Effect of Sliding Speed on the Thermal Stresses of Single-Disk Friction Clutches

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Abstract The friction clutch is considered an essential machine element in the power transmission system, whereas the friction clutches are used widely in many applications of mechanical engineering, especially in the automotive vehicles. Most of failures happen in the elements surfaces of the clutch system due to the excessive heat generated through the early stage of the engagement period. In this paper, the finite element technique is applied to compute the variation of the heat generated due to friction, temperature and thermal stresses through the heating stage of the friction clutch. The simulation of working of the friction clutch has been accomplished using developed axisymmetric finite element models. The results present the distributions of frictional heat generated and thermal stress during the sliding period. The results proved that the non-uniformity of the pressure distribution is responsible for generating the high local frictional heat generated in some zones of nominal contact area through the heating stage and this will lead to a high increase in the thermal stresses.

Introduction

Previous analyses about the effective parameters affecting the thermo-elastic behavior of the friction clutch underlined the need of friction clutch with high resistance to high thermal stresses and wear. The high thermal stresses were considered the main cause that lead to failure in the contact surfaces of clutches such as surface cracks and dishing, material transfer and bond failure. When the friction clutch works under high thermal stresses for enough time, it may cause the premature damage to the clutch parts. Figure 1 illustrates the main elements of the dry friction clutch system responsible to transfer the torque from the driving shaft to the driven shaft.

Barber et al. [1–7] examined different techniques to solve the thermo-elastic contact problem which included the frictional heat generated. One- and two-dimensional finite element models were developed to investigate thermal-contact problem of two disks through the sliding. Two cases are assumed: The first case when the speed is constant with time and the second case when the speed varies with time. The irregular form is obtained using the particular solution when the critical speed of the sliding system is the default one.

The thermal behavior, thermal stresses, energy dissipated and performance of different configurations of the

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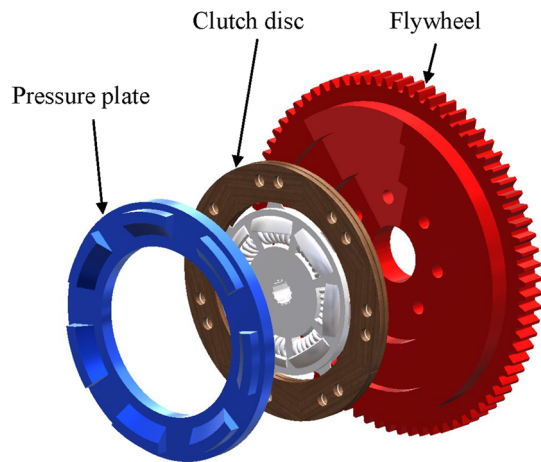
Fig. 1 The main components of the frictional clutch system

Fig. 2 Finite element models for the single-disk clutch

automotive clutches and brakes were studied which operate under different working conditions. Also, the thermal stresses of the multi-disk clutch during a single engagement were investigated. Their analyses presented comprehensive understanding about the computations of the energy balance in the clutch system at any time during the slipping period [8–15].

Topczewska [16, 17] obtained analytical models to calculate thermal stresses distributions in the friction elements of braking systems for different braking modes. For this purpose, the solutions of the boundary quasi-static problems of thermo-elasticity were received for different specific friction power, based on Timoshenko's model and the corresponding temperature fields. The last mentioned were calculated from one-dimensional problem of heat conduction. It was established that the values and distribution of the thermal stresses initiated by frictional heating depend mainly on the temporal profile of a specific friction power.

It can be considered that the sliding speed is the most effective design parameter on the magnitude of the thermal stresses in the sliding systems such as friction clutches. In this paper, the effect of the sliding speed on the distribution of frictional heat generated and thermal stresses in the friction clutch at the first phase (sliding period) of engagement has been investigated. The heat transfer by convection was included in the developed analysis.

Finite Element Formulation

Two-dimensional finite element models (axisymmetric) to represent the friction clutch system working in the dry condition have been developed. This kind of model has been proved as an effective one to reduce the complexity of
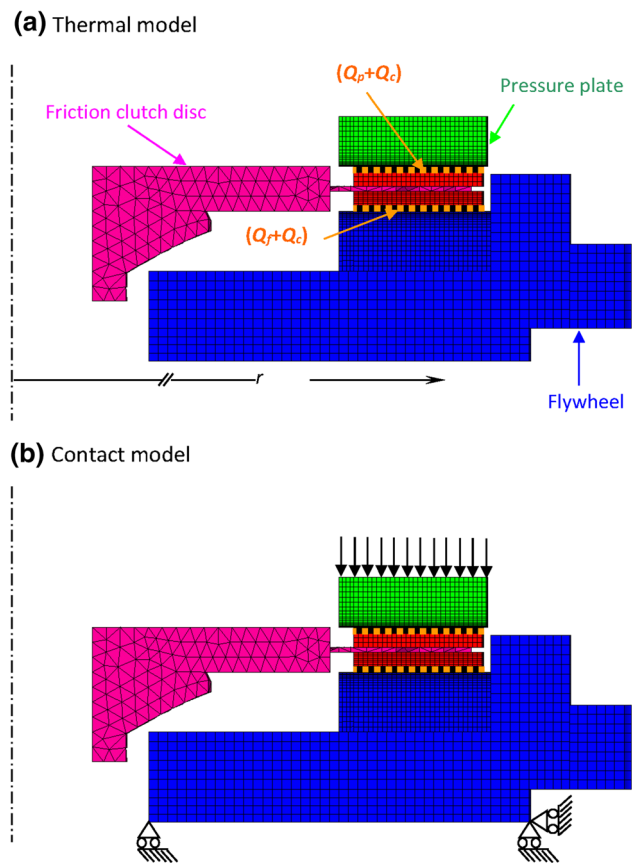
the clutch model and the ensuing computational time. The design of friction clutch selected in this work has no groove; therefore, the axisymmetric model is valid. Figure 2 depicts the finite element models (thermal and elastic) with the boundary conditions.

In Fig. 2, Q_{f} is the heat rate which enters into flywheel, Q_{c} is the heat rate which enters into clutch disk and Q_{p} is the heat rate which enters into the pressure plate. h is the convection heat transfer coefficient. The heat generated due to friction between two contacting surfaces at any time can be calculated as follows:

$$q(r, t) = \mu p \omega_{\mathrm{r}} r, \quad 0 \leq t \leq t_{\mathrm{s}} \tag{Eq 1}$$

where t_{s} is the sliding time; μ the coefficient of friction; p the contact pressure; ω_{r} the angular sliding speed; and r the disk radius.

It was assumed that the angular speed slipping diminishes linearly with time as follows:

$$\omega_{\mathrm{r}}(t) = \omega_{ro}\left(1 - \frac{t}{t_{\mathrm{s}}}\right), \quad 0 \leq t \leq t_{\mathrm{s}} \tag{Eq 2}$$

where ω_{ro} is the initial angular sliding speed.

The first step is to find the solution of the simultaneous thermal-elastic coupling problem, and then, the whole

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solution of the thermo-elastic problem of the friction clutch system can be found. Where, it can be found the distribution of contact pressure $p(r, t)$ when obtain the solution of the elastic model applying the temperature distribution $T(r,z,t)$ as input for the program. This assumption is based on Hook's law, where the thermal strain form is written as follows [18]:

$$\varepsilon_{ij} = \frac{(1 + v)}{E} \sigma_{ij} - \left(\frac{v}{E} \sigma_{mm} + \alpha T\right) \delta_{ij} \qquad \text{(Eq 3)}$$

where v is Poisson's ratio; E Young's modulus (N/m2); σ stress components (N/m2); α thermal expansion (K−1); δ_{ij} Kronecker delta.

The equilibrium form of the stress is as follows [13]:

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

The distribution of frictional heat generation $q(r,t)$ on the surfaces of contact [Eq. (1)] can be calculated based on the results of the elastic analysis (the contact pressure distribution). The next step is to use the results of the frictional

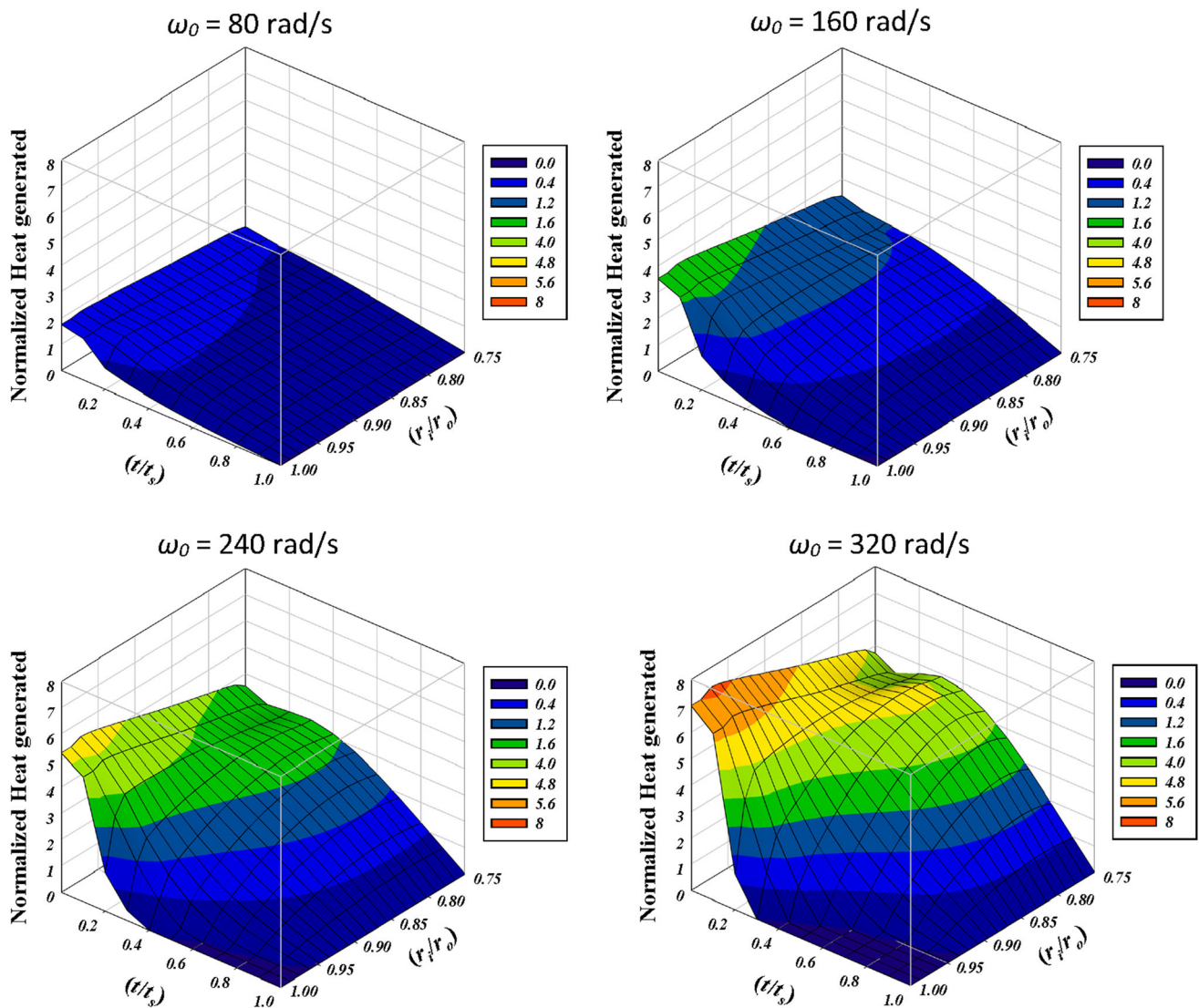Fig. 3 Schematic of thermal stresses analysis using finite element technique
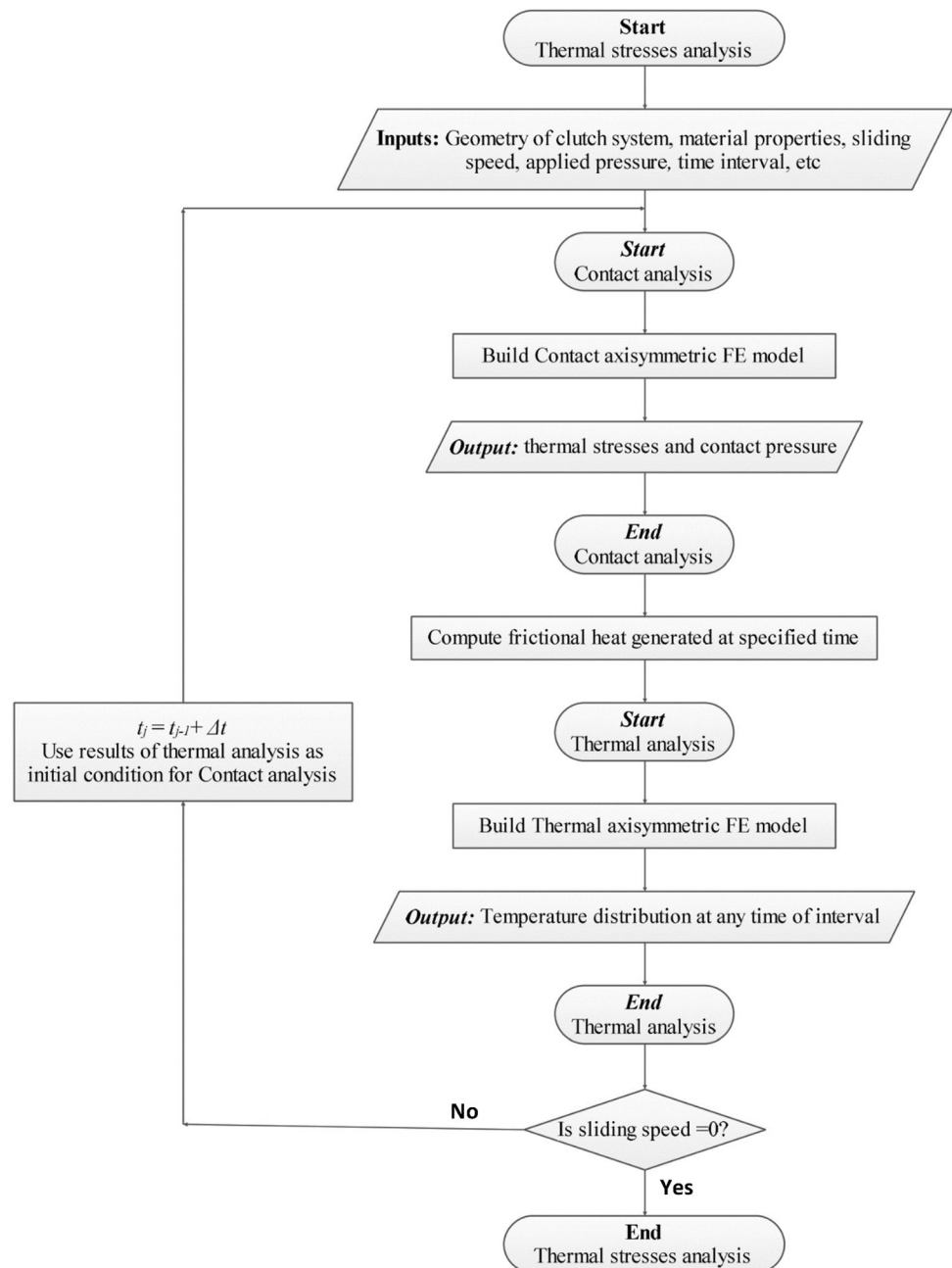
Fig. 4 The distributions of the frictional heat generated over the clutch disk surface during the sliding period when applying different sliding speeds

heat generation as input (thermal load on the contacting surfaces) in the thermal analysis (transient heat conduction).

The final step is to find the solution of the thermal problem to find the distribution of temperature as follows:

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

where k is thermal diffusivity ($k = K/\rho\,c$); K conductivity (W/mK); ρ density (kg/m^3); c specific heat (J/kg K).

The steps of the developed approach in this paper to obtain the solution to the coupling problem (temperature and stress fields) of the friction clutch system are summarized in Fig. 3.

The proposed approach consists of two analyses; the first one is the contact analysis (elastic analysis) and the second one is the transient thermal analysis. The contact analysis was used to find the distribution of the contact pressure and thermal stresses, whereas the transient thermal analysis was used to compute the distribution of temperature at any time during the slipping phase. Numerical models (finite element) were developed to represent the friction clutch during the slipping period. It was built a code using ANSYS/APDL to achieve the required numerical tasks.

PLANE55 was used as element for the transient thermal model, where it has four nodes with one degree of freedom (temperature). The type of the element used to build the main contact model of the clutch system (flywheel, clutch
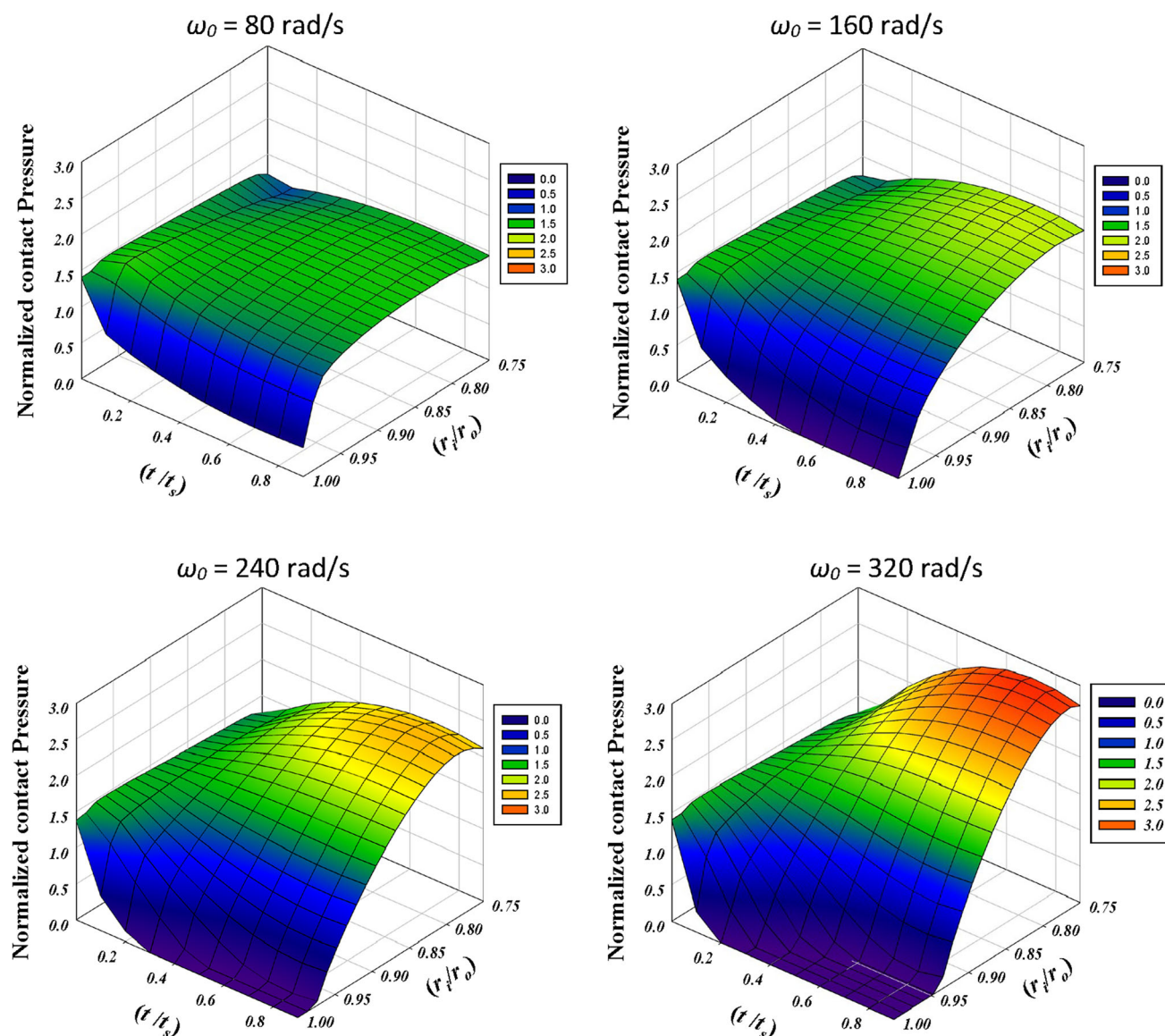
Fig. 5 The distributions of the contact pressure over the clutch disk surface during the sliding period when applying different sliding speeds

disk and pressure plate) is Plan13, where this element consists of four nodes with up to 4 degree of freedom (deformations and temperature). Two types of elements were used to represent the contact zone, which are Conta172 (contact surfaces) and Targe169 (target surfaces).

The Case Study

It was assumed that the materials used to accomplish the numerical analyses are isotropic and homogeneous. It was assumed that the value of the coefficient of convection is 40.89 (W/m^2 K) according to Ref. [8]. During the slipping phase, the convection coefficient was constant over all exposed surfaces of the clutch system. The details of the

geometry for clutch system (number of friction surfaces = 2) are as follows:

- The friction clutch disk: the inner and outer disk radii 0.0925 and 0.125 m, respectively. The thicknesses of the frictional facing and axial cushion are 0.0025 and 0.0016 m, respectively.
- The pressure plate: the inner and outer disk radii 0.06 and 0.092 m, respectively. The thickness of the pressure plate is 0.0097 m.
- The flywheel: the inner and outer disk radii 0.049 and 0.097 m, respectively. The thickness of the flywheel is 0.0194 m.

Applied pressure is 0.9 MPa.

The material properties are as follows:

- The values of Young's module for friction material and steel are 0.30 and 125 Gpa, respectively.
- The value of Poisson's ratio for friction material and steel is 0.25.
- The values of density for friction material and steel are 2000 and 7800 kg/m^3, respectively.
- The values of specific heat for friction material and steel are 120 and 532 J/kg K, respectively.
- The values of conductivity for friction material and steel are 1 and 54 W/m K, respectively.
- The value of thermal expansion for friction material and steel is 12×10^{-6}.

Coefficient of friction is 0.25. Slipping time is $t_s = 0.5$ s.

Results and Discussion

The effects of the sliding speed have been investigated in detail on the frictional heat generated, contact pressure and thermal stresses. Four cases were selected to conduct the numerical analyses in this research paper with different initial angular sliding speeds ($\omega_{0r} = 80$, $\omega_{0r} = 160$, $\omega_{0r} = 240$ and $\omega_{0r} = 320$ rad/s).

The results of these 4 cases provided a better understanding of the complex interaction between the influences of sliding speed against applied pressure on the thermoelastic behavior of the friction clutches.

Figure 4 demonstrates the distribution of the frictional heat generated over the clutch disk surface during the engagement period with different values of sliding speed ($\omega_0 = 80$, 160, 240 and 320 rad/s). It was found that the highest value of the frictional heat occurred when $\omega_0 = 320$ rad/s and the lowest one occurred when ω_0 is equal to 80 rad/s. It can be noticed that when the sliding speed increased the distribution of the frictional heat generated changed dramatically from semi-uniform to non-uniform. High amount of frictional heat on a specific zone on the nominal contact area is the main disadvantage that occurs when the applied sliding speed increases. Under such circumstances, the contact surfaces will be quickly damaged.

Figure 5 presents the normalized contact pressure on the frictional surfaces during the complete sliding process. It can be seen that when applying low sliding speed, the contact pressure distribution is semi-uniform and when the sliding increases, the distribution changes dramatically to be non-uniform. At high sliding speed (320 rad/s), it can be noticed that the heist contact pressure focused near the inner radius and the separation started between the

contacting surfaces at the outer radius. This is considered a strong indication that the status of the sliding system changed from stable to unstable.

Figure 6 exhibits the history of the maximum values of temperature of the single-disk clutch system. It can be concluded that the values of temperature increased with time and reached the peak at ($t = 0.6\ t_s$) and then decreased to the final values at the end of the sliding period. It can be also seen that the temperature increased significantly when the sliding speed increased. The maximum increments in the temperatures are found to be 35.1 and 165.4 K corresponding to the lowest and highest values of sliding speed, respectively. The maximum temperature occurred at the inner disk radius.
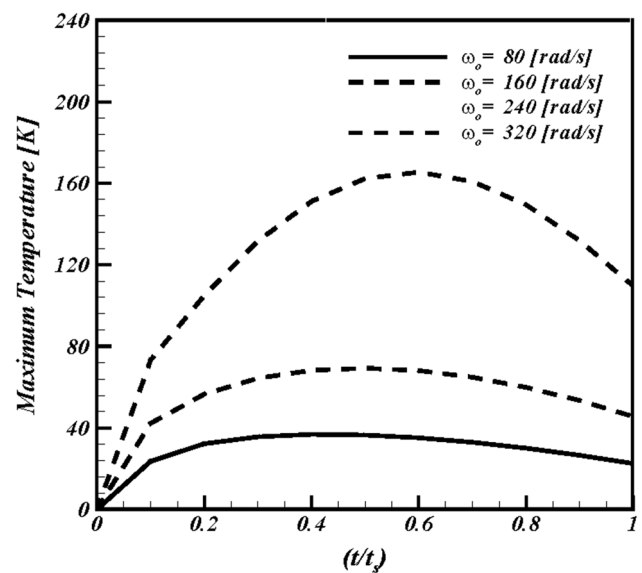

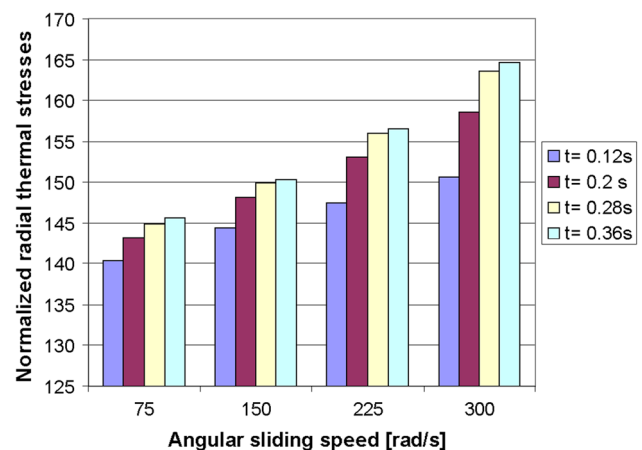**Fig. 6** The maximum temperature history of the single-disk clutch system

Fig. 7 Normalized radial thermal stresses when applying different sliding speeds

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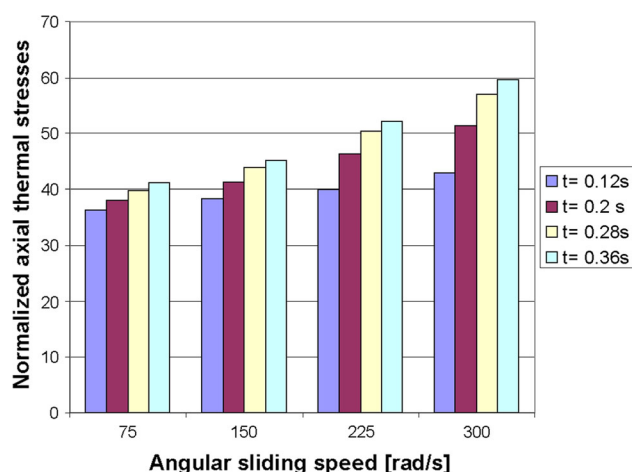J Fail. Anal. and Preven. (2020) 20:1534–1540

Fig. 8 Normalized axial thermal stresses when applying different sliding speeds

Figure 7 illustrates the maximum values of the normalized radial thermal stresses at the selected intervals during the engagement period. It can be observed that the values of radial thermal stresses increased with sliding speed and with time of sliding too. It was found that all values of the stresses are tensile.

It can be seen from Fig. 8 where it was selected the same intervals during the sliding period, also the same behavior where the axial thermal stresses increased too with sliding speed and the time of sliding. All axial values of thermal stresses (maximum stresses) are found compressive stresses. Generally, the values of tensile stresses are higher than the compressive one during the whole period of engagement.

Conclusion and remarks

The thermal stress problems in the sliding systems as friction clutches and brakes are significantly affected by the magnitude of the sliding speed. The stability status of friction clutch system may be changed under certain conditions from stable to unstable. The main reason behind such a change is the magnitude of the sliding speed. Therefore, the stability criterion of the sliding systems is determined based on the critical speed ratio (sliding sped of the system/critical speed). If this ratio is greater than 1, then the sliding system is unstable, and if the ratio less than 1, then the sliding system is stable.

In this work, the developed numerical approach based on finite element method was used to explore the influence of the sliding speed on the thermal stresses problem of the friction clutch system (single disk with both active surfaces).

The most important finding is that the stability of the sliding system (friction clutches) is largely depending on

the value of the sliding speed. By carefully following the obtained results, it can be observed that the values of thermal stresses increase significantly when the sliding speed overcomes critical threshold.

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