre *g*, with radius *g a*, describe the curve of the float.

The number of buckets N is given by the rule

$$N = \frac{8}{5} d + 16,$$

for wheels of from 10 to 20 feet in diameter.

In these illustrations only a few of the buckets or floats are drawn, but it will be understood that the circumference of the wheel has symmetrically spaced buckets all round it, as in the portion in which such buckets are shown.

XIII.

CENTRIFUGAL PUMPS.

Introductory.

The centrifugal pump is not an apparatus for generating mechanical power, but on the contrary is usually employed to give to water potential energy, utilising for this purpose the mechanical energy of a steam-engine or other prime mover. In principle and construction it resembles the turbine so closely, however, that the two can be most conveniently studied together, and as the water is usually not guided at entrance it is somewhat simpler to study than the turbine, hence we here devote a chapter to this—in some respects the most important—machine for raising or circulating water.

It is not necessary to dwell on the history of the development of the centrifugal pump. Euler, the great mathematician, brought out a crude form of centrifugal pump, of little practical value, an account of which was published in 1754.

In 1830, a centrifugal pump, the patent of a Mr. M'Carty, was used in the United States Navy Yard at New York. Several pumps were tried by French engineers, but the appliance only came into commercial use after the great Exhibition in London in 1851, when the Appold pump was brought prominently into notice, with an efficiency about three times that of any other exhibited.

Mr. Appold made many experiments, some of which seem to show the greater efficiency of curved vanes over radial ones. There are, however, many things to be considered, radial vane pumps being now made (mainly by Continental makers) with good efficiency.

There is no doubt, however, that the late Mr. Appold, in connection mainly with improvements in the revolving part or fan, and Professor James Thomson, in regard to the whirlpool chamber, did more than any others to make the centrifugal pump an apparatus of great practical and commercial value.

Water cannot pass along a path of suddenly changing curvature without loss of energy, which loss is greater, the greater the velocity of the water. This fact must be borne in mind in designing machines like centrifugal pumps or turbines, to act on, or be acted upon, by water. It is absolutely impossible for a frictionless liquid to flow in a path discontinuous as to curvature. If water be compelled to flow, say, along a pipe which suddenly changes in diameter, it produces for itself little whirls or eddies, which act as wheels to help it round the corners, just as one puts rollers under a log of wood to get it moved along more easily. Wherever such eddies are set up energy is wasted, not only by the whirls in the corners, but by smaller eddies set up and carried along by the water. If we compel water to flow in a path like BAC (Fig. 78), it creates eddies to carry it by a path of continuous change of curvature from B to C.

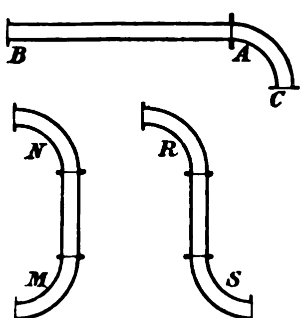


FIG. 78.

The probable truth of this conception of eddies can be shown experimentally, for if the water flows along N, and you make it pass round a similar curve M, you do not get as much waste of energy in the second operation as the first; whereas, if it first pass along R and then round S, a similar curve, but bent the opposite way, you get fully as much waste at the second bend as the first. This seems to indicate that the little eddies created at N are available at M, but those produced at R have to be destroyed and new ones created, rotating in the opposite sense, in order to carry the fluid round S. It is evident then, that great care must be exercised, in designing a centrifugal pump or turbine, to provide for the water a path free from abrupt changes in direction. The vanes and other guiding surfaces have to be placed at the proper angles, so that the water may pass into or out of the wheel without sudden change in direction or velocity, and all curves should have a gradual change of curvature, such as may be obtained by using an elastic strip as a template.

A pump is designed to add to the store of energy possessed by every pound of water passing through it. The calculation of the addition, positive or negative, which the vanes of any pump or wheel give to each pound of water is not difficult, though writers on this subject have confused the issues, and frightened students, by endeavouring to use mathematics to find out things which cannot be calculated properly at all. The leading principle on which we depend in designing these machines has been very lucidly explained by Professor Perry in an illustration like the following:—Suppose a man jumps into an American railway train, and after wandering about through it anywhere, jumps off again; how do we calculate the energy, positive or negative, the train has given to him? The answer is: find his momentum in the direction of the train's motion just before he alights on the train, and also find his momentum, in the same direction, just before he leaves it. The difference of these is the momentum he gives to the train, and "momentum per second" is force. Suppose a number of men could perform the feat every second, following each other with the greatest regularity, then the momentum given to the train in one second could be readily calculated, or the force which the men exert on the train could be found. This force, multiplied by the distance passed through by the train in one second ($= v$ where v is the velocity of the train) would represent the energy given per second to the train, or by the train to the men, as the case may be.

Now, if we wish the men to enter the train without receiving shocks from the partitions, it is evident that we should shape those partitions in a peculiar way.

It may be well to first of all consider this illustration as bearing on the action of water in the centrifugal pump, in which case the water is *not* guided before it enters.

An example will best illustrate this. Suppose water flowing radially with a velocity of 4 feet per second into a wheel rotating, at the point where the water enters, with a linear velocity of 8 feet per second, how ought the vane to be shaped so as to allow the water to enter with as little shock as possible?

Let A B, Fig. 79, be the curve of the wheel. Draw C D normal to A B, and make it 4 units long to represent the radial velocity 4. Draw C E tangential, and 8 units long, then the direction of the resultant C F is the proper direction of the vane just at the tip. It is the direction in which the partitions of the carriages in the American railway train ought to be sloped so as to give as little shock as possible to the men entering it. But we also see that the man ought not to try to enter at right angles to the direction of the train's motion. Hence in turbines the water is guided in the proper direction before it enters the revolving wheel. This is one reason—though a minor one—why turbines are more efficient than centrifugal pumps.

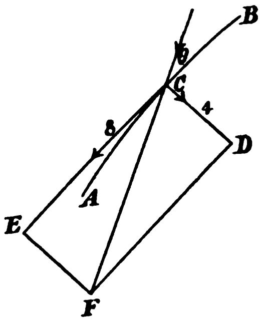


FIG. 79.

Take a simple case (Fig. 80): the water had *no* momentum in the direction of the motion of the wheel before it entered it, at A; having entered, it now moves along the vane A B, gradually attaining the velocity of the wheel, and then it finally leaves at B. If the vane is radial at B it has the same velocity as the wheel just before it leaves. Let this velocity be v feet per second. Then every pound of water leaving B, leaves with a tangential momentum $\frac{v}{32 \cdot 2}$, and retards the wheel with a force of this amount acting at B. This force $\times\, v$ is the energy the one pound of water receives per second from the wheel $= \frac{v^2}{32 \cdot 2}$.

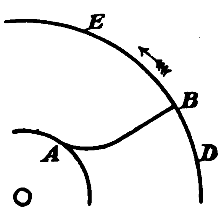


FIG. 80.

One pound of water in the discharge chamber of the pump has gained this much energy from the time it left the supply-pipe, except that it lost some of its energy by friction. If the vanes were bent backwards towards D, the water would receive less energy than this, and if they were bent forwards towards E, it would receive more.

The water gets the energy to squander or store as it pleases. It does squander a good deal of it in friction. But if it converted it all into potential energy it would raise it to a height $\frac{v^2}{32 \cdot 2}$ feet; in other words, it would be lifted to a height, above the pond from which it was taken, of $\frac{v^2}{32 \cdot 2}$, or to a height *twice that due to the velocity of the circumference of the wheel* (since the velocity v is due to a height given by the rule $v^2 = 2gh$, or $h = \frac{v^2}{64 \cdot 4}$). Suppose the rim of the wheel had a velocity of 46·8 feet per second, a stone would fall freely through 34 feet to acquire this velocity, hence the total lift of the pump (if perfect as our rule assumes) would be 68 feet.

The real height to which the water is lifted, divided by the ideal height $\frac{v^2}{32 \cdot 2}$, is the efficiency of the wheel, or rather of the water passages all through the pump.

The wheel gets energy from an engine, and the energy given out by the engine *per pound of water lifted* is a measure of the efficiency of the shafting, belting, and wheel.

It may be well now, having considered some of the elementary laws governing the action of the centrifugal pump, to go a little more fully into the considerations influencing the sizes and shapes of the pump passages, and vanes.

With a steady flow of water

$$\frac{v^2}{2g} + \frac{P}{w} + h = \text{constant}$$

= the total store of energy of 1 lb.

In pipes, and in fact, wherever water flows, its total store of energy is gradually diminished by friction. The object of a pump is to *increase* this total store.

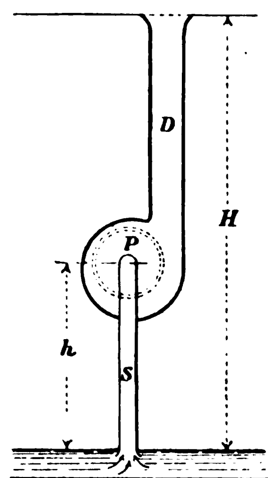


FIG. 81.

In Fig. 81 is roughly shown the general arrangement of a centrifugal pump. H is the total height to which the water is to be lifted, i.e. the total potential energy which every pound of water is to receive. Theoretically, h may be anything under 32 feet; in practice it is best not to have it more than from 6 to 10 feet. P is the pump, S the suction pipe, and D the delivery pipe; the water enters the pump at the centre, being drawn in by the partial vacuum produced, is whirled round in the revolving wheel or fan F, passing into the whirlpool chamber or diffusor W, and volute or discharge chamber D (Fig. 82), leaving the wheel with kinetic energy $= \frac{v_2^2}{g}$, where v_2 is the circumferential velocity of the outside of fan (radius r_2). This large store of kinetic energy is gradually changed into pressure energy in the whirlpool chamber, and by the time the water reaches the delivery pipe it has a sufficient pressure to force it up the pipe, in which ascent almost all its energy is gradually changed into potential energy.

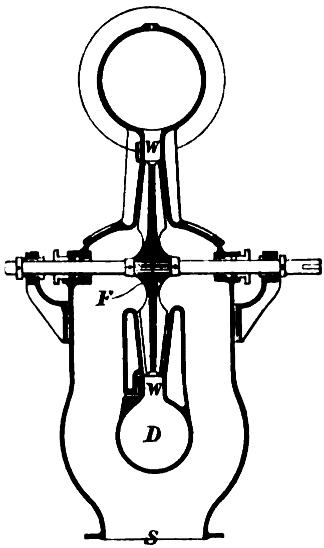


FIG. 82.

It is necessary that the water should receive as little shock as possible in entering the revolving wheel, hence the vanes are shaped as shown in Fig. 83 (where only a few of the vanes are shown), so that the direction of flow is as nearly as possible the same the instant after entering the wheel as it was the instant before.

If this is to be accomplished, evidently if v_1 is the velocity of the inner circumference, and v_r the radial velocity of the water, Fig. 84 shows that by measuring off A B $= v_r$ and A D (at right angles to A B) $= v_1$, and completing the parallelogram of velocities, θ is the angle required.

And we see that $$\tan \theta = \frac{v_r}{v_1}.$$

This is the angle of the vane just where it joins the inner circumference.

In many turbines and some centrifugal pumps (which as far as theory goes are merely reversed turbines) the radial velocity of the water is constant through the wheel, this necessitating that the area of the openings through which the water flows shall be the same everywhere. This could be accomplished by making $b_2 r_2 = b_1 r_1$, b_1 and b_2 being the breadths at radii r_1 and r_2.

These dimensions are, however, modified as experience and experiment indicate. In many good pumps like the Appold pump (see Fig. 87, p. 133) the outside area is much greater than the inner, and thus the water leaves with less radial velocity and greater pressure, so that a much smaller whirlpool chamber suffices.

In our radial vane pump—to keep to the easy illustration for the moment—if Q be the quantity of water passing through the pump per second, $Q = 2 \pi r_2 b_2$ minus a certain allowance for the thickness of the vanes. If v_r be the radial velocity of the water at the outer rim, and A the clear area of the openings by which it leaves the rim, $\frac{Q}{A} = v_r$.

If w be the weight of 1 cubic foot of the liquid, since change of momentum per second is force, and momentum

$$= \text{mass} \times \text{velocity} = \frac{w\,Q}{g} \times v_2,$$

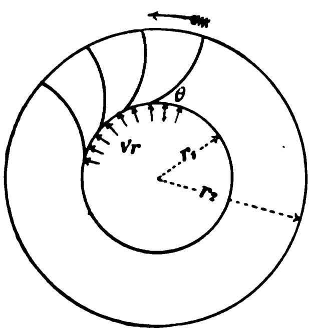


FIG. 83.

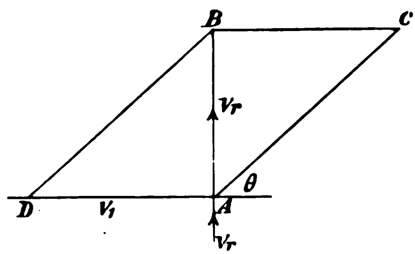


FIG. 84.

∴ the force exerted on the water = gain in momentum per second

$$= \frac{w\,Q}{g} v_2,$$

and force × velocity per second = work done per second.

$$\therefore \quad \frac{w\,Q}{g} v_2^2 = \text{the energy given to the water in ft.-lbs. per second.}$$

The total weight of water passing per second is w Q, and the energy imparted per second to 1 lb. is $\frac{v_2^2}{g}$, and it is lifted H feet altogether.

∴ neglecting friction, total store =

or

$$v_2^2 = g\text{H},$$

or

$$v_2 = \sqrt{gH} = \sqrt{2g\left(\frac{H}{2}\right)}$$

—a law like that for the velocity of a body falling freely. Hence we see *the velocity of the rim is equal to that of a stone which has fallen freely through a height = half the total lift of the pump*, or is the velocity due to half the head H.

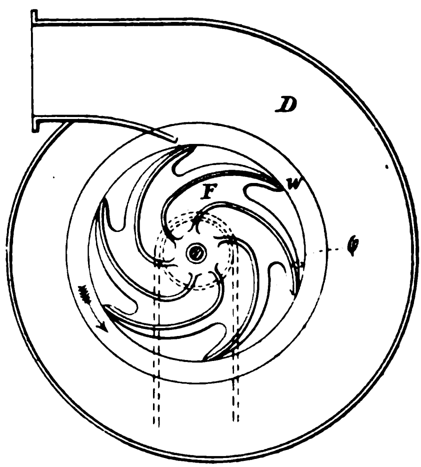


Fig. 85.

This law is of considerable importance in the case of turbines. It is only true neglecting friction. As a matter of fact, the pump has to be driven at a greater speed than this. It is also only true if the pump is delivering little water, but just keeping the water at a constant level in the discharge pipe. If the pump were gradually to slow down the water would fall and drive the pump as a turbine—supposing a sufficient supply available.

The foregoing, viz. that the velocity v_2 of the water is that of the circumference of the fan, is only true for pumps in which, as our figures indicate, the vanes are radial to the outside circumference of the fan. Very often, in fact nearly always in practice, the vanes are curved as in Fig. 85.

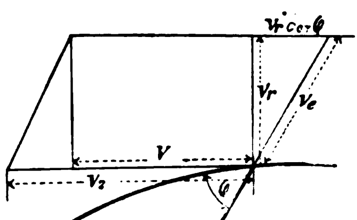


Fig. 86.

If v_r (Fig. 86) is the radial velocity and v_e the velocity along the vane, v_2 being the velocity of the rim of the wheel, then the tangential velocity imparted to the water = V, which is found as indicated in the figure.

$$\frac{Qw}{g} V = \text{the tangential force exerted on the water}$$

and

$$\frac{Qw}{g} v_2 V = \text{the energy given to it per second,}$$

there being wQ lbs. passing per second; hence the work imparted to each pound of water $= \frac{v_2 V}{g}$, which is less than $\frac{v_2^2}{g}$, since V is less than v_2.

If ϕ is the angle which the vane makes with the outer circumference (Figs. 85 and 86), $v_r \cot \phi$ is the backward tangential velocity, and the forward tangential velocity is evidently

$$v_2 - v_r \cot \phi.$$

$\therefore \quad (v_2 - v_r \cot \phi)\frac{Qw}{g}$ = momentum given per second = tangential force at rim of wheel, and

$v_2 (v_2 - v_r \cot \phi) \frac{Qw}{g}$ = energy given per second to wQ lbs. of water, or

$v_2 (v_2 - v_r \cot \phi) \frac{1}{g}$ = H = the energy given to 1 lb. of water neglecting friction.

Really

$$\frac{g \times H}{v_2 (v_2 - v_r \cot \phi)} = \eta \text{ (the hydraulic efficiency).}$$

In the case of radial vanes the total energy given to the water in the wheel is made up of half kinetic and half pressure energy. For total energy given to 1 lb. $= \frac{v_2^2}{g}$, kinetic energy $= \frac{v_2^2}{2g}$ $\therefore$ pressure energy $= \frac{v_2^2}{2g}$ also.

In the case just considered, with the backward sloping vane, the kinetic energy given to 1 lb.

$$= \frac{(v_2 - v_r \cot \phi)^2}{2g},$$

and the total energy

$$= v_2 (v_2 - v_r \cot \phi) \frac{1}{g}.$$

In the pump shown in Figs. 82 and 85, the whirlpool chamber is seen in section at W, F being the impeller or fan driven by an engine or outside motor, and S the suction pipe. The use of a large whirlpool chamber is to allow the kinetic energy to gradually die out as the water recedes from the vanes. The use of the whirlpool chamber was first pointed out clearly by the late Professor James Thomson, whose name will always have a chief place as a pioneer in this branch of engineering.

In pumps with backward sloping vanes, the water leaves the fan with comparatively little kinetic energy, and the whirlpool chamber may be small. This will be seen in the Appold pump, Fig. 87, where it will also be observed that lateral easement is given to the water as well, so that its velocity, and hence its kinetic energy, may be small on leaving the fan.

In order to get the greatest efficiency out of a pump of this kind, it is necessary—all other things being equal—to have the value of ϕ that which will make the *total energy* a maximum and the *kinetic energy* a minimum.

This will best be seen from an example.

Let the total lift be 15 feet.

$$\text{Circumferential velocity } v_2 = \sqrt{2g \times 7\tfrac{1}{2}} = 22.$$

Let the radial velocity be $\frac{1}{8}$ of that due to the total lift.

$$\therefore \quad v_r = \tfrac{1}{8}\sqrt{2g \times 15} = 4, \text{ say.}$$

$$\text{Total energy} = 22\,(22 - 4\cot\phi)\frac{1}{g}.$$

$$\text{Kinetic energy} = \frac{(22 - 4\cot\phi)^2}{2g}.$$

Tabulate as below.

Angle ϕ	$g \times$ total energy.	$g \times$ kinetic energy.	$g \times$ pressure energy.
90	484	242	242
60	433	194	239
45	396	162	234
30	330	112	218
20	222	61	161
15	156	25	131

This calculation is based on the assumption that the circumferential velocity is in every case the same for the same lift, being obtained from $v_2 = \sqrt{gH}$.

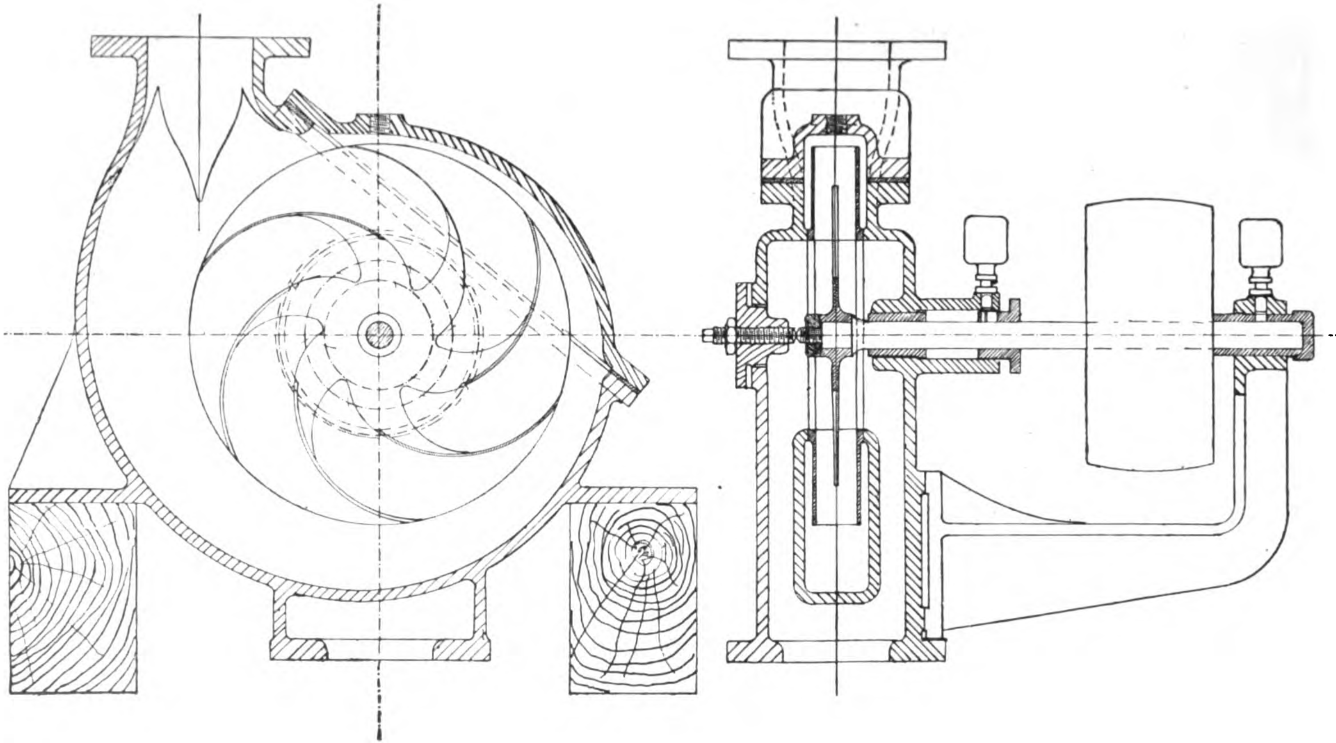


FIG. 87.

If the angle ϕ be too small the vanes are too long and friction is increased, also if we go much below 30° there is a rapid falling off of the total energy. The right angle in the above example seems to be between 30° and 20°.

THE WHIRLPOOL CHAMBER.

In the pump, in order that there shall be a minimum waste of energy in the whirlpool chamber, the water must follow the *law of natural or free vortex flow.*

Let the pump be horizontal (h remains constant in our equation of constant energy per lb.). Let there be a large whirlpool chamber.

$\frac{dp}{dr}$ is the rate of change of pressure as we go further out.

Neglecting h,

$$\frac{v^2}{2g} + \frac{P}{w} = \text{constant.}$$

Differentiating,

$$\frac{v}{g} \cdot \frac{dv}{dr} + \frac{1}{w} \cdot \frac{dP}{dr} = 0.$$

But

$$\frac{dP}{dr} = \frac{w}{g}\,\frac{v^2}{r}$$

(neglecting gravity, and here we are going across circumferential stream lines).

Substituting value of $\frac{dP}{dr}$, we get

$$\frac{v}{g} \cdot \frac{dv}{dr} + \frac{1}{g} \cdot \frac{v^2}{r} = 0.$$

Dividing across by $\frac{v^2}{g}$, and arranging,

$$\frac{dv}{v} + \frac{dr}{r} = 0.$$

Integrating,

$$\log v + \log r = C \text{ (a constant)};$$

$$\therefore \quad vr = \text{antilog } C,$$

or

$$v \propto \frac{1}{r}.$$

This is the law of natural flow. A particle of water travels round in a spiral path, its velocity decreasing as its distance from the centre increases.

This is the sort of flow water naturally assumes, and it is seen, but in a reverse order, on pulling the central plug out of a wash-hand basin. In no other sort of flow is so little energy wasted, but the flow through the hole is small. Hence, a pump with a very large whirlpool chamber may give a good efficiency and a high lift, but may not work well if a large flow is required, with comparatively small lift.

As a matter of fact, the whirlpool chamber, though a splendid arrangement from the theoretic point of view and correct in principle, would require to be very large to realise our idea of natural flow. This large size would cause inconvenience and expense, whilst the greater surfaces exposed to the moving water would give rise to considerable waste of energy by friction. Hence it is probable that the common-sense solution of the problem, due to numberless experiments by makers, is the best.

The wheel having larger orifices at its outer than its inner circumference, and the backward sloping of the vanes, allows the kinetic energy to be small, and to be, to a great extent, converted into pressure energy without the use of a large whirlpool chamber.

Hence the path of the water particle, instead of being that of a spiral starting with the point at which it leaves the vanes, is much more direct, and space is saved, a larger flow with a smaller pump is possible, whilst nearly if not quite as high efficiency is obtained as it is possible to have, even with a Thomson chamber of practical dimensions.

The backward sloping of the vanes does *not* add to the efficiency of the pump unless the whirlpool and discharge chambers are of fixed size and too small to realise our ideal of natural flow. Given a chamber of proper size and shape, the radial vane pump is probably as efficient as any other. Recent radial vane pumps (by Messrs. Farcot) have given a high efficiency. They do not require to run at such a high speed as sloping vane pumps. Radial vane fans also are very efficient.

The way in which the foregoing principles have been carried out in typical pumps of English make, will be understood from a study of Figs. 87, 88 and 89, where are shown, respectively, the "Appold" pump of Messrs. Easton, Anderson and Goolden, the "Conqueror" pump of Messrs. W. H. Allen and Co., and a pump somewhat resembling the "Invincible" pumps of Messrs. Gwynne. The last is a working drawing with some dimensions, which may be of use to the young designer. It may be necessary to remind such that usually a foot-valve is provided at the bottom of the suction pipe, also an orifice in the pump cover to allow the pump to be filled with water or steam at starting, so that the pump may commence to "draw."

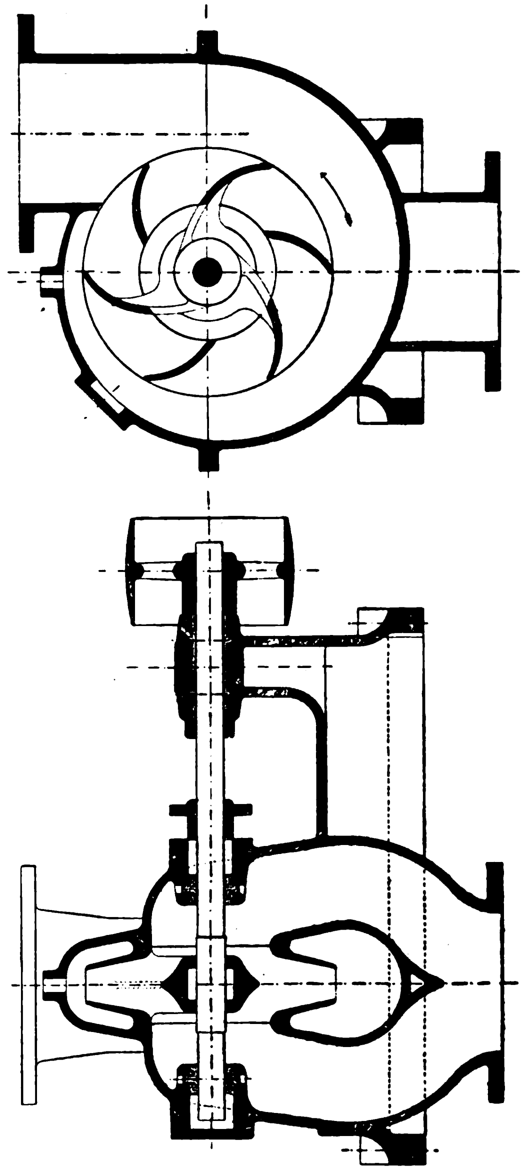


Fig. 88.

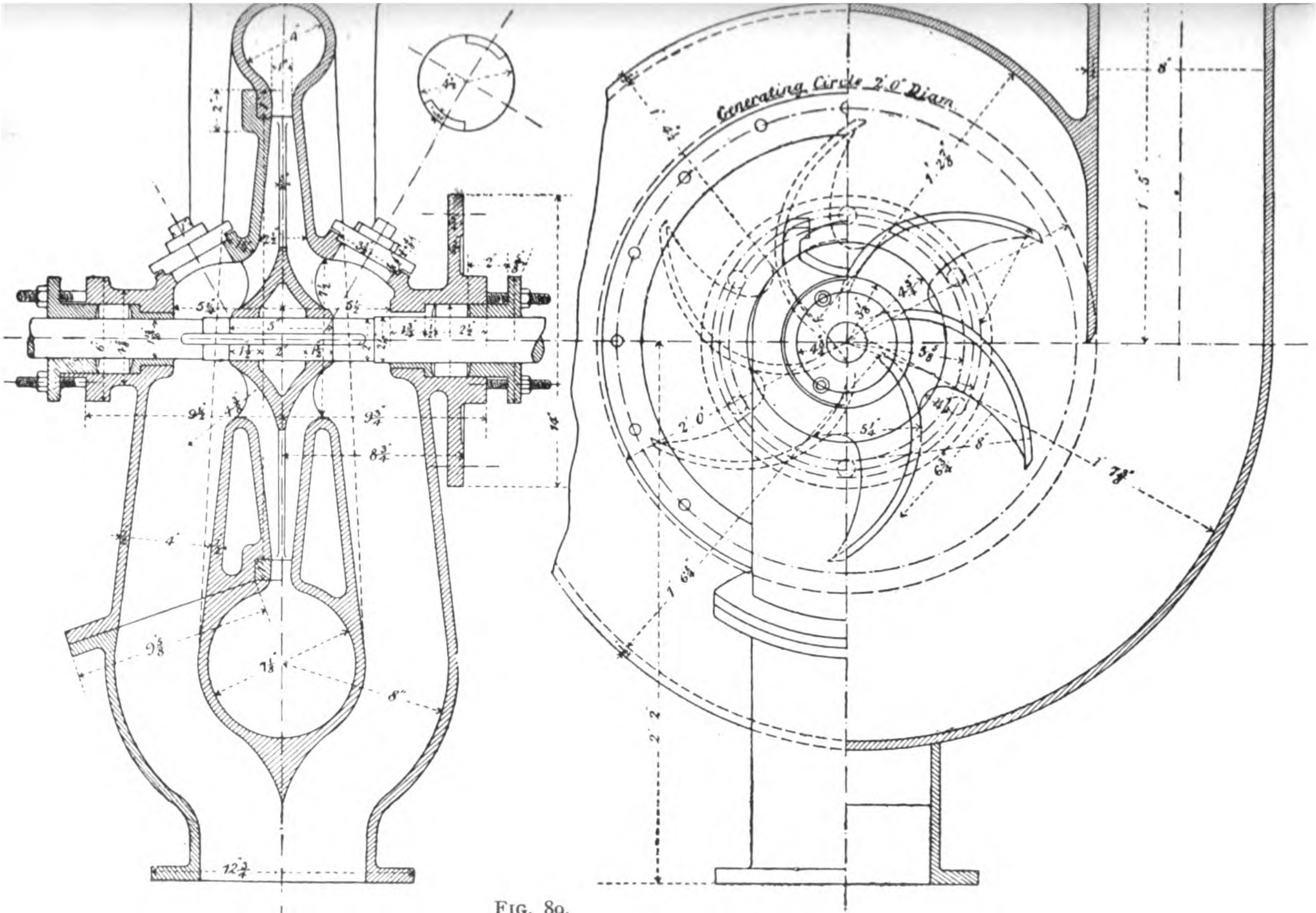


FIG. 89.

Change of Pressure inside the Revolving Wheel.

If we imagine the wheel to be horizontal and neglect gravity, which we may do if speed and radius are sufficiently great, we get the law already worked out for change of pressure with cylindric level surfaces (see p. 22).

The law is, that

$$P_2 - P_1 = \frac{w a^2}{2g}(r_2^2 - r_1^2),$$

P_2 being the pressure at radius r_2, P_1 that at radius r_1, a the angular velocity, and w the weight of unit volume of the water. If we assume that the pump has radial vanes, the whole change of pressure is

$$P - P_0 = \frac{w a^2}{2g}(r_2^2 - r_1^2) = w\frac{(v_2^2 - v_1^2)}{2g},$$

where r_2 and r_1 are the outside and inside radii, v_2 and v_1 the corresponding linear velocities.

But the total gain of energy per pound (neglecting resistances) is the gain of pressure energy + the gain of kinetic energy, and the gain of pressure energy is—to stick to our easy rule—2·3 times the gain of pressure in *lbs. per square inch*, being

$$\frac{2\cdot3}{144}(P - P_0) = \frac{2\cdot3 \times 62\cdot4}{144} \times \frac{(v_2^2 - v_1^2)}{2g} = \frac{v_2^2 - v_1^2}{2g},$$

which is also the gain of kinetic energy per lb. Pressures being in lbs. per square foot, the kinetic energy will be expressed in ft.-lbs.

Hence the gain of pressure energy and the gain of kinetic energy are equal. The total gain of energy is therefore twice the gain of pressure energy, or twice the gain of kinetic energy.

This law is also nearly true in a *sloping-vane* pump, if the pump is delivering very little water.

Change of Pressure in the Whirlpool Chamber.

Although probably never attained in practice, it may be of interest to study the law of change of pressure in a perfect whirlpool chamber, where the water follows the *law of natural flow*.

Since

$$v \propto \frac{1}{r}, \qquad v = \frac{K}{r}, \qquad \text{or} \qquad v^2 = \frac{K^2}{r^2},$$

where K is a constant.

Neglecting differences of level,

$$\frac{v^2}{2g} + \frac{P}{w} = \text{a constant.}$$

Substituting for v^2, we have

$$\frac{K^2}{2gr^2} + \frac{P}{w} = \text{a constant.}$$

or

$$P = \text{a constant} - \frac{K^2 w}{2gr^2}.$$

P_2 is pressure where radius is r_2;

$$\therefore\quad P_2 = \text{a constant} - \frac{K^2 w}{2gr_2^2},$$

and

$$P = \text{the same constant} - \frac{K^2 w}{2gr^2}.$$

Therefore, subtracting,

$$P - P_2 = \frac{K^2 w}{2gr_2^2} - \frac{K^2 w}{2gr^2},$$

or

$$P = P_2 + c\left\{\frac{1}{r_2^2} - \frac{1}{r_2}\right\},$$

where $c = \dfrac{K^2 w}{2g}$.

To find value of c or K.

$$c = \frac{K^2 w}{2g} \quad\text{and}\quad K^2 = v^2 r^2;$$

$$\therefore\quad c = \frac{v^2 r^2 w}{2g};$$

$$\therefore\quad P = P_2 + \frac{v^2 r^2 w}{2g}\left\{\frac{1}{r_2^2} - \frac{1}{r_1^2}\right\} = P_2 + \frac{v^2 r^2 w}{2g}\left\{\frac{r^2 - r_2^2}{r_2^2 r^2}\right\}$$

$$= P_2 + \frac{v^2 w}{2g}\left\{\frac{r^2 - r_2^2}{r_2^2}\right\}.$$

But

$$v r = K = v_2 r_2;$$

$$\therefore\quad v = \frac{v_2 r_2}{r},$$

and

$$v^2 = \frac{v_2^2 r_2^2}{r^2}.$$

$$(1) \quad \therefore \quad P = P_2 + \frac{v_2^2 w}{2 g r^2} (r^2 - r_2^2),$$

or

$$(2) \quad P = P_2 + \frac{v_2^2 w}{2 g}\left(1 - \frac{r_2^2}{r^2}\right).$$

Since

$$\frac{w}{2g} = \frac{62 \cdot 4}{64 \cdot 4} = 0 \cdot 969$$

the law is practically

$$(3) \quad P = P_2 + 0 \cdot 97\, v_2^2 \left(1 - \frac{r_2^2}{r^2}\right).$$

Example.—Take $P_0 = 2116$ lbs. per square foot atmospheric pressure (it should be less than this by an amount equivalent to suction height), $r_0 = \frac{1}{2}$ foot, r variable up to 1 foot, speed 300 revolutions per minute, plot pressure curve for inside of wheel.

Continuing the example for whirlpool chamber, $P_2 = 2833$, $r_2 = 1$ foot ($v_2^2 = 985 \cdot 96$), we plot the values of p and r.

The result is shown in the table.

Values of r (feet).	Values of P (lbs. per sq. ft.).	Values of r (feet).	Values of P (lbs. per sq. ft.).
0·5	2116	1·1	2999
0·6	2220	1·2	3125
0·7	2345	1·3	3223
0·8	2490	1·4	3302
0·9	2650	1·5	3364
1·0	2833		

The left-hand set of values are obtained from the law for the inside of the revolving wheel, the right-hand set from the law above.

These results are shown in curves at Fig. 90, there being *two* curves, each with its own law, these curves joining at A B.

In the foregoing the elementary conception of radial vanes has been adhered to as giving rise to less cumbrous expressions, but the change required to render the work applicable to the ordinary pump with sloping vanes is easily made, as indicated at page 131.

In many modern pumps, especially of moderate size, the water enters at *one* side only. This gives an endlong pressure along the impeller axis, which may be balanced hydraulically or taken up by a suitable bearing. For the purposes of the student, the symmetrical ›rms referred to are of more importance.

Centrifugal Pumps in Series.

Centrifugal pumps are sometimes worked in series. One case is recorded by Barr in which two centrifugal pumps working thus—the first discharging into the suction pipe of the second—raised water through a height of 150 feet. The efficiency of the combination in this case was small. High lift pumps are, however, now made to give a good efficiency.

Efficiency of Centrifugal Pumps.

The centrifugal pump does not compare well with the turbine as regards efficiency, partly on account of the non-guiding of the water at entry in the case of the former, but mainly owing to the fact that in the latter the conversion of the kinetic energy of the water into the

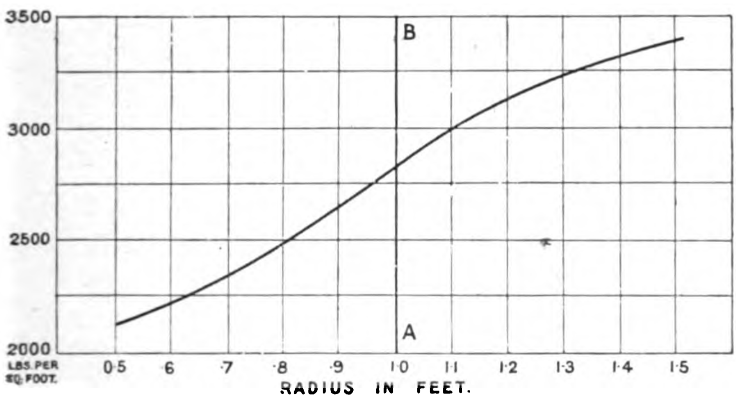


Fig. 90.

pressure, and finally the potential, form is not easily accomplished without considerable waste. Some of the methods of diminishing this loss which have been employed, have already been referred to. More fully stated they are (1) the use of a large whirlpool chamber, as in the Thomson pump; (2) the use of discharge passages of gradually increasing area, as in the Mather-Reynolds pump; (3) the use of backward-sloping vanes, as adopted by Mr. Appold in 1851, and by many makers since then; and (4) the use of guide vanes in the discharge chamber, as tried by Dr. Stanton.*

* Proc. Inst. C.E., Feb. 1903.

Methods (1) and (2) have not been widely adopted, mainly from constructional reasons, great size and consequently increased friction in the first, and the difficulty of altering the discharge area to suit variable flows in the second, being adverse factors. Method (3) is of course, common, but the speed has to be increased for the same lift as compared with radial-vaned pumps, and even a moderate increase of speed involves large increase of waste by friction. Whilst pumps with curved vanes are certainly superior to those with radial vanes for moderate speeds, their superiority is doubtful for high speeds, and the best Continental makers now seem to favour the latter form. Method (4) has been tried experimentally by Dr. Stanton, with apparently good results. The gain in efficiency is probably due to diminished slip and friction, and to the gain in pressure as compared with a pump discharging into a free vortex. The advantage of the

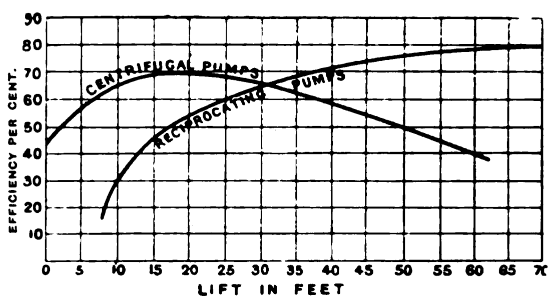


FIG. 91.

guides is most marked when the wheel has radial vanes. It was found that the number of guide passages should not be less than four; their areas being such that the velocity of flow into the passages is equal to the velocity of discharge from the wheel.

In regard to the usual methods of measuring efficiency practically, efficiency is often taken to mean $\frac{\text{water horse-power}}{\text{brake horse-power}}$ in the case of pumps driven by a belt; but in direct-driven pumps it is often taken to mean $\frac{\text{water horse-power}}{\text{indicated horse-power}}$, as the brake horse-power is not easily obtained. Some of Gwynne's pumps have given 65 per cent. by the latter method of measurement, which of course includes the mechanical efficiency of the engine.

The curves shown in Fig. 91 give a comparative view of the efficiencies of centrifugal and reciprocating pumps, on the authority of Mr. Webber.*

Professor Unwin has shown that the friction of the water in the narrow space between the revolving fan and the casing has a good deal to do with the loss of efficiency. The internal surfaces, especially those which touch rapidly moving water, should be as smooth as possible.

RÉSUMÉ.

The following rules, collected, or deduced from the foregoing, enable some points of the design to be settled, it being understood that proportions can best be obtained from the drawings of a good pump.

Let N = speed in revolutions per minute.
b_2 = clear breadth of passages at outside of disc.
t = thickness of vane.
n = number of vanes.

Then $b_2 t \operatorname{cosec} \phi$ = area of vane where it meets outer surface, and hence clear area $= 2\pi r_2 b_2 - n b_2 t \operatorname{cosec} \phi$.

H and Q are given; N and ϕ can be fixed.

$Q = \dfrac{G}{60 \times 6 \cdot 25}$ where G = number of gallons raised per minute; also $v_2 = \sqrt{2gH}$ to $1 \cdot 3 \sqrt{2gH}$ in good pumps, the lower value corresponding approximately with $\phi = 30°$, and the higher with $\phi = 15°$.

At the outside the radial velocity

$$v_r = \frac{Q}{(2\pi r_2 b_2 - n b_2 t \operatorname{cosec} \phi)\, C}$$

where C is a coefficient, usually about $0 \cdot 9$.

Also $v_2 = 2\pi r_2 \dfrac{N}{60}$, and $V = v_2 - v_r \cot \phi$.

The radial velocity at inner circumference $= v_r \times \dfrac{A_2}{A_1}$; A_2 being the outer, and A_1 the inner clear area.

The radial velocity at the eye of the disc is often taken as approximately $= 0 \cdot 25 \sqrt{2gH}$.

* 'Transactions of the American Society of Mechanical Engineers,' vols. vii. and ix.

Fix r_1; then $v_1 = 2\pi r_1 \frac{N}{60}$, and hence θ (angle of vane at inner end) can be found as indicated at p. 128.

The efficiency, neglecting loss at entrance, etc. $= \frac{gH}{v_2 V}$.

XIV.

TURBINES.

THE reader who has followed carefully the reasoning in the case of centrifugal pumps will have no difficulty in understanding all the theoretical considerations which are of much importance in the case of the turbine. A turbine is simply a centrifugal pump reversed; but the turbine is usually furnished with curved guide vanes to guide the water as it enters the wheel.

Remember, if water moves from one place to another under the action of gravity alone, 1 lb. of it has the following store of energy:

h ft.-lbs. of energy (potential), being h feet above datum level.

$\frac{v^2}{2g}$ ft.-lbs. of energy (kinetic), because of its velocity of v feet per second.

$2{\cdot}3\,p$ ft.-lbs. of energy (pressure), because of its pressure of p lbs. per square inch.

Now water-wheels, turbines, water-pressure engines, hoists, etc., take part of its store of energy *from* every pound of water and give it to machinery or to goods or people. As a simple case of the abstraction of energy, the action of the turbine may be readily understood by the illustration of the railway train given at page 125.

Another illustration is furnished by the suggestion of some one that the stations of the Underground Railway in London should be furnished with large circular platforms, kept moving so that their circumferences should go at a known speed, say, 5 miles an hour. The trains would not have to stop, but merely slow down to the speed of the periphery of the platform, when the passengers could alight and walk towards the centre of the platform, gradually losing their kinetic energy, and finally finding their way by a spiral staircase at the centre up to the street. If a steady stream of people could relied on, no driving mechanism would be necessary, as each passenger on alighting would give up the momentum, received by him from the train, to the platform, thus contributing to the driving force required. Of course this is impracticable, but it is a good illustration of what takes place in a turbine. Each pound of water gives up its momentum to the turbine, and it should drop out, after passing through the turbine, with *no* momentum in the direction of the turbine's motion, like the man going up the spiral staircase to the street.

Turbines may be roughly divided into two classes—reaction turbines and impulse turbines—in the first of which the wheel passages are always full and therefore the water under pressure; and in the second the passages are not usually filled.

In considering the action of the turbine, it may be well to study the inward-flow wheel of Professor James Thomson, as the theory of the turbine is in this case most clearly exemplified. Water flows from a pen-trough through cast-iron pipes to A (Fig. 92). Remember the pipe should be bell-mouthed and as large as convenience will allow.

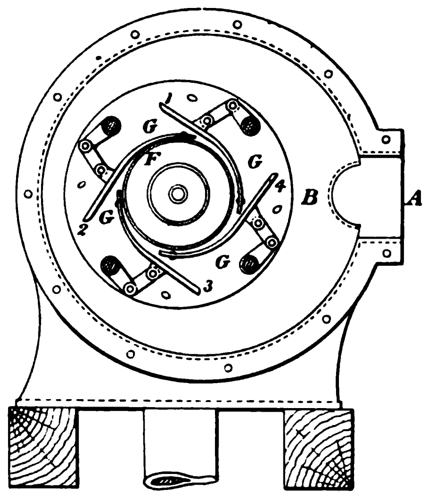


FIG. 92.

Figs. 92, and the enlarged view 93, show a plan of the chamber B into which the water flows. This chamber is so large that the velocity here is small, and the water finds its way equally readily into the central space, where it flows guided by the guide blades 1, 2, 3, 4, into the revolving wheel.

At the last, just before it enters the wheel, it has a very considerable velocity as the space is small, the guide-blade chamber being narrow. The guide-blades cause the water to flow radially as well as tangentially, the *tangential* velocity being approximately equal to that of the wheel.

If you wanted to enter a moving railway train without shock, you would be wise to get up a velocity equal to that of the train in the same direction before jumping in. Hence the *tangential* component of the water's velocity should be that of the circumference of the moving wheel.

It is easy to find the angle θ the guide vanes make with the wheel circumference; as in the case of the radial-vane centrifugal pump,

$$\tan \theta = \frac{v_r}{v_2}.$$

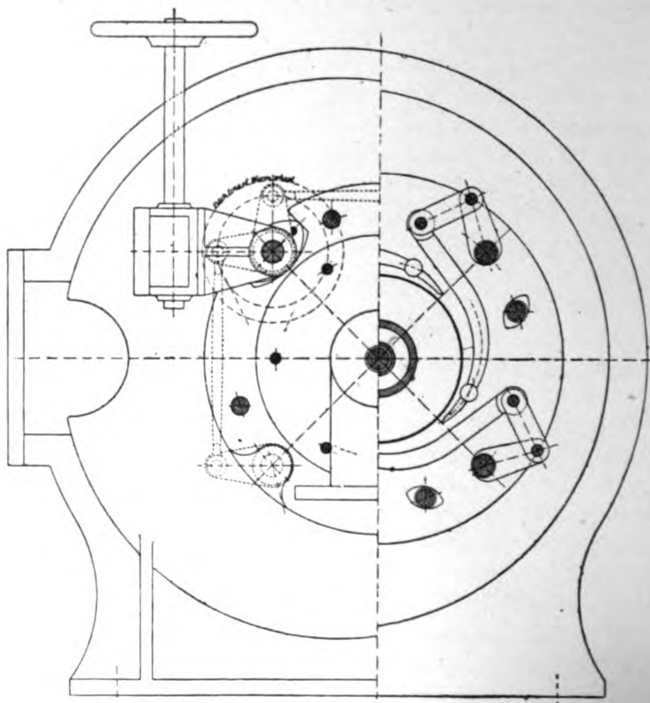


FIG. 93.

The shapes of the wheel vanes are shown in the left-hand view in Fig. 94, and a part section by a plane through the axis is shown to the right. The outside breadth b_2 is about $\frac{1}{2}\,b_1$, and r_2 about twice r_1, which is $= b_1$, thus giving a constant area through the wheel. The radial velocity through the turbine is often taken as about one-eighth of that due to the total head, or

$$v_r = \tfrac{1}{8}\sqrt{2\,g\,\mathrm{H}};$$

also

$$\mathrm{Q} = \mathrm{A}\,v_r = 2\,\pi\,r_2\,b_2 \times v_r,$$

also

or

$$= 2\pi(2r_1) \times \frac{r_1}{2} \times v_r,$$

or

$$Q = v_r \times 2\pi r_1^2.$$

In practice, allowance must be made for the thickness of the vanes. Neglecting this

$$r_1 = \sqrt{\frac{Q}{2\pi v_r}}.$$

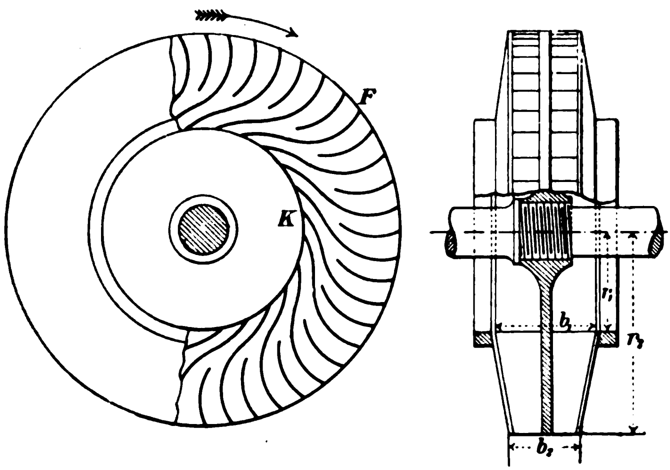


FIG. 94.

The horse-power of the turbine, neglecting all losses, is given by the rule

$$\text{H.P.} = \frac{\text{H} . \text{Q} \times 60 \times 62 \cdot 4}{33000} = 0 \cdot 1134 . \text{H} . \text{Q} = \text{H G} \times 0 \cdot 708,$$

where G is the number of gallons passing per second.

The *useful* horse-power = 0·085 H Q = 0·531 H G at 75 per cent. efficiency.

In the foregoing, for the sake of simplifying the expressions, the wheel vanes are supposed to be normal to the outside or inlet circumference. As will be seen from the figure, they are not quite normal, but slope, so as to more readily admit the water. Thus the water the instant after it enters the wheel has *not* the same but a *greater* velocity, in the direction of the circumference, than the wheel, as it is moving *forward* as well as inward along the vane.

The forward component of its velocity must be added to the velocity of the rim.

The construction may be simply given as follows:—Draw E D to represent the tangent, and B D the vane (Fig. 95) at the point where the latter meets the inlet surface of the wheel. At D draw D C at right angles to E D and of length to represent the radial velocity of the water. Draw C B parallel to D E and produce it, making A B to represent v_2, the linear velocity of the outer circumference.

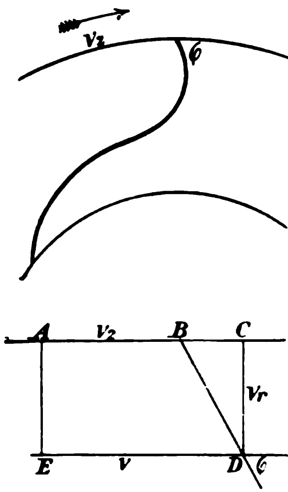


FIG. 95.

Complete the rectangle A E D C, then E D represents the *actual* velocity v of the water in the direction of the tangent. Evidently $v = v_2 + v_r \cot \phi$.

If Q be the quantity passing per second through the wheel, the momentum given per second by the water to the wheel is

$$\frac{Q w}{g} (v_2 + v_r \cot \phi),$$

which is the force acting on the wheel, and this multiplied by v_2 represents the energy given to the wheel per second by the mass $\frac{w Q}{g}$ of water.

The energy given per second by 1 lb. is

$$\frac{1}{g} (v_2 + v_r \cot \phi)\, v_2 \text{ ft.-lbs.}$$

This appears to be greater than $\frac{v_2^2}{g}$, the energy given per second by 1 lb. in the case of radial vanes. It *is* greater if v_2 is obtained as before (see rule for circumferential velocity of centrifugal pumps if vanes are radial, p. 129), by equating $\frac{v_2^2}{g}$ to H, the total fall.

But in this case the velocity v_2 would not be strictly the same as before, for to get it we must put the energy given per second by 1 lb. of water equal to the potential energy lost, i.e. H.

Or, neglecting hydraulic losses as before,

$$\frac{1}{g}(v_2 + v_r \cot \phi) v_2 = \text{H},^*$$

$$v_2^2 + v_2 v_r \cot \phi = g\text{H},$$

or

$$v_2^2 = g\text{H} - v_2 v_r \cot \phi.$$

For radial vanes, $v_2^2 = g\text{H}$.

Evidently v_2 is less than in the case of radial vanes, but as $\cot \phi$ is usually small, $v_2 v_r \cot \phi$ may often be neglected. A calculation shows that for an angle of 60° and head of 60 ft., v_2 is about $6\frac{1}{2}$ per cent. less than in the case of radial vanes.

Angle of Vanes at Outlet Surface.

The condition determining the angle which the vane should make at the outlet surface of the wheel is that the water should leave with *no* tangential velocity, therefore with no kinetic energy in the direction of the wheel's motion.

In the case of the inward-flow turbine we are now considering, it is not difficult to see how the necessary construction is obtained.

Draw v_r (Fig. 96) normal to the inner circumference, and make it, say, $= \frac{1}{8}$ of the velocity due to the total head $= \frac{1}{8}\sqrt{2g\text{H}}$.

Draw v_1 tangential to the inner circumference, and make it $= 0 \cdot 66 \sqrt{2g\text{H}} \times \frac{r_1}{r_2}$ as the best practical value; complete the parallelogram; then the water has a radial flow v_r of its own, also a forward velocity v_1 due to the wheel, and we want it to issue with no tangential velocity relative to the earth. Evidently the actual velocity it has (represented by v_0) should have a backward component = the forward resultant velocity along the same line. If a is the angle required, then $v_0 \cos a = v_1$. Thus a is obtained, since

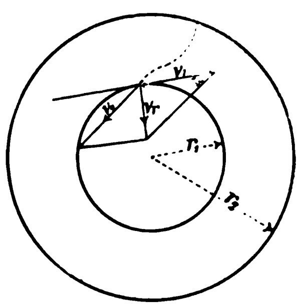


Fig. 96.

$\tan a = \frac{v_r}{v_1}$, v_r and v_1 being at right angles to one another.

* Really the left-hand expression divided by the right = η (the hydraulic efficiency); or the equation should be

$$\frac{1}{g}(v_2 + v_r \cot \phi) v_2 = \eta \text{H}.$$

A similar construction will be required in axial-flow turbines, v_r being drawn normal to the *outlet surface of the wheel*, whatever it may be.

RADIAL-FLOW TURBINES.—SHAPES OF VANES.

The shape of vane adopted in practice is like that shown in Fig. 94, a contrary curvature being adopted. This is to avoid as far as possible the contraction of the vein of water flowing from the buckets of the wheel, for it has already been pointed out that wherever a vein of water issues from an orifice which is of such a shape that it *contracts* the jet, energy is wasted; and at p. 58 coefficients of *hydraulic resistance* are given enabling us to calculate the waste of energy in some cases. Now the shaping of the vanes as here shown gives a closer approach to a uniform depth of channel for the water near the

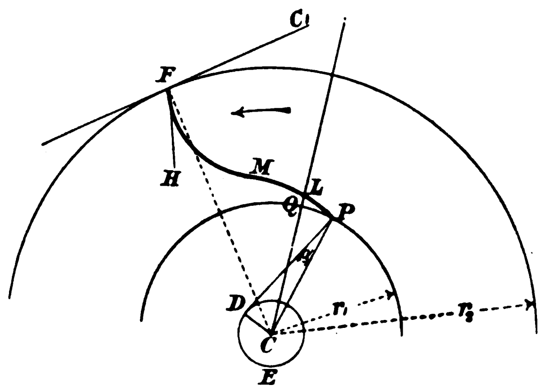


FIG. 97.

inner circumference, and hence less contraction and less waste of energy. A very marked contrary curvature would be objectionable, for reasons already explained.

It can be shown that if the *angles* of the vanes are properly arranged and the proportions of the turbine correct, the shape of the vane itself is not of great importance, as turbines with widely differing shapes of vanes give nearly the same efficiency. The surfaces of the vanes should be smooth and the vanes thin, all *abrupt* changes in the curvature of the water-path being carefully avoided. A good construction for the vane outline is as follows (Fig. 97) :—

Construction.

With centre C and radii r_1 and r_2 draw the outline of the inner and outer circumferences of the wheel.

Take any point P on the inner circle and make C P D = a.

Draw C D perpendicular to P D, and a circle from C with C D as radius. Make P Q = the inner pitch, and draw a radius C Q L.

Imagine a thread wound round circle C D and fastened, say, at E, this thread bearing a pencil at P. Let the thread be unwound; the pencil will trace out a curve like P L M, M being a little to the left of C Q L.

Draw F C_1 a tangent at point where vane cuts the outer circumference, and make the angle C_1 F H = $180° - \phi$.*

Then complete the vane from F to M by hand, or with an arc of a circle; the wheel revolves in the sense indicated by the arrow.

Radial Outward-Flow Turbines.

Outward-flow turbines were used before those with inward flow, but the construction is very similar in the two cases. The Fourneyron

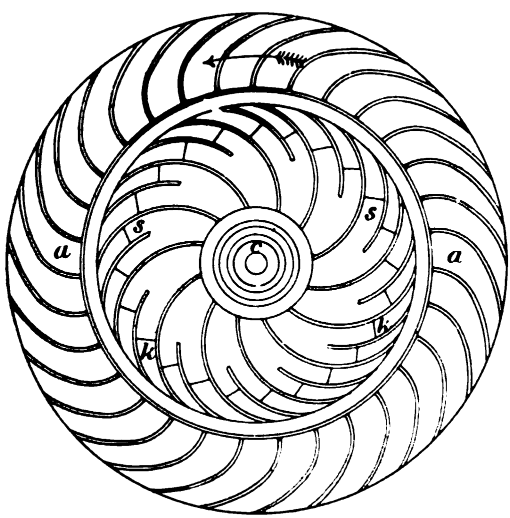


Fig. 98.

turbine is one of the best known of this class. The water, under pressure due to the head, enters at the centre c (Fig. 98), passes

* In the Thomson turbine ϕ is usually about 60°.

through the guide passages *s k* into the wheel *a*, which is driven round in the direction indicated by the arrow. This wheel is not in any sense self-governing, like the Thomson turbine, as an increase of speed causes an increase of centrifugal force, which in this case acts with the flow and tends to *increase* the power and speed of the turbine.

The regulation is usually effected by a cylindrical sluice gate, and often the wheel is divided by horizontal partitions into parts which are, in fact, separate turbines, the water being then shut off from these portions successively, if diminished output be required.

This turbine is *not* used with a suction tube, whereas the Thomson turbine and many others have such a tube, which often adds 4 per cent. to the efficiency of the turbine, and allows it to be placed at any convenient height (within certain limits) above tail water.

A complete section of a Fourneyron turbine, of recent date, is shown in Fig. 115. This type of wheel is now rarely used.

In the rules for radial-flow turbines deduced above, the radial velocity of the water is assumed. If this somewhat approximate method be not accurate enough, it is easy to get out exact trigonometrical relations between the various angles, speeds, etc. Our space, however, does not admit of an explanation of this somewhat tedious method. Graphic solutions, where possible, are preferable.

Axial-Flow Turbines.

As the name implies, the water in these turbines enters the guide vanes in a direction *parallel* to the axis of the wheel. Hence the guide vanes A D, B C, etc. (Fig. 99), should be normal at A and B, and nearly parallel straight lines making the proper angle with K N at D and C. Thus we have the following construction for the shape of guide vanes.

Make D C equal to the distance from centre to centre of vanes at middle radius of outlet guide surface. Make the angle C D F $= \theta$, calculated as in the Thomson turbine. Draw C F G perpendicular to D F, and thus find G the centre from which the arc F A is drawn. The other vanes are shaped in the same way.

The guide vanes are secured between two concentric casings, the water passing through the guide passages A D C B, etc., into the wheel passages. The wheel vanes are not quite normal to the inlet surface L M, and they may be drawn as follows:—Make R S equal to the pitch, and draw S P and R Q, making the outlet angle α calculated as before. Next, to find J, the point in which the vane outline meets the inlet surface L M. Draw any line H I making the inlet angle ϕ with L M, and make I H O a right angle. To find ϕ, set off along the vane outline a distance, say D F, to represent the initial inflow velocity of the water, draw F E horizontally to represent the velocity of the wheel; then E D makes with L M the required angle. Bisect the angle R O H (R O being at right angles to S P), and through P draw P J perpendicular to this bisector. This determines the point J. Draw J X parallel to H O to meet R O in X, which is the centre of the arc J P. One vane outline being found, all the others can be drawn by template. It should be noted that the arc J P should be "eased off" near its junction with the straight line P S, so that the change of curvature may be gradual. This can be done by using an elastic strip as a ruler.

This figure is supposed to show part of a development of the

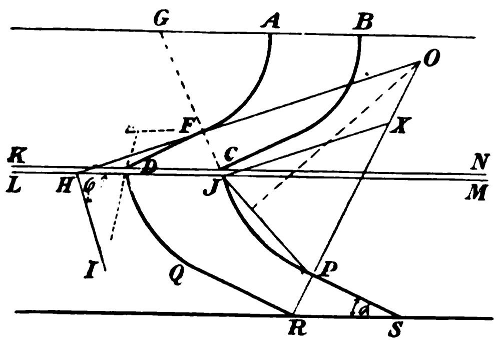


FIG. 99.

cylindric surface concentric with the shaft axis, and passing through the point of mean radius of the guide and wheel passages.

The method of finding the various angles has been fully explained in connection with the Thomson turbine. It is evident that points on the inlet surface of different radii have different velocities. Hence the value of v_2 taken is that for the mean radius, the surfaces being usually made slightly helical to agree with the change in v_2. In impulse wheels the surfaces are not helical.

The velocity of flow into the wheel is given by the rule $v = K\sqrt{2gH}$, where K is about 0·67 for Jonval turbines. A summary of the usual values of K for different wheels is given at page 167.

The ratio of the area of the guide passages to that of the outflow wheel orifices is also required. This is nearly unity in Jonval turbines. The ratio of outside radius to breadth of wheel must be fixed. This, in the Jonval turbine, described later on, is $\frac{48 \cdot 22}{17 \cdot 72}$, or about $2\frac{1}{4}$ to 1.

These data fix the main points of the design. For further details the student should consult standard works on the turbine.

Some authorities give the guide vane and wheel vane angles as measured from a *normal* to the inlet and outlet surfaces, or in other words, the complement of the angle here taken.

GRAPHIC METHOD IN TURBINE DESIGN. PRESSURE TURBINES.

The fundamental equation is

$$\eta H = \frac{w_1 V}{g}$$

where η is the hydraulic efficiency, H the total head available for driving the turbine after frictional losses outside the turbine have been deducted. w_1 = velocity of whirl = horizontal component of initial velocity (v_1) of water entering the wheel. V = velocity of wheel at inlet surface (usual notation is here adopted). This velocity was denoted by v_2 in previous text. Since the wheel passages are always full of water under pressure, the vertical component (in parallel flow turbines) of the water's velocity is constant, and is the velocity with which the water is rejected into the tail-race—which velocity should be small. Suppose the waste of energy in this way to be, say, 8 per cent. of that due to the total head. Let H = 30 feet, say, $\frac{8}{100} \times 30 = 2 \cdot 4$ and $\sqrt{2g \times 2 \cdot 4} = 8 \cdot 02 \sqrt{2 \cdot 4} = 12 \cdot 41$ feet per second. Express this in terms of our unit $\sqrt{2gH}$. $12 \cdot 41 = x\sqrt{2g \times 30}$, whence $x = 0 \cdot 28$ or $u = 0 \cdot 28 \sqrt{2gH}$.

Imagine 12 per cent. of the available energy wasted in friction, we have, therefore, 80 per cent. remaining. The velocity due to 80 per cent. of 30 feet is $39 \cdot 2$ feet per second $= 0 \cdot 89 \sqrt{2gH}$. This 80 per cent. is spent in producing the horizontal velocity and in producing pressure in the clearance space. Of the $0 \cdot 89$ times our unit we may take $0 \cdot 66$ as the horizontal component w_1 and hence, by construction, we can find the required angles of guides and wheel vanes.

Set up A B (Fig. 100) to represent $0 \cdot 28$ and lay off the horizontal distance B D to represent 0·66,* this gives A D representing v_i, the direction of inflow velocity and parallel to direction of guide vanes at end nearest wheel. To find the proper velocity for wheel, apply the principle of momentum. The water enters the wheel with a horizontal velocity $w_1 = 0{\cdot}66\sqrt{2gH}$. It leaves with *no* horizontal velocity or momentum, hence, since force = change of momentum per second, the horizontal force exerted by each 1 lb. of water is $\frac{w_1}{g}$ lbs. weight. If V = the velocity of the wheel at inflow circumference, since force × velocity per second = work per second, $\frac{w_1 V}{g}$ ft.-lbs. is work per second in this case.

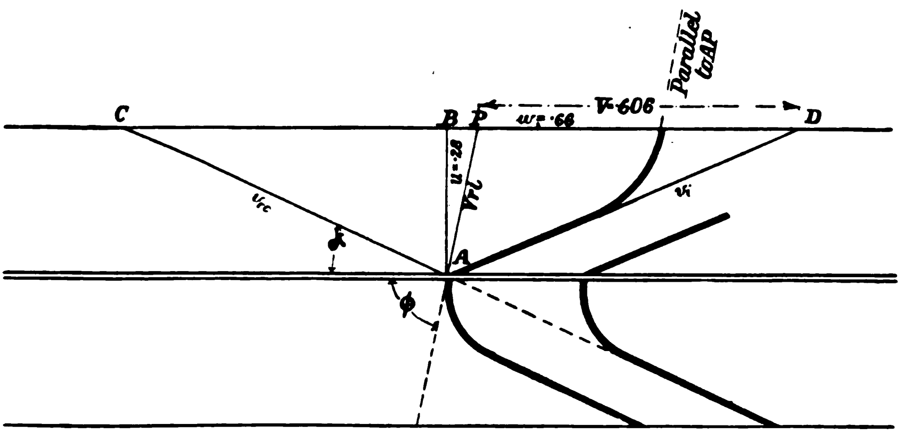


FIG. 100.

Hence

$$\frac{w_1 V}{g} = 0{\cdot}80\ H$$

or

$$V = \frac{0{\cdot}80\,gH}{0{\cdot}66\sqrt{2gH}} = \frac{0{\cdot}40 \times 2gH}{0{\cdot}66\sqrt{2gH}} = 0{\cdot}606\sqrt{2gH}.$$

Measure back from D a distance D P to represent 0·606; then, since A D represents v_i, we have P A the direction of the wheel vane at the inlet end. To get the angle of the wheel vane at the outlet end, combine *u* and V in a similar way. This may be done by setting off B C to represent V, and then C A is the direction of the vane at

* In these constructions the turbine-rod method of Von-Reiche as developed by Professor Unwin in his Inst. C.E. lecture on "Water Motors" is followed.

the outlet circumference. With regard to the combined area A of the passages, $A = \frac{Q}{v}$, where Q is the flow through the turbine in cubic feet per second and v is the velocity normal to the plane of A.

Radial-flow turbines may be designed by setting out graphically as above, and then, in the drawing, projecting from straight lines to circles occupying corresponding positions.

The method already given at pages 149 and 150 is, however, simpler in the case of radial-flow turbines, than the graphic method here referred to.

Impulse Turbines.

A reaction turbine is designed to work always full of water, continuity of flow being essential to efficient working. To obviate the difficulty of low efficiency with variable flow, experienced when reaction turbines are used with very variable loads, *impulse* turbines have been introduced, in which the wheel passages are supposed never to be filled with water, the water in the wheel being under atmospheric pressure only. In a reaction turbine continuity of flow necessitates very careful design of passages as to section, etc.

In designing an impulse turbine, since the passages are not to be filled, there is much more latitude for the designer, and any dimensions (within wide limits) may be chosen which seem convenient. In these turbines the quantity of water passing does not so much affect the efficiency, hence speed regulation may be effected by partially closing the guide passages. Turbines of this class, often called *Girard* turbines after the inventor, are much used now, having in many cases, especially on the Continent of Europe, displaced the older reaction turbines.

The velocity of flow into the wheel in an impulse turbine is determined by the rule $v_i = K\sqrt{2gH}$. K usually ranges in value from 0·9 to 0·95.

For a very neat construction giving vane angles, etc., of an axial-flow impulse turbine, refer to Fig. 101, where the unit is $\sqrt{2gH}$. First decide the energy to be rejected into the tail-race per lb. flowing. Let it be 10 per cent.; the velocity corresponding to $\frac{1}{10}$ H is v_p (or u) $= 0 \cdot 32\sqrt{2gH}$. Draw the triangle of velocities C A B so that v_p (here assumed constant) is the vertical component of the initial velocity v_i; then C A is the direction in which the water enters the wheel; and hence the direction of the guide vane here, the other end being vertical. Bisect C B in D. Then C D represents the proper velocity of the wheel (see page 159), and A D is the direction of relative motion of water and wheel, and is tangential to the wheel vane at its inlet end. The relative velocity remains unchanged in an impulse turbine. Set off B E = V, the velocity of the wheel; then A E (= D A) is the direction of the relative motion of water leaving the wheel, and hence tangential to the vane at outlet end. The three

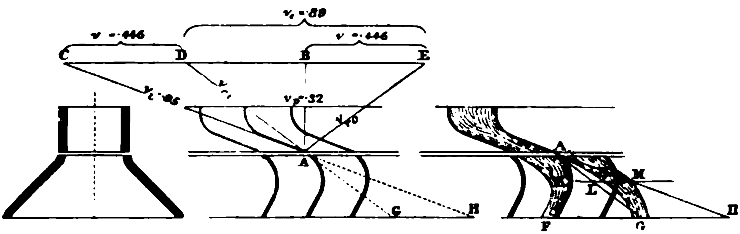


FIG. 101.

angles required are thus determined. Since the relative velocity v_{ri} is changed to v_{ro} in passing through the wheel, ∴ D E (representing V_e) shows the velocity utilised in the wheel. Hence each pound loses H ft.-lbs. of energy, of which $\frac{V_e^2}{2g}$ is utilised. Hence

$$\eta = \frac{V_e^2}{2gH} = 0 \cdot 89^2 \text{ (from figure } V_e = 0 \cdot 89 \sqrt{2gH}) = 0 \cdot 79.$$

The wheel vanes are usually widened out to allow free deviation, as shown in the cross-section to the left.

It is sometimes useful to determine the exact absolute path of the water. If there were no wheel vanes the absolute path would be

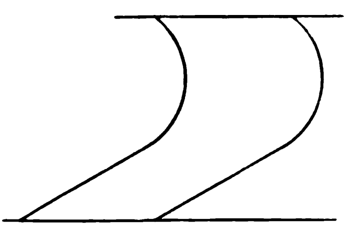


FIG. 102.

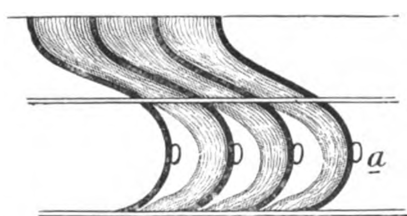


FIG. 103.

A H, and the relative path A G. The wheel vanes, however, deviate the water the distance L K from A G. Set off M N = L K. Then N is a point in the absolute path. Other points may be found in a similar way, and the absolute path drawn. Or the absolute path being chosen, the wheel outline can be found. The absolute path

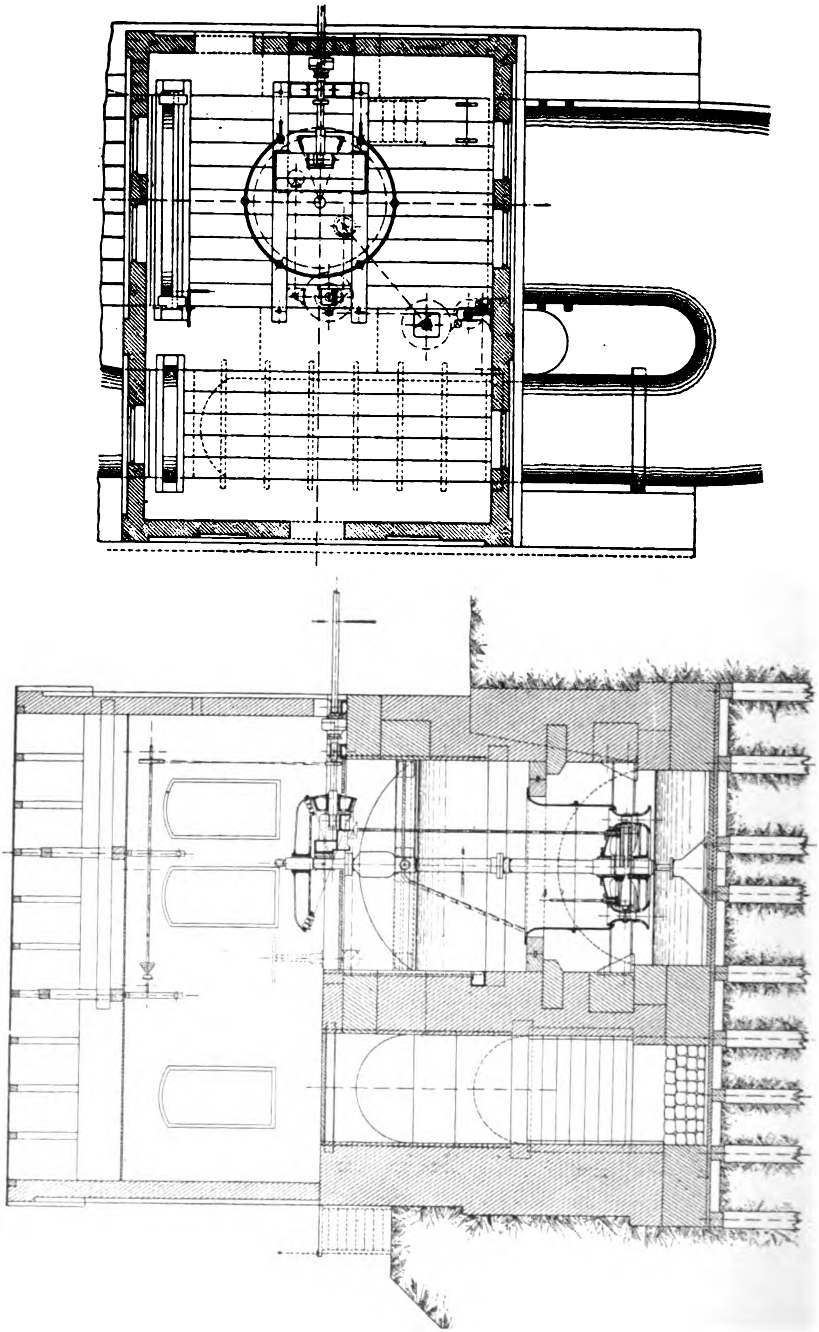


FIG. 104.

should be of continuous and tolerably uniform curvature, and the water-stream a convergent rather than a divergent one.

The sectional area of the *guide* passages is determined from $Q = A\,v$, since they are always full. Since the moving wheel vanes *obstruct* the flow from the guides, it is usual to find A (the outflow area of the guide passages) from the rule $v = 0 \cdot 85 \sqrt{2\,g\,H}$, instead of $0 \cdot 90$ or $0 \cdot 95 \times \sqrt{2\,g\,H}$.

In axial impulse turbines the angles just referred to are usually the same at the inner as at the outer circumference, helical surfaces not being adopted here.

The shapes of the wheel vanes are shown roughly in Fig. 102, the upper or inlet ends being very much hooked or scoop-shaped, the

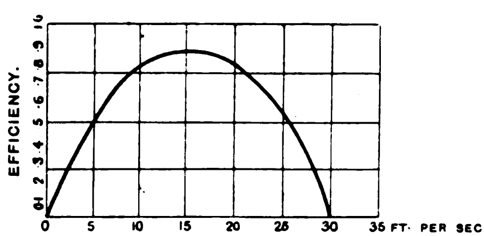


FIG. 105.

lower end straight for some distance. The number of vanes is always *greater* in the guide apparatus than in the wheel, to allow free deviation, as already described.

Fig. 103 shows the path followed by the water, the wheel being supposed at rest. The water passes out of the upper guide portion into the lower wheel buckets, which are *not filled.* The method of ventilating the buckets is shown by the apertures, *a*, etc.

Fig. 104 shows a section of a Girard axial-flow impulse turbine, made by the continental firm whose Jonval turbine is shown in Fig. 115. The figure shows the construction of the wheel and the method of supporting and fixing the same.

Efficiency and Velocity.

Fig. 105 shows roughly how the efficiency of an inward-flow impulse turbine is affected by varying the speed. It will be seen that the best velocity is very nearly half that at which the wheel would run if unloaded.

Mixed-Flow Turbines.

In many typical mixed-flow turbines of American design, the water follows an inward and downward course through the wheel. At entrance the water moves nearly radially, the motion having a small downward component, but as it passes through the wheel the downward component becomes more important. A section of such a wheel is given on Fig. 106.

The Victor Turbine.

This turbine has been employed in many important installations; at Montmorency Falls in Canada, at Kern River, California, where power is generated to be conveyed electrically to Los Angeles, a distance of 125 miles, and in many other places in the United States as well as in this country.

In the section (Fig. 106) will be seen G the guides, T the turbine wheel vanes, L the lifting or regulating gate, R rack attached to the same, moved by the pinions P, driven by the bevel gearing B from the governor. S is the turbine shaft, K a wood bearing, *s* a removable shoe, B the bearing surface of footstep, where W represents the lignum vitæ or other wooden fixed part of friction pair, F being the adjustable lower extremity of footstep. The pulley on left end of pinion shaft is connected to a balance weight which is employed to counterpoise the weight of the gate.

This is the standard vertical form of turbine, but where the wheel is direct-coupled to an electric generator, and in many other cases, the wheel is placed with its axis horizontal. In large turbines, such as those at Montmorency Falls, there are guide passages at two opposite portions of the inlet circumference only, and a sectorial gate moved by the governor opens a greater or less number of these passages as required by the exigencies of load.

The Victor turbine has a high efficiency, as shown by the following data.

Efficiency.

In usual practice the makers guarantee an efficiency of 80 per cent. at full gate. Actual tests made at the Holyoke Testing Flume show for a small 12 inch 25 horse-power wheel, an average efficiency (for 9 tests) at full gate of 80·84 per cent., at $\frac{8}{10}$ full gate (7 tests) of 77·31 per cent., at $\frac{6}{10}$ full gate (7 tests) of 70·40 per cent., and at half gate of 65·64 per cent. A pull of $\frac{1}{2}$ lb. applied at a distance

1·6 feet from the centre of the shaft was sufficient to start the turbine when empty.

A larger wheel developing at full gate 252 horse-power, gave 80·62

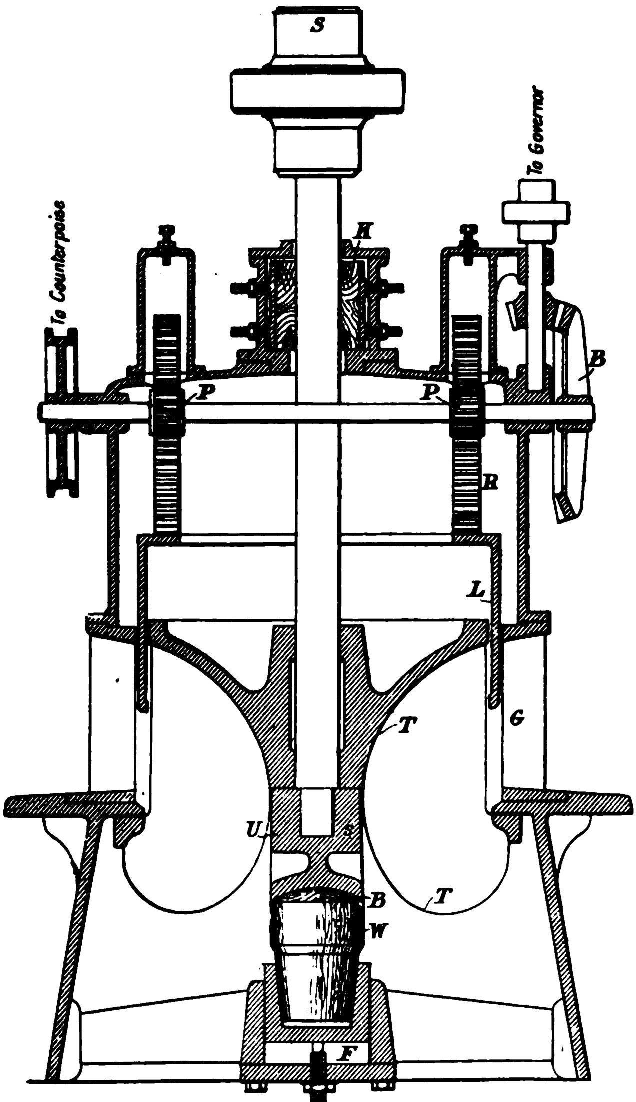


FIG. 106.

per cent. at full gate, 79·73 per cent. at $\frac{8}{10}$ full gate, 71·06 per cent. at $\frac{6}{10}$ full gate, and 65·04 at half gate. In this case a torque of 36 lb.-ft. sufficed to start the wheel when the flume was empty, the weight of the dynamometer and that part of the shaft which was above the lowest coupling being suspended by a ball bearing.

The results of efficiency tests of the 1000 horse-power units installed at the Montmorency Falls, Quebec, are shown in Fig. 107.*

These are certified by the chief engineer of the Quebec Railway Light and Power Company, and in them the efficiency losses in the direct-coupled generator have been allowed for.

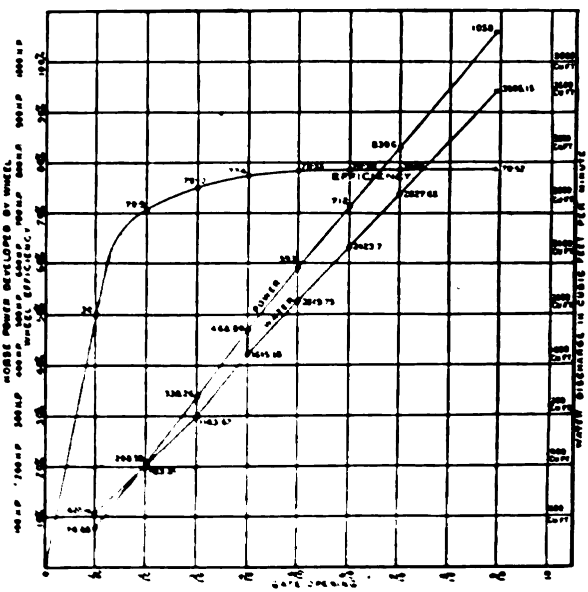


FIG. 107.

The "Hercules," another wheel of similar type, is shown in Fig. 108. In regard to the velocity of the wheel, we find, from data published by the makers, that a wheel 2 feet in diameter, suited to work with a head of 40 feet, should revolve 308 times per minute. This gives rise to the rule $v = 0 \cdot 635 \sqrt{2gH}$, v being the circumferential velocity of the wheel. The rule agrees very nearly with that given by Professor Unwin for a Thomson turbine. The curves of vanes etc., are the result of many experiments, and the shapes arrived at

* Published in the "Engineer," January 11, 1904.

give a high efficiency, especially at less than full flow. The gate is placed between the guide wheel and turbine wheel, the vanes of the latter being provided with several projections to take—to some extent—the place of sub-divisions, and thus improve the efficiency at part gate flow.

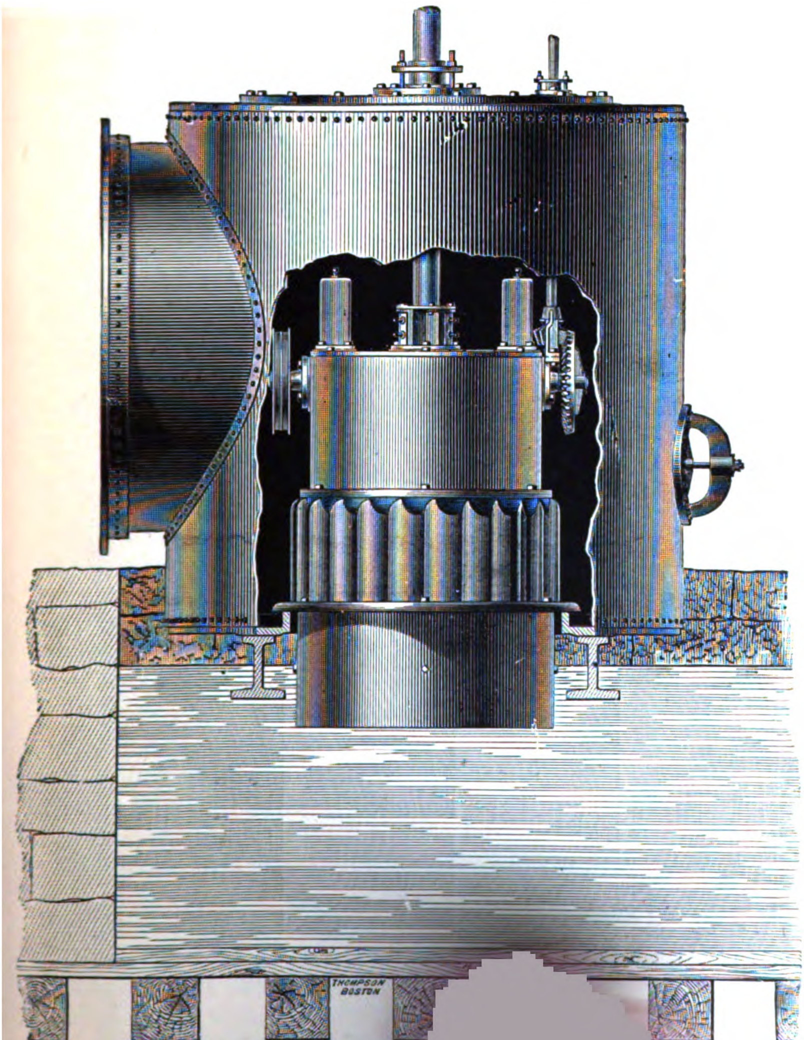


FIG. 1 ... M 2

The results plotted in Fig. 109, certified by Professor Thurston, are remarkable, showing a maximum efficiency of 87 per cent., with 70 per cent. when the gate opening was less than one-half of the full amount, or the flow 60 per cent. of that when the gate was full open. These figures show incidentally, what the reader has no doubt already noticed, that half gate opening does *not* mean half of full flow, but something considerably more.

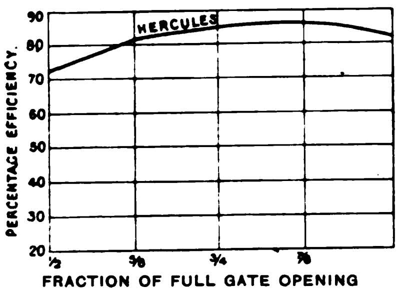


FIG. 109.

Another advantage claimed for the "Hercules" turbine is that, owing to the wide wheel passages, objects may pass through the wheel with but very little injury. Also the pressure on the footstep bearing is small owing to the direction of flow being largely radial.

The "British Register-gate" turbine is of British construction, and very similar to the "Hercules," but does not give quite so high an efficiency.

TURBINES FOR LOW FALLS.

Pressure turbines seem on many accounts to be most suitable for low falls, but in some cases impulse turbines may be employed with advantage. Everything depends on the conditions to be met. For instance, if the water supply varies from say 2000 cubic feet per minute in winter to 400 cubic feet per minute in summer, and the average efficiency is to be fairly high in all cases, impulse turbines may be necessary, as it is essential to the proper and economical working of pressure turbines that they shall always run full.

Of pressure turbines, a carefully designed parallel-flow wheel with several compartments which can be closed in succession offers many advantages with very variable flows, though its efficiency may be lower than that of a good radial-flow turbine running under favourable conditions.

Turbines of the parallel-flow type referred to have been employed with success on the Continent and in this country under the conditions obtaining in many English rivers, with falls as low as 2 feet. The following peculiarities should be noted:—

(1) With low falls a great variation of effective head may be anticipated, as the tail-water usually rises more than the head-water

in time of flood, and when quantity is greatest effective head is least. Almost constant power may be obtained with properly designed turbines.

(2) Usually no artificial storage of water is possible owing to the large supply necessary.

(3) Turbines for low falls are more costly per horse-power than for high falls, and proper design to suit all the circumstances is more necessary in the case of low falls.

Impulse motors introduced by Poncelet, when he put properly curved floats on his undershot water wheel, were further improved by Girard, when he adopted the same principle in his turbine and introduced free deviation, as already described at page 157. To use these turbines to advantage, it is necessary that they shall not work submerged, and this fact limits their use in the case of low falls with variable tail-water level.

Haenel has to some extent got over the difficulty by adopting a Girard turbine in which the water in passing through with a constant relative velocity *fills* the buckets entirely. This turbine is the connecting link between impulse and reaction turbines, and its use for low falls with variable tail-water level, but with fairly constant effective head, has been successful, but the speed must be kept nearly constant for good results with impulse turbines. Pressure turbines must however be regarded as the most generally suitable for the low falls utilisable in this country. They have the great advantage of working with a suction tube, and hence can be placed above tail-water. The inward-flow turbine of the Thomson type built in several stages to meet the variable water supply, or with movable guide-blades regulated by a proper governor, has been successfully employed, but the parallel-flow pressure turbine of the Jonval type built in compartments has, perhaps, been brought to the greatest stage of perfection for this purpose. In general, the outermost compartment utilises the minimum supply with maximum head, whilst the inner compartment gives practically the same power under a reduced effective fall, either alone or together with the outer compartment.

Each compartment is regulated independently by opening the requisite number of guide buckets in one compartment after the other; thus the power and speed may be kept nearly constant.

Such a turbine at the Zurich waterworks with three compartments, the effective fall varying from 10 feet 6 inches to 4 feet 9 inches, gave the following results: with the outer compartment alone and maximum fall ($10\frac{1}{2}$ feet), at normal speed of 25 revolutions per minute, the efficiency was 73·7 per cent.; the outer and middle compartments together with fall 7 feet gave 75·4 per cent., and all three compartments together gave 80·7 per cent. Turbines of this class have been successfully employed on falls as low as 1 foot 6 inches.

Messrs. Escher, Wyss and Co. have also introduced a "cone" turbine with inflow outline conical in shape and with a diagonal flow, which can be subdivided into compartments, and is said to give good results. It may be considered as intermediate between parallel and radial flow types, the mixed flow turbines being less definite and inclining to one or other of the main types. Large turbines of this class have been installed at Lyons, to suit a fall varying from 27 to 40 feet, the speed of each being about 120 revolutions per minute, and the useful horse-power 1250. In Italy, turbines of this class give 300 brake horse-power at 70 revolutions per minute under a fall varying from 14 feet 9 inches to 12 feet 6 inches.

To the different kinds of gates employed our space only admits of a passing reference. Cylindrical gates for radial-flow wheels should be employed only when the wheel is built in compartments, and the gate is used to close altogether one or more compartments, as a partly open gate of this type is very wasteful of energy. If, however, it is only desired to regulate the wheel for varying *load* as distinguished from varying water supply, this type of gate may be employed. The question of promptitude of action has to be considered, especially when the turbines are employed to drive electric generators. Revolving gates placed outside the guide passages, or better still between the guide wheel and turbine proper, have been used. By turning such a gate all the guide-passages are simultaneously altered in area, hence the ratio of the guide area to wheel area, or the degree of reaction, is varied. The most correct and efficient method of regulating radial-flow wheels, however, is that designed by Professor James Thomson nearly forty years ago, viz. the method of having guide-vanes movable on hinges, so that not only the area of the guide-passages but also the angle of the guide-vanes relative to a tangent to the wheel may be varied. With such a method of regulation as high an efficiency as 70 per cent. with only quarter-gate opening has been obtained.*

Summary of Rules as to Velocity, etc.

As one of the first things to be determined is the velocity of the inflow circumference, the following are the best average values of various authorities:—

* Proceedings of the Institution of Civil Engineers, vol. clvii.

Name of Wheel.	Velocity of Inflow Circumference.
Reaction turbines :—	
Thomson (inward radial flow) ..	$0.66\sqrt{2gH}$
Jonval (axial flow)	$0.64\sqrt{2gH}$
Fourneyron (outward radial flow) ..	$0.625\sqrt{2gH}$
Mixed flow (various)	0.63 to $0.77\sqrt{2gH}$
Impulse turbines	0.45 to $0.5\sqrt{2gH}$

Velocity of Flow from Guide Passages into Wheel.

	Velocity of Flow.
Axial-flow turbines (like Jonval) ..	$0.67\sqrt{2gH}$
Fourneyron	$0.75\sqrt{2gH}$
Mixed-flow turbines	$0.63\sqrt{2gH}$
Impulse turbines	0.9 to $0.95\sqrt{2gH}$

Classification of Turbines.

	Reaction or Pressure Turbines.	*Impulse Turbines.*
Complete Admission.	Wheel passages filled; pressure in clearance space.	Wheel passages usually not filled; free deviation; no pressure in clearance space.
	Inward, axial, diagonal, or mixed flow; discharge usually below tail water or into suction tubes.	Discharge above tail water. No suction tube.
	Outward flow; discharge usually above tail water, and without suction tube.	Inward, outward, or axial flow, with complete or partial admission.

XV.

SOME TURBINES AND TURBINE-POWER INSTALLATIONS.

Power from Niagara Falls.

To utilise at least a small part of the immense power running to waste at Niagara Falls has been the dream of engineers for many years. The late Sir William Siemens, in his address as President of the Iron and Steel Institute in 1877, gave expression to the general feeling amongst engineers that in the near future advantage would have to be taken of the great water-power stores of energy provided by nature, and he computed that all the coal raised throughout the world barely represented the power of Niagara. In Continental countries, such as Switzerland and Sweden, much had then, and more had since, been done in this direction, and now, in the beginning of a new century, the greatest water-power installation in the world is approaching completion at Niagara. Those completed and the installations now in progress, will total over half a million horse-power.

The work of the Cataract Construction Company, or Niagara Falls Power Company—a pioneer company of wealthy capitalists guided by some of the foremost engineers of the time—aimed at the useful development of some 100,000 horse-power out of the seven millions or so said to be represented at the Falls. A full description of this great work is beyond our province, but a brief outline of the plan adopted may be of interest. It consists in arrangements for tapping the Niagara river, at a place about a mile above the Falls, by a canal 250 feet wide at its junction with the river, and of an average depth of 12 feet.

This canal is lined with masonry and contains 10 inlets, by which the water is taken to the wheel-pit. This pit is 178 feet in depth, and connected with the river below the falls by a tail-race, consisting of a tunnel 7000 feet long, 21 feet high, and 18 feet wide at its largest part, with a net section of 386 square feet. Over 1000 men were employed for three years in constructing the tunnel, in which more than 16,000,000 bricks were used for lining. The water brought by the canal from the higher portion of the river is employed to drive twin-turbines; in power house No. 1, ten units of 5000 horse-power each, with a fall of 140 feet. They are of the Fourneyron type, but of special design. The water passes from the canal to each set of turbines, through a penstock or riveted iron tube 7½ feet in diameter. Each of these twin wheels is three stories high, and is surrounded by a cylindrical gate or sluice worked by a governor of special design. The water from the penstock passes up through the guide portion of the upper wheel, acting *upward* on the cover of that wheel, which forms a balancing piston, as shown in Fig. 110; whereas in the lower wheel of the pair the water acts as in the ordinary radial outward-flow wheel. It is calculated that there is thus obtained a resultant upward force or pressure from 149,000 to 155,000 lbs., depending on the gate openings. Each pair of wheels is connected to a great vertical shaft, consisting of a steel tube 38 inches in diameter, but solid at the journals, where it is 11 inches in diameter, the upper end of this shaft bearing the revolving field of an immense alternator dynamo machine. The weight of the shaft and revolving part of dynamo (which forms a fly-wheel) is 152,000 lbs., so that the bearings

—a Fontaine oil bearing or step at the lower end and one or two thrust bearings near the top of the shaft—have only to deal with the difference of this weight and the upward force of the water when the wheels are at work.

The wheels are to discharge 430 cubic feet of water per second, making 250 revolutions per minute, giving out, at 75 per cent. efficiency, 5000 horse-power. On the Canadian side larger units (10,000 horse-power inward flow) are now being installed.

Each dynamo or alternator weighs about 170,000 lbs., the revolving part, or field-ring, weighing 79,000 lbs., the armature being stationary. The current is given off at a pressure of about 2400 effective volts, with the low frequency of 25 cycles per second; "step-up" and "step-down" transformers being used at the near and remote ends of the longer circuits.

The wheel-pit is really a long slot cut in the rock, and at about 140 feet from the surface the turbines are placed on plate girders, there being thus one clear tail-race, connected with the main tail-race tunnel, under all the turbines.

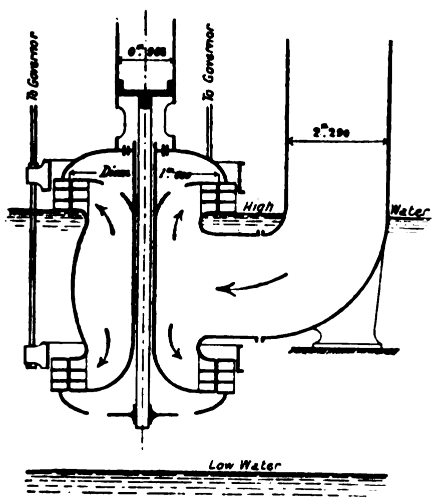


FIG. 110.

Fig. 110 shows an outline sketch of the turbine with regulating sluice. This sketch shows very clearly how the water is admitted through the aperture in the upper part of the casing to the balancing piston; by the aid of the pressure of water on this the great weight of the shaft is nearly balanced.

Fig. 111 gives a good general idea of the arrangement of the turbines, turbine shafts, dynamos and regulators; whilst Fig. 112 shows an accurate section of the upper turbine from a working drawing kindly supplied by the makers, the I. P. Morris Co. of Philadelphia. This illustra tion will be readily understood when compared with Fig. 110. The balancing piston gives, at the normal speed of 250 revolutions per minute, and under the normal head of water, a slightly preponderant upward pressure on the collar bearings of the main shaft.

In power house No. 2 there are 11 units each of 5500 horse-power. They are of the inflow type with twin draft-tubes in the centre, at the junction of which, is a balancing piston.

The Niagara Falls Paper Company's installation, described more fully on the next pages, is an entirely separate undertaking, with its own supply, but discharging its waste water into the Cataract Company's tunnel.

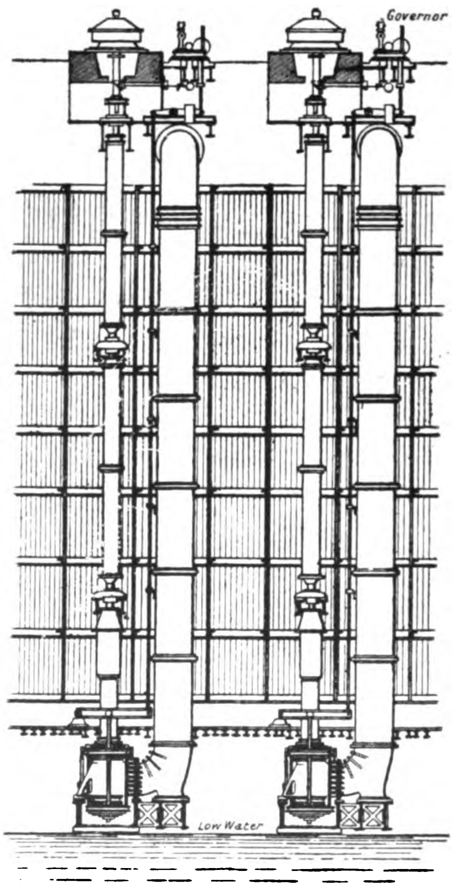
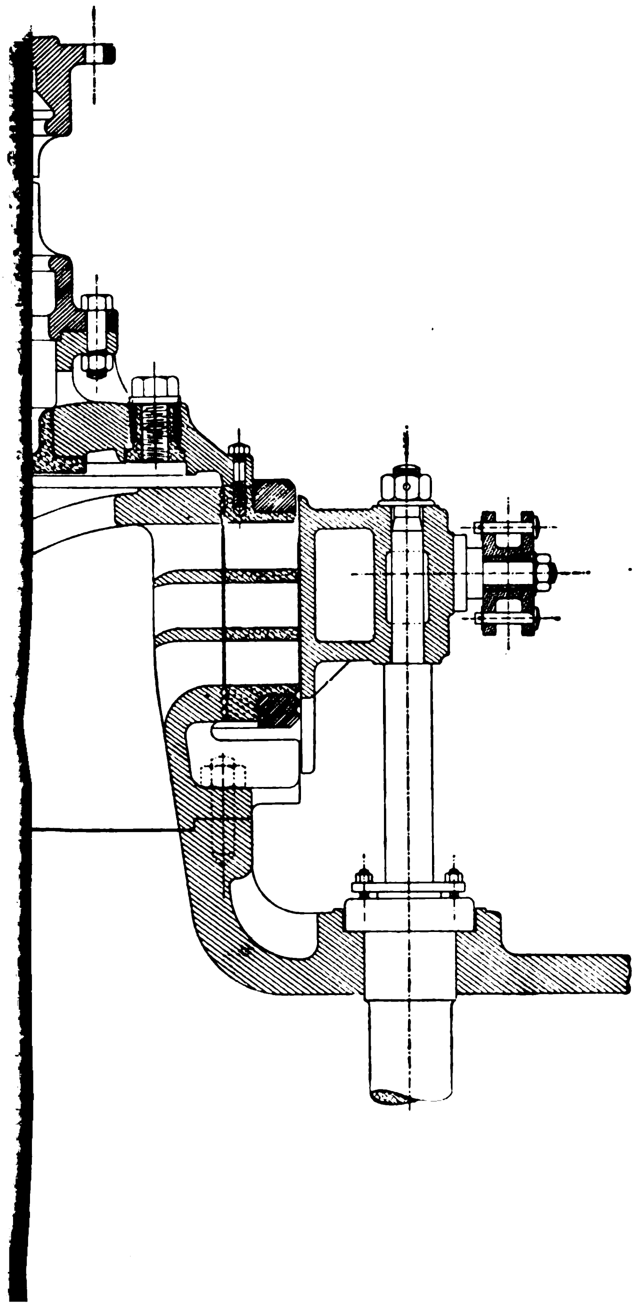


FIG. 111.

TURBINES OF THE NIAGARA FALLS PAPER COMPANY.

Already the idea of utilising some of the power of Niagara Falls has been so far successfully carried out, that the Niagara Paper Company has had its turbines at work for some years. These turbines are shown in Fig. 113, where an elevation of the penstock with a section of three of the turbines is given to the right, and to the left a plan showing the position of the turbines with respect to the penstocks which convey the water to the wheels. The greater penstock is an immense tube 13 feet 6 inches in diameter, the metal being $\frac{7}{8}$-inch thick, riveted together *in situ* by hydraulic riveters, and it supplies four turbines of the Jonval type, each of 1100 horse-power, the head being 140 feet. The smaller penstock is 9 feet in diameter, supplying two turbines. The turbines were made and the whole of the hydraulic installation carried out, by Mess[illegible] Woo[illegible]



[*Face page* 170.

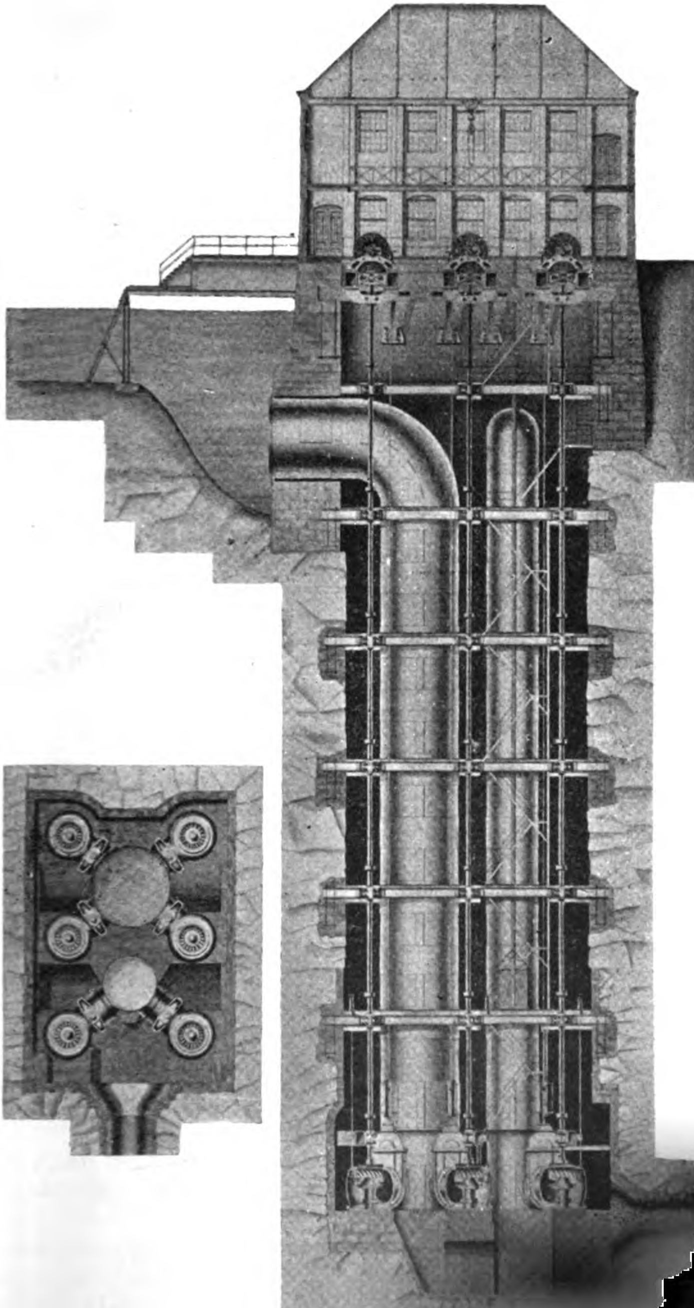


FIG. 113.

of Philadelphia. This firm usually guarantees a turbine efficiency of 80 per cent. The flow required for each turbine is 86 cubic feet per second, or nearly 144 tons of water per minute.

As the large penstock supplies four turbines, the flow will be about 344 cubic feet per second, which, with a diameter of 13½ feet or an area of 143 square feet, gives a velocity of rather less than 2½ feet per second.

The wheels are fitted with gates which are the patent of Mr. Geyelin, the engineer for Messrs. Wood, who has designed and carried out the installation. These are more fully shown in the section of the wheel given at Fig. 114.

The difficulties met with, and successfully overcome, in this case were of no ordinary kind. For instance, every engineer knows the trouble there is in properly supporting a very large vertical shaft.

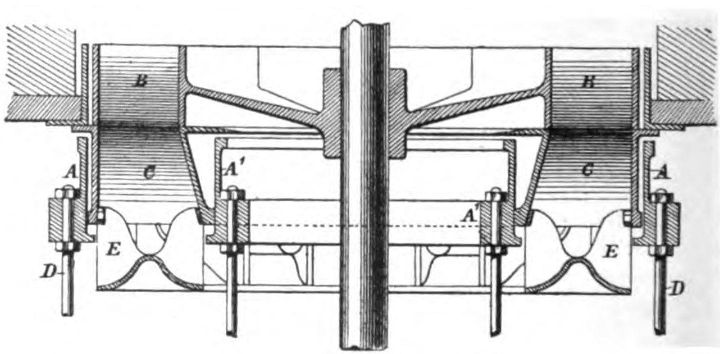


FIG. 114.

But ordinary vertical shafts are very small things compared with these immense columns of forged iron, 10 inches in diameter and 144 feet long, each therefore weighing *about 38,410 lbs. or 17·1 tons.* It is not only the mere supporting of this weight, but the weight of the wheel 4 feet 8 inches in diameter which it bears, together with—in case of the turbines being placed and driven as usual—the weight of the immense column of water 13½ feet in diameter and 145 feet in height above the wheel. Provision must be made, not only to support this weight, but to allow the shaft to revolve freely and steadily, transmitting 1100 horse-power at the speed of 260 revolutions per minute.

An inspection of the picture will show that each shaft is supported by a collar bearing under its bevel wheel (this bearing being somewhat like the thrust bearing of a propeller shaft), together with a footstep under the turbine. But Mr. Geyelin does not trust to any bearings for the support of a shaft like this when transmitting power.

His solution of the difficulty is simply to invert each turbine, the water therefore entering below each wheel, passes *up* through it, and at a gate opening of two-thirds of the full amount the upward pressure due to gravity acting on the water exactly balances the downward force of gravity on the shaft, whilst at full gate opening there is a preponderant *upward* pressure, and at any other opening a pressure which is only the *difference* of the two. This solution has proved as successful in practice as it is correct in theory.

Each shaft bears at its upper end a steel bevel wheel 4 feet 9¾ inches in pitch diameter, with a pitch of 5½ inches, gearing with a mortise bevel wheel 6 feet 3⅓ inches in diameter, the wheels running at the high pitch line velocity of 4000 feet per minute.

It will thus be seen that the horizontal shaft driven by each turbine revolves 200 times per minute.

The wheels themselves, which are of the Jonval type, are, with the regulating sluice or gate, shown in section in Fig. 114, where B is the revolving wheel, seen in section, and C the stationary or guide wheel.

The cut also shows the Geyelin patent gates. A A′ is a circular sleeve or gate, shown open and in section; E being a double hood, against which the gates close. At D is shown the opposite portion of the circular sleeve or gate, also open. Each sleeve, which weighs 2800 lbs., its weight being counterpoised, is operated by suitable levers and wire rope connections from the power house, to which the power is, in the first instance, transmitted.

Largest Turbine in the World.

The largest water turbine yet constructed is now being erected, by the I. P. Morris Co., at Shawinigan, on the St. Maurice River, 84 miles from Montreal. The wheel, which is 30 feet from base to top, is of the horizontal shaft inflow type, and is to give 10,500 horse-power. It is, however, only a little higher in power than the units now being installed for the Ontario Power Co. at Niagara, the hydraulic governor of which is described at p. 197. The shaft of the Shawinigan turbine is 32 feet 3½ inches long, 22 inches in diameter at the centre, tapering to 16 inches at the generator side and 10 inches at the other side. The "runner" or revolving part of the turbine is of bronze and weighs 5 tons, and the shaft weighs 10 tons. They are shown in the lower portion of Fig. 114*a*. When under full load the turbine, which is furnished with two draft-tubes (as shown in Fig. 114*a*), will take about 400,000 gallons of water per minute, the head being 140 feet. The water enters by the inlet shown, which is 10½ feet in diameter, flows round spirally and then enters through a set of 24 distributor vanes controlled by pistons, inside the cylinders seen on the top of the turbine casing, which are operated by hydraulic pressure and are under the control of the governor. These vanes are hollow castings of bronze, and have a small angular motion about steel pins, in accordance with Thomson's method already described. They are connected by links to two rings (one on either side), which move on ball-bearing seats, the rings being in turn connected by links to the pistons. The water is discharged at the centre, to right and left, into the two draft-tubes at the bends of which are stuffing-boxes for the shaft, the bearings for the latter being on separate pedestals not shown. The current from the generator is to be stepped up from 2200 volts 2 phase to 50,000 volts 3 phase, for transmission to Montreal.

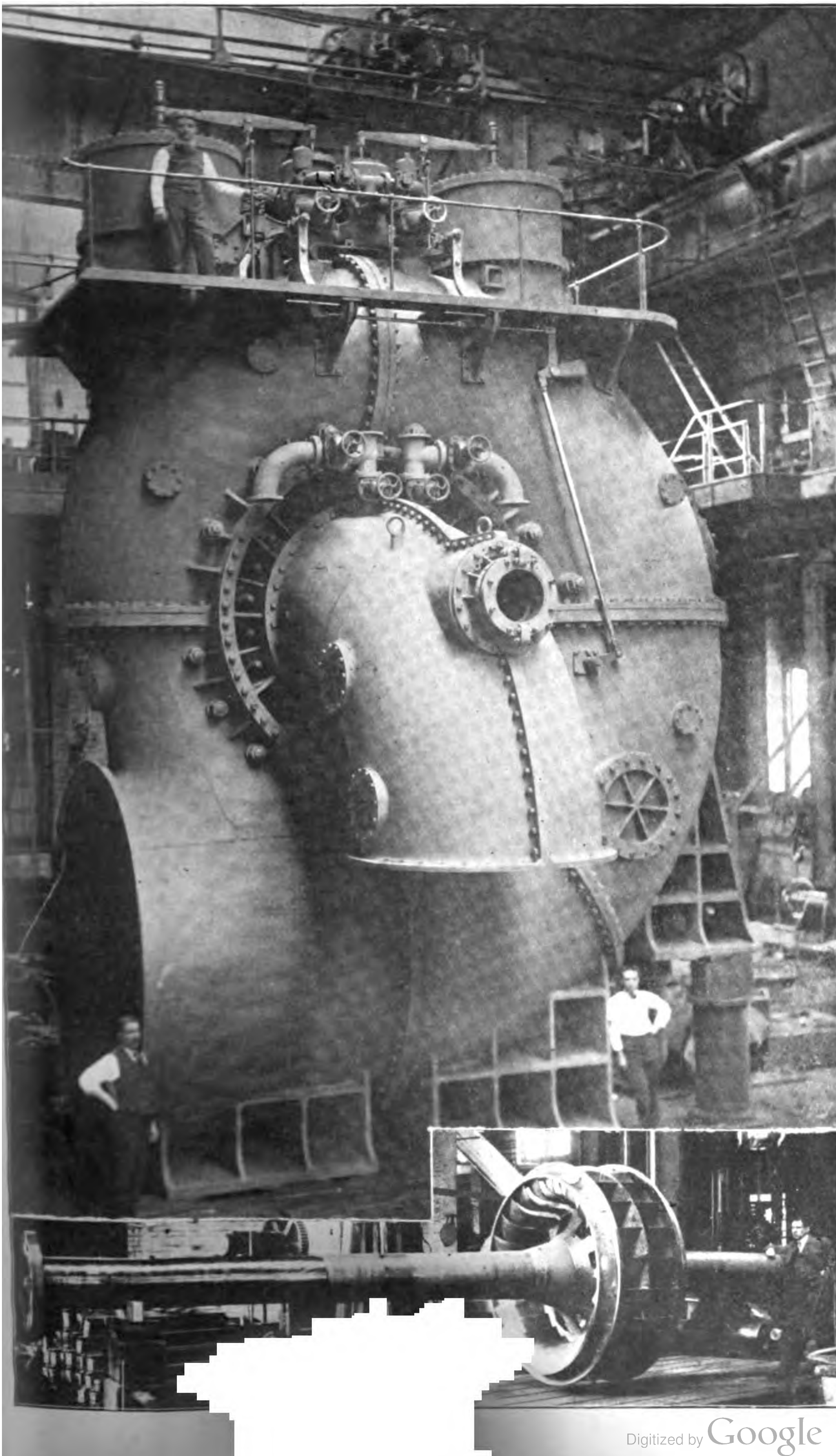
Turbines of Continental Makers.

One of the best known of the Continental firms devoting themselves largely to the construction of turbines is the Augsburg Maschinenfabrik. This firm has fitted up the turbines of many important water-power installations. The picture (Fig. 115) shows a recent form of Jonval turbine supplied by this company for driving the spinning and weaving machinery of the Kraehnholm manufactory, at Narva, near St. Petersburg.

The turbines are of 1250 effective horse-power each, five turbines having up to the present been fixed in these works. The first installation took place in 1867, when a turbine was supplied to take the place of an iron water-wheel which required frequent repairs. That turbine has been running since, without, it is said, any important repairs, and at present, owing to an increase of the fall, the wheel is giving out more than the power originally intended.

That turbine and those recently fixed, Fig. 115 being from one of them, are intended for a supply of 570 cubic feet (= 16 cubic metres) of water per second, with a fall of 25 feet (7600 mm.). With an efficiency of 77⅓ per cent., each would develop 1250 horse-power, but the maximum efficiency is higher than this. With the minimum efficiency —probably about 75 perc ent.—the effective horse-power is 1211. The turbines are, according to the Jonval system, arranged with suction pipe. A circular balanced sluice or gate serves for starting or stopping easily, as well as to some extent for speed regulation.

The sluice is placed at the discharge end of the suction pipe, the



flow being thus roughly regulated. The wheels are of 12·3 feet (= $3\frac{3}{4}$ metres) outside diameter, and make 50 revolutions per minute. A pivot or footstep bearing, of a very good design, is provided to resist the heavy water pressure. By means of bevel wheels the power is transmitted to the horizontal shaft, shown in left-hand figures, which makes 70 revolutions per minute, the further transmission being partly by gearing and partly by hemp ropes. The weight of each turbine complete with casing and all appurtenances as shown is about 140 tons. For closing the turbine chambers themselves there are provided at each inlet two shutters, which are driven either through spur-wheels by hand, or through water pressure engines with pistons.

This regulating arrangement is provided on the top of the guide wheel: by it half the guide passages can be closed in pairs, and provision is made for the free admission of air to them by pipes passing through the hinges of the flaps or shutters.

No complete test of the efficiency of this installation is available, but a similar installation by the same firm at the sewing-thread factory, Göggingen, has been most completely tested by Professor Schoter, the flow of water being measured by the Q = A V method described at page 73, the effective power being measured by a Prony brake.

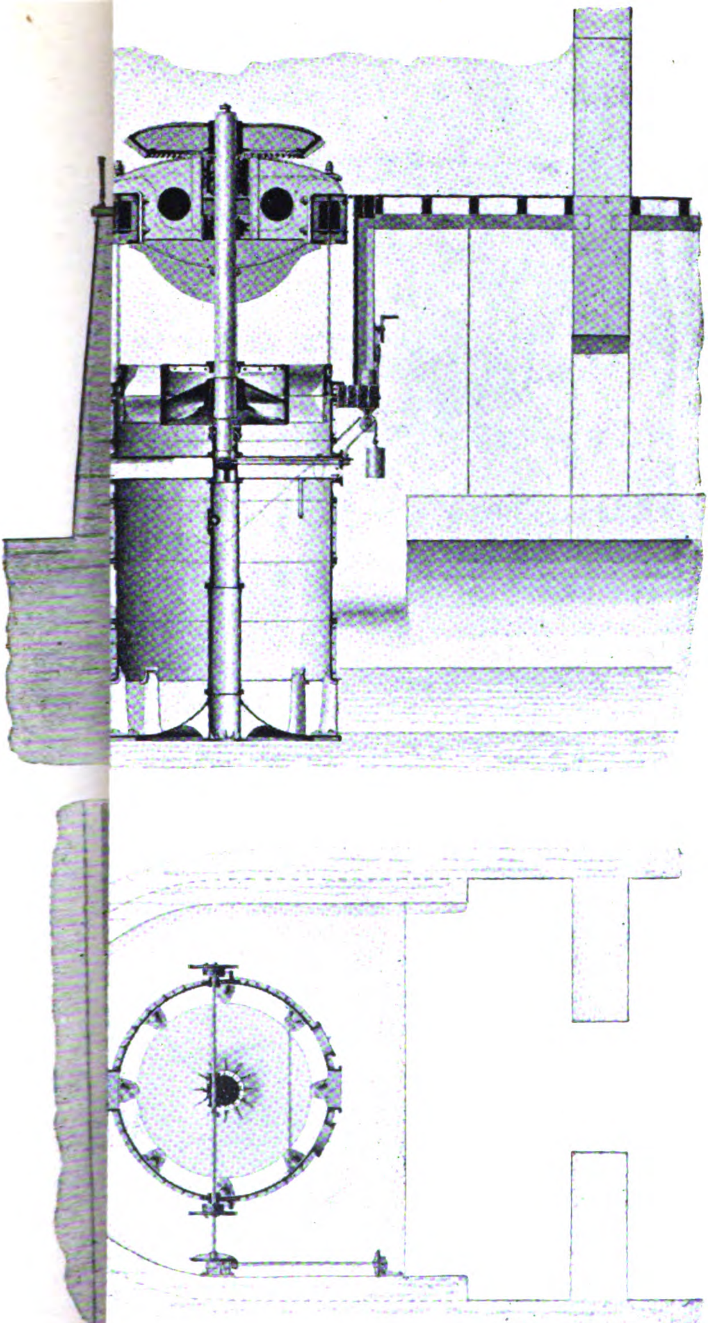
Professor Schoter's results show an efficiency of 82 per cent. for full gate opening, and with half the guide passages open an efficiency of 75 per cent.

It will be interesting for the reader to compare the American Jonval wheel with comparatively small diameter and high speed. with this large wheel of the same type going at a lower speed, constructed in accordance with what seems to be the Continental usage.

Many other very large installations might be described, but readers in this country may wish to have some details of smaller plant utilising a low fall such as is most frequently available. A good installation of this kind will now be referred to.

Newry Turbine Installation.

This is a good modern example of the utilisation of a low fall with turbines and overhead electric transmission for lighting and power purposes. The fall is about 5 feet 6 inches and an old mill-race would have been utilised, but it was, after consideration by Messrs. Moorhead the owners, and Mr. Ball the engineer, decided to erect the turbine generating station at the old weir formerly employed to divert water from the Newry river to a mill a quarter of a mile distant.



[*Face page* 176.

The turbine is a radial inward-flow or central-flow wheel of the type shown on page 145, supplied by Messrs. Escher, Wyss and Co., of Zurich. The variation of head due to frequent floods and the consequent rise of the tail-water owing to a somewhat restricted lower reach of river had to be considered, the result as to maximum power conditions being fully worked out on page 87. The average water supply is about 6200 cubic feet per minute, which, with an efficiency

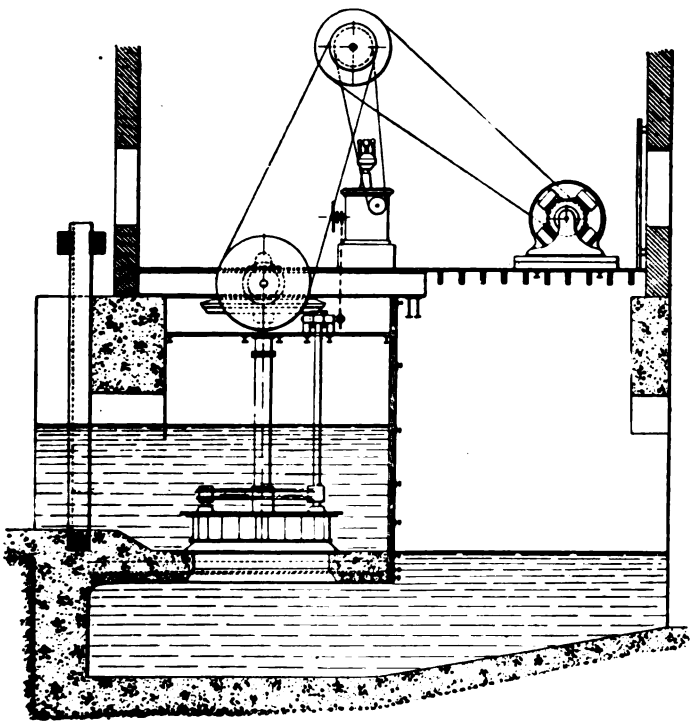


FIG. 116.

of 76 per cent., will develop 49 brake horse-power. The wheel is suspended from a ring-step bearing which is fastened to a cast-iron bridge carried by two steel joists placed across the wheel chamber at the floor level as shown in Fig. 116. The bearing-surfaces are immersed in oil which constantly circulates between the fixed and movable step-plates. These plates being of hard metal require little adjustment.

The horizontal shaft, driven by bevel gearing from the turbine revolves three times as fast as the turbine. From this shaft an overhead countershaft is driven at 300 revolutions per minute, which, in turn, drives the compound-wound 4-pole direct-current dynamo at 650 revolutions per minute, giving a current of 130 amperes at 230 volts, the guaranteed electrical efficiency at full-load being 90·2 per cent. and 89 per cent. at half load.

The arrangements of foundations, turbine pit, etc., will be gathered from the illustration, the foundation being carried down to solid rock at the tail-race end, the concrete employed being one part of Portland cement to two of sand and four of granite chips. The wheel is supported on a floor consisting of a grill of rails filled in and overlaid with concrete, and the water back is made of 4-inch tongued and grooved planks caulked with hemp and coated with tar, the hydrostatic thrust being taken by rails let into the concrete walls at both ends. The gate closing the entrance to the turbine pit is 12 feet wide, and it is raised or lowered by a worm and worm-wheel. Two 6 feet gates open into a by-wash to take the water when the turbines are not working, also to take the superfluous flood waters. The loss in transmission has been found to be only 7·2 per cent. at full load. This is an interesting installation, as it comprises many features which are of importance in this country where low falls are most plentiful, and comparatively small powers are often required.*

XVI.

SPEED REGULATION.

THE governing or speed regulation of water-wheels, including turbines, is effected either by hand or by a governor, which acts indirectly on the sluice or gate, the frictional and other resistances being considerable. The governor adopted by Fairbairn in the case of ordinary water-wheels is shown in outline in Fig. 117.

It consists of an ordinary Watt centrifugal governor, rotating about a vertical axis, and driven by the water-wheel through the shaft M and bevel wheels Q and P. M is hollow, and has inside it the shaft *a* which also carries the clutch Y Y, driven with it by a feather, but movable axially on it. Z is a bevel wheel loose on *a*, and gearing

* Figs. 116 and 132 inserted by courtesy of "Engineering."

with P. When clutch Y is moved to the right it engages Q, and shaft *a* rotates with Q. If, on the other hand, Y is moved to the left, Z and *a* rotate together. In this way shaft *a* and the worm and worm-wheel W and T may be driven in opposite directions. The clutch is actuated by the governor in the following way:—When the

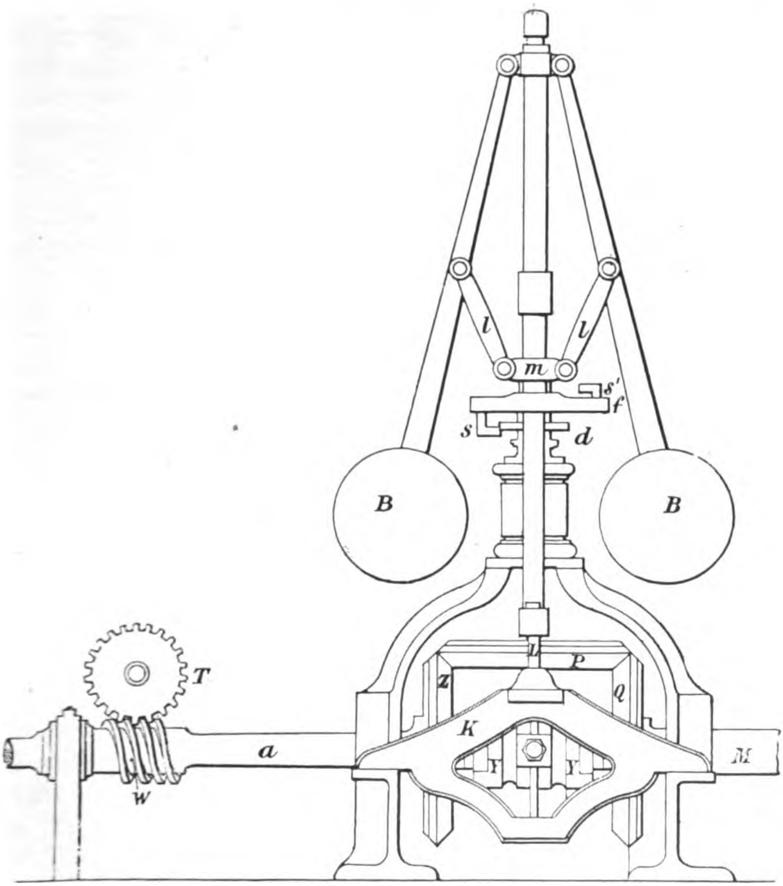


FIG. 117.

governor balls B B diverge, they raise the sleeve *m* through the links *l l*, and with it the cam *d*, which is on a brass slide attached to the sleeve *m*. The fork *f*, which carries two knee-irons *s* and *s'*, capable of coming into contact with cam *d*, is attached to the bent lever L, pivoted to framing K, this lever having a somewhat similar fork at its lower end capable of moving the clutch. The cam *d* being brought into contact with *s'*, the fork *f* is moved over by the rotation of the cam, and the clutch is put into contact with one of the bevel wheels When, on the other hand, the motor and governor go too slow, the balls converge, *d* is lowered into contact with *s*, the fork and clutch are moved in the opposite direction, and shaft *a* gets its motion from the other bevel wheel, hence rotates in the opposite sense. Thus, in the one case, the worm-wheel T, which is keyed to the same shaft as the pinion which moves the sluice, is rotated so as to close the gate; in the other case to open it as required. A form of governor often adopted for this purpose has the bevel wheels Q and Z placed loose on the governor spindle, with the clutch Y between them; this clutch being moved directly from the governor sleeve, determines the motion of a bevel wheel attached to shaft *a*. The defect of such an arrangement is that the force necessary to move the clutch and hold it in position is due to the centrifugal force of the *governor balls*, which is small, as the governor rotates slowly.

In Fairbairn's arrangement, just described, the force required to move the clutch and hold it where required is obtained through the cam from the *motor*, not from the governor balls, the latter being required only to set the cam and knee-irons in proper relative position, so that the water wheel can act on the clutch. Provision is made to return the clutch to its central or non-effective position as soon as the cam *d* goes out of contact with both *s* and *s'*, in which case *a* remains at rest. There is also a slotted link, not shown, which prevents the balls rising too high, so as to carry *d* over the upper edge of *s'*.

Vortex Turbines and Speed Regulation.

Professor James Thomson's turbine, to which reference has so frequently been made in the foregoing, is often called a "vortex" turbine, since the water flows in a converging path to the central discharge orifice.

Turbines of this class have one great advantage, viz. they are, to some extent, self-governing. Centrifugal force acts against the flow of the water, hence, if the velocity of the wheel increases above the normal amount, the velocity of the water is more or less checked, the wheel receives less water per second, and gives out less power.

Another important point about the Thomson vortex turbine is the method of having *adjustable* guide blades, which can be moved to suit a varying supply of water. These guide blades B B, as will be seen from Figs. 118 and 119, are pivoted near their points, and conduct the water into the revolving wheel A. D D are shafts with bell cranks, which move the guide blades B B so as to make a different

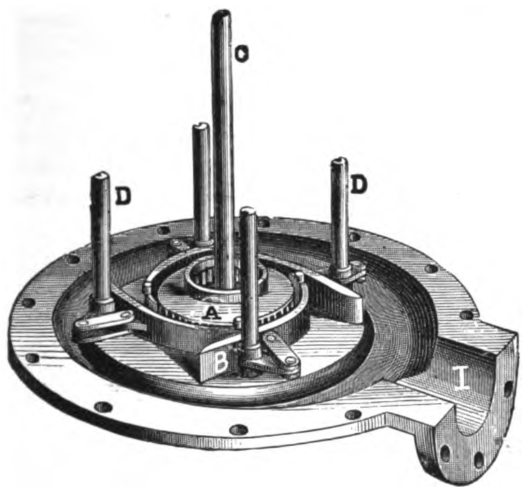


FIG. 118.

angle with the wheel. E E (Fig. 119) are the outside coupling-rods connecting D D together, so that they may be moved simultaneously

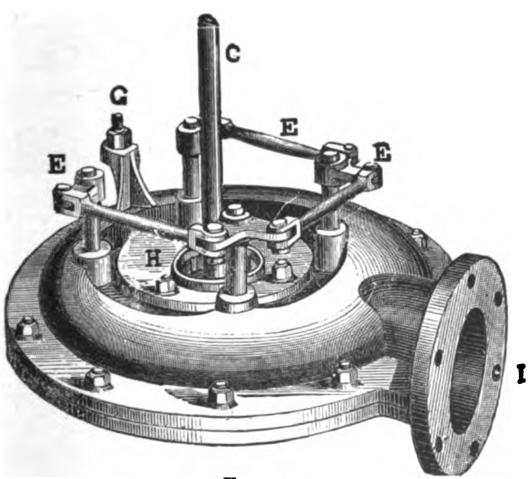


FIG. 119.

either by hand or governor. It will be seen that a very small amount of turning of the shafts D D will considerably alter the angle of the blades B B. If the wheel runs with light load, so that v_2 is greater than usual, centrifugal force partly stops the flow, the radial velocity is less than that for which θ (see p. 146) is calculated; hence the angle θ is varied to suit the new conditions of load, without the efficiency of the wheel being much affected.

This turbine can also be placed with its axis horizontal, as shown in Fig. 120, where I is the inlet, and K K the suction pipes, which, since the height of the wheel above the water in the tail-race is less than 32 feet, are full of water during the time the wheel is at work.

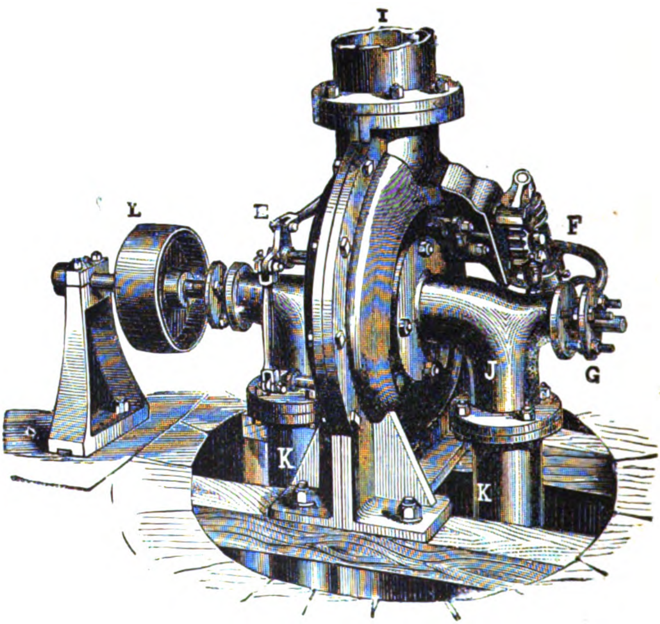


FIG. 120.

F is the hand gear for altering the guide blades, and L the driving pulley, which, being thus conveniently placed with its axis horizontal, is easily connected to any machine to be driven.

Axial-flow turbines have not the same facility of governing as the "vortex" wheel, and being usually placed with their axes vertical, there is a certain end thrust on the shaft. This thrust may, however, in some cases be made of great service in counteracting the dead weight of the shaft.

MURRAY'S GOVERNOR.

The governor of Murray, shown in Fig. 121, designed for use with turbines, is a good example of a relay governor. Figs. 1 and 2 are sections at right angles to one another of a four-way valve, actuated by the governor I (Fig. 3). The outer casing A has a supply port B in it, and an escape or exhaust port C, also passages D and E leading respectively to the top and bottom of a regulating cylinder, from which the supply of water to the wheel is controlled. Within the casing A is a sleeve F, with ports communicating (as

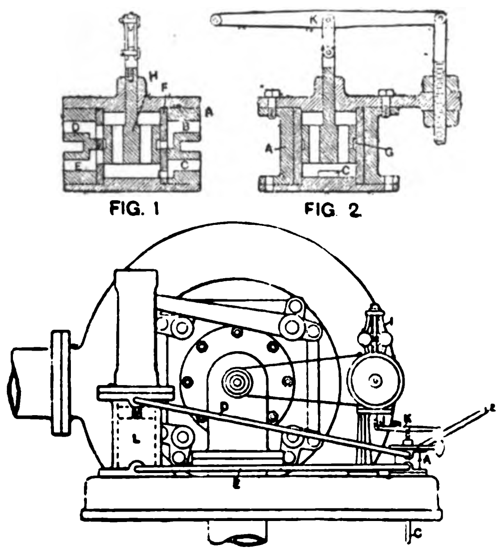


FIG. 121.

shown in Fig. 1) with an annular passage G (Fig. 2) in the piston H, which works within F. The breadth of this passage G is a little less than the distance between the inside edges of the ports D and E, so that when the piston H is in the central position the ports D and E are closed to supply but opened slightly to exhaust.

Referring to Fig. 3, I is the centrifugal governor, driven from the water-wheel or turbine, and connected by a rod J to a lever K, which acts on the piston H through a short link. When the speed of th motor increases the governor balls diverge, H is raised, and passes by the passage D to the upper part of the regulating cylinder L (Fig. 3), acting on a piston M therein in such a way as to diminish the supply of water to the motor, M being connected to the sluice. If the speed decreases below normal, the opposite action takes place, water entering by E, the other end of the cylinder L, raising M and admitting more water. It will thus be seen that the governor, like all good modern governors which have considerable work to do, acts through a fluid *relay*, the centrifugal force of the governor being utilised merely to control *the relay*, and not to do the work of setting a heavy valve which moves under considerable pressure, and probably with a good deal of friction.*

The Governing of Turbines.

In all cases where the governing of water-actuated motors is effected by diminishing the supply, it may be laid down as a fundamental principle that it is better to close some inlet apertures altogether than to partially close all. There are many devices for effecting one or other of these objects in the case of turbines. In one arrangement little paddles fixed to vertical rods move over the inlet guide apertures. There are rollers at the upper ends of these rods, which rollers rest on an inclined ring or spiral surface. When this ring is rotated by hand, or the governor, the paddles are moved so as to open or close the apertures, the best arrangement being that in which only alternate openings are thus covered. The total cutting-off of supply is effected by a separate sluice. Then there is another arrangement in which annular strips of leather are used, the ends being fixed, two to the guide apparatus and two to conical rollers, which can rotate about their geometrical axes (slightly inclined to the horizontal), also about the axis of the turbine (vertical), so as to wind on or let off the strips. A vertical shaft carries a pinion, which engages a toothed sector whose function is to effect the motions of the rollers, and thus close the guide apertures. Various types of sliding sluice are used. Thus in one form used with an axial-flow turbine, the guide passages form two semicircular sets, two sliding discs close an equal number of apertures on each side of the centre, the discs being moved by a rack and worm.

A sluice at the bottom of the suction tube is also used in some cases to regulate the speed; a very good example of a ring sluice is seen in the section of Mr. Geyelin's turbine; and the method,

* The author is indebted to Messrs. Gilbert Gilkes & Co., of Kendal, for illustrations of the Thomson turbine and Murray governor.

adopted in the turbines of the Niagara Power Company, of building the turbine, so to speak, in several stories, and cutting off the supply completely from one or more of these, is good for preserving a high efficiency with partial output.

Snow Governor for Turbines.

A form of governor used with the "Hercules" turbine is shown in Fig. 122.

It is a centrifugal governor with toothed sectors attached to the upper ends of the ball arms, so that as the balls rise or fall the central spindle falls or rises.

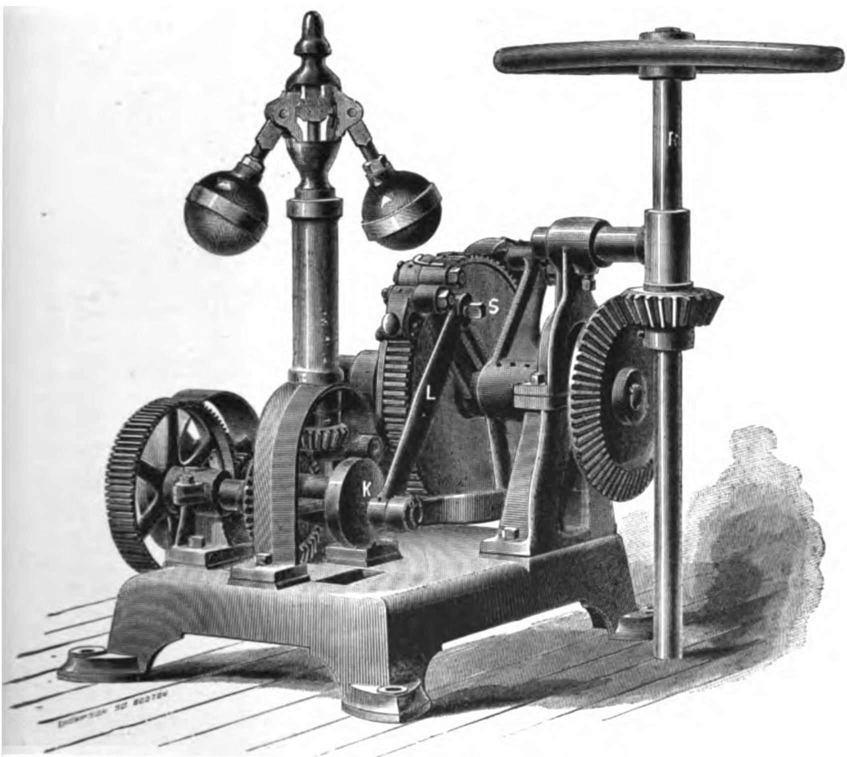


Fig. 122.

There is a connection between this spindle and a sector controlling two pawls which engage the teeth of the large central spur wheel S. By means of a disc crank K, seen to the right of the governor bevel gear, a reciprocating motion is given to the inclined link L, at the upper end of which are two pawls engaging the teeth of S. One of these pawls closes the sluice by moving S in one direction, the other opens it by moving S in the opposite direction. The sector, or pawl shifter, part of the edge of which is seen to the left of S and under the nearest pawl, would prevent either pawl from acting but for a depression in it, as indicated in Fig. 123.

If the governor balls rise, the sector is rotated like the hands of a watch, allowing the "closing" pawl to operate, and lifting the other out of gear. If, on the other hand, the governor goes at less than the normal speed, the sector is moved in the reverse direction by the upward movement of the governor spindle, the opening pawl gears, and the closing pawl is lifted. Thus the bevel wheel on the same shaft as S turns R, the sluice shaft, in one direction or the other, as long

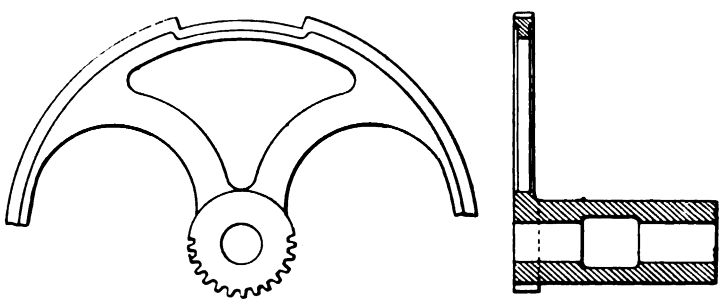


FIG. 123.

as either pawl is in gear, each revolution of the crank K giving a short motion to S through the pawl in gear with it. When the speed is normal the sector lifts both pawls, and the governor ceases to operate the gate. An arrangement is provided for lifting the pawls when the gate is fully open, but the turbine going at less than normal speed, owing to low water supply or excessive load.

It should be mentioned that the gate can be operated by hand. The largest water-power installation at present in Britain (viz. 1200 horse-power) is that near Aberdeen, where "Hercules" turbines are used.

KING'S GOVERNOR.

A neat arrangment of pawl governor is due to Mr. King, of Newmarket, Gloucestershire. Fig. 124, which shows the ratchet arrangement best, is a float governor designed to keep the water in the head-race at a constant height, so that the wheel may work under a constant head. Referring to the figure, the toothed wheel replaces, or is used in conjunction with, the hand-wheel usually employed to work the sluice or gate.

The teeth of the wheel are "masked" for a portion of the circumference by a cam 4, on the boss of which is a pinion gearing with a sector (shown dotted), which in its turn is connected to a float 5 in a

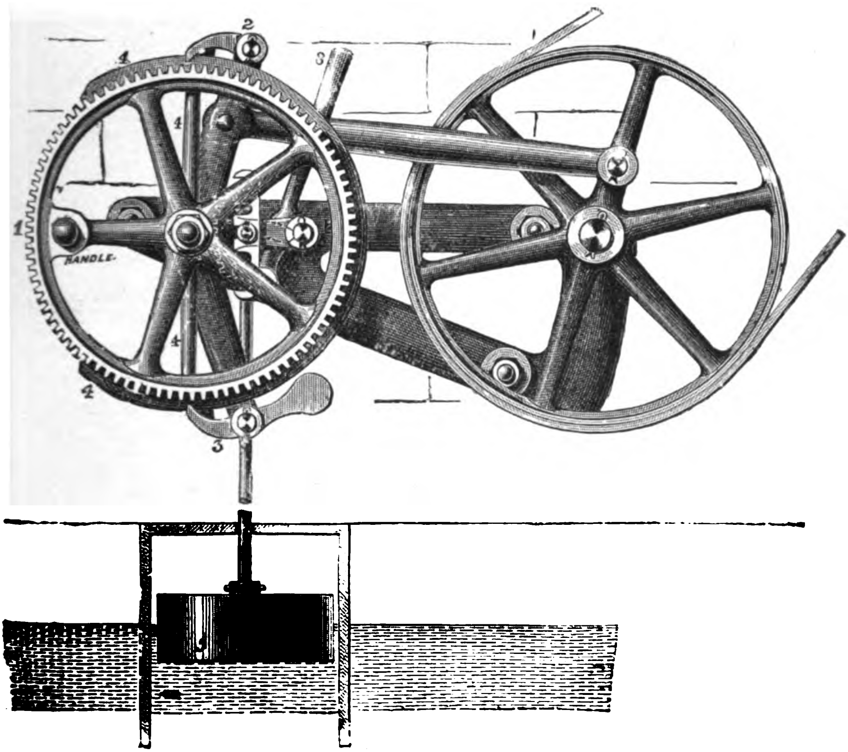


FIG. 124.

tank communicating with the head water. The pawls 2 and 3, and the bent lever carrying them, are worked by a connecting rod from a pin in one of the arms of the belt pulley, driven by the water-wheel or turbine. In the illustration the float is stationary, and the water at its proper maximum height, the pawls being out of gear. But if the level of the water lowers, the float falls, turning the masking ca[illegible] to one side, leaving the pawl 2 free to act on the teeth of th[illegible] the sluice being thus slowly closed. If the water rises, the float unmasks the lower side of the wheel 1, and the gate is opened by pawl 3. If a number of water-wheels work from one source, this

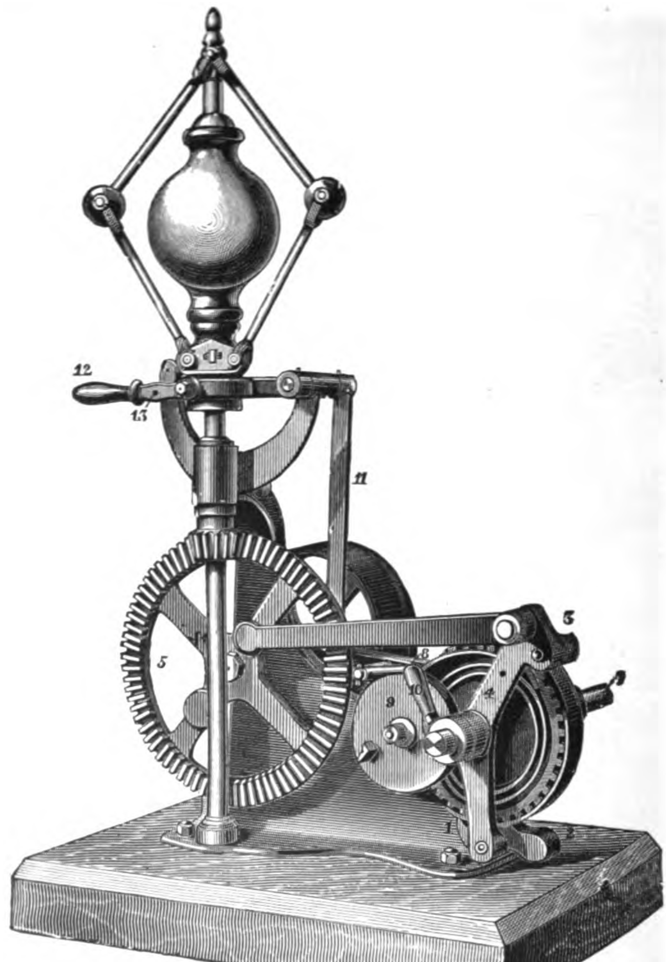


FIG. 125.

governor attached to one of them will govern all, by keeping a constant head. The float may be replaced by a centrifugal governor, as shown in Fig. 125. The gate is worked by the pawls and masking cam, as before, the latter being connected to the governor instead of to the float, increase of speed causing one pawl to act, and decrease of speed the other. Sometimes a turbine drives shafting in conjunction with a steam engine, in which case Mr. King's clutch, which drives only in one direction, is useful; for when the turbine lags behind through deficient water supply, the clutch does not act and the turbine furnishes no power, but is not a drag on the engine.

When the turbine again gets up speed to the engine standard, the clutch engages, and the water-motor takes its share of the work.

Hett's Governor.

Mr. Hett's governor, shown in Fig. 126, consists of a loaded governor of the Porter type, which actuates a belt shifter or fork so as to move the crossed belt to different positions on the two tapered driving cones shown. When the governor sleeve is in mid position, the belt is on the middle of the two cones, which therefore rotate at the same speed. An increase of speed causes the governor to move the belt to the right, and thus drive the upper cone at a lower speed relative to the governor. But the lower cone drives the governor spindle through the bevel wheels shown; the upper cone drives a sleeve which is loose on that spindle. The sleeve carries the upper bevel wheel, and the spindle the lower bevel wheel, of the central nest of differential gearing. When the sleeve and spindle revolve at the same speed—in other words, when the belt is on the centre of the cones—the differential gearing which connects spindle and sleeve is inoperative, the vertical bevel wheels acting merely as idle wheels, rotating, but not changing the direction of their axes. If, however, spindle and sleeve rotate at different speeds, the vertical bevel wheels rotate not merely about their axes, but run round the other wheels,

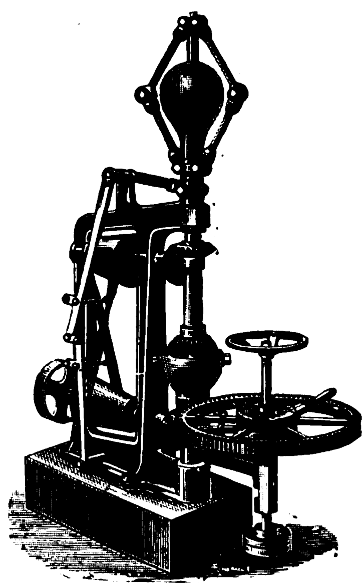


Fig. 126.

their axes rotating, and it is this differential motion which is transmitted to the gate by the pinion and spur-wheel shown. In all these cases the gate is also fitted with a hand-wheel, by which it can be opened and closed by hand.

"Hunting" in Governors.

Unless some special feature be introduced in the design of a governor—especially a comparatively slow-moving one—there is great risk of that peculiar and unstable condition as regards speed called "hunting" being experienced. For instance, the speed, we will say, is normal, and the revolving masses occupy their mean position; now let a change of load occur, say the load is reduced, the governor balls move outwards, the mechanism proceeds to close the gate. This movement continues as long as the speed is above the normal, but the gate has now been partly closed, and the turbine, no longer receiving so great a supply, drops in speed; this causes a collapse of the balls, followed by the action of the gear to open the gate, the cycle continuing in this way, each change of gate being a little too late, and always "hunting" or trying to follow the oscillatory speed movements of the governor.

To obviate this many devices have been adopted; but in most of the successful ones there is a portion of the mechanism whose function is, to be always trying to return the valve controlling the mechanism acting on gate, to its mean position. This is further referred to in connection with the Lombard Governor (page 195). A governor of a very quick-moving type, such as that described on the next pages, is not so likely to suffer from this defect, and if it does, probably the introduction of a dash-pot will be a sufficient remedy, whereas with a slow-moving governor a dash-pot will probably accentuate the evil.

Mechanical Governor of Niagara Turbines.

To govern turbines of 5000 horse-power so that a change in power given out, from 4000 to 2700 horse-power, shall be accompanied by a change of speed of only 2·8 per cent., and a change from 5000 to 1500 horse-power by only 5·2 per cent., with changes of speed produced by ordinary working changes of output of less than 1 per cent., may be regarded as the most satisfactory result yet obtained by mechanical water-wheel governors. These results have been obtained in experiments with Messrs. Piccard and Pictet's governor at Niagara. Throttling the tremendous flow of water moving under a great head necessitates a mechanism of no ordinary power, whilst the result shows sensitiveness equal to that of a small steam engine governor.

Of course, in a governor like this the centrifugal portion cannot do the work of moving the heavy gates against the force of the flowing

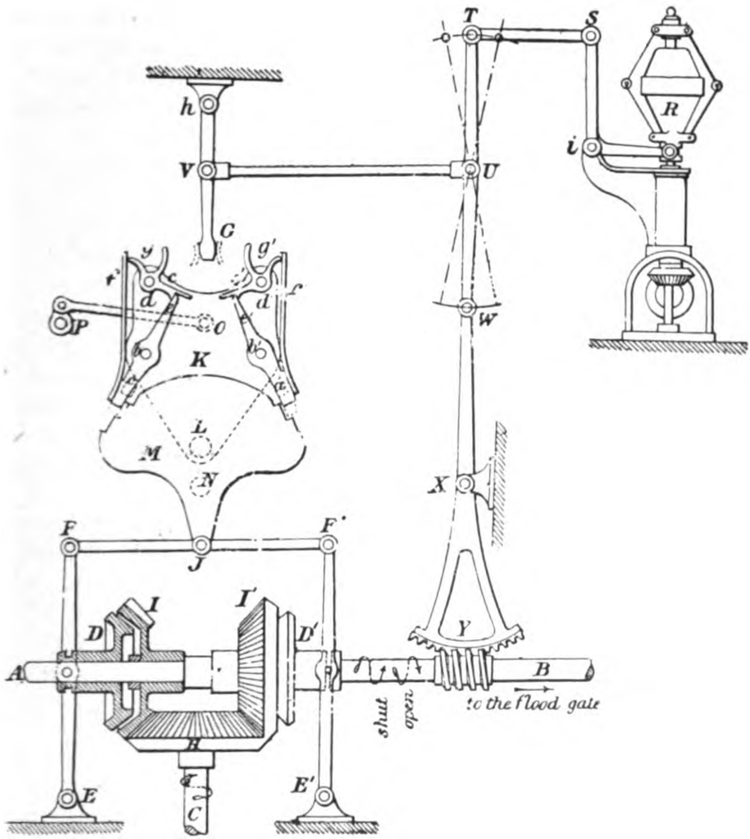


FIG. 127.

water, the motor itself must move the gate. The governor of Messrs. Piccard and Pictet is a pawl and clutch governor of a most ingenious kind. The principle on which it acts will be best understood from the elementary drawing shown in Fig. 127.*

The sluice or gate, which is a plain cylindric rim, is moved by the

* Through the courtesy of the designers, Messrs. Piccard and Pictet, of Geneva.

shaft A B which is turned in either direction by the bevel wheels I I′ (driven through C by the turbine), according as either is engaged by the friction cone D or its fellow D′. The bevel wheels are loose on the shaft A B; the friction cones turning the shaft by feathers, but being capable of moving along the shaft.

D and D′ are actuated by the links E F, E′ F′ respectively, which in turn are moved by their connection at J with the sector M pivoted at N. Above this sector is a piece K which is oscillated by the crank P turned by the turbine, so that K, and its latches *g c d*, *g′ c′ d′*, with corresponding indent pieces or pawls *a e*, *a′ e′*, are always kept oscillating about centre L as long as the turbine revolves.

As shown in the figure, the latches *c c′* are engaging the pawls so that the latter are free of the sector M, and the cones D D′ are out of gear.

When, say, an increase of speed takes place through part of the load on the turbine being thrown off, the centrifugal governor R moves G to the left as will be readily seen, and the latch *c* is lifted, the piece *a* falling into gear with the teeth on the left side of M, through the action of the spring *f*. The oscillation of K now moves sector M in the direction opposite to that of the hands of a watch; the point J is moved to the right and the cone D is thrown into gear, causing the shaft A B to revolve in the same sense as the bevel wheel I, thus partially closing the gate.

As soon as the gate has sufficiently closed to lower the speed of the turbine, R falls, G is moved to the right, D is released, and, if this movement continues sufficiently long, D′ is thus thrown into gear, and A B turned in the opposite sense.

It is worthy of note that when *a*, for example, is in contact with a tooth of M, its extremity moves with M about N as centre, whilst its own centre of oscillation is, like the whole of K, at L. This eccentricity of movement has the effect of lifting the other end *e* of the pawl into a position to be engaged by *c*, should the latter be freed by a releasing motion of G.

An important feature to be noticed is, that the pin W, which may be regarded as the fulcrum of the lever WUT, is *not* a fixed point but is moved by the action of the worm Y in such a way as to tend to prevent that lag of the motion of the gate behind that of the motor which is noticeable in most pawl governors. In the case of an ordinary governor of this kind, if the motor increases in speed beyond the limit necessary to overcome the friction of the governing gear and sufficiently to move its change mechanism, the pawl is thrown into gear, which closes the gate; the action is continued till the gate is partially closed, but now the motor is no longer receiving sufficient water, and its speed has begun to fall, which in due time causes the other pawl to act, opening the gate. Thus the action of the governor on the gate, taking some time, is always a little too late, resulting in a periodic fluctuation of speed which may render the apparatus useless.

To show that the arrangement here adopted prevents this, we may point out that in order to bring the apparatus to rest, it is only necessary that the motion of the point W be proportional, and in the opposite sense, to that of T. The motions of W are like those of the gate (increased or reduced), and the motions of T follow those of the tachometer R: hence by this arrangement the peculiar state of instability as regards speed, known as "hunting," is prevented.

Promptitude is secured by the fact that the crank P makes 200 revolutions per minute, giving more than six blows of the pawl per second, thus the regulating mechanism has six chances of acting every second, one way or the other.

The driving force required is taken from the motor, and the tachometer has only to exert a very small force, the gate of the Niagara turbines being closed completely or completely opened in twelve seconds with a force of *about* 5 *tons*, thus enabling it to easily cut through débris which may be in the way.

The governor as applied to the turbines of the Niagara Cataract Construction Company is shown in section in Fig. 128.* It is identical with that described, except that the friction cones are replaced by brake wheels and gearing.

Though this is a most ingenious piece of mechanism, it is not now used, having been replaced by a hydraulic governor. One of these latter, applied to a newer turbine of 5000 brake horse-power, showed a speed variation of only 3·8 per cent. when *the whole load* was switched off.

HYDRAULIC GOVERNORS.

Hydraulic governors for turbines usually consist of a pendulum governor of ordinary centrifugal type, a steering valve, and a servo motor, or in some cases a hydraulic piston connected mechanically to the gate of the turbine. The least change of speed moves the steering valve, admitting water to or exhausting it from the piston of the servo motor, the movement of which is communicated to the gate. The movement of the piston returns the valve to its normal position,

* Figs. 128, 262–264 and 272 are inserted by the courtesy of the proprietors and editors of 'Cassier's Magazine.'

cutting off the working pressure fluid and stopping all further movement of the piston until a fresh change of speed takes place. If the

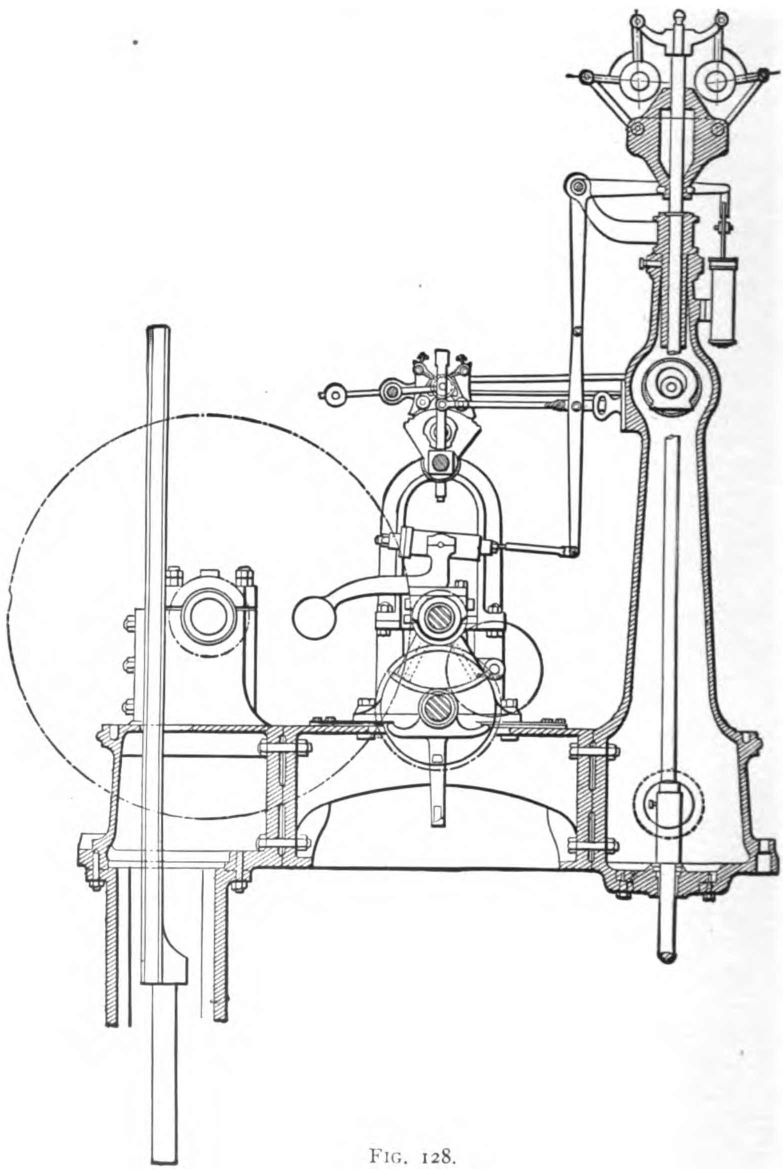


FIG. 128.

fall is 40 feet or more the ordinary pressure of the water may be enough to work the governor, but if not, a pump is used to produce the necessary hydraulic pressure, and in this case oil is often used instead of water.

Hydraulic Governor of Messrs. Escher Wyss & Co.

A good governor of this type is that of Escher Wyss & Co. It consists of a centrifugal governor of the inverted form. To a point on the governor stem which moves up or down with the motion of the governor balls, the end A of a horizontal link A B is attached, the end B being borne on a tappet resting on an inclined plane attached to the rod of the main piston of the governor. This rod actuates the gate through a crank. A point on A B near A gives motion to the controlling or steering valve, which admits fluid under pressure to one side or the other of the main piston. If the movement of the centrifugal governor is such as to raise A, fluid is admitted to the face of the piston which moves forward, the tappet moving down the inclined plane lowers B, thus returning the valve to its mean position. If the fluid is admitted to the annular area of the piston the opposite action takes place, the tappet moving *up* the incline, thus returning the valve to the closed position until a further change of speed takes place. When the head is great the closing of the gate is accompanied by the opening of a valve allowing water to pass to the tail-race without acting on the wheel.

Lombard Hydraulic Governor.

One of the most successful of the type referred to is the Lombard governor (made in Boston, Mass.), which has been installed in plants aggregating over half a million horse-power. The D type of this governor is illustrated in Fig. 129. It consists of a centrifugal governor of the Pickering type, which operates a valve admitting or exhausting water or oil under pressure. It will be seen that under the bed of the governor is a cylindric tank which is divided into two parts by a partition indicated by the row of rivet-heads.

The larger or left-hand portion of the tank is about half full of oil, the upper part of this portion containing air under a pressure of about 200 lbs. per square inch. The smaller or right-hand end is exhausted so as to have a fairly good vacuum in it. The pressure and vacuum are maintained by a pump, driven from the large pulley shown on the right-hand side of the governor. The gates are moved by the movement of a piston which is acted on at one side by the oil from the pressure tank, whilst the other side of the piston communicates with the vacuum tank, the oil on this side, when it runs into the vacuum tank, is immediately pumped into the pressure tank. Increase of speed causes the governor balls to diverge, as in the case of a steam-engine governor, the top plate to which the flat springs of the governors are attached being depressed, and depressing a rod which runs down the centre of the apparatus between the standards, seen to the right in the illustration. This rod is attached to a small vertical piston valve. At normal speed the balls occupy such a position that this valve is closed, but the moment the speed increases above or decreases below the normal amount the valve moves down or up from its closed position, admitting oil or water, as the case may be, to one side or other of the regulating valve controlling the main piston of the governor. This piston, by a rack and gearing, moves the regulating gate of the turbine. There is a very interesting "floating gear" designed to prevent hunting, which would take place if the working valve and mechanism acted slowly or too long in one direction.

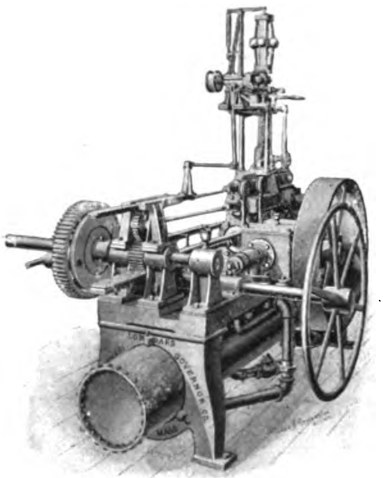


FIG. 129.

The centre cylinder contains the large balanced valve which admits the working fluid. This valve is under the control of the small upper cylinder as well as the working cylinder. The relation of the floating gear to the rack above and the pinion below is such that a movement of the upper rack outwards will carry the floating gear and consequently the valve stem in the same direction, which, through the displacement of the balanced valve referred to from its mid position, will immediately cause the main rack to move out. The outward movement of this rack, rotating the pinion, will move the floating gear backward until the main rack has moved out exactly as far as the upper rack, when the valve will once more have assumed its mid position. Thus every movement of the upper rack is duplicated immediately by the lower one, the movements being so nearly simultaneous that an observer can hardly tell which moves first. This rapid return of the working valve to its mean position prevents hunting. The small cylinder is controlled by the piston valve moved by the governor balls.

The rod connecting the balls to the valve is in two pieces connected by a screwed sleeve, which can be rotated by hand so as to alter the normal speed of the governor. A dash-pot is also provided and forms part of the regulating gear, the whole action of the anti-hunting mechanism being somewhat difficult to explain without several drawings, but it acts very effectively, and it is said that it can be made so "dead-beat" that it will make but one stroke to set the gates correctly for almost any change of load. If there is a tendency for the governor to carry the gates beyond the proper position, and then oscillate several times after a change of load, it is a sign that the dash-pot is not quite sluggish enough in its action, and the remedy is to slightly close a valve in the dash-pot. If the governor does not move the gates quite far enough, the fore-named valve should be opened a little.

The governor is certainly very interesting as a piece of mechanism and very effective in its operations.

Relay-Valve Governor to Control Ten Thousand Horse-Power.

In the very large power units now being installed at Niagara, special governors are to be employed. The relay-valve type of the Lombard hydraulic governor, to be used in connection with the Ontario Power Company's generators, and to control a 10,000 horse-power unit, is shown in Fig. 130. The main cylinders are 14 inches in diameter and the stroke is 2 feet. They operate with oil or water at a pressure of 200 lbs. per square inch. The relay-valve mechanism, which is very ingenious, will be understood by a reference to Fig. 130A, where A is the main cylinder, in which works the piston *b* with its rod *a*. Pressure fluid is admitted to, or exhausted from, this cylinder by the ports *q* and *r*, which, by the main relay-valve B, are put in communication respectively with the inlet *u* or the outlets *t* and *v*, one port communicating with inlet when the other communicates with outlet. The main valve B is balanced, and consists of two pistons with their tail-rods *s* and *x*, the left-hand one *x*, being larger than *s*. *s* is acted upon by a *steady* hydraulic pressure supplied through the passage O, whilst *x* is acted on by a *variable* pressure furnished through the inlet *g*, the pipes *h* and *f*, and the valve K, which is controlled by the centrifugal governor. The pipe *f* communicates with the compensating cylinder D, in which works the piston *e*, the rod of which is connected to the main piston-rod *a* by the yoke Y. The capacity

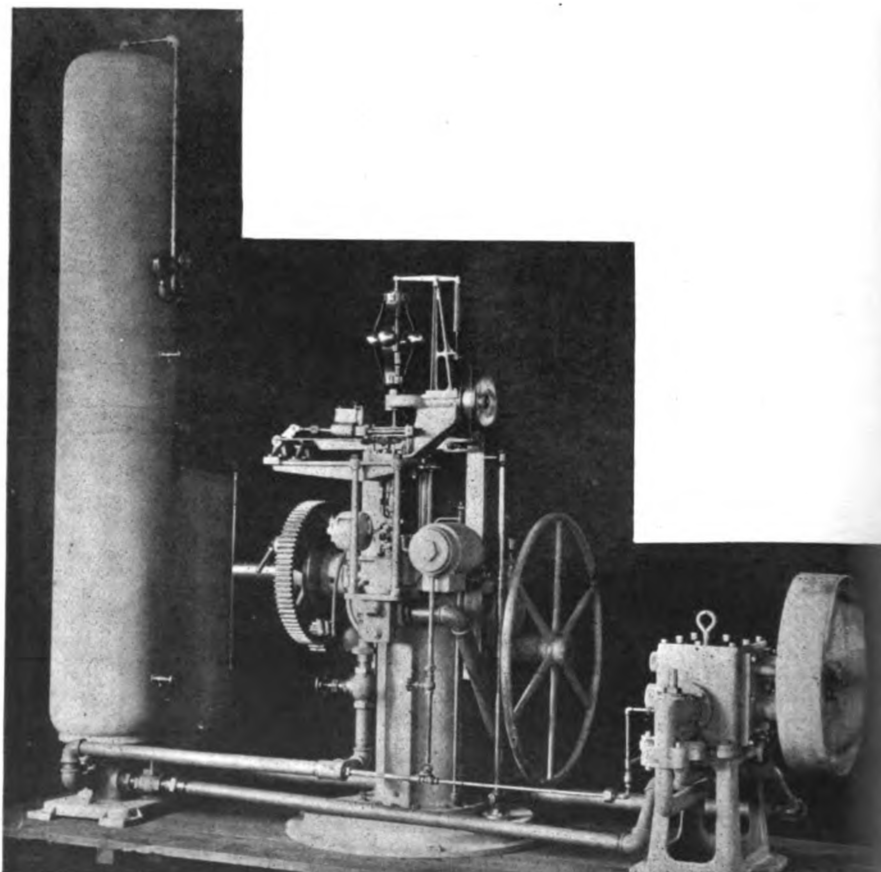


FIG. 130.

of the compensating cylinder to the right of *e* varies, therefore, with the position of *e*.

The pressure supply is admitted to the system by the three-way balanced piston-valve K, working in a cylinder which has ports *n*, *l*, and *m*. Valve K has a little lap over the two outer ports *m* and *n*, but the reduced stem of K is always in communication through the annular space round it, with port *l*.

Fluid under pressure is admitted through *m* (usually from the same source which supplies the valve chamber w_1) and is exhausted through *n*. The action of the mechanism is as follows:—When the

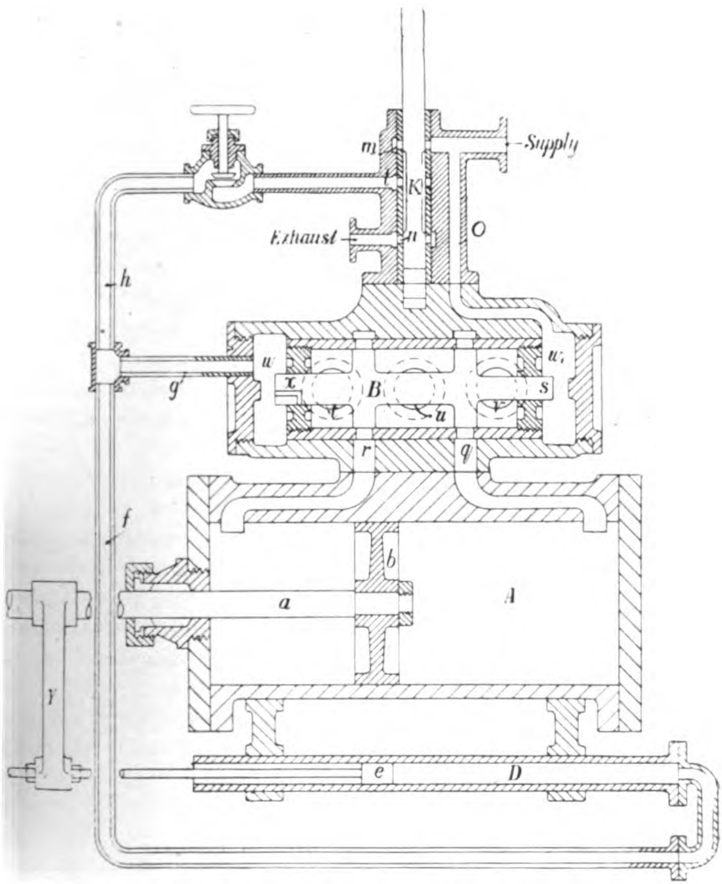


FIG. 130A.

valve K moves up, the fluid is admitted by *m* to the space round K, thence by port *l* and pipe *h* to chamber *w*. The fluid in *w* is therefore at about the same pressure as that in w_1, but owing to the greater area of *x* over *s* the main valve B moves to the right, thereby opening port *r* to the main exhaust *t*, and port *q* to pressure supply *u*, thus admitting pressure supply to the right-hand end of the main cylinder A, so that *b* moves to the left. This movement is accompanied by the simultaneous movement of *e*, causing an increase in the capacity of the compensating cylinder D, and hence the pressure in *w* falls. Thus very quickly a balance of forces is reached, by which the valve B is, for an instant, held open a certain definite amount, the closing of valve K allowing *e* to catch up on it as it were, so that the room found in D is just sufficient to allow a balance of forces. During this operation the valve B travels to the left (under the action of the constant pressure in w_1 acting on *s*), thus closing ports *q* and *r*, about in the same ratio as that in which the valve K closes the inlet port *m*. Thus the valve K and the main valve B come to rest together. All the passages are now full of fluid which cannot escape, and there is no motion until K is again opened. A *downward* motion of K causes the opposite series of operations to take place, but the important thing to notice is that by the action of the compensating cylinder D the main valve B, which was moved to the right by the upward motion of the pilot-valve K, is afterwards returned towards the left; thus the movement of the main piston *b* is restricted to such dimensions as shall re-set up the condition of equilibrium in the system, and thus hunting is obviated.

The pilot-valve K need only be small. It is found that to supply a main valve B, sufficient to operate a main piston of 16 inches diameter and 2 feet stroke, with working fluid at 200 lbs. per square inch, K need not be more than three-eighths of an inch in diameter, with a travel of only *one-eighth of an inch*. It should be stated that a spring may be used to urge B in one direction, and there are various other ways in which the inventors secure the desired motion, but that described here is the simplest used for high powers.

Governors of this type have been constructed to exert a force of over 50,000 lbs., and to perform a complete stroke in about one second.

Other Hydraulic Governors.

In the Newry installation (page 177), two rotary oil pumps are driven from the countershaft by a belt. The circulation of oil within a closed chamber is regulated through a steering valve by the position of the revolving masses in a pendulum governor, by which one or other of the pumps is brought into operation according as the turbine tends to go above or below normal speed. The action is communicated through a worm and sector to the guide vanes of the turbine itself. These vanes are balanced, and adjustment to suit the new conditions of load is quickly effected.

In some cases part of the same oil supply under pressure which actuates the speed controlling apparatus is used in the footstep bearing which supports the turbine, and thus relieves the pressure of the bearing surfaces.

Differential Governor for Pelton Wheels.

The Pelton wheel is a very efficient motor, but it has the defect of being sensitive to changes of load. When these wheels were used to drive an electric power or light installation, this defect gave rise, in the earlier history of the development of the motor, to some difficulty. Often the regulation was effected by a man who sat with his hand on the lever of the supply jet, at the same time keeping his eye on the speed indicator or voltmeter. The differential governor shown in Fig. 131 is a successful apparatus designed for this purpose. Two 18-inch pulleys revolve loosely, in opposite directions, upon a shaft, one being driven by the motor to be "governed" and the other by a separate wheel working against a constant resistance. The two pulleys have bevel wheels attached to them, which gear into two other bevel wheels on a shaft which is at right angles to that on which the pulleys revolve, and forms one piece with it. To the left of the pulleys will be seen a pinion loose on the latter shaft, and with ratchet teeth cut in opposite directions on either side of its boss or hub, with corresponding circular ratchets gearing with the teeth.

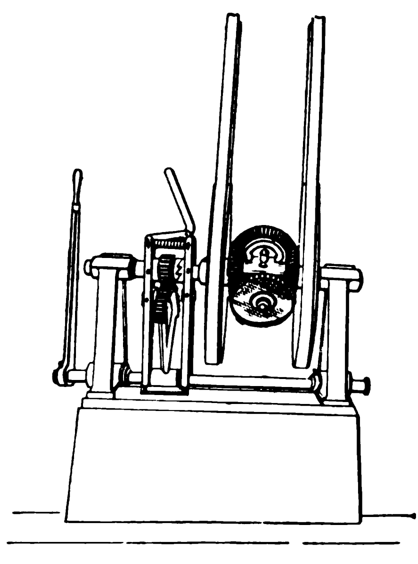


Fig. 131.

These ratchets are keyed to the shaft but free to move along it, and are thrown in or out of gear by a short lever with a spring. The pinion engages a sector connected by suitable rods and levers to the valves which regulate the jet or jets of water. If the two pulleys revolve in opposite directions at exactly the same speed, there is no motion of the central shaft which carries the two loose bevel wheels, hence no motion is communicated to the pinion or sector. If, however, the working wheels—driving the electric installation say—increase in speed, owing to part of their load being thrown off, there is a difference of speed of the two loose bevels; hence the central gear shaft changes its position, acting on the pinion and sector and partially closing the valves. The opposite action takes place if the speed of the working motors decreases below that of the constant speed motor.

The ratchets are thrown out of gear on starting, and when the working wheels have reached the normal speed, the governor is thrown into gear.

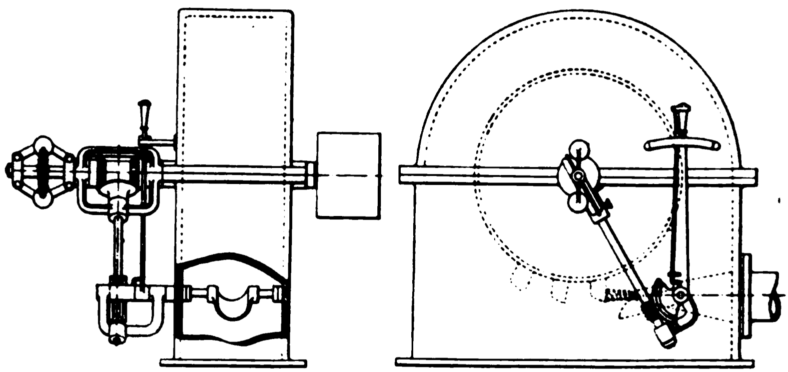


FIG. 132.

This is an interesting application of the differential system in governing, as well as an important apparatus in itself.

It may also be mentioned that another arrangement Mr. Hett has patented, is a nozzle with an internal conical spear or plug, which may be moved by hand or governor so as to alter the power of the jet, and of the motor driven by it. In another case a movable shutter is employed to cut off a portion of the jet.

PITMAN GOVERNOR FOR PELTON WHEELS.

Throttling governors are not very suitable as the pressures are usually high, and with long pipes, if the load is thrown off suddenly, throttling or stopping the flow may cause sudden and serious rise in pressure due to the "water-hammer" action.

Many governors act by deflecting the jet through a movement of the nozzle, this movement being controlled by a governor; this secures good regulation, of course without any attempt to economise water.

The trouble of making a satisfactory water-joint in the nozzle is, however, considerable.

In the Pitman governor, two views of which as applied to a Pelton wheel are shown in Fig. 132, a separate deflector is used, this deflector being controlled by a centrifugal governor. This deflector divides the jet, deflecting part of it, or it may deflect the whole of it, if the load change necessitates this. The deflector is moved by a worm segment shown in the right-hand view, this segment being moved by a worm actuated by the bevel gear shown in the left-hand figure. A light friction clutch actuated by the governor locks one or other of the two bevel wheels on the main shaft with that shaft according as the speed rises above or falls below the normal; thus the worm is driven in one or the other direction, and more or less of the jet is cut off and deflected from the wheel cups. A small Pelton wheel fitted with this governor showed on trial a rise of only 3 per cent. in speed when the whole load was thrown off. The quadrant can be disconnected from the deflector when hand regulation only is desired.

The invention is due to Mr. Pitman, of Ledbury.

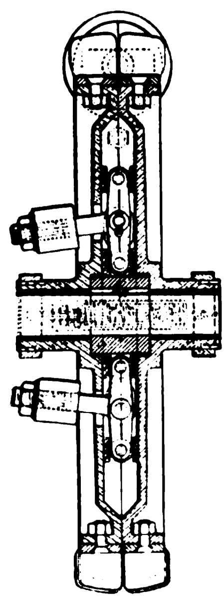


FIG. 133.

In the Cassel wheel excellent regulation is effected without, however, any attempt to save water at lower than normal powers. The wheel consists, as shown in section in Fig. 133, of two portions or discs meeting along a central plane at right angles to the centre line of shaft. These portions are held together by springs, and have buckets round their peripheries like the Pelton wheel. The discs are free to move endwise along the shaft and turn it by projecting lugs, which bear on arms forming part of the central spider which is keyed on the shaft. This spider also bears the pivots on which turn T levers with weights fixed at their outer ends. When these weights, owing to increase of speed above the normal, move outward radially the upper or table part of each T lever separates the discs to a smaller or larger extent, depending on the excess of speed. Hence the jet passes either partially or wholly through between the discs, and of course gives up only a portion, or none of its power to the wheel. The regulating masses are so proportioned that at the normal speed their centrifugal force just balances the tension of the springs; any increase of speed, therefore, results in a separation of the discs, and a consequent quick return to the normal speed again. As good regulation is usually more important, in these machines, than saving of water, this seems a likely and successful method of regulating whereby no attempt is made to interfere with the flow of water in the jet.

Governors for Pelton Wheels of Large Powers.

The speed regulation of most of the Pelton wheels developing large powers for purposes such that good regulation is essential, is effected by hydraulic governors.

A successful form is applied to a nozzle of rectangular shape, the governor moving a shutter, the edge of which forms one side of the rectangle. Thus, if the load diminishes, the area of the jet is decreased to suit the conditions of load at the same time that part of the water is diverted to exhaust without acting on the wheel; thus obviating, to a considerable extent, undue increase of pressure in the supply pipe.*

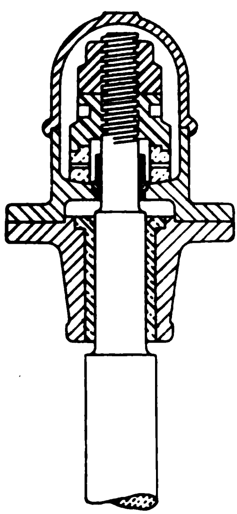


Fig. 134.

Footstep Bearing for Turbines.

In most turbines, and especially in the larger wheels used for low falls, or where a very long and heavy vertical shaft is used, with perhaps a dynamo attached directly to its upper end, a very heavy weight has to be borne, and the design of a proper footstep or similar bearing becomes an important matter. In some cases part of the weight may be balanced as already described, but in many cases a heavy weight must be supported on the footstep. In some

* Unless some such relief be provided, if the time taken to stop the jet entirely be less than the quotient of twice the length of the supply pipe by the velocity of sound in water, the increase of pressure will give an *increase* of kinetic power during the early stages of the closing motion, and the speed of the wheel will go up instead of down.

cases these are placed below the turbine shaft and immersed in water, *lignum vitæ* being employed for the bearing surface. There are drawbacks in such an arrangement, due to wear and difficulty of access. An overhead step is now more usually employed, a good modern example being shown in Fig. 134. The upper cup is entirely closed and filled with oil. Thus the step-plates, which take the pressure of the cap screwed on to the top of the turbine shaft, and thus the pressure due to the weight of the turbine and shaft, have the oil freely circulating around and between them. The cap is held on by a nut which can be turned to readjust the turbine after wear of the step-plates, or to lower it for inspection or repair. The pressure on the bearing surfaces may be relieved by passing oil at a pressure of 200 lbs. per square inch or so through suitable holes in the contact surfaces. In some cases this oil is taken from the governing apparatus.

XVII.

HYDRAULIC PRESSING MACHINERY.

THE HYDRAULIC PRESS.

THE principle underlying the action of hydraulic pressing and lifting machinery is said to have been discovered by Stevinus, but was enunciated by Pascal 150 years before Joseph Bramah made a practical use of the principle. Pascal's statement is that "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 it exactly, then a man pushing the small piston will equilibrate that (?) of 100 men pushing the piston which is 100 times as large, and will overcome that of 99." In other words, there will be equilibrium if the forces are inversely as the areas of the pistons.

This is a direct consequence of the law—proved at page 5—that in a fluid, if gravity be neglected, the intensity of pressure is everywhere the same.

This result may be obtained in another way as an illustration of the "law of work," which may be stated as follows:—"The work given to any machine, or done *on* the machine, is exactly equal to that *obtained from* or done *by* the machine, if there is no waste and no storage of energy, and if the machine works at a steady speed"

To apply this law to a case which illustrates Pascal's principle exactly: in Fig. 135 are shown two vessels E and D connected by the pipe S, and therefore fulfilling the conditions of Pascal's one vessel, the vessels and pipe being filled with water, except where the space is occupied by the ram R and the plunger P. Suppose the vessels and pipe to be watertight, and that water is incompressible: hence if l inches of the plunger enter the water, $a\,l$ cubic inches of water are displaced by it, its cross-section being a square inches. This water tries to escape, but if nothing yields or breaks it cannot do so. Hence it must find room by forcing the ram R (and its load) up

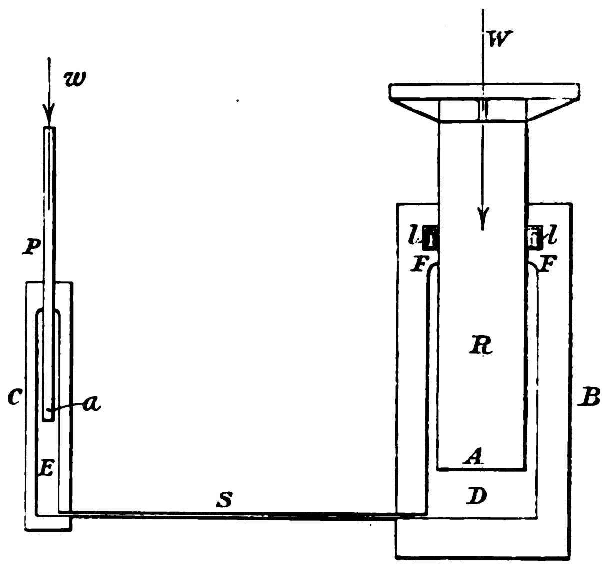


FIG. 135.

through a distance c inches. Hence, from the law of work, neglecting friction, $w\,l = W\,c$; if the area of the ram is A square inches, $\frac{l}{c} = \frac{A}{a}$, and as l divided by c is the *relative* motion of P and R, evidently the velocity ratio of the machine is the *ratio of the area of the ram to that of the pump plunger.*

As an illustration, if A is 100 square inches, and a is 1 square inch, then when P moves in say 100 inches, 100 cubic inches of water are displaced, and will find room by moving R up *one* inch.

It should be borne in mind, however, that in all machines *less* energy is obtained from the machine than is put into it; the ratio of the latter amount to the former (under conditions of steady speed and no storage of energy) is called the efficiency of the machine. In the hydraulic press this may be considerably over 90 per cent.

To be accurate, the energy exerted by P is equal to that necessary to overcome friction, together with that spent in raising W.

Packing Leathers.

It may not be out of place here to direct attention to the method of packing the ram R so as to allow it to move in and out of the press cylinder watertight. In the cylinder is a rectangular recess in which a tunnel-shaped piece of leather is inserted, as shown at *l l* (Fig. 135). Some of the water in D finds its way past F F; this water, getting inside the leather *l*, forces the latter more and more

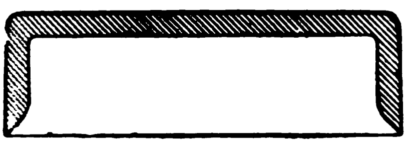


Fig. 136.

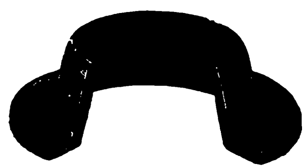


Fig. 137.

tightly against the ram as the pressure becomes greater and greater, thus preventing leakage. This, in fact, constitutes the most important part of Bramah's invention.

Leather packings are of different shapes. Sometimes they are cup-shaped, as shown in Fig. 136, this form being used in the hydraulic jack. Sometimes the shape is that of a hat with straight brim and no crown, as in Fig. 137, or they may be U-shaped, as in Fig. 138, which is the form usually employed for warehouse press rams. Such packings

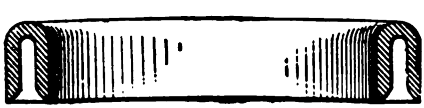


Fig. 138.

are made by soaking a disc of leather of the proper size in hot water until the leather is soft, and then pressing it into a mould of the required shape by a corresponding core, which is forced and held down by a screw till the leather is dry, or in cases where a large number of such leathers have to be made, by the ram of a small hydraulic press.

In the case of the U-leather the pressing may be done in two stages: first it is pressed into the cup shape, and then into the U-shape, the central disc being afterwards cut out. The recess in the press cylinder in which such leathers sit should be lined with gun-metal, and in many cases that portion of the ram which comes into contact with the leather is also covered with gun-metal or copper.

Friction of Leather Packings.

The friction of such packings as those referred to above has been the subject of a considerable number of experiments. Mr. Hick, of Bolton, found that the law of friction in such cases is a simple one, showing friction proportional to total load on the ram, and inversely proportional to the diameter of the latter.

The law can be expressed approximately as follows:—

$$f = 0 \cdot 04 \frac{P}{D},$$

P being the total load on the ram of which D is the diameter in *inches*. If, for instance, the diameter of the ram is 8 inches and the total load 50 tons, the force necessary to overcome the friction of the leather,

$$f = \frac{0 \cdot 04 \times 50}{8} = 0 \cdot 25 \text{ ton} = 560 \text{ lbs.}$$

The formula may be readily put into the following shape. Since $0 \cdot 7854\ D^2 p$ may be taken $= P$, $f = 0 \cdot 0314\ Dp$ for well lubricated leathers, where p is the pressure of the water in lbs. per square inch. With new or badly lubricated leathers the coefficient is $0 \cdot 0471$.

The U-packing can be more readily placed in position, if the cylinder, as is usually the case, be fitted with a removable ring or gland. Sometimes a brass ring is inserted inside the leather to keep it up to its work as at A (Fig. 139). The gland in this case is not very tightly screwed up. The lower end of the ram should be well rounded, as the leather is usually a little smaller in inside diameter than the outside of the ram.

For small rams or pistons, strips of leather wound spirally are used as packing, or cup leathers may be used as at B. In this case a ring or core is placed inside the leather, and fastened to the piston with set screws or bolts. As there is a tendency for leakage to take place round the piston-rod, the seat is usually left a trifle higher near the rod to ensure that the leather shall bear tightly on its seat there. India-rubber packing is also sometimes employed.

HEMP AND OTHER ROPE PACKINGS.

These are now used for hydraulic cylinders on account of their comparatively small cost. The hemp must be compressed with a great pressure, sufficient to make a joint against the ram, watertight under the *highest* pressures, hence the friction of such packings is high even when small pressures are sometimes used. The gland and stuffing-box are similar to those employed on steam-engine cylinders. The friction of such packings cannot be so accurately expressed as in the case of leather packings. It is said that if well lubricated the rule $f = 0 \cdot 1 \times p D$ may be employed, which gives $f = 1782$ lbs. in

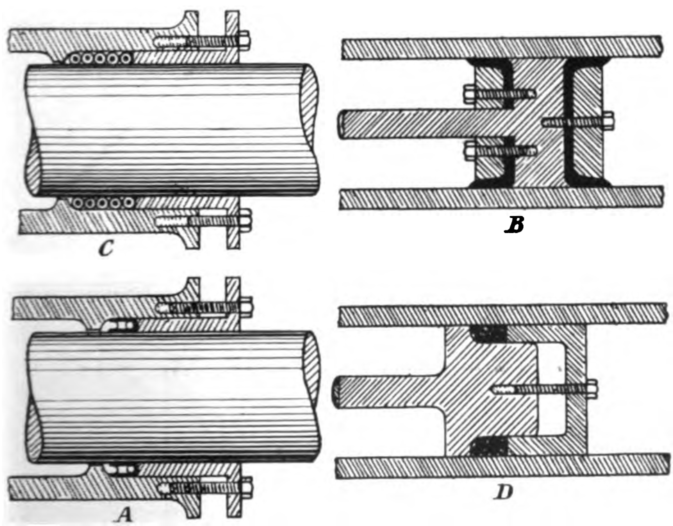


FIG. 139.

our example, or over three times that of the leather. This formula is, however, doubtful, for in this case friction is too dependent on lubrication, the tightness of packing, etc., to be readily expressible in an easy rule.

It is possible to make such packings tight for very high pressures.

In some cases a packing consisting of an india-rubber core with a flax or yarn covering is employed as at C (Fig. 139).

The pressure of the gland squeezes the packing into an oval form, and thus a tight joint is secured without excessive pressure on the ram or piston-rod.

For small pistons or moderate pressures the rope packing shown at D (Fig. 139) is simple and inexpensive.

Hand Press.

The action of the press will be understood from an examination of Fig. 140, where a section of a hand press is shown. The pump plunger C on its upward stroke draws in water through the upward opening valve F; in the down stroke F closes and G opens, allowing the water to pass along the pipe E to the press cylinder D, which is

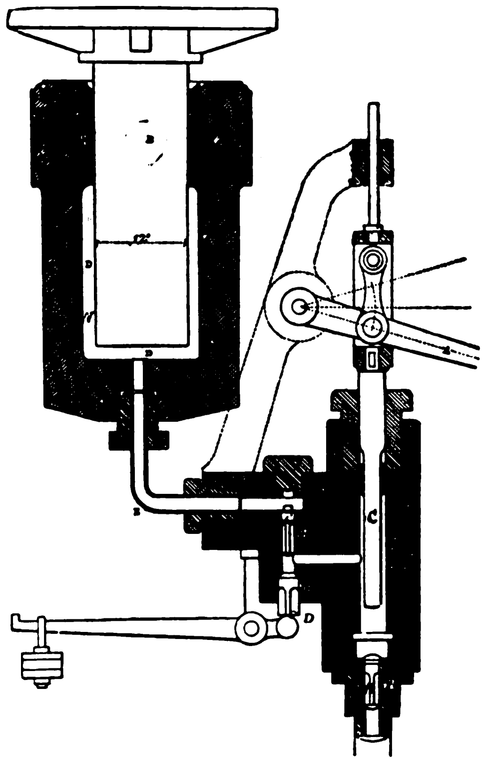


Fig. 140.

already filled with water. The influx of this new supply, due to the downward stroke of C, causes the ram B to be raised; thus goods resting on the platform attached to B are pressed on the continuance of the operation.

The safety valve H opens and allows some water to escape, should the pressure accidentally exceed the limit which has been fixed as the greatest consistent with safety. Other details are apparent from the figure.

This machine, formerly much used, has now to a large extent been superseded by the power press, worked either from private steam pumps or public hydraulic power mains.

Even with a machine of this kind great forces may be exerted, two men working the pump being able to crush into shapeless masses great blocks of oak, and even to reduce large cubes of glass or stone to powder.

Such machines are very efficient.

One reason why the frictional waste of energy is so small in the case of the hydraulic press, is that the motion of the water is very slow, for, as already proved, in fluids the friction depends very much on velocity, and is indefinitely small when the motion is very slow. Whatever loss there is from this cause, occurs in the *narrow* passages rather than in the press itself. The solid friction is mainly at the fulcrum of the lever, and at the glands where leather or hemp rub on metals, this quasi-solid friction being proportional to load. Hence we might expect to find—what experimenters have found—that the total friction is about proportional to the total load. With fluids like petroleum oils the friction would be less, but the difficulty of packing greater, whereas with fluids like tar, honey, etc., it would be necessary, in order to get a high efficiency, to make, perhaps, only one stroke per hour.

We have assumed that in the hydraulic press there is no storage of energy; this is hardly correct, even if we disregard the compressibility of the water. The lifting of the ram is, in fact, a storing of energy which is almost all given out again as the ram descends. It is usual to regard this lifting of the ram as an absolute small waste of energy, but if the load raised be, say, less than the weight of the ram itself, it becomes necessary to take it into account. This is the case in warehouse and hotel hoists or lifts, which will be referred to more fully in a later section. One difficulty which presents itself in attempting to take the weight of the ram into account is, that as the ram rises its apparent weight, i.e. the part of its weight to be overcome, increases. You know that a stone, when immersed in water, is easier to lift than when it is in air; and just so here, as the ram rises, more and more of it is in air and less in water, hence it is harder and harder to lift. Usually the loads on a hydraulic press are so great compared with the weight of the ram that the latter may be neglected; but in lifts it has often to be taken into account and balanced in ways which will be explained.

Press Details. Variation of Pressure.

In the case of hydraulic jacks, the load on the ram is the same throughout the whole operation, but this is not the case in the hydraulic press when used for baling operations. In the case of bales of cotton which are brought to England *via* the Suez Canal, it is necessary to compress the cotton so tightly that it looks like a piece of oak when cut, and, indeed, can be planed up like oak. In pressing material of this kind, there is, during the early part of the operation, comparatively little pressure on the ram; but it is the *greatest* total pressure to be exerted which determines the relative sizes of ram and plunger. It is obvious that if the ram were to rise quickly during the early part of the operation under small pressure, and then more slowly and under greater pressure towards the end, a saving of time and a more regular expenditure of energy would be effected.

This object is, to a certain extent, carried out by different arrangements. In hand presses, for instance, the fulcrum of the lever is in some cases changed, so as to give a greater mechanical advantage towards the end; or a large pump may be used at the beginning, and a small one at the end; or two equal pumps may be used first, and only one afterwards.

In another form of baling machine twelve pump plungers are attached to the cross-heads of steam-engines. At the beginning of the operation all twelve are working and the pressure is small. As the pressure gets greater one set of four pumps is detached, so that they merely pump water back into the tank from which they take it, hence expend very little energy. Eight pumps are now forcing water into the press, which rises much more slowly than before, but as the eight have nearly the whole horse-power of the engines acting on them, they are able to give to the water a far higher pressure. Near the close of the operation four more are thrown out of gear, and the pressure is correspondingly increased. It is now more common to use sets of six pumps and throw them out of gear two at a time. In some cases more than one press and ram are used, the extra rams commencing to act when the greater pressure is required.

Or an accumulator (p. 233), supplying water at, say, ½ ton per square inch, may be used for the earlier part of the operation, involving 70 to 80 per cent. of the total lift of the ram, the remaining 20 per cent. movement being effected by the action of water direct from the pumps, rising to a final pressure of 2½ tons per square inch or even higher. This method is very convenient, saves time, and requires smaller engine power; but it is not economical, as the full accumulator pressure is employed at the earlier stages, where 100 or 200 lbs. per square inch would be sufficient. In some modern presses the hydraulic intensifier is employed, and the pressure of the water supplying the press varied in this way; but this has the same disadvantage of want of economy during the earlier portions of the operation. To obviate to some extent this difficulty, Mr. Bellhouse * introduced his hydraulic intensifier (page 425), used first as a diminisher, giving a pressure of 224 lbs. per square inch, the accumulator pressure of ½ ton per square inch being next employed, and afterwards the operation is completed by the help of the intensifier used as an intensifier, and giving 2½ tons per square inch. Probably the further development of the intensifier method will lead to still greater economy.

The change of pressure required during the operation of pressing one class of Manchester goods is clearly shown by the curve in Fig. 141, whilst in Fig. 142 are given curves which show the pressures required per square foot of platten, or bottom of baling box, necessary to bale the given materials to the stated density. In these the friction against the sides of baling box is not taken into account. This may require an addition of 25 per cent. for bales up to 40 lbs. per cubic foot and as much as 30 or 40 per cent. for denser baling. The baling box should be 1¼ inch less in length, breadth and depth than the size of bale required.

Pumps for Press Work.

In connection with modern presses direct-acting pumps are most usually employed. This system was first applied by Messrs. Nasmyth, Wilson & Co. In their system direct-acting pumps without fly-wheels are used, the cranks being set at quarter-centres; the engines move whenever water is required, the steam being used unexpansively and throttled to agree with the requirements of the load. This system is economical, but requires large plant.

Direct-acting pumps with fly-wheels are much used, it being more usual to have fly-wheels than to trust to the automatic reversal of motion.

Details of Presses.

Press cylinders were formerly made of cast iron only. Presses from 9 to 14 inches in diameter, with a thickness in the latter case of 10 inches, gave little, if any, factor of safety at the highest pressures. Thus the 14-inch press was stressed to 4 tons per square inch at its

* 'Minutes of Proceedings of Institution of Civil Engineers,' vol. xcix.

innermost layers, and even with chilled castings this gave almost a dangerous stress. The maximum squeeze exerted by the ram was 460 tons. Owing to the wish of exporters to pack tighter and tighter, and thus reduce freight charges, steel was tried as a material for press cylinders, and has now come into general use.

The ultimate tensile stress in this case is 25 to 35 tons per square

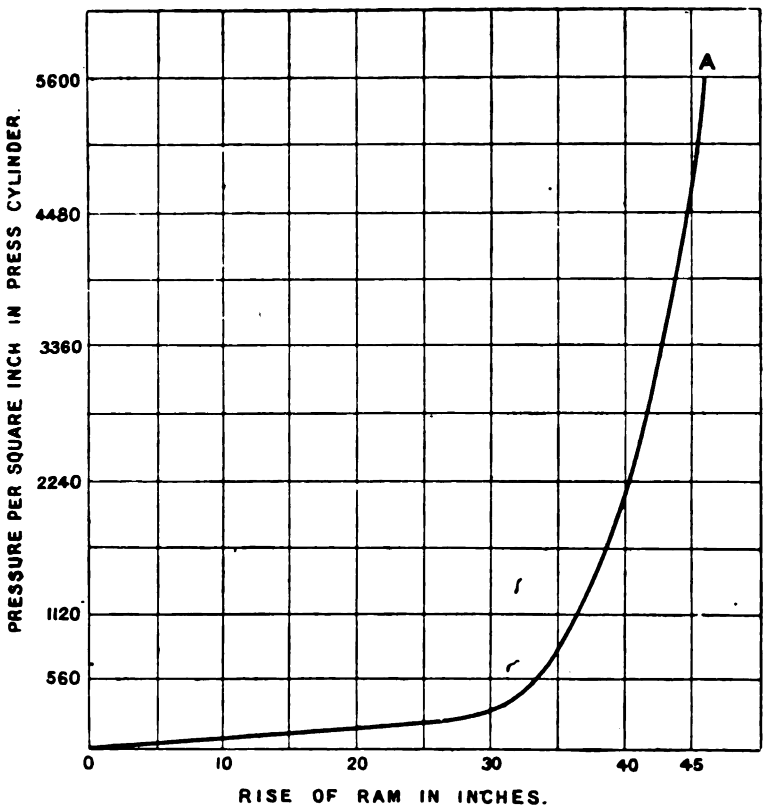


FIG. 141.

inch, the usual thickness for a 14-inch press being about $2\frac{3}{4}$ inches, thus giving a factor of safety of 3 or more, at usual pressures of $2\frac{1}{2}$ or $2\frac{3}{4}$ tons per square inch, and permitting a maximum squeeze of 700 tons.

The ram acts of course in compression, hence cast iron is good enough. It is usual to case the upper part for 42 to 48 inches with

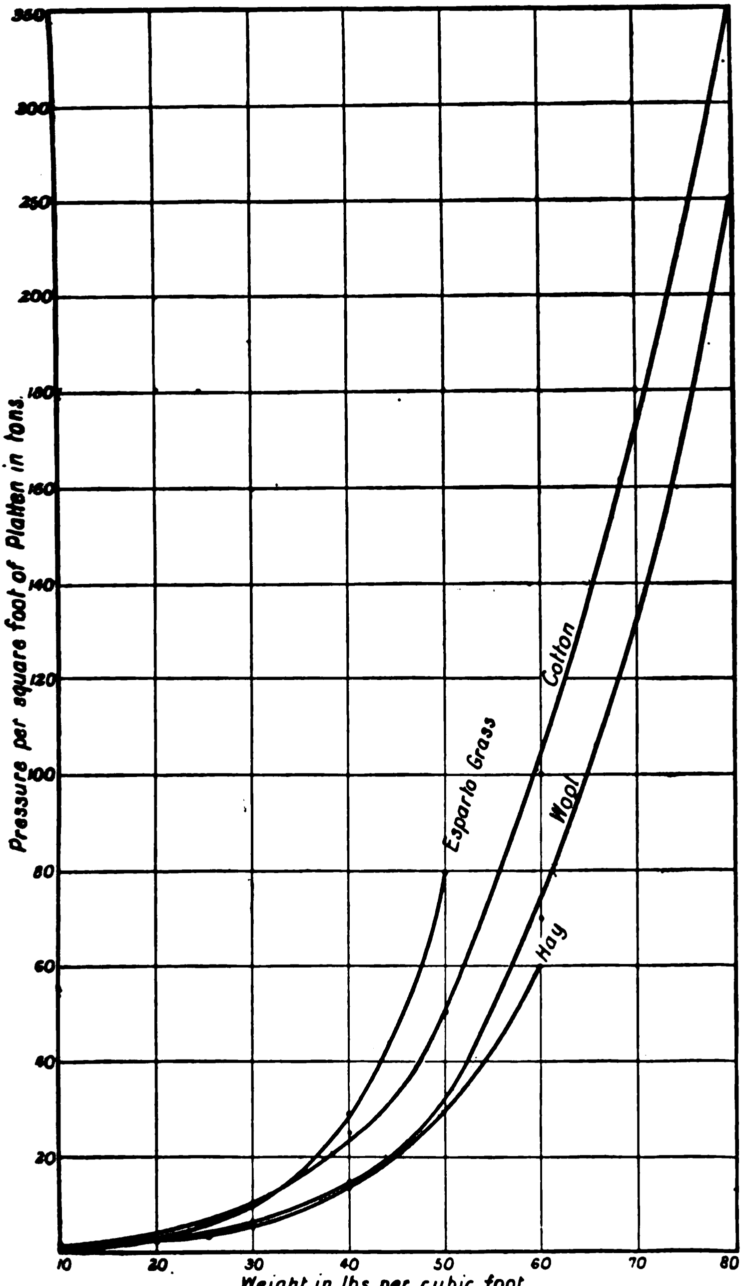

Pressure per square foot of Platten in tons.
350
300
250
200
180
160
140
120
100
80
60
40
20
Esparto Grass
Cotton
Wool
Hay
10
20
30
40
50
60
70
80
Weight in lbs. per cubic foot

FIG. 142.

a brass hoop, to reduce friction and prevent deterioration of the leather packing, or in some cases a copper covering, deposited by electrolytic methods, is used.

The platten or table is guided to move vertically by four rollers, working on accurately turned pillars.

It will be understood that the table has to be removed whenever a new packing leather is introduced, hence facilities for this removal must be provided in the design. The press tops and bottoms are made of cast iron, and are in reality compound beams strengthened by flanges, a continuous top flange of greater cross-section than the somewhat similar bottom flange giving the increased tensile strength which the material lacks, and which experience has shown to be necessary. The designer must also, in this and other hydraulic machines of the same class, provide a means of emptying the cylinders of water in time of frost, when the machine is out of use.

A reference to a drawing of a good modern press will show how these matters are all carefully adjusted.

Piping.—Steel piping is now always used for the conveyance of the *high-pressure* water supplying the press, its greater tensile strength and smaller liability to corrosion rendering it much more suitable than either cast or wrought iron.

Efficiency.—With small pressures and low speeds, the efficiency of an accumulator or press with constant load, may be determined approximately by observing the pressures by gauge, as the ram rises and as it falls. An efficiency of about 98 per cent. may be expected, but with high pressures or speeds the method cannot be accurately carried out. A series of actual tests for efficiency at, or including, high pressures, would be most interesting.

Modern Hydraulic Press.

Fig. 143 shows a good modern hydraulic press. It has a top and bottom platten, also three massive columns, a bottom cylinder with its ram, and *two* top cylinders with their rams; there are also the top and bottom followers with lashing plates and revolving boxes —the parts below the baling platform are not shown in the figure— the function of the whole being as follows:

After the sliding boxes have been filled, say with cotton, from the upper or filling floor, the first box is brought over an opening in the raised platform, seen underneath the upper floor, and its contents are discharged into one of the revolving boxes of the press (seen nearly under the press), which is at that time brought below that opening.

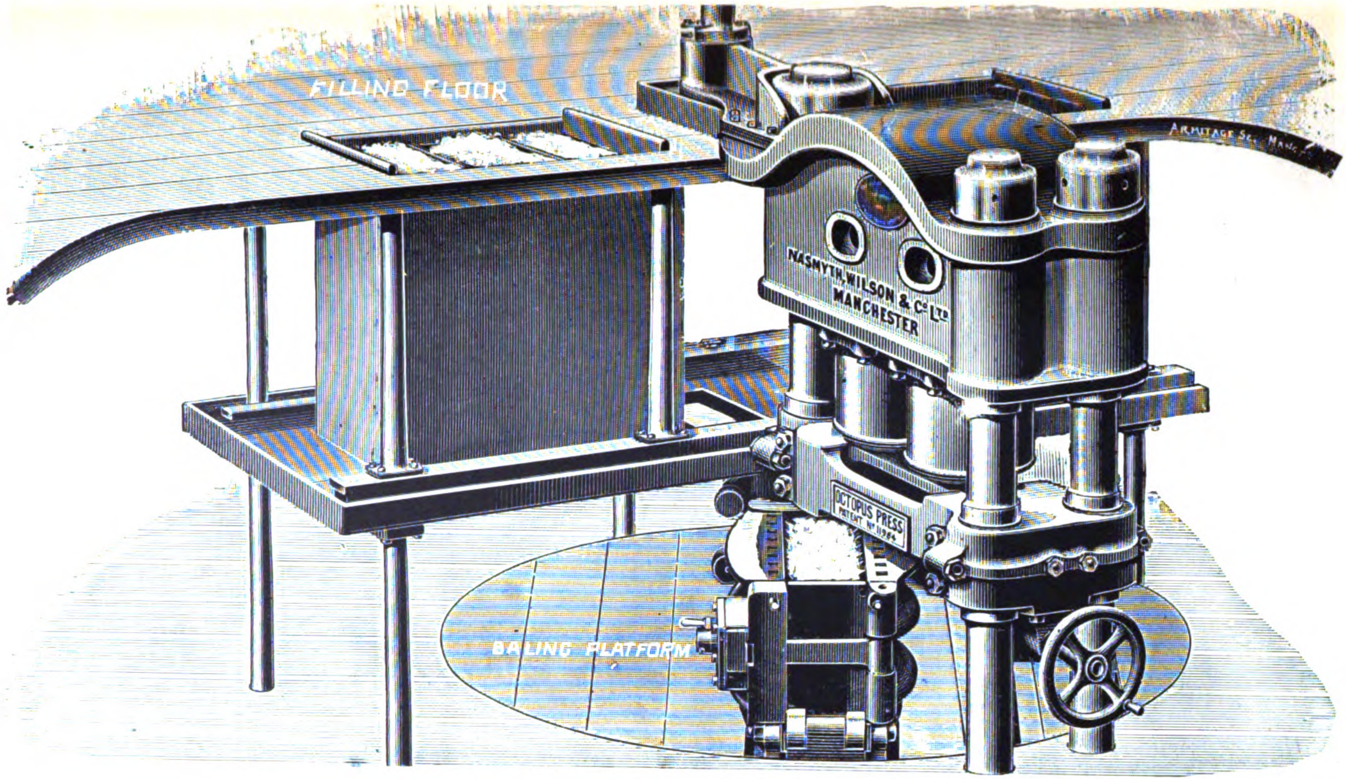

FILLING FLOOR
NASMYTH, WILSON & Co Ltd
MANCHESTER
OCTOPUS PRESS
BALING PLATFORM

FIG. 143.

The frame is then moved, and the second filling box is brought over the opening in the platform, and its contents pressed into the revolving box by means of a "treader" attached to the ram of the hydraulic cylinder provided for that purpose. The treader is then withdrawn, and the cotton being held down by an automatic apparatus, the third box is brought into position and its contents forced in by the treader. The revolving box having now been filled and the cotton held by the automatic apparatus, it is turned round and brought with its contents into the position to be compressed and finished by the rams of the press, whilst the other revolving box is moved under the opening in the platform, and undergoes a similar filling simultaneously with the pressing and finishing of the first bale. There are three rams to each press, the bottom one doing the preliminary pressing, the addition of the other two larger top rams giving the necessary finishing force. The lower end of the bottom ram, and the working surface of the top rams, are covered with gun-metal to diminish friction and abrasion of cup-leathers.

By this arrangement a considerable saving in the time necessary for making a bale is effected, and bales of great weight can be made in a press of ordinary dimensions.

XVIII.

HYDRAULIC JACK.

THIS apparatus can be most conveniently studied here, though belonging to the class of *lifting* rather than pressing machines.

It is one of the most useful of the portable machines for raising loads, and it is rapidly displacing—indeed has already in a large measure displaced—the older and less efficient screw jack. Fig. 144 shows a section of the best-known form of the jack. Here the ram is stationary, and the casing or press moves; the ram being packed in a watertight manner by a cup leather of the kind already described, fastened on the top of the ram by a washer and set-screw as shown. The handle N works the pump plunger by means of the crank K, whose length is K O. When the handle is raised, water enters from the cistern C, by the inward opening valve S, the space under the plunger. If the handle be now pushed down, part of the water under the plunger finds its way through the downward-opening

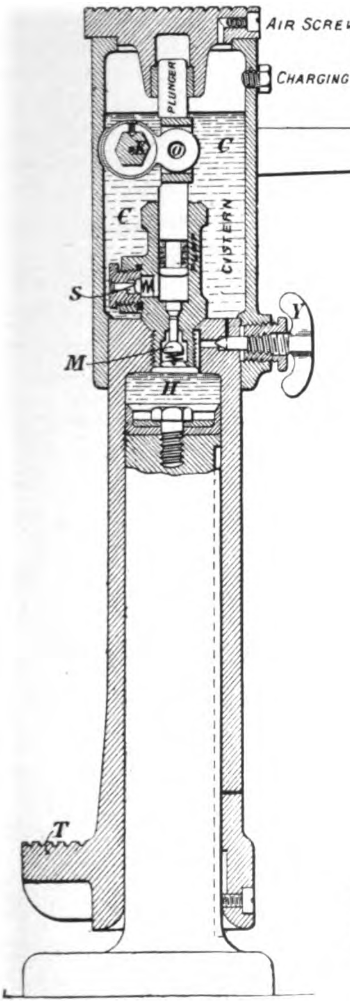


FIG. 144.

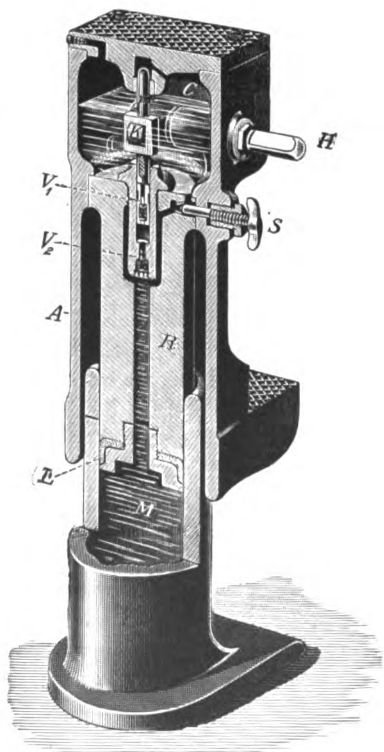


FIG. 145.

valve M into the space H above the ram, and as more and more water is forced into this space the casing rises on the ram, thus raising any load which may be resting on the casing of the jack.

The toe T may be employed for lifting rails or other low-lying loads. The load is lowered by slackening the lowering screw Y, which opens a passage from H to the cistern C, and the load on the jack forces the water from H back into C, thus diminishing the volume in H and lowering the load. A set-screw or other projection in the casing of the jack works in a vertical slot in the ram to prevent the former from turning on the latter as it rises or descends.

To prevent the casing from being raised too high on the ram, there is a little hole in the former, which, at a certain height, allows some water to escape.

A newer and improved form of the jack, for raising very large loads, is shown in Fig. 145.

The plunger in rising creates a partial vacuum under it, the water entering through the grating and by the valve V_1, which is in the plunger itself. On the down stroke of the plunger this valve is closed and V_2 opened, the water is therefore forced through the passage in the ram R into the space M. As more and more water is forced into M the ram R and casing A are raised, and with them the load on the casing.

In this form of jack the ram is protected from injury, which is not the case in the older and commoner forms of the apparatus, and the cup-leather packing L is kept moist more readily. If, from any cause, leakage occurs in the older form of jack, the water all escapes and the leather becomes hard and dry; whereas in this form, even if leakage does take place, the leather is still immersed. A horizontal section of the ram is *not* circular, but has a flat side, the casing being, of course, of the same shape, hence the groove and set-screw are not required. These and other improvements show the evolution of this appliance in the capable hands of Messrs. Tangye.

Of the practical utility of the hydraulic jack as a portable machine for raising weights, it is not necessary to say much. Our great advance in hand load-raising appliances is evident from the facts, that half a century ago Le Bas required the help of *480 men* working capstans to raise the Luxor Obelisk in Paris, whilst twenty-six years ago Cleopatra's Needle was raised to its present position on the Thames Embankment by *four men, each working one hydraulic jack.*

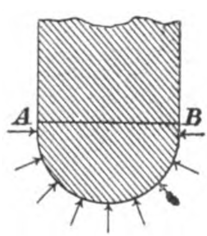


FIG. 146.

Some readers may think that in these machines the shape of the end of the ram has some effect on the total force with which the ram is pressed upwards. This is not so; the fact to be borne in mind is that water is almost frictionless, and hence can press only normally on any surface confining it.

We may imagine the water particles to be little bodies very well greased, each particle pressing on its neighbours because they all press on it, but it presses and is pressed equally in all directions. When, therefore, it comes in contact with a wall or boundary of a vessel, the pressure must be normal on the boundary, and the same on *every unit of area* at the same depth. This is also true of any interface separating two portions of the water.

The only thing to consider then, is whether the ram in moving up one inch leaves the same empty space behind whatever the shape of the end may be, and a little consideration of such a figure as Fig. 146 will show that this is true.

Efficiency of the Hydraulic Jack.

It is rather difficult to find the law of efficiency of an apparatus like this, where the motion is reciprocating, as a weight has to be applied to the handle, and this weight must be lifted by hand in the upward stroke of the handle.

The following experiment with a 3-ton jack—not, however, in very good order—was carried out by an evening student (Mr. J. W. Kearton) at the Technical College, Finsbury. Great care was taken to get approximately accurate results, the load being applied by a long lever, and the handle replaced by a pulley, so that the arm of the applied force might be constant.

The results shown on Fig. 147 were obtained.

Efficiency of a Hydraulic Jack. Experimental Results.

Three-ton hydraulic jack. Mechanical advantage of pulley used instead of handle, $14\frac{3}{4}$; diameter of ram, 2 inches; diameter of plunger, 1 inch. ∴ velocity ratio of jack $= 14{\cdot}75 \times 4 = 59$.

$\text{Efficiency} = \dfrac{\text{work given out}}{\text{work put in}}$. Let W be raised 1 foot. P must move through 59 feet. ∴ work given out $= W \times 1$, work put in $= P \times 59$, or efficiency $= \dfrac{W}{59 \times P}$, or generally $= \dfrac{W}{r \times P}$, where r is the velocity ratio.

The smallest force P at the handle necessary to raise steadily a

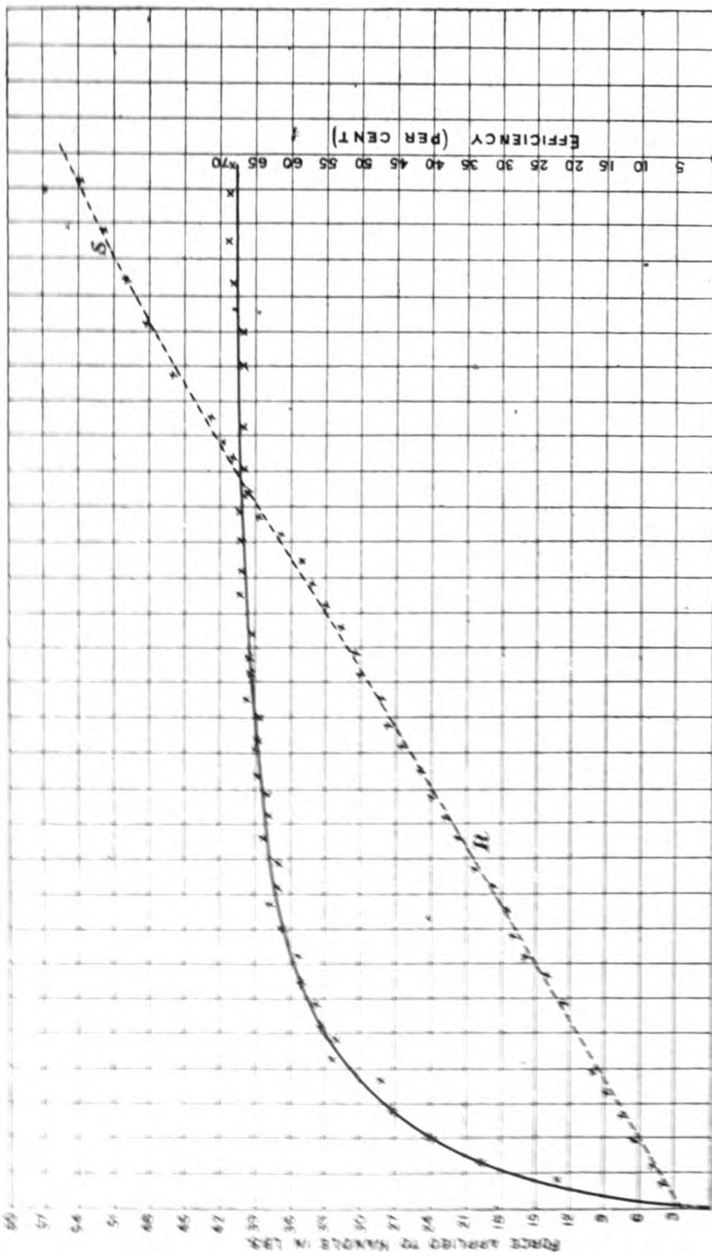

EFFICIENCY (PER CENT)
LOAD IN LBS
FORCE APPLIED TO HANDLE IN LBS

FIG. 147.

load W was in each case observed. Values of P and W are shown in the lower curve (Fig. 147).

It will be seen that the corresponding values of P and W are connected by a "straight line law." In other words,

$$P = aW + c,$$

where a and c are constants.

To find their values in this case, take two points, say R and S, on *the curve*.

At R,

$$P = 21, \quad W = 780,$$

and at S,

$$P = 51, \quad W = 2025.$$

Putting these values into the general equation,

$$(1) \quad 21 = a \times 780 + c,$$
$$(2) \quad 51 = a \times 2025 + c,$$

whence by subtraction,

$$30 = a \times 1245, \quad \text{or} \quad a = 0{\cdot}0241,$$

and this value of a substituted in (1) gives

$$c = 2{\cdot}202.$$

Hence the law of the machine is

$$P = 0{\cdot}0241\,W + 2{\cdot}202.$$

The law connecting efficiency (E) and load (W) is

$$E = \frac{W}{59\,(0{\cdot}0241\,W + 2{\cdot}202)} = \frac{1}{59\left(0{\cdot}0241 + \frac{2{\cdot}202}{W}\right)},$$

which, when W becomes great, and hence $\frac{2{\cdot}202}{W}$ negligible, gives

$$E = \frac{1}{59 \times 0{\cdot}0241} = 0{\cdot}7 \text{ as the greatest possible efficiency.}$$

NOTE.—The student, in using this method of finding maximum efficiency, should carefully consider whether the term here represented by $\frac{2{\cdot}202}{W}$ is really negligible, by substituting for W the greatest load the machine will safely lift.

XIX.

APPLICATIONS OF THE HYDRAULIC PRESS.

Hydraulic presses are now used for a great variety of purposes—in fact, almost all pressing operations are performed by a modification of Bramah's famous machine.

Oils are expressed from seeds, porous materials are freed from moisture and consolidated, and even metals are forced to pass through orifices and assume given shapes by the great pressure of a hydraulic press. A treatise might be written on the various modifications of the hydraulic press, but our limited space permits only a reference to one or two forms which seem most interesting, and which may not be familiar to the reader. Of these probably a

Hydraulic Press for making Lead Pipes

is one of the most curious. The fact that metals like lead *flow* when subjected to great pressure, is referred to at page 4, M. Tresca's famous experiments showing this very clearly.

This fact is taken advantage of in making the ordinary lead pipes with which we are so familiar in connection with the fitting up of new dwelling houses in towns, and the visits of the plumber after a severe frost.

Fig. 148 shows clearly the arrangement as employed in the best works where lead pipes are made.

To the left of the figure are seen a small steam engine and pump by which water is forced into the press or water cylinder, seen in the lower portion of the central figure. The upper part of the ram of this press, shown in section, bears a thick vessel called a container, which can be detached and moved by means of the derrick (seen above) to the stove on the right. This container is placed on the stove and heated to 300° or 400° F.; it is then replaced on its ram, and molten lead is run in from the pot on the right. All scum or dirt having been removed from the surface of the molten lead, the container and the hollow lead ram are placed in proper position, as shown in the figure, the lead being allowed to cool and consolidate. The pump is now put into operation, and the container is forced upwards on its small hollow ram, shown in the central figure. This ram has an annular die in its lower end, with core fastened to the container, the section of the annular hole in this die being that of the required lead pipe. As the container moves up under the great pressure of its supporting ram, the lead is forced through this die in a continuous stream, which is the lead pipe, this pipe being wound

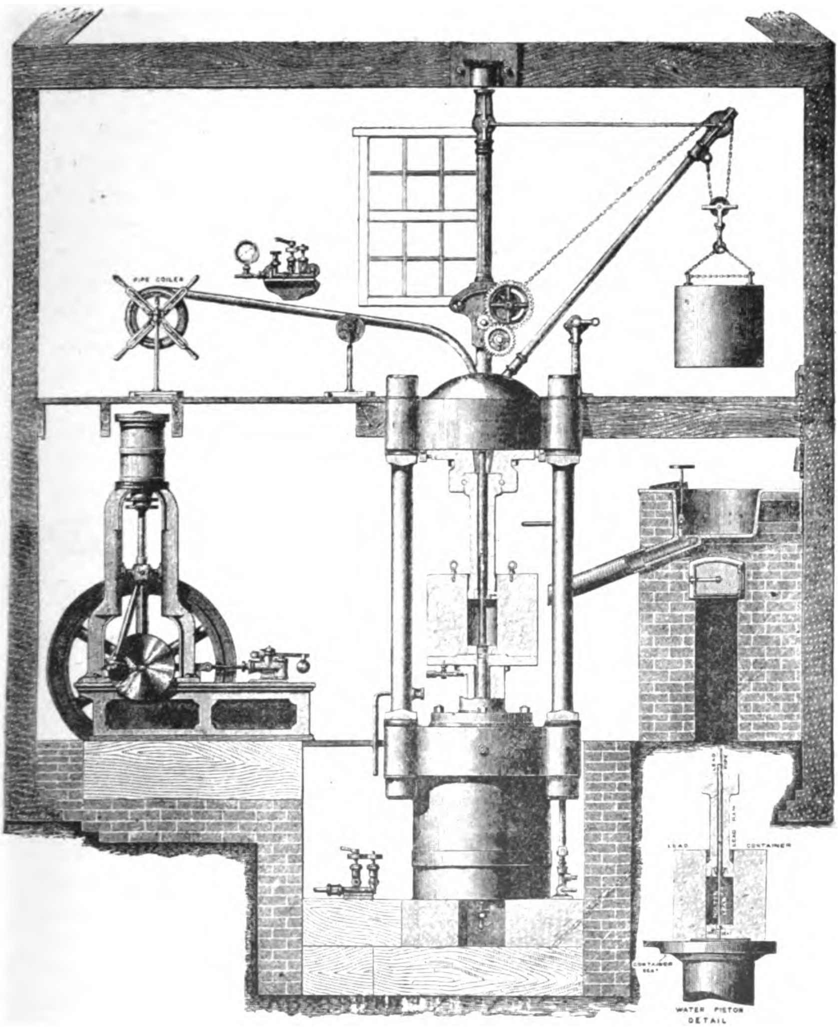


FIG. 148.

on the hand wheel shown in the top left-hand portion of the figure Larger pipes are supported from the ceiling by small pulley blocks and ropes, and are not wound on the reel.

The pressure required for making lead pipes is from one to two tons per square inch, according to the size of pipe. Composition pipes, and pipes made of tin and covered with lead, are constructed, also electric light cables covered with lead, by a modification of this machine. The arrangement for covering cables is shown in Figs. 149 and 150. In this case the container is entered by a *solid* ram, and the cable is led through laterally in a way that will readily be understood from the plan (Fig. 150). The cable, in passing through the container, which contains lead under high pressure, emerges at the other side with the required covering of lead firmly adhering to it. In this case there is a continuous circulation of cold water in a special chamber round the hollow block through which the cable runs.

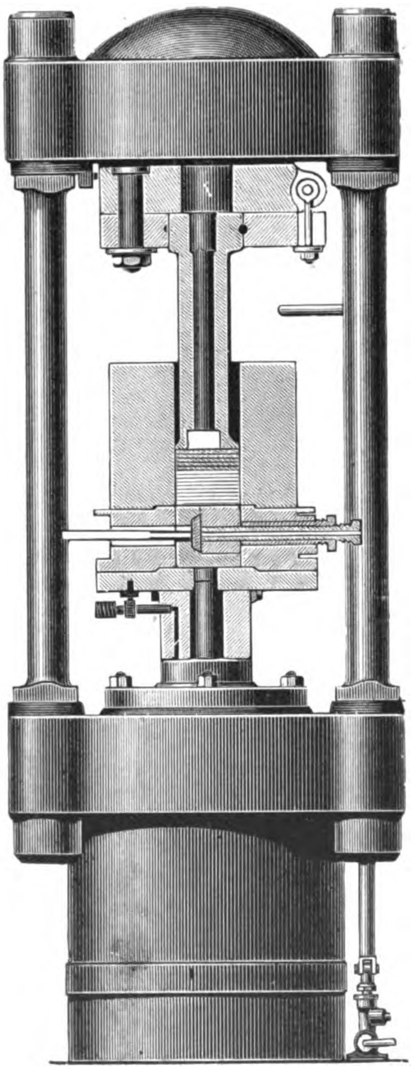


FIG. 149.

The dies for regulating the thickness of covering are placed in the hollow block, and the guiding cores are secured by regulating screws to the container. These cores are either hollow or double-cased, to allow of the flow of water. To prevent the metal passing through the side of the hollow block, the core is screwed up against the regulating ring or die, the cable is passed through the core, and when the metal is solidified in the first charge, the core is unscrewed back as desired, to allow the metal to pass round and cover the cable. Thus the cable is only exposed at the point of contact, a very important matter. Many of the above devices and improvements are due to Mr. Alexander Wylie, of Johnstone, near Glasgow.

Oil Presses.

A useful application of the hydraulic press is afforded in the operation of expressing oils from seeds. This oil-pressing business has now become widespread, and the kinds of seeds treated very numerous.

Taking an important example, linseed, the method of treatment is briefly as follows :—The seed is first cleaned and separated from

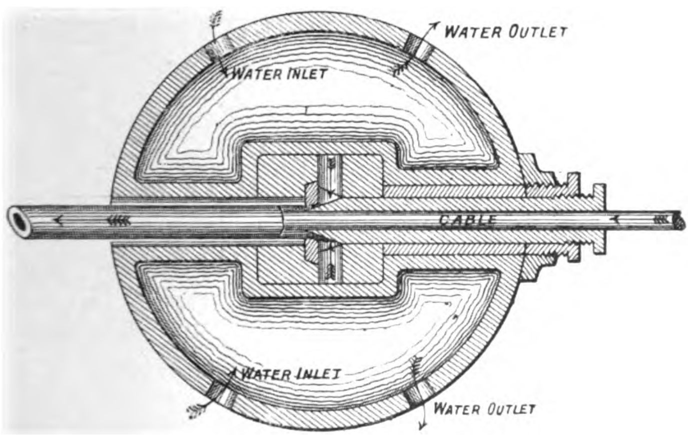


Fig. 150.

impurities ; it is then crushed between rolls, usually five in height, three being of larger and two of smaller diameter ; the seed thus receives four crushings.

The crushed seed is then heated in a vessel called a kettle, in which live steam is injected into the mass, whilst it is thoroughly stirred by machinery, thus making it hot and moist. The stuff is now "moulded" in a hinged bottomless box, usually about 29 inches by 13 inches, and 3½ inches deep. A tray, with a piece of cloth about 6 feet long and 13½ inches wide on it, having been previously placed under this box, the loose ends of the cloth are folded over the seed, and the whole pushed forward over the ram of a small hydraulic press in the moulding machine. This forward motion automatically opens the valve of the press, and the mould of seed is subjected to pressure by the ram, which reduces the cake of seed from $3\frac{1}{2}$ inches to $1\frac{1}{4}$ inch in thickness, or to a point at which the oil is just ready to flow, but is not actually expressed. Whilst one cake is being thus moulded, the attendant is preparing another as described.

The semi-solid moulded cake is now ready for the oil press (Fig. 151). It, and others like it, as soon as ready, are taken one at

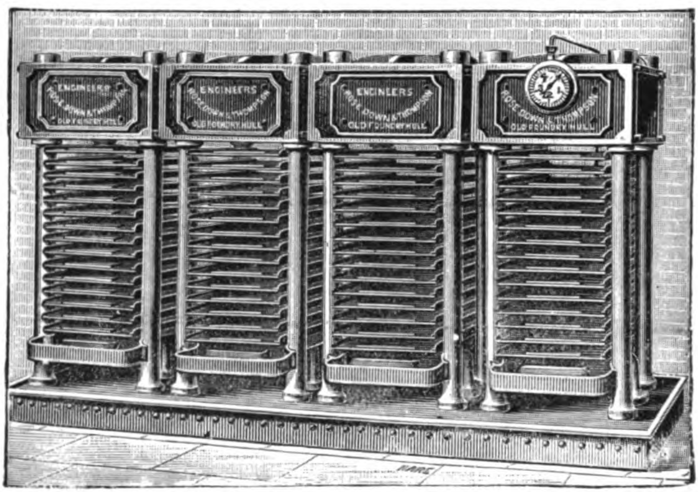


FIG. 151.

a time and placed in the press, each cake between two of the iron plates seen in the illustration. These plates are usually corrugated, and may bear any trade mark or legend desired.

When the spaces between the plates have been filled, the plates are pressed closer together by the ram of a hydraulic press emerging from its cylinder, in a way which will readily be understood from the section, Fig. 152. Thus the sixteen cakes in each press are subjected to a pressure gradually increasing to $1\frac{3}{4}$ tons per square inch, provision being made for conducting the expressed oil to suitable cisterns for filtering, or wherever it is required.

The oil is used for many purposes, such as painting, and the cake, after the oil is extracted from it, forms a valuable food for cattle.

Mr. Greathead's Shield.

A very interesting application of the hydraulic press is seen in the shield employed by Mr. Greathead in the construction of the City and South London Railway, and other tunnels under the Thames.

Figs. 153 and 154 show the construction of the shield, and enable its action to be readily understood. The shield consists of a cylinder composed of two plies of steel plate, each ½ inch thick, riveted together with countersunk rivets. This cylinder carries at its front end a strong ring of cast iron, to which are bolted the plates and channel irons forming the face, with steel cutters for excavating the tunnel, which is made either equal to, or a little greater in diameter than the cylinder, depending on whether the tunnel is, or is not, straight at the point in question.

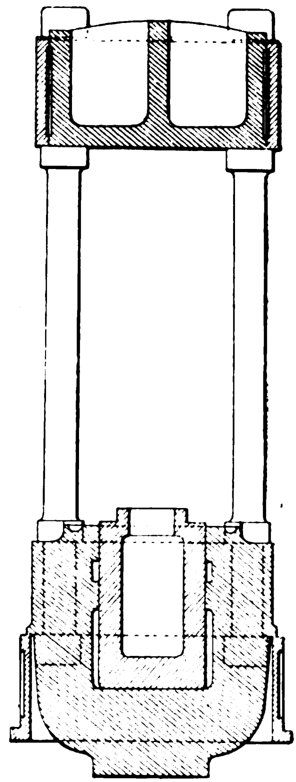


Fig. 152.

The inside of the cylinder in the rear of the face is lined with cast-iron segments, to which are fastened six hydraulic presses as shown in Fig. 154, one of these being shown in section in Fig. 153. These presses are supplied with water from pumps by pipes not shown in the illustrations.

When the material is excavated, the pumps are put into operation, and the rams of the presses force the shield forward. The rear end of the shield for a length of 2 feet 8 inches consists of the steel cylinder only, as shown in Fig. 153, and within it the tunnel lining, consisting of massive cast-iron segments, is put together; this cylinder being moved forward by admitting pressure water behind the pistons of the presses.

A proper door must be constructed in the face of the shield, and air locks provided where compressed air is used. The space between the segments and earth may be filled in with cement grouting, a special apparatus having been designed by Mr. Greathead for this purpose.

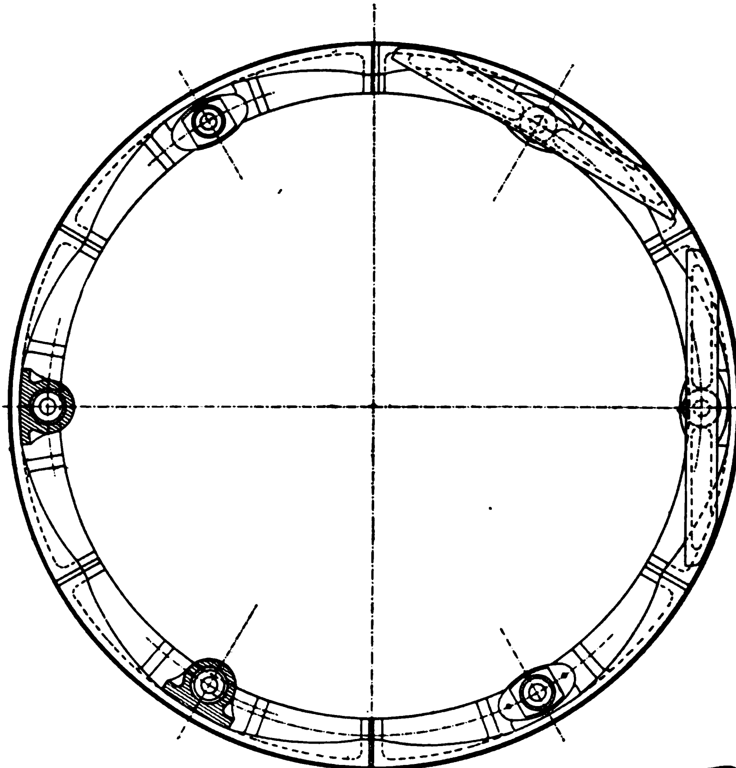


FIG. 154.

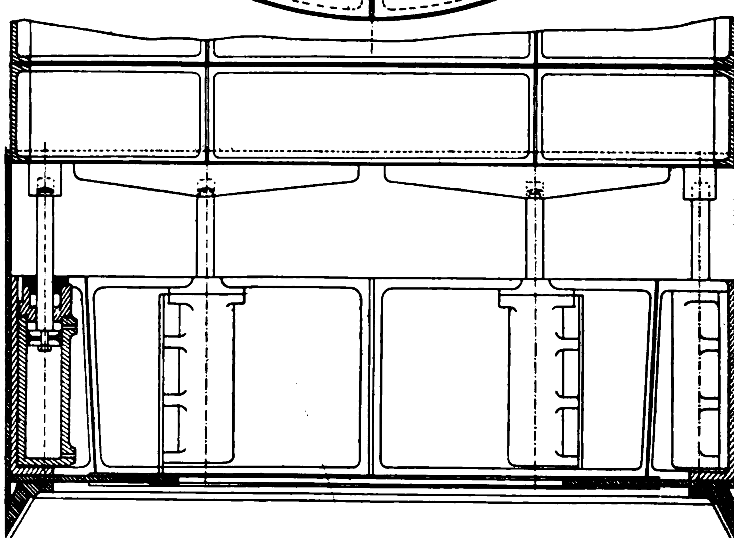


FIG. 153.

HYDRAULIC HOOP-TIGHTENING PRESS.

Hydraulic machinery has been used for various processes connected with cask-making, though to a somewhat limited extent. Fig. 155* shows a novel application of a hydraulic press to the tightening of the permanent hoops. The cask, with its hoops partly on, is placed on a table or platten over the ram of a hydraulic press. Above this table is a casting which carries a series of steel driving-arms, connected together in such a way that when one is pulled outwards they all open, and when released they fall together again, being weighted. The cask being placed in position, pressure water is admitted to the press, and the ram and table ascend: the driving-arms catching the hoops force them on tigntly. A relief valve is provided in the supply pipe, so that when the hoop becomes tight enough, and hence the pressure reaches a given intensity, the valve opens, and the relative motion of hoop and cask ceases. Each hoop is thus driven uniformly to the proper degree of tightness and with less risk of breaking the hoop.

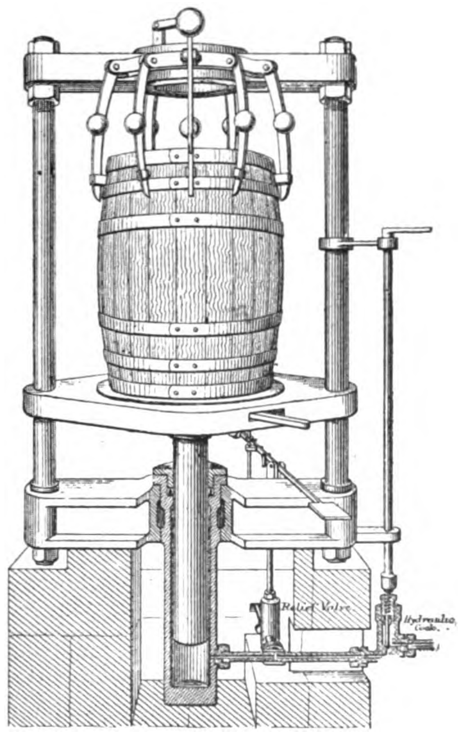


FIG. 155.

There are many other applications of the hydraulic press to which space does not permit a reference; those indicated seem to be of considerable interest, and are not described in the usual text-books.

* From the 'Minutes of Proceedings of the Institution of Civil Engineers,' vol. cxv., by permission.

XX.

HYDRAULIC TRANSMISSION OF POWER.

THE practical success of almost every class of appliance by which the resources of nature have been developed and man enriched, has been due largely to the genius and effort of one man. For instance, what Watt did for the steam engine, Benjamin Huntsman for crucible, and Henry Bessemer for mild steels, Lord Armstrong may be said to have done for hydraulic machinery for the storage and transmission of power.

It must not be imagined that such inventors in all cases began professional life as, or had the training of, engineers. They had, however, nature's endowment of inventive genius. Watt was a mathematical instrument maker when he hit on the great discovery of a separate condenser, and Armstrong was a solicitor when he set his inventive faculties to work to devise more efficient means than then existed for utilising great natural heads of water. In a letter to the 'Mechanics' Magazine' of 1840, he pointed out some of the advantages of water as a medium for the storage and transmission of power, and showed that if water were pumped by a steam engine into an elevated reservoir, a large portion of the potential energy thus given to the water could be obtained from it again. Further, a small engine pumping continuously could thus supply many machines working intermittently. After inventing a water-wheel for high falls, which, though efficient, never came into practical use, he invented the hydraulic crane, an apparatus which has ever since had his name associated with it. This machine is fully referred to in a succeeding section.

The first hydraulic crane was erected on Newcastle Quay in 1846, being served by water from the ordinary town mains. Cranes erected at Liverpool and elsewhere were similarly driven, but cases soon occurred in which the town supply was at insufficient or too variable pressure, and pumps were employed in conjunction with air-chambers to give the necessary pressure. This not proving satisfactory, a tank on a high tower—about 200 feet high—was erected to obtain the necessary head. In one case, in 1850, it was found impossible, except at great expense, owing to the treacherous nature of the foundations, to build a tower of the requisite height; hence Mr. Armstrong was compelled to adopt some other plan, and he invented the ***hydraulic accumulator***, an appliance which has, more than any other, contributed to the success of hydraulic systems of power transmission from central stations.

The Hydraulic Accumulator.

The action of the accumulator is very simple, and will be readily understood from a glance at the illustration. Fig. 156 shows one of the usual forms. It consists of a wrought-iron weight case W filled with slag, concrete, or ballast, this case resting on and being attached to a ram R, which can rise and fall in its press P, R passing water-tight through a gland and stuffing-box, as shown. The ram is attached to a cross-head B, which in turn is fastened to the weight-case. The press P is in communication with the pressure mains, so that when there is a demand for water, weight W is usually either rising or falling, means being provided to ensure that the ram shall always rest on water. The pumping engine can start in any position, and there is an automatic means of communication between the accumulator and engine, such as is shown in Fig. 157.

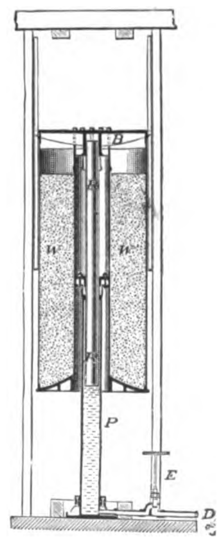


Fig. 156.

When the accumulator weight rises to the top of its stroke, it lifts weight A, which slackens the chain B B, allowing lever C D to fall, shutting off the steam by the butterfly valve D.

When W falls, the weight A raises lever C, opening the steam valve and starting the engine. The movable pulley is introduced at B, so that A may travel twice as far as C, thus ensuring a gradual stoppage of the engine, and enabling a small weight to be used; but this may be dispensed with, in which case it is best to have A greater than C.

If the accumulator should rise too high through the failure of valve D, or any part of the connection, the stop P—which consists of a plate with a hole in it through which the chain passes—catches a projection M on chain FR, and thus lifts the momentum valve T, allowing the water to escape from the pressure pipe, either to exhaust or to another accumulator weighted to give a lower pressure, so that the accumulator may not rise higher.

When the accumulator is too far from the engine for this method of connection, a hydraulic gear is used which consists of a small hydraulic cylinder in the engine room, the piston of which controls the starting and stopping gear of the engine. The valve for admitting pressure water to, or exhausting it from, this cylinder is fixed against the accumulator framing, and is operated by the weight case when near the top of its stroke. The valve is connected by a small pipe with the hydraulic cylinder referred to, and this pipe may be of any

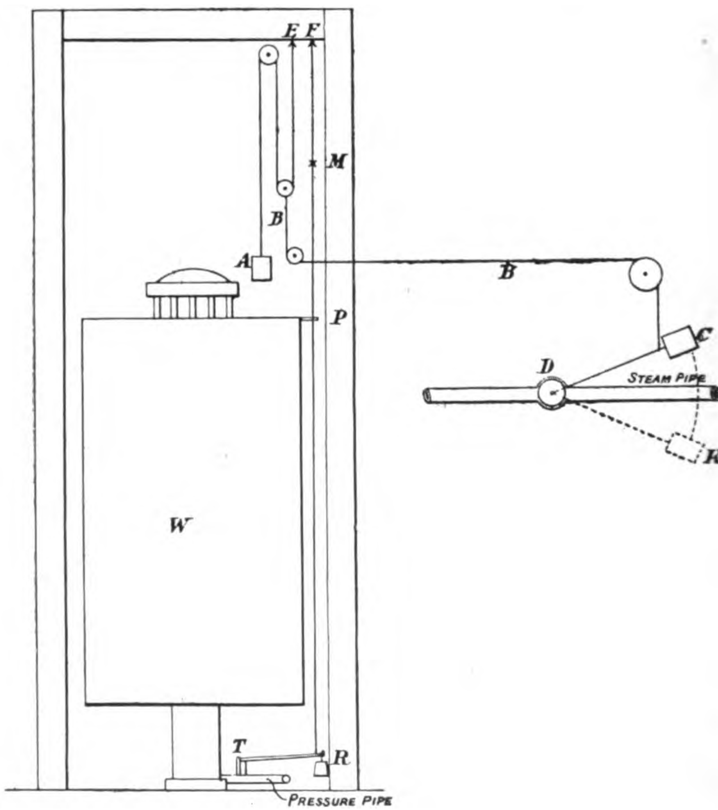


FIG. 157.

length. In the case of compound engines there is a special arrangement of valves for starting the engine, which is referred to in the description of pumping engines for accumulator work, given at p. 422.

An accumulator is a very efficient store-house for energy. Mr. Tweddell found that an accumulator charged at 1250 lbs. per square inch discharged at 1225 lbs. per square inch, friction being overcome by a pressure of $12\frac{1}{2}$ lbs. per square inch, 1 per cent. of the energy being wasted in charging, and 1 per cent. in discharging, or *the efficiency of the accumulator as a store-house for energy is* 98 *per cent.*

Storage Capacity of Accumulators.

The energy stored when the weight W lbs. is at the top of its stroke of h feet is W h ft.-lbs.

If the accumulator discharges at p lbs. per square inch, its usual storage is $62 \cdot 4 \times 2 \cdot 3\, p\, A\, h$ ft.-lbs., A being the area of the ram cross-section in square feet, and its capacity $\frac{62 \cdot 4 \times 2 \cdot 3}{33{,}000 \times 60} \times p\, A\, h$

$\frac{p\, A\, h}{13{,}800}$ horse-power hours, nearly.

An accumulator performs several useful functions. Thus it acts as a store-house for energy, but is useful more in the nature of a fly-wheel to level up inequalities of supply and demand than to provide a large store. Thus each large accumulator of the London Hydraulic Power Company stores $106 \times 2240 \times 23 = 5{,}461{,}120$ ft.-lbs., but since 1 horse-power for an hour requires 1,980,000 ft.-lbs., the accumulator only has a storage capacity of about 2·8 horse-power hours. It can, however, give out a great power for a short time, say, 160 horse-power for one minute. Compare even a number of these accumulators with a storage reservoir, such as that at Zurich, which contains 353,000 cubic feet at an elevation of 475 feet above the motors, having therefore a capacity of 5284 horse-power hours.

The accumulator acts as a pressure regulator, keeping the pressure in the mains constant and nearly equal (in lbs. per square inch) to $\frac{W}{a}$, where a is the area of the ram in square inches, W being in lbs. It provides the elastic arrangement necessary in such a system, for if all the machines in the system stop suddenly, the pumps merely raise the accumulator weight, whereas if no such arrangement were provided, something must be broken.

It also has the fourth use of providing an automatic system, the pumps being controlled by the accumulator as already described.

Differential Accumulator.

An accumulator of the differential type used by Mr. Tweddell is shown in Fig. 158. Here the weight contains a heavy press, in which is a ram A, which in this case *is fixed*, the weight and press rising and falling. The ram admits water through its centre; there are two glands, and the ram is continued right through the cylinder, being of smaller diameter above. Hence the weight may be regarded as giving pressure, not to the area of cross-section of ram, but to the *difference of cross-section* of the two parts of the ram.

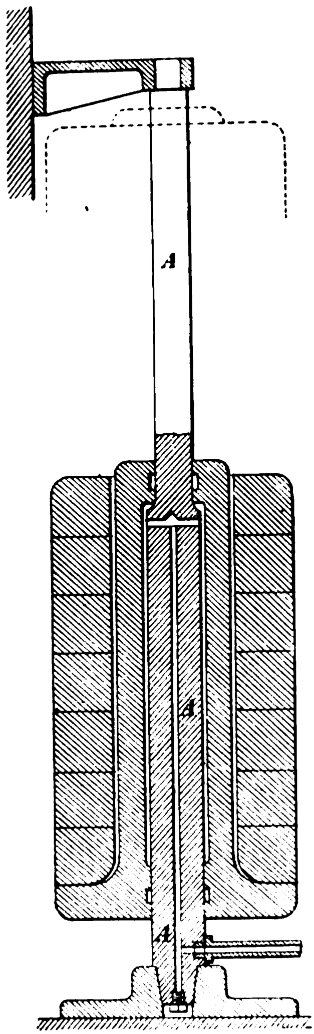


Fig. 158.

By this means a very great pressure may be obtained from a comparatively small weight, and a rigid guide is afforded for the weight to move on. This accumulator will store the same amount of energy as any other in which the weight and stroke are the same, the storage in any case being W × H ft.-lbs.; but since it contains only a small *volume* of water when the ram is at the top of its stroke, it will give out its store very quickly, and hence is used mainly with small machines.

Another form of differential or intensifying accumulator is shown in Fig. 159. A pressure of about 2000 lbs. per square inch was required in shops, where a supply at a very much smaller pressure was available. Mr. Tweddell allowed water from a tank to act on the under side of the piston C attached to the ram D, of much smaller area. The water in the press E thus had a greater pressure—neglecting friction—just in the ratio of the area of C to that of D. Water from the low pressure supply or pumps passes in at G, and is taken off at H, C being used merely as a regulator of pressure. The water acting on C is not wasted, but may be obtained from a tank, to which it is returned as C falls. If the supply pipe A is large, with easy bends, it is even possible to get a momentum effect which was wanted for riveting. The intensifier (see page 425) acts on a similar principle to this accumulator.

Hydraulic Power Supply.

Systems for the supply of hydraulic power from central stations have developed rapidly within recent years. London, Hull, Liverpool, Birmingham, Melbourne, Sydney, Antwerp, and Glasgow have now central station hydraulic supply systems, but London is the most important and extensive example in the world; hence it may be sufficient here to refer briefly to London as a typical case.

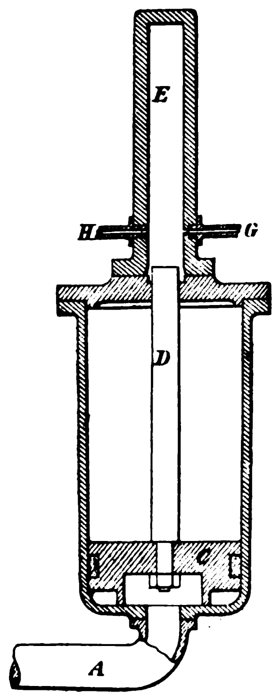


Fig. 159.

The London Hydraulic Power Company's works were established in 1884, and have now 150 miles of mains laid in the streets, the internal diameter of the largest main being 7 inches, and the pressure in the mains 750 lbs. per square inch. At Manchester and Glasgow the pressure is 1120 lbs. per square inch.

The success of hydraulic power cooperation in London—and indeed in some other towns—is due largely to the engineer, Mr. E. B. Ellington, whose name is mentioned later on in connection with hydraulic balances for lifts.

The London Company have several pumping stations, as it has been found that a station capable of giving out about 1200 horse-power is most economical, both in capital expenditure and working expenses.

The stations are situated at Falcon Wharf, Blackfriars; at Millbank, near the Houses of Parliament; at Wapping; and at Wharf Road, City Road, with a small accumulator station at Philip Lane, E.C. A small accumulator station also exists at Kensington Court, where accumulators, loaded to 450 lbs. per square inch, are worked from the mains.

Engines.

In describing very briefly some of the details of the system which has been so successfully adopted in London, the engines, pumps and boilers may be first referred to. Figs. 160 and 161 show a side and front elevation of Mr. Ellington's engines, made by the Hydraulic Engineering Co., of Chester. Each engine has three inverted cylinders, the piston of each cylinder working directly a single-acting pump.

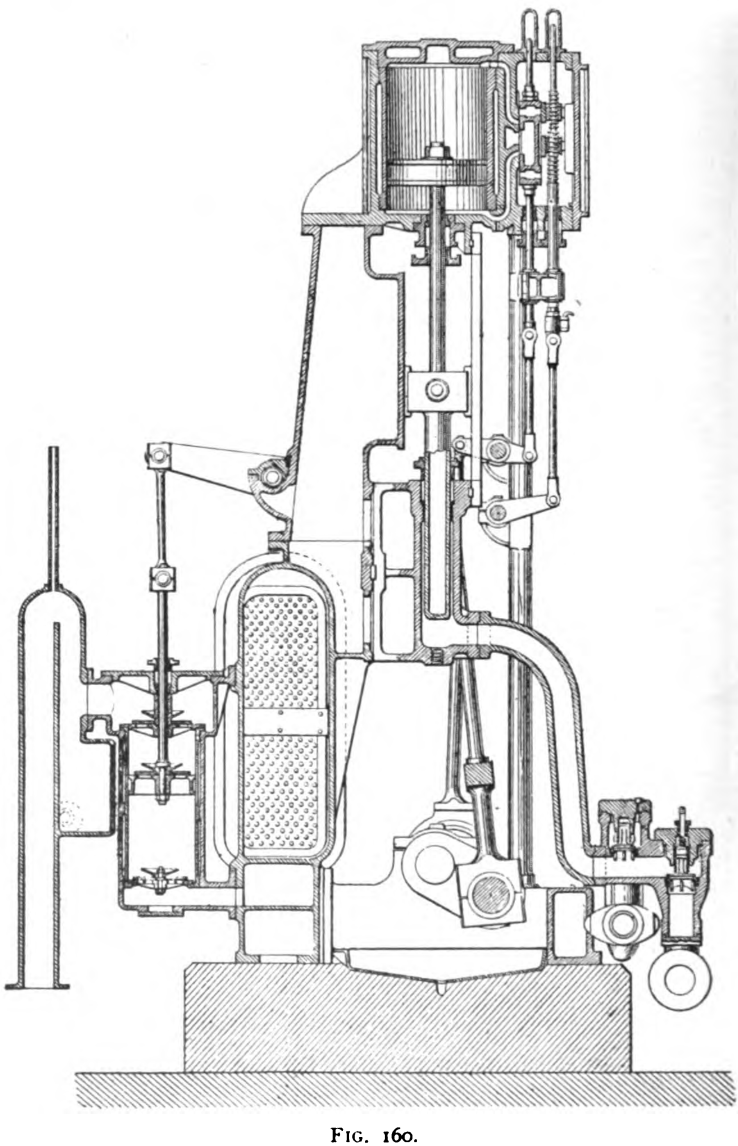


Fig. 160.

The connecting-rods span the pump barrels, and work on crank-pins on a three-throw crank shaft, the cranks being set at angles of 120°; there is thus no dead point in the stroke, and there is good

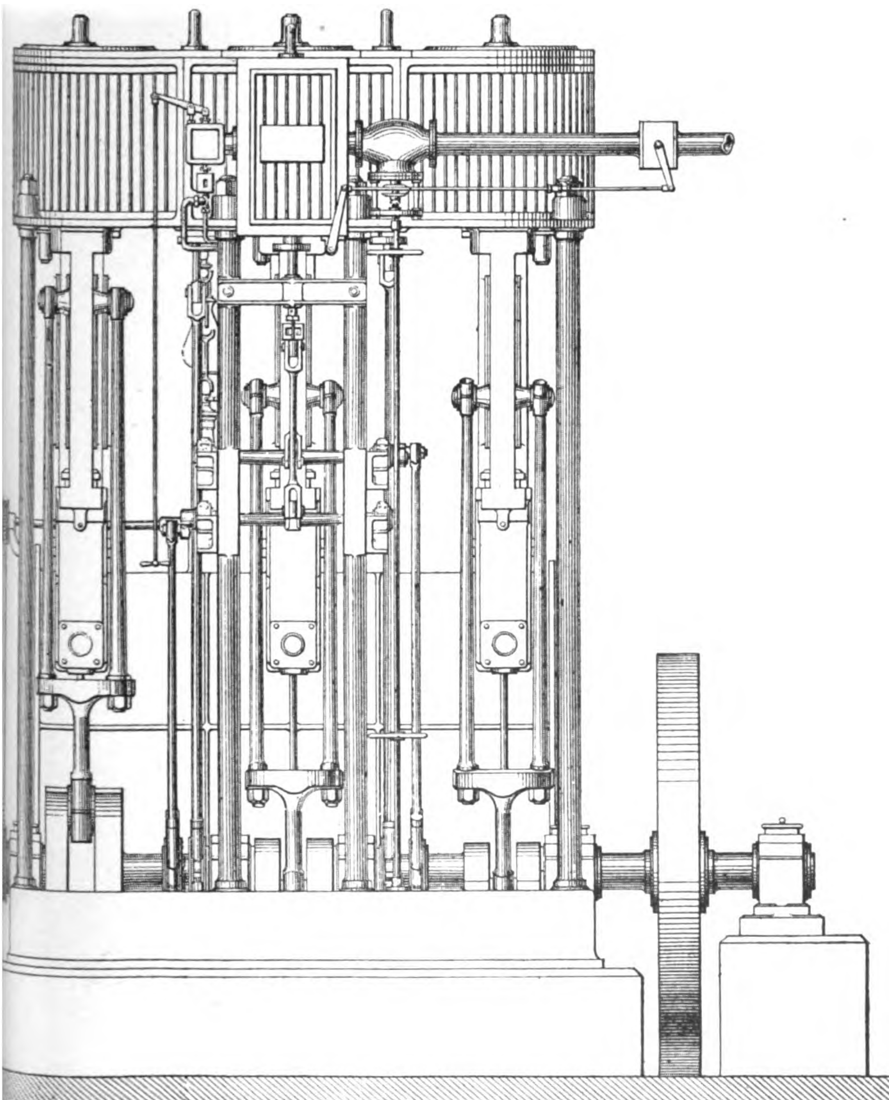


FIG. 161.

balance of reciprocating parts. The high-pressure cylinder has a variable expansion valve on the back of the main slide, adjustable whilst the engine is running, thus enabling the point of cut-off to be varied to suit the work and steam pressure.

All the cylinders are steam-jacketed and provided with connections which allow the water of condensation to be returned to the boilers by gravitation. (The student is not to suppose that water can usually be forced *into* the boilers by gravitation—it is, however, in this case, as both the steam and return water pipes are under the boiler pressure.)

The cylinders are carried on four steel columns at the front, and three cast-iron columns at the back, incorporated with and forming part of the condenser. The pumps are attached to the cast-iron columns and stayed to the cylinders by steel bars, which also act as front guides for the engine cross-heads.

The air, circulating, and feed pumps are placed behind the condenser, and worked from the piston of the intermediate cylinder by rocking levers and links.

The engine is controlled by a high-speed governor; automatic gear is also provided by which the engine is controlled by the accumulator, somewhat in the way already described.

At a nine-hours' trial of a set of these engines at the Wapping station, with high-pressure cylinder 15 inches, intermediate cylinder 22 inches, and low-pressure 36 inches in diameter, all 2 feet stroke, 160,880 gallons of water were pumped against an accumulator pressure of 795 lbs. per square inch. The indicated horse-power of the engines was 206·55, steam consumption 15·26 lbs. per horse-power per hour, and coal consumption 1·49 lbs. per indicated horse-power per hour.

Of this steam consumption 1·16 lbs. per indicated horse-power per hour were accounted for by steam-pipe condensation and loss. The coal used contained 8½ per cent. of clinker and ash.

The London Hydraulic Power Company have nineteen sets of engines of this type at work. They are very compact, and it will be seen from the results given above that the efficiency is high.

Another type of engine—made by the same firm—is shown in elevation and plan in Figs. 162 and 163, and is the engine adopted at the Hull Docks to supply water for the mains of the Dock machinery, the pressure being 800 lbs. per square inch. This, as will be seen, is a horizontal engine with two high-pressure cylinders 20 inches in diameter, and two low-pressure cylinders 38 inches in diameter, arranged tandem fashion, with pumps of the piston and

ELEVATION.

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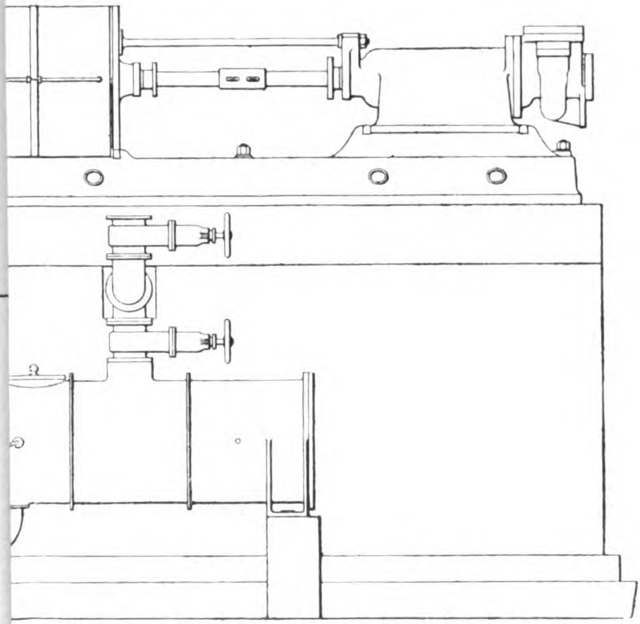
SCALE



PLAN.

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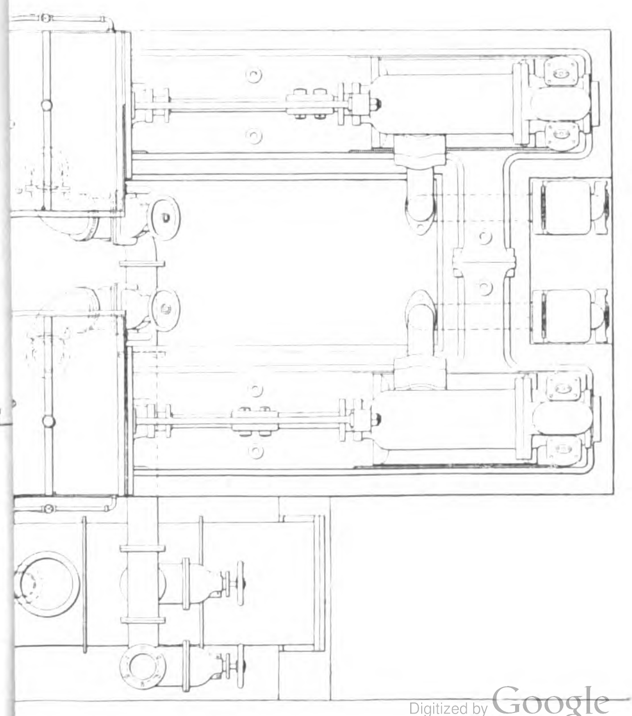

[Face page 240.

plunger type $7\frac{1}{2}$ inches in diameter, placed behind the cylinders and worked direct from the piston rods, the stroke being 3 feet 2 inches.

The high-pressure cylinders have variable expansion valves on the back of the main slides, adjustable whilst the engine is running. The low-pressure cylinders are steam-jacketed. The pumps are fitted with double suction and delivery valves and separate pump barrels, arranged to be interchangeable, and so that they can be removed and replaced in a short time. The steps of the crank shaft are of the four-part type, adjustable both vertically and horizontally. The condenser is of the *surface* type, circular in form, and fitted with brass tube plates and tubes $\frac{3}{4}$ inch diameter, through which the circulation water is passed by a separate pumping engine. The air-pump is of the single-acting type, and is worked from the left-hand crank-pin. From the right-hand crank-pin a pump is worked to return the water after use in the hydraulic machinery to the supply tank.

These engines deliver 400 gallons of water per minute against an accumulator pressure of 800 lbs. per square inch, the steam pressure being 80 lbs. per square inch. The engines are strong and massive in design, each weighing about 75 tons. Breakdowns are practically unknown.

These illustrations, and those on pages 238 and 239, give a very good idea of the types of engines most in use in hydraulic supply stations.

Boilers.

The boilers need not be fully described here. At Wapping, the Fairbairn-Beely boiler is adopted. This is a double boiler, like two Cornish boilers one above the other, the lower one containing a large flue and a number of small tubes, the upper one containing part of the water and all the steam space. At Millbank, Lancashire boilers are used, a Green's economiser, consisting of 96 tubes, being fitted at the back of the boilers, to utilise the heat of the waste gases of combustion in warming the feed-water. Vicars' mechanical stokers are employed, the hoppers being kept full by an attendant, who fills a hopper from which the coal is distributed by a "creeper," consisting of a trough and worm. The economiser and stoking-gear are driven by a Brotherhood three-cylinder hydraulic engine (more fully described at page 320).

It has been found, from careful trials, that at Wapping the average thermal efficiency of the engines was 15·25 per cent., that of the boiler and economiser 78·2 per cent., rising to 86·7 per cent. at full power. Similar trials show 12·24 per cent. and 78·7 per cent. respectively for the Millbank boilers.*

ACCUMULATORS.

It is not necessary to give a separate picture of the accumulators, as that shown at Fig. 156, though not exactly like those used in the London stations, gives a good idea of the general features of such an apparatus. The accumulators used in London have wrought-iron cases filled with iron slag. There are two at Falcon Wharf and two at Wapping Station, each with a ram of 20 inches diameter and 23 feet stroke; one at Millbank 18 inches diameter and 20 feet stroke, and a similar one at Philip Lane. The total weight of the larger accumulator load is 106 tons, but the load can be increased so as to give a pressure of 800 lbs. per square inch. The capacity of these accumulators is 1600 gallons, whilst the pumping plant can supply 3500 gallons per minute, showing that the accumulators act mainly as regulators of pressure, their power-storage capacity being insignificant.

SUPPLY FOR THE PUMPS.

The main portion of the water used at Falcon Wharf is taken from the Thames, headings having been driven under the bed of the river to ensure a sufficient supply at all states of the tide. The water after precipitation is filtered in tanks containing cast iron perforated plates and charcoal filter beds. At Millbank and Wapping wells supply the water, the Porter-Clark process of filtering and softening being used at the former station. The cost is about 1*d.* per 1000 gallons treated.

Variations in the demand for power are met mainly at Falcon Wharf.

DUTY OF ENGINES.

It may be of interest to point out that a high duty is obtained from the engines. Thus on a certain date 7000 gallons were pumped, to a pressure of 800 lbs. per square inch, per hundredweight of coal burnt. This gives as duty $7000 \times 10 \times 800 \times 2 \cdot 3 = 128{,}800{,}000$ ft.-lbs. An average figure seems to be 5500 gallons, equivalent to a duty of 101,200,000, which is very good considering the intermittent character of the pumping.

* For further details consult the report of Mr. Ellington's paper on "Hydraulic Power Supply in London," 'Minutes of Proceedings of the Institution of Civil Engineers,' vol. cxv.

COST OF POWER.

This is a most important matter and has been very fully gone into by Mr. Ellington. The cost of production depends largely on the *load-factor*, that is, *the ratio of the average output per hour to the maximum output in any one hour.* The annual load factor for 1904 was 0·3715, and the heaviest day load-factor for the same year 0·494. The actual station cost of 1000 gallons at a pressure of 750 lbs. per square inch increased from 5·172*d.* in 1893 to 5·696*d.* in 1904, owing mainly to the increased price of coal. Mr. Ellington some years ago made a comparison which showed that the station cost of an equivalent amount of electric energy as produced by the Westminster Electric Supply Corporation was 9·014*d.*; one Board of Trade unit or its equivalent costing 1·383*d.* electrically and 0·793*d.* hydraulically. Of course, mere station cost of production is a small item in the actual cost of power to the consumer. The London Hydraulic Power Company pumped 878 million gallons in 1904, and supply power at the price of from 2*s.* to 1*s.* 6*d.* per 1000 gallons, the latter being equivalent to 2·76*d.* per Board of Trade unit, or about 2·07*d.* per brake horse-power per hour, obtained from hydraulic motors running fairly regularly during several hours per day. Electric companies charge from 2*d.* to 6*d.* per Board of Trade unit. This comparatively low cost of power has led Mr. Ellington to try whether hydraulic power might not be advantageously used for the production of electric current. Pelton wheels—which are probably better suited to the high pressure of hydraulic mains than any other motor—driven by the water from the mains, were employed to drive dynamos, about 66 per cent. of the hydraulic energy being realisable electrically, the cost being, at 3*d.* per Board of Trade unit for pressure water, $\frac{100}{66} \times 3 = 4 \cdot 5d.$ per Board of Trade unit electrically developed.

This current, if used to drive electric motors of 70 per cent. efficiency, would work out at nearly 6½*d.* per brake horse-power per hour. There seems to be some room here for the utilisation of hydraulic power, especially for electric lighting in suitable small installations. It would of course be much cheaper, for power purposes, to use the hydraulic power direct in hydraulic motors, but cases may occur in which electric motors, owing to their high speeds and ease with which they can be moved about, may be preferable.

There has been considerable discussion as to the reasons of the small cost of hydraulic power in comparison with electric; the *small amount of leakage* (over 90 per cent. of all the water pumped being usually accounted for), the high efficiency of hydraulic accumulators, both as store-houses of energy and as pressure regulators, and the comparatively high load-factors, are, in the author's opinion, the chief reasons of the comparatively favourable results.

Given the pressure or "head" of the supply and the cost of a given volume of the water, it is sometimes useful to be able, readily, to calculate the cost of 1 horse-power hour in water actually supplied. It will readily be seen that this cost is

$$\frac{c \times 198}{\text{head in feet}}, \quad \text{or} \quad \frac{c \times 86}{\text{pressure per square inch}},$$

where c is the cost of 1000 gallons. If c is in pence the answer will also be in pence. Mr. Ellington thinks it is not possible, profitably, to supply power thus, for less than *2d.* per horse-power hour, but it should be mentioned that the power is now supplied in London, in some cases, for *1s. 3d.* per 1000 gallons, being at the rate of 1·72*d.* per horse-power hour.

EXAMPLES.

1. At Southampton it has been proposed to use the ordinary town supply for power purposes. If the "head" in the lower parts of the town is 138 feet, and the proposed charge 3*d.* per 1000 gallons, find the cost per horse-power hour. *Ans.* 4*d.* nearly.

2. Compare this with the cost in London where the pressure is 700 lbs. per square inch and charge 1*s.* 6*d.* per 1000 gallons.

Ans. Cost 2·2*d.* per horse-power hour in London. Comparative cost $\frac{\text{Southampton}}{\text{London}} = \frac{1 \cdot 93}{1}$.

3. What will be the charge per 1000 gallons in Glasgow where the pressure is 1120 lbs. per square inch, if the consumer is to have power at the same rate as in London? *Ans.* 28·6*d.*

PIPES AND PIPE JOINTS.

The pipes usually employed are of cast iron, 6 inches internal diameter being the largest size formerly used, but pipes of 7 inches diameter are now employed. Of course, pipes to stand a pressure of 750 lbs. per square inch, or the greater pressure 1120 lbs. per square inch employed at Glasgow and Manchester, require to be very thick. The diagram given at A, Fig. 164, shows for a pipe 6 inches in internal diameter, the increase of thickness as the pressure is increased. The question of the strength of thick pipes is one of considerable interest.

STRENGTH OF PIPES.

In calculating the strength of a thick pipe like a hydraulic main, it will not do to assume—as we do in the case of a pipe, the thickness of which is small in comparison with its diameter—that the stress is the same all across a section of the pipe. The stress is evidently greater near the inside of the metal than at a point further

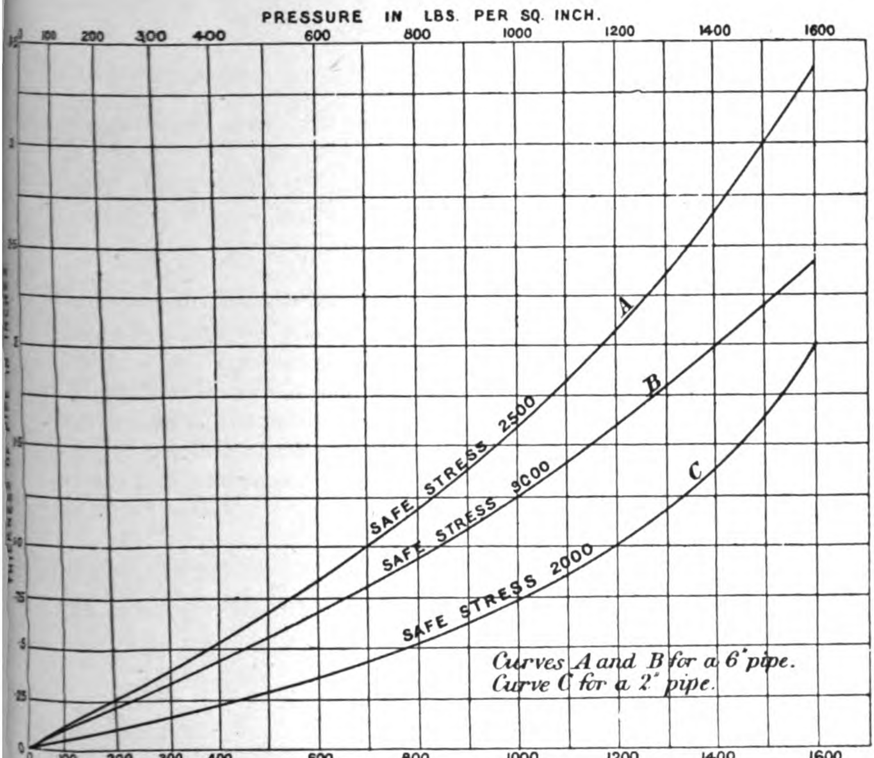

FIG. 164.

out; the exact law by which the stress falls off is not accurately known, but the following rule seems to give results best in accord with actual facts. The proof of this rule, which Professor Perry has worked out for his classes, is as follows:—Let (as in Fig. 165) r_1 = inside and r_2 = outside radius of cylinder, p_1 the intensity of pressure inside, and p_2 that outside the cylinder. Let the circles near together represent a very thin ring of metal of unit length, whose internal radius is r and outside radius $r + \delta r$, corresponding pressures being p and $p + \delta p$ respectively. Consider the strength of this ring, p_1 being greater than p_2.

The upper half of the ring tends to fly away from the lower half. $2 rp - 2(r + \delta r)(p + \delta p)$ is the total upward pressure acting on the ring, and $2 \delta r f$ is the resistance offered by the metal, f being the tensile stress in it. Hence, dividing by 2 all across,

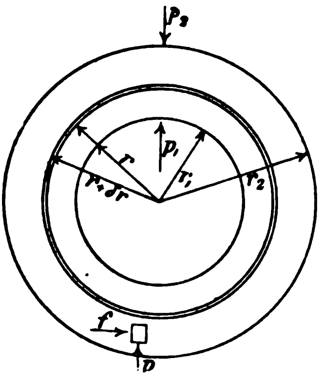

FIG. 165.

$$rp - (r + \delta r)(p + \delta p) = \delta r f.$$

By making the ring of metal thinner and thinner we may neglect $\delta r \,.\, \delta p$ in comparison with other terms, hence transposing we have

$$(f + p)\,.\,\delta r + r \delta p = 0 \quad (1)$$

The metal is acted on by crushing and tensile stresses. On account of the crushing stress of p lbs. per square inch, there is a lengthening at right angles to the paper of $p b$,* and similarly a shortening of amount $f b$ at right angles to the paper, on account of the tensile stress f lbs. per square inch; hence the total lengthening in a direction at right angles to the plane of the paper is $p b - f b$.

Assuming that a plane section remains plane,

$$p b - f b = \text{a constant};$$

$$\therefore \quad p - f = \text{a constant } c,$$

or

$$(2) \quad f = p - c.$$

Substituting this value of f in equation (1), we have

$$(2p - c)\,\delta r + r \delta p = 0,$$

or

$$\frac{\delta r}{r} + \frac{\delta p}{2p - c} = 0.$$

* b is the lateral strain corresponding to a longitudinal strain $a \left(= \frac{1}{E}\right)$ or a longitudinal stress of 1 lb. per square inch.

Allowing our increments of radius and pressure to diminish indefinitely,

$$\int \frac{dr}{r} + \int \frac{dp}{2p - c} = \text{a constant},$$

whence

$$\log r + \tfrac{1}{2} \log (2p - c) = \text{a constant},$$

or

$$2 \log r + \log (2p - c) = \text{a constant},$$

or

$$\log r^2 + \log (2p - c) = \text{a constant}.$$

$$\therefore \quad \log \{r^2 (2p - c)\} = \text{a constant},$$

or

$$2p - c = \frac{\text{a constant}}{r^2}.$$

Let K = this constant ÷ 2.

$$(3) \quad \therefore \quad p = \frac{K}{r^2} + \frac{c}{2},$$

and

$$(4) \quad f = \frac{K}{r_2} - \frac{c}{2},$$

since $p - f = c$.

When $\qquad r = r_1, \qquad p = p_1,$

and when $\qquad r = r_2, \qquad p = p_2;$

therefore, putting in these values and getting K and $\frac{c}{2}$ in terms of r_1, r_2, p_1 and p_2, we have

$$(5) \quad f = \frac{p_1 (r_2^2 + r_1^2) - 2 p_2 r_2^2}{r_2^2 - r_1^2}.$$

This is the general law, but if we neglect the outside pressure, as we may usually do in comparison with the great internal pressure, the rule becomes

$$(6) \quad f = \frac{p_1 (r_2^2 + r_1^2)}{r_2^2 - r_1^2},$$

or the internal pressure (per square inch) multiplied by the sum of the squares of the internal and external radii or diameters is equal to the tensile stress the material will stand multiplied by the difference of the squares of the same two radii or diameters. This is the well-known rule which is usually employed for calculating the strength of thick pipes.

For calculation purposes the rule is often more convenient if written in the form

$$D = d\sqrt{\frac{f+p}{f-p}};$$

D being the external and d the internal diameter. The student will see that the thickness $= \frac{D - d}{2}$.

We have found that a 6-inch pipe designed for a pressure of 800 lbs. per square inch is usually $1\frac{1}{8}$ inch thick, corresponding to a safe stress of 2596 lbs. per square inch. For a pressure of 1120 lbs. per square

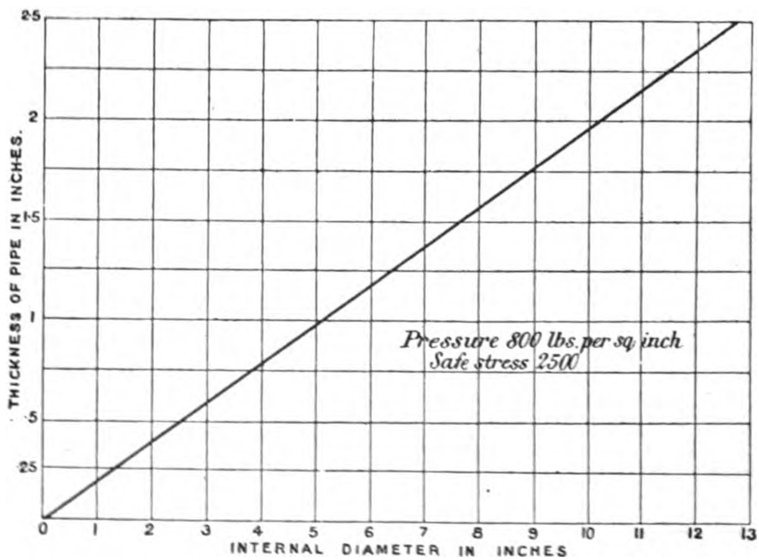

FIG. 166.

inch the thickness is $1\frac{3}{8}$ inch, agreeing with a stress of 2912 lbs. per square inch. Small pipes are made thicker to allow for defects at joints, corrosion, etc., thus a safe stress of about 2000 lbs. per square inch is considered best for pipes 2 inches in internal diameter. Results worked out from these data are shown in the convenient form of curves (Fig. 164), giving the connection between thickness of pipe and pressure of water in each case.

To complete the series the curve shown in Fig. 166 has been plotted. This shows the connection between thickness and internal diameter for pipes to withstand a pressure of 800 lbs. per square inch, taking 2500 as the safe stress. That figure gives about 6 as a factor of safety. These curves enable any one at a glance to determine the proper thickness of metal for a hydraulic main when the internal diameter and pressure are known. The conditions which determine what the internal diameter ought to be are discussed at page 451. For a given pressure the horse-power transmitted through the pipe is the factor which determines the diameter of the pipe. The curves show that for higher pressures a great increase of thickness is required for a given increase of pressure; in fact, as the pressure increases we soon reach a point (when $f = p$) at which no amount of increase in thickness will make the pipe strong enough. This points to some other material than cast iron as the proper one for mains to withstand great pressures, in fact steel pipes from 5 to 12 inches in diameter have been used at Antwerp to convey water at a pressure of 750 lbs. per square inch. Those pipes have stamped steel flanges.

Pipe Joints.

The joints used by Mr. Ellington at London, and elsewhere, are shown in side elevation and section in the left-hand portion, and in end elevation in the right-hand portion of Fig. 167.

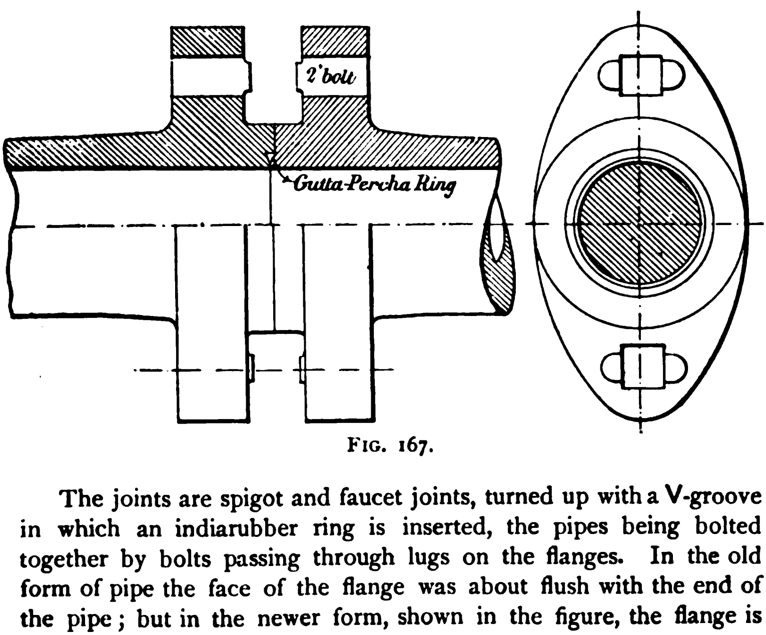

Fig. 167.

The joints are spigot and faucet joints, turned up with a V-groove in which an indiarubber ring is inserted, the pipes being bolted together by bolts passing through lugs on the flanges. In the old form of pipe the face of the flange was about flush with the end of the pipe; but in the newer form, shown in the figure, the flange is set back some distance from the end. It has been found that this has increased the strength at least 35 per cent., and only one failure of flange has occurred since this form was introduced, whereas with the old form of flange failures were not uncommon.

With this form of joint, leakage is almost unknown, notwithstanding the high pressures used.

Pressure Due to Shock.

If we have a column of water moving with a given velocity along a pipe, a very important problem is to find the pressure produced by the sudden stoppage of the water. This problem has been worked out by Professor Perry, and, without troubling the reader with the mathematics, we give the very important result arrived at by the investigation.

Given a column of fluid of density ρ_o grammes per cubic centimetre, moving with a velocity of V_o centimetres per second, on sudden stoppage there is a wave of compression travelling along with the velocity of sound in the fluid. The increase of pressure, from the stopped end to the front of the wave, is f dynes per square centimetre, where f is obtained by the law

$$f = V_o K^{\frac{1}{2}} \rho_o^{\frac{1}{2}}.$$

K is the modulus of cubic elasticity $= 2 \times 10^{10}$ dynes per square centimetre, and $\rho_o = 1$ gramme per cubic centimetre for water.

The law expressed in British units is

$$p = V \times 62 \cdot 46$$

p being in lbs. per square inch, and V in feet per second; or if K be taken as 300,000 lbs. per square inch, the law is $p = 63 \cdot 5$ V.

In this rule the pipe is supposed to be perfectly rigid.

In practice the "ram" pressure never rises so high as that given by the above rule, because it is impossible to stop the flow with infinite rapidity, and also because the pipe stretches a little.

Mr. Weston has made some useful experiments on the actual pressures produced in pipes of various diameters by stopping the flow in about 0·15 of a second.

A series of experiments were carried out by the author at the Technical College, Finsbury, with the object of determining how the "ram" pressure varied with rate of stoppage.

The paper barrel of a Crosby steam-engine indicator fixed on the pipe was driven at a constant speed, and a valve in the pipe beyond the indicator was closed at different rates, the pencil of the indicator in each case drawing a curve, of which ordinates showed pressure in pipe and abscissæ time taken to close the valve. The pressure was found to rise slowly at first and then with increasing rapidity as the valve became more and more nearly closed, the maximum pressure being in each case nearly proportional to rate of closing. The time occupied in closing the valve varied from half a second to $\frac{1}{80}$ of a second, the ram pressures varying from 0 to 50 lbs. per square inch; but the experiments are not yet sufficiently complete to warrant the formulation of a law. Important experiments by Professor Goodman are referred to on page 414.

XXI.

HYDRAULIC CRANES.

Lord Armstrong (then Mr. Armstrong), the pioneer in the designing of high-pressure hydraulic power machinery, soon saw the peculiar fitness of pressure-water for working cranes. Being fully aware that this motive power is best adapted for giving a slow steady motion, he devised a method of lifting the load at one stroke of a ram or piston, multiplying the motion sufficiently by means of pulleys. The arrangement adopted by him, and still employed, is like an ordinary pulley tackle worked the reverse way—in other words, the ram acts where, in pulleys, the weight is usually attached, the load on the crane occupying the place of the usual applied force. Thus it is necessary to apply to the ram or piston of the hydraulic crane a force four, six, or eight times that due to the weight of the load raised, but the ram moves correspondingly slower.

A crane was designed by Mr. Armstrong on these lines, with an additional cylinder for slewing or turning the crane and load, the piston of this cylinder carrying a rack working into a spur wheel on the base of the crane. By the courtesy of Lord Armstrong's firm we are able to give, in Fig. 168, a correct illustration of the crane. The reader will be struck with the excellence of the design, made fifty years ago, and scarcely surpassed even now.

It will be noticed that there are *three* working cylinders, one, two, or all of which may be used at will, thus altering the power of the crane, and the consumption of water, to suit the load.

Every well-designed machine of this kind is provided with a relie valve, to prevent injury to the cylinders or mechanism if the momen

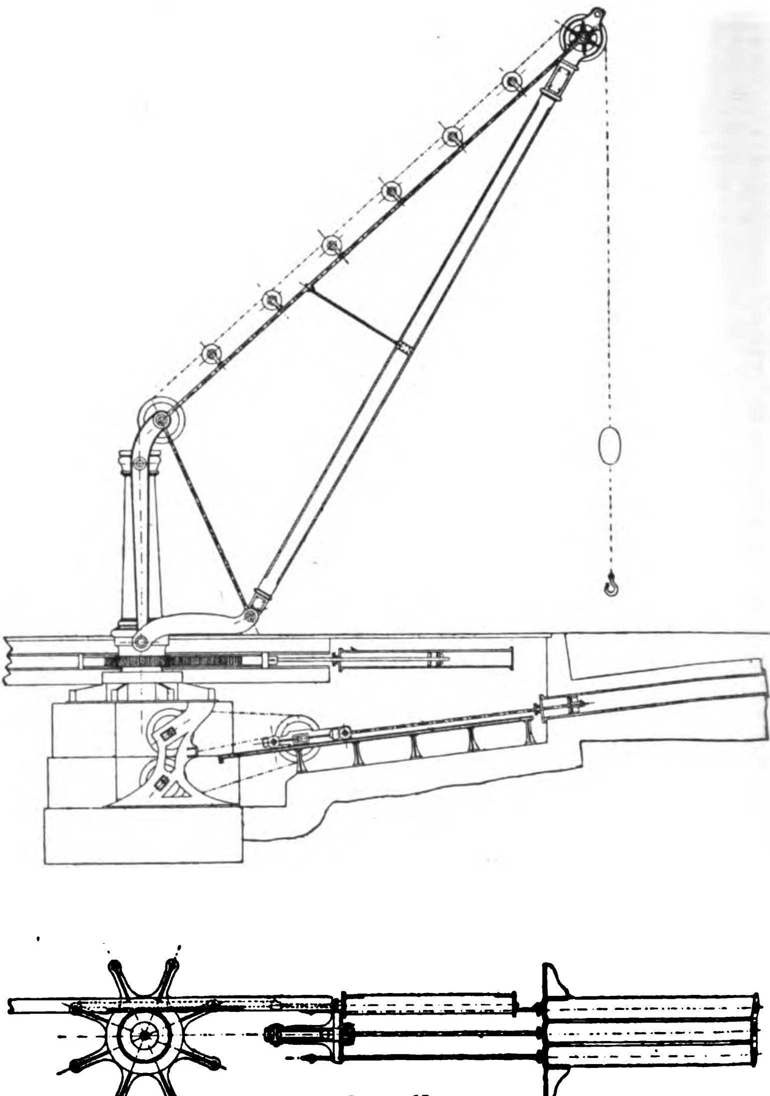

FIG. 168.

tum of a moving load be suddenly resisted. Thus, if the load on a crane is being rapidly lowered, and the exhaust valve is suddenly closed, the momentum of the moving parts, resisted by unyielding water, is likely to cause severe shock, unless some means be provided by which water can escape as soon as the pressure exceeds a given limit. Fig. 169 shows diagrammatically how this can be accomplished in the case of a crane. S is the supply pipe through which the pressure water passes to the lifting cylinder C; M is the regulating valve admitting the supply when necessary, and R the exhaust valve. M is usually closed, when R is opened. If R be suddenly closed, the inertia of the moving parts prevents them from coming immediately to rest, but when the pressure exceeds a certain limit, the relief valve K, which is weighted so as to open only at or above that pressure, opens, and some water finds its way back to S through the by-pass F. The actual arrangement adopted by Armstrong in cranes in which the regulating valve is a slide valve, is shown in Fig. 170*, where spaces M M communicate with the supply pipe S, and E E with the exhaust R. When the slide valve is moved in the direction of the arrow, the pressure supply is first cut off from the port Y by the lap of the valve, the port X being still open to exhaust. At this instant the flap valve

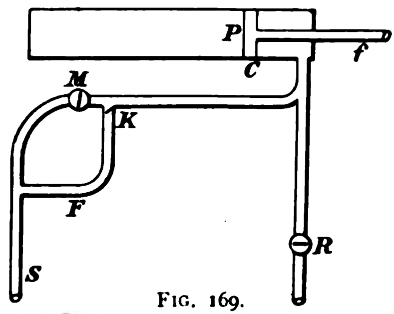

FIG. 169.

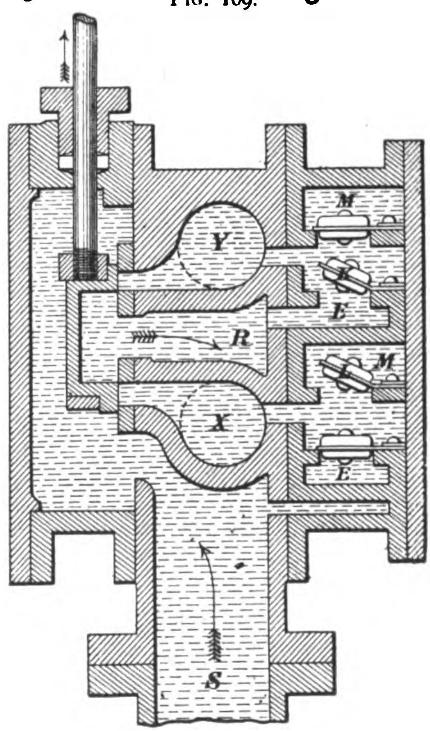

FIG. 170.

* From Proceedings, Inst.M.E., 1858.

K opens upwards and allows a small quantity of water to pass fro the exhaust port R into Y, to follow up the ram until brought to re When the valve arrives at the mid-position, as shown in the figu X is closed to exhaust, and the pressure in X being further increas by the motion of the ram before it is completely stopped, the secon relief valve L opens and a small quantity of water passes by M int S. When the slide valve moves the opposite way, the two remainin relief valves come into action in the same way. This gives perfe control of the crane, with freedom from shocks. This crane (Fig 168), though thoroughly tested and found efficient, was not erecte publicly for some time.

In 1846, one of these cranes was erected on Newcastle Quay, an excited great interest, engineers coming from all parts to see it, amon others the engineer of the Liverpool Docks, who ordered some crane of a similar kind for Liverpool, and the great system of hydrauli dock cranage was successfully started. In 1850 cases were met with in which the pressure of the ordinary town supply was insufficient, or of too variable a nature to be relied on, and the expense of erecting towers with elevated tanks to supply the head necessary would have been too great, owing to the nature of the foundations; hence steam pumps were used in conjunction with air chambers to give the necessary pressure. The air vessels not proving a very good arrangement, Mr. Armstrong devised the hydraulic accumulator—one of the most important features in a hydraulic supply system. Hydraulic cranes have now come into common use for dock, railway, warehouse and other purposes. The first hydraulic crane used for railway station work was erected at Newcastle, on the North-Eastern Railway, in 1848.

Railway Station Cranes.

Fig. 171 shows a small hydraulic crane, such as is used at large railway stations. It is capable of lifting 30 cwt.

The lifting cylinder here is in the centre of the revolving pillar or crane post, which is turned by a pair of hydraulic plungers, with multiplying sheaves and chains attached to a drum on the bottom of the pillar. The lifting, lowering and turning valves are placed side by side, the pressure water being conveyed to the lifting cylinder through an oscillating joint in the bottom socket of the pillar. This crane is used where the building in which it is placed allows of the top socket in which the pillar turns being attached to the roof or an overhead floor. The next illustration, on the other hand, is that of a crane which is independent of such support. In some cranes the lifting cylinder itself forms the pillar, but even for light cranes it is more usual, now, to construct the pillar of rolled steel or iron, and to place the cylinder within it, as shown in the figure.

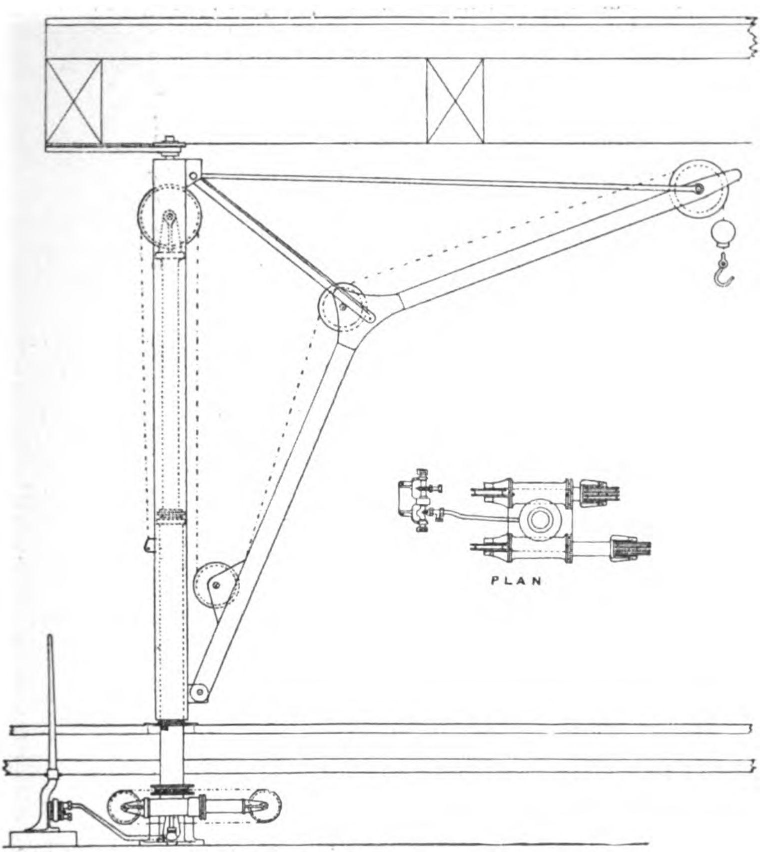

FIG. 171.

The crane shown in Fig. 172 is of the older type, the ram being in the centre of the crane post, and when it is forced out of its cylinder by pressure water from the hydraulic mains, it pulls in the chain to which the load is attached. There being one fixed and two movable pulleys, the velocity ratio of this machine is 4 to 1. The rams for slewing are seen at the base of the crane post.

Dock or Quay Cranes.

Cranes for loading or discharging cargo are now usually of the movable type, this type possessing obvious advantages over fixed cranes. It must be remembered that rapidity of loading and unloading are of first importance in quay and dock work, and with movable cranes, four or five can be brought to bear on one vessel, being placed in positions to suit the several hatches. For ordinary loading purposes a crane capable of lifting from 30 cwt. to 2 tons is found

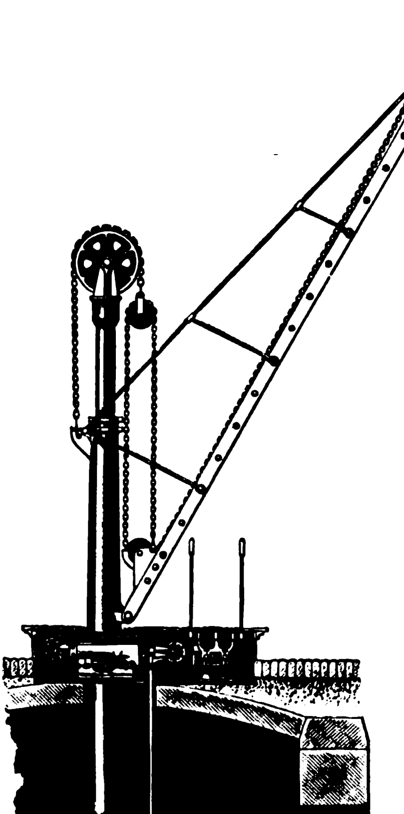

Fig. 172.

to be most convenient, and many patterns of cranes of this power are used. Views of some of these are shown in Figs. 173 and 174, the lifting cylinder being within the pillar in each case. The multiplying power of the sheaves is usually 6 to 1, the length of the lift being 50 to 60 feet.

The turning machinery is in the pedestal at the foot of the pillar, and is similar to that of the cranes already described, the turning

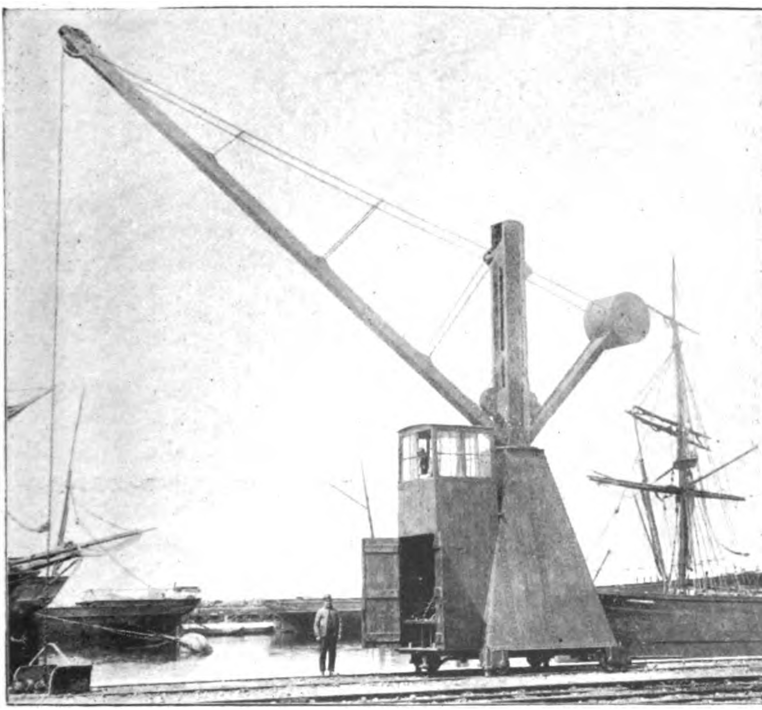

FIG. 173.

cylinders, however, being of greater power on account of the greater rake or radius of the jib. The valves are worked from a cabin, sometimes placed on the pedestal. Sometimes two cabins are used, one on each side of the pedestal, with two sets of valves, from either of which the crane can be worked, but often one cabin only is used, and it is placed in the revolving part of the crane, as shown in Fig. 174. In this case the turning cylinder is fixed on an inclined frame at the back of the pillar, the drum which is cupped to receive the links of the chain is fixed to the upper bearing in which the pillar revolves. The pedestals of these cranes are frequently made with archways through them, to economise space by allowing a road or line of rails to be made through them. Screw chocks are fitted at each corner of the pedestal, to relieve the wheels of the weight of the

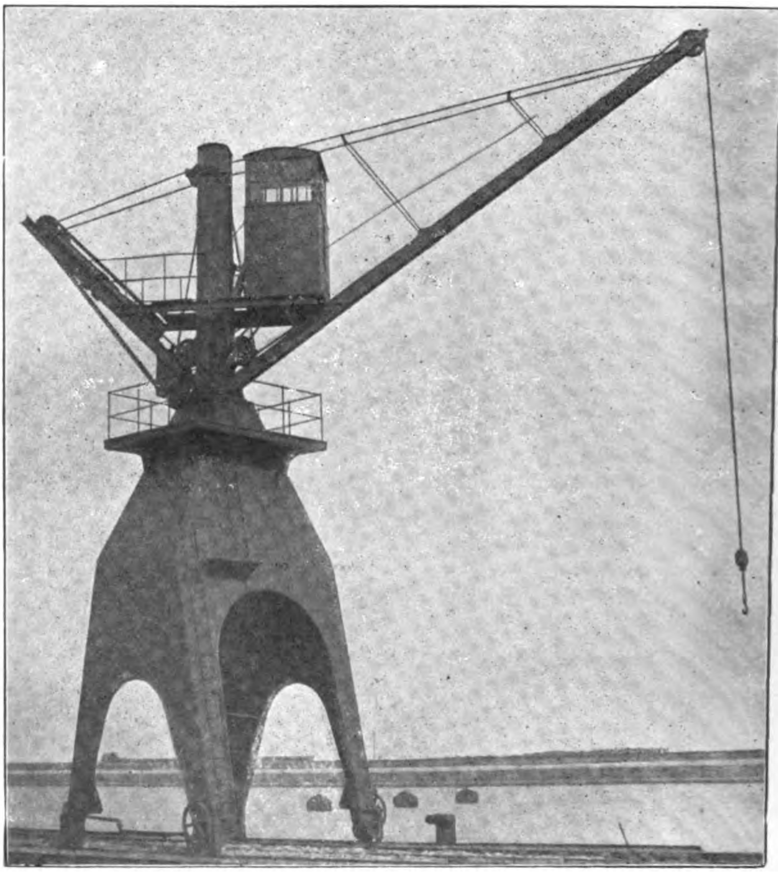

FIG. 174.

crane, and hand gear is often fitted for moving the crane along the quay. There is usually a frame on the back of the pillar, carrying sufficient counterbalance weights to render the crane stable without any attachment to the rails, even if loaded with twice the highest load it is designed to lift.

In cases where an exceptionally long rake or radius of jib is required, the jib is fitted with a topping or derricking motion as shown in Figs. 175 and 176. Cranes intended for dealing with coal or other like minerals are often fitted with weighing gear, which may be of the hydrostatic kind (the chain passing over a pulley attached to a small plunger which rests on water in its cylinder, the pressure of this water showing the weight lifted), or on the steel-yard principle.

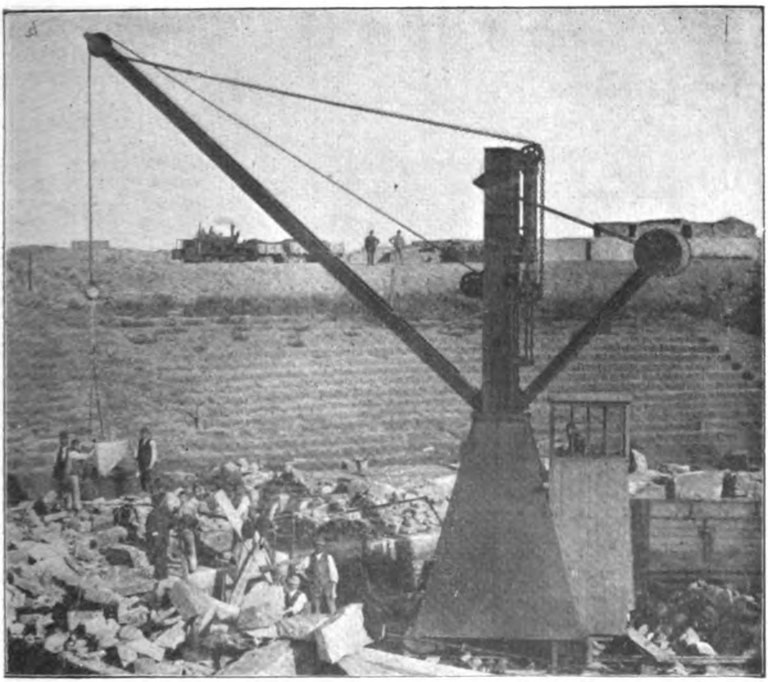

FIG. 175.

In this case the jib-head sheave is mounted, not directly on the jib itself, but on a steel-yard supported on knife-edges near the end of the jib, as shown in Fig. 177; the inner end of the steel-yard being attached to a spring balance, which shows the weight on the crane. The counterbalance weight, balancing the weight of the steel-yard, is shown in the figure, on a short lever whose fulcrum is on the jib.

Cranes with Variable Power.

In cranes lifting small loads, no attempt is made to economise pressure water by using a smaller amount when small loads are dealt with, nor is the power of the crane varied. In the case of 40-ton

Fig. 176.

quay cranes, however, the power of the crane, and the amount of pressure water used per lift, are varied to suit, to some extent, the load raised.

This variation of power is effected in different ways. In one class of crane two powers are obtained by having the lifting ram shaped like a piston, with a very thick piston rod. Full power is obtained by admitting the pressure water to the front or full area of piston, the

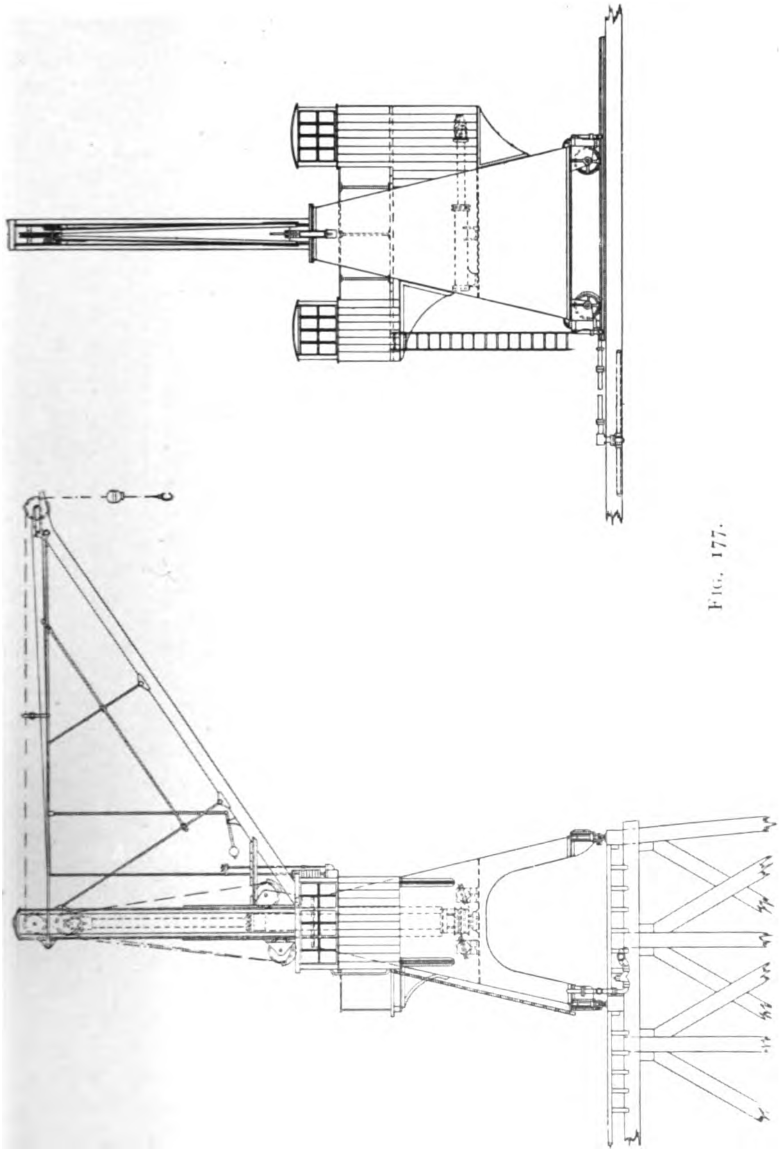

FIG. 177.

back being open to exhaust. When the lower power is required, the pressure supply is admitted to both front and back of the piston, the pressure on the full area predominating, the water on the other side of the piston is forced back into the pressure mains, and thus only a net amount of water, equal to the difference of the displacement of the full piston and of the annular space round the piston rod or plunger, is used, the effective force being that due to the area of the plunger only. The arrangement of valves for such a crane is shown in Fig. 178, which fully explains itself.

Another arrangement for obtaining varying powers is that in

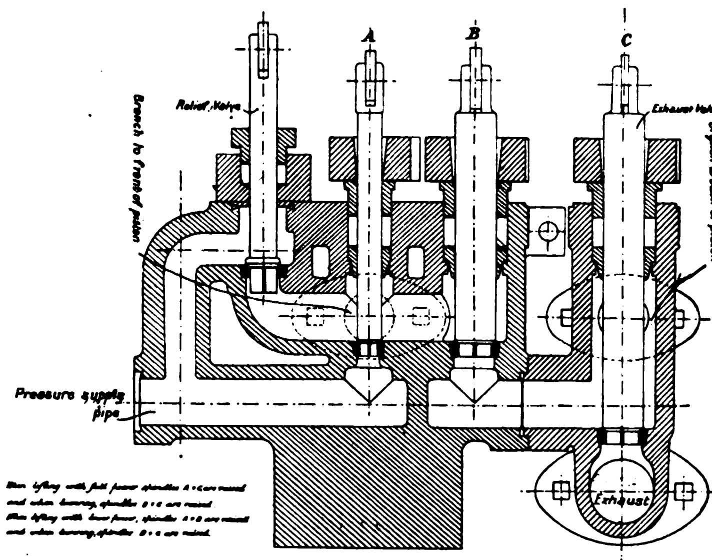

Fig. 178.

which three cylinders are used, these cylinders being placed alongside of each other and close together, the three plungers working in these cylinders being attached to one cross-head, as shown in the plan of the first Armstrong crane (Fig. 168). The valve, very similar to that shown in Fig. 178, is so arranged that the pressure water can be admitted either to the centre plunger, to the two outside plungers, or to all three together.

This crane lifts with three different powers, using proportionate amounts of water. Neither of these methods is now much in favour with the best makers.

It is very seldom that more than two powers are required, and in such cases it is found to be better to employ a cylinder with two concentric plungers, the outer plunger being hollow and forming the cylinder for the inner plunger. Clamps are provided for holding back the outer plunger when the lower power only is required. When full power is necessary, the clamps are thrown out of action, and the two plungers move together. The outer plunger is, of course, fitted with a stuffing-box similar to that of the cylinder in which it moves. In cranes of this class the valves are the same as for an ordinary single-powered crane, and of the same type as that shown in Fig. 178, but with two spindles only, one for pressure and one for exhaust. Slide-valves are used in some cranes, as described at page 275.

Heavy Quay Cranes.

Fig. 179 is a typical example of a heavy fixed pedestal crane, such as referred to above, being capable of lifting 40 tons with 50 feet lift. The pillar containing the lifting machinery is mounted on a pedestal secured to the quay by eight or more strong holding-down bolts. The machinery is inside the pillar as before. It consists of the ordinary hydraulic cylinder with ram or plunger and multiplying sheaves, but, as is usual in heavy cranes, the plunger instead of being a simple one, as in the lighter cranes, consists of two concentric plungers, the inner one working through a stuffing-box in the outer end of the large one. Clips are fitted to the pillar, which can be thrown into action by a hand lever on the side of the pillar. These clips lock the outer plunger, so that the smaller plunger alone acts, when comparatively light loads are being lifted. When heavy loads are being raised, the larger plunger acts on the lifting chain, the inner one remaining inert. Thus with light loads less pressure water is used. The lifting chain, as will be seen from the figure, is double. This is to avoid the use of a very large chain, the pulley above the lifting eye being merely provided to allow for unequal stretch of the two portions of chain, which is doubled in a "bight" over this pulley, and passes thence over the multiplying sheaves in connection with the lifting cylinder, both ends being attached to the cylinder. The same range of power and lift could be obtained by fixing one end only of the chain to the cylinder, increasing the multiplying power of the sheaves and using a running block instead of the equalising pulley described, but the method adopted reduces friction and wear of the chain.

The pillar is turned by a pair of hydraulic cylinders, placed horizontally side by side in a casting on one side of the pedestal, with chains, multiplying sheaves and a drum fixed on the bottom of

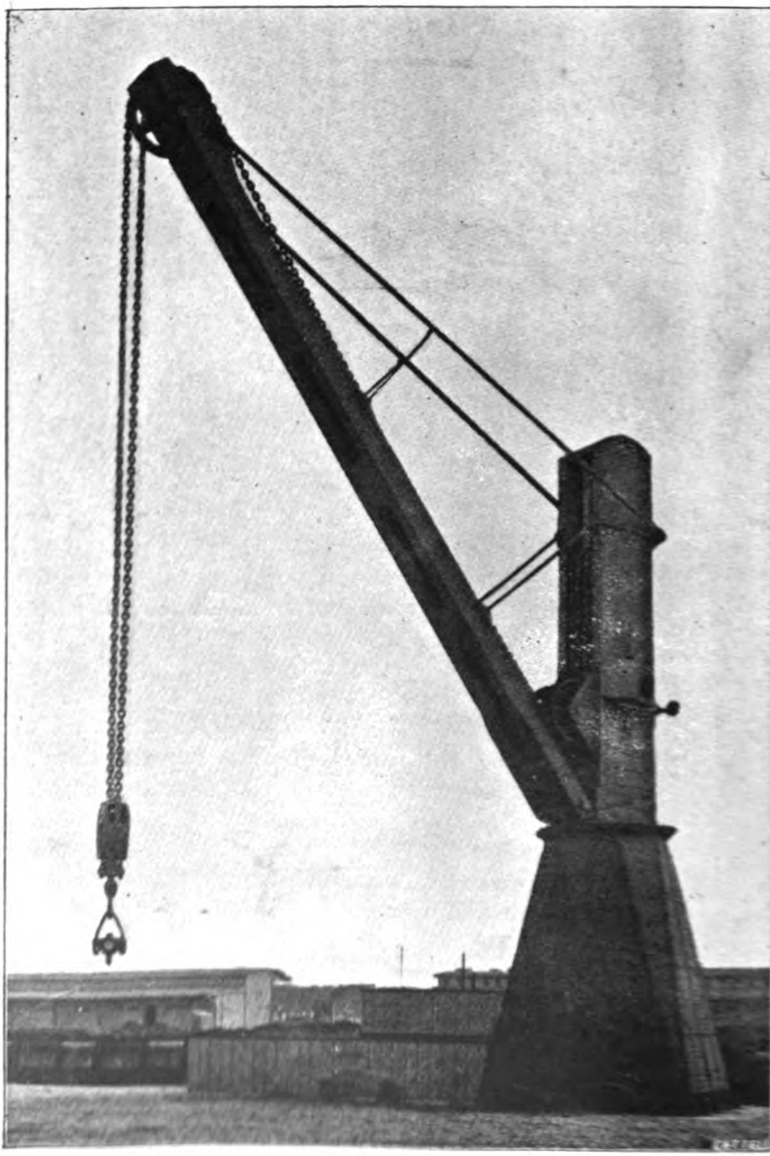

FIG. 179.

the pillar as before. The top bearing of the pillar is usually fitted with "live" or anti-friction rollers. The turning machinery is, in some cases, placed in a chamber under the level of the quay roadway, being covered in by cast-iron plates.

Sometimes these cranes are made to lift 50 tons, and have an archway through the pedestal capable of allowing a locomotive to pass through, as shown in Fig. 180. This crane was designed to serve two graving docks lying side by side.

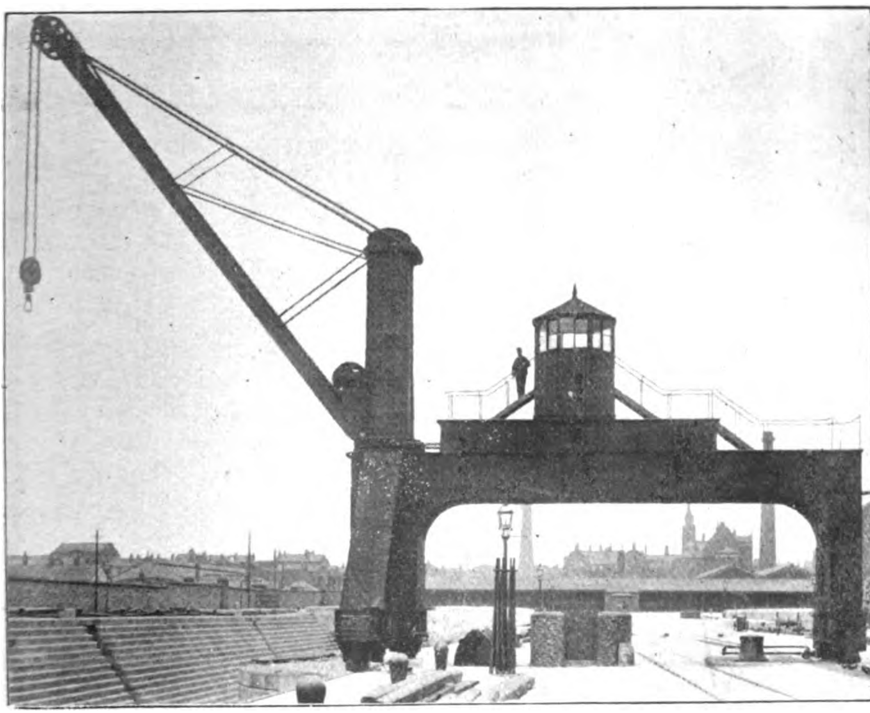

Fig. 180.

When the crane has travelled to the end of these two docks, the leg of the pedestal farthest from the dock edge is secured in a temporary pivot on the quay, and the crane turns round the pivot. It can then be moved along the side of the other dock, the distance between the two docks being rather more than twice the span of the crane pedestal.

With cranes above 50 tons lifting power the "roller path" type of crane is usually adopted (see Fig. 181), either with direct-acting lifting cylinder as shown, or with chains or wire ropes, as before. The crane shown in the illustration can lift 100 tons; the lift of the direct-acting cylinder being 50 feet, and the radius of load 55 feet. The crane is mounted on a ring of "live" or anti-friction rollers, and is turned by a hydraulic engine, with gearing acting on a circular rack.

An auxiliary chain purchase capable of lifting 30 tons through 90 feet is provided, the direct-acting cylinder being swung in towards the jib, when this purchase is in use.

Coaling Cranes.

A very important application of hydraulic power is that which deals with the shipment of coal. Fig. 182 shows a movable crane designed for taking up a wagon of coal and emptying it into a ship.

The apparatus consists of three parts, the crane, the cradle, and the cradle seat. The crane is very much like some of those described, with the addition of a special hydraulic cylinder for tipping the truck. The cradle and seat are so designed that there is no breach in the continuity of the line of rails. The cradle sitting directly on the rails, is guided into place by the seat, which is fitted with ramps at each end so that trucks can easily be run on and off the cradle. The truck is prevented from running forward on the cradle, when the latter is tipped over the ship's side by a pair of hooks engaging one of the axles, as shown in the illustration. These hooks are mounted on a shaft, which can be turned down to disengage the truck. The trucks are hauled by hydraulic capstans, which can also move the crane.

Coal-Hoists.

This is another method of shipping coal. The most usual pattern of hoist is shown in Fig. 183.

The loaded truck is run on to a cradle at the quay level, and is raised by direct-acting hydraulic cylinders, placed directly under the cradle to such a height as may be necessary.

The cradle is fitted with a tipping frame which carries rails, on which the truck rests. This frame is hinged at the front of the cradle, and at the rear end is connected to the plunger of an oscillating hydraulic cylinder fixed to the under-side of the cradle, so that when this cylinder is brought into operation, the truck is tipped into the shoot on the front of the hoist framing. This shoot is adjustable to suit the level of the ship. The framing for guiding the cradle and

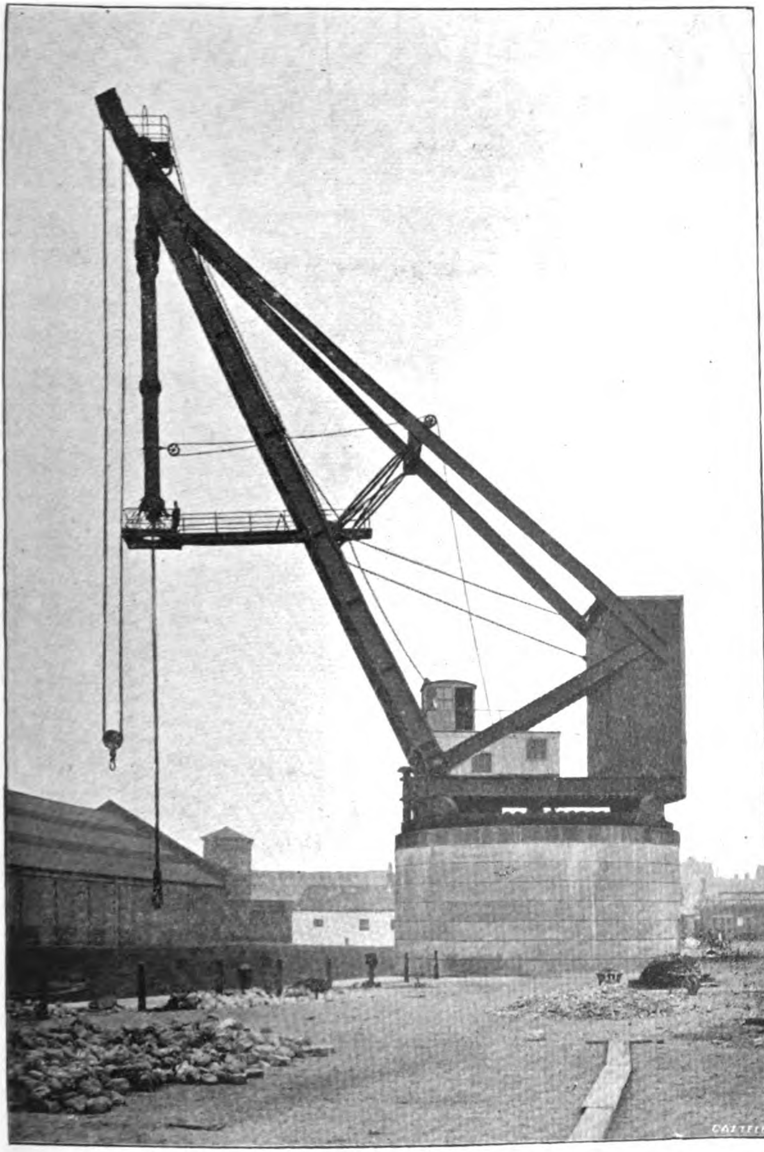

FIG. 181.

carrying the shoot is of iron or steel, and on one side of it is fitted an auxiliary or anti-breakage crane, by which when first starting to load a ship the coal, if of a friable nature, can be lowered in a box from

FIG. 182.

the nose of the shoot to the bottom of the ship, until a sufficient cone of coal is formed in the hold to break the fall of the remainder.

The empty trucks in this case are run off on a viaduct, which is connected with an inclined plane, and are not lowered directly to the quay level.

Where hopper wagons have to be dealt with, the under-side of

FIG. 183.

the cradle is fitted with an inclined hopper or shoot, and the cradle is raised until the nose of this shoot is in line with the rear end of the main shoot. In some cases the cradles are lifted by chains and ropes. Provision is usually made in this, as in all good hoists, for balancing the dead weight of the cradle, and sometimes that of the truck also. Sometimes coal-hoists are made movable along the quay to suit the various hatches of the ship.

Foundry Cranes.

The cranes referred to in the foregoing pages are all by Messrs. Sir W. G. Armstrong, Whitworth & Co. There are, of course, many other forms of hydraulic crane, and other makers. In some foundry cranes, for instance, a horizontal tie, similar to that shown in Fig. 171, is employed, the load being suspended from a little carriage which can run along the tie, pressure water being used to lift the load, which is "traversed" by hand, as in the crane depicted in Fig. 184 ; or in some cranes the three operations of lifting, traversing and sluing are performed by three separate cylinders.

The crane depicted is one in the foundry of the Glenfield Company, Kilmarnock. It is capable of lifting 5 tons 14 feet high, at a radius of 15 feet. The frame-work consists of steel I beams and channels with the lifting cylinder placed between the cheeks of the masts. It is fitted with hand gear for "racking" the load out or in, the bottom step of the crane being fitted with ball bearings, so that the crane is easily slued by hand.

In the background of the ilustration is seen a double powered crane capable of lifting 4 or 9 tons through 17 feet at a radius of 23½ feet. The latter crane has hydraulic sluing and racking cylinders. It is provided with ball bearings at the hook block. The platform on which all the valves are placed is fixed to the mast of the crane and revolves with it.

The crane most fully illustrated is chosen as being of a type in most general use in foundries.

Hydraulic Centre Casting Crane.

When the load has only to be raised a comparatively short distance, and steadiness of motion is essential, as in the case when the load consists of a ladle containing molten metal, a type of crane known as a "centre" crane, is used. In this crane the ram is rigidly fixed to the upper portion of the crane which terminates in a cylindrical mast. The whole upper portion of the crane, bearing the ladle rises and falls with the ram, working in the cylinder, which is placed below ground level.

A good modern example of such a crane, made by Messrs. Henry Berry and Co., of Leeds, is shown in Figs. 185 and 186. The illus-

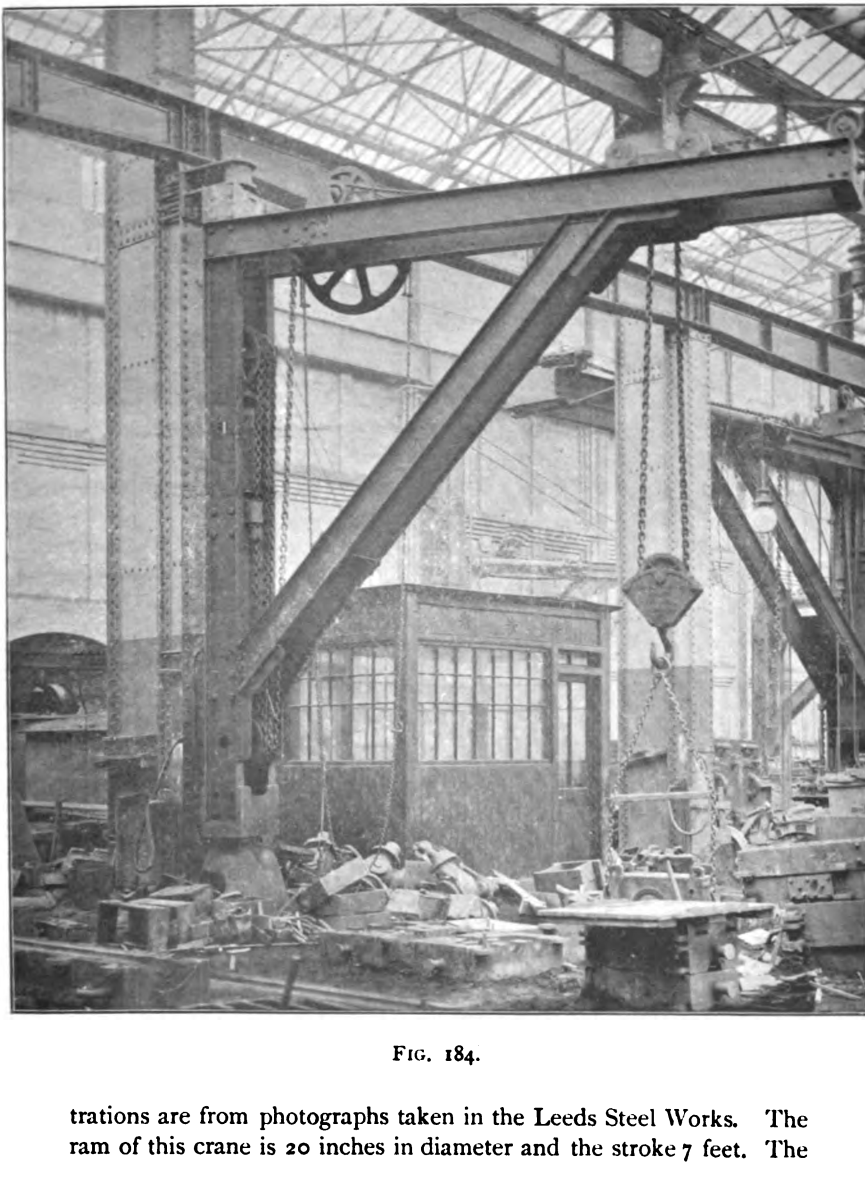

FIG. 184.

trations are from photographs taken in the Leeds Steel Works. The ram of this crane is 20 inches in diameter and the stroke 7 feet. The ladle is supported on the arm of the crane at a distance of 22 feet from the ram. The crane is also fitted with a 9-inch diameter hydraulic

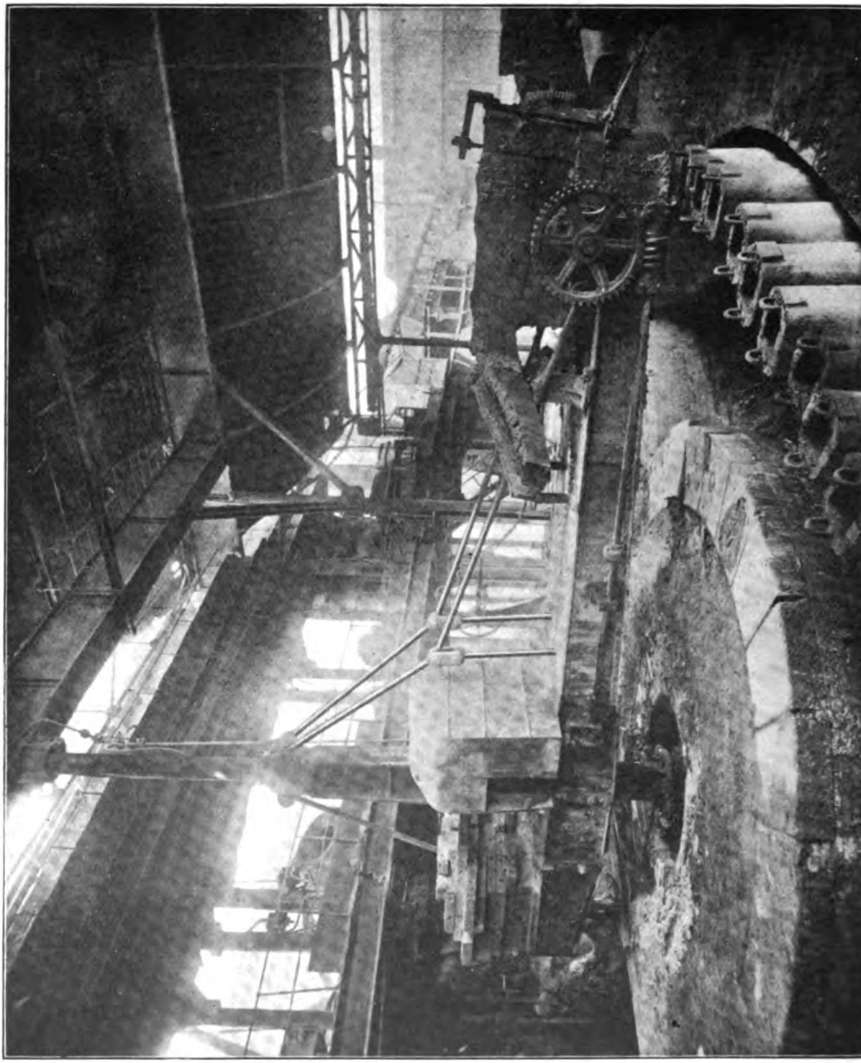

Fig. 185.

cylinder and ram, placed horizontally between the beams bearing the ladle, for altering the position of the ladle in or out. The turning over of the ladle is effected by a small hydraulic engine having rams 3 inches in diameter. This engine is under the sheet steel cover near the central cylinder, and gives motion to a shaft fitted with

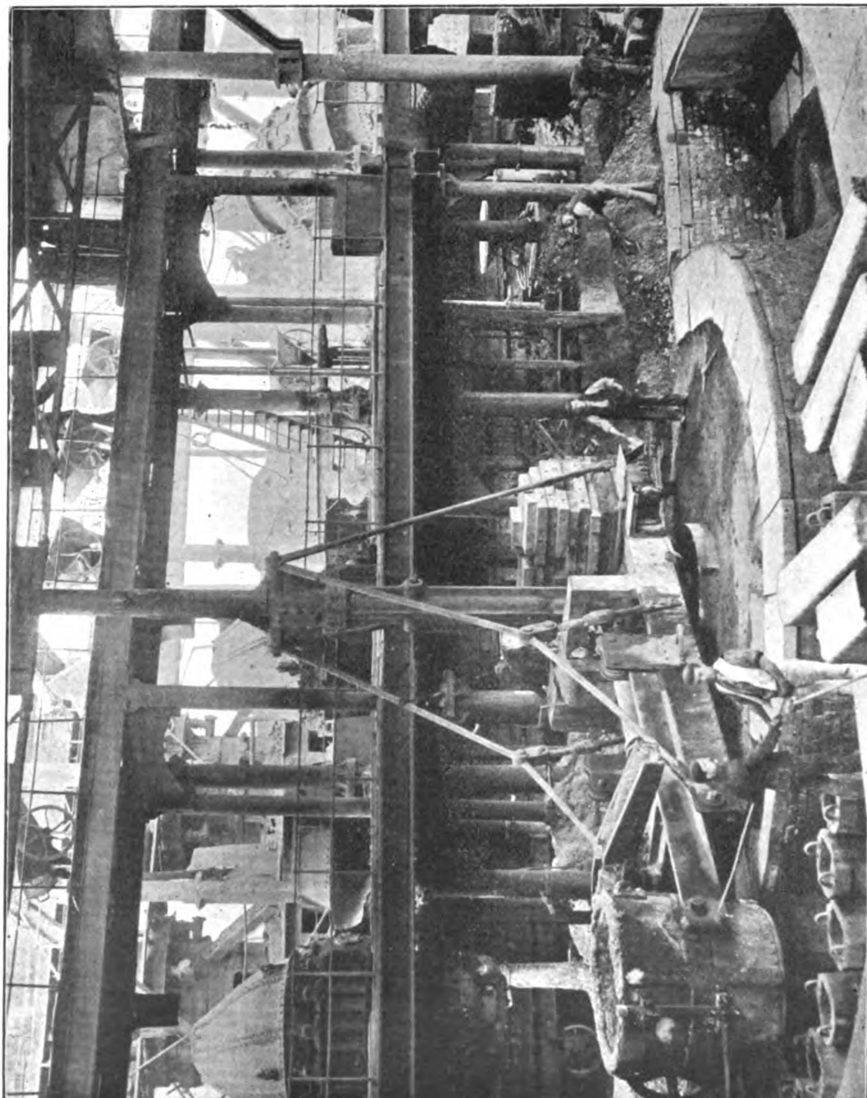

FIG. 186.

worm and worm wheel gear for turning the ladle as shown. The pressure of the water is usually about 620 lbs. per square inch, and the ladle takes 14 tons of molten metal.

This type of crane is known as a top supported crane, the top of the mast passing through, and being capable of rising and falling in, a guide in a strong box girder, securely fastened to the roof of the building. In this form of crane the bending stress on the ram is greatly reduced on account of the support afforded by the mast passing through the girder, and the vertical movement of the ram is consequently effected with a minimum of friction.

The crane is considered a great improvement on the old form, with an unsupported ram which was liable to very heavy combined stresses, owing to the difficulty of always having the centre of gravity of the entire load on the ram exactly in line with the axis of the

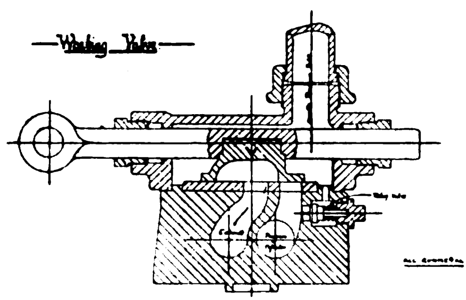

FIG. 187.

latter, the ladle and its contents, some 14 tons, acting at the end of the long arm carrying it, being variable in radius.

The ladle when full of molten steel is supported by the pressure on the ram, and the crane is pulled round the circle of the casting pit by hand (in some cases this is done by a special hydraulic cylinder), the ladle being brought over each ingot mould in turn.

The molten steel is run from the ladle through a nozzle in the bottom of it, the flow of the steel being regulated by a stopper made of refractory material which is submerged in the fluid metal to close the nozzle. This stopper is worked by a lever moving it up and down through guides, so that the stream of metal can be shut off when desired in passing from mould to mould.

When the casting operation is finished, the ladle is immediately turned over by means of the small three cylinder hydraulic engine, so that all scoria or slag may fall out, or be removed from the ladle, which is then fitted with a new nozzle and stopper rod preparatory for the next cast.

The spout shown is merely for running off the slag, which is poured from the converter together with the steel. As the ladle gets filled with steel, the lighter slag floats on the top and runs off through the spout.

The older type of hydraulic "centre crane" of Messrs. Tannett, Walker & Co., is ingenious in that the central ram supports the weight, and acts as a guide in lowering and raising the load, there being two smaller side rams, which are employed to lift the load. In other words, the pressure water acting on the central ram supports the dead weight of the crane, etc., whilst the pressure acting on the side rams overcomes the net load to be raised.. Thus water is economised, for the water under the central ram is returned again to

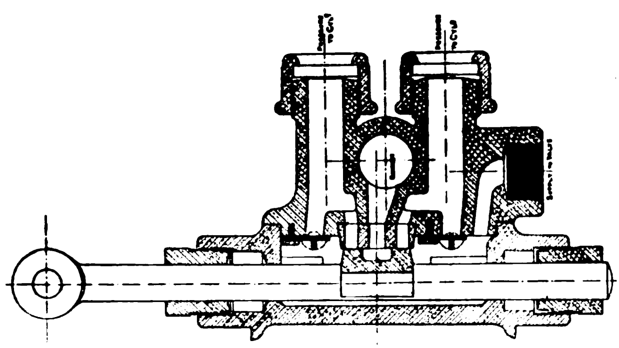

FIG. 188.

the mains, as the crane is lowered. One side ram only may be used to lift small loads, both acting when the full load is raised, and thus a variable power and water consumption are obtained.

CRANE VALVES.

In addition to the reference already made on page 262, it should be noted that for small cranes, or where the pressure is not too high, a simple slide-valve, such as that shown in Fig. 187, is often employed as the working valve. This valve acts in the following way :

When lifting, the valve V is moved to the left, and pressure-supply from the pipe at the top of the figure passes to the pressure cylinder past the right-hand end of the valve. When lowering, V is moved to the right so that water passes from the pressure cylinder back through the inside of the valve to exhaust.

When a heavy load is being quickly lowered, if the valve V be suddenly moved to the position shown in the figure, the water would attain a dangerous pressure were it not able to return to the supply pipe by the relief-valve shown.

The type of valve often used for controlling the turning or sluing motion is shown in Fig. 188. The supply enters by the pipe shown at the right-hand side of the figure, the valve, being a slide-valve, if

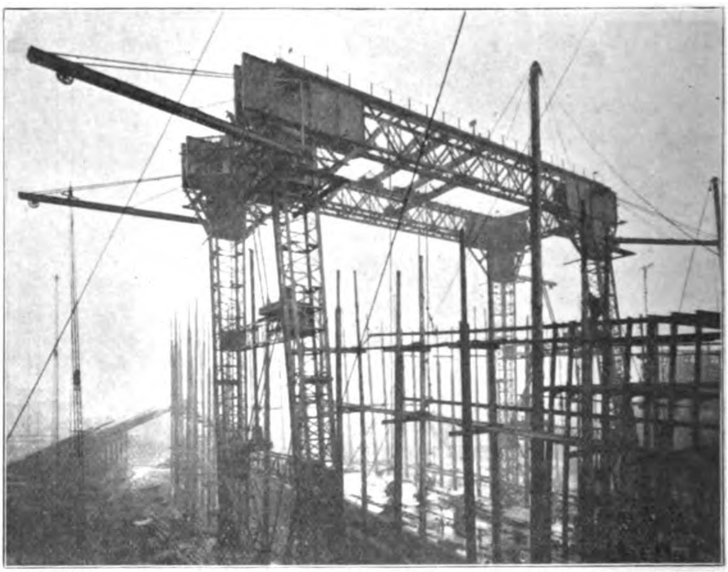

FIG. 189.

moved one way admits supply to one sluing cylinder, if the other way, admits supply to the other cylinder, at the same time opening communication through the hollow space in back of valve between the former cylinder and exhaust.

HYDRAULIC GANTRY.

One of the most interesting among recent applications of hydraulic power is seen in the large hydraulic travelling gantry, erected in the shipbuilding works of Messrs. Harland and Wolff. The total weight of this huge apparatus is 540 tons; it is 100 feet in span from centre to centre of each double pair of rails on which it rests; the clear height from the rail level to the under-side of the cross girders of the gantry is 98 feet, and the clear space between the vertical legs is 95 feet. A view of the complete gantry is given in Fig. 189, whilst Fig. 190 shows the lower portion of one side, with lift cylinders and hydraulic engines for moving the gantry. The entire apparatus is built of steel; the use of lattice bracing, with rigidly braced plate-girders at the corners, gives the necessary rigidity. At the four corners are fixed four travelling jib cranes for suspending

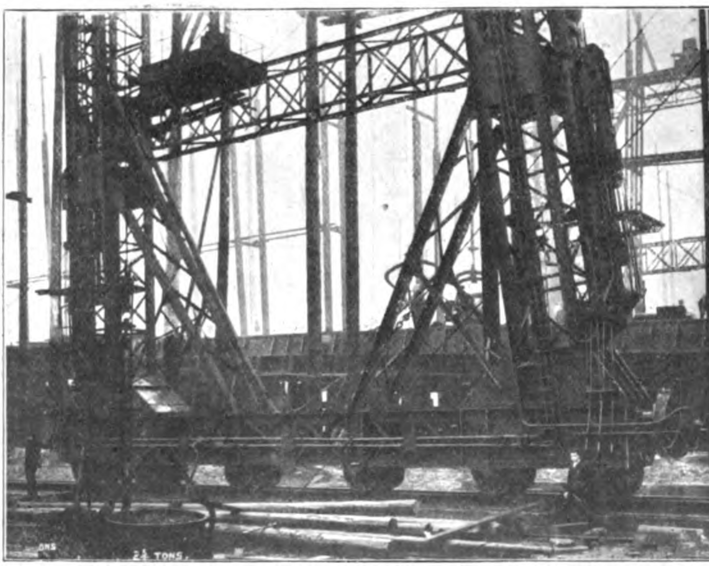

FIG. 190.

portable punchers and riveters and for handling material. Each is capable of lifting a load of about 4 tons through a height of 80 feet at a rake of 40 feet, and of turning through an angle of 180°, all these motions being performed by hydraulic power, the hydraulic cylinders and valves controlling the motions being fixed to the legs of the gantry near the ground, as shown in Fig. 190, steel wire ropes being employed between the cylinders and the cranes, and for lifting the loads. In addition to these jib cranes, the gantry also supports three hydraulic travelling cranes, working at the top of the gantry, two at a higher level than the other. These cranes support, by hydraulic lifts suspended from the travelling carriages of the cranes, portable riveting machines, etc., the motion of the travellers being controlled from the level at which the riveters are worked by hand chains connected to sprocket wheels on the cranes. The gantry is propelled up or down its double set of rails by two hydraulic engines, seen at the extreme right in the second illustration, which act by gearing on the wheels of the gantry. The rails consist of strong steel bars of H section with flat steel bars riveted upon the upper surface. The two H sections are braced together and are bedded upon concrete beds carried on piles, the total length of the rails being 650 feet. We understand that Messrs. Harland and Wolff have now three of these gantries in use.

XXII.

HYDRAULIC LIFTS.

Passenger Lifts.

The increasing value of land in cities, necessitating increased height of houses, has rendered a lift an almost indispensable adjunct of modern civilisation. A good lift, well designed in every detail, and well constructed, provides for almost every contingency which can arise in connection with its use. Failure of the valve, over-travel, failure or fracture of the suspending ropes or supporting ram, leakage, too high speed, are all taken into account and provided against in a high-class lift. But these provisions cost money, hence a really good lift cannot be had at a low price.

To erect a cheap unsafe passenger-lift is not only to be "penny wise and pound foolish," but it is morally criminal. A merchant who would not trust his own person inside an unsafe vehicle, has no right to expect his employés to use daily a lift which is dangerous to life and limb. Proprietors of hotels and other places of public resort are likely, for their own credit, to be careful in this matter, but *all* lifts for the use of passengers should be inspected and licensed by some competent authority, such as the Board of Trade. If this were done, the familiar heading, "Another Lift Accident," would probably soon disappear from the daily newspapers.

Hydraulic Lifts.

The old mechanical lift, worked by ropes or pitch-chain and pulleys, continually moving up and down so long as the shafting driving it revolved, had its day, and despite its dangerous character is still used. For purposes where safety is a first consideration, however, the hydraulic lift alone is employed.

It is difficult now to determine with certainty who first constructed a hydraulic lift; probably Lord Armstrong, who is said to have employed one of his cranes to raise and lower in the vertical shaft of a warehouse or workshop, a cage containing goods. The extension of our systems of hydraulic power co-operation has rendered the hydraulic hoist, or lift, one of the most popular and common of the many applications of pressure water to ordinary operations. The principle of a hydraulic lift is exceedingly simple, yet a perfect and safe lift is by no means easy to construct.

In our pressure mains is water, every pound of which at any given point possesses energy $h + \frac{v^2}{2g} + 2 \cdot 3 p$ ft.-lbs., because it is h feet above datum, is moving at a velocity of v feet per second, and is at a pressure of p lbs. per square inch.

The energy represented by the first two terms is not available, to any considerable extent, for doing work in a cylinder of a lift. We may take the last term as representing the energy of each pound. A hydraulic lift is an appliance which takes this energy and transforms it into the potential energy of goods or people at the higher levels served by the lift. Both these are high forms of energy, so we expect, and indeed obtain, a high efficiency from the apparatus.

Direct-acting Hydraulic Passenger-Lift.

There is no doubt that most people feel more safe, in a lift, if they can see how it is supported and moved. The feeling of safety is greatly increased, if the passenger can see that the cage is *pushed* up by a ram instead of being *pulled* up by chains or ropes, because in the latter case the thought of what would occur if the ropes were to break, interferes with one's comfort. It is quite true that the direct-acting lift, without any balancing arrangement, consisting of a cage or room resting on and attached to a ram, which works vertically in a long cylinder, and is supported by water, is a very safe and simple appliance. Unfortunately such an arrangement is usually expensive, for each lift of the cage will use as much pressure water as is required to fill the space left by the rising ram. On account of this expense, and the first cost entailed by sinking a well for the long cylinder to a depth of 60 or 100 feet, to say nothing of the making and jointing of the ram and cylinder (these items of first cost being common to all direct-acting lifts), this simple form of lift is not so much used as one might expect. Where a plentiful supply of water at a comparatively low pressure is available, it offers the advantages of simplicity and safety. The direct-acting lift, with proper balancing arrangements, is very much used in cities provided with high-pressure supply, and is probably nearly as safe as a good suspended lift. It is worthy of note, however, in passing, that one or two recent fatal lift accidents have happened to direct-acting lifts. The feeling of greater safety in a direct-acting lift due to the fact that the passenger can see how he is pushed up by a substantial-looking ram, is more or less illusory, if the comparison be made with a high-class suspended lift. For one thing, the direct-acting lift hardly affords the same facility for the use of a safety gear.

In designing one of the simple direct-acting lifts the principal calculations are easy. Thus if the weight of the ram—i.e. the force necessary to support it when as far out of the press as it can go—is W lbs., that of the cage w lbs., and the load L lbs., the pressure of the water being p lbs. per square inch, then the cross-section of the ram, a square inches, is given by

$$W + w + L = p\,a$$

if friction be neglected; or, if the average force necessary to overcome friction be 20 per cent. of the total load,

$$\frac{120}{100}(W + w + L) = p\,a.$$

Suppose, for example, W is 1500 lbs., w 1220 lbs., the load to be raised 1120 lbs., and the least diameter of ram consistent with its use as a strut is 8 inches; find the pressure of the water.

$$\frac{12}{10}\,\frac{(1500 + 1220 + 1120)}{0{\cdot}7854 \times 8^2} = p,$$

or p is 91·67 lbs. per square inch. A slightly greater pressure in the supply pipes would be necessary, owing to the resistance of the valve, etc.

If we attempt to use this lift with a high-pressure supply of, say, 700 lbs. per square inch, we find that the area of the ram is

$$\frac{12}{10}\,\frac{(1500 + 1220 + 1120)}{700} = a,$$

or a is 6·58 square inches or its diameter is 2·89 inches, which would be much too thin for a long column such as this ram.

This shows the necessity for some sort of pressure-diminishing arrangement where high-pressure supply is used.

In places where the lift can be worked, not from pressure mains connected with an accumulator, but from a tank or the ordinary domestic supply, the objection of the too great ram diameter necessary for strength giving too high a force, does not apply.

For instance, in the above case, if we have a comparatively low lift, and know that from strength considerations the ram should be 5 inches in diameter, the pressure of our water will be $= \frac{3840}{0 \cdot 7854 \times 5^2}$ = 195·5 lbs. per square inch neglecting friction, or 234·6 lbs. per square inch, allowing 20 per cent. for friction. If this water comes from a tank or reservoir, the height of the water in it must be at least 234·6 × 2·3, or 539½ feet above the level of the hoist. If the actual head available is 200 feet, we must increase the diameter of our ram to 8·27 inches.

The ram diameter can thus be determined for the pressure available, but if too great will give rise to constructive difficulties and expense, as well as greater friction.

We are here neglecting the fact that the apparent weight of the ram which has to be considered, is not really a constant thing at all, but becomes greater, the further the ram emerges from its cylinder. You know that a stone is more easily lifted if immersed in water; that, in fact, any body seems to lose, through immersion in water, *an amount of weight equal to the weight of the water it displaces.* This should be taken into account in any perfect balancing arrangement, but if the pressure is high, it is not of much practical importance.

You see, then, that with high-pressure water, if we make the ram of sufficiently large diameter to stand the stresses to which it is subjected as a strut, the pressure on it will be much too great, causing too rapid motion, unless we reduce the pressure of the water somehow, say, by the wasteful method of throttling.

Again, it would be very wasteful to take water from our pressure mains and employ it to raise a dead weight, such as that of the ram and cage, which, in descending, would give back none of the energy spent on it; and repeat this operation hundreds of times a day. Some sort of balance, therefore, becomes a necessity. Various balancing devices will be described presently. Fig. 191 shows the general arrangement of a good modern direct-acting lift.

The room, or cage (usually called "the lift") consists of a well-

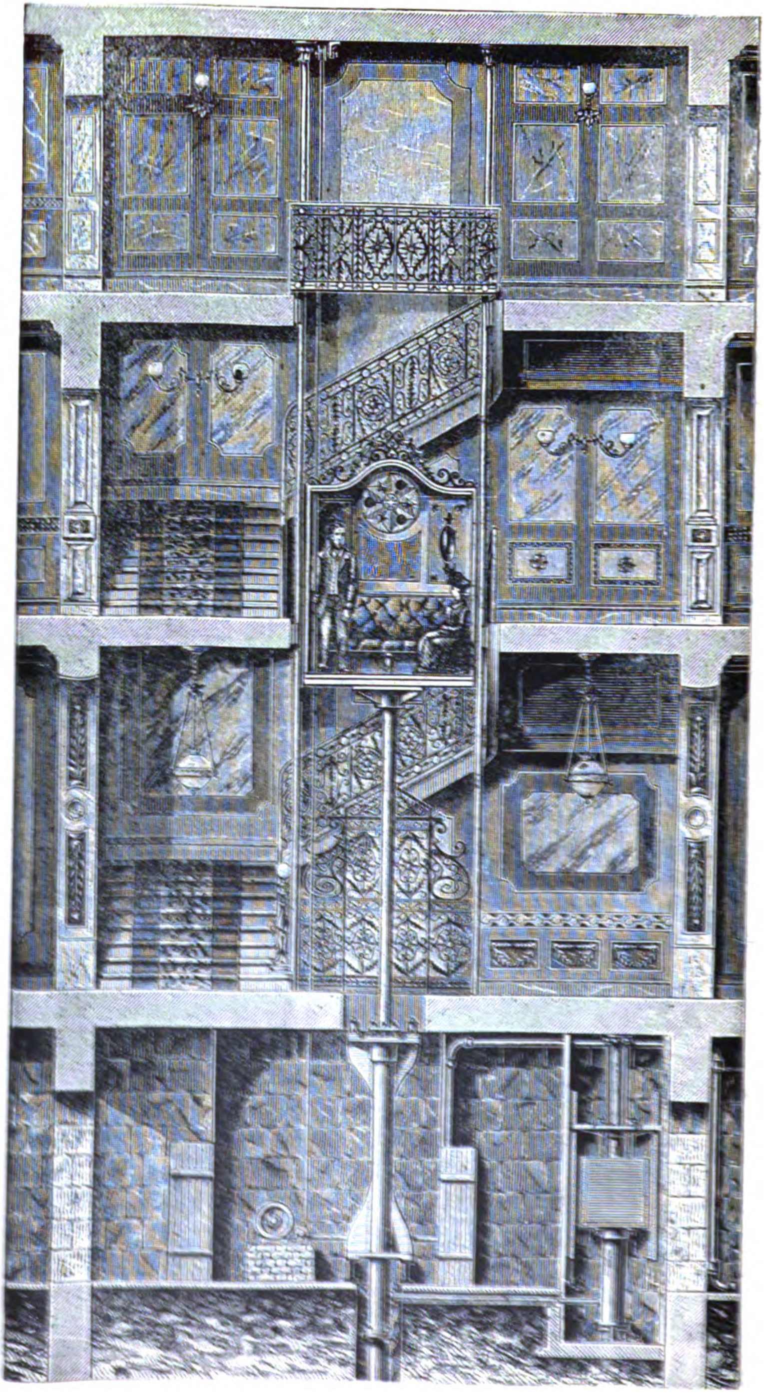

Fig. 191.

finished and comfortably upholstered room, moving vertically in a square shaft or well-hole adjoining the stairs of the building. The cage or room is securely fastened to the top of a long ram, which works watertight through the gland and stuffing-box—very similar to that of a steam-engine cylinder—of a press sunk deeply into the earth. In the cellar to the right is seen the hydraulic balance (to be more fully described), with its feed-pipe from the mains and the service pipe from it to the lift cylinder.

The valve by which the motion of the lift is controlled is shown worked by a rope passing up through the cage.

Hydraulic Suspended Lifts.

Before referring more fully to methods of balancing, it may be well to consider a good modern suspended lift. Fig. 192 shows the kind of lift which has been fitted throughout the Imperial Institute, London, by Messrs. Archibald Smith and Stevens. The cage is suspended by four steel wire ropes, each of which is of sufficient

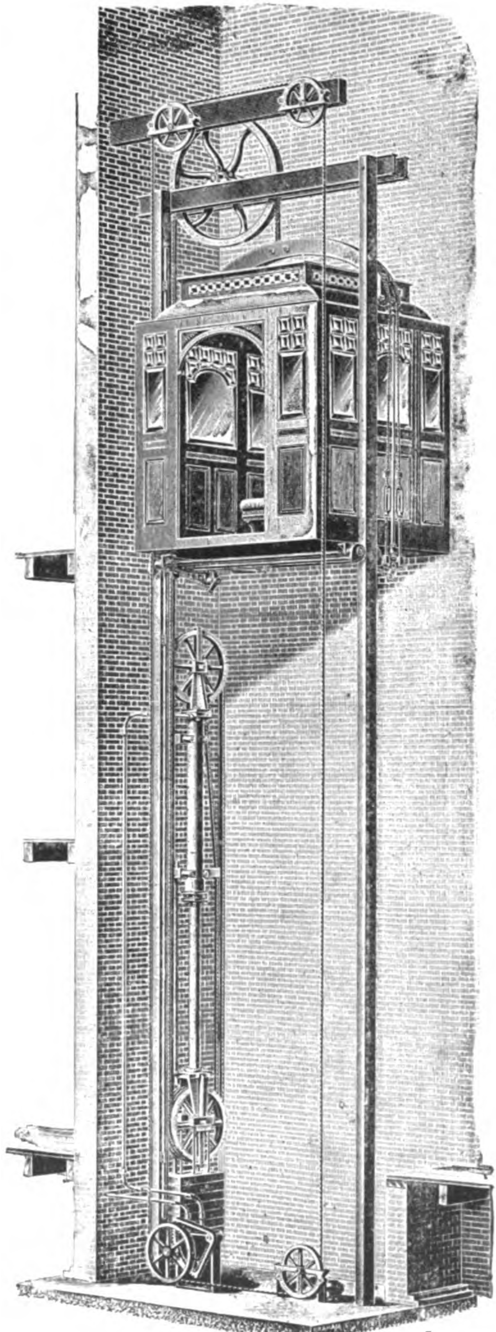

Fig. 192.

strength to bear the entire load with perfect safety. The cage is carried in an iron cradle, and is fitted with a safety gear of a very ingenious kind, which comes into action stopping the cage, if any one of the ropes stretches or breaks. The speed can be increased to 500 feet per minute—which would be impossible with direct-acting lifts—but 200 feet per minute is the limit recommended. The lift is simple, and no deep bore-hole for the lift cylinder is required, as it can be placed in any convenient position, the motion of the ram being magnified by pulleys as shown. As regards cost, it is said that in London the cost of moving a load of 10 cwt. up and down three times does not exceed one penny.

Arrangements of the "Otis Elevator."

The Otis Elevator Company—the pioneers in the branch of engineering devoted to lift construction—adopt an arrangement of cylinder and valves which is worthy of careful consideration.

The arrangement is shown diagrammatically in Fig. 193, where A is the cage, B the wire rope carrying it (in practice there are *four* ropes), passing over the guide pulley C and the pulley D on the end of the rod of the piston F, which works in the hydraulic lift cylinder G.

It is evident that to lift the cage A the piston F must be forced downwards. The pressure water—either from pressure mains or a tank at a sufficient height—is admitted at H to the valve box, in which the slide valve K works. F moving downwards, it is evident that the water under F must escape, which it does by the escape or discharge pipe J, passing to J through the inner or exhaust space of the valve.

The position of the valve shown in Fig. α corresponds with the upward journey of the cage A; that in Fig. β, where the admission port 2 and the escape pipe 4 are both covered by the flanges of the valve, represents the position of things when the cage is stationary; and that shown in Fig. γ represents the position for the downward journey of the cage. The cage which, even when empty, is sufficiently heavy to draw the piston F up, does so, and the water above the latter passes through the passages 2 and 3 into the lower end of the cylinder, the escape port 4 and the inlet 2 being both closed to pressure supply.

It is evident that, by slightly opening the passages 2 and 4, the motion of the cage may be varied at will within certain limits. This throttling, however, is wasteful of energy. The movement of the valve is effected by the hand-rope N, passing up through the cage.

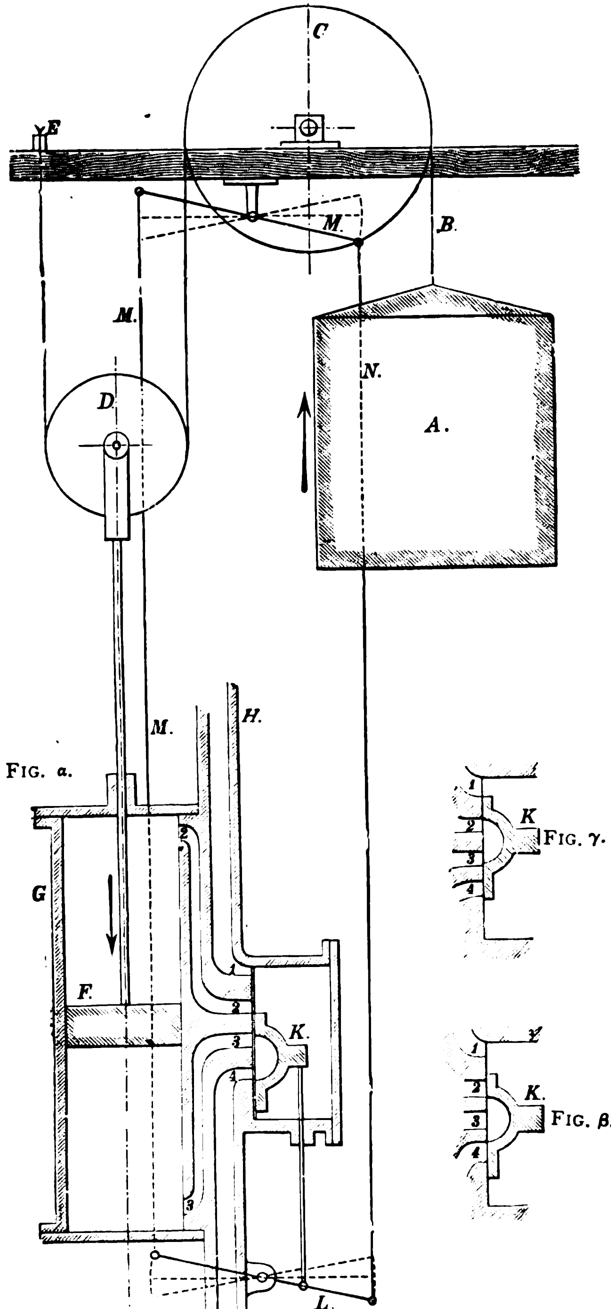

Fig. α.

Fig. γ.

Fig. β.

Fig. 193.

This acts on the lever L, and on the valve, the rope being kept taut by the lever M.

By gradually closing the ports the motion of the cage can be reduced very gradually. If by inadvertence the inlet port is not closed, when the cage reaches the top of its ascent, the piston F, in its further descent, covers the outlet passage 3, the piston being forced to stop, having a quantity of unyielding water in front of it. In a similar way, if the valve is not moved to the proper position in the downward journey, the piston covers passage 2 and is compelled to stop. A safety valve is provided to prevent undue pressure due to sudden closing of the passages, by hand or otherwise. This arrangement of allowing the exhaust water, during descent, to merely pass round into the other end of the cylinder, is a most important feature, because it is hardly possible that this transfer can take place with sufficient rapidity to permit a very sudden descent of the cage. With such an arrangement, an accident like that which took place recently in London, referred to later on, is impossible, at least during descent.

This arrangement also obviates the difficulty met with in some suspended hydraulic lifts, where, if the cage in its downward course, meets with an obstruction, the ram and ropes go on slacking out, and the sudden removal of the obstacle may cause an accident on account of the quantity of rope which has been paid out. Even if an accident do not occur, the ropes are liable to leave the pulleys when very slack. The Otis arrangement prevents these troubles, as the stoppage of the cage merely stops the ram, since it is the excess weight of cage which pulls the ram up, and if this excess weight be removed the piston stops, both sides of it being subjected to the same water pressure.

The actual construction of the lift differs from that shown in detail, but the principle is the same. Thus, there are four ropes instead of one as shown, two piston rods alongside of each other, joined to the one piston, thus facilitating the fixing of the loose pulley D; and a balance weight, nearly balancing the weight of the cage, is fixed under D. The valve, instead of being a flat slide valve as shown in the figure, is a piston valve, which, of course, works much more freely under heavy pressure, and also allows a series of perforations in the covers of the ports, which are gradually closed by the valve. The hand-rope acts on a rack and pinion, instead of on the lever shown, but the principle is that adopted by the company for lifts not over, say, 66 feet.

For higher lifts the column of water under the piston would exceed 33 feet, and would no longer be supported by atmospheric pressure. The lift cylinder can, however, be placed horizontally and further guide pulleys employed, or the cylinder can be sunk under the level of the escape water, for higher elevation. The Otis Company have over 2500 "elevators" in New York alone, of which at least 50 have lifts of from 225 to 250 feet.

In the Empire Building, Broadway, there are ten Otis hydraulic lifts working side by side, with a rise of about 250 feet.

So far as the author is aware, this company does not use hydraulic balances, employing the dead weight already noted. The annular piston area only is employed for lifting, hence the hydraulic balance is less required in this than in direct-acting lifts.

The method of reducing the pressure by throttling, already noticed, though very convenient, and a valuable means of adding to the smoothness of the motion at stopping or starting, is wasteful, and not, in principle, a desirable thing.

The annular volume has to be filled with water each time the piston descends or the load is raised; this water passes round to the other end of the cylinder as the cage is lowered, and to the exhaust on next lift of cage. Hence a volume of pressure water equal to the annular volume is used each lift of the cage. The annular volume may, however, be made sufficiently small to prevent excessive cost of water.

These lifts are most frequently actuated by water from the ordinary domestic supply mains, from a tank, or from a private pumping plant on the premises.

The arrangements for packing the piston and piston rods are good. The piston being at the bottom of its stroke, the cover of the lower end of the cylinder is taken off and new rings put in. The piston rods are packed in the usual way with glands and stuffing-boxes, which are of course readily accessible.

The safety-gear of this lift is described under the separate heading "Safety-Gears."

In the Commercial Buildings, New York, more than 67,000 persons are raised and lowered per day by Otis lifts. This company also supplied the principal lifts of the Eiffel Tower, and more recently the lifts of the Glasgow Harbour tunnel, of which a brief description is given at page 308.

Safety Gears.

Many safety gears or safety catches for lifts have been designed. only two or three of which, however, are really reliable. The commonest form of safety gear for a suspended lift is one in which the suspending ropes are attached to levers, which bear cams or latches at their outer ends, these cams being kept from coming in contact with the guides by the pull of the suspending ropes. A spring is usually provided which brings the cam into contact as soon as the rope breaks or slacks sufficiently. The difficulty with such an arrangement is that, if the springs are strong enough to make the cams act as really effective brakes on the cage in case of fracture of the ropes, they will probably cause the safety apparatus to act each time the cage starts downwards. Since the cage, load and ram constitute a considerable mass, and have therefore great inertia, they resist being set in motion even by gravity. Hence, when the cage is stationary, just before descending, the suspending ropes must slack out freely in order that the cage may promptly start its downward journey. This is the opportunity of the spring-impelled safety catch, and if sufficiently active to be a real safeguard, it is difficult to prevent it from acting when not required.

It must be remembered too, that ropes seldom break quite through, and the back drag of a portion of a broken rope, with a gear of this sort, may be sufficient to prevent it from acting even when the cage is falling.

Safety Gear of the Reliance Lift.

The safety gear of the class of lift erected at the Imperial Institute is shown in Fig. 194. Hundreds of these lifts have been erected in Great Britain, the Colonies and India, with a record of absolute immunity from accident. The gear, as will be seen from an examination of the figure, has no springs, and it is *actuated by the entire suspended weight.* The lift is suspended by four ropes, two on each side of the cage, each rope A being secured to the horizontal arm of a bell crank B. The vertical arms of the two bell cranks—referring to one side of the cage as shown in the figure—are joined together by a bar C, so that the bell cranks must move simultaneously. Under ordinary conditions, the two ropes being equally tight, the horizontal forces of the two cranks inward are equal, and hence these forces neutralising each other the cage is suspended symmetrically with respect to the guides G G. Two cams F F are provided having projecting lugs, which engage with corresponding projections on the bar C. It will also be noticed that the cam-shafts E E are coupled together, so that if one cam engages with the guide all must grip together. If a cam once comes in contact with the guide, any tendency of the cage to descend only intensifies the grip. Suppose one of the ropes A to slacken, then the other rope having more than its share of the weight to support, causes an extra horizontal force on the bar C, which, acting against the diminished force due to the other lightly loaded crank, moves the cage over, and the cam on the former side comes in contact with the guide, immediately causing all the cams to grip, and arresting the cage. Thus the gear does not usually wait till the rope actually breaks, but acts on that stretching

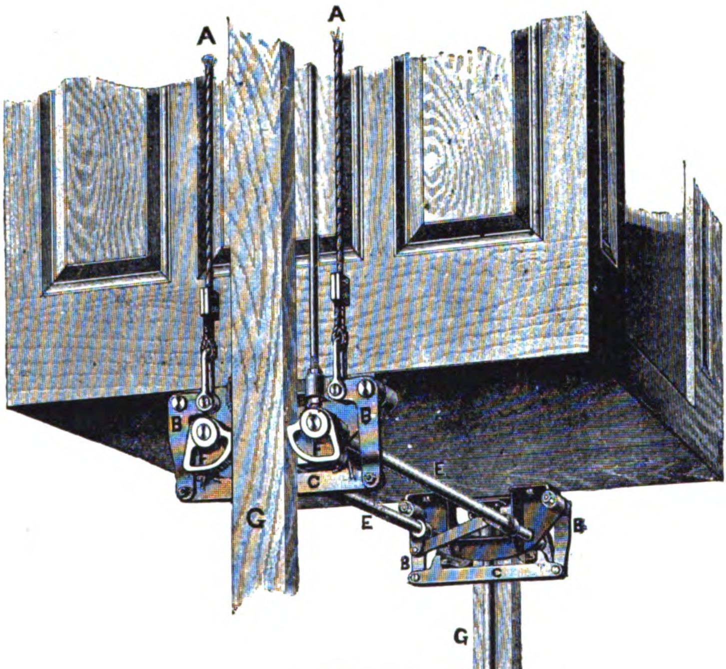

FIG. 194.

taking place which usually precedes fracture. This gear often acts by the stretching of a sound rope under ordinary conditions, but this only calls attention to the fact that the rope requires tightening by the means provided for that purpose. If the gear acts through the stretching or breaking of a rope, the cage can readily be drawn up to the floor above so that the passengers may alight, the cams only preventing ***downward*** motion. If all the ropes broke absolutely simultaneously, the gear might not prevent an accident, if the cage were to drop without one or other of the cams touching. But such a thing is almost impossible, and the addition of a spring—which, however, it is as well to avoid for the reasons already given—would provide for even this exceedingly remote contingency. Any safety gear which only comes into use in case of an accident, and is never used under the regular working conditions, generally fails when required, having become clogged or oxidised through want of use. The common test of cutting the ropes immediately above the cage, is one that a very imperfect gear will satisfy, since the back drag of a portion of a broken rope, which would prevent the gear from acting during an accident, is absent.

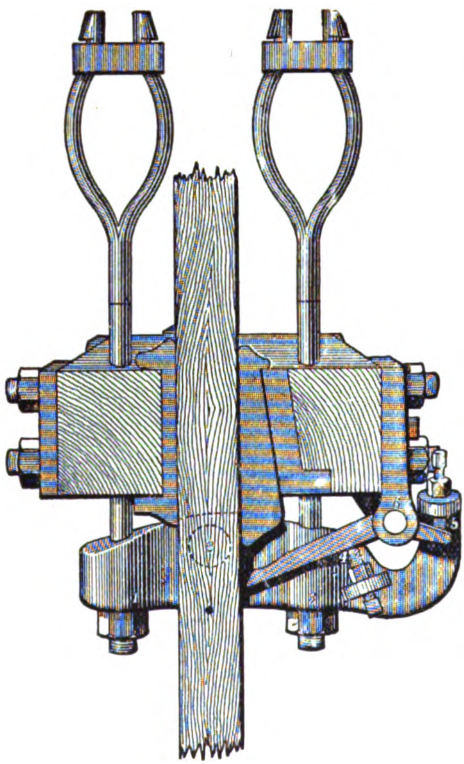

FIG. 195.

SAFETY GEAR OR CATCH OF THE OTIS LIFT.

The cage, as already stated, is carried by four ropes. These ropes pass first to an iron yoke which surmounts the cage, and thence in two pairs down two opposite sides of the cage to suspending eyes, which carry the cross-piece on which the cage rests. This cross-piece, the top yoke and oblique ties, form the framework containing the cage, this framework carrying four gun-metal guide sleeves, which slide up and down on planed hardwood guides.

The ropes are not attached directly to the lower cross-piece, but to a lever or balance-beam, which has its centre of oscillation 2 (Fig. 195) in the cross-piece, and its ends 1 and 3 receive the pull of the two ropes directly. If any one of the ropes stretches, the balance beam rises at one side or the other, any such rise causing the catch-wedge to grip and arrest the cage in its descent. For instance, if the rope at 1 is a little too tight, the end 1 of the balance beam rises above its normal horizontal position, causing its head 5 to strike the gripping lever 6, 4, and so forcing the wedge at the right of the guide to grip. Each time this wedge is driven home that in the opposite end of the cross-beam is also tightened as the spindle 6 of the gripping lever passes through to the other side of the cage, there carrying a similar lever, which, being like this one, fast on the spindle, must act when the nearer one acts. If rope 3 is too tight, 3 on rising strikes the gripping lever at 4, also causing the wedge to act. When the cage is thus arrested it is only necessary to move the valve so as to raise the cage a little in order to free the wedges and allow ascent of the cage, a weak spring relieving the wedge. This is one of the few safety devices which is really safe.

Safety Gear to prevent Excessive Speed.

Both in this, and the Reliance lift already described, there is one possible but extremely improbable kind of accident which the gear does not seem quite able to meet, that is the simultaneous breakage of *all* the ropes. Such an accident is unheard of, as *each* rope will bear the whole load on the four with a factor of safety of about 14, but in some of the Otis lifts even this contingency is provided for by a separate safety device, which is intended to prevent excessive speed either of ascent or descent.

It is possible that a person unaccustomed to the handling of the lift might, with an excessive load, open the valve too far and allow too rapid a descent.

This "regulator" consists of a centrifugal governor, which is actuated by a light endless wire-rope or belt attached to the cage, the rope above and below running over idle pulleys. This rope can be gripped and held fast by a clamp near the regulator; the clamp acts as soon as a certain pre-arranged speed is exceeded. The rope being held fast, is left behind by the cage in its descent, and a nut fastened on the rope strikes a lever, which in turn acts on the safety wedge causing the latter to grip, and stop the cage.

In the illustration (Fig. 196) the regulator is shown in the position it would assume on the speed exceeding the allowed limit. The governor balls diverging, have actuated the sleeve lever and crank which cause the rope to be caught just to the left of the upper guide pulley. The way in which the relative motion of the cage to the now stationary, or even slowly moving, rope, is made use of to actuate the gripping wedge is readily seen from the figure, the little projecting toe *a*, seen to the left, being fast on the common axle of the nearer

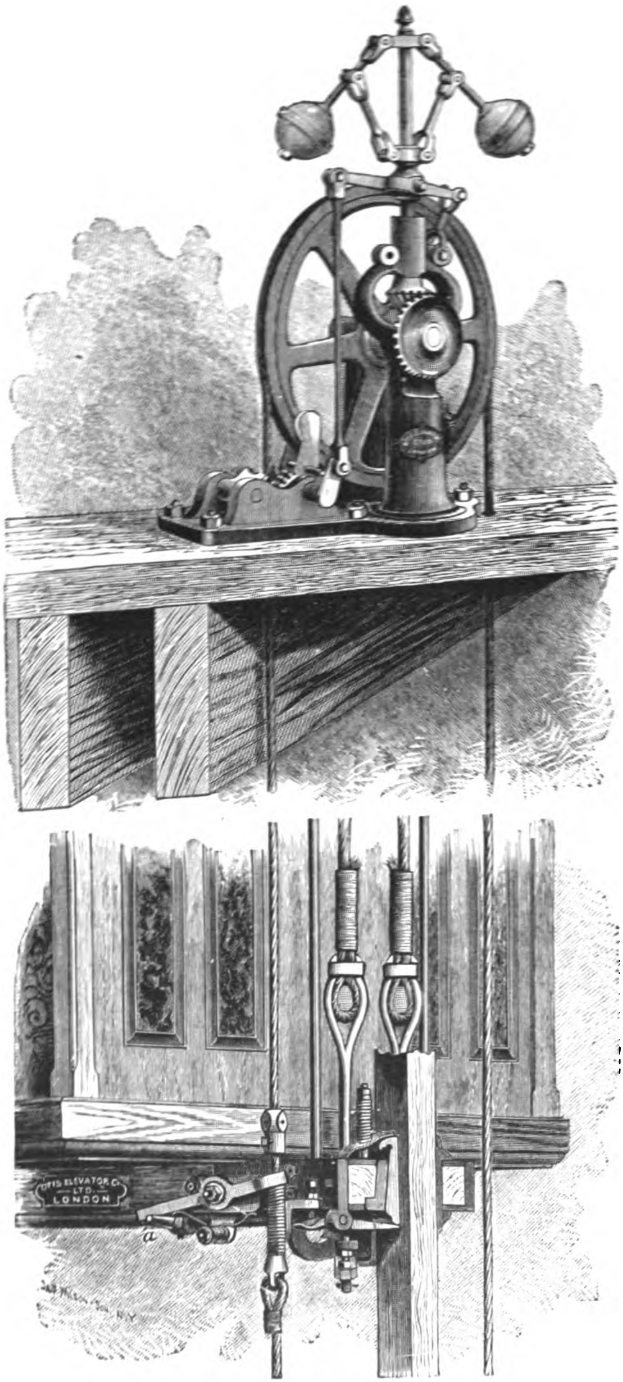

FIG. 196.

pair of gripping levers. Such an arrangement should be fixed to all *direct-acting* hoists, and then an accident like that described at page 302 would be impossible. It is likely, however, that direct-acting lifts, now that their absolute safety is no longer certain, will be less used in future, a first-class suspended lift possessing many advantages. Many suspended lifts of the cheaper kind are, however, very dangerous, and ought not to be used by passengers.

BALANCING ARRANGEMENTS.

As a considerable portion of the load raised each upward journey is a dead load consisting of the weight of the cage and at least part of the ram, it is evident that it would be a very wasteful method, to abstract each time as much energy from our pressure mains as would raise this load, without receiving any return. Hence some system of balancing becomes a necessity, by means of which the constant load may be eliminated from the bill of energy required. There are many methods of balancing, the simplest being by the use of a counterweight; in other words dead weights are made to balance dead weights, or the lift is made fairly balanced when unloaded. For the direct-acting lift a counterweight balance is very easy to arrange.

COUNTERWEIGHT BALANCES.

Fig. 197 shows such a balance. There may be two fixed pulleys at the top instead of one as here shown. If we imagine the cage and ram to weigh w lbs.—supposing the apparent weight of the latter to be constant, which is not true—then if our counterweight is w lbs. there will be balance in the position shown, though in other positions the weight of the chain attaching the counterweight to the cage may introduce discrepancies. In practice it is found best not to balance *all* the dead weight, or in other words, to have the balance weight a little too small so that the cage may readily be lowered when empty. The apparent weight of the ram is *not*, however, a constant thing at all, as has already been observed, and it is easy to counteract this want of uniformity of weight by the varying balancing force due to the chain. As the ram rises out of the water the weight to be balanced *increases*, this increase being *the weight of the column of water* which now occupies the place lately occupied by the ram. Thus if the ram rises one foot, a column of water one foot long and of the same diameter as the outside of the ram must enter the cylinder. The weight of this column is the weight apparently added to the ram.

But at the same time our ram, in moving up one foot, moves the chain so that A (Fig. 197) is shortened one foot, whilst B is lengthened one foot, the balancing effect being the same as if the chain had remained stationary, and two feet more of the same chain had been fastened on at B. To have perfect balance, therefore, two feet of chain should weigh the same as the water column displaced by one foot of the ram. It is not difficult to construct such a chain or rope. Suppose we take a 7-inch ram, the weight of a column of water one foot long and 7 inches in diameter is $0 \cdot 7854 \times 7^2 \times 12 \times 0 \cdot 036 = 16 \cdot 62$ lbs., and our chain or rope must weigh $8 \cdot 31$ lbs. per foot.

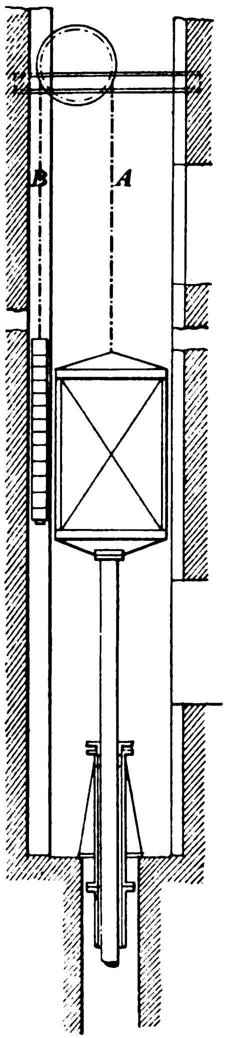

Fig. 197.

In the case of low pressure lifts where the diameter of the ram is considerable, this matter of the varying apparent weight of the ram must be attended to, but in high-pressure lifts, especially suspended lifts, it is not of much importance, though usually given great prominence in books. For instance, take our 5-inch ram, let it be 60 feet long, the extreme variation in weight is only $508 \cdot 8$ lbs., and if the weight of ram and cage is, say, two tons, the variation is about one-eighth of the dead weight to be balanced—a not very serious fraction. If, however, a suspended lift be employed, the ram being vertical as in the direct lift, the length of ram necessary would probably not be more than one-sixth of the lift, and the variation of its apparent weight is of very little consequence. In using counterweight balances it is best to have *two* balance weights and two ropes or chains so as to avoid overhead gear.

One rather grave defect introduced by such balances is the production of tension in the upper part of the ram. The ram and cage seem to be pushed up from beneath, but as a matter of fact, the ram is pushed up from beneath by the water pressure, and pulled up at the top by the balance weight. The water pressure has only to overcome the net load, whilst the balance weight has to overcome the dead weights, much greater in amount. To cite one usual case, the ram is pushed up from beneath by a force of half a ton, and pulled up at the top by a force of 2 tons. It is evident therefore, that the upper part of the ram, together with the fastenings joining the ram to the cage, are in tension, and that the neutral surface dividing the region of tension in the ram from that of compression, is continually altering down and up as the ram moves up and down. The lift is, therefore, really a ***suspended*** one with tension ***in the ram***, which is not so well calculated to bear tension as the wire ropes of a suspended lift. Should the ram or fastenings fail, the cage will be pulled up by the balance weight, and a serious accident will ensue—such an accident has happened, and is referred to later on.

The method of applying a balance weight adopted in the case of the Otis elevator, however, completely obviates this difficulty.

Hydraulic Balances.

It is now the almost universal custom in the highest class of English lifts—especially if served from high-pressure street mains—to have hydraulic balances. Most of those in use are due to Mr. Ellington. The simplest is the ***dead-weight hydraulic balance*** shown in Fig. 198. It consists of a wide cylinder B with a piston A and a very large piston rod bearing cylindric weights W. The pressure water from the mains enters, at P, the annular space above A and presses A downwards, the water below A being forced out at Q into the lift cylinder. In fact A in its downward stroke displaces as much water as fills the lift cylinder, which on the fall of the lift is forced back again under A. It is evident that the arrangement acts as a "diminisher" (a reversed intensifier), for, neglecting friction, the pressure per square inch below A, multiplied by the full area of A, is equal to the intensity of pressure above A, multiplied by the annular area of the upper side of A. In other words, if the pres-

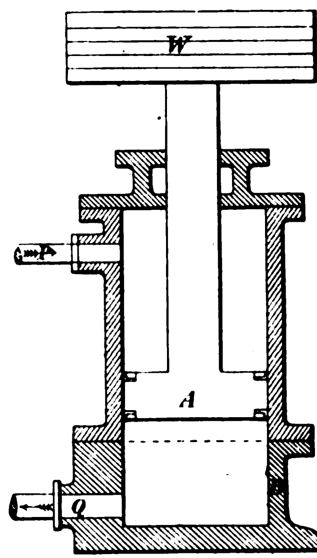

Fig. 198.

sure from the mains is, say, 1120 lbs. per square inch, and the annular area half the full area of A, the pressure in the lift cylinder will be about half 1120 or 560 lbs. per sq. inch. This is a great advantage when using high pressure water, as the ram can now be made of sufficient diameter to give the requisite stiffness.

As the lift descends, A and W are raised, the water above A passing out to the exhaust (not shown); and when the starting-valve is opened—the exhaust being closed—the balance piston and weight fall, raising the lift, ram and cage.

The annular area above A is calculated so that the high-pressure water acting on it will about raise the net load in the cage, the weight W and piston nearly balancing the weight of the lift-ram and cage. Hence (approximately) energy equivalent only to the raising of the *net* load is required each journey, whereas, if no balance were used, the full amount of gross load would have to be raised by the water from the pressure mains. The second useful function of acting as a diminisher is also fulfilled, and the head of water over A increasing slightly as A falls or the lift rises gives a slight increase of pressure; but not usually enough to balance completely the increase of apparent weight of the lift-ram. It will be seen that A is packed by "cup" leathers, and whilst this gives rise to only a small amount of friction, it has the practical disadvantage of all internal packings, requiring the removal of the balance weights and piston before the packing can be renewed.

For this reason balances with external packings are generally preferred. The piston-rod packing is not shown, but may be either of the well-known gland and stuffing-box kind, or leathers may be used.

Movable Cylinder Balances.

The balance shown in Fig. 199 is also often used with high-pressure supply. It consists of a cylinder F, divided into two parts and movable vertically on two hollow rams P and Q. Heavy weights (not shown) are hung from F. The supply enters by Q, and P leads to the lift cylinder. The water in the lower part of F does not pass away to exhaust, but a portion of it simply moves backwards and forwards to and from the lift cylinder, as the cage ascends and descends, sufficient high-pressure water entering at Q to supply the deficiency that would otherwise be left in the upper part of F each downward journey which it makes, i.e. on each upward journey of the lift. On the downward journey of the lift F moves up, and the surplus water in the upper part passes off to exhaust. It will be seen that the arrangement acts as a diminisher; for if the internal cross-section of Q be a square inches, that at P being A square inches, the pressure of water in Q = q lbs. per square inch, that in P p lbs. per square inch, then, neglecting friction, $a \times q = A \times p$, or $p = \frac{a}{A} q$, the lift supply being equal to the high-pressure supply multiplied by the ratio of the smaller area to the greater.

This balance has the great practical advantage of being packed with *external packings*, which, as will be seen from the figure, are of the usual gland and stuffing-box order. Thus the packings of the balance cylinder can be renewed without the necessity for removing any weights or withdrawing a ram, and hence the lift will only be out of use from this cause for a very short time, or the packing can be done after business hours.

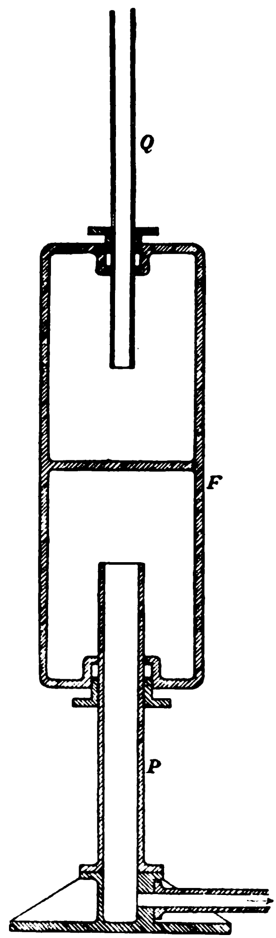

FIG. 199.

A form of balance very similar in principle to the above, but which is more in favour with the best makers, is shown in Fig. 200, where, as before, there is a movable cylinder with balance weights attached to it, moving on two fixed hollow rams. In this case, however, the lower ram embraces the cylinder, instead of entering it, as in the last case. The figure shows clearly the construction and action of the balance, and it will be seen that the packings are external, as before. This is the form of balance usually employed with high-class lifts like the "Reliance" lift, already described. In none of these is any attempt made to balance completely the varying weight of the ram due to varying immersion, though the varying head of water in the balance cylinder, as has already been pointed out, effects a small change of pressure tending in the right direction.

In the form of balance shown in Fig. 201, this varying apparent weight is balanced completely. The lift cylinder is in communication, by the pipe H, with the balance cylinder M, below which is a wider cylinder N. There is a piston to each, the two pistons

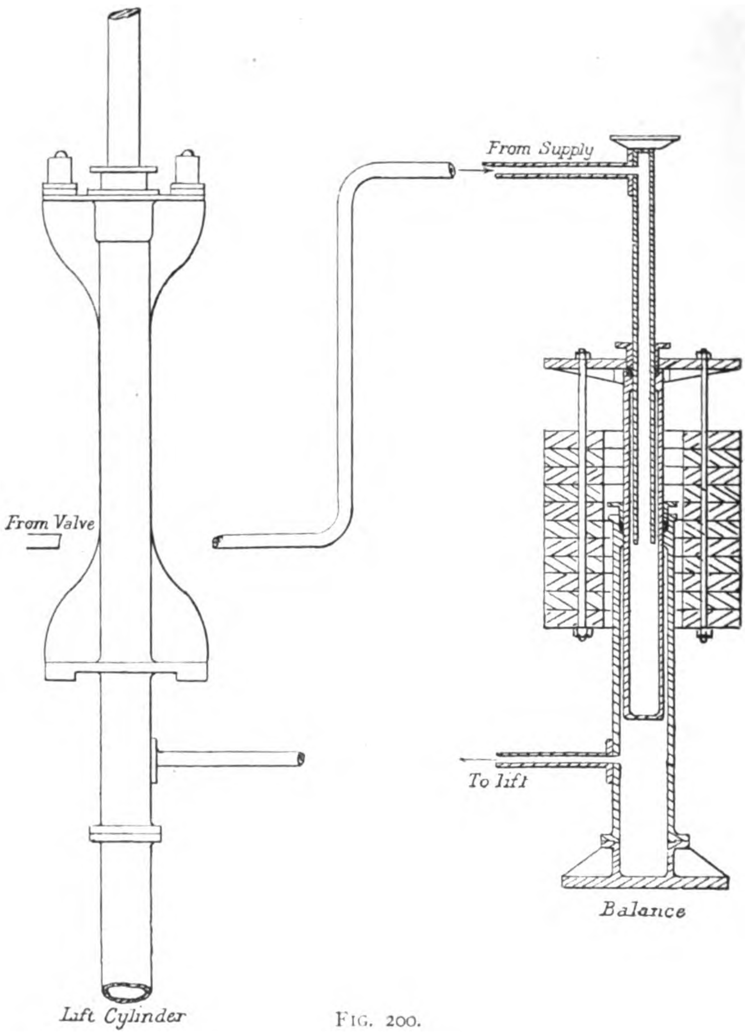

FIG. 200.

being connected by a common rod D. The capacity of the annular space E E, below the upper piston, is, when that piston is at the top of its stroke, equal to the displacement of the lift ram. The annular area L of the lower piston, round D, is sufficient, when subjected to the working pressure, to lift the *net* load and overcome friction; whilst the full area of the upper piston G is calculated, when subjected to the same pressure, to balance approximately the weight of the cage and ram.

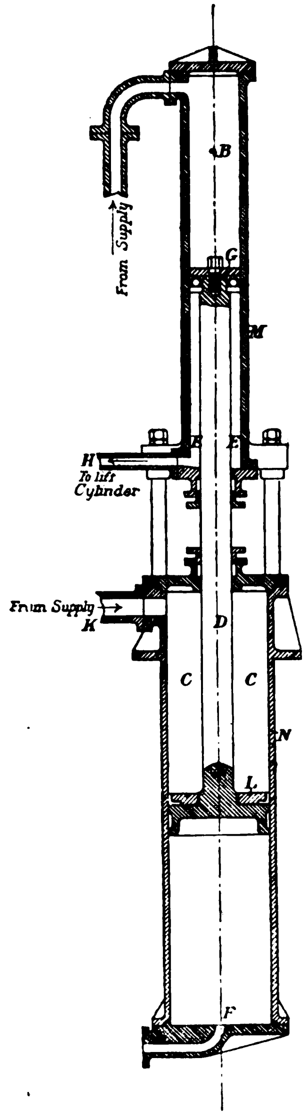

FIG. 201.

Suppose the lift ram to be at the bottom of its stroke; then, when the starting valve is opened by the attendant in the lift pulling the hand-rope, pressure water is admitted to the two cylinders at B and K. The pressure on the two pistons G and L causes them to descend, forcing water from E E to the lift cylinder through H. The lift-ram ascends and in doing so gets heavier, but the two balance pistons are descending, and the weight of water over each increasing, so that the pressure through H is increasing just in the right proportion if all the areas and sizes have been properly calculated. When the exhaust is opened, the *water from* C C *only* passes away, the weight of the ram and cage forcing the water from the lift cylinder back into E E. To make good leakage, the pressure water can be admitted by F under the lower piston, thus raising it. The lift ram being at the bottom of its stroke, water will flow past the leathers into the annular space and supply the deficiency.

In usual working the pipe F is left open to the atmosphere. By closing it, however, a partial vacuum may be formed under L, which may be utilised on the next ascent of the lift to assist in raising the load, supposing that load to be a little in excess of the normal amount for which the lift is designed. If the lift is used for *lowering* goods only, this vacuum may be utilised, in some cases, to raise the empty cage without the expenditure of any supply water at all, though this is very seldom done, as it is not often that the packings are sufficiently air-tight for this purpose.

It will be seen that this balance, unlike the others, acts as an *intensifier*; for, supposing the supply to act only on G, the pressure of the lift cylinder acting at E E, or on the annular area of G, must be greater than the supply, to say nothing of the pressure on L. Hence this form of balance is usually employed with lifts worked by a *low-pressure* supply, and when direct-acting lifts are employed, as in that case only is it necessary to take the varying weight of the lift ram into account.

The balance, as will be seen, is furnished with *internal packings*, and is somewhat complicated; hence it is not so much used as some of the others described, though many writers refer to it as if it were the only form of hydraulic balance worthy of notice.

VALVES FOR LIFTS.

The principal or controlling valve, by which the lift is raised and lowered at pleasure, is usually a slide valve; this form of valve giving only a gradual opening or closing, and hence avoiding shock, relief valves being rarely necessary for high pressure work.

The form of valve used by some makers is shown in Fig. 202,

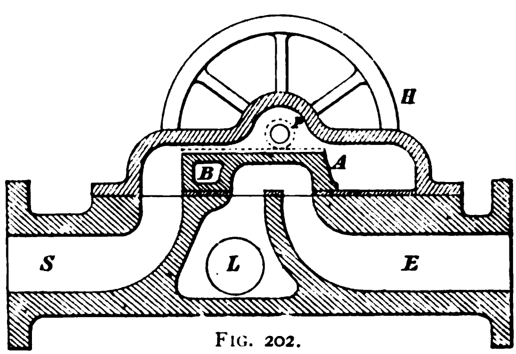

FIG. 202.

where A is the valve worked by a rack on its upper side, engaging a pinion P, which is fast on the axle of the rope-wheel H. An endless rope engages H, and one portion of it passes up through the cage. When H is turned by the attendant pulling this rope, the slide A is removed as required. In the position shown in the figure, the lift branch pipe L is in communication with the exhaust E, and hence lowering takes place. When the slide is moved over to the right, the supply S and lift L communicate, exhaust being closed, and hence the lift rises. When the slide is moved over till B covers the port above L, the lift is stationary, as water cannot enter nor leave L. Another advantage of using a slide valve is that it may be only partially opened, thus checking the motion of the lift and preventing too rapid descent. The slide A is usually of gun-metal, and works on a gun-metal seat. Means are provided for closing the valve automatically when the cage reaches the proper position; thus, should the attendant neglect his duty, over-travel is prevented.

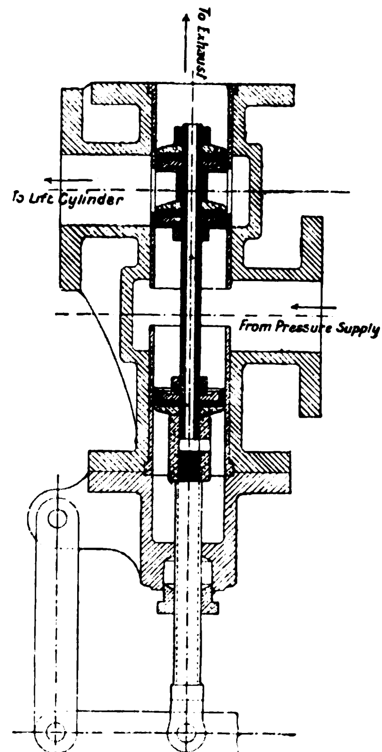

FIG. 203.

In low-pressure lifts an air-chamber is usually inserted between the supply and the starting valve, to guard against shocks.

Another form of slide valve, more in favour with the best makers, is shown in Fig. 203. It will be seen that this is an equilibrium valve of the piston type, and the figure shows clearly both the construction of the valve and the way in which it is moved. In the position shown the lift is stationary. If the valve is moved downwards, communication is opened between the lift cylinder and exhaust; if moved upwards, pressure supply has access to the lift.

TELESCOPIC LIFTS.

These lifts are intended to obviate the difficulty of sinking a deep bore hole for a direct-acting lift, and yet not have a suspended cage. They were more noticed some years ago, before suspended lifts came into repute, than at present. The ram is made telescopic. At the bottom of each length a projection engages the next length of ram, within which it works, pulling it up, and it in turn pulls up the next, and so on. This catching of each length at a particular point in the stroke causes an unpleasant jerk, and the number of packing glands greatly increases the friction. Hence these lifts are not much in favour. Richmond's method of causing all the rams to advance together, the pressure water being allowed to pass, by a special passage, from the inside of one ram to the base of that working within it, is very ingenious, and prevents the jerky motion already referred to. The telescopic lift has too many internal packings.

Rack lifts have also been used, the pressure water being used to drive a small motor, which actuates a pinion working into a rack supporting the cage of the lift. These lifts have not, however, come into general use.

LIFT ACCIDENTS.

It has been shown that a really good lift, constructed on proper principles by a firm of repute, is as safe as any other means of locomotion in common use. In fact there are makers of first-class lifts who have supplied thousands of them, and who can boast after many years' experience that *no accident* of a severe or fatal nature has ever happened through using one of their lifts. But lift accidents *do* occur, and mention may be made of one or two notable instances, which seem to serve as warnings to owners and users of lifts.

An accident happened some years ago to a lift in the Grand Hotel, Paris, where the connection between the cage and the ram gave way, and the cage was dragged upwards by the heavy counterweight balance, causing the death of several people, not by the cage *falling*, as timid people usually dread, but by its being dashed against the top of the lift well-hole. This accident shows the one defect of a balance of this kind, but it is quite possible by good workmanship and careful design to render such an accident well-nigh impossible, even with such a balance. It will be noticed that this accident happened to a *direct-acting* lift.

Another sad accident of a somewhat unusual character is of special importance to users and repairers of lifts.

On the 25th of February, 1895, Mr. T. C. Read, the assistant chief surveyor at Lloyd's Registry, was killed by the falling of a lift at the offices of the Registry, White Lion Court, London. The lift was a direct-acting one of 45 feet stroke, with a hydraulic deadweight balance, similar to that shown in Fig. 198. A by-pass pipe was provided by which the pressure water could pass direct from the mains (of the London Hydraulic Power Company) to the lift, when, as on the fatal occasion referred to, the balance required re-packing. As it turned out, it is a pity that this pipe was ever put in, as, had it not been provided, the lift must have remained idle whilst repairs were in progress. On the date referred to, the packings of the hydraulic balance were being renewed, and the men were actually engaged on the work whilst the lift was working with water direct from the mains. Mr. Read had entered the lift, which ascended as far as the first floor, when the pipe joining the starting valve to the hydraulic balance suddenly *broke*, the water escaped rapidly, and the lift fell, Mr. Read being killed.

The piping had been tested to 2500 lbs. per square inch internal fluid pressure, whilst the pressure on this occasion was only 700 lbs. per square inch. It is believed that the pipe had been subjected to some kind of transverse stress, probably by allowing a weight to fall upon it, and the sudden shock of a passenger closing the valve quickly produced the fracture. The lessons of this accident are, *first*, avoid balances with internal packings, as the removal of the balance weights in this case may have had something to do with the accident, or, at any rate, with a packing of the external kind, renewal would have been the work of a few minutes only, and would probably have been undertaken at a time when the lift was not required; *second*, do not allow the lift to be used at all when undergoing repairs; *third*, have a lift with proper safety gear, and preferably a suspended lift; and *fourth*, do not allow passengers to work the lift themselves, but have an attendant who will carefully move the valves, and save his wages by the longer life of the lift. If this is absolutely impossible, have starting valves and relief valves or air-chambers of a peculiar kind on the lift, which will make it impossible to produce shocks by sudden stoppage of the water moving under high pressure.

People sometimes fall down the lift well on account of a door not having been properly fastened. This can hardly be properly termed a lift accident. Usually the doors of lifts are fastened from the inside, and can be opened and closed only by some one in the lift. Devices for automatically closing and opening the doors by the motion of the lift have been arranged, one form of which will now be referred to.

Automatic Gate for Lifts.

To prevent the kind of accident referred to, the wells of lifts which are not in charge of an attendant should be fitted with automatically closing doors. There are many of these, but that of

Mr. John Botterill, of Leeds, shown in Figs. 204 and 205, is one of the most successful. The doors of the lift well are closed and opened by the motion of the lift.

As will be seen from the illustrations, a curved plate or cam of wrought iron, shaped somewhat like a carriage spring with exaggerated curvature, is attached to the side of the lift cage. To the framing of the lift well-hole, or to the slides which guide the cage, are attached, about three feet above each floor, a pair of cross-bars or rails, on which runs a little carriage with four wheels or rollers,

FIG. 204.

FIG. 205.

resting two on each rail. To the back of the carriage, or side nearest the lift, is attached a bowl-shaped piece of iron, whilst on the nearer or outer side there is a small chain pulley. One end of a chain is attached to the lift framing, the chain is passed round the pulley just mentioned, over another pulley on the framing, then over two pulleys fastened to the ceiling, and from thence to the lift gate.

Fig. 204 shows the lift descending, the curved cam on it just coming into contact with the bowl on the inside of the carriage. As the lift descends, the carriage is pushed backwards by the cam, the chain is pulled in, and the gate is fully opened at the instant when the carriage reaches such a point of its path that the floor of the lift is level with that of the building, as shown in Fig. 205.

On the further descent of the lift, the weight of the gate causes it to close as the carriage follows the upper portion of the cam. This action is repeated at every floor of the building.

It should be noticed that the gate does not begin to open till the lift has half filled the doorway; further, that the motion of the gate is twice as fast as that of the lift, and could be made even faster by the use of more multiplying pulleys; but it must be pointed out that the introduction of further pulleys to secure quicker opening and closing would, other things being equal, necessarily increase the force required to move the apparatus.

Fig. 206.

The gate is sufficiently high to prevent anyone from falling over it, and it cannot be moved by anyone in the lift. It closes without shock, and requires no floor space, with little danger of anyone being caught between it and the lift. These are matters which seem to render the arrangement a good one.

Part of the weight of the gate can be counterbalanced, and thus little additional force is required to work the lift.

The opening arrangements here shown at the side of the lift can readily be placed at the back, so as to be quite out of the way, or the reach of any mischievous person; or they can be completely covered over, as indicated in Fig. 205.

A neater and simpler arrangement, patented by Mr. Botterill, is shown in Fig. 206, in which the little bogie is dispensed with; a pulley at the end of a vibrating bar being used instead. An opening resembling points on a railway is also provided on each side of the cam; this aperture can be opened from the inside of the lift, so as to allow the pulley to pass through when it is desirable to pass a particular door without opening it. The aperture is closed automatically by a spring.

Efficiency of Hydraulic Lifts.

The efficiency should always be calculated for a complete cycle or journey. It is evidently $\frac{W \times H}{w \times 2 \cdot 3 \times p}$, where W is the net load raised through the lift H feet, *w* being the net weight of pressure water used, and *p* the intensity of the pressure in lbs. per square inch, W and *w* being in lbs. Direct-acting lifts with good balancing arrangements may give 90 per cent., and suspended lifts from 70 to 85 per cent. efficiency. Direct-acting lifts without counterbalances have a low efficiency. For instance, one with a 5-inch ram, using water at 150 lbs. per square inch, and raising a net load of half a ton, gives, by the above rule, about 30 per cent. efficiency. The rule is also applicable to hydraulic cranes.

In regard to these, only a general rule, in which *p is the actual pressure in the cylinder*, has been given. The highest authorities state that the efficiency varies so widely with the type of crane, the state of packing, and the number of pulleys, that no general rule, beyond that given in the text, is possible. Though some writers give data making the efficiency depend only on the number of pulleys, these are misleading. The efficiency probably varies from 50 to nearly 90 per cent.

LIFTS FOR VEHICLES.

Hydraulic Wagon Lifts.

These lifts usually present few features requiring special notice. A simple form of the lift is shown in Fig. 207, where the platform bearing the wagon is attached directly to the top of the ram, the dead weight of the lift being balanced by a counter-weight as shown.

The form adopted at the Somers Town (London) goods station of the Midland Railway is of a somewhat novel character. It is shown diagrammatically in Fig. 208, where the upper triangular portion of the figure represents the lift, which is supported by three rams, each working in its own cylinder, the central one B being the largest. Ram A is always connected to pressure, and simply balances the weight of the lift itself. Ram C is used to lift loads of 5 tons or under, B being then in communication with a return water tank, which is at a higher level than the lift, to keep its cylinder always full of water. When a load of *over* 5 tons is being lowered, the water which is under C is forced back into the pressure mains. Thus when, as at Somers Town, the lift is used most frequently to lower full wagons and lift empty ones, no pressure water is ultimately wasted. If the descending load is less than 5 tons, C must communicate with the return water tank, and in this case pressure water is lost. Ram B alone, if under pressure, raises loads up to 15 tons. It must always discharge its water into the tank. Rams B

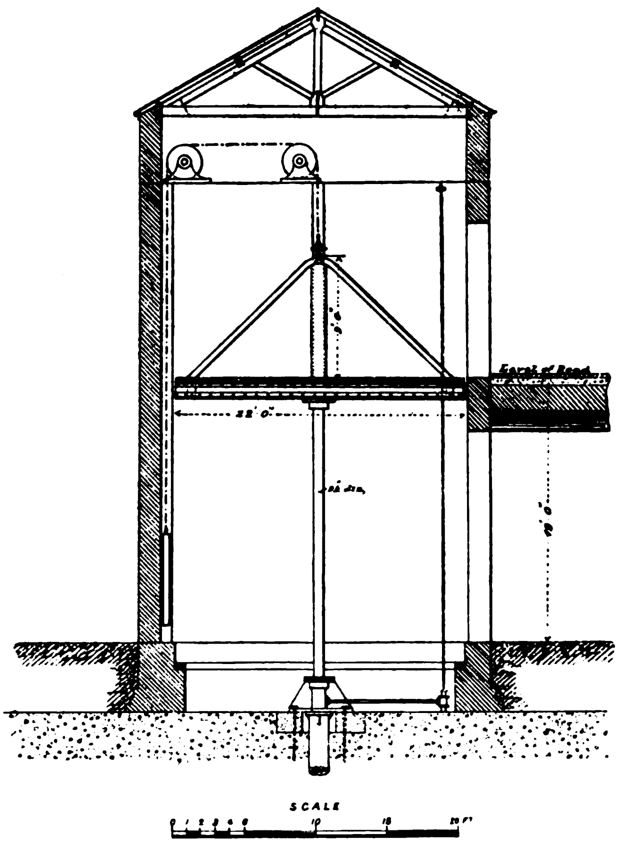

FIG. 207.

and C together lift, under pressure, 20 tons. Thus interesting combinations are possible by this system, with a considerable saving of pressure water.

Coal-tips on a similar principle are used at some of the stations on this railway, a truck being elevated and "tipped" by the action of three rams similar to those described.

Lifts of the Glasgow Harbour Tunnel.

These lifts, constructed by the Otis Elevator Co., Limited, and on their well-known principle, are worked from a complete high-pressure hydraulic installation, the maximum load being 6 tons and lift 72 feet. There are six elevators or lifts, three for raising and

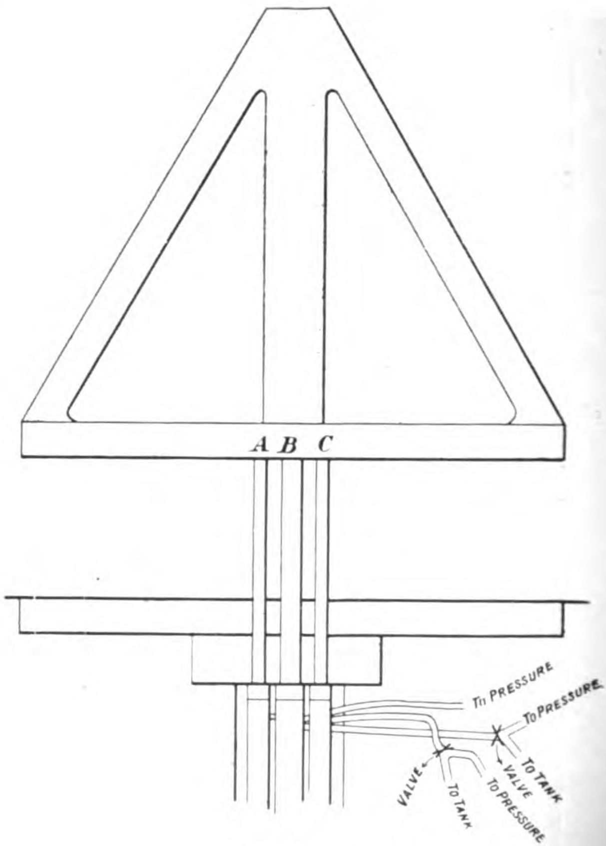

Fig. 208.

three for lowering vehicles for each of the two traffic tunnels, there being a separate tunnel for passengers. The lowering lifts can also be used for *raising* loads if required.

The elevating cylinders are 13 inches in diameter with a minimum thickness of 2½ inches of cast iron, and have been tested to 1800 lbs.

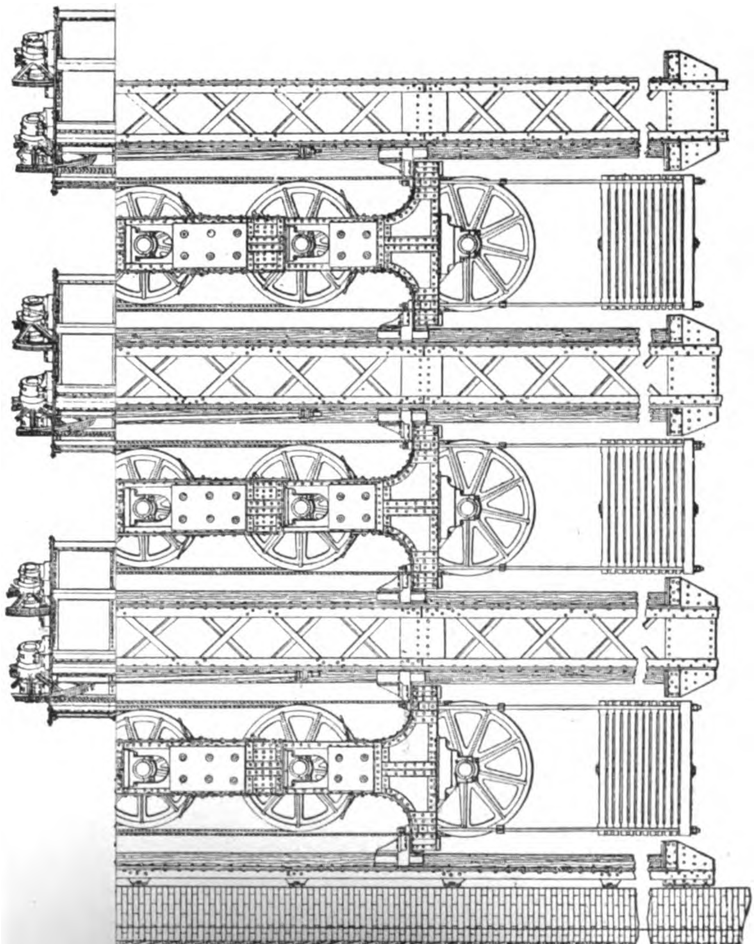

FIG. 209.

[*Face page* 308.

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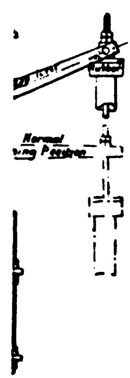

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per square inch. The lowering cylinders are 11½ inches in diameter and of similar construction. The stroke of the working rams or pistons is one-sixth of the lift, the magnifying gear consisting of three

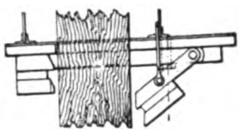


FIG. 210.

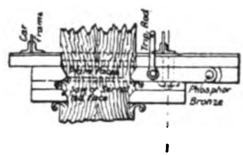


FIG. 211.

tandem sheaves mounted in a travelling frame, as shown in Fig. 209. The main and counterbalance sheaves are placed overhead as shown in the illustration, and beneath the travelling sheaves balance weights are suspended by steel rods; the travelling sheave frames having also slabs of cast iron bolted to them, thus bringing the total balance weight up to 16,000 lbs. to partially balance the dead weight of the lift cars. The main suspending rope is ⅞ inch in diameter, consisting of six strands each of 18 steel wires, the strands being wound on a hemp core. Each rope is tested up to a load of 24 tons, and there are four ropes to each lift attached to clevises and adjustment rods. Each clevis-screw is attached to opposite ends of a double-fulcrumed lever, which acts on the safety-grips should a rope break.

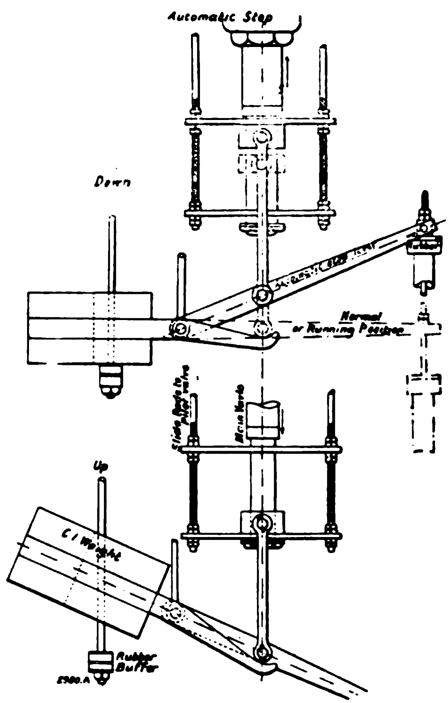


FIG. 212.

These safety-grips have teeth which engage the pitch-pine guides sideways, the guides being 5¾ inches wide and the gripping teeth only 5 inches wide when in action. They are drawn into action, if the speed becomes excessive, by a governor placed on the overhead framework.

Fig. 210 shows the grips in their usual position, whilst Fig. 211 shows them in action. These have been subjected to the severe test of a load of 31,851 lbs., which after a free fall of 13½ inches, was arrested by them in a total drop of the car of 2 feet 10 inches, the average resisting force of each safety grip being 26,415 lbs.

The main valve works on the differential principle, the pressure acting constantly on its lower area which is half the upper area. When the top is open to exhaust, the valve rises; when the top is put under the pressure it falls, these operations being controlled by a pilot valve. An automatic balanced lever arrangement is provided to stop the car, when it has reached the end of its journey, as shown in Fig. 212; the stop motion lever acting on the pilot valve, and if that fails to respond, on the main valve, moving the latter into the position required to bring the car to rest.

For further details of the lifts the reader is referred to issues of 'Engineering' for May and June 1895.

XXIII.

HYDRAULIC CANAL LIFTS, AND GRAVING DOCKS.

In Fig. 213 is shown probably the most interesting canal lift in this country. It is the lift of Messrs. Clark and Standfield, erected under the superintendence of Mr. Leader Williams, at Anderton, on the river Weaver, in Cheshire. The canal is 50 feet above the river, and the lift is constructed to raise boats from the river to the canal, and lower boats from the canal to the river—in fact, both operations are performed at the same time. There are two great wrought-iron tanks, each 75 feet long and 15½ feet wide, each carried on the top of a ram 3 feet in diameter, which can rise and fall in a hydraulic press as shown by the right-hand figure, which is a cross-section. It is needless to say that the girder work necessary to carry this immense trough, and its load of water and boat, must be very strong and well designed. Each ram has to bear a load of 240 tons, giving a pressure of about a quarter of a ton per square inch in the press. Each end of a trough is furnished with a gate. When a trough is up, one gate is opened and the trough forms part of the

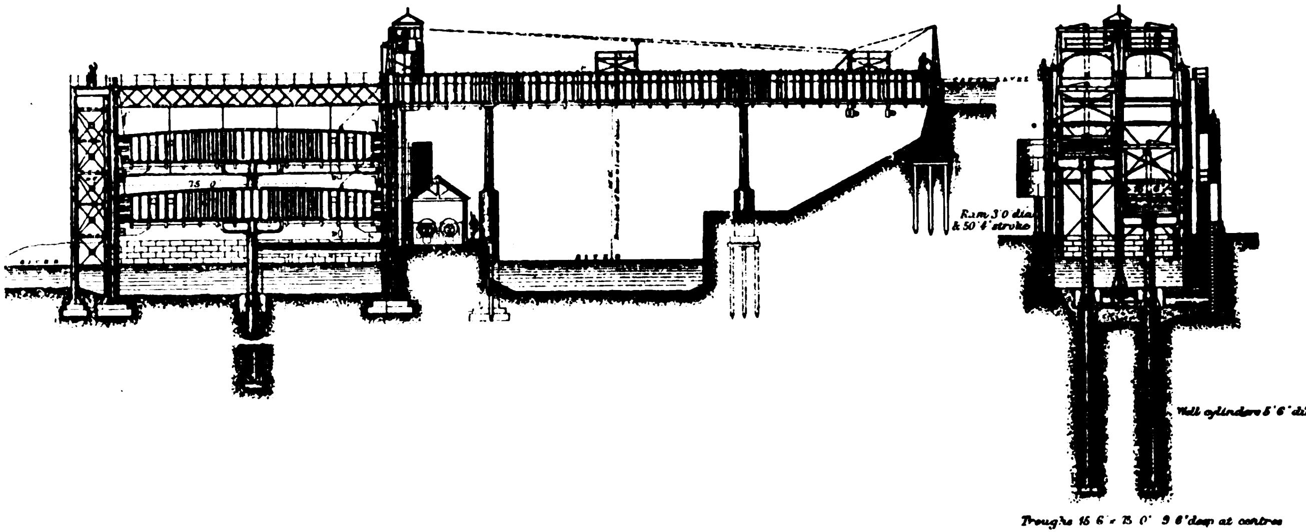

75 0
Ram 3'0" dia & 50'4" stroke
Well cylinders 5'6" dia.
Troughs 15'6" × 75'0" 9'6" deep at centre

FIG. 213.

aqueduct or canal, a barge floats into it, and the gate is closed. The trough is lowered, and when it reaches the lower level the other gate is opened and the trough forms part of the river. The full details of how the great presses are sunk into the ground, how a tunnel has been constructed to enable the packing-leathers to be examined, and how the columns acting as guides to the troughs are stayed and supported, may be read in Mr. Duer's paper in the 'Proceedings of the Institution of Civil Engineers,' vol. xlv. The reader will see that, in a case of this kind, the total weight to be lifted depends only on the height of the water in the trough. Suppose we put a boat into any vessel containing enough water to float it, then if we take out this boat and put in another, whether of the same or a different weight, if sufficient water be added or taken away to bring the level to what it was before, the total weight of boat and water is the same. This follows from the well-known fact in hydrostatics that a floating body displaces its own weight of the fluid in which it floats; hence, if a heavier boat be put in, it displaces more water, the excess being just equal to the excess of weight of the new boat over the old. In fact, if the level of the water be kept constant, the total weight is the same whether there be any boat in it or not. This suggested to the designers an ingenious method of supplying part of the energy required to raise the lift.

It will be seen from the section that one trough A rises, whilst the other B falls, there being a means of communication between the two presses. Suppose A to be up and forming part of the canal, B being down. It is of no consequence whether each contains a boat or not. Suppose there is 5 feet of water in A, and only 4 feet 6 inches of water in B, it is evident that the heavier weight A will fall, and the lighter B rise; the ram under A forcing its water into the press under B, since the pressure on the former is the greater. B will go on rising till the increased weight of its ram, as it emerges from the press, together with the diminished weight of A's ram owing to greater immersion, balance the extra load A. In the actual case at Anderton, about eleven-twelfths of the work required to perform one lift is obtained by having the level of water 6 inches less in the ascending trough than in the descending one; the remaining one-twelfth is obtained from the accumulator, part of which will be seen in both views.

The accumulator has a stroke of 13 feet 6 inches, with a ram 1 foot 9 inches in diameter. The engine and accumulator alone are capable of lifting one trough and its load, which may either be two small barges of 40 tons or one large one of 100 tons. The depth of water in the troughs is regulated by siphons, there being twelve to each trough, which reduce the level of water in the ascending trough to 4 feet 6 inches and keep the water level, and hence the load to be lifted, always of the proper amount, whatever the weight of the boat may be. The gates at the ends of the troughs are lifting gates, moving into position near similar gates in the canal and river. The gates are easily moved, when the water is allowed to pass through a valve into the space between the two gates, i.e. the gate closing the aqueduct and that closing the end of the trough fitting it.

The lift is capable of taking eight barges up and eight down per hour, and it can raise one load in three minutes. If this were done by locks, it would take about one and a half hour per barge, using, of course, a large amount of water. The lifting of one trough separately by the engine and accumulator alone takes about half an hour. Without the aid of the accumulator the 6 inches of extra depth of water in the descending trough will lift the load within 4 or 5 feet of the top. The waste in the operation—for it must not be supposed that there is *no* waste, although the ascending and descending weights may be equal—is, first, 6 inches of water over the area of one trough falling 50 feet = 1,800,000 ft.-lbs. of energy. Second, the accumulator raises one trough through, say, the last 4 feet 6 inches of its lift = 240 tons raised through 4½ feet, equivalent to 2,419,200 ft.-lbs. The total waste in one operation is therefore 4,219,200 ft.-lbs. If the same operation were performed by locks under favourable circumstances, they would require 14½ feet depth of water of the area of one lock, falling 50 feet, equalling an expenditure of 51,500,000 ft.-lbs. of energy, about twelve times that required by the hydraulic lifts. Probably, however, the saving in time is the most important item, if the canal has a plentiful supply of water.

The designer has stated that if he were designing another similar lift, he would use only one ram and trough, with an engine and large accumulator, performing the operation directly, thus greatly reducing the first cost. It will be seen that in the double arrangement each trough in its turn acts as accumulator to the other.

Canal Lift at Les Fontinettes.

A somewhat similar lift has been erected at Les Fontinettes, near St. Omer, in France, by the same firm, to replace a flight of five locks, with a total fall of 43 feet. A general view of the lift is shown in Fig. 214. The troughs are about twice as long as the last, and of a greater depth, each supporting ram being 6 feet 6 inches in

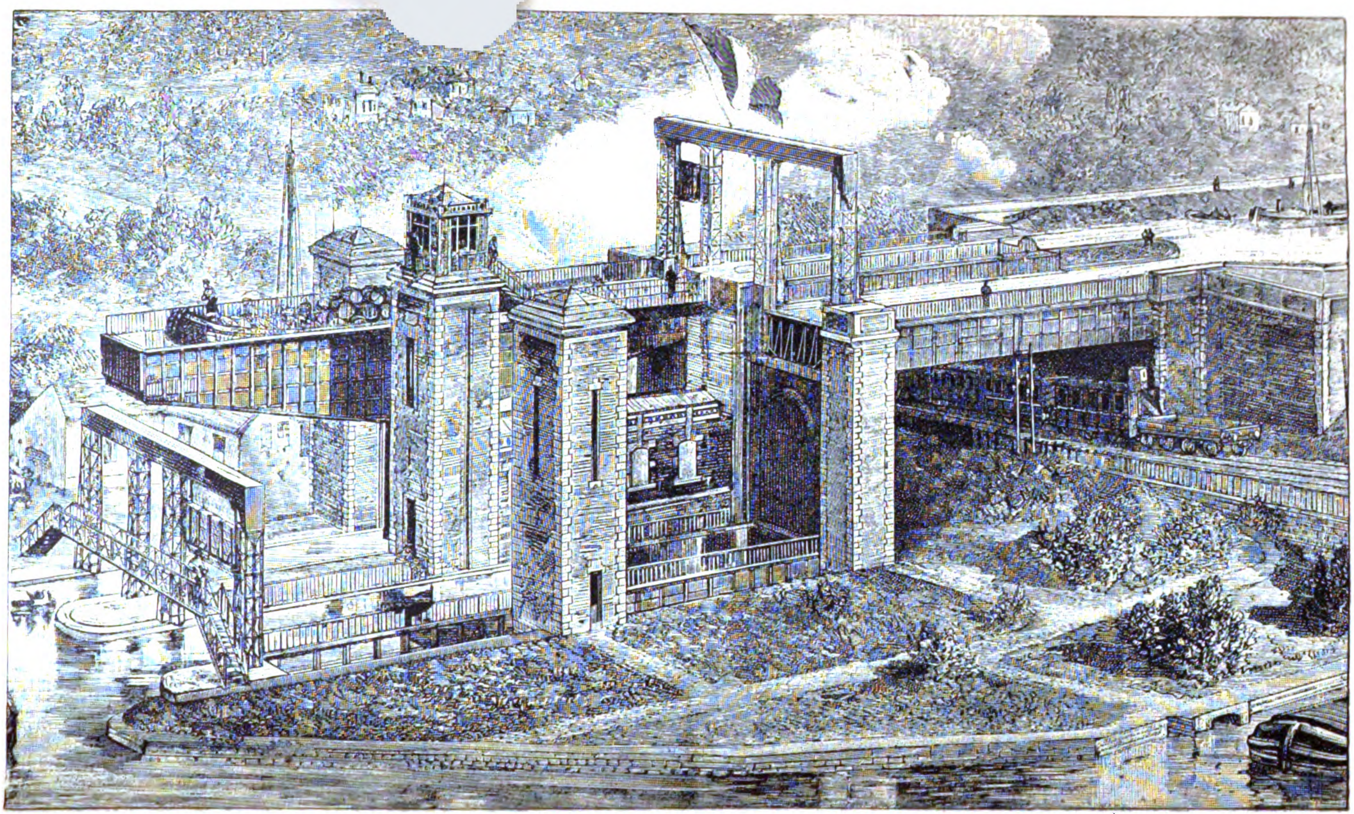


Fig. 214.

diameter. There are compensating reservoirs provided, from one of which water flows to the descending trough, thus increasing its weight just by the amount required to balance the *diminution* in apparent weight of its ram owing to increasing immersion in the press.

In the illustration the further trough is shown at the top, forming virtually a part of the high-level canal; the nearer trough is at the bottom, and forms part of the lower reach of canal. The gates of both troughs are lifted, a barge having just left the lower, and another just entered the upper trough. When a barge is in position, the gates are lowered, and each trough becomes an isolated tank. The excess weight required in a descending trough is supplied by stopping the rising pontoon a little *before* it arrives at the level of the canal, so that when the gates are opened it receives about 16 inches more than the normal depth of water. This excess is run off when the pontoon reaches the lower level by stopping it a little too soon, and when the gates are opened the extra water flows away. As it is impossible to stop the pontoons exactly at the proper place, and also to make good leakage, as well as to open and close the gates, a small accumulator is provided.

The gates are made watertight by filling the space between the pontoon and the aqueduct with an indiarubber hose, which can be inflated with air to a pressure of 22 lbs. per square inch. This joint is made tight after the trough reaches its position. Central guides are employed to cause a vertical motion of the troughs.

There is a further improvement: the descending trough does not descend into water—which made the trough get light too soon at Anderton—but into a dry chamber, and only becomes a portion of the canal when the gates are opened. Thus the falling of one trough can lift the other the whole way except for friction and leakage, which are provided for by the accumulator. Although the troughs have more than twice the capacity of those at Anderton, a single operation causes a loss of only 20 tons of water from the higher to the lower level, equivalent to about 2,000,000 ft.-lbs. of energy.

La Louvière Lift.

It is not necessary to refer in detail to other lifts of a similar class. The lifts by the same firm, designed to connect the Mons and Condé canal with a branch of the Charleroi and Brussels Canal in Belgium, are among the most famous.

The work involved first of all the construction of $9\frac{1}{3}$ miles of branch canal to connect the two main canals, the difference of which is no less than 293 feet 6 inches, with very little water available. The barges navigating the canals are 128 feet long and require 8 feet of water, so that the waste by locks would be very great. The plan finally adopted was to overcome part (76 feet) of this difference of level by six locks, with four canal lifts each of about 50 feet stroke, and resembling, in general features, the Anderton or Fontinettes lifts, to deal with the remaining levels.

Owing to the failure of one of the cast-iron presses at Anderton the presses of these lifts are completely surrounded by steel hoops shrunk on very tightly, with a circular supply pipe from which the pressure water enters each press by smaller radiating pipes.

Clark and Standfield's Ram-balancing Arrangements.

One of the methods adopted by this firm for balancing the change due to the varying displacement of the rams, where the falling of an accumulator ram causes the rise of another ram, has already been noticed. But they have adopted other very ingenious methods. Thus, in one case, a ram rises vertically above the accumulator, passing watertight into a tank of water. The part of it in the tank is always covered with water, and, as the accumulator descends, its increased weight compensates for the displacement effects of all the other rams in the system requiring balancing. Another method they have adopted is to have a tank of water as part of the load of the accumulator, the water-level in it being kept the same as that of a neighbouring fixed tank by means of siphons, so that the load on the accumulator increases as it falls, thus balancing the diminution of weight of its ram. By properly varying the shape of this tank they can get, within limits, any required variation in the pressure of the water supplied.

This is a very ingenious method of balancing, though for many operations this and kindred methods are not so much considered as formerly.

Hydraulic Graving Docks.

By the aid of hydraulic power the old and tedious method of docking a ship for repairs may be superseded by a method both rapid and comparatively inexpensive. Instead of floating the ship into a reservoir which must be pumped dry, the ship is lifted out of water by hydraulic presses, acting on a grid and pontoon, over which the ship had previously been placed. Less time is thus required for the operation, and a smaller capital outlay than would be necessary for an ordinary dock of the same capacity.

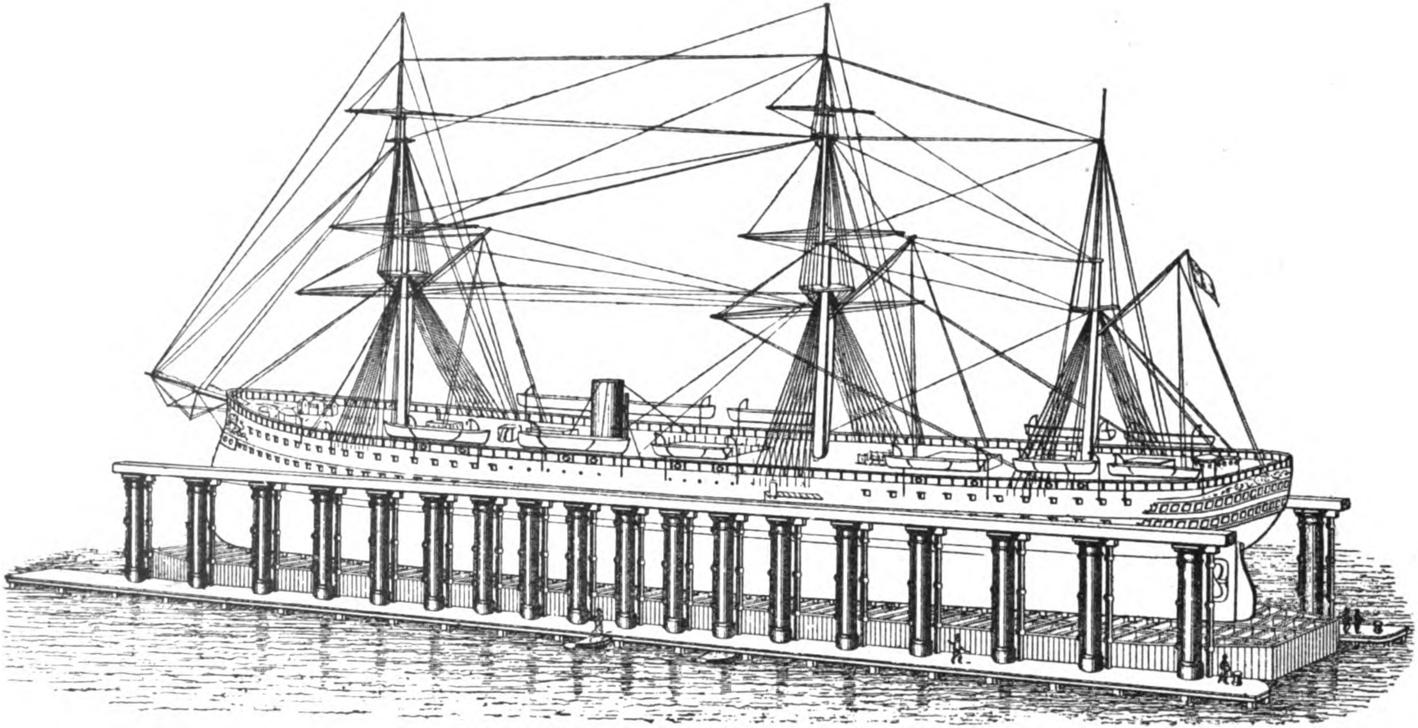


FIG. 215.

The largest dock of this kind yet erected is that designed by the late Mr. Edwin Clark and erected by his firm at Bombay. The length of the dock is 400 feet, clear width 88 feet, and it will lift vessels up to 6500 tons, drawing 30 feet of water. It consists of two parallel rows of cylindrical cast-iron columns, shown in Fig. 215, eighteen columns to each row, their distance apart gradually increasing from 18 feet at the middle to 24 feet at the ends. The clear distance between the rows is 88 feet. Each of the columns is

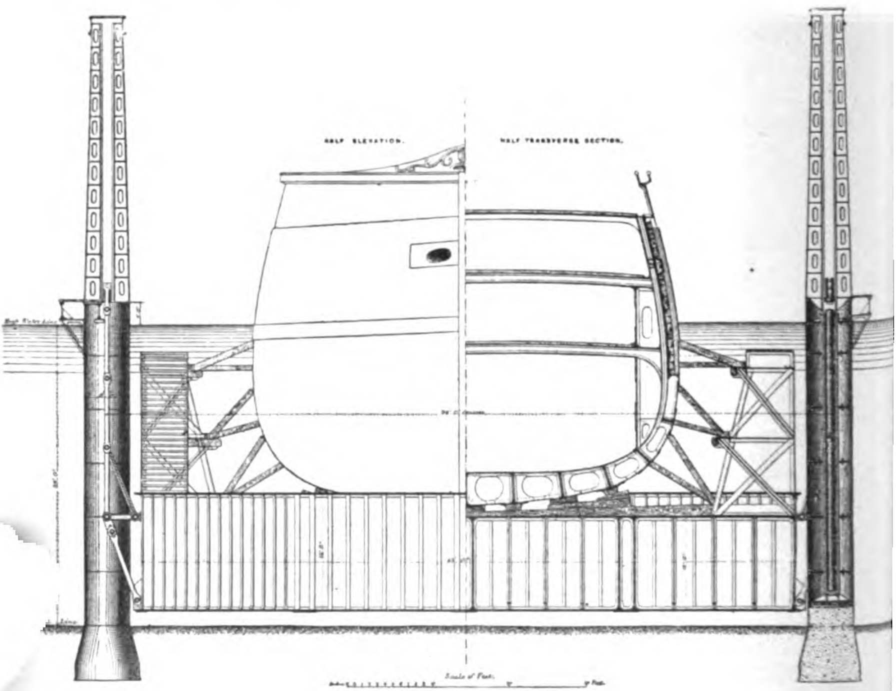


FIG. 216.

7 feet 6 inches in diameter at its base, and 6 feet 6 inches at the top; some of them are 109 feet long, their tops only appearing above high water. They are firmly fixed to the rock, and in cement concrete. In each column are enclosed two hydraulic presses which rest on the rock, having solid rams 14 inches in diameter and 35 feet stroke; these rams support a cross-head working in slots in the top part of the column (see Fig. 216). The column therefore supports none of the weight of the ship, which is borne by the rams.

From the ends of each cross-head are suspended two rods, one on each side of the column, and to the lower extremities of the corresponding rods of two opposite columns are attached lattice girders, so that the rams of two columns support two girders. The girders are 10 feet 8 inches deep, and there is no longitudinal connection between them. They form what may be called the "grid." The tops of each row of columns are connected by two girders 2 feet 7 inches deep; on these run four 25-ton travelling cranes for raising the rams, &c., for repair. There is also a gangway two feet above high water for workmen and seamen. Extra columns are provided at each end of the lift furnished with bollards or capstans for working vessels from.

The pontoon or great iron raft consists of a framework of cross girders, divided into thirty-six water-tight compartments, in the bottom of each of which is a valve. The pontoon weighs 1610 tons; it is 380 feet long, 85 feet broad, 9 feet 6 inches deep at the outside; the upper side, which is not covered in, slopes gradually towards the centre, where the depth is 6 feet 6 inches. This pontoon, when completely emptied of water, can support 6500 tons, and it is fitted with keel-blocks, shores, and the most approved appliances for cradling ships. The valve-house is on the pier head; it contains three small wheel valves, each of which controls one of the three groups into which the presses are divided. Lifting power can thus be applied at either end or at the side of the pontoon as required. There is also a means provided, in the valve-house, for cutting off the action of the presses in any pair of opposite columns, so that if one, or even three or four presses fail, the operation may continue.

The method of docking a vessel is as follows:—The pontoon is placed in position on the grid, and sunk to the required depth by admitting water by the valves already noticed. The ship is then floated into position over the pontoon and moored. The presses are now set to work, and directly the vessel rests on the keel-blocks on the rising pontoon, the various sliding blocks and shoring appliances are adjusted, and the vessel and pontoon are raised out of the water. The water is now run out of the pontoon, and the valves in the bottom closed. The grid is lowered, and the pontoon, being now buoyant, is left floating with the ship on it, and can be towed to any required spot for the repair or examination of the latter.

It should be mentioned that the valves referred to are opened and closed by hydraulic power.

Several pontoons may be employed, and hence one lift may dock several ships.

In the hydraulic graving dock formerly in use at Victoria Docks, London—the first of the series of similar docks constructed by Mr. Clark—the ship was raised by a grid similar to that employed at Bombay, but here eight pontoons were provided. The dock was 310 feet long, 62 feet wide, and capable of raising a ship of 3000 tons with a draught of 18 feet at ordinary high water. Later a large pontoon was built, with high sides and a gate at one end. This was lowered under water by the lift, and the ship was floated in. The power of the lift was sufficient to raise the sides of the pontoon out of the water, when, the gate being closed, the water was removed and the bottom of the ship exposed. In this way the power of the dock would be increased to 4000 tons. The presses were arranged in three groups as before to guard against tilting, which would ensue if all the presses were actuated from one supply and one part of the ship happened to be heavier than another. There were also a few guide-columns provided for the grid to slide against in its ascent or descent. The details are very similar to those of the Bombay Dock.

At Malta a dock of a similar class has also been provided.

XXIV.

HYDRAULIC ENGINES.

In most hydraulic engines a reciprocating motion is produced by the action of the working fluid, water, on a piston, and this is afterwards converted into a rotary motion by a crank, as in the steam engine; in fact, many hydraulic engines are simply modified steam engines.

The Brotherhood Hydraulic Engine.

One of the most successful of the hydraulic engines in use in this country is the three-cylinder engine of Mr. Brotherhood, which is well represented in Fig. 217, which is a perspective sectional view, and in Fig. 218, which shows a section through the central lines of the cylinders.

Three trunk pistons P P P, working in cylinders A A A, open at their inner ends, are connected to one crank-pin R. The pistons are pushed forward towards the central chamber by the pressure water which is admitted to each in turn by a passage V, leading from the rotary valve seen to the right in Fig. 217, and exhausted by the same passage, when an opening in the valve allows communication with the exhaust. The engine is single-acting.

The method of packing the piston by U leathers is shown on the upper piston in Fig. 218, and a newer method by cup leathers in Fig. 224, this method allowing easier adjustment and renewal of leathers. A separate section through the valve and seat and a view of the face of the valve are shown in Fig. 219, where A is the seat with

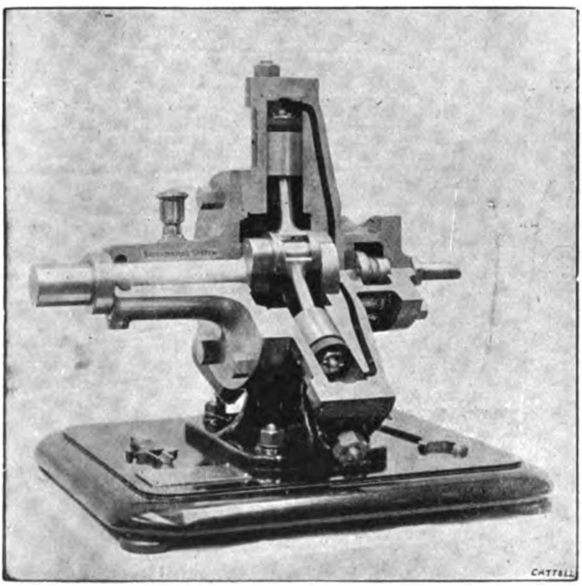


FIG. 217.

passage V, the section of which is shown as an opening V in the last figure. The darkly coloured portion C is the valve which is rotated from the shaft S, the square end of which fits into the square hole in the valve. This shaft is driven from the main crank by the pin K; as shown, the passage V is open to exhaust. The space B B is connected to supply which finds its way through the openings provided in C, and through one of the segmental openings seen in the valve-face to the cylinder when that opening at the proper time comes over a port. The valve is of phosphor bronze, and revolves in the valve-chest which is bolted to A. There being no dead centre, the engine will start in any position, the arrangement being equivalent to three cranks at 120° to each other. Uniformity of motion is thus secured, and a fairly uniform flow of water to the engine. In a single-acting engine with one cylinder the flow of water would be very variable, as the motion of the piston is variable; but in this and similar engines the three cylinders equalise the demand so as to secure a fair, and at no time too rapid, velocity of the supply—a matter of considerable importance, as the hydraulic losses are about proportional to the square of the velocity. Also a rapid velocity is usually accompanied by a diminution of pressure if the accumulator

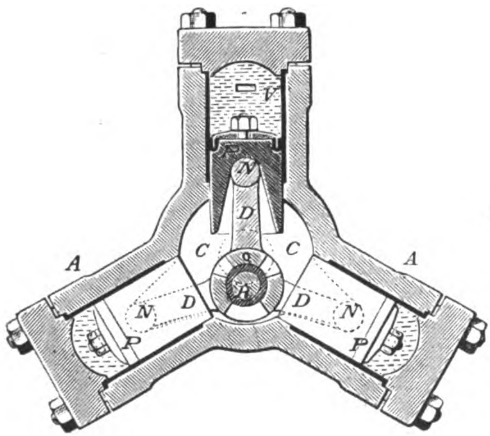


FIG. 218.

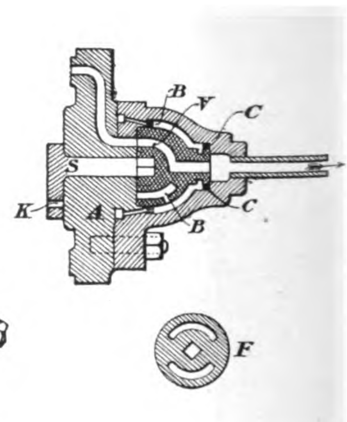


FIG. 219.

is at some distance from the engine. The engine is a good one, wears well, is capable of running at high speed, and hence is suitable for direct driving, which is now so much in favour.

The efficiency of the engine is high, if worked at or near full load. The "indicated" horse-power of the engine is easily calculated. If the average pressure during a working stroke be p lbs. per square inch, a square inches being the cross-sectional area of the cylinder, the horse-power is $3 \times \frac{plan}{33,000}$, where l is the length of the stroke in feet, and n the number of *revolutions* per minute.

ARMSTRONG'S HYDRAULIC ENGINES.

The introduction of hydraulic capstans at Paddington Station about 1851, called Mr. Armstrong's attention to the design of hydraulic engines. It did not appear to him that the various forms of rotary engines which had failed with steam were likely to succeed with water. An engine with reciprocating piston seemed necessary, and the form consisting of three oscillating cylinders with pistons acting on a three-throw crank, and with mitre-valves worked by cams from a shaft turned by the engine, was adopted. The type afterwards employed and still used for small powers has, as stated, three oscillating cylinders, each cylinder being fitted with a combined ram and piston, the upper side of the piston presenting only one-half the effective area of the lower side, the pressure on this half area being always constant, as this side is always open to supply, the other

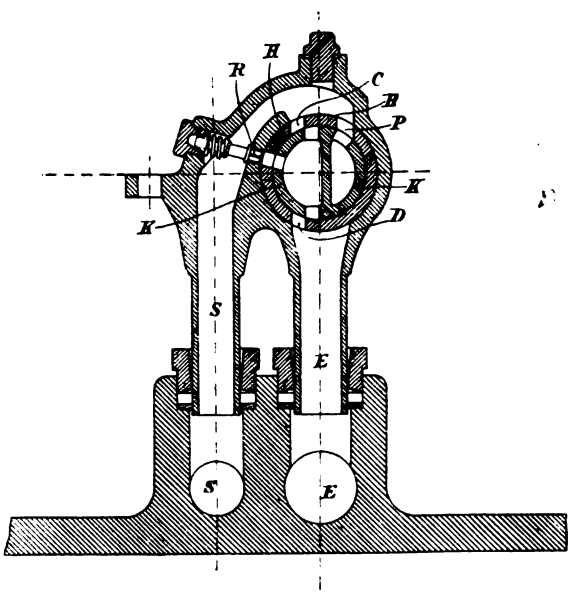
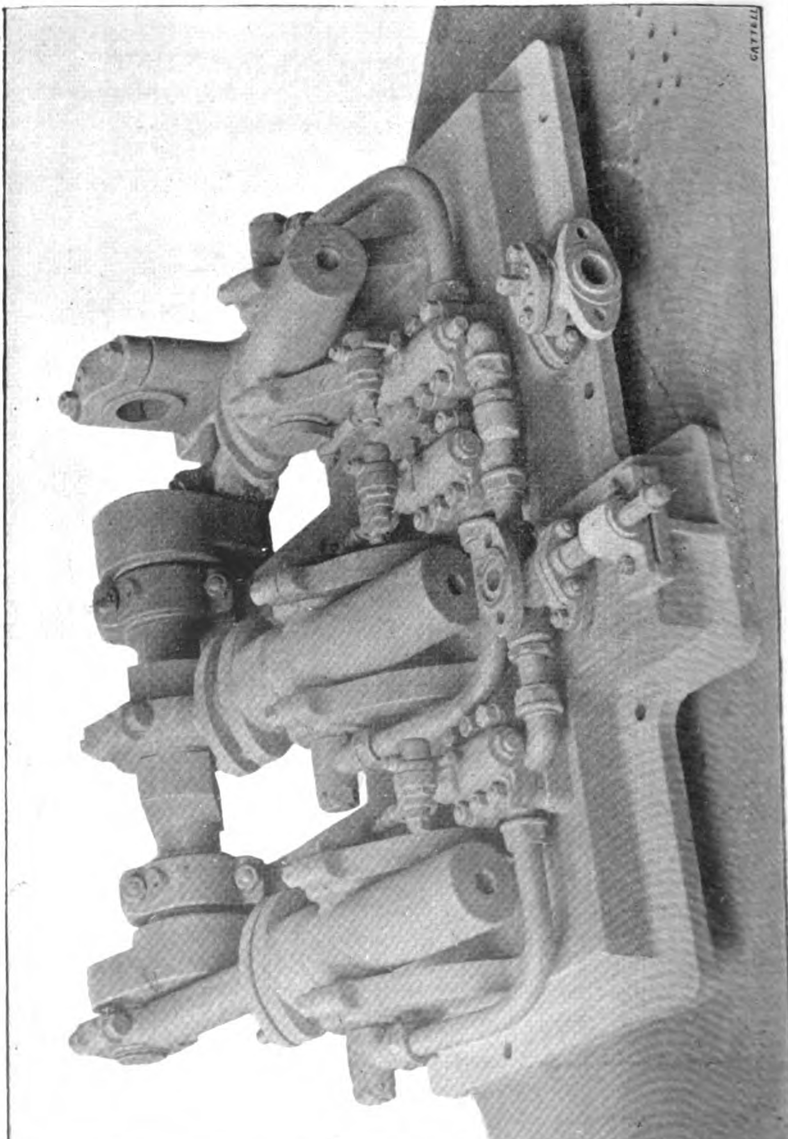


FIG. 220.

side, of full area, communicating alternately with pressure and exhaust; so that in one stroke the movement is due to difference of pressure, whilst in the next it is due to unopposed pressure. The action is similar to that of the Armstrong pump referred to at p. 424.

The valve is a two-port cylindrical slide valve, placed either within the trunnion on which the cylinder oscillates, or in a prolongation of it. The valve is worked direct by the oscillation of the cylinder, a relief valve being provided to prevent concussion by giving a means of escape for shut-in water.

The valve will readily be understood from Fig. 220. S is the supply passage, P the constant-pressure port always open to the upper



or annular side of the piston; C is the pressure port to under or full-area side of the piston; D is the exhaust port, and E the discharge passage from the engine.

K is a ring of hard metal forming the fixed working face for the valve, the upper segment of which H H is free to press up against the rubbing surface, as it wears away, being kept in contact by the pressure of the water.

The inner ring shows the trunnion in section, with the pressure and exhaust ports in its upper and lower sides respectively. The port of the relief valve is always open when that valve is required to act.

When the engine is required to reverse, if the type of valve here shown be used, two valves are provided for each cylinder, one for right-hand revolution in one trunnion, the other for left-hand revolution in the other, but an ordinary slide valve with reversing gear is simpler.

The engine more usually made now, is shown in Fig. 221. It has three oscillating cylinders as before, but the cylinders are fitted with plain rams, this arrangement requiring no internal packings for the pistons, which is an advantage. The engine is, of course, single-acting, the valve being like that shown in Fig. 220, except that the constant-pressure port and passage in valve are omitted. This engine is made from 1 to 70 horse-power.

In the largest size of engines, such as those used at the Tower Bridge, the three-cylinder arrangement is adhered to, but the cylinders are fixed, the ends of the plungers being guided and fitted with connecting rods as in a steam engine.

Haag's Hydraulic Engine.

This compact little motor has been a good deal used, especially on the Continent. It is, as will be seen from Fig. 222, of the oscillating cylinder type, the cylinder rocking on large trunnions through ports in which the pressure water is admitted and exhausted. The action will be better understood from an examination of Fig. 223, which shows two sections of a smaller form of the engine.*

The left-hand figure represents a section through the axis of the cylinder, whilst the right-hand figure shows a section at right angles to this through the centre of one trunnion. The passage S is open to supply and E to exhaust, P_1 and P_2 being ports through

* By Messrs. W. H. Bailey & Co., of Salford.

which the water is admitted by p_1 and p_2 to the two ends of the cylinder. As the trunnion and cylinder oscillate, owing to the revolution of the crank C, say in the direction indicated by the arrow a, the port P_1 is brought opposite supply S, and water is admitted by p_1

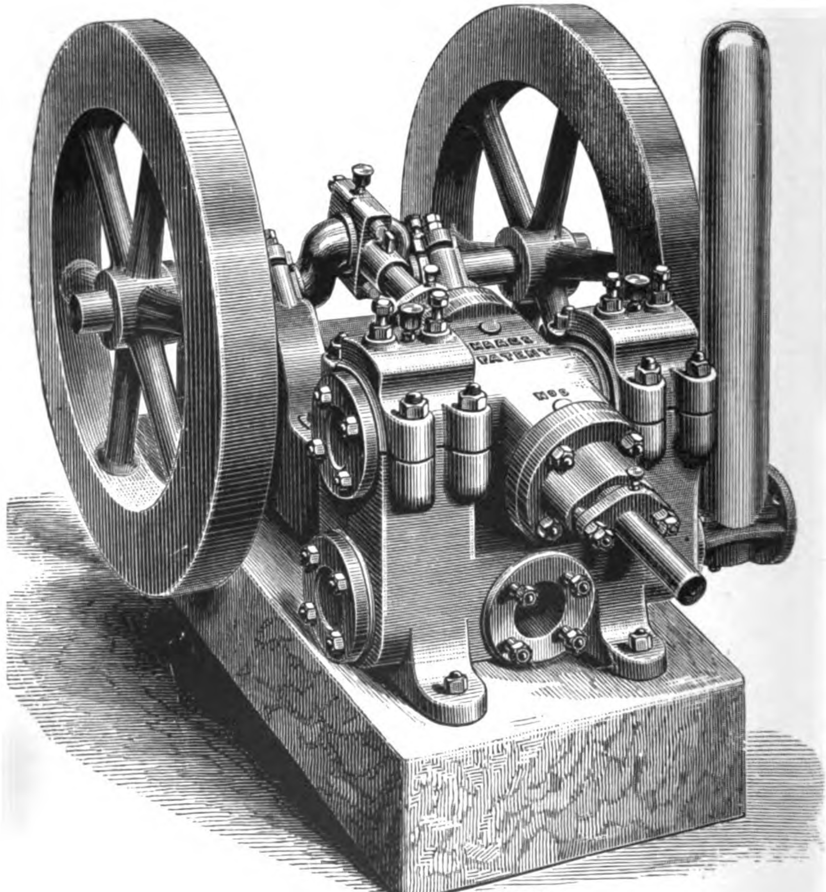


Fig. 222

behind the piston Q, which is thus moved forward. This motion, causing the cylinder to oscillate in the opposite direction, presently brings P_1 opposite exhaust and P_2 to supply, when the back stroke commences, and the cycle is repeated.

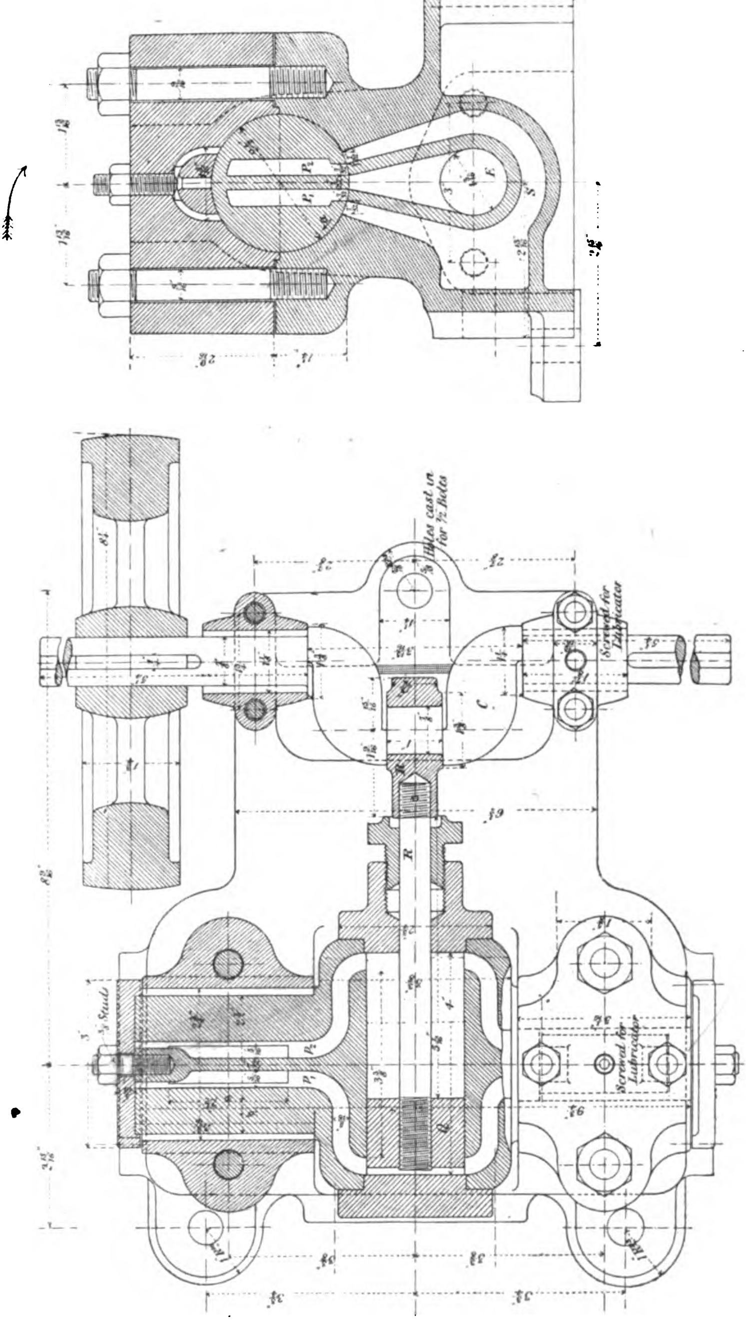

a
Holes cast in for ½ Bolts
Screwed for Lubricator
Screwed for Lubricator
½ Stud

FIG. 223.

This gives a neat and compact little engine, which for full loads works with a good efficiency, say from 80 to 85 per cent., with heads of water varying from 80 to 300 feet. With lower pressures the efficiency is somewhat reduced.

For the benefit of the student a dimensioned section is shown in Fig. 223.

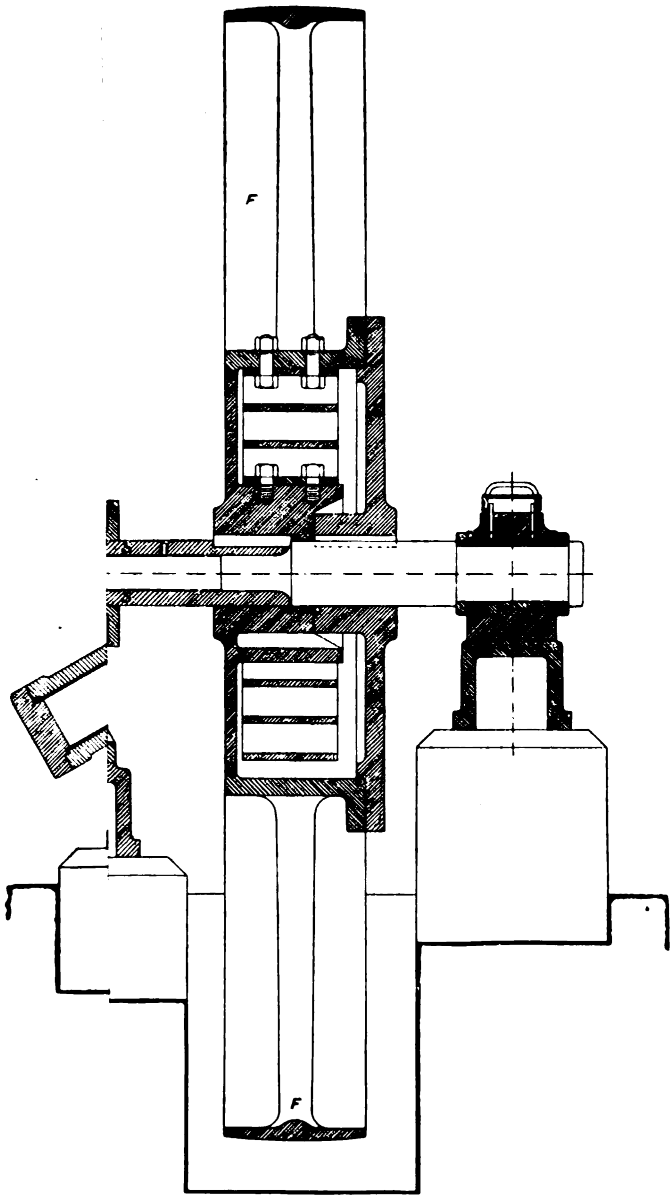
VARIABLE-POWER ENGINES.

THE BROTHERHOOD-HASTIE HYDRAULIC ENGINE.

Since it is impossible in hydraulic engines to obtain the advantages arising from the expansion of a working fluid like steam, other methods have been adopted for varying the power of the motor, and the consequent consumption of water, in accordance with the demand to be met by the engine. Cutting off the pressure before the end of the stroke, and letting the remaining piston displacement be filled with low-pressure water from a tank, suggests itself as one method of accomplishing the end in view; but this has not proved very successful. Usually the required variation is produced by varying the length of the stroke. This is done in the Brotherhood-Hastie engine, shown in Fig. 224, which is really a Brotherhood engine fitted with Hastie's arrangement for varying the throw of the crank. Instead of the connecting rods D D D taking hold of a crank having a fixed throw they are coupled to a crank-pin which, by means of a cam acting on a slide, can be adjusted to various distances from the centre of the crank-shaft. It will be seen that the shaft S is hollow, and is traversed by a central spindle, having fixed on it the adjusting cam just referred to.

The crank-shaft carries a fly-wheel F, of which the central portion is split, one half being connected to the hollow crank-shaft S, the other to the central spindle, the two parts being connected together by the strong volute (or flat spiral) spring, like a clock spring, shown in section.

The arrangement is such that the crank-shaft can only transmit driving power to the fly-wheel through the spring, thus coiling it, more or less, according to the resistance of the load. But this coiling of the spiral spring means a rotating movement of the hollow crank-shaft relative to the inner spindle, and a consequent movement of the adjusting cam C (small figure) relative to the crank disc, which contains a slot in it shown at M, in which the cam moves K (carrying the crank-pin) further from or nearer to the centre A, depending on whether the load increases or decreases. K is shown



[Face page 328.

on the highest part of the cam, so that the stroke has its greatest value. When the load is removed, the cam is turned by the spring in the opposite or left-hand direction, until the depression V is met with, the stroke being then shortest. The dotted lines show another portion of the cam which is used when the engine runs in the reverse direction. The power of the engine can thus be varied from a maximum to a minimum in the ratio of about 3 to 1.

Though this is an exceedingly ingenious arrangement, with all the constructive details well thought out, it is now very seldom employed. The makers (the Hydraulic Engineering Company of Chester) say that they now rarely make the engine.

Rigg's Variable Stroke Rotary Hydraulic Engine.

Of the many variable power hydraulic engines which have been devised, that of Mr. Arthur Rigg is probably the only one which has attained practical success.

To construct a hydraulic engine such that the consumption of water may agree fairly with the power given out without any great loss of efficiency, and to do this automatically, has been the goal for which many inventors have striven. When the pressure-water costs 1*s.* to 2*s.* per thousand gallons, it is evident that an engine which, driving a capstan for instance, uses as much water in hauling in the unloaded rope as in pulling its maximum load, is a very wasteful and expensive motor for varying powers. In Rigg's engine the stroke, and hence the water consumption, is varied, either by hand or governor, to agree with the power demand, this variation being easily effected whilst the engine is running. The speed of the engine can also be varied within much wider limits than are possible with reciprocating engines, the engines being capable of running at a very high speed without undue shock or vibration—600 revolutions per minute being often attained.

The general principle of the Rigg engine is not very easy to explain without a model. Perhaps the simplest way of regarding the matter is to think first of all of any three-cylinder engine, such as the Brotherhood, where there are three cylinders at 120° to each other, driving a common crank, with the approximately uniform turning moment which such a system provides. If instead of bolting the bed of such an engine down to its foundations, we imagine the main shaft fixed, and the three cylinders and bed to revolve round that shaft, a general idea of the action of the Rigg engine may be obtained.

Fig. 225 will illustrate the action. Here a number of cylinders—say three—revolve round a common centre E, whilst the pistons (plain plungers) revolve round the centre O. At the position 1 A the cylinder and piston circles coincide, and at 5 B are farthest apart; whilst during travel from A to B there is a gradual increase of the distance 2 *c*, 3 *d*, etc., and a gradual diminution of the distance from B round by 6 *g* to A. Imagine three cylinders fixed symmetrically round the inside of the outer circle, which takes the place of the fly-wheel of an ordinary engine, the cylinders being fitted with plungers, all capable of moving round a pin at O. Then as the fly-wheel and cylinders revolve, the pistons will travel regularly in and out of the cylinders as in an ordinary engine.

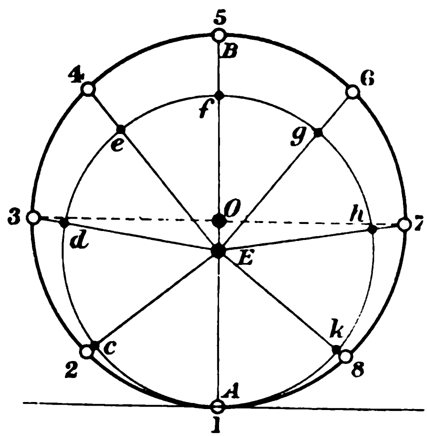


Fig. 225.

The actual arrangement of the cylinders and pistons is shown in Fig. 226, which gives a front view. The cylinders are balanced against each other during construction, as are also the pistons. It is true the pistons and cylinders possess sliding movements relative to each other, as well as accelerations and retardations as regards angular movement, with a small change per revolution of the position of the mean centre of gravity of the entire revolving mass; but these changes are small, and are carefully balanced for mean speed and stroke, the unbalanced force, which is taken up by the bedplate and foundations, being very small compared with the totals of such forces met with in other types of engines.

Assuming a constant speed of fly-wheel, it will be seen that as the pistons move *out of* their cylinders their mean angular velocity decreases, whilst as they return their mean angular velocity *increases*. These changes do not affect the regularity of motion, because, as one cylinder and piston becomes a driver in so far as their angular velocity diminishes, so the opposite pair becomes driven in an equal degree. These internal forces, mutually balancing, do not appear outside. All the cylinders revolve at the same distance from their common centre, and contain ordinary plungers with usual packings, hence do not suffer from many of the practical defects which have hitherto rendered rotary engines impossible.

Fig. 226 represents, partly in elevation and partly in section, one of these engines, of about 30 horse-power, designed for working at a pressure of 700 lbs. per square inch. In this illustration the cover

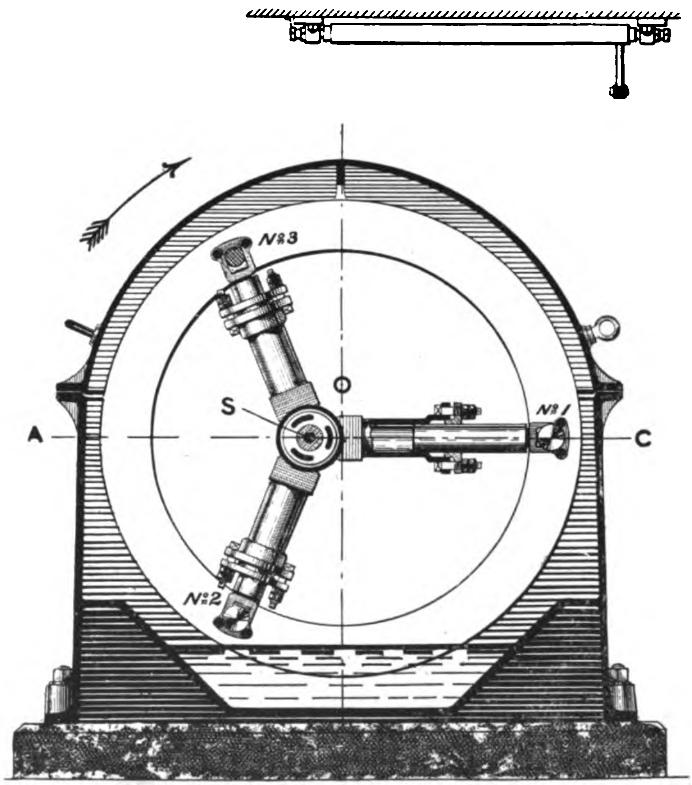


FIG. 226.

is removed. This cover performs no part in the working of the engine, but is used merely for protection, and for retaining the oil for general lubrication.

The figure shows the main driving cylinders with the valve-chest removed, and Fig. 227 gives a section of the relay engine, whereby the stroke of the main engine is controlled by the governor or by hand.

Returning to Fig. 226, all the cylinders ar[illegible]sition for full stroke in the direction indicated by the arrow. No. 1 cylinder is a section, and it will be seen that it consists of the ordinary hydraulic cylinder with its plunger or ram, the gland and packing being clearly shown.

The cylinder is cast in one piece with a circular valve, with which it revolves on the main stud S, as do the other cylinders Nos. 2 and 3. The plungers, connected to the disc crank, revolve round centre O,

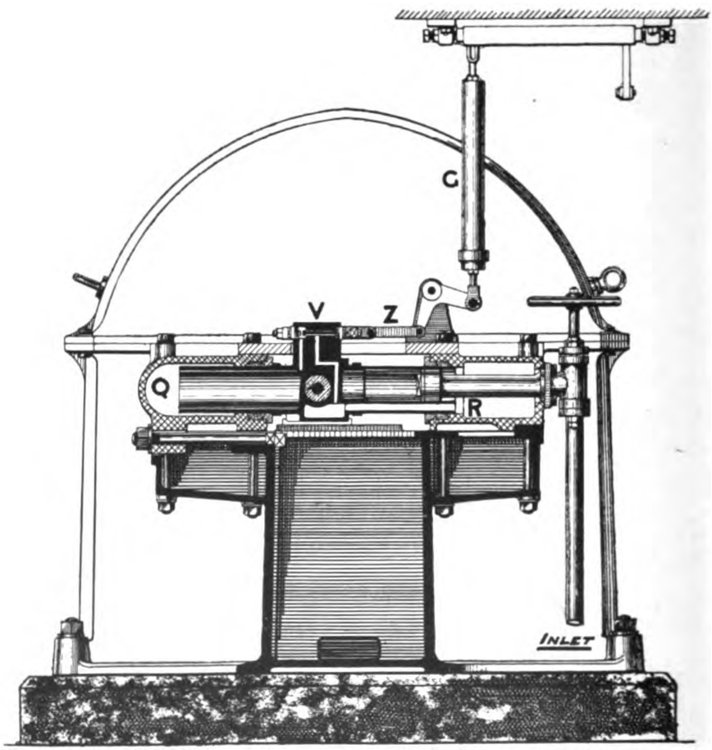


FIG. 227.

which is also the centre of the main shaft ; the distance O S therefore constitutes the length of the crank, or half the stroke. This crank does not turn round as in an ordinary engine, so it can easily be altered in length. The exhaust opens for each cylinder when it comes into the position of No. 1 cylinder, the extremity of the outer stroke being here completed. At the position A the circular valve provides for the admission of pressure-water, and the inflow continues throughout the stroke until the cylinder again arrives at the first position. Any movement of the cylinder stud S alters the stroke, and if S be moved over to coincide with O, the stroke is zero. If the movement of S be continued beyond O towards C, the stroke increases again, but the engine runs in the opposite direction.

Hence the engine, being variable in stroke and reversible as regards direction of motion, is very suitable for driving capstans, etc. It is best regulated by hand for reversing, but by governor for winding, as regards speed and power given out. The very ingenious method of altering the stroke will now be more fully described.

RELAY ENGINE.

All good governing mechanisms which have to exert considerable forces or to keep steady any portion of a mechanism against varying and considerable forces must act through a relay.

The relay engine of Mr. Rigg's combination is shown in Fig. 227. It consists mainly of two rams, Q and R, of different diameters, cast in one piece with the valve-chest V; the smaller ram R being subjected to a constant pressure, admitting supply to the inlet port for driving the main engine. The large ram Q is provided with two valves, whereby the pressure can be admitted to, or exhausted from it. Only the exhaust valve is shown in complete section; the inlet valve, lying behind, cannot be seen, but they are alike.

These valves are of ordinary construction, and are carried along with the valve-chest, so that a push or pull from the stop-lever Z connected with the governor G gives rise to a movement in one direction or the other, according to the extent of such push. This action results in the relay engine starting, stopping, or moving in exact obedience to the extent of the impulse it receives from the governor in either direction, carrying the stud S (Fig. 226) attached to it, further from or nearer to the fly-wheel centre; thus altering the stroke of the main engine.

The governor—of a well-known type—is contained in this case within the driving pulley, as shown in Fig. 228, and its connection with the valves of the relay engine—omitted in the figure—is so simple as to require no description.

The outside appearance of the relay is shown in Fig. 229.

The general function of an engine is to turn a shaft. This engine, however, seems to be capable of application to many purposes to which hydraulic engines are not now applied. Applied to a capstan, the advantages of good governing and variable stroke are evident.

At first the speed is rapid as the slack rope is drawn in, and the governor shortens the stroke till very little water is consumed; then as the load comes on the rope the stroke increases until the engine

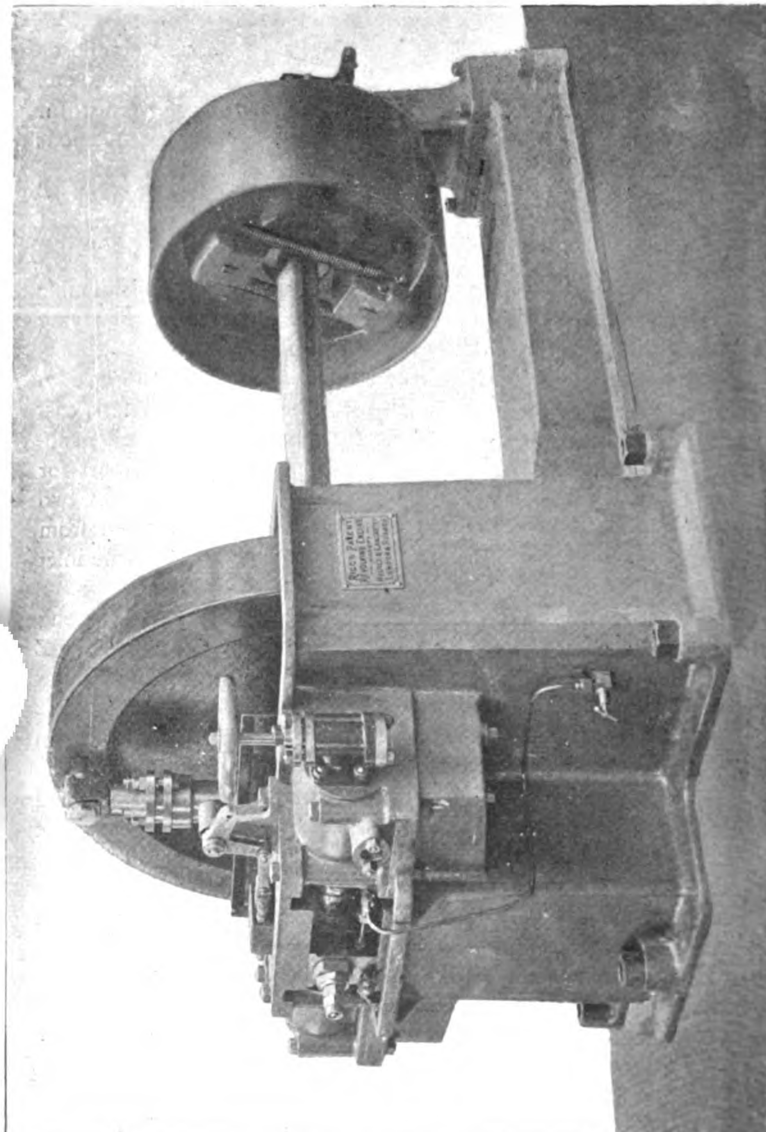


FIG. 228.

works at full power. In an operation of this kind probably less than half the water is used that would be required by an ordinary non-variable hydraulic engine. Where fairly continuous power of variable amount is required for long periods the saving of water may be half of the entire amount required by an ordinary engine. The reader can easily figure out the saving in cost for a given average power. The

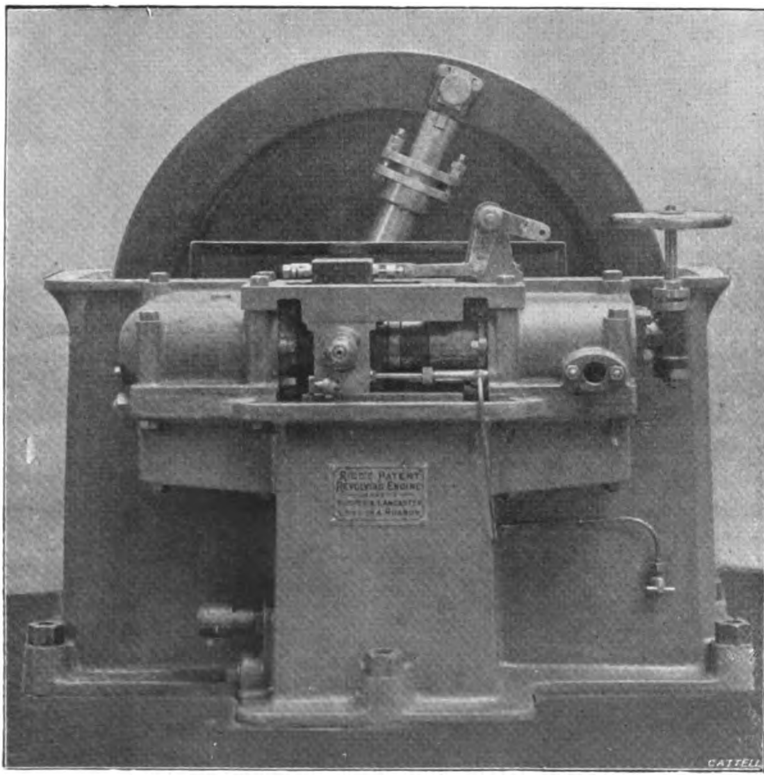


FIG. 229.

successful application of one of these engines (of about 10 horse-power) to the movement of a heavy draw-bridge is referred to in the section on Movable Bridges.

The engine is carefully thought out and well designed, the patentee, Mr. Rigg, having the requisite practical and theoretical knowledge, and having devoted many years to the perfecting of the engine.

Capstan Engines.

An interesting application of the Rigg engine for driving a capstan is shown in Fig. 230. The engine is horizontal, and drives a shaft which is geared to that bearing the steel bollard. The engine turns readily in either direction, and its stroke is regulated by a

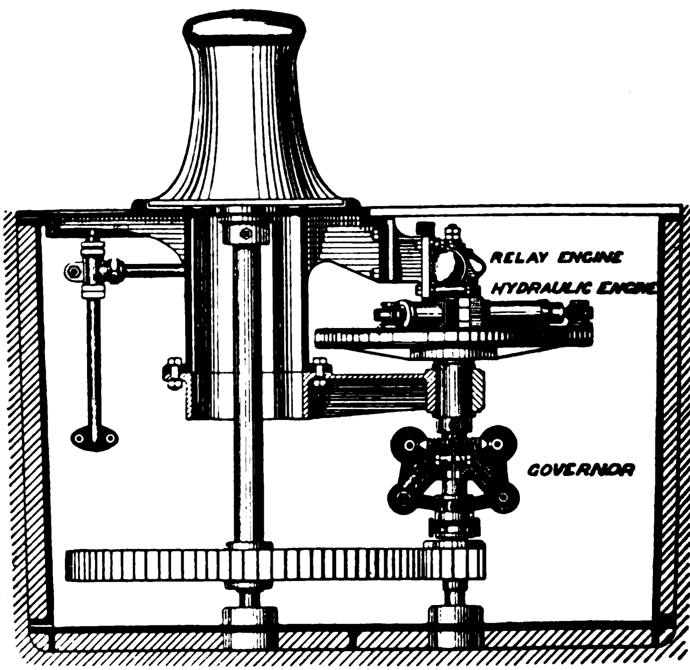


Fig. 230.

governor acting through the relay, as already described. This forms a compact arrangement, and the engine can go at any required speed up to 500 revolutions per minute or more, using pressure-water proportionate to the work done.

Other capstan engine combinations are shown at pp. 366 and 376.

XXV.

BRIDGE AND DOCK-GATE MACHINERY.

BRIDGES which can be opened for the passage of ships are of different types. Of those worked by hydraulic power we have (1) swing-bridges, (2) draw-bridges, (3) bascule bridges. A type of swing-bridge for a double line of rails, leaving two passages for ships when open, is shown in Fig. 231. The picture is that of a comparatively small bridge, but gives a good general view of such a bridge, showing how the opening portion moves. A larger and heavier bridge of a similar kind crosses the river Tyne at Newcastle.

SWING-BRIDGE OVER THE TYNE.

The swinging portion of this bridge, which is carried on a central pier of masonry, is 280 feet long, and spans two passages each 104 feet wide clear of the fenders, one on either side of the central pier. The bridge has a roadway 22 feet wide between the two main girders, and two footways, each 8 feet wide, one on either side. These are carried by cantilevers from the outside of the main girders.

The main girders themselves are of triangular construction, the upper and lower booms being of trough section, and the vertical and diagonal members of H or box section.

The general arrangement of the machinery for turning the bridge is shown in Fig. 232.

This machinery is placed in the central pier. There are two small vertical steam engines, with boilers, supplying accumulators from which a pressure of 700 lbs. per square inch is obtained for working the hydraulic machinery.

To prevent interference with the working of the bridge, the accumulators are placed and work entirely below the upper level of the pier, being sunk in two cast-iron cylinders embedded in the masonry. The illustration explains itself. To the right and left are the accumulators for storing the water pumped by the steam engines. Full details of one of these accumulators are given in Fig. 233, which is reproduced from the working drawing.

The accumulators are joined by piping, and are also connected by smaller piping to the hydraulic engines, the positions of which

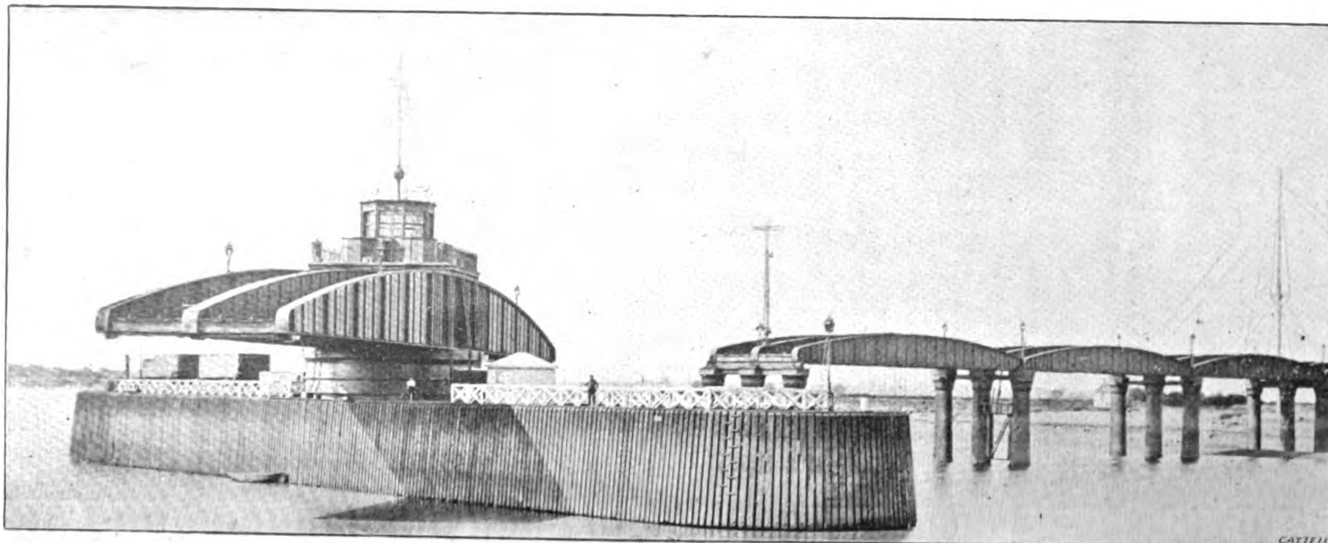
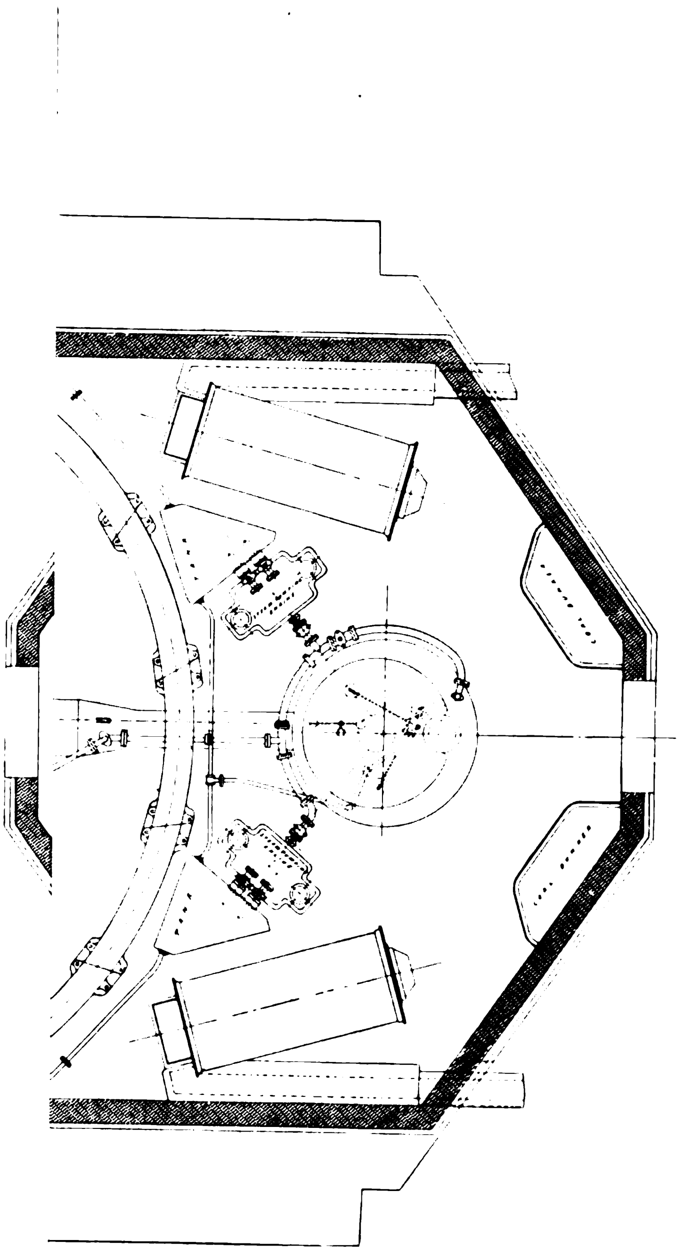


FIG. 231.



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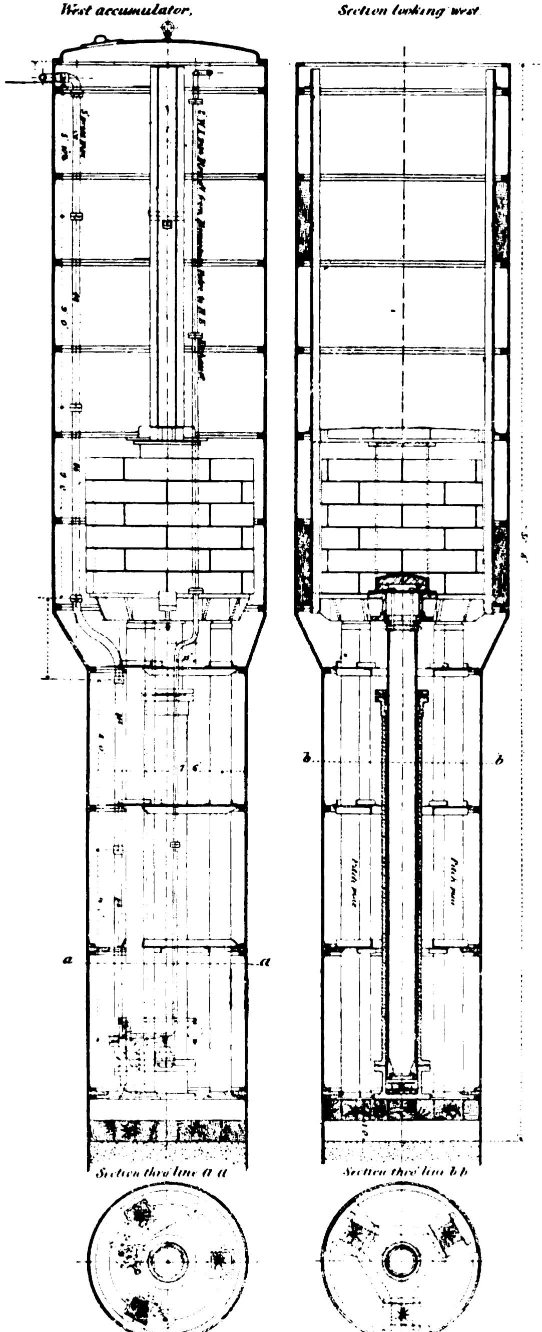


West accumulator.
Section looking west.
b
b
a
a
Pitch pine
Pitch pine
Section thro' line a a
Section thro' line b b

FIG. 233.

are shown on the left. These engines, two in number, are of the well-known three-cylinder type depicted in Fig. 221, and described at page 322. They exhaust into pipes, which convey the water to the tanks from which the supply for the pumping engines is drawn.

They are placed in a chamber, as shown, near the centre of the bridge, and are connected by gearing with a pinion working into a turning rack fixed to the upper roller path.

The bridge turns on a ring of "live" rollers running between this upper and a lower path. A section of the path showing this anti-friction method of support is shown in Fig. 234.

The rollers carry about one-third of the total weight of the opening portion of the bridge; the remainder being taken by a hydraulic press, indicated at the centre of Fig. 232, which forms the pivot.

Sliding blocks worked by hydraulic cylinders are provided at the two ends of the bridge for taking the weight at these points when the bridge is open for road traffic. Direct-acting cylinders are fixed in the ends of the main girders, by means of which the ends of the bridge can be lifted so that the sliding blocks may be withdrawn, when the bridge is required to be opened for river traffic. The bridge can turn in either direction and the range of travel is not limited. The hydraulic gear is controlled from a cabin carried upon the roadway at the centre of the bridge, from which position the man in charge has a good view.

The total weight of the swinging portion of the bridge is 1400 tons. The hydraulic machinery is by Messrs. Sir W. G. Armstrong & Co., the engineers of the bridge being Mr. Ure and his successor, Mr. Messent.

Draw-Bridges.

An interesting example of a draw-bridge worked by hydraulic power is shown in Fig. 235. Formerly this bridge was worked by two pinions turned through gearing by a hydraulic engine, the pinions working into two racks fastened to the bridge.

This not proving altogether satisfactory the system has been adopted which is illustrated in Figs. 235 and 236.

The bridge is supported by wheels running on rails, and is moved by a steel wire rope, one inch in diameter, which is fastened at one end to the bridge as shown, being wound on a drum as indicated in Fig. 236; the other portion of the rope being fastened to the other end of the bridge and wound on the drum in the opposite direction.

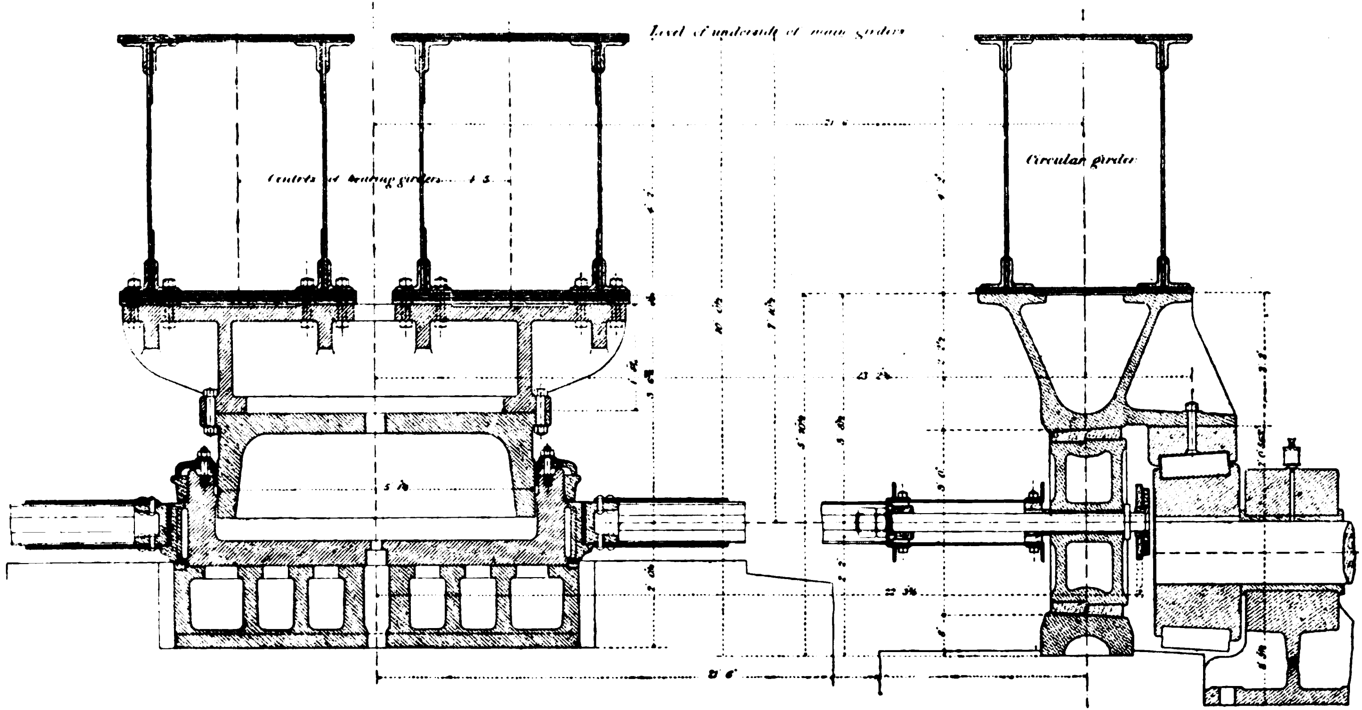


FIG. 234.

Hence when one rope is wound on to the drum, the other is unwound from it.

The drum is turned by a Rigg's hydraulic engine, which seems specially adapted for this work. It will readily be understood that

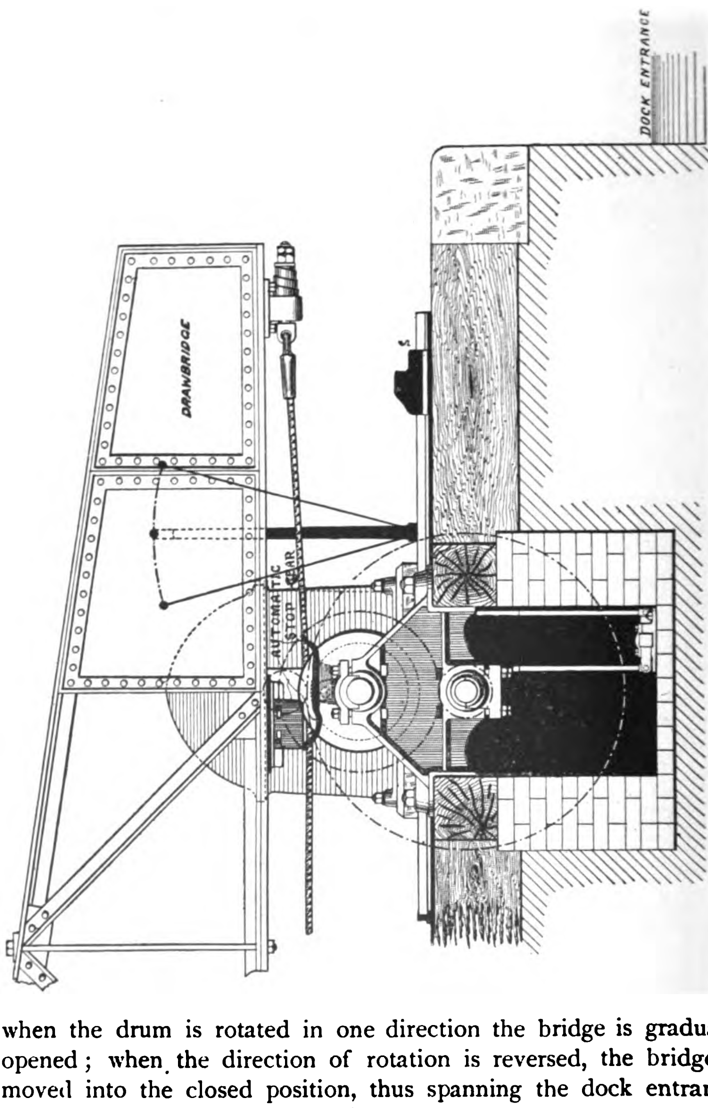


FIG. 235.

when the drum is rotated in one direction the bridge is gradually opened; when the direction of rotation is reversed, the bridge is moved into the closed position, thus spanning the dock entrance. The maximum pull on the rope is 3 tons.

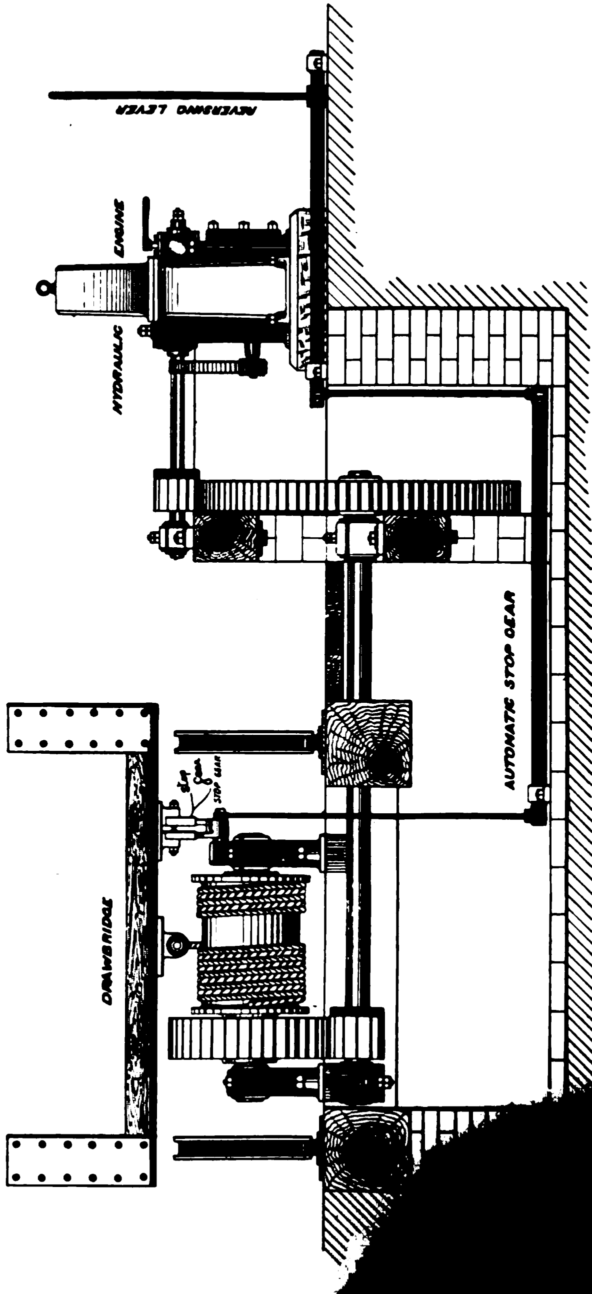

REVERSING LEVER
HYDRAULIC ENGINE
AUTOMATIC STOP GEAR
STOP GEAR
DRAWBRIDGE

FIG. 236.

A volute spring is provided at each end of the rope as shown in Fig. 235, to allow the pull necessary to start the bridge to be applied more gradually.

An ingenious stop-gear is provided to prevent over-winding of the bridge as the stops S on the rails might not be sufficient, if the engine were driven at high speed.

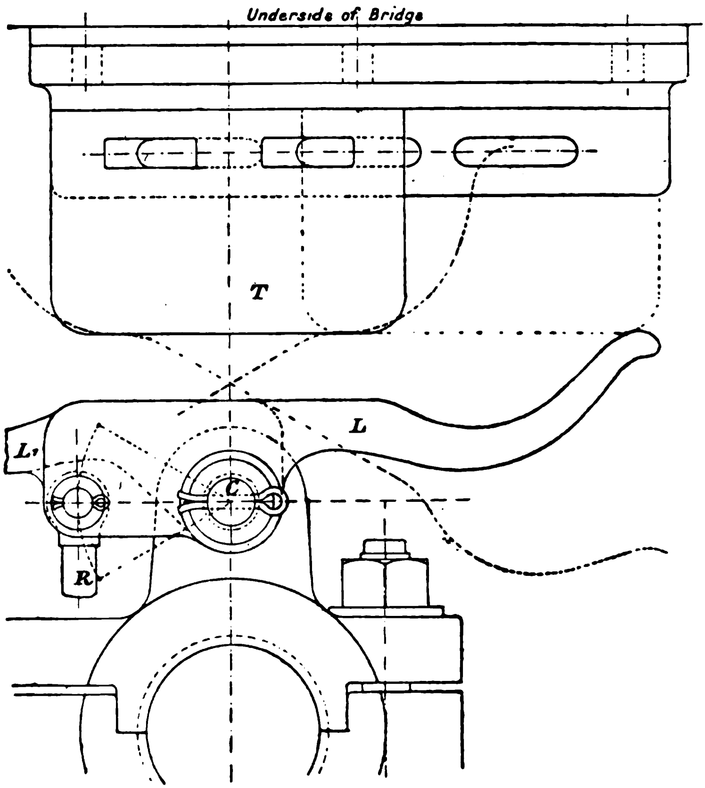


FIG. 237

A separate view of this automatic arrangement is shown in Fig. 237.

The curved lever L, the piece carrying it, and its fellow L_1 can oscillate on the pin C, and it works the reversing lever of the engine through the rod R, as indicated on the other drawings. The projection T on the bridge engages the bent lever L, when the latter occupies the inclined position shown by the upper right-hand dotted line, moving it down to the horizontal position if the bridge be moving from left to right, thus putting the reversing lever into the mean position and stopping the engine by bringing the crank-pin of the engine into the central position, as already explained. The stop-gear can be moved again to the proper position by hand in order to allow the engine to be started, in the following way.

There are two levers L and L_1 in different planes; one is acted on when the motion is, as above, from left to right, the other when the motion of the bridge is from right to left—in other words, when the bridge reaches the closed position L is depressed, whereas, when it reaches the open position L_1 is depressed, by a projection similar to T (but in the same plane as L_1) on the other end of the bridge. In the position of L shown in Fig. 237 the bridge has reached the closed position; if it be necessary to open it, L_1 is elevated by hand and the reversing lever moved into the position, giving motion in the required direction.

This method of moving a draw-bridge is in some respects novel, and has proved highly successful in this case, the bridge being now opened or closed in less than one minute.

Bascule Bridges.—The Tower Brilge, London.

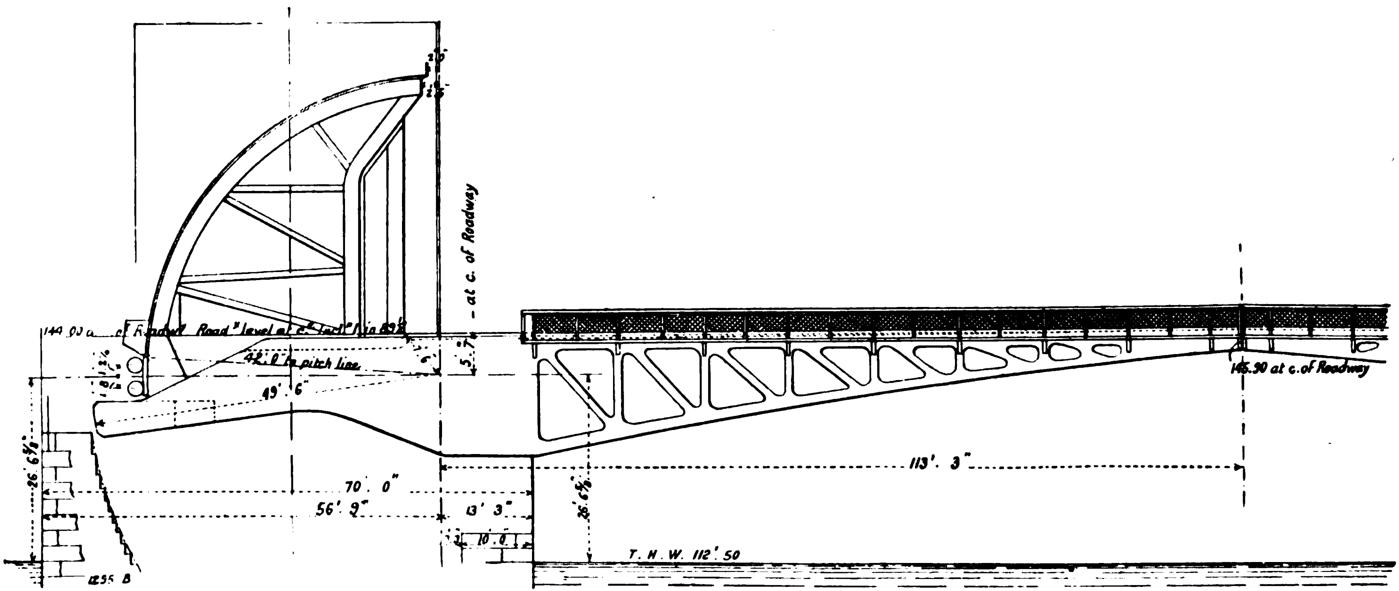
Space does not permit a lengthened reference to bascule bridges, but a short notice of the Tower Bridge, as the best example of such a bridge extant, may be interesting.

The construction of this great bridge was commenced in September 1886, and the bridge was opened for traffic by the Prince of Wales on June 30th, 1894. The total cost, including price of land, etc., was estimated at £1,900,000. The bridge connects by easy gradients of 1 in 60 on the north side and 1 in 40 on the south side, a point a short distance eastward of the Tower of London, opposite the Royal Mint, with a point near the Horselydown stairs, on the south side of the river Thames.

The depth of water under the central or opening span is 33½ feet at high water, with a headway of 20 feet near the abutments, increasing to 29 feet at the centre of the span. The piers rest on caissons filled with cement sunk 25 feet below the bed of the river, the superstructure consisting of elaborate stonework with a framework of steel inside.

The roadway is 35 feet wide, with two footways each 12½ feet wide on the approaches, reduced to 32 feet and 8½ feet respectively on the centre span.

FIG. 218.



42' 0" to pitch line
49' 6"
70' 0"
56' 9"
13' 3"
113' 3"
5' 7" - at c. of Roadway
145.30 at c. of Roadway
T. H. W. 112'.50

FIG. 239.

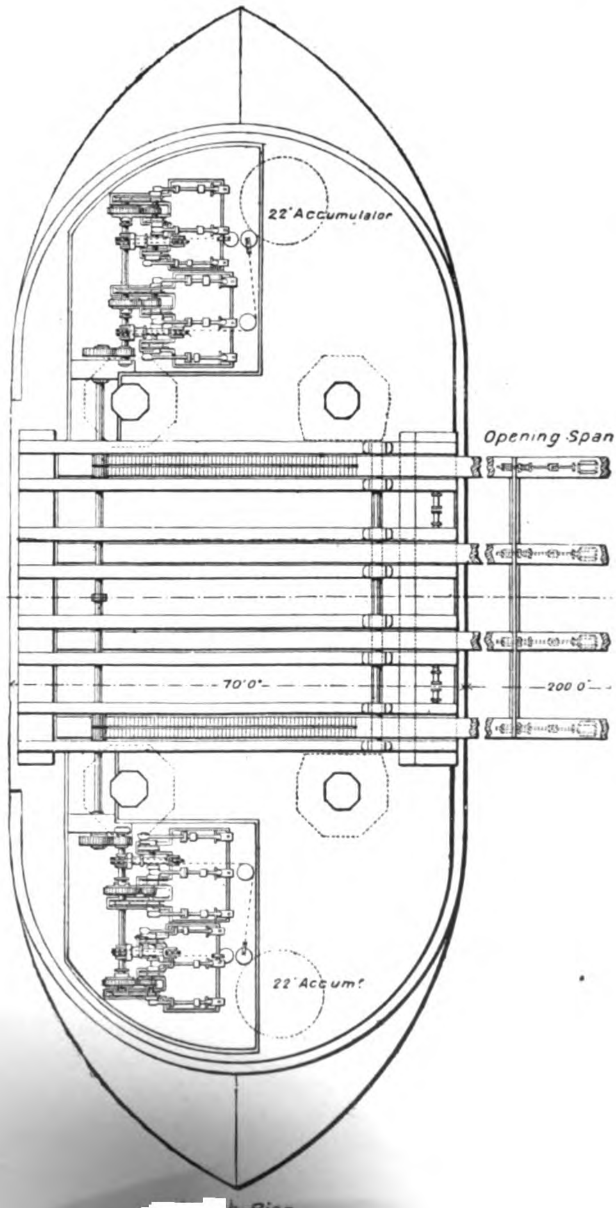
The general appearance of the bridge, and method of opening the bascule span, are shown in Fig. 238. Each leaf or half of the bascule span is 113 feet long from the centre of the horizontal hinge or pivot (1 foot 9 inches in diameter, resting on roller bearings) on which it turns. The short arm of this portion is 50 feet long, so that the entire length of each leaf is about 163 feet and its width 50 feet; total weight about 1200 tons. The inner or shorter arm of each bascule is loaded with lead, so as to bring the centre of gravity of the opening portion to the pivot; thus the main resistances to be overcome in opening the bridge are those due to wind pressure, inertia, and friction. Hydraulic lifts are provided to take foot passengers to the high-level roadway during the period when the bridge is open for river traffic.

The Hydraulic Machinery.

The method of opening the bridge will be understood from an examination of Fig. 239, which shows one half or leaf of the opening span in elevation. It will be seen that attached to the two main girders of each leaf of the opening span are great quadrants bearing toothed racks, each composed of steel, being of the shape of the arc of a circle of 42 feet radius. Each rack consists of eleven segments 6 feet long and 17 inches wide, with teeth of 5·93 inches pitch. Two rows of these racks are fixed side by side to each quadrant. Into these racks gear pinions 24½ inches in diameter, driven by hydraulic engines; there are two pinions to each rack. The hydraulic motive power is supplied at a pressure of 700 lbs. per square inch by accumulators, two being placed in each pier and two in a separate accumulator house on the Surrey shore. The pressure-water for the northern half of the opening span is conveyed by pipes laid in the high-level footway. The machinery is all in duplicate, each set including two compound steam pumping engines of the Armstrong type, each of 360 horse-power.

The hydraulic motors, also in duplicate, are placed in chambers in each pier, the arrangement being shown in Fig. 240, a dotted circle indicating each accumulator with ram 1 foot 6 inches in diameter and 22 feet stroke; the hydraulic engines are worked by the pressure-water from these accumulators, actuating through spur gearing the pinions gearing with the racks already referred to. An end view of the racks and pinions is shown in the lower portion of Fig. 241.

By an automatic arrangement, the pressure water is shut off from the hydraulic engines, when the operation of raising or lowering each


22" Accumulator
Opening Span
70' 0"
200' 0"
22" Accum!
Pier

leaf has been completed, so that if the man in charge neglects his duty, the leaves gradually come to rest in the vertical or horizontal positions. As a further precaution, hydraulic buffers are fixed in such positions that, even if the machinery lost its control of the bridge, the impact due to the leaves being brought to rest would be absorbed without injurious shock, in a manner very similar to that employed in absorbing the recoil of heavy guns. The hydraulic lifts for passen-

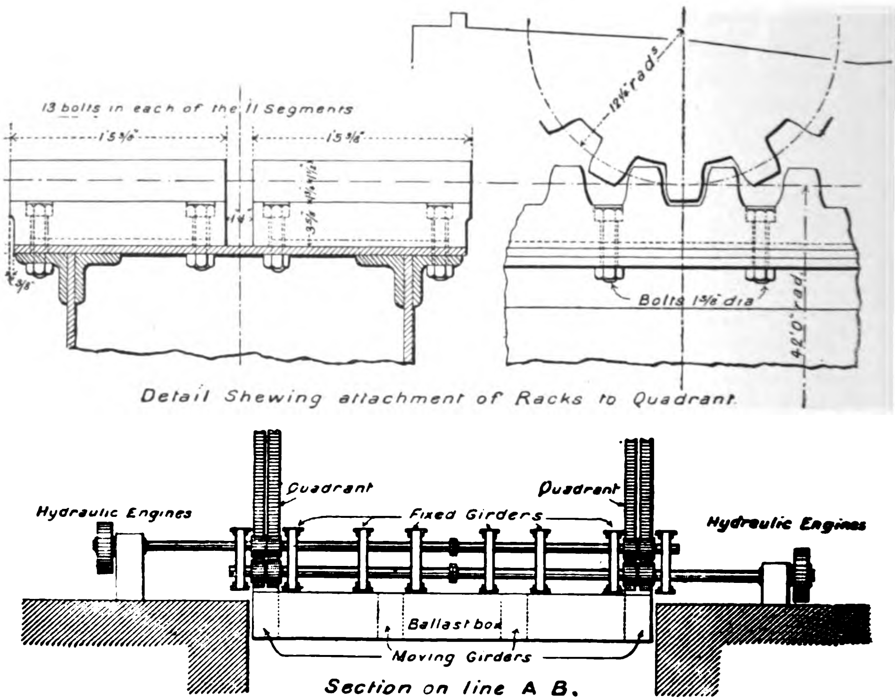


FIG. 241.

gers are of the suspended type, two to each pier, safety catches being provided in a way very similar to that already described. The time occupied in opening the bridge is about a minute and a half. The whole time taken to clear the bridge, open it, allow a ship to pass, and re-close it, is about five minutes.

Sir John Wolfe Barry, C.B. was the designer of the bridge, and engineer during construction, the hydraulic machinery being by Messrs. Sir W. G. Armstrong & Co.

Dock-Gate Machinery.

Formerly dock-gates were usually opened and closed by chains worked from crabs or winches, turned, in the older docks by hand, but more recently by hydraulic power. Two chains, or sets of chains, passed from the gate to the winch—one directly, the other over an opposite drum. Thus the rotation of the winch in one direction pulled in, say, the closing chain, slackening out the other, whilst rotation in the opposite direction pulled in the indirect-acting chain, slackening the former, thus moving the gate in the opposite direction. In some cases the winch or crab was actuated by an endless chain, moved from drums worked by hydraulic engines, the same chain also actuating capstans; in other cases the engines worked the crab through shafting, whilst a better and more compact method was found in working the crabs directly from hydraulic engines, usually of the oscillating type.

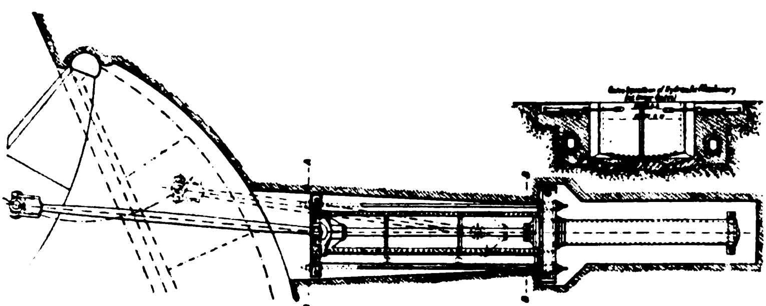


FIG. 242.

Coming to later times, the chains by which the gate is opened and closed are actuated by the motion of a hydraulic ram working in or out of its press, the motion being magnified by pulleys, as in hydraulic cranes, and the crab being dispensed with.

Thus, on pressure-water being admitted to the opening cylinder, the ram of this cylinder is forced out, dragging in a chain which, being passed over multiplying pulleys, acts on the gate so as to open it. The gate is closed by admitting the pressure water to another similar cylinder the ram of which acts on the closing chain, the former cylinder being in communication with the exhaust. These cylinders may be at opposite sides of the dock entrance, and thus a carrying drum is dispensed with.

This method is simple, and the machinery less liable to get out of order than in the others mentioned.

Recently, at the Windsor lock of the Barry Docks, a simpler and more direct system of hydraulic machinery has been employed to open and close the gates.

The ram of a double-acting cylinder acts directly, through a connecting-rod, on the gate, as shown in Fig. 242.

The cylinder is of cast iron, and is 19 inches internal diameter, the ram or piston having a stroke of 14 feet.

The piston rod is 7 inches in diameter, and is covered with copper, being connected to a cross-head which moves between steel guides. The connecting-rod is of wrought iron, joined at its inner end through a forged steel gimbal to the cross-head, the outer end being connected to the gate by a similar gimbal. This arrangement permits angular movement both vertically and horizontally.

The front end of the hydraulic cylinder is fixed to a framework of steel plates and angles, which is secured to the masonry of the lock wall, the framework taking the thrust when the gates are moved, the back end of the cylinder being supported by a cast-iron saddle. The inner ends of the guides are attached to this framing, whilst their outer ends are secured to a girder built into the masonry.

The working valves are spindle valves of the type used in hydraulic cranes, whilst a reducing valve is employed to regulate the pressure, which may be anything from 200 to 700 lbs. per square inch. Automatic cut-off gear is also employed to prevent over-movement of the gate.

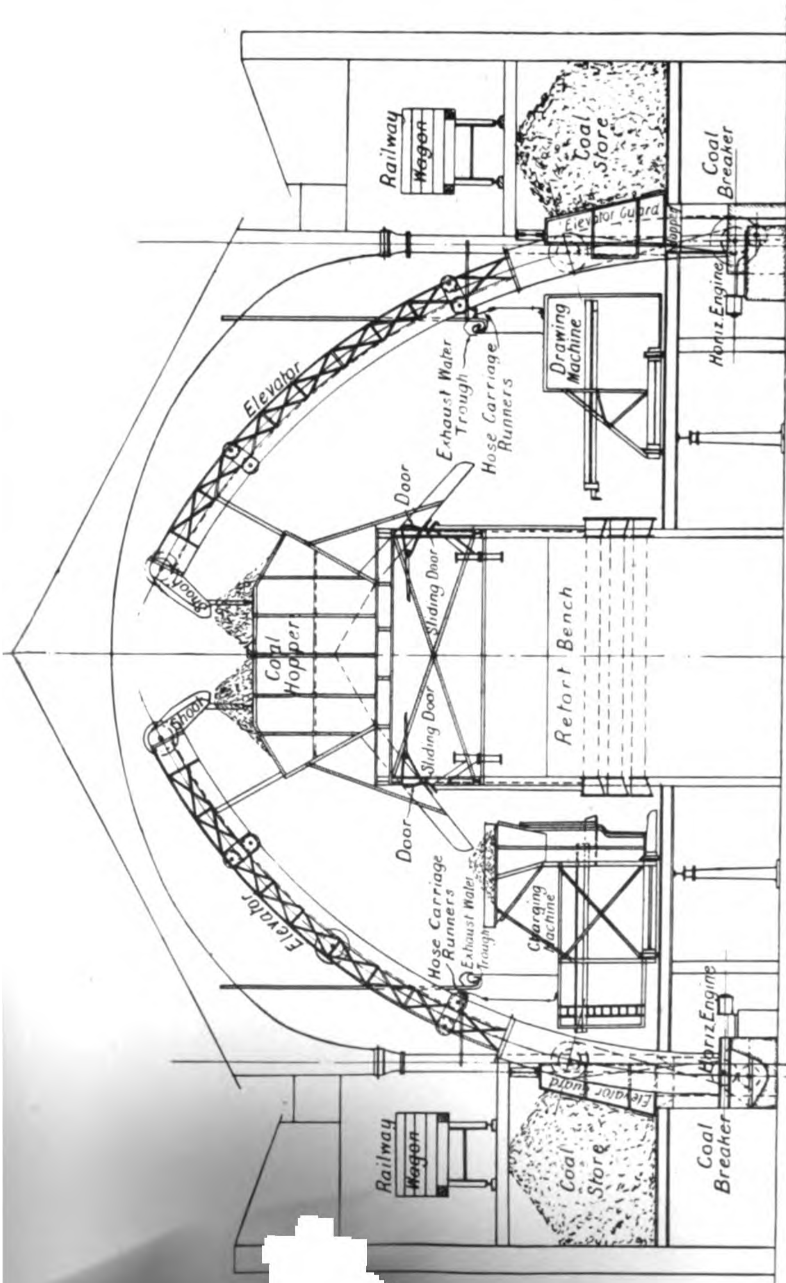
The position of the cylinders relative to the gate are shown in the smaller figure, the former being placed in recesses in the walls, as shown.

Capstans, operated directly by oscillating hydraulic engines, in the way usually adopted by the makers of the whole of the machinery, Messrs. Armstrong, Whitworth & Co., are provided for the service of the lock, the pressure-water for these being taken from the same supply.

XXVI.

HYDRAULIC GAS-STOKING MACHINERY.

Various operations carried on in gasworks can now be performed nearly automatically, by hydraulic machinery. The general arrangement of such machinery for charging and drawing the charges from the retorts in a large London gasworks is shown in Fig. 243 (cross section of Beckton gasworks). The coal is delivered direct f



SCALE 1/16" = 1 FOOT

FIG. 243.

trucks, from whence it passes to the breakers, which consist of rolls, the first pair having suitable claws for drawing in the coal. The broken coal is then raised by elevators to a large hopper, from which the small hoppers of the charging machines for charging the retorts,

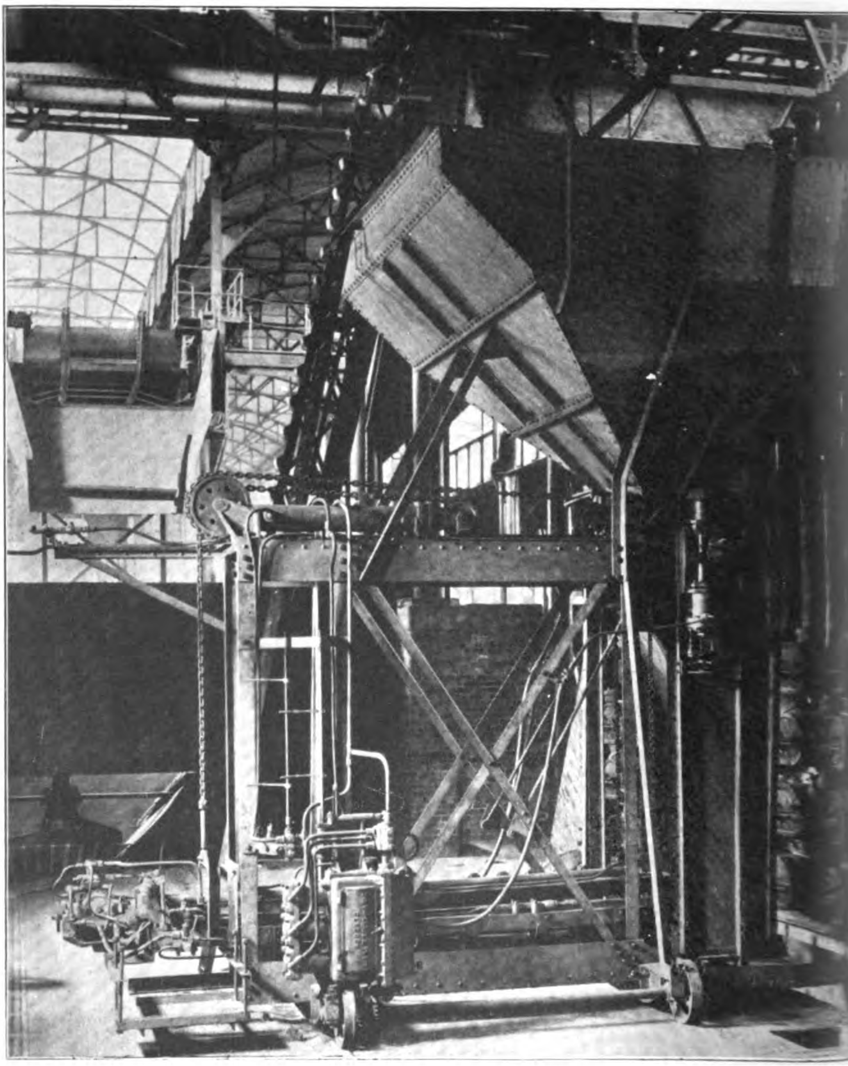


FIG. 244.

are supplied. These machines work automatically, a certain quantity of coal, from the hopper, being dropped in front of a pusher-plate by which it is pushed into the retort, about equal quantities being delivered into equidistant positions by a series of pushes, which are due to the action of two hydraulic plungers or rams, one giving the forward, the other the backward, motion.

The Arrol-Foulis charging machine is shown in Fig. 244. The hopper on the top of the machine is built of angles and plates into the frame of the machine, and has a capacity of 4 to 8 tons. The coal in falling to the retorts passes through a regulating drum. The amount of coal dropped by the drum, in a given time, can be readily adjusted. A shoot pan on a rising and falling charging beam receives the coal from the coal drum, and, on being pushed forward, bridges the distance between the machine and the retort mouth. The charge is now laid in the retort by hydraulic cylinders carried on the beam, which actuate a long

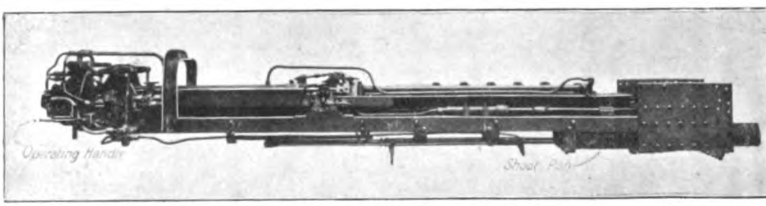
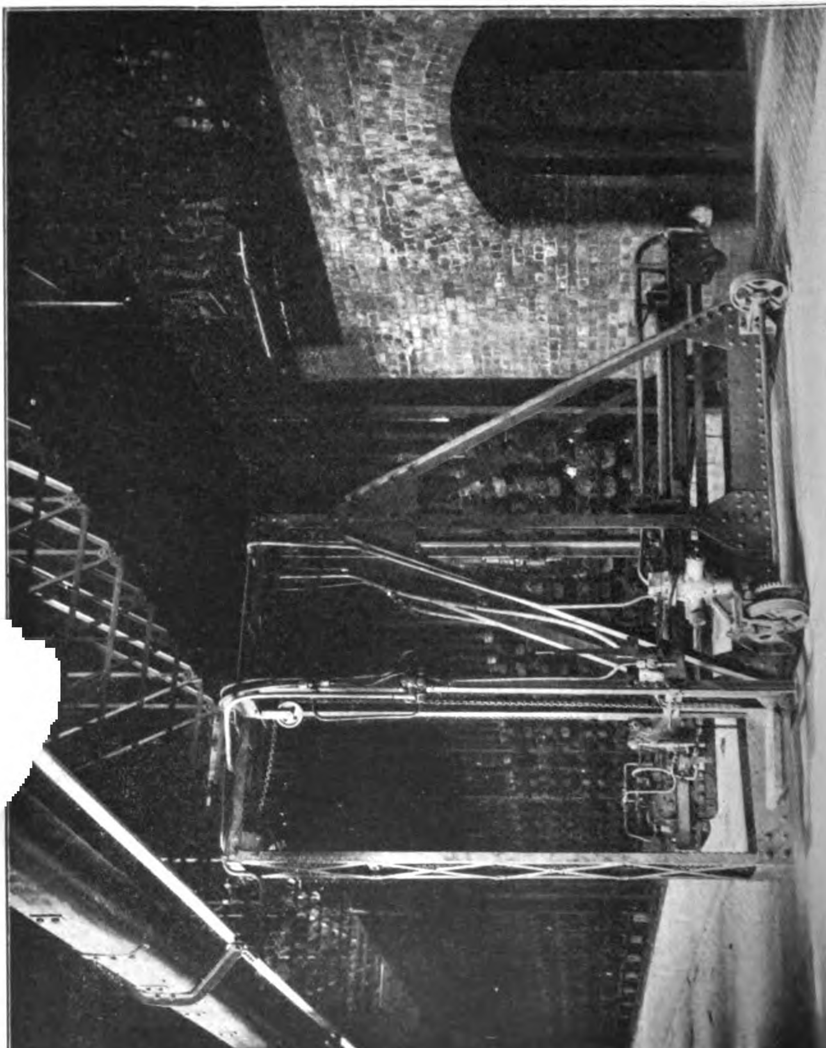


FIG. 245.

rod with front pusher-plate. The coal is first swept in, to within 12 inches of the back end of the retort, then the charges of coal are successively deposited at about 18 inches apart, the last charge leaving the retort filled to a depth of about 8 inches. The distances apart of the successive deposits are fixed by a bar on the beam, which engages with the charging ram in succession, the action being automatic when once the machine is set. All the motions are controlled readily by one man on the platform of the machine, and are all performed by hydraulic power, the machine itself being traversed along the house by a hydraulic motor. The pressure is from 400 to 700 lbs. per square inch, the water being conveyed by a flexible wired hose-pipe, sufficiently long to carry the machine over three to five beds of retorts. The hose is then shifted to the next swivel joint and the machine carried on. The frame of the machine is of plates and angles, the bottom carriage brackets being of cast steel, the whole machine weighing about 9 tons. A front view of the charging beam itself is seen in Fig. 245; the stoppers, regulating the distance apart of deposits, are seen to the right, just over the top of the beam.



The framework of the drawing machine is of the same construction, the beam rising or falling to suit the varying heights of the retorts, as in the charging machine. The beam carries a spear rod, with a drawing plate or clawt head. This plate can take up either the horizontal or vertical position. It is tripped into the horizontal position whilst the rod is being inserted in the retort above the glowing coke. When the rods begin to be withdrawn, the plate assumes the vertical position and cuts into the coke, bringing out the portion in front of it. Pushing and drawing rams, as before, give the required motions, the action being automatic when once the machine is adjusted. The total weight of the machine, which is shown in Fig. 246 is about 4 tons, and the time taken to draw a charge is about one minute. These machines (by Sir Wm. Arrol & Co., of Glasgow) are worked regularly on 48 retorts per hour in some of the London gasworks.

Thus, hydraulic machinery is made to perform operations which are known to be about the most trying which workmen can be called upon to perform. Before the introduction of these machines, the very slow and trying manual process alone was available, but in these days it is surprising that the machines are not used in all cases.

XXVII.

HYDRAULIC MACHINERY ON BOARD SHIPS.

The operations of loading, discharging, and storing cargo in, as well as the steering of ships, offer scope for the use of hydraulic machinery; whilst the manipulation of heavy guns and other appliances on board men-of-war renders it a necessity.

Messrs. Armstrong, Whitworth & Co. in the case of warships, and Mr. A. Betts-Brown in the case of ordinary vessels, may be mentioned as those to whom the success attained in these directions is mainly due.

It is evident that in passenger ships the adoption of hydraulic machinery for the performance of certain operations offers advantages in its silence, and the absence of that heat, dirt, and dust, which are the concomitants of steam machinery.

If hydraulic machinery is to be used with advantage on shipboard, one of the essential things required is a suitable accumulator or its equivalent, a sensitive hydraulic pressure governor. Deadweight accumulators may be ruled out as unsuitable, for evident reasons.

Steam Accumulator.

The steam accumulator of Mr. Betts-Brown, shown in Fig. 247, fulfils the requirements of the case. It may be described as a steam intensifier or pressure regulator. It consists of a steam cylinder A, with its piston B attached to a ram C passing through a stuffing-box into a hydraulic cylinder D. The steam pressure being, say, 70 to 80 lbs. per square inch, the area of the steam piston is about ten times that of the ram, hence the pressure of the water beyond the ram is from 700 to 800 lbs. per square inch. Steam is admitted to the accumulator through the pipe E, and exhausted by F, Q being the main exhaust pipe of the engines.

Steam is admitted to the pumping engines by N. The steam acting on B merely performs the function of the dead-weight in an ordinary accumulator. There is less inertia, and therefore less necessity for safety-valves on the hydraulic mains supplied from such an accumulator. When a safety-valve on ordinary mains acts some of the pressure-water is lost, or its energy is greatly diminished; hence this method avoids the loss referred to, and it has the further advantage that the pumps can act more quickly in starting the accumulator ram in what may be called its upward motion. The pumps are attached directly to the accumulator as shown, thus space is saved, and less rigid foundations are required, the whole weight being only a small fraction of that of an ordinary accumulator of similar kind. These are probably the most important features of the invention. Against these advantages may be put the loss due to leakage and condensation of the steam, but this loss is reduced by the use of separate admission and exhaust passages, as shown in the illustration. Compactness is probably the most valuable feature of such contrivances when used on board ships.

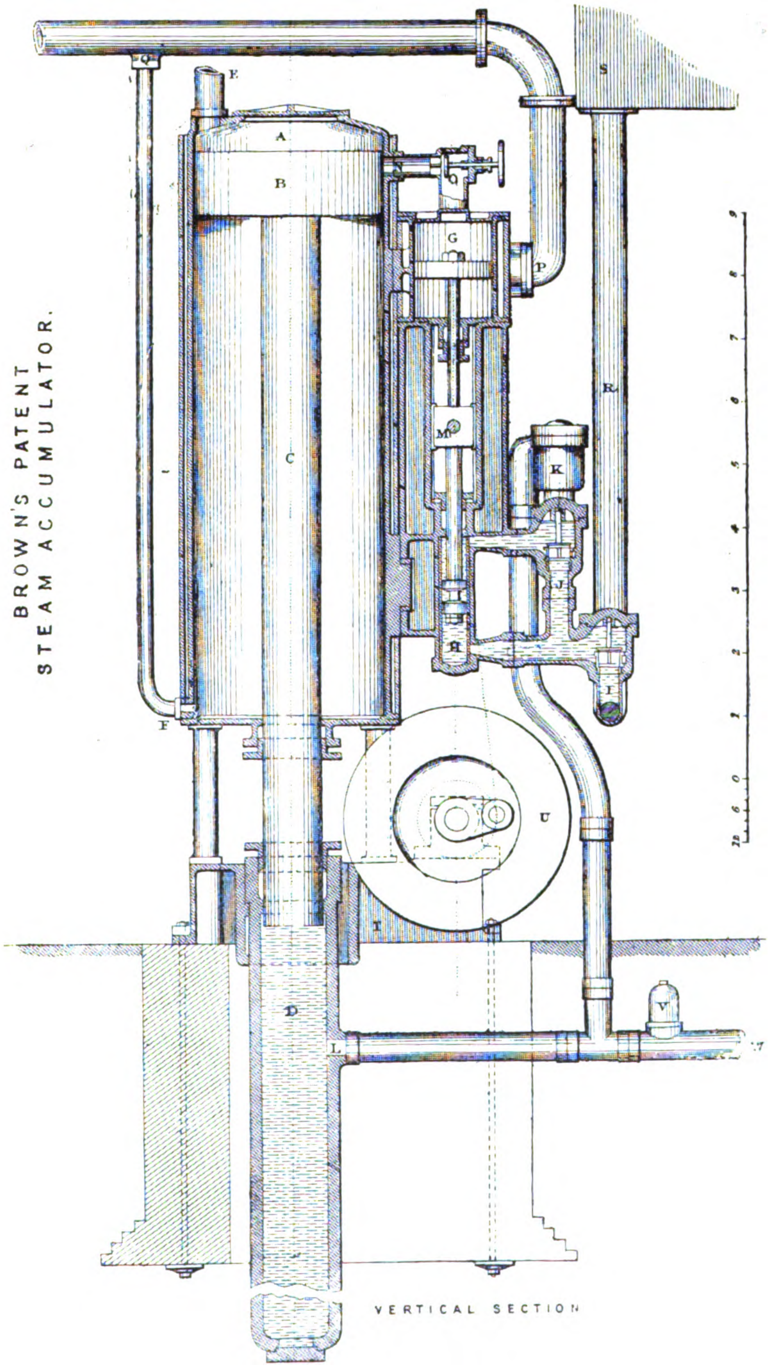
Other methods of obtaining the requisite "head" or load on accumulators on shipboard have been devised.

Compressed air has been used instead of steam, and springs have been employed to a considerable extent on British men-of-war.

Spring accumulators, i.e. accumulators in which the ram is pressed down by a spring, are naturally of very limited capacity, their stroke being short, and they do not give a constant pressure. They are more properly described as "equalisers of pressure," acting much like an air-chamber, giving an *average* pressure which for certain limits of stroke may be calculated.

By the introduction of an excellent hydraulic pressure governor (fully described later on) Messrs. Armstrong have been enabled to dispense with accumulators, or even "equalisers," on ships of war.

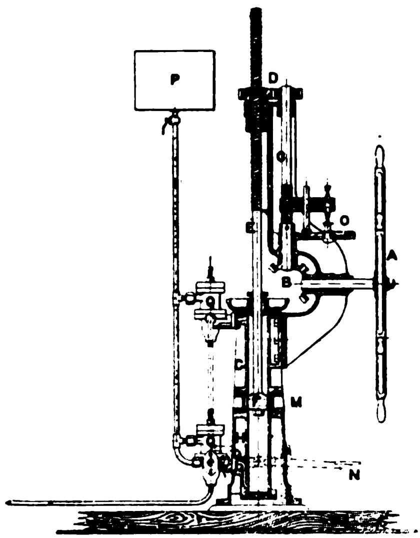
BROWN'S PATENT
STEAM ACCUMULATOR.



VERTICAL SECTION

Brown's Telemotor and Steering Gear.

This apparatus provides a hydraulic means of communication between the bridge and the steering engine, which in modern ships is often attached directly to the rudder-head to obviate the risk of the breakage of chains. Fig. 248 shows a vertical section of that part of the apparatus usually placed on the bridge. The steering wheel A drives a pair of bevel-wheels B, which actuate shaft C and the pinions D, one of which acts as a nut for the screwed piston-rod E. The view shows the piston F in mid position, within a cylindric distance piece forming part of the upper and lower cylinders G and H, this piece allowing free passage for the water from H to G. On turning the wheel A the piston F will travel one way or the other, and after passing one of the annular ports will act as a pump.



TELEMOTOR ON BRIDGE

Fig. 248.

In Fig. 249 the aft portion of the telemotor is shown close to the steering gear. It consists of a double-acting hydraulic cylinder I, with piston J and rod, which is connected to cross-head K, and from there to the lever S of the steering gear. The after steering wheel T being disengaged, the springs L L tend to force the piston J into mid stroke, and consequently bring the rudder amidships. The cylinder G on the bridge is connected to the corresponding end G of the cylinder aft by a pipe, also H forward to H aft as shown. The pipes, etc., being full of water, if piston F is forced up or down beyond the $\frac{1}{16}$ of an inch required to cover the annular port, piston J moves in a similar way. Thus the steersman on the bridge acts on the steering gear, through the piston J, its rod acting on the lever S of the steering engine valve.

The steam steering gear is shown in elevation in Fig. 250, and in plan in Fig. 249. In Fig. 250 A is a cast-iron tiller keyed to the rudder-head B, having at its after end a double jaw C fitted with bearings, and at the other end D a toothed segment. A toothed quadrant E is bolted to the deck; into this a pinion F gears. Shaft G is fastened to this, and bears a clutch-wheel H. The worm-wheel I embraces the clutch-wheel, the necessary friction for driving G being effected by the expanding clutch-wheel acting against a spring. The

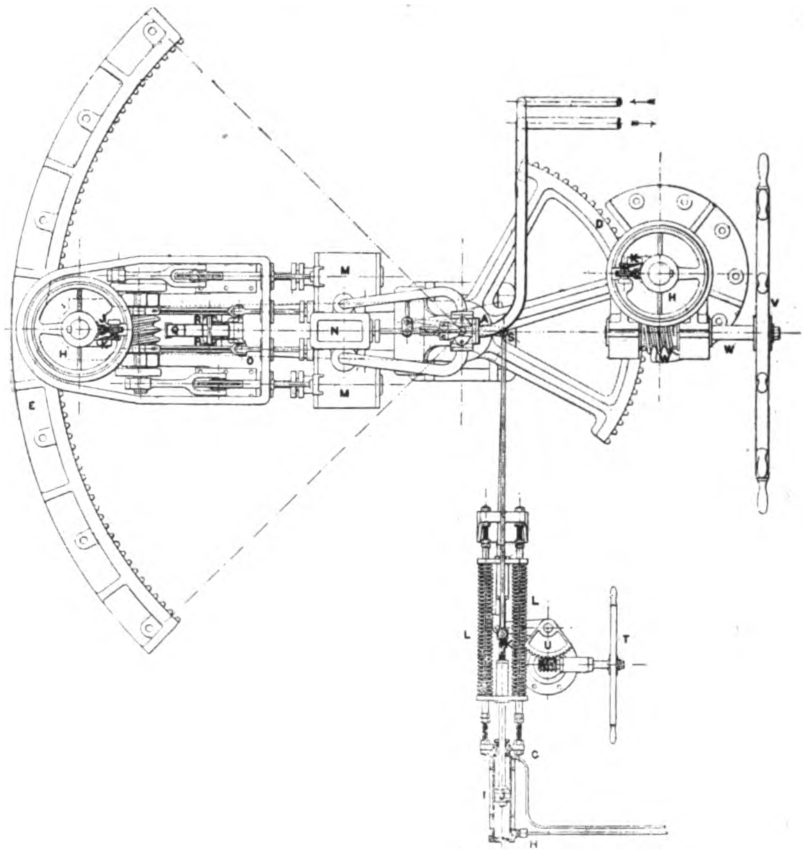
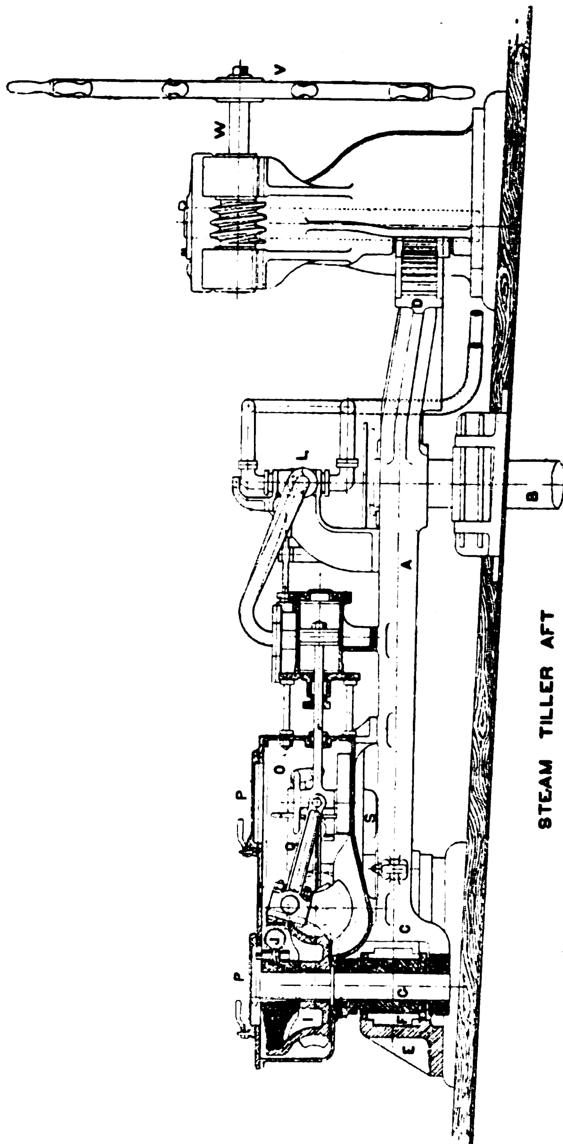


FIG. 249.

steering engines are carried on the tiller and move round with it, receiving and exhausting their steam through a double stuffing-box L on the rudder-head (see A, Fig. 249). M M are the steam cylinders, the slide valve N reversing the motion. This will show how the hydraulic arrangement acts on the steering engine.

The hydraulic communication has the advantage of being readily conveyed round corners and in out-of-the-way places, whilst by its use the noise of chains is avoided.



STEAM TILLER AFT

FIG. 250.

Special means are provided to supply any deficiency of water due to leakage, and an indicator is provided on the bridge to show the

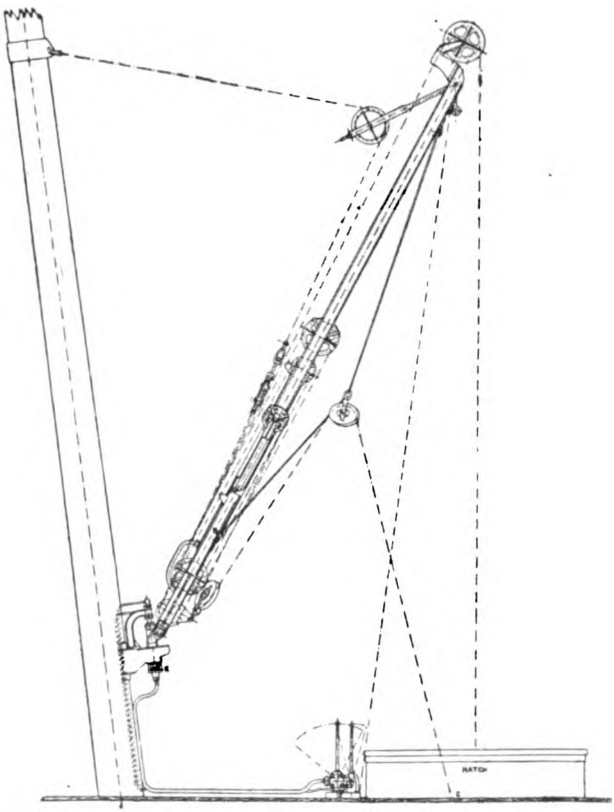


FIG. 251.

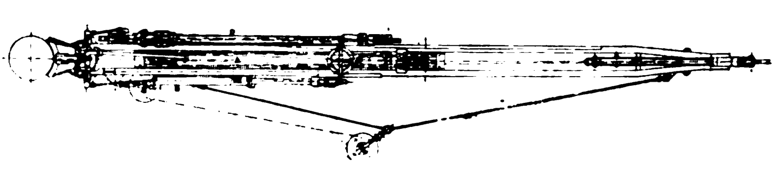


FIG. 252.

position of the helm at any instant. The way in which the rectification of the liquid in the pipes and cylinders is effected each time the tiller comes amidships is very ingenious. It may be indicated briefly as follows:—Suppose the gear by leakage or other cause gets out of correspondence, the hand-wheel being "hard over" forward whilst the springs have brought the gear aft to the amidships position; water is now sent into that part of the cylinder aft lettered H, which drives the piston J hard over towards the stuffing-box end of the cylinder, moving the piston F. When the piston F enters the distance-piece free communication is established between the cylinders on the bridge and both ends of the cylinder I aft. The springs then exert their force and put the gear amidships, the whole being again in correspondence.

Hydraulic Derrick.

Coming now to apparatus for loading and unloading ships, Figs. 251 and 252 show A, the lifting cylinder of a hydraulic derrick, with its ram and pulleys, the sluing cylinder B with its ram, pulleys and rope tackle made fast at C, the derrick being pulled in, and falling out by gravity owing to the rake of the mast. Fig. 252 shows a plan of the arrangement.

A derrick which can be actuated either by steam or pressure-water has been recently brought out, as the cost of fitting hydraulic appliances in some cases prevented the use of the hydraulic form of the apparatus.

Hydraulic Winch.

This apparatus is shown in detail in Figs. 253 and 254. Two winding barrels A A (Fig. 253), on their frames B B, with warping ends C C, are driven by the rams of three oscillating cylinders D (Fig. 254). These cylinders receive and exhaust their water through the trunnions by means of partially balanced cylindric valves E, somewhat like those of the Armstrong engine (Fig. 221), their casing remaining stationary, whilst the valve or plug E moves with the cylinder. The action of the rams on the crank is best understood from Fig. 254. In order that the power of the winch may be adapted to some extent to the load, an arrangement is provided for changing the throw of the crank, even whilst the winch is at work. The crank-pin is fixed in two discs F, which are placed eccentrically with respect to the axis of the winch, and each of these discs revolves within a recessed face-plate fixed upon the winch-shaft, and is connected to by sliding bolts I (Fig. 253). A series of corresponding h the two discs F are provided for the bolts so as to secure different positions; the two bolts are withdrawn simulta lever, and pushed in again by springs. Thus the st

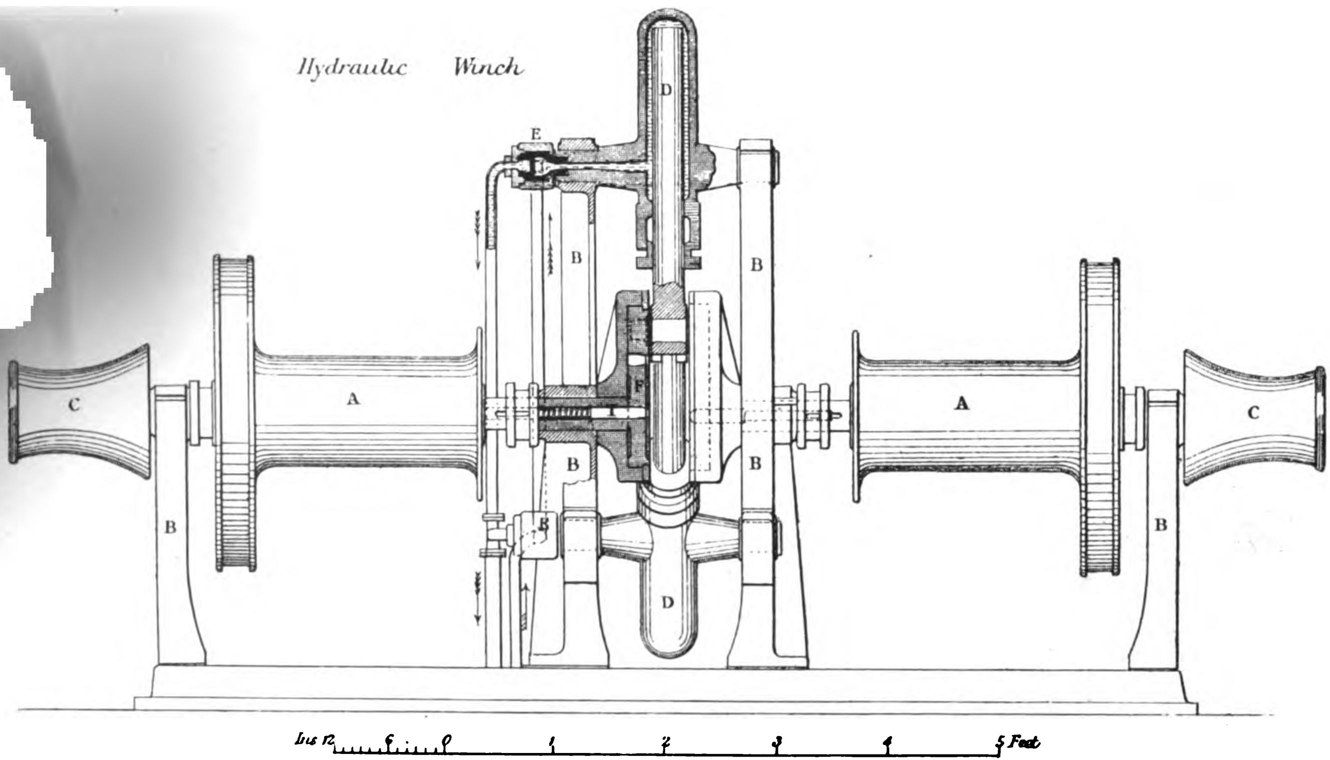


FIG. 253.

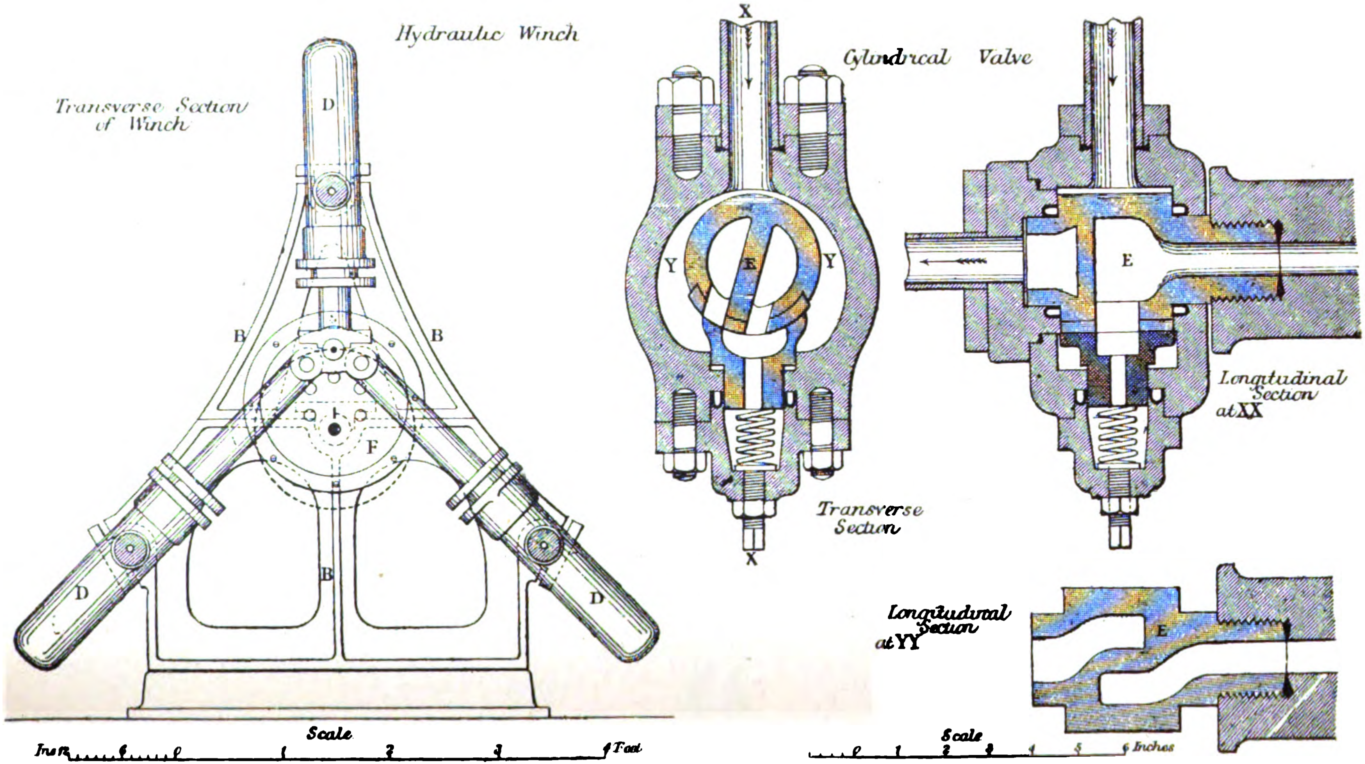

Hydraulic Winch
Transverse Section of Winch
D
B
B
F
B
D
D
Scale
Inch
1
2
3
4 Feet
Cylindrical Valve
X
Y
E
Y
X
Transverse Section
E
Longitudinal Section at XX
Longitudinal Section at YY
E
Scale
1
2
3
4
5
6 Inches

FIG. 254.

from 6 inches to 18 inches to suit the load. The values of the stroke may be 6, 9, 12, 15 or 18 inches, depending on the positions of the bolts. For the heaviest loads the speed is only about 20 revolutions per minute, and in that case the force on each ram is 3 tons, which with a stroke of 18 inches gives sufficient power without gearing, the rams acting directly on the winding barrel. Thus there is less noise and vibration than in the case of steam winches.*

HYDRAULIC APPLIANCES FOR SHIPS OF WAR.

The manipulation of heavy guns, and other operations which must be performed on board war-ships, render a complete hydraulic system necessary. For working gun-mountings hydraulic machinery possesses many advantages over that worked by steam, compressed air, or electricity, for the following reasons:—*Pipes conveying pressure-water can be led over a ship without causing heat or risk of fire. If a pipe be damaged no explosion takes place, and the place of damage is easily located. Hydraulic power can be applied directly to work presses, lifts, etc., and it works silently; and the hydraulic system is best suited to arrangements combining recoil-absorbers with means for running the guns out and in.* The perfection of the system now fitted in British war-ships is due almost entirely to the enterprise and talent with which the matter has been taken up at Elswick Works.

It may be well, in noticing a few of the recent developments in this field of marine engineering, to begin with the source of power.

HYDRAULIC PUMPING ENGINES OF A BRITISH WAR-SHIP.

Fig. 255 † shows a section of a pair of engines such as may be seen on a modern war-ship. They consist of two compound engines, each with a high and a low pressure cylinder, arranged in tandem. The piston-rods of these cylinders are connected to cranks which drive a shaft fitted with eccentrics for the regulation of the slide-valves, admitting steam to and exhausting it from the steam cylinders; the piston-rods being prolonged and forming pump-rods as shown in the illustration.

* For further information on this subject, see Mr. A. Betts-Brown's paper on 'Hydraulic Power for Loading and Discharging Steamships,' read before the Institution of Naval Architects, 1890.

† Figs. 255–257 are from drawings kindly supplied by Messrs. Armstrong, Whitworth & Co., and are similar to illustrations in 'Artillery. Its Progress and Present Position.' (J. Griffin & Co., Portsmouth.)

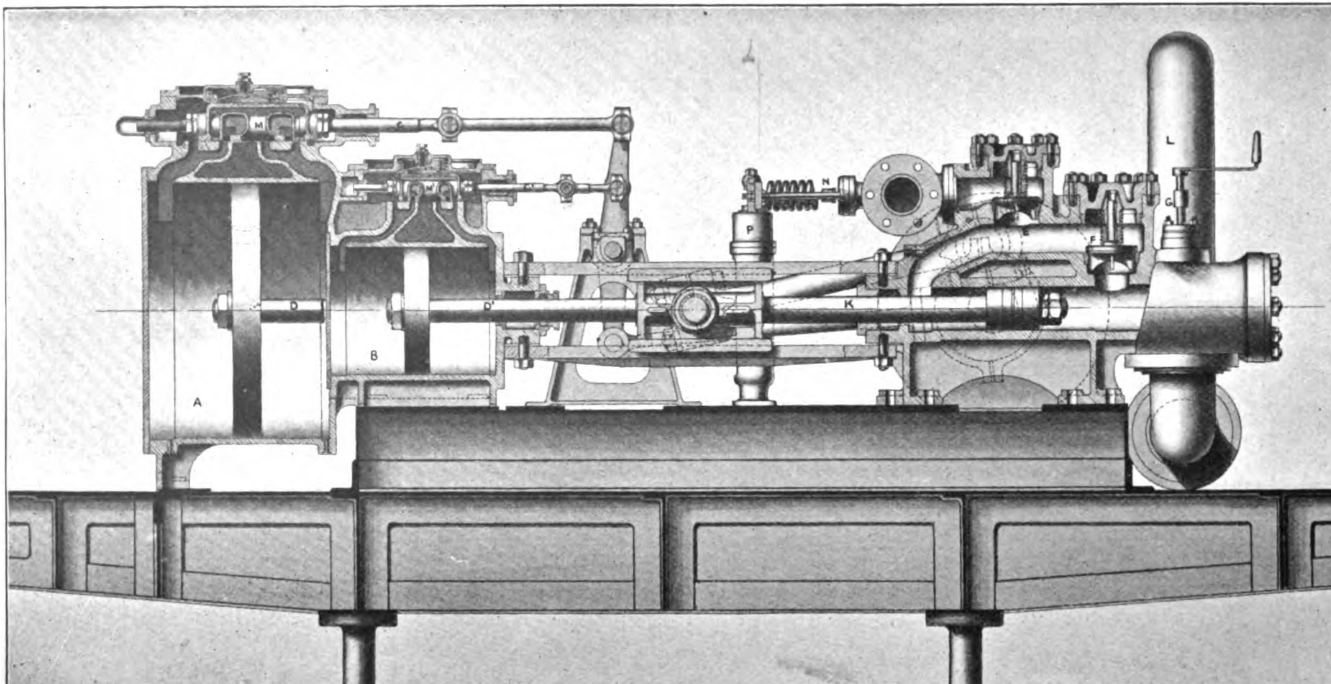


FIG. 255.

Each pump is provided with two delivery and one suction-valve. As the pump-rod moves in one direction, water is drawn into the pump through the suction-valve, and on the return stroke the charge of water is forced through the first delivery valve. After this the water may be said to divide, half passing through the second delivery valve into the pressure main, the other half passing again into the pump behind the piston, the rod of which is of half the sectional area of the pump cylinder. The next stroke may be said to force this half charge through the valve into the main; thus the pumps, being in duplicate, two for each engine, at each revolution of the crank-shaft there are delivered four half-charges or two complete charges of a pump. This well-known form of pump is fully described at p. 422.

Referring to the figure, A and B are respectively the low and high-pressure cylinders of one engine. The low-pressure piston is connected by rods D to the same cross-head as D′ the high-pressure piston rod, the cross-head being also attached to the pump-rod K. On the forward stroke of the plunger a charge of water is forced through the first delivery valve F, but since half this amount is required to fill the space left by the plunger as it advances, half only of the charge is forced through the second delivery valve E into the mains. On the return stroke the water on the other side of the plunger is forced through E, thus the whole charge has now been forced into the mains, a new charge being drawn into the pump. The sectional area of K is half that of the pump cylinder. The slide valves M M′ are worked by eccentric gearing from a shaft driven by connecting-rods from the cross-heads.

The hydraulic governor P is most ingenious. It is in communication with the pressure main, the water in which acts on a small plunger of the governor, which, when the pressure exceeds the normal intensity, rises against a spiral spring, this motion regulating the steam throttle-valve. To guard against the possibility of the engine and pumps racing, through, say, the fracture of a pressure main reducing the pressure against which the pumps act, a Murdock's speed governor is also attached. Two separate sets of pumps are provided, each capable of supplying the whole installation. This fact, and the excellent hydraulic governor, render an accumulator unnecessary.

Hydraulic Recoil Buffer.

Coming now to some of the operations effecte d by the use o pressure-water, we may first notice the guns. Gun mountings have many duties to perform, one very important one being the absorption of at least the greater part of the energy of the recoil of the gun. The hydraulic system of doing this is in universal use for guns varying from the small 3-pounder to the immense 110-ton gun. The suggestion of the use of hydraulic resistance for this purpose is due to the late Sir W. Siemens, but the first recoil buffer was made at Woolwich. The principle was employed by Lord Armstrong in one of his earliest hydraulic cranes. The Woolwich buffer was first intended to act as an auxiliary to frictional compressors. The principle of recoil buffers is to provide a resistance to recoil, due to the passage of a fluid through a small orifice or orifices at a high speed. Thus in the Woolwich buffer a closed cylinder was used, fitted with a packed piston-rod and a piston which had holes in it. The cylinder being filled with a liquid such as oil, the piston could only move by the passage of the liquid through the holes, hence when the movement of the recoiling gun was transmitted to the piston, the higher the velocity of the latter the greater the resistance offered by the piston. Mr. Vavasseur's improved form had a port which was fitted with a valve gradually closing the orifice, the movement of the valve being effected through the medium of studs fitting spiral groves in the cylinder.

The Elswick Buffer,

shown in Fig. 256, is that of a quick-firing gun. T is the piston, U the valve key. The piston-rod is attached to a horn θ on the gun. The recoil cylinder X is made of forged steel, and is screwed into a bracket Y. V is the controlling ram, W a plug for adjusting the action of V, and Z a small radial hole to admit the liquid into the cavity in the centre of the piston-rod during recoil. β is a bronze ring to prevent seizing between piston and cylinder. It is easy to see how the appliance acts.

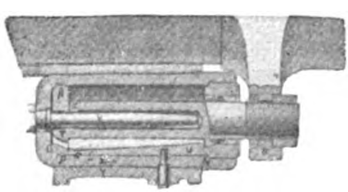


Fig. 256.

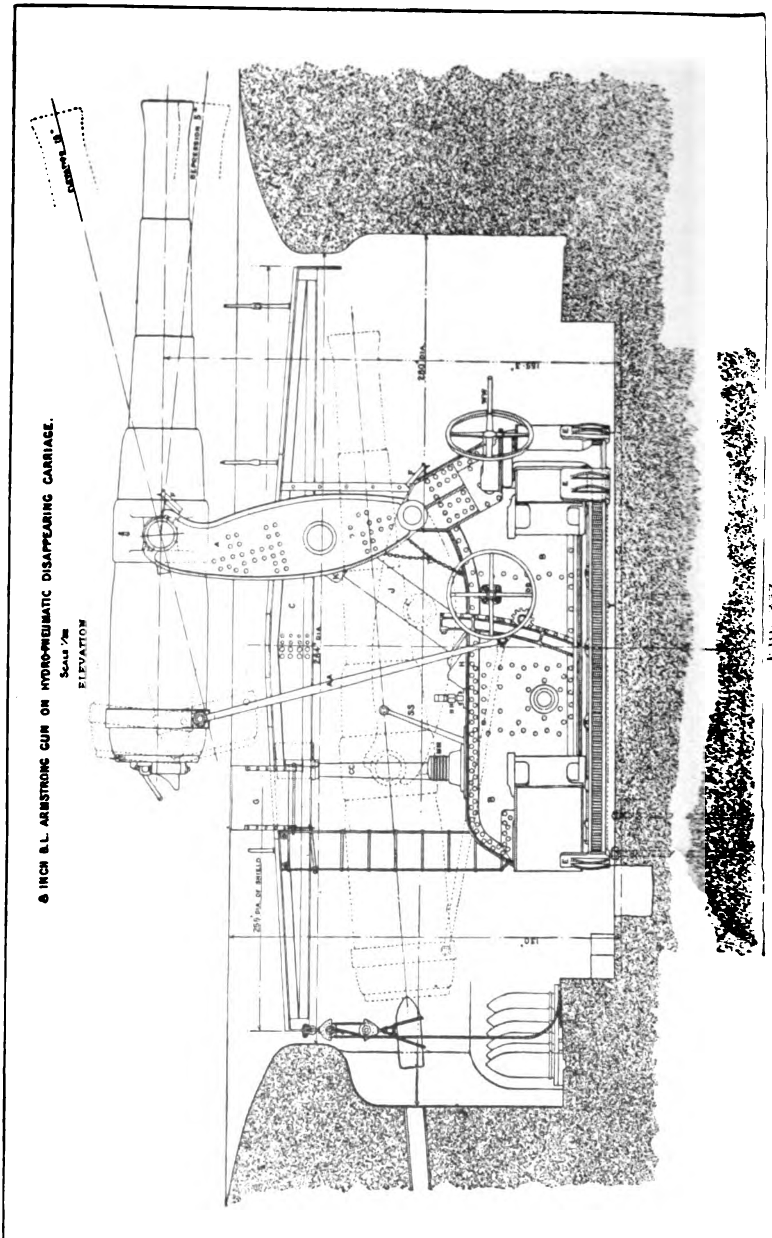
It is on the *tension* principle—in other words, the recoil pressure acts on the annular area of the piston round the rod, which is in tension, the fluid finding its way through Z into the cavity left behind and round V, during recoil. The use of the controlling ram is very important, regulating the speed of the "running out." As the gun runs out again (from right to left) after recoil, the liquid escapes from the central cavity by a passage which is gradually closing by the ingress of the taper ram V, and thus the gun runs out with a decreasing speed, arriving gently at the front buffers or stops.

Hydro-Pneumatic Disappearing Mounting for Guns.

This is one of the most important of the applications of fluid pressure to gun manipulation; though given here, it is more usually employed on land, the gun being fired and loaded from a pit. It will be understood from an examination of Fig. 257. It acts somewhat as follows:—When the gun is in the firing position it is kept up by the liquid, which is at a sufficient pressure to force out the ram J of the recoil press H, shown underneath the breech of the gun. This pressure is obtained by compressed air acting on the liquid. When the gun is fired the liquid is forced through recoil valves, *not* into an exhaust chamber, but into the compressed air chamber, further compressing the air, thus assisting to a small extent to absorb the recoil, which is mainly, however, absorbed by a recoil apparatus. When the energy of recoil is absorbed, the gun has descended into the lower or loading position, the air being much compressed, but it cannot force the liquid back into the recoil press since the valves are "non-return" valves. If the charge be reduced, the gun is not brought quite down. A pump is provided for completing the lowering, or to bring the gun down without firing, if required.

When the gun is loaded, and all is ready, a valve is opened, and the liquid passes from the air compartment into the recoil cylinder, the ram is forced out and the gun rises, coming gradually to rest in the proper firing position, owing to the gradual automatic closing of the inlet valve. The gun having already been properly trained on the target, it can be fired immediately it is up, thus it only remains in an exposed position for a very short interval of time. This mounting, though originally designed for 6-inch guns, has been applied with the greatest success to 68-ton ($13\frac{1}{2}$-inch) guns, the energy of recoil being in this case 730 ft.-tons.

Referring to the figure, it will be seen that the recoil press has trunnions which rest on bearings in the platform. Thus the press can accommodate itself to the circular path of the upper end of the ram. Two rods connect the breech end of the gun to the gear. The dotted lines show the position of the gun when down. The lower ends of the rods can be set when the gun is down, so as to give any required elevation to the gun when it rises. The gear has a friction clutch in it which allows relative motion when the gear is subjected



8 INCH B.L. ARMSTRONG GUN ON HYDRO-PNEUMATIC DISAPPEARING CARRIAGE.
Scale 1/32
ELEVATION

to an *extra* strain, which sometimes is experienced on firing. There is also an automatic brake fitted to the elevating gear. Sights are provided altogether independent of the gun, which can be "laid" when down, and the usual sights, as well as reflecting sights, are provided, to be used if required. A pump is said to give the requisite initial pressure of about 1500 lbs. per square inch, for a 9·2-inch gun. The pit in which the gun is mounted is provided with an overhead shield to prevent the entrance of shells. This disappearing mounting has met with great success, and has been adopted or copied by many nations. Spring mountings have been used with some success.

Hydraulic Disappearing Mountings.

These have also been constructed. The principle will be understood from Fig. 258. G is the gun resting on its platform, which rises and falls on the ram A. The accumulator has two large rams A A, and a much smaller one B. These three rams rise and fall

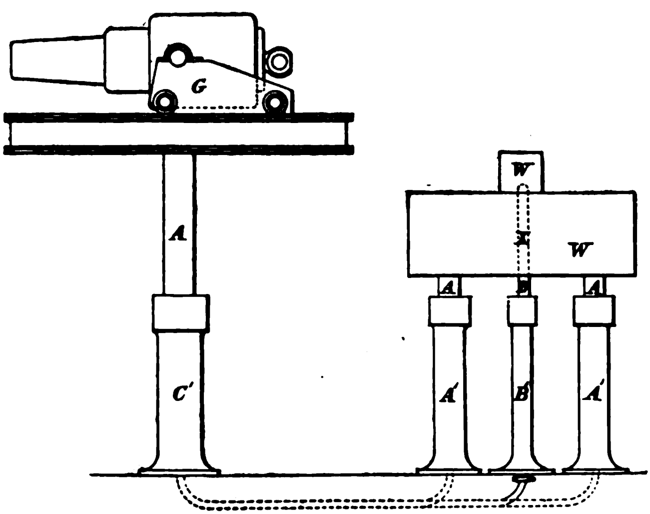


Fig. 258.

ther, bearing the accumulator load. Now if press B′ merely municates with a neighbouring tank, whilst A′A′ form the presses ng pressure-water to the gun-press C′, the pressure in A′ and C′ will sufficient to raise the gun. Now let B′ no longer open into a low-pressure cistern, but let it, too, act as one of the accumulator presses; then the pressure of the water per square inch will be less than before, since the weight W is supported by a greater plunger area; the consequence is the gun will sink downwards with its platform. As water goes into the cistern every time B falls, water must be supplied by a pump, to enable the accumulator to lift the gun to its proper position every time. It will be seen, however, that the arrangement acts somewhat like the balance of a hydraulic hotel lift, only a very small quantity of water being lost during each operation. The expense of this somewhat cumbrous arrangement for raising and lowering guns, and the comparative slowness of the action, have prevented it from being so largely used as might have been expected.

Hydraulic Engine for Moving Turrets.

Returning again to gun manipulation on ships of war, we notice that in many modern war-ships the gun, or a number of guns, are placed in a turret, which is made to revolve by hydraulic power. The movement of the "training gear" is one well adapted for hydraulic machinery, considerable rapidity being necessary with a

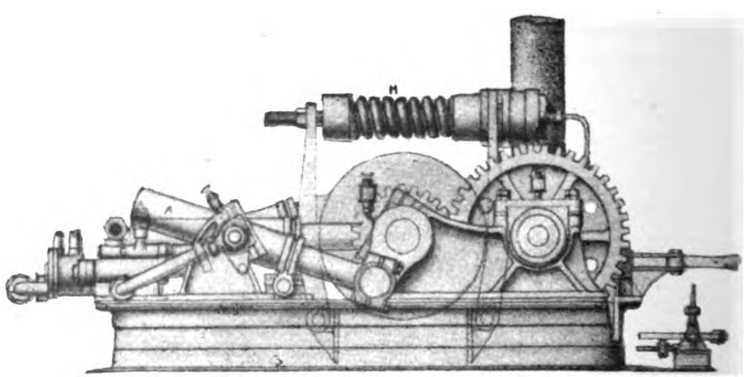


Fig. 259.

steady creeping movement. Sometimes the training engine is placed inside the turret, to which pressure-water is led by properly jointed pipes, the engine working a pinion which gears with a fixed rack. Often, however, the engine is placed outside and drives a pinion gearing with a rack fixed to the turret, this being probably the better method. One of the engines is shown in Fig. 259. It will be seen that it has three oscillating cylinders with rams operating cranks set at 120° to each other. The slide valves of the cylinders are worked by rods from an eccentric point in the trunnions, about which the cylinders oscillate. Each cylinder has a reversing valve, but all three reversing valves are worked from one shaft, this shaft being controlled by cylinders termed the "starting and reversing" cylinders. A hand-wheel in the sighting station of the turret sets a small slide-valve which admits water to one end of the above cylinders; the pistons of those cylinders move and turn the shaft which is connected to the reversing valves. This shaft is also connected to the small slide-valve just mentioned. As soon as the pistons of the "starting and reversing" cylinders have come to the point necessary to give the desired speed to the engines, the small slide-valve is closed, the engines continuing to work until the small slide-valve is set in its central or "cut-off" position. It is sometimes necessary to stop the turret quickly, hence a powerful brake is provided, by means of a strong spiral spring, which is always trying to press a block-brake on the training engine shaft, but it is kept from doing this by the water in a hydraulic cylinder connected with the brake. The same action of admitting water to the engines releases this block from the shaft, hence directly the supply is cut off from the engines the brake is applied.

Referring to the figure, a hand-wheel in the turret actuates the small slide-valve referred to, pressure-water being admitted to the "starting and reversing" cylinders G shown to the left, the pistons of which act on the valves which start or reverse the main cylinders A. Thus the amount of movement of the valves of the "starting and reversing cylinders," and therefore the speed of the engine, depends on the amount of opening of the slide valve by hand, though it is almost immediately afterwards closed automatically.

One trunnion of each oscillating cylinder works a slide valve for the admission of pressure or the release of water through the *opposite* trunnion. Pressure is also admitted—at the same time as to AAA—to the small cylinder containing the piston which prevents the brake from acting. H is the large spring for closing the brake.

Foreign vessels are sometimes fitted with training gear similar to that found on cranes. Two hydraulic cylinders placed near the turret have their rams connected by a pitch chain, which passes round the base of the turret; hence when one cylinder is opened to pressure and the other to exhaust the turret revolves. This arrangement requires considerable space, and the stretching of the chain gives rise to unsteadiness and difficulty.

It may be mentioned here that manganese-bronze pipes are now used for the conveyance of pressure-water in these installations, such pipes being stronger, weight for weight, than copper.

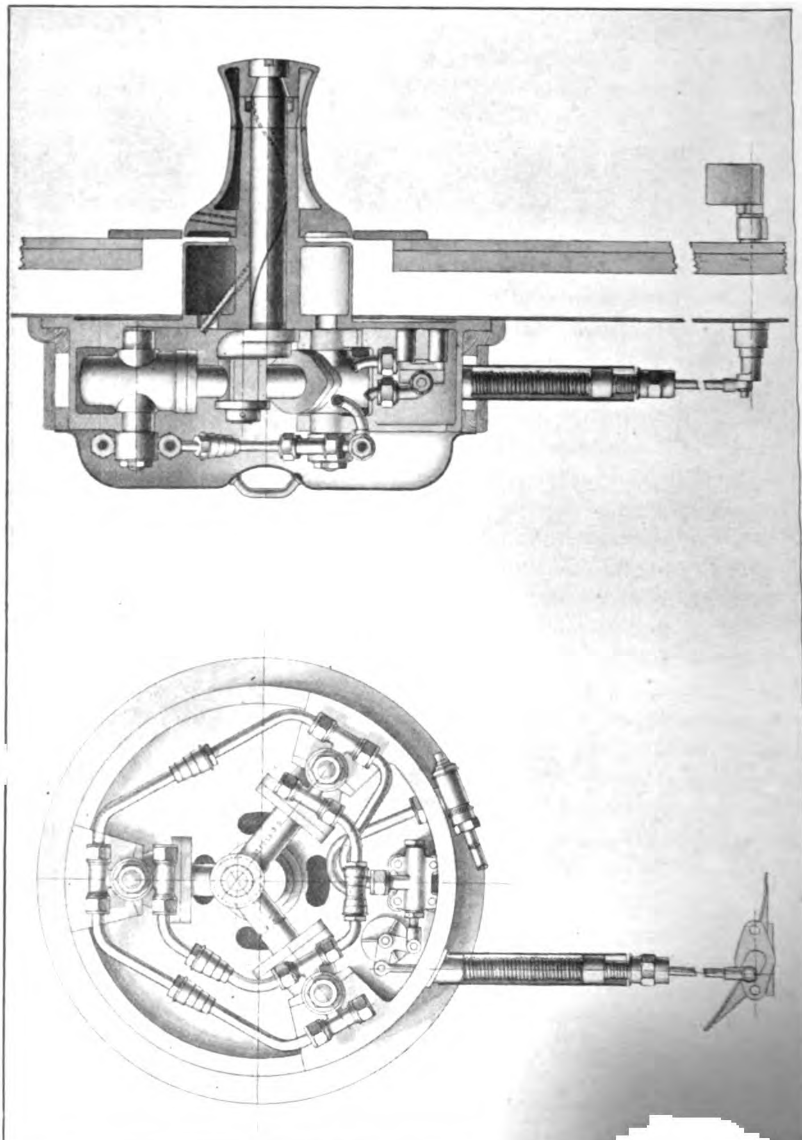


FIG. 260.

Hydraulic Capstan for Magazines.

Space does not admit a reference to more than a few of the purposes to which hydraulic power is applied on such ships as the most modern war-vessels of the British navy. An interesting case must close our brief notice. The use of a hydraulic capstan for moving heavy ammunition in magazines saves much labour.

The magazines are fitted with a complete system of machines. The way in which the ammunition is stowed—the shell, etc., being in long "bays" divided into compartments—admits of the use of an overhead travelling crane for fore and aft transport, as well as hydraulic purchases—very similar to hydraulic cranes—for hoisting and traversing. The projectiles are thus hoisted and traversed over to bogies which run on rails; they are then brought to the charging cages which work up and down the ammunition shafts. These cages are fitted with safety gear, the suspending wire rope being fastened to levers which bear teeth at their other ends, which, if the wire breaks, are forced against the guides and support the cages. The "hydraulic rammer," a beautiful contrivance, takes the ammunition from the cage and forces it into the gun.

The hydraulic capstan forms a useful feature of the magazine machinery, by which the haulage of the bogies containing powder-cases, etc., can be readily effected. Fig. 260 shows the arrangement of one of these, consisting of a bollard worked by a small oscillating three-cylinder hydraulic engine, the cylinders being set at 120° to each other. The pressure-water is admitted by a throttle-valve worked by the foot, a spiral spring closing it when the foot is removed. The trunnions of the cylinders have ports cut in them, and these ports are brought into the pressure and exhaust positions by the oscillation of the cylinder, in a way fully described under the heading "Hydraulic Engines."

XXVIII.

HYDRAULIC MACHINE TOOLS.

Probably in no department of engineering has the use of hydraulic power met with more success than in its application to certain machine tools. This success is due to the peculiar suitability of pressure-water as the motive agent for the performance of a certain class of operations requiring the exertion of a great force with comparatively slow motion, as in punching, riveting and the like.

The widespread and successful use of hydraulic machine tools in our time is largely due to the talent, energy, and persevering initiative of the late Mr. Ralph Hart Tweddell, M. Inst. C.E., who may be called the father of the system. Mr. Tweddell was educated at Cheltenham College, serving his apprenticeship with the firm of Messrs. R. and W. Hawthorn, of Newcastle-on-Tyne. Whilst there, Mr. Tweddell gave evidence of his inventive powers, and took out his first patent at the early age of 20. His attention was soon attracted to the wasteful and imperfect methods of transmitting power, then exclusively employed in engineering works. In large works, often covering many acres, either wasteful steam pipes to operate motors, or the usual cumbrous and noisy shafting and gearing were alone employed to transmit power, often over long distances. Especially when this power was used intermittently, and yet, to be always available, the shafting must continually revolve, as when used for such operations as riveting, punching, shearing, etc., it became evident to Mr. Tweddell that a more efficient system was necessary. Yet the first successful attempts to transmit power by hydraulic means, and utilise it for this class of work, were made rather with a view to improvement of the work than greater efficiency.

The first hydraulic machine tool was a small one, constructed for the purpose of tightening the ends of the tubes of marine boilers in their tube-plates. This proving successful, the inventor's attention was next directed to some of the other weak points in the processes then employed in boiler construction, and the bad character of the riveting seemed to offer a promising field for improvement.

Steam pressures increasing meant thicker plates and more of them to be riveted together with larger rivets, and as hand-riveting did not improve, hydraulic power was tried. This proved a great success, Tweddell's first stationary riveter, made and used by Messrs. Thompson and Boyd, of Newcastle, proving the advantages of the new system. Mr. Tweddell patented his first stationary riveter in 1865, and his portable riveter in 1872. Thousands of hydraulic riveters have since been made, and this method of riveting is now specified for in all high-class work in which it is possible to employ it.

The peculiar fitness of hydraulic power for such work is evident, when we consider that in ... say, rail ... irders, a pressure of 40 tons—equal to the wei... e lo... —has to be applied to the hot rivet in order ... pro... the head of the rivet, but at the same time to bring the plates into the closest contact, and cause the rivet to fill every part of the hole. The pressure of the water being known, the greatest force the hydraulic riveter can exert can easily be calculated, and this force cannot be exceeded, nor the machine strained, as an ordinary riveter would be if forced beyond its capacity.

This is where the great advantage of hydraulic riveting comes in, some of our largest riveting machines exerting on the plates a final squeeze of as much as 150 or 200 tons; thus a large number of plates can be brought very closely together and the rivets, often 9 or 10 inches long, forced into every portion of the holes.

Another important advantage is that the stroke can be varied as required, this variable range of travel being practically unobtainable in geared machines.

After the excellence of the work done by hydraulic riveters was clearly demonstrated, punching, shearing, and other machines actuated in the same way quickly came into use, and the great efficiency of such a system of machines driven by water from one central source, became apparent. The long lines of shafting, with their attendant dirt, noise, and danger, in many cases gave place to lines of piping, buried out of sight in the ground, meandering in and out by curious paths to the machines to be supplied. The power thus available is, as it were, stored up in the pipes and accumulator, and when there is no demand for it there is no waste—unlike the revolving shafts and gearing which are always wasting energy if available for immediate work.

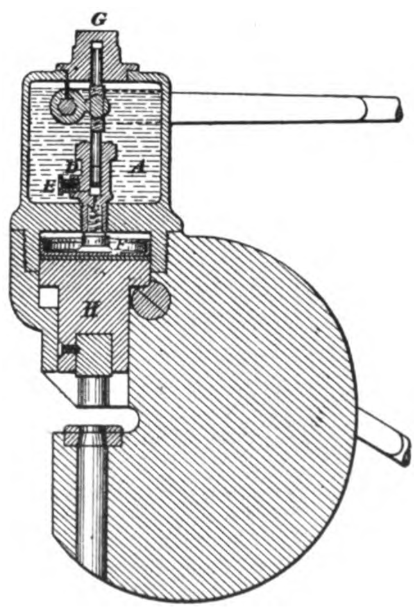


FIG. 261.

PORTABLE MACHINES.

In what may be termed the manual hydraulic epoch, hydraulic machines somewhat like the hydraulic jack were employed. Fig. 261 shows a section of one of these machines, a hand punching "bear." H is a stout ram capable of moving in a cylinder formed in the casting which constitutes the framework of the machine. This ram is rendered watertight by a cup-leather F, which is fastened to the top

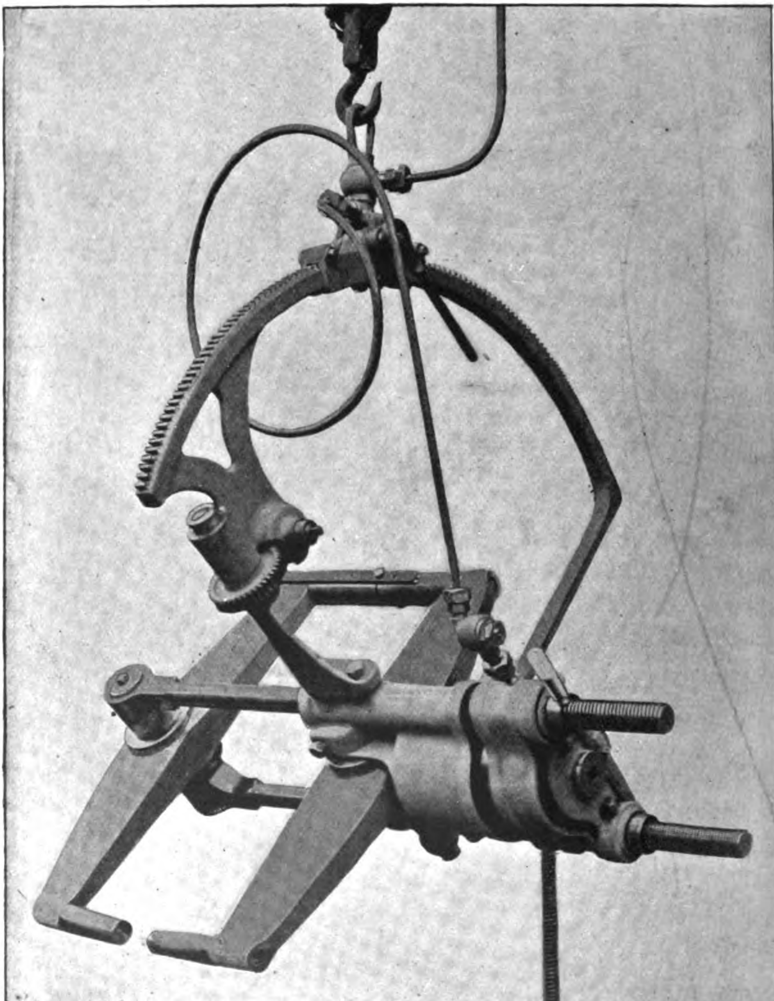


FIG. 262.

of the ram by a set-screw and washer. Above the ram is a casing containing a reservoir A, usually filled with water. On working the upper handle, water is first drawn in, and then forced down by the pump-plunger, seen inside its barrel D, into the space above the ram. Thus the ram, with the punch let into it as shown, is urged downwards and the punch forced through the plate placed in the space or jaws underneath.

The lower handle is provided for withdrawing the punch by admitting pressure-water to the annular space under the head of the ram.

This machine is slow, and its power very limited, though it proved, in those early days, a useful portable apparatus for small jobs.

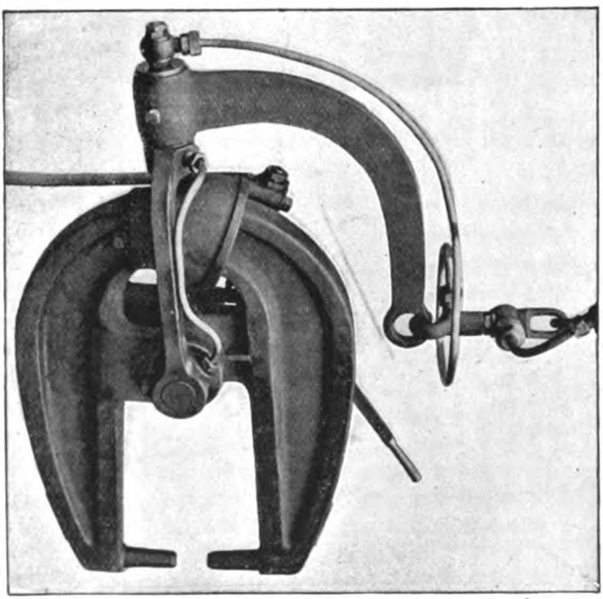


FIG. 263.

The first portable riveter supplied from pressure mains is illustrated in Fig. 262. It will be seen that this machine consists of two strong jaws movable about a hinge at one end, having the riveting dies for shaping the rivet at their other and opposite extremities. The jaws and dies are closed on the rivet by the ram of a small press which forms one extremity of shackles embracing the centre of the riveter. The pressure-water is led to this press by a more or less flexible pipe, and as the ram emerges from its press the shackles press the dies tightly on the rivet and plates. Worm and worm-wheel gearing are provided for turning the machine into various positions. Of course in riveting—unlike punching—very little power is required for a back stroke. The machine can readily be altered so as to rivet at either end, or can be made to suit different sizes of work. This riveter, patented in 1871, was used to rivet up ships' frames. It was somewhat expensive, and not so well suited for heavy work as a direct-acting machine. In the following year the first bridge was riveted *in situ* on an English railway by portable machines.

The frames and keels of our great Atlantic liners were soon

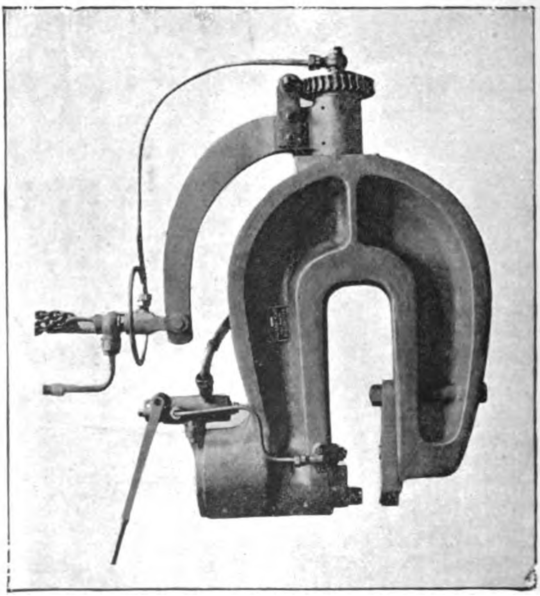


FIG. 264.

riveted almost exclusively in this way, and portable riveters gradually developed into neater and more perfect forms. Figs. 263 and 264 show typical modern instances, and modifications by others, of the Tweddell system. In Fig. 263 the jaws are movable about a pivot or gudgeon placed nearly centrally, the hydraulic cylinder and ram being (in section) curved to arcs of circles with the axis of this gudgeon as centre, and thus a very compact machine is obtained; the egress of the ram under the influence of the pressure water, which is led from pressure mains to the curved cylinder by the flexible pipe shown, causing the closing of the dies on the rivet and plates.

Another form of portable machine, in this case direct-acting, is shown in Fig. 264. Here the framework of the riveter is a solid piece of metal with the press immediately over one of the dies, to which the ram is attached. The provision for slewing is shown in the figure, the flexible pipe allowing any necessary motion. This is really a small stationary machine, supported and moved by suitable gear. The pressure-water which is used to work the machine is also employed to actuate the suspending gear. This is an advantage, as many of these portable machines weigh several tons, and are capable of exerting a force of 30 tons on the rivet, closing three or four

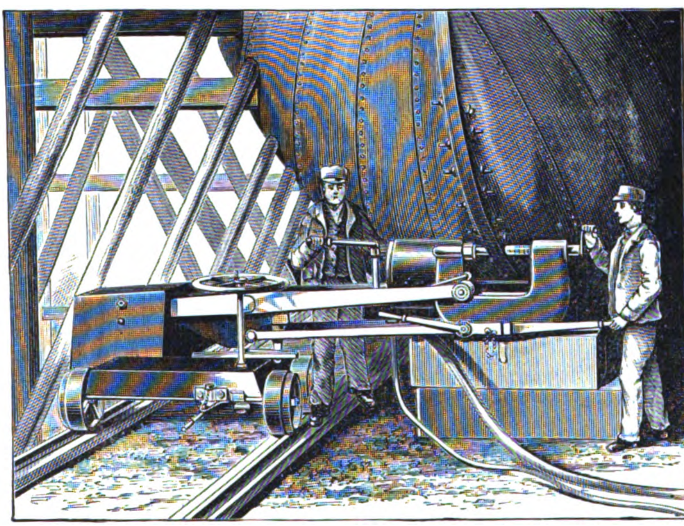


FIG. 265.

rivets per minute. Fig. 265 shows an interesting application of such machines to rivet up the keel of an Atlantic liner.

In Fig. 266 is shown a section of a portable riveter. The die D is forced out by pressure-water acting on the main ram R, which contains the plate-closing ram P; to the latter is attached the plate-closer C. Q is a ram under constant pressure which withdraws R and P, when the cylinders of the latter are opened to exhaust.

Water is admitted to act on P, through the telescopic joint formed by T and E, whilst pipe S conducts water to act on R, and the constant pressure ram is acted on by water admitted at A.

The riveter is slung by L, and is allowed to swivel by the joint at J.

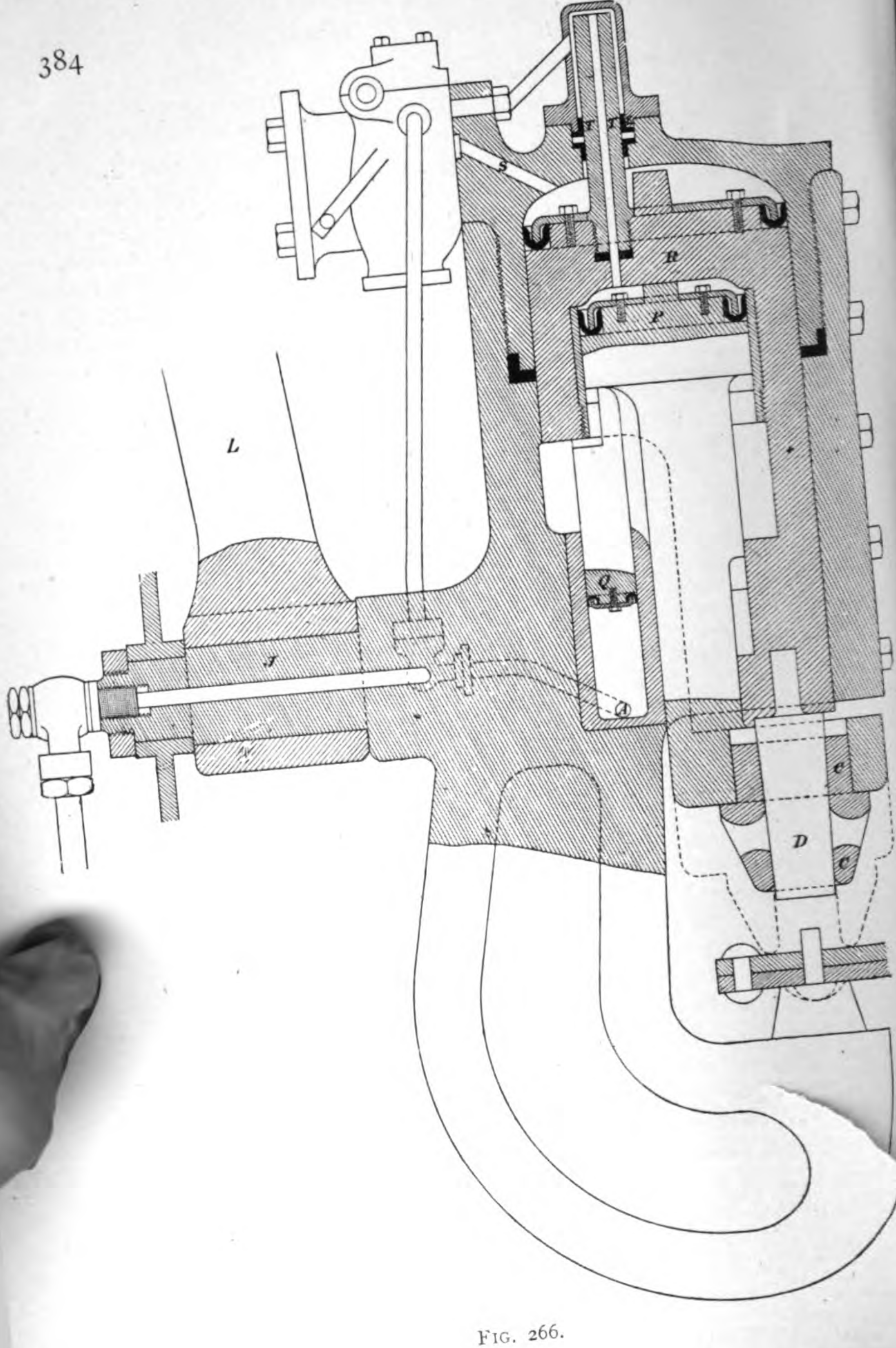


FIG. 266.

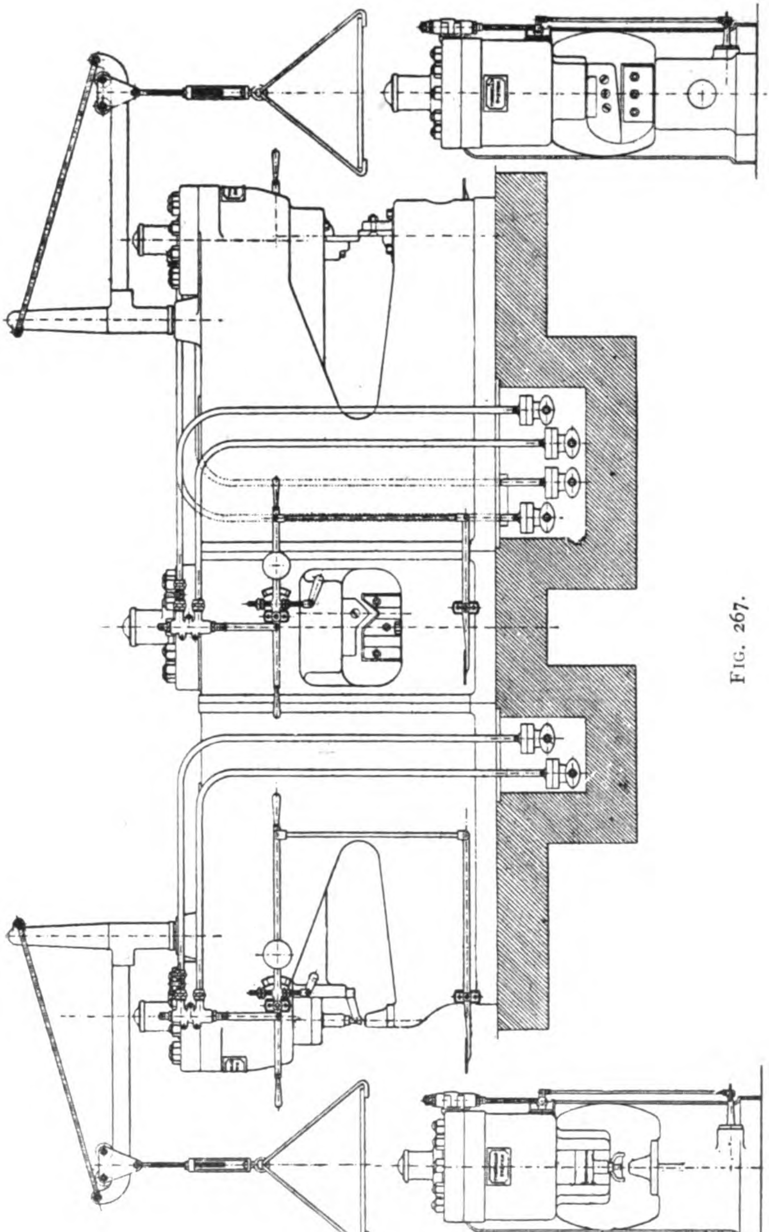


FIG. 267.

STATIONARY MACHINES.

Fig. 267 shows a combined punching and shearing machine of the stationary type. There is a cylinder and ram for punching, also a separate cylinder and ram for shearing, and an additional set for angle cutting. Two pipes communicate with each cylinder, one for pressure and one for exhaust. Piston valves, worked by the levers shown, control the motion of each ram. When the pressure-water is exhausted from each cylinder, a piston, which is always under constant pressure, causes the working ram to rise. There is an automatic cut-off gear for restoring the levers to their shut-off position when the main ram reaches the extremity of its downward stroke. Small cranes are provided for handling the plates.

Fig. 268 gives a view of a large stationary riveter, capable of exerting on the rivet and plates a closing pressure of 150 tons.

The next figure but one gives a view of the valves and other internal parts of a similar machine.

INTERNAL ARRANGEMENTS OF STATIONARY PUNCHING MACHINE.

A section of one of Tweddell's hydraulic punching machines is shown in Fig. 269. B is the main ram, to which is attached the punch P. When B is forced out of its cylinder A by the ingress of pressure-water at C, the punch is forced through the plate, which has previously been placed over the die D. K is a small piston whose rod is R. This piston is used for withdrawing the punch, the space under K being always open to pressure. The upper side of K is open to the atmosphere. The spaces at S are filled with hemp or other packing. The method of making B and R water-tight by U leathers is clearly shown in the illustration. The lever L is connected to the hand lever which works the controlling valve, so that when the punch has pierced the plate the space C is opened to exhaust, and piston K withdraws the ram B.

SECTION OF STATIONARY RIVETER.

In Fig. 270 are seen three views, including a section, of a large hydraulic riveter, such as that shown in Fig. 268, with Tweddell's patent water-saving device. The arrangement may be briefly described as follows :—

The cylinder M, working on the fixed ram K, bears the riveting die D, whilst inside M works the piston R, which is connected to the annular plate-closing tool C. The heated rivet having been placed in position in its hole, D and C are advanced to the work by the automatic drawback piston S, pressure-water being admitted to both sides of it by the two pipes P P′, the advance being due mainly to

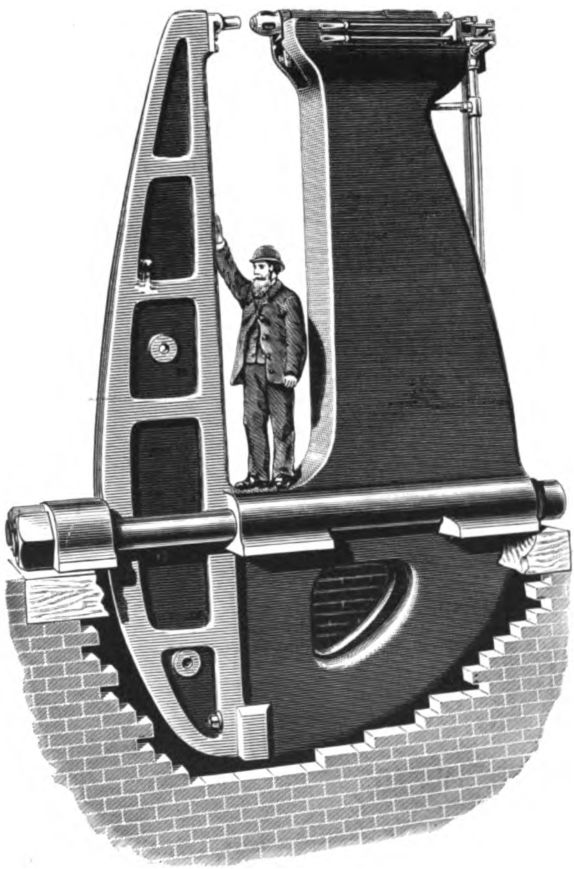


FIG. 268.

the preponderance of pressure on the full or face area over that on the annular area of this piston, but assisted somewhat by low-pressure water, which is, at the same time, taken into the main and plate-closing cylinders from a tank at a height of about 20 feet above the machine. The cupping die and plate-closing tool having been brought up to the

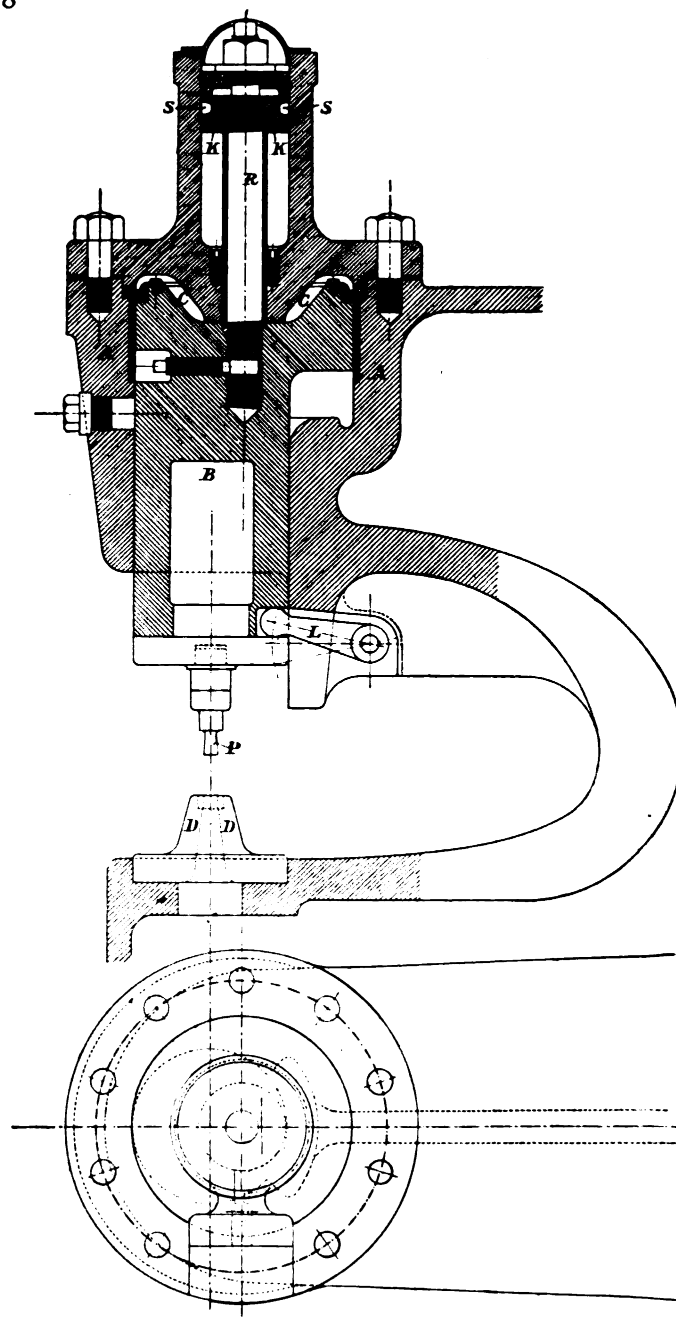


FIG. 269.

work, pressure-water is now admitted to the plate-closing cylinder, the annular tool C pressing the plates closely together. Pressure-water now enters the main cylinder through the lower of the three valves

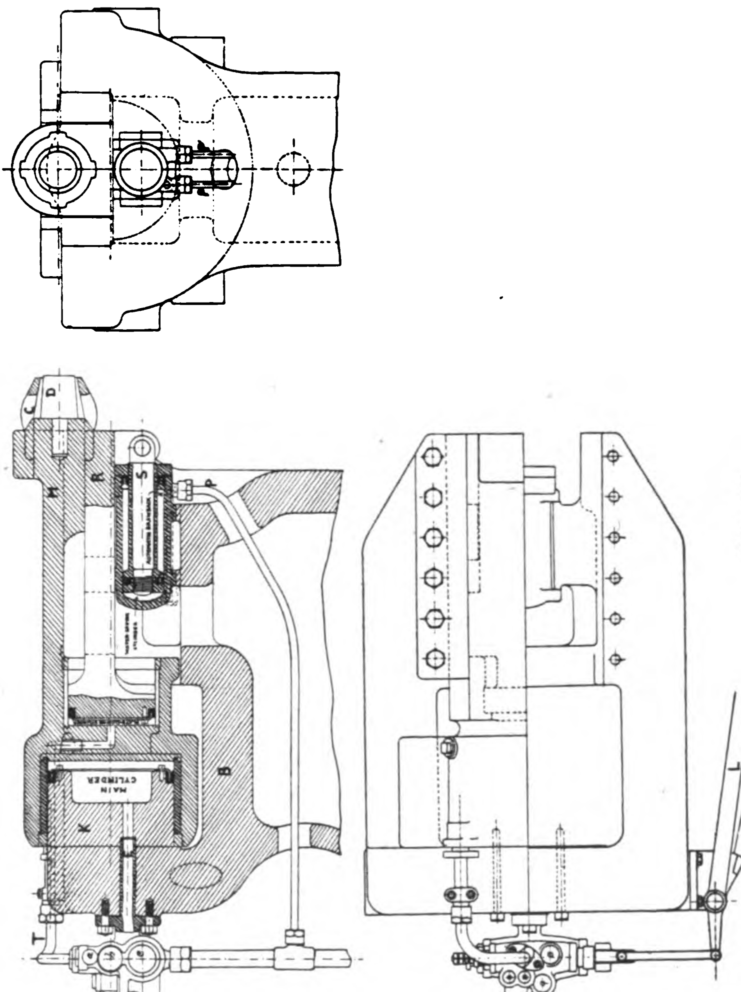


FIG. 270.

a, *b*, *c*, the die D closing the rivet with a pressure due to the difference of the areas of the rams K and R, some water passing from the plate-closing to the main cylinder through the common pressure pipe, to allow the latter cylinder to move relative to R. The pressure is kept on the rivet for a little while until it cools somewhat, when the water from K and R is exhausted and goes back into the tank, the pressure on the back of S returning everything to its normal position, the face of S being open to exhaust.

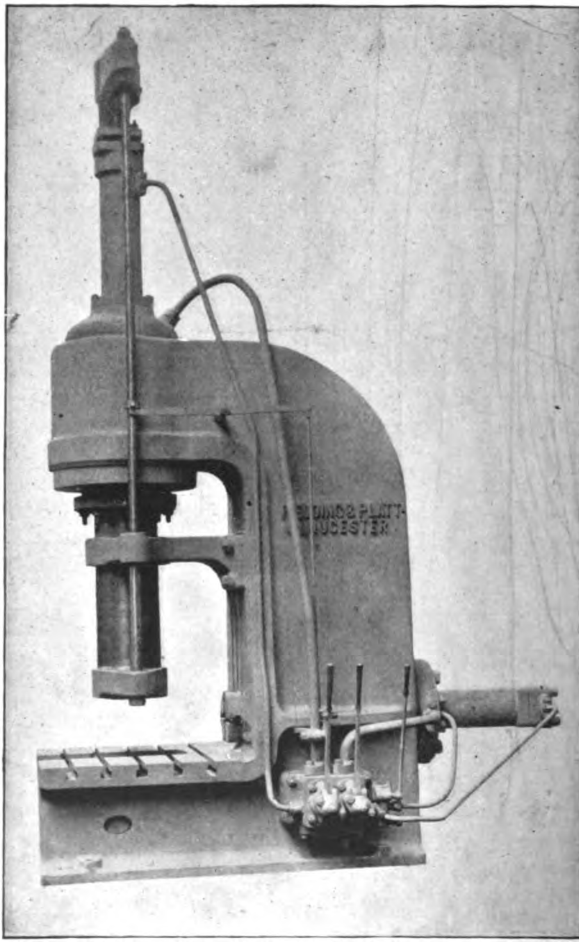


FIG. 271.

To avoid complication, all the details of valves are not shown in the figures, but it will be understood that there are three levers at L, which work the valves.

Hydraulic Forging Press.

The hydraulic forging press was first introduced at the works of Sir Joseph Whitworth & Co. Sir Joseph Whitworth suggested the application of high pressure to steel castings of all shapes, as well as to ingots. This was found impracticable, but the managing director

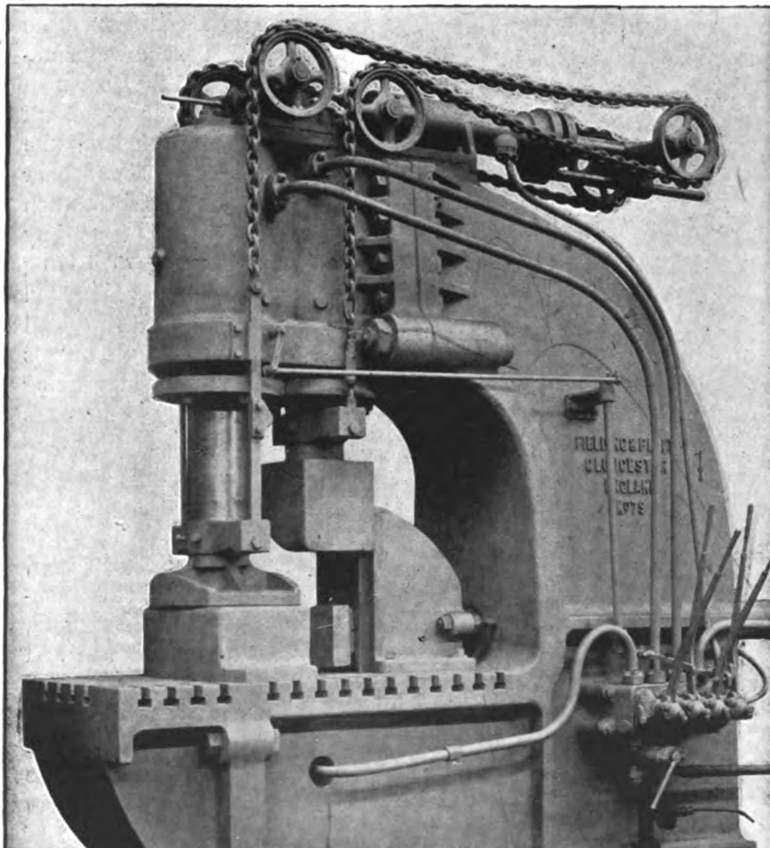


Fig. 272.

of the company, Mr. Gledhill, tried the forging of ingots by a suitable press after compression.

These forging presses are now used in many of the leading railway and other works. Fig. 271 shows one of these machines. They can be made to exert a force of about 10,000 tons at each blow or squeeze.

Hydraulic Flanging Press.

A large flanging press of Tweddell's system is shown in Fig. 272. Large machines formerly in use exerted a force of, say, 650 tons, finishing the operation of flanging in one heat, and giving products which—passing through the same dies—were identical.

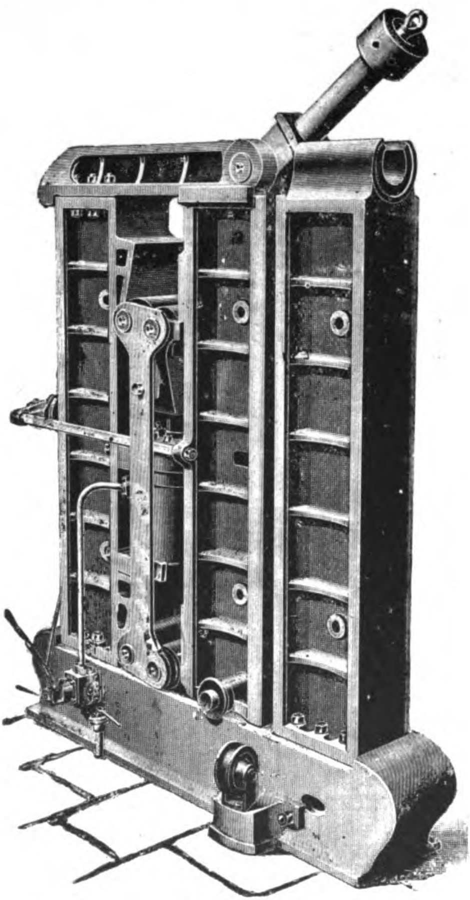


Fig. 273.

For the heaviest marine work this machine is unsuitable, and the cost of a machine to do the work on this principle would be prohibitive. The flanger shown in the illustration does the work "step by step." Instead of having dies and blocks the full size of the work, a small segment of a circle only is used, and the plate, having been properly centred, is turned round, and having been seized and held firmly in place on this segment by one ram, another descends and turns the flange over, the operation being completed by a third ram, which comes forward and squares up the flange. The operation is thus performed in a manner similar to that practised by hand, but the blow or pressure is from 50 to 100 tons, instead of numberless small blows being delivered by hammers. As before, the rams move in the exhaust direction by the action of a separate ram or piston, which is under constant pressure. The illustration shows clearly the position of the rams and the general construction of the machine.

Hydraulic Plate Bender.

One of Tweddell's hydraulic plate-bending presses is shown in Fig. 273. Bending boiler plates in hydraulic presses is not a new operation, Messrs. Eltringham, of South Shields, being probably the first to adopt this method. The method is much superior to rolls, as there is much less risk of fracture of the plate. Besides, the cost of rolls sufficiently powerful to bend plates of $1\frac{3}{8}$-inch thickness is great, and there is difficulty in bending the plate to the true curve required, at the end which last leaves the rolls.

Mr. Tweddell's bender has the advantage over others of a similar type of having the girders which bend the plates placed vertically so that the plate is fed as shown in horizontal section in Fig. 274, and is thus easily handled.

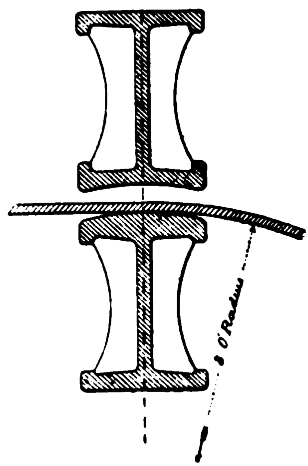


Fig. 274.

The very ingenious parallel motion used in the multiple punching and plate-shearing machines of Tweddell's system is used to work the moving girder.

The machine consists, as will be seen from the illustration, of two vertical fixed girders united by a common bed-plate, and also a top girder. In Fig. 273 a portion of this girder is moved upwards out of its seat: The inner edge of the outer girder is convex (Fig. 274), the opposite edge of the moving girder being concave; hence, when the latter approaches the former, it bends the plate to the required radius. It is not necessary that the dies should come together so that the plate fits accurately the space between them, as it has been found that it is sufficient if the dies bear on three points, but there must be a suitable gear to regulate the moving die to travel the proper distance at each stroke, this distance being constant throughout the operation, so as to ensure uniform curvature. Mr. Tweddell's parallel motion secures this. The hydraulic cylinder is fixed to the back girder, the top of the ram being attached to a cross-head carrying two rollers. By means of two side rods two similar rollers are raised, simultaneously with the top ones. Two straight facing pieces are on the moving girder, whilst on the back standard are two inclined planes. The rollers, which are in contact with one another and with the above bearing surfaces, are pushed up by the ram, force the moving girder forward, and press the plate, the return motion being effected by gravity, assisted by a hydraulic cylinder on the back girder.

These machines will take in and bend cold, steel plates 1½ inch thick and 13 feet wide, to the final curvature at a speed of from 2 to 2½ feet per minute. The length of the plates is, of course, unlimited.

Advantages of Hydraulic Riveting.

In riveting plates together a frictional resistance to sliding is produced, almost equal to welding, owing to the tightness with which the plates are held together by the rivets. In fact, this frictional resistance has a good deal to do, in the first instance, with preventing deformation.

The limit of this resistance is variable; it depends on the nature of the material of the rivets, the temperatures at which the operation of riveting is commenced and finished, and the method of riveting.

For a joint consisting of not less than three rivets the following resistances (experimentally determined) may be relied on, the limit of elasticity of the rivets employed being quoted.

Frictional Resistance to Sliding.

Limit of Elasticity of Metal forming the Rivet per square inch.	Method of Riveting.	Temperature at which the Rivet is Closed.	Resistance per square inch of Sections of Rivet to be Sheared.
tons.			tons.
11·4 (iron) . .	Hand	Bright-red heat	2·5
14·0 ,, . .	,,	,, ,,	3·0
11·4 ,, . .	Hydraulic	White heat	3·2
14·0 ,, . .	,,	,,	3·7
14·0 (steel) . .	Hand	Bright-red heat	2·85
14·6 ,, . .	,,	,, ,,	3·2
14·0 ,, . .	Hydraulic	White heat	3·8
14·6 ,, . .	,,	,,	4·2

The ultimate shear stress of the rivets was found to be not less than three-fourths of their ultimate tensile stress. The above experiments show the superiority of hydraulic riveting.

Cost of Riveting.

The following facts in regard to riveting in shipbuilding are interesting :—

Hand Riveting.		Hydraulic Riveting.	Relative Cost per 100 Rivets Formed. Hydraulic / Hand
Riveting of keels—			
Rivets closed in 9 hours,	100	200 (same size of rivets)	$\frac{1}{2}$
Frame riveting—			
Rivets closed in 9 hours,	300	1000 ,, ,,	$\frac{1}{3}$
Beams ,, ,, ,,	400	1000 ,, ,,	$\frac{3}{8}$

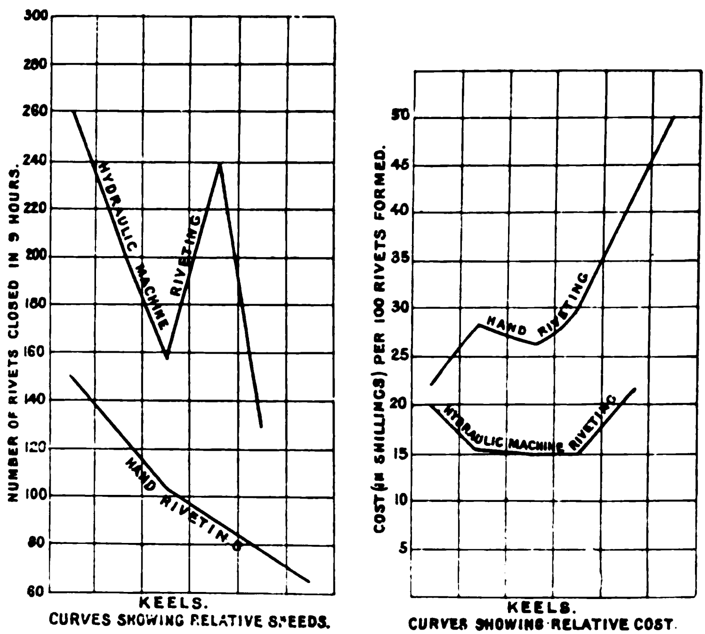


Fig. 275.

The relative advantages of hydraulic and hand riveting as regards cost and speed may be exhibited in the form of curves, as in Figs. 275 and 276.

It will be understood that there is here shown by each pair of corresponding points * on the two curves, say, the cost in two cases

* The actual points plotted are mainly at the apexes.

of forming 100 rivets under similar conditions; points on the other pair of curves showing, in the same way, the number of rivets formed in a given time.

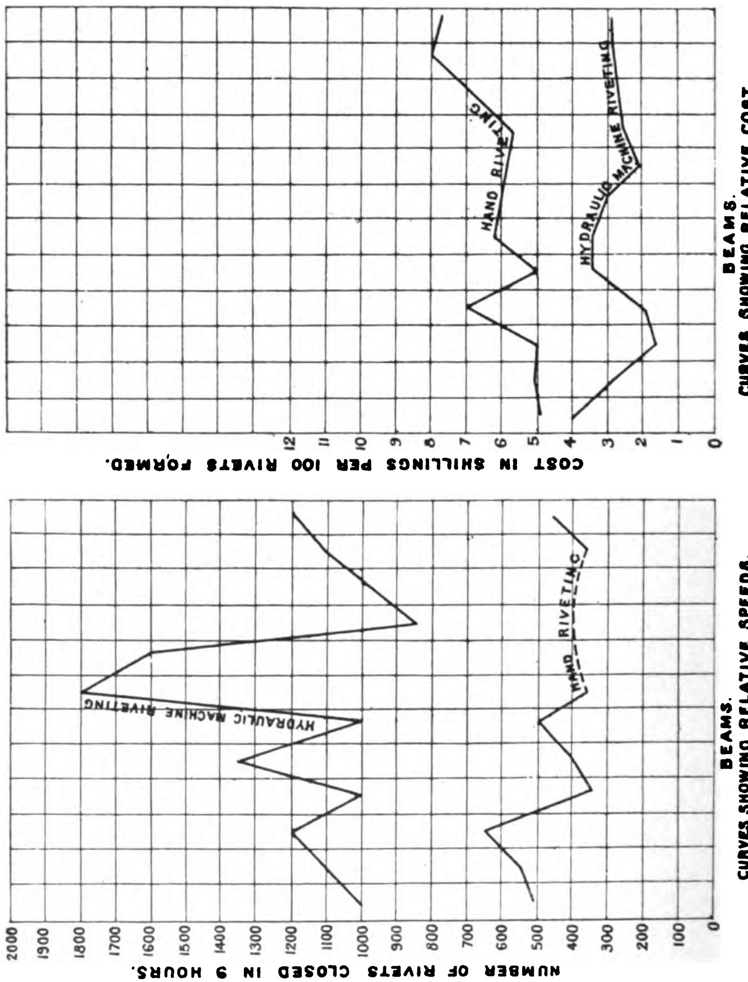


Fig. 276.

Many other curves of a similar kind might be given, but the student is referred to Mr. Tweddell's numerous papers on the subject, and to the record of Professor Kennedy's experiments on the strength of riveted joints, published in the 'Proceedings of the Institution of Mechanical Engineers,' for further information.

XXIX.

PUMPS.

DEVICES for raising water are both numerous and interesting, and, from the historical point of view, should have precedence over other hydraulic appliances. Many of the more ancient of these are described in Ewbank's 'Hydraulics,' and our space does not admit of a lengthened reference to them.

A pump has been defined as an apparatus for lifting water by the motion of a piston in a cylinder. This definition does not, however, include such important appliances as the centrifugal pump, nor pumps of the "pulsometer" class.

The Syringe, a very ancient apparatus, is no doubt the parent of the modern pump.

The Suction, or Atmospheric Pump, is a contrivance for producing a more or less complete vacuum in a pipe, the lower end of which is immersed in water, and thus raising water to the bucket. Such a pump requires two valves, one opening upwards, allowing the water to pass through, but not to return, the other for exhausting the air and lifting the water raised by suction.

In 1641 a Florentine pump-maker constructed an atmospheric or "sucking" pump, and tried to raise water to a height of 50 or 60 feet. The attempt proved abortive, though an examination showed the pump to be perfect. After numerous attempts, the difficulty was submitted to Galileo, who was a native of Florence. The action of such pumps was, in those days, explained on the principle that Nature "abhorred a void," and by some occult means tried to prevent such being formed, or to fill it if formed with whatever was most convenient.

The limit of this occult power was not surmised, and Galileo, then an old man of eighty, did nothing more to solve the difficulty than to state that this law was "limited, and ceased to operate for heights above 32 or 33 feet."

Torricelli, in 1643, announced his great discovery that water is raised in pumps by the pressure of the air, and by exact experiments he determined the intensity of that pressure.

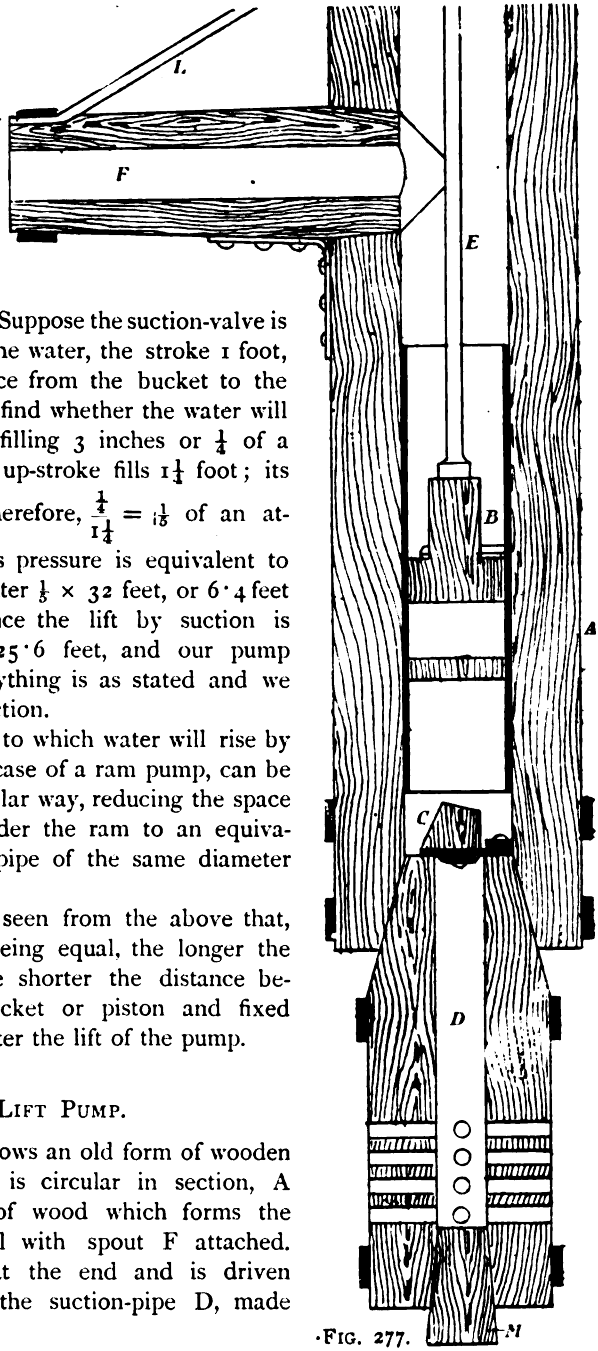
Pascal, shortly afterwards, silenced objectors to the new law by showing that the height of the barometric column was, in accordance with Torricelli's discovery, less at the top of a mountain than at its base. The discovery of Torricelli led to the construction of the air pump, and in 1654 Otto Guerricke, of Magdeburg, made experiments with it before the German Emperor and others.

The exact pressure of the atmosphere being known, and hence the limit of the height to which water can be raised by "suction," the suction pump came into universal use. Since the pressure of the atmosphere is usually about 14·7 lbs. per square inch, if it were possible to produce a perfect vacuum in a pipe—for convenience the beginner may imagine it to be one square foot in section and bent round under the water so as to present a horizontal end to the pressure of the air and a slight layer of water over it—this air-pressure acting on the water overlying the pipe would, neglecting friction, compel it to ascend the pipe to such a height as to give a balancing pressure of 14·7 lbs. per square inch. Since a cubic foot of water weighs 62·4 lbs., evidently $14{\cdot}7 \times 144 = 62{\cdot}4 \times h$, where h is the height of the column of water balancing the air-pressure. From this it is evident that h is 33·8 feet.

The pressure of the air varies from day to day with the height of the barometer, and it is best to take the case of least pressure, hence for calculation purposes it may be taken that a column of water 32 feet in height will balance the pressure of the atmosphere. This marks the extreme limit to which water can be raised by suction, but as it is impossible to produce a perfect vacuum in a common pump, and on account of friction, from 25 to 28 feet is found to be the practical limit. If, however, some air be allowed to mingle with the ascending column of water, this column being lighter than if composed solely of water, may be raised to a greater height than 32 feet, but the stream will be discontinuous and the action of the pump very imperfect.

The height to which water will rise may be calculated, in a given case, somewhat as follows: suppose the fixed clack to be, say, 1 foot from the bucket or moving part, when the latter is at the bottom of its stroke. Then, if the stroke be, say, 2 feet, the air which occupied 1 foot in length of barrel will, after an up-stroke, occupy 3 feet, or will be at $\frac{1}{3}$ its former pressure. Since a perfect vacuum represents 32 feet rise, this, which may be called $\frac{2}{3}$ of a perfect vacuum, will give a lift by suction of $32 \times \frac{2}{3} = 21\frac{1}{3}$ feet. This water will only be raised to the clack, but a very slight increase of stroke will raise it to the bucket. This calculation is, of course, made on the supposition that it is impossible to produce a greater vacuum under the clack or suction-valve than can be produced above it in one stroke, if the clack does not open. The lift will thus be not more than

32 − h, where h is the height of a column of water equivalent to the rarefied air-pressure in the barrel.



·FIG. 277.

Example.—Suppose the suction-valve is 25 feet above the water, the stroke 1 foot, and the distance from the bucket to the clack 3 inches, find whether the water will rise. The air filling 3 inches or $\frac{1}{4}$ of a foot, after the up-stroke fills $1\frac{1}{4}$ foot; its pressure is, therefore, $\frac{\frac{1}{4}}{1\frac{1}{4}} = \frac{1}{5}$ of an atmosphere; this pressure is equivalent to a column of water $\frac{1}{5} \times 32$ feet, or 6·4 feet in height, hence the lift by suction is 32 − 6·4 or 25·6 feet, and our pump will do if everything is as stated and we can neglect friction.

The height to which water will rise by suction in the case of a ram pump, can be found in a similar way, reducing the space around and under the ram to an equivalent length of pipe of the same diameter as the ram.

It will be seen from the above that, other things being equal, the longer the stroke and the shorter the distance between the bucket or piston and fixed valve, the greater the lift of the pump.

Lift Pump.

Fig. 277 shows an old form of wooden lift-pump. It is circular in section, A being a log of wood which forms the working barrel with spout F attached. A is coned at the end and is driven tightly on to the suction-pipe D, made with a corresponding taper and plugged at M, the water entering by the side holes shown. A is the working barrel bored out smoothly, or sometimes fitted with a metal liner as shown, containing the bucket B, moved by the pump-rod E. C is the suction-valve or clack fastened to the top of D, consisting of a flap of leather weighted with heavy wood or lead. The bucket valve on its upper side has a similar flap. Once the water passes the bucket it is simply raised like an ordinary load.

Let l be the length of the stroke and d the diameter of the barrel in feet, then the quantity of water lifted each stroke is (neglecting slip) $0 \cdot 7854\, d^2 l$ cubic feet. If the pump makes n strokes per minute, and the total lift be h feet, the horse-power represented in water raised is

$$\frac{0 \cdot 7854 \times 62 \cdot 4 \times d^2 l n h}{33{,}000} = 0 \cdot 00148\, d^2 l n h.$$

This rule also applies to a single-acting, and multiplied by 2, to a double-acting force-pump, d being the diameter of the plunger or piston. The ratio of this power to that given to the pump is its efficiency, which usually increases with the lift h in pumps of this class, as many of the resistances are nearly constant. A centrifugal pump, on the other hand, *decreases* in efficiency beyond a certain value of h.

The quantity discharged per hour (in gallons) is, on the same assumption $= 294\, d^2 l n$ for a single-acting pump.

In calculations of suction height, like those given above, but of a more practical nature, it will also be necessary to make allowance for the resistance of bends, orifices, etc., and the slip of water past the valve will take probably at least 6 inches off the height of the column of water capable of being raised. The actual horse-power given out will be less than that given by the rule, on account of these and other disturbing influences.

Plunger Force-Pump.

When the height to which water has to be raised is considerable, it may be forced part of the way, or when the water has to be forced into a vessel against pressure, a piston or plunger force-pump is usually employed. This must be of a much stronger construction than the lift-pump.

Fig. 278 shows a small hand force-pump. Instead of a bucket a plunger is here adopted; this plunger passing water and air-tight through a stuffing-box. When the plunger is raised, water is drawn into the pump barrel by suction, and on the down-stroke the valve A closes and C opens, the water being forced up the pipe to the spout, which may be at any required height above the plunger. The hydrostatic pressure the pump must withstand may be calculated as the pressure due to a column of water of the height of the spout, and the amount of water entering the delivery pipe each stroke as that of a column of water of the cross-section of the *plunger*, and same length as the stroke of the latter.

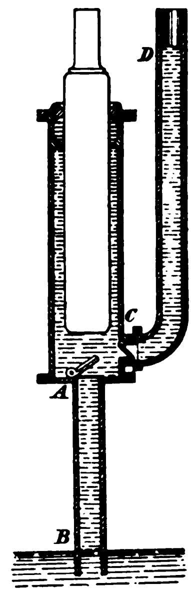


FIG. 278.

Care must be taken that the pump-plunger does not move faster than the water can enter behind it, else the plunger on returning will meet the water, and a shock will result. The pressure produced by such shock may be great, and is a matter of extreme importance in all calculations of the strength of pump-barrels, pipes, etc. (See p. 414.)

A form of plunger-pump much in use is shown at Fig. 279, where R is the horizontal plunger or piston, moving water-tight through the gland G, but not necessarily filling the barrel B. A is the suction-valve up through which the water ascends on the backward stroke of R; on the forward stroke water is forced through the delivery-valves C and D and the delivery-pipe P. Two valves are provided to ensure the non-return of the water in the event of one valve sticking, also to enable an examination or repair, of that part of the pump outside C, to be effected. The valves are ball-valves, which are now much used. This form of pump is convenient, and is much used for forcing the feed-water into steam boilers, etc. The pump is single-acting.

THREE-THROW PUMPS.

A combination of three pumps, generally single-acting, all driven from one crank-shaft by cranks set at angles of 120° to each other, is often employed; offering a more uniform resistance than single or double pumps, and giving a nearly constant discharge.

Fig. 280 shows a neat little combination of a three-throw Gould pump driven by a Robinson hot-air engine. Each pump is single-acting, its piston being of the "trunk" type; i.e. it is a cylinder closed at one end, to which is attached the piston-rod, which also forms the connecting-rod, and which oscillates inside the piston as the crank revolves. The suction-valves are at one side of the base,

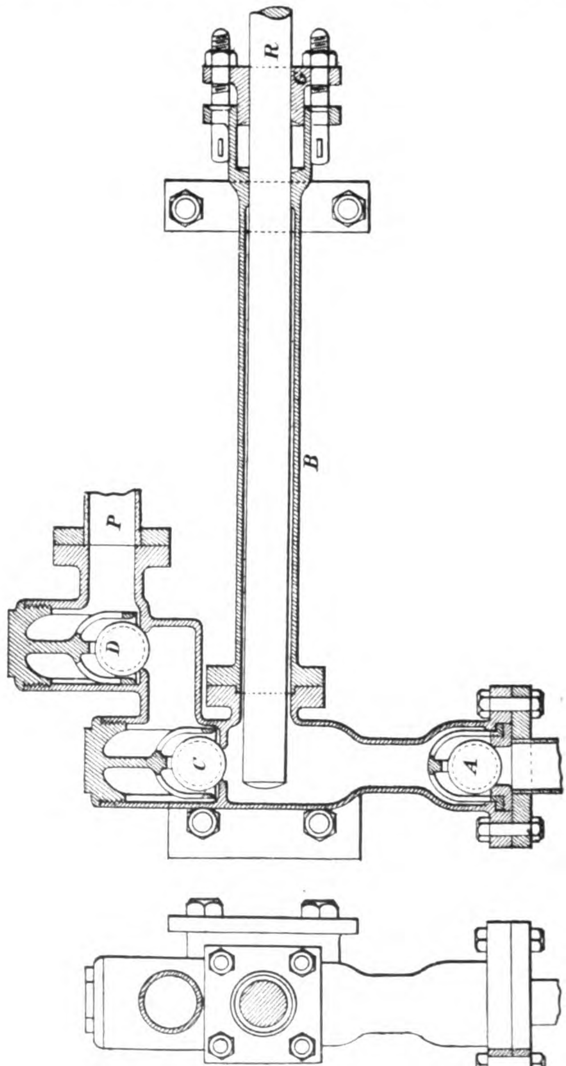


FIG. 279.

and the delivery-valves at the other side. Instead of an engine an electric motor is often employed to drive such pumps, where current is readily obtainable.

Double-acting Pump.

Fig. 281 shows a form of double-acting pump, in which A is the working barrel, with one suction and one delivery pipe. When B moves forward F opens, admitting water to A, G also opening to allow the water above B to pass to the delivery branch K. On the

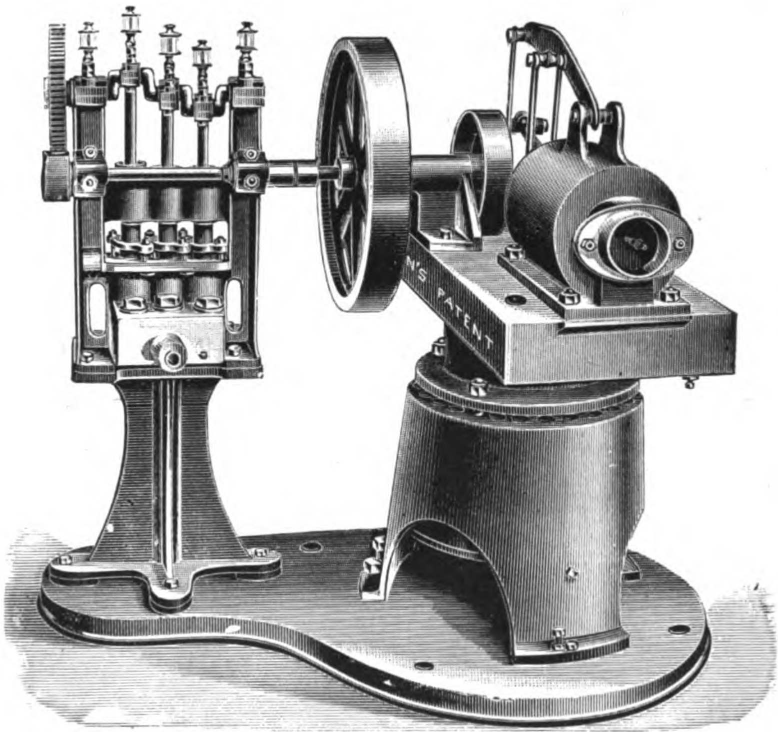


Fig. 280.

down-stroke F and G close and E and H open, part of the water in A being forced through H, and water entering by E behind B as required. There are many forms of double-acting pump, but this will explain the action.

The Davidson Steam Pump.

There are, of course, many forms of steam pump. A good, and somewhat peculiar form of single-cylinder steam pump, the Davidson steam pump, is shown in Fig. 282.

It has one steam and one water cylinder, and is double-acting. The peculiarity of the pump is its valve-motion; the valve—a cylindrical valve—is moved both by steam pressure and by mechanical means.

The valve, as will be seen from the figure, is attached to two pistons which assist in moving it. In the exhaust passage is a cam

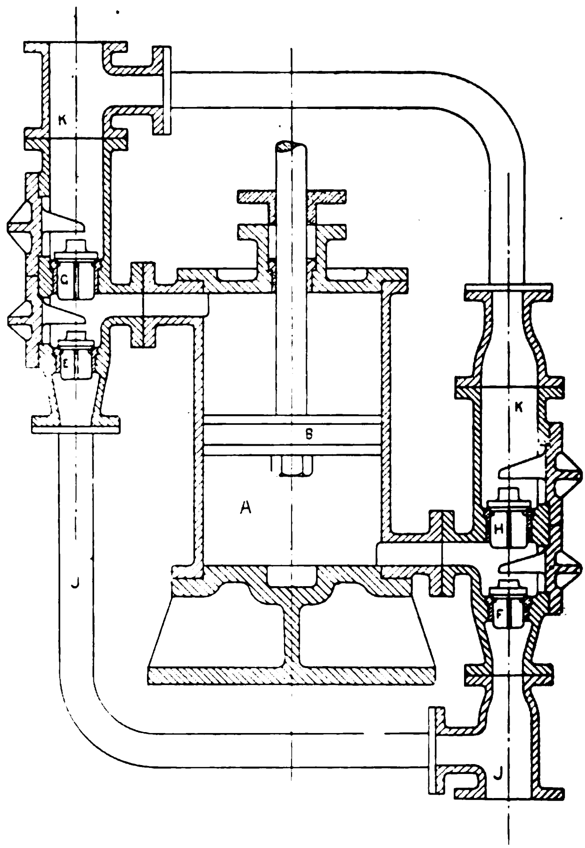


FIG. 281.

which is rocked by an arm connected with the piston-rod. This cam engages a steel pin attached to the valve, and it not only moves the valve axially, but also rocks it on its axis. By moving it round on its axis, passages are opened and closed which admit steam to or exhaust it from one end or other of the valve-chest. If we imagine one main steam port completely open, the first motion of the cam will be to turn the valve on its axis, thus, in due time, bringing it into position to admit steam to the auxiliary piston, and so moving the valve. The valve will then be closed *mechanically* slightly before the end of the main piston stroke, so as to cushion the steam in the main cylinder and bring the piston to rest. The next motion of the cam will be to open the other auxiliary port to steam, and the last to exhaust, and so on. The pump has thus no dead centre, the auxiliary ports being opened whenever the main ports are closed.

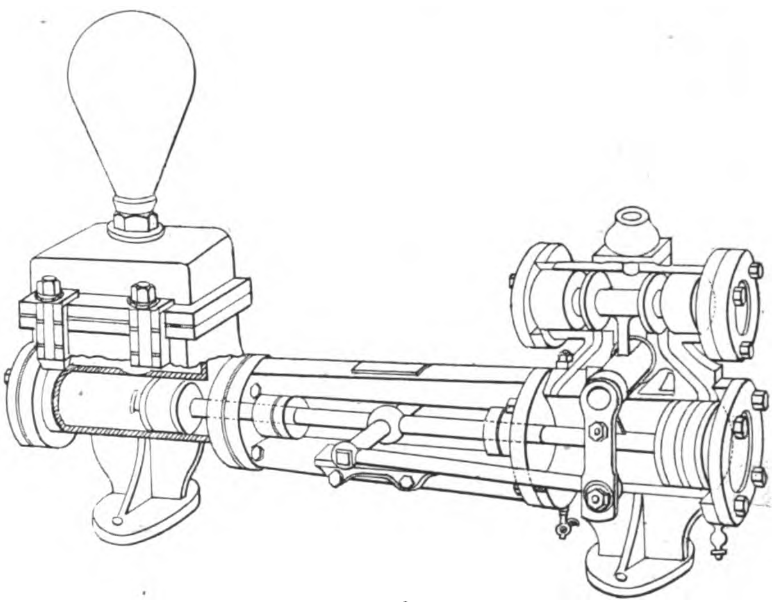


FIG. 282.

The mechanical movement of the valve ensures uniform length of stroke, and also ensures that, at a high rate of speed, the valve shall be carried by a mechanical connection.

DUPLEX PUMPS.

A duplex pump consists of *two sets*, each consisting of a steam piston and a pump-plunger, usually connected directly together, and each working in its own cylinder. The great feature of a duplex pump is that the movement of the steam-valve of one steam cylinder is effected by the piston of the other steam cylinder. This arrangement allows the water-pistons to stop or pause momentarily at the end of each stroke; the water-valves thus find their seats without shock, and the pump wears well. The pistons whilst in motion move with a nearly uniform velocity, not like fly-wheel pumps, in which the pistons being driven from a *crank* going round uniformly, having a *varying* velocity. Water is delivered by duplex pumps in a constant stream, and without that concussion noticeable in other kinds of pumps. The duplex pump is the invention of Mr. Henry R. Worthington.

The Worthington Steam Pump.

Fig. 283 gives a good example of a modern duplex pump, showing one side or set of steam and pump-pistons and valves, of the simple form of the pump.

The steam-piston A and the pump-plunger B are on the same rod which gives motion to a swinging arm F, at the other end of the spindle

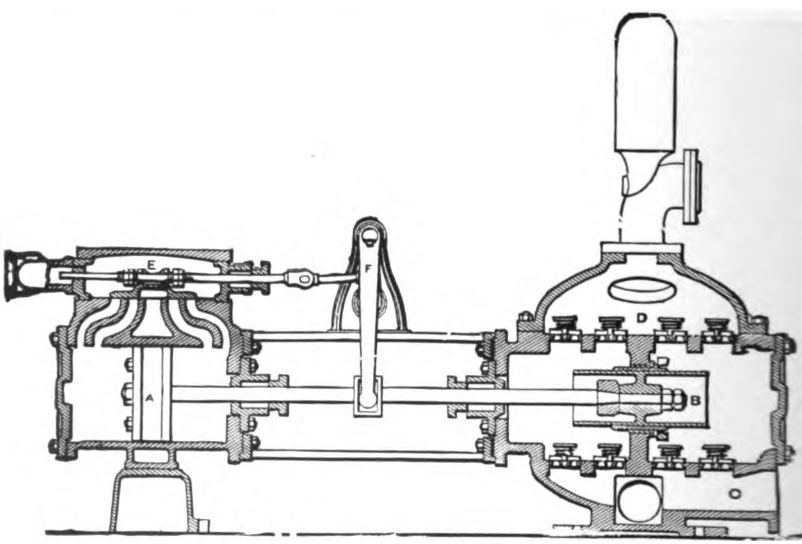


Fig. 283.

of which is a similar but shorter arm connected to the valve of the other steam cylinder. E, the valve of the cylinder shown, is moved by the piston-rod of the further set. Thus, when A moves forward to the end of the stroke, A′ (its neighbour steam-piston) should be at the back end of its stroke, which will move E to its back position, opening the forward port in order to force A back again. The steam-valve has neither lap nor lead, but as soon as the piston covers the first port, which is the exhaust, the steam in the cylinder is cushioned in front of H to prevent it striking the cylinder-cover. Each piston when it reaches the end of its stroke waits for its valve to be moved over by the other steam-piston before making the return stroke, there being, as the cut shows, a certain amount of freedom or slack between the nuts on the valve-spindle, which allows a spindle to move some distance before acting on its valve. A similar motion is obtained where Corliss valves are used, by having the lever working the valves attached to a pin working in a slot. In this way the pump-valves get time to seat themselves quietly, and wear well, there being smooth and natural motion. As one or other steam-valve must be always open, there is no dead point, and the pump is always ready to start. India-rubber valves are shown on the pump end, but it must be remembered that this material is unsuitable for such purposes as forcing water into steam boilers, for working against heavy pressures, or for pumping hot water.

In the latter case the water, if over 150° F. in temperature, should, if possible, be allowed to flow into the pump-barrel, as it is difficult to lift hot water by suction, a sufficient decrease of pressure causing boiling of the water. The difficulty increases with the temperature of the water.

Rankine's formulæ connecting the pressure and temperature of the boiling-point of water, are as follows:—

$$\log P = A - \frac{B}{T} - \frac{C}{T^2},$$

where P is the pressure per square foot, and T the absolute Fahrenheit temperature (= ordinary Fahrenheit temperature + 461).

This formula may be used when T is known and P is required. If P is known, T can be found from the inverse formula, which is

$$T = 1 \div \left\{ \sqrt{\left(\frac{A - \log P}{C} + \frac{B^2}{4C^2} \right)} - \frac{B}{2C} \right\},$$

where

$$A = 8{\cdot}2591,$$

$$\log B = 3{\cdot}43642,$$

$$\log C = 5{\cdot}59873; \quad \therefore \quad C = 396,945.$$

$$\frac{B}{2C} = 0{\cdot}003441, \text{ and } \frac{B^2}{4C^2} = 0{\cdot}00001184.$$

The pressure produced above the hot water may be calculated as before, knowing the dimensions of the pump, when the temperature of the boiling-point of the water can be determined from the second formula. This temperature must be considerably greater than the actual temperature of the water if the pump is to work properly.

Example.—An ordinary suction-pump is applied to lift water at a temperature of 120° F. The distance between the bucket and the suction-valve at the bottom of the stroke is 3 inches; the stroke being 6 inches. Imagining the suction-pipe to be very short, and that the pump is filled with air to begin with, find if the water will be lifted or if it will boil; also the temperature at which it will boil.

Ans.—It will be lifted. It boils at 161° F.

Worthington Compound High-Duty Pumping Engine.

This engine, shown in section in Fig. 284, has been brought to great perfection. It consists of two pairs of tandem cylinders, one pair being shown in the cut, with similar ones at the back of these. Each engine works its fellow's main steam-valves. These steam-valves are of the cylindric semi-rotative or Corliss type, placed at either end of each cylinder, the clearance being small. They work in chests cast on the cylinders, and are actuated by an arm from the piston-rod of the *other* engine, after the duplex method already explained, but the cut-off valves are actuated by an arm attached to the piston-rod of *their own* engine, these valves being partially balanced. As the steam passes from the high-pressure cylinder A to the low-pressure cylinder B it passes through a re-heater R kept hot by steam at boiler-pressure. The condenser is usually underneath, as also the air-pump (shown in the cut). The steam thus passes from the boiler to A, from A through the re-heater to B, and from B to the condenser, from which the water and vapour are withdrawn by the air-pump K.

The pump end of the engine consists of two water-cylinders C placed side by side—one is shown in the figure—with suction-chambers below. They are connected by a cross-suction pipe with a single opening to the suction-main. They have also a cross delivery pipe from the delivery chambers, which are above the water cylinders with a single opening to the delivery main, the cross-delivery pipe being surmounted by an air-chamber, kept at proper pressure by a small independent air-compressor.

Each water-cylinder has a diaphragm cast in the centre of it, with rings lined with composition metal, through which the double-acting pump-plungers—each of which is attached directly to the steam

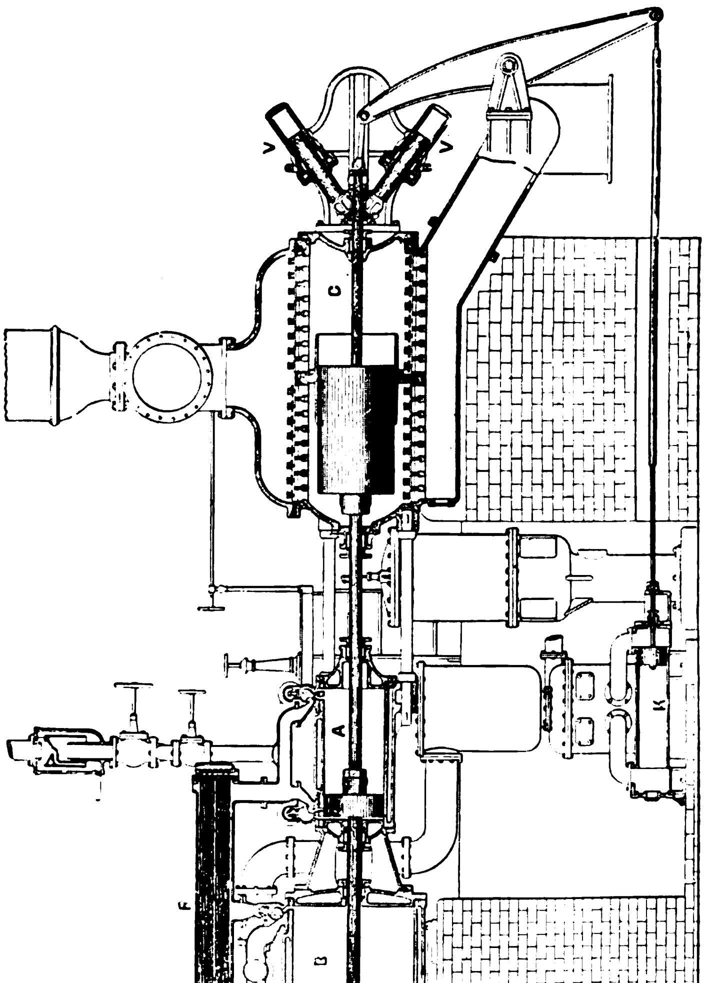


Fig. 284.

piston-rod of its engine—work. The valves of the water-cylinders, usually india-rubber discs, are held down by spiral springs.

HIGH-DUTY COMPENSATING ATTACHMENT.

This attachment constitutes a feature of the modern pump. It is necessary to high economy of steam that it be used expansively, but this gives high pressure at the beginning and low pressure during the later part of the steam-piston stroke. To compensate for this the cylinders V V are provided, which are connected by a pipe passing through the hollow trunnions on which they oscillate. These cylinders contain water at a certain pressure (say 200 lbs. per square

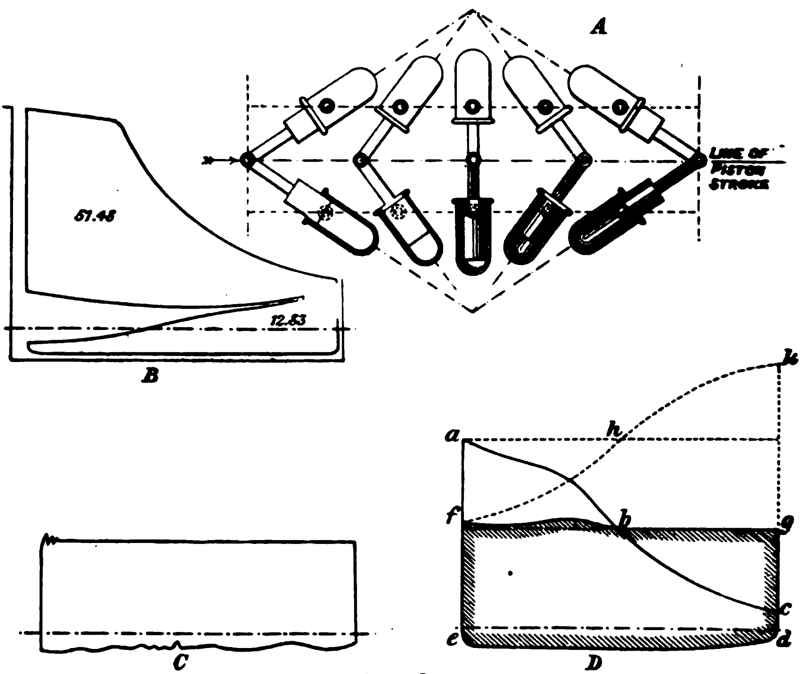


FIG. 285.

inch), and the plungers which work in these cylinders are attached to the main piston-rod or pump-rod, and work in a manner explained by the separate cut. (Fig. 285.)

The pressure in these cylinders acts against the steam pressure in the first part of the stroke, and *with* it in the latter part of it. It will be understood that the cylinders communicate with each other and with a pipe in which water is kept at a fairly constant pressure by a small intensifier worked from the air-vessel of the pump. The diagrams in Fig. 285 show very clearly the function effected by this arrangement.

Part B shows indicator diagrams (reduced to low-pressure piston area) taken from the high and low-pressure cylinders of one engine. C is from the pump of the same engine, whilst in D the curve *f*, *h*, *k* shows the resultant pressure due to the compensating pistons as they assume the angles shown in A, and combining this with the effective pressure line *a*, *b*, *c* of the steam cylinders, we get the line *f*, *b*, *g*, showing the resultant pressure transmitted to the pump-plunger, which agrees very well with C.

The attachment also acts as a safety device or governor, for if a main breaks, the pressure on the compensating pistons being reduced or taken off, the stroke of the steam-pistons is shortened, or in some cases they are brought quietly to rest, being unable to complete the stroke without the help of the auxiliary piston.

Duty. Efficiency.

The reader will notice that these are called "high-duty" engines. The "duty" of an engine—a term not so much used as formerly, but still applied to pumping engines—means the number of ft.-lbs. of work given out per hundredweight of coal (usually best Welsh) burnt in the boiler furnaces. In the United States the term is usually employed to denote the number of ft.-lbs. given out per 1000 lbs. of dry steam supplied to the engine, the pressure of the steam being specified. Professor Unwin, in 1888, made some tests of one of these engines at Hampton, and obtained a duty of 110,000,000; whilst Professor Kennedy's tests, in 1894, at Hornsey Sluice, of a triple-expansion engine by a London maker, differing slightly in detail from that shown in the section, gave the very high duty of 139,500,000; the coal burnt per pump horse-power-hour being in that case only 1·59 lb.

The efficiency of the boiler, or fraction of total heat of combustion of the coal taken up by water in boiler, was 80·4 per cent., whilst the mechanical efficiency of the engine, i. e. the ratio of the pump horse-power to the indicated horse-power, was 84·4 per cent.

The engines of the Boston (U.S.) waterworks at recent trials gave a duty of 178,497,000 ft.-lbs. per 1000 lbs. of dry steam at a pressure of 185 lbs. per square inch. A duty of 160,000,000 ft.-lbs. per 1000 lbs. of steam at 150 lbs. per square inch has been specified in some cases; but this is rather high, 140,000,000 ft.-lbs. being more usual.

Some Useful Memoranda.

It is impossible, in our brief space, to go into all the details of pump design, but a few useful data may be given. Some different forms of valves have been shown, and books may be consulted as to other forms.

If the valve area be too small for the plunger area, the pump will be noisy. The area of clear water-way through a set of valves should not be less than about 40 per cent. of the plunger area if the plunger speed is 100 feet per minute. At a speed of 125 feet per minute the valve area should be increased to 50 per cent., and so on, till at 200 feet per minute the valve area will equal the plunger area. This area may be obtained by making few valves of large area, but it must be borne in mind that whilst the valve area (usually a circle) increases as the square of the diameter, the circumference—which determines for a given lift the discharge area under the valve—increases only as the diameter. Hence, if we make few valves, we must increase the lift of each so as to give sufficient discharge area, and this consideration limits the diminution in the number of valves. Mechanically operated valves are now coming into use for high speeds.

The necessity for an air-chamber is apparent on single-acting pumps, but even on duplex pumps it is always found. Its form is not of much importance, but its size is of great importance. For high-speed pumps like fire-engine pumps it should be five or six times the capacity of the water displaced by a stroke of the pump plunger, whilst for double-acting duplex pumps half of this capacity will do. It should be on the highest part of the pump, over the delivery opening. There is usually on large pumps a separate air-pump to keep the air in the chamber at the proper pressure.

Apparatus for Injecting Air into Air-Vessels of Pumps.

Fig. 286 shows Wipperman and Lewis's apparatus for this purpose. A is a vessel partially filled with water, having a regulating cock C at its lower end. C is connected to the pump valve-box between the suction and delivery valves.

At the top of A is fixed a small gun-metal valve-box D, with inlet and outlet air-valves, D being connected by pipe E with the pump air-vessel G. When the main pump draws water it partially empties A of water, drawing in air, as shown by gauge F, regulation of the amount being effected by C. On the return stroke of the main pump plunger the air previously drawn into A is forced into G. Thus there is no piston in the pump supplying G, little friction, few parts, and the cost of the apparatus is small.

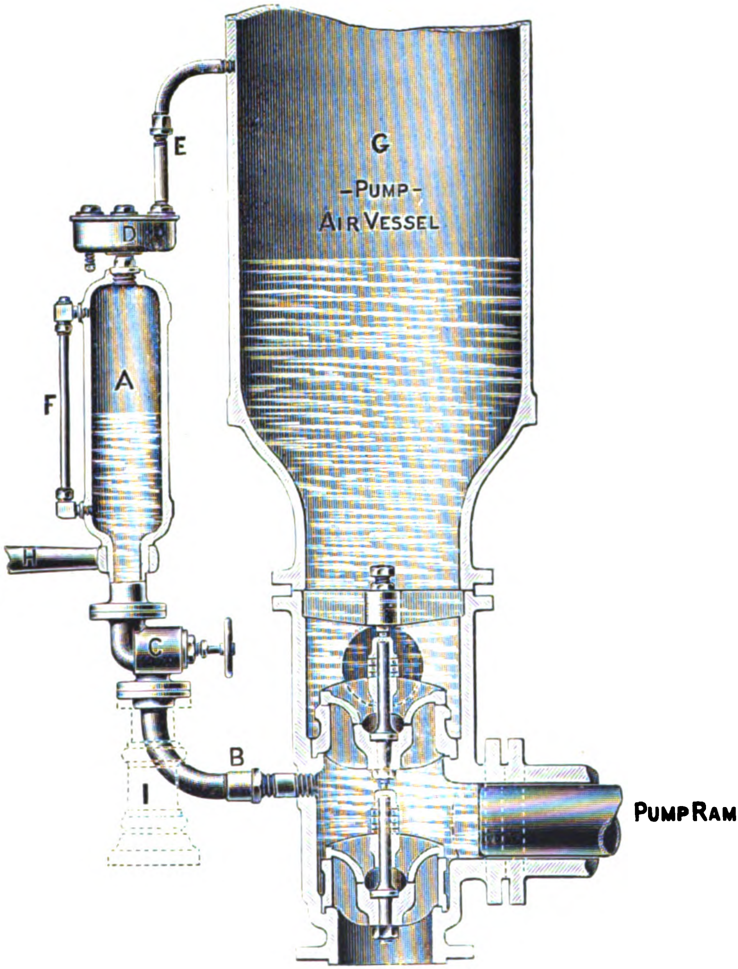


FIG. 286.

Discussions have taken place as to the utility of an air-chamber on the *suction* side of a pump. The flow of water into a pump is often continuous, whilst the discharge is non-continuous; hence an air-chamber *is* of use to obviate the "water-hammer action" and other disturbing effects due to an attempt to reconcile the two kinds of flow. Its size may be about half the capacity of the discharge air-chamber.

When suction-pipes are long and crooked this "vacuum" chamber is a necessity, and should be placed as near the pump as possible. The velocity of flow in a suction-pipe should not exceed 200 feet per minute—on good pumps it is much less—and the pipe should be everywhere of the same diameter, with few bends.

Professor Goodman's Experiments.

Professor Goodman made recently important experiments on the discharge from, and the fluctuations of pressure in, the suction and discharge pipes of a plunger pump.* He found, amongst other things, that there is a critical speed above which if the pump be run the water does not follow the plunger on entering the pump barrel, and when later on the speed of the plunger is reduced, the water overtakes the plunger, causing a more or less violent blow or impact. When the speed is near this critical point the discharge coefficient for the pump delivery is very unstable, and varies as widely as 20 or 30 per cent. for very slight disturbances in the conditions—usually increasing rather suddenly at the critical speed. At speeds above this critical speed the coefficient remains fairly steady, increasing with the outlet pressure. The sudden increase of discharge coefficient is much greater at low than at high delivery pressures; it is also greater with a long than with a short suction pipe, and practically disappears at delivery pressures of 100 lbs. per square inch. The discussion of the probable reasons of this strange variation would take up too much space here, but some important suggestions are made in the article referred to.

The fitting of a vacuum vessel on the suction pipe obviated almost entirely the violent banging before experienced, and allowed smooth working for much higher speeds than were before admissible; also the discharge coefficient was now nearly constant.

Another curious thing noticed was that the pump actually delivered, under some circumstances, considerably—even in a few cases more than 50 per cent.—more water than the plunger displacement, but of course the delivery in horse-power never becomes equal to that indicated in the pump barrel.

This phenomenon is due to the inertia pressure in the suction pipe forcing open the delivery valve before the completion of the suction stroke and also increasing the pressure and delivery during the delivery

* 'Engineering,' February and March, 1903.

stroke. The column of water in the suction pipe has to be suddenly stopped and reversed in direction at the beginning of the delivery stroke, and if the pressure this reversal induces is greater than the resistance of the valve, the water passes up the delivery pipe. This "water-ram" pressure is of great importance, and Professor Goodman's experiments only confirm the views the author formed as the result of some experiments carried out before the first edition of the present work appeared. What the pressure due to the absolutely sudden stoppage of a column of moving water will be has been investigated; the rule is given at page 250, and Professor Goodman's curves show actual pressures in his experiments as compared with this absolutely sudden stoppage pressure. What the actual "water-ram" pressure in any given case will be, taking time of stoppage, friction, and also stretching of pipe into account, has never, so far as the author is aware, been completely and correctly worked out. It is a problem of considerable complexity having to do largely with wave motions, but all the supposed solutions which he has seen are wrong. Probably the rise of pressure will not be serious if the time taken to close the valve be greater than twice the length of the pipe divided by the velocity of sound in water. If less than this the compressibility of the water and extensibility of the pipe come into play; but unless some method of closing the valve *according to a special law* be adopted, the retardation will be non-uniform, and the rise of pressure in the earlier part of the operation will be due mainly to a disturbance of the condition of steady flow in the pipe. The rapid increase of pressure just before the valve closes, rising to a maximum shortly after it is closed, is usually due mainly to the compression wave, and the action is different from that during the earlier period, being of the nature of impact.

To consider the whole operation as following one law can only give a very rough approximation, and it is surprising that so few writers deal in a logical way with so important a subject.

One leading writer refers to the time element in the question by stating that the increase of pressure due to the stoppage in t seconds of the water in a pipe l feet long, d feet in diameter, moving at a velocity of v feet per second, is $\frac{\pi}{4}\frac{w\,l\,d^2\,v}{g\,t}$ lbs.

Now we know that the *average* force during the interval of time t required to destroy a certain momentum is—if the *acceleration be uniform*—momentum $\div$ time, but Professor Goodman's curves, and those obtained by the author in 1896, show that the pressure rises slowly at first, but with great rapidity near the end of the little interval of time taken to close the valve, indicating a non-uniform retardation. The *average* increase of pressure is of little value to us; it is the *greatest* pressure that we must in most cases consider, and this is not twice the average, nor is it easily obtained from the average increase of pressure, even if we knew that. It may, however, be obtained by finding $\frac{dv}{dt}$ at each increment of time if we know v, the velocity of the water at different times and stages of the closing operation of valve orifice, since $\frac{dv}{dt}$ = acceleration, and acceleration × mass = force. The stretching of the pipe introduces a modifying factor, which again depends on the suddenness of the rise of pressure towards the end of the operation. The problem is one awaiting the experimenter who has time and a sufficient command of mathematics.

Professor Goodman gives some useful formulæ. The delivery in a pump of the type used (without a vacuum vessel on the suction-pipe) depends on the speed of the pump and on a "coefficient of discharge," which may be taken as about 0·94 with normal working.

This coefficient must only be used when the delivery pressure is greater than $0{\cdot}000148\, L\, R\, N^2 \frac{A_p}{A_s}$, and the speed N (revolutions per minute) less than

$$54{\cdot}5 \sqrt{\frac{(34 - h_s - h_f)\, A_s}{L\, R\left(1 - \frac{1}{n}\right) A_p}}$$

where R is the radius of the crank, L the length of the connecting rod (both in feet), $n = \frac{L}{R}$, A_s = area of suction pipe, A_p = area of plunger; h_s = suction head below pump, h_f = loss of head due to friction in pipes and passages; 34 being taken as the barometric height (all in feet of water).'

THE "PULSOMETER" PUMP.

Those who are familiar with the history of the steam engine will remember that, towards the end of the seventeenth century, Captain Savery raised water by condensing steam in a closed vessel connected by a pipe with the well. This vessel being not more than 32 feet above the surface of the water, the vessel was filled with water on the formation of a more or less complete vacuum by the condensation of the steam. The water was then raised to a much greater height by the pressure of steam from a "strong boiler" acting on the surface of the water. This is the principle employed in pumps of the "pulsometer" class. There is necessarily a good deal of condensation during the pressure portion of the operation. Hence a high efficiency is impossible with such a pump; but the simplicity of the

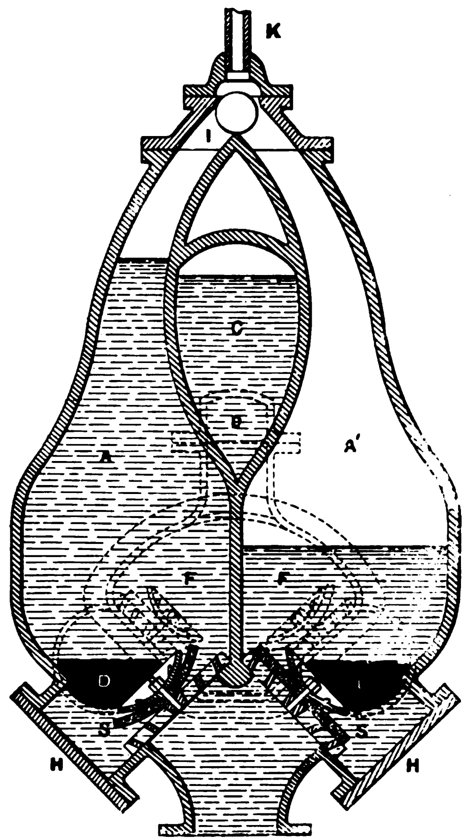


FIG. 287.

pump, the absence of a working piston or parts readily choked by sand and dirt, and its portable nature, render it a useful apparatus in many cases, and readily applicable in emergencies.

A section of the best known form of this pump is shown in Fig. 287. It consists of a casing with two chambers A A′ side by side, meeting in a neck containing a ball-valve I which admits the steam alternately to either chamber. There is also a central air-chamber C communicating with the discharge valves F F and the discharge openings D D. H H are planed covers, S S the suction valves. Suppose the three vessels to be sufficiently filled with water through an opening in C, this water being prevented from escaping by a foot-valve; steam enters through I, displacing the water through the right-hand opening D by pressure. As soon as the water is lowered below the upper surface of the opening D, steam blows through with some violence, causing rapid condensation in A′. The ball is now drawn to the right-hand side, water rises in the right-hand chamber A′, the steam entering the left-hand chamber, forcing the water there through D, the action being repeated as above. Though a good deal is made of the shape of the passage D being such as to secure rapid condensation by exposing the maximum cross-section for the smallest fall of water surface, as a matter of fact it is probable that in steady working the surface of the water never gets as low as D in either chamber.

The vessel C assists in promoting the steady flow of water through the discharge valves F F, by providing a small head of water under which the discharge takes place. Air-cocks, kept slightly open, are provided on the vessels to prevent shock.

Often there is placed over I a valve called the "grel," which forms a very important feature of the newer forms of the pump.

The "Grel" Valve.

The use of this valve has added greatly to the economy of the pump. It admits of the expansion of the steam, which is no longer allowed to follow the water during the whole stroke, but is cut off about half stroke.

Fig. 288 shows this arrangement. There is a modified ball-valve A instead of the ordinary upper valve, and over this is the cut-off valve B, A and B corresponding to the main and cut-off valves in a Meyer's expansion gear for the steam engine.

The valve B is so constructed that its lower portion forms a piston working in the cylinder C, differences in pressure within and without this cylinder actuating the piston and valve. The cylinder C is connected with the steam and pump chambers by the holes D D. The action of the apparatus is somewhat as follows. When steam is turned on B is opened and the steam flows past A into one of the pump-chambers, partly driving out the water there. When about half the water is driven out the pressure in C becomes sufficient to lift the valve B, closing the steam opening, and keeping it closed till A has moved over to the other side, when the pressure in C falls, owing to its connection with the second pump-chamber, and B is opened, the cycle proceeding as before. It will then be seen that after about half-stroke the remainder of the work of that stroke is performed by the expanding steam shut in the chamber by the closing of the cut-off valve B.

This arrangement adds greatly to the economy of the pump (from 25 to 50 per cent. it is said) and it also renders it impossible for live

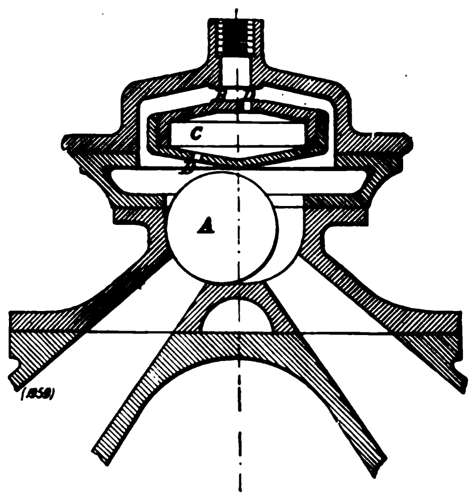


FIG. 288.

steam to blow straight into the rising main at the end of each stroke, as sometimes happens in badly designed pumps of this class.

In tests by Professor Hudson Beare, of a pump fitted with this arrangement, about 13,500 ft.-lbs. of work, actually spent in lifting water, were obtained from 1 lb. of steam at a pressure of 55 lbs. per square inch.

Bailey's "Aqua-Thruster," a pump of the pulsometer class, is shown in Fig. 289.

CHAIN PUMPS.

Where very dirty liquids have to be pumped, such as slurry at brickworks, gas tar, liquid manure, and other liquids or semi-liquids, which would clog the valves of an ordinary pump, chain pumps are used. They are really elevators used for fluids. The illustration (Fig. 290) shows a hand pump of this type. It consists of a chain passing round a pulley, and having discs attached to it at intervals. These discs fit

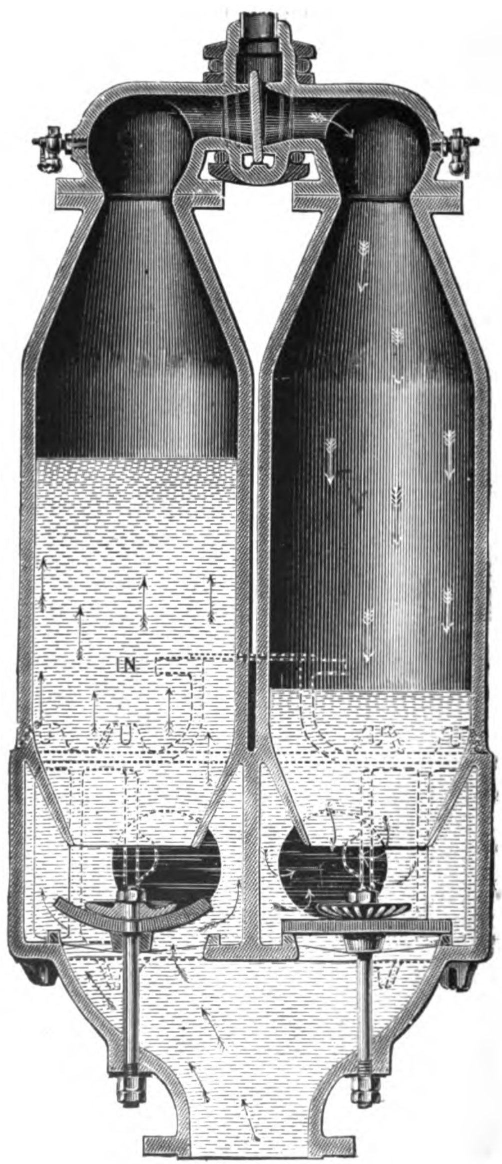


FIG. 289.

the pump barrel on the ascending side fairly, but not tightly. The action of the pump is, of course, simply that of an elevator. On account of the considerable leakage past the discs they are only used for small lifts, and a liberal allowance for "slip" must be made in calculating the probable discharge of a given pump. The chain wheel is made with recesses into which the discs pass, as seen in the illustration.

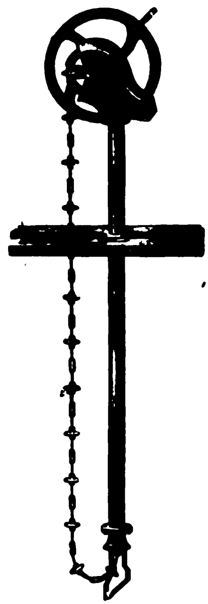


FIG. 290.

HYDRAULIC PUMPS.

The term hydraulic pump is here applied to a pump which is actuated by pressure water, in much the same way as a steam pump is actuated by steam. Some steam pumps, such as the Worthington pump, can, by a proper modification of the valves, be used as hydraulic pumps.

Fig. 291 shows a good form of hydraulic pump designed by Mr. Ellington, and used at the London Hydraulic Power Company's pumping stations, and at the Buenos Aires sewage pumping works. They are single-acting, with plungers 30 inches in diameter and 3 feet (some 4 feet) stroke, speed 10 double strokes per minute. The pressure-water is admitted by the valve seen at the top of the right-hand illustration, to the centre of the plunger to force it down, and to overcome the resistance of the constant pressure underneath the side rams attached to the plunger cross-head, which raise the plunger when the valve controlling the flow of water to the centre plunger is open to exhaust. It will be understood that only the downward is a working stroke, in the upward stroke only the weight of the plunger, etc., together with friction and suction, have to be overcome, hence the rams acting during this stroke are small. There is very little suction in the Buenos Aires pumps; the sewage to be pumped runs into the pump by gravity. The exhaust pressure-water is discharged into the pump cylinders, thus assisting to keep the plungers clean. The pump is started and stopped automatically by a float in the cistern, to which the water is pumped in the case of the London station, and in the sump *from* which the sewage is pumped in the Buenos Aires works. The efficiency in the case of some of the London pumps, working against a head of 80 feet, has been found to be over 75 per cent.

ACCUMULATOR PUMPS.

One or two good forms of pumping engine for supplying hydraulic pressure mains have already been described at pages 238 and 240. Fig. 292 gives a perspective view of an engine by Messrs. Armstrong, Whitworth & Co., which is much in favour for such work.

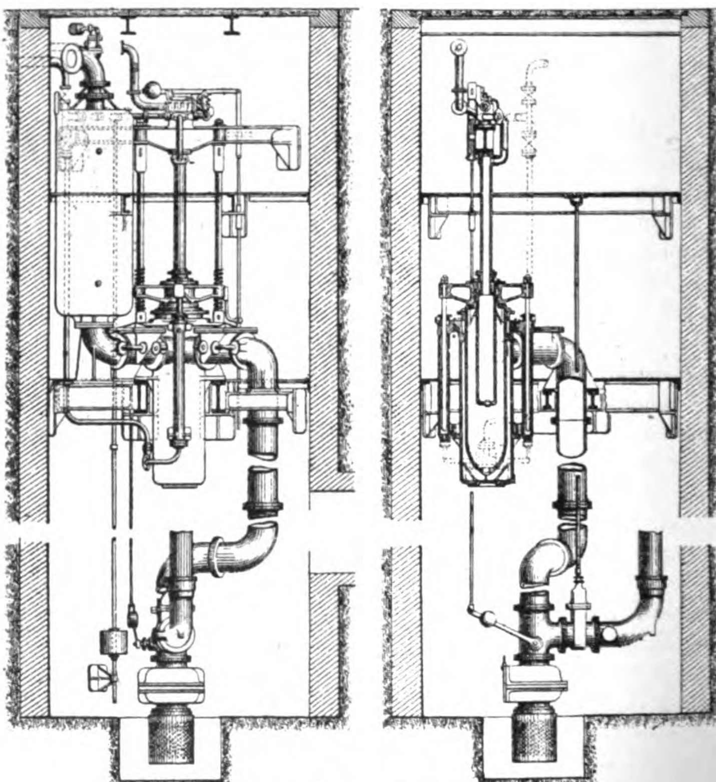


FIG. 29 .

The engine is a direct-acting double tandem-compound, i. e. it consists of two compound engines, the cylinders of each being arranged tandem fashion, with one piston-rod for the high and low-pressure cylinders, the pump plunger being a prolongation of this rod.

The high-pressure cylinders are fitted with double slide-valves,

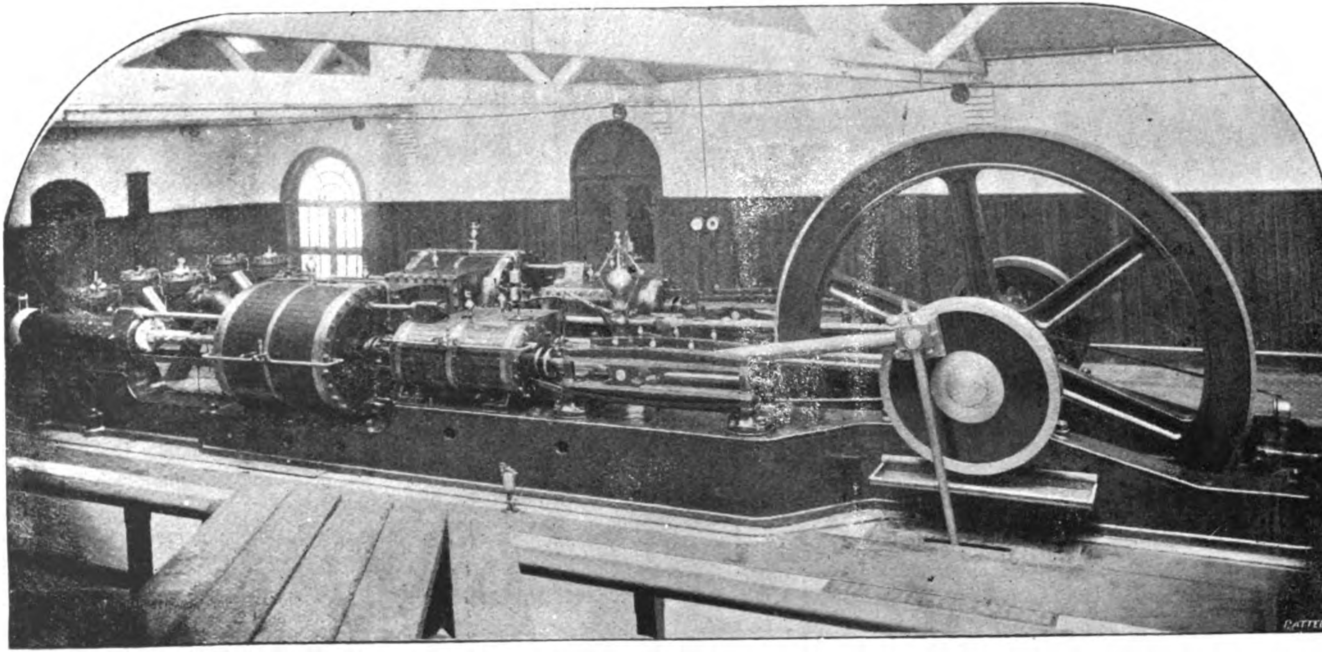


FIG. 292.

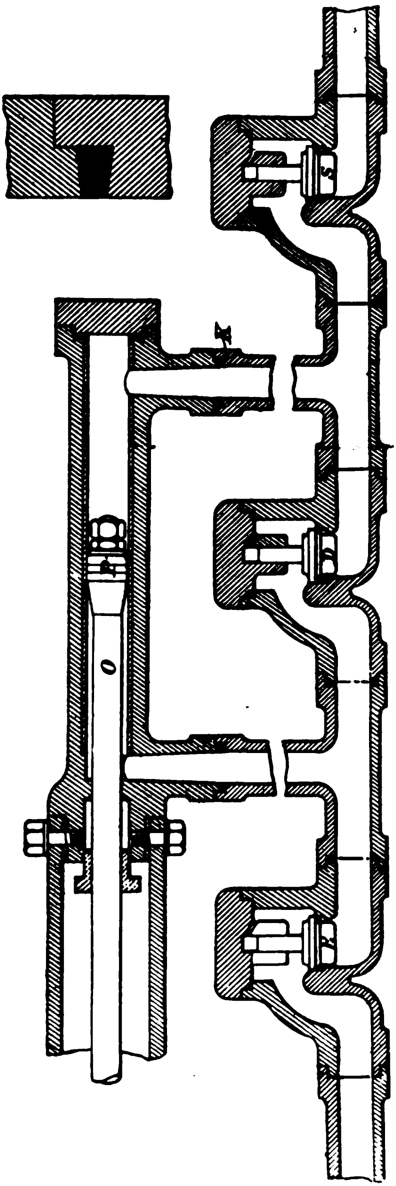


Fig. 293.

the upper slide being so arranged that the expansion can readily be varied by hand — since the constant load makes governor regulation of expansion no advantage—to suit the load or steam pressure, or in case the condenser is out of action. The maximum grade of expansion adopted is from 10 to 16 volumes, i. e. the steam is not allowed to expand to more than from 10 to 16 times its volume at admission.

The arrangement of pump and valves is shown in Fig. 293. In the backward stroke of P water is drawn in through the suction-valve S to fill the space left to the right of P, whilst in the forward stroke of P the water is discharged through the delivery valve D, half of it finding its way to the annular space round O, which is just half the cross-sectional area of P. Each stroke of the pump, by this differential arrangement, delivers the same quantity of water (equal to half the displacement of P) to the accumulator. E is a check delivery valve, always now used. This arrangement gives easy access to the valves. The two delivery valves and suction-valve for each pump are now frequently, as in the engine shown in Fig. 292, included in one valve-box.

In all these engines a throttle-valve is placed in the steam supply pipe, this valve being actuated by the accumulator load in such a way that when the accumulator is fully charged the valve closes, and when a small portion of the charge is withdrawn the valve is again opened. In compound engines working with high grades of expansion an automatic arrangement, which admits the steam direct to the low-pressure cylinder, is provided to assist the engine to start after being stopped or slowed down by the accumulator.

XXX.

THE HYDRAULIC INTENSIFIER.

PROBABLY the intensifier, as such, is due to Mr. Ashcroft, whose patent bears date of 1869. The same principle has, however, been made use of by many others, at different times, before and since this early date. Tweddell's intensifying accumulator, described at p. 236, though used primarily for a different purpose, acts in much the same way as the intensifier. The intensifier is an apparatus for increasing the pressure of water in hydraulic mains, pipes, or machines, using only the energy of the pressure-water itself to effect the change in pressure. But for this distinction a steam pump would be an intensifier. An intensifier worked the reverse way is a "diminisher," as a hydraulic pump usually is, giving a reduced pressure. The intensifier is in some respects analagous to the electric transformer.

THE BELLHOUSE INTENSIFIER.

This apparatus, the invention of Mr. Bellhouse, is shown in section in Fig. 294. It is much used in Manchester, where changes of pressure are required for presses worked from hydraulic or town mains.

The Bellhouse intensifier is single-acting. The high pressure can only be obtained in the up-stroke, and consequently on the return of the ram the "slack" water must either be allowed to run to waste or be made use of in the press; it takes up that amount of clearance between the goods and top of the press, hence called "slack." It also does the same amount of pressing, as the pressure due to the small ram on the larger one is about 200 lbs. per square inch. There are therefore two methods of using this single-acting intensifier, first without, and second with, arrangements for using the slack water for pressing.

The most simple valve for the first method is a single stop and let-off valve in connection with the larger ram and pressure supply. In this arrangement the pressure is constantly on the small ram. Assuming the large ram to be at the top of its stroke, on releasing the pressure on the large ram the pressure due to the area of the small ram forces out the slack water to waste. On reversing the

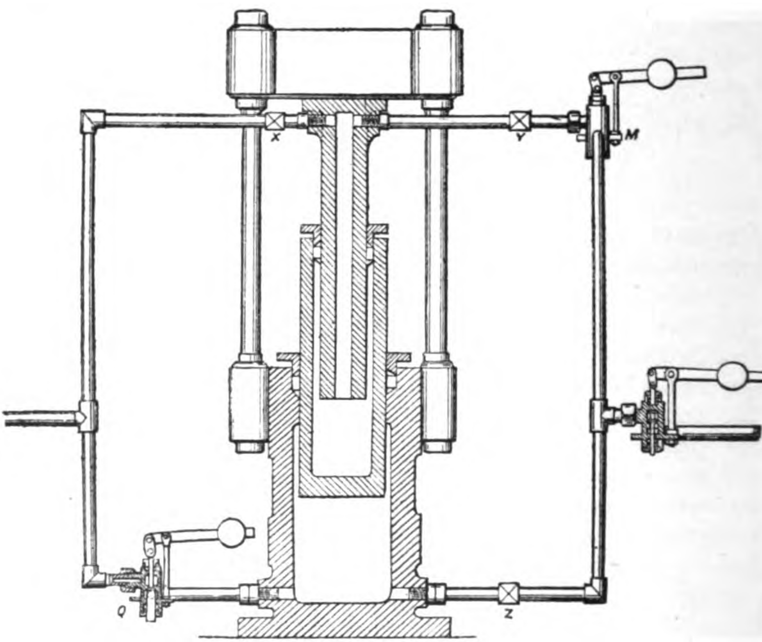


FIG. 294.

action, and applying the pressure on the large ram, the pressure becomes intensified in the ratio of the areas of the two rams. These alternating valves may be made to work automatically. In the second arrangement, Q and M (Fig. 294) are inverted weighted valves and X Y Z back-pressure valves. Supposing large ram at top of its stroke, the pressure being admitted to the small ram forces the slack water into the pipes until such resistance is met with that the inverted valve M opens, and in the second operation admits the pressure water directly upon the full area of the packing press ram. The third action takes place when the still more heavily weighted valve Q opens and admits the pressure on the large ram, which continues to rise until the desired pressure is obtained.

For fuller information the reader should consult a paper on 'Hydraulic Power,' by Mr. Gilbert Lewis, read before the Manchester Association of Engineers.

Ordinary Intensifier.

A more usual type of intensifier is shown in Fig. 295; * the action of the apparatus is somewhat as follows. Imagine the ram A to be at the bottom of its stroke (not as shown). The valve B (see plan at bottom of figure) is opened which admits water from the mains, raising the ram A, forcing water out through the valve D and filling the interior of A with water at the normal pressure. Valve C is then opened, the back-pressure valve G closes, and the pressure inside the ram is intensified in the ratio of the area of ram A to that of the hollow fixed ram E. In other words, the supply pressure acts on the area of A, and the new intensified pressure on the ram E. The fact that E is hollow need not confuse the student, as the water in this hollow space is at the same pressure as that acting on E, hence its resistance acts as that of any part of the solid annular end of E. This "intensified" water passes away through the centre of E and the pipe F to the machines supplied.

In one case the sizes were as follows:—

Diameter of A 15½ inches,
" E 6 inches,
Stroke of E 13 feet;

hence

$$\text{Intensifying ratio} = \frac{(15 \cdot 5)^2}{6^2} = 6 \cdot 67.$$

Pressure of supply 700 lbs. per square inch.

Intensified pressure $= 700 \times 6 \cdot 67 = 4669$ lbs. per square inch, minus an allowance for overcoming the friction of the apparatus. The actual intensified pressure was 4500 lbs. per square inch.

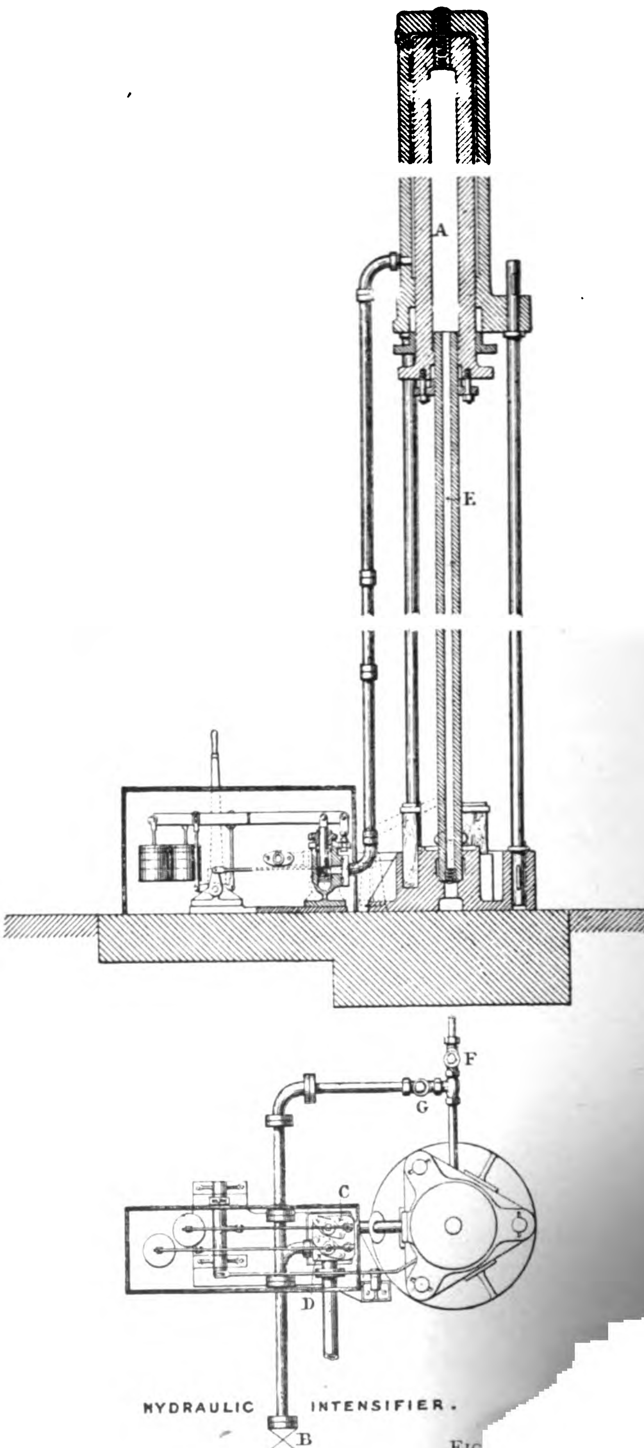
The water used per stroke was,

at 700 lbs. per square inch, 106·5 gallons;

that given off

at 4500 lbs. per square inch, 16 gallons.

* This, and Fig. 291, from Proc. Inst. C.E., vols. xciv. and cv.



HYDRAULIC INTENSIFIER.

Fi

Energy wasted per stroke,

$$2 \cdot 3 \{700 \times 106 \cdot 5 - 4500 \times 16\} \times 10 = 58,650 \text{ ft.-lbs.}$$

Energy received per stroke,

$$2 \cdot 3 \times 700 \times 106 \cdot 5 \times 10 = 1,714,650 \quad \text{,,} \quad \text{,,}$$

Hence efficiency of apparatus is

$$\frac{1,656,000}{1,714,650} = 0 \cdot 96.$$

or 96 per cent.

XXXI.

HYDRAULIC RAMS.

AMONG hydraulic machines this apparatus is unique, as its action is different from that of all other hydraulic apparatus and depends on a different principle.

As the accounts usually given of this principle are imperfect, and very often quite incorrect, and as these machines have recently received a considerable extension of their scope, it will be worth while to consider them in some detail.

In their generic form all hydraulic rams consist of a pipe of some length conveying water from a higher to a lower level, and a valve at the foot of this pipe which alternately allows the water in the pipe to flow and prevents its flowing. These are the only features common to all forms of this machine. The object aimed at is to use the fall of a quantity of water either to raise part of it to a higher level than the origin, or to compress air.

Considering only the first use at present, if it is desired to raise the water only to a slight height (say as high above the origin as the origin is above the tail race), the only other parts required are a smaller pipe rising from the lower end of the main pipe B into a delivery trough E (see Fig. 296). The action is as follows:—While the water escapes freely through the valve C, the water in pipe B is acquiring velocity and momentum. On closing of the valve C the momentum of the column of water in B causes it to continue to flow and rise up in pipe D, and some of it (if the pipes are suitably proportioned) rises to the top of pipe D and overflows into the delivery trough E. The momentum of the water is thus expending in raising

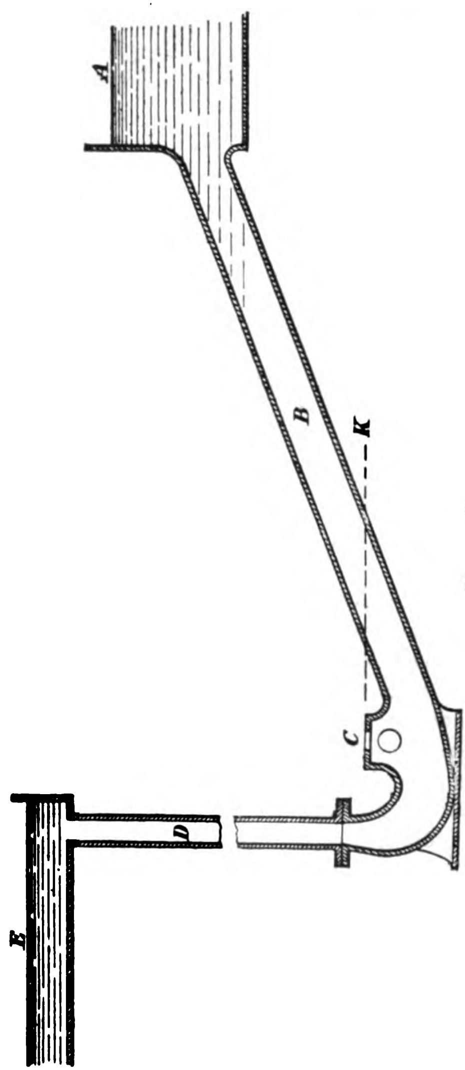


FIG. 296

part of itself, and when it is entirely expended, the whole mass of the water comes to rest. The extra head of water in pipe D then causes flow to take place in the contrary direction, some of the water being returned to the source A. The valve C being then reopened, the same cycle of operations is repeated. This obviously forms an exceedingly simple apparatus for purposes such as irrigation, and has been used for this purpose in France. An apparatus of this kind was exhibited in the Paris Universal Exposition of 1889. Excepting for very low lifts, this is, however, an inefficient arrangement, as a great part of the energy of the water is wasted in forming eddies during the flow and reflux. When the lift is considerable, the form of the apparatus is therefore as in Fig. 297, a delivery valve F and an air-vessel G being added, which makes the flow of water fairly constant through the rising main H, instead of intermittent.

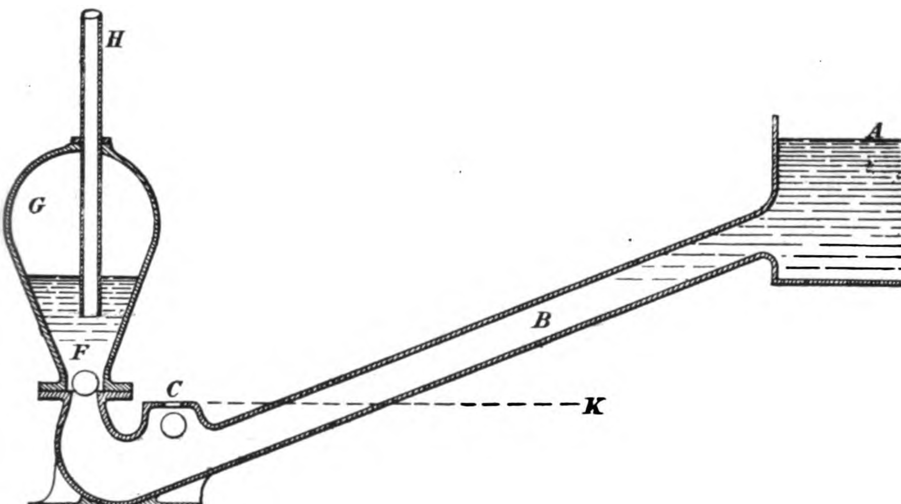


FIG. 297.

It is also possible to use a fall of impure water to raise pure water from a well. Hydraulic rams may, therefore, be classified as follows: (1) those with no air-vessel directly communicating with the drive-pipe B; (2) those with such an air-vessel; (3) pumping rams which utilise a fall of impure water to raise pure water from a well; and (4) a kind of ram working without the violent concussion noticeable in the others, and which may be used to compress air, and is sometimes called a hydraulic ram engine.

The general principles of the action of a simple ram of class (1) have already been indicated. In the case of class (2) the action is as follows: Water from A (Fig. 297) flows through the drive-pipe B (Fig. 298), say with increasing velocity. When the velocity reaches a certain limit the dash-valve C is lifted suddenly, closing the outlet and giving a suddenly increased pressure to the water, which enables it to lift the valve F against whatever pressure there may be in the air-vessel V (F is often a ground-in gun metal valve with feather guides and stop). The backward movement of the water now reduces the pressure on C, which opens—F closing—and the water flows on in B as before till C is again closed and the cycle of operations repeated. In some good modern rams an additional pulse-valve H is supplied, the valve being forced downwards by a spiral spring S, which can be compressed

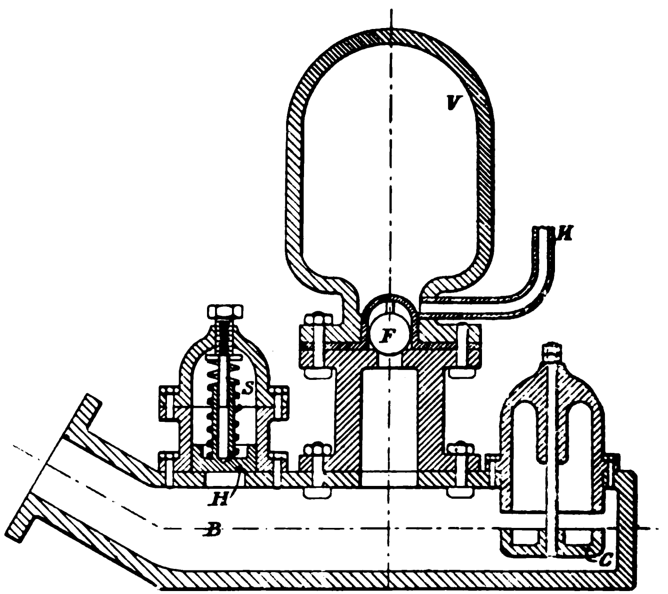


FIG. 298.

to the proper initial amount. This spring is compressed and H moves upwards under the ram pressure, thus storing up a certain amount of energy which is given out again on the reflex motion of the water, thus aiding the pulsing action so essential to success. The pressure in V is, or soon becomes, sufficient to force the water up the delivery pipe K, which is fitted with a proper stop valve,* not shown in the illustration. There is also a small snifting-valve, which

* **In starting the ram this valve is closed, the delivery-pipe filled with water, and a pressure produced in the air-vessel by a pump equivalent to the head of this water; the stop-valve being then opened, the ram is ready to start.**

may be a small tubular hole about the size of a pin in the trunk near F, and just below the delivery valve, with an enlargement in it in which is a small loose piston. When the dash-valve C closes, the water that forces open the delivery-valve F acts also on the small piston closing the small inlet, but when the water recoils the little piston is drawn back and a small quantity of air enters the air-vessel, bubbling upwards through the water. Thus the vessel V is kept supplied with air. A good example of a pumping ram (class 3) is shown in Fig. 299. The dash-valve C closing as before produces a sudden pressure of the water in B and in the cylinder R attached to B. In the cylinder R is a piston P which can move up and down water-tight, and which is connected by the link Q to a smaller piston *p*. The pistons are pushed down by the arm D, which is hinged at E, and also by a pair of counterweights, only one of which, W, is shown. S is a suction-valve attached to the suction-pipe reaching down into the well, F being the valve into the air-vessel V, and K the delivery pipe as before. The action is as follows: the increase of pressure in R due to the closing of C raises the pistons P and *p*, and also the counterweights W. In rising, *p* pushes water through F into the air-vessel V. As soon as the shock or increased pressure due to the closing of C is exhausted, the counterweights cause the pistons to descend, leaving a partial vacuum above *p*, which raises the water through the suction-valve S. It will be noticed that in this most useful form of ram the pure water cannot mix with the impure water, so that any stream much too impure for household purposes may be utilised to supply pure water by actuating the pumping ram. The illustration shows clearly how the pistons are packed, and gives other details which are of interest. If the supply of water be insufficient to work a ram continuously, it may be accumulated in a dam or large tank, communication between which and the drive-pipe is opened automatically by a float or similar contrivance when the water rises to a sufficient height.

Some Details of Hydraulic Rams.

It may be of interest to the young designer to have some useful proportions for parts of ordinary rams. For instance, the cubic content of the air-vessel should be about equal to twice that of a portion of the delivery pipe whose length is equal to, and the length of the drive pipe in small rams should be about equal to, the vertical height to which the water has to be raised. With high heads proportionally shorter, and with low heads longer, drive-pipes are necessary. It is said that

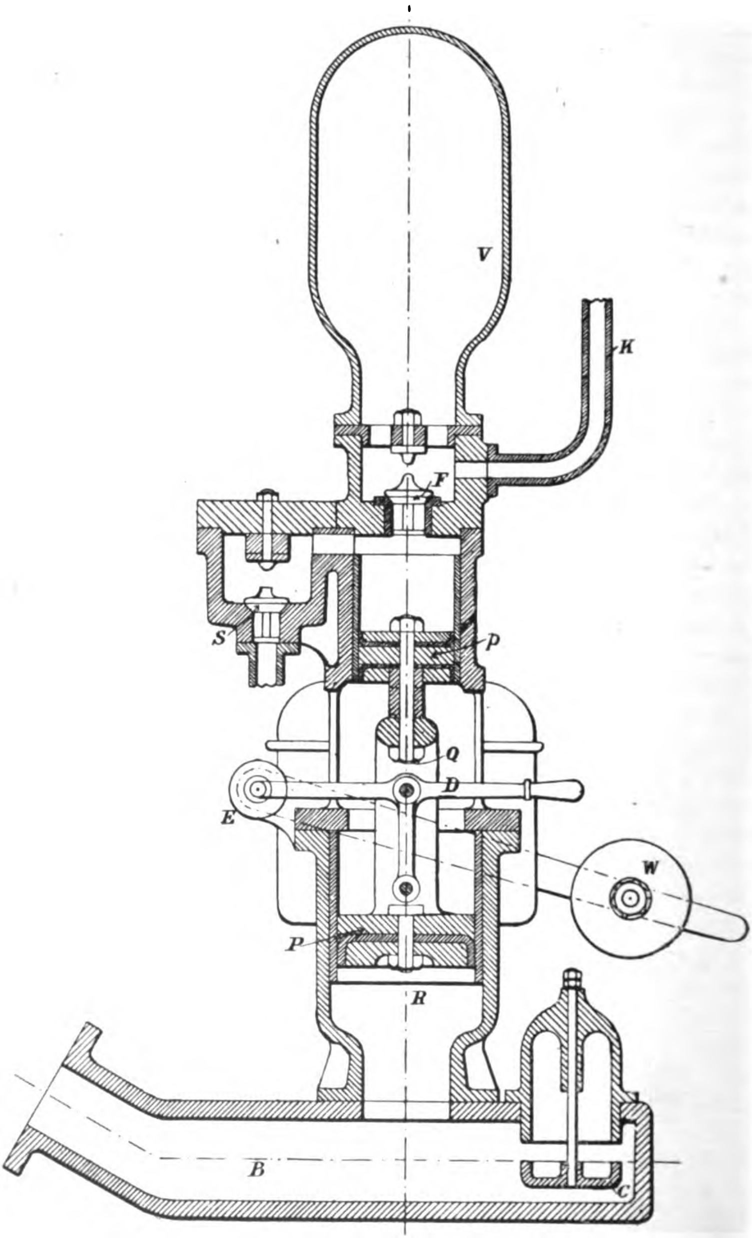


FIG. 299.

the delivery-pipe should be at least half the diameter of the drive-pipe. Other proportions for valves, etc., are best obtained from a drawing of a good ram. It may be noted in regard to the number of beats per minute, that with a short drive-pipe the beats are quicker than with a long one. In one case with short drive-pipe the beats varied from 100 to 150 per minute, and with a long drive-pipe the beats were from 28 to 35 per minute. In this case the short drive-pipe was about 8 feet and the long one 60 feet long, but the experiment shows the effect of length of drive-pipe on rapidity of beating, the working head being unaltered.

In regard to efficiency its average value was not very different in the two cases, but the longer drive-pipe gave more even results, the efficiency being more nearly constant.

It has been found that it is inadvisable to have a ram deliver water directly into a dwelling house, as the pulsating noise conveyed by the pipe and water causes considerable annoyance.

In introducing class 4, or what have been erroneously styled hydraulic ram engines, it may be necessary to correct a mistake as regards the theory of the action of hydraulic rams which is very prevalent.

Referring to what has been mistakenly called the theory of a simple ram, and confining our remarks to the form shown in Fig. 297, it is sometimes said the cause of water entering the air-vessel (where the pressure may be many times that of the head A K) is the "suddenness" of the closing of the valve C.

From what has been already said, and the example with which we began, it will hardly be necessary to point out the inaccuracy of this, or the still greater absurdity of the statement frequently made, that the ram "works by a blow." This statement is not made in any good text-book, but it is still current in popular explanations, and seems to receive some endorsement from the very name of these machines, which seems to imply violence.

The popular idea is, however, not without some foundation. In all, or nearly all, actual rams constructed up to a few years ago, there was very evident violence, violence so great that it had come to be accepted as certain that these machines could only be used on a very small scale. One great source of this violence was seen to be the sudden closing of the valve C which produced an actual blow or knock, but it was often believed that besides this the water impacted on the valve F with a second or even severer blow.

Many makers tried to diminish the blow of the valve on its seat by adding counterweights or springs. These devices did diminish the blow, obviously by retarding the shutting, but as the efficiency of the ram was then greatly diminished, this confirmed them in the idea that suddenness was of the essence of the action of a ram.

We have therefore this apparent contradiction, that, theoretically, suddenness is not essential, but that, practically, it is.

The explanation is exceedingly simple, but as for want of it the development of these useful machines has been retarded for many years, it forms a useful lesson in the disadvantage of slipshod reasoning.

The blow was, in fact, merely an accident of the particular construction adopted, and not an essential accompaniment of the principle of its action.

It is clear that if we shut the valve C *slowly enough*, we may dissipate the whole energy of the flowing water by fluid friction through the valve-orifice, leaving *none* to cause entry into the air-vessel. Hence, slow shutting of the valve must of course (other things being the same), diminish the efficiency of the ram. This loss by fluid friction is, however, quite sufficient to account for the loss of efficiency, without the necessity of assuming that the action of the water is in some mysterious way different when it presses on the delivery valve F gradually or suddenly.

If we could shut the valve C slowly *without choking the escape*, there is therefore every reason to expect that the efficiency of the ram would *not* be diminished, and even that it would be increased, for the valve C, as ordinarily made, does choke the escape somewhat, even when closing very quickly.

This is the essence of the important improvement now to be referred to. It is simply a device for shutting the valve C in such a way that the shutting does not check the free flow of the water.

The results of this alteration have been great. The most important is that it has made it possible to construct hydraulic rams of large size. Of course, for the complete attainment of this object, other alterations in the ancient design had also to be made, tending to greater efficiency; but it is this device, and its consequent abolition of all violence, which has made the other features practicable. Whereas, before this improvement, the largest practicable size of efficient rams was that with flow-pipe 4 or 5 inches in diameter, rams have since been made with pipes 2 feet in diameter, the velocity of flow being in some cases, 5 feet per second, instead of about 1 foot, as in the older forms.

Until lately rams were only useful for what are called domestic purposes—e.g. supply of water to single houses; they are now found to be suitable for the largest water powers, such as for water supply to cities.

Similar machines have also been made for compressing air directly by water power.

The absence of the imagined "water ram" action in these machines is shown by pressure diagrams taken from their interior, of which Fig. 300 is a specimen. Their efficiency, instead of being less than that of the violent rams, is greater, varying from 70 per cent. to over 80 per cent.

Small rams, say up to 4 inches diameter of pipe, are still made exclusively on the general type shown in Figs. 297 and 298, and this is justifiable, because for such small machinery a certain violence, although somewhat objectionable, is allowable, and it is therefore not worth while to increase the expense of the apparatus to avoid it. The new type of ram is therefore adopted exclusively for larger machines.

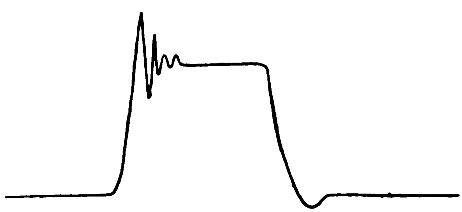


FIG. 300.

Fig. 301 shows the main features of one of these machines, omitting details.

In the figure the letters on the parts, corresponding to those of an old-type ram, are the same as those in Figs. 296 and 297. The additional parts are an antechamber M and the air-valve N. The form of the valve C and the methods of moving it vary with the size of the machine and the kind of work to be done. In this example (which is of a machine with pipe 2 feet in diameter) the valve C is an annular slide valve, and is moved by mechanism outside the ram, which is not shown in the figure.

The part played by the antechamber M and air-valve N is as follows:—When the valve C rises water begins to escape through it, the water in chamber M of course also escaping. This causes valve N to open and admit air to the chamber. Then, when valve C is again closed, instead of such closing at once shutting off the flow in pipe B, it merely diverts its course; while the valve C is shutting that part of the water which cannot readily escape through the narrowing orifice of the valve, flows into and fills the antechamber M, and the flow is therefore not in any way checked. When the

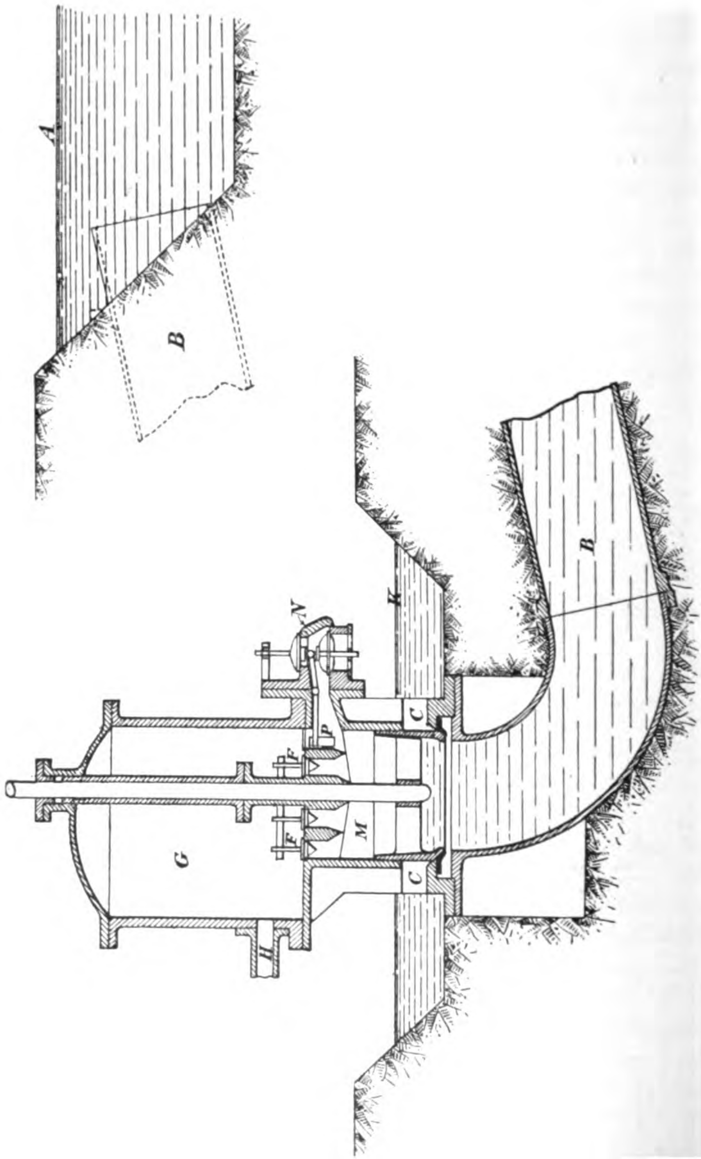


FIG. 301.

chamber is thus filled (which happens an instant after the main valve C is completely closed), the air-valve closes. The pressure in the antechamber then rises, and the delivery valves F opening, some of the water flows into the air-vessel against whatever pressure there may be there. In almost all cases, however, it is advantageous to trap a small part of the air in the antechamber, and to compress and enter this with the water. Where the main valve is worked by an outside motor, this air is made use of to work the valve.

The closing of the air-valve is accomplished in various ways, according to circumstances. In the example it is closed by the flow of the water in the antechamber past a float P, which is connected to the air-valve by a lever. There are means for readily adjusting the exact position of this float, so as to cause the closing of the air-valve at the instant required.

Machines of this type, when intended to compress air instead of pumping water, have the antechamber large enough, not only to contain the water which flows during the closing of valve C, but also the air which is to be compressed at each stroke, and the float P is then placed further below the roof of the antechamber than shown in the figure. The machine is very highly spoken of by Prof. Unwin and many other authorities, giving an efficiency of from 70 to 75 per cent. for pumping, and 80 per cent. when compressing air. Considering first cost and small amount of wear and tear; it compares very favourably with any other method of raising water used by engineers.

The above ram and other improvements in water-raising appliances are due to Mr. H. D. Pearsall, Assoc. Mem. Inst. C.E.

The Efficiency of Hydraulic Rams.

This has already been referred to in dealing with class 4, but it may be well to devote a little space to the question of the efficiency of the ordinary hydraulic ram such as that included in class 2. In all cases the efficiency $= \frac{Q_1 h_1}{Q h}$, where Q_1 is the quantity raised through a lift h_1, Q being the quantity used, and h the fall. It has been found by experiment that the efficiency depends on the ratio $\frac{h_1}{h}$. Eytelwein from 1123 experiments deduced the formula,

$$\text{Efficiency} = 1 \cdot 12 - 0 \cdot 2 \sqrt{\frac{h_1}{h}}.$$

This gives an efficiency of 0·77 when $\frac{h_1}{h} = 3$,

and an efficiency of 0·226 when $\frac{h_1}{h} = 20$.

If the ratio $\frac{h_1}{h}$ exceeds 31·36 the formula gives the efficiency as *a negative quantity.*

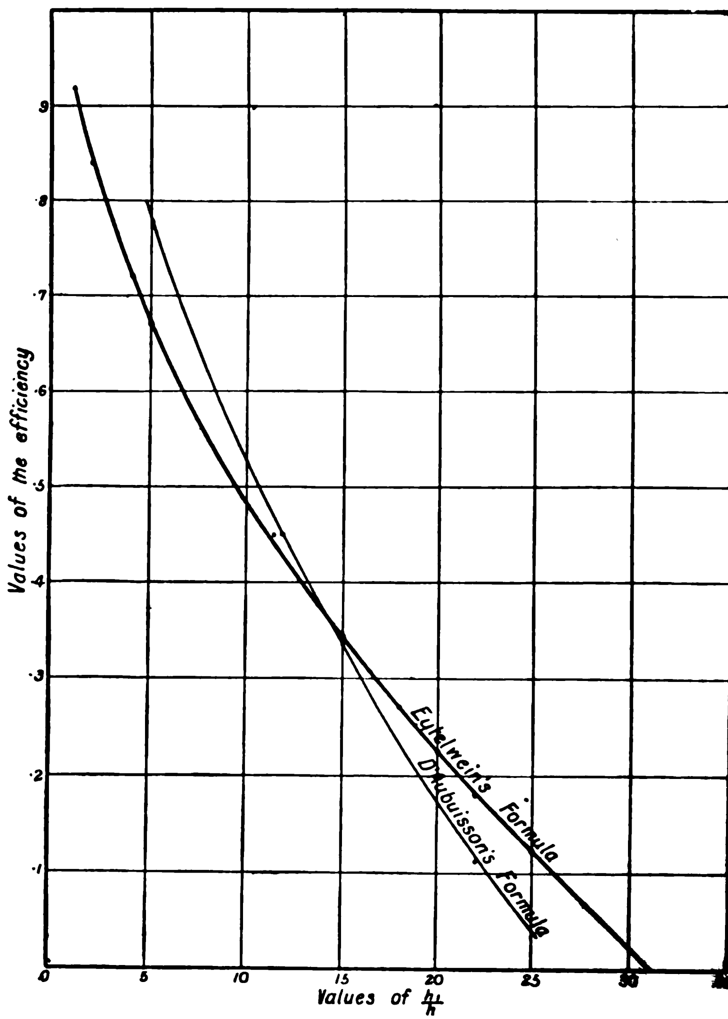


FIG. 302.

The formula of D'Aubuisson has also been used, giving an efficiency of 0·79 when $\frac{h^1}{h} = 5$, and 0·03 when the ratio is 28. Both these formulæ are shown plotted as curves in Fig. 302.

Example.—A hydraulic ram is employed in a stream where the fall is 20 feet and the amount available for working the ram 500,000 gallons per day. Find the quantity the ram will raise per hour to a height of 200 feet above its own level. The ratio is here 10, and referring to the Eytelwein curve (Fig. 302), we see that the efficiency is 0·485.

$$500{,}000 \text{ gallons per 24 hours} = \frac{500{,}000}{24} = 20{,}833 \text{ gallons per hour};$$

hence

$$\text{Since efficiency} = \frac{Q_1 h_1}{Q h} \qquad 0{\cdot}485 = \frac{Q_1 \times 200}{20{,}833 \times 20},$$

or

$$\frac{0{\cdot}485 \times 20{,}833}{10} = Q_1,$$

hence

$$Q_1 = 1010{\cdot}4 \text{ gallons per hour.}$$

XXXII.

THE SIPHON.

THE head h (= the difference of level of B and D) is that which overcomes the friction of the siphon pipe, and gives the observed flow at D. The part A B of the pipe which is in the water is not usually considered, but whilst this is correct when the water is at rest, with the water in motion the internal friction of this portion of the pipe, like that of the remainder, must be overcome. The theory of the siphon is often put somewhat as follows:—Imagine the siphon full of water and a diaphragm closing the pipe at D, a similar one being at B. The pressure on that at D is due to the column of water of height C d in the outer leg, whilst that at B sustains a pressure due to the shorter column in C B of height C b. Hence, if the diaphragms be removed, the greater pressure at D causes motion towards D, as under a head due to a column of water $b\,d$. The height of C above B is limited by the vacuum which can be obtained, which depends largely on the amount of air which collects at C—an air-valve there being often necessary. It is evident that if the siphon be as in Fig. 302*a*, the water, once it reaches C, will fall down C D readily, hence the problem often reduces itself to deciding whether, and with what velocity, the water will ascend the inner leg A C. To do this, if C *d* be more than 34 feet, we may take the head due to atmospheric pressure (less an allowance for contained air) minus the "rise" C *b*, and find the flow at C through A C due to this head; then consider the flow at D through pipe C D to be due to head = C *d* minus the head representing the back pressure of the atmosphere at D, plus any pressure transmitted from A C. If the inner leg be long, the flow through it will be less than that given by the above calculation for C D, and no pressure will

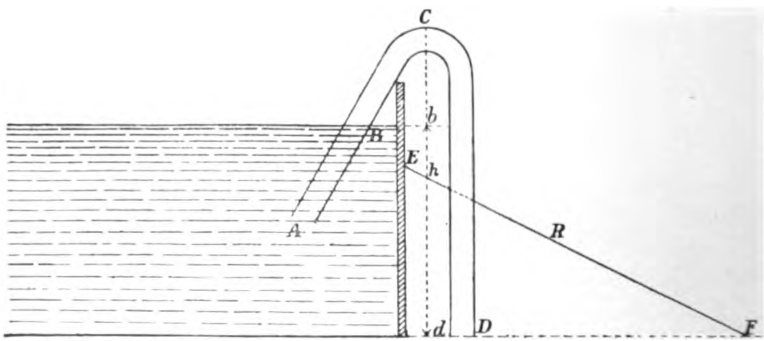


FIG. 302*a*.

be transmitted to the outer leg, but if the inner leg be direct and the outer one long, some pressure may be transmitted to assist the flow in C D. This consideration shows that A C should be made as short and direct as possible, and all joints should be made air-tight. In using cast-iron pipes a coating of coal-tar pitch, or similar solution, both within and without, is recommended by authorities on this subject. The actual flow in a siphon is usually less than that found by calculation, considering the siphon as an ordinary pipe with water in it flowing under head *b d*. This may be due to the air which collects at C, and to the inner leg being too long, so that the flow through it is not sufficient to keep the outer leg always filled.

If we know the probable velocity, V, in the siphon, and if we neglect entrapped air we may find it approximately from the rule $\frac{V^2}{2g} = \frac{h}{1 + 4f\frac{l}{d}}$, where l is the total length of pipe and d its diameter.

Measure B E vertically downwards to represent $\frac{V^2}{2g}$, and draw E F, making an angle with the horizontal whose sine $= \frac{4f}{d} \times \frac{V^2}{2g}$, where f is D'Arcy's coefficient ($= 16 \cdot 1 \times \lambda$ as given on page 44). Find a point R on E F such that E R = A C, then the siphon will not act properly if C is more than 34 feet above R, even if there be no air entrapped at C.

XXXIII.

HYDRAULIC BRAKE.

A Hydraulic brake is an apparatus for absorbing energy by fluid friction, developed mainly during passage of the fluid past an obstacle. The simplest form of the apparatus is the dash-pot so often employed for stilling vibrations.

Such a form as that shown in Fig. 303 is often employed, the body whose kinetic energy is to be wholly or partially absorbed acting on the piston-rod D, moving the piston, thus causing the fluid

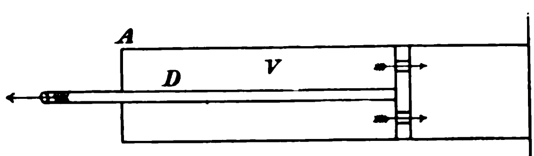


Fig. 303.

to pass through the holes in it. The arrows indicate the direction of flow relative to the piston for the given direction of motion of the latter.

A better form is shown in Fig. 304, where the piston fits the cylinder and the fluid passes from one end to the other, on the motion of the piston, through the pipe R and an orifice which can be closed to a greater or less extent by the tap C. The form devised by

Mr. Langley, for absorbing the energy of a moving train as it is brought to rest at a station, is illustrated in Fig. 305. In this case the buffer-stop is attached to a piston P, which fits the cylinder C fairly well, except at two portions, where rectangular strips are attached to the cylinder. These strips taper in thickness, so that when the stop is full out towards the right there is a considerable space for the

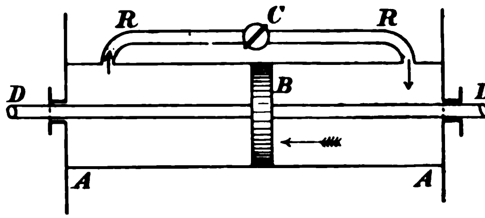


FIG. 304.

passage of the fluid past the piston; but as the piston travels to the left the orifices close gradually, as indicated by the sections, thus giving a nearly constant resistance. The piston is brought back to the full out position by a chain and counterweight, when the colliding body is removed. The hydraulic resistance is of the same nature as that met with in a pipe of suddenly varying diameter, and

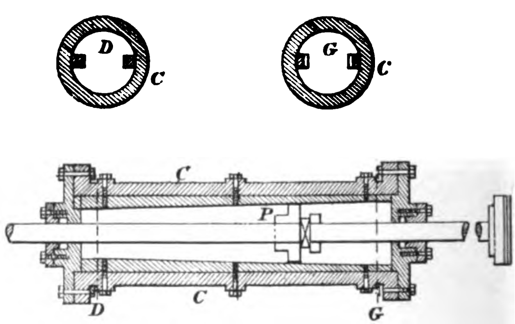


FIG. 305.

hence is usually assumed to be proportional to the square of the velocity of the water or of the piston, since—in the first two forms at any rate—the velocity of the water is proportional to the speed of the piston.

Just at first the motion is very rapid, and towards the end of the stroke very slow, and the assumption may not be correct in these cases, but on the whole the result arrived at from this basis of reasoning will probably not be far wrong.

Let $\frac{W V^2}{2g}$ be that portion of the kinetic energy of the colliding body which is to be absorbed by the apparatus, S the stroke, F the hydraulic resistance opposing the piston, the area of which is r times the effective area of the orifices, v being the velocity of the piston.

Then the loss of head is, as in the case of a pipe of suddenly varying diameter (neglecting the small loss due to contraction of stream), $h_1 - h_2$, or

$$\frac{P_1 - P_2}{w} = (r - 1)^2 \frac{v^2}{2g},$$

where P_1 and P_2 are the pressures per unit area on the two sides of the piston, and w the weight of unit volume of the fluid.

From this it is evident that

$$F = w A (r - 1)^2 \frac{v^2}{2g},$$

A representing the piston area.

Water being the fluid,

$$\text{the total resistance} = 62 \cdot 4 \, A (r - 1)^2 \frac{v^2}{2g} + R,$$

where R is the resistance due to solid friction.

Also,

$$\frac{W V^2}{2g} = (\text{average value of } F + R) S.$$

In illustration of the law it may be useful to plot a curve showing, in a given case, the variation of resistance as the piston moves forward.

Take the following numbers:

$$A = 0 \cdot 75,$$
$$r = 41 \cdot 25,$$
$$2g = 64 \cdot 4,$$
$$R = 50,$$

and assume a constant retardation if there were no solid friction, i. e. $v \propto \sqrt{x}$, where x is the distance from the end of the stroke.

(For convenience in plotting, suppose the motion to be in the opposite sense to that which is usual, the resistance being now replaced by the pull necessary to give the velocity v to the piston.)

The total resistance

$$y = 1177 \cdot 4\, v^2 + 50,$$

and $v \propto \sqrt{x}$;

$$\text{also } v = 14\tfrac{2}{3} \text{ when } x = 6,$$

or speed of train 10 miles an hour before collision, stroke 6 feet.

The law now becomes

$$y = 1177 \cdot 4 \times 36 x + 50.$$

This is evidently a straight line law, which when plotted gives the upper curve in Fig. 306.

If it were possible to have $v \propto x$, the law would be

$$y = 1177 \cdot 4\, (2 \cdot 44)^2 x^2 + 50.$$

This curve is the lower one in Fig. 306.

The average resistance in the first case is 127,141 lbs. With 6 feet stroke this one cylinder will absorb the kinetic energy of a train weighing 102 tons, moving at the given speed. In the second case the average resistance is evidently much less than in the first. It is easy to see that if the resistance—neglecting R—is to be constant

$$(r - 1)\, v \text{ must be constant,}$$

$$\text{i.e. } (r - 1) \sqrt{x} \text{ constant if } v \propto \sqrt{x},$$

$$\text{or } (r - 1)^2 x \text{ constant.}$$

From this law the area of the waterway may be designed. The application of a similar apparatus to the absorption of the energy of recoil of guns has already been referred to.

Hydraulic Dynamometer or Brake.

In the foregoing, reference has been made to methods of absorbing energy by fluid friction. The energy thus absorbed is not, in those cases, measured. The hydraulic dynamometer, now to be briefly described, not only absorbs energy, mainly by fluid friction, but measures the amount thus dissipated.

The use of an apparatus of this kind for measuring large powers was first proposed by Froude, but to Professor Osborne Reynolds is due the credit of having made the dynamometer a practical success. It consists of a hollow bronze wheel keyed on the shaft which supplies the power to be measured, the interior of the wheel being furnished with vanes inclined forward in the direction of motion. The wheel is surrounded by a bronze casing containing similar vanes inclined in the direction to oppose motion, the casing being supported

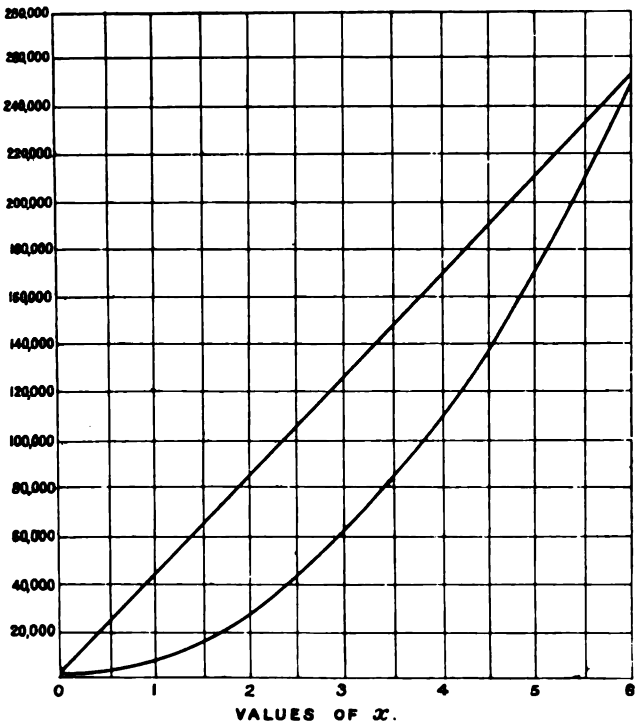


FIG. 306.

by the shaft, but capable of rotating on it as axis. Water under head enters the brake through a passage cut in the boss of the brake-wheel, flowing into the interior of the wheel and thence to the circumference under the action of centrifugal force. When the water arrives at the circumference, it impinges on the vanes of the casing, which deprive it of its rotational motion, returning it again into the spaces between the vanes of the wheel, and so on; thus, if the exit passage be sufficiently restricted, exercising a turning effect on the casing. The water, after having completed its functions in the wheel, escapes between the circumference and the casing into an external chamber, from which it finally escapes by a pipe containing a valve, by which the flow from the brake can be regulated at will. There are also air and overflow passages, which need not be further described here. If the exit passage is full open, the water leaves the brake as fast as it enters it, and very little resistance is offered to

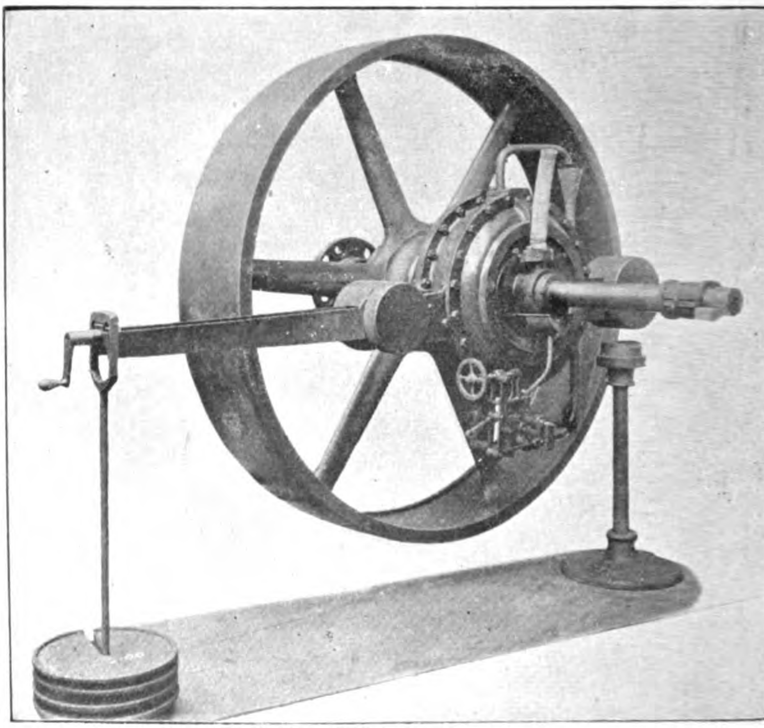


FIG. 307.

the rotation of the shaft; but if the exit valve is partially closed, the brake gradually fills, centrifugal force increasing the pressure on the exit valve until, finally, the discharge equals the supply, the resistance offered by the brake being then constant as long as the speed is constant. The brake has thus the important advantage of giving a constant resistance of—within wide limits—any required amount, this resistance being readily varied by simply turning a tap.

The turning moment or torque exerted by the brake on the casing is measured by a graduated lever rigidly attached to the casing, and provided with a jockey weight, which is moved along by a hand-wheel and double-threaded screw. The distance between two successive graduations represents one pound-foot of resisting torque offered by the brake. A brake of medium size will absorb from $\frac{1}{8}$ to 150 horse-power. The flow through the brake may be automatically regulated from the brake lever, thus giving a constant resistance with *varying* speeds. The apparatus has for some time been in successful use; that depicted in Fig. 307 being in use in the Whitworth Engineering Laboratory of Owens College, Manchester.

The largest dynamometer of this kind constructed up to the present, so far as the author knows, is one made in 1894 by Messrs. Mather and Platt for Messrs. Willans and Robinson, and designed to work up to a maximum turning effort of 26,400 lb.-feet, or about 1000 horse-power, at, say, 200 revolutions per minute.

XXXIV.

WASTE OF POWER IN HYDRAULIC MAINS.

A LARGE portion of the power generated at the central station of a hydraulic supply company is spent in overcoming frictional and hydraulic resistances in the mains. It is important, therefore, to be able to calculate approximately the amount of this waste in any given case, and if possible, in the case of new mains, to find the most economical diameter for a given power and pressure. We have already seen that the energy of 1 lb. of water at a pressure of p lbs. per square inch may be taken as $2 \cdot 3\,p$ ft.-lbs., the other items of the total energy being of little importance. Assuming that the weight of a cubic foot of water is still $62 \cdot 4$ lbs.—it is really a little more—then every cubic foot of water has a store of $62 \cdot 4 \times 2 \cdot 3\,p = 144p$ ft.-lbs.

Let the flow be Q cubic feet per second; the energy per second is $144\,p\,Q$ ft.-lbs., and if p be the pressure at entrance, the horse-power entering the pipe (call it E) is

$$\frac{144\,p\,Q}{550} = 0 \cdot 2605\,p\,Q \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Assuming D'Arcy's law for frictional waste to be true for the high pressures we are dealing with, that law gives loss of energy of

1 lb. as $4f\frac{L}{d}\frac{v^2}{2g}$, where $f = 0\cdot005\left(1 + \frac{1}{12\,d}\right)$ for smooth pipes.

The waste in Q cubic feet, or Q × 62·4 lbs., is therefore

$$4 \times 62\cdot4 \times f\frac{L}{d} \times \frac{v^2}{2g}Q,$$

and since Q is the quantity passing a given section in one second, the waste of energy per second by friction in L feet of straight pipe *d* feet in diameter is evidently this amount, or the horse-power wasted (call it W) is

$$\frac{4 \times 62\cdot4}{550} \times f\frac{L}{d} \times \frac{v^2 Q}{2g},$$

where v is the velocity of the water in feet per second. To eliminate v we have,

$$\frac{\pi}{4}d^2 v = Q;$$

$$\therefore \quad v^2 = \frac{16\,Q^2}{\pi^2 d^4},$$

also

$$Q = \frac{E}{0\cdot2605p} \text{ from (1)},$$

whence

$$v^2 = \frac{16\,E^2}{(0\cdot2605)^2 p^2 \pi^2 d^4};$$

hence

$$(2) \quad W = \left\{\frac{4 \times 62\cdot4}{64\cdot4 \times 550} \times \frac{16}{(0\cdot2605)^3 \times (3\cdot1416)^2}\right\} f\frac{L}{d^5}\frac{E^3}{p^3}$$

$$= 0\cdot646\,f\frac{L\,E^3}{p^3\,d^5}.$$

For a pipe 6 inches in internal diameter $f = 0\cdot0058$, hence for this diameter

$$W = \frac{0\cdot00374\ L\ E^3}{p^3\,d^5} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This rule is often employed for other diameters, as the change in the coefficient is not great for any likely change in *d*. The energy wasted at bends and junctions can be calculated from the rules given at page 58.

It is evident from (3) that the waste is greatly diminished by increasing *p*, or, better still, by increasing *d*, but the larger pipes necessarily cost more; hence it is a very interesting problem to find what is the best diameter, having regard on the one hand to frictional waste of energy, and on the other to greater cost of pipes. A similar question occurs in electric transmission of power by continuous or direct currents, but the use of alternating currents of high pressure (or voltage) renders the matter of economy in the use of copper for conductors a comparatively unimportant one.* These things will be referred to more fully later on. The waste of power may be looked at from the point of view of the power which actually *arrives* at the distant station, instead of that which is sent in.

Let D be the horse-power delivered, then evidently $E = D + W$, and this value of E must be substituted in (3), which gives an equation containing cubes, etc., of W, capable of being solved by trial or by any of the approximate methods given in treatises on algebra.

Suppose, for instance, we wish to *deliver* 100 horse-power at a place one mile distant through a 4-inch pipe, the pressure at entrance being 700 lbs. per square inch. Here

$$W = \frac{0 \cdot 00374 \times 5280 \times (100 + W)^3}{700^3 \times (\frac{1}{3})^5},$$

and simplifying,

$$542{,}798\, W - (D + W)^3 = 0.$$

Let the left-hand side be denoted by f(W).

A very good practical method of finding a solution is to choose various values of W, calculate f(W), and tabulate as follows:

W.	f(W).
34	− 57,444
36	− 28,648
38	− 3,108
40	+ 19,120

* The whole matter is fully discussed in an article on "Hydraulic and Electric Transmission of Power," by the author, published in 'Engineering' of May 22nd and June 5th, 1891.

Plotting these values on squared paper, putting values of f (W) vertically or as ordinates, and letting the horizontal axis be taken across the middle of the sheet, we get a curve which crosses the latter axis at the point which gives the value of W, making f (W) = 0, the solution required.

In this case it will be found that W is 38·25; in other words, we must send in 138·25 horse-power, so that 100 may arrive at the distant end. If we send in 100 horse-power, only 14·48 are wasted, and 85·52 arrive. A somewhat extreme case has been taken to show the difference in the two methods, the pipe being too small for the power.

The following tables show the amount of power wasted in various cases:

TABLE I.—Pressure at entrance 700 lbs. per square inch.

Horse-power sent in.	Horse-power lost in one mile of straight pipe.			Horse-power lost in five miles.		
	8-inch Pipe.	6-inch Pipe.	4-inch Pipe.	8-inch Pipe.	6-inch Pipe.	4-inch Pipe.
100	0·422	1·84	14·48	2·11	9·2	72·4
200	3·37	14·72	115·84	16·8	73·6	..
400	26·9	117·76	..	134·4	..	..
500	52·7	175	..	263·5	..	..

TABLE II.—Pressure at entrance 1120 lbs. per square inch.

Horse-power sent in.	Horse-power lost in one mile.			Horse-power lost in five miles.		
	8-inch Pipe.	6-inch Pipe.	4-inch Pipe.	8-inch Pipe.	6-inch Pipe.	4-inch Pipe.
100	0·103	0·448	3·63	0·507	2·24	17·64
200	0·819	3·59	28·23	4·09	17·94	141·18
400	6·55	28·7	226·26	32·81	143·75	..
500	12·87	42·71	439·5	64·35	213·58	..
1000	102·93	341·25	..	..	..	..

If the percentage of the entering power which may be wasted is determined beforehand, we have the following tables:

TABLE III.—DISTANCES TO WHICH 500 HORSE-POWER MAY BE TRANSMITTED WITH A GIVEN LOSS. Pressure at entrance 700 lbs. per square inch.

Percentage of Entering Power lost in Transmission.	8-inch Pipe.	6-inch Pipe.	4-inch Pipe.	3-inch Pipe.
per cent.	feet	feet	feet	feet
10	5,004·9	1,146·4	147·03	31
20	10,009·8	2,292·8	294·06	62
40	20,019·6	4,585·6	588·12	124
50	25,024·5	5,732	735·15	155
80	40,039·2	9,171·2	1176·24	248
100	50,049	11,464	1470·3	310

TABLE IV.—DISTANCES TO WHICH 500 HORSE-POWER MAY BE TRANSMITTED WITH A GIVEN LOSS. Pressure at entrance 1120 lbs. per square inch.

Percentage of Entering Power lost in Transmission.	8-inch Pipe.	6-inch Pipe.	4-inch Pipe.	3-inch Pipe.
per cent.	miles	miles	miles	feet
10	3·916	0·87	0·11	127·04
20	7·833	1·74	0·225	254·08
40	15·66	3·48	0·45	508·17
50	19·5	4·35	0·56	635·2
80	31·3	6·93	0·90	1012·3
100	39·16	8·7	1·1	1270·4

Take 3*d.* per horse-power-hour as an average cost, 1 horse-power day and night for a year comes to 110*l.*, though if the power be taken continuously probably this is too much. The cost of horse-power wasted therefore works out to

$$\frac{110 \times 0\cdot00374\,E^3}{p^3\,d^5} = \frac{0\cdot411\,E^3}{p^3\,d^5}.$$

Adding to this the annual interest on cost of 1 foot of pipe, differentiating and equating to zero, a value of d in terms of E for any given pressure and safe stress can be found. The result obtained in the article referred to is that for a pressure of 700 lbs. per square inch $d = 0\cdot079\,E^{\frac{3}{7}}$ is the rule for most economical diameter.

One way of obtaining a roughly approximate solution is to assume the price of pipes proportional to the weight of metal. This is not accurate, because different safe stresses are taken for different sizes of pipes, and there is more trouble and expense per ton in casting small than large pipes. These two items, however, tend to neutralise one another.

Assume price per foot proportional to weight or area of cross section, i.e.

$$\text{Price} \propto \frac{\pi}{4}(D^2 - d^2) = K \times \frac{\pi}{4}(D^2 - d^2).$$

Since 1 foot of 6-inch pipe costs when laid about 0·35*l.*—this, however, including expenses *not* proportional to weight—and taking 2500 lbs. per square inch as the working stress, we have by a reference to the curve on page 245 the thickness for a pressure p of 700 = 1 inch.

Therefore for this pressure,

$$\text{Price } 0 \cdot 35 = K'\left(\frac{7^2}{144} - \frac{6^2}{144}\right),$$

or

$$K' = 3 \cdot 87;$$

hence we assume that for any pressure the price = 3·87 $(D^2 - d^2)$. Now, to get D in terms of p,

$$D^2 = d^2 \frac{(f+p)}{f-p}.$$

$$\text{Price} = 3 \cdot 87 \left\{ d^2 \frac{(f+p)}{f-p} - d^2 \right\} = 3 \cdot 87\, d^2 \left\{ \frac{f+p}{f-p} - 1 \right\} = \frac{7 \cdot 74\, d^2 p}{f-p}.$$

llowing 12 per cent. per annum as interest, including depreciation, .tc., we have the total cost per foot of pipe per annum :

$$\frac{0 \cdot 411\, E^3}{p^3 d^5} + \frac{0 \cdot 12 \times 7 \cdot 74\, p\, d^2}{f-p}$$

+, possibly, a term depending upon repair expenses, etc., which we will neglect.

Assuming values of f and p, differentiating and equating to zero, we get the value of d in terms of E, which makes this cost a minimum.

For instance, if $p = 700, f = 2500$, we have

$$d = 0 \cdot 07\, E^{\frac{3}{7}},$$

or a 6-inch pipe is right for 100 horse-power at this pressure, but it is wasteful to force a greater power through it. The solution given in the article in 'Engineering' leads to an almost identical result, though worked out in quite a different way.

If p = 1120, the rule becomes

$$d = 0\cdot05\ E^{\frac{3}{7}}.$$

It has already been pointed out that these rules are only approximate, but they serve to show the importance of having the pipe of sufficient diameter for the horse-power transmitted through it. The following table, giving the most economical diameter for certain powers, as compiled from the above rules, may be useful.

TABLE V.

Pressure at Entrance 700 lbs. per square inch.		Pressure at Entrance 1120 lbs. per square inch.	
Diameter of Pipe.	Horse-power sent in.	Diameter of Pipe.	Horse-power sent in.
feet		feet	
0·37	50	0·267	50
0·503	100	0.359	100
0·679	200	0·484	200
0·913	400	0·717	500
1·35	1000	0·965	1000

If we could determine what voltage agrees with an assigned hydraulic pressure, we might make a comparison between the two systems as regards conductor waste of power. This we cannot accurately do, as we should have to compare things of different kinds, but we may look at the matter from the following point of view.

One ampere at a pressure of one volt conveys $\frac{33,000}{746} = 44\cdot23$ ft.-lbs. of energy per minute.

One cubic foot at a pressure of 1 lb. per square inch conveys $2\cdot3 \times 62\cdot4 = 143\cdot52$ ft.-lbs. of energy per minute.

We may assume, therefore, that a pressure of 1 lb. per square inch agrees with $\frac{143\cdot52}{44\cdot23} = \frac{3\cdot24}{1}$ volts. Or we may suppose the units of pressure to agree, and then our units of quantity would be to each other in the above ratio, one cubic foot being analogous to 3·24 amperes.

Lord Kelvin and others have deduced rules for the most economical area of electrical conductor under given circumstances. A current density of about 380 amperes per square inch is often taken as giving the best result.

Before leaving this subject it may be well to mention that the rules for most economical area of conductor, deduced, from the point of view of the power which *arrives*, by Professors Ayrton and Perry (and given in the 'Electrician' for March 1886) may be applied to the case where E horse-power are *sent in*, the horse-power wasted per mile being

$$W_1 = \frac{E \sin \theta}{n + \sin \theta},$$

where θ is the angle whose tangent is $\frac{n t}{P}$, n being the number of *miles* of conductor, P the pressure at entrance in volts, and t a constant depending on the price of copper, the cost of one electrical horse-power, etc., and often taken as about 17. The value of t corresponding to a current density of 380 amperes per square inch is 16·636.

Using Lord Kelvin's rule for area of conductor, and giving a current density of 380 amperes per square inch, the power wasted is

$$W = 16{\cdot}636 \frac{E}{P} \times l,$$

l being the length of conductor in miles.

Table VI., compiled from these rules, is interesting. In all cases the pipe or conductor is of that area or diameter which is most consistent with economy. The electric pressure of 2000 volts is taken instead of 700 × 3·24, for the sake of round numbers, and there is a return conductor. Returning to hydraulic transmission, if the coefficient for a 6-inch pipe be taken as correct for all diameters, a simple rule can be obtained for power waste when the pipe is properly proportioned. Thus for a pressure of 700 lbs. per square inch $d = 0{\cdot}07\ E^{\frac{3}{7}}$, and the wasted horse power per mile is

$$W_1 = \frac{0{\cdot}00374 \times L E^3}{700^3\ \ (0{\cdot}07\ E^{\frac{3}{7}})} = \frac{0{\cdot}00374 \times 5280 E}{(0{\cdot}07)^5 \times 700^3} = 0{\cdot}034\ E^{\frac{6}{7}}.$$

The similar rule for a pressure of 1120 lbs. per square inch is

$$W_1 = 0{\cdot}045\ E^{\frac{6}{7}}.$$

TABLE VI.

Horse-power sent in.	Hydraulic Transmission. Pressure at Entrance 700 lbs. per square inch. Horse-power wasted. Distance 1 mile.	Horse-power wasted. Distance 5 miles.	Horse-power wasted. Distance 20 miles.	Hydraulic Transmission. Pressure at Entrance 1120 lbs. per square inch. Horse-power wasted. Distance 1 mile.	Horse-power wasted. Distance 5 miles.	Horse power wasted. Distance 20 miles	Electric Transmission. Pressure at Entrance 700 volts. Conductor designed by Lord Kelvin's Rule. Horse-power wasted. Distance 1 mile.	Horse-power wasted. Distance 5 miles.	Horse-power wasted. Distance 20 miles.	Electric Transmission. Pressure at Entrance 2000 volts. Conductor designed by Lord Kelvin's Rule. Horse-power wasted. Distance 1 mile.	Horse-power wasted. Distance 5 miles.	Horse-power wasted. Distance 20 miles.	Electric Transmission. Pressure at Entrance 2000 volts. Conductor designed by Ayrton and Perry's Rules. Horse-power wasted. Distance 1 mile.	Horse-power wasted Distance 5 miles.	Horse-power wasted. Distance 20 miles.
[illegible]	0·972	4·860	19·44	1·286	6·430	25·72	2·38	11·90	47·6	0·832	4·160	16·64	0·822	4·106	15·64
[illegible]	[illegible]	8·805	35·22	2·33	11·65	46·60	4·76	23·80	95·2	1·663	8·318	33·27	1·644	8·212	31·28
[illegible]	[illegible]	15·95	63·80	4·22	21·10	84·40	9·52	47·60	190·4	3·327	16·636	66·54	3·28	16·42	62·56
[illegible]	[illegible]	22·57	90·3	5·78	28·90	115·60	14·28	71·40	285·6	4·989	24·945	99·78	4·93	24·64	93·84
[illegible]	[illegible]	[illegible]8·90	115·6	7·65	38·25	153·00	19·04	95·20	380·8	6·654	33·27	133·08	6·57	32·85	125·12
[illegible]	[illegible]	[illegible]4·95	139·8	9·25	46·25	185·00	23·80	119·0	476·0	8·318	41·59	166·36	8·22	41·06	156·4
[illegible]	[illegible]	[illegible]3·40	253·6	16·78	83·90	335·6	47·6	238·0	952·0	16·636	83·18	332·72	16·44	82·12	312·8

This coefficient is not really accurate, as the variation of D'Arcy's coefficient should be taken into account. That, however, would make only a small difference in the results for any ordinary difference in the diameters, and to avoid complication it is here neglected. It may seem at first sight wrong to have a greater horse-power wasted when a higher pressure is used, but it must be remembered that the pipe is in this case much thicker and more costly—although of smaller diameter—than that required for lower pressures.

If the pipe be designed without regard to economy, the waste increases rapidly with the power transmitted after the proper power for its diameter has been reached.

Thus, if the pipe be 6 inches in diameter, it is all right, at 700 lbs. per square inch, for powers up to 100, but for 200 horse-power the waste is 14·72 in one mile, 73·6 in 5 miles, and so on. If the pipe be properly designed it is only 8·8 instead of 73·6 in 5 miles. If we attempt to force 500 horse-power through such a pipe we find that 175 are wasted in the first mile, and that it is impossible to transmit any of the power beyond a distance of 2·8 miles, whereas, if properly proportioned, the waste is only 6·9 horse-power in the first mile instead of 175. Enough has been given to show the great importance of not having the pipe too small for the power it conveys; the remedy, if a large pipe be objectionable, lies in duplicating or triplicating the pipe. The use of some other material than cast iron will probably, in the near future, allow this frictional waste of power to be greatly reduced; but the solution here given can be made applicable to the new material by the substitution of the new cost of 1 foot of pipe and the new safe stress.

It is not in our province to enter into a complete comparison of the hydraulic and electric systems. The limit of pressure is soon reached in hydraulic work, hence for long distances the electric system practically holds the field, not only on account of the high pressures which can be used, and hence the comparatively small cost of conductors, but also on account of the ease with which conductors can be fixed in out-of-the-way places, and the efficiency of electro-motors when running either with full or partial loads. However, in towns, and for comparatively short distances, the hydraulic system compares very favourably with any other as regards efficiency, and supplies probably the best means of working lifts, cranes and other machines of that kind. Recent improvements, such as those of Mr. Rigg, show that the provision of a hydraulic motor of high efficiency at all loads, and which will run at constant speed, is a possibility of the near future, if indeed it has not already been constructed; this being the only thing wanted to render hydraulic power in many respects the best for intermittent business, domestic, and power, operations in cities. Thus, water, the commonest gift of nature, becomes the most ready means of obtaining power in some places, and the most efficient means of *transmitting* power for comparatively short distances in all. It is certain that the branch of engineering herein briefly referred to, will, as our coal becomes more expensive, assume more and more important dimensions; and London, as it ought, in some respects has shown the way.

APPENDIX.

WASTE OF ENERGY AT CURVED BENDS.

THE following rule has been deduced* from the most recent experiments on the head (h_B) wasted at curved bends :

$$h_B = a\,l\,\rho^p\,v^n \qquad \rho = \frac{r}{R},$$

in which r = radius of pipe, R = radius of centre-line of bend, and l = length of bend measured along centre-line of pipe; where a, p and n are constants, of which a few values are given in the accompanying table.

a	p	n
For pipes of varnished wood, ρ not greater than 0·2, a = 0·00827	0·83	1·777
Ditto, where ρ is not less than 0·2 and not greater than 0·5, a = 0·1245.	2·5	1·777
For drawn brass pipes, a = 0·00857	0·83	1·76
For cast-iron asphalted pipes 12 in. to 16 in. diam., a = 0·00415 to 0·00275.	0·83	1·78 to 1·86
Ditto, 30 in. diam., a = 0·001503	0·83	2·0

* Simplified from the rules given in a paper by Mr. Alexander, published in the *Proc. Inst. C.E.* for April, 1905.

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