"FRICTION OF FABRIC MATERIAL AND BEARING METALS"

(a) Investigation of Friction and Wear of Fabric Linings.

(b) Investigation of Lubricants and Friction Bearing Metals.

by

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INVESTIGATION OF FRICTION AND WEAR

OF

FABRIC LININGS.

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ABSTRACT

This thesis discusses the results obtained by experiments carried out on the friction of 'Fabric' material. It presents a general analysis of the suitability of cotton, asbestos, and metals as friction materials, and the combination of these materials in bonded form as brake and clutch linings. The tests to be described were carried out in order to determine the coefficient of friction, wear properties, and general behaviour of a few of the many fabric materials under varying conditions of load, rubbing speed, and temperature.

Special study has been given to the variation of frictional resistance at speeds varying from 0.5 to 50 feet per minute. It has been shown, that with a cotton bonded fabric the frictional resistance is very high at "crawl speeds", and gradually diminishes as the speed increases. The effect of temperature on the wear of fabric material and on the destruction of the binder used has also been studied. Throughout the experiments it has been noted that a thin film of iron oxide changes the value of the coefficient of friction almost as much as a film of oil.

INTRODUCTION

The desire to have a brake or clutch lining which would give a higher coefficient of friction, a greater dissipation of heat without burning or sparking, and as great if not greater durability than cast iron or steel, has been the incentive towards the manufacture of fabric materials. Especially has the development of electric trains and motor cars produced a great demand for dependable fabric linings. Although such linings have been shown in practice to possess economic and mechanical advantages, little has been published concerning their fundamental properties.

The increasing need for verification by actual experiment of the results claimed by various makers has prompted this research work. The different fabrics tested are given by their trade names - (i) 'Ferodo Fibre' which is a woven impregnated cotton; (ii) 'Ferobestos' is a woven impregnated asbestos; (iii) 'Ferodo Bonded Asbestos', (iv) 'Breko', (v) Checko, (vi) Raybestos which are made mainly of bonded asbestos woven on to a copper, brass, or white-metal wire reinforcement, and impregnated with a binding material; (vii) Ioco, (viii) Fabroil, which are made by hydraulically pressing together impregnated layers of woven cotton or duck. In addition to these eight materials which form the main test materials to be examined, comparative tests were made on bronze, wood, leather and 'gatex' (special belting material).

A microscopic examination of the material showed that item (iii) contained probably up to 30 per cent of cotton; item (iv) was almost entirely pure asbestos woven with brass wire reinforcement. The amount of wire is a varying quantity not only in the different fabrics, but also on the successive layers of the same fabric, (see figure 2a).

The items (vii) and (viii) are fabricated material used for wheel teeth and friction discs, manufactured to take the place of rawhide and compressed paper. These materials are used for bearing bushes where water is used as the lubricant.

The surface against which all these materials were run was cast iron. This cast iron surface was maintained at constant temperature either by a supply of cooling water, or by heat from gas burners.

PRELIMINARY TESTS

In order to recognize the main differences between the asbestos, cotton, and cotton-asbestos lining material preliminary microscopic examination was made. Examples of Canadian and Russian mined asbestos were examined. Fig. 1 (a) shows the material as mined with its long straight slippery fibres. The micro-photographs, Fig. 1 (b) and Fig. 1 (c) show respectively the cotton and asbestos partly finished ready for spinning into threads for weaving. Fig. 2 illustrates brake linings, one of pure cotton which is readily recognized by its twisted hard short fibre, and the other a 30% cotton-asbestos woven material.

(a) To compare pure asbestos with cotton (Fig. 1 (b) and Fig. 1 (c)) as a braking material, a pad of asbestos was used as a brake to stop a small heavy rotating fly-wheel. The pressure was applied to the pad by springs of known strength. A similar pad of cotton was substituted for the asbestos and further tests carried out under the same pressure condition. In all the experiments the cotton pad took a more sudden grip, causing vibrations or jerky motion in the slowing up of the flywheel. It was also found that the cotton surface became glazed. A rubbing wear test was carried out on the same materials, and it was observed that, with temperatures of rim about 180° F. the wear on the cotton pad piece became very rapid. The wear of the asbestos pad was inappreciable. These preliminary experiments showed that cotton as a friction-fabric material might possess unsuitable qualities.

(b) Comparison was made of the different binding material used. Samples of the fabrics (i.e. viii) were

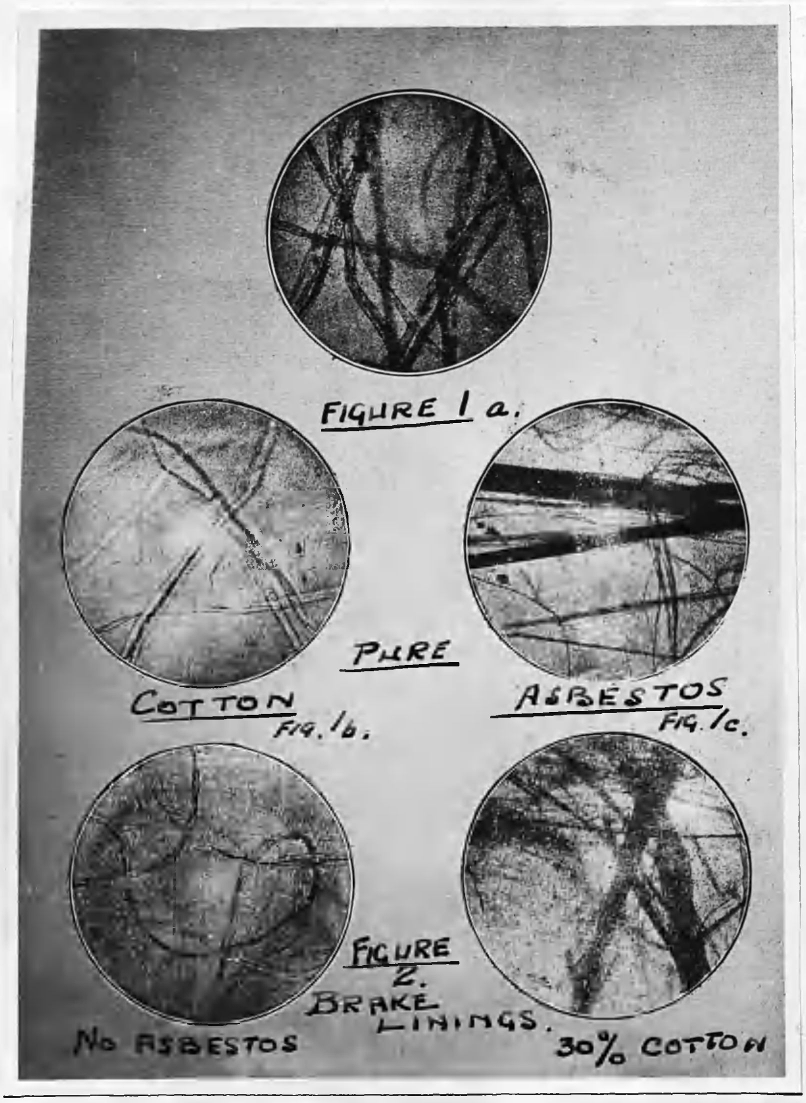

FIGURE 1 a.

PURE

COTTON Fig. 1b.

ASBESTOS Fig. 1c.

FIGURE 2. BRAKE LININGS.

No ASBESTOS

30% COTTON

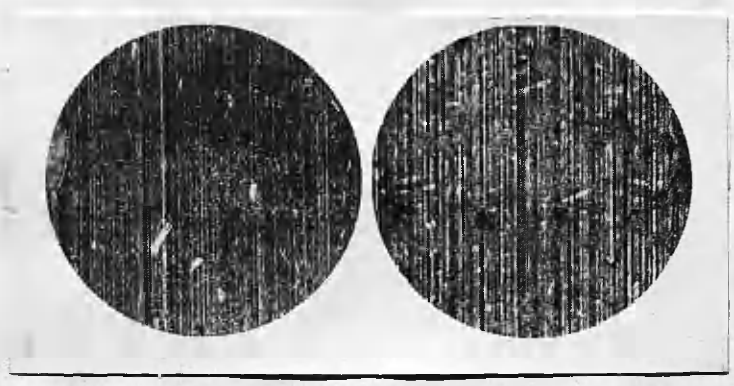

FIGURE 2a.

SHOWING SCORED SURFACES OF BRAKE LINING

placed in a vessel which was supplied with steam at a pressure of 80 to 100 lb./in^2. The steam did not burn the cotton fabrics, but after a short period of its application the binding material was destroyed. The samples of fabrics were also placed in cold water, and in hot water with rust in suspension. The results showed that the synthetic resin used had formed a good weather-proofing and rust-resisting surface. Similar tests with sulphuric and nitric acid showed that only the latter had any effect on the surface of the brake linings.

(c) Carbonization tests were carried out in a small electric furnace in order to test how much combustible material each fabric contained. The weight at 100°, 200°, 300° and 400° F. was noted. At 500° F. the impregnation material became plastic, and failed as a binding substance at 600° F. The pure asbestos brake linings showed no change of weight at 400° F., but the weight of others varied from 7 to 30 per cent. of initial weight, the inflammable material being cotton used in the weaving process.

(d) The tensile stress of the various friction fabrics varied from 3500 to 5000 lb. per in^2; the compressive pressure with measurable reduction in thickness was 1,000 to 2,000 lb. per in^2; the weight per cubic inch was 30 grammes for material woven on brass wire, and for material without metal 15 to 20 grammes. At 400° F., and with a pressure of only 300 lb. per sq. in., carbonization took place and the lining charred.

(e) Laboratory tests using flat smooth cast surface (horizontal and inclined) also specially designed apparatus to give constant pull are described later, Fig. 7. (p.12a.)

(f) Abrasion tests were carried out with the several brake linings; it was found that the grinding could only be carried out successfully on the fabrics which were woven on brass wire.

More complete tests were carried out on a drop stamp, where a weight of one ton was lifted through 15 feet at a speed of 1000 ft. per min., by means of a ferodo belt pressing on the rim of a friction wheel with an average pressure of 100 lb. per sq. in. These tests lasted only for 20 to 30 minutes. In this time the friction wheel which was air-cooled, changed its temperature from 80° (Smithy temperature) to 520° F. When the work done exceeded 50,000 ft. lb. per minute per sq. in. of ferodo excessive wear took place. The author was also able to make a comparison between a one ton Massey stamp using (a) water-cooled drum with wood blocks as brake material, and (b) fabric lining with air-cooling only. Experiments were carried out at the works of the Gartsherrie Engineering Co. For continuous working the brake linings containing cotton were most unsatisfactory and the drop stamp had to stand until the brake cooled. From measurements of the wear during these tests it was found that taking cast iron as the standard, the following wear ratios were obtained:- cast iron on cast iron 1/1; asbestos on cast iron 12/1; cotton on cast iron 26/1.

TESTING MACHINE

A machine was designed on which three series of tests could be carried out; the first of the series covering a range of speed from 0.26 to 2000 feet per minute; special attention being given to the speeds between 0.6 and 55 ft. per minute ("crawl speeds"). The amount of drag on the fabric material was measured.

In the second series the same speeds were employed but the temperature was to be varied from 55° to $500^\circ F$. This could be done either by a variation of the applied pressure between the fabric and the wheel, or by the application of external heating. The temperature of the rubbing surface was measured by a thermo element in which a small copper wire is embedded. The performance is a measure of $P_t = \mu p F_b$. $\mu =$ coefficient of friction, $p =$ radial pressure on contact surface, $F_b =$ brake covering or rubbing surface; the average slip velocity is measured. The third series is a wear or fatigue test, measurements for which may be in thousandths of an inch of wear, or grammes weight loss of material.

The machine, on which the first series of tests was carried out, is illustrated in Figs. 3 and 4, also Figs. 5(a) and 5(b). This machine has been fully described in a previous work(9) and only the main features are mentioned here. Referring to these figures, D is a cast iron wheel driven at a uniform speed by an electric motor, the drive being taken by an arrangement of belt drives, an epicyclic gear, and a specially geared lathe running headstock. Pressing on the rim of the brake-wheel is a block E which carries four pads of friction fabric material, F, and the variable load, W. This pad holder floats on the rim of the wheel and is connected by means of links to bell cranks, G, which are fitted with well lubricated ball bearings. The other ends of the bell- crank levers are connected to a dashpot H, and a spring balance, S.

The object of the dashpot was to damp out any vibration which might be set up, and also to balance the linkages attached to the spring balance. The position of the fabric friction material on the brake wheel could readily be adjusted at each change of conditions. It will be seen that with this arrangement the pull exerted by the friction material/

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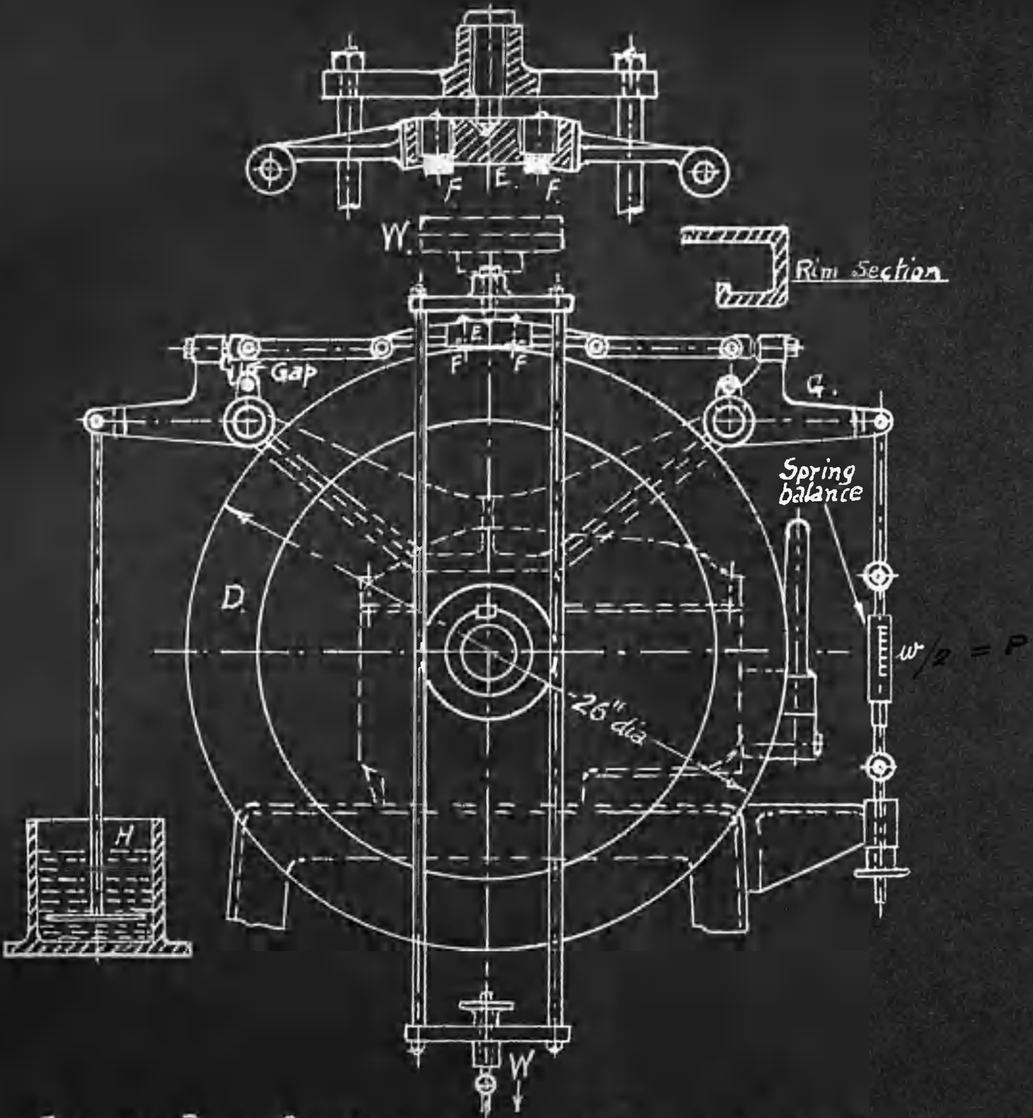


FIGURE 3.— GENERAL ARRANGEMENT OF LEVER SYSTEM AND DETAIL OF FRICTION PAD.

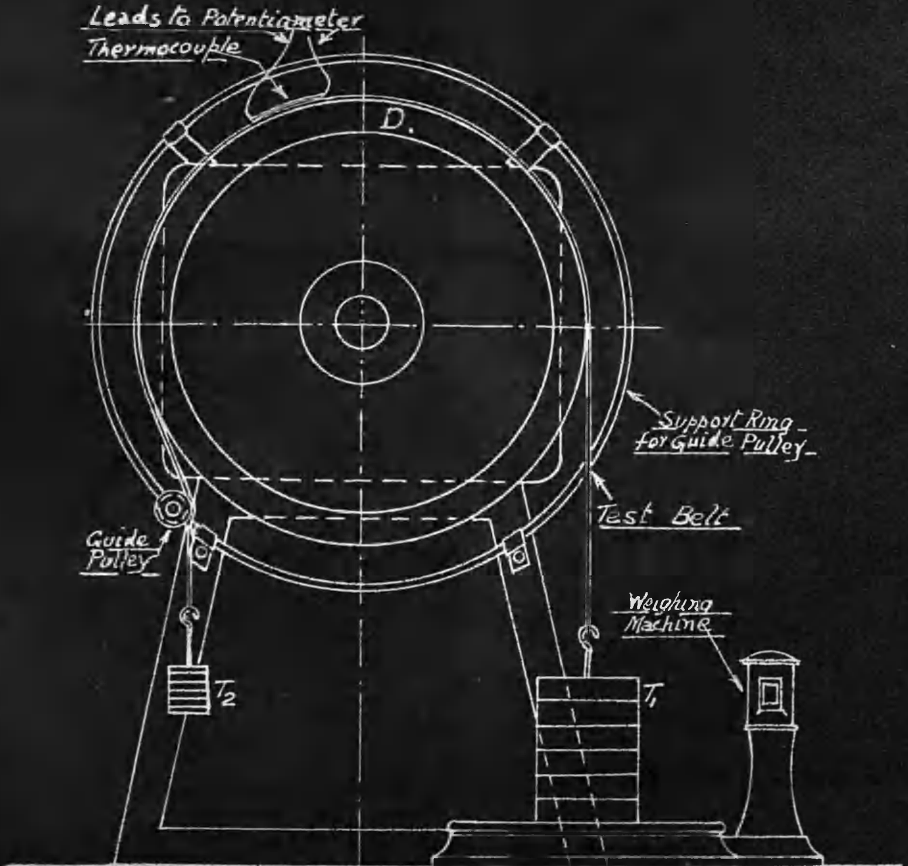


FIGURE 4— ARRANGEMENT FOR BELTS AND COTTON DUCK TESTING

material, when the wheel is rotating against it, is transmitted to the spring balance; (a weighing machine was substituted for the spring balance in the lubricated material tests.). The arms of the bell-cranks, G, are of unequal lengths, the leverage being 2 to 1, making the pull on the spring balance $P = {}^{W}/_{2}$ lb.

The advantages claimed for the machine as designed are direct application of load with no twisting action; direct readings of pull with no tilting action; easy adjustment of loading and change of speed. This machine can readily be adapted to carry out static friction tests. The brake wheel can be fixed to keep it from turning and the friction material allowed to slip on the rim. The wear which takes place in the material can readily be measured even with the machine running during endurance tests. Either a supply of cooling water was applied to the underside if the rim of the brake wheel, or a supply of heat from bunsen burners applied to the channel section rim, Fig.3, page 11.

A diagramatic plan view of the drive of the machine is shown in Fig. 5(b). The main shaft, M.S., is driven by the motor which runs at approximately constant speed. The drive is taken by either of two belts (1 and 2) to the counter shaft, C.S. The pulley of belt, 2, contains an epicyclic gear which may be used either direct, or with 100 to 1 reduction. Thus in this gear three speeds may be obtained, represented by 1, 2, and e. The drive from the countershaft to the gear box is taken by a three speed cone pulley, giving a further three speed, a, b, and c. Finally, the turret-head-lathe gear box gives a further three/

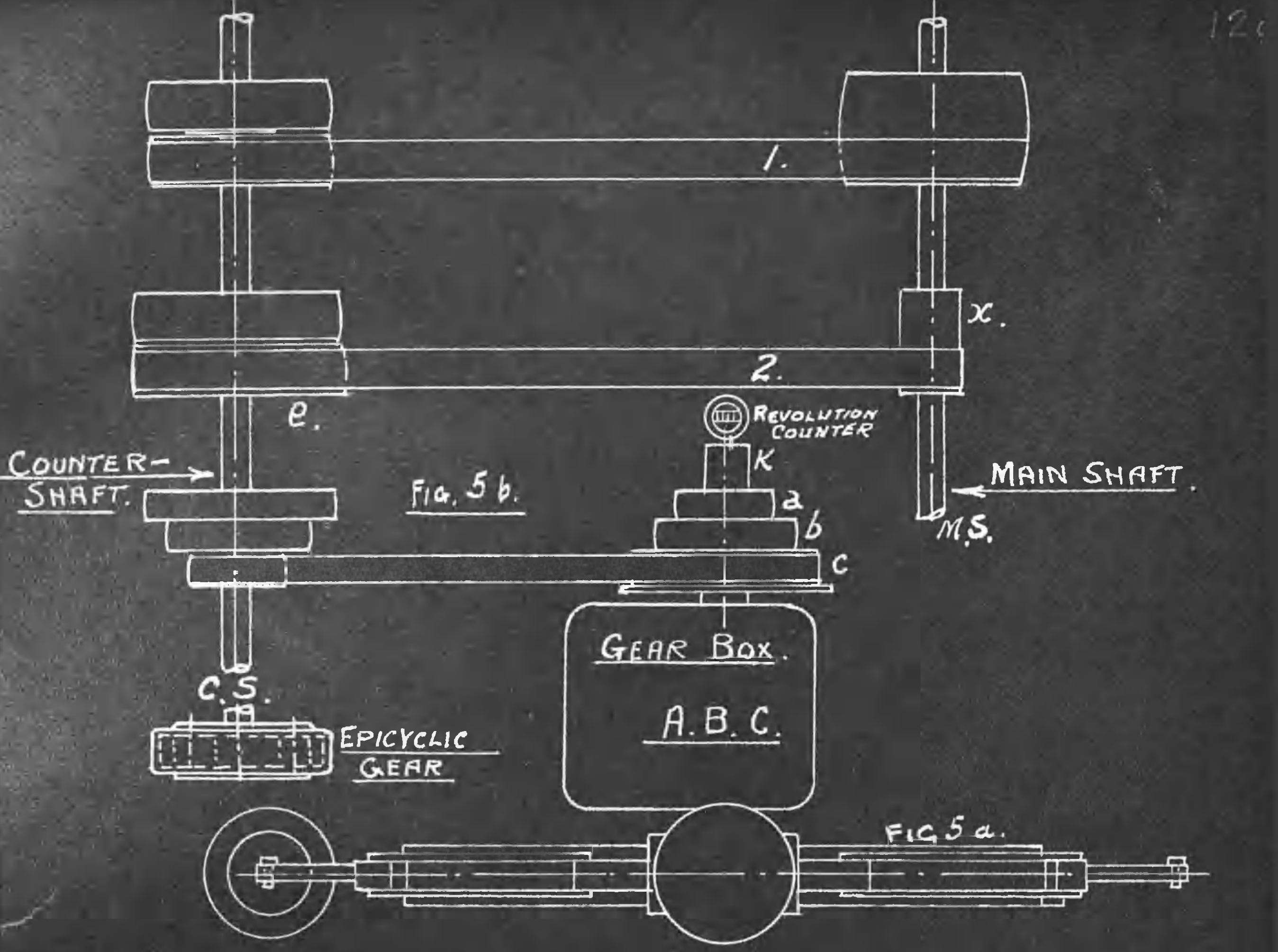


FIGURE 5 — PLAN VIEW OF BELT DRIVES AND GEARS

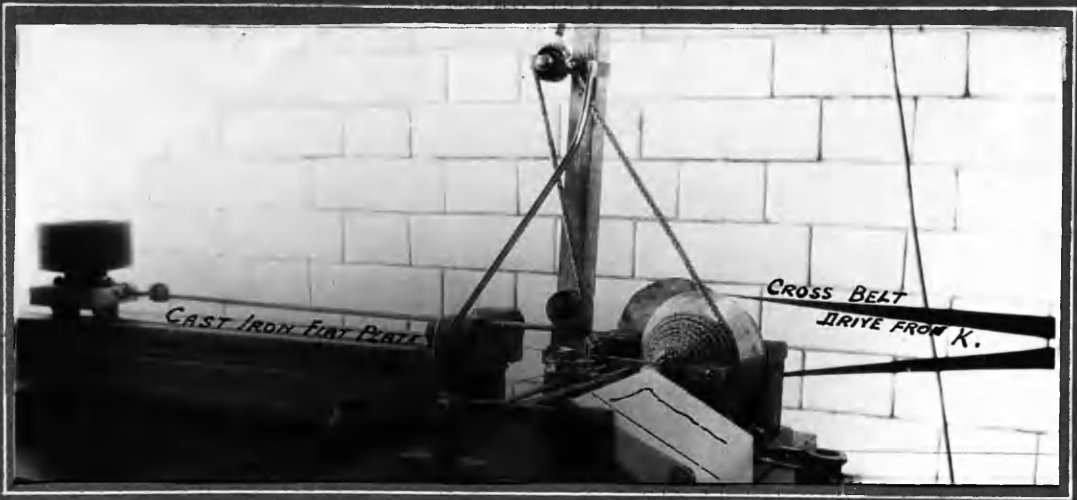
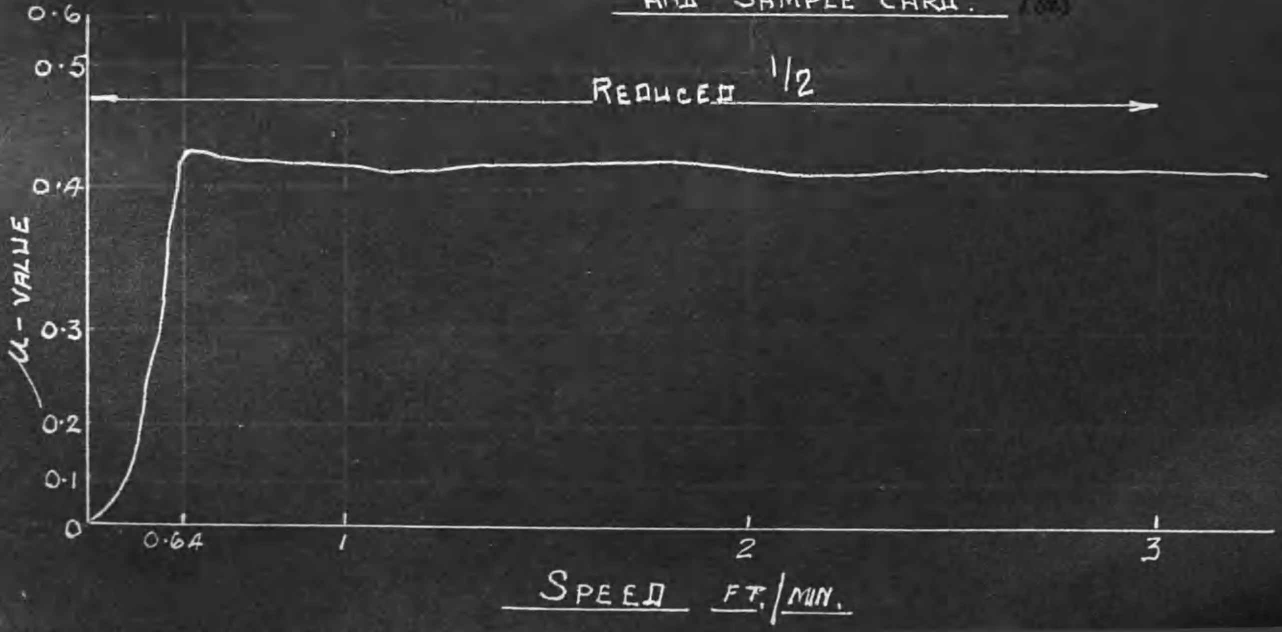


FIGURE 7 — AUXILIARY TESTING MACHINE, AND SAMPLE CARD. 7(a)



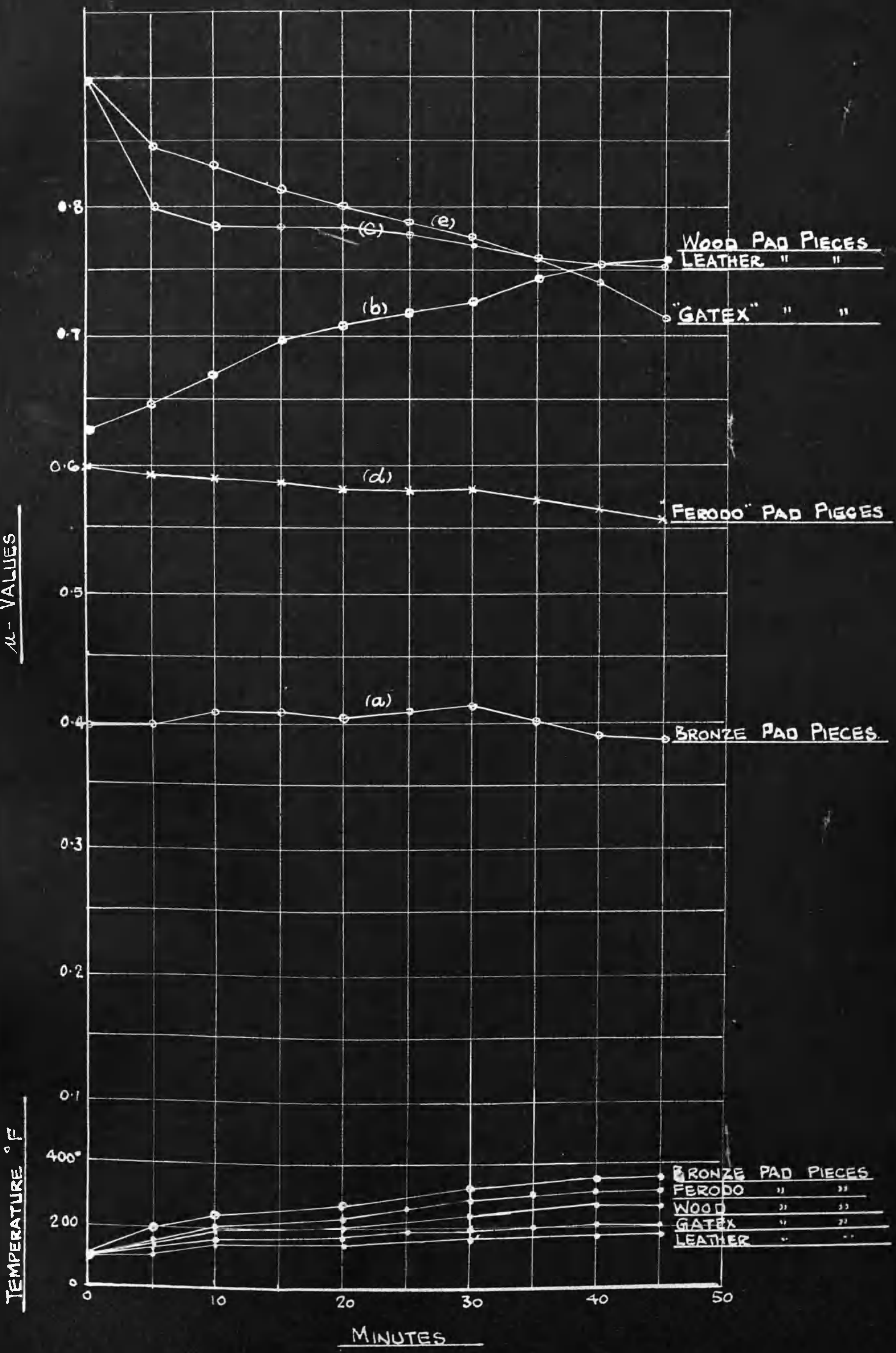
three speed variation, A, B, and C., so that by taking all the possible arrangements of these drives 27 speeds may be obtained. Gaps in the speeds may be bridged by changing the speed of the motor by adjustment of the brushes.

On the 26 inch diameter brake wheel the minimum rim speed was 0·26 ft. per min., and the maximum nearly 2000 ft. per min. To avoid changing the speed of the motor, pulley, x, was replaced by one four times the size, and in this way a very complete and close range of speeds could be obtained, especially at the lower speeds.

Temperatures at the interface were measured by calibrated iron-eureka thermocouples. The wire was not allowed to rub on the surface of the wheel D, but two wires passed into the fabric friction material very closely to the outer surface, and two touched the inner surface. The wires were insulated from the cast iron brake block by porcelain tubes. Leads to the potentiometer were so arranged as to cause no difficulty with friction test.

TEST PROCEDURE

Static Friction and Low Speed Tests - Before any running tests were carried out, an attempt was made to determine the static friction of the different fabrics. The surface of the brake rim was washed with petroleum ether. The pads of brake material had been previously fitted to the curvature of the wheel rim. A long run was then made, heavy loads were applied by means of a central spring balance attachment, the temperature rose gradually to 550°F. The comparative values of the coefficient of friction, with the different materials during this preliminary running, are shown in the upper part of Fig.6., page 14, plotted on a time base/



GRAPHS OF μ-VALUES, ALSO TEMPERATURE RISE ON TIME BASE

FIGURE 6.

base, and shown as a matter of interest.

When a good rubbing surface was obtained, i.e. the friction fabric embedded on the wheel, the wheel was locked so that no motion could take place. The balance and dashpot rods were removed leaving only the block E with links and bell-cranks. Varying loads, W, were placed on the block E, and known weights were hung on the end of the right-hand bell-crank until the block began to move. The largest weight, P, which could be applied to the bell-crank and cause uniform motion of the block, E, was taken as the static pull, Effective pull, w = 2P This was verified by applying weights to the left-hand bell-crank until motion took place in the opposite direction. The difference in the value of the weights on the right and left hand levers was taken as the pull and the average of a number of such readings was taken as the final reading.

<u>Tables for Static Friction Tests.</u> (Table 1).

II. 'Ferodo' Bonded Asbestos

W lb./in^2	w lb.	
7	3·95	0·31
14	5·2	0·37
28	9·2	0·33
42	14·8	0·35
56	20·8	0·37
66·2	22·4	0·32
76·4	24·4	0·32
86.6	26.8	0.31

I. 'Ferodo' Fibre

W lb./in^2	w lb.	
7	4·0	0·57
12·1	6·8	0·56
17·2	9·9	0·57
22·3	13·2	0·57
27·4	15·0	0·55
32·5	18·2	0·55
35·4	23·2	0·66
44.6	29.0	0.65

IV. 'Breko'

7	2·6	0·37
12·1	4·2	0·35
17·2	6·2	0·34
22·3	7·6	0·34
27·4	9·2	0·34
32·5	11·2	0·345
35·4	10·4	0·31
44·6	12·8	0·29

VI. Raybestos

7	3·0	0·43
12·1	4·4	0·36
17·2	6·8	0·40
22·3	8·4	0·38
27·4	10·4	0·38
32·5	12·0	0·37
35·4	10·0	0·29
44·6	12·8	0·29

These/

These experimental results have been carried out under the usual very difficult means of the investigator having to tell when uniform motion begins and whether it is continuous.

Further static friction tests were made using a long cast iron runway, the surface of which was much the same as the surface of the brake wheel. This gave a much longer distance to travel, but the same difficulties were inherent. As a check on previous results this test proved quite satisfactory, showing that the two cast iron surfaces were very similar.

Auxiliary Testing Machine - In order to make certain that the coefficient of friction at very slow speeds was correct, a small variable gear machine was put together, and this was used with an automatic recording arrangement. This auxiliary machine was driven from the main shaft of the friction machine, and could be made to give continuously varying rubbing velocity.

Fig. 7 shows a conical grooved drum driven at a constant speed by a narrow belt from the end of the gear-lathe headstock. The drum shaft has been extended to form a roller over which a strip of paper travels. The pull cord passes over a guide pulley mounted on a swivel bearing suspended from a spring. The pull on the trolley cord gives a deflection of the pulley varying in accordance with the magnitude of the pull on the friction fabric. These spring deflections are magnified up by means of a lever arrangement, and are transmitted /

transmitted by a cord and a counterweight to a carriage which carries a pencil.

Since the conical drum rotates at constant speed, the cord is pulled in at a continuously increasing speed, and the magnitude of the pull will be recorded on the strip of moving paper. The base of the curve traced will be in terms of the speed, and a suitable calibration of the ordinates in terms of the coefficient of friction of the material under test.

The speeds used were 0·64 to 3·2 ft. per min.
2·6 to 13 ft. per min.
12·0 to 60 ft. per min.
with loads varying from 16 to 32 lb.

This preliminary speed test confirms the values of those derived from the Friction Testing Machine and a sample curve obtained is shown, Fig 7(a), p 12(a).

As a final preliminary test the Friction Machine was made to run at a constant speed with varying loads. This was done by placing a variable resistance in series with the motor windings. The selected speed was 27·2 ft. per min. and loads on the friction fabric were varied from 7 lb. per sq.in. to almost the destructive crushing pressure. It is found that Ferodo Fibre did not conform to the law $\frac{F}{R} = \mu$, but gave results which when plotted gave a law more nearly $F = aN \pm b$ N = Normal load, 7lb. minimum
F = friction force, lb.

$$= 0{\cdot}348N - 0{\cdot}75 \text{ lb.}$$

When the point at which F is not altered by load, is reached, the coefficient has changed from 0·329 to 0·29. The Ferodo's appear to be the more homogeneous of all the friction materials under test, but they also show irregularities in structure, which show themselves during long running tests. According to Loney the friction increases at a greater rate than the normal pressure up to a certain point.

RUNNING TESTS

RUNNING TESTS.

Calibration of Machine. - In all friction machines it is necessary to have uniformity of conditions when taking readings. A zero reading or machine multiplier must be predetermined. The machine is fitted with an adjusting screw in the form of a right and left hand screwed turnbuckle and locking arrangement. By means of this adjustment the measurements between the stops and the right-angle facing or projection on the bell-cranks can be made equal. A gauge is fitted and can be used throughout any series of tests; the gap being adjusted before any reading was taken. The best form of gauge was found to be a hardened steel roller. The reading which then appeared on the springbalance, 2·8 lb. was taken as the zero, and had to be subtracted from subsequent readings.

To test the accuracy with which the springbalance pulls could be *read*, the machine was run at 60 revs. per min. and loaded until *the belt drive* could not be depended upon to keep this constant speed. The friction fabric carrier weighs 7 lb. and to this at regular intervals of 2 lb. 20 loads were added and the corresponding readings taken of pull recorded at springbalance.

Table (2)

60 R.P.M. Loads being added - 'up'. Load being taken off - 'down'

Up lb. b.	1·3	1·65	2	2·5	3	3·5	4	4·6	5·1	5·6	6+	7·15	8-	10·5	10·28+
Down lb ~~lb~~.	1·3	1·65	2	2·5	3·15	3·7	4·2	4·7	5·25	5·7	6·15	7·15	8·15	10·75	10·2

Table (3)

87 R.P.M.

Up lb. b.	1·25	1·6	2	2·6	3·15	3·6	4·15	4·6	5·1	6	8·5	*up lb.*
Down lb. ~~lb~~.	1·3	1·6	2-	2·5	3	3·4	3·9	4·45	5	5·8	8·25	*down lb.*

The/

The second place of decimal is only approximate.

The Speed is taken by means of a revolution counter driven from the main shaft of the geared headstock.
In the meantime calling the steps, 'Low', 'Mid', 'High', using the Epicyclic and 4" pulley -

Low - 1 turn in 24¼ mins. = 0·28 ft. per min.
Mid - 1 turn in 8¼ mins. = 0·825 ft. per min.
High - 1 turn in 2·43 mins. = 2·7 ft. per min.

Low - 8 r.p.m. = 54.5 ft.per min. without epicyclic gear
Mid - 24 r.p.m. = 163·5 ft.per min.
High - 73 r.p.m. = 496 ft. per min.

The Applied Loads are 7, 14, 21 , 28, 35·65, 45·85, 61.5, 71.7 lb. These consisted of carrier block; 4 rods and hanger flanges, pivoted to hang plumb, and series of applied dead loads. It was considered much better to use dead loads, and to dispense with central balance when determining the coefficient of the fabric material. On the wear and temperature effect tests the central springbalance was found useful.

Range of loads at slow speeds. Epicyclic Gear in use.

Table (4)

(SEE ATTACHED SHEET)

Range of Loads at slow speeds. Epicyclic Gear in use.

Ferodo Bonded asbestos (30% cotton) **TABLE (4)**

Best working pressure

Speed ft./min.	7 lb.		14 lb.		21.38 lb.		28 lb.		35.65 lb.		45.85 lb.		61.5 lb.		71.7 lb.	
	Pull	μ	Pull	μ	Pull	μ	Pull	μ	Pull	μ	Pull	μ	Pull	μ	Pull	μ
0.28	1.08	0.31	1.91	0.28	4.78	0.45	4.95	0.35	6.1	0.34	8.3	0.34	10.5	0.34	12.2	0.34
0.35	0.89	0.25	2.15	0.31	5	0.47	4.68	0.34	6.7	0.38	8.2	0.34	10.3	0.33	11.9	0.33
0.566	0.86	0.25	2.52	0.36	3.9	0.37	4.75	0.34	6.1	0.34	7.3	0.30	9.5	0.31	10.7	0.30
0.825	1.05	0.3	2.04	0.29	4.8	0.45	5	0.36	7.5	0.42	8.6	0.36	10.7	0.34	12.7	0.35
1.15	0.95	0.27	2.15	0.31	4.86	0.46	4.88	0.35	6.1	0.34	8.6	0.35	10.5	0.34	12.2	0.34
1.56	1.01	0.29	2.57	0.37	4.82	0.45	4.9	0.35	6.2	0.35	8.1	0.33	10.0	0.33	11.2	0.31
2.7	0.96	0.27	2.2	0.31	5.05	0.47	5.45	0.39	6.8	0.38	-	-	-	-	-	-
3.4	1.09	0.31	2.55	0.37	5.07	0.47	5.2	0.37	-	-	-	-	-	-	-	-
5.12	1.22	0.35	2.92	0.41	4.7	0.44	-	-	-	-	-	-	-	-	-	-

About 20 lb/in² pressure
gives best μ-value

The machine can be depended upon to give good average values of μ up to 55 lb. per sq.in. when using the 4" dia. pulley drive through the Epicyclic Gear. Above this load motion is rather unsteady. With care the higher loads 60 to 75 lb. were applied and the readings obtained were checked on the auxiliary testing machine already described.

Range of Loads when small pulley is Driving.

Table (5)

	Pull	14.2	13.5	13.15	13.75	13.4	13.4	13.35	13.25	11.2
~~lb.~~	Loads 48·5 lb									
	μ	0.62	0.59	0.57	0.6	0.58	0.58	0.58	0.58	0.49
~~lb.~~	Pull	15.7	15.6	15	16.3	15.1	15.6	15.85	14.95	13.8
	Loads 56·05 lb.									
	μ	0.56	0.55	0.53	0.58	0.54	0.55	0.57	0.53	4.9
	Pull	18.45	18.35	17.85	19.3	18.2	15.9	18.8	18.1	17.2
~~lb.~~	Loads 66·25 lb.									
	μ	0.56	0.55	0.54	0.58	0.55	0.48	0.57	0.54	0.52

The speed constant for the machine is 6.8, i.e. revs. per min. x 6.8 = ft. per min.

With Large Pulley Drive and load 28 lb. Zero 2.8 lb.

Table (6)

R.P.M.	Speed ft./min.	Actual Pull lb.	Dry Atmos. μ	Moist Atmos. μ	Brake Wheel Rusting. μ
14	95	8.15	0.582	0.56	0.36
20	136	8.2	0.586	0.56	0.36
28.75	195	8.35	0.596	0.56	0.36
41	279	7.85	0.561	0.52	0.3
60^+	410	7.7	0.55	0.52	0.3
84	572	7.35	0.525	0.52	0.36
125	850	6.55	0.468	0.46	0.4
180	1224	6.35	0.44	0.44	0.4
250	1700	6.2	0.425	0.42	0.4

With/

With Small Pulley Drive and Load 28 lb. Zero 2.8 lb.

R.P.M.	Speed ft./min.	Actual Pull lb.	Dry Atmos. μ.	Moist Atmos. μ.	Brake Wheel Rusting. μ.
4	27.2	8.35	0.596	0.38	0.28
5.6	38	8.2	0.584	0.38	0.26
8	54.4	8.1^{+}	0.58	0.38	0.26
12	81.6	8.1^{-}	0.578	0.38	0.28
17	115.5	8.02	0.572	0.45	0.29
24	166	7.95	0.568	0.45	0.3
36	243.8	7.85	0.56	0.45	0.3
51	340	7.65	0.546	0.45	0.3
71	480	7.15	0.51	0.45	0.3

With Small 4" dia. Pulley and Epicyclic Gear Drive; Load 28 lb. Zero 2.8 lb.

			μ.
1/24.2	0.28	6.1	0.436
	0.35	6.45	0.46
	0.56	6.65	0.48
1/8.2	0.825	6.9	0.50
	1.115	6.75	0.488
	1.56	7.05	0.506
1/2.43	2.7	7.3	0.528
	3.4	7.65	0.554
0.8	5.12	7.85	0.564

Values for Coefficient of Friction about 1/3 dry.

It will be seen from these Speeds that there is a fairly large gap between 5.12 and 27.2 ft. per min. and to span this gap the 4" dia. pulley was replaced by one approximately 16" dia., giving nine speeds, minimum 1.18 ft./min. and maximum 20.48 ft./min. These were rather dangerous speeds at which to run the Epicyclic Gear. The vibration which was set up caused disturbance in the rooms about the Laboratory, the Epicyclic Gear being close up to the roof of the Laboratory. This vibration had no appreciable effect on the working of the fabric friction machine. The readings had to be taken after 6 p.m. or during the week ends, but as far as possible check readings at the other speeds were taken to keep the tests as closely as possible under similar conditions.

From the number of changes of gear which must be made at each test it was necessary to form a system in taking readings. Speeds by means of the revolution counter were converted/

converted into peripheral speeds, and arranged in ascending order; those are shown in Table (6). The arrangement of drives need only be noted, and the speed was taken from the above results. This saved much time and reduced chances of error. Check results during the test showed any changes too small to be appreciable.

The running in of the friction surfaces needs special care. The author suggests that the same care should be given to newly fitted, moulded fabrics, brake linings. If this is not done very erratic values may be expected, when a motor car or any machine is being brought to stand still from high speed in a short period of time. (See Appendix 2)

The humidity of the air as a factor in friction experiments has been clearly illustrated by Dr. Macaulay[4], also by Dr. Rankine [5]. One of the most familiar instruments for measuring atmosphere humidity is the wet and dry bulb hygrometer, and it will be well to remember that, by means of this instrument, allowance could be made for readings taken during the summer and winter. A laboratory may have a much drier atmosphere during the cold weather when heat is supplied by hot air. In the summer the air may be drawn through a water spray and delivered to the rooms loaded with moisture.

<u>Calibration of Heating Arrangement</u> - It was found necessary to employ six bunsen burners to raise the temperature of the rubbing surface from 120°F. to 550°F. and time required was about 20 minutes at 115 revs. per min. of brake wheel. By applying a separate friction pad to the wheel, and adding two blow lamps to the heating arrangements the temperature could be raised fairly rapidly. At the very high speeds it was found impossible to heat the rim as the/

the heat was dissipated quickly. Slow heating caused the coefficient of friction of a friction fabric to fall from 0·52 to 0·34, while μ for the same fabric dropped from 0·52 to 0·2 when the wheel was heated rapidly.

Calibration for Wear Test - The material in this test is 'Breko' Fabric Brake Lining. See Fig. 3, p. 11. Measurements, ff,
Micrometer Readings, $(\overset{(ff)_1}{1{\cdot}904} + \overset{(ff)_2}{1{\cdot}8955} + \overset{(ff)_3}{1{\cdot}905} + \overset{(ff)_4}{1{\cdot}904}) \div 4$
= 1·902 average. (measured in cold).
Applied load = 28 lb. with machine constant 3·25 lb.
Test started at 12 o'clock noon and finished at 4 p.m., i.e. - 4 hours duration of test.

	Average R.P.M.	Load lb.	Spring Balance Pull lb.	Effective Pull lb.	x	Leverage Ratio	μ
1st Hour	80	28	7·75	4·5	x	2	0·32
2nd Hour	81	28	7·5	4·25	x	2	0·31
3rd Hour	82	28	7·4	4·15	x	2	0·29
4th Hour	84	28	7·3	4·05	x	2	0·29

Micrometer Readings of lining thickness

$= (1{\cdot}8925 + 1{\cdot}8916 + 1{\cdot}90 + 1{\cdot}8915) \div 4$

= 1·8939 inches (measured when cold)

Therefore wear in four hours' running = 1·9021 - 1·8939
= 0·0082 inches.

which is a Wear of 0·002 cub. inches per hour.

Second Test,	Wear	of	0·0018	cub.	inches	per	hour.
Third Test,	"	"	0·0015	"	"	"	"
Fourth Test,	"	"	0·0016	"	"	"	"
Fifth Test,	"	"	0·0016	"	"	"	"
Sixth Test,	"	"	0·0016	"	"	"	"

On an average the time to wear Breko Brake Lining (½" thick) to ¼" thick (half original thickness) is 416 hours continuous running.

The/

The specific gravity of the material (Breko) is about 30 grms. per cub. inch. and the wear by weight is approximately 0·0012 gr./hr.cm . Such a test can only be carried out over a long period of time as the lining requires to be weighed before and after test. Check weights are not so easily carried out as check measurements.

Friction tests of fabric material would not be complete without reference to rubber treated cotton fibres; cotton cloth impregnated with Bakelite, of which one firm in Britain uses 15,000 lb. per year. Bakelite is now used for many engineering parts. It is a chemical combination of phenol and formaldehyde, forming a synthetic resin or resinoid, quite different in physical characteristics to natural resin. Mixed with wood meal light brake material can be made, chiefly used for friction discs. Screw threads can be moulded from bakelite compounds. Raw hide still accounts for 15 per cent. of the silent running gear wheels, although 'Ioco' and 'Fabroil' are very much improved non-metallic gear material.

The advantage of keeping a constant supply of heat to the brake rim can be made use of in testing the make up materials employed in the manufacture of brake linings. The heat generated by friction is carried away by the rotating brake wheel. Ventilation conditions are almost perfect in the laboratory. In the case of haulage-way brakes, where the momentum of the running machine can only be checked by the brake action, which converts the energy developed into heat; the wood blocks on which the ferodo lining was fixed became charred and was destroyed inside four months. The heat was so great that it has passed through the asbestos lining and burned the wood.

It has been found difficult to obtain all the possible working conditions, and to carry out long continuous tests on the compounded fabric material. The tests may only show the average/

average coefficient values and these obtained under ideal conditions of working.

TESTS AND TEST RESULTS.

Series (1)

A comparative test was carried out between the working materials, Bronze, Wood, and Leather, and compared with the fabric materials 'Ferasbestos' and 'Gatex' in FIG.6.

The Bronze packing pieces in the carrier were used as a templet from which to form the pad pieces. These pads were carefully adjusted to the curvature of the brake wheel. The conditions of the test are such that no material is overheated.

	Bronze	Ferasbestos	Wood	Leather	Gatex
Pressure lb./in^2	28	28	28	14	14
Rubbing Speed ft./min.	115	115	86	86	86
Max. Temp. °F.	350	300	250	200	175

The heat was supplied by four bunsen burners and regulated to give the required heat to the rim of the wheel so that the temperature of the pad pieces at the brake surface could be read on the potentiometer. A graph was used for the potentiometer which gave readings in degrees F. This graph was furnished by the Natural Philosophy Department of the Royal Technical College and was used throughout this research work. The instrument was checked from time to time during the test period.

For each test the speed was kept constant and the pad pieces were applied to the rim but with only 7 lb. per sq.in. pressure to ensure perfect fitting of brake material, the temperature being raised gently to 100°F. Then the specific load was applied as shown above. The temperature changes were also gradual as shown on graph, Fig. (6).

(a)/

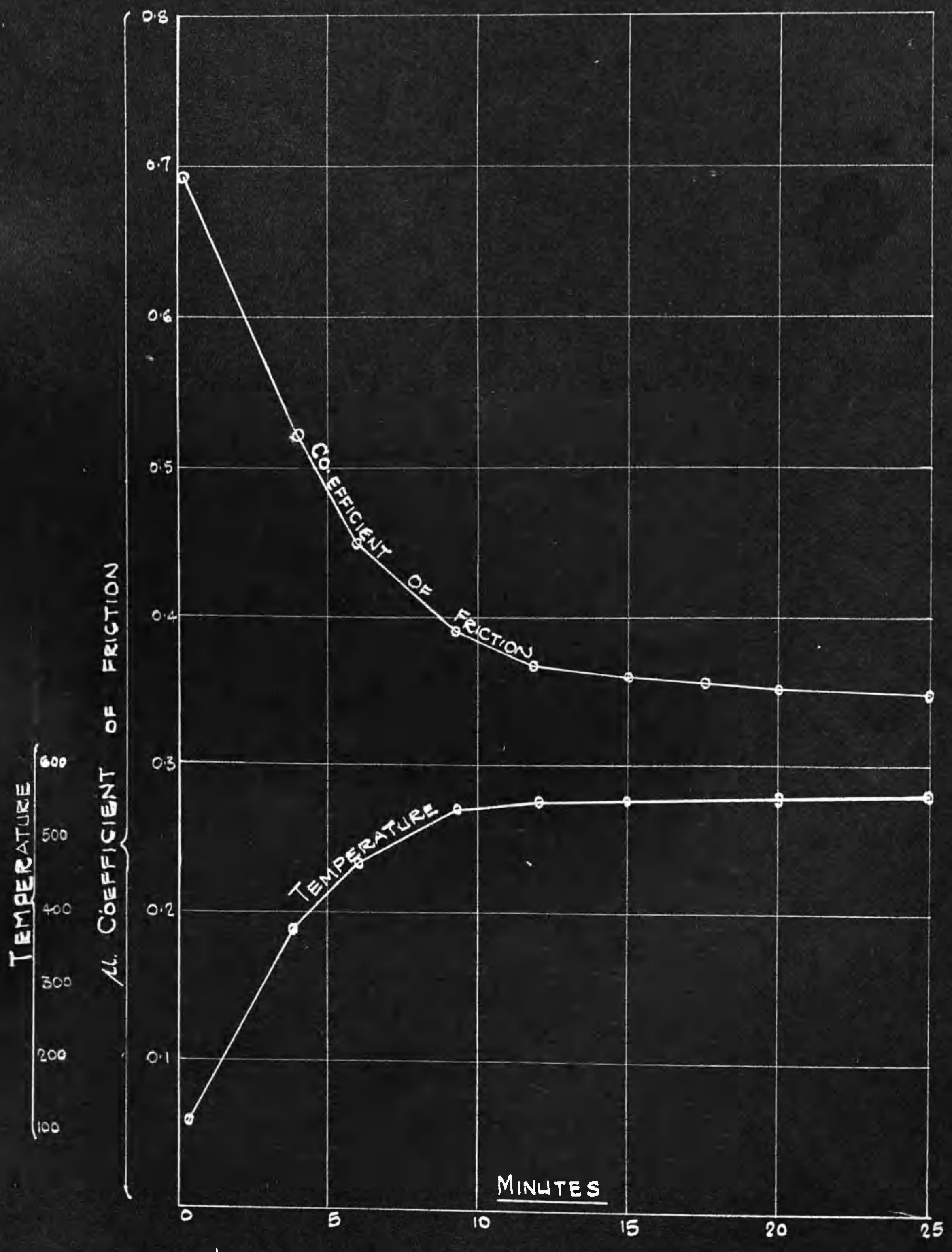
(a) With the Bronze pads scoring of the wheel would seem almost certain if retained in action for any length of time. The coefficient of friction kept fairly constant.

(b) The Wood brake pads showed just that feature which caused them to be discarded in practice where a water cooled drum could not be incorporated, such as in the 'Massey' drop-stamp hammer. Even with the smaller specific loading and lower temperature there is a great variation in the friction coefficient. Charring of the wood pad pieces soon takes place but the brake rim is in no way destroyed; it becomes rather more polished.

(c) The Leather pad pieces had to be retained carefully in the holder, that is, they were not allowed to project more than was necessary to make the cast iron carrier, E, clear the rim. Leather under pressure and temperature does not hold its high μ-value for any length of time, changing from almost 0·9 to 0·75 as the temperature changes from 100° to 175°F. The heat appears to spoil the dressing material of the leather.

(d) The first composite material on the list shows properties which certainly are to be desired. Unlike leather it is not easily deformed by pressure at high temperatures. Heat transmission from wear surface to carrier is readily found by measuring the rise in temperature of the outside of the carrier block, also the inside on non-wearing surface of the brake lining. 'Ferasbestos' is a Ferodo material made by teasing out the single silky-like fibres of the mined asbestos. Each fibre can be treated separately by soaking in a resinoid compound. Asbestos flock may be spun and then treated with fluid gumming material, bakelite, or sugar compound. Pressed lightly into form and baked in an electric drying oven, the lining is then compressed by hydraulic press. The moulds are electrically heated and the pressure may be 15 tons per sq.in.

The/



LOAD ON BRAKE 87 LB OR PRESSURE 87 LBS/SQ INCH.

RUBBING SPEED 25 FEET PER MINUTE

FIGURE 2

The method of manufacture as well as the materials used are brought out in the test results shown in curves (d) Fig. (6), p.14. The brake rim is only slightly polished and the brake pads show no change in surface. The small serrations on the face of the pads used [in Table (1), I - Ferodo Bonded Asbestos], do not appear with the Ferobestos.

(e) 'Gatex' is really a belting material, but as it has a very compact structure and could be cut into the small pad pieces required without excessive fraying of the edges, it was put under test. The load applied was 14 lb.per sq.inch. It shows a very high coefficient of friction much similar to that of leather. It is made of a woven yarn treated with a rubber latrex and the whole is vulcanized into a solid fabric. Very little sign of change in structure could be discerned by the rise from 100° to 200°F. and a pressure of 14 lb. per sq.in. is certainly not excessive.

The author would have liked to carry out research work on the newer form of belting material when used for brake linings. The cotton duck and cotton fibres have given way to woven hair material specially treated so as to have no internal shear among the fibres. Belting problems appear to give a mixture of fluid and solid friction which has given great difficulties to investigators, but in fabric linings, certainly, the manufacturer has overcome many of the early difficulties by special processes . Belting owing to the travel has more cooling surface than brake or clutch material.

<u>Series (2).</u>

These tests were carried out using four different brakes of friction brake linings; they are the same group as used in making the preliminary tests, (I) 'Ferodo Fibre', (II) 'Ferodo Bonded Asbestos', (III) 'Breko', and (IV) Raybestos.

The three last named being very much similar in structure, but made by different manufacturers, who recommend special working pressures to be used for their material. The Author has already written on this subject [6] of Ferodo Brake Linings, but this research work provides an opportunity of comparison without writing unfavourably of any manufacturer's product.

The pressure ranges from 7 to 28 lb. per sq.in. using pads 1 sq.inch area, and to permit of higher pressures special pads of ½ sq.in. are used. The temperature range is from room temperature 55°F. to 120°F. No external heat is applied.

From the earlier tests experience had been gained in running the machine. The object now aimed at is continuous running over the whole speed range with absolute minimum of stops. It was found that, except at the highest speeds, the necessary changes could be made without any damage to the gearing. The rim was marked where gear changing could be made, and it soon became a matter of practice to make a smooth change; the teeth were given a small entrance angle.

Each test was started at top speed and subsequent readings taken in a generally decreasing order of speed. By adopting these methods very few complete stops required to be made. The machine was allowed time to assume constant conditions after each change. This was necessary not only for the speed function but also to permit of the steady-up of the conditions of the brake linings. At each change of load the machine was brought to high speed before proceeding to add load. Up-load and down-load readings were taken for each material.

A rough graph of the readings was made on a load base, also on a time base. This graph enabled a check reading to be made of any doubtful point. At first an average/

average of a number of readings was taken as the final reading, but the machine could be depended upon to give a fair estimate of μ without much care. The principal point to watch is the adjustment of the position on brake rim, and to see that the hangers are not fouling on the sides of the brake wheel.

The results of these tests are shown in Tables (7) to (14) and the curves have been plotted to show the coefficient of friction, μ-value, on a base of specific speed, Figs (6) to (15). For the purpose of comparison the curves have been plotted for separate loadings, Figs. (9a,b,c,d) These curves may be analysed to bring out the respective features inherent in the fabric linings.

Fabric Friction.

Table 7. 'Ferodo' Fibre. Constant of Machine 2·8 lb.

		Pull in lb.				
No. of Drives	Speed ft./min.	7 lb load 7 lb/in^2	17·2 lb 17·2 lb/in^2	27·4 lb 27·4 lb/in^2	17·2 lb 34·4 lb/in^2	22·3 lb 44·6 lb/in^2
e c C	0·258	5·6	12·2	18·1	10·4	13·2
e b C	0·38	5·4	12·2	18·8	10·2	13·4
e a C	0·52	5·4	12·8	19·0	11·8	14·0
e c B	0·75	5·4	12·2	18·9	11·0	13·8
e b B	1·16	5·6	12·6	18·6	11·2	14·2
e a B	1·56	5·8	13·0	19·1	11·4	15·0
e C A	2·45	5·4	12·3	18·4	11·6	15
e b A	3·40	5·6	12·2	19·0	12·2	15
e a A	4·76	5·1	11·1	18·8	12·0	-
		Machine stopped to take out the Epicyclic Train.				
2 c C	25·8	4·4	10·0	14·8	10·8	13·2
2 b C	38	4·0	9·2	13·6	9·8	12·2
2 a C	52	4·0	8·6	13·4	9·2	11·6
2 c B	75	3·8	8·6	13·4	9·2	11·8
1 c C	93·5	3·1	8·6	13·0	9·0	11·7
2 b B	116	3·0	8·1	12·8	8·8	11·4
1 b C	131	3·1	8·2	13·0	9·0	11·6
2 a B	156	3·4	8·4	13·2	9·0	11·4
1 a C	191	3·4	8·2	13·0	8·8	11·6
2 c A	245	4·2	9·4	14·2	9·7	12·8
1 c B	270	3·6	8·6	13·6	9·4	12·0
2 b A	340	4·1	9·6	14·2	10·6	13·4
1 b B	381	3·4	8·4	13·6	9·8	12·2
2 a A	476	4·1	9·6	-	-	-
1 a B	530	3·6	9·0	14·0	9·6	12·2
1 c A	830	3·6	9·6	14·2	9·8	12·6
1 b A	1220	4·0	9·1	14·2	10·0	12·6
1 a A	1700	4·0	9·2	14·0	9·8	12·0

(i) 'Ferodo' Fibre. Constant for machine 2.8 lb.

Table VIII

Coefficient of Friction μ.

Speed ft./min.	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34.4 lb/in^2	44.6 lb/in^2
0·258	0·80	0·70	0·66	0·61	0·50
0.38	0.77	0.71	0.68	0.59	0.59
0.52	0.77	0.74	0.69	0.68	0.65
0.75	0.77	0.71	0.69	0.64	0.62
1.16	0.80	0.73	0.68	0.65	0.64
1.56	0.83	0.76	0.69	0.69	0.70
2.45	0.77	0.72	0.67	0.68	0.67
3.40	0.80	0.71	0.69	0.70	-
4.76	0.72	0.65	0.69	0.68	-

Machine stopped to take out the Epicyclic Gear

Speed ft./min.	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34.4 lb/in^2	44.6 lb/in^2
25.8	0.63	0.58	0.54	0.63	0.59
38	0.57	0.54	0.50	0.57	0.55
52	0.57	0.51	0.49	0.54	0.52
75	0.54	0.51	0.49	0.53	0.53
93.5	0.44	0.51	0.47	0.52	0.52
116	0.43	0.47	0.46	0.51	0.51
131	0.49	0.48	0.47	0.51	0.52
156	0.49	0.49	0.48	0.52	0.51
191	0.49	0.48	0.48	0.51	0.52
245	0.58	0.54	0.51	0.56	0.57
270	0.56	0.50	0.50	0.55	0.55
340	0.58	0.55	0.51	0.61	0.60
381	0.59	0.49	0.50	0.57	0.55
476	0.59	0.55	-	-	-
530	0.59	0.52	0.51	0.56	0.55
830	0.52	0.56	0.52	0.57	0.57
1220	0.57	0.53	0.52	0.58	0.56
1700	0.51	0.53	0.51	0.57	0.54

Temperature variation 60° to 120°F; this refers to last nine readings.

Table IX

(i) 'Ferodo' Bonded Asbestos. Constant for the machine 2.8 lb.

Speed ft./min.	Pull in lb.				Temp.
	7 lb Load 7 lb/in^2	14 lb Load 14 lb/in^2	28 lb Load 28 lb/in^2	17 lb Load 34 lb/in^2	
0.258	2.6	2.8	11.6	7.4	60°F.
0.38	2.5	3.2	10.6	7.4	
0.52	2.8	2.0	10.8	7.6	
0.75	2.8	3.0	11	7.8	
1.16	3.2	3.2	10.6	8.0	
1.56	2.6	3.2	11.4	8.0	
2.45	2.2	3.4	11.2	8.4	
3.40	2.4	3.0	11.4	8.0	
4.76	2.8	3.6	11.8	7.8	

Machine stopped, Epicyclic Gear disengaged.

Speed ft./min.	Pull in lb.				Temp.
	7 lb Load 7 lb/in^2	14 lb Load 14 lb/in^2	28 lb Load 28 lb/in^2	17 lb Load 34 lb/in^2	
25.8	3.0	4.2	12.6	9.8	60°F.
38	3.0	4.2	12.4	9.2	
52	3.0	4.0	12.4	9.2	
75	3.2	4.0	12.4	8.8	
93.5	2.8	4.0	12.8	8.8	
116	2.6	3.8	12.2	8.4	
131	3.0	3.6	12.4	8.6	
156	2.6	3.6	12.0	8.6	80°F.
191	2.8	3.6	11.9	8.4	
245	2.6	4.0	13.4	9.0	
270	2.8	3.5	11.8	8.4	
340	2.6	3.8	13.4	8.8	
381	3.4	3.2	11.6	8.2	
476	2.4	3.6	-	8.6	100°F.
530	3.6	3.0	11.4	8.0	
830	2.4	3.0	12.2	8.6	
1120	2.6	2.8	12.0	8.2	120°F.
1700	3.0	2.8	11.6	7.8	

Diagrams to show the
appearance of Fabric
after test.

Table X.

(ii) 'Ferodo' Bonded Asbestos. Machine No. 2.8 lb.

Speed ft./min.	Coefficient of Friction μ				Temp.
	7 lb/in^2	14 lb/in^2	28 lb/in^2	34 lb/in^2	
0.258	0.37	0.40	0.42	0.43	
0.38	0.37	0.46	0.39	0.43	
0.52	0.40	0.39	0.39	0.38	
0.75	0.40	0.43	0.40	0.45	
1.16	0.46	0.46	0.39	0.46	60°F.
1.56	0.37	0.46	0.38	0.47	
2.45	0.31	0.48	0.40	0.48	
3.40	0.34	0.43	0.38	0.46	
4.76	0.40	0.51	0.43	0.45	
Machine stopped, Epicyclic Gear locked.					
25.8	0.43	0.60	0.46	0.57	60°F.
38	0.43	0.60	0.45	0.53	
52	0.43	0.57	0.45	0.53	
75	0.45	0.57	0.45	0.51	
93.5	0.40	0.57	0.45	0.51	
116	0.37	0.54	0.45	0.49	
131	0.43	0.51	0.45	0.49	80°F.
156	0.40	0.51	0.44	0.50	
191	0.37	0.51	0.43	0.49	
245	0.40	0.57	0.48	0.52	
270	0.37	0.50	0.43	0.49	
340	0.48	0.54	0.48	0.51	
381	0.34	0.46	0.42	0.48	
476	0.43	0.51	-	0.50	
530	0.34	0.43	0.38	0.46	
830	0.37	0.43	0.44	0.50	
1220	0.43	0.40	0.43	0.48	120°F.
1700	0.43	0.40	0.42	0.45	

Speed ft./min.	Pull in lb.				Temp.
	7 lb Load 7 lb/in^2	14 lb Load 14 lb/in^2	28 lb Load 28 lb/in^2	17 lb Load 34 lb/in^2	
25.8	3.0	4.2	12.6	9.8	60°F.
38	3.0	4.2	12.4	9.2	
52	3.0	4.0	12.4	9.2	
75	3.2	4.0	12.4	8.8	
93.5	2.8	4.0	12.8	8.8	
116	2.6	3.8	12.2	8.4	
131	3.0	3.6	12.4	8.6	
156	2.6	3.6	12.0	8.6	80°F.
191	2.8	3.6	11.9	8.4	
245	2.6	4.0	13.4	9.0	
270	2.8	3.5	11.8	8.4	
340	2.6	3.8	13.4	8.8	
381	3.4	3.2	11.6	8.2	
476	2.4	3.6	-	8.6	100°F.
530	3.6	3.0	11.4	8.0	
830	2.4	3.0	12.2	8.6	
1120	2.6	2.8	12.0	8.2	120°F.
1700	3.0	2.8	11.6	7.8	

Diagrams to show the
appearance of Fabric
after test.

Table X.

(ii) 'Ferodo' Bonded Asbestos. Machine No. 2.8 lb.

Speed ft./min.	Coefficient of Friction μ				Temp.
	7 lb/in^2	14 lb/in^2	28 lb/in^2	34 lb/in^2	
0.258	0.37	0.40	0.42	0.43	
0.38	0.37	0.46	0.39	0.43	
0.52	0.40	0.39	0.39	0.38	
0.75	0.40	0.43	0.40	0.45	
1.16	0.46	0.46	0.39	0.46	60°F.
1.56	0.37	0.46	0.38	0.47	
2.45	0.31	0.48	0.40	0.48	
3.40	0.34	0.43	0.38	0.46	
4.76	0.40	0.51	0.43	0.45	
	Machine stopped, Epicyclic Gear locked.				
25.8	0.43	0.60	0.46	0.57	60°F.
38	0.43	0.60	0.45	0.53	
52	0.43	0.57	0.45	0.53	
75	0.45	0.57	0.45	0.51	
93.5	0.40	0.57	0.45	0.51	
116	0.37	0.54	0.45	0.49	
131	0.43	0.51	0.45	0.49	80°F.
156	0.40	0.51	0.44	0.50	
191	0.37	0.51	0.43	0.49	
245	0.40	0.57	0.48	0.52	
270	0.37	0.50	0.43	0.49	
340	0.48	0.54	0.48	0.51	
381	0.34	0.46	0.42	0.48	
476	0.43	0.51	-	0.50	
530	0.34	0.43	0.38	0.46	
830	0.37	0.43	0.44	0.50	
1220	0.43	0.40	0.43	0.48	120°F.
1700	0.43	0.40	0.42	0.45	

During the working with this material it was found that the machine showed up a fierceness of grip at slow speeds.

Table X

(ii) 'Breko' Bonded Asbestos. Constant for machine 2.8 lb.

Speed ft./min.	Pull in lb.				
	7 lb load	17.2 lb load	27.4 lb load	17.2 lb load	22.3 lb load
	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34.4 lb/in^2	44.6 lb/in^2
0	2.6	8.2	12.8	7.1	9.4
0	2.6	7.8	12.4	7.6	9.8
0	3.0	7.8	12.2	7.8	10.4
0	2.8	7.8	13.0	7.0	9.0
1.16	2.8	8.0	13.2	7.6	10.0
1.50	2.4	7.2	12.4	7.6	10.0
2.45	3.0	8.6	13.6	6.2	8.6
3.4	2.6	7.8	13.6	7.0	9.6
4.76	2.1	7.0	13.5	8.0	10.0
Machine stopped to lock Epicyclic Gear					
25.8	2.4	6.6	10.8	6.0	7.8
38	2.4	6.8	10.8	5.8	7.6
52	2.2	6.4	10.4	5.8	7.6
75	2.0	6.4	9.8	5.8	7.6
95.5	2.1	6.0	9.6	5.6	7.2
116	2.0	5.8	9.4	5.4	7.2
131	2.0	6.0	9.4	5.6	7.1
156	1.8	6.4	9.2	5.4	7.0
191	1.8	6.4	9.2	5.6	7.0
245	2.2	5.8	10.6	5.4	7.6
270	2.0	6.6	9.4	5.4	7.0
340	2.4	5.6	10.1	5.8	7.8
381	1.8	6.2	9.2	5.4	6.8
476	2.6	5.8	-	6.2	-
530	1.8	5.6	8.8	5.0	6.8
830	1.8	5.6	9.3	5.4	7.2
1220	2.0	3.8	9.0	5.6	7.6

Note: With this material the friction machine worked very smoothly.

Table XI

(iii) 'Breko' Bonded Asbestos. Constant for machine 2.8 lb.

Speed ft./min.	Coefficient of Friction μ				
	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34 lb/in^2	44.6 lb/in^2
0.258	0.37	0.48	0.47	0.41	0.42
0.38	0.37	0.45	0.45	0.44	0.44
0.52	0.42	0.45	0.45	0.45	0.46
0.75	0.40	0.45	0.47	0.41	0.41
1.16	0.40	0.46	0.48	0.44	0.45
1.56	0.34	0.42	0.45	0.44	0.45
2.45	0.42	0.50	0.49	0.41	0.43
3.40	0.37	0.45	0.49	0.41	0.43
4.76	0.30	0.41	0.48	0.46	0.44

Machine stopped to lock Epicyclic Gear.

Speed ft./min.	Coefficient of Friction μ				
	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34 lb/in^2	44.6 lb/in^2
25.8	0.34	0.38	0.39	0.35	0.35
38	0.34	0.39	0.39	0.34	0.34
52	0.31	0.37	0.38	0.34	0.34
75	0.29	0.37	0.35	0.34	0.34
95.5	0.30	0.37	0.35	0.33	0.32
116	0.29	0.35	0.34	0.31	0.32
131	0.29	0.34	0.34	0.32	0.32
156	0.26	0.35	0.34	0.31	0.31
191	0.26	0.35	0.34	0.31	0.31
245	0.31	0.37	0.38	0.33	0.31
270	0.29	0.34	0.34	0.31	0.34
340	0.34	0.38	0.37	0.33	0.32
381	0.35	0.33	0.34	0.30	0.30
476	0.37	0.36	-	0.36	-
530	0.26	0.30	0.32	0.30	0.31
830	0.26	0.33	0.34	0.31	0.32
1220	0.28	0.34	0.34	0.31	0.32
1700	0.28	0.34	0.33	0.32	0.34

There was practically no change in temperature throughout the test.

Table XII

(IV) 'Raybestos' Bonded Asbestos. Constant for machine 2.8 lb.

Speed ft./min.	Pull in lb.				
	7 lb load 7 lb/in^2	17.2 lb load 17.2 lb/in^2	27.4 lb load 27.4 lb/in^2	17.2 lb load 34.4 lb/in^2	22.3 lb load 44.6 lb/in^2
0.258	2.6	7.2	11.4	7.0	8.8
0.38	2.6	7.0	11.0	7.0	9.2
0.52	2.8	7.4	12.2	7.0	8.6
0.75	3.0	7.4	12.0	7.0	9.2
1.16	2.4	7.2	11.4	7.6	9.5
1.56	3.2	7.6	12.0	7.0	9.0
2.45	3.0	7.4	11.8	7.6	10.4
3.40	2.6	7.0	11.0	7.4	9.6
4.76	3.2	8.2	12.2	7.8	10.4
Machine stopped to lock Epicyclic Gear					
25.8	2.4	7.6	12.2	7.4	9.6
38	2.4	7.0	11.4	7.0	9.2
52	2.6	7.0	11.0	6.8	9.0
75	2.4	7.2	11.2	7.2	9.6
93.5	2.5	7.0	11.2	6.8	9.2
116	2.6	6.4	10.6	6.5	9.0
131	2.4	6.6	11.0	6.6	8.8
156	2.0	6.0	10.6	6.2	8.4
191	2.2	6.2	10.4	6.4	8.4
245	2.6	7.2	11.4	7.4	9.6
270	2.4	6.6	10.8	6.8	8.8
340	2.8	7.2	10.4	7.4	9.4
381	2.2	6.2	10.0	6.4	8.4
476	2.4	6.8	-	-	-
530	2.2	6.2	10.0	6.0	7.6
830	2.6	6.6	10.8	6.4	8.6
1220	2.2	6.4	10.4	6.2	8.4
1700	2.0	6.0	10.4	6.3	8.6

(IV) 'Raybestos' Bonded Asbestos. Constant for machine 2.8 lb

Speed ft./min.	Coefficient of friction μ				
	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34.4 lb/in^2	44.6 lb/in^2
0.258	0.37	0.48	0.38	0.41	0.39
0.38	0.37	0.41	0.40	0.41	0.41
0.52	0.40	0.43	0.44	0.41	0.39
0.75	0.42	0.43	0.44	0.41	0.41
1.16	0.38	0.42	0.38	0.44	0.42
1.56	0.45	0.44	0.44	0.41	0.40
2.45	0.43	0.43	0.40	0.44	0.46
3.40	0.38	0.41	0.40	0.43	0.43
4.76	0.45	0.47	0.44	0.45	0.46
Machine stopped to lock Epicyclic Gear.					
25.8	0.34	0.44	0.4	0.43	0.43
38	0.34	0.41	0.4	0.41	0.41
52	0.37	0.41	0.4	0.31	0.40
75	0.34	0.42	0.4	0.41	0.43
93.5	0.36	0.41	0.4	0.31	0.41
116	0.37	0.37	0.39	0.37	0.40
131	0.34	0.38	0.40	0.38	0.39
156	0.29	0.35	0.39	0.36	0.38
191	0.31	0.36	0.38	0.37	0.38
245	0.37	0.42	0.42	0.43	0.43
270	0.34	0.38	0.39	0.40	0.39
340	0.40	0.41	0.38	0.43	0.47
381	0.32	0.36	0.37	0.37	0.38
476	0.34	0.39	-	-	-
530	0.32	0.36	0.37	0.35	0.34
830	0.37	0.38	0.39	0.37	0.39
1220	0.32	0.37	0.38	0.36	0.38
1700	0.30	0.35	0.38	0.37	0.38

The machine worked smoothly with this material. There is little or no rise in temperature, 60° to 100°F.

(V) The brake lining material as now manufactured is a very much improved product. A test was carried out on friction fabric which was found to be composed of Asbestos, Magnesium Compound, Fuller's Earth, and about 30 per cent. of cotton, the whole impregnated with what the maker called a secret 'Friction'. It is at least 20 years old and had been sent to the Laboratory of the Mechanics Department of the Royal Technical College for testing purposes, and had to be given up as no consistent results could be obtained. The makers claimed a coefficient of 0.4.

Using the Friction Machine the following results were obtained.

Table XIII. Constant for machine 2.8 lb.

Static μ	Speed ft./min.	7 lb/in^2		14 lb/in^2		17.05 lb/in^2	
		Pull lb.	μ	Pull lb.	μ	Pull lb.	μ
	25.8	2.8	0.80	4.50	0.64	59.0	0.69
	38	2.65	0.76	4.80	0.67	5.80	0.68
0.3	52	2.45	0.70	4.80	0.68	4.97	0.59
	75	2.6	0.74	4.55	0.65	5.85	0.68
to	95.5	2.75	0.78	4.90	0.70	5.87	0.69
	116	-	-	-	-	6.05	0.71
0.4	131	2.7	0.77	4.85	0.69	5.65	0.66
	156	-	-	-	-	6.00	0.71
Using	191	2.6	0.74	4.65	0.66	6.05	0.71
	245	-	-	-	-	5.90	0.69
Brake	270	2.55	0.73	4.50	0.64	5.85	0.68
	340	-	-	-	-	5.75	0.68
Wheel.	381	2.45	0.7	4.20	0.60	5.57	0.65
	476	-	-	-	-	5.40	0.63
	530	-	-	-	-	5.20	0.62
	830	-	-	-	-	5.10	0.60
	1220	-	-	-	-	5.05	0.59
	1700	-	-	-	-	4.35	0.5

A change in the temperature from 60° to 250°F. lowered the μ from 0.7 to 0.2 when taken in conjunction with the 17,05 lb./in^2 pressure. Running at this temperature, 250°F. for five minutes destroyed the fabric material. It would certainly be dangerous to use this friction fabric, for continuous lowering purposes on a mine incline. The material appeared fairly hard but a soaking in water almost destroyed its form, and even at low temperature the wear was excessive.

(VI) After the usual tests were carried out on this pure asbestos friction brake lining, 'Chekko', the small driving pulley was replaced by a 22-inch pulley. Calibration of the speeds of the machine gave the following:-

	Speed - ft. per min.								
With Epicyclic Gear	1.37	1.94	2.83	4.24	5.2	8.5	12.4	17.3	25.5
Without Epicyclic Gear	142	175	305	421	587	815	1360	2040	2580

It will be seen on examining the previous tables of speeds that in the portion marked 'Machine stopped to lock Epicyclic/

Epicyclic Gear' that the speed changes from 4.76 to 25.8 ft. per min. This rather big change in crawl speeds has been beautifully spanned by the last five speeds in the top line of table, namely 5.2 to 25.5 ft. per min. There is also a gain of three new top speeds 1360 to 2580 ft. per min. The author does not recommend the frequent use of these high speeds as the Epicyclic Gear was designed to run at very slow speeds.

(VI) 'Chekko' Bonded Pure Asbestos, Light Brass Bonding.

Table XIV

Speeds ft./min.	Coefficient of Friction, μ			
	21.37 lb/in^2	28 lb/in^2	35.65 lb/in^2	45.85 lb/in^2
1.37	0.46	0.43	0.54	0.54
1.94	0.47	0.46	0.55	0.54
2.83	0.48	0.47	0.56	0.54
4.24	0.48	0.47	0.57	0.56
5.2	0.48	0.49	0.61	0.58
8.5	0.49	0.51	0.63	0.58
12.4	0.49	0.53	0.61	0.58
17.3	0.50	0.54	0.60	0.58
25.5	0.55	0.55	0.60	0.59
Machine stopped to lock Epicyclic Gear.				
96	0.63	0.64	0.61	0.60
136	0.62	0.62	0.65	0.58
142	0.56	0.63	0.66	0.60
175	0.58	0.66	0.56	0.59
305	0.64	0.60	0.60	0.55
421	0.58	0.55	0.62	0.57
587	0.56	0.49	0.53	0.50
815	0.54	0.47	0.53	0.48
1360	0.53	0.45	0.54	0.51
2040	0.48	0.43	0.52	0.49
2580	0.46	0.41	0.50	0.49

From 1⅝ inch diameter pulley driven from the D.C. motor, the other seven speeds of this series being neglected.

Wear Test on 'Chekko' Bonded Asbestos.

Room Temperature at 11.55 a.m. 49°F.

Micrometer Readings before Test 1.9125 (cold)
After Test 1.9093 (cold)

Difference 0.0032 inches = wear

Pull on Springbalance, 9.2 lb. Zero Reading 2.8 lb.
Balance arm, 2 : 1.

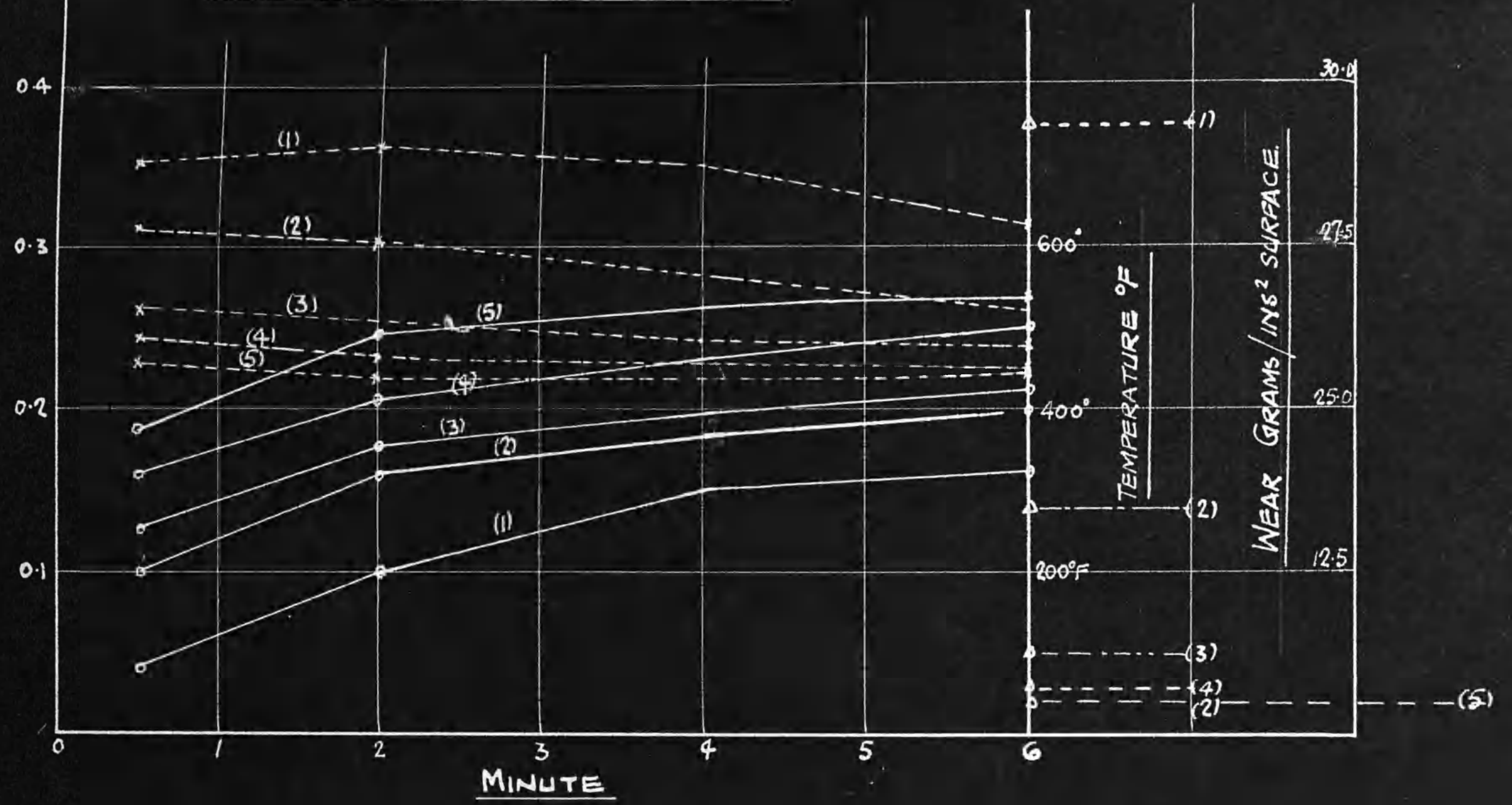
Speed 2000 ft. per minute. Load 21⅜ lb.

"CHEKKO" BONDED ASBESTOS

PRESSURE 120 LBS/□"

RUBBING SPEED 550 FT/MIN.

WEAR AREA CONSIDERED = 1 SQ. IN.



TEMPERATE, CO-EFFICIENT AND WEAR TEST

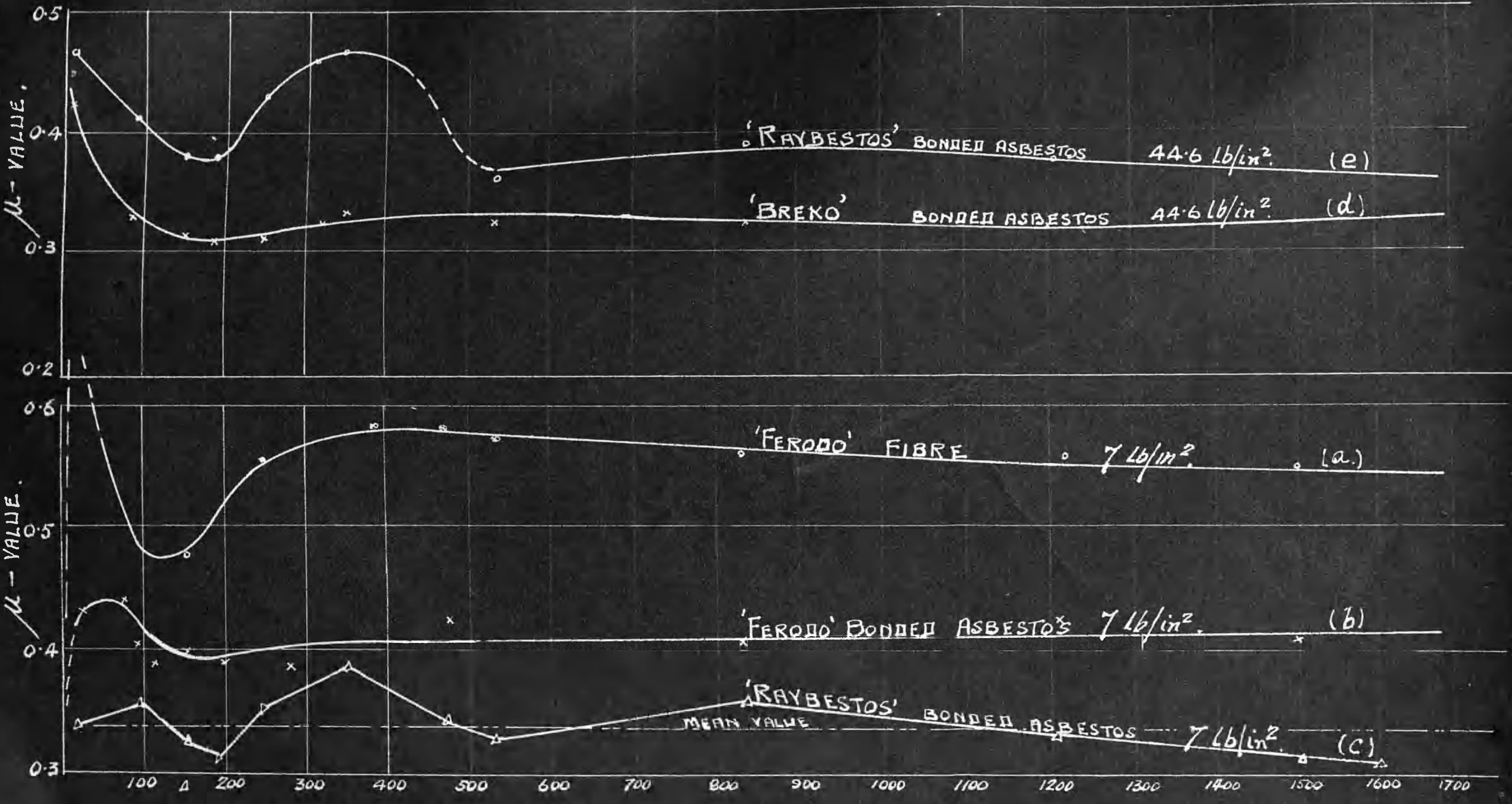


FIGURE 9 — a, b, c, d, e. RUBBING SPEED FT/MIN.

Time	Spring Balance Pull, lb.	Temperature	value. μ
11.55 a.m.	9.2	49°F.	0.6
12	9.4		
12.15 p.m.	9.35		
12.30	9.25	65°F.	
12.45	9.15		
1	9.15	70°F.	0.58
1.15	9.15		
1.30	9.15	75°F.	
1.45	9.15		
2	9.15	75°F.	
2.15	9.10		
2.30	9.05		
2.35	9.05	76°F.	0.56

Load changed to 36.65 lb. μ

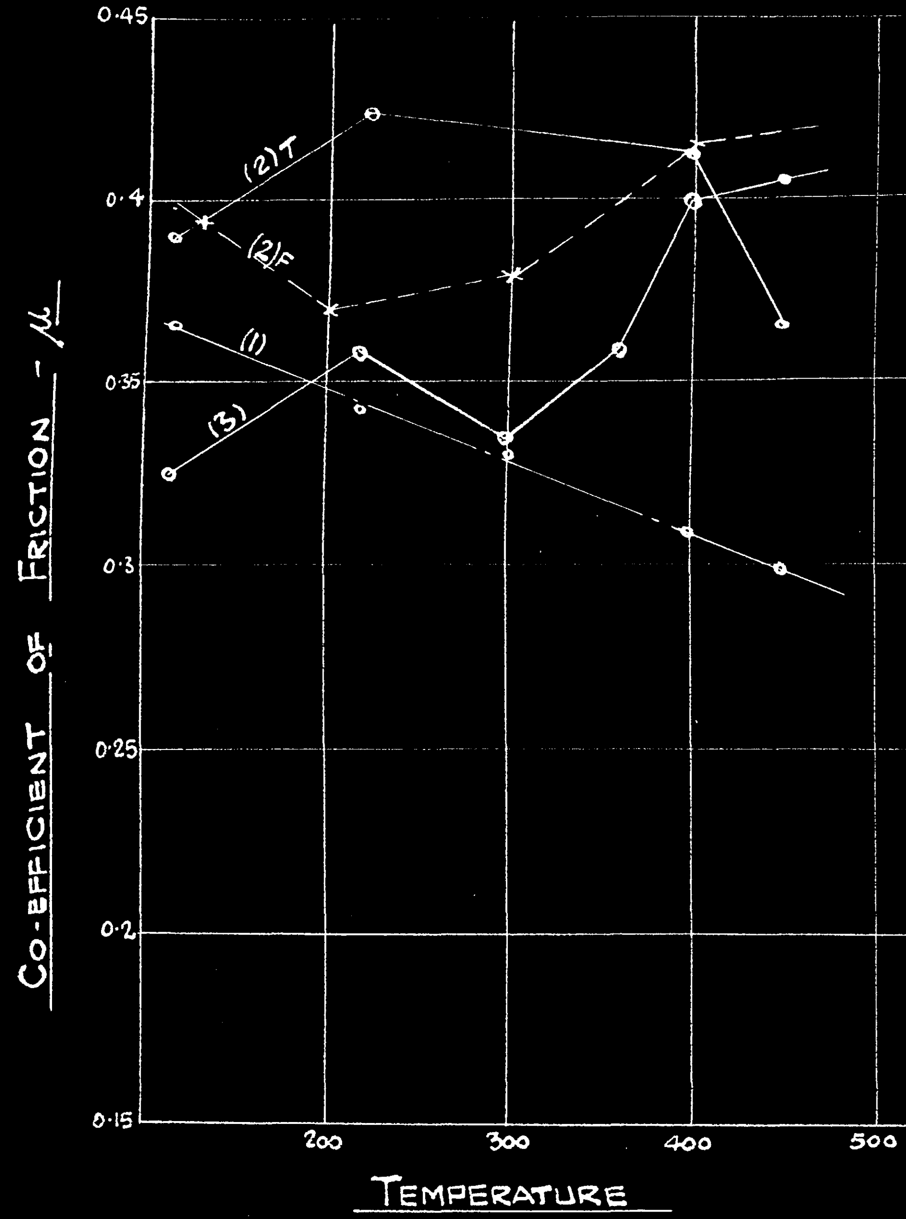
2.50 p.m.	12.45	75°F.	0.53
3	12.55	80°F.	
3.10	12.58	85°F.	0.55
3.40	12.55	95°F.	
4	12.55	100°F.	0.54
4.20	12.5	105°F.	
4.40	12.45	110°F.	0.53
4.50	12.45	110°F.	
5	12.45	110°F.	0.54

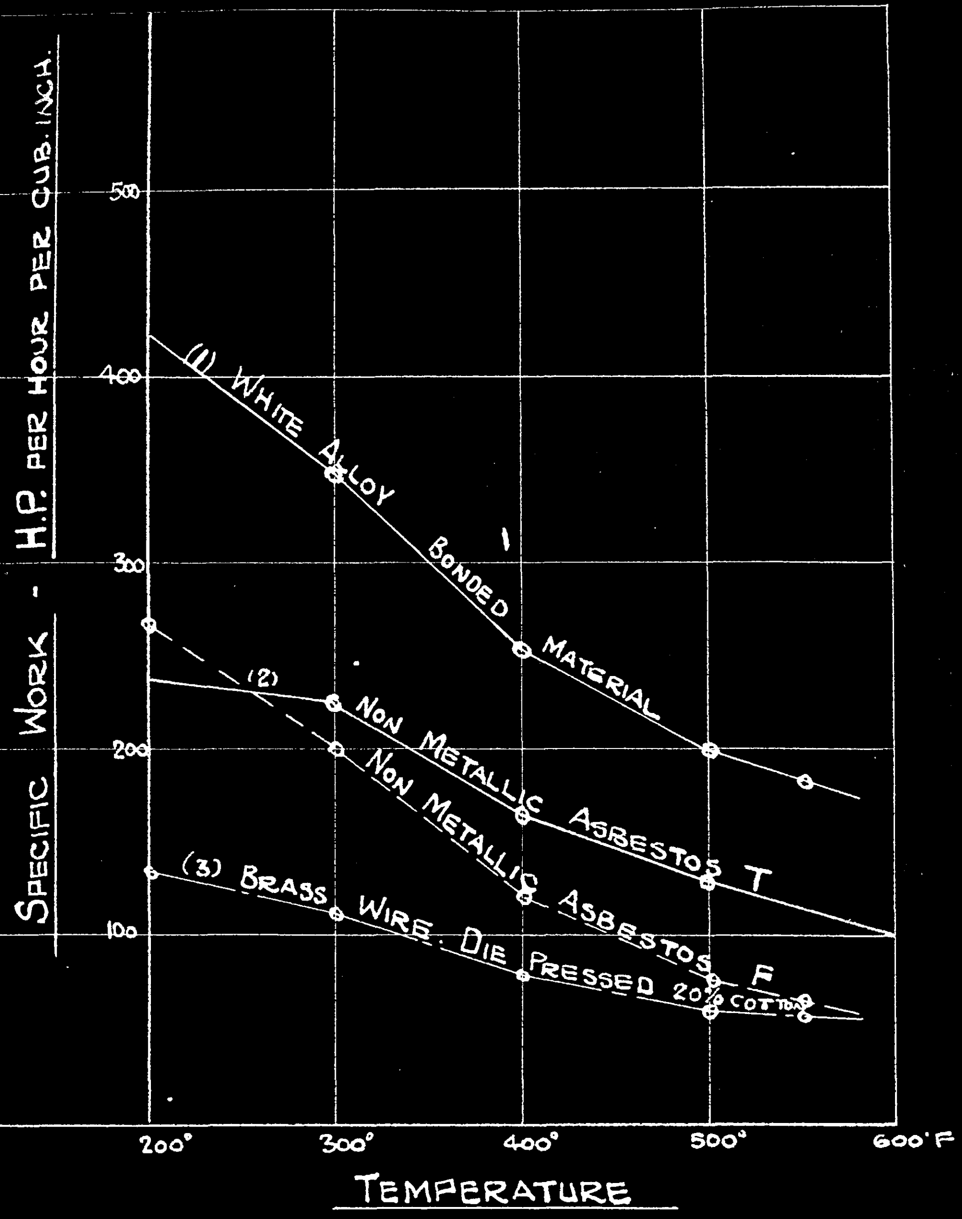
Room Temperature 52°F. Micrometer (1.9093 - 1.9089)
= 0.004 inches.

When the 'Chekko' material is to be supplied for clutches the makers prescribe a specific loading 12 to 20 lb/in^2. In order to carry out temperature tests, taking two different loads and speeds, four bunsen flames were arranged round the interior of the rim supplied the heat. It was afterwards found that these required to be augmented by the aid of two blow lamps, and the brake wheel had to be shielded from the cold draughts as much as possible.

'Chekko' Bonded Asbestos. Micro. readings, 1.9207.

Time	Temperature of Rim °F.	Coefficient of Friction, μ	Remarks Dia. 26"
2.45 p.m.	110	0.58	Surface dry, Speed 1360 ft/min.
2.51	196	0.56	Specific load, $21\frac{3}{8}$ lb/in^2
3	238	0.53	Wheel partly enclosed
3.20 p.m.	242	0.52	Speed 587 ft./min.
4.10	163	0.70	Specific Load 7 lb/in^2
4.18	70	0.647	Wheel not enclosed.





WEAR IN TERMS OF HORSE POWER HOUR PER UNIT VOLUME

The excessive wear during the first period of wear-tests is more noticeable with the woven material than with the highly pressed (under temperature) moulded material. The latter can be machined and does not require much running-in, since the bearing surface is more regular than with the more soft woven and impregnated material.

From the average values of a large number of tests on woven fabric pure asbestos the curves in Fig. 9 have been plotted. These was excessive wear at first, which gradually dropped down to an almost stationary value of wear; the curves show that the rise in temperature is almost constant; while the value of the coefficient of friction falls uniformly over the first four tests, 24 minutes, and then remains almost constant over a long period test. The period of time between each test was a half minute, and the friction wheel was protected from cold currents.

Between room temperature and 300°F. the wear appears to be directly in proportion to the amount of work dissipated at the brake wheel rim. Above 300°F. the wear is excessive and grows rapidly for all Woven Bonded Fabrics; it is in this region that fabrics containing cotton become charred. The wear for nearly all fabrics is twice as great at 520° as at 260°F. for similar speeds.

Effect of Change of Pressure and Speed, but pv = constant.

During the tests with constant temperature regulated by the bunsen gas flame, the amount of wear was unchanged by adding load and decreasing speed, or decreasing load and increasing speed, to give equal works (p.v.). With a fabric material having a low μ-value it is found better to work with/

with a low p-value, say 10 to 12 lb/in^2 maximum and a high rubbing speed V. With the high coefficient of friction fabrics a pressure of 30 lb/in^2 is found to be more suitable with lower rubbing speed. Tests carried out on a drop-hammer ferodo brake lifting tup showed that, when 50,000 ft. lb. per sq.in. per minute was demanded from lining, the material wore out very rapidly with air cooled brake rim pulley, but with water cooled brake pulley the wear was not excessive. All tests carried out on the various fabrics show that it is not advisable to design for more than 12,500 ft.lb. per sq.in. per minute. A graph showing specific work, H.P.hr. per sq.in. plotted on temperature °F. is given at Fig. 10.

Effect of Rapid Heating on μ-value.

A new brake lining having a μ-value = 0.48 at°350 F. is applied to the brake wheel which when running has the temperature raised to nearly 350°F. before applying the brake. After the brake is applied the temperature rises rapidly to 575°F., time approximately 10 minutes. The μ-value of the material is reduced to 0.18 and may be taken on average as 0.2 for 20 minutes before disintegration of lining material takes place. The same material at 425°F. has μ-value = 0.4 but when raised to 575°F. in 10 minutes has still a μ-value = 0.35, and never goes under μ = 0.3. In both cases the μ-value increases as disintegration of lining takes place caused by the binding material becoming plastic. It was also found that the more slowly a brake lining is heated up the less likely is the bonding material to leave the fabric. The graph, Fig, 11 , shows how the coefficient of friction varies with temperature in a number of fabrics. Taking an average of five different makers' linings, the maximum T-value is about 250°F. and the minimum 400°F.

Effect/

Effect of Roughening Surface of Brake Rim. -

Some cotton and duck fabrics have been tested which were made with a sulphur greasy compound incorporated while under pressure. It was found that oxidization took place, and smoothness of action soon disappeared. Where cast iron drums have been roughened by a file it has been observed that the fabric lining suffers very severely, and after a relatively short time, it is unfit for use. An iron cement was applied to a small pulley rim and Ferodo Fibre was used as a brake. Under the application of a very light pressure wear was excessive and heat generated rapidly, showing that in brake linings abrasion material would be detrimental.

Effect of Applying Water or Lubricants to Brake Wheel.-

In this set of experiments, "Raybestos", bonded asbestos brake lining is used. The speed is kept (approx.) constant at 1000 ft. per minute, V, and with a pressure of 45 lb/in^2 the μ - value is 0.38. Raising the temperature of the brake rim is followed by a drop in the value of the coefficient of friction. Removing the bunsen burners and applying water to the surface of the brake rim lowers the μ - value considerably and as long as the water is allowed to wet the brake surface of the wheel this coefficient becomes smaller. On removing the water and again applying heat the μ-values rises as the moisture dries off and is noted to be greater than original starting value. A series of these changes are shown, Fig. 12, in order to depict the various changes; shocks or shudders are seen on the brake wheel's motion.

When moist conditions prevail during tests the following changes are noticeable:

(1) that the high rubbing value of the friction fabric materials change when working on a water lubricated surface ;

(2) that μ - values for all materials are lowered by one third dry test values;

(3)/

(3) That a thin coating of rust or iron-oxide has been found to lower the μ-value by one half.

The pad pieces, in one test, after being fitted to the curvature of the brake drum were allowed to soak in water for three days. The fabric material was dryed on blotting paper and placed in the holder. It only required a second or two to reach the maximum μ-value yet there certainly was a distinct lag in reaching μ-value found in previous tests. The lining was not destroyed by the moisture, and it did not appear to affect the lining after the first test run which pointed to the fact that the surface moisture, rather than penetration of water into the bonded fabric, was the cause of the temporary change in the value of the friction coefficient.

To prevent oil or water drip reaching the rubbing surface, rubber coating in the form of a solution has been tried on the brake lining leading edge. i.e. the first edge opposing the motion of wheel. Experiments on pads with rubber solution on edges were not carried to any length, but showed clearly that fluid could be guided past the brake surface by this simple contrivance.

Effect of Lubricant on Fabric Material.

Apart from the lowering of the μ-value, lubrication did not destroy in any way the die pressed fabric material, which, freed from oil, soon regained its normal coefficient of friction value. On heating up the brake wheel when tallow had been applied to the brake-pad surfaces, very erratic gripping was shown on the wheel's motion. The passing from lubrication film to boundary lubrication and then to dry condition could clearly be seen. It was not intended in this investigation to try out the different lubricating oils tested in the author's second investigation, but as a matter of interest a set of brake pads was sparingly treated/

treated with Bayonne oil, castor oil, and tallow. The mineral oil was soon rubbed off and the coefficient of friction of the fabric changed rapidly from 0.1 to 0.35. The tallow appeared to last longer as a lubricating medium with a coefficient of 0.15 rising to 0.35. The castor oil did appear to affect the lining but in rather a peculiar way as it gave a friction coefficient reading of nearly 0.2 and at one time this value reached 0.41 showing that this oil had acted as a dressing; on running for some time the value fell to 0.35 which was the dry coefficient value of the fabric undergoing test.

On the friction machine experiments have been carried out with dry and tallow coated cotton or duck belting. As this is a fabric, which when impregnated with rubber, has been used for brake linings, a few readings are given; fuller tests on this subject from a different point of view have been made at the Royal Technical College by Laird[7]. This paper deals with "Viscous effects in Dynamometer Belts", and the author in reading over this paper for publication purposes soon discovered that Laird's investigations pointed to the fact that the following formula

$$T_1 = T_e\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right), \text{ where } T_e = T_1 - T_2$$

effective tension, was not easy of manipulation, since the difference in tensions depends on belt creep and slip, arc of contact between belt and pulley face, character of pulley surface and belt surface, also velocity of belt. In the dry cotton belt tests a slight formation of oxide on the brake rim gave variable readings; the humidity of the atmosphere, or the least trace of moisture in the room caused variations in the readings. On the introduction of another variable, namely a tallow coating on the cotton brake band, made $T_1 - T_2$ still more variable, and the usual belt/

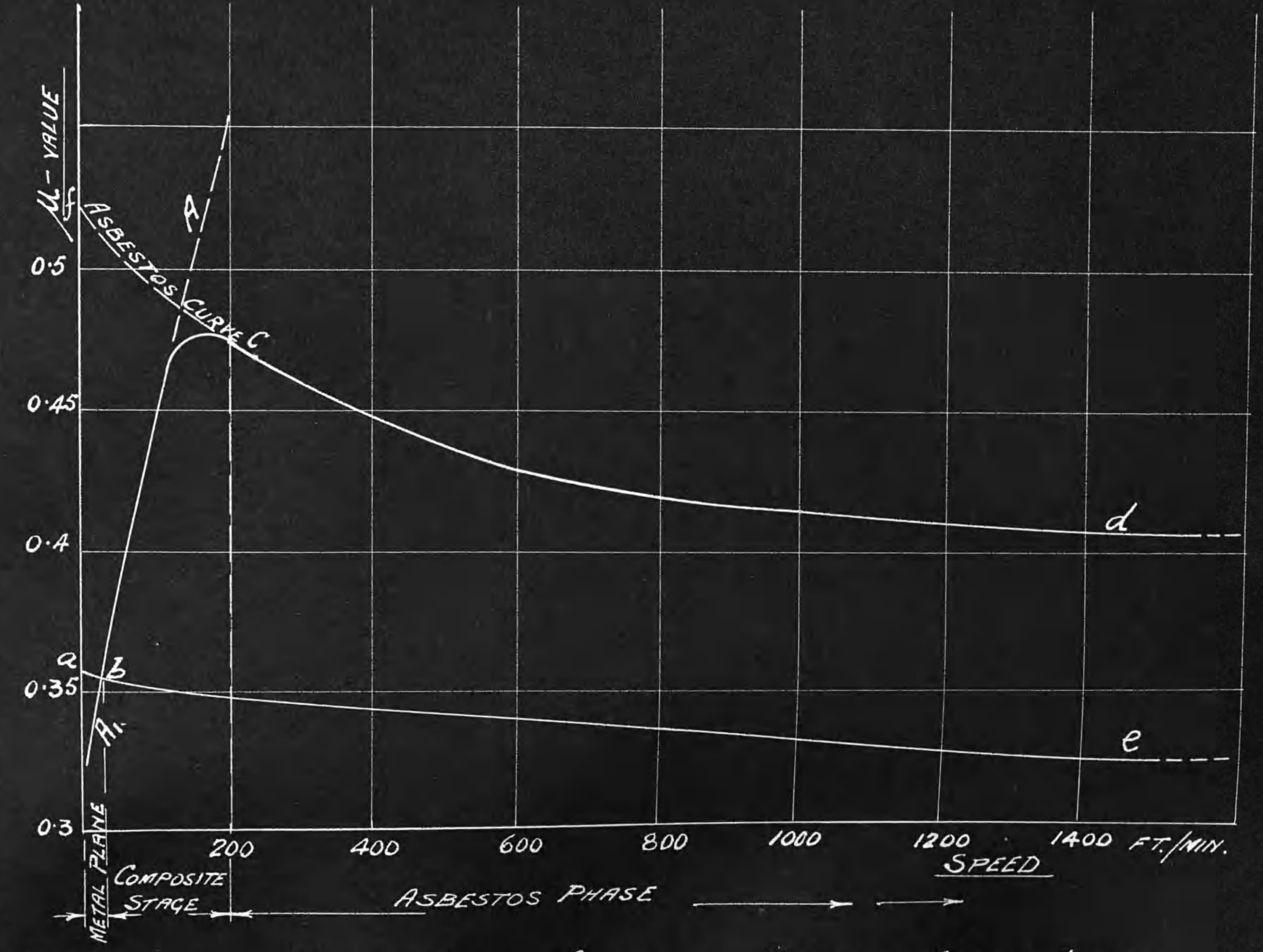


FIGURE THREE PHASE CURVE FOR BONDED FABRIC LINING.

Fig. 13

belt formula $\frac{T_1}{T_2} = e^{\mu\theta}$ was not applicable as the test results show. From the curve plotted from experimental data Laird gave the formula $T_1 - T_2 = K(\lambda A V W)^{\frac{1}{2}}$ where λ = coefficient of viscosity; A is the area of film in shear or area of contact between the belt and the wheel; V is the speed of slipping; and W is calculated from end tensions as $\left(\frac{T_1 + T_2}{2}\right)\theta$.

The resemblance between the formula given in the author's investigation of lubricants submitted as part of this thesis is very noticeable.

(1) Laird's $(T_1 - T_2) = K(\lambda A V W)^{\frac{1}{2}}$

(2) author's $\mu = K\left(\frac{\lambda V}{2P}\right)^{\frac{1}{2}}$ lubricated bearing.

(3) Boswall's $\mu = K\left(\frac{ZN}{P}\right)^{\cdot 58}$ for clearance bearings.

(4) Dry belting $(T_1 - T_2) = e^{\mu\theta}$

If the testing of lubricated cotton belting entails the consideration of film lubrication with all its inherent difficulties. Fabric Brake Linings, which are made from bonded asbestos (plus in some cases 10 to 30 per cent of cotton), brass wire (which does not always have the same bearing surface), and bonding material (with varying viscosity), added to these the possibility of oil or water lubrication on brake wheel rim, have given to this investigation a most difficult number of variations and complicated combinations of variables.

Considering the curve, Fig. 13 which has been taken from a bonded asbestos lining, there are three distinct phases in this curve which may be denoted by the (1) Brass or Metal plane, (2) Composition Stage, (3) Asbestos Plane; this is assuming no cotton mixture in fabric.

The portion f.d. represents the curve of asbestos alone.

The portion a.e. represents the curve of brass or metal alone, where a,b,c,d is the actual curve obtained for the material. The first portion of the curve, namely line AA appears/

appears to be determined by the intensity of the loading, being near to the axis the higher the loading becomes. The metal curve shows little change with increase of speed of rubbing, but in the "composite stage" the μ-value increases rapidly with the increase of speed. The asbestos part of curve decreases with the increase of speed, very rapidly at first and then becomes more nearly constant.

(1) for metal μ = 0.3 to 0.38 average 0.34 depending on the metal used.

(2) composite material (metal, asbestos plus bonding material) μ increases at the rate of 0.15 per 100 ft. per minute.

(3) for asbestos $\mu = 0.43 \rightarrow \text{speed}^{0.03}$, this being an average derived from curves with 7 lb and 28 lb per sq.in. pressure loading; the equations for the curves, Fig. 13 a. on Fig 20 being (a) log μ = -0.35 -0.02 log speed.

(b) log μ = -0.38 -0.04 log speed.

Using as a lubricant machine oil changes the coefficient in the phase (1) from 0.3 to 0.1 in four minutes, and the asbestos when lubricated still lowers this value and at the end of another period of five minutes the μ-value has become 0.02. In the case of pressed block linings the lowest value reached was 0.08 which is almost equal to metal on metal. Curves, Fig. 14 [p.45] have been drawn to illustrate the changes which take place in temperature and μ-values when lubricant is used on the Brake Wheel.

CONSIDERATION OF RESULTS ON BRAKE LININGS

From the foregoing experimental results given in this investigation it will be seen that a wide variation of results is due to the variable nature of the combined materials and their method of manufacture. The Asbestos Fibres are washed with mineral acids in order to make the fibres more absorbent, and also to prevent co-agulation of rubber while they are being impregnated with latex. The percentage mixture is usually/

usually 80 asbestos to 20 rubber; the latex usually forms 30 per cent. solid content, and vegetable fibres may replace asbestos in part. Sulphur may also be added. This in the 'Investigation on Lubricants' has been placed among the materials useful for carrying away heat from an over-heated journal bearing. The mixture of materials, Asbestos, Rubber, Sulphur, etc. is made into a dough; this in the case of the Asbestos Bonded Brake Linings is forced into the meshes of the wire fabric, and put into moulds in the case of moulded linings without metal. After being allowed to set at a moderate heat, 150°F., in a vacuum drier, the slab or mould is compressed to about $^1/_3$ of its initial thickness. The slabs and the moulds are kept heated while under hydraulic pressure, and this tends towards constancy of finished material. The temperature of the moulds vary from 275 to 400°F. and the hydraulic load applied may be such as to give 50 tons.

(1) These linings have a hardness of Rochwell No.B. 95 and can be machined. Thus when a brake strap arrangement is properly designed to give uniform pressure, the lining may be made to fit closely on to the brake wheel. A fault found in carrying out experiments with fabric material made in 1926 was the lack of smooth surface.

(2) The discarding of the metal binder is another disadvantage brought out by this investigation owing to quantity of metal surface exposed to rubbing wear not being uniform.

(3) The material as now manufactured is not destroyed by exposure to water or oil, but this does not mean that an oiled surface can be used for a heavy load brake or friction clutch. The fabric material behaves in the same manner as all lubricated metal surfaces subjected to running speed and pressure/

pressure. The asbestos composition can be used for clutches designed for working in oil. This is the material used in combined clutch and change-speed lever gear.

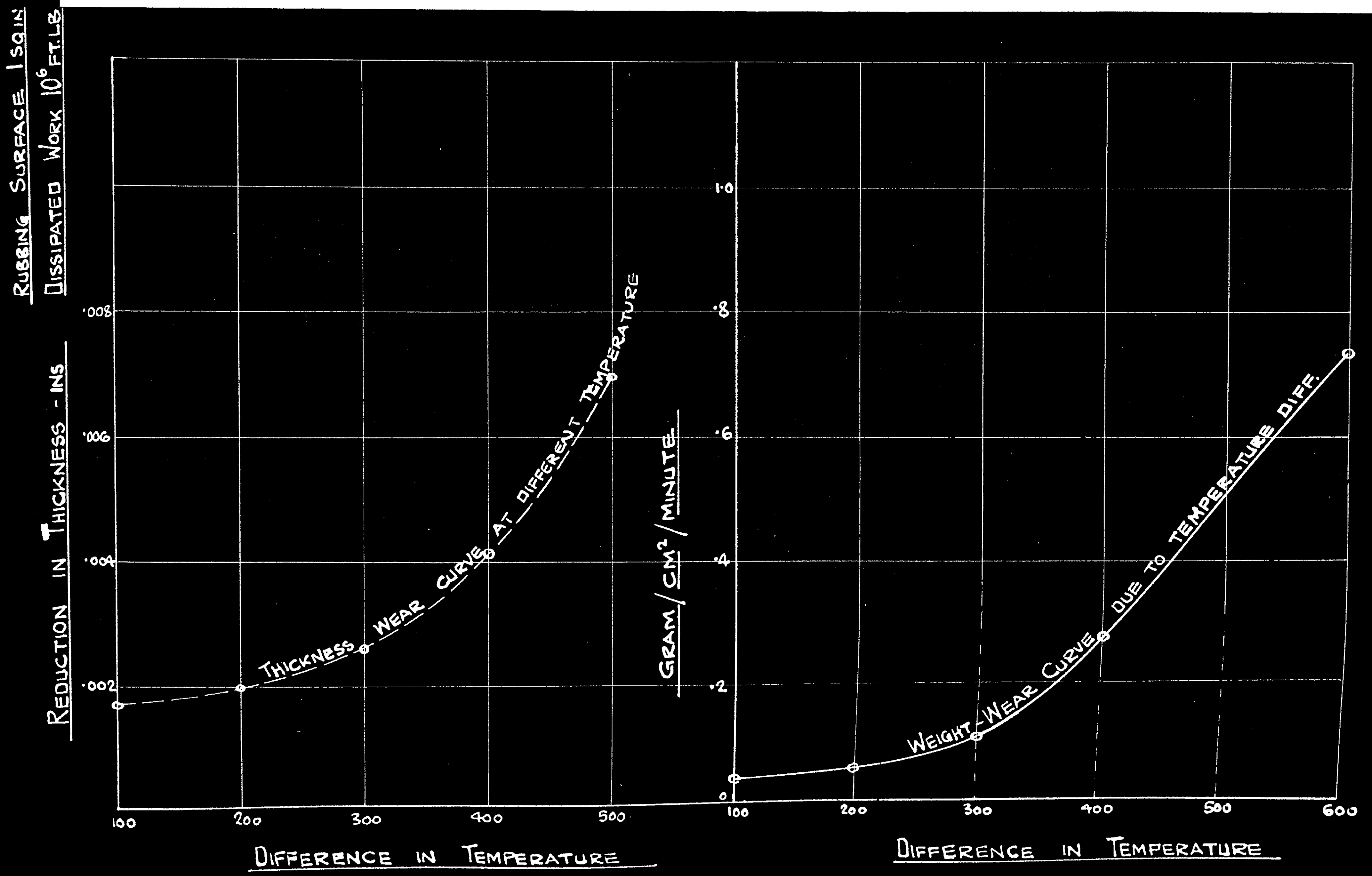
(4) Temperature effects are such that with many built up fabrics excessive wear takes place at high temperatures. The temperature, with the newer hot-pressed linings, may now reach 500°F. without destruction.

(5) Wear, which was measured by change of thickness in a given time, or by change of weight over a given period of rubbing, has been measured by absorption of energy per unit of volume (one cubic centimetre) and expressed in horse-power hours per cubic centimetre or ft.lb. per min. per sq.in. Fig 15

This method of measuring wear could be stated as an index of the pefformance of brake linings, and could be made a basis of comparison, horse power hours per cubic centimetre, but must not be taken when making a comparison between service brakes fitted to machines where excessive pressure may be applied intermittently and for short periods. The brakes are never allowed to cool but are used at peak temperatures where wear is excessive.

(6) The nature of the surface of the brake rim is a deciding factor. In the main tests a cast iron brake drum was used, but moulded type of brake lining has been proved more suitable to use with low carbon steel pressed drums than bonded asbestos. Metal, introduced to strengthen the yarn, causes seizure to take place which is followed by a fluctuating frictional value which results in 'snatch' or 'shudder' and brake drum scoring.

At slow crawl speeds 'snatch' and 'shudder' is very noticeable in the case of high μ-value fabric material. While the low μ-value material such as Breko, with its low steady frictional value of 0.28 to 0.25, provides a smooth and fairly/


RUBBING SURFACE 1 SQ IN
DISSIPATED WORK 10⁶ FT. LB.
REDUCTION IN THICKNESS - INS
·002
·004
·006
·008
THICKNESS WEAR CURVE AT DIFFERENT TEMPERATURE
100
200
300
400
500
DIFFERENCE IN TEMPERATURE
GRAM/CM²/MINUTE
0
·2
·4
·6
·8
1·0
WEIGHT-WEAR CURVE DUE TO TEMPERATURE DIFF.
100
200
300
400
500
600
DIFFERENCE IN TEMPERATURE

WEAR CURVES.

fairly rapid retardation. This class of material, used with a low bearing pressure, was found to be good for electric lifting cranes and hoisting machinery, a fact confirmed by tests carried out by Messrs. Adamson at their crane works.

CONCLUSIONS ON BRAKE LININGS

This investigation had for its aim the comparison of fabric linings as manufactured, a grouping together of test results as required by the engineer, an encouragement to standardization of tests, and quality of material used. Since the first results of these tests[6] were published 1930, manufacturers have improved their running tests, and guided by results obtained by these, and other experimenters' results on durability when consideration is taken of duty imposed they have gone on creating new and better materials suited for carrying heavier loads, increasing speeds, and stopping powers when suddenly applied to moving machinery.

Much of the data given in this investigation are applicable to Friction Clutches and Friction Spur and bevel wheels. The only difference is in the range of pressures applied in clutches and wheels which may vary from 25 to 75 lb. per sq.in., whereas in brake linings the pressure on an average is about 15 lb. per sq.in. with a maximum of 30 lb. per sq.in. Even a lower pressure 12 lb. per sq.in. has been used by crane makers, and suggested by test results obtained in this investigation. The brake drums and brake shoes used in practice, even when made of pressings are not flexible. The brake linings in all the tests described are "worn in", that is, in applying the brake all points must touch the drum at the same moment. In applying the brake the lining is being compressed, the compression being in proportion to the pressure which is exerted on the lining (Hooke's Law), or the sum of all friction forces $\Sigma F = \mu \Sigma N$.

The higher the self-actuation the more sensitive is a brake to change in friction coefficient. Therefore, if too high/

high a degree of self-actuation is used in a brake design, the brake is liable to "grab" in some cases, while it will not hold sufficiently in others. That is why it is much more important to obtain a brake lining which is affected as little as possible by temperature, moisture, speed, etc. than to obtain a high frictional coefficient. There appears to be no way of judging the self-actuation of a brake shoe. Different designs show different features and makers of brake linings are now inclined to specific brake linings to be used with certain designs.

Fabric Material used for Gearing.

When gear wheels are made from fabrics, such as 'Ioco' and 'Fabroil', consideration must be given to the Tensile Strength of the material, to the hardness of the fabric on the root surface of the tooth and to the wearing surface of the material. The duty of wheel teeth is to transmit power with least possible loss. The efficiency of the gearing depends on lubrication, which is quite different from Fabric Brake Lining which seek to use the dry rubbing surfaces, sliding friction, without seeking to use seizure action which would destroy the surfaces. In gear teeth the action is part rolling and part sliding friction with line contact.

Comparison is made between material used for wheel blanks.

(1) Fabroil pinions which are made from compressed cotton (specially prepared cotton fibres). This material takes the place of compressed paper and wood pulp blanks. In this case there is no binder used and compressor therefore has to be maintained by means of steel side plates, or shrouds; if these are removed the Fabroil is useless. In this class raw-hide pinions and pinions made from Ioco sheet may be classed.

The nature of these fabrics made testing on the Friction/

Friction Machine difficult as the cutting up into ½-in. squares to fit the fabric holder, Fig. 3 was almost impossible. Test carried out on this material showed that the dry friction was rather erratic, varying from static friction 0.25 to running friction 0.6, and again falling rapidly to 0.25 as heat brought out the dressing. When lubricant was applied the μ- value dropped by about $\frac{2}{3}$ giving a μ-value of 0.16 to 0.2. (Tallow lubricant). Castor oil dressing when applied to the friction pads was inclined to cause a viscous drag.

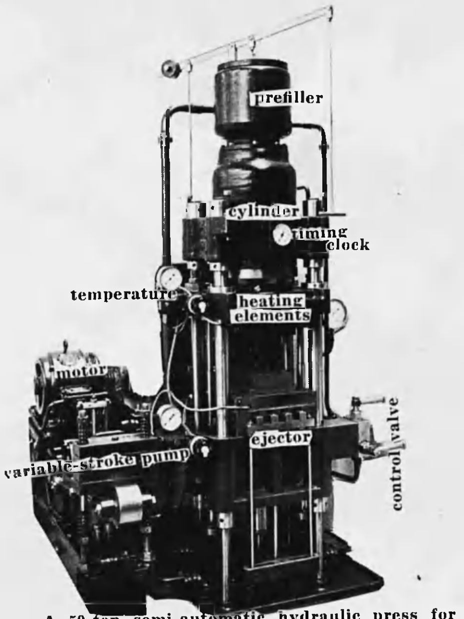
(2) Fabroil-A cut very easily into ½-in. squares and when run on the Friction Machine gave smooth running and taking on a beautiful polished surface which worked well with cast iron. The lubricated friction tests gave a value for μ = 0.1 to 0.06. A high pressure could be applied to this material, and oil did not tend to destroy its structure even when running against the grain of the texture, yet it does not appear to have the strength of the shrouded pinion.

(3) Ferodo bonded asbestos could not be used for gearing but for light duty Ferodo fibre could be used.

(4) Ioco is made in much the same manner as Fabroil the only difference being that in a semi-plastic state the woven cotton duck strip is impregnated with synthetic resin and compressed in steam heated moulds under pressure. It is finished as it comes from the moulds except for slight trimming at the parting line in gear blanks, and curing for about 20 hours at 212°F. Arrangement of Press is shown in Fig. 16.

(5) Cotton cloth impregnated with Bakelite is pressed into heated moulds to make turning gears for oil pumps. Duck cloth has usually a tensile strength of 24 to 30 lb. per inch width per ply.

Table/



A 50-ton semi-automatic hydraulic press for plastic moulding, with motor-driven variable-stroke automatic pump.

Figure 16

Gear Wheel
Built up
and shrouded

Rockwell Tests on
10 Blanks
Max - 86 B. } average 71
Min - 69 B }

Figures 17

Table of comparative strength and hardness.

Material	Ultimate Tensile Strength	Izod ft lb.	Hardness Max.	Hardness Min.
Fabroil-A $\frac{7}{8}$" thick	4 tons/in^2	2	48.8	44.4 Brinell
Formapex $\frac{1}{2}$" thick	Tensile 5½ tons/in^2	3	36	34 "
Micarta	7 tons/in^2		(45	41 "
	(Shear) 2 tons/in^2		(99	93 Rockwell B
Ferodo Fibre $\frac{1}{2}$" thick	2 tons/in^2	1	98	Rockwell B
Formapex Gear Wheel Blank	3 tons/in^2 at boss and rim	1.5	78	77 Rockwell B.
	5 tons at web.	2.5	82	73.5 "
Vulcanized Fibre	7.5 tons/in^2	3.5	75	70 Rockwell B

The Young's Modulus for all fabrics is approximately 1×10^6. The blanks can be machined by using tools with large top rake; circular milling hobs with steep top rake are used for cutting teeth in blanks, and grinding may be done preferably by a rapidly rotating fine-grained emery wheel, (e.g. No. 40 emery, 5 inch diam, 3000 r.p.m.), taking care to avoid undue heating of the fabric. These features were clearly brought out during the machining of the various test pieces.

Cross-Breaking Strength or Maximum Fibre Stress.

For most practical purposes this is defined as the load, P, required to fracture, by bending, a bar-shaped test-piece of given rectangular cross section (b = breadth, d = depth), when this is supported or held horizontally and the load applied vertically in one of the following forms of test:-

(a) Cantilever Test: The bar is held by a clamp at one end and load applied at a distance, L, from the near edge of the clamp. Stress = $\frac{6PL}{bd^2}$. Ioco, 0.5" x .75" x 2.5". Safe load, 25 lb.

(b)/

(b) 3-point Loading test: The bar is supported at two points a given distance, L, apart, and the load applied midway between them. Stress = $3PL/2bd^2$, Fabroil-A = 0.5" x .75" x 2.5", Safe Load, 95 lb.

Impact Strength

(c) This is expressed as energy per unit area of cross section of the test piece where fracture occurs, but this test varies with the shape of the cross section. This gave approximately 2 to 3 ft.lb. for all fabric material.

Plastic Yield Test - (d): Test as in (b) at a given temperature (200°F.) for a given time 9 minutes, or 80°F. for a period not less than 12 hours. One of the Fabroil-A pieces was turned to ¼" diameter at ends and a plain portion ⅛" diameter for 2" long. This test piece was subjected to torsion and gave on an average 10000 lb/in^2 (max. 13000 lb/in^2, min. 9000 lb/in^2).

Absorption Tests - The Vulcanized Fibre, although like the other fibres tested was insoluble in ordinary solvents, was seen to absorb water (swells). In this respect Ioco and Fabroil materials are better than Vulcanized Fibre, wood, raw-hide, etc.

Formapex Miocarta and Fabroil-A gears and blanks are sent out from factory with metal centres which are permanently held in position without rivets. Sketches of a blank and a cut gear are shown in Fig.17 . These can be made much stronger and lighter than when cut from slabs.

From the formula $HP = \frac{0.0001 \times V \times f \times b \times y}{D.P}$ derived for Ioco material, let breadth of face of wheel b = 1 inch and the diametrical pitch D.P. = 1, then the formula for horse power becomes $HP = .0001 \times V \times f \times y$ per inch of face per 1 diametrical pitch. Assume pressure angle $14\frac{1}{2}^{\circ}$, then $\log HP = \log 0.0001 + \log V + \log f + \log y$. To make an alignment chart to suit this logarithmic formula of/

of four variables, let log r = log V + log f.

If V varies from 100 to 2000 ft. per sec. then

log 20000 - log 100 = 1.301. magnify this by 5, say 6.505"

The safe stress of the material for different speed may vary from 1000 to 4000.

Then log 4000 - log 1000 = 0.602. magnify by 10, say 6.02".

The form factor will vary from 0.075 to 0.118.

log 0.118 - log 0.075 = 0.2. magnify by 1/30 say 6"

If the distance between scale V, (S_v), and scale f, (S_f), = 3"

then scale of r will be distant from V by e"

$$\text{when } e = \frac{S_f}{S_v + S_f} \times 3 = \frac{\frac{1}{10}}{\frac{1}{5} + \frac{1}{10}} \times 3 = \frac{\frac{3}{10}}{\frac{3}{10}} = 1''$$

and scale of r, (S_r), = $\frac{3}{10}$

Again place scale y at distance 3" from scale r,

$$\text{then } e = \frac{S_y}{S_r + S_y} \times 3 = \frac{\frac{1}{30}}{\frac{3}{10} + \frac{1}{30}} \times 3 = \frac{3}{10} = 0.3$$

Then scale for H.P. (S_{HP}) = $\frac{1}{3}$

$$\text{Substitute in formula HP} = 0.0001 \times \overset{V}{2000} \times \overset{f}{1000} \times .118$$

$$= 23.6$$

$$\text{and HP} = 0.0001 \times 100 \times 4000 \times .075$$

$$= 3.$$

The chart has now been drawn, Fig. 18 from this data and gives any of the four variables when the others are known. This chart has been improved by adding the scale for the pitch diameter of the wheels, the number of teeth and the diametrial pitch. The formula which gives these dimensions and numbers are

(1) $\frac{(\text{diam. of gear})\ \text{r.p.m.}}{12}$ = Velocity in ft.per min.

= 0.26 d x r.p.m. = V

(2) circumferential pitch p = $\frac{\pi}{D.P.}$ and n p = πd

Therefore n = d x D.P.

From/

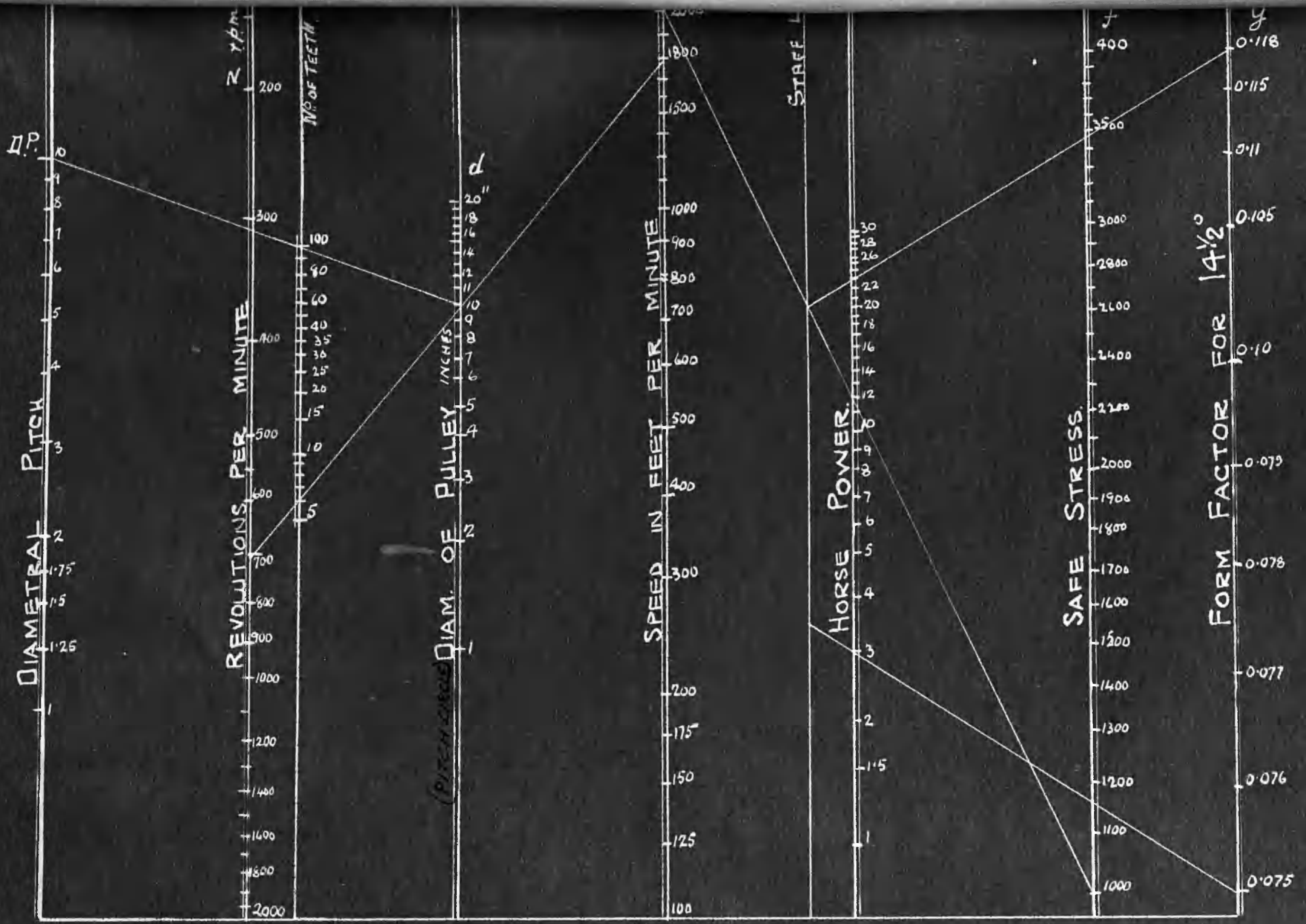


FIGURE 18 — NOMOGRAM FOR "FORMAPEX" MIOCARTA (BAKELITE) GEAR WHEEL —

From (1) $\log d = \log V - \log .26 - \log N$

From (2) $\log D.P. = \log n - \log d$

By the same method the scales N and D.P. have been added. These could have been drawn separately but by the addition of key-lines the scales are clearly connected.

The author was asked by Professor Mellanby to furnish the manufacturers of Ioco Fabric with a formula to suit their material, and this has been added as a matter of interest, and is a practical illustration of the ease with which nomograms can be adopted to test results. An experimental apparatus was also designed to test the endurance of fabric gearing when running in conjunction with cast iron wheels.

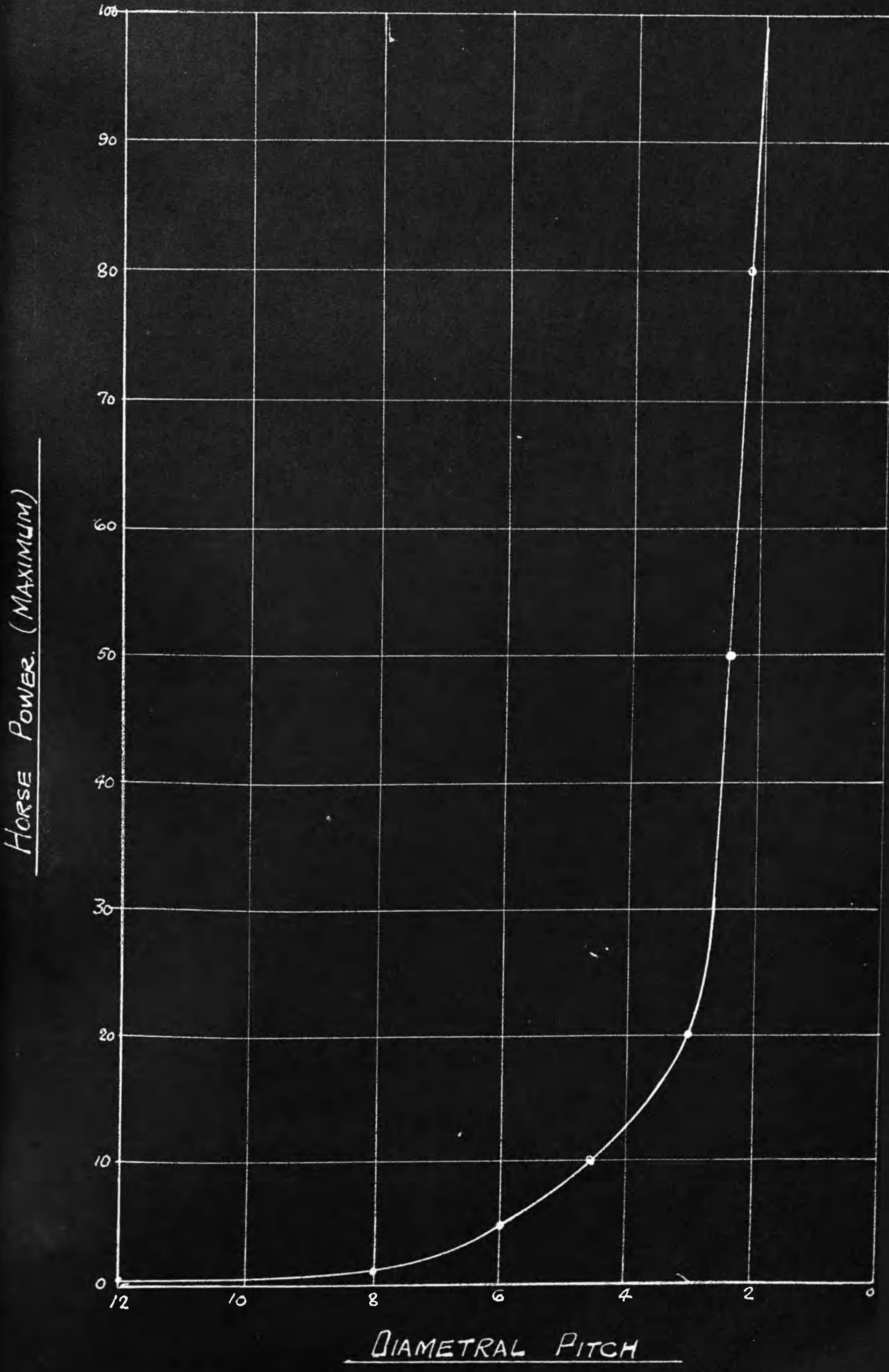
For the first run-in of fabric gearing it is advisable to use a solid lubricant such as a mixture of vaseline and graphite, or colloidal graphite. A thin coating of this lubricant during the run-in period helps to make a polished smooth surface, afterwards the gears may be run in a bath of good lubricating oil.

The fabrics might be made to suit the chuck of a reversal of bending stress machine, and could be subjected to a sliding motion abrasion as a speedier method of testing the gear blank pressed fabric material. This would certainly give a very much shorter test than was recommended to the Ioco Company.

From the alignment chart the table and graph horse powers for various diametrical pitch have been drawn up for easy application, Fig 19.

Diametrial pitch	10	8	6	5	4	3	2½	2
H.P.	¼ - ¾	1-2	3-5	7.5	10	15-25	50-60	75-100

This investigation was supplemented by work done on the strength of various plastic moulded material for hardness and strength/



FABRIC MOULDED CUT GEAR BLANKS.

FIGURE 19

strength, the number of which has considerably increased since this research began in 1929, but as the Thesis has become rather lengthy it was thought advisable to leave this for a future research.

In previous papers the author has thanked Professors A.L. Mellanby and Professor Wm. Kerr for permission to carry out these tests and here again he desires to thank them for guidance and encouragement during the many tests carried out. To the different manufacturers of fabrics supplied for these tests the author tenders his thanks, and also to the Governors of the Royal Technical College for the freedom to use the power and plant erected in the College Laboratories of the Mechanical Engineering Department.

APPENDIX (application of data to road and rail)

The importance of Braking Systems can be well understood when the problem is considered of arresting as much as 50,000,000 ft.lb. of energy at 90 m.p.h. in the space of 40 sec. (or about ½ mile). The value of μ varies widely with speed, decreasing rapidly at the higher speeds where it is most needed.

(1) Percentage Braking $= \dfrac{\text{road or rail}}{\text{shoe}} = \dfrac{\mu_r}{\mu_s}$

varies for dry or wet surface of rail, and on surface material of road (concrete, macadam, asphalt) and also on these being dry or wet.

Friction on track $F \leqq \mu_r$ (normal load) $\leqq \mu_r N = 0.3\ N$ (steel rail)

This is approximately $= \mu_s$ Brake applied pressure $= \mu_s B$.

Equivalent rail or road friction $\mu_e \times N = F = \mu_s B$

so that $F = \mu_e N < \mu_r N$ or $\mu_e < \mu_r$ to prevent slippage.

$$\mu_s = \mu_r \left(\frac{1 + 0.014\ V}{1 + 0.075\ V}\right)$$

This/

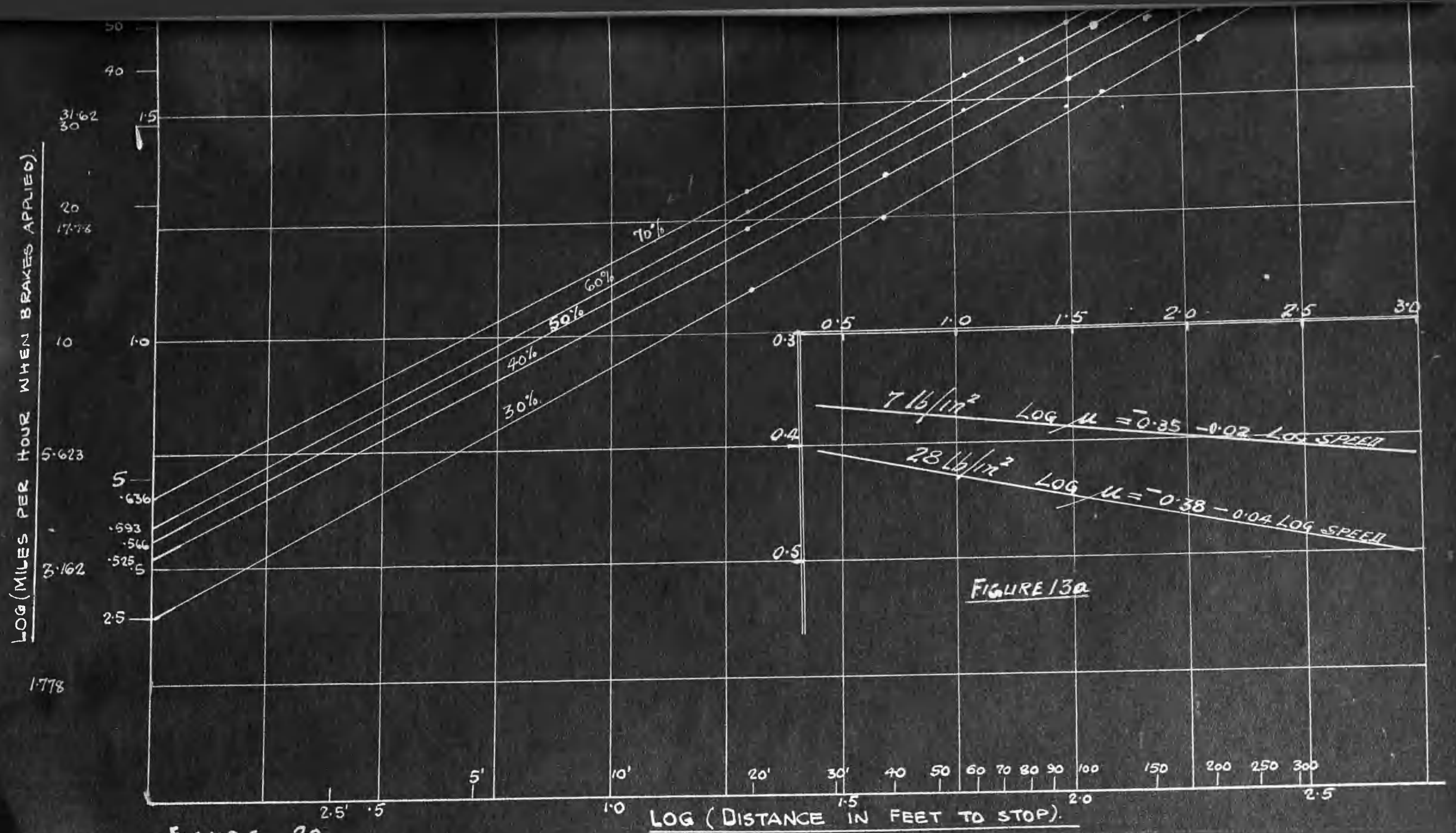

LOG (MILES PER HOUR WHEN BRAKES APPLIED).
50
40
31·62
30
1·5
20
17·78
10
1·0
5·623
5
·636
·593
·566
·525
·5
3·162
2·5
1·778
70%
60%
50%
40%
30%
2·5'
·5
5'
1·0
10'
20'
1·5
30'
40
50
60
70
80
90
2·0
100
150
200
250
300
2·5
LOG (DISTANCE IN FEET TO STOP).
0·5
1·0
1·5
2·0
2·5
3·0
0·3
0·4
0·5
7 lb/in²
LOG μ = ‾0·35 − 0·02 LOG SPEED
28 lb/in²
LOG μ = ‾0·38 − 0·04 LOG SPEED
FIGURE 13a

FIGURE 20

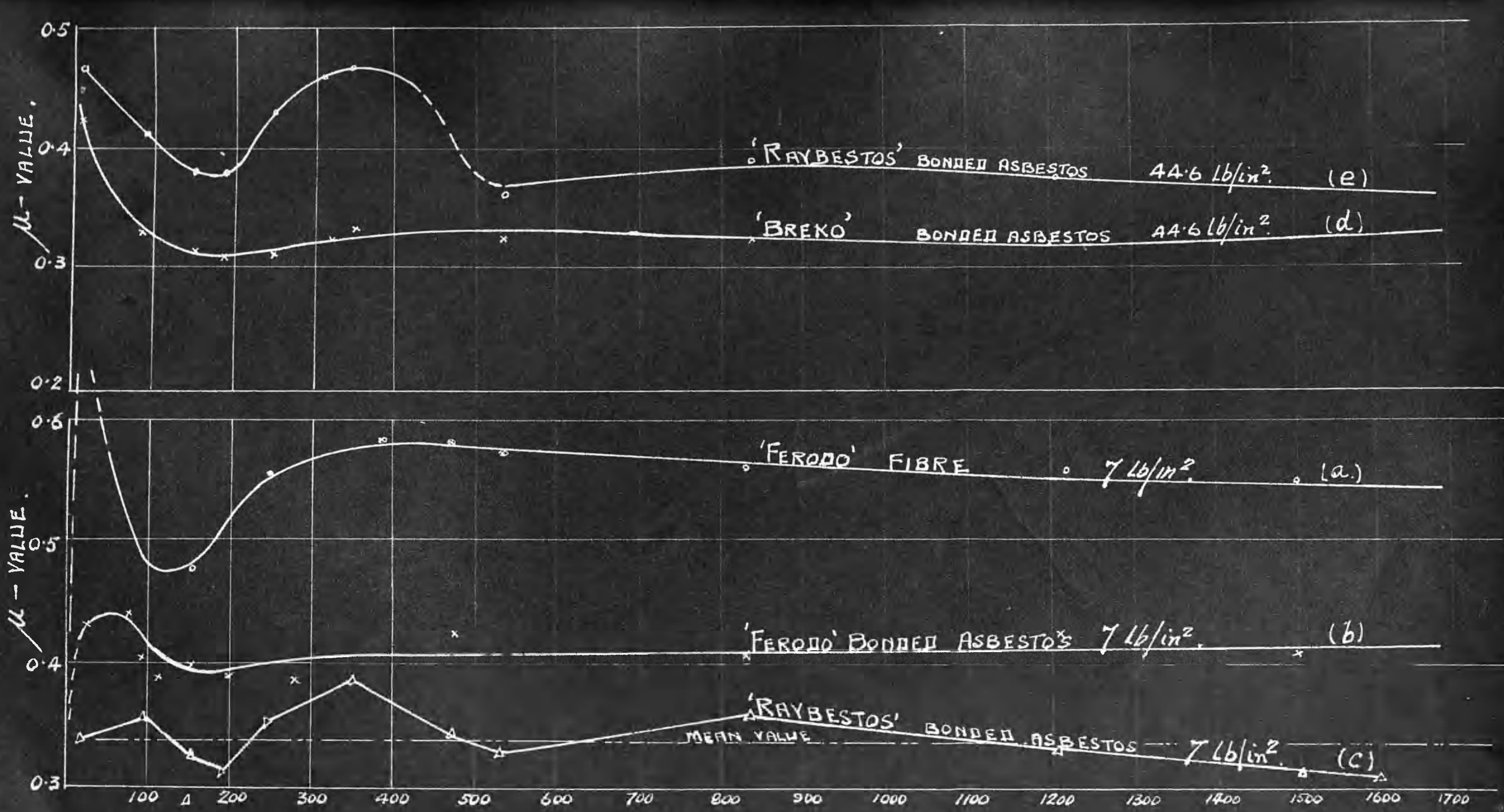


FIGURE 9 — a.b.c.d.e.

Time	Spring Balance Pull, lb.	Temperature	value. μ
11.55 a.m.	9.2	49°F.	0.6
12	9.4		
12.15 p.m.	9.35		
12.30	9.25	65°F.	
12.45	9.15		
1	9.15	70°F.	0.58
1.15	9.15		
1.30	9.15	75°F.	
1.45	9.15		
2	9.15	75°F.	
2.15	9.10		
2.30	9.05		
2.35	9.05	76°F.	0.56

Load changed to 36.65 lb. μ

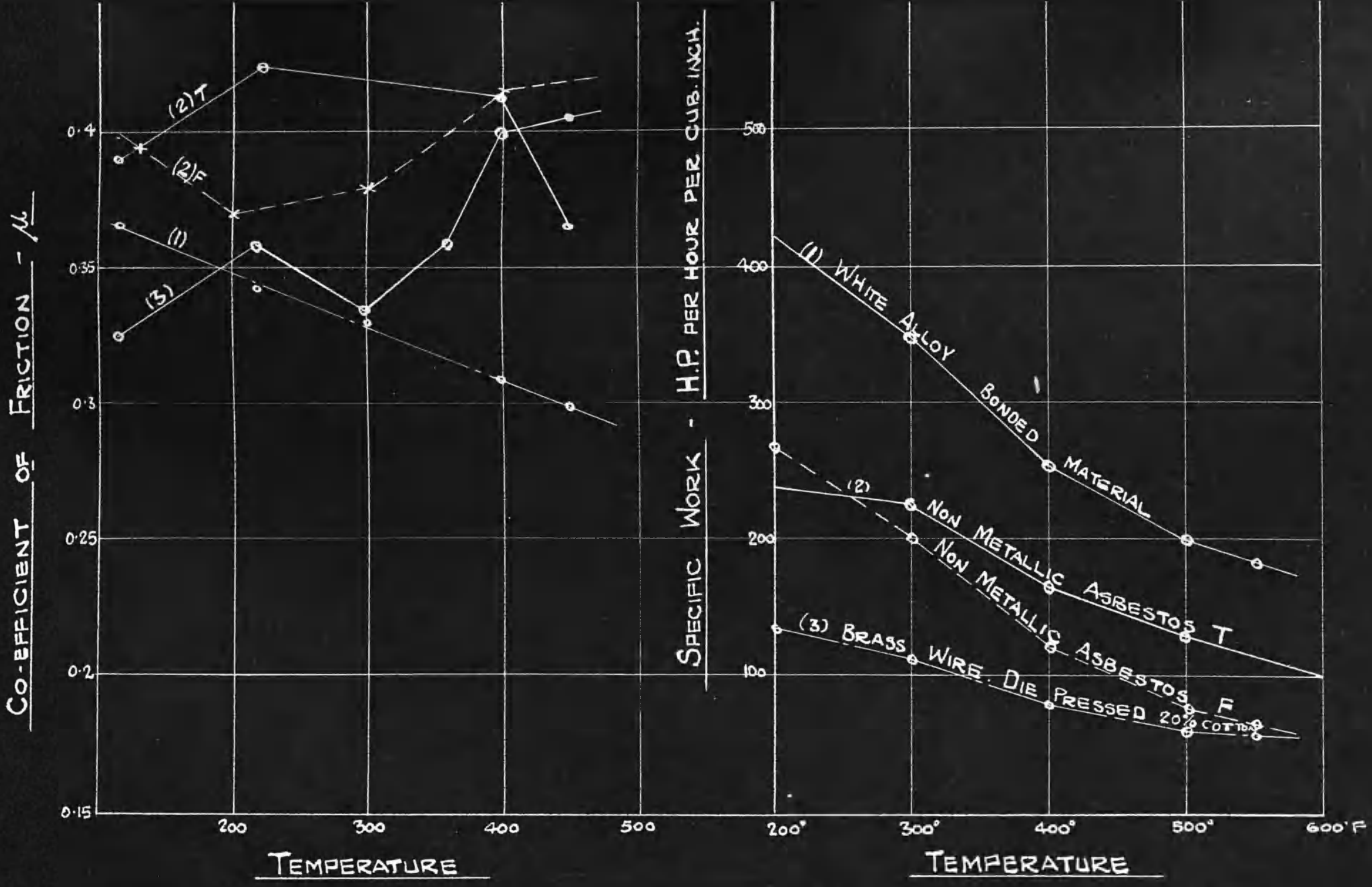
2.50 p.m.	12.45	75°F.	0.53
3	12.55	80°F.	
3.10	12.58	85°F.	0.55
3.40	12.55	95°F.	
4	12.55	100°F.	0.54
4.20	12.5	105°F.	
4.40	12.45	110°F.	0.53
4.50	12.45	110°F.	
5	12.45	110°F.	0.54

Room Temperature 52°F. Micrometer (1.9093 - 1.9089) = 0.004 inches.

When the 'Chekko' material is to be supplied for clutches the makers prescribe a specific loading 12 to 20 lb/in^2. In order to carry out temperature tests, taking two different loads and speeds, four bunsen flames were arranged round the interior of the rim supplied the heat. It was afterwards found that these required to be augmented by the aid of two blow lamps, and the brake wheel had to be shielded from the cold draughts as much as possible.

'Chekko' Bonded Asbestos. Micro. readings, 1.9207.

Time	Temperature of Rim °F.	Coefficient of Friction, μ	Remarks Dia. 26"
2.45 p.m.	110	0.58	Surface dry, Speed 1360 ft/min.
2.51	196	0.56	Specific load, $21\frac{3}{8}$ lb/in^2
3	238	0.53	Wheel partly enclosed
3.20 p.m.	242	0.52	
4.10	163	0.70	Speed 587 ft./min.
4.18	70	0.647	Specific Load 7 lb/in^2 Wheel not enclosed.



Wear in terms of Horse Power Hour per unit volume

FIGURE ~~10~~ 11

The excessive wear during the first period of wear-tests is more noticeable with the woven material than with the highly pressed (under temperature) moulded material. The latter can be machined and does not require much running-in, since the bearing surface is more regular than with the more soft woven and impregnated material.

From the average values of a large number of tests on woven fabric pure asbestos the curves in Fig. 9 have been plotted. These was excessive wear at first, which gradually dropped down to an almost stationary value of wear; the curves show that the rise in temperature is almost constant; while the value of the coefficient of friction falls uniformly over the first four tests, 24 minutes, and then remains almost constant over a long period test. The period of time between each test was a half minute, and the friction wheel was protected from cold currents.

Between room temperature and 300°F. the wear appears to be directly in proportion to the amount of work dissipated at the brake wheel rim. Above 300°F. the wear is excessive and grows rapidly for all Woven Bonded Fabrics; it is in this region that fabrics containing cotton become charred. The wear for nearly all fabrics is twice as great at 520° as at 260°F. for similar speeds.

Effect of Change of Pressure and Speed, but pv = constant.

During the tests with constant temperature regulated by the bunsen gas flame, the amount of wear was unchanged by adding load and decreasing speed, or decreasing load and increasing speed, to give equal works (p.v.). With a fabric material having a low μ-value it is found better to work with/

with a low p-value, say 10 to 12 lb/in^2 maximum and a high rubbing speed V. With the high coefficient of friction fabrics a pressure of 30 lb/in^2 is found to be more suitable with lower rubbing speed. Tests carried out on a drop-hammer ferodo brake lifting tup showed that, when 50,000 ft. lb. per sq.in. per minute was demanded from lining, the material wore out very rapidly with air cooled brake rim pulley, but with water cooled brake pulley the wear was not excessive. All tests carried out on the various fabrics show that it is not advisable to design for more than 12,500 ft.lb. per sq.in. per minute. A graph showing specific work, H.P.hr. per sq.in. plotted on temperature °F. is given at Fig. 10.

Effect of Rapid Heating on μ-value.

A new brake lining having a μ-value = 0.48 at 350 F. is applied to the brake wheel which when running has the temperature raised to nearly 350°F. before applying the brake. After the brake is applied the temperature rises rapidly to 575°F., time approximately 10 minutes. The μ-value of the material is reduced to 0.18 and may be taken on average as 0.2 for 20 minutes before disintegration of lining material takes place. The same material at 425°F. has μ-value = 0.4 but when raised to 575°F. in 10 minutes has still a μ-value = 0.35, and never goes under μ = 0.3. In both cases the μ-value increases as disintegration of lining takes place caused by the binding material becoming plastic. It was also found that the more slowly a brake lining is heated up the less likely is the bonding material to leave the fabric. The graph, Fig, 11, shows how the coefficient of friction varies with temperature in a number of fabrics. Taking an average of five different makers' linings, the maximum T-value is about 250°F. and the minimum 400°F.

Effect/

Effect of Roughening Surface of Brake Rim. -

Some cotton and duck fabrics have been tested which were made with a sulphur greasy compound incorporated while under pressure. It was found that oxidization took place, and smoothness of action soon disappeared. Where cast iron drums have been roughened by a file it has been observed that the fabric lining suffers very severely, and after a relatively short time, it is unfit for use. An iron cement was applied to a small pulley rim and Ferodo Fibre was used as a brake. Under the application of a very light pressure wear was excessive and heat generated rapidly, showing that in brake linings abrasion material would be detrimental.

Effect of Applying Water or Lubricants to Brake Wheel.-

In this set of experiments, "Raybestos", bonded asbestos brake lining is used. The speed is kept (approx.) constant at 1000 ft. per minute, V, and with a pressure of 45 lb/in^2 the μ - value is 0.38. Raising the temperature of the brake rim is followed by a drop in the value of the coefficient of friction. Removing the bunsen burners and applying water to the surface of the brake rim lowers the μ - value considerably and as long as the water is allowed to wet the brake surface of the wheel this coefficient becomes smaller. On removing the water and again applying heat the μ-values rises as the moisture dries off and is noted to be greater than original starting value. A series of these changes are shown, Fig. 12, in order to depict the various changes; shocks or shudders are seen on the brake wheel's motion.

When moist conditions prevail during tests the following changes are noticeable:

(1) that the high rubbing value of the friction fabric materials change when working on a water lubricated surface ;

(2) that μ - values for all materials are lowered by one third dry test values;

(3)/

(3) That a thin coating of rust or iron-oxide has been found to lower the μ-value by one half.

The pad pieces, in one test, after being fitted to the curvature of the brake drum were allowed to soak in water for three days. The fabric material was dryed on blotting paper and placed in the holder. It only required a second or two to reach the maximum μ-value yet there certainly was a distinct lag in reaching μ-value found in previous tests. The lining was not destroyed by the moisture and it did not appear to affect the lining after the first test run which pointed to the fact that the surface moisture, rather than penetration of water into the bonded fabric, was the cause of the temporary change in the value of the friction coefficient.

To prevent oil or water drip reaching the rubbing surface, rubber coating in the form of a solution has been tried on the brake lining leading edge. i.e. the first edge opposing the motion of wheel. Experiments on pads with rubber solution on edges were not carried to any length, but showed clearly that fluid could be guided past the brake surface by this simple contrivance.

Effect of Lubricant on Fabric Material.

Apart from the lowering of the μ-value, lubrication did not destroy in any way the die pressed fabric material, which, freed from oil, soon regained its normal coefficient of friction value. On heating up the brake wheel when tallow had been applied to the brake-pad surfaces, very erratic gripping was shown on the wheel's motion. The passing from lubrication film to boundary lubrication and then to dry condition could clearly be seen. It was not intended in this investigation to try out the different lubricating oils tested in the author's second investigation, but as a matter of interest a set of brake pads was sparingly treated/

treated with Bayonne oil, castor oil, and tallow. The mineral oil was soon rubbed off and the coefficient of friction of the fabric changed rapidly from 0.1 to 0.35. The tallow appeared to last longer as a lubricating medium with a coefficient of 0.15 rising to 0.35. The castor oil did appear to affect the lining but in rather a peculiar way as it gave a friction coefficient reading of nearly 0.2 and at one time this value reached 0.41 showing that this oil had acted as a dressing; on running for some time the value fell to 0.35 which was the dry coefficient value of the fabric undergoing test.

On the friction machine experiments have been carried out with dry and tallow coated cotton or duck belting. As this is a fabric, which when impregnated with rubber, has been used for brake linings, a few readings are given; fuller tests on this subject from a different point of view have been made at the Royal Technical College by Laird [7]. This paper deals with "Viscous effects in Dynamometer Belts", and the author in reading over this paper for publication purposes soon discovered that Laird's investigations pointed to the fact that the following formula

$$T_1 = T_e\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right), \text{ where } T_e = T_1 - T_2$$

effective tension, was not easy of manipulation, since the difference in tensions depends on belt creep and slip, arc of contact between belt and pulley face, character of pulley surface and belt surface, also velocity of belt. In the dry cotton belt tests a slight formation of oxide on the brake rim gave variable readings; the humidity of the atmosphere, or the least trace of moisture in the room caused variations in the readings. On the introduction of another variable, namely a tallow coating on the cotton brake band, made $T_1 - T_2$ still more variable, and the usual belt/

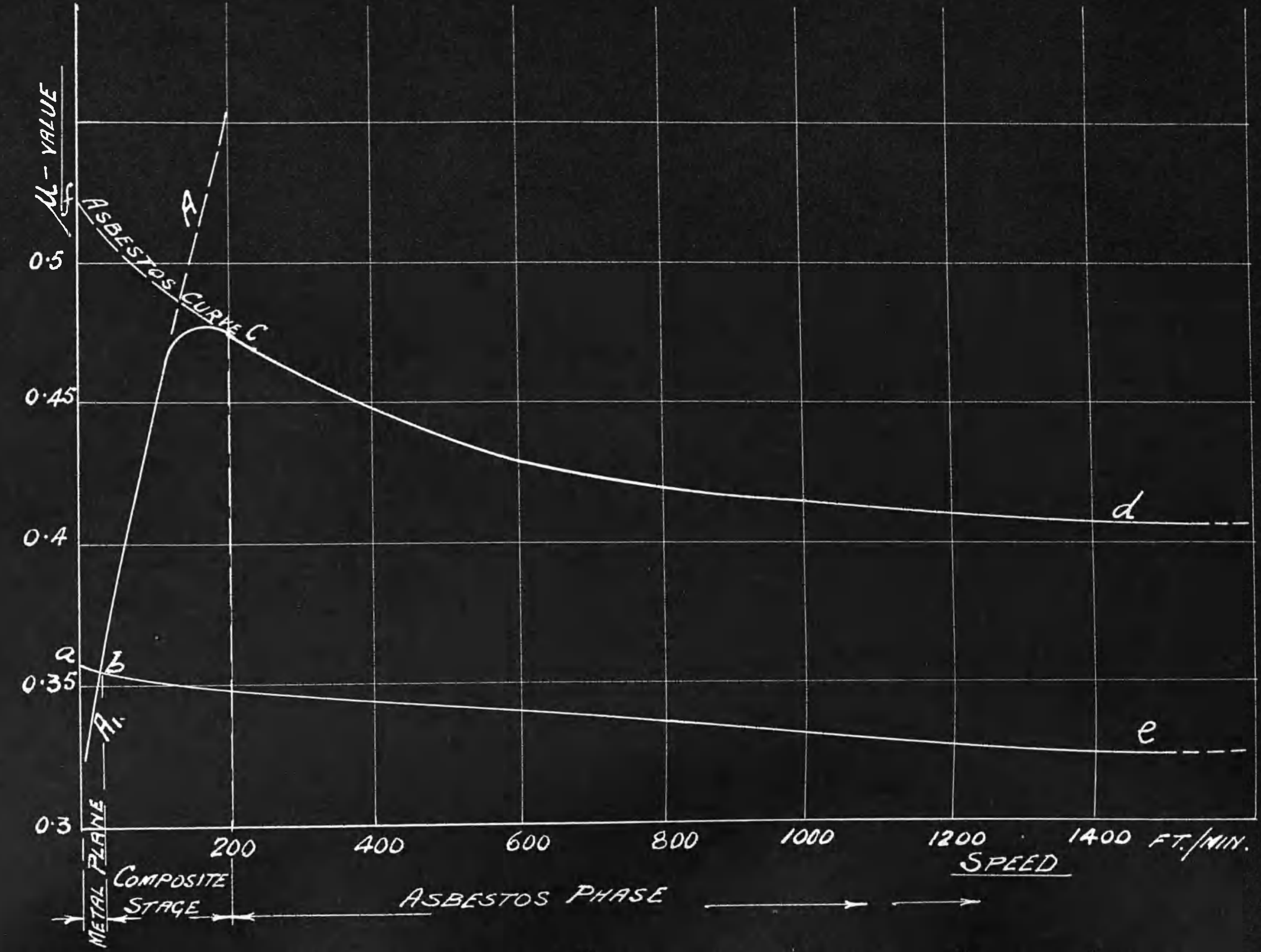


FIGURE THREE PHASE CURVE FOR BONDED FABRIC LINING.

Fig. 13.

belt formula $\frac{T_1}{T_2} = e^{\mu\theta}$ was not applicable as the test results show. From the curve plotted from experimental data Laird gave the formula $T_1 - T_2 = K(\lambda A V W)^{\frac{1}{2}}$ where λ = coefficient of viscosity; A is the area of film in shear or area of contact between the belt and the wheel; V is the speed of slipping; and W is calculated from end tensions as $\left(\frac{T_1 + T_2}{2}\right)\theta$.

The resemblance between the formula given in the author's investigation of lubricants submitted as part of this thesis is very noticeable.

(1) Laird's $(T_1 - T_2) = K(\lambda A V W)^{\frac{1}{2}}$

(2) author's $\mu = K\left(\frac{\lambda V}{2P}\right)^{\frac{1}{2}}$ lubricated bearing.

(3) Boswall's $\mu = K\left(\frac{ZN}{P}\right)^{\cdot 58}$ for clearance bearings.

(4) Dry belting $(T_1 - T_2) = e^{\mu\theta}$

If the testing of lubricated cotton belting entails the consideration of film lubrication with all its inherent difficulties. Fabric Brake Linings, which are made from bonded asbestos (plus in some cases 10 to 30 per cent of cotton), brass wire (which does not always have the same bearing surface), and bonding material (with varying viscosity), added to these the possibility of oil or water lubrication on brake wheel rim, have given to this investigation a most difficult number of variations and complicated combinations of variables.

Considering the curve, Fig. 13 which has been taken from a bonded asbestos lining, there are three distinct phases in this curve which may be denoted by the (1) Brass or Metal plane, (2) Composition Stage, (3) Asbestos Plane; this is assuming no cotton mixture in fabric.

The portion f.d. represents the curve of asbestos alone.

The portion a.e. represents the curve of brass or metal alone, where a,b,c,d is the actual curve obtained for the material. The first portion of the curve, namely line AA appears/

appears to be determined by the intensity of the loading, being near to the axis the higher the loading becomes. The metal curve shows little change with increase of speed of rubbing, but in the "composite stage" the μ-value increases rapidly with the increase of speed. The asbestos part of curve decreases with the increase of speed, very rapidly at first and then becomes more nearly constant.

(1) for metal μ = 0.3 to 0.38 average 0.34 depending on the metal used.

(2) composite material (metal, asbestos plus bonding material) μ increases at the rate of 0.15 per 100 ft. per minute.

(3) for asbestos $\mu = 0.43 \div \text{speed}^{0.03}$, this being an average derived from curves with 7 lb and 28 lb per sq.in. pressure loading; the equations for the curves, Fig. 13 a. on Fig 20 being

$$\text{(a)} \quad \log \mu = -0.35 - 0.02 \log \text{speed}.$$

$$\text{(b)} \quad \log \mu = -0.38 - 0.04 \log \text{speed}.$$

Using as a lubricant machine oil changes the coefficient in the phase (1) from 0.3 to 0.1 in four minutes, and the asbestos when lubricated still lowers this value and at the end of another period of five minutes the μ-value has become 0.02. In the case of pressed block linings the lowest value reached was 0.08 which is almost equal to metal on metal. Curves, Fig. 14 [p.45] have been drawn to illustrate the changes which take place in temperature and μ-values when lubricant is used on the Brake Wheel.

CONSIDERATION OF RESULTS ON BRAKE LININGS

From the foregoing experimental results given in this investigation it will be seen that a wide variation of results is due to the variable nature of the combined materials and their method of manufacture. The Asbestos Fibres are washed with mineral acids in order to make the fibres more absorbent, and also to prevent co-agulation of rubber while they are being impregnated with latex. The percentage mixture is usually/

usually 80 asbestos to 20 rubber; the latex usually forms 30 per cent. solid content, and vegetable fibres may replace asbestos in part. Sulphur may also be added. This in the 'Investigation on Lubricants' has been placed among the materials useful for carrying away heat from an over-heated journal bearing. The mixture of materials, Asbestos, Rubber, Sulphur, etc. is made into a dough; this in the case of the Asbestos Bonded Brake Linings is forced into the meshes of the wire fabric, and put into moulds in the case of moulded linings without metal. After being allowed to set at a moderate heat, 150°F., in a vacuum drier, the slab or mould is compressed to about 1/3 of its initial thickness. The slabs and the moulds are kept heated while under hydraulic pressure, and this tends towards constancy of finished material. The temperature of the moulds vary from 275 to 400°F. and the hydraulic load applied may be such as to give 50 tons.

(1) These linings have a hardness of Rochwell No.B. 95 and can be machined. Thus when a brake strap arrangement is properly designed to give uniform pressure, the lining may be made to fit closely on to the brake wheel. A fault found in carrying out experiments with fabric material made in 1926 was the lack of smooth surface.

(2) The discarding of the metal binder is another disadvantage brought out by this investigation owing to quantity of metal surface exposed to rubbing wear not being uniform.

(3) The material as now manufactured is not destroyed by exposure to water or oil, but this does not mean that an oiled surface can be used for a heavy load brake or friction clutch. The fabric material behaves in the same manner as all lubricated metal surfaces subjected to running speed and pressure/

pressure. The asbestos composition can be used for clutches designed for working in oil. This is the material used in combined clutch and change-speed lever gear.

(4) Temperature effects are such that with many built up fabrics excessive wear takes place at high temperatures. The temperature, with the newer hot-pressed linings, may now reach 500°F. without destruction.

(5) Wear, which was measured by change of thickness in a given time, or by change of weight over a given period of rubbing, has been measured by absorption of energy per unit of volume (one cubic centimetre) and expressed in horse-power hours per cubic centimetre or ft.lb. per min. per sq.in. Fig 15

This method of measuring wear could be stated as an index of the pefformance of brake linings, and could be made a basis of comparison, horse power hours per cubic centimetre, but must not be taken when making a comparison between service brakes fitted to machines where excessive pressure may be applied intermittently and for short periods. The brakes are never allowed to cool but are used at peak temperatures where wear is excessive.

(6) The nature of the surface of the brake rim is a deciding factor. In the main tests a cast iron brake drum was used, but moulded type of brake lining has been proved more suitable to use with low carbon steel pressed drums than bonded asbestos. Metal, introduced to strengthen the yarn, causes seizure to take place which is followed by a fluctuating frictional value which results in 'snatch' or 'shudder' ahd brake drum scoring.

At slow crawl speeds 'snatch' and 'shudder' is very noticeable in the case of high μ-value fabric material. While the low μ-value material such as Breko, with its low steady frictional value of 0.28 to 0.25, provides a smooth and fairly/

WEAR CURVES.

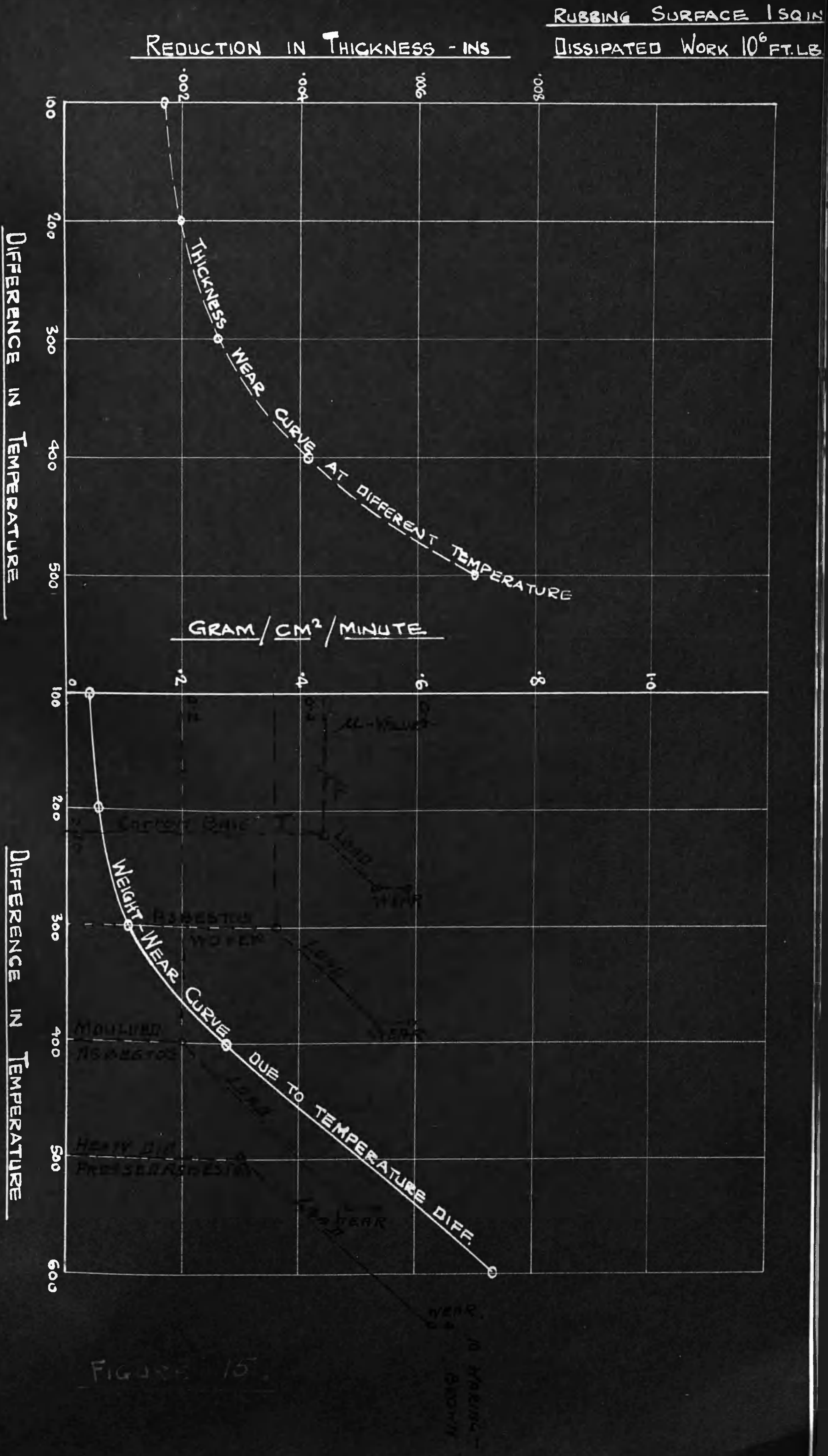
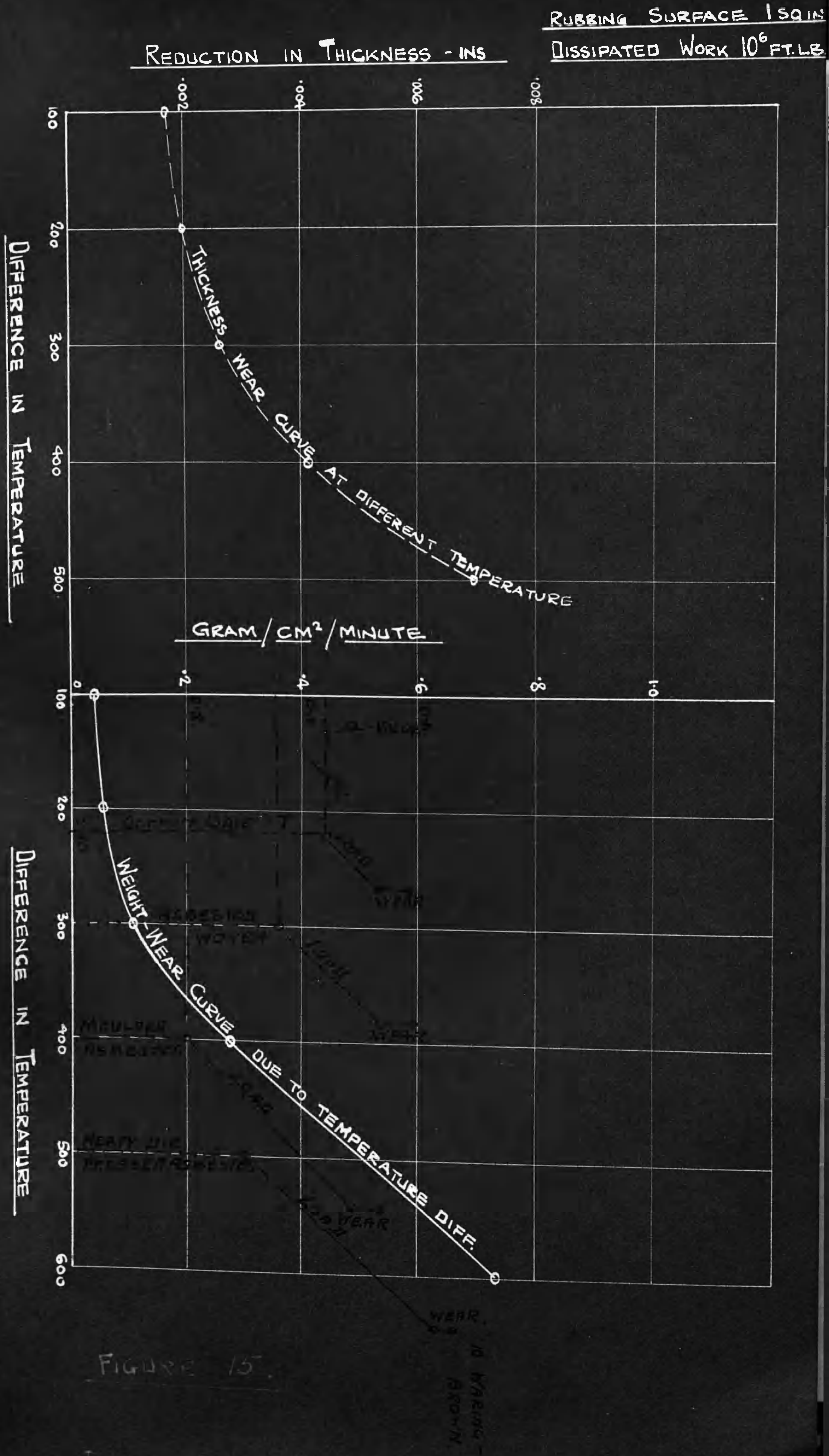


FIGURE 15.



WEAR CURVES.

FIGURE 15.

fairly rapid retardation. This class of material, used with a low bearing pressure, was found to be good for electric lifting cranes and hoisting machinery, a fact confirmed by tests carried out by Messrs. Adamson at their crane works.

CONCLUSIONS ON BRAKE LININGS

This investigation had for its aim the comparison of fabric linings as manufactured, a grouping together of test results as required by the engineer, an encouragement to standardization of tests, and quality of material used. Since the first results of these tests[6] were published 1930, manufacturers have improved their running tests, and guided by results obtained by these, and other experimenters' results on durability when consideration is taken of duty imposed they have gone on creating new and better materials suited for carrying heavier loads, increasing speeds, and stopping powers when suddenly applied to moving machinery.

Much of the data given in this investigation are applicable to Friction Clutches and Friction Spur and bevel wheels. The only difference is in the range of pressures applied in clutches and wheels which may vary from 25 to 75 lb. per sq.in., whereas in brake linings the pressure on an average is about 15 lb. per sq.in. with a maximum of 30 lb. per sq.in. Even a lower pressure 12 lb. per sq.in. has been used by crane makers, and suggested by test results obtained in this investigation. .The brake drums and brake shoes used in practice, even when made of pressings are not flexible. The brake linings in all the tests described are "worn in", that is, in applying the brake all points must touch the drum at the same moment. In applying the brake the lining is being compressed, the compression being in proportion to the pressure which is exerted on the lining (Hooke's Law), or the sum of all friction forces $\Sigma F = \mu \Sigma N$.

The higher the self-actuation the more sensitive is a brake to change in friction coefficient. Therefore, if too high/

high a degree of self-actuation is used in a brake design, the brake is liable to "grab" in some cases, while it will not hold sufficiently in others. That is why it is much more important to obtain a brake lining which is affected as little as possible by temperature, moisture, speed, etc. than to obtain a high frictional coefficient. There appears to be no way of judging the self-actuation of a brake shoe. Different designs show different features and makers of brake linings are now inclined to specific brake linings to be used with certain designs.

Fabric Material used for Gearing.

When gear wheels are made from fabrics, such as 'Ioco' and 'Fabroil', consideration must be given to the Tensile Strength of the material, to the hardness of the fabric on the root surface of the tooth and to the wearing surface of the material. The duty of wheel teeth is to transmit power with least possible loss. The efficiency of the gearing depends on lubrication, which is quite different from Fabric Brake Lining which seek to use the dry rubbing surfaces, sliding friction, without seeking to use seizure action which would destroy the surfaces. In gear teeth the action is part rolling and part sliding friction with line contact.

Comparison is made between material used for wheel blanks.

(1) Fabroil pinions which are made from compressed cotton (specially prepared cotton fibres). This material takes the place of compressed paper and wood pulp blanks. In this case there is no binder used and compressor therefore has to be maintained by means of steel side plates, or shrouds; if these are removed the Fabroil is useless. In this class raw-hide pinions and pinions made from Ioco sheet may be classed.

The nature of these fabrics made testing on the Friction/

Friction Machine difficult as the cutting up into ½-in. squares to fit the fabric holder, Fig. 3 was almost impossible. Test carried out on this material showed that the dry friction was rather erratic, varying from static friction 0.25 to running friction 0.6, and again falling rapidly to 0.25 as heat brought out the dressing. When lubricant was applied the μ- value dropped by about $\frac{2}{3}$ giving a μ-value of 0.16 to 0.2. (Tallow lubricant). Castor oil dressing when applied to the friction pads was inclined to cause a viscous drag.

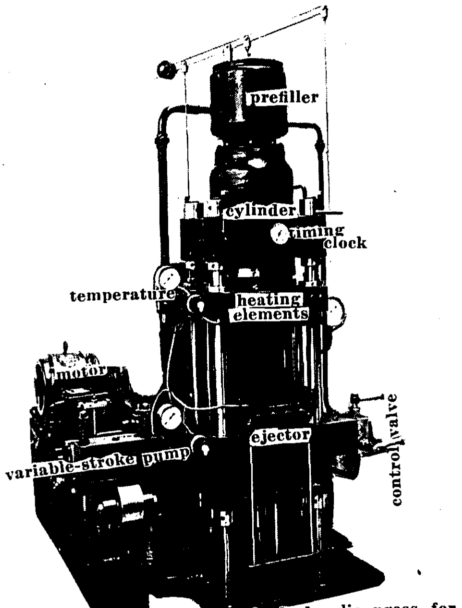
(2) Fabroil-A cut very easily into ½-in. squares and when run on the Friction Machine gave smooth running and taking on a beautiful polished surface which worked well with cast iron. The lubricated friction tests gave a value for μ = 0.1 to 0.06. A high pressure could be applied to this material, and oil did not tend to destroy its structure even when running against the grain of the texture, yet it does not appear to have the strength of the shrouded pinion.

(3) Ferodo bonded asbestos could not be used for gearing but for light duty Ferodo fibre could be used.

(4) Ioco is made in much the same manner as Fabroil the only difference being that in a semi-plastic state the woven cotton duck strip is impregnated with synthetic resin and compressed in steam heated moulds under pressure. It is finished as it comes from the moulds except for slight trimming at the parting line in gear blanks, and curing for about 20 hours at 212°F. Arrangement of Press is shown in Fig. 16.

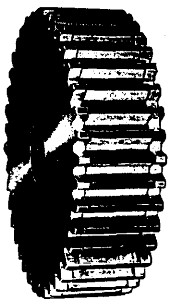
(5) Cotton cloth impregnated with Bakelite is pressed into heated moulds to make turning gears for oil pumps. Duck cloth has usually a tensile strength of 24 to 30 lb. per inch width per ply.

Table/



A 50-ton semi-automatic hydraulic press for plastic moulding, with motor-driven variable-stroke automatic pump.

Figure 16



Gear Wheel
Built up
and Shrouded

Rockwell Tests on
10 Blanks
Max - 86 B. } average 71
Min - 69 B }

Figure 17

Table of comparative strength and hardness.

Material	Ultimate Tensile Strength	Izod ft lb.	Hardness Max.	Min.
Fabroil-A $\frac{7}{8}$" thick	4 tons/in^2	2	48.8	44.4 Brinell
Formapex ½" thick Micarta	Tensile 5½ tons/in^2 7 tons/in^2 (Shear) 2 tons/in^2	3	36 (45 (99	34 " 41 " 93 Rockwell B
Ferodo Fibre ½" thick	2 tons/in^2	1	98	Rockwell B
Formapex Gear Wheel Blank	3 tons/in^2 at boss and rim 5 tons at web.	1.5 2.5	78 82	77 Rockwell B. 73.5 "
Vulcanized Fibre	7.5 tons/in^2	3.5	75	70 Rockwell B

The Young's Modulus for all fabrics is approximately 1×10^6. The blanks can be machined by using tools with large top rake; circular milling hobs with steep top rake are used for cutting teeth in blanks, and grinding may be done preferably by a rapidly rotating fine-grained emery wheel, (e.g. No. 40 emery, 5 inch diam, 3000 r.p.m.), taking care to avoid undue heating of the fabric. These features were clearly brought out during the machining of the various test pieces.

Cross-Breaking Strength or Maximum Fibre Stress.

For most practical purposes this is defined as the load, P, required to fracture, by bending, a bar-shaped test-piece of given rectangular cross section (b = breadth, d = depth), when this is supported or held horizontally and the load applied vertically in one of the following forms of test:-

(a) Cantilever Test: The bar is held by a clamp at one end and load applied at a distance, L, from the near edge of the clamp. Stress = $\frac{6PL}{bd^2}$. Ioco, 0.5" x .75" x 2.5". Safe load, 25 lb.

(b)/

(b) 3-point Loading test: The bar is supported at two points a given distance, L, apart, and the load applied midway between them. Stress = $3PL/2bd^2$, Fabroil-A = 0.5" x .75" x 2.5", Safe Load, 95 lb.

Impact Strength

(c) This is expressed as energy per unit area of cross section of the test piece where fracture occurs, but this test varies with the shape of the cross section. This gave approximately 2 to 3 ft.lb. for all fabric material.

Plastic Yield Test - (d): Test as in (b) at a given temperature (200°F.) for a given time 9 minutes, or 80°F. for a period not less than 12 hours. One of the Fabroil-A pieces was turned to ½" diameter at ends and a plain portion ⅛" diameter for 2" long. This test piece was subjected to torsion and gave on an average 10000 lb/in^2 (max. 13000 lb/in^2, min. 9000 lb/in^2).

Absorption Tests - The Vulcanized Fibre, although like the other fibres tested was insoluble in ordinary solvents, was seen to absorb water (swells). In this respect Ioco and Fabroil materials are better than Vulcanized Fibre, wood, raw-hide, etc.

Formapex Miocarta and Fabroil-A gears and blanks are sent out from factory with metal centres which are permanently held in position without rivets. Sketches of a blank and a cut gear are shown in Fig. 17 . These can be made much stronger and lighter than when cut from slabs.

From the formula $HP = \frac{0.0001 \times V \times f \times b \times y}{D.P}$

derived for Ioco material, let breadth of face of wheel b = 1 inch and the diametrical pitch D.P. = 1, then the formula for horse power becomes $HP = .0001 \times V \times f \times y$ per inch of face per 1 diametrical pitch. Assume pressure angle 14½°, then $\log HP = \log 0.0001 + \log V + \log f + \log y$. To make an alignment chart to suit this logarithmic formula of/

of four variables, let log r = log V + log f.

If V varies from 100 to 2000 ft. per sec. then log 20000 - log 100 = 1.301. magnify this by 5, say 6.505"

The safe stress of the material for different speed may vary from 1000 to 4000.

Then log 4000 - log 1000 = 0.602. magnify by 10, say 6.02".

The form factor will vary from 0.075 to 0.118.

log 0.118 - log 0.075 = 0.2. magnify by 1/30 say 6"

If the distance between scale V, (S_V), and scale f, (S_f), = 3"

then scale of r will be distant from V by e"

$$\text{when } e = \frac{S_f}{S_V + S_f} \times 3 = \frac{\frac{1}{10}}{\frac{1}{5} + \frac{1}{10}} \times 3 = \frac{\frac{3}{10}}{\frac{3}{10}} = 1"$$

and scale of r, (Sr), = $\frac{3}{10}$

Again place scale y at distance 3" from scale r,

$$\text{then } e = \frac{S_y}{S_r + S_y} \times 3 = \frac{\frac{1}{30}}{\frac{3}{10} + \frac{1}{30}} \times 3 = \frac{3}{10} = 0.3$$

Then scale for H.P. (S_{HP}) = 1/3

$$\text{Substitute in formula } HP = 0.0001 \times \overset{V}{2000} \times \overset{f}{1000} \times .118$$

$$= 23.6$$

$$\text{and } HP = 0.0001 \times 100 \times 4000 \times .075$$

$$= 3.$$

The chart has now been drawn, Fig.18 from this data and gives any of the four variables when the others are known. This chart has been improved by adding the scale for the pitch diameter of the wheels, the number of teeth and the diametrial pitch. The formula which gives these dimensions and numbers are

(1) $\frac{\text{(diam. of gear) r.p.m.}}{12}$ = Velocity in ft.per min.

$= 0.26\ d \times \text{r.p.m.} = V$

(2) circumferential pitch $p = \frac{\pi}{D.P.}$ and $np = \pi d$

Therefore $n = d \times D.P.$

From/

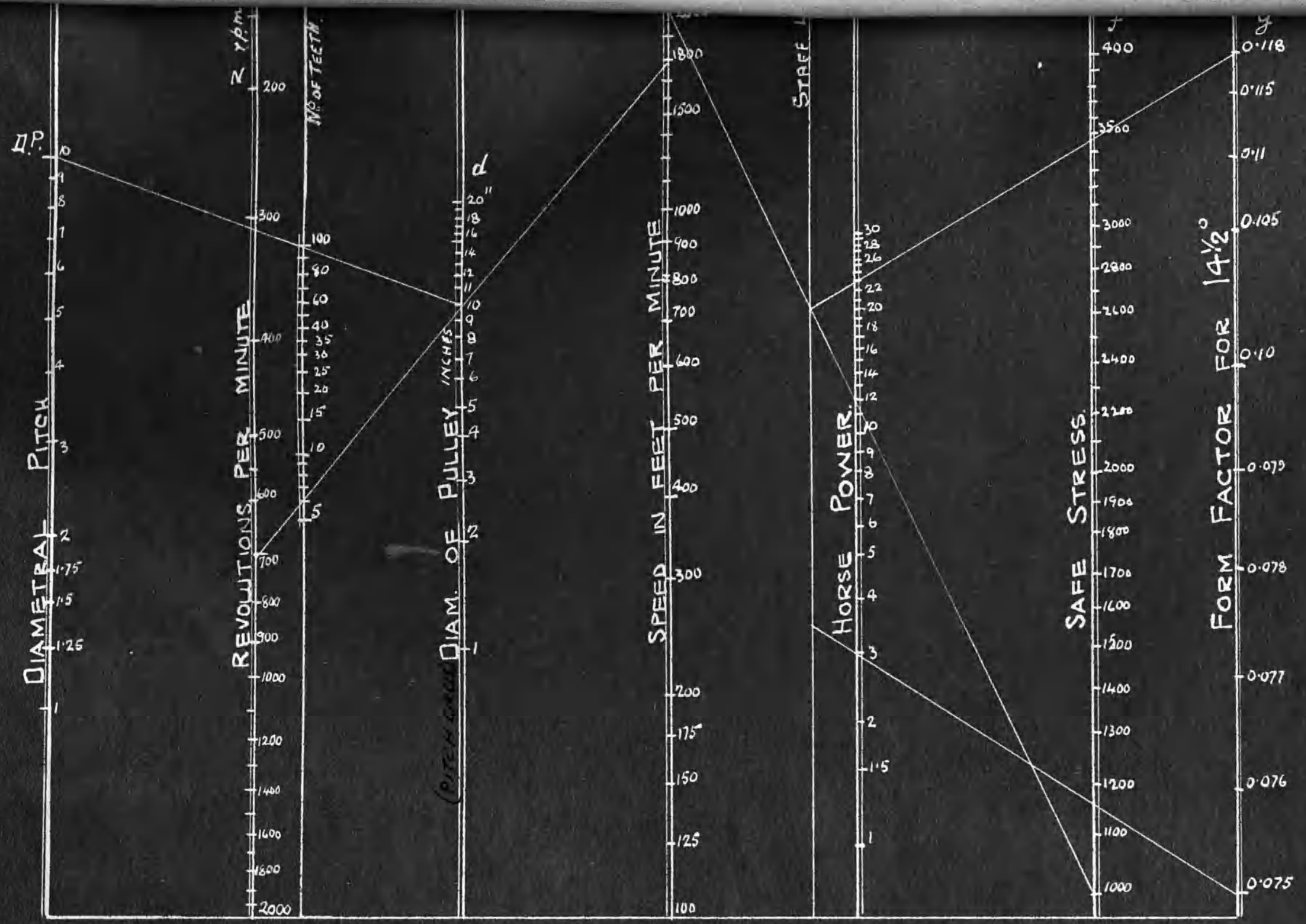


FIGURE 18 — NOMOGRAM FOR "FORMAPEX" MIOCARTA (BAKELITE) GEAR WHEEL —

From (1) $\log d = \log V - \log .26 - \log N$

From (2) $\log D.P. = \log n - \log d$

By the same method the scales N and D.P. have been added. These could have been drawn separately but by the addition of key-lines the scales are clearly connected.

The author was asked by Professor Mellanby to furnish the manufacturers of Ioco Fabric with a formula to suit their material, and this has been added as a matter of interest, and is a practical illustration of the ease with which nomograms can be adopted to test results. An experimental apparatus was also designed to test the endurance of fabric gearing when running in conjunction with cast iron wheels.

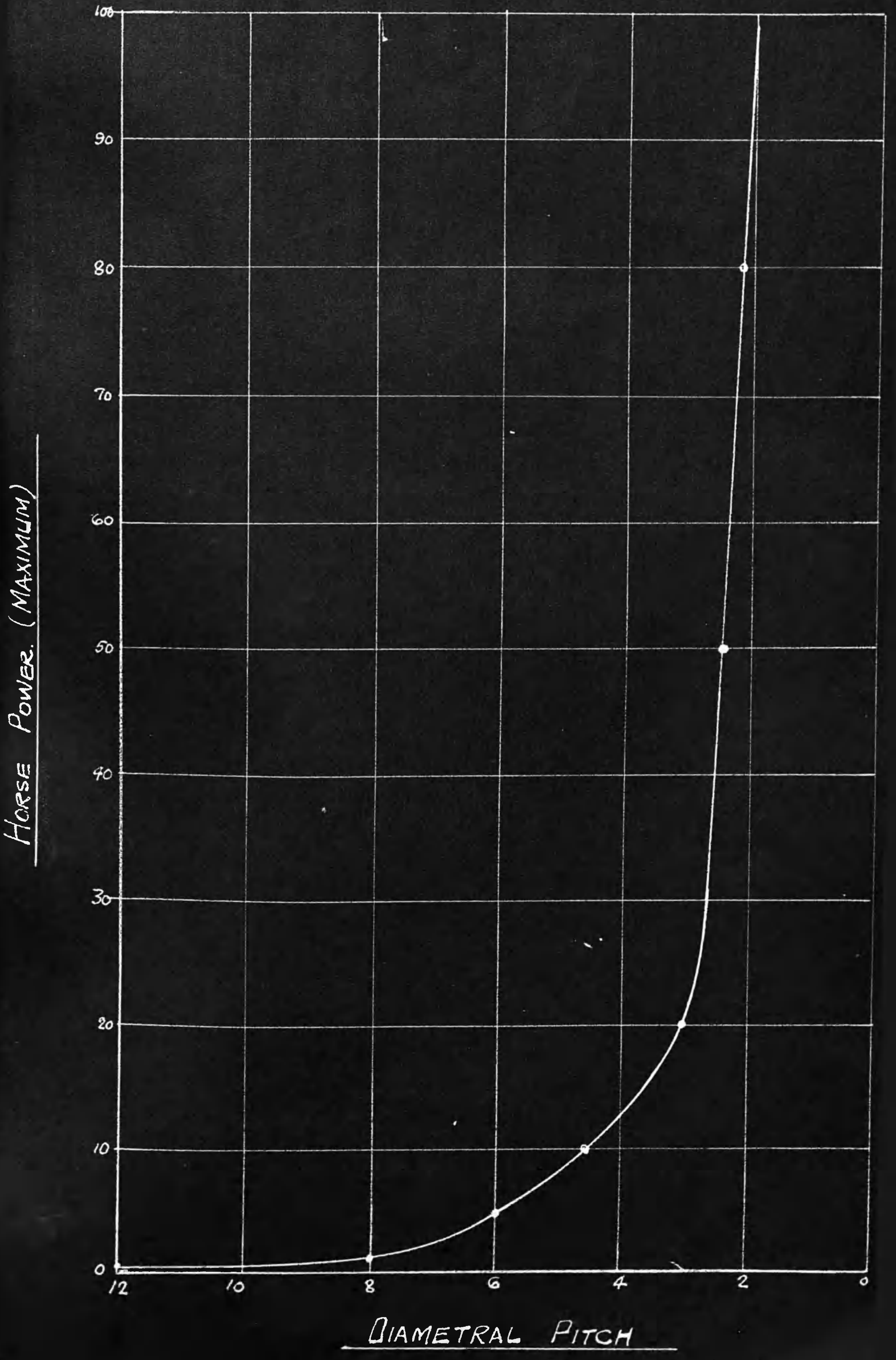
For the first run-in of fabric gearing it is advisable to use a solid lubricant such as a mixture of vaseline and graphite, or colloidal graphite. A thin coating of this lubricant during the run-in period helps to make a polished smooth surface, afterwards the gears may be run in a bath of good lubricating oil.

The fabrics might be made to suit the chuck of a reversal of bending stress machine, and could be subjected to a sliding motion abrasion as a speedier method of testing the gear blank pressed fabric material. This would certainly give a very much shorter test than was recommended to the Ioco Company.

From the alignment chart the table and graph horse powers for various diametrical pitch have been drawn up for easy application, Fig 19.

Diametrial pitch	10	8	6	5	4	3	2½	2
H.P.	¼ - ¾	1-2	3-5	7.5	10	15-25	50-60	75-100

This investigation was supplemented by work done on the strength of various plastic moulded material for hardness and strength/



FABRIC MOULDED CUT GEAR BLANKS.

FIGURE 19

strength, the number of which has considerably increased since this research began in 1929, but as the Thesis has become rather lengthy it was thought advisable to leave this for a future research.

In previous papers the author has thanked Professors A.L. Mellanby and Professor Wm. Kerr for permission to carry out these tests and here again he desires to thank them for guidance and encourggement during the many tests carried out. To the different manufacturers of fabrics supplied for these tests the author tenders his thanks, and also to the Governors of the Royal Technical College for the freedom to use the power and plant erected in the College Laboratories of the Mechanical Engineering Department.

APPENDIX (application of data to road and rail)

The importance of Braking Systems can be well understood when the problem is considered of arresting as much as 50,000,000 ft.lb. of energy at 90 m.p.h. in the space of 40 sec. (or about ½ mile). The value of μ – varies widely with apeed, decreasing rapidly at the higher speeas where it is most needed.

load) $\leqq \mu_r$ N = 0.3 (steel r

(1) Percentage Braking $= \dfrac{\text{road or rail}}{\text{shoe}} = \dfrac{\mu_r}{\mu_s}$

varies for dry or wet surface of rail, and on surface material of road (concrete, macadam, asphalt) and also on these being dry or wet.

Friction on track F $\leqq \mu_r$ (normal load) $\leqq \mu_r$ N = 0.3 N (steel rail)

Friction on track F $\leqq \mu_r : \mu_s$ Brake applied pressure $= \mu_s B$.

Equivalent rail or road friction $\mu_e \times N = F = \mu_s B$

so that F $= \mu_e$ N $< \mu_r$N or $\mu_e < \mu_r$ to prevent slippage.

$$\mu_s = \mu_r \left(\frac{1 + 0.014\ V}{1 + 0.075\ V}\right)$$

This/

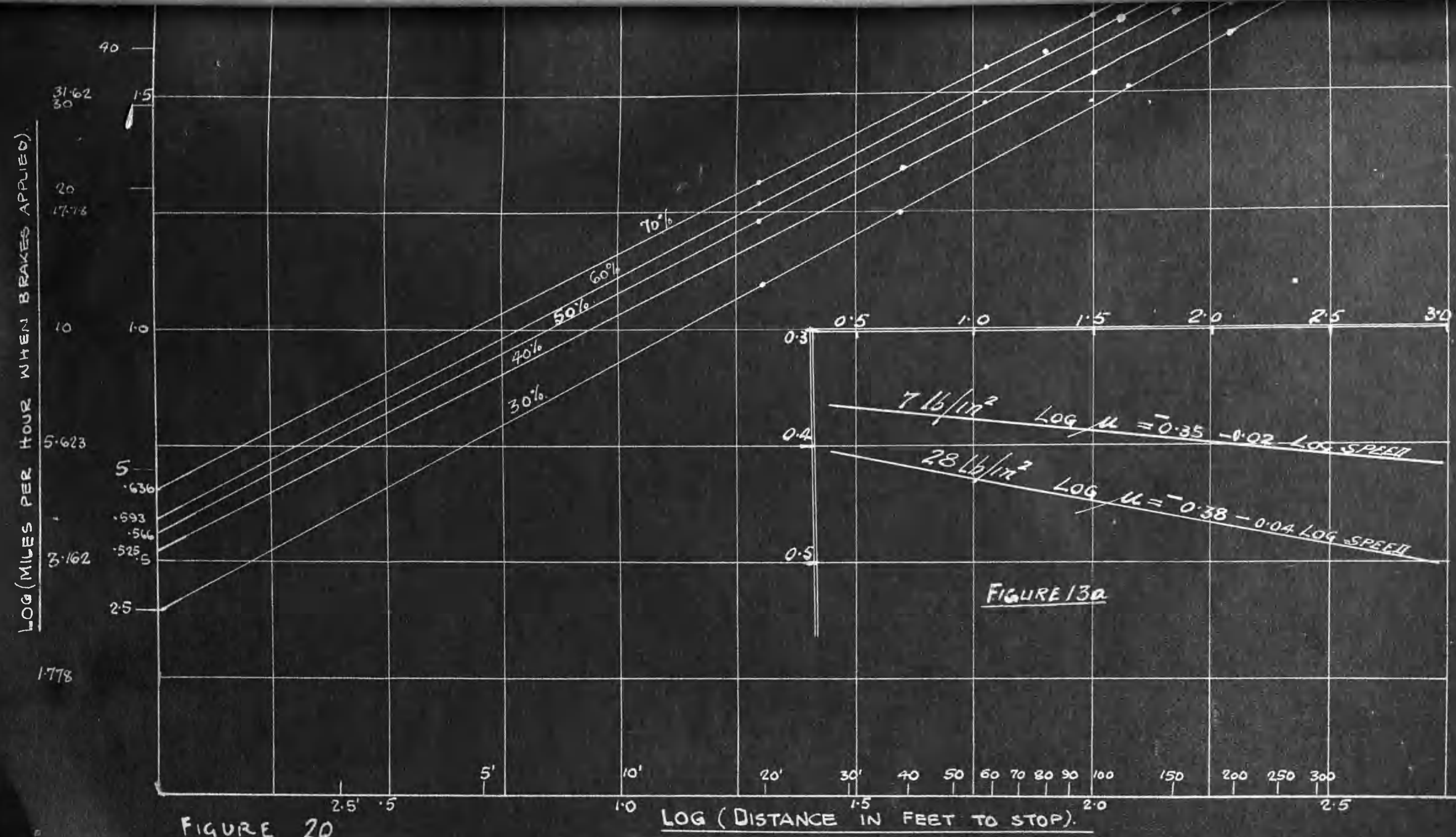

LOG (MILES PER HOUR WHEN BRAKES APPLIED).
40
31·62
30
1·5
20
17·78
10
1·0
5·623
5
·636
·593
·566
·525
·5
3·162
2·5
1·778
70%
60%
50%
40%
30%
0·5
1·0
1·5
2·0
2·5
3·0
0·3
0·4
0·5
7 lb/in²
LOG μ = $\bar{0}$·35 − 0·02 LOG SPEED
28 lb/in²
LOG μ = $\bar{0}$·38 − 0·04 LOG SPEED
FIGURE 13a
5'
10'
20'
30'
40
50
60
70
80
90
100
150
200
250
300
2·5'
·5
1·0
1·5
2·0
2·5
LOG (DISTANCE IN FEET TO STOP).

FIGURE 20

This would give for μ_s at 60 m.p.h. a value 0.096,
for μ_s at 30 m.p.h. a value 0.15,
and for μ_s at 20 m.p.h. a value 0.17
for ordinary steel rail.

(2) Brake efficiency testers as applied to motor cars.

In place of giving a relation between the coefficient of friction of the Brake Shoe and the Rail or Road Coefficient the Siemen's Brake Meter (a) and the Ferodo Brake Efficiency Indicator (a_1) give a relationship between the miles per hour of car's motion and the stopping distance in feet.

(a) is simply a pendulum in the form of a capillary tube (with Pitot tube end) containing a coloured fluid. The column shows the retardation (b) in yards per sec.per sec., and where (S) represents the braking distance in yards and (v) the speed of the vehicle in yards per sec.[8]

$$S = \frac{V^2}{2b}.$$

The brakes are applied evenly on a level dry road when motor is moving at say 20 miles per hour.

Derived from $V^2 = u^2 - 2fs$ and $v = u \pm ft$

(a_1) In this tester the speed of motor is 20 miles per hour when the brakes are applied. There are three dials, 20 per cent for a stop or draw up in 67 ft., 30 per cent for stop in 45 ft., and 50 per cent for a stop in 27 ft.[8]

TEST FIGURES FROM A CAR ON MACADAM ROAD, Fig.20.

Brake Efficiency per cent.	V = miles per hour S = stopping distance in feet	Practically
30	$\log V = .40 + .536 \log S$ or $V = 2.51S^{.536}$	$V^2 = 9S$
40	$= .525 + .506 \log S$ or $V = 3.31S^{.506}$	$= 12S$
50	$= .566 + .516 \log S$ or $V = 3.68S^{.516}$	$= 14.2S$
60	$= .593 + .515 \log S$ or $V = 3.92S^{.515}$	$= 18S$
70	$= .66 + .506 \log S$ or $V = 4.57S^{.506}$	$= 21S$

Brake Efficiency	30	40	50	60	70 per cent.
@ 30 m.p.h.					
Distance to draw up	100	75	63	50	43 feet.
@ 25 m.p.h.	70	52	44	35	30 feet.

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INVESTIGATION OF LUBRICANTS

AND

FRICTION BEARING METALS.

INDEX

INDEX (continued)

INVESTIGATION OF LUBRICANTS, AND FRICTION OF BEARING MATERIAL.

ABSTRACT

In this investigation the lubricating properties of various oils at low rubbing speeds are considered.
A comparison is made between pure vegetable, mineral and animal oils when used as lubricants, with cast iron, phosphor bronze, white metal, mild steel, and nickel-chrome heat treated steel, as bearing materials.
Some consideration is given to the blending of machine oil, lubricants for use in forging and drawing, and the effect of introduction of mercury and chlorine with lubrication mixtures. The finish of the bearing material is considered in the reduction of friction effect. Endurance tests are carried out on various lubricants at different temperatures, and the wear of bearing materials is noted, condition of surfaces are examined after failure of lubrication, or restriction of lubricant supply. The results of experiments have been made use of in Bearing design.

INTRODUCTION

(a) Some notes on the metals used in Tests.

Cast Iron and Mild Steel form the greatest portion of bearing metals; of the alloys, phosphor bronzes and White Metal are the chief. Crank shafts are made of special Steel in which nickel and chromium are employed to form a steel alloy. In this investigation the metals used are (1) Cast Iron on Cast Iron, on Phosphor Bronze, and on White Metal; Nickel Chrome Steel on Cast Iron, ob Bronze and on White Metal.

The approximate chemical analysis of the material used.

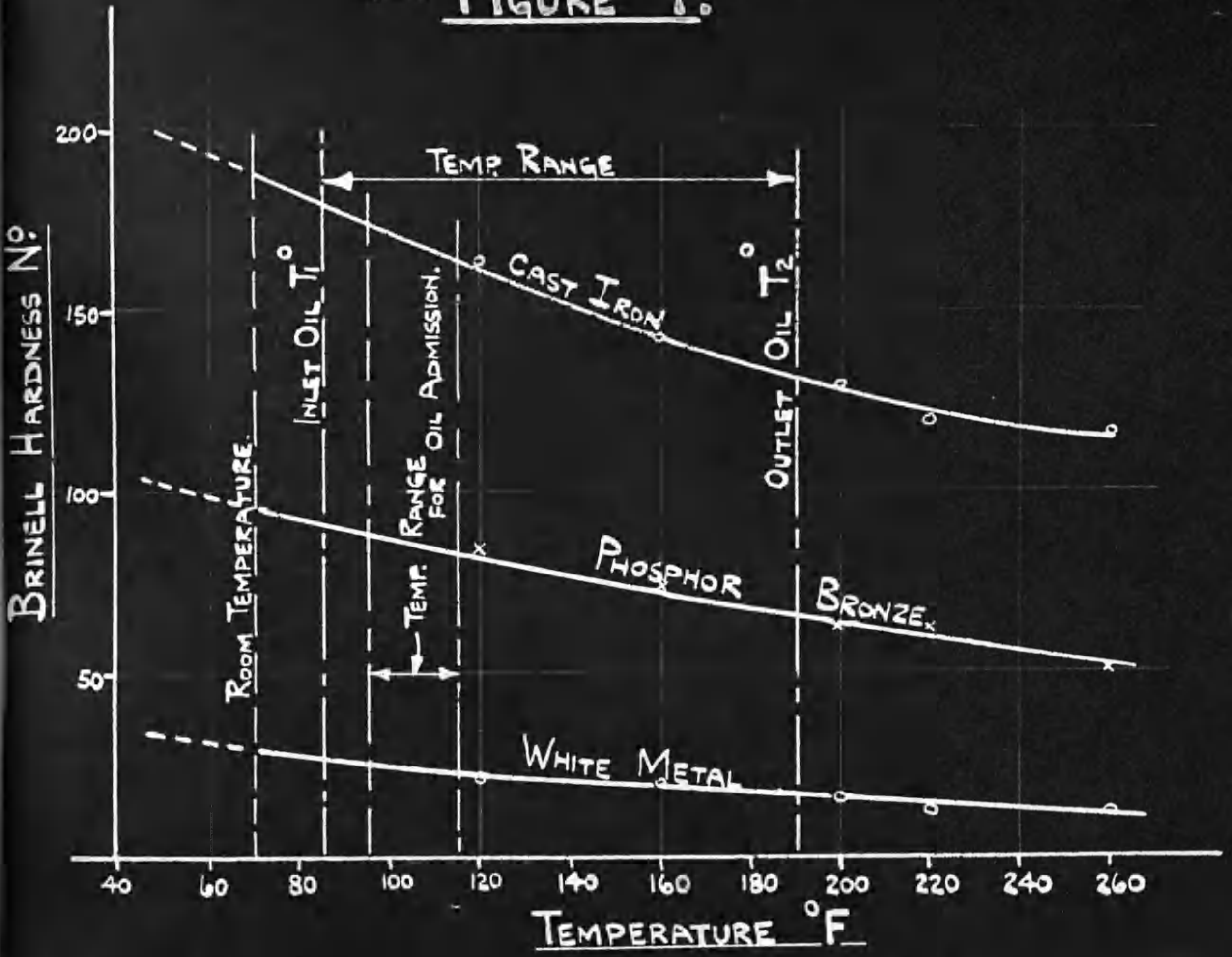
	C	Mn	Si	S	P	Ni	Cr	P_b	Sn	Cu	Sb
Cast Iron	3	0.6	1.3	0.05	0.2	3					
Nickel Chrome Steel	0.31	0.65	0.15	0.03	0.03	3.3	0.7				
Phosphor Bronze					0.4			10	9.5	80	
White Metal								1.5	78	7.5	13

The approximate physical properties.

	lb. per sq.in.			
	Tensile Stress.	Compressive	Fatigue ±limit.	Brinell Hardness. Ball = 10mm. Dia. Pressure = 3000 Kgr.
Cast Iron	3.2×10^4	12.8×10^4	1.4×10^4	190
Nickel Chrome Steel (oil hardened)	12.7×10^4		4.4×10^4	290
Bronze	7×10^4		1.5×10^4	93
White Metal	2.3×10^4			29 at 70°F. 12 at 212°F.

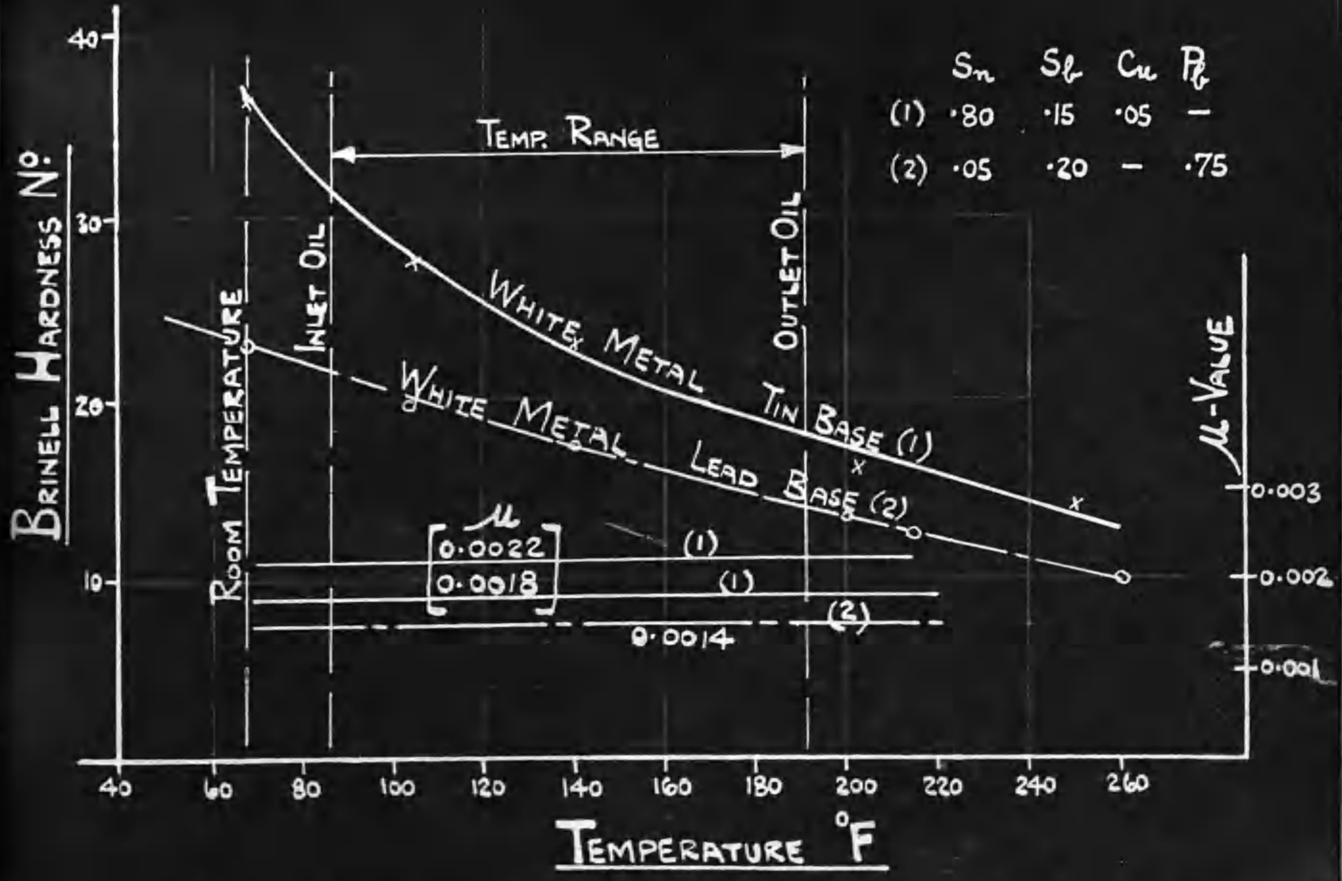
The cast iron, Phosphor Bronze, and White Metal discs were wound with an electric-heating element, and six hardness readings at temperatures ranging from room temperature 56°F. to 250°F. were taken. Brinell Nos. have been plotted on a base of Temperature. Fig.1. On this diagram there is shown the average temperature range for a number of bearings. The best temperature/

— FIGURE 1. —



BALL HARDNESS. TEST OF PLATES.

TIME OF LOADING ½ MINUTE.



BALL HARDNESS OF WHITE METAL TEST PLATES.

AND BEARING METAL

temperature inlet appears to be approximately 86°F. giving a safe range of 100°F. rise in oil temperature and therefore approximate bearing temperature. White Metal lining shows a reduction in Brinell hardness by 50 per cent. during the rise of 100°F. This change in Brinell number, along with heavy load which has given rise to the temperature change, assists the plastic flow of the White Metal.

(b) Some notes on the Lubricants used in Tests.

In a paper already published / on lubricating properties of various oils, data will be found concerning some of the lubricants used in this investigation. The following lubricants are considered, (1) three forms of machine oils are compared to show how the main features can be examined by means of simple apparatus, (2) Neatsfoot oil, Graphited Spindle oil plus 1 per cent. of oil dag, Diesel oil, and Lape oil, (3) Tallow (pure), Tallow 10 per cent White Lead, Tallow and Castor oil.

Engineers are accustomed to the use of "Specific Gravity" and realize that a unit volume of oil weighs more as the numerical value of the specific gravity increases, and $S\,\frac{60^{o}F}{60^{o}F}$ is the ratio arbitrarily chosen, or $S\,\frac{15.5^{o}C.}{15.5^{o}C.}$ The weight of one gallon of oil is ten times its specific gravity but this unit has no relation to the lubricating quality of an oil. However, the lower the specific gravity of the oil the easier will it be to free the oil from water by centrifugal means. The values of the specific gravity were obtained by determining the density of the oil.

50 c.cs of oil were carefully weighed at 60°F.
i.e. weight of 50 c.cs oil plus weight of specific gravity bottle = 63.52 gms.
weight of specific gravity bottle = 16.83 gms.
Therefore weight of 50 c.cs of oil = 46.69 gms
Therefore density at 60°F. = 0.934
and specific gravity = 0.934 [9.34 lb/gallon]

The/

The density at any other temperature was obtained from the formula $D = d - kt$ ....(1)

where D = density of oil at required temperature
d = " " " " 60°F.
t = number of degrees above 60°F.
k = a constant = .00035 for most lubricating oils.

If the specific gravity was required to very great accuracy a correction for the buoyancy would require to be made. This is not required in simple lubrication problems, but a curve of correction for temperature rise, which is to be added when the temperature is above 60°F. and substituted when the temperature is below 60°F. is given, Fig.2., for various specific gravity oils.

In this investigation the viscosity of the oils under test was obtained by use of a Standard Redwood Viscometer 50 m l and converted into C.G.S. units (Poises) by the formula given by Higgins.

$$\eta = \rho \left(0.0026\,T - \frac{1.715}{T}\right) \text{ poises } ....(2)$$

where η = absolute viscosity in C.G.S. units (poises)
T = No. of seconds for outflow of 5 c.cs.
ρ = Density of oil at temperature of test.

Hence from (1) and (2) the values of absolute viscosity are obtained.

Temp. °F.	Redwood Times Seconds	Density from 1 D	Viscosity from 2 (poises)	Rape Oil Units
60.5	3855	0.934	9.34	725
90	900	0.923	2.16	193
120	320	0.913	0.755	59
150	158	0.902	0.363	30
180	75	0.892	0.170	17

For the purpose of comparison a column showing "rape oil units" has been added to the table.

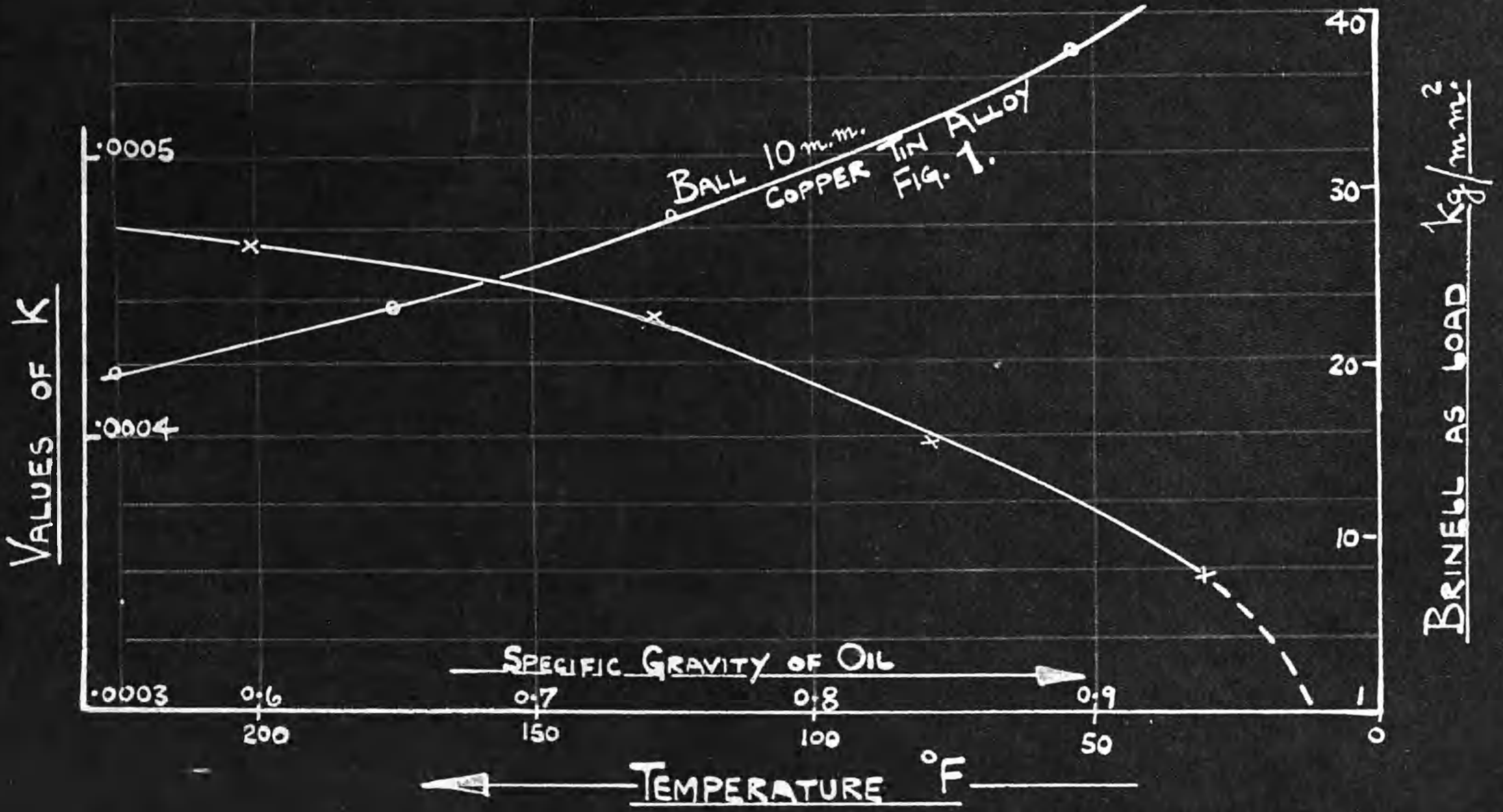
$$\text{Rape oil units} = \frac{T}{535} \times \frac{\text{sp.gr. of oil at } t^{\circ}\text{F.}}{\text{sp.gr. of refined rape oil at } 60^{\circ}\text{F}} \times 100$$

$$= \frac{T \times St^{\circ}F \times 100}{535 \times 0.915} = \frac{T \times St^{\circ}F.}{4.895}$$

535 and 0.915 are, respectively, the Redwood No.1. time of flow and specific gravity of refined rape oil at 60°F.

During/

FIGURE 6.



VALUES OF K FOR VARIOUS SPECIFIC GRAVITY.

During this investigation reference was made to viscosities given on the Saybolt and the Engler viscometer, and a blending chart is given in Fig.3. which shows the three viscosity scales in general use. The absolute centipoises (assuming Sp.Gravity 0.9) are also shown. The blending charts have been found necessary when an oil is to be made to specification drawn up from detailed examination of lubrication requirements.

(c) Some notes on the surface finish of the Bearing Metal.- The gun-metal bush was machined with smooth finish and test made on the Bearing Machine . The bearing surface was then hand-scraped and second test carried out. An expanding mandril finish was then put on metal and adjustment for clearance made; the clearance was kept as near 1/1000 as possible and a test carried out. The machine was then run for several hours as a "running in" process before the final test.

Much the same procedure was gone through in obtaining the different finishes on the Phosphor Bronze and White-Metal plates used in the Deeley Oil Testing Machine.

A white-metal bearing bush was made by fixing the plummer block on the shaft and casting the bush into the block. As a finish the bearing machine shaft was made to rotate slowly just before the white-metal had cooled. The bush was then removed and cut out to suit the oil rings similar to the gun-metal bush, Fig.4(a), as far as possible the same area was maintained. A sketch of lines of oil flow is shown Fig.4(b). The white-metal is then given a rub with a wire brush to clear off scale after which a test was carried out. This was American practice with long bearings of small diameter; for example, the main spindle of Pickering governors. A test was carried out with the metal as described and then a second test after eight hours running in. A section was taken off the material from a very smooth portion of the bush and a second at a part where evidently seizure had taken place, Figs. 5(a) and 5(b).

The/

FIGURE 3.

BLENDING DIAGRAM.

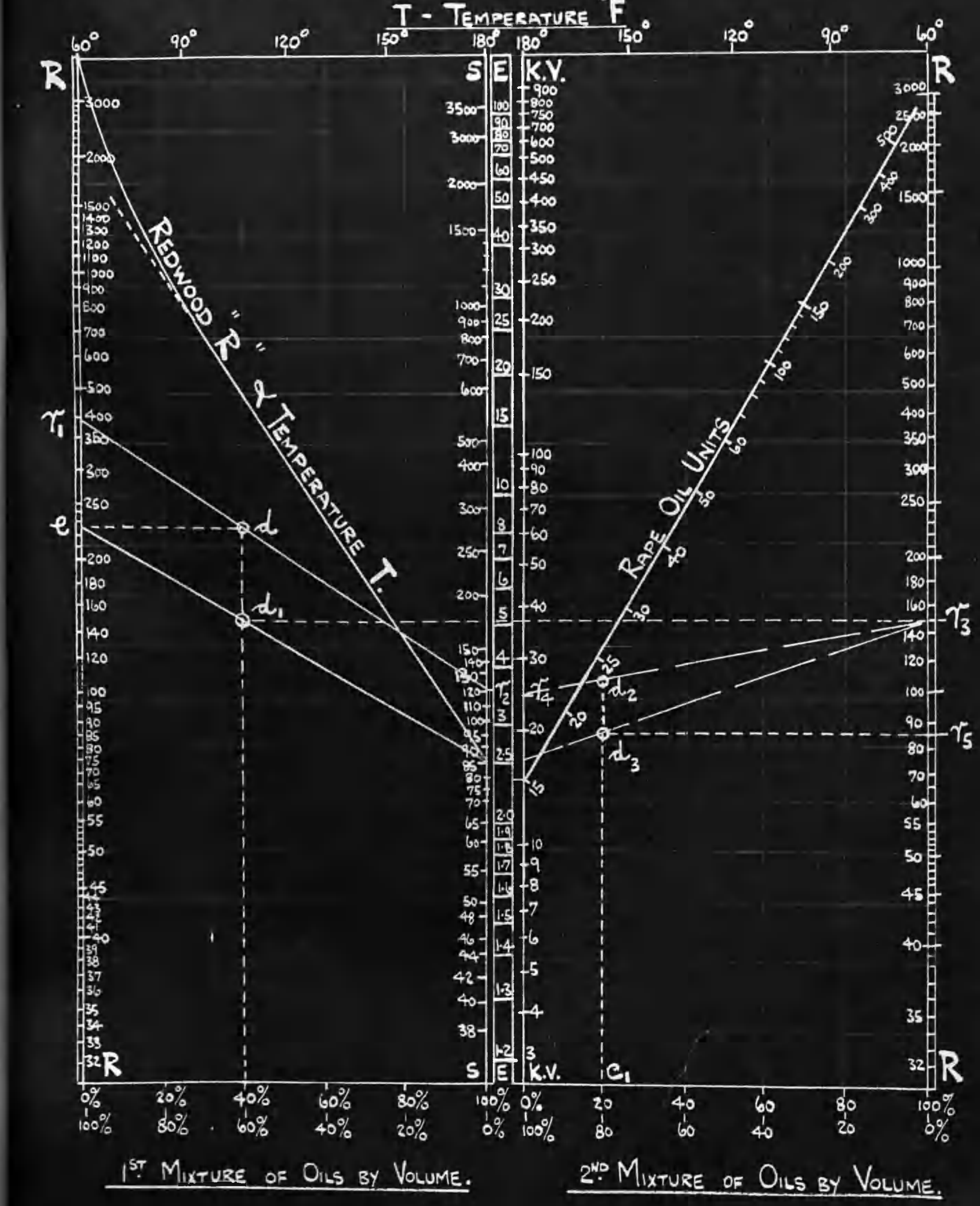
EXAMPLE:- $40\%\ R_{400} + 60\%\ R_{100}$ @ 108°F. GIVES R_{150}

r_1 to r_2 d_1 to r_3

BLEND OF 3-OILS. $80\%\ R_{150} + 20\%\ R_{100}$ @ 158°F. GIVES R_{84}

r_2 to r_4 d_3 to r_5

R = REDWOOD. S = SAYBOLT. E = ENGLER. K.V. = KILO VOLTS.



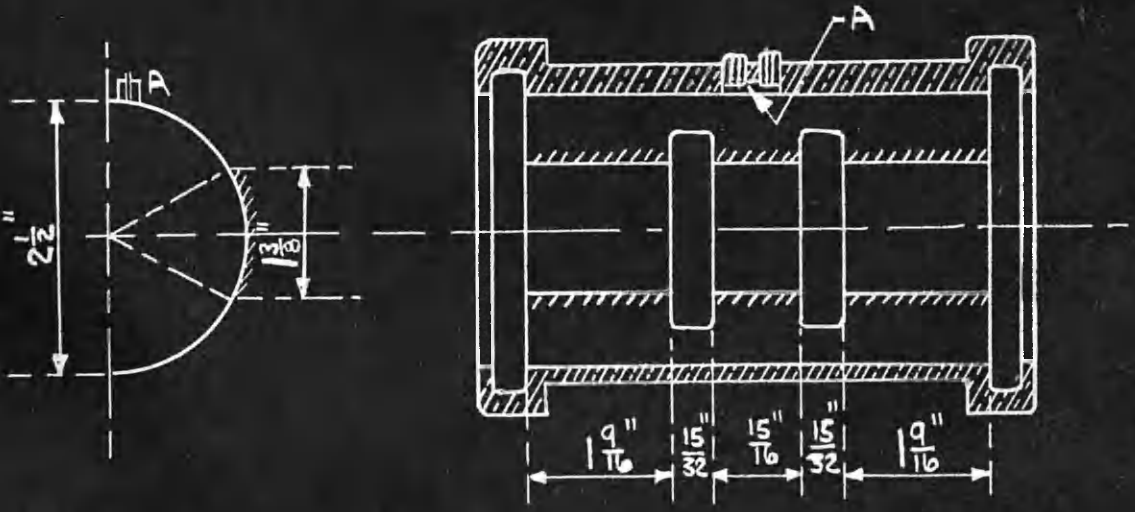


FIG. 4a. SURFACE BEARING DIMENSIONS.

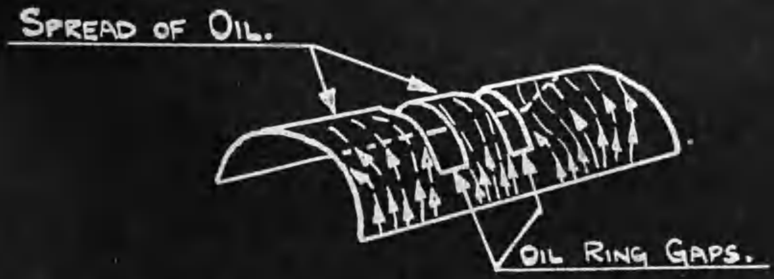


FIG. 4b. OIL STREAM FLOW.

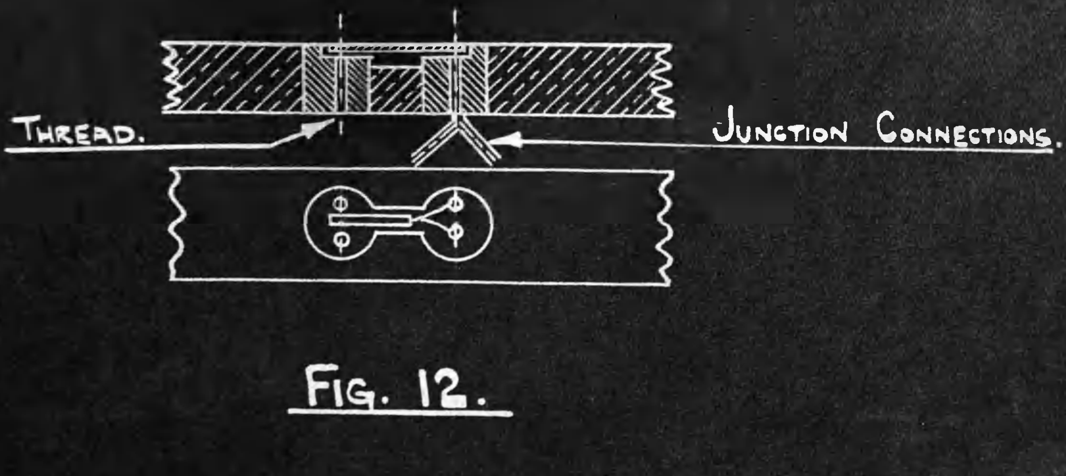
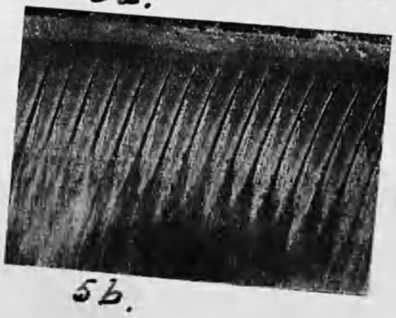


FIG. 12.

5a.

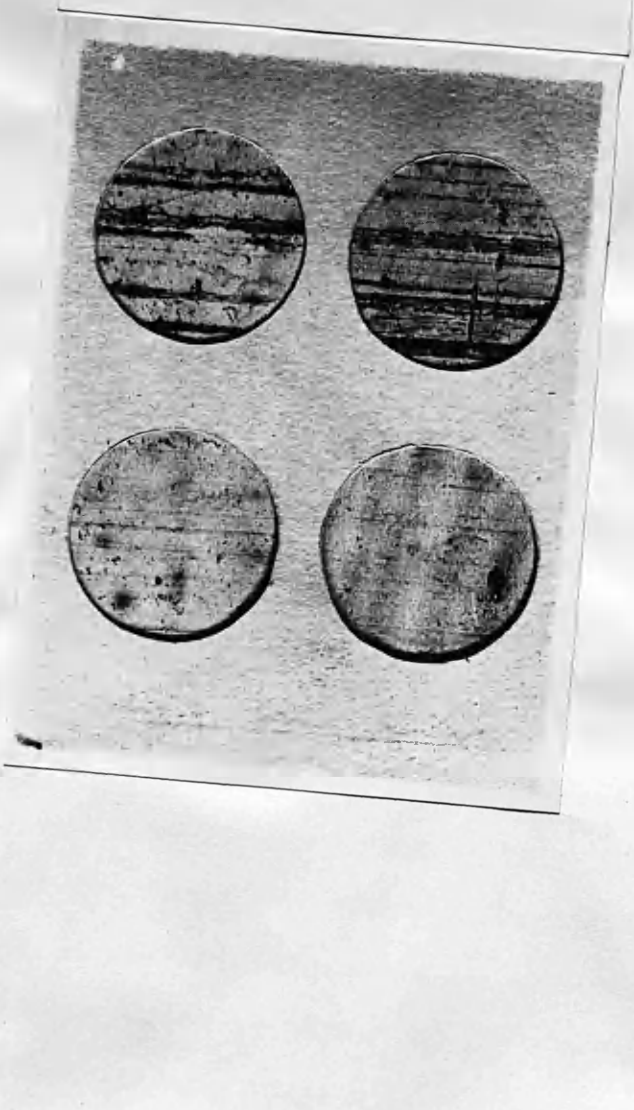
METAL SMOOTH SURFACE



5b.

METAL ROUGHENED SURFACE

FIGURE 10



The bush was adjusted and a smooth cut was taken in Lathe; a further test was then run. For comparison a spring-tool was used in turning out the bore, this took off the spiral finish of the more pointed turning tool, and again friction results were obtained. Next a grinder surface finish was tried, and then the same followed by a roller tool. The last surface was that obtained from lathe spring scraper followed by roller tool. The latter tool had the effect of hardening the white metal giving a Brinell hardness of 31 in place of 29 at 70°F.

Boundary Conditions in Lubricated Elements.

With these remarks on the various lubricants, bearing metals, and finish of surface, it is now proposed to demonstrate that mineral oils have a smaller surface tension than water, and fatty oils have a still lower value than mineral oils. Boundary friction is shown to be a function of the chemical constitution of the lubricant, and the nature of the solid surfaces, since these properties affect the attraction between both.

Bachmann and Brever demonstrated that the higher the lubricating power of an oil, the higher is the heat of wetting against finely divided copper. Taking 100 grams of copper, the heats of wetting were:

	Castor oil	Linseed oil	Liquid Paraffin	Petroleum	Petroleum Acid 1%
Calories	12.1	13.8	3.8	5.7	21.3

The efficiency of the addition of small quantities of free organic acids in lowering the surface tension of petroleum lubricants, and thus enhancing their "oiliness" properties, is clearly demonstrated. For boundary conditions a good lubricant is one which is strongly attracted by the solid whereas a poor lubricant is one which is attracted less strongly, (reference to paper on Lubricating Properties of Oils).

A/

A paper on "The Elastic Range of Friction" by J.S. Rankin [2] and an article on "The Effects of Gases, Vapours and Liquids on the Limiting Friction between Solid Surfaces" by J.M. MacAulay [3] show clearly the effect of excessive pressure and the failure of lubrication in friction when boundary conditions prevail.

Lubricants with Force Fits.- That Viscosity varies with pressure can be clearly seen by examination of test figures obtained in lubrication of plugs used for "Force Fit Experiments" also "Contact Film Resistance in Rail Wheel Force Fits". [4.] In Fig. 6(a), with a Mineral Oil, the push off force never reaches the push on pressure, showing that the film has not broken down, whereas in Fig.6(b) the tallow and white lead curves show a failure of the film and a push off pressure greater than the push on pressure. The viscosity of the mineral oil has increased nearly ten times under radial pressure; at the same time the Viscosity of the tallow has increased four times. There is a cold-working condition of the material under test. This has been clearly established by Dr. Russell, but this investigator has failed to point out the effect of the different film strength of the various lubricants used. Tallow has an approximate film strength of 40,000 lb. and a friction coefficient of 0.005. Castor oil has a film strength of 60,000 lb. and a friction coefficient of 0.05, and its wetting value is not nearly as good as tallow.

Figs. 7 to 9 show comparisons between vegetable and animal oils. The fatty acid added to the pure rape does not show a great improvement. These are compared in 8(a) and 8(b). When selecting a lubricant to be used for press on or force fits the author would recommend that the lubricants be a mixture of the lubricants which will combine film strength (i.e. castor oil) and the higher wetting properties, and lower coefficient of friction (i.e. Tallow).

Lubricants with Pressings.- If a corrosive substance added to the lubricant would not be detrimental, then the introduction of a volatile substance, such as carbon tetrachloride, has been found/

LUBRICANTS WITH FORCE FITS.

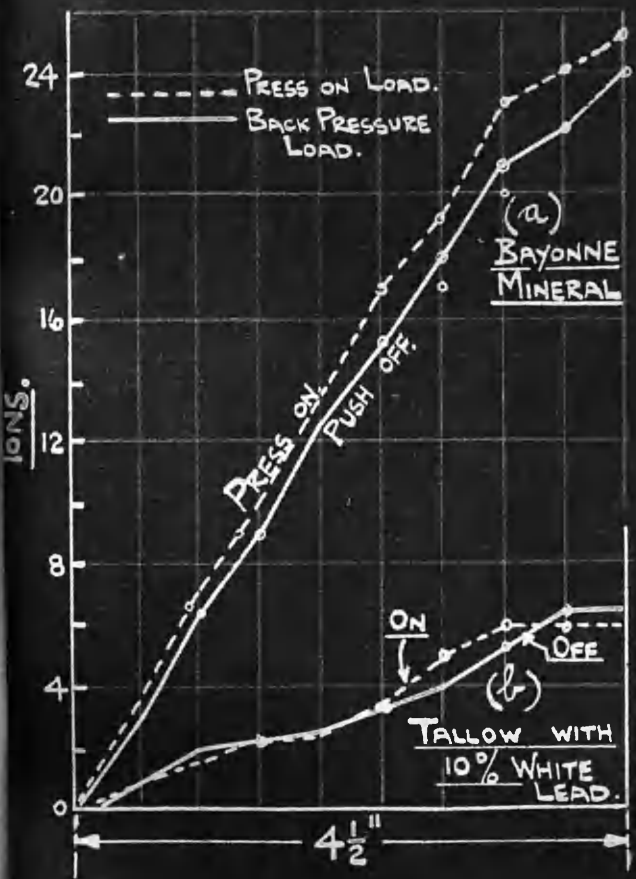
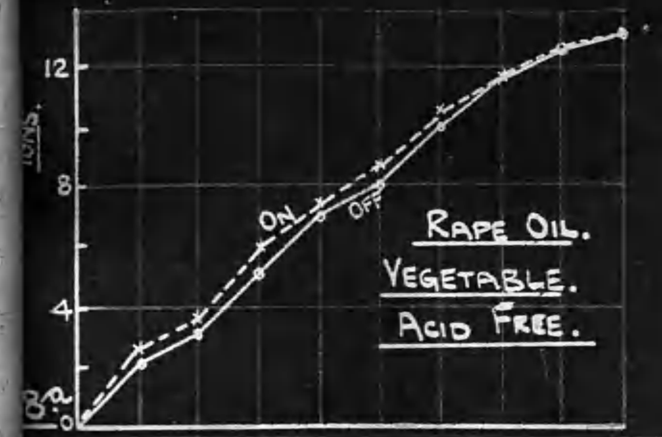
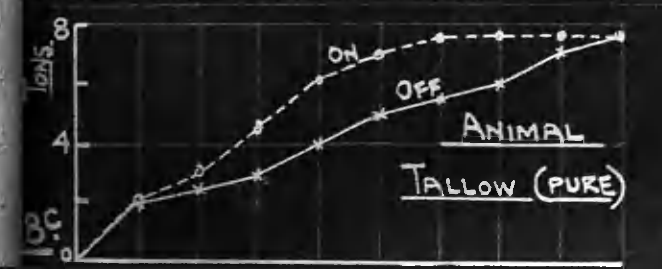


FIG. 6.

8b.



8a.



8c.

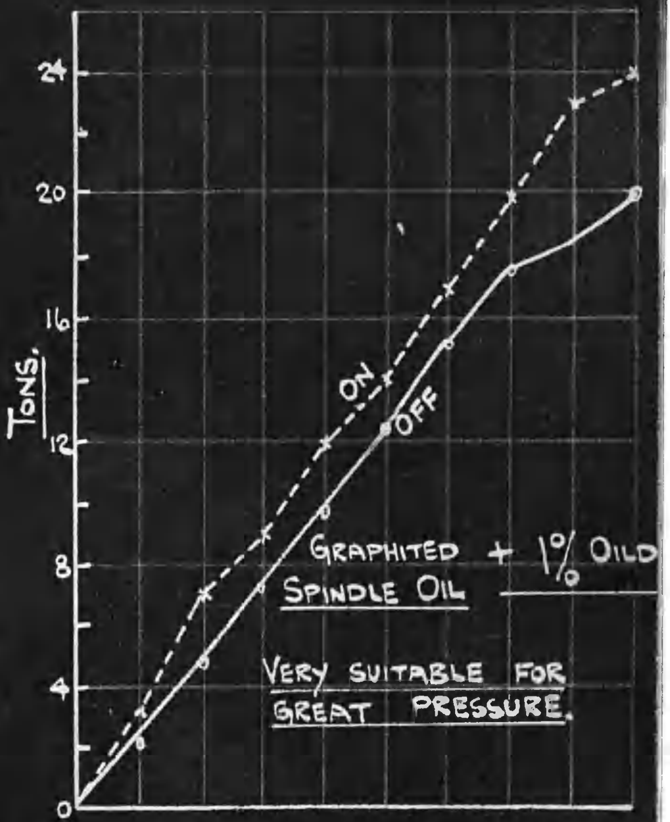


FIG. 7.

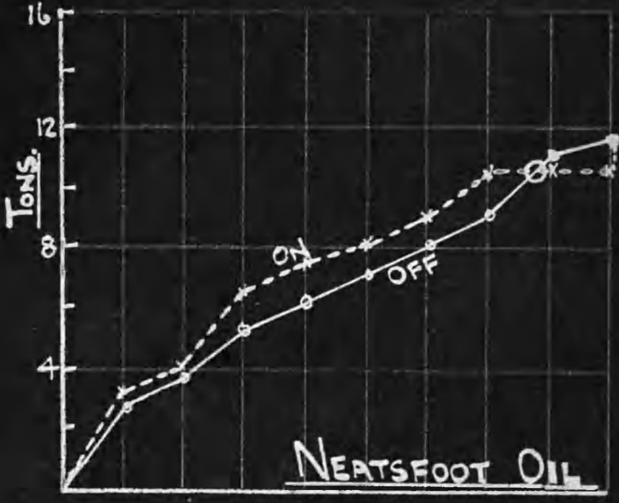


FIG. 9.

found to so fortify the film strength as to make it almost indestructible. The application of such a compounded lubricant, castor oil, tallow, and carbon tetrachloride, to the surface of a sheet steel plate pressing has reduced the power required by nearly 30 per cent, and the withdrawal of the press tool from the pressing and the pressing from the die was very much simplified. Advantage of the lubricating properties have been taken to make material flow into a difficult die formation, and at some parts of the metal sheet to be pressed no lubrication has been applied, and the metal has been found to hold with little or no movement.

The author had the opportunity of carrying out many experiments in the Gartsherrie Engineering Company and Weldless Chain works at Coatbridge with pressing and stripping tools.

The dies were found to last longer if a lubricant of high film strength, low coefficient of friction, and good wetting properties, was used.

Lubricants with Bearings.- While considering the effect of boundary lubrication and semi-lubrication, tests were carried out on lubricating oil drawn from the crank case of a motor car after different periods of use. The oil was allowed to stand in a test-tube until all sediment had separated, and then the usual tests were carried out on the Deeley and Boult testing machines. Very little difference, if any, could be detected between the new and the old oil. In some cases the old oil appeared to possess lubricating qualities superior to the new oil. It was found, however, that the pegs on the Deeley machine and the steel plate had their surfaces destroyed when left in the oil under heavy pressure; Also, that in starting up and stopping the Boult machine bearing surfaces were being destroyed by the crank-case oil.

A chemical test was applied to the old and the new oils, and it was found that an appreciable amount of sulphuric acid had been added to the lubricating oil in the crank-case.

The/

The H_2SO_4 had not dissolved in the oil but was held in suspension, and could be separated by a centrifugal separator. The ageing test on the Deeley showed signs of the corrosive effect of the acid. The oil film had been broken by the application of heavy loads and where this had taken place the acid coming in direct contact with the metal attacked it with vigour.

On bearings $F_e + H_2SO_4 \longrightarrow F_eSO_4 + H_2$

and is followed by a secondary reaction

$$F_eSO_4 + H_2 \longrightarrow F_eS + H_2O$$

The Bearing Machine which was designed in the Royal Technical College is fitted with a double ring oiler bearing 2½ in. diameter and 5 in. long. By indenting into the top bearing bush pieces of material to be tested, the question of friction with semi-lubrication and the wear of material under different conditions can be considered. The rings may both be removed or one may be allowed to carry oil to the shaft, thus supplying oil to the friction pad piece under test. In this way an abundant, a restricted supply, or no oil supply can be obtained. The rubbing speed, V, and the pressure, p, per unit area of rubbing surface, was altered throughout the tests, also the temperatures range from 66 to 200°F. and was kept constant at a predetermined condition. The critical condition is reached when $V/\sqrt{p}$ is greater than 30 in lb.ft.min. units as the coefficient of friction rapidly increases. When 30 lb.ft.min. unit has been exceeded the 'semi-fluid' phase is attained. Although the friction is less when $V/\sqrt{p} < 30$ in lb.ft.min. units, it would appear that the chance of seizure is greater and the wear is also more rapid. In the semi-fluid condition the coefficient is independent of viscosity.

<u>Tests and Test Results:</u> <u>Deeley Machine.</u>

<u>Series 1.</u>- Nickel-Chrome Steel on White Metal.

To find the effect of machine finish or smoothness of surfaces on Static Friction.

The/

NICKEL-CHROME STEEL ON WHITE METAL

Table showing μ - value for different finishes

Load on Bearing Material lb.	Disc Turned Effort ÷	Disc Turned and Scraped Effort ÷	Disc Turned Scraped and Burnished. First Polish	Second Polish	(a) Disc run in First Finish	(b) Disc run in Second Finish	(c) Disc run in Third Finish	(d) Disc run in Average 4th & 5th Finish
10	1 = 12.8	1 = 9.5	8.7	8.3	7.8	7.5	6.75	6.75
20	2 = 13.4	2 = 11.5	10.0	9.7	8.35	8.0	7.2	7.1
30	3 = 14.9	3 = 10.7	10.0	9.5	8.9	8.3	8.07	8.0
40	4 = 14.0	4 = 10.0	10.0	10.1	8.45	8.25	8.25	8.2
50	5 = 13.8	5 = 9.6	9.7	10.2	8.9	8.8	8.3	8.25
60	6 = 13.9	6 = 9.8	9.6	10.1	9.6	9.2	8.6	8.4
70	7 = 14.4	7 = 9.7	9.7	9.8	9.7	9.6	8.9	8.5
80	8 = 15.3	8 = 10.3	10.3	10.0	10.0	9.75	8.85	8.75
90	9 = 16.6	9 = 10.4	10.4	9.9	9.8	9.4	9.15	9.0
100	10 = 17.5	10 = 10.6	10.0	9.75	9.55	9.45	9.3	9.15
Average values	14.66	10.2	9.84	9.73	9.2	8.83	8.33	8.21
μ - values	0.330	0.234	0.225	0.223	0.221	0.205	0.191	0.188

Surfaces separated every five minutes and oil caused to flow between them to ensure clean rubbing. This method proved quite successful.

Processes (a),(b),(c), and (d) carried out on Deeley machine, rubbing times, ½-hour, average pressure, 50 lb. per sq.in.

The Oil in use is Castrol XL. and the only alteration made to the Standard Deeley machine was the addition of a heating coil, so that the temperature of the oil might be varied or maintained. The properties of the nickel-chrome steel and the white metal have already been given, p.4.

The steel pegs have been finished carefully, and lapped to give a smooth surface which is perfectly flat to bear against a white-metal surface.

The white metal surface is finished in the following manner:-

(1) Surface finish - fine or smooth turned.
(2) " " - after smooth turning, scraped.
(3) " " - after turning and scraping, the white metal is burnished which caused flowing and has a hardening effect.
(4) " " - same as (3) but with second burnishing.
(5) Test materials rubbed against each other under light pressure. The effect produced is similar to "running-in" of a journal bearing.

(6) Is the same as (5) but with longer period of running-in.

(7) Is the same as (5) and (6) but with special finish.

The effect of the finish is clearly shown in table of results, by the gradual lowering of the static value of the coefficient μ. The average value of μ becomes quite steady after (3), showing the value of running-in.

(See Table of Results on separate sheet)

<u>Series II.-</u> Nickel Chrome Steel on Phosphor Bronze.

In this case the resultant changes are not quite so marked. The first turning finish is much better than that of the white-metal. The turning tool does not cause so much chattering or knurling on the surface of the harder phosphor bronze plate. The harder material, Brinell No. 93, 10 mm. ball and 1000 Kg, corresponding to Rockwell R_b 50, $\frac{1}{16}$ in. ball and 100 Kg. takes on a much better finish with apparently less care than the white-metal, which when tested for hardness on the Rockwell machine/

machine specially equipped with an $\frac{1}{8}$ in. diameter ball and a pressure of 60 Kg. This special equipment gave a measure of the hardness of fabric and soft material when read on the R_b scale, and corresponded in the case of white metal R_b 50, to Brinell No.29, $22\frac{1}{2}$ mm. ball and 500 Kg.

A note on the various methods of testing for hardness, and on the connection between the hardness numbers has not been fully dealt with in this investigation. This subject is worthy of a thorough investigation. The hardness number has been made use of in the selecting of the best speeds and feeds for machining.

Table showing μ-value for different finishes.

Nickel-chrome steel on Phosphor Bronze.

First Finish - Machined; Second Finish - Machined and scraped; Third Finish - Machined, scraped and run-in.

Load on Bearing Material	First Finish Effort -			Second Finish Effort -			Third Finish Effort -		
10	9.5	or ÷ 1 =	9.5	8.75	or ÷ 1 =	8.75	8.5	or ÷ 1 =	8.50
20	18.25	" 2 =	9.12	18.1	" 2 =	9.05	17.5	" 2 =	8.75
30	27.3	" 3 =	9.10	26.7	" 3 =	8.90	26.0	" 3 =	8.66
40	35.6	" 4 =	8.90	35.1	" 4 =	8.77	34.9	" 4 =	8.72
50	43.25	" 5 =	8.65	43.0	" 5 =	8.60	42.5	" 5 =	8.50
60	51.80	" 6 =	8.65	51.5	" 6 =	8.59	51.0	" 6 =	8.50
70	61.75	" 7 =	8.82	60.0	" 7 =	8.55	60.0	" 7 =	8.57
80	70.25	" 8 =	10.03	70.0	" 8 =	8.75	70.0	" 8 =	8.75
90	82.25	" 9 =	9.14	80.1	" 9 =	8.90	80.0	" 9 =	8.90
100	94.0	"10 =	9.40	90.0	"10 =	9.00	88.0	"10 =	8.80

One point which is clearly brought out in this test is the great advantage of running-in bearings at light loads. There was a tendency for the bearing surfaces to seize at heavy load, if the surfaces were not perfectly smooth, even although the precaution of inserting a film of oil had been taken.

Example (a) - White-metal turned (only) and mating with smooth finished Nickel-Chrome steel at the bearing pressure of 100 lb. per sq.in. gave a reading 170 on the Deeley machine, or a static coefficient of $\frac{170 \times 10}{100 \times 43.58} = \frac{17}{43.58} = 0.387.$

After/

After perfectly smooth surfaces were obtained with polishing the rubbing-in process applied followed by the cleaning process referred to in some of the previous papers in the use of the Deeley machine. The value of μ becomes

$$\frac{9.3}{43.58} = 0.213$$ which is a decrease in the static friction value of 45%.

Example (b) - Phosphor Bronze (Turned smoothly not polished) at high load with nickel-chrome steel, pressure 100 lb.per sq.in. gave

$$\mu = \frac{94 \times 10}{10 \times 43.58} = 0.26$$ for the value of static friction, which changed to

$$\mu = \frac{88 \times 10}{10 \times 43.58} = 0.22$$ a lowering of the static friction coefficient value by 22.3% by lapping and polishing the surfaces of the bearing materials.

<u>Series III.</u> - Castor Oil being the lubricant and bearing surfaces Nickel-Chrome Steel on Cast Iron.

Table of μ - values for different finishes:-

Load on Bearing Material	Smooth Turned and scraped		Smooth Turned and polished		Run-in on light load metal on metal lubricants	
10	6.5	1 = 6.5	6	1 = 6	5.7	1 = 5.7
20	12.75	2 = 6.37	11.4	2 = 5.7	11.6	2 = 5.8
30	19.20	3 = 6.4	17.35	3 = 5.78	17.25	3 = 5.75
40	25.8	4 = 6.45	23	4 = 5.75	22.6	4 = 5.65
50	31.5	5 = 6.3	28.8	5 = 5.76	28.75	5 = 5.75
60	38.0	6 = 6.35	34.2	6 = 5.7	34.5	6 = 5.75
70	44.0	7 = 6.3	40.1	7 = 5.73	40.25	7 = 5.75
80	50	8 = 6.25	45.6	8 = 5.7	45.25	8 = 5.66
90	55.8	9 = 6.2	51.25	9 = 5.7	50.8	9 = 5.64
100	60.2	10 = 6.02	56.7	10 = 5.67	55.5	10 = 5.55
Static Friction	$\mu = 0.145$		$\mu = 0.132$		$\mu = 0.131$	

Example (c) - Cast Iron (smooth turned and scraped not polished) at high load with Nickel-Chrome Steel at a pressure 100 lb.per sq.in gave $$\mu = \frac{60.2 \times 10}{10 \times 43.58} = \frac{60.2}{43.58} = 0.138$$

and/

and when the bearing surfaces had been polished, rubbed-in on oil

$$\mu = \frac{5.55 \times 10}{10 \times 43.58} = \frac{5.55}{43.58} = 0.127$$

which shows a lowering of the static friction value of only 8%.

Using the Brinell No. as a basis of hardness for test material, white-metal, 29; phosphor bronze, 93; cast iron, 190. The results confirm that the hardness of a material even when only smooth finished has resisting properties on the tool ridges. The ridges, left by the turning and spring scraper, in the case of white-metal, have a tendency to cause the bearing materials to seize. It is well known that white-metal flows under pressure and that the oil film in a journal bearing is effected by this flow. The metal follows the eccentricity of the loading and this is followed by a reduction of the clearance.

<u>Series IV.</u> - Lubricating Oil Castor XL with different bearing metals and varying loads. The velocity or rubbing speed was kept as constant as possible throughout the tests.

Load	(1) Cast Iron on White Metal				(2) Cast Iron on Cast Iron			
10	6.8	7.9	7.1	1 = 7.1	5.8	1 = 5.85	5.7	1 = 5.7
20	14.2	15.5	14.8	2 = 7.4	11.75	2 = 5.87	11.5	2 = 5.75
30	22.5	24.5	23.0	3 = 7.6	17.65	3 = 5.88	18.6	3 = 6.2
40	35	36	35.5	4 = 8.9	24.5	4 = 6.12	24.5	4 = 6.37
50	43	44	43.25	5 = 8.65	31.25	5 = 6.25	31.7	5 = 6.34
60	52.5	54	53.0	6 = 8.8	37.1	6 = 6.2	36.3	6 = 6.05
70	62	62	62	7 = 8.85	43.5	7 = 6.21	42.6	7 = 6.08
80	75	73	70	8 = 8.7	48.5	8 = 6.06	47.5	8 = 5.94
90	84	80	81	9 = 9.0	54.6	9 = 6.06	52.1	9 = 5.8
100	100	97	89	10 = 8.9	59.2	10 = 5.92	58.6	10 = 5.86

Static value of $\mu = 0.194$ $\qquad$ $\mu = \frac{6.02}{43.58} = 0.137$

Oil	2(a) M.A. on C.I.	2(b) M.S. on P.B.	2(c) M.S. on M.S.
Bayonne oil	0.216	0.226	0.267
Bayonne 2% Rape	0.186	0.196	0.214
Castrol XL	0.198	0.208	0.241
Average	$\mu = 0.20$	$\mu = 0.21$	$\mu = 0.24$

Load	(3) Cast Iron on Phosphor Bronze				
10	8.5	8.2	1	=	8.3
20	17.6	16.8	2	=	8.6
30	27.5	25.5	3	=	8.8
40	36.2	34.5	4	=	8.8
50	45.2	43.7	5	=	8.9
60	54.2	52.5	6	=	8.9
70	63.5	61.75	7	=	8.98
80	70.0	69.5	8	=	8.75
90	78.5	77.5	9	=	8.66
100	84.8	83.38	10	=	8.52

$$\mu = \frac{8.6}{43.58} = 0.198$$

(1) Cast Iron on White-Metal.- In this test the same difficulty of obtaining a correct value at high loads was in evidence, the cast iron having to be run against the white-metal surface (at light loads) until a perfect smooth surface had been obtained. As in the previous tests the maximum load is 100 lb.per sq.inch.

Maximum $\mu = \frac{10}{43.58} = 0.221$ and minimum $\mu = \frac{8.9}{43.58}$ = 0.202, a difference of 8.18% due to difference of surface conditions. The surface of the white metal had been polished, and effectively cleaned in the way already explained.

(2) Cast Iron on Cast Iron.- The values obtained for the static friction are extremely regular. The same test had been carried out with these two metal surfaces more than eight months previously, and the μ - value obtained was 0.131; in the present test $\mu = 0.137$. The increase in the coefficient due to ageing had been noticed among the first experiments where the period of ageing was ten days, and this test was put on to confirm the statement that ageing alters the value of the static μ . 2(c) High μ value 0.34,seizure occurs.

(3) Cast Iron on Phosphor Bronze. - There is not the same tendency to seize at high loads, but still there was a grip between the surfaces giving a higher value of μ , which decreased/

decreased with rubbing-in of the surfaces. The values given in the table are for the condition after a true bearing surface had been obtained. For curves see Caldwell and Thomson.[7]

Fine Finish to Turned Surfaces

In making new test plates for the Deeley machine the problem of finish presented a considerable difficulty. The plates were smooth turned. This surface under the microscope presented a most irregular appearance. A spring scraper was used and the ridges partly disappeared, but still the surface was very irregular. Various lubricants were tried as coolants for the lathe tool, the shape of the tool being kept the same throughout. It was found that a small drop of glycerine and water gave the best finish and avoided the use of a carbonundum oil slab. The use of the grinder would have introduced an unknown factor into the friction tests. Pure oil of turpentine, was tried on the spring scraper but the surface obtained was rougher.

Tests and Test Results: Bearing Machine.

The machine as designed did not come up to the requirements of research work in all the points of lubrication of journal bearings, and was replaced by Dr. Thomson by a machine described in an article[6] on Lubrication and Bearing Design. The machine, however, is quite suitable for tests on effect of smoothness of machine surface using phosphor bronze and white metal bushes or pad pieces. It may also be used as a lubricant and wear tester in conjunction with the Disc Machine used for fabric material, giving an excellent *means* of comparing oils or friction material when running against a lubricated or wetted surface.

The friction at various rubbing speeds can be determined. The temperature of the lubricating oil is kept constant, and the load is varied over the test, but may be kept constant over a range of speeds.

Series I/

Series I. test (a) - Temperature kept constant at 66°F.
Speed varied from 0 - 400 r.p.m. or 0 to 314 ft.per min.
to 1280 ft. per min.
Load varied from 56 - 160 lb. per in^2 and then to 228 lb/in^2

Oil used - Gas Engine Lubricant.

See separate sheet for Table.

The journal bearing had already been running for eight hours before test readings were taken. The surface of the material was extra smooth and at the end of test was in perfect condition. (Fig.10) p. 12.

Series I. test (b) - Temperature kept constant at 138°F.
Speed varied from 0 to 400 r.p.m. to 1600 r.p.m. or 1280 ft. per min.
Load varied from 0 to 228 $lb./in^2$.
Oil used - Gas Engine Lubricant.
See separate sheet for table.

Conditions were difficult to keep constant during the latter period of this test. An examination of the bearing showed signs of seizure of bearing surface. Z is viscosity in poise.

Series I. test (c) - Temperature kept constant at 190°F.
Speed varied from 0 to 1280 ft. per min.
Load varied from 0 to 228 lb. per in^2.
Oil used - Gas Engine Lubricant.
See separate sheet for table, p. 26.

As in Test (c) the bearing showed tendency to lack of oil supply and seizure of bearing surfaces.

Although this investigation deals chiefly with the effect on friction of different finishes of materials, and different oils/

Temp. of Oil Film	Load lb/in^2	Rev. per min.	w_1 lb	w_2 lb	$w_1 + w_2$ $= w$ lb	T = friction Torque $= \frac{1}{2}$ W x	$\mu = \frac{T}{1.25 XL}$	$V/\sqrt{P}$ lb.ft.min.
66°F.	56	50	0.5	0.1	0.6	6.75	0.016	5.52
"	"	100	0.55	0.3	0.85	9.56	0.0229	10.5
"	"	200	0.7	0.5	1.2	13.5	0.0323	21.0
"	"	250	0.8	0.6	1.4	15.74	0.0377	26.3
"	"	300	0.8	0.65	1.45	16.3	0.039	31.4
"	"	350	0.9	0.65	1.55	17.42	0.0418	37.1
"	"	400	0.95	0.65	1.6	18.0	0.0431	42.0
66°F.	100	50	0.1	0.5	0.6	6.75	0.0095	3.97
"	"	100	0.2	0.6	0.8	9.0	0.0.26	7.85
"	"	200	0.4	0.85	1.25	14.05	0.0197	15.7
"	"	250	0.5	0.9	1.4	15.72	0.0214	19.67
"	"	300	0.6	0.95	1.55	17.42	0.0244	23.6
"	"	350	0.6	1.0	1.6	18.0	0.0252	27.5
"	"	400	0.65	1.0	1.65	18.55	0.026	31.4
66°F.	160	50	0.7	0.0	0.7	7.87	0.007	3.13
"	"	100	0.9	0.1	1.0	11.25	0.01	6.16
"	"	200	1.2	0.3	1.5	16.87	0.015	12.8
"	"	250	1.3	0.4	1.7	19.1	0.017	15.5
"	"	300	1.4	0.45	1.85	20.8	0.0185	18.65
"	"	350	1.45	0.45	1.9	21.4	0.019	20.9
"	"	400	1.5	0.5	2.0	22.5	0.02	24.8
66°F.	228	465					0.022	25.0
"	"	700					0.023	39.0
"	"	932	Large pulley on Counter Shaft, Bearing kept cool.				0.026	48.5
"	"	1160					0.031	60.5
"	"	1400					0.037	72.5
"	"	1600					0.046	84.5

Temp. of Oil Film	Load lb/in^2	Revs. per min.	w_1 lb.	w_2 lb.	$w_1 + w_2 = w.$	Friction Torque.	$\mu = \frac{T}{1.25 L}$	$\frac{P}{ZN}$
138°F.	56	50	0.1	0.25	0.35	3.94	0.0094	1.8
"	"	100	0.2	0.45	0.65	7.3	0.0174	0.94
Z = 0.6	"	200	0.4	0.6	1.0	11.25	0.0268	0.47
poise	"	250	0.5	0.7	1.2	13.5	0.0322	0.37
	"	300	0.6	0.7	1.3	14.6	0.0349	0.37
	"	350	0.65	0.75	1.4	15.75	0.0375	0.27
	"	400	0.7	0.8	1.5	16.9	0.04	0.23
138°F.	100	50	0.1	0.35	0.45	5.06	0.0071	3.3
	"	100	0.2	0.45	0.65	7.31	0.0103	1.6
	"	200	0.3	0.6	0.9	10.1	0.0142	0.8
	"	250	0.4	0.7	1.1	12.34	0.0174	0.66
	"	300	0.5	0.75	1.25	14.05	0.0197	0.53
	"	350	0.55	0.8	1.35	15.2	0.0213	0.47
	"	400	0.6	0.85	1.45	16.3	0.0228	0.4
138°F.	160	50	0.1	0.3	0.4	4.5	0.004	5.3
	"	100	0.2	0.5	0.7	7.88	0.007	2.6
	"	200	0.2	0.6	0.8	9.00	0.008	1.3
	"	250	0.2	0.65	0.85	9.56	0.0085	1.06
	"	300	0.2	0.7	0.9	10.1	0.009	0.9
	"	350	0.25	0.75	1.0	11.25	0.01	0.75
	"	400	0.3	0.8	1.1	12.4	0.012	0.67
138°F.	228	465	0.35	0.85	1.4	15.6	0.015	0.82
	"	700	0.9	1.0	1.9	21.7	0.021	0.54
	"	932	0.9	1.25	2.15	23.8	0.023	0.41
	"	1160	1.0	1.45	2.45	27.0	0.026	0.33
140°F.	"	1400	0.9	1.55	2.45	27.0	0.026	0.27
145°F.	"	1600	0.7	1.65	2.35	26.0	0.025	0.23

Temp. of Oil Film	Load lb/in^2	Revs. per min.	w_1 lb.	w_2 lb.	$w_1 + w_2 = w$	Friction Torque.	$\mu = \frac{T}{1.25L}$	$\frac{P}{ZN}$
190°F.	56	50	0.3	0.2	0.5	5.62	0.0134	9.4
	"	100	- 0.5	0.9	0.4	4.5	0.010	4.7
Z = 0.16	"	200	- 0.7	1.0	0.3	3.37	0.008	2.35
	"	250	- 0.5	0.9	0.4	4.5	0.0108	1.9
	"	300	- 0.5	1.0	0.5	5.63	0.0135	1.56
	"	350	- 0.3	1.1	0.8	9.0	0.0215	1.35
	"	400	- 0.2	1.05	0.9	10.1	0.0242	1.18
190°F.	100	50	0.1	0.3	0.4	4.5	0.0063	12.5
	"	100	0.05	0.25	0.3	3.37	0.0047	6.2
	"	200	0.15	0.45	0.6	6.75	0.0095	3.1
	"	250	0.18	0.54	0.7	7.88	0.0138	2.5
	"	300	0.20	0.6	0.8	9.0	0.0126	2.1
	"	350	0.15	0.55	1.1	12.4	0.0111	1.7
	"	400	0.2	0.7	1.2	13.5	0.0126	1.5
190°F.	160	50	0.1	0.5	0.6	6.75	0.006	20.0
	"	100	0.1	0.3	0.4	4.5	0.004	10.0
	"	200	0.15	0.5	0.64	7.2	0.0064	5.0
	"	250	0.15	0.55	0.7	7.88	0.007	4.0
	"	300	0.2	0.6	0.8	9.0	0.008	3.3
	"	350	0.2	0.7	0.9	10.1	0.009	2.8
	"	400	0.2	0.8	1.0	11.25	0.01	2.5
190°F.	228	465	1.90	1.6	3.5	39.5	0.035	3.07
	"	700	1.50	1.5	3.0	33.7	0.03	2.05
	"	932	1.35	1.35	2.7	30.5	0.027	1.54
	"	1160	1.15	1.25	2.4	27.0	0.024	1.24
	"	1400	1.02	1.25	2.3	26.0	0.023	1.02
	"	1600	1.0	1.2	2.2	25.0	0.022	0.90

average coefficient values and these obtained under ideal conditions of working.

TESTS AND TEST RESULTS.

Series (1)

A comparative test was carried out between the working materials, Bronze, Wood, and Leather, and compared with the fabric materials 'Ferasbestos' and 'Gatex' in Fig. 6.

The Bronze packing pieces in the carrier were used as a templet from which to form the pad pieces. These pads were carefully adjusted to the curvature of the brake wheel. The conditions of the test are such that no material is overheated.

	Bronze	Ferasbestos	Wood	Leather	Gatex
Pressure lb./in^2	28	28	28	14	14
Rubbing Speed ft./min.	115	115	86	86	86
Max. Temp. ^{o}F.	350	300	250	200	175

The heat was supplied by four bunsen burners and regulated to give the required heat to the rim of the wheel so that the temperature of the pad pieces at the brake surface could be read on the potentiometer. A graph was used for the potentiometer which gave readings in degrees F. This graph was furnished by the Natural Philosophy Department of the Royal Technical College and was used throughout this research work. The instrument was checked from time to time during the test period.

For each test the speed was kept constant and the pad pieces were applied to the rim but with only 7 lb. per sq.in. pressure to ensure perfect fitting of brake material, the temperature being raised gently to 100^{o}F. Then the specific load was applied as shown above. The temperature changes were also gradual as shown on graph, Fig. (6).

(a)/

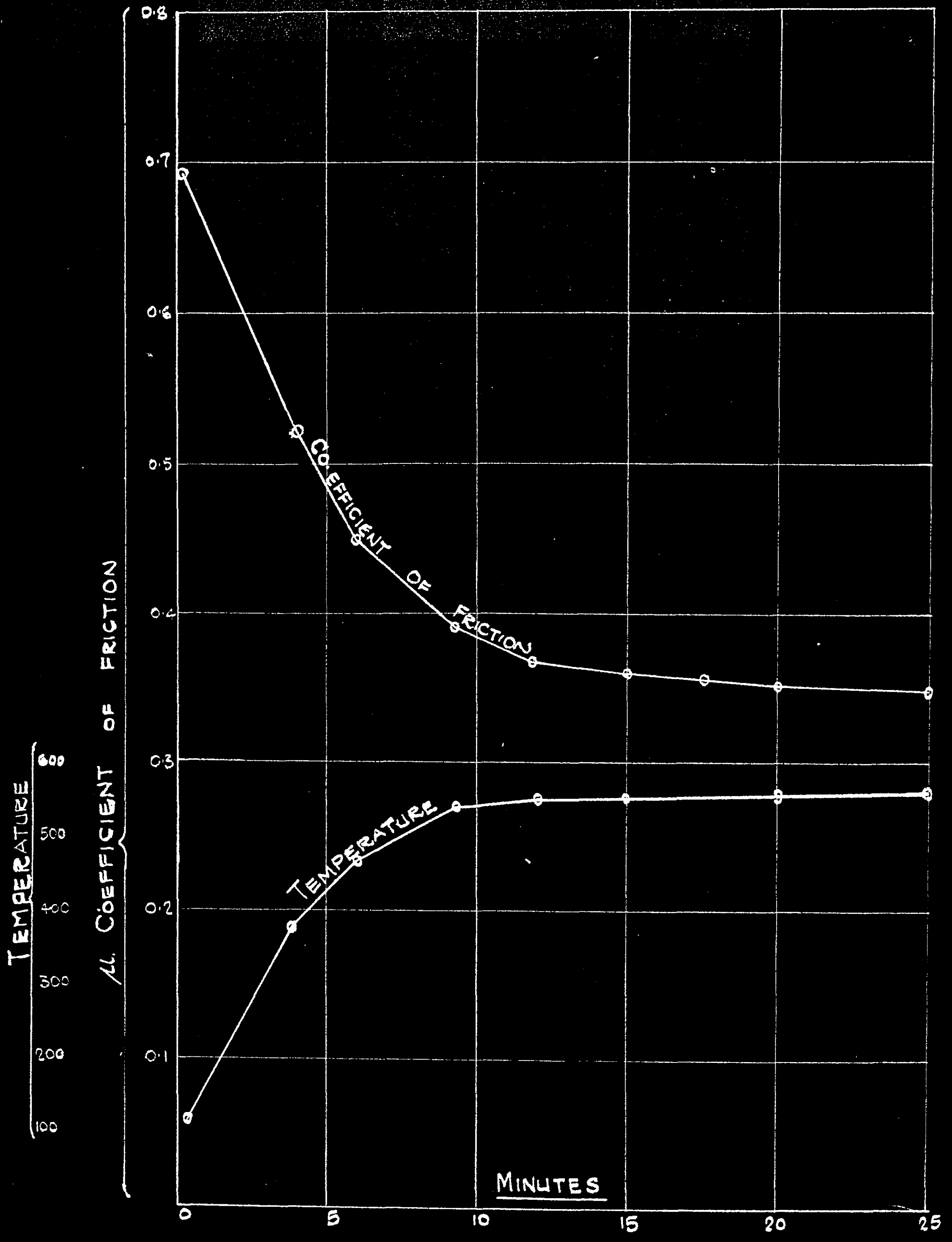
(a) With the Bronze pads scoring of the wheel would seem almost certain if retained in action for any length of time. The coefficient of friction kept fairly constant.

(b) The Wood brake pads showed just that feature which caused them to be discarded in practice where a water cooled drum could not be incorporated, such as in the 'Massey' drop-stamp hammer. Even with the smaller specific loading and lower temperature there is a great variation in the friction coefficient. Charring of the wood pad pieces soon takes place but the brake rim is in no way destroyed; it becomes rather more polished.

(c) The Leather pad pieces had to be retained carefully in the holder, that is, they were not allowed to project more than was necessary to make the cast iron carrier, E, clear the rim. Leather under pressure and temperature does not hold its high μ-value for any length of time, changing from almost 0·9 to 0·75 as the temperature changes from 100° to 175°F. The heat appears to spoil the dressing material of the leather.

(d) The first composite material on the list shows properties which certainly are to be desired. Unlike leather it is not easily deformed by pressure at high temperatures. Heat transmission from wear surface to carrier is readily found by measuring the rise in temperature of the outside of the carrier block, also the inside on non-wearing surface of the brake lining. 'Ferasbestos' is a Ferodo material made by teasing out the single silky-like fibres of the mined asbestos. Each fibre can be treated separately by soaking in a resinoid compound. Asbestos flock may be spun and then treated with fluid gumming material, bakelite, or sugar compound. Pressed lightly into form and baked in an electric drying oven, the lining is then compressed by hydraulic press. The moulds are electrically heated and the pressure may be 15 tons per sq.in.

The/



LOAD ON BRAKE 87 LB OR PRESSURE 87 LBS/SQ. INCH.

RUBBING SPEED 25 FEET PER MINUTE

The method of manufacture as well as the materials used are brought out in the test results shown in curves (d) Fig. (6), p.14. The brake rim is only slightly polished and the brake pads show no change in surface. The small serrations on the face of the pads used [in Table (1), I - Ferodo Bonded Asbestos], do not appear with the Ferobestos.

(e) 'Gatex' is really a belting material, but as it has a very compact structure and could be cut into the small pad pieces required without excessive fraying of the edges, it was put under test. The load applied was 14 lb.per sq.inch. It shows a very high coefficient of friction much similar to that of leather. It is made of a woven yarn treated with a rubber latrex and the whole is vulcanized into a solid fabric. Very little sign of change in structure could be discerned by the rise from 100° to 200°F. and a pressure of 14 lb. per sq.in. is certainly not excessive.

The author would have liked to carry out research work on the newer form of belting material when used for brake linings. The cotton duck and cotton fibres have given way to woven hair material specially treated so as to have no internal shear among the fibres. Belting problems appear to give a mixture of fluid and solid friction which has given great difficulties to investigators, but in fabric linings, certainly, the manufacturer has overcome many of the early difficulties by special processes . Belting owing to the travel has more cooling surface than brake or clutch material.

Series (2).

These tests were carried out using four different brakes of friction brake linings; they are the same group as used in making the preliminary tests, (I) 'Ferodo Fibre', (II) 'Ferodo Bonded Asbestos', (III) 'Breko', and (IV) Raybestos.

The three last named being very much similar in structure, but made by different manufacturers, who recommend special working pressures to be used for their material. The Author has already written on this subject [6] of Ferodo Brake Linings, but this research work provides an opportunity of comparison without writing unfavourably of any manufacturer's product.

The pressure ranges from 7 to 28 lb. per sq.in. using pads 1 sq.inch area, and to permit of higher pressures special pads of ½ sq.in. are used. The temperature range is from room temperature 55°F. to 120°F. No external heat is applied.

From the earlier tests experience had been gained in running the machine. The object now aimed at is continuous running over the whole speed range with absolute minimum of stops. It was found that, except at the highest speeds, the necessary changes could be made without any damage to the gearing. The rim was marked where gear changing could be made, and it soon became a matter of practice to make a smooth change; the teeth were given a small entrance angle.

Each test was started at top speed and subsequent readings taken in a generally decreasing order of speed. By adopting these methods very few complete stops required to be made. The machine was allowed time to assume constant conditions after each change. This was necessary not only for the speed function but also to permit of the steady-up of the conditions of the brake linings. At each change of load the machine was brought to high speed before proceeding to add load. Up-load and down-load readings were taken for each material.

A rough graph of the readings was made on a load base, also on a time base. This graph enabled a check reading to be made of any doubtful point. At first an average/

average of a number of readings was taken as the final reading, but the machine could be depended upon to give a fair estimate of μ without much care. The principal point to watch is the adjustment of the position on brake rim, and to see that the hangers are not fouling on the sides of the brake wheel.

The results of these tests are shown in Tables (7) to (14) and the curves have been plotted to show the coefficient of friction, μ-value, on a base of specific speed, Figs (6) to (15). For the purpose of comparison the curves have been plotted for separate loadings, Figs. (9a,b,c,d) These curves may be analysed to bring out the respective features inherent in the fabric linings.

Fabric Friction.

Table 7. 'Ferodo' Fibre. Constant of Machine 2·8 lb.

		Pull in lb.				
No. of Drives	Speed ft./min.	7 lb load 7 lb/in^2	17·2 lb 17·2 lb/in^2	27·4 lb 27·4 lb/in^2	17·2 lb 34·4 lb/in^2	22·3 lb 44·6 lb/in^2
e c C	0·258	5·6	12·2	18·1	10·4	13·2
e b C	0·38	5·4	12·2	18·8	10·2	13·4
e a C	0·52	5·4	12·8	19·0	11·8	14·0
e c B	0·75	5·4	12·2	18·9	11·0	13·8
e b B	1·16	5·6	12·6	18·6	11·2	14·2
e a B	1·56	5·8	13·0	19·1	11·4	15·0
e C A	2·45	5·4	12·3	18·4	11·6	15
e b A	3·40	5·6	12·2	19·0	12·2	15
e a A	4·76	5·1	11·1	18·8	12·0	-
Machine stopped to take out the Epicyclic Train.						
2 c C	25·8	4·4	10·0	14·8	10·8	13·2
2 b C	38	4·0	9·2	13·6	9·8	12·2
2 a C	52	4·0	8·6	13·4	9·2	11·6
2 c B	75	3·8	8·6	13·4	9·2	11·8
1 c C	93·5	3·1	8·6	13·0	9·0	11·7
2 b B	116	3·0	8·1	12·8	8·8	11·4
1 b C	131	3·1	8·2	13·0	9·0	11·6
2 a B	156	3·4	8·4	13·2	9·0	11·4
1 a C	191	3·4	8·2	13·0	8·8	11·6
2 c A	245	4·2	9·4	14·2	9·7	12·8
1 c B	270	3·6	8·6	13·6	9·4	12·0
2 b A	340	4·1	9·6	14·2	10·6	13·4
1 b B	381	3·4	8·4	13·6	9·8	12·2
2 a A	476	4·1	9·6	-	-	-
1 a B	530	3·6	9·0	14·0	9·6	12·2
1 c A	830	3·6	9·6	14·2	9·8	12·6
1 b A	1220	4·0	9·1	14·2	10·0	12·6
1 a A	1700	4·0	9·2	14·0	9·8	12·0

(i) 'Ferodo' Fibre. Constant for machine 2.8 lb.

able VIII

Coefficient of Friction μ.

Speed ft./min.	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34.4 lb/in^2	44.6 lb/in^2
0·258	0·80	0·70	0·66	0·61	0·50
0.38	0.77	0.71	0.68	0.59	0.59
0.52	0.77	0.74	0.69	0.68	0.65
0.75	0.77	0.71	0.69	0.64	0.62
1.16	0.80	0.73	0.68	0.65	0.64
1.56	0.83	0.76	0.69	0.69	0.70
2.45	0.77	0.72	0.67	0.68	0.67
3.40	0.80	0.71	0.69	0.70	-
4.76	0.72	0.65	0.69	0.68	-

Machine stopped to take out the Epicyclic Gear

25.8	0.63	0.58	0.54	0.63	0.59
38	0.57	0.54	0.50	0.57	0.55
52	0.57	0.51	0.49	0.54	0.52
75	0.54	0.51	0.49	0.53	0.53
93.5	0.44	0.51	0.47	0.52	0.52
116	0.43	0.47	0.46	0.51	0.51
131	0.49	0.48	0.47	0.51	0.52
156	0.49	0.49	0.48	0.52	0.51
191	0.49	0.48	0.48	0.51	0.52
245	0.58	0.54	0.51	0.56	0.57
270	0.56	0.50	0.50	0.55	0.55
340	0.58	0.55	0.51	0.61	0.60
381	0.59	0.49	0.50	0.57	0.55
476	0.59	0.55	-	-	-
530	0.59	0.52	0.51	0.56	0.55
830	0.52	0.56	0.52	0.57	0.57
220	0.57	0.53	0.52	0.58	0.56
700	0.51	0.53	0.51	0.57	0.54

Temperature variation 60° to 120°F; this refers to last nine readings.

Table IX

(i) 'Ferodo' Bonded Asbestos. Constant for the machine 2.8 lb.

Speed t./min.	Pull in lb. 7 lb Load 7 lb/in^2	14 lb Load 14 lb/in^2	28 lb Load 28 lb/in^2	17 lb Load 34 lb/in^2	Temp.
0.258	2.6	2.8	11.6	7.4	
0.38	2.5	3.2	10.6	7.4	
0.52	2.8	2.0	10.8	7.6	
0.75	2.8	3.0	11	7.8	60°F.
1.16	3.2	3.2	10.6	8.0	
1.56	2.6	3.2	11.4	8.0	
2.45	2.2	3.4	11.2	8.4	
3.40	2.4	3.0	11.4	8.0	
4.76	2.8	3.6	11.8	7.8	

Machine stopped, Epicyclic Gear disengaged.

Speed ft./min.	Pull in lb.				Temp.
	7 lb Load 7 lb/in^2	14 lb Load 14 lb/in^2	28 lb Load 28 lb/in^2	17 lb Load 34 lb/in^2	
25.8	3.0	4.2	12.6	9.8	60°F.
38	3.0	4.2	12.4	9.2	
52	3.0	4.0	12.4	9.2	
75	3.2	4.0	12.4	8.8	
93.5	2.8	4.0	12.8	8.8	
116	2.6	3.8	12.2	8.4	
131	3.0	3.6	12.4	8.6	
156	2.6	3.6	12.0	8.6	80°F.
191	2.8	3.6	11.9	8.4	
245	2.6	4.0	13.4	9.0	
270	2.8	3.5	11.8	8.4	
340	2.6	3.8	13.4	8.8	
381	3.4	3.2	11.6	8.2	
476	2.4	3.6	-	8.6	100°F.
530	3.6	3.0	11.4	8.0	
830	2.4	3.0	12.2	8.6	
1120	2.6	2.8	12.0	8.2	120°F.
1700	3.0	2.8	11.6	7.8	

Diagrams to show the
appearance of Fabric
after test.

Table X.

(ii) 'Ferodo' Bonded Asbestos. Machine No. 2.8 lb.

Speed ft./min.	Coefficient of Friction μ				Temp.
	7 lb/in^2	14 lb/in^2	28 lb/in^2	34 lb/in^2	
0.258	0.37	0.40	0.42	0.43	
0.38	0.37	0.46	0.39	0.43	
0.52	0.40	0.39	0.39	0.38	
0.75	0.40	0.43	0.40	0.45	
1.16	0.46	0.46	0.39	0.46	60°F.
1.56	0.37	0.46	0.38	0.47	
2.45	0.31	0.48	0.40	0.48	
3.40	0.34	0.43	0.38	0.46	
4.76	0.40	0.51	0.43	0.45	
	Machine stopped, Epicyclic Gear locked.				
25.8	0.43	0.60	0.46	0.57	60°F.
38	0.43	0.60	0.45	0.53	
52	0.43	0.57	0.45	0.53	
75	0.45	0.57	0.45	0.51	
93.5	0.40	0.57	0.45	0.51	
116	0.37	0.54	0.45	0.49	
131	0.43	0.51	0.45	0.49	80°F.
156	0.40	0.51	0.44	0.50	
191	0.37	0.51	0.43	0.49	
245	0.40	0.57	0.48	0.52	
270	0.37	0.50	0.43	0.49	
340	0.48	0.54	0.48	0.51	
381	0.34	0.46	0.42	0.48	
476	0.43	0.51	-	0.50	
530	0.34	0.43	0.38	0.46	
830	0.37	0.43	0.44	0.50	
1220	0.43	0.40	0.43	0.48	120°F.
1700	0.43	0.40	0.42	0.45	

During the working with this material it was found that the machine showed up a fierceness of grip at slow speeds.

Table X

(ii) 'Breko' Bonded Asbestos. Constant for machine 2.8 lb.

Speed ft./min.	Pull in lb.				
	7 lb load 7 lb/in^2	17.2 lb load 17.2 lb/in^2	27.4 lb load 27.4 lb/in^2	17.2 lb load 34.4 lb/in^2	22.3 lb load 44.6 lb/in^2
0	2.6	8.2	12.8	7.1	9.4
0	2.6	7.8	12.4	7.6	9.8
0	3.0	7.8	12.2	7.8	10.4
0	2.8	7.8	13.0	7.0	9.0
1.16	2.8	8.0	13.2	7.6	10.0
1.50	2.4	7.2	12.4	7.6	10.0
2.45	3.0	8.6	13.6	6.2	8.6
3.4	2.6	7.8	13.6	7.0	9.6
4.76	2.1	7.0	13.5	8.0	10.0
Machine stopped to lock Epicyclic Gear					
25.8	2.4	6.6	10.8	6.0	7.8
38	2.4	6.8	10.8	5.8	7.6
52	2.2	6.4	10.4	5.8	7.6
75	2.0	6.4	9.8	5.8	7.6
95.5	2.1	6.0	9.6	5.6	7.2
116	2.0	5.8	9.4	5.4	7.2
131	2.0	6.0	9.4	5.6	7.1
156	1.8	6.4	9.2	5.4	7.0
191	1.8	6.4	9.2	5.6	7.0
245	2.2	5.8	10.6	5.4	7.6
270	2.0	6.6	9.4	5.4	7.0
340	2.4	5.6	10.1	5.8	7.8
381	1.8	6.2	9.2	5.4	6.8
476	2.6	5.8	-	6.2	-
530	1.8	5.6	8.8	5.0	6.8
830	1.8	5.6	9.3	5.4	7.2
1220	2.0	3.8	9.0	5.6	7.6

Note: With this material the friction machine worked very smoothly.

Table XI

(iii) 'Breko' Bonded Asbestos. Constant for machine 2.8 lb.

Speed ft./min.	Coefficient of Friction μ				
	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34 lb/in^2	44.6 lb/in^2
0.258	0.37	0.48	0.47	0.41	0.42
0.38	0.37	0.45	0.45	0.44	0.44
0.52	0.42	0.45	0.45	0.45	0.46
0.75	0.40	0.45	0.47	0.41	0.41
1.16	0.40	0.46	0.48	0.44	0.45
1.56	0.34	0.42	0.45	0.44	0.45
2.45	0.42	0.50	0.49	0.41	0.43
3.40	0.37	0.45	0.49	0.41	0.43
4.76	0.30	0.41	0.48	0.46	0.44

Machine stopped to lock Epicyclic Gear.

Speed ft./min.	Coefficient of Friction μ				
	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34 lb/in^2	44.6 lb/in^2
25.8	0.34	0.38	0.39	0.35	0.35
38	0.34	0.39	0.39	0.34	0.34
52	0.31	0.37	0.38	0.34	0.34
75	0.29	0.37	0.35	0.34	0.34
95.5	0.30	0.37	0.35	0.33	0.32
116	0.29	0.35	0.34	0.31	0.32
131	0.29	0.34	0.34	0.32	0.32
156	0.26	0.35	0.34	0.31	0.31
191	0.26	0.35	0.34	0.31	0.31
245	0.31	0.37	0.38	0.33	0.31
270	0.29	0.34	0.34	0.31	0.34
340	0.34	0.38	0.37	0.33	0.32
381	0.35	0.33	0.34	0.30	0.30
476	0.37	0.36	-	0.36	-
530	0.26	0.30	0.32	0.30	0.31
830	0.26	0.33	0.34	0.31	0.32
1220	0.28	0.34	0.34	0.31	0.32
1700	0.28	0.34	0.33	0.32	0.34

There was practically no change in temperature throughout the test.

Table XII

(IV) 'Raybestos' Bonded Asbestos. Constant for machine 2.8 lb.

Speed ft./min.	Pull in lb.				
	7 lb load 7 lb/in^2	17.2 lb load 17.2 lb/in^2	27.4 lb load 27.4 lb/in^2	17.2 lb load 34.4 lb/in^2	22.3 lb load 44.6 lb/in^2
0.258	2.6	7.2	11.4	7.0	8.8
0.38	2.6	7.0	11.0	7.0	9.2
0.52	2.8	7.4	12.2	7.0	8.6
0.75	3.0	7.4	12.0	7.0	9.2
1.16	2.4	7.2	11.4	7.6	9.5
1.56	3.2	7.6	12.0	7.0	9.0
2.45	3.0	7.4	11.8	7.6	10.4
3.40	2.6	7.0	11.0	7.4	9.6
4.76	3.2	8.2	12.2	7.8	10.4
Machine stopped to lock Epicyclic Gear					
25.8	2.4	7.6	12.2	7.4	9.6
38	2.4	7.0	11.4	7.0	9.2
52	2.6	7.0	11.0	6.8	8.0
75	2.4	7.2	11.2	7.2	9.6
93.5	2.5	7.0	11.2	6.8	9.2
116	2.6	6.4	10.6	6.5	9.0
131	2.4	6.6	11.0	6.6	8.8
156	2.0	6.0	10.6	6.2	8.4
191	2.2	6.2	10.4	6.4	8.4
245	2.6	7.2	11.4	7.4	9.6
270	2.4	6.6	10.8	6.8	8.8
340	2.8	7.2	10.4	7.4	9.4
381	2.2	6.2	10.0	6.4	8.4
476	2.4	6.8	-	-	-
530	2.2	6.2	10.0	6.0	7.6
830	2.6	6.6	10.8	6.4	8.6
1220	2.2	6.4	10.4	6.2	8.4
1700	2.0	6.0	10.4	6.3	8.6

(IV) 'Raybestos' Bonded Asbestos. Constant for machine 2.8 lb

Speed ft./min.	Coefficient of friction μ				
	7 lb/in^2	17.2 lb/in^2	27.4 lb/in^2	34.4 lb/in^2	44.6 lb/in^2
0.258	0.37	0.48	0.38	0.41	0.39
0.38	0.37	0.41	0.40	0.41	0.41
0.52	0.40	0.43	0.44	0.41	0.39
0.75	0.42	0.43	0.44	0.41	0.41
1.16	0.38	0.42	0.38	0.44	0.42
1.56	0.45	0.44	0.44	0.41	0.40
2.45	0.43	0.43	0.40	0.44	0.46
3.40	0.38	0.41	0.40	0.43	0.43
4.76	0.45	0.47	0.44	0.45	0.46
	Machine stopped to lock Epicyclic Gear.				
25.8	0.34	0.44	0.4	0.43	0.43
38	0.34	0.41	0.4	0.41	0.41
52	0.37	0.41	0.4	0.31	0.40
75	0.34	0.42	0.4	0.41	0.43
93.5	0.36	0.41	0.4	0.31	0.41
116	0.37	0.37	0.39	0.37	0.40
131	0.34	0.38	0.40	0.38	0.39
156	0.29	0.35	0.39	0.36	0.38
191	0.31	0.36	0.38	0.37	0.38
245	0.37	0.42	0.42	0.43	0.43
270	0.34	0.38	0.39	0.40	0.39
340	0.40	0.41	0.38	0.43	0.47
381	0.32	0.36	0.37	0.37	0.38
476	0.34	0.39	-	-	-
530	0.32	0.36	0.37	0.35	0.34
830	0.37	0.38	0.39	0.37	0.39
1220	0.32	0.37	0.38	0.36	0.38
1700	0.30	0.35	0.38	0.37	0.38

The machine worked smoothly with this material. There is little or no rise in temperature, 60° to 100°F.

(V) The brake lining material as now manufactured is a very much improved product. A test was carried out on friction fabric which was found to be composed of Asbestos, Magnesium Compound, Fuller's Earth, and about 30 per cent. of cotton, the whole impregnated with what the maker called a secret 'Friction'. It is at least 20 years old and had been sent to the Laboratory of the Mechanics Department of the Royal Technical College for testing purposes, and had to be given up as no consistent results could be obtained. The makers claimed a coefficient of 0.4.
Using the Friction Machine the following results were obtained.

Table XIII. Constant for machine 2.8 lb.

Static μ	Speed ft./min.	7 lb/in^2 Pull lb.	7 lb/in^2 μ	14 lb/in^2 Pull lb.	14 lb/in^2 μ	17.05 lb/in^2 Pull lb.	17.05 lb/in^2 μ
	25.8	2.8	0.80	4.50	0.64	59.0	0.69
	38	2.65	0.76	4.80	0.67	5.80	0.68
0.3	52	2.45	0.70	4.80	0.68	4.97	0.59
	75	2.6	0.74	4.55	0.65	5.85	0.68
to	95.5	2.75	0.78	4.90	0.70	5.87	0.69
	116	-	-	-	-	6.05	0.71
0.4	131	2.7	0.77	4.85	0.69	5.65	0.66
	156	-	-	-	-	6.00	0.71
Using	191	2.6	0.74	4.65	0.66	6.05	0.71
	245	-	-	-	-	5.90	0.69
Brake	270	2.55	0.73	4.50	0.64	5.85	0.68
	340	-	-	-	-	5.75	0.68
Wheel.	381	2.45	0.7	4.20	0.60	5.57	0.65
	476	-	-	-	-	5.40	0.63
	530	-	-	-	-	5.20	0.62
	830	-	-	-	-	5.10	0.60
	1220	-	-	-	-	5.05	0.59
	1700	-	-	-	-	4.35	0.5

A change in the temperature from 60° to 250°F. lowered the μ - from 0.7 to 0.2 when taken in conjunction with the 17,05 lb./in^2 pressure. Running at this temperature, 250°F. for five minutes destroyed the fabric material. It would certainly be dangerous to use this friction fabric, for continuous lowering purposes on a mine incline. The material appeared fairly hard but a soaking in water almost destroyed its form, and even at low temperature the wear was excessive.

(VI) After the usual tests were carried out on this pure asbestos friction brake lining, 'Chekko', the small driving pulley was replaced by a 22-inch pulley. Calibration of the speeds of the machine gave the following:-

	Speed - ft. per min.								
With Epicyclic Gear	1.37	1.94	2.83	4.24	5.2	8.5	12.4	17.3	25.5
Without Epicyclic Gear	142	175	305	421	587	815	1360	2040	2580

It will be seen on examining the previous tables of speeds that in the portion marked 'Machine stopped to lock Epicyclic/

Epicyclic Gear' that the speed changes from 4.76 to 25.8 ft. per min. This rather big change in crawl speeds has been beautifully spanned by the last five speeds in the top line of table, namely 5.2 to 25.5 ft. per min. There is also a gain of three new top speeds 1360 to 2580 ft. per min. The author does not recommend the frequent use of these high speeds as the Epicyclic Gear was designed to run at very slow speeds.

(VI) 'Chekko' Bonded Pure Asbestos, Light Brass Bonding.

Table XIV

Speeds ft./min.	Coefficient of Friction, μ			
	21.37 lb/in^2	28 lb/in^2	35.65 lb/in^2	45.85 lb/in^2
1.37	0.46	0.43	0.54	0.54
1.94	0.47	0.46	0.55	0.54
2.83	0.48	0.47	0.56	0.54
4.24	0.48	0.47	0.57	0.56
5.2	0.48	0.49	0.61	0.58
8.5	0.49	0.51	0.63	0.58
12.4	0.49	0.53	0.61	0.58
17.3	0.50	0.54	0.60	0.58
25.5	0.55	0.55	0.60	0.59
	Machine stopped to lock Epicyclic Gear.			
96	0.63	0.64	0.61	0.60
136	0.62	0.62	0.65	0.58
142	0.56	0.63	0.66	0.60
175	0.58	0.66	0.56	0.59
305	0.64	0.60	0.60	0.55
421	0.58	0.55	0.62	0.57
587	0.56	0.49	0.53	0.50
815	0.54	0.47	0.53	0.48
1360	0.53	0.45	0.54	0.51
2040	0.48	0.43	0.52	0.49
2580	0.46	0.41	0.50	0.49

From 18 inch diameter pulley driven from the D.C. motor, the other seven speeds of this series being neglected.

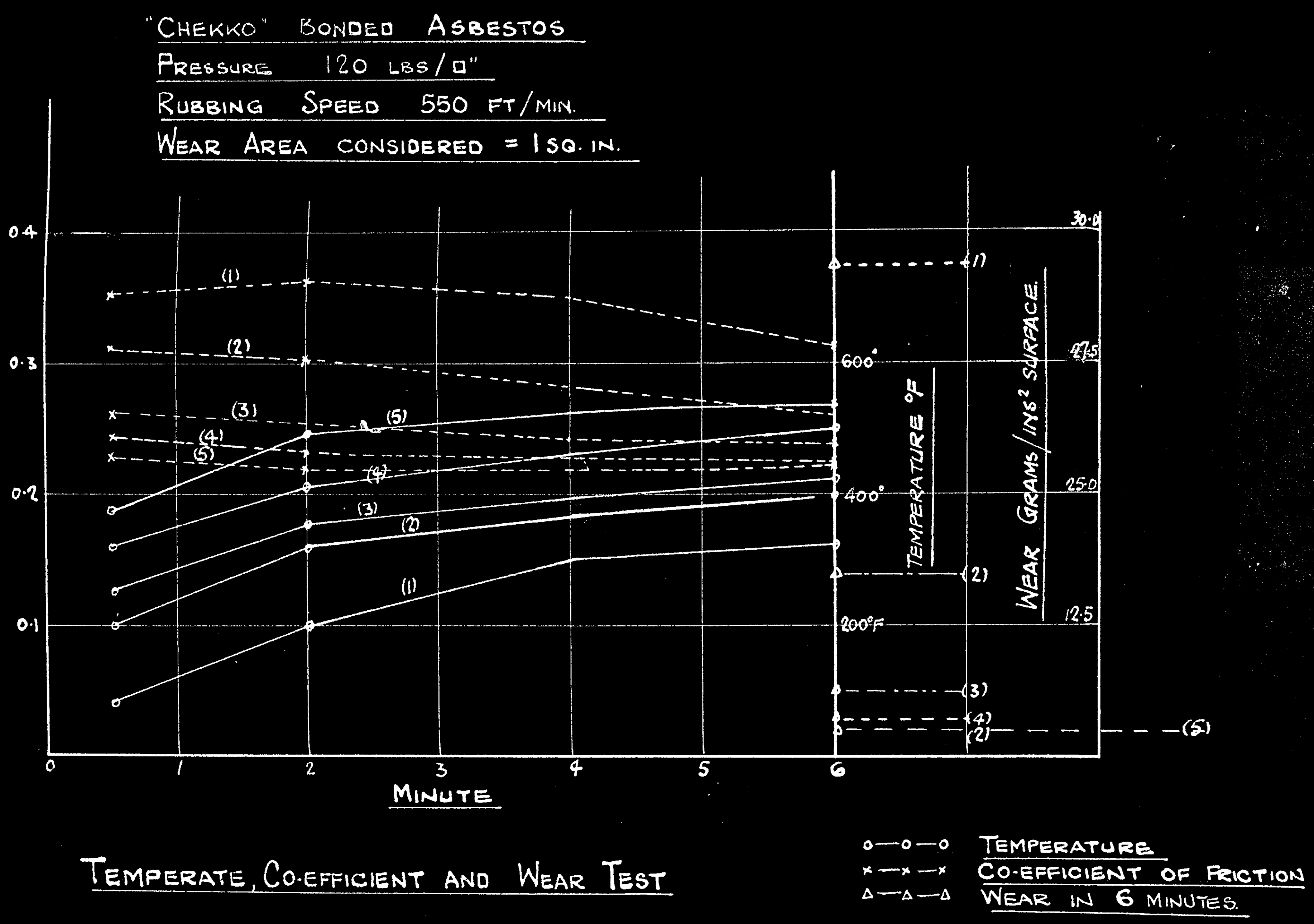
Wear Test on 'Chekko' Bonded Asbestos.

Room Temperature at 11.55 a.m. 49°F.

Micrometer Readings	before Test	1.9125	(cold)
	After Test	1.9093	(cold)
	Difference	0.0032	inches = wear

Pull on Springbalance, 9.2 lb. Zero Reading 2.8 lb.
Balance arm, 2 : 1.

Speed 2000 ft. per minute. Load 21⅜ lb.



"CHEKKO" BONDED ASBESTOS
PRESSURE 120 LBS/□"
RUBBING SPEED 550 FT/MIN.
WEAR AREA CONSIDERED = 1 SQ. IN.

TEMPERATE, CO-EFFICIENT AND WEAR TEST

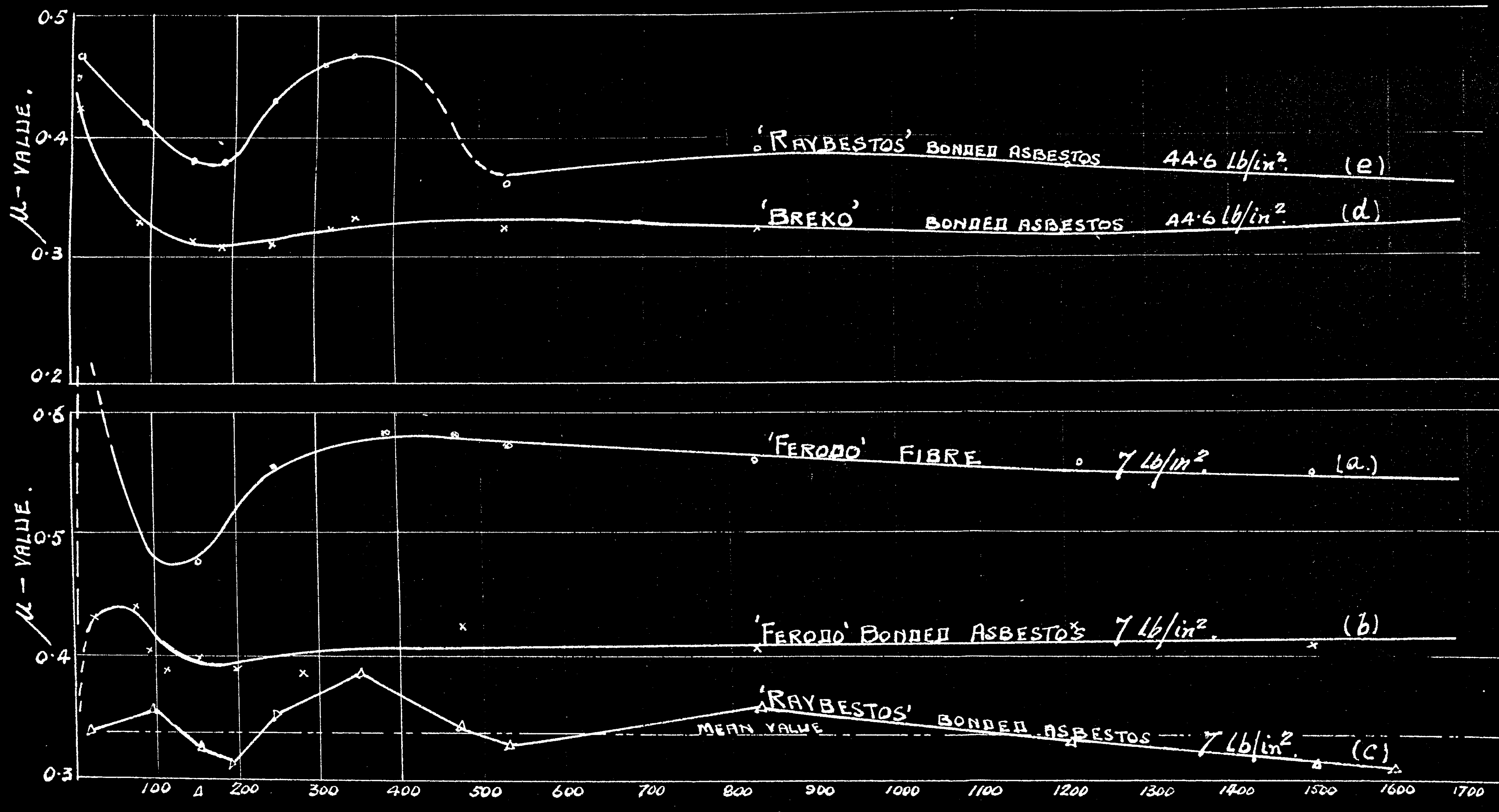


FIGURE 9 — a.b.c.d.e.

Time	Spring Balance Pull, lb.	Temperature	value. μ
11.55 a.m.	9.2	49°F.	0.6
12	9.4		
12.15 p.m.	9.35		
12.30	9.25	65°F.	
12.45	9.15		
1	9.15	70°F.	0.58
1.15	9.15		
1.30	9.15	75°F.	
1.45	9.15		
2	9.15	75°F.	
2.15	9.10		
2.30	9.05		
2.35	9.05	76°F.	0.56

Load changed to 36.65 lb. μ

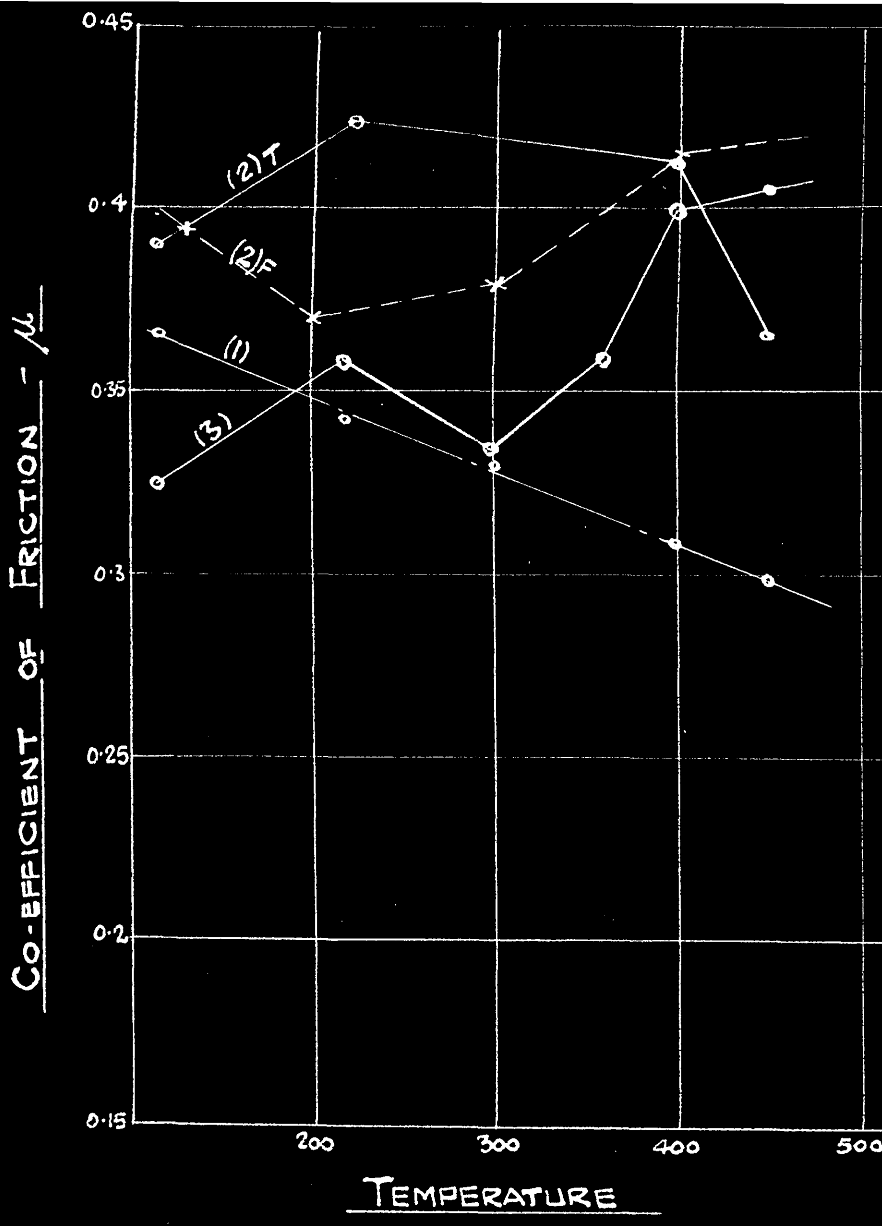
2.50 p.m.	12.45	75°F.	0.53
3	12.55	80°F.	
3.10	12.58	85°F.	0.55
3.40	12.55	95°F.	
4	12.55	100°F.	0.54
4.20	12.5	105°F.	
4.40	12.45	110°F.	0.53
4.50	12.45	110°F.	
5	12.45	110°F.	0.54

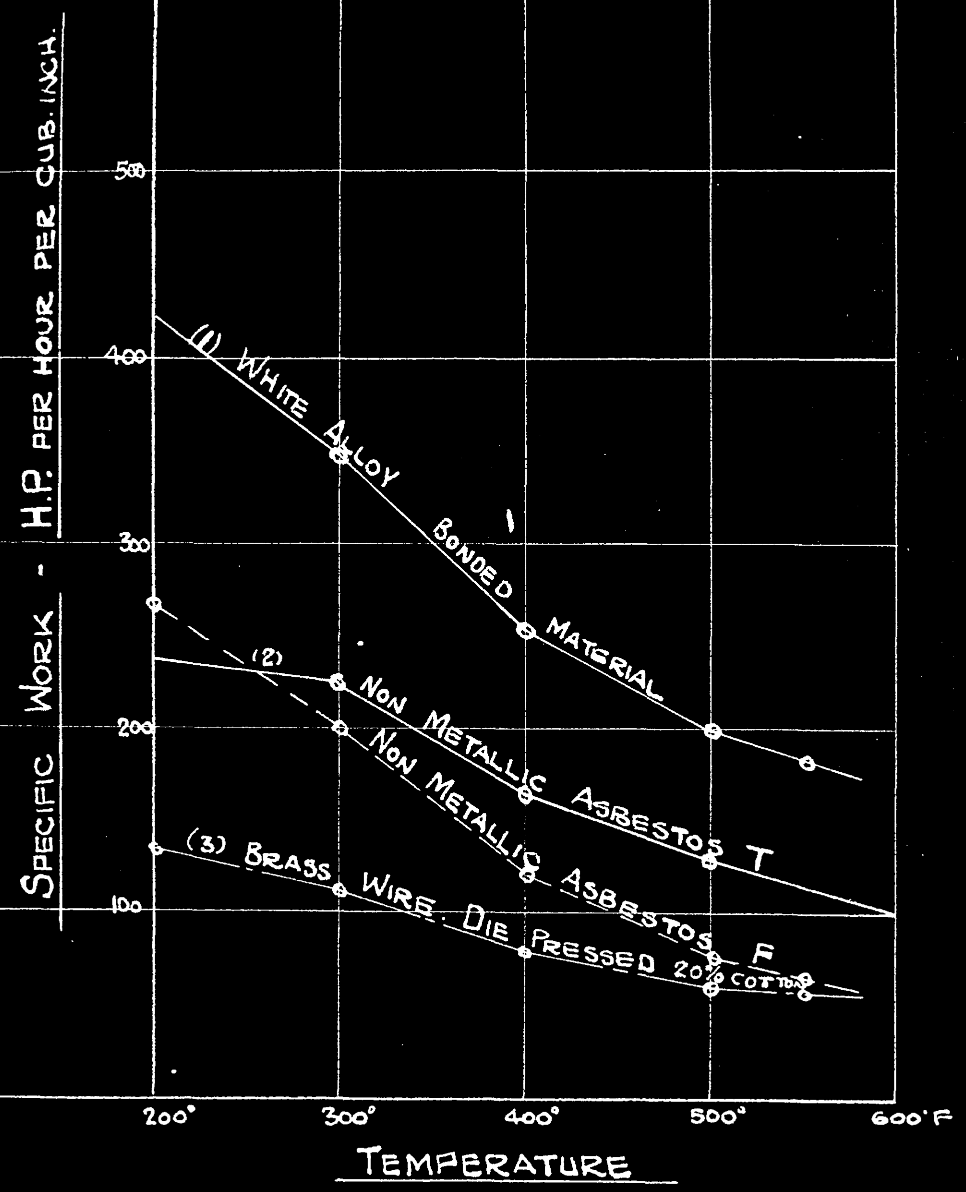
Room Temperature 52°F. Micrometer (1.9093 - 1.9089)
= 0.004 inches.

When the 'Chekko' material is to be supplied for clutches the makers prescribe a specific loading 12 to 20 lb/in². In order to carry out temperature tests, taking two different loads and speeds, four bunsen flames were arranged round the interior of the rim supplied the heat. It was afterwards found that these required to be augmented by the aid of two blow lamps, and the brake wheel had to be shielded from the cold draughts as much as possible.

'Chekko' Bonded Asbestos. Micro. readings, 1.9207.

Time	Temperature of Rim °F.	Coefficient of Friction, μ	Remarks Dia. 26"
2.45 p.m.	110	0.58	Surface dry, Speed 1360 ft/min.
2.51	196	0.56	Specific load, $21\frac{3}{8}$ lb/in^2
3	238	0.53	Wheel partly enclosed
3.20 p.m.	242	0.52	Speed 587 ft./min.
4.10	163	0.70	Specific Load 7 lb/in^2
4.18	70	0.647	Wheel not enclosed.





Wear in terms of Horse Power Hour per unit volume.

The excessive wear during the first period of wear-tests is more noticeable with the woven material than with the highly pressed(under temperature) moulded material. The latter can be machined and does not require much running-in, since the bearing surface is more regular than with the more soft woven and impregnated material.

From the average values of a large number of tests on woven fabric pure asbestos the curves in Fig. 9 have been plotted. These was excessive wear at first, which gradually dropped down to an almost stationary value of wear; the curves show that the rise in temperature is almost constant; while the value of the coefficient of friction falls uniformly over the first four tests, 24 minutes, and then remains almost constant over a long period test. The period of time between each test was a half minute, and the friction wheel was protected from cold currents.

Between room temperature and 300°F. the wear appears to be directly in proportion to the amount of work dissipated at the brake wheel rim. Above 300°F. the wear is excessive and grows rapidly for all Woven Bonded Fabrics; it is in this region that fabrics containing cotton become charred. The wear for nearly all fabrics is twice as great at 520° as at 260°F. for similar speeds.

Effect of Change of Pressure and Speed, but pv = constant.

During the tests with constant temperature regulated by the bunsen gas flame, the amount of wear was unchanged by adding load and decreasing speed, or decreasing load and increasing speed, to give equal works (p.v.). With a fabric material having a low μ-value it is found better to work with/

with a low p-value, say 10 to 12 lb/in^2 maximum and a high rubbing speed V. With the high coefficient of friction fabrics a pressure of 30 lb/in^2 is found to be more suitable with lower rubbing speed. Tests carried out on a drop-hammer ferodo brake lifting tup showed that, when 50,000 ft. lb. per sq.in. per minute was demanded from lining, the material wore out very rapidly with air cooled brake rim pulley, but with water cooled brake pulley the wear was not excessive. All tests carried out on the various fabrics show that it is not advisable to design for more than 12,500 ft.lb. per sq.in. per minute. A graph showing specific work, H.P.hr. per sq.in. plotted on temperature $^{\circ}$F. is given at Fig. 10.

Effect of Rapid Heating on μ-value.

A new brake lining having a μ-value = 0.48 at$^{\circ}$350 F. is applied to the brake wheel which when running has the temperature raised to nearly 350°F. before applying the brake. After the brake is applied the temperature rises rapidly to 575°F., time approximately 10 minutes. The μ-value of the material is reduced to 0.18 and may be taken on average as 0.2 for 20 minutes before disintegration of lining material takes place. The same material at 425°F. has μ-value = 0.4 but when raised to 575°F. in 10 minutes has still a μ-value = 0.35, and never goes under μ = 0.3. In both cases the μ-value increases as disintegration of lining takes place caused by the binding material becoming plastic. It was also found that the more slowly a brake lining is heated up the less likely is the bonding material to leave the fabric. The graph, Fig, 11 , shows how the coefficient of friction varies with temperature in a number of fabrics. Taking an average of five different makers' linings, the maximum T-value is about 250°F. and the minimum 400°F.

Effect/

Effect of Roughening Surface of Brake Rim. -

Some cotton and duck fabrics have been tested which were made with a sulphur greasy compound incorporated while under pressure. It was found that oxidization took place, and smoothness of action soon disappeared. Where cast iron drums have been roughened by a file it has been observed that the fabric lining suffers very severely, and after a relatively short time, it is unfit for use. An iron cement was applied to a small pulley rim and Ferodo Fibre was used as a brake. Under the application of a very light pressure wear was excessive and heat generated rapidly, showing that in brake linings abrasion material would be detrimental.

Effect of Applying Water or Lubricants to Brake Wheel.-

In this set of experiments,"Raybestos", bonded asbestos brake lining is used. The speed is kept (approx.) constant at 1000 ft. per minute, V, and with a pressure of 45 lb/in^2 the μ - value is 0.38. Raising the temperature of the brake rim is followed by a drop in the value of the coefficient of friction. Removing the bunsen burners and applying water to the surface of the brake rim lowers the μ - value considerably and as long as the water is allowed to wet the brake surface of the wheel this coefficient becomes smaller. On removing the water and again applying heat the μ-values rises as the moisture dries off and is noted to be greater than original starting value. A series of these changes are shown, Fig.12, in order to depict the various changes; shocks or shudders are seen on the brake wheel's motion.

When moist conditions prevail during tests the following changes are noticeable:

(1) that the high rubbing value of the friction fabric materials change when working on a water lubricated surface ;

(2) that μ - values for all materials are lowered by one third dry test values;

(3)/

(3) That a thin coating of rust or iron-oxide has been found to lower the μ-value by one half.

The pad pieces, in one test, after being fitted to the curvature of the brake drum were allowed to soak in water for three days. The fabric material was dryed on blotting paper and placed in the holder. It only required a second or two to reach the maximum μ-value yet there certainly was a distinct lag in reaching μ-value found in previous tests. The lining was not destroyed by the moisture and it did not appear to affect the lining after the first test run which pointed to the fact that the surface moisture, rather than penetration of water into the bonded fabric, was the cause of the temporary change in the value of the friction coefficient.

To prevent oil or water drip reaching the rubbing surface, rubber coating in the form of a solution has been tried on the brake lining leading edge. i.e. the first edge opposing the motion of wheel. Experiments on pads with rubber solution on edges were not carried to any length, but showed clearly that fluid could be guided past the brake surface by this simple contrivance.

Effect of Lubricant on Fabric Material.

Apart from the lowering of the μ-value, lubrication did not destroy in any way the die pressed fabric material, which, freed from oil, soon regained its normal coefficient of friction value. On heating up the brake wheel when tallow had been applied to the brake-pad surfaces, very erratic gripping was shown on the wheel's motion. The passing from lubrication film to boundary lubrication and then to dry condition could clearly be seen. It was not intended in this investigation to try out the different lubricating oils tested in the author's second investigation, but as a matter of interest a set of brake pads was sparingly treated/

treated with Bayonne oil, castor oil, and tallow. The mineral oil was soon rubbed off and the coefficient of friction of the fabric changed rapidly from 0.1 to 0.35. The tallow appeared to last longer as a lubricating medium with a coefficient of 0.15 rising to 0.35. The castor oil did appear to affect the lining but in rather a peculiar way as it gave a friction coefficient reading of nearly 0.2 and at one time this value reached 0.41 showing that this oil had acted as a dressing; on running for some time the value fell to 0.35 which was the dry coefficient value of the fabric undergoing test.

On the friction machine experiments have been carried out with dry and tallow coated cotton or duck belting. As this is a fabric, which when impregnated with rubber, has been used for brake linings, a few readings are given; fuller tests on this subject from a different point of view have been made at the Royal Technical College by Laird[7]. This paper deals with "Viscous effects in Dynamometer Belts", and the author in reading over this paper for publication purposes soon discovered that Laird's investigations pointed to the fact that the following formula

$$T_1 = T_e\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right), \text{ where } T_e = T_1 - T_2$$

effective tension, was not easy of manipulation, since the difference in tensions depends on belt creep and slip, arc of contact between belt and pulley face, character of pulley surface and belt surface, also velocity of belt. In the dry cotton belt tests a slight formation of oxide on the brake rim gave variable readings; the humidity of the atmosphere, or the least trace of moisture in the room caused variations in the readings. On the introduction of another variable, namely a tallow coating on the cotton brake band, made $T_1 - T_2$ still more variable, and the usual belt/

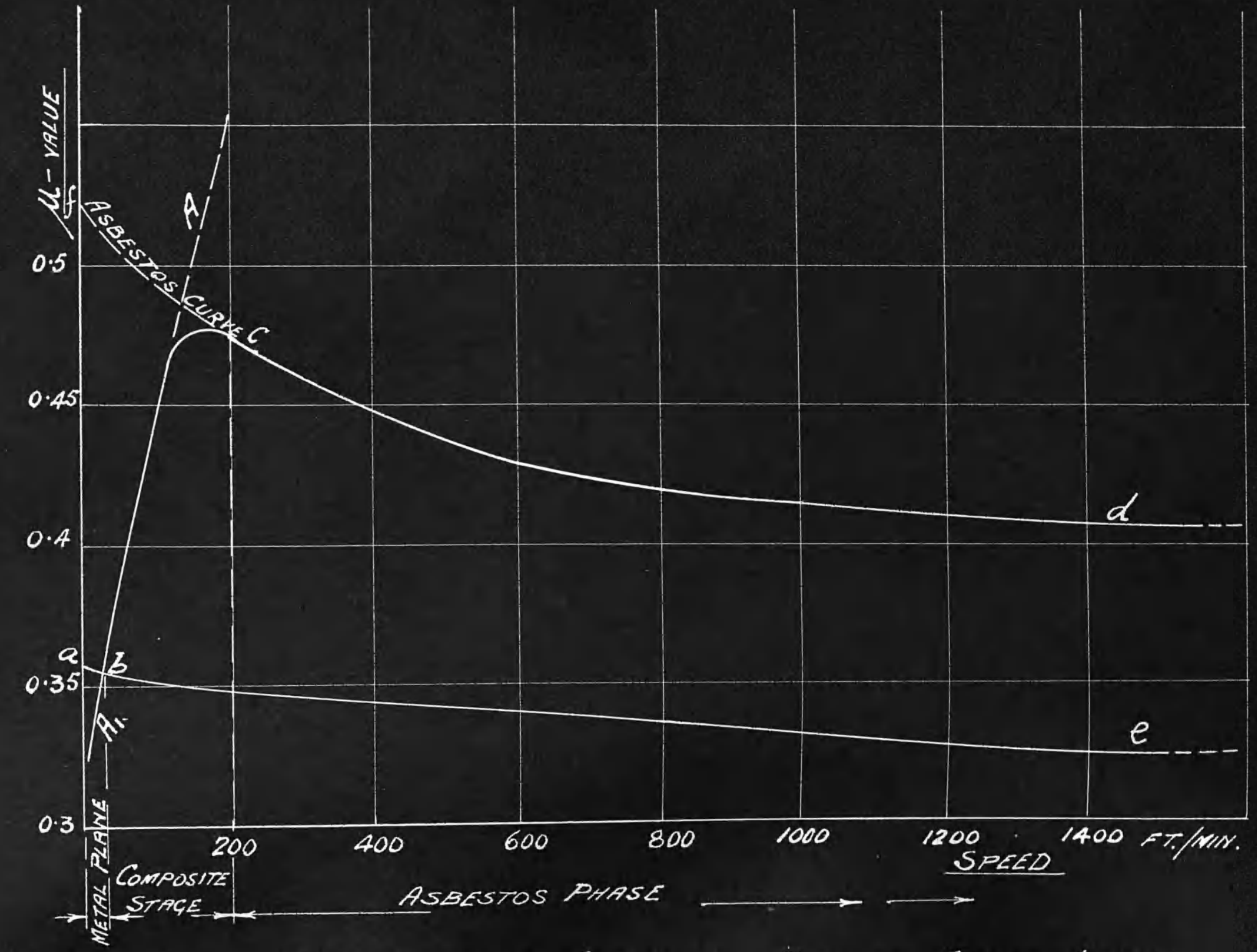


FIGURE THREE PHASE CURVE FOR BONDED FABRIC LINING.

Fig. 13.

belt formula $\frac{T_1}{T_2} = e^{\mu\theta}$ was not applicable as the test results show. From the curve plotted from experimental data Laird gave the formula $T_1 - T_2 = K(\lambda A V W)^{\frac{1}{2}}$ where λ = coefficient of viscosity; A is the area of film in shear or area of contact between the belt and the wheel; V is the speed of slipping; and W is calculated from end tensions as $\left(\frac{T_1 + T_2}{2}\right)\theta$.

The resemblance between the formula given in the author's investigation of lubricants submitted as part of this thesis is very noticeable.

(1) Laird's $(T_1 - T_2) = K(\lambda A V W)^{\frac{1}{2}}$

(2) author's $\mu = K\left(\frac{\lambda V}{2P}\right)^{\frac{1}{2}}$ lubricated bearing.

(3) Boswall's $\mu = K\left(\frac{ZN}{P}\right)^{\cdot 58}$ for clearance bearings.

(4) Dry belting $(T_1 - T_2) = e^{\mu\theta}$

If the testing of lubricated cotton belting entails the consideration of film lubrication with all its inherent difficulties. Fabric Brake Linings, which are made from bonded asbestos (plus in some cases 10 to 30 per cent of cotton), brass wire (which does not always have the same bearing surface), and bonding material (with varying viscosity), added to these the possibility of oil or water lubrication on brake wheel rim, have given to this investigation a most difficult number of variations and complicated combinations of variables.

Considering the curve, Fig. 13 which has been taken from a bonded asbestos lining, there are three distinct phases in this curve which may be denoted by the (1) Brass or Metal plane, (2) Composition Stage, (3) Asbestos Plane; this is assuming no cotton mixture in fabric.

The portion f.d. represents the curve of asbestos alone.

The portion a.e. represents the curve of brass or metal alone, where a,b,c,d is the actual curve obtained for the material. The first portion of the curve, namely line AA appears/

appears to be determined by the intensity of the loading, being near to the axis the higher the loading becomes. The metal curve shows little change with increase of speed of rubbing, but in the "composite stage" the μ-value increases rapidly with the increase of speed. The asbestos part of curve decreases with the increase of speed, very rapidly at first and then becomes more nearly constant.

(1) for metal μ = 0.3 to 0.38 average 0.34 depending on the metal used.

(2) composite material (metal, asbestos plus bonding material) μ increases at the rate of 0.15 per 100 ft. per minute.

(3) for asbestos $\mu = 0.43 \div \text{speed}^{0.03}$, this being an average derived from curves with 7 lb and 28 lb per sq.in. pressure loading; the equations for the curves, Fig. 13 a. on Fig 20 being

(a) $\log \mu = -0.35 - 0.02 \log \text{speed}$.

(b) $\log \mu = -0.38 - 0.04 \log \text{speed}$.

Using as a lubricant machine oil changes the coefficient in the phase (1) from 0.3 to 0.1 in four minutes, and the asbestos when lubricated still lowers this value and at the end of another period of five minutes the μ-value has become 0.02. In the case of pressed block linings the lowest value reached was 0.08 which is almost equal to metal on metal. Curves, Fig. 14 [p.45] have been drawn to illustrate the changes which take place in temperature and μ- values when lubricant is used on the Brake Wheel.

CONSIDERATION OF RESULTS ON BRAKE LININGS

From the foregoing experimental results given in this investigation it will be seen that a wide variation of results is due to the variable nature of the combined materials and their method of manufacture. The Asbestos Fibres are washed with mineral acids in order to make the fibres more absorbent, and also to prevent co-agulation of rubber while they are being impregnated with latex. The percentage mixture is usually/

usually 80 asbestos to 20 rubber; the latex usually forms 30 per cent. solid content, and vegetable fibres may replace asbestos in part. Sulphur may also be added. This in the 'Investigation on Lubricants' has been placed among the materials useful for carrying away heat from an over-heated journal bearing. The mixture of materials, Asbestos, Rubber, Sulphur, etc. is made into a dough; this in the case of the Asbestos Bonded Brake Linings is forced into the meshes of the wire fabric, and put into moulds in the case of moulded linings without metal. After being allowed to set at a moderate heat, 150°F., in a vacuum drier, the slab or mould is compressed to about $^1/_3$ of its initial thickness. The slabs and the moulds are kept heated while under hydraulic pressure, and this tends towards constancy of finished material. The temperature of the moulds/vary from 275 to 400°F. and the hydraulic load applied may be such as to give 50 tons.

(1) These linings have a hardness of Rochwell No.B. 95 and can be machined. Thus when a brake strap arrangement is properly designed to give uniform pressure, the lining may be made to fit closely on to the brake wheel. A fault found in carrying out experiments with fabric material made in 1926 was the lack of smooth surface.

(2) The discarding of the metal binder is another disadvantage brought out by this investigation owing to quantity of metal surface exposed to rubbing wear not being uniform.

(3) The material as now manufactured is not destroyed by exposure to water or oil, but this does not mean that an oiled surface can be used for a heavy load brake or friction clutch. The fabric material behaves in the same manner as all lubricated metal surfaces subjected to running speed and pressure/

pressure. The asbestos composition can be used for clutches designed for working in oil. This is the material used in combined clutch and change-speed lever gear.

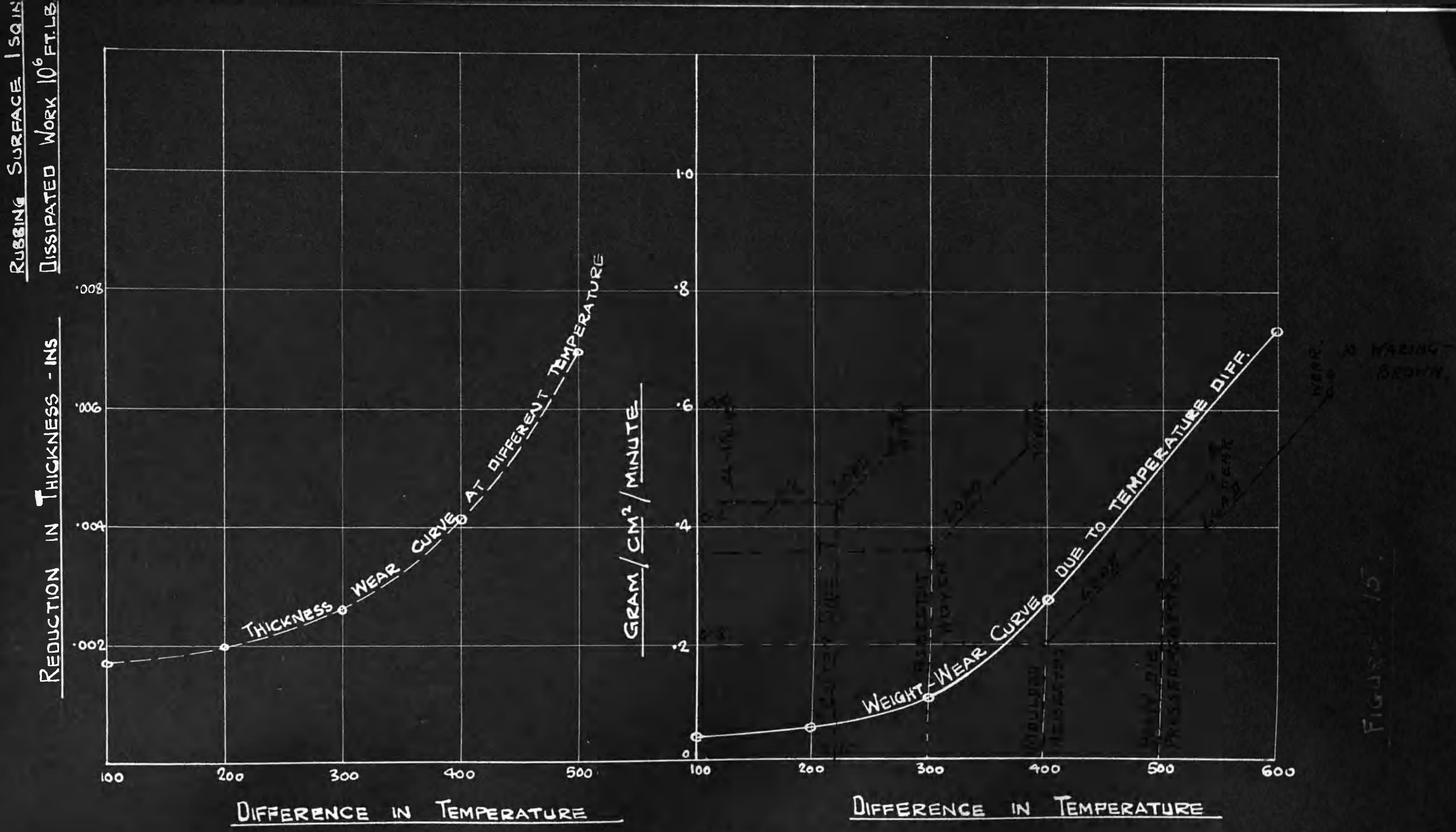
(4) Temperature effects are such that with many built up fabrics excessive wear takes place at high temperatures. The temperature, with the newer hot-pressed linings, may now reach 500°F. without destruction.

(5) Wear, which was measured by change of thickness in a given time, or by change of weight over a given period of rubbing, has been measured by absorption of energy per unit of volume (one cubic centimetre) and expressed in horse-power hours per cubic centimetre or ft.lb. per min. per sq.in. Fig 15

This method of measuring wear could be stated as an index of the performance of brake linings, and could be made a basis of comparison, horse power hours per cubic centimetre, but must not be taken when making a comparison between service brakes fitted to machines where excessive pressure may be applied intermittently and for short periods. The brakes are never allowed to cool but are used at peak temperatures where wear is excessive.

(6) The nature of the surface of the brake rim is a deciding factor. In the main tests a cast iron brake drum was used, but moulded type of brake lining has been proved more suitable to use with low carbon steel pressed drums than bonded asbestos. Metal, introduced to strengthen the yarn, causes seizure to take place which is followed by a fluctuating frictional value which results in 'snatch' or 'shudder' and brake drum scoring.

At slow crawl speeds 'snatch' and 'shudder' is very noticeable in the case of high μ-value fabric material. While the low μ-value material such as Breko, with its low steady frictional value of 0.28 to 0.25, provides a smooth and fairly/



WEAR CURVES.

Figure 15

fairly rapid retardation. This class of material, used with a low bearing pressure, was found to be good for electric lifting cranes and hoisting machinery, a fact confirmed by tests carried out by Messrs. Adamson at their crane works.

CONCLUSIONS ON BRAKE LININGS

This investigation had for its aim the comparison of fabric linings as manufactured, a grouping together of test results as required by the engineer, an encouragement to standardization of tests, and quality of material used. Since the first results of these tests[6] were published 1930, manufacturers have improved their running tests, and guided by results obtained by these, and other experimenters' results on durability when consideration is taken of duty imposed they have gone on creating new and better materials suited for carrying heavier loads, increasing speeds, and stopping powers when suddenly applied to moving machinery.

Much of the data given in this investigation are applicable to Friction Clutches and Friction Spur and bevel wheels. The only difference is in the range of pressures applied in clutches and wheels which may vary from 25 to 75 lb. per sq.in., whereas in brake linings the pressure on an average is about 15 lb. per sq.in. with a maximum of 30 lb. per sq.in. Even a lower pressure 12 lb. per sq.in. has been used by crane makers, and suggested by test results obtained in this investigation. .The brake drums and brake shoes used in practice, even when made of pressings are not flexible. The brake linings in all the tests described are "worn in", that is, in applying the brake all points must touch the drum at the same moment. In applying the brake the lining is being compressed, the compression being in proportion to the pressure which is exerted on the lining (Hooke's Law), or the sum of all friction forces $\Sigma F = \mu \Sigma N$.

The higher the self-actuation the more sensitive is a brake to change in friction coefficient. Therefore, if too high/

high a degree of self-actuation is used in a brake design, the brake is liable to "grab" in some cases, while it will not hold sufficiently in others. That is why it is much more important to obtain a brake lining which is affected as little as possible by temperature, moisture, speed, etc. than to obtain a high frictional coefficient. There appears to be no way of judging the self-actuation of a brake shoe. Different designs show different features and makers of brake linings are now inclined to specific brake linings to be used with certain designs.

Fabric Material used for Gearing.

When gear wheels are made from fabrics, such as 'Ioco' and 'Fabroil', consideration must be given to the Tensile Strength of the material, to the hardness of the fabric on the root surface of the tooth and to the wearing surface of the material. The duty of wheel teeth is to transmit power with least possible loss. The efficiency of the gearing depends on lubrication, which is quite different from Fabric Brake Lining which seek to use the dry rubbing surfaces, sliding friction, without seeking to use seizure action which would destroy the surfaces. In gear teeth the action is part rolling and part sliding friction with line contact.

Comparison is made between material used for wheel blanks.

(1) Fabroil pinions which are made from compressed cotton (specially prepared cotton fibres). This material takes the place of compressed paper and wood pulp blanks. In this case there is no binder used and compressor therefore has to be maintained by means of steel side plates, or shrouds; if these are removed the Fabroil is useless. In this class raw-hide pinions and pinions made from Ioco sheet may be classed.

The nature of these fabrics made testing on the Friction/

Friction Machine difficult as the cutting up into ½-in. squares to fit the fabric holder, Fig. 3 was almost impossible. Test carried out on this material showed that the dry friction was rather erratic, varying from static friction 0.25 to running friction 0.6, and again falling rapidly to 0.25 as heat brought out the dressing.
When lubricant was applied the μ- value dropped by about $\frac{2}{3}$ giving a μ-value of 0.16 to 0.2. (Tallow lubricant). Castor oil dressing when applied to the friction pads was inclined to cause a viscous drag.

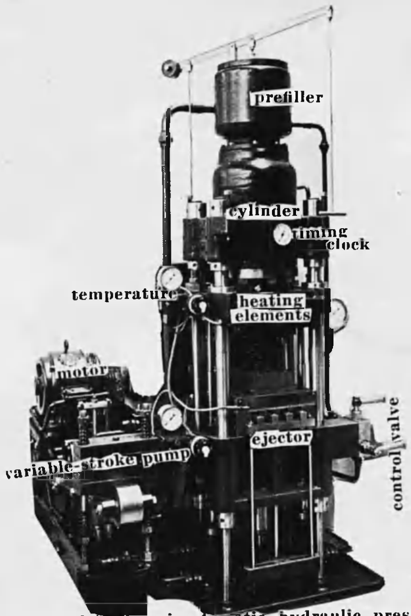
(2) Fabroil-A cut very easily into ½-in. squares and when run on the Friction Machine gave smooth running and taking on a beautiful polished surface which worked well with cast iron. The lubricated friction tests gave a value for μ = 0.1 to 0.06. A high pressure could be applied to this material, and oil did not tend to destroy its structure even when running against the grain of the texture, yet it does not appear to have the strength of the shrouded pinion.

(3) Ferodo bonded asbestos could not be used for gearing but for light duty Ferodo fibre could be used.

(4) Ioco is made in much the same manner as Fabroil the only difference being that in a semi-plastic state the woven cotton duck strip is impregnated with synthetic resin and compressed in steam heated moulds under pressure.
It is finished as it comes from the moulds except for slight trimming at the parting line in gear blanks, and curing for about 20 hours at 212°F. Arrangement of Press is shown in Fig. 16.

(5) Cotton cloth impregnated with Bakelite is pressed into heated moulds to make turning gears for oil pumps. Duck cloth has usually a tensile strength of 24 to 30 lb. per inch width per ply.

Table/



A 50-ton semi-automatic hydraulic press for plastic moulding, with motor-driven variable-stroke automatic pump.

Figure 16

Gear Wheel
Built ups
and Shrouded

Rockwell Tests on
10 Blanks
Max -86 B. } average 71
Min -69 B }

Figure 17

Table of comparative strength and hardness.

Material	Ultimate Tensile Strength	Izod ft lb.	Hardness	
			Max.	Min.
Fabroil-A $\frac{7}{8}$" thick	4 tons/in^2	2	48.8	44.4 Brinell
Formapex $\frac{1}{2}$" thick	Tensile 5$\frac{1}{2}$ tons/in^2	3	36	34 "
Micarta	7 tons/in^2		(45	41 "
	(Shear) 2 tons/in^2		(99	93 Rockwell B
Ferodo Fibre $\frac{1}{2}$" thick	2 tons/in^2	1	98	Rockwell B
Formapex Gear Wheel Blank	3 tons/in^2 at boss and rim	1.5	78	77 Rockwell B.
	5 tons at web.	2.5	82	73.5 "
Vulcanized Fibre	7.5 tons/in^2	3.5	75	70 Rockwell B

The Young's Modulus for all fabrics is approximately 1×10^6. The blanks can be machined by using tools with large top rake; circular milling hobs with steep top rake are used for cutting teeth in blanks, and grinding may be done preferably by a rapidly rotating fine-grained emery wheel, (e.g. No. 40 emery, 5 inch diam, 3000 r.p.m.), taking care to avoid undue heating of the fabric. These features were clearly brought out during the machining of the various test pieces.

Cross-Breaking Strength or Maximum Fibre Stress.

For most practical purposes this is defined as the load, P, required to fracture, by bending, a bar-shaped test-piece of given rectangular cross section (b = breadth, d = depth), when this is supported or held horizontally and the load applied vertically in one of the following forms of test:-

(a) Cantilever Test: The bar is held by a clamp at one end and load applied at a distance, L, from the near edge of the clamp. Stress = $\frac{6PL}{bd^2}$. Iozo, 0.5" x .75" x 2.5". Safe load, 25 lb.

(b)/

(b) 3-point Loading test: The bar is supported at two points a given distance, L, apart, and the load applied midway between them. Stress = $3PL/2bd^2$, Fabroil-A = 0.5" x .75" x 2.5", Safe Load, 95 lb.

Impact Strength

(c) This is expressed as energy per unit area of cross section of the test piece where fracture occurs, but this test varies with the shape of the cross section. This gave approximately 2 to 3 ft.lb. for all fabric material.

Plastic Yield Test - (d): Test as in (b) at a given temperature (200°F.) for a given time 9 minutes, or 80°F. for a period not less than 12 hours. One of the Fabroil-A pieces was turned to $\frac{1}{4}$" diameter at ends and a plain portion $\frac{1}{8}$" diameter for 2" long. This test piece was subjected to torsion and gave on an average 10000 lb/in^2 (max. 13000 lb/in^2, min. 9000 lb/in^2).

Absorption Tests - The Vulcanized Fibre, although like the other fibres tested was insoluble in ordinary solvents, was seen to absorb water (swells). In this respect Ioco and Fabroil materials are better than Vulcanized Fibre, wood, raw-hide, etc.

Formapex Miocarta and Fabroil-A gears and blanks are sent out from factory with metal centres which are permanently held in position without rivets. Sketches of a blank and a cut gear are shown in Fig. 17. These can be made much stronger and lighter than when cut from slabs.

From the formula $HP = \frac{0.0001 \times V \times f \times b \times y}{D.P}$ derived for Ioco material, let breadth of face of wheel b = 1 inch and the diametrical pitch D.P. = 1, then the formula for horse power becomes HP = .0001 x V x f x y per inch of face per 1 diametrical pitch. Assume pressure angle $14\frac{1}{2}^{\circ}$, then $\log HP = \log 0.0001 + \log V + \log f + \log y$. To make an alignment chart to suit this logarithmic formula of/

of four variables, let log r = log V + log f.

If V varies from 100 to 2000 ft. per sec. then log 20000 - log 100 = 1.301. magnify this by 5, say 6.505"

The safe stress of the material for different speed may vary from 1000 to 4000.

Then log 4000 - log 1000 = 0.602. magnify by 10, say 6.02".

The form factor will vary from 0.075 to 0.118.

log 0.118 - log 0.075 = 0.2. magnify by $1/30$ say 6"

If the distance between scale V, (S_v), and scale f, (S_f), = 3"

then scale of r will be distant from V by e"

$$\text{when } e = \frac{S_f}{S_v + S_f} \times 3 = \frac{\frac{1}{10}}{\frac{1}{5} + \frac{1}{10}} \times 3 = \frac{\frac{3}{10}}{\frac{3}{10}} = 1''$$

and scale of r, (S_r), = $\frac{3}{10}$

Again place scale y at distance 3" from scale r,

$$\text{then } e = \frac{S_y}{S_r + S_y} \times 3 = \frac{\frac{1}{30}}{\frac{3}{10} + \frac{1}{30}} \times 3 = \frac{3}{10} = 0.3$$

Then scale for H.P. (S_{HP}) = $1/3$

$$\text{Substitute in formula } HP = 0.0001 \times \overset{V}{2000} \times \overset{f}{1000} \times .118$$

$$= 23.6$$

$$\text{and } HP = 0.0001 \times 100 \times 4000 \times .075$$

$$= 3.$$

The chart has now been drawn, Fig. 18 from this data and gives any of the four variables when the others are known. This chart has been improved by adding the scale for the pitch diameter of the wheels, the number of teeth and the diametrial pitch. The formula which gives these dimensions and numbers are

(1) $\frac{(\text{diam. of gear})\ \text{r.p.m.}}{12}$ = Velocity in ft.per min.

$= 0.26\ d \times \text{r.p.m.} = V$

(2) circumferential pitch $p = \frac{\pi}{D.P.}$ and $np = \pi d$

Therefore $n = d \times D.P.$

From/

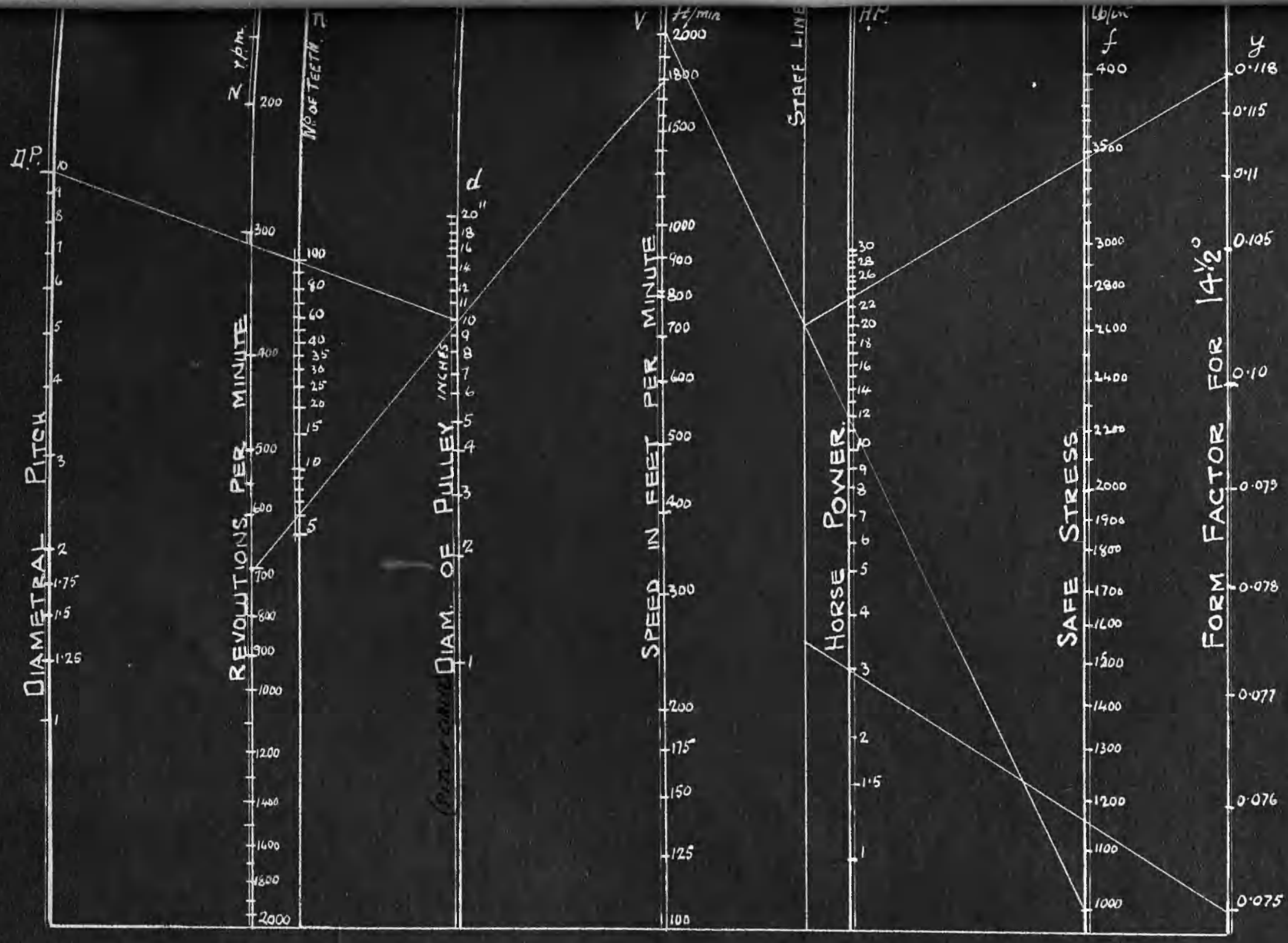


FIGURE 18 — NOMOGRAM FOR "FORMAPEX" MIOCARTA (BAKELITE) GEAR WHEEL —

From (1) $\log d = \log V - \log .26 - \log N$

From (2) $\log D.P. = \log n - \log d$

By the same method the scales N and D.P. have been added. These could have been drawn separately but by the addition of key-lines the scales are clearly connected.

The author was asked by Professor Mellanby to furnish the manufacturers of Ioco Fabric with a formula to suit their material, and this has been added as a matter of interest, and is a practical illustration of the ease with which nomograms can be adopted to test results. An experimental apparatus was also designed to test the endurance of fabric gearing when running in conjunction with cast iron wheels.

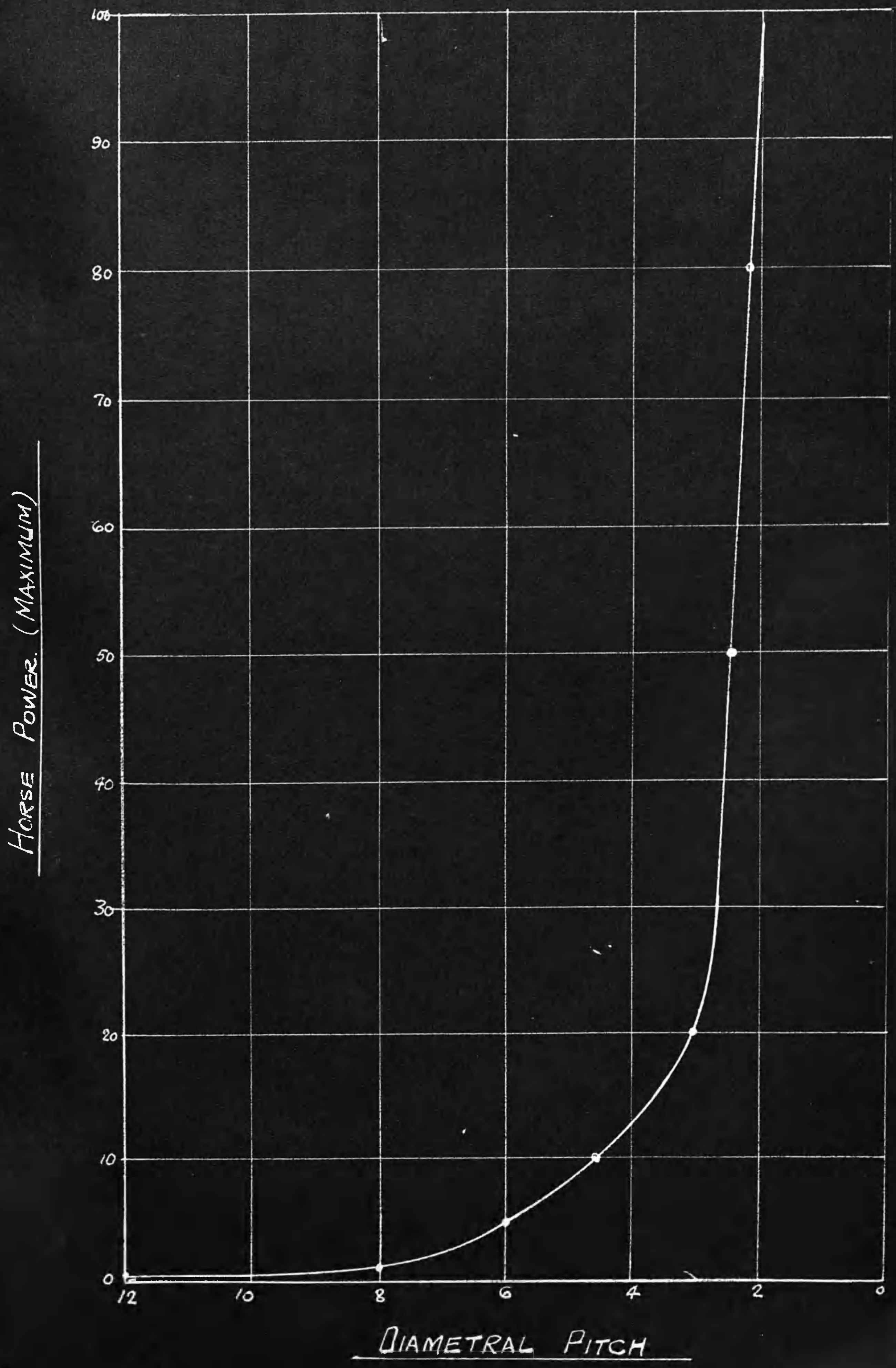
For the first run-in of fabric gearing it is advisable to use a solid lubricant such as a mixture of vaseline and graphite, or colloidal graphite. A thin coating of this lubricant during the run-in period helps to make a polished smooth surface, afterwards the gears may be run in a bath of good lubricating oil.

The fabrics might be made to suit the chuck of a reversal of bending stress machine, and could be subjected to a sliding motion abrasion as a speedier method of testing the gear blank pressed fabric material. This would certainly give a very much shorter test than was recommended to the Ioco Company.

From the alignment chart the table and graph horse powers for various diametrical pitch have been drawn up for easy application, Fig 19.

Diametrial pitch	10	8	6	5	4	3	2½	2
H.P.	¼-¾	1-2	3-5	7.5	10	15-25	50-60	75-100

This investigation was supplemented by work done on the strength of various plastic moulded material for hardness and strength/



FABRIC MOULDED CUT GEAR BLANKS.

FIGURE 19

strength, the number of which has considerably increased since this research began in 1929, but as the Thesis has become rather lengthy it was thought advisable to leave this for a future research.

In previous papers the author has thanked Professors A.L. Mellanby and Professor Wm. Kerr for permission to carry out these tests and here again he desires to thank them for guidance and encourggement during the many tests carried out. To the different manufacturers of fabrics supplied for these tests the author tenders his thanks, and also to the Governors of the Royal Technical College for the freedom to use the power and plant erected in the College Laboratories of the Mechanical Engineering Department.

APPENDIX (application of data to road and rail)

The importance of Braking Systems can be well understood when the problem is considered of arresting as much as 50,000,000 ft.lb. of energy at 90 m.p.h. in the space of 40 sec. (or about ½ mile). The value of μ – varies widely with apeed, decreasing rapidly at the higher speeds where it is most needed.

(1) Percentage Braking $= \dfrac{\text{road or rail}}{\text{shoe}} = \dfrac{\mu_r}{\mu_s}$

varies for dry or wet surface of rail, and on surface material of road (concrete, macadam, asphalt) and also on these being dry or wet.

Friction on track $F \leqq \mu_r$ (normal load) $\leqq \mu_r N = 0.3\ N$ (steel rail)

This is approximately $= \mu_s$ Brake applied pressure $= \mu_s B$.

Equivalent rail or road friction $\mu_e \times N = F = \mu_s B$

so that $F = \mu_e N < \mu_r N$ or $\mu_e < \mu_r$ to prevent slippage.

$$\mu_s = \mu_r \left(\frac{1 + 0.014\ V}{1 + 0.075\ V}\right)$$

This/

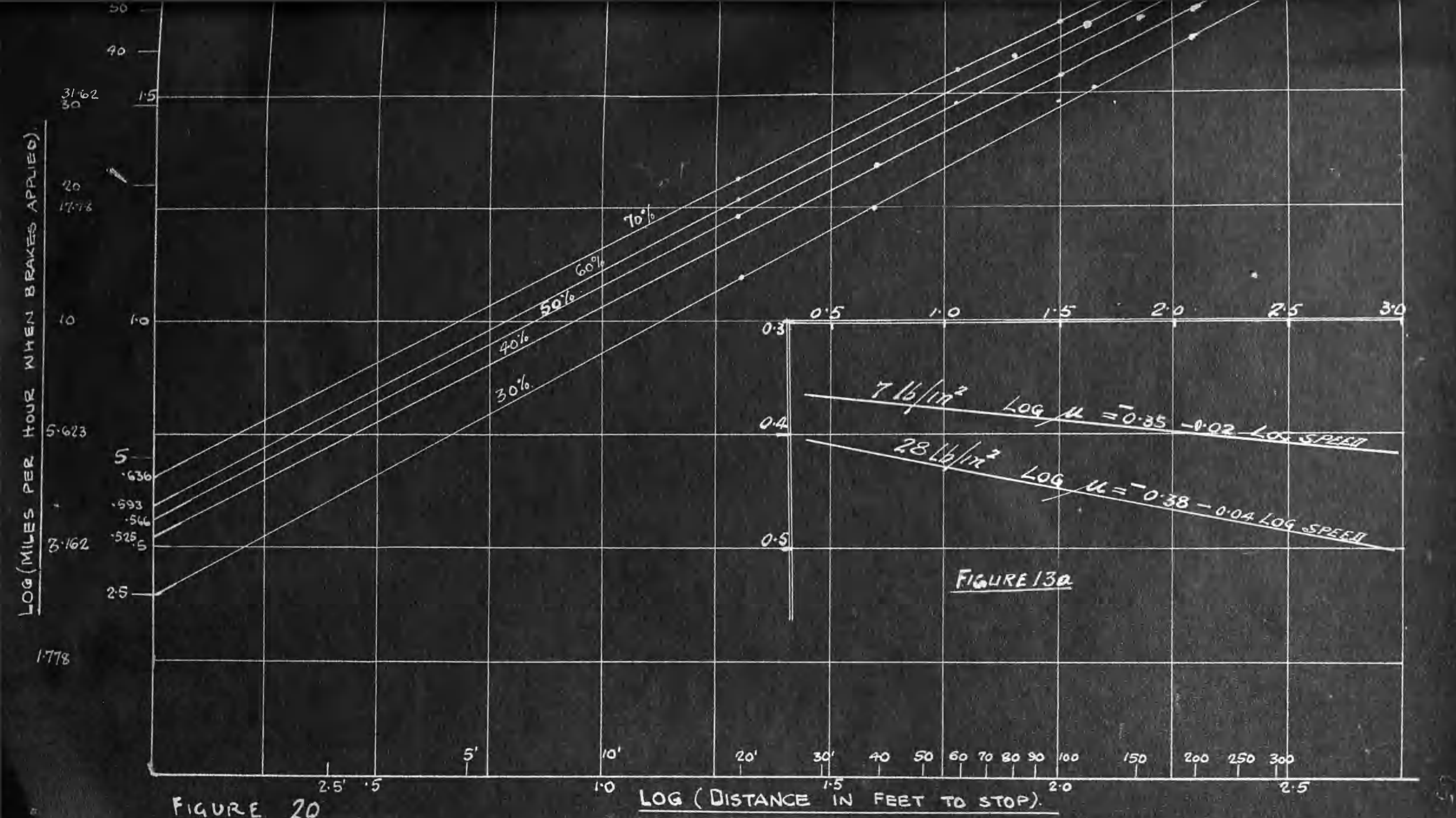


FIGURE 13a

FIGURE 20

This would give for μ_s at 60 m.p.h. a value 0.096,
for μ_s at 30 m.p.h. a value 0.15,
and for μ_s at 20 m.p.h. a value 0.17
for ordinary steel rail.

(2) Brake efficiency testers as applied to motor cars.

In place of giving a relation between the coefficient of friction of the Brake Shoe and the Rail or Road Coefficient the Siemen's Brake Meter (a) and the Ferodo Brake Efficiency Indicator (a_1) give a relationship between the miles per hour of car's motion and the stopping distance in feet.

(a) is simply a pendulum in the form of a capillary tube (with Pitot tube end) containing a coloured fluid. The column shows the retardation (b) in yards per sec.per sec., and where (S) represents the braking distance in yards and (v) the speed of the vehicle in yards per sec.[8]

$$S = \frac{v^2}{2b}.$$

The brakes are applied evenly on a level dry road when motor is moving at say 20 miles per hour.

Derived from $V^2 = u^2 - 2fs$ and $v = u \pm ft$

(a_1) In this tester the speed of motor is 20 miles per hour when the brakes are applied. There are three dials, 20 per cent for a stop or draw up in 67 ft., 30 per cent for stop in 45 ft., and 50 per cent for a stop in 27 ft.[8]

TEST FIGURES FROM A CAR ON MACADAM ROAD, Fig.20.

Brake Efficiency per cent.	V = miles per hour S = stopping distance in feet		Practically
30	$\log V = .40 + .536 \log S$	or $V = 2.51 S^{.536}$	$V^2 = 9S$
40	$= .525 + .506 \log S$	or $V = 3.31 S^{.506}$	$= 12S$
50	$= .566 + .516 \log S$	or $V = 3.68 S^{.516}$	$= 14.2S$
60	$= .593 + .515 \log S$	or $V = 3.92 S^{.515}$	$= 18S$
70	$= .66 + .506 \log S$	or $V = 4.57 S^{.506}$	$= 21S$

Brake Efficiency @ 30 m.p.h.	30	40	50	60	70 per cent.
Distance to draw up	100	75	63	50	43 feet.
@ 25 m.p.h.	70	52	44	35	30 feet.

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INVESTIGATION OF LUBRICANTS

AND

FRICTION BEARING METALS.

I N D E X

INDEX (continued)

INVESTIGATION OF LUBRICANTS, AND FRICTION OF BEARING MATERIAL.

ABSTRACT

In this investigation the lubricating properties of various oils at low rubbing speeds are considered.
A comparison is made between pure vegetable, mineral and animal oils when used as lubricants, with cast iron, phosphor bronze, white metal, mild steel, and nickel-chrome heat treated steel, as bearing materials.
Some consideration is given to the blending of machine oil, lubricants for use in forging and drawing, and the effect of introduction of mercury and chlorine with lubrication mixtures. The finish of the bearing material is considered in the reduction of friction effect. Endurance tests are carried out on various lubricants at different temperatures, and the wear of bearing materials is noted, condition of surfaces are examined after failure of lubrication, or restriction of lubricant supply. The results of experiments have been made use of in Bearing design.

INTRODUCTION

(a) Some notes on the metals used in Tests.

Cast Iron and Mild Steel form the greatest portion of bearing metals; of the alloys, phosphor bronzes and White Metal are the chief. Crank shafts are made of special Steel in which nickel and chromium are employed to form a steel alloy. In this investigation the metals used are (1) Cast Iron on Cast Iron, on Phosphor Bronze, and on White Metal; Nickel Chrome Steel on Cast Iron, on Bronze and on White Metal.

The approximate chemical analysis of the material used.

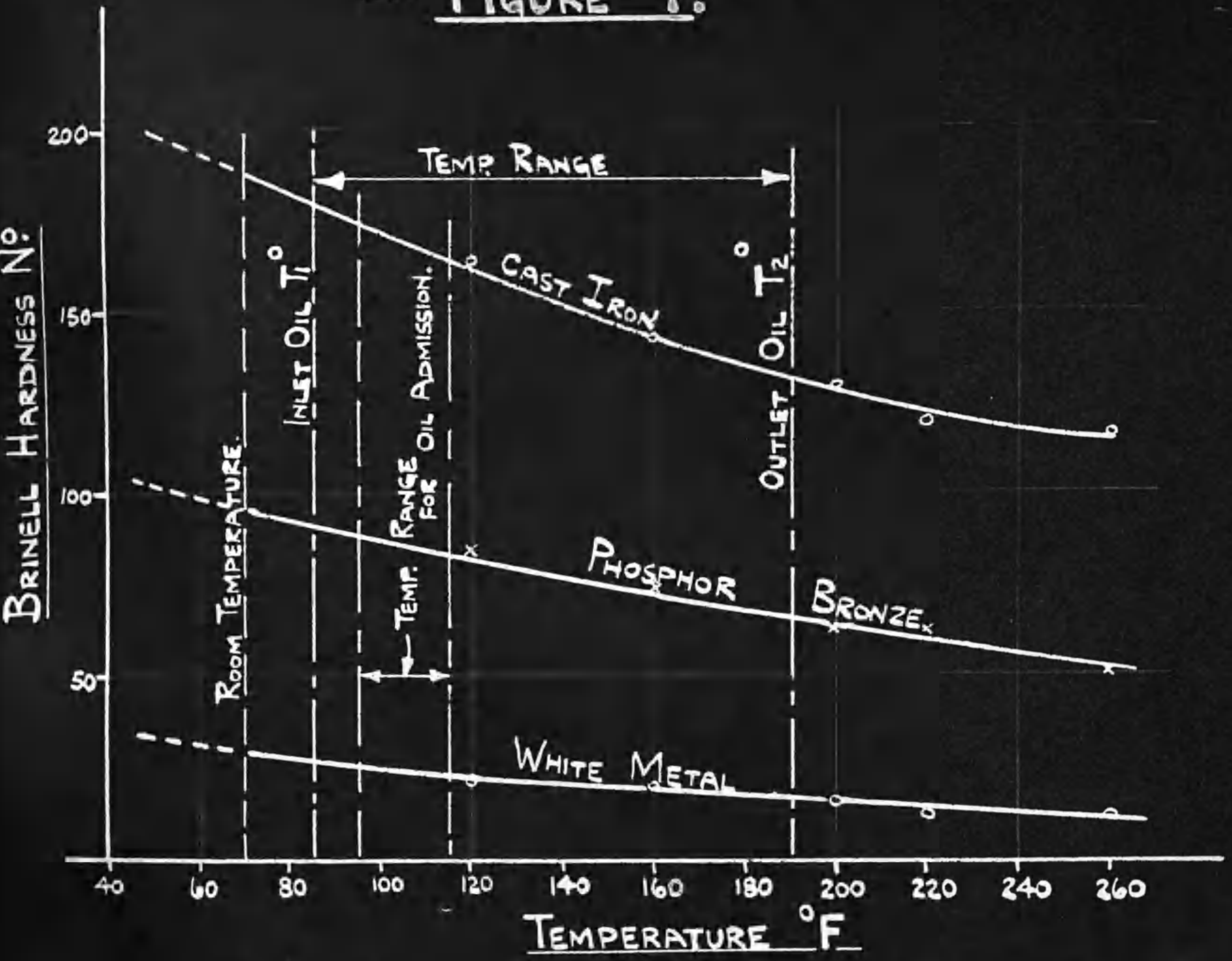
	C	Mn	Si	S	P	Ni	Cr	P_b	Sn	Cu	Sb
Cast Iron	3	0.6	1.3	0.05	0.2	3					
Nickel Chrome Steel	0.31	0.65	0.15	0.03	0.03	3.3	0.7				
Phosphor Bronze					0.4			10	9.5	80	
White Metal								1.5	78	7.5	13

The approximate physical properties.

		lb. per sq.in.		
	Tensile Stress.	Compressive	Fatigue ±limit.	Brinell Hardness. Ball = 10mm. Dia. Pressure = 3000 Kgr.
Cast Iron	3.2×10^4	12.8×10^4	1.4×10^4	190
Nickel Chrome Steel (oil hardened)	12.7×10^4		4.4×10^4	290
Bronze	7×10^4		1.5×10^4	93
White Metal	2.3×10^4			29 at 70°F. 12 at 212°F.

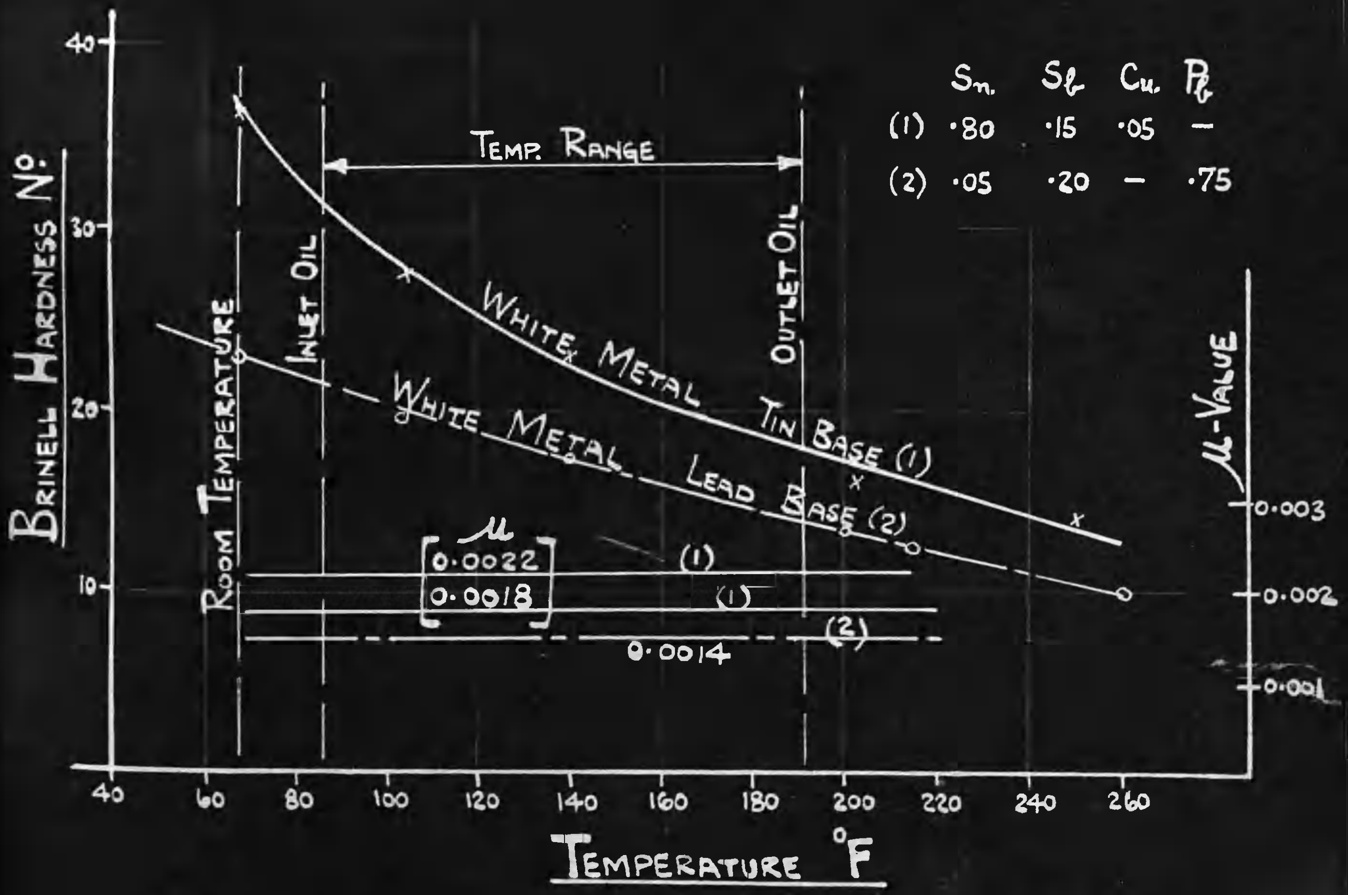
The cast iron, Phosphor Bronze, and White Metal discs were wound with an electric-heating element, and six hardness readings at temperatures ranging from room temperature 56°F. to 250°F. were taken. Brinell Nos. have been plotted on a base of Temperature. Fig.1. On this diagram there is shown the average temperature range for a number of bearings. The best temperature/

— FIGURE 1. —



BALL HARDNESS. TEST OF PLATES.

TIME OF LOADING ½ MINUTE.



BALL HARDNESS OF WHITE METAL TEST PLATES.

AND BEARING METAL

temperature inlet appears to be approximately 86°F. giving a safe range of 100°F. rise in oil temperature and therefore approximate bearing temperature. White Metal lining shows a reduction in Brinell hardness by 50 per cent. during the rise of 100°F. This change in Brinell number, along with heavy load which has given rise to the temperature change, assists the plastic flow of the White Metal.

(b) Some notes on the Lubricants used in Tests.

In a paper already published / on lubricating properties of various oils, data will be found concerning some of the lubricants used in this investigation. The following lubricants are considered, (1) three forms of machine oils are compared to show how the main features can be examined by means of simple apparatus, (2) Neatsfoot oil, Graphited Spindle oil plus 1 per cent. of oil dag, Diesel oil, and Lape oil, (3) Tallow (pure), Tallow 10 per cent White Lead, Tallow and Castor oil.

Engineers are accustomed to the use of "Specific Gravity" and realize that a unit volume of oil weighs more as the numerical value of the specific gravity increases, and S $\frac{60^{o}F}{60^{o}F}$ is the ratio arbitrarily chosen, or S $\frac{15.5^{o}C.}{15.5^{o}C.}$ The weight of one gallon of oil is ten times its specific gravity but this unit has no relation to the lubricating quality of an oil. However, the lower the specific gravity of the oil the easier will it be to free the oil from water by centrifugal means. The values of the specific gravity were obtained by determining the density of the oil.

50 c.cs of oil were carefully weighed at 60°F.
i.e. weight of 50 c.cs oil plus weight of specific gravity bottle = 63.52 gms.
weight of specific gravity bottle = 16.83 gms.
Therefore weight of 50 c.cs of oil = 46.69 gms
Therefore density at 60°F. = 0.934
and specific gravity = 0.934 [9·34 lb/gallon]

The/

The density at any other temperature was obtained from the formula $D = d - kt$ ....(1)

where D = density of oil at required temperature
d = " " " " 60°F.
t = number of degrees above 60°F.
k = a constant = .00035 for most lubricating oils.

If the specific gravity was required to very great accuracy a correction for the buoyancy would require to be made. This is not required in simple lubrication problems, but a curve of correction for temperature rise, which is to be added when the temperature is above 60°F. and substituted when the temperature is below 60°F. is given, Fig.2., for various specific gravity oils.

In this investigation the viscosity of the oils under test was obtained by use of a Standard Redwood Viscometer 50 m l and converted into C.G.S. units (Poises) by the formula given by Higgins.

$$\mu = \rho \left(0.0026\,T - \frac{1.715}{T}\right) \text{ poises } ....(2)$$

where μ = absolute viscosity in C.G.S. units (poises)
T = No. of seconds for outflow of 5 c.cs.
ρ = Density of oil at temperature of test.

Hence from (1) and (2) the values of absolute viscosity are obtained.

Temp. °F.	Redwood Times Seconds	Density from 1 D	Viscosity from 2 (poises)	Rape Oil Units
60.5	3855	0.934	9.34	725
90	900	0.923	2.16	193
120	320	0.913	0.755	59
150	158	0.902	0.363	30
180	75	0.892	0.170	17

For the purpose of comparison a column showing "rape oil units" has been added to the table.

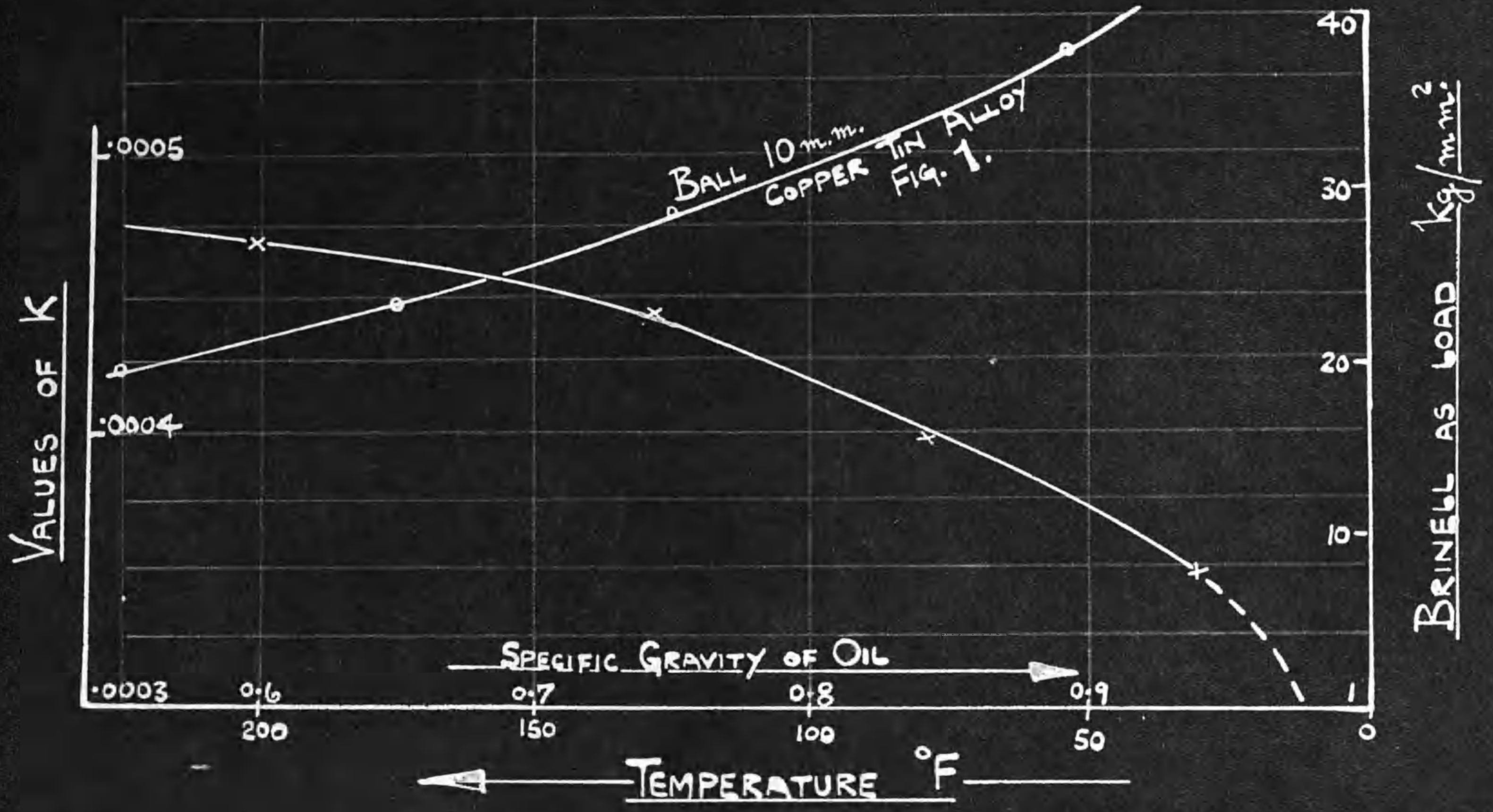
$$\text{Rape oil units} = \frac{T}{535} \times \frac{\text{sp.gr. of oil at } t^{\circ}\text{F.}}{\text{sp.gr. of refined rape oil at } 60^{\circ}\text{F}} \times 100$$

$$= \frac{T \times St^{\circ}F \times 100}{535 \times 0.915} = \frac{T \times St^{\circ}F.}{4.895}$$

535 and 0.915 are, respectively, the Redwood No.1. time of flow and specific gravity of refined rape oil at 60°F.

During/

FIGURE 2.



VALUES OF K FOR VARIOUS SPECIFIC GRAVITY.

During this investigation reference was made to viscosities given on the Saybolt and the Engler viscometer, and a blending chart is given in Fig.3. which shows the three viscosity scales in general use. The absolute centipoises (assuming Sp.Gravity 0.9) are also shown. The blending charts have been found necessary when an oil is to be made to specification drawn up from detailed examination of lubrication requirements.

(c) Some notes on the surface finish of the Bearing Metal.- The gun-metal bush was machined with smooth finish and test made on the Bearing Machine . The bearing surface was then hand-scraped and second test carried out. An expanding mandril finish was then put on metal and adjustment for clearance made; the clearance was kept as near 1/1000 as possible and a test carried out. The machine was then run for several hours as a "running in" process before the final test.

Much the same procedure was gone through in obtaining the different finishes on the Phosphor Bronze and White-Metal plates used in the Deeley Oil Testing Machine.

A white-metal bearing bush was made by fixing the plummer block on the shaft and casting the bush into the block. As a finish the bearing machine shaft was made to rotate slowly just before the white-metal had cooled. The bush was then removed and cut out to suit the oil rings similar to the gun-metal bush, Fig.4(a), as far as possible the same area was maintained. A sketch of lines of oil flow is shown Fig.4(b). The white-metal is then given a rub with a wire brush to clear off scale after which a test was carried out. This was American practice with long bearings of small diameter; for example, the main spindle of Pickering governors. A test was carried out with the metal as described and then a second test after eight hours running in. A section was taken off the material from a very smooth portion of the bush and a second at a part where evidently seizure had taken place, Figs. 5(a) and 5(b).

The/

FIGURE 3.

BLENDING DIAGRAM.

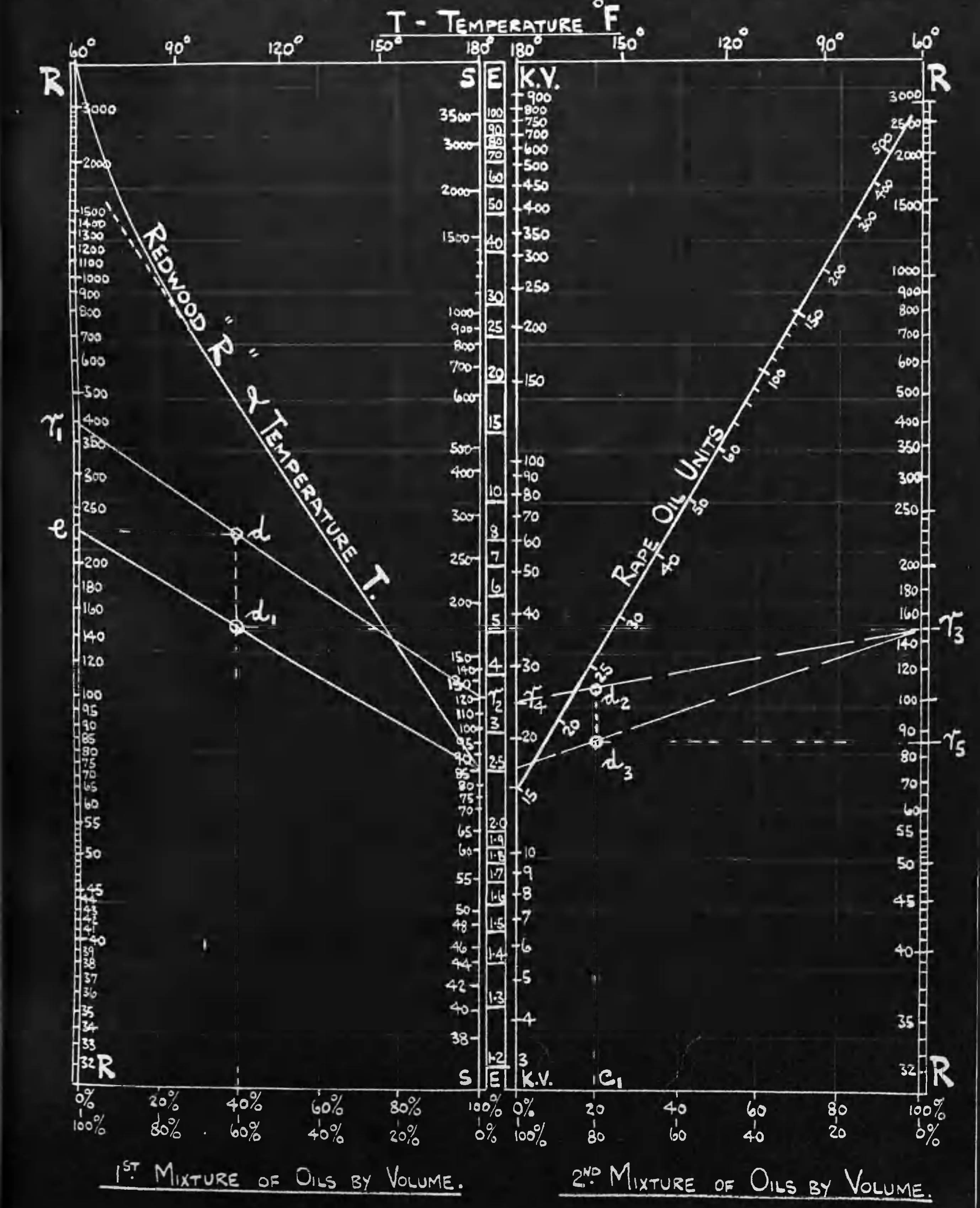
EXAMPLE:- 40% R_{400} + 60% R_{100} @ 108°F. GIVES R_{150}

r_1 to r_2 d_1 to r_3

BLEND OF 3-OILS. 80% R_{150} + 20% R_{100} @ 158°F. GIVES R_{84}

r_2 to r_4 d_3 to r_5

R = REDWOOD. S = SAYBOLT. E = ENGLER. K.V. = KILO VOLTS.



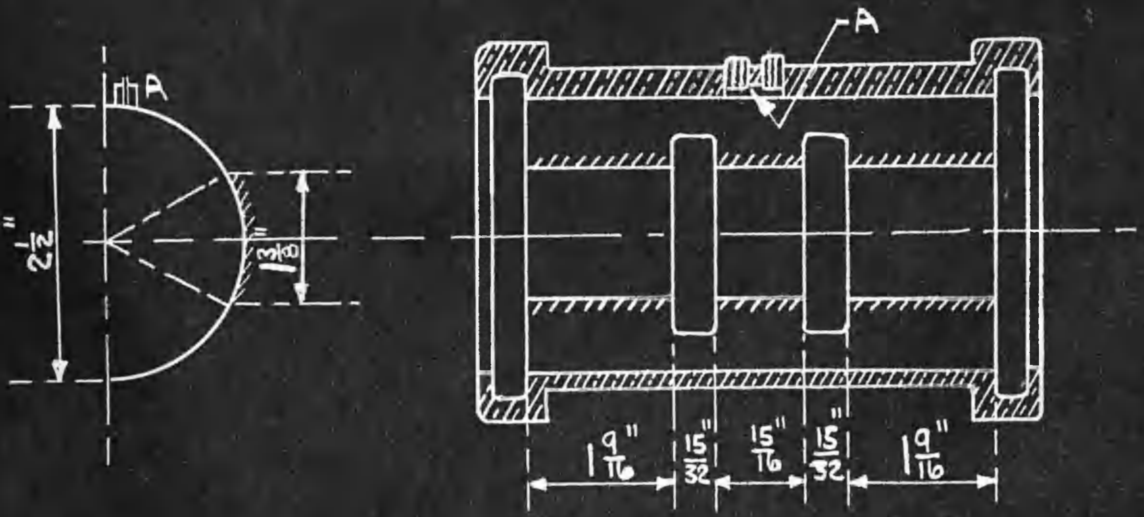


FIG. 4a. SURFACE BEARING DIMENSIONS.

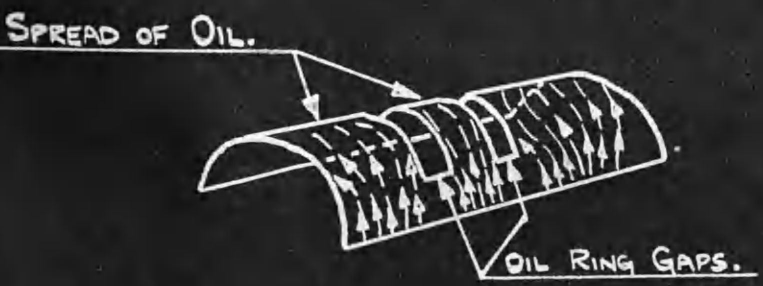


FIG. 4b. OIL STREAM FLOW.

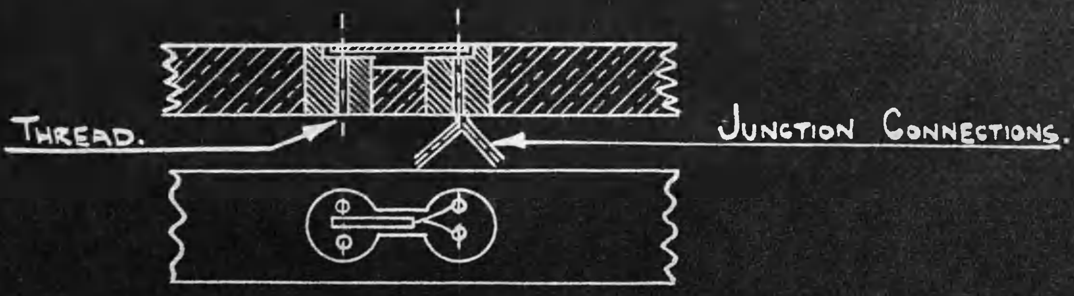


FIG. 12.

5a.

METAL SMOOTH SURFACE

5b.

METAL ROUGHENED SURFACE

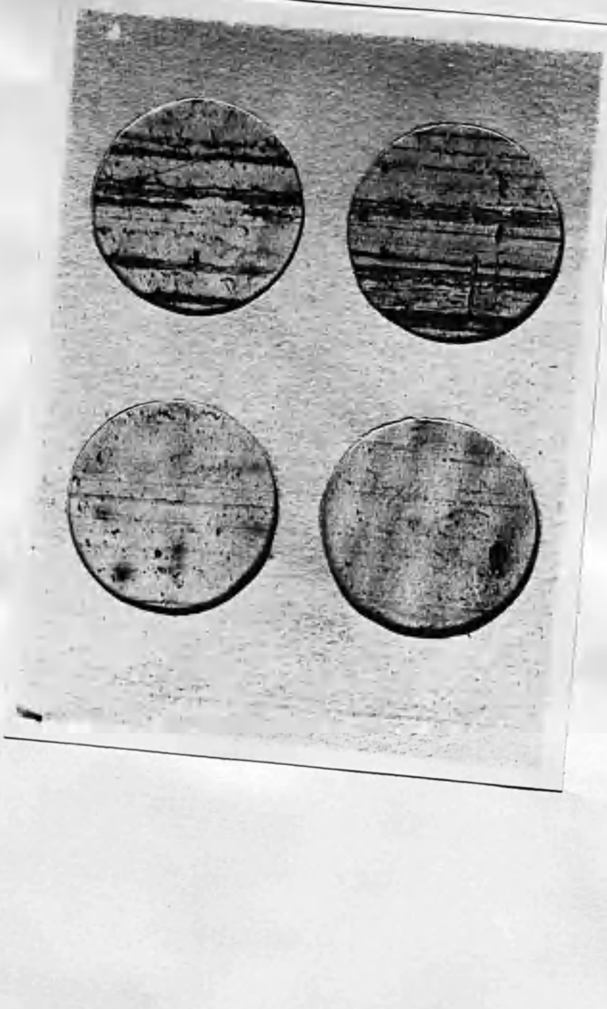


FIGURE 10.

The bush was adjusted and a smooth cut was taken in Lathe; a further test was then run. For comparison a spring-tool was used in turning out the bore, this took off the spiral finish of the more pointed turning tool, and again friction results were obtained. Next a grinder surface finish was tried, and then the same followed by a roller tool. The last surface was that obtained from lathe spring scraper followed by roller tool. The latter tool had the effect of hardening the white metal giving a Brinell hardness of 31 in place of 29 at 70°F.

Boundary Conditions in Lubricated Elements.

With these remarks on the various lubricants, bearing metals, and finish of surface, it is now proposed to demonstrate that mineral oils have a smaller surface tension than water, and fatty oils have a still lower value than mineral oils. Boundary friction is shown to be a function of the chemical constitution of the lubricant, and the nature of the solid surfaces, since these properties affect the attraction between both.

Bachmann and Brever demonstrated that the higher the lubricating power of an oil, the higher is the heat of wetting against finely divided copper. Taking 100 grams of copper, the heats of wetting were:

	Castor oil	Linseed oil	Liquid Paraffin	Petroleum	Petroleum 1% Acid
Calories	12.1	13.8	3.8	5.7	21.3

The efficiency of the addition of small quantities of free organic acids in lowering the surface tension of petroleum lubricants, and thus enhancing their "oiliness" properties, is clearly demonstrated. For boundary conditions a good lubricant is one which is strongly attracted by the solid whereas a poor lubricant is one which is attracted less strongly, (reference to paper on Lubricating Properties of Oils).

A/

A paper on "The Elastic Range of Friction" by J.S. Rankin [2] and an article on "The Effects of Gases, Vapours and Liquids on the Limiting Friction between Solid Surfaces" by J.M. MacAulay [3] show clearly the effect of excessive pressure and the failure of lubrication in friction when boundary conditions prevail.

Lubricants with Force Fits.- That Viscosity varies with pressure can be clearly seen by examination of test figures obtained in lubrication of plugs used for "Force Fit Experiments" also "Contact Film Resistance in Rail Wheel Force Fits". [4.] In Fig. 6(a), with a Mineral Oil, the push off force never reaches the push on pressure, showing that the film has not broken down, whereas in Fig.6(b) the tallow and white lead curves show a failure of the film and a push off pressure greater than the push on pressure. The viscosity of the mineral oil has increased nearly ten times under radial pressure; at the same time the viscosity of the tallow has increased four times. There is a cold-working condition of the material under test. This has been clearly established by Dr. Russell, but this investigator has failed to point out the effect of the different film strength of the various lubricants used. Tallow has an approximate film strength of 40,000 lb. and a friction coefficient of 0.005. Castor oil has a film strength of 60,000 lb. and a friction coefficient of 0.05, and its wetting value is not nearly as good as tallow.

Figs. 7 to 9 show comparisons between vegetable and animal oils. The fatty acid added to the pure rape does not show a great improvement. These are compared in 8(a) and 8(b). When selecting a lubricant to be used for press on or force fits the author would recommend that the lubricants be a mixture of the lubricants which will combine film strength (i.e. castor oil) and the higher wetting properties, and lower coefficient of friction (i.e. Tallow).

Lubricants with Pressings.- If a corrosive substance added to the lubricant would not be detrimental, them the introduction of a volatile substance, such as carbon tetrachloride, has been found/

LUBRICANTS WITH FORCE FITS.

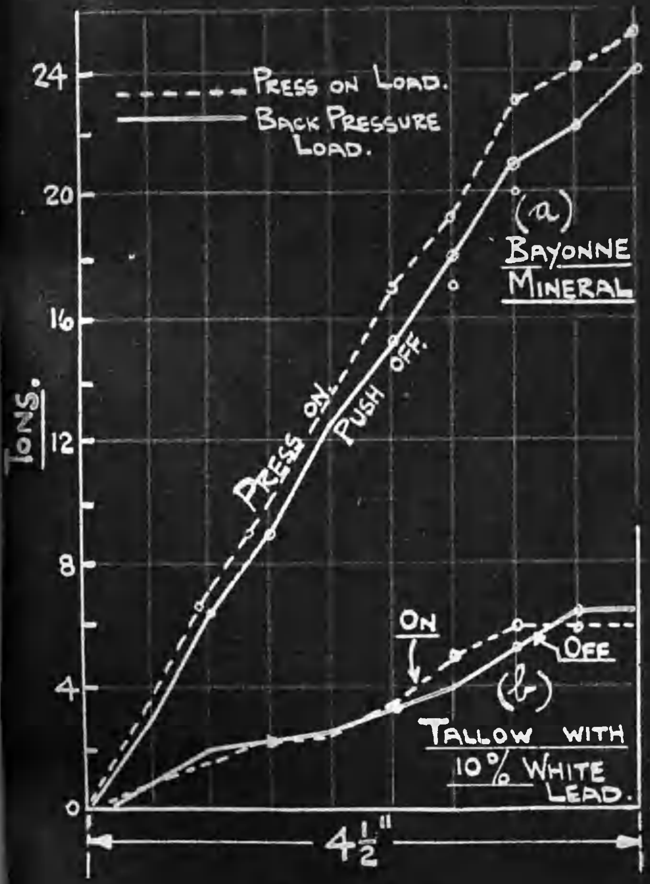
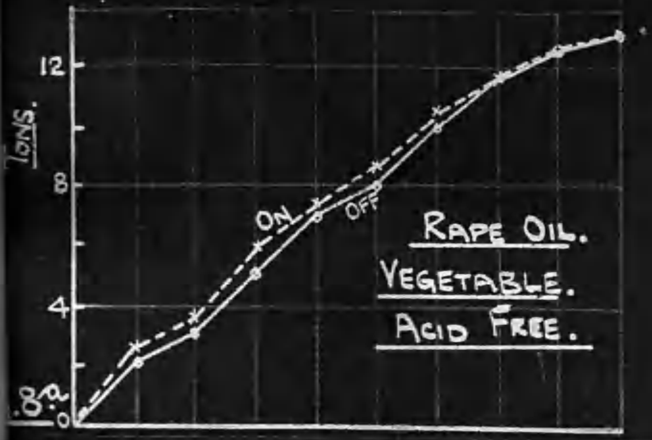
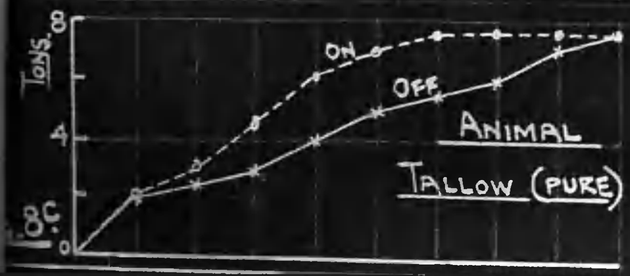


Fig. 6.





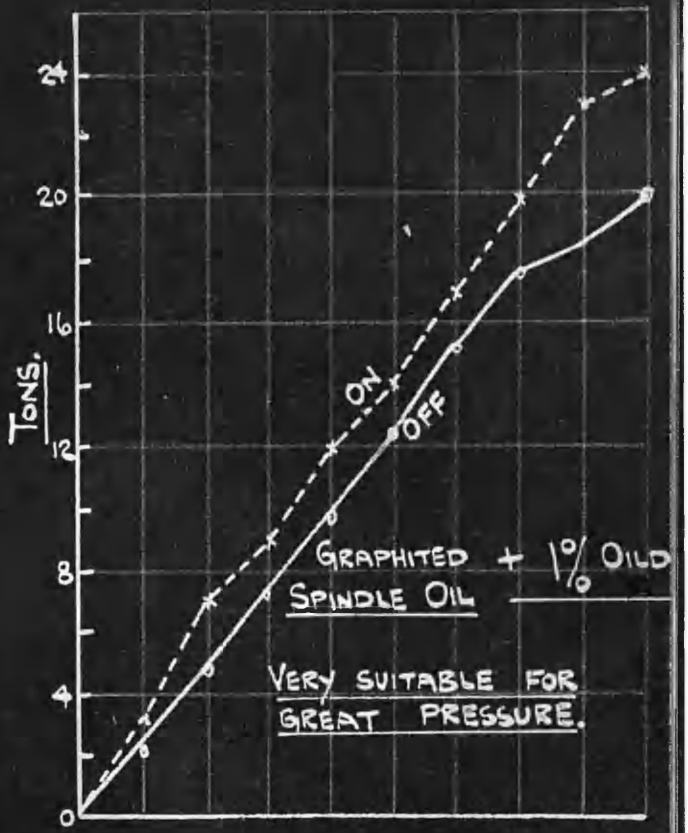


Fig. 7.

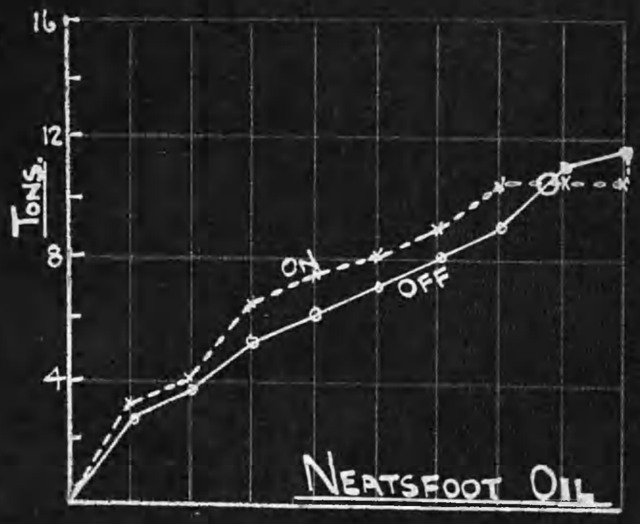


Fig. 9.

found to so fortify the film strength as to make it almost indestructible. The application of such a compounded lubricant, castor oil, tallow, and carbon tetrachloride, to the surface of a sheet steel plate pressing has reduced the power required by nearly 30 per cent, and the withdrawal of the press tool from the pressing and the pressing from the die was very much simplified. Advantage of the lubricating properties have been taken to make material flow into a difficult die formation, and at some parts of the metal sheet to be pressed no lubrication has been applied, and the metal has been found to hold with little or no movement.

The author had the opportunity of carrying out many experiments in the Gartsherrie Engineering Company and Weldless Chain works at Coatbridge with pressing and stripping tools.

The dies were found to last longer if a lubricant of high film strength, low coefficient of friction, and good wetting properties, was used.

<u>Lubricants with Bearings.</u>- While considering the effect of boundary lubrication and semi-lubrication, tests were carried out on lubricating oil drawn from the crank case of a motor car after different periods of use. The oil was allowed to stand in a test-tube until all sediment had separated, and then the usual tests were carried out on the Deeley and Boult testing machines. Very little difference, if any, could be detected between the new and the old oil. In some cases the old oil appeared to possess lubricating qualities superior to the new oil. It was found, however, that the pegs on the Deeley machine and the steel plate had their surfaces destroyed when left in the oil under heavy pressure; Also, that in starting up and stopping the Boult machine bearing surfaces were being destroyed by the crank-case oil.

A chemical test was applied to the old and the new oils, and it was found that an appreciable amount of sulphuric acid had been added to the lubricating oil in the crank-case.

The/

The H_2SO_4 had not dissolved in the oil but was held in suspension, and could be separated by a centrifugal separator. The ageing test on the Deeley showed signs of the corrosive effect of the acid. The oil film had been broken by the application of heavy loads and where this had taken place the acid coming in direct contact with the metal attacked it with vigour.

On bearings $F_e + H_2SO_4 \longrightarrow F_eSO_4 + H_2$
and is followed by a secondary reaction

$$F_eSO_4 + H_2 \longrightarrow F_eS + H_2O$$

The Bearing Machine which was designed in the Royal Technical College is fitted with a double ring oiler bearing 2½ in. diameter and 5 in. long. By indenting into the top bearing bush pieces of material to be tested, the question of friction with semi-lubrication and the wear of material under different conditions can be considered. The rings may both be removed or one may be allowed to carry oil to the shaft, thus supplying oil to the friction pad piece under test. In this way an abundant, a restricted supply, or no oil supply can be obtained. The rubbing speed, V, and the pressure, p, per unit area of rubbing surface, was altered throughout the tests, also the temperatures range from 66 to 200°F. and was kept constant at a predetermined condition. The critical condition is reached when $V/\sqrt{p}$ is greater than 30 in lb.ft.min. units as the coefficient of friction rapidly increases. When 30 lb.ft.min. unit has been exceeded the 'semi-fluid' phase is attained. Although the friction is less when $V/\sqrt{p} < 30$ in lb.ft.min. units, it would appear that the chance of seizure is greater and the wear is also more rapid. In the semi-fluid condition the coefficient is independent of viscosity.

<u>Tests and Test Results:</u> <u>Deeley Machine.</u>

<u>Series 1.</u>- Nickel-Chrome Steel on White Metal.
To find the effect of machine finish or smoothness of surfaces on Static Friction.

The/

NICKEL-CHROME STEEL ON WHITE METAL

Table showing μ - value for different finishes

Load on Bearing Material lb.	Disc Turned Effort ÷	Disc Turned and Scraped Effort ÷	Disc Turned Scraped and Burnished. First Polish	Disc Turned Scraped and Burnished. Second Polish	(a) Disc run in First Finish	(b) Disc run in Second Finish	(c) Disc run in Third Finish	(d) Disc run in Average 4th & 5th Finish
10	1 = 12.8	1 = 9.5	8.7	8.3	7.8	7.5	6.75	6.75
20	2 = 13.4	2 = 11.5	10.0	9.7	8.35	8.0	7.2	7.1
30	3 = 14.9	3 = 10.7	10.0	9.5	8.9	8.3	8.07	8.0
40	4 = 14.0	4 = 10.0	10.0	10.1	8.45	8.25	8.25	8.2
50	5 = 13.8	5 = 9.6	9.7	10.2	8.9	8.8	8.3	8.25
60	6 = 13.9	6 = 9.8	9.6	10.1	9.6	9.2	8.6	8.4
70	7 = 14.4	7 = 9.7	9.7	9.8	9.7	9.6	8.9	8.5
80	8 = 15.3	8 = 10.8	10.3	10.0	10.0	9.75	8.85	8.75
90	9 = 16.6	9 = 10.4	10.4	9.9	9.8	9.4	9.15	9.0
100	10 = 17.5	10 = 10.6	10.0	9.75	9.55	9.45	9.3	9.15
Average values	14.66	10.2	9.84	9.73	9.2	8.83	8.33	8.21
μ - values	0.330	0.234	0.225	0.223	0.221	0.205	0.191	0.188

Surfaces separated every five minutes and oil caused to flow between them to ensure clean rubbing. This method proved quite successful.

Processes (a), (b), (c), and (d) carried out on Deeley machine, rubbing times, $\frac{1}{2}$-hour, average pressure, 50 lb. per sq.in.

The Oil in use is Castrol XL. and the only alteration made to the Standard Deeley machine was the addition of a heating coil, so that the temperature of the oil might be varied or maintained. The properties of the nickel-chrome steel and the white metal have already been given, p.4.

The steel pegs have been finished carefully, and lapped to give a smooth surface which is perfectly flat to bear against a white-metal surface.

The white metal surface is finished in the following manner:-

(1) Surface finish - fine or smooth turned.
(2) " " - after smooth turning, scraped.
(3) " " - after turning and scraping, the white metal is burnished which caused flowing and has a hardening effect.
(4) " " - same as (3) but with second burnishing.
(5) Test materials rubbed against each other under light pressure. The effect produced is similar to "running-in" of a journal bearing.

(6) Is the same as (5) but with longer period of running-in.

(7) Is the same as (5) and (6) but with special finish.

The effect of the finish is clearly shown in table of results, by the gradual lowering of the static value of the coefficient μ. The average value of μ becomes quite steady after (3), showing the value of running-in.

(See Table of Results on separate sheet)

<u>Series II.-</u> Nickel Chrome Steel on Phosphor Bronze.

In this case the resultant changes are not quite so marked. The first turning finish is much better than that of the white-metal. The turning tool does not cause so much chattering or knurling on the surface of the harder phosphor bronze plate. The harder material, Brinell No. 93, 10 mm. ball and 1000 Kg, corresponding to Rockwell R_b 50, $\frac{1}{16}$ in. ball and 100 Kg. takes on a much better finish with apparently less care than the white-metal, which when tested for hardness on the Rockwell machine/

machine specially equipped with an $\frac{1}{8}$ in. diameter ball and a pressure of 60 Kg. This special equipment gave a measure of the hardness of fabric and soft material when read on the R_b scale, and corresponded in the case of white metal R_b 50, to Brinell No.29, 22½ mm. ball and 500 Kg.

A note on the various methods of testing for hardness, and on the connection between the hardness numbers has not been fully dealt with in this investigation. This subject is worthy of a thorough investigation. The hardness number has been made use of in the selecting of the best speeds and feeds for machining.

Table showing μ-value for different finishes.

Nickel-chrome steel on Phosphor Bronze.

First Finish - Machined; Second Finish - Machined and scraped; Third Finish - Machined, scraped and run-in.

Load on Bearing Material	First Finish Effort -			Second Finish Effort -			Third Finish Effort -		
10	9.5	or ÷ 1	= 9.5	8.75	or ÷ 1	= 8.75	8.5	or ÷ 1	= 8.50
20	18.25	" 2	= 9.12	18.1	" 2	= 9.05	17.5	" 2	= 8.75
30	27.3	" 3	= 9.10	26.7	" 3	= 8.90	26.0	" 3	= 8.66
40	35.6	" 4	= 8.90	35.1	" 4	= 8.77	34.9	" 4	= 8.72
50	43.25	" 5	= 8.65	43.0	" 5	= 8.60	42.5	" 5	= 8.50
60	51.80	" 6	= 8.65	51.5	" 6	= 8.59	51.0	" 6	= 8.50
70	61.75	" 7	= 8.82	60.0	" 7	= 8.55	60.0	" 7	= 8.57
80	70.25	" 8	= 10.03	70.0	" 8	= 8.75	70.0	" 8	= 8.75
90	82.25	" 9	= 9.14	80.1	" 9	= 8.90	80.0	" 9	= 8.90
100	94.0	" 10	= 9.40	90.0	" 10	= 9.00	88.0	" 10	= 8.80

One point which is clearly brought out in this test is the great advantage of running-in bearings at light loads. There was a tendency for the bearing surfaces to seize at heavy load, if the surfaces were not perfectly smooth, even although the precaution of inserting a film of oil had been taken.

Example (a) - White-metal turned (only) and mating with smooth finished Nickel-Chrome steel at the bearing pressure of 100 lb. per sq.in. gave a reading 170 on the Deeley machine, or a static coefficient of $\frac{170 \times 10}{100 \times 43.58} = \frac{17}{43.58} = 0.387$.

After/

After perfectly smooth surfaces were obtained with polishing the rubbing-in process applied followed by the cleaning process referred to in some of the previous papers in the use of the Deeley machine. The value of μ becomes

$\frac{9.3}{43.58} = 0.213$ which is a decrease in the static friction value of 45%.

Example (b) - Phosphor Bronze (Turned smoothly not polished) at high load with nickel-chrome steel, pressure 100 lb.per sq.in. gave

$$\mu = \frac{94 \times 10}{10 \times 43.58} = 0.26$$ for the value of static friction, which changed to

$$\mu = \frac{88 \times 10}{10 \times 43.58} = 0.22$$ a lowering of the static friction coefficient value by 22.3% by lapping and polishing the surfaces of the bearing materials.

<u>Series III.</u> - Castor Oil being the lubricant and bearing surfaces Nickel-Chrome Steel on Cast Iron.

Table of μ- values for different finishes:-

Load on Bearing Material	Smooth Turned and scraped		Smooth Turned and polished		Run-in on light load metal on metal lubricants	
10	6.5	1 = 6.5	6	1 = 6	5.7	1 = 5.7
20	12.75	2 = 6.37	11.4	2 = 5.7	11.6	2 = 5.8
30	19.20	3 = 6.4	17.35	3 = 5.78	17.25	3 = 5.75
40	25.8	4 = 6.45	23	4 = 5.75	22.6	4 = 5.65
50	31.5	5 = 6.3	28.8	5 = 5.76	28.75	5 = 5.75
60	38.0	6 = 6.35	34.2	6 = 5.7	34.5	6 = 5.75
70	44.0	7 = 6.3	40.1	7 = 5.73	40.25	7 = 5.75
80	50	8 = 6.25	45.6	8 = 5.7	45.25	8 = 5.66
90	55.8	9 = 6.2	51.25	9 = 5.7	50.8	9 = 5.64
100	60.2	10 = 6.02	56.7	10 = 5.67	55.5	10 = 5.55
Static Friction	μ = 0.145		μ = 0.132		μ = 0.131	

Example (c) - Cast Iron (smooth turned and scraped not polished) at high load with Nickel-Chrome Steel at a pressure 100 lb.per sq.in gave $\mu = \frac{60.2 \times 10}{10 \times 43.58} = \frac{60.2}{43.58} = 0.138$

and/

and when the bearing surfaces had been polished, rubbed-in on oil

$$\mu = \frac{5.55 \times 10}{10 \times 43.58} = \frac{5.55}{43.58} = 0.127$$

which shows a lowering of the static friction value of only 8%.

Using the Brinell No. as a basis of hardness for test material, white-metal, 29; phosphor bronze, 93; cast iron, 190. The results confirm that the hardness of a material even when only smooth finished has resisting properties on the tool ridges. The ridges, left by the turning and spring scraper, in the case of white-metal, have a tendency to cause the bearing materials to seize. It is well known that white-metal flows under pressure and that the oil film in a journal bearing is effected by this flow. The metal follows the eccentricity of the loading and this is followed by a reduction of the clearance.

Series IV. - Lubricating Oil Castor XL with different bearing metals and varying loads. The velocity or rubbing speed was kept as constant as possible throughout the tests.

Load	(1) Cast Iron on White Metal				(2) Cast Iron on Cast Iron			
10	6.8	7.9	7.1	1 = 7.1	5.8	1 = 5.85	5.7	1 = 5.7
20	14.2	15.5	14.8	2 = 7.4	11.75	2 = 5.87	11.5	2 = 5.75
30	22.5	24.5	23.0	3 = 7.6	17.65	3 = 5.88	18.6	3 = 6.2
40	35	36	35.5	4 = 8.9	24.5	4 = 6.12	24.5	4 = 6.37
50	43	44	43.25	5 = 8.65	31.25	5 = 6.25	31.7	5 = 6.34
60	52.5	54	53.0	6 = 8.8	37.1	6 = 6.2	36.3	6 = 6.05
70	62	62	62	7 = 8.85	43.5	7 = 6.21	42.6	7 = 6.08
80	75	73	70	8 = 8.7	48.5	8 = 6.06	47.5	8 = 5.94
90	84	80	81	9 = 9.0	54.6	9 = 6.06	52.1	9 = 5.8
100	100	97	89	10 = 8.9	59.2	10 = 5.92	58.6	10 = 5.86

Static value of $\mu = 0.194$ $\qquad$ $\mu = \frac{6.02}{43.58} = 0.137$

Oil	2(a) M.A. on C.I.	2(b) M.S. on P.B.	2(c) M.S. on M.S.
Bayonne oil	0.216	0.226	0.267
Bayonne 2% Rape	0.186	0.196	0.214
Castrol XL	0.198	0.208	0.241
Average	$\mu = 0.20$	$\mu = 0.21$	$\mu = 0.24$

Load	(3) Cast Iron on Phosphor Bronze		
10	8.5	8.2	1 = 8.3
20	17.6	16.8	2 = 8.6
30	27.5	25.5	3 = 8.8
40	36.2	34.5	4 = 8.8
50	45.2	43.7	5 = 8.9
60	54.2	52.5	6 = 8.9
70	63.5	61.75	7 = 8.98
80	70.0	69.5	8 = 8.75
90	78.5	77.5	9 = 8.66
100	84.8	83.38	10 = 8.52

$$\mu = \frac{8.6}{43.58} = 0.198$$

(1) Cast Iron on White-Metal.- In this test the same difficulty of obtaining a correct value at high loads was in evidence, the cast iron having to be run against the white-metal surface (at light loads) until a perfect smooth surface had been obtained. As in the previous tests the maximum load is 100 lb.per sq.inch.

Maximum $\mu = \frac{10}{43.58} = 0.221$ and minimum $\mu = \frac{8.9}{43.58}$ = 0.202, a difference of 8.18% due to difference of surface conditions. The surface of the white metal had been polished, and effectively cleaned in the way already explained.

(2) Cast Iron on Cast Iron.- The values obtained for the static friction are extremely regular. The same test had been carried out with these two metal surfaces more than eight months previously, and the μ - value obtained was 0.131; in the present test μ = 0.137. The increase in the coefficient due to ageing had been noticed among the first experiments where the period of ageing was ten days, and this test was put on to confirm the statement that ageing alters the value of the static μ . 2(c) High μ value 0.34,seizure occurs.

(3) Cast Iron on Phosphor Bronze. - There is not the same tendency to seize at high loads, but still there was a grip between the surfaces giving a higher value of μ , which decreased/

decreased with rubbing-in of the surfaces. The values given in the table are for the condition after a true bearing surface had been obtained. For curves see Caldwell and Thomson.[7]

Fine Finish to Turned Surfaces

In making new test plates for the Deeley machine the problem of finish presented a considerable difficulty. The plates were smooth turned. This surface under the microscope presented a most irregular appearance. A spring scraper was used and the ridges partly disappeared, but still the surface was very irregular. Various lubricants were tried as coolants for the lathe tool, the shape of the tool being kept the same throughout. It was found that a small drop of glycerine and water gave the best finish and avoided the use of a carbonundum oil slab. The use of the grinder would have introduced an unknown factor into the friction tests. Pure oil of turpentine was tried on the spring scraper but the surface obtained was rougher.

Tests and Test Results; Bearing Machine.

The machine as designed did not come up to the requirements of research work in all the points of lubrication of journal bearings, and was replaced by Dr. Thomson by a machine described in an article[6] on Lubrication and Bearing Design. The machine, however, is quite suitable for tests on effect of smoothness of machine surface using phosphor bronze and white metal bushes or pad pieces. It may also be used as a lubricant and wear tester in conjunction with the Disc Machine used for fabric material, giving an excellent means of comparing oils or friction material when running against a lubricated or wetted surface.

The friction at various rubbing speeds can be determined. The temperature of the lubricating oil is kept constant, and the load is varied over the test, but may be kept constant over a range of speeds.

Series I/

Series I. test (a) - Temperature kept constant at 66°F.
Speed varied from 0 - 400 r.p.m. or 0 to 314 ft.per min. to 1280 ft. per min.
Load varied from 56 - 160 lb. per in^2 and then to 228 lb/in^2
Oil used - Gas Engine Lubricant.

See separate sheet for Table.

The journal bearing had already been running for eight hours before test readings were taken. The surface of the material was extra smooth and at the end of test was in perfect condition. (Fig.10) p. 12.

Series I. test (b) - Temperature kept constant at 138°F.
Speed varied from 0 to 400 r.p.m. to 1600 r.p.m. or 1280 ft. per min.
Load varied from 0 to 228 $lb./in^2$.
Oil used - Gas Engine Lubricant.

See separate sheet for table.

Conditions were difficult to keep constant during the latter period of this test. An examination of the bearing showed signs of seizure of bearing surface. Z is viscosity in poise.

Series I. test (c) - Temperature kept constant at 190°F.
Speed varied from 0 to 1280 ft. per min.
Load varied from 0 to 228 lb. per in^2.
Oil used - Gas Engine Lubricant.

See separate sheet for table, p. 26.

As in Test (c) the bearing showed tendency to lack of oil supply and seizure of bearing surfaces.

Although this investigation deals chiefly with the effect on friction of different finishes of materials, and different oils/

Temp. of Oil Film	Load lb/in^2	Rev. per min.	W_1 lb	W_2 lb	$W_1 + W_2$ = W lb	T = friction Torque $= \frac{1}{2}$ W x	$\mu = \frac{T}{1.25\ XL}$	$V/\sqrt{P}$ lb.ft.min.
66°F.	56	50	0.5	0.1	0.6	6.75	0.016	5.52
"	"	100	0.55	0.3	0.85	9.56	0.0229	10.5
"	"	200	0.7	0.5	1.2	13.5	0.0323	21.0
"	"	250	0.8	0.6	1.4	15.74	0.0377	26.3
"	"	300	0.8	0.65	1.45	16.3	0.039	31.4
"	"	350	0.9	0.65	1.55	17.42	0.0418	37.1
"	"	400	0.95	0.65	1.6	18.0	0.0431	42.0
66°F.	100	50	0.1	0.5	0.6	6.75	0.0095	3.97
"	"	100	0.2	0.6	0.8	9.0	0.0.26	7.85
"	"	200	0.4	0.85	1.25	14.05	0.0197	15.7
"	"	250	0.5	0.9	1.4	15.72	0.0214	19.67
"	"	300	0.6	0.95	1.55	17.42	0.0244	23.6
"	"	350	0.6	1.0	1.6	18.0	0.0252	27.5
"	"	400	0.65	1.0	1.65	18.55	0.026	31.4
66°F.	160	50	0.7	0.0	0.7	7.87	0.007	3.13
"	"	100	0.9	0.1	1.0	11.25	0.01	6.16
"	"	200	1.2	0.3	1.5	16.87	0.015	12.8
"	"	250	1.3	0.4	1.7	19.1	0.017	15.5
"	"	300	1.4	0.45	1.85	20.8	0.0185	18.65
"	"	350	1.45	0.45	1.9	21.4	0.019	20.9
"	"	400	1.5	0.5	2.0	22.5	0.02	24.8
66°F.	228	465					0.022	25.0
"	"	700					0.023	39.0
"	"	932	Large pulley on Counter Shaft, Bearing kept cool.				0.026	48.5
"	"	1160					0.031	60.5
"	"	1400					0.037	72.5
"	"	1600					0.046	84.5

Temp. of Oil Film	Load lb/in^2	Revs. per min.	W_1 lb.	W_2 lb.	$W_1 + W_2 = W.$	Friction Torque.	$\mu = \frac{T}{1.25\ L}$	$\frac{P}{ZN}$
138°F.	56	50	0.1	0.25	0.35	3.94	0.0094	1.8
"	"	100	0.2	0.45	0.65	7.3	0.0174	0.94
Z = 0.6	"	200	0.4	0.6	1.0	11.25	0.0268	0.47
poise	"	250	0.5	0.7	1.2	13.5	0.0322	0.37
	"	300	0.6	0.7	1.3	14.6	0.0349	0.37
	"	350	0.65	0.75	1.4	15.75	0.0375	0.27
	"	400	0.7	0.8	1.5	16.9	0.04	0.23
138°F.	100	50	0.1	0.35	0.45	5.06	0.0071	3.3
	"	100	0.2	0.45	0.65	7.31	0.0103	1.6
	"	200	0.3	0.6	0.9	10.1	0.0142	0.8
	"	250	0.4	0.7	1.1	12.34	0.0174	0.66
	"	300	0.5	0.75	1.25	14.05	0.0197	0.53
	"	350	0.55	0.8	1.35	15.2	0.0213	0.47
	"	400	0.6	0.85	1.45	16.3	0.0228	0.4
138°F.	160	50	0.1	0.3	0.4	4.5	0.004	5.3
	"	100	0.2	0.5	0.7	7.88	0.007	2.6
	"	200	0.2	0.6	0.8	9.00	0.008	1.3
	"	250	0.2	0.65	0.85	9.56	0.0085	1.06
	"	300	0.2	0.7	0.9	10.1	0.009	0.9
	"	350	0.25	0.75	1.0	11.25	0.01	0.75
	"	400	0.3	0.8	1.1	12.4	0.012	0.67
138°F.	228	465	0.35	0.85	1.4	15.6	0.015	0.82
	"	700	0.9	1.0	1.9	21.7	0.021	0.54
	"	932	0.9	1.25	2.15	23.8	0.023	0.41
	"	1160	1.0	1.45	2.45	27.0	0.026	0.33
140°F.	"	1400	0.9	1.55	2.45	27.0	0.026	0.27
145°F.	"	1600	0.7	1.65	2.35	26.0	0.025	0.23

Temp. of Oil Film	Load lb/in^2	Revs. per min.	w_1 lb.	w_2 lb.	$w_1 + w_2 = w$	Friction Torque.	$\mu = \frac{T}{1.25 L}$	$\frac{P}{ZN}$
190°F.	56	50	0.3	0.2	0.5	5.62	0.0134	9.4
	"	100	- 0.5	0.9	0.4	4.5	0.010	4.7
Z = 0.16	"	200	- 0.7	1.0	0.3	3.37	0.008	2.35
	"	250	- 0.5	0.9	0.4	4.5	0.0108	1.9
	"	300	- 0.5	1.0	0.5	5.63	0.0135	1.56
	"	350	- 0.3	1.1	0.8	9.0	0.0215	1.35
	"	400	- 0.2	1.05	0.9	10.1	0.0242	1.18
190°F.	100	50	0.1	0.3	0.4	4.5	0.0063	12.5
	"	100	0.05	0.25	0.3	3.37	0.0047	6.2
	"	200	0.15	0.45	0.6	6.75	0.0095	3.1
	"	250	0.18	0.54	0.7	7.88	0.0138	2.5
	"	300	0.20	0.6	0.8	9.0	0.0126	2.1
	"	350	0.15	0.55	1.1	12.4	0.0111	1.7
	"	400	0.2	0.7	1.2	13.5	0.0126	1.5
190°F.	160	50	0.1	0.5	0.6	6.75	0.006	20.0
	"	100	0.1	0.3	0.4	4.5	0.004	10.0
	"	200	0.15	0.5	0.64	7.2	0.0064	5.0
	"	250	0.15	0.55	0.7	7.88	0.007	4.0
	"	300	0.2	0.6	0.8	9.0	0.008	3.3
	"	350	0.2	0.7	0.9	10.1	0.009	2.8
	"	400	0.2	0.8	1.0	11.25	0.01	2.5
190°F.	228	465	1.90	1.6	3.5	39.5	0.035	3.07
	"	700	1.50	1.5	3.0	33.7	0.03	2.05
	"	932	1.35	1.35	2.7	30.5	0.027	1.54
	"	1160	1.15	1.25	2.4	27.0	0.024	1.24
	"	1400	1.02	1.25	2.3	26.0	0.023	1.02
	"	1600	1.0	1.2	2.2	25.0	0.022	0.90

oils used as lubricants, it would be of interest to examine the results in the light of these obtained by the new Bearing Testing Machine used by Thomson[6]. Due to the method of loading the ring lubrication almost failed to act, and the displacement of the Bearing on the shaft endways had to be made use of to assist film lubrication. It was found almost impossible to measure angular displacement or eccentricity, but it was easy to obtain 'greasy' lubrication conditions, and conditions of bearing at low peripheral velocity of the shaft. The temperature could be limited, or the heat generated easily accounted for, and put in the form of Heat Generated $= \mu \frac{W V}{J} = k.D.L.$ $L \times D$ bearing projected area. The k-value is simply obtained, as the base and cover is much similar to that of a heated rectangular cast iron bar. $\frac{W}{DL} = P =$ load, lb/in^2 of bearing surface, and therefore $PV = \frac{kJ}{\mu}$. With the machine in use, it is not certain that an oil film is obtained or maintained, but with the new machine at starting and stopping the film can hardly be in existence over a certain period unless special provision is made for its formation and maintenance. The two aims of reliability and reduction in friction, at start and finish of the period of running, are in opposition in certain respects since reliability and provision against overloads and reduced speeds entails the use of oil of high viscosity, working at a low value of P/ZN, Consequently, with a large value of minimum oil thickness and high μ - value. A low coefficient of friction necessitates the use of oil possessing low viscosity and working at a high value of P/ZN, resulting in smqll value of minimum oil thickness.

To design a good working bearing investigators have stated that it is necessary to choose a journal diameter, clearance, bearing, etc., transverse width of bearing, ahd position of bearing with respect to load line, in addition to other details such as method of supply of oil, provision of oil grooves, nature of bearing surfaces, and general arrangement/

arrangement of bearing. This appears to leave out the main points, - what is the relation between the static and moving friction? how long is it before static and moving friction become normal? An engine on standing has both its static and moving friction growing rapidly during the first 30 seconds, this approaching its ultimate value after 1 minute. At the starting up of engines similar to the ROYAL TECHNICAL COLL. National Gas Engine, where the moving parts are excessively heavy, bearings have been destroyed by static friction due to faulty lubrication. The same thing is liable to occur in stopping this class of engine. Film lubrication on which the design of the bearing has been based has, under these circumstances, failed, and seizure of the bearing has ensued. The static torque when the engine is lightly loaded may be four times the moving, a figure determined from a 40 B.H.P. Gas Engine. A combination of clearance and bedded bearing was applied with success to guard against the danger zone which occurs between the limits of 0 and 80 r.p.m.

Considered from the point of failing lubrication, the bearing machine used gives valuable information as regards the lubricating values of mineral, vegetable or fatty oils, comparison between new and worn oil, temperature of oil film, effect of rubbing speed and loading pressure, closeness of fit and nature of bearing surfaces. The machine can be made to give solid friction, boundary or greasy lubrication, and, with careful manipulation of bearing film lubrication.

The viscosity tests applied to several of the oils used in these tests are shown in Figs. 11 (a), (b) and (c). The blended oils used were made from non-oxidizable mineral oil. The air in the room in which the tests were carried out was fairly free from grit and dust, and therefore the wear/

FIGURE 11a.

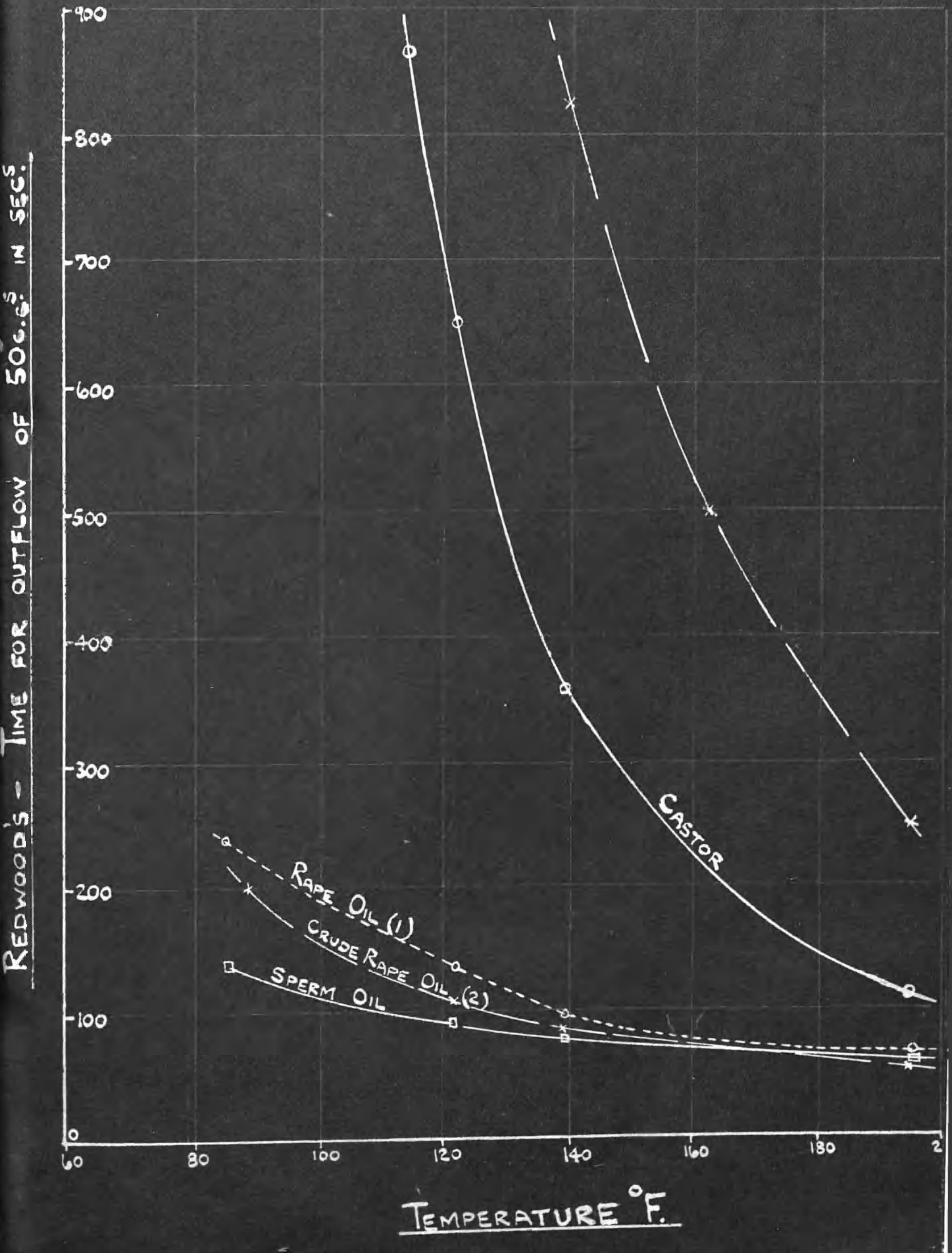

REDWOOD'S - TIME FOR OUTFLOW OF 50c.c.s IN SECS.
900
800
700
600
500
400
300
200
100
0
60
80
100
120
140
160
180
2
TEMPERATURE °F.
CASTOR
RAPE OIL (1)
CRUDE RAPE OIL (2)
SPERM OIL

FIGURE 11b.

COMPARATIVE VISCOSITIES IN REDWOOD TIMES O

MACHINE OILS - M, V, AND G.
BUCHANAN ENGINE OIL - B.
DIESEL ENGINE OIL - D.
GAS ENGINE LUBRICATING OIL - V1.

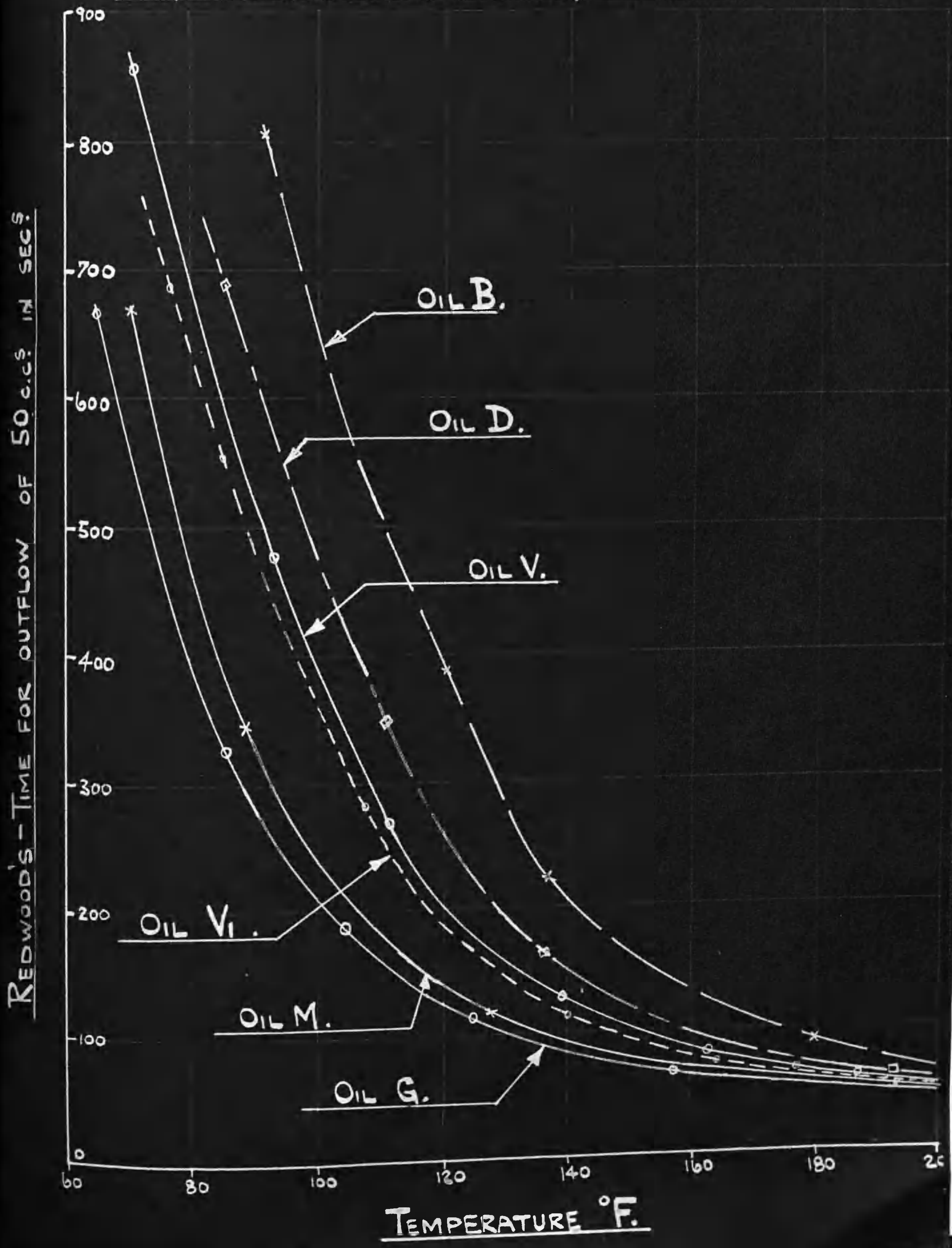
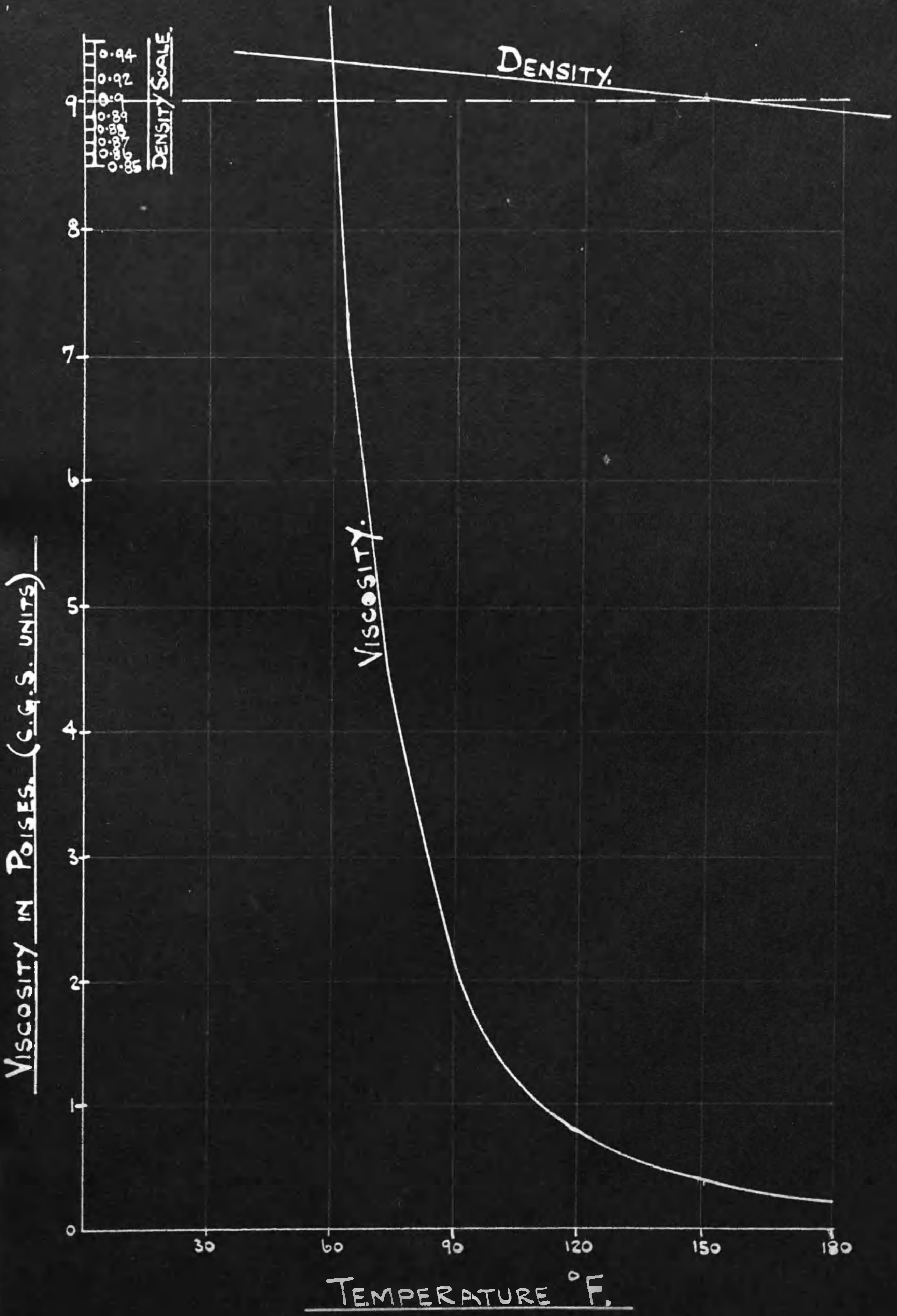


FIGURE 11c.



CHANGE OF VISCOSITY AND DENSITY WITH TEMP^R CHANGES FOR MACHINE OIL - B.

wear qualities of the oils were obtained under ideal conditions. It is certainly a defect that in addition to the change of pressure which takes place between the inlet and the outlet edges of the bearing surface, the temperature variations which are sure to take place have not been recorded as these alter the viscosity and density of the oil. In this bearing there is no doubt about where the maximum pressure and temperature occurs, and this is shown by the wear zone on the bearing surface. Referring to figure 4 (b), which shows a diagramatic oil flow for the bearing, the arc of embrace can be varied in this Series i (a), (b) and (c). The arc is 65°. The bearing surface is not continuous as was shown in Figure 4 (a), but is made up thus.- 1.375" ($1\frac{9}{16}$" + $\frac{15}{16}$" + $1\frac{9}{16}$") = 1.375 x 4.06 = 5.6 in^2.

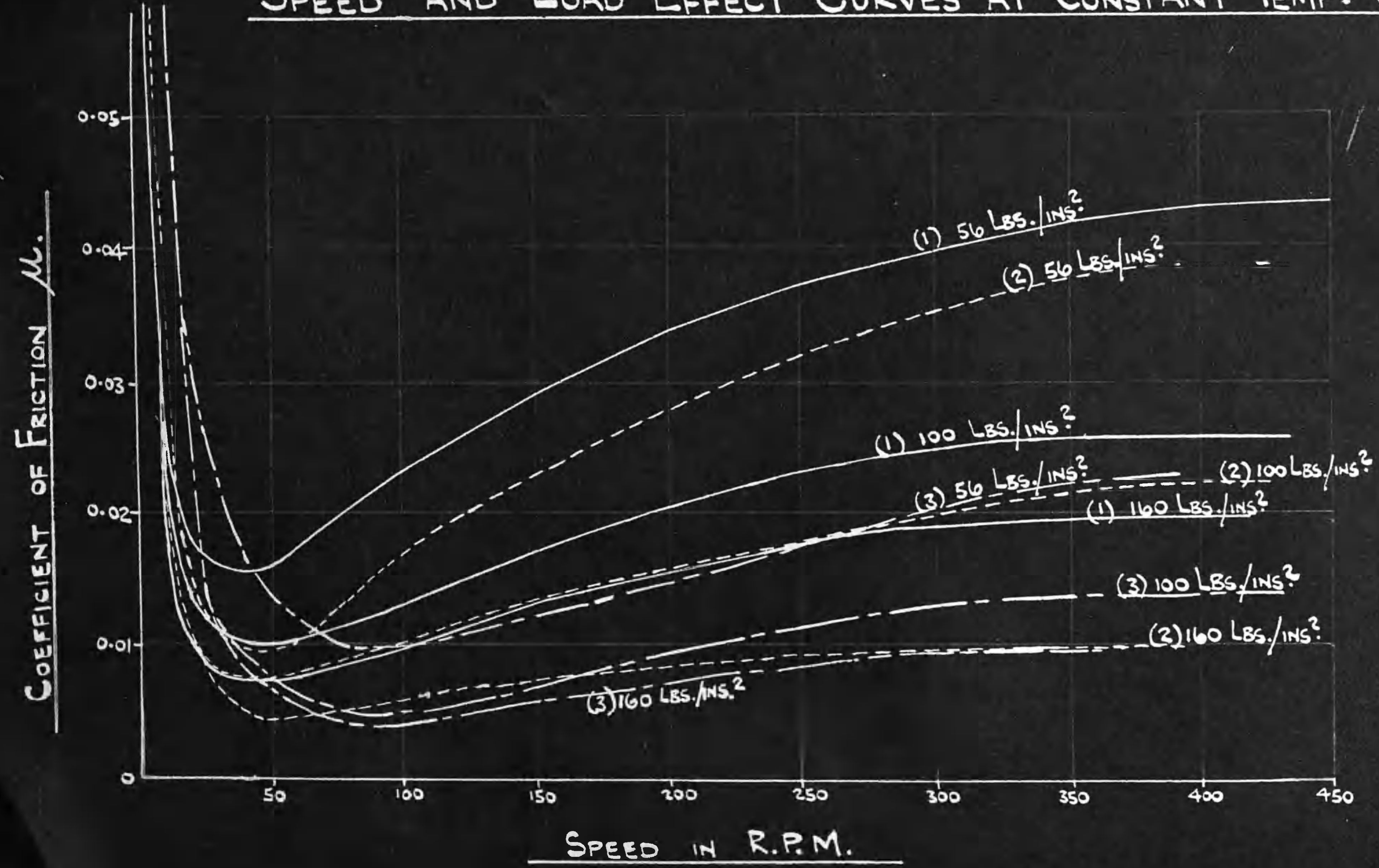
Actual loading 280 + 36 = 316 lb. giving $\frac{316}{5.6}$ = 56 lb/in^2
280 + 36 + 244 = 560 lb. giving = 100 lb/in^2
280 − 468 + 102 + 50 = 900 lb. giving = 160 lb/in^2
280 + 468 + 102 + 254 + 56 = 1160 lb. giving = 228 lb/in^2

The temperature of the oil film was determined by fitting an iron-constantin electro couple junction at the side of the bearing block. Two ¼-in. diameter holes are drilled in the side of the brass and connected at the surface by means of a shallow groove. Two insulating plugs ¼-in. diameter, and drilled with a very small spiral drill, were driven into the bush and the thermo couple connected, Figure 12. p.11 A sensitive millivoltmeter was connected to the other end of the junction, and the results balanced with a standard cell.

The Effect of Temperature is seen from the tables given but for convenience these have been graphed in Figs. 13 and 14. It is seen for all speeds that the value of decreases, due to the decreased viscosity of the oil with rise of temperature. This is less noticeable with lower speeds, and at about 50 r.p.m. the value of μ increases if the temperature is raised above 120°F. These curves show clearly/

FIGURE 13.

SPEED AND LOAD EFFECT CURVES AT CONSTANT TEMPR. OF :-



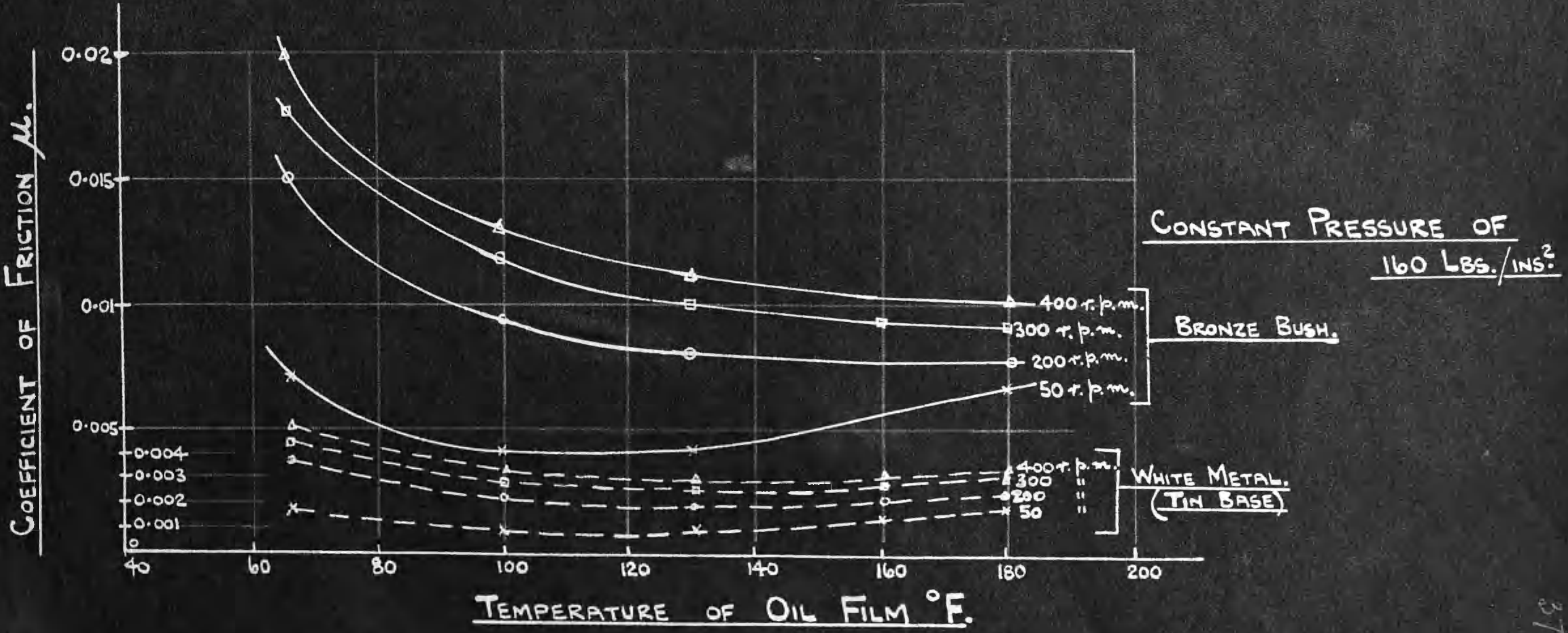
clearly the danger of running a National Gas Engine[8] at slow spped (starting up or slowing down). The rotating masses of this class of engine are very heavy and film lubrication is difficult to obtain and maintain. The bearing when slowing down after running has the disadvantage of being in the region of 100° or 120°F. and since at this temperature the viscosity has decreased until the oil film cannot support the load. The curves show that a certain increase in temperature above normal, when the bearing is running at speeds above 150 r.p.m., is not only beneficial but advisable and economical.

A bearing will function with less loss at 40°F. above room temperature due to decrease in the μ - value. It is here that the tests as carried out would assist in the design of intercooled bearings. The effect of increasing the clearance space from 0.001 per inch diameter to 0.004D. The friction is noticed to be less with a flooded bearing, and large clearance, but is subject to 'knock', most unsuitable for electrical machinery, yet necessary in the crank pins of a reciprocating engine. The author stripped a crank pin bearing, bedded carefully the bearing surface, carefully cleaned out the oil grooves and gutters, gave the usual clearance space, ran the engine on light load, yet after leaving port this engine had to be stopped and the clearance increased due to malalignment.

To design a bearing on a series of tests run on a small experimental bearing under perfect conditions would be to court disaster, yet the data when used judiciously will prove valuable. Film thickness under normal load and speed conditions must be sufficient to guard against the possibility of contact if vibration be set up due to external influence. Dr. Shamnon has shown how lubrication is/

FIGURE 14.

SPEED AND TEMPERATURE EFFECT CURVES AT CONSTANT LOADS.



is used to damp out vibrations. It would appear that the designer of bearings must make provision for the use of lubricating oil as a buffer. Shaft vibrations can be traced as the cause of the destruction and seizure of surface of journal and bearing materials.

Radiation of Heat. Using the curve for average bearings as derived by Lasche[9] the formula for which is approximately - ft.lb. radiated per sq.in. projected area, per deg.F. temp. diff. per min.

$$= 0.01\ (t^{o}oil - t^{o}air) + 1.9$$

$$= 0.01\ (190 - 70\ ) + 1.9 = 3.1$$

The radiating capacity of the bearing block and bearing material test piecesC must be greater than or equal to

$$\frac{51V}{(t_1 - 32)(t_1 - t_a)} = \frac{51 \quad x \quad 1281}{158 \quad x \quad 120} = 3.45$$

In this case, when the room temperature is 10°F. the oil temperature, 190°F., and the r.p.m. 1600, the maximum allowable heating has been obtained. This confirms the tendency to seizure in the series i (c). The point brought out here is one which is often given little consideration in design, namely the dissipation of heat from the bearing block. This may either be done by an automatic increase in the oil supplied to the bearing, say by rheostat control, or by supplying water to a jacket surrounding the bearing. Conducted heat through mild steel usually taken as 0.9 to 1.1 watts per sq.in. for 1°C. rise, and for cast iron 1 watt per sq.in. per 1°C.

The designer must make provision for the momentarily changes in load and speed, followed by increase in temperature which brings with it a change of viscosity of the lubricant. In film lubrication there is just the possibility that contact between the bearing and journal material may take place due to vibrations being set up by external influences.

Hard/

Hard foreign material may pass through the oil filter and be swirled round by the oil film, thus causing wear which must be considered. These facts must be considered when choosing the value $\frac{P}{ZN}$ or $\frac{ZN}{P}$ suggested by investigators who submit experimental data as a method of design. Almost all mechanical engineering consists, when applied to actual designing, in an approximate application of "exact science", the value of the approximation being fixed by the sound judgment of the designer.

The formula, based on a film shear resistance of 0.45 lb. per sq.in. for 1 B.Th.U. lost in journal friction per hour = 190ℓ "d" v(ft.per sec.) $\div$ $t^{0} - 32$, has given way to the formula by Boswall $^{10}\mu = (\frac{ZN}{P})^{.58}$, or that adopted by Thomson $\mu = H(\frac{V}{P})^{\frac{1}{2}}$ where H varies with loading.

The Effect of Speed of Rubbing Surface

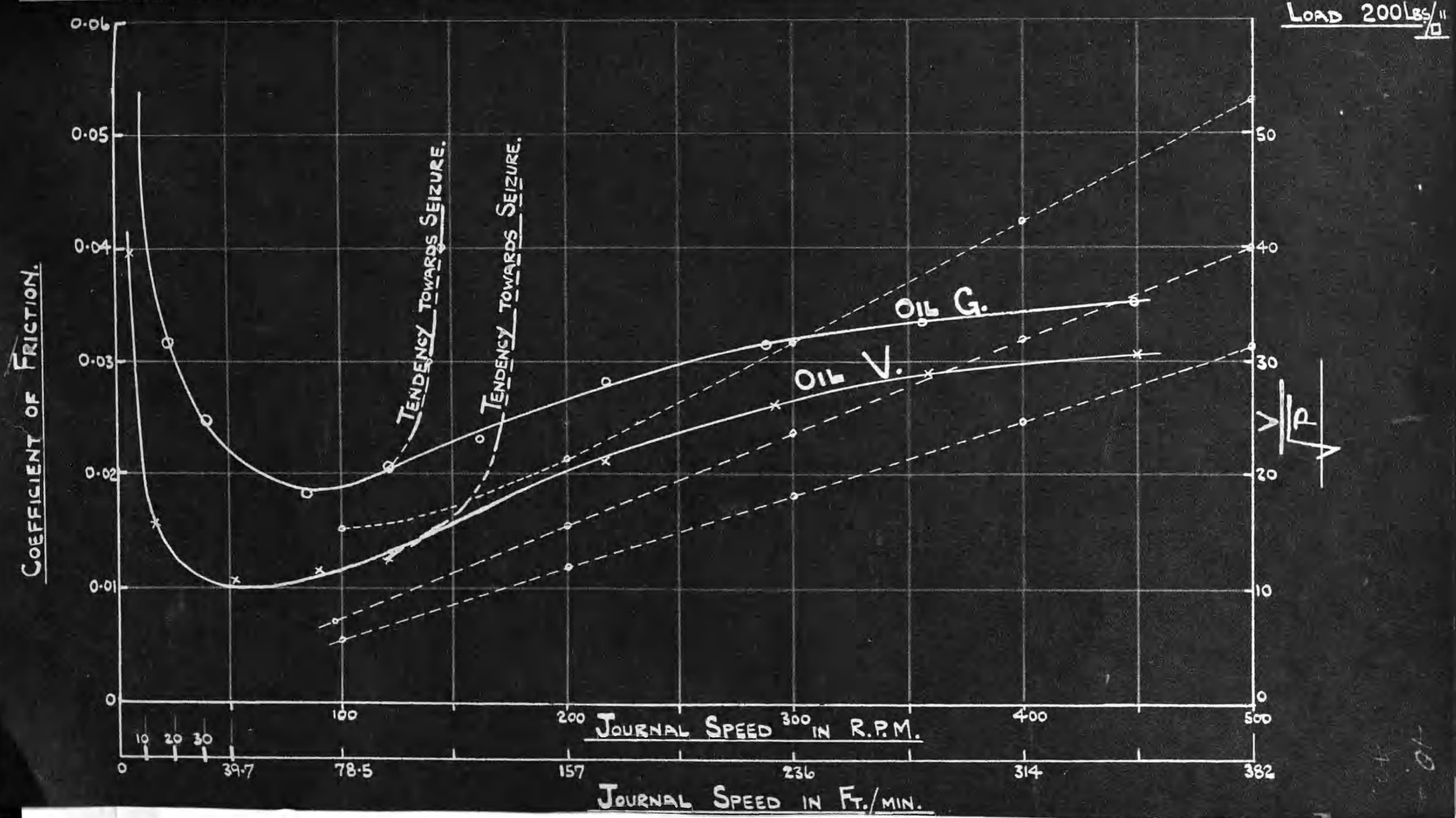
Fig. 15 in addition to showing the temperature effects shows that with higher running speeds (which are proportional to speeds of rubbing surfaces) there is less tendency for the viscous film to break down. This is an important conclusion and indicates that the viscous film is being constantly repaired and replaced at high speeds. The effect of speed on the value of μ at constant temperatures is shown by curves, Figs. 13 - 15, indicating clearly that with increasing speed three conditions of lubrication are successively passed.

i. - At zero speed a condition of no lubrication exists, and with perfectly cleaned bearing surfaces solid friction exists. The absorbed oil film on the bearing surface gives some lubricating effect, but seizure soon takes place. The steel journal as well as the bronze bush being effected, and their smooth surface is easily damaged.

ii. - At speeds up to 40 r.p.m. the value of μ decreased to a minimum value due to the forcing of more and more oil between the surfaces until perfect viscous film lubrication was obtained. The minimum value of μ in each case would be reached/

FIGURE

BEARING MACHINE. COEFFICIENT OF FRICTION – SPEED CURVES FOR OILS V AND G. TEMP^R. 100°F



reached just when the wedging action of the oil is sufficient to separate the rubbing surfaces.

iii. - At speeds above 40 r.p.m. the value of increased to an indefinite limiting value in each case. The increasing μ-value with increase of speed is due entirely to the faster rate of shear obtaining within the viscous oil film.

An interesting result is obtained by plotting μ against speed, N, logarithmically, and as will be seen from Fig.15_a a straight line law is obtained for speeds above 50 r.p.m., i.e. when viscous film condition holds.

Relation between μ and N as shown by graph is

$$\log \mu = \text{constant} + 0.5 \log N = 0.5(\log N - 2C)$$

but $2C = 2 \log C_1$

$$\therefore \log \mu = \cdot 5\left(\frac{\log N}{2 \log C_1}\right)$$

as the temperature was kept constant, viscosity, Z, is constant, and with constant pressure $\frac{Z}{P}$ is constant.

$$\therefore \mu = K\left(\frac{ZN}{P}\right)^{\frac{1}{2}}$$ Z-value in poises.

$K = 0{\cdot}005$ for $P = 56$ lb/in^2; $K = 0{\cdot}005^2$ for $P = 100$ lb/in^2

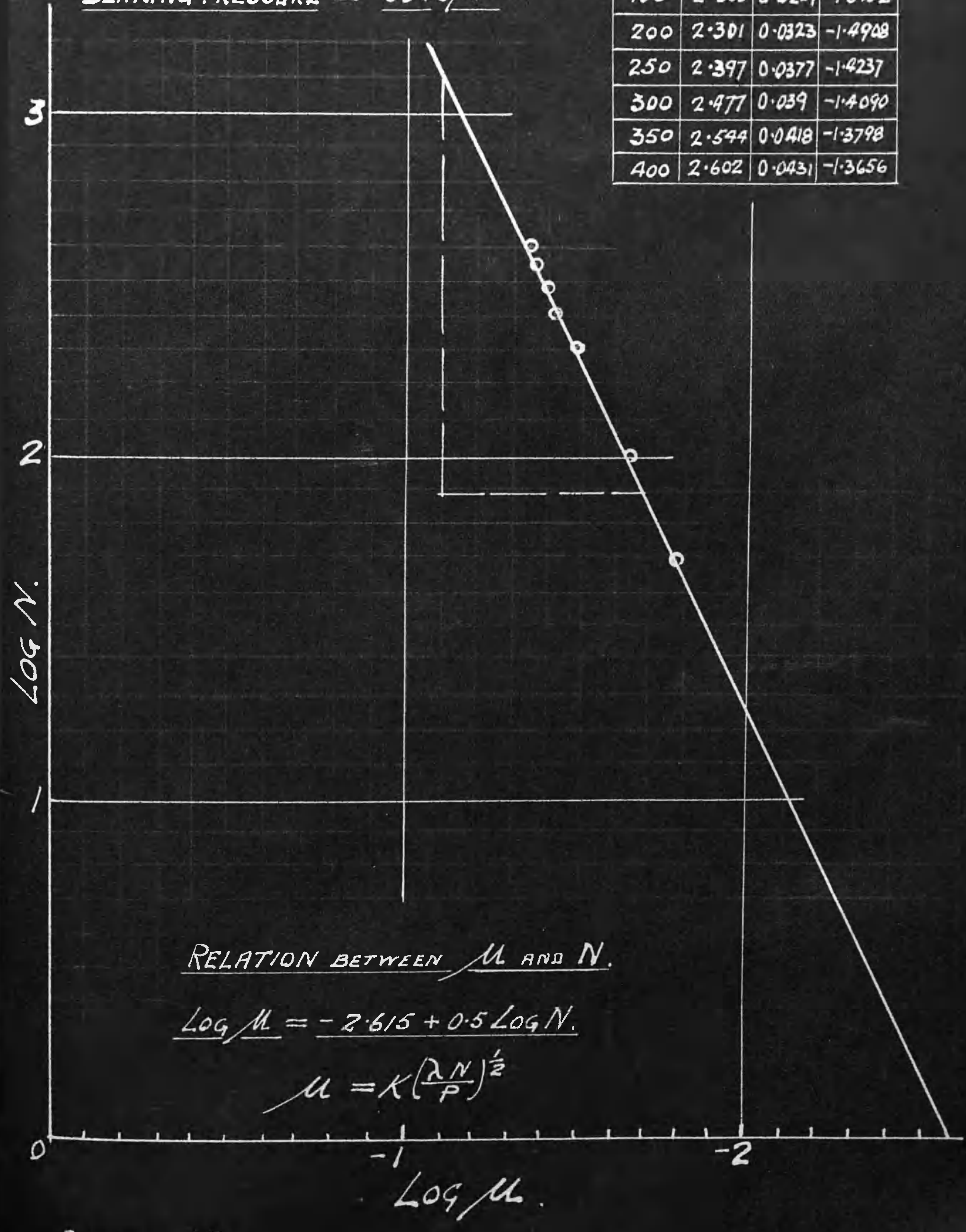
$K = 0{\cdot}005^5$ for $P = 160$ lb/in^2; $K = 0{\cdot}006^5$ for $P = 228$ lb/in^2

The equation is based on the existence of a condition of viscous film lubrication, and its limits are fixed by the capacity of the oil film (a) to remain continuous at the given speed, (b) to remain continuous at the given load. As soon as the oil film begins to rupture and boundary lubrication commences, then at that point the equation ceases to be valid.

This semi-lubrication period has been divided into two portions 'unctuous' and 'semi-fluid' phases. The first where $V/\sqrt{P}$ is less than 0.075 lb.ft.sec. units, the coefficient/

TEMPERATURE = 65°F.
VISCOSITY OF OIL = 8·5 POISES
BEARING PRESSURE = 56 LB/IN^2

N r.p.m.	Log r.p.m.	μ	Log μ
50	1·699	0·016	-1·7960
100	2·000	0·0229	-1·6402
200	2·301	0·0323	-1·4908
250	2·397	0·0377	-1·4237
300	2·477	0·039	-1·4090
350	2·544	0·0418	-1·3798
400	2·602	0·0431	-1·3656



RELATION BETWEEN μ AND N.

$$\text{Log}\,\mu = -2{\cdot}615 + 0{\cdot}5\,\text{Log}\,N.$$

$$\mu = K\left(\frac{\lambda N}{P}\right)^{\frac{1}{2}}$$

FIGURE 15a.

coefficient of friction is independent of the pressure, and diminishes slightly with increase of temperature; viscosity plays no part in this diminution, which is due to molecular influences; the coefficient of friction diminishes as the speed increases, the curve of variation being of similar form to that obtained with dry surfaces. The latter, where $V/\sqrt{P}$ exceeds 0.075, the coefficient of friction increases rapidly; it is better for the designer to aim at keeping a bearing in this phase as the risk of seizure is less, and wear not so rapid, while the coefficient of friction is independent of viscosity.

THE LOADING PRESSURE

The remarks made in connection with temperature effects and speed effects have in some measure covered the influence of the load pressure on lubrication. It is seen that with excessive pressures the oil film is unable to remain continuous. This gives rise to increased friction as the surfaces are brought into intimate contact, more noticeable at low than high speeds. The limit of the pressure with perfect film lubrication, high speed and perfect cooling, is the compressive stress of the bearing surfaces.

This investigation does not deal with this extreme case, but an example of an extra heavily loaded bearing is found on the roller-tracks of high-lift dock cranes. Special high pressure oil pumps are used, not only to circulate oil in the bearings, but to ensure that there is a film of oil between the loaded surfaces before starting. 1500 to 3000 lb/in^2 oil pressure applied to the zone of greatest pressure, so that the bearing load is lifted free from the shaft, thus eliminating all metallic contact, may be used. The power lost in starting, when high pressure lubrication is used, is about 1/3 that of the ring bearing lubrication. Under heavy load, and ring lubrication, the temperature rose 82 to 84°F. in 10 minutes, while with a high pressure lubricator/

lubricator it might only be 50 to 54^{o}F. in 10 minutes.

This point is most noticeable in journal bearings and is reflected in the general use of heavy oils and greases of extremely high viscosity on bearings of this nature. There is a great loss of power in starting up shop shafting due to the desire to save the shaft and its bearing from seizure. Also, by the use of heavy greases and tallows, railway wagons, reversing gears, and all types of machinery in intermittent action, are safeguarded from excessive bearing friction and wear at the instant of starting up. This is certainly an important factor which must be recognized and covered for in all design, - The ~~bearing~~ shaft must not only run but start safely under load.

In the machine under test the static friction, even at light loads, is at least four times the moving friction, and on stopping the shaft's rotation, the static and the moving friction grow rapidly. The value of the cranking torque, from 0 to 100 r.p.m., must be of the form $T = A + B\sqrt{NZ}$.

EFFECT OF WIDTH OF OIL-RING AND DEPTH OF IMMERSION

Very little has been done on the friction in actual oil ring bearings, while experimental and theoretical determination of friction in tubular bearings are available. The power necessary to drive the oil rings is quite small. Experiments on a 27-in. diameter ring, ½-in. wide, shows that the speed is almost proportional to the speed of the journal and reaches 0.15 H.P. at 1000 r.p.m.

For a ½-in. width, $Q = .0008\ N - .22.$

For a ¼-in. width, $Q = .0004\ N - .09.$

Q = gallons per min.
N = r.p.m. of shaft.

It is found that the largest quantity for any width is reached when the ring is immersed in the oil with the inside of the ring just covered. Light rings made of aluminium and even aluminium-bronze are noticed to slip, decreasing their efficiency/

efficiency, yet a heavy lead ring does not show any advantage over a hard rolled brass ring. The light rings at the high speeds become quite ineffective, but at the speeds used the film does not appear to have formed between the ring and the journal. Only that oil carried inside the ring is supposed to be used. More than this quantity can be depended upon to reach the shaft. The oil delivery is imrpoved by allowing the ring to ride over the outer surface of the bearing in addition to its contact with the shaft. Side scrapers are fairly effective, although each type of scraper will cause slip of ring.

The size of the side reliefs, and the design of the top bearing, require careful consideration. The average ratio of the diameter of ring to shaft is 1 to 0.6. In this investigation the shaft is 2½-in. diameter and 5-in. long, using a diametrial clearance of 0.002 per inch diameter. Then the radial clearance is 0.0025, and assume that the velocity distribution, where the minimum film thickness is 0.001 in. is linear, then the quantity of oil required at this point will be the product of the semi-peripheral velocity of the shaft, the film thickness and the transverse width of the bearing resulting in a quantity of 3 cubic centimetres per sec. Each ring actually delivers -

Q = .0004 x 900 - 0.09 = 0.2 gallons per min.
= 3.1 cub.cms.

That is, the two rings supply double the quantity required, which is satisfactory, taking into account the fact that the actual quantity of oil required will be greater than this amount on account of end leakage in the bearing up to the point of minimum film thickness. Alteration of the ring thickness and corresponding weight will influence the oil delivered.

In this test bearing the rings were made ⅜-in. wide but the/

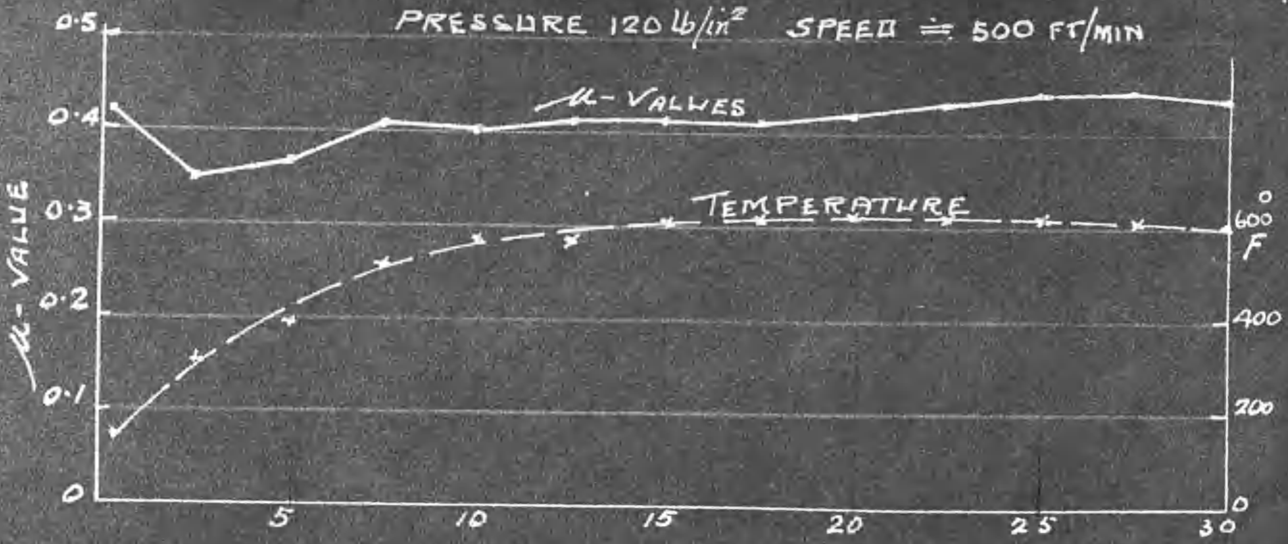


FIGURE 11 CURVE SHOWING EFFECT OF SLOWLY HEATING

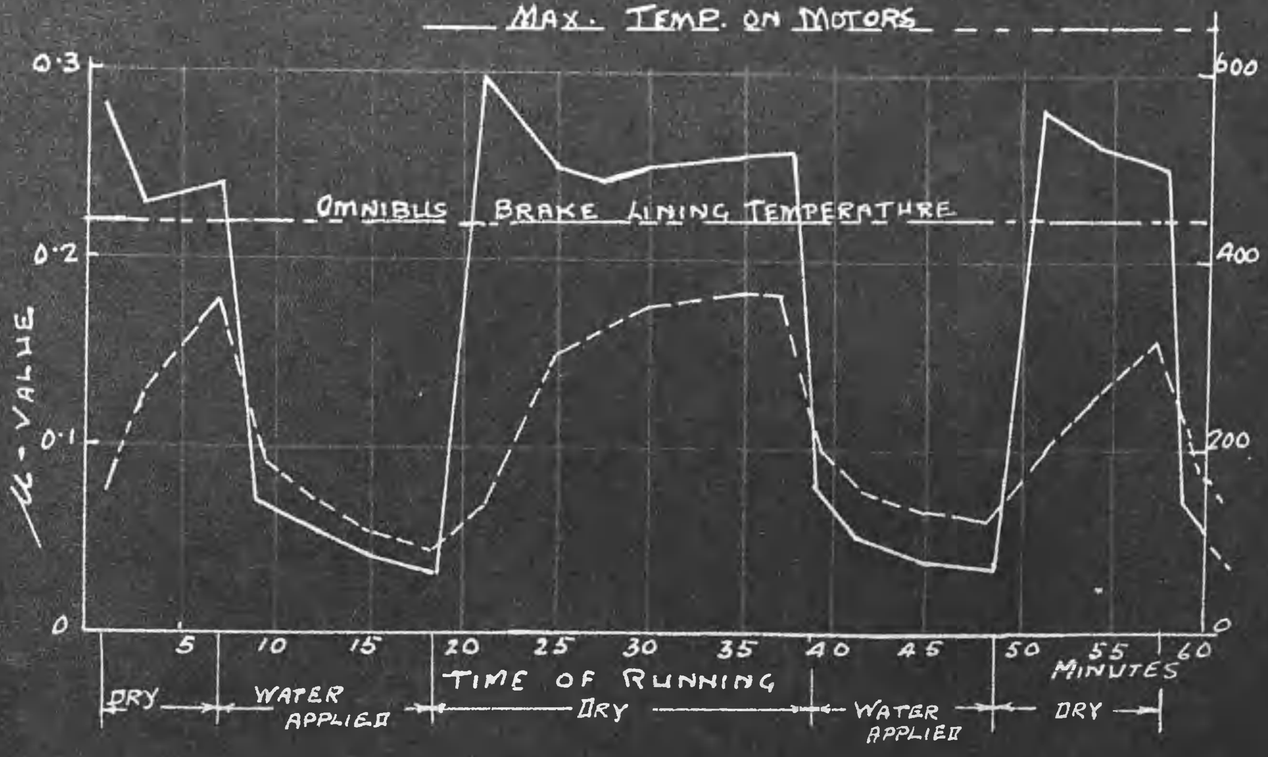


FIGURE 12 — EFFECT OF WATER APPLIED TO BRAKE WHEEL

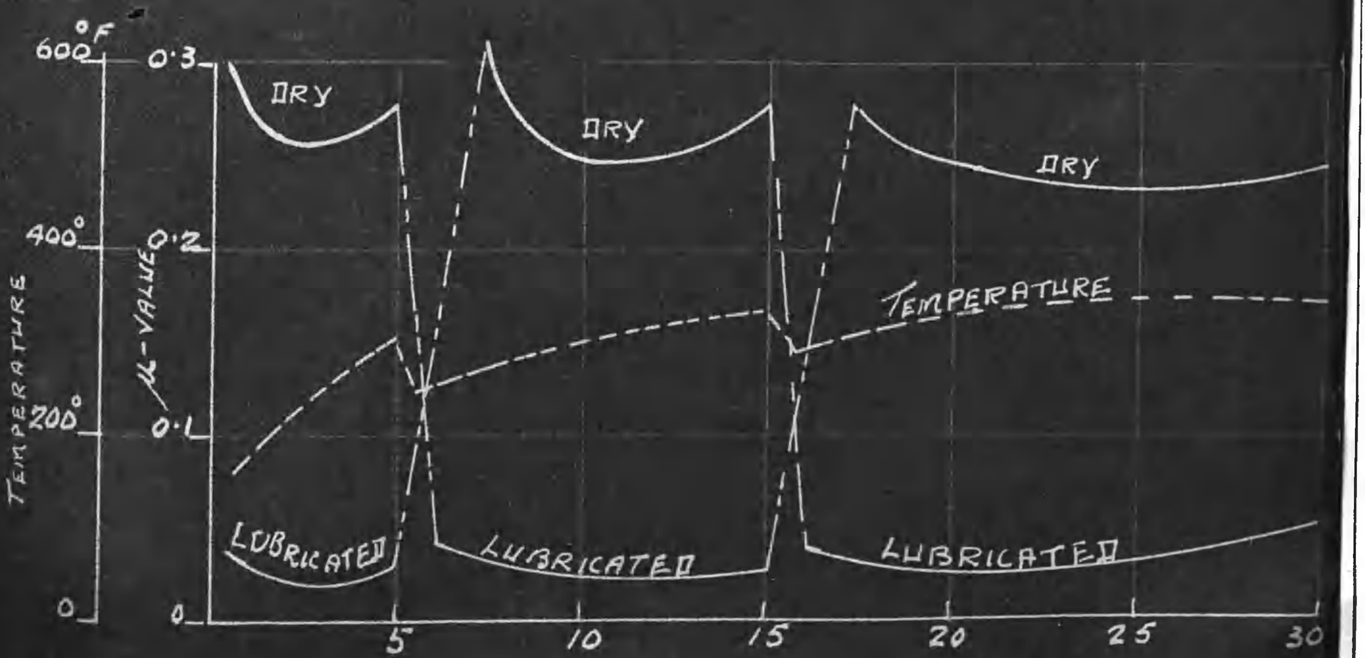


FIGURE 14. BRAKE RIM LUBRICATED WITH MACHINE OIL.

the thickness was reduced to keep the weight the same as a ring of ¼" x ¼" cross section. The friction of the bearing is not independent of the form of the end seals used to prevent leakage.

EFFECT OF FINISH OF BEARING SURFACES.

Although a number of oil testing machines, which will give quickly the lubricating qualities of oil, have been put on the market, measuring the finish of metal bearing surfaces is rather a different process. Quantitative evaluation of finish has still to be developed. Comparison tests, such as have already been described when using the Deeley machine, have been carried out on the bearing machine.

i. - The phosphor bronze bush was turned to fit the bearing block and then bored and machine scraped to leave 0.002 tolerance for running fit; the journal was oil lapped.

ii.- Bearing bush was stripped and diamond scraped in machine to leave 0.0015 tolerance.

iii. - Bearing bush was stripped, afterwards an expanding mandril was used, and then a ball or sphere finishing tool was used. This latter tool appreciably hardened the surface of the bearing material as shown by higher Brinell. Tolerance 0.001.

Note. - Tool ridges could without difficulty be seen on No.i., but with No.iii, finish it was quite difficult to discern high and low portions due to spiral feed.

The table shows the effect on starting and running of the machine.

Class of Finish	Coefficient of Friction		Temperature °F.		Remarks
	Starting	Running Average	Maximum Average	Room	
i Turned & scraped	0.277	0.018	168	55 to 65	Time of Test - 5 hours (each) 200 r.p.m. 160 lb/in^2
ii Special scraped	0.235	0.010	152		
iii Rolled	0.175	0.009	145		

This/

This test does not give a true comparison of finish as the oil film thickness varies with the clearance, but the clearance, or tolerance allowance, has been kept to the usual running fit allowance. The oil used is the same as that used in long running tests.

With finish (i) the journal was slightly damaged, and this after all care being taken to ensure greasy lubrication at start. It would appear that the wear of the ridges caused the accumulation of grit which worked along the bearing surface causing longitudinal scoring of journal, and pitting in bronze bush.

EFFECT OF SPEED AND LOAD INCREASE, TEMPERATURE CONSTANT ON μ-VALUE

	Temp. °F	Viscosity Z	Pressure lb/in^2	r.p.m. N	Coeff. friction	$\frac{P}{ZN}$	$\frac{V}{\sqrt{P}}$
(a)	86	2.6	120	135	0.017	0.62	0.128
			180	300	0.028	0.23	0.136
			240	915	0.032	0.09	0.70
			320	1540	0.033	0.08	0.97
(b)	140	0.5	120	135	0.014	1.8	
			180	300	0.015	1.2	
			240	975	0.016	0.49	
			320	1540	0.016	0.42	
(c)	195	0.1	120	135	0.012	8.9	
			180	300	0.014	6.0	
			240	975	0.016	2.5	
			320	1540	0.021	2.1	
(d)	245	0.08	120	135	0.017	11.0	
			180	300	0.016	7.5	
			240	975	0.015	3.1	
			320	1540	0.016	2.6	

(a) - With an oil temperature of 86°F., a rubbing speed of 10.6 ft.per sec., and pressures varying from 120 to 335 lb/in^2, the μ-values, 0.022 to 0.028, increase rapidly with increase of speed, and decrease slowly as speed decreases.

(b) - T = 140 F, V = 10.6 ft./sec., with pressures 120 to 180 lb/in^2, the μ-value is 0.018 to 0.02, much the same/

same as (a); at pressures 240 to 300 the decrease is much slower than (a) with decrease of speed; but at pressure 300 increase in μ takes place with either increase or decrease of speed.

(c) - With T = 195, V = 10.6 ft./sec., and pressure 120 to 180 lb., varies from 0.018 to 0.02 (max.) and critical speed appears at 3.38 ft./sec; at pressure 240 to 300 the μ-value has risen to 0.23 to 0.26, and at 17.2 ft./sec. μ falls slightly with increase of speed, but rises rapidly with decrease of speed. Wear takes place quite rapidly under this latter condition, and the swing of bearing is appreciable.

(d) - At T = 245°F., V = 17.2 ft./sec., μ = .018 is a critical position, μ increasing with either a decrease or increase of speed; between 200 and 335, μ = 0.023 to 0.03 and increased rapidly with decrease of speed; seizure would take place on stopping.

The conditions in (c) and (d) with the bearing pressure above 100 lb/in^2 would be unsuitable unless the oil could be supplied at pressure enough to give film lubrication, as greasy or semi-lubrication fails to prevent seizure when running up to speed or on stopping without first effectively cooling the bearing and journal. It is under these conditions that water jacketing of the plummer block would require careful consideration.

OIL AS A CONDUCTOR OF HEAT AND COOLANT

	Viscosity			Spec. Gravity	Spec. Heat.		Thermal Conductivity	
Mineral Oil	100	130	210°F	d	100	210°F	100°	210°F
Light oil	111	69	40.5	0.9129	0.45	0.514	0.872	0.844
Heavy oil	209	104	45.5	0.9159	0.449	0.516	0.869	0.840

Sp.ht. = $(t + 670)(2.10 - d)/2030$
Thermal Conductivity $k = 0.813/d\,[1 - 0.0003(t - 32)]$
d = sp.gr. of oil at 60 F, t = temp. in °F.
and k is in units of B.Th.U. per hr. per sq.ft. per in. per° F.

The/

The quantity of oil Q in lb/min with rubbing speeds from 5 to 50 ft./sec. and at constant pressure 100 lb/in^2 follows approximately the law $Q = 0.04V + 2.16$ lb/min. This quantity can be supplied by the rotation of the oil rings, but the quantity falls off rapidly with reduction in speed below 5 ft./sec. $Q\ gal/min = 0{\cdot}0008 V \times \frac{d}{2} \times l \left(\frac{ZN}{p}\right)^{\cdot 562}$ for Test Bearing.

The heat conductivity through the cast iron is in the region of $k = 21.6$ B.Th.U. per sq.ft. per °F. per hour.

The coolant property of the oil was much increased by the use of powdered graphite. For comparison, tests in the semi-lubrication phase were carried out with sulphurized mineral oil, and colloidal graphite compound, also one test with pine oil containing a high sulphur content.
The main function of sulphur, which has been used for reducing the temperatures in main bearings while running in, is to act as a coolant. Sulphur combines in three states, unstable, fixed, and semi-fixed; the last becomes unstable after a period of time, or under heat or pressure.

It was noticed that with a 5 to 15 per cent. mixture of flowers of sulphur the tendency to seizure was much reduced, and the torque required was reduced 60 per cent. The effect on the journal finish was noted, and when surface was examined under the microscope it was found to have a better finish than when test began. The film strength and adhesiveness of the lubricating medium permits of at least 20 per cent. higher loads being used with no sign of seizure at low speeds.

TESTS ON DIFFERENT MATERIALS

Conditions of these tests are in the semi-lubrication phase, and are run for about 10 minutes with failing lubrication, the oil rings having been removed. The effective scoring of the bearing surface gives an indication of the nearness to which the critical condition has been approached.

Lubricant Machine oil	Material	Speed ft/min.	Pressure lb/in^2	μ	Pressure lb/in^2	μ
190°F.	Gun Metal	3.5	100	0.016	175	0.018
	Lead Bronze		200	0.015	260	0.017
	White Metal		300	0.0130	320	0.0135

This table shows the average value for μ from a large number of tests, and these are the approximate critical loading points, also either an increase or a decrease of speed may cause seizure. Oscillation of the bearing machine on its journal took place near the critical positions.

Seizure of the Lead Bronze is quite positive and tends to destroy the shaft. It is found difficult to get a correct copper -tin - lead mixture with a hardness suitably adjusted to the mating journal.

Because of a high coefficient of expansion these materials required to be put on a tin steel back, the clearance space to allow of perfect running requires to be appreciably greater than with gun metal.

New Steel Shaft and White Metal Bush

As the shaft of the bearing machine was bent and due to permanent set was running out of truth, a new length of shaft was turned. Sample analysis of cuttings showed C = 0.35, Mn = 0.8, Si = 0.2, S = 0.037, P = 0.039. This is much the same as that used for railway axles; the Brinell Np. is 187. Two white metals were used on steel backs with similar bearing areas to that of gun metal bush and with total area = 5.6 sq.ins. The first lining (a), was high in lead value, Pb 75, Sb 21, Su 5, and the second (b), was high in tin, Su 80, Sb 15, Cu 5.

Brinell	Approx. Temperatures °F.	Material and H.
10 mm ball	86, 140, 195, 245.	(a) 21, 37, 13, 11.
500 kg.		(b) 31, 22, 17, 14.

The/

The tin-base alloy (b) is much stronger than (a), and is preferred to the lead-base alloy since its surface is not so easily damaged by any 'knock' in the bearing. The bearing is machined as follows:- (1) Lathe finish; (2) Grinding; (3) Lathe spring tool scraper followed by roller, and during the tests perfect lubrication maintained, care being taken at the starting up periods when ring lubrication was not effective. The speed is kept constant at approximately 570 revs. per minute and the load applied gives a pressure on bearing surface of 70 lb./in^2; duration of test in each case, 4½ hours.

Figure 16. shows the results of these tests plotted on a base of time against increase on temperature. Readings were taken every half-hour, but during the first period 5-minute readings were taken. From the plotted curves the following table of results are obtained and these are again plotted in Figure 17, p.57, with temperature and μ-values as base and abscissa.

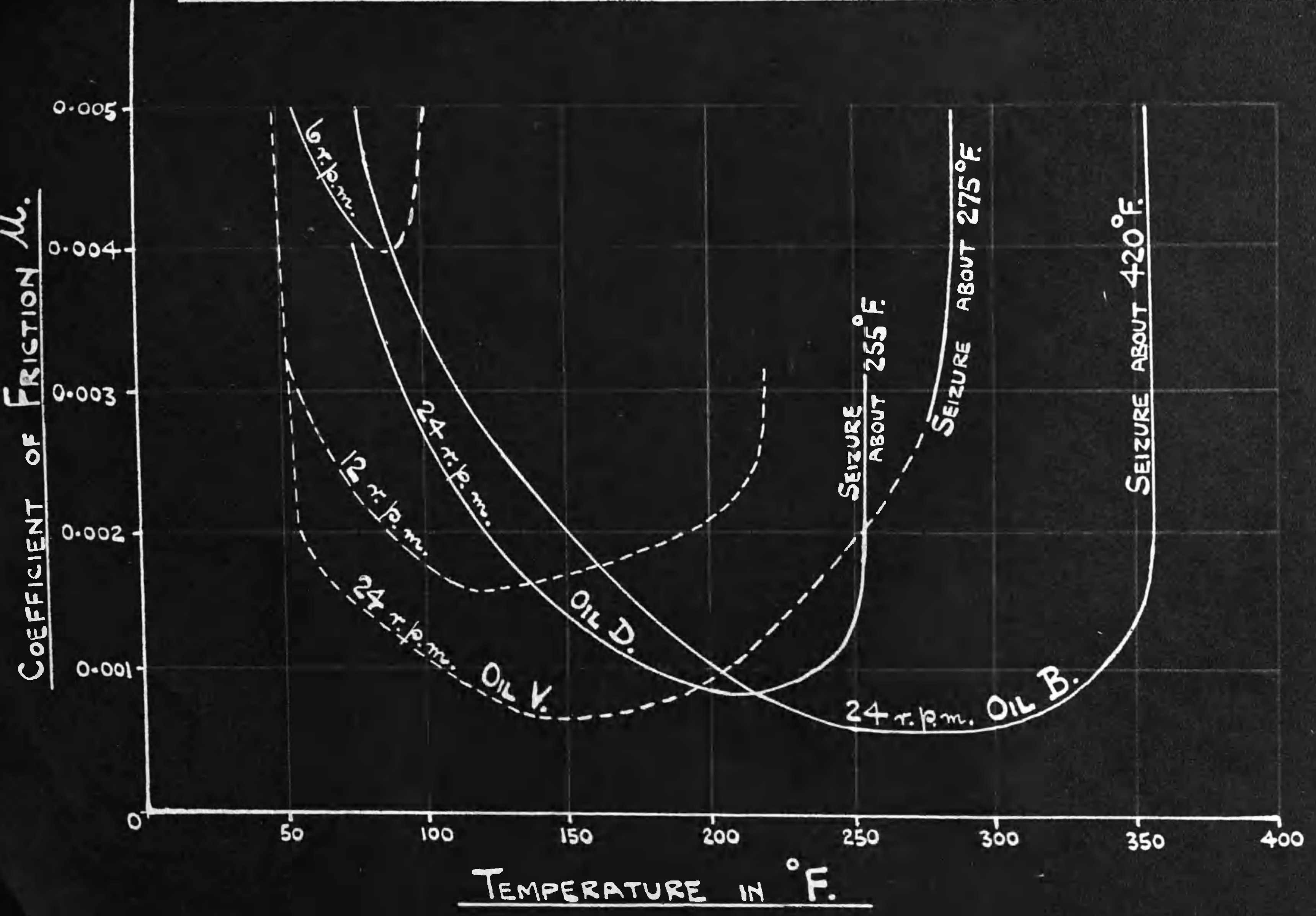
Tin-base white metal (curve — — dotted —)

Degree of Finish	Coeff. of Friction Starting	Coeff. of Friction Running	Max. Temp. °F.	Temp. Rise °F.	Clearance Increase
(1) Lathe finish	0.18	0.0032	210	142	0.002"
(2) Grinding	0.15	0.0018	156	87	0.0015"
(3) Mandril or roller	0.09	0.0015	142	73	0.0002"

It will be seen that the lead-base alloy had a lower friction value and a smaller temperature rise with the three classes of finish. The softness of this alloy may be got over by the addition of calcium, sodium or barium, but the addition of these may also cause disintegration when coming in contact with oxygen which is always present in lubricating oils. Any attempt to harden the bearing metal by the use of tin is to be avoided. Small test pieces used on the Friction Machine show up local high spots after running for a short period. The lead-base alloy bearing/

FIGURE 16.

FAILING LUBRICATION FOLLOWED BY SEIZURE.



bearing showed a considerable amount of wear, but was not injurious to the journal; the lead mixed with the oil and formed a fine lapping material which provided a high polish to the steel surface.

Importance of Surface Quality

The running fit tolerance must bear some relation to the quality of the surface. Where the methods of machining produce rough surfaces, it is of little use to specify small tolerances. The higher parts of the surface lie within the tolerance, and the remaining part of the surface is outside the limits. Lathe finish, even with spring scraper, and grinding, leave surfaces which have a short life before the bearing becomes unserviceable through wear.

Wear in itself is a fairly simple condition to determine but has several directions. Wear in a combination of directions may cause an unexpected smash. Where shafting is concerned the chief danger lies at the bearing, since the bearing metal soon becomes reduced in dimensions beyond the efficient performance. At the bearing there is a combination of wear since the bearing becomes big while the shaft journal decreases in diameter; the sum of this wear soon passes the proper clearance. Owing to large eccentricity, or shift, either bearing element becomes eccentric, scored, and overheated, since the axes of the shaft and bearing have not remained parallel; disalignment of the bearings may cause undue stress by withdrawal of the support. It was such a combination of wear that caused the deflection of the bearing machine shaft. The ball race owing to faulty lubrication had excessive wear and showed signs of pitting.

A new lining was cast into the bearing bushes, approximately of the same analysis as that tested in the Deeley machine, p.4, namely— P_b, 1.5; Sn, 78; Cu, 7.5; Sb, 13. This high tin alloy bearing after being cast had its surface brushed with a wire brush to reduce the high spots. No work such as filing, was done/

done on the material. This brushing of the material reduced by 40 per cent. the value obtained when running the journal on the white metal as cast round the shaft. Only short tests were made under these conditions as the heat generated was excessive.

The bearing was now hand-scraped and the μ-value obtained; this was quite high but the heat generated was not so great, the bearing running much cooler. In this case the projected area of the bearing surface is more nearly that of a machined surface.

<u>Machined Surface.</u> - After stripping the bearing it was (i) bored in the lathe and spring-tool scraped, then tested. It was not possible to do much in the way of determining the best clearance to suit the class of finish, but the clearance was gradually reduced by the removal of thin paper adjusting strips until the usual running conditions were reached.
(ii) After stripping the bearing bush was bored and scraped, then finished off with an expanding roller mandril.

No attempt was made to apply hone finish to the white metal surface, as the hardness number is only about one quarter of that of the bronze bush. The data given in the table, showing the effect of finish on the reduction of friction torque, or μ-value, is after a period of running-in.

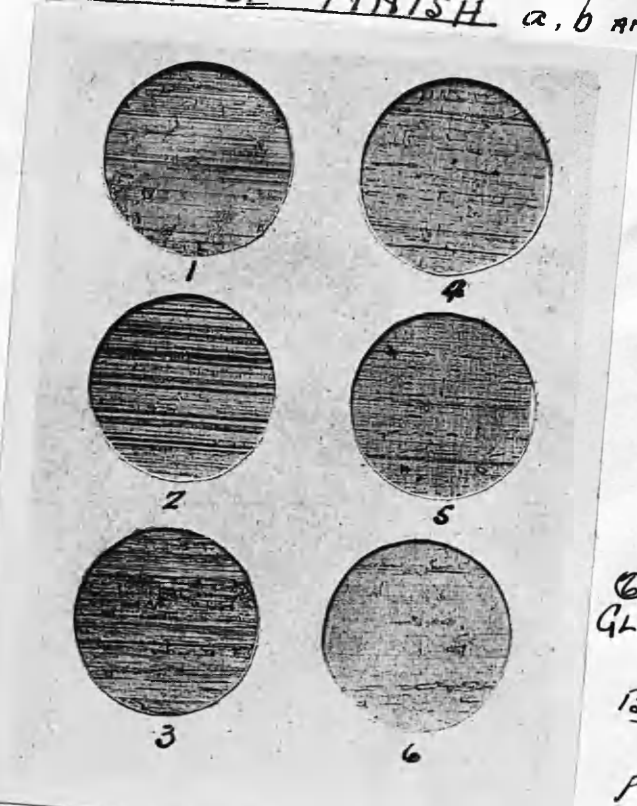
(a)	Class of Finish	Coeff. of Friction Starting	Running	Temp. °F. Max.	Room	Remarks
1	Cast round shaft	0.181	0.0033	180		
2	Wire Brushed	0.153	0.0022	155	55 to 65	Time of tests (each) 5 hours.
3	Hand Scraped	0.150	0.0022	155		
4	Machine Finish	0.150	0.0018	145		
5	Expanding Mandril or Splice Finish	0.090	0.0015	135		

(b) Test at 300 r.p.m. on White-Metal Bush

1	Rough Casting	77°F.	Temp.	diff.	μ-value	0.003
2	Wire Brushed. or Hand Scraped.	76°F.	"	"	"	0.002
3	Machined and Expanded to size	73°F.	"	"	"	0.018

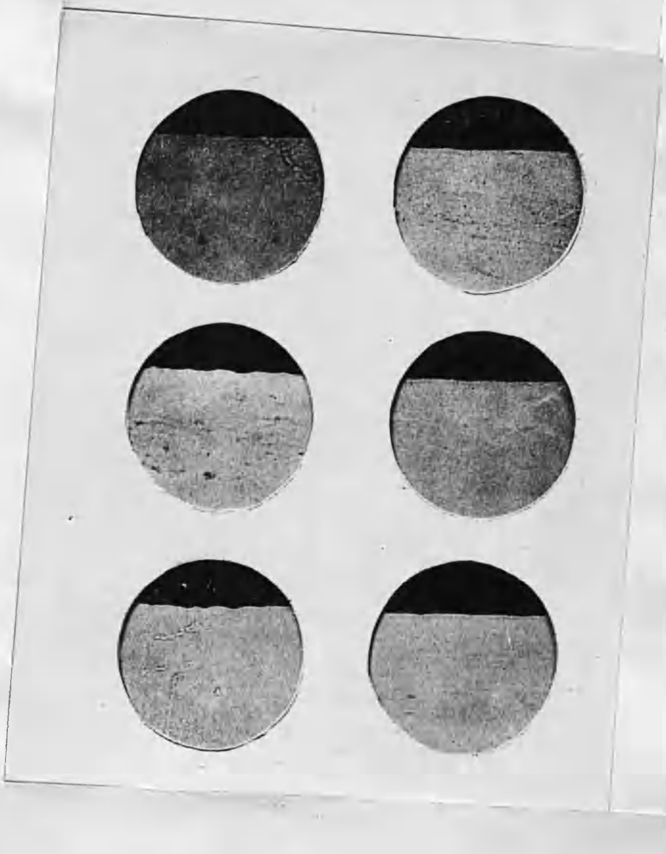
(c)/

"SURFACE FINISH" a, b AND C, p. 54.



⑥ GLASS FINISH, WITH BURNISHER

FIGURE 20a.



TRANSVERSE SECTION

THE LUBRICANT HAS IMPREGNATED THE MATERIAL.

FIGURE 20 b.

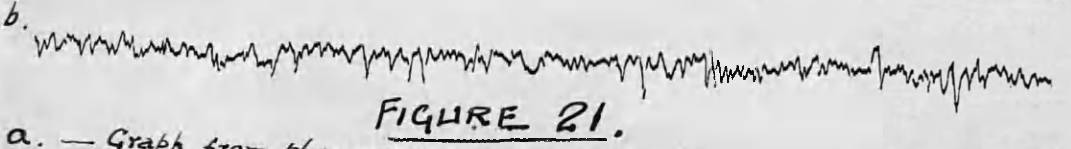


FIGURE 21.

a. — Graph from three Roller Burnished Surface.
b. — " " Spring Scraper Surface.

(c) Effect of reduction in bearing afea, Finish (3)

Area of Surface sq.in.	Pressure lb/in^2	Temperature °F.		Coefficient of Friction
		Rise	Difference	
5.5	215	65	43	0.0039
5				
4.5	272	65	42	0.0032
4				
3.5	360	66.5	42.5	0.0028
3				
2.5	590	67.5	43	0.0022
2				
1.8	600	68.5	43	0.0020

In Fig. 16, p. 52, the effect of finish is shown by the temperature rise in a given time, and the change in μ-value during the test is also plotted against time. As a comparison, μ is plotted on a base of temperature, T, in Fig. 17, p. 57. The average values for a short test, 1½ hours, are shown on Fig. 18, p. 58. The speed in this test was varied, that is, speeded up and reduced to zero by means of the variable gear. An attempt was made to note the change in temperature due to change variations insspeed. In all these curves the advantage of the roller finished material is clearly demonstrated.

The author has had the opportunity of experimenting with roller bars, having rollers ½ in. in dia. and ½ in. long set parallel to the bar and others with the axis of the rollers at an angle to the bar.. It was found possible to do with a much rougher cut surface from the lathe tool when followed by the expanding roller mandril with the inclined rollers, which turned more freely on their own axis.

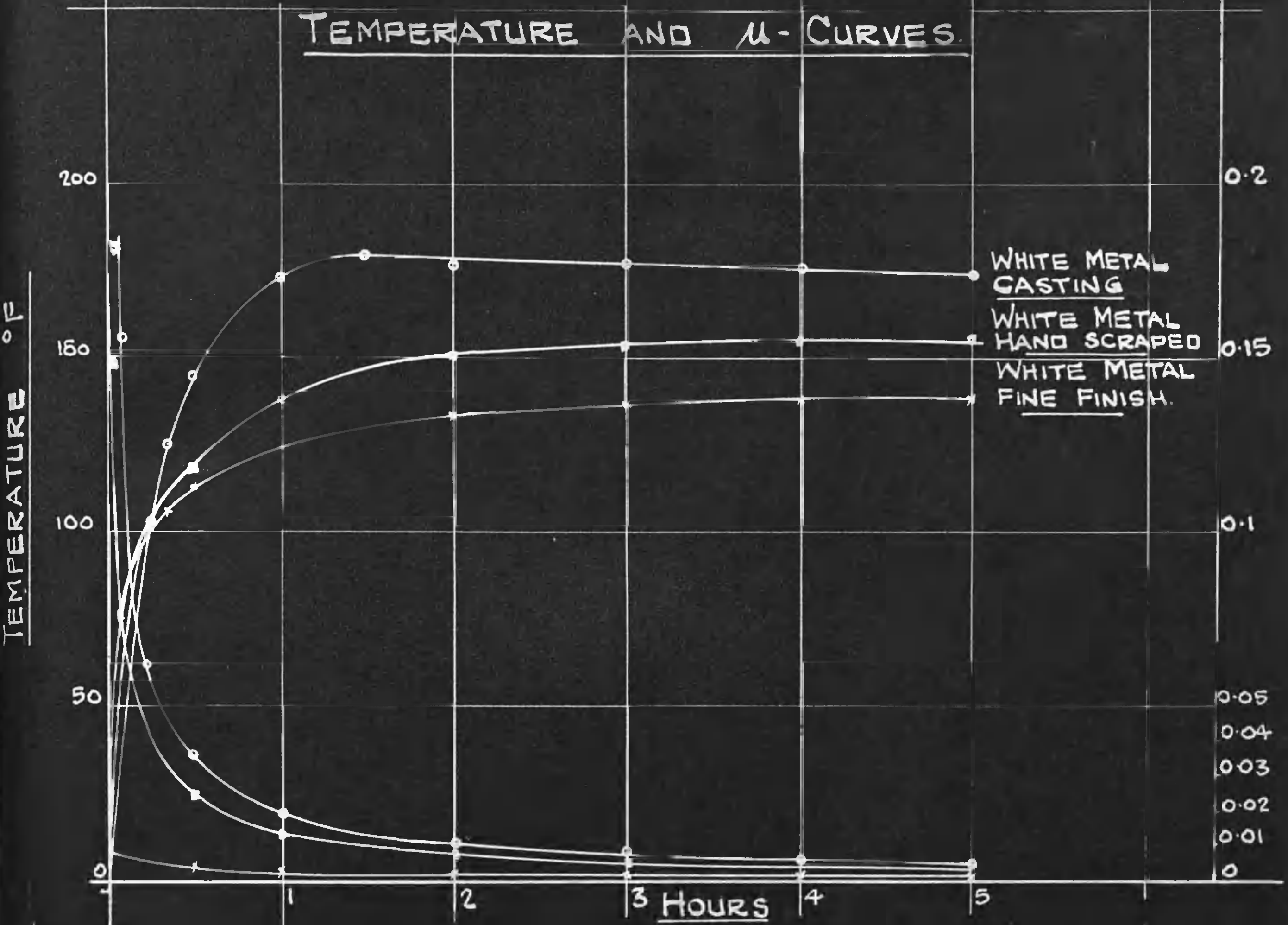
On repeating the tests, shown in Fig. 18, it is found that even the running-in effect of the earlier tests have not improved the surfaces enough to make them all be classed as surfaces of equal degrees of smoothness. The expanded material of the roller finished bearing remains superior to all other machined surfaces, Fig. 19, p. 57

A/

EFFECT OF MACHINE FINISH

TEST No 1.

TEMPERATURE AND μ - CURVES.



DURATION OF TEST 1

FIGURE 17A

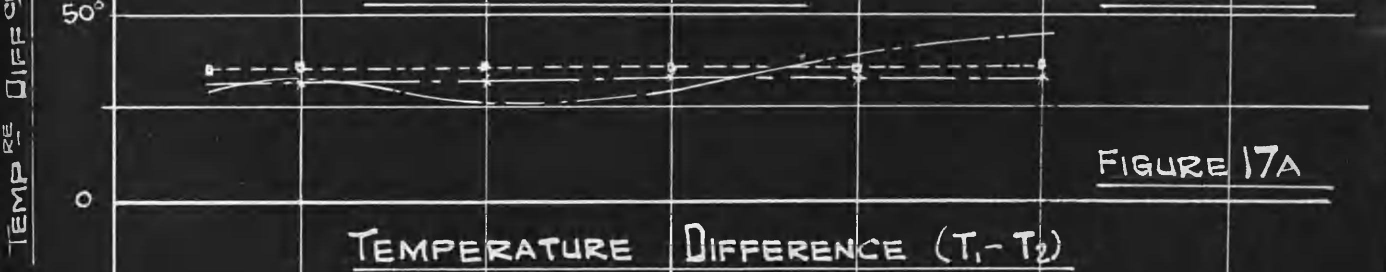


FIGURE 17A

TEMPERATURE DIFFERENCE $(T_1 - T_2)$

T_1 = TEMP^{RE} DURING 1ST RUN } SAME
T_2 = " " 2ND " } CONDITIONS.

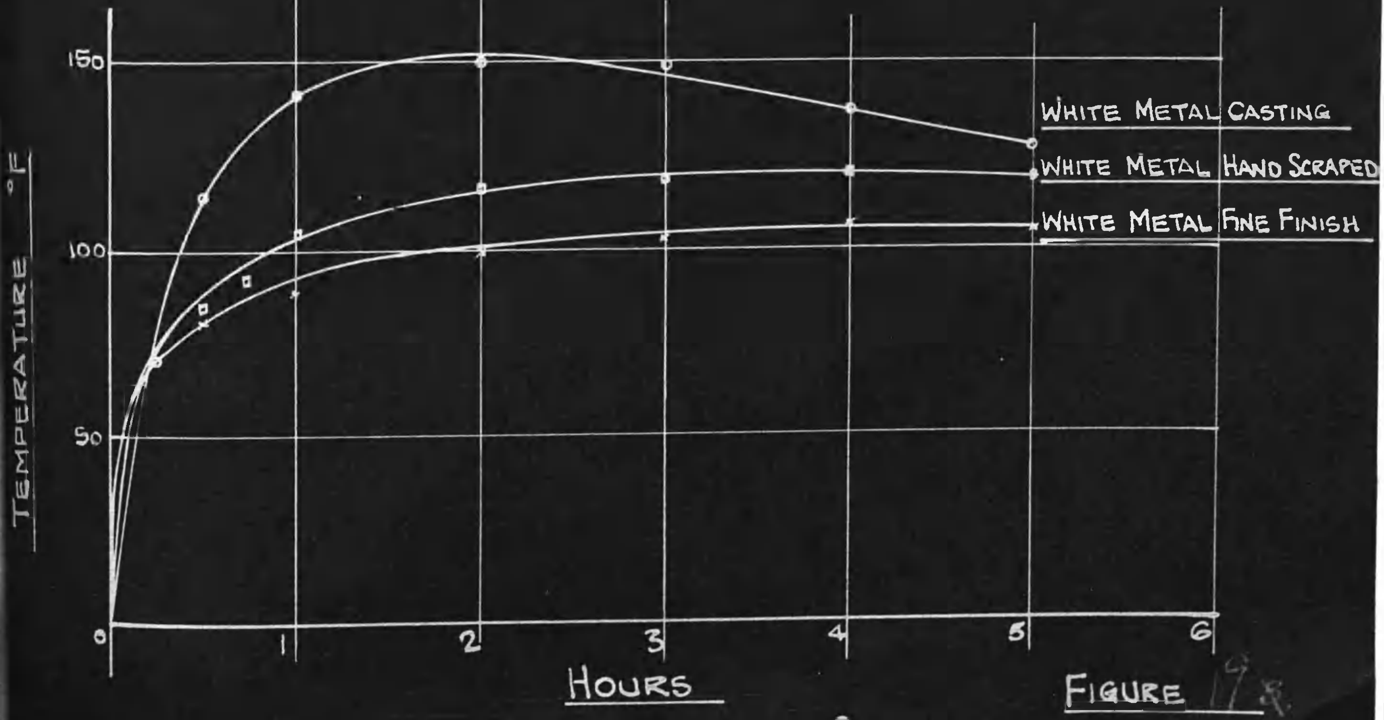
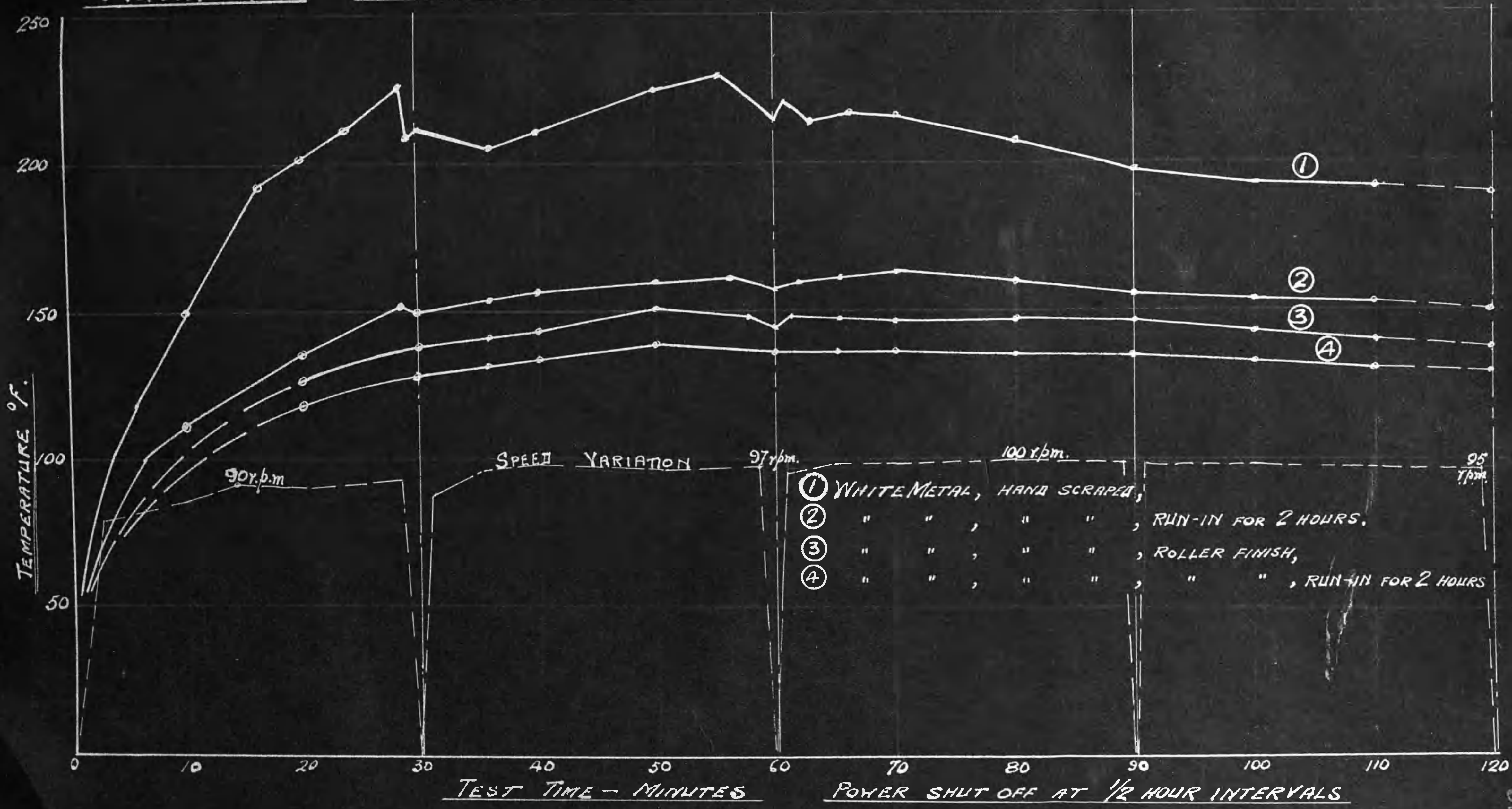


FIGURE 17B

DURATION OF TEST 2
TEMPERATURE CURVES

FIGURE 18. EFFECT OF MACHINE FINISH ON RUN-IN TIME



A comparison of the surface finish was made by taking micro-photographs and these are shown with magnification 100 times. <u>Fig. 20 (a) and (b)</u>/ p.55. (a) is a lathe tool finish and (b) is the same lathe tool finish followed by an expanding roller finish. (c) is a spring-tool scraped finish and (d) is the same as (c) followed by roller finish. (e) is grinding machine finish, and (f) is the same as (c) followed by roller finish. In each case the roller finished material shows a distinct advantage in fineness of structure, which would reduce the running-in time. The change in structure of the white metal due to different methods of machining is seen in cross-section micro-photographs. This change is affected by the amount left for expanding. An excessive expanding allowance was found to cause a brittleness in the bearing metal which, under expansion due to rise in temperature, might cause failure of the material.

CONCLUSIONS.

The investigation has clearly brought out the beneficial effects of certain lubricants on the starting up torques and under running conditions. Heavy oils while lessening the risk of seizure when starting up a machine, or line of shafting, causes a big loss of power due to the increase of friction work or viscosity effect. The experimental results show how this frictional work decreases as the temperature of the bearing oil increases.

Many of the tests were carried out while considering the theory of lubrication, but the main investigation was concerned with the design of the bearing block and its capability of assisting radiation and conduction of heat from the bearing surface. An attempt being made to show when water or fan cooling would be necessary.

Where single bearings are lubricated by oil having graphite or sulphur in suspension, the use of these as lubricants/

lubricants and coolants are found to improve mineral oil. That certain oils will carry as much as 25 per cent. of flowers of sulphur without becoming difficult to feed to a bearing was tested out on the bearing machine; pine oil was found to carry sulphur more easily than Bayonne oil.

A comparison of many lubricating blended oils shows them to possess similar features and to give very similar results under test. Many of the poorer classes of lubricating oils give quite good results on the short period tests.

That machine finish or smoothness of surface will reduce loss due to friction and decrease chances of seizure, or failure of lubrication, is shown. Also the wear per unit length of time running is shown to be decreased by roller surface finish which reduces the coefficient of friction at starting and running. The time spent on machine shop finish by the additional operation of rolling a bearing to size, reduces time allocated to the usual running-in process on a test bed. The process of rolling decreases the necessity of large tolerance clearances, and reduces ridge wear to a minimum.

The author has under consideration the design of a combined micrometer and optical ridge testing machine for testing smoothness, before removing a bearing from the lathe or boring machine. Such a machine could be used to test the effect of different lubricants used during the rolling process, and would have saved time in investigation of different smoothness of finish. This investigation has shown that standardization in finish is much required in machining operations, but an effective test of smoothness of surface finish is necessary before this can be done. Surface-measuring instruments can now detect irregularities of 0.000001 inches and this is what can be expected on lapped and mirror finished material. An out of centre gauge/

gauge, fitted with a diamond pointed tracer, having a magnification mirror for vertical movements (2500 magnifications) and for horizontal movements (20 magnifications), could be used as a high and low limit gauge of surface finish. The limits could be set so as to reduce the running in time by half, and the danger of seizure on starting up would also be reduced by standard finish of bearings, i.e. a tolerance of finish is necessary. Fig. 21, p.55, shows the diagram of smoothness obtained from such an apparatus. When a bearing is run in there is a decided wear, and the metal removed can cause pitting when the machine bearing is slowing down before stopping; a slight oiliness will promote grinding by holding the particles.

Suggested finishes for materials used in tests are .-

Steel Journal: Rough and Finish turn, scrape or grind, burnish (with three roller burnishes).

White Metal: Rough and Finish Bore, Reaming and Rolling.

Lead Bronze: Rough and Finish Bore, Reaming and Honing.

Tin-base Bronze: Rough and Finish Bore, Reaming and Rough Honing, Finish Honing and Mirror Honing.

The lubricant to be used in machining - a free-from-acid rape oil.

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