

Vol. 40, No. 1 (2018) 92-99, DOI: 10.24874/ti.2018.40.01.08

Tribology in Industry

www.tribology.rs



The Temperature Distribution in Friction Clutch Disc under Successive Engagements

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Keywords:

Dry friction clutch Successive engagements, Temperature distribution Finite element analysis

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A B S T R A C T

The repeated engagements were made by the friction clutch in the realistic applications to reach the desirable speed. The traditional theoretical method to compute the temperature field specifically the maximum temperature appeared in the surfaces of the friction clutch disc only during the single engagement is bootlicker to obtain a successful design which meets the demands today. The temperature fields were computed during 6 consecutive engagements when the full engagement period was 5s. Three dimensional model of the friction surface of clutch disc was build based on finite element technique to determine the thermal behavior of friction clutch system. The contact pressure between the contact surfaces was supposed uniform during all engagements. It was found that repeated engagements have a great influence on the surface temperature.

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1. INTRODUCTION

Friction clutches and brakes are considered to be the most common type used in automotive application. Two or more surfaces are pressed together by a normal force to create a friction torque. The friction surfaces could be flat and perpendicular to the axis of rotation. Figure 1 shows the main parts of typical single-disc clutch system during engagement and disengagement operations. The requirement for an accurate estimation of the surface temperature of the friction clutch increases for different applications in the mechanical engineering field to avoid the early failure or damage in the contact surfaces.

Faidh-Allah [1] developed a thermolelastic finite element model of the friction clutch (single clutch disc). The effect of different friction materials (material properties) on the thermal and mechanical behavior was investigated intensely. Organic and Sintered friction materials was used as a friction facing of the clutch disc. The results showed the temperature distribution, the heat flux due to friction and the contact distribution on the contact surfaces of the friction clutch at any time during engagement. It was found that the highest temperature and contact pressure occurred when using Organic friction material.

Ye et al. [2] studied the thermodynamic process and dynamic behaviors in the friction clutches under different working circumstances. A mathematical model of a friction clutch was built. The dynamic characteristics were investigated based on LuGre friction model. Advanced Modeling Environment for performing Simulations of engineering systems (AMESim) was used to analyze thermal and dynamic process of a friction clutch.

The obtained results were compared with other approaches and the comparison proved that the clutch model was reliable.

Bogdanovich and Tkachuk [3] developed new solutions of the friction heat generated problems in the sliding systems. The solutions were introduced based on the Blok assumptions to find the friction heat generated and the flash temperature. The parameters affected values and distributions of the flash temperature on the contact area were investigated. The wear mechanisms of the contact sliding materials were studied taking into consideration the heat generation due to the friction.

Myklebust and Eriksson [4] studied experimentally the transmitted torque of a heavy duty dry friction clutch of truck (HDT). The results showed that the characteristic of transmitted torque depends slightly (ineffective) on the slipping speed. The dynamic model has been built and validated with experimental results. The errors in the dynamic results were reduced with 4 % using the new dynamic model of a friction clutch.

Pisaturo and Senatore [5] studied the thermal behavior of the friction clutch when the level of the temperature was higher than 300 °C. The effect of the temperature on the coefficient of friction was investigated. A modified new control algorithm to compute heat generated during vehicle launch and up-shift maneuves was presented. The heat generated was used as thermal load in the finite element simulation to find the temperature distribution during the multi-engagements. The results showed that the surface temperatures reached the crucial level of working temperature (around 300 °C) after a few engagements. The analysis proved that the working temperature affected the frictional behavior.

Zhao et al. [6] investigated the coupling of thermal and mechanical problem of multi-disc

clutch where the carbon-carbon composite material was used to achieve the analysis. Finite element method was used to obtain the thermal behavior of the friction clutch during the sliding period. Axisymmetric model was used to find the temperature and deformation fields. The results illustrated that high amount of thermal stresses appeared due to the excessive surface temperature. Furthermore, thickness and thermal properties of the friction material were investigated. The results demonstrated that the highest value of stresses was the hoop stresses.

Wenbin et al. [7] developed a finite element model to study the thermal performance of a carbon fabric wet clutch. The heat convection effect was taken into consideration in the developed model. The obtained results were compared with experimental results measured from thermometers. The influence of the thermal parameters on the surface temperature during engagements and failure of the carbon fabric composites were investigated. The results indicated that the level of the surface temperatures were low in case where the specific heat was low value and thermal conductivity was high value.

This paper highlights the importance of the effect of repeated engagements on the thermal behavior (distribution of surface temperature) of the friction facing of the dry clutches when the pressure on the surface of contact is uniform. The friction clutch disc was simulated with Finite element method to analyze the surface temperature during multi-engagements.

2. ENERGY CONSIDERATIONS

Heat Transfer is a kind of energy transportation, where the heat will be transferred by conduction, convection and radiation. The relative motion that occurred between the clutch parts, due to this fact one can be neglect the effect of radiation [1-4]. The input and output powers to the clutch system are explained in Figure 2. It was clear from mentioned figure that the energy was classified into two categories. The first category was the mechanical energy ($P_{m. input}$ and $P_{m. output}$) and the second one was the thermal energy ($P_{t. output}$). The heat energy represented the result of subtracting input mechanical power from output mechanical power. This heat energy converts to heat generation on the contacting area of the clutch elements. The surface temperature of the clutch surfaces will be increasing due to heat generation. During the initial time before the clutch reaches the full engagement, the maximum amount of heat will be generated.



Fig. 1. Major parts of a single-disc clutch system



Fig. 2. Input and output powers of the clutch system

The variations of the rotating speeds and the temperatures during the engagement cycle of the friction clutch are shown in Fig. 3. Basically the engagement cycle consists of two periods; the first one is called the engage period where the engine rotates with n_e and the clutch disc rotates with n_c . The second period is called engaged period, the clutch disc will be rotated at the same speed with engine and there is no slipping any more between them. In other words, the critical period in the clutch engagement cycle is the first period (slipping period) when the sliding occurs between contact surfaces and the friction heat generation will appear in the system.



Fig. 3. The variation of rotating speeds and temperature of the friction clutch system during a single cycle.

The energy balance of the friction clutch system during the slipping period can be written as follows [2],

$$\Delta Q = Q_g - Q_c, \qquad 0 \le t \le t_{sl.} \qquad (1)$$

where ΔQ , Q_g , and Q_c are the change in the internal energy of the friction clutch system, the friction heat generation during the sliding period and the amount of the convectional heat transfer, respectively. t_{sl} is the sliding time until reach the clutch disc the engine speed. One can find the internal energy of the friction clutch system at any time by using the following formula [2],

$$Q_{I.E.} = Q_{I.I.E.} + \Delta Q, \qquad 0 \le t \le t_{sl.} \quad (2)$$

where $Q_{LL\ E}$ is the internal energy of the friction clutch system at the initial condition. The entire amount of the heat generation through the sliding time is [8],

$$Q_g = \mu p_c \,\omega_r \, r, \qquad 0 \le t \le t_{sl.} \tag{3}$$

where μ , p_c , ω_s and r are the frictional coefficient, the contact pressure, the relative angular velocity and friction clutch disc, respectively. The function of the relative angular velocity is assumed as follows [1],

$$\omega_r(t) = \omega_i . (1 - \frac{t}{t_{sl.}}), \qquad 0 \le t \le t_{sl.}$$
(4)

where ω_i represents the relative angular velocity when the clutch starts sliding (t = 0). The variation of the heat generation on the friction clutch surfaces is [1],

$$Q_{g.c} = P_f \ \mu \ p \ r \ \omega_i (1 - \frac{t}{t_{sl.}}), \ 0 \le t \le t_{sl.}$$
(5)

where P_f is the partition factor of the heat generation partition ratio that represents the factor divides the heat generation into the friction clutch parts (friction clutch material, the pressure steel plate and the flywheel. In this work both of the pressure steel plate and the flywheel were made from the same material. One can write the formula of the partition factor as follows [9],

$$p_{f} = \frac{\sqrt{K_{c} \rho_{c} c_{c}}}{\sqrt{K_{c} \rho_{c} c_{c}} + \sqrt{K_{pr} \rho_{pr} c_{pr}}} = \frac{\sqrt{K_{c} \rho_{c} c_{c}}}{\sqrt{K_{c} \rho_{c} c_{c}} + \sqrt{K_{g} \rho_{g} c_{g}}}$$
(6)

where *K*, ρ and *c* are the material properties (conductivity, density and specific heat). The material properties of the pressure plate, flywheel and friction material were denoted with the subscript of *pr*, *fl* and *c* respectively.

3. FINITE ELEMENT FORMULATIONS

Thermal analysis using finite element method is considered efficient and common method in the different field of engineering and there are many researchers is used this technique [10-13]. The finite element equation which used to find the solution of eat conduction problem in any structure can be written as follows [14],

$$[C]{T} + [K]{T} = {R}$$
(7)

where [*C*], [*K*], {*T*}, {*T*} and {*R*} are matrix of the specific heat, matrix of the conductivity, nodal temperatures vector and the first derivative of temperature respect to time and the heat load.

The significant issue that should take into consideration to calculate the temperature distribution of the friction clutch during the transient period (slipping phase) with high precision is the element size of the finite element model. In this work, a suitable element size was selected after testing the results of the model. The Crank-Nicolson technique was chosen as an unconditionally stable scheme.

ANSYS15 (APDL) has been used to developed numerical model of the friction disc during the six successive engagements. It was assumed as uniform wear of the clutch surfaces for the complete calculations. All used materials (friction and steel) were isotropic and homogeneous, Table. 1 recorded all materials properties and operational parameters.

Table. 1. Model parameters and material propertiesused in the thermal analysis [1].

Internal radius of friction clutch disc [m]	0.075
External radius of friction clutch disc [m]	0.15
Friction material's thickness [m]	0.003
Axial cushion's thickness [m]	0.0015
Maximum pressure [MPa]	0.255
Coefficient of friction	0.3
No. of the frictional surfaces	2
Torque [N.m]	840
Initial angular sliding speed [rad/sec]	200
The conductivity (friction material)[W/mK]	0.5
The conductivity (pressure plate, flywheel and axial cushion) [W/mK]	54
The density (friction material) [kg/m ³]	1200
The density (pressure plate, flywheel and axial cushion) [kg/m ³]	7200
The specific heat (friction material) [J/kg K]	1550
The specific heat (pressure plate, flywheel and axial cushion) [J/kg K]	480
Initial temperature, T _i [K]	300
time step [s]	0.0005

It was assumed in the present simulation that the coefficient of heat transfer by convection is equal to $40.89 \text{ W/m}^2 \text{ K}$ [1] for all expose surfaces. The

slipping time was 0.5 s and the full engagement time was 2 s of each successive engagement. The 3-dimentional finite element model of the friction clutch disc (without grooving) is shown in Fig. 4.



Fig. 4. Thermal boundary conditions of a friction clutch system (a), and the finite element model of the frictional clutch disc (b).

The selected element to achieve the finite element simulation was SOLID90, where this kind of element has twenty nodes with a single degree of freedom (temperature). High accuracy of results was obtained in different applications especially with curved boundaries using SOLID90 element, and this was the reason to build the numerical model based on the mentioned element. The optimal mesh was selected based on the mesh sensitivity technique.

4. RESULTS AND DISCUSSIONS

A number of computational runs were made using numerical approach to investigate the effect of the successive engagements on the thermal behaviour of the dry friction clutches. The temperature distributions were found at any instant during the six successive engagements, where each engagement consists of two phases are slipping phase (0.5 s) and complete engagement phase (2 s).

Table 2 shows the results of the maximum temperatures of the friction clutch based on the approach was used in this work and Fu et al. [15]. The maximum difference between the present results and Fu et al. [15] does not exceed (1 %). The material properties and dimensions of the verification case study is listed in Table 3.

Table 2.	The results	of the	verification	case.
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	Maximum Temperature [K]
Present work	648.8
Ref. [15]	642.6
[%]	0.96

Table 3. Parameters and material properties of the verification case [15].

Properties	Friction material properties	Steel properties		
Modulus of elasticity [GPa]	0.07	200		
Thermal conductivity [W/mK]	0.25	48		
Specific heat capacity [J/kg K]	1337.6	480		
Density [kg/m ³]	1300	7800		
Dimensions of friction lining: inner and outer diameters = 252 mm & 386mm; Thickness = 5 mm				
Dimensions of steel disc: Thicknesses = 10mm & 5mm				



Fig. 5. The temperature distribution of friction clutch at different time steps during the first engagement.

The temperature distributions of the friction clutch at different time steps during the first and sixth engagements exhibited in Figs. 5 and 6. It was found that the temperature distribution were uniform over the contact surfaces of the friction clutch at any time during all engagements, except the small zones near the inside and outside disc radius of the clutch disc where the values of temperature were a little bit less than the surface temperature because of the convection effect.



Fig. 6. The temperature distribution of friction clutch at different time steps during the sixth engagement.

In all engagements, the maximum surface temperature happened at the middle time of the sliding time, the times of the maximum temperatures were 0.25 s, 2.75 s, 5.25 s, 7.75 s, 10.25 s and 12.75 s corresponding to the six successive engagements, respectively.



Fig. 7. The maximum temperatures at the inner, mean and outer radii during the sixth engagement.

Figure 7 shows the maximum surface temperatures at the inner, mean and outer radii during the sixth successive engagements. It can be seen that the maximum temperatures occurred at the mean radius at nay time during all engagements. Also, it's clear that the temperatures at the inner radius approximately equal to the temperatures at the outer radius.

Figure illustrated 8 the temperatures distributions through the thickness of the friction clutch disc at the mean radius during the 1st, 4th and 6th engagements. It can be noticed that there is no thermal effect on the mid thickness of the clutch disc until the 4th engagement (t=7.75 s). The reason behind theses results is the poor thermal properties of the friction materials that lead to concentrate the highest temperature at the surface layer of the lining friction material. Therefore, it's recommended to select the friction material with good thermal properties to avoid the rapid wear in the contact surfaces.



Fig. 8. The temperatures through the thickness of the friction clutch disc at mean radius.



Fig. 9. The maximum surface temperature through six successive engagements.

The variation of the maximum temperature of a friction clutch disc through the six successive engagements at different levels (in the direction of depth which started from the friction clutch surface until 0.5509 mm depth from the surface) is shown in Fig. 9. It can be seen for all the successive engagements that the values of temperature decrease rapidly when move away from the surface in the direction of the depth of the friction material. In the first engagement, the

maximum increments in the temperature at the clutch surface and at 0.55 mm from the clutch surface were 39 K and 3.6 K, respectively. While in the last engagement (6th), the maximum increments in the temperature at the clutch surface and at 0.55 mm from the clutch surface were 68.9 K and 31.6 K, respectively. It can be noticed, during all the successive engagements the temperatures raise dramatically with the engagements number.

5. CONCLUSIONS AND REMARKS

In this research paper, a finite element model was developed to study and analyze the thermal problem of the friction clutch disc throughout six successive engagements assuming a uniform wear between the contact surfaces. The friction clutch disc was simulated numerically using a threedimensional finite element model. The complete thermal behaviour during the six successive engagements were recorded and analyzed.

The essential point was that the maximum surface temperature is function of the number of the successive engagements, where the surface temperature grown very fast during engagements. Therefore, in the design of friction clutch it should take into consideration the maximum number of successive engagements under the specific condition and what is the maximum surface temperatures expected during these engagements. Under the heavy duty conditions, the maximum surface temperature expected should be lower than the allowable working temperature of the selected friction material, and the difference between them will depend on the chosen safety factor of the design operation.

Also, it was found that the critical time for the friction clutch system occurred at the mid time of the sliding period for all successive engagements, where the maximum temperature appeared in the contact surface at this moment.

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LIST OF SYMBOLS

- $\Delta Q \qquad \begin{array}{c} \text{change in the internal energy of the friction} \\ \text{clutch system} \\ Q_G \qquad \begin{array}{c} \text{the friction heat generation during the sliding} \\ \text{or which} \end{array}$
- Q_c period Q_c the amount of the convectional heat transfer
- t_{sl} Sliding time
- $Q_{LL\,E}$ Internal energy of the friction clutch system at the initial condition
- μ Coefficient of frictional,
- *pc* Contact pressure
- *ωs* Relative angular velocity
- p_f the partition factor of heat generation
- *k* Thermal conductivity
- ho Density
- c Specific heat