Proceedings of the 6th International Conference of Fluid Flow, Heat and Mass Transfer (FFHMT'19)
Ottawa, Canada – June, 2019
Paper No. FFHMT 120
DOI: 10.11159/ffhmt19.120

Modelling and Simulation of Lubricant Flow Fluid in Wet Friction Pair

Liyong Wang[1, 2], Qian Wang[1], Le Li[1], Changliang Tang[1]
[1]The Ministry of Education Key Laboratory of Modern Measurement and Control Technology, Beijing Information Science and Technology University
Beijing, China
wangliyong@bistu.edu.cn; wangqian_9212@163.com; Lile086@126.com
[2]Collaborative Innovation Center of Electric Vehicle in Beijing, Beijing Information Science and Technology University
No.12 Qinghe Xiaoying East Road, Haidian District, Beijing, China

Abstract **-** The lubricant flow field between the disks of the wet friction pair has been simulated under different lubricant inlet flow rate. The temperature changing on the surfaces of the disks during the wet friction pair operation is investigated. The temperature curve which was exported from the thermo-mechanical coupling simulation for the wet friction pair, is used to define the thermal field of the lubricant flow calculation filed. The viscosity-temperature characteristics of the lubricant is considered in this research as well. The simulation results of the lubricant flow fluid based on the different temperature and different flow velocity distribution show that grooves effect is the very important factor of the lubricant flow fluid. It is found that the flow velocity in the radial grooves and the zone near the walls contact to friction disk which has been grooved is larger while the temperature is lower. It is also shown that the temperature field and flow velocity field are distributing circularly and the closer to the outer edge the higher the flow rate and the higher the temperature is.

Keywords**:** Wet friction pair, Lubricant flow fluid, Temperature distribution, Flow velocity.

1. Introduction

Wet friction pairs have been extensively concerned due to its wide application in transmission components such as wet clutches, wet brakes, and liquid-adhesive speed-regulating clutches. A wet friction pair consists of a dual disk and a friction disk on which various groove patterns have been employed to improve its performance. During the engagement process, the relative velocity reduces to zero rapidly within a few seconds. In the meantime, a large amount of frictional heat is generated, leading to thermal failure of the friction pair surface and causing friction coefficient change sharply [1]. To provide effective cooling, the lubricant is forced to flow into the space between the friction disk and dual disk from the inner edge continuously [2]. In this paper, the flow fluid of the lubricant has been investigated.

Focus on the thermal and flow fluid of lubricant, domestic and foreign scholars have conducted extensive research based on transmission components. Payvar [3] developed a numerical solution to compute the heat transfer coefficient in the radial grooved of wet clutch, and compared the numerical to the experimental results which showed good agreement. Jang and Khonsari [4-5] presented a comprehensive Reynolds equation for modelling the thermal aspects of the engagement process in a wet clutch, considering the viscous heat dissipation in the fluid and heat transfer into the separator as well as the effect of roughness, permeability, centrifugal force, deformability, and groove geometry. Some scholars have studied the flow field under different working conditions of wet friction pair. Zazzaque and Kato [6] studied the effect of radial and off radial grooves on the wet clutch at disengaged state by presented the fluid dynamics model focusing only on the hydrodynamic behaviour of the wet clutch. Then Zazzaque and Kato [7] developed an analytical model to investigate the transient engagement characteristic of the wet clutch and the influence of the oil groove guiding, depth and area ratio applying the narrow groove theory. Some scholars have investigated the characteristic of wet friction pair lubrication friction. Marklund and Larsson [8-9] developed an engineering tool to simulated the friction characteristics of wet clutch working under boundary lubrication conditions and a simulation model in which the temperature and the torque transfer was estimated under limited slip conditions. Some scholars [10] research on two-phase flow within the disengaged wet clutch using numerical simulation and experimental method as well.

In the present paper, the lubricant flow fluid between friction disk and dual disk of the wet friction pair which temperature changes with time under specific working condition is focused on. The model takes into account the viscosity-temperature characteristics of the lubricant and the different inlet flow of the lubricant.

2. Modelling Method

The wet friction pair had been simplified before analysed. In this paper, we investigate the performance of radial groove pattern.

2.1. Geometric model

The structure schematic model of two wet friction disks separated by a thin film of lubricant is shown in Fig. 1. There are several radial grooves evenly distributed in the friction material. The lubricant between the two disks is forced to flow from inner edge to the outer edge under the centrifugal force and the pressure difference between the inner and outer diameters. In the meantime, the lubricant transfers heat with the surface of the disks to reduce the friction pair temperature.

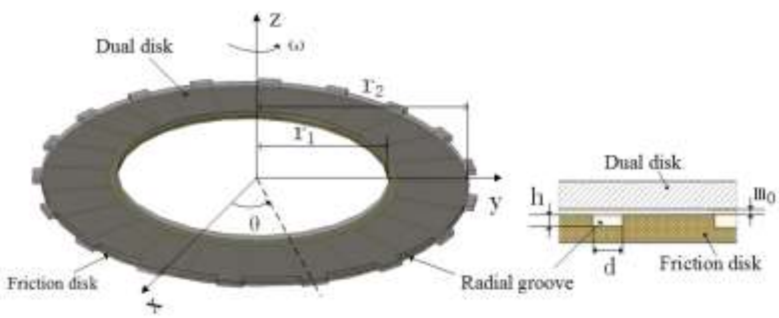

Fig. 1: Wet friction pair model.

2.2. Lubricant model

It is assumed that the lubricating is filled with the gap between the friction disk and the dual disk during the process. And the calculation field of the lubricant flow field between the disks is established as shown in Fig. 2(a). As we all see, the calculation field had been blocked for five parts which would be loaded thermal boundary conditions respectively in latter time. The meshed calculation field and some load boundary conditions is shown in Fig. 2(b). The periodic rotational symmetry method is applied on the simulation model. The lubricant inlet from inner edge and its temperature is taken as the ambient temperature. The walls of the lubricant calculation field contact to the surfaces of the disks are set to no slip boundary conditions.

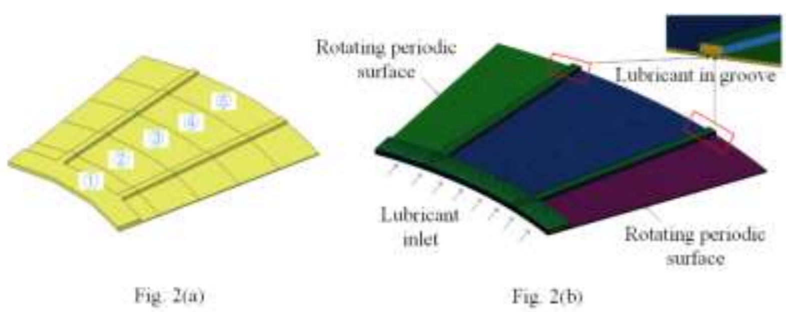

Fig. 2: Calculation field of the lubricant flow field and boundary condition.

2.3. Heat transfer model

There is a relative rotational speed between the friction disk and the dual disk during the operation, generating a large large amount of frictional heat, which is mainly diffused by heat conduction and heat convection with the lubricant. According to the theory of heat transfer, there is the established heat transfer equation of the friction pair in the Cartesian coordinate system:

$$\frac{\rho_i \cdot c_i}{k_i}\frac{\partial T_i}{\partial t} = \frac{\partial}{\partial x}\left(\frac{\partial T_i}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\partial T_i}{\partial y}\right) + \frac{\partial}{\partial z}\left(\frac{\partial T_i}{\partial z}\right) \tag{1}$$

Where ρ_i is the density, c_i is the specific heat, k_i is the thermal conductivity, T_i is the temperature of the friction element, t is the sliding friction time, and x, y, z are coordinates of the point of friction. The i takes a value of either 1 or 2, corresponding to the dual disk or the friction disk respectively.

3. Simulation

The thermal boundary condition applied on the calculation field of the lubricant flow field in this paper follow the numerical simulation model developed by Li [1]. The inlet lubricant temperature is set as ambient temperature 26 ° C and its flow rate is 5 L/min, 10 L/min, 20 L/min, and 30 L/min respectively.

3.1. Thermal condition

The temperature field distribution results of the analytical model simulated the friction process of friction pairs under the specific working condition that the rotational speed is 700 rpm is shown in Fig. 3. The temperature values of the temperature points on the steel sheet surface which named as i – j shown in Fig. 3(a) are exported in Fig. 3.

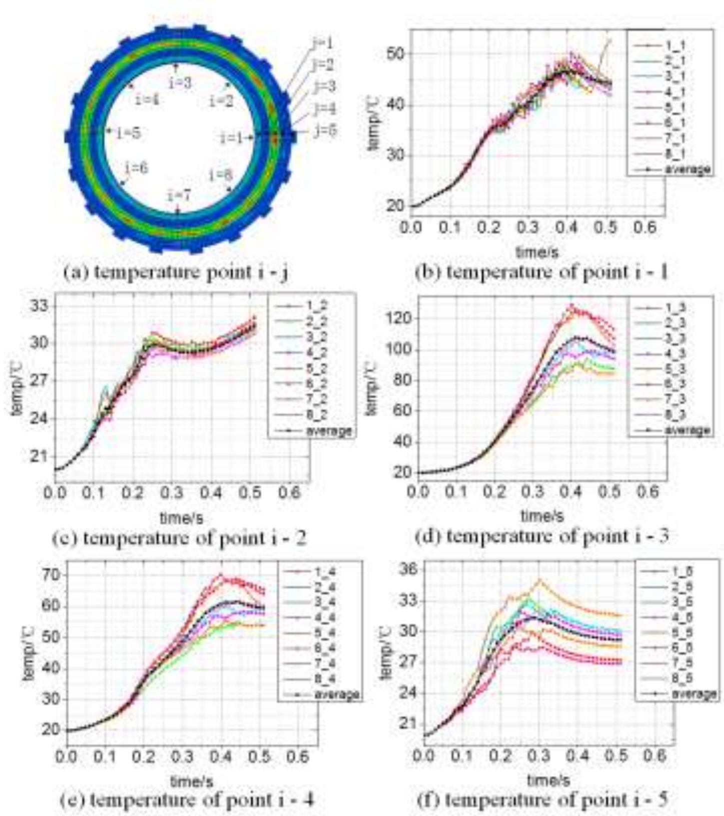

Fig. 3: Temperature curve of selected points on dual disk surface.

The temperature of each point on each circle changes with time as shown in Fig. 3(b-f). It is shown that the temperature on the surface of the dual disk increases gradually in the initial stage of sliding friction, and distributes in a ring shape. The average temperature curve of each circle is loaded on the each blocked calculation field of the lubricant as thermal boundary condition considering the hot spot areas on the surface of wet friction pair which is rotating as.

3.2. Lubricant material

The lubricant is assumed as incompressible Newtonian fluid in this paper, and its material properties are shown in Table. 1.

Table 1: Calculation parameters of lubricant.

Density ρ $(\mathrm{kg \cdot m^{-3}})$	Thermal conductivity k $(\mathrm{W \cdot (m \cdot {}^\circ C)^{-1}})$	Kinematic viscosity μ $(10^{-6}\mathrm{m^2 \cdot s^{-1}})$	Specific heat c $\mathrm{J \cdot (kg \cdot {}^\circ C)^{-1}}$	Pr
850	0.1~0.4	28.8~35.2	2231	1797

The viscosity-temperature characteristics of the lubricant has been taken into account considering the high temperature has a great influence on the viscosity of the lubricant [11]. The viscosity-temperature curve as shown in Fig. 4. And a user-defined function is used to apply the material properties on the calculation field of the lubricant.

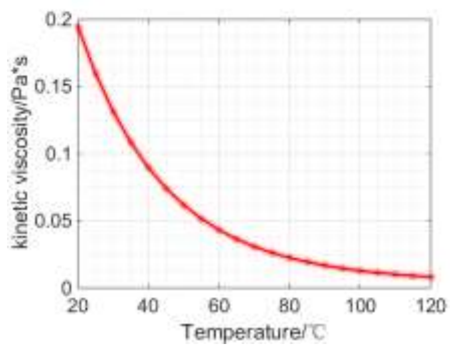

Fig. 4: The viscosity-temperature characteristics of the lubricant.

4. Result and Discussion

The simulation results of the lubricant flow field under different lubricant inlet conditions are shown in Fig. 5 and Fig. 6 respectively.

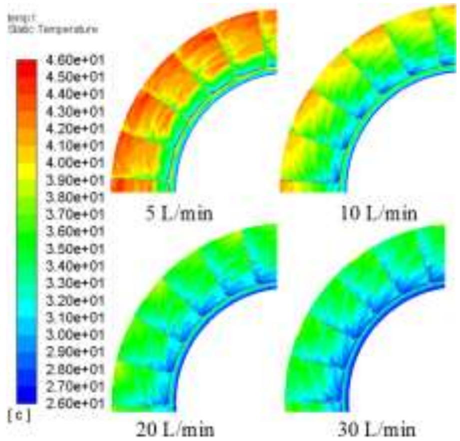

Fig. 5(a): Temperature distribution - 1.

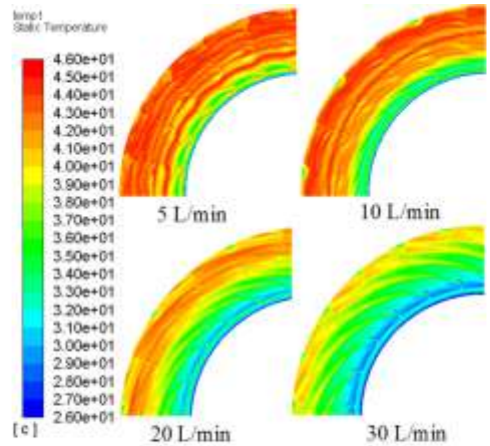

Fig. 5(b): Temperature distribution - 2.

Fig. 5(a) and Fig. 5(b) show the temperature distribution of the lubricant field contact to the grooved friction disk and and the dual disk. It can be seen from Fig. 5(a) that the lubricant flow field temperature is higher and almost all over the entire flow field when the lubricant inlet is 5 L/min. The temperature inside the grooves is clearly distinguishable from the temperature in un-grooved area which displays a certain cooling effect of the grooves. The flow field temperature is reducing while lubricant inlet increasing.

Fig. 5(b) shows that the temperature of the lubricant flow field is unevenly distributed due to the uneven temperature distribution on the wall of the disks. With the gradual increase of the lubricant inlet, the high temperature zone of the flow field gradually spreads outward. The high temperature annular band distribution occupies about one-half of the flow field as the lubricant inlet is 20 L/min and less than one third as 30 L/min. Comparing Fig. 5(a) with Fig. 5(b), it can be found that the temperature distribution at the surface contacted with the friction disk is significantly lower than that contacted with the dual disk.

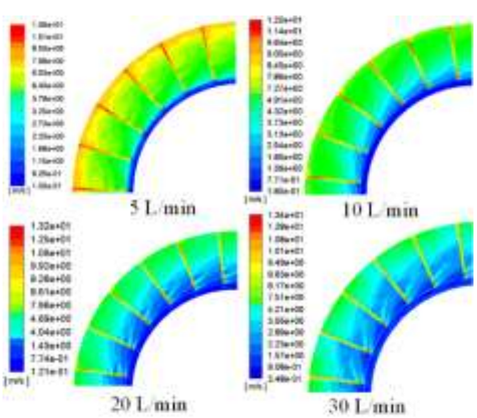

Fig. 6(a): Flow velocity distribution - 1.

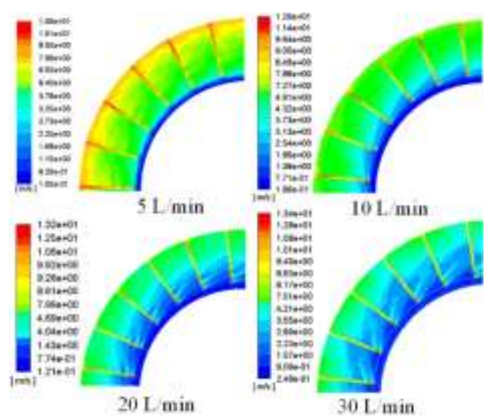

Fig. 6(b): Flow velocity distribution - 2.

The flow velocity distribution of the lubricant field contact to the grooved friction disk and the dual disk is shown in Fig. 6(a) and Fig. 6(b). It can be seen from Fig. 6(a) that the flow velocity in grooves and near the outer diameter is larger. The lubricant in grooves is pumped out under the centrifugal force due to the rotation of the frictional pair. In Figure 6(b), since the dual disk is not grooved, the flow velocity distribution of the lubricant contacted with the dual disk is uniform relatively. It can be seen that the grooves have a certain influence on the flow field velocity distribution.

It can be seen in Fig. 6(b) that the lubricant flow rate is also an annular band distribution, and the largest flow velocity is on the outermost zone while the smallest is on the innermost. The larger flow zone is gradually distributed to the outer diameter as the inlet flow rate increasing.

5. Conclusion

The lubricant flow fluid within two disks of the wet friction pair has been simulated as the lubricant inlet is different. It is shown that the thermal effect and the grooves effect play important roles on the lubricant flow fluid, based on the simulation results of the temperature and flow velocity distribution.

The flow velocity in the radial grooves near the walls contact to friction disk is larger while the temperature is lower at the same lubricant inlet condition. In the meantime, the temperature field and flow velocity field are distributing circularly. The closer to the outer edge, the higher the flow rate and the higher the temperature is. As the inlet flow rate of the lubricant increasing, the flow velocity gradually increases, the temperature gradually decreases, and the distribution of the flow velocity and temperature also changes accordingly.

The simulation model in this paper is established through the simplification of practical engineering problems, which can preliminarily characterize the lubricant flow fluid within two disks of the wet friction pair. The distribution of lubricant flow fluid and temperature field can provide a reference for the reliable operation of the wet friction pair.

Acknowledgements

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work was supported by the National Natural Science Foundation of China (No. 51605035) and completed with support from the Beijing Finance Fund for Science and Technology Planning Project (No. KZ201611232032) (No.KM201811232023), and Beijing Excellent Talent Training Project (No.2017000020124G019).

References

[1] L. Li, "Numerical analysis of dynamic characteristics of wet friction temperature fields," *J. Advances in Mechanical Engineering*., vol. 9, no. 12, pp. 1-14, 2017.

[2] E. H. Zhao, "Influence of Rotating Speed on Local Lubrication and Friction Characteristics of Wet Clutch," *J. Acta Armamentar*, vol. 38, no. 4, pp. 625-633, 2017.

[3] P. Payvar, "Laminar heat transfer in the oil groove of a wet clutch," *J. Heat Mass Transfer*., vol. 34, no. 7, pp. 1791-1798, 1991.

[4] J. Y. Jang and M. M. Khonsari, "Thermal Characteristics of a Wet Clutch," *J. Journal of Tribology.*, vol. 121, no. 1, pp. 610-617, 1999.

[5] J. Y. Jang and M. M. Khonsari, "Three-Dimensional Thermohydrodynamic Analysis of a Wet Clutch with Consideration of Grooved Friction Surfaces," *J. Journal of Tribology*, vol. 133, no. 1, pp. 1-12, 2011.

[6] M. M. Razzaque and T. Kato., "Effects of Groove Orientation on Hydrodynamic Behaviour of Wet Clutch Coolant Films," *J. Journal of Tribology*, vol. 121, no. 1, pp. 56-61, 1999.

[7] M. M. Razzaque and T. Kato, "Effects of a groove on the behaviour of a squeeze film between a grooved and a plain rotating annular disk," *J. Journal of Tribology*., vol. 121, no. 4, pp. 808-815, 1999.

[8] P. Marklund and F. Sahlin, "Modelling and simulation of thermal effects in wet clutches operating under boundary lubrication conditions," *J. Journal of Engineering Tribology.*, vol. 223, no. 8, pp. 1129-1141, 2009.

[9] P. Marklund and R. Larsson, "Wet clutch under limited slip conditions - simplified testing and simulation," *J. Journal of Engineering Tribology*., vol. 221, no. 5, pp. 545-551, 2007.

[10] Z. Xiong and S. H. Yuan, "Numerical Investigation of the Air-oil Two-phase Flow inside a Wet Clutch," *J. Journal of Mechanical Engineering*, vol. 52, no. 16, pp. 117-123, 2016.

[11] C. B. Xiong and B. Ma, "Experimental Study and Thermal Analysis on the Buckling of Friction Components in Multi-Disc Clutch," *J. Journal of Thermal Stresses*, vol. 38, no. 11, pp. 1325-1345, 2015.