Vol. 34, No. 4 (2012) 206-216

Tribology in Industry

www.tribology.fink.rs

RESEARCH

Finite Element Analysis of Temperature Field in Automotive Dry Friction Clutch

O.I. Abdullah [a], J. Schlattmann [a]

[a] System Technology and Mechanical Design Methodology, Hamburg University of Technology, Germany

Keywords:

Dry friction clutch
Frictional heating
Temperature field
2D FEM

Corresponding author:

Oday I. Abdullah
System Technology and Mechanical Design Methodology
Hamburg University of Technology
Germany
E-mail:
oday.abdullah@tu-harburg.de

A B S T R A C T

The friction clutch design is strongly dependent upon the frictional heat generated between contact surfaces during the slipping at beginning of engagement. Because of that the frictional heat generated firstly will reduce the performance of clutch system and then will lead to premature failure in some cases. Finite element method was used to investigate an effect of thermal load type on the temperature field of the clutch system. Two-dimensional axisymmetric model was used to study the temperature distribution for the clutch system (pressure plate, clutch disc and flywheel) during heating phase (slipping period) and in the cooling phase (full engagement period). Depending on basic friction clutch design two types of thermal loads were applied; load type A (uniform pressure) and load type B (uniform wear). Repeated engagements made at regular interval were considered in this work. ANSYS13 has been used to perform the numerical calculation in this paper.

1. INTRODUCTION

Automotive clutches and brakes are frequently subjected to high thermal loads, and this load will occur as a result of slipping between contact surfaces. The surface temperatures may cause serious disadvantages such as thermal cracks and permanent deformation, and these conditions will lead to premature failure for the sliding systems in some cases.

Al-Shabibi and Barber [1] investigated alternative method to solve the thermoelastic contact problem with frictional heat generation. Two-dimensional axisymmetric finite element model built to study the temperature field and pressure distribution of two sliding disks. Constant and varying speeds were considered in this analysis. The results show that the initial temperature is shown to be crucial since it represents the particular solution, which can have quite irregular form, this situation especially true when the system operates above the critical speed.

Ivanović et al. [2] presented a pragmatic semi-physical approach to modeling the thermal dynamic behaviour of wet clutch. The thermal energy balance was considered the base to investigate the heat transfer mechanisms in the separator plate. The coefficient of friction behaviour and thermal dynamics are considered the most important parameters which effect on the wet clutch dynamics response. Moreover, the effect of coefficient of friction on the clutch slips speed, applied force and friction surface temperatures have been studied. The results of the dynamic thermal model were experimentally validated.

Zagrodzki [3] studied the frictional heating in the sliding systems and the effect of sliding speed on the stability of the system when the sliding speed exceeds the critical value. Finite element model used to investigate the transient thermo elastic process, spatial discretization and modal superposition is presented. Constant sliding speed was performed in this analysis. The transient salutation includes both of homogenous part (corresponding to the initial condition) and non-homogenous part (represent the background process). The results show that the important parameters which contributors in the background process are the nominal process equivalent to uniform pressure distribution in isothermal case and the other is the pressure variation caused by geometric imperfection or by design features.

Jang and Khonsari [4] presented new developed model to predicting the rate of growth of instability in a conductor-insulator system in the presence of a liquid lubricant and found a relation between the wave speed and the configurations of hot spots. The analysis includes term for roughness of surface and is capable of treating bodies of finite thickness with or without lubrication. The results show that the shape of hot spots is more likely to be circular with an appreciable penetration depth when the wave speed is small, and the hot spots is elongated when the wave speed is much slower than the operating speed

Zhang et al. [5] developed new model for wet clutches in hydrodynamic machineries to computes the thermal behaviour with high accuracy. Matlab/Simulink was used to build the model and find the temperature field during engagement. The shift schedule, piston pressure and relative velocity parameters considered in this simulink model. Two-dimensional heat conduction model was used to calculate the temperature distribution along the axial direction and radial direction on the sliding interface between contact surfaces. The results have shown a good agreement with the experimental work.

Seo et al. [6] suggested thermal model to estimate the temperature distribution for wet clutch in 4WD coupling to avoid thermal failure for clutch plate during operating condition of vehicle. The results of this model are validated experimental of actual 4WD vehicle under limitation of torque 900 [N.m]. The theoretical results have shown a good agreement with the experimental results.

Czél et al. [7] used finite element method for dry ceramic clutch disk to find the thermal behaviour and comparison with experimental measurement results. The heat partition factor and heat convection coefficient were assumed to be changing in time and space. A distributed heat source was applied for modeling heat generation. The thermal results have shown a good agreement with the measured temperature.

This paper presents the full details of temperature distribution for dry friction system (flywheel, clutch disc and pressure plate) during repeated engagements. The finite element method used to compute the temperature field for clutch disc under different pressure distributions. Two types of pressure distributions were applied between contact surfaces uniform pressure and the linear pressure with disc radius (maximum at inner radius and minimum at outer radius) based on theory of clutch design. The effect of convection is considered in both cases.

2. FRICTIONAL HEAT GENERATED BETWEEN CONTACT SURFACES

The friction clutch system consists of pressure plate, clutch disc and flywheel as shown in Fig. 1 and when the clutch start to engagement the slipping will occurs between contact surfaces due to the difference in the velocities between their. A high amount of the kinetic energy converted into heat energy at interfaces according to the first low of thermodynamics. The heat generated between contact surfaces will dissipate by the conduction between friction clutch components and by convection to environment. It can represent the friction clutch system (without grooves) with two-dimensional axisymmetric model due to the symmetry.

In this work, it was assumed that the thermal properties of materials are isotropic and independent of temperature. The actual contact area is equal to the nominal contact area. Two types of pressure distribution are applied between contact surfaces, the uniform pressure (load tape A) and the linear pressure distribution (load type B) according to the basic design theory of the friction clutch. Assuming that the total friction energy is converted into heat, the total heat generated during the slipping is given as follows [8],

$$Q_t(r,t) = Q_{gen.f} + Q_{gen.c} = Q_{gen.p} + Q_{gen.c} = \mu\, p V_s; \quad 0 \le t \le t_s \tag{1}$$

Where, $Q_{gen.f}$, $Q_{gen.c}$, $Q_{gen.p}$ and V_s ($V_s=\omega_s$ r) are the heat generated on the flywheel, heat generated on the clutch disc, heat generated on the pressure plate and the sliding velocity and ω_s is the sliding angular velocity (rad/sec) respectively. Assuming that the sliding angular velocity decreases linearly with time as,

$$\omega_s(t) = \omega_o\left(1-\frac{t}{t_s}\right), \quad 0 \le t \le t_s \tag{2}$$

Where ω_o is the initial sliding angular velocity when the clutch starts to slip (t=0). The total heat generated between contact surfaces when assume the pressure distribution on contact surfaces is uniform (load A) at any time of slipping is (Fig. 2),

$$Q_{t.u}(r,t) = \mu\, p\, r\, \omega_o\left(1-\frac{t}{t_s}\right); \quad 0 \le t \le t_s \tag{3}$$

And the total heat generated between contact surfaces when assume the pressure distribution on contact surfaces is linear with disc radius (load B) at any time of slipping is (Fig. 3),

$$Q_{t.l}(t) = \mu\, C\, \omega_o\left(1-\frac{t}{t_s}\right); \quad 0 \le t \le t_s \tag{4}$$

Where C = $p_{max.}\, r_i$. The heat conduction equation for an axisymmetric problem in the cylindrical coordinate (r, z) used to obtained temperature distributions with time for the friction clutch system,

$$\frac{\partial}{\partial r}\left(K\frac{\partial T}{\partial r}\right) + \frac{K}{r}\frac{\partial T}{\partial r} + \frac{\partial}{\partial r}\left(K\frac{\partial T}{\partial z}\right) = \rho c\,\frac{\partial T}{\partial t}; \; t>0 \tag{5}$$

Where: r is the radial coordinate, z is the axial coordinate, K is the thermal conductivity, ρ is the density and c is the specific heat. There are two methods used to calculate the temperature field in the automotive clutches and brakes are:

1. Using a heat partition ratio to compute the heat generated for each part individually [9].
2. Apply the total heat generated for whole model by using the contact model [10].

The contact model's approach is used in this work, and assumes perfect thermal contact between contact surfaces or in the other word the surface temperatures are equal in the interface for both bodies.

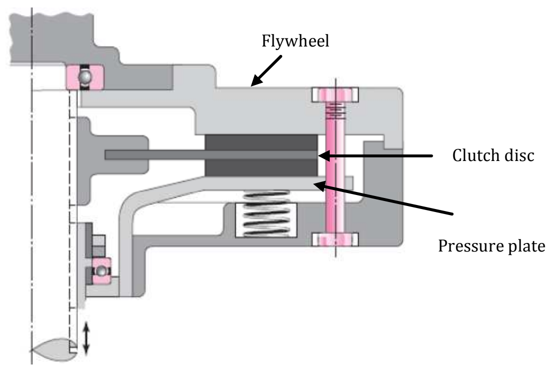


Fig. 1. The friction clutch elements.

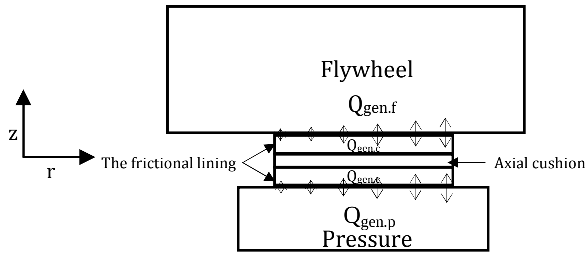


Fig. 2. The Contact model for clutch system (Uniform pressure/ load A).

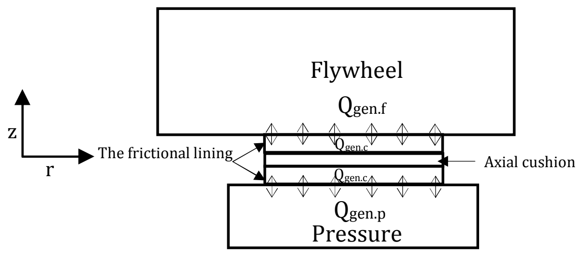


Fig. 3. The Contact model for clutch system (Uniform wear/ load B).

3. FINITE ELEMENT FORMULATION

Transient condition involved time dependent function of the heat transfer analysis. During the transient condition, the temperature change in a unit volume of material is resisted by thermal mass that depends on the mass density ρ of the material and its specific heat c. The finite element formulation can be expressed as [11],

$$[C]\{\dot{T}\}+[K]\{T\}=\{F\} \qquad (6)$$

Where, [C] is the specific heat matrix, [K] the conductivity matrix, {T} the vector of nodal temperatures, $\{\dot{T}\}$ is the derivative of temperature with time ($\partial\dot{T}/\partial t$), and {F} the applied heat flows. The axisymmetric finite element model of the friction clutch system with boundary conditions is shown in Fig. 4. In this paper ANSYS software was used to investigate transient thermal response behavior for dry friction clutch. Four-noded thermal element (PLANE55) was used in this analysis. A mesh sensitivity study was done to choose the optimum mesh from computational accuracy point of view (Fig. 5). The Crank-Nicolson method was selected as an unconditionally stable scheme. In all computations for the friction clutch model, it has been assumed a homogeneous and isotropic material and all parameters and materials properties are listed in Table 1. The heat transfer coefficient has been taken as 40.89 W/m^2 K [12] and is assumed to be constant over all exposed surfaces, and the slipping time is 0.4 s for all engagements.

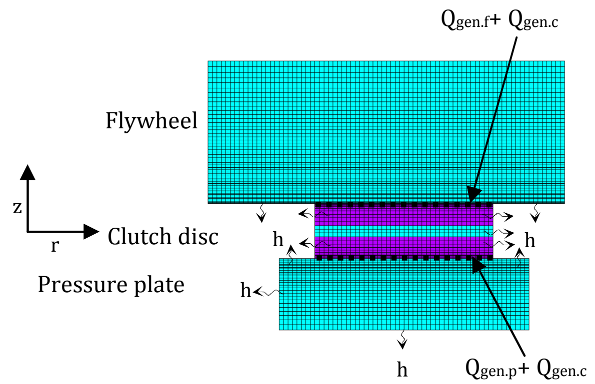


Fig. 4. FE models with the boundary conditions. (No. of elements = 6488).

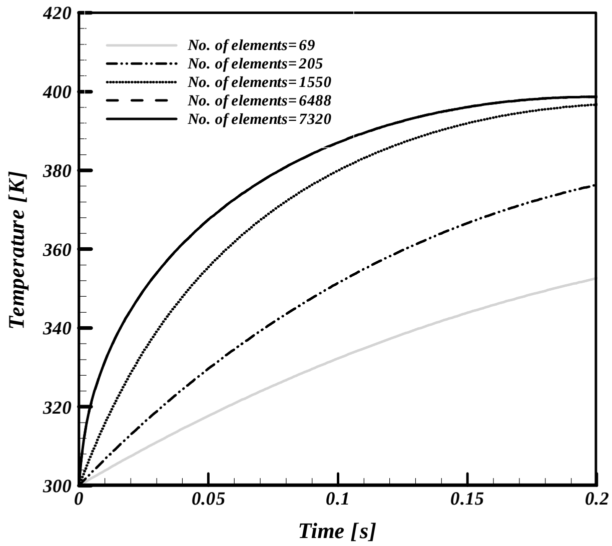


Fig. 5. Variation of maximum temperature with time for different No. of elements.

Table 1. The model parameters and material properties.

Parameters	Values
Inner radius of friction material & axial cushion , r_i [m]	0.06298
Outer radius of friction material & axial cushion , r_o [m]	0.08721
Thickness of friction material [m]	0.003
Thickness of the axial cushion [m]	0.0015
Inner radius of pressure plate [m]	0.05814
Outer radius of pressure plate [m]	0.09205
Thickness of the pressure plate [m]	0.00969
Inner radius of flywheel [m]	0.04845
Outer radius of flywheel [m]	0.0969
Thickness of the flywheel [m]	0.01938
Maximum pressure, $p_{max.}$ [MPa]	1
Coefficient of friction, μ	0.3
Number of friction surfaces	2
Torque [Nm]	432
Maximum angular slipping speed, ω_o (rad/sec)	200
Conductivity for friction material, (W/mK)	0.6
Conductivity for pressure plate & flywheel, Kp & Kf (W/mK)	42
Density for friction material, (kg/m^3)	1570
Density for pressure plate, flywheel & axial coushion, (kg/m^3)	7800
Specific heat for friction material, (J/kg K)	534
Specific heat for pressure plate, flywheel & axial coushion, (J/kg K)	450
Intial temperature, Ti [K]	300
Time step, Δt (s)	0.001
Number of engagemants, n	10

3. RESULTS AND DISCUSSIONS

In the practical application the friction clutch makes repeated engagements, and the temperature fields (especially the maximum temperature) during these engagements are consider essential for designer. The temperature distributions were computed during 10 repeated engagements at regular interval (5 S) for the same energy dissipation. Two types of load are considered in this work, load type (A) when assume the pressure is uniform between contact surfaces and load type (B) when assume the rate of wear is uniform.

The approach of present work was compared with the numerical results of Ref. [13] to find the maximum temperature ($T_{max.}$) at inner and outer radius of friction clutch. Table 3 shows the current results with numerical results of [13], and the values of percentage difference with numerical results. In this table, the maximum difference not exceeds (1 %). The data for the verification case are shown in Table 2.

Table 2. The parameters and material properties for verification case [13].

Properties	Friction material properties	Steel properties
Dimensional parameters	interior diameter, D1=252 mm, external diameter, D2=386mm; thickness, t2=5 mm	thickness t1=10mm t3=5mm
Modulus of elasticity (MPa)	70	70 2×10^5
Thermal conductivity (W/mK)	0.25	48
Specific heat capacity (J/kg K)	1337.6	480
Density, (kg/m^3)	1300	7800

Table 3. The values of maximum temperature at inner and outer radius.

	$T_{max.}$ at r_i [K]	$T_{max.}$ at r_o [K]
Present work	533.3	639.5
Ref. [13]	536.4	642.6
% Difference	0.57	0.48

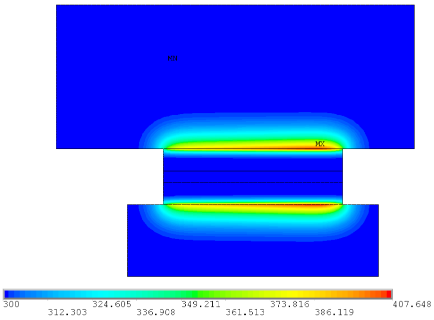
The temperature distributions of clutch elements for both cases (load A & B) during repeated engagements are shown in Fig. (6).

From this figure, it can be seen that the values of maximum temperatures when applied load (A) are greater than the values of maximum temperature when applied load (B) during all engagements and under the same conditions. The maximum temperatures when applied load (A) are located near the outer radius (r_o) and the temperatures when applied load (B) are distributed uniformly in most contact area. The temperature decreases in regions which located at inner and outer radius of clutch disc because of the effect of convection on these regions.

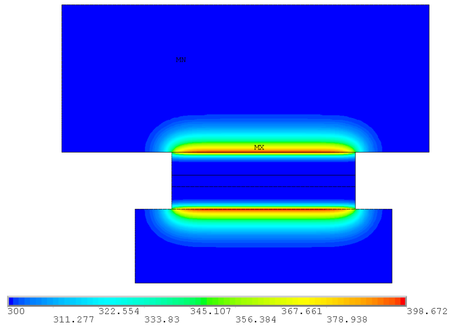
Figures 7 and 8 show the temperature distributions in different time (0.2, 11 & 48.8 s) over cross-section of friction clutch elements at r_i when applied loads A & B. It can be seen for both figures that the temperatures at pressure plate and flywheel interfaces are equal approximately in the first engagement (the difference between them less than 0.5 %) but after the first engagement the temperatures of pressure plate interface are greater than the temperatures at flywheel interface; this results because of low thermal capacity of pressure plate compared with the flywheel, and this situation will lead to the temperature of pressure plate increases rapidly and reaches to high values with short time. Also, it's clear that the values of temperatures when applied load (B) are greater than the values of temperature when applied load (A) during all engagements.

The maximum values of temperature change from (342.6 K & 350.5 K) during the first engagement to the (470.1 K & 480 K) during the tenth engagement corresponding to load A and load B respectively.

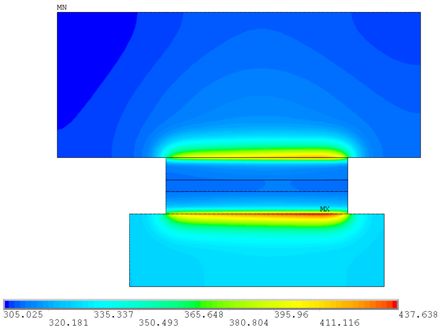
Figures 9 and 10 present the temperature distributions over cross-section of friction clutch elements at mean disc radius (r_m) for different time (0.2, 11 & 48.8 s) when applied load A & B. It can be seen from these figures that the values of temperatures when applied load (A) are approximately equal to the values of temperature when applied load (B) at any time (the difference between them less than 0.5 %). The maximum values of temperature change from 398 K during the first engagement to the 528 K during the tenth engagement for both cases (A &B).



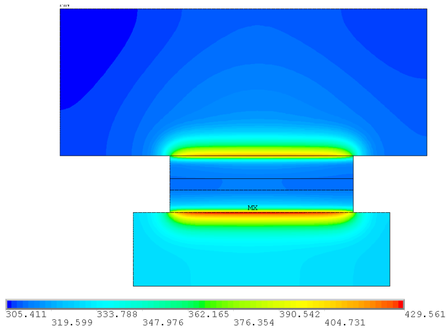
(a) n=1 (t=0.2 s) /load A



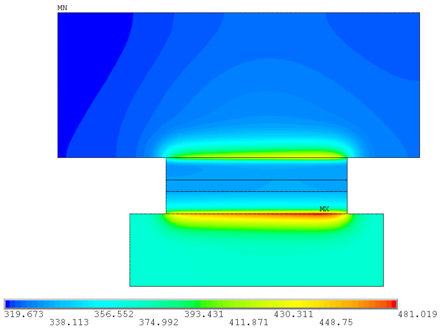
(b) n=1 (t=0.2 s) /load B



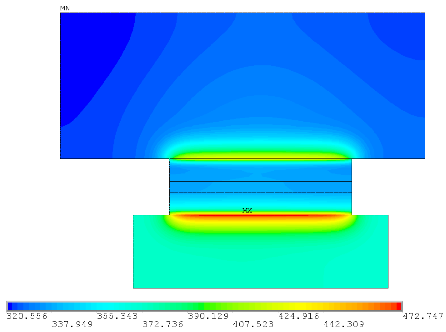
(c) n=3 (t=11 s) /load A



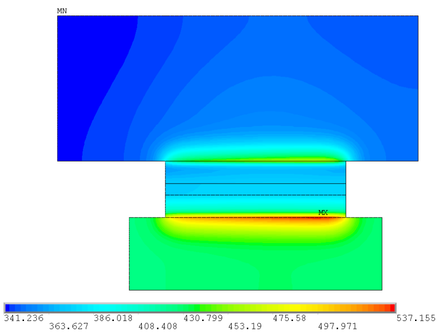
(d) n=3 (t=11 s) /load B



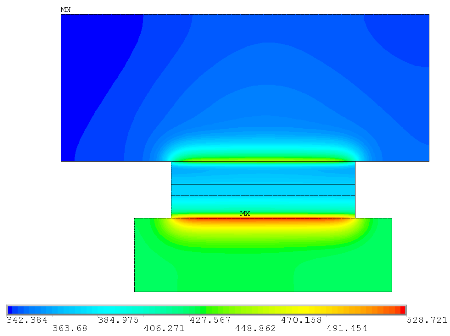
(e) n=6 (t=27.2 s) /load A



(f) n=6 (t=27.2 s) /load B



(g) n=10 (t=48.8 s) /load A



(h) n=10 (t=48.8 s) /load B

Fig. 6. Temperature distribution of the friction clutch elements at different number of engagement.

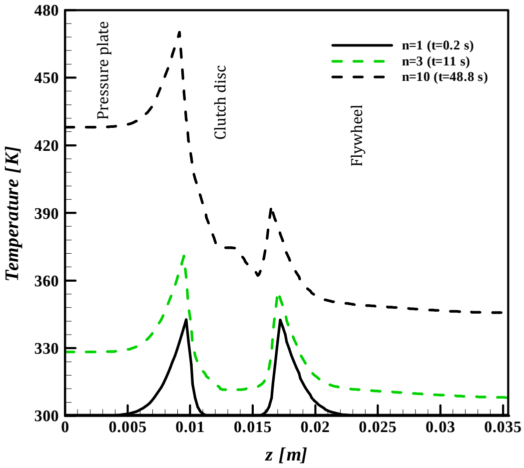


Fig. 7. Temperature of distribution in cross-section at r_i of friction clutch elements (load A).

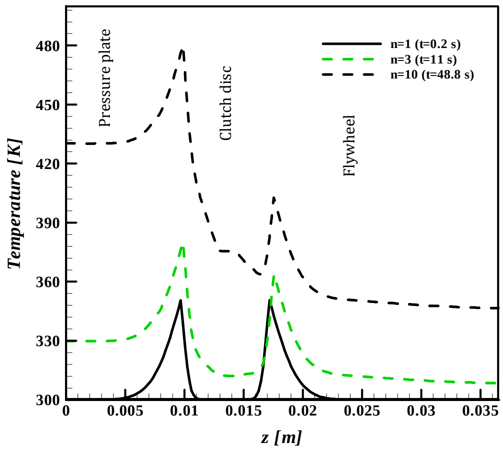


Fig. 8. Temperature of distribution in cross-section at r_i of friction clutch elements (load B).

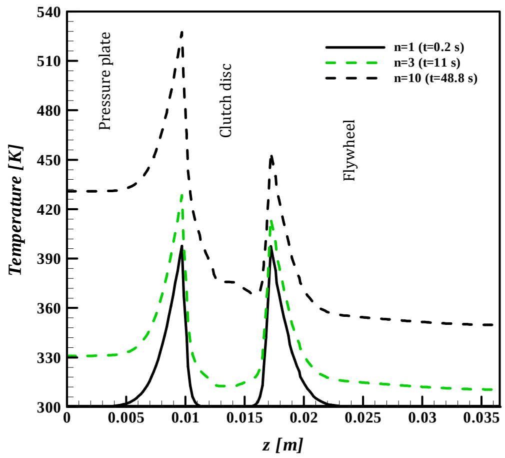


Fig. 9. Temperature of distribution in the cross-section at r_m of friction clutch elements (load A).

The temperature distributions over cross-section of friction clutch elements at r_o for different time (0.2, 11 & 48.8 s) are shown in Figs. 11 and 12 for both types of load (A & B). It can be seen from these figures that the values of temperatures when applied load (A) are greater than the values of temperature when applied load (B) during all engagements. The maximum values of temperature change from (356.7 K & 349.7 K) during the first engagement to the (486.1 K & 477.5 K) during the tenth engagement corresponding to the load A and load B respectively.

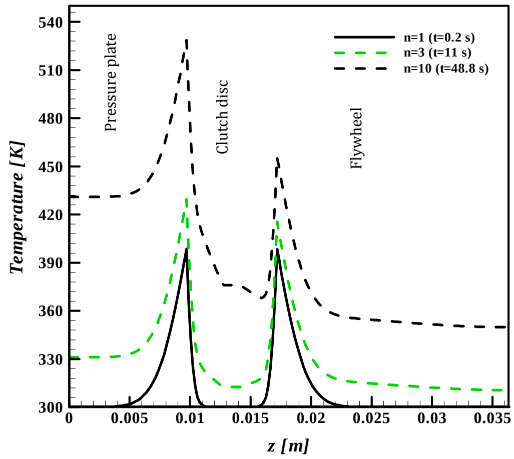


Fig. 10 Temperature of distribution in the cross-section at r_m of friction clutch elements (load B).

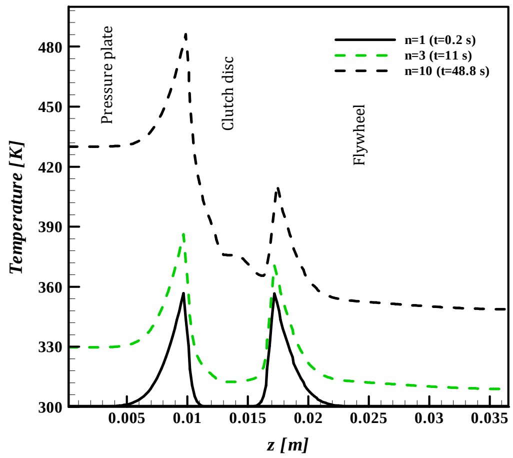


Fig. 11. Temperature of distribution in the cross-section at r_o of friction clutch elements (load A).

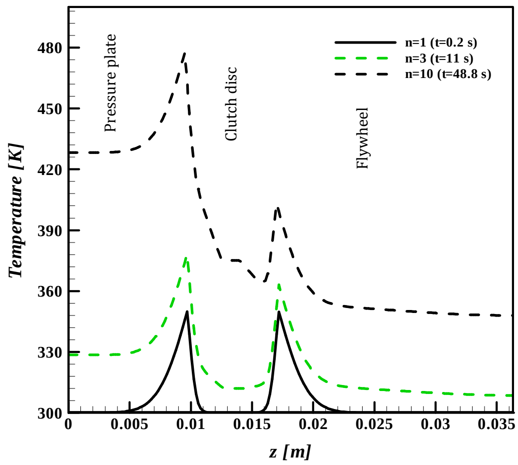


Fig. 12. Temperature of distribution in the cross-section at r_o of friction clutch elements (load B).

Figs. 13 and 14 demonstrate the temperature distributions with disc radius at contact region between flywheel and clutch disc at different engagements for load types A & B. It can be seen from these figures, that the values of temperatures when applied load (A) are greater than the values of temperatures when applied load (B) during all engagements. The maximum temperature when applied load (A) is located at r = 0.95 r_o, also it can be seen when applies load (B) that the temperatures are uniformly distributed except the regions near the inner and outer radius due to the effect of convection. The maximum values of temperature change from (407.5 K & 398.6 K) during the first engagement to the (463.8 K & 456.1 K) during the tenth engagement corresponding to load A and load B respectively.

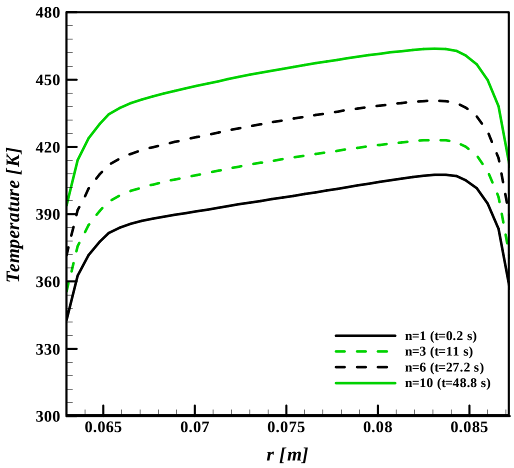


Fig. 13. Temperature of distribution with disc radius (flywheel/ clutch disc- load A).

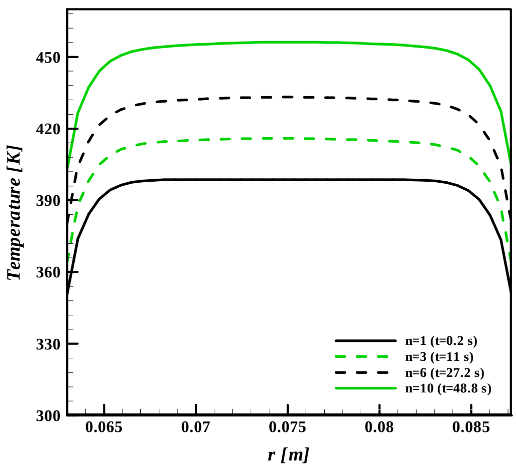


Fig. 14. Temperature of distribution with disc radius (flywheel/ clutch disc- load B).

The temperature distributions with disc radius at contact region between the pressure plate and the clutch disc at different time are shown in Figs. 15 and 16. These figures have the same behaviors with Figs. 13 and 14 but the range of the values of temperature are higher for both types of load, these results because of the lower thermal capacity of pressure plate compared with the flywheel. The maximum values of temperature change from (407.5 K & 398.6 K) during the first engagement to the (537.09 K & 528.7 K) during the tenth engagement corresponding to load A and load B respectively.

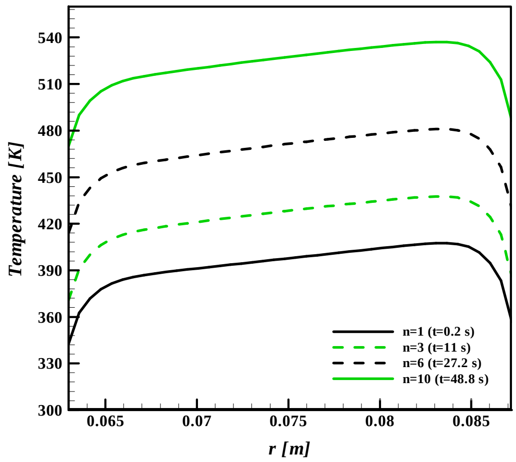


Fig. 15. Temperature of distribution with disc radius (pressure plate/ clutch disc- load A).

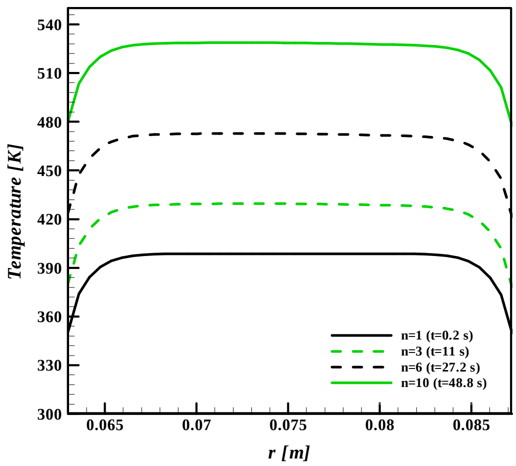


Fig. 16. Temperature of distribution with disc radius (pressure plate/ clutch disc- load B).

Figures 17 and 18 show the variation of temperature with time during 10 engagements at three locations of the contact area between the flywheel and clutch disc (r_i, r_m and r_o) when applied loads A and B. During all engagements, it can be observed that the differences between the maximum temperatures of both cases (A & B) are very small (less than 1 K). The maximum temperatures reached after 10 engagements are

455.2 K and 456.1 K corresponding to the load (A) and load (B) respectively.

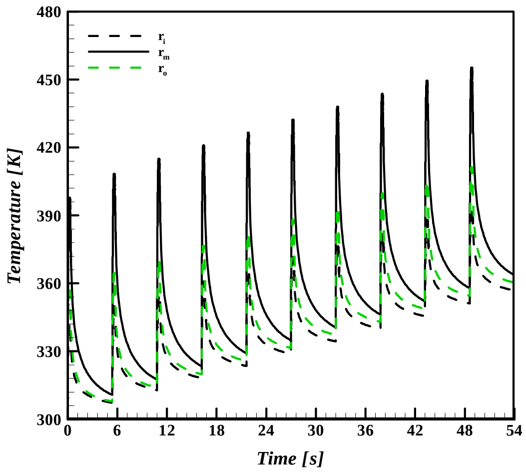


Fig. 17. The variation of temperature with time (Flywheel/ clutch disc-load A).

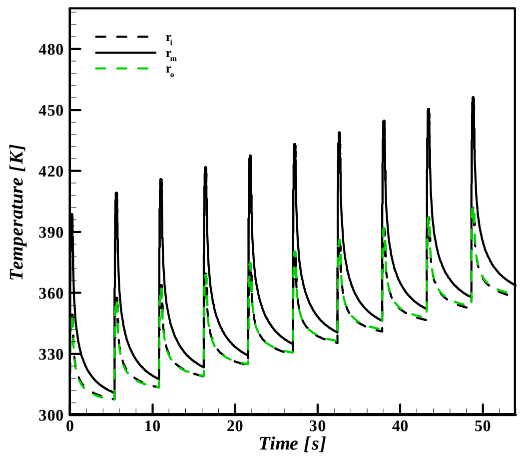


Fig. 18. The variation of temperature with time (Flywheel/ clutch disc-load B).

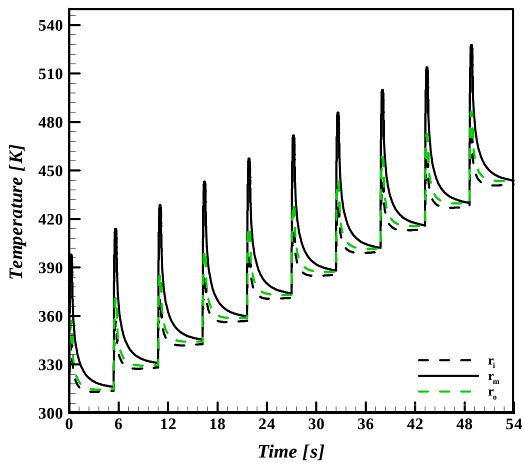


Fig. 19. The variation of temperature with time (Pressure plate/ clutch disc-load A).

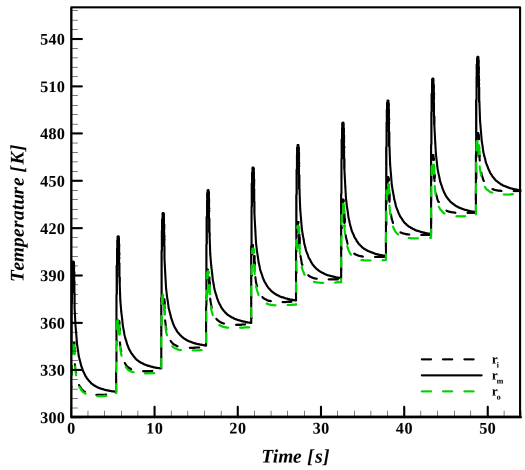


Fig. 20. The variation of temperature with time (Pressure plate/ clutch disc-load B).

The variation of temperature during 10 engagements for different location of disc radius at the contact area between the pressure plate and clutch disc for loads (A & B) are shown in Figs. 19 and 20. During all engagements, it can be observed that the same behaviour with Figs. 17 & 18 but the range of temperatures are higher. The maximum temperatures reached after 10 engagements are 527.6 K and 528.5 K corresponding to the load (A) and load (B) respectively.

4. CONCLUSIONS AND REMARKS

In this paper the transient thermal analysis of a dry friction clutch system during 10 repeated engagements based on the uniform pressure and uniform wear theories was performed. Two-dimensional axisymmetric model was built to obtain the numerical simulation for friction clutch elements during the slipping.

The temperature will increase rapidly when the number of engagements increase and in some cases the temperature exceeds the maximum limit of temperature, this situation lead to friction clutch failure before the expected lifetime, therefore the study of the temperature field of contact surfaces during repeated engagements operation is necessary to give an indication about the maximum temperature during the engagements.

It can be seen from results, that the values of maximum temperature when applied load (A) are higher than the values of temperature when applied load (B) and the difference between them is approximately 8 k at any engagement, these results because of the load (A) is function of radius and the intensity of thermal load will focusing near outer radius. Furthermore, it can be noticed when applied load (B) the uniform temperature on the contact surface because of the thermal load is constant with radial direction. Generally, the difference between the maximum temperatures between cases (A & B) depends on the disc radius ratio R ($R=r_i/r_o$), when R is small the difference between the maximum temperatures between two cases (A & B) is high and when the R increases the difference between them will decreases. The maximum temperature for both cases occurs approximately in the middle of slipping time for all engagements.

The maximum temperatures occur on the pressure plate and flywheel are approximately equal at the first engagement but in the subsequent engagements the maximum temperature of the pressure plate will be higher than the maximum temperature of flywheel and the difference between them will increase with increases of engagement's number. The difference between maximum temperatures of pressure plate and flywheel are (14.5 K, 40.3 K & 73.3 K) corresponding to the engagement number (3, 6 & 10) respectively when applied load type (A), and when applied load (B) the difference between the maximum temperatures of pressure plate and flywheel are (13.7 K, 39.6 K & 72.5 K) corresponding to the engagement number (3, 6 & 10) respectively, the reason for these differences in temperatures is the low thermal capacity of pressure plate compared with the flywheel, and to reduce the temperatures range of pressure plate have to increase the thickness of pressure plate and this solution is considered very expensive from point of view of production. Increases the quantity of heat transfer by convection is consider the other solution to reduce the temperature of friction clutch system, therefore it's important to select the suitable design for clutch to increase the exposed area of the clutch (e.g grooves) to increase the heat transfer to environment.

The sliding speed plays an important role in determining the amount of heat generated between contact surfaces; therefore it's necessary to know the maximum sliding velocity in application, to select the suitable friction material to withstand the high temperature which produces by frictional heating during the slipping.

The temperature distributions of friction clutch are considered essential first step to calculate the amount of internal energy of the friction clutch due to the slipping during a process of engagement. The precise calculation for each of the rate of wear [14-16] and the internal energy of friction clutch will lead to Knowledge the lifetime of friction material with high accuracy.

REFERENCES

[1] Abdullah M. Al-Shabibi and James R. Barber: *Transient Solution of the Unperturbed Thermoelastic Contact Problem*, Journal of Thermal Stresses, Vol. 32, No. 3, pp. 226-243, 2009.

[2] Vladimir Ivanović, Zvonko Herold and Joško Deur: *Experimental Characterization of Wet Clutch Friction Behaviors Including Thermal Dynamics*, SAE International Journal of Engines, Vol. 2, No. 1, pp. 1211-1220, 2009.

[3] Przemyslaw Zagrodzki: *Thermoelastic instability in friction clutches and brakes- Transient modal analysis revealing mechanisms of excitation of unstable modes*, Int. J. of Solids and Structures, Vol. 46, pp. 2463–2476, 2009.

[4] J.Y. Jang and M.M. Khonsari: *On the Growth Rate of Thermoelastic Instability*, Transactions of the ASME, Vol. 126, pp. 50-55, 2004.

[5] Jin-Le Zhang, Biao Ma, Ying-Feng Zhang and He-Yan Li: *Simulation and Experimental Studies on the Temperature Field of a Wet Shift Clutch during one Engagement*, in: *International Conference of Computational Intelligence and Software Eng*, CiSE) 2009 IEEE, 11-13.12.2009, pp. 1-5.

[6] Howon Seo, Chunhua Zheng, Wonsik Lim, Suk Won Cha and Sangchull Han: *The results have shown a good agreement with the experimental work*, in: *Vehicle Power and Propulsion Conference (VPPC)*, 2011 IEEE.

[7] Balázs Czél, Károly Váradi, Albert Albers and Michael Mitariu: *Fe thermal analysis of a ceramic clutch*, Tribology International, Vol 42, No. 5, pp. 714–723, 2009.

[8] Oday I. Abdullah and Josef Schlattmann: *The Effect of Disc Radius on Heat Flux and Temperature Distribution in Friction Clutches*, J. Advanced Materials Research, Vol. 505, pp. 154-164, 2012.

[9] Oday I. Abdullah and Josef Schlattmann: *Finite Element Analysis of Dry Friction Clutch with Radial and Circumferential Grooves*, in: *World Academy of Science, Engineering and Technology conference*, April 25-26 2012, Paris, France, pp. 1279-1291.

[10] L. Wawrzonek, R.A. Bialecki: *Temperature in a disk brake, simulation and experimental verification*, Int. J. of Numerical Methods for Heat & Fluid Flow, Vol. 18, No. 3-4, pp. 387-400, 2008.

[11] Robert Cook: *Finite Element Modeling for Stress Analysis*, John Wiley & Sons, 1995.

[12] Oday I. Abdullah and Josef 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.

[13] Hua Fu, Li Fu, Ai-ping Liu and Guan-Lei Zhang: *Finite Element Analysis of temperature field of clutch in Tunnel Boring Machine*, in: *2010 WASE International Conference on Information Engineering*, 14-15 Aug. 2010, pp. 166-169.

[14] M.A. Chowdhury, D.M. Nuruzzaman, A.H. Mia, M.L. Rahaman: *Friction Coefficient of Different Material Pairs under Different Normal Loads and Sliding Velocities*, Tribology in Industry, Vol. 34, No. 1, pp. 18-23, 2012.

[15] Birleanu Corina and Sucala Felicia: *About the tribological behavior of ceramic materials*, Tribology in Industry, Vol. 30, No. 3&4, pp. 10-14, 2008.

[16] V. Bria, D. Dima, G. Andrei, I.-G. Birsan, A. Circiumaru: *Tribological and Wear Properties of Multi-Layered Materials*, Tribology in Industry, Vol. 33, No. 3, pp. 104-109, 2011.