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Effect of Surface Roughness on the Thermoelastic Behaviour of Friction Clutches

The accurate computation of the contact pressure distribution is considered the main key to obtain the temperature distribution of the contact surfaces of clutch with high accuracy. High number of researchers in the thermoelastic field assumed that the contact surfaces of the automotive clutches and brakes are flat, and they do not take the actual surface roughness into consideration in the numerical models. In this paper, a new model of rough clutch disc has achieved to show the actual contact pressure of the new and used friction facing of clutch disc. The effect of the surface roughness of the used friction facing on the heat generated and the temperature fields are investigated as well. The comparison has been made between the results when assuming the surface is rough and when it's flat. Axisymmetric finite element model has been developed to study the thermal behaviour of a single-disc clutch system (pressure plate, clutch disc and flywheel) during the sliding period.

Keywords: *dry friction clutch, contact, frictional heating, temperature field, FEM.*

1. INTRODUCTION

A number of researchers assumed that the contact surfaces of the automotive clutches and brakes are flat in the numerical analysis of contact problem of their systems. In reality the contact surfaces are rough on the microscopic scale. This fact will affect the contact phenomenon, and it consequently will affect the performance, efficiency and wear characteristics of the sliding system. Generally, the total actual contact area of the surface consists of a number of microscopic sub-contact areas located asperities of the rough surface. Common solutions of contact problem are assumed that the actual contact surface as a statistical distribution of asperities of prescribed shape. Owing to this assumption, the sum of the individual loads on the contact asperities is equal to the total load, each of which compresses a distance depending on its initial height [1].

The early investigation in this field was made by Greenwood and Williamson [2]. They discovered that many important properties of the contact are almost independent of the local asperity behaviour if the asperity height distribution in Gaussian. When the distribution of the identical asperities is exponential, they found linear relations between the total load, thermal and electrical contact conductance and the total contact area, regardless the constitutive law describing of the contact process of the actual contact area. The most resent developments have a tendency to focus on describing the contact surfaces as a stationary random process.

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Greenwood [3] found the relations between various treatments of this type and discussed the correlation between the profilometer measurements and the properties of the surfaces. In the past few years, a fast progress in the experimental approach to investigate the measurements of the surface profile revealed the existence of hierarchy of scale up to the limits of the experimental discrimination.

Abdullah and Schlattmann [4-8] investigated the temperature field and the energy dissipated of dry friction clutches during a single and repeated engagement under uniform pressure and uniform wear conditions. They also studied the effect of the contact pressure between contacting surfaces when torque is varying with time on the temperature field and the internal energy of clutch disc using two approaches. The heat partition ratio approach which used to compute the heat generated for each part individually, whereas the second applies the total heat generated for the whole model using contact model. Furthermore, they studied the effect of engagement time, sliding speed, thermal load and dimensionless disc radius (inner disc radius/outer disc radius) on the thermal behaviour of the friction clutch during the beginning of engagement.

2. EXPERIMENTAL WORK

The actual surface roughness of the new frictional facing of clutch disc is measured using MarSurf UD 120/LD 120 device. The method is used in this device is called "Stylus method". The procedure is used to find that the surface roughness of the specimen starts with fixing the specimen, setup the device and use the stylus to traverse across the work-piece surface. The vertical displacement of the stylus will measure with a conversion into electric signal, and this signal will be amplifying. The results are conversed into digital data, this data will be used by the special software to analyze this data to present the surface profile and calculate the standard roughness parameters. Figure 1 shows MarSurf UD 120/LD 120 device which used to measure the surface roughness. Also, Figure 1 shows the surface roughness with disc radius for three samples at different locations of the frictional facing.

Where R_a is the arithmetical mean deviation of the roughness profile, R_z is the maximum height of the roughness profile, $R_{\max}$ is the maximum roughness depth, and R_t is the maximum distance between the highest peak and lowest groove over a specified distance. It can be seen from this figure that the surface roughness at each location has different behaviour. The differences between the values of the standard roughness parameters are, however, small.

3. FINITE ELEMENT FORMULATION

In this paper, the sequentially coupled thermal-mechanical approach will be used to study the thermo-mechanical behaviour of a single-disc clutch. The sequentially coupled thermal-mechanical approach used two different models. One is used to solve the elastic problem to yield the displacement field and the contact pressure distribution whereas the other model is used to solve the transient thermal problem to account for the change in the temperature field. Both models are however coupled to each other since the contact pressure from the first model is needed to define the frictional heat flux for the second model. Furthermore, the temperature field from the thermal model is required for computation of the contact pressure from the elastic model for the next load step. To account for the coupling and variation of the sliding speed with time, clutch engagement time is divided into small time steps. At each time step, the instantaneous nodal temperatures are used in the elastic contact model to determine the contact pressure distribution. The pressure distribution is assumed to remain constant during subsequent time.

A transient finite element simulation is developed for axisymmetric thermoelastic contact problem of a single-disc dry clutches.

Figure 2 shows the flowchart of the finite element simulation of a coupled field problem of clutches. If the temperature field $T(x, y, z, t)$ is known, the thermoelastic contact problem can be solved to obtain the contact pressure $p(x, y, t)$. The heat generated due to friction is:

$$q_{\mathrm{f}} = \mu \, p V_{\mathrm{s}} \, , \tag{1}$$

where: μ is the coefficient of friction, p is the contact pressure, and V_s is the sliding speed between the clutch parts.

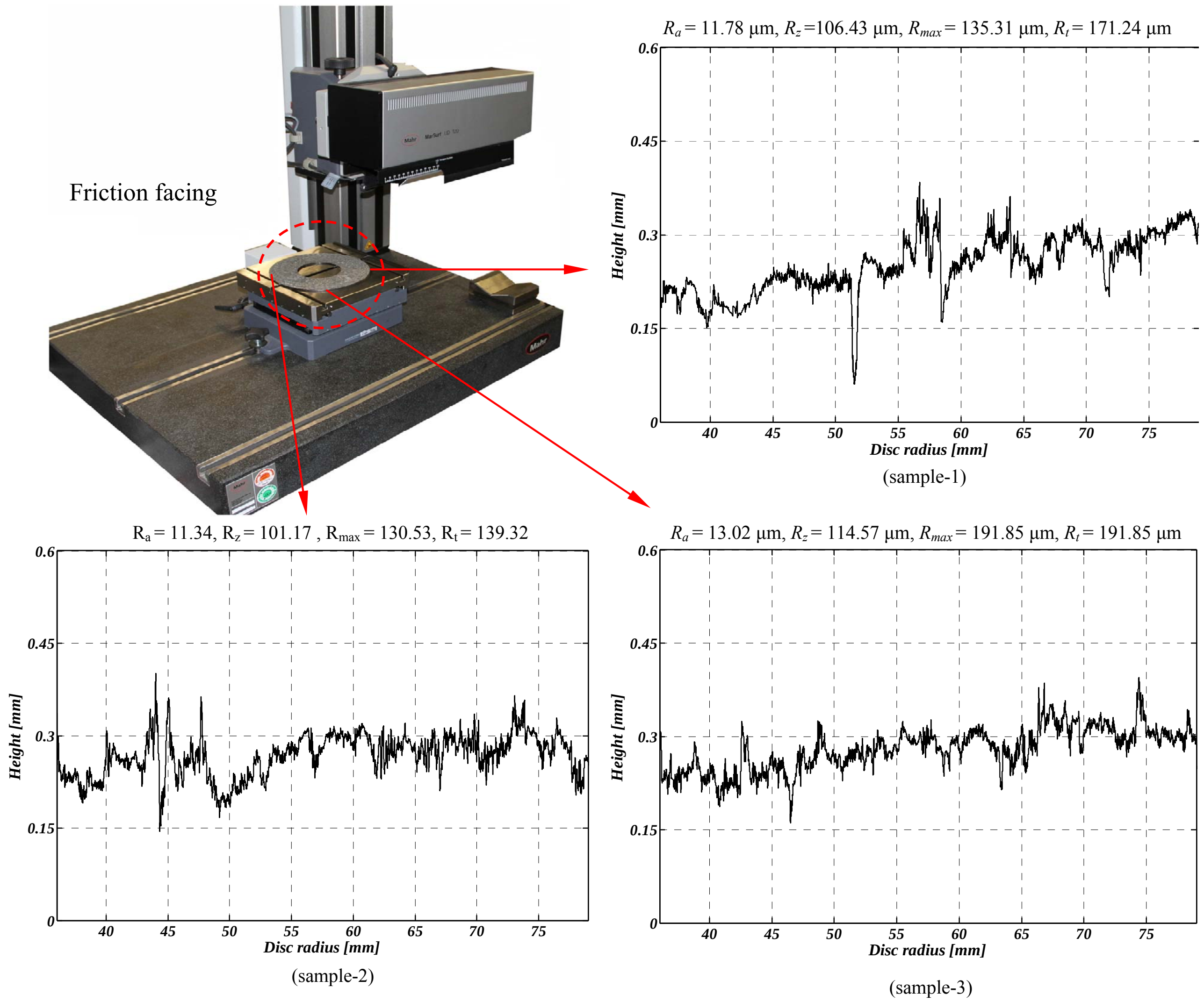

Figure 1. The details of MarSurf UD 120/LD 120 device and the measured surface roughness of the frictional facing

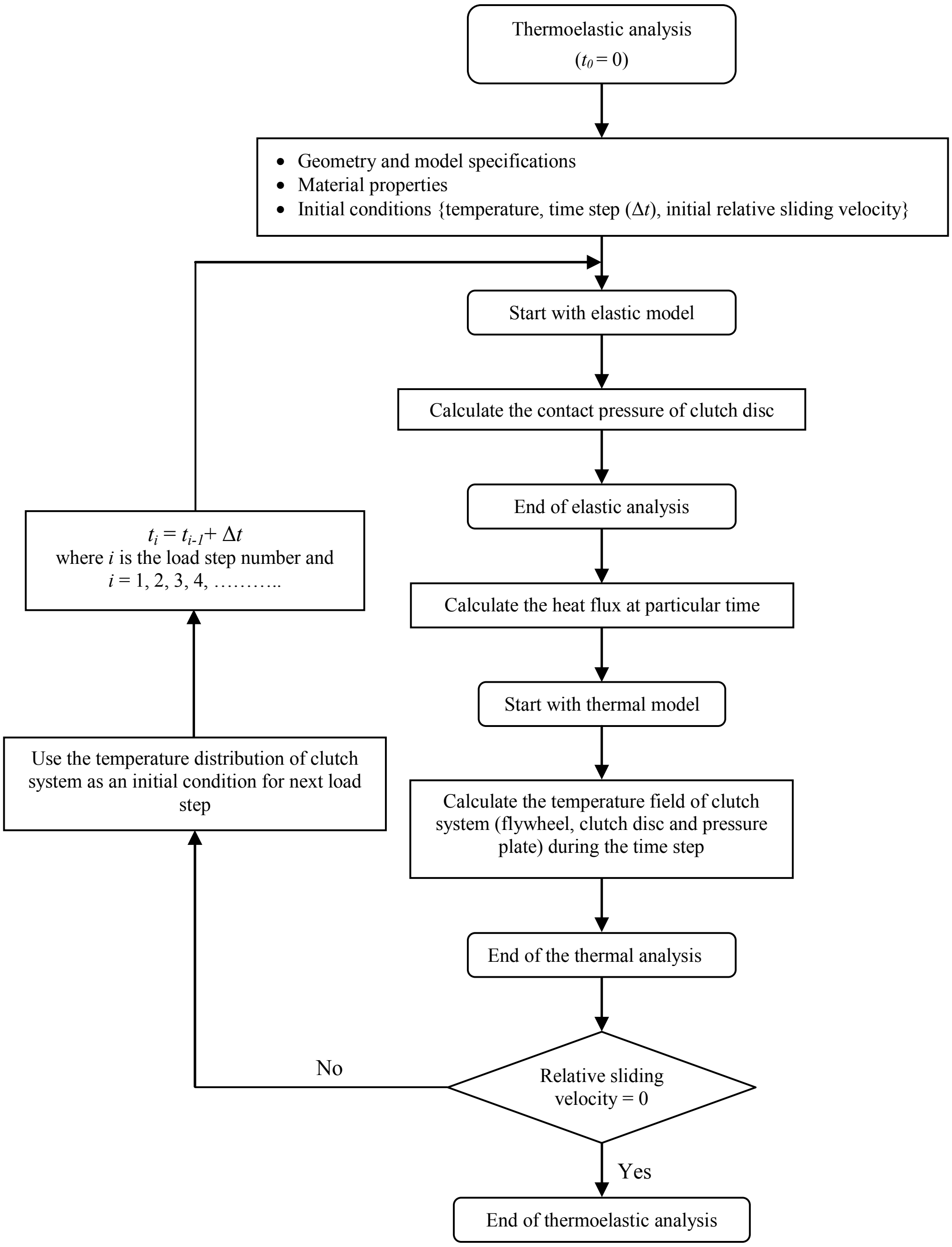

Figure 2. Flowchart of finite element simulation of sequentially coupled thermal-mechanical approach [10]

The contact pressure in (1) can be defined as the set of nodal contact forces. Therefore, the frictional heating q_{f} can be represented by the heat sources at nodes. These data is used as an input into the thermal finite element simulation. The next step of this procedure is the heat conduction equation represented by the thermal model, which can be solved to obtain the new temperature field $T(x, y, z, t+\Delta t)$ of the system. In modelling of contact problems, a special attention is required because the actual contact area between the contacting bodies is usually not known beforehand. In contact problems, unlike other boundary problems, nodes on the contact surface do not have prescribed displacements or tractions. Instead, they must satisfy two relationships: (1) Continuity of normal displacements on the contact surface (no overlap condition of contact area); (2) Equilibrium conditions (equal and opposite tractions).

Even if the contacting bodies are linear materials, contact problems are nonlinear because the contact area does not change linearly with the applied load. Accordingly, iterative or incremental schemes are needed to obtain accurate solutions of contact problems.

The iterations to obtain the actual contact surface are finished when all of these conditions are met [11]. Figure 3 shows the interfaces of two adjacent subregions i and j of elastic bodies. The elastic contact problem is treated as quasi-static with standard unilateral contact conditions at the interfaces.

The following constraint conditions of displacements are imposed on each interface:

$$w_i = w_j \,, \text{ if } \; p > 0 \tag{2}$$

$$w_i \le w_j \,, \text{ if } \; p = 0 \,, \tag{3}$$

where: p is the normal pressure on the friction surfaces.

The radial component of the sliding speed resulting from the deformations is considerably smaller than the circumferential component. Therefore, the frictional forces in radial direction on the friction surfaces are disregarded in this study [12]. Figure 3b shows thermal phenomena of two adjacent subregions of bodies. The interfacial thermal boundary conditions depend on the state of mechanical contact. Two unknown terms q_{ni} and q_{nj} exist on each interface. In order to fully define the heat transfer problem, two additional conditions are required on each contact interface. If the surfaces are in contact, the temperature continuity condition and the heat balance condition are imposed on each interface:

$$T_i = T_j \,, \text{ if } \; p > 0 \,, \tag{4}$$

$$q_{\mathrm{f}} = \mu p r \omega = -\sum q_n = -(q_{ni} + q_{nj}) \,, \text{ if } \; p > 0 \,, \tag{5}$$

where: ω is the angular sliding speed.

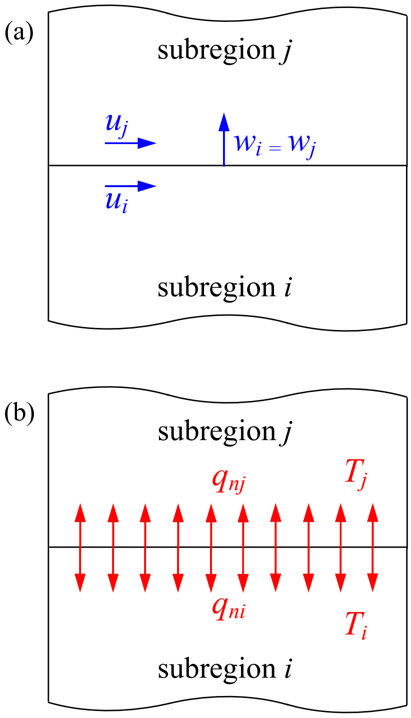

Figure 3. Contact model for: (a) elastic and (b) transient thermal problem in two adjacent subregions

Using the aforementioned conditions, equations of one node from each pair of contact nodes are removed. If the surfaces are not in contact, the separated surfaces are treated as an adiabatic condition:

$$q_{\mathrm{f}} = 0 = q_{ni} = q_{nj} \,, \text{ if } \; p = 0 \,. \tag{6}$$

Distribution of normal pressure p in (5) can be obtained by solving the mechanical problem occurring in the clutch disc. Assume the sliding angular speed decreases linearly with time as:

$$\omega(t) = \omega_o (1 - \frac{t}{t_{\mathrm{s}}}) \,, \quad 0 \le t \le t_{\mathrm{s}} \,, \tag{7}$$

where: ω_o is the initial sliding angular speed when the clutch starts to slip ($t = 0$).

A mesh sensitivity study was done to choose the optimum mesh from computational accuracy point of view. The full Newton-Raphson with unsymmetric matrices of elements used in this analysis assuming a large-deflection effect. In all computations for the friction clutch model, material has been assumed a homogeneous and isotropic material and all parameters and material properties are listed in Table 1. Analysis is conducted by assuming there are no cracks in the contact surfaces. Axisymmetric finite element model of a single-disc clutch system with boundary conditions is shown in Figure 4. In this model, the surfaces of clutch disc are considered rough.

Table 1. The properties of materials and operations

Parameters	Values
Inner radius of friction material and axial cushion, r_{i} [m]	0.035
Outer radius of friction material and axial cushion, r_{o} [m]	0.08
Thickness of friction material, t_{l} [m]	0.0035
Thickness of the axial cushion, t_{axi} [m]	0.002
Inner radius of pressure plate, r_{ip} [m]	0.03
Outer radius of pressure plate, r_{op} [m]	0.085
Thickness of the pressure plate, t_{p} [m]	0.01018
Inner radius of flywheel, r_{if} [m]	0.01
Outer radius of flywheel, r_{of} [m]	0.095
Thickness of the flywheel, t_{f} [m]	0.0358
Applied pressure, p_{a} [MPa]	1
Coefficient of friction, μ	0.2
Number of friction surfaces, n_{p}	2
Initial angular sliding speed, ω_o [rad/s]	300
Young's modulus of friction material, E_{l} [GPa]	0.3
Young's modulus of pressure plate, flywheel and axial cushion, E_{p}, E_{f}, and E_{axi} [GPa]	210
Poisson's ratio of friction material, v_{l}	0.25
Poisson's ratio of pressure plate, flywheel and axial cushion, v_{p}, v_{f}, and v_{axi}	0.28
Density of friction material, ρ_{l} [kg/m^3]	2000
Density of pressure plate, flywheel and axial cushion, ρ_{p}, ρ_{f}, and ρ_{axi} [kg/m^3]	7800
Conductivity of friction material, K_{l} [W/mK]	1
Conductivity of pressure plate, flywheel and axial cushion, K_{p}, K_{f}, and K_{axi} [W/mK]	54
Specific heat of friction material, c_{l} [J/kgK]	120
Specific heat of pressure plate, flywheel and axial cushion, c_{p}, c_{f}, and c_{axi} [J/kgK]	532
Thermal expansion of friction material and steel, α [K^{-1}]	12E– 6

4. RESULTS AND DISCUSSIONS

Transient thermoelastic analysis has been performed to study the effect of the surface roughness on the contact pressure, the frictional heat generation (heat flux) and

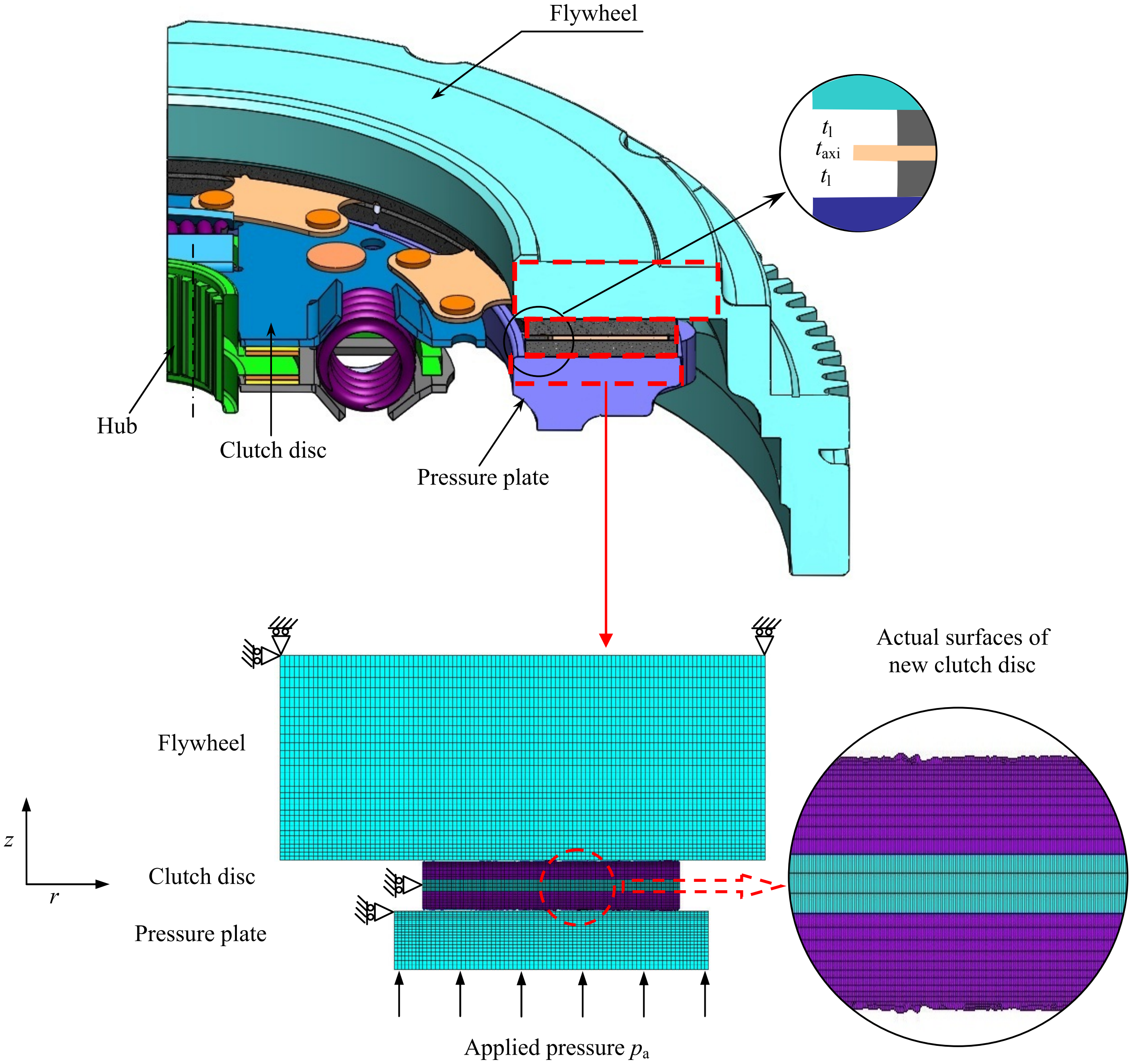

Figure 4. Axisymmetric finite element model with boundary conditions of the rough surface of the frictional facing

the temperature field during the slipping period of a single-disc clutch system.

The variations of the normalized contact pressure with disc radius for both sides of the new clutch disc (actual surface roughness) are shown in Figures 5 and 6. It can be seen that the contact pressure will focus on a very small zone (highest points in the rough contact surfaces) of the nominal contact area during the first engagement of the new friction facing. This situation will not continue for a long time due to the fast wear in peak points of the frictional facing, subsequently the frictional facing will be less rough and the actual contact area will increase with time until a certain time. After a large number of applications, the wear that occurred in the contact surfaces will decrease the actual contact area of the frictional facing and this will lead to failure of the friction material.

In the second part of the results, the effect of surface roughness on the contact pressure, the heat generated and the surface temperature will investigate the used clutch disc. It will assume that the maximum distance between the highest peak and a lowest groove over a specified distance (R_t) is assumed equal to 35 % of the value which measured a new friction disc.

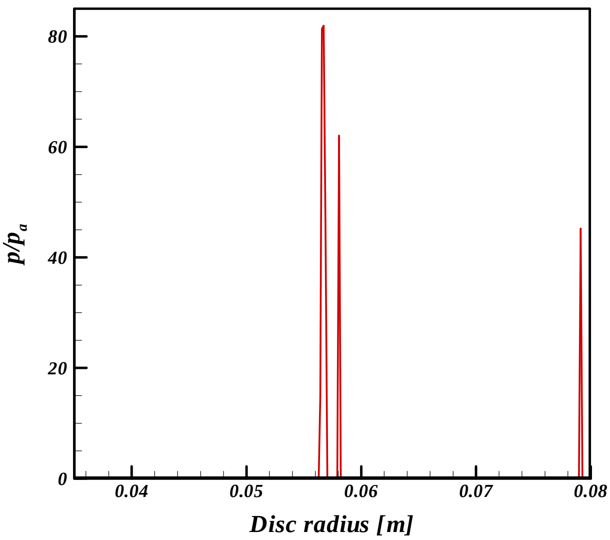

Figure 5. Variation of the normalized contact pressure with disc radius (flywheel side)

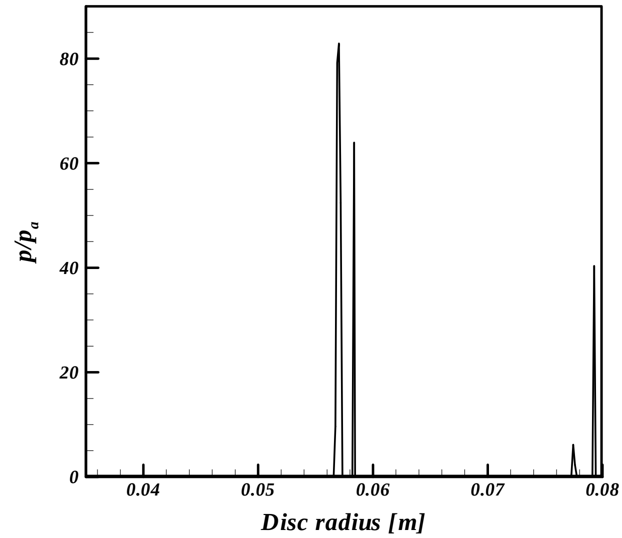

Figure 6. Variation of the normalized contact pressure with disc radius (pressure plate side)

Figures 7, 8 and 9 show the variation of the contact pressure distribution of the rough and flat friction clutch surfaces at selected time intervals during the sliding period. It can be seen from these figures that the values of the contact pressure increases with time. The reason of these results is due to the increasing of the thermal deformations in the contacting surfaces, and the actual contact area will decrease with time. Also, it can be seen from all these figures that the values of the maximum contact pressure of the rough surface are higher than those of the flat surface. The maximum values of contact pressure occur in the area located between the inner and mean disc radii of the clutch discs for all cases. The percentage of increasing in maximum contact pressure is found to be 124 % at t = 0.36 s (t = 0.9 t_s) when considered the rough surface instead of the flat one.

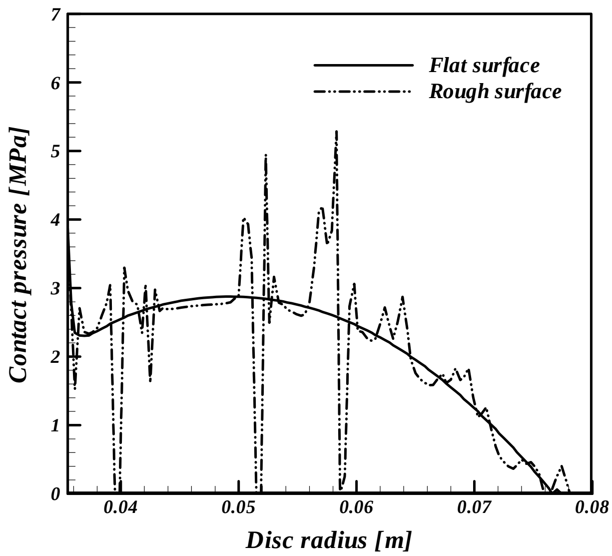

Figure 7. Variation of the contact pressure with disc radius of a single-disc clutch (pressure plate side, *t* = 0.04 s)

Figures 10 and 11 demonstrate the variation of heat flux with disc radius at selected time intervals for both models (rough and flat clutch discs). It can be seen that the magnitude of the frictional heat generated decrease with time due to the decreasing occurrence in the sliding speed.

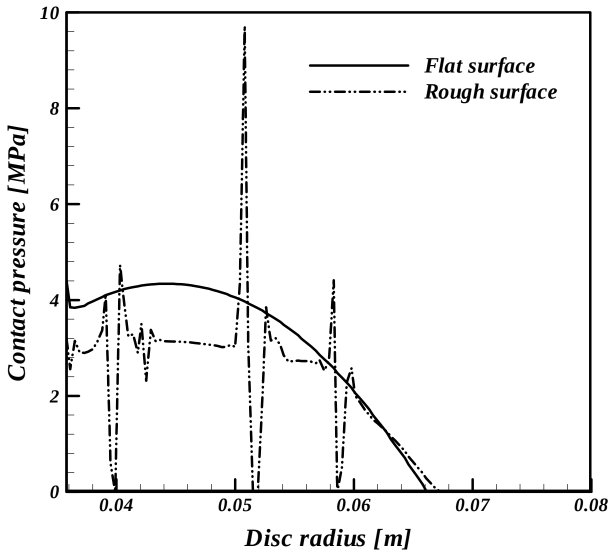

Figure 8. Variation of the contact pressure with disc radius of a single-disc clutch (pressure plate side, *t* = 0.2 s)

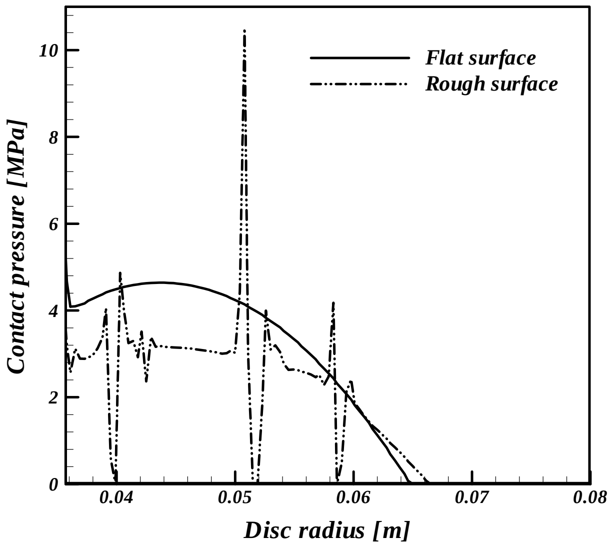

Figure 9. Variation of the contact pressure with disc radius of a single-disc clutch (pressure plate side, *t* = 0.36 s)

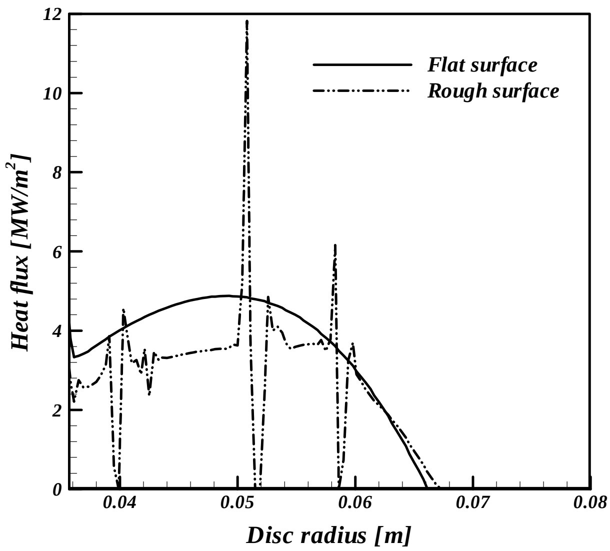

Figure 10. Variation of the heat flux with disc radius of a single-disc clutch (pressure plate side, *t* = 0.24 s)

The frictional heat generated is proportional to the sliding speed, when the sliding speed increases the amount of heat flux increases too, see (1). Also, it can be observed that the locations of the highest values of the heat flux correspond to the locations of the highest values of the contact pressure.

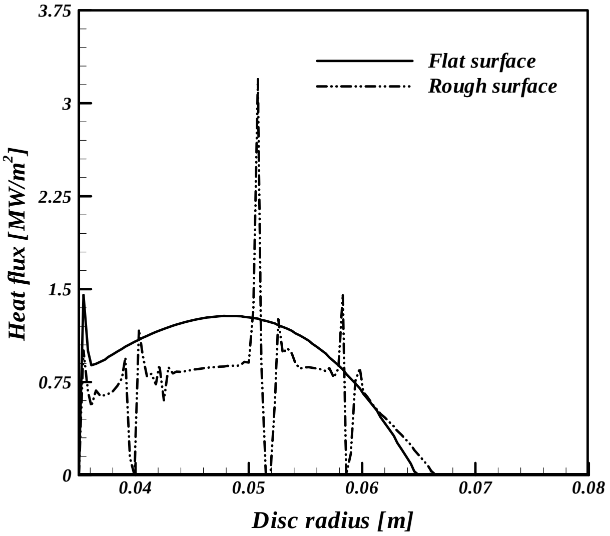

Figure 11. Variation of the heat flux with disc radius of a single-disc clutch (pressure plate side, *t* = 0.36 s)

At the end of the sliding period the magnitude of heat flux becomes zero and the clutch system enters into a full engagement period (all the parts of the clutch system will rotate as one unit).

Figures 12 and 13 show the temperature distributions over the friction disc at selected time interval of the sliding period. It can be seen that the temperature values of the rough surface are higher than those of flat surface in some locations, while in the other locations the temperature values of the flat surface are higher than those of rough surface.

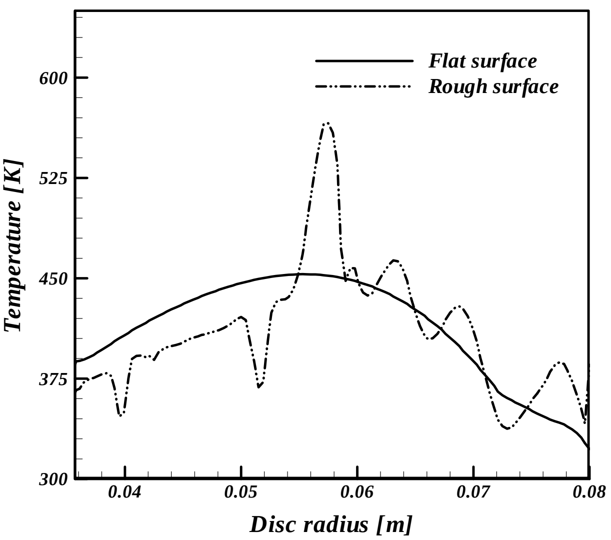

Figure 12. Variation of the surface temperature with disc radius of a single-disc clutch (pressure plate side, *t* = 0.1 s)

The reason of these results is the values and the distribution of the surface temperature as a function of surface roughness. In other words, in the highest peak point in the contact surface the highest temperature will occur there, and in the lowest groove of the contact surface the minimum temperature will occur at this point.

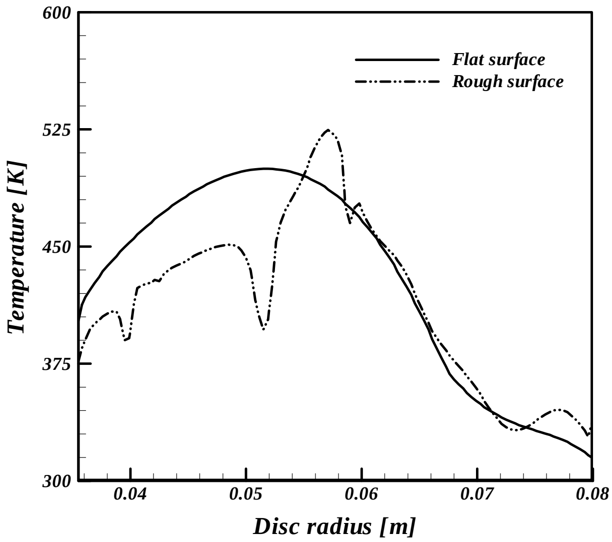

Figure 13. Variation of the surface temperature with disc radius of a single-disc clutch (pressure plate side, *t* = 0.2 s)

5. CONCLUSIONS AND REMARKS

In this paper, the solution of the transient thermoelastic problem of a single-disc clutch system during the beginning of engagement (slipping period) has been performed. Axisymmetric finite element model was built to obtain the contact pressure, the frictional heat generation (heat flux) and the temperature field.

The surface roughness which is considered an important issue should be taken into consideration. The magnitude and the distribution of the contact pressure are very sensitive to the surface roughness as shown in the results of this paper. The initial roughness of the frictional facing will decrease with time due to the fast wear process between the contact surfaces at the beginning of engagement, and the rate of wear will be faster in the soft material (friction material). The damaged or incorrectly machined flywheel (such as large deformation, thermal cracks, etc.) causes many problems. One of them is the contact pressure on the small regions of nominal frictional interface (e.g. bands and/or spots). Owing to this problem, it is essential when fitting a new clutch to a vehicle to ensure that the flywheel is in perfect condition, in order to prevent any possible premature damage occurrence in the clutch system.

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УТИЦАЈ ХРАПАВОСТИ ПОВРШИНЕ НА ТЕРМОЕЛАСТИЧНО ПОНАШАЊЕ ФРИКЦИОНОГ ДИСКА КВАЧИЛА

Одаи И. Абдулах, Јозеф Шлатман, Михаел Литкин

Прецизан прорачун расподеле контактног притиска се сматра најважнијим условом за добијање расподела температура контактних површина квачила са високом прецизношћу. Велики број истраживача који се баве термоеластичношћу претпоставља да су контактне површине у квачилима и кочницама идеално равне, и у нумеричким моделима не узимају у обзир храпавост површина. У овом раду је примењен нов модел који узима у обзир храпавост фрикционог диска квачила, који је успео да прикаже стварни контактни притисак на радну површину диска у два случаја: нов диск и коришћен диск. Поред тога је испитан и утицај храпавости радне површине коришћеног диска на развој топлоте, односно на температурско поље површине диска. Извршено је поређење резултата добијених под претпоставком да је испитивана површина идеално равна и резултата добијених под претпоставком да је та површина храпава. Осносиметрични модел са коначним елементима је направљен како би се извршило проучавање термичког понашања квачила са једним диском, током клизања.