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Heat generation and transfer in automotive dry clutch engagement

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Abstract

Dynamic behaviour of automotive dry clutches depends on the frictional characteristics of the contact between the friction lining material, the flywheel and the pressure plate during the clutch engagement process. During engagement due to high interfacial slip and relatively high contact pressures, generated friction gives rise to contact heat, which affects the material behaviour and the associated frictional characteristics. In practice excess interfacial slipping and generated heat during torque transmission can result in wear of the lining, thermal distortion of the friction disc and reduced useful life of the clutch.

This paper provides measurement of friction lining characteristics for dry clutches for new and worn state under representative operating conditions pertaining to interfacial slipping during clutch engagement, applied contact pressures and generated temperatures. An analytical thermal partitioning network model of the clutch assembly, incorporating the flywheel, friction lining and the pressure plate is presented, based upon the principle of conservation of energy. The results of the analysis show a higher coefficient of friction for the new lining material which reduces the extent of interfacial slipping during clutch engagement, thus reducing the frictional power loss and generated interfacial heating. The generated heat is removed less efficiently from worn lining. This might be affected by different factors observed such as the reduced lining thickness and the reduction of density of the material but mainly due to poorer thermal conductivity due to the depletion of copper particles in its microstructure as the result of wear.

The study integrates frictional characteristics, microstructural composition, mechanisms of heat generation, effect of lining wear and heat transfer in a fundamental manner, an approach not hitherto reported in literature.

Keywords: Automotive clutch; Thermal network model; Clutch lining Temperature; Friction; Tribometry; Lining material properties

Nomenclature

a	Thermal partitioning coefficient
c	Specific heat

d_{air}	Density of air
F	Friction
Gz	Graetz number
k	Thermal conductivity
L	Characteristic length
N	Normal load
Nu	Nusselt number
Pr	Prandtl number
r_m	Mean radius
R	Thermal resistance
m	Mass
Re	Reynolds number
t	Time
T	Temperature
T_f	Friction torque
v_{air}	Kinematic viscosity of air
Z	Number of friction interfaces

Greek symbols

θ	Angular displacement
μ	Coefficient of friction
ω	Angular velocity

Abbreviations

FEA	Finite Element Analysis

1. Introduction

A major reason for dry clutch failure is the disproportionate heat generation via friction for prolonged periods of time [1-3]. Generated heat also changes the friction characteristics of the lining material through thermal degradation and wear, as well as distortion of the friction disc through conversion of generated heat into thermal stresses. The changes in the friction characteristics of the clutch lining can lead to a greater propensity to judder [4-7]. Distortion of the friction disc causes its uneven contact with the pressure plate and the flywheel, resulting in localised contacts, the hot-spotting phenomenon [8] and thermoelastic instability [9].

The heat is generated during the clutch engagement process as the kinetic energy converts into thermal energy. The main parameters affecting this process are the interfacial sliding velocity, the applied contact pressure (clamp load), the coefficient of friction of the contacting surfaces and the thermal properties of the materials in contact. The changes in friction characteristics with pressure and interfacial slip speed (e.g. a negative gradient of the coefficient of friction with slip speed) can then lead to coupling instabilities [10] manifested in the form of judder vibrations [4, 5] in the vehicle. Therefore, a better understanding of the effect of the aforementioned parameters during clutch engagement is essential in clutch system design for improved operational performance (e.g. noise and vibration refinement) and system reliability (avoiding wear, distortion and other failure mechanisms due to thermal stressing).

In an investigation into the thermal properties of clutch lining material Khamlichi et al [11] found that the fibre content in the matrix did not affect the temperature rise in a single engagement test, but had a significant effect in multiple engagements of the clutch. Higher fibre content in the matrix allowed heat to dissipate faster, thus avoiding high levels of thermal stressing.

Various numerical models are reported in order to study the thermal behaviour of the clutch system. El-Sherbiny and Newcomb [12] calculated the temperature distribution in the friction disc, the flywheel and the pressure plate, using a finite difference method. Different band - widths were chosen in the calculation process so that the effect of contact area could be analysed. Maximum temperatures of 100-120ºC for a single engagement and 280-300ºC for multiple engagements under severe conditions were predicted. The sever conditions involved a clutch slipping time of up to 2 seconds. No modelling based on thermal partitioning was considered, including the thermal resistances in the clutch interfacial contacts. This is an essential approach for a better estimate of prevailing conditions as shown by Olver [13] and Paouris et al [14] for the case of gear contacts and Morris et al [15] for the case of piston compression ring.

Finite element (FE) methods have been used to study the thermo-mechanical behaviour of clutch contacts. Zhao et al [16] created an FE model to study the thermo-mechanical response of a carbon-carbon dry multi-disc clutch system. Their model was used to study issues of contact mechanics, heat transfer and thermo-elastic distortion. Contact pressure and temperature distributions on the friction interfaces were obtained. For a clutch engagement time of 4 seconds, the highest temperatures were observed halfway into the engagement cycle. It was shown that increasing the thickness of the clutch friction disc reduced the peak temperature. Nevertheless, in the absence of any experimental results, the validity of the predictions could not be completely ascertained. Czel et al [17] considered ceramic clutches and compared their results with experimental measurements. Their model incorporated a heat partitioned network with equal contact temperatures at the frictional interfaces. A dedicated clutch rig was devised and measurements of the generated heat were acquired and used in the FE model for thermal loading. A relatively good agreement was found between the experimental results and the predictions, although a constant interfacial coefficient of friction was assumed. This assumption ignores the inherent dependence of the frictional behaviour of the clutch lining material on temperature, applied pressure and slip velocity.

Recently, Pisaturo and Senatore [18] investigated the frictional behaviour of dry clutches at high temperatures (250-300ºC). Their FE model predicted the temperature field during different manoeuvres. A high interfacial temperature rise of 30-35ºC was obtained even for a single engagement. Then, after only five engagements the temperature raised to 160-180ºC, which could be attributed to rather prolonged engagement times of 2.7-3.45 seconds.

It is shown that the coefficient of friction in dry clutch contact is dependent upon contact temperature [19, 20]. Due to the direct contact of clutch faces, it is almost impossible to determine the contact temperature *in-situ*. Therefore, the temperature is usually measured at a point away from the contact. However, depending on the structure of the vehicle or the test rig and due to heat losses in the transfer path, the same readings from an actual vehicle and a test rig may not entirely correlate. In addition, the effect of surface thermal phenomena such as flash temperature rise cannot be directly measured. Therefore, along with a tribodynamics model, a thermal model should also be developed to enable prediction of contact temperature, particularly when the thermal conditions in an actual clutch contact in a vehicle is intended to be simulated.

This paper presents an analytical thermal model for automotive dry clutch assembly. The model is based on the principle of conservation of energy and includes the main parts of the clutch assembly such as the contact of friction lining with the flywheel and the pressure plate. It incorporates experimentally measured coefficients of friction through pin-on-disc tribometry under real-clutch operating conditions. Temperature rise in the clutch assembly is presented for two conditions of the same friction lining material: new and worn. The importance of thermal transport properties of the friction material is noted. The current study aims to highlight the importance of thermal effects in tribodynamics analysis of the clutch, and potentially the brake systems.

2. Thermal Model

An analytical thermal model for the study of clutch system is developed based on the principle of conservation of energy. Therefore, any changes in the temperature of each component is as the result of the amount of thermal energy gain or loss as [21]:

$$m_i c_i \frac{\partial T}{\partial t} = \sum \dot{q} \tag{1}$$

The three main components of the clutch system are the flywheel (fw), friction lining (fl) and the pressure plate (pp) .There are two sources of heat generation as the result of friction: (a) the contact between the flywheel and the friction lining, (b) the contact between the friction lining and the pressure plate.

Figure 1 demonstrates a schematic representation of the studied automotive dry clutch assembly system.

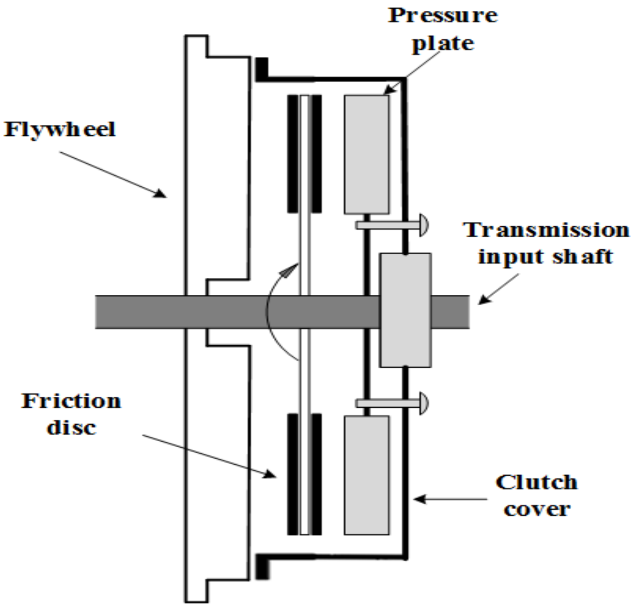


Figure 1: Schematic representation of the studied clutch assembly

Figure 2 shows the detailed developed analytical thermal network model based on the equivalent thermal resistance circuit of the clutch assembly. The generated contact heat at each of the friction lining interfaces, with the flywheel and the pressure plate, is conducted to their bodies and then either conducted through other adjacent bodies (e.g. the crankshaft, pressure plate, engine, transmission) or convected away through ambient air. Portion of the generated heat conducted to each contacting body from the contact should be determined using a heat partitioning method.

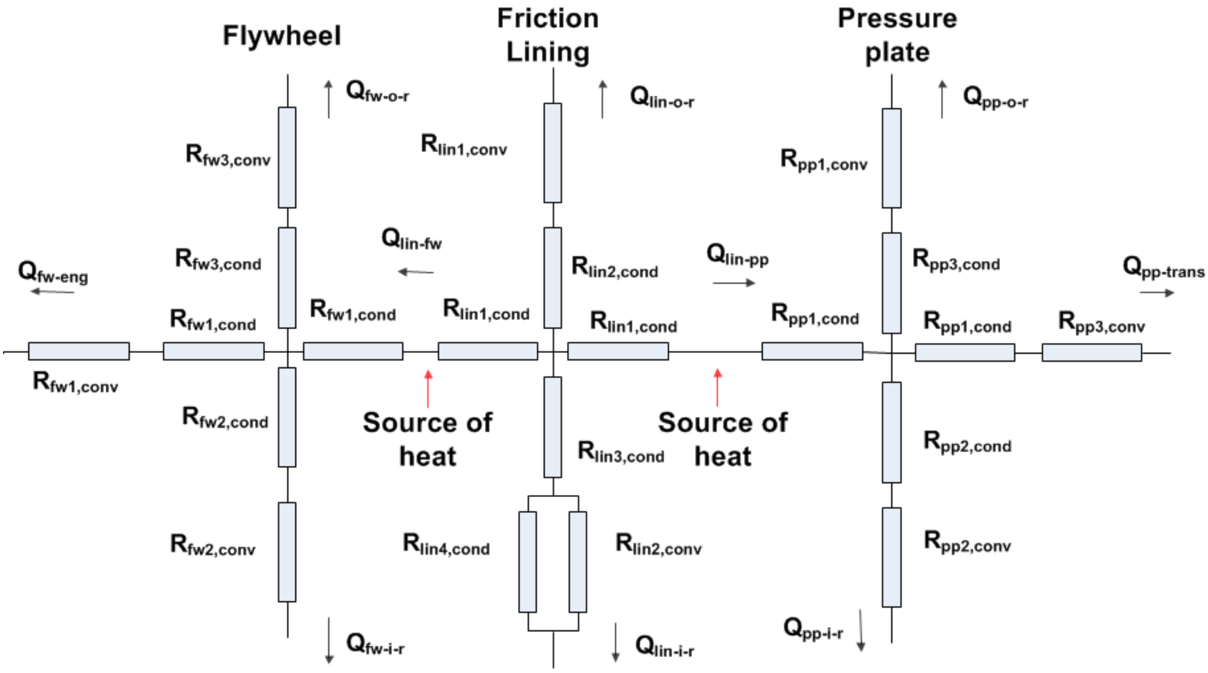


Figure 2: The equivalent thermal resistant network

2.1. Determination of generated contact heat

The generated heat in the contact between the mating surfaces (i.e. the friction lining and pressure plate and flywheel surfaces) is calculated based on the produced frictional power losses of the corresponding contacts. Assuming that all the frictional losses in the contact convert to thermal energy, the frictional power loss generated between the mating surfaces becomes [17]:

$$\dot{q} = Zfr_m\omega \tag{2}$$

where, Z is the number of friction interfaces (two for the current analysis) and r_m is the mean radius of contact. Friction on any element of contact at any position (r, θ) is obtained as:

$$f(r,\theta) = \mu(r,\theta)N(r,\theta) \tag{3}$$

where, $N(r,\theta)$ is the elemental normal applied load at any position (r,θ) as:

$$N(r,\theta) = p(r,\theta)rdrd\theta \tag{4}$$

where, $p(r,\theta)$ is the generated pressure. By assuming a uniformly applied clamp load and taking into account the contact surface area taken by grooves and rivets, the contact pressure at any point can be obtained as:

$$p(r,\theta) = \frac{W}{\pi(r_o^2 - r_i^2)} \tag{5}$$

where: $W = \iint N(r,\theta) r dr d\theta$.

Thus, elemental contact friction becomes:

$$f(r,\theta) = \mu(r,\theta) \frac{W}{\pi(r_o^2 - r_i^2)} r dr d\theta \tag{6}$$

where, $\mu(r,\theta)$ is the coefficient of friction between the mating surfaces. It is important to note that the interfacial coefficient of friction within the contact domain, in general, is not constant and varies with the relative localised slip speed of the surfaces, localised elemental applied load and the operating contact temperature.

Replacing this into the relationship given for power loss, the generated heat in the contact can be obtained as:

$$\dot{q} = Z \frac{W\omega}{\pi(r_o^2 - r_i^2)} \int_0^{2\pi} \int_{r_i}^{r_o} \mu(r,\theta) r^2 dr\, d\theta \tag{7}$$

By assuming a constant value for the instantaneous coefficient of friction for the contact, integrating of equation (7) yields:

$$\dot{q} = Z \frac{2\mu W\omega(r_o^3 - r_i^3)}{3(r_o^2 - r_i^2)} \tag{8}$$

During clutch engagement, the slip speed is assumed to reduce linearly with time (see Figure 3). The relationship for a linear reduction of the slip speed is [22, 23] :

$$\omega_s = \omega_0 \left(1 - \frac{t}{t_s}\right), \quad 0 \le t \le t_s \tag{9}$$

In the above expression, ω_0 is the angular velocity of the flywheel at the instance of initial contact and t_s is the duration of slipping.

In the current analysis, it is assumed that the sliding (relative) speed between the flywheel and the friction plate is the same for both sides of the friction disc surface.

The variations in relative angular velocity between the friction lining and flywheel/pressure plate are shown in Figure 3. The clutch engagement process as seen in Figure 3 comprises the slipping state during the engagement process and stiction in the final lock-up state.

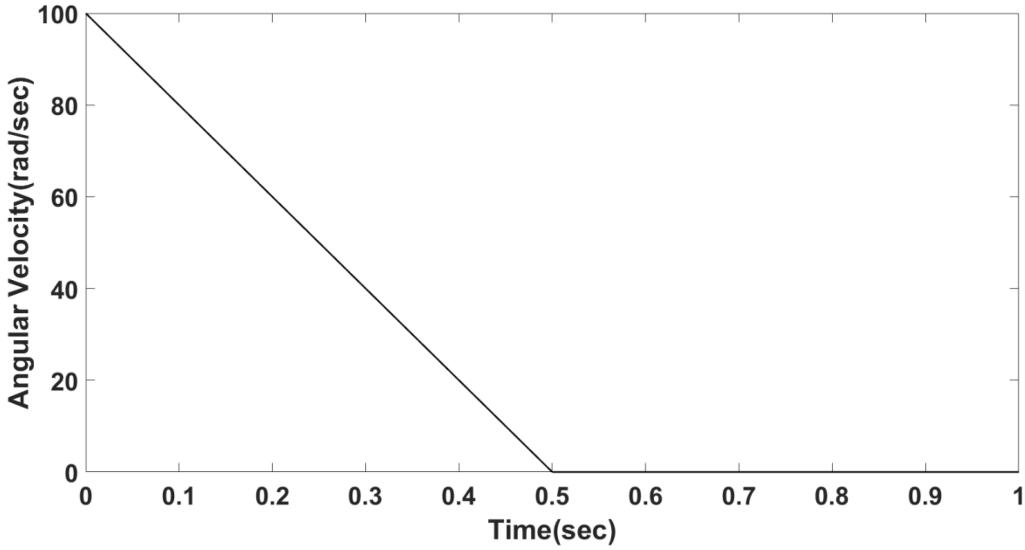


Figure 3: Variations in sliding speed during clutch engagement

2.2. Interfacial thermal balance

The heat transfer problem is analysed using a lumped parameter model (Figure 4). As already stated, the principle of conservation of energy is used for the interfaces between the flywheel and the friction lining and between the latter and the pressure plate. Two mechanisms of heat transfer are considered; conduction through the solid contacting bodies and convection through ambient air. In Figure 4, heat transfer through conduction is represented by dash-dot arrows and that through convection by dotted arrows.

Conduction takes place between the friction lining and the flywheel, and between the friction lining and the pressure plate. Furthermore, conduction heat transfer takes place between the flywheel and the crankshaft and from the friction disc through its retaining hub. Convection heat transfer occurs from all of the three contacting bodies in the clutch assembly. Convection from the flywheel surface takes place from all its exposed (non-contacting) regions facing the engine, and also on the clutch side. From the friction disc, convection occurs from its inner and outer radii beyond its contacting domain. In addition, convection occurs through the friction disc hub, splined to the transmission input shaft. In the pressure plate, convection takes place from its inner and outer radii. Finally, pressure plate convects heat from its exposed side facing the transmission housing.

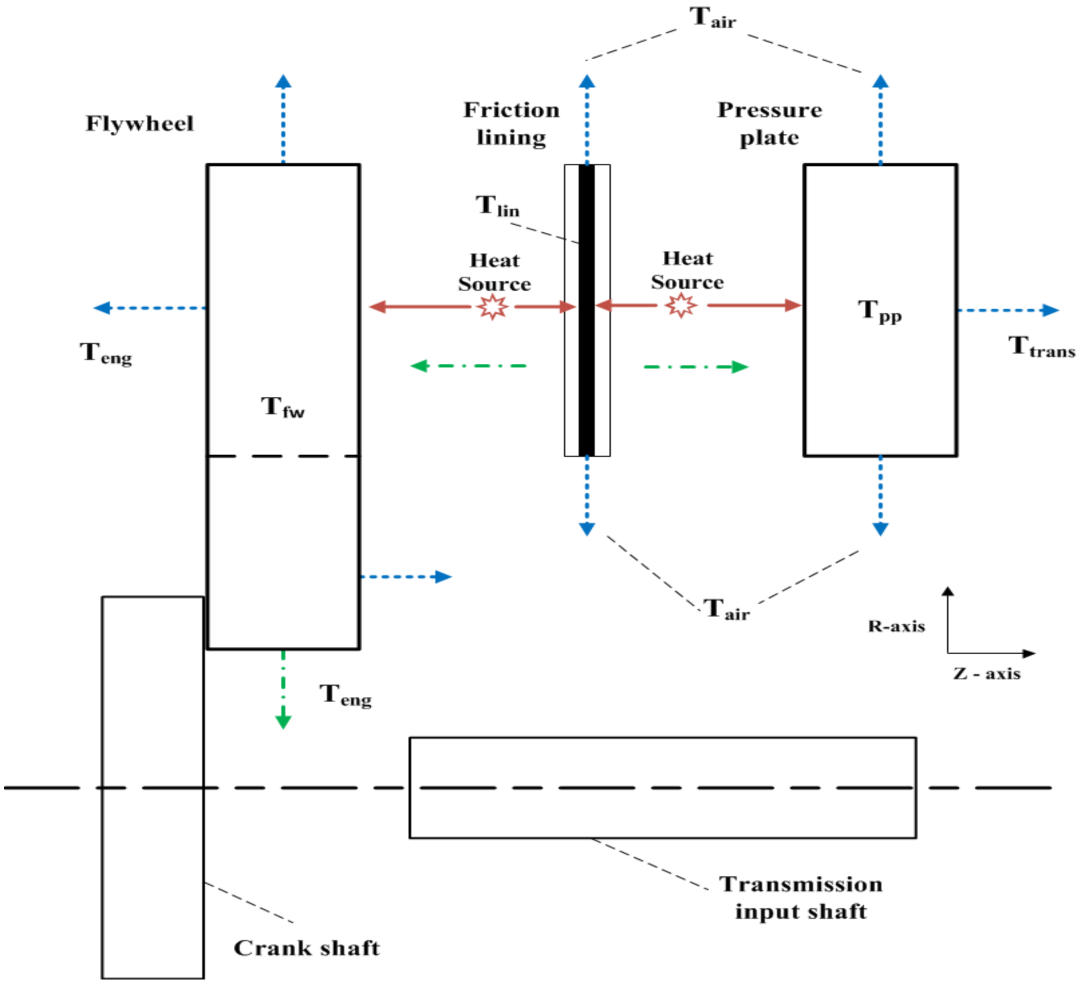


Figure 4: The clutch thermal model using lumped thermal elements

Applying equation (1) for the heat fluxes shown in Figure 4, three differential equations for each part of the main clutch assembly can be derived.

(a)- Flywheel

For the flywheel, if $\dot{q}_{tot-fw}$ represents the proportion of generated heat and transferred to it, the thermal energy balance can be written as:

$$(mc)_{fw}\frac{\partial T}{\partial t} = \dot{q}_{tot-fw} + \dot{q}_{lin-fw} - \dot{q}_{fw,i,r} - \dot{q}_{fw,o,r} - \dot{q}_{fw,eng} \tag{10}$$

where, $\dot{q}_{lin-fw}$ is the heat transferred from the friction lining to the flywheel surface, $\dot{q}_{fw,o,r}$ and $\dot{q}_{fw,i,r}$ represent the heat dissipated through the outer and inner radii of the flywheel, and $\dot{q}_{fw,eng}$ represents the heat dissipated towards the engine.

The amount of heat conducted to the flywheel from the contact area can be expressed as:

$$\dot{q}_{tot-fw} = \alpha_{fw/fl}\frac{\dot{q}_{tot}}{2} \tag{11}$$

where, $\alpha_{fw/fl}$ represents the total generated heat in the contact of the flywheel and the friction lining, apportioned to the flywheel.

(b)- Friction lining

The proportion of heat generated in the contact between the flywheel and friction lining material, transferred to the lining surface is:

$$\dot{q}_{fw/fl} = (1 - \alpha_{fw/fl}) \frac{\dot{q}_{tot}}{2} \quad (12)$$

In addition, a portion of the heat generated in the contact between the friction lining surface and the pressure plate is also conducted into the friction lining material. Assuming that the portion of the heat transferred to the friction lining is $\left(1 - \alpha_{pp/fl}\right) \dot{q}_{tot}/2$, then the corresponding thermal conduction equation can be written as:

$$\dot{q}_{pp/fl} = \left(1 - \alpha_{pp/fl}\right) \frac{\dot{q}_{tot}}{2} \quad (13)$$

where, $\dot{q}_{pp/fl}$ is the heat generated in the contact between the friction lining and the pressure plate. The thermal balance equation for the friction lining material becomes:

$$(mc)_{fl} \frac{\partial T}{\partial t} = \dot{q}_{fw/fl} + \dot{q}_{pp/fl} - \dot{q}_{lin-fw} - \dot{q}_{lin-pp} - \dot{q}_{lin,i,r} - \dot{q}_{lin,o,r} \quad (14)$$

in which, $\dot{q}_{fw/fl}$ is the heat conducted to the flywheel and $\dot{q}_{pp/fl}$ is the heat conducted to the pressure plate. $\dot{q}_{lin,i,r}$ includes the transferred heat (through conduction and convection), which is dissipated through the inner radius of the friction lining surface and $\dot{q}_{lin,o,r}$ is that dissipated through its outer radius.

(c)- Pressure plate

The proportion of heat generated in the contact between the pressure plate and friction lining material and transferred to the pressure plate is $\dot{q}_{tot-pp}$. This can be described by the thermal conductivity equation as:

$$\dot{q}_{tot-pp} = \alpha_{pp/fl} \frac{\dot{q}_{tot}}{2} \quad (15)$$

The thermal balance equation for the pressure plate can then be written as:

$$(mc)_{pp} \frac{\partial T}{\partial t} = \dot{q}_{tot-pp} + \dot{q}_{lin-pp} - \dot{q}_{pp,i,r} - \dot{q}_{pp,o,r} - \dot{q}_{pp,trans} \quad (16)$$

where, $\dot{q}_{lin-pp}$ is the heat transferred from the friction lining to the pressure plate. The heat dissipated through the outer and inner radii are denoted by $\dot{q}_{pp,o,r}$ and $\dot{q}_{pp,i,r}$. The heat dissipated from the transmission side of the pressure plate is denoted by $\dot{q}_{pp,trans}$.

2.3. Calculation of individual heat fluxes

To evaluate the heat fluxes described above, there are two main cases which need to be addressed. These are: (a) conduction between two bounding solid surfaces, and (b) convection through the interaction of a heated solid surface with any adjacent fluid (ambient air in this case). The heat transfer components in equation (10) are represented by:

$$\dot{q} = \frac{T_A - T_B}{R_{AB}} \tag{17}$$

In the general case of conduction, the thermal resistance is considered to be:

$$R_{AB} = \frac{L_{AB}}{kA_{AB}} \tag{18}$$

in which, L_{AB} is the characteristic length, k is thermal conductivity and A_{AB} is the surface area. In the general case of convection, the thermal resistance is given by:

$$R_{AB} = \frac{1}{hA_{AB}} \tag{19}$$

where, h is the convective heat transfer coefficient. The convective heat transfer coefficient over the internal radius of the friction lining and the pressure plate can be generally described as:

$$h = \frac{Nuk_{air}}{L} \tag{20}$$

where Nu is the Nusselt number, k_{air} is the thermal conductivity of air and L is the characteristic length. Flow from the internal radius of the friction lining and the pressure plate is considered to be analogous to flow over a flat plate due to the existence of a significant clearance. The characteristic length for the friction lining and the pressure plate is given by:

$$L = 2\pi r_i \tag{21}$$

where, r_i is the internal radii of the friction disc and the pressure plate.

The Nusselt number for this case becomes [24]:

$$\mathrm{Nu} = 0.664\mathrm{Re}^{1/2}\mathrm{Pr}^{1/3} \tag{22}$$

where, Re is the Reynolds number and Pr, the Prandtl number.

The Reynolds number is defined as:

$$\mathrm{Re} = \frac{\omega r_{in} L}{v} \tag{23}$$

where, ω is the angular velocity of a rotating disc (in this case the friction disc and the pressure plate) and v is the kinematic viscosity of air.

The Prandtl number is defined as:

$$\mathrm{Pr} = \frac{c_p \eta}{k_{air}} \tag{24}$$

where, c_p is the specific heat and η is the dynamic viscosity of air.

For the external radius of the flywheel, air flow is assumed to take place through a circular annulus bounded by the flywheel/clutch cover interface and the external diameter of flywheel. The Nusselt number for this case is defined as [25]:

$$\mathrm{Nu} = 3.563 + \frac{0.0668\mathrm{Gz}^{1/3}}{0.04+\mathrm{Gz}^{-2/3}} \tag{25}$$

where, Gz is Graetz number, given as:

$$\mathrm{Gz} = \mathrm{RePr}\frac{D_h}{L} \tag{26}$$

In equation (26) D_h is the hydraulic diameter of the circular annulus (equivalent tube), and L is the characteristic length as stated in equation (21). For the clutch assembly elements exhibit convective heat transfer via the external radius. Reynolds number in this case can be defined as:

$$\mathrm{Re} = \frac{\omega r_{ex} D_h}{v} \tag{27}$$

where, r_{ex} is the external radius of the rotating discs (in this case the flywheel, the friction disc and the pressure plate) and D_h is the hydraulic diameter.

Convective heat transfer in the case of the exposed flywheel side (facing the engine) is parallel to the surface of the flywheel. Experimental studies have yielded empirical expressions for the Nusselt number for such cases [26] as:

$$\mathrm{Nu} = 1.94\mathrm{Re}^{0.382} \tag{28}$$

where, the Reynolds number is defined as:

$$\mathrm{Re} = \frac{r_{fw}^2 \omega}{v} \tag{29}$$

.3. Friction and Thermal Measurements

In the dry contact of a sliding pair, such as in the interfacial contact of the clutch friction disc faces with the flywheel and the pressure plate, generated heat is due to boundary friction interactions. Boundary friction is dependent on surface topographies and materials of counter face surfaces. The interfacial coefficient of friction also varies with the operating conditions such as sliding speed, generated contact pressure and flash surface temperature.

It is difficult to accurately measure the in situ coefficient of friction in a clutch system in a repeatable manner. This is attributed to a variety of causes such as surface roughness variation, surface contamination, interfacial misalignments, to name but a few. These factors in uncontrolled testing environment contribute to accumulating errors, making for unreliable and poorly repeatable measurements.

Tribometry can be used to obtain the interfacial frictional characteristics under realistic representative conditions. In this study, a pin-on-disc tribometer is utilised. Use of pin-on-disc tribometers for the study of performance of various clutch systems has already been reported in literature [27-30]. Representative conditions in real clutch interfacial contacts are replicated in specially designed pin-on-disc tribometer [20] in line with standard testing protocols; ASTM G99 and DIN 50324. The pin-on-disc tribometer is shown in Figure 5.

The flat friction lining samples are cut and attached firmly to the contact face of the pin. The disc is made of the same steel grade as that used for the pressure plate or the flywheel, ensuring for the same surface topography in each case (through grinding and ascertained through measurement of statistical topographical parameters, using a white light interferometer). The pin with the attached lining sample is loaded onto the surface of the rotating disc. A range of loads is used, which result in representative contact pressures experienced in the interfacial contact of friction lining during the clutch engagement process. The contact sliding velocity is also altered according to linear reduction of clutch angular velocity during the engagement process. A copper cartridge heating element is built into the underside of the rotating disc to raise the temperature of its surface to representative temperatures measured from clutch interfaces under various manoeuvres. K-type thermocouples and a temperature control unit are used to maintain the desired bulk surface temperature of the disc. The thermocouples were placed in close proximity to the disc and at the same time temperatures were measured in the contact before and after the measurement.

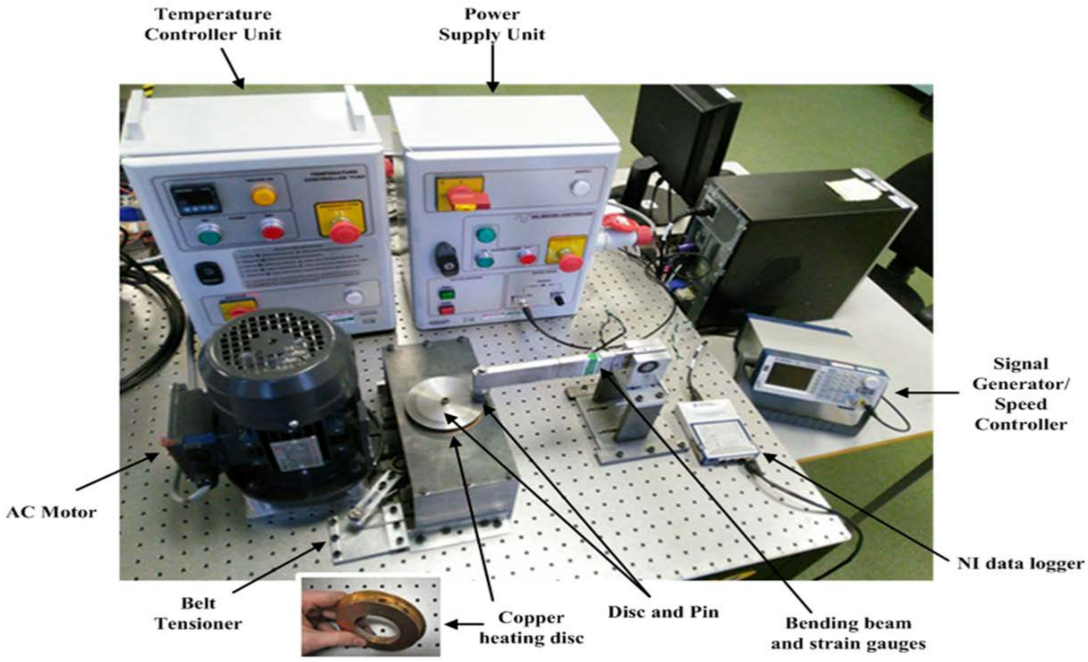


Figure 5: A fully instrumented pin-on-disc tribometer

For the thermal measurements, the Hot Disk TPS 2500 S thermal constant analyser is used to measure the desired properties as shown in Figure 6. The Hot Disk TPS 2500 S utilises the transient plane source method. A thermal conductive plane sensor acting additionally as a heat source is placed between the two samples of the material under examination. As the temperature of sensor/source increases the heat dissipates to the material at a rate, depending on its thermal transport properties.

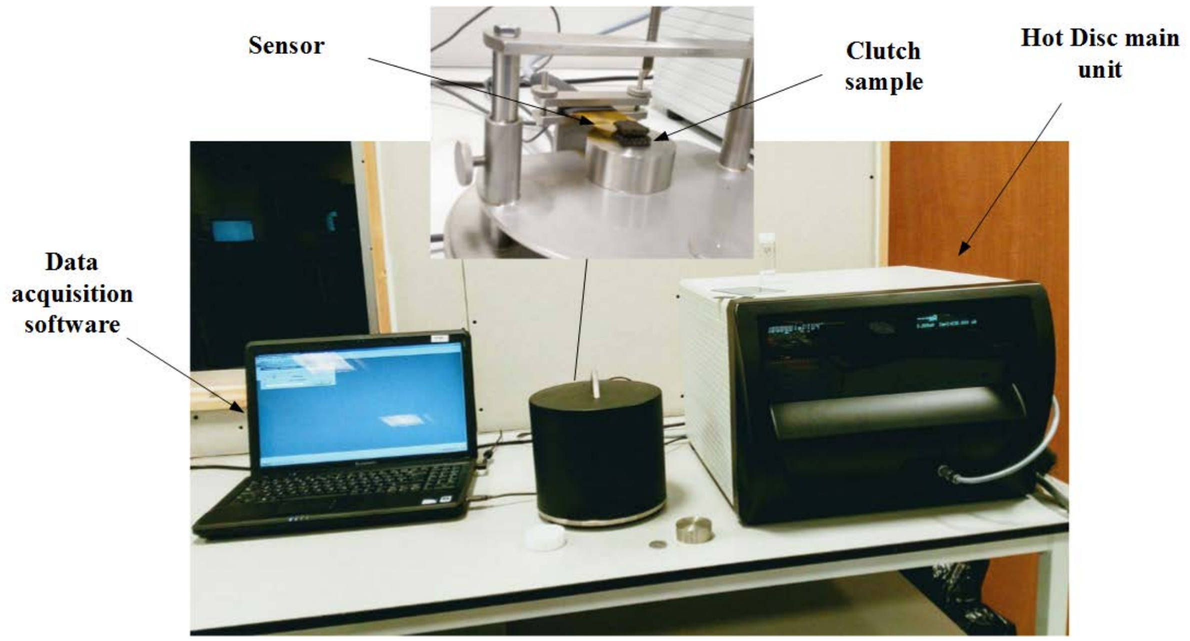


Figure 6: Hot Disk TPS 2500 S thermal conductivity analyser

The frictional and thermal measurements are performed for both cases of a new and worn friction lining material. The worn clutch lining material was extracted from a worn real clutch system after 50,000 km use. The clutch was replaced due to its poor performance. For precise measurements a strict testing protocol is followed, involving several surface cleaning steps with a number of solvents to ensure degreasing, removal of any cutting fluid residue, debris or other contaminants. These are highlighted in [19-20, 30]. The procedure also involves constant monitoring of surface topography of all the contacting pairs and replacement of the same to ensure testing repeatability in line with the aforementioned standards. All friction measurements are carried out for the same predetermined sliding distance of 300 m. For all friction measurements a new sample pair is used.

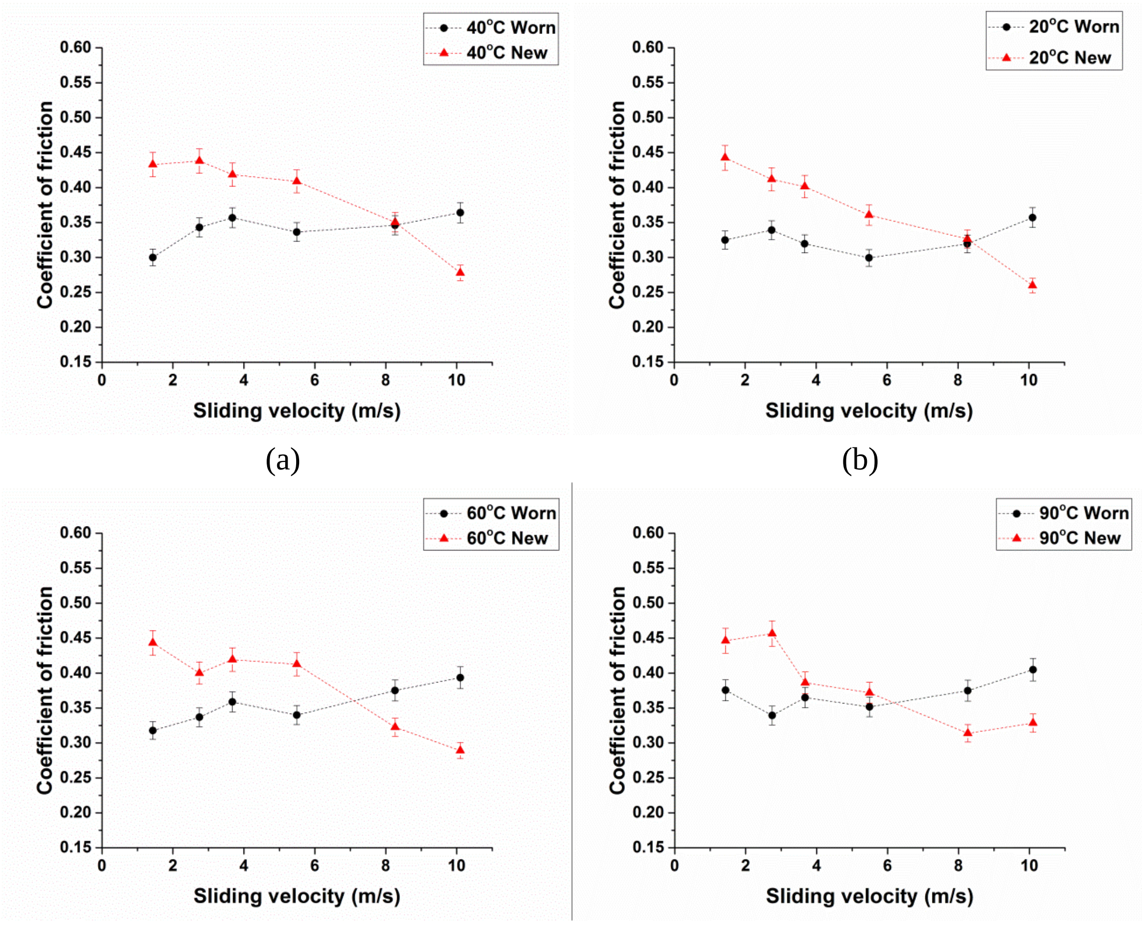
The test conditions for friction measurements are aimed to represent the real clutch applications during a typical engagement process. Contact pressures, sliding velocities and surface temperature were provided by the friction lining manufacturer and also measured in some cases by vehicle field trials. As is customary in such studies, it is assumed that the sliding velocity reduces linearly with respect to the interfacial slip time, t_s [22]. This is expressed in equation (9). In all cases, the sliding time (time of engagement process/clutch pedal actuation), t, was set to 0.5 seconds. This value represents a reasonable optimum engagement time (clutch actuation time) as also noted in [19, 29-30]. Reduced clutch actuation times can lead to loss of clamp load, which is one cause of clutch judder [5, 6]. Longer actuation times can lead to prolonged clutch slip, leading to undue wear of the lining material. Therefore, a test matrix of combinations of sliding speeds and contact pressures is created (Table 1). The table contains values for the real clutch applications and their corresponding values for the pin-on-disc tribometry.

Table 1: Comparison of clutch engagement conditions with the equivalent pin-on-disc tests

Test time (s)	**Clutch condition**		**Equivalent pin-on-disc condition**	
	Flywheel speed (m/s)	**Clamp load (N)**	**Speed (m/s) / (RPM)**	**Load (kg)**
30	10.1	400	10.1 / 2412	0.21
37	8.3	900	8.3 / 1972	0.48
55	5.5	2250	5.5 / 1311	1.12
82	3.7	3750	3.7 / 878	2.00
110	2.7	5000	2.7 / 656	2.66
210	1.4	7000	1.4 / 342	3.72
1257	0.2	10000	0.2 / 57	5.32

The initial flywheel angular velocity, ω_o, is at the idling speed of the engine, which corresponds to the transition in engine speed from idle to first gear at an initial speed of 100 rad/s. The temperature range of interest in the current investigation (20°C-90°C) is chosen, based on the recommended range by the lining manufacturer for the optimum lining performance.

Using tribometry, frictional characteristics are measured for new and worn clutch lining materials under various operating conditions. Figure 7 shows the variations of interfacial coefficient of friction with contact sliding velocity at different disc bulk temperatures. It should be noted that the contact pressure for each data point in Figure 7 is at the corresponding value encountered in a real clutch application and at the same sliding speed (see Table 1). Therefore, it is not considered as an independent variable. It is also important to note that although bulk temperature does not indicate the actual contact surface (flash) temperature, in practice the measured bulk temperature is used for the characterisation of the clutch lining materials.



(c) (d)

Figure 7: Variation of coefficient of friction with interfacial slip speed for new and worn clutch samples at various temperatures: (a) 20°C, (b) 40°C, (c) 60°C and (d) 90°C

Figure 7 also shows the various stages in the engagement process. At the instance of clutch actuation the temperature of surfaces is considered to be at the environmental temperature of 20°C (Figure 7(a)). As the engagement commences, the temperature in the contact starts to rise rather rapidly. Figures 7 (b) to (d) depict the changes in the coefficient of friction as the bulk surface temperature rises for both new and worn clutch samples. Table 2 provides a list of extrapolated equations through regression analysis for the variations of coefficient of friction with speed at different operating temperatures. Each temperature relates to different stages of the clutch engagement process. It must be noted that the coefficient of friction also alters with the sliding speed and applied contact pressure (corresponding to the rising clutch clamp load).

Table 2: Variation of coefficient of friction for the new and worn clutch lining material with slip speed, v, at different temperatures

Temperature	**Extrapolated equation for new friction material**	**Extrapolated equation for worn friction material**
20℃	$0.48 - 0.021v$	$0.32 - 0.00189v$
40℃	$0.49 - 0.0191v$	$0.32 - 0.00452v$

60℃	$0.48 - 0.0178v$	$0.31 - 0.00761v$
90℃	$0.47 - 0.0167v$	$0.34 - 0.00436v$

In Table 2 the coefficient of friction has the form: $\mu = \mu_s + mv$, where the intercept μ_s is the static coefficient of friction (fully clamped clutch condition) and m depicts the gradient of kinetic coefficient of friction with slip speed. This is an important parameter as Centea et al [5, 6] have shown that a negative slope (i.e. $m < 0$) increases the propensity of the clutch to judder during the engagement process. The results also show that the static coefficient of friction for both new and worn lining material does not vary significantly with bulk temperature, although there is a difference between these. This confirms the inferior frictional characteristics of the worn material. Through analysis of the slope of the kinetic coefficient of friction for both material conditions, it can be seen that for the new material the slope is temperature dependent and decreases with a rise in temperature. Therefore, there is a greater tendency for clutch judder at higher operating conditions. However, for the worn lining, the slope does not alter appreciably with bulk temperature. Thus, the new lining material is more responsive to the operational conditions such as with slip speed, applied load and temperature.

Thermal properties of the clutch lining material are measured by the transient plane source method for new and worn samples. The properties measured are thermal conductivity, specific heat capacity and thermal diffusivity. These are listed in Table 3.

Table 3: Friction lining thermal properties for new and worn clutch lining material

Material Thermal Properties	**Condition**	
	New	**Worn**
Thermal conductivity ($\boldsymbol{W/mK}$)	0.652 ± 0.13	0.353 ± 0.17
Specific heat ($\boldsymbol{J/kgK}$)	542 ± 35	614 ± 39
Thermal diffusivity ($\boldsymbol{m^2/s}$)	$8.3 \times 10^{-7} \pm 0.3 \times 10^{-7}$	$4.6 \times 10^{-7} \pm 0.35 \times 10^{-7}$

The results in Table 3 show that significant differences exist between the new and worn (after 50,000 km use) friction lining material. A 46% reduction in the thermal conductivity is in line with the microscopic observations in Figure 8, showing the depletion of copper particles, which enhance heat removal, on the surface of the worn specimen.

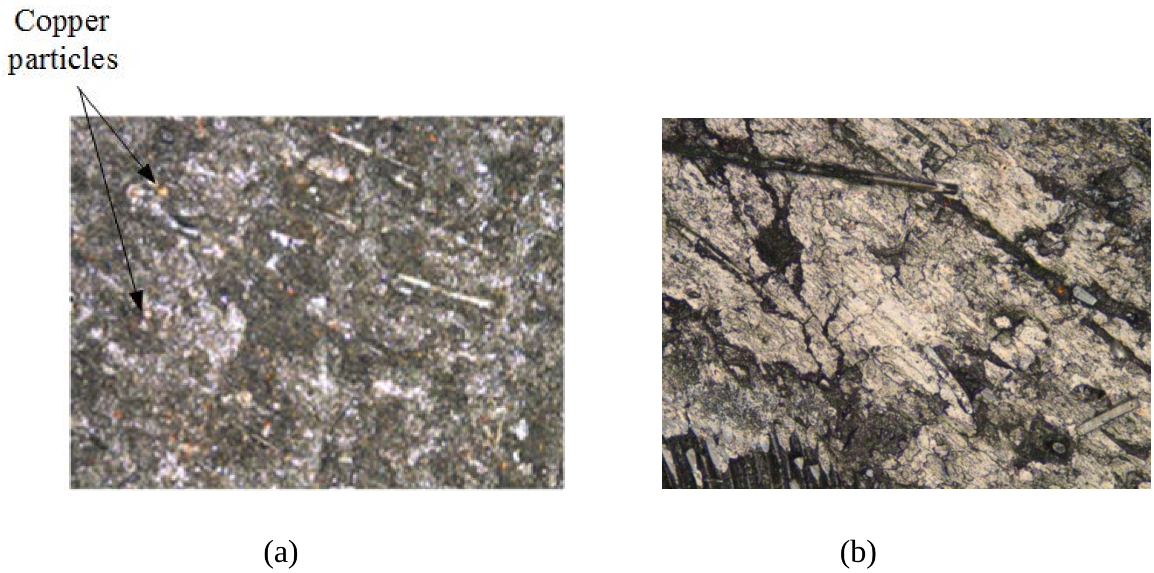


Figure 8: Microscopic images of the (a) new and (b) worn clutch lining materials

4. Predictions of the Analytical Thermal Model

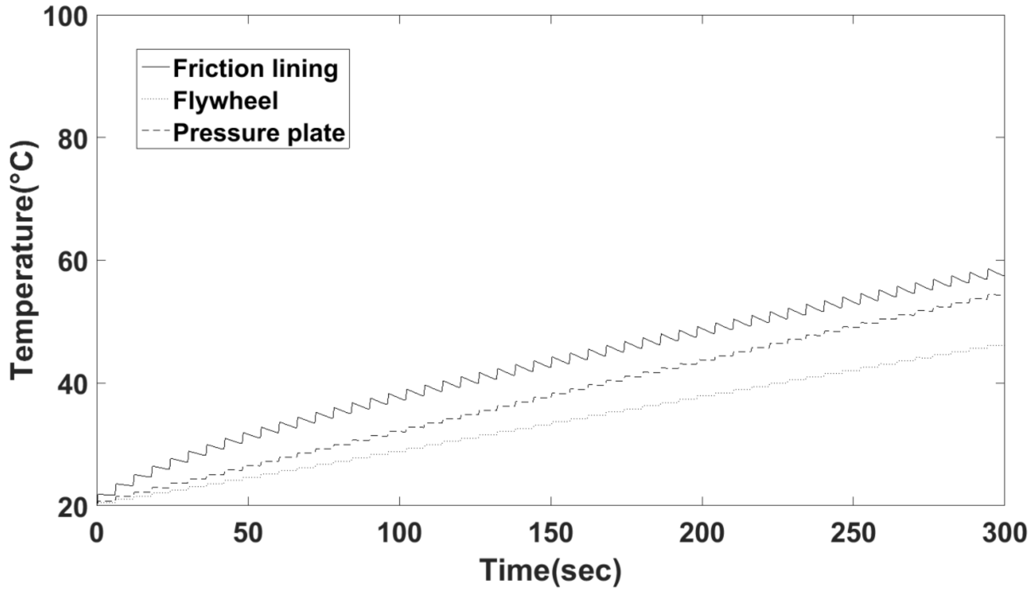
For the numerical simulations two cases of clutch engagement are considered. In both these cases constant length cycles of engagement and disengagement occur for a total duration of 300s.

Case 1: This case corresponds to 0.5s engagement time, followed by a dwell time of 5s in the fully clamped state.

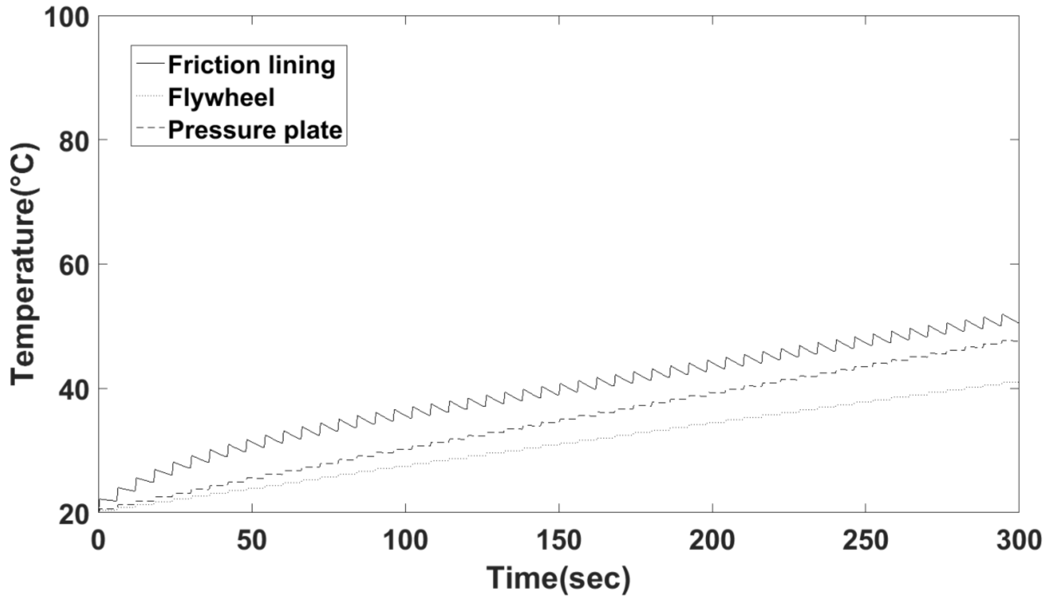
Case 2: In this case the engagement time is increased to 1s, representative of prolonged slipping during clutch actuation. The aim is to obtain an understanding of how the clutch assembly behaves under rather extreme conditions.

For this study, the disengagement period is assumed to be instantaneous. The above cases are performed for both the new and worn dry clutch materials. Furthermore, several actuation times are considered to produce an actuation time-temperature graph which demonstrates the relationship between the clutch actuation time and temperature rise in the friction disc.

Figures 9a and b show the variations in the interfacial temperature for new and worn friction lining materials based on the aforementioned Case1 study. As it can be seen in Figure 9a, the bulk temperature of the lining material reaches a value around 70°C for constant engagements with a realistic engagement time. Pressure plate reaches a temperature around 65°C, whilst the flywheel temperature ascends to 55°C. The difference in temperature between the pressure plate and the flywheel surfaces is because of the exposed free surface geometry of the flywheel facing the engine, leading to a greater convective heat transfer.



(a)

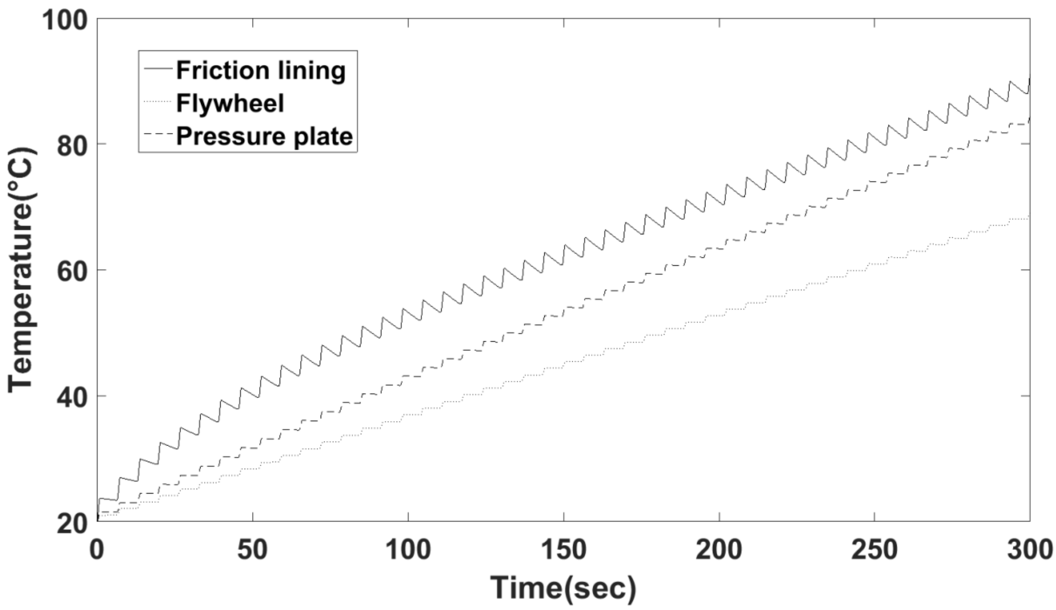


(b)

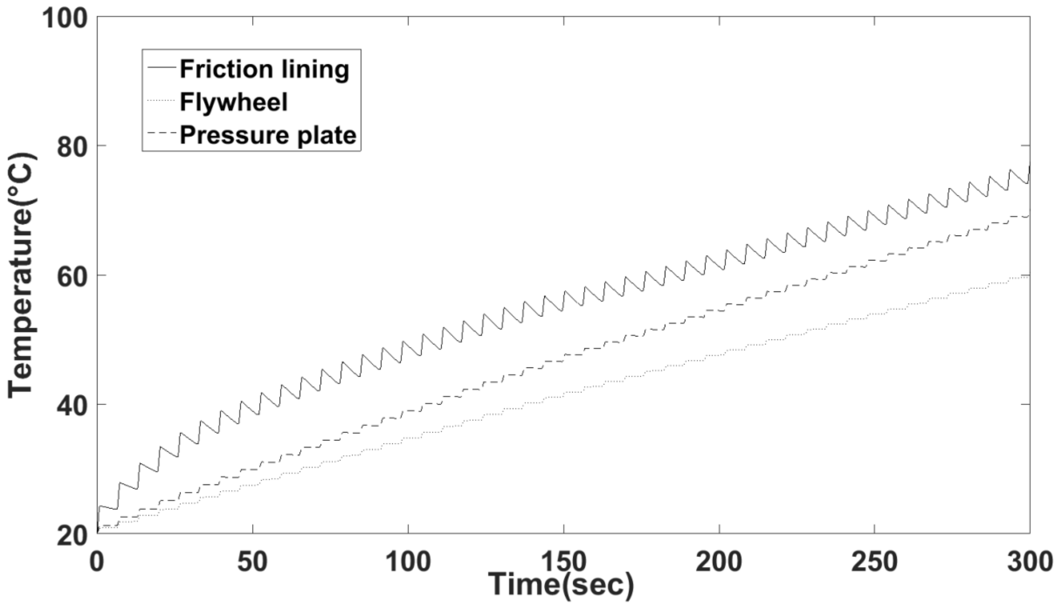
Figure 9: Temperature rise in constant engagement for the clutch system with (a) new and (b) worn friction lining materials (Case 1)

The results in Figure 10 correspond to the aforementioned Case 2 study, where the clutch system is subjected to prolonged slipping at the interfacial contacts owing to a longer clutch engagement. As it would be expected, the final attained temperatures nearly double in value, compared with the previous case, for both the clutch linings; new and worn.

For the worn friction material and with the prolonged engagement times (Figure 10b) the trend observed is similar to the previous case with only a difference in the final temperature reached. The observed temperature difference is around 30°C between the 0.5s engagement and that with a 1s engagement time for the new clutch material. A similar difference is also noted for the temperature difference for the pressure plate and the flywheel in both cases. Similar temperature differences are also observed when using the worn friction lining material.



(a)



(b)

Figure 10: Temperature rise due to a constant engagement for (a) new and (b) worn friction lining materials with prolonged engagement (Case 2)

Figure 11 shows the variation of friction lining temperature with engagement time for both new and worn states. The results show that the temperature of the new friction material remains higher than that of the worn case. This is because of higher generated friction with the latter, thus a greater generated heat. Additionally, it can also be observed that the temperature difference tends to increase with the engagement time, indicating the susceptibility of the new lining material to prolonged clutch activation. The lining material is designed to withstand engagement times from 0.15 to 7s. However, the optimum clutch engagement time would be around 0.5s. It is clear that engagement times exceeding 0.5s leads to excessive interfacial slipping and a higher generated friction which increases the lining temperature. Increased friction and generated heat can result in undue wear rate as well as friction disc distortion, noted in practice. Shorter clutch engagement times, as in hurried release of the clutch pedal, result in some degree of loss of clamp load, thus reduced pressure loading. One result of this is reduced friction. In practice, this leads to clutch take-up judder [4-7], even though reduced interfacial temperature would be advantageous in terms of wear and durability.

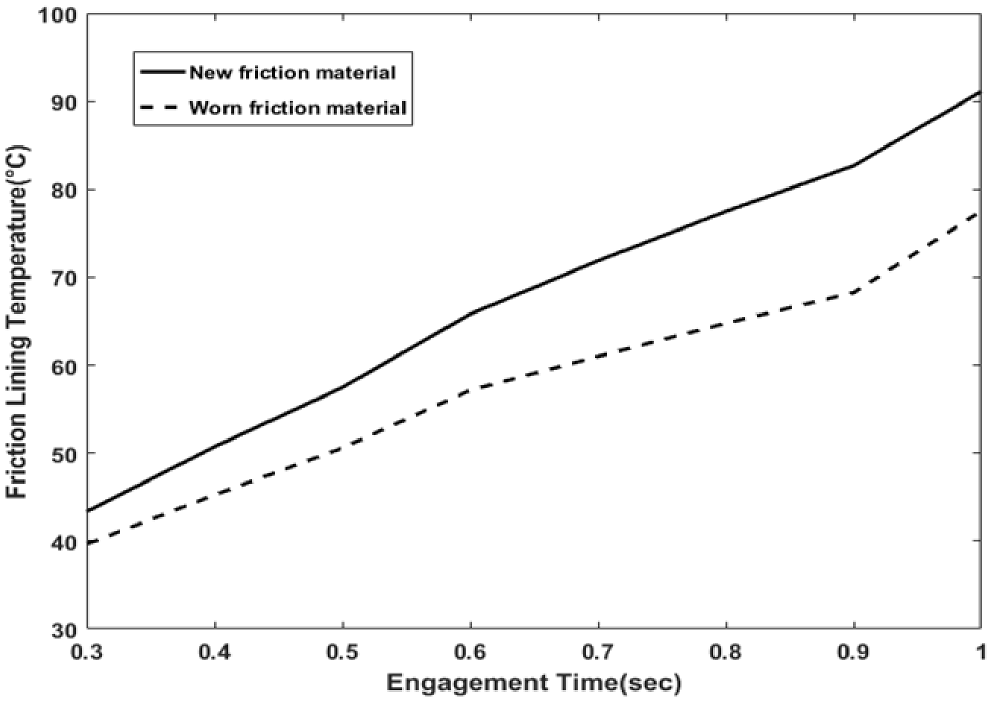


Figure 11: Temperature rise of friction lining materials (new and worn) for different clutch engagement times

5. Conclusions

The study presents an analytic thermal model based on the principle of conservation of energy between interfacial frictional losses and generated heat for the dry clutch assembly. The clutch friction material under investigation is tested under two conditions; new and worn state (after 50,000 km use). Experimental measurements of the coefficient of friction and the thermal properties of these lining states are used in an analytical thermal network model to study the evolution of temperature in the clutch assembly during various clutch engagements, typical of cases of driver behaviour. Case1 is typical of the engagement time for which the lining material is designed. Case 2 is for prolonged clutch engagement time, leading to excess interfacial slipping, thus increased friction and heat generation.

Experimentally measured $\mu - v$ characteristics show the temperature-sensitive nature of the new friction lining material, whilst the same is not noted for the worn lining. A microstructural study of the worn lining indicates depletion of copper particles from the lattice structure, thus the reduced ability of the worn lining in conductive heat transfer. This is also established through measurement of thermal conductivity of the lining material variants used in this study.

The results of measurements using the tribometer show a higher coefficient of friction for the new lining material than the worn variety. This reduces the extent of interfacial slipping during clutch engagement, thus reduced frictional power loss through conversion into generated heat. Furthermore, the generated heat is removed from the new lining material due to its superior thermal conductivity, as shown by the developed thermal model.

Finally, the study integrates frictional characteristics, microstructural composition, mechanisms of heat generation, effect of lining wear and heat transfer in a fundamental manner, an approach not hitherto reported in literature.

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