Finite Element Simulation of Thermal Behavior of Dry Friction Clutch System during the Slipping Period

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Abstract

Most of failures in the friction clutches occur due to the excessive heat generated due to friction between various parts, and this heat causes high temperatures leading to high thermal stresses. In the present research paper, numerical simulation had been developed using finite element method to simulate the thermal behavior of the dry friction clutch. Three-dimensional finite element model was made and analyzed using ANSYS/Workbench sofware18. The friction clutch system was firstly modeled mathematically and solved numerically to determine the transient thermal response of the clutch disc. The two fundamental methods of uniform wear and uniform pressure are assumed. The applied torque during the sliding period was constant. The temperature and heat generated were estimated for each clutch part (pressure plate, clutch disc and flywheel) using heat partition ratio. The assumptions that are inherent in the derivation of the governing equations are presented which followed up by the appropriate boundary conditions. The results show that the maximum temperature values for uniform pressure condition are greater than those for uniform wear condition. Also, the temperature value increased with time and approximately reaches the highest value at the middle of the sliding period when the applied torque is constant with time and then decreased to the final values at the end of slipping period.

Keywords: Dry friction clutch, thermal analysis, 3D FEM

I.   Introduction

The temperatures of the rubbing surfaces increased due to the heat generated during sliding between two contacting surfaces. Temperature rise during the sliding period has a considerable influence on the lifetime of the sliding surfaces of clutch. High thermal stresses will be produced at the contact surface due to increased values of surface temperatures. When the clutch continues working under these conditions, this situation causes surface cracks and permanent distortions. Therefore it is necessary to know the magnitude of the maximum surface temperature to prevent

clutch failure before the lifetime and how this depends on known conditions of loading.

Reference [III] applied the FEM to investigate the influence of the thermo mechanical loads on the pressure and hub plates. Three different loads have been implemented: the thermal load for the slipping, the contact pressure of diaphragm spring and centrifugal force. The results of the investigation showed that the thermal load has a significant effect on the values of temperatures and stresses; therefore, it is recommended to use a pressure plate with higher thickness in order to have larger heat capacity which reduces the thermal stresses. Reference [V] implemented Green's function method to propose an analytical solution to determine the thermal performance for the pad and the disc for unsteady conditions with heat generation as a function of space and time. The research assumed that generated heat between the surfaces is dissipated to the surrounding in order to prevent the friction coefficient between elements of the brake from reduction and surface temperature from increasing. Reference [I] applied the transient finite technique to study the thermal behavior of a dry ceramic clutch disc. It has been hypothesized that the heat distribution factor and heat convection coefficient are functions of space and time. The generated heat on the contact surface was modeled using a distributed heat source. Thermal results showed an agreeable convention with the measured temperature. Reference [XI] presented analytical and numerical solution for the transient heat conduction problem. The surface temperature of the tribos was calculated for the disc surface with time variant speed. The thermal stresses were modeled by using two dimensional finite elements to investigate the influence of the boundary conditions on the temperature of the tribological system. It was found that the value of thermal stress declines as the braking time increases. Reference [II] used the finite element method in the investigation of thermal stresses in the brake system. They estimated the contact pressure on pads of the brake and stress fields failure built up in the disc. Reference [VII] investigated the heat generation in the friction surfaces and the temperature profile by using finite element method. The investigation was for the case of contact bands between pressure plate and clutch disc and for the case of contact bands between the clutch disc and the flywheel. Also the influence of pressure capacity on the temperature was studied as the pressure is varies with time. The results proved that both the sliding time evolution and the contact area ratio massively affect disc clutch temperature fields in time domain. It was concluded that the temperature distribution is the highest for case of constant pressure as compared with other pressure types. This can be attributed to the short time of applying thermal load as compared with other thermal loads types.

Later, Reference [VIII] estimated the transient surface temperature of a dry friction clutch of a single plate assuming that the torque is time dependent and the wear is uniform. It was presented the equations for energy dissipation between surfaces in contact. The temperature distributions were compared for cases when the torque is constant and time dependent. It was found that the temperature is higher for the case of constant torque because the short time of thermal load connecting. Furthermore, they introduced mathematical models for to predict the heat and

temperature distribution during the period of slipping. In their work that the heat flux and wear assumed to be uniform distributed over the clutch contact surface at any time [IX]. They also Reference [X] investigated the influence of friction material thickness of clutch disc that consists of multiple friction discs on the values and distribution of the contact pressure for each surface of friction discs at the beginning of engagement using finite element method. The results showed that the thickness of the frictional lining material of a clutch disc is a significant parameter that influences the elastic and thermal behaviors of a dry friction clutch. The results proved that the magnitudes of the contact pressure increased dramatically when the thickness of the friction facing decreases. Reference [IV] studied the temperature distribution for three models of clutches during the sliding time and for a single engagement. It was used different types of friction materials, the values of initial angular velocity and the pressure exerted by the pressure plate to investigate the influence of these variations on the sliding system. Also, it was analyzed and modifies the model of clutch disc to minimize the friction energy that converted to heat due to conduction and convection in order to obtain a better clutch model. Reference [XII] used FEM to analyze the deformation, stress concentration, elastic strain, thermal gradient and heat flux distribution of a copper alloy friction lining and structural steel friction lining of a clutch plate. It was observed that copper alloy frictional lining material of clutch plate dissipated frictional heat at a faster rate than structural steel frictional lining of clutch disc. The design was accomplished using Solid Works 2016 and the finite element analysis was carried out using ANSYS 16.0. Reference [VI] enhanced the analytical solution to determine heat flux dissipated during the sliding period between two rubbing surface of dry friction clutch at the first engagement using equation of motion of two inertia system. It was assumed a uniform pressure condition. Also, it was investigated the effect of variation of torque and angular sliding velocity with time during the slip period. The results showed that the heat flux is increased with disc radius when assumed the pressure is uniformly distributed over the contact surface.

The aim of the present research paper is to determine the frictional heat generated and temperature history for the clutch disc, flywheel and pressure plate; during the slipping period using ANSYS/Workbench software 18 depending on the fundamental design theories for friction clutch (uniform wear and uniform pressure). The torque during the sliding period was assumed constant with time.

II.   Problem Formulation

The heat generated due to friction between friction clutch components is dissipated to environment by convection and between the clutch's parts by conduction. In this analysis, a mathematical model was proposed and solved numerically with the uniform pressure and uniform wear conditions. Isotropic materials were assumed with independence of temperature thermal properties.

During slipping period, total heat generated is given as:

$$Q_t = Q_{gen.f} + Q_{gen.c} = Q_{gen.p} + Q_{gen.c} = \mu p v \quad ; \quad 0 \leq t \leq t_s \qquad (1)$$

$$= 0; t > t_s$$

Where;

$$\text{v} = \omega_r * \text{r} \qquad (2)$$

Assuming linear decrease with time relation for the relative angular velocity; that is,

$$\omega_r(t) = \omega_{ro}(1 - \frac{t}{t_s}) \qquad (3)$$

Assuming that the pressure is uniformly distributed on the rubbing surfaces, the total heat generated based on the friction force between rubbing surfaces during the sliding period is defined as **[VII]**:

$$Q(t,r) = \mu pr\omega_{ro}(1 - \frac{t}{t_s}); \;\; 0 \le t \le t_s \qquad (4)$$

$$p = \frac{3T_o}{2\pi\mu n_p(r_o^3 - r_i^3)} \qquad (5)$$

And when assume uniform wear distribution, the total heat dissipated is function of time only and does not depend on clutch disc radius and it is uniformly distributed over it [VII]:

$$Q(t) = \mu C\omega_{ro}(1 - \frac{t}{t_s}); \;\; 0 \le t \le t_s \qquad (6)$$

Where;

$$C = p_{max} * r_i \qquad (7)$$

$$p_{max} = P\frac{r}{r_i} \qquad (8)$$

$$p = \frac{T_o}{\pi\mu \, n_p r(r_o^2 - r_i^2)} \qquad (9)$$

The total interface heat Q_t is divided between contacting surfaces as shown in Fig.1. The amounts of this heat flux entering into each parts of clutch are depending on the heat partition ratio (γ) [IX]. The heat flux enters into the clutch disc is:

$$Q_{gen.c} = \gamma Q_t \qquad (10)$$

The heat flux enters into the pressure plate and flywheel is,

$$Q_{gen.f} = Q_{gen.p} = (1 - \gamma)Q_t \qquad (11)$$

The formula that used to calculate heat partition ratio for automotive clutches is based on Charron's formula:

$$\gamma = \frac{\sqrt{K_1\rho_1 c_1}}{\sqrt{K_1\rho_1 c_1} + \sqrt{(K_2\rho_2 c_2)}} \qquad (12)$$

Subscripts (1 and 2)refer to the clutch disc, pressure plate or flywheel respectively (assume that the flywheel and pressure plate has the same material properties). The

heat conduction equation for three dimensional unsteady in the cylindrical coordinates (r, z, θ, t) was used to calculate the temperature history of the clutch elements.

III. Numerical Analysis

The FE technique was used to investigate the problem; it was used ANSYS/Workbench 18 to achieve the tasks of the research paper. The analysis covers different cases of working conditions.

The models of flywheel, frictional disks and pressure plate were prepared for simulations using ANSYS Design Modeler (DM) and SOLIDWORKS 2016 based on the shape and dimensions of actual clutch system. The parts of clutch can be seen in Fig.1. In the simulations, each part was investigated individually, where the applied boundary conditions satisfied the real case study. Due to symmetry, it was selected quarter part of each of the flywheel, frictional disc and pressure plate as shown in Fig.2.

One of the most important parts in any numerical simulation is the meshing. It can affect the accuracy and the time of the solution. Higher number of elements requires long time, and too few numbers of elements may lead to inaccurate results. The meshing tool of Ansys balances between the number of the elements and the accuracy to generate the suitable element grid for each simulation. The mesh generated for the models and enhancement of boundary layers grid with nodes statistics are explained in Fig.3. The mesh generated for each part within approximately 1,200,000 tetrahedral cells and with a high quality grid with element refinement for the faces exposed to heat flux condition.

IV. Case Study

The details of the selected cases that analyzed in this work can be summarized as follows:

- Inner disc radius, r_i = 0.064 m;
- Outer disc radius, r_o = 0.091m;
- Friction material thickness, t_c =0.003 m;
- Torque Capacity, T = 580 N.m;
- Friction coefficient, μ = 0.3;
- Maximum pressure, p_{max} = 1 MN/m2;
- Friction surfaces number, n_p = 2;
- Maximum angular slipping speed, ω_{ro} = 200 rad/s;
- Friction material conductivity, K_c= 0.75 W/m.K;
- Pressure plate and flywheel conductivity. K_p, and K_f = 56W/m.K;

- Friction material density, ρ_c =1300 kg/m3;
- Pressure plate and flywheel density, P_p and ρ_f =7200 kg/m3;
- Friction material specific heat, C_c=1400 J/kg.K;
- Pressure plate and flywheel specific heat, C_p and C_f =450 J/kg.K;
- Time step = 0.05s.
- Slip period ts = 0.4 s;
- Initial temperature Ti = 22°C at t = 0 s;
- The heat flux applied for a friction surfaces and convection surfaces as shown in Fig. 4 (a, b and c).

Figs. 5 and 6 show the heat flux distribution with time for clutch disc and flywheel or pressure plate respectively assuming uniform wear condition. Figs. 7 and 8 illustrate the heat distribution over the friction clutch cross section and flywheel or pressure plate respectively during the slipping period assuming that pressure is uniformly distributed over the cross section for the clutch element.

IV. Results and Discussions

A series of compactions were achieved to explore the problem under different working conditions.

Figs. 9 and 10 demonstrate the contour for temperature distribution on the frictional clutch disc with time at the end of slipping time for uniform wear and uniform pressure, respectively. Figs. 11 and 12 show the surface temperature history of clutch disc during slipping period under the same assumptions. It can be observed from these figures that the highest temperature values occurred for both assumptions (uniform wear and uniform pressure) at the middle of slip period (t_s = 0.2 s). In the case of uniform wear, the value of the maximum temperature is found T = 105.37°C at ts = 0.2s a, while the value of the maximum temperature at the slipping period end (t_s = 0.4 s) is found as 81.241°C.

In the case of uniform pressure, the value of the maximum temperature at the slipping period end (t_s = 0.4 s) is found as 109.32°C.

It is clear from the obtained results that the surface temperature values increased with time and approximately reaches the highest value in the middle of the sliding period when the applied torque is constant with time and then decreased to its final values at the slipping period end (t_s= 0.4 s). Also, it can be observed from these results that the highest value of temperature obtained when assumed the uniform pressure compared with uniform wear assumption under the same boundary condition.

Figs. 13- 16 show that the heat generated due to friction at the interface contact between clutch parts surfaces under uniform wear is less than those under the uniform pressure. The reason behind these results is thermal load under the uniform pressure is a function of disc radius, where the highest value of friction heat flux

occurred on the outer disc radius. These figures exhibit the heat generated on the frictional faces of the clutch disc during slipping period.

Figs. 17 and 18 show the temperature distribution over the clutch disc radius at the end of slipping period (t_s=0.4s) under the assumptions of uniform wear and uniform pressure distribution over the rubbing surfaces respectively. It can be observed from these results that the temperatures under the uniform wear condition were distributed uniformly over the clutch disc radius except the regions near the inner and outer radius due to the effect of convection. While, in the uniform pressure condition the temperature of the friction clutches is proportional to the disc radius (r). The minimum values of temperature obtained at inner disc radius and maximum values obtained at outer radius. It can be observed from results of uniform pressure condition that the temperature raised from T = 79°C at inner disc radius to T= 108 °C at outer disc radius, where the difference between the temperatures at inner and outer disc radius is 29°C.

Figs. 19- 22 demonstrate the temperature distribution during the slipping time and at the end of slip time over cross-section of flywheel part of frictional clutches. It is clear from these figures that the highest values of temperature approximately occurred at the middle of slipping period are found (T = 149.02°C) and (T = 120.08°C) according to uniform pressure and uniform wear conditions respectively. While at the end of slipping period the temperature decreased to the lowest values, which is (T = 97.428°C) for uniform pressure and (T = 96.006°C) for uniform wear condition.

Figs. 23- 26 present the temperature distribution during the slipping time and at the end of slip time over cross-section of pressure plate part of frictional clutches. It can be seen that the maximum values of temperature approximately occurred at the middle of slipping period are (T = 152.91°C) and (T = 108.59°C) for uniform pressure and uniform wear condition respectively. While, at the end of slip period the temperature of pressure plate surface decreased to the final value is (T = 141.38°C) for uniform pressure and (T = 100.92°C) for uniform wear condition.

Figs. 27 and 28 show the temperature distribution at t_s=0.2525s over cross section for the pressure plate surface at inner and outer radius under uniform wear and uniform pressure load. It is clear from these figures that the temperatures over flywheel and pressure plate surface when applied uniform pressure condition is higher than those appeared under the uniform wear theory.

V. Conclusions and Remarks

Numerical simulation has been developed using finite element method to study the transient thermal problem of a dry friction clutch assembly (pressure plate, clutch disc, and flywheel) during the sliding period (single engagement). It was assumed a constant torque with time function to obtain the numerical solution. It is concluded from the obtained results that the maximum temperature values for uniform pressure condition are greater than those for uniform wear condition. This is because the uniform pressure is a function of disc radius and the intensity of thermal

load is focused near outer radius. Also, for uniform wear assumption the frictional heat generated and the temperature distribution is function of contact pressure and sliding velocity for all cases. The temperature is distributed uniformly over the frictional clutch disc in the case of uniform wear, and it is not function of disc radius. Also, it was concluded that the temperature values increased with time and approximately reaches the highest value at the middle of the sliding period when the applied torque is constant with time and then decreased to the final values at the end of slipping period.
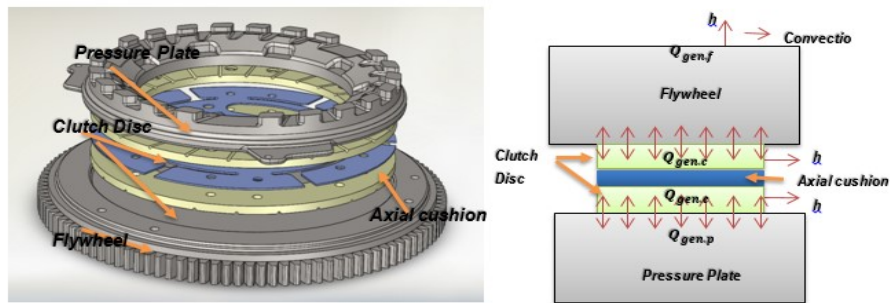
Fig.1: Main Parts of a Single-Disc Friction Clutch System with Boundary Condition
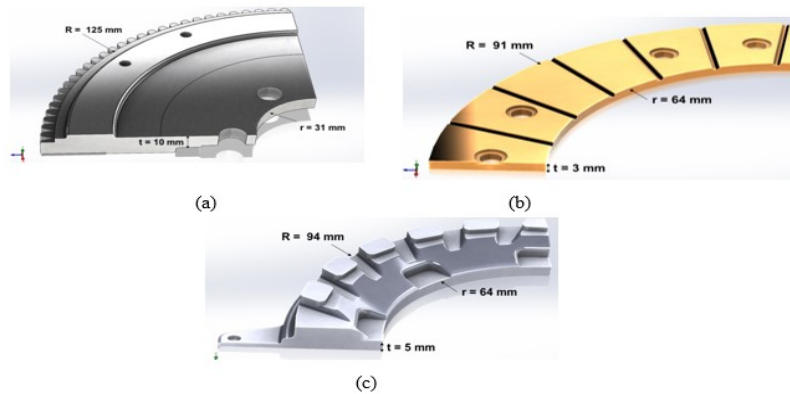
Fig.2: Clutch Components with Dimensions (a) Flywheel (b) Clutch Disk and (c) Pressure Plate.
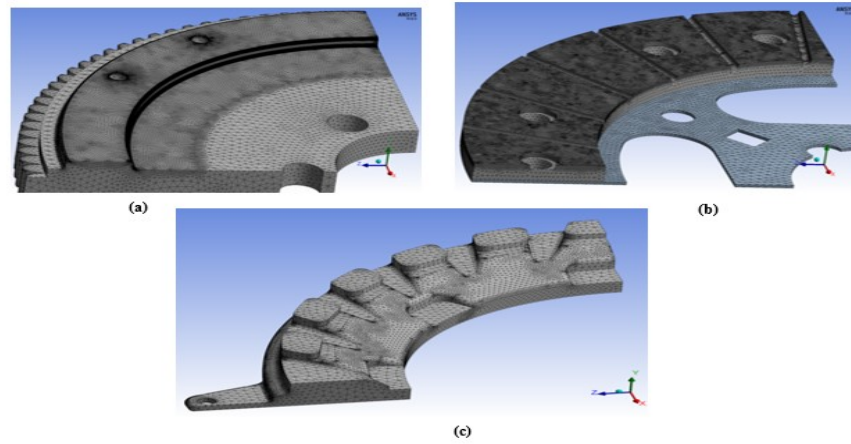
Fig.3: Grid Generation for Clutch Components with Mesh Statistics (a) Flywheel (b) Clutch Disk and (c) Pressure Plate.
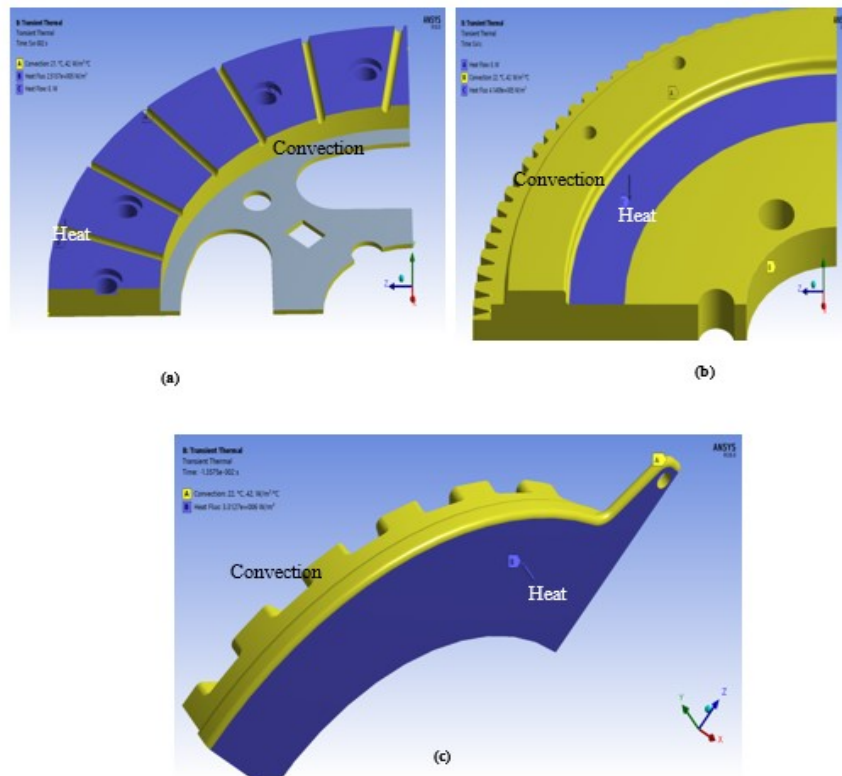
Fig.4: Boundary Conditions and Loading (a) Flywheel (b) Clutch Disk and (c) Pressure Plate
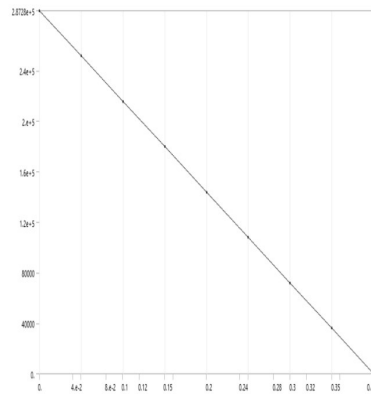
Fig.5: Evolution of Thermal Heat Flux in proportion to the
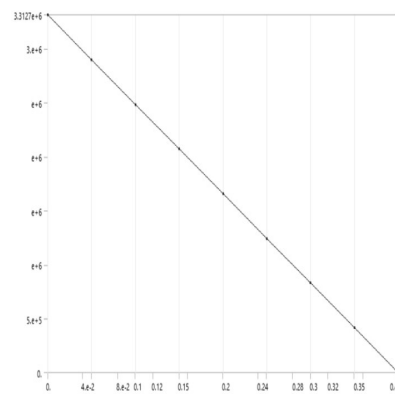
Fig.6: Evolution of Thermal HeatFlux in Proportion to the
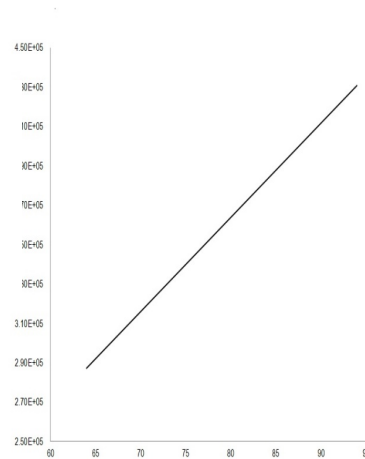
Fig.7: Evolution of Thermal HeatFlux in proportion to the
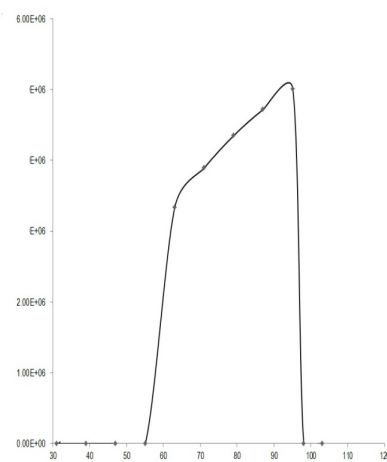
Fig.8: Evolution of Thermal Heat Flux in Proportion to
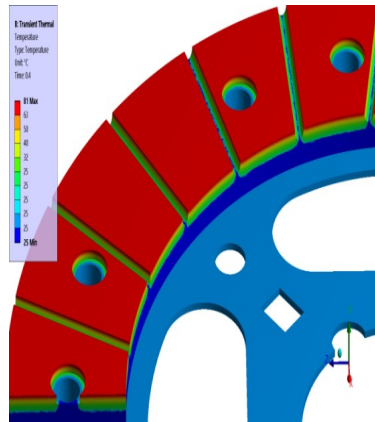
**Fig.9: Temp. Distribution
at the End of Slipping**
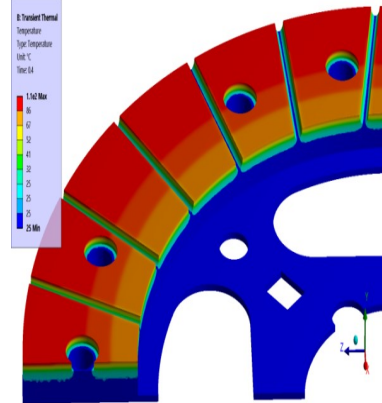
**Fig.10: Temp. Distribution
at the End of Slipping**
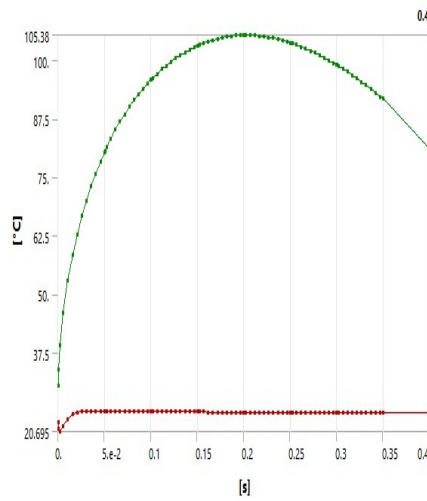
**Fig.11: Max. & Min.
Temperature at the**
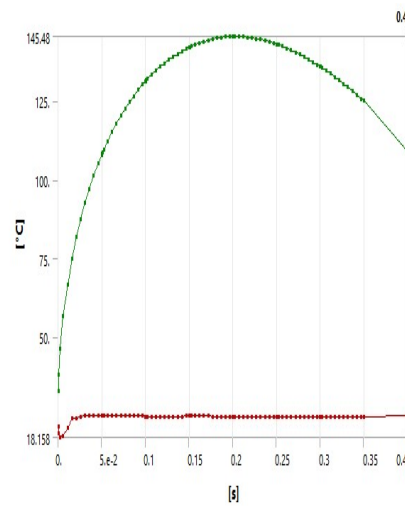
**Fig.12: Max. & Min.
Temperature at the**
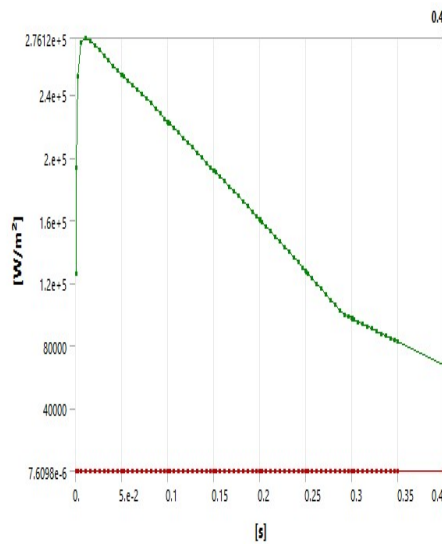
Fig.13: Total Heat Flux at the Slipping Time (Uniform
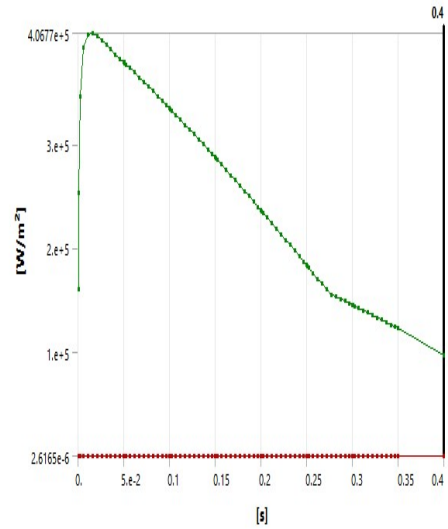
Fig.14: Total Heat Flux at the Slipping Time at Outer disc
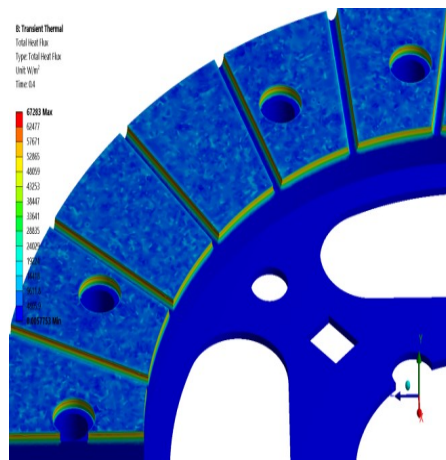
Fig.15: Heat Flux Distribution at the End of
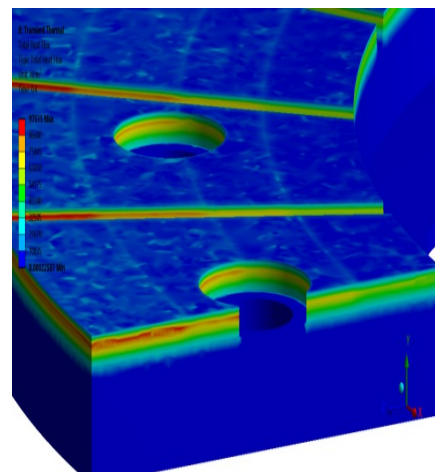
Fig16: Heat Flux Distribution at the End of

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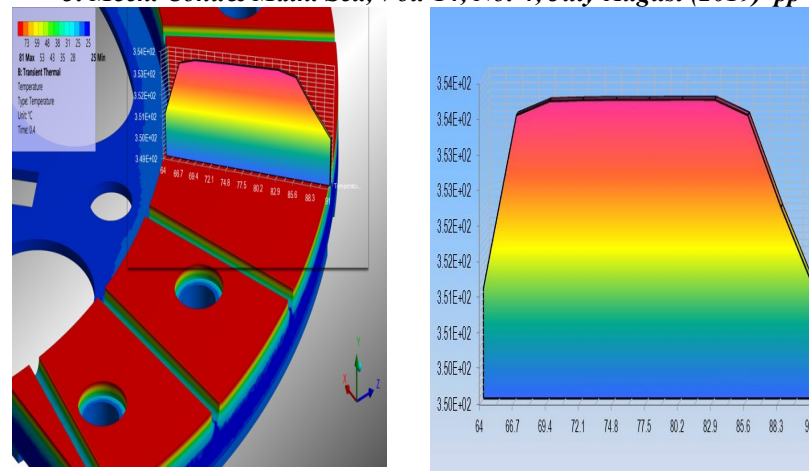
**Fig.17: Temperature Distribution
over Clutch Disc Radius during the**
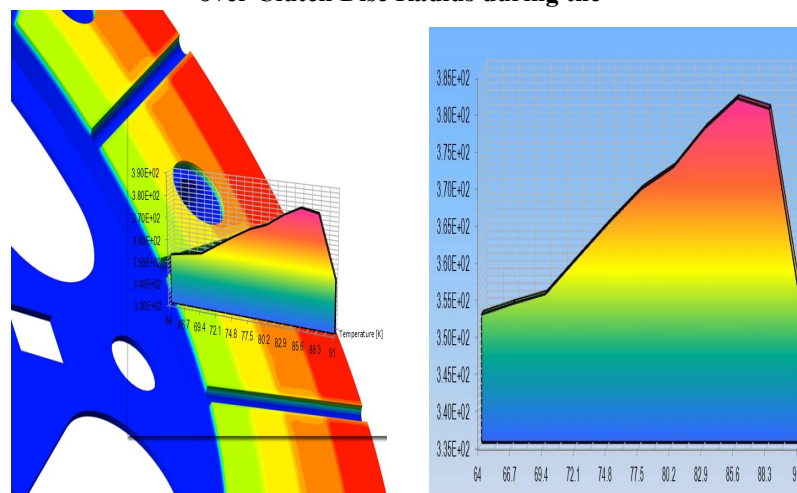
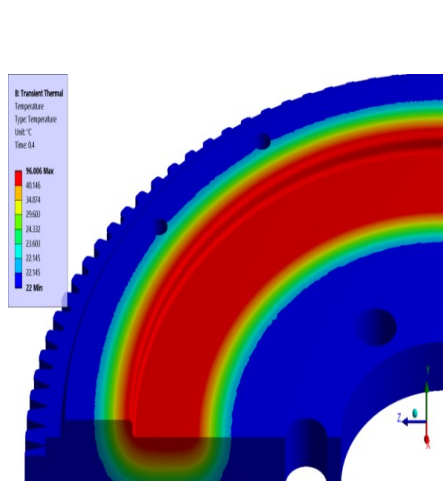**Fig.18: Temperature Distribution over
Clutch Disc Radius during the End of**
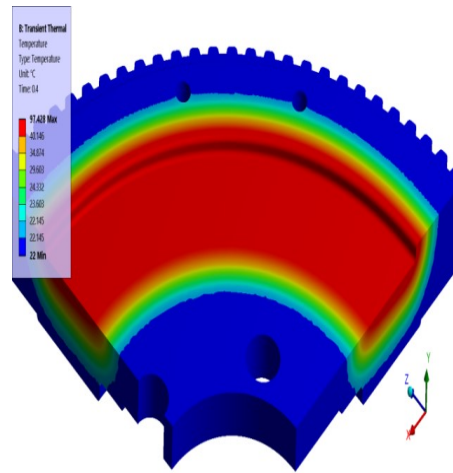
Fig.19: Temperature Distribution at the End of

Fig.20: Temp. Distribution at the End of slipping Time
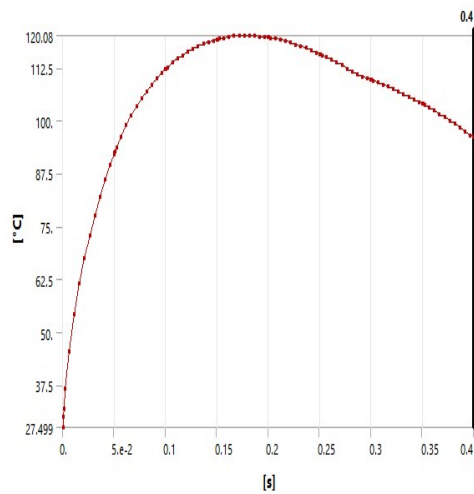

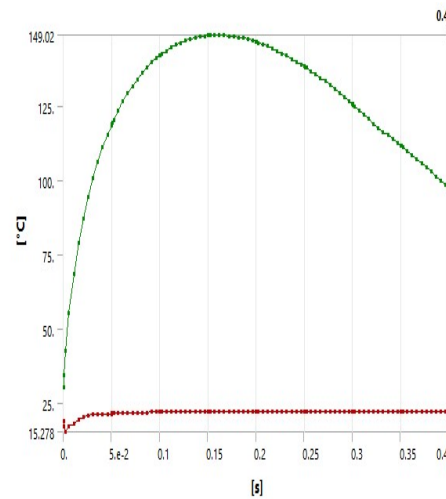**Fig.21: Max. & Min. Temperature at the Slipping**

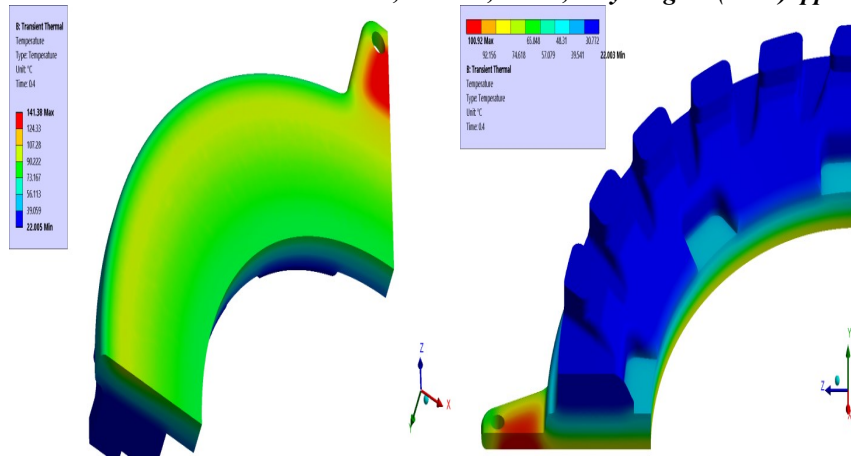**Fig.22: Max. & Min. Temperature at the Slipping**

Fig.23: Temperature Distribution at the End of
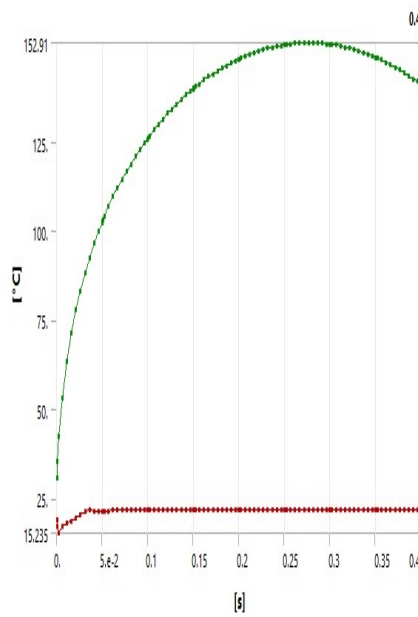

Fig.24: Temperature Distribution at the End of
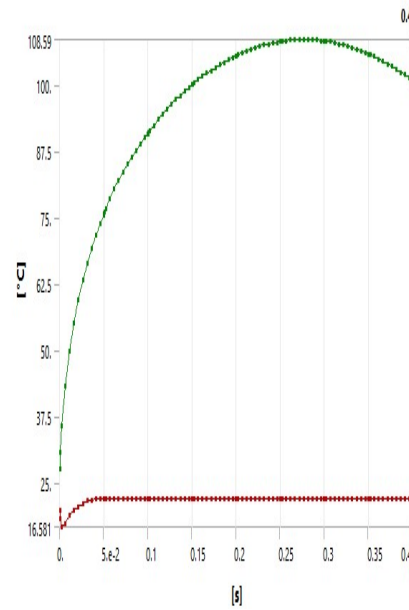

Fig.25: Max. & Min. Temperature during the

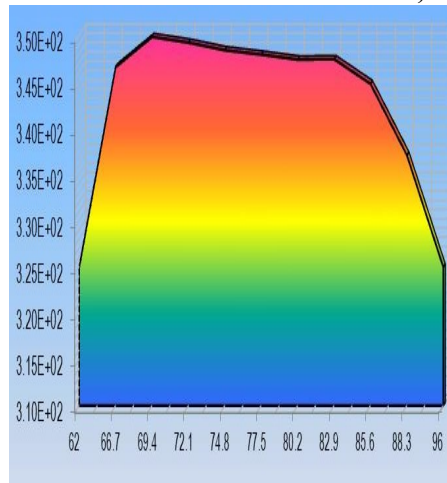**Fig.26:Max. & Min. Temperature during the**

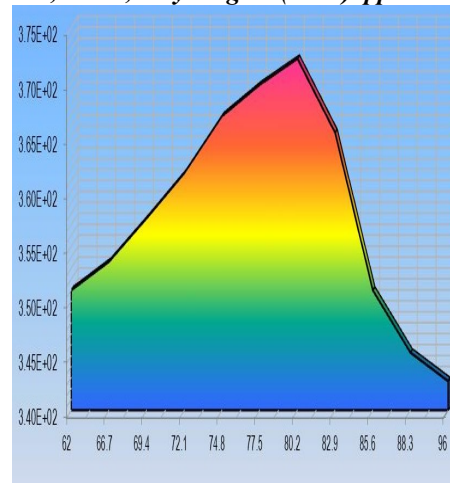**Fig.27: Temperature Distribution Verses Radius at**

Fig.28: Temperature Distribution Verses Radius at

Nomenclature

A_t : Total area of the friction faces of clutch (m²)

C: Specific heat (kJ/kg.k) also defined by equation (7)

k: Thermal conductivity (W/m.k)

n_p : Friction surfaces number

p : Normal pressure on the friction surface (Pa)

p_{max} : Maximum interface Pressure (Pa)

Q_t : Total heat generated (W)

$Q(t, r)$: Total heat generated is function of time and disc radius (W)

$Q_{gen.f}$: Heat generated on flywheel

$Q_{gen.c}$: Heat generated on clutch disc (W)

$Q_{gen.p}$: Heat generated on pressure plate (W)

r : Disc radius (m)

r_i : Inner disc radius (m)

r_o: Outer disc radius (m)

t: Time (sec)

t_s: slipping time (sec)

T_o: Torque capacity of clutch (N.m)

V : Velocity (m/s)

μ: Coefficient of friction

γ : Heat partition ratio

ρ: Density (kg/m^3)

ω_r: Relative angular speed (rad /sec)

ω_{ro}: Maximum initial relative angular speed at t_s=0 (rad /sec)

References

I.   Balázs Czél, Károly Váradi, Albert Albers, and Michael Mitariu, "Fe thermal analysis of aceramic clutch", Journal of Tribology International, 42(5):714–723, 2009.

II.   Belhocine, Ali, and Mostefa Bouchetara, "Thermomechanical modeling of dry contacts in automotive disc brake", International Journal of Thermal Sciences 60:161-170, 2012.

III. Choon Yeol Lee, Il Sup Chung, and Young Suck Chai, "Finite element analysis of an automobile clutch system", Journal Key Eng. Materials, 353-358:2707-2711, 2007.

IV.   E. Mouffak, M. Bouchetara, "Transient thermal behavior of automotive dry clutch discs by using Ansys software", ISSN 1392-1207. MECHANIKA, Vol. 22, No. 6, pp. 562−570 (2016).

V.   Faramarz Talati , Salman Jalalifar, "Analysis of heat conduction in a disk brake system", jornal of Heat Mass Transfer 45:1047–1059, 2009.

VI.   Jenan S. Sherza, Ihsan Y. Hussain, Oday I. Abdullah. "Heat flux in friction clutch with time dependent torque and angular velocity", International Conference on Advanced Science and Engineering (ICOASE), 2018.

VII. Oday I. Abdullah, J. Schlattmann, "Effect of band contact on the temperature distribution for dry friction clutch", World Acad. Sci., Eng. and Technol., Int. Sci. Index 6.9 (2012): 150-160.

VIII.   Oday I. Abdullah, Josef Schlattmann , "computation of surface temperatures and energy dissipation in dry friction clutch for varying torque with time", International Journal of Automotive Technology, Vol. 15, No. 5, pp. 733−740 (2014).

IX.   Oday I. Abdullah, Josef Schlattmann , "Thermal behavior of friction clutch disc based on uniform pressure and uniform wear assumptions", Friction 4(3): 228–237 (2016).

X.   Oday I. Abdullah, Josef Schlattmann , "Thermal behavior of friction clutch disc based on uniform pressure and uniform wear assumptions", FME Transactions vol.46, pp.33–38 (2018).

XI.   Yevtushenko, Kuciej A. A., M., and Yevtushenko O., "Three-element model of frictional heating during braking with contact thermal resistance and time-dependent pressure", International Journal of Thermal Sciences, 50(6):1116-1124, 2011.

XII.   Yogesh Emeerith, Dr. Rabindra Nath Barman, "Structural and Thermal Analysis of a Single Plate Dry Friction Clutch Using Finite Element Method (Fem)" IDL - International Digital Library Of Technology & Research, volume 1, Issue 5, May (2017).