

**Study and comparison of torque transmitting characteristics of Asbestos disc clutch and 6 pad ceramic clutch in Ashok Leyland 370 engine**Sindavalam Sharath¹, Dr. Manish Kumar²¹The Institution of Engineers (India), Kolkata, India²Dept. of Production and Industrial Engg., MBM Engineering College, Jodhpur, India

Abstract— Study has been carried out on the torque transmitting characteristics of clutch plates, one having Asbestos as friction material and the other as ceramic pads, when fitted on Ashok Leyland 360 engine.. Results thus obtained are compared to ascertain the reliable one.

Index Terms— Asbestos disc clutch, ceramic clutch plate, coefficient of friction, torque transmitted.

I. INTRODUCTION

Conventional clutch plates are made of Asbestos either moulded around brass, copper or zinc wires or woven and impregnated with rubber or asphalt. Use of asbestos in clutch linings is being prohibited by many countries as asbestos dust if inhaled may lead to cancer. Thus ceramic and sintered metals (bronze base and iron base) have replaced asbestos linings. These materials possess higher wear resistance, higher coefficient of friction and can be used at higher temperatures [1]. Numerical analysis is carried out to determine the torque transmitting abilities of the clutch plates using uniform pressure theory and uniform wear theory.

Statement of the case study is- “Can we replace asbestos disc clutch with ceramic clutch plate of similar dimensions for better reliability?”

II. STRUCTURE

Study of the case is carried out on the basis of three different cases. They are:

Case I:

The torque transmitted by conventional asbestos disc clutch is determined for uniform pressure and uniform wear conditions. Torque transmitted by the clutch for both the conditions if the asbestos lining is replaced with ceramic material is calculated. The results thus obtained are analyzed to judge which of the two materials is superior to transmit torque

Case II:

The torque transmitted by 6- pad ceramic clutch plate is calculated for uniform pressure and uniform wear conditions. Factor of safety is determined and compared with that of conventional asbestos clutch disc.

Case III:

As the clutch linings wear out, the axial load exerted by compressed springs reduces as the initial compression of springs is relieved by the amount of wear out. The torque transmission characteristics of both asbestos clutch disc and 6- pad ceramic clutch plate are calculated for the condition when linings have worn out to the permissible limits as stated by OEM. Results are compared with the maximum torque developed by engine to state whether there does exist any clutch slip.

III. INPUT DATA**Ashok Leyland 370 Engine:**

The leading particulars of Ashok Leyland 370 Engine are as follows [2]:

Bore	103.28 mm
Stroke	120.65 mm
Number of cylinders	06
Maximum output	809.60 KW at 2400 RPM
Maximum torque	368.86 N-m at 1600 RPM

Asbestos Disc Clutch:

The specifications of asbestos disc clutch are as follows [2]:

Outer diameter	356 mm
Frictional area (Both sides)	1238 cm ²

Thickness of single lining	6.35 mm
Permissible wear out of lining	3.81 mm

6- Pad Ceramic Clutch Plate:

The specifications of 6- pad ceramic clutch plate are as follows [3]:

Outer diameter	356 mm
Frictional area of each pad	41.54 cm ²
Number of pads on either side	06
Thickness of friction pad	6.35 mm

Pressure Plate:

The specifications of pressure plate are as follows [2]:

Number of springs	12 (3 groups of 4 springs)
Total spring force	8289.45 N
Free length of spring	11.88 cm
Stiffness of spring	120.76 N/cm

IV. NUMERICAL ANALYSIS

The material used for friction pads in both the clutch plates are known. Thus the range of values of coefficient of friction [1] of asbestos lining and ceramic pads are as given below:

Type of material	Coefficient of friction, μ
Asbestos lining	0.20 – 0.30
Ceramic composites	0.40 – 0.60

In the further study, it is presumed that the coefficient of friction of asbestos lining is 0.25 and for ceramic pads is 0.60.

CASE I:

TORQUE TRANSMITTED BY ASBESTOS DISC CLUTCH

Given,

Frictional area, $A = 619 \text{ cm}^2$ (Single side of clutch) [2]

Axial force, $W = 8289.45 \text{ N}$ [2]

Uniform Pressure Theory:

Frictional torque,

$$T = n\mu WR \quad (1)$$

Where, n = Number of contact sides = 2

μ = Coefficient of friction (0.25) [1]

R = Mean radius of friction surface which is given by [4]

$$R = \frac{2}{3} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right]$$

Where, r_1 and r_2 are external and internal radii of friction faces.

Outer radius, $r_1 = 17.8 \text{ cm}$

Area of friction surface, $A = \pi(r_1^2 - r_2^2)$
 $619 = \pi(17.8^2 - r_2^2)$

$r_2 = 10.94 \text{ cm}$

Thus, $R = 14.64 \text{ cm}$ or 0.1464 m

Frictional torque, $T = 2 \times 0.25 \times 8289.45 \times 0.1464$
 $= 606.787 \text{ N-m}$

Factor of safety, F_s is given by

$$F_s = \frac{\text{Frictional torque}}{\text{Maximum torque produced by engine}} \quad (2)$$

$$F_s = \frac{606.787}{368.86} = 1.645$$

Uniform Wear Theory:

Frictional torque, $T = n\mu WR$

Where, R = Mean radius of friction surface which is given by [4]

$$R = \left[\frac{r_1 + r_2}{2} \right]$$

Thus, $R = 14.37$ cm or 0.1437 m

Frictional torque, $T = 2 \times 0.25 \times 8289.45 \times 0.1437$
 $= 595.596$ N-m

Factor of safety for this case would be,

$$F_s = \frac{595.596}{368.86} = 1.615$$

TORQUE TRANSMITTED BY CLUTCH DISC WHEN ASBESTOS LININGS ARE REPLACED WITH CERAMIC MATERIAL

For ceramic material, it is presumed that the value of coefficient of friction μ is 0.60 .

Uniform Pressure Theory:

Frictional torque, $T = n\mu WR$

$$= 2 \times 0.6 \times 8289.45 \times 0.14$$

$$= 1456.29 \text{ N-m}$$

Factor of safety is,

$$F_s = \frac{1456.29}{368.86} = 3.948$$

Uniform Wear Theory:

Frictional torque, $T = n\mu WR$

$$= 2 \times 0.6 \times 8289.45 \times 0.1437$$

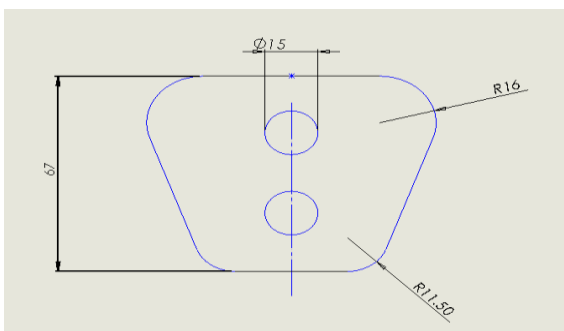
$$= 1429.43 \text{ N-m}$$

Factor of safety is,

$$F_s = \frac{1429.43}{368.86} = 3.875$$

CASE II:

TORQUE TRANSMITTED BY 6- PAD CERAMIC CLUTCH PLATE



Dimensions of ceramic friction pad

Given [3],

Outer radius of clutch = 17.8 cm

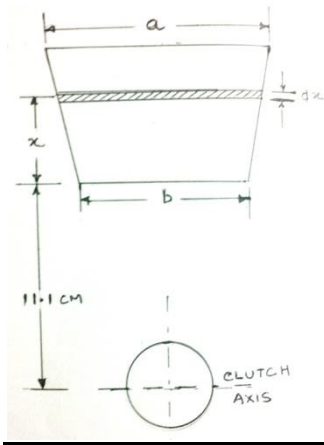
Width of the pad = 6.7 cm

Distance from centre of the clutch to the inner side of friction pad = 11.1 cm

Frictional area of each pad, A_s (calculated) = 41.54 cm^2

The pad of given area can be approximated to a shape of trapezium having the dimensions of parallel sides as 7.7 cm and 4.7 cm and height of 6.7 cm.

Uniform Pressure Theory:



Let the length of larger parallel side be ' a ' and length of smaller parallel side be ' b '. Let height of the pad be ' l '. Length of the parallel side at a distance x from the smaller parallel side is given by

$$b + \frac{(a-b)}{l} x$$

Area of the elementary section of thickness dx at a distance of x from the smaller parallel side is given by

$$\left[b + \frac{(a-b)}{l} x \right] dx$$

Normal or axial force on the elementary area of single pad [4],

$$\begin{aligned} \delta W &= \text{Pressure} \times \text{Area} \\ &= p \times \left[b + \frac{(a-b)}{l} x \right] dx \end{aligned}$$

Let $\frac{(a-b)}{l}$ be ' k '.

Frictional force on the elementary area acting tangentially at a distance $(x + 0.111)$ m,

$$F_r = \mu \times \Delta w \quad (3)$$

Frictional torque acting on the elementary area,

$$T_r = F_r \times (x + 0.111)$$

Total frictional torque acting on the friction surface of single pad,

$$\begin{aligned} T_s &= \int_{0.111}^{0.178} \mu p (b + kx) (x + 0.111) dx \\ &= \mu p \int_{0.111}^{0.178} (bx + 0.111b + kx^2 + 0.111kx) dx \\ &= \mu p [0.0485x^2 + 0.005217x + 0.15x^3]_{0.111}^{0.178} \\ &= \mu p \times 1.92946 \times 10^{-3} \\ &= \mu \times \frac{W}{A_s} \times 1.92946 \times 10^{-3} \end{aligned}$$

Axial force, $W = 8289.45$ N [2]

Area of 6 pads, $A = 6 \times 0.004154 \text{ m}^2 = 0.024924 \text{ m}^2$

Total frictional torque acting on the clutch plate,

$$T = n\mu \times \frac{W}{A} \times 1.92946 \times 10^{-3}$$

Where, n = Number of contact sides = 2

For $\mu = 0.6$ [1],

Total frictional torque acting on the clutch, $T = 770.06$ N-m

Factor of safety would be,

$$F_s = \frac{770.06}{368.86} = 2.09$$

Uniform Wear Theory:

While designing machine parts which are subjected to wear due to sliding friction, it is assumed that the normal wear is proportional to the work of friction. The work of friction is proportional to the product of normal pressure (p) and the sliding velocity ($V = 2\pi rN$) [4]. Therefore,

Normal wear $\propto$ Work of friction $\propto pV$

or $pV = K$ (a constant)

or $p = K/V$

When the friction surface is new, there is a uniform pressure distribution over the entire contact surface. This pressure will wear most rapidly where the sliding velocity is maximum and this will reduce the pressure between the friction surfaces. This wearing- in process continues until the product pV is constant over the entire surface. After this, the wear will be uniform.

Let p be the normal intensity of pressure at a distance $(x + 0.111)$ m from the axis of the clutch. Since the intensity of pressure varies inversely with the distance, therefore

$$p(x + 0.111) = C \text{ (A constant)}$$

$$\text{or } p = C/(x + 0.111)$$

Normal force on the elementary area,

$$\delta W = p \times \left[b + \frac{(a-b)}{l} x \right] dx$$

Total force acting on the friction surface of single pad,

$$\begin{aligned} W_s &= \int_{0.111}^{0.178} \frac{C}{(x + 0.111)} (b + kx) dx \\ &= C \int_{0.111}^{0.178} \frac{1}{(x + 0.111)} (b + kx) dx \end{aligned}$$

Total force acting on 6 ceramic pads is,

$$W = 6C \int_{0.111}^{0.178} \frac{1}{(x + 0.111)} (b + kx) dx$$

We know that total axial force, W exerted by the springs on the friction surface is 8289.45 N. Thus,

$$8289.45 = 6C \int_{0.111}^{0.178} \frac{1}{(x + 0.111)} (b + kx) dx$$

Let,

$$x + 0.111 = t$$

$$dx = dt$$

$$\text{At } x = 0.111, t = 0.222,$$

$$x = 0.178, t = 0.289$$

$$8289.45 = 6C \int_{0.222}^{0.289} \frac{1}{t} [b + k(t - 0.111)] dt$$

Solving the above equation, we get

$$W = 0.1764 C$$

$$C = 46992.347 \text{ N/m}$$

Total frictional force acting on the clutch plate,

$$T = n \int_{0.111}^{0.178} \mu p (b + kx) (x + 0.111) dx$$

$$T = n\mu C \int_{0.111}^{0.178} \frac{1}{(x + 0.111)} (b + kx) (x + 0.111) dx$$

$$T = 2 \times 0.6 \times C \int_{0.111}^{0.178} (b + kx) dx$$

$$T = 1.2 \times C \left[bx + \frac{kx^2}{2} \right]_{0.111}^{0.178}$$

$$T = 1.2 \times C \times [0.047x + 0.225x^2]_{0.111}^{0.178}$$

$$T = C \times 9.00681 \times 10^{-3}$$

$$T = 423.25 \text{ N-m}$$

Factor of safety for this case is,

$$F_s = \frac{423.25}{368.86} = 1.15$$

CASE III:

CONDITIONS OF CLUTCH SLIP

The axial or normal pressure exerted by the springs provides a frictional force in the circumferential direction when the relative motion between the driving and driven members tends to take place. The clutch would slip when [1]:

1. The torque due to frictional force is less than the torque to be transmitted.
2. The wear of frictional lining exceeds the limiting value. This amount of wear decompresses the springs and thus axial pressure exerted would reduce causing slip of clutch.
3. The driving shaft and driven shaft rotate at different speeds. Clutch would slip until the angular velocities of the two shafts become equal or the relative velocity becomes zero.

Condition I (Torque due to frictional force):

The frictional force exerted by the clutch discs in all the above cases is more than the maximum torque developed by the engine or in other words the factor of safety for all the cases is more than unity. Thus, there exists no clutch slip due to this frictional force.

For the condition of engine when producing maximum output of 110 HP or 809.60 KW at 2400 rpm, torque developed is given by,

$$T = \frac{P}{\omega} = \frac{P \times 60}{2\pi N} \quad (4)$$

$$= \frac{80960 \times 60}{2\pi \times 2400} = 322.30 \text{ N-m}$$

The torque developed at maximum output is also less than the frictional torque acting on the clutch plate and thus clutch slip doesn't occur at this condition.

Condition II (Wear of frictional lining):

Given [2],

Free length of the spring = 11.88 cm

Stiffness of the spring = 120.767 N/cm

Total axial load exerted by the springs, $W = 8289.45 \text{ N}$

Total stiffness of 12 springs = $120.767 \times 12 = 1449.204 \text{ N/cm}$

$$\begin{aligned} \text{Initial compression of springs} &= \frac{\text{Axial load, } W}{\text{Total stiffness of springs}} \quad (5) \\ &= \frac{8289.45}{1449.204} \\ &= 5.72 \text{ cm} \end{aligned}$$

The permissible wear limit of single clutch lining is given as 3.81 mm [2]. When considering both the contact surfaces, the maximum permissible wear limit would be 7.62 mm or 0.762 cm. Thus we need to calculate the torque transmission capacity of both the clutch plates when the initial compression of spring is relieved by 0.762 cm or in other words when the springs are compressed by 4.958 cm.

For asbestos disc clutch:

We shall consider the uniform wear theory as the clutch plate is worn out.

Total axial force exerted by the springs,

= Stiffness of 12 springs $\times$ Compression of spring

= 1449.204 $\times$ 4.958

= 7185.1534 N

Torque transmitted by the clutch (uniform wear) = $n\mu WR$ [4]

= 2 $\times$ 0.25 $\times$ 7185.1534 $\times$ 0.1437

= 516.2532 N-m

We know that maximum torque developed by the engine is 368.856 N-m [1]. The torque transmitted by worn out asbestos clutch disc under uniform wear condition is more than maximum torque developed by engine. Thus clutch wouldn't slip even when the linings have worn out up to the permissible limit. OEM might have decided this limit to provide clearance between clutch release levers and clutch release bearing.

For 6- pad ceramic clutch plate:

Total axial force exerted by the springs (when linings are worn out), $W = 7185.1534$ N

Earlier we had derived for uniform wear conditions,

$$W = 0.1764 C$$

$$\text{or } C = \frac{7185.1534}{0.1764} = 40732.162$$

Torque transmitted by the clutch = $C \times 9.00681 \times 10^{-3}$

= 366.867 N-m

In this case the frictional torque acting on the friction surfaces of clutch plate is less than the maximum torque developed by the engine. Thus at the condition when engine is developing maximum torque and the linings have worn out to the permissible limit, the clutch plate slips. Hence 6-pad ceramic clutch plate must be replaced when the pads get worn more than the said permissible limit.

Condition III (Relative velocity of driving and driven shafts):

The difference in velocities of driving and driven shafts is known as relative velocity at time t and is given by [1],

$$\omega = \frac{d\theta}{dt} = \frac{d\theta_1}{dt} - \frac{d\theta_2}{dt} \quad (6)$$

$$\text{or } \omega = (\omega_1 - \omega_2) - T \left(\frac{I_1 + I_2}{I_1 I_2} \right) t \quad (7)$$

Where, θ_1 and θ_2 are angular displacements of driving and driven shafts (rad)

ω_1 and ω_2 are angular velocity of driving and driven shafts (rad/s²)

I_1 and I_2 are moments of inertia of driving and driven shaft (Kg-m²)

T is clutch torque (N-m)

The clutching operation is complete at the instance when the relative velocity becomes zero. Let at this instance time is t_1 . Equating relative velocity to zero and substituting $t = t_1$, we get

$$t_1 = \frac{(\omega_1 - \omega_2) I_1 I_2}{(I_1 + I_2) T} \quad (8)$$

It is observed that time required for engaging the clutch or duration of clutch slip is directly proportional to the difference in the angular velocities of both the shafts and inversely proportional to the torque.

Also the total energy dissipated in the form of frictional heat during clutching operation is given by [1],

$$E = \frac{1}{2} \frac{(\omega_1 - \omega_2)^2 I_1 I_2}{(I_1 + I_2)} \quad (9)$$

Thus the energy dissipated during clutch slip is independent of the torque and directly proportional to the square of relative velocity of driving and driven shafts. Also the frequency of clutching operation is also an important parameter as the temperature of friction pads increases and requires time to dissipate heat.

The duration of clutch slip and energy dissipated could be calculated if the values of other parameters are known.

V. FINAL RESULTS

The torque transmission characteristics and state of clutch slip for both the clutch plates have been analyzed. The results obtained for various cases are tabulated as given below:

CASE I:

Type of theory	Torque transmitted (N-m)			
	Asbestos disc clutch ($\mu=0.25$)	Factor of safety	When asbestos lining replaced with ceramic material ($\mu=0.6$)	Factor of safety
Uniform pressure theory	606.787	1.645	1456.29	3.948
Uniform wear theory	595.596	1.615	1429.43	3.875

CASE II:

Type of theory	Torque transmitted (N-m)			
	Asbestos disc clutch ($\mu=0.25$)	Factor of safety	6- Pad Ceramic clutch plate ($\mu=0.6$)	Factor of safety
Uniform pressure theory	606.787	1.645	770.06	2.09
Uniform wear theory	595.596	1.615	423.25	1.15

CASE III:

Type of clutch plate	Torque transmitted (N-m) by clutch when lining has worn out by 7.62mm [2]	Maximum torque developed by engine (N-m) [2]	Clutch slip
Asbestos disc clutch	516.253	368.856	Doesn't exist
6- pad ceramic clutch plate	366.867		Exists

VI. CONCLUSION

After the analysis is carried out on asbestos disc clutch and 6- pad ceramic clutch plate, the following statements are made after comparing the results:

1. Conventional asbestos disc clutch is possessing better factor of safety. But if the asbestos linings are replaced with ceramic material, the torque transmitted and factor of safety achieved is much higher.
2. When coefficient of friction for ceramic pad is presumed to be 0.6, the clutch plate serves the need though the factor of safety is less when compared with asbestos disc clutch.

3. When the engine is developing maximum torque, clutch slip doesn't occur in asbestos disc clutch even though the linings have worn out to the permissible limits.

4. The 6- pad ceramic clutch plate slips when the engine is developing maximum torque and the linings have worn out to permissible limits i.e., 7.62 mm.

Finally, it is concluded that as ceramic material has better torque transmission characteristics, if the asbestos linings are replaced with ceramic material, then the clutch plate would yield greater factor of safety. But in case of 6- pad ceramic clutch plate, the design criteria of ceramic pads needs to be revisited in order to increase the factor of safety and also to eliminate clutch slip.

As the wear rate or life of asbestos lining and ceramic pads is not tested, the time scale for the material to wear up to permissible limit couldn't be determined. Thus with the above carried study it is not possible to comment on the reliability of these clutch plates.

VII. SCOPE OF FUTURE STUDY

Though study could be conducted on the torque transmission characteristics of various friction materials, it is obvious that wear rate needs to be known to precisely define reliability. For a fixed amount of braking the amount of wear of asbestos material tends to remain fairly constant or increase slightly with respect to temperature, but once the temperature reaches $>200^{\circ}\text{C}$, the wear increases exponentially with increasing temperature because of thermal degradation of blend components and other chemical changes. Whereas the cermet materials are more thermally resistant than asbestos. The clutch discs need to be tested on a full brake Dynamometer test bed to determine the wear rate for similar clutching operations which reflects the actual performance of the friction material [5]. Also in the study, the values of coefficient of friction of frictional material are presumed, as exact values of the asbestos lining and ceramic composite pads are not known. The ceramic composite material has coefficient of friction values more than 0.4 and is insensitive to grabbing [6].

With the aid of computer simulation software and exact values of coefficient of friction of friction pads, better analysis could be carried out. An optimized solution thus arrived could replace conventional components with more efficient and eco-friendly ones.

REFERENCES

- [1] VB Bhandari, Design of machine elements 3e, Mc Graw Hill Education Pvt. Ltd., Published 2010, Friction clutches, page No. 448 - 471.
- [2] Ashok Leyland Service Manual, Published 2009, Leading particulars of vehicle.
- [3] Clutch Auto India Ltd., Parts Catalogue, Published 2015.
- [4] RS Khurmi and JK Gupta, A text book of machine design 13e, S Chand and Company Ltd., Published 2002, Clutches, page No. 802 – 829.
- [5] http://www.mkautoindia.com/products/brake_linings_rolled_linings.html.
- [6] Walter Krenkel, Ceramic Matrix Composites: Fiber Reinforced Ceramics and their Applications, Wiley VCH, 21 May 2008, Ceramic Clutches, page No. 403-404.