Effects of Frictional Facing Thickness on the Heat Generated in Dry Friction Clutches

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Abstract. The accurate calculation of the frictional heat generated between the contacting surfaces of friction clutch elements during the slipping period due to the relative motion between the driving and driven parts is considered to be an essential factor for automotive designers. A numerical model of a single-disc clutch was built using the finite element method to investigate the effect of frictional facing thickness on the heat generation and distribution between the contacting surfaces of the flywheel, friction clutch disc and pressure plate. A sequentially coupled thermal-mechanical approach was used to simulate the thermal-elastic coupling that occurs in dry friction clutches during the slipping period. An axisymmetric finite element model was used to represent the single-disc clutch system at the start of the engagement. The results approved that frictional facing thickness has a significant effect on the distribution of the frictional heat generated and the actual contact area of friction clutches.

1. Introduction

The friction clutch is considered to be an important facet of a standard power transmission system. The contact surfaces of the friction clutch system are subjected to high temperatures during the starting period of engagement as they attempt to transfer the required torque to the driven shaft. The interaction between the contact pressure, sliding speed, and friction characteristics of the contact surfaces thus determines the distribution of heat generated, and consequently the surface temperature distribution. Thus, accurate calculation of the heat generated on the contact surface is the key to creating successful designs for friction clutches.

Figure 1 illustrates a single-disc friction clutch system during the engaging and disengaging processes. The main elements of the single-disc clutch system are the pressure plate, friction clutch disc and flywheel. This type of friction clutch has two effective friction surfaces, one on each sides. When force is applied to start the engagement between the driving and driven parts, heat is generated due to the difference in the speeds between the driving and driven parts. The surface temperature increasea dramatically during this phase (Figure1 a). Once, all parts are rotating together at the same speed the heat generated decreases and drop to zero. This phase is called full engagement as shown in Figure 2.

Abdullah et al. [1-8] investigated thermal and thermo-elastic problems in dry friction clutches during the sliding phase by means of several different approaches. The effects of changing design parameters such as sliding speed and applied pressure on temperature distribution were studied using a finite element method and experimental work. Two and three-dimensional finite element models were developed to achieve numerical simulation. The distributions of contact pressure between the elements of the friction clutch were computed during the engagement period when various functions of torque were applied, while in the experimental work, a pin-on-disc test rig with infrared camera was used to find the maximum surface temperature under various boundary conditions. The effects of sliding time, surface roughness and restricted conditions on the thermal behaviours of friction clutches through the beginning of engagement were investigated in detail.

In [9] experimental outcomes for the frictional response of current materials used for brake and clutch facings over adequate ranges of slip speed and contact pressure were analysed, and models were created to achieve benefit from frictional maps in terms of developing control algorithms.

In [10], a two-dimensional closed-form solution for the temperature field at the interface between the metal and lining surfaces was proposed by using a moving slab theoretical model subject to heat sources from friction energy dissipation. The setting of boundary conditions involved linear application of wall heat flux that was dependent on the wall temperature and allowed the investigator to take into account both Dirichlet and Neumann boundary conditions.

In [11] a detailed FEA model was developed to achieve temperature field in an automotive-dry-clutch system during both start-up and gearshiftmanoeuvres. Such analysis highlights the fact that temperature increase can be as high as 30 to 40 K in a single manoeuvre; thus, repeated launch manoeuvres could trigger critical values for clutch facing materials.

In [12,13], the FE analysis aimed at deepening understanding of the temperature rise ensuing after several clutch slipping operations. Results from the latter paper proved that even in the engaged phase without any slip between clutch surfaces and flywheel/ pressure plate, the convective flow toward the clutch housing air environment is not adequate to completely dissipate the thermal energy generated by earlier engagement.

The present research paper thus aims to investigate the effects of frictional facing thickness on heat generated during the sliding period. An axisymmetric finite element model was developed to seek at solution of the thermo-elastic problem of a single-disc friction clutch working under dry conditions. Constant applied pressure was assumed during the whole process of engagement and the convection effect was also taken into consideration.

2. Finite Element Simulation

The simulation of the thermo-elastic problem in dry friction clutches using the finite element method will be explained in more detail in this section. In order to reduce the time consuming for calculations and the complexity of the finite element model, an axisymmetric finite element model was selected to represent the friction clutch system. This model is valid because of the symmetry existing in the boundary conditions and geometry (where the selected friction clutch disc is without grooves). Figure 2 and 3 show the thermal and elastic finite element models with boundary conditions where $Q_{gen.f}$, $Q_{gen.c}$ and $Q_{gen.p}$ are the rate of heat entering into the flywheel, clutch disc and pressure plate, respectively. Here, h is the convective heat transfer coefficient, and the rate of the heat generated between two rubbing surfaces per unit area at any time is [8]

$$q(r,t) = \mu \; p \; \omega \; r \qquad ,0 \le t \le t_s \tag{1}$$

where t_s, μ, p, ω and r *are* the sliding time, coefficient of the friction, contact pressure, angular sliding speed and radius, respectively. The sliding angular speed decreases linearly with time according to the following formula

$$\omega(t) = \omega_o (1 - \frac{t}{t_s}) \qquad ,0 \le t \le t_s \tag{2}$$

where ω_o is the initial angular sliding speed. Transient thermal conduction and the elastic coupled problem must be solved simultaneously to obtain a solution for the thermo-elastic problem in the friction clutches. The contact pressure distribution p (r, t) can then be computed when given solution of the elastic model using the given temperature distribution $T(r,z,t)$. This computation is based on Hook's law with a thermal strain relationship as follows [14]

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$$\varepsilon_{ij} = \frac{(1+\nu)}{E}\sigma_{ij} - \left(\frac{\nu}{E}\sigma_{mm} + \alpha T\right)\delta_{ij} \tag{3}$$

where υ, E, σ, α and δ_{ij} are Poisson's ratio, Young's modulus [N/m^2], a stress component [N/m^2], thermal expansion [K^{-1}] and the Kronecker delta, respectively. The equilibrium stress is [14],

$$\frac{\partial \sigma_{ij}}{\partial x_j} = 0 \tag{4}$$

The obtained contact pressure $p(r,t)$ from the elastic analysis can be used to calculate the frictional heat generated $q(r,t)$ on the contacting surface [equation (1)]. The obtained heat generated can then be used in the transient heat conduction analysis to represent the Thermal Load on the contact surfaces. The next step is to find the solution of the equation for transient heat conduction to obtain temperature distribution $T(r, z, t + \delta t)$ as,

$$\nabla^2 T = \frac{1}{k}\frac{\partial T}{\partial t} \tag{5}$$

where k is the thermal diffusivity ($k = K/\rho\, c$). K, ρ and c are the conductivity, density and specific heat, respectively. Figure 4 shows a flowchart of the developed approach using a finite element method to find the solution to the coupling problem (temperature and stress fields) of a single-disc friction clutch system. The developed approach consisted of two simulations; the elastic contact simulation was used to compute contact pressure distribution and thermal stresses. While the transient thermal simulation was used to calculate the temperature distribution at each instant during the heating phase. Finite element models of the single-disc clutch were thus developed by using ANSYS APDL to conduct numerical analysis. The selected materials simulated for the friction clutch system were homogeneous and isotropic; Table 1 lists the functional parameters, dimensions and material properties of the selected materials. The convection coefficient was 40.89 W/m^2 K [8] and this was held constant over all exposed surfaces. The element used for the thermal model was PLANE55, an element that consists of four-nodes; for each node one degree of freedom (Temperature). For the elastic model, the element has 4 nodes coupled-field (thermal and structural fields) solid element with upto 4 DOF per node. Conta172 and Targe169 elements were used for the contact surfaces ad target surfaces, respectively.

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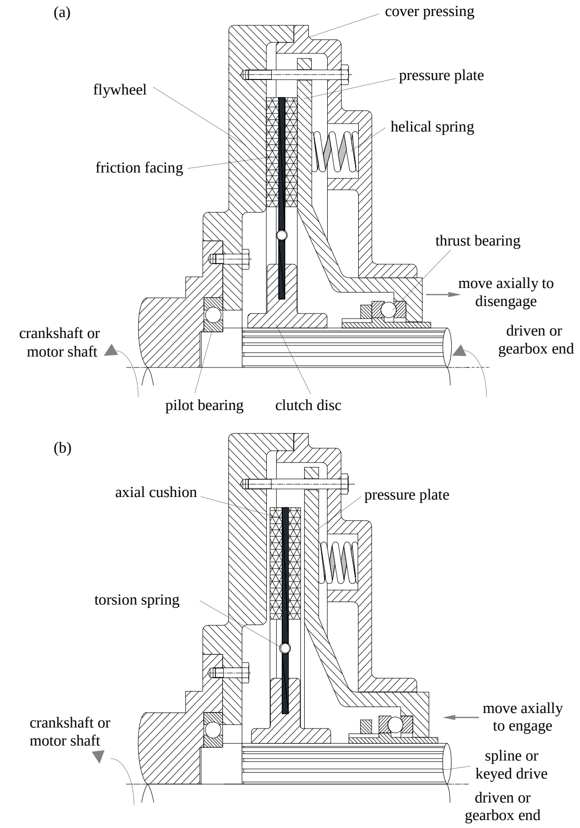

Figure 1. Engagement and disengagement in single-disc clutch system
(a) Engaged, (b) Disengaged

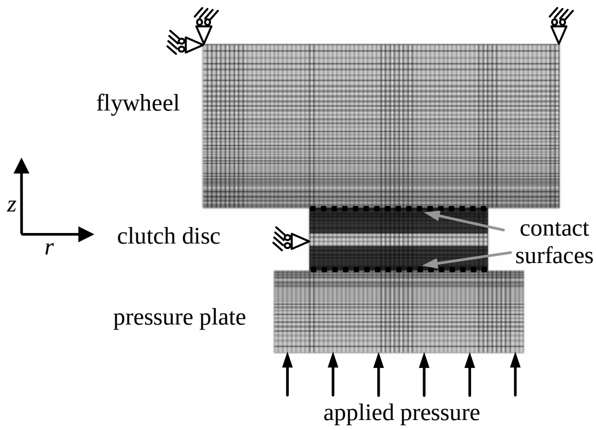

Figure 2. Finite element model with boundary conditions (contact model). [No. of elements = 5488]

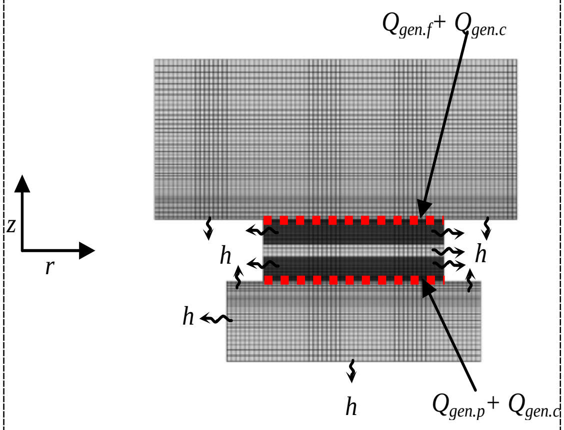

Figure 3. Finite element model with boundary conditions (thermal model). [No. of elements = 5488]

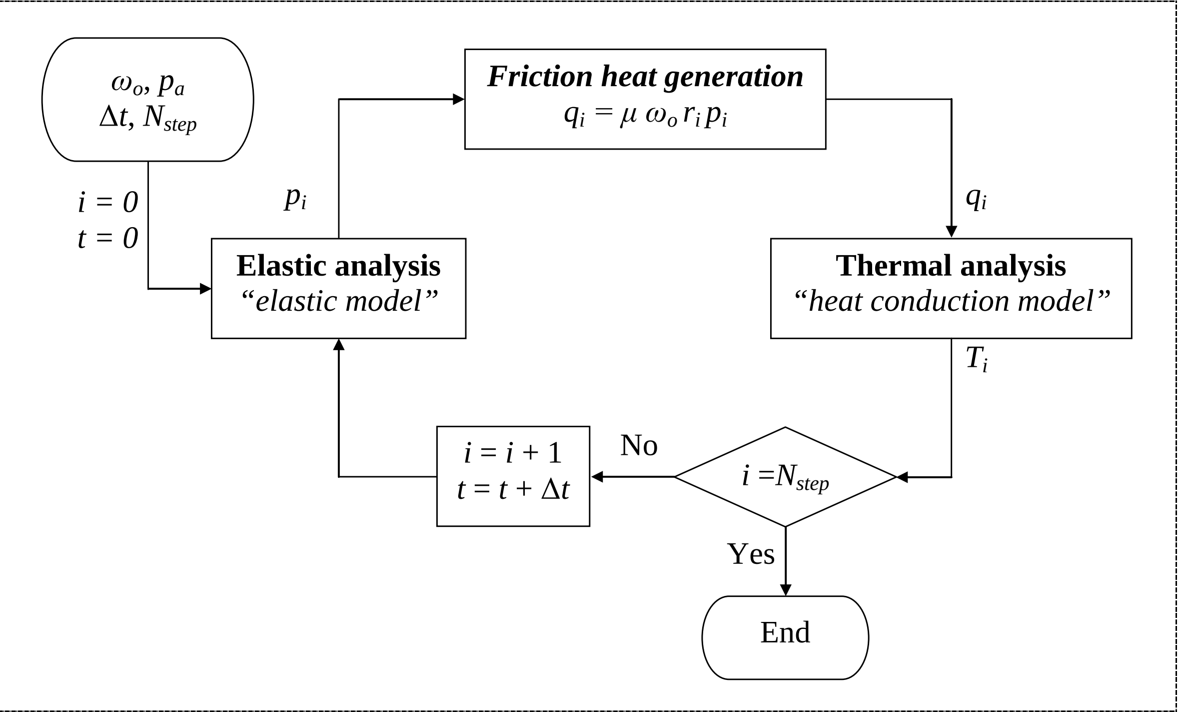

Figure 4. Flowchart: thermal - elastic coupling.

Table 1. Operational parameters and material properties of the friction clutch system [4]

Parameters	Values
Inner radius of friction material and axial cushion, [m]	0.064
Outer radius of friction material and axial cushion, [m]	0.087
Thickness of axial cushion, [m]	0.0015
Inner radius of pressure plate, [m]	0.058
Outer radius of pressure plate, [m]	0.092
Thickness of pressure plate, [m]	0.0097
Inner radius of flywheel, [m]	0.0485
Outer radius of flywheel, [m]	0.097
Thickness of flywheel, [m]	0.0194
Applied pressure, p_a [MPa]	1
Coefficient of friction, μ	0.2
Maximum angular slipping speed, ω_o [rad/sec]	290
Number of friction surfaces, n	2
Young's modulus of friction material, [GPa]	0.30
Young's modulus of pressure plate, flywheel and axial cushion, [GPa]	200
Poisson's ratio of friction material, [GPa]	0.25
Poisson's ratio of pressure plate, flywheel and axial cushion [1]	0.25
Density of friction material, [kg/m^3]	2000
Density of pressure plate, flywheel and axial cushion, [kg/m^3]	7800

Specific heat of friction material, [J/kg K]	120
Specific heat of pressure plate, flywheel and axial cushion, [J/kg K]	532
Conductivity of friction material, [W/mK]	1
Conductivity of pressure plate, flywheel and axial cushion, [W/mK]	54
Thermal expansion of friction material and steel, [K^{-1}]	12×10^{-6}
Slipping time, t_s [s]	0.4

3. Results and Discussions

Analysis of the thermo-elastic problem in the dry friction clutches was done to study the effects of frictional facing thickness on the heat generated during the beginning of engagement (heating phase). Three different frictional facing thicknesses were selected (1, 2 and 3mm) to achieve the numerical simulations discussed in this paper.

Figures 5-7 illustrate the distribution of the frictional heat generated during the heating phase using different frictional facing thickness. It can be noticed that a huge amount of heat is generated in a semi-uniform distribution at the starting time (t = 0) due to the high contact pressure and sliding speed. After a very short time when the effect of thermal-contact coupling begins, the distribution of the heat generated changes, and the highest values occurred in the zone located between the inner and mean radii. While the lowest values appeared in the zone located between the mean and outer radii. The maximum values of frictional heat generated occured at approximately at the middle time of the sliding period ($t \approx 0.2t_s$) which is located at the inner radius for all cases. At the end of the heating phase (t = 0.4s), all parts of the friction clutch system rotate together at the same speed, consequently the frictional heat generated reached zero at the end of the heating phase. This can be attributed the considerable change in the distribution of the heat generated during the heating phase, due to the high thermal deformations and contact pressure which occurred during this phase. It was found that the percentage of reduction in frictional heat generated was 6.5% when the frictional facing thickness was amended from 1 mm to 3 mm.

Also, it can be seen from the results that the actual contact area increases significantly when the frictional facing thickness increases. Figure 8 shows the percentage of actual contact area when using different frictional facing thicknesses. The actual contact area increases by approximately 20% when the frictional facing thickness is changed from 1mm to 3mm.

Figure 9 illustrates the variation of maximum surface temperature on the clutch disc’s surface which faces the pressure plate throughout the heating phase. It was found that the highest temperature occurred when frictional facing with 1mm thickness was used, and the lowest temperatures occurred when used frictional facing with 3mm thickness. The difference between temperatures using frictional facings of 1 mm and 3 mm thicknesses increased with time until t= 0.28 s, then decreased. The maximum difference of 30 K occurred at t= 0.28 s when the frictional facing thickness was changed from 1 mm to 3 mm.

Generally, when the frictional facing thickness is thin, thermal deformation will have a more significant impact on the distribution of contact pressure, leading to concentrate the pressure in certain zones from the nominal contact area. As a result of this situation, the frictional heat generated (temperature) will be increased and the contact area will be decreased. On the other hand, when the frictional facing thickness is thick, the distribution of contact pressure will be more uniform over the contact surfaces and thermal deformations will have less effect on the contact pressure, leading to a reduction in the level of frictional heat generated (temperature) at the contact surfaces of the clutch.

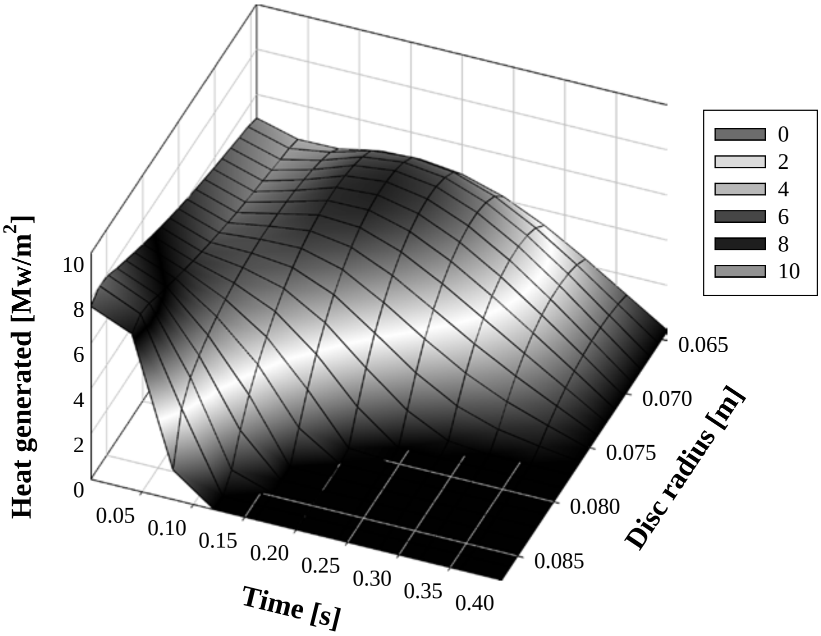

Figure 5. Variation of frictional heat generated on the friction clutch disc with radius and time (pressure plate side, t_f=1mm).

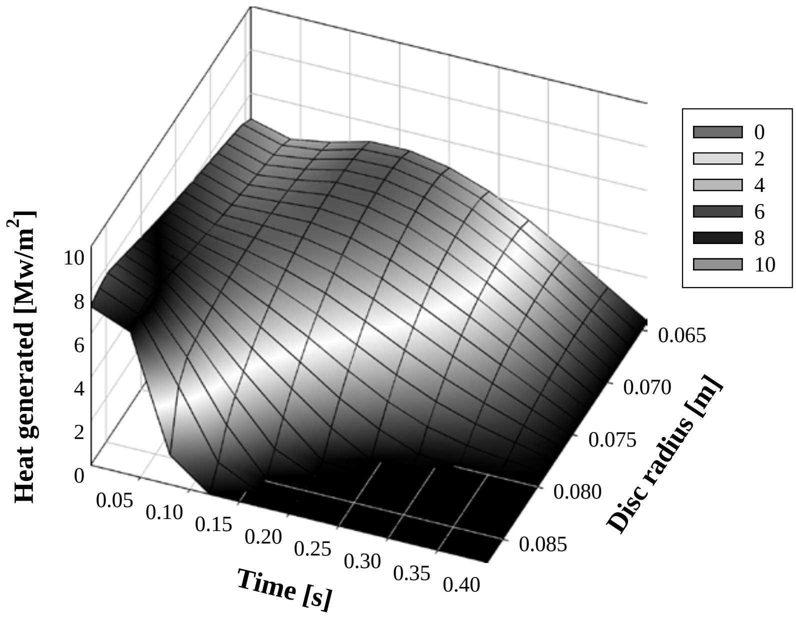

Figure 6. Variation of frictional heat generated on the friction clutch disc with radius and time (pressure plate side, t_f=2mm).

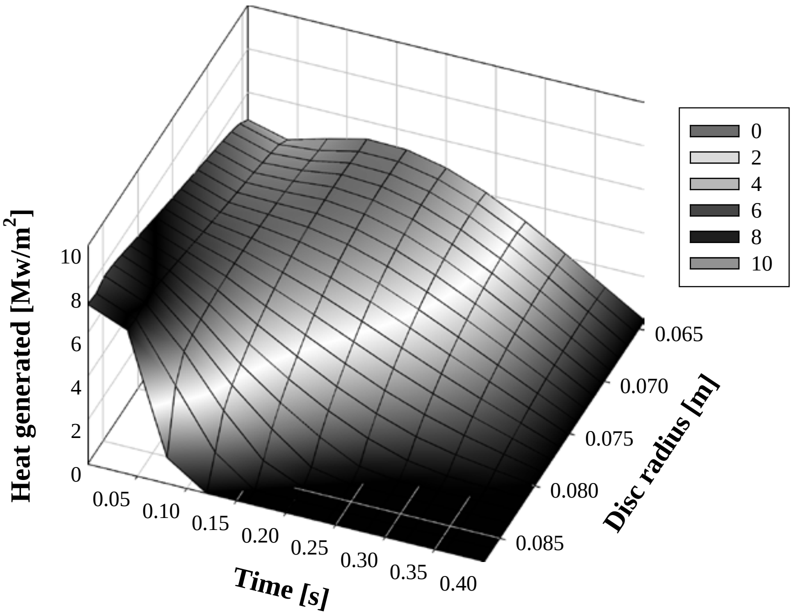

Figure 7. Variation of frictional heat generated on the friction clutch disc with radius and time (pressure plate side, t_f=3mm).

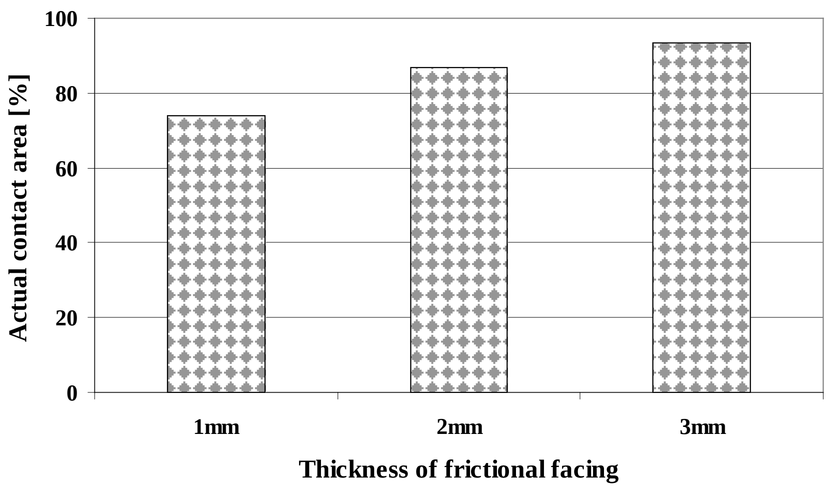

Figure 8. Percentage of actual contact area with different frictional facing thickness (t =0.2s).

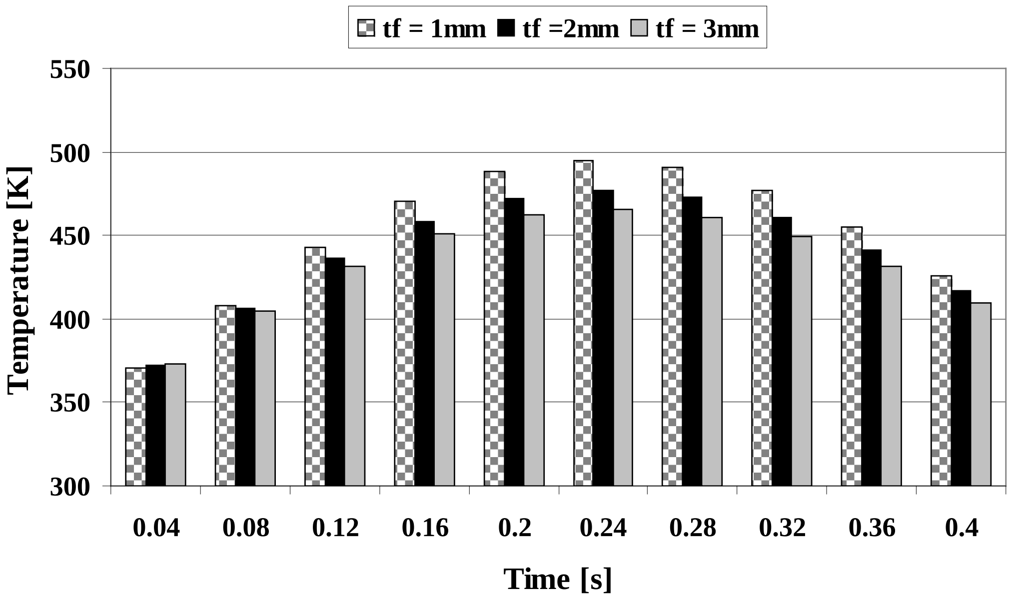

Figure 9. Maximum surface temperature of the friction clutch disc with different frictional facing thicknesses.

4. Conclusions and Remarks

This research paper has presented a solution of the thermal-elastic coupling problem which occurrs during the heating phase of dry friction clutches. Finite element modelling was applied using the axisymmetric model developed to conduct numerical simulation. The results presented the distribution of heat generated, temperatures and actual contact area. The rapid growth in thermal deformations at the beginning of the heating phase were seen to lead to changes in contact pressure, and consequently the distribution of heat generated changed from semi-uniform to non-uniform on the contact surface.

The thickness of the friction facing was found to a significant factor that dramatically affects the distribution of the heat generated, contact pressure, temperature and actual contact area. Thus, it should be considered vital to determine the critical frictional facing thickness for each specific case, taking into consideration factors such as the power transmitted, maximum sliding speed, and allowable stresses.

5. References

[1] Abdullah O and Schlattmann J 2017 *Tribol T.* **60(6)** 1011-1021
[2] Abdullah O and Schlattmann J 2016 *Friction* **4(3)**1-10
[3] Abdullah O and Schlattmann J 2016 *Friction* **4(2)**1-9
[4] Abdullah O , Sabri L, and Al-Sahb W 2016 *Tribologia* **2016(2)** 9-24
[5] Abdullah, O, Schlattmann J and Lytkin M. 2015 *FME Transactions* **43(3)** 241-248
[6] Abdullah O and Schlattmann J 2014 *J Tribol* **136(3)** 1-6
[7] Abdullah O and Schlattmann J 2014 *Int J Auto Tech-Kor* **15(**5**)**733-740
[8] Abdullah O and Schlattmann J 2014 *J Balk Tribol Assoc* **20(2)** 184-198
[9] D'Agostino V, Cappetti N, Pisaturo M and Senatore A 2012 *Proc. Int. Conf. on Mechanical Engineering Congress & Exposition*, Transportation Systems (Texas, USA) vol 11, p9
[10] Senatore A, D'Agostino V, Giuda R Di and Petrone V 2011 *Tribol. Int.* **44(10)** 1199-1207
[11] Cuccurullo G., D'Agostino V., Giuda R. Di and Senatore A. 2011 *Meccanica* **46** 589-595
[12] Pisaturo M and Senatore A 2016 *Appl. Therm. Eng.* **93** 958 - 966
[13] Pisaturo M, Senatore A and D'Agostino V 2016 *Proc. Int. Conf. on Intelligent Engineering Systems* (Budapest) p 69
[14] Boresi A P, Schmidt R.J and Sidebottom OM 1993 Advanced Mechanics of Materials, *Wiley* (New York)