Proceedings of the 9th International Conference on Fluid Flow, Heat and Mass Transfer (FFHMT'22)
Niagara Falls, Canada – June 08-10, 2022
Paper No. 191
DOI: 10.11159/ffhmt22.191

Numerical Investigation on the Flows within Friction Pairs of a Wet Clutch

Qian Wang[1], Biao Ma[1], Changsong Zheng[1], Liang Yu[1], Liyong Wang[2], Liangjie Zhen[1]
[1] School of Mechanical Engineering, Beijing Institute of Technology
No.5 Yard, Zhong Guan Cun South Street, Hai Dian District, Beijing, China
3120195217@bit.edu.cn; mabiao@bit.edu.cn; zhengchangsong@bit.edu.cn; 3120170250@bit.edu.cn
[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
wangliyong@bistu.edu.cn

Abstract **-** It is of great importance to study the lubricating oil flow field within the friction pairs of wet multi-disc clutches. Consideration of the oil inlet flow rate will be important for the cooling of friction pairs. In this study, a numerical model is developed with radial oil grooves taken into account, and the effect of the oil inlet flow rate and relative rotational speed between the friction pairs are analyzed. The results show that the lubricating oil flow velocity gradually increases with the increase of the inlet flow rate. However, when the relative rotational speed is low, the flow velocity gradually decreases in the radial direction, while it increases gradually under high rotational speed. Moreover, When the friction disc has radial oil grooves, the flow rate of lubricating oil in the oil groove is obviously greater than that of the oil film. The oil film flow velocity is smaller than that without the oil groove. This paper has an important reference value for improving the heat dissipation performance of wet clutch friction elements.

Keywords**:** Wet multi-disc clutch, Friction pairs, Lubricating oil, Flow velocity

1. Introduction

Wet multi-disc clutches are commonly used in high-power transmission systems. During the clutch engagement process, a large amount of frictional heat is generated on the interfaces of the friction components, which usually leads to a thermal failure of the friction surface and causes the friction coefficient change sharply [1-2]. In order to dissipate this friction heat, the lubricating oil is continuously pumped to flow through this friction pairs in radial direction [3]. It is of great importance to study the oil flow field within the friction pairs.

Regarding the research on the thermal effect of the flow field between the wet friction plates, Payvar developed a heat transfer model of the friction plate with radial grooves and solved the heat transfer coefficient in the groove [4]. Jang and Khonsari [5-6] found that thermal effects have an important impact on the transmission torque and clutch engagement time taking the friction surface permeability and roughness into consideration. Some scholars have also studied the flow field of the wet friction pair under different working operations. Based on fluid dynamics, Razzaque and Kato [7] studied the effect of different inclination angles of radial grooves on drag torque in idling wet clutch, as well as the number of oil grooves and flow rate of ATF. In view of the pressing process of the friction pair [8], the transient meshing characteristics of the wet clutch were studied by using the narrow groove theory, and the influence of the oil groove orientation, depth and area ratio on the pressing time of the friction pair of the clutch. Marklund and Larsson [9-10] simulated the friction characteristics of wet clutches under boundary lubrication conditions, using the technique of measuring boundary lubrication friction coefficients to predict wet clutches under limited slip conditions operating temperature and torque transfer. For the flow field between wet friction pairs, domestic and foreign scholars [11-12] also carried out theoretical numerical simulation and experimental research on the oil-gas mixed two-phase flow.

Based on simplification of the actual flow field in friction pairs, this paper focuses on the numerical solution of the flow situation of lubricating oil. The flow velocity distribution is complicated with and without radial oil grooves. The effect of relative rotational speed between the friction pairs and the amount of oil inlet flow rate is analysed. This study is extremely significant for improving the cooling performance of the friction pair in wet multi-disc clutches.

2. Modelling Method

2.1. Geometric model

In order to study the lubricating oil flow situation between the friction pairs, it is assumed that the lubricating oil is filled the plates gap during the sliding process, and a numerical calculation model is established theoretically. Figure 1 shows the schematic diagram of the friction pair and the force analysis of the oil unit between the friction pairs.

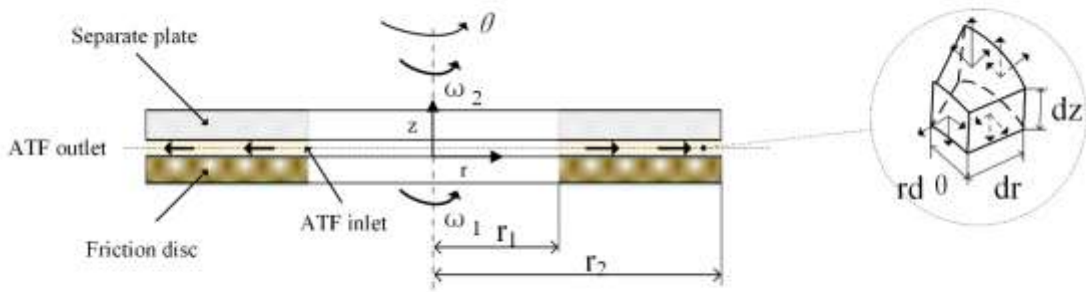


Fig. 1: Wet friction pair model.

In this Figure, r, θ, z are the respective coordinates of the cylindrical coordinate system; r1 and r2 are the inner and outer diameters of friction pair, respectively; ω1 and ω2 are the rotational angular velocities of friction disc and separate plate, respectively. Based on an assumption of the lubricating oil, a simplified Wiener-Stokes equation can be obtained as:

$$\begin{cases} -\dfrac{1}{\rho}\dfrac{\partial p}{\partial r}+\dfrac{\mu(t)}{\rho}\left(\dfrac{\partial^2 u_r}{\partial r^2}+\dfrac{1}{r}\dfrac{\partial u_r}{\partial r}-\dfrac{u_r}{r^2}+\dfrac{\partial^2 u_r}{\partial z^2}\right)=u_r\dfrac{\partial u_r}{\partial r}-\dfrac{u_\theta^2}{r} \\ \dfrac{\mu(t)}{\rho}\dfrac{\partial^2 u_\theta}{\partial z^2}=2\dfrac{u_r u_\theta}{r} \\ \dfrac{\partial p}{\partial z}=0 \end{cases} \tag{1}$$

The simplified continuity equation is:

$$\frac{\partial u_r}{\partial r}+\frac{u_r}{r}=0 \tag{2}$$

In the formula, p is the oil film pressure; u_r, u_θ, and u_z, are the radial, circumferential and axial velocity components of the lubricating oil, respectively; μ(t) is the dynamic viscosity of the lubricating oil.

2.2. Model solution

The boundary conditions are assumed as:

$$\begin{cases} u_r(r,0)=u_r(r,h)=0 \\ u_\theta(r,0)=\omega_1 r, u_\theta(r,h)=\omega_2 r \end{cases} \tag{3}$$

The oil inlet pressure is the largest, which is mainly determined by the relief valve, and the oil outlet is connected to the atmosphere. Therefore, the boundary conditions of the oil inlet and outlet pressure are as follow:

$$\begin{cases} p_{\max}\big|_{r=r_1} = p_0 \\ p\big|_{r=r_2} = 0 \end{cases} \tag{4}$$

Simultaneously simplify the Wiener-Stokes equation and the continuity equation, solve them and bring in the boundary conditions. Finally, the velocity components of the lubricating oil fluid along the radial and circumferential directions can be obtained as:

$$\begin{aligned} u_r = &-\frac{1}{60}\frac{r\rho z(\omega_2-\omega_1)}{\mu(t)h^2}\left[5(\omega_2-\omega_1)z^3+20\omega_1 hz^2+2(2\omega_2+3\omega_1)h^3-3(3\omega_2+7\omega_1)h^2 z\right] \\ &-\frac{3}{140}\frac{Q_0^2\rho z}{\mu(t)\pi^2 r^3 h^6}\left(14z^5-42hz^4+35h^2z^3-9h^4z+2h^5\right)-\frac{3Q_0(z-h)z}{h^3\pi r} \end{aligned} \tag{5}$$

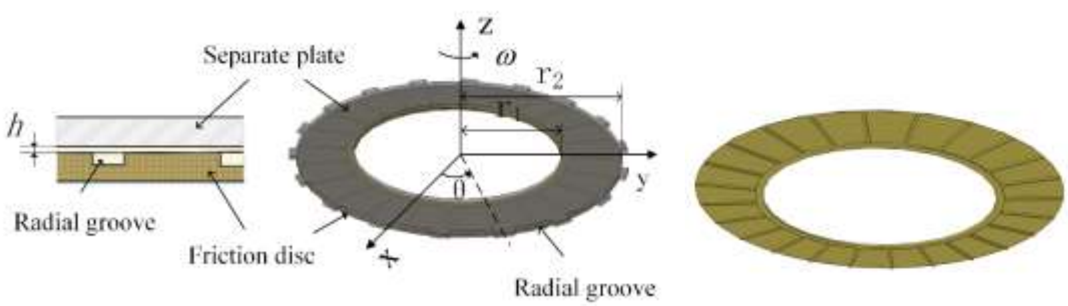
$$\begin{aligned} u_\theta = &\frac{Q_r\rho}{\pi r h^3\mu(t)}\left[\frac{3}{10}\frac{\omega_2-\omega_1}{h}z^5-\frac{1}{2}(\omega_2-2\omega_1)z^4-\omega_1 hz^3+\frac{2\omega_2+3\omega_1}{10}h^3 z\right] \\ &+\frac{\omega_2-\omega_1}{h}rz+\omega_1 r \end{aligned} \tag{6}$$

Therefore, the flow rate of lubricating oil between the wet friction pairs is:

$$u = \sqrt{u_r^2+u_\theta^2} \tag{7}$$

3. Simulation

A friction pair is simplified, as shown in Figure 2(a). The friction disc is made with radial oil grooves, whose depth is 0.75mm. The lubricating oil flow field is shown in Figure 2(b), where h is the oil film thickness, take as 0.3mm. The lubricating oil model without oil grooves is like this, and its thickness is the same, 0.3mm.



(a) Friction disc with radial grooves (b) Lubricating oil model

Fig.2: Friction pair with radial grooves

In order to compare the flow field distribution of the lubricating oil under different operating conditions, four lubricating oil inlet flow rates are set: 1, 5, 10, and 20 L/min, and four relative rotational speeds between the friction pairs are set: 20, 100, 500, and 1000 rpm. Taking the lubricating oil at 40°C for calculation, its material properties are shown in Table 1.

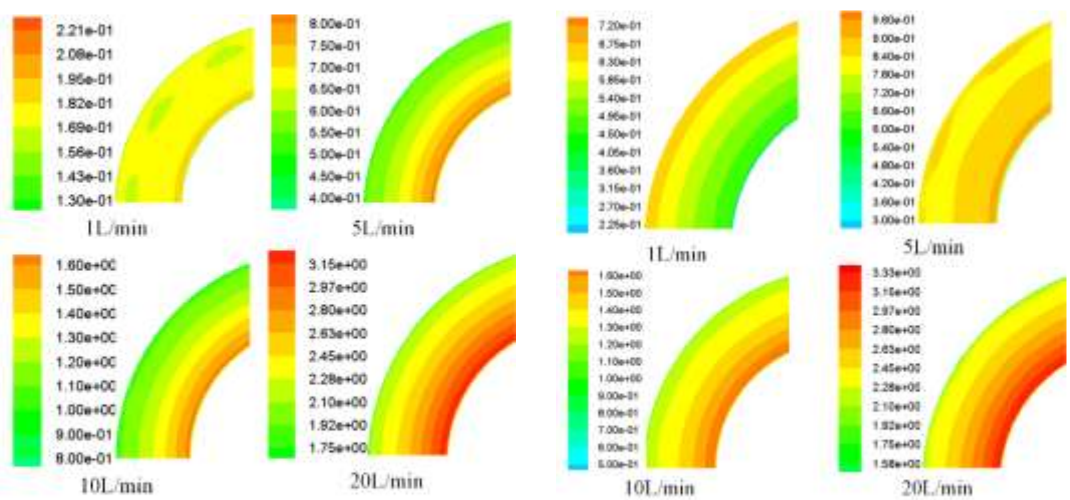
Table 1: Calculation parameters of lubricant.

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

4. Result and Discussion

4.1. Calculation results without oil grooves

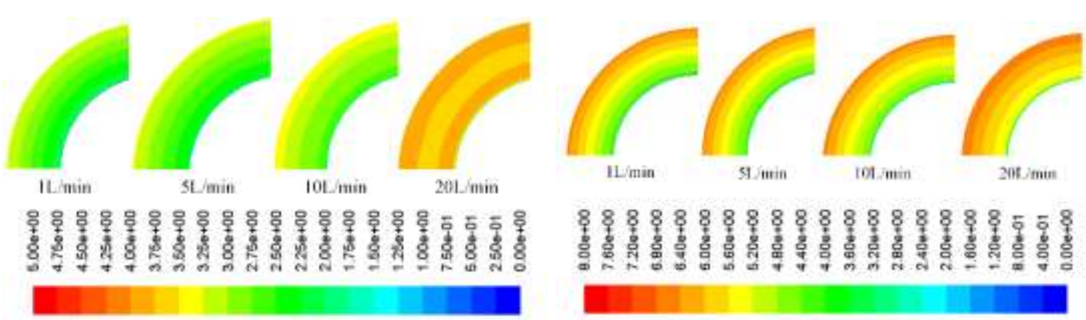
Figures 3(a) and (b) show the flow field distribution when the relative rotational speed are 20 and 100 rpm, respectively. The flow velocity of the lubricating oil gradually increases with the increase of inlet flow rate. The flow velocity at the inner diameter is the largest, and it gradually decreases along the radial direction. Notably, when the relative rotational speed is 20 rpm and the inlet flow rate is 1L/min, the flow velocity distribution is relatively uniform. When the relative speed is 100rpm and the inlet flow rate is 5 L/min, the lubricating oil velocity distribution has no obvious change. But under other conditions, the lubricating oil velocity has obvious gradient changes.



(a) 20 rpm (b) 100 rpm

Fig.3: Lubricating oil flow field rate distribution

Figure 4 (a) and (b) show the flow field distribution as the relative rotational speed are 500 and 1000 rpm, respectively. The flow velocity at the inner diameter is the lowest, and it gradually increases along the radial direction. At the 500rpm condition, the lubricating oil flow velocity changes little until oil inlet increase to 20 L/min, where the lubricating flow velocity is almost equal in the inner and outer edges. While at the 1000rpm condition, the lubricating flow velocity has no obvious change even the inlet flow rate increases to 20 L/min.

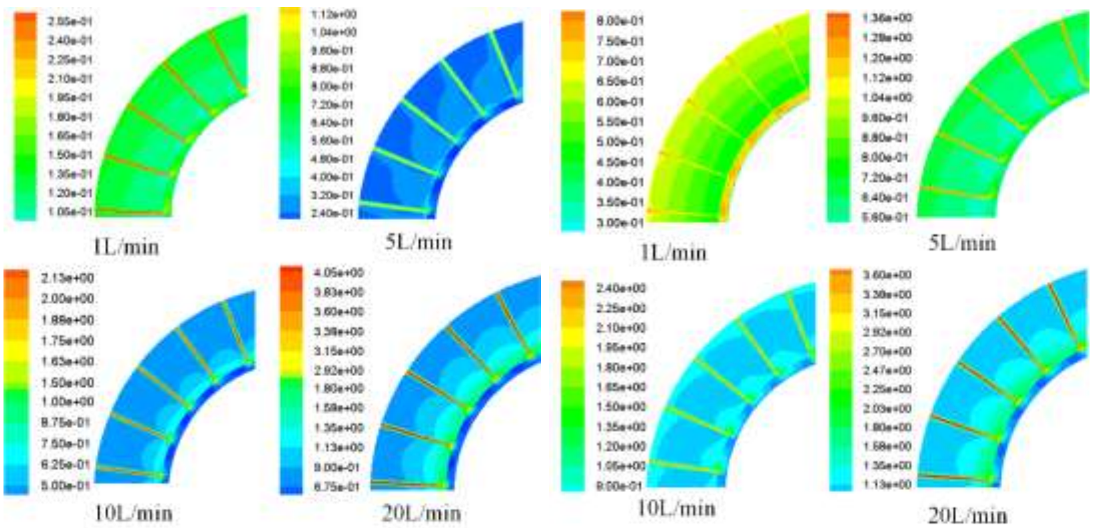


(a) 500 rpm (b) 1000 rpm

Fig.4: Lubricating oil flow field rate distribution

4.2. Calculation results with radial oil grooves

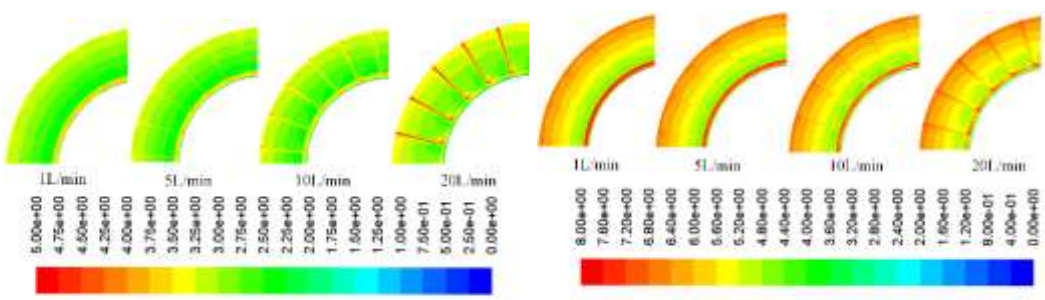
Figure 5 (a) and (b) show the flow field distribution when the relative rotational speed is 20 and 100 rpm, respectively. It can be seen that the flow velocity of lubricating oil in the oil groove is significantly greater than that of oil film. At a lower speed of 20 rpm, when the inlet flow rate is 1 L/min, the oil film velocity distribution is relatively uniform; when the inlet flow rate is 10 and 20 L/min, the velocity distribution is gradually smaller along the radial direction. At a higher speed of 100rpm, when the inlet flow rate is 5 L/min, the oil film velocity distribution is relatively uniform, and the increase in flow rate has a greater impact on the lubricating oil velocity distribution. Moreover, when the inlet flow rate is large, it has a weak effect on the distribution of the lubricating oil flow rate, mainly on the size of the flow rate.



(a) 20 rpm (b) 100 rpm

Fig.5: Lubricating oil flow field rate distribution

Figure 6 (a) and (b) show the flow field distribution when the relative rotational speed is 500 and 1000 rpm. The lubricating oil flow rate gradually increases in the radial direction. At inlet flow rates of 1 and 5 L/min, the flow rate did not change significantly with increasing oil inlet flow rate. When the relative rotational speed is 500rpm and the inlet flow rate is increased from 10 to 20 L/min, the flow velocity distribution changes greatly. However, when the relative rotational speed is 1000rpm, when the inlet flow rate increases from 10 to 20 L/min, the change of lubricating oil flow velocity distribution is small.



(a) 500 rpm (b) 1000 rpm

Fig.6: Lubricating oil flow field rate distribution

5. Conclusion

Numerical modelling of the lubricating oil flow field between the friction pairs in a wet multi-disc clutch is carried out, and the flow velocity distribution of the lubricating oil is calculated with the radial oil grooves. Different inlet flow rates and relative rotational speed conditions are taken into consideration. The main conclusions are as follow:

As the inlet flow rate increases, the lubricating oil flow velocity gradually increases. When the relative rotational speed is low, the flow velocity gradually decreases in the radial direction, and the flow velocity is greatly affected by the inlet flow. When the relative rotational speed is high, the flow velocity increases gradually along the radial direction, and the increasing gradient becomes larger and larger. As the rotational speed increases, the flow velocity distribution is gradually reduced by the influence of the lubricating oil inlet flow, which is mainly affected by the flow velocity.

When the friction plate has radial oil grooves, the flow rate of lubricating oil in the oil groove is obviously greater than that of the oil film. The oil film flow velocity is smaller than that without the oil groove, while the flow velocity in the oil groove is greater than the oil film flow velocity without the oil groove. This paper has important reference value for improving the heat dissipation performance of wet clutch friction elements.

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. 52175037, No. 51975047, and No. 51805289) and completed with support from the Beijing Key Laboratory Foundation (No. KF20212223201).

References

[1] Z. Zhang, X. Shi, and D. Guo, "Dynamic Temperature Rise Mechanism and Some Controlling Factors of Wet Clutch Engagement," *Mathematical Problems in Engineering*, vol. 2016, pp. 1-12, 2016.

[2] C. Xiong, B. Ma, H. Li, F. Zhang, and D. Wu, "Experimental Study and Thermal Analysis on the Buckling of Friction Components in Multi-Disc Clutch," *Journal of Thermal Stresses*, vol. 38, no. 11, pp. 1323-1343, 2015.

[3] E. Zhao, B. Ma, and H. Li, "The Tribological Characteristics of Cu-Based Friction Pairs in a Wet Multi-disk Clutch Under Nonuniform Contact," *Journal of Tribology*, vol. 140, no. 1, 2018.

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

[5] J. Y. Jang, M. M. Khonsari, "Thermal Characteristics of a Wet Clutch," *Journal of Tribology*, vol. 121, no. 3, pp. 610-617, 1999.

[6] J. Y. Jang, M. M. Khonsari, and R. Maki, "Three-Dimensional Thermohydrodynamic Analysis of a Wet Clutch With Consideration of Grooved Friction," *Journal of Tribology*, vol. 133, no. 1, pp. 12-11703, 2011.

[7] R. M. Mahbubur and K. Takahisa, "Effects of Groove Orientation on Hydrodynamic Behavior of Wet Clutch Coolant Films," *Journal of Tribology*, vol. 121, no. 1, pp. 56-61, 1999.

[8] R. M. Mahbubur and K. Takahisa, "Effects of a Groove on the Behavior of a Squeeze Film Between a Grooved and a Plain Rotating Annular Disk," *Journal of Tribology*, vol. 121, no. 4, pp. 808-815, 1999.

[9] P. Marklund, F. Sahlin, and R. Larsson, "Modelling and simulation of thermal effects in wet clutches operating under boundary lubrication conditions," *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, vol. 223, no. 8, pp. 1129-1141, 2009.

[10] P. Marklund and R. Larsson, "Wet clutch under limited slip conditions - simplified testing and simulation," *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, vol. 221, no. 5, pp. 545-551, 2007.

[11] Y. Yuan, P. Attibele, and Y. Dong, "CFD Simulation of the Flows Within Disengaged Wet Clutches of an Automatic Transmission," *SAE International*, vol. 320, no. 1, pp. 3-6, 2003.

[12] B. Xiao, W. Wu, J. Hu, and S. Yuan, "Fluid-Solid Coupled Heat Transfer Investigation of Wet Clutches," *SAE International*, 2017.