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SMALL INERTIA-TYPE MACHINE FOR TESTING BRAKE LINING

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ABSTRACT

This paper describes a machine of the inertia type which has been developed and built for testing small samples of brake lining. The operation of the machine consists in bringing a flywheel up to a predetermined speed at equal intervals of time and stopping it by means of a brake mechanism which employs the lining under test. The machine is fully automatic in operation and was designed to permit the measurement of the coefficient of friction and wear of brake lining under a variety of test conditions simulating those encountered in service. The interval between stops, the speed from which stops are made, the pressure of the lining on the drum, and the moment of inertia of the flywheel may all be varied within limits. The machine has been used to determine the coefficient of friction and the rate of wear of a large number of brake linings. The temperature of the brake drum and lining probably affects the coefficient of friction more than any other factor. Tests have been made under three sets of conditions: (1) those in which the brake drum reaches a temperature of about 300° F; (2) those in which it reaches a temperature of about 500° F; and (3) those designed to show the effect of moisture.

Some data are presented to show the effect of the smoothness of the surface of the brake drum on the rate of wear of brake lining.

CONTENTS

I. INTRODUCTION

This paper describes a machine developed at the National Bureau of Standards for testing brake lining. The intention was to construct a machine which would (1) be suitable for measuring the coefficient of friction and wear characteristics of brake lining under different conditions, (2) operate at low cost, (3) simulate the action of brakes in service, (4) give results comparable with service in a minimum amount of time with a minimum amount of attention, and (5) be adaptable for use in purchase specifications.

The design of this machine is based upon inertia-testing machines such as are used by manufacturers of brake lining.[1] It is smaller in size, however, and is designed to employ relatively small samples of lining instead of complete brakes, as is common with the large machines. Tests have been made on a variety of types and kinds of lining and tentative test conditions have been established for the routine testing of linings.[2]

II. DESCRIPTION AND OPERATION OF MACHINE

The machine is of the inertia type, and the operation consists essentially in bringing a flywheel up to a predetermined speed at equal intervals of time and stopping it by means of one of the two brake mechanisms which operate in the brake drums at the ends of the flywheel shaft.

The machine and its control apparatus are shown in figure 1, and a supplementary diagrammatic sketch of the hydraulic-brake system is shown in figure 2.

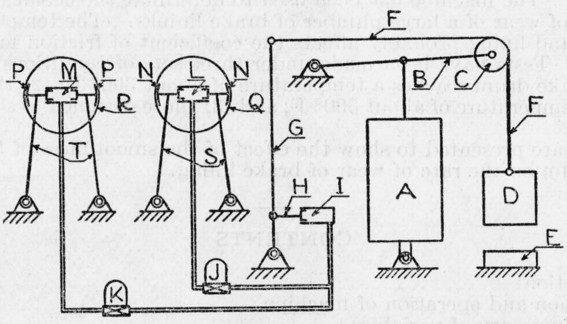

FIGURE 2.—*Diagrammatic sketch of hydraulic-brake system.*

Thrustor *A* raises thrustor lever arm *B* and sprocket *C* to lift weight *D* off its base *E*. Weight *D* then pulls on chain *F*, which pulls on lever arm *G* pushing piston *H* into master cylinder *I*. This forces brake fluid through solenoid valve *J* or *K* to wheel cylinder *L* or *M*, thus forcing lining *N* or *P* against drum *Q* or *R*. Lining *N* is carried on brake arms *S* and lining *P* on brake arms *T*.

A complete and separate brake unit is mounted on a tail stock at each end of the machine (fig. 1), enabling two samples of lining to be tested over the same period of time. These units are usually operated to make alternate brake applications. Either unit, however, may be operated to make consecutive applications when only one sample is being tested. The actual braking is accomplished by hydraulically forcing two small pieces of lining against a standard brake drum.

This machine was designed so that at comparable speeds the energy absorbed per square inch of lining in stopping the flywheel could be made equal to the energy absorbed in stopping an automobile. A comparison between selected test conditions on one hand and service conditions of a passenger car on the other is given in table 1. Al-

[1] Bockius and Hunt, *Brake drum and lining development*, SAE Journal, **37**, 250 (July 1935).
E. W. Sisman, *Brake performance*, Inst. of Automobile Engrs. Journal, **5**, No. 3, 20 (Dec. 1936).
[2] Procurement Division Specification No. 369 for brake lining.

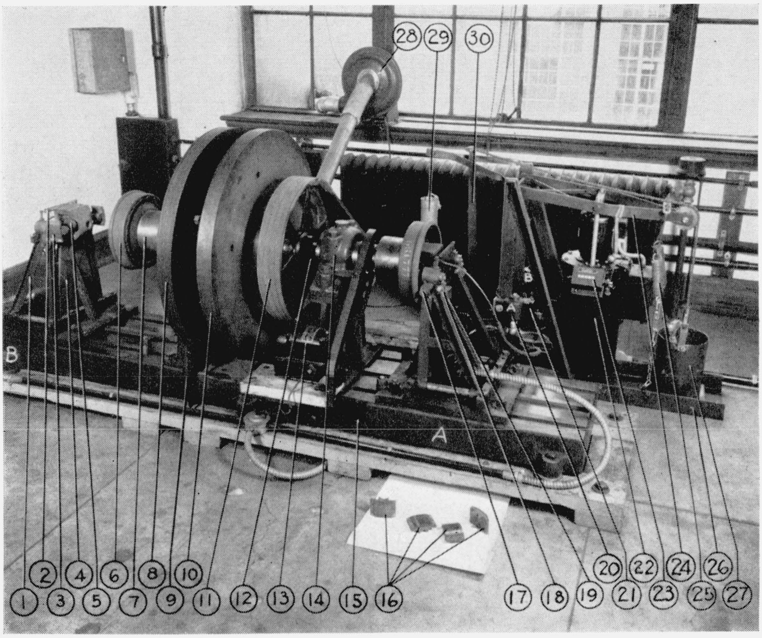

FIGURE 1.—*Inertia machine for testing brake lining.*

1. Brake arm.
2. Brake-retracting spring.
3. Brake shoe.
4. Tail stock.
5. Wheel cylinder.
6. Brake drum.
7. Brake-drum hub.
8. Extra flywheel weights.
9. Flywheel weight frame.
10. Flywheel.
11. V-belt drive from main motor.
12. Main shaft.
13. Tachometer generator.
14. Main bearing.
15. Base.
16. Extra brake shoes.
17. Stop bolt for brake arm.
18. Shoe adjustment.
19. Automatic cut-off switch.
20. Solenoid valves *A* and *B* for opening pressure line to wheel cylinder.
21. Master cylinder.
22. Thrustor.
23. Counter.
24. Thrustor lever-arm.
25. Adjustment spring for thrustor lever-arm.
26. Thrustor weight frame.
27. Weight bucket.
28. Auxiliary fan for cooling main motor.
29. Weight to release brake lever-arm.
30. Brake lever-arm.

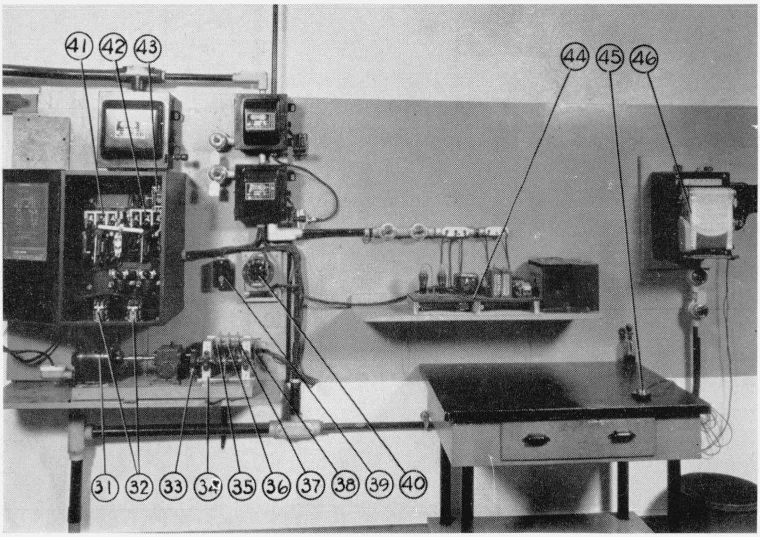

FIGURE 1 (Continued).—*Controls for automatic operation of machine for testing brake lining.*

31. Synchronous motor for driving timing unit.
32. Auxiliary relays.
33. Gear train.
34. Timing unit.
35. Cam-operated mercury switch for starting main motor.
36. Cam-operated mercury switch for opening solenoid valve *A*.
37. Cam-operated mercury switch for opening solenoid valve *B*.
38. Jack switch for selecting solenoid valve operation.
39. Start-stop switch for automatic cut-off relay.
40. Resistance for controlling speed from which stops are made.
41. Automatic switch for thrustor.
42. Automatic switch for main motor.
43. Automatic cut-off relay.
44. Vacuum-tube control unit.
45. Push button for driving chart at high speed.
46. Speed-time recorder.

though most tests are made under the conditions indicated in this table, the machine incorporates features which enable tests approximating many other service conditions to be made.

TABLE 1.—*Comparison of test conditions with service conditions*

Condition	Conditions of test on machine	Service conditions with automobile
Weight of car		3,600 lb.
Moment of inertia of flywheel	11 slug ft ²	
Total *effective* area of lining used in making a stop	8 in.²	120 in.² ª.
Effective area of lining in each brake	8 in.²	30 in.².
Diameter of brake drum	10 in	10 in.
Effective diameter of tire		29. 3 in.
Speed	600 rpm	52. 2 mph.
Brake-drum speed	600 rpm	600 rpm.
Average effective pressure of lining on brake drum	75 lb/in.²	75 lb/in.² ª.
Total kinetic energy at given speed	21,700 ft-lb	325,500 ft-lb.
Kinetic energy absorbed per square inch of lining during time of brake application	2,712.5 ft-lb	2,712.5 ft-lb.
Assumed coefficient of friction of lining	0.40	0.40.
Time of stopping	6.9 sec	6.9 sec.
Deceleration	9.1 radians/sec ²	11.1 ft/sec ².
Deceleration of brake drum	9.1 radians/sec ²	9.1 radians/sec ².
Temperature of brake drum	Frequency of stops can be regulated to produce any estimated service temperature.	Depends on service conditions.

ª Estimated.

1. SUMMARY OF VARIABLES

The principal adjustments which may be made are as follows:

1. The moment of inertia of the flywheel may be varied from 11 slug ft² to approximately 20 slug ft² by adding weights (8)[3] to the flywheel.

2. The force on the brake shoe may be varied from 0 to approximately 600 lb.

3. The speed from which stops are made may be set at any point up to 600 rpm.

4. The intervals between brake applications may be made ½, ¾, 1, 1½, or 2 min.

2. FLYWHEEL AND DRIVING MOTOR

The flywheel (10) is essentially a steel disk 30 in. in diameter and 1.9 in. thick, mounted, together with the pulley for a V-belt drive (11), on a 2.5-in. shaft between two self-alining roller bearings (14). It is driven by means of 4 V-belts from a 220-volt, a-c, 3-phase, induction motor, rated at 10 hp and 1,750 rpm, capable of being started by a line switch (42). An auxiliary cooling unit (28) is provided for the driving motor, to prevent overheating under the frequent starting and stopping of the machine. This unit consists of a 10-in. blower, driven by a ⅛-hp motor, so connected that the artificial ventilation aids the natural ventilation.

The moment of inertia of the rotating system can be increased from its minimum value of 11 slug ft² in steps of approximately 1.25 slug ft² to a maximum of about 20 slug ft² by adding the requisite number of disks (8). These disks are ¼ in. thick and 30 in. in diameter and are stored on a frame attached to the base of the machine. They may be slipped over the flywheel hub and bolted to the flywheel.

[3] Numbers in parentheses correspond to the numbers shown in figure 1.

3. BRAKE UNITS

The two brake units are identical. The drums (6) of centrifused-cast iron, 10 in. in diameter, and designed for lining 2 in. wide, are bolted to hubs (7) on the ends of the flywheel shaft. Each brake is mounted on a tail stock which may be pulled away from the drums to give access to the brake shoes. Each brake employs two shoes (3), which are rigidly bolted to the brake arms (1) in such a way that they will fit into the brake drum and be diametrically opposite. Each brake arm consists of an A-shaped frame, which is pivoted on a shaft at the base of the tail stock. This shaft is parallel to the main shaft. The shaft for each arm acts as a bearing and anchor about which the arm may turn, and is centered directly below the midcontact point of the lining with the drum. If small samples of lining are employed, this type of mounting reduces the effect of self-energization [4] to a negligible amount (see section V) and makes a measurement of coefficient of friction possible.

One piece of lining is fastened to each shoe. Each piece (16) ordinarily has 4 in.^{2} of contact surface and is cut with a small additional lip on each end to permit it to be clamped onto the shoe.

A retracting spring (2) is connected across the top of the two brake arms on each unit to insure brake release. Each spring exerts a pull of about 50 lb.

By providing appropriate hubs, other brake drums may be used. If the diameter of the brake drum is changed, it is necessary to use shoes with the proper radius of curvature, to shift the bearings of the brake arms to lie below the midcontact points of the lining with the drum, and to install longer pressure arms for the wheel cylinder.

4. BRAKE-ACTUATING SYSTEM

The brakes are actuated by a hydraulic system, shown schematically in figure 2. This system consists of a standard automobile master cylinder (21), two standard straight-bore wheel cylinders (5), two high-pressure solenoid valves, and the necessary connecting tubing. The master cylinder is actuated by a dead weight (27) connected to the brake lever-arm (30) by means of a chain and sprocket. This weight rests on a base until brake application is desired, at which time an oil-pump device, termed a thrustor (22), pushes up the thrustor lever-arm (24) and the sprocket sufficiently to lift the weight off its base. The force due to the weight is then transmitted through the chain to the lever-arm (30) and hence to the piston of the master cylinder. Thus, the brake is applied and the flywheel is brought to rest. The force exerted on the drum by the lining is dependent upon this weight and is limited in amount by the load the thrustor will lift. The maximum value is approximately 600 lb per shoe, or 150 lb/in.^{2}, based on 4 in.^{2} of lining. The force generally used is 300 lb per shoe, or 75 lb/in.^{2}

Each wheel cylinder is connected through a separate solenoid valve to the one master cylinder. These solenoid valves are closed

[4] If the frictional force on the brake shoe exerts a moment about the bearing of the brake arm, the braking force is increased or decreased depending on the direction of the moment. This effect is known as "self-energization."

unless energized. Thus, either brake may be applied as desired by energizing the proper valve.[5]

In order to cut down the shock of brake application and to increase the uniformity of force on the brake, two adjustable springs (25) are connected between the thrustor lever-arm (24) and the thrustor weight frame (26). These springs can be adjusted so that at the maximum lift of the thrustor lever-arm the chain will be at right angles to the brake lever-arm (30).

5. CONTROL APPARATUS

The control apparatus consists of a timing unit (34), a tachometer generator (13), and a vacuum-tube control unit (44), together with the necessary relays and switches.

The operation of the machine is fully automatic once the control circuits are closed, except that to obtain records of deceleration for calculating the coefficient of friction the high-speed chart drive on the recorder must be manually operated. As will be seen in section VI, these records usually need be taken only during the first part of a series of stops while the brake lining and drum are heating up. When records for determining the coefficient of friction are not desired, an unlimited number of stops can be made at equal intervals without attention.

The cycle of operations is as follows:

1. The timing unit (34) starts the main motor, which drives the flywheel (10) by means of the V-belt drive (11):
2. While the motor is bringing the machine up to speed, the timing unit opens the proper valve (20) to cause application of the desired brake.
3. When the machine gets up to speed, the vacuum-tube control unit (44), responding to the increased voltage from the generator (13) cuts off the driving motor and starts the thrustor (22); applying one brake.
4. When the machine stops, the control unit, responding to the reduction of the voltage from the generator to zero, interrupts the thrustor circuit, releasing the brake.
5. About 20 sec after the brake has been applied, a period sufficiently long to permit any lining under a force of 300 lb to stop the flywheel from 600 rpm, the timing unit allows the solenoid valve to close.
6. The cycle is then repeated as many times as desired.

The timing unit consists essentially of three mercury switches operated by three cams. One switch (35) starts the driving motor and the other two (36, 37) open and close the respective solenoid valves (20) in the hydraulic circuit at the proper time. The cams are all mounted on the same shaft and are driven by a synchronous motor through a reduction gear box and an adjustable gear train (33). Brake applications may be made at different time cycles by changing the gears on this train. For normal testing, the machine is started once every ¾ min and each brake is applied alternately once every 1½ min.

[5] The present valves have a maximum recommended working pressure of 300 lb/in.^{2} Valves designed for greater working pressures would be desirable, but would probably have to be specially constructed, since they do not seem to be stock items.

The tachometer generator is driven [6] from the flywheel shaft by a 1 to 1 ratio. Internal gears in the generator drive the armature shaft at twice the flywheel speed. A potential of approximately 18 volts is generated when the flywheel is running at 600 rpm.

The vacuum-tube control unit consists of a cutoff circuit and a brake-release circuit. Each contains a number 57 vacuum tube and a sensitive relay. Each relay operates when the plate current in the tube of its particular circuit reaches about 3 milliamperes. The circuits are so designed that the voltage from the tachometer generator regulates the plate current in both tubes. In the cutoff circuit the plate current increases as the tachometer voltage increases, and in the brake-release circuit the plate current decreases as the tachometer voltage increases. An adjustable resistance (40) regulates the plate current in the cutoff circuit and hence the speed from which stops are made.

The relays in the control circuit operate two auxiliary relays (32) which control the main switches (41, 42) for the thrustor and the driving motor. The cutoff relay operates the auxiliary relay which cuts off the main motor and energizes the thrustor. The brake-release relay operates the auxiliary relay which cuts off the thrustor. The starting of the main motor is accomplished by the timing unit, as previously explained.

In addition to the regular controls for operating the machine, there are a number of safety controls designed to prevent damage in case of any trouble. A stop bolt (17) is placed in each brake arm to limit the arm movement and thus prevent the piston of the wheel cylinder from pushing too far out, in case a condition should arise in which the brake drum does not limit the movement of the shoe. An automatic cutoff switch (19) is connected to each brake arm. Its function is to open the automatic cutoff relay (43) if a lining comes off or if the lining wear becomes great enough to permit the brake arms to be forced out beyond a certain predetermined limit. The opening of this relay stops the timing unit and consequently the functioning of the machine. A thermal guard in the driving motor also opens this relay if the motor gets too hot. A mechanical interlock between the driving motor and thrustor automatic switches prevents the motor and the thrustor from operating at the same time.

6. RECORDING APPARATUS

The recording apparatus is a speed-time recorder with a record chart that can be driven at two different speeds. The chart is normally driven at ¾ in./hr by a synchronous clock; but when a record of the deceleration is desired, the chart may be driven at ¾ in/sec. by a synchronous motor which is cut in by a manually operated switch. The recording pen is controlled by the voltage from the tachometer generator.

[6] The chain drive shown in fig. 1 has been replaced by gears since the photograph was made.

III. METHODS OF CALIBRATING MACHINE

1. MOMENT OF INERTIA OF FLYWHEEL

The calibration of the flywheel for moment of inertia was made by application of data measured with a prony brake to the following formulas:

Let T_0=total retarding torque in pound feet

T_f=retarding torque in pound feet due to friction and windage.

T_1, T_2, T_3, etc.=applied prony brake torque in pound feet.

I=moment of inertia of flywheel in slug feet squared.

α_1, α_2, α_3, etc.=deceleration of flywheel in radians per second per second when torques T_1, T_2, T_3, etc., are applied.

From the general formula for torque and angular motion the total retarding torque is

$$T_0=T_1+T_f=I\alpha_1, \tag{1}$$

or

$$T_1=I\alpha_1-T_f. \tag{2}$$

Similarly,

$$T_2=I\alpha_2-T_f, \tag{3}$$

and

$$T_3=I\alpha_3-T_f. \tag{4}$$

Therefore, assuming T_f constant, any two values of T may be combined to eliminate T_f, for instance

$$T_1-T_2=I(\alpha_1-\alpha_2),$$

or

$$I=(T_1-T_2)/(\alpha_1-\alpha_2), \tag{5}$$

in which case formula 5 gives a true value for I. Actually, T_f decreases slightly with a decrease in speed; but by selecting large values of T_1, T_2, etc., relative to T_f, the error introduced is negligible.

2. FORCE ON BRAKE SHOES

The force on the brake shoes (F_1) was measured by means of two calibrated test rings arranged with linkages in such a way that actual brake application forced the shoes against the links and deflected the rings. Lead weights were added to or taken from the weight bucket (27) and the corresponding deflections noted.

IV. MEASUREMENT OF COEFFICIENT OF FRICTION

The values of coefficient of friction sought for brake linings are not the constant values obtained under static conditions but rather the variable values obtained under different operating conditions. Consequently, consideration should be given to its value during each stop of a series of successive stops. The actual determination of these characteristics is made by applying data measured on the machine to the following formulas.

Let μ=the coefficient of friction at any instant during the braking period.

μ_a = the average coefficient of friction during the braking period.
F_1 = the force on the shoe in pounds per shoe, a constant during the braking period.
F_2 = the force due to friction in pounds per shoe at any instant.
$T_0 = T_1 + T_f$ = total retarding torque in pound feet.
T_f = the average friction and windage torque in pound feet.
I = the moment of inertia of the flywheel in slug feet squared.
α = the deceleration of the flywheel in radians per second per second at any instant.
α_a = the average deceleration in radians per second per second during the braking period.
$\Delta\omega$ = the change of speed of flywheel in revolutions per minute during the time Δt.
Δt = the time in seconds required for a change in speed of $\Delta\omega$.
t = the time in seconds for a complete stop.
D = the diameter of brake drum in feet.

By definition,

$$\mu = F_2/F_1 \tag{6}$$

From a consideration of mechanics and the machine construction the brake torque

$$T_1 = 2F_2D/2 = F_2D \tag{7}$$

Then from formula 1 in section III–1,

$$T_0 = T_1 + T_f = I\alpha \tag{8}$$

Therefore,

$$F_2D = I\alpha - T_f \text{ or } F_2 = (I\alpha - T_f)/D. \tag{9}$$

Then by substituting this value of F_2 in formula 6,

$$\mu = (I\alpha - T_f)/DF_1 = (I\alpha/DF_1) - (T_f/DF_1). \tag{10}$$

Formula 10 is a general formula from which calculations of coefficient of friction may be made. In all regular tests the values I, D, and F_1 are maintained constant and α is determined from the speed-time curve recorded by the speed recorder. T_f was found to be 5.5/lb-ft and was determined as subsequently described. If average values of coefficient of friction are sought, formula 10 may be rewritten as

$$\mu_a = (I\alpha_a/DF_1) - (T_f/DF_1). \tag{11}$$

But

$$\alpha_a = 2\pi\Delta\omega/60\Delta t, \tag{12}$$

which may be substituted in formula 11 to get

$$\mu_a = (2\pi I\Delta\omega/60DF_1\Delta t) - T_f/DF_1. \tag{13}$$

Formula 13 is the one actually used to calculate the coefficient of friction, and it may be simplified by putting in the constant values for I, D, F_1 and T_f. Most tests at the present time are made with I=11 slug ft^2, D=5/6 ft, F_1=300 lb, and T_f=5.5 lb-ft. It is important to note that here, as in the calibration of the flywheel for moment of inertia, T_f may be considered constant when, and only when, the retarding torque due to brake application is large in comparison with that due to friction and windage. If this were not the case, the variations in T_f would have to be taken into account in the following formulas. Thus, formula 13 becomes

$$\mu_a = \frac{2\times\pi\times 11}{60\times(5/6)\times 300}\times\frac{\Delta\omega}{\Delta t} - \frac{5.5}{(5/6)\times 300} = 0.00461\frac{\Delta\omega}{\Delta t} - 0.022. \quad (14)$$

If stops are made from 600 rpm, the average coefficient of friction for a complete stop becomes

$$\mu_a = \frac{2.765}{t} - 0.022. \quad (15)$$

If made from 450 rpm, it becomes

$$\mu_a = \frac{2.073}{t} - 0.022. \quad (16)$$

The relation between the coefficient of friction, μ_a, as calculated from formulas 15 and 16, and the time, t, for a complete stop is shown

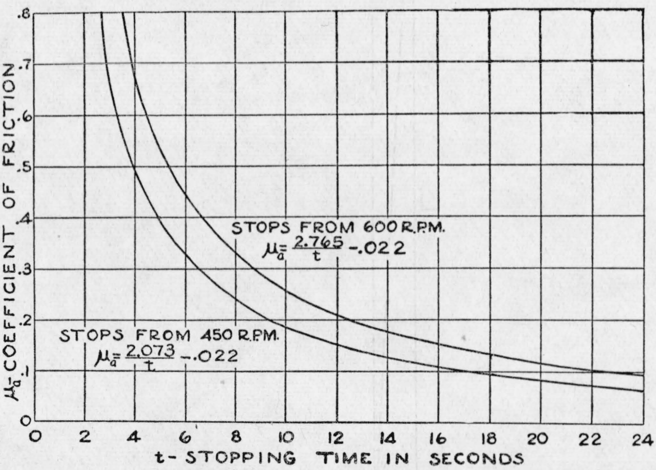

FIGURE 3.—*Calculated curves for determining coefficient of friction from the stopping time for stops from 600 and 450 rpm.*

in figure 3. The coefficient of friction corresponding with the time for stopping read from the speed-time curve may then be taken directly from figure 3.

V. EFFECT OF SELF-ENERGIZATION

The effect of self-energization on the machine may be determined from a consideration of the forces and moments in the brakes and of the schematic diagram shown in figure 4.

Let o=center of the brake drum.

b=center of surface of contact between lining and drum.

a=anchor point of brake shoe.
r=radius of brake drum.
L=length ab, which is vertical and is perpendicular to ob.
w=width of lining.
$2c$=length of lining.
P=applied pressure in pounds per square inch assumed to be normal between the lining and the drum.

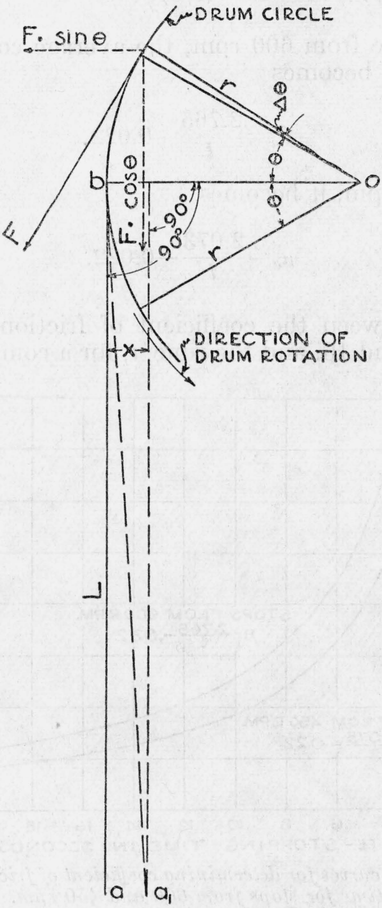

FIGURE 4.—*Schematic diagram of the brake forces about the anchor point of the shoe.*

o, center of drum; **b**, center of contact surface of lining with drum; **a**, anchor point of brake shoe; **r**, radius of drum; **L**, length **ab**; **F**, tangential or frictional force of the drum on the lining per unit length.

μ=coefficient of friction between the lining and drum.
$F=wP\mu$=tangential or frictional force of the drum on the lining per unit length of lining.

For any increment length of lining $r\Delta\theta$, the frictional force is $Fr\Delta\theta$. The moment of the force about the point a (fig. 4) can be obtained by the use of the horizontal and vertical components of the force, respectively, $Fr \sin \theta \, \Delta\theta$ and $Fr \cos \theta \, \Delta\theta$. The horizontal component gives a moment

$$Fr \sin \theta (L + r \sin \theta) \Delta\theta \tag{17}$$

which is counterclockwise for positive values of θ and clockwise for negative values of θ. The angle θ is taken as positive when it is measured upward from the line *ob*. The vertical component

$$Fr \cos \theta (r - r \cos \theta) \Delta\theta \tag{18}$$

always gives a clockwise moment. Combining these moments and taking counterclockwise moments as positive, one obtains

$$Fr(L \sin \theta - r \cos \theta + r) \Delta\theta. \tag{19}$$

The total moment about the point *a*, for the entire block of lining of length $2c$, is given by

$$\int_{\theta=-c/r}^{\theta=+c/r} Fr(L \sin \theta - r \cos \theta + r) d\theta. \tag{20}$$

Integrating formula 20, one obtains

$$2Fr[c - r \sin (c/r)]. \tag{21}$$

But one may write

$$\sin (c/r) = c/r - 1/6(c/r)^3 + 1/120(c/r)^5 - + \cdot \cdot \cdot \tag{22}$$

and since c/r is considerably less than unity, terms containing c/r to powers of 5 or higher are negligible. Therefore, the total moments due to self-energization may be written as follows:

$$2Fr[c - c + 1/6(c^3/r^2)],$$

or

$$1/3(Fc^3/r). \tag{23}$$

Substituting the actual numerical values from the following,

$F = wP\mu = 2 \times 75 \times 0.4 = 60$
$c = 1$ in.
$r = 5$ in.
$w = 2$ in.
$P = 75$ lb/in.^{2}
$\mu = 0.40$
$L = 13.8$ in.

the moment becomes

$$(1/3)60(1/5) = 4 \text{ lb-in.} \tag{24}$$

The increase of horizontal force of the lining against the drum which would produce an equivalent moment is

$$4/L = (4 \text{ lb-in.})/(13.8 \text{ in.}) = 0.29 \text{ lb.} \tag{25}$$

But this force is negligible in comparison with the 300-lb force on the shoe normally used, and, furthermore, it is well within the limit of accuracy in measuring the force on the shoe; hence one may conclude that the effect of self-energization is negligible.

The calculations were only for the shoe at which the motion of the drum surface is downward. However, both sides of the brake are symmetrical, and it may be shown that the effect of self-energization is the same in value for the shoe at which the motion of the drum surface is upward; but its sign is opposite, and it will be subtracted from the force on the shoe instead of added to it.

If the anchor point (a) (fig. 4) were shifted by a small amount (x) to some other point (a_1), so that a_1b is not perpendicular to bo, the foregoing calculations do not hold. By a similar mathematical development, however, it may be shown that the self-energization for this case becomes

$$Fc^3/3r+Fx(2c-c^3/3r^2), \qquad (26)$$

where x is the displacement of the anchor point from a to a_1 in inches. If the same numerical values used in calculating eq 24 are substituted in eq 26, one obtains a moment of

$$(4+119.2x) \text{ lb-in.} \qquad (27)$$

about a_1 or an equivalent horizontal force of

$$(0.29+8.75x) \text{ lb} \qquad (28)$$

acting with or against the 300-lb applied force, depending on which shoe is considered.

VI. DISCUSSION OF TESTS AND RESULTS

1. TEMPERATURE OF BRAKE LINING AND DRUM DURING TESTS

During the operation of the brake-lining test machine, the factor which probably causes more variation in the coefficient of friction than any other is the variation in temperature.

If a test is started with the brake lining and drum at room temperature and stops are made at equal intervals of time from a definite speed of the flywheel, the temperature of the brake lining and drum will get successively higher with each stop until a temperature is reached at which the energy dissipated by conduction, convection, and radiation of heat from the lining and drum is equal to the energy given up by the flywheel. If the interval between stops or other conditions are changed, a higher or lower temperature may be reached. In the results shown in this section, the interval between stops has been adjusted to obtain a particular temperature. No external heat was used. The temperatures here given were measured with an iron-constantan thermocouple hard-soldered to a small piece of spring brass which rode lightly on the drum directly behind the brake shoe. This method of measuring temperatures gives values which are relative but are probably slightly less than the true temperature at the face of the lining during its contact with the drum. Measurements of temperatures by this method agree closely with values obtained by placing a thermocouple between the lining and the drum under pressure at the instant the drum is brought to rest or by placing a thermocouple in the lining with the junction exactly at the face of the lining. Furthermore, readings are readily reproduced and are more convenient to make.

2. VARIATIONS OF COEFFICIENT OF FRICTION OF BRAKE LINING

The coefficient of friction of a sample of brake lining was ascertained by three conditions of test, each of which was designed to show a particular tendency. (1) Several series of stops were made from 600 rpm at 1½-min intervals to show how the friction varied under normal conditions. During each series the temperature increased from that of the room to approximately 300° F. This temperature was about the maximum at which most linings could be tested and still give comparable values of coefficient of friction over a series of runs as the temperature increased from that of the room to the maximum. (2) Forty stops were made from 600 rpm at ½-min intervals to show the tendency of a lining to fade[7] when operating under more severe

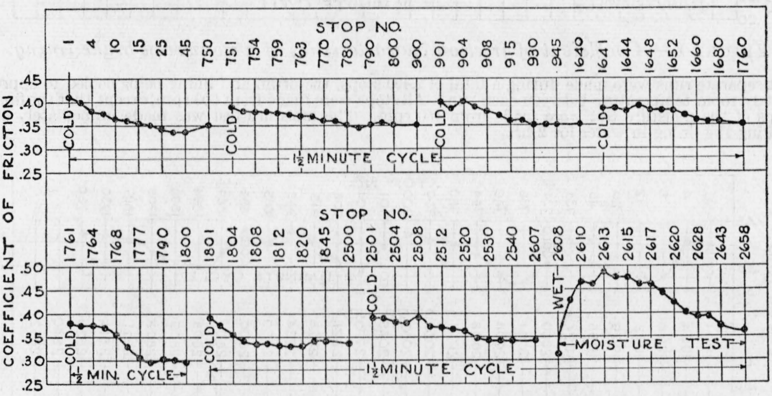

FIGURE 5.—*Coefficient of friction characteristics of a woven brake lining.*

Eight separate runs were made during a total of 2,658 stops, the drum and lining being cooled to approximately room temperature between each run. All stops were made from 600 r pm, except that the first 10 stops of the moisture test were made from 450 rpm. The moisture test was made immediately after soaking lining in water for 2 hrs.

conditions. The temperature rose to approximately 500° F during this part of the test; this temperature is considered to be critical for many linings. (3) The effect of moisture on brake lining was determined by making 100 consecutive stops at 1½-min intervals immediately after immersing the lining in water for 2 hr. The purpose of the moisture test was to determine the tendency of a lining to grab.[8] But the amount of moisture required to show this tendency was usually very critical, and the test had to be performed in such a way that the lining would dry slowly and not pass completely through the critical condition during one or two brake applications. Consequently, the severity of the test was reduced by making the first 10 stops from 450 rpm instead of 600. The remainder of the stops were made from 600 rpm to show how quickly the friction characteristic returns to normal.

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[7] A lining is said to fade when the coefficient of friction decreases rapidly to a low value.
[8] A lining is said to grab when the coefficient of friction gets high enough to cause excessive brake-retarding torque with small brake pressures.

Figures 5, 6, and 7 illustrate the results obtained for three brake linings tested in this way. The variations in the coefficient of friction shown are typical of those usually found for brake linings. As judged by a proposed specification,[9] the linings represented by figures 5 and 6

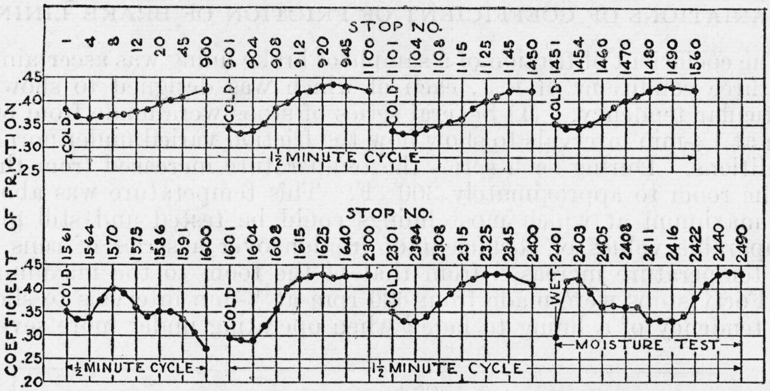

FIGURE 6.—*Coefficient of friction characteristics of a nonwoven brake lining.*

Eight separate runs were made during a total of 2,490 stops, the drum and lining being cooled to approximately room temperature between each run. All stops were made from 600 rpm, except that the first 10 stops of the moisture test were made from 450 rpm. The moisture test was made immediately after soaking the lining in water for 2 hr.

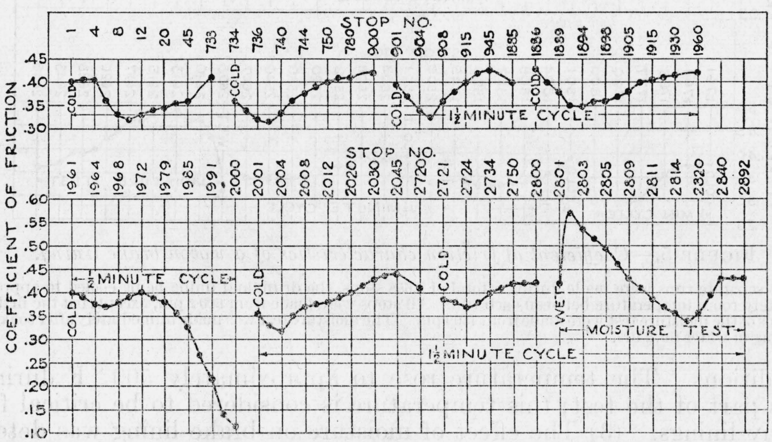

FIGURE 7.—*Coefficient of friction characteristics of a woven brake lining.*

Eight separate runs were made during 2,892 stops, the drum and lining being cooled to approximately room temperature before each run. All stops were made from 600 rpm, except that the first 10 stops of the moisture test were made from 450 rpm. The moisture test was made immediately after soaking in water for 2 hours. Comparison with figure 6 shows that the coefficient of friction is subject to larger variations.

are considered to have reasonably uniform friction while that represented by figure 7 has relatively irregular friction. The range of the coefficient of friction of the latter for the 1½-min-cycle runs is too great; it fades too much on the ½-min cycle and goes too high on the moisture part of the test.

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[9] Procurement Division Specification No. 369 for brake lining.

Apparently the way in which the coefficient of friction varies as consecutive stops are made is a function of temperature and is characteristic for each lining. In general, the coefficient of friction of a nonwoven lining will increase with a moderate rise in temperature while that of a woven lining will decrease. This is not always the case, however, as evidenced by figure 7. The coefficient of friction obtained during the 1½-min-cycle series of stops repeated itself quite closely for each run as the temperature of the lining and drum increased. During the ½-min-cycle stops, however, the temperature reached was sufficient to alter the surface of many linings and a temporary change in the friction took place until the affected part of the lining was worn away. Temperature ceased to be the predominating factor for the first part of the moisture test and the effect obtained was due mostly to water. This condition prevailed until the lining was well dried and heated, at which time the effect of temperature on the coefficient of friction again became apparent.

3. WEAR OF BRAKE LININGS

Wear is a function of the drum surface, and some reliable standard must be established if a wear test is to be made with assurance of its value.

At the present time, all tests are made with drums that have been wet-ground by means of a vitrified silicon-carbide wheel. These drums all seem to be of about the same relative smoothness, and the indications are that they give uniformly relative results.

The rate of wear was calculated by dividing the change in thickness of the lining during the test by the number of stops made. Thickness measurements were made with a ratchet micrometer (caliper) at four designated points on each piece of lining at the beginning and at the end of each test. At least 2,000 stops were made during each test.

Wear tests have been made with new drums, drums turned in a lathe, and ground drums of various degrees of smoothness. Of these, the tests made with turned drums gave the most variable results as well as the greatest wear. Figure 8 shows the results of tests made on five different linings with three distinctly different drum surfaces. These results indicate the wide variation that may be expected with nonuniform drum surfaces.

Smooth drums will cause a lining to wear much less rapidly than rough ones and, in general, will give more nearly uniform results. Furthermore, the change in the rate of wear during a test is considerably greater with a rough drum. Sometimes as much as 4,000 or 5,000 stops are required before the drum will be smoothed up enough to give consistent results. On the other hand, a relatively smooth surface, such as surface *C* figure 8, will cause much less wear, the change in the rate of wear will be much less, and only a few hundred stops are required before consistent results will be obtained.

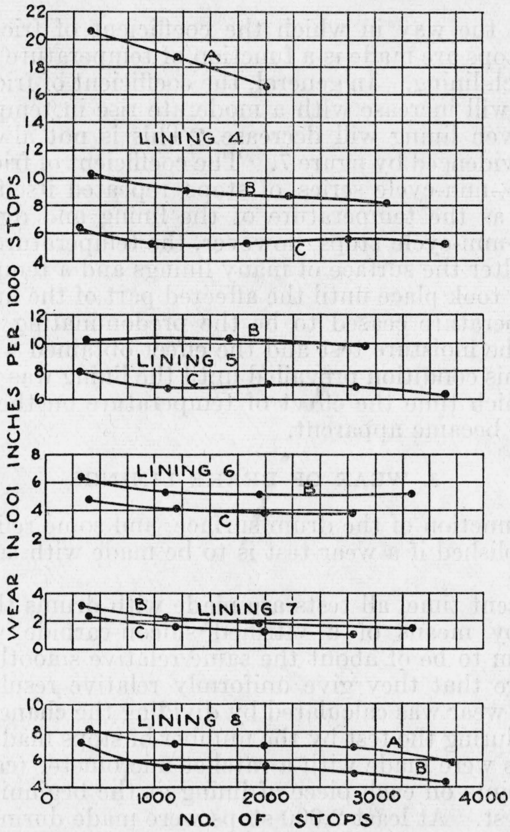

FIGURE 8.—*The effect of drum surfaces on the wear of five brake linings.*

The rates of wear shown by curves *A*, *B*, *C* are, respectively, for new, dry-ground and sanded, and wet-ground drum surfaces.

The advice and cooperation of members of the Brake Lining Manufacturers Association in the design of the machine, and the assistance of Lawrence A. Wood in laying out the control circuits are gratefully acknowledged.

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