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DYNAMOMETERS

AND

THE MEASUREMENT OF POWER:

A TREATISE ON THE CONSTRUCTION AND APPLICATION OF DYNAMOMETERS;

WITH A DESCRIPTION OF THE METHODS AND APPARATUS EMPLOYED IN MEASURING WATER- AND ELECTRIC-POWER.

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PREFACE.

THE aim in the following pages has been to pre in convenient form, for the use of Technical Stud and Engineers, a description of the construction principles of action of the various types of Dynam eter employed in the measurement of power.

A chronological presentation of the subject has been attempted, as many of the forms once used h entirely disappeared; with very few exceptions t various types discussed are those now in use.

In the measurement of the mechanical horse-pow of a hydraulic motor the effective power may be certained by means of a friction-brake, or other dyn mometer, under any given conditions; but as the may be such that the maximum power of the wheel not developed, it remains for the Engineer to determin those conditions best suited to the wheel under cor sideration. In Chapter V is given a discussion of th methods and apparatus in use for ascertaining the effi ciency of a given wheel, including the determinatio of flow in rivers and streams.

The work here presented has been used as the basi of a course of lectures to Engineering students, and i the outgrowth of a series of articles published in the *American Machinist.*

In its preparation free use has been made of numerous publications relating to the subject, and references for further information are given in foot-notes throughout the text.

Special mention is due Professor Andrew Jamieson, of Glasgow, for use of matter and figures from his "Steam and Steam-engines" (London: Chas. Griffin & Co.). The author is also under obligations to Professors Jas. E. Denton, J. Burkitt Webb, Dr. Mansfield Merriman, and others.

J. J. FLATHER.

LAFAYETTE, IND., Aug. 1, 1892.

NOTE TO THE SECOND EDITION.

THE present edition is not merely a revision of the previous work. Five new chapters have been added and some of the old ones have been rewritten; its scope has thus been greatly enlarged, and it is believed that in its present form the book will find an increased field of usefulness among shopmen as well as engineers generally.

In Chapter IV several new forms of transmission-dynamometers are discussed, and Chapter V, on the Measurement of Water-power, contains some new material on the Venturi meter and meter testing, together with illustrated descriptions of the mechanism and principles of action of various water-meters.

Of the new chapters which have been added, one is devoted to the Power required to drive Machinery, and four are on the Measurement of Electrical Power; these include a general consideration of the subject, together with a discussion of electrical measuring instruments and methods in use for determining the power and efficiency of electric motors.

The chapter on the Power required to drive Machinery consists of tabulated results of observed horse-power required to operate machines of various kinds and sizes working under different conditions; these embrace a wide range, including all ordinary machine-shop tools as well as the most common forms of wood-working machinery.

The plan of giving references for further information has been extended, and the investigator will find much additional material in the foot-notes.

In the preparation of the work on electrical measurements, the author has received many helpful suggestions from Mr. Frank W. Springer, of the University of Minnesota, to whom, and to all others who have assisted in any way, he desires to acknowledge his indebtedness.

J. J. FLATHER.

MINNEAPOLIS, MINN., December 31, 1899.

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DYNAMOMETERS

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THE MEASUREMENT OF POWER.

CHAPTER I.

DETERMINATION OF DRIVING POWER.

In designing a modern machine-shop or manufactory, and in estimating the cost of power for its working plant, an accurate knowledge of the amount of power absorbed by the different machines is not only desirable, but essential to economy and efficiency.

If the power required is not known, the engine or motor provided may prove incapable of driving the work; or, on the other hand, the motive power may be largely in excess of that required: in either case there is an unnecessary expense—in the first case, in remedying the evil, and, in the second, in the daily expenditure of fuel for excess of power.

So also in fitting up a factory: if a more accurate knowledge of the power required to drive machine-tools were known, there would be a greater economy in running them. The writer has in mind a case that came under his notice a few years ago, in which a certain wood-planer had its countershaft changed three times—different diameters of pulley, and different widths of belt, and finally a heavier countershaft being used—before it would work satisfactorily. Machines are largely belted by guesswork. If the pulleys guessed at are *nearly* large enough to do the work, the workman stretches his belt to its utmost, and manages to run the machine by taking light cuts; if, however, the belt has a velocity and width barely sufficient to run the machine, and an ordinary cut will throw off the belt, then, if split pulleys are not in use, a length of shafting is taken down and a larger pulley put in the place of the one which has shown itself to be insufficient to drive. Sometimes both a greater width of belt and a larger pulley have to be resorted to.

Another case was where a 6-inch belt running over a 36-inch pulley at 120 revolutions per minute had been put in, and had been running for more than a year driving a roomful of high-velocity machines used for covering magnet-wire. Evenness of motion is specially desirable in this class of machinery; and yet, when all the machines were on, the shaft would vary from 80 revolutions per minute to its normal speed.

Rosin or belt-oil was in order every few days, and when the slip became too great the engine had to be shut down and the belt relaced. The relacing was done by the use of clamps, and the belt finally became so taut that the increased friction on the bearings near the driving-pulley kept the boxes and shaft constantly hot. This belt was a continual source of annoyance and expense; but, because it had been deemed large enough to furnish power for the forty or more machines in the room, no change had ever been made, and it had run for over a year in this same manner.

A 9-inch belt was put on, and no trouble was afterwards experienced, though it has now been running for several years. It is easy enough to remedy a defect like this; but prevention would have been better, and would have considerably reduced the expense account. The knowledge of the power required to drive the machinery was wanting. The question arises: How can this power be estimated?

In the discussion of an inquiry as to the power required to drive machine-tools, Mr. G. H. Babcock stated at a meeting of the American Society of Mechanical Engineers* that for a general rule in ordinary machine-work we may take roughly one horsepower as sufficient to drive machine-tools necessary to keep ten men at work; but this, he adds, does not necessarily include shafting, engine, etc., nor blowers for foundry work.

Expressed algebraically, this rule of thumb would be

$$\frac{N}{10} = H.P.,$$

where N equals the number of men employed; or, if we let $10 = C$, a constant, we have

$$\frac{N}{H.P.} = C.$$

* Trans. A. S. M. E., vol. VIII.

The above as it stands is of little or no value; in the first place, C is too large, as will be shown by the following data,* obtained from a large number of representative machine-shops; and, in the second place, the power required to drive the machinery varies between such wide limits, even in the same class of work, that separate values of C cannot be determined which may be relied upon as giving even a rough approximation.

Such a formula is of value only when C has been determined for two or more similar plants, and applied to another equipment working under similar conditions—and this indeed is rarely met with.

By a reference to the above table it will be noticed that two firms on exactly the same line of work—that of manufacturing machine-screws—require a total horse-power such that the number of men employed per horse-power is in the one case 2, and in the other 0.62; or as 3 to 1. The small value of C in both cases is evidently due to the nature of the machinery, which is largely automatic, one man being able to feed several machines. A comparison of the values of C (obtained for total power used) of two well-known firms, the Pratt & Whitney Manufacturing Co. and the Brown & Sharpe Manufacturing Co., shows that the latter employ only 3.91 men per horse-power, while Pratt & Whitney employ 6.04; and yet these shops may be considered to belong to the same class.

Another comparison of two firms running about the same class of machinery is that of the Baldwin Locomotive Works and William Sellers & Co. The Baldwin Locomotive Works give a value of

* Compiled by author in 1889.

TABLE I.

HORSE-POWER; FRICTION; MEN EMPLOYED.

Name of Firm.	Nature of Work.	Horse-power. Total.	Horse-power. Req. to Drive Shafting.	Horse-power. Req. to Drive Machinery.	Horse-power. Per Ct. to Drive Shafting.	Number of Men.	C for Total H. P.	C_1 for Effective H. P.
Lane & Bodley	Engines and wood-working machinery	58				132	2.27	
J. A. Fay & Co	Wood-working machinery	100	15	85	.15	300	3.00	3.53
Union Iron Works	Engines, mining machinery	400	95	305	.23	1600	4.00	5.24
Frontier Iron and Brass Works	Marine engines, etc	25	8	17	.32	150	6.00	8.82
Taylor Manufacturing Co	Automatic engines	95				230	2.42	
Baldwin Locomotive Works	Locomotives	2500	2000	500	.80	4100	1.64	8.20
W. Sellers & Co. (one department)	Heavy machinery	102.45	40.89	61.56	.40	300	2.93	4.87
Pond Machine-Tool Co.	Machine-tools	180	75	105	.41	432	2.40	4.11
Pratt & Whitney Co	" "	120				725	6.04	
Brown & Sharpe Co	" "	230				900	3.91	
Yale & Towne Co	Cranes and locks	135.05	66.81	68.24	.49	700	5.11	10.25
Ferracute Machine Co	Presses and dies	35	11	24	.31	90	2.57	3.75
T. B. Wood's Sons	Pulleys and shafting	12				30	2.50	
Bridgeport Forge Co	Heavy forgings	150	75	75	.50	130	0.86	1.73
Singer Manufacturing Co	Sewing-machines	1300				3500	2.69	
Howe Manufacturing Co	" "	350				1500	4.28	
Worcester Machine Screw Co	Machine-screws	40				80	2.00	
Hartford Machine Screw Co.	" "	400	100	300	.25	250	0.62	0.83
Nicholson File Co	Files, etc	350				400	1.14	
	Averages	346.4			38.6	818.3	2.96	5.13

1.64 for C per total horse-power, and Sellers & Co. give 2.93. If we deduct the power required to run the shafting in each works, the values become, respectively,

$$C_1 = 8.20,$$
$$C_1 = 4.87.$$

A closer result obtains between the Pond Machine-Tool Works and William Sellers & Co. throughout all the data given. The percentage of power required to drive shafting is in one case 41 per cent., and in the other 40 per cent. The values of C and C_1 are as follows:

	C.	C_1.
Pond Machine-Tool Works	2.4	4.11
William Sellers & Co..........	2.93	4.87
Average	2.66	4.49

These values are sufficiently close to enable one to deduce an approximate value for C and C_1 which would apply to either case, but when used in connection with other and similar shops the results could not be depended upon, even roughly.

Thus the average value of C, as shown above, is 2.66 per total horse-power. Applied to Wm. Sellers & Co. this gives

$$\frac{N}{H.P.} = C; \quad \therefore N = C \times H.P.,$$

$$= 2.66 \times 102.45,$$
$$= 273 \text{ men employed.}$$

Applied to Pond Tool Works:

$$N = 2.66 \times 180,$$
$$= 478 \text{ men employed.}$$

In the former case N should equal 300, and in the latter 432. These results are only rough estimates, but are correct to within 10 per cent of the actual number employed. If we apply the formula to the Pratt & Whitney Co., manufacturers of machine-tools, we obtain:

$$N = 2.66 \times 120,$$
$$= 319 \text{ men;}$$

whereas we find from the table that 725 are employed.

Looked at from its most favorable standpoint by comparing those values for similar grades which most nearly agree, it will be seen that the formula is really of no practical value, and much less can it have any weight when looked at from an engineering standpoint.

A look at some of the results obtained from the data given may prove to be interesting.

It has been stated that ten is too large a value for the number of men employed per horse-power when applied to machine-tools, even neglecting the power required to drive the shafting, etc.

The average value of C_1 obtained from table is 5.13, or say 5.0; while the minimum value is 0.833, the maximum being 10.25. It would be out of the question to apply the formula with the average value of C_1, viz., 5.0, to either of these cases; nor is it practicable to apply any other value of C_1 either to determine the horse-power from the number of men employed, or, *vice versa*, to obtain the number of men employed from the horse-power furnished.

It will be noticed in the sixth column, headed "Per cent of power required to drive shafting," that very wide differences occur. The maximum is that used by the Baldwin Locomotive Works, viz., 80 per cent—an extremely large factor;—while the minimum given by J. A. Fay & Co. is only 15 per cent; the average, 38.6 per cent, corresponds to that quoted by William Sellers & Co. within less than 1½ per cent.

Mr. J. T. Henthorn, in a paper read before the American Society of Mechanical Engineers, states that the friction of the shafting and engine in a print-mill should not exceed 19 per cent of the full power. Out of fifty-five examples of a miscellaneous character which he has tabulated, seven cases are below 20 per cent, twenty vary from 20 per cent to 25 per cent, fifteen from 25 per cent to 30 per cent, eleven from 30 per cent to 35 per cent, and two above 35 per cent, while the average of the total number is 25.9 per cent.

Mr. Barrus, speaking of this subject, quotes eight cases, the data of which were obtained from tests made by himself in various New England cotton-mills, in which the minimum percentage was 18, and the maximum 25.7, the total average being 22 per cent.

Mr. Samuel Webber states that 16 per cent of the total power of a mill is sufficient to overcome the friction of shafting and engine—10 per cent for the shafting alone. But in this estimate Mr. Webber does not include the loss due to the belts running upon loose pulleys, which he does not consider to be part of the shafting, as they are not so running while the machinery is in operation: and when it is not, they may be thrown off as well as not, except for convenience. He further estimates, both from his own experience and the observations of others, that the power consumed by the machine-belts on the loose pulleys in a large cotton-mill is about 8 per cent of the whole. This 8 per cent added to the 16 per cent loss due to shafting and engine will give 24 per cent of the total power—a result which agrees closely with the average values given above.

The writer believes that for shops using heavy machinery the percentage of power required to drive the shafting will average from 40 to 50 per cent of the total power expended. This presupposes that under the head of shafting are included elevators, fans, and blowers.

In shops using lighter machinery and with foundry connected the power percentage will be about the same as above; but, if the foundry work is done outside, the power required to drive the shafting will not average so high, the range being about 10 per cent less, or from 30 to 40 per cent of the total.

In machine-shops with a line of main shafting running down the centre of a room, connected by short belts with innumerable countershafts on either side, often by more than one belt, and, as frequently happens, also connected to one or more auxiliary shafts which drive other countershafts, we can see why the power required to drive this shafting in machine-shops should be greater than that found in cotton and print mills, the machinery of which is in general driven from the main lines of shafting. Nor can we neglect the loss due to belts upon loose pulleys, as with the numerous clutches and countershafts in use the conditions more nearly approach those which exist when the machinery is in operation. There is no doubt, however, that a large percentage of the power now spent in overcoming the friction of shafting in ordinary practice could be made available for useful work if wider and looser belts were employed, or, what would have the same effect, if the belts were slackened and their speed increased; and also if more attention were paid to lubrication. The use of roller bearings for line shafting obtains to a limited extent only, but the success attained with this form of bearing indicates that it will be used to an increasing extent when its advantages are more widely known.

As the power required to drive the machinery in a modern plant cannot be even approximately ascertained from its relation to the number of men employed, the question still remains open: How can this power be measured?

One method frequently used is that by which the power required is ascertained from the velocity and width of driving-belt. Different rules have been given in our text-books and engineering journals in order to estimate the driving power of a belt from its width and velocity. A rule which the writer has used in his practice when the difference in diameters of pulleys is not very great is: Every inch in width of a single cemented belt, having a velocity of 800 feet per minute, will transmit one horse-power up to a velocity of about 5000 feet per minute; beyond 5000 the centrifugal force of the belt largely diminishes its effective tension. Expressed as a formula we have

$$\frac{bV}{800} = H.P.,$$

in which b equals breadth of belt in inches, and V equals velocity of belt in feet per minute. To illustrate this, let us look at an example. Suppose the main shaft of a factory runs at 125 revolutions per minute, and a 12-inch pulley on this shaft drives a 10-inch pulley on the counter of a 16-inch lathe through a 3-inch belt; the lathe is driven by a $2\frac{1}{2}$-inch belt running from a 10-inch step to an 8-inch. What power does the lathe absorb when the belt is taxed to its limit? The speed of the belt is 392 feet per minute; if we disregard slip, which is about two per cent of the total velocity, this would give

$$\frac{2.5 \times 392}{800} = 1.22\ H.P.$$

Now, as shown, the belt will transmit according to our formula 1.22 H. P., and by calculating H. P. for the different machines in the factory a measure of the driving power may be obtained, to which a certain per cent should be added for power required to drive the shafting.

This process might give an approximation somewhat nearer the truth than the method previously discussed, but as the formula is based on a certain permissible stress in the belt-fibres, which stress is well within the limit of safety, we do not know how much *more* power the belt is exerting, nor do we know that it *is* exerting as much as the formula calls for. Although we can calculate what the width of a belt *ought* to be to transmit a given horse-power, x, at a given velocity, the stress in the belt may be greater or less than that on which our formula is based, and the resulting horse-power transmitted may be $x \pm y$.

In order to measure the amount of driving power from the velocity and width of belting, the tension on the tight and slack sides of a belt, the arc of contact α between belt and pulley, and the coefficient of friction ϕ are all necessary.

The width of a belt of thickness t must be such that its cross-section multiplied by its permissible working stress f is capable of resisting the maximum tension T_1 in the driving side of the belt, or $T_1 = btf$.

We have the general equation

$$H.P. = \frac{PV}{33\,000}.$$

Now if we let $P = \frac{T_1}{m}$, where m is a function of the arc of contact α, and coefficient of friction ϕ, we obtain

$$H.P. = \frac{btfV}{33\,000m}.$$

If $bt =$ one square inch, the horse-power transmitted per 1000 feet per minute is expressed by $H_0 = \frac{f}{33m}$.

This Reuleaux calls the *specific duty* of a belt, the value of which he gives for leather belting, as varying from 5.3 to 9.8; hence

$$\frac{f}{33m} = 5.3 \text{ to } 9.8.$$

As f varies in different belts, and m varies with α and ϕ, it is seen that any general formula, whether rational or empirical, is not trustworthy when the total amount of power absorbed is desired—however satisfactory such a formula may be when used to calculate the width of belt to transmit safely a given horse-power. The only reliable method of determining this transmission of power is by the use of some form of dynamometer.

Where power is rented from one firm to another, the necessity of obtaining correct estimates of the amount consumed is apparent. A case in point is that of the Lowell Hosiery Co., which rented an estimated total of 13½ horse-power, for which $125 per horse-power per annum was paid.

A dynamometer test being made, it was ascertained that 28½ H. P. was being used—more than double the amount paid for. Another case is that of the Northampton Tape Co., whose lease called for 30 H. P.; a dynamometer being applied to the shaft it was found that 11 H. P. was the maximum transmitted. Still another case is that of a company in Worcester, which hired rooms and power, the basis of rent being estimated at 13 H. P. Forty horse-power was actually used, as shown by a dynamometer test, and the rent was increased accordingly.

Many such instances could be cited to show that very wide differences exist between the amount of estimated power and the amount actually developed as determined by an accurate dynamometer. Such wild estimates are at first sight difficult to account for, since there are several good rules in use for ascertaining the width of belt to transmit a given horse-power; however, as already shown, as these rules do not take into account the individual differences in belt-tension, there will result, with variations of velocity and tension, corresponding variations of power transmitted.

Mr. Henry R. Towne's experiments with leather belting show that the ultimate strength of a laced belt $\frac{7}{32}''$ thick is about 200 pounds per inch of width; assuming a factor of safety of 3, this gives 66 pounds as the allowable strain per inch of width in single belting (Morin assumes 55 pounds). For a cemented or riveted joint the permissible strain may be taken one-third greater, or about 90 pounds per inch of width.

The following table, compiled by Mr. Nagle, gives a list of belts in use, and the actual horse-power transmitted by them, compared with which are calculated widths by the formulæ of Webber and Nagle. The widths in the ninth and tenth columns have been calculated by the writer and added to Mr. Nagle's table; the handy rule referred to in the last column being the one previously mentioned, namely,

$$b = \frac{800\,H.P.}{V}.$$

TABLE II.

WIDTH AND VELOCITY OF BELTING.

Horse-power.	Velocity in feet per minute.	Diameter of small pulley in inches.	Belt-pull per inch, width in lbs.	Thickness.	Width of Belt.				
					In use.	Webber's Rule.	Nagle's Rule.	$\frac{1000\ H.P.}{1.06\ V}$.	Handy Rule.
375	5,600	60	98	Double	24	22	34	31½	27
250	3,080	84	58	4-ply	48	50	28	28	23
220	2,451	42	135	Single	22	98	31	84	70
175	3,179	72	93	Double	19½	15½	25	26	22
175	3,629	115½	55	"	29	15	22	23	20
130	2,117	70	113	"	18	18	22	29	24½
125	3,490	84	82	"	14½	8	17	17	14½
90	2,860	60	87	"	12	10	15	15	12½
77	2,268	60	77	"	14½	12	12	16	13½
45	2,000	48	37	Single	20	21	15	21	18
49	2,111	72	24	"	14	21	18	21	18½
43	1,800	60	44	"	18	20	14	23	19
41	1,809	60	42	"	17½	12	16	21	18
40	2,000	72	37	"	8	14	13	19	16
18	850	22	116	Double	6	19	8	10	8½
8	942	30	40	Single	7	12	8	8	7

In the formula used by Mr. Nagle, the coefficient of friction deduced from the experiments of Mr. Towne is assumed at 42 per cent, but as this coefficient was deduced from the results of experiments having a large slip it is probable that the value is somewhat higher than average practice would warrant. The experiments made by Mr. Wilfred Lewis * with belting running at an average velocity of 800 feet per minute, give coefficients varying from 25 per cent to 100 per cent. Rankine assumes 15 per cent as the coefficient of friction, but the results of all other investigators show this

* Trans. A. S. M. E., vol. VII. p. 549.

value to be too low. Morin gives 28 per cent as an average value for dry belts on smooth cast-iron pulleys, and 12 per cent for very greasy shop belts on cast-iron pulleys—the mean of these being 20 per cent. Recent investigations at the Massachusetts Institute of Technology show that this mean value is a little low, but probably nearer the truth than either Towne's or Rankine's coefficient. According to these later experiments * the value 27 per cent was chosen as being the best under the average conditions to which an ordinary belt is subjected in practice—allowing $2\frac{1}{2}$ per cent for slip—and this value has been used in calculating the widths given in column 9, the formula used being

$$b = \frac{1000\ H.P.}{1.06\ V};$$

Reuleaux' formula, in which $\phi = 0.28$, instead of 0.27 as here used, is

$$b = \frac{1100\ H.P.}{V},$$

V being in feet per minute. The average arc of contact, α, on the smaller pulley being equal to $.95\pi$, or a little less than 180°.

Upon consideration it will be seen that the rule

$$b = \frac{800\ H.P.}{V},$$

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* Trans. A. S. M. E., vol. VII.

commonly used in the machine-shops, differs somewhat from

$$b = \frac{1000\ H.P.}{1.06\ V},$$

which takes into account the arc of contact and the coefficient of friction, the average values $\alpha = .95\pi$ and $\phi = .27$ being used.

Reduced to an equivalent form this equation becomes

$$b = \frac{943\ H.P.}{V},$$

which will give a width of belt a little greater than that obtained by using the shop rule referred to.

As the arc of contact on smaller pulley decreases, the width of belt will have to increase; thus for an arc of contact of 120° the width of belt should be 25 per cent greater than that found from the above rule.

As these formulas are based on a given thickness of belt, t, if we increase this thickness the power transmitted ought to increase in proportion, and for double belts we should have half the width required for a single belt under the same conditions. With large pulleys and moderate velocities of belt it is probable that this holds good, and this value has been used in those cases in the table where double belts are employed.

With small pulleys, however, when a double belt is used there is not such perfect contact between the pulley-face and the belt—due to the rigidity of the latter—and more work is necessary to bend the belt-fibres than when a thinner and more pliable belt is used. Moreover, the centrifugal force tending to throw the belt from the pulley increases in a greater degree than the effective tension in the belt, and for these reasons the width of a double belt required to transmit a given horse-power when used with small pulleys is generally assumed not less than seven-tenths the width of a single belt to transmit the same power.

An inspection of the fourth column in Table II shows that the actual stress or belt-pull for single belting varies from 24 to 135 pounds per inch of width. Considering these varying tensions and comparing the calculated width with those found in actual practice, we arrive at the same conclusion previously reached, viz., that the driving power of a belt is not directly determinable by the use of a formula unless the belt-pull or stress is known for each particular case.

CHAPTER II.

FRICTION-BRAKES.

WE have already stated that the only satisfactory method of ascertaining the amount of power is by the use of some form of dynamometer—by which we mean an instrument or machine for measuring the power exerted by a prime mover, or the amount of power consumed in driving a machine or machinery plant.

Although the engine-indicator is an instrument for measuring power, and is thus a dynamometer, as it neither transmits nor absorbs the power, its discussion will not be entered into in these pages. The use of the engine-indicator in connection with a new form of transmission-dynamometer designed by the writer will be shown farther on.

Among the many machines and devices for measuring power one of the simplest is the Prony friction-brake; and but for certain disadvantages attendant on its use it would possess a superiority to all other contrivances.

Primarily this consists of a lever *L*, Fig. 1, connected to a revolving shaft or pulley in such a manner that the friction induced between the surfaces in contact will tend to rotate the arm in the direction in which the shaft revolves. This rotation is balanced by weights *P*, hung in the scale-pan at the end of the lever. A counterpoise attached to the brake-arm is often used in order to balance it before adding weights in the scale-pan. If not balanced, the weight of the lever-arm must be ascertained and its moment added to the total moment of the weight in order to obtain an accurate measure of the friction. In order to measure the power for a given number of revolutions of pulley, we add weights to the scale-pan and screw up on bolts *b b*, until the friction induced balances the

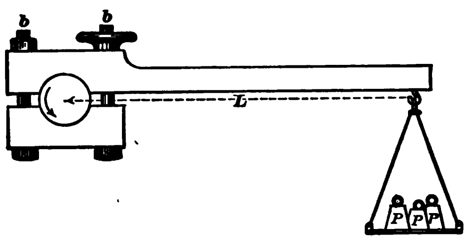

FIG. 1.

weights and the lever is maintained in its horizontal position, while the revolutions of shaft per minute remain constant. That this measure of the friction is equivalent to a measure of the work of the shaft will be seen when we consider that the entire driving power of the shaft is expended in producing this friction at the required number of revolutions per minute —and this driving power is equal to the mechanical effect of the shaft when running at the same speed in the performance of useful work.

With the ordinary form of lever-brake, in order to maintain a stable equilibrium of the lever the weight should be supported on a knife-edge and act below the centre of the shaft. In this case, when the weight falls or rises, through any irregularity of the brake, the lever-arm is decreased or increased, and the slight irregularity is overcome by a corresponding change of moment; whereas, if the weight act above the axis, any increase or decrease in weight will cause it to act through a longer or shorter arm, as the case may be, and the lever cannot of itself come back to its horizontal position. This does not apply to that form of

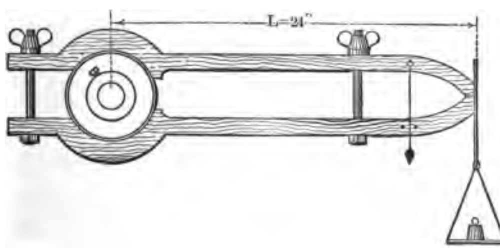

FIG. 2.

brake where the force is measured on a platform scale, as it is evident the lever-arm is practically constant. Although the construction of the lever is of great importance, Mr. Heinrichs has shown that the proportions of the brake for different horse-powers are even more important in order to obtain uniformity of test.

From a number of experiments made with a Prony brake of the design shown in Fig. 2, Mr. Heinrichs gives the following dimensions as being most suitable for the horse-powers designated: *

* Mechanics, 1884.

TABLE III.

DIMENSIONS FOR PRONY BRAKE.

To measure	Number of Revolutions per Minute of Brake-pulley.	Size of Brake.		Diameter of Brake-pulley.
		Length.	Width.	
		in.	in.	in.
From 2 to 5 horse-power.	1200 to 1800	24	$1\frac{1}{2}$	4
	700 to 1200	24	$2\frac{1}{2}$	6
From 5 to 8 horse-power.	1200 to 1600	24	$2\frac{1}{2}$	6
	800 to 1200	24	4	6

A regulator or dash-pot attached to the end of the lever-arm or scale-beam may be used with the Prony brake—and other various forms of dynamometer in which the pressure is weighed—in order to maintain a more even balance and to prevent vibrations and sudden shocks due to momentary slip of the belt or inefficient lubrication of the brake.

This dash-pot is generally in the form of a cylinder from 4 to 6 inches in diameter, partly filled with oil or water in which a piston about $\frac{1}{16}$ inch less in diameter is submerged. This piston will allow the oil to pass freely around it as it rises or falls with a slow motion, but will oppose a resistance to any sudden movement. An adjustable piston by which the motion of the oil can be regulated as desired is sometimes an advantage. This can be readily made by turning two disks to fit the bore of the cylinder and drilling several holes through both disks by clamping together.

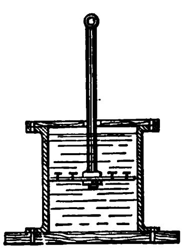

FIG. 3.

By connecting these disks to a stem with a shoulder and nut, any desired area of opening between the disks can be obtained by turning one upon the other and tightening the nut. The piston should be attached to the scale-beam by an eye and pin so as to move freely, and the beam should be balanced and adjusted with the piston in place in the liquid before beginning to weigh.

A dash-pot, Fig. 3, used in the Lowell hydraulic tests, was made with a thin disk of iron turned to fit loosely in its cylinder; six $\frac{3}{8}$-inch holes were drilled and tapped in it and fitted with brass thumb-screws, any or all of which could be removed if desired to allow a freer passage of the water contained in the cylinder; the screw being left on the plate in order to maintain the original balance.

Instead of hanging weights in a scale-pan, as in Fig. 1, the friction may be weighed on a platform-scale; in this case the direction of rotation being the same, the lever-arm will be on the opposite side of the shaft.

A modification of this brake, in which the lever acts on a platform-scale, is that in use in the Sibley College Engineering Laboratory, and is shown in Fig. 4. The brake-wheel is keyed to the shaft, and its rim is provided with inner flanges about two inches deep, which form an annular trough for the retention of water to keep the pulley from heating. A small stream of water constantly discharges into the trough and revolves with the pulley—the centrifugal force of the particles of water overcoming the action of gravity; a waste-pipe *p*, with its end flattened, is so placed in the trough that it acts as a scoop, and removes all surplus water.

The brake consists of a flexible metal strap to which are fitted blocks of wood forming the rubbing surface; the ends of the strap are connected by an adjustable

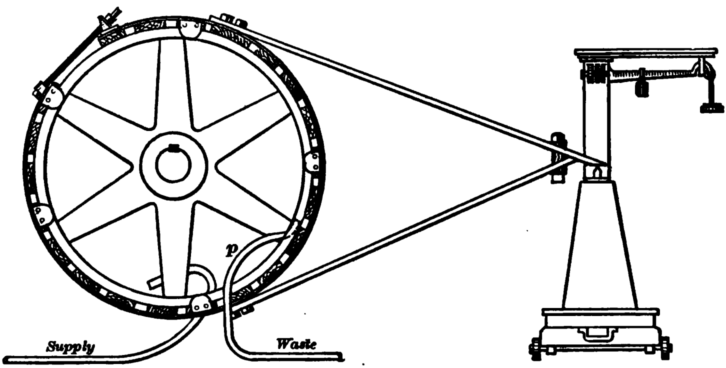

FIG. 4.

bolt-clamp, by means of which any desired tension may be obtained.

The horse-power or work of the shaft is determined from the following:

Let $W =$ work of shaft in foot-pounds per minute, equals power absorbed per minute;

$P =$ unbalanced pressure or weight in pounds, acting on lever-arm at distance L;

$L =$ length of lever-arm in feet from centre of shaft;

$V =$ velocity of a point in feet per minute at distance L, if arm were allowed to rotate;

$N =$ number of revolutions per minute.;

$H.P. =$ horse-power.

Then will $W = PV = 2\pi LNP$.

Since $H.P. = \dfrac{PV}{33\,000}$, we have

$$H.P. = \frac{2\pi LNP}{33\,000}.$$

If $L = \dfrac{33}{2\pi}$, we obtain

$$H.P. = \frac{NP}{1000}.$$

$\dfrac{33}{2\pi} = 63.024$ inches,—practically 5 feet 3 inches—a value often used in practice for the length of arm.

It will be noticed that neither the diameter of the pulley nor the pressure and weight of the friction-blocks on the same, nor the coefficient of friction enter into the formula for obtaining the horse-power. As previously noted, the friction induced between the brake-blocks and the rim of the pulley tends to rotate the brake in the direction in which the shaft revolves; this rotation is counterbalanced by the weight acting upon the arm of the brake, and when the system is in equilibrium the moments are equal; that is, if $F =$ friction between blocks and pulley acting at radius $= R_1$, and $P =$ counterbalance acting at distance L from centre of shaft, we shall have

$$FR_1 = PL.$$

Multiplying each member of the equation by $2\pi N$, where $N =$ number of revolutions of shaft per minute, we obtain

$$2\pi N \times FR_1 = 2\pi NPL = W.$$

That is, the work absorbed per minute by friction equals the work in foot-pounds per minute at the end of the lever-arm. And since we have the means of obtaining this work W when the weight P and arm L are known, it will readily be seen that the friction and radius of brake-pulley do not have to be considered in obtaining the measure of the power of a rotating shaft.

If, however, the coefficient of friction, ϕ, between the rubbing surfaces be known, we may obtain from the above equation an expression for the pressure exerted on the pulley-rim by the brake:

Let $F = \frac{PL}{R_1}$ represent the force of friction between the surfaces in contact at the pulley-rim, then $\frac{F}{\phi}$ will equal the pressure exerted upon the pulley necessary to produce the force F.

The coefficient of friction varies from .06 to .50, depending upon the different materials in contact and upon the lubrication of the surfaces. Within certain limits, the more perfect the lubrication the smaller the coefficient between any two materials.

A brake-dynamometer similar to the one shown in Fig. 4 is used by the Westinghouse Machine Co., in testing their engines before being sent out of the factory. For engines above 125 horse-power and under 250 a brake-wheel is used which is 48 inches in diameter and 24 inches face, with internal flanges about 3½ inches deep, carrying a stream of water about 2 inches deep, fed by a ½-inch pipe, the overflow being removed as shown in figure by means of the scoop-pipe.

The rubbing surface is composed of 28 hard-wood blocks, oak or hickory, which are each $3\frac{1}{2}$ inches wide, spaced $1\frac{7}{8}$ inches apart. These blocks are lubricated with fat pork or suet, which is packed in against the flat face of the wheel between the blocks. The lever-arm is $63\frac{1}{8}$ inches long.

For smaller engines a brake-wheel 48 inches in diameter by 13 inches face is used, the details being the same as in the larger wheel except the brake-arm, which in this case is shorter, being $27\frac{3}{4}$ inches long.

Even with the sizes given a brake-rim occasionally catches fire, the cooling water not being sufficient to carry off the heat quickly enough.

The following reports of tests made with these brakes were furnished to the writer through the courtesy of the Westinghouse Machine Co., both tests being on their Automatic Compound Engines:

Size of engine	16 & 27 × 16	8 & 13 × 8
Initial steam-pressure	93	96
Terminal steam-pressure	13	14
High-pressure M. E. P	45.37	49.87
Low-pressure M. E. P	19.75	22.12
Indicated horse-power	205.5	41.73
Brake horse-power	183.48	38.25
Loss or friction	22.02	3.48
Percentage of loss	10.7	8.3
Gross indicated water-rate	23.95	24.77
Gross brake water-rate	26.83	27.03
Revolutions per minute	249	378
Brake-load (pounds)	785	241
Dead weight on scales	50	11
Radius of brake (inches)	$63\frac{1}{8}$	$27\frac{3}{4}$
Duration of test (minutes)	8	15

The arm of the brake is often omitted, in which case the friction is induced either by the use of a flexible brake-strap supplied with wooden blocks, or simply by the use of a band or ropes thrown over the pulley.

For small powers ordinary leather belting from two to four inches wide is generally used, but care should be taken that the belt is not sticky: a well-worn flexible belt free to slip on the pulley-face will give the most uniform results.

The belt should be narrower than the pulley-face,

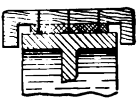

FIG. 5.

and, in order to provide against its slipping off the rim sideways, it should be tacked to three or four light strips of wood placed across the face of the pulley: these strips being cut out to receive the pulley-rim and leaving a projection of about $\frac{3}{4}$ inch on each side of the rim, as shown in cross-section in Fig. 5.

In Fig. 6 is seen the general arrangement of this method, the belt being carried over the pulley on the motor to be tested and one end secured to the floor by any convenient means. The other end is provided with a scale-pan or flat wooden box to carry the weights. A wire or stout cord attached to the bottom of the box and secured to the floor will prevent the accidental pulling of the box over the shaft while making the test. This wire must necessarily remain slack when the weights are in the box.

With this form of brake the power is measured as with the lever-brake; that is, the work, W, of shaft in foot-pounds per minute equals the product of the weight, P, in the scale-box multiplied by the velocity, V, in feet per minute of the lever-arm of the weight (see page 24), which in this case is equal to the radius of the pulley plus half the thickness of the belt. If we neglect the belt-thickness, the velocity V will equal the circumferential velocity of the pulley, hence

$$W = PV = 2\pi RNP,$$

or the horse-power

$$= \frac{2\pi RNP}{33\,000} = 0.0001\,904\,RNP,$$

where R is radius of arm in feet, and $N =$ number of

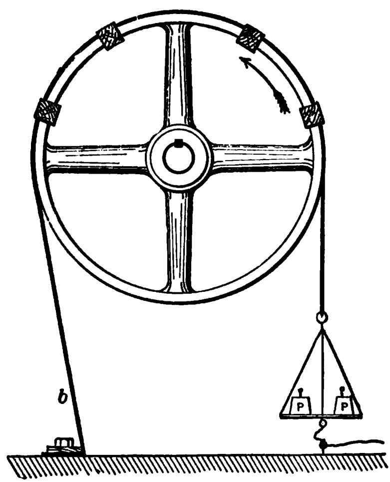

FIG. 6.

revolutions per minute. If we take radius of arm, r, in inches we shall obtain

$$H.P. = \frac{2\pi rNP}{33\,000 \times 12} = 0.00\,001\,586\,rNP.$$

In working with this belt-brake, in order to obtain accurate results the weights should be so adjusted that there shall be no tension in the end of the belt which is secured to the floor. A common error is to overload the scale-box and create a pull on the end *b* which will cause an indication of power in excess of its true value. A spring-scale or balance interposed between the end *b* and the floor, as shown in Fig. 7, will give

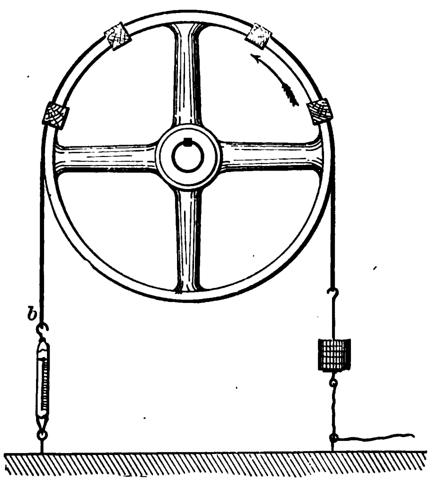

FIG. 7.

the amount of the pull, if any exists, which pull should be deducted from the weight in the scale-box. It is evident that the weight of the spring-scale should be added to the pull which it indicates in order to obtain the total tension in the end *b*. Another method is to scrape the belt, thus causing a greater adhesion to the pulley-face; this will pull the belt around in the direction of the arrow, tending to lift the weight in the scale-box, thus producing a slackness at the end secured to the floor. With care in the weighting, if sufficiently small weights are provided there need be little or no

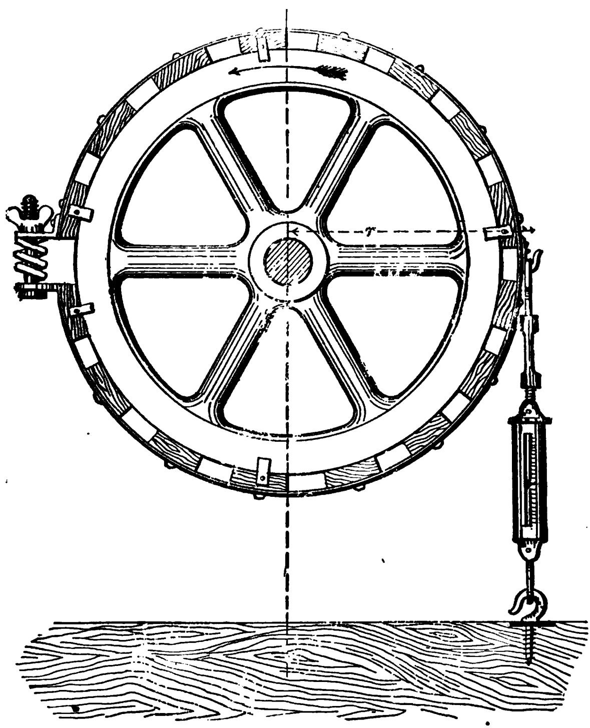

Fig. 8.

tension at *b*. A 3-inch belt over a 24-inch pulley running at 200 revolutions per minute, with a weight of 50 pounds in the scale-box, will measure about 2 horse-power.

For larger powers the brake-strap is lined with wooden blocks and encircles the pulley, the friction being measured either by attaching weights to a hook or scale-pan and screwing up on an adjusting bolt which brings the two ends of the strap together; or a spring balance is used in connection with the adjusting screw, as shown in Fig. 8.

In the Brauer compensating brake, the band which encircles the pulley is of thin rolled iron when the pulley-rim is flat; wire ropes are used for a grooved pulley.

For small forces Mr. Gisbert Kapp has advantageously employed the arrangement represented in Fig. 9. The brake-cord, which embraces half the pulley

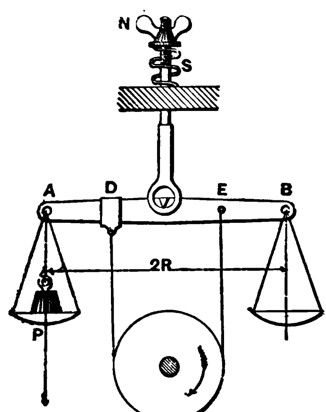

FIG. 9.

circumference, is attached at E on a level with the knife-edge of the scale-beam, and at D in a point somewhat below, so that the lever-arm of D is increased while that of E is diminished, thus forming a compensating device. The spring S and nut N allow an adjustment of the tension in the cord after the scale-pan is weighted.

The brake recommended by the Royal Agricultural Society,* designed by Mr. C. E. Amos and Mr. Appold,

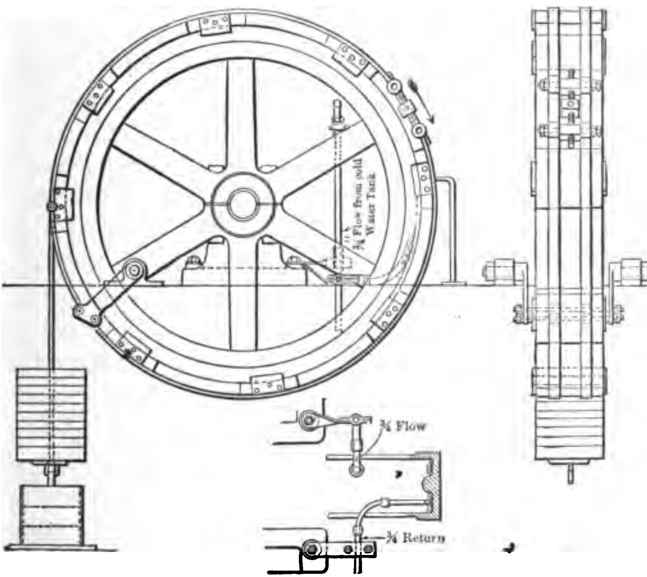

FIG. 10.

is somewhat similar to those already described, but, as will be noticed, Fig. 10, this brake is provided with a self-acting system of levers which are arranged to adjust the tension in order to compensate for the variations in the moment of friction.

* Proc. British Inst. C. E., vol. xcv, 1888–89.

In this brake the strap is made in two parts to which blocks of wood are secured, and at a convenient point the two portions are joined by a right-and-left-hand adjusting screw.

The other ends of the strap are jointed to a double swinging lever in such a manner that the radii of the two ends of the strap from the centre of oscillation of the lever are unequal.

If, through deficiency of lubrication or other causes, the wheel should tend to carry the strap around with it in the direction of the arrow, the greater radius of the end nearer the weight would effect a loosening of the strap and a diminution of the friction; whereas if the friction is momentarily insufficient to sustain the weight, it will in falling tighten the strap, and thus maintain automatically a fairly constant moment.

This form of brake, like that of Appold, can only be used for measuring small horse-powers, unless we take into account the reaction at the point of suspension of the lever.

So long as the friction between the wooden blocks and wheel is such that the weight of the brake-strap and suspended weight is sufficient, at the required speed, to carry the load without tightening the adjusting screw to any extent, the lever does not affect the results—the conditions being similar to those which would obtain if the brake were without compensating lever, and the strap so slack that the bottom-blocks barely touch the wheel. That the resultant of the tensions in the brake-band resolved along the lever affects the measure of the power can be shown by means of the following figure (11).

Let the lever ECD be in the position shown, and the system in equilibrium. The tensions of the brake-blocks on the lever towards the right at C, and left at D, are represented in the figure by T_1 and T_2. On the other hand, the reactions of the lever on the brake-blocks are T_1 towards the left at C, and T_2 towards the right at D; then, since there is equilibrium in the sys-

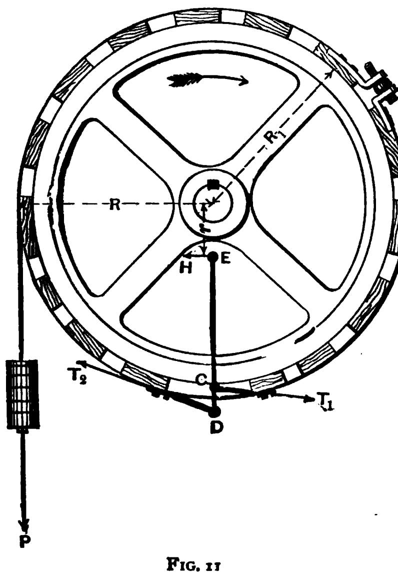

FIG. 11

tem, the algebraic sum of the moments taken about the centre of shaft must equal zero.

The resultant of the forces T_1 and T_2, which we may call Q, must pass through the point of suspension, E, of the lever. Resolving this force Q into its vertical and horizontal components acting at the point E, which is directly under the centre of the shaft or centre of moments, we have the moment of the vertical component equal to zero. Calling the horizontal component h, and the vertical component v, we have the sum of the moments about the centre of rotation:

$$PR - FR_1 - hr - v0 = 0,$$

or

$$PR = FR_1 + hr;$$

in which P is the weight acting on brake at radius R; F is the friction between brake-blocks and rim of pulley acting at radius R_1; and h is that component of the reaction at the point of support of the lever which tends to produce a rotation about the centre of shaft; its lever-arm $= r$.

Since $FR_1 = PR - hr$, we have

$$2\pi NFR_1 = 2\pi N(PR - hr);$$

that is, the work absorbed by friction equals the work of the shaft in foot-pounds per minute (when R_1, R, and r are in feet, and $N =$ revolutions per minute), or, as previously found,

$$W = 2\pi N(PR - hr),$$

and

$$\text{horse-power} = \frac{2\pi N}{33\,000}(PR - hr).$$

The amount of the force h is best obtained by the use of a spring-balance. With a high coefficient of friction the force h may be small, and might be disregarded in approximate measurements, but in every case where accuracy is desired its moment must be considered.

Ropes used as brake-straps have given very satisfactory results.

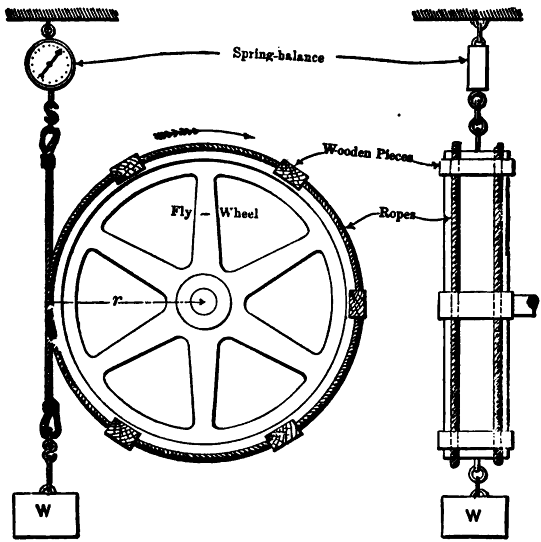

Fig. 12.

Prof. Andrew Jamieson, of the Glasgow College of Science and Arts, states that he prefers a rope brake to any one of the numerous forms which he has tried, and believes that it could be adopted for large powers and for long continuous runs, for the following reasons:

"It could be constructed on very short notice from materials always at hand in every factory, and at very little expense. It is so self-adjusting that no accurate fitting is required. It can be put on and taken off in a moment; is very light and of small bulk. It needs little or no attention for lubrication. The back-pull registered by the spring-balance is steady, and might be made a minimum by properly adjusting the weight. For larger powers only more, or larger, or flatter ropes, or a larger brake-wheel, would be required."

Fig. 12 represents a rope-brake used by Prof. Jamieson to indicate a gas-engine of fifteen brake horse-power. In this test the diameter of ropes was 0.6 inch, working over a 5-foot fly-wheel.

The following are some of the conditions under which the test was made:*

Mean revolutions of brake-wheel per minute.....	205
Weight, *P*, in lbs..............................	157
Mean back-pull on balance, in lbs...............	4
Mean brake H. P. during two hours' run.........	15.23
Gas-consumption per brake H. P. in cu. ft. per hr.	24.3
" " " indic. " " "	18.9

More recently Prof. Jamieson has used the forms of rope-brake shown in Fig. 13. These are of the same kind employed in the trials of gas-engines under the auspices of the Society of Arts, London, and give much more satisfactory results than any other form of brake hitherto devised for light work. The substitution of the spring-balance in the right-hand figure for the weight shown at the left of the figure is a decided ad-

* See paper by W. W. Beaumont in Proc. British Inst. C. E., 1888–9; also Jamieson's Steam and Steam-engines (London, 1890).

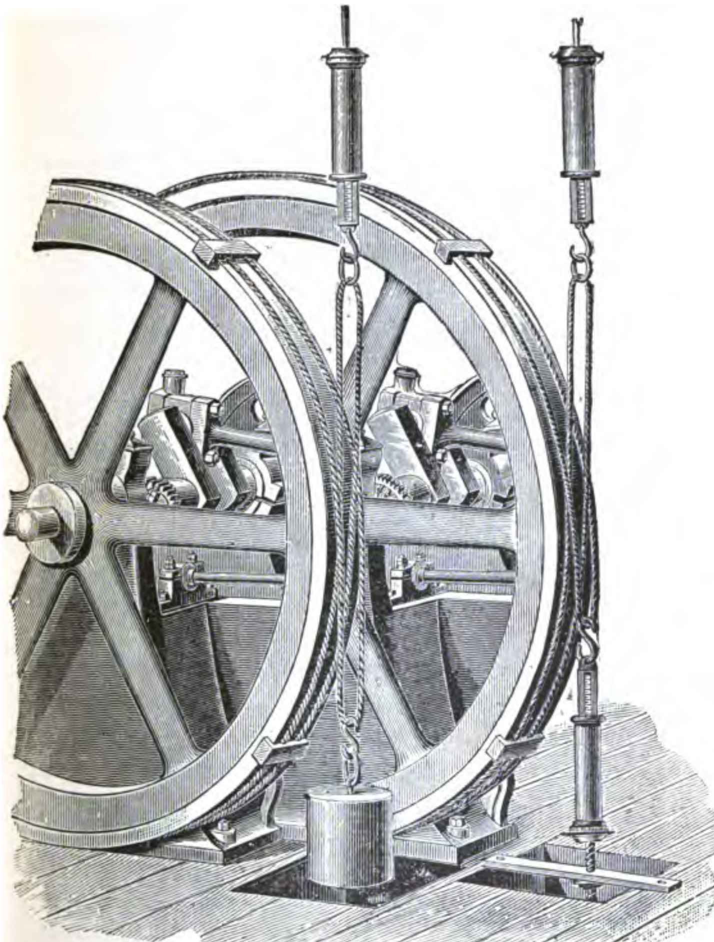

FIG. 13.—TWO FORMS OF ROPE-BRAKE USED BY PROF. JAMIESON.

vantage, since by the use of two spring-balances of different periods of oscillation the "hunting" action of the brake is effectually minimized, enabling observations to be taken with great precision. To obtain the brake-load it is only necessary to add the weight of the hanging part of the lower balance to its own reading, and subtract from this sum the back-pull as registered by the reading of the upper scale.

This form of brake deserves to be better known; for with it no lubrication whatever is required, and continuous runs of any desired length of time may be carried out without any fear of overheating or requiring to stop for adjustment.

With this brake Prof. Jamieson conducted a five-hour continuous test of Brown's Rotary Engine,* and obtained for speeds varying from 560 to 600 revolutions per minnte an average brake horse-power of 20.78. As the brake-wheel used was 4 feet in diameter it will be seen that the average surface velocity was nearly 7300 feet per minute—a very unsatisfactory speed for friction-brakes.

An interesting form of rope-brake dynamometer is that shown in Fig. 13*a*.

This brake was designed to measure 125 horse-power when placed upon a 72-inch fly-wheel running at 150 revolutions per minute. The frame *E*, which stands upon the platform scales *F*, is made of such a size that the two upright pieces pass each side of the fly-wheel *A*. At the lower portion of the frame there

* Trans. Inst. Engineers and Shipbuilders in Scotland, Nov. 1891.

is a cross-piece which holds the lower ends of the two ropes.

The ropes pass around the wheel, and the upper ends are spliced so as to pass over a pair of hooks at the end of the screw *D*.

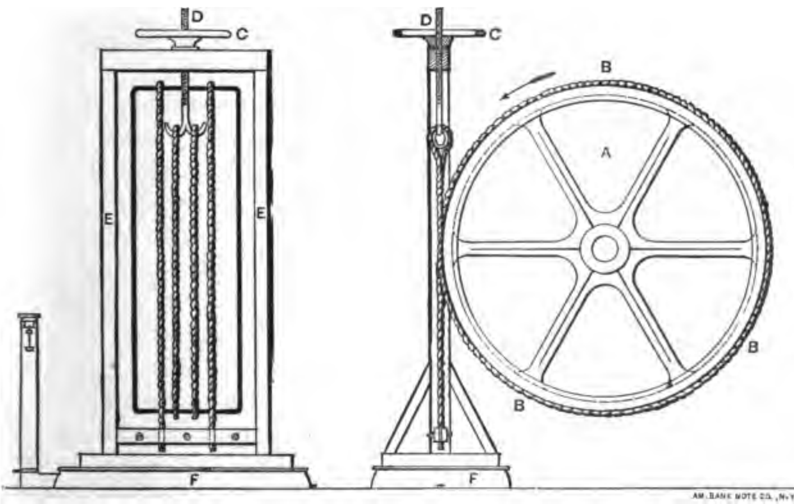

FIG. 13*a*.—ROPE-BRAKE USED TO MEASURE 125 HORSE-POWER.

The difference in tension which the friction produces in the ropes is measured directly on the platform scales, no correction being necessary for back pull.

Prof. D. S. Jacobus * states that in one case a test was made with this brake of six hours' duration without any trouble whatever.

During a subsequent test the load was run up to 145 brake horse-power and maintained at this figure. The total variation as registered by a box-counter, read

* Trans. A. S. M. E., vol. XVI. p. 820.

every two or three minutes, was only three revolutions, showing how steadily a rope-brake can be run even with heavy loads.

In the test mentioned, the inside of the fly-wheel was cast with flanges on either side, thus forming a hollow space which was filled with water.

When loaded from 120 to 150 horse-power a little water was also added on the outside of the wheel in order to prevent the charring of the ropes; this precaution is, however, usually not necessary. Brakes of this character have been used very satisfactorily for five- and six-hour runs over pulleys no larger than 42 inches in diameter, measuring up to 75 or 80 horse-power.

In using a rope-brake of any kind one half the diameter of the rope should be added to the radius of the pulley in order to obtain the effective radius of the brake. The writer's experience with rope-brakes would indicate that the diameter of rope should be as large as can be conveniently handled. Ropes one inch or more in diameter are preferable to $\frac{5}{8}$-inch or $\frac{3}{4}$-inch, as these wear out quickly with heavy stresses.

An interesting form of brake-dynamometer invented by M. Rappard, modified in order to adapt its use to large forces and high-speed machinery, is thus described in *La Lumière Electrique:* *

"One of these improvements consists in the substitution, for the rubbing surfaces, of linen bands secured to metallic straps, instead of the ordinary belts usually employed. In this way a composition belt is obtained

* August 1, 1891.

which is entirely inextensible, very strong and perfectly free to allow water to pass to cool the surfaces.

"A strip of brass .08 inch thick covered with bands of linen .04 inch thick constitutes a very desirable form of belt for this work.

"It is by the use of this new form of inextensible strap that M. Rappard has been able to construct the machines represented by Figs. 14 to 16.

The apparatus represented by Fig. 14 consists—

"1st. Of a brake shaft connected by a universal joint to the motor to be tested.

"2d. Of a drum mounted mid-length of the brake-shaft, and of two loose pulleys placed on each side of the drum, upon the hubs of which the arms of a forked balance-yoke are supported.

"3d. Of three metallic straps, two for the loose pulleys and the other for the drum: this last, which produces the friction, is covered with a band of linen; from the forked yoke to which it is attached this strap passes over the drum and descends vertically to the lower cross-bar of the frame.

"The two other straps, also attached to the forked yoke, envelop the lower surface of the loose pulleys, from which they rise vertically and are attached to the upper cross-bar of the frame.

"This vertical frame of wood (it would be better to construct it in part of wrought-iron pipe) carries at top and bottom two strong cross-bars, through which pass the bolts which receive the ends of the straps.

"These bolts are for regulating the tension of the straps so as to produce the necessary friction to balance the load of the brake.

"The whole apparatus is suspended by a chain which, after passing over a pulley rigidly supported above the

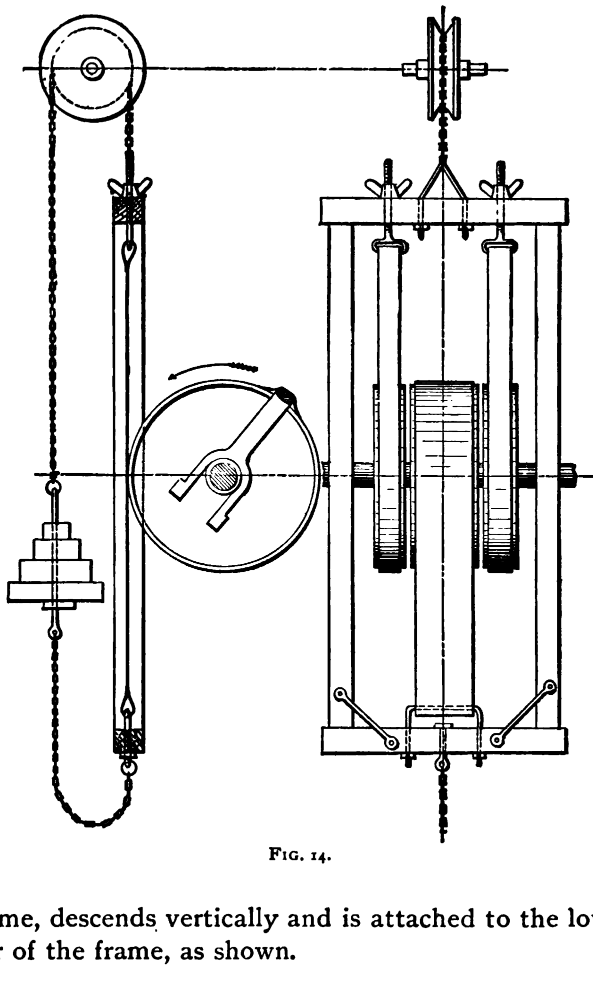

FIG. 14.

frame, descends vertically and is attached to the lower bar of the frame, as shown.

"This arrangement is used to insure an equal rolling and unrolling of the belts on the pulleys and drum, in order to maintain a constant load on the brake notwithstanding the vertical movement. The weight of the frame and the brake-load are carried upon a rod situated in the centre of the vertical portion of the chain.

"There will often be an advantage in placing the apparatus horizontally; in this case the plane of the bands is placed tangentially to the upper part of the drum, the horizontal motion being obtained by means of small friction rollers placed under the frame. At each end of the frame there is a chain which, after being stretched horizontally, passes under the pulley at an angle and descends vertically to the floor. The chains should be long enough so that they do not leave the floor whatever the motion of the frame.

"The brake-load is placed upon that one of the two chains which is connected to the cross-bar of the frame to which are attached the bands from the loose pulleys.

"Fig. 15 is another arrangement of the Rappard balance-dynamometer which permits placing the brake-shaft nearer the floor. The centre strap, covered with canvas, and which forms the rubbing surface, passes downwards and under a guide-pulley, thence upwards to the rod which receives the weights.

"The two other bands, after passing under the loose pulleys, ascend, and are carried over guide-pulleys, thence downwards, and are attached to the extremities of a short beam, the centre of which receives the eye of the rod which carries the load.

"Tension in the straps is obtained by means of two screws and nuts which allow the shaft of the guide-pulleys to be raised or lowered.

"The water necessary for cooling the straps of the

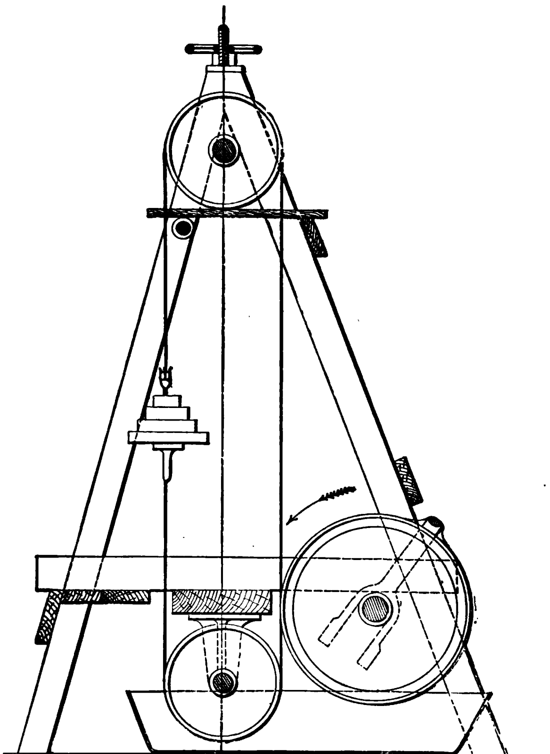

FIG. 15.

brake instead of falling upon the exterior surface, is delivered to the interior of the drum by two small pipes passing between the drum and the two loose pulleys. The water is retained in the interior of the drum by two narrow flanges, and is distributed over all the surface centrifugally; perforations across the face of the drum allow the water to lubricate the strap.

"These automatic-balance-brakes permit very accurate results, for there is only the friction of the brake-shaft bearings to be deducted from the total measure of

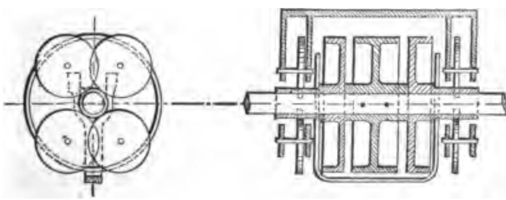

FIG. 16.

the work; however, this friction is very small, since, on an average, it does not rise above one fourth per cent of the total work in an apparatus measuring 50 horse-power.

"Still this cause of error can readily be overcome if desired, by mounting on friction rollers. In this case the brake-shaft bearings are replaced by the lengthened hubs of the loose pulleys, which are supported by four pairs of rollers as shown in Fig. 16; as will be noticed, the hubs of the loose pulleys do not revolve, and only follow the angular displacement of the forked yoke caused by the variations of friction."

If we wish to determine the horse-power of a vertical shaft—for instance that of a turbine—by means of a friction-brake, we can no longer suspend the weight directly from the bar or lever, but must insert a bent lever, so that the vertical direction of the weight may be converted into a horizontal force.

Fig. 17 represents a friction-brake for a vertical shaft which was used by Francis in his Lowell hydraulic experiments in testing a 150-horse-power turbine. The brake-wheel rim *A* is of cast-iron $5\frac{1}{2}$ feet in diameter and 24 inches width of face. This rim is 3 inches thick, and is cast with internal lugs which permit it

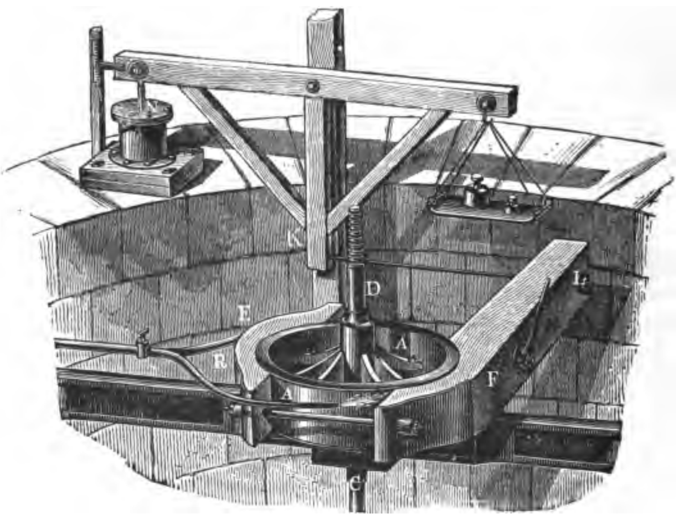

FIG. 17.

to be bolted to a spider keyed to the turbine shaft *D*, provision being made for a slight expansion between the end of the arms and the brake-rim, which is flanged to receive the brake-shoes. The brakes, *E* and *F*, are of maple, and are tightened by two 2-inch square bolts; one of the brake-arms, *F*, is connected to the swinging lever, *K*, by means of the rod *KL*, as shown. From one end of this lever the scale-pan is hung, and to the other end is connected a hydraulic regulator, *N* (see Fig. 3), which consists of an iron plate, half an inch thick, turned $\frac{1}{16}$ inch less than diameter of cylinder, free to move up and down in a cylinder filled with water, so that it acts, as previously noted, as a moderator in controlling any sudden vibrations of the lever-arm. The brake is cooled by means of a forked pipe, *R*, which throws jets of water against the wheel, the quantity of cooling water being about .17 cubic feet per minute. When running slow the lubrication was with linseed and resin oil; water, however, was preferred for the faster speeds—about 60 revolutions per minute.

Mr. Samuel Webber, in 1884, had occasion to test a large turbine at Augusta, Ga., and for this purpose had a brake made similar in appearance to the one shown in Fig. 4, page 24, but arranged horizontally with a bent lever like the one just shown. In this brake the friction-pulley was 7 feet in diameter and 24 inches face. The brake-lever was of oak, 16 inches square, reaching 15.91 feet from the centre of shaft to the point of connection with the bent-lever scale-beam, which latter had a leverage of two to one to reduce the amount of weights to be handled. Lubrication was supplied by strong soap-suds fed from three large cans placed at intervals around the brake. Besides this a thin jet of water was thrown upon the brake through a flattened nozzle.

The apparatus worked perfectly, and a steady test was obtained of 475 H. P. at 76 revolutions of the wheel per minute.

This is probably the heaviest test of a single motor ever made with a brake at this speed.

The strap of the brake was made of boiler-iron lined with blocks of soft wood, and the pulley had deep flanges, so that the brake set into it like a saddle. The iron clamp was in two pieces hinged together at a point opposite the adjusting bolt.

In connection with this brake Mr. Webber used a hydraulic regulator for the scale-beam, the cylinder of which was 18 inches diameter and the piston 17⅞ inches.

In using any form of friction-brake, if the surface in contact with the pulley be too large, it will be found that a considerable weight may be added to the scale-pan without materially altering the position of the lever-arm; but if, on the other hand, this rubbing surface be too small, the resulting friction will show great irregularity—probably on account of insufficient lubrication—the jaws being allowed to seize the pulley, thus producing shocks and sudden vibrations of the lever-arm. The material in contact with brake-pulley, no doubt, enters largely into the question of smooth running, especially if the lubrication be not of the best. Soft woods, such as bass, plane-tree, beech, poplar, or maple, are generally to be preferred to the harder woods for brake-blocks. Old leather belting, secured to wooden blocks, forms a good rubbing surface, provided the leather is not sticky or gummy, and maintains a very regular motion of the brake if properly lubricated.

For high speeds and small powers the writer has found strong soap-suds very efficient for this purpose. A convenient method of supplying the lubricant to small brakes is to place a large can, provided with a pet-cock, directly above the brake, allowing the soapy water to trickle down two or more wires which lead to the pulley-surface. A trough and shield can be suitably arranged to catch the excess of water thrown from the pulley.

For light tests Mr. Webber has found that cork gives a very good rubbing surface.

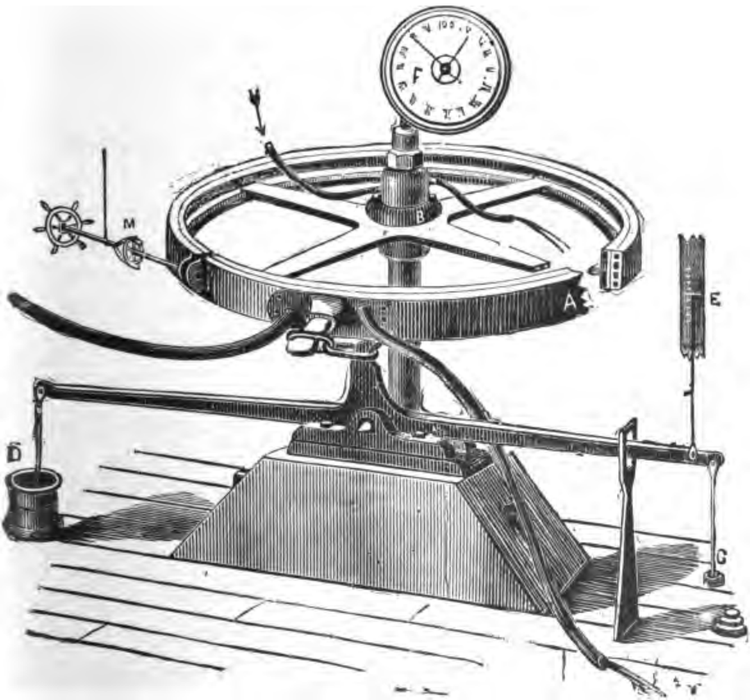

Fig. 18.

Babbitt metal has also been used for this purpose—the pulley being grooved and the Babbitt shoes cast to fit it. There is no doubt that this material would give excellent results as a brake-rubbing surface if properly lubricated.

Self-cooling brakes, Fig. 18, where both the rim of the pulley and the brake-strap were hollow, with a stream of cold water flowing through them, were used by Mr. Emerson at Lowell in 1869, oil being used on the metallic rubbing surfaces as the lubricant. In this brake the wheel *B* is made of cast-iron, and the friction-band of composition or gun-bronze, the hollow band being supplied with water from the outside, while the rim of pulley is kept cool by means of water which enters the hub and is delivered through the hollow arms to the rim.

Mr. W. W. Beaumont, in his excellent paper on "The Friction-brake Dynamometer," previously referred to, has given a formula by means of which the relative capacities of brakes can be compared, judging from the amount of horse-power ascertained by their use:

If $W =$ width of rubbing surface on brake-wheel in inches;

$V =$ velocity of point on circumference of wheel in feet per minute;

$K =$ coefficient,—then

$$K = \frac{WV}{H.P.}$$

The average of three brakes cited by Mr. Beaumont gives the value of K as 860.

In Table IV is given a number of tests and the size of brake used, from which the value of K has been calculated, as shown in the last column. These figures have been collected from various sources and represent varied practice.

TABLE IV.

CAPACITY OF FRICTION-BRAKES.

Horse-power.	R.P.M. Brake-pulley.	Brake-pulley. Face in inches.	Brake-pulley. Dia. in feet.	Length of arm in inches.	Design of Brake.	Value of *K*.
21	150	7	5	33	Royal Ag. Soc., compensating	785
19	148.5	7	5	33.38	McLaren, compensating	858
20	146	7	5	32.19	McLaren, water-cooled and compensating...	802
40	180	10.5	5	32	Garrett, water-cooled and compensating...	741
33	150	10.5	5	32	Garrett, water-cooled and compensating...	749
150	150	10	9	... .	Schoenheyder, water-cooled..............	282
24	142	12	6	38.31	Balk, compensating....	1385
180	100	24	5	126.1	Gately & Kletsch, water-cooled..............	209
475	76.2	24	7	15.91ft.	Webber, water-cooled..	84.7
125 } 250 }	290 } 250 }	24	4	63	Westinghouse, water-cooled..............	465
40 } 125 }	322 } 290 }	13	4	$27\frac{1}{2}$	Westinghouse, water-cooled..............	847

By referring to the table it will be seen that the above calculations for eleven brakes give values of K varying from 84.7 to 1385 for actual horse-powers tested, the average being $K = 655$. By a comparison of the sizes and speeds given by Mr. Heinrichs (see Table II), K is found to average 895 for small horse-powers varying from 2 to 8. From the nature of the device, these latter brakes are not water-cooled.

In the Gately & Kletsch water-cooled brake (for description of which see article by Prof. R. H. Thurston in Jour. Franklin Inst., April 1886) the wheel was designed to measure 540 horse-power, but it does not appear to have been used to indicate more than 180 horse-power.

For this number $K = 209$. In the Schoenheyder water-cooled brake $K = 282$; in the large Westinghouse brake K varied from 288 to 709 for actual horse-powers tested, averaging 465.

For the smaller Westinghouse brake, K averaged 847, which seems to be the only case in which the value of the coefficient for non-compensating brake exceeds that ascertained for compensating brakes. The average value of K for the several water-cooled non-compensating brakes is 377, and for the compensating brakes $K = 853$. Neglecting the extreme value as given for the Balk brake, K will equal 762.

From these deductions it would appear that when the brake-strap is provided with some form of compensating device (as, for instance, that shown in Fig. 10) by which a self-acting adjustment of the tension of the strap is supposed to maintain a nearly constant moment of friction, the rubbing surface is generally greater than when such device is not employed. Instead, therefore, of assuming an average coefficient of 860, the writer would propose the following:

$K = 400$ for water-cooled brake non-compensating;
$K = 750$ for water-cooled brake compensating;
$K = 900$ for non-cooled brake with or without compensating device.

For metal brake-shoes the value of K could probably be much less, as the radiation of heat from the metallic surfaces would be greater.

From the above values of K the width of brake-wheel can be obtained for the different types:

$$W = \frac{400\ H.P.}{V};$$

$$W = \frac{750\ H.P.}{V};$$

$$W = \frac{900\ H.P.}{V};$$

in which, as before, $W =$ width of bearing surface in inches on pulley, and $V =$ velocity of a point on circumference of pulley in feet per minute.

In the different forms of Prony and friction brake, it is evident that as the work of the shaft is all spent in overcoming the resistance due to friction, no useful work is done. The friction-brake is thus an absorbing dynamometer.

CHAPTER III.

ABSORPTION-DYNAMOMETERS.

ANOTHER form of absorbing dynamometer is that designed by Prof. C. B. Richards, of the Sheffield Scientific School of Yale University. It consists of a tank, *AB* (Fig. 19), within which two paddle-wheels revolve

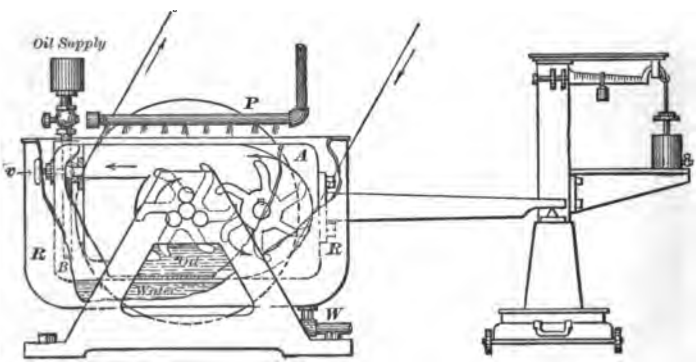

FIG. 19.

in oil, thus producing a resistance and a tendency to rotate the whole tank, which is mounted on friction-rollers. This tendency to rotate is measured by the lever-arm acting on a platform-scale. By means of the valve *v* the oil in the tank can be allowed to circulate with greater or less freedom; by closing the valve a pressure is brought to bear on the oil in the tank, so that the resistance to the rotation of the inner wheels thus becomes a drag on the driving power; when the maximum resistance is obtained without decreasing the number of revolutions per minute of the shaft, the force of resistance, measured on the scale-beam, will enable us to calculate the horse-power consumed. In order to prevent any change of temperature in the oil, a constant stream of water is discharged onto the tank through a perforated pipe, *P*, above it. Beneath the tank proper a metal receiver, *R*, catches the water, which is then carried off by the waste-pipe *W*, shown at the bottom of the receiver.

Part of the tank *AB*, and also of the outside receiver *R*, is torn away in the figure, in order to show more clearly the circulation of oil and position of the paddle-wheels. One of these latter is mounted on the pulley-shaft, and has the same direction of rotation as the belt-pulley; the other is driven by a gear (not shown), and revolves in the opposite direction. A casing at each end of the tank fits close to the paddle-wheels, the blades of which roll on each other. In this respect the internal arrangement is similar to that c various rotary engines and blowers. In order th there should be a minimum amount of vibration of he scale-beam while weighing the pressures, a rod a dash-pot were used—the latter being supported by he arm attached to the side of the scales. out

The size of this dynamometer was 30 × 1 *F F.* inches, and would measure from ¼ to 14 horse- on shell

With this apparatus, as with the Prony bra to which be seen that an absorbing dynamomete nd passing used to determine the power which is actually transmitted to a machine; it can only measure the power which is produced in circumstances as similar as possible to those under which the machine is operated; and this power is assumed equivalent to that consumed by the machine. About the year 1873, Prof. Richards used this principle of measuring the tendency of the belt to rotate a body about its axis, and designed a stand or cradle upon which the machine itself was suspended on trunnions. When the machine to be tested was put in motion, its tendency to rotate thus became a measure of the resistance.

This same principle was introduced by Prof. Brackett, of Princeton, a number of years later, in his cradle-dynamometer, which is now very generally and successfully used in testing dynamos and electric motors.

A little consideration will show that the cradle-dynamometer measures the actual power transmitted to the machine or developed by the motor, and is thus a transmitting dynamometer. As such it will be considered subsequently.

An absorption-dynamometer, by which also any esired load can be maintained on the engine, is the vention of Prof. Alden, of Worcester. This dynaiometer is essentially a friction brake in which the in oessure causing the friction is distributed over a comrotatatively large area, thus giving a low intensity of rollers.ure between the rubbing surfaces.
lever-arm friction is produced by the pressure of water valve v the city pipes acting upon two copper plates in with greateith a smooth cast-iron disk keyed to the shaft

which revolves in a bath of oil between the plates. These latter are secured by a water-tight joint to a casing which does not revolve, and to which is bolted a lever-arm carrying weights as in an ordinary Prony brake. The shell or casing is so constructed that it permits an equal pressure of water upon both sides of the disk—a sufficient quantity of the water being allowed to pass through the machine to carry off the heat due to the energy absorbed.

An ingenious form of valve operated by the slight angular motion of the dynamometer varies the supply of water, and consequently the pressure between the frictional surfaces, thus securing automatic regulation. Referring to Figs. 20 to 24, *A* (Fig. 20) is an iron disk keyed to the crank-shaft *B*. The sides of this disk are finished smooth, and each side has one or more shallow radial grooves, as shown at *X* (Fig. 21). The outer shell consists of two pieces of cast-iron, *C C*, bolted together, but held at a fixed distance apart by the iron ring *D*—whose thickness is the same as that of the disk *A*—and by the edges of the copper plates *E E*. Each of these plates at its inner edge makes with the cast-iron shell a water-tight joint by being "spun" out into a cavity in the iron and held by driven rings *F F*. Thus between each copper plate and its cast-iron shell there is a water-tight compartment, *WW*, into which water from the city pipes is admitted at *G*, and passing

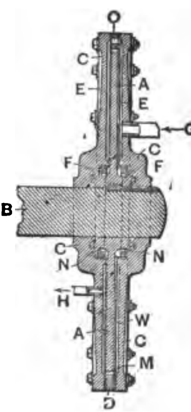

FIG. 20.

to the opposite compartment through passages, as shown at *O*, is discharged through a small outlet at *H*.

The chamber *MNN* is filled with oil, which finds its way from *N* to *M* along the grooves in the disk *A*.

The shaft is free to revolve in the bearings of the cast-iron shell *CC*. The shell has an arm carrying weights, as shown in Fig. 21. The arm has its angular motion limited by stops at *P* and *Q*.

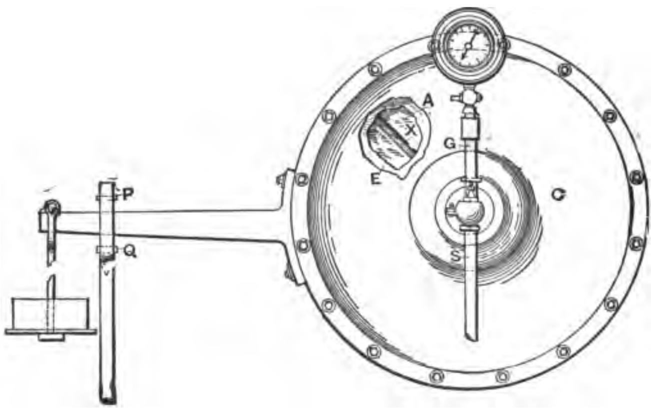

FIG. 21.

An automatic valve at *V* (Fig. 22—and shown in sections, Figs. 23 and 24) regulates the supply of water to the machine.

The valve consists of two brass tubes fitted one inside the other, but free to revolve relatively to one another. The inside tube has one end closed. Each tube has slots parallel, or nearly parallel, to its axis. One tube connects with the supply-pipe *S*, the other with a pipe rigidly fixed to the brake and communicating with one of the compartments *W*. A flexible tube, *R*, encloses the whole. The valve is so adjusted that a slight angular motion of the brake varies the free water passage through the slots (see Fig. 23); and the aperture at *H*, through which the water is discharged, being small and constant, the press-

FIG. 22.

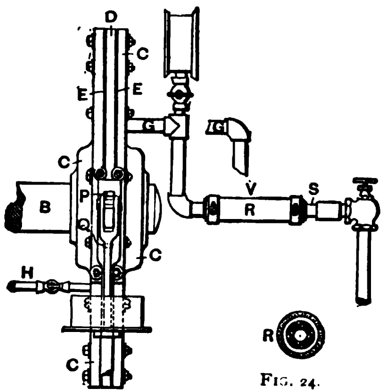

FIG. 24.

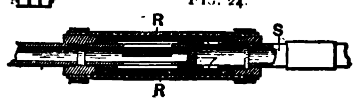

FIG. 23.

ure of the water in the chambers *W W* is thus automatically varied.

The dynamometer is operated as follows: The chamber *NNM* being filled with oil, weights are suspended from the arm to give the desired load. The engine is started, and when up to speed a valve is suitably opened in the water-pipe leading to the automatic valve (*V*), which latter being open, allows water to pass to the compartments *W W*. The pressure of this water forces the copper plates against the sides of the revolving disk *A*—with which they were already in contact—causing sufficient friction to balance the weights upon the arm, which then rises. This motion operates the automatic valve, checking the flow of water to the brake and regulating the moment of the friction on the disk to the moment of the weights applied to the arm of the brake. The first trial of the machine gave remarkable results, the arm standing midway between the stops, with only a slight and slow vibration, and this without the use of a dash-pot. The water seems a little sluggish in its action in response to the motion of the regulating valve, so that there is no sudden vibration of the arm, and the load is practically constant.*

In experimenting with a 50-horse-power Alden brake, Prof. Goss, of Purdue University, has found that the operation of the brake is very materially improved by cutting spiral grooves on each face of the revolving plate and connecting the inner compartment between the copper disks with two pipes—the one near the hub, and the other at the outer circumference of the shell. This admits of a better distribution and circulation of the oil, which is fed from the pipe connected to the chamber near the hub. From this chamber the oil is carried to the circumference, both by the radial grooves and by the spiral groove which crosses the former, thus ensuring a very even and uniform distribution of the oil, which then passes out at the circumference into a strainer situated

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* Trans. A. S. M. E., vol. VI.

above the oil feed-pipe, whence it is again carried to the central chamber at the hub, and the process repeated as long as the machine is in operation.

An interesting application of the Alden brake has been made in the Experimental Laboratory of Purdue University by which the power of an eight-wheeled passenger locomotive is absorbed. In this arrangement, Fig. 25,* the locomotive, weighing 43 tons, is mounted with its drivers, which are 63 inches in diameter, upon heavy supporting wheels, of the same diameter, free to revolve by contact with the drivers in either direction: the prolonged axles of the supporting wheels are each provided with a large flat cast-iron disk keyed to the shaft, which is allowed to rotate in a closed case between plates of copper, about three-sixteenths inch thick, which can be forced against the rotating disk by hydraulic pressure as in the Alden dynamometer. Each brake was designed for a load of 200 H. P. under a moment of 10 500 foot-pounds, with a maximum water-pressure of 40 pounds per square inch. The shaft to which the disk is keyed is $7\frac{1}{8}$ inches in diameter. The disk is 56 inches in diameter and $2\frac{3}{4}$ inches thick; it is provided with thirty-two radial oil-grooves on each face, besides which a spiral groove of about 4 inches pitch is cut across the face intersecting the radial grooves, thus thoroughly distributing and circulating the oil as in the smaller brake previously alluded to. The locomotive is free to move forward or backward only through a very small distance (about a quarter of an inch), its tendency to motion in either

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* From *Am. Machinist*, April 28, 1892.

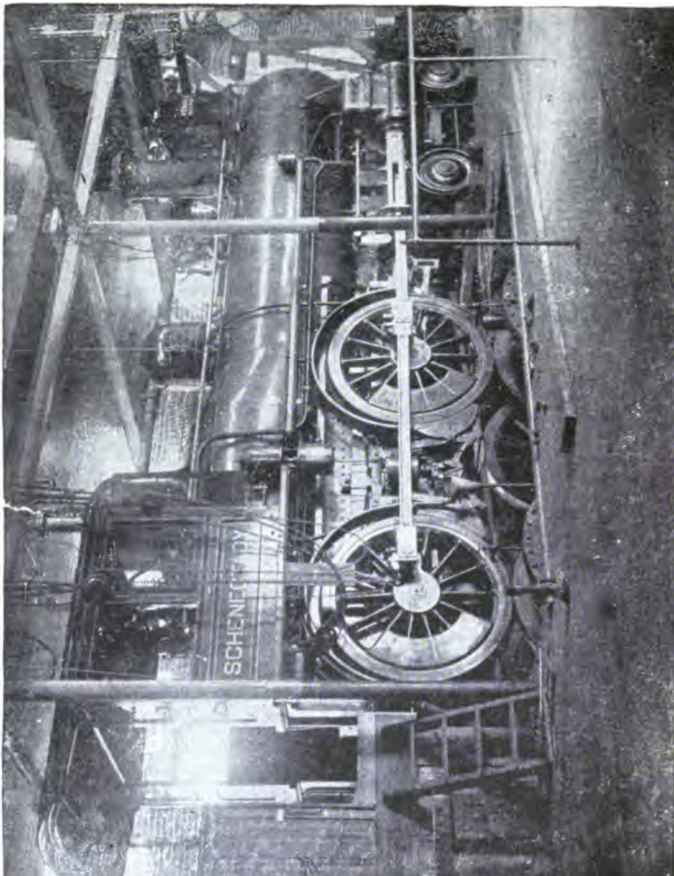

FIG. 25.—APPLICATION OF ALDEN BRAKE TO LOCOMOTIVE.

direction being measured by a system of levers and weights connected to the draw-bar by which the traction of the engine can readily be weighed. Any desired load and speed can be maintained by means of the powerful friction-brakes which are bolted securely to stone foundations—in this respect differing from the Alden dynamometer, which is free to rotate through a small arc. The smoke is exhausted through the roof of the building by a Sturtevant blower which is placed above the smoke-stack, but not in connection with it.

An absorption-dynamometer, designed by Mr. Wm. Froude * to measure the power of large marine engines is essentially another form of water-brake.

In this arrangement, the engine in delivering its power may be assumed to be winding up a weight out of indefinite depth, but the weight instead of being constant and assigned (as in the case of the suspended weight on a friction-brake) will vary with the speed of rotation much in the same way as the resistance of the propeller itself does; and thus the work performed by the engine under trial will more closely resemble its natural work, though the same circumstance renders necessary an automatic method of recording the variations of the resistance which occurs during the trial. The reaction, as will be shown, instead of arising from the friction of two solid surfaces, will consist of a series of fluid jets which are maintained in a condition of intensified speed by a sort of turbine revolving within a casing filled with water, both turbine and casing being mounted on the end of the screw-shaft in place of the

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* Proc. Brit. Inst. M. E., vol. for 1877.

screw; the turbine revolving while the casing is dynamometrically held stationary. The jets are alternately dashed forward from projections in the turbine against counter-projections in the interior of the casing, tending to impress forward rotation upon the casing, and are in turn dashed back from the projections in the casing against those in the turbine, tending to resist the turbine's rotation. The important point is that the speed of jets is intensified by the reactions to which they are alternately subjected; and thus, in virtue of this circumstance, a total reaction of very great magnitude is maintained within a casing of comparatively very limited dimensions.

The nature of this arrangement will be understood by referring to the following figures, which represent the dynamometer as designed to measure 2000 H. P.

In Fig. 26, *A* represents the screw end of the screw-shaft; *BB* shows in section what has been termed "the turbine"; it is a disk or circular plate 5 feet in diameter, with central hub keyed to the shaft in place of the screw, and revolving with the shaft. The disk is not flat throughout its entire zone, being shaped into a semi-oval section which sweeps around the whole circumference concentric with the axis. In Fig. 27 Fig. 26 is repeated and the "casing" is added, *CC* representing the front and *DD* the back.

The face is shaped into a channel the counterpart of that in the turbine disk, so that the two semi-oval channels in effect form one complete channel. The back of the casing encloses the turbine entirely, but without touching it. The casing is also provided with a hub, which is an easy fit over that of the turbine, so that the latter is free to revolve within the casing, which is stationary. Both casing and turbine are provided with a series of twelve fixed diaphragms, one of which is shown in Fig. 28. These diaphragms cut the channel obliquely, being semicircular in outline, so that when set at an angle, as shown in side-view (Fig. 29), their circular edges fit the

FIG. 26.

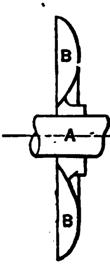


FIG. 27.

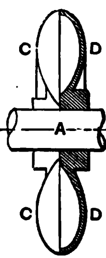


FIG. 28.

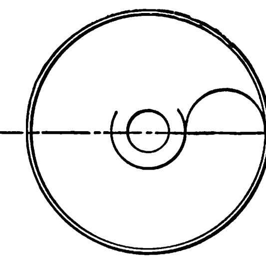
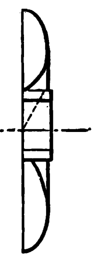


FIG. 29.



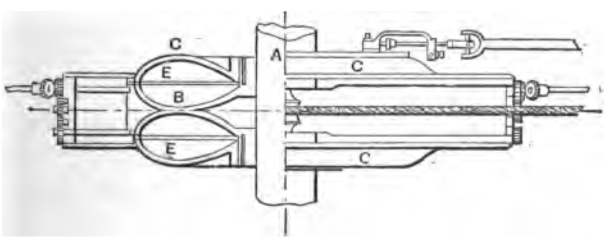


FIG. 30.
FROUDE'S MARINE-ENGINE DYNAMOMETER.

oval bottom of the channel, while their diameters span the major axis of the oval. Thus is formed by casing and turbine, when the diaphragms are opposite to each other, a series of cells; and as the function of the turbine is to rotate while the casing remains at rest, one half of each cell is moving past the other half in such a manner that the moving half, if viewed from its stationary counterpart, would appear to be advancing antagonistically towards it. The effectiveness of this combination to resist rotation will be seen to depend essentially on this assumed antagonistic motion. The channel and casing is filled with water, and the turbine is made to rotate as described. When the turbine is thus put in motion, the water contained in its half-cells is urged outward by centrifugal force, and in obeying this impulse it forces inward the water contained in the half-cells of the stationary casing, and thus a continuous current is established—outward in the turbine's half-cells, and inward in those of the casing.

The current, though in fact originated solely by centrifugal force, possesses, when once called into existence, a vitality and power of growth quite independent of centrifugal force and dependent on, what has been called, the virtually antagonistic attitude or motion of the two sets of diaphragms and the cells of which they are the boundaries.*

It can be shown that, with a dynamometer of given dimensions, the reactions which tend to stop rotation of the turbine and to give rotation to the casing will be as the square of the speed of rotation of the shaft to which it is attached; and that by comparing two similar, but differently-dimensioned turbines, their respective *moments of reaction* for the same speed of rotation should be as the fifth powers of their respective diameters.

Mr. Froude constructed an experimental pair of dynamometers in which the turbine diameters were re-

* For discussion of principles involved, see Appendix in vol. for 1877 Proc. Brit. Inst. M. E.

spectively 12 inches and 9.1 inches. Now $\left(\frac{12}{9.1}\right)^5 = 4$, and therefore the ratio of moments of the two instruments at a given speed should also have been 4. The ratio determined by experiment was in fact 3.86, but the small difference is referable to the circumstance

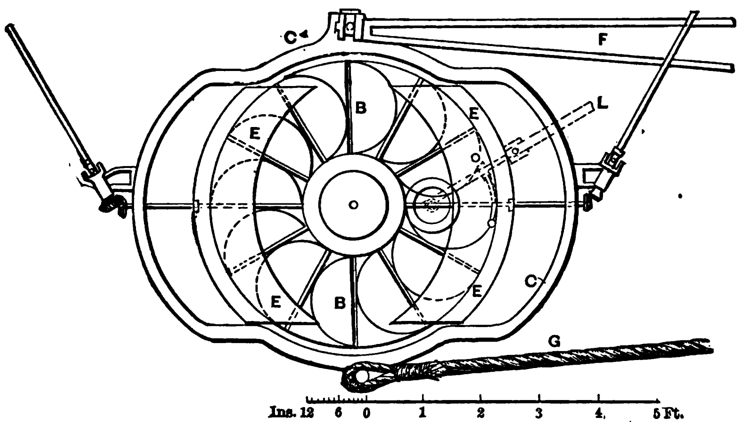


FIG. 31.—FROUDE'S MARINE-ENGINE DYNAMOMETER.

that in the larger of the two instruments the internal surface was rougher and the friction of the water greater. The data thus obtained not only verify the scale of comparison based on the fifth power of the respective diameters, but also furnish a starting-point by which to proportion the dimensions of an instrument required to deal with any given horse-power delivered at a given speed.

It thus appears that an instrument similar to that shown in Figs. 30 and 31 will measure 2000 H. P. at 90 revolutions per minute; the turbine being 5 feet in diameter, and formed with two faces, with a double-sided casing to match. This double arrangement, it may be added, while it supplies a double circumferential reaction with a given diameter, has the advantage of obliterating all mutual thrust on working parts. In order to adapt this dynamometer to measure varying horse-powers—that is, to produce readily a greater or less reaction with a given number of revolutions—two sliding shutters, *E E*, of thin metal, fitted between the turbine and casing, are arranged so that each shutter may be carried forward by a screw-motion governed from the outside.

By this means the internal water-ways or passages through the cells are contracted and the reactions greatly reduced.

The experiments with the models showed that, with any given speed of turbine, the reaction could be reduced with a perfectly graduated progression in any required ratio down to one-fourteenth.

The intensity of reaction is thus easily brought under the control of the operator within a wide range. The brake represented in the figures, and designed, as stated, for an engine of 2000 H. P. at 90 revolutions per minute, is also capable of dealing with one of 340 H. P. making 120 revolutions per minute.

The mechanical reaction due to friction in the working parts of the instrument, while of relatively small amount, is in effect wholly incorporated with the hydrodynamical reaction, and is thus taken account of.

In applying this dynamometer to measure the power of a ship's engines the instrument is mounted upon the screw-shaft in place of the screw, as shown in Fig. 32.

The casing is provided with proper apertures, capable of being closed at will, to permit the egress of air and ingress of water.

If the moment to be measured and recorded be regarded as the product of two factors, force and leverage, of which the one varies inversely as the other, it is plainly a question to be settled by considerations of

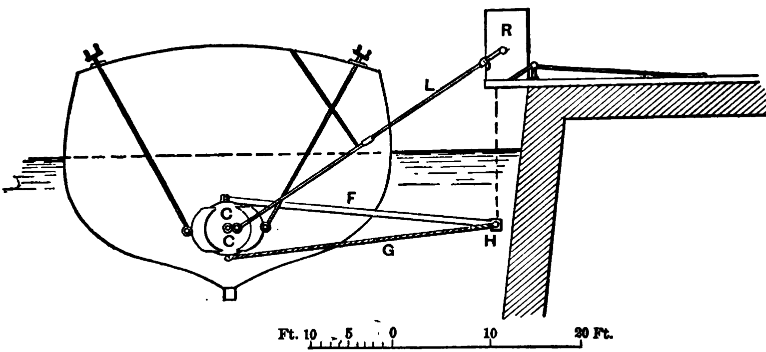


FIG. 32.—MARINE-ENGINE DYNAMOMETER. MODE OF APPLICATION.

convenience, whether the record shall involve a large force delivered at short leverage, or *vice versa*. In the present case it will be seen that a large leverage is desirable; for, if we assume the force to be acting at the circumference of the casing, say 3 feet from centre of shaft, there will be required $\frac{2000 \times 33000}{2 \times 3.14 \times 3 \times 90} = 38904$ lbs.—a force which will bear large reduction. In the arrangement shown (Fig. 32) the leverage has been increased in the ratio of 10 to 1.

The lever here shown consists of a rod *F* and wire rope *G*, connected to the casing *C* at one end and unit-ing in H at the other. As the force at H acts downwards, it will be seen that F is in compression and G in tension.

A suitable weighing apparatus, consisting of a system of flat springs and levers, is provided for ascertaining the load, to which is attached a recording device connected to the screw-shaft through the rod L, which takes its motion by bevel gears directly from the shaft.

More recently, Prof. Osborne Reynolds,* of Owens College, Manchester, has constructed several of these water-brakes for experimental purposes; and as the result of his experience he finds that air is drawn from the water and accumulates in the centre of the cells, occupying water space and diminishing resistance, besides producing an irregular motion. This would be prevented if passages could be provided through the outside to the axis of vortex within, carrying a supply of water at or above atmospheric pressure, so as to prevent the pressure at this point falling below that of the atmosphere. This was accomplished by Prof. Reynolds by perforating the vanes of the turbine and supplying water through the perforations. It also appeared that by having similar perforations in the casing open to the atmosphere the pressure at centre of vortex could be rendered constant, whatever the supply of water and speed of wheel, so that it would then be possible to run the brake partially full and regular; resistance from nothing to maximum, without sluices.

These conclusions being verified on a small model (4-inch turbine), three larger brakes with 18-inch wheels

* See Proc. Brit. Inst. C. E., vol. XCIX, also Van Nostrand's Science Series, No. 99.

were constructed. These brakes proved everything desirable except when running under a constant load with varying speeds. This matter was considered during their construction, and an arrangement was devised by which the supply and exit of water to and from the brake was automatically controlled; the lifting of the lever opening the exit and closing the supply so as to diminish the quantity of water in the brake, and *vice versa.*

During the twelve months these brakes have been in use they have received no attention whatever. The casing is provided with a lever 4 feet long from centre of shaft to the weight. When the speed of engines reaches about 20 revolutions per minute the levers rise (whatever load they have on), and though always in slight motion, they do not vary half an inch until the engines stop; during the run, the load on the brakes may be altered at will without any other adjustment.

The engines to which the brakes were connected were each designed to work with any steam-pressure up to 200 pounds per square inch, at any piston-speed up to 1000 feet per minute, and to have expansion-gear to cut off from 0 up to two-thirds stroke.

Each engine was furnished with a fly-wheel weighing 1200 pounds.

The dimensions of engines were as follows:

High-pressure..	5 inches	diameter,	10 inches	stroke	
Intermediate...	8 "	"	10 "	"	
Low-pressure...	12 "	"	15 "	"	

All the cylinders were steam-jacketed, but arranged so that any or all of the jackets could be cut out.

CHAPTER IV.

TRANSMITTING-DYNAMOMETERS.

Half a century ago, Morin gave as the requirements of a dynamometer the following, viz.:

First. The sensibility of the instrument should be proportioned to the intensity of efforts to be measured, and should not be liable to alterations by use.

Second. The indications of flexures should be obtained by methods independent of the attendance, fancies, or prepossessions of the observer, and should consequently be furnished by the instrument itself, by means of tracings, or material results, remaining after the experiments.

Third. We should be able to ascertain the effort exerted at each point of the path described by the point of application of the effort, or, in certain cases, at each instant in the period of observations.

Fourth. If the experiment from its nature must be continued a long time, the apparatus should be such as can easily determine the total quantity of work expended by the motor.

To meet these conditions, Morin made the spring-dynamometer, in order to obtain the magnitude of a force, as, for instance, the traction of a horse on a loaded wagon or canal-boat.

In this dynamometer a force was measured by the flexure produced by it on two springs connected at their ends and loaded in the middle, Fig. 32*a*. The force P is applied at the centre M of spring AB, and its magnitude is determined by the increase of the distance MN between the two springs AB and CD

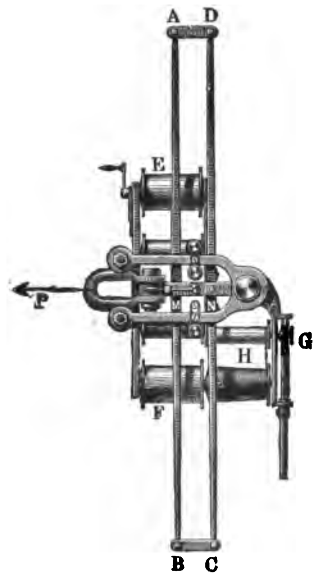


FIG. 32*a*.—TRACTION-DYNAMOMETER.

when the latter is attached to its load at centre N. When a steel bar of rectangular cross-section is placed freely upon two supports, and subjected in the middle to a force P perpendicular to its length, its flexure, d, so long as it does not exceed the limits of elasticity, will be

$$d = \frac{1}{48} \frac{Pl^3}{IE};$$

in which

d = deflection of bar,

l = length of bar between supports,

I = moment of inertia for rectangular section,

E = modulus of elasticity for material of the bar.

If b = breadth of the bar, we shall have, since $I = \frac{bh^3}{12}$,

$$d = \frac{1}{4} \frac{Pl^3}{Ebh^3}.$$

Since there are two springs in this case, the total deflection will be

$$d = \frac{1}{2} \frac{Pl^3}{Ebh^3}.$$

Now if the longitudinal profile of the bar be parabolic, the flexure will be double that of a spring of uniform thickness, while the strength remains the same; hence we have

$$d = \frac{Pl^3}{Ebh^3} = nP,$$

where n is a number to be determined by experiment. If in the construction of a spring-dynamometer known weights be applied, and the deflection, d, observed, the number, n, can be calculated and used in the construction of a scale.

Morin found that with good steel the deflection may reach one-tenth of the length of the spring before the relation between it and the force changes.

In order to determine the relative strength of men and animals Regnier constructed the spring-dynamometer shown in Fig. 32*b*, in which two elliptical steel springs about 12 inches long are connected at

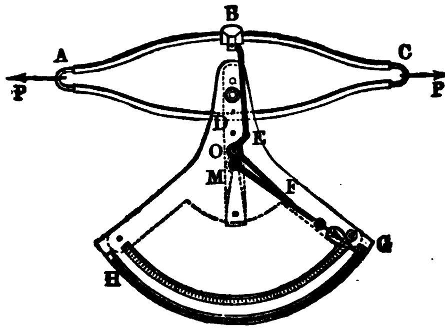


FIG. 32*b*.—SPRING-DYNAMOMETER.

A and *C*, forming a closed ring. *DEGH* is a sector suitably graduated and rigidly attached to the lower spring; the lever *F*, pivoted at *O*, is caused to move when the two springs are brought together either by pulling on the ends or by compressing at *B* and *D*.

Above the lever *F* is a self-registering index-finger *MG* which is actuated by a pin attached to the hand *F*. When force is applied to the dynamometer the hand *F* presses against the index-finger and moves it to a position corresponding to the magnitude of the force. A friction washer at *M* causes a sufficient amount of friction to hold the pointer in position after the force is withdrawn.

In order to meet the second, third, and fourth requirements mentioned in the beginning of this chapter, Morin designed a self-registering recording dynamometer, by which the work performed was traced upon a continuous roll of paper, set in motion by suitable wheelwork as shown at *GH*, Fig. 32*a*.

More recently Mr. C. M. Giddings of Rockford, Ill.,

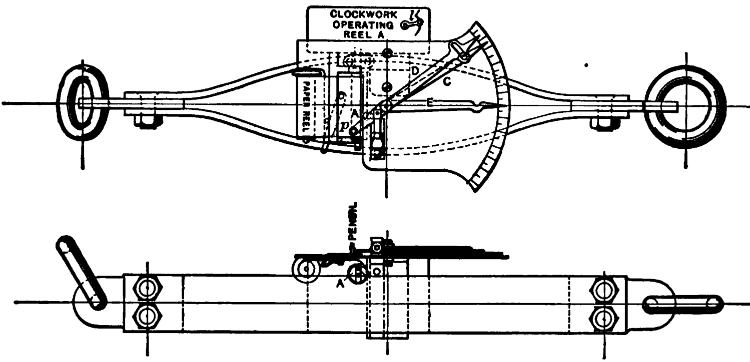


FIG. 32*c*.—GIDDINGS TRACTION-DYNAMOMETER.

has constructed such an apparatus, Fig. 32*c*, especially designed for obtaining the draught-power of traction-engines and horses on roads or in the field. This dynamometer consists of a pair of elliptical springs joined at the ends with links for attachment to load. To one of these springs is attached the dial-plate, and also the pivoted index-hands.

The pointer *C*, carrying a pencil at *p*, is attached to the opposite spring by means of an adjustable connection. The slotted index-finger *D* has a slight friction on the pivot, and serves as an averaging hand to the reading of the pointer *C*.

A pin projects through the end of *C* both above and below; the upper portion works in a slot in the averaging hand *D*, which is a dark color to distinguish it from the other when in motion. The lower end of the pin comes in contact with the pointer *E*, and carries it up to the point of maximum effort, where it remains.

To the spring which carries the dial-plate is also attached a clock movement that rotates spool *A*; the lever *l* permits this spool to be started and stopped as desired.

The record-paper is placed upon a wooden roller and is wound on spool *A* when the machine is in use. Fig. 32*d* is reproduced from an actual diagram taken on

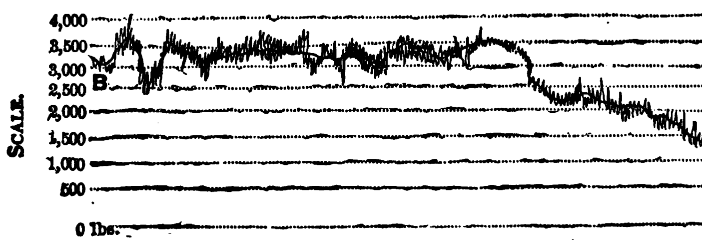


FIG. 32*d*.—DIAGRAM FROM GIDDINGS TRACTION-DYNAMOMETER.

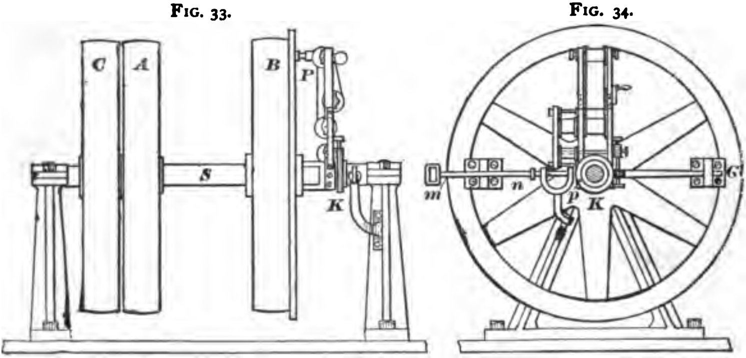
a Giddings traction-dynamometer during a trial of steam-ploughs at Massillon, Ohio.

In this case the abscissæ represent time; but if the operating mechanism be driven from the wagon or traction wheels, the abscissæ will be proportional to the space passed over, and the area of the diagram will then represent work done.

When required to determine the force of rotation of a shaft or pulley the preceding forms of dynamometer require modification; the essential features, however, remain the same. The following description illustrates the application of the foregoing principles applied to the Morin rotation-dynamometer.

Upon a shaft resting on two cast-iron supports are three pulleys of the same diameter, Figs. 33 and 34. *A* is fixed, *C* is loose, and *B* is movable around the shaft between the limits which we shall indicate. This apparatus being placed between the driving-shaft and a machine whose resistance is to be measured, the loose pulley *C* receives the power from the driving-shaft by

FIG. 33. FIG. 34.



MORIN'S TRANSMISSION-DYNAMOMETER.

means of a belt, which, when transferred to *A*, sets the shaft *S* in motion.

The pulley *B* is free on the shaft, and is connected to it by means of two parabolic springs which are fastened to the shaft, and at the end *G* to the rim of *B*. These springs turning with the shaft deflect more or less, according to the resistance encountered, and when the resistance to flexure overcomes the resistance of the machine, motion is transmitted through the springs to *B*.

Upon the shaft is a worm, *K*, having a stop, *p*, so that

7

by means of a sliding bar, *mn*, it may be prevented from revolving with the shaft during the experiment. By a suitably arranged train of gearing a series of drums is set in motion, by means of which a roll of paper is caused to pass under a pencil, *P*, attached to one of the arms of pulley *B*, thus recording the resistances, and giving a measure of the work performed.

Using the same notation previously given, and substituting *R*—radius of path in feet—for *L*, we have the work done

$$W = 2\pi RNP;$$

where $P =$ resistance overcome in the machine driven by the dynamometer. *P* can be readily ascertained when deflection of spring is known. One of the principal objections to the use of this instrument is that the centrifugal force of the rotating pieces enters as a factor into the final result; for accurate work this will necessitate corrections for different speeds, and in this respect Morin's dynamometer does not fulfil his third requirement of a good instrument, viz.: "We should be able to ascertain the effort exerted at each point of the path described by the point of application of the effort, or, in certain cases, at each instant in the period of observations."

Another form of transmitting-dynamometer, sometimes called the differential dynamometer, was introduced into this country by Mr. Samuel Batchelder, of Saco, Maine, in 1836. The principle of this machine is, that to hold a weight by the radius of a circle in a horizontal position takes as much power as to lift the same weight through the distance which would be traversed by it in any given number of revolutions if

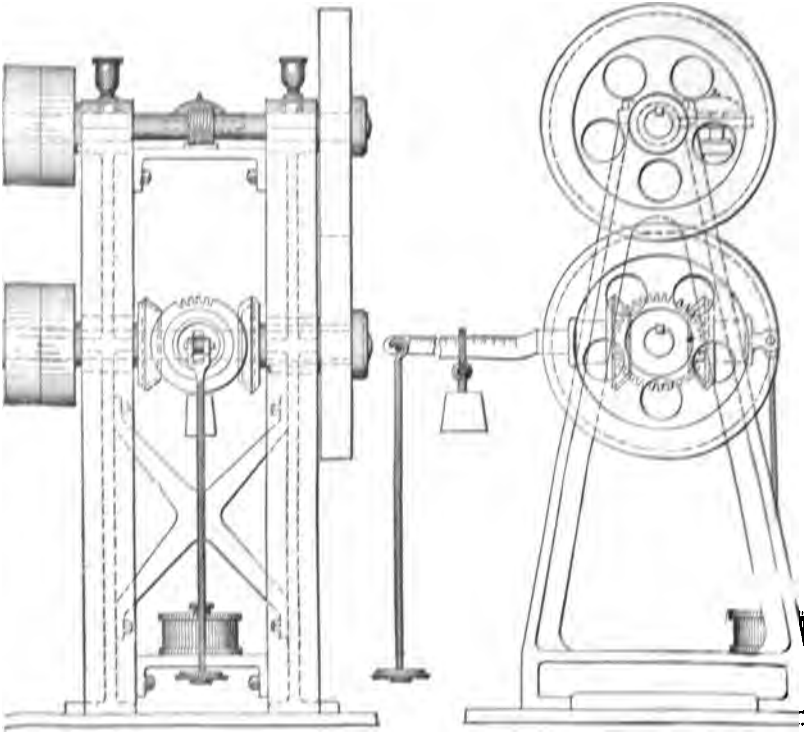


FIG. 55.—WEBBER BALANCE-DYNAMOMETER.

rotated in the circle and in the time required for such number of revolutions. We have already seen that this is the governing principle of the Prony brake where the lever is maintained in a horizontal position, the work being estimated as though the weight suspended at the end of the lever rotated in a circle whose radius was equal to the length of arm *L*. Though alike in principle, the methods by which these two dynamometers operate are radically different. The Batchelder instrument, improved and modified, is now made by the Lawrence Machine Co., and known as the Webber balance-dynamometer.

The following description of this machine (see Fig. 35) is taken from a paper by S. S. and W. O. Webber, read before the Society of Mechanical Engineers.*

"On the receiving-shaft are fixed a pair of fast-and-loose pulleys at one end, and a spur-gear at the other. This spur-gear drives a corresponding gear of the same size and number of teeth, which is fixed on the end of a sleeve or collar, having on its other end a bevel-gear which forms one side of what is known as a 'box' or 'compound' gear. A corresponding gear on the opposite side of the 'box' is fixed on the delivering-shaft which passes through the sleeve above mentioned, and also through the fulcrum of the scale-beam. The two remaining sides of the box are composed of a pair of equal and similar gears, which revolve freely around the scale-beam on either side of the fulcrum. "One would really be sufficient for the purpose, but a pair is used in order to preserve a balance. When motion is given to the shafts by means of a belt to the

* Trans. A. S. M. E., vol. IV.

receiving-pulley, the intermediate gears revolve about the scale-beam without effect; but when a belt is carried from the delivering-pulley to the machine to be tested, the resistance causes the intermediates to act with the effect of levers on the scale-beam, and would put the latter in revolution about its axis or fulcrum if it were not restrained by the weights, which are to be added, and adjusted until a balance has been obtained. It will be readily seen that the real motion of the scale-beam, were it free to move, would only be one half that of the shafts, and the weights in actual use are therefore double their apparent value—or in other words, the weight marked 1000 pounds is in reality two pounds instead of one."

The circumference of the circle through which the weight would travel, were it free to move, is ten feet, therefore we can readily calculate the horse-power from the following:

$$H.P. = \frac{Pv}{33\,000} = \frac{P \times 2\pi RN}{33\,000};$$

since $2\pi R = 10$, we have

$$H.P. = \frac{10PN}{33\,000},$$

in which, as in our former notation, $P =$ pounds weight, $N =$ revolutions per minute, and $v =$ velocity in feet per minute.

The weights are marked for $N = 100$.

One of the older forms of dynamometer in which the tendency to rotate a shaft was weighed on an arm, as in the present case, was the Hachette steelyard-dynamometer,* a machine now seldom used.

A recent modification of this apparatus is known as the Riehlé-Robinson dynamometer designed by Prof. S. W. Robinson.

This machine, Fig. 35*a*, consists essentially of a supporting frame, or pedestal, a T-shaped arm carrying the driving mechanism, and a graduated scale or weighing-apparatus.

In use the dynamometer is made fast to the floor and the lower pulley belted to the machine to be tested, while a second belt connects the upper pulley on the dynamometer with the pulley on the power shaft.

The two pulleys of the dynamometer are mounted on a strong cross-tree bar so that they both overhang and can be swung around to any position.

This overhang and swing makes it convenient to put either belt on or off without unlacing, and to swing the pulleys either way for tightening or loosening both belts.

The pulleys have each a gear on the end of the hub, both of which mesh into a smaller gear between, the latter being supported on a pin made fast as a crank-pin in an arm attached to a shaft which passes through the centre of the hub of the cross-tree. To the opposite end of this shaft the poise-bar is made fast by a set-screw in a boss to which the poise-bar is secured.

* Weisbach, vol. II, Hydraulics, p. 47.

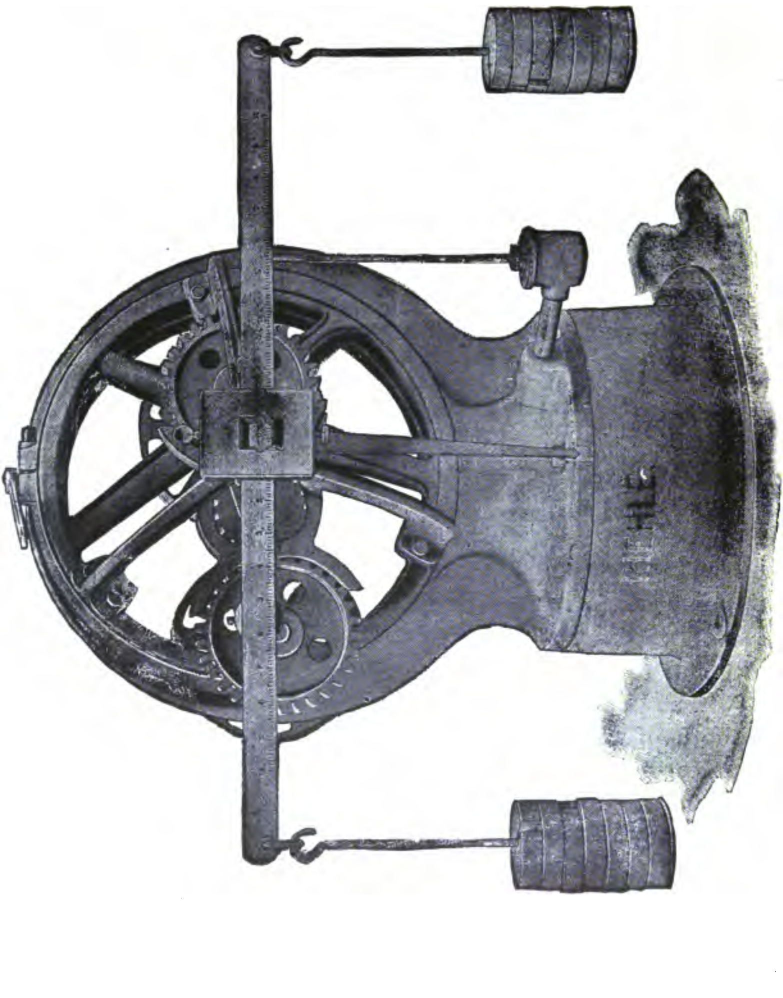


Fig. 35a.—Robinson-Riehlé Dynamometer.

As work is transmitted from one pulley to the other through the intermediate gear on the crank-pin, the latter will be thrust to one side with a force proportional to the effort transmitted; this tends to tip the poise-bar, to prevent which weights are applied.

The machine is so proportioned that each 5-lbs. weight in the scale-pan represents 100 foot-pounds of work per revolution.

The equilibrating weights on the poise-bar, together with the speed, furnish data for calculation of the work being transmitted.

The value thus obtained must be corrected by the use of a diagram or table of constants which have been previously determined by calibrating the machine.

Another form is that known as the belt transmission-dynamometer, used by Dr. Hopkinson in his tests with the Siemens dynamo-electric machines.

The principle involved is the weighing of the resulting stress from a deflected belt, and by this means ascertaining the direct stress upon the belt itself.

As previously intimated, the power exerted by a belt is the difference of strain on the two sides of the belt, multiplied by the velocity of a point on the belt. A belt connecting two shafts, when at rest, has the same tension in all its parts, but as soon as work is performed by the belt this uniform tension ceases, the driving-shaft exerts a pull on the driven proportional to the resistance overcome, and as the adhesion of the belt is brought into play, one side—that on which the pull is exerted—is tightened, while the other is correspondingly slackened.*

To obtain a measure of this difference in belt-strain, the dynamometer shown in Fig. 36 was designed by Mr. Robert Briggs.

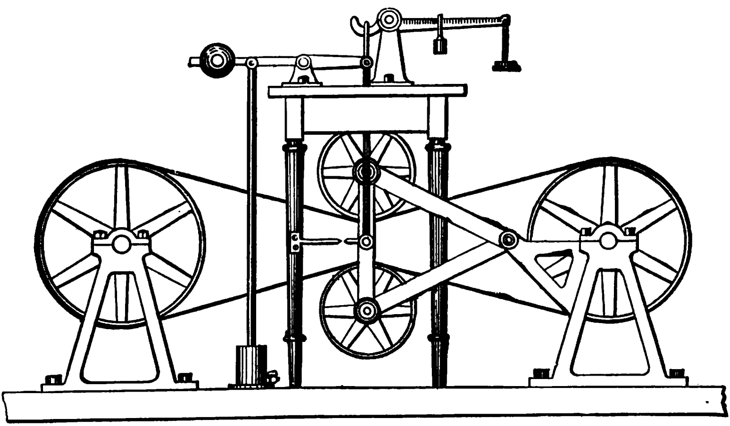


FIG. 36.—BRIGGS BELT-DYNAMOMETER.

In this arrangement it is evident that when at rest, or running with no resistance, the system will come into equilibrium with equal but opposite angles for both the lower and upper belt—provided the weight of the carrier-pulleys, the frame supporting same, and the weight

* In a series of experiments on leather belting made by Wm. Sellers & Co. in 1885 it was shown that the sum of the belt-tensions is not constant, but increases with the load. This is contrary to the generally accepted theory that the sum is constant, but subsequent experiments have shown that the total tension actually increases as the difference increases, whether the belt be horizontal or vertical.

of the belt are balanced. It can be shown that the resultant of strain from the deflected belt varies as the cosine of the angle which the belt makes with the ver-

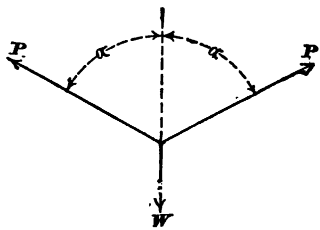


FIG. 37.

tical, or $W = 2P \cos \alpha$, (Fig. 37); therefore, if we make the angle $75° \ 31'$, the cosine will equal 0.250, or

$$\cos^{-1} \tfrac{1}{4} = 75° \ 31'.$$

Let $\cos \alpha = .25$; tension on tight side of belt $= T_1$; tension on slack side of belt $= T_2$; weight $= W$; force transmitted $= P$; then will $T_1 - T_2 = P$. Now, since $W = 2P \cos \alpha$, and $\cos \alpha = 0.25$, we have $W = 2P \times \tfrac{1}{4}$, hence

$$P = 2W.$$

If, therefore, a weight w be applied on the scale-beam so that it exerts a force W, acting downwards, there will be transmitted by the belt a force $P = 2W$, in order to maintain the system in its central position; and this force is a measure of the driving power of the belt.

Accepting these relations of angles and force, the following diagram, Fig. 38, will show the relative positions of the arrangement employed. An allowance of $\frac{1}{10}$ inch has been made for half the thickness of belt when the radii of the line of the belt on the two pulleys become as shown, 8.1 and 12.1 for the 16- and 24-inch pulleys respectively.

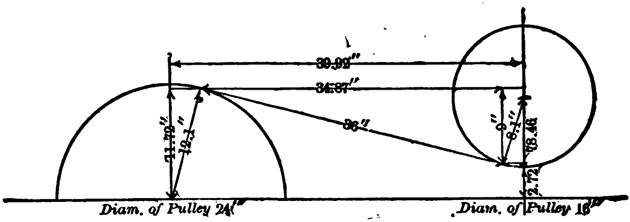


FIG. 38.

A modification of this dynamometer was designed by Prof. Elihu Thomson, in which the angle α was made equal to 60°.

Another form, in which the difference in tension of the slack and driving sides of the belt is exerted to vibrate a system of lever-arms and scale-beam, is that designed by Mr. W. P. Tatham of Philadelphia, and constructed for the use of the Franklin Institute.*

This machine, Fig. 39, consists of a double gallows-frame constructed of wood, framed together at the foot, and sustaining at the top a cross-block, from which the scale-beam is suspended. This beam is capable of weighing 300 lbs., and is graduated to 25 lbs. by pounds and tenths.

When the indicator is employed, a spring-balance is

* See Journal Franklin Institute, Dec. 1882.

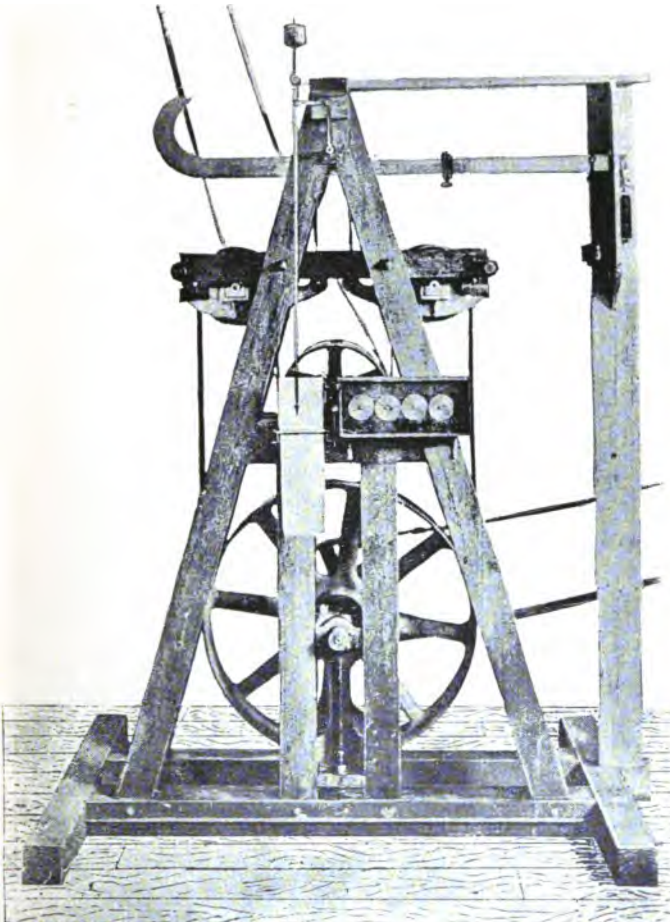


FIG. 39.—TATHAM DYNAMOMETER.

attached near the extreme end of the beam so as to exhibit 25 lbs. by pounds and tenths.

On each side of the principal centre of the beam and 1.9 inches therefrom (unseen in the figure) are knife-edges, from which hang two links suspending the free-moving ends of two cast-iron lever-frames, whose fulcrums are outside knife-edges which rest upon two iron plates bolted to the gallows-frames.

Each of these lever-frames carries a pulley whose face is 7 inches and whose average radius is 4.30 inches.

The axis of the pulley is placed 8.78 inches from the link knife-edge and 4.39 inches from a line joining the fulcrum knife-edges. The effective radius of the pulley is found by experiment to be 4.38 inches.

The middle pulley, partially obscured by the counter and indicator-card, represents the machine on trial.

Its shaft is produced towards the observer, and by means of a clutch and sleeve carrying a small spur-wheel and worm-screw, the counter and card are put in or out of gear at pleasure. The shaft, produced towards the rear, carries an outside pulley and may be coupled directly to the machine on trial, or connected with it by a belt.

The middle pulley has a face of 7 inches and an average circumference of 38.594 inches. Careful measurements showed that the actual delivery of belt per revolution was 39.595 inches, or about .005 inch less than the 3.3 feet desired.

The larger lower pulley, 30 inches diameter and 7 inches face, is the driver on the first-motion shaft. It is on a shaft which receives power from an outside source, and runs in journals on a frame adjusted vertically in slides by means of set-screws, so as to tighten the belt.

The belt runs in the direction of the arrows on the outside, down on the left and up on the right. But in describing its operation, it is best to follow the tension of the belt in a direction contrary to the motion of the belt itself.

The tension, originating at the lower driving-wheel, acts vertically upon the left-hand idler-pulley at the extremity of its effective radius, and in a line joining the two knife-edges of the fulcrum, and therefore the effect of this part of the belt upon the scale-beam is *nil*.

Losing enough force to overcome the friction of the idler-pulley, the remaining tension acts vertically: *first*, by reaction upon the lever-frame carrying the idler-pulley, at a point corresponding to the extremity of the inside effective radius of the pulley, and thence through the link, upon the positive side of the scale-beam; and *second*, upon the middle pulley representing the machine on trial. These forces are equal and opposite.

The tension acting upon the middle pulley, there performs the work which is to be measured and is reduced thereby. The remainder acts, *first*, by reaction on the middle pulley, and, *second*, directly upon the lever-frame carrying the right-hand idler-pulley as before, and thence through the link to the negative side of the scale-beam. These two forces are equal and opposite.

The tension then passes over the idler through the fulcrum, as before, to the place of beginning. The outside slack tension has therefore no influence on the scale-beam.

It is evident from this description that the only forces bearing upon the scale-beam are the tension of the tight belt on the positive side of the beam, and the tension of the slack belt on the negative side. The scale-beam therefore weighs the difference between the two.

The horse-power absorbed by the machine being tested may be found from the general formula

$$\frac{Pv}{33\,000} = H.P.$$

P in this case equals the number of pounds shown on scale-beam, v = velocity of belt = $2\pi RN$, where N is the number of revolutions per minute; but, as previously shown, the velocity of the belt is 3.3 feet per revolution, therefore the equation for this particular machine becomes

$$H.P. = \frac{PN}{10\,000}.$$

The principal centre of the scale-beam is lengthened towards the observer, and at its nearest extremity carries a vertical lever-arm attached to a horizontal link connecting it with a long vertical index-lever which carries a pencil at its lower end, moving horizontally as the end of the beam vibrates vertically. This pencil marks upon a ribbon of paper caused to move vertically between two revolving rollers, which are driven by the worm-screw upon the prolongation of the shaft of the middle pulley before mentioned. One hundred revolutions of this worm cause one revolution of the worm-wheel upon one of the rollers.

The scale-beam being attached to a spring-balance when the indicator is used, the ordinates of the curve traced by the pencil, plus the weights hanging on the scale-beam, will represent the force employed, while the abscissas will represent the motion.

The Tatham dynamometer, which measured the power consumed by the dynamo-electrical machines tested by a committee of judges in June, 1885 (see report in supplement to the Franklin Institute Journal, Nov. 1885), is capable of measuring 100 horse-power. The largest machine then measured required 70 H. P.; the smallest, 0.23 H. P. This machine, Fig. 40, occupies a floor-space of about 6 by 4 feet, and is $7\frac{1}{2}$ feet high. Upon the cast-iron bed-plate, which is provided with levelling-screws, are erected the two main frames, bolted together and united at the top by an arch from which the scale-beam is suspended. A movable A-frame in two parts is hinged to the bed-plate, and when in position holds firmly the journal-boxes of the outside bearing of the two middle shafts.

When opened, it gives liberty to change the outside pulleys, or the belts which run upon them.

This dynamometer is upon the same principle as the machine represented in Fig. 39, but differs from it in that the single pulley upon the first-motion shaft of the latter is replaced by three pulleys in the present machine. See skeleton diagram Fig. 41.

All of the pulleys are cast-iron plate-pulleys, turned all over and accurately balanced. They are $12\frac{1}{2}$ inches

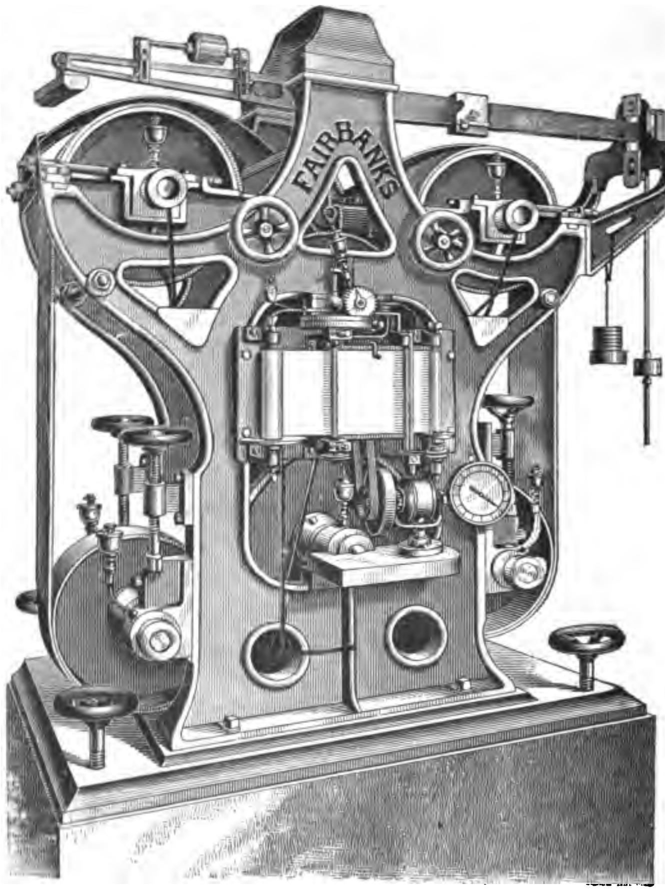


FIG. 40.—THE TATHAM DYNAMOMETER. IMPROVED FORM.

face, and are upon steel shafts 2 inches diameter, running in brass boxes which are from 6 to 8 inches in length.

The pulley *D* is 25 inches diameter, crowned and placed upon the first-motion shaft, which receives

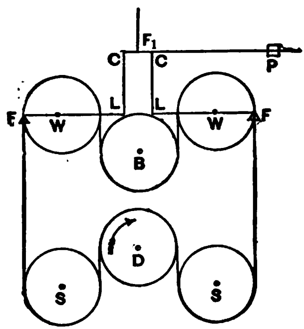


Fig. 41.

power from an outside belt. The pulley *B*, 25 inches diameter, ground perfectly true and flat, is upon a shaft which conveys the power to the machine to be tested. In measuring a motor, its power is applied to the pulley *B*.

The two pulleys *S* are crowned, 21 inches diameter, and their shafts run in bearings which are upon vertical slides regulated by screws. The vertical movement of these pulleys regulates the tension of the belt. The pulleys *W* and *W* are 21 inches diameter, slightly crowned, and their shafts run in bearings upon the two lever-frames *LF* and *LF*, each of which has its fulcrum in a pair of knife-edges at *F* resting upon the main frame. The inside ends of the lever-frames are suspended by links *LC* and *LC* to the scale-beam *FP* at equal distances on either side of the principal centre of the beam. There are two adjustments to each of these lever-frames. (1) Two micrometer-screws adjust the position of the centre of the pulley, so that the line of effect of a belt hung on it on the outside will pass through the fulcrum, and no addition of weight to the belt will affect the scale-beam; which is experimentally proved. (2) The position of the knife-edge suspended to the link is adjusted so that the scale-beam weighs accurately any weight suspended by a piece of belt hung over the inside of the pulley.

The endless belt used was a four-ply gum belt, 12 inches wide and 0.26 inch thick.

If the belt were 0.21 inch thick, its delivery would have been 6.6 feet per revolution of the pulley *B*.

It will be seen by the construction that the pulley *B* is actuated by the difference of the tensions of the two parts of the belt tangent to it, and that the scale-beam weighs the same difference of tensions of the same parts tangent to the pulleys *W* and *W*.

The scale-beam was graduated in 600 divisions of $\frac{1}{16}$ inch, each representing a half-pound with the travelling poise used. On this poise is a small beam graduated in hundredths, so that the small poise upon the small beam is capable of weighing $\frac{1}{200}$ of a pound when the machine is in motion. The more rapid the motion the more delicately can the weighing be accomplished.

In testing dynamo-electrical machines, the resistance measured being very uniform, it was only necessary that the belts used should be of even thickness and free from lumpy splicings, to get rid altogether of the tendency to dance which otherwise afflicts the beams of belt-dynamometers.

The fastest speed made by the dynamometer during the tests was 1700 revolutions per minute, which gave the belt a speed of $2\frac{1}{8}$ miles per minute. The fastest speed of any test was about 1400 revolutions (9240 feet of belt) per minute continued for ten consecutive hours, during which the belt ran over 1000 miles.

The centrifugal force tending to break the belt at this speed is about 1350 pounds on each part, but this force does not come on the journals or pulleys; it is confined to the belt itself, and stretches it until it becomes slack. The slack is taken up by screwing down either of the pulleys *S*, and when the machine slows or stops the belt *is* tight.

In getting the "friction" of the pulley *B*, after a test, the machine was run light at the same speed that it had run loaded during the test; thus comprehending in similar measure all sources of resistance whether from friction proper, bending and straightening the belt, or air-currents. The force required to bend and straighten the belt was sensibly affected by the temperature of the air.

Before the dynamo tests began it was observed that the air-currents, caused by the rapid movement of the belt, interfered with the functions of the scale-beam, and it was found necessary to place sheet-iron roofs over the upper pulleys. The lubrication is accomplished by an automatic feed.

A suitable counter is provided to register the number of revolutions, which latter can be observed to within a fraction of one revolution.

It is also provided with apparatus to record the power measured. This, however, was not used during the tests referred to, as direct weighing was found so convenient, and the results could be so quickly calculated. At the end of the scale-beam is a vertical rod attached below to an iron cylinder which floats in mercury in a cylindrical iron pipe. The beam being balanced, any force tending to raise it lifts the cylinder out of the mercury proportionally. This motion, multiplied by levers, is communicated to a pencil-point which moves vertically $\frac{1}{8}$ of an inch to the pound and records the weight upon a paper band moving horizontally one inch for every 100 revolutions and recording them. This automatic registration of weight is applied only to the fractions of weight between the even fifty pounds, the principal part of the weight being hung at the end of the scale-beam in the usual way.

By confining the registration to this small excess, it is registered on the large scale above mentioned. The method of calculating the H. P. is similar to that used in the smaller machine; the formula being

$$H.P. = \frac{PN}{10\,000}.$$

P is in half-pounds since, the delivery of belt per revolution of B is 6.6 feet. This, however, supposes a belt $\frac{21}{100}$ of an inch thick. A thicker belt requires a correction in accurate work.

Not the least interesting portion of the report of the committee referred to is that relating to the "Calibration of the Dynamometer." In order to prove whether or not the dynamometer measured correctly the power transmitted through it, it was used in the determination of the mechanical equivalent of heat on a large scale. The water-churn used was a cylinder, 3 feet diameter and 3 feet long, holding 1223 pounds of water. In the continuous method, devised by Professor Marks, the water entered the churn at nearly uniform temperature and left it at nearly uniform temperature, about 15.5° Centigrade higher than it entered. The operation continued for three hours. The first half-hour was occupied in bringing the exit-water to uniform temperature, when the experiment proper began and continued for two hours and a half, during which over five tons of water passed through the churn and was raised about 15.5° Centigrade by the continued exertion of about 46 horse-power.*

The result as calculated was:

Mechanical equivalent for 1° Centigrade.......	1391.05 foot-pounds.
" " " 1° Fahrenheit......	772.81 " "

Still another modification of the belt dynamometer —which has in its favor simplicity—is that shown in Fig. 42. Instead of employing a scale-beam with movable weights, the force is measured by difference in actual weight of the machine when at rest and when in motion—the driving side of the belt being on the lower idle pulley. In this case the dynamometer is placed upon an ordinary platform-scale, and the base filled with iron or other suitable material which will

*See article by W. P. Tatham in Journal of the Franklin Institute, Dec. 1885.

outweigh the pull of the belt. This is weighed (after the belt is put on ready for running), and when work is performed the resistance of the driven shaft tends to straighten out the belt, and thus to lift the weight in

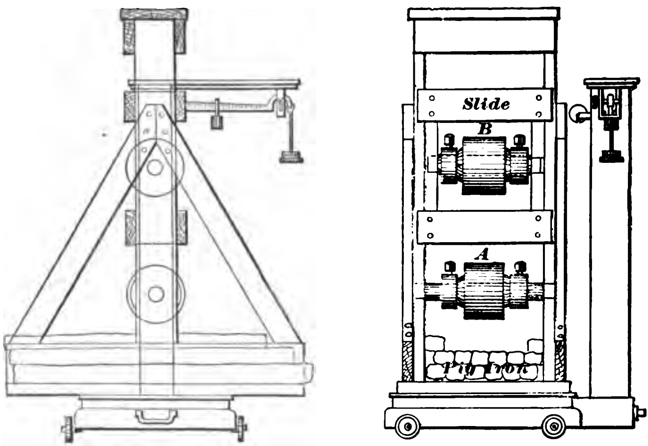


FIG. 42.

the base, so that, if weighed when the maximum resistance is reached, the difference in weight will equal W.

As before, the driving force is equal to $P = T_1 - T_2$, the difference in tension between the tight and slack sides of the belt.

From an inspection of the accompanying diagram, Fig. 43, it will be seen that as the pulley B is free to move up and down, the angle β will be less the greater the tension in T_1, for the greater the tension in T_1 the less (proportionally) there will be in T_2, and in consequence the weight of B will cause the slide to which B is attached to drop.

As the tension in T_1 is equal to the weight acting at

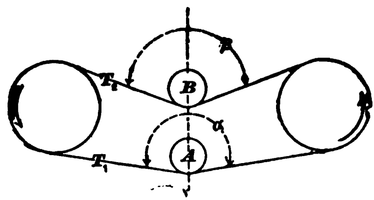


FIG. 43.

A divided by twice the cosine of the angle which it makes with the vertical, or, in the present case, with $\frac{\alpha}{2}$, we have

$$T_1 = \frac{W}{2\cos\frac{\alpha}{2}}.$$

If $\frac{\beta}{2}$ equal the angle which T_2 makes with the vertical, we have in like manner

$$T_2 = \frac{w}{2\cos\frac{\beta}{2}};$$

where w equals the weight of the slide and is constant. The weight W acting at A is the difference in weight of the machine when at rest and when work is performed. The angles α and β are variable, their magnitude depending upon the load passing through the belts. A convenient method of obtaining these angles is by the use of a jointed gauge free to open to any desired angle. This gauge consists of two thin strips of metal or wood hinged at one end, and provided with a clamp-screw or thumb-nut; by placing the gauge parallel to the edge of the belt the angle α, or β, made by the belt can readily be obtained by adjusting the legs of the gauge to correspond to the angle of the belt; by transferring this angle to paper its magnitude may be measured by means of a protractor.

A belt-dynamometer designed by Messrs. Geo. Wales and F. M. Leavitt and built under the direction of Prof. Jas. E. Denton, in 1883, for the use of the Chicago Railroad Exhibit Committee appointed to test dynamometers, is shown in Fig. 44.

This apparatus was designed to make an autographic portable dynamometer on the belt-angle principle, using the angle of the belt as the primary element of force measured. The belt could be drawn to any angle by a wrench applied to a chain-winding pulley *G*, and the ratio of the belt-angle to the difference of tension was given by the gauge *a*; that is, the reading of the point *d* on the semicircular scale gave a constant which, multiplied into the height of the pencil on the paper-drum *h* and into the scale of the spring *L*,—which was variable over a large range by sliding the spring and arm *N* along the levers *H* and *I*,—gave the differences of tension.

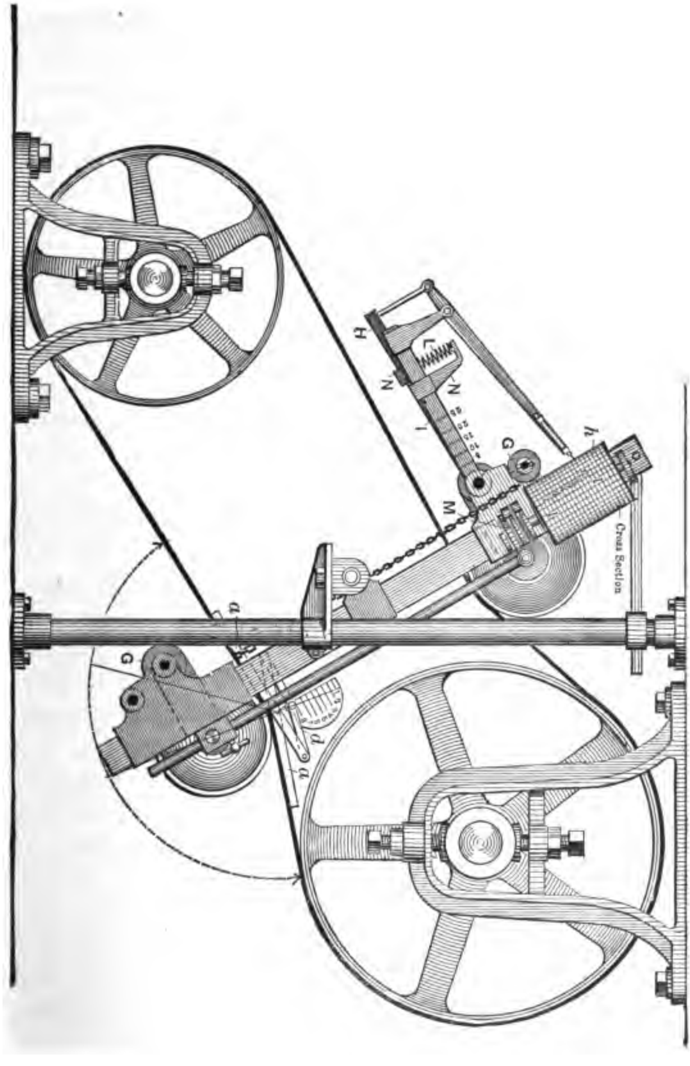


FIG. 44.—REGISTERING BELT-DYNAMOMETER.

The length of paper revolved in a given time gave the space moved through by the belt. The train of differential gearing *M*, driving the paper-drum, gave a wide range of speeds to the drum. The instrument was successfully applied to the measurement of power on a 200 incandescent-light dynamo; and with a tight belt 7 inches wide which did not violently vibrate, a very perfect trace, under varying loads, could be obtained on the paper by the use of a flexible pencil-point.

The time was indicated each minute by perforating the paper by an electric spark.

The disadvantage in using this apparatus and, in fact, most forms of belt-dynamometer, is the tendency of the belt to produce excessive vibration, thus causing very irregular readings.

For small machines which can be mounted on skids or other supports, and placed on a pair of platform-scales, the driving power can be obtained directly from the difference in weight when at rest and when performing work, provided the driving shaft be placed vertically over the driven shaft of the machine: in this case a dash-pot connected with the scale-beam would be an advantage in obtaining steadiness of readings.

As previously shown (page 56), the power transmitted to a machine, or given out by a motor, can be determined by supporting the machine upon trunnions and measuring its torque, or turning moment.

In the Brackett cradle-dynamometer the torque is determined by suitably mounting the machine to be tested upon a swinging platform suspended from knife-edges and supplied with a scale-beam and sliding weight; the tendency of the driving belt to rotate the machine may be weighed on the scale-beam, and will give a measure of the power.

This dynamometer in its modified and improved

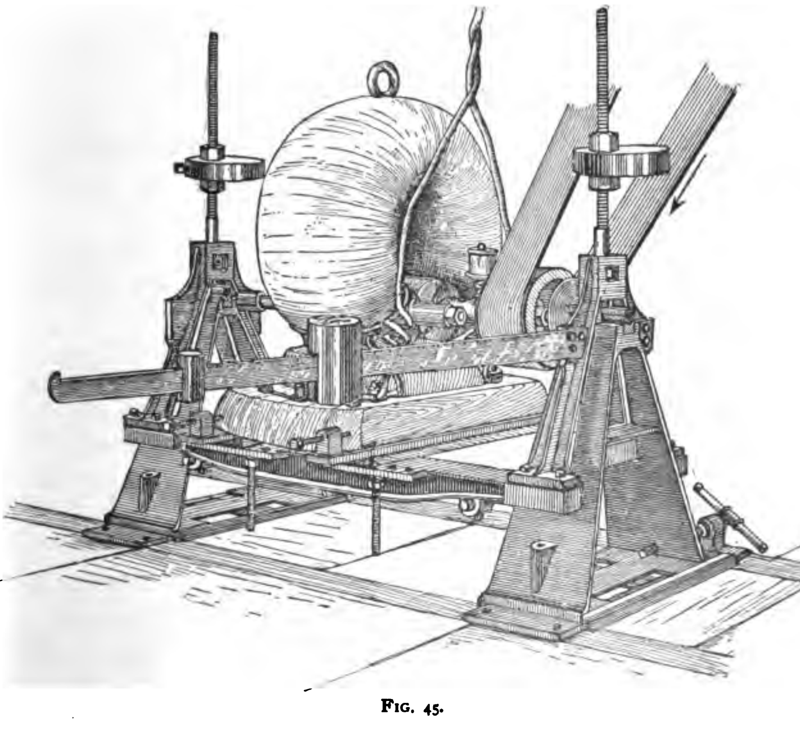


FIG. 45.

form as manufactured by Queen & Co., of Philadelphia, consists essentially of a strong stiff platform, Fig. 45, furnished with two rigid uprights in each of which is fixed a heavy steel knife-edge from which the platform is suspended. These knife-edges rest upon firm supports bolted to the floor and so constructed that a slight swinging motion is allowed to the platform upon which the machine to be tested is mounted.

To one of the swinging uprights, near the knife-edge, is fixed a graduated horizontal lever which carries a sliding weight. Adjusting screws are provided, by means of which the axis of rotation of the armature of any given machine may be made to coincide with the axis of oscillation of the cradle, viz., the line which passes through the two knife-edges. In this way machines of various makes and sizes can readily be put in position and their data determined. Provision is made to enable the experimenter to determine when this adjustment is secured by use of a circular plate of metal fixed to the inner end of each of the knife-edges, so that its centre coincides very closely with the axis of oscillation of the cradle; when, by means of the adjusting screws, the armature-shaft is made concentric with the circular plates, as determined by means of a gauge, the machine is in the proper position.

Counterweights are also provided which can easily be set so that the centre of gravity of the system, including the machine and the cradle together with its attachments, shall nearly coincide with the axis of oscillation, as is done in the common balance.

Suppose the dynamo-machine placed upon the cradle and the whole adjusted as above, and that we wish to determine the mechanical energy requisite to produce a given current. The machine being at rest with its driving belt off, the cradle is brought to equilibrium by means of the sliding weight, and the position of the latter is noted. The machine is then belted and driven at the proper speed, the circuit being closed to produce the required current. In consequence of the interaction between the armature and the field-magnets equilibrium will be destroyed, and the sliding weight must be moved to a new position in order to restore it. This done, the new position is noted. The difference between this and the former position is the length of the effective arm of the couple which acts against journal friction and the resistance of the armature to motion, due to the interaction between itself and the field-magnets.

If, as before, we represent the length of this effective arm in feet by L, the weight of sliding balance in pounds by P; the number of revolutions per minute by N, then the mechanical effect, W, in foot-pounds per minute absorbed by the dynamo or given out by the motor will be

$$W = 2\pi LNP,$$

an equation similar to that previously found in the discussion on friction-brakes. It must be noticed, however, that here the length of arm L is not the total horizontal distance from centre of sliding weight to centre of shaft (centre of suspension), as in the Prony and band brakes, but is the difference of lever-arms L_1 and L_2 measured when the machine is in open circuit (L_1) and when the circuit is closed (L_2), so that $L = L_2 - L_1$.

It is obvious that the cradle-dynamometer can be used to measure the work absorbed by any machine which can be conveniently mounted on the swinging platform; the effective lever-arm, L, being obtained by subtracting the arm L_1, obtained by running the machine light, from L_2, obtained when the machine is performing useful work.

In any case the mechanical horse-power can be obtained by dividing W by 33 000.

Besides the energy required to turn the armature journals in their bearings and to produce the current, some is necessarily spent in producing disturbance in the air about the armature. The amount may be determined by means of the cradle-dynamometer if desired, and, when found, if it be added to that determined as above, the total energy expended upon the machine will be known. The energy expended upon the air is in most machines very small, and may be neglected without serious error.

The manner of using the cradle-dynamometer to measure the energy developed by the dynamo-machine when used as a motor will immediately be obvious, since no new principle is involved.

A cradle-dynamometer designed for a capacity of from $\frac{1}{3}$ to 33 horse-power (250 to 25 000 watts) will weigh about 1200 lbs., and occupies a floor-space of 4 by 6 feet.

Since its first introduction by Prof. Brackett, the cradle-dynamometer has been largely used for measuring the power of dynamos and motors. Experience has shown that where the power of a single dynamo is concerned and a vertical driving belt can be used, this dynamometer is sufficiently accurate for all practical purposes, the variations of successive measurements being easily kept within a twentieth of a horse-power. We know of no successful attempt, however, to use a belt in any other direction, the difficulties being that the knife-edges are liable to slip sideways on their supports, and that the pull of the belt causes a deflection of the whole apparatus, which interferes with the adjustments. In fact, the machine should be driven from beneath and not from above, to be able to make the latter easily and satisfactorily.

To set the axis of the machine in line with the knife-edges a special device is used in the Experimental Laboratory of the Stevens Institute, and is regarded as the most reliable method of adjustment in use. If an error is made of one-thousandth of a foot sideways in setting the machine and the pull of the belt is, say, 500 pounds, a substitution of these quantities, with 1000 revolutions per minute, in the formula shows that an error of one-tenth of a horse-power will be caused thereby. The device mentioned—first used by Prof. J. E. Denton in 1883, when tests were made for the Chicago Railway Commission—consists of two equal weights hanging from each end of a piece of belt which is hung over the pulley of the machine. The weights together should, preferably, be about equal to the pull of the belt. The machine is adjusted so that hanging these weights on causes no change in the position of the scale-beam, and any irregularity in the pulley is eliminated by trying it with the pulley in positions 180° apart.

The next apparatus to be described is the Floating Dynamometer, the invention of Prof. J. Burkitt Webb of Stevens Institute.

This dynamometer, Figs. 46 and 47, consists of the approximately square *tank A*, containing water or some heavier liquid, as brine. In this is floated the nearly square *caisson C*, which is a negative tank, i.e., one having its water-tight surface outside. The tank rests either

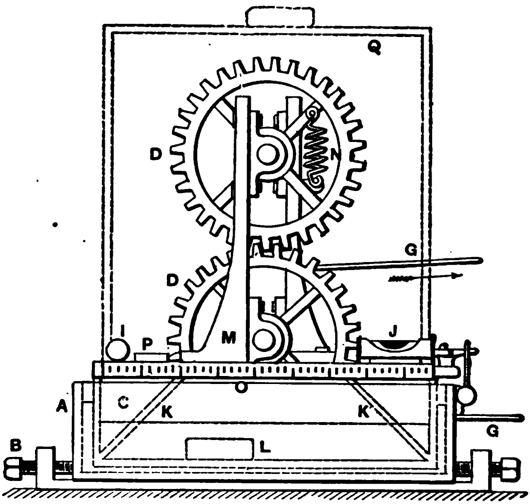


FIG. 46.—WEBB'S FLOATING DYNAMOMETER—ELEVATION.

on the floor or upon *skids*, *B*, with screws for adjusting the tank a small distance sideways. On the caisson is mounted any suitable machine whose plus or minus consumption of power is to be measured. The figure shows four gear-wheels mounted in such a way that their running-friction may be measured in an exact manner. The power absorbed or developed by other rotating machines—as steam-engines, electric motors, dynamos—may thus be measured.

The caisson, together with whatever is mounted upon it, is termed the *float*, and the weight of liquid displaced by it must of course be equal to its own weight. In addition to this, the proportions of the caisson must be such as to bring its *metacentre* very near to the *centre of gravity* of the float.

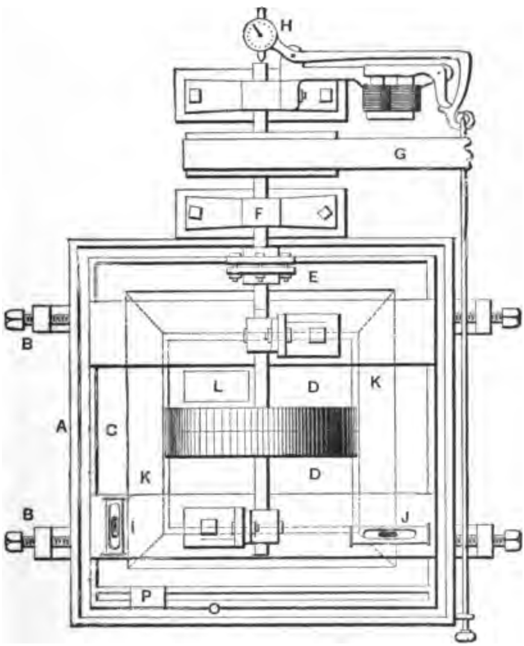


FIG. 47.—WEBB'S FLOATING DYNAMOMETER—GROUND PLAN.

The machine to be tested (i.e., the combination of gear-wheels *D*, or other machine) is driven, by a *coupling*, *E*, from the *countershaft* *F*. The latter sustains the pull and vibrations of the driving belt *G* and transmits to the machine a pure moment only. The connecting coupling *E* is of a special design, and has the property of acting as a semi-rigid universal joint. As a universal joint it allows the float to deviate from its normal position, when forced to do so, without injury to the apparatus, while its rigidity acts automatically to keep the machine-shaft in line with the countershaft.

The countershaft is furnished with a *speed-counter*, *H*, mounted upon a lever, which can be operated from the front of the dynamometer to throw the counter in and out of gear. This lever is furnished with a special *electric-lock*, which holds it securely in or out of gear, except at regular intervals when the electric circuit is broken by a clock and the lever unlocked, by a spring not shown in the figure. The period of time during which the counter is operating must therefore be an integral number of these intervals; in default of some such arrangement for securing a high degree of precision in measuring the speed, this part of the work would be considerably less precise than that of measuring the moment. The electric-lock may be incorporated in the construction of the counter, or a "card" may be run from the countershaft and intervals of time marked upon it automatically, thus recording the exact number of revolutions during each interval.

For measuring the driving moment the caisson is furnished with two adjustable levels, a *shaft-level*, *I*, and a *moment-level*, *J*, and with a *scale*, *O*, and sliding weight, or *pee*, *P*. The shaft-level is set so as to show when the machine-shaft is level, and therefore in the same horizontal plane as the horizontal countershaft; a deviation therefrom being corrected by varying the quantity of water in the tank. This adjustment requires no great accuracy. The moderately sensitive moment-level, J, is set to a central position when the coupling is disconnected and the P at one end of the scale, then, when the machine is running, it is brought again to the centre by shifting the P until the caisson assumes its former position. The product of the number of pounds of the P multiplied by the difference in feet between its two positions gives the driving moment, which, when multiplied, as usual, by 6.2832 times the number of turns per minute and divided by 33 000, gives the horse-power. Putting L for the number of feet between the two positions of the P, and N for the number of revolutions of the shaft per minute, we have as before

$$\text{Horse-power} = \frac{2\pi LNP}{33\,000}.$$

It is customary to make the weight of P such that the horse-power per hundred or per thousand revolutions can be read directly from the scale. Thus the scale may be divided into feet and decimals of a foot and P may be made equal to $33 \div 2\pi$, or about $5\frac{1}{4}$ lbs.; then every hundredth of a foot that P is moved means a thousandth of a horse-power for every hundred revolutions.

When the machine is not heavy enough to sink the caisson to its full depth, only enough water is put in the tank to just float the caisson, so as to have but a thin sheet of water between the caisson and tank bottoms; this acts as a dash-pot and prevents troublesome oscillations. The exact levelling of the float is completed by means of some loose *ballast*, L, and the stability is reduced, to the small amount required, by raising some of the ballast to the top of the caisson or to a (removable) *shelf*, *Q*, over the machine. See Fig. 46.

For dynamometers of large size the use of heavy ballast can be avoided by the *stability-trough K*. This is a trough of triangular section running entirely round the inside of the caisson and partly filled with the same liquid as is in the tank; by putting more liquid in the trough the stability is decreased. The machine should be put on in the first place at about the right height; the higher it is the less the stability. The right height can be determined by a simple calculation of the centre of gravity of the float, but this is not necessary, for the machine may be at once mounted and blocked up until the float shows signs of instability, when the final adjustment can be made with the ballast. The countershaft should then be blocked up to about the same height. Instead of setting a light machine high on the caisson to secure small stability, loose ballast can be put at once on the shelf *Q* until the proper stability is obtained.

By a suitable arrangement of mechanism upon the caisson power may be transmitted from one machine to another, both standing upon the floor, and the machine becomes a transmitting-dynamometer.

A very high degree of precision can be attained with this dynamometer. In fact, as there is practically no friction in the liquid to interfere with the action of the caisson, almost any degree of precision may be reached. Experience has shown that in close comparative tests of machines, or in measuring their friction or air-resistance, there is no other dynamometer with which the (almost unavoidable) accidental errors may not cover up or reverse the results. The particular machine illustrated in the figures as mounted for the purpose of being tested will serve as an illustration of this fact. In attempting to measure the friction of gearing it has been customary to measure the power supplied to the gears and that received from them, and to take the difference of these two quantities as the amount lost in friction; but the unavoidable error in measuring these two quantities by ordinary means is such as to introduce much uncertainty into the value obtained for the friction, and in many cases to render it entirely valueless.

In explanation of the particular method illustrated for measuring the friction of gears, it may be further explained that the lower shaft has two gears fast to it, while on the upper shaft the forward gear is free to turn on the shaft. The upper gears are connected by an adjustable spring, *N*, by means of which the loose wheel is powerfully rotated so as to bring the teeth of the upper and lower wheels in contact, with a known and adjustable pressure. By this arrangement it must be evident, upon examination, that the horse-power or energy transmitted by the gears is carried around in a circuit only through the gears themselves, and does not at all embarrass the direct measurement of the loss due to friction. The gear at the back, or countershaft side, on the lower shaft drives the gear above it, communicating to it a certain horse-power, dependent upon the velocity of the teeth and the pressure between them. This gear drives the one in front of it on the upper shaft, by means of the spring *N*, and then this gear drives the gear beneath it, thus returning the horse-power to the lower shaft, less the loss by friction. The upper shaft is adjustable on the standards *M*.

The dynamometer, therefore, is called upon to measure the friction only, and no such reliable determination of frictional losses can be made by a measurement of gross and net horse-powers, where the small quantity lost must be obtained as the difference between the relatively large gross and net quantities. This *differential method* of measuring friction was first published by Professor Webb in the Transactions of the American Society of Mechanical Engineers.*

In measuring the friction and internal air-resistance of any machine no special precautions are needed; the machine is simply run for that purpose and the measurement made in the same way as any other measurements of the power absorbed by the machine. It is, however, to be noted that, in attempting to do this with a dynamo, the residual magnetism will cause a waste of power in Foucault currents, which will be included in and may invalidate the measured result. This residual magnetism, however, may be nearly eliminated by means of a current from a battery, or from another machine, which is passed through the field and successively reversed and reduced by means of a rheostat. There is no way of separating the internal air-resistance from the friction, except to get rid of it by running the machine in vacuo.

If it be the external air-resistance or "fanning" that

* Vol. IX. p. 213.

is in question, the machine is to be run as a motor, the belt having been removed, and the measurement is to be made in the same way as before, but with a degree of care suited to the small quantity to be determined.

A dynamometer for measuring 40 to 50 H. P. occupies, without the countershaft, a floor-space of about ninety inches square, and may, if desired, be built beneath the floor, so as to have the general appearance and convenience of platform-scales.

In the experiments of Hartig a dynamometer was used in which, by means of a series of gears, the rotating force is made to act upon a pair of springs, one of which is furnished with a pencil which describes a curve as a roll of paper is caused to move before it. The principle of action will be understood from the following, which is abstracted from Weisbach's Mechanics.*

To the interior of the wheel CA, Fig. 48, upon which the rotating force P acts at A, is bolted an annular gear which engages at D and D_1 with two equal gears, DE and D_1E, both of which act upon a third gear, EE. This last gear revolves freely upon the shaft C of the wheel DD_1, and is firmly attached to the drum BC upon which the resistance Q acts, while the other two gears, DE and D_1E, have their axes supported by a lever, FCF_1, which revolves freely about C. On the hub of this lever is a band, one end of which is fastened to the dynamometer-springs H H, which latter are bolted at M to the floor. We see that here the rotat-

* Dubois' translation, vol. II. part I.

ing force P is held in equilibrium by two forces, R and $-R$, that out of these last arises a couple, $-R,R$, which holds the force of resistance Q in equilibrium, and that therefore the forces $2R$ and $-2R$ act at F and F_1 and stretch the springs H H with a certain force Z.

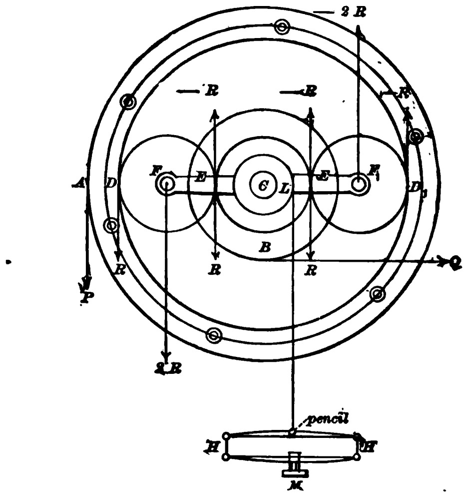


FIG. 48.—ARRANGEMENT USED BY HARTIG.

Let $a =$ lever-arm CA of the force P;
$b =$ lever-arm CB of the resistance Q;
$r =$ radius CD of large annular gear;
$r_1 =$ radius CE of centre gear-wheel, and hence
$\dfrac{r - r_1}{2} =$ radius FD of intermediate gear,
$c =$ lever-arm of the force $Z = CL$.

Then we shall have

$$Pa - Rr = Rr, \quad \text{or} \quad Pa = 2Rr;$$

also,

$$Qb = 2Rr_1, \quad \text{and} \quad Zc - Qb = Pa.$$

Substituting above values for Qb and Pa, we have

$$Zc = 2R(r + r_1);$$

hence

$$\frac{Pa}{Qb} = \frac{2Rr}{2Rr_1}, \qquad \text{or} \quad \frac{P}{Q} = \frac{b}{a} \times \frac{r}{r_1},$$

and

$$\frac{Pa}{Zc} = \frac{2Rr}{2R(r + r_1)}, \quad \text{or} \quad \frac{P}{Z} = \frac{c}{a} \times \frac{r}{r + r_1}$$

A transmission-dynamometer which has been used with very satisfactory results is that shown in Figs. 48 *a*,

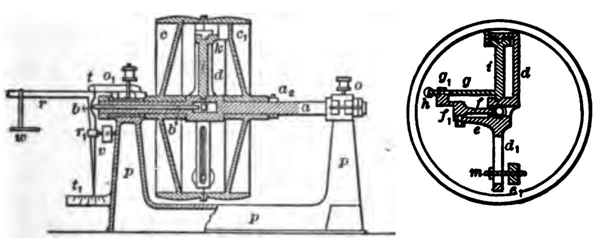


FIG. 48*a*.

b, *c*, and *d*. It was due largely to this piece of apparatus that certain power tests of generators and electric motors were made at the Frankfort Electrical Exhibition in 1891.*

This machine, designed by Mr. Fischinger, consists of a continuous, partly hollow shaft *a*, Fig. 48*a*, which runs in the journals o o_1 of the frame p. The three arms d d_1 e

* Electrical World, 1892, vol. XIX. page 400.

form one piece with the shaft and stand out from it at right angles. The smaller shaft i with the lever-arm g has one bearing in the enlarged portion of the main shaft, its upper end running on a pin of the arm d. This upper end of the shaft i carries a lever of the first class k. The two arms of the latter engage with two projections, nn_1 and vv_1, cast on the inside faces of the pulleys c and c_1, Fig. 48*b*. On a pin at the extreme end of arm e turns the double lever ff_1. A pin at the end of arm f_1 runs in a slot g_1 of the lever-arm g. The outer end of this arm g carries two threaded pins with adjusting nuts h h_1. To avoid sudden shocks when the dynamometer is set in motion these two adjusting nuts strike against two rubber cushions on the inside faces of the two pulleys. The arm d has a long slot in which runs the bolt m carrying a weight e'.

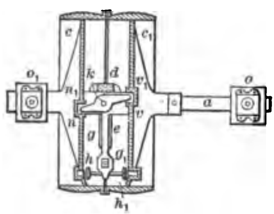


FIG. 48*b*.

By a proper adjustment of this weight the centre of gravity of the whole lever system can be brought to coincidence with the centre of the main shaft aa.

A rod b runs through the centre of the hollow part of shaft a. This rod b pushes with one end against the end of lever-arm f, while the other end touches the arm r_1 of lever r_1r. An adjustable weight v serves to balance the weight of arm r. The lever r_1r turns around the pin t; the arm r_1, which acts as an index finger, moves over a scale t_1 at the lower end of the frame.

The whole system of levers is enclosed between the two movable pulleys c c_1; it is, however, very easily accessible by loosening the collar a_2 and shifting the pulley c_1 toward o.

The action of this dynamometer is as follows: The power is transmitted from the power generator by a belt to the pulley c_1, Fig. 48*c*, by means of the lever kk_1

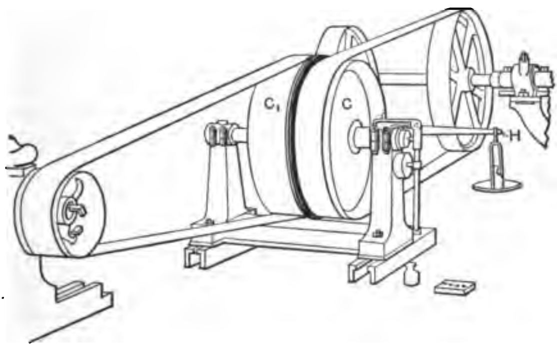


FIG. 48*c*.—FISCHINGER'S DYNAMOMETER.

to the pulley c, and from this pulley by another belt to the machine under test. The rotary motion is therefore transmitted from one pulley to the other by the lever kk_1 and shaft i. But since the two pulleys are loose on the shaft they will tend to shift their position relatively to each other. This amount of shifting is communicated by the lever kk_1 to shaft and lever-arm g_1, thence to the double lever ff_1. The end of lever ff_1 pushes against the rod b, and this in turn against arm r_1 of lever rr_1. The tendency to shift between the two pulleys is therefore indicated by the index r_1 on the scale t_1.

In order to find the force which tries to turn the lever kk_1 about its fulcrum i, weights are placed in the scale-pan at the end of lever-arm r until the index r_1

returns to zero, in which case the system is in equilibrium. The levers have such position relative to each other that the forces transmitted through them all act perpendicularly to the arms, thus allowing the relations between them to be easily calculated. The ratio of the weights on the scale-pan to the pull on the circumference of the pulley depends upon the ratio of the lever-arms. This ratio is made 10 to 1; the pull on the circumference is therefore 10 times the weight in the scale-pan. It must be taken into consideration, however, that in this case the weight in the scale-pan will indicate the total work done, including that which it takes to run the dynamometer without a load. The weight corresponding to this latter amount of work will, of course, differ for different velocities, and can very easily be determined by throwing off the belt from the driven pulley and by balancing the system when running at the new velocity which is desired for a certain test. By making records of the different balancing weights for different velocities this can be kept in the form of a table for future reference. The diagram Fig. 48*d* shows the system of levers which can easily be measured with great accuracy. To determine the proper point of suspension of the scale-pan, the lengths of the different lever-arms are accurately measured, and the point x found by calculation.

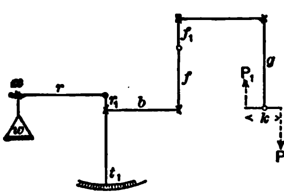


FIG. 48*d*.

For a convenient calculation of the work done or horse-power transmitted, the pulleys are given such diameters that their circumferences are expressed by even numbers. The following table gives some of the data of five different types of the dynamometer:

	I.	II.	III.	IV.	V.
Circumference of pulleys in metres.	1.5	2	3	3.5	4
Maximum load on scale-pan in kilogr.	6.0	12.0	22.0	50.0	75.0
Max. number of revolutions p m....	1200	900	600	510	450
Max. number of h. p. at a circumferential velocity of 30 metres per second	24	48	88	200	300

In this form of dynamometer great sensitiveness is obtained by proper balancing and minimum friction in the lever system, accomplished by the use of knife-edges wherever possible.

A very commendable feature is its large range for any given size. This is shown in the following results of tests with a No. III dynamometer, which measures up to 88 horse-power maximum:

Current strength in amperes.	E M.F. in volts.	No. of revolutions of dynamo.	Velocity of belt per second $= v$.	Pull on belt in kilog. $= k$.	Horse-power h. p. $= \frac{v.\ k.}{75}$.
215	65	830	15.1	122	24.6
158	68.5	850	15.5	96.5	19.9
103	70.5	870	15.75	71.0	14.9
31	74.0	875	16.1	34.5	7.4
0	75.0	885	16.25	17.5	3.8
0	2.5	890	16.25	6.	1.3
245	63.0	840	15.75	14.1	29.5
0	67.0	835	15.25	14.5	2.95

Among other forms of dynamometer not already discussed is the Emerson Power-scale—an instrument which is connected directly to the revolving shaft without the interposition of belts, except that used to drive the shaft itself. The machine in principle is a rotary scale, and its construction closely resembles the well-known Fairbanks platform-scales. This dynamometer is largely used in cotton-mills to determine the power consumed by the individual machines, and when used with care forms an excellent instrument for the purpose, being self-contained and readily applied. In this machine, the pulley which receives the power is loose on the shaft, and is connected with the latter by means of a spider which is keyed to the shaft, the hub of the spider forming one of the guides to the position of the pulley (not shown in the figure). Around this spider is a rim free to rotate, and from which studs project and serve to connect it to the pulley. In transmitting power from the shaft to the loosened pulley the tendency of the rim to rotate on the spider is resisted by a system of levers which communicate with a pendulum balance-weight. A dash-pot filled with oil is connected to the long lever and chain-rod to prevent unnecessary oscillations of the pendulum. These instruments are made in halves, so that they may be readily applied without disarranging pulleys or line-shafting.

The cotton-mill scale shown in Fig. 49 is fitted with special clutch and split bushings to fit shafts varying from $\frac{3}{4}$ inch to $1\frac{1}{4}$ inches, being secured in position by nut *B*. In this form of scale two sets of prime levers, *K K*, are used, so as to operate without change when running in either direction. Two studs, one of which is shown at *C*, are used to connect the loose driving pulley with the spider which is keyed to the shaft. These

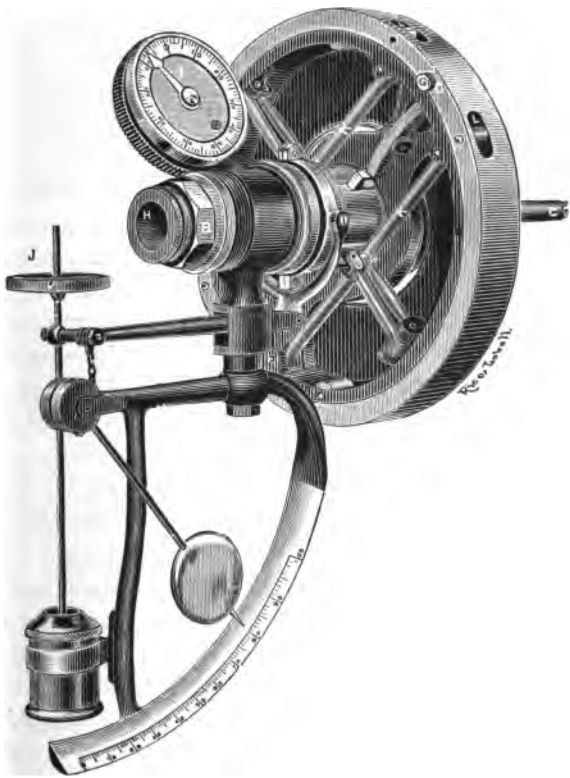


FIG. 49.—EMERSON POWER-SCALE.—One fourth size.

studs are screwed into a plate with projecting lug which drives the spider by means of the pin *G* in the rim. When the slide *H* is pushed in as shown in the figure, the stop *G* is thrown out of gear with the loose plate, and this latter is free to revolve on the hub of the spider, being driven by the loose pulley.

These scales are constructed so that the pivots in the ends of the levers at *L* describe a circle whose circumference is two feet, and the quadrants are graduated to read pounds; if the graduations are insufficient, weights may be added at *J*, the leverage of the scales being such that an actual weight of one pound placed at *J* has the effect of fifty pounds on the quadrant.

In the larger power-scales the centre of pivots of the prime levers (*K*) is always taken at such a distance from the centre that the distance passed through in one revolution is equal to a given number of feet. Thus in the scale designed to weigh 65 horse-power, the greatest diameter of the machine is 38 inches and the space passed through by pivot *L* in one revolution is 9 feet.

To ascertain the number of horse-power by means of an Emerson power-scale it is first necessary to find the centrifugal force of the unbalanced moving parts of the scale. This is obtained by running the belt on the tight pulley, the loose plate being disconnected from the spider, then note the reading as shown by the position of the pendulum on the quadrant.

This amount will be small for slow speeds, and below a certain minimum speed will be zero; but as it varies with the square of the number of revolutions, it should in every case be determined at the same velocity at which the total force is determined. In a test by Mr. Channing Whitaker to determine the effect of a cotton-mill scale it was found that at a speed of 416 revolutions per minute the reading was one-half pound, but at the speed of 1000 revolutions per minute it amounted to thirty-six pounds.

Having ascertained the amount to be deducted for a given speed, which is in fact equivalent to balancing the scale, we can find the horse-power developed from our general formula $\frac{PV}{33\,000} = H.P.$

If $F =$ total pounds indicated on the quadrant,
$f =$ pounds necessary to balance at given speed,
$N =$ number of revolutions per minute,
$C =$ path in feet of end of lever (K),
then

$$F - f = P, \quad \text{and} \quad N \times C = V,$$

which substituted in above formula will give net horse-power. The observed data of a test with a cotton-mill scale was as follows:

The gross indicated force $= 83$ pounds; the tare or balancing force $= 23$ pounds; revolutions per minute $= 791$. The path of end of lever being 2 feet, we obtain

$$\frac{(83 - 23)\,791 \times 2}{33\,000} = 2.87 \text{ horse-power.}$$

Another form of shaft-dynamometer is the Power-meter which has recently been patented by Mr. Franklin Van Winkle. This is a rotary transmitting-dynamometer which is especially adapted for adjustment to any shaft or pulley for measuring power transmitted by a shaft to a pulley, or *vice versa*, in this respect resembling the Emerson power-scale.

Helical pull-springs are employed for weighing the amount of force transmitted from the driving to the driven portion of the dynamometer.

Figs. 50–58 will illustrate the construction and application of this dynamometer.

Figs. 50–55 are illustrative more particularly of the "light portable" style. The construction and opera-

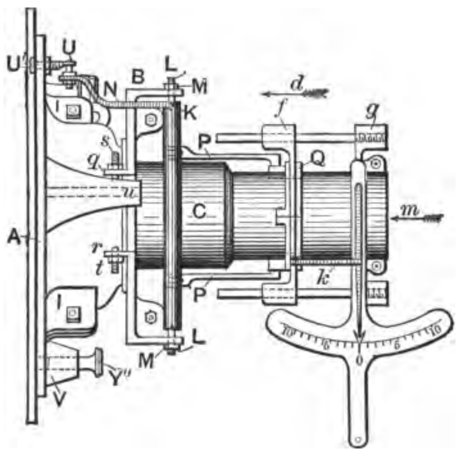


Fig. 50.

tion of all other styles will, however, be understood from this description, as it embraces the features of the others.

Similar letters refer to similar parts throughout the several views.

To facilitate the application of the dynamometer to a shaft, the main framework and all parts which surround the shaft are made in halves, in order that the dynamometer may be mounted on the shaft in the manner of a split or separable pulley.

Split bushings are used for reducing the bore on any shaft smaller than the hole in the hub. For bushing machines employing four springs it is immaterial whether or not the machine is concentric with the shaft; hence rough wood-bushings may be employed.

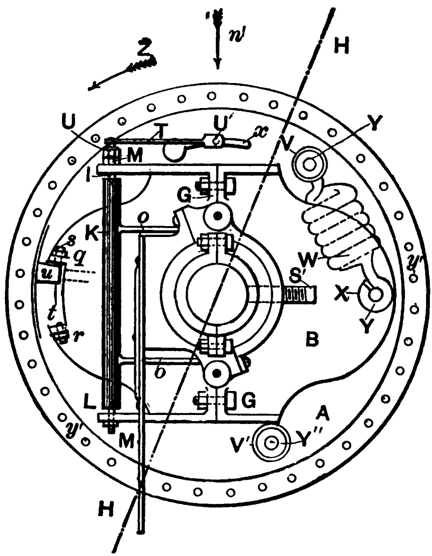


FIG. 51.

The main hubs of all machines are chambered in the middle of their length, in order to leave room for cord to be tied around the outside of wood bushings or lagging of any kind that is convenient for building up the size of the shaft. Thin manila drawing-paper wrapped around the shaft answers the purpose admirably.

The main framework consists in an elliptical plate

B, the outline of which is best shown in Fig. 51. This plate has a central hub *C*, prolonged on one side, with the grooved collar *e* near its end; this hub projects a short distance to the other side of the plate, with a spherical exterior surface *D* (see Figs. 52 and 53) terminating in the plain collar *E*, the plate and hubs being made in halves and held together by bolts passing through the projecting lugs

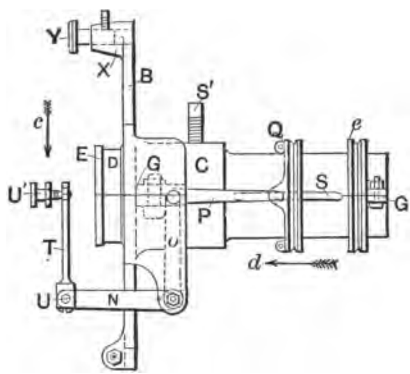


FIG. 52.

G G. The central hollow hub of this main framework is recessed along a portion of its prolonged end and bored out in the remaining portions of its length to receive the shaft upon which it may be placed, as shown in Fig. 53.

A is a circular plate the middle portion of which is dished or crowning. This plate is made in halves coming together along the line *HH*, Fig. 51, and held together by bolts passing through projecting lugs *I I*, Fig. 50, the plate being provided with a short central hollow hub which is bored out to fit loosely around the spherical portion *D* of the central hub of the main framework.

K is a rock-shaft the ends of which have countersunk recesses, by means of which it is mounted on conically-pointed screws *L L*, which pass through lugs projecting from *B* and are held firmly in place by lock-nuts *M M*. *N* is an arm on *K* projecting toward the plate *A*, and *o* and *o* are parallel arms projecting from *K* at right angles to *N*, one over each side of the hub *C*.

P P are links connecting by pivotal screws the ends of the parallel arms *o o* to the grooved collar *Q*, which, being made in halves, encircles the reduced portion of hub *C*, being free to slide along *C*, and provided with feathers which project into the slots *S*, Fig. 52.

T is a connecting-rod having spherical socket-ends with detachable caps.

U and *U'* are spherical or ball-ended stud-bolts set in the end of the arm *N*, and in the curved slot *x* in the plate *A*, respectively, and connected by the connecting-rod *T* (see Fig. 51). When the plate *A*, rock-shaft *K*, and collar *Q* are mounted on the framework of the dynamometer and connected as described, then any change of relative angular position between the plates *A* and *B* around the axis of hub *C* will cause *Q* to move along the hub, the direction and degree of travel being dependent upon the relative direction and degree of motion between the two plates *A* and *B*.

V V' are bosses on one side of the plate *A*, projecting toward *B*. *X* is a similar boss on the plate *B*, projecting toward the plate *A*.

W, Fig. 51, is a helical pull-spring connecting the plates *A* and *B*. The material of the helix forming *W* is turned up in eyes at both ends, through which the suspending pins *Y* and *Y′* pass, the pins being held in place by set-screws, as shown in Fig. 53. If the plate *A* be rotated on its axis in the direction indicated by

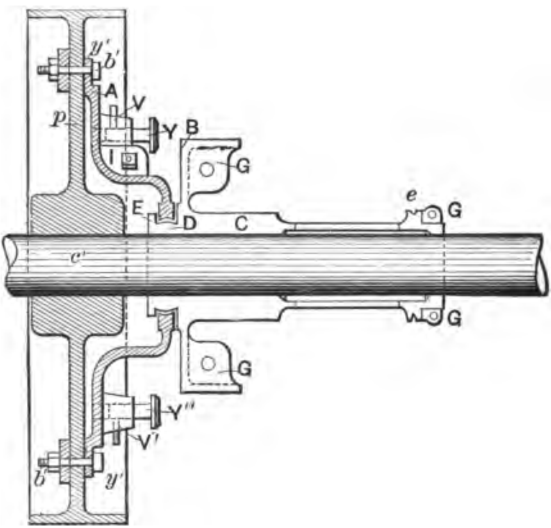


FIG. 53.

the arrow *Z*, Fig. 51, any resistance offered to such rotation by *B* will be transmitted through *W* to *A*, causing *W* to elongate, and thereby permitting *A* to advance in its relative angular position with respect to *B* in direction of the arrow *Z* until the resistance offered by *B* is overcome by *W*. Then *B* follows along in the rotation primarily imparted to *A*. The direction of arrow *c* in Figs. 52 and 54 is the same with respect to *B* as arrow *Z* in Fig. 51. Consequently, when *A* advances in rotation with respect to *B* in the manner described and the ball-stud *U'* is fixed in the slot *x* of the plate *A*, then *U'*, *T*, and *U* and the end of the arm *N* are carried in direction of arrow *c*, imparting a partial revolution of the shaft *K* on its axis, resulting finally in movement of the parallel arms *o*, links *P*, and collar Q in direction of arrow *d*. If, during such rotation of *A*, the resistance offered by *B* lessens, then the spring, by reason of transmitting a lesser strain, resiles to such length as may correspond to the reduced resistance and carries *B* forward in the direction of rotation toward its original position with respect to *A*. Each particular degree of resistance transmitted by the spring from one plate to the other produces its particular position of the sliding collar *Q* on the hub, and will be indicated upon the scale by the pointer *n*, as follows:

In Fig. 50 *f* and *g* are rings made in halves, the interior surfaces of which are suitably bevelled for fitting loosely around the grooves in the sliding collar *Q* and fixed collar *e*. These rings are each provided with two bosses for the purpose of receiving the guide-rods *i i*, one end of each being screwed into *g*, while *f* is free to slide over the remaining portion of the rods. The ring *f* has a projection *j*, see Figs. 54 and 55, to which is pivoted the link *k*, and the ring *g* has projecting from it the scale-plate *l*, the lower portion of which is made in form of a segmental arc, on which is laid off a scale.

In Fig. 54 *n* is a pointer-hand pivoted at *p* and connected by the link *k* to the ring *f*, its free end being carried over the scale in accordance with the motion of the ring *f*, in the sliding collar *Q* along the hub-

When the hub is in rotation, the scale-rings f and g may be prevented from rotating with the hub, and caused to remain stationary by holding the downward. extending portion of the scale-plate in the hand, or by securing the lower portion of the scale-plate by twine or otherwise to a stationary object. The position assumed by the pointer-hand on the scale may be noted while the hub is in rotation.

If the plate B receives the primary rotation instead of A, but in a direction opposite to that indicated by the arrow Z, Fig. 51, and such rotation be resisted by the plate A, then the spring W will be similarly elongated, and n will be carried over the scale in the same manner.

The periphery of the plate B has cut out of it a gap or notch bounded by projecting lugs q and r, through which pass the set-screws s and t. u is the stop-bracket projecting from the plate A through the gap in B. When the spring in its normal length and without any strain upon it connects A and B, as previously described, then A is to be turned past B sufficiently to take up any lost motion between the suspending pins and the plates A and B or between the suspending pins and the eyes of the spring. The screw s is then to be set down and secured in contact with u. With u and s thus in contact the dynamometer may be driven backward without injury to or derangement of its parts. When driven in the direction which tends to elongate the spring, the maximum relative motion between the two plates and consequently the maximum elongation of the spring are both limited by u coming in contact with the end of the screw *t*. If, however, it be desired to measure resistance transmitted between the plates when the relative directions of rotation are opposite to those described, then in order that such resistance may be transmitted in a manner tending to elongate *W* it is necessary for *W* to be connected from the projecting hub *X* of the plate *B* to the projecting hub *V'* of the plate *A* by means of the suspending pins *Y* and *Y''* When the spring is connected without strain, as shown in Fig. 51, the proportions of the dynamometer are such that the distance from *X* to *V'* is greater than the distance from *X* to *V* by such an amount that in order to connect *W* without strain from *X* to *V'* it is first necessary to rotate *A* around *B* in the direction of arrow *Z* a sufficient distance to bring the side of the stop-bracket *u* against the end of *t*. *t* is then in position to operate as a backward stop, while *s* becomes the forward stop. When *W* is thus changed about, the partial rotation of *A* past *B*, which is incidental thereto, results in carrying the pointer-hand to a point to the left of the zero of the scale—that is to say, in direction of the arrow *d*. The pointer-hand may be returned to zero by loosening the ball-stud *U'* in the slot *x* of the plate *A* and moving it along the slot to such position that *n* again indicates zero, in which position *U'* may be again secured to *A*. When thus adjusted, any resistance to rotation between the plates *A* and *B*, causing the spring to elongate, will cause *n* to assume a position to the right of the zero of the scale.

Successive positions which the end of the pointer-hand *n* will assume on the arc of the scale-plate for different numbers of horse-powers or foot-pounds per minute transmitted from A to B or B to A, employing a given spring, may be determined for a given speed of rotation of the dynamometer, inasmuch as the degree of elongation of the spring is ascertainable for any degree of resistance to rotation which the plate A may offer to the plate B, or *vice versa*. Thus if the distance from centre of shaft to centre of the suspending pins of the spring W be 5 inches, and the elongation of the spring be 1 inch under a pull of 200 pounds, the horse-power transmitted at 100 revolutions per minute for an elongation of half an inch will be

$$H.P. = \frac{2\pi RNP}{33000} = \frac{2\pi \times 5 \times 100 \times 100}{33\,000} = 9.52.$$

For a given extension of the spring, which represents a corresponding force P, the pointer-hand will assume a definite position, and if the lever-arms of the instrument be suitably proportioned, the arc may be so graduated that for a given spring the successive divisions will represent horse-powers and decimals for a fixed number of revolutions per minute. Fig. 50 illustrates the appearance of a divided and figured scale laid off on l in the manner described for a stated speed of rotation of the dynamometer—as, for instance, one hundred revolutions per minute—using always the same spring. In dynamometers heretofore made this has been the only type of scale provided, and when such a scale is used at any other speed of rotation, or when any change is made in the spring employed different from those for which the scale is especially constructed, then, in order to arrive at the true number of horse-powers, the operator must make calculations for every reading.

To obviate this the Van Winkle dynamometer is provided with a "differential" scale-plate by which the horse-power for varying speeds is indicated directly

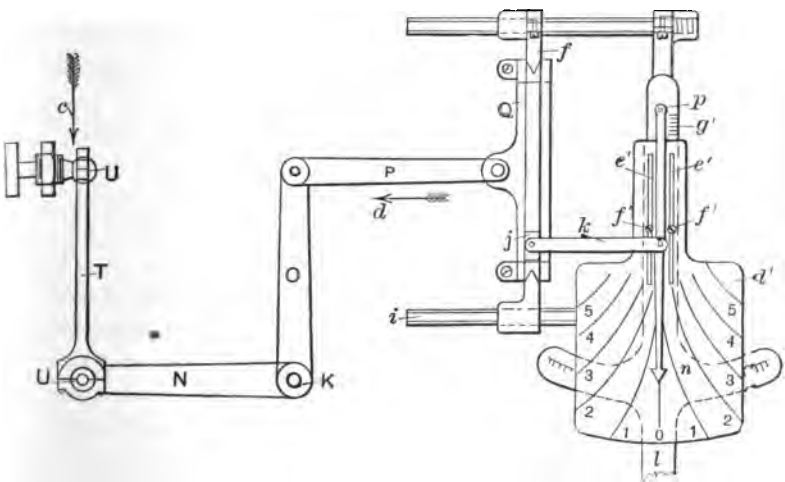


FIG. 54—DIFFERENTIAL SCALE-PLATE.

by the pointer, within the limits of the spring used. This consists of a flat spade-shaped plate *d'* of form shown in Fig. 54, the narrow upper portion of which has parallel slotted openings *e'*. The differential plate when used is to be laid upon the face of *l* under the pointer-hand, as shown, and may be secured in different positions of vertical adjustment by means of screws *f' f'*, which pass through the slots into the upper part of *l*. When using always the same spring or equal springs, arbitrarily-spaced lines, each representative of a speed of rotation, are made upon the face of the upper portion of *l* in form of a scale *g'*, and so laid off that the upper edge of *d'* may be brought opposite to any division on *g'*. As shown in Fig. 54, the face of *d'* is laid off with a central zero-line, to the right and left of which are curved lines marked "1, 2, 3, 4," etc. These "differential curves" are of such form that when the upper edge of *d'* is set opposite to that division of *g'* corresponding to any given speed of rotation made by the dynamometer, then the end of the pointer-hand will be on the curve marked "1" when one horse-power is being transmitted, on curve "2" for two horse-powers, "3" for three horse-powers, etc., to the right or left of the zero-line, according to the direction of resistance for which the spring and pointer-hand may have been adjusted.

When the dynamometer is always to be used at a constant speed of rotation and for the purpose of greater or less sensitiveness of action, different strengths of springs are employed at different times. Then in a similar manner the differential scale-plate may be laid off in curves to be used for indicating different horse-powers, the divisions of the scale *g'* being then taken as representative of different strengths of springs instead of different speeds of rotation.

For springs offering different degrees of resistance to elongation, each of which may be used in the dynamometer at different speeds of rotation of the latter, the same general form of differential scale-plate is employed in conjunction with the compounding scale-plate *j'*, Fig. 55. The scale-divisions *g'* are then discarded, excepting a single division-line *h'*, which, for greater explicitness of location, is marked with an arrow-head. By means of the binding-screw *k'*, which passes through the slotted projecting portion of the

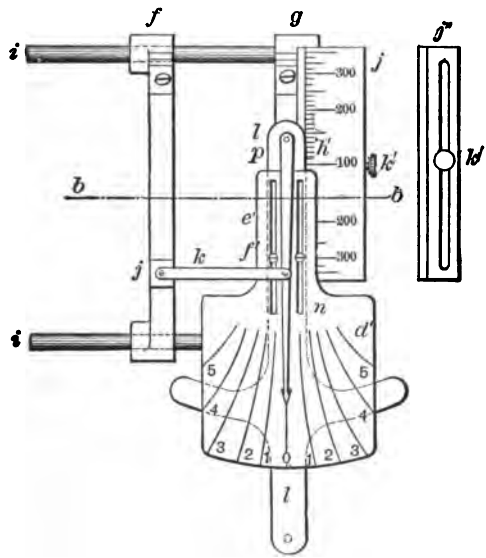


FIG. 55—COMPOUNDING SCALE-PLATE.

back of *j'*, the latter may besecured in different vertical adjustments with reference to *l*, so that any one of the division-lines drawn on the face of *j'* may be opposite to *h'*.

Beginning at a certain division on *j'*, as, for example, that marked "100," and proceeding upward, lines are drawn each of which is representative of the number of pounds required to elongate different springs

1 inch, as one hundred, two hundred, three hundred, etc.; and similarly beginning at "100" on j' and proceeding downward, lines are drawn representative of different speeds of rotation of the dynamometer, as, for instance, one hundred, two hundred, three hundred, etc., revolutions per minute.

The spacings of the divisions of the compounding scale-plate j' and the curves on the differential scale-plate, when used in conjunction with the compounding scale, are such that the pointer-hand will indicate the correct number of horse-powers on the differential scale-plate provided j' is so adjusted that the division representing the strength of spring employed is opposite the division-line h' marked with an arrow-head, and the upper edge of d' is fixed opposite to that division on j' representing the speed of rotation.

In order to apply the dynamometer, as, for instance, for the purpose of measuring the horse-power taken by a pulley p', Fig. 53, from line-shaft c', the pulley is first loosened from the shaft by removing set-screws, keys, or other means of fastening, making of it a loose pulley. A short distance one side or the other of the pulley the plate B is mounted on the shaft, the halves being secured together by the bolts through lugs G. Then the rock-shaft K and its connections, the scale-ring and scale employed, and finally the plate A are all mounted on B. The weighing spring W is then connected according to the direction of motion, and the ball-stud U' is set in the slot x, so as to bring the pointer n to indicate zero when the stop-bracket u is against the end of the screw s or t for a back-stop. The dynamometer is then moved along the shaft until the periphery of the plate *A* comes against the arms of the pulley. The periphery of *A* has a projecting surface *y′*, which is perforated all around by holes, as shown in Fig. 51. The plate *A* is secured to the pulley by means of bolts and straps which pass over the arms. The plate *B* is then secured to the shaft by the set-screw *S′* (shown in Figs. 51 and 52). When the shaft is set in motion, any power taken from it by the pulley for driving other machinery by means of a belt will be transmitted from the plate *B* to the pulley through the spring *W*, resulting in a greater or less elongation of *W*, and consequent movement of the pointer-hand to a position on the scale-plate employed, and the latter, as previously noted, may be held stationary or secured to a stationary object to prevent its turning with the dynamometer.

It will be noticed that the performance of the weighing springs is transmitted by positive mechanism to the pointer-hand on the dial; no wrapping connectors of any kind being employed.

The Light Portable style weighs, complete, 62½ lbs., and, as intimated in the description, it is supplied with springs of different degrees of sensitiveness, up to a capacity of 20 horse-power at 100 revolutions per minute.

Another form of the Van Winkle dynamometer has two or more springs similarly connected in a series. The pins for suspending the weighing springs are the same distance apart and at a uniform distance from the axis of the dynamometer. One of the springs operates the same as the single spring employed in the Light Portable style, taking up the load from the beginning.

This is called the initial weighing spring. The remaining springs have looped eyes at one end, the loops being of such different lengths that after the initial spring has been loaded to a portion of its capacity the

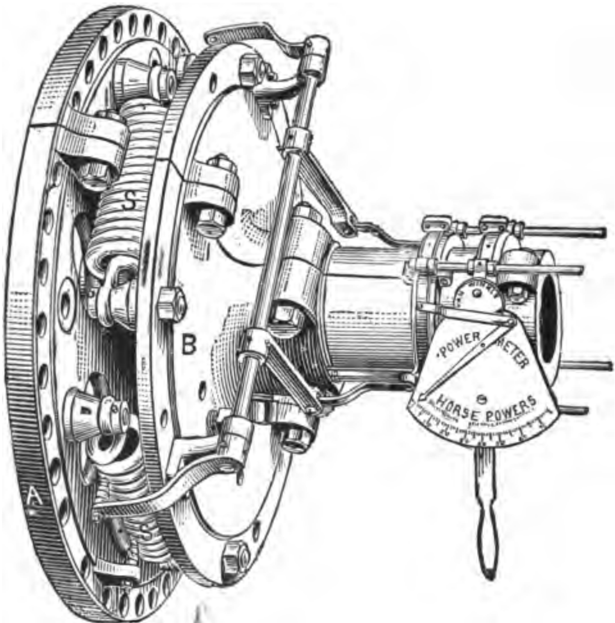


FIG. 56.—VAN WINKLE DYNAMOMETER.

looped springs assist the initial, drawing on their suspending pins, one after another.

This system of suspending the weighing springs greatly facilitates accuracy in construction, application, and use of the dynamometer.

The holes for the suspending pins being equally spaced, the springs are interchangeable in location; and the amplitude of swing, or change of relative angular position between the plates, being dependent upon the resistance of only one, or as many springs as may be necessary for transmitting any load, the divisions of the zero end of the scale are coarser than if all springs started to pull at the beginning, and admit of smaller fractional sub-division.

Still another form is shown in Figs. 56, 57, and 58. in which *A* is the pulley-plate and *B* the plate which i

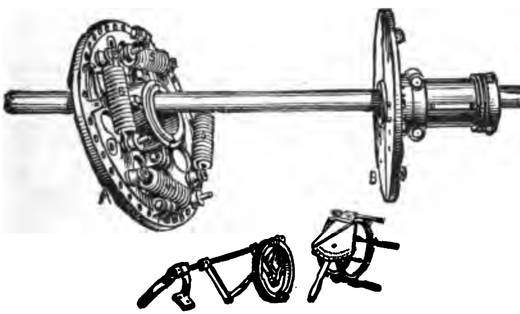


FIG. 57.

secured to the shaft. It differs from the Light Portable style in requiring the connecting-rod, which operates the rock-shaft, to be changed to the other side of the machine, in order to use the dynamometer in a reverse direction: and the weighing springs being suspended from stud-bolts, the springs, together with their stud-bolts, must be removed from the plates *A* and *B*, and the studs inserted each in the opposite plate in holes provided for the purpose.

Fig. 57 illustrates the parts of the "Standard Portable" dynamometer, and Fig. 58 illustrates it applied to a shaft and pulley.

It is claimed by the manufacturer that these dynamometers are only about one-half the weight of other types for equal capacities. The dynamometers have been tested after several years' use and the weighing springs have been found to retain their original strength.

No allowance for centrifugal, frictional, or other

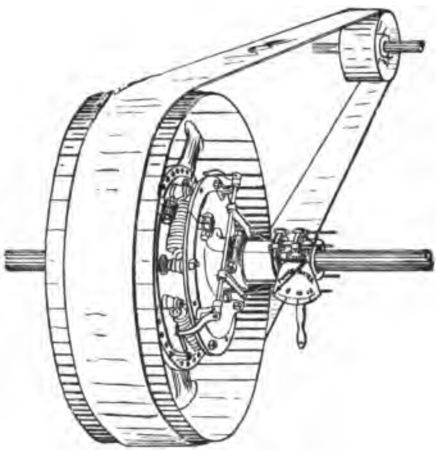


FIG. 58—VAN WINKLE DYNAMOMETER APPLIED TO SHAFT.

error is made in using this instrument. The weighing springs are so proportioned that, as has been proven in tests of the dynamometer, the indications of the scale are unaffected by centrifugal disturbance beyond the highest speeds of shaft in practice.

The transmitting mechanism is so proportioned and balanced that any tendency to centrifugal disturbance is avoided.

The only sources of frictional disturbance are in the pivotal joints of the rock-shaft motion and the friction due to the weight of the scale. When, however, the dynamometer is employed for measuring the amount of power which a pulley receives from a shaft, then this small amount of friction is not thrown on the weighing springs.

There is no friction between the pulley and the shaft, nor between that portion of the dynamometer which is secured to the pulley and the rest of the instrument. This, at first sight, may not appear to be the case; tests of the dynamometer show, however, that no friction exists in cases where the pulley when loosened from the shaft is loose enough to be turned on the shaft by hand. The explanation furnished for the absence of friction is, that the belt drawing the pulley always in the same direction, accompanied by rotation of both shaft and pulley, any change of load permits of the pulley assuming an appropriate angular position with respect to the shaft by rolling on the shaft; consequently, in the standardizing and operation of these dynamometers centrifugal force and friction are neglected because they can exert no appreciable influence; the actual resistance which the springs offer for various degrees of elongation being the one thing which is taken into account in calibrating any scale.

Different sizes of these dynamometers have been employed in the measurement of power required by mills, tenants, and machines, requiring from a fraction of a horse-power up to several hundred horse-powers, and under the widest range of circumstances.

The facility with which they may be applied, and their precision in indication of the lighest to the heaviest loads, have earned for them a prominent place among dynamometers manufactured for general use.

Two Van Winkle dynamometers furnished to a firm in Antofagasta, Chili, respectively of 450 and of 600 horse-power capacity, at 120 revolutions per minute transmit the power of two 9-inch shafts. They are believed to be the most powerful rotary transmitting-dynamometers, of any type, ever constructed.

While investigating the subject of power transmission as applied to milling-machines, the writer constructed an apparatus shown in Fig. 59 by which he

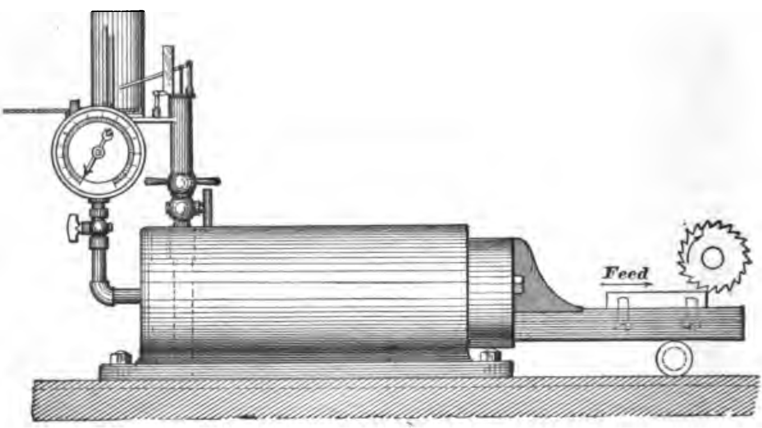


FIG. 59.

proposed to measure the magnitude of the force exerted by the teeth of the cutter, but the results were not wholly satisfactory when applied to a milling-machine. Used on a planer, however, a measure of the useful work was readily obtained from the card taken from the indicator attached.

Prof. L. P. Breckenridge, of the University of Illinois, had previously made some interesting experiments with a similar apparatus for determining the pressure exerted by a drill working under similar conditions, and later had very successfully applied the apparatus to planer tools. (See *American Machinist*, August 14, 1890.) The action will be understood by an inspection of the figure. The thrust of the tool acts upon the plunger of the cylinder, thereby forcing the contained oil into the pressure-gauge and into the cylinder of the indicator. By a suitable arrangement of cords, the drum of the indicator is made to revolve synchronously with the stroke of the tool or with the work; and as the pencil is forced upwards by the pressure exerted at the point of the tool, it will be seen that a measure of the work performed can be obtained from the card. The gauge is simply a check on the indicator. It is evident that the total work performed cannot be obtained by this means, as the force required to drive the machine itself is disregarded.

To obtain the total work, and at the same time the useful effect, the plan was adopted of mounting the cylinder upon a rotating pulley and forcing the oil through the centre of the shaft, but in order to balance the pulleys two cylinders were used as shown in Fig. 60. To maintain the lever-arm constant, the cylinder through which the transmitting force acts should not be bolted rigidly to the pulley-arm, but should be pivoted in such a manner as to obtain a constant lever-arm. The action of the Flather dynamometer is this:

The pulley L, which receives the driving belt, is loose on the shaft and free to turn within certain limits. F, secured to the shaft, is belted to the machine to be

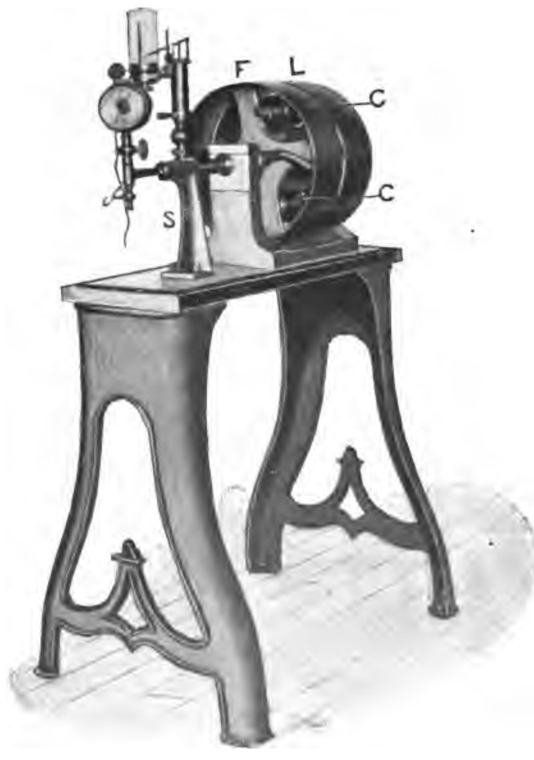


Fig. 60.—Flather Hydraulic Dynamometer.

tested, and carries a pair of cylinders, C, which are supported on trunnions; these cylinders are partially filled with oil, and connected to centre of shaft by a small flexible tube. The end of the shaft is bored out and is provided with a hollow steel tube free to revolve in the spindle and fitted with gland and stuffing-box nut to prevent leakage of oil. Connected with this steel tube is the stand *S*, carrying a pressure-gauge and an indicator. When motion is given to the pulley *L* it revolves through a small arc until a steel pin in its arm comes in contact with the plunger in cylinder *C*. If there is no resistance to be overcome, the indicator-pencil and gauge finger remain at zero; but as soon as resistance occurs the plunger is forced inwards. When the power overcomes the resistance, motion is communicated to the pulley *F*, and the machine is driven through the force transmitted by the oil.

As the plunger is forced inwards, the indicator-pencil and gauge-finger will in consequence rise, and the amount of rise will determine the pressure per square inch acting on the plunger. If the distance of the lever-arm acting on the plunger is known. the power can be readily ascertained from the formula

$$H.P. = \frac{P \times 2\pi rN}{12 \times 33\,000},$$

in which r equals radius in inches of path traversed, and N equals number of revolutions per minute. If the construction of the machine be such that r is constant, as in the present case, and equal to C, the formula becomes:

$$H.P. = \frac{0.0\,001\,904PNC}{12} = 0.00\,001\,586PNC.$$

In later machines the point of application of the force P was such that one revolution of the pulleys caused this point to move through a space equal to two feet; in which case

$$HP = \frac{2PN}{33000}.$$

In the experimental machine first constructed the pulleys were each 12 inches in diameter and 3½ inches face; the cylinders were 1.954 inches in diameter, presenting an area of three square inches. The plungers were of hard bronze, and were kept tight by leather cup-washers secured to the end as shown in Fig. 63. A 5-lb. spring was used in the indicator, as with stronger springs the cards obtained in some of the tests were not sufficiently large to show the small differences of power which it was desired to determine.

It would seem that the centrifugal force of the plunger would materially affect the true value of the force transmitted, but a careful examination of this force with varying lever-arms, corresponding to different positions of the plunger, shows that the actual effect is very small.

As the lever-arm of the driving force is constant, the centre of gravity of the plunger will have a varying arm dependent upon its position relative to the cylinder. This is shown in Fig. 63, in which G, G', and G'' are three positions of the centre of the plunger, the lever-arms of which are 3.67 inches, 3.87 inches, and 3.50 inches respectively—the radius of the driving force being constant and equal to 3.6 inches.

The centrifugal force, f, of the plunger can be calculated from the formula

$$f = 0.0\,000\,284\,WrN^2,$$

W being the weight in pounds, r the radius in inches, and N the number of revolutions per minute. In the case before us $W = 1.75$ pounds. If we assume N to equal 100, 150, 200, 250, we obtain for f the values shown in the following diagrams, Figs. 64, 65, 66.

As the centrifugal force acts along the radial line through the centre of gravity of the plunger, it will be seen that only the horizontal component can be considered as a force acting in the direction of motion. If we assume the average radius of the centre of gravity of the plunger (3.67 inches), and the average number of revolutions per minute to be 175 we find from Fig. 64 that the corresponding value of f is 5.7 pounds, as shown by dotted coördinates. If we decompose this force into its two components (see Fig. 63), we find the horizontal component is 1.37 pounds, the vertical being 5 pounds; now the vertical component produces a certain amount of friction against the walls of the cylinder which retards the motion of the plunger; if we take the coefficient of friction in this case to equal seven per cent, we obtain 0.35 pound for the friction which acts in the opposite direction to that of motion, hence $1.37 - .35 = 1.02$ pounds equals the effective component of the centrifugal force. As this acts on an area of 3 square inches, the effective component at the given speed is only 0.34 pound per square inch, which, if

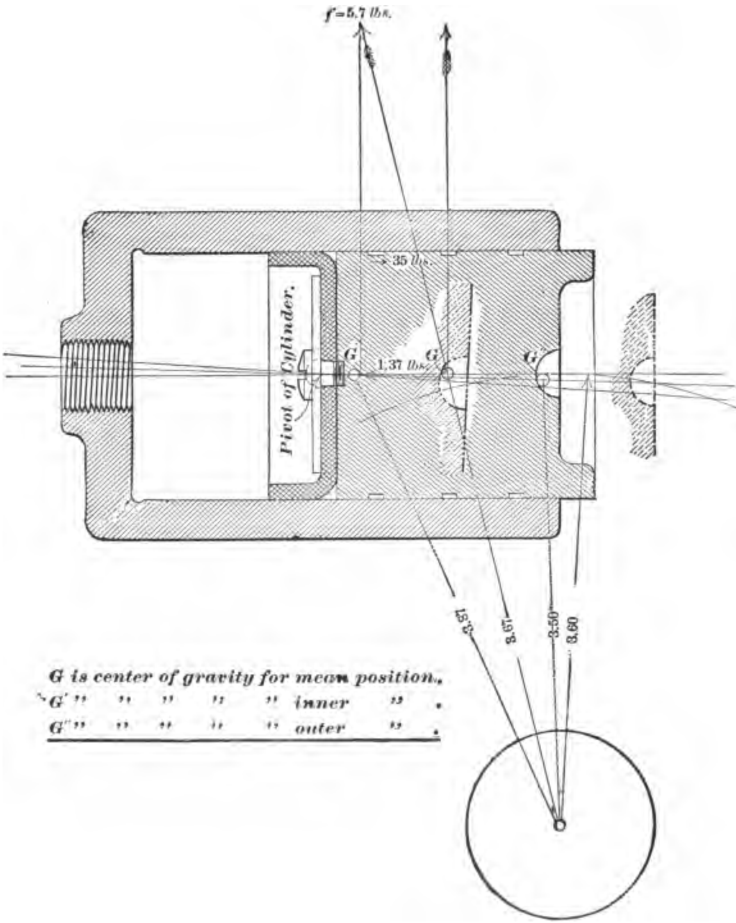

f=5.7 lbs.
35 lbs.
Pivot of Cylinder.
G
G'
G''
1.37 lbs.
3.57
3.67
3.50
3.60
G is center of gravity for mean position.
G' " " " " " inner " .
G'' " " " " " outer " .

FIG. 63.

there were no friction in the machine, would have to be deducted from the actual force as indicated on the pressure-gauge or card.

Experiments to ascertain the friction of the machine showed that 0.425 foot-pound, only, was necessary to overcome the friction when the machine was at rest with belts thrown off. With the sensitive spring used

FIG. 64.

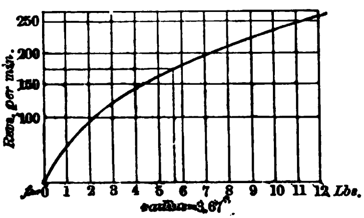
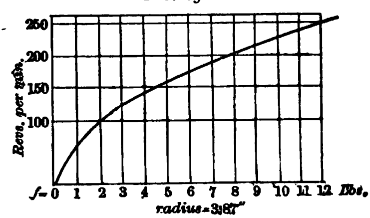


FIG. 65.



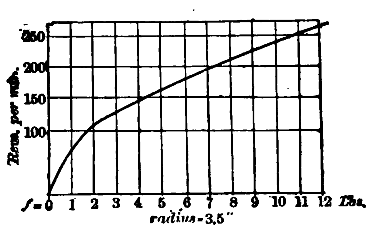


FIG. 66.

in the indicator very small changes in the forces were determinable. At the ordinary speeds at which the machine was run there was no appreciable difference in pressure, as shown by the zero line, whether the machine was stopped or running with the transmitting belt thrown off; with both belts on, however, a resistance at the plunger as small as three-fourths pound could readily be determined, as will be shown subsequently. From this it was concluded that for ordinary speeds the effect of the centrifugal force of the plunger was neutralized by the friction in the dynamometer; in any case it could be neglected by obtaining the zero line when running free at a given speed.

The small coiled spring outside of the cylinder, Fig. 60, connecting the arms of the loose pulley *L* with the bracket *b*, keeps the pin *p* in contact with the plunger; the action of this spring is to force the plunger into the cylinder, and thus raise the pressure on the gauge; this force is, however, counteracted by another spring inside the cylinder, which resists the inward motion of the plunger, yet acts with an equal force to keep the plunger in contact with the pin. When resistance is applied to the pulley *F* the plunger is forced into the cylinder until this resistance is overcome; the inner spring is thereby compressed and presents a resistance

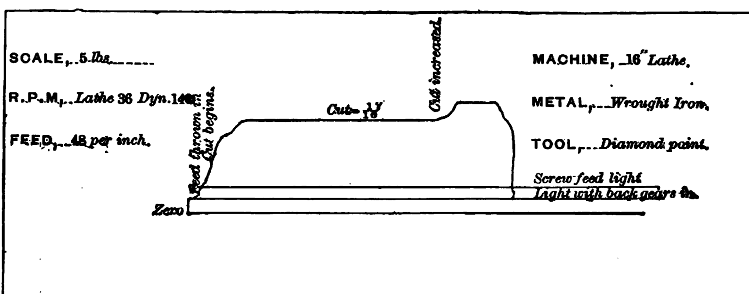


FIG. 67.—INDICATOR-CARD FROM HYDRAULIC DYNAMOMETER.

to the motion of the piston. As the spring is very light, and the compression seldom exceeds half an inch, it will be seen that this force of resistance is hardly appreciable.

An examination of an indicator-card, Fig. 67, from this dynamometer shows that the power required to drive a 16-inch Flather lathe with back gears in, well lubricated, and running light at 36 revolutions per minute, is

$$H.P. = \frac{PV}{33\,000}$$

$$= \frac{0.11 \times 2(5 \times 3 \square'') \times 2\pi \times 3.6'' \times 140}{33\,000 \times 12} = 0.026$$

$= 868.6$ foot-pounds,

where 0.11 equals the height of card in inches; 5 pounds equals the spring used; area of cylinder-piston equals 3 inches; radius of arm equals 3.6 inches; the revolutions of dynamometer being 140 per minute. With the screw feed in, still running light, the power

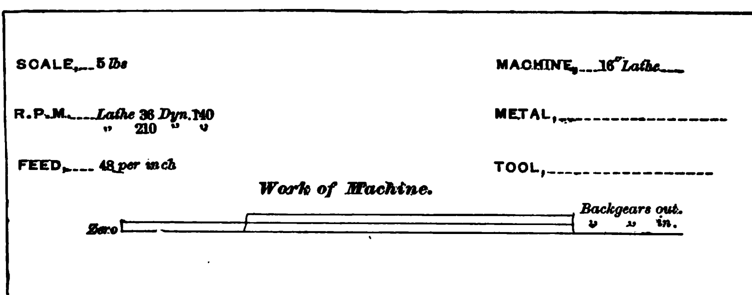


FIG. 68.—FRICTION-CARD FOR 16-INCH LATHE.

was found to be 0.040 H. P., or 1421 foot-pounds; with the load on, which was a light cut $\frac{1}{16}$ inch deep on a round bar of wrought-iron with diamond-pointed tool, the maximum power registered was 0.204 H. P., or 6948 foot-pounds.

An interesting result, shown in Fig. 68, was obtained on several cards, this being a greater amount of power used to drive the lathe running free without back-gears than under the same conditions with the back-gears thrown in.

A somewhat similar occurrence, repeatedly confirmed, was noticed by Mr. Wilfred Lewis in the course of some experiments with a 48-inch lathe.

The probable reason for this is that the work of friction in the spindle-brasses and other bearings is much less at the lower velocity. With the belt on any given step of a four-stepped cone-pulley the reduction of velocity in the main spindle-journals, when back-gears are thrown in, will be about nine to one, which reduces the work of friction very materially; the superior lubrication of the cone-pulley due to the revolving spindle also reduces its friction below that required to drive the spindle at the greater velocity without the back-gears, and, with the ratio of speeds as great as that ordinarily employed, this reduction in journal-friction more than compensates for the work spent in overcoming the resistance due to the gearing.

The machine just described enables one to use any make of indicator by attaching it to the stand *S*, but it was thought desirable to make the dynamometer self-contained and capable of giving a continuous record. With this object in view the writer designed his recording hydraulic dynamometer, a number of which are in successful operation.

In its latest form, Figs. 68 *a*, *b*, *c*, this machine consists essentially of three pulleys, 18 inches in diameter,

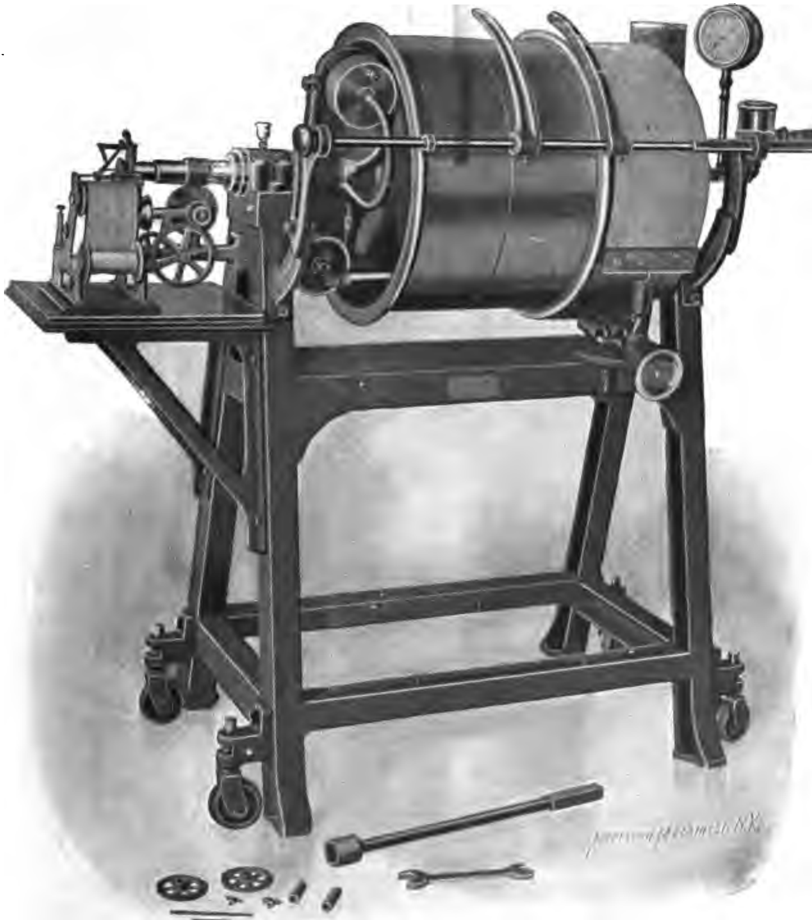


FIG. 68a.—PRESENT FORM OF FLATHER TRANSMISSION- AND ABSORPTION-DYNAMOMETER.

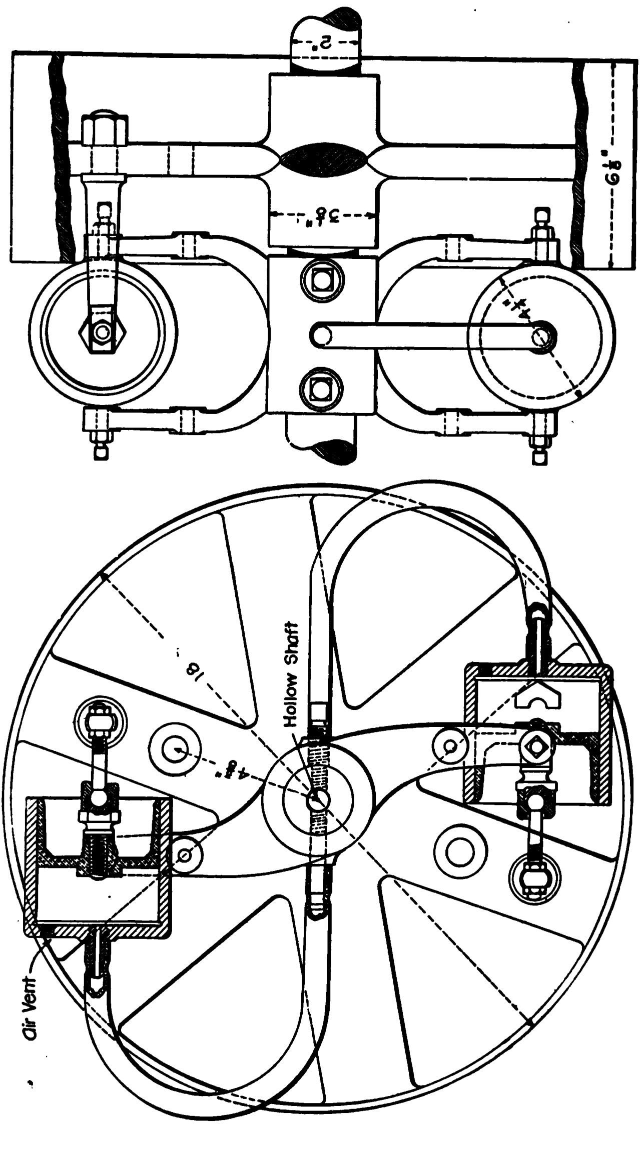


Fig. 68b.—Arrangement of Cylinders and Pulley.

carried by a hollow shaft which runs upon roller-bearings mounted on a stiff frame as shown.

The pulley on the right is fixed to the shaft and may be used for the belt which connects the dynamometer to the machine to be tested, or it may be used for a brake-band; the middle pulley is an idler and runs loose on the shaft; while the one to the left receives the driving-belt and is free to turn only within given limits. This latter pulley is connected to the shaft through the hydraulic cylinders and cylinder-carrier, shown in Fig. 68*b*.

The cylinders and shaft are filled with oil, so that any resistance to turning produces a pressure in the cylinders. This pressure is transmitted by the oil through the hollow shaft, and is recorded by the movement of an indicator-piston which is fitted into the end of the shaft; the pressure is also shown by the gauge at the opposite end.

In addition to the force curve traced by the indicator-pencil, the zero or datum line is traced by another pencil which is adjustable in position and may be located at pleasure on a line parallel with the axis of the shaft. Thus one is enabled to measure the total power required to run a given machine or any subdivided part of it.

Two sets of cylinders are provided with the machine; in one case the area of each cylinder is 10 inches, and in the other the area is 2½ inches. This permits a much wider range to the instrument and secures in all cases a suitable height of card. Each pair of cylinders is provided with bronze pistons that work without cup-leathers. These are actuated by steel plunger-pins which may be located in either of two positions, the inner for ordinary work and the outer for extra-heavy work or when the speed of rotation is slow.

Springs of various intensities are used to transmit the pressure from the shaft to the indicator-pencil,

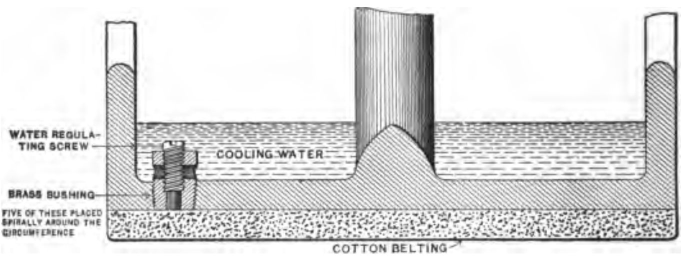


Fig. 68*c*.—Section of Pulley Rim.

depending upon the pressure recorded. These springs are changed as in any ordinary indicator. Adjustable casters permit the dynamometer to be moved about or to rest firmly on the floor, as desired. One of the interesting features about this machine is its adaptability to either transmit or absorb power.

The fixed pulley which is ordinarily used to connect the dynamometer to the machine to be driven has internal flanges for carrying the cooling water, and the rim is furnished with a number of adjustable bushings, as shown in Fig. 68*c*, by which the quantity of water admitted to the brake-band is readily controlled. The cooling water is supplied directly to the wheel from a small circular tank attached to the machine and provided with a regulating-valve. The brake-band is a heavy cotton belt, 7 inches wide, connected through two sets of tension-springs to the brake-regulating screw. It will be readily seen that the brake-

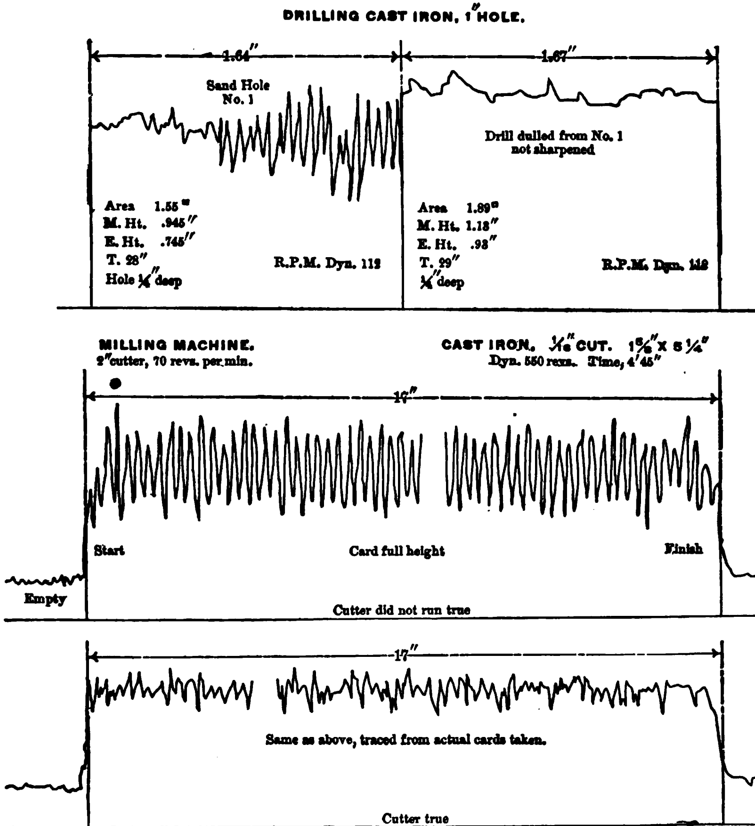


FIG. 68*d*.—DYNAMOMETER CARDS FROM DRILL PRESS AND MILLING MACHINE.

band may be quickly removed, thus converting the machine into a transmission-dynamometer. In either case an automatic record is traced upon a roll of paper which can be stopped or started at pleasure. The feed mechanism is provided with change-gears, so that three different speeds can be given to the paper roll, depending on the work to be done.

In Fig. 68*d* several diagrams from this dynamometer are presented which may be of interest.* It will

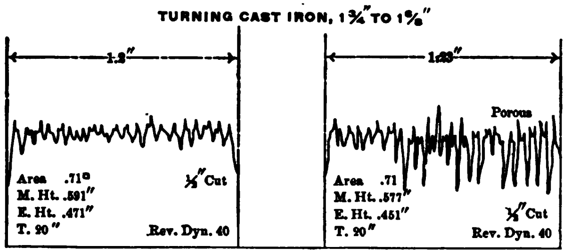


FIG. 68*e*.—DYNAMOMETER CARD FROM LATHE.

be noticed from an examination of these cards that the power required to drive the machine empty can readily be determined if desired, for all that is necessary is to run the machine at the same speed without any pressure on the cutting tool, and the resulting card will give a measure of the work. The same is true of the power required to operate the feed mechanism of the machine, or the dynamometer itself.

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* See article by Mr. J. D. Hoffman, Trans. A. S. M. E., vol. XVII. p. 471.

CHAPTER V.

MEASUREMENT OF WATER-POWER.

IN testing a hydraulic motor a friction-brake or other absorbing dynamometer applied to pulley on the driving-shaft, as already described, will give the power developed by the motor under the given conditions, but this power may be less than that which it is possible to attain, or which might be developed by the wheel when running at a greater or even a lesser speed: for if the velocity of the wheel be reduced to zero, there will be no power developed; and if, on the other hand, the speed be excessive, the water will flow through the motor, giving up but little of its energy to the wheel.

In making a test of a hydraulic motor, therefore, it will be necessary to find the available energy of the water which passes through the wheel in a unit of time, and also the power developed by the motor in the same time while running at different velocities and with different quantities of water.

A wheel may be working under conditions which will develop a maximum power, but the efficiency of the motor may not be so great as when developing a lesser power. The problem then presents itself to de termine the speed of wheel and quantity of water which will give the maximum amount of power; and secondly, to determine that speed and quantity of water which will give the maximum efficiency.

The efficiency of the motor in any case will be the ratio of the useful work performed, as determined by a dynamometer, to the theoretical or available work due to the energy of the water; that is,

$$\eta = \frac{P}{Wh},$$

where

η = efficiency;
P = effective work of the wheel in foot-pounds per unit time;
W = weight of water passing the wheel per unit time;
h = available head of water above the motor in feet.

If h is the total height of fall from upper level in head-race to lower level in tail-race, or if it is the difference in levels between reservoir and the discharge-pipe of the motor when the latter is supplied by a system of pipes, we shall obtain in $\eta = \frac{P}{Wh}$ an expression for the efficiency of the fall; but if h is only the height from the level of head-race to the motor, in the one case, and the effective pressure-head, as determined by a gauge in the supply-pipe at a point near the motor, in the other, then this expression will give the efficiency of the motor.

It will be apparent that to obtain the greatest efficiency of the fall, the wheel should be placed as near as possible to the level of the water in the tail-race; and that in the case of motors supplied by systems of piping, the latter should be arranged to reduce the available head as little as possible.

In determining the available energy of a fall of water, the most important and at the same time most difficult measurement to be made is that of the quantity of water delivered to the motor in a given time.

If the volume be small, the most reliable method of measurement is to weigh the water discharged into a tank or barrel placed upon platform-scales, as shown in Fig. 70, which represents an arrangement used at the Lehigh University for testing a small hydraulic motor.

For larger quantities the water discharged may be collected into a receiving-tank of known capacity, and its weight determined from its volume. Sometimes two tanks are employed, with an automatic arrangement by which each tank is filled and emptied alternately. A counter attached to the apparatus gives the number of times each tank has been filled. In ordinary tests the weight of a cubic foot of water may be assumed with sufficient accuracy at 62.5 pounds.

With the arrangement shown in Fig. 70 the quantity of water passing the wheel per minute was 428 pounds or 51 gallons under a pressure at the gauge of 65 pounds per square inch, the diameter of nozzle being $\frac{1}{2}$ inch.

The head corresponding to this pressure is 150 feet (see page 203); and since the weight of water passing the wheel per minute is known, the theoretical horse-power may be obtained from

$$H.P. = \frac{Wh}{33\,000} = \frac{428 \times 150}{33\,000} = 1.94.$$

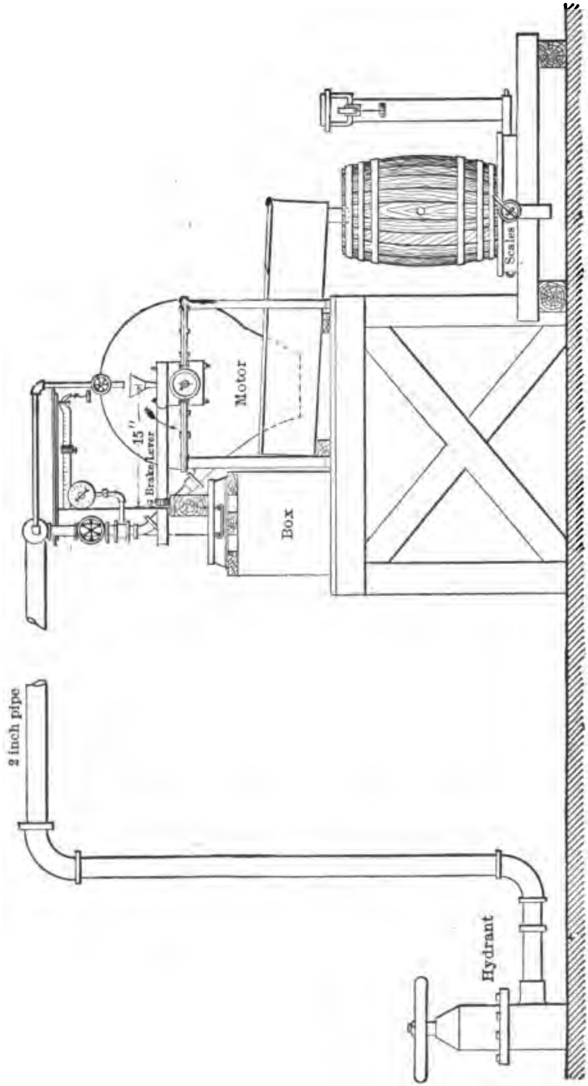

Fig. 70.—Arrangement used for Testing Small Hydraulic Motors.

With the motor running at 526 revolutions per minute and an unbalanced pressure of $6\frac{1}{4}$ pounds on the scales, the lever-arm of the brake being 15 inches, the brake horse-power is

$$B.H.P. = 0.0001\,904\,PRN$$
$$= 0.0001\,904 \times 6.25 \times \frac{15}{12} \times 526 = 0.78\,;$$

therefore the efficiency under the given condition was

$$\eta = \frac{0.78}{1.94} = 40 \text{ per cent.}$$

By lowering the pressure to 30 pounds per square inch (equals 69 feet head), the quantity of water passing the wheel per minute, with the same nozzle, was decreased to 295 pounds, corresponding to which the theoretical horse-power is 0.62.

The speed of wheel also being decreased to 354 revolutions per minute, the brake horse-power was only 0.40, but the efficiency has been increased to

$$\frac{40}{62} = 64 \text{ per cent.}$$

From this it will be seen, as previously noted, that a wheel may develop a maximum horse-power under given conditions, but the efficiency may be much less than that obtained under different conditions when the horse-power is not so great. The effect of varying the size of nozzle with varying head and load may be seen from the following tabulated results from tests made on a small motor by Mr. J. C. Escobar, the pressure ranging from 30 to 75 pounds per square inch.

TABLE V.

Test of a Small Hydraulic Motor.

Diameter of Nozzle.	Gauge Pressure in lbs. per sq. inch.	Head of Water on Wheel in feet.	Weight on Scales in lbs.	Weight of Water passing Wheel per minute.	Revolutions of Wheel per minute.	Theoretic Horse-power of Water.	Effective or Brake Horse-power.	Efficiency of Wheel, per cent.
½	65	150	0.5	428	668	1.94	.08	4
	65	150	6.25	428	526	1.94	.78	40
	55	127	6.0	391	404	1.50	.57	38
	45	104	6.0	346	341	1.09	.48	44
	35	81	5.0	310	341	0.78	.41	52
	30	69	4.75	295	354	0.62	.40	64
⅜	75	173	0.5	276	697	1.45	.08	6
	75	173	6.25	276	544	1.45	.81	56
	65	150	6.0	255	468	1.25	.67	60
	55	127	4.5	238	571	0.91	.61	66
	45	104	4.0	220	521	0.69	.49	71
	35	81	3.5	198	375	0.48	.31	63
	30	69	3.0	172	412	0.36	.22	61
¼	75	173	0.5	119	812	0.62	.09	15
	75	173	3.5	119	428	0.62	.35	57
	65	150	3.5	112	380	0.41	.31	75
	55	127	3.0	104	400	0.40	.28	71
	45	104	2.75	96	334	0.33	.22	65

The following results will show very clearly the effect of varying the load for the same head and diameter of nozzle. It will be noticed that as the load increases the speed decreases, and that the power developed increases with the load up to a given point; beyond this, however, the power, and hence the efficiency, decreases as the load is increased.

TABLE VI.

EFFECT OF INCREASING LOAD FOR A GIVEN NOZZLE.

Weight on Scales in lbs.	Revolutions of Wheel per minute.	Theoretic Horse-power of Water.	Effective or Brake Horse-power.	Efficiency of Wheel, per cent.
0.5	651	0.912	0.07	8
1.0	637	"	.15	16
1.5	624	"	.22	24
2.0	618	"	.29	32
2.5	612	"	.36	39
3.0	600	"	.42	46
3.5	594	"	.49	53
4.0	588	"	.36	61
4.5	571	"	.61	66
5.0	544	"	.65	70
5.5	517	"	.68	73
5.75	483	"	.66	72

Gauge pressure = 55 pounds, corresponding to a head of 127 feet; diameter of nozzle = $\frac{3}{8}$ inch.

For larger wheels the following methods for determining the quantity of water are employed: determination of the velocity of flow in a conduit of known cross-section by means of floats or current-meters; direct measurement by various forms of water-meter; measurement over weirs and through orifices.

When current-meters are used it is customary to divide a section of the stream (taken at right angles to the general direction of flow) into a number of parts, preferably of equal area, and to observe the velocity, as indicated by the current-meter, in each of these parts. From the mean of the observed velocities at different depths in each subdivision of the section the average velocity of the whole section is obtained; by multiplying the area of cross-section by the velocity per second the quantity of water passing through the section per second will be obtained.

If A = area of channel at the given section in square feet;
V_m = average velocity of current in feet per second;
Q = cubic feet of water passing through the channel per second,—

then

$$AV_m = Q.$$

A closer determination may be made by ascertaining the discharge of each subdivision from its area and mean velocity; the discharge of the stream will then be the sum of the discharges thus found.

It is evident that the mean velocity of each subdivision, and hence of the entire section, will be more closely determined the greater the number of vertical stations across the stream.

A very accurate method of obtaining the area at the given section in narrow streams or small navigable rivers is to run a cord or wire across the channel at right angles to the stream, and to take a number of soundings at equal intervals measured along the wire.

The lead for the soundings should be of sufficient weight to insure a vertical measurement in every case; its weight varies from five pounds for shallow, still water to twenty pounds for deep and swift currents. A long cylindrical shape, similar to a sash-weight, offering little resistance to the water, is suitable for the purpose.*

It is essential that the cord attached to the lead should be thoroughly stretched before being graduated. The graduations are placed one foot apart and indicated by a small strip of cotton attached to the line, every five feet being denoted by a leather strip.

Sounding-poles are preferable for shallow channels, and should be graduated to feet and tenths.

When float-measurements are used to ascertain the velocity of the current, it is advisable to take soundings in two sections, in order to determine accurately the discharge of the stream.

If a sufficient number of soundings be made, and the results plotted on section paper, the free-hand curve joining the lower ends of the vertical ordinates will represent very closely the contour of the bed of the channel, from which the area of the section may be obtained, either by the use of a planimeter or by one of the approximate methods.

For subsequent use in determining the height of the water, a permanent bench-mark, as for instance a spike driven into a tree-stump, should be established in the immediate vicinity and a water-gauge located near by. For this purpose a white-painted board, graduated to feet and tenths plainly marked in black, is fastened to a stake or post firmly set at the edge of the water; the zero-point of the scale is located with reference to the bench-mark previously set, which also provides a

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* Johnson's Surveying. Wiley & Sons, 1890.

means of resetting the gauge in case of disturbance or renewal.

The current-meter used at the present time is generally some modification of Woltmann's Mill or Tachometer shown in Fig. 71, which consists of a small wheel with inclined floats or vanes, *F*, held in the current,

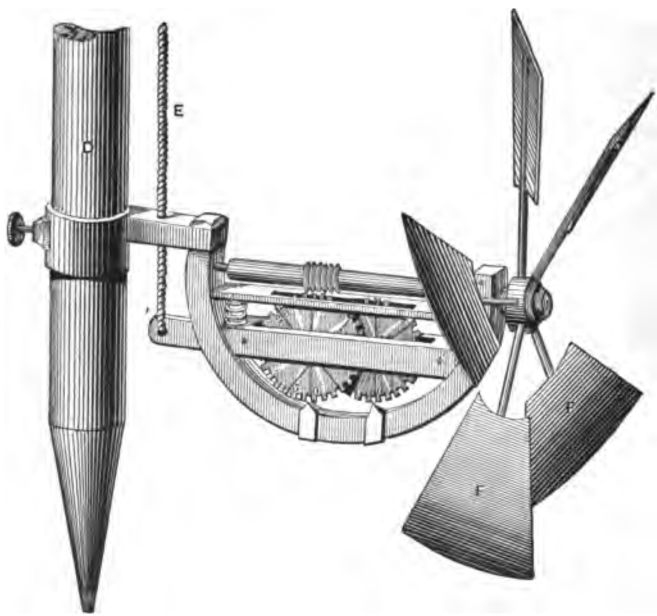

Fig. 71.—Woltmann's Mill.

which causes it to revolve at a speed nearly proportional to the velocity of the water passing it. By a suitable arrangement of gearing connection is made with an indicator which records the number of revolutions. Sometimes a rudder is attached to cause the wheel to face the current. The apparatus is either held at the extremity of a pole, *D*, or, by being adjustable along a vertical rod fixed in the bed, it may be set at any desired depth below the surface.

That the exact number of revolutions in a given time may be obtained, the instrument is arranged with a cord and spring so that the recording device may be thrown in or out of gear at any instant.

In some of the more recent instruments electrical connection is made with the rotating shaft by a "make-and-break contact," and the number of revolutions are shown on a registering apparatus on shore or at the surface of the water in a boat, as the case may be.

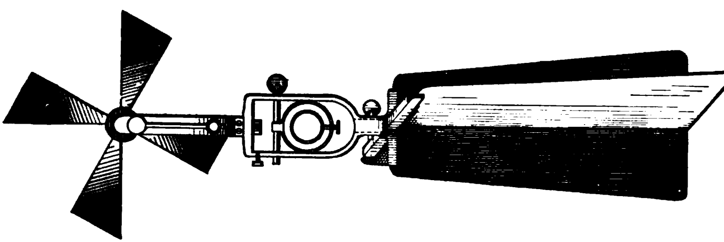

FIG. 72.—CURRENT-METER.

The form of current-meter shown in Fig. 72 * was used upon the gauging of the Connecticut River, and was designed particularly to avoid the catching of floating substances, such as leaves and grass, upon either the vanes or the axis, and to render the record of the instrument independent of the position of its axis with respect to the line of the current; also to get less friction upon the axis, so as to measure low velocities accurately.

* Made by Buff & Berger, Boston.

This current-meter is also adapted to be used with an electric register for showing the number of revolutions of the wheel. It is constructed upon the principle of Robinson's anemometer, turning by the difference of pressure upon opposite vanes of the wheel. The vanes of this meter, however, instead of being hemispherical cups with a straight stem, are made conical at the ends, and are hollow and taper to the central hub, so as to offer no obstruction to the slipping off of straws, leaves, or grass as the wheel revolves. The central hub is made tapering, so that any object can slide off easily, and it extends over the joints at the ends of the axis, so as to enclose and protect them from floating substances.

The axis runs in agates, through which a fine platinum wire connects with the metal of the frame.

The forward end of the frame which carries the wheel can be turned and secured in any position so that the wheel can be horizontal, vertical, or at any desired angle.

The electrical connection is made by carrying an insulated wire from near the centre of the instrument, where the insulated wire from the battery is attached to it when in use, out to the end of one arm of the wheel-frame, where it ends in a fine platinum wire resting upon a ring in the hub of the wheel. This ring is made of alternate interchangeable sections of silver and hard rubber, secured in place by screws, so that their position can be changed to register whole or part revolutions as desired.

There is also a socket and set-screw in the body of the frame near the centre, for the return-current, which can be carried most conveniently through a plain wire slightly twisted around the insulated wire so as to form one cord. If the instrument is run upon a wire, or has a metallic connection with the surface, the return-current can be made through that.

This meter can be used in connection with any apparatus for registering the revolutions of the wheel by the breaks in the electric circuit.

The Price current-meter,* which is used to a considerable extent by the U. S. Coast and Geodetic Survey, is shown in Fig. 73.

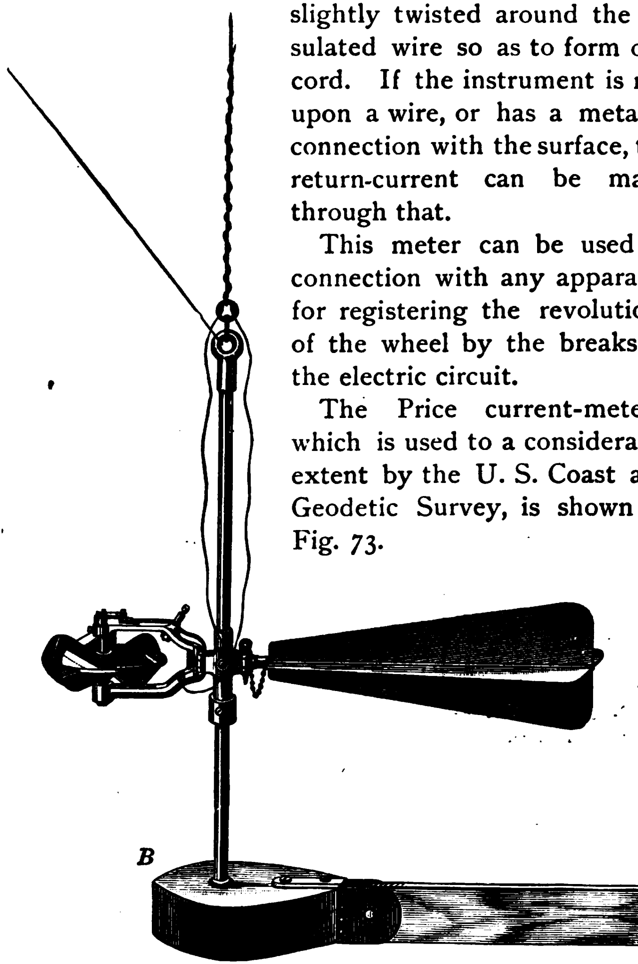

FIG. 73.—PRICE'S CURRENT-METER.

* Made by W. & L. E. Gurley, Troy, N. Y.

The wheel of this meter carries five conical buckets, very strongly and compactly formed so as to be able to resist injury from floating driftwood, while at the same time it is so designed as not to be liable to obstruction from leaves or grass.

A hollow trunnion fitting freely upon the rod supports the frame by a pivot on each side, and thus by the rod and pivots the meter is free to move both horizontally and vertically, and so adjust itself to the direction of the current.

The rod is of brass, $\frac{3}{4}$ inch in diameter and 2 feet long, its upper end having an eye of brass screwed firmly on and pinned, and its lower end screwed into a brass socket in the weight *B*, and secured by a nut. The weight *B* is of lead and weighs about sixty pounds; it has a rudder of wood, which can be set at any angle with the weight, or turned up parallel with the rod when not in use. This weight is only used for deep-water and harbor surveying where the currents are very strong. For shallower waters the meter is used upon a rod of wood or metal.

FIG. 74.

The electric register used with this instrument is shown in Fig. 74.

Before using a current-meter it will be necessary to calibrate it in order to ascertain the number of revolutions of the wheel with known velocities of current. The calibration of the instrument is most readily obtained by causing it to pass through a measured distance at a uniform velocity in still water not less than 5 feet deep. To secure a good rating there should be no wind and the meter should be immersed to a depth of about 2 feet below the surface. The usual method of obtaining the velocities for rating the meter is to attach the instrument to a vertical rod which projects 2 or 3 feet in front of the bow of a small boat, and to either row or pull the boat, by means of cords, at as uniform a rate as possible, over a measured course of about 200 feet, observers on shore noting the exact time of passing the range-lines. By varying the speed of the boat for successive passages, which should be at a uniform rate, a table of constants may be computed from which the velocity of any current can be determined from the number of revolutions per minute as shown on the dial.

Frequent ratings of a meter while in use will insure reliability in its readings.

In the method of measurement by surface-floats the velocity is obtained by observing the time of transit of a light floating body, such as a flat disk, or ball of wood, over a known distance. By placing several floats across a stream and noting their velocities, the average surface-velocity may be approximately computed, but this method is apt to be very inaccurate when there are any local disturbances due to wind or eddies in the current. The use of double floats presents a much more reliable means of obtaining the velocity.

As in the method with current-meters the velocity of the filaments should be ascertained in several verticals across the stream and at various depths below the surface. For this purpose a body slightly heavier than the water is suspended at the desired depth from another body floating at or just beneath the surface, and of such a form and size as to offer less resistance to the stream than the first, so that without sensible error the velocity with which the floats are carried along by the current is that of the submerged body and of the stream at the particular depth below the surface at which it is placed.

For the surface-float a block or ball of wood is often used, but hollow floats, such as glass or metallic balls, are preferred by many engineers, as they may be partially filled with water and sunk just below the surface, where they are less affected by the wind. A small flag or other suitable indication will locate the position of the float.

For the sub-surface-floats metallic balls have been used from 6 to 8 inches in diameter. Humphreys and Abbot, in their work on the Mississippi, used small kegs without top or bottom, ballasted with strips of lead so as to sink and remain upright; these kegs were 9 inches high and 6 inches in diameter, but for depths greater than 5 feet below the surface a larger size, 12 inches high and 8 inches in diameter, was used.

A very convenient form of float is made by joining two sheets of galvanized iron at right angles, intersecting in their centre lines, and weighting the lower edges with lead. This maintains the float in an upright position and gives the required tension on the connecting cord. The vanes should be from 8 to 20 inches high, depending upon the depth of stream in which they are to be used. Cylindrical air-cavities are provided along the upper edges of the vanes.

By connecting the upper and lower floats with a fine wire, chain, or, preferably, a braided silk cord, and varying its length, we shall obtain the several velocities at varying depths. The mean of all these observed velocities may be assumed to be the average velocity of the current.

To obtain the mean velocity in a perpendicular by a single measurement, a floating rod is employed. This rod may be either of wood or tin in sections screwed together for convenience; the lower section being fitted with a hollow metal cap which is filled with enough shot or gravel to cause it to sink to the required depth and to maintain a nearly vertical position. The immersion of the rod should be at least nine tenths of the depth of the water, which should not be more than 20 to 25 feet.

If the channel were of uniform depth, and the rod reached to the bottom without actually touching, then the velocity of the rod would be very nearly the mean velocity of all the filaments in the vertical plane through which the rod passes. As the rod does not reach the bottom, its velocity can only record the mean velocity of the filaments in a vertical plane to a depth equal to its immersion.

In general the rod-float will give for small channels more reliable results than those obtained by the use of the double ball-float.

To obtain the velocity or rate of motion of floats, two parallel range lines are laid off on shore, from 100 to 200 feet apart, and the float placed in the current at some distance above the first range-lines. Two transits are usually employed for timing the floats, one being set on each range. In addition two time-keepers will be required to take the exact time on stop-watches when signalled by the observer at the transit.

If the stream is not too wide, the passage of the float across the fixed ranges may be noted by a single observer using only a stop-watch and, if occasion require it, a field-glass. The watch is started when the float crosses the first line, then the observer walks to the lower station and stops the watch the instant the float passes the range-line. The total distance, s, divided by the number of seconds, t, will give the mean observed velocity, v, of the float, or

$$v = \frac{s}{t}.$$

On account of the uncertainty of float-measurements, due to action of the wind, local currents, eddies, and other causes, several observations should be taken to obtain a fair average value of the velocity.

Approximate determinations of the mean velocity of a stream in any vertical may be made from a single measurement by obtaining either the mid-depth velocity or the surface-velocity, and multiplying such velocity by a coefficient.

It has been shown that the curve plotted for the velocity of the filaments in a vertical will, in general, be represented by a parabola whose axis is parallel to and beneath the surface, except when the wind is down-stream with a rate equal to or greater than the velocity of the current. According to Humphreys and Abbot the axis of the parabola, or filament of maximum velocity, will approach the surface or recede from it, depending upon the direction and intensity of the wind. This is shown in Fig. 75, which is taken from the report of Humphreys and Abbot.*

When the air is calm the axis will be found to lie about 0.3 of the entire depth of stream beneath the surface. A down-stream wind brings the axis nearly

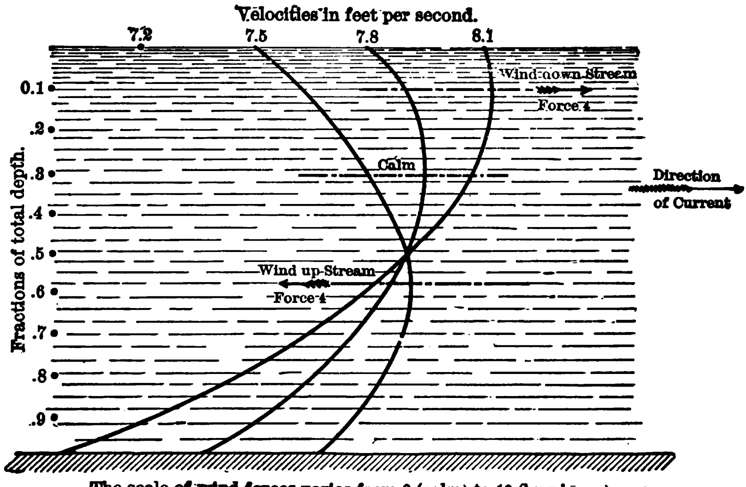

FIG. 75.—VELOCITY OF CURRENT AT VARYING DEPTHS.

to the surface, while with the wind up-stream it is found below the mean depth. It will be noticed that these curves intersect at about mid-depth, from which it is inferred that the velocity of the mid-depth filament is not affected by the wind.†

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* Physics and Hydraulics of the Mississippi River.

† For an exhaustive discussion of velocity of currents, see Hering & Trautwine's translation of Ganguillet & Kutter's General Formula for Flow of Water in Rivers.

This mid-depth velocity will represent very closely the mean velocity of the vertical, being from one to six per cent greater, according to the velocity of stream, depth, and roughness of bed. Hence by taking the different station or division mid-depth velocities and applying a coefficient of from 0.96 to 0.98, the mean velocity of the sub-section will be obtained.

The other method—that of measurement from surface velocity alone—has been employed to a considerable extent, but it must be remembered that the results are only approximate, and for this reason should be used only for rough estimates. From many experiments to determine the mean velocity in a vertical from its measured surface velocity, it has been found that if the observation be taken when there is no sensible wind, the mean velocity of the current may vary from 0.8 to 0.9 of the surface velocity. If a mean value of 0.85 be used for the coefficient, the discharge calculated from the average of all the surface velocities thus obtained may be assumed to approximate the actual discharge within a limit of from ten to twenty per cent. For obtaining the surface velocity a current-meter should be used.

A very old instrument for measuring velocities, invented or used by Pitot, consisted simply of a vertical glass tube with a right-angled bend, placed so that its mouth was normal to the direction of flow (Fig. 75*a*).

The impulse of the stream balances a column in the tube, the height of which is $h = \frac{Kv^2}{2g}$; where v is the velocity at the mouth of the tube, and K is a coefficient determined by experiment.

According to Weisbach's experiments a mean value of K for velocities varying from 1 to 4 feet per second is 1.2.

Pitot expanded the mouth of the tube so as to form a funnel or bell-mouth. In that case he found $K = 1.5$. The objection to this is that the motion of the stream is interfered with, and the velocity in front of the tube is probably not the same as that of the unobstructed stream.

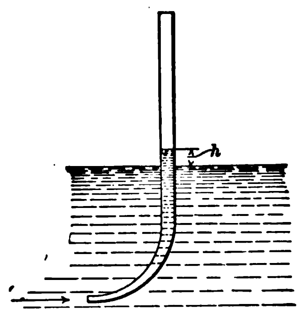

FIG. 75*a*.—PITOT TUBE.

For this reason the tube was made converging at the end so as to avoid interference with the stream, and also to prevent oscillations of the water-column.

One objection of the Pitot tube in its original form was the great difficulty and inconvenience of reading the height h in the immediate neighborhood of the stream-surface. This is obviated in the Darcy gauge, which can be removed from the stream to be read.

This instrument consists of two Pitot tubes having their mouths at right angles, one of which is placed normal to the stream. With the tubes placed in this way it is found that the difference of level, h, is sensibly equal to $\frac{v^2}{2g}$, from which the velocity at a definite point in the stream may be ascertained. The tubes are so arranged that both can be closed at any instant and the instrument lifted out of the water, thus permitting the difference of level to be readily determined.

These readings are, however, not uniformly constant, and a number of observations are required to determine an average. In order to facilitate the readings the tubes are so connected at their upper ends that a portion of the air may be sucked out: this will raise the water in both columns to an amount proportional to the difference between atmospheric pressure and that in the tubes. It is obvious that the difference of level will remain the same.* The use of Pitot's tube is accompanied with many disadvantages, and it is now rarely employed as an instrument for velocity-determinations in running streams.

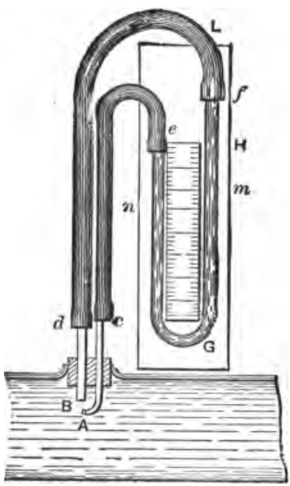

FIG. 75*b*.—MODIFIED PITOT'S TUBE.

A modified form of Pitot's tube is sometimes used for measuring the velocity of liquids or gases flowing under pressure. This is shown in Fig. 75*b*, which will be readily understood. The difference in the two legs of the U tube is due to the velocity of the current, since both branches are subjected to equal pres-

* See article on Hydromechanics, Enc. Brit., vol. XII.

sure. In this case also, $v = K\sqrt{2gh}$, in which K is to be determined for any given tube.*

The method of measurement by water-meters is often employed to ascertain the quantity of water used by a motor when the latter is supplied through a pipe.

In general water-meters may be divided into two classes: the positive or volume meter, and the inferential or velocity meter.

The positive includes all forms of piston-meters, whether reciprocating or rotary, together with a numerous type known as disk-meters. The second class, or inferential meter, is usually some form of turbine, screw, or rotating vanes. The distinctive difference between the two is that the positive meter measures water by means of a chamber alternately filled and emptied; whereas in the inferential meter the flowage is determined by the rate of revolution produced by the current upon the rotating device.

Such meters may, however, be very unsatisfactory; for when the flow is small the force may be insufficient to overcome the friction in the meter and cause the turbine to revolve, in which case there is no record of the water that passes through.

Among the older forms of positive water-meter which have been satisfactorily used for a number of years is the Worthington piston-meter, shown in Fig. 75*c*.

This meter is of the reciprocating-piston type, and

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* Carpenter's Experimental Engineering, p. 266.

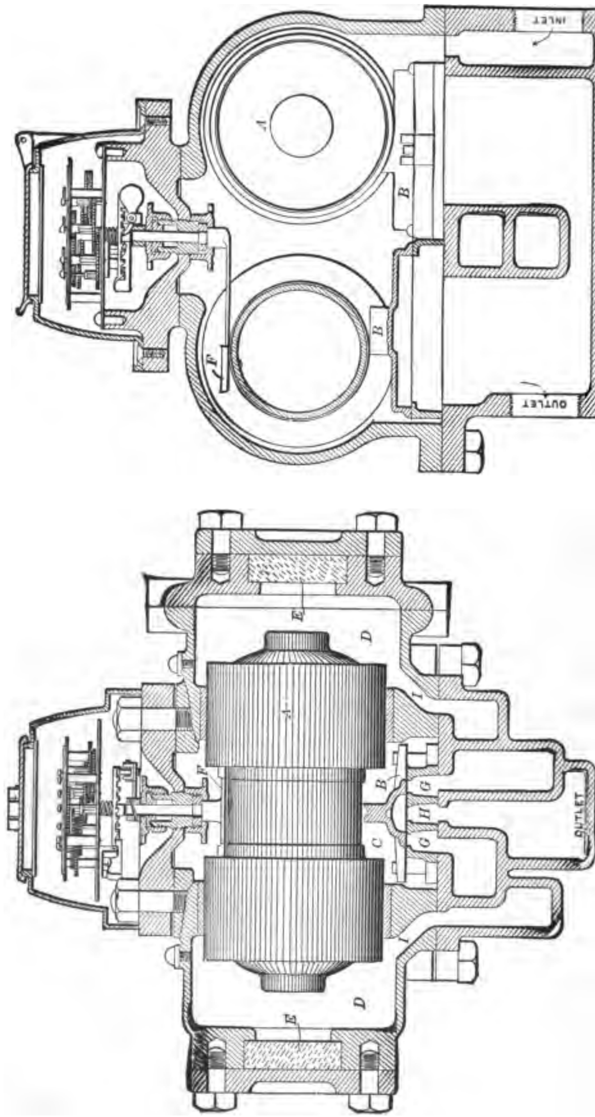

FIG. 75c.—THE WORTHINGTON WATER-METER.

will be readily understood from the following description.

The plungers *A A* are closely fitted in parallel rings. The water passes through the inlet and port *I*, and is admitted under pressure into chamber *D*, at one end of each plunger alternately, while the connection is made between the chamber at the other end of the outlet. Thus the plunger in moving displaces its volume, discharging it through the outlet. The arrangement is such that the stroke of the two plungers alternates, the valve actuated by one admitting pressure to the other. The plungers are brought to rest at the end of the stroke by the rubber buffers *E E*. One plunger imparts a reciprocating motion to the lever *F*, which operates the counter-movement through the spindle and ratchet-gear as shown. Thus it will be seen that the counter is arranged to move the dial-pointers once for every four strokes or displacements, and unless there be leakage in the parts no water should pass through the meter without registration, for, in order to pass through, it must be displaced by the plungers and therefore recorded by the movement of the lever and counter mechanism. The counter usually registers in cubic feet, one cubic foot being 7.48 U. S. gallons. The counter is read in the same way as registers of gas-meters.

The following example and directions may be of use to those unacquainted with the method: If the pointer is between two figures, the smaller one must invariably be taken; suppose the pointers of the dials stand as shown in Fig. 75*d*; starting at the dial marked 10 cubic feet, we get the figure 4; from the next, marked 100 cubic feet, the figure 7; from the next, marked 1000 cubic feet, the figure 8; and from the next, marked 10,000 cubic feet, the figure 6; the pointer on the 100,000-cubic-foot dial being between the 10 and the 1 indicates nothing. The reading is therefore 6874 cubic feet.

FIG. 75*d*.

The Hersey disk-meter is a very satisfactory illustration of the modern type of positive piston rotaries. The piston of this meter, Fig. 75*e*, is a vulcanized rubber disk, provided with a spherical bearing in the centre. The chamber in which it operates is made to conform to the motions of the disk; the surface of one section is a portion of a sphere, the other two sections are a conical plate and a straight plate, respectively; the central part of the two plates being formed to accommodate the spherical bearing of the disk, which latter divides the cylinder at all points in its movement into receiving and discharging spaces.

In operation the water in the receiving space exerts a pressure on one side of the disk, and at the same time the pressure is relieved in the discharging space on the opposite side; this causes the piston to gyrate, and as an amount of water is discharged at each complete movement equal to the contents of the entire cylinder, the number of gyrations of the disk will indicate the

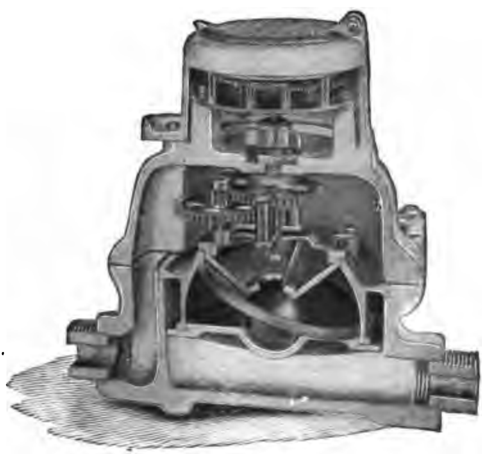

FIG. 75e.—THE HERSEY DISK-METER.

amount of water passing through the meter. The movement of the disk is communicated to a dial which indicates in the desired unit of measurement.

The disk operates as its own valve to control the water, the edge of the disk passing over openings in the spherical walls of the chamber in such a way as to admit and discharge the water at the proper time. The water enters the lower half of the meter, passes through the side wall of the cylinder, through the measuring chamber into the upper half, and thence out through the passage on the side of the meter.

Fig. 75*f*.—The Lambert Water-meter.

The Lambert meter illustrated in Fig. 75*f* represents another form of disk-meter in which the exterior confining wall assumes the form of a spherical zone. The principle of action is similar to that in the Hersey, but the details and construction are somewhat different, as will be noticed by reference to the cut.

A recent form of water-meter has been invented by Mr. Clemens Herschel,* in which a compound tube provided with piezometers is used to determine the discharge. This apparatus is constructed upon the results of experiments by Venturi which show that when water flows through a pipe of which the section is contracted and subsequently gradually increased, the pressure in the smallest section is much less than in the largest on either side of the contraction, and may with suitable proportions sink below the atmospheric pressure, so that it can be measured by a vacuum-gauge. The velocity in the smallest section is theoretically that due to the effective head corresponding to the difference between the pressure in the largest section before the contraction and that in the smallest section, plus the influence of the velocity in the largest section, generally very slight. To obtain the actual velocity, the theoretical quantity has to be multiplied by a coefficient to be determined by experiment.

The Venturi meter, Fig. 75*g*, is formed of two truncated cones joined at their smallest diameters by a short throat-piece. At the upstream end and at the throat there are encircling pressure-chambers as shown at *U* and *T*, in which the piezometer-tubes are inserted. The difference of pressure at these points is always the same for the same velocity of flow, whatever the total

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* The Venturi Water-meter. Trans. A. S. C. E., vol. XXII, Dec. 1887. Also Hydraulic Motors, by G. R. Bodmer, p. 322.

or hydraulic pressure may be; in practice this has been found to be very nearly that obtained from the fundamental formula $h = \frac{v^2}{2g}$, where h corresponds to the difference in pressure (in feet of water) at U and T, and v is the velocity of flow in the throat. If no

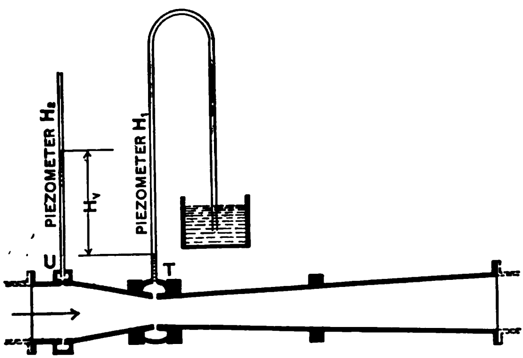

FIG. 75g.—THE VENTURI TUBE.

losses of head occur, the difference of pressure or "head on the venturi" in terms of the height of a column of water will be $H_v = \frac{v_1^2 - v_2^2}{2g}$, in which v_1 = velocity in the throat, and v_2 = velocity in the tube at the point U.

In the construction of the Venturi meter the area a_2 of the tube at U is 9 times greater than the area a_1 at the throat T; hence the head on the venturi is

$$H_v = \frac{80}{81} \frac{v_1^2}{2g},$$

and the theoretical velocity through the throat is

$$v_1 = \sqrt{2gH_v \times \tfrac{81}{80}} = 1.0062\sqrt{2gH_v}.$$

Since the quantity of water, Q, passing through the venturi is equal to the area multiplied by the velocity of water,

$$Q = a_1 v_1 = 1.0062 a_1 \sqrt{2gH_v}.$$

Owing to the losses of head which actually exist, this expression must be multiplied by a coefficient, C, in order to obtain the real discharge. The actual delivery will then be

$$Q_a = 1.0062 C a_1 \sqrt{2gH_v}.$$

A careful and exhaustive series of experiments by Herschel gave C values ranging from 0.94 to 1.04 throughout a test of fifty-five experiments with a meter inserted in a 12-inch tube; of these only two gave values less than 0.96, and only four were greater than 1.01. In another set of experiments with a meter inserted in a tube 9 feet in diameter, twenty-eight tests gave values ranging from 0.95 to 0.99, the highest coefficients being those obtained with the lowest velocities.

With any other ratio of areas the theoretical discharge, Q, is obtained by substituting the values of v_1 and v_2 in terms of the respective areas in

$$H_v = \frac{v_1^2 - v_2^2}{2g};$$

$v_1 = \dfrac{Q}{a_1}$, and $v_2 = \dfrac{Q}{a_2}$, hence

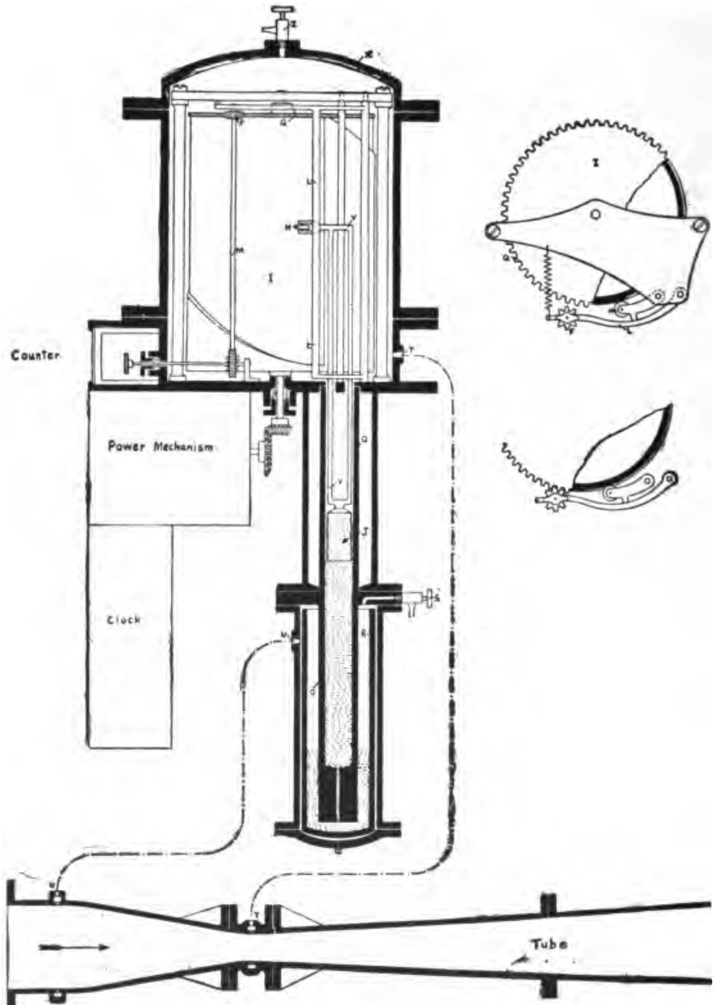

Fig. 75h.—The Venturi Meter.

$$H_v = \frac{Q^2(a_2^2 - a_1^2)}{2ga_1^2a_2^2} \quad \text{or}$$

$$Q = \frac{a_1a_2\sqrt{2gH_v}}{\sqrt{a_2^2 - a_1^2}};$$

from which the actual discharge may be determined by introducing the coefficient C as before, that is,

$$Q_a = C\frac{a_1a_2\sqrt{2gH_v}}{\sqrt{a_2^2 - a_1^2}}.$$

In all cases H_v is the difference of level in two columns of water measured in the piezometers H_2 and H_1 inserted in the tubes and throat respectively. If the depression of the venturi be sufficient, a vacuum will be obtained in the piezometer H_1, and the suction will support a column of water h_1 feet in height in the inverted leg of the piezometer-tube. The head on the venturi, H_v, will then be h_1 plus the height of a column of water in piezometer H_2 measured above the interior crown of throat.

The Venturi meter may be connected to a registering apparatus and the quantity of water which passes through read off from a dial as in the case of other meters previously described.

In this case the pressures existing in the tube and throat are transmitted by small pipes T, U, to the register,* Fig. 75*h*, where they oppose one another, and are balanced by displacement of level of two columns of mercury in cylindrical tubes, one within the other. The inner mercury column carries a float,

* Made by Builders' Iron Foundry, Providence, R. I.

FIG. 75*i*.—THE VENTURI METER IN POWER STATION.

J, *V*, the position of which is dependent on, and is an indication of, the velocity of water flowing through the tube. The position assumed by an idler wheel *H*, carried by this float, relative to an intermittently revolving integrating drum *I*, determines the duration of contact of gears *G* and *F* connecting the drum and counter, by which the flow for successive intervals is registered.

The tube is usually laid as a part of the pipe line and requires no more attention than the line itself. Fig. 75*i* shows an application of the Venturi meter for ascertaining the quantity of water delivered to the wheels at the power station of the Pioneer Electric Power Co., Ogden, Utah.*

The success which has already attended the use of the Venturi meter indicates that this meter may be more satisfactorily used in power determinations involving large quantities of water under varying conditions than any other method.

The experiments of Herschel with velocities varying from six inches to six feet per second through the tubes show the coefficient of discharge to be remarkably constant, and it is probable that in most cases the error will fall well within a limit of two per cent.

The accuracy of water-meters depends primarily upon the rate of flow, although the constructional features and the condition of the wearing parts has an appreciable effect upon their sensibility. In all cases, therefore, where such meters are to be used for power measurement the instrument should be carefully calibrated under different rates of discharge. Any error

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* *Engineering News*, July 8, 1897.

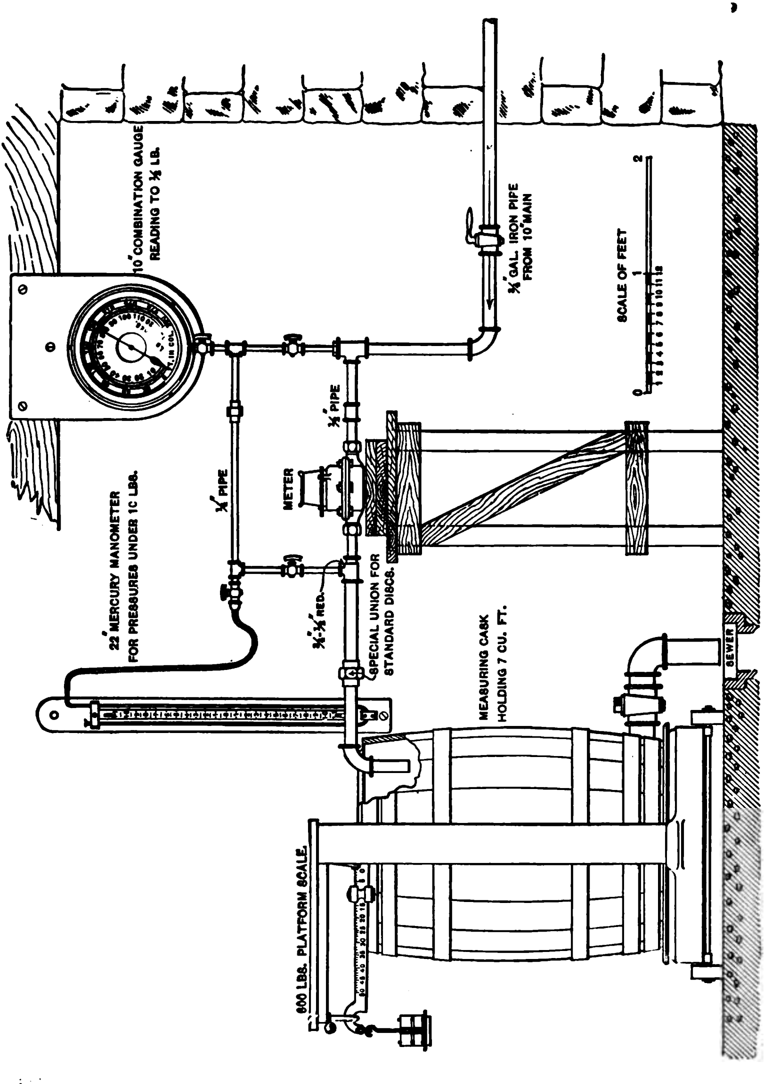

FIG. 75j.—APPARATUS FOR TESTING WATER-METERS.

thus ascertained will furnish a constant for correction when the meter is in use.

Fig. 75*j* illustrates an admirable method for testing water-meters which readily permits the measurement of the discharge and the indication of the pressures in the inlet and outlet pipes of the meter.*

In this arrangement varying rates of discharge are obtained through perforations in thin brass disks placed in the outlet connection of the meter. The disks are held against a rubber gasket in a brass union recessed as shown in detail in Fig. 75*k*. The apertures in the disks vary from $\frac{1}{32}$ inch to $\frac{1}{2}$ inch in diameter.

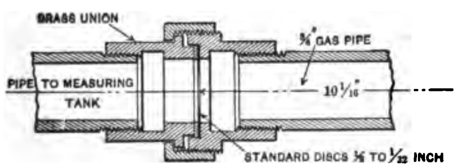

FIG. 75*k*.—DETAIL OF UNION AND STANDARD ORIFICE.

During the tests referred to, the stop-cock which controls the supply from the main was set at full opening, and the flow of water through the meter controlled entirely by the standard orifice in the discharge-pipe. Before commencing a test each meter was operated some time at full opening to work out the air.

The results of the tests made by Mr. Hill indicate that disk-meters consume somewhat less of the static head on the service-pipe and therefore realize a larger percentage of reduced capacity than the piston or

* See paper by Mr. John Hill on Tests of Water-meters, Trans. A. S. C. E., vol. XLI, Feb. 1899.

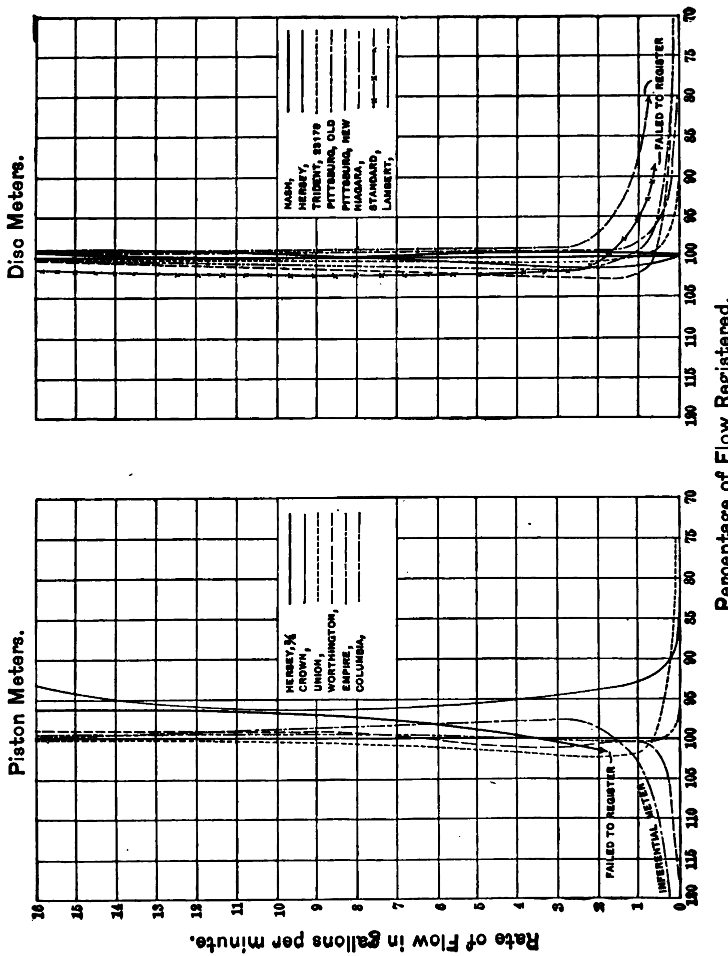

FIG. 75½.—ACCURACY TEST OF ⅝-INCH METERS.

inferential meters. The loss of head and reduced discharge by the introduction of a water-meter may, in some instances, be a serious drawback, especially with motors running under full load when the consumption should fully equal the capacity of the service-pipe.

The tests show that the actual loss of head for $\frac{5}{8}$-inch piston-meters when placed in a $\frac{3}{4}$-inch service-pipe was about 84% on Worthington, Crown, and Empire, and 72% for Union rotary with $\frac{1}{2}$-inch orifices; 17% for Worthington and Crown, 15% for Empire, and 8% for Union rotary with $\frac{1}{4}$-inch orifices; and from 1.18 to 0.82 for all four meters with $\frac{1}{8}$-inch orifices. A $\frac{3}{4}$-inch Hersey piston-meter showed better results, as would be expected with a meter of the same size as the service-pipes. For the eight $\frac{5}{8}$-inch disk-meters the loss of head was not far from 75% for $\frac{1}{2}$-inch; 9 or 10% for $\frac{1}{4}$-inch, these being averages; and generally considerably below 1% for $\frac{1}{8}$-inch orifices. The $\frac{5}{8}$-inch Columbia inferential meter gave 91% less for $\frac{1}{2}$-inch orifice, 25% for $\frac{1}{4}$-inch, and 3% for $\frac{1}{8}$-inch.

A total of 17 different meters was tested by Mr. Hill, and the trifling average variation in errors of registry for rates of discharge, varying in some examples as high as 90 to 1, is surprising. While the maximum error with the low rate of discharge obtained through the $\frac{1}{32}$-inch opening was in some cases excessive, it is to be noted, Fig. 75*l*, that for ordinary rates of consumption the average error is negligible.

The method of measuring the flow of water over weirs is that which has usually been employed in testing the larger hydraulic motors, for the reason that it has generally been considered the most convenient and practicable.

The coefficients which enter into the formula for discharge over a weir vary to such an extent with differences in proportions that accurate results are only attainable by closely following the proportional dimensions of the apparatus and constructing the weir and its approaches in accordance with the experiments for which a given set of coefficients have been determined.

By this method the stream to be measured is dammed by a weir and all the water compelled to flow through a rectangular opening at the top. Occasionally the weir is suppressed or drowned and the water is allowed to fall over the whole length of the weir, in which case the sides of the conduit or head-race should be parallel and vertical for some distance up-stream above the weir.

If h denote the height in feet of the water-level above the edge of weir,—measured a few feet back from the sill before the sheet of water begins to curve downwards,—L the length of the weir-opening in feet, then the theoretical quantity of water discharged per second can be shown to be

$$Q = \tfrac{2}{3}Lh\sqrt{2gh}.$$

The results of experiments, however, show that the actual quantity is less than this, therefore a coefficient must be used to determine the correct amount.

It has been found that the coefficient varies with the following dimensions and conditions:

Length of weir;

Height of water over weir;

Width of canal of approach or head-race;

Nature and thickness of edges of weir;

Distance from bottom of weir to bottom of conduit.

When the length of opening in weir is less than the width of head-race, so that the opening has thin edges at the ends, it is said to have end-contractions, since the thin ends cause the stream of water to contract in flowing through.

The formula deduced by Francis * from experiments on weirs from ten to twenty feet wide and from seven to nineteen inches depth of water over crest, is

$$Q = 0.623 \times \tfrac{2}{3} Lh \sqrt{2gh};$$

or, as it is generally written,

$$Q = 3.33 L \sqrt{h^3}$$

when there are no end-contractions, and

$$Q = 3.33(L - 0.1nh) \sqrt{h^3}$$

when end-contraction occurs and n (usually 2) is the number of end-contractions.

To secure accurate results, the up-stream edge of the crest of the weir is made straight, sharp, and smooth—usually by constructing it with an iron edge, bevelled sharply, and fitting similar apertures at the sides. The depth of the water below the crest of the weir should be not less than one-third of the length of the weir; otherwise the velocity of approach must be

* Lowell Hydraulic Experiments.

considered, as this tends to increase the volume of water carried over. Air should have free access to the space under the sheet as it flows over the crest.

A more exact determination may be obtained by the use of the following coefficient tables computed by Mr. Hamilton Smith, Jr.,* from the experiments of Poncelet, Lesbros, Francis, Fteley & Stearns, and others, in which a separate coefficient is given for varying lengths of weir and under different heights above crest of weir.

TABLE VII.

COEFFICIENTS FOR DISCHARGE OVER WEIRS: TWO END-CONTRACTIONS.

Coefficient = c in formula $Q = c \times \frac{2}{3} Lh \sqrt{2gh}$.

Effective Head in Feet.	L = Length of Weir in Feet.									
	.66	1	2	3	4	5	7	10	15	19
.1	.632	.639	.646	.652	.653	.653	.654	.655	.655	.656
.15	.619	.625	.634	.638	.639	.640	.640	.641	.642	.642
.2	.611	.618	.626	.630	.631	.631	.632	.633	.634	.634
.25	.605	.612	.621	.624	.625	.626	.627	.628	.628	.629
.3	.601	.608	.616	.619	.621	.621	.623	.624	.624	.625
.4	.595	.601	.609	.613	.614	.615	.617	.618	.619	.620
.5	.590	.596	.605	.608	.610	.611	.613	.615	.616	.617
.6	.587	.593	.601	.605	.607	.608	.611	.613	.614	.615
.7	.585	.590	.598	.603	.604	.606	.609	.612	.613	.614
.8			.595	.600	.602	.604	.607	.611	.612	.613
.9			.592	.598	.600	.603	.606	.609	.611	.612
1.0			.590	.595	.598	.601	.604	.608	.610	.611
1.1			.587	.593	.596	.599	.603	.606	.609	.610
1.2			.585	.591	.594	.597	.601	.605	.608	.610
1.3			.582	.589	.592	.596	.599	.604	.607	.609
1.4			.580	.587	.590	.594	.598	.602	.606	.609
1.5				.585	.589	.592	.596	.601	.605	.608
1.6				.582	.587	.591	.595	.600	.604	.607
1.7							.594	.599	.603	.607

* Smith's Hydraulics. Wiley & Sons, 1886.

The heads in the first column are the effective heads, and in such cases when the water approaches the weir with a sensible velocity the head h' due to that velocity $\left(h' = \frac{v^2}{2g}\right)$ must be used in connection with the head h over the weir, in which case the effective head may be considered equal to $h + 1.4h'$; hence

$$Q = c \times \tfrac{2}{3} L \sqrt{2g(h + 1.4h')^3}.$$

As an example of the application of the table, the following is taken from Merriman's "Hydraulics." Let it be required to find the discharge per second over a weir 4 feet long when the head h is 0.457 foot, there being no velocity of approach.

From the table the coefficient of discharge is 0.614 for $h = 0.4$, and 0.610 for $h = 0.5$, which gives about 0.612 for $h = 0.457$. Then the discharge per second is

$$Q = .612 \times \tfrac{2}{3} \times 8.02 \times 4 \times \sqrt{(.457)^3} = 4.04 \text{ cubic feet.}$$

If the width of the feeding-canal be 7 feet and its depth below the crest be 1.5 feet, the velocity-head will be

$$h' = \frac{v^2}{2g} = 0.01555v^2;$$

but the velocity $v =$ quantity of water discharged, divided by the area of the stream; hence

$$v^2 = \frac{Q^2}{A^2} = \frac{(4.04)^2}{(7 \times (1.5 + .457))^2};$$

the velocity-head then becomes

$$h' = 0.01555\left(\frac{4.04}{7 \times 1.96}\right)^2 = 0.00134 \text{ foot.}$$

The effective head now becomes $h + 1.4h' = 0.459$ foot, and the discharge per second is

$$Q = .612 \times \tfrac{2}{3} \times 8.02 \times 4 \times \sqrt{(.459)^3} = 4.07 \text{ cubic feet.}$$

As this result depends upon the degree of accuracy with which the quantities used are ascertained, and also upon a possible slight error in obtaining the coefficient, it may be assumed that a probable error of at least one per cent exists in the final result.

It will be evident that as the velocity-head h' is small compared with the head over weir, the latter may be used as the effective head, with no appreciable error, in selecting a coefficient from the table for the first approximation.

The method of measurement over weirs is often employed in testing-flumes by constructing the tail-race with a rectangular opening, through which the discharge which flows from the motor is measured in the manner just described.

The determination of the height of water over weirs requires considerable care for accurate tests, on account of the small height generally involved. For this purpose some form of the Boyden hook-gauge is usually employed.

The instrument,* shown in Fig. 76, consists of a

* Made by W. & L. E. Gurley, Troy, N. Y.

wooden frame 3 feet long and 4 inches wide, in a rectangular groove of which another piece is made to slide carrying a metallic scale divided to feet and hundredths, and figured from 0 to $2\frac{2}{10}$ feet, as shown.

FIG. 76.

Connected with the scale is a brass screw passing through a socket, fastened to another shorter sliding piece, shown above, which can be clamped at any point on the frame, and the scale with hook moved in either direction by the milled-head nut.

There is also a vernier attached to the frame, and movable under the screw-heads which secure it, in order to adjust its zero to correspond with the point of the hook when setting the gauge. The vernier reads to thousandths of a foot.

The form of hook-gauge designed by Emerson and used in the Holyoke testing-flume is shown in Fig. 77. In this gauge a small gear operated by a worm engages a finely cut rack on the back of the scale-rod which permits a very close adjustment of the hook, the vernier being arranged to read to ten-thousandths of a foot.

When in use the hook is raised from below the level of the water until its point barely pricks the surface, when it will be noticed that a slight swell and distortion of the reflected light is caused just above the point of the hook; by carefully lowering the hook until this distortion disappears, the point may be assumed to be at the level of the water, which can then be read from the vernier. The instrument is supposed to have been previously set with its vernier at zero, when the point of the hook was exactly on a level with the sill of the weir.

The hook-gauge is generally enclosed in a wooden case or box open at the top, and provided with a small inlet at the bottom, in order to prevent any disturbance of the water in the vicinity of the hook. As previously stated, the measurement of the head over the weir must be taken several feet back of the crest, where the water is level.

To allow the observations to be taken more readily, the water may be led by a hose or other pipe from the bottom of the race-way above the weir (up-stream) into the hook-gauge box, which may be placed at any convenient point near by.

Very accurate results may be obtained by the use of a good levelling-rod with a hook secured to the foot; the slide may be operated by a small gear and rack which can be attached to the rod to allow a fine adjustment of the hook.

The total head or fall can be obtained very precisely by the use of a hook, at the level of the water in the tail-race, secured to a graduated rod placed beside

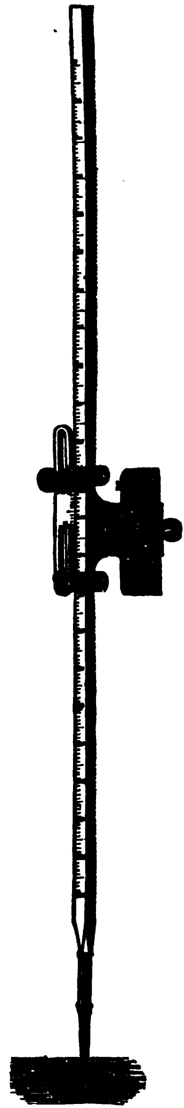

FIG. 77.

a fixed cylinder with glass tube attached, which is connected by means of a rubber hose with the upper water-level. By this arrangement the reading on the scale at the water-level in the glass tube will give the total height between the two levels—the graduations on the rod being a measure of the distance from the point of the hook.*

Another method of obtaining the total head is to run a line of levels from one to the other. Permanent bench-marks being established, gauges can then be set in the head and tail races, and graduated so that their zero-points will be at some datum below the tail-race level. The difference in readings will give the required total head.†

Simpler methods will suggest themselves and may be used where less accuracy is required, as in rough estimates of water-power.

As water in most cases where available for power has a commercial value, the most advantageous and profitable use of it should be considered.

In this relation not only the efficiency of the motor employed, but the pipes which supply the motive power have an important bearing upon the result.

When water is delivered to a hydraulic motor through a pipe or nozzle, as in the numerous class of small motors fed from a city main, the diameter and length of pipe, as well as the size and shape of nozzle, largely affects the work done on the motor.

The head is not that due to difference in levels

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* R. H. Thurston. Trans. A. S. M. E., vol. VIII.

† Merriman's Hydraulics, p. 288.

between the reservoir and the motor, but is much less on account of losses in transmission due to friction in the pipes, loss at entrance, loss due to bends and angles in the pipe, changes in cross-section, and other causes.

As any loss of head is a direct loss of power, such loss should be prevented as much as the circumstances in the case will justify. Where both the water-supply and head are limited, such pipe should be put down as will avoid, as far as possible, serious loss in the head or supply; where, on the other hand, water is abundant and a very considerable head can be obtained, a loss in this way may be justified to a larger extent to save cost of pipe.

The greatest loss in long pipes is that due to friction, which loss may be deduced approximately from the following formula:

$$h_1 = f \frac{l}{d} \frac{v^2}{2g},$$

in which h_1 = height of resistance of friction,* f a coefficient obtained by experiment for different conditions, l the length of pipe in feet, d its diameter in feet, and v the velocity of water in feet per second. The coefficient of friction, f, is not constant, but varies with the velocity and with the diameter and internal condition of the pipe.

From this it will be seen that the loss due to friction is independent of the pressure of the water; that it is proportional to the length of pipe; that it increases nearly with the square of the velocity; that it is in-

* Weisbach.

versely proportional to diameter of pipe; and that it decreases with the smoothness of the pipes and joints.

The coefficient, f, varies, according to Merriman, from 0.01 to 0.05 and is often assumed in approximate calculations at 0.02.

The following table of coefficients for smooth clean iron pipes obtained from deductions of Fanning, Smith, and others has been compiled by Prof. Merriman, and will give the value to use in any particular case, from which the probable loss due to friction may be obtained.

TABLE VIII.

FRICTION FACTORS FOR SMOOTH, CLEAN IRON PIPES.

Coefficient $= f$ in formula $h_1 = f\frac{l}{d}\frac{v^2}{2g}$.

Diameter in Feet.	Velocity in Feet per Second.						
	1	2	3	4	6	10	15
0.05	0.047	0.041	0.037	0.034	0.031	0.029	0.028
0.1	.038	.032	.030	.028	.026	.024	.023
0.25	.032	.028	.026	.025	.024	.022	.021
0.5	.028	.026	.025	.023	.022	.020	.019
0.75	.026	.025	.024	.022	.021	.019	.018
1.	.025	.024	.023	.022	.020	.018	.017
1.25	.024	.023	.022	.021	.019	.017	.016
1.5	.023	.022	.021	.020	.018	.016	.015
1.75	.022	.021	.020	.018	.017	.015	.014
2.	.021	.020	.019	.017	.016	.014	.013
2.5	.020	.019	.018	.016	.015	.013	.012
3.	.019	.018	.016	.015	.014	.013	.012
3.5	.018	.017	.016	.014	.013	.012	
4.	.017	.016	.015	.013	.012	.011	
5.	.016	.015	.014	.013	.012		
6.	.015	.014	.013	.012	.011		

The loss of head due to resistance as the water enters a pipe will vary with the form of mouthpiece employed, and may be taken as

$$h_2 = 0.5 \frac{v^2}{2g}$$

for average cases, although with a perfect bell-shaped mouthpiece this loss will be zero.

For long pipes the loss due to entrance is very slight, as compared with the loss due to friction.

The other losses which occur, such as those due to change of cross-section, angular connections, curvature of bends, and resistance of valves, are not so readily obtainable.

Where the radius of curvature is great as compared to the diameter of pipe, and few bends occur in the pipe, the loss will be small; also where conical reducers are used when changing from one diameter to another, the loss for each change will be barely appreciable and may be neglected.

Moreover, as the actual conditions are generally unknown, these latter losses will have to be neglected in ordinary computations, and the formula for the velocity will then be that obtained for pipes comparatively straight, smooth, and of essentially the same diameter.

Assuming the general formula

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

we can obtain the velocity of flow corresponding to a given hydrostatic head from $V = \sqrt{2gh'}$ if no losses occur in the pipe; but if the velocity-head of the issuing stream equal $\frac{v^2}{2g}$, the losses in the pipe will then be equal to the hydrostatic head minus the velocity-head, hence equal to $h' - \frac{v^2}{2g}$.

This loss must be equal to the sum of the losses due to friction, and to entrance (provided the lesser resistances due to curvature and other causes be neglected); therefore

$$h' - \frac{v^2}{2g} = h_1 + h_2.$$

By substituting the values previously found for h_1 and h_2 there is obtained

$$h' - \frac{v^2}{2g} = f\frac{l}{d} \times \frac{v^2}{2g} + 0.5\frac{v^2}{2g},$$

or

$$v = \sqrt{\frac{2gh'}{1.5 + f\frac{l}{d}}};$$

which is a convenient formula to use in obtaining the velocity of flow in straight pipes of uniform diameter, from which the head corresponding to this velocity may be obtained by substitution in the general formula $h = \frac{V^2}{2g}$. The head, h', necessary to overcome the resistances in a given length and diameter of pipe and to maintain the velocity, v, may be calculated from

$$h' = \frac{v^2}{2g}\left(1.5 + f\frac{l}{d}\right).*$$

If a given supply of water, Q, be required per second, the theoretical area of pipe will be $A = \frac{1}{4}\pi d^2$; therefore the velocity in the pipe will be $v = \frac{Q}{A} = \frac{4Q}{\pi d^2}$; hence the theoretical head required to maintain the flow will be

$$h' = \frac{\left[\frac{4Q}{\pi d^2}\right]^2 \times \left(1.5 + f\frac{l}{d}\right)}{2g},$$

provided the inner surface of the pipe be reasonably smooth. If an iron pipe be unprotected by any surface coating it will in time become coated with scale or lime deposits and more or less tuberculated. These deposits affect the discharge in a twofold manner: first by reducing the area of pipe, and secondly by increasing the roughness. Therefore to reduce the loss as much as possible it will be an advantage to cover the inner surface with coal-tar varnish or some other suitable coating.

In any case the velocity-head of the issuing jet will equal $\frac{v^2}{2g}$; hence if the discharge, Q, and also the area, a, of jet be known, the velocity, v, can be determined from

* For a discussion of the loss due to bends, curvature, reduction in area, resistances in valves and cocks, see Weisbach, Coxe's translation, pages 874 *et seq.*

$v = \frac{Q}{a}$; therefore $\frac{v^2}{2g} = \frac{Q^2}{2ga^2}$, and the velocity-head will then be

$$h_0 = \frac{Q^2}{2ga^2}.$$

In the determination of the value of the area, *a*, of issuing stream the general method employed is to caliper the jet at its least cross-section. By carefully ascertaining the diameter of jet for a given orifice or tube and comparing the area of latter with the area of jet there is obtained a value, C', which may be used as a coefficient in obtaining actual contraction for a given opening.

Thus if a equals the area of jet, and A equals area of circular orifice in a thin plate, there is obtained

$$C' = \frac{a}{A} = \frac{\frac{1}{4}\pi d^2}{\frac{1}{4}\pi D^2} = \left(\frac{d}{D}\right)^2;$$

where $d =$ diameter of jet and $D =$ diameter of orifice. The average value of C' thus found $= 0.62$.

If the orifice have rounded or curved edges, the contraction will be very much diminished and the coefficient will be found to vary from 0.62 to 1.0.

If the actual quantity of water which flows through an orifice in a given time be measured, there will be found, as in the case of flow over weirs, that this quantity is much less than the theoretical discharge calculated for the area of opening under the given head; therefore the theoretical discharge must be multiplied by a coefficient C in order to determine the actual discharge.

This coefficient of discharge varies from about 0.59 to 0.64 for circular orifices in a thin plate, depending upon the size of orifice and the head. For ordinary cases the coefficient of discharge, C, = 0.61 may be assumed.

It can be shown further that the velocity of flow through an orifice in a thin plate is diminished about two per cent by friction, and that the theoretical velocity must be multiplied by a coefficient to obtain the actual velocity. This coefficient of velocity C_1 will vary slightly, increasing with the head, but 0.98 may be assumed to meet most conditions.

It will be noticed that the coefficient of velocity is equal to the ratio

$$\frac{\text{coefficient of discharge}}{\text{coefficient of contraction}} = \frac{C}{C'} = C_1.$$

From these considerations it will be seen that the circular orifice in a thin plate offers another method of ascertaining the discharge. If the area of reservoir or supply-tank be large compared to area of orifice, and if the head, h, at centre of orifice in a vertical plane is large compared with the diameter of opening, the theoretical discharge may be assumed equal to

$$Q' = \frac{\pi d^2}{4} \sqrt{2gh};$$

therefore the actual discharge will be $CQ' = Q$, hence

$$Q = \frac{0.61 \pi d^2}{4} \sqrt{2gh}.$$

As it is impracticable to place the buckets or vanes of a water-motor sufficiently near an orifice to utilize the energy of the jet, short tubes, nozzles, or tips are used for this purpose, and for these separate coefficients will have to be determined.

When the discharge takes place through a short cylindrical tube whose length is about three times its diameter, it will be found that under ordinary conditions there is no contraction of the jet, but the velocity of the stream is diminished about 18 per cent; hence the coefficient of velocity C_1 may be assumed to be 0.82 for such short pipe when the inner corners are not rounded. When there is no contraction, that is when $C' = 1$, the coefficient of discharge C will equal the coefficient of velocity, since $\frac{C}{C'} = C_1$; hence the coefficient of discharge in this case will equal 0.82. It has been found that if a conical converging tube be used, the coefficient of velocity and of discharge are both very much higher than for a straight tube, and for this reason such tubes or mouthpieces are used, with certain modifications, when it is desired to utilize the energy of flow to the best advantage. From experiments by D'Aubuisson and Castel* on conical tubes with varying angles of convergence and with square corners at entrance, the coefficient of discharge attained its maximum value of 0.946 for a tube whose sides converge at an angle of $13\frac{1}{2}°$; but the coefficient of velocity increased continually as the angle increased; for a tube whose angle was 48° 50′ the coefficient of velocity was 0.984. In these experiments the tube

* Weisbach.

was $\frac{5}{8}$ inch diameter at small end, and its length 2.6 times its diameter.

The results of Castel's experiments also show that under varied heads the coefficients of discharge and of velocity were practically constant for the same mouthpiece.

Some experiments by Lespinasse on the canal of Languedoc * show the great advantage in using converging mouthpieces to effect an increase in the discharge; the mouthpieces employed were truncated rectangular pyramids 9.59 feet long, the dimensions at one end 2.4 by 3.2 feet, at the other .44 by .62 foot; their opposite faces were inclined at angles of 11° 38′ and 15° 18′, and the head employed was 9.59 feet. The experiments resulted in determining a coefficient of discharge varying from 0.976 to 0.987.

If the motor to be tested be supplied with several conical or curved mouth-pieces, it is advisable to calibrate each one in order to obtain its coefficient of discharge, C.

By inserting a gauge near the discharge, in the supply-pipe, which should be large relatively to the nozzle, the hydrostatic head may be obtained, by multiplying the gauge-pressure by 2.304 as shown hereafter. By this means the actual discharge may be obtained by noting the pressure and calculating the theoretical discharge for the given mouth-piece from

$$Q' = \frac{\pi d^2}{4} \sqrt{2gh};$$

then $Q' \times C$ will equal the actual discharge.

* Jackson's Hydraulic Manual.

If we assume the weight of a cubic foot of water to be 62.5 pounds, and the height of a column of water to be h feet, the total pressure, P, per square foot will be

$$P = 62.5h,$$

and

$$h = .016P.$$

As the pressure is ordinarily given in pounds per square inch, the above will become

$$p = 0.434h,$$

and

$$h = 2.304p.$$

This will give the available hydrostatic head corresponding to a given pressure, p, in the pipe as ascertained by the reading of a pressure-gauge inserted near the nozzle (see Fig. 70, page 168), the reading of the gauge to be taken when the water in the pipe has no velocity.

In obtaining the pressure from a gauge in order to determine the effective head available at the motor, the pipe to which the gauge is connected should be inserted in the supply-pipe near the entrance to the nozzle, at right angles to the axis of the supply-pipe, and, preferably, the latter should be tapped on one side rather than on top. If the gauge-tube be inclined toward the stream in the pipe when the water is flowing through, the tendency will be to increase the pressure-head; and if it be inclined in the opposite direction, the reverse will be the case.

If the reading of gauge be taken when the motor is running, there will be a certain diminution of head, as indicated by the gauge, due to the velocity of the water in the pipe. When the diameter of supply-pipe is large compared to that of the nozzle at discharge—which is usually the case, as the velocity and energy of the water is best utilized by such arrangement—the reduction of pressure-head is barely appreciable and may be neglected, as the error from this cause is well within the limits of the degree of accuracy attained in determining other quantities involved.

When, however, the supply-pipe and nozzle do not greatly differ in size, the velocity in the pipe approaches more nearly to the velocity at the nozzle, and the pressure-head may in such cases differ materially from the effective head.

If h equals the known pressure-head, and h_e equals the head due to the velocity, V, in the pipe, then the effective head will be

$$h'' = h + \frac{V^2}{2g};$$

the velocity v at the end of the nozzle will then be

$$v = C_1 \sqrt{2gh''} = C_1 \sqrt{2g\left(h + \frac{V^2}{2g}\right)};$$

but since the same quantity of water which discharges from the nozzle must pass through the pipe, the respective velocities will be inversely proportional to the sectional areas, and hence to the squares of their respective diameters; that is,

$$\frac{v}{V} = \frac{A}{a} = \frac{D^2}{d^2},$$

or

$$V = v \times \frac{d^2}{D^2};$$

in which V, A, and D represent the velocity of flow, area of cross-section, and diameter of pipe; and v, a, and d are the corresponding values at the outlet of nozzle.

Substituting this value of V in the expression for v, on the opposite page, there is obtained

$$v = C_1 \sqrt{2g\left(h + \frac{v^2}{2g} \times \frac{d^4}{D^4}\right)},$$

or

$$v = \sqrt{\frac{C_1^2 \times 2gh}{1 - C_1^2\left(\frac{d}{D}\right)^4}},$$

from which the effective head, $h''\left(= \frac{v^2}{2g}\right)$, may be calculated. When the diameter of pipe is large compared to the diameter of nozzle at discharge, the ratio $\frac{d^4}{D^4}$ will be very small, in which case $v = C_1\sqrt{2gh}$ will approach $v = C_1\sqrt{2gh''}$; if the ratio of sectional areas, or $\frac{d^2}{D^2}$, is less than one to ten, the error in using h for h'' will be less than one-half of one per cent, and when the ratio is less than one to twenty, h may be assumed to equal h'' within a limit of .025 of one per cent,—a greater degree of accuracy than can be obtained from the other factors involved.

As an example, a motor is supplied by a pipe two inches in diameter having a nozzle whose diameter at discharge equals half an inch, the gauge-pressure in the pipe near the entrance to the motor equals 43 pounds, and the coefficient of velocity = 0.98.

According to the exact formula,

$$v = \sqrt{\frac{C_1^2 \times 2gh}{1 - C_1^2 \frac{d^4}{D^4}}} = \sqrt{\frac{0.96 \times 64.4 \times 99.07}{1 - .96 \times \frac{(0.5)^4}{(2)^4}}}$$

$$= \sqrt{64.4 \times 95.49} = 78.41 \text{ feet per second.}$$

From the approximate formula

$$v_1 = C_1 \sqrt{2gh''}$$

we find, by assuming the effective head, h'', equal to the pressure-head, h,

$$v_1 = .98 \sqrt{64.4 \times 99.07} = \sqrt{64.4 \times 95.11} = 78.26;$$

that is, the gain in using the exact formula will only be

$$\frac{v}{v_1} = \frac{78.41}{78.26} = 1.0019,$$

or about two tenths of one per cent, which can readily be neglected without sensibly affecting the result.

CHAPTER VI.

MEASUREMENT OF ELECTRICAL POWER. GENERAL CONSIDERATIONS.

WHERE suitable current is available an electric motor may be used as a dynamometer with a very high degree of accuracy, provided the motor be properly calibrated and the efficiency factors thus obtained used in determining the power transmitted to the machine to be tested. Too frequently this is not taken into account, and we find the electrical input used as a measure of the power required to drive a given machine, irrespective of the load on the motor or its corresponding efficiency. Owing to the internal losses in the motor due to resistance of conductors, eddy currents, magnetic hysteresis and friction, it is impossible to transform into useful work all the energy supplied to the motor; therefore it is incorrect to assume the electrical power delivered to the motor as equivalent to the mechanical power absorbed by the machine under test. Especially is this true in those cases where a motor is used to transmit power very much below its rated capacity,—or rather that load for which the efficiency is a maximum.

The actual efficiency of a motor is a variable ratio for any given machine, and usually increases directly with the load.

If η = actual efficiency of motor,
P_e = electric power in watts supplied to motor,
w = useful mechanical effect of the motor, then

$$\eta = \frac{w}{P_e} = \frac{\text{Output}}{\text{Intake}}.$$

This efficiency is commonly referred to as the commercial efficiency, and is obviously less with an increase in the various losses which enter into the running of a motor.

These losses are sometimes divided and those due to eddy currents, hysteresis, and friction enter as a factor in determining the efficiency of conversion ; while the loss due to resistance in the field and armature eonductors affects the electrical efficiency. The product of these two efficiencies is equal to the commercial efficiency of the motor.

In the measurement of power the latter is of chief importance, and is easily obtained since the total losses are readily determined.

The electrical efficiencies of a motor have little value from a commercial standpoint. A machine may have a high electrical efficiency, yet on account of mechanical imperfections, friction, eddy currents, hysteresis and leakage, the net or commercial efficiency may be very low.

For our present purposes the commercial efficiency will be considered as the efficiency of the machine.

The efficiency of a motor depends somewhat upon its capacity, and we should expect the efficiency of a 50-H.P. motor to be materially greater than that of a 2-H.P. motor, both working at the same percentage of load.

This is clearly shown in Fig. 78, which represents the efficiency curves obtained from tests on a 50-H.P.

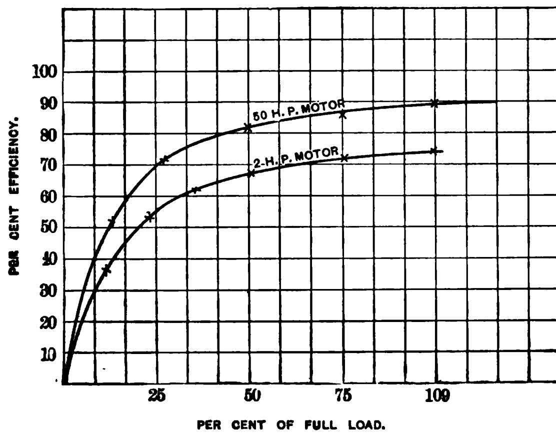

FIG. 78.—EFFICIENCY CURVES OF ELECTRIC MOTORS.

and a 2-H.P. motor respectively. Here, it will be noted, the full load efficiency in the case of the larger motor is 88 per cent, whereas in the smaller machine it is only 74 per cent; at half loads this is reduced to 82 per cent and 67 per cent respectively, while at one-fourth loads the relative efficiencies are 69 and 55 per cent.

It has been stated that the efficiency of a motor usually increases with the load. While this may be true in general, it frequently happens that the efficiency at full load is less than that at some fractional part of the load. If the work required of the motor varies between wide limits, it is evident that the most economical performance should occur at some point between no load and full load; that is, the efficiency at average load should be a maximum.

Under certain conditions the average load may be less than one-half the full load, as, for instance, in the operation of various machine tools and cranes driven by independent motors; in other cases the average load may be as high as three-fourths or even equal to full load. In any case, the economy of operation will be greatest when the maximum efficiency occurs at average load.

In Fig. 79 is shown an efficiency curve made from a test of a slow-speed motor used on an electric travelling crane. This motor was designed for an overload of 50 per cent, and it was assumed that the average load would be considerably less than full load. It will be noticed that the efficiency at full load is 79 per cent, while that at half load is about 80 per cent; between these values the efficiency rises to 82 per cent at two-thirds the full load, and diminishes on either side.

The wide range of loads with little change in the efficiencies is effected by the slow speed of the armature, which reduces the losses due to friction, eddy currents, and hysteresis; but this is obtained at the expense of a relatively low full-load efficiency due to the increased conductor resistance.

In the testing of electric motors it is customary to give both the horse-power and the torque exerted by the shaft. Torque is a convenient term to express the

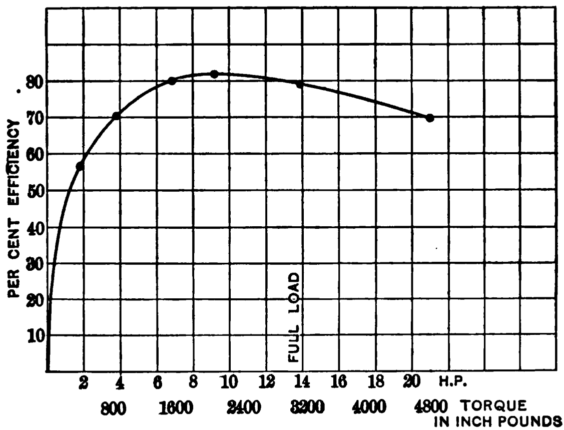

FIG. 79.—EFFICIENCY CURVE OF SLOW-SPEED MOTOR.

twisting effect, and is simply the turning moment or angular force which produces, or tends to produce, torsion around an axis. In the general expression for the performance of a shaft (see page 29) the work in foot-pounds per minute is

$$W = 2\pi RNP.$$

Let $\tau = PR$ = torque in foot-pounds,

$\omega = 2\pi N$ = angular velocity per minute, then

$$W = 2\pi N\tau = \omega\tau.$$

Since the horse-power exerted by the shaft is

$$H.P. = \frac{W}{33000},$$

we shall have

$$H.P. = \frac{2\pi N\tau}{33000},$$

hence

$$\tau = \frac{33000 \times H.P.}{2\pi N}.$$

Both mechanical energy and electrical energy may be expressed in terms of the same units, either in horse-power or in watts or kilowatts—the practical unit of electrical power.*

In the former case it is the product of the torque times the velocity, as already shown ; in the latter it is the product of the motor's electromotive force (i.e., the counter-electromotive force opposing the electro

* For the purposes of the physicist the torque may be expressed in dyne-centimeters if the force is measured in dynes and the radius in centimeters.

In the same way the speed may be given in radians per second and the work done will be expressed in ergs per second.

The practical unit of electromotive force, the volt, is one hundred million times greater than the absolute or C. G. S. unit, and the ampere is equal to one-tenth of the unit of current in the C. G. S. system ; the watt, therefore, is equal to 10^7 C. G. S. units of work per second $= 10^7$ ergs per second $= \frac{10^7}{981}$ gramme-centimeters per second. See S. P. Thomson's "Dynamo-Electric Machinery."

motive force of supply), times the current flowing in the armature.

Let $E =$ electromotive force in volts,

$I =$ current in amperes,

$P_e =$ number of watts delivered to motor;

$H.P. = 746$ watts; then

$$P_e = EI,$$

and

$$H.P. = \frac{P_e}{746} = \frac{EI}{746};$$

but

$$H.P. = \frac{2\pi N\tau}{33000},$$

therefore

$$EI = \frac{746}{33000} 2\pi N\tau = 0.0226 \times 2\pi N\tau.$$

If the speed of rotation, n, be given in seconds, then

$$EI = \frac{746}{550} 2\pi n\tau = 1.356 \times 2\pi n\tau.$$

CHAPTER VII.

INSTRUMENTS FOR MEASURING ELECTRICAL POWER.

THE determination of the efficiency of a motor involves, in its simplest form, both mechanical and electrical measurements.

The output is usually determined by means of some form of friction-brake or dynamometer which absorbs the power as already described. The power may also be absorbed by running a dynamo from the motor, which in turn receives the greater part of its electrical energy from the dynamo thus operated. This method will be described later.

The intake is ascertained by the use of electrical measuring-instruments such as galvanometers, including voltmeters and ammeters, electro-dynamometers, electrothermal and electrostatic devices, and other apparatus.

It is only of recent years that commercial testing instruments have been made of a suitable degree of accuracy to insure their reliability for the purpose of the engineer. The galvanometer in its simple form has long been in use in the laboratory, but it is not particularly well adapted to the requirements of shop testing.

The Thomson reflecting galvanometer is frequently employed when great accuracy is required and when very high resistances have to be measured; but for most other purposes the work can be more satisfactorily accomplished by the use of voltmeters and ammeters or some other instrument better adapted to the conditions.

The Thomson galvanometer* consists essentially of a very small magnetic needle, about $\frac{3}{8}$ inch long, fixed to the back of a small circular mirror, whose diameter is about equal to the length of the magnet. This mirror, which is a plano-convex lens, of about six feet focus, is suspended from its circumference by a silk fibre devoid of torsion, the magnetic needle being at right angles to the fibre. The mirror is placed in the axis of a large coil of wire, which completely surrounds it, so that the needle is always under the influence of the coil at whatever angle it is deflected. A beam of light from a lamp placed behind a screen, about three feet distant from the coil, falls on the mirror, and is reflected back onto a graduated scale placed just above the point where the beam emerges from the lamp. The scale is straight, and is usually graduated to 360 divisions on either side of the zero-point.

It is not absolutely necessary that the working zero be the middle or zero-point of the scale; it is a very common practice to adjust the instrument so that the reflected beam of light normally falls near the end of

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* For detailed description see "Handbook of Electrical Testing," by H. P. Kempe.

the scale. By this adjustment an extreme range of 720 divisions can be obtained.

The Thomson galvanometer, on account of its delicacy of construction, is extremely sensitive to vibration and to magnetic influence. Compared to the Thomson, the D'Arsonval galvanometer (Fig. 80) pos-

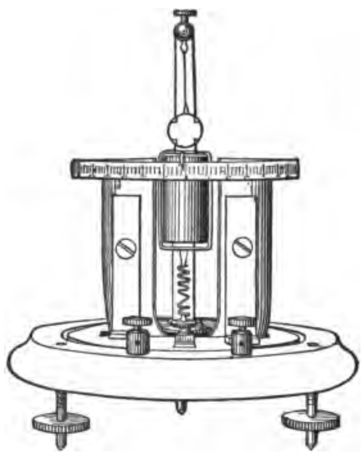

FIG. 80.—D'ARSONVAL GALVANOMETER.

sesses the great advantage that it is free from ordinary magnetic disturbances, and it is also less susceptible to vibration; but on the other hand it is less sensitive and not so satisfactory for measuring high resistances. The distinctive feature of this instrument is that the coil is movable and the magnet is fixed; whereas in almost all other forms there is a fixed coil and a movable magnetic needle.

The movable coil, which is usually wound on a light metallic frame, is suspended by a small wire or phosphor-bronze strip which forms one terminal, the other being made through a light spiral spring below. A small mirror is attached to the suspended coil which also carries a pointer free to rotate through a small angle over a graduated scale. In order to strengthen the magnetic field, a soft-iron core is placed within the coil, which is free to turn in the narrow intense field between the poles of the magnet and the iron core. As there is no iron in the movable parts, the external magnetic forces do not disturb the zero of the instrument; it can be used in any location if placed not less than ten feet from a dynamo, and it is practically independent of the earth's magnetic field. The instrument is also "dead beat," that is, the intensity of the magnetic field in which the coil is situated is such that the induced currents produce a damping effect and cause the indicator to come to rest at its final position of deflection almost instantly.

The freedom from magnetic disturbances renders this instrument particularly useful for shop measurements of resistance and location of faults in circuits and electrical apparatus.

The well-known direct-current ammeters and voltmeters of Weston are a modification of the D'Arsonval galvanometer.

In these instruments the magnetic field is produced by the poles of a strong horseshoe magnet to which a cylindrical core of soft iron (Fig. 81) is suitably connected; an annular space is left between the core and pole pieces-which constitutes the magnetic field in which the coil rotates. This coil consists of fine wire wound upon a light rectangular frame of aluminum or copper, mounted upon pivots which turn in jewelled bearings.

Flat spiral springs of a non-magnetic alloy are connected to the frame above and below it, and serve the

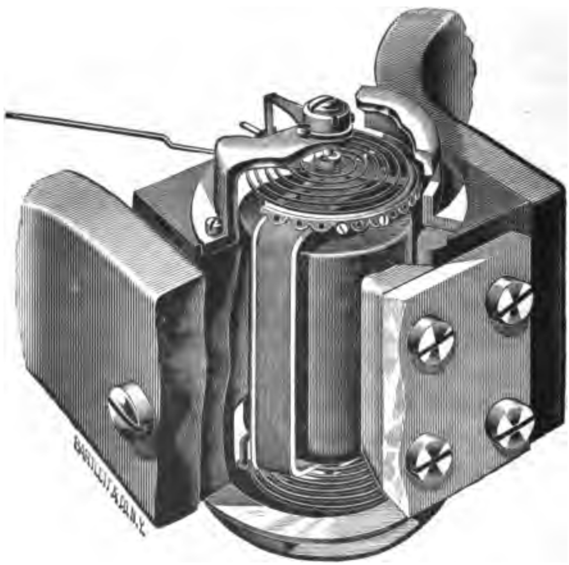

FIG. 81.—MAGNET AND COIL FOR WESTON VOLTMETER.

double purpose of conducting the current to the moving coil as well as opposing its tendency to rotate. A pointer connected to the upper end of the coil and moving over a properly graduated scale indicates the position of the coil and therefore the strength of the current to be measured.

Most of the voltmeters contain, besides the movable coil, a separate resistance coil located underneath the scale-plate.

In the ammeters a shunt is likewise placed within the case and connected to the ends of the coil.

These instruments maintain a high degree of accuracy and are convenient to use not only for practical shop tests, but for a wide range of laboratory experiments.*

Fig. 82 represents a type of Weston voltmeter

FIG. 82.—WESTON VOLTMETER.

which is very satisfactory for general purposes. Three binding-posts are shown which connect with two coils of different lengths and resistances, thus giving two grades of sensibility. The long coil of the higher resistance is used for all excepting special purposes.

In using these instruments it is to be noted that the

* See series of articles on construction and application of Weston Voltmeters and Ammeters, *Electrical World*, vol. XIX. p. 217.

voltmeter may be inserted in the circuit for any length of time without sensibly affecting the accuracy of its indications. With the ammeters, however, the independence of outside temperature is not so satisfactorily maintained; for this reason an ammeter should not be kept in circuit for any length of time if great accuracy is desired.

In general, there is no essential difference between a voltmeter and an ammeter. A galvanometer, for instance, if the resistance in its circuit remains constant, may be used to measure the current strength or voltage, since the current is proportional to the voltage.

In like manner any voltmeter which allows current to pass through it can be calibrated to indicate amperes as well as volts. Usually, however, an ammeter is made with a very low resistance, since it is placed directly in the main circuit, and any increased resistance reduces the watts in the line.

On the other hand the resistance of a voltmeter ought to be as high as possible, in order that the current taken by it shall be insignificant. However, an ordinary ammeter may be calibrated as a milli-voltmeter, because there is a certain voltage corresponding to each value of the current; that is, since the current strength, I, varies directly as the electromotive force, E, and inversely as the resistance, R, we have, according to Ohm's law,

$$I = \frac{E}{R}, \quad \text{and} \quad E = IR.$$

In a similar manner a voltmeter, unless electrostatic, may be converted into a milli-ammeter.*

If it is desired to measure larger currents than the ammeter is arranged for, a known resistance may be inserted as a shunt between the binding-posts.

If I = actual range of instrument in amperes,
I_1 = range desired,
r = internal resistance of instrument, in ohms,
R = resistance of shunt to be used; then

$$R = \frac{I}{I_1 - I} \times r.$$

Thus if $I = 150$ amperes, $I_1 = 300$, $r = 0.00022$ ohm, then $R = 0.00022$ ohm also; each scale division therefore represents $\frac{300}{150} = 2$ amperes, that is, the readings must be multiplied by 2.

Owing to the increased resistance due to imperfect contact, the length of wire for a given resistance as determined by calculation may have to be modified to obtain a desired result.

The following practical method as used in the electrical laboratory at the University of Minnesota gives accurate results and is convenient to use.

A wire of sufficient carrying capacity and any convenient length is connected in shunt around the ammeter terminals as shown in Fig. 83. In order to insure as perfect contact as possible, brass plugs are turned down on one end to fit the ammeter binding-

* See chapter on Electrical Measuring-instruments in "Electric Lighting," by F. B. Crocker.

posts; the other end is drilled and furnished with binding-screws for both the main circuit and the shunt.

The ammeter is placed in the circuit with the shunt resistance disconnected. A constant load, as, for instance, a number of lamps, is thrown on the circuit and a reading taken; it is to be observed that the load should be as large as possible within the limits of the

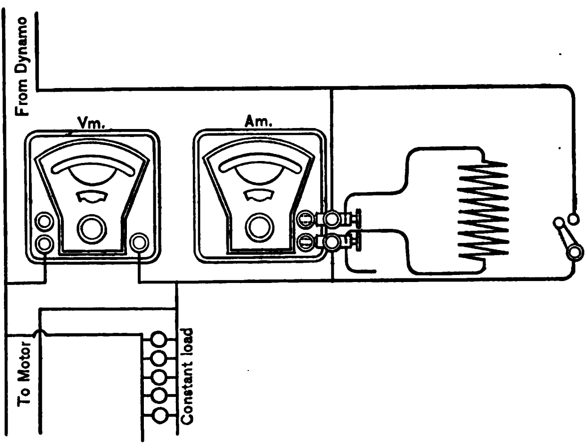

FIG. 83.—USE OF AMMETER WITH SHUNT RESISTANCE.

instrument; now the shunt wire is connected to the ammeter through the brass plugs, and adjusted in length until the reading on the instrument is a convenient fractional part of the previous reading, the voltage being the same in each case. It will be evident that, without actually knowing the magnitude of the resistance thus introduced, any subsequent readings that may be taken with the shunt in circuit will be directly proportional to the ratio assumed.

Thus if it is desired to measure a current of 120 amperes with a 50-ampere instrument, a constant load of, say, 48 amperes is first measured, then the shunt wire is inserted and adjusted till the ammeter shows only one-third the current, or 16 amperes, with the same voltage; that is, two-thirds of the current is shunted through the resistance, and the ammeter readings should be multiplied by three in order to obtain the true values.

By a method somewhat analogous to this, high resistances may be determined by the use of an ordinary voltmeter ranging from 0 to 150 volts. In this case, the voltage, E, of the line is ascertained, then the resistance, R, to be measured is inserted in the circuit, as shown in Fig. 84, and a second reading taken; if the

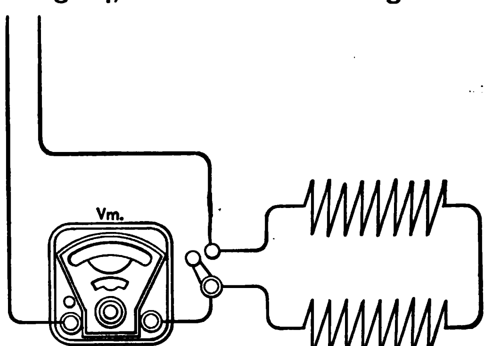

FIG. 84.—MEASURING RESISTANCE WITH VOLTMETERS.

potential difference with the resistance in circuit equal E_1 volts, then

$$R = r\left(\frac{E}{E_1} - 1\right).$$

If, for instance, $E = 100$ volts, $E_1 = 40$ volts and $r = 18000$ ohms, then

$$R = 18000\left(\frac{100}{40} - 1\right) = 27000 \text{ ohms.}$$

The resistance r is usually known for each instrument.

Among the electromagnetic devices for measuring current the ammeters used with the Brush arc machines and the earlier form of Westinghouse ammeters are familiar examples.

The latter consists of a bundle of soft-iron wires hung from one end of a lever to which a long pointer is attached; a movable counterweight serves as a balancing force when the current is being measured.

The iron core is suspended within a coil of large copper wire through which the current passes. This instrument is also adapted to alternating currents by insulating the wires in the core in order to prevent the circulation of induced currents.

A modification of this instrument is also used as a voltmeter. In the present form of Westinghouse round-pattern instruments a flat armature or vane of soft iron, to which an aluminum pointer is attached, replaces the iron core in the earlier form.

This vane extends through the interior of a heavy copper coil, and is so mounted that one edge of the vane becomes the axis of rotation, the other edge being free to oscillate.

When no current is flowing the free edge hangs toward the centre of the coil, but when current is passing the vane is magnetized and the free edge approaches the side of the coil. The amount of rotation depends upon the strength of the current, which is shown by the pointer on a graduated scale.

The steelyard ammeter, illustrated in Fig. 85, is an interesting piece of apparatus whose principle of action is similar to that of the earlier Westinghouse ammeters. In this case a soft-iron core inside the copper coil attracts an armature attached to the lever as shown. A

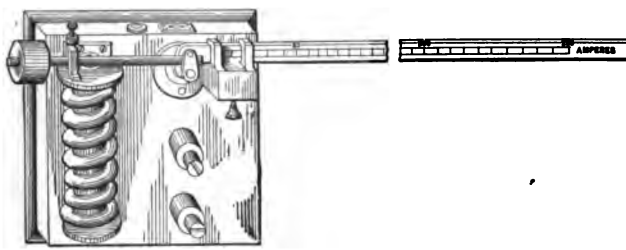

FIG. 85.—STEELYARD AMMETER.

sliding weight on the other end of the lever serves to maintain a balance when current is passing through the coil.

The advantages possessed by the various forms of electromagnetic measuring-instruments are their cheapness and simplicity of construction; but they do not indicate definitely near the zero-point, and with direct currents the hysteresis of the iron core causes the reading to be less for a given current when the latter is increasing than when it is decreasing.*

This error is not material in a station ammeter, but

* Crocker Electric Lighting, p. 418.

for accurate power measurements it is too large to be neglected.

The Cardew "hot wire" voltmeter deserves mention among the various electro-thermal devices which have been employed with more or less success. This instrument has been used extensively in commercial testing, especially in England, where it has been received with much favor.*

In this voltmeter the current is passed through a long high-resistance wire suitably arranged in a brass tube; the expansion due to heating the wire actuates a small shaft connecting through a multiplying gear to the index-finger, and a graduated dial permits a direct reading of the voltage.

The self-induction in this instrument is very small which is an advantage when measuring alternating currents. It is also dead beat, and is not disturbed by magnetic influences; but it absorbs considerable energy and is not satisfactory at low voltages.

Differences in external temperatures also affect the resistance of the wire, but this may be overcome by calibrating at extreme temperatures and using a correction factor.

Among the numerous devices which have been employed for measuring alternating currents few have been more satisfactory than the various electrostatic instruments in use. Of these Kelvin's vertical electrostatic voltmeter illustrated in Fig. 86 is one of the simplest.

* See paper on "The Measurement of Electric Current," by Jas. Swinburne, Proc. British Inst. C. E., vol. cx, 1892.

These voltmeters may be used on either direct or alternating systems, and being electrostatic they consume no energy, nor require a temperature correction,

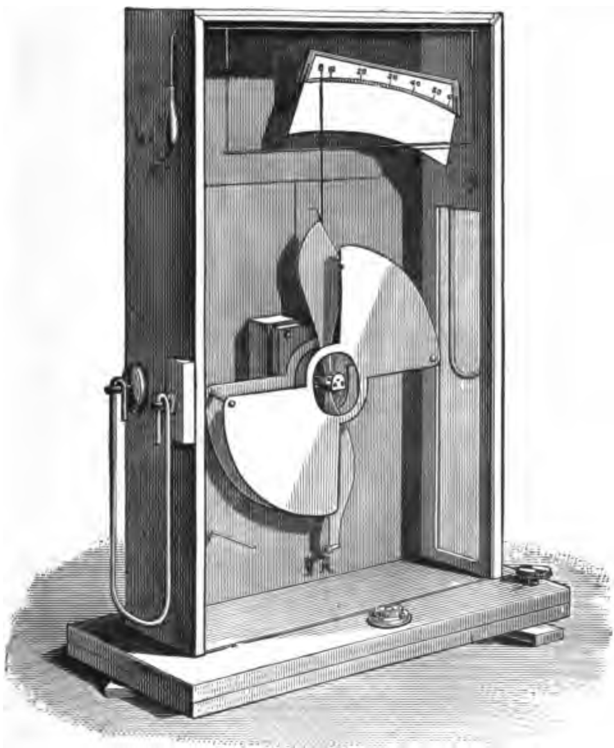

FIG. 86.—ELECTROSTATIC VOLTMETER.

as no current flows through them. They are direct-reading and uninfluenced by external magnetic fields unless rapidly alternating: in this case eddy currents may introduce an error in the readings.

The instrument is essentially an air-condenser having one of its parts movable about an axis so as to increase or diminish the capacity. The condenser is enclosed in a metal case, for the double purpose of protecting the movable part from air-currents and from the disturbing influence of any electrified body, other than the fixed portion, differing from it in potential.

It consists of two quadrant-shaped brass plates, between which is suspended an aluminum plate carrying a pointer which indicates the difference of potential existing in the plates. When the fixed and movable plates are connected respectively to two points of a circuit between which a difference of potential exists the plates become oppositely charged; mutual attraction is therefore set up, and the movable plate is deflected. The magnitude of the displacing force is proportional to the square of the potential difference, and this attraction is balanced by a series of weights hung on a knife-edge attached to the lower part of the movable plate.

This type of electrostatic voltmeter is capable of measuring differences of potential from 200 to 20,000 volts, and for this reason it is especially well adapted for high-pressure alternating currents, although it may be used with the lower voltages of both direct and alternating currents. It is to be noted that any one instrument is not adapted to the entire range; for instance, one size is designed to measure from 200 to 4000 volts, another from 1000 to 20,000, while intermediate sizes are adapted to the intermediate pressures.

For general work the Kelvin multicellular electrostatic voltmeter shown in Fig. 87 is a convenient prac-

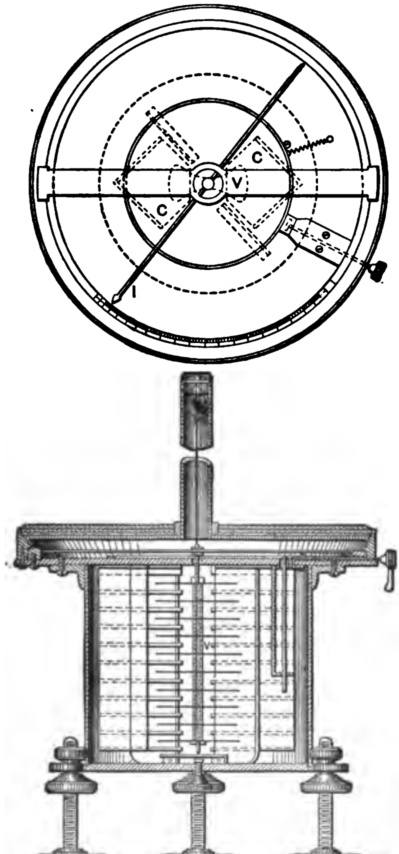

FIG. 87.—MULTICELLULAR ELECTROSTATIC VOLTMETER.

tical instrument well adapted to the needs of the engineer. This instrument is intended for measuring differences in potential, varying from 40 to 1600 volts distributed over a series of five separate ranges; it may also be used to measure current strength by connecting it in shunt to a known resistance, but this latter is not entirely satisfactory. In this instrument a number of vanes, *V*, arranged in two sets, are placed parallel to each other upon a spindle which is suspended by a fine iridio-platinum wire from a torsion head at the top of a vertical brass tube. These vanes move freely between insulated cells, *C*, formed of brass plates in metallic connection with each other and arranged in equidistant horizontal planes; an aluminum pointer, *I*, attached to the top of the spindle indicates the difference of potential between the movable and stationary portions of the instrument by direct readings in volts.

If a single cell were used, the suspending wire would have to be very fine or very long—an undesirable feature in either case. The use of several cells produces sufficient torque to permit the employment of a suspension wire of practical dimensions.

The action of the instrument is identical with that of the vertical electrostatic voltmeter already described. In certain forms of this instrument a vane is attached to the moving armature and immersed in oil; this forms a damping device and prevents too violent vibrations of the pointer.

Another arrangement of this voltmeter has a horizontal dial similar to that shown, to which an inclined mirror is attached, thus permitting readings to be made at a distance from the instrument.

A very satisfactory instrument in many respects for laboratory or shop use is the Siemens electro-dynamometer, shown in Fig. 88.

This apparatus, like other instruments already discussed, may be used for measuring both direct and alternating currents.

FIG. 88.—SIEMENS ELECTRO-DYNAMOMETER.

The construction of the electro-dynamometer is based upon the mutual action of currents upon one another and is illustrated in Fig. 89. *AB*, *ab* are two rectangular coils placed at right angles, one of them being fixed to the framework and the other free to turn about a vertical axis. The stationary coil, *AB*, is within the movable one, which latter is suspended by a thread or supported on a pivot; electrical connection with the movable coil is made through mercury-cups into which the ends of the coil dip. Two stationary coils are generally used, one consisting of a few turns of large wire, the other of a greater number of turns of finer wire. One end of each is attached to the binding-posts 1 and 2 respectively, Fig. 89, and the other ends are connected

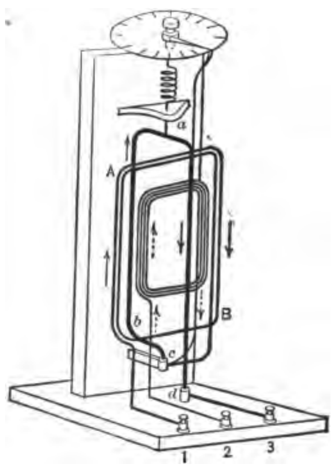

FIG. 89.—DIAGRAM OF ELECTRO-DYNAMOMETER.

to the mercury-cup *c*; a conductor leads from the lower mercury-cup, *d*, to a third binding-post, 3, thus completing the circuit whether the current enters through the heavy or the light wire; in any case it passes out through the movable coil *ab*. When currents are sent through the two coils the magnetic stresses tend to bring them into the same plane; this tendency in the present case is resisted by a spiral spring one end of which is attached to the suspended coil, the other being connected to a knurled head which is provided with a pointer that may be moved about a graduated dial-plate. A zero indicator is also connected to the movable coil.

When the moving coil is deflected the knurled head is turned until the pointer on the coil stands at zero; the number of degrees or divisions of the dial may then be read from the torsion-indicator and the strength of the current determined. Since the same current flows through both coils, the resultant magnetic effect between them must be proportional to the product of the current in each; that is, the force tending to turn the moving coil varies as the square of the current; the force of the spring varies directly as the angle through which it is turned: therefore the current strength is proportional to the square root of the angle through which the torsion-head is turned.

Every instrument must be calibrated and a table of values of current strength made out for various angles of torsion.

If θ = angle of deflection,
I = current in amperes,
k = constant determined by experiment, then

$$I = k\sqrt{\theta}.$$

It is advisable to recalibrate the instrument from time to time, for, while good springs are practically permanent, a poor one, according to Swinburne,* may change as much as one per cent in a month's time.

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* Electrical Measurements, by Jas. Swinburne, p. 44.

In using the electro-dynamometer care should be taken that it is kept accurately levelled and free from outside disturbing magnetic influences. It should therefore be so placed that when the suspended coil swings freely its plane lies in that of the magnetic meridian. The Siemens electro-dynamometer is, for moderate currents, a very accurate instrument when used with care, but the loss of time in taking readings is a serious disadvantage, for in every case the moving coil must first be brought to zero, the amount of torsion noted, and then the reference table consulted to determine the current strength. An additional disadvantage is that the instrument is not susceptible of great accuracy when the readings are low.

For these reasons the Siemens electro-dynamometer has in many cases been superseded for shop use except for the measurement of alternating currents, in which field it still occupies an important position.

It is evident from what has preceded that if both the fixed and the movable coils be wound with fine wire, and a resistance connected in series with the instrument, the latter will be sensitive to small currents and may be used as a voltmeter.

The Weston alternating-current voltmeter is a modication of this instrument, in which the coils are so shaped that the movable coil may be rotated through a wide angle, while a pointer attached to it travels across a quadrant scale which gives a direct reading of the voltage.

The electro-dynamometer may also be wound and used as a wattmeter. Since the magnetic force of each coil is proportional to the current through it, if one coil be wound with a great many turns of fine wire and connected in shunt across the circuit so that the current received by it shall be proportional to the voltage, and, further, if the other or heavy wire be connected as before, in the main circuit, so as to carry the total current, the torque will be proportional to the product of the voltage times the current flowing in the circuit, that is, proportional to the watts.

For power and efficiency tests in direct-current work the use of the wattmeter is not essential, since in any case the power in watts may always be obtained by multiplying the readings of the voltmeters and ammeters; but with alternating currents this is generally not the case; the instant of maximum current rarely coincides with that of maximum pressure, since, in a reactive circuit, the current is caused to lag behind or lead the pressure.

The average value of the watts expended in a circuit during a complete cycle is equal to the average of all the instantaneous products; therefore in order to obtain the true watts the product of the instantaneous volts and amperes must be integrated over a complete cycle.

If E = electromotive force in volts,

I = current in amperes,

P_e = watts expended in an alternating circuit,

ϕ = angle of lag,

then it can be shown that $P_e = EI \cos \phi$, that is, the true watts are obtained by multiplying the apparent watts, as determined from the voltmeter and ammeter readings, by a factor which is equal to the cosine of the angle of lag.* If the true watts be divided by the apparent watts, a quotient will be obtained which has been called the "power factor" of the circuit; it is evidently equal to cosine ϕ. However, it is not necessary to determine the angle of lag, since the true watts are obtained directly by the use of a wattmeter. In an ordinary wattmeter the force acting on the moving coil at any instant varies, as we have seen, directly as the product of the instantaneous pressure and the instantaneous current, so that if the moving coil has too much inertia to vibrate with the varying force, the average force will vary as the average product of the electromotive force and current. A wattmeter, assuming there is no self- or mutual induction, therefore measures the power, or true watts.

The Siemens electro-dynamometer used as a wattmeter is a very useful piece of apparatus, but its readings need correction. There are two principal causes of error which limit the application of this instrument: the first, which is the same for continuous as for alternating currents, is the loss in the wattmeter itself; the second is that due to the self-induction of the shunt circuit when used with alternating currents.

A third error is that due to the disturbing influences of external magnetic fields.

In connecting a wattmeter the shunt or pressure coil may be connected to the line ahead of the series or

* See Kapp's Alternating Currents of Machinery, p. 43; also Jackson's Alternating Currents, p. 109.

current coil, as shown at *a* in Fig. 90, or it may be connected on the other side of the current coil, at *c* as in Fig. 91; in either case if the power delivered to the work circuit *ce* is to be determined the reading will be

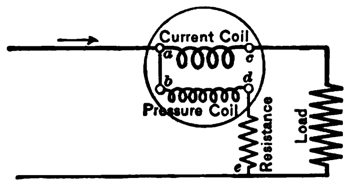

FIG. 90.—WATTMETER CONNECTIONS.

too high, since the wattmeter really measures the watts expended in the circuit as well as that spent in one of its own coils. The error will, however, be negligible except in the measurement of small powers, but

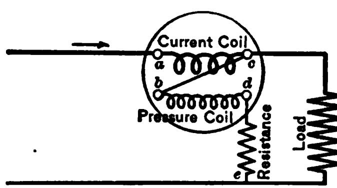

FIG. 91.—WATTMETER CONNECTIONS.

it may be practically eliminated by connecting the pressure coil at *c*, as in the second case, and employing a compensating coil which will neutralize the field to an extent proportional to the current in the pressure coil. Thus if a fine wire in series with the pressure coil, and having the same number of turns as the current coil, be wound around the latter from *c* towards *a*, Fig. 92, so that the current flow will be in opposite directions, the readings obtained by the wattmeter will now indicate the number of watts delivered to the work circuit,—if we neglect the inappreciable loss of current in the resistance coil, which,

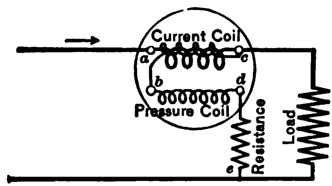

FIG. 92.—COMPENSATING COIL.

however, may be readily calculated, and subtracted from the reading.

If on the other hand, it be desired to measure the power received from a generator or other source of electrical energy, the connections should be made as in Fig. 90, which reduces the error to the same extent as in the previous case; but under these conditions the readings must be increased.

If the wattmeter has been calibrated with direct currents the readings will not be correct on an inductive circuit, but with proper precautions the error from this cause may be made very small if the voltage is not too low.* It is necessary that the self-induction of the pressure coil be small as compared with its resistance; the current coil, therefore, must have a strong field, and the moving coil should be wound with compara-

* *L'Electricien*, March 3, 1894; also Swinburne in Proc. Brit. Inst. C. E., vol. cx.

tively few turns of fine wire and connected in series with a high non-inductive resistance.

In order to eliminate any errors due to magnetic disturbances when measuring direct-current power, the instrument should be placed in a fixed position and readings taken with the current flowing in one direction; then by reversing the direction in both the current and pressure coils and taking another reading, the correct value will be the mean of the two readings.

It is advisable to keep the instrument at least 10 feet away from large iron masses, such as dynamos, running or idle, transformers, steam pipes and radiators, or from bus-bars carrying heavy currents.

Various modifications of the electro-dynamometer wattmeter have been constructed, notable among which are the Weston portable and the Thomson inclined-coil. These are indicating wattmeters and are especially adapted to the requirements of the engineer, since they can be used on either direct- or alternating-current circuits, and are essentially dead-beat instruments.

The Weston wattmeter consists of a stationary current coil wound with heavy wire and made in two sections which are separated a short distance from each other. A circular pressure coil, composed of fine insulated wire, is suspended between the sections of the field coil. Fastened to opposite vertical diameters of this circular coil is a steel pivot to which the inner end of a non-magnetic spiral spring is attached in a manner similar to that used in the Weston voltmeter, Fig. 80.

When no current is flowing these springs will keep the coil in a certain zero position, from which it will be deflected when current is caused to pass through the coils.

In using the wattmeter the terminals of the pressure coil are connected in shunt to the circuit, while the current coil is connected in series. When current is passing, the mutual action of the coils produces a torque which is proportional to the product of the currents in each coil; as the current in the pressure coil is proportional to the voltage, it is evident that the torque produced will be proportional to the watts expended in the circuit—assuming no self-induction in the shunt coil. A pointer is connected to the upper end of the moving coil, and a graduated dial is calibrated to give a direct reading in watts.

To secure quick readings and to prevent injury to the pivots of the pressure coil by reason of violent shocks to the moving system, caused by suddenly closing the circuit, the instrument is provided with a damping device. This consists of a light aluminum disk mounted on the axis of the moving coil and acted upon by a spring-brake. A button is provided which frees the pointer only when fully depressed; when the contact is released the needle is held at the point of last indication and permits very rapid readings to be taken.

In order to eliminate any error due to self-induction the instrument is provided with a non-inductive low-capacity resistance, which consists of a number of sheets of mica wound with a thin strip of high-resistance alloy wire connected in series with the pressure coil, the whole being contained in the wattmeter-case. The alloy used has a very low temperature coefficient, so that the instrument may be left in circuit, if desired, without appreciably affecting the accuracy of its indications.

The current coil is provided with a compensating coil

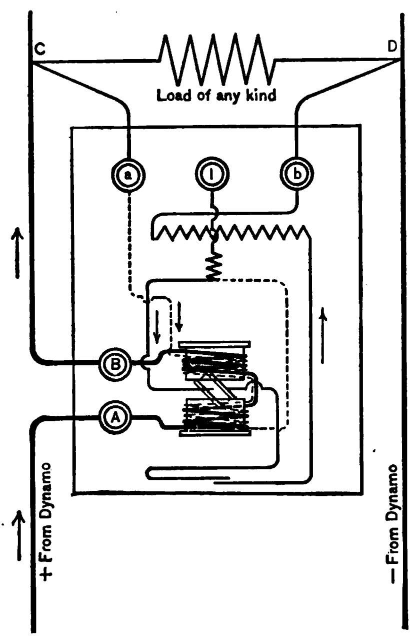

FIG. 93.—DIAGRAM OF WESTON WATTMETER.

as shown in diagram Fig. 93. This consists of a fine wire wound over the current coil in series with the pressure coil as previously explained. An independent binding-post, *I*, permits this compensating coil to be cut out when it is desired to use the instrument with two independent circuits, or when two separate sources of current are used for calibration. In this case an equivalent resistance may be substituted for the compensating coil.

The load, of whatever nature, is connected across the mains from *C* to *D*.

With high-pressure currents *C* should always be connected to *A* or *B*; in fact it is generally advisable to make the connections in this way instead of joining *D* with *A* or *B*, since this latter will bring full pressure between the voltage coil and the current coil in the instrument. If it is necessary to reverse the current, it is preferable to change *A* and *B* instead of crossing *C* and *D*.

The range of a wattmeter may be extended by inserting in series with the pressure circuit a separate non-inductive resistance called a "multiplier." This does not increase the current-carrying capacity of the instrument, but simply affects the voltage. In using a multiplier with a wattmeter it is a matter of importance that the multiplier be connected between the voltage coil and the circuit to be measured so that there will be practically no difference of potential between the voltage coil and the series coil.

Such an arrangement is shown in Fig. 94. If the multiplier were connected in on the same side of the circuit as the series coil, there would be a serious difference in potential in the wattmeter coils, which if excessive would destroy the instrument.

Another method of using a wattmeter on a high-potential circuit is shown in Fig. 95, in which the voltage coil is connected to the low-pressure coil of a wattmeter-transformer, the primary of which is connected so as to receive the full voltage of the circuit. In

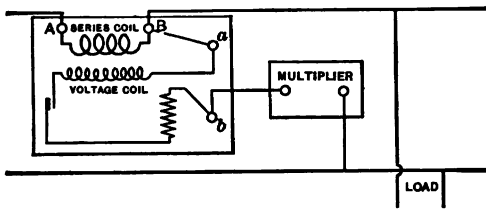

FIG. 94.—MULTIPLIER WITH WATTMETER.

order to insure that no dangerous difference of potential exists between the two circuits of the wattmeter, Professor Ryan * has shown that they should be con-

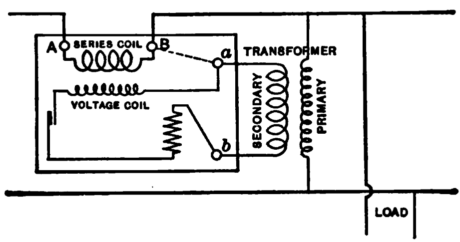

FIG. 95.—WATTMETER-TRANSFORMER.

nected by a balancing-wire as indicated by dotted lines in the figure.

In certain classes of commercial work it is very desirable to obtain a continuous record of the power in watts delivered by a generator or supplied to a motor or other circuit. To meet this requirement various forms of recording or integrating wattmeters have been

* Bedell in *Sibley Journal of Eng.*, vol. XI. 237.

designed, many of which are highly successful in operation and admirably adapted to their purpose.

There are, broadly speaking, three classes of electrical-energy meters of this character which depend upon or are actuated by (1) chemical action; (2) clockwork or weights; (3) electric currents. In the first case a definite portion of the current to be measured is shunted through an electrolytic cell and the amount of metal deposited, or the quantity of electrolyte decomposed, gives a measure of the current passing. While this form of meter is accurate, it possesses the disadvantage that the plates have to be weighed; there is no recording feature, nor can it be used except on continuous currents.

In the second class a clock is made to vary its time by the action of the current on a magnet attached to the pendulum. The best known form is the Aron, in which two clocks are used, one of which keeps accurate time, while the other is accelerated by the action of the current. The difference in time of the two clocks integrates the work done and gives a measure of the power. In a later form the action of the current winds the clock so that no attention is necessary for this purpose. These meters are direct-reading and are said to be very accurate when used on continuous-current work.*

The most important class is included under the third heading and comprises those meters which operate as motors. Of these the Thomson, Shallenberger, and Duncan are well-known instruments.

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* See Wordingham on Meters in Proc. Inst. C. E., vol. CX.

The Thomson meter may be regarded as a form of electro-dynamometer so arranged that instead of merely indicating power it integrates it with regard to time, and in this way measures the amount of energy that has passed. The general principle of all such meters is that of employing a small fraction of the power to be integrated to produce motion in an armature. If the velocity of the armature can be made exactly proportional to the total energy, then the number of revolutions of the armature in a given time is a correct measure of the power.

In this meter, Fig. 96, two fixed coils of heavy conductor are placed one on either side of a drum armature which is wound without any iron in its core. The main current passes through the fixed coils in series, and the armature forms a part of the high-resistance pressure-circuit which is shunted across the mains.

A non-inductive resistance is also placed in the pressure circuit so that the meter may be used with either direct or alternating currents. Since there is no iron in the magnetic circuit, the driving torque is proportional to the product of pressure and current and is therefore proportional to the watts. If the retarding force were strictly proportional to the speed, this would give correct readings at once, but friction at the bearings introduces a disturbing influence and must be counteracted. This is effected by superior mechanical construction of the bearings, and also compensating what remains by means of an additional permanent driving torque, produced by inserting a fine wire field-coil in series with the pressure-circuit, so that with no current passing through the series coil, but with the pressure-coil connected in the circuit, the friction is exactly balanced. The vertical shaft of the armature receives the current through a small silver commutator and two delicate brushes. Attached to the lower end

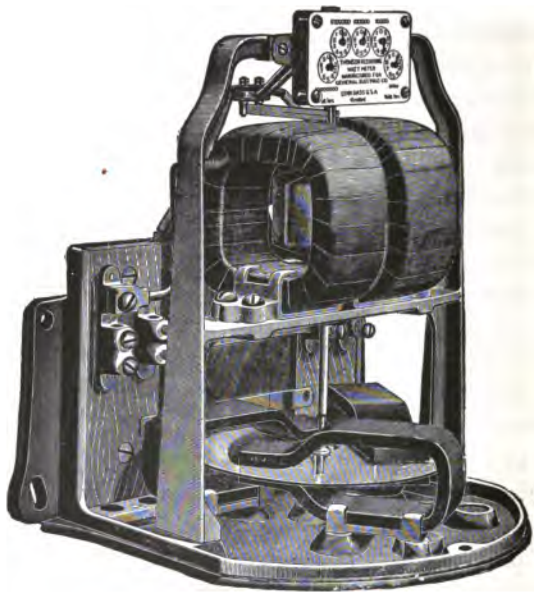

FIG. 96.—THOMSON RECORDING WATTMETER.

of the spindle is a large copper disk, mounted horizontally so as to rotate between the poles of permanent magnets as shown. The reaction between the eddy currents induced in the disk and the magnetic field acts as a brake and constitutes the retarding force. The recording mechanism gives the reading in watts if used with a varying potential, or if used on constant potential mains it may be arranged to read directly in ampere-hours.

Another instrument which has been largely used on the latter service is the Shallenberger ampere-hour meter.

This instrument, Fig. 97, is essentially a small induction or rotary field motor, in which the moving parts, while having no contact with the actuating circuit, are carried around by the rotating magnetic field which surrounds the iron-ring armature on the motor shaft.

The meter is designed to be connected directly in series with the circuit, and the entire current to be measured passes through a few turns of heavy wire, called the primary coil, within which, and at an angle to it, is placed a closed copper coil, known as the secondary; inside of this is a thin metallic disk mounted upon an upright spindle, connected at its upper end with a train of recording gears, and at the other with a set of four aluminum fan-blades.

When an alternating current passes through the primary coil, an alternating field is developed in the direction of the axis of the primary coil. At the same time a current is induced in the secondary coil, and this induced current develops another field in the direction of the axis of the secondary coil, that is to say, at an angle to the first. These two alternating fields combine to produce a resultant field, but as the alternations of the two are not coincident in time, the direction of the maximum effect of the resultant field is constantly shifting or moving in a circle, more exactly an ellipse,* and what is termed a revolving field is produced.

The metallic disk and its spindle tend to revolve in

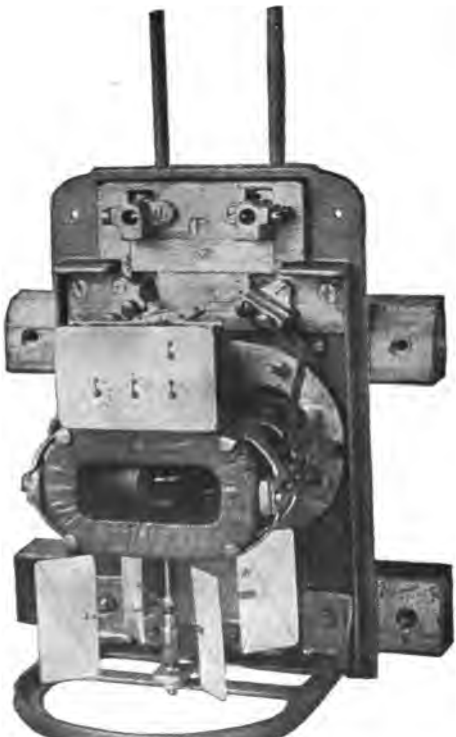

FIG. 97.—SHALLENBERGER AMPERE-HOUR METER.

unison with the rotary field, the torque or driving force being proportional to the square of the current.

* See Polyphase Electric Currents, by S. P. Thompson, p. 64.

The fan-blades produce a retarding effect proportional to the square of the speed, so that, omitting bearing-friction, the driving and retarding forces balance each other at all speeds; that is, the speed of the meter varies directly as the current.

On account of the general introduction of alternating-current motors taking currents which may not be proportional to the power delivered, the Shallenberger integrating wattmeter was designed.

In this meter, Fig. 98, a thin aluminum disk constitutes the movable element, in which eddy currents are induced by the current to be measured. This latter current passes through stationary coils placed below the disk, and a shunt current, proportional to the

FIG. 98.—SHALLENBERGER INTEGRATING WATTMETER.

electromotive force, passes through another coil located above the disk.

The design of the meter is such that the torque exerted upon the disk is always proportional to the product of the current and electromotive force multiplied by the power factor (the ratio of true energy to apparent energy). In other words, for any power factor the torque varies directly as the power transmitted through the circuit.

As the torque is proportional to the power passing the meter (and not to the square of the current), the wattmeter starts with less current than is required in the current meter.

Two permanent magnets embrace the disk between their poles and exert a retarding force directly proportional to the rate of rotation. The speed is therefore proportional to the power.

The constant of the instrument is not affected by changes of temperature, since the retarding eddy currents induced by the permanent magnets are affected by changes of the temperature of the disk in exactly the same ratio as those induced by the actuating currents, so that no change of speed results.

In order that an induction wattmeter may operate accurately on loads having different power factors, the shunt field must be exactly in quadrature with the shunt electromotive force. With the usual form of inductive shunt circuit this is impossible, as the loss in the shunt circuit can never be made zero. With favorable conditions it may be made very small compared with the apparent energy expended. Meters have been made in which the ratio of true loss to apparent loss in the shunt circuit was 0.09, or the lag of shunt current was nearly 85 degrees behind the shunt electromotive force. With this shunt circuit a meter will register with fair accuracy on any current commonly met with in commercial practice. Several methods have been used to secure a greater degree of accuracy; that generally employed in the Shallenberger meters consists of a small closed secondary within the shunt coil, which is adjustable either in position or in resistance. Currents induced in this secondary lag in phase behind the magnetic field inducing them, and the new component added to the original field produces a resultant field which, by adjustment of the secondary, may be brought in exact quadrature with the shunt electromotive force. Thus the effective field of the shunt coil is made to lag behind the current in that coil.

CHAPTER VIII.

METHODS OF MEASURING THE POWER OF DIRECT-CURRENT MOTORS.

It has been stated that in order to determine the efficiency of a motor under varying loads it is necessary to ascertain both the intake and the output, and it has been shown that the commercial efficiency is equal to the ratio:

$$\frac{\text{output}}{\text{intake}} = \eta = \frac{w}{P_e}.$$

It is evident that the difference between the intake and the output equals the losses, so that the efficiency may also be represented by the ratio:

$$\frac{\text{intake} - \text{losses}}{\text{intake}}.$$

Various methods have been adopted for measuring the losses separately, but in most cases the results are only approximate.

The losses which have to be determined are those due to conductor resistances in armature and fields; eddy currents; hysteresis and friction of various kinds, including brush and air resistance as well as that due to the bearings. Besides these there is a loss due to commutator brush-contact resistance which varies inversely with the current density; in low-voltage machines (110 to 125 volts pressure) this may reduce the efficiency from $1\frac{1}{2}$ to 2 per cent.*

The loss in the armature conductors varies as the product of the square of the current times the resistance in the wire $= I^2R$; this is not great for light loads but increases rapidly with the load. The same is true of the loss in the field-coils; but if the pressure in the field-coils of a shunt-wound machine is not constant, the loss will vary with the difference of the square of the voltage.

The losses due to eddy currents increase with the square of the speed, while the other losses, including hysteresis and friction, are assumed to vary directly with the speed.

As far as friction is concerned this may be a correct assumption, but in many cases it is far from exact.

In certain machines the fields are so placed as to exert a magnetic attraction on the armature shaft which tends to reduce the weight on the bearings; as the intensity of the field varies this pull varies also. In other cases the conditions are exactly the reverse of this.

An unsymmetrical field whether produced by design or wear has often a marked effect upon the friction losses at different loads; in such cases it is incorrect to

* *Electrical World and Engineer.* Vol. XXXIV. pp. 406, 417.

assume that this loss is constant. Moreover the character and condition of the lubricant, influenced by the temperature of bearings, materially affects the coefficient of friction and hence the friction losses.

The various losses due to eddy currents, hysteresis, and friction are frequently classed as stray power. Since it is difficult to separate these losses with any

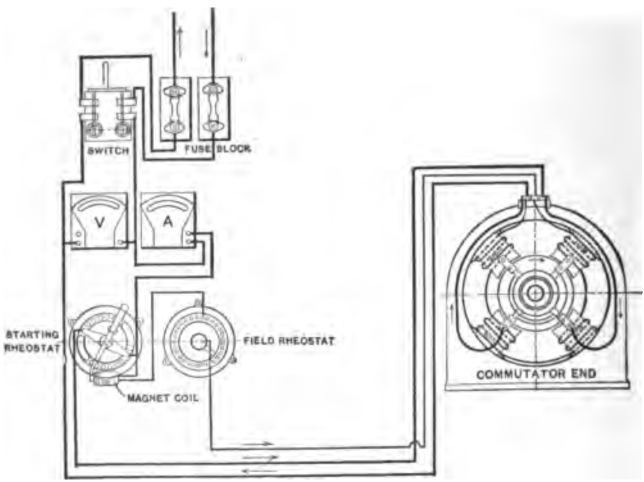

FIG. 99.—CONNECTIONS FOR TESTING DIRECT-CURRENT MOTORS.

degree of certainty, it is better to group them together and determine the total losses due to the stray power and conductor resistance.

Usually it will be found most practicable to determine the output directly by means of some form of friction brake or dynamometer. This method is generally susceptible of greater accuracy than any other, and involves the simplest measurements of the electrical power given to the motor and the mechanical power delivered by the motor; it also permits the efficiency to be determined independently of belt-pull or increased bearing friction due to the latter, which should not be charged to the motor. In testing a direct-current motor by means of a friction dynamometer, a voltmeter is connected across the mains, while an ammeter is placed in series as shown in Fig. 99. The belt-pulley is furnished with a Prony brake or other friction dynamometer, and the motor is run under varying loads as desired. Readings are taken from the voltmeter and ammeter at stated intervals, while at the same time the speed of the machine and the weight acting on the dynamometer are determined.

Since the electrical input in watts is equal to the product of the electromotive force in volts times the current in amperes, and since 746 watts equals one horse-power, we have

$$\frac{EI}{746} = \text{intake in horse-power};$$

the brake load or output in horse-power equals $\frac{2\pi LNP}{33000}$ (see page 29); therefore

$$\eta = \frac{\text{output}}{\text{intake}} = \frac{2\pi LNP}{33000} \times \frac{746}{EI} = 0.142\,\frac{LNP}{EI},$$

in which $L =$ radius of brake-arm,

$N =$ number of revolutions of armature per minute,
$P =$ pressure in pounds acting on brake-arm,
$E =$ electromotive force in volts,
$I =$ current in amperes.

Before taking readings for any given load the motor should be run at such load for a sufficient length of time to obtain the heat effect, which materially influences the result.

The resistance of both armature and field-magnet coils changes with the temperature, so that an increase of heat means a larger conductor loss. Hysteresis and eddy-current losses also vary with the heat of the machine; the friction of the shaft may or may not be increased, depending largely upon the character of the lubricant; in any case it is important in testing the motor that it be warmed up so as to approach the normal conditions under which it is to operate.

The Committee on Standardization, appointed by the American Institute of Electrical Engineers,* recommends that all losses should be measured at or reduced to the temperature assumed in continuous operations, or in operation under specified conditions; the standard conditions of room-temperature being 25° C. (77° F.) under a barometric pressure of 760 mm. (29.92 inches). If the room-temperature during the test differs from 25° C., the observed rise of temperature should be corrected by ½ per cent for each degree C.

* Transactions, vol. XVI; 1899.

It is always advisable to run the motor with both increasing and decreasing loads, since the effect of residual magnetism in the field, influenced more or less by temperature variations, will often produce a slight difference in the input for a given brake-load.

The voltage at the motor terminals is maintained constant at its normal pressure by means of a suitable resistance placed in series with the main circuit; or, if the current is obtained direct from an independent dynamo, the voltage may be regulated by varying the strength of the dynamo fields.

If it is desired to determine the efficiency of the motor by measuring the losses, instead of the output, it will be necessary to ascertain, first, the conductor losses; and secondly, the stray power; the sum of these represents the total loss, which, subtracted from the intake, will give approximately the power delivered at the motor pulley.

Let P_c = loss in conductors in watts = I^2R;
P_s = stray power in watts;
R = total conductor resistance;
R_a = conductor resistance in armature;
R_f = conductor resistance in field-coils;
I = total current delivered to motor;
I_a = current in armature in amperes;
I_f = current in field-coils in amperes.

With these assumptions the general formula for efficiency will be

$$\eta = \frac{\text{intake} - \text{losses}}{\text{intake}} = \frac{EI - (P_c + P_s)}{EI}.$$

In order to determine P_c it is necessary to know the armature and field resistances R_a and R_f; of these, the former may be found by disconnecting the armature from the circuit and sending through it a strong current while at rest. A suitable resistance, such as a water rheostat, should be placed in the external circuit so as to limit the current to the safe carrying capacity of the armature.

The drop in potential, measured across the armature terminals, divided by the current in amperes will give the armature resistance; that is $R_a = \frac{E}{I}$. For a low-resistance series field-coil R_f is obtained in a similar manner; but for a shunt field-coil it is not necessary to place any resistance in the circuit, since its own resistance is comparatively high. If I_a represents the current flowing through the armature, and I_f that in the field-coils, then

$$\begin{aligned} P_c = I^2R &= I_a^2R_a + I_f^2R_f \\ &= I^2(R_a + R_f) \end{aligned}$$

for a series-wound motor, and

$$\left(I - \frac{E}{R_f}\right)^2 R_a + \frac{E^2}{R_f}$$

for a shunt motor, since

$$I_a = I - \frac{E}{R_f}, \quad \text{and} \quad I_f = \frac{E}{R_f}.$$

The armature conductor loss, if temperature variations be neglected, equals I^2R_a and may be calculated for any given load when the current is known; on the other hand, the field conductor loss for a given excitation will be practically constant.

If the motor be run at no load, the stray power, P_s, will equal the total watts supplied minus the conductor losses, or

$$P_s = EI - P_c.$$

The stray-power losses are assumed to be approximately constant for constant speed, so that the total losses may be obtained by adding the calculated conductor losses to the stray power thus determined for any known load.

Usually with constant-potential motors the speed varies somewhat with the load, and under these conditions a correction is frequently made for the stray-power losses which are affected by the speed.

A convenient method of obtaining the effect of speed variation, proposed by Kapp and also by Housman, consists in separately exciting the field-magnets to a constant value; then the motor is run without load at different speeds by varying the voltage of the current delivered to the armature. By noting the corresponding values of speed, voltage, and amperes, a curve may be plotted, with the speeds as abscissæ and the watts as ordinates, which represents in a simple manner the combined eddy currents, hysteresis, and frictional losses when the motor is running without load.

The values thus obtained are to be added to the I^2R losses when the speed corresponding to the current I is known; from which the total power and the efficiencies may be calculated.

A more satisfactory method is to operate the motor as a dynamo on open circuit by means of an accurately calibrated motor-dynamometer, the fields being separately excited to their normal value as in the previous case. By running the motor in this way at various speeds within the ordinary limits, and noting the power required, a series of values of the stray-power losses will be obtained, which may be plotted into a curve for use in determining the efficiency from the previous equations.

However, where the speed variation is not excessive this is a refinement which is questionable in many cases, since it is probable that the hysteresis and friction losses increase somewhat with a heavier load and thus offset any gain due to fall of speed.

The method of testing motors by means of two or more similar machines, first proposed by Hopkinson,* may often be used with very satisfactory results. In this method two machines of similar size and type are coupled together mechanically, either by direct connection or by belt; one machine acts as a dynamo and generates power that is delivered electrically to the other machine, which runs as a motor and assists in driving the dynamo. The losses are made up by supplying power from an outside source.

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* Phil. Trans., 1886. Reprints.

The extra power delivered to the system is measured by means of a transmission-dynamometer, and as this is relatively small compared to the total power of the motor, the results will be sensibly accurate if ordinary care be exercised.

Kapp's modification of this method consists in supplying electrical power from another source to balance the losses in the two machines. In this case (Fig. 100)

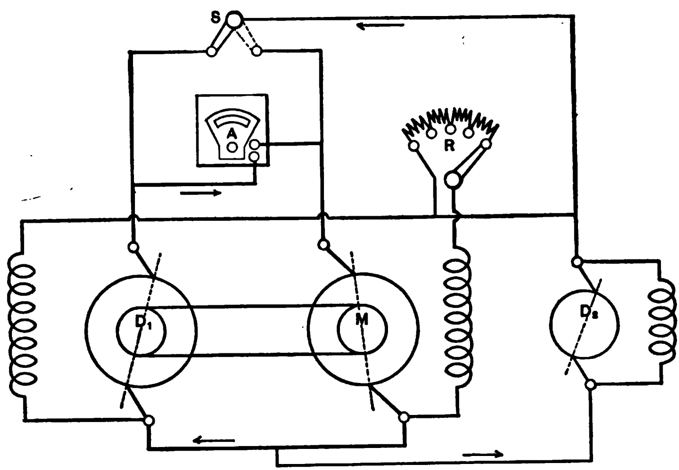

FIG. 100.—KAPP'S THREE-MACHINE METHOD.

two machines of similar type and approximately the same power are coupled together so that D_1 runs as a dynamo and supplies current to the motor M; an auxiliary machine, D_2, of the same voltage is connected in parallel to the other machines and furnishes current for making up the difference in current between D_1 and

M, besides exciting the fields.* A rheostat is used to weaken the field of *M* so it may run as a motor. An ammeter is inserted in the circuit from one brush of D_1 to one of *M*, and a switch, *S*, permits the supply from the auxiliary circuit to be led in on the right and left respectively. Readings are taken with the switch first on one side and then on the other; as the voltage is the same in each case, the ratio of the two currents is the efficiency of the combination of the two machines. Assuming the losses to be the same in each machine, the efficiencies will be equal: the square root of the ratio of the two readings will therefore be the efficiency of either machine, since the final efficiency, in any case where two efficiencies are involved, is equal to the product of the separate efficiencies.

The method employed in the General Electric Company's works at Schenectady is an obvious modification of the above and gives a convenient arrangement for starting which is a matter of considerable importance. In this case the voltage of the three machines is regulated by rheostats introduced into each field-circuit. The machines are placed as in Fig. 100.

An ammeter and starting rheostat are located between the loss supply D_2 and motor *M*; a second ammeter is connected through a switch placed in the circuit between the motor *M* and dynamo D_1.

The loss supply D_2 having been brought up to voltage, the motor *M* is started.

This necessarily runs the armature of the dynamo D_1

* S. P. Thomson's Dynamo-Elect. Machines, p. 760.

to which M is connected by a belt; D_1 then picks up, generating the field-current. The field-rheostat is then regulated so that the voltage across the brushes of D_1 is about three volts higher than that of the motor and the loss supply. When the difference of potential across the switch is about three volts, if the speed is correct, the switch is closed and the machines "pump back" on each other. Then to bring the machines up to the desired load the motor-field is weakened and the rheostats otherwise adjusted to maintain the proper speed.

All types of direct-current machines may be tested in this manner by varying the conditions somewhat to suit each case.

Machines as large as 2100 kilowatts capacity have been run at full load in this way with an expenditure of only 400 horse-power of supply.*

In other cases 100 horse-power of supply have been used to test to full load at the same time about 1000 horse-power of electrical machinery.

A very satisfactory method which permits all of the measurements to be made electrically, but does not necessarily involve current of the same kind nor equal voltage, is the employment of an additional motor, of any convenient size, which has been previously calibrated so that it may be used as a transmission-dynamometer.

In this case (Fig. 101) two similar machines are coupled together as in the Hopkinson method, prefer-

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* Th. Straus in *Electrical Engineer*, vol. XXII, p. 272.

ably by direct connection, and power is supplied from one to the other, assisted by the auxiliary motor which furnishes an amount of power equal to the combined losses in the two machines. The watts delivered to the motor-dynamometer, multiplied by its efficiency under

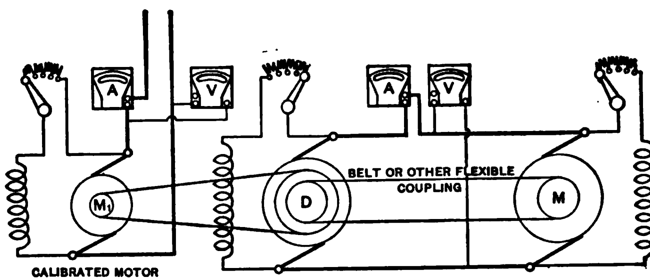

FIG. 101.—USE OF CALIBRATED MOTOR.

the given conditions, is evidently the power absorbed, from which the efficiency of either machine may be readily determined when the voltage and current in the motor-dynamo circuit is known.

Various methods are in use for testing motors by weighing the torque without using a friction brake, which is accomplished by means of certain forms of dynamometers; of these the Brackett cradle- and Webb floating-dynamometers are well suited to the purpose.

Their use involves obvious modifications of the preceding electrical methods together with those mentioned in the descriptions of the machines already given on pages 106 to 118.

In testing railway motors a method frequently adopted consists in connecting two similar machines to one axle by gears, in a manner similar to that employed when the motors are permanently set up in the cars. One of the machines is then run both in forward and reversed directions as a motor, while the other, which acts as a generator, sends its current into a water-rheostat, thereby loading the motor. After running this way for some time under different loads and speeds, during which the motor is given a heavy overload, the combination is reversed, and the machine which was previously run as a motor now becomes the generator and the test is conducted as in the first case.

The water-rheostats employed for such purposes are of various forms, consisting essentially of a water-tight barrel or box in which two iron plates are suspended in a very dilute acid bath or salt-water solution, either or both plates being adjustable so as to vary the extent of immersion or distance apart. Cast-iron plates are probably the most satisfactory, but ordinarily heavy sheet iron or boiler-plate is used as being more generally available.

The tank or box may be either of iron or wood. A cast-iron box suitably mounted on porcelain insulators makes a good tank which is free from leakage, but care must be exercised in handling the plates lest they be short-circuited; on this account a wooden tank is generally preferred. The common form of barrel rheostat, with horizontal plates submerged in salt water, is not entirely satisfactory for heavy currents, owing to the liability of the under side of the top plate to become covered with a gaseous scum which makes the resistance quite variable.

A perforated plate tends to prevent the accumulation of the gas; if, in addition to this, carbonate of soda (washing-soda) be used instead of sodium chloride (common salt), better results will be obtained.

Mr. E. J. Willis suggests that the electrodes be made from galvanized-iron sheets rolled and riveted into a cylindrical form, the diameter of inside cylinder being about six inches less than the outer.*

The size of tank for a given load depends largely upon the quantity of water which is used. If an unfailing supply of salt water is at hand, as for instance a dock adjacent to the power house, the current density may be as great as 100 amperes per square foot of active surface of both plates.

Such a source of supply is not, however, often available and a lesser current density must be used. As the amount of salt in the water is increased from zero to saturation the maximum current-carrying capacity is also increased, but owing to the rapidly changing resistance of the plates and the convenience of varying the resistance by increasing or decreasing the distance between them, any rules for size of water-rheostat based upon area of plate and percentage of solution are susceptible of widely varying results.

A water-rheostat is simply a means of dissipating heat, and its capacity is therefore dependent upon the quantity of heat that may be taken up by the water in a given time.

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* *American Electrician*, vol. IX. 1897, p. 468.

This may be predetermined when the total watts to be dissipated and the range of temperature are known.

Since the energy expended in raising one pound of pure water 1° Fahr. at or near 60° F. is equal to 778 foot-pounds, it follows that the energy consumed in raising the temperature of 57 pounds of water per minute 1° F. is equivalent to one kilowatt—44,220 foot-pounds per minute.

If w = pounds of water supplied per minute;
T_1 = initial temperature of water in deg. Fahr.;
T_2 = final temperature of water in deg. Fahr.;
P_w = power in kilowatts; then

$$w = 57 \frac{P_w}{T_2 - T_1}.$$

Assuming an initial temperature of 70° F. it will be seen that if a constant temperature of 184° F. be maintained in the tank it will require one-half pound of water to be supplied per minute for every kilowatt of energy dissipated.

A low temperature will permit a slow evaporation in the form of a hardly distinguishable vapor; whereas a high temperature may result in a violent ebullition with disagreeable fumes and irregular action. In the first case it is only necessary to supply sufficient water to take the place of that evaporated. Where the boiling is excessive it is advisable to deliver more water than is evaporated, in order to lower the temperature; the excess water being allowed to pass off through a suitable overflow. If the amount of water running through is large, the solution may become too weak; in this case a very small quantity of saturated solution, added gradually to the tank, will permit a fairly constant resistance to be maintained, for any length of time. Under the first-named conditions a given water-rheostat will carry only one-half the current that can be comfortably taken care of with running water. In the latter case about 15 kilowatts per square foot of total plate surface can be easily handled provided about two cubic feet of tank volume be allowed for every square foot of plate surface.

Where a more uniform load is desired than is obtainable with a water-rheostat an immersed wire resistance may be used. This consists of German silver or copper wire wound upon a wooden frame which is immersed in a vessel of running water. By controlling the supply of water so that a constant temperature is maintained within the tank a sensibly constant resistance will be obtained. Since the resistance increases about one per cent for every five degrees (Fahr.) rise in temperature, it will be evident that within given limits the load may also be varied by regulating the temperature of the water. A number of such rheostats made of about 300 feet of No. 12 bare copper wire are used in one of the power stations of the Consolidated Traction Company of Pittsburg, and it is found that each will safely dissipate 120 kilowatts — 225 amperes at 550 volts.

CHAPTER IX.

METHODS OF MEASURING THE POWER OF ALTERNATING-CURRENT MOTORS.

IT will be readily understood from what has preceded that the measurement of alternating currents involves much more complication than obtains with direct currents, since the power is not directly proportional to the product of the readings of the volt and ammeter placed in the circuit. (See page 253.)

With periodically varying currents and pressures the product of the current and electromotive force at any instant is the rate at which energy is being expended on the circuit at that instant.

By placing a non-inductive resistance in series with the circuit to be measured, the instantaneous values of electromotive force and current may be determined by one of the various methods of instantaneous contacts.*

If the values of the current and pressure thus obtained be plotted as in Fig. 102, the products of the corresponding ordinates will represent the power curve, as shown in P, P', P'', the mean ordinate of

* See Nichols' Laboratory Manual, vol. II. Trans. A. I. E. E., vol. IX. p. 179; vol. X. p. 503.

which will give the average value of the power expended in the circuit in each cycle $= \frac{ei}{2} \cos \phi$, where e and i are maximum instantaneous values of the current and electromotive force during a complete period.*

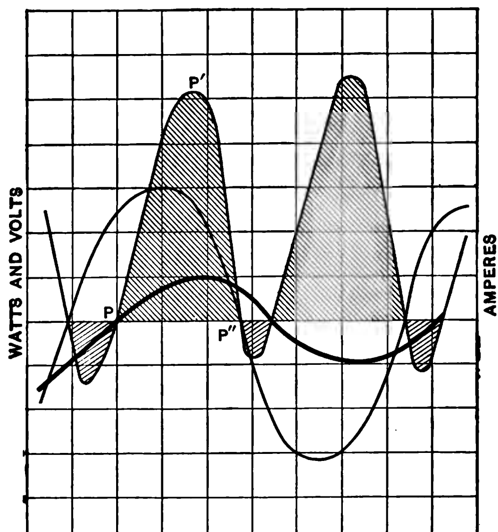

FIG. 102.—VOLTAGE, CURRENT, AND POWER CURVE.

While measurements of instantaneous values of alternating currents are very useful in furnishing data to the designer, or for special investigations, they have little value in commercial power determinations, since such measurements may be obtained more satisfac-

* Fleming, Alternate Current Transformer, vol. I. p. 124.

torily and conveniently by other methods. Among these the 3-voltmeter method has been used to some extent, but it has some disadvantages in practice, since a small error in the readings may produce a comparatively large percentage error in the results; moreover it involves a source of current having an electromotive force practically twice as great as that in the circuit to be measured.

The method consists essentially of a non-inductive resistance, such as a bank of incandescent lamps, placed in series with the inductive circuit to be meas-

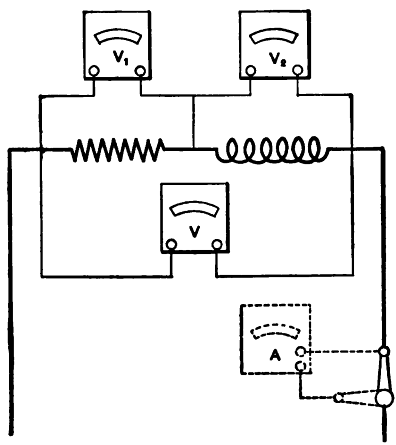

FIG. 103.—THREE-VOLTMETER METHOD.

ured. Three electrostatic or other voltmeters suitable for measuring alternating currents are connected up as in Fig. 103, and readings taken of the difference of potential, E_2 in the inductive circuit; E_1 in the non-inductive circuit, and E in the two combined. If

$I =$ current in amperes obtained by an electrodynamometer or ammeter, then the mean power in the circuit will be

$$P_e = \frac{I}{2E_1}(E^2 - E_1^2 - E_2^2).$$

The most satisfactory results will be obtained when E_1 is about equal to E_2. If the value of the non-inductive resistance is known, the current need not be measured, since $I = \frac{E_1}{R_1}$; therefore

$$P_e = \frac{1}{2R_1}(E^2 - E_1^2 - E_2^2).*$$

In order to overcome some of the disadvantages of this method Prof. Fleming suggested the use of three ammeters or low-pressure electrodynamometers connected as in Fig. 104, the power in watts given to the circuit being measured by the three ammeter readings.

If the resistance R_1 is known, then the power will be

$$P_e = \frac{R_1}{2}(I^2 - I_1^2 - I_2^2).$$

If R_1 is unknown, the electromotive force E may be obtained by placing a voltmeter in the circuit; in this case, since $E = R_1 I_1$, we have

$$P_e = \frac{E}{2I_1}(I^2 - I_1^2 - I_2^2).$$

* See paper by Prof. Ayrton and Dr. Sumpner before Physical Society, London, 1891; also *Electrical World*, vol. XVIII, 1891, p. 131.

The most favorable condition for accuracy is that in which the current I_1 is nearly the same as I_2.

In this method, as in the previous one, slight errors in the instruments may introduce quite large errors in the final values, since the formulas contain a function

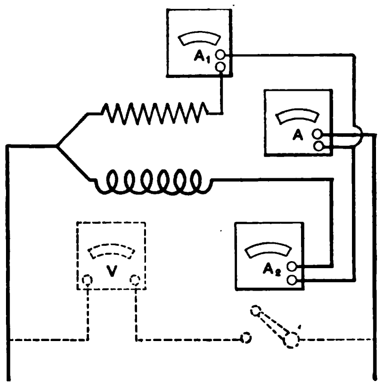

FIG. 104.—THREE-AMMETER METHOD.

which is the difference of the squares of several quantities.*

In order to eliminate some of the errors in these methods various modifications have been used,—sometimes two voltmeters and one ammeter, and again one voltmeter and two ammeters,—but none of these methods are altogether free from error. Probably the latter combination, in which a voltmeter is used in parallel with the non-inductive resistance, will give the most satisfactory results. In this method (Fig. 105) $\left(\frac{E}{R_1}\right)^2$ replaces

* Fleming, *Electrician*, May 8, 1891; also *Phil. Mag.*, Aug. 1891, p. 204.

I_1^2 in the three-ammeter method, and the power is therefore

$$P_e = \frac{1}{2}R_1\left[I^2 - \left(\frac{E}{R_1}\right)^2 - I_2^2\right].$$

R_1 is obtained by opening the circuit at S_2 and measuring the current through the non-inductive resistance together with that through the voltmeter; the reading of the latter instrument divided by the current in amperes gives the resistance R_1.

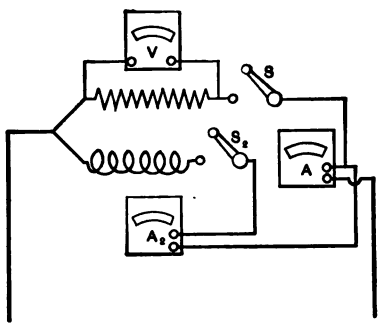

FIG. 105.—COMBINED VOLT- AND AMMETER METHOD.

A switch placed at S permits the readings of the ammeters to be compared with each other.

These methods, are often impracticable for commercial testing, as it may be impossible to obtain the conditions which are necessary to accurate measurements.

By far the most satisfactory results are those obtained by the use of a wattmeter placed in the circuit, which, as we have shown (page 254), gives the true power direct in watts. The methods of using this instrument, previously discussed, apply to alternating as well as continuous currents.

In testing single-phase motors the output should preferably be determined by the use of a friction dynamometer, in a precisely similar manner to that employed with direct-current motors. In this case, however, the intake should be measured with a wattmeter, for, owing to the current lag, the product of the effective electromotive force and current, as determined by the voltmeter and ammeter readings, does not give the true watts, but is almost invariably too high.

A voltmeter and ammeter may, however, be used together with the wattmeter, as shown in Fig. 106, in order to determine the power factor, which is the ratio of the true watts to the apparent watts; that is, since the true watts or power $= EI \cos \phi$,

$$\cos \phi = \text{power factor,}$$
$$= \frac{\text{true watts}}{EI}.$$

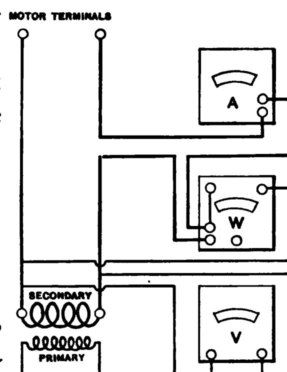

FIG. 106.—CONNECTIONS FOR SINGLE-PHASE MOTORS.

If, instead of a motor circuit, the load should consist of a number of incandescent lamps, the angle of lag is so small that $\cos \phi$, that is, the power factor, practically equals unity.

The various losses in the motor may be determined by simple modifications of the stray-power methods employed in direct-current work, in which purely electrical measurements are made.

However, for commercial determinations of power and efficiency it will be found more practicable, and usually more accurate also, to measure the output directly by means of a friction dynamometer.

In testing a two- or three-phase motor some complication exists.

In a two-phase system with independent circuits, as in Fig. 107, the power may be readily determined by

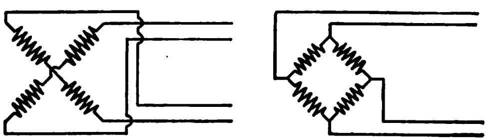

FIG. 107.—STAR AND MESH CONNECTIONS, FOUR-WIRE TWO-PHASE CIRCUIT.

connecting a wattmeter in each of the circuits and noting the readings; the sum of the watts in the two circuits is a measure of the real power in the system. If the circuits are balanced, the power will be twice that in either circuit, but this condition cannot be depended upon; therefore two wattmeters, or their equivalent, must be used. One instrument, however, may be connected successively in each circuit provided it be done rapidly so that the load will be practically constant during the two readings. It can be shown * that the

* See articles by A. D. Lunt, *Elect. World*, vol. XXIII, 1894, p. 771 *et seq.*; also M. Blondel in Proc. Electrical Congress, Chicago, 1893, p. 112; also Görges in *Elect. Rev.*, London, vol. XXVIII.

real power expended in a polyphased system is equal to the summation of a series of quantities, each found by obtaining the mean product of the current in each line conductor by the difference of potential between the terminal of that conductor and the terminal of some one conductor nominally selected for reference. Thus in a system of n wires there will be $n - 1$ such products necessary to represent the real power. In order then to obtain the real or mean power expended in the system, it is only necessary to ascertain the mean value of each of the separate products by the use of a wattmeter, or any other means of measuring the real watts in a simple alternating current.

Moreover, since in any polyphased system the algebraic sum of the currents at any instant is equal to zero, therefore in the two-phase 3-wire system, and in the three-phase 3-wire systems, Fig. 108, the current in any one wire is equal to the algebraic sum of the

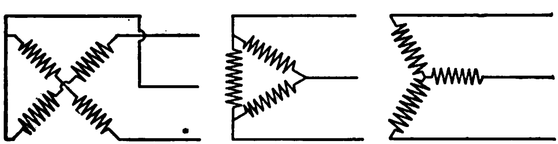

FIG. 108.—THREE-WIRE CIRCUITS.

currents in the other two. It thus becomes apparent that the real power of either system may be measured in identically the same manner, so that in each case, given the three line wires and the three terminals, the measurement may be carried out with entire disregard as to whether a two- or three-phased system is being dealt with.

Thus in Fig. 109 two wattmeters are inserted in the circuits *A*, *B*, *C* in the two-phase 3-wire system; and in like manner two wattmeters are placed in the three-phase circuits *A′*, *B′*, *C′*.

In both cases the connections are identical, and the

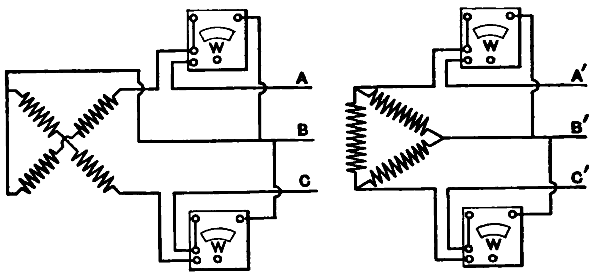

FIG. 109.—TWO-WATTMETER METHOD.

algebraic sum of the wattmeter readings in each system will give the real power in that system irrespective of the number of phases.

In the two-phase system if the series-coil of the wattmeter be connected to the outside wires and the return to the pressure-coil as shown, the total power will also be equal to the arithmetic sum of the watts in each leg. On the other hand, in the three-phase system the total power will be equal to the arithmetic sum only when the power factor is greater than 0.50, that is, when ϕ is less than 60°. If the angle of current lag is just equal to 60°, one of the wattmeters will read zero, and the other will give the total power in the system; for angles of lag greater than 60° the readings of one of the wattmeters will be negative; in this case the connections between the terminals of the series-coils will have to be reversed: the arithmetic difference of the two readings will then give the real power in the system.

If it is not readily apparent which condition exists, this may be ascertained by sending a continuous current through the wattmeter and noting the sense of the deflection; if the same relative connections be observed in subsequent use of the instrument, opposite deflections will be negative and must be subtracted. For proper relative connections see Fig. 93.

In order to ascertain the apparent watts and the power factor for different loads it will be necessary to insert a voltmeter and an ammeter in each circuit in a manner similar to that employed in the case of single-phase alternating currents. The power factor will then equal the total real power divided by the total apparent power. It rarely happens that a sufficient number of instruments are available for this purpose, but this is not a very serious difficulty, for one set of instruments may be connected to the circuits so as to give all the data required, and if the loads are fairly constant during the readings accurate results will be obtained.

For temporary or occasional testing a very satisfactory arrangement is that given in Fig. 110.

The illustration shows the connections for both two- and three-phase systems. Referring to the three-phase as being perfectly general, it will be seen that the transformer secondaries are connected with the motor terminals through a number of mercury-cups so arranged as to permit either direct connection with

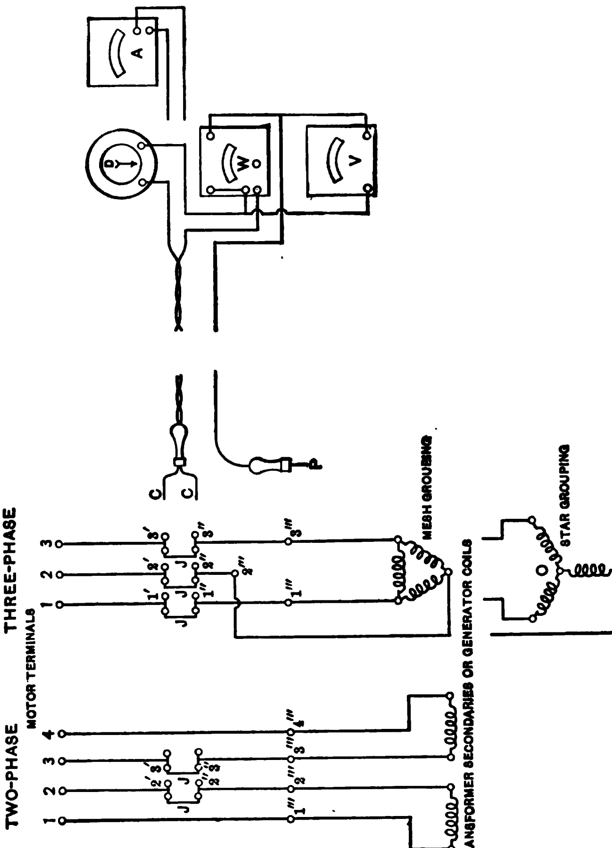

Fig. 110.—General Method with One Set of Instruments.

the motor, or through the instruments, as desired. Flexible conductors are used between the instruments and the circuits, to which they are successively connected when readings are to be taken; jumpers of suitable size ordinarily constitute a part of each circuit, but are to be removed when the instruments are connected in. Where great accuracy is required it is advisable to insert a small resistance in the jumpers during the test in order to balance the resistances in the instruments.

While two wattmeter readings are sufficient to determine the real power, the arrangement shown readily permits a simultaneous reading from each of the instruments, so that the apparent watts as well as the true watts may be obtained at once if independent measurements be made for each of the three circuits.

It will be noticed that one terminal of the pressure-circuit is connected to the current conductor and therefore is always in the line whose current is to be measured. If the current conductor, *C*, be connected in any two of the lines successively, and the pressure-terminal inserted in the remaining line, the algebraic sum of the wattmeter readings thus obtained will give the true watts in the system. Thus if the pressure-terminal *P* be connected in $2'''$, while the current terminals *CC* are inserted in $1'-1''$ and $3'-3''$ successively, the respective wattmeter readings will give a measure of the true power in the three circuits. Since it is immaterial which two of the three circuits be taken for the series-coils connections, it is evident that three groups of wattmeter readings may be obtained, and these should be equal to each other.

In order to obtain the apparent watts in the system it is necessary to determine the sum of the products of the current in each line by the voltage between the lines, which must be divided by $\sqrt{3} = 1.73$. That is, the apparent watts

$$= \frac{\text{sum of } EI \text{ products in the three circuits}}{\sqrt{3}}.$$

With the connections as shown in the diagram (Fig. 110), readings may be obtained with the current conductor, C, successively inserted in $1'-1''$, $2'-2''$, and $3'-3''$, while the pressure terminal, P, is successively connected across each of the remaining circuits.

By grouping the readings in tabular form it will be seen that six sets of values are obtained, three of which are sufficient to determine the power; but as the remaining values afford a check on the work it is desirable to take all of the readings, as little additional labor is involved.

The various connections are as follows:

C in line No. 1, and P in 2 or 3, gives effective current in 1 and voltage between 1 and 2 or 1 and 3.

C in line No. 2, and P in 3 or 1, gives effective current in 2 and voltage between 2 and 3 or 2 and 1.

C in line No. 3, and P in 1 or 2, gives effective current in 3 and voltage between 3 and 1 or 3 and 2.

It will be observed that the above grouping also contains the connections for the wattmeter terminals.

By noting the wattmeter readings in each case and suitably grouping the corresponding values the true watts may be readily determined from any one of the three sets of values thus obtained. In combining the various readings it is only necessary to observe that each set of values should be selected with *C* connected alternately in two of the lines, while *P* is inserted in the third.

The following grouping of wattmeter-readings corresponds to the connections as above:

C in 1 and *P* in 2 (voltage between 1–2) " " 3 " " " 2 (" " 3–2)	algebraic sum of readings gives true watts.
C in 2 and *P* in 3 (voltage between 2–3) " " 1 " " " 3 (" " 1–3)	algebraic sum of readings gives true watts.
C in 3 and *P* in 1 (voltage between 3–1) " " 2 " " " 1 (" " 2–1)	algebraic sum of readings gives true watts.

While the true watts in a 3-phase system are thus obtained directly from the wattmeter-readings, it is necessary to divide the products of the effective currents and pressure by $\sqrt{3}$ in order to determine the apparent watts—unless an available neutral point, or its equivalent, will permit readings to be obtained in the coils themselves instead of in the line wires.

That this is true whether the generator coils are joined in star or in mesh grouping will be readily understood from the following diagrams, Figs. 111 and 112. If *a*, *b*, and *c* represent the voltage in the three coils pO, qO, and rO in a balanced three-phase system, Fig. 111, the pressure between the lines 1 and 2 equals that between the coils measured across the terminals *p* and *r*; moreover, it is evident that the difference of potential between p and r is equal to the resultant $OR = 2a \cos 30^\circ = a\sqrt{3}$, which is 30° in advance of a.

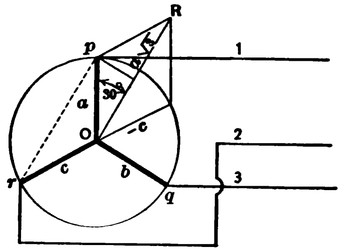

FIG. 111.

The line current with this grouping is evidently the same as that in the coil to which it is connected; therefore if $E =$ the effective voltage measured across the lines, and $I =$ effective current in the line, then

$$\frac{EI}{\sqrt{3}} = \text{apparent watts in the circuit.}$$

Considering the mesh grouping it will be seen, Fig. 112, that the voltage between the lines is the same as

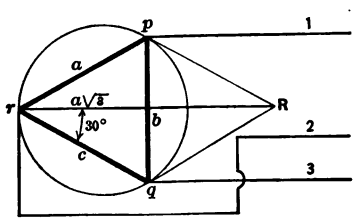

FIG. 112.

that existing in the coil to which the lines are joined; the current in the line in this case, however, will be equal to the resultant of the current in two adjacent coils. If a, b, c represent the currents in the coils, the current in line No. 1 will equal $rR = 2a \cos 30^\circ = a\sqrt{3}$ as before; that is, the line-wire current is 1.73 times as great as the current in the coil. Thus it will be seen that, in star grouping, the voltage in the line circuit is greater than at the coil terminals, while the current is the same; with mesh grouping, on the other hand, the current in the line wire is greater than in the coil, but the voltage remains the same. In either case the apparent watts in any circuit connected as in Fig. 110 will equal $\frac{EI}{\sqrt{3}}$.

In a three-phase balanced system connected as above, the total apparent watts will equal

$$P_e = \frac{3EI}{\sqrt{3}} = \sqrt{3}EI;$$

in an unbalanced system

$$P_e = \frac{\text{sum of } EI \text{ products}}{\sqrt{3}}.$$

Where the instruments may be inserted directly in the coil circuits of the machine, or transformer secondaries, or where a neutral point is accessible, the sum of the wattmeter-readings in each of the three legs will give the true power in the system; in the same way the sum of the products of the effective current and voltage in each coil will give the apparent watts,—since in each case the current and electromotive force in the individual coils is measured independently. If in Fig. 110, for instance, the common point *O* in the star grouping be accessible, and if the current-conductor of the wattmeter be inserted successively in the three circuits 1, 2, and 3, while the pressure terminals be connected across *O*—1, *O*—2, and *O*—3, respectively, the sum of the readings thus obtained will give the true watts; so also the sum of the products of the effective current in the line, as measured by an ammeter or an electrodynamometer, multiplied by the voltmeter-readings, taken across the ends of each coil, will give the apparent watts in the system.

Where much testing is to be done it will be found advisable to erect a substantial testing-table equipped with suitable instruments so connected that any desired combination may be obtained conveniently and rapidly by means of switches and plugs.

Such a testing outfit is shown in Fig. 113, which represents the arrangement used by the Westinghouse Electric and Manufacturing Company at their works in East Pittsburg.

The switches shown in the diagram are double-throw, double-pole, dynamo-changing switches, but for convenience of illustration they have been drawn as knife-switches; *Sm* and *Sm* being arranged so that they may be short-circuited across the ends if desired. By a proper arrangement of the switches and plugs the instruments may be connected to any leg as desired,—care being taken that the remaining legs are closed meanwhile.

Either two- or three-phase motors may be calibrated with connections as shown. Ordinarily in the case of two-phase motors the two windings of the primary are

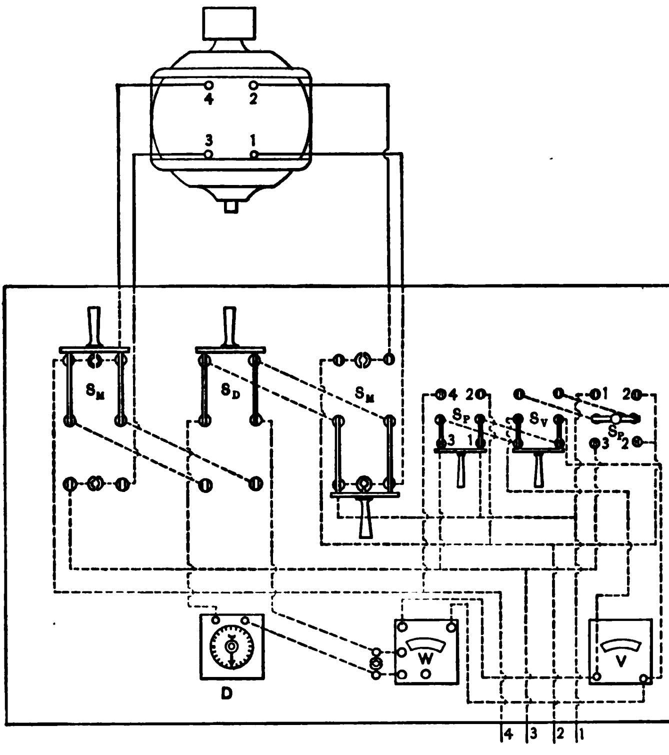

FIG. 113.—TESTING-TABLE FOR TWO- OR THREE-PHASE CIRCUITS.

entirely distinct and separate. The apparent watts then equal the sum of the products obtained by multiplying the current in each circuit by the voltage of that circuit. The true watts are obtained as before by taking the sum of the wattmeter readings in the two circuits. For a three-phase motor the connections are precisely similar to those previously given in Fig. 110, and the measurements are made in the same way. In each case the total current equals the sum of the effective currents in the three legs divided by $\sqrt{3}$.

The connections are

$$\left\{\begin{matrix} 1(1-3), \\ 2(2-3), \end{matrix}\right\} \text{ or } \left\{\begin{matrix} 1(1-2), \\ 3(2-3), \end{matrix}\right\} \text{ or } \left\{\begin{matrix} 2(1-2); \\ 3(1-3); \end{matrix}\right\}$$

where the quantity inside the parenthesis represents the pressure connections, and that outside is the leg in which the current is simultaneously read; the algebraic sum of the wattmeter readings in any one of the three groupings above gives the true watts in the system. The electrodynamometer is read in all three legs simultaneously with the wattmeter readings: when corrected to give amperes, their sum divided by $\sqrt{3}$ and multiplied by the mean voltage, read across the three phases, will give the apparent watts, that is,

$$\text{apparent watts} = \frac{\text{sum of currents in the three legs} \times \text{mean volts}}{\sqrt{3}}.$$

In the Westinghouse Company's works the output for efficiency tests is obtained by the use of a Prony brake and flanged pulley to which a constant supply of cooling water is furnished.

For the larger sizes of motor the brake-strap is of the ordinary "cider-press" type somewhat similar to that shown in Fig. 4, but in this case the wooden blocks do not come in contact with the pulley surface. The rubbing surface is a flexible band of sheet brass fastened at one end to one of the blocks to which the clamping-screw is attached, the other end being free; between this band and the wooden blocks is a layer of asbestos to prevent the blocks taking fire.

The lubricant employed is the heavy dope, or grease, used in street-railway motor journals, mixed with a

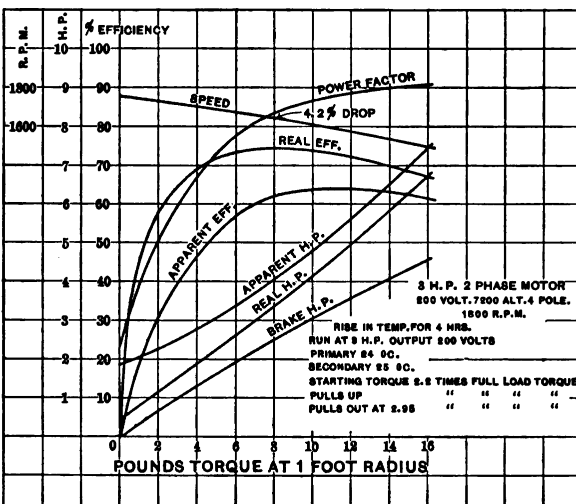

FIG. 114.—CURVES OF TEST OF 3-H.P. TWO-PHASE MOTOR.

little dynamo oil or graphite. Motors of 500 horse-power have been tested in this way up to full rated load.

With any form of brake-strap which is not compensating it is a difficult matter to maintain an absolutely constant load on a motor for any length of time; therefore the brake test is continued at full load only for a short time except with small motors—up to 5 or 6 horse-power.

For continuous runs, as for heat tests, the motors are

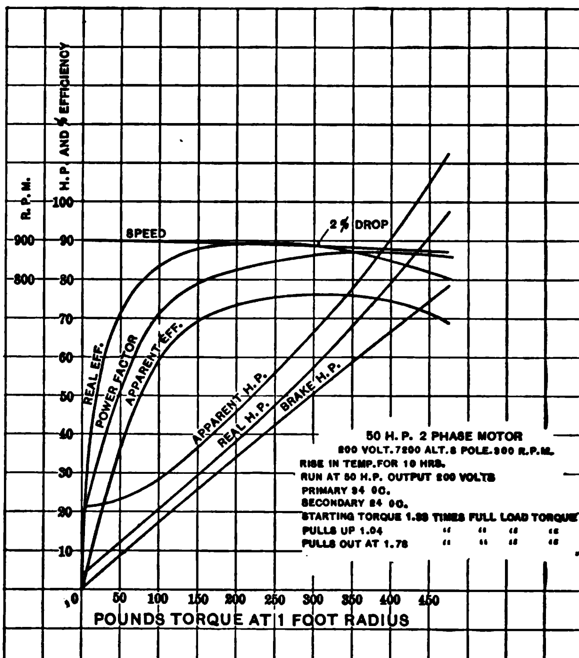

FIG. 115.—CURVES OF TEST OF 50-H.P. TWO-PHASE MOTOR.

belted or direct-connected to generators, which are in turn loaded as already indicated.

The complete sets of curves shown in Figs. 114 and 115 illustrate very fully the results of power tests made on a 3-H.P. two-phase motor, and also on a 50-H.P. two-phase motor. The machines both operate on a 200-volt system at 7200 alternations per minute. The starting torque of the smaller machine is 2.2 times the full-load torque; while that of the larger is 1.33 times the full-load torque.

In these curves it will be understood that the real horse-power is the sum of the true watts delivered to the motor divided by 746; and the real efficiency is the brake horse-power divided by the real horse-power; in the same way the apparent efficiency is the brake power divided by the apparent input.

In the foregoing matter on electrical measurements the testing of dynamos has not been entered upon, because a general treatment of this subject would involve questions which are beyond the scope and purpose of the present work. It will be apparent, however, that for actual determinations of the power and efficiency of dynamos the preceding methods and apparatus used for testing motors may readily be modified to include power measurements of dynamos, and this is all the present work aims to accomplish.

CHAPTER X.

POWER REQUIRED TO DRIVE MACHINERY.

NOW that the electric motor is such an economic factor in power distribution, a knowledge of the power required to drive each separate machine or tool is of especial importance in the selection of motors to be used in any given case.

For isolated machines and for heavy machines that may be in occasional use the individual motor is particularly well adapted, as it consumes power only when in operation. It is, however, necessary that each motor thus connected shall be capable of supplying sufficient power to operate its machine under the heaviest as well as lightest loads.

In certain cases, moreover, the load is liable to very great irregularity, as, for instance, in metal-working planers, in which the resistance offered by the machine at the moment of reversal of the platen is far higher than at other times, and may be so great as to endanger the armature of the motor. Under these conditions it is necessary to use a motor of much larger capacity than the average load would indicate.

Fortunately with electric motors the rated capacity is usually less than the safe maximum load, which is determined either by the heating of the conductors tending to break down the insulation, or by excessive sparking at the brushes.

For momentary overloads relatively large currents may pass through the coils without injury to the insulation, since the temperature effect is cumulative and requires time for its operation. However, for continuous periods of considerable length it is usually unsafe to operate the motor much above its rated output.

Ordinarily in machine-driving the motor is shunt-wound, and the current through the field-coils is constant under all conditions of load; but to obtain the best results with this class of machinery, in which the load is intermittent and subject to sudden variations, the motor should be compound-wound so as to increase the torque without an excessive increase of current in the armature.

In many cases with individual motors, owing to wide variations in power required, the average efficiency of the motor may be very low; for this reason a careful consideration of the conditions governing each case indicates that for ordinary machine-driving, especially with small machines, short lengths of light shafting may be frequently employed to good advantage and the various machines, arranged in groups, may be driven from one motor. By this method fewer motors are required and each may be so proportioned to the average load that it may be run most of the time at its maximum efficiency. When short lengths of shafting are employed the alignment of any section is very little affected by local settling of beams or columns, and since a relatively small amount of power is transmitted by each section, the shaft may be reduced in size, thus decreasing the friction loss. Moreover, with this arrangement, as also with the independent motor, the machinery may often be placed to better advantage in order to suit a given process of manufacture; shafts may be placed at any angle without the usual complicated and often unsatisfactory devices, and setting-up room may be provided in any suitable location as required, without carrying long lines of shafting through space. This is an important consideration, for not only is the running expense reduced thereby, but the clear head-room thus obtained free from shafting, belts, ropes, pulleys, and other transmitting devices, can be more easily utilized for hoists and cranes, which have so largely come to be recognized as essential to economical manufacture.

In arranging such a system of power distribution the average power required to drive is of as much importance as the maximum, for in a properly arranged group system the motor capacity need not be the equivalent of the total maximum power required to operate the several machines in the group, but may be taken at some value less than the total, depending upon the number of the machines and the average period of operation. On the other hand, as already shown, the motor capacity of independently driven machines must not only equal the maximum power required to drive the machine at full load, but it must be capable of exerting a greatly increased momentary torque. In any case large units should be avoided, for the multiplication of machines driven from one motor entails additional shafting, countershafts, and belting which may readily cause the transmission losses to be greater than that obtained with engines and shafting alone, besides frustrating some of the principal objects of this method of transmission.

As far as the efficiency of transmission is concerned it is doubtful whether in a large number of cases motor-driving is any more efficient than well-arranged engines and shafting.

As pointed out by Mr. Geo. Gibbs,* the principal thing to be kept in mind is a desired increase in efficiency of the shop plant in turning out product, with a reduction in the time and labor items, without especial reference to the fuel items involved in the power production. While the question of power loss, due to friction of shafting on the one hand, or electrical transmission and conversion on the other, should generally be of secondary importance compared with the increased facility of production, it is nevertheless necessary, in order to properly equip a plant that shall be most economical in all respects, to know how much power is required to operate each machine or group of machines under various conditions.

This may be determined experimentally by the methods previously discussed, but in many cases facilities for such experiments are lacking, and usually one is dependent upon the results of tests made by others.

A large number of such tests have been made and

* Proc. N. Y. Railway Club, March, 1898.

published, especially during the past few years, so that at the present time the power required to drive the ordinary machinery in iron- and wood-working establishments is fairly well determinable when the conditions are known.

In order to facilitate such determination the author has tabulated the results of numerous experiments upon machines of various kinds working under different conditions; these have been suitably classified for convenient reference, and form the subject-matter of the present chapter.

In addition to a large number of tests made by the author, the data for this compilation include results obtained by Hartig,* Vauclain and Halsey,† Pike,‡ Dodge,§ Crocker,‖ Cox,¶ Wise,** Benjamin,†† Carlsen and Lemon,‡‡ and others.

By far the most comprehensive of such tests are those made by Dr. Hartig. In these experiments sixty-nine separate machines were operated through a recording transmission-dynamometer; from five to fifty tests were made on each machine, and the horse-power

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* Versuche über Leistung und Arbeits-Verbrauch der Werkzeug-maschinen, Leipzig.

† *American Machinist*, Feb. 6, 1896.

‡ *Ibid.*, Sept. 24, 1896.

§ *Ibid.*, Aug. 6 and Oct. 8, 1896.

‖ Trans. A. I. E. E., vol. XII. p. 299.

¶ *American Machinist*, May 7, 1896.

** *Ibid.*, June 25, 1896.

†† *Machinery*, March, 1899.

‡‡ Graduating Thesis, University of Wisconsin, 1896. (Not published.) Courtesy of Professors Storm Bull and D. C. Jackson.

required to drive the machine idle as well as that required to do useful work under various conditions was carefully determined.

The character and extent of work done was observed in each case, and it is much to be regretted that later experimenters have not always been careful to note with the same degree of care what Hartig calls the performance of the machine, which, in metal- or wood-removing machinery, is determined from the cross-section of cut or chip, and involves the rate of feed and cutting speed: these and the general dimensions and velocities of each machine enable different sizes to be more readily compared.

Another valuable series of power tests is that conducted by Messrs. Carlsen and Lemon. In these tests over 120 machines or groups were experimented upon; in each case the power was determined by means of a 10 H. P. motor, belted to the machine or countershaft; a recording ammeter was placed in the circuit, and this was checked by a wattmeter read at suitable intervals. The voltage was maintained constant during each test. The machine, or group, was run for a definite length of time, so that the watts noted represent the average power delivered to the motor. The motor was calibrated both before and after the tests, and the efficiencies thus obtained were used in determining the actual horse-power delivered to the machine under test.

The tests made by Prof. Dodge also take account of the motor losses, and both the electrical input and the mechanical output are recorded.

In many cases where electrical measurements have been employed the efficiency of the motor has not been considered; the effect of this is to increase the apparent power required to operate the machine itself under the stated conditions; that is, a part of the observed power is spent in overcoming the resistances of the motor and should not ordinarily be charged to the machine, although often of considerable value in determining the electrical equipment.

In those cases where the rated capacity of motor is known the net horse-power delivered to the machine has been estimated; in a number of instances the data have permitted fairly accurate efficiency curves to be constructed, and these have been used in determining the net power. In any case where the actual power delivered to the machine has been estimated, such fact is indicated in the tables.

Where extreme overloading exists no attempt has been made to determine the real power delivered, but it is certainly very small as compared with the electrical input, owing to the relatively large I^2R losses.

It will be noted that some of the tests indicate vastly more power than might be expected or is shown with similar machines.

There are several reasons for this: In some shops the machines are worked to their utmost capacity and cuts are taken out of all proportion to those which ordinarily obtain; moreover in certain cases, in locomotive shops, for instance, many of the machines are arranged with two cutting-tools, thus greatly increasing the power required; then again electrical measurements may include the losses in the motor and shafting, which are not included in the results obtained by dynamometer tests.

It is also probable that in some cases the object was to ascertain what the motor would do under the severe conditions imposed, rather than to discover how much power was required to operate a machine under normal loads.

While the values presented have been accurately determined either by dynamometer or electrical measurements, it must be understood that in estimating the power required to drive a certain machine or group the values given for the individual machines are not necessarily representative, as the power required to drive a machine is dependent so largely upon a number of factors which from their nature it is impossible to include in a brief summary. However, it is believed that the number of examples is sufficiently large to form a satisfactory basis of calculation for the power required to drive a given machine, or to determine the motor equipment in any desired case.

RECORDS OF TESTS.

IRON-WORKING MACHINERY.	Observed Horse-power.	
Lathes.	Motor Input.	Delivered to Machine.
Lathe, 13-inch (Zimmermann); speeds from 5 to 188 r.p.m.		
Running idle (from .13 to .34 H.P.)....		.18
Turning wrought iron, $\frac{8}{32}$ inch depth of cut; breadth .02 inch; cutting speed 25 ft.p.m., 11½ lbs. chips per hour....		.41
Lathe, 16-inch (Flather).		
Running idle, with back gears in, 40 r.p.m.		.04
Turning wrought iron, ⅛ inch depth of cut; feed .02 inch..........		.40
Two other 16-inch lathes, required .19 and .22 H.P. respectively; the latter was turning cast iron 2 inches dia., $\frac{1}{16}$ inch depth of cut.		
Lathe, 16-inch by 8 feet (Putnam).		
Running empty..........		.08
Turning machine-steel, 5½ lbs. chips per hour..........		.54
Same, 9 lbs. chips per hour		.71
Countershaft not included.		
Lathe, 17½-inch; speeds from 6½ to 183 r.p.m.		
Running idle (from .16 to .46 H.P.)....		.21
Turning wrought iron, $\frac{7}{32}$ inch depth of cut; breadth .04 inch; cutting speed 16 ft.p.m., 25 lbs. chips per hour.....		.87
Lathe, 18-inch (Lodge & Davis); turning cast-iron pieces 3 inches dia.		
Average power for 5-hour run, steady work..........		.43
Average power while working........		.45
Maximum power during test..........		.91

LATHES—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Lathe, 20-inch (Fitchburg); speeds from $8\frac{1}{2}$ to 375 r.p.m.		
Running idle (from .1 to .3 H.P.)......		.15
Boring cast iron, 4-inch bore, $\frac{5}{32}$ inch depth of cut; .02 inch feed..........		.33
Lathe, 21-inch; turning and boring $11\frac{1}{2}$-inch cast-iron collars, 7-inch bore; cut $\frac{3}{16}$ inch deep, $\frac{1}{30}$ inch feed.		
Average power for 5-hour run.........		.35
Maximum power during test..........		1.04
Lathe, 22-inch by 6 feet (Flather).		
Running empty from .07 to .3 H.P....		.07
Turning cast iron, 9 lbs. chips per hour		.26
Same, 20 lbs. chips per hour		.81
Gap-lathe, 26-inch; speeds from 2.3 to 88 r.p.m.		
Running idle (from .04 to .33 H.P.)....		.05
Turning cast iron, $\frac{3}{32}$ inch depth of cut; breadth $\frac{1}{32}$ inch; dia. of work 31 inches; cutting velocity 32 ft.p.m. 11 lbs chips removed per hour		.46
Polishing 34-inch disk with emery stick, velocity 64 r.p.m..................		1.26
Lathe, 30-inch (Prentice); turning 3-step cone pulley 24-, $18\frac{1}{2}$-, and 12-inch diameters, $6\frac{5}{8}$-inch face.		
Average power for 5-hour run.........		.69
Average power while working.........		.80
Maximum power occurred while filing 6-inch arbor.......................		1.8
Gap-lathe, 50-inch, eight speeds from $\frac{1}{2}$ to 11 r.p.m., face-plate 80 inches diameter.		
Running idle (from .18 to .72 H.P.)....		.18
Turning cast iron, $\frac{3}{32}$ inch depth of cut; breadth $\frac{1}{16}$ inch; dia. of work $8\frac{1}{2}$ ft.;		

Lathes—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
cutting speed 13½ ft.-p.m., 12½ lbs. chips removed per hour.............		.53
Wheel-lathe, 60-inch, speeds from ¾ to 18 r.p.m. Running idle............ No cuts were taken.		.22 to 3.4
Facing-lathe, 68-inch; twelve speeds from 1.2 to 140 r.p.m.		
Running idle (from .37 to .81 H.P.)....		.37
Turning cast-iron, 7/32 inch depth of cut; breadth .04 inch; dia. of work 10 inches, cutting speed 16½ ft.p.m.....		.91
Wheel-lathe, 84-inch; turning large wheel centres, two wheels on axle; one tool to each wheel in all cases. Power was measured when turning cast-iron centres, the power required being greater with centres than with tires, on account of heavier cut.		
Light cut............................	2.9	
Extreme cut..........................	7.9	
Average cut	6.1	
Another 84-inch swing-lathe, turning 32-inch centres.		
No cut................................	.5	
Cut ½ inch deep, feed 1/16 inch..........	4.5	
Cut ½ inch deep, feed 3/32 inch..........	5.3	
Cut ½ inch deep, feed ⅛ inch...........	5.8	
Lathe, 90-inch (Bement & Miles); two tool-heads, both cutting; turning cast-steel driving-wheels, 67½ inches diameter; ½ revolution per minute; no lubricant; 8-H.P. Gibbs motor.		
Motor only...........................	1.30	
Driving countershaft.................	1.43	.13 est.
Driving countershaft and idle machine	2.10	.75 "
Depth of cut ½ inch; feed 1/32 inch, heavy cut—two tools..............	6.38	4.5 "

TURRET LATHES AND SCREW MACHINES.	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Turret Lathe, 21-inch (Gisholt, size H); facing cast-iron, width of cut $1\frac{1}{8}$ inch.		
Average power for 3-hour run........		.37
Maximum power during test..........		.91
Turret Lathe, 24-inch (Gisholt, size I); turning and finishing cast-iron rings $7\frac{1}{2}$ inches dia., $4\frac{1}{8}$-inch bore, 1-inch face, turned to 7 inches dia. and faced; 3 finished per hour.		
Average power for $4\frac{1}{2}$-hour run........		.32
Average power while working........		.40
Turret Lathe, 28-inch (Gisholt, size J); work on large gear-blank; hub cored $5\frac{1}{16}$, finished to $5\frac{1}{2}$ inches dia., $4\frac{3}{4}$ inches long; a sweep-facing cut 5 inches wide was taken off face.		
Average power for 5-hour run........		.78
Maximum during test.................		2.88
Tests on three other 28-inch Gisholt lathes gave the following mean values:		
Average power for 5-hour run........		.44
Average power while working		.56
Maximum power, mean of three machines............................		1.57
Screw Machine (Warner & Swasey, No. 3, with friction head); making small screws.		
Average power for 5-hour run........		.21
ROLL-TURNING LATHES.		
Roll-turning Lathe, 40-inch (Garrison); steel roll 39 inches dia., $5\frac{1}{2}$ feet long; roll supported in housings, speed of roll 1 revolution in 1 minute 15		

Roll-turning Lathes—*Continued.*	Observed Horse-power. Motor Input.	Delivered to Machine.
seconds; 10-H.P. shunt-wound motor, 800 r.p.m., voltage 200.		
Driving machine with roll, no cut.....	1.6	
Maximum cut $2\frac{1}{4}$ inches wide, $\frac{1}{32}$ inch deep, one tool.....................	6.7	4.8 est.
Roll-turning Lathe, 61-inch (Cambria); soft cast-iron roll 35 inches dia., supported in housings; speed of roll, 1 revolution in 1 minute 25 seconds; 5-H.P. shunt motor, 1000 r.p.m., 210 volts.		
Driving machine with roll, no cut.....	1 to 1.1	
Average cut $2\frac{1}{4}$ inches wide, $\frac{3}{64}$ inch deep............................	2.5 to 3.4	1.7 to 2.4 est.
Excessive cut, $2\frac{1}{4}$ inches wide, $\frac{5}{64}$ inch deep............................	2.8 to 4.8	1.9 to 3.5 "
Roll-turning Lathe, 63-inch (Bement); double back gear; steel roll 35 inches dia., $5\frac{1}{2}$ feet long; roll supported in housings; speed of roll, 1 revolution in 1 minute 30 seconds; 10-H.P. shunt motor, 1000 r.p.m., 200 volts.		
Driving machine with roll, no cut.....	2.4	
Maximum cut, $1\frac{3}{4}$ inches wide, $\frac{1}{8}$ inch deep............................	10.2	7.8 est.
Roll-turning Lathe, 73-inch (Bement); turning 48-inch steel pinion without sink head carried on centres; dia. at journal being turned, 28 inches; 10-H.P. shunt motor, 210 volts.		
Driving machine with work, no cut, starting............................	10.4	8.1 est.
Driving machine with work, no cut....	1.4	
Cut 2 inches wide, $\frac{5}{64}$ inch deep.......	3.9 to 4.8	2.5 to 3.3 est.
Large Roll Lathe, with two tools turning steel rolls, $33\frac{1}{2}$ inches dia., driven by line shaft 15 feet long, 2 inches dia., with 3 hangers; speed of line		

Roll-turning Lathes—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
shaft 500 r.p.m.; 10-H.P. shunt-wound motor coupled direct to line shaft; 220 volts.		
Motor, line shaft, and lathe..........	2.3	
Driving line shaft and two tools $4\frac{3}{4}$ inches wide, cutting at full capacity.	5.9 to 14.4	4.2 to 11.5 est.
Planers.		
Planer, 17-inch by 4 ft. (Bement & Miles; planing cast iron, stroke during test from 4 to 30 inches; feed from $\frac{1}{16}$ to $\frac{3}{32}$ inch; cut $\frac{3}{32}$ to $\frac{3}{16}$ inch deep.		
Average power for 5-hour run.........		.27
Agerage power while actually working		.91
During reversal.....................		1.33
Planer, 24-inch by 6 ft. (Gray); running empty from .3 to 1.25 H.P.; this decreases as the length of stroke increases.		
Planing machine-steel, heavy cut......		1.35
In this case there were 21 lbs. metal removed per hour.		
Running empty, same conditions, average...............................		.48
Maximum power at reversal..........		4.9
Planer, 24-inch, 13-ft. table (Bement & Miles); two tool-heads, both cutting; planing steel connecting-rod; no lubricant; 12 H.P. Gibbs shunt-wound motor, 240 volts.		
Motor only..........................	1.5	
Driving countershaft.................	1.95	.45 est.
Driving countershaft and idle machine:		
Forward, 18 ft.p.m., 6.5 amperes.....	2.1	.6 "
During reversal, 60 amperes........	19.5	
Backward, 60 ft.p.m., 20 amperes...	6.5	4.4 "
Cut $\frac{1}{4}$ inch deep, feed $\frac{1}{8}$ inch...........	8.0	5.6
The heavy rush of current at reversal,		

Planers—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
60 amperes, lasts but a few seconds and then quickly and quite steadily falls to the normal value for the backward run.		
Planer, 30-inch by 8 ft. (Putnam); planing cast iron; return speed same as forward.		
Cut $\frac{5}{32}$ inch deep, $\frac{1}{10}$-inch feed		1.33
Cut $\frac{3}{16}$ inch deep, $\frac{1}{10}$-inch feed		1.47
Cut $\frac{1}{4}$ inch deep, $\frac{1}{10}$-inch feed		1.96
Average power for $3\frac{1}{2}$-hour run		1.18
Planer, 31-inch by 10 ft. (Bement & Miles); planing cast iron, cut from $\frac{1}{16}$ to $\frac{1}{4}$ inch deep; feed $\frac{1}{16}$ inch.		
Average power for run of 4 hours		.48
Average power while working		,83
Power required to reverse		5.95
Planer, 34-inch by 9 ft. (Zimmermann); weight of table 2600 lbs.; cutting speed 11 ft.p.m.; return 17 ft.p.m.		
Running idle		.27
Planing cast iron, cut .16 inch deep; breadth .05 inch; $13\frac{1}{4}$ lbs. chips per hour		.84
Planer, 36-inch, 16 ft. (Niles); return 4 to 1; planing 13 ft. length with two tools, down-feed $\frac{1}{2}$ inch per hour; total weight on table 4700 lbs.		
Average power for 5-hour run		1.24
Average power while working		2.5
Reversing		14.89
Planer, 36-inch, 18-ft. table (Sellers), belt reversal; two tool-heads, both cutting; planing wrought iron with water; 19-H.P. Gibbs shunt-wound motor; 240 volts.		
Motor only	2.4	

PLANERS—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Driving countershaft..................	3.2	.8 est.
Driving countershaft and idle machine:		
Forward, 20 ft. per minute, 12 amperes	3.9	1.4 "
During reversal, 70 amperes........	22.1	17.6 "
Return 80 ft.p.m., 15 amperes.	4.7	2.2 "
Depth of cut 1 inch; feed ⅛ inch	16.7	12.5
Another 36-inch planer, 12-ft. table.		
Motor and countershaft...............	2.7	
Empty table forward...................	3.4	
Empty table at moment of reversal....	10.7	
Planing wrought iron, 1 tool..........	7.4	
Same with two tools..................	12.5	
Planer, 48-inch, 16 ft. (Pond); return 3 to 1; planing Gisholt lathe-beds, weight 3750 lbs. Cuts varied from ⅝ inch to 1 inch, feeds from $\frac{5}{64}$ to .2 inch. Part of the time two tools were used, cuts ¼ inch deep, feed .14 inch.		
Average power for 14-hour run........		3.78
Average power while working (for entire run)...........................		6.1
Heaviest work on one tool, 1 inch deep; $\frac{5}{64}$-inch feed, 20 minutes		9.9
Maximum power at reversal		17.6
Planer, 48-inch, 15-foot bed (Betts); cutting speed 9¾ ft.p.m.; return 3 to 1; 7½ H.P. motor belted to countershaft.		
Cut ⅛ inch, feed $\frac{1}{16}$ inch, load on bed 4	2.5	1.6
tons..............................	4.0	3.1
Instant of reversal....................	3.1	2.1
Return at speed........................		
The speed of platen in this case was very slow, and it is fair to assume that the belt-speed was slow also; this accounts for the relatively small amount of power at reversal.		
Planer, 56-inch, 35-foot table (Sellers);		

PLANERS—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
quick-return pattern with clutch reversal; two tool-heads, both cutting, planing wrought-iron locomotive frames with water; 20-H.P. U. S. motor, shunt-wound.		
Motor only	1.6	
Driving countershaft	4.6	2.5 est.
Driving countershaft and idle machine:		
Forward, $18\frac{1}{4}$ ft.p.m	7.1	5.0 "
Moment of reversal	19.8	16.0 "
Return, 111 ft.p.m	12.7	9.5 "
Depth of cut $\frac{3}{8}$ inch; feed $\frac{5}{16}$ inch	13.4	10.3 "
Planer, 56-inch, 24-foot table (Sellers); shifting belt reversal; two tool-heads, both cutting; planing wrought iron with water; 24-H.P. Gibbs motor, shunt wound.		
Motor only	3.25	
Driving countershaft	4.56	1.3 "
Countershaft and idle machine:		
Forward, 19 ft.p.m	4.8	1.5 "
During reversal	31.1	23.0 "
Return, 70 ft.p.m	7.15	3.6 "
Depth of cut $\frac{3}{8}$ inch; feed $\frac{3}{16}$ inch	16.0	11.2 "

This and the previous test give an instructive comparison between the action of the shifting belt and clutch reversal. The clutch planer has a return of 111 ft.p.m. and a table 35 feet long, which requires at reversal 15.2 H.P., neglecting countershaft; on the other hand, the shifting-belt machine of approximately the same size with a velocity of 70 ft.p.m. and a table 24 ft. long requires 26.5 H.P. during reversal.

Planer, 62-inch, 35-foot table (Sellers); quick return, 4 to 1; planing wrought-iron locomotive frames.		
Motor only	2.3	
Driving countershaft	4.4	
Empty table forward	11.0	

Planers—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Empty table return....................	11.8	
Empty table at moment of reversing ..	18.0	
Cut ⅝ inch deep, ¼ inch feed, two heads, two tools..........................	21.1	
Planer, 76-inch, 38 feet long (Zimmermann); weight of table 33,000 lbs.; cutting speed 13 ft.p.m.; return 30 ft.p.m.		
Running idle, mean power............		.60
Planing cast iron, depth of cut ⅝ inch, breadth .05 inch, weight of casting 9400 lbs.; 52 lbs. chips per hour.....		1.47
Planer, 84-inch, 15-foot bed (Sellers); cutting speed 18 ft.p.m.; return 3 to 1; 7½ H.P. motor, belted to countershaft.		
Driving countershaft and loose pulleys.	2.25	
Driving planer-bed forward, no load...	2.5	
Reversal.............................	8.9	
Return at speed......................	2.8	
Reversal.............................	5.7	
Load on bed 12,600 lbs.		
Driving planer-bed, forward	2.8	
Reversal.............................	9.6	
Return at speed	4.3	
Reversal.............................	5.9	
The above tests show how little the weight of work on the platen affects the power required to operate the machine.		
Planer, 96-inch, 20-foot bed (Betts); cutting speed 12 ft.p.m., return 3 to 1; countershaft speed, 250 r.p.m.; 7½ H.P. motor, belted to countershaft.		
Driving countershaft with machine belts on loose pulleys...............	1.69	.73
Running machine idle with 15 tons on bed: Forward.........................	1.73	.80
Instant of reversal............	5.70	4.93

	Observed Horse-power.	
PLANERS—*Continued.*	Motor Input.	Delivered to Machine.
Return at speed	2.60	1.73
Planing steel: $\frac{5}{16}$ inch cut, .02 inch feed.	3.45	2.53
$\frac{3}{4}$ inch cut, .02 inch feed.	3.65	2.73
The cutting tests were made on a duplicate machine with a load of 12,600 lbs. on the bed.		
Planer, 10 feet wide by 9 feet high, 22-foot bed; 15-H.P., C. & C. motor.		
Average power	5.11	3.2 est.
Power for reversal from cutting to return stroke	21.65	
Power during cutting stroke	4.76	2.9 "
Power during return stroke	5.73	3.7 "
SHAPERS.		
Shaper, 9½-inch, with traversing-head (Richards' pattern). Capacity: length 23 inches, breadth 9½ inches. Four speeds, from 15 to 27 strokes per minute; return 2 to 1		
Running empty (from .07 to .12 H.P.).		.07
Planing cast iron, cut .2 inch deep, breadth .02 inch; 10 ft.p.m., 5.2 lbs. chips per hour		.24
Shaper, 15-inch (Gould & Eberhardt); power required to run light varied with stroke and speed.		
A 6-inch stroke speeded to 12, 23½, 41, and 70 strokes per minute required from		.01 to .17
A 15-inch stroke, same speeds		.03 to .47
Planing wrought iron 7 ft.p.m., cut .22 inch deep, .01 inch feed		.17
Planing wrought iron 14 ft.p.m., cut .17 inch deep, .01 inch feed		.27
Planing bronze, 46 ft.p.m., cut .23 inch deep, .01 inch feed		.64
Another 15-inch shaper (Flather) plan-		

Shapers—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
ing cast iron and machine steel, stroke about 8 inches		.3 to .5
Shaper, 16-inch (Hendey).		
Running empty, stroke 2½ inches		1.57
Running empty, stroke 13½ inches		.15
Cutting cast iron, 10½ lbs. metal removed per hour		.44
Power at reversal		1.8
Shaper, 17-inch (Putnam); 72 strokes per minute, 4½-inch stroke; planing brass, ⅛ inch depth of cut, $\frac{1}{32}$-inch feed.		
Average power for 5-hour run		.54
Power when actually working		1.29
The short stroke and rapid reversals account for the large power required.		
Shaper, 19-inch (Bement & Miles); return 2 to 1; one tool-head; planing wrought iron with soda-water; 8-H.P. Gibbs shunt motor; voltage 240; speed forward at maximum stroke 16 ft.p.m.		
Motor only	1.44	
Driving countershaft	1.6	
Driving countershaft and idle machine	1.8	
Cut ⅛ inch deep, feed $\frac{1}{16}$ inch, stroke 8 inches	4.8	3.1 est.
Cut ¾ inch deep, ⅛-inch feed, very heavy cut	9.7	7.8 "
Shaper, 29-inch, with traversing-head (Richards' pattern). Capacity: length 90 inches, breadth 29 inches. Five speeds from 4 to 33 strokes per minute; return 2 to 1.		
Running empty (from .15 to .73 H.P., depending on number of strokes)		.26
Planing wrought iron, depth of cut .28 inch, breadth .05 inch; 18 ft.p.m., 17½ lbs. chips per hour		1.14

Slotting-machines.	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Slotting-machine, 9½-inch stroke; diameter of table 24 inches; largest diameter of work 40 inches; speeds from 12 to 32 strokes per minute.		
Running empty (from .22 to .64 H.P.)..		.22
Cutting cast iron, 9½-inch stroke; depth of cut .16 inch, breadth .02; velocity 14 ft.p.m.; 4¾ lbs. iron removed per hour. Average power....................		.44
Wide variation due to reversals.		
Slotting-machine, 12-inch stroke, 36-inch table. Ordinary machine-shop pattern of slotter with goose-neck frame, crank motion; cutting wrought iron.		
Motor and countershaft..............	1.5	
Slotter empty......	1.55	
12-inch stroke........................	4.4	
8-inch stroke.........................	4.7	
4-inch stroke.........................	6.3	
Cuts are the heaviest the belt would pull.		
Slotting-machine, 15-inch stroke (Sellers); diameter of table 40 inches, cutting wrought iron with water; 18 strokes per minute; return speed 3 to 1; 8-H.P. Gibbs motor, shunt-wound, 240 volts.		
Motor only...........................	1.45	
Driving countershaft.................	1.8	.3 est.
Driving countershaft and idle machine, downward stroke 22½ feet per minute	2.2	.7 "
Depth of cut ⅛ inch, feed $\frac{3}{16}$ inch; heavy cut..................................	7.3	5.5
Slotting-machine, 15-inch stroke (Zimmermann). Diameter of table 45 inches; largest diameter of work 70 inches; speeds from 4 to 42 strokes		

SLOTTING-MACHINES—*Continued.*	Observed Horse-power. Motor Input.	Delivered to Machine.
per minute; back-geared; return 2 to 1.		
Running empty (from .43 to .94 H.P.)..		.57
Cutting cast iron, 9½-inch stroke, 21 strokes per minute; depth of cut $\frac{5}{16}$ inch, breadth $\frac{1}{32}$ inch; cutting velocity 28 ft.p.m., 17½ lbs. chips per hour.		
Average power........................		.95
Wide variation due to reversals.		
Locomotive-frame Slotting-machine; crank motion with quick reverse; stroke 8 inches. These machines have two heads with a tool to each; independent motor to each head. The horse-power is given for one motor.		
Motor only...........................	2.3	
Motor and empty machine	3.	
Heavy cut in wrought iron............	8.3 to 10.3	
Belt slipped with 10.3 H.P.		
Locomotive-frame Slotting-machine; planer motion with dogs and shifting belt; two heads as before, with motor to each head.		
The horse-power of one motor was:		
Motor only...........................	2.3	
Heavy cuts...........................	10.5 to 11.8	
Empty machine at moment of reversing	15.4	
The cut requiring 11.8 H.P. slipped the belt.		
Double-head Locomotive-frame Slotting-machine (Bement & Miles); width between housings 30 inches; length 24 feet; slotting two wrought-iron locomotive frames with water; length of stroke 9 inches; 15½ strokes per minute; maximum stroke 16 inches; separate motor to each tool. The power given is for one motor only.		

	Observed Horse-power.	
SLOTTING-MACHINES—*Continued.*	Motor Input.	Delivered to Machine.
12-H.P. Gibbs motor, shunt-wound; 240 volts.		
Motor only (1)	1.5	
Motor and countershaft (1)	1.9	.4 est.
Motor countershaft and idle machine:		
Downward stroke, cutting speed 21 ft.p.m.	2.2	.7 "
Return stroke, speed 61 ft. p. m.	2.2	
Depth of cut $\frac{1}{8}$ inch, feed $\frac{1}{16}$ inch.	8.4	6.5 "
Double-head Frame-slotting Machine; width between housings 48 inches; length 48 feet; slotting wrought-iron locomotive frames; separate motor to each tool. The readings from which the horse-power was calculated were taken for one motor only, (*a*) being from one head and (*b*) from the other; 19-H.P. Gibbs shunt motor, 240 volts.		
a.		
Motor only	2.4	
Countershaft	3.2	.8 est.
Countershaft and idle machine:		
Downward, 12 feet per minute	5.5	3.0 "
Reversal	17.8	15.0 "
Depth of cut $\frac{3}{4}$ inch, feed $\frac{1}{32}$ inch	7.6	5.0 "
b.		
Countershaft	3.2	.8 "
Countershaft and idle machine:		
Downward, 12 feet per minute	4.8	2.5 "
Reversal	21.6	17.5 "
Return stroke, 36 feet per minute	6.4	3.5 "
Depth of cut $\frac{1}{16}$ inch, feed $\frac{3}{4}$ inch	11.4	8.5 "
DRILLING-MACHINES.		
Drill-press, 18-inch (Barnes); reaming $\frac{7}{16}$-inch holes in steel to $\frac{17}{32}$ inch.		
Average power for run of 3 hours		.16

DRILLING-MACHINES—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Maximum power during test.......... The variation from the average was not great.		.57
Drill press, 24-inch (Prentice).		
Ordinary work, average..............		.11
Maximum work, drilling $\frac{15}{16}$-inch hole in machinery-steel, drill crowded....		.75
Drill-press, 24-inch (Bickford); drilling $1\frac{7}{8}$-inch holes in $\frac{1}{2}$-inch soft steel, 20 holes per hour.		
Average power for $2\frac{1}{2}$-hour run........		.32
Maximum power during test..........		.51
Drill-press, 30-inch (Gould & Eberhardt); drilling cast iron. Drills from $\frac{3}{8}$ inch to $1\frac{1}{2}$ inches were used.		
Average power for 5-hour run........		.11
Average power while working........		.48
Maximum power during test..........		1.06
Drill-press, 34-inch (Bickford); drilling steel, $\frac{1}{2}$-inch hole, also 2-inch hole in cast iron.		
Average power during 4-hour test.....		.52
Average power during test with 2-inch drill............................		.61
Radial Drill, 42-inch arm (Bement & Miles); drilling in wrought iron with soda-water; 42 r. p. m., machine not working to full capacity on account of feed-gearing being out of order; 5-H.P. motor, 240 volts.		
Motor only...........................	.8	
Driving countershaft..................	.96	.15 est.
Driving countershaft and idle machine	1.1	.28 "
Feed $\frac{1}{4}$ inch per minute, 2 inch-drill...	2.1	1.1 "
Radial Drill, 5-foot arm (Universal Radial Drill Co.); drilling from $\frac{1}{8}$-		

Drilling-machines—*Continued.*	Observed Horse-power. Motor Input.	Delivered to Machine.
to 1¼-inch holes and reaming 1¼ to 2¼ inches.		
Average power for 10-hour run........		.21
Average power while actually working		.43
Maximum power, drilling 1⅝-inch hole 6¼ inches deep....................		1.45
An increased depth of hole requires more power owing to the clogging of chips.		
Radial Drill, maximum radius 6 feet (Zimmermann); speeds from 3 to 169 r.p.m.; back-geared.		
Running empty (from .10 to .44 H.P.)..		.44
Drilling ¼-inch holes in cast iron, 150 r.p.m., feed .52 inch per minute......		.53
Radial Drill, maximum radius 8⅓ feet (Hartmann); speeds from 3½ to 134 r.p.m., back-geared.		
Running empty (from .14 to .78 H.P.)..		.30
Drilling 2-inch hole in cast iron (flat drill), 36 r.p.m., feed .16 inch per minute; 8 lbs. chips per hour; very light feed.........		.67
Two-spindle Drilling-machine (Harrington & Sons); reach 22 feet; drilling in wrought iron with soda-water; 80 r.p.m.; 5-H.P. motor, 240 volts.		
Motor only.........................	.97	
Driving countershaft.................	1.1	.13 est
Driving countershaft and idle machine	1.45	.45 "
Feed $\frac{9}{10}$ inch per minute:		
Using one 1-inch drill...............	2.7	1.7 "
Using one 1¼-inch drill.............	3.7	2.6 "
Using both drills...................	5.15	3.9 "
Three-spindle Drilling-machines (Bement & Miles); length 21 feet; drilling in wrought iron with soap and oil; 86 r.p.m.; 8-H.P. motor, 245 volts.		
Motor only.........................	1.6	

DRILLING-MACHINES—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Driving countershaft	2.1	.5 est.
Driving countershaft and idle machine	2.6	.8 "
Feed .84 inch per minute:		
One $\frac{7}{8}$-inch drill, average	3.6	1.6 "
Two $\frac{7}{8}$-inch drills, average	4.1	2.0 "
Three $\frac{7}{8}$-inch drills	4.8	2.7 "
Same machine as last, doing same character of work, but with larger drills representing the maximum work for which the machine was designed; 78 r.p.m.		
Three 2-inch drills, feed .45 inch per minute	8.0	6.0 est.
Multiple - spindle Drilling - machines. Three of these machines were operated through a line shaft 70 feet long, $2\frac{3}{8}$ inches diameter, with ten hangers; speed of shaft 200 r.p.m.; 15-H.P. shunt motor, 800 r.p.m., 220 volts.		
Running shaft and drill-presses idle	4.4	
Drilling seven $\frac{3}{4}$-inch holes in thin part of rails	11.7 to 17.6	
Drilling seven $\frac{3}{4}$-inch holes in thick part of rails, fuse blown	22.	

Eight-head Multiple-drill (Gantry); driven by line shaft 15 feet long, $1\frac{3}{4}$ inches diameter, with eight hangers; speed of line shaft 850 r.p.m. The line shaft is the main driving-shaft of the machine, from which the drill-heads are driven. The work of the machine is the drilling and reaming of holes in steel girders. Holes punched $\frac{3}{4}$ inch and enlarged by drills to $\frac{15}{16}$ inch; 10-H.P. shunt-wound motor, 220 volts, direct-connected to shaft.

Motor and line shaft with friction wheels in contact—starting	28.8 to 43.2	
5 drills at work	20.4	
6 drills at work	24.8	
7 drills at work	27.6	
8 drills at work	28.8 to 34.6	
Motor was not of sufficient capacity. A 25-H.P. motor was substituted.		

	Observed Horse-power.	
DRILLING-MACHINES—*Continued.*	Motor Input.	Delivered to Machine.
Rail Drill-presses. Two Sellers rail drill-presses were operated through a line shaft 70 feet long, 2½ inches diameter, with ten hangers; speed of shaft 200 r.p.m.; 15-H.P. shunt motor, 800 r.p.m., 220 volts.		
Motor and line shaft (belts on loose pulleys)............................	4.4	
Drilling six $1\frac{1}{16}$-inch holes in web of rail, ½ inch thick..................	16.3	
Drilling six $1\frac{1}{16}$-inch holes in web of rail, ¾ inch thick..................	20.7	
The increase of power in the latter case is due to the clogging of the chips below ½ inch depth.		
BORING-MILLS.		
Horizontal Boring and Drilling Machine; capacity 11½ inches diameter, 15 inches long; speed of boring-bar from 4 to 160 r.p.m.; back-geared.		
Running light (from .10 to .25 H.P.)...		12
Enlarging $\frac{15}{16}$-inch hole to 2 inches; feed .33 inch per minute; cutting velocity 31 ft.p.m. at 60 r.p.m.; 10 lbs. wrought iron per hour......................		.93
Horizontal Boring-machine (Pratt & Whitney, No. 2 double-head traverse drill).		
Drilling $1\frac{5}{16}$-inch holes in hard tool-steel, average power for 5-hour run..		.47
Average power while working........		.65
Maximum power during test..........		2.17
Horizontal Boring and Drilling Machine (Bement & Miles, No. 2); boring, reaming, and facing 7-inch to 9-inch holes; run of 17 hours.		
Boring, average power...............		.34

Boring-mills—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
Reaming, average power..............		.71
Maximum power during test..........		1.49
Horizontal Boring-mill; 78-inch swing; light pattern; boring hubs of 56-inch driving-wheel centers, using single lathe-pattern boring-tool,—not a boring head.		
Boring hole 6⅞ inches dia., heavy cut..	4.5	
Another 78-inch swing, very heavy pattern. (Same work as last.)		
Boring hole 6⅞ inches diameter........	6.9	
Facing hub, cut ⅛ inch deep...........	6.4	
This mill often takes heavier cuts.		
Horizontal Boring, Milling, and Drilling Machine (Bement & Miles, No. 10); motor belted to cone shaft when moving table, and to countershaft when running drill.		
Moving table longitudinally, and transversely, while rotating, no load on table; highest speed of cone shaft out of gear..........................	3.01	2.07
Running machine idle:		
Low speed of cone shaft, intermediate gear............................	1.72	.93
High speed of cone shaft, intermediate gear........................	2.88	2.00
High speed of cone shaft, out of gear	5.45	4.53
Cylinder Boring-machine; motor connected to one line-shaft, which drives several machines; all work on 18-inch locomotive cylinders.		
Motor only	1.8	
Motor and shafting..................	6.4	
Single machine boring, roughing cut, about ½ inch deep, ⅛ inch feed..	18	
Facing flanges, two tools, heavy cut...	25.5	

BORING-MILLS—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Another machine, facing flanges, two tools	32	
Both machines boring, average cut	31	
Vertical Boring-mill (Sellers); table 4½ feet diameter; one tool-head; boring cast-steel driver tires, 3½ ft. diameter; speed 8 r.p.m.; 5-H.P. Gibbs motor, 240 volts.		
Motor only	1.75	
Driving countershaft	2.1	3 est.
Driving countershaft and idle machine	2.4	.6 "
Depth of cut ¼ inch, feed $\frac{1}{16}$ inch	4.6	2.7
Vertical Boring-mill (Baldwin Locomotive Works); table 5½ ft. diameter; one tool-head; boring cast-iron driver hub; 8-H.P. Gibbs motor, 240 volts.		
Motor only	1.4	
Driving countershaft	1.6	.2 est.
Driving countershaft and idle machine	2.4	1.0 "
Depth of cut ⅛ inch; feed $\frac{1}{16}$ inch at 3 inches from centre	4.2	2.3 "
Depth of cut ⅛ inch; feed $\frac{1}{16}$ inch at 7 inches from centre	4.8	2.9 "

In the following tests of boring-mills readings were taken, when cutting, from the first one only. The readings from the first machine give figures for a considerable range of feeds; by adding the net amounts of power for the cuts to the readings for the larger machines when not cutting, figures may be obtained for the probable power of these machines when cutting.

Vertical Boring-mill (Pond); table 8 ft. diameter; speed of countershaft 180 r.p.m.; speed of cone shaft, in seven steps, from 90 to 360 r.p.m.; 7½-H.P. motor belted to 30-inch (largest) cone pulley on countershaft; starting torque of cone shaft in gear, 98 ft.-lbs.; starting torque of cone shaft out of gear, 125 ft.-lbs.		
Running idle in gear, lowest speed	1.45	.53

Boring-mills—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Running idle in gear, next to highest speed	1.87	.93
Running idle out of gear, lowest speed.	1.76	.8
Running idle out of gear, next to highest speed	4.17	3.27
Cutting tests on low steel; weight on bed about 100 lbs.; cutting speed 8 ft. per minute.		
Running machine idle, lowest speed, in gear	1.77	.8
1 tool, ¼-inch cut, $\frac{1}{32}$-inch feed	2.24	1.33
2 tools, ¼-inch cut, $\frac{1}{32}$-inch feed	2.48	1.60
2 tools, ¼-inch cut, $\frac{1}{16}$-inch feed	3.31	2.40
1 tool, ¼-inch cut, $\frac{3}{32}$-inch feed	2.77	1.87
2 tools, ¼-inch cut, $\frac{3}{32}$-inch feed	3.67	2.80
When belted for highest speed it would require a 7½-H.P. motor to start this machine. A motor of this size would otherwise be unnecessarily large.		
Vertical Boring-mill (Pond); table 10 feet diameter; speed of countershaft 195 r.p.m.; speed of cone shaft, in seven steps, from 81 to 468 revolutions per minute; motor belted to 30-inch (largest) pulley on countershaft; 7½-H.P. shunt motor; starting torque of cone shaft in gear 98 ft.-lbs.		
Running idle in gear, lowest speed	1.81	.86
Running idle in gear, next to highest speed	2.00	1.07
Running idle out of gear, lowest speed.	1.90	1.00
Running idle out of gear, next to highest speed	4.09	3.20
Load on bed about 100 lbs. No accurate determination could be made of the power necessary to start this machine out of gear, as a starting-bar had to be used. Belts otherwise slip and run off.		

	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
BORING-MILLS—*Continued.*		
Vertical Boring-mill (Pond); table 12 feet diameter; speed of countershaft 180 r.p.m.; nine speeds of cone shaft in regular steps, from 90 to 360 r.p.m.; motor belted to 34-inch pulley (next to largest) on countershaft; 7½-H.P. shunt-wound motor; starting torque of cone shaft in gear 223 ft.-lbs.		
Running idle in gear, lowest speed....	1.86	.93
Running idle in gear, third from highest speed..........................	2.67	1.73
Running idle out of gear, lowest speed..	3.02	2.07
Running idle out of gear, third from highest speed......................	6.72	5.80
Load on bed, about 3000 lbs.		
MILLING-MACHINES AND GEAR-CUTTERS.		
Universal Milling-machine, small (No. 1, Brown & Sharpe); milling cast iron 70 r.p.m., cutting speed 32 ft.p.m. Power required to run machine empty varied greatly with the rate of feed (from .002 to .13 H.P.)..............		.01
Cut .10 inch deep, 1⅛ inches wide, medium feed.........................		.2
Cut .15 inch deep, 1⅛ inches wide, medium feed.........................		.3
Another similar machine, milling cast iron, $\frac{1}{16}$ inch cut, 1⅝ inches wide, feed 1 inch per minute..................		.25
Milling-machine (Brown & Sharpe, No. 3); milling ½-inch slot $\frac{3}{16}$ inch deep in cast iron; fastest feed.		
Average power for 2-hour run.........		.14
Milling-machine (Brown and Sharpe,		

MILLING-MACHINES AND GEAR-CUTTERS—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
No. 4, plain miller); cutting steel gears 4 pitch.		
Average power for 5-hour run........		.31
Average power while working........		.45
Milling-machine (Brown & Sharpe, No. 5, plain miller); milling brass, light work.		
Average power for 5-hour run........		.35
Small Traversing-head Milling-machine (Hartmann). Will take work $8\frac{1}{2}$ inches high, 20 inches long. Dia. cutter-head 11 inches; velocity of cutter 13 and 19 ft. per minute; inserted teeth.		
Running idle........................		.10
Face-milling, soft cast iron, face $2\frac{1}{2}$ inches; $\frac{1}{8}$ inch depth of cut, feed 2 inches per minute; $4\frac{1}{2}$ lbs. chips per hour........................		.18
Same, with skin cut, but less than $\frac{1}{16}$ inch deep........................		.34
Milling-machine, large (Zimmermann). Will take work 12 inches high, 84 inches long. Diameter cutter-head 13 inches, 12 cutters; speed 5, 7, $10\frac{1}{2}$ r.p.m.		
Running idle (from .26 to .54 H.P.)....		.26
Milling cast iron, face $7\frac{1}{2}$ inches; depth of cut .15 inch; feed $\frac{3}{8}$ inch per minute; $9\frac{1}{2}$ lbs. chips per hour; velocity of cutters 17 ft.p.m.............		.66
Gear cutter, 21-inch (Zimmermann).		
Running idle........................		.10
Cutting 10-inch cast-iron spur-gear, 5 pitch, feed $\frac{15}{16}$ inch per minute; $1\frac{1}{2}$ lbs. metal removed per hour; velocity of cutter 40 ft.p.m.................		.28

	Observed Horse-power.	
MILLING-MACHINES AND GEAR-CUTTERS—*Continued.*	Motor Input.	Delivered to Machine.
Gear-cutter, 50-inch automatic (Gould & Eberhardt, old style); cutting 5-pitch gear, machine-steel; face 2½ inches; feed $\frac{7}{16}$ inch per minute.		
Steady work for 5 hours..............		.60
Milling-machine for steel plates, geared to shunt-wound motor; diameter of cutters 6 inches; face 18 inches; feed 2 inches per minute; 5-H.P. 220-volt motor. In use at the Pencoyd Steel Works.		
Motor and empty machine............	1.2	
Motor and one tool cutting 13 inches wide, $\frac{8}{64}$ inch deep..................	8.8	
Motor and one tool cutting 13 inches wide, $\frac{1}{16}$ inch deep..................	11.8	
Motor was not capable of doing the work and heated badly, owing to a constant excessive load.		
MILLING-SAWS.		
Cold-saw (Newton), 20-inch diameter, $\frac{3}{16}$ inch thick; speed 8 r.p.m.; 192 teeth; 2-H.P. individual shunt motor, 1000 r.p.m., 220 volts.		
Sawing 9-inch rails, 108 lbs. per yard..	2.4 to 3.8	
Three Milling-saws, 30-inch diameter, ¼ inch thick, driven by line shaft 35 feet long, 1⅞ inch diameter, 7 hangers; speed of line shaft 860 r.p.m.; 10-H.P. shunt-wound motor, 220 volts, coupled direct to line shaft.		
Motor and line shaft (belts on loose pulleys)...........................	2.5	
Driving line shaft and one saw, sawing 7 × ⅜-inch I beams.................	3.1	
Driving line shaft, two saws, each sawing two 12 × ⅜-inch channel beams,		

	Observed Horse-power.	
MILLING-SAWS—*Continued.*	Motor Input.	Delivered to Machine.
and one saw, sawing 7 × ⅜-inch I beams..............................	6.7	
Hot-saw for Structural Steel; 48-inch diameter, ¼ inch thick; speed 1800 r.p.m.; short counter-shaft between motor and saw-shaft; 125-H.P. motor, 220 volts, current from 15 to 300 amp.		
Motor, countershaft, and saw running idle..............................	7.4	
Cutting one 4 × 4 × ½-inch angle......	29.3	20 est.
Cutting one 10 × ⅜-inch flat............	65.7	54 "
Cutting one 9 × ½-inch I beam.........	80.4	68 "
Cutting 20 × ½-inch I beams, maximum and minimum values for each cut:		
First cut, 2½ minutes................	30.8 to 53.9	22 to 42 est.
Another cut, 55 seconds.............	30.8 to 92.5	22 to 79 "
Hot-saw, 30-inch diameter, ⅛ inch thick; speed 8000 ft.p.m.		
Running idle..........................		.61
Cutting 5-inch round iron (red-hot), feed .4 feet per minute..............		4.12
BOLT CUTTERS, NUT-TAPPING AND PIPE-THREADING MACHINES.		
Single Bolt and Nut Machine; speed of die 76 r.p.m.; cutting ¾-inch iron bolts 2⅝ inches long, 8 finished in 5 minutes.		
Average power for 3-hour test........		.37
Bolt-cutter (Zimmermann-Sellers type). Will cut bolts from ¼ to 1¼ inch; speed of dies 16 to 58 r.p.m.		
Running idle (from .12 to .2 H.P.).....		.18
Cutting 1¼-inch wrought-iron bolts, Whitworth thread, one cut, speed of dies 54 r.p.m........................		1.32
The same, but speed of dies 36 r.p.m...		.81
The same, but speed of dies 16 r.p.m...		.53

	Observed Horse-power.	
BOLT-CUTTERS, NUT-TAPPING AND PIPE-THREADING MACHINES—*Continued.*	Motor Input.	Delivered to Machine.
Cutting $\frac{3}{4}$-inch bolts, wrought iron, speed of dies 58 r.p.m.		.56
Cutting $\frac{3}{4}$-inch bolts, wrought iron, speed of dies 17 r.p.m.		.30
Nut-tapping Machine, three taps, 160 r.p.m.; 300 $\frac{7}{16}$-inch nuts tapped per hour; oil bath.		
Average power for 4-hour test		.45
Pipe-threading Machine, 3-inch; 3-H.P. Crocker-Wheeler motors, shunt-wound, speed 950 r.p.m., belted to countershaft.		
Running idle	1.2	
Cutting $1\frac{1}{2}$-inch pipe, 40 r.p.m.	2.0	1.3 est.
Threading $1\frac{1}{2}$-inch pipe, 45 r.p.m.	2.5	1.8 "
Threading 3-inch, 21 r.p.m.	2.5	1.8 "
Threading 3-inch, 32 r.p.m.	4.8	3.4 "
Speed was too great in this latter case; dies were burned.		
Pipe-threading Machine, 8-inch, same motor as above.		
Running idle, 12 r.p.m.	1.5	
Cutting 4-inch pipe, 12 r.p.m.	2.8	2 "
Threading 4-inch pipe, same speed	5.0 est.	
Threading $5\frac{5}{8}$-inch pipe, same speed	3.4	2.5 "
The $5\frac{5}{8}$-inch is casing-pipe with fine threads—cut not so heavy as 4-inch regular pipe: this latter took more than 25 amperes—the capacity of the ammeter.		
PUNCHING AND SHEARING MACHINES.		
Double Punch and Shear (Long & Allstatter); motor belted direct; shearing $\frac{7}{8}$-inch round steel, also punching $\frac{7}{16}$-inch round holes in $\frac{1}{4}$-inch channel iron—4 at a time.		
Average power for $3\frac{1}{2}$-hour run		.19

PUNCHING AND SHEARING MACHINES—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Maximum hourly reading		.24
Maximum power shearing $1\frac{1}{4}$-inch round iron rapidly		2.28
Punching and Shearing Machine (Ferracute, No. 4); speed 108 r.p.m., motor belted direct; shearing steel $1\frac{1}{4} \times \frac{5}{16}$ inch, wrought iron $1 \times \frac{1}{2}$ inch (short pieces), steel $3 \times \frac{1}{4}$ inch for cultivator shovels, steel $5 \times \frac{3}{16}$ inch, diamond-shaped, for shovels.		
Average power for 4-hour run		.46
Maximum momentary power		2.49
Very little variation in current except when cutting short pieces rapidly. This gave the maximum power for the wrought-iron pieces.		
Punch-press, large (Hartmann); overreach 28 inches, stroke 3 inches; 8 strokes per minute.		
Machine running idle		1.0
Punching $1\frac{1}{2}$-inch square holes in 1-inch plate		4.41
Punch-press, large, fitted with two fly-wheels weighing 1000 lbs. each; machine geared direct to 10-H.P. shunt-wound motor, 220 volts; current varied from 5 to 150 amperes.		
Motor and punch empty, starting	13.1	
Motor and punch running idle	1.5	
Motor and punch at the time of punching $1\frac{5}{8}$-inch holes through 1-inch steel plate (regular work of the machine is much lighter)	43.2	
The work was too severe for the motor, and injured the armature after punching about 200 holes consecutively.		
Punch for Rails; 34 strokes per minute;		

PUNCHING AND SHEARING MACHINES—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
5-H.P. individual shunt-motor belted to machine; speed of motor 1000 r.p.m., 220 volts.		
Starting punch........................	8.9	
Punching three $1\frac{1}{4}$-inch holes in neck of rails, $\frac{1}{2}$ inch thick....................	1.8 to 3.	
Two Angle-punches and one Double Plate-planer; driven by line shaft 45 feet long, $2\frac{3}{16}$-inch diameter, with seven hangers; speed of line shaft 750 r.p.m; 25-H.P. shunt-wound motor coupled direct to line shaft; 220 volts, 10 to 110 amperes; punching done cold.		
Motor and line shaft (belts on loose pulleys)............................	3	
One machine punching holes $\frac{15}{16}$ inch diameter in $\frac{1}{2}$-inch angles............	5.9	3.5 est.
Two machines punching holes $\frac{15}{16}$ inch diameter in $\frac{1}{2}$-inch angles............	8.8	5.7 "
Two punches and double planer, working..................................	13.3	9.3 "
Moment of reversing planer...........	26.0 to 31.7	21 to 24 est.
The planer and punches are belted from countershaft. Each punch has two 1000-lb. fly-wheels.		
Small Cropping-shears geared to motor; average character of work, shearing hot steel 1 inch square; 1-H.P. motor, 220 volts; current varied from 2 to $9\frac{1}{2}$ amperes.		
Motor and shears, running............	.6	
Motor and shears at the moment of cutting (with shunt field-coils).......	2.9	
Motor and shears at the moment of cutting (with compound field-coils)..	1.7	
Motor was not of sufficient capacity to do the work. After compounding the field-coils of the motor the current		

Punching and Shearing Machines—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
was reduced about 4 amperes, but the machine still remained insufficient.		
Scrap-shears, connected direct to motor; pinion fitted with slipping clutch; shearing done cold; 15-H.P. shunt-wound motor, 220 volts; current varied from 10 to 73 amperes.		
Motor and shears, starting............	20.6	
Running idle, at speed................	2.9	
Cutting 4 × ½-inch steel plate..........	5.9	
Cutting 7 × ½-inch steel plate..........	8.8	
Cutting 11 × ¾-inch steel plate........	17.7	
Shearing steel rails 3 inches high, 2¾-inch base, 1⅞-inch tread.............	15.63 to 16.21	
Shearing steel bars 2½ inches diameter.	21.5	
Scrap-shears, same machine as preceding, connected direct to compound-wound motor; shearing done cold; motor the same as preceding with winding changed; voltage 220, current from 10 to 50 amperes.		
Motor and shears, starting............	14.7	
Running idle at speed................	2.9	
Shearing 6½ × ½-inch plate............	5.0	
Shearing 11 × ¾-inch plate............	13.3	
Shearing steel rails same as preceding.	12.4	
Shearing steel bars 2½ inches diameter.	14.7	

	Motor Input.	
	Shunt-wound.	Compound-wound.
Double Angle-shears (Hilles & Jones), connected direct to motor. Comparison of shunt and compound winding; 40-H.P. motor; voltage 220; shearing done cold.		
Motor and shears, starting............	38	35
Motor and shears, running............	5.3–7.4	5.9
Cutting 6 × 3½ × ½-inch steel angles....	10.3	8.8

PUNCHING AND SHEARING MACHINES—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
Plate Shears, large (Hartmann); 28-inch blades; stroke 3 inches; 7 strokes per minute.		
Machine running idle..................		.67
Cutting 1-inch iron plate, continuous cut at rate of 3$\frac{1}{2}$ ft.p.m........		7.12
Shear for Cold Steel Plates, geared to 10-H. P. motor, shunt-wound, 220 volts, current from 5 to 77 amperes.		
Motor and shears, starting............	14.6	
Running idle when at speed...........	1.5	
Cutting 3$\frac{1}{2}$ × $\frac{3}{8}$-inch plate	5.9	
Cutting 8 × $\frac{5}{8}$-inch plate	11.2	
Cutting 14 × $\frac{3}{4}$-inch plate	22.2	
The flow of current was excessive and frequently injured the armature. A slipping clutch was subsequently provided for pinion. No injury to armature occurred after the clutch was added. The results of tests made on the machine thus equipped with the slipping clutch were as follows:		
Motor and shears, starting..........	11.7	
Running idle when once up to speed.	1.5	
Cutting 8 × 1-inch plate............	10.3	
Cutting 9 × 1$\frac{1}{4}$-inch plate...........	17.5	
Cutting 19 × $\frac{15}{16}$-inch plate..........	20.2	
Angle-shears, with fly-wheel of 1000 pounds; driven by countershaft 6 feet long, 2$\frac{1}{4}$-inch diameter, 700 r.p.m., 2 hangers; motor coupled direct to countershaft; 15-H. P. motor, 225 volts, from 4.6 to 40 amperes.		
Motor and countershaft (belts on loose pulleys)............................	1.4	
Shearing one 6$\frac{1}{2}$ × 4 × $\frac{1}{2}$-inch steel angle, cold.........................	12.1	9.2 est.
Hydraulic Shearing-machine (Tangye Bros.); 17-inch blades; 3$\frac{1}{2}$-inch stroke		
Machine running idle		.37
Cutting 2$\frac{1}{4}$ × 5-inch flat iron		1.52

Drop and other Power Hammers.	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
Trip-hammer, 40 lbs. (Bradley).		
Light work, average for 2-hour run....		.88
For full stroke, steady hammering, maximum		3.02
Trip-hammer, 60 lbs. (Bradley).		
Work light; running all the time, 4 hours:		
Average		1.11
Maximum..........................		3.08
Trip-hammer, 200 lbs. (Bradley).		
Average power for 3-hour run		2.85
Average power while in operation after starting............................		6.34
Maximum record during the test......		9.6
Spring-hammer, 55 lbs. (Justice); motor belted direct; 360 strokes per minute; welding wheel-rims, 30 per hour for 2 hours.		
Average		.41
Maximum for starting hammer........		3.45
Maximum while working..............		2.27
Drop-hammer (Williams, White & Co.); weight of hammer-head 525 lbs.; weight of die 300 lbs.; lift 11 inches; straightening 1¼-inch round iron.		
Average power during 3-hour test		1.08
Maximum while lifting		3.09
Grinding-machinery.		
Emery-grinder, small, double-head, for tools; diameter of wheel 7½ inches; face ⅞ inch; speed 2250 r.p.m		
Grinding planer-knives, average for 1-hour run..........................		.39
Emery-grinder, large, double-head; motor belted direct; diameter of		

GRINDING-MACHINERY—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
wheels 9 and 11 inches; 2-inch face; speed 1500 r.p.m.		
Two men working, light work, average for 1 hour..........................		1.11
Two men working, heavy work		3.76
Grinder for Planer- and Shaper-knives (Kerrick); cup emery-wheel, 8 inches diameter, 1800 r.p.m.; motor belted direct; table travel 18 inches in one case and 11 inches in another.		
Average power in 2-hour test		.52
Tool-grinder (Gisholt); 11-inch wheel.		
Grinding shop-tools, average for 2-hour run..................................		.67
Average while working...............		.8
Grinding-machine (Landis); grinding hardened steel cylinder $1\frac{7}{16}$-inch hole; $1\frac{1}{8}$-inch wheel.		
Average power for $2\frac{1}{2}$ hours...........		.35
Grinding-machine (Brown & Sharpe, No. 1); 7-inch wheel; grinding cast iron 2 inches diameter, 3 inches long; also hardened steel $1\frac{1}{2}$ inches diameter, 8 inches long.		
Average power for 5-hour run		.35
Grinding-machine (Brown & Sharpe, No. 2).		
Average power for 2-hour run with 11-inch wheel		.25
Another similar machine grinding hardened steel; wheel speed of 2200 r.p.m.		
Average power for $2\frac{1}{2}$ hours		.49
Saw-grinder; diameter of emery-wheel		

Grinding-machinery—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
11½ inches; thickness ½ inch; velocity 5500 ft.p.m. at 1850 r.p.m.		
Running idle........................		.40
Sharpening saw......................		.56
Polishing-machine, double; motor belted direct; 14-inch wheels; work crowded heavily.		
Average power for 1 hour............		2.62
Maximum for ordinary working....... Variatious sudden and large.		5.6
Grindstone, fine-grained, for tools; 31 inches diameter, 6-inch face; run with cone pulley at 240, 140 and 85 r.p.m.; corresponding velocities 2000, 1200, 700 ft.p.m.		
Running idle at 85 and 140 r.p.m. (with countershaft)......................		.32 to .39
Countershaft alone..................		.16
Grinding wrought-iron stock 2½ inches broad, pressure of 56 lbs.; velocity of stone 700 ft.p.m..................		1.55
Grindstone, coarse-grained, 43 inches diameter, 11½ inches face, 150 r.p.m., velocity 1700 ft.p.m.		
Running idle........................		.24
Grinding wrought-iron stock 3 inches wide, with a pressure of 130 lbs. against the stone..................		3.11
Grindstone, 7 ft. in diameter, 85 r.p.m., cutting velocity 1900 ft.p.m.		
Grinding plows, average power for 4-hour test.........................		3.65
Average power while working........		4.35
Maximum during test................		10.36

WOOD-WORKING MACHINERY.	Observed Horse-power.	
WOOD-LATHES.	Motor Input.	Delivered to Machine.
Pattern-maker's lathe, small swing but heavy pattern; speed 900 r.p.m.; motor belted direct to lathe.		
Motor idle..............................	.96	
Driving lathe idle.....................	2	.9
Turning seasoned poplar, 12 inches diameter, ½-inch cut..................	3.2	2
Pattern-maker's Facing-lathe, 32-inch face-plate; will swing 100 inches; speed from 44 to 480 r.p.m.		
Running idle (from .15 to 1.14 H.P.)...		.63
Facing 40-inch disk, pine, $\frac{1}{16}$-inch depth of cut; breadth .02 inch; velocity of rim 2500 ft.p.m.....................		.93
Copying-lathe (Blanchard type) for turning hammer-handles, spokes, etc.; will swing 5 inches diameter, 45 inches long; speed of work 6 to 40 r.p.m.; of cutter, 2300.		
Running idle (from .17 to .44 H.P.)....		.18
Turning ash axe-handles..............		.43
WOOD-PLANERS, MATCHERS, AND SHAPING-MACHINES.		
Wood-planer, 17-inch; speed of knives 2350 r.p.m. or 4700 ft. per minute circumferential velocity.		
Running idle........................		1.25
Planing 11-inch pine boards, depth of cut .42 inch, feed 14 ft. per minute..		4.63
Wood-planer or jointer, 24-inch (cylinder machine); hand feed; belt pulley 4-inch diameter, 5-inch face; speed of knives 3200 r.p.m., cutting speed 4000 ft.p.m.; motor belted direct to knife-cylinder.		
Motor only..........................	.96	

Wood-planers, Matchers, and Shaping-machines—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Driving machine idle.................	2.40	1.4
Planing white pine, cut .11 inch deep by 18 inches wide, feed 25 ft. per minute............................	4.80	3.8
Wood-planer, 24-inch (cylinder pattern); belt pulley 5-inch diameter, 5-inch face; 2250 r.p.m.; cutting speed 3100 ft.p.m.; power feed; motor belted to knife-cylinder.		
Motor only...........................	.96	
Driving machine idle.................	2.40	1.4
Planing pine, cut $\frac{1}{16}$-inch deep 18 inches wide, 11 ft. per minute. .... Planing oak, cut $\frac{1}{16}$ inch deep, $6\frac{1}{2}$ inches wide, 11 ft. per minute......	3.6	2.5
Wood-planer and Molder; 15-inch heavy 4-side molder with roll feed; lower knives 1800 r.p.m., upper 2700, sides 1800.		
Running idle		4.18
Planing 14-inch by $3\frac{1}{2}$-inch boards, depth of cut $\frac{8}{16}$ inch, feed $4\frac{1}{2}$ feet per minute		6.91
Wood-planer (Daniels pattern); cutters 30 inches from centre to centre; 700 revolutions of cutter-head per minute; cutting velocity 5400 ft.p.m.; feed 7 ft.p.m.		
Running idle		1.45
Planing 15-inch boards (red beech), cut .2 inch deep		3.2
Wood-planer (Daniels pattern); machine bed $2\frac{1}{2}$ feet by $21\frac{1}{2}$ feet; belt pulley 13-inch diameter, $5\frac{1}{2}$-inch face; speed 350 r.p.m.; velocity of cutters 10400 ft.p.m.; power feed; motor belted to countershaft.		
Motor only	.96	
Driving machine idle.................	3.9	2.8

Wood-planers, Matchers, and Shaping-machines—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machines.
Planing seasoned oak, cut $\frac{3}{16}$ inch deep 20 inches wide, 12 ft.p.m.	6.2	5.1
Wood-surfacer, 24-inch; Whitney's pattern with elevating table and roll feed; speed of knives 1875 r.p.m. or 2250 ft.p.m. cutting speed.		
Running idle		1.42
Planing 11-inch pine boards, depth of cut $\frac{3}{32}$ inch, feed 10 ft.p.m.		3.03
Wood-planer, 26-inch (Witherby, Rugg & Richardson); speed of knives 3500 r.p.m.		
Planing Georgia pine and maple 2 by 4 inches, average power; 3-hour test, variations sudden and large		8.53
Maximum power		15.5
Wood-planer and Matcher, 6 × 14-inch (Glen Cove); shaft 700 r.p.m.; top cutters 5 inches diameter, side 6 inches diameter; speed 3600 r.p.m.; 19 cubic feet hemlock chips per hour. The hemlock was quite wet.		7.5 to 8.2
Wood-planer and Molder; 8-inch 4-side molding-machine (Egan); roll feed; driving shaft 1100 r.p.m., top cutter 5-inch diameter, 4000 r.p.m.; 15.5 cubic feet white-pine chips per hour, feed 25 ft.p.m.		4.0
Wood-planer and Molder; 13-inch heavy 4-side molding machine with roll feed (Robinson); 1750 r.p.m. of the horizontal knives, 1800 r.p.m. of the vertical.		
Running idle		3.35
Surfacing 9-inch pine board, depth of cut $\frac{1}{8}$ inch, feed 7 ft.p.m.		4.24
Edge Molding and Shaping Machine (Zimmermann); vertical spindle 2000 r.p.m.; cutting velocity 2020 feet per		

Wood-planers, Matchers, and Shaping-machines—*Continued.*	Observed Horse-power. Motor Input.	Delivered to Machine.
minute; cutter 3¾ inches dia., 1¼-inch face; 6 cutting-faces, 3 for right- and 3 for left-hand rotation.		
Running idle........................		
Cutting soft wood molding, feed 4 ft. p.m..............................		1.3
Carver and Molder; speed of cutter 5250 r.p.m.; motor belted direct to cutter-spindle.		2.0
Motor only..........................	.96	
Driving cutter idle..................	2.8	1.7
Cutting groove, circular sector, 2 inches wide, ¾ inch deep, 3½ feet per minute in white pine......................	3.9	2.9
Wood-shaper (Buss Machine Works); two heads; knives 3½ inches dia.; cutting 2⅜-inch hard maple; shaping to slight curve on template; both heads in use.		
Average power for 4-hour test........		2.87
Wood-boring, Mortising, and Tenoning Machines.		
Wood-boring Machine; speed of bit 375 r.p.m.; hand feed; motor belted to bit-shaft.		
Motor only..........................	.96	
Driving machine idle..................	1.7	.5
Boring 4-inch hole in seasoned oak, 10 feet per minute..................	2.3	1.2
Wood-boring Machine; belt pulley 8 inches diameter, 3-inch face; speed 750 r.p.m.; hand feed; motor belted to machine-shaft.		
Motor only..........................	.96	
Driving machine idle................	1.9	.8
Boring 1-inch hole in oak, feed 45 inches per minute................ ; Boring 1⅜-inch hole in oak, feed 8½ inches per minute...	2.2	1.1

WOOD-BORING, MORTISING, AND TENONING MACHINES—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
Gang Boring-machine (Williams, White & Co.); motor belted direct; five bits driven by 6-inch belt; speed of bits 2000 r.p.m.; $\frac{13}{32}$-, $\frac{7}{16}$-, and $\frac{1}{2}$-inch bits.		
Boring hard maple $1\frac{3}{4}$ inches thick, feed 6 ft. per minute		5.75
First ten minutes of run the average power was		6.54
The ammeter showed no appreciable difference whether bits were working or running idle.		
Horizontal Wood-mortiser and Boring-machine; will mortise $8\frac{1}{2}$ inches deep by 11 inches long; 86 strokes per minute.		
Running empty......................		.34
Cutting mortise in soft wood $1\frac{1}{4}$ inches wide, thickness of cut $\frac{1}{8}$ inch, mean cutting velocity 90 ft. per minute....		.49
Horizontal Wood-mortiser and Boring-machine; will mortise $8\frac{1}{2}$ inches deep, $11\frac{1}{2}$ inches long; speeds from 400 to 1550 r. p. m.		
Running idle (from .65 to 2.0 H. P.)...		1.67
Boring 4-inch hole in pine, 1000 r.p.m., feed $7\frac{1}{2}$ inches per minute..........		3.68
Tenon and Mortiser (Zimmermann); will cut tenon $6\frac{1}{2}$ inches long; speed of cutters for tenon 1450 r. p. m., cutting velocity 4500 ft. per minute.		
Running idle........................		2.17
Cutting tenon in pine $5\frac{3}{4}$ inches long by $\frac{3}{4}$ inch deep; feed (by hand) from 6 inches to 2 ft. per minute.........		2.25
Tenon and Mortiser (Hartmann); will cut tenon $8\frac{1}{2}$ inches; speed of knives 2000 r.p.m.		
Running idle........................		1.42

Circular Saws and Band Saws—*Continued.*	Observed Horse-power. Motor Input.	Delivered to Machine.
Cutting tenon in pine 4½ inches long by ½ inch depth of cut; feed 3 ft. per minute..........		2.11
Tenon and Mortiser (Zimmermann); will cut tenon 11 inches; speed 1900 r.p.m.; cutting velocity 5900 ft. per minute.		
Running idle..........		.61
Cutting tenon in pine 3 inches long, depth 1⅞ inches; feed 8½ inches per minute..........		2.73
Circular Saws and Band Saws.		
Rip- and Cross-cut Circular Saw, 12 inches diameter; speed 3300 r.p.m., corresponding to cutting velocity of 4100 ft.p.m.		
Sawing 2-inch elm, work crowded.....		2.48
Circular Rip-saw, 12 inches diameter; speed 2200 r.p.m. or 6900 ft.p.m.; hand feed; belt pulley 3½ inches diameter, 3-inch face; 7½-H.P. motor belted direct to pulley on saw-arbor; saw set to wobble for cutting grooves.		
Motor only..........	.96	
Driving saw idle..........	2.2	1
Cutting groove in seasoned walnut ⅜ × ⅞ inch, 12 ft.p.m..........	3.6	2.6
Circular rip-saw, 13 inches diameter; 1930 r.p.m. or 6500 ft.p.m.; motor belted direct.		
Sawing hard maple plank 2½ to 3 inches thick, machine crowded hard for 3 hours, average power..........		3.73
Variations were sudden; maximum power..........		15
Circular rip-saw, 14 inches diameter;		

Circular Saws and Band Saws—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
speed 2200 r.p.m. or 8100 ft.p.m.; arbor-pulley 3-inch diameter, 5-inch face; hand feed; 7½-H.P. shunt-wound motor belted direct to saw-arbor.		
Motor only..........................	.96	
Motor and saw idle..................	2.7	1.5
Ripping seasoned heart-oak 3½ inches thick, 12 ft.p.m....................	6.3	5.3
Cross-cut Circular Saw, 16 inches diameter; 1780 r.p.m. or 7500 ft.p.m.		
Sawing pine 1¼ × 12 inches; average power in 1 hour's run..............		1.85
Sawing maple 3 × 12 inches; average power in 1 hour's run		2.41
Variations sudden; maximum power..		10.5
Circular Saw, 24 inches diameter; thickness .08 inch; speed 7400 ft.p.m.		
Running idle........................		.7
Ripping 7½-inch pine, 7 ft.p.m.........		3.23
Ripping 7½-inch ash, 4 ft.p.m..........		4.50
Circular Rip-saw, 24 inches diameter; speed 1500 r.p.m. or 9400 ft.p.m.; hand feed; motor belted direct to 7-inch pulley on saw-shaft.		
Motor driving saw, idle...............	3.2	.8
Ripping seasoned heart-oak 6 inches thick, 10 ft.p.m....................	12.8	11.
Ripping seasoned white pine 6½ inches thick, 15 ft.p.m..................	9.4	7.3
Ripping seasoned yellow pine 2 inches thick, 45 ft.p.m..................	10.7	8.9
Circular Rip-saw, 28 inches diameter; speed 1200 r.p.m. or 8800 ft.p.m.; arbor-pulley 5½-inch diameter by 8½-inch face; hand feed; motor belted to saw-shaft.		
Motor and saw idle...................	3.4	2.1

CIRCULAR SAWS AND BAND SAWS—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Ripping seasoned heart-oak 7⅞ inches thick; feed 10 ft.p.m.	19.3	18.8
Circular Saw, 35 inches diameter; thickness .12 inch; speed 7750 ft.p.m.		
Running idle		1.16
Ripping 7½-inch pine, 9 feet per minute		5.6
Ripping 7½-inch ash, 5 feet per minute.		6.7
Circular Saw, 60 inches diameter.		
Sawing 21-inch red-oak logs, feed 33 ft.p.m., 62 square feet of surface cut per minute		35.
Sawing 15-inch ash logs, feed 43 ft.p.m., 54 square feet of surface cut per minute		37.
Same with feed of 45 ft.p.m.		40.
Sawing 6-inch red-oak logs, feed 70 ft.p.m., 35 square feet surface cut per minute		14.6
Band Saw, 28-inch band wheels; speed 480 r.p.m. or 3500 lineal feet per minute; belt-pulley 12 inches diameter, 3½-inch face; hand feed; motor belted to saw-shaft.		
Motor only	.96	
Motor and saw idle	1.7	.90
Ripping seasoned oak 3 inches thick, feed 2½ feet per minute Ripping seasoned pine 3 inches thick, feed 4 feet per minute Cross-cut seasoned oak 3½ inches thick, feed 4 feet per minute	2.3	1.3
Band Saw, 34-inch band wheels; saw-blade $\frac{1}{16}$ inch by ⅜ inch; cutting speed 1350 ft.p.m.		
Running idle		.2
Sawing dry oak 9½ inches thick with feed of 1⅜ ft.p.m., 82 square feet surface per hour		.96

CIRCULAR SAWS AND BAND SAWS—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Band Saw, 36-inch band wheels; motor belted direct; 310 r.p.m. or 2900 ft. p.m. cutting speed; sawing 2⅝-inch hard maple, work crowded.		
Average power for 4½ hours' test......		.66
Variations were sudden; maximum power..............................		2.27
Band Saw, 42-inch band wheels; speed 350 r.p.m. or 3850 ft.p.m.; belt-pulley 16 inches dia., 5-inch face; hand feed; motor belted direct to saw-shaft.		
Motor only..........................	.96	
Motor and saw idle....................	2.9	1.9
Ripping seasoned oak 12 inches thick, feed 3 feet per minute...... Cross-cutting seasoned oak, 8 inches thick, feed 5 feet per minute...... Ripping live oak 10 inches thick, feed 3.2 feet per minute..........	5.7	3.6
Band Saw (Resaw), 72-inch band wheels; speed 160 r.p.m. or 3000 ft.p.m.; belt-pulley 30 inches dia., 8-inch face; power feed; motor belted to counter-shaft.		
Motor and saw idle..................	12.1	10.4
Ripping seasoned ash 10⅝ inches thick, feed 6 feet per minute...............	16.1	14.8
Ripping seasoned white pine 16⅛ inches thick, feed 10 feet per minute........	16.1	14.8
Ripping yellow pine 12 inches thick, 20 feet per minute..................	18.8	17.6
Saw Frame, 24-inch.		
Machine running idle............. ...	9.75	
Starting..................	23.0	
Twelve saws cutting birch logs, 16 inches deep, feed 6 inches per minute	17.6	
Fourteen saws cutting ash logs, 15 inches deep, feed 7 inches per minute	20.7	
Twenty-eight saws cutting yellow pine, 23 inches deep, feed 4 inches per min.	29.1	

MISCELLANEOUS IRON-WORKING MACHINES.

Rolling-mill for Sheet Metal (No. 4, Mossberg & Granville); rolls 5 inches diameter, 8 inches long, 14 r.p.m., 18 ft.p.m. lineal velocity. The journals ($3\frac{3}{8} \times 4\frac{1}{2}$) were fitted with plain bronze bearings, as well as roller bearings, and separate tests were made in each case. Rolling brass, copper, and steel from 2 inches to 3 inches wide and from $\frac{1}{16}$ to $\frac{3}{8}$ inch thick. The amount of reduction per pass varied with the pressure on rolls and the number of passes before annealing the stock.

	Observed Horse-power Delivered to Machine.	
Running machine idle	.12	
	With Plain Bearings.	With Roller Bearings.
Total pressure on rolls 20,000 lbs	3.24	.84
" " " " 40,000 "	4.82	1.71
" " " " 50,000 "	7.21	2.27

It will be noticed that the average power in the above tests is about three times greater with the plain bearings than with the roller bearings.

	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
Plate-bending Machine; rolls 13 inches diameter, $9\frac{1}{2}$ ft. long; speed of rolls 2.86 ft. per minute.		
Running idle		.54
Bending $\frac{9}{16}$-inch plate, $4\frac{1}{2}$ ft. wide by 9 ft. long, into a cylinder, 10 passes per plate. Plates were red-hot.		2.70
Plate-bending Rolls, large size.		
Running idle	5.5	
Raising top roll	8.5	
Lowering top roll	7.0	
Rolling plate .45 inch thick, $4\frac{1}{2}$ ft. wide, $16\frac{1}{2}$ ft. long; endwise on ($4\frac{1}{2}$ ft.)	6.9	

Miscellaneous Iron-working Machines—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
Sideways on (16 ft.)	12.2	
Another plate .55 inch thick 4 ft. 8 inches × 21 ft. long, rolled sideways	19.3	
Plate-planing Machine.		
Running idle	.6	
During reversal	1.8	
Planing 1-inch × 14-ft. plate	3.1	
Foundry Tumbling-barrel, 20 inches diameter by 2 ft. long.		
Average power for 1-hour test with box full		.92 H.P.
Foundry Tumbling-barrel, 36 inches diameter by 4 ft. long, 12 r.p.m.		
Average power for 1-hour test		2.43
The tumbling-barrel was only half full, making a heavier pull than when more fully loaded.		
Cranes and Elevators.		
2-ton Elevator (Reedy, Chicago); speed of elevator 50 feet per minute; 10-H.P. Fort Wayne motor (smooth-core armature), 1800 r.p.m., belted to countershaft.		
Running empty, elevator at rest	1.5	.7
Running empty, up	4.8	3.4
Running empty, down	2.1	1.2
Raising ½ ton (motor 1600 r.p.m.)	7.	5.3
Lowering ½ ton (motor 1780 r.p.m.)	1.2	.5
Raising 1 ton (motor 1580 r.p.m.)	10.3	8.4
Lowering 1 ton (motor 1840 r.p.m.)	.9	
Raising 2 tons	15.5	11.6
Lowering 2 tons	.3	
5-ton Electric Travelling Crane.		
No load:		
Running motor only	1.3	

Cranes and Elevators—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Motor with clutches................	2.	
Bridge, starting....................	13.3	
Bridge, running 300 feet per minute.	6.4	
Traversing trolley..................	2.7	
Raising hook........................	3.	
Lowering hook.......................	2.7	
Loaded with 8500 pounds:		% efficiency.
Lifting at 8 feet per minute........	9.9	48
Lowering at 8 feet per minute.......	4.4	
Traversing trolley..................	3.4	
Bridge, starting....................	20.8	
Bridge, running 300 feet per minute.	6.2	
6-ton Jib Crane.		
No load: running motor and machinery only:		
Clutches open.......................	1.	
Raising hook........................	3.5	
Lowering hook.......................	1.5	
Traversing trolley..................	1.5	
Rotating crane......................	2.1	
Loaded with 11,800 pounds:		
Raising load........................	11.6	
Lowering load.......................	3.	
Traversing trolley..................	5.3	
Rotating crane......................	2.4	
10-ton Electric Travelling Crane.		% efficiency
Raising 22,000 lbs. at 10 feet per minute	14.7	45
Raising 10,200 lbs. at 20 feet per minute	12.4	50
Raising 10,200 lbs. at 20 feet and traversing trolley at 200 feet per minute.	13.3	
Lifting 6000 lbs. at 40 feet per minute..	16.3	45
Lifting 6000 lbs. at 40 feet and traversing trolley 200 feet per minute......	20.9	
10-ton Jib Crane.		
No load: running motor and machinery only:		
Clutches open.......................	1.2	
Rotating crane......................	1.8	
Traversing trolley..................	1.5	
Raising hook 40 feet per minute.....	5.8	

Cranes and Elevators—*Continued.*	Observed Horse-power. Motor Input.	Observed Horse-power. Delivered to Machine.
Lowering hook same speed..........	3.	
Raising hook 10 feet per minute.....	2.4	
Lowering hook same speed	1.5	
Loaded with 7.23 tons:		% efficiency.
Raising load 10 feet per minute......	11.1	39
Lowering load at same speed........	3.	
Traversing trolley...	5.8	
Rotating crane	2.4	
Loaded with 9.43 tons:		% efficiency.
Raising load 10 feet per minute	13.	44
Lowering load at same speed	2.1	
Traversing trolley..................	7.	
Rotating crane	2.3	
15-ton Electric Travelling Crane (Shaw); three motors; 14-H.P. motor on hoisting-train, 275 r.p.m. at full load; 82% efficiency at a little below full load, 80% at full load.		
Raising hook 30.60 ft. per minute.....	6.19	4.8
Lowering hook 19.7 " " "	9.14	7.5
Raising 20,000 lbs. 13.2 ft.p.m.......	14.45	11.4
Lowering 20,000 lbs. 29.6 "	5.75	4.4
Raising 30,000 lbs. 10.9 "	17.98	13.4
Lowering 30,000 lbs. 28.7 "	6.49	5.2
Raising 40,000 lbs. 9.7 "	22.86	14.8
Lowering 40,000 lbs. 28.7 "	7.74	6.2
Traversing trolley, light, 125 ft.p.m.	1.64	
" " 20,000 lbs. 102 "	1.67	
" " 30,000 lbs. 103 "	1.79	
" " 40,000 lbs. 92 "	1.89	
Traversing bridge, light, 200 ft.p.m.	7.52	
" " 20,000 lbs. 193 "	8.45	
" " 30,000 lbs. 186 "	8.40	
" " 40,000 lbs. 185 "	8.86	
The mechanical efficiency of hoisting-train at full load is 68.7%; since the efficiency of hoisting-motor is 80% at full load, the total efficiency is 55%—a very high efficiency for machinery of this character.		

Cranes and Elevators—*Continued.*	Observed Horse-power.	
	Motor Input.	Delivered to Machine.
30-ton Electric Travelling Crane (Sellers); 3 motors; transformer system, loaded with 63,000 pounds; speed of hoist 20 feet per minute; 25-H.P. motor for hoisting, 25-H.P. motor for bridge travel, 5-H.P. motor for trolley travel; speed of bridge travel 350 feet per minute.		
Starting hoist	30.6	
Hoisting	23.	
Lowering load	4.	
Starting bridge	9.7	

Machine Groups.

A few groups of machines operated from a shaft driven by a single motor are presented as interesting illustrations of the well-known fact that the average power required to run a number of machines grouped together may be very much less than the aggregate power required to operate the individual machines.

Group of 5 machines run by a 3-H.P. 110-volt motor: one 17-inch lathe, three 12- to 14-inch lathes, and one slotter. The average electrical input was	1.03 H.P.

Group of 5 machines driven by 10-H.P. 500-volt motor: Planer 16 inches by 3 feet (New Haven Mfg. Co.); return stroke same as forward; light cuts in cast iron and steel. Turret lathe, 21-inch (Gisholt, size H), working on $1\frac{1}{2}$-inch brass valves. Tool-grinder (Gisholt); 6-inch wheel, 2300 r.p.m. (Not much used during test.) Lathe, 14-inch (Blaisdell); variable work. Drill-press, 10-inch. (Not much used during test.)

	Horse-power Delivered to Machines.
Average power for 4-hour run	1.50
Maximum power during test	2.33
Average power for shafting, belts, and pulleys	1.38

Group of 5 machines driven by 10-H. P. 500-volt motor: Bolt-header working on $\frac{7}{16}$-inch iron. Drill-press, 18-inch; $\frac{5}{8}$- and $\frac{1}{2}$-inch drills, drilling steel. Double punch and shearing-machine; $\frac{7}{16}$-inch punch, $\frac{5}{8}$ steel plate. Punch and shear; $\frac{1}{2}$-inch square punch, $\frac{5}{16}$-inch steel plate. Punch and shear; shearing $\frac{5}{8}$-inch round iron, four at a time.

Average power delivered to the group during 4-hour run....................	2.05 H. P.

Group of 20 machines in the Crocker-Wheeler factory, driven by 10-H. P. motor, 110 volts, 750 r.p.m. This group consisted of 8 lathes, 4 milling-machines, 3 planers and shapers, 2 grindstones, 1 vertical-shaft grinding-machine, and 3 small tools. The average number of machines running was 13, most of the lathes being in use the greater part of the time.

	Motor Input.	Delivered to Machines.
Average power for 5-hour test	4.78	3.3 est.

Group of 16 machines and 90 feet of 2-inch shafting running at 200 r.p.m. The machines consisted of 9 lathes of different sizes, ranging from 12- to 22-inch swing; 1 planer, 24 inches by 6 feet; 2 shapers, 15-inch stroke; 2 upright drill-presses, one of them a 24-inch; 1 No. 1. B. & S. milling-machine, and several small tools. When the group was run at its full capacity, all the larger machines being in use, the work was distributed as follows:

	Horse-power Delivered to Machines.
Line shaft only (by dynamometer)....	.84
Thirty belts and loose pulleys.........	1.73
Sixteen machines, empty..............	3.45
Cutting metal........................	4.25
Total	10.27

Of which 2.57 H. P. is required for the shafting and 7.65 H. P. for the machines at work. In this test the machines were working above their normal rate.

Group of 15 machines and small lighting dynamo. The group consisted of one 36-inch by 8 ft. planer, planing steel castings, cut $\frac{1}{4}$ inch to $\frac{3}{8}$ inch deep, feed $\frac{1}{16}$ inch, cutting speed 18 ft. per minute; 15-inch shaper planing cast iron, light cut; 30-inch lathe, boring 5-inch dia. $\frac{1}{8}$-inch cut; three 16-inch lathes, variable work; two 12-inch speed-lathes not working; 15-inch turret lathe, light work; 28-inch drill-press drilling $\frac{3}{4}$-inch holes in steel, work crowded; small sensitive drill (Slate); No. 2 universal milling-machine; punch-press, will punch 12-inch disk, light sheet-iron work; 36-inch grindstone, tool-grinding, work crowded; emery grinder and buffing wheel.

An 8-K.W. 110-volt dynamo was also run in this unit; this machine furnished current for 4 arc lights and 40 16 c.p. incandescent lamps, all running during the tests.

The unit was driven by a 15-H.P 500-volt shunt-wound motor, 1200 r.p.m., connected to line shaft by 6-inch belt. The shaft was $2\frac{1}{4}$ inch dia., 60 feet long, 9 hangers, and ran at 220 r.p.m.

All the above machines were driven through countershafts except the grindstone, punch-press, drill-press, and dynamo.

	Motor Input.
Maximum power required to operate this group, including dynamo, was..	15.2 to 17.1 H.P.
The above power was measured when the work was considerably above the usual rate; more machines were in operation and the dynamo was run at its full load.	
Dynamo-belt thrown off...............	5.8 to 7.3
Planer, 30-inch lathe, and two 16-inch lathes, other belts thrown off, light work and intermittent..............	3.3
When reversing planer, momentary...	8.1
Motor and line shaft, belts off........	2.3

CHAPTER XI.

AN ANALYSIS OF THE POWER REQUIRED TO DRIVE LATHES.

By a comparison of the preceding data it will be observed that the total power may be divided into two parts, one of which is necessary to drive the machine itself; the other, to perform useful work. The latter depends upon a number of varying conditions, the principal factors being, in most cases, the nature and quantity of material removed. If this be referred to a common standard, as for instance the weight of metal or volume of chips removed per hour, then a measure of the power required for the same or similar machines may readily be calculated when any other rate of production is involved. The following analysis of this work, as carried out for a number of lathes, indicates that the method might profitably be extended to include machines of all kinds, or at least those machines which use cutting-tools in one way or another; for it is evident that the power required to operate mere form-changing machines can not be readily compared by referring to their output.

Among cutting-tools the drilling-machine seems to offer an exception to the general rule that the power required to remove metal is proportional to the volume or weight of chips cut in a given time. In this case the depth of hole has much to do with the power required to operate, since an increase in depth may very materially increase the friction of the drill in the hole unless special provision be made to remove the chips.

On the other hand, as the diameter of drill increases the power decreases, owing to the greater drawing-in action of the lips of the drill which relieves the pressure on the drill-spindle and feed mechanism.*

It has been stated that the power required to run a small lathe while taking a light cut in wrought-iron was one-tenth horse-power.

By a comparison of data on the subject it will be seen that the actual power consumed is quite variable, and that the power required to turn off metal may be much less than that required to file or polish the same in the lathe, or even to run the lathe empty.

It is evident that the power required to do useful work varies with the depth and breadth of chip, with the shape of tool, and with the nature and density of metal operated upon; and while it would also appear that the power required to run a machine empty should be constant for a given speed, a little investigation will show that this latter is often a variable quantity.

In the case of a lathe, for instance, when the machine is new, the working parts have not become worn or fitted to each other as they will be after running a few months; and at first, the length of time depending on the frequency of its use, the lathe will run hard, in which condition the power required will be greater than will be the case after the running parts have become worn.

---

* Prof. L. P. Breckenridge in Jour. Eng. Society, Lehigh University, 1888; also *American Machinist*, vol. XXII.

Another cause of variation in this portion of the power absorbed will be found in the driving belt: a tight belt will increase the friction very considerably. Hence to obtain the greatest efficiency of a machine, that is, the ratio of useful work to total power absorbed, we should use wide belts, and run them just tight enough to prevent slip. The belts should also be soft and pliable, otherwise power is consumed in bending them to the curvature of the pulleys.

Another point in this connection, sometimes overlooked, is the relative diameter of cone-pulleys. A belt may be wide enough and loose enough to run well on the larger steps of the driven shaft, but on the higher speeds may be altogether too tight. The writer has in mind a small lathe in which it was necessary to let out the belt three quarters of an inch when changing from the largest to the smallest step on the cone-pulley.

A third cause is the variation of journal-friction, due to slacking up or tightening the cap-screws, and also the end-thrust bearing screw. When one man runs a lathe so that a pull on the belt will revolve the spindle half a dozen times, and another man screws down the boxes of the same lathe so that he can only move the spindle by a series of tugs with both hands on the belt, no dynamometer is needed to show that the power absorbed will be different in the two cases.

The power necessary to drive the lathe over and above that required to turn off metal or do useful work is often increased by setting up the tail-stock centre too hard, or by letting the centre run dry. In one of the writer's experiments it was noticed that the power absorbed constantly increased, and with no perceptible change in thickness of chip or condition of tool. The cut was smooth and clean, yet the dynamometer showed nearly three times the power ordinarily required. The tool was withdrawn from cut with very little change in the pressure; a drop of oil on the dead-centre, however, instantly caused the line to fall, but the normal pressure was reached only when the centre was eased a little. Subsequent trials showed conclusively that the ordinary running power could be more than doubled by carelessness at this point.

Hartig's* investigations show that it requires less total power to turn off a given weight of metal in a given time than it does to plane off the same amount; and also that the power is less for large than for small diameters. This latter fact is readily understood when we consider that the faster we run a lathe-spindle the more power it requires, provided we do not consider the intervention of intermediate gear-shafts; when back gears are used, the power will be less with the belt running upon a given step of the cone-pulley than if the gears were thrown out, although such power may be greater than that required to produce a greater number of revolutions of the spindle when running without back gears.

This is clearly shown in Table IX, which gives the actual (measured) horse-power required to drive a lathe empty at varying numbers of revolutions of main spindle.

* **Versuche über Leistung und Arbeits-Verbrauch der Werkzeugmaschinen.**

TABLE IX.

HORSE-POWER FOR SMALL LATHES.

Without Back Gears.		With Back Gears.		Remarks.
Revolutions of Spindle per minute.	Horse-power required to drive empty.	Revolutions of Spindle per minute.	Horse-power required to drive empty.	
47.0 80.5 138.0	0.101 .118 .147	29.0 49.7 85.3	0.113 .128 .177	12" lathe built by Zimmermann, of Chemnitz, Germany.
47.4 80.4 125.0 188.0	.159 .202 .259 .339	4.84 8.18 12.8 19.2	.132 .150 .187 .230	Small lathe (13½") built by Zimmermann. New machine.
54.6 82.2 122.0 183.0	.206 .260 .339 .455	6.61 9.95 14.8 22.1	.157 .177 .206 .249	17½" lathe built by Zimmermann. New machine.
81.11 132.72 219.08 365.00	.126 .145 .197 .310	8.34 14.6 24.33 38.42	*.133 .126 .141 .274	20" Fitchburg lathe.
18.8 33.5 54.6 82.2	.086 .137 .210 .326	2.31 4.12 6.72 10.8	.035 .047 .063 .087	26" lathe built by Zimmermann.

* Horse-power greater than should be the case, on account of belt rubbing on back-gear hollow shaft.

The only exception is the series of values given for the 12-inch lathe, which show a larger horse-power when the back gears are thrown in.

It will be noticed that the relative speeds of the spindle in this lathe are not properly proportioned, there being, practically, only four variations of speed, whereas there should be six.

The ratio of the smallest number of turns of the cone to the smallest number of turns of the spindle is $\frac{47}{29}$, or 1.62 to 1, instead of 8 or 9 to 1 as in the other cases, from which fact we are led to conclude that if the ratio be small between the speed of the cone (which revolves freely on the spindle when back gears are used) and the speed of the spindle, the work spent in driving the spindle through the back gears will be greater than the work saved by reduced friction in the bearings.

From this it is apparent that there will be a certain ratio of reduction for which the power required to run the lathe with or without back gears will be the same: with a greater ratio it will take less power with the back gears in, and with a lesser ratio more power will be required.

Assuming the lathe to be in good condition, the brasses a good running fit, and the belt not too tight, we see from these results that, in order to estimate the total horse-power required to do a certain amount of work, we must know something about the speed at which the lathe will be run, which speed is, of course, dependent upon the diameter and nature of work.

If we plot the curve of horse-power and revolutions from the above table, we shall obtain straight lines, or approximately straight lines, as shown in Fig. 116, which is drawn for four different lathes varying in size from 12 to 20 inches swing—the power necessary to drive with back gears in not being considered in this diagram.

Here the number of revolutions of spindle per minute is given on the extreme left, and the horse-power on the lower line. All the lines emanate near the point A, and diverge with different degrees of rapidity. The lines for the 12-and 20-inch lathes are parallel for a portion of their length, and show the least increase in horse-power for a given increase in speed. The line for the 17½-inch lathe falls away very

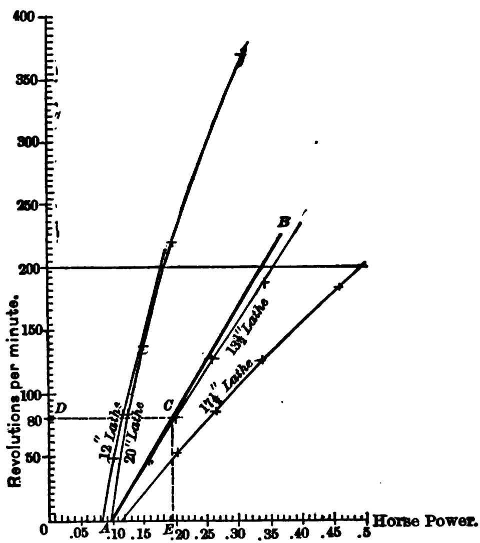

FIG. 116.—HORSE-POWER REQUIRED TO DRIVE SMALL LATHES.

quickly, which shows a rapid increase in power for a given increase in number of revolutions. The average for the four lathes represented is given in the line AB, which strikes the base-line for horse-power at the point about .095, corresponding to which the revolutions per minute = 0.

If we wish to find the horse-power for any given speed, say 80 revolutions per minute, we have simply to draw a line parallel to the base line from the point 80 on the revolution scale, until it cuts the line AB; the distance cut off, DC, or what is the same thing, oE, will give the horse-power direct. In the assumed case it is .19 +. As AB was chosen as the mean of all the lines represented, the horse power measured upon it can only approximate the actual, which, in the extreme cases, 12- and 17½-inch lathes, will vary more and more as the speed increases. If we let $H.P._0$ = horse-power necessary to drive lathe empty, and N = number of revolutions per minute, then the equation for the line AB will be

$$H.P._0 = 0.095 + 0.0012N.$$

In the same way we can plot similar curves for the power necessary to drive the lathes empty when the back gears are in; from this we can assume an average and find the equation of the line as before. With the same notation previously given, this equation for lathes under 20 inches swing is

$$H.P._0 = 0.10 + 0.006N.$$

The larger lathes vary so much in construction and detail that no general rule can be obtained which will give, even approximately, the power required to run them, and although the formulas just obtained show that at least 0.095 horse-power is needed to start the small lathes, for which the line AB has been drawn, Fig. 69, there are unquestionably many American lathes under 20 inches swing working on a consumption of less than .05 horse-power.

The amount of power required to remove metal in a machine is also variable, but determinable within more accurate limits. The shape and condition of tool, the hardness of material to be cut, the rate of feed and depth of cut, all affect the final result.

Every machinist has some special form of lathe-tool, ground to some particular angle, and with a given amount of rake—not measured, but ground so it will look right—which will give the cleanest cut, and turn off the metal quicker than any other tool in the shop.

One man uses a round-nose tool with no top-rake for cast-iron, and with a coarse feed and square chip (depth of chip equal to the breadth) turns out considerable work during the day.

Another uses a diamond point, or some other form, which he grinds and sets a little differently from every one else, and he, too, turns out a large quantity of work. For wrought-iron, steel, or brass, it is the same; every man has his own form of tool which he considers will do the best and quickest work.

Without entering into the question of which is the better form of tool to use in a given case, we shall assume ordinary conditions, and try to ascertain how much power is absorbed by a lathe in removing metal; the power required to run the lathe will not be included in this discussion. As a means of convenient comparison, the work done has been reduced to the weight of metal removed, or that would be removed, in one hour, provided all conditions remained the same.

Referring again to Dr. Hartig's researches, we find, in connection with a 13½-inch lathe, that the greatest amount of work done per hour was 11.55 pounds of wrought-iron removed under the following conditions:

Cutting speed = 24.6 feet per minute; breadth of cut = .017 inch; depth of cut = .08 inch; horse-power ($H.P._1$) required to turn off the metal = .230.

As a result of twenty-three experiments on this lathe, we find that

$$H.P._1 = CW,$$

where C is a constant, and W the weight of chips removed per hour;

$$C = .032 \text{ for wrought-iron, and}$$
$$C = .025 \text{ for cast-iron.}$$

The greatest amount of wrought-iron removed per hour on a 17½-inch lathe was 25 pounds; the cutting speed was 15.88 feet per minute; breadth of cut was .04 inch, and thickness of chip = .20 inch; horse-power required = .66.

The following is the result of forty-one experiments on this lathe:

$$H.P._1 = CW,$$

where, as before, W = weight of chips per hour;

$$C = .045 \text{ for steel,}$$
$$= .027 \text{ for wrought-iron,}$$
$$= .03 \text{ for cast-iron.}$$

On a 26-inch lathe the weight of chips removed per hour = 10.93 pounds; the cutting speed = 32 feet per minute; breadth of cut = .024 inch, and thickness of cut .08 inch; horse-power required = .413. From ten measurements the constant C was found to be .04 for cast-iron; as this is 25 per cent higher than ordinarily obtained, we infer that the iron was much harder, or the tool in poor condition, or perhaps both affect the result.

For a heavy turning and facing lathe 50 inches swing, treble geared, as a result of twenty experiments, C was found to be .027 for cast-iron.

For another heavy lathe, 56 inches diameter of face-plate, C was found to be .031 for cast-iron, as a result of twenty-four experiments.

A small lathe of 12 inches swing, turning wrought-iron at the rate of 4.88 pounds per hour, at a velocity of 23.4 feet per minute, breadth of cut = .018 inch, thickness of chip = .062 inch, required .21 horse-power. For this lathe

$$C = .045 \text{ for wrought-iron,}$$
$$= .028 \text{ for cast-iron.}$$

From these and other data we find that the horse-power required to turn off metal can be obtained, if we know the amount of chips removed per hour, by using the formula

$$H.P._1 = CW,$$

in which suitable values of C obtained from the foregoing are

.030 for cast-iron,
.032 for wrought-iron,
.047 for steel.

As we should infer, the size of lathe, and therefore the diameter of work, has no apparent effect on the cutting power, as shown by the constant C. If the lathe be heavy the cut can be increased, and consequently the weight of chips increased, but the value of

C appears to be about the same for a given metal through several varying sizes of lathes.

Mr. J. F. Hobart, working on this line a few years ago, published some interesting results in the *American Machinist*,* from which the writer has computed the average weight of metal removed per hour, and the corresponding useful horse-power—the horse-power required to run lathe empty being neglected,—from which the values of *C*, in the subjoined Table No. X, have been obtained.

All the experiments were conducted on a 20-inch Fitchburg lathe (previously mentioned in Table IX), and the metal cut throughout was cast-iron.

TABLE X.

HORSE-POWER REQUIRED TO REMOVE METAL IN A 20-INCH LATHE.

Descriptive Number.	Number of Trials.	Tool used.	Average Cutting Speed in Feet per Minute.	Depth of Cut in Inches.	Average Breadth of Cut in Inches.	Average H. P. required to remove Metal.	Average Lbs. of Metal turned off per Hour.	Value of Constant *C*.
1	22	Side tool..........	37.90	.125	.015	.342	13.30	.025
2	15	Diamond..........	30.50	.125	.015	.218	10.70	.020
3	17	Round-nose........	42.61	.125	.015	352	14.95	.023
4	2	Left-hand round-nose............	26.29	.125	.015	.237	9.22	.026
5	4	Square-faced tool ½" broad.........	25.82	.015	.125	.255	9.06	.028
6	1	Square-faced tool ½" broad.........	25.27	.048	.048	.200	10.89	.018
7	1	Square-faced tool ½" broad.........	25.64	.125	.015	.246	8.99	.027

* See *American Machinist*, Sept. 11 and 18, 1886.

An examination of the above table shows that an average of .26 horse-power is required to turn off 10 pounds of cast-iron per hour, from which we obtain the average value of the constant $C = .024$.

As will be noticed, most of the cuts were taken so that the metal would be reduced $\frac{1}{4}''$ in diameter; with a broad surface cut and a coarse feed, as in No. 5, the power required per pound of chips removed in a given time was a maximum; the least power per unit of weight removed being required when the chip was square, as in No. 6.

The work of R. H. Smith,* who conducted similar experiments in England on a 29-inch lathe ($14\frac{1}{2}$-inch centre), has also been cast into the same form with the average results shown in Table XI.

TABLE XI.

HORSE-POWER REQUIRED TO REMOVE METAL IN A 29-INCH LATHE.

Number of Experiments.	Metal.	Cutting Speed.	Depth of Cut.	Average Breadth of Cut.	Average Horse-power required to remove Metal.	Average Pounds Metal removed per Hour.	Value of C.
4	Cast-iron	12.7	.05	.046	.105	5.49	.019
4	Cast-iron	11.1	.135	.046	.217	12.96	.017
2	Cast-iron	12.85	.04	.038	.098	3.66	.027
4	Wrought-iron	9.6	.03	.046	.059	2.49	.023
4	Wrought-iron	9.1	.06	.046	.138	4.72	.029
4	Wrought-iron	7.9	.14	.046	.186	9.56	.019
2	Wrought-iron	9.35	.045	.038	.092	2.99	.031
4	Steel	6	.02	.046	.043	1.03	.042
4	Steel	5.8	.04	.046	.085	2.0	.042
4	Steel	5.1	.06	.046	.108	2.64	.040

* See Cutting Tools, by R. H. Smith, Cassell & Co., London, 1884.

Besides the general results which we here obtain, the relative cutting speeds and amount of metal turned off per hour is quite significant. With the American lathe the cutting speeds on cast-iron were about 30 feet per minute; the German experiments on the same metal averaged 19 feet, while the English speeds on cast-iron were always much less, varying from 14 to $2\frac{1}{2}$ feet per minute.

The small values of C, .017 and .019, obtained for cast-iron, are probably due to two reasons: the iron was soft and of fine quality, known as pulley-metal, requiring less power to cut; and, as Prof. Smith remarks, a lower cutting speed also takes less horse-power.

In summing up the results here presented, if we omit for the present the power necessary to overcome the internal friction of the lathe, there would seem to be no good reason why an average of the cases cited should not be taken as representing average practice. Hardness of metals and forms of tools vary, otherwise the amount of chips turned out per hour per horse-power would be practically constant, the higher cutting speeds decreasing but slightly the visible work done.

Taking into account these variations, we find that the weight of metal removed per hour, multiplied by a certain constant, is equal to the power necessary to do the work.

This constant we have deduced as follows:

	Cast-iron.	Wrought-iron.	Steel.
Hartig	0.030	0.032	0.047
Smith	.023	.028	.042
Hobart	.024		
Average	.026	.030	.044

For a cut under ordinary conditions which would remove 6 pounds of cast-iron, 5 pounds of wrought-iron, or $3\frac{1}{2}$ pounds of steel chips per hour, the horse-power necessary would be practically the same.

$$
\begin{aligned}
C \times W &= .026 \times 6 = .15 \ H.P. \\
C \times W &= .030 \times 5 = .15 \ H.P. \\
C \times W &= .044 \times 3\tfrac{1}{2} = .15 \ H.P.
\end{aligned}
$$

As previously shown, the power necessary to run the lathe empty will vary from about .05 to .3 H.P., which should be ascertained and added to the useful horse-power, to obtain the total power expended.

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