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ORIGINAL ARTICLE

Wear Based Lifetime Estimation of a Clutch Facing using Coupled Field Analysis

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ABSTRACT – Repetitive use of the clutch, over a period of time, causes the friction material at the contact surfaces (clutch facing and flywheel/pressure plate) to wear, thus deteriorating its performance and usable life. The working life of a rigid clutch is the limiting factor when it comes to extracting maximum performance from a dual mass flywheel system, which is used in a lot of modern vehicles nowadays to lower fuel consumption and improve ride quality. In this study, we investigate the influence of different groove patterns on wear in rigid clutch facings and estimate their life using a comprehensive finite element model. The wear is calculated and analysed for five different groove patterns across two different inorganic materials, namely FTL180 and TF1600-MC2, using Archard's Adhesive Wear Model. Coupled multi-physics elements are employed in the analysis to capture the effect of frictional heat generation on wear. We found that the Waffle pattern offered a decrease of 10.4% in volumetric wear loss, a 5.78% decrease in maximum wear thickness and an increase of 11.51% in the average working life is used in city like conditions with frequent engagements. This work sheds light on the impact of groove patterns on clutch facing wear and opens a new path for the design and development of more resilient rigid clutches.

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INTRODUCTION

At present, modern engines generate relatively higher torques when they are operated at low engine speeds. This is attributed to higher magnitudes of noise and vibrations at lower engine speeds. The dual-mass flywheel (DMF) facilitates the driving in an operational range that offers optimised fuel consumption with a high level of ride and noise comfort along with reduced CO_2 emissions [1]. In addition, DMF helps attain the operational smoothness much needed for the modern economic engines. The technological advantages such as fuel savings and emission benefits of this system will not let the automobile manufacturers discard their presence in future automobiles. Despite several distinct advantages, higher cost and more complicated assembly of DMF system are main drawbacks. Furthermore, it cannot be rebuilt or even resurfaced to eliminate hotspots and worn areas, thus necessitating a complete replacement. Most manufacturers recommended that DMF must be replaced along with the clutch. This, in turn, results in a considerable increase in replacement cost compared to traditional flywheel and clutch replacement.

The average service life for the majority of DMFs is up to 150000 miles, while the stock/standard clutch facing's life is in the range of 60000-100000 miles which is further dependent upon operating conditions, drivers' usage and regular maintenance [2]. A key symptom of wear in this system is a clutch slip or a clutch facing failure. The clutch plate should last as long as the DMF to maximise the system efficiency, but the stock/standard clutch plates failed to do so. The faulty clutches should be patched as soon as possible to prevent harm to the system. The repetition of the same problem will lead to consumer dissatisfaction and costlier warranty claims. The high investment cost incurred during the purchase/replacement of DMF is essentially recovered in approximately a year's worth of cruising by reducing the fuel consumption by about 21% [3]. This also implies that there will be a significant reduction in CO_2 emissions per km and thus improving the ecological footprint of the vehicle.

Modern gearboxes incorporating DMF are seeing increased use of rigid clutch discs because of their particularly designed cushion deflection and cost-effectiveness. For these clutches, the friction materials are made to have a high burst strength, which stays consistent for an extended working range while still being slip-free and smooth. These particular composite materials were designed with several specific properties, such as several reinforcing fibres, wear-resistant binders, and lubricants. Thermoelastic instability (TEI) and thermal buckling may develop during sliding contact between friction pairs when the operating speed or temperature reaches specific threshold levels [4]. There are hotspots because the pressure across the friction plate is not consistent. They reduce the life of the friction lining on top of everything else, further cutting down on the lifetime of the lining. It is vital to know the temperature distribution as well as the highest temperatures obtained in operation to prevent the clutch failure before the estimated lifespan. These numbers vary based on different factors, viz., the varying loads, the qualities of the clutch, the clutch plate size, and the degree of air cooling.

A significant proportion of the driving range is covered at the clutch temperatures of 100°C or below, while it may peak to 400°C under heavy vehicle load, which then results in an exponential increase of wear—a high quality facing exhibits low wear rate in a broad spectrum of time and over different temperatures. A high clamp load for an average

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clutch size would lead to higher specific facing unit pressure. Under this condition, the clutch facing temperatures rise quickly enough to deteriorate the base material and reduce its frictional coefficient. The lower thermal conductivity of a facing material initially causes surface damage. Thus, a clutch facing should be designed to withstand all the high-load scenarios without significantly reducing component life. Rigid clutch discs in modern transmissions have seen the increasing use of DMF systems due to their specially tuned cushion deflection and cost-effectiveness [4]. Friction materials in those clutches are designed to have high burst strength, constant frictional coefficient over a wide operating range, smooth and slip-free engagement. These requirements are met by employing robust multiphase composites consisting of the binder resin, reinforcing fibres, solid lubricants, abrasives, wear-resistant space fillers, and other friction modifiers [4].

Abdullah and Schlattmann [5] used Lagrange and penalty contact algorithms to analyse the engagement period of the clutches. Their study on contact pressure and penetration using the above algorithms allowed them to demonstrate the significance of the contact stiffness in-between the frictional contact areas of the clutch components. These points were accounted for based on their results on variations in contact pressure, penetration, and contact stresses. Vadiraj [6] conducted a brief investigation on the engagement characteristics and failure analysis of friction pad and pressure plate materials tested on a test track in a commercial vehicle. This study developed a correlation based on energy dissipated and frictional wear loss during clutch engagement to determine the clutch facing's useful working life in terms of the total number of engagements. The clutch engagements were thoroughly tested in the dynamometer to understand their features better and predict their useful life in terms of engagements.

Finite element analysis of the temperature distribution around radial, circumferential grooves on a dry friction clutch disc is presented by Abdullah and Schlattmann [7]. The radial, circumferential grooves were looked at, as well as the additional friction clutches with and without those grooves. To determine maximum temperature and total heat transported by convection, a variety of groove area ratios were evaluated. In addition, they discovered that the groove area ratio relates to the quantity of energy transported by convection. Different groove ratios were expressed in terms of changes in internal energy over time. To confirm the experimental results, the thermal behaviour of a dry ceramic disc was modelled by connecting two separate FE models, which each took place in time and space, according to the study by Czel et al. [8]. Thermal modelling like this would be used to modify the clutch components to capture as much thermal energy as possible and allow for efficient heat flow. Friction material, rotational velocity, and pressure imposed by the pressure plate were varied to evaluate temperature distribution at a single engagement in a clutch simulation that was carried out by Mouffak et al. [9]. The thermal and material behaviour of clutch facing was examined using models with zero, four, and eight grooves. Minimising frictional energy due to convection and conduction was shown to be a crucial component in improving lifespan.

According to Jabbar et al. [10], frictional heat created between contact surfaces during the slipping has a significant impact on the thermal behaviour of dry friction clutches. The thermal and thermoelastic assessments of friction clutch systems were completely reviewed by the authors. Gkinis et al. [11] tested the dry clutches under representative working conditions that include the interfacial slippage during the engagement, applied contact pressure and generated temperatures. The test results revealed that the higher coefficient of friction associated with the new lining material minimises the amount of frictional power loss and interfacial heating generated during the clutch engagement. A single-plate dry clutch utilised in passenger cars, trucks, and buses was modelled in a 1-D thermal model by Paulraj and co-author [12]. Through simulation, they estimated the temperature increase on the clutch facing and clutch housing. This was used to predict clutch life at an early design stage, in conjunction with data provided by the supplier (facing temperature vs facing wear ratio). In both rapid and routine vehicle testing, clutch life/wear can be predicted using clutch facing temperature.

Bao et al. [13] determined that three different groove-shaped friction pairs wear at various rates based on their calculations. To acquire the actual wear quantity and evaluate the accuracy and validity of the mathematical model, the SAE#2 testing equipment was used. There was a significant difference in wear between the three types of grooved discs with waffle grooves and those with three-way parallel grooves. Two-way parallel grooves design was ascertained as the best, and the wear from one-time contact can better reflect changes in friction pair dynamics.

The most common problems that cause clutch slippage are worn friction material and inadequate friction or clamp load. To minimise these problems, clutch facings should withstand high loads and any severe events without compromising any significant decrease in a lifetime. Due to more powerful engines with smaller installation space requirements, these engines, in turn, increase power density for the clutch facings. The researcher's control strategies lead to conflicting objectives such as small facing wear, small friction losses, the minimum time needed for the engagement, and regulation of the slip acceleration at the lock-up point for reducing the undesirable driveline oscillation [14]. A multi-faceted study analysing the dynamic performance of a clutch such as the one presented in this paper is not yet available online in a detailed form. Research on empirical relations for variation of contact surface wear, wear thickness, and other clutch design parameters are lacking in the literature. The present literature does not clarify the function of the specific groove design on the friction properties of the dry clutch and a large majority of current studies focused on the chemical composition, wear properties and friction of materials used mostly in a wet clutch. Extensive research has not yet been carried out on the dry clutch and comparative material.

Wear in plastic contact primarily takes place via two modes: abrasive wear and adhesive wear. In a clutch system [25], wear is predominant due to the presence of high contact pressures. In 1953, Holm and Archard [26] developed a simple and useful wear equation that models the slow progressive wear of the abrasive grains grinding against a softer material as a function of the normal load, the real area of contact, and the sliding distance. It proposes that the asperities on the contact surface are spherical. During sliding, a certain fraction of these asperities deforms plastically under high local contact pressures and fractures. The differential form of Archard's Wear equation is especially helpful for calculating wear at local nodes provided contact pressure is known, thus making it easy to implement in FEA. In the majority of the papers, there is no wear effect considered in the transient analysis to get combined results. But clutch wear is an influential factor for the performance of the automotive clutch. Collective interactions between contact pressure, temperature, and wear need to be considered in the problem. Consequently, an analysis of the Archard wear model is also used to obtain the optimal findings for analysis. Here, wear is analytically studied in clutch facing with varying groove patterns. We compare the results across two inorganic materials developed by Friction Technology Limited (FTL-180) and ThermoFiber (TF 16000-MC2) and show similar trends in properties based on Archard's interpretation within a range of applied load. We finally provide possible explanations for why this is the case and potential improvements to the proposed models.

A lot of studies have been conducted to investigate the issues concerning the relationship of dynamics, contact phenomena, and accompanying tribological processes observed in multiple friction contacts in transmissions, bearings, clutches, brakes, and so on. The studies performed on the systems with frictional contact have resulted in numerous mathematical models to predict the dynamic behaviour of the components. The literature survey performed by the authors made them conclude that wear mechanisms are often neglected in those mathematical models, the ecological impact and working life of the components have hitherto not been undertaken. Based on the literature gaps found, the objectives of the present study have been formulated. The objectives of the present study would be (i) to identify the optimal groove pattern for clutch facing, which minimises the temperature and wear under full engagement (ii) analyse and compare the performance of inorganic materials used in modern clutches (iii) graphical representative trends between the dynamic behaviour of the clutch and the various design parameters.

DESIGN APPROACH

The clutch lining is one of the highly stressed power transmission elements, which is riveted to the clutch disc and produces an initial slipping. This is then followed by an adhesive friction mechanism in combination with the clutch pressure plate and flywheel. If the torsional vibrations in the powertrain are minimised by utilising a DMF, the clutch discs can be used with or without a torsional damper. To meet the highest criteria for comfort, often, the combination of DMF and clutch disc with a single-stage torsion damper is used. The most cost-effective option is rigid clutch discs or clutch discs with offset correction for lower requirements. These discs have specially tuned cushion deflection and are used for vehicles with a DMF. Due to their lighter weight and simpler geometry, they provide smooth build-up of torque when starting and ensure safer torque transmission through partial compensation of the temperature deformation of the flywheel and pressure plate. The following points are key goals to design a clutch facing for required application: (i) lifetime, and operational robustness, (ii) lower weight and cost, (iii) increased torque capacity and lower torsional vibrations, (iv) lower slip rate, and (v) efficient cooling and thermal stability. Figure 1 shows the dimensions of the clutch assembly being investigated.

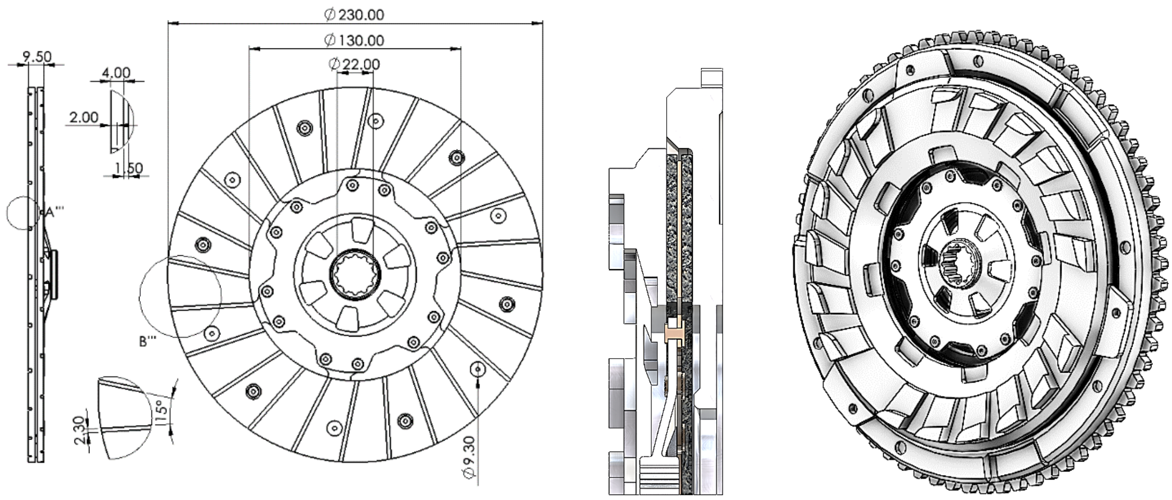


Figure 1. Dimensions and features of the clutch geometry.

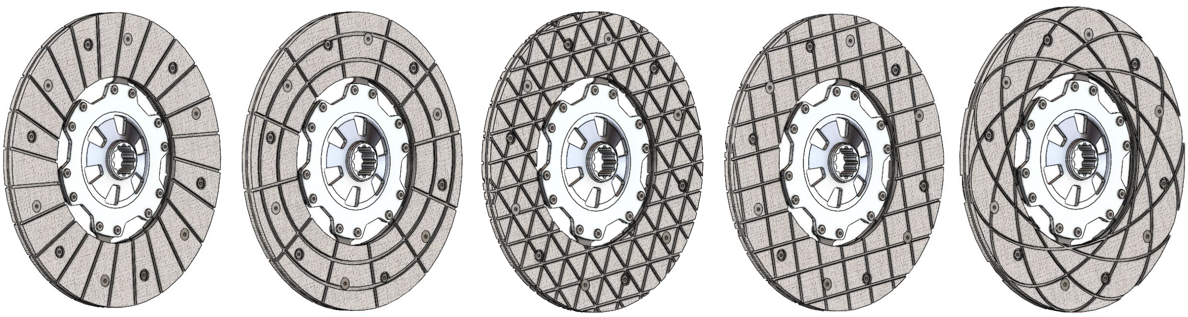
The regulations of Indian Standards (IS 3649) for the clutch facing design are adopted for the design variants investigated in this research. The specified standard covers the requirements of an automotive clutch facing for single plate and multi-plate applications under dry conditions [15]. The operational robustness of a clutch facing is proportional to its temperature robustness [16]. Mouffak and Bouchetara determined that at the end of the slip time, increasing the number of friction surfaces of the disc linings greatly reduces the temperature level and thus reduces thermal stress [9].

The clutch torque capacity is directly proportional to the clamping load, frictional coefficient of the material, mean effective radius of the clutch facing, and no. of clutch discs (as required). Thus, a clutch with a lower mean effective radius results in lower clamping force due to smaller diaphragm springs. These two factors do not significantly reduce the torque capacity, but adding an extra clutch disc doubles the torque capacity. Higher clamping loads can induce higher pressure loads on crankshaft thrust bearings. Also, an increase in repeated engagements led to higher surface wear which can result in loss of life. The wear behaviour of a clutch facing is strongly dependent on its temperature, friction power, and clamp force. The temperature variation during the vehicle operation influences the cushion spring compression behaviour due to frictional heat generation of the clutch facings with the flywheel and pressure plate surfaces during the engagement phase. The facings are subjected to a wide range of centrifugal forces and temperatures during operation. Thus, the clutch facing should have the higher structural integrity to operate under all conditions. Brust speed and tensile strength are important in maintaining structural stability, but irregular higher temperatures affect the facing strength. The clutch facing must possess high material strength to withstand thermal stresses even after its thermal limit, thus ensuring better thermal stability. The design parameters of the clutch are presented in Table 1.

Table 1. Clutch design parameters.

Parameters	Value
Number of friction surfaces	2
Service factor (b)	1.4
Mean radius of friction disc (mm)	89
Clutch friction plate outer radius (mm)	115
Clutch friction plate inner radius (mm)	65
Thickness of clutch plate (mm)	9.5
Thickness of clutch facing (mm)	4
Thickness of the axial cushion (mm)	1.5
Number of rivets	12
Outer diameter of rivet (mm)	9.3
Inner diameter of rivet (mm)	4.7
Head Depth (mm)	1.25
Number of Splines	12
Maximum angular slipping speed, ω_o (rad/sec)	104.72
Slipping Period, t_s (sec)	0.5
Initial temperature, T_i (°C)	25

Thus, to maximise the heat dissipation, friction, and grip to the DMF, a wide range of design variations in the groove area could be followed. The different designs adopted are presented in Figure 2. It is also employed as a depository of waste or metal chips from the flywheel or clutch and has many perforated holes. The clutch hub riveting to the central plate is the driving shaft slot. They are designed to handle structural loads and thermal stress and are tailored to reduce weight and moment of inertia. For greater heat dissipation, the S/V ratio of the clutch hub is optimised [4]. In the uniform wear hypothesis, the parameters are specified by a service factor 1.4. In an application where the engine torque is between 270 and 370 Nm, the considered clutch system is used [4]. This clutch system may generally be utilised in combination with 1.5-1.8 L engines and 6-speed DCT and Automated manual transmission (AMT) with DMF.



(a) Stock (b) Ellipse (c) Trigonal (d) Waffle (e) Sunburst

Figure 2. Illustration of the developed groove patterns.

MATERIALS

For the selection of clutch-type (with or without damper) and clutch material (organic or inorganic) for the intended application, it is important to consider necessary factors: (i) wear characteristics, (ii) operating temperature range, (iii) facing design, (iv) clutch feel, (v) strength. The materials considered in this work are used only to analyse the facing design and effect of groove pattern design and area on its structural and thermal performance. Two inorganic paper friction materials, namely FTL180 and TF1600-MC2, are considered in this study. Both have great performance, high friction, and a high proportion of aramid fibre, which are non-metal composites. They may be seen in sintered metal materials as an alternative and provide several advantages. They are resistant to large energy inputs and are ideal for dry and oil applications. They are not abrasive to the counter material, are silent in operation, and resist high pressures. The wear rate is consistent even at high temperatures. The recommended mating surface for both is fine-grained, pearlitic cast iron or cold-rolled steel with a Brinell hardness between 150 – 200. In terms of the bonding material used, it is necessary to use a thermosetting adhesive to obtain the best results. Also, according to the manufacturer's guidelines, no backing plate is required for most clutch applications. Organic friction facings offer a compromise between tribological and structural properties. Moreover, their properties have interactions. Despite their smooth engagement, lower tolerance to repeated engagements and a broad range of operating temperatures limited their applications in the relevant field. Organic materials bonded by resins often tend to lose friction as the temperature rises. This is because the resin melts and starts acting like a lubricant. At this point, any increase in clamp load or the clutch material coefficient of friction will simply elevate the shock load to the gears. The required material properties for the pressure plate and DMF to maximise compatibility are obtained by regulating chemical composition, cooling rate, and heat treatment of grey cast iron accordingly. There are very slight differences in each of the materials when it comes to frictional coefficient and the wear rate, as seen in the table below. The important material properties of these two inorganic clutches facing materials are presented in Table 2 from the manufacturer's site.

Table 2. Material properties [17,18].

Parameters	FTL180	TF1600MC2
Material type	Paper friction	Paper friction
Static friction coefficient, μ	0.41	0.30
Fading temperature (°C)	395	423
Wear rate (mm^3/kwh)	53	30
Dynamic friction coefficient, μ_d	0.36	0.4
Hardness (DIN53505)	82	85
Density (g/cm^3)	1.21	1.30
Thermal conductivity (W/m°K)	0.24	0.26
Poisson coefficient	0.28	0.30
Tensile strength (ASTM D638) - (N/mm^2)	72	70
Compressive strength (UNE 53205) - (N/mm^2)	308	306
Young modulus (ASTMD 638-10)- (N/mm^2)	7290	7260
Continuous operation, T_{max} (°C)	360	360
Intermittent operation, T_{max} (°C)	400	400
Max. allowable pressure (MPa)	1.034	1.034
Burst resistant (200×137×3.5) 200°C (rpm)	18180	18300

ANALYTICAL FORMULATION

A basic clutch disc with a simple outer diameter of 230 mm, facing width of 50 mm is designed according to IS 3649: 2018 regulations. The clutch virtually transforms the engine torque into heat during the slippage just before maximum engagement, and the heat generated must be dissipated into the surroundings; hence the thermal efficiency of the clutch is of considerable significance. The relative sliding between flywheel and clutch plate during engagement also results in wear in the clutch components. Based on the driver's way of operation, the impact energy of the engagement also causes vibrations leading to noise and judder. In this project, all the aforementioned phenomena are investigated with the help of FEA. The initial results are used to develop new groove patterns. The simulations are then run on these geometries to generate data sets for recording any noticeable trends in various properties. Additionally, two different inorganic materials, namely; TF1600-MC2 and FTL180, are used for each of varying geometries to reinforce the consistencies of the trends.

Frictional Torque Model

During the clutch disc design, the adoption of uniform wear theory is recommended as uniform pressure assumption gives a higher torque in most cases. Thus, a clutch disc design based on uniform pressure will result in the clutch slip when it becomes old. Assuming constant wear rate, i.e., pr = C, the elemental clamping force on clutch facing is given by Eq. (1).

$$\boldsymbol{\delta W} = 2\pi r p \delta r \tag{1}$$

$$\boldsymbol{\delta W} = 2\pi C \delta r \tag{2}$$

where, $\boldsymbol{W}$ – clamp load, C – wear rate (constant), p – pressure, r - radius. Integrating the element clamping force will give the total clamping force W as in Eq. (3).

$$\int_{\mathbf{0}}^{\boldsymbol{W}} \boldsymbol{\delta W} = \int_{r_i}^{r_o} 2\pi C \delta r \tag{3}$$

$$\mathbf{C} = \frac{\boldsymbol{W}}{2\pi(r_o - r_i)} \tag{4}$$

The total frictional torque, T, is given by integration of the product of the clamping force given by ($\boldsymbol{\mu\delta W}$) and the radius, r, between the limits of the clutch facing, r_i and r_o, as in Eq. (5).

$$\mathbf{T} = 2\pi\mu p r^2 \delta r \tag{5}$$

$$\mathbf{T} = 2\pi\mu C \int_{r_i}^{r_o} r\delta r \tag{6}$$

$$\mathbf{T} = \pi\mu C(r_o^2 - r_i^2) \tag{7}$$

$$\mathbf{T} = z\mu\mathbf{W}r_m \text{ , where } r_m = (r_o + r_i)/2 \tag{8}$$

For a single plate clutch having one pair of contact surfaces, the total torque capacity is given by Eq. (8). Where z – number of frictional surfaces, μ- coefficient of friction, r_m – mean radius of the friction clutch and the average pressure (p_{avg}) applied on the frictional contact is given by Eq. (9).

$$\boldsymbol{P_{avg}} = \frac{\boldsymbol{W}}{\pi(r_o^2 - r_i^2)} \tag{9}$$

Thermal Model

For the thermal model, the following assumptions are considered: (i) clamping force on the clutch facing is uniform (ii) constant convective heat transfer coefficient during clutch engagement are employed to avoid complexity. During the engagement period, due to the speed difference of DMF and clutch plate, heat is generated between frictional surfaces till relative sliding velocity falls to zero. The heat generated between components of the clutch is dissipated through convection. The different groove patterns on the clutch facings affect the thermal dissipation through convection accordingly. The total heat produced during slipping is given as Eq. (10) [19].

$$\boldsymbol{q_f} = \begin{cases} \mu p \boldsymbol{V_s}; & 0 \le t \le t_s \\ 0; & t > t_s \end{cases} \tag{10}$$

$$\boldsymbol{V_s} = \omega_s r \tag{11}$$

$$\boldsymbol{\omega_s(t)} = \boldsymbol{\omega_0}\left(1 - \frac{t}{t_s}\right); 0 \le t \le t_s \tag{12}$$

where, q_f – Total thermal heat flux generated, V_s ($V_s = \omega_s r$) – sliding velocity and ω_s – sliding angular velocity, ω_o – relative sliding angular velocity when the clutch starts to slip (t=0). In this study, under uniform wear conditions, the friction torque and the heat flux generated are given in Eq. (13) and Eq. (15), respectively.

$$\boldsymbol{T} = \pi p_{max}\mu r_i(r_o^2 - r_i^2) \tag{13}$$

$$\boldsymbol{q_f} = \frac{T\omega_s}{A_f} \tag{14}$$

$$\boldsymbol{q_f} = \mu C\omega_0\left(1 - \frac{t}{t_s}\right) \; where, \mathrm{C} = p_{max}.r_i \tag{15}$$

Where ($C = p_{max}.\ r_i$), p_{max} - maximum pressure which determines the maximum transmittable torque; A_f - total friction surface area; r_i – inner radius, r_o – outer radius, ω_0 – angular velocity of sliding of the clutch disc. The rate of heat energy generated is a function of time only in the case of uniform wear [20]. The heat generated will be distributed in the clutch components by conduction and dissipated to the surrounding air by convection. To examine the temperature distribution in the clutch facing, 3D models of the clutch disc are simulated by applying the thermal load with boundary conditions during the engagement. The model with imposed loads and boundary conditions is presented in Figure 3.

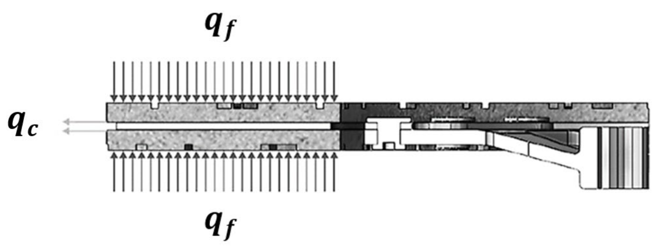


Figure 3. Loads and boundary conditions for the thermal model.

Surface Wear Model

Surface degradation caused by the removal of material during relative movement between one surface and another contacting body is known as wear. It can be approximated by models that relate various quantities at the contact surface to material loss despite its complex nature that involves both mechanical and chemical processes. Repositioning the contact nodes at the contact surface in ANSYS approximates material loss due to wearing. A wear model determines the updated coordinates of the nodes. Based on the contact readings at the contact nodes, the wear models compute how much and in which direction a contact node should be moved to simulate wear. According to Archard's original model, the rate of volume loss due to wear is proportional to the contact pressure and sliding velocity at the contact surface [21]. The software, by default, implements a generalised version of this model that incorporates proper law dependence on contact pressure and velocity.

$$\dot{w} = \frac{K}{H} P^m v_{rel}^n \tag{16}$$

where, K-wear coefficient, H-material hardness, P-contact pressure (obtained from an indentation test), V_{rel} - the relative sliding velocity, m-pressure exponent, n-velocity exponent. The values of K, m and n in the Archard wear model are typically obtained via empirical lab tests by experimentation on sample sets and curve fitting of the generated data sets.

METHODOLOGY

For mesh generation, standard practices are followed to get high-quality solid elements. A hexahedral meshing method is adopted for the target body and a tetrahedral meshing for the contact body. Linear elements are preferred for this analysis instead of quadratic elements due to their apparent higher stiffness values. This prevents them from becoming distorted easily during the change in contract status. The elements on the contact body (friction lining) are made finer to capture the wear more accurately. On average, a total of 11473 nodes and 51373 elements are generated across various geometries. This includes the aforementioned CONTA174 contact elements. Since wear requires repositioning the surface nodes to simulate material loss, with increasing wear, the element quality of the solid elements underlying the contact elements gradually becomes worse. Because of element distortions, the analysis can eventually terminate. In such a case, to strengthen the mesh and continue the analysis, we can adopt manual rezoning or nonlinear mesh adaptivity. Both these methods require substantial processing power and rebuild the mesh midway through the solution whenever the set user criterion (e.g., element quality) crosses a certain critical value, thus avoiding solution divergence. This makes the FEA model more robust and versatile by avoiding possible solution divergence in certain scenarios. The schematic of the workflow adopted in the present work is depicted in Figure 4.

In our case, to model the thermal contact, we have to use node-to-surface contact elements combined with thermal, structural, coupled solid elements to visualise and model the temperature distribution at the contact surface on the friction lining. To analyse the thermal contact behaviour with volume wear analysis, we have to assign frictional sliding at the contact. The dissipated frictional energy generates heat in contact surfaces, i.e., friction lining surfaces. The contact surface temperature is needed for interface heat conduction, convection, or radiation. To take into account the conduction and convection between the contact and target surfaces, we need to assign thermal conductance coefficient and heat convection coefficient, which can be constant value or function of time, temperature, or location. To model radiative heat transfer, we need emissivity value, Stefan-Boltzmann constant, offset temperature, and environment (ambient) temperature. We have to apply uniform heat flux on the contact elements, not on the target elements. To contribute to target elements for near-field contact, external flux can be applied. For modelling of heat generation due to friction, transient thermal, structural analysis results and effects are needed. These settings in the ANSYS Workbench can give the desired outputs.

The surface wear calculations are activated in a coupled field analysis to take the temperature effect into account. During the steady-state period (full engagement phase), it was anticipated that two types of load conditions act on the clutch system: the contact pressure between clutch elements due to the axial force of the diaphragm spring and the centrifugal force owing to rotation. The force applied is ramped up to the steady value to allow the contact and the target elements to find each other better. To capture the wear more accurately, the wear calculations are turned on right at the start of the first sub-step. In a frictional type of contact, the elements at the contact surface may experience an abrupt change in status (when they come into or out of contact), causing a sudden change in the stiffness matrix of the solver equation. This is the primary cause of non-linearity and hence solution convergence problems in contact analysis. Due to this high degree of non-linearity, contact simulations require a substantial amount of computational power. Thus, we have simplified the geometry to a great extent by removing small features such as fillets with small radii. The contact surfaces of rigid-flexible contact are set to the deformable body, while the target surfaces are set to the rigid body.

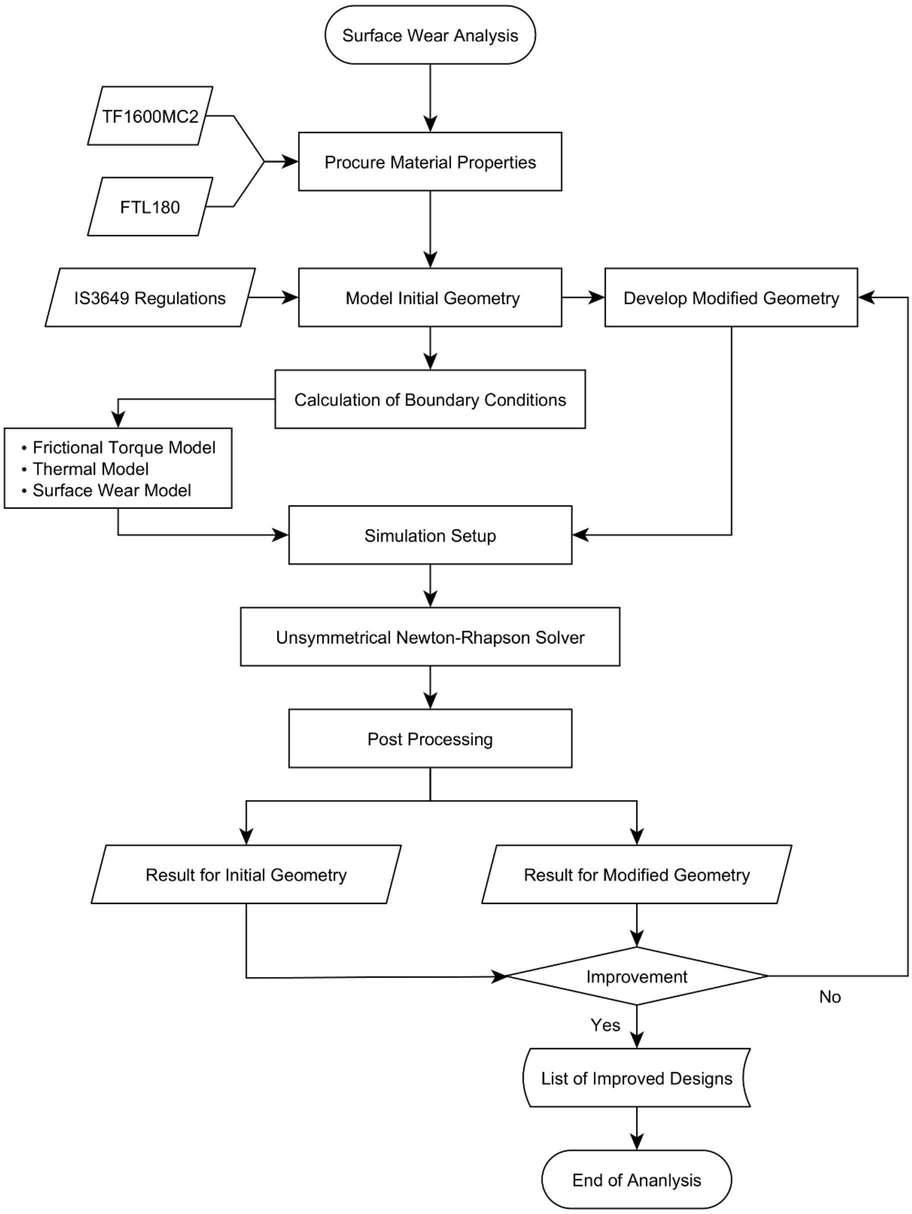


Figure 4. Workflow schematic.

In the case of contact formulation algorithm, only Augmented Lagrangian and Penalty Function algorithms are recommended since the default Pure Lagrangian algorithm can result in convergence problems. We have used the Augmented Lagrangian formulation here as it is more robust in comparison to, and is not recommended. The location of the contact detection point is set to normal to the target surface since, in this scenario, the target body (flywheel/ pressure plate) is fixed. Asymmetric contact is used as the flywheel/pressure plate does not wear significantly in comparison to the clutch plate. The Coulomb friction between the clutch face and the pressure plate produces heat. The pre-calculated heat flux on the contact region of the clutch face is taken into account. Heat loss by convection happens on air-exposed surfaces. The value convection is derived in the ANSYS database for stagnant air scenario-vertical planes. The radiative heat transfer is considered to be negligible.

In ANSYS, surface wear can only be activated for quasi-static and transient dynamic analyses. Contact surface wear is not natively available in the ANSYS Mechanical interface and thus has to be activated by invoking APDL (Ansys Parametric Design Language) commands. We use the CMROTATE command to impart rotational velocity components to the clutch plate instead of providing rotational velocity in mechanical because the latter indirectly invokes CGOMGA or CMOMEGA, which only takes into account the centrifugal forces and while ignoring the velocity components. The KBC command is used to step apply the required rotational velocity at the first sub-step. To initiate wear calculation right away, TB, WEAR is used in conjunction with TBFIELD, TIME. The problem is simulated in four load steps. In the first load step, pressure is ramped to the desired level. To avoid Solver Pivot Error, the NCNV command is called to raise the upper limit for the nodal DOF solution to 1.0E30. Using the NEQIT command, the maximum number of equilibrium iterations for nonlinear analyses is set to 50 to improve the chances of solver convergence.

RESULTS AND DISCUSSION

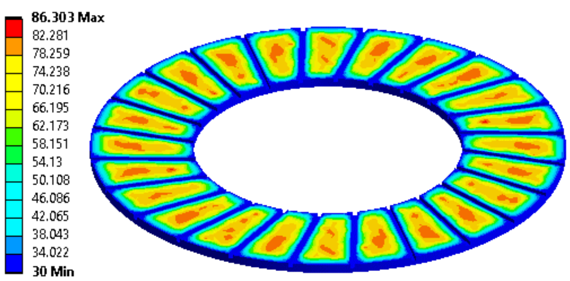
Slippage in the clutch has three major causes (i) facing wear (i.e., past the wear limit), (ii) improper contact, and (iii) burnt or disintegrated material. These root causes are frequently observed in standard-facing designs. The design approach for improving clutch geometry is dependent on temperature distribution, volumetric wear, wear thickness, and slip rate observed during an engagement scenario. Rusli et al. derived that the normal contact stiffness and friction coefficient are affected significantly by velocity and clamping force [22]. The values of these characteristics for the stock pattern are considered as a base reference throughout this simulation. The activation of the Archard wear model in ANSYS enables a set of miscellaneous results in the post-processing section, as shown in Table 3. These results are not natively available in the mechanical interface. These commands provide desired node-specific results at each element.

Table 3. Miscellaneous wear parameters obtainable in ANSYS Mechanical.

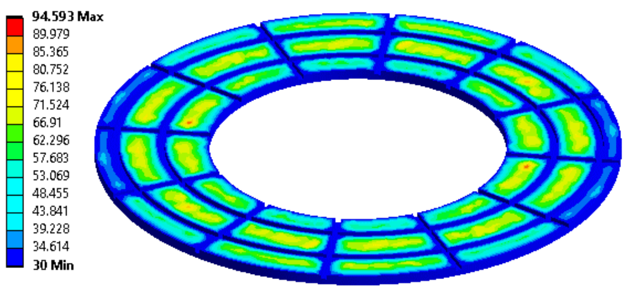
Name	Definition	Function	Nodes				
			E	I	J	K	L
VWEAR	Volume lost due to wear	CONTNMISC	189	-	-	-	-
VREL	Equivalent sliding velocity	CONTNMISC	-	156	157	158	159
WEARZ	Wear correction-Z component	CONTNMISC	-	180	181	182	183

Temperature

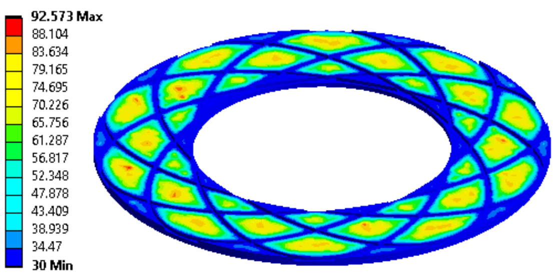
Clutch uses friction to transmit the torque, causing it to generate heat through slippage. At such elevated temperatures, the coefficient of friction may change, affecting the torque output. It can also distort the plates due to uneven distribution of heat; hence there is an absolute need to consider the thermal performance while discussing its behaviour. Figure 5 indicates the maximum temperature distributions of FLT180 on the contact region at the time of first engagement. No irregularity in the temperature contours is evident, which could affect the thermal stability of the clutch facing. The temperatures are found to rise during consecutive engagements as there is not enough time for the convective heat transfer medium to cool down the clutch facing fully at the next engagement. Hence, it is observed that the overall symmetry also considerably influences temperature distribution, while the patterns with aspect ratios closer to unity provide the lowest temperatures. Almost similar trends are observed for both the materials considered in this investigation.



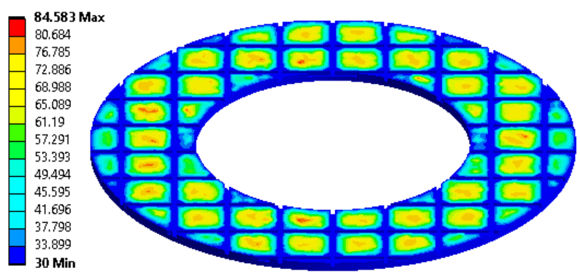
(a) Stock



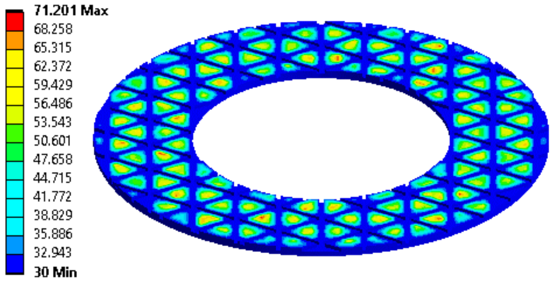
(b) Ellipse



(c) Sunburst



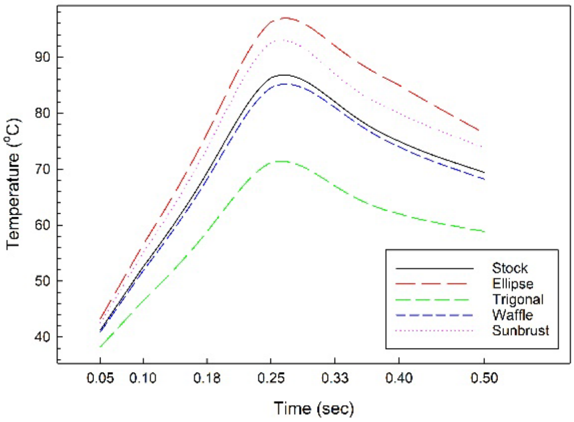
(d) Waffle



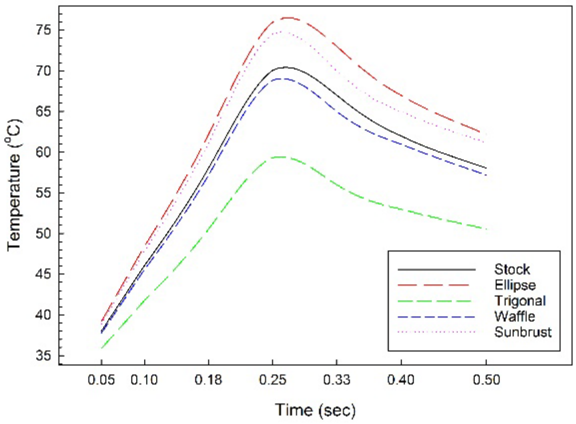
(e) Trigonal

Figure 5. Temperature distribution for the investigated clutch geometries.

Figure 6 shows a bell curve trend for the temperature-time graphs of all different clutch geometries made of inorganic materials. A similar trend is observed by the previous researchers [9,19,22,23,24]. The temperature continues to rise as the frictional heat generation peaks during the slipping period (t = 0.25 sec), and once the clutch is facing sticks to the flywheel/pressure plate, convective heat transfer increases, thus causing the cooling. Out of different patterns considered in this investigation, the Trigonal is providing the best cooling performance, followed by the Waffle pattern. Due to the lower thermal conductivity of the friction material, the hotspots tend to form over a large area. This is evident from the temperature contour plot for the stock pattern (as well as the outer contact surface of the Sunburst pattern).



(a)



(b)

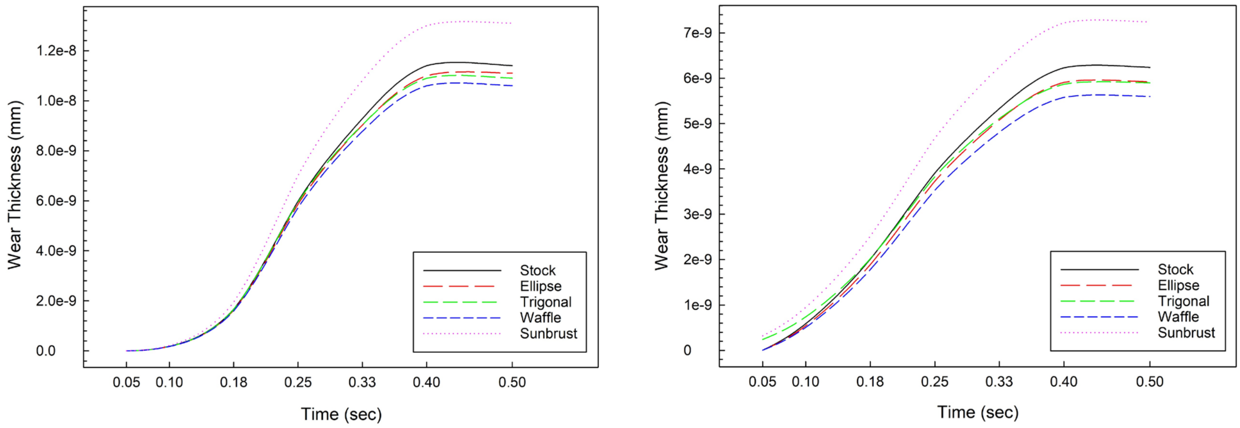
Figure 6. Temperature over time for different clutch geometries with (a) FTL 180 (b) TF1600-MC2.

Wear Thickness

When the friction material on the clutch disc wears beyond a certain point, the disc is no longer thick enough for it to be sufficiently clamped between the pressure plate and flywheel. The clutch then slips and loses the ability to transmit power to the transmission. If the surface wear is ignored, there may be a point where the rivets attaching the clutch material to the metal backing plate become exposed and scratch the flywheel causing further damage. A similarity is observed in the graphs of volumetric wear loss and wear thickness. During the initial contact phase, more wear is observed in TF1600-MC2 in comparison to FTL180. The Waffle pattern is found to exhibit the lowest level of material wear thickness. The wear thickness variation over time for different clutch geometries with the investigated materials is presented in Figure 7.

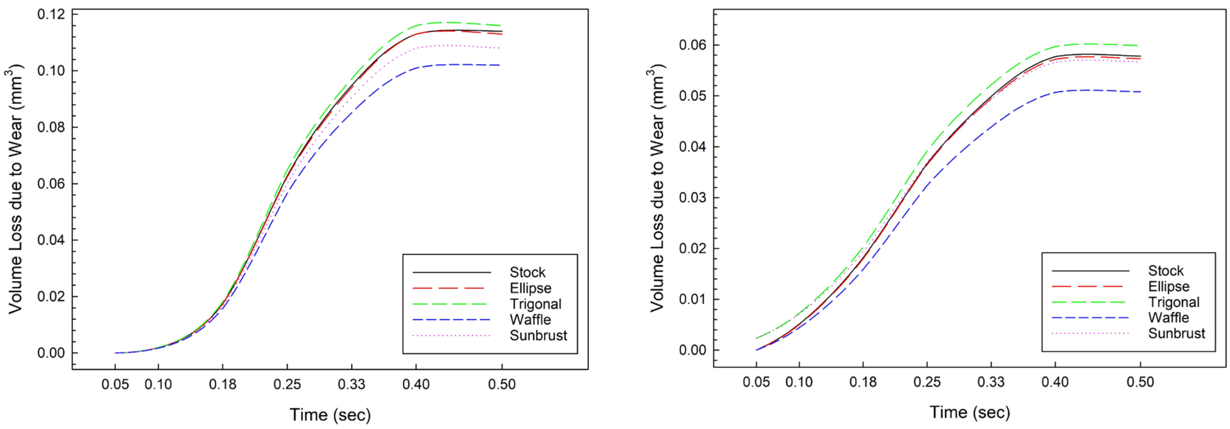
Volume loss due to wear

The volume lost during an engagement is an important criterion used to calculate and predict the estimated life of the particular clutch geometry. As shown in Figure 8, the volumetric wear loss is more linear in TF1600-MC2 as compared to FTL180. Thus, a clutch plate made of TF1600-MC2 will wear more uniformly during the initial engagement phase. The waffle pattern shows the lowest level of material wear compared to the stock clutch. Due to the higher rate of volumetric wear in the stock pattern, the facing will exceed its wear limit faster. As a consequence, the amount of clamp load available at the contact patch will drop to suboptimal values.



(a) (b)

Figure 7. Wear thickness over time for different clutch geometries with (a) FTL 180 (b) TF1600-MC2.

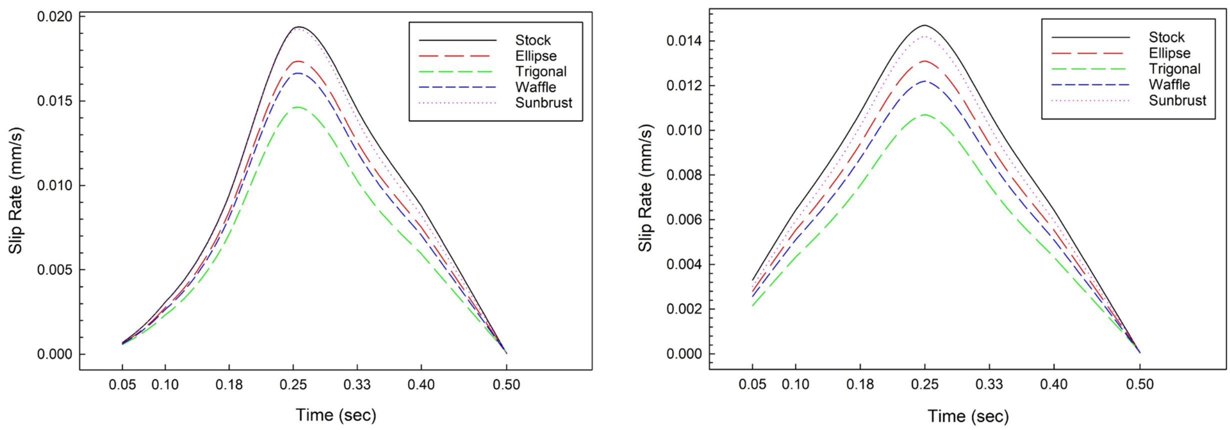


(a) (b)

Figure 8. Volumetric wear loss over time for different clutch geometries with (a) FTL 180 (b) TF1600-MC2.

Slip Rate

Slip rate can be defined as the equivalent sliding velocity at the contact surfaces relative to each other. From Figure 9, it is inferred that slip rates in all cases are found to be within an acceptable range. All of the developed patterns provide slip rates lower than that observed in the stock pattern. Since torque transferring capacity is directly proportional to the observed slip rate, it is safe to say that all the developed patterns provide higher torque transfer efficiency during the engagement. Here, it is also observed that relative slipping occurring during the initial engagement in TF1600-MC2 is higher than that of FTL180.



(a) (b)

Figure 9. Slip-rate over time for different clutch geometries with (a) FTL 180 (b) TF1600-MC2.

Lifetime

The wear limit of the clutch is defined as the maximum thickness of the facing that can wear before the clutch performance starts deteriorating. This limit varies from design to design based on the amount of material above the rivets or depth of grooves in the friction material, whichever comes first. In our case, the depth of grooves is the limiting factor with a value of 2 mm. Considering this value as the wear limit and dividing it by overall cumulative wear, one can estimate the total number of engagements for which the clutch will provide optimum performance. Then, by assuming the rate of clutch usage per km (based on the habits of an average driver in city conditions), one could calculate the total driving distance before the clutch exceeds its wear limit. There will always be certain scenarios bundled where the data from this simulation may prove to be unsatisfactory. Especially when it comes to surface wear since a major factor is the driver's habit and level of skill which cannot be accurately accounted for. The estimated life with different clutch geometries and materials is presented in Figure 10. There may be other factors such as vehicle weight, gearing, and terrain as well that can lead to deviations into predicted service life.

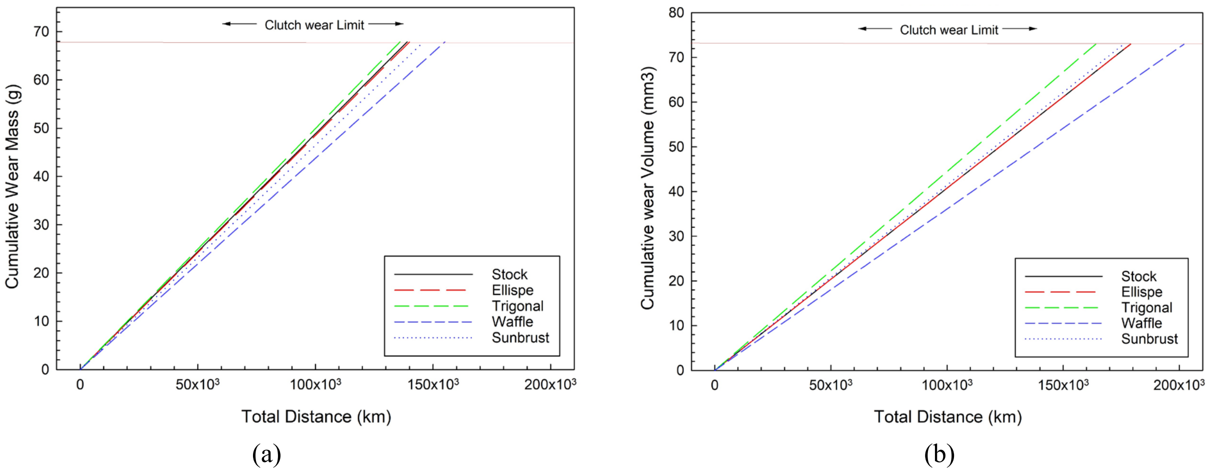


Figure 10. Estimated lifetime of the different clutch geometries with (a) FTL 180 (b) TF1600-MC2.

CONCLUSION

In this research, a simulation study is performed to model and analyse wear in clutch facings with various groove patterns. The wear characteristics of each of the patterns are studied by using a coupled field analysis in ANSYS. APDL commands are invoked to activate wear calculations via the Archard wear model in the mechanical interface. All the patterns are tested with two different inorganic materials with slightly different coefficients of friction to understand the trend in wear parameters better. From the study, the following observations are made.

i. The trigonal pattern showed a temperature reduction of about 17.61% compared to the stock pattern under identical operating conditions.
ii. The waffle pattern offered a decrease of about 10.4% in volumetric wear loss and a 5.78% in maximum wear thickness over the stock pattern.
iii. The waffle pattern exhibited an estimated increase of about 11.51% in the average working life when the clutch is used in high-traffic, city-like conditions.
iv. The wear rate, lifetime comparison of two inorganic material-based clutches facing different groove patterns revealed the supremacy of TF-1600MC2 over FTL 180.

The above results provide evidence that usage of the waffle pattern in a clutch facing provides improved performance in terms of wear and working life when coupled with inorganic materials. The clutch facing wears out faster than the DMF. During replacement of the clutch facing at the end of its service life, the entire set is needed to be replaced due to the complexity of the assembly. In such a case, the customer often finds the option of switching to solid mass flywheel logical. However, in the long term, this increases fuel consumption and indirectly CO_2 emissions. Increasing the life of the clutch facing improves the cost to performance ratio, thereby making the option of continuing with DMF more lucrative while saving money in terms of fuel during its extended lifetime.

The scope of this work can be extended in the future by (i) simulating and analysing the optimised facing on repeated engagements with organic and inorganic materials, (ii) obtaining a more accurate convection film coefficient using ANSYS CFX and validating the results by running lab tests, (iii) estimating the life of the clutch for standardised drive cycles like Modified Indian Driving Cycle (MIDC) and (iv) conducting lab experiments on the optimised clutch facing for performance analysis with required materials. Based on these data from physical lab tests can be used to implement and extract the advantages of the optimised facings.

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REFERENCES

[1] Valeoscope product focus on transmission systems dual-mass flywheel (998120). [Online]. Available: https://valeoservice.cld.bz/VS-Transmission-Systems-Dual-Mass-Flywheel-DMF-valeoscope/16; [Accessed: April 14, 2021]

[2] User data on flywheel replacement. [Online]. Available: http://forums.pelicanparts.com/boxster-cayman-forum/552741-typical-lifetime-stock-dual-mass-flywheel.html; [Accessed: April 14, 2021]

[3] Clutches article on dual-mass flywheel. [Online]. Available: https://www.a1-clutches.co.uk/dual-mass-flywheels/ [Accessed: April 14, 2021]

[4] M.R. Ramesh *et al.,* "Optimising thermal performance of a dry rigid clutch by varying groove pattern and friction material," *Mater. Today: Proc.,* vol. 46, no. 17, pp. 7459-7467, 2021, doi: 10.1016/j.matpr.2021.01.130.

[5] O.I. Abdullah, and J. Schlattmann, "Contact analysis of a dry friction clutch system", I*nt. Sch. Res. Notices*, vol. 2013, 9 pages, 2013, doi: 10.1155/2013/495918

[6] A. Vadiraj, "Engagement characteristics of a friction pad for commercial vehicle clutch system.", *Sadhana*, vol. 35, no. 5, pp. 585–595, 2010, doi: 10.1007/s12046-010-0042-9

[7] O. I. Abdullah, and J. Schlattmann. "An investigation into the thermal behavior of the grooved dry friction clutch." *J. Tribol.*, vol. 136, no. 3, 6 pages, 2014, doi: 10.1115/1.4027388

[8] B. Czél *et al.*, "FE thermal analysis of a ceramic clutch", *Tribol. Int.,* vol. 42, no. 5, pp. 714-723, 2009, doi: 10.1016/j.triboint.2008.10.006.

[9] E. Mouffak and M. Bouchetara, "Transient thermal behavior of automotive dry clutch discs by using ANSYS software." *Mechanics,* vol. 22, no. 6, pp. 562-570, 2016, doi: 10.5755/j01.mech.22.6.17277

[10] N.A. Jabbar, I.Y. Hussain, and O.I. Abdullah, "Thermal and thermoelastic problems in dry friction clutch: a comprehensive review", *Heat Transfer*, 1- 24, 2021, doi: 10.1002/htj.22257

[11] T. Gkinis *et al.*, "Heat generation and transfer in automotive dry clutch engagement", *J. Zhejiang Univ. Sci. A*, vol. 19, no. 3, pp. 175–188, 2018, doi: 10.1631/jzus.A1700481

[12] S. Paulraj and S. Muthiah, "Estimation of clutch life for manual transmission vehicle through thermal modeling of clutch housing and clutch facing," *SAE Technical Paper* 2017-01-2439, 2017, doi: 10.4271/2017-01-2439.

[13] H. Bao *et al.*, "Wear characteristics of different groove-shaped friction pairs of a friction clutch", *Appl. Sci.*, vol. 11, no. 1, 2021, doi: 10.3390/app11010284

[14] Ibrahim *et al.*, "Review on development progress of automatic manual transmissions control." *J. Chongqing Univ.,* vol. 3, no. 2, pp. 38-42, 2004

[15] IS 3649: 2018, Automotive vehicles - Clutch facing for automotive transmission (First Revision). [Online]. Available: https://www.services.bis.gov.in:8071/php/BIS_2.0/bisconnect/standard_review/Standard_review/Isdetails?ID=MTAyMzY%3D; [Accessed: March 15, 2021].

[16] K. Arunachalam and G. S. Pandian. "Modeling and analysis of clutch facing made up of biodegradable coir fibre-based composite material." *Polym. Polym. Compos.*, vol. 24, no. 7, pp. 463-468, 2016, doi: 10.1177/096739111602400703

[17] Clutch material properties. [Online]. Available: https://www.protecfriction.com/tf1600-mc2-high-performance-kevlar; [Accessed: February 28, 2021].

[18] Clutch material properties. [Online]. Available: https://s3-eu-west-1.amazonaws.com/frictiontechnology/pdfs/FTL180.pdf; [Accessed: February 28, 2021].

[19] A. Anosh *et al.,* "Dry friction clutch disc of an automobile under transient thermal load: a comparison of friction lining materials." *MATEC Web of Conferences*, vol. 124, 2017, doi: 10.1051/matecconf/201712407003

[20] O.I. Abdullah and J. Schlattmann, "The correction factor for rate of energy generated in the friction clutches under uniform pressure condition", *J. Adv. Theor. Appl. Mech.*, vol. 5, no.6, pp. 277-290, 2012.

[21] Ansys help. [Online]. Available: https://www.ansys.com/en-in/academic/students; [Accessed: February 28, 2021].

[22] M. Rusli *et al.*, "Squeal noise analysis using a combination of nonlinear friction contact model." *Int. J. Automot. Mech. Eng.*, vol. 17, no.3, pp. 8160-8167, 2020, doi: 10.15282/ijame.17.3.2020.09.0613

[23] O.I. Abdullah, J. Schlattmann, and A.M. Al-Shabibi, "Thermo-mechanical analysis of the dry clutches under different boundary conditions", *Tribol. Ind.*, vol. 36, no. 2, pp. 172-180, 2014.

[24] O.I. Abdullah, L. A. Sabri, and W. A. Al-Sahb, "Finite element analysis of the thermal behaviour of single-disc clutches during repeated engagements." *Tribologia,* vol. 266, no. 2, pp 9-24, 2016, doi: 10.5604/01.3001.0010.7559

[25] L. Frérot, R. Aghababaei, and J.-F. Molinari, "A mechanistic understanding of the wear coefficient: From single to multiple asperities contact", *J. Mech. Phys. Solids*, vol. 114, pp. 172-184, 2018, doi: 10.1016/j.jmps.2018.02.015

[26] J. Archard, "Contact and rubbing of flat surfaces", *J. Appl. Phys*., vol. 24, no.8, pp. 981-988, 1953, doi:10.1063/1.1721448