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STUDY THE THERMAL PROBLEM IN THE FRICTION CLUTCHES APPLYING A DIFFERENT APPLIED PRESSURE FUNCTIONS

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Abstract - The thermal problem in the friction clutches is considered an essential issue that affects the performance and lifetime of the contacting surfaces of elements of the clutch system. Therefore, this paper focused on an essential factor affect the thermal behavior of the friction clutches, this factor is the function of applied pressure during the sliding period. The effect of the applied pressure function in the dry friction clutches is investigated theoretically. Two functions of applied pressure were assumed; constant and increasing linearly with the sliding time. A finite element technique has been used to compute the temperature distribution of the contacting surfaces of the friction clutch disc at any instant during a single engagement. Axisymmetric finite element models were developed to find the solution of theromelstic coupling problem in the automotive clutches. The numerical results showed that the highest temperature appeared when applying a constant pressure where as the lower values of temperatures appeared when applying the linear function of applied pressure.

Keywords -Thermo-elastic problem; Single-disc friction clutch; Applied pressure; Thermal analysis; Finite element technique.

I. INTRODUCTION

The sufficient knowledge of the factors affecting the thermoelastic behavior of a friction clutch is considered the main key to the automobile engineers to obtain a friction clutch with high resistance to high temperature and wear. The high temperatures were considered the major source of the critical drawbacks such as surface cracks and plastic deformations, and when the friction clutch works under these conditions, may cause the premature damage to the clutch parts.

Figure 1 illustrates the system of a single-disc friction clutch during the engaging and disengaging processes. The main elements of the single-disc clutch system are pressure plate, friction clutch disc and flywheel. This kind of friction clutch has two effective friction surfaces on both sides. When applied the force to start the engagement between the driving and driven parts, the heat will be generated due to the difference in the speeds between the driving and driven parts. The surface temperature will increase dramatically during this phase (figure1 a). Later on, when all parts rotate together at the same speed the heat generated will decrease and reach the zero, this phase is called full engagement.

Barber et al. [1-7] examined different technique to solve the thermoelastic contact problem with frictional heat generation. Axisymmetric finite element models were built to study the thermal and the contact problems of two sliding disks. In their analysis, the speeds were assumed constant and varying during the sliding operation. They concluded that the particular solution has quite irregular form based on the initial temperature appeared especially when the sliding system works higher than the critical speed.

The thermal behavior, energy dissipated and performance of different kinds of dry friction clutches under different working conditions were studied by Abdullah et al. [8-12]. Also, they investigated the effect of surface roughness of the friction clutch discs on the thermoelastic phenomenon under a single and multi- engagements. The deep understanding was presented about the calculations of the energy dissipated and stored in the clutch system during and after the sliding period.

This research paper spots light on the effect of function for applied pressure on the thermal behavior of the friction clutches under dry condition. Two functions of applied pressure were assumed; the first function of applied pressure was constant with time (P = 0.5MPa), while the second one increased linearly with time from P= 0 at (t = 0) to the final value (P = 0.5MPa) at (t_s= 0). The slipping times for both cases were calculated based on the equations of motion [10], where the slipping times to reach the full engagement phase (when all parts rotate together at the same speed without slipping between each other) of each case are 0.4s and 0.8s corresponding to the constant and linear applied pressure functions, respectively.

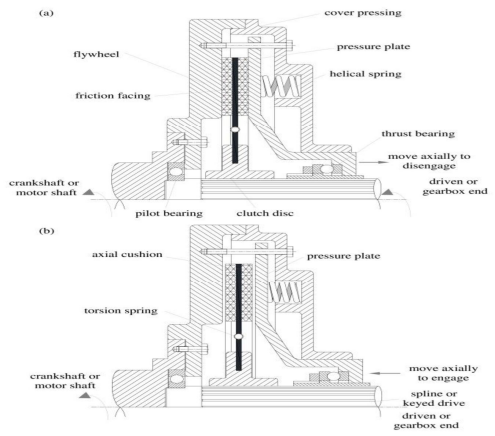

Fig. 1. Engagement and disengagement in single-disc clutch system (a) Engaged, (b) Disengaged

II. FINITE ELEMENT SIMULATION

In order to reduce the time consuming for calculations and the complexity of the finite element models, 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 shows 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. The rate of the heat generated between two rubbing surfaces per unit area at any time is,

$$q(r,t) = \mu\ p\ \omega\ r \qquad ,0 \le t \le t_s \quad (1)$$

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

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

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 thermoelastic problem in the friction clutches. The contact pressure distribution p(r, t) can 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 [13]

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

whereυ, E, σ, α and δ_{ij} are Poisson's ratio, Young's modulus [N/m^2], stress components [N/m^2], thermal expansion [K^{-1}] and the Kronecker delta, respectively. The equilibrium stress is [13],

$$\frac{\partial \sigma_{ij}}{\partial x_j} = 0 \qquad (4)$$

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,

$$\nabla^2 T = \frac{1}{k}\frac{\partial T}{\partial t} \qquad (5)$$

wherek is the thermal diffusivity (k = K/ρc). K,ρ and c are conductivity, density and specific heat, respectively. Figure 3 illustrates the flowchart of the developed sequentially 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 developed using ANSYS APDL to conduct numerical analysis. The selected materials which used to simulate 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 of the system. The element used for the thermal model was PLANE55, this element consists of four-nodes; for each node one degree of freedom (Temperature). Plan13 element was used for the all elastic model (flywheel, clutch disc and pressure plate), this element has 4 nodes coupled-field (thermal and structural fields) solid element with up to 4 DOF per node. Conta172 and Targe169 elements were used for the contact surfaces and target surfaces, respectively.

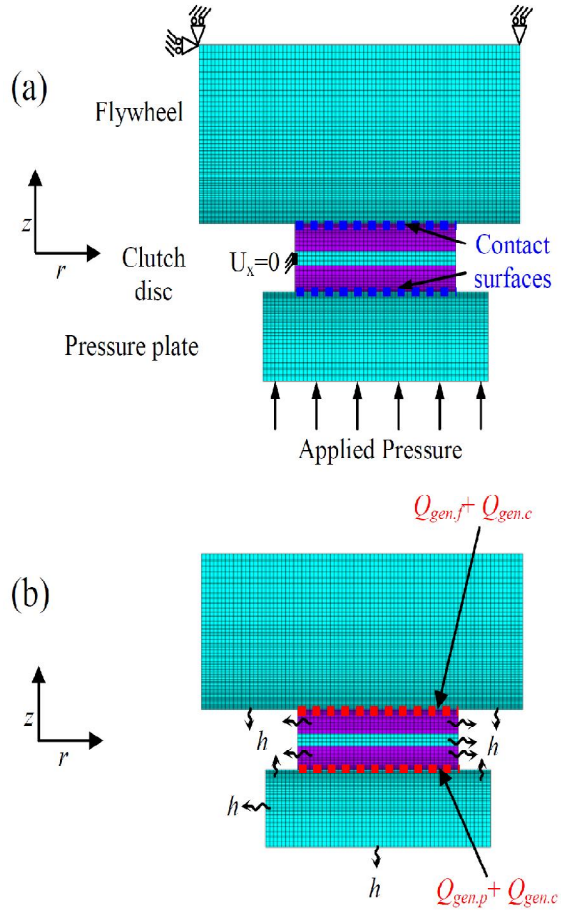

Fig. 2. FE models with the boundary conditions [(a) Elastic model; (b) Thermal model]

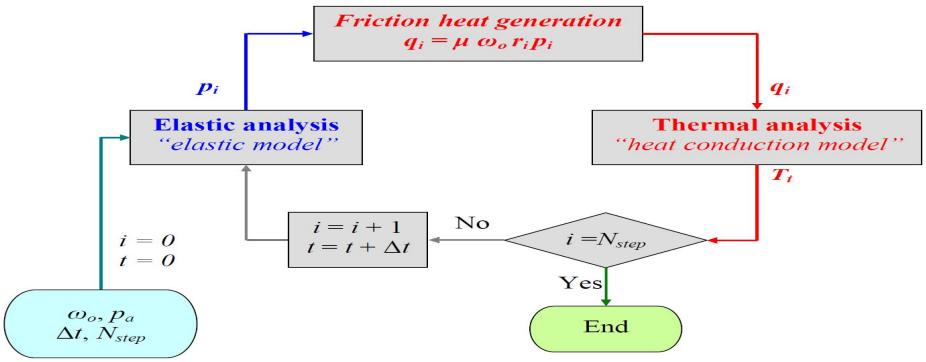

Fig. 3. Flowchart of sequentially coupled thermal-mechanical approach

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

Parameters	Values
Inner radius of friction disc and axial cushion, [m]	0.065
Outer radius of friction disc and axial cushion, [m]	0.09
Thickness of axial cushion, [m]	0.0017
Inner radius of pressure plate, [m]	0.06
Outer radius of pressure plate, [m]	0.095
Thickness of pressure plate, [m]	0.0095
Inner radius of flywheel, [m]	0.049
Outer radius of flywheel, [m]	0.095
Thickness of flywheel, [m]	0.019
Coefficient of friction, μ	0.3
Number of friction surfaces, n	2
Young's modulus of friction material, [GPa]	0.3
Young's modulus of pressure plate, flywheel and axial cushion, [GPa]	200
Poisson's ratio of friction material, [GPa]	0.12
Poisson's ratio of pressure plate, flywheel and axial cushion [1]	0.3
Density of friction material, [kg/m^3]	846
Density of pressure plate, flywheel and axial cushion, [kg/m^3]	7800
Specific heat of friction material, [J/kg K]	1610
Specific heat of pressure plate, flywheel and axial cushion, [J/kg K]	452
Conductivity of friction material, [W/mK]	0.241
Conductivity of pressure plate, flywheel and axial cushion, [W/mK]	42
Thermal expansion of friction material, [K^{-1}]	14×10^{-6}
Thermal expansion of pressure plate, flywheel and axial cushion, [K^{-1}]	12×10^{-6}
Slipping time (constant applied pressure function), t_s [s]	0.4
Slipping time (linear applied pressure function),, t_s [s]	0.8

III. RESULTS AND DISCUSSIONS

In this paper, the temperature distribution of the friction clutch disc when applied different pressure function has been investigated. Two cases of applied pressure functions during the sliding period have been considered, a constant applied pressure with time and while the second applied pressure function increased linearly with sliding time.

The variation of surface temperature with disc radius and time during a single engagement applying the constant and linearly functions of applied pressure are

shown in figures 4 and 5. It can be seen for both cases that the temperatures increase with time until reach the maximum values approximately after the middle time of the slipping period. Also, the maximum values appeared near the inner radius of friction disc.

The maximum value of surface temperature (pressure plate side) is found to be T = 382.39K at r = 0.0678m (t = $0.6t_s$) when the applied a constant pressure function. However, the maximum value of surface temperature (at the pressure plate side) when applied a linear function of applied pressure is found to be T = 350.03K at r = 0.0678m (t = $0.7t_s$).

CONCLUSIONS AND REMARKS

The transient thermoelastic analysis of the dry friction clutches during a single engagement was performed. Axisymmetric models were built to obtain the numerical simulation for thermal problem which appeared between the elements of the friction clutch system during slipping period.

The results showed that the influence of the function of applied pressure during the sliding period is significant on the temperature field in the domain of time. It was found, that the results of temperature distribution applying the constant pressure are higher than the other one when applying the linear function of pressure, because of the high amount of heat will generated in very short time according to the constant pressure function compared with the linear function of applied pressure.

The study of temperature field of the contact surfaces during the beginning of engagement operation is necessary to give indication about the maximum temperature occurred during the sliding period, and then it can be evaluate the status of the clutch system if it's safe under a certain thermal condition or not. This investigation is considered essential to the automotive engineers to obtain the most efficient design of the friction clutch and to estimate the lifetime of the contacting elements of the clutch system.

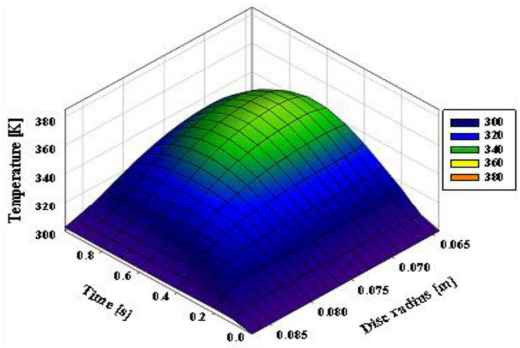

Fig. 4. Surface temperature distribution of clutch disc applying a linear function of pressure

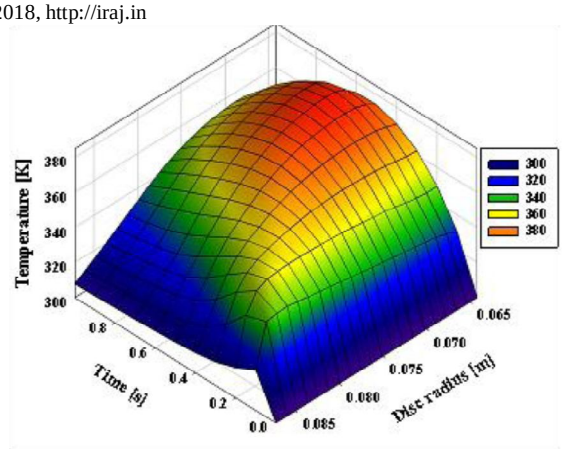

Fig. 5. Surface temperature distribution of clutch disc applying a constant pressure

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