THE
INSTITUTION
OF
CIVIL ENGINEERS.

SESSION 1888-89.—PART I.

SECT. I.—MINUTES OF PROCEEDINGS.

13 November, 1888.

SIR GEORGE B. BRUCE, President,
in the Chair.

The Telford and Watt Medals, the George Stephenson Medal, the Telford Premiums, the Manby Premium, and the Miller Prizes, awarded for the Session 1887-88 (vol. xciv. pp. 152, 153) were presented by the President to the various recipients.

(Paper No. 2328.)

“Friction-Brake Dynamometers.”

By WILLIAM WORBY BEAUMONT, M. Inst. C.E.

ALTHOUGH the friction-brake dynamometer can never give a scientifically accurate measurement of the rate of absorption of the work done upon it by any motor, it has been much used for that purpose, with results not sufficiently incorrect to affect materially the truth of calculations based upon them.

The indications of any apparatus depending on friction must be as variable as the causes of friction and the conditions affecting it, and hence the value of the measurements, obtained by means of a friction-brake dynamometer, must depend on the completeness with which these causes and conditions are taken into consideration or are eliminated.

In a friction-brake the causes of friction are similar to those which generally obtain in other applications of materials when brought into rubbing contact; but the conditions are more variable, and are chiefly due to variation in pressure and in lubrication, both these being affected by variation in the rate of work-absorption and temperature.

From time to time the accuracy of the measurements of power obtained with friction-dynamometers has been questioned; and this Paper is written with the object of placing such facts and considerations before the Institution as will elicit the views which are supported by practice among engineers.

[THE INST. C.E. VOL. XCV.]

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Friction-brakes in use, and recognized as satisfactory instruments for the measurement of power, or as the best available, are very few; but as the friction-dynamometer is an instrument of great importance to engineers and to the users of steam and other motors, some of them will be described, with a view to comment on those questions of principle of construction and of working, which are primary to confidence in their indications.

The friction-brake dynamometer in nearly all its forms is essentially that devised by Prony, an outline diagram of which is shown in Fig. 1. Here the friction between the surface of a wheel A, rotated by the motor whose power is to be measured, and

FIG. 1.

the wood blocks W above and below the wheel, is employed in maintaining a weight P, suspended at a point on a horizontal line level with the centre of the wheel. The point is at some distance above the lever B, by which the load is supported, and, in order that the distance R from the centre of the wheel to the vertical centre of the load may remain constant, the load is suspended from a quadrant whose radius is R. This brake is capable of sufficiently exact determinations of power, ranging from 5 to 200 HP., for most practical purposes, where more than a mere check upon calculations is required; but it presents some of the elements of inaccuracy which pertain to the friction type of absorption-

dynamometers, especially when used for the measurements of variable powers. For measuring the power of a motor, capable of running at a uniform speed with a constant load, the inaccuracy of its indications may be very small and often insignificant; but when the power varies, the inertia of the lever B, and the work done in moving it and the load P, count without record against the motor. Moreover, a small load due to the lever, though balanced while the lever is horizontal, constitutes a load against the motor when the lever rises; and the work done when this load is lifted is also neither recorded nor measured. With a truly circular wheel and uniform turning power, this brake gives very nearly accurate results, provided the lubrication between the rubbing surfaces be

FIG. 2.

uniformly maintained; but slight variations in this respect, due to variation in quantity, quality, and temperature of the lubricant and rubbing surfaces, make it difficult to keep the tension of the strap E in strict accord with the total friction necessary to maintain the load at a constant level. The frequent change of this tension by the screws C for the purpose of meeting these variations introduces further inaccuracy.

A simple form of friction-brake, much used as a dynamometer in portable engine-building establishments, for testing engines by running them against a known load before sending them out, consists simply of a thin iron or steel strap, or a pair of straps, E, Fig. 2, to which are attached a number of blocks W of wood. Sometimes a leather strap is used. At H a hook is fastened to the straps for the suspension of the load P, and at S the ends of the

strap are connected by a right- and left-handed screw for the adjustment of the tension on the strap or pressure of the blocks W upon the wheel A, so as to obtain the necessary frictional grip to carry the otherwise unsupported load P. With a truly turned wheel A, and with uniform lubrication, this brake will run for hours with a variation of but a few inches in the level of the load P, if the engine under test is of good design and has a fly-wheel A of the weight many makers adopt. The variations that do occur from several causes are, however, sufficient to make adjustment by the screw S necessary, sometimes frequently. The errors in estimating the work done by the engine, which result from these causes, are generally small; but it is desirable to remove them, if possible, when very accurate tests are required. It has been with this object that devices for automatically varying the tension in the belt E with the variation in the total friction have been introduced by numerous experimenters; and at an early date a brake-wheel with an internal water channel was used to avoid the mechanical difficulties which resulted from the heating of the brake-wheel and the variations due to the heating of the lubricant.

With either of the brakes mentioned, the work in HP. done in supporting the load P in lbs. at the distance R in feet from the centre of the wheel A, making N revolutions in time T in minutes will be—

$$HP = \frac{(R \times 2) \times \pi \times P \times N}{33,000 \times T},$$

or taking C = circumference of circle of radius R, and V = velocity in feet per minute of circumference, then $V = CN$, and

$$\begin{aligned} HP &= \frac{CNP}{33,000T} &= \frac{VP}{33,000T} \\ P &= \frac{33,000HP}{CN} &= \frac{33,000HP}{V} \\ C &= \frac{33,000HP}{NP}, \text{ and } N = \frac{33,000HP}{CP}. \end{aligned}$$

One form of automatically-adjusting or compensating brake, suitable for small powers, is due to Mr. Deprez, and is shown at Fig. 3. In this¹ a brake-wheel A is attached to a disk B, and to the disk are pivoted levers EE. These levers are connected by a link F and lever G loaded by a weight Q. The weight of the levers EE is balanced by the quadrant-shaped counterpoise C, so that the

¹ See "Guide pour l'essai des machines à vapeur," par J. Buchetti.

weight P is the representative of the work done by the motor, the weight Q having no varying effect on the work done by the motor, or on the position of the load P, so long as the lever G remains horizontal; but as it is by the influence of Q and the lever G that the friction-blocks W are brought to bear upon the brake-wheel A, and as that influence decreases as G leaves the horizontal and approaches the vertical, the weight which the frictional grip will support decreases. Hence the brake becomes automatic in its adjustment of the friction, the descent of the weight P increasing the frictional grip, and correspondingly the lifting of the weight causes a decrease in the friction, so that the weight again falls, a

FIG. 3.

position of equilibrium being ultimately taken, in which load and friction are equal. It will be readily seen, that with any of the causes enumerated, variation of speed of the motor, or oscillation of the load P, the latter must be raised through some distance, or the lever G must pass through a considerable angle before the frictional grip of the brake-blocks is modified. The work of lifting the load being lost work is an objection to this brake. The system of compensation does not, however, contain any element that will affect the load actually carried by the motor, and the only objection to the brake as a dynamometer for small powers is that the compensation takes place so slowly that any of the errors attaching to the simple brake (Fig. 1) remain, and adjustment must fre-

quently be made so that the point of suspension of Q remains truly at the centre of the brake-wheel.

A simple form of self-adjusting brake-dynamometer, due to Mr. J. Imray, M. Inst. C.E., is shown by Fig. 4. A quadrant Q , balanced by a weight w , receives the brake-strap carrying P . A smaller weight p is attached to the other end of the brake-strap. The compensating action is due to the increase in total friction which accompanies increase of circumferential surfaces in contact. If the motor lifts the weight P , the frictional grip is lessened as the arc BH is lessened, and P returns or finds a position of equilibrium.

FIG. 4.

What is spoken of as a modification of this brake consists¹ in using a spring-balance instead of the small weight p ; and in dispensing with the quadrant Q , the wood blocks being continued on the P side of the band, and the lower end of the spring being fixed. This arrangement, also shown on Fig. 4, acts much in the same way as Mr. Imray's brake, but the range of movement necessary for self-adjustment will be less as the pull of the spring

¹ Various attributed to Navier and to Messrs. Easton and Anderson, and to one of Messrs. Ransomes, Sims and Head's managers; but due, the Author believes, to Mr. H. A. Byng of the latter firm, who used it first in Paris in 1867; at Brussels in 1868; at Santiago, Chile, in 1869; and at Cairo in 1874, where it was seen by Sir Frederick Bramwell.

is lessened or increased with any movement, while the weight p in the other case remains the same, and considerable movement must take place so as to make a sensible difference in the total friction due to difference of arc in contact. The spring arrangement is more accurate, as every rise or fall of the weight P decreases or increases the tension on the spring, and thus a very small range of movement of P is sufficient to satisfy small variations in turning moment or frictional grip.

One of the best known forms of friction-brake dynamometers, fitted with a compensating device, is that designed by Mr. C. E. Amos and Mr. Appold, and is the form used for the larger powers by the Royal Agricultural Society. It is similar to that shown by Fig. 2, but, besides the hand-adjusting screw S , it is provided

FIG. 5.

with a compensating lever K as shown in Fig. 5, by means of which the rise or fall of the load P is supposed to be attended with a decrease, or increase, in tension on the brake-strap, so that a position of equilibrium is automatically attained without causing inaccuracy in the indications. With a given tension in the belt, and with the load P carried so that its point of suspension H is opposite the pointer T , the lever K takes a vertical position; but as soon as the load P is lifted, as in Fig. 6, the lever pivoted at X moves with, and virtually increases the length of, the belt, and thus slackens it, allowing the load again to descend. If, on the other hand, the total friction decreases and is insufficient to carry the load in its normal position, the descent of the load carries the compensating lever to a position such as is shown by Fig. 7, thus

tightening the belt and increasing the frictional grip until the conditions are again such as will enable the load to reassume the medial position. If the change in the position of the point of

FIG. 6.

suspension of the load has been due to a temporary cause, this automatic action may restore the balance without further adjustment; but if the departure from the medial position is not small,

FIG. 7.

then the adjustment by the hand-screw S must be resorted to. It will be seen that the compensating action cannot come into play except by the rise or fall of the weight from its proper position,

and hence the value of the device is confined to its power of limiting that rise and fall. In practice, generally speaking, the adjustment required by means of the screw S is as necessary with the compensating lever as without it, and its value may therefore be questioned for this reason alone. A further reason, however, for questioning the value of this compensating lever is that it introduces an element of error, which may be small or considerable, but which must be variable almost directly in proportion to the extent to which the lever comes into play. With a heavily-loaded brake the error must exist, and with a lubricant

FIG. 8.

FIG. 9.

which materially lessens the total friction of the wood blocks upon the wheel, it must amount to a considerable part of the whole indicated power, unless adequate allowance is made for it. This allowance has seldom been made, since it has not been necessary with the conditions under which these friction-brakes have generally been used, namely, with the brake-strap slack as shown exaggerated at N, Fig. 9.

When the lever does act in fulfilling its purpose, the tension in the brake-strap causes it to press against the pivot X with a pressure proportionate to that tension. This will be more fully referred to hereafter.

In a better form of compensating brake designed by Mr. Balk, and used by Messrs. Ransomes, Sims and Jefferies, Fig. 10, the compensating lever is outside, instead of within, the circumference of the brake-strap. It is connected at B and at C to the ends of the strap, and a fixed pin F passes through a slot at the outer end of the lever. To the latter is suspended a scale-pan and a weight p sufficient to keep the lever floating with the pin F midway between the two sides of the slot. This weight p becomes a measure of the tension upon the belt, at least at the parts to which it is attached; but it must be varied with change of condition of the brake-blocks, the lubricant and the temperature of the blocks and wheel, and as it must be taken as acting at the radius O F in favour of the weight P, these variations become troublesome by virtually making P a variant.

FIG. 10.

Fig. 11 shows the larger friction-brake dynamometer used by Messrs. Ransomes, Sims and Jefferies, and is precisely similar to the first one of the kind which Mr. Balk perfected. The Author is indebted to Mr. John Jefferies for this drawing, and for the following tabulated particulars showing the gross load, the weight on the end of the tension or compensating lever, and the variation of the latter with different speeds and powers. These particulars have been taken from the records of numerous experiments made with Messrs. Ransomes' engines by Mr. H. A. Byng. The wheel is 6 feet in diameter, and 1 foot in width, the load being suspended from flat steel springs or tapes, forming a tangent to a circle exactly

20 feet in circumference. A counter is attached to the side of the frame, and thrown in or out of gear by the clutch handle H. The great advantage of this brake over all others is that the experimenter can always ascertain the actual load, although the tension-lever acts as a compensating lever. There is considerable variation in the scale-weights with the same gross load. This, Mr. Byng explains, is due to the heating of the brake-wheel. When much heated, more grease has to be used, and in consequence the coefficient of friction is reduced, and the weights in the scale have to be augmented to give more tension in the brake-straps. The figures appear to show that, the larger the number of revolutions, the smaller the tension for a given total friction, and this infer-

FIG. 11.

ence is supported by experience. Mr. Byng has found that with the higher speeds less weight is required on the scale for a given load, or, in other words, the higher the speed the less the necessary tension in the brake-belt. By means of the Balk brake the tension at the ends of the brake-belt is measured directly, and for accurate trials a higher speed is preferred with less gross load and with a scale-weight below 7 lbs.

The wood blocks, beech or plane-tree, of the brake above referred to, have been in use a long time, and are now semi-charred and saturated with grease; the surface next the wheel has been several times coated with black-lead, and some of the blocks have become

TABLE I.—THE BALK FRICTION-BRAKE DYNAMOMETER.

The following Table shows the gross load and counterbalance weights used in trials, at various speeds and HP.

Gross Load.	On Scale.	Effective Load.	Revolutions.	HP.	Gross Load.	On Scale.	Effective Load.	Revolutions.	HP.
Cwt. qrs. lbs.	Lbs.	Lbs.			Cwt. qrs. lbs.	Lbs.	Lbs.		
1 0 1	4	105	130·20	8·28	2 1 14	17	232	134·42	18·90
0 3 22	2	104	152·10	9·40	2 1 22	11	252	125·20	19·12
1 1 8	6	136	128·50	10·59	2 2 20	20	260	123·50	19·46
1 0 24	5½	125	145·60	11·00	2 1 16	16	236	136·75	19·56
1 1 17	7½	142	140·00	12·00	2 1 18	19	232	139·59	19·62
1 1 10	3	144	138·30	12·07	2 1 16	16	236	144·00	20·60
1 2 1	6½	156	131·25	12·40	2 1 8	12	246	150·00	21·45
1 2 14	7	168	136·92	13·94	2 1 16	16	236	160·00	22·93
1 2 4	3½	165	139·53	13·95	2 1 16	16	236	155·00	22·17
1 2 18	2	182	134·74	14·86	2 1 12	14	236	160·00	23·00
2 0 10	16	202	127·75	15·64	2 3 10	19	280	135·79	23·04
2 0 10	16	202	131·30	16·07	2 3 14	24	274	142·45	24·00
2 0 0	14	202	134·00	16·40	3 1 0	25	314	136·00	25·88
2 0 0	11	202	137·00	16·78	3 1 0	25	314	140·40	26·72
2 0 12	17	202	135·25	16·55	3 0 14	18	314	146·00	27·78
2 0 0	11	202	136·88	16·75	3 0 14	11	328	140·89	28·00
2 0 8	15	202	145·75	17·84	3 2 21	14	385	141·50	33·00
1 3 24	9	202	148·69	18·20	3 3 18	25	388	151·22	35·56
2 0 0	9½	205	146·22	18·16					

almost metallic on the surface. In starting for a set of trials, the wheel and the blocks are thoroughly cleansed and left free of adherent grease. As soon as the wheel gets warm, their saturated condition supplies the necessary lubrication. For loads up to 20 HP. there is no difficulty, the Author is informed, in running this brake all day with perhaps two alterations of the scale weights, and by applying a very small quantity of cold tallow to the wheel and rubbing off the excess immediately with dry waste. Mr. Byng is of opinion that the blocks should be of metal, say Babbit, properly ground to fit the wheel and jointed together. With such a brake-strap he thinks some definite relation between load, tension and velocity might be obtained.

A water-cooled brake used by Messrs. Richard Garrett and Sons,

Leiston, is shown by Figs. 12, 13 and 14. The wheel is 5 feet in

FIG. 12.

FIG. 13.

diameter, and 11 inches wide between the flanges within which the wood blocks run. An annular trough is formed by internally projecting flanges 3·5 inches in depth. The straight form of the Appold compensating lever is employed, though under conditions which seldom, if ever, bring it into play sufficiently to affect materially the accuracy of the indication of the brake. The Author is indebted to Mr. Frank Garrett, M. Inst. C.E., for some interesting particulars concerning the working of this dynamometer. No lubricant is now used; but the blocks, which are of beech, are probably thoroughly saturated with years of previous use on a wheel not cooled by water. It has been ascertained that the brake-wheel with the water contained in it absorbs about 0·75 indicated HP. at 180 revolutions per minute. The circumference of the load circle is 17·5 feet, and the maximum load

FIG. 14.

carried 418 lbs. Three experimental runs with this brake were recently made by Mr. Garrett with a view to obtain figures as to the quantity of water used, its rise in temperature, and the relation between the work mechanically done and thermodynamically accounted for by the heat conveyed to the water. Before starting the engine-water was put into the wheel to the full depth of the annular trough; this water was weighed and called A. As soon as the engine was started water was added, which was also weighed. This second quantity, occupying a depth of 2 inches in the internal annular trough, was called B. The supply is kept up under ordinary circumstances by a pipe constantly passing a small stream into the wheel. During the experiments referred to, however, the water was all weighed and added by hand, the depth of 2 inches being as far as practicable maintained. This quantity may be called C. When the engine is stopped at the end of the run all the water falls out except a quantity equal to the first quantity mentioned, namely, that which fills the lower part of the wheel to the full depth of the trough. During the runs the water was carefully maintained at the depth of 2 inches, so that the quantity added during the run may be taken as evaporated. The engine ran at 130 revolutions per minute.

TABLE II.—EXPERIMENTS WITH GARRETT'S WATER-COOLED BRAKE.

Trial No.	1	2	3
Time minutes	150	155	154
Brake HP.	31	28	18
Water A	24 lbs.	24 lbs.	24 lbs.
" B	180 "	180 "	180 "
" C evaporated	108 "	90 "	64 "
Temperature <i>t</i> at start . . .	54° Fahr.	58° Fahr.	52° Fahr.
" T at end	174° "	162° "	148° "
Heat-units expended, raising A+B+C to T	37,440	30,576	25,728
Heat-units expended in eva- porating C from T	107,244	90,090	64,704
Total heat-units	144,684	120,666	90,432
Mechanical equivalent foot-lbs.	111,696,048	93,054,152	69,813,504
HP. minutes	3,384·4	2,819·8	2,115·5
HP.	22·53	18·29	13·73
Per cent. of brake HP. . . .	72·6	65·0	76·27

When the loss by radiation of the frictionally developed heat is considered, the heat represented by the water raised in temperature and evaporated, a mean of 71·3 per cent., must be admitted to be a large proportion even if some deduction be made for slight loss of water by spray.

The Author has since made an experimental run with this brake. The figures obtained show that the evaporation takes place at a mean temperature of about 160° , and that of the mechanical work done a mean of about 72 per cent. is accounted for by the water heated and evaporated in the wheel-trough.

A form of brake used by Messrs. J. and H. McLaren, of Leeds, as devised by Mr. Druitt Halpin, M. Inst. C.E., is shown in Figs. 15 and 16. The engine upon which it was used gave off about 20 HP.

The brake-wheel is 5 feet in diameter, and 7 inches in width. The radius of the load circle is 32.19 inches. Water is constantly supplied to the trough, and constantly taken away by a scoop pipe. In 1886 Mr. Halpin proposed to use the Appold compensating lever, but has since abandoned it, as introducing an error varying with the coefficient of friction of the brake-blocks on the wheel.

As a brake which within a limited range of power might be considered automatic, and without the errors introduced by some of the forms of compensating action, the form shown at Fig. 17

may be suggested. In this the ends of the brake-strap $E E'$ instead of being connected are led over the rollers $R R$, which roll on a path $T T$, and are connected by links to a lever L pivoted and connected to the weight P . In the ordinary working of such a brake the lower blocks would touch the wheel with small pressure, and the weights $p p'$ would be adapted to the total friction required. The whole of the parts would be put in balance by the movable weight w . With variations in the turning moment or the lubricant P would rise and fall, but the rise and fall would be of short range as the separation or approach of the rollers $R R$ would rapidly change the extent of surface in contact.

FIG. 17.

Special reference should be made to a very simple form of friction-brake dynamometer, first proposed by Professor James Thomson, consisting of a cord or rope passed over the upper half-circumference once, or taking one complete turn round a smooth wheel, the one end carrying a weight P , and the other attached to a spring-balance in a manner similar to that shown in Fig. 15, the rope bearing directly upon the wheel without the intervention of blocks. This brake works exceedingly well for small powers, and there seems to be no reason against its use for large powers if a number of separate ropes be used. All the ropes might be attached to a cross-head at either end, from which the weight P would hang, and to which the spring-balance could be attached. The pull on

the spring-balance would be deducted from the weight P for the actual load carried, as in the case of Figs. 14 or 15. During the recent trials of gas- and steam-engines under the auspices of the Society of Arts, Professor Kennedy, M. Inst. C.E., used two hemp ropes, of about $1\frac{3}{4}$ inch circumference, passed once round the fly-wheel of the engine under trial, and these were found to be quite sufficient to absorb 20 HP., a water-trough wheel being employed and very little unguent.

PROPORTIONS AND DIMENSIONS OF BRAKES.

In English practice there has not been much diversity in the proportions given to brakes of these several forms for the absorption of a given power or the measurement of a given quantity of work; but, in the United States, a Prony brake has been used of dimensions supposed to have been sufficient to enable it to absorb about 500 HP. It was made with a wheel 5 feet in diameter, having a rim of trough-shaped section supplied with constantly renewed cooling water.

A comparison may now be made by an examination of the proportions of the brakes described, and by reference to the results of their working.

The brake used by several engineering firms is of the type Fig. 5, and of the size used by the Royal Agricultural Society at Newcastle in 1887, for a maximum of about 20 HP. at 130 revolutions per minute, and fitted with a compensating device, Figs. 5, 6, and 7. The wheel is 5 feet in diameter and 7 inches in width, and the radius R of the point of suspension of the load about 2 feet 9 inches, C being 23·75 lbs. With a load $P=270$ lbs., and the number of revolutions per minute $N=150$, this brake gives 21·17 HP. The lubricant used for the brake-blocks was tallow and water.

The similar brake used by Messrs. McLaren gave HP. = 19·1 with $N=148\cdot5$ and $R=2$ feet 9·38 inches; and another of similar dimensions, Fig. 12, but with water-cooled wheel, also used by Messrs. McLaren, gave 20·2 HP. with $P=271$ lbs., $N=145\cdot7$ and $R=2$ feet 8·19 inches; but it can be used for much higher powers. Tallow was the lubricant employed.

The water-cooled brake of Messrs. Garrett, for powers up to a maximum of 40 HP., has a rim of the section shown in Fig. 14, capable of taking friction blocks of about 10·5 inches in width, the width between the exterior flanges of the wheel being 11 inches. With this brake, $R=2\cdot8$ feet, maximum load $P=418$ lbs., and $C=17\cdot5$ lbs., HP.= 40 with $N=180$, Mr. Garrett finds that the

brake runs most smoothly at 180 revolutions per minute; but, as it is connected by universal joints to the engine to be tested, it must run at the speed of the engine. Taking $N = 150$ for this brake with the same load $P = 418$, the HP. = $33\cdot4$.

The Royal Agricultural Society has a brake with three wheels of the diameter and width of the single brake already mentioned, and this with $N = 150$ is intended to measure a maximum of 100 HP., or about $33\cdot3$ HP. per wheel.

As a convenient means of comparing the relative capacity of these brakes, judging by the amount of work for which they have been designed, or to which they have been put, a coefficient $K = \frac{W V'}{H P}$ may be employed, W being the width of the wheel in inches, and V' = the velocity of the periphery of the wheel in feet per minute. This gives for the

Royal Agricultural Society's single brake	$K = 824$
" " " treble "	$K = 495$
Garrett's water-cooled brake . .	$K = 740$
Ransomes', Balk's brake . . .	$K = 1,020$

or a mean of 860, omitting the Royal Agricultural Society's treble brake.

Compared in this same manner the Prony brake described by Professor R. H. Thurston,¹ as devised for measuring a maximum of 540 HP., the wheel being 5 feet in diameter and 2 feet wide, and $N = 100$, gives $K =$ only 75. This brake is stated to have been freely lubricated with beef tallow, plumbago and lard oil, and, although designed for a maximum of 540 HP., it does not appear to have worked above 200 HP., and at this power $K = 188$.

The Royal Agricultural Society's brake, the similar one of Messrs. McLaren, and that of Messrs. Garrett, are all fitted with compensating-levers. Without this arrangement there is no doubt that, although the brakes are not cooled with water, they would work with accuracy to higher powers, and would probably give satisfactory results with a constant $K = 850$, although, for long-continued runs, $K = 900$ to $K = 950$ would be better.

Even with the high coefficient $K = 1,020$ obtained from Messrs. Ransomes', Sims and Jefferies' brake with maximum load, the wheel heats sufficiently to make much lubrication necessary during long runs. This is shown by the variation in the scale weights with the same gross load.

It would appear that the water-cooled brake may be made to

¹ Journal of the Franklin Institute, April 1886. 3rd Series. Vol. xci. p. 290

work with a very much lower coefficient; probably a coefficient of 450 would be high enough for a water-cooled brake if not fitted with the compensating-levers.

The water-cooled brake described by Professor Thurston provides an exceptional case; but, although this brake was designed for 540 HP. as a maximum, it does not appear to have been employed for more than 174·5 HP., and it is questionable whether its performance at the proposed maximum would have been at all satisfactory, or even at half the maximum, especially when it is remembered that only two bands 3 inches wide were used to carry the wood blocks, which though 24 inches in length were only 2·5 inches in thickness. The flexure of the blocks, under the maximum tension of the belts, would have been sufficient to have made the pressure immediately under the belts excessive; while the pressure between the belts and at the ends of the blocks would be very small, and the effect of this on the wood and lubricant could not have conduced to good running. Even with the load carried, Professor Thurston remarks that: "the friction between the brake-blocks and the face of the pulley was reduced to a minimum by effective lubrication." This seems to show that the brake was not strong enough for the power measured, for it is hardly a desirable thing to have to reduce the friction to a minimum. Taking 180 HP. as the greatest for which this brake was used, then the coefficient $K = 209$.

The determination of the diameter and width of a brake-wheel for a given power is often affected by prior fixed circumstances and conditions, but an appeal to experience leads to the conclusion that a somewhat greater width of wheel than has commonly been used for the higher powers is advisable. Taking the Royal Agricultural Society's single-wheel brake, for width, $W = \frac{KHP}{C'N} = \frac{824 \times 20}{15 \cdot 7 \times 150} = 7$ inches, C' being the circumference of the wheel; but if the compensating-levers were not used, a rather larger power might be measured on this brake, and using the proposed constant for long runs, $K = 900$, then with a diameter of wheel remaining the same, *i.e.*, 5 feet, W would according to the above $= 7 \cdot 6$ inches. For such powers, however, it may be questioned whether a wheel of lesser diameter and greater width would not be preferable, although the pressure upon the wood blocks would have to be greater. For 20 HP. a wheel 3 feet 6 inches in diameter would require a width of 10·9 inches.

For a water-cooled brake the proportions adopted in this respect

by Messrs. Ransomes', Sims and Jefferies for their brake of Balk's design, or by Messrs. Garrett and Sons for their brake as described, would seem to be satisfactory.

For several purposes in connection with these brakes, it is necessary to consider the tension in the brake-strap or straps, or belt, and the pressure upon the wood-blocks.

For a given power the total frictional resistance F at the face of the wheel must be proportional to the velocity V' of the periphery of the wheel of diameter D in feet, or $F = \frac{HP \times 33,000}{D \pi N} = \frac{HP \times 33,000}{V'}$, and taking T , the maximum tension in the brake strap,

t = the minimum tension.

a = the fraction of the circumference of the wheel embraced by the brake-blocks,

f = the coefficient of friction, then following Rankine,¹

$F = T - t$ and the ratio $T : t$ is the number whose common logarithm is $2.729 fa$,

or
$$\frac{T}{t} = 10^{2.729 fa} = n.$$

This may be put

$$\text{Log } \frac{T}{t} = 2.729 fa = n,$$

and
$$T = F \left(1 + \frac{t}{T - t} \right), \text{ or } T = F \cdot \frac{n}{n - 1}.$$

In these brakes $a = 1$, and by way of illustration f may be taken as 0.2. Then, in the case of the large Prony brake already mentioned as described by Professor Thurston—

$$\text{Log } \frac{T}{t} = 2.729 fa = 0.5458,$$

and
$$n = 3.5.$$

For the maximum HP. for which the brake was calculated

$$F = \frac{540 \times 33,000}{1,570.8} = 11,345 \text{ lbs.}$$

$$T = 11,345 \times 1.4 = 15,833 \text{ lbs.}$$

$$t = \frac{11,345}{3.5} = 3,241 \text{ lbs.};$$

but taking the maximum HP. at which the brake was worked = 200 HP., $T = 5,902$ lbs. and $t = 1,204$ lbs.

¹ "Machinery and Millwork," p. 403.

If in the same way the tensions on the Royal Agricultural Society's single brake are taken as used at Newcastle last year with about 20 HP., $F = 270$ lbs., $T = 378$ lbs., and $t = 77$ lbs. If f is taken as 0.3, then with the same brake-load $T = 333.7$ lbs., and $t = 51.6$ lbs.

For a comparative approximation to the maximum pressure p per square foot of wood block upon the wheel surface, W' being the width of the wheel in feet, p may be taken

$$T + t = D p W', \text{ or } T = \frac{D p W'}{2},$$

and
$$p = \frac{T + t}{D W'}.$$

If this be allowed as giving figures which will permit the pressures on the wood blocks of the several brakes to be compared, or as an approximation to the pressure, then the greatest pressure per square foot of surface of Professor Thurston's Prony brake, if it had been used as proposed for 540 HP., would have been, assuming the coefficient $f = 0.2$,

$$p = \frac{15,883 \times 2}{5 \times 2} = 3,176 \text{ lbs.}$$

Assuming the blocks to be only four-fifths of the total area of wheel-face, then $p = 3,811$ lbs. At 200 HP., which is more than the actual power exerted, $p = 1,180$ lbs.

In the same way, and with the same coefficient, the greatest pressure per square foot on the blocks of the Royal Agricultural Society's brake at Newcastle would be $p = 260$ lbs. with the blocks close, or 312 lbs. with the blocks covering 0.8 of the surface of the wheel.

With the brake used by Messrs. McLaren, it would be the same if the same coefficient obtained.

With Messrs. Garretts' brake, $p = 269$ lbs., or with the blocks covering 0.8 of the surface $p = 323$ lbs.

With Messrs. Ransomes', Sims and Jefferies' brake, the wheel of which is 6 feet in diameter and 1 foot wide, and where the wood blocks are close together, Fig. 11, $p = 110$ lbs.

This is only intended as a rough approximation to the greatest pressure per square foot of rubbing-surface of wood, but it affords a means of comparison of the pressure with the different brakes. The mean pressures are probably somewhat less than half those given. From these figures the advantages of width of wheel will be seen. The pressure for a given load will, of course, be less for

a given power absorption with increase in the diameter of the wheel. The limit of velocity has, however, here to be considered, especially with the wheels not cooled with water by the aid of an internal trough. What that limit is cannot now be said, but it is noteworthy that Messrs. Garrett find their brake runs more smoothly at 180 revolutions per minute than at any lower speed, presumably for the same power.

THE APPOLD FRICTION-BRAKE DYNAMOMETER.

It is necessary now to consider the effect of the introduction of the Appold compensating-lever, at least to do so sufficiently to find whether it is a desirable feature or not.

In illustration of this point, it may be conceded that, if the friction on the one hand be conceived to be almost infinitely small, the load P would have to be infinitely great, and the reaction at X would also be infinite. If, on the other hand, the frictional grip be increased so as to approach adhesion, then with any load P that could be put upon the brake, while it still acted as a dynamometer, the reaction at the pivot at the end X of the compensating lever may be infinitely small, supposing the lever to come into play. It may therefore be seen that the reaction at X for any given brake and given power, may vary with the frictional grip, as affected by lubrication, from zero to any maximum. The greater the difference between what may be called the capacity of the brake, and the power of the motor, the less will a given reduction in the friction affect the compensating-lever, because, with the belt ends working almost slack in their connections, the greater may be the increase in the tension in the belt, before it becomes sufficient to make the reaction at X a quantity sufficiently appreciable to affect the indications.

It may, however, be easily seen that under the ordinary working conditions, that is to say, with the size of engine usually tested on the brakes, such as those used by the Royal Agricultural Society, the effect of the lever on the accuracy of the registration may be very small.

If, for instance, the lever be of the proportions shown in Fig. 8 (p. 9), and the tension on the two ends A of the strap be assumed to be 100, then the reaction at X will be equal to the difference of the reactions at A and B , or about 11.

This will act in helping to lift the weight, its effective help in this direction being proportional to the relation between the distance from the centre of the wheel to the point X , and the distance from the centre of the wheel to the point of suspension

of the weight, and, if this relation be as 1 to 2, then the reaction tending to raise the weight will be 5·5, and may be, with the ordinary relations between the weight P and the tensions in the ends of the brake-strap, an entirely negligible quantity, and of practically no numerical effect on the HP. indicated by the brake. If, however, a brake is used in such a way that the coefficient of friction is very small, and anything like the above assumed tension in the strap-ends be allowed to arise, then the error may become an important quantity.

With the brake, for instance, of the type shown by Fig. 5, as used by the Royal Agricultural Society at Newcastle, with an engine exerting 20 HP., and a weight P of about 270 lbs., if the tension at the ends of the brake-strap were as much as 100 lbs., and the reaction as much as 11 lbs., reducing the effective weight to 264·5 lbs., the difference, 5·5, is 2 per cent. of the whole. Neglecting the effect of the compensating lever, and taking the gross load on the brake as 270 lbs., the circumference of the load-circle being 17·25 feet, and the revolutions 135 per minute, the engine would apparently exert 19·05 HP. If, however, the tensions at the ends of the strap, where connected to the lever, be, to take an extreme example, as much as 100 lbs., and assuming the reaction to be as above, 11 lbs., and the gross load thus reduced by 5·5 lbs., or to 264·5 lbs., then the engine would have exerted only 18·66 HP., the difference being 0·39 HP. This with an engine using 30 lbs. of steam per brake HP. per hour is equal to 11·7 lbs. of feed-water per hour, or, with a boiler evaporating 10 lbs. of water per 1 lb. of coal, it represents 1·17 lb. of coal per hour, or 3·8 lbs. in a run of four hours of an engine exerting 19 brake HP.

Under the same circumstances and assumptions, but with the engine running at 150 revolutions, the HP. would be 21·16, neglecting the effect of the lever, and 20·73 when correction is made for it, making a difference of 0·43 HP.

If, however, the tension of the brake-strap be taken, as calculated by the expression given on p. 22, and an assumed coefficient $f = 0·2$, there is obtained by approximation an error of 0·34 HP. instead of 0·43 HP. In the same way, for $f = 0·3$ the result is an error of 0·23 HP., *i.e.*, 20·83 HP. instead of 21·16 HP.

It is, however, an objection that these calculations proceed on the assumption that the relation between the gross load P, and the tension T in the belt, remains the same for the velocity concerned, *i.e.*, 2,355 feet per minute at the periphery of the wheel, as for the low velocity at which the brake would just slip, when tightened up on the fixed wheel.

If the belt of blocks is set so that it will move at a low velocity under the influence of a weight suspended, as in Fig. 2, when the wheel is fixed, the tension in the belt and the load that will overcome the friction of rest being thus ascertained, a coefficient of friction for that speed may be at once deduced. It would, however, be useless to apply this coefficient to any calculations regarding the brake when at work under ordinary conditions, inasmuch as the coefficient so found will not apply at the high velocity of wheel-periphery commonly to be dealt with. This is shown by the facts given on p. 11. The tension in the belt, as controlled by the adjusting-screw S, must be much greater to support a given weight when the wheel is running than when the wheel is at rest. The velocity very materially affects the friction, and no demonstration of the dynamics of the compensating-lever of the Appold brake is possible without taking these facts into consideration. To be able to do this, however, experimental inquiry is necessary.

Some useful information on this point was, however, obtained by Messrs. J. and H. McLaren, who attached a spring-balance to the upper end X of the compensating-lever of the brake, which they made of the same dimensions as that of the Royal Agricultural Society. By this means they ascertained the variation of the reaction at X with variation of velocity of wheel-periphery, and of lubrication of the wood-blocks. They found that the pull on the spring-balance increased as the speed of the engine fell, when the steam-pressure decreased during the clinking of the grate. This is precisely what would be expected, from what has already been said as to the greater tension in the brake-strap with a very low velocity, and it is clear that with the falling speed of the engine the weight P descended a little and increased the tension in the brake-strap, and therefore the reaction at X. Their experiments conclusively show the direct relation between tension in the brake-strap and the lubrication. In some cases, when using tallow, they got 24 lbs. pull on the spring-balance at X, and when using water the pull was 258 lbs. When they got the 24 lbs. pull there was a considerable quantity of water getting in between the brake-blocks and the wheel, but in some of the tests when, to use their own words, they "fairly smothered the brake in tallow," so that the water could not get in between the blocks and the wheel-face, they measured 20 HP. with only 4 to 5 lbs. pull on the spring-balance. Their experiments lead them to conclude that, with an internally cooled brake-wheel of the size employed, compensating-levers could be used without error up to 20 HP.

So long as the Appold brake, like that of the Royal Agricultural Society, is not used for more than 15 HP., and is sufficiently, but still sparingly, lubricated with tallow or suet, the friction between the wood and iron is such that the weight of the brake-strap and blocks with the suspended load is sufficient, at the ordinary speeds of the engines tested, to carry the load without screwing up the belt so that there is more than a few lbs. tension at the compensating-lever. Under such conditions the lever does not affect results, and adjustment of the frictional grip and position at which the load is carried has to be made by the hand-screw S. The conditions are the same as, or very similar to,

FIG. 18.

those which would obtain if the brake were as shown by Fig. 2, that is to say, without compensating-lever, but with a belt so slack that the bottom blocks barely touch the wheel.

The following investigation on this subject,¹ which has been kindly placed at the Author's disposal by the Consulting Engineers of the Royal Agricultural Society of England, supports the views herein expressed:—

Let W = load on brake-strap (Fig. 18);

T_1 T_4 = tensions just above and below the points of suspension of W ;

¹ The Journal of the Royal Agricultural Society of England. October 1888. pp. 672-78.

$T_2 T_3$ = tensions at two ends, C and D, of strap connected to lower ends of compensating levers;

P = pull on upper ends of these levers;

a, a' = radii of brake-strap and wheel respectively;

O D = d , C D = b , and O E = c ;

F_1 = friction of A B C,

F_2 = friction of A D, and F = total friction of brake-strap.

The portion A B C of the brake-strap is kept in equilibrium by the tensions T_1 and T_2 at its ends, the friction exerted on it by the wheel and by its own weight. This gives, taking moments about O:—

$$T_1 a = T_2 O M + F_1 a' + \text{moment of weight of A B C.}$$

Considering the portion A D in the same way—

$$T_3 O N = T_4 a + F_2 a' - \text{moment of weight of A D.}$$

Adding these two equations, then:—

$$T_1 a + T_3 O N = T_2 O M + T_4 a + (F_1 + F_2) a' + \text{difference of moments of weight of A B C and A D.}$$

But the moment of weight of A B C = moment of weight of A D because the strap is accurately balanced.

$$\text{Therefore } (T_1 - T_4) a = T_2 O M - T_3 O N + F a'.$$

$$\text{Now } O M = O C \sin O C M = (d - b) \sin O C M,$$

$$\text{and } O N = O D \sin O D N = d \sin O D N,$$

$$\text{and } T_1 - T_4 = W.$$

Substituting these values in the above equation:—

$$W a = T_2 (d - b) \sin O C M - T_3 d \sin O D N + F a';$$

$$= d \{T_2 \sin O C M - T_3 \sin O D N\} - b T_2 \sin O C M + F a'.$$

But because the lever is in equilibrium, considering the forces at right-angles to it:—

$$T_2 \sin O C M = P + T_3 \sin O D N,$$

$$\text{or } P = T_2 \sin O C M - T_3 \sin O D N;$$

and by taking moments about D of all the forces acting on the levers,

$$D E \times P = C D \sin O C M T_2$$

$$\text{or } (d - c) P = b T_2 \sin O C M.$$

Substituting these two relations in the above equation, then:—

$$W a = d P - (d - c) P + F a';$$

$$W a = c P + F a';$$

$$\text{or } W - \frac{c}{a} P = \frac{a'}{a} F.$$

that is to say, in order to get the effective value of W , the pull at E , diminished in the ratio of c to a , must be deducted.

This result agrees with the principle that the external forces acting on the system should balance. Since it is at rest, and the tensions are internal forces, the load, the pull at E , and the friction are the only external forces that have to be considered, as the point of support is in the centre of gravity of the strap. This gives at once, by taking moments about O :—

$$a W = c P + F a' ;$$

as before. Also because

$$P = \frac{b T_2 \sin O C M}{d - c},$$

it follows that P depends upon the tension of the strap, and upon the proportion which $C D$ bears to the whole length of the lever. Now the tension of the strap depends upon the lubrication; hence, the more efficient that is, the greater will be the inaccuracy of the brake.

It is evident that, in order to determine the probable amount of error in the Cardiff and Newcastle trials, the pull upon the upper ends of the levers must be ascertained when the brakes are running under exactly the same conditions as to power, speed, temperature of air and lubrication. When the lubrication is constant, the pull on the upper ends of the levers always bears a constant ratio to the load on the brake.

The following Table gives the results obtained by Messrs. McLaren, by means of the water-cooled brake, and by the Engineers to the Royal Agricultural Society with the same engine at Newcastle.

TABLE III.

	Messrs. McLaren's Trial.		R. A. S. Trial at Newcastle.
	Halpin's Brake.	Brake made like R. A. S.	
Indicated HP.	23·70	22·20	24·020
Brake HP.	20·20	19·10	20·770
Coal per brake HP. per hour	2·11	2·14	2·267
Feed-water per brake HP. per hour . .	22·10	22·00	21·530
Mechanical efficiency.	0·85	0·86	0·860

It will be seen that the figures agree very closely.

The Author has not here described more of the many forms of friction-brake dynamometers than are necessary to comprise most of those of a typical character. Reference may be made to an essay,¹ by Professor E. Brauer, on various other forms, including numerous devices for arriving at a compensating action, none of which, however, appear to be free from the objections herein referred to.

It may be suggested that a very useful dynamometer might be obtained by a combination of absorption and transmission dynamometers.

The Paper is accompanied by three sheets of drawings, from which the Figs. in the text have been reproduced.

¹ Zeitschrift des Vereines deutscher Ingenieure. Band xxxii. Seite 56 1888; and Minutes of Proceedings Inst. C.E., vol. lxxx. p. 266.