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Thermal stress analysis of the multi-disc friction clutches

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KEYWORD

Thermal stresses
Dry friction
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Finite element method

ABSTRACT

The accurate thermal stresses estimation of any machine element that is subjected to high temperature gradient is considered as essential matter for the mechanical engineers to obtain a successful design. Friction clutches are subjected periodically to high thermal stresses during the beginning of engagement due to the relative motion between driving and driven parts. Axisymmetric finite element models were developed in the present work to simulate the multi-disc clutches during the initial period of engagement to compute thermal stresses, heat generation, and temperature field. Axisymmetric models were used to simulate a multi-disc clutch works under the dry condition during the sliding period. The results indicate that the highest values of the thermal stresses appeared on the last friction disc (5th) and less effect on the other friction discs. This study presents a promising design tool to investigate the effect of materials, surface roughness, sliding speed and boundary conditions on the thermal stresses induced in the contacting surfaces of the friction clutches.

1.0 INTRODUCTION

High thermal stresses are considered as one of the main reasons which lead the friction clutches to failure before the expected lifetime. The thermal stresses may cause serious drawbacks such as thermal cracks and permanent deformations, and when the friction clutch works under these conditions for sufficient time, this will lead the contacting surfaces to premature failure in some cases. The main parts of the multi-disc friction clutch system consist of pressure plate, clutch discs, plate separators and piston as shown in Figure 1.

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Zagrodaki (Zagrodzki, 1985; Zagrodzki, 1990) investigated the thermal stresses and the temperature field in the wet multi-disc clutches that occurred during the beginning of engagement. The thermal properties of the discs were assumed non-homogenous in the analysis. Two numerical approaches were selected (Finite element and finite difference methods) to build the models to conduct the numerical analysis. The main conclusion was; when the temperatures in the radial direction decreased, this will affect gradually on the thermal stresses adversely.

Barber et al. (Yeo and Barber, 1996; Du et al., 1997; Al-Shabibi and Barber, 2002; Al-Bahkali and Barber, 2006; Al-Shabibi and Barber 2009) studied the thermo-mechanical problem in the automotive friction clutches and brakes using analytical and numerical solutions. It was developed the reduced order model approximation to represent the mathematical model using one or more dominant perturbation (eigen-functions). They proved the ability to obtain acceptable results of contact pressure and temperature based on the developed models of the sliding system with a modest degree of freedom. The most important conclusion was that the reduced order models have very good approximations in the initial period of engagement for the automotive brake or clutch when the sliding speed is higher than the critical sliding speed of the sliding system.

Abdullah and et al. (Abdullah and Schlattmann 2016a; Abdullah and Schlattmann 2016b; Abdullah et al. 2018a; Abdullah et al. 2018b) investigated the temperature field and energy generated in the dry friction clutches during a single and repeated engagement under uniform pressure and uniform wear conditions using different theoretical methods. They 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 the clutch disc using analytical and numerical solutions. Furthermore, it was studied the effect of engagement time, sliding speed and thermal load on the thermal behavior of the friction clutches during the beginning of the engagement.

The coefficient of the friction affected by the temperature, because of the high temperatures occurs due to the friction was considered the most parameter affect significantly the operating variables. The history of the contact surfaces is very effective on the coefficient of friction behavior; therefore, the friction/temperature curve for any individual material needs to be interpreted in the relation to the nature of the proposal device and anticipation (Baker 1992). Figure 2 shows the variation of coefficient of the friction with temperature for the good quality woven material. It can be seen that the relatively considerable effect of temperature on the coefficient of friction during the working range of the temperature reduced the coefficient of friction at the upper temperature limit. Materials such this is suitable only for duty up to modest working temperature of about 225 ˚C. The maximum temperature for cotton fiber based material is around 175 ˚C (Baker 1992). Moulded materials have different behaviors with temperature. Figure 3 shows the variation of the coefficient of friction with temperature of the typical moulded material, used for industry. It can be noticed for this type of material that the coefficient of friction is more stable during increasing of the working temperature (Baker 1992).

Also, the effect of sliding speed on the frictional characteristics of the contacting bodies is very effective on the wear, thermal and contact results. Where, each frictional material has a specific behavior according to the sliding speed and the working condition (dry or wet).

It was clear that the values of the coefficient of friction affected significantly by the high temperature and sliding speed. In this paper, the effect of temperature and sliding speed on the coefficient of friction wasn't included, because of the analysis covered only a single-engagement. Where, the period of engagement is very short is less than (1s). It should be considered the effect of temperature and sliding speed on the coefficient of friction when the analysis studies the wear

and thermal behavior of the friction clutches during the multi-engagements to obtain the accurate results of wear, temperature field and thermal stresses.

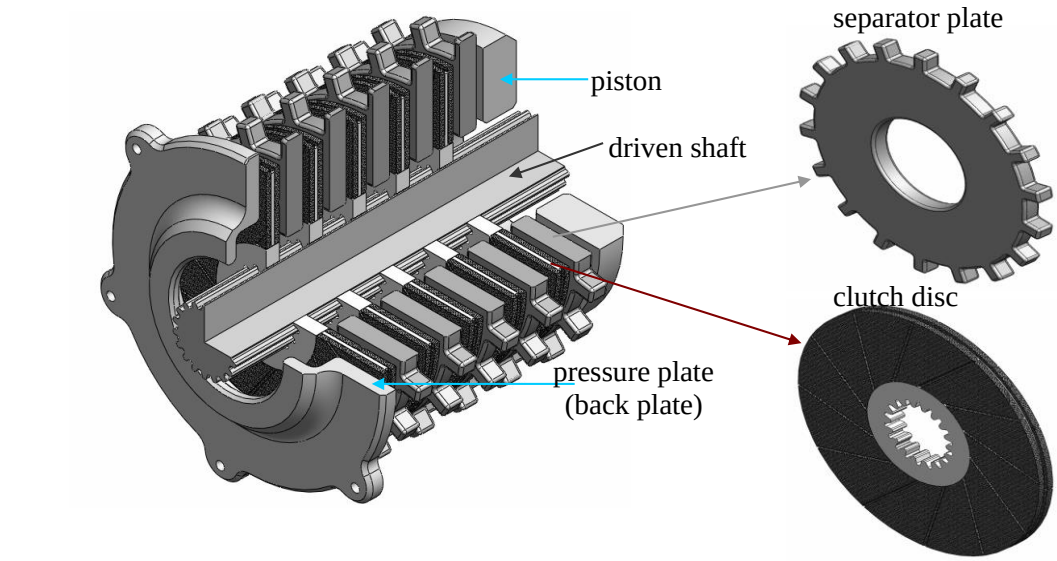

Figure 1: Main parts of a multi-disc clutch system.

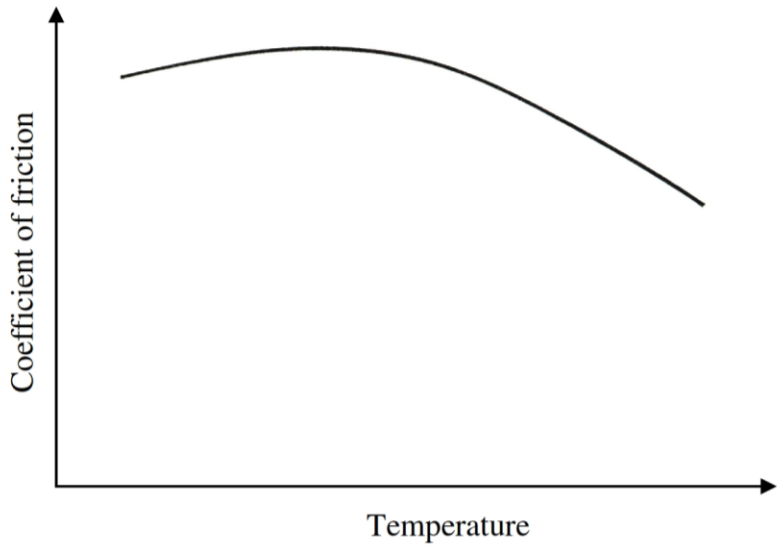

Figure 2: Typical curve of friction/temperature of a good quality woven material (Baker, 1992).

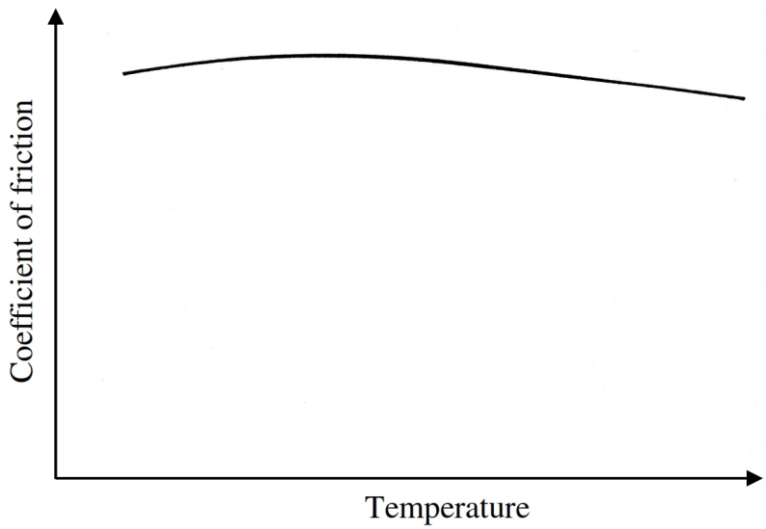

Figure 3: Typical curve of friction/temperature of a good quality moulded material (Baker, 1992).

2.0 FINITE ELEMENT FORMULATION

In the present work, the thermoelastic model of a multi-disc clutch was developed numerically to investigate the heat generated, temperature field and thermal stresses during the sliding period. The sequentially coupled thermal-mechanical approach has been used to investigate the thermoelastic problem involves thermo-mechanical coupling that occurred in the friction clutches. This approach used two different models; the first one is used to solve the elastic problem to yield the displacement field, contact pressure distribution and thermal stresses. Whereas the second model is used to solve the transient thermal problem in order to compute 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. Given the temperature field *T(r,z,t),* the thermoelastic contact problem is solved for the contact pressure *p(r,t)*. This is done by solving Hook's law with thermal strain relations;

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

and the equilibrium equation;

$$\frac{\partial \sigma_{ij}}{\partial x_j} = 0 \tag{2}$$

The next step is the application of the boundary conditions that involve the exposed surfaces and displacement constraints. The contact pressure from the thermoelastic problem is used to define the frictional heat flux;

$$q = \mu\, p\, \omega_r\, r \tag{3}$$

where p, ω_r and r are the contact pressure, angular sliding speed and disc radius, respectively. The second problem that needs to be solved is the transient heat conduction equation;

$$\nabla^2 T = \frac{1}{k}\frac{\partial T}{\partial t} \tag{4}$$

in order to yield a new temperature field $T(r,z,t+\delta t)$. The boundary conditions of this problem involve the heat flux $q(r)$ given at the contact interface and some other boundaries, which can consist of convective and insulated surfaces. It was assumed a homogeneous and isotropic material for all computations of the friction clutch model. All parameters and materials properties are listed in Table. 1. In the analysis, it is also assumed that there are no cracks in the contacting surfaces.

Figure 4 illustrates the flowchart of the developed approach using a finite element method to find the solution to the coupling problem (temperature and stress fields) of the multi-disc friction clutch system. The developed approach consists 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 multi-disc clutch were thus developed by using ANSYS APDL to conduct numerical analysis. The boundary conditions of the thermal and elastic (finite element model) models of the multi-disc clutch are shown in Figures 5 and 6.

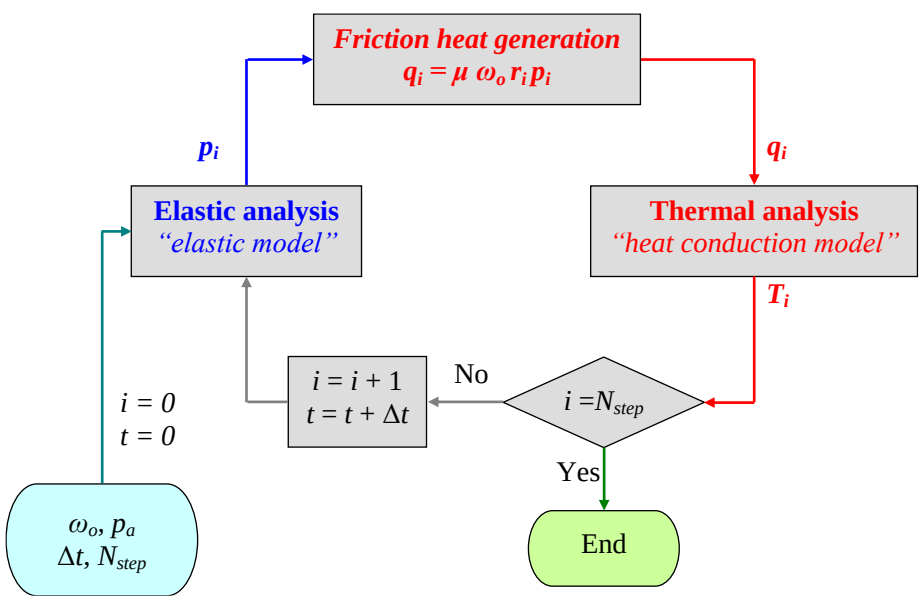

Figure 4: Flowchart of sequentially coupled thermal-mechanical approach.

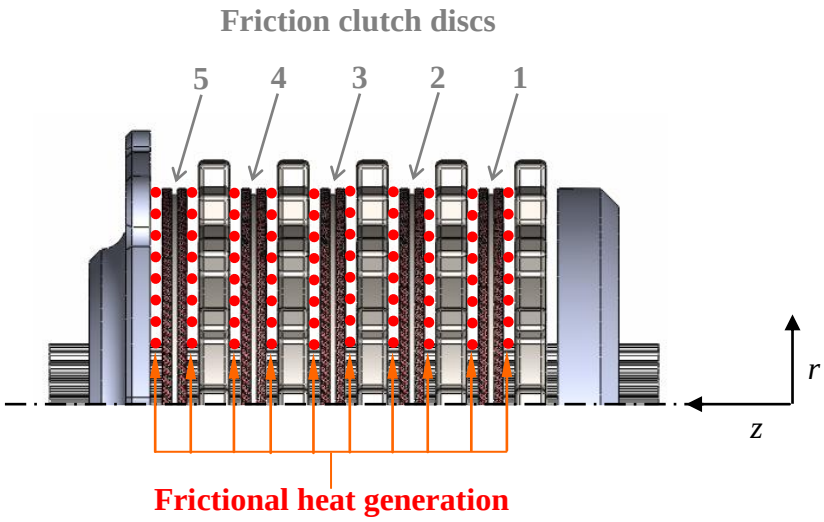

Figure 5: Thermal model with boundary conditions of a multi-disc clutch system (adiabatic conditions are applied to all boundaries of the system).

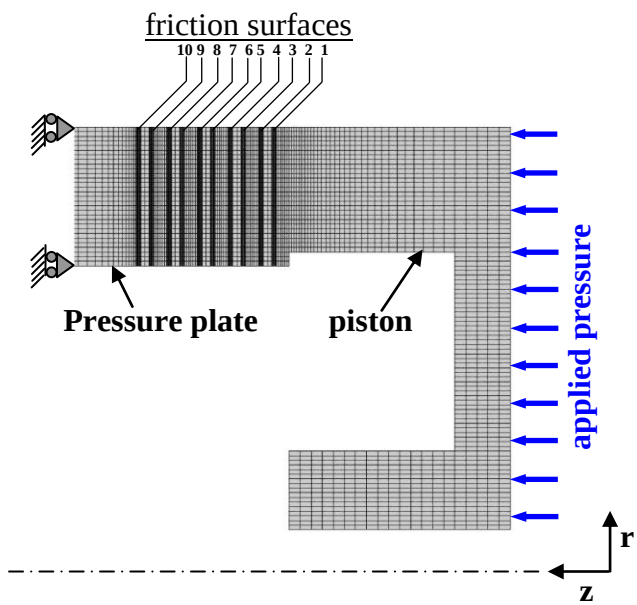

Figure 6: FE model of a multi-disc clutch system with elastic boundary condition (No. of elements= 4428).

Table 1: The properties of materials and operations.

Parameters	Values
Inner and outer radii of friction clutch disc [m]	0.052 0.067
Thickness of clutch disc including the friction surfaces [m]	0.00193
Thickness of friction material [m]	0.00053
Inner and outer radii of pressure plate [m]	0.052 0.067
Thickness of pressure plate [m]	0.0074
Inner radii of piston [m]	0.0235 0.0535
Outer radii of piston [m]	0.032 0.067
Thicknesses of the piston [m]	0.006 0.024
Applied pressure, P [MPa]	1
Coefficient of friction, μ	0.2
Number of friction clutch disc, n	5
Maximum angular slipping speed [rad/sec]	300
Young's modulus of friction material [GPa]	0.30
Young's modulus of pressure plate, plate separator, piston and clutch plate [Gpa]	125
Poisson's ratio of friction material,	0.25
Poisson's ratio of pressure plate, plate separator, piston and clutch plate	0.25
Density of friction material [kg/m^3]	2000
Density of pressure plate, plate separator, piston and clutch plate [kg/m^3]	7800
Specific heat of friction material and steel [J/kg K]	120 & 532
Conductivity of friction material and steel, [W/mK]	1 & 54
Thermal expansion of friction material and steel [K^{-1}]	12 ×10^{-6}
Slipping time, t_s [s]	0.5

3.0 RESULTS AND DISCUSSIONS

The numerical simulation of the multi-disc clutch system is achieved using finite element technique to study the heat generated, temperature and thermal stresses. The results were obtained based on two developed numerical models. The first model is the contact model (elastic model) that used to compute the thermal stresses of all contacting surfaces. While the second model is the thermal model (heat conduction model) used to compute the temperature

distribution of all system parts of the multi-disc clutch at any instant during the sliding period.

Figure 7 shows the variation of the maximum surface temperature during the sliding period. It can be noticed, that the temperature starts from initial value (T_i) at beginning of slipping (t = 0) and increases to maximum value (T_{max} = 891.8 K) approximately after the mid time of slipping period ($t \approx$ 0.4 s), and then it gradually decreases from T_{max} to the final temperature (T_f = 864.3 K) at end of slipping period (t_s = 0.5). The maximum temperature occurred in the last contact surface (10th friction surface in the 5th clutch disc).

Figure 8 shows the distribution of the heat generated with disc radius of the last contact surface of 5th disc clutch at different time intervals. It can be seen that the hot spot was generated near the mean disc radius.

Figure 9 shows the contours of radial thermal stresses at the selected intervals during the sliding period. It can be seen, that the maximum radial thermal stresses occurred for all cases in 5th clutch disc and maximum value of the stress (tensile) was at the mid time of the sliding period (t_s= 0.25s). In general, the tensile stresses are higher than the compressive one during the whole period of sliding.

It can be seen from Figure 10 a different behaviour in the contours of the axial thermal stresses at the same selected intervals during the sliding period. Whereas, the maximum axial thermal stresses occurred for all cases in 5th clutch disc and maximum value of the stress (compressive) was at (t_s = 0.4s). It can be noticed that the values of stresses in 5th disc (compressive zone) increases with time and it is very clear before the end of the sliding period.

Figure 11 illustrates the contours of Von Mises stresses at the selected intervals during the sliding period. The values of the Mises stresses increased with sliding period due to the change that occurred in the contact pressure from the uniform to non-uniform distribution in addition to the thermal effect from the frictional sliding. The highest values of the Von Mises stresses appeared in 2nd, 3rd and 4th friction clutch discs, while the maximum values during whole the period occurred in the 4th friction clutch disc.

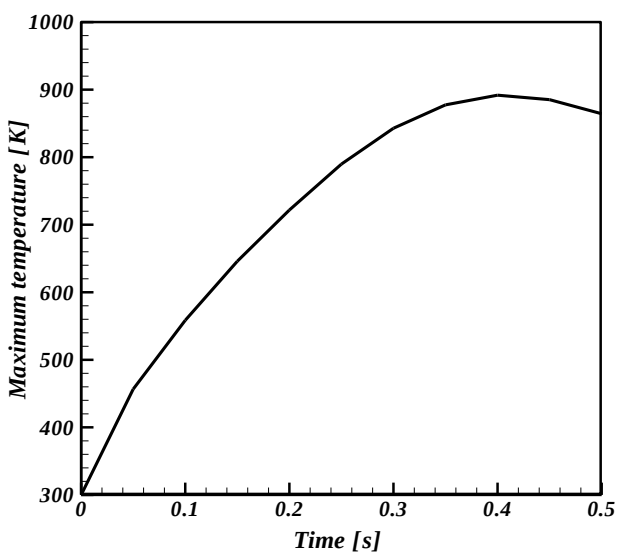

Figure 7: Maximum surface temperature during the sliding period.

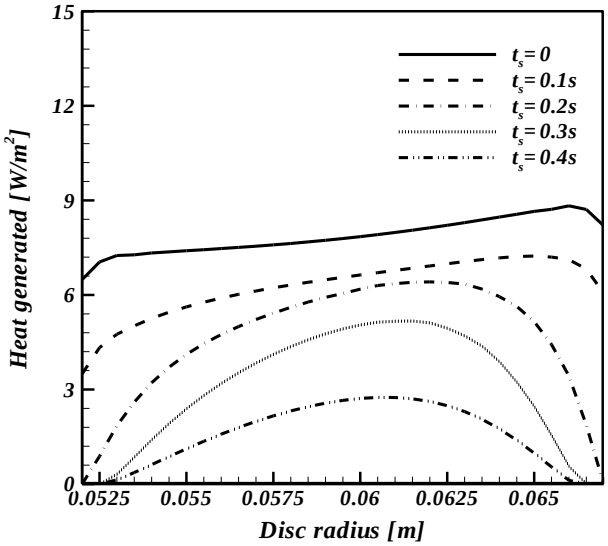

Figure 8: Frictional heat generated on10th friction surface during the sliding period.

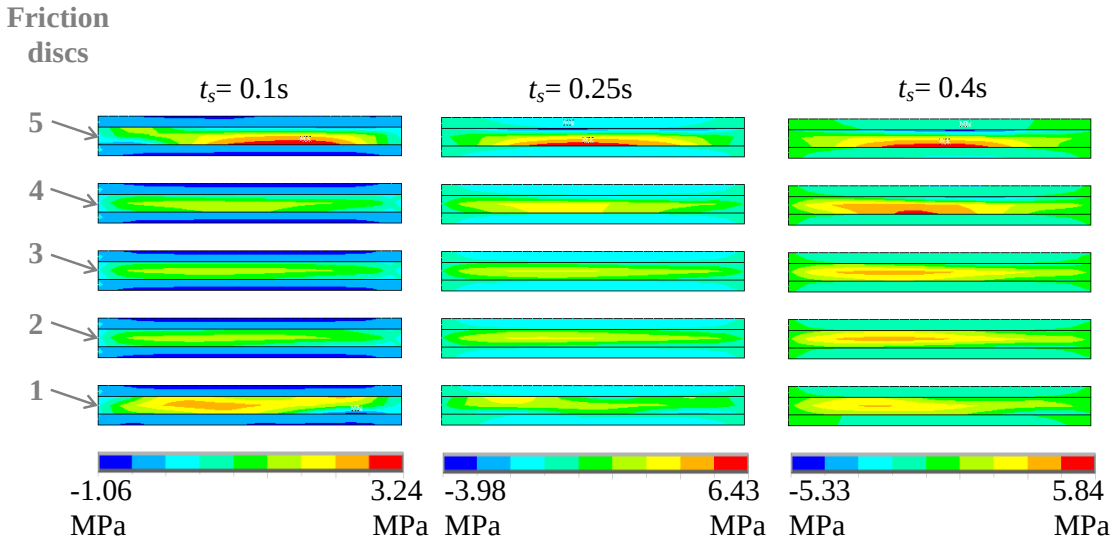

Figure 9: Radial thermal stresses in the friction clutch discs at different time of sliding period.

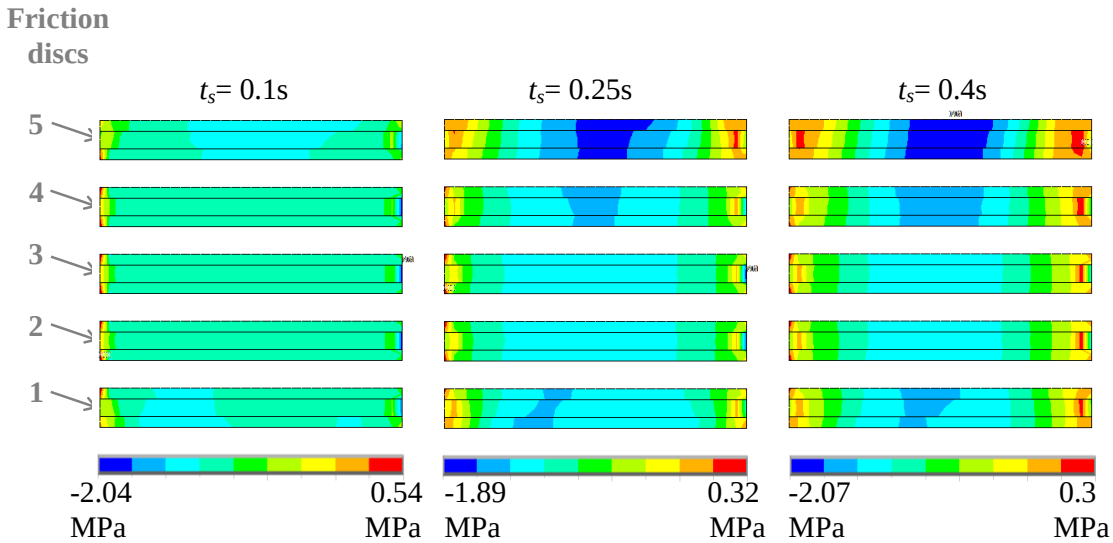

Figure 10: Axial thermal stresses in the friction clutch discs at different time of sliding period.

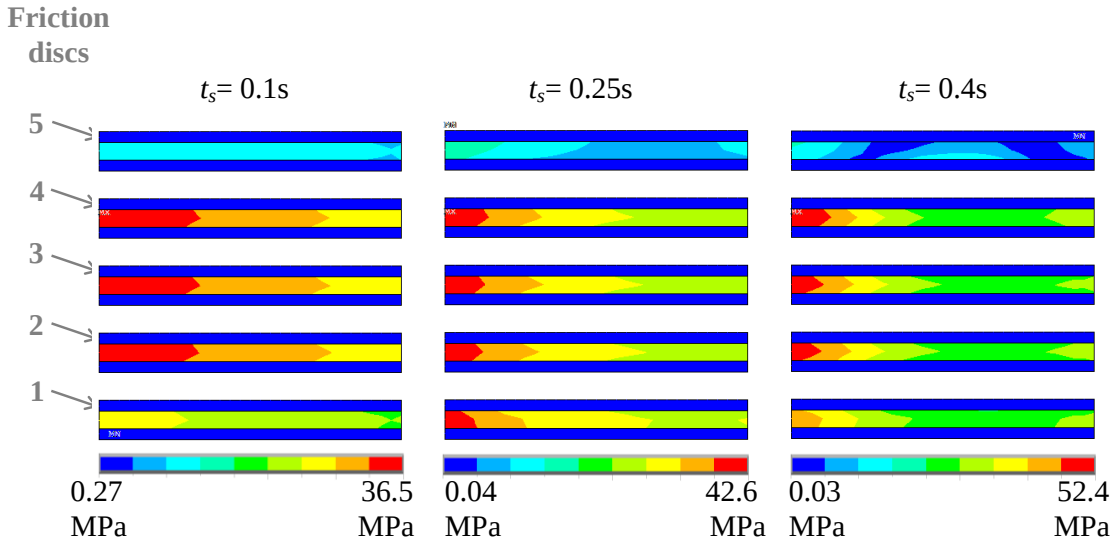

Figure 11: Von Mises stress in the friction clutch discs at different time of sliding period.

4.0 CONCLUSIONS AND RECOMMENDATIONS

The thermal stress analysis of the multi-disc friction clutches working under dry condition during the sliding period has been achieved in this research paper. The developed numerical models specified the complex interaction that exists among the contact pressure, the frictional heat generated and thermal stresses to compute these values at any instant during the initial period of engagement. The highest values of the thermal stresses appeared on the last friction disc (5th) and less effect on the other friction discs. The results proved that the distribution of the thermal stresses in the friction clutch is function of the temperature field in a certain application. The present work presents a promising design tool to investigate the effect of materials, surface roughness, sliding speed and boundary conditions on the thermal stresses induced in the contacting surfaces of the friction clutches.

A further future work may be recommended to investigate the effect of temperature and sliding speed on the friction factor. Also, the effect of materials, surface roughness, sliding speed and boundary conditions on the design of the clutch and the induced stresses may be investigated.

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