Treatise on mills and millwork

On the principles of mechanism and on prime movers

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Section III. On prime-movers.

SECTION III.

ON PRIME-MOVERS.

CHAPTER I.

ON THE ACCUMULATION OF WATER AS A SOURCE OF MOTIVE POWER.

THE machinery of mills, as a whole, may be generally divided into three classes; — the *prime-movers*, from which the power is derived for keeping the machinery of the mill in motion; the *transmissive* machinery or *millwork* (shafting, gearing, &c.), by which the power obtained through the prime-mover is distributed over the different parts of the mill, so that it may be applied at the most convenient place and at the required velocity; and lastly, the *machines*, technically so called, by which the special operations of the mill in the preparation of its manufactures are carried out. It will be convenient to treat of these divisions in separate sections and in the order just named.

Prime-movers are those combinations of mechanism which *receive motion and force* directly from some natural source of power, and convert it into that condition in which it is applicable to the purposes of manufacture. Thus the water-wheel takes from the falling water a part of the *work* accumulated in it, and imparts it as a rotatory motion to the machinery of the mill; and, similarly in the steam-engine, the heat force of the fuel is converted through the medium of the pressure of the steam into motive power in a condition for producing work or mechanical effect. Also the force of currents in the atmosphere impinging upon the expanded sails of windmills, has been in former days extensively employed as a motive power. From these three sources, falling or moving water, the combustion of coal in the production of steam, and wind, we derive almost exclusively at the present time the motive power necessary for carrying on our immense mining and manufacturing systems.

It is only of late years that in this country the steam-engine has nearly superseded the use of air and water as a prime-mover. Until recently steam has been auxiliary to water, it is now the principal source of power, and waterfalls are of comparatively small value, except in certain districts. So long as water was depended upon, the mills of Great Britain and Ireland were necessarily circumscribed in their operations and diminutive in size; they have now become so colossal, that they require steam-engines of much greater power than the largest water-wheels, and there appears to exist no limit to the magnitude and importance to which they may yet attain.

Water-wheels, therefore, are those prime-movers which receive a certain portion of their energy from falling or flowing water, and their power or dynamic effect clearly depends upon the amount of water supplied and the height through which it falls, or its velocity at the point of application. Hence water-wheels are usually placed on the banks of rivers where a large body of water is at hand, and near some considerable natural or artificial fall in the bed of the stream.

In establishments where manufacturing processes are carried on, and a large body of men employed, it is essential to success that there should be no stoppages, and that there should be always at command a uniform power, equal to the requirements of the mill. Now, as the quantity of water in rivers varies considerably at different periods of the year and in different conditions of weather, it has been found necessary, in many instances, to impound the water by means of *reservoirs* placed at the sources or the higher portions of the river, so as to retain the waters of wet seasons, and to part with them again in periods of drought and deficiency. To a small extent this may be effected by weirs thrown across the river, so as to retain the water which comes down at night, for the use of the mill during the day. But in many instances large reservoirs of a hundred or more acres in extent, and containing when full several million cubic feet of water, have been constructed. In these the drainage from a large extent of country is collected during the rainy seasons, and remains stored for use, whenever the supply of water in the river becomes inadequate; in this way damage from floods is prevented on one hand, and on the other the supply for an indefinite period of time is equalised. Among the large works of this kind are the Shaws' waterworks at Greenock, and the Lough Island Reavy or Bann reservoirs in the county of Down, in the north-east of Ireland, together with later works of the same kind for the supply of water to the cities of Glasgow and Manchester, Melbourne, &c.

Reservoirs are best placed in hilly districts, at the bottom of a valley into which the water drains from a considerable extent of country. In selecting a site for reservoirs regard must first be had to the value of the land. They should be placed in retired valleys, where the cost of the land does not bear a high ratio to the cost of construction, and should there exist a natural lake it may be converted into a reservoir with greatly increased economy. Regard must also be had to the nature of the site. The reservoir should be restrained as far as possible by the natural rise of the ground around it, in order that as few embankments as possible may be requisite for the retention of the water. Again, the geological structure of the country must be examined, as the quantity of water to be expected to flow from a flat country, well clothed with vegetation, will be very different from that which will pour in torrents down the steep declivities of uncovered mountains. In districts of limestone, abounding in vertical fissures and subterranean cavities, a very much smaller quantity of water will drain off the higher districts than from a non-absorbent formation of primitive rock; or where the beds are horizontal and impervious. The steeper the district and the more rapidly the water is discharged to the reservoir, the less will be lost by evaporation and absorption.

It is necessary in constructing reservoirs, to obtain some measure of the quantity of water which may be expected to accumulate annually, in order to provide sufficient storage. For this purpose it is most important to determine the area of land draining into the valley chosen for the formation of the reservoir, and the average annual rainfall of the district, with, if possible, the probable loss or waste arising from the re-evaporation and absorption by vegetation, &c.

To ascertain the drainage area it is sufficient to determine the *summit level* or *watershed*, i. e. the ridge surrounding the valley which marks the line at which the streams flow in opposite directions into contiguous valleys. This may be determined by a special survey, with a careful examination of an accurate chart like the Ordnance map, on which the contour of the country, brooks, &c. are plainly marked. The whole of the basin included within the watershed is termed the catchment basin. In the case of the Bann reservoirs it amounts to 3,300 statute acres in extent; in that of the Greenock reservoirs to 5,000 acres; at the Manchester waterworks to 19,000 acres.

An immense number of experiments have been made of late years on the rainfall in different parts of Europe, and with considerable success in determining the laws of rain distribution. For England the annual average rainfall amounts to about 36 in. in depth over the entire surface, distributed throughout the year as in the following table:—

Table of Mean Rainfall at London and Manchester.

Month.	Greenwich.*			Manchester.†		
	Average for 34 Years in ins.	Greatest fall in one month.	Least fall in one month.	Average for 64 years in ins.	Greatest fall in one month.	Least fall in one month.
January . . .	1·68	4·83	0·30	2·4915	5·85	0·32
February . . .	1·58	3·69	0·04	2·4190	6·56	0·44
March	1·61	3·45	0·40	2·2588	6·03	0·18
April	1·73	4·79	0·06	2·0225	4·75	0·16
May	1·96	4·16	0·50	2·3746	8·00	0·09
June	1·83	4·26	0·59	2·8483	7·05	0·20
July	2·37	6·65	0·10	3·7231	11·48	0·29
August . . .	2·40	4·65	0·07	3·5715	8·74	0·73
September . .	2·40	4·79	0·40	3·1353	9·00	0·24
October . . .	2·67	5·37	0·53	3·8404	9·00	0·60
November. . .	2·53	4·33	0·85	3·5682	7·37	0·62
December. . .	2·02	4·72	0·08	3·3088	9·50	0·07
Mean annual depth	24·781			35·5620		

* J. H. Belville.

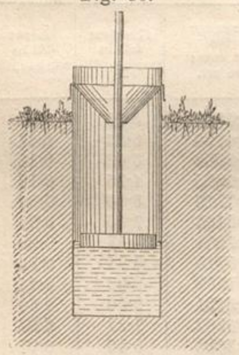
† Principally from Dr. Dalton; see Manchester Memoirs.

This would give a mean of 30 inches, but it must be borne in mind that in the lake districts and all along the west coast, there is an annual fall of rain greatly exceeding that amount, and in some places in the higher districts in Cumberland the returns have been as high as 180 to 200 inches; from this it will be seen that 36 inches is a fair average for the whole surface of Great Britain.

It is, however, important in the construction of reservoirs to have observations of the rainfall in the district in which they are to be placed. Local causes greatly influence the quantity of rain; thus the average fall in Essex is about 20 in., whilst at Keswick, in Cumberland, it is as much as 67·5 in, and at Seathwaite, in the same county, it averages the enormous quantity of 141·5 in.

The method of determining the rainfall is very simple. A cylindrical vessel, of the form shown in section in fig. 83, is placed on the ground or sunk into it in such a manner that its mouth is about 12 in. above the surface. Sometimes a permanent rod and float is added, by means of which the depth of rain received on the funnel and preserved in the vessel is read off at sight, and for ordinary purposes this is probably the best plan. But where the greatest accuracy is requisite it is necessary either to tie down the rod or to remove it altogether after making an observation, as otherwise the rain in driving obliquely impinges upon the rod instead of passing over the funnel, and a slight excess is in this way registered above the true rainfall upon the area of the vessel. It is most accurate, however, to draw off the rain and measure it in a graduated glass tube. By placing two or three of these rain-gauges at different elevations around the site of a proposed reservoir, and examining them at convenient intervals of a week or month, it is easy to estimate the exact quantity of rain which falls upon the catchment basin in the course of one year; which with certain deductions is the quantity to be

Fig. 83.



provided for in the reservoir. The precaution of placing the gauges within 5 in. or a foot of the ground is important, as, in accordance with an ill understood law, the quantity of rain rapidly decreases even at slight elevations from the ground, and it is also important to place the gauge where no artificial currents of air are created, as by the sloping side of the roof of a house. This subject was fully investigated several years ago by Mr. J. F. Bateman and a committee of the Manchester Philosophical Society. Observations had been made on and near the lines of the Ashton and Peak Forest Canals, about the accuracy of which, from their disagreement, doubts had arisen. The gauges in these observations were placed on the ridging of the roofs of the houses of the various lock-keepers, under the impression that, from the exposure of the position, all the rain which fell must there be caught. New gauges were placed in the same localities, but at the surface of the ground, and the results of these experiments were as follows:—

Locality.	Gauge on roofs.	Gauge on ground.	Excess per cent. on ground.
	in.	*in.*	
Near Middleton . . .	18·14	28·8	58·76
Near Rochdale . . .	20·50	30·3	47·8
Whiteholm Reservoir . .	22·64	35·1	55·0
Blackstone Edge . . .	23·45	34·2	45·84
Blackhouse . . .	24·89	35·9	44·23
Sowerby Bridge . . .	16·77	23·8	41·92

This enormous difference, amounting to 50 per cent. on the average, fully proves the unfitness of the roofs of houses for registering the rainfall. The upward currents of wind created by the sloping roof appear to have carried the raindrops over the edge of the gauge.

Dr. Heberden found the annual fall of rain at the top of Westminster Abbey to be 12·099 in. On the top of a house close by of much inferior altitude 18·139 in.; on the ground 22·608 in.

Mr. Phillips, at York, found the total fall for three years at an altitude of 213 feet to be 38·972 in.; at 44 feet, 52·169 in.; and on ground 65·430 in.

Notwithstanding the explanations of these facts which have been offered, Sir J. F. W. Herschel has within the last year asserted that the cause is yet to seek. The raindrops certainly appear to increase in size in the moist lower strata of the atmosphere.

Mr. Phillips's explanation has been accepted by some Meteorologists, that this augmentation is caused by the deposition of moisture on the surface of the drop, in consequence of its temperature being lower than that of the moist strata of air through which it passes. But this does not appear to be consistent with the fact, that in the condensation of vapour a large amount of latent heat would be liberated. Mr. Baxendale, who pointed this out, estimates from Professor Phillips's observations that in the condensation of the amount of water which corresponds to the augmentation of the rain drop in a fall of 213 feet, sufficient heat would be liberated to raise the temperature of the drop to 434° F.

The quantity of rain which falls in twenty-four hours, is about 1 in. at the maximum in average districts in England, although in the remarkably exceptional district in Borrowdale, already alluded to, 6·7 in. have been known to fall in the same period. The western coasts generally receive a larger proportion of water than other districts. Mountainous districts in this country, to an elevation of 2000 feet, receive a larger proportion of rain than lowlands. According to the late Dr. Miller there fell in twenty-one months in the lake district:—

In the valley, 160 feet above the sea	. . .	170·55 inches.
Styehead, 1290	„	185·74 „
Scatoller, 1334	„	180·28 „
Sparkling Tarn, 1900	„	207·91 „
Great Gable, 2925	„	136·98 „
Scawfell, 3166	„	128·15 „

Mr. Bateman's observations agree with these results, in proving the increase of rainfall corresponding to increased elevation*,

* The increase of rainfall in passing from the valley to the mountain must be carefully distinguished from the decrease as we ascend upward into the atmosphere, as shown in Mr. Phillips's observations.

as shown by the following figures, representing the rainfall near Glossop in one year:—

Westerly foot of hills, 500 feet above the sea . .	45·0 inches.
„ edge of table-land, 1500 „	67·8 „
Easterly edge of Kinderscout, 1600 „	77·45 „
„ foot of hills	40·85 „

After having determined from these considerations the quantity of water annually falling on the drainage district of a proposed reservoir, it is necessary in the next place to ascertain the probable loss from evaporation and other causes during the transmission to the reservoir. The numerous experiments on evaporation made upon small surfaces of water and of earth may be dismissed as having afforded too inconsistent results to be of any practical value.* Dr. Dalton's experiments are accurate and valuable as far as they go, but they are deficient in points of application to practical investigations. The area from which evaporation takes place is identical neither with the area of the catchment basin nor with the reservoir surface; but is a variable quantity depending on the season, the climate and the locality. It appears to me that the evaporation from a surface of water in low flat land charged with moisture, or a level vegetated surface, is very different from the evaporation in mountainous districts where there are precipitous descents to the brooks. In the former case the waters are retained and remain for weeks more or less exposed to the solar rays and the drying influences of wind. In the latter the rain pours in torrents down the barren hill-sides, and is launched into the valley where the principal evaporation takes place upon a very limited area of surface.

So also in tropical countries; the evaporation from a surface of water is greater than the rainfall upon the same surface, but then the rain falls in torrents, and is rapidly carried away to its

* Dr. Dalton gives the annual evaporation from a surface of water as 25·158 inches; Dr. Dobson, 36·78 inches; Dr. Thomson, 32 inches. The above views in regard to these experiments I expressed in a report on the Bann reservoirs in 1836. Mr. Conybeare gives the evaporation from a surface of water at Greenwich Observatory 5 feet, at Bombay 8 feet, and at Calcutta 15 feet, per annum.

natural or artificial reservoirs, and then the evaporation takes place from a very small area of surface.

Since the establishment of reservoirs and the carrying out of large drainage operations, opportunities of estimating the relation of the rainfall to the discharge by rivers have been generally available, and several important experiments have been made in this way. The method of arriving at results is to ascertain the rainfall over a catchment basin the area of which is known. The whole of the water discharged by brooks, &c., is then conveyed over a rectangular weir or waste board, and the mean velocity of the current and its breadth and depth determined by observations made once or twice every day. The comparison of the amount of water discharged with the total fall will afford the data for ascertaining the amount of evaporation.

Observations of this kind were made by Mr. Bateman with great care in the years 1845, 1846, 1847, with reference to the construction of reservoirs for the supply of Manchester with water, from the Derbyshire hills beyond Staleybridge and Mottram. Gauges were placed at the bottom of the Swineshaw valley (through which flows a tributary of the Tame), and near the summit of Windyate edge, and for some time a gauge was placed midway between these places. Similar gauges were placed in Longendale valley, and the stream in each was measured two or three times a day. From these observations the following table is compiled:—

Locality.	Year.	Mean rain.	Mean discharge.	Waste or loss by evaporation.
		in.	in.	in.
Swineshaw Brook	1845	59·8*	40·70	19·10
	1846	42·6	33·24	9·36
	1847	49·3	37·10	12·20
Longendale Valley	1847	55·2	49·46	5·74

The first was a wet year, the second one of the dryest on record, the third an average year.

* The rainfall possibly somewhat too high. Manchester Memoirs, vol. ix. p. 17.

By uniting the observations at the Swineshaw and the Longendale valleys, we get the following general table of the monthly fall and flow for three years:—

Month.	Rain.	Discharge.	Difference.
	in.	*in.*	*in.*
January	2·36	2·85	−0·49
February	4·30	4·10	+0·20
March	1·70	1·30	+0·40
April	5·22	4·12	+1·10
May	6·48	4·75	+1·73
June	3·40	1·65	+1·75
July	1·52	0·99	+0·53
August	4·32	1·24	+3·08
September	7·38	5·12	+2·26
October	4·66	5·67	−1·01
November	5·48	6·25	−0·77
December	6·74	8·55	−1·81
	53·56	46·59	+6·97

In the following table I have collected the most reliable results on the relations of discharge, rainfall, and evaporation:—

District.	Year.	Area of country drained in acres.	Rain-fall in inches.	Discharge in inches.	Differences or loss by evaporation	Remarks.
Bute	1826–7	—	45·4	23·9	21·5	Dry year, Mr. Thom.
Greenock	1828	—	60·0	41·0	19·0	Mr. Thom.
Gorbals	1852	2,750	60·0	48.0	12·0	
Swineshaw Brook	1845–7	1,250	50·58	37·01	13·5	Mr. Bateman.
Rivington Pike	1847	10,000	56·5	44·0	12·5	Mr. Hawksley.
Lough Mask, Ireland	1851–2	70,000	49·34	28·59	20·75	Flat country, Mr. Betagh.

The above table shows a loss of from 12 to 20 in., or an average waste of 16 in. of rainfall arising out of re-evaporation and other causes of absorption.

"The storage requisite for equalising the supply of water between dry and wet years should be provided with a due reference to the continuance of drought, and the quantity of water which will flow off the ground: in extreme wet seasons no water should be allowed to run to waste. Experience has shown that in the regions of comparatively moderate rain in this country, the storage to effect this object should vary from 20,000 or 30,000 to 50,000 or 60,000 cubic feet for each acre of collecting ground, the smaller quantity being about sufficient for an available rainfall of perhaps 18 in. and the larger for one of about 36 to 40 in."* 80,000 cubic feet per acre of collecting ground are provided at Lough Island Reavy; 60,000 at the Gorbals reservoirs, Glasgow; 49,000 at Rivington Pike, and 34,000 at Manchester; at the last, the whole fall not being impounded.

I proceed, neglecting further details on this subject, which belongs rather to the province of the civil than the mechanical Engineer, to give an example of the carrying out of these views, of the utility and importance of reservoirs in districts abounding with waterfalls, and where mills are numerous and depending in whole or in part on a steady and regular supply of water.

In 1836 I was called upon to report upon the best means of regulating the water supply upon the river Bann, which from its excessive variations of flow was a source of great inconvenience to the manufacturers on its banks. The river Bann rises among the lofty bare summits of the Mourne mountains, in the north-east of Ireland, where there is a heavy rainfall, and in consequence devastating floods frequently poured down its channel, carrying bridges, embankments, and other obstructions before them. On the other hand, during the summer months, the ordinary supply of water was totally inadequate to the demands of the mills; whilst the flourishing state of the linen trade called for an extended application of power, in a district where steam was not available as a motive power unless at great cost. Hence, in co-operation with Mr. Bateman, the ground was surveyed, and two reservoirs erected in the upper part of the river, by which these evils were removed, and a continuous and adequate supply of water rendered available.

* Report of the British Association, "On the Supply of Water to Towns." By J. F. Bateman, C.E. 1858.

Lough Island Reavy, the site selected for the principal reservoir, was a natural lake, bounded on the north and south by land of considerable elevation, which although having a comparatively small extent of drainage (3,300 acres ultimately) was supplied by good feeders, which, united to the surplus waters of the river Muddock, would fill the reservoir at least once or twice a year. The original surface of the Lough, fig. 84, was 92½ acres in extent; on this it was proposed to raise a depth of 35 feet more water, by the aid of embankments, and to draw off at a depth of 40 feet under that height. The area thus enlarged would be 253 statute acres, and the capacity of the reservoir is 287,278,200 cubic feet.

Corbet Lough was the second site, and although at first abandoned from its proximity to the town of Banbridge, was afterwards adopted. At a small expenditure for embankments, Corbet Lough was raised 18 feet above its summer level, so as to cover 74½ acres, and to have a capacity of 46,783,440 cubic feet of water.

A third site was selected further up amongst the mountains, but at this part the works were never executed.

It is understood that 12 cubic feet of water per second falling one foot, will, in its best application on a water-wheel, afford a force equivalent to 33,000 lbs. raised one foot high per minute, or one horse power. Now supposing the reservoirs to discharge 40 cubic feet of water per second, the fall from the lowest point of outlet at Lough Island Reavy, to the tail water of the lowest mill on the Bann, being 350 feet, we have a total force of 1166 horses available for mill purposes, or in other words, the millowners will derive an average advantage of 3·3 horse power for every foot of fall. This, it must be observed, is not a supposed quantity, but the result of certain data, taken by calculation from the waters of the Bann. It must be noticed further that this supply of 2400 cubic feet per minute is not the whole power. The calculations are for one half, the river supplying the remainder, except in extremely low water, when the demands from the reservoir may be increased to meet the emergency.

From the estimates made at the time, the expenditure to secure this result would be

	£	s.	d.
For Lough Island Reavy	12,600	0	0
„ Corbet Lough	3,512	0	0

At Lough Island Reavy it was necessary to construct four embankments, marked A, B, C and D, in fig. 84.

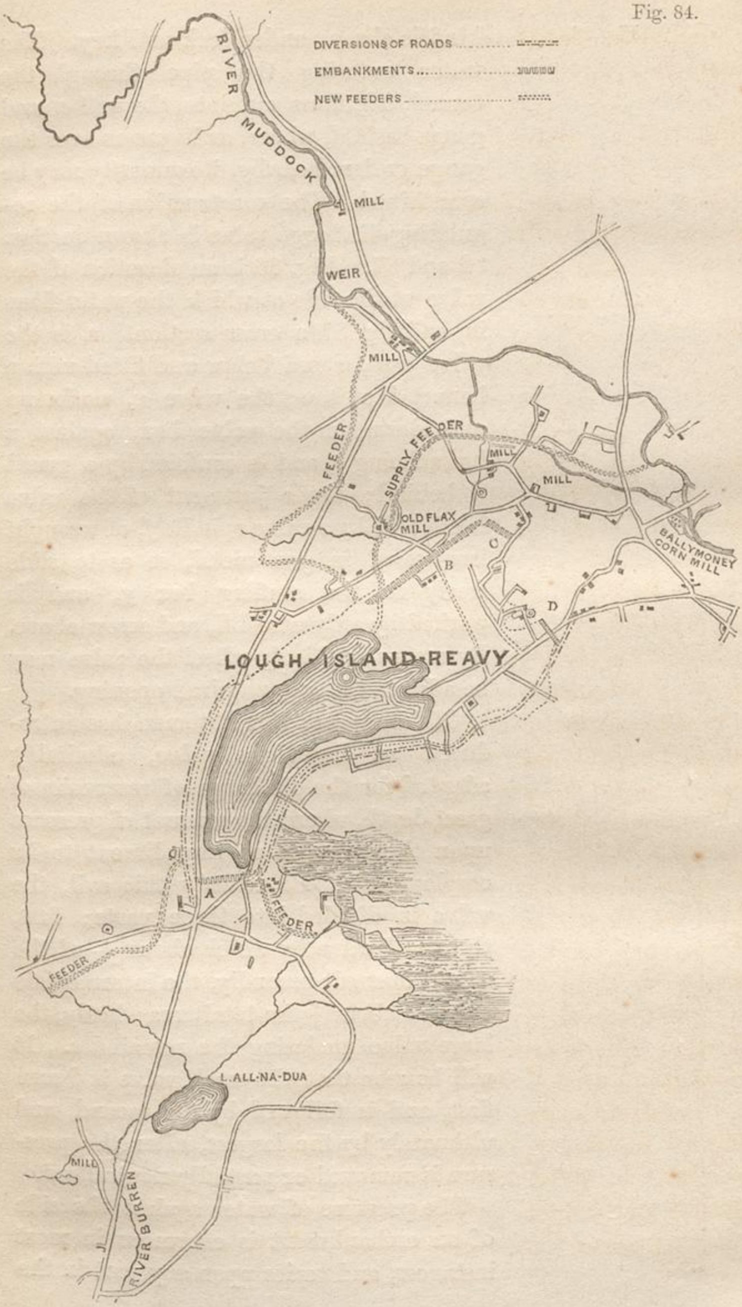
The principal S.W. side . . .	137,400	cubic yards.
Small do. . . .	17,400	„
„ N.W. end . . .	5,200	„
„ E. end . . .	99,781	„
Total .	259,781	„

The substratum of the valley being water-tight, the footing for the puddle was easily obtained by sinking a trench into the water-tight stratum, whence the puddle wall was carried up vertically with the bank to the required height. It was 12 feet in width, at 40 feet below the top, diminishing to 8 feet wide at the summit. A layer of peat was brought up on the inside of the puddle, and a similar layer on the face of the slope. Above the peat a layer of three feet of gravel was laid, and on that the stone pitching forming the inner side of the bank. The inner slopes of the embankments were $2\frac{1}{2}$ horizontal to 1 vertical, and 3 horizontal to 1 vertical. The outer slopes 2 horizontal to 1 vertical, and $2\frac{1}{2}$ horizontal to 1 vertical. The discharge pipes, two in number, each 18 in. in diameter, were placed at the bottom of a stone culvert, at the lowest part of the embankment, with suitable discharge valves, &c. The rainfall for the district amounted to from 72 to 74 in. annually, of which at least 48 in. found its way to the reservoirs.

Fig. 84 is a plan of the original disposition of Lough Reavy and its feeders. The original area of the lake is shaded, and its present area is indicated by the dotted line connecting the embankments A, B, C and D. The diversions of roads and new feeders rendered necessary are also indicated.

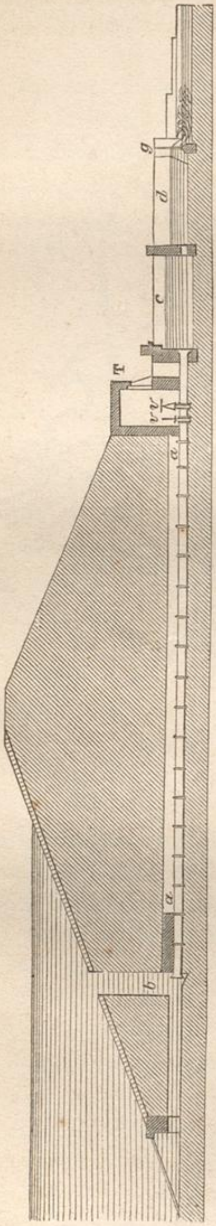
Fig. 85 represents a section of the embankment of the Belmont reservoir, which will sufficiently explain the arrangement of the culvert and discharge pipe, *a a*, with the stop and discharge valves *v v*, in the valve house T, which in works of this

Fig. 84.



Map of Lough Island Reavy Reservoir.

Fig. 85.



kind is always under lock and key. The water entering the pipe through the tunnel, *b b*, flows out into the well *c*, and gauge basin *d*, where, as it passes over the gauge or dam board *g*, its quantity may be ascertained. The construction of the regulating discharge valve is shown in figs. 86 and 87. Fig. 86 is an elevation of the valve at the side at which the water flows in, and fig. 87 a cross section. A is the valve case, closed below and fitted with a bonnet *b* at top; the valve *v*, works up and down in the valve box, against a brass facing *c*, and is confined by a guard *d* behind; the adjustment of the valve is effected by a valve spindle *f*, of wrought iron, cased in gun metal, so as to slide freely in the stuffing box *g*, and is worked by a fly-wheel *h*, and screw above. By means of this fly-wheel the valve may be adjusted to any required opening.

Of late years Mr. Bateman has introduced an ingenious valve, admirably adapted for the discharge of reservoirs of great depth, where the amount of pressure upon the valve is an impediment to its employment. To remedy this evil, the valve is divided into three parts; first, the small valve by which about $\frac{1}{10}$ of the area is opened; secondly, the intermediate valve of about $\frac{1}{4}$ the total area; lastly, the large valve unclosing the remainder. It will be seen that the small valve is drawn first, and is followed by the second, and ultimately by the largest, after the pressure is removed or partially neutralised.

The pressure of water against the side of an embankment is enormous in most instances, and varies upon any part in the ratio of its depth below the surface. Let h = the depth of water in a reservoir; A = the area in square feet of a vertical

Fig. 86.

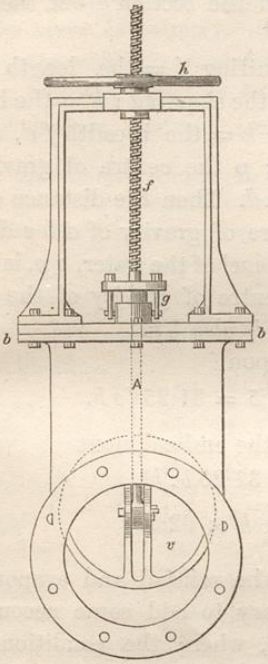
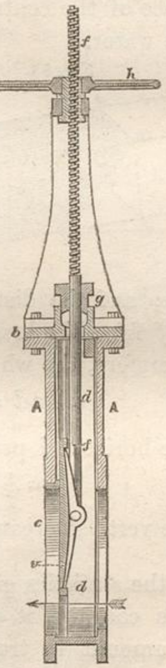


Fig. 87.



section of the embankment of the depth h; then the lateral pressure upon the embankment in a horizontal direction is, in lbs.

$$P = \tfrac{1}{2}\, h \times 62\tfrac{1}{2} \times A = 31{\cdot}25\, A \,.\, h,$$

a cubic foot of water weighing $62\frac{1}{2}$ lbs.

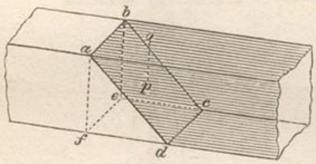
Or, generally, the whole pressure of water upon a submerged plane surface is equivalent to the area of the surface, multiplied by the weight per cubic unit of the fluid, and by the head of water measured from the centre of gravity of the submerged surface. That is, for water

$$P = 62\tfrac{1}{2} \,.\, A_1 \,.\, h_1.$$

Where h_1 = the depth of the centre of gravity below the level of the water in feet; A_1 the area of the surface in feet; P = the pressure on the surface in lbs.

And the whole pressure in any one direction is equal to the area of a section of the fluid vertical to that direction, multiplied by the weight of a cubic unit of the fluid and by the distance of the centre of gravity of the section from the level of the water.

Fig. 88.



Putting $l =$ the length $a\,b$; $s =$ the slope $a\,d$; $h =$ the height $a\,f$; $k =$ the breadth $f\,d$, all in feet; p the centre of gravity of $a\,b\,c\,d$. Then the distance of the centre of gravity of $a\,b\,c\,d$ from the level of the water, $o\,p$, is equal to $\frac{1}{2}\,h$; and the distance of the centre of gravity of the plane $a\,b\,e\,f$ from the level of the water is also $\frac{1}{2}\,h$.

Therefore, the whole pressure upon

$$a\,b\,c\,d = \tfrac{1}{2}\,l\,s\,h \times 62{\cdot}5 = 31{\cdot}25\,l\,s\,h.$$

The horizontal pressure against the embankment

$$= \tfrac{1}{2}\,l\,.\,h^2 \times 62{\cdot}5 = 31{\cdot}25\,l\,.\,h^2.$$

The vertical pressure $= l \times k \times h \times 62{\cdot}5$.

To the statistics given above of the rainfall and evaporation in this country, it will be necessary to add some account of their amount in tropical climates, where the conditions are essentially different. In such climates for three quarters of the year the rain never falls, and the whole quantity for the annual consumption falls during the remaining quarter.

At the Bombay Water Works constructed by Mr. Conybeare, the annual rainfall is 124 inches, of which $\frac{6}{10}$ are assumed to be available for storeage. The area draining into the basin is 3948 acres, so that the supply is upwards of 6,600,000,000 gallons. The storeage capacity of the reservoir is 10,800,000,000 gallons, or 1,733,000,000 cubic feet.*

At the Melbourne Water Works constructed under the direction of Mr. Matthew Bullock Jackson, the area of the reservoir when full is 1303 acres, greatest depth 25 feet 6 inches, average depth 18 feet, and capacity 6,400,000,000 gallons. The area of the natural Catchwater basin is 4650 acres, together with

* Minutes of Proceedings of Institute of Civil Engineers, vol. xvii. p. 560.

600 acres drained by a water course. This area, however, may be increased if a larger supply is necessary. This watercourse at the same time opens a connection with the River Plenty, through which flows the water drained from an extent of 40,000 acres of country. This watercourse is opened during the winter to fill the reservoir from this source. The following table gives the detail of the rainfall and evaporation observed by Mr. Jackson during the construction of the works:—

TABLE, SHOWING THE AMOUNT OF SPONTANEOUS EVAPORATION AND RAINFALL FOR TWELVE MONTHS ENDING 31ST JANUARY 1858.

Months.	Rainfall at				Spontaneous Evaporation at Melbourne.
	Melbourne, 94½ feet above the level of the sea.	Yan Yean.	Geelong, 96 feet above the level of the sea.	Ballarat, 1438 feet above the level of the sea.	
	inches.	*inches.*	*inches.*	*inches.*	*inches.*
February . .	3·98	1·33	2·39	0·23	8·14
March . .	3·80	3·61	1·99	3·75	5·10
April . . .	0·99	0·78	1·07	1·55	4·25
May . . .	2·00	2·05	1·72	1·85	1·97
June . . .	1·99	1·89	1·58	0·00	1·50
July . . .	1·16	2·39	1·15	0·00	1·86
August . .	1·69	2·42	1·14	2·42	2·57
September . .	3·83	3·70	3·19	2·68	3·76
October . .	5·28	4·70	2·63	4·63	4·23
November . .	2·12	1·80	3·15	2·27	5·84
December . .	0·83	1·76	0·33	0·73	11·01
January . .	0·88	1·07	0·00	0·00	11·23
Totals . .	28·55	27·50	20·34	20·11	61·46

It is to be presumed that the evaporation given above as nearly three times the rainfall is the evaporation from a surface of water such as that of the reservoir itself. The rain, however, is collected from a surface thirty-five times as great as that of the reservoir when at its maximum height.

Weirs or *Dams*, thrown across the beds of rivers, have always been employed in order to raise the head of water in the river bed, and to divert a portion of it for the purposes of the mill. We have now to consider how most economically to secure a sufficient fall, and to protect the dam from the destructive effects of floods.

There is hardly any department of engineering which requires more careful consideration than that of forming barriers to large quantities of moving water; and when the nature of rivers carrying off the drainage from a large area is considered, and the enormous power of suddenly accumulating floods, the nature of the resistance required from a dam may be easily conceived, and when all the care of the engineer has been exercised, it nevertheless sometimes occurs that the torrents tear up and destroy in a night the work which was intended to perform the quiet industrial duties of a mill for ages, leaving, in place of the well turned arch across the stream, only the horns of the abutments and an indistinguishable mass of rubbish mingled with the mountain debris of the flood.

Such is frequently the case with weir constructions, particularly those across the rapids of mountain torrents, and this not unfrequently causes the construction of a temporary dyke of boulder stones capable of withstanding the ordinary action of the river, and easily replaced when floods have caused its partial destruction. This description of weir is carried diagonally across the stream at *a* (fig. 89), and being considerably longer than its breadth forces part of the water into the conduit *b*, and passes the remainder over the top in a thin sheet, which does little or no damage to the banks below. In the above description of weir it seldom happens that much fall can be obtained, and they are

Fig. 89.

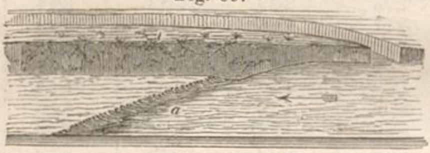
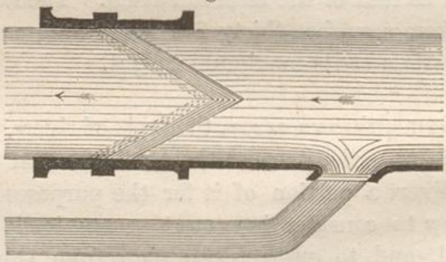


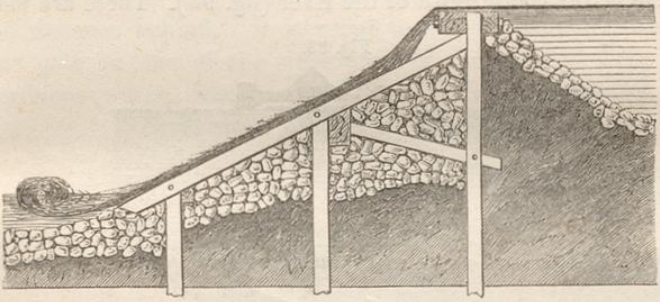
Fig. 90.



therefore adopted where there is a large supply of water employed upon an undershot wheel.

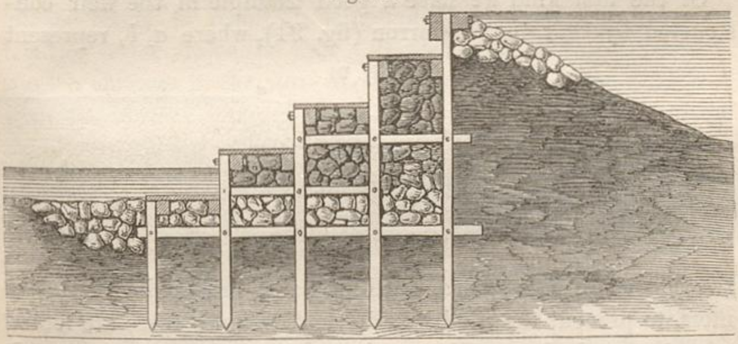
Another description of weir, which is generally employed on moderate sized rivers, is the V form constructed across the bed of the river, as shown in fig. 90, in plan. The object of adopting this form of weir is to increase its resisting powers, and by spreading the fall of water over a large surface, to diminish its destructive effects upon the apron below; the descending currents meeting in the angle of the V neutralise their effects on the foundations, and do less injury to the banks on either side. This weir is generally formed of piles (fig. 91), with an open

Fig. 91.



frame of timber, into which are inserted large boulder stones, forming a compact mass of boulder sheeting resting on gravel, and nearly impervious to water. Another weir, preferred to most others where timber is plentiful, is formed into a series of

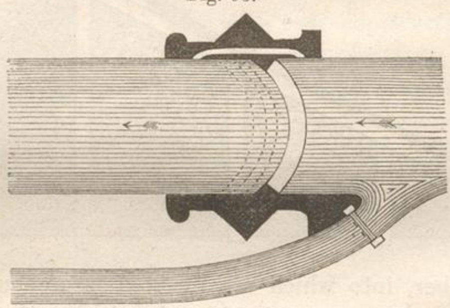
Fig. 92.



steps (fig. 92), over which the water falls in cascades, which destroys its injurious effect on the foundations; it is composed of piles placed at right angles with the direction of the stream, and placed in rows properly stayed and covered with planking firmly nailed to the horizontal and vertical timbers. When it is necessary to have the structure watertight a line of sheet piling is usually driven in, in the line of the weir across the whole breadth of the stream, and these again, supported by foot piles and stays at different distances, form a perfectly tight and very durable weir.

The most perfect weirs, however, are formed of stone, built of solid ashlar, and usually forming part of the segment of a circle across the breadth of the river (fig. 93). These are made,

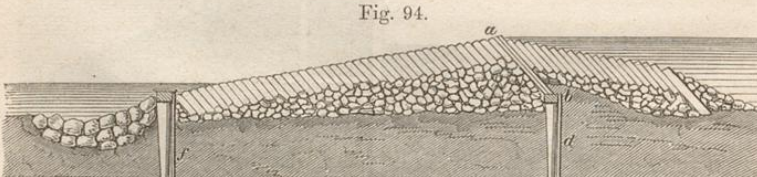
Fig. 93.



1st, with long inclined slopes on either side; 2nd, solid, with nearly perpendicular walls; or 3rd, with a curved apron to break the force of the fall.

Of the first kind we have a good example in the weir constructed by Smeaton at Carron (fig. 94), where *a*, *b*, represent

Fig. 94.

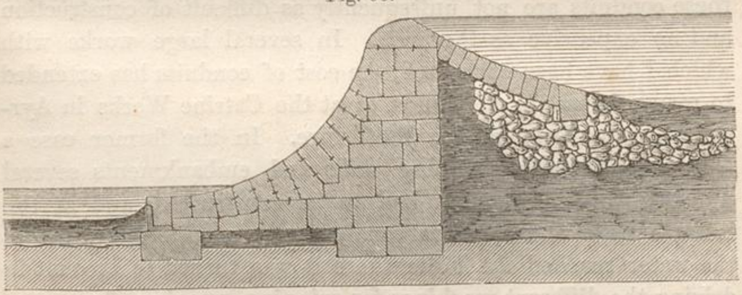


two courses of flag stones, breaking joint, and packed with live moss, to prevent the silt being driven through; these are footed upon grooved sheet piling with bearing piles and stringer *d*, the flags being supported on rubble; at the foot of the dam is another row of sheet piling *f*, similarly supported and protected by a fir plank at top from the action of the water. Over the rubble is placed a row of regular stones, laid endways so as to be perfectly secure from derangements by floods.

The second description of stone weir is a solid ashlar wall having its convex side to the current (fig. 93) and abutting upon heavy masses of masonry on each side of the stream. Fig. 95, exhibits this weir modified by having a curved apron, so as gradually to convert the vertical fall of the water into a horizontal flow in the direction of the stream.

It will suffice to observe further, that the head of water immediately over the crest is less than the head of water at some distance behind. It is usual to cut a channel, with a sluice gate in one of the wing walls of the weir, to draw off superfluous water, when requisite. The utmost caution is

Fig. 95.



needed, both in observing the conditions of the river and the effects likely to result in times of flood from the increased head of water above the weir. Rapid rising of the waters and sudden changes in the state of the river are too often neglected with disastrous consequences to works of this kind, just on the eve of completion, or to the lands above the dam in consequence of flooding caused by the obstruction of the dam. In cases where this last danger is apprehended, a self-acting dam has sometimes been employed, consisting of a massive frame of planks carried across the river and attached by hinges to the crest of the dam. This plank is maintained in a vertical position in ordinary conditions of flow by balance weights attached or hung over wheels upon the wing walls, so as to retain the maximum desirable head of water. In floods the increased pressure of the overflowing water overcomes the balance weights and throws down the plank into a horizontal position, opening a free passage for the water.

Conduits. — Having thus considered the means of accumulating water power and regulating its supply by means of reservoirs and weirs, we have yet to consider the formation of conduits or lades, as they are called in some places, for the actual discharge of the water upon the water wheel or other machine by which its power is to be utilised. By the construction of a weir we may have dammed back the water half a mile or a mile, and formed the upper part of the stream into a reserve from which the supply of water can be drawn and two or three feet or more of fall gained; but unless the mill is built close up to the banks of the stream head courses, canals and tail races have to be cut in order to make the fall available, and these conduits are not unfrequently as difficult of construction and as expensive as the weir. In several large works with which I have been connected, the cost of conduits has extended to many thousands of pounds, as at the Catrine Works in Ayrshire, or the Deanston in Perthshire. In the former case a large tunnel, with retaining walls and embankments several hundred yards in extent, had to be constructed, and at the latter a wide and spacious canal, nearly a mile long, before the water reached the mills where it was turned to account in driving the different machines for spinning, weaving, &c.

The large expenditure in these and similar works, operates much against the economy of water power, and when the extremes of floods and droughts, including the interest of capital sunk, is considered, it will be seen that it frequently happens that steam power might have been purchased and maintained at as economical a rate. Let us take, for example, the Catrine Mills, at which there is a fall of forty-eight feet, and a power of 200 horses, nearly constant throughout the year. In this establishment there are two colossal water wheels, each fifty feet in diameter and twelve feet wide. Now taking the weir, the tunnel, the upper conduit, tail race, &c., arched to a distance of a third of a mile down the river, we may estimate the ultimate cost, approximately, as follows:—

Water privileges and land	£4,000
Cost of weir	1,000
Head race, tunnel, and canal	3,000
Archways, cisterns, sluices, &c.	1,000
Wheelhouse and foundations	1,500
Tail-race	1,500
Water-wheels and erection	4,500
Contingencies	1,500
Total	£18,000

The cost of power independent of mill-work equivalent to an annual rental for interest of capital, repairs, and wear and tear, at 7 per cent., amounting to 1260*l.*

This may be contrasted with steam power in a district where coal can be purchased at 7*s.* per ton, and we have,

Cost of engines of 100 nominal horse power	£4,000
Engine-house, foundations	1,500
Contingencies	500
Total	£6,000

This at 10 per cent. for interest of capital, repairs, and renewals, will be equivalent to

An annual rental of	£600
Add consumption of coal at 4lbs. per indicated horse power per hour, engineers' wages, &c.	900
Total	£1,400

Against the higher rental in the case of steam, must be set the cost of transit of the raw material and products of the mill, which must be transported to and from the market at a greatly increased cost, as in the case of the Catrine works, with the risk of stoppage also from want of water in long continued drought or frost. It is true that labour may be had cheaper in the country than in towns, but that is no counterpoise for want of skill amongst the operatives, or for the loss of those numerous conveniences which are to be obtained in the great foci of labour where the whole powers and energy of the country have been concentrated.

On the whole, there appears (in the present improved state of the steam engine and the price of coal) to be no advantage in this country in water power as applied to manufactures, and it is only at out districts, and where the mere wants of the inhabitants have to be supplied, that water mills can be used with profit. Before the introduction of the steam engine, water power was invaluable, but we now see that it cannot at all times be depended upon, and that in most cases where a large amount of power is required, the chief source from which it must be derived is steam.

CHAP. II.

ON THE FLOW AND DISCHARGE OF WATER, AND THE ESTIMATION OF WATER POWER.

In the present chapter it is proposed to enter only so far into those questions of Hydrodynamics which relate to the measurement of the discharge of water, and the estimation of water power, as it is necessary they should be understood by the practical millwright, in order that he may be at no loss in comparing the efficiency of various forms of water machinery, calculating their power, and proportioning them to their position and their work. For minute and accurate mathematical investigations, the reader is referred to special treatises on hydrodynamics, in which the subject is treated from another point of view.

From the nature of a non-elastic fluid such as water, in which the particles are free to move over one another without friction, the following relations hold between pressure, velocity and discharge.

1st. The *pressure* P upon a unit of area at the depth h beneath a fluid surface is equal to the weight of a column of the liquid h units high; that is, if W be the weight of a unit of volume,

$$\text{P} = \text{W}\, h \ldots (1).$$

And therefore the pressure on a horizontal surface of a units area $= \text{W}\, h\, a$.

2nd. The *velocity* with which a fluid flows from a small orifice at the depth h beneath the surface, is the same as the velocity it would have acquired in falling freely the same distance under the action of gravity, if we neglect those causes of retardation to be considered presently. If we take $v =$ mean velocity of the effluent water; $h =$ mean depth of orifice beneath the surface, or in other words the *head* of fluid; $g = 32{\cdot}1908 =$ the velocity generated in a falling body in one second, then by the laws of accelerating motion,

$$v = \sqrt{2gh}*, \ldots (2)$$

that is, the theoretical velocity of effluent water is equal to the square root of 64·38 times the mean head; understanding by mean head the head measured from the centre of the orifice. Thus we have in the following table the theoretical velocity at various heads.

TABLE I.—THEORETICAL VELOCITY OF EFFLUENT WATER.

Head.	Velocity per second.	Head.	Velocity per second.	Head.	Velocity per second.	Head.	Velocity per second.	Head.	Velocity per second.
ft.	*ft.*	*ft.*	*ft.*	*ft.*	*ft.*	*ft.*	*ft.*	*ft.*	*ft.*
1	8·02	11	26·6	21	36·8	31	44·6	41	51·3
2	11·34	12	27·8	22	37·6	32	45·4	42	52·0
3	13·90	13	28·9	23	38·5	33	46·1	43	52·6
4	16·04	14	30·0	24	39·3	34	46·8	44	53·2
5	17·93	15	31·1	25	40·1	35	47·4	45	53·8
6	19·64	16	32·1	26	40·9	36	48·1	46	54·4
7	21·21	17	33·1	27	41·7	37	48·8	47	55·0
8	22·68	18	34·0	28	42·4	38	49·4	48	55·6
9	24·09	19	34·9	29	43·2	39	50·1	49	56·1
10	25·38	20	35·9	30	43·9	40	50·7	50	56·7

3rd. The *quantity* of water which issues from an orifice at a depth h beneath the surface of a fluid is equal to the area of the orifice multiplied by the velocity of the effluent water, that is neglecting the diminution from the *venâ contractâ* to be mentioned shortly.

Let Q = units of volume discharged per second
a = area of orifice
v = velocity of effluent water
$Q = av = a\sqrt{2gh} \ldots (3)$.

And in t seconds $Qt = avt$ will be discharged.

Where Q is called the *theoretical discharge*, and is found by multiplying the area of the orifice in feet by the velocity of the effluent water in feet per second found as above.

4th. If the orifice instead of opening freely into the air as supposed above, opens into another reservoir of fluid, we must substitute in the above equations the difference of level of fluid

* Or in feet $v = 8{\cdot}03\sqrt{h}$.

in the two reservoirs for the head above the centre of the orifice. Let h' be the head above the centre of the orifice in the higher reservoir and h'' in the lower, then the effective head $h = h' - h''$ ... (4).

5th. If the water escape by a rectangular *notch* instead of an orifice, that is an aperture such that the upper level surface of the water does not come in contact with the sides of the vessel, falling freely in the air, the theoretical discharge is two thirds of the area of the effluent vein multiplied by the velocity of efflux, or more accurately if h = head of water, b = *breadth of notch*

$$Q = \tfrac{2}{3}\, b \,.\, h \,.\, \sqrt{2\, g\, h} \;\ldots\; (5).$$

We must next examine certain properties of fluid motion which cause the *actual* or *effective discharge* to differ materially from the theoretical discharge given in the above equations, although in a constant ratio, so that the one may always be calculated from the other.

1st. *Thick-lipped orifices* or mouth-pieces. For smooth orifices, the length of which is about twice or three times the smallest diameter, the *actual* does not widely differ from the *theoretical* discharge. The velocity of the effluent current is however never so great as that in equation (2), but is diminished for a constant ratio for each kind of orifice, and the discharge is less in the same proportion. For a simple cylin-

Fig. 96.

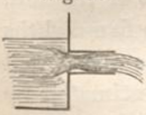


Fig. 97.

Fig. 98.

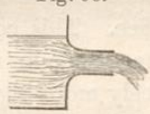
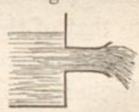


Fig. 99.



drical tube, fig. 94, of about $1\frac{1}{2}$ diameters in length, the velocity of the effluent water is equal to $0{\cdot}8\, v = 0{\cdot}8 \sqrt{2\, g\, h}$ and the actual discharge $= a \times 0{\cdot}8 \times v = 0{\cdot}8$ Q. Where the interior angle of the tube is rounded, as in fig. 96, the velocity amounts to as much as $0{\cdot}96\, v$ to $0{\cdot}98\, v$, and hence the discharge to $0{\cdot}96$ Q to $0{\cdot}98$ Q, where Q as before is the theoretical discharge given by the above formulæ. This constant ratio is called the coefficient of velocity.

Hence we have this rule for determining the quantity of water discharged by a thick-lipped orifice; seek first in Table I. the velocity corresponding to the given head of water, measured from the centre of the orifice; multiply this velocity by the area of the orifice in square feet, and the product will be the theoretical discharge in cubic feet per second. For the actual discharge, this must be multiplied by 0·7, if the orifice be of the form of fig. 97; by 0·8, if the orifice be of the form of fig. 96, and by 0·97 if it be of the form of fig. 98. The importance of the form of orifice is manifest, and hence a trumpet mouth * should be employed in all water pipes, wherein a maximum discharge is desirable, the quantity being increased which ever way the trumpet mouth is turned, whether as in fig. 98 or 99, but most in the former case. For conical converging tubes, d'Aubuisson found the coefficient of efflux to vary from 0·829 to 0·946 as the lateral convergence increased from 0° 0′ to 13° 24′, and from 0·946 to 0·847 as the convergence increased from 13° 24′ to 48° 50′; the area of the orifice being measured at the small extremity. For tubes which at first converge and then diverge, so as to take the form of the fluid vein, the coefficient of discharge is 1·55, that is, of course, taking the minimum area of the tube.

2nd. *Thin-lipped orifices*, the fluid escaping freely into the air. With orifices of this nature, the fluid vein contracts very remarkably at a short distance beyond the orifice, and the discharge is diminished in the ratio of the least area of the vein to that of the orifice. This contraction amounts to five-eighths of the area of the orifice in most cases, and hence the actual discharge is scarcely more than five-eighths that estimated by equation (3). Putting $m =$ the coefficient of contraction, we have the actual discharge from an orifice $= m\,\mathrm{Q} = m\,a\,\sqrt{2\,g\,h}$.

The velocity of the effluent vein is also diminished in a slight degree, perhaps by three or five per cent., but it will be most convenient to combine the coefficients of contraction and velocity together, and to call m the coefficient of discharge or ratio of actual to theoretical discharge.

* As for instance in the reservoir drawing, fig. 84.

A very large number of experiments have been made upon the values of the co-efficient *m* for various forms of orifices, the most important of which we owe to Michelotti, Castel, Bidone, Bossut, Rennie and others. But by far the most important and complete are those conducted by MM. Poncelet and Lesbros under the auspices of the French government, and all interested in hydraulic investigations must feel indebted to them for the skill, perseverance and accuracy with which they have registered so large a body of results. These determinations go to show that the value of the coefficient of discharge ranges between 0·58 and 0·7 *, being greater for small orifices and small velocities and less for large orifices and high velocities.† For heads of three and four feet and upwards, the coefficient of discharge may be taken at 0·6.

Mr. Rennie's results give the following values of *m*.

	Head of 4 feet.	Head of 1 foot.
Circular orifices . . .	0·621	0·645
Triangular orifices . . .	0·593	0·596
Rectangular orifices . . .	0·593	0·616

For more accurate calculations I have abridged the following tables of M. Poncelet's results from the "Aide-Memoire" of M. Morin, reducing the measures to the English standard.

TABLE II. — COEFFICIENTS OF DISCHARGE OF VERTICAL RECTANGULAR ORIFICES, THIN-LIPPED, WITH COMPLETE CONTRACTION. THE HEADS OF WATER MEASURED AT A POINT OF THE RESERVOIR WHERE THE LIQUID WAS PERFECTLY STAGNANT.

Head or summit of orifice. ins.	Coefficients of discharge for orifices of a height of					
	7·9 ins.	3·9 ins.	1·9 ins.	1·18 ins.	0·78 ins.	0·39 ins.
0·79	0·572	0·596	0·615	0·634	0·659	0·694
1·9	0·585	0·605	0·625	0·640	0·658	0·679
3·9	0·592	0·611	0·630	0·637	0·654	0·666
7·9	0·598	0·615	0·630	0·633	0·648	0·655
11·8	0·600	0·616	0·629	0·632	0·644	0·650
15·7	0·602	0·617	0·628	0·631	0·542	0·647
39·4	0·605	0·615	0·626	0·628	0·633	0·632
59·1	0·602	0·611	0·620	0·620	0·619	0·615
78·7	0·601	0·607	0·613	0·612	0·612	0·611
118·1	0·601	0·603	0·606	0·608	0·610	0·609

* Rankine. † Weisbach.

TABLE III. — COEFFICIENTS OF DISCHARGE OF VERTICAL, THIN-LIPPED, RECTANGULAR ORIFICES, WITH COMPLETE CONTRACTION. THE HEADS OF WATER MEASURED IMMEDIATELY OVER THE ORIFICE.

Heads or summit of orifice in ins.	Coefficient of discharge for orifices of a height of					
	7·9 ins.	3·9 ins.	1·9 ins.	1·18 ins.	0·78 ins.	0·39 ins.
0·78	0·594	0·614	0·638	0·668	0·697	0·729
1·97	0·593	0·614	0·636	0·651	0·672	0·686
3·94	0·595	9·614	0·634	0·640	0·657	0·669
7·87	0·599	0·615	0·630	0·633	0·649	0·656
11·81	0·601	0·616	0·629	0·632	0·644	0·651
15·74	0·600	0·616	0·630	0·632	0·646	0·653
39·37	0·605	0·615	0·626	0·628	0·633	0·632
59·05	0·602	0·611	0·620	0·620	0·619	0·615
78·74	0·601	0·607	0·614	0·612	0·612	0·611
118·11	0·601	0·603	0·606	0·608	0·610	0·609

Thence we derive this rule for estimating the discharge of water from thin-lipped orifices; seek in Table I. the velocity corresponding to the head of water measured from the centre of the effluent vein; multiply this by the area of the orifice in square feet, and the product is the theoretical discharge. Five eighths of the theoretical discharge will give the actual discharge in cubic feet a second, if a rough approximation only is required. If the estimation is to be accurate, seek in Tables II. and III. the coefficient of discharge most nearly corresponding to the given head and area of orifice, and multiply the theoretical discharge by the coefficient so found. Thus the following table has been calculated.

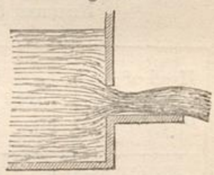
TABLE IV. — THEORETICAL AND ACTUAL DISCHARGE FROM A THIN-LIPPED ORIFICE OF A SECTIONAL AREA OF ONE SQUARE FOOT.

h Head.	v Table I.	m Table II.	$a \times v$ Theoretical Discharge per second.	$m\,a\,v$ Actual discharge per second.	h Head.	v Table I.	m Table II.	$a \times v$ Theoretical discharge per second.	$m\,a\,v$ Actual discharge per second.
ft.	*ft.*	*ft.*	*ft.*	*cub. ft.*	*ft.*	*ft.*		*cub. ft.*	*cub. ft.*
1	8·02	0·60	8·02	4·812	11	26·6	0·60	26·6	15·96
2	11·34	0.60	11·34	6·804	12	27·8	0·60	27·8	16·68
3	13·90	0·60	13·90	8·340	13	28·9	0·60	28·9	17·34
4	16·04	0·60	16·04	9·624	14	30·0	0·60	30·0	18·00
5	17·93	0·60	17·93	10·75	15	31·1	0·60	31·1	18·66
6	19·64	0·60	19·64	11·78	16	32·1	0·60	32·1	19·26
7	21·21	0·60	21·21	12·73	17	33·1	0·60	33·1	19·86
8	22·68	0·60	22·68	13·61	18	34·0	0·60	34·0	20·40
9	24·10	0·60	24·10	14·46	19	34·9	0·60	34·9	20·94
10	25·40	0·60	25·40	15·24	20	35·0	0·60	35·9	21·54

The above table is given as a sample of the method of calculation according to the above rule. For other areas and different heads the calculation may very easily be performed.

3rd. *Discharge with incomplete contraction.*— It is very frequently the case in practice that one of the sides of a thin-lipped orifice is prolonged, so that the vein of fluid no longer contracts upon all sides, as in fig. 99A. In this case the coefficients in Tables II. and III. give too low a result. M. Morin gives the following rule for discharge with incomplete contraction: —

Fig. 99A.



Multiply the coefficient of discharge for complete contraction found as above by

1·035	when the vein contracts on	3 sides	only
1·072	„ „	2 sides	„
1·125	„ „	1 side	„

in order to obtain the true coefficient by which the theoretical discharge must be multiplied to give the actual discharge. Hence, for an approximate calculation, we may multiply the theoretical discharge ($= a \times v$) by 0·63, when the orifice is prolonged upon one side; by 0·66 when it is prolonged on two sides, and by 0·69 when it is prolonged on three sides. When all four sides are prolonged, the thick-lipped orifice, fig. 94, is formed of which the coefficient of efflux is 0·8.

4th. *Discharge from rectangular notches, waste-boards, and weirs.* — In this case the theoretical discharge =

$$\left.\begin{aligned} Q &= \tfrac{2}{3}\, b \,.\, h \,.\, \sqrt{2\,g\,h} \\ &= \tfrac{2}{3}\, a\, v \end{aligned}\right\} \ldots\ldots(5)$$

where Q = discharge in cubic feet per second.

b = breadth of notch or weir.

h = head of water, measured at some distance behind the crest of the weir.

v = velocity in feet per second = $\sqrt{2\,g\,h}$...(2)

a = area of effluent vein = $b \times h$.

The actual discharge is found by multiplying the theoretical discharge by the coefficient of efflux m, which varies under different circumstances. The millwright must select this coefficient from the following tables, so as to suit the particular case to which it is to be applied.

TABLE V.—COEFFICIENT OF DISCHARGE FOR WEIRS, FROM EXPERIMENTS ON NOTCHES 8 INCHES BROAD, BY PONCELET AND LESBROS.

Head of Water in inches.	0·89	0·78	1·18	1·57	2·36	3·15	3·93	5·90	7·86	8·65
Coefficient of discharge $= \frac{2}{3} m$.	0·424	0·417	0·412	0·407	0·401	0·397	0·395	0·393	0·390	0·385

This gives a mean value of 0·4 or $\frac{2}{5}$ths for $\frac{2}{3} m$, and hence we may approximately find the discharge from a waste board by multiplying the head in feet by the breadth of the notch in feet, and by the velocity due to the head found by equation (2) or Table I. Two-fifths of this product will be the discharge in cubic feet per second.

In 1852 the council of the Institution of Civil Engineers awarded to Mr. Blackwell a premium for a valuable series of experiments on the discharge of water from weirs made on a very large scale, and with various conditions of head and with different kinds of overfall bars. What constitutes the principal value of Mr. Blackwell's paper is the scale on which the experiments were made, and their close approximation to actual practice. It must be borne in mind, however, that in calculating the quantity of water discharged in the flow of rivers over weirs, reference must be had to the form of the top, in order to ascertain the state of the overfall as compared with those in the following table, from which the coefficient is taken for calculation.

TABLE VI. — COEFFICIENTS OF DISCHARGE FROM WEIRS, FROM EXPERIMENTS BY MR. BLACKWELL.

No. of Trials.	Description of Overfalls.	Head in ins.	$\frac{2}{3}$ m.
6	Thin plate 8 feet long	1 to 3 3 to 6	·440 ·402
11	Thin plate 10 feet long	1 to 3 4 to 6 6 to 9	·501 ·435 ·370
23	Plank 2 inches thick, with notch 3 feet broad .	1 to 3 3 to 6 6 to 10	·342 ·384 ·406
56	Plank 2 inches thick, with notch 6 feet broad .	1 to 3 3 to 6 6 to 9 9 to 14	·359 ·396 ·392 ·358
40	Plank 2 inches thick, with notch 10 feet broad .	1 to 3 3 to 6 6 to 9 9 to 12	·346 ·397 ·374 ·336
4	Plank 2 ins. thick, notch 10 ft. broad, with wings	1 to 2 4 to 5	·476 ·442
7	Overfall with a crest, 3 feet wide, sloping 1 in 12, 3 feet long like a weir . . .	1 to 3 3 to 6 6 to 9	·342 ·328 ·311
9	Overfall with a crest, 3 feet wide, sloping 1 in 18, 3 feet long, like a weir . .	1 to 3 3 to 6 6 to 9	·362 ·345 ·332
6	Overfall with a crest, 3 feet wide, sloping 1 in 18 and 10 feet long . . .	1 to 4 4 to 8	·328 ·350
14	Overfall with a level crest 3 feet wide by 6 long	1 to 3 3 to 6 6 to 9	·305 ·311 ·318
15	Overfall with level crest, 6 feet long by 3 broad	3 to 7 7 to 12	·330 ·310
12	Overfall with level crest, 3 feet wide by 10 long	1 to 5 5 to 8 8 to 10	·306 ·327 ·313
61	At Chew Magna, overfall bar 10 feet long, 2 inches thick	1 to 8 3 to 6 6 to 9	·437 ·499 ·505

The most important of the generalisations from this table are—

1st. That the discharge is decreased in proportion to the breadth and inclination of the crest, being least when the crest is level.

2nd. That converging wing walls, above the overfall, increase the discharge.

Where, as in the case of a river, the water approaches the weir with a certain velocity, it should be taken separately into account, the above coefficients being deduced from experiments on reservoirs so large that the water was approximately stagnant.

Let k = height of head due to velocity v of water as it approaches the weir; that is, let $k = \frac{v^2}{64 \cdot 38}$;

the effective discharge is then =

$$Q = \tfrac{2}{3} m \,.\, b \,.\, \sqrt{2g}\, \left[(h+k)^{\frac{3}{2}} - k^{\frac{3}{2}} \right]^{*} \; \dots (6).$$

TABLE VII.—EXAMPLES OF ESTIMATION OF DISCHARGE FROM WEIRS.

Head of water.	v. Velocity due to head. Table I.	$\frac{2}{3}m$ Coefficient of discharge. Tables V. VI.	b. Breadth.	$\frac{2}{3} b \,.\, h \,.\, v$. Theoretical discharge per second.	$\frac{2}{3} m \,.\, b \,.\, h \,.\, v$. Actual discharge per second.	Remarks.
inches.				*cubic feet.*	*cubic feet.*	
1	2·32	0·412	1 ft.	1·283	·793	Thin lipped waste board.
6	5·67	0·393	1 ft.	1·890	1·114	
12	8·02	0·380	1 ft.	5·35	3·047	
6	5·67	0·350	10 ft.	18·90	9·922	Weir with crest.

According to the above formula I have computed the following table, showing at a glance the velocity in feet, the theoretical discharge in cubic feet per second, and the actual discharge of water over a thin-edged notch or weir for various heads from $\frac{1}{2}$ an inch to 6 feet—the water approaching the weir with no perceptible current.

* Weisbach.

TABLE VIII.—DISCHARGE OF WATER OVER A THIN-EDGED NOTCH OR WEIR FOR EVERY FOOT IN BREADTH OF THE STREAM IN CUBIC FEET PER SECOND.

h. Head in feet.	*v.* Velocity per second.	$\frac{2}{3}m$. Coefficient of discharge. Table V.	$\frac{2}{3}b.h.v.$ Theoretical discharge per second.	$\frac{2}{3}m.b.h.v.$ Actual discharge per second.	*h.* Head in feet.	*v.* Velocity per second.	$\frac{2}{3}m$. Coefficient of discharge.	$\frac{2}{3}b.h.v.$ Theoretical discharge per second.	$\frac{2}{3}m.b.h.v.$ Actual discharge per second.
	ft.		*cub. ft.*	*cub. ft.*		*ft.*		*cub. ft.*	*cub. ft.*
·05	1·79	0·42	·0596	·0376	2·1	11·62	0·35	16·94	8·890
·1	2·53	0·41	·168	·1037	2·2	11·90	0·35	17·44	9·163
·2	3·58	0·40	·478	·286	2·3	12·17	0·35	18·66	9·796
·3	4·39	0·40	·878	·527	2·4	12·43	0·34	19·88	10·142
·4	5·07	0·40	1·352	·811	2·5	12·68	0·34	21·14	10·778
·5	5·67	0·39	1·890	1·105	2·6	12·94	0·34	22·42	11·439
·6	6·20	0·89	2·414	1·411	2·8	13·42	0·34	25·04	12·774
·7	6·71	0·39	3·130	1·832	3·0	13·89	0·34	27·78	13·751
·8	7·17	0·38	3·824	2·179	3·2	14·35	0·33	30·60	15·153
·9	7·31	0·38	4·386	2·500	3·4	14·79	0·33	33·52	16·593
1·0	8·02	0·38	5·34	3·047	3·6	15·22	0·33	36·53	18·081
1·1	8·42	0·38	6·18	3·519	3·8	15·64	0·33	39·62	19·612
1·2	8·82	0·37	7·04	3·914	4·0	16·04	0·33	42·78	21·173
1·3	9·15	0·37	7·98	4·399	4·2	16·24	0·33	46·03	22·786
1·4	9·47	0·37	8·84	4·906	4·4	16·82	0·33	49·33	24·422
1·5	9·79	0·37	9·78	5·430	4·6	17·20	0·33	52·74	26·109
1·6	10·11	0·36	10·78	5·821	4·8	17·57	0·33	56·22	27·829
1·7	10·67	0·36	12·09	6·529	5·0	17·93	0·33	59·76	29·564
1·8	10·75	0·36	12·90	6·966	5·2	18·29	0·33	63·40	31·385
1·9	10 99	0·36	13·92	7·516	5·4	18·64	0·33	67·11	33·216
2·0	11·34	0·36	15·12	8·164	6·0	19·64	0·33	78·56	38·887

Mr. Sang of Kirkcaldy has proposed a very ingenious arrangement for the approximate measurement of the flow of water over a rectangular notch in a waste board, particularly applicable in cases where the flow has to be frequently registered, as in the daily observations by which drainage is estimated. He employs a scale graduated variably, so as to give at once the number of cubic feet per minute of water to every inch in breadth of the rectangular notch. Hence instead of employing a complex formula, nothing more is required than to observe at what number the water stands on the rule, and to multiply that by the number of inches of breadth of the notch, and we obtain at once the discharge in cubic feet per minute. He proposes that such a scale engraved on paper should be placed in a glass tube, hermetically sealed, and permanently fixed in a suitable position on the weir. The following table calculated by Mr. Sang shows the position of the divisions of the scale corresponding to cubic feet, and tenths of cubic feet discharge:—

Division.	Distance from Zero of scale in inches.	Division.	Distance from Zero of scale in inches.	Division.	Distance from Zero of scale in inches.	Division.	Distance from Zero of scale in inches.
·0	·000	2·6	3·327	5·1	5·214	7·6	6·803
·1	·379	2·7	3·412	5·2	5·282	7·7	6·862
·2	·602	2·8	3·496	5·3	5·350	7·8	6·921
·3	·789	2·9	3·579	5·4	5·417	7·9	6·980
·4	·955	3·0	3·661	5·5	5·483	8·0	7·039
·5	1·109	3·1	3·741	5·6	5·549	8·1	7·098
·6	1·252	3·2	3·822	5·7	5·615	8·2	7·156
·7	1·387	3·3	3·901	5·8	5·681	8·3	7·214
·8	1·516	3·4	3·979	5·9	5·746	8·4	7·272
·9	1·641	3·5	4·057	6·0	5·811	8·5	7·329
1·0	1·760	3·6	4·134	6·1	5·875	8·6	7·387
1·1	1·875	3·7	4·210	6·2	5·939	8·7	7·444
1·2	1·987	3·8	4·285	6·3	6·003	8·8	7·501
1·3	2·096	3·9	4·360	6·4	6·066	8·9	7·558
1·4	2·202	4·0	4·434	6·5	6·129	9·0	7·614
1·5	2·306	4·1	4·508	6·6	6·192	9·1	7·671
1·6	2·407	4·2	4·581	6·7	6·254	9·2	7·727
1·7	2·507	4·3	4·654	6·8	6·316	9·3	7·783
1·8	2·604	4·4	4·726	6·9	6·378	9·4	7·838
1·9	2·700	4·5	4·797	7·0	6·440	9·5	7·894
2·0	2·794	4·6	4·868	7·1	6·501	9·6	7·949
2·1	2·886	4·7	4·938	7·2	6·562	9·7	8·004
2·2	2·977	4·8	5·007	7·3	6·622	9·8	8·059
2·3	3·066	4·9	5·077	7·4	6·683	9·9	8·114
2·4	3·155	5·0	5·146	7·5	6·743	10·0	8·168
2·5	3·242						

5. *Friction of fluids in conduits and pipes.*—In long tubes an increased retardation arises, which must be ascribed to the friction of the fluid against the sides, and it has been ascertained that this element of retardation, whilst independent of the pressure of the fluid, increases in the ratio of the length of the tube, and decreases in the ratio of the width or diameter. It also increases nearly as the square of the velocity.

For pipes of uniform size and with no considerable amount of bending, it may be shown that the velocity of discharge

$$= v = \sqrt{\frac{2380\,h\,d}{l + 54\,d}} - \frac{1}{12} \cdot \frac{l}{l + 54\,d} \cdots (7)$$

or if h be not very small, neglecting the last term,

$$\sqrt{\frac{2380\,h\,d}{l + 54\,d}} \cdots (8)$$

and if the pipes be very long,

$$v = \sqrt{\frac{2380\, h\, d}{l + 54\, d}} - \frac{1}{12} \ldots (9)$$

where l = length of pipe in feet; d = diameter in feet; h = head in feet; and the constants have been derived from the experiments of Prony and d'Aubuisson.

Formula (8) very nearly coincides with that given by Poncelet, namely,

$$v = 47{\cdot}9 \sqrt{\frac{d\, h}{l + 54 d}} = \sqrt{\frac{2300\, d\, h}{l + 54\, d}}$$

The most convenient way in practice of estimating the retardation of friction in the pipes is to measure the head of water which is requisite to overcome the friction, without increasing the velocity of the current. In calculations of quantity when the head h, necessary to generate the required velocity of exit, has been estimated by the rule for thick-lipped orifices already given, another head h_1 must be added as necessary to overcome the friction, if the orifice is prolonged into a tube.

The height to overcome friction may be calculated from the formula

$$h_1 = n \frac{l}{d} \cdot \frac{v^2}{2\, g} \ldots (10)$$

where n is the coefficient of friction derived from experiment. Hence, putting Q for the discharge,

$$v = \frac{4\, Q}{\pi\, d^2} \ldots (11)$$

when the pipe is cylindrical.

We may combine this formula with the preceding formula for the discharge from a thick-lipped orifice, putting m = coefficient of resistance for the portion of tube next the cistern, and n, coefficient of resistance for the remainder of the tube, we then have for the whole head of water

$$h = m \frac{v^2}{2\, g} + n \,.\, \frac{l}{d} \cdot \frac{v^2}{2\, g} + \frac{v^2}{2\, g} \ldots (12)$$

or $$h = \left(1 + m + n \frac{l}{d}\right) \frac{v^2}{2\, g} \ldots (13)$$

and $$v = \frac{\sqrt{2\, g\, h}}{\sqrt{1 + m + n \frac{l}{d}}} \ldots (14)$$

and putting a = area of orifice, the discharge =

$$Q = a\,v \ldots (15).$$

If there be bends in the tube an increased element of resistance is introduced; if we put p for the sum of the resistances due to this source,

$$h = \left(1 + m + n\frac{l}{d} + p\right)\frac{v^2}{2\,g} \ldots (16)$$

$$= \left(1 + m + n\frac{l}{d} + p\right)\left(\frac{4\,Q}{\pi}\right)^2\frac{1}{2\,g\,d^4} \ldots (17).$$

These formulæ we have from Weisbach's "Mechanics of Engineering," from which the following table has been reduced and adapted to English measures.

Table of the Value of the Coefficient of Friction n for different Velocities.

Inches.	Velocity in Feet.											
	0	1	2	3	4	5	6	7	8	9	10	11
0	∞	·0316	·0265	·0244	·0229	·0220	·0215	·0209	·0205	·0202	·0198	·0195
4	·0443	·0293	·0257	·0239	·0226	·0218	·0213	·0208	·0204	·0201	·0197	·0195
8	·0355	·0277	·0250	·0233	0223	·0217	·0211	·0206	·0203	·0200	·0196	·0194
12	·0316	·0265	·0244	·0229	·0220	·0215	·0209	·0205	·0202	·0199	·0195	·0193

To use this table look in the horizontal line at top for the nearest velocity in feet, in the vertical column underneath and opposite the nearest number of inches will be found the value of n required. Thus for a velocity of 6 feet 8 inches per second, the coefficient n will be found to be 0·211, being under 6 and opposite 8.

From sixty-three experiments Weisbach deduces another general formula for the flow of water in tubes which is very convenient for calculation. It is based on the hypothesis that the resistance to friction increases simultaneously as the square, and as the square root of the cube of the velocity, and is of the form

$$h_1 = \left(0{\cdot}01482 + \frac{0{\cdot}017963}{\sqrt{v}}\right)\frac{l}{d}\cdot\frac{v^2}{2\,g}, \ldots (18)$$

which gives the head due to friction in feet. From this formula, Mr. James Thomson, M.A., C.E., and Mr. George Fuller, C.E., have calculated the following very complete and convenient table.

In laying pipes the following directions are not unimportant; the mouth, both for ingress and egress, should be trumpet-shaped; bends should be as far as possible avoided, and especially sharp angular bends; at junctions the smaller pipe should be brought round in a curve to agree in direction with the main. And, lastly, where a pipe rises and falls much, air is apt to collect in the upper parts of the bends, and thus reduce the section at that part, and it is advisable to make provision by a cock or otherwise for drawing it off at intervals.

Flow of water in open channels.—This is a question of importance, and requires careful consideration on the part of the engineer, as it is a case of frequent occurrence in calculations of the flow of water. It is often, from various circumstances, impossible to throw even a temporary waste board or weir across a stream the quantity of discharge from which it is desirable to ascertain, and hence it becomes necessary to determine a formula which takes into account the friction of the river sides.

In estimating the velocity of a stream on a canal or river, by throwing in floating bodies of nearly the same specific gravity as the water, and estimating the time they require to pass a given measured distance, it must be borne in mind that the velocity is greatest in the centre of the stream and near the surface, and less at the bottom and near the sides. It is generally most convenient to ascertain the velocity at the centre, where the stream is fleetest, but it is essential in calculations to know the mean velocity, or the velocity of a stream of the same section, discharging the same quantity of water, but unaffected by friction at the sides. In practice it will be sufficient to assume that the mean velocity of a stream is equal to 0·83 per cent. or $\frac{4}{5}$ of the velocity at the surface.

Or we may use an empirical formula of Prony's, putting v for the mean velocity, and v for the surface velocity, measured by a floating body at the middle of the stream

$$\text{v} = \frac{v\,(v + 7{\cdot}77)}{v + 10{\cdot}33} \ldots (19)$$

For small streams the most accurate method of measurement is the formation of a temporary weir by a vertical board thrown

FRICTION OF WATER IN PIPES.

Table calculated from a Formula of Julius Weisbach, to show, for Pipes 100 feet in Length, the relation between 1st. The Velocity of the Water, in feet per second.—2nd. The Internal Diameter of the Pipe.—3rd. The Head to overcome the Friction, in feet.—4th. The number of Cubic Feet of Water delivered per Minute; so that when any two of these four quantities are given, the remaining two can be found.*

By James Thomson, A.M., C.E., and George Fuller, C.E., Belfast.

		Internal Diameter of the Pipes in Inches.											
		3		3½		4		4½		5		6	
Velocity of Water, in feet per second.	Head to produce velocity, in feet.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.
2·0	·062	·659	5·89	·565	8·02	·494	10·4	·439	13·2	·395	16·3	·329	23·5
2·2	·075	·780	6·48	·669	8·82	·585	11·5	·520	14·6	·468	18·0	·390	25·9
2·4	·090	·911	7·07	·781	9·62	·683	12·5	·607	15·9	·547	19·6	·456	28·2
2·6	·105	1·05	7·65	·901	10·4	·788	13·6	·701	17·2	·631	21·3	·526	30·6
2·8	·122	1·20	8·24	1·03	11·2	·900	14·6	·800	18·5	·720	22·9	·600	32·9
3·0	·140	1·35	8·83	1·16	12·0	1·02	15·7	·905	19·8	·815	24·5	·679	35·3
3·2	·160	1·52	9·42	1·31	12·8	1·14	16·7	1·02	21·2	·915	26·2	·763	37·7
3·4	·180	1·70	10·0	1·46	13·6	1·27	17·8	1·13	22·5	1·02	27·8	·851	40·0
3·6	·202	1·89	10·6	1·62	14·4	1·41	18·8	1·26	23·8	1·13	29·4	·943	42·4
3·8	·225	2·08	11·2	1·78	15·2	1·56	19·9	1·39	25·2	1·25	31·0	1·04	44·7
4·0	·250	2·28	11·8	1·96	16·0	1·71	20·9	1·52	26·5	1·37	32·7	1·14	47·1
4·2	·275	2·49	12·3	2·14	16·8	1·87	22·0	1·66	27·8	1·50	34·3	1·25	49·5
4·4	·302	2·71	12·9	2·33	17·6	2·03	23·0	1·81	29·1	1·63	36·0	1·35	51·8
4·6	·330	2·94	13·5	2·52	18·4	2·21	24·0	1·96	30·4	1·76	37·6	1·47	54·1
4·8	·360	3·18	14·1	2·72	19·2	2·38	25·1	2·12	31·8	1·91	39·2	1·59	56·5
5·0	·390	3·43	14·7	2·94	20·0	2·57	26·2	2·28	33·1	2·05	40·9	1·71	58·9
5·2	·422	3·68	15·3	3·15	20·8	2·76	27·2	2·45	34·4	2·21	42·5	1·84	61·2
5·4	·455	3·94	15·9	3·38	21·6	2·96	28·2	2·63	35·8	2·37	44·2	1·97	63·6
5·6	·490	4·22	16·5	3·61	22·4	3·16	29·3	2·81	37·1	2·53	45·8	2·11	65·9
5·8	·525	4·50	17·1	3·85	23·2	3·37	30·3	3·00	38·4	2·70	47·4	2·25	68·3
6·0	·562	4·78	17·7	4·10	24·0	3·59	31·4	3·19	39·7	2·87	49·1	2·39	70·7
6·2	·600	5·08	18·2	4·36	24·8	3·81	32·4	3·39	41·0	3·05	50·7	2·54	73·0
6·4	·640	5·39	18·8	4·62	25·6	4·04	33·5	3·59	42·4	3·23	52·3	2·69	75·4
6·6	·680	5·70	19·4	4·89	26·4	4·28	34·5	3·80	43·7	3·42	54·0	2·85	77·7
6·8	·722	6·02	20·0	5·16	27·3	4·52	35·6	4·01	45·0	3·61	55·6	3·01	80·1
7·0	·765	6·35	20·6	5·45	28·0	4·77	36·6	4·24	46·4	3·81	57·2	3·18	82·4

	Internal Diameter of the Pipes in Inches.													
	7		8		9		10		11		12		13	
Velocity of Water, in feet per second.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.	Head to overcome friction, in feet.	Cubic feet per minute.
2·0	·282	32·0	·247	41·9	·220	53·0	·198	65·4	·180	79·2	·165	94·2	·152	110
2·2	·334	35·3	·293	46·1	·260	58·3	·234	72·0	·213	87·1	·195	103	·180	121
2·4	·390	38·5	·342	50·2	·304	63·6	·273	78·5	·248	95·0	·228	113	·210	133
2·6	·450	41·7	·394	54·4	·350	68·9	·315	85·1	·287	103	·263	122	·242	144
2·8	·514	44·9	·450	58·6	·400	74·2	·360	91·6	·327	111	·300	132	·277	156
3·0	·582	48·1	·509	62·3	·453	79·5	·407	98·2	·370	119	·339	141	·313	166
3·2	·654	51·3	·572	67·0	·508	84·8	·458	105	·416	127	·381	151	·352	177
3·4	·729	54·5	·638	71·2	·567	90·1	·510	111	·464	134	·425	160	·393	188
3·6	·808	57·7	·707	75·4	·629	95·4	·566	118	·514	142	·472	169	·435	199
3·8	·892	60·9	·780	79·6	·693	101	·624	124	·567	150	·520	179	·480	210
4·0	·979	64·1	·856	83·7	·761	106	·685	131	·623	158	·571	188	·527	221
4·2	1·07	67·3	·935	87·0	·832	111	·748	137	·680	166	·624	198	·576	232
4·4	1·16	70·5	1·02	92·1	·905	116	·814	144	·740	174	·679	207	·626	243
4·6	1·26	73·7	1·10	96·3	·981	122	·883	150	·803	182	·736	217	·679	254
4·8	1·36	76·9	1·19	100	1·06	127	·954	157	·867	190	·795	226	·734	265
5·0	1·47	80·2	1·28	105	1·14	132	1·03	163	·935	198	·857	235	·791	276
5·2	1·58	83·3	1·38	109	1·23	138	1·10	170	1·00	206	·920	245	·850	287
5·4	1·69	86·6	1·48	113	1·31	143	1·18	177	1·07	214	·986	254	·910	298
5·6	1·81	89·8	1·58	117	1·40	148	1·26	183	1·15	222	1·05	264	·973	309
5·8	1·93	93·0	1·68	121	1·50	154	1·35	190	1·22	229	1·12	273	1·04	321
6·0	2·05	96·2	1·79	125	1·59	159	1·43	196	1·30	237	1·19	283	1·10	332
6·2	2·18	99·4	1·90	130	1·69	164	1·52	203	1·38	245	1·27	292	1·17	343
6·4	2·31	102	2·02	134	1·79	169	1·61	209	1·47	253	1·35	301	1·24	354
6·6	2·44	106	2·14	138	1·90	175	1·71	216	1·55	261	1·42	311	1·31	365
6·8	2·58	109	2·26	142	2·01	180	1·81	222	1·64	269	1·50	320	1·39	376
7·0	2·72	112	2·38	146	2·12	185	1·90	229	1·73	277	1·59	330	1·46	387

Internal Diameter of the Pipes in Inches.

Velocity of Water, in feet per second.	Head to produce this velocity, in feet.	14		15		16		17		18		19	
		Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.
2·0	·062	·141	128	·132	147	·123	167	·116	189	·110	212	·104	236
2·2	·075	·167	141	·156	162	·146	184	·138	208	·130	233	123	260
2·4	·090	·195	154	·182	176	·171	201	·161	227	·152	254	·144	283
2·6	·105	·225	167	·210	191	·197	218	·185	246	·175	275	·166	307
2·8	·122	·257	179	·240	206	·225	234	·212	265	·200	297	·189	331
3·0	·140	·291	192	·271	221	·255	251	·240	284	·226	318	·214	354
3·2	·160	·327	205	·305	235	·286	268	·269	302	·254	339	·241	378
3·4	·180	·365	218	·340	250	·319	284	·300	321	·283	360	·269	401
3·6	·202	·404	231	·377	265	·354	301	·333	340	·314	382	·298	425
3·8	·225	·446	243	·416	280	·390	318	·367	359	·347	403	·328	449
4·0	·250	·489	256	·457	294	·428	335	·403	378	·380	424	·360	472
4·2	·275	·534	269	·499	309	·468	352	·440	397	·416	445	·394	496
4·4	·302	·582	282	·543	324	·509	368	·479	416	·452	466	·429	519
4·6	·330	·631	295	·589	339	·552	385	·519	435	·490	488	·465	543
4·8	·360	·682	308	·636	353	·596	402	·561	454	·530	509	·502	567
5·0	·390	·734	321	·685	368	·642	419	·605	473	·571	530	·541	590
5·2	·422	·789	333	·736	383	·690	435	·650	492	·614	551	·581	614
5·4	·455	·845	346	·789	397	·740	452	·696	511	·657	572	·623	638
5·6	·490	·903	359	·843	412	·791	469	·744	529	·703	594	·666	661
5·8	·525	·964	372	·899	427	843	486	·793	548	·749	615	·710	685
6·9	·562	1·02	385	·957	442	·897	502	·844	567	·798	636	·755	709
6·2	·600	1·09	397	1·01	456	·953	519	·897	586	·847	657	·802	732
6·4	·640	1·15	410	1·08	471	1·01	536	·951	605	·898	678	·851	756
6·6	·680	1·22	423	1·14	486	1·07	553	1·01	624	·950	700	·990	780
6·8	·722	1·29	436	1·20	500	1·13	569	1·06	643	1·00	721	·951	803
7·0	·765	1·36	449	1·27	515	1·19	586	1·12	662	1·06	742	1·00	827

Velocity of Water, in feet per second.	20		22		24		26		28		30	
	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.	Head to over-come friction, in feet.	Cubic feet per minute.
2·0	·099	262	·090	316	·082	377	·076	442	·070	513	·066	589
2·2	·117	288	·106	348	·097	414	·090	486	·083	564	·078	648
2·4	·137	314	·124	380	·114	452	·105	531	·097	616	·091	707
2·6	·158	340	·143	412	·131	490	·121	575	·112	667	·105	766
2·8	·180	366	·164	443	·150	528	·138	619	·128	718	·120	824
3·0	·204	393	·185	475	·170	565	·157	663	·145	770	·136	883
3·2	·229	419	·208	507	·191	603	·176	708	·163	821	·152	942
3·4	·255	445	·232	538	·213	641	·196	752	·182	872	·170	1001
3·6	·283	471	·257	570	·236	678	·218	796	·202	923	·189	1060
3·8	·312	497	·284	601	·260	716	·240	840	·223	974	·208	1119
4·0	·342	523	·311	633	·285	754	·263	885	·244	1026	·228	1178
4·2	·374	550	·340	665	·312	791	·288	929	·267	1077	·249	1237
4·4	·407	576	·370	697	·339	829	·313	973	·290	1129	·271	1296
4·6	·441	602	·401	728	·368	867	·339	1017	·315	1180	·294	1355
4·8	·477	628	·434	760	·397	905	·367	1062	·341	1231	·318	1414
5·0	·514	654	·467	792	·428	942	·395	1106	·367	1283	·343	1472
5·2	·552	680	·502	823	·460	980	·425	1150	·394	1334	·368	1531
5·4	·592	707	·538	855	·493	1018	·455	1194	·423	1385	·394	1590
5·6	·632	733	·575	887	·527	1055	·486	1239	·452	1437	·422	1649
5·8	·674	759	·613	918	·562	1093	·519	1283	·482	1488	·450	1708
6·9	·718	785	·652	950	·598	1131	·552	1327	·513	1539	·478	1767
6·2	·762	811	·693	982	·635	1168	·586	1371	·544	1590	·508	1826
6·4	·808	838	·735	1013	·673	1206	·622	1416	·577	1642	·539	1885
6·6	·855	864	·778	1045	·713	1244	·658	1460	·611	1693	·570	1943
6·8	·904	890	·821	1077	·753	1282	·695	1504	·645	1744	·602	2003
7·0	·953	916	·867	1109	·794	1319	·733	1548	·681	1796	·635	2061

* This Table also gives, by the first and second columns, the relation between the velocity and the head required to produce the velocity, calculated by the laws of falling bodies, independently of friction, contraction of the vein, or other retarding causes.

The formula of WEISBACH used for this Table is given in his "Ingenieur-und-Maschinen-Mechanik," vol. 1. p. 434, and when reduced to English measure is as follows :

$$H = \left\{ 0{\cdot}0144 + \frac{0{\cdot}01716}{\sqrt{V}} \right\} \frac{L}{D} \times \frac{V^2}{64{\cdot}4};$$

where H = head to overcome the friction, in English feet ; L = length of pipe, in English feet; D = internal diameter of pipe, in English feet; and V = velocity of water, in English feet per second.

across the stream and carefully puddled at the edges. A rectangular notch of sufficient capacity to pass the water must be cut in the middle portion. The height of the water above the level of the notch should then be measured either at its crest, or better still at some distance behind, where the water is nearly still, and the constant for calculation will be found in Table V. or VI. as the case may be.

But if a waste board or weir cannot be employed, we may find the surface velocity, and from that obtain its mean velocity by the methods given above. If then we take the depth of the stream at various parts of its breadth and so calculate its sectional area, we may find the cubic feet of water discharged per second by multiplying the mean velocity (in feet per second) by the area so found (in square feet).

Thus, if a body floats along the surface of a stream 300 feet in a minute, its maximum velocity $= \frac{300}{60} = 5$ feet per second, and its mean velocity, according to Prony, $= \frac{5\ (5 + 7{\cdot}77)}{5 + 10{\cdot}33} = 4{\cdot}16$. Now let the depth of the stream, 16 feet broad, measured at equal distances of two feet apart, be 0, 1, $2\frac{1}{2}$, 3, 3, $3\frac{3}{4}$, 3, $1\frac{3}{4}$, 0 feet respectively, then the area =

$$2 \times (1 + 2\tfrac{1}{2} + 3 + 3 + 3\tfrac{3}{4} + 3 + 1\tfrac{3}{4}) = 36 \text{ sq. ft.}$$

$\therefore$ Cubic feet of water discharged per second

$$= 36 \times 4{\cdot}16 = 149{\cdot}76 \text{ cubic feet.}$$

In rivers, the coefficient of friction n in formulæ (12) (13) and (14) may be taken at 0·0075 ; it varies according to Weisbach from 0·00811 to 0·00748 as the velocity increases from 0·1 to 1·0 metres, and from 0·00748 to 0·00743 as the velocity increases from 1 to 3 metres.

The following formulæ express the relations of velocity, fall, and discharge, when the flow of the stream is uniform :

$$h = \zeta \cdot \frac{l\,p}{F} \cdot \frac{c^2}{2\,g}, \ldots (20)$$

$$c = \sqrt{\frac{F}{\zeta\,l\,p} \cdot 2\,g\,h}, \ldots (21)$$

$$Q = F\,c \ldots (3).^*$$

* Weisbach, vol. i. p. 493.

Where h = whole fall, Q = discharge, F = transverse section of stream, c = mean velocity of stream, l = distance which the river flows for a fall h, $\frac{p}{F}$ the perimeter of the water profile, and ζ the coefficient of friction.

The best form of section must be that which presents the least resistance to a given quantity of water flowing through the channel. Now it has been shown that the resistance of friction varies directly as the wetted perimeter and inversely as the area of the section, and when the area is constant it will therefore vary directly as the wetted perimeter. Consequently the best form of section will be that with the least perimeter for a given area. Hence for open channels in which the upper water line is not part of the wetted perimeter, the half square is the best rectangular section, and the semihexagon the best trapezoidal section. For equal flows of water the semicircle will have less friction than the semihexagon, and the semihexagon than the semisquare. In designing conduits, for instance, the head race or tail race of water wheels, not only must the sectional form be attended to, but bends must be avoided as much as possible.

Estimation of Water Power.—Where a natural reservoir of mechanical power is employed through the medium of a prime-mover in overcoming resistances, in sawing, grinding, &c., we term the moving force the power, and the resistances overcome the work.

The dynamic unit by which we estimate force or resistance is the *foot-pound*, or the unit of force which is capable of lifting a weight of one pound one foot high. A second unit is employed when estimating large expenditure of force, namely, the *horse-power*. One horse-power, according to the estimate of Watt, was equivalent to 33,000 lbs. raised one foot high in a minute or 550 foot-pounds per second.

It is evident that a power exerted by a weight of water falling a given number of feet is capable of raising an equal weight the same number of feet. The power expended must equal the resistances overcome. In transmitting power through a prime-mover, however, a certain loss necessarily takes place, arising (1) from the loss or waste of the power by spilling, leakage, &c., and (2) from the absorption of a part of the power in overcoming the resistances of the prime-mover itself, friction, &c. Hence the work accomplished by a prime-mover is never equivalent to the power expended on it; the useful effect is always only a certain percentage of the power, and this percentage is called the efficiency or modulus of the machine.

Now for a water-wheel on which a stream of water acts by gravity alone:—

Let h = height of fall in feet.
W = weight of water delivered on the wheel per second.
n = the number of cubic feet per second.
P = the dynamic force of the falling water in foot-pounds.
P_1 = P reduced to horses power.
U = the useful effect of the machine in foot-pounds and
$U_1 = U$ to reduced H. P.
u = the modulus of the machine.

Then for the total water power of the fall we have, in foot-pounds,

$$P = W h \ldots (1)$$

and water weighing 62·5 lbs. per cubic foot,

$$P = 62{\cdot}5 \; n \; h$$

or, in horses power,

$$P_1 = \frac{W h}{550} = \frac{62{\cdot}5 \, n h}{550} = \frac{n h}{8{\cdot}8}$$

Hence, for every foot of fall 8·88 cubic feet of water per second, or 1·47 tons per minute, theoretically afford an available force of one horse power.

But by definition,

$$\frac{u P}{100} = U \ldots (22).$$

$$\therefore U = \frac{u W h}{100} \text{ in foot-pounds.}$$

$$\therefore U_1 = \frac{u \; W \; h}{55000} \text{ in horses power.}$$

and,

$\dfrac{100 - u}{100} P$ = the sum of the resistances from friction, &c., and the loss from wasted water, in accumulating and transmitting the power.

CHAP. III.

ON THE CONSTRUCTION OF WATER WHEELS.

In the present age, the same importance is not attached to water power as before the introduction of steam, as has been already shown. Nevertheless, since water is still largely employed in some districts and for certain kinds of work, it is of importance that the machinery for rendering it useful should be constructed upon the best principles so as to secure a maximum effect. In numerous localities in Europe and America, water is the principal motive agent by which manufacturing processes are carried on; and the time has not yet arrived when it can be dispensed with even in our own country. We shall endeavour to point out the difference between the ordinary and improved forms of water wheels, and to lay down sound principles of construction, accompanied by examples for the guidance of the millwright.

CLASSIFICATION OF WATER MACHINES.

Water may be expended upon water machines, 1st. By gravitation, as in vertical wheels generally; 2nd. By pressure simply, as in the water pressure engine, where the water acts on a reciprocating piston; 3rd. By the impulse of effluent water striking float boards, as in the Poncelet wheel; 4th. By the reaction of effluent water issuing from an orifice, as in the Barker's mill and Whitelaw's turbine; or lastly, by momentum, as in the case of the water ram.

It is not, however, always possible in practice to classify water machines according to the mode in which the water expends its force, and hence it will be more convenient to divide them according to the point at which the water is applied, and the direction in which it passes through the wheel, as in the following summary:—

1st. *Vertical Water Wheels*, the plane of rotation being vertical and the water received and afterwards discharged at the same orifice on the external periphery. These may be subdivided into :—

a. Overshot wheels, where the water is applied over the crest, or near the upper extremity of the vertical diameter.

b. Breast wheels, where the water is applied below the crest at the side of the wheel.

c. Undershot wheels, where the water is applied near the bottom of the wheel, and acts, 1. By gravitation as in the improved undershot wheel; or 2. By impulse as in the ordinary undershot and Poncelet wheels.

2nd. *Horizontal Wheels*, the plane of rotation being horizontal and the water passing through the wheel from one side to the other. These may be subdivided into :—

a. Horizontal wheels strictly so called, in which the water passes vertically down through the wheel, acting as it passes on curved buckets.

b. Turbines, annular wheels in which the water enters the buckets at the internal periphery, and passing horizontally is discharged at the external periphery.

c. Vortex wheels, in which the water entering at the external periphery flows horizontally and is discharged at the internal periphery.

3rd. *Reciprocating Engines*, in which the water is applied upon a piston and regulated by valves on the same principle as the steam engine.

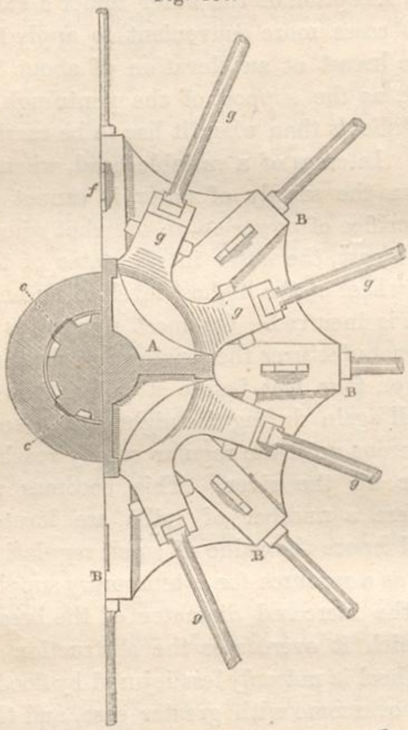
The improvements of the Vertical Wheel.—In the present chapter it will be convenient to enter on the consideration of the construction of vertical wheels. Since the time of Smeaton's experiments in 1759, the principle on which vertical water wheels have been constructed has undergone no important change, although considerable improvements have been effected in the details. The substitution of iron for wood has afforded opportunities for extensive changes in their forms, particularly in the shape and arrangement of the buckets, and has given a lighter and more permanent character to the machine than had previously been attained. A curvilinear form for the buckets has been adopted, the sheet iron of which they are composed affording great facility for being moulded into the required shape. It is not the object of the present treatise to enter into the dates of past improvements, but it will suffice to observe that the breast wheel has taken precedence of the overshot wheel, probably from the increased facilities which a wheel of this description affords for the reception of the water under a varying head. It is in most cases more convenient to apply the water of high falls on the breast at an elevation of about 30° from the vertical diameter, as the support of the pentrough is much less expensive and difficult than when it has to be carried over the top of the wheel. In cases of a variable head, when it is desirable to work down the supply of water, it cannot be accomplished without a sacrifice of power on an overshot wheel; but when applied at the breast, the water in all states of the river is received upon the wheel at the highest level of its head at the time, and no waste is incurred. On most rivers this is important, as it gives the manufacturer the privilege of drawing down the reservoir three or four feet before stopping time in the evening, in order to fill again during the night; or to keep the mill at work in dry seasons until the regular supply reaches it from the mills higher up the river. This becomes an essential arrangement where a number of mills are located upon the same stream, and hence the value of small regulating reservoirs behind the mill as a resource for a temporary supply.

Another advantage of the increased diameter of the breast wheel is the ease with which it overcomes the obstruction of back water. The breast wheel is not only less injured by floods, but the retarding force is overcome with greater ease, and the wheel works in a greater depth of back water.

Component parts of Water Wheels.—Vertical water wheels consist essentially of a main axis resting on masonry foundations, and together with arms and braces forming the means of support for the machine. Chambers for the reception of the water constructed of shrouding, sole-plate, and buckets. A pentrough with sluice for laying on the water, and a tail-race for conveying it away; and an internal or external geared spur wheel and pinion for transmitting the power. These parts we shall treat of successively, before describing the modifications of the vertical wheel.

The main axis is a large and heavy cast-iron shaft carried upon plummer blocks bolted to the masonry foundations of the wheel house. It sustains the weight of all the moving parts of the wheel, and in some cases the power is taken from it, when it is subjected to a force of torsion. It is usually cast with deep

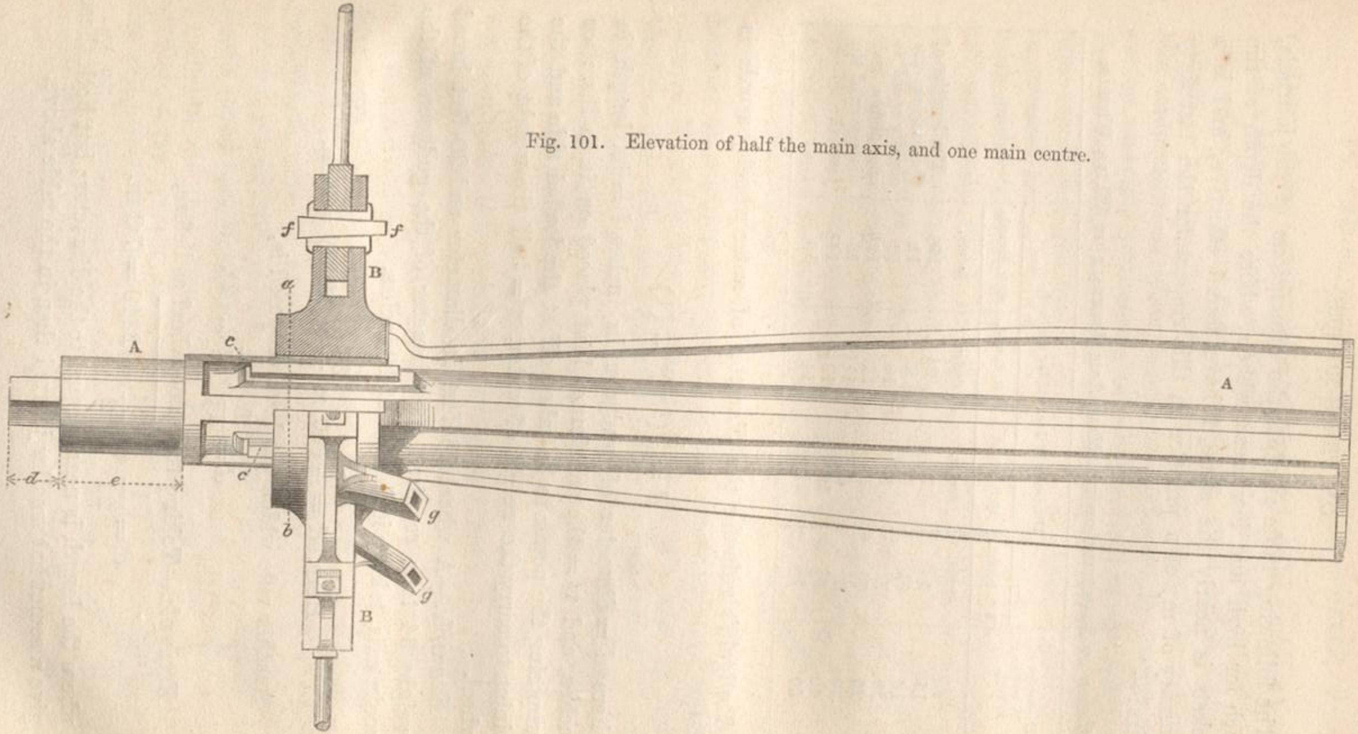
Fig. 100.



ribs or wings, calculated to resist the tensile and compressive strain to which they are alternately subjected as the wheel revolves. A section and elevation of the main-axis of a water wheel 20 feet in diameter and 22 feet wide are shown in figs. 100 and 101.* A A is one half the main-axis with its four deep ribs. The part *e* is the journal on which the wheel revolves, and *d* is left square, for the convenience of fixing a screw-jack should the wheel require raising. B, B, B are the recesses for the radial arms of 2½-inch round iron fixed by the keys *f*, *f*, *g*, *g* the corresponding recesses for the braces which pass diagonally across the wheel and alternate with the arms; *c*, *c* are the key beds on the main axis for fixing the main centre. It is difficult to estimate the strain on this shaft when the wheel is on the suspension principle, although the work it has to perform is trifling compared with what it would have to sustain in the event of the power being taken from the axle. In the latter case the wheel has to sustain not only the weight of the wheel and the water in the buckets, but also the force of torsion, as the power is transmitted from the periphery through the arms and axle to the main gearing of the mill.

* The wheel is shown in Plate IV. Fig. 110 is also an enlarged detail drawing of this wheel.

Fig. 101. Elevation of half the main axis, and one main centre.



The following table exhibits the dimensions of the journals, which for high and low breast wheels, where the depth of the buckets is nearly the same, I have found effective, and is a summary of my own practice in this respect for the last forty years:—

TABLE OF DIAMETERS OF THE MAIN AXIS JOURNALS OF WATER WHEELS.

Diameter of Wheels in feet.	Diameter of Journal for a wheel				
	5 ft. broad.	10 ft broad.	15 ft. broad.	20 ft. broad.	
	inches.	*inches.*	*inches.*	*inches.*	
15	6	7	8½	10	The lengths of the bearings are usually equal to one and a half diameters of the journal.
18	6½	7½	9	11	
20	7	8	10	12	
25	7½	8½	11	12½	
30	8	9	11½	13	
40	8¾	10	12½	14½	
50	9½	11	14	16	

Tredgold's rule for the diameter of water wheel journals is that,

$$d = \frac{1}{9}(l\,\mathrm{W})^{\frac{1}{3}} \ldots (1)$$

where d = diameter of gudgeon in inches, l = its length in inches, and W = the maximum load placed on it in lbs.; or supposing the power to be taken off at the loaded side and the pinion to carry the weight of water, W = half the weight of the wheel.

Example.—A wheel 18 feet in diameter and 20 feet broad weighs 34 tons, required the diameter of the gudgeon of the main axis, taking its length at 10 inches.

$$\text{Here, } d = \frac{1}{9}\left(10 \times \frac{34}{2} \times 2240\right)^{\frac{1}{3}} = 8 \text{ in.}$$

Another rule which has been proposed is,

$$d = \frac{1}{25}\sqrt{\mathrm{W}} \ldots (2).$$

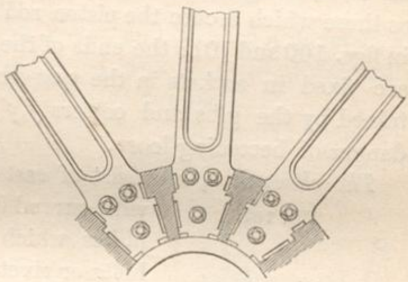
Example.—Taking the same wheel as before,

$$d = \frac{1}{25}\sqrt{\frac{34}{2} \times 2240} = 7{\cdot}8 \text{ in.}$$

where the length is nearly equal to the diameter; but both these give a somewhat smaller journal than in the table above.

There exists a wide difference of principle amongst mill-wrights as to the mode of attaching the wheel to the axis. It may either be rigidly fixed by cast-iron arms which resist its weight, as a series of columns alternately exposed to a tensile and compressive strain, or it may be supported by tension rods on the principle now most generally practised in the construction of improved iron water wheels. In the former case the arms are of cast-iron fixed in recesses in a cast-iron main centre, to which they are accurately fitted on chipping strips, and then bolted as shown in fig. 102. Flat wrought-iron arms are sometimes riveted to the main centre in a somewhat similar manner.

Fig. 102.



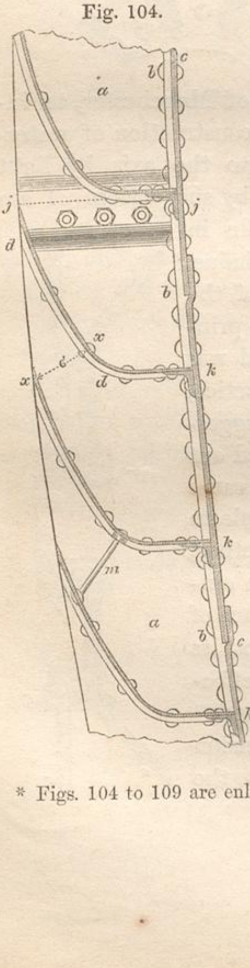
It was reserved for Mr. T. C. Hewes, of Manchester, to introduce an entirely new system in the construction of water-wheels, in which the wheels, attached to the axis by light wrought-iron rods, are supported simply by suspension. I am informed that a wheel on this principle in Ireland was actually constructed with chains, with which, however, from the pliancy of the links, there was some difficulty. But the principle on which this wheel was constructed was as sound in theory as economical in practice, and is due originally, it is said, to the suggestion of Mr. William Strutt, and was carried out fifty years ago by Mr. Hewes, whilst at the same time Mr. Henry Strutt applied the principle to cart-wheels, some of which, thus put together, were for a long time in use. Mr. Hewes employed round bars of malleable iron in place of the chains, and this arrangement has kept its ground to the present time, as the most effective and perfect that has yet been introduced.

Fig. 103.

In the earlier construction of suspension wheels the arms and braces were attached to the centre by screws and nuts, as shown in fig. 103. The arms *c*, *c* passed through the rim *b b*, and the braces *e*, *e* are set diagonally in the angle of the rim. This arrangement, although convenient for tightening up the arms and braces, was liable to many objections; the nuts were subject to become loose from the vibration in working, so as to endanger the wheel, and to create a difficulty in keeping it truly circular in form. To obviate this, in 1824, I substituted gibs and cotters, on the same principle as those which secure the piston rod of a steam engine, as shown in figs. 100 and 101: the ends of the arms are forged square, and are fixed in sockets in the cast-iron centre, and are there retained by the gibs and cotters *f*, *f* in perfect security from the danger of becoming loose.

The shrouds a, a consist of cast-iron plates cast in segments with curved flanches to receive the bucket plates which are attached to them by bolts or rivets (*d*, *d*, fig. 104), and round the inner periphery a projecting flanch (*b*, figs. 104 and 105) is formed for the reception of the sole plates (*c*). Fig. 104 is a side elevation, and fig. 105 a section of a large shrouding of this description, 15 inches deep. *a a* the cast-iron segmental plate of the shroud; *b* the flanch to which the sole plate *c c* is riveted; *d d* the curved flanches and bucket plates; B the bucket. The segments of the shrouds are bolted together by overlap joints, *j*, *j*, shown also in section in fig. 106. The overlap is placed on the bucket side of the shroud to preserve a smooth face on the outside of the wheel. The arms are attached to the shrouds either by riveting, or, according to my own practice, by dovetailing into recesses cast upon the inner face of the shroud. Fig. 107 * represents this arrangement in section, and fig. 108 in plan. The ends of

Fig. 104.



* Figs. 104 to 109 are enlarged details of the Catrine Wheels, Plates I. and II.

the arm *c c* are forged into a **T** form, and are fitted into a similar shaped recess on the shroud. To retain the arms in position, it

Fig. 105.

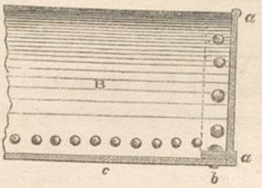
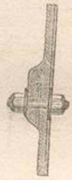


Fig. 106.



is only requisite to give to the recess and **T**-head a dovetail, as shown at *d*. The boss on the shroud must be tapered gradually

Fig. 107.

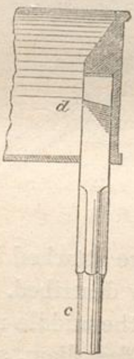
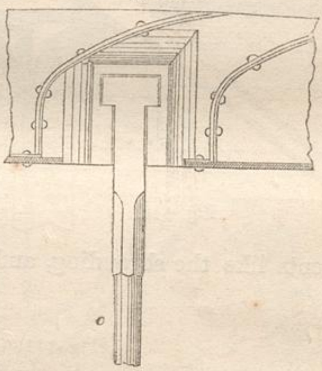


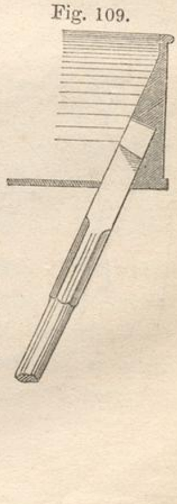
Fig. 108.



down, to avoid injury in casting from unequal contraction in cooling. The arms are usually 2 to $2\frac{1}{2}$ inches in diameter for almost all wheels, and the braces $1\frac{3}{4}$ to 2 inches.

To strengthen the wheel laterally, diagonal arms, called braces, are used (*g g*, *g g*, figs. 100 and 101), and where the wheel is not of great width, these braces pass from the main centre on one side to the shroud on the opposite side of the wheel, alternating with the radial arms and fixed in the same manner (fig. 109). Where the wheel is broad I prefer to attach the braces to a middle ring of cast-iron, riveted to the interior of the sole plates in their centre between the shroudings. This ring strengthens the wheel in an important degree, by supporting the bucket and sole plate at their weakest part, where they are liable to

Fig. 109.



yield to the weight of the water. The middle ring is cast in

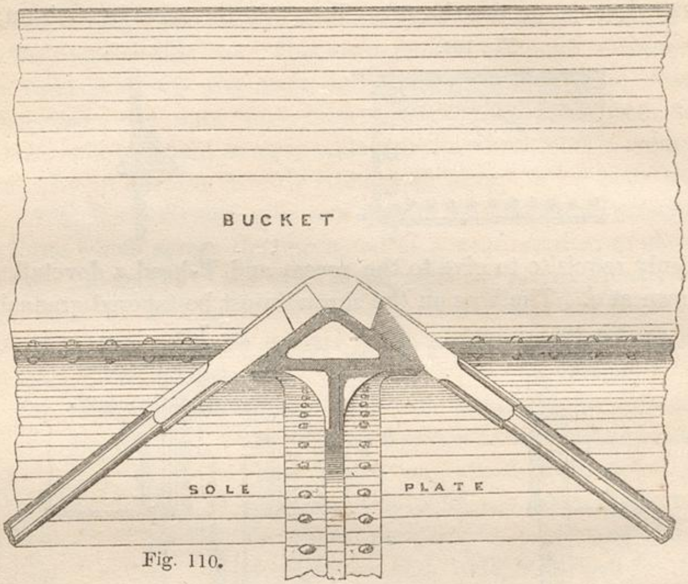


Fig. 110.

segments like the shrouding, and the braces are attached in the way already described. Fig. 110 shows the middle ring of a wheel 20 feet diam. and 22 feet broad.

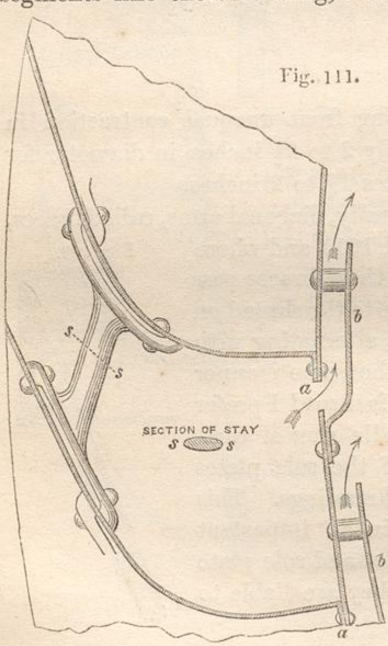
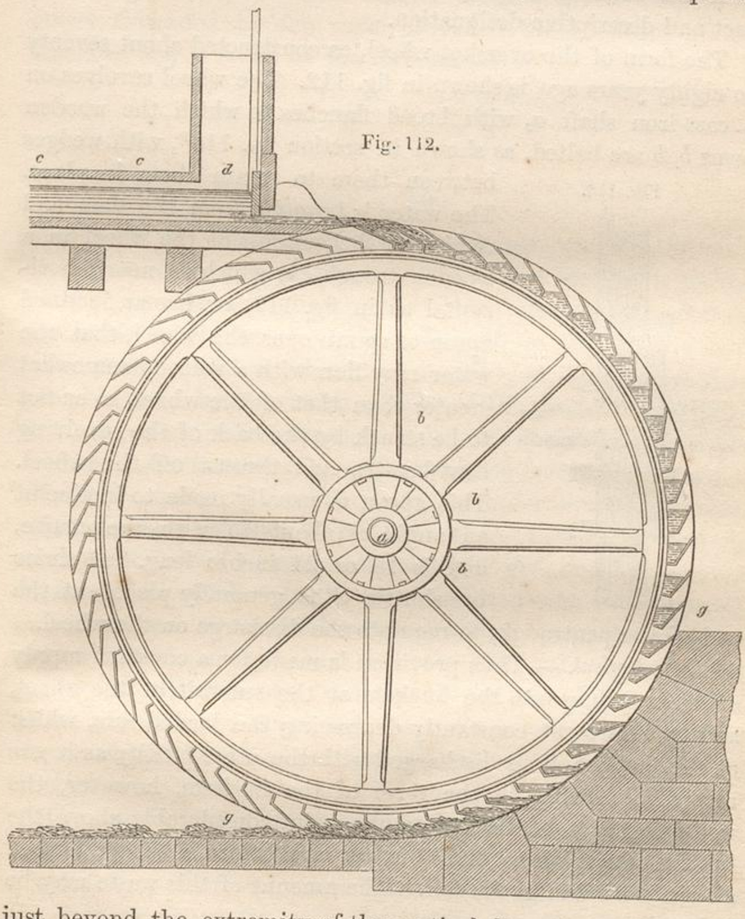


Fig. 111.

The sole plates are of wrought-iron, $\frac{1}{8}$th inch thick (No. 10 Wire Gauge) riveted together with lap-joints. The buckets are riveted throughout their whole length to the sole plate by a bend at the bottom, or in some cases by a small angle iron (*k k*, fig. 104). For the further support of the bucket plates, at every two feet of their length they are riveted to bucket stays forming a complete ring of auxiliary columns round the wheel at every two feet of its breadth. These bucket stays may be of wrought-iron, turned, with two collars, and riveted through each bucket plate, as at *m*, fig. 104, or else of cast-iron, as at *s s*, fig. 111.

The overshot water wheel.—By the overshot water wheel was originally intended that form of wheel in which the stream of water was led *over the summit* of the wheel, and thrown upon it

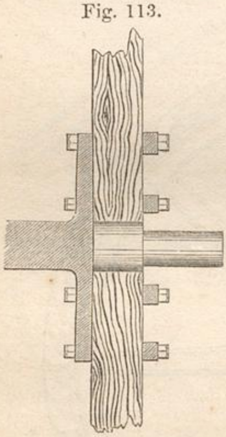
Fig. 112.



just beyond the extremity of the vertical diameter. The water is retained upon the wheel in troughs or buckets, and by its weight continuously depresses the loaded side of the wheel so as to create a motion of revolution. By a convenient modification of the mode of applying the water, however, the stream was laid on to the wheel upon the same side as it approached, by reversing the direction of the spout or sluice, and for this form the name of pitch-back over-shot wheel was employed. In present use the term overshot is no longer used strictly, but is arbitrarily applied to all wheels in which the water is laid on near the summit, although high-breast is perhaps a more correct and descriptive designation.

The form of the overshot wheel, as constructed about seventy to eighty years ago, is shown in fig. 112. The wheel revolves on a cast-iron shaft *a*, with broad flanches to which the wooden arms *b*, *b* are bolted, as shown in section fig. 113*, with wedges between them to retain them in place. The water is brought from the dam and carried to the summit of the wheel in a wooden trough *c c*, which is nearly horizontal as in fig. 112, or has an inclined apron or spout over the wheel, that the water may flow with a velocity somewhat greater than that of the wheel, so as not to be struck by the back of the revolving float-boards, and thrown off the wheel. This apron is usually made to incline at an angle of about 15° with the course, and is 18 or 24 inches long. A sluice or shuttle *d* is generally placed at the end of the pentrough, to regulate the discharge on the wheel.

Fig. 113.



Useful effect. — Thus provision is made for a constant supply of water falling into the buckets at the summit of the wheel, and by its weight constantly depressing the loaded side, whilst at the bottom it is discharged with the same facility as it was received. Owing to the form of the buckets, however, the water begins to be discharged at a point considerably above the bottom of the wheel, and thus escapes before it has performed all the work due to the fall. The amount of this waste may be reduced —

1. By adopting a curvilinear form of bucket.

* In earlier wheels, in which the main axis is of wood instead of iron, the principal arms are usually placed in parallel pairs extending across the main axis to the shrouding on either side.

2. By only partially filling the buckets.
3. By a close-fitting breast to retain the water on the wheel.

But when decreased as far as possible, this waste is still an important item in the performance of the wheel, and hence the useful effect secured is never equal to the work of the water due to the space through which it falls. The fraction expressing the percentage of useful effect derived from a given quantity of power expended by the water is called the efficiency of the machine, and is found by the formula —

$$m = \frac{100\,\mathrm{U}}{u} = \frac{100\,w\,h}{\mathrm{W\,H}}$$

$$\mathrm{U} = \frac{m\,\mathrm{W\,H}}{100}$$

where m is the efficiency of the machine per cent.; u the work of the water employed per minute, or the weight W of the water in pounds multiplied by the fall H in feet, measured from the surface of the water in the pentrough to that in the tail-race; U the useful effect of the machine, or the pressure p in pounds moved by the working point of the machine multiplied by h, the space in feet through which this point is moved per minute, or the number of pounds raised one foot high by the machine per minute. In ordinary overshot water wheels, the useful effect amounts to about 60 per cent. of the power; or a supply of 12 cubic feet of water per second will give one horse-power for every foot of fall. In the improved iron high-breast wheels, as I have been in the habit of constructing them, the efficiency amounts to 75 per cent., in which case 10·8 cubic feet of water per second will give one horse-power per foot of fall. This is about a maximum effect for water machines, and hence the improved high-breast wheel may be considered as nearly perfect as a water machine.

The waste of water from spilling may to a certain extent be reduced by decreasing the opening of the buckets, but with the disadvantage of at the same time increasing the difficulty of the exit of the water at the bottom of the wheel, and of its entrance at the summit. The waste may be further lessened in an important degree by increasing the breadth of the wheel and the capacity of the buckets, and in general it is not advisable that the buckets should ever be more than two-thirds filled with the average supply of water. The buckets then reach a much lower position before they begin to discharge than when they have been nearly filled. The third means of preventing the spilling of the water is by a curved breast fitting closely to the wheel, as shown in fig. 112, *g g*, and serving, when accurately fitted, to retain the water on the buckets. With low falls this breast is of considerable importance, and secures a considerable increase of efficiency. But with large wheels for high falls, with small openings in the buckets, it is of no value, and does not compensate for its cost, when the buckets are made of the best form, so as to retain the water as long as possible upon the wheel; and in these cases the breast is invariably dispensed with.

The pitch-back wheel.—The most important modification of the old over-shot wheel is known as the pitch-back wheel, in which the course of the current of water is reversed in the pentrough, and laid on the wheel from the same side at which it approaches. In old wheels it was essential, as the wheel generally worked more or less frequently in back water, that the tail-race should always lie in the same direction as the revolution of the wheel. Hence when the position of the waste water culvert was fixed by other circumstances, it often happened that the millwright was driven to the use of the pitch-back wheel to meet the conditions of the case, and the advantages of this form of wheel were thus forced on his notice. It was perceived that by increasing the diameter of the wheel, the water might be laid on at a distance from the summit, and it was shown theoretically that a larger useful effect would be secured by laying it on at about 25° to 30° from the summit, than if it took the water over the top. And in this way, when the introduction of iron gave sufficient facility for the construction of wheels of large diameter, the high breast wheel was adopted, and has maintained its ground to the present time as one of the most perfect and economical machines.

Direction of tail-race.—It is no longer necessary that the flow of the tail water should be in the direction of the wheel's revolution. On the contrary, I frequently take it in the opposite direction or at the side, according as the circumstances of the case determine the position of the wheel, and the point of discharge. The old plan of setting the wheel parallel with the stream is no longer requisite, provided proper care is taken to give a sufficient outlet to the water. To effect that object it is essential to sink the bottom of the tail-race two or two and a-half feet beneath the bottom of the wheel, and that depth should be continued to the river, so as to form the tail-race into a canal, with the water flowing gently, and with a comparatively slow motion from the wheel. In this arrangement the bottom of the wheel, when standing in an ordinary condition of the river, is 8 or 9 inches above the water in the tail-race, so that its motion cannot be impeded, and there is left ample space for the rise occasioned by the continuous discharge from the buckets during the working of the wheel. To show how immaterial is the direction of the tail-race, I may add that I have in some cases formed the tail-race into an underground tunnel, in the shape of an inverted syphon. Fig. 114 shows this arrangement as adopted for a mill in 1832, to secure an increase of fall. A shows the wheel and wheel-house, in which originally the wheel was 24 feet in diameter, the fall 22 feet, and the tail-

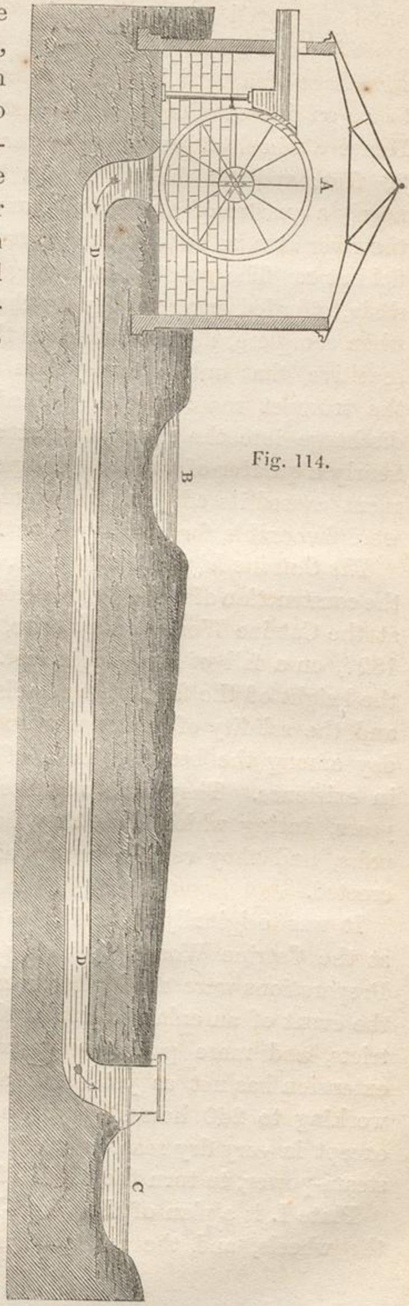


Fig. 114.

water conveyed direct into the river Eagley, at B. When replacing this wheel by a new one, it was found that by taking advantage of a bend in the river, and conveying the tail water to C, an increase of about 6 feet of fall could be obtained. Hence a wheel 32 feet in diameter was adopted, with a fall of 28 feet; and for the tail-race a tunnel D D was constructed, nearly a quarter of a mile long, and passing under the bed of the river at B, so as to meet the stream on the other side of the field at C. The substratum being composed of hard rock and shale, afforded every facility for the drifting of the tunnel, and when complete, the flow of water through it was so exceedingly sensitive, that only a few gallons falling from the wheel into the trumpet mouth at A, immediately caused a perceptible discharge into the river at C, at a distance already stated, of nearly a quarter of a mile. The perfect success of this arrangement caused its adoption in other cases, where the conditions were favourable for carrying it out.

The Catrine high-breast wheels.—Plates I. and II. illustrate the construction of the improved iron high-breast wheel as applied at the Catrine Works in Ayrshire, between the years 1825 and 1827, on a fall of forty-eight feet. Taking into consideration the height of the fall, these wheels, both as regards their power and the solidity of their construction, are even at the present day among the best and most effective structures of the kind in existence. They have now been at work upwards of thirty years, during which time they have required little or no repairs, and they remain nearly as perfect as when they were erected.

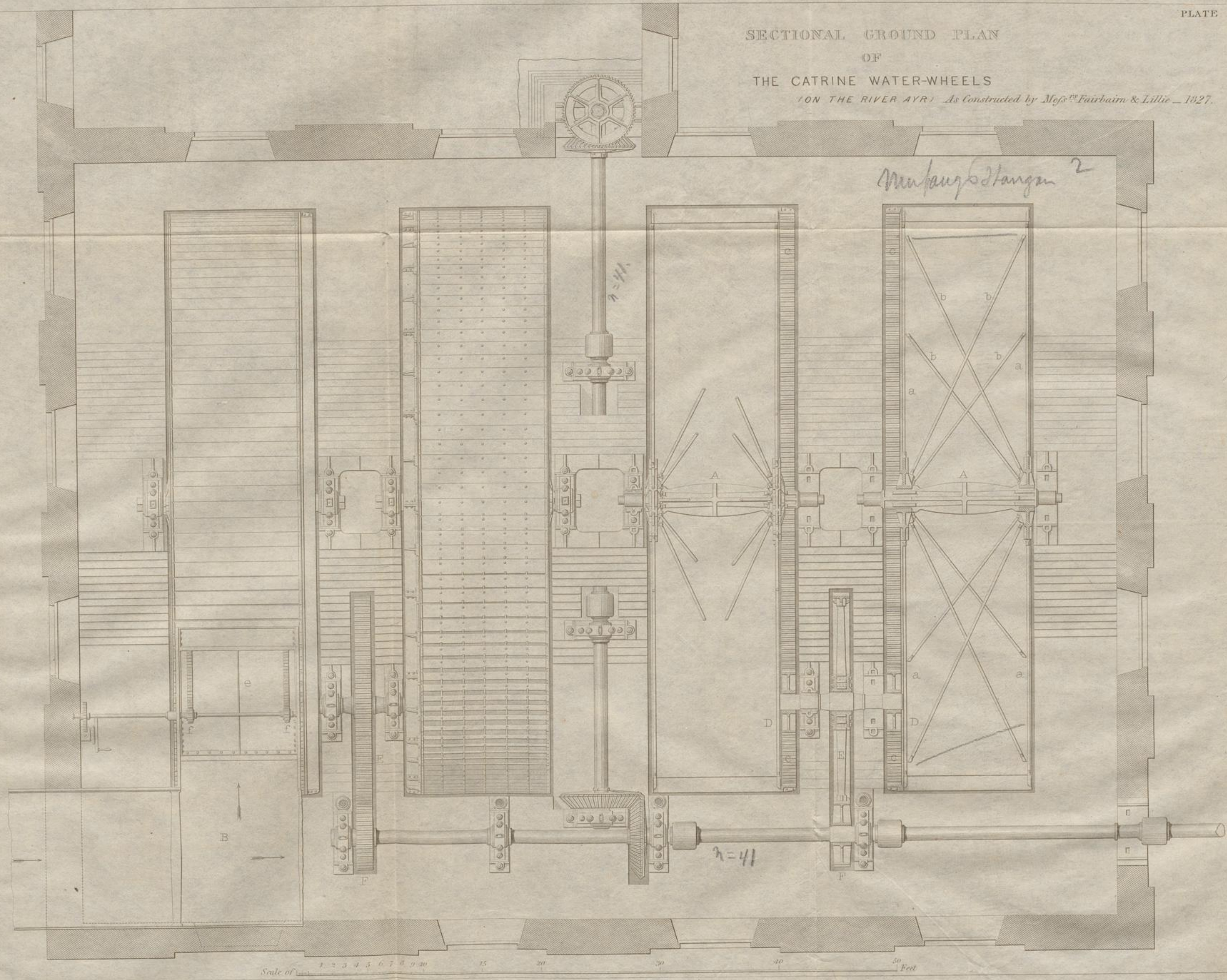
It was originally intended to erect four of these wheels at the Catrine Works, but only two have been constructed. Preparations were made, however, for receiving two others in the event of an enlargement of the reservoirs in the hill districts, and more power being required for the mills. This extension has not as yet been wanted, as these two wheels are working to 240 horses' power, and are sufficiently powerful, except in very dry seasons, when they are assisted by auxiliary steam-power, to turn the whole of the mills.

Plate I. is a plan of the wheel-house, showing the position of the wheels, and the arrangement of the main gearing. The

PLATE 1

SECTIONAL GROUND PLAN OF THE CATRINE WATER-WHEELS

(ON THE RIVER AYR) As Constructed by Mefs.rs Fairbairn & Lillie _ 1827.



Wm Fairbairn del. London Longman & Co. W. A. Beever first pair of wheels is shown in section, to exhibit the main axle, arms, braces, spur segments, and pinions. The other pair are shown in plan, one exhibiting the buckets, and five rows of bucket-stays, while the pentrough, sluice, and regulating gear are shown on the other. It will be seen that the motion of each pair of wheels is transmitted through a common pinion shaft, and thence by another pinion and spur-wheel, by which the velocity is increased to the first motion shaft of one mill, whilst between the two pairs of wheels there is the first motion shaft of another mill geared into the preceding shaft by a pair of large bevil wheels.

Plate II. is an elevation of the wheel house, with the masonry for supporting the wheels, tail-race, tunnel, &c. The right half of the wheel is shown in section, and the left half in elevation, and there is a section of the pentrough, sluice, and plates, to guide the water into the buckets.

The following are the references to the different parts of the wheel:—

A, main axis.

a a a, arms. *c*, segments.

b b b, braces. *d*, joints of segments.

B, pentrough.

e, sluice with racks. *f*, pinion connected with governor.

C, tunnel running through the wheel-house, and acting as the tail-race.

D, pinion gearing into internal segmental spur-wheel on shrouds.

E, wheel on the same axis as D, and communicating the power to the pinion F on the first motion shaft.

G, galleries to obtain access to the pentrough and other parts of the wheel.

The water is brought from the reservoirs in a tunnel 10 feet in diameter through the hill part, and thence in a conduit 12 feet wide, and 5 feet deep, arched over. The reservoirs cover 120 Scotch acres, of an average depth of 8 or 10 feet, giving storeage room for a large supply of water; and the sill of the reservoir sluice, from which the aqueduct bottom is carried level to the pentrough, is 16 inches above the lowest overflow of the sluice on the wheel; hence in dry seasons the water may be drawn off to within 16 inches of the bottom of the lade. At the same time the pentrough is made of a depth of six feet, in order that in seasons of plentiful supply the water may be drawn off at the highest level, and the entire fall, as far as possible, rendered effective.

The total supply of water requisite to work the mills when the wheels were started was about 60 tons or 2150 cubic feet per minute, the wheels revolving at a circumferential velocity of four feet a second, or 182 buckets passing each sluice per minute. This gives $\frac{2150}{182 \times 2} = 5{\cdot}9$ ft. or 6 cubic feet of water nearly for each bucket of the wheels. The whole capacity of each bucket is $17\frac{1}{4}$ cubic feet, hence when thus working the buckets were just one-third filled.

$v = 4$
$n =$
$m = 1/3$

When working to their full power of 240 horses, however, the fall being 48 feet, this pair of wheels would require,

$$\frac{100 \times 33000 \times 240}{75 \times 2240 \times 48} = 98{\cdot}2 \text{ tons of water per minute,}$$

if we suppose the useful effect to be 75 per cent. of the water power expended. Now if we take the circumferential velocity at 5 feet per second, at which the wheel should then run, this would give 7·7 cubic feet of water per bucket, or $\frac{7{\cdot}7}{17{\cdot}25} = \frac{10}{22}$ or one-half nearly, as the ratio of the quantity of water in the buckets to their capacity.

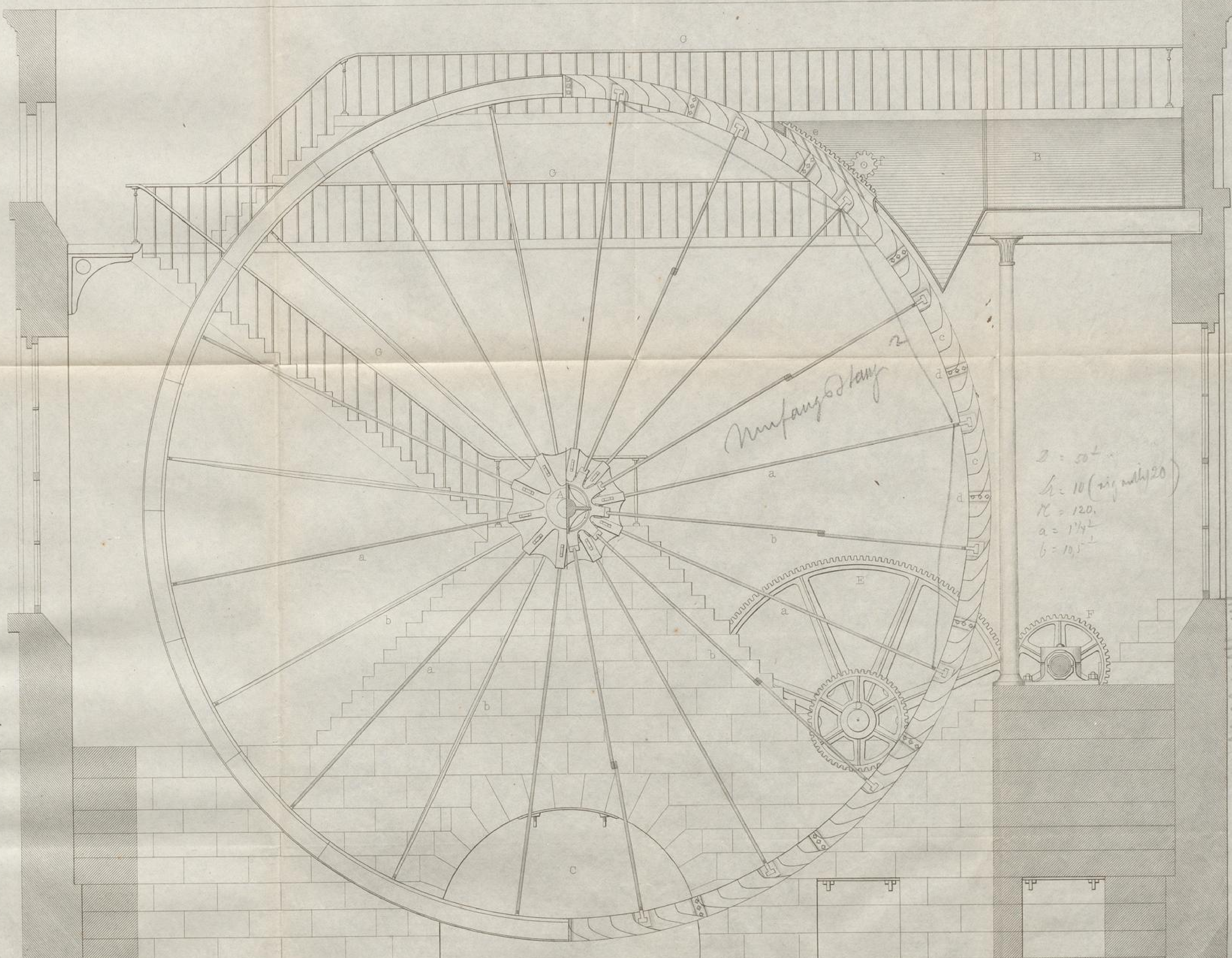
Between these limits these water wheels act effectively and economically.

$D = 50$
$b = 10.5$
$a = 1'4$
$D_1 = 48.5$
$t =$

The wheels are 50 feet in diameter, 10 feet 6 inches wide inside the bucket, and 15 inches deep on the shroud; the buckets are 120 in number, and have an opening of 6 inches; the internal spur segments are 48 feet 6 inches diameter, 3 inches and a quarter in pitch, 15 inches broad, and have 560 teeth. The pinions are the same width and pitch, and are 5 feet 6 inches in diameter. The intermediate wheel between the pair of segment pinions is 18 feet $3\frac{3}{4}$ inches in diameter, 16 inches broad, and $3\frac{1}{4}$ inches pitch; and the large bevil-

SECTION AND ELEVATION OF THE CATRINE WATER WHEEL.

PLATE 2.



Wm Fairbairn del.

London. Longman & Co

H. Adlard sc.

wheels are 7 feet in diameter, $3\frac{1}{2}$ inches pitch, and 18 inches broad on the cog, so as to be of sufficient strength to convey, if necessary, the united power of the four water wheels.

When viewed from the entrance, the two wheels already completed have a very imposing effect, from their elevation on stone piers. And as the whole of the cisterns, sluices, winding apparatus, galleries, &c., are considerably elevated, they are conveniently approached in every part. Under the wheels there is a capacious tunnel, terminating at a considerable distance down the river and conveying away the tail water from the wheels.

TABLE OF SPEEDS.

Water wheel 50 ft. 0 in. = 1·5 revolutions = 4 ft. per second.

	Diameter.	Revs.		Diameter.		Revs.
Segments,	48 ft. 6 in. and	1·5	into wheel	5 ft. 6 in.	=	13·3 of shaft.
Wheels A,	18 ft. $3\frac{3}{4}$ in. and	13·3	into pinion	5 ft. 6 in.	=	44 of main shaft to mill.
Wheels B,	7 ft. 0 in. and	44	into wheel	7 ft. 0 in.	=	44 of shaft to new mill.
Wheels S,	5 ft. 9 in. and	44	into wheel	4 ft. 0 in.	=	63 of upright in new mill.

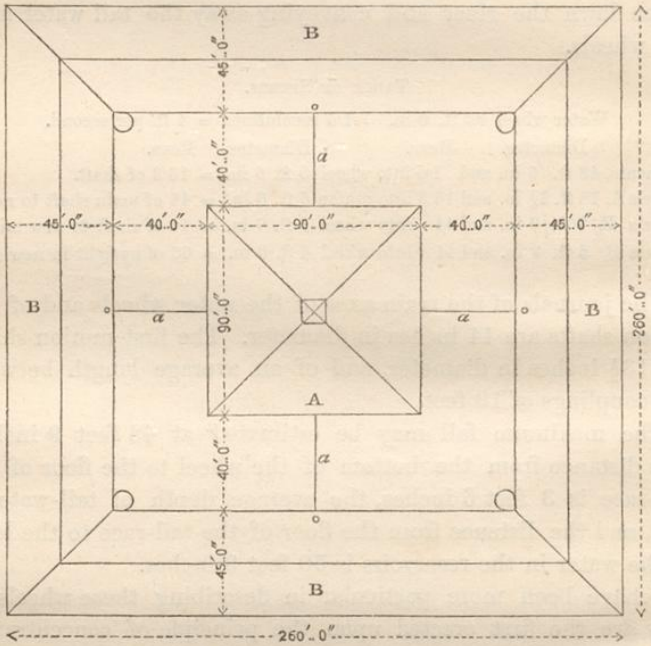
The journals of the main axes of the water wheels and of the pinion shafts are 14 inches in diameter. The first motion shafts are $13\frac{1}{2}$ inches in diameter, and of an average length between the couplings of 19 feet.

The maximum fall may be estimated at 48 feet 9 inches. The distance from the bottom of the wheel to the floor of the tail-race is 3 feet 6 inches, the average depth of tail-water 2 feet, and the distance from the floor of the tail-race to the level of the water in the reservoirs is 50 feet 9 inches.

I have been more particular in describing these wheels, as they are the first erected upon the principle of concentration and combined action. In former cases it had been the custom to erect the wheels near where the work was required, so that it was not unusual to have three or four wheels at a short distance from each other, working independently. This was the case at the Catrine works before the large wheels were erected. It was found desirable, however, in extending the works to have the whole power concentrated in one wheel-house, with a uniform fall, so as to simplify the transmission of the power to the different parts of the mills. This was effected in the manner already described with great success, and the result has been a continuous and efficient supply of power from 1827 to the present time.

Immediately following the erection of the Catrine wheels, those of Deanston, belonging to the same proprietary, were commenced. The Deanston Works were designed upon a much larger scale than even those at Catrine, as it was intended to erect eight powerful water wheels instead of four, as in the

Fig. 115.



works in Ayrshire. The Deanston Works were erected with two water wheels in the bottom room of the factory about the year 1780, and came into the hands of their present proprietors about 1798 or 1800. After the completion of the alterations in Ayrshire, a similar concentration of the power was desired for Perthshire, and I was requested to prepare both for a renewal of the old wheels and the erection of new ones on a larger and more comprehensive scale. In obedience to these instructions, an entirely new site was selected for the water power, close to the old mills on the River Teith, and provision made for an increased fall, and an improved application of the water power.

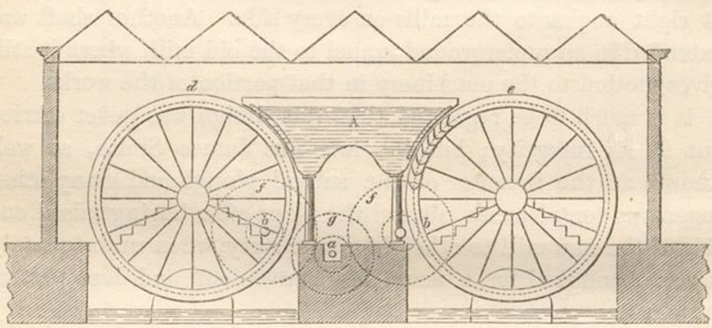
The new wheels as then designed were eight in number, and were placed together in a rectangular building adjoining the old mill, but arranged to afford power to an entirely new establishment surrounding the wheel-house, according to the annexed plan (fig. 115), in which the centre building A is the wheel-house, and the buildings B B B B, surrounding it on all four sides, and three stories in height, contained the machinery driven by the wheels. From this design it will be seen how the power, amounting to 800 horses, was given out on each side by the shafting *a a a a*, radiating from the centre of the wheel-house at right angles to the mills on every side. Another shaft was extended in an underground tunnel to the old mill, where it still gives motion to the machinery in that portion of the works.

It is much to be regretted that this design was never carried out in its integrity; but the late Mr. James Smith, so well known as the inventor of the subsoil plough and many other ingenious contrivances, altered the plans after having raised one side of the new mill to a height of one story, when unfortunately it was abandoned for a much less convenient and less perfect structure.

As respects the water wheels, the two first were erected by myself — then in partnership with my much respected friend, Mr. James Lillie,—and the last two by Mr. Smith, who, with the Cotton Mills, has since carried on considerable engineering works. The remainder have never been erected. There were, however, several novelties in the arrangement of these wheels which it may be desirable to describe at greater length. The River Teith is the principal feeder, and falls into the Forth about a mile above Stirling. The supply in ordinary seasons is about 260 cubic feet per second, and for many months in the year more than double that quantity. The original fall was about 18 feet; but, by the erection of a weir higher up the stream, and the construction of a canal three-quarters of a mile long, it was increased to 33 feet, so as to afford, except in very dry seasons, nearly 800 available horses' power. Of late years this has been increased by a copious supply from Loch Vennaquar, the surface of which has been raised at the cost of the Corporation of Glasgow, as a compensation for the water taken from Loch Katrine; which falls into the Teith for consumption in the city. From Loch Vennaquar, therefore, there is a continuous supply at all seasons.

The augmentation of the fall from 18 to 33 feet nearly doubled the power for the mills, and also the supply of water which was conveyed direct from the weir to the new wheels in the rectangular building. The water flowed into a wrought-iron pentrough A, fig. 116, supported on iron columns, and delivered the water into the wheels on each side. The wheels were 36 feet in diameter, and of the same construction as

Fig. 116.



those at Catrine. Those on one side of the pentrough, *d d d d*, gave off their power by an internal spur gearing, and those on the other, *e e e e*, by an external spur gearing on the shrouds of the water wheels; the shafts carrying the pinions, *b b*, gearing into the water wheel segments, carried also a spur-wheel, *f f*, 18 feet diameter, gearing into a common pinion *g*. This last pinion was on the central shaft *a a*, passing along the centre of the wheel-house, and giving off motion to the shafts *á á* by the bevel wheels *k*, at the centre of the wheel-house (fig. 117).

Water wheel, 4·1526 ft. circumferential velocity = 2·203 revolutions.

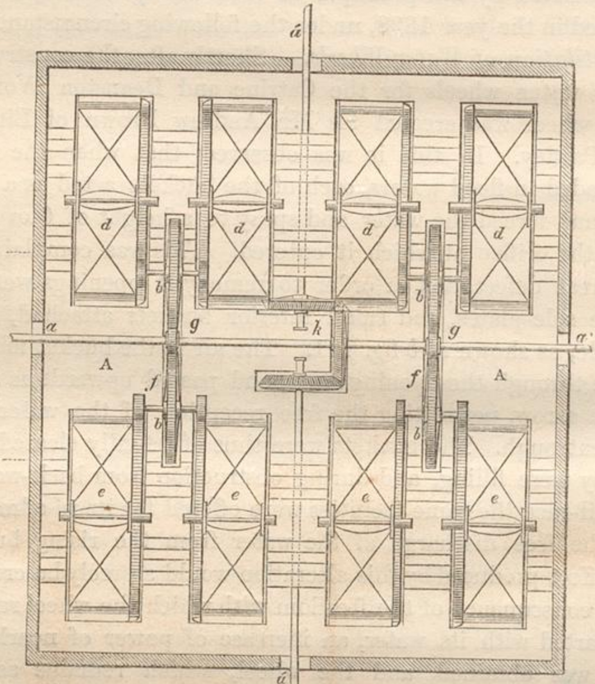
	Ft.	in.		Revs.		Ft.	in.			Revs.	
Segments,	33	8½	and	2·203	into	5	6	pinion	=	13·59	cross shaft.
Wheels *b b*,	18	2½	and	13·59	into	5	6	pinion	=	45·3	of main shaft to mill.
Wheels	6	0	and	45·3	into	4	2½	wheel	=	65	of upright in mill.

It will be observed that these high-breast wheels have the peculiar advantage of permitting the use of a sliding or folding sluice, for the admission of the water, which can be adjusted to a very variable fall. So that, at whatever height the water may stand, the velocity at which it enters the wheel will be the same, because it falls over the top of the adjusted sluice. But with this advantage they are apt to become liable to the defect of admitting the water with too much difficulty, a defect which was remedied by the principle of ventilation, which I first introduced in the year 1828, under the following circumstances:—

Ventilation of Water Wheels.—Shortly after the construction of the water wheels for the Catrine and Deanston Works, a breast wheel was erected for Mr. Andrew Brown of Linwood near Paisley. In this it was observed, that when the wheel was loaded in flood waters, each of the buckets acted as a water blast, and forced the water and spray to a height of 6 or 8 feet above the orifice at which it entered. This was complained of as a great defect, and in order to remedy it openings were cut in the sole-plates, and small interior buckets attached, inside the sole, as shown at *b* fig. 111. The air in the bucket made its escape through the openings *a*, *a*, and passed upwards as shown by the arrow, permitting the free reception of the water from the pentrough. The buckets were thus effectually cleared of air as they were filling, and during obstruction from back-water in the tail-race the same facilities were offered for its re-admission, and the free discharge of the water from the rising buckets. The effect produced by this alteration would scarcely be credited, as, in consequence of the freedom with which the wheel received and parted with its water, an increase of power of nearly one-third was obtained, and the wheel, which remains as then altered, continues, in all states of the river, to perform its duties satisfactorily.

This difficulty in the admission of the water had often been noticed by the early millwrights, and where it interfered with the working of the wheel, their remedy was to bore holes for the escape of the air in the sole-plate or the start of each bucket. Thus, in his "Mechanical Philosophy," Dr. Robison gives a similar instance to that of Mr. Brown; a wheel 14 feet in diameter and 12 feet wide was working in 3 feet of back-water and labouring prodigiously; three holes, each one inch diameter, were made in each bucket, when the wheel ceased to labour, and its power was increased one-fourth. The objection to holes in the sole-plate or buckets is a certain spilling of the water over the interior of the wheel, which cannot be avoided. But it must be remembered that air being 800 times rarer than water will escape through a hole at least thirty times faster with

Fig. 117.



the same pressure. Hence, the area for the escape of the air may be made very much smaller than the opening of the bucket.

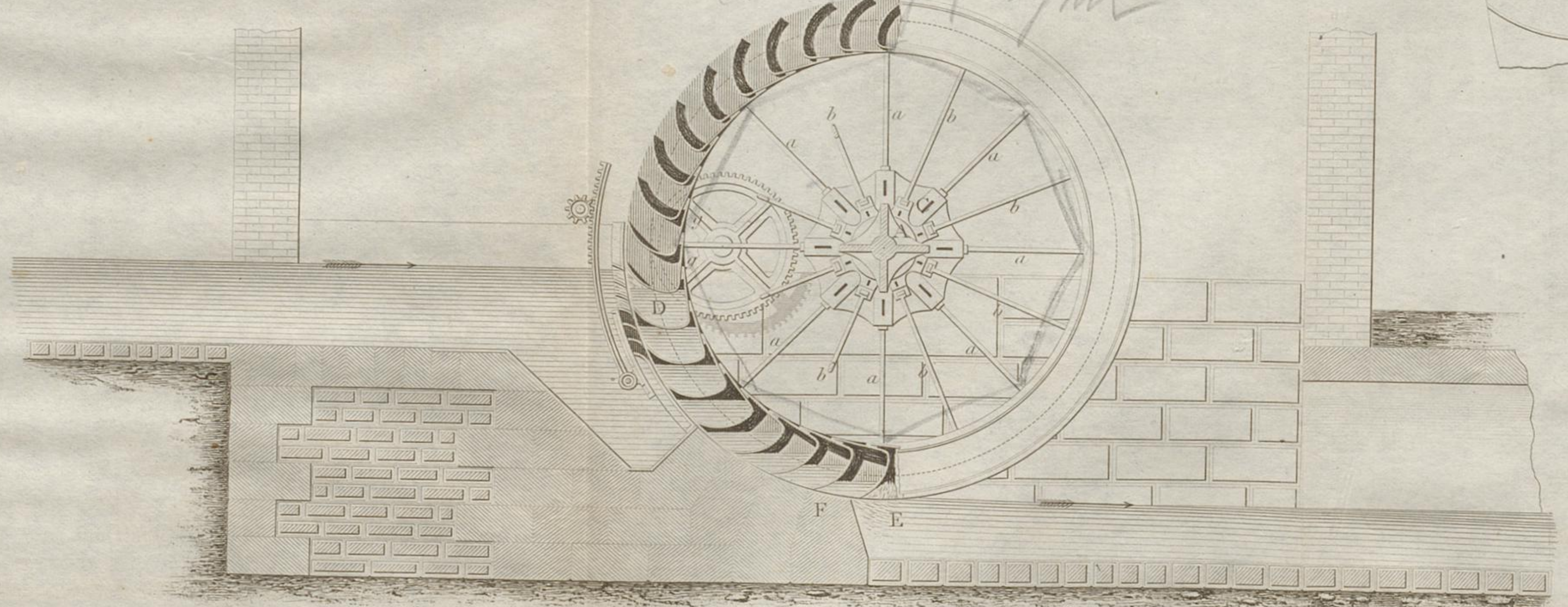
The amount of power gained, and the beneficial effects produced upon Mr. Brown's wheel, induced the adoption of the ventilating principle as a permanent modification of construction. The first wheel thus designed was erected at Wilmslow in Cheshire, and was started in 1828. It was identically the same with that shown in Plate III., and it was closely followed by a further improvement, as shown in Plate IV.

SECTIONAL VIEWS OF Mr. FAIRBAIRN'S IMPROVED WATER WHEEL.

WITH VENTILATED BUCKETS, WITHOUT SOLE-PLATE.

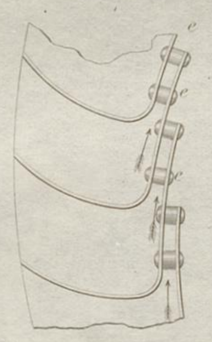
SIDE VIEW & CROSS-SECTION

Fig.1.



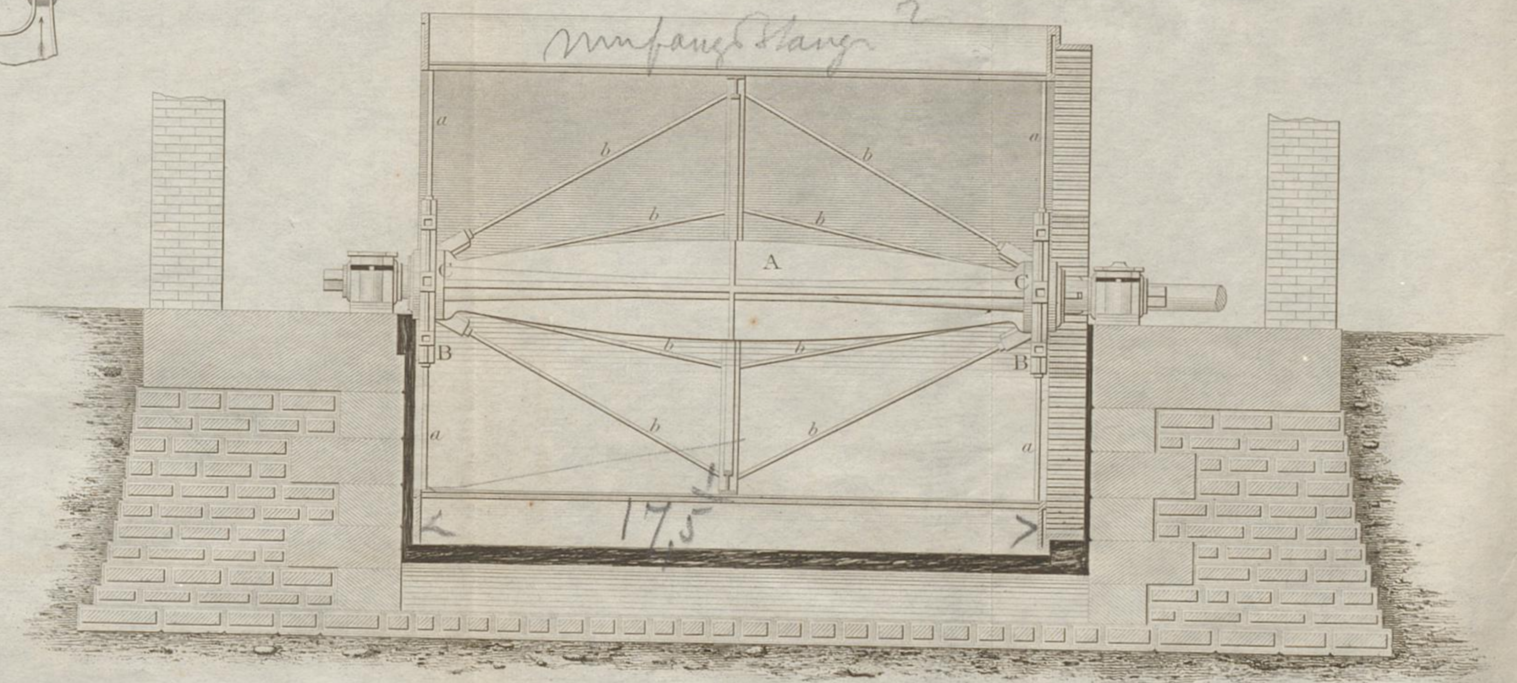
ENLARGED VIEW OF BUCKETS

Fig.3.



LONGITUDINAL SECTION

Fig.2.



Wm. Fairbairn del.

SCALE OF 0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 FEET

W. A. Beever, sc.

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Close-bucketed wheels labour under great disadvantages when receiving the water through the same orifice at which the air escapes. When, as is frequently the case, the water is discharged upon the wheel in a sheet of greater depth than the opening between two buckets, the air is thus suddenly condensed in the bucket, and re-acting by its elastic force throws back the water upon the orifice of the cistern, and thus allows the buckets to pass imperfectly filled. A similar obstruction occurred whenever the wheel worked in backwater, the water being lifted in the rising buckets, the mouths of which being under water the entrance of air was effectually prevented; and the deeper the backwater the more completely they filled with water and the greater became the difficulty in discharging. Many millwrights to remedy this were in the habit of boring holes in the sole near the start of the bucket, and of narrowing the spout or sluice so as to leave room on each side of the buckets for the escape of the air, means which to some extent remedied the evil of the spilling and sputtering of the water, but in most cases occasioned considerable waste of power, from the water being driven through the openings and falling over the interior of the wheel.

Other remedies have been attempted, such as circular tubes and boxes attached to the sole-plates; but these plans have been generally unsuccessful, owing to the complexity of their structure and the inadequate manner in which they attained the object contemplated. In fact, in wheels of this description it has been found more satisfactory to submit to acknowledged defects, than to incur the trouble and expense of partial and imperfect remedies. In the ventilated wheels about to be described, the perfect escape of the air is effected by very simple means, and great success has attended their application in situations where interruptions frequently arise from excess of backwater or a deficiency of supply.

Low-breast ventilated Wheel.—Plate III. represents a front and side view of a water wheel with ventilated buckets. Portions of the shrouding and segments are removed in order to show a section of the buckets, and the position in which they receive the water.

A is the axle or ribbed shaft, supporting the two main axes, C C, from which the wheel is suspended; B B are the projecting sockets into which the ends of the malleable arms *a a* and the diagonal braces *b b* are keyed. The arms are 2 inches, and the braces $1\frac{3}{4}$ inches in diameter. D represents the buckets, with the shuttle which regulates the admission of the water, and which is made to slide downwards. F the termination of the stone breast, and E the tail-race. This wheel, it will be seen, is arranged as a low-breast.

The principle of the construction of the buckets is more clearly shown on an enlarged scale in fig. 3, Plate III., the sole-plate being abandoned and the bucket plates bent round and prolonged upwards so as to overlap one another, leaving an opening, indicated by the arrows, for the escape of the inclosed air. The bucket plates are connected together by tubular ferules, or stays, through which a rivet is passed, and riveted on each side.

The wheel should always, as in this plate, be placed above the tail water, and not, as in the older forms of wheels (fig. 112), be carried down to the level of the tail-race floor; and the breast of wood, iron, or stone, but usually the latter, which is of so much importance for low falls in retaining the water on the wheel, should break off about ten inches from the extremity of a vertical diameter of the wheel. In fact, the benefits of this form of breast and tail-race are so great they should be strictly carried out where it is desirable to make effective use of the fall.

In high-breast wheels of twenty-five feet in diameter, and upwards, the breast is not required, as the buckets having narrower openings, and their lips extended nearer to the back of the following buckets, retain the water longer on the wheel. In this case the loss from spilling constitutes too small a percentage of the power to compensate for the expense of a lofty and close-fitting breast. In some cases the breasts have been composed of iron and wood, but in the best constructed they are of masonry, and allow little or no space between them and the wheels. It is, however, necessary to be cautious that extraneous matters do not in that case gain admission to the buckets, as by jamming between the buckets and the curb they might cause disaster.

The preceding statements, so far as relates to the method of ventilation, have been principally confined to the form of bucket and description of water wheel suitable for low falls. It will now be necessary to describe the best form of breast wheels for high falls, or falls of from one half to three-fourths of the diameter of the wheel.

High-breast ventilated Wheel.—A water wheel of this kind, constructed for T. Ainsworth, Esq., of Cleator, near Whitehaven, is represented in Plate IV. It is twenty feet in diameter, twenty-two feet wide inside the bucket, and twenty-two inches deep on the shroud. It has a close riveted sole, composed of No. 10 wire gauge iron plate, and the buckets are ventilated from one to the other, as shown on a larger scale in fig. 3. The fall is seventeen feet, and the water is discharged upon the wheel by a circular shuttle, A, which is raised and lowered by a governor as circumstances require. By this arrangement the whole height of fall is rendered available, and the water in dry seasons may be drawn off three or four feet, in order to afford time for the dam to fill in the periods during which the mill is stopped.

The power is taken from each side by two pinions working into the internal spur segments B B, and these again give motion to shafts and wheels at C C, which communicate with the machinery of two different mills, at some distance from each other.

Arrangement of Gearing.—The position of the pinion, or the point where it gears into the spur segments on the water wheel, whether internal or external, is of importance in every water wheel, but pre-eminently so in those constructed on the suspension principle, which are indifferently prepared to resist the torsive strain to which they would be subjected if the power were taken from the unloaded arc of the wheel. Water wheels of this construction, with malleable iron rods only two inches in diameter for their support, could not resist the strain, but would twist round upon the axle, and destroy the wheel.

It is necessary, therefore, in every case, to take the power from the loaded side of the wheel, as near the circumference as possible, in order to throw the weight of the water directly upon the pinion without transmitting it through a larger arc of the wheel than is absolutely necessary. For this purpose the spur pinion should be below the centre of gravity of the water on the wheel, and therefore more or less below the extremity of the horizontal diameter.

In the old water wheels, where the power was generally taken from the axle, the whole of the force passed through the arms to the point, and afterwards by a pit-wheel by some multiplier of speed to the machinery of the mill. In the improved wheels this is no longer the case: the arms, braces, and axle have only to sustain the weight of the wheel, and to keep it in shape, and the power being taken from the circumference, considerable complexity is avoided, and the requisite speed far more easily obtained.

Speed of Water Wheels.— I have usually made breast wheels for high and low falls, with a velocity between 4 and 6 feet per second at the periphery, and between these limits water wheels may be worked with economy. But for a minimum velocity I have taken 3 feet 6 inches per second, for falls of from 40 to 45 feet, and for a maximum velocity, 7 feet per second, for falls of 5 or 6 feet. The higher velocities, namely, from 5 to 6 feet per second, are now very generally adopted for the best constructed wheels, not indeed on the score of economy in the expenditure of water, but for the purpose of obtaining more easily the requisite speed under the variable conditions of supply. In this climate, where the atmosphere is so much charged with moisture, the rivers, for eight months in the year, generally afford an ample supply of water. It is for this reason that an increased velocity is given to the wheel, in order to increase the power in average conditions of supply, so as to work off the surplus rather than adapt the wheel to the minimum expenditure. It would, however, be advantageous to increase the capacity of the wheel, and work at a velocity of four feet, or at most four feet six inches per second.

Area of opening of Bucket.—The width of the opening of the bucket varies according to the point at which the water is laid on. I have made them with openings as low as 4 inches wide and as much as 20 inches, the first being for very high breast and the latter for undershot wheels, but ordinarily the width is from $5\frac{1}{2}$ to 8 inches for high breast and from 9 to 12 inches for low breast wheels. In this matter the millwright must exercise his own judgment, taking into account, 1st, the quantity of water to be delivered upon the wheel; 2nd, the position on the circumference at which the water is to be delivered, a wider opening being necessary for low-breast than for high; and 3rd, he must consider whether the circumstances of

SECTIONAL VIEWS OF Mr FAIRBAIRN'S IMPROVED WATER WHEEL

WITH VENTILATED BUCKETS & SOLE-PLATE.

AS ERECTED FOR THOs AINSWORTH, ESQr

CLEATOR NEAR WHITEHAVEN.

SCALE

0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 FEET

Fig. 1.

FRONT VIEW & SECTION.

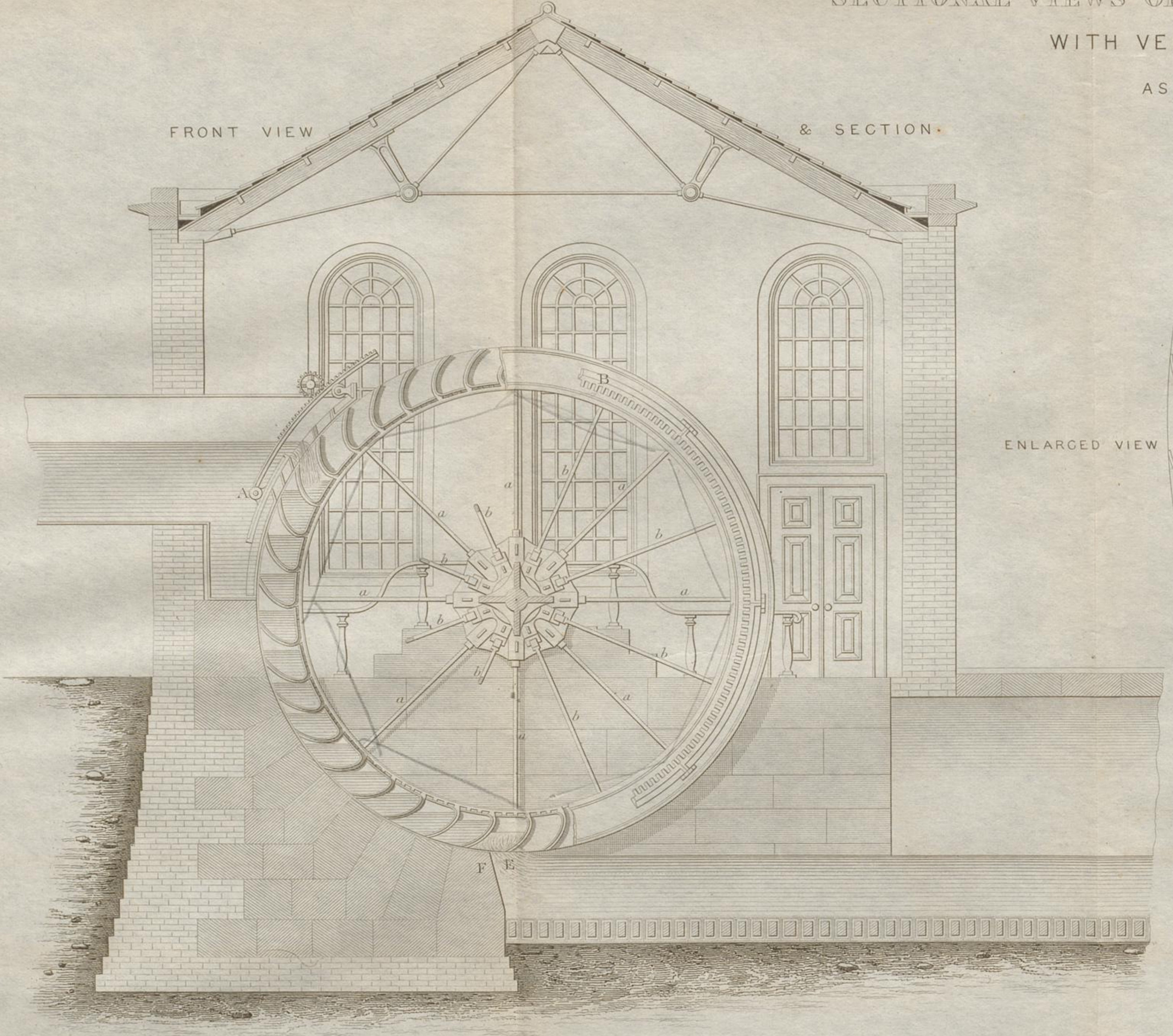


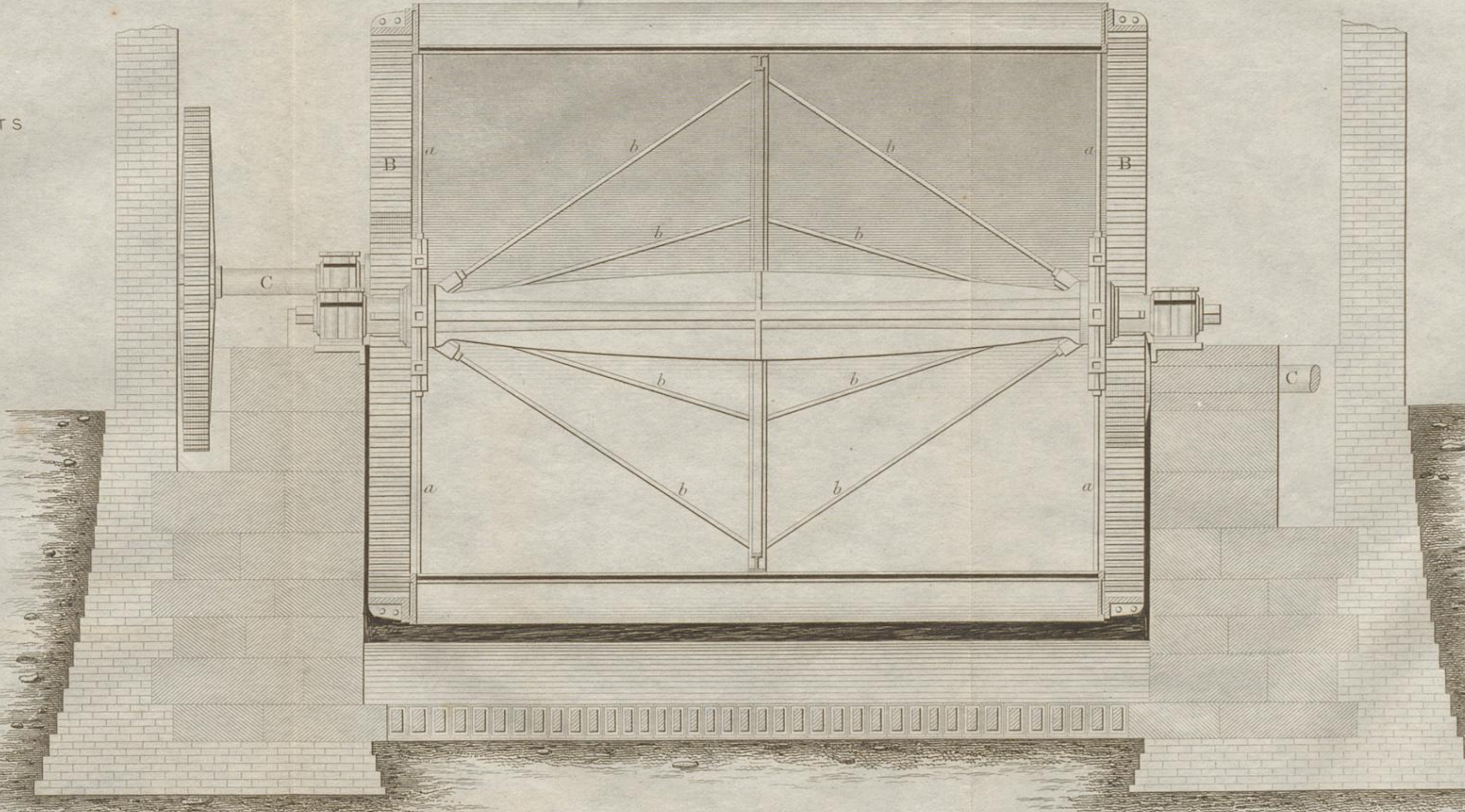
Fig. 3.

ENLARGED VIEW OF BUCKETS

LONGITUDINAL SECTION

Umfangsklangen ?

Fig. 2.



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W. A. Beever, sc.

the case in any degree limit the width of the wheel. The width of the opening must be measured perpendicularly to the direction in which the water enters the wheel; thus in fig. 104, $x\,x$ is the width of opening.

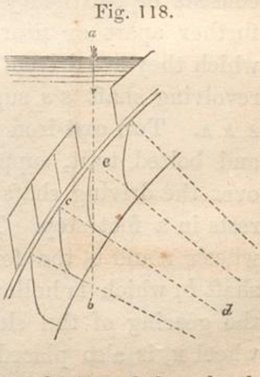
For high falls, the best proportion of the *area* of opening of the bucket, that is, the width multiplied by the length between the shrouds, is found to be such that 5 square feet of sectional area of opening is allowed for 25 cubic feet capacity in the bucket. But in breast wheels which receive the water at a height of not more than 10 degrees above the horizontal diameter, 8 square feet should be allowed for the same capacity. With these proportions the depth of the shrouding is assumed to be about 2 or 2½ times the width of the opening.

The distance of the buckets apart, measured upon the external periphery of the wheel, I have been accustomed to make from 1 foot to 1 foot 6 inches, low breast being somewhat further apart in general than high breast. This proportion fixes the number of buckets in the wheel according to the following table:—

				No. of buckets.		
For wheels	10 feet	diameter,	from	20	to	30.
,,	20	,,	,,	40	,,	60.
,,	30	,,	,,	60	,,	90.
,,	40	,,	,,	88	,,	120.
,,	50	,,	,,	120	,,	150.
,,	60	,,	,,	130	,,	180.

In setting out the curve of the water wheel bucket in breast wheels, a line $a\,b$ may be drawn cutting the external periphery of the shroud at the point and in the direction in which it is intended that the water shall strike the wheel after passing the guide plates of the pen-trough sluice. If we then measure a distance c equal to the distance of the buckets apart, and from the centre e draw the radius $d\,c$, the line $a\,b$ will be nearly the direction of the lip of the bucket and $c\,d$ the direction of its start, and the curve must be drawn connecting these lines according to the judgment of the millwright, making some allowance for the velocity of the wheel.

Fig. 118.



The Shuttle.—The shuttle of these wheels requires a slight notice. The front of the pentrough is of cast-iron, in the form of an arc closely fitting the periphery of the wheel, with an opening extending from side to side for the passage of the water to the buckets. This opening is made of such a breadth and is placed in such a position that when the water in the pentrough is highest it will flow upon the wheel near the top, and when the water is lowest it will still be able to enter the buckets near the bottom. This opening is then fitted with inclined guide plates, arranged so as to prevent the water in entering striking against the sole plate or the back of the succeeding bucket. Over the guide plates is a door, or closely fitting sluice, which slides up or down, according to the height of the water in the pentrough, so as to admit a thin sheet of water flowing over its upper edge through the guide plates into the buckets of the wheel. By this arrangement it will be seen that the water is always drawn off at its highest level and the fall economised to the utmost extent. Racks are fitted to the back of the sluice with pinions, by which its position is altered, and the quantity of water flowing on the wheel adjusted.

In the Catrine wheel, Plates I. and II., the pentrough consists of cast-iron plates bolted together and resting on beams supported on one side by the wall of the wheel-house and on the other on columns.

Figs. 119 and 120 represent the water wheel governor, a very ingenious arrangement, similar in principle to that of the steam-engine, but adapted in its details to a different purpose. It consists of two heavy balls which in revolving take a position further apart or nearer together, according to the velocity at which they are driven. These balls are swung upon the vertical revolving shaft *s s* supported in the strong cast-iron framing A A A. Two cast-iron brackets B B on either side of the frame, and bolted to it, support between them a bridge C C, passing over the driving shaft and clutch box, on which the shaft *s s* rests in a foot step. This vertical shaft is driven by the bevel wheels F and G, the former of which is keyed on the driving shaft *b*, which is hollow, to allow the shaft *a a* connected with the gearing of the sluice to pass through it. A third bevel wheel H, is also placed on a hollow shaft, and is driven by the

Fig. 119.

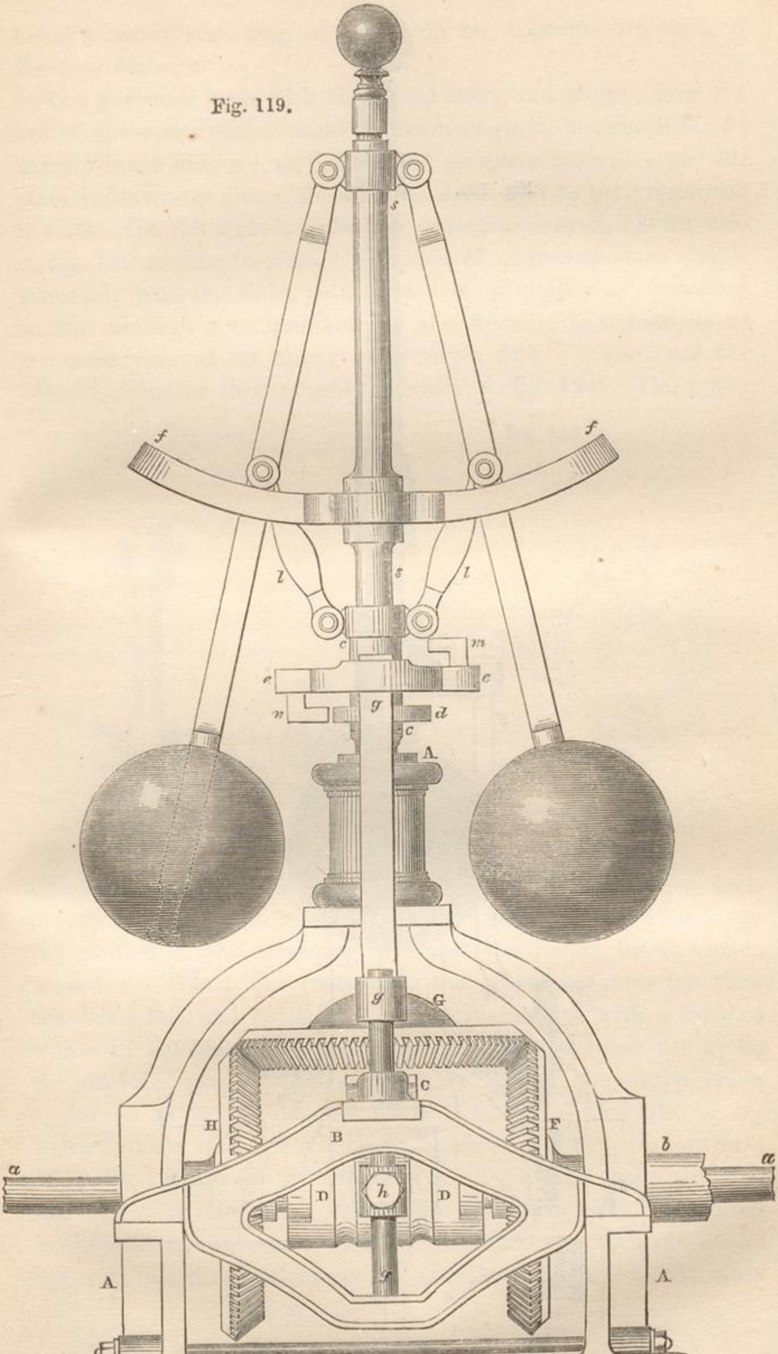
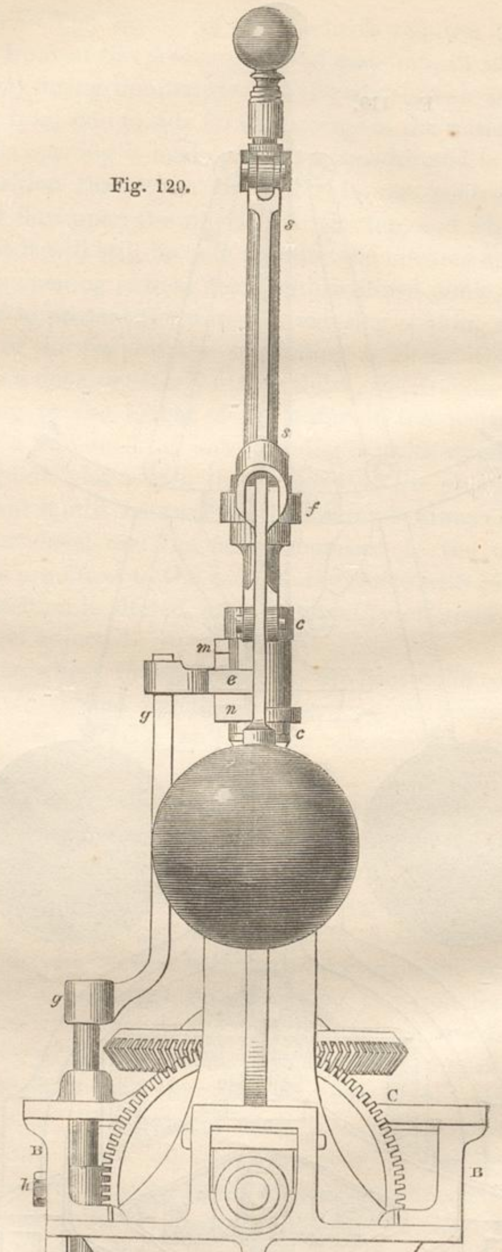


Fig. 120.



bevel wheel G, revolving of course in an opposite direction to the first wheel F.

The governor balls with the radial arms and slotted arcs *f f* are of the construction usual in steam-engines, but the links *l l* carry a brass slide *c c*, so that, as the governor balls diverge, this slide is drawn up along the vertical shaft, and as they approach it falls. On the slide is fixed the eccentric cam *d*, shown also in fig. 122 as seen in plan. This cam of course revolves simultaneously with the slide, balls, and vertical shaft *s s*. Attached to the bracket B on one side of the framing is a bent lever *g g g* carrying at its upper extremity a fork *e e*, and near the bottom a similar fork placed vertically, *h*, fig. 121. The upper

Fig. 121.

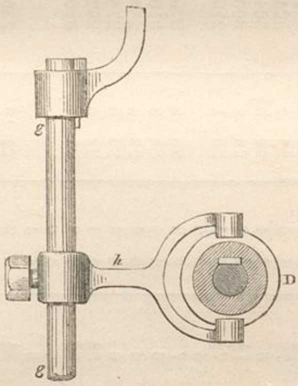
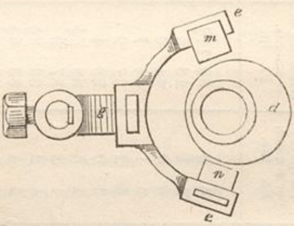


Fig. 122.



fork is moved by the revolving eccentric *d*, the lower fork moves a clutch box which slides backwards and forwards on the shaft *a a*, and engaging alternately with the bevel wheels H and F. When the motion of the wheel becomes too slow the balls fall and bring the cam *d* in contact with a knee of iron *n* in the upper fork *e e;* this causes the clutch D D to be thrown into gear with the bevel wheel H, and the clutch being keyed so as to slide on the shaft *a a*, causes that also to revolve and the sluice or shuttle to be lowered. On the contrary, when the motion of the wheel is too rapid, the balls diverge, the cam *d* is raised and strikes the upper knee *m*; the clutch is then thrown into gear with F; the shaft *a a* revolves in the opposite direction and causes the shuttle to be raised. At other

TABLE OF PROPORTIONS OF WATER WHEELS.

Diameter of shrouds. Ft. in.	Fall. Ft. in.	Depth of shrouds in inches.	No. of buckets.	Opening in buckets in inches.	Speed of periphery per second. Ft. ins.	Diameter of segments. Ft. in.	No. of Cogs.	Pitch in inches.	Breadth in inches.	Remarks.
60 4	56 4	14	144	3¾	4 0	59 11¾	912	2½	10	External spur.
50 0	48 0	15	120	6	4 1	48 6	560	3¼	15	
46 0	42 0	10	144	4½	4 0	44 6	560	3	10	
46 0	41 10	13	120	4½	4 1	44 7	480	3½	12	
40 0	36 0	18	80	6½	4 2	37 9	432	3½	14	
39 9	35 4	12	112	5	4 0	39 4	592	2½	8	External spur.
40 0	36 2	10	112	4¾	4 3	38 5¼	480	3	9	
40 0	36 0	18	80	6½	4 2	37 9	432	3¼	14	
36 0	32 0	18	80	7	4 2	33 8½	384	3¼	15	
36 0	31 8	16	80	7	4 4	33 6¼	384	3¼	14	
32 10	28 3	13	96	5½	4 4	30 7½	384	3	9	
30 0	27 0	12	96	5½	4 4	28 9	360	3	10	
30 0	24 0	18	72	7	4 6	28 2	324	3¼	14	
30 6	27 0	15	84	6	4 4	29 0½	364	3	12	
28 6	23 0	18	60	7½	4 6	26 2	300	3¼	13	
28 0	24 9	22	60	7½	4 5	25 1	216	4⅜	14	
26 0	22 8	10	70	5	4 6					
26 0	24 0	18	60	7	4 2	23 8	270	3¼	12	
26 3	22 0	14	70	6	4 4	24 8	310	3	10	
24 0	20 0	14	70	6	4 6	22 4	280	3	10	
24 0	21 0	14	60	6	4 6	22 0	276	3	10	
24 0	20 2	20	48	8½	4 4	22 0	252	3¼	14	
22 9	19 6	8	80	3¾	4 6	21 6½	390	2⅝	6	
22 6	18 10	14	60	6	4 0					
22 3	20 0	14	70	6	4 4					
22 0	20 0	10	70	4¼	4 6	21 10	330	2½	6½	External spur.
22 0	18 3	19	48	7	4 8					
22 0	19 6	19	48	8	4 10	20 3	232	3¼	12	
21 0	17 5	16	50	7	4 7	20 0	250	3	9	External spur.
20 0	17 0	20	48	7½	4 6					
20 0	16 0	22	40	8	4 4	17 10½	224	3	12	
20 0	17 6	14	56	5½	4 6	18 6¼	232	3	9	
19 9	17 0	10	60	4½	4 7	18 6½	280	2½	8	
19 0	16 4	12	48	5	4 10	17 7⅜	264	2½	10	
19 0	16 0	22	42	10	5 6	16 1⅞	174	3½	13	
18 0¼	15 0	15	48	6	4 9					
18 0	14 0	20	32	13	5 4					Ventilated.
18 0	15 0	20	32	11½	5 10	16 0	200	3	14	Ventilated.
18 0	15 9	12	48	5½	4 3	16 6	248	2½	8	
18 0	13 4	22½	32	21	6 0					Ventilated.
18 8	15 0	20	40	15½	5 6	16 9	192	3¼	14	Ventilated.
16 6	13 0	24	36	12½	5 6	12 3	132	3½	10	Ventilated.
16 0	10 6	18	40	15	5 6	15 4¼	192	3	12	Floats, ventilated, external spur.
16 0	12 8	15	40	9	4 10					
16 0	14 0	16	40	7¼	5 0	14 0¾	176	3	12	
16 0	12 0	24	32	16½	5 8					Ventilated.
16 0	11 4	21	32	17	6 0	14 0¾	176	3	9	Ventilated.
16 0	12 8	28	32	17	6 4					Ventilated.
16 0	11 9	24	32	15	5 10					
15 6	12 0	20	40	10½	5 8					Ventilated.
15 0	12 0	16	32	10½	5 8	14 6	168	3¼	10	
15 0	13 2	12	40	6	4 8					
15 0	13 0	18	36	7	5 3	13 10	174	3	10	Ventilated.
14 0	10 0	15	36	9	5 6					
14 0	8 6	15	36	9¼	5 10					Ventilated.
14 0	11 8	24	36	4¼	5 2					Ventilated.
12 6	10 0	10	36	4½	5 0					
12 1	8 3	12	30	6	5 1	11 6	174	2½	7	
12 0	8 0	15	36	6¼	5 0	10 10	162	2¼	7	
12 0	10 4	8	36	4	4 8	11 3½	282	1½	4½	
10 0	8 0	8	36	3½	4 6					
7 0	5 0	6	32	2⅞	5 0	6 5½	248	1	2½	

times, when the motion does not require adjustment, the clutch is disengaged from both wheels and the whole of the winding apparatus is stationary.

This arrangement of governor is exceedingly compact and effective and a great improvement on the original condition in which I first found it, with rollers and reversing pulleys. It is free from the objection to which those governors are open which directly bring the sluice gearing into operation and retain it so by their momentum.

As examples of the speed at which this part of the machinery is worked, I subjoin a few examples that are working successfully:—

Governor shaft, . . 36 revolutions per minute.
Rack shaft, from 0·0314 to 0·058 revolutions per min.

There is usually a worm on the shaft *a a*, working into a wheel on a cross shaft; on the cross shaft a second worm working into a wheel on the rack shaft; and a small pinion 8 inches in diameter on the rack shaft gears into the rack upon the sluice. This rack should be jointed to the sluice at the middle, and should be of such a length that the rack shaft and pinion can be placed out of water above the pentrough. But the details of the gearing and shafting by which the motion of the governor is transmitted to the sluice vary with the position of the governor and the circumstances of each particular locality, and they must therefore be left to the millwright's own judgment. Only it is important to observe that the motion of the sluice should in every case be slow, as in the above examples, or the acceleration or retardation in the supply of water will cause an irregular motion first faster and then slower in the wheel, conditions inadmissible where machinery is employed.

In designing a water wheel the first important consideration is the height of the fall; this taken in conjunction with the intended outlay will fix the diameter of the wheel. We must next determine the form of bucket as already detailed. Then the quantity of water per second in cubic feet must be ascertained, and this will determine the necessary capacity of the bucket and the consequent breadth of the wheel. Here we have to consider also, 1st, that the bucket is not to be more than $\frac{1}{3}$ or $\frac{1}{2}$ filled; and, 2nd, the rate of revolution of the wheel which determines the number of buckets passing the shuttle per second. (p. 137.)

Suppose a wheel, having 5 feet peripheral velocity per second, supplied with 3000 cubic feet of water per minute, and the breadth of which has to be determined so that the buckets shall be only one-half filled:—

Let depth of shroud = . . . 14 inches.
distance between buckets = . . 14 inches.
section of water in bucket when full,
at the pentrough = . . . 144 sq. ins.

Here five buckets pass the sluice per second, and each must contain $\frac{3000}{5 \times 60} = 10$ cubic feet of water per second; but they are to be only one-half filled when containing this quantity of water, hence their capacity must be 20 cubic feet. Their sectional area is 1 square foot, and hence 20 feet is the breadth necessary for the wheel.

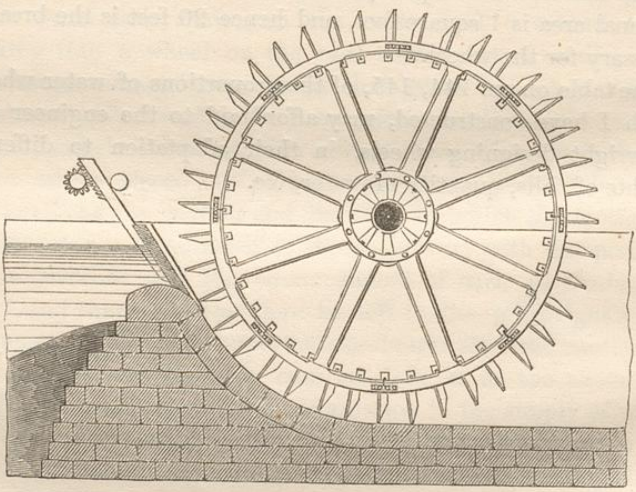
The table on pp. 144, 145, of the proportions of water wheels which I have constructed, may afford aid to the engineer and millwright designing wheels, in their adaptation to different heights of falls, quantity of water, &c.

CHAP. IV.

ON THE UNDERSHOT WATER WHEEL.

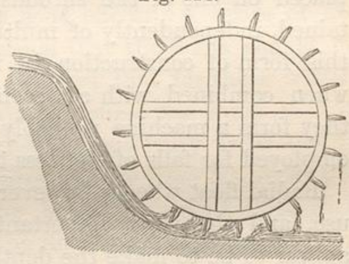
Before the introduction of iron, undershot water wheels were frequently employed, and were in almost every instance constructed with straight radial floats, as in the annexed sketch,

Fig. 123.



the water being discharged against the float-boards, as it rushed with considerable velocity underneath the shuttle. This was the invariable practice down to Smeaton's time even, the principle being to employ the impulse of the fluid stream, and not its gravity or weight. Indeed, there appeared to be an impression that this was the more effective and economical mode of application, and probably arose out of the circumstances of the original employment of water as a moving power. The earliest wheels of which we read are undershot wheels placed between two boats in a flowing stream, and driven by its impulse, and in Smeaton's own time the works for the supply of water to London obtained their power from some magnificent examples of precisely similar wheels, placed in the tidal stream rushing between the clumsy piers of the old London Bridge. In the old time it was no doubt an advantage to have the prime mover working at a considerable velocity, and an overshot wheel will not do this effectively. Hence wheels were sometimes built of the form shown in fig. 124, the water being carried down from the top of the fall so as to strike the radial floats of the wheel at a very high velocity. Such a wheel is described in Smeaton's Reports.

Fig. 124.



The earliest great advance in the perfecting of the water wheel was effected mainly by Smeaton, and we owe to him the first experimental inquiries on the effect and proper velocity and proportions of water wheels. In all the various applications of water, experimental researches have hitherto been the principal means of advance, and in no department has more labour and talent been expended in such inquiries; the result is, that our hydraulic machinery of the present day is as perfect, and yields as high a proportion of the power to the actual fall of water, as we can ever hope to obtain.

In my own practice I have been accustomed to employ water even for very low falls, solely by gravity, using the arrangement already described, as a low breast wheel, when treating of ventilation, and which is shown in detail in Plate III. This wheel is 16 feet in diameter, 17 feet 6 inches between the shrouds, and is adapted to a fall varying from 5 to 8 feet, according to the condition of the river. The water flows into the wheel at its highest level, over a sliding sluice of precisely the same construction as in high breast wheels; it is retained in the buckets to the bottom of the fall, by the cast-iron and stone breast fitting accurately to the edge of the buckets. The advantages of this construction are manifest, as the water expends its full force on the wheel from the very top of the fall, the buckets being well ventilated, and having a curvature adapted to the position in which they receive the water. By these means, a greatly increased duty is obtained as compared with the wheels with radial floats acted upon by impulse or gravity, or by both. Besides, with this form of wheel, the spider, or suspension principle of construction, may be adopted, and the power taken off at once from an internal segmental spur-wheel, placed on one of the shrouds, and a high velocity at once obtained, independently of multiplying gear. The advantages of this form of construction in iron wheels are very great, and, when combined with an economical application of the water, they form a machine probably as effective as any which can be employed for falls of not less than 5 feet.

Radial float wheels, however, constructed of wood are still in use, and the most important directions in respect to these appear to be to make the depth of the floats large, as compared with the thickness of the lamina of water which strikes them; to place the sluice as close as practicable to the floats; to contract somewhat the aperture of the sluice, and to expand the tail-race immediately beyond the vertical plane passing through the axis, to allow the water escaping from the floats to diffuse itself in the tail-race, and pass freely away. These directions, with the following practical formula for fixing the diameter of the wheel, we have from the dissertation on water wheels in the Engineer and Machinist's Assistant.

Let u = the velocity of the extremity of the floats; N the number of turns desired per minute; h = fall in feet. Assume $u = 2{\cdot}4 \sqrt{h}$ for a maximum effect, then the diameter expressed in terms of the velocity and height of fall will be $19{\cdot}1 \times \frac{2{\cdot}4 \sqrt{h}}{N} = \frac{46}{N} \sqrt{h}$ nearly. Thus supposing the height of fall = h = 4 feet; number of turns required per minute = N = 8; then the diameter = $\frac{46}{8} \sqrt{4} = 11\frac{1}{2}$ feet nearly. Twelve to twenty-five feet is the usual range of diameter for undershot wheels, and the same writer considers 12 to 16 feet to be the most effective; in my own practice, I have found from 14 to 18 feet perform the best duty. Feathering, or inclining the floats, does not appear to increase the useful effect.

The number of floats is usually equal to $\frac{4}{3}d + 12$, where d is the diameter in feet. The thickness of the vein of fluid striking the floats may be from 6 to 9 inches, and the depth of the floats from 18 inches to 2 feet.

M. Poncelet, one of the first authorities on Hydraulic Machines, and the first writer on Turbines, has contrived a very important modification of the undershot wheel, which has been used on the Continent with very good effect. A series of experiments led him to the conclusion that the floats should be curved instead of plane, and he deduced that for these wheels the velocity which gives a maximum effect was equal to 0·55 the velocity of the current, whilst it may vary from 0·5 to 0·6. He found the dynamic effect to vary from 50 to 60 per cent. of that of the water, being better for small falls with large openings at the bottom of the flood gate, and less for deep falls with small openings.

For describing the curve of Poncelet's floats, let $c\,c$ be the external circumference, and $a\,r$ the radius of the wheel; take $a\,b = \frac{1}{3}$ to $\frac{1}{4}$ the fall, and draw the inner circumference of the shrouding; let the water first strike the bucket at the point a and in the direction $d\,a$, draw $a\,e$ perpendicular to $d\,a$, so that the angle $e\,a\,r$ will be 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.

Fig. 125.

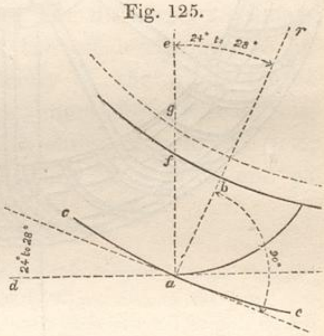
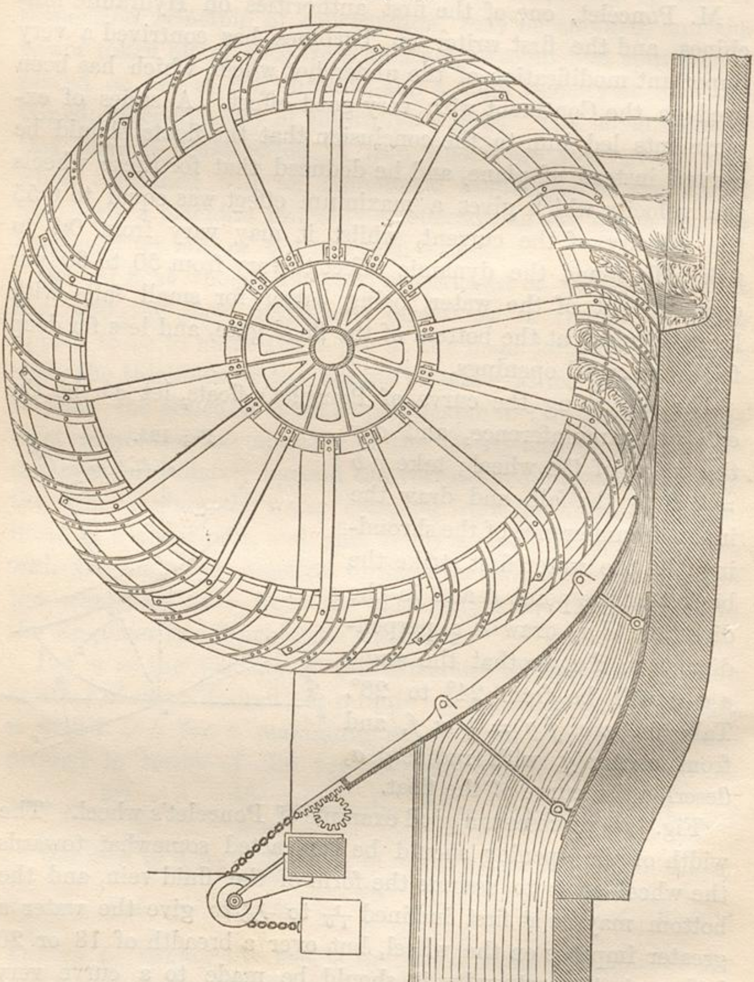


Fig. 125 represents a good example of Poncelet's wheel. The width of the opening should be contracted somewhat towards the wheel so as to assume the form of the fluid vein, and the bottom may be at first inclined $\frac{1}{10}$ to $\frac{1}{15}$, to give the water a greater impetus on the wheel, but over a breadth of 18 or 20 inches at the extremity it should be made to a curve very accurately fitting the periphery of the wheel.

So also the tail-race may be expanded in width and depth to keep the wheel clear of backwater. The buckets are made of wrought iron of the requisite curve, riveted to the shrouds on each side, and the sole plate is altogether dispensed with ; as no resistance is opposed by the air, the buckets are made more numerous than in breast or undershot wheels, and as the wheel

Fig. 126.



carries no weight of water, it may be made comparatively light. For the number of buckets for wheels of from 10 to 20 feet in diameter, we may take

$$n = \frac{8}{5}d + 16$$

Thus for a wheel 15 feet in diameter,

$$n = \frac{8 \times 15}{5} + 16 = 40$$

The wheel shown in the figure is 16 feet 8 inches in diameter, and 30 feet wide, and is driven by a fall 6 feet 6 inches high, yielding 20,000 cubic feet per minute. With a circumferential velocity of 11 or 12 feet per second, it afforded 140 horses' power.

This wheel gives a useful effect of 50 to 60 per cent. of the water power employed when well constructed, and may be used with advantage for falls not greater than about 6 feet. Above this the low breast wheel is certainly more advantageous and costs less.

Poncelet made some experiments on wheels of this class, with the friction break. The wheel was 11 feet diameter, 28 inches wide, and with 30 floats. He found the efficiency equal to 52 per cent. when the ratio of the velocity of the wheel to the water was 0·52. Morin has also experimented on these wheels, and for falls of from 3 to $4\frac{1}{2}$ feet, with sluice openings of 6, 8, 10, and 11 inches, he found the efficiency 52, 57, 60, and 62 per cent. respectively.*

* In a conversation with General Poncelet on this subject I found that the wheel which bears his name gives a duty of nearly 60 per cent. of the water employed. This is about the same as my own wheel with ventilated buckets for low falls, where the sole is entirely dispensed with. There is, however, this difference, namely, that in the Poncelet wheel the water is discharged upon the floats from *under* the sluice, whereas, in that of the ventilated wheel, it is discharged into buckets *over* the sluice from the upper surface of the fall.

CHAP. V.

ON TURBINES.

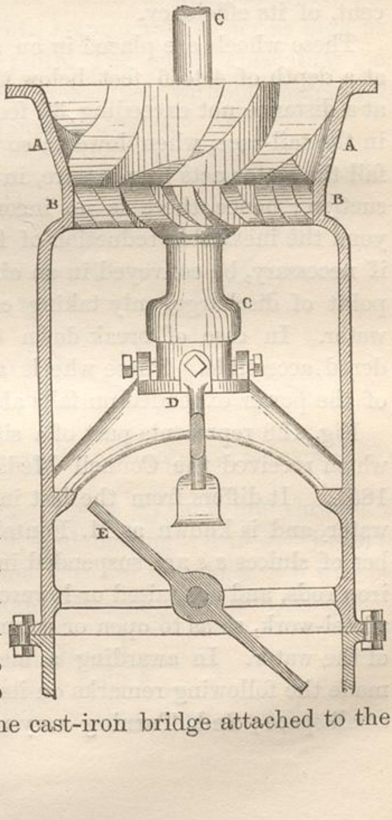
It will be impossible in the present work to enter into details on the theory and construction of the immense variety of prime-movers known under the name of turbines, the development of the principles of which we owe chiefly to continental mathematicians. Two varieties of horizontal wheels or turbines have long been employed on the Continent, which although ill-devised and ineffective, yet presented evident advantages in their small size, cheapness, and simplicity of construction. These are known in France as *roues à cuves* and *rouets volants*, the former being a small wheel revolving on a vertical axis, and having inclined curved vanes or buckets arranged radially. It is placed in a pit so that the water passing vertically through it should act by pressure and reaction on the buckets. The *rouet volant* differs from this in having the water applied to the wheel at a small part only of the periphery, so as to drive the wheel by impulse. These wheels of from 3 to 5 feet in diameter with nine to twelve buckets are usually made of cast-iron, and fixed upon a lever foot bridge, so that they can be slightly raised or depressed. The running millstone is fixed on the upper extremity of the vertical axis, so as to obviate the use of any gearing or belting. In regard to efficiency, the *roues à cuves* yield about 27 per cent. and the *rouets volants* about 30 to 40 per cent. of the water used.

General Poncelet was the first to demonstrate the principle and superior advantages of the turbine, and in 1827 M. Fourneyron recalled public attention in France very forcibly to the construction of the horizontal wheels by a turbine very happily conceived and executed. For this invention he received in 1833 a prize of 6000 francs; and the principles of his machine have been investigated, and its superiority proved, by the ablest continental experimenters on hydraulics. In its present form it is equal in efficiency to the best hydraulic machines, and in many circumstances is very advantageously employed. Since then the manufacture of these turbines in countries where water power is much depended upon has assumed considerable importance, and very numerous modifications of its form and construction have been adopted.

1. *Turbines in which the water passes vertically through the wheel.*

Wheels of this class are composed of two annular cylinders, the upper fixed and the lower revolving on a vertical axis. The upper is fitted with guides to direct the water most effectively against similar curved vanes or buckets, turned in the opposite direction, in the lower wheel. The water passes from the reservoir or cistern placed over the upper cylinder, vertically downwards, acting on the revolving wheel by pressure as it glides over the surface of the vanes.

Fig. 127.



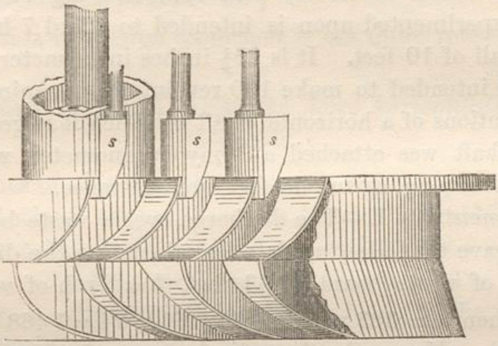
Burdin about 1826 invented a turbine of this description (*turbine à evacuation alternative*), the efficiency of which was as much as 67 per cent. of the water power expended.

Fig. 127 represents Feu Jonval's turbine (known also as the Koechlin turbine). The fixed wheel is shown at A A, the revolving wheel at B B. The wheels consist of cast-iron rims having wrought-iron guides grooved and riveted to them. The running wheel is keyed on the shaft C C, which is supported on a step D, firmly fixed by screws on the cast-iron bridge attached to the cylinder forming the tail-race. The regulation of the water is effected partly by a valve E resembling the throttle-valve of a steam-engine and placed beneath the wheel, or in some cases by a sluice at the opening of the conduit into the tail-race. This method, when much variation of power is required, reduces the efficiency of the wheel, but it has the merit of great simplicity and facility of construction. In the construction represented the vane carries outside the cylinder in which it is placed a wheel, acted on by a worm from a hand-wheel placed at any convenient point above the upper cistern. There are also employed movable divisions by which part of the inner periphery of the revolving wheel is enclosed, and the water passes through a narrower annular aperture on the external periphery. This arrangement is said to have operated effectively in America, so that a wheel giving 60 H. P. in wet seasons can work at 40 H. P. in dry seasons, without losing more than 15 or 16 per cent. of its efficiency.

These wheels are placed in an air-tight cylinder, for low falls at a depth of 4 to 6 feet below the surface, and for high falls at a distance not exceeding 30 feet above the level of the water in the tail-race, when lowest; so that in the upper part of the fall the water acts by pressure, in the part below the wheel by suction; hence, there is no inconvenience from backwater beyond the inevitable reduction of fall, and the waste water may, if necessary, be conveyed in an air-tight pipe to any convenient point of discharge, only taking care that its mouth be under water. In case of break down the wheel is very easily rendered accessible. These wheels are said to yield 75 per cent. of the power expended on falls above 12 feet.

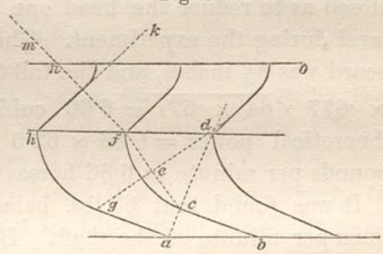
Fig. 128 represents part of a similar turbine by M. Fromont, which received the Council Medal at the Great Exhibition of 1851. It differs from the last in the method of regulating the water, and is known as M. Fontaine Baron's turbine. A number of sluices *s s* are suspended in the fixed wheel by wrought-iron rods, and are raised or lowered simultaneously by means of wheel-work, so as to open or contract the orifices for the passage of the water. In awarding a medal to this turbine, the jury made the following remarks on its merits:—“ 1st. It occupies a small space; 2nd. Turning very rapidly, it may, when used for grinding flour, be made to communicate the motion directly to the millstones; 3rd. It works equally well under great and

Fig. 128.



small falls of water; 4th. It yields, when properly constructed, and with the supply of water for which it was constructed, a useful effect of 68 to 70 per cent., being an efficiency as high as any other hydraulic machine; 5th. The same wheel may be made to work at very different velocities, without materially altering its useful effect."—*Reports of Juries.*

Fig. 129.



In designing a wheel of this description, we must take a distance $a\ b$ equal to the distance between the floats, or $\frac{6{\cdot}29 \times \text{radius}}{\text{No. of vanes}}$. Take the angle $a\ b\ c = 15°$ to $20°$, and draw $a c$ perpendicular to $c\ b$. Lay off $d\ c f$ equal to $\frac{\delta + \beta}{2}$ where δ is the angle $a\ b\ c$, and β is taken arbitrarily equal to $100°$ to $110°$. Bisect $c f$ in e, and through e draw $g\ d$ perpendicular to $c\ f$, and cutting $c\ d$ in d. From d with radius $d\ c$, or $d\ f$, draw the arc to which $c\ b$ will be a tangent. For the guides, take the angle $f\ h\ k = a$, so that

$$\text{Cot } a = \text{Cot } \beta + \frac{1}{\sin \delta}$$

draw $f\ m$ perpendicular to $h\ k$, cutting the top of the guide wheel $n\ o$ in n; from n draw arc, touching $h\ k$. These directions are from Weisbach.

In America the Koechlin turbine has been experimented upon by the Franklin Institute, with the following results:—The turbine experimented upon is intended to afford 7 horse power under a fall of 10 feet. It is $21\frac{1}{4}$ inches in diameter, $3\frac{1}{2}$ inches deep, and intended to make 190 revolutions per minute, giving $63\frac{1}{3}$ revolutions of a horizontal shaft to which it is geared 3 to 1. To this shaft was attached a Prony dynanometer, whose lever was 7·96 feet long, giving 50 feet circumference.

Experiment No. 1.—The discharge over a waste board in the tail-race gave the following data for calculating the discharge:—L=width of waste board=3·83 feet, h=depth of water on it, 0·74. Then $Q = \cdot 383 \times 3\cdot 83 \times \cdot 74 \sqrt{64 \times \cdot 74} = 7\cdot 468$ cubic feet per second. Hence the theoretical power$=7\cdot 468 \times 62\cdot 5 \times 9\cdot 34 \times 60 = 261{,}537$ foot-pounds per minute, $=7\cdot 92$ horses power.

It was found that at 63 revolutions per minute of the horizontal shaft 63 lbs. balanced the lever. Hence the power developed by the wheel was $63 \times 63 \times 50 = 198{,}450$ lbs. $=6\cdot 014$ horses power.

Experiment 2.—The gates from the head race were so far closed as to reduce the head one foot, and maintain it at that level during the experiment. The depth of water on the waste board was $8\frac{1}{8}$ inches, and the fall 8·41 feet. $\therefore Q = 0\cdot 39 \times 3\cdot 83 \times \cdot 677 \sqrt{64 \times \cdot 677} = 6\cdot 66$ cubic feet per second. Hence theoretical power $= 6\cdot 66 \times 62\cdot 5 \times 8\cdot 41 \times 60 = 210{,}000$ foot-pounds per minute $= 6\cdot 36$ horses power.

It was found that 63 lbs. balanced the lever at 49 revolutions per minute of the shaft. Hence the power developed by the wheel was $49 \times 63 \times 50 = 164{,}350$ lbs. $= 4\cdot 98$ horses power.

The coefficients are then for No. 1, $\frac{6\cdot 014}{7\cdot 92} = 0\cdot 76.$

" " " " 2, $\frac{4\cdot 98}{6\cdot 66} = 0\cdot 78.$

And making allowance for leakage round the waste board, the experimenters conclude that the wheel yielded 75 per cent. of the power expended.

Another experiment on a 60-horse power turbine gave the following results:—

$$\left.\begin{array}{l}\text{Effective power } 56{\cdot}30\\ \text{Theoretical power } 63{\cdot}92\end{array}\right\} 0{\cdot}88.$$

Perhaps this very large coefficient is not quite reliable.

2. *Turbines in which the water flows horizontally and outwards.*

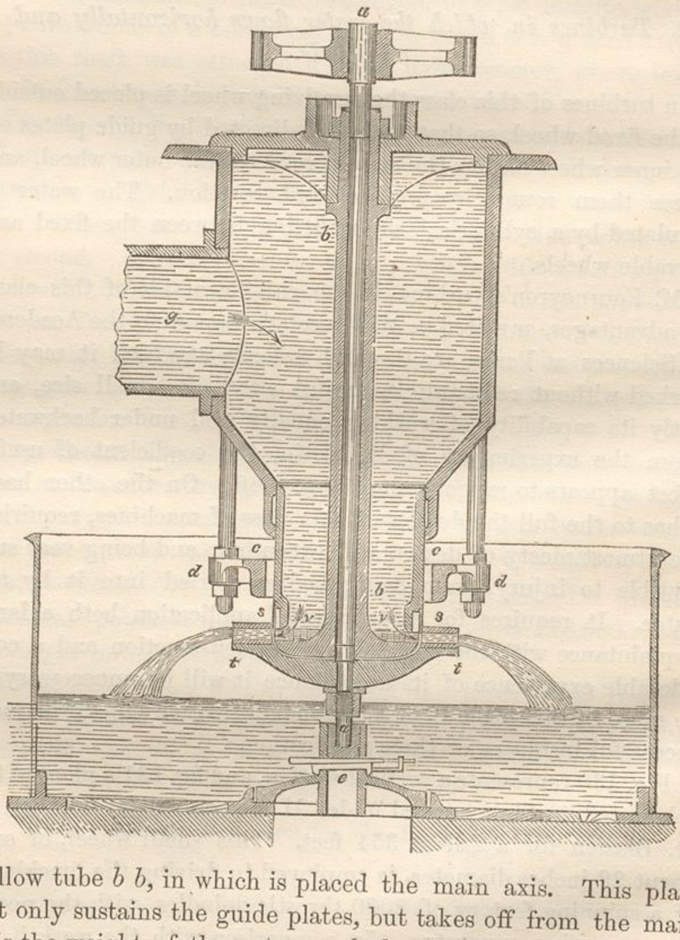
In turbines of this class the revolving wheel is placed outside of the fixed wheel, so that the water directed by guide plates on the inner wheel strikes the curved vanes of the outer wheel, and forces them round by pressure and reaction. The water is regulated by a cylindrical sluice fitting between the fixed and movable wheels.

M. Fourneyron's turbine is the chief example of this class. Its advantages, as stated in M. Poncelet's Report to the Academy of Sciences at Paris, are the high velocity at which it may be worked without reducing its useful effect, its small size, and lastly its capability of working equally well under backwater. From the experiments of M. Morin, the coefficient of useful effect appears to range from 0·60 to 0·80. On the other hand it has to the full the defects of this class of machines, requiring the utmost nicety of design and execution, and being very susceptible to injury, from small bodies carried into it by the water. It requires for its successful application both a large acquaintance with the principles of its construction and a considerable experience of its use: hence it will be unnecessary to do more in this place than select for illustration one of the most successful instances of their application.

Fig. 130 represents a vertical section, and fig. 131 a plan, of the celebrated turbine erected under M. Fourneyron's direction at St. Blazien for a fall of 354 feet. This small wheel, of only about 26 inches diameter, is employed in driving the machinery of a spinning factory of 8000 throstle-spindles, with the necessary preparing apparatus. In comparison with the work it has to perform, it is therefore of a size altogether unique.

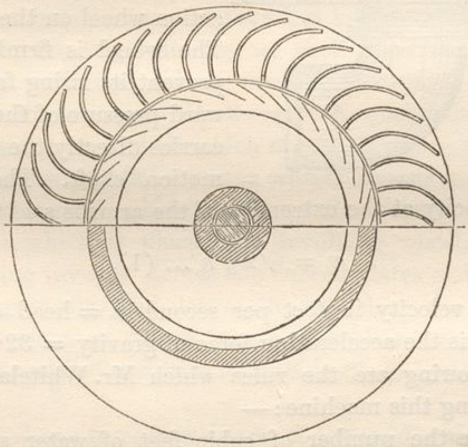
The wheel consists of a cast-iron concave plate *t t*, keyed on the main axis *a a*; on this is fixed the annular wheel *s s*, consisting of an upper and lower plate of wrought iron, in which are fixed the 36 curved diaphrams seen in the plan, fig. 131. Opposite each of these curved plates on the outer revolving wheel, there is a similar guide on the inner fixed wheel *v v*, which are carried on a massive cast-iron plate attached to the

Fig. 130.



hollow tube *b b*, in which is placed the main axis. This plate not only sustains the guide plates, but takes off from the main axis the weight of the water, and thus reduces the friction on the foot step. The cylinder *c c* slides up and down in the larger water cistern, and forms a circular sluice between the revolving and fixed wheel, by which, within certain limits, the discharge of water and velocity of the turbine can be regulated. This sluice is raised or lowered by 4 rods, *d d*, which are screwed above into the eyes of 4 pinions (not shown). These pinions all gear into one larger wheel, and in this way the 4 rods may be raised or lowered simultaneously. The supply of water is brought to the cistern by a pipe *g* of $16\frac{1}{2}$ inches

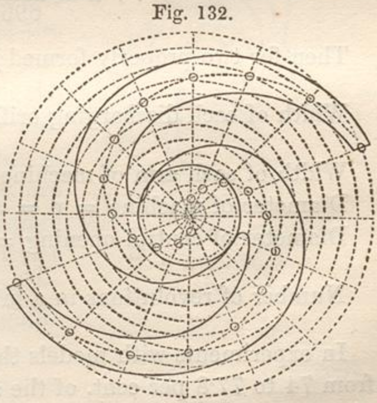
Fig. 131.



diameter, and 1200 feet in length. The spindle works on a steel pivot in a footstep adjusted by gibs and cotters *e*. This turbine makes from 2200 to 2300 revolutions per minute.

Another form of turbine, in which the flow is horizontally outwards, has been made to some extent in this country by Messrs. Whitelaw and Stirrat, and is sometimes called the Scotch turbine or reaction wheel. It is precisely on the principle of Barker's mill, and works by reaction. The principal improvement effected by Mr. Whitelaw is the form of the arms, which are curved in an archimedean spiral. Fig. 132 shows the method of striking these curves, the centre line of the arm being first drawn

Fig. 132.



and half the breadth set off on each side of it, so that the capacity of the arm increases from the extremity towards the centre in the inverse ratio of the velocity at each point.

Fig. 133.

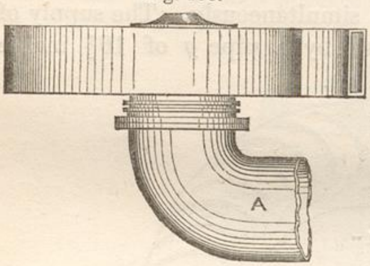


Fig. 133 shows the arrangement of this wheel, the water being brought in a pipe A, curved at bottom, so as to enter the reaction wheel on the under side. The wheel is firmly stayed to prevent its rising from the upward pressure of the water, and carries directly the vertical first motion shaft. The most effective velocity at the extremity of the arms is said to be nearly

$$v = \sqrt{2\,g\,h} \;\dots\; (1),$$

where v = velocity in feet per second, h = head of water in feet, and g is the accelerating force of gravity = 32·19.

The following are the rules which Mr. Whitelaw gives for proportioning this machine:—

Let Q be the number of cubic feet of water supplied per minute.

H, the height of the fall or head of water.

E, the useful work in units of horse-power.

$$\therefore\ \text{E} = \frac{\text{Q H}}{696{\cdot}73} \;\dots\; (1).$$

Then for two properly formed jets:—

Width of each discharging orifice $= w_1 = \sqrt{\dfrac{135\,\text{E}}{1000\,\text{H}\sqrt{\text{H}}}}$.

Width of each arm of machine $= 4\,w_1 = w_2$.

Diameter of machine $= \text{D} = 50\,w_1$.

Diameter of central opening $= 10\,w_1$.

Number of revolutions in a minute $= \dfrac{149{\cdot}4338\sqrt{\text{H}}}{\text{D}}$.

In experiments with models the wheel is said to have realised from 74 to 77·8 per cent. of the available power.

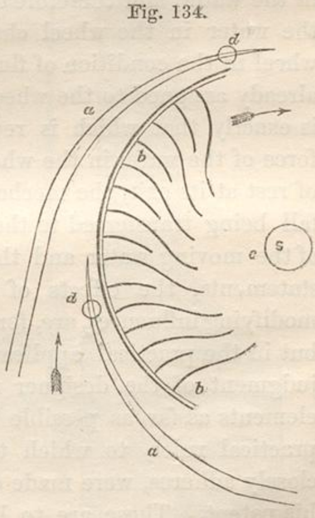
3. *Turbines in which the water flows horizontally inwards; vortex wheels.*

We owe the invention of this class of turbines to one of my own pupils, Mr. James Thomson, C. E. of Belfast, and probably no turbines are more efficient or capable of more general application to every variety of fall than the vortex wheels which he has constructed. For this reason, and also because from their recent introduction they are less known than the varieties which have been longer in use, we shall illustrate them rather more fully with the aid of working drawings, supplied by Messrs. Williamson and Brothers of Kendal, who we believe have at present erected all which are employed in this country.

The peculiarity of these vortex wheels consists in the arrangement of the fixed guide blades on the outside of a circular chamber in which is placed the revolving wheel, so that the water flowing inwards strikes the curved plates of the revolving wheel tangentially, and leaves the wheel at the centre at a minimum velocity; the whirlpool created in the wheel chamber giving to this description of turbine its designation of vortex wheel.

Fig. 134 shows the general form of the guides and passages of a vortex wheel; *a, a* are the fixed guides, four in number, which direct the water tangentially into the passages of the wheel *b b;* after having done its work in these, the water leaves the wheel at the open passage at the centre *c; s* is the vertical shaft carrying the wheel and communicating its motion to the mill. The chamber in which the guide blades *a, a* are fixed, forms part of the supply chamber, and the supply of water to the wheel may be regulated by altering the position of the guide blades, and thus diminishing or increasing the area of opening between them. For this purpose the guide blades are fixed on

Fig. 134.

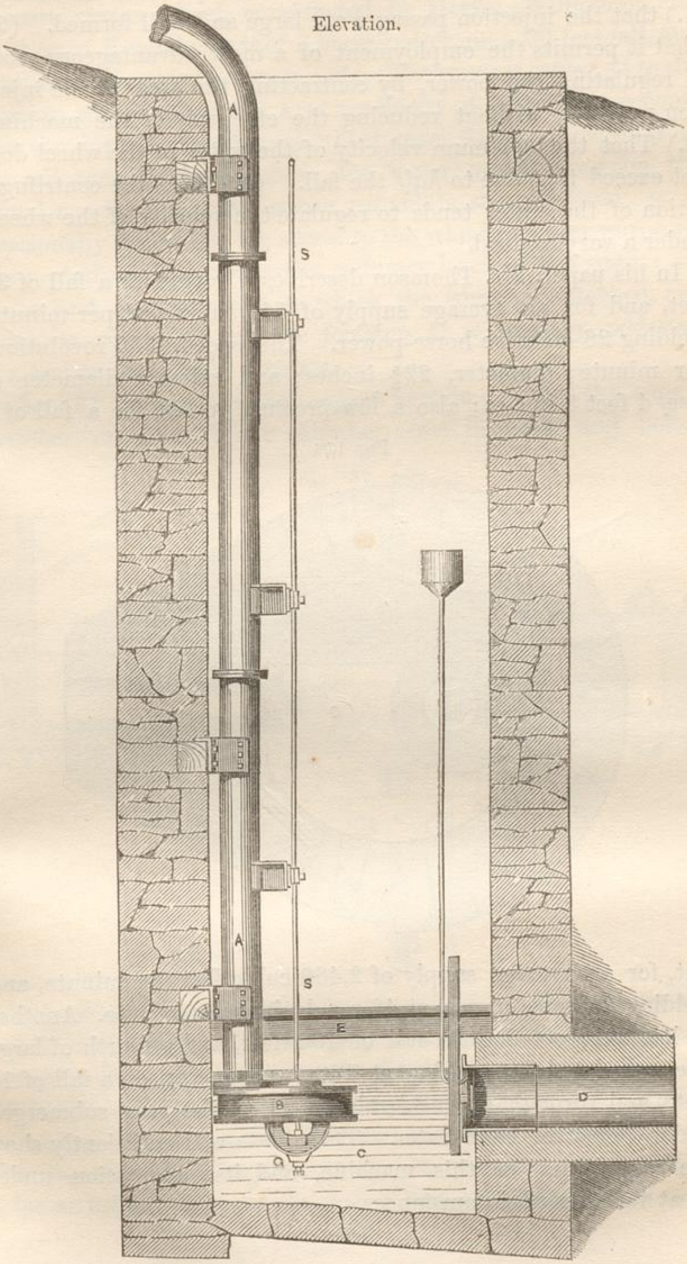


gudgeons *d*, *d*, near their extremities, and are connected by levers and links so that they may be shifted simultaneously by a spindle. The inner radius of the wheel is usually half the external radius, and the obliquity of the inner ends of the vanes 30° to 45°.

The general principles of these turbines Mr. Thomson thus explained at the meeting of the British Association in 1852 :— "The velocity of the circumference is made the same as that of the entering water, and thus there is no impact between the water and the wheel ; but, on the contrary, the water enters the radiating conduits of the wheel gently, that is to say, with scarcely any motion in relation to their mouths. In order to attain the equalisation of these velocities, it is necessary that the circumference of the wheel should move with the velocity which a heavy body would attain in falling through a vertical space equal to half the vertical fall of water, or, in other words, with a velocity due to half the fall, and that the orifices through which the water is injected into the wheel chamber should be conjointly of such area, that when all the water required is flowing through them it may also have a velocity due to half the fall. Thus one half only of the fall is employed in producing velocity in the water, and therefore the other half still remains acting on the water in the wheel chamber at the circumference of the wheel in the condition of fluid pressure. Now, with the velocity already assigned to the wheel, it is found that this fluid pressure is exactly that which is requisite to overcome the centrifugal force of the water in the wheel, and to bring the water to a state of rest at its exit, the mechanical work due to both halves of the fall being transferred to the wheel during the combined action of the moving water and the moving wheel. In the foregoing statements, the effects of fluid friction, and of some other modifying influences, are, for simplicity, left out of consideration ; but in the practical application of the principles, the skill and judgment of the designer must be exercised in taking all such elements as far as possible into account. To aid in this, some practical rules, to which the author (Mr. Thomson) as yet closely adheres, were made out by him previously to the date of his patent. These are to be found in the specification of the patent, published in the *Mechanics' Magazine* for January 18 and January 25, 1851."

Fig. 135.

Elevation.

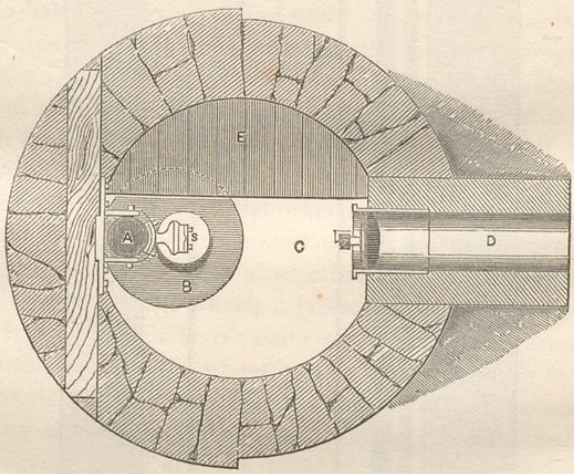


$\frac{1}{4}$ in. = 1 ft.

Mr. Thomson claims for his wheel the peculiar advantages (1.) that the injection passages are large and well formed. (2.) That it permits the employment of a most advantageous mode of regulating the power, by contracting the areas of the injection passages, without reducing the efficiency of the machine. (3.) That the maximum velocity of the water in the wheel does not exceed that due to *half* the fall. (4.) That the centrifugal action of the water tends to regulate the velocity of the wheels under a varying load.

In his paper, Mr. Thomson describes a vortex for a fall of 37 feet, and for an average supply of 540 cubic feet per minute, yielding 28 effective horse-power. The speed, 355 revolutions per minute; diameter, $22\frac{5}{8}$ inches; and extreme diameter of case, 4 feet 8 inches; also a low-pressure vortex for a fall of 7

Fig. 136.

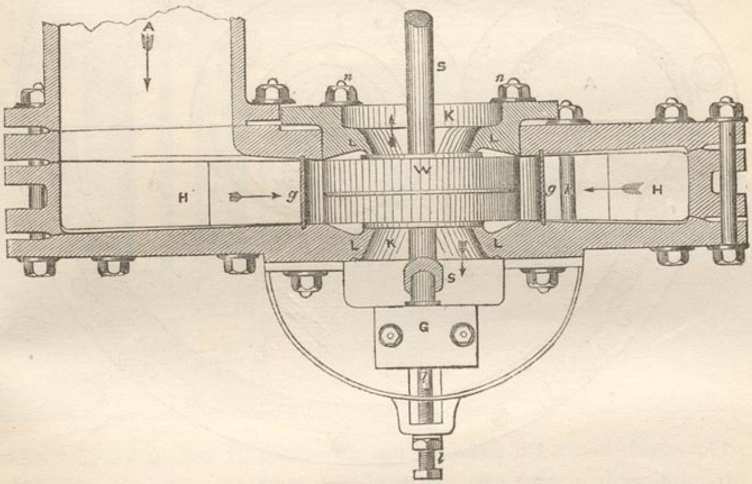


feet, for an average supply of 2,460 cubic feet per minute, and yielding 24 horse-power, at 48 revolutions per minute. Another he has constructed for a fall of 100 feet, and a fourth of large size, calculated for working at 150 horse-power, on a fall of 14 feet, and through a considerable part of the year submerged under 7 feet of back-water. These data will sufficiently show the capabilities of this machine, and its adaptation under great varieties of circumstances.

Figs. 135 and 136 exhibit an elevation and plan of a high-pressure vortex wheel, constructed by Messrs. Williamson and Brothers of Kendal. It is of 5 horse-power, on a 30 feet fall, and consumes 118 cubic feet per minute. The water is conveyed to the wheel in the 9-inch pipe A A, at a velocity of 4·4 feet per second. B is the supply chamber, or wheel case, fixed on masonry in the tail race C, from which the water passes away by the tunnel D. In the drawing the tunnel is shown closed, as is occasionally necessary, for access to the wheel or other purposes. E is a platform just above the ordinary level of the water; S S is the first motion shaft, to which the wheel is attached, and which is supported on the footstep at G, and by pedestals attached to the supply pipe A A.

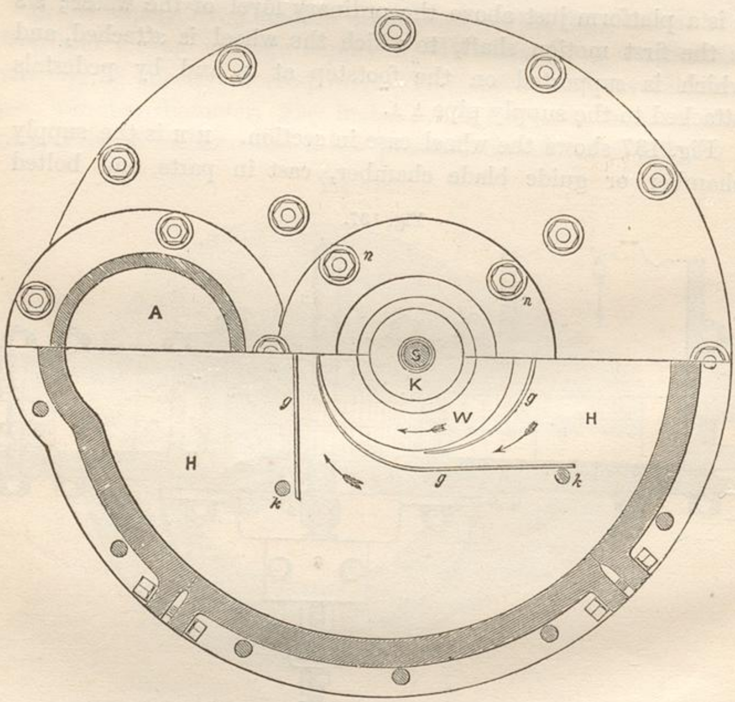
Fig. 137 shows the wheel case in section. H H is the supply chamber or guide blade chamber, cast in parts and bolted

Fig. 137.



together as shown. W, the wheel itself, about 10 inches in diameter, and composed of wrought iron plates with wrought iron curved vanes; *g*, *g* the four guide blades, in this wheel fixed and let into grooves cast in the cover and bottom of the chamber; K, K, four bolts tying the cover and bottom of the supply chamber together to strengthen it against pressure. A the supply pipe as before, and K, K the openings in the centre of the wheel for the escape of the waste water after it has done its work on the wheel. The joint between the wheel and its case is made by means of the accurately fitting annular parts L, L, adjusted for the wheel to run without friction by bolts n, n in the upper piece. S S is the first motion shaft resting on a lignum vitæ pivot firmly fixed in the foot bridge G, which is bolted on below the supply chamber, the height of the pivot as it wears being adjusted by the screw l l. The pivot is lubricated by the water

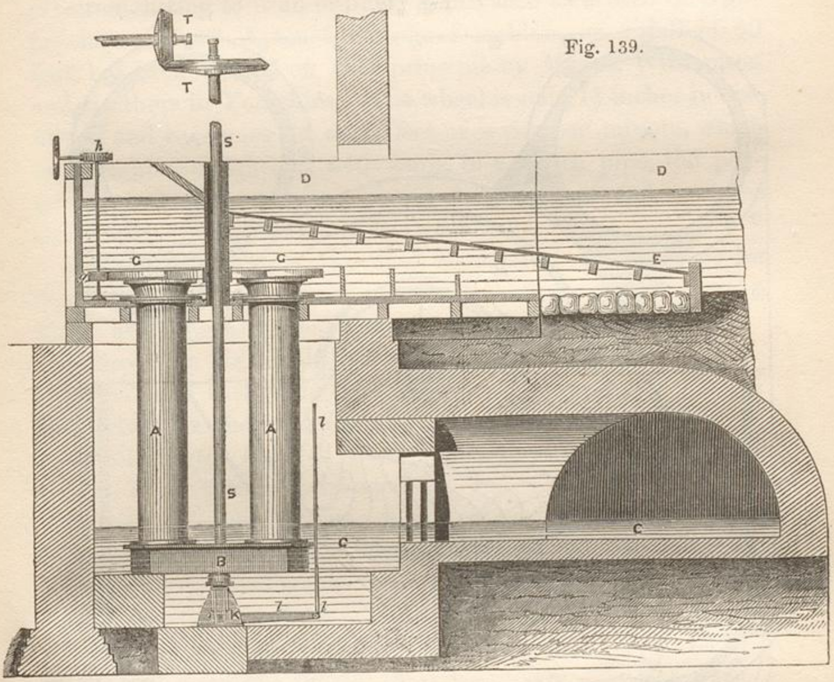
Fig. 138.



in which it works spread over it by a radial groove. In other cases Mr. Thomson makes the shaft to terminate in an inverted cup containing a concave brass disc working on a fixed steel pivot, with a radial groove for spreading the water. He does not consider the lubrication with oil so essential as other engineers insist, and believes the cases in which turbine pivots have been rapidly destroyed to be attributed to the absence of a proper provision for the escape of the air between the rubbing surfaces. Fig. 138 represents a half-plan and half-horizontal section of the same wheel, the same letters of reference being used as in fig. 137.

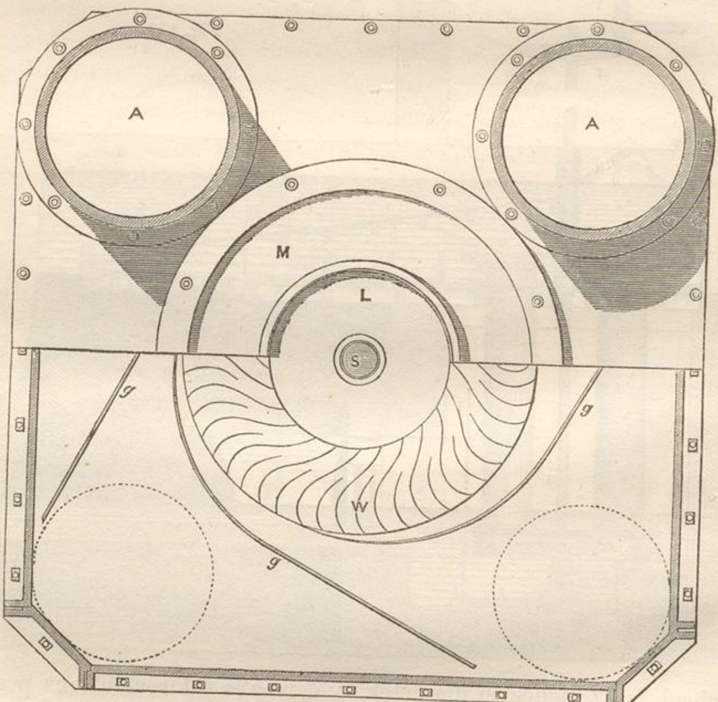
Fig. 139 exhibits the arrangement, in sectional elevation, of a low-pressure vortex wheel with its pentrough and tail race. This wheel is of 34 horse-power, with an effective fall of 14 feet 3 inches, and a supply of 1680 cubic feet per minute. The

Fig. 139.



wheel is 52 inches in diameter, and makes 94 revolutions per minute. A, A are the four supply pipes, two feet in diameter, so that the water in them has a velocity of about 2·3 feet per second. B is the square supply chamber, C C the tail race, and D D the conduit and pentrough. The water as it arrives passes through a perforated metal strainer E E, to prevent the choking of the narrow passages of the wheel by floating leaves, &c. Over the four supply pipes A, A, is fixed a circular cast iron plate G G, with four holes corresponding in one position with the trumpet mouths of the supply pipes. On the edge of this plate is a rack into which the pinion *g* gears, so that by moving the worm and wheel *h* the sluice plate G G may be revolved and the entrance for the admission of the water to the wheel more or less closed or opened. This is an effective and inexpensive means of regulating the power of the wheel, where the supply of water is abundant and it is not necessary to economise its

Fig. 140.



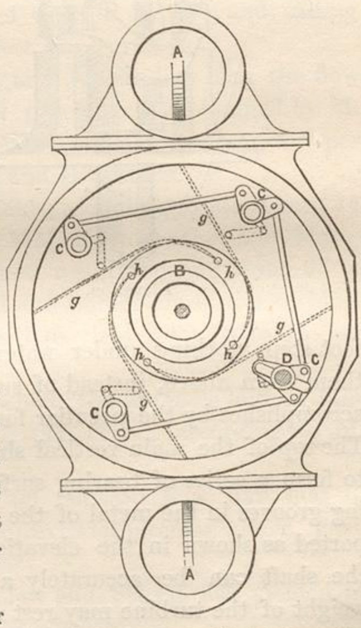
Scale ½ in. = 1 ft.

expenditure to the utmost extent. S S is the first motion shaft, and T, T the bevel wheels by which it gives off the power of the wheel to the mill. K is the footbridge carrying a step which can be raised by the lever *l l* as it wears away. Fig. 140 exhibits a half-plan and half-section of the wheel and supply chamber. A, A, as before, supply pipes, W, the wheel itself, *g*, *g*, *g*, *fixed* guide blades, the regulation of the wheel being effected by the sluice as before described. S, first motion shaft, L central opening for the escape of the water, M wheel cover, forming at its inner periphery a close and accurate joint with the revolving wheel.

Another plan which has been adopted with these wheels for regulating the speed, when they are applied to high falls, is to bring the supply pipe when near the wheel into a horizontal direction, fitting to it an ordinary sluice such as is used in high-pressure mains. A ten horse-power turbine, on a fall of 80 feet, has been erected on this principle by Messrs. Williamson and Brothers in Yorkshire. The wheel is only 13 inches in diameter, and consumes 44 cubic feet of water per minute, which is brought a distance of 340 yards, in a 9-inch pipe, the pen-trough and strainer being placed at the upper end, and the sluice at the bottom, close to the wheel.

But beyond question the most economical arrangement for regulating the expenditure of water, although somewhat more complicated in its details, is the adjustment of the guide blades themselves in the manner already alluded to. Fig. 141 shows a plan of a turbine, arranged with *movable* guide blades. A, A, two supply pipes, B, wheel cover; C, C, C, C, bell cranks connected together by links. The whole of these bell cranks are worked by a vertical spindle, D, and worm and wheel in the mill; they carry in the supply chamber links shown by the dotted lines, by which the guide blades *g*, *g*, *g*, *g*, movable on centres at *h*, *h*, *h*, *h*, can be opened or closed.

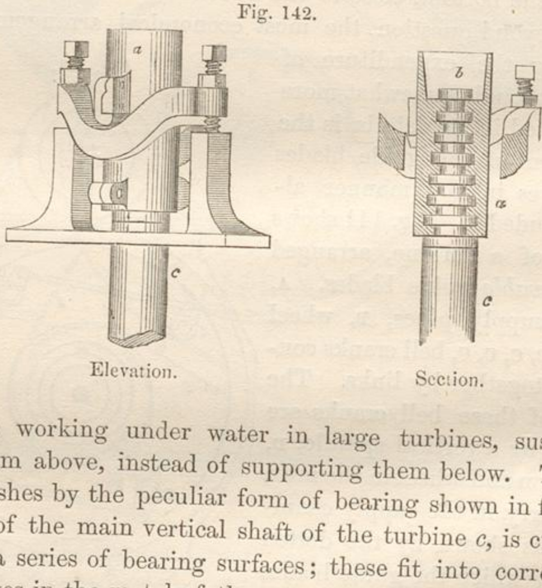
Fig. 141.



These turbines yield 75 per cent. of the power expended, and are therefore as efficient as the best water wheels or turbines. They work equally well under backwater, and if it be necessary they can be placed at any height less than 30 feet above the water in the tail race, the lower part of the fall being made to do its work on the wheel by suction in pipes descending from the central discharge orifice, and terminating in the water of the tail race.

In America, turbines of various kinds have come into extensive use, and some erected there are of unprecedentedly large size. The better forms have been copied in their main features, from European machines already described, with some variation in the constructive details. Thus Mr. Boyden has introduced a diffuser, or annular mouth-piece, round the outer or revolving wheel of the Fourneyron turbine, instead of permitting the water to escape into the free space of the tail water. He has also, to avoid the difficulties arising from the rapid wear of the

Fig. 142.



Elevation. Section.

foot-step working under water in large turbines, suspended them from above, instead of supporting them below. This he accomplishes by the peculiar form of bearing shown in fig. 142. The top of the main vertical shaft of the turbine *c*, is cut so as to form a series of bearing surfaces; these fit into corresponding grooves in the metal of the suspension box *a*, which is supported as shown in the elevation by gimbals. The height of the shaft can be accurately adjusted by screws, so that the weight of the turbine may rest on the collars in the suspension shaft and the lower bearing beneath the water serve merely to retain the shaft in its place. By lining the suspension box *a* with a soft metal, principally tin, melted and poured in with the necks in place, sufficient accuracy can be attained to prevent any undue strain on particular collars. This form of bearing is said to have been successfully employed, and to obviate the difficulties of oiling beneath the water.

Efficiency of Turbines.— It may be useful to revert here for a moment to the experiments which have been made upon different forms of turbine to ascertain their relative efficiency. In all these machines, the useful work rendered, is less than the entire force of the fall of water which acts upon them by the loss of work expended in overcoming the friction and inertia of the machine, together with the loss from the *vis viva* expended in shocks and impact and passing away in the water of the tail race, and from other causes in special cases. The fraction which expresses the ratio of the total work expended by the water to the useful work returned by the machine is the *efficiency* of the machine. Commonly we express this ratio in a percentage, taking the work of the fall as 100, and calling the work accomplished useful effect or return.

For the turbines of Fontaine and Jonval, in which the flow is vertical, a return of 70 to 72 per cent. was obtained by M. Morin; 67 per cent. by MM. Alcau and Grouvelle; 74·5 per cent. by MM. Hulze, Borneman, and Bruckman.

The turbine of Fourneyron yields, according to M. Morin, 74 per cent.; but 64½ according to MM. Redtenbacher and Marozeau; M. Fourneyron has obtained results varying from 65 to 80 per cent. according to the fall and immersion of the turbine. The turbine of St. Blazier is said to yield from 70 to 75 per cent.

The turbine of Poncelet, in which the water is laid on tangentially, yields from 65 to 75 per cent.; according to M. Hulze, 70 per cent.

The turbine of Cadiat, with an outward flow like that of Fourneyron, but regulated by an exterior circular sluice, gave 65 per cent. to M. Redtenbacher.

The reaction wheel of Whitelaw and Stirrat has yielded in experiments with models 70 to 78 per cent.

Mr. Thomson's vortex wheel yields according to his experiments 75 per cent.

All these returns appear to approximate closely to the duty performed by water wheels; probably not so high as that given by a well-constructed iron water wheel, but the difference is inconsiderable. Smeaton's experiments gave, on his overshot wheels, as much as 76 per cent., and the results obtained from experiments on the breast wheel, with ventilated buckets on a large scale, gave nearly 78 per cent. of the actual power of the water employed.

Certain advantages, it must be admitted, are obtained by the turbine in certain localities under certain conditions; but it is very doubtful whether they are equal, either on the score of expense or ultimate efficiency, to well-constructed water wheels. In some situations favourable for their reception they are doubtless preferable in effecting a reduction of the original cost, but taking to account the conveyance of the water in pipes and other charges, it will be found as a general rule that the difference is not considerable, and that a well-constructed water wheel of 50 years' duration is an effective and excellent substitute for the turbine.

4. *Water Pressure Engines.*

In the water pressure engine the power obtained from the *pressure* of a column of water is employed in generating a reciprocating instead of a rotatory motion. Engines of this description have long been employed in the mining districts of the Continent, but in England their use appears to date from 1765, when a single-acting water pressure engine was erected for draining a mine in Northumberland by Mr. Westgarth.

For the most successful application of these engines, as regards efficiency, it is necessary that the motion of the water should be slow, and as far as possible without shock. Three to six strokes per minute, or a velocity for the piston of one foot per second, is about the ordinary speed. The stroke also should be long, and therefore "the most advantageous use to which a water pressure engine can be put is the pumping of water, to which slow motion and a long stroke are well adapted, because they are favourable to efficiency, not only in the engine but in the pump which it works."—*Rankine.*

The valves now usually employed in these engines are solid pistons working in the supply pipe, with leather or metal packings. Figs. 143 and 144 showing the valves for a single-acting engine will sufficiently indicate the principle. A A is the supply pipe, B B the entrance to the cylinder, and C C the eduction pipe. When the cylinder is being filled, fig. 144, the valve D is below the entrance and closes the eduction pipe. When, however, the cylinder is emptying, fig. 143, the valve is raised and then closes the supply pipe. Deep notches are cut in these valves in order that they may very gradually open and close the passages to prevent shock.

Fig. 143. Fig. 144.

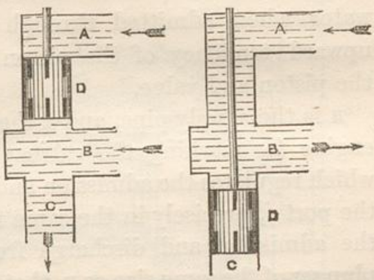
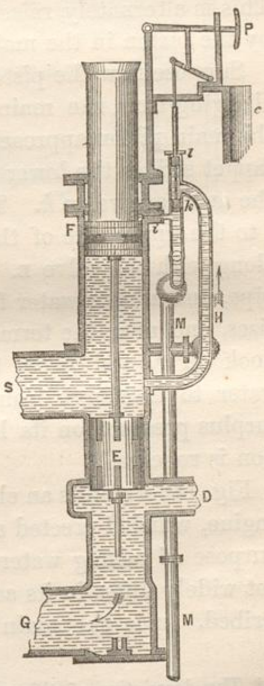


Fig. 145.



These valves are usually worked by a small water pressure engine, acting in the reverse direction to the general engine, and worked from it by tappets. Fig. 145 shows such an arrangement, from the single-acting engine of M. Junker.

In this drawing *c* represents the upper edge of the main cylinder, S the supply pipe, D the port connecting the main cylinder with the valve chest, G the discharge pipe: E is the valve, which when above D, as in fig. 145, permits the water to escape from the cylinder, and when below D, closes the discharge pipe and opens a passage from the supply pipe. The area of the valve E, is made less than that of the piston F, with which it is connected by a rigid rod. Hence the pressure of the water between E and F tends to raise them both.

The upper side of F is provided with a trunk working in a stuffing box in the top of the valve cylinder. The use of this is to diminish the effective area of the upper side of the piston F, so that it shall not be more than is requisite to enable the water when admitted through the port *i* to overcome the upward tendency of the piston together with the friction of the piston and valve.

H is the supply pipe, and M the discharge pipe of the auxiliary engine for working the valves; K is the valve of this engine which regulates the admission and discharge of the water through the port *i*, precisely in the same manner as the valve E regulates the admission and discharge from the main cylinder. *l* is a plunger of the same size as K, that the pressure between them may be equalised and not tend to move K upwards or downwards. The rod to which *k* and *l* are fixed is connected by means of a train of levers and link work with a lever carrying the crutch P. This is alternately raised and depressed by a tappet rod carried by the piston in the main cylinder *c*.

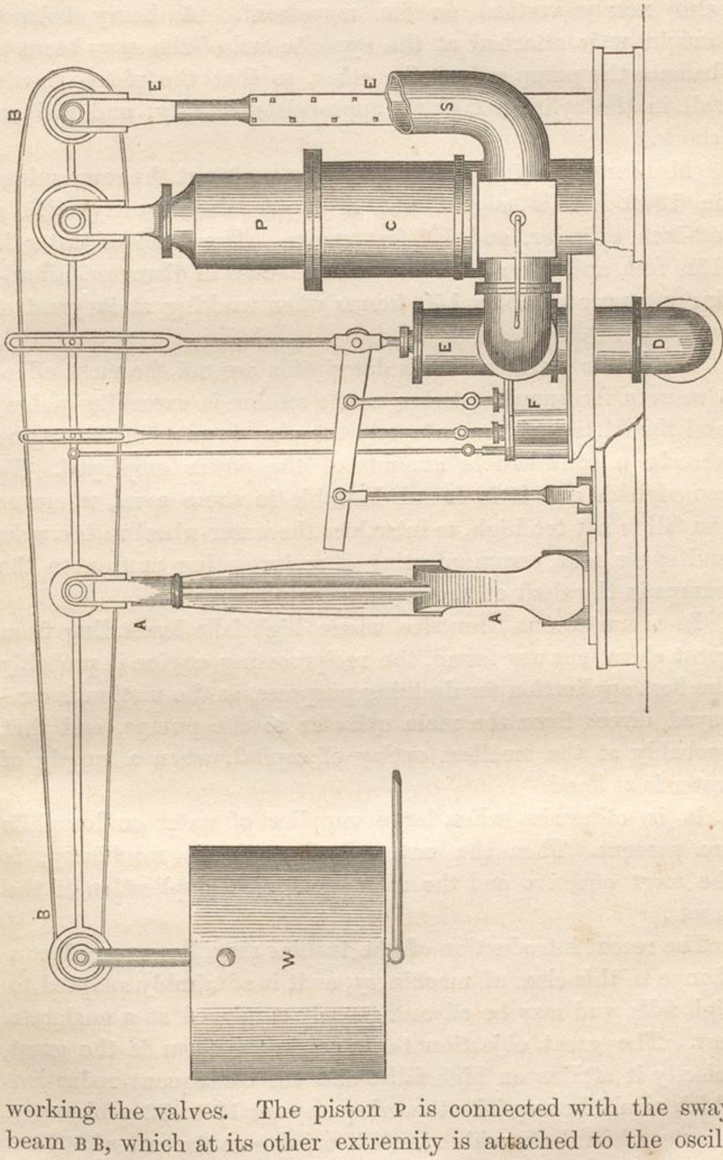
Suppose now the piston valve E is raised, and the water discharging from the main cylinder, as shown in fig. 145. When the main piston approaches the bottom of its stroke, the upper tappet strikes the lower hook on P and depresses it, along with the auxiliary valve *k*. This admits water from S through H and *i* to the upper side of the counter piston F, so as to depress it along with the valve E. The valve E then closes the discharge pipe, and admits water from S to the main cylinder; the piston rises, and near the termination of its stroke strikes the upper hook on P, and raises the auxiliary valve *k*. This allows the water to discharge from the upper side of F, and then the surplus pressure on its lower side lifts it with E, and the operation is repeated.*

Fig. 146 exhibits an elevation of a single-acting water pressure engine, which I erected some years since in Derbyshire for the purpose of raising water from the Alport lead mines. It does not widely differ in its action from that of M. Junker just described. C is the main cylinder, and P its piston or plunger.

* The description of this valve is abridged from Mr. Rankine's and Prof. Weisbach's Treatises.

S the supply pipe, and D the discharge pipe, connected with the valve apparatus E. F is the cataract or auxiliary engine for

Fig. 146.



working the valves. The piston P is connected with the sway beam B B, which at its other extremity is attached to the oscillating connecting rod A A, which is fixed on a pivot or joint at its lower extremity. By this arrangement the piston is permitted to rise vertically, and the spear rod of the pumps E is also nearly vertical in its movement. A heavy balance weight W is attached at the opposite end of the sway beam to balance the pump rods at the other, so that the piston should fall in the cylinder C at an appropriate velocity, and without shock.

Mr. Joseph Glynn erected a similar engine at the same mines in 1842. This engine was of larger size, namely with a 50-inch cylinder, and 10 feet stroke. The head of water is 132 feet, and lifts a plunger rod 42 inches in diameter, affording a power of about 150 horses when working at its greatest velocity.

Hydraulic engines of this description are not the most effective even for pumping water, as the motion is exceedingly slow, and the friction of the water and the organic parts of the engine absorbs a considerable amount of the power employed. To remedy this evil it is found desirable in some cases, wherever the fall is not too high, to introduce the water-wheel with cranks and spear rods, communicating a reciprocating motion to the pumps in the shaft of the mine.

In mountainous countries, where high falls descending from great elevations are found, the reciprocating engine is probably the best application for draining purposes, as the motion is conveyed direct from the main cylinder to the pumps, and that probably at the smallest outlay of capital, when a supply of water is at hand.

It is otherwise when large supplies of water on low falls are present. Then the water-wheel, with its machinery, is the most effective and the most economical application of the power.

The recent introduction of the turbine may, however, effect a change in this class of machinery, as it is admirably adapted to high falls, and may be advantageously employed at a moderate cost. The great objection to its use in this form is the great velocity it attains on high falls, and the consequent reduction which would be requisite to work pumps at 10 to 12 strokes per minute, when the machine itself is moving at the rate of 400 to

500 revolutions per minute. This appears to be the only drawback, and it is not improbable that the simple cylinder here described may, under certain conditions, be best adapted to meet all the requirements of raising water from deep mines with the aid of convenient streams on high falls.

CHAP. VI.

ON THE PROPERTIES OF STEAM.

BEFORE considering the application of the steam-engine as a prime-mover, it may be interesting to know something of the properties of steam by which it is moved, in regard to pressure, temperature, and density, as ascertained by various philosophers since the days of Newcomen and Watt. Of late years a great change has gradually taken place in the system of working the steam-engine. At the time of the introduction of the double acting engine of Watt, the makers of engines never dreamed of employing steam at a greater pressure than 10 lbs. on the square inch, and up to 1840 that was the maximum pressure at which steam-engines were worked, with the exception of a few constructed on Wolf's principle of double cylinders, where the steam is first admitted to the piston of the smaller cylinder at a pressure of 30 to 40 lbs. per square inch, and after having performed its office there, is allowed to expand into the second cylinder of three or four times greater capacity, and thus to unite its force with that of the small cylinder, as it moved from one extremity of the stroke to the other. To work this description of engine with high-pressure steam, it was necessary to proportion the strength of the parts of the engine as well as the boiler to a much greater extent of pressure than in the double-acting engine of Watt. Hence it was soon found that the waggon form for the latter, as employed by Watt, was not calculated to resist a pressure exceeding 10 or 12 lbs. per square inch without the introduction of numerous wrought-iron stays to retain it in form. To raise steam for the compound engine such a boiler was wholly inadequate, and a series of small boilers, with hemispherical ends, were introduced in its stead wherever steam of high-pressure was required.

The single pumping engines of Watt, and the compound engines of Wolf, employed at the mines in Cornwall, gave, however, extraordinary results as regards the work accomplished for the quantity of coal consumed, which was less than half the quantity used in the rotative engines employed in mills. It was also asserted that the double cylinder engine in use on the Continent (but chiefly made in this country) was performing a more satisfactory duty than could possibly be attained by the single cylinder low-pressure engine.

These assertions, often repeated, and the returns of Cornish engines, published from year to year, led to a close inquiry into the subject, first in my own works at Manchester, and subsequently before the British Association for the Advancement of Science, where the whole question was ably discussed, and ultimately led to a better system of working in factory engines, with a saving of one-half the fuel formerly consumed in effecting the same quantity of work. In these investigations it was found that the compound engine had no advantage over the single cylinder engine, as constructed by Watt, when worked at the same pressure of steam and the same rate of expansion; that is, a single cylinder engine, with properly constructed valves, having the power of cutting off the steam at any point of the stroke, is quite as effective, and more simple in construction, than the double cylinder engine. It is true, that at first the double cylinder engine had an advantage over the single cylinder engine in its greater uniformity of motion, but this is no longer the case, as an increase of the velocity of the piston from 240 to 320 and 360 feet per minute effectually remedies that evil, and increases the power of the engine in the ratio of the increase of speed.

Thus it will be seen that a great change has come over the system of employing steam; the pressure is quadrupled in factory engines, and more than doubled in marine engines. Every engine of recent construction is provided with boilers of great resisting powers, and on an average cuts off the steam in the cylinder at one-fourth, and at other times one-fifth or one-sixth of the stroke, the steam acting by expansion alone during the remaining three-fourths, four-fifths, or five-sixths, as the case may be. This system is found to be of great value, as the quantity of fuel consumed does about double the amount of work which could be got out of it on the low-pressure principle.

The important results already obtained by a judicious system of working steam expansively has given a powerful stimulus to the extension of our commerce and manufactures, and the question naturally arises, whether or no we have attained the full benefit from the introduction of the methods of working now employed, or whether we may not reap a still greater advantage from progressing in the same direction and using steam of higher pressure, expanded to still greater lengths than has yet been attained in our present practice. This is a question which remains for solution, and it appears most desirable that we should ascertain by direct experiments to what extent of pressure and expansive action we may safely venture with perfect security to the boilers and the working parts of the engines. Assuming for a moment that an increased pressure, accompanied by increased expansion, would in the same proportion increase the economy of working, we have then to consider the capabilities of our vessels for resisting those pressures. And lastly, the observation of the action of steam in expanding has led many to expect still further advantage from the use of superheated or gaseous steam. To make sure progress in either of the directions here indicated two things are necessary: we must cultivate a more intimate acquaintance with the resisting powers of materials, and the strength of vessels of different forms before we can assure ourselves of success; and we must attain increased and increasing knowledge of the properties of the agent we employ under the various conditions of expansion and superheating. In regard to the first of these requisites a steady progress has been made, and experimental inquiries have been extensively carried on in regard to the resisting powers of vessels and the causes of their failure, and the difficulty of constructing boilers to resist very high pressures has been greatly diminished. Our knowledge of steam has also rapidly increased, and many of the necessary questions relating to its properties have been for ever set at rest by the recent and classical labours of Regnault, carried on at the instance and with the assistance of the French Government. The questions of the density and law of expansion of steam, however, still require solution.

They are being investigated, from a theoretical point of view, with considerable success, by Mr. Rankine of Glasgow. The experimental inquiry I have undertaken in conjunction with my friend Mr. Tate, and a part of the results, comprising experiments up to a pressure of 60 lbs. per square inch, will appear in the Transactions of the Royal Society. We are now preparing to enter on the more arduous and dangerous task of ascertaining the density, volume, &c. at much higher pressures. The accumulation of facts on this subject, bearing directly upon the application of steam, cannot be otherwise than acceptable to the general reader, and I shall, therefore, without further preface, insert such an abstract as bears directly on the subject under consideration.

General Laws of Vaporisation.

When a liquid is heated in any vessel, its temperature progressively rises up to a certain point, at which it becomes perfectly stationary. At that point the heat continuously absorbed becomes *latent*, or is no longer registered by the thermometer; ebullition commences, and vapour, of a bulk enormously greater than that of the liquid from which it is formed, rises in bubbles and fills the vessel. In this condition the temperature of the liquid is perfectly constant; no urging of the fire will cause it to rise; the heat, absorbed continuously, expands itself in effecting that change in the state of aggregation of the liquid which we know as vaporisation.

This remarkable constancy in the temperature of liquids undergoing vaporisation in open vessels has long been known and applied to the graduation of thermometers. The point at which a liquid boils in an open vessel is called its boiling point. The following table gives the boiling points of some of the more important liquids:—

	Boiling Point. Fahr.	Authority.
Water	212°·0	
Ether	94·8	Kopp.
Alcohol	173·1	Pierre.
Sulphuric Acid	640·0	Marignac.
Mercury	662·0	Regnault.

We have said that the boiling point of a liquid is constant when in an *open vessel*, that is, when subject to the atmospheric pressure. If we change the pressure the temperature of ebullition changes also. Thus, if we place a vessel of hot but not boiling water under the receiver of an air-pump, and rapidly exhaust the air, the liquid will after some time begin to boil, and we may notice that the lower its temperature the more perfect must we make the vacuum before ebullition commences. Or again, if water be subject to pressure greater than that of the atmosphere, its temperature must be raised higher than 212° before it will boil. Experiment, therefore, shows that the boiling point, constant at the same pressure, varies at different pressures, rising higher as the pressure increases, and *vice versâ*.

Strictly speaking, the pressure of the atmosphere is not always the same; it varies within narrow limits from day to day; it decreases as we ascend higher into it, and hence there will be a small but corresponding variation in the boiling point at different times and places. This last fact has afforded the means of measuring the altitude of mountains, by determining the difference of the boiling point at their base and their summit. Measuring the atmospheric pressure by the column it supports in the barometer, we may draw up the following table of the relation of the boiling point to the height of the barometer column and the altitude of the observer, assuming that the barometer stands at 29·922 inches, and water boils at 212° Fahr. at the level of the sea.

TABLE I.—EXHIBITING THE INFLUENCE OF CHANGES OF ATMOSPHERIC PRESSURE ON THE BOILING POINT OF WATER, AND THE BOILING POINT AT DIFFERENT ALTITUDES.

Height of Barometer in inches.	Altitude in feet.	Boiling Point of Water. Fahr.	Height of Barometer in inches.	Altitude in feet.	Boiling Point of Water. Fahr.
29·922	0	212°·0	25·888	3,926	204°·9
29·396	462	211·1	25·468	4,460	204·0
28·774	933	210·0	25·014	5,000	203·0
28·559	1,411	209·3	24·046	6,111	201·2
27·846	1,897	208·5	23·454	7,263	200·0
27·348	2,392	207·6	18·992	13,700	190·0
26·852	2,895	206·7	15·135	18,000	180·0
26·372	3,407	205·8	12·145	26,000	170·0

But, besides pressure, certain other circumstances exercise a slight but sensible influence on the boiling point. In a glass vessel the boiling point of water is about two degrees higher than in a metal one, owing apparently to some adhesion between the glass and the liquid. Dr. Miller states that if the glass be varnished with shellac the temperature of the water may be raised to 221° in the open air, when a sudden burst of steam will take place, during which the temperature falls to 212°. From a similar cause the presence of salts in solution raises the boiling point in some cases considerably. A saturated solution of common salt boils at 227° Fahr., and a saturated solution of chloride of calcium, which has an enormous affinity for water, does not boil at a less temperature than 355° Fahr.

There is yet one other remarkable condition of evaporation which should be noticed here. If water be dropped upon a clean metallic surface heated sufficiently high, instead of entering into ebullition it assumes a globular form, and rolls about very slowly and quietly evaporates. This condition, known as the spheroidal state, has been investigated by Mr. Boutigny. He finds that the temperature of the liquid globule never rises so high as its boiling point, being indeed usually 5° to 10° below it; that the temperature of the plate necessary to cause the spheroidal state varies with different liquids, and depends in part on the conducting power of the plate; and he considers the temperature of the spheroid to be constant, being for water 205°·7; for alcohol 167°·9, and for ether 93°·6.

If, whilst the spheroid is rolling upon the metal plate, the temperature of the plate is allowed to fall below a certain temperature (340° for water), the spheroid breaks, and is suddenly dispersed in vapour.

The temperature of the vapour rising from a liquid is necessarily identical with that of the liquid from which it rises, except in those cases in which the boiling point has been affected by adhesion, when the vapour at once adjusts itself to the normal temperature at that pressure.

So long as vapour is in contact with the liquid from which it has been formed its temperature continues the same as that of the liquid, for if it be heated it takes up fresh liquid, and the temperature falls from the absorption of the heat rendered latent, until the normal temperature of the boiling point is regained.

The Vaporisation of Water and the Formation of Steam.

The temperature at which water boils is therefore constant at each pressure, and in consequence the temperature of the steam itself, when in contact with water, is constant at each pressure. The relation between the temperature and pressure of steam has been ascertained by experiment.

When in contact with the water producing it, steam is at the maximum density consistent with that temperature and pressure, and is then called *saturated* steam, or vaporous steam, and its temperature is called the maximum temperature of saturation at the given pressure. Usually when the *pressure of steam* is spoken of, the pressure of saturated steam is intended.

When isolated from the water producing it and heated, the steam expands, and decreases in density if the pressure be constant, or if the volume be constant it increases in pressure; it is then called variously anhydrous, gaseous, or superheated steam. The rate of expansion of superheated steam must be determined by experiment.

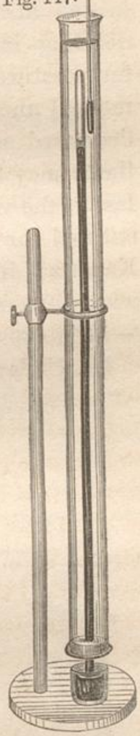
By the *density* of steam we mean the relative weight of a unit of volume. The *specific volume* of the steam is the reciprocal of the density, or the ratio of the volume of the steam to that of the volume of water which produced it. The density of saturated steam is constant at each temperature, and must be determined by experiment.

The *latent heat* of evaporation of steam is the quantity of heat which disappears in effecting the conversion of the water into vapour, or which reappears in the condensation of the steam. The latent heat of evaporation added to the sensible heat, or heat required to raise the temperature of the water up to the temperature of ebullition, is called the total heat of the steam.

The Relation between the Pressure and Temperature of Saturated Steam.

Probably the earliest experiments on this subject were made by Watt *, who tells us that when inventing the separate condensation he made some trials (in 1774) from which he constructed a curve, of which the ordinates represented the pressures, and the abscissæ the temperatures of the steam, and thus enabled him to calculate the one from the other at sight, with sufficient accuracy for his purposes. Watt first surmised that the elastic force or pressure of the steam increased in a geometric progression for temperatures increasing in an arithmetical progression.

Fig. 147.



Robison † made experiments upon the same subject at elevated temperatures, ascertaining the temperature at which the steam began to blow off from a safety valve loaded with weights, a proceeding susceptible of little accuracy. Dalton ‡, however, was more successful in devising an accurate method. He employed a barometer carefully purged of air, into which he introduced a small quantity of water. The barometer was surrounded by an outer water bath, by which the vapour in its chamber was heated to various temperatures. The mercury in the barometer tube adjusted itself so as to be in equilibrium at each temperature between the pressure of the atmosphere on the outside and the pressure of the vapour within, and the column fell as the temperature rose to an extent which is an exact measure of the pressure of the vapour within. The difference of height of an ordinary barometer and the barometer containing the water gives directly the pressure of the steam, so that by a series of careful measurements of a humid barometer and ordinary dry barometer, the pressures corresponding to various temperatures may be observed.

This method, under various modifications, has been frequently employed, both for water and other liquids, at pressures which

* Muirhead's Life of Watt, p. 76.

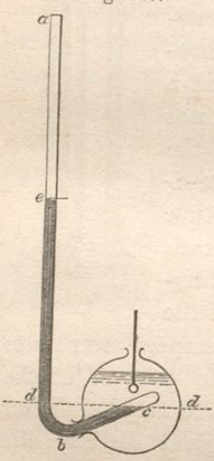
† Mechanical Philosophy, vol. ii. p. 23.

‡ Memoirs of the Manchester Literary and Philosophical Society, vol. xv. p. 409. New Series, vol. v. p. 553.

are less than that of the atmosphere. The chief difficulty is to maintain the liquid in the bath by which the barometer is heated at a uniform temperature, and to prevent it from dividing into strata unequally heated. To obviate this, the temperature may be observed at various depths and the arithmetical mean taken, or the length of the barometer may be decreased as the temperature rises. Or a barometer with two limbs may be employed, as in the researches of Dr. Ure, or, lastly, the varying temperature of the atmosphere may be substituted for that of the liquid bath, as in the experiments of Kaemtz *, intended to supply data for meteorological purposes, which extended over a period of two years, and ranged from $-15°$ to $+80°$ Fahr.

Dr. Ure's † modification enables the experiments to be carried to pressures higher than that of the atmosphere. The space in the barometer tube occupied by the vapour need never be large, and the increase of elastic force is measured by the quantity of mercury which must be added to a second limb of the barometer in order to maintain the quicksilver in the first at a constant level. Thus, in fig. 148, *a b c* is the bent barometer tube for experiments above the atmospheric pressure, the shorter limb being enclosed in a glass vessel, which can be filled with oil and heated progressively to any required temperature. Fine rings of platinum wire are firmly fixed round the tubes at the level *d d*, and as the temperature rises the mercury in the limb in the bath is maintained at this level by adding mercury in the other limb, when the column *d e*, supported by the steam, measures its elastic force.

Fig. 148.



Dalton, whose experiments were, on the whole, accurate, inferred from the results which he obtained with water and alcohol, that the tension of all vapours was equal at temperatures

* Traité de Météorologie, vol. i. p. 290.
† Phil. Trans., 1818, p. 338.

equally distant from their boiling-points under atmospheric pressure. This law, which has since borne his name, has not been confirmed by experiments on a larger number of liquids. For many liquids, however, it is nearly true, at small distances above the boiling point. Thus:—

Degrees from the boiling point. Fahr.	Elasticity in inches of Mercury. Water.	Alcohol.	Ether.
+ 30	52·90	56·60	50·90
+ 20	44·06	46·30	42·64
+ 10	36·47	37·00	35·20
0	30·00	30·00	30·00
− 10	24·50	24·20	24·70

In the above table the boiling-point of water is 212°, of alcohol 173°, and of ether 104°.

The following table gives a few of Dalton's results for comparison with those of other experimenters which will be given presently.

TABLE II.—ELASTIC FORCE OF THE VAPOUR OF WATER, ACCORDING TO DALTON.

Temperature. Fahr.	Elastic Force of Vapour. In inches Mercury.	In lbs. per square inch.
0	·066	·033
10	·090	·045
20	·129	·064
40	·263	·131
60	·524	·262
80	1·000	·500
100	1·86	·93
150	7·42	3·71
212	30·00	15·00

In 1823 the French Government, then legislating on the subject of steam, and requiring some further knowledge of its properties, intrusted to the French Academy the conduct of some important experiments on this subject. The Academy appointed a Commission, consisting of MM. Prony, Arago, Girard, and Dulong, to investigate the subject, and their report was published in the Memoirs of the Academy for 1831.* The experiments detailed in this Report were made chiefly by MM. Dulong and Arago, by a new method, and with all the care

* Mémoires de l'Institut, tom. x. p. 194, and Annales de Chimie et de Physique, tom. xliii. p. 74.

and accuracy which was possible in the state of science at the time. They were also on a scale which is only possible where private effort is seconded by the munificence of the Government.

Their apparatus consisted of, 1st, a boiler to generate the steam, 2nd, a manometer to measure the pressure.

The pressure, which extended to twenty-four atmospheres, was in fact measured by the column of mercury it would support in an open glass tube, but as the length of tube necessary for this purpose rendered it very inconvenient, they employed an intermediate measurer, consisting of a closed air manometer, graduated by experiment with the open mercury column. At the centre of the tower of the ancient church of St. Geneviève they erected a firmly supported wooden column, to which they attached the glass tubes containing the mercury column. These tubes, thirteen in number, were each 6½ feet in length, so that the mercury column for the graduation of the manometer could be as much as 86 feet in height, corresponding to a pressure of thirty atmospheres, or 450 lbs. per square inch. This column was adjusted precisely vertical, and communicated with a cistern containing 100 lbs. of mercury. The manometer, which consisted of a carefully dried glass tube, closed at the upper extremity, and 67 inches long, communicated with the same cistern, and was maintained at a uniform temperature by a stream of water circulating round it. The height of its mercury was read by means of a vernier, similar to that of a standard barometer. It is easy to see how, by means of a force pump, the pressure in the cistern of mercury could be increased at pleasure, and how the pressure could be registered by reading off simultaneously the height of the mercury in the open tube and its corresponding level in the manometer. When the value of the divisions of the manometer had been thus determined up to twenty-seven atmospheres, it became an instrument for measuring pressure of as great accuracy and delicacy as could be desired.

The boiler for generating the steam was of a capacity of 17·6 gallons, to ensure a uniform temperature, and communicated with the manometer by a tube filled with water, cooled by a refrigeratory apparatus. The temperature was measured by means of mercurial thermometers, placed in thin metal tubes, containing mercury, to protect them from pressure.

The boiler being charged, and a convenient quantity of fuel introduced into the furnace, the temperature was allowed to rise until it nearly attained a maximum. A series of readings were then taken simultaneously from the manometer and four thermometers, until the temperature passed its maximum, and began sensibly to decrease. The readings at the maximum were alone retained for calculation. Fresh fuel was then added, and a second experiment obtained.

The method, carried out with the skill for which MM. Arago and Dulong have earned so high a reputation, possesses most of the essentials of complete accuracy. Its chief defect, as M. Regnault has pointed out, lies in this, that when the pressure and temperature are changing, however slowly, it is impossible to be absolutely certain that the thermometers have followed that change with the necessary rapidity, and that they do really register the temperature at the time the observation is made. There is in these experiments one other source of possible error, namely, the use of the mercurial thermometer, which, in the higher parts of its scale, does not possess the accuracy necessary in experiments of this nature. Be this as it may, these experiments are of high value and permanent importance. The results obtained in thirty experiments are given in Table IV. on next page.

Next to the experiments of the French Academy, the most important experiments on the relation of temperature and pressure of steam were those of the Franklin Institute in America. They differed considerably from those of the French physicists, and are probably less reliable. The following table gives an abstract of the results:—

Table III.—Elastic Force of Steam from the Experiments of the Franklin Institute.

Pressure in Atmospheres.	Temperature in degrees Fahr.	Pressure in Atmospheres.	Temperature in degrees Fahr.	Pressure in Atmospheres.	Temperature in degrees Fahr.
1	212°	$4\frac{1}{2}$	298°·5	8	336°
$1\frac{1}{2}$	235	5	304·5	$8\frac{1}{2}$	$340\frac{1}{2}$
2	250	$5\frac{1}{2}$	310	9	345
$2\frac{1}{2}$	264	6	315·5	$9\frac{1}{2}$	349
3	275	$6\frac{1}{2}$	321	10	$352\frac{1}{2}$
$3\frac{1}{2}$	284	7	326		
4	291·5	$7\frac{1}{2}$	331		

TABLE IV. — RESULTS OF MM. ARAGO AND DULONG'S EXPERIMENTS ON THE RELATION OF PRESSURE AND TEMPERATURE OF SATURATED STEAM.

No. of Experiment.	Temperature observed. Cent.		Mean Temperature reduced to Fahr. scale.	Elastic force of Steam in Metres of Mercury, at 0° C.	Elastic force in inches of Mercury.	Elastic force in Atmospheres of 29·922 inches.
	Small Thermometer.	Large Thermometer.				
	°	°	°			
1	122·97	123·7	253·99	1·62916	64·141	2·14
2	132·58	132·82	270·86	2·1767	85·698	2·87
3	132·64	133·3	271·34	2·1816	85·891	2·88
4	137·70	138·3	280·40	2·5386	99·947	3·348
5	149·54	149·7	301·31	3·4759	136·85	4·584
6	151·87	151·9	305·38	3·6868	145·15	4·86
7	153·64	153·7	308·60	3·881	152·80	5·12
8	163·00	163·4	315·76	4·9383	194·42	6·51
9	168·40	168·5	335·21	5·6054	220·69	7·391
10	169·57	169·4	337·06	5·7737	227·31	7·613
11	171·88	172·34	341·79	6·151	242·17	8·114
12	180·71	180·7	357·26	7·5001	295·29	9·893
13	183·70	183·7	362·66	8·0352	316·35	10·60
14	186·80	187·1	366·51	8·6995	342·51	11·48
15	188·30	188·5	371·12	8·840	348·04	11·66
16	193·70	193·7	380·66	9·9989	393·66	13·19
17	198·55	198·5	389·33	11·019	432·83	14·53
18	202·00	201·75	395·36	11·862	467·02	15·65
19	203·40	204·17	398·80	12·2903	483·88	16·21
20	206·17	206·10	403·03	12·9872	511·32	17·13
21	206·40	206·8	403·88	13·061	514·22	17·23
22	207·09	207·4	405·03	13·1276	516·84	17·30
23	208·45	208·9	407·62	13·6843	538·76	18·05
24	209·10	209·13	408·39	13·769	542·10	18·16
25	210·47	210·5	410·86	14·0634	552·41	18·55
26	215·07	215·3	419·32	15·4995	610·23	20·44
27	217·23	217·5	423·25	16·1528	635·95	21·31
28	218·3	218·4	425·03	16·3816	644·96	21·60
29	220·4	220·8	429·08	17·1826	676·50	22·66
30	223·88	224·15	435·22	18·1894	716·13	23·994

The experiments of Arago and Dulong give a temperature of 358°·88 Fahr. for a pressure of ten atmospheres, or 6°·38 higher than that of the American Institute. This notable difference, too great to be merely accidental, Regnault, whose experience in the matter entitles him to speak with certainty, attributes to the use of mercurial thermometers, which, although agreeing perfectly between 32° and 212°, often present at elevated temperatures a difference of many degrees. Regnault's own experiments give 356·54 on the air-thermometer as the temperature at the pressure of ten atmospheres, which is 2°·34 lower than the French Academy, and 4° higher than the Franklin Institute. As at this temperature the mercurial-thermometer gives higher indications than the air-thermometer, MM. Arago and Dulong's experiments appear the most reliable.

The uncertainty arising from the discordance of the numerical results of the different physicists who had studied this question, and especially the difference above noted, called for a new investigation.

The experiments of M. Regnault on the reliability of the various instruments employed in measuring temperature, led him to the conclusion that at elevated temperatures the indications of different mercurial-thermometers were too variable to be trusted, unless they were made of the same description of glass, and that even in that case they require reduction to the absolute temperature of the air-thermometer. The following table gives some of the results obtained :—

Temperature by the Air Thermometer.	Temperatures by Mercurial Thermometers.			
	Crystal of Choisi-le-Roi.	Ordinary Glass.	Green Glass.	Swedish Glass.
100°	100°	100°	100°	100°
130	130·20	129·91	130·14	130·07
150	150·40	149·80	150·30	150·15
180	180·80	179·63	180·60	180·33
200	201·25	199·70	200·80	200·50
250	253·00	250·05	251·85	251·44
300	305·72	301·08		
350	360·50	354·00		

The above numbers are in centigrade degrees. They show for the thermometer of ordinary glass, when its indications are reduced to Fahrenheit's scale, the following divergences from the true temperature shown by the air thermometer : —

Temperature by Air Thermometer. Fahr.°	Temperature by Mercurial Thermometer. Fahr.°
302	301·64
392	391·46
482	482·09
572	573·94
662	669·20

To M. Regnault the French Government committed, on the proposition of M. Legrand, the task of carrying on a series of experiments to determine, with the greatest precision, the principal laws and numerical data which enter into the calculation of the duty of steam-engines, and supplied the funds for fulfilling its intentions on a scale such as would have been impossible in any private enterprise. The papers, which were the result of this munificence, are amongst the most important in the recent history of science. They cover a large ground, and possess a precision and completeness before scarcely ever attained in researches of this kind.

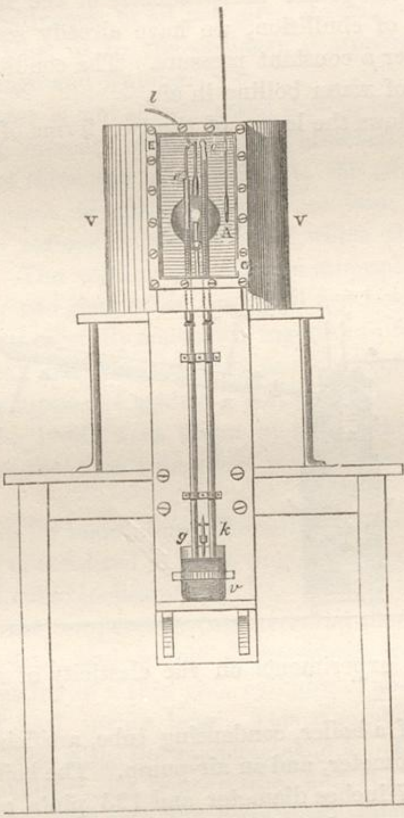
In relation to the steam-engine, the most important questions which M. Regnault has set himself to solve are—1st. The elastic force of the vapour of water, both at pressures lower than that of the atmosphere, and at high pressures, up to 400 lbs. per square inch. 2nd. The latent heat of the vapour of water through a similar range of temperature and pressure. 3rd. The specific heat of liquid water. The laws of the density and expansion of steam, it will be observed, Regnault did not touch, but, on the subjects above named, his researches are not likely to be superseded in accuracy or extent.

To ascertain the relation of temperature and volume of steam at low temperatures, Regnault adopted the plan of employing two barometers, placed side by side, under precisely similar circumstances, into one of which was introduced a portion of water perfectly freed from air. The upper part of these barometers was surrounded by a large bath of water, maintained by agitation at a constant temperature. The difference of level of the mercury in the humid and dry barometers gave directly the elasticity of the steam at the temperature of the bath.

Fig. 149, shows one of the forms of apparatus employed. The two barometers, *e g*, *o k*, were plunged in the same cistern *v*, and maintained vertical against a firm board. In the form shown, the moist barometer, *e g*, communicated with a glass globe A, of a capacity of 80 cubic inches, and exhausted of air by means of an air-pump, after which the tube *l* was hermetically sealed The tension of the air remaining was accurately ascertained, and did not exceed 1 to 2 millimetres. The bath of galvanised iron, V V, was of a capacity of about 2746 cubic inches; over a rectangular opening opposite the barometers the plate of glass, E G, was fixed, and through this the readings were taken, after the error arising from refraction had been determined. By means of a lamp placed underneath, a constant temperature could be maintained in this bath as long as was necessary to take a series of readings. When these were complete, water was withdrawn from the bath and replaced by boiling water, then, when a constant temperature was again arrived at, a new series of readings could be obtained.

These methods answered with perfect accuracy up to about 150° Fahr., above this the tendency of the water to separate into strata of unequal temperature began to manifest itself so as to

Fig. 149.

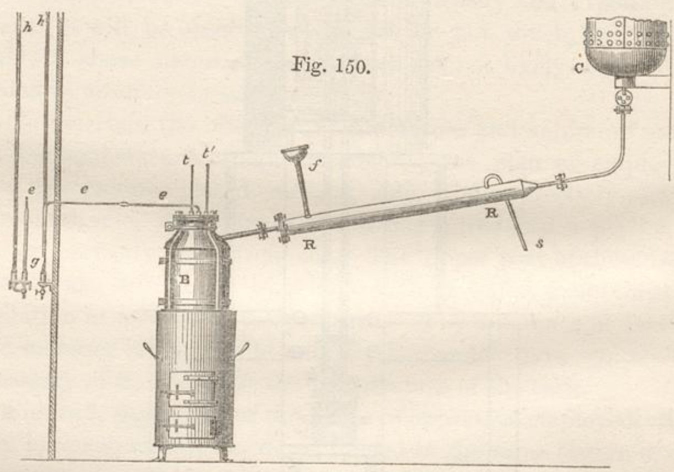


introduce errors into the experiments. Regnault, therefore, had recourse to the plan of observing the temperature at which water boils at determined pressures. This was the proceeding adopted by Arago and Dulong, but with a new precaution.

Those physicists were, by the method they adopted, compelled to regulate their experiments by the condition of their fire, so that it was impossible to maintain a constant temperature for any great length of time. By adding to their apparatus a large vessel containing air and acting as an artificial atmosphere, together with some large air-pumps, by which the pressure on the water could be predetermined and maintained perfectly constant, Regnault, in fact, obtained means for regulating his experiments altogether independently of the furnace, for the temperature of ebullition, we have already seen, is perfectly constant under a constant pressure. The conditions were identically those of water boiling in air.

Fig. 150 shows the larger of the two forms of apparatus em-

Fig. 150.



ployed in the experiments on the elasticity of steam at high temperatures.

It consists of a boiler, condensing tube, artificial atmosphere, mercurial manometer, and an air-pump. The boiler, B, is of red copper, of 13·7 inches diameter and 123 pints capacity. The cover carries two tubes, in which were placed mercurial thermometers protected from pressure, and a third for an air thermometer. The boiler was strengthened by iron rings bolted round it. The refrigerator, R R, of a copper tube 5 feet long, communicating with the boiler, surrounded by a larger tube, was arranged so that a continuous stream of cold water flowed into the funnel f, and away by the siphon, s. The reservoir of air forming the artificial atmosphere for maintaining a constant pressure was formed of a cylinder, C, of 62 gallons capacity, and riveted and brazed so as to be perfectly air-tight. The manometer for ascertaining the pressure consisted of an open mercurial column in every instance, than which no more perfect instrument, or one more free from corrections depending on theoretical calculations, could be devised. The indications are of equal sensitiveness at all pressures, instead of continually decreasing in value, as in the ordinary compressed air-gauge employed by Arago and Dulong.

This manometer is not shown in the sketch, but it consisted of a cistern of mercury with a pipe attached having four openings; one of these was usually closed; the other three were for the attachment of glass tubes to contain the mercury columns in the various experiments in which this apparatus was employed. The column open to the atmosphere consisted of ten to twenty-two glass tubes, nearly 10 feet long and $\frac{4}{10}$ inch bore. These were carefully connected together to form a column perfectly vertical, and from 40 to 80 or more feet in height, as required, being supported against a vertical wall. Up to fifteen atmospheres the levels were taken by two cathetometers; for higher pressures the glass tube itself was graduated into millimetres.

The air-pump for maintaining the artificial atmosphere in the large cylinder, C, consisted of three single-acting cylinders, each discharging 42 cubic inches per stroke.

To measure the temperature two mercurial thermometers, t, t, perfectly accordant, and an air thermometer, were employed; the latter consisting of a thin glass cylinder of about 1·17 inches diameter, and 11·7 inches long. This communicated by the capillary tube, e e, with the manometer, g h. The capacity of the air thermometer, its temperature, and the pressure were therefore known, and with the corrections which M. Regnault applied, he considers that it indicated sensibly $\frac{1}{25}$ of a degree.

It will be evident how, with this apparatus, a continuous and energetic ebullition was maintained in the boiler, B, under any pressure at which the observer wished to determine the temperature of the steam. Condensation went on at the same time with a corresponding rapidity in the cooling apparatus, R R; the pressure being maintained constant by the air-pump, the thermometers would in time become identical in temperature with the steam in the boiler B; simultaneous observations at this period of the thermometers and the manometer gave the relation of temperature and pressure which was required.

We have now described all the principal methods by which it has been sought to determine, experimentally, the relation of the pressure and temperature of saturated steam. It is necessary that we should next consider how they may be expressed in a formula suitable for calculation.

The law to which Watt was led, and which is usually known as Dalton's, from the care with which he verified it, so far as his experiments went, is that which, in general terms, most nearly expresses this relation; it is, that the elastic force of vapours increases in a geometrical progression, for a series of temperatures increasing in an arithmetical progression, and many of the formulæ which have been constructed to express the results of experiments, have been based upon it. Strictly speaking, it is, however, only an approximate expression of the true law.

One of the earliest and best of the formulæ which have been proposed, is that first applied to steam by M. Prony, of the form

$$F = a\alpha^t + b\,\beta^t + c\gamma^t + \ldots \quad (1)$$

where F is the elastic force, and t the temperature. The other quantities are constants derived from experiment. This formula is accurate, but requires a large amount of calculation.

Dr. Young proposed the formula

$$F = (a + bt)^m \ \ldots \ (2)$$

which has been the basis of several formulæ employed for interpolation by physicists. Thus MM. Arago and Dulong give from their own experiments the following constants:—

$$e = (1 + 0{\cdot}7153\ T)^5 \ \ldots \ (3)$$

where e expresses the elasticity in atmospheres of 29·922 inches of mercury; T the temperature in centigrade degrees reckoned from 100°, positive above that point and negative below, taking for unity an interval of 100°. For pressures greater than one atmosphere this formula is satisfactory, but it deviates greatly from experiment at lower pressures. At the same time it has a great simplicity.

The formula proposed by the Franklin Institute was of the same kind, the constants being

$$e=(\cdot 00333\,\mathrm{T}+1)^6 \;\ldots\; (4)$$

where e is the pressure in atmospheres of 30 inches of mercury, and T the excess of the temperature above 212° in Fahrenheit. This formula also does not apply below atmospheric pressure with accuracy. For calculation we may write it

$$\log e=6 \log \{\cdot 00333\,(t-212^\circ)+1\} \;\ldots\; (5)$$

and for calculating the temperature from the pressure,

$$=\frac{\sqrt[6]{e}-1}{\cdot 00333}+t\ 212 \;\ldots\; (6)$$

Another form of expression was given by M. Biot in 1844, viz.

$$\log \mathrm{F}=a+b\,a^t+c\,\beta^t \;\ldots\; (7)$$

The constants for this formula M. Regnault has calculated from the following values, obtained from the graphic curve, which represented his experiments:—

$t_0 = 0°$	$\mathrm{F}_0 = 4\cdot60$ mm.
$t_1 = 25$	$\mathrm{F}_1 = 23\cdot55$
$t_2 = 50$	$\mathrm{F}_2 = 91\cdot98$
$t_3 = 75$	$\mathrm{F}_3 = 288\cdot50$
$t_4 = 100$	$\mathrm{F}_4 = 760\cdot00$

whence he deduced

$$\begin{aligned} \log a_1 &= 0\cdot006865036 \\ \log \beta_1 &= \bar{1}\cdot9967249 \\ \log b &= \bar{2}\cdot1340339 \\ \log c &= 0\cdot6116485 \\ a &= +4\cdot7384380 \end{aligned}$$

where the third term $c\beta_1{}^t$ is negative. Between 0° and 100° centigrade, this expresses M. Regnault's results with very great exactness. Above this temperature, it gives results which become sensibly different from those of experiment.

For temperatures between 100° and 230° centigrade, M. Regnault obtained the following values for the constants in M. Biot's formula:—

	For the Air Thermometer.	For the Mercurial Thermometer.
$\log \alpha_1$	$= \bar{1}\cdot997412127$	$= 1\cdot997443007$
$\log \beta_1$	$= 0\cdot007590697$	$= 0\cdot01182377$
$\log b$	$= 0\cdot4121470$	$= 0\cdot4163766$
$\log c$	$= \bar{3}\cdot7448901$	$= \bar{4}\cdot9731198$
a	$= 5\cdot4583895$	$= 5\cdot4882878$

Where in the formula the second term is negative, and

$$\log F = a - b\,\alpha_1^{x} + c\,\beta_1^{x} \quad \ldots (8)$$

$$x = t^{\circ} - 100^{\circ}$$

which gives the relation of temperature in centigrade degrees and pressure in millimetres. The mercurial thermometer was constructed of crystal of Choisi-le-Roi.

Mr. Rankine, in 1849, urged some theoretical objections to the formulæ employed by M. Regnault, and proposed the following:—

$$\log p = A - \frac{B}{\tau} - \frac{C}{\tau^2} \quad \cdots (9)$$

$$\tau = 1 \div \left\{ \sqrt{\left(\frac{A - \log p}{C} + \frac{B^2}{4\,C^2}\right)} - \frac{B}{2\,C} \right\} \quad \cdots (10)$$

when $\tau = T + 461^{\circ}\cdot 2$ Fahr.

The constants for this formula are —

$$\left.\begin{array}{rcl} A & = & 8\cdot2591 \\ \log B & = & 3\cdot43642 \\ \log C & = & 5\cdot59873 \\ \dfrac{B}{2\,C} & = & 0\cdot003441 \\ \dfrac{B^2}{4\,C^2} & = & 0\cdot00001184 \end{array}\right\} \text{giving } p \text{ in lbs. per square foot.}$$

$A = 6\cdot4095$ giving p in inches of mercury.
$A = 6\cdot1007$ giving p in lbs. per square inch.

For accuracy this formula leaves little to be desired, but it requires considerable calculation, especially for finding the temperature from the pressure. Where so great accuracy is not required, the following simple formula gives results that may be relied upon for practical purposes over a large range of the scale:—

$$\log p = \frac{5\,(\text{T}-212)}{\text{T}+367} \;\dots\; (11)$$

$$\text{T} = \frac{2895}{5-\log p} - 367 \;\dots\; (12)$$

which gives the pressure p in atmospheres of 29·922 inches of mercury or 14·7 lbs. per square inch.

Example 1. — For instance, let T be given = 230° Fahr.,

$$\text{then } \log p = \frac{5\times 18}{597} = 0{\cdot}15075 = \log 1{\cdot}415.$$

At 230° Fahr. therefore the pressure is 1·415 atmosphere = $1{\cdot}415 \times 29{\cdot}922 = 42{\cdot}339$ inches of mercury = $1{\cdot}415 \times 14{\cdot}7$ = 20·8 lbs. per square inch, or $20{\cdot}8 - 14{\cdot}7 = 6{\cdot}1$ lbs. above the atmospheric pressure.

Example 2. — Again, let p = ·691 atmosphere

$$= \frac{2895}{5-\bar{1}{\cdot}8395} - 367 =$$

$$\frac{2895}{5{\cdot}1605} - 367 = 193^\circ{\cdot}99 \text{ Fahr.}$$

Example 3. — We may also calculate the case given in Example (1) by Mr. Rankine's formula; here $\tau = \text{T} + 461{\cdot}2$ = $230^\circ + 461^\circ{\cdot}2 = 691^\circ{\cdot}2$.

$$\begin{aligned} \log \text{B} &= 3{\cdot}43642 \\ -\log \tau &= \underline{2{\cdot}83960} = 0{\cdot}59682 = \log\, 3{\cdot}9521 \\[1ex] \log \text{C} &= 5{\cdot}59873 \\ -2\log \tau &= \underline{5{\cdot}67921} = \bar{1}{\cdot}91952 = \log.\ \underline{0{\cdot}8308} \\ &\qquad\qquad\qquad\qquad\qquad\quad 4{\cdot}7829 \end{aligned}$$

For lbs. per square inch $6{\cdot}1007 - 4{\cdot}7829 = 1{\cdot}3178$ = log 20·79 lbs.

For inches of mercury $6{\cdot}4095 - 4{\cdot}7829 = 1{\cdot}6266$ = log 42·33 inches.

These results are almost identical with those given by the preceding formula.

The following table may serve as a guide in the use of these formulæ, showing how far they are accurate and within what limits on the scale they may be used with safety:—

Temperature Fahr.	Pressure of Steam.				
	Regnault's Tables.	Tate's Formula. (11.)		Rankine's formula. (9.)	
	Inches.	Inches.	Error.	Inches.	Error.
− 25·6	·0126	·0099	− ·0027	·01113	− ·0015
+ 32·0	·1811	·1661	− ·0150	·1734	− ·0077
69·8	·7265	·7051	− ·0214	·7200	− ·0065
100·4	1·9410	1·915	− ·0260	1·936	− ·0050
150·8	7·6791	7·674	− ·0051	7·695	+ ·0159
212·0	29·9218	29·922	0	29·922	0
257·0	68·658	68·640	− ·018	68·65	− ·008
302·0	140·995	140·81	− ·185	140·87	− ·125
347·0	264·471	263·86	− ·611	264·20	− ·271
392·0	460·204	458·98	− 1·224	459·90	− ·304
437·0	751·866	750·31	− 1·556	751·98	+ ·12

The errors of the numbers given by the formulæ are placed beside them for comparison.

Table V.—Of the Pressure and Corresponding Temperature of Saturated Steam, obtained from the Tables of M. Regnault by Interpolation and Reduction to English Measures.

Pressure in lbs. per square inch.	Temperature in degrees Fahr.	Rise of Temperature for 1 lb. Pressure.	Pressure in lbs. per square inch.	Temperature in degrees Fahr.	Rise of Temperature for 1 lb. Pressure.	Pressure in lbs. per square inch.	Temperature in degrees Fahr.	Rise of Temperature for 1 lb. Pressure.
1	101·98	24·28	31	252·09	1·85	70	302·71	0·93
2	126·26	15·35	32	253·94	1·76	75	307·38	0·89
3	141·61	11·47	33	255·70	1·77	80	311·83	0·85
4	153·08	9·25	34	257·47	1·68	85	316·00	0·81
5	162·33	7·79	35	259·15	1·68	90	320·03	0·77
6	170·12	6·78	36	260·83	1·61	95	323·87	0·74
7	176·90	6·00	37	262·44	1·60	100	327·56	0·71
8	182·90	5·41	38	264·04	1·54	105	331·10	0·68
9	188·31	4·92	39	265·58	1·53	110	334·51	0·66
10	193·23	4·54	40	267·12	1·49	115	337·84	0·63
11	197·77	4·19	41	268·60	1·47	120	340·99	0·61
12	201·96	3·92	42	270·07	1·43	125	344·06	0·59
13	205·88	3·67	43	271·50	1·41	130	347·05	0·57
14	209·55		44	272·91	1·39	135	349·93	0·57
14·7	212·00	3·47	45	274·30	1·35	140	352·76	0·56
15	213·02	3·27	46	275·65	1·34	145	355·6	0·55
16	216·29	3·14	47	276·99	1·31	150	358·3	0·51
17	219·42	2·96	48	278·30	1·29	160	363·4	0·48
18	222·37	2·82	49	279·59	1·26	170	368·2	0·47
19	225·19	2·72	50	280·85	1·25	180	372·9	0·46
20	227·91	2·63	51	282·60	1·22	190	377·5	0·43
21	230·54	2·54	52	283·32	1·21	200	381·8	0·42
22	233·08	2·35	53	284·53	1·20	210	386·0	0·39
23	235·43	2·32	54	285·73	1·17	220	389·9	0·39
24	237·75	2·25	55	286·90	1·15	230	393·8	0·37
25	240·00	2·16	56	288·05	1·14	240	397·5	0·36
26	242·16	2·10	57	289·19	1·12	250	401·1	0·34
27	244·26	2·04	58	290·31	1·11	260	404·5	0·34
28	246·32	1·98	59	291·42	1·09	270	407·9	0·33
29	248·30	1·93	60	292·51	1·05	280	411·2	0·32
30	250·23	1·86	65	297·77	1·01	290	414·4	0·31
31	252·09		70	302·71		300	417·5	

On the Relation of Temperature and Density of Saturated Steam.

Notwithstanding the very numerous experimental researches on the relation of pressure and temperature of steam, the relation of temperature and density, which is equally important in the calculations of the steam-engine, has, till recently, been investigated by theoretical investigations alone. By the method of Dumas it was found that, in becoming vapour, a cubic unit of water expanded to 1669 cubic units of steam, and from this single datum the density and volume at all other temperatures has been calculated, on the assumption that steam follows the same laws of expansion and contraction, under the influence of temperature and pressure, as a perfect gas.

The gaseous laws, or the laws of the relation of volume, pressure, and temperature of a perfect gas may be enumerated as follows:—

1. Mariotte's or Boyle's law; the pressure or elasticity is inversely as the volume when the temperature remains the same. That is, if a volume of gas of 10 cubic feet volume, under a pressure of 15 lbs. per square inch, be subjected to a pressure of 30 lbs. per square inch, the volume will be diminished to 5 cubic feet; or, on the other hand, if the pressure be decreased to $7\frac{1}{2}$ lbs. per square inch, the volume will increase to 20 cubic feet. Expressed in a formula, putting P for the pressure when the volume is V, P_1 the pressure when the volume is V_1,—

$$\frac{P}{P_1} = \frac{V_1}{V} \dots (13).$$

2. Gay-Lussac's or Dalton's law; the expansion of a given weight of an elastic fluid under a constant pressure is $\frac{1}{459}$th part of its volume at 0^0 Fahr. for every degree of increase of temperature. Expressed in a formula this law is,—

$$\frac{V}{V_1} = \frac{459 + t}{459 + t_1} \dots (14).$$

Hence, also, if the volume be constant,

$$\frac{P}{P_1} = \frac{459 + t}{459 + t_1} \dots (15)$$

and combining the two formulæ

$$\frac{V \times P}{V_1 \times P_1} = \frac{459 + t}{459 + t_1} \dots (16)$$

that is, the product of the volume and pressure at one temperature, is to that product at another temperature, as the temperature in the first case to the temperature in the second, the temperatures being counted from the absolute zero, or a temperature of $-459°$ Fahr.

Now we have seen, that it has been determined experimentally for steam that when $t_1 = 212°$ Fahr., $P_1 = 14{\cdot}7$ and $V_1 = 1669$, and if we assume that steam is strictly gaseous, these data suffice for calculating the volume or density of the same weight of steam at any other temperature and pressure; substituting in (16) we get

$$V = 1669 \times 14{\cdot}7 \times \frac{459 + t}{671 \times P}$$

$$= 36{\cdot}5 \; \frac{459 + t}{P} \dots (17).$$

Thus, if we take from the preceding table of the relation of temperature and pressure the corresponding numbers, and substitute them for t and P in the above formula, we shall get the theoretical volume at that temperature and pressure. Thus from Table V. we have $t = 281°$ when $P = 50$ lbs., then

$$V = 36{\cdot}5 \; \frac{459 + 281}{50} = 540$$

that is, a volume of 1669 cubic feet at 212°, would be reduced to 540 at 281°, and of course the density increased in the inverse ratio.

From this well-known formula all the tables of the density of steam, with one recent exception, have been deduced, on which calculations of the duty of the steam-engine have been founded.

Although experimentalists have for some time questioned the truth of this theoretical formula, yet, up to a recent time, no reliable direct experiments had been made to test its truth. Yet a few years since, Dr. Joule and Professor Thomson announced, as the result of the application of the dynamical theory of heat, that for temperatures above 212° Fahr. there would prove to be a considerable deviation from the gaseous laws in the case of steam. In 1855, Professor Rankine gave a theoretical formula for the density of steam, confirmatory of Professor Thomson's views.* This formula deduces the volume from the latent heat, and is of the form

$$v - \mathrm{V}' = \frac{\mathrm{H}}{\mathrm{L}} \ldots (17)$$

where L is the latent heat of evaporation per cubic foot in foot pounds of energy, and H the latent heat of evaporation of one pound of steam in units of energy, and $v - v'$ is the increase of volume of one pound of the fluid in evaporating. As we have as yet not considered the subject of latent heat, we may express Professor Rankine's formula in another form, as giving the volume from the pressure and temperature. It is then

$$\mathrm{V}' = \frac{772\,\{1091{\cdot}7 - {\cdot}7\,(\mathrm{T} - 32)\} \times (\mathrm{T} + 461{\cdot}2)^2}{2{\cdot}3026\, v'\, \mathrm{p}\,\{\mathrm{B}\,(\mathrm{T} + 461{\cdot}2) + 2\,\mathrm{C}\}} + 1.$$

where $\frac{v}{v'}$ is the specific volume of the steam, v' the volume of one pound of water at the temperature T; p the pressure of the steam at T temperature in pounds on the square foot; log B = 3·43642 ; log C = 5·59873.

About the same time Mr. Tate made some experiments with ether, which led him to the conclusion that, at pressures somewhat above the atmospheric, the vapour of this substance does not follow the gaseous laws. These experiments led to a comprehensive series of researches, undertaken by Mr. Tate in conjunction with myself, to ascertain the density of saturated steam at all pressures, by a new and original method.

The general features of our method of ascertaining the density of steam, consist in vaporising a known weight of water in a

* Proc. Roy. Soc. Edinb. 1855. These views have been further developed by Mr. Rankine in his Manual of the Steam Engine and other Prime Movers, in which are given full tables of the density of steam, agreeing well with the experimental results about to be detailed.

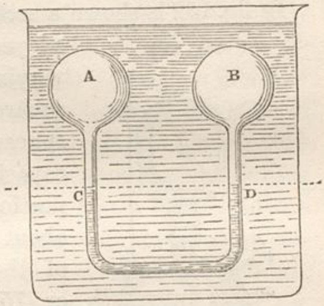
large glass globe—with a stem—of known capacity and devoid of air, and observing the exact temperature at which the whole of the water is just vaporised. Then, knowing the weight, volume, and temperature of the steam, its specific gravity may be calculated. In order to pursue this method with safety and with the requisite amount of accuracy, the following peculiarities of construction of the apparatus were adopted.

First, in order to secure the thin globe from bursting, and at the same time to have it uniformly heated, it is placed in a strong closed copper steam bath, having a thermometer and pressure gauge attached, and a strong glass tube, closed at its exterior extremity, for receiving the stem of the globe. By this arrangement the glass globe is secured from bursting, for whatever may be the elasticity of the steam, the internal pressure in the globe is balanced by the external pressure in the steam bath.

Second, when a given weight of water is vaporised in a closed vessel devoid of air, the steam is said to be in a state of saturation so long as any portion of the liquid remains in the vessel. But after all the water is vaporised, heat being still applied, the steam becomes superheated, or heated beyond the temperature just requisite for vaporising all the water. By way of distinction we call this point the maximum temperature of saturation. Now as we have to find by observation the temperature of the steam exactly at the point when the whole of the water is vaporised, the determination of this with sufficient accuracy and delicacy has hitherto formed the great practical difficulty attending experimental researches on the density of vapours. We have overcome this difficulty by using what may be called a *saturation gauge*, the form of which varies according to circumstances, but the principle on which it is constructed may be illustrated as follows:

Imagine two globes, A, B, fig. 151, connected by a bent tube containing mercury, and immersed in a large bath of liquid to secure uniformity of temperature; suppose these globes devoid of air but containing weighed portions of water, say twenty grains in A and thirty in B. If heat be now applied to the liquid bath so as to increase progressively the temperature of the globes, this weighed portion of water will gradually pass into steam, and the elastic force in each globe will increase in a ratio corresponding with the temperature, but without in the least affecting the uniformity of level of the mercury columns C and D, because the pressure on each side will be the same. But when the whole of the water in globe A has been evaporated, this equality of pressure will no longer exist and the column C will rise. The pressure in B increases in the ratio for saturated steam, whilst that in A increases in the much smaller ratio of superheated steam, and hence the difference of level of the columns. The instant at which the columns begin to rise on one side and fall on the other, is the point at which the whole of the water in A is converted into steam; and the temperature then noted is the maximum temperature of saturation. The following theoretical table gives approximately the rise of the mercury column at several temperatures: —

Fig. 151.



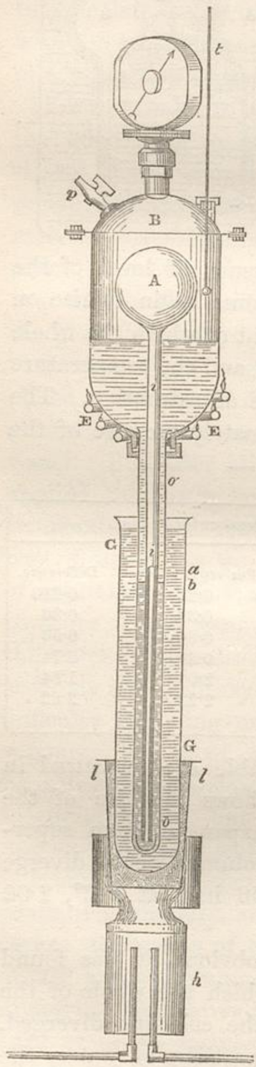
Saturated Steam.		Increments of Pressure for 1° Fahr.		
Pressure.	Temperature.	For expansion.	For vaporisation.	Difference.
At 4 lbs. and	152°	0·012	0·222	0·210
7 ,,	176°	0·022	0·32	0·30
15 ,,	213°	0·044	0·60	0·56
20 ,,	228°	0·060	0·80	0·74
61 ,,	295°	0·160	2·00	1·84
74 ,,	308°	0·200	2·22	2·02

The increments of pressure in this table are measured in inches of mercury. Their difference shows the rise of the mercury column on the side on which expansion from superheating is taking place. That is, the columns would diverge from the level ·210 inch at 152° F., 0·56 inch at 213°, 2·02 inch at 308°, and so on.

For reasons which will hereafter be obvious, it was found impossible to determine the instant at which the whole of the water in the globe was vaporised and the columns diverged. The cohesion of the glass to the last particles of water, the foggy condition of the steam, and other causes, rendered it necessary to superheat the steam a few degrees, and then having very carefully determined the difference of level of the columns, to estimate from these data the maximum temperature of saturation.

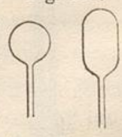
In fig. 152 is shown a sectional elevation of the apparatus employed in these researches for pressures varying from 15 lbs. to 70 lbs. on the square inch, or from one to five atmospheres. A is the glass globe of measured capacity for the reception of the weighed portion of water, drawn out into a stem about 32 inches long. The average size of the globes was $5\frac{1}{4}$ inches diameter or 75 cubic inches capacity; the stems were $\frac{3}{8}$ to $\frac{7}{16}$ inch bore. B B is the copper boiler, or steam bath in which the globe was heated uniformly throughout. The copper bath is prolonged by a strong glass tube, *o o*, $1\frac{1}{4}$ inches in diameter, and closed at the bottom; this tube is fixed to the boiler by a stuffing box, its upper part being trumpet-mouthed to prevent its being forced out by the pressure. The joint in the stuffing box was made by a ring of vulcanised india-rubber, which at the temperatures required in this series of experiments, answered its purpose perfectly. To heat this outer glass tube, which was peculiarly liable to explode, and, in fact, on two occasions did so, an outer oil bath, G G, was used, made of blown glass, twenty inches long, and resting in a sand bath, *l l*. This bath was supported on a tripod. The copper bath was heated by a coil of gas jets, E E; and the oil bath by a large wire gauge lamp, *h*, protected from draughts by a muffle, K K. The temperature thus obtained and distributed uniformly throughout the glass tube and steam bath by convection, was measured by a thermometer in the

Fig. 152.



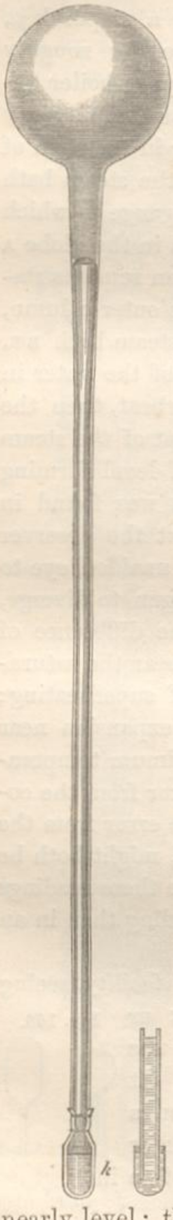
oil bath, and another *t*, exposed naked on the steam bath, and fixed in a stuffing box. Opposite the thermometer is a stopcock *p*, and on the top of the boiler a pressure gauge, for roughly indicating the pressure in the boiler. The copper boiler replaced the globe B in the diagram, fig. 151. The two mercury columns, the outer in the tube *o o*, and the inner in the stem of the globe *i i*, separate the vapour and water in the steam bath from that in the globe, and form the saturation gauge to which reference has been made. So long as the steam in the globe A remains in a state of saturation, the inner column remains stationary at a point a little above the level of the outer column, so as to balance the column of water in the steam bath B B. But when in raising the temperature the whole of the water in A is evaporated, and the steam begins to superheat, then the pressure of the steam in A no longer balances that of the steam in B, and the columns diverge: the difference of level forming a measure of the expansion of the steam. It was found in practice a matter of the utmost importance that the observer should not, in these experiments, trust to the unaided eye to determine the point at which the columns began to diverge, but that a careful series of measurements of the difference of level of the columns should be made, not only near the saturation point, but also at various temperatures of superheating; thus affording data for determining the law of expansion near the saturation point, and for estimating the maximum temperature of saturation from a point at which the error from the cohesion between the water and the glass, and the error from the retention of portions of water in the steam itself, might both be eliminated. It was also found advisable to take these readings of the levels of the columns, rather in a descending than in an ascending series of temperatures.

Fig. 153.



To read the column levels with rapidity and facility, seeing that they could not be approached within six or eight inches, a simple form of cathetometer was devised, sufficiently accurate for the purpose. It consisted of a telescope with cross wires sliding on a vertical graduated iron stem, and carrying a vernier for reading off the levels to the one-hundredth of an inch.

Fig. 154.



The steps in the process for determining the specific gravity of steam by this apparatus were as follows: —

A glass globule of a size to contain, as nearly as might be, the required quantity of water for vaporisation, was selected from a series (fig. 153). These globules had open stems, and after being filled with water were immersed hot in a cup of mercury, so that, in cooling, the mercury should rise into and fill the capillary stem. The weight of the water introduced was easily ascertained by deducting from the weight after filling, the weight of the dry cup, globule, and mercury. In this state the cup of mercury was transferred, and the globule passed into the large globe, in which a Torricellian vacuum had been previously formed.

To form the Torricellian vacuum, the globe, dried and filled with warm mercury, was heated on a sand bath until the mercury boiled; the stem was then filled with dry mercury, and the globe inverted, with its stem inserted in a basin of mercury. The globule was then introduced into the stem, and allowed to ascend into the globe. In order to transfer the globe from the basin to its place in the steam bath, a cup *k*, fig. 154, filled with mercury was suspended from the stem by an indian-rubber strap, a platinum wire being inserted between the cup and globe stem to ensure free passage for the mercury. The cover of the boiler B B being then taken off, and the outer tube *o o* dried and partially filled with dry mercury, the globe was raised and inserted into its place, resting on a tripod in the boiler. The cover was then fixed with a flax and red lead joint, and the cock *p* connected with an air pump. Exhaustion was effected, so that the columns in the globe stem and outer tube stood nearly level; the air pump was then removed, and a portion of water allowed to enter through the cock. The gas lights were then kindled; and until the water attained the boiling point, the columns were maintained level by means of the air pump, to prevent the possible entrance of water into the globe. After boiling for a time the cock p was closed, and the process of vaporisation went on simultaneously in the bath and globe, the temperature being kept sufficiently high in the oil bath G G to maintain the water in the outer tube in a state of ebullition. The temperature of the baths is slowly and uniformly raised, until the temperature of the vapour in the globe is considerably above the maximum temperature of saturation. After having been maintained for a considerable period at this temperature, the levels of the columns were observed; then the temperature being allowed to sink some degrees, the operation was repeated, and the temperature again reduced; and so on until the columns became stationary, indicating saturated steam in the globe as well as on the boiler. A series of readings was taken at each temperature, to make sure that the globe had attained a uniformity of temperature. At the same time the levels of some file marks on the stem were taken, by which the capacity of the globe in each position of the mercury column could be determined. All the elements were thus obtained for calculating the density of the steam.

Let W be put for the weight of distilled water at 39°·1 Fahr., filling the globe to the point at which the mercury columns stood at the maximum temperature of saturation. Let w be the weight of water vaporised; v the specific volume of the steam, or the number of times the volume of steam exceeds the volume of the water from which it is raised, then:—

$$v = \frac{W}{w} \dots (19).$$

By at once superheating the steam in the globe, and then slowly reducing the temperature until the maximum temperature of saturation is reached, we secure the following advantages:— The cohesion of the water to the surface of the glass being overcome, that force, it may be presumed, cannot be regained until the glass again becomes wet, which can only occur on condensation, that is, by the reduction of the temperature below that which corresponds to the maximum temperature of saturation.

Moreover, the observation of the columns at different temperatures of superheating, not only supplies us with data for ascertaining the maximum temperature of saturation, but also for determining the law of expansion of superheated steam near the saturation point.

The following table gives the temperature of saturation deduced from the experiments with the above apparatus from the two highest temperatures of superheating attained in each case. Where the lower of these temperatures is manifestly within the limits of imperfect expansion, the reduction from higher temperature only has been retained.

TABLE VI.—RESULTS OF EXPERIMENTS ON THE DENSITY OF STEAM AT PRESSURES OF FROM 15 TO 70 LBS. PER SQUARE INCH.

Number of Exper.	Maximum Temperature of Saturation, Fahr.		Pressure of Steam in Inches of Mercury.		Specific Volume of the Steam.
1	° 242·89 242·92	242·90	53·60 53·63	53·61	943·1
2	244·90 244·74	244·82	55·60 55·44	55·52	908·0
3	245·42 245·02	245·22	56·08 55·70	55·89	892·5
4	255·37 255·62	255·50	66·70 66·97	66·84	759·4
5	263·20 263·09	263·14	76·26 76·13	76·20	649·2
6	267·35 267·08	267·21	81·71 81·36	81·53	635·3
7	269·24 269·16	269·20	84·36 84·25	84·20	605·7
8	274·76		92·23		584·4
9	273·30		90·08		543·2
10	279·42		99·68		515·0
11	282·55 282·61	282·58	104·48 104·60	104·54	497·2
12	287·49 287·00	287·25	112·82 112·75	112·78	458·3
13	292·53		122·25		433·1
14	288·25		114·25		449·6

A similar series of experiments was obtained at pressures less than 15 lbs. per square inch, but in this case the saturation gauge was abandoned. The stem of the globe was immersed at bottom into a cistern of mercury open to the atmosphere; in other respects the method of the experiment was precisely the same. The water was introduced, the globe heated; and as vaporisation went on, the mercury column descended in proportion to the increase of the elasticity of the vapour. Simultaneous readings of a barometer were taken; and by deducting from the height of the mercurial column in the barometer, the height of that in the globe stem. So long as the vapour in the globe was in a condition of saturation, its elasticity thus found corresponded with that in M. Regnault's tables. When it became superheated, the ratio of increase of elasticity was very greatly reduced, and the column became almost stationary. The superheating was carried in these experiments to twenty or thirty degrees above the saturation point. The principle of the experiments was therefore entirely unchanged, the only alteration being that the elasticity of the saturated steam was obtained from previous experiments, and that of superheated steam observed, and the difference of level of saturated and superheated steam obtained by subtracting the one from the other, instead of being directly observed.

The following table gives the results obtained in this series of experiments reduced on the same principle as the last:—

TABLE VII.—THE RESULTS OF EXPERIMENTS ON THE DENSITY OF STEAM AT PRESSURES BELOW THAT OF THE ATMOSPHERE.

Number of Exper.	Maximum Temperature of Saturation, Fahr.	Pressure of Steam in Inches of Mercury.	Specific Volume of the Steam.
1	° 136·85, 136·70 } 136·77	5·36, 5·34 } 5·35	8275·3
2	155·38, 155·28 } 155·33	8·64, 8·61 } 8·62	5333·5
3	159·35, 159·35, 159·40 } 159·36	9·45, 9·45, 9·46 } 9·45	4920·2
4	170·88, 170·96 } 170·92	12·46, 12·48 } 12·47	3722·6
5	171·52, 171·44 } 171·48	12·63, 12·60 } 12·61	3715·1
6	174·92	13·62	3438·1
7	182·26, 182·34 } 182·30	16·01, 16·02 } 16·01	3051·0
8	188·30	18·36	2623·4
9	198·78	22·88	2149·5

These results show that the density of saturated steam at all temperatures above as well as below 212°, is invariably greater than that derived by calculation from the gaseous laws.

As we propose extending these experiments to higher pressures, it is premature to venture on any elaborate generalisation of the results we have attained. The following formulæ, however, express with much exactness the relation between temperature and volume, and between pressure and volume, as indicated by our experiments.

Let v be the specific volume of saturated steam, at the pressure P, measured by a column of mercury in inches; then

$$v = 25{\cdot}62 + \frac{49513}{\text{P} - 72} \;\ldots\; (20)$$

$$\text{P} = \frac{49513}{v - 25{\cdot}62} - 0{\cdot}72 \;\ldots\; (21)$$

The following numbers show the agreement of these formulæ with the experimental results: —

Temperature Fahr.	Specific Volume		Proportional Error of Formula.
	By Experiment.	By Formula.	
° 136·77	8275·3	8183	$-\frac{1}{90}$
155·33	5333·5	5326	$-\frac{1}{762}$
159·36	4920·2	4900	$-\frac{1}{246}$
170·92	3722·6	3766	$+\frac{1}{87}$
171·48	3715·1	3740	$+\frac{1}{149}$
174·92	3438·1	3478	$+\frac{1}{86}$
182·30	3051·0	2985	$-\frac{1}{46}$
188·30	2623·4	2620	$+\frac{1}{874}$
198·78	2149·5	2124	$-\frac{1}{90}$
242·90	943·1	937	$-\frac{1}{157}$
244·82	908·0	906	$-\frac{1}{451}$
245·22	892·5	900	$+\frac{1}{111}$
255·50	759·4	758	$-\frac{1}{759}$
263·14	649·2	669	$+\frac{1}{32}$
267·21	635·3	628	$-\frac{1}{91}$
269·20	605·7	608	$+\frac{1}{304}$
274·76	584·4	562	$-\frac{1}{26}$
273·30	543·2	545	$+\frac{1}{271}$
279·42	513·0	519	$+\frac{1}{128}$
282·58	497·2	496	$-\frac{1}{497}$
287·25	458·3	461	$+\frac{1}{152}$
292·53	433·1	428	$-\frac{1}{86}$
288·25	449·6	456	$+\frac{1}{75}$

We have also computed the following table from the experimental formula, which exhibits at a glance the pressure, volume, and weight of saturated steam, and will enable the reader to ascertain the necessary data for calculations at all pressures from 1 to 250 lbs. per square inch :—

General Table (VIII.) of the Relation of Pressure, Volume, and Weight of Saturated Steam deduced from experimental Data.

Pressure. In lbs. per sq. inch.	Pressure. In Inches of Mercury.	Specific Volume.	Decrease of specific Volume per lb. Pressure.	Weight of a Cubic Foot of Steam.
1	2·0361	17990·6	7633·0	·00347
2	4·0722	10357·6	3081·0	·00602
3	6·1083	7276·6	1666·0	·00858
4	8·1444	5610·6	1042·5	·01112
5	10·1805	4568·1	715·5	·01258
6	12·217	3852·6	521·0	·01620
7	14·253	3331·6	395·0	·01874
8	16·289	2936·6	311·2	·02126
9	18·325	2625·4	251·1	·02377
10	20·361	2374·3	206·9	·02630
11	22·397	2167·4	173·4	·02880
12	24·433	1994·0	147·3	·03131
13	26·469	1846·7	126·9	·03380
14	28·505	1719·8		·03630
Atmospheric 14·7 Pressure.	29·922	1641·5	110·4	·03803
15	30·541	1609·4	96·8	·03878
16	32·577	1512·6	85·7	·04127
17	34·614	1426·9	76·3	·04375
18	36·650	1350·6	68·5	·04622
19	38·686	1282·1	62·1	·04869
20	40·722	1220·0	55·6	·05117
21	42·758	1164·4	50·9	·05361
22	44·794	1113·5	46·6	·05606
23	46·830	1066·9	42·8	·05851
24	48·866	1024·1	39·3	·06096
25	50·902	984·8	36·4	·06339
26	52·938	948·4	33·8	·06582
27	54·975	914·6	31·4	·06826
28	57·011	883·2	29·2	·07068
29	59·047	854·0	27·2	·07310
30	61·083	826·8	25·6	·07550
31	63·119	801·2	24·0	·07792
32	65·155	777·2	22·5	·08032
33	67·191	754·7	21·2	·08272
34	69·227	733·5	18·9	·08510
35	71·264	713·4	17·9	·08751
36	73·299	694·5	16·9	·08988
37	75·336	676·6	16·1	·09227
38	77·372	659·7	15·4	·09463
39	79·408	643·6	14·8	·09700
40	81·444	628·2	14·1	·09937
41	83·480	613·4	13·4	·10040
42	85·516	599·3	12·1	·10416
43	87·552	585·9	12·0	·10654
44	89·588	573·8	11·4	·10880
45	91·625	561·8	10·9	·11111
46	93·607	550·4	10·5	·11342
47	95·697	539·5	10·4	·11571
48	97·733	529·0	10·1	·11801
49	99·769	518·6	9·4	·12037
50	101·805	508·5		·12276

Pressure.		Specific Volume.	Decrease of specific Volume per lb. Pressure.	Weight of a Cubic Foot of Steam.
In lbs. per sq. inch.	In Inches of Mercury.			
51	103·841	499·1	9·0	·12508
52	105·877	490·1	8·7	·12737
53	107·913	481·4	8·5	·12967
54	109·949	472·9	8·2	·13200
55	111·985	464·7	7·7	·13434
56	114·022	457·0	7·4	·13660
57	116·058	449·6	7·2	·13884
58	118·094	442·4	7·1	·14111
59	120·130	435·3	6·8	·14341
60	122·166	428·5	6·5	·14568
61	124·202	422·0	6·4	·14793
62	126·238	415·6	6·2	·15021
63	128·274	409·4	5·9	·15248
64	130·310	403·5	5·8	·15471
65	132·346	397·7	5·6	·15697
66	134·383	392·1	5·5	·15921
67	136·419	386·6	5·3	·16147
68	138·455	381·3	5·2	·16372
69	140·491	376·1	4·9	·16598
70	142·527	371·2	4·8	·16817
71	144·563	366·4	4·7	·17038
72	146·599	361·7	4·6	·17259
73	148·635	357·1	4·5	·17481
74	150·671	352·6	4·3	·17704
75	152·708	348·3	4·2	·17923
76	154·744	344·1	4·1	·18142
77	156·780	340·0	4·0	·18360
78	158·816	336·0	3·9	·18579
79	160·852	332·1	3·8	·18797
80	162·888	328·3	3·7	·19015
81	164·924	324·6	3·7	·19232
82	166·960	320·9	3·6	·19454
83	168·996	317·3	3·5	·19674
84	171·032	313·9	3·4	·19887
85	173·069	310·5	3·3	·20105
86	175·105	307·2	3·2	·20321
87	177·141	304·0	3·2	·20535
88	179·177	300·8	3·1	·20753
89	181·213	297·7	3·0	·20970
90	183·249	294·7	2·9	·21183
91	185·285	291·8	2·9	·21393
92	187·321	288·9	2·8	·21608
93	189·357	286·1	2·8	·21819
94	191·393	283·3	2·7	·22045
95	193·429	280·6	2·6	·22247
96	195·466	278·0	2·6	·22455
97	197·502	275·4	2·6	·22667
98	199·538	272·8	2·5	·22883
99	201·574	270·3	2·4	·23095
100	203·610	267·9	2·19	·23302
110	223·971	246·0	1·84	·25376
120	244·332	227·6	1·54	·27428
130	264·693	212·2	1·33	·29419
140	285·054	198·9	1·20	·31386
150	305·415	186·9	0·96	·33400

Pressure. In lbs. per sq. inch.	Pressure. In Inches of Mercury.	Specific Volume.	Decrease of specific Volume per lb. Pressure.	Weight of a Cubic Foot of Steam.
160	325·776	177·3		·35209
170	346·137	168·3	0·90	·37092
180	366·498	160·5	0·78	·38895
190	386·859	153·3	0·72	·40722
200	407·220	146·9	0·64	·42496
210	427·581	141·2	0·57	·44211
220	447·942	135·9	0·53	·45935
230	468·303	131·2	0·47	·47581
240	488·664	126·8	0·44	·49232
250	509·025	122·7	0·41	·50877

In the above table, which is probably the first calculated from direct experimental data, the third column is calculated by means of formula (20), and the last by dividing the weight of a cubic foot of water, at the temperature of 39°·1 Fahr., by the specific volume of the steam, that is, by the volume to which a cubic foot of water expands when converted into steam.

It will be interesting to compare the numbers given by the above table with those which are obtained in an entirely independent manner from Mr. Rankine's formula in which the volume is deduced from the latent heat. The following numbers show a near agreement in the results. With these has been placed at the same time a column giving the results deduced from the gaseous laws as they have hitherto been generally received.

Comparison of the Values of the specific Volume of Saturated Steam, from the Formulæ of Mr. Fairbairn and Mr. Tate, Mr. Rankine, and from the Gaseous Laws.

Temperature, Fahr.	Pressure in Inches of Mercury.	Specific Volume. Fairbairn and Tate.	Specific Volume. Rankine.	Specific Volume. Gaseous Laws.
104°	2·1618	17207	$19520 + \frac{1}{7}$	$19390 + \frac{1}{7}$
140	5·8578	7553	$7620 + \frac{1}{116}$	$7611 + \frac{1}{130}$
176	13·9621	3397	$3367 - \frac{1}{113}$	$3385 - \frac{1}{283}$
212	29·9218	1641·5	$1645 + \frac{1}{415}$	$1700 + \frac{1}{28}$
248	58·7116	858·7	$874 + \frac{1}{57}$	$941{\cdot}8 + \frac{1}{10}$
284	106·9930	485·3	$498 + \frac{1}{37}$	$516{\cdot}8 + \frac{1}{15}$
320	183·1342	294·9	$301 + \frac{1}{35}$	$316{\cdot}6 + \frac{1}{13}$
356	297·1013	191·9	$191 - \frac{1}{213}$	$204{\cdot}1 + \frac{1}{15}$

The fractions in the two last columns indicate the proportional deviation from the experimental formula. Within the limits of the experiments below the atmospheric pressure, that is, between 136° and 212°, both Mr. Rankine's and the gaseous results agree closely with our experiments. Above 212° both Mr. Rankine's and our own results deviate considerably from those obtained on the assumption of the gaseous laws, whilst at the same time the two former approximate nearly in value.

Fig. 155, represents graphically the relations which have been described in this section, the spheres representing the volume assumed by the same weight of steam at the respective pressures shown on the figure. It may also serve to show how the density increases and the volume decreases with the pressure according to the law determined by our experiments.

On the Latent Heat of Steam at different Pressures.

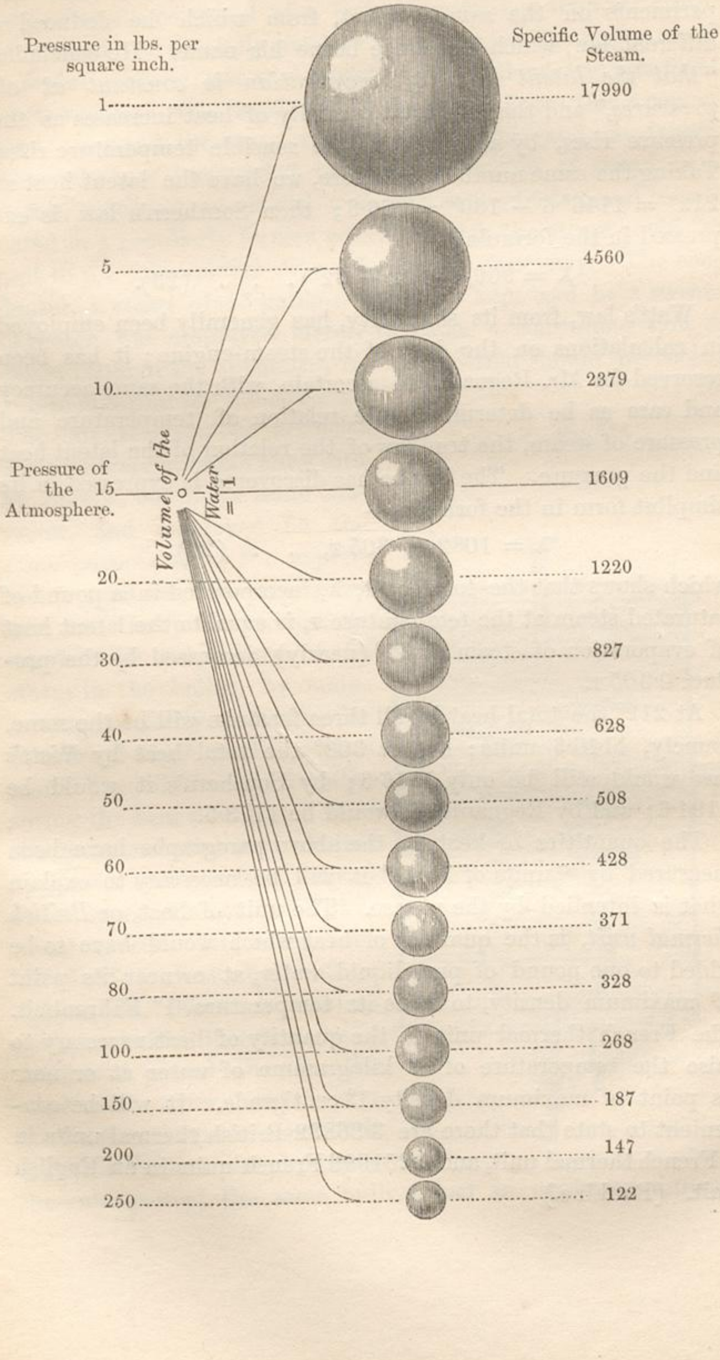
It has already been explained that in all changes in the state of aggregation of bodies heat becomes latent or sensible. If a body passes from the solid to the liquid, or from the liquid to the gaseous state, heat becomes latent; in the inverse process an equal amount of heat becomes sensible.

Black determined the amount of increase of heat in the water surrounding the worm of a still by the condensation of a weighed portion of steam, and found that the condensation of one pound of steam raised the temperature of an equal quantity of water 954° Fahr., an estimate which has since proved too low.

Watt investigated this subject in relation to the action of his condenser, and from his experiments concluded "*that the quantity of heat necessary to convert one pound of water at 32° into steam at any pressure is constant.*" That is, that the latent heat of steam decreases as the pressure rises, by as much as the sensible heat increases, the total heat being constant. If we take the total units of heat of steam at 212° to be 1146·6, and if λ be put for the total units of heat, and λ^1 for the latent heat at any other temperature T, then the law of Watt will be expressed by the formula —

$$\lambda = \lambda^1 + T = 1146{\cdot}6. \quad . \quad . \quad (22).$$

Fig. 155.


Pressure in lbs. per square inch.
Specific Volume of the Steam.
1
17990
5
4560
10
2379
Pressure of the 15 Atmosphere.
Volume of the Water = 1
1609
20
1220
30
827
40
628
50
508
60
428
70
371
80
328
100
268
150
187
200
147
250
122

Subsequently to Watt, in 1803, Southern made some experiments on the same subject, from which he deduced a different law, which has since borne his name. He concluded "*that the latent heat of evaporation is constant at all pressures*," and that the total quantity of heat increases as the pressure rises, by as much as the sensible temperature rises. Taking the same numbers as before, we have the latent heat at $212° = 1146°{\cdot}6 - 180° = 966{\cdot}6$; then Southern's law is expressed in the formula:—

$$\lambda = 966{\cdot}6 + (\text{T} - 32°) \quad . \quad . \quad . \quad (23).$$

Watt's law, from its simplicity, has generally been employed in calculations on the duty of the steam-engine; it has been reserved for Mr. Regnault to ascertain, with the same accuracy and care as he determined the relation of temperature and pressure of steam, the true law of the relation of the latent heat and the pressure. The law he has discovered is expressed in its simplest form in the formula—

$$\lambda = 1082 + {\cdot}305\,\text{T}, \quad . \quad . \quad (24)$$

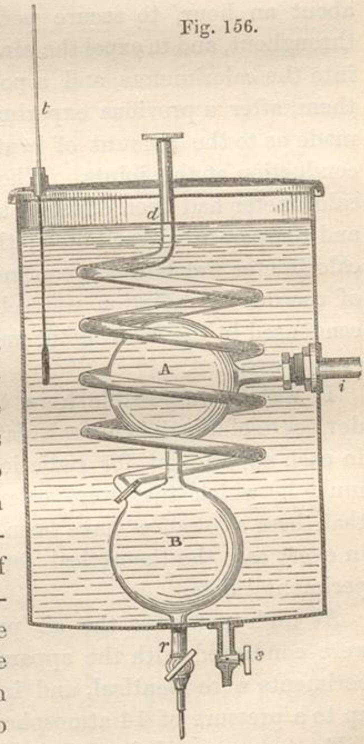
which shows that the total heat, λ, incorporated in a pound of saturated steam at the temperature T, is equal to the latent heat of evaporation of steam at 32° (nearly), increased by the product 0·305 T.

At 212° the total heat by all three formulæ will be the same, namely, 1146·6 units; but at 300° the total heat by Watt's law would still be only 1146·6; by Southern's it would be 1194·6; and by Regnault's it would be 1173·5.

The quantities of heat in the above paragraphs have been measured by "units of heat:" it will be necessary to explain what is intended by the phrase. The unit of heat, or *British thermal unit*, is the quantity of heat which would have to be added to one pound of pure liquid water, at or near its point of maximum density, to raise its temperature 1° Fahrenheit. The French thermal unit is the quantity of heat necessary to raise the temperature of 1 kilogramme of water at or near its point of maximum density 1° centigrade. It will be convenient to state that there are 3·96832 British thermal units in a French thermal unit, and 0·251996 French units in an English unit. (Rankine.)

The apparatus employed by Regnault to determine the latent heat of steam consisted of a boiler, condenser, artificial atmosphere and two precisely similar calorimeters. The boiler, made immensely strong, was of a capacity of 66 gallons, and was filled when required with newly-distilled water. From this a copper tube conveyed the steam to the calorimeters and condenser. This tube was surrounded by a dense stratum of steam in a jacket, which communicated with the same boiler, and terminated in a peculiarly formed cock, by which the steam could be sent to the calorimeters or condenser, as necessary. The condenser, a vessel of 13 gallons' capacity, kept cool by a stream of water, was employed merely to cause a continuous flow of steam through the apparatus, lest any portion should be cooled before entering the calorimeters.

Fig. 156.



The air receiver, or artificial atmosphere, communicating with air pumps, was of the same character, and employed for the same purpose as in the experiments upon the relation of pressure and temperature, viz. to regulate the temperature of the steam in the boiler, by maintaining perfectly uniform the pressure at which ebullition takes place. The calorimeters for measuring the heat disengaged were the most essential part of this apparatus, and consisted of two red copper cylinders, with thin metal covers. The worm consisted of a first bulb A, of red copper, ·078 inch in thickness, into which the steam to be condensed passed directly; the condensed water and steam thence passed into a second bulb B, with a cock r placed outside the calorimeter; the same bulb B, had an upper tubulure, by which it communicated with a copper worm. An agitator, or fan, of two discs of fluted copper, served to blend together the strata of water in the calorimeter during the experiment. The same volume of water was introduced into the calorimeters at every experiment, being measured in a gauging vessel. The mercurial manometer and forcing pumps were identical with those employed in the experiments on the relation of pressure and temperature. When complete, the apparatus was tested with a pressure of air of ten atmospheres, and every leak perfectly closed.

For experiments at the atmospheric pressure, every part of the apparatus exposed to condensation was covered with flannel. The distributing cock was placed so that no steam reached the calorimeters, and distillation was commenced and carried on for about an hour, to secure perfect uniformity of temperature throughout, and to expel the air. Cold water was then introduced into the calorimeters, and a portion of the steam sent through them, after a previous experiment for five minutes had been made as to the amount of heat communicated to the water by conduction at the joints. When sufficient condensation in the calorimeter had been effected, the cock was closed, and the time and temperature were noted; the agitation of the water in the calorimeter was however continued, and observations of the rate of cooling by radiation were obtained. The quantity of water condensed in the calorimeter was allowed to flow out at the cock *r*, and weighed.

It will be impossible here to enter in detail into all the devices to obtain data for calculating the corrections to be applied in each experiment for radiation, conduction, &c; nor the formulæ by which they were calculated. It is certain, however, that these allowances have been made with very great accuracy; in every case the theoretical formula has been checked by experimental data.

At high pressures the air pumps and artificial atmosphere were connected with the apparatus; in other respects the experiments were identical, and in this way results were obtained up to a pressure of 14 atmospheres, or 205 lbs. per square inch.

For pressures below that of the atmosphere the forcing pumps were replaced by the ordinary exhausting air pump, communicating with the reservoir of air. In this way experiments were obtained at pressures varying from 0·22 atmosphere to 0·64 atmosphere.

The most accurate formula for the latent heat of evaporation is Mr. Rankine's: —

$$l = 1091{\cdot}7 - 0{\cdot}695\ (\text{T} - 32^\circ) - 0{\cdot}000000103\ (\text{T} - 39^\circ{\cdot}1)^3$$

but for practical purposes the last factor may be omitted. In this way the following table of latent and total heat has been computed. The latent heat being calculated for a pound weight of steam, the units of heat required to raise the water from 32° to the boiling point are very nearly T — 32°.

Table IX.—The Latent and Total Heat of Steam from 1 lb. to 150 lbs. per Square Inch.

Pressure in lbs. per sq. inch.	Corresponding Temperature, Fahr.	Latent Heat of Evaporation in British thermal Units.	Total Heat from 32° Fahr. in British thermal Units.	Increment of total Heat per lb. Pressure.
1	101·98	1043·1	1113·1	7·3
2	126·26	1026·1	1120·4	4·6
3	141·61	1015·4	1125·0	3·5
4	153·08	1007·4	1128·5	2·7
5	162·33	1006·3	1131·2	2·4
6	170·12	995·5	1133·6	2·1
7	176·90	990·8	1135·7	1·9
8	182·90	986·6	1137·6	1·5
9	188·31	982·8	1139·1	1·5
10	193·23	979·4	1140·6	1·4
11	197·77	976·1	1142·0	1·3
12	201·96	973·2	1143·2	1·2
13	205·88	970·5	1144·4	1·0
14	209·55	967·8	1145·4	
14·7	212·00	966·1	1146·1	1·0
15	213·02	965·4	1146·4	
16	216·29	963·1	1147·3	·9
17	219·42	960·8	1148·2	·9
18	222·37	958·8	1149·1	·9
19	225·19	956·9	1150·1	·8
20	227·91	955·0	1150·8	·7
21	230·54	953·0	1151·5	·7
22	233·08	951·3	1152·3	·7
23	235·43	949·6	1153·0	·7
24	237·75	948·0	1153·7	·7
25	240·00	946·4	1154·4	·7
26	242·16	944·9	1155·1	·7
27	244·26	943·4	1155·7	·6
28	246·32	942·0	1156·3	·6
29	248·30	940·5	1156·9	·6
30	250·23	939·1	1157·5	·6
31	252·08	937·8	1158·0	·5
32	253·94	937·4	1158·3	·5
33	255·70	935·2	1159·0	·5

Pressure in lbs. per sq. inch.	Corresponding Temperature, Fahr.	Latent Heat of Evaporation in British thermal Units.	Total Heat from 32° Fahr. in British thermal Units.	Increment of total Heat per lb. Pressure.
34	257·47	934·0	1159·5	
35	259·65	932·8	1160·0	·5
36	260·83	931·6	1160·4	·5
37	262·43	930·4	1160·8	·5
38	264·04	929·3	1161·3	·5
39	265·57	928·2	1161·8	·4
40	267·11	927·1	1162·2	·4
41	268·59	926·1	1162·7	·4
42	270·07	925·1	1163·2	·4
43	271·49	924·0	1163·6	·4
44	272·91	923·0	1163·9	·4
45	274·28	922·0	1164·3	·4
46	275·65	921·1	1164·7	·4
47	276·97	920·1	1165·1	·35
48	278·30	919·2	1165·5	·35
49	279·57	918·2	1165·8	·3
50	280·85	917·3	1166·1	·3
52	283·32	915·6	1166·9	·3
54	285·73	913·9	1167·6	·3
56	288·05	912·2	1168·2	·3
58	290·31	910·6	1168·9	·3
60	292·51	909·1	1169·6	·3
65	297·77	905·3	1171·1	·3
70	302·71	901·8	1172·5	·3
75	307·38	898·5	1173·8	·3
80	311·83	895·1	1175·1	·26
85	316·00	892·2	1176·2	·26
90	320·03	889·2	1177·3	·22
95	323·87	883·9	1178·5	·22
100	327·56	881·3	1179·5	·22
105	331·10	878·9	1180·4	·20
110	334·51	876·5	1181·4	·20
115	337·84	874·2	1182·3	·20
120	340·99	872·0	1183·1	·20
125	344·06	872·0	1184·0	·19
130	347·05	869·9	1184·9	·18
135	349·93	867·8	1185·7	·16
140	352·76	865·7	1186·5	·16
150	358·3	861·7	1188·0	·15
200	381·7	845·0	1194·7	·13
250	401·1	831·2	1200·3	·11

On the Law of Expansion of Superheated Steam.

When steam is isolated from water and heated, it expands and decreases in density if the pressure be constant, or increases in pressure if the density (volume) be constant. In this state it is said to be *surcharged* or *superheated*, or, perhaps better, *gaseous* steam.

The earliest experiments on superheated steam are perhaps those of Frost in America, which, however, do not appear to be reliable. Mr. Siemens adopted a simple apparatus not long since, with which he sought to determine the rate of expansion of steam isolated from water, and which gave a high rate of expansion, but his conclusions are not at all borne out by my own experiments, and his apparatus does not appear calculated to yield very accurate results.

We are at this time prosecuting some researches on this subject, which are not yet advanced sufficiently for publication. In the meantime some results within a small range of superheating, obtained during the experiments on the density of saturated steam, approximate so closely to what might have been expected to be the law in this case, whilst at the same time they were made with great care, that I believe they are entitled to greater confidence than any previous attempts at the determination of this question.

Mr. Rankine, in the absence of data, has taken as the basis of his calculations on superheated steam, in his recently published "Manual of the Steam Engine," the assumption that superheated steam follows precisely the gaseous laws in its expansion under the influence of heat, that is, that

$$\frac{\text{v}}{\text{v}_1} = \frac{461{\cdot}2 + t}{461{\cdot}2 + t_1}$$

Mr. Siemens' experiments do not at all agree with this assumption; they would give a higher rate of expansion. But, with a certain proviso, my own results accord with it very nearly, and would seem to show that superheated steam expands at the same rate as a perfect gas.

The proviso to which I allude is this, that within a short distance of the temperature of maximum saturation, not exceeding about 20° Fahr., the rate of expansion is variable; close to the saturation point it is much higher than that of a perfect gas, but it rapidly decreases till, at a point at no great distance above the temperature of saturation, it becomes sensibly identical with that of a perfect gas. These results, however, do not extend over a sufficient range of temperature at present for us to deduce the true law, although their entirely independent coincidence with the laws already known to physicists is some guarantee for their accuracy.

By the rate of expansion we mean here the fraction expressing the increment of volume for one degree of temperature Fahrenheit; for air this fraction is:

$$r = \frac{1}{459 + t}$$

where t is the temperature of the gas. Thus at 212° the rate of expansion of a perfect gas is $\frac{1}{671}$, at 300° it is $\frac{1}{759}$, at 400 it is $\frac{1}{859}$; and so on at other temperatures.

Now in our experiments we may deduce the rate of expansion in a similar way, assuming it to be uniform for small increments of temperature; thus in experiment 6, in which the maximum temperature of saturation is 174°·92 Fahr., the coefficient of expansion for the steam between that temperature and 180° Fahr. is $\frac{1}{190}$, or three times that of air; whereas between 180° and 200° the coefficient is very nearly the same as that of air, being $\frac{1}{637}$ when air would be $\frac{1}{639}$; and the same rule is found in every experiment. The mean coefficient at zero of temperature from seven experiments below the atmospheric pressure, and calculated from a point several degrees above that of saturation, is $\frac{1}{438}$, whereas for air it is $\frac{1}{459}$, so that within the range of superheating obtained in these experiments the formula of expansion would be,

$$\frac{v}{v_1} = \frac{438 + t}{438 + t_1}.$$

The experiments seem to indicate that if the superheating had been carried further, the coefficient would have still more closely agreed with that which applies to incondensible gases. The following table gives the results upon which the previous generalisations have been founded, and which seem for the present the most reliable results we possess upon this subject. Before long I hope that we shall be able to lay before the public some direct experiments upon this subject, carried to a high degree of superheating.

The following table gives the value of the coefficient of expansion for superheated steam, taken at different intervals of temperature from the maximum temperature of saturation:

TABLE SHOWING THE COEFFICIENT OF EXPANSION OF SUPERHEATED STEAM.

No. of the Exper.	Maximum temperature of saturation.	Temperatures between which the expansion is taken.		Coefficient of Expansion of Steam.	Coefficient of Expansion of Air.
1	136°·77	140°	170°	$\frac{1}{593}$	$\frac{1}{399}$
2	155 ·33	160	190	$\frac{1}{336}$	$\frac{1}{619}$
3	159 ·36	159 ·36	170·2	$\frac{1}{150}$	$\frac{1}{618}$
		170 ·2	209·9	$\frac{1}{624}$	$\frac{1}{629}$
5	171 ·48	171 ·48	180	$\frac{1}{200}$	$\frac{1}{630}$
		180	200	$\frac{1}{604}$	$\frac{1}{639}$
6	174 ·92	174 ·92	180	$\frac{1}{190}$	$\frac{1}{634}$
		180	200	$\frac{1}{637}$	$\frac{1}{639}$
7	182 ·30	182 ·3	186	$\frac{1}{230}$	$\frac{1}{641}$
		186	209·5	$\frac{1}{630}$	$\frac{1}{643}$
8	188 ·30	191	211	$\frac{1}{604}$	$\frac{1}{650}$
1	242 ·9	243	249	$\frac{1}{317}$	$\frac{1}{702}$
4	255 ·5	257	259	$\frac{1}{392}$	$\frac{1}{716}$
		257	264	$\frac{1}{600}$	$\frac{1}{716}$
6	267 ·21	268	271	$\frac{1}{210}$	$\frac{1}{727}$
		271	279	$\frac{1}{640}$	$\frac{1}{730}$
7	269 ·2	271	273	$\frac{1}{232}$	$\frac{1}{730}$
		273	279	$\frac{1}{331}$	$\frac{1}{733}$
9	279 ·42	283	285	$\frac{1}{298}$	$\frac{1}{742}$
		285	289	$\frac{1}{333}$	$\frac{1}{744}$
13	292 ·53	297	299	$\frac{1}{281}$	$\frac{1}{756}$
		299	302	$\frac{1}{638}$	$\frac{1}{758}$

The limited extent to which the superheating is carried leaves the question of efficiency at higher temperatures unsolved. We believe we are perfectly conversant with the costly machinery that is used for this purpose on board ship and in other places, but until more reliable data have been determined by direct experiment, it would be premature to pronounce by what law the advantages assumed to result from its use are produced. We hope in entering on this inquiry to arrive at conclusions founded on the unerring principles of physical truth.

CHAP. VII.

VARIETIES OF STATIONARY STEAM ENGINES.

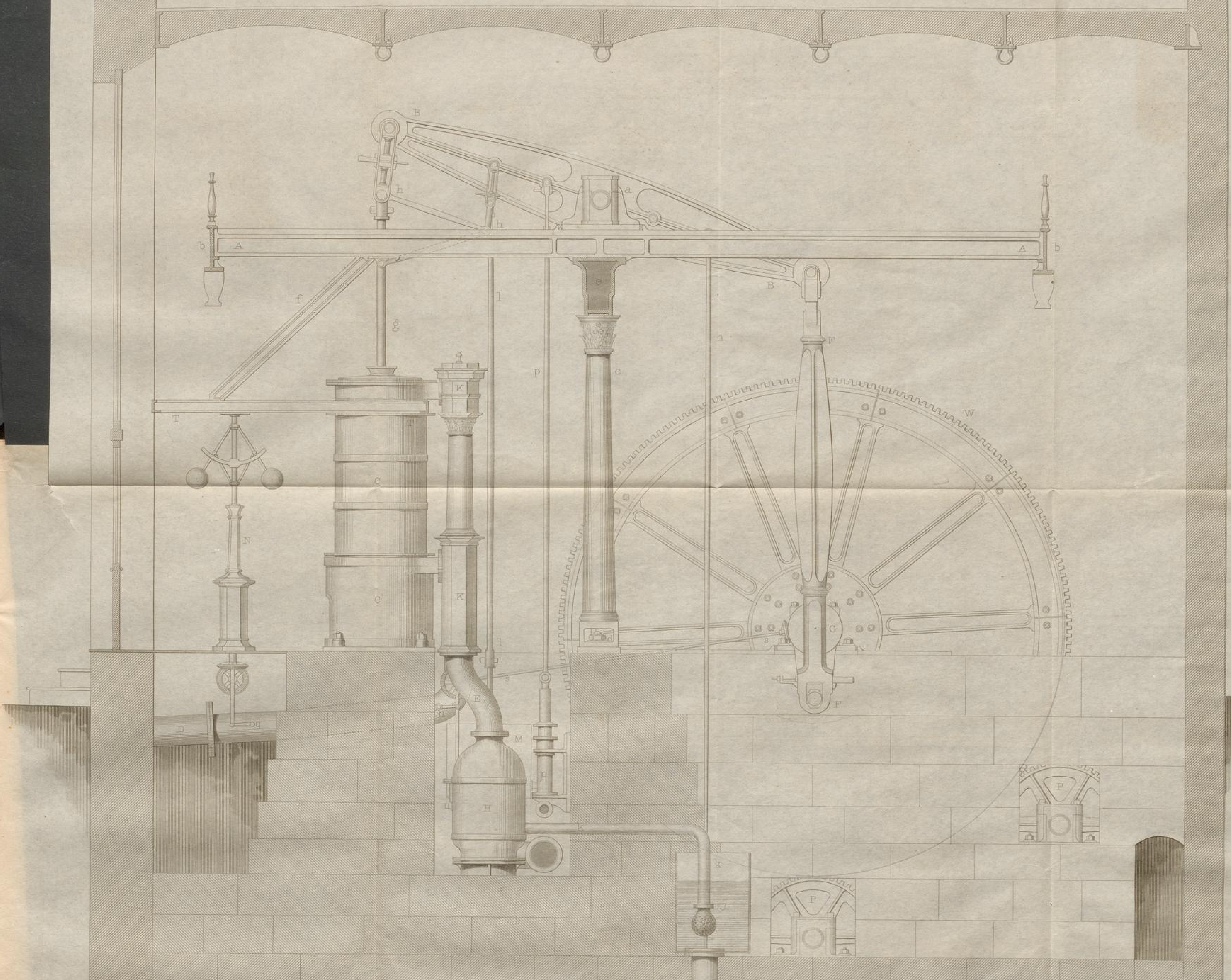
The steam engine as an instrument of propulsion is at the present time of such vast importance as to sink into insignificance every other known agent as a motive power. We have already considered the best methods by which the power of water can be utilised, but the whole of the water power in Great Britain falls immeasurably short of that obtained from steam, in every department of useful art. If we were to stop for a moment to compare the amount of steam power employed in industrial operations, with that of wind or water, we should find that the latter were mere fractions in the sum; and looking forward to still further developments in its application, I have taken some pains in the preceding chapter in giving a concise account of the properties of water when converted by the agency of heat into vapour or steam. I have considered these facts of vital importance to a knowledge of its economical employment and application, and I have dwelt longer on the inquiry than I originally intended, in order that I might have an opportunity of rendering accessible the results of experiments on the density of steam, and that the subject might be clearly and distinctly understood before treating of the construction of the steam engine.

It is not my object in this treatise to follow historically the many changes and improvements which have been effected in the steam engine since it left the hands of Watt. Suffice it to observe, that there has been no change in the principles of its action, unless we are to reckon as such the recent employment of gaseous instead of saturated steam. All the other improve-

Plate V

ELEVATION OF 100 HORSES POWER BEAM ENGINE
AT SALTAIRE.

Constructed by Mess^rs William Fairbairn & Sons

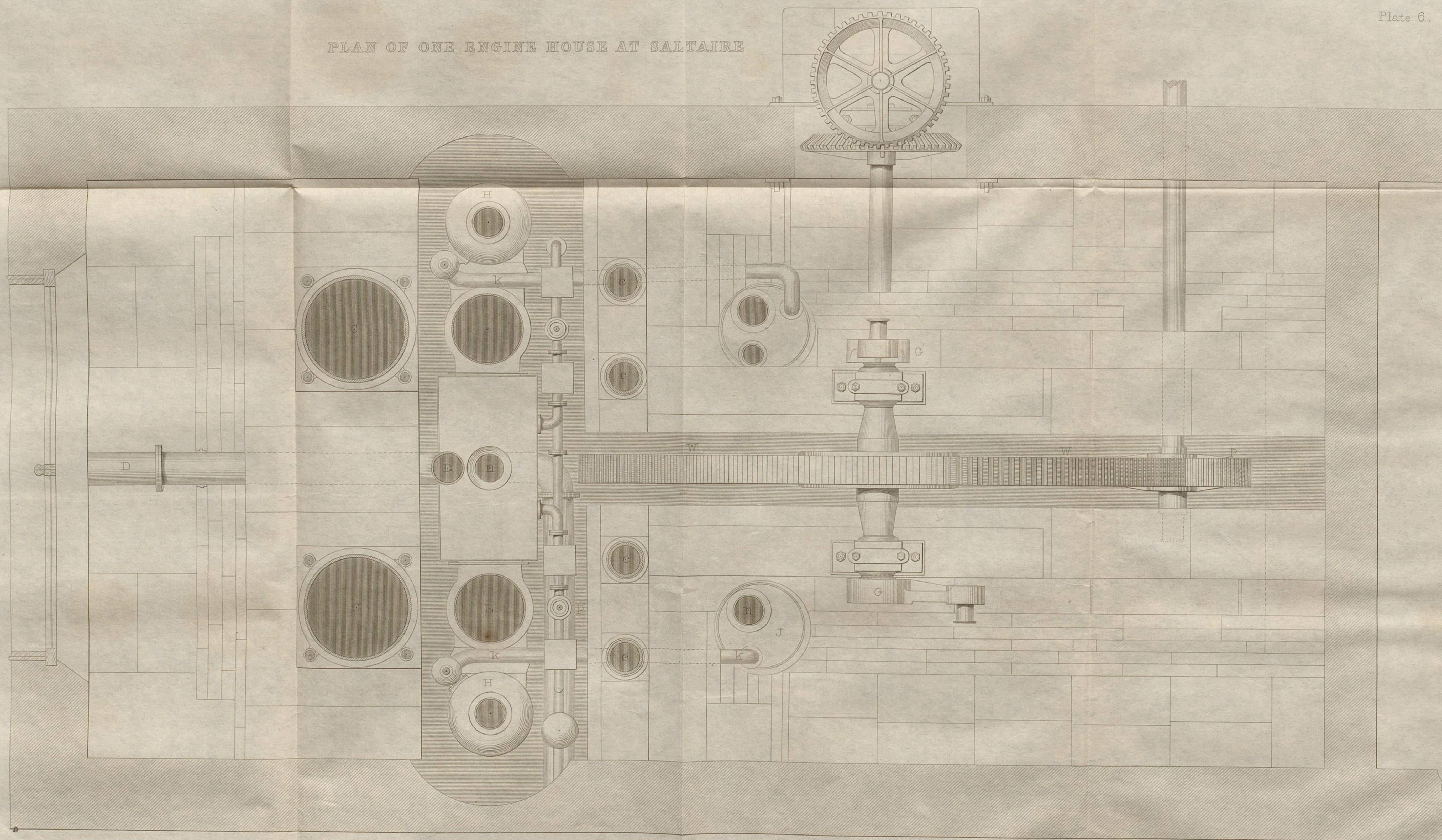


R. Adlard, sc.

Inches 12 0 1 2 3 4 5 6 7 8 9 10 11 12 Feet

London: Longman & C^o

PLAN OF ONE ENGINE HOUSE AT SALTAIRE



H. Adlard, sc.

Inches 12 0 1 2 3 4 5 6 7 8 9 10 11 12 Feet

London Longman & Co.

ments, in whatever form they present themselves, are confined to alterations of the organic parts of the engine, but have effected no change in the principle of action.

Taking then the condensing engine in its best and most economical form, I shall endeavour to lay before the reader some examples of the best and most recent construction, adapted for mill purposes in all the conditions of manufacture to which they are applied. In making a selection of those of medium size, I have chosen for illustration those at Saltaire, near Bradford, of 100 nominal horse-power each. Those engines are the best known for mills and factories, and the description of them will include the essential features of every other condensing beam engine.

Stationary Beam Engines.

At Saltaire the engines required to drive the machinery consist of two pairs of condensing beam engines, each engine being of 100 nominal horse-power, or collectively 400 nominal horse-power. They are placed on either side of the principal entrance to the mills, and are supplied with steam from boilers placed underground, at a short distance in front of the mill.

Plate V. contains a side elevation of one of these engines, and Plate VI. a plan of one engine-house with its pair of engines. The general arrangement will be understood when it is noticed that the power generated in the cylinders C, and transmitted through the working beam B B, to the large spur fly-wheel W, 24 feet in diameter, is taken direct from its circumference by the pinions P P, which give it off at the required velocity to the shafting of the mill.

The working beam B B is supported on two massive columns *c* 16 feet high, 14½ inches in least diameter, and 1½ inch thick of metal; these columns are bolted down beneath the whole mass of masonry supporting the engine. The heavy entablature *e* bolted to each column, and to the columns of the adjoining engine, is firmly fixed in the walls of the engine-house on each side, and the spring beams A A over this and at right angles with it are similarly attached to the cross beams *b b*. In this way an exceedingly strong and rigid support is secured for the main centre of the engine, which, resting in its pedestal *a*, has to sustain the principal strain of working. The spaces between the spring beams and the walls, excepting where the main beam vibrates, are filled with ornamental perforated metal plates, forming the beam-room, approached by the staircase *f*, for the purpose of oiling the centres, repairs, &c. The working-beam receives its motion from the piston-rod *g*, through the parallel motion *h h*, and transmits it by the connecting-rod F and crank G to the fly-wheel W.

The steam is brought from the boilers through a prolongation of the tunnel or flue in which the smoke passes to the chimney, and enters the engine-house by the pipe D. Having thence been admitted to the cylinder through the valve chests K K, it repasses after it has completed its work to the condenser H, through the eduction pipe E, in the usual way. The condenser is supplied with cold water from the river Aire, by the pipe *k k*, which communicates with the cold-water cistern J; the injecter through which the water enters the condenser is in these engines 6 inches bore, but the supply of water may be diminished if necessary by the injection gear hereafter described. Beside the condenser is the air-pump for pumping out the water and the air which enters with the water into the condenser, and is worked by the rod *l l* from the beam through a part of the parallel motion. A pump to supply the cold-water cistern is worked by the rod *n*, and another pump is worked by the rod *p p*, by which part of the hot water from the condenser is pumped back again for the supply of the boilers, in proportion as the water in them is decreased by its evaporation into steam. The supply of steam to the engine is regulated by the governor N acting on the throttle valve *q*, and thus the speed of the engine is kept uniform. A shaft *s s*, receiving motion from a bevel wheel on the crank shaft, works the equilibrium valves in the chests K K, as will be described; T T is a flooring or stage by which access is gained to the cylinder covers for oiling and cleaning. The cylinder is 50 inches in internal diameter, and has 7-feet stroke; it stands on the circular cylinder bottom C′, which is firmly bolted to the masonry by the long holding down bolts *r r*.

The length of the engine-house is 50 feet, and its breadth 24 feet. It will be seen that the two engines are combined so as to act in concert upon the same crank shaft and fly-wheel, the cranks being placed at right angles to each other, that when one engine is passing its top and bottom centres, and exerting least power, the other is in mid stroke and exerting its whole power upon the full leverage of the crank. In this way the action of the engines is equalised, and the motion rendered smoother than is possible with an independent engine, whilst, in case of accident to either of the pair, its fellow may be employed alone until the damage is made good.

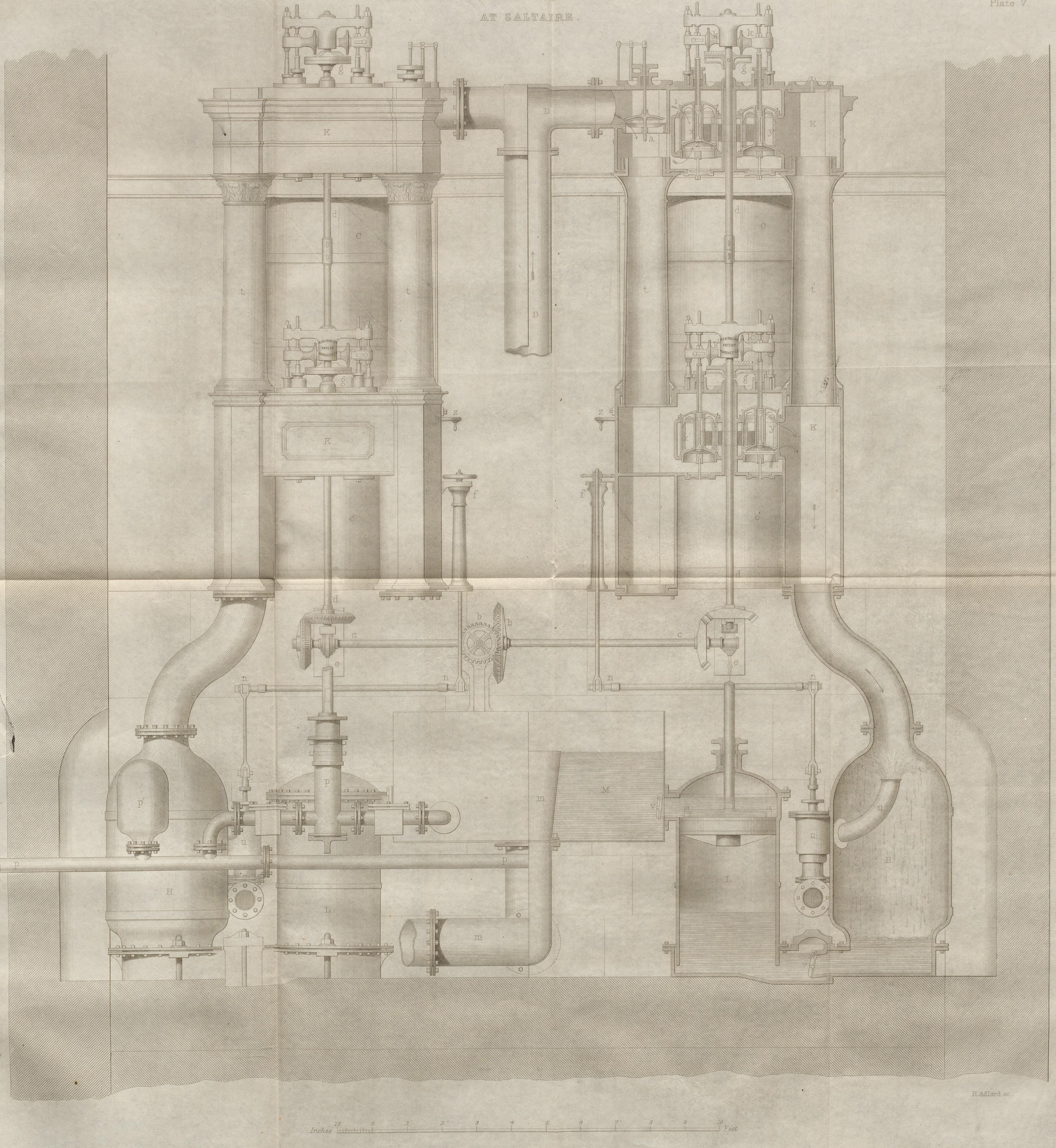
Plate VII. exhibits a half-elevation and half-section of the valve chests, condensers, air pumps, &c., of a pair of engines, showing the valves and the manner of working them. As before, C C are the cylinders, C^1 C^1 the cylinder bottoms, K K the upper, and K^1 K^1 the lower valve chests, fixed right over the cylinder ports and communicating by the side pipes t t^1. D D is the steam pipe, H H the condensers, L L air pumps, with their valves v v v; M the hot-well into which the air-pump lifts the water accumulating in the condenser. This water passes away by the overflow pipe m; p p p are feed pumps for supplying the boilers, with an air vessel p^1, for equalising the pressure and preventing any sudden shocks in the pipes; u, injection cock and injecter, the quantity of water admitted being regulated by the injection cock worked by the hand wheel f, through the medium of the small shafts and bell cranks n n.

The valves in these engines are of a peculiar construction, being modifications of the double beat or equilibrium valve, invented by Mr. Hornblower, and generally employed in the mining engines in Cornwall, where the high price of coal has led to that rigid economy for which its engineers have long been so justly famous. Most of the appliances for using steam expansively in rotative engines (*i. e.* in mill engines as distinguished from pumping engines), are open to the objections,—1st, of wire-drawing the steam; 2nd, of cutting it off too slowly; and 3rd, of leaving too much space between the cut-off valve and the cylinder, whereby much steam is wasted without producing its due mechanical effect. To remedy these defects I have employed the particular arrangement of valves shown in the plate, which are applicable to all rotative engines working expansively, whether with high or low-pressure steam.

The steam entering the upper steam chest K, through the stop valve *a*, has free access also to the lower steam chest K^1, through the side pipe *t*; whilst the exhaust steam has also clear access to the condenser, through the other side pipe t^1. The steam is admitted to the cylinder from the valve boxes by means of the valves x and x^1; and after having completed its work it passes through the exhaust valves y and y^1 to the condenser, these valves being opened and shut, alternately, at the right instant by an apparatus yet to be described. Each of the valves consists of two single conical valves x, 1 and 2, carefully secured together and accurately fitting their seats; the lower valve is slightly smaller than the upper. The steam is admitted on the upper and lower side of each of these pairs of valves and presses in opposite directions, so that the downward pressure on the upper valve is neutralised by the upward pressure on the lower, excepting that a slight preponderance is given to the former in consequence of the difference of area in the valves, in order to aid in keeping the valves firmly pressed upon their seats when released by the cams. Hence they lift with the greatest ease and expose any required opening for the admission or exit of the steam.

The mode of working these valves is very simple; a shaft *s s* (Plate V.), receives motion from the crank shaft and imparts it by the bevel wheels *b b*, to the horizontal shaft *c c*; this in turn gives motion to the valve spindles *d d*, which pass continuously through bearings in the valve chests and are supported on footsteps on the brackets *e e*. Upon each of these spindles are fixed two discs *g g*, carrying cams upon their upper surfaces, so arranged as to lift and release each valve at the proper instant of time. This is effected by a direct and simple action; the height of the cam corresponds with the lift of the valve, its length with the duration of the lift, and its position on the cam disc, which makes one revolution for every stroke, regulates the instant of time in the course of the stroke at which the valve is opened and shut. The action of the cams is transferred to the valves through the medium of friction pulleys *k k k k*, fixed upon small cross-heads, which are guided in their upward

CROSS SECTION OF ONE ENGINE HOUSE
AT SALTAIRE.



H. Adlard sc.

London Longman & C°

and downward motion by the brass standards in which they work. In the case of the steam valve these pullies are capable of adjustment by sliding them along the cross-head, towards or away from the valve spindle, so as to bring them over different parts of the cam, which is so arranged that the steam may be cut off at $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{4}$, or any required portion of the stroke, the remainder being effected by the expansion of the steam.

The exhaust steam requiring a full opening into the condenser, it is desirable to retain the exhaust valve fully open during the whole length of the stroke. By the present arrangement this is effected with a greater degree of certainty than by any other means hitherto proposed. The exhaust valves rise suddenly on the short inclined planes of the cams, and having allowed time for the escape of the steam through a wide passage to the condenser, they fall with equal celerity by their own weight; thus a more complete vacuum is formed under the piston than is perhaps possible to obtain by any other process.

The stop valve *a* is a simple conical valve, worked by a lever and hand wheel *z*, fixed by a bracket to the side of the steam chests, and is chiefly used for shutting off the steam from the engine.

The following diagrams were taken from these engines on May 4th, 1859. The engines were then working at 25 revolutions per minute, and one pair with part of the load off:—

Diameter of cylinder	50 ins.
Area "	1963·50 ins.
Speed of piston . .	350 feet per minute.
Scale of diagrams .	$\frac{1}{16}$ inch per lb. pressure.

Engine A.

From this diagram we get:—

	Lbs. per sq. in.
Mean pressure of steam . . . =	7·1684
Deduct for friction, &c. . . . =	2·0000
Effective pressure . . . =	5·1684

∴ Actual horses-power = 107·63.

INDICATOR DIAGRAMS.

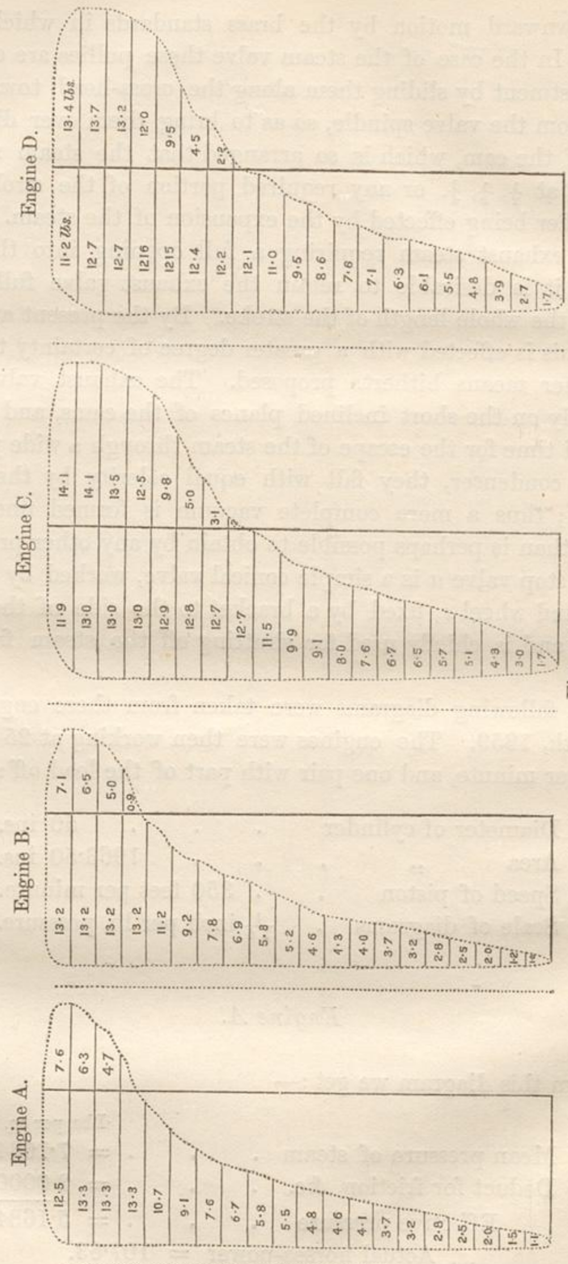


Fig. 157.

Engine B.

From this diagram we get:—

Mean pressure of steam . . .	=	7·3646
Deduct for friction, &c. . . .	=	2·0000
Effective pressure . . .	=	5·3646

∴ Horses-power = 111·46.

Engine C.

From this diagram we get:—

Mean pressure of steam . . .	=	13.301
Deduct for friction, &c. . . .	=	2·000
Effective pressure . . .	=	11·301

∴ Horses-power = 235·34.

Engine D.

From this diagram we get:—

Mean pressure of steam . . .	=	12·946
Deduct for friction, &c. . . .	=	2·000
Effective pressure . . .	=	10·946

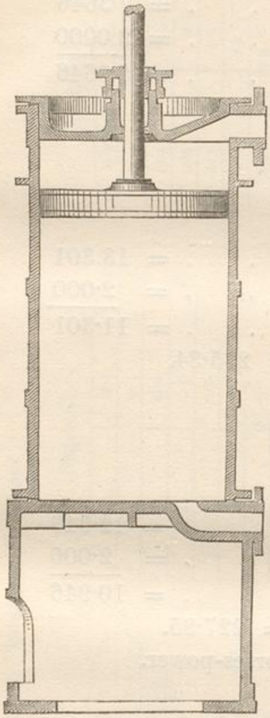
∴ Horses-power = 227·95.

Collectively 682·38 horses-power.

With a higher pressure of steam, or a shorter expansion, these engines will work to nearly double the above or 1200 horses-power.

Fig. 158 represents the cylinder, and its base, of the Saltaire engines. The cylinders are 50 inches internal diameter, with 7 feet stroke, of metal $1\frac{1}{2}$ inches thick. The ports are twenty inches wide, by 6 inches deep, so as to give $\frac{1}{16}$ the area of the piston for the admission and exit of the steam. The equilibrium valves have the upper disc 12 inches diameter, or 113 inches area, the lower disc $10\frac{1}{2}$ inches diameter, or $86\frac{1}{2}$ inches area, lift of steam valves $1\frac{5}{8}$ inches; of exhaust valves $1\frac{7}{8}$ inches. Steam pipes, $12\frac{1}{2}$ inches diameter; exhaust, 13 inches; condenser, 40 inches; air pump, $33\frac{1}{2}$ inches, and 3 feet 6 inches stroke. Beam 21 feet 6 inches long between the end centres, or over three times the stroke; and $3\frac{1}{2}$ feet deep in the middle, or $\frac{1}{6}$ the length. Main centre of wrought iron 12 inches diameter in the beam and 9 inches in the bearings. Spur fly wheel 24 feet 5 inches in diameter, with 230 cogs on the rim, 14 inches broad, 4 inches pitch; the rim is in 10 segments and has a sectional area of 200 square inches.

Fig. 158.



It is now more than thirty years since it was found desirable to increase the power of the steam engines employed in manufacture, and instead of engines of from 20 to 50 nominal horse-power, as much as 100, and in some cases 200 horse-power were required to meet the demand. To keep pace with the rapid extension of our manufactures, not only was the power itself doubled, and in some cases quadrupled, but a new class of men was brought into existence as mechanical engineers, and these, with the facilities afforded by new constructions and improvements of tools, gave to the manufacture of steam engines, and machinery of every description, an impetus that in a few years produced steam engines in an accelerated ratio of ten to one.

For some years previous to the great demand for power, the mills were driven by single engines, some as much as 50 or 60 horse-power, but these had soon to give place to others of much greater force, or, what was found to answer much better, two were employed coupled together as described above. Working in pairs, they were found to afford greater uniformity of action from the cranks being placed at right angles. Again, it was found that the speed of 240 feet per minute, considered as the maximum by Watt, was insufficient with the increasing demand for power, and speeds of 320 to 350 feet per minute are now become general. In some of the old engines, however, with such an increase of speed, the breakages became so numerous as to cause a retrograde movement, and a return to the old speed.

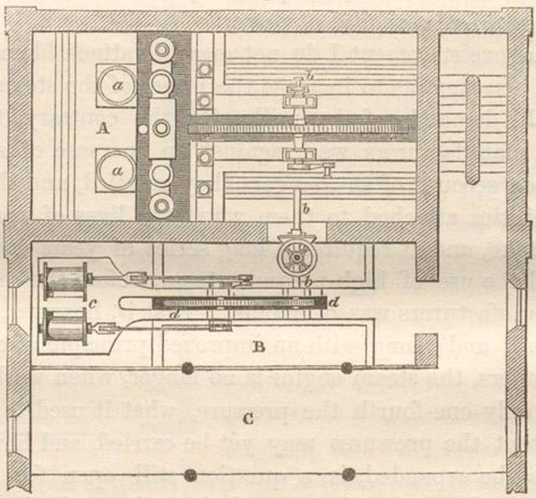
The increase of speed was, however, inadequate to meet the requirements for power in many cases, and the next resource was to increase the pressure of the steam. Unfortunately many of the boilers and engines were not calculated to withstand the forces to which they were thus subjected, and the result was an increase of the number of breakages and explosions to an extent that was ruinous to life and property. The ultimatum of all this was to increase the number of steam engines with an entirely new description of boiler, calculated to withstand higher pressures, and maintain the speed required to work the engine up to the required standard of power.

In the above statement I do not mean to attach blame to any person in his attempts to increase the power of the steam engine to meet the demands of the mill. On the contrary, the majority of manufacturers were against an increase of speed or pressure on account of the dangers they entailed, and the heavy responsibilities attached to them when the lives of workpeople were at stake, and it required a long series of years, in which I advocated the use of high-pressure steam, before the reluctance of the manufacturers was overcome. That is, however, now accomplished, and along with an improved principle of construction in boilers, the steam engine is no longer, when worked with steam of only one-fourth the pressure, what it used to be. To what extent the pressures may yet be carried, and how far the steam may be expanded, is a question still open for solution. But judging from what has already been done, the inference is that we have not as yet attained the maximum pressure, nor the rate of expansion calculated to afford the greatest economy in the use of steam as a source of power.

To accomplish the increase of pressure no change has taken place in the engine itself, beyond the strengthening of the parts, and the substitution of wrought-iron and steel for parts which were before considered sufficiently strong of cast-iron.

Where additional power was required in mills which could not be obtained by increasing the speed and pressure of the old engines, horizontal, high-pressure non-condensing engines were sometimes introduced, and these, in the manufacturing districts, are commonly called *Thrutchers.* These Thrutchers have been largely employed in Staleybridge and the surrounding district, and although not of great value on the score of economy, they are, nevertheless, important as auxiliary to the larger condensing engines. They are generally attached to the main gearing or first motion wheels, and the steam, which enters their cylinder at 50lbs. pressure, exhausts into the cylinder of the condensing engine, and is there expanded and worked over again. This system of double action would appear favourable to the expansive process, but unfortunately the distance

Fig. 159.

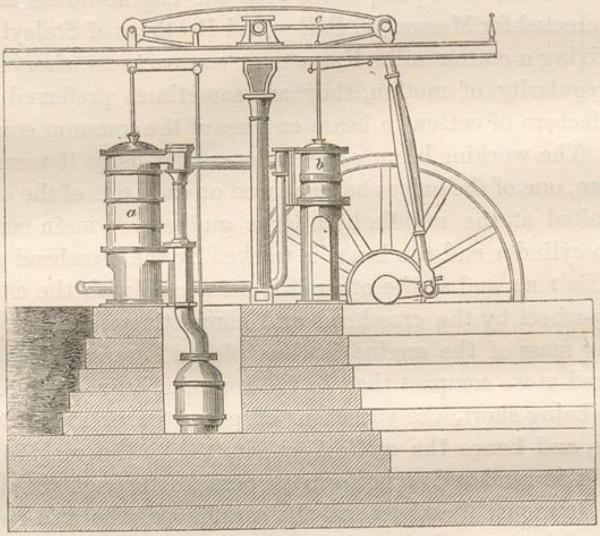


of the high-pressure cylinders from those of the condensing engine, and the consequent loss of heat by radiation and the retardation from friction, including the complicated nature of the connections, &c. is so great as to neutralise or destroy the economy of fuel which would otherwise have been secured.

There is, however, a considerable saving in original cost, which to a certain extent balances this drawback and renders the Thrutchers valuable as an auxiliary power. In cases where engines are overloaded and where it is impossible for want of space to erect new engines on the first principle of construction, the horizontal non-condensing engines are admissible, and may be used with advantage.

The annexed sketch (fig. 159) will explain the mode of connecting the horizontal high-pressure with the vertical condensing engines. At A are shown a pair of 60 horse-power engines with the cylinder *a a*, and fly-wheel *e e*. At B are the double high-pressure engines with their cylinders *c*, and their connection with the main shaft *b b*, by a spur fly-wheel and pinion *d d*. In this way all the four engines are united; a certain portion of the lower part of the mill being occupied with the auxiliary engines B,

Fig. 160.



and with a new set of boilers to supply the steam at the requisite pressure.

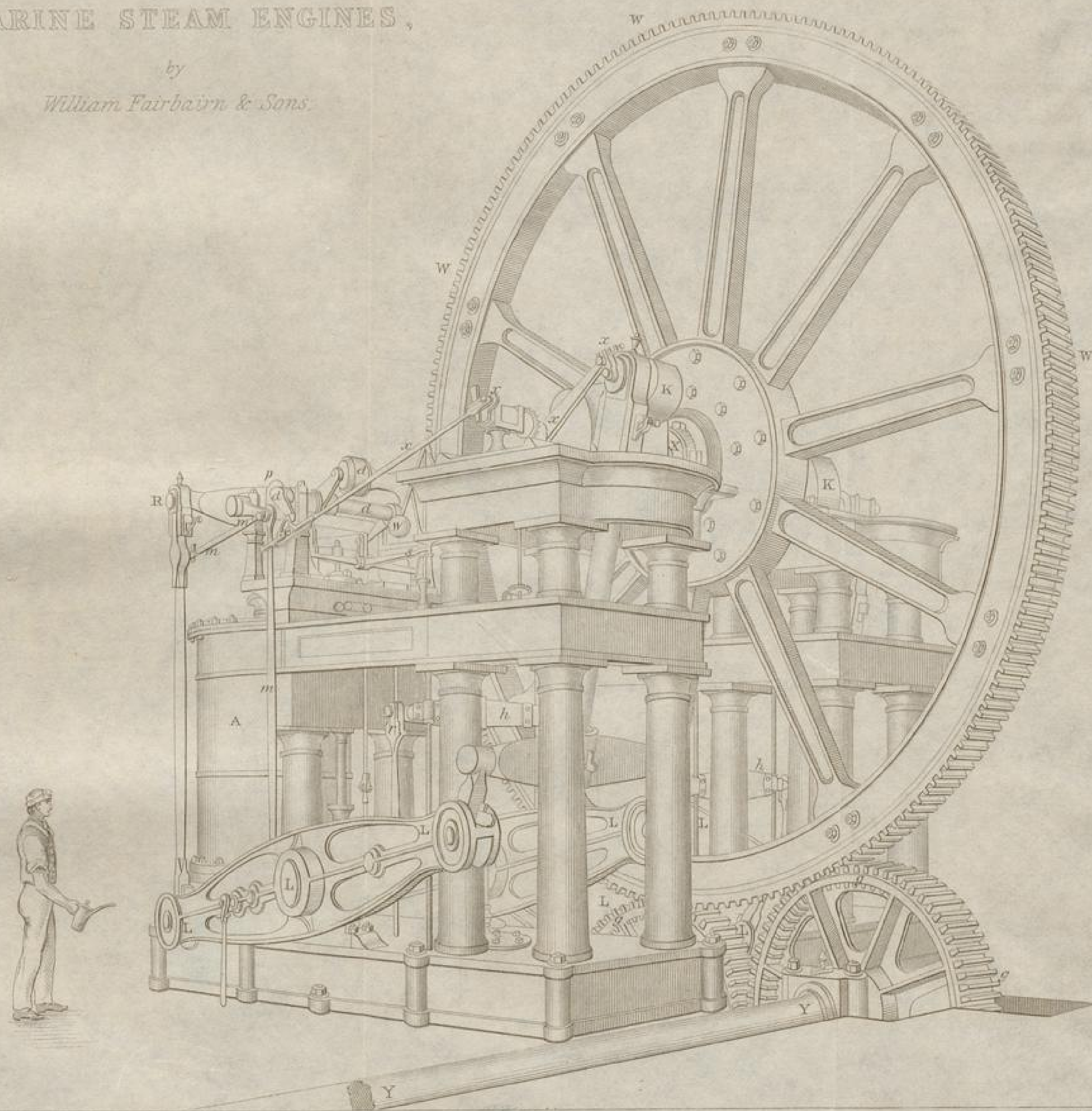
The great deficiency of power severely felt in many establishments, has been supplied in some cases by another method, without resorting to the erection of new engines, and that is by *McNaughting* the old ones. This process, the invention of Mr. McNaught, consists in increasing, or even nearly doubling, the power of a condensing engine by the introduction of a high-pressure cylinder attached to the same working beam, that is, provided the beam is strong enough to bear the increase of strain. This system of Mr. McNaught is simple and as effective as the Thrutchers, but it has unfortunately the same drawbacks as regards loss of heat and expenditure of fuel, although it must be admitted that the power of the engine is increased to a large degree. The mode of *McNaughting* is shown in fig. 160, where *a* is the cylinder of the condensing engine, *b* the high-pressure cylinder mounted on a pedestal to a convenient height and worked half stroke from the main beam at *c*. This it will be observed is a simple process, as it does not interfere with the main gearing of the mill, and in other respects also it has advantages over the Thrutchers already described.

Fig. 161 shows in perspective view the engine-house and engines erected for Messrs. W. Bailey and Brothers of Staleybridge, for driving a cotton mill. From their compactness, short stroke, and regularity of motion, they are sometimes preferred in the manufacture of cotton to beam engines of the common construction. The working beam or great lever L L L, is, as it were, split into two, one of the halves being placed on each side of the engine, but united at the middle by a large gudgeon or main centre L. At the cylinder end the beam is worked by the crosshead *p*, and side rods L R; and at the other it is connected with the crank of the fly-wheel by the crosshead and connecting rod L K L. The moving mass of the engine is thus placed lower and the whole rendered more compact than in the common beam engine. The stroke being short, the variations of power occur at shorter intervals, and hence the motion transmitted to the machinery of the mill is rendered as uniform as possible through the agency of the immense fly-wheel and the coupling of the two engines. The striking peculiarity of these engines is the large geared fly-wheel W W W W, formed of toothed segments, receiving the power of both engines, correcting its irregularities, and giving it out directly at its periphery, and at a high velocity, to the first motion shaft of the mill Y Y. Not only is the requisite speed of revolution very quickly attained in this way, but all intermediate trains of wheels are entirely dispensed with, and durability with simplicity secured in the highest degree. The pinion *g* which receives the power from the fly-wheel is geared with hornbeam teeth as an additional precaution for rendering the motion perfectly smooth and noiseless.

MARINE STEAM ENGINES,

by

William Fairbairn & Sons.



H. Adlard sc.

London: Longman & C°.

The steam pipe conducts the steam into an outer steam jacket round the cylinder A A; from this it enters the valve boxes similarly constructed to those of the Saltaire engines, but with short D valves instead of the equilibrium valves with which those engines are fitted. The piston is packed with metallic rings, and the steam ports are formed in the cover and bottom plate of the cylinder. The condenser is placed below the bottom valve chest, and near it the air-pump worked by a cross-head seen at *h*.

The valves receive motion by the following arrangement; a stud in the crank pin K carries round a small radius rod X X on an axis concentric with the crank; a smaller crank on this axis has a length equal to half the throw of the valve, or equal to that which would be given to the ordinary eccentric; and by a bar *x x* similar to the eccentric rod, the valve is moved by this lesser crank in the same way as by an eccentric; *m m m m* are the links of the parallel motion, *w* the governor, *s* the throttle valve.

From the above description it will be seen that the marine engine has some advantages over the long stroke beam engine as applied to mills. At an early period of my own practice I introduced it on an extensive scale, and there are numbers now at work, exclusive of those erected for Messrs. Bailey and Brothers, that are performing an efficient duty, and giving entire satisfaction.

With regard to economy of fuel, the marine engine is equal to the beam engine when worked on the same principle of expansion with equilibrium valves. It has besides the advantage of taking up less room in the mill, and having the whole of the action upon a lower basement, to which the frame of the engine is securely bolted. Every other engine, whether vertical, having a downward motion, or horizontal, has an advantage over the beam engine as regards space, and the distribution of its force through its organic parts direct upon the solid foundation.

But notwithstanding these advantages of marine and direct acting engines, they have not gained upon the old plan of a stationary beam engine for employment as the chief prime mover in mills. The reason that the beam engine has not been supplanted, appears to be, that its simplicity of construction and the facility of getting to every part in case of repairs being necessary, give it a superiority over every other form, however perfect and compact. Besides, the engineers (or *engine tenters*, as they are called in the manufacturing districts) find there is less trouble in cleaning, and there is therefore a desire on their part to have the old construction in preference to every other. From these considerations the old Boulton and Watt form of engine, strengthened and improved by being adapted to work expansively, is now the favourite, and is likely to maintain its ground as long as steam is depended on as the source of power in mills.

In the consideration of steam as a prime mover, it would be unjust to omit to notice the modification of Woolf, so extensively used on the Continent, where fuel is expensive, and where the greatest economy in its use has been an object of serious consideration.

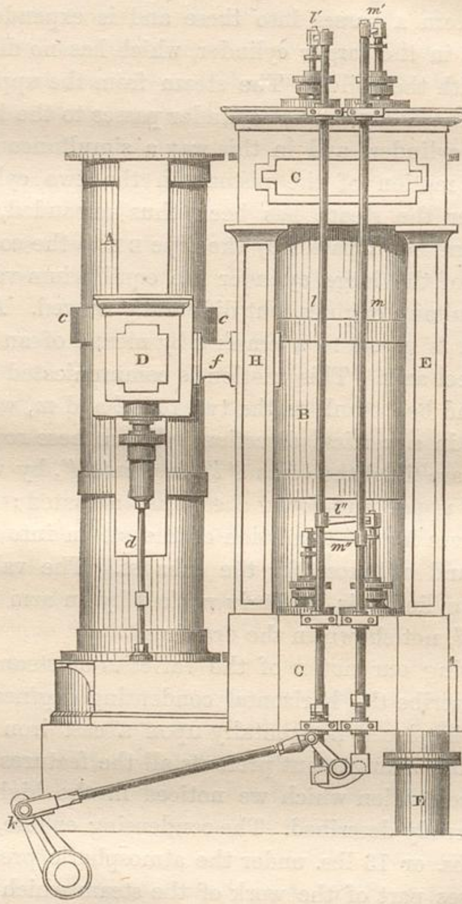
For the last half century Woolf's engine has been preferred in France and other countries on the continent of Europe, and this has arisen from the fact that until the last fifteen years the single cylinder engine has been worked with low pressure steam only, without expansion. Now it is evident that the single cylinder engine worked with full steam throughout the stroke, will require a larger expenditure of fuel than another engine worked expansively. Thus the double cylinder or compound engine, in which high pressure steam was employed, expanded through three-fourths of the stroke, appeared to effect a considerable saving of fuel; but taking both engines worked alike, with steam of the same pressure similarly expanded, as is now the case in the best single cylinder engines, there appears to be no advantage in the compound over the simple single cylinder engine. On the contrary, there is a loss in the original cost of the engine, and the complexity of the one as compared with the other. I have therefore no hesitation in recommending the single cylinder engine worked expansively, as an efficient competitor of the compound engine.

Fig. 161 is a view of the two cylinders and valve chests of a Woolf's engine. The small or high pressure cylinder is shown at A, 23 inches in diameter and 6 feet stroke, and the large or low pressure cylinder at B, 40 inches in diameter and 8 feet stroke; their contents being as 1 to 4. The steam is brought from the boiler by a pipe not shown in the drawing, which admits the steam into the annular passage *c c*, whence it passes into a valve chest D of the ordinary construction. This valve chest communicates by the passages *f* and H with the two valve chests C C of the large or low pressure cylinder B, and the exhaust steam from A passes into these and is expanded to four times its bulk in the larger cylinder, which has no direct communication with the boiler. The steam from the upper side of the piston of the high pressure cylinder passes to the lower part of the larger cylinder, and in this way a simultaneous upward or downward motion of the pistons in the two cylinders is secured. After the steam has been thus expanded, and the work so economised, it passes by the pipe E E to the condenser.

The valves of the large cylinder are equilibrium valves, like those of the Saltaire engines, but differently moved. A reciprocating motion is given to a crank *k* by means of an eccentric on the fly-wheel axis. This motion is communicated by a connecting rod and bell crank to the two rods *l* and *m*, which slide with freedom in a vertical direction. Upon these rods at suitable heights are fixed the arms $l' \, l''$ and $m' \, m''$, by which the valves in the upper and lower chests are actuated; l' and m'' lifting the valves for the admission of the steam into the cylinder, and l'' and m' those for the exhaust. The valve of the high pressure cylinder is similarly worked by an arm connected with the rod *d*, not shown in the drawing.

Before closing our notice of the varieties of steam engines, we have to describe the horizontal condensing engines in which the cylinder is placed horizontally upon a cast iron frame or bed plate. This arrangement presents all the features of cheapness and concentration which we noticed in the high pressure *Thrutchers* already described. The condensing engine, however, works to 12 lbs. or 13 lbs. under the atmospheric pressure, and thus economises part of the work of the steam which is lost in the *Thrutchers*, with the additional disadvantage that they work against a considerable back pressure. It is for these reasons that the high pressure non-condensing engines are not in demand where a large amount of power is required. They are, however, simple and effective, excepting as regards economy of fuel. In some cases they are preferable to the condensing engine, and that is in small establishments and as auxiliaries to water wheels when the supply of water fluctuates, and a small engine is needed when the supply is deficient. In large esta-

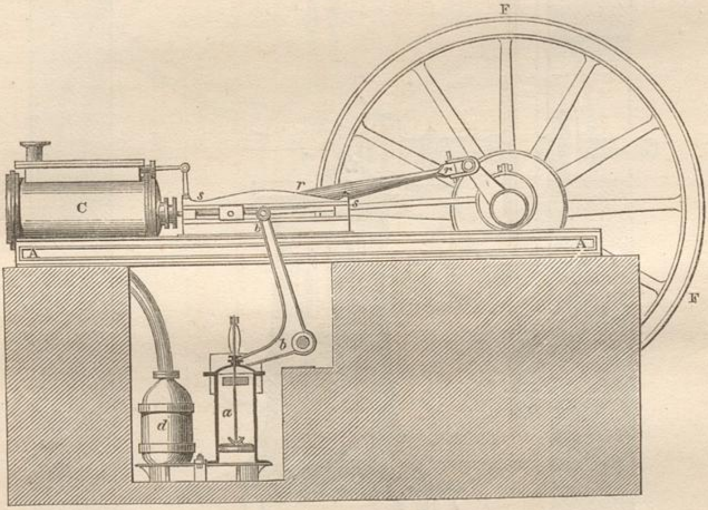
Fig. 161.



blishments, and more especially where coal is dear, the high pressure engine must give place to the condensing. The horizontal condensing engine presents some of the advantages of both classes of engines, as it is economical as regards fuel, and at the same time is lighter, more compact, requires less space, and costs less money than a beam engine. These are its merits. Its drawbacks are the horizontal position of the cylinder, involving unequal wear of the parts, and the tendency of the cylinder to become oval.

Fig. 162 represents a horizontal condensing engine, in which C is the cylinder, *s s* slide bars carrying the cross-head on the piston rod, from which is worked the fly wheel F F by the connecting rod *r r*, and the bell crank *b b*, by a short link. This bell crank transmits motion to the piston of the air-pump *a*,

Fig. 162.

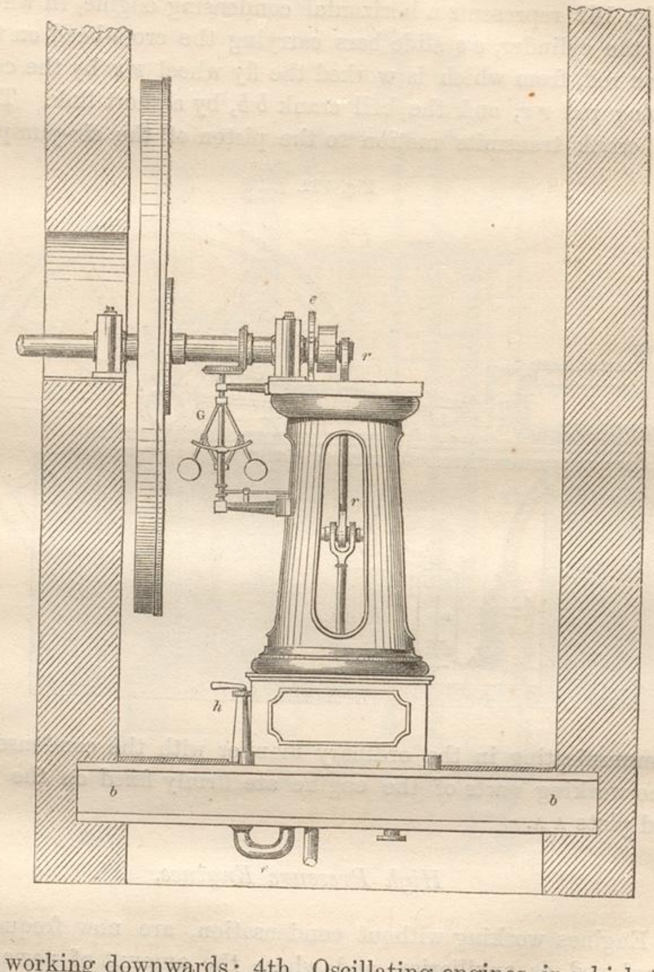


communicating in the ordinary manner with the condenser *d*. The working parts of the engine are firmly fixed on the iron bed plate A A.

High Pressure Engines.

Engines working without condensation, are now frequently employed as auxiliaries, and where the amount of power required is not large. It will not be necessary to describe here all their varieties, but they may be briefly enumerated as, 1st. Horizontal engines, like the last described, but without the condenser; 2nd. Columnar engines, in which the action is vertical, the framing being in the form of a hollow cylindrical column; 3rd. Vertical engines, with the cylinder placed above the crank

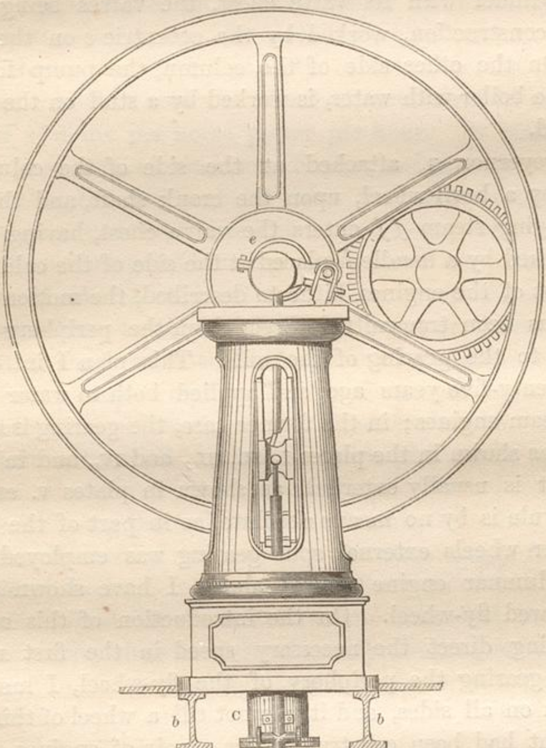
Fig. 163.



and working downwards; 4th. Oscillating engines, in which the crank is worked direct from the piston rod, and the cylinder oscillates upon trunnions near its centre, to allow of the requisite vibration of the crank; 5th. Steeple engines, in which the piston rod carries a cross-head, from which the connecting rod works downwards to the crank.

One example of the best of these varieties will be sufficient for our purposes in this place. Figs. 163 and 164 represent the columnar engine, which from its simple, compact, and neat form, is probably superior to most other constructions; it combines

Fig. 164.



several advantages from its vertical position, and the ease with which it is supported on two iron beams, *b b*, built into the walls of the engine-house for that purpose. In this way the necessity for heavy foundations is entirely done away with, and the boiler may be placed immediately below the engine, in order to be close to its work, and to save space. The annexed drawings represent a front and side elevation of an engine of this description of six horses' power, but the same principle has been successfully extended to engines of thirty horses' power. The piston rod is cottered in the usual way into a cross-head, carrying blocks which slide in fixed guides on each side of the columnar framing of the engine; the connecting rod *r r* is attached to this cross-head, and also to the crank overhead. C is the cylinder with its valve chest, the valves being of the short D construction, worked by the eccentric *e* on the crank shaft. On the other side of the column, the pump for supplying the boiler with water, is worked by a stud on the piston cross-head.

The governor G, attached at the side of the column, is worked by a bevil wheel, upon the crank shaft, and the pipe for supplying steam *f f* enters the valve chest, having a stop valve worked by a handle *h* placed at the side of the column.

In most of the engines hitherto described, the motion of the engine has been transmitted direct from the periphery of the fly-wheel to the gearing of the mill. This plan I introduced nearly twenty-five years ago, and applied both to water wheels and to steam engines; in the former case, the gearing is usually internal, as shown in the plates I., II., III., and IV., and in steam engines it is usually external, as shown in plates V. and VII. But this rule is by no means absolute, as in part of the Deanston water wheels external spur gearing was employed; and in the columnar engine figured above, I have shown an internal geared fly-wheel. On the introduction of this method of obtaining direct the necessary speed in the first motion shaft, by gearing the periphery of the fly-wheel, I met with opposition on all sides, and it was not till a wheel of thirty-six tons weight had been constructed for a pair of engines of 240 horses' power, that the more sceptical were convinced that the regularity of motion was not impaired, and that the train of geared wheels had been abandoned, with a material saving of power, and economy of prime cost. This system of connecting the prime mover with the machinery of the mill, greatly simplifies the motion, by attaining the required velocity at once, without the aid of speed gearing necessary where the motion is taken from the fly-wheel shaft. This system has now become universal in stationary steam engines.

The duty of engines, or amount of work done for a given quantity of coal consumed, has gradually improved as the engine itself has been modified. Smeaton's table of the effect of fifteen atmospheric engines at work at Newcastle in 1769 gives a mean of 5,590,000 lbs. raised one foot per bushel of coals per hour. This is equivalent to an expenditure of 29·76 lbs. of coal per horse power per hour. In Smeaton's own engine, erected at Long Benton, an improved duty of 9,450,000 lbs. raised one foot per bushel of coal was obtained, equivalent to an expenditure of 17·6 lbs. per horse power per hour. In Watt's engines the expenditure of fuel was further reduced to so large an extent that the payment for them was made proportional to their economy, one-third of the annual saving of fuel obtained by their use being paid during the term of the patent. In the earliest of Watt's engines without expansion the expenditure appears to have been about 8½ lbs. to 9 lbs. per horse power per hour. At the present time, in condensing engines working expansively, a duty as high as 2·6 lbs. of coal per horse power per hour has been obtained, or 11½ times the amount of work for the same consumption of fuel as in the early atmospheric engines.

The following tables have been carefully compiled by Mr. H. Harman, late chief inspector of the Association for the Prevention of Boiler Explosions in the Districts of Lancashire and Yorkshire, from the extensive returns furnished to that association. They show most significantly the progressive economy arising from the use of high-pressure steam, and from a long expansion.

TABLE FOR THE YEAR 1858-9.

Description of Engine.	Observed Pressure of Steam per Square Inch in the Boilers.															Maximum consumption of coal per horse power per hour.	Minimum consumption of coal per horse power per hour.
	15 lbs. and under.			16 lbs. to 30 lbs.			31 lbs. to 45 lbs.			46 lbs. to 60 lbs.			Above 60 lbs.				
	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.		
Condensing - -	20	969	10·6	193	24674	6·4	88	12338	5·7	6	679	3·8	—	—	—	17·0	2·6
Non-condensing -	—	—	—	3	36	14·0	10	509	9·9	6	399	8·7	—	—	—	23·0	5·5
Working compound	—	—	—	12	822	6·4	105	10340	5·8	201	23688	4·9	49	6796	3·9	11·0	3·0
Condensing. Steam cut off before $\frac{1}{4}$ stroke	—	—	—	37	4297	6·1	15	2320	5·2	2	175	5·2	—	—	—	9·6	2·6
Condensing. Do. $\frac{1}{4}$ to $\frac{1}{2}$ stroke	13	671	10·3	135	18837	6·1	75	10183	5·6	4	504	4·2	—	—	—	17·0	2·9
Condensing. Do. later than $\frac{1}{2}$ stroke - -	6	245	10·6	16	1577	6·7	—	—	—	—	—	—	—	—	—	16·7	3·0

TABLE FOR THE YEAR 1859-60.

Description of Engines.	Observed Pressure of Steam per Square Inch in the Boilers. 15 lbs. and under.			16lbs. to 30lbs.			31 lbs. to 45 lbs.			46 lbs. to 60 lbs.			Above 60 lbs.			Maximum consumption of coal per horse power per hour.	Minimum consumption of coal per horse power per hour.
	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.	No. of Engines.	Indicated horse power.	Average consumption per horse power per hour.		
Condensing - - -	4	203	8·3	35	4228	5·8	19	2480	5·8	2	79	4·5	—	—	—	9·8	3·4
Non-condensing -	—	—	—	1	15	—	5	332	9·4	3	194	6·8	—	—	—	11·9	5·1
Working compound	—	—	—	1	146	5·0	15	1938	4·5	18	1935	5·1	5	1012	4·1	8·0	3·0
Condensing. Steam cut off before ¼ stroke	1	28	—	15	1572	5·7	6	878	5·0	2	80	4·5	—	—	—	8·0	3·4
Condensing. Do. ¼ to ½ stroke	1	51	6·8	16	2429	6·0	13	1605	5·8	—	—	—	—	—	—	8·6	3·6
Condensing. Do. later than ½ stroke - - -	2	124	9·8	4	236	6·0	—	—	—	—	—	—	—	—	—	—	—

In reference to the apparent superiority of the compound engines in the last table, Mr. Harman observes "that owing to increased friction, &c., engines which have been compounded invariably indicate more horses' power than before, the machinery remaining the same; hence arises an advantage, apparent and not real, in calculating the consumption of fuel. . . . . Consideration has led me to conclude that the gross amount of power exhibited by compound diagrams as at present calculated is fallacious."

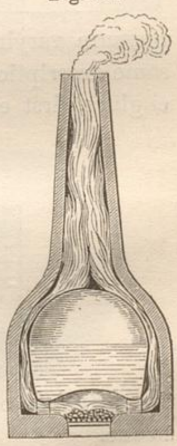
CHAP. VIII.

ON BOILERS.

Vessels for the generation of steam for supplying steam engines are of a great variety of forms, and are usually denominated boilers. These vessels require great care and judgment in their construction, in order that the fuel may be most economically applied, the waste and nuisance of smoke avoided, and the enormous force which steam is capable of exerting at high temperatures, safely restrained.

The boiler is, in fact, to the steam engine what the living principle is to animated existence. Like the stomach, it requires food to maintain the temperature, circulation, and constant action, which constitute the energy of the steam engine as a motive power. To keep up the temperature we have to feed, stoke, and replenish the furnace with fuel, and we may safely consider it a large digester, endowed with the functions of producing that supply of force required in the maintenance of the action of the steam engine.

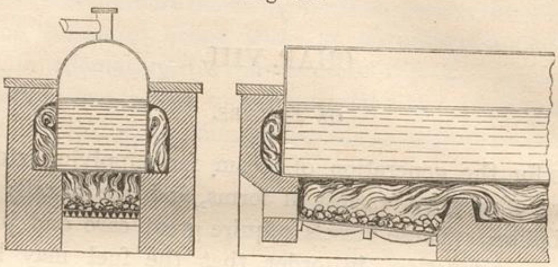
Fig. 165.



The boiler has undergone great changes of form and construction to adapt it to use. At first it was hemispherical, fig. 165, as when employed by Newcomen, which shape was retained for many years with certain modifications. Subsequently it was altered by Watt to the form of a parallelopipedon with a semi-cylindrical top, as shown in fig. 166. This form of boiler was extensively used by Watt in the early stages of his steam engines, and continued to take precedence of every other description of vessel employed for the production of steam. It was, however, modified by the introduction of a central flue, and a slight modification of its exterior shape to enable it to withstand greater pressures. Fig.

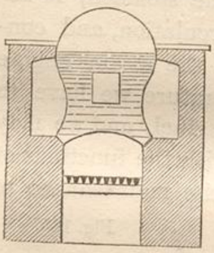
167 represents the improved form of boiler, in which, in addition to the curvature introduced along the bottom, the sides

Fig. 166.



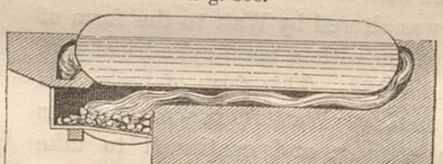
were also constructed in that form, the better to resist the pressure of the steam, which at that time was increased from 7 lbs. to 10 lbs. on the square inch.

Fig. 167.



Simultaneously with these improvements, Woolf and Hornblower introduced high pressure boilers of the cylindrical form, some with hemispherical ends, fig. 168, and others with flat ends having a cylindrical flue through the centre, fig. 169. Boilers of this sort were extensively used in Cornwall, where the pumping of the mines by steam engines on Woolf's plan required a pressure varying from 30 lbs. to 40 lbs. per square inch. The same description of boiler was adopted by Watt in his pumping engines, first erected in Cornwall, and worked expansively on

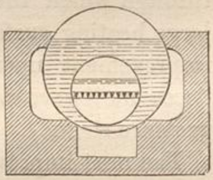
Fig. 168.



the principle of cutting off the steam at an early point in the stroke.

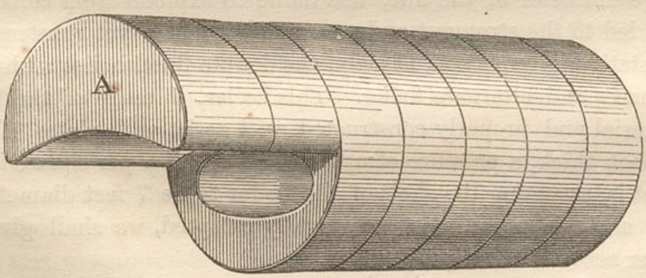
It was in the Cornish districts that the first great improvements in boilers took place. The high price of coal became an important item in the use of pumps for draining the mines, and every measure of economy was resorted to for a reduction of the cost. The result was the introduction of the cylindrical boiler, fig. 169, with a central flue, and a system of premiums to the superintendents for every bushel of coal saved in raising a given weight of water out of the mines. This system of premiums worked well in Cornwall, and I apprehend the steam engines of those districts are still worked with greater economy than in any other part of the kingdom. The Cornish miners pay more attention to their engines, are more careful of their boilers, and are stimulated to a more rigid economy than in any other part of the kingdom. They are never short of boiler space, and never force their fires or increase the power of their engines without increasing the capacity of their boilers. These conditions give to the Cornish engines the advantages which are lost sight of in other districts, to such an extent, in some instances, as to increase the consumption beyond all reasonable bounds. Of late years great improvements have been effected in this respect, and further progress in the same direction will doubtless lead to similar results in the economical use of steam.

Fig. 169.



Exclusive of the Cornish principle of construction, boilers have been introduced of a cylindrical form, with a central ellip-

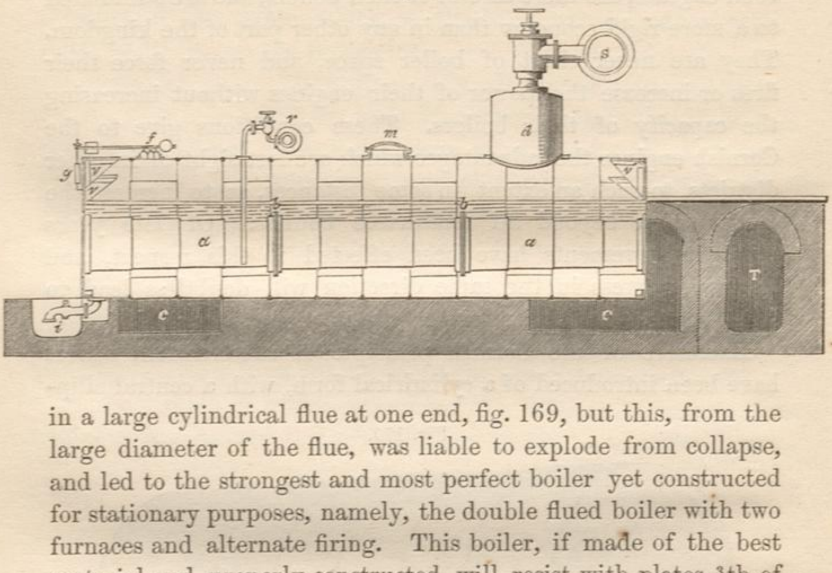
Fig. 170.



tical flue, and with the bottom cut away at one end to a distance of 8 feet, as shown in fig. 170, to admit a large furnace under that part. It will be seen that this boiler from the large concave arch at A, and the elliptical flue, inherits all the defects of the waggon form, fig. 167, and could not therefore be employed without danger of explosion, at pressures above 12 lbs. per square inch. From its peculiar shape it took the name of the *Whistlemouth*, or *Butterley* boiler, and with its internal flue, through which the products of combustion pass, it presents a large heating surface, and was for several years considered an improvement upon Boulton and Watt's boiler. Like its predecessors it gave way to others better calculated to generate high pressure steam.

The next improvement was the Cornish boiler, with a furnace

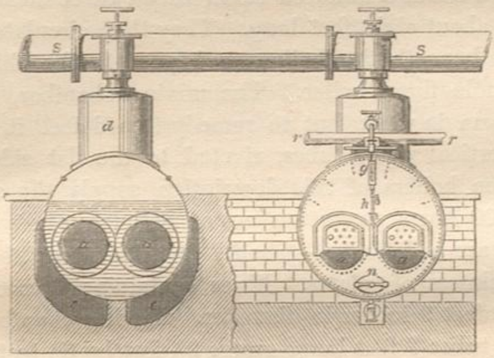
Fig. 171.



in a large cylindrical flue at one end, fig. 169, but this, from the large diameter of the flue, was liable to explode from collapse, and led to the strongest and most perfect boiler yet constructed for stationary purposes, namely, the double flued boiler with two furnaces and alternate firing. This boiler, if made of the best material and properly constructed, will resist with plates $\frac{3}{8}$th of an inch thick, a pressure of upwards of 300 lbs. per square inch (that is, assuming the shell of the boiler to be 7 feet diameter), and as it is now almost universally adopted, we shall give a fuller account of its proportions.

Fig. 171 exhibits a longitudinal section, and fig. 172 a front view, and a cross section of a double-flued stationary boiler, as I have been accustomed to construct it. It was originally devised with a view to alternate firing in the two furnaces, in order to prevent the formation of smoke; it consists of an external cylindrical shell with flat ends, and has two similar cylindrical flues, *a a*, passing through the water space of the boiler. The shell is six feet to seven feet in diameter, and twenty to thirty feet in length, and composed of $\frac{5}{16}$ to $\frac{3}{8}$ inch plates, riveted with lap joints, according to the pressure it is required to stand. The flues are usually two feet six inches, or two feet nine inches in diameter, leaving a space of about six inches all round, for the circulation of the water. These flues are strengthened by ribs *b b*, to prevent collapse, according to

Fig. 172.



the principles developed in my researches on that subject.* The boiler has flat ends, stayed by triangular gussets *v, v*. The furnaces are within the flues, and the products of combustion after passing through these, unite and pass beneath the boiler, towards the front, in the flues *c c*, where they turn and pass back again on the other side to the chimney flue T.

The ordinary fittings of these boilers consist of a stem dome *d*, on which is placed a nozzle or stop valve, communicating with the large steam feed pipe S, with which all the boilers communicate. Man-holes *m n*, are fitted to the boiler for cleaning and examination. The pipe *r* brings the feed water from the hot well of the engine in most cases, whence it passes

* Vide "Useful Information for Engineers," Second Series, pp. 1—45.

through a small stop valve down to the bottom of the boiler near the furnace end. Safety valves are shown at *f*, two having fixed weights, and a third being pressed down by a spring balance *g*. At the bottom of the boiler is a mud cock *i*, and there are usually a steam gauge for registering the pressure, and a glass water gauge *h*, for indicating the amount of water in the boiler.

Another form of boiler frequently employed for mills, is in part multitubular. There are two furnaces in two cylindrical flues, precisely similar to those in the preceding boiler, but immediately beyond the furnace; these flues unite into one chamber, in which the gases mix, and thence the gaseous products pass through about a hundred small tubes, three inches in diameter, and about eight feet long. They then circulate in brick work flues beneath the boiler, and pass to the chimney as in the double flued boiler. The mixing chamber, from its elliptical form is weak, but to remedy this defect it is stayed by three vertical water tubes, riveted to the flat sides of the ellipse. Ten boilers of this description supply the steam for the four 100 horse power engines at Saltaire; their principal dimensions are as follows:— Shell 7 feet diameter, 24 feet long, and $\frac{5}{16}$ thickness of plates. Flues containing furnaces 9 feet long, and 2 feet 9 inches in diameter. Mixing chamber 8 feet long, small tubes 7 feet long, grates $2\frac{1}{2}$ feet by $6\frac{1}{2}$ feet.

The heating surface in one of these boilers is as follows:—

Area of furnace flues	135	square feet.
„ of mixing chamber	102	„
„ of three vertical tubes	28	„
„ of small tubes	550	„
„ of exterior flues	285*	„
Total	1100	„

Area of firegrate, $33\frac{1}{2}$ square feet, being in the ratio of 1 to 32.

Messrs. J. and W. Galloway have patented a boiler in which a number of vertical, conical, water tubes, five or six inches in diameter at bottom, and twice as much at top, are introduced

* The area of the flues, 285 feet, under the exterior of the boiler is of little value, as the greater portion of the heat is absorbed by the time it has passed through the three-inch tubes.

into an elliptical flue passing through the boiler. The flame and heated gases circulate amongst these tubes and impinge against their sides. The dimensions of one of these boilers is as follows:—Length 24 feet; diameter 7 feet; greatest diameter of main flue, 5 feet 7 inches; the flue contains 21 vertical water tubes, acting as stays to prevent collapse, $11\frac{1}{2}$ inches in diameter at top, and 6 inches at bottom. These tubes are welded and placed zig-zag fashion, so that a man may creep along each side of the flue to clean or examine it. The two furnaces are each $7\frac{1}{2}$ feet long, $2\frac{3}{4}$ feet diameter.

Another boiler, known as the French or elephant boiler, is sometimes used. It consists of three cylindrical tubes with hemispherical ends, one larger than the other two, and placed above them. The smaller boilers communicate with the upper one by conical water tubes. The furnace is under the lower tubes, and the gases after passing the length of these, return underneath the larger boiler above, round which they circulate three times.

Messrs. Dunn and Co. manufacture what they term a retort boiler, in which the steam is generated in a number of retorts, or cylindrical tubes, about 9 feet long by 18 inches in diameter, placed *transversely* to the furnace. These all communicate with a large steam chamber above. This boiler is chiefly intended for exportation, being light and convenient for carriage in new countries without roads, or the usual means of conveyance.

Computation of the Power of Boilers.—In our attempts to give any definite rules on this question, we must state that there is hardly any branch of practical science so exceedingly anomalous and unsatisfactory as that of boiler power. In fact, there is no definite rule for our guidance, on the contrary, the whole is a jumble of guesses, and for years I have laboured in vain to reduce our past experience to something like a system; or to some reliable and definite rules calculated to guide us to correct results. This however appears to be impossible, as we are in a constant state of transition with a long vista of speculations before us, which seem likely only to lead to the point from which we started. It is like the smoke question, where every man is his own doctor, and promises much, while nothing is done. The only sound and definite principles of construction we have arrived at, pertain to the locomotive boiler, where the area of heating surface, capacity of fire box and grate area necessary to generate, with the aid of the blast pipe, the required quantity of steam, are well ascertained. But in land and marine boilers we have not as yet come to an agreement, and probably for this reason, that we have not the energy of an artificial draught, which in the locomotive increases or diminishes in proportion to the speed of the engine, and the strength of the blast respectively. Now this is not the case with the condensing engine, either on shore or afloat, and notwithstanding that there are many efficient and well-proportioned boilers doing their work well, we are nevertheless deficient in the knowledge of which under any given circumstances are the best construction and the most economical proportions.

We are still in want of an experimental investigation calculated to supply data on this subject. The trials hitherto made have been too partial and under too variable circumstances to be relied upon. As it is we must be content to take them as they now exist, under the hope that time may elicit greater certainty in the improved conditions now in progress. On some future occasion it is possible we may return to the subject, as it is one of deep importance in forwarding the manufacturing interests of the country.

Horses Power of Boilers.—The horses power of boilers is dependent in part on the capacity of the boiler itself, in part on the heating surface, and in part on the area of grate and the consumption of coal per hour. The common rule for estimating the horse power, is as follows:—Calculate the "*effective section*" of the boiler by adding to the diameter of the boiler the diameters of any internal flues and multiplying by the length of the boiler, and divide the product by the constant 5·5, 5·75, or 6, according to the practice of different engineers.

For condensing engines I have usually allowed about twelve square feet of "effective section" for each nominal horse power of the engine, although in practice many conditions necessitate the alteration of this proportion to suit circumstances. Now, as engines are at present constructed, working at from two to three times their nominal horses power, this is equivalent to an allowance of 5 square feet of "effective section" per indicated horse power, and hence agrees approximately with the rule given above. But this empirical rule is not at all to be relied upon, as it gives erroneous results with boilers of different forms and proportions.

The true method of calculating the proper proportion of boiler for any given engine is, however, to estimate the actual amount of steam required, which can easily be done with the aid of the tables, already given, of the weight and density of steam. Then provide a boiler capable of evaporating that weight of water, according to the data obtained in experiments with boilers of the particular construction employed. Some data of this kind will be given below. It being borne in mind that more heat is required, and less water evaporated with a given weight of coal, the higher the pressure at which the steam is employed.

Area of Heating Surface. — The total area of metal exposed to the flame and hot gases is called the *total heating* surface of the boiler, and is usually expressed in terms of the grate-bar surface. This unit of comparison has, however, been rendered ambiguous by the employment of another unit called the *efficient heating surface.* The efficient heating surface is obtained by deducting from the total heating surface one-half the area of vertical portions, and one-half the area of horizontal cylindrical flues, on the supposition that the vertical heating surfaces and the under side of flues and tubes act less efficiently in absorbing heat than horizontal surfaces above the flame.

A common allowance of effective heating surface for stationary boilers has been 10 to 15 square feet per square foot of grate area, and one square foot of grate is required per nominal horse power of the engine. I have usually allowed 16 or 17 square feet of effective heating surface; and in Cornish boilers 25 square feet is allowed. In general practice it will, however, be found that such a proportion as 17 will better serve the interests of the employers of steam engines than the extreme limits of 1 in 10 or 1 in 25; at least this is the best proportion for cylindrical flued boilers. The limits which define the amount of efficient heating surface are on the one hand the temperature of the gases escaping into the chimney, which should be as low as possible, and on the other the temperature of the boiler bottom, at which soot is deposited. If the gases escape at a higher temperature than is necessary to create a sufficient draught, heat is wasted by dissipation in the atmosphere, in consequence of insufficient heating surface. On the other hand, if the boiler is unduly increased, so that part of the heating surface is coated with soot, and the absorption of heat prevented, not only is boiler space wasted, but heat is lost by radiation.

In the Saltaire boilers the proportions of the heating surface may be estimated as follows: —

	Total heating surface in sq. ft.	Efficient heating surface in sq. ft.
Furnaces	135	68
Mixing chambers	102	51
Vertical tubes	28	14
Three-inch tubes	550	275
Exterior flues	285	192
	1100	600

Area of firegrate 33·5 square feet.

That is, 17 square feet of effective and 32 square feet of total heating surface per square foot of grate.

Again, in a double-flued tubular boiler, 30 feet long, 7 feet diameter, with two flues each 2 feet 8 inches in diameter, we have the following proportions: —

	Total heating surface.	Efficient heating surface.
Internal flues	504	252
Exterior flues	390	318
	894	570

Area of grates = 33 square feet.

Hence, there would be 27 square feet of total heating surface, and 17 feet of effective heating surface, per square foot of grate area.

Boiler Capacity.—In my practice I have always advocated large boilers. I have said before that boilers of limited capacity, when overworked, must be forced, and this forcing is the gangrene which corrupts and festers the whole system of operations. Under such circumstances perfect combustion is out of the question, and every attempt at economy fails. Usually with flued boilers I have allowed 15 to 20 cubic feet of boiler space per indicated horse power after deducting the flues. Mr. Armstrong contends for 27 cubic feet, of which one half is steam, and the other half water room. I have allowed one-third for steam and two-thirds for water where the boiler is fitted with a dome. When the steam-room is too small, the boiler primes, or water is carried over from the boiler with the steam.

Area of Grate-bar Surface. — The area of the grate depends upon the quantity and quality of the fuel to be burnt. In Cornish boilers, in which the combustion is slowest, only 6 to 10 lbs. of fuel are burnt per square foot of grate bar per hour; and in ordinary factory boilers about 14 to 16 lbs. is the average quantity. In marine boilers the combustion is still more rapid, and in locomotives it rises as high as 40 to 120 lbs. per square foot per hour.

The grate bars are ordinarily made to slope, to facilitate the pushing back of the fuel which has been partially coked on the dead plate. This slope varies from 1 in 5 to 1 in 25, being in cylindrical flued boilers somewhat restricted by the form of the flue. The firegrate terminates in a brick bridge, over which the flame and products of combustion pass into the flues. These bridges distribute the flame over the boiler bottom, and cause an eddy which facilitates the mixture and combination of the gaseous products.

Mr. D. K. Clarke has very carefully investigated the relations of grate-bar surface,heating surface, and consumption of coal, and has arrived at the following relations :—1st. For a given area of grate the total hourly consumption of fuel should vary as the square of the total heating surface; that is to say, if the heating surface be doubled, the total consumption of fuel might be increased four times, whilst the same evaporative efficiency would be maintained. 2nd. For a given extent of heating surface, the total hourly consumption should vary inversely as the area of the grate. For instance, if the grate surface were increased to twice the area, the total hourly consumption of fuel should be reduced to one-half, in order to maintain the same efficiency. 3rd. For a given hourly consumption of fuel, the area of the firegrate will vary as the square of the heating surface in maintaining the same efficiency. For example, if twice the heating surface be employed, the grate may be extended to four times. Conversely if half the heating surface be removed, the grate must be reduced to one-fourth of its area. It is apparent from these relations, as Mr. Clarke has observed, that a superfluous size of grate is detrimental to the power of the boiler, unless at a sacrifice of fuel. On the contrary, an extension of heating surface adds a still greater proportion to the power of the boiler, whilst the efficiency of the fuel is maintained. The general formula embodying these relations is $F = C\frac{H^2}{G}$, in which F is the quantity of fuel consumed per hour, H the area of heating surface, G the grate area, and C a constant varying for each kind of boiler.

Grates for burning wood require to be constructed on different principles from those for the consumption of coal. In this case, from the rapid ignition of the material, the furnace must be constructed capaciously, whilst at the same time the area for the admission of air must be reduced. In Russia, where nearly the whole of the coal used in manufacture is imported from this country, it is usual to have the boilers constructed on the same principle as has already been described. It, however, sometimes happens, as in the case of the late war, that the supply of coal ceases, and the owners of mills are in this emergency under the necessity of burning wood, which even in Russia at the present time is more expensive than imported coal. When driven to its use, all that is done is to remove the coal grate and furnace bars, and substitute an iron gridiron, laid on the bottom of the internal flues, which increases the capacity of the furnace and decreases the grate area. The boiler is then as efficient with wood as it was before with coal. In other cases the wood is supplied by a hopper, in which it descends as it burns away at the bottom.

Evaporative Power of Boilers.—Good coal liberates in combustion sufficient heat to evaporate from 14 to 15½ lbs. of water, and good coke to evaporate about 13 lbs. of water per pound of the fuel. Wood evaporates only 6 to 7½ lbs. per pound of fuel.

The actual evaporation in engine boilers falls far short of this theoretical result, owing to the heat carried off by the chimney, imperfect combustion, radiation, &c.

In 1858 a report was published by Mr. Armstrong, Mr. Longridge, and Mr. Richardson, detailing the results of extensive experiments on the evaporative power of steam coals. These experiments were made with a multitubular boiler, with two furnaces and 135 tubes, $5\frac{1}{2}$ feet long and 3 inches in diameter. With this boiler they first determined a standard of evaporative power when the boiler was worked on the ordinary system, every care being taken to obtain the maximum of work out of the boiler, by keeping the fires clear and by frequent stoking. No air was admitted except through the firegrates. As the economic effect of the fuel increases when the ratio of the firegrate surface to the absorbing surface is diminished, they adopted two sizes of firegrates, and obtained in consequence two standards of reference. With the larger firegrate the amount of work done by the boiler per hour was greatest, but this was accomplished at a relative loss of economic value of the fuel, as compared with the smaller grate. The one gave the standard of maximum evaporative power of the boiler,—the other the standard of economic effect of the fuel. The grate areas were $28\frac{1}{2}$ and $19\frac{1}{4}$ square feet respectively. The heating surface of the boiler was 749 square feet. The results obtained are given in the following table:—

	Firegrate $28\frac{1}{2}$ sq. ft.		Firegrate $19\frac{1}{4}$ sq. ft.	
	A.	B.	A.	B.
Economic value, or pounds of water evaporated from 212° by 1 lb. of coal	9·41	11·15	10·06	12·58
Rate of combustion, or pounds of coal burned per hour per square foot of grate . . .	21·15	19·00	21·00	17·25
Rate of evaporation per square foot of firegrate per hour, in cubic feet of water, from 60° .	2·62	2·93	2·909	2·995
Total evaporation per hour in cubic feet of water from 60°	74·80	79·12	56·01	57·78

The columns marked A give the general results, much smoke being often evolved; those marked B, the mean of the best results obtained in the experiments when making no smoke. The coal employed in these experiments, viz., the Hartley's, is very superior to that ordinarily employed in factory boilers.

By an apparatus constructed by Mr. Wright, of Westminster, the same experimenters determined the absolute heating effect of this coal, and of some similar coal from Wales, to be as follows: —

Welsh coal	14·30 lbs. from 212°
Hartley coal	14·63 " "

To give some idea of the practical economic effect of coal in stationary engine boilers, we may transcribe here some results obtained with care by Mr. John Graham, of Manchester, not as completely applying to ordinary practice, but as affording useful guidance when taken in conjunction with the preceding results on a better description of coal. The water was measured by a meter.

	Pounds of water evaporated from 212° by 1 lb. of coal.
Boiler with two internal furnaces, known as "breeches boiler" .	6·88
Waggon boiler	10·26
Cylindrical boiler with external furnace	7·54
Butterley boiler	9·72

Mr. Longridge has found 6 to $7\frac{1}{2}$ lbs. of water evaporated from 62° Fahrenheit, and converted into steam at 20 to 55 lbs. pressure, per pound of coal, by two flued boilers.

Mr. Rankine's formula for the efficiency of ordinary stationary boilers without a feed-water heater and with chimney draught is,

$$\frac{E'}{E} = \frac{\frac{11}{12} S}{S + \frac{1}{2} F} \dots (1);$$

where E' is the available evaporative power of one pound of fuel in a boiler furnace, E the theoretical evaporative power, S area of heating surface, and F the number of pounds of fuel burnt per hour per square foot of grate. But the reader must be referred to his own work on the steam engine for a discussion of the constants to be employed under different circumstances.

Strength of Boilers. — To be of maximum strength, both the external shell and the internal flues should be as far as possible cylindrical. Where this is impossible and flat surfaces are necessary, careful staying by gussets or longitudinal stays is essential to safety. The cylindrical portions of boilers can be very easily proportioned to the steam they have to bear by the formulæ which will be given below. Certain restrictions are placed upon the proportions of boilers by the nature of the riveted joints. That these may be steam and water tight under pressure, and at the same time not unnecessarily weakened by rivets, it has been found best to use plates of about $\frac{5}{16}$ or $\frac{3}{8}$ inch thick, and plates of other dimensions are very seldom employed in the construction of boilers. Thick plates are inefficiently riveted, thin ones inefficiently caulked, and this restricts the available thickness for the plates within nearly the limits which have been stated. It is necessary, therefore, in proportioning boilers, having given the working pressure, to choose the diameter which is suitable for such a thickness of plates, lessening the diameter for high-pressure boilers and increasing it for low-pressure ones. Length does not affect the strength in vessels subject to internal pressure, and hence the diameter is the variable quantity over which we have most control in proportioning the external shell. But the flues, as will be shown, decrease in strength with their length, and this dimension is in that case more easily modified than the diameter.

The general equation expressing the resistance of thin hollow vessels to internal strain is, for spherical vessels,

$$\text{P} = \frac{4\,c\,t}{d}(\text{nearly}) \ldots (2);$$

and for cylindrical vessels, bursting longitudinally,

$$\text{P} = \frac{2\,c\,t}{d} \ldots (3).$$

This equation gives the bursting pressure in lbs. per square inch, when the thickness of the plates c in inches, the tenacity of the joints t in lbs. per square inch, and the diameter d in inches, are given.

Thus, for a boiler 7 feet or 84 inches diameter, $\frac{3}{8}$ inch or ·375 inch thick, and with joints having a tenacity of 34,000 lbs. per square inch, the bursting pressure

$$= \text{P} = \frac{2 \times 34000 \times {\cdot}375}{84} = 303\tfrac{1}{2} \text{ lbs.}$$

The value of t for various materials is given in the following table: —

Without joints:	
Wrought-iron plates	50,000
Steel plates	100,000 to 130,000
Copper, sheet	30,000
Glass	4200 to 6000
With joints:	
Wrought-iron plates, double-riveted	35,700
Wrought-iron plates, single-riveted	28,600
Wrought-iron boiler plates, with single joints, crossed	34,000

In the case of well-constructed wrought-iron stationary boilers, I have been accustomed to take $t = 34000$, and in this case the bursting pressure of cylindrical vessels is, as was taken above,

$$\text{P} = \frac{68000\,c}{d} \ldots (4).$$

But with boilers the factor of safety is ordinarily taken at six, or the working pressure is not allowed to exceed $\frac{1}{6}$th of the bursting pressure, and in this case the maximum strain on the iron per square inch of section is 5666 lbs. Putting p for the safe working pressure,

$$p = \frac{11333\,c}{d} \ldots (5);$$

or in the case of the seven-foot boiler taken above,

$$p = \frac{11333 \times \cdot 375}{84} = 50\cdot 6 \text{ lbs.},$$

equivalent to one-sixth of $303\frac{1}{2}$, the bursting pressure.

For half-inch plates we get from formula (5),

$$p = \frac{5666\,c}{d}, \text{ and } d = \frac{5666}{p};$$

for three-eighths inch plate we have similarly,

$$p = \frac{4250}{d}, \text{ and } d = \frac{4250}{p};$$

and lastly for five-sixteenths inch plate,

$$p = \frac{3541}{d}, \text{ and } d = \frac{3541}{p}.$$

That is, in words, to find the safe working pressure of a boiler, divide 5666 for $\frac{1}{2}$-inch plates, 4250 for $\frac{3}{8}$-inch plates, and 3541 for $\frac{5}{16}$-inch plates, by the diameter in inches. Similarly, to find the safe diameter for a given working pressure, divide the same numbers by that working pressure in lbs. per square inch.

For flues subjected to an external pressure, I have deduced experimentally a formula the data of which are given in "Useful Information for Engineers," Second Series. Putting P for the collapsing pressure and p for the safe working pressure as before, C thickness of plates in inches, D diameter in inches, L length in feet,

$$\text{P} = 806300 \frac{\text{C}^{2 \cdot 19}}{\text{L D}} \ldots (6);$$

$$p = \frac{\text{P}}{6} = 134400 \frac{\text{C}^{2 \cdot 19}}{\text{L D}} \ldots (7).$$

For practical purposes we may substitute for the power 2·19 the square of the thickness. But it is better to employ a table of logarithms, when we get

$$\text{P} = 1 \cdot 5265 + 2 \cdot 19 \text{ log. } (100 \text{ C}) - \text{log. L D}.$$

Thus, for example, to find the collapsing pressure of a flue 10 feet long, 36 inches in diameter, and composed of $\frac{1}{2}$-inch plates, we have approximately,

$$\text{P} = 806300 \times \frac{(\frac{1}{2})^2}{36 \times 10} = 560 \text{ lbs.};$$

or, more accurately, by logarithms,

$$\text{log. P} = 1 \cdot 5265 + 2 \cdot 19 \text{ log. } 50 - \text{log. } 360 = \text{log. } 502 \text{ lbs.}$$

The safe working pressure of this flue would be $\frac{502}{6} = 74$ lbs.

This formula shows that with vessels subjected to external pressure the strength varies inversely as the length. That is, a flue 20 feet long will be only one-half the strength of one 10 feet long. This remarkable law enables us to proportion the strength of boiler flues with great ease. By introducing rigid angle or T iron ribs riveted round the exterior of the flues, we virtually decrease the length and increase the strength in the same proportion. Two or three such rings on the flues of boilers, constructed of plates equal in thickness to those of the shell, will usually render the resistance to collapse equal to that of bursting.

Accessories of Boilers. The Feed Pump. — Boilers require replenishing with water in proportion to the waste by evaporation. For this purpose, in the early boilers, working at very low pressures, an open stand pipe was employed with a valve at top, for the admission of the water from a reservoir regulated by a float in the boiler. With the increase of pressure at which steam engines are worked, this stand pipe has been abandoned and replaced by the feed pump, either attached to the engine or worked by a donkey engine attached to the boiler. The capacity of this pump must be such as to discharge into the boiler two to three times the quantity of water required by the engine in the shape of steam. The ample tables we have already given of the density of steam will enable this to be calculated with perfect ease. We have only to find the volume of steam required by the engine at each stroke (depending on the rate of expansion at which it works), and the pressure of the steam being known we have to seek its weight in the table of density, page 215, and provision must be made for the discharge of two or three times this quantity into the boiler at each stroke of the engine.

Back Pressure Valves.—To prevent accident in case of stoppage or fracture of the feed apparatus, there should always be placed on the feed pipe between the boiler and the regulating valve a self-acting valve to prevent the return of the water. Supposing the feed pipe accidentally broken, the water in the boiler would be forced back by the pressure of the steam, and expose the boiler to injury by overheating. In such a case the back pressure valve is of great service in preventing the escape of the water when acted on by the pressure of the steam.

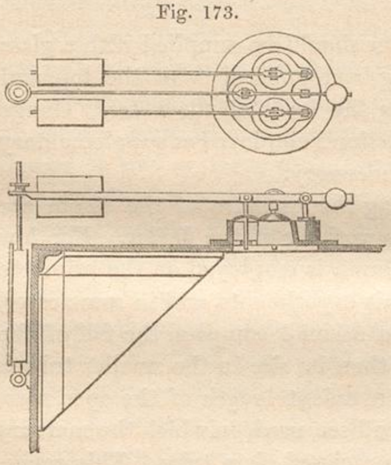
Feed-water Heating Apparatus.—When the products of combustion escape into the chimney at an elevated temperature, the heat may be utilised by the employment of water-tubes through which the feed water is introduced on its way to the boiler. Of the arrangements adopted for this purpose the best is that of passing the feed water through a wrought-iron pipe or supplementary boiler, placed in the main flues immediately behind the boiler, where the water is heated to the boiling point after leaving the pump. A more complete apparatus is that of Mr. Green, of Wakefield, known as the "Fuel Economiser." It consists of a series of upright tubes through which the feed water passes on its way to the boiler, and is heated above the boiling point, and steam in part generated. The formation of soot on the pipes was the source of the ill success of previous attempts in this direction. This difficulty Mr. Green has overcome by an apparatus of scrapers or cleaners, consisting of rings encircling the pipes, and maintained in constant but slow motion by chains and pulleys driven by a belt from the engine. With this apparatus it is found that when the waste gases escape at a temperature of 400° to 500°, the feed water can be heated to an average of 225°, the temperature of the gases after leaving the pipes being reduced to 250°. To produce this effect 10 square feet of heating surface are provided for each horse power.

Water Gauges.—Every boiler should be supplied with a glass tube, fixed in suitable stuffing boxes, and open at the top and bottom to the boiler to show the level of the water. Gauge cocks at various levels are sometimes employed as supplementary to the glass gauge: both are necessary.

Steam Gauges for indicating the pressure of the steam are also indispensable to the safe working of the boiler. For low pressures an open mercury column is employed on the principle of that used by Regnault in his experiments, and in some cases, to bring the indications within a small compass, the fall of the mercury in the cistern rather than its rise in the smaller tube is observed. To avoid the inconvenient length of the open mercury column, the air gauge has been used, in which the mercury in its rise condenses the air in a closed glass tube. This gauge, accurate and sensitive, has yet the fault that the indications decrease in length as the pressure increases, and there is also some difficulty in preserving the quantity of air in the gauge constant. Recently Mr. Allan has overcome these difficulties by the use of a conical air chamber so arranged that the indications of the gauge shall be uniform at all pressures, and the air can be renewed at any instant. In M. Bourdon's gauge a curved metallic tube, communicating at one end with the steam boiler, and at the other closed, is used. The curvature of this tube decreases with the increase of pressure in its interior, and the closed end being free to move is connected with an arrow moving over a graduated arc and marking the pressure. In Schaeffer and Budenberg's gauge the pressure acts on a flat corrugated plate of steel which expands and raises a rack acting on a toothed wheel, and carrying a similar arrow to indicate the amount of pressure. In Smith's gauge a flat spiral spring is used, against which the pressure acts through the medium of a plate of india-rubber. All these gauges should be fixed on the boiler with a siphon in which water from the boiler may condense. In this way the pressure in the boiler is transmitted through the column of water, and at the same time the gauge is unaffected by the temperature of the steam.

Safety Valves.—I usually place three safety valves on boilers, as shown in fig. 173; two of these have fixed weights on their levers, and the other is pressed down by a spring balance, and serves to regulate the working pressure of the boiler. The two larger valves for a fifty horse boiler have each an area of 12 square inches. The third is of only 5 square inches area. These valves are fixed to a common valve seating. The bearing surfaces of the valves are made either flat, conical, or spherical. Flat valves have a tendency to blow off at too low a pressure, from the steam getting between the bearing surfaces. These valves should always be open to the atmosphere that they may be seen.

Fig. 173.



Man Holes are required to obtain access to the boiler for purposes of examination and cleaning. In double-flued boilers one must be placed beneath the flues as well as above them. *Mud Cocks* are placed near the bottom of boilers for the discharge of water and sediment.

Fusible Plugs are portions of metal fusing at a temperature not greatly exceeding the maximum working temperature of the steam, and fixed in that portion of the boiler most liable to be overheated from deficiency of water. These plugs are of pure lead or of an alloy of bismuth, lead and tin, according to the temperature they are required to melt at, and they are thought to prevent danger by relieving the pressure of the boiler, and putting out the fire before the plates are injured by overheating. These plugs, however, tend to lose their fusibility, and to become coated with a protecting coat of oxide or sediment, which prevents the communication of heat. They are not a very reliable provision.

Plans for the prevention of Smoke.—Amongst the earliest of these we may class those which depend on mechanical means for the supply of the fuel.

Of this class is the earliest patent for smoke prevention taken out by James Watt in 1785. By this plan the fire is supplied from above downwards by a reservoir of fuel in contact with the burning mass, the combustion of which is supported by a strong lateral current of air passing direct through the fire to a flue on the other side aided by a slight downward current beside or through the fuel, which last descends by its own gravity as it is consumed. For the purpose of intercepting and completing the combustion a clear fire is maintained at the entrance into the flues, so that the products of the first fire, being subjected to the intense heat of the second and mingled with atmospheric air, may be effectually consumed.

Apart from the external reservoir, we owe to Watt the dead plate very generally adopted in stationary boilers. The fresh fuel is thrown upon the dead plate, where it gradually cokes the more volatile constituents distilling over and being consumed by the bright fire beyond. Then the coked fuel is pushed back on to the bars and a new supply introduced in front. This plan, where proper provision is made for the supply of the necessary quantity of air, obviates the production of smoke as effectually as many more complicated contrivances.

The succeeding patentees of the principle of mechanical feeding as a substitute for hand labour, have followed two different plans. Some have made the grate itself to carry forward the fuel, either by revolving horizontally or by rolling forward longitudinally, the grate-bars being connected together to form an endless chain, or by the oscillation of the alternate bars causing the thrusting forward of the fuel by what has been called a peristaltic movement. Others have made the grate stationary and have used fans revolving horizontally to distribute the fuel over the grate-bars. In all these cases the coal is supplied slowly and uniformly from a hopper. There is no doubt that the uniform distribution of the fuel over the whole surface of the grate-bars, so far as it is secured by these systems, must be to a large extent advantageous in the diminution of smoke and economy of fuel. At one time they were extensively used, but the complication and expense of the apparatus has led to their general abandonment and the return to hand-feeding.

Other plans for the prevention of smoke depend on a double furnace with alternate firing.

Double furnaces patented by Mr. Losh were in use as early as 1815, and in various modifications have been employed ever since. The principle of double furnaces within the same boiler was first introduced by myself; and the plan adopted has already been described as the double-flued boiler. The two flues enable the stoker to fire alternately, and so maintain a more uniform generation of steam than with a single flue, and the flame passing from one flue mingling with the gases from the other, assists in their combustion. I believe that this simple system of alternate firing, when conjoined with the requisites of the economical generation of steam, viz. plenty of capacity in the boiler, sufficient admission of air, and, what is quite as necessary, careful and attentive stoking, will effect the prevention of smoke without any costly apparatus, so far as that is possible with any given description of fuel. There is this further advantage in double furnaces, that the air required for combustion is necessarily variable. Now a double furnace tends to equalise the supply. The two furnaces fed alternately will not require a maximum or a minimum quantity at the same time, and as the two currents of gaseous products mingle, the surplus air of the one furnace will supply the deficiencies of the other. In this way the tendency is to compensate the supply and demand, and prevent waste from too large or too small a quantity in either furnace.

Others in seeking the prevention of smoke have introduced an additional supply of air over the fire.

Mr. C. Wye Williams was one of the earliest, as he has been the most pertinacious and consistent, advocate of the introduction of a large additional volume of air into the furnace, and we have to thank him for the labour he has expended in proving the necessity for air as one of the prime conditions of economy of fuel and success in the prevention of smoke. Mr. Wye Williams contends for a uniform admission of cold air to the furnace, relying upon frequent thin feeding to equalise the needs of the furnace. The peculiar principle of his plan is the mechanical division of the air by causing it to enter the furnace through what he terms a diffusion plate, or partition perforated with numerous small apertures. This is usually placed behind the bridge where the gases needing combustion pass into the flues. There is no doubt this is a convenient method for the introduction of air, and has in many instances effectually prevented the formation of smoke.

Mr. Syme Prideaux contends for a variable admission of air, greatest when the fuel is first thrown on, and decreasing to the ordinary supply through the grate-bars as the fire burns clear. For this purpose he constructs his furnace doors with metal Venetians, which open by a self-acting apparatus when the fuel is supplied. They then gradually close at a regulated speed, altogether independent of the care of the fireman. The air entering through the door is, by an arrangement of plates, warmed as it enters the furnace, and carries back the heat radiating from the door.

All these systems are more or less effective, but I am inclined to think that a judicious engineer with a careful stoker or fireman, will effect all the objects to be attained with the means placed at his disposal, in a well constructed boiler of sufficient capacity, and with a simple furnace such as has been described in the foregoing chapter, as completely as can be done by any one of the numerous nostrums held forth as the only antidotes for smoke, and promising great economy of fuel.

CHAP. IX.

ON WINDMILLS.

Atmospheric disturbances causing wind have from a high antiquity been employed as a motive power, and probably the earliest application of this force was the propulsion of ships by sails. Amongst the most primitive races, long before their intercourse with civilisation, this power was applied in the navigation of small vessels; and the ancient Phœnicians, Greeks, and Romans were all of them well acquainted with this mode of employing the force of the wind for purposes of human industry. It is to be regretted that we have no records of the time when it was applied as a motive power in mills; this event is lost in the oblivion of the past, and it was not till early in the thirteenth century that we find the Dutch and French employed in the construction of windmills adapted to the wants of an energetic and industrious population. These times were marked by a growing intelligence that encouraged and fostered inventive talent, and the Dutch millwrights and engineers were long celebrated for their skill and knowledge in every art that had for its object the improvement of the industrial resources of the people.

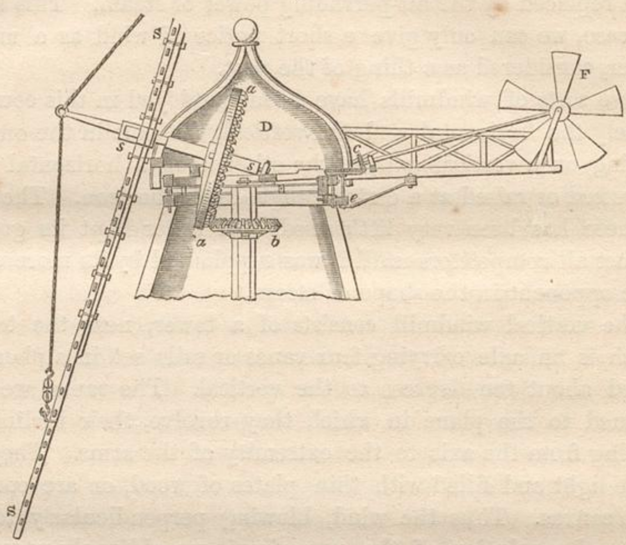
It was from Holland that our knowledge of windmills and wind as a motive power was first imported, and it is within my own recollection that the whole of the eastern coasts of England and Scotland were studded with windmills; and that for a considerable distance into the interior of the country. Half a century ago, nearly the whole of the grinding, stamping, sawing, and draining was done by wind in the flat countries, and no one could enter any of the towns in Northumberland, Lincolnshire, Yorkshire, or Norfolk, but must have remarked the numerous windmills spreading their sails to catch the breeze. Such was the state of our mechanism sixty years since, and nearly the whole of our machinery depended on wind, or on water where the necessary fall could be secured. Now both sources of power are also abandoned in this country, having been replaced by the all-pervading power of steam. This being the case, we can only give a short notice of wind as a motive power, considered as a thing of the past.

Two sorts of windmills have been employed in this country, namely the horizontal and the vertical, the sails in the one revolving on a vertical axis, in the other upon a horizontal axis, depressed or raised at a certain angle to the horizon. The first of these has been very little used; the latter kept its ground against all competitors until it was supplanted by its more energetic opponent in the shape of steam.

The vertical windmill consists of a tower, near the top of which is an axle carrying four vanes or sails set in a plane inclined about ten degrees to the vertical. The vanes are also inclined to the plane in which they revolve, their inclination varying from the axis to the extremity of the arms. They are made light and filled with thin plates of wood, or are covered with canvas. Thus the wind, blowing perpendicularly to the plane of revolution of the arms, impinges obliquely upon the broad sails, and a rotatory movement is generated, which, transmitted by bevel gearing, works the millstones, scampers, and machinery of every sort contained in the mill.

The mill-sails require to be placed perpendicularly to the direction of the wind, and for this purpose, in the older mills, the whole upper part of the tower containing the machinery is turned round by manual labour. In more modern constructions, however, a dome or cap carrying the sails is fixed on the summit of the tower, and is turned by a self-acting fly with four or more oblique vanes, similar to a smoke jack, which, acted upon by the changing currents of wind, gives motion to the cap of the tower, carrying round with it the wind axis and sails, keeping them perpendicular to the direction of the wind. Such a mill is shown in fig. 174, where D is the cap moving on rollers, *s* the shaft carrying the sails S S, and the bevel wheel *a a*, gearing into another bevel wheel *b*, on the millstone shaft. The wind acting on the fan F, communicates motion to the bevel wheel and spur pinion *e*, which, acting on a spur wheel or rack fixed on the summit of the tower, causes the revolution of the cap. The sails of the fan are constructed so that when they lie in

Fig. 174.



the plane of the wind, they are not affected; but as the wind shifts, it strikes them obliquely, and causes the revolution of the cap till they are again in the plane of the wind.

Of experiments upon windmills by far the most important are those of Smeaton, communicated to the Royal Society in 1759. The inclination of the sail to the plane of revolution he found should vary in the following ratio, where the radius is supposed to be divided into six equal parts, and the angle of the sail given at each point:—

	Angle with the plane of motion.	
0	—	centre.
1	18°	
2	19	
3	18	middle.
4	16	
5	$12\frac{1}{2}$	
6	7	extremity.

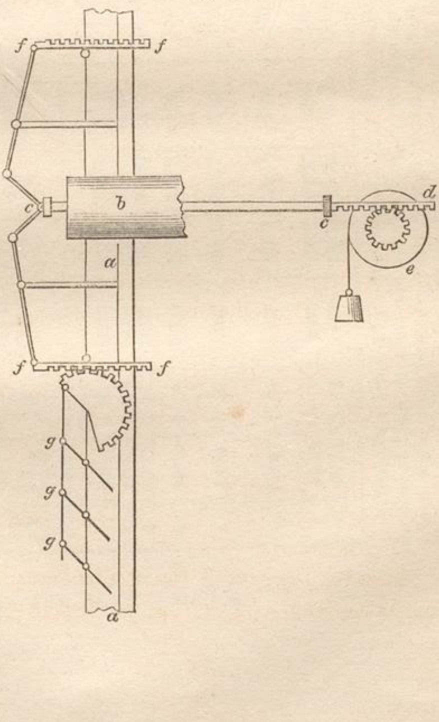
This inclination of the sail to the plane of revolution is known as its *weather*. Sails before Smeaton's time were simple parallelograms; he found, however, that advantage was gained by adding a triangular sail to the leading edge of the radius or whip *a a*, so placed that the sail was broadest at its periphery. The extreme breadth of the sail, *c b*, was then made equal to one third of the radius or whip, and of this $\frac{5}{8}$, or $\frac{5}{24}$ of the radius, was the breadth of the ordinary sail, *a b*, and the remaining $\frac{3}{8}$, or $\frac{3}{24}$ of the radius, was the breadth of the triangular leading sail, *a c*, as shown in fig. 175. The ordinary length of the whip of the sail is 30 feet.

Fig. 175.

Regulation of the Speed of Windmills.—This is best effected in the case of windmills with cloth sails, by a plan of Mr. Bywater, in which a series of racks and pinions cause the cloth to roll or unroll according to the strength of the wind.

Another plan, suggested by Mr. (now Sir William) Cubitt at the beginning of this century, is shown at fig. 176, applied to sails which have movable boards or thin plates instead of sail-cloth. *a a* is the whip; *b*, the axis on which the sails are carried, and which is hollow to receive the rod *c c*. At the extremity of this is a rack, *c d*, gearing in a pinion, *e*, which is connected with a pulley over which is hung a weight so as to press the rod *c c* outwards with a constant force. *g*, *g*, *g*, are the boards which form the surface of the sail, and which are connected together so as to open or shut like the bars of a Venetian blind. On the last board of each sail is a toothed segment, in

Fig. 176.



which works a rack, $f\,f$, connected by levers with the rod $c\ c$, as shown. By this arrangement the force of the wind, as it varies, opens or shuts the boards of the sail, so as to keep the total pressure on the sails equivalent to the force exerted by the balance-weight hung over the pulley e.