International Communications in Heat and Mass Transfer 71 (2016) 180–187

Contents lists available at ScienceDirect

International Communications in Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ichmt

Simulation and application of temperature field of carbon fabric wet clutch during engagement based on finite element analysis☆

Li Wenbin [a], Huang Jianfeng [a,*], Fei Jie [a], Cao Liyun [a], Yao Chunyan [b]

[a] School of Materials Science & Engineering, Shaanxi University of Science & Technology, Xi'an, Shaanxi 710021, PR China
[b] Culture and Communication School, Shaanxi University of Science & Technology, Xi'an, Shaanxi 710021, PR China

ARTICLE INFO

Available online 30 December 2015

Keywords:
Boundary lubrication thermal effects
Thermal analysis
Heat capacity
Thermal conductivity
Automatic transmissions
Self-lubricating composites

ABSTRACT

The temperature plays a significant role in the tribology properties and failure of friction materials during engagement of wet clutch. In order to obtain the temperature field of carbon fabric wet clutch, the thermal model was developed and the finite element analysis was conducted with the heat flux, convective and conductive heat-transfer taken into account. The predicted temperatures of thermometer hole were compared with experimental values. The effects of the thermal parameters on the temperatures of engagement and the damage of carbon fabric composites were investigated. Results show the thermal is evaluated as effective and can well predict the temperature field. The lower skeletal density, lower specific heat capacity and higher thermal conductivity are indispensable for the purpose of lowering the temperature of engagement. The highest temperature appears at R = 0.0509 m, where the damage of friction lining easily occurs.

1. Introduction

Wet clutches are used in a variety of machines, especially in automatic transmissions [1]. Large amounts of heat generated within a relatively short period of time cause a considerable and non-uniform temperature increase. The temperature always varies with the heat flux produced by friction and heat-transfer among the friction lining, lubrication oil, core disc and separator disc during engagement of wet clutch [2,3].

Numerous researches have been conducted on the temperature field of wet clutch during engagement. Based on the derived heat conduction equations, P. Zagrodzki et al. modeled the temperature field and thermal stresses of a sintered bronze wet clutch by numerical analysis method [4]. However, the predicted data of temperature field were not compared with the measured data. With the combined theory and the separation of variables technique taken into consideration, T.-C. Jen et al. developed a complimentary numerical model to simulate the temperature rise and temperature distribution [5]. However, the two-dimensional entity model was established and the constant energy engagement was assumed. A general thermal numerical model was presented by R.A. Tatara et al. based on the transient numerical solution of the heat conduction equation in two dimensions [6]. To predict the transient thermal response of wet clutch during engagement, a three-dimensional finite-volume-based numerical method was developed by Y.G. Lai et al. [7]. Subsequently, the model was applied to groove design. Therefore, the temperature field was usually simulated by numerical method and the friction lining was usually paper-based or bronze-based friction material.

The temperature has a very important influence on the torque transfer of the engagement of wet clutch. The high temperature can soften the friction materials and decrease the viscosity of oil, leading to the decrease of dynamic coefficient of friction. Therefore, the low and stable temperature contributes to the occurrence of the smooth torque transfer with little vibration and jerk. Many researchers have investigated the influence by simulation methods. To simulate the effects of temperature on torque transfer, C.L. Davis et al. developed a simplified isothermal model [8] and P. Marklund et al. developed a friction model based on boundary lubrication regime [9]. To model the influences of thermal on engagement process, a comprehensive formulation was presented by J.Y. Jang et al. with the governing equations, boundary conditions and numerical solution technique considered [10].

Moreover, the temperature has a very important effect on the tribology properties and failure of friction materials. The effects of operating conditions (interface pressure, rotating speed and total inertia) on the friction and wear performances of friction materials can be usually attributed to the temperature change during sliding [11–14]. The tribology behaviors are affected by the declining mechanical properties (elastic modulus and hardness) and the lubricating properties generated by the rise of interface temperature [15–17]. The excessive temperature is liable to burn the surface of friction lining and damage machine parts as well as lubrication oil, which would lead to unstable transmission.

☆ Communicated by Dr. W. J. Minkowycz.

* Corresponding author.
E-mail address: huangjf@sust.edu.cn (H. Jianfeng).

http://dx.doi.org/10.1016/j.icheatmasstransfer.2015.12.026
0735-1933/

What's more, the excessive temperature gradient easily generates the large thermal stress, which would accelerate the damage of friction lining.

The influences of temperature on surface carbonization were studied with the numerical method by H. Osanai et al. [18] and Y. Yang et al. [19]. J. Fei et al. explored the thermal properties of carbon fiber reinforced paper-based friction material (CFRPF) with TG-DTG analysis [20]. It was found that there are three stages in the thermal degradation of CFRPF, and the resin content has a great effect on the degradation temperature of the second stage. H. Zhao et al. introduced a new parameter to better investigate the effects of temperature, speed, and load on the friction coefficient of friction materials [21]. Moreover, the temperature can affect the viscosity of lubricating oil and then affect the shear stress produced by oil film. The uneven temperature would induce the uneven lining wear and the “hot spots” of separator plate. Therefore, it is very significant to get insight into the temperature field of engagement.

As new and important wet friction materials, carbon fabric reinforced phenolic composites have been widely used as friction lining in the wet clutch [22,23]. Compared with paper-based friction materials, carbon fabric composites exhibit the unique combination of wear-resistance [24], good load-carrying capacity [25], self-lubricating [26] and thermal stability properties [27] because of their orderly aligned structure. Numerous researches have been carried out on the tribology properties of carbon fabric composites [23,28,29]. However, the simulation of the temperature field of carbon fabric wet clutch was seldom reported.

In this paper, the thermal model of carbon fabric wet clutch was developed. The temperature field and temperature gradient were obtained by finite element analysis. Besides, the effects of thermal parameters on the temperatures of engagement and the damage of carbon fabric composites were explored. The damage is defined as the increasingly rough surface in the macro photography photographs and the appearances of matrix cracks and fiber breakages in SEM micrographs. The model can help design the clutch to limit the temperature to a safe level with no need of loosing torque. Meanwhile, the model can make it possible to simulate the torque transfer in application and extreme working conditions, which can't be effectively performed in a test rig.

2. Thermal model

The thermal model was developed to simulate the temperature field of carbon fabric wet clutch, when the surface of friction lining is in contact with the surface of separator disc, as shown Fig. 1. The lubrication oil can flow through the gap between the friction lining and the separator disc. Meanwhile, much of heat generated in the sliding interface is

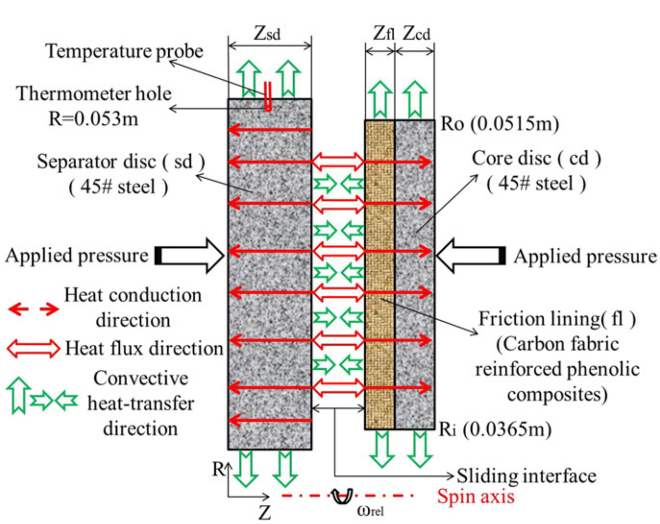

Fig. 1. Schematic sketch of computation domain.

transferred out from the clutch by lubrication oil. In the computations, there is also a certain amount of lubrication oil along the inner and outer circumference, which represents the lubrication oil sump. The applied pressure firstly increases to a set value in the initial stage and then remains consistent till the end of engagement process. The applied pressure is usually regarded as the set value during engagement in the previous researches [13,30], because the time of initial stage is very brief and the applied pressure of initial stage is lesser than the set value.

The heat flux, convective and conductive heat-transfer are the factors mainly considered in the thermal model. The thermal radiation is ignored because of its little influence on the thermal model. The simulated friction disc located in a clutch consisting of several similar discs, the heat conduction over the edges in axial direction is also neglected [4]. Thus, the outer edges ($z = 0$ and $z = Z_{sd} + Z_{fl} + Z_{cd}$) are assumed as insulated and Neumann boundary conditions (The normal derivative of function is constant for the boundary of carbon fabric wet clutch). Notably, the heat conduction can be calculated by the formulas that come from ANSYS 14.5 software, as shown in Eqs. (1)–(3).

$$\frac{\partial T_{sd}}{\partial t} = \frac{k_{sd}}{\rho_{sd} C_{p-sd}} \left(\frac{\partial^2 T_{sd}}{\partial r^2} + \frac{1}{r}\frac{\partial T_{sd}}{\partial r} + \frac{\partial^2 T_{sd}}{\partial {Z_{sd}}^2} \right) \tag{1}$$

$$\frac{\partial T_{fl}}{\partial t} = \frac{k_{fl}}{\rho_{fl} C_{p-fl}} \left(\frac{\partial^2 T_{fl}}{\partial r^2} + \frac{1}{r}\frac{\partial T_{fl}}{\partial r} + \frac{\partial^2 T_{fl}}{\partial {Z_{fl}}^2} \right) \tag{2}$$

$$\frac{\partial T_{cd}}{\partial t} = \frac{k_{cd}}{\rho_{cd} C_{p-cd}} \left(\frac{\partial^2 T_{cd}}{\partial r^2} + \frac{1}{r}\frac{\partial T_{cd}}{\partial r} + \frac{\partial^2 T_{cd}}{\partial {Z_{cd}}^2} \right) \tag{3}$$

where T_{sd}, T_{fl}, T_{cd} are the temperatures of separator disc, friction lining and core disk, respectively; t is the engagement time; r is the radius of the clutch discs; ρ_{sd}, C_{p-sd}, k_{sd} are the density, the specific heat capacity and the thermal conductivity of separator disc, respectively; ρ_{fl}, C_{p-fl}, k_{fl} are the skeletal density, the specific heat capacity and the thermal conductivity of friction lining, respectively. ρ_{cd}, C_{p-cd}, k_{cd} are the density, the specific heat capacity and the thermal conductivity of core disc, respectively; Z_{sd}, Z_{fl}, Z_{cd} are the thickness of separator disc, friction lining and core disc, respectively.

2.1. Heat flux

The heat flux is generated by friction work in the sliding interfaces of friction lining and separator disc. Based on abundant experiment data, the energy conversion factor (θ) is obtained with the energy loss (especially the friction work converted to mechanical energy and friction noise) taken into account during engagement. The heat flux is calculated by Eqs. (4)–(11). The dynamic coefficient of friction is fitted as a function of sliding speed and applied pressure according to the experimental data obtained from the MM1000-II wet friction performance tester, as shown in Eq. (4). The rise of applied pressure enlarges asperity contact area, which can be regarded as the smoother surface. Thus, the dynamic coefficient of friction decreases with the increase of applied pressure and the relationship corresponds to an inverse proportional function, which has been illustrated in our previous investigations [31]. In addition, the total energy of the system increases and the braking time is prolonged with the rise of rotating speed, which greatly increases the interface temperature. The increasing temperature decreases the oil viscosity and then decreases the shear stress produced by oil film. Besides, the rise of interface temperature can soften the friction lining, leading to the decrease of the dynamic coefficient of friction. Thus, the dynamic coefficient of friction decreases with the increase of sliding speed and the link is in agreement with a logarithmic function, which is in accordance with the previous researches of Refs. [13,32]. As seen in Fig. 2, the fitting curve from Eq. (4) has a good consistency with the experimental data, when the applied pressure is 1.0 MPa. Thus, the model can also be used at different sliding speeds and in