SCHOESSEL

Design of a Brake &
Clutch Testing Machine

Mechanical Engineering

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DESIGN OF A BRAKE AND CLUTCH TESTING MACHINE

BY

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

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ENTITLED Design of a Brake and Clutch Testing Machine

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

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A BRAKE AND CLUTCH TESTING MACHINE.

Introduction.

The object of this thesis is to design a suitable machine for testing various types of friction clutches and prony brakes more or less completely. Up to the present time there have been but very few machines made with this idea in mind. The machines already in existence are so made as to test one type of clutch only and hence are of little practical importance when tests on different types are required. An attempt has been made to get a machine that will accommodate various types of clutches from the smallest size up to those capable of transmitting approximately seventy horse power, and prony brakes having a face of sixteen inches and a diameter not greater than forty-two inches.

With the machine as designed the following important points may be investigated:-

First: To determine the forces required to throw in the shifter lever when the clutch is in motion and when the clutch is at rest, also before and after the load has caused the clutch to slip.

Second: To determine the power transmitted for different adjustments of the clutch corresponding to the different forces required to throw in the shifter lever, including the maximum
power which the clutch is capable of transmitting, and the maximum power it is capable of picking up from rest.

Third: To determine the relation between the maximum forces applied at the end of the shifter lever and the corresponding axial forces, and the maximum power transmitted by the clutch for the same adjustments.

Fourth: To determine the material best suited for use as friction surfaces for either clutches or prony brakes; this would include the determination of the relations existing between the coefficient of friction, unit pressure, and speed.
CHAPTER I

Description of the Machine.

1.- Specifications. The machine primarily must be of good rigid construction so as to eliminate all jar and vibration. Friction in the machine itself should be reduced to a minimum so that as much as possible of the power put into the driving pulley be delivered at the brake. For this reason ball bearings should be used on the shafts.

The machine should be so arranged that one clutch may be taken out and another readily put into its place with a minimum of time required.

All bearings and parts subject to wear should be easily accessible for oiling and repair.

A rigid connection should be provided between the driving and driven pulleys when investigating prony brakes. This is used instead of transmitting power through a clutch in order to eliminate the possibility of slippage.

The prony brake must measure accurately the power transmitted by the clutch.

The instruments and operating levers and handwheels should be so placed that the operator can control the machine without being required to change his position very much.
Instruments must be provided for obtaining the effort required to throw in the shifter lever, to determine the total and instantaneous slippage and for securing the weight on the brake.

The driving end must be so arranged that the driving pulley may be removed and one of different size readily put on in its place.

2. General Description. By reference to the assembly of the machine, page 19, it will be seen that the power is delivered to the driving pulley A. It is then transmitted through the main shaft and first coupling B to the clutch C; through the clutch and is finally absorbed by the prony brake D. The prony brake arm rests upon a platform scale (not shown) by means of which the power transmitted is measured. The mechanism E is a device used for adjusting the brakes to the various loads applied.

At F is shown the apparatus designed to give the effort required to throw in or hold in the clutch. A screw passes through the end of the shifter lever, the other end of which is held solid by an adjustable pivot at G, thence through the shifting lever bracket which is drilled to receive it. A spring fits around the screw and rests against a shoulder inside of the bracket. By means of either of two handwheels the spring is put in compression which action throws the clutch in.

Thus to test a clutch, it is mounted between the shaft couplings B and B', the shifting lever is then adjusted to suit; the power is next thrown on, and the clutch is engaged by means of.
the shifting lever. The prony brake is now adjusted for the particular power that it is desired to transmit and readings of the several instruments are taken and recorded.

All journals are equipped with Hess Bright ball bearings so as to reduce internal friction of the machine to a minimum. These bearings are supported in heavy pillow blocks fastened rigidly to pedestals mounted on the structural steel bed of the machine. The pedestals in addition to three steel channels bolted across at right angles to the main channels, H and K, serve to prevent vibration.

It will be noticed that the operator can stand at L and control the entire machine after it is once started.

The capacity of the machine is rated at seventy horse power at one hundred revolutions per minute of the driving shaft. This maximum horse power was determined since the largest clutch for a three inch shaft will transmit approximately seventy horse power. The main shaft, belt pulley, and brake, however, will all stand larger loads and a greater number of revolutions per minute since a liberal factor of safety is used.
CHAPTER II

Calculations.

3.- Belt. Since belts having cemented joints are stronger and run more quietly than those having laced joints, the former type was selected. A double thickness belt, transmitting seventy horse power at two hundred revolutions per minute was decided upon since under certain conditions the speed of the machine may be increased as when brakes are being tested. To determine the size of belt to deliver the above horse power at the given speed the formula derived in our text on Mechanics of Machinery, Part I, was used. This formula is as follows:

\[ P = bt(m - \frac{12wy^2}{g})(\frac{ue}{eue} - 1) \]  

in which

- \( b \) represents the width of belt.
- \( t \) is the thickness of belt, \( \frac{7}{16} \) in this case.
- \( m - \frac{12wy^2}{g} \) and \( \frac{ue}{eue} - 1 \) are factors, the values of which are taken directly from the tables in the above mentioned text.

Let us assume that the driving pulley is 3 feet in diameter and that the arc of contact \( e \) is approximately 180 degrees, and, furthermore, that the coefficient of friction \( u \) is 0.35. From the table on page 48 in the above text we find that for
\[ e = 180 \text{ degrees and } u = 0.35. \]

\[ \frac{e^{ue\cdot l}}{e^{ue}} = 0.67. \]

From the table on page 49 the value of

\[ (m - \frac{12vy^2}{c}) = 389. \]

for the velocity of 1886 feet per minute which is the speed of the pulley at 200 revolutions per minute.

Now the tangential force \( P \) may be found from the horse power transmitted as follows:

\[ P = \frac{33000H}{V} \]  

in which

\( H \) represents the horse power transmitted, and \( V \) " velocity in feet per minute.

Substituting the proper values in (2) we find

\[ P = 1230 \text{ lb.} \]

Solving for the width \( b \) by evaluating (1) we get

\[ b = \frac{1230}{389 \cdot 0.67 \cdot 0.43} = 11". \]

Hence, a belt eleven inches wide and of double thickness was selected.

4.- Belt Pulley. The belt pulley should transmit seventy horse power at 200 revolutions per minute. A pulley three feet in diameter having six arms and a twelve inch face will be chosen. To determine the proportions of the arms and hub, the
formula quoted by Unwin in his text on machine design will be used. The formula for the proportions of the arm of a pulley carrying a double belt is as follows:

\[ d = 0.798 \sqrt[3]{\frac{3Bd}{n}} \]  

(3)

in which

- \( d \) represents the depth of arm,
- \( n \) " " number of arms,
- \( B \) " " face of the pulley, in inches,
- \( D \) " " diameter of the pulley, in inches.

Substituting the proper values in (3), we have

\[ d = 0.798 \sqrt[3]{\frac{3 \cdot 12.36}{6}} = 3\frac{2}{3}'' \]  

approximately.

The thickness of the arm is usually made 0.4 of the depth; hence, in this case the thickness becomes \( 1\frac{3}{8}'' \). The proportions of the arm as calculated are shown in Fig. 1.

The dimensions of the hub are generally arrived at by using good judgement. Fig. 2 shows the proportions chosen.
As in the case of the hub the rim is proportioned by judgement, the dimensions chosen being shown in Fig. 3.

5. Prony Brake. The function of the prony brake is to absorb the energy put into the machine by the source of power. To determine the size and proportions of the brake pulley a method as suggested in Carpenter's Experimental Engineering was used. The power to be absorbed, number of revolutions and diameter of the wheel are known. In order to obtain the resistance of the brake the following formula was used.

\[ F = \frac{33000H}{\pi DN} \]  

(4)
For example, in which

F represents the resistance of the brake,
H " " horse power,
D " " diameter, 3\(\frac{1}{2}\) feet in this case,
N " " number of revolutions per minute.

Knowing the values of H, D and N we can solve for F.

\[
F = \frac{33000 \cdot 70}{3.1416 \cdot 3.5 \cdot 200} = 1050 \text{ lb.}
\]

Now the strains in brake straps are essentially the same as those in a belt, hence the same principle can be used.

\[
F = T_1 - T_2 \quad \text{(5)}
\]

in which

\(T_1\) represents the greatest tension, and
\(T_2\) " least tension.

\[
\frac{F}{T_2} = 10^{2.7288fc} \quad \text{(6)}
\]

in which

\(f\) represents the coefficients of friction, and
\(c\) " percentage which the arc of contact bears to the entire circumference.

\[
T_1 = \frac{FB}{B-1} \quad \text{(7)}
\]

\[
T_2 = \frac{F}{B-1} \quad \text{(8)}
\]

in which

B equals the number whose log. is \(2.7288fc\)
\[
B = 10^{2.7288fc} \quad \text{(9)}
\]

\[
B = 10^{2.7288 \cdot 0.15} = 2.529
\]
Taking this value of B and substituting in equations (7) and (8) above,

\[ T_1 = 1050 \left( \frac{2.529}{1.529} \right) = 1740 \text{ lb.} \]

\[ T_2 = \frac{1050}{1.529} = 677 \text{ lb.} \]

From the value of \( T_1 \) we compute the required area of the brake straps, using 10,000 lb. as the safe working stress.

Section of brake straps equals \( \frac{1740}{10000} = 0.175 \text{ sq. in.} \)

A constant \( K \) depending upon the width of brake, velocity of the periphery and the horse power is given by the formula

\[ K = \frac{WY}{H} \]

in which

\( w \) represents the width of the pulley, and

\( v \) " " " velocity of the periphery in feet per minute.

In good practice an average value of \( K \) may be assumed as 500.

The velocity in this case equals 2200 feet per minute. Solving,

\[ w = \frac{KH}{v} \]

\[ w = \frac{500 \cdot 70}{2200} = 15 \text{ inches.} \]

Hence, a brake pulley having a face of fifteen inches and a forty-two inch diameter will be used.

As in the case of the belt pulley, Unwin's formula was used to calculate the proportion of the arms.

\[ d = 0.798 \sqrt[3]{\frac{HD}{N}} \]

(12)
in which

\[ B, D \text{ and } W \text{ are as before.} \]

\[ d = 0.798 \sqrt[3]{\frac{15.42}{6}} = 3.78 \text{ or } 3\frac{3}{4}". \]

Using \( b = 0.4d \) as before, we find \( b = 1\frac{1}{2}". \)

It is common practice to use a double row of arms when the power transmitted is large and the face of the pulley is wide.

The proportions of the pulley are shown in Fig. 5.
6. - Spring. This spring forms a part of the mechanism that measures the effort required to throw in the shifter lever of the clutch. It is designed for a load of two hundred and twenty-five pounds, which load was considered sufficiently large to meet all requirements. The method as given in Kent's Mechanical Engineer's Pocket Book was used in determining the various dimensions.

The outside diameter was taken \( \frac{13}{4} \)" as being a convenient size.

Now the deflection of one coil given by Kent is

\[
F = \frac{8P(D-d)^3}{Ed^4} \tag{13}
\]

in which

- \( P \) represents the load in lb. on spring,
- \( D \) " outside diameter,
- \( d \) " diameter of steel, and
- \( E \) " torsional modulus of elasticity.

\[
F = 8 \left( \frac{225(1.75-0.25)}{12000000 \cdot 0.25} \right)^3 = 0.129 \text{ inches.}
\]

When the spring is compressed there are four coils per inch, when expanded the total length should be seven inches. Hence, the expanded length of one inch of spring

\[
e = 1 + (4 \cdot 0.129) = 1.52 \text{ inches.}
\]

Solid length \( = \frac{7}{1.52} \approx 4\frac{5}{8} \) inches.

Pitch for winding \( = \frac{4}{1.52} = 2.63 \) coils per inch.

\[
\frac{1.52 - 1}{4} = 0.13 \text{ inches, which is the distance between coils when the spring is expanded.}
\]
7. Shafts. The size of the shafts was really fixed when the capacity of the machine was decided upon, that is, a clutch having a three inch bore was to be the maximum size, hence the main shafts were made to fit this clutch.

A consideration of the torsional and bending stresses in the shafts will show whether or not they are large enough.

Considering the combined twisting and bending,

\[ M_e = \sqrt{M^2 + T^2} \quad (14) \]

in which

- \( M \) represents the bending moment on the shaft,
- \( T \) represents the twisting moment on the shaft,
- \( M_e \) represents the equivalent moment on the shaft.

In Fig. 6 is shown the driven shaft.

![Fig. 6](image)

The twisting moment in the shaft,

\[ T = 63030 \frac{H \cdot P}{\pi} \quad (15) \]

in which

- \( H.P. \) represents the horse power,
- \( N \) represents the revolutions per minute,

\[ T = 63030 \frac{70}{200} = 22050 \text{ lb. in.} \]

\[ P = \frac{T}{R} \quad (16) \]

in which

- \( R \) represents the radius of the pulley,

\[ P = \frac{22050}{21} = 1050 \text{ lb.} \]

\[ F = T_1 + T_2 \quad (17) \]
in which

\[ F \] represents the load on the shaft, and

\[ T_1 \text{ and } T_2 \] are as before.

Solving graphically from the values of \( T_1 \) and \( T_2 \) obtained in Art. 5,

\[ F = 1840 \text{ lb.} \]
\[ R_1 = R_2 = \frac{1340 \cdot 11.25}{22.5} = 920 \text{ lb.} \]
\[ M_1 = 920 \cdot 11.25 = 10350 \text{ lb. in.} \]

In Fig. 7 is shown the driving shaft.

\[ P = T_1 - T_2 = \frac{22050}{18} = 1230 \text{ lb.} \]
\[ \frac{T_1}{T_2} = e u e = 3.03 \]  
\[ T_1 - T_2 = 1230. \quad \frac{T_1}{T_2} = 3.03. \]

Solving simultaneously,

\[ T_1 = 1836 \text{ lb. and } T_2 = 606 \text{ lb.} \]
\[ F = 1836 + 606 = 2442 \text{ lb.} \]
\[ R_1 = \frac{6.5 \cdot 2442}{23.5} = 884 \text{ lb.} \quad R_2 = \frac{32 \cdot 2442}{23.5} = 3320 \text{ lb.} \]
\[ M_2 = 2442 \cdot 8.5 = 20800 \text{ lb. in.} \]
\[ M_e = 10000 \sqrt{2.205^2 + 2.08^2} = 30280 \text{ lb. in.} \]

From the table on page 60 in the text it is found that the
section modulus for a three inch shaft is 2.7. Hence,

\[ S = \frac{30280}{2.7} = 11400 \text{ lb. per sq. in.} \]

Since this working stress is well below the allowable stress, the shaft is amply large.

8.- Ball Bearings. The balls and ball races are standard as manufactured by the Hess Bright Manufacturing Company. The size of the bearings was determined by a consideration of the reactions at the bearings. The greatest load on any one bearing is 3320 pounds as obtained in the shaft calculations. Now since there are two ball races in each bearing, dividing 3320 by two will give 1660 pounds as the load on each race.

A ball race capable of withstanding this load would be too small from a practical standpoint, hence one having 8400 pounds as its maximum capacity was selected. This corresponds to their radial bearing number 321.

In order to provide for the end thrust, a combination of radial bearing number 321 and thrust bearing number 1119 was used on the clutch end of the driven shaft. The end thrust will vary considerably with different types of clutches; hence the selection of the bearing is a matter of judgment.

The pillow blocks were proportioned by judgment and not by a rigid force analysis. The latter would call for rather thin walls which would not cast well.
CHAPTER III

Special Fixtures.

9.- Total Slip. In order to obtain the total slip in the clutch, a revolution counter of convenient size, preferably small, is attached to both the driving and driven end of the main shaft. The difference in readings between the two instruments for any given interval of time will give the total slip in the clutch.

10.- Instantaneous Slip. The problem of obtaining the instantaneous slip is a more difficult one and an attempt to solve it has been made by using an electrical device. Two small magnetos are either geared or driven by means of chains from the driving and driven shaft. The magnetos are first made to give the same terminal electro-motive force at equal speeds. This can be done by shunting a piece of iron across the terminals of the permanent magnet of the magneto giving the higher voltage, and reducing its area by filing so as to reduce the flux through it. By this means the two magnetos can be made to give the same terminal electro-motive force. Their armature currents are then connected in opposition and a voltmeter connected in the circuit. The instrument, which will read the excess of one electro-motive force over the other is then calibrated to read in per cent slip. This calibration can easily be made since the terminal electro-motive force
is directly proportional to the armature speed, or in other words to the rate of the cutting of the flux.

11.- Device for Measuring Effort Required to Throw in Shifting Lever. An alternate design used for measuring the effort required to throw in the clutch was devised and is shown in detail in Fig. 8. As before a screw is used to obtain the required motion of the shifter lever. Instead of a spring, however, fluid pressure is used. A grooved piston A, fits into a cylinder B, filled with oil. This cylinder is securely fastened by means of a bracket C to the frame of the machine. On the end of the screw D is a yoke E which fits over the cylinder in such a manner that a pull upon the screw exerts a pressure upon the piston. By means of suitable piping the cylinder is connected to a Bristol recording gauge upon which are recorded the pressure equivalent to the effort required to engage the clutch.

Fig. 8
**BILL OF MATERIAL.**

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<th>Name of Part</th>
<th>No. Wanted</th>
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<td>5&quot; x 1&quot; Sq. Hd. Cap Screws</td>
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