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Dry Friction Clutch Disc of an Automobile under Transient Thermal Load: A Comparison of Friction Lining Materials

Anosh Ali[1], Liaqat Ali[2], Samiur Rahman Shah[2], Mushtaq Khan[2], Syed Husain Imran[2] and Shahid Ikramullah Butt[2]

[1]*Department of Mechatronics Engineering, College of Electrical and Mechanical Engineering, National University of Sciences and Technology (NUST), Islamabad 44000, Pakistan*
[2]*School of Mechanical and Manufacturing Engineering (SMME), National University of Sciences and Technology (NUST), Islamabad 44000, Pakistan*

Abstract. This paper shows the comparison of temperatures produced in a dry friction clutch disc with different materials during a single engagement to assist in clutch plate design and analysis. A study of usage of different materials for friction lining of clutch disc is required, which will provide improved performance and enhanced life. This investigation is modelled mathematically and solved numerically using finite element method. ANSYS® 15.0 is a dedicated finite element package used for determining the temperature distribution across a clutch disc. In the present work, an investigation of a conventionally used harmful friction lining material asbestos is compared with carbon-carbon composite, S2-glass fibre and aluminium metal matrix composite. The transient thermal analysis of a clutch disc with different materials is performed and the temperature distribution on the clutch system is compared. Simulation results indicate that all the values of the temperature obtained from the analysis of aluminium metal matrix are less than those of asbestos based lining material, therefore clutch disc made up of aluminium metal matrix composite will assure the extended service life and the longer stability due to the fact that the temperature responsible for the wear and tear has been reduced. Furthermore, the slipping time is also considered in this investigation.

1 Introduction

A clutch is commonly used in automotive vehicles for power transfer from the engine to the transmission system. Since a clutch has to be replaced many times before the car reaches end of its lifetime, so it becomes necessary to enhance its functional life. The lifetime of a clutch is highly dependent on the life and performance of the clutch plate. An automobile dry friction clutch system consists of a flywheel, clutch plate and a pressure plate.

Clutch disc comes in direct contact with the flywheel of the engine for power transmission. It consists of frictional material on both sides and an axial cushion is sandwiched between them. Friction lining is used to prevent the slippage between the two contacting surfaces. The engine side face of the friction lining couples with the flywheel while the transmission side couples with the pressure plate. When a clutch is engaged, the relative motion due to slippage between the contacting surfaces of a clutch system results in high surface temperatures which may lead to the surface cracks, enhanced wear rate as well as permanent distortions which can cause premature failure of the contacting surfaces [1-2].

Slippage occurring between the two contacting surfaces results in heat generation and consequently, increase in temperature. Increased temperature in clutch disc results in elastic deformations and thermal cracking which is responsible for the failure of the friction material before the expected lifespan of the clutch [3]. The generated temperature will be conducted between the clutch system as well as dissipated to the atmosphere [4].

Recently, thermal problems due to sliding of dry friction clutches and brakes are being extensively studied. It is found that the temperatures fields of the brake disc are highly effected by the contact pressure with specific material properties [5].

For a clutch disc, a study of using new friction materials is needed for improved performance and enhanced life of plate as well as clutch itself. Conventional materials are being replaced by the composites in both the clutch and brake applications. Aluminium composite reinforced with natural fibre is a good option in the application of automotive brake pads [6-7]. Behaviour of Aluminium metal matrix composites and high strength glass fibres and towards friction and wear is proved to be promising. S2 glass fibre is proved to be a suitable material for braking operation as a result of transient thermos-elastic analysis of brake disc for various materials in frequent braking operation [8]. Thermal resistance of glass fibres is more as compared to conventionally used asbestos [9]. There is a lot of research still to be done in the area of composite friction materials [4].

The study of thermo-mechanical characteristics reveals the dependence of heat transfer on slippage time and contact area ratio. Since, the pressure is maximum in

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constant pressure, hence, heat distribution is also maximum in this case. The maximum temperature appear at the half of the slippage time. [1, 10].

During clutch engagement, temperature distribution under constant pressure is higher than pressures of other type and the uniform temperature in radial direction due to continuous contact surface [11]. The one dimensional time variant heat transfer is dependent on friction in two sliding contact brake disk [12]. Expressions for determining the heat dissipation rate during single clutch engagement are a function of thermal conductivity, elastic modulus, and coefficient of thermal expansion [13].

In this study, the clutch lining temperature during contact between clutch disc and flywheel and also between clutch disc and pressure plate during single engagement is evaluated by finite element method. Transient thermal module in ANSYS® 15.0 is used to determine the temperatures produced in the friction lining of a clutch disc for different materials and the results obtained are compared.

2 Problem formulation

During slipping period (t_s), a lot of kinetic energy produced is transformed into thermal energy. In the present work, it is assumed that because of slipping, the energy due to friction between two sliding surfaces is converted into heat energy at the interface [14]. During slipping, the total heat produced is given as

$$Q(r,t)=\begin{cases}\mu p V_s & ; \quad 0\le t\le t_s\\ 0 & ; \quad t>t_s\end{cases} \tag{1}$$

$$V_s=\omega_s r \tag{2}$$

Here sliding velocity is denoted by V_s whereas ω_s is used for the sliding angular velocity, and is assumed to decrease linearly with time as,

$$\omega_s(t)=\omega_0\left(1-\frac{t}{t_s}\right) \quad ; \quad 0\le t\le t_s \tag{3}$$

The initial sliding angular velocity is ω_0 (at $t=0$) at the start of the slippage. The heat produced at any time of slipping on the clutch surface is

$$Q_c(r,t)=f_c\mu p r\omega_0\left(1-\frac{t}{t_s}\right) \quad ; \quad 0\le t\le t_s \tag{4}$$

The heat dissipation between friction lining and pressure plate/flywheel is given by f_c which is the heat partition ratio and is calculated as

$$f_c=\frac{\sqrt{k_c\rho_c c_c}}{\sqrt{k_c\rho_c c_c}+\sqrt{k_f\rho_f c_f}}=\frac{\sqrt{k_c\rho_c c_c}}{\sqrt{k_c\rho_c c_c}+\sqrt{k_p\rho_p c_p}} \tag{5}$$

where the indexes c, f and p denotes friction lining, flywheel and pressure plate [11].

During start of clutch engagement process, there will be a difference of velocities between friction lining and flywheel i.e. between driven and the driving shaft and as a result, slippage occurs between the contacting surfaces and it will be responsible for the heat generation [15].

This heat produced is dissipated through the process of conduction between components of the clutch system and also through the process of convection to the surrounding environment [13] as shown in Fig.1.

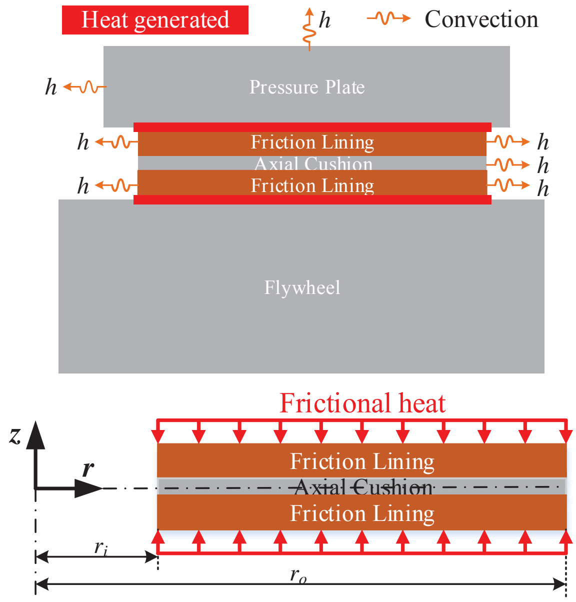


Figure 1. Heat Transfer.

Clutch parameters taken for the analysis are shown in Table 1 [16] and the three-dimensional model for the clutch system developed with the help of these parameters is shown in Fig. 2.

Table 1. Clutch Parameters

Parameters	Values
Inner radius of friction lining and axial cushion, r_i [m]	0.06298
Outer radius of friction lining and axial cushion, r_o [m]	0.08721
Inner radius of pressure plate [m]	0.05814
Outer radius of pressure plate [m]	0.09205
Inner radius of flywheel [m]	0.04845
Outer radius of flywheel [m]	0.0969
Thickness of friction lining, [m]	0.003
Thickness of axial cushion [m]	0.0015
Thickness of pressure plate [m]	0.00969
Thickness of flywheel [m]	0.01938
Slippage time, t_s [s]	0.4
Maximum pressure, p_o [Pa]	1×10^6
Initial relative angular velocity, ω_{ro} [rad s^{-1}]	200
Ambient temperature, T_a [K]	295.15

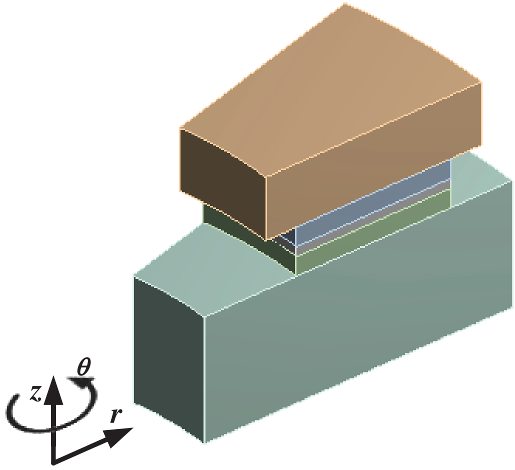


Figure 2. Coordinate system of analysis.

For the temperature field analysis of the clutch system, the starting point is the parabolic heat conduction equation in the cylindrical coordinate system, (r-radial coordinate (m), θ-circumferential coordinate (rad), z-axial coordinate (m)), which is centred in the axis of the disc and z refers to its thickness [1, 17].

$$\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}=\frac{1}{\alpha}\frac{\partial T}{\partial t} \quad ; \qquad (6)$$
$$r_i \le r \le r_o, \quad 0 \le \theta \le 2\pi, \quad 0 \le z \le \delta, \quad t>0$$

Here α represents the thermal diffusivity.

In the present work, the temperature in z-axis is symmetric due to same applied load value and also same heat transfer coefficient on friction clutch surfaces. Convection takes place from the exposed surfaces of the clutch model with the heat transfer coefficient taken as independent of the temperature. The material properties of the friction material, flywheel, pressure plate and axial cushion are assumed to be isotropic and independent of the temperature [18]. The temperature fields in the clutch are found with the help of the below mentioned three-dimensional boundary-value problems.

$$K_{cu}\left.\frac{\partial T}{\partial r}\right|_{r=r_0}=h\left[T(r_0,z,t)-T_a\right] \quad ; \qquad (7)$$
$$0 \le \theta \le 2\pi, \quad 0 \le z \le \frac{t_{cu}}{2}, \quad t \ge 0$$

Where h is the convection heat transfer and T_a is the ambient temperature.

$$K_c\left.\frac{\partial T}{\partial z}\right|_{z=\frac{t_{cu}}{2}+t_c}=Q_c(r,t) \quad ; \qquad (8)$$
$$r_i \le r \le r_0, \quad 0 \le \theta \le 2\pi, \quad 0 \le t \le t_s$$

$$K_c\left.\frac{\partial T}{\partial r}\right|_{r=r_i}=h\left[T(r_0,z,t)-T_a\right] \quad ; \qquad (9)$$
$$0 \le \theta \le 2\pi, \quad \frac{t_{cu}}{2} \le z \le \frac{t_{cu}}{2}+t_c, \quad t \ge 0$$

The initial temperature is given as

$$T(r,\theta,z,0)=T_i \quad ; \qquad (10)$$
$$r_i \le r \le r_o \quad 0 \le \theta \le 2\pi, \quad 0 \le z \le t_c+\frac{t_{cu}}{2}$$

$$\left.\frac{\partial T}{\partial r}\right|_{r=r_i}=0 \quad ; \qquad (11)$$
$$0 \le z \le \frac{t_{cu}}{2}, \quad t \ge 0$$

$$\left.\frac{\partial T}{\partial z}\right|_{z=0}=0 \quad ; \qquad (12)$$
$$r_i \le r \le r_o, \quad 0 \le \theta \le 2\pi, \quad t \ge 0$$

Rate of convection heat transfer $Q_{conv.}$ is

$$Q_{conv.}=h[T(r,z,t)-T_a]; \quad t \ge 0 \qquad (13)$$

During launch, the total clutch energy Q produced as shown in Fig. 1 can be calculated by

$$Q(t)=\int_0^{t_s}\tau\omega_r(t)\cdot dt \qquad (14)$$

Initially pressure is uniform for new plates. Wear at outer radius will be more and as a result, the shape will be changed and pressure will be redistributed .i.e. less at outer radius and more at inner radius. Torque on friction lining is

$$\tau=r\mu F=2\pi\mu p_o r_i\int_{r_i}^{r_o} r\cdot dr \qquad (15)$$

Heat generated between flywheel/pressure plate and friction lining due to relative angular velocity is

$$Q_c(r,t)=\tau\omega_r(t)=2\pi\mu p_o r_i\omega_r(t)\int_{r_i}^{r_o} r\cdot dr \qquad (16)$$

$$Q(r,t)=2\pi\mu p_o r_i\omega_{ro}\left(1-\frac{t}{t_s}\right)\int_{r_i}^{r_o} r\cdot dr \qquad (17)$$

$$Q(t)=\pi\mu p_o r_i\omega_{ro}\left(r_o^2-r_i^2\right)\left(1-\frac{t}{t_s}\right) \qquad (18)$$

$$Q(t)=\begin{cases}\pi\mu p_o r_i\omega_{ro}\left(r_o^2-r_i^2\right)\left(1-\dfrac{t}{t_s}\right) & ; \quad 0 \le t \le t_s \\ 0 & ; \quad t>t_s\end{cases} \qquad (19)$$

The heat generation with respect to time on the clutch plate lining and pressure plate/flywheel interaction is calculated as above.

3 Finite element formulation

The finite element (FE) modelling is carried out with ANSYS Meshing as shown in Fig. 3 using 3-D 20-node thermal solid element SOLID90, whereas, the contact between friction lining and pressure plate/flywheel is modelled with 3-D target surface TARGE170 and 3-D 8-

node surface-to-surface contact element CONTA174. The convection and heat generation surfaces are modelled with 3-D thermal surface effect element SURF152. The model is discretised into 127,650 nodes and 335,494 elements to assure seamless transfer of heat throughout the model. The mesh is kept uniform by mapped face meshing and relevance between the contacts of friction lining and pressure plate/flywheel is set to 100% to avoid any discontinuity in heat transfer. This relevance is not affecting the contacts between friction lining and axial cushion, since the heat is dissipated through flywheel and pressure plate, not through axial cushion.

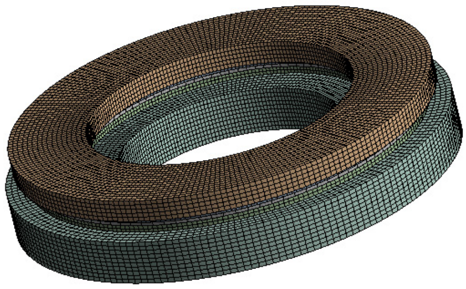
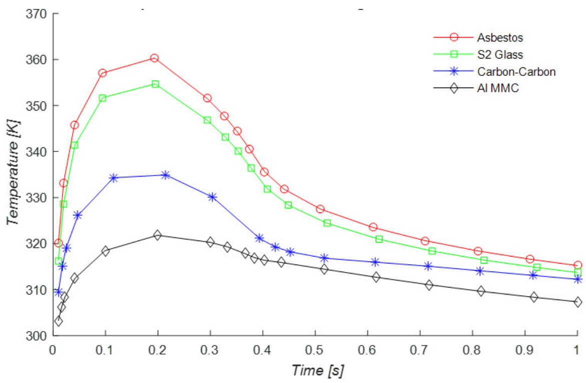


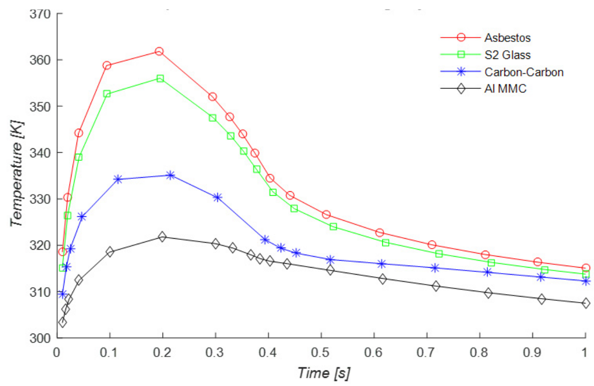
Figure 3. Finite Element Model.

4 Results

The slippage time taken in the simulation is 0.4 second hence the maximum temperature i.e. 361.83 K is occurring at half of the slippage time. The same trend is observed if the friction lining of asbestos is replaced by carbon-carbon composite, S2-glass fibre and aluminium metal matrix composite as shown in Fig. 4.



(a)



(b)

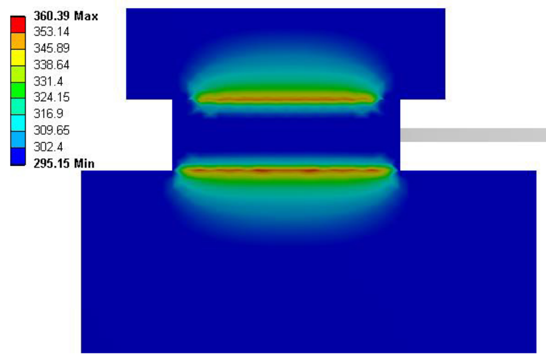
Figure 4. Temperature Profile of: (a) Friction Lining/Pressure Plate side and (b) Friction Lining/Flywheel side

A comparison of the maximum temperature of various materials as a result of the transient thermal analysis is given in the Table 2.

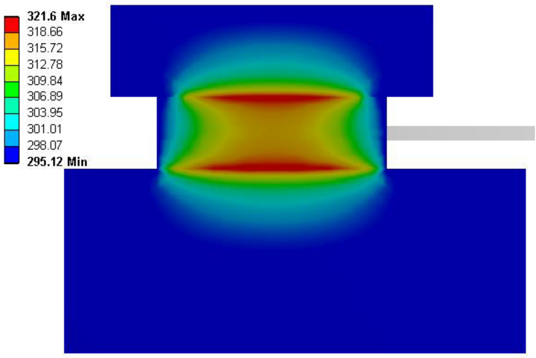
Table 2. Maximum temperature for the tested friction materials

Material	Temperature at Pressure Plate side (K)	Temperature at Flywheel side (K)
Asbestos	360.39	361.83
S2 Glass Fibre	354.73	356.04
Carbon-Carbon Composite	334. 64	335.12
Aluminium Metal Matrix Composite	321. 6	321.79

The temperature is found maximum in conventional friction lining material asbestos and minimum in aluminium metal matrix composite as shown in Fig. 5. The heat dissipation trend is established in the opposite manner, i.e. maximum for aluminium metal matrix composite and minimum for asbestos.



(a)



(b)

Figure 5. Temperature Distribution at clutch system with a friction lining of: (a) Asbestos and (b) Aluminium Metal Matrix Composite

5 Conclusions and future scope

In this paper, transient thermal analysis of three different materials for the replacement of Asbestos based clutch lining is performed. The analysis is done to assure the extended functional life and longer stability of the clutch disc and ultimately the clutch itself. Simulation results successfully indicate that Aluminium Metal Matrix composite is better material selection for friction lining in order to reduce rise in temperature and to maximize heat dissipation as compared to Asbestos based friction lining.

Future investigation will attempt to develop a system that will take initial design parameters and material properties of friction lining as an input, analyze clutch disc (structurally and thermally) and will give optimized design parameters and friction lining material for specific purpose. Also, other non-asbestos materials can also be compared for the particular application.

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