Study on engaging characteristics of wet friction pair with copper-based powder metallurgical friction material

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Abstract: Copper-based powder metallurgy (Cu-PM) has become quite common as the frictional material for clutches in transmission system. However the traditional research on friction pair is still limited to the static, qualitative and semi empirical studies. Based on the study of the physical properties of Cu-PM materials, the mathematical model of the wet friction pair is established. Through a simulation example, the transmission torque and the oil film pressure are obtained. Furthermore, a test bench has been set up to test the wet friction pair with Cu-PM friction materials. The correctness of the mathematical model was proved by comparing the experimental data and simulation data.

1. Introduction

The friction pair is one of the key parts of the wet clutch in the transmission. It consist of friction discs and steel discs [1-2]. Under the action of pressure, the friction disc joins the steel disc to transfer the torque of the transmission system. The performance of the frictional pair directly determines the design level and performance of the transmission system. The structure of the friction pair is shown in figure 1, and the friction discs and steel discs are alternately arranged. In the process of wet clutch engagement, axial pressure makes the friction sheet and dual steel sheet stick together. Torque transfer from input shaft to output shaft by friction force on the contact surface between the discs and viscous shear force of the lubricating oil film.

Figure 1. The structure of the friction pairs

The mechanism of torque transfer in the dynamic process of engagement friction pair has been studied step by step. Burton [3-5] pointed out that the friction torque is produced by the shear torque of the lubricating oil when the oil film thickness is thicker, so the friction material and its lubrication characteristics have a decisive influence on the friction performance of the friction pair. Matsumoto [6-7] studied the influence of elastic deformation and permeability of friction materials on the engagement performance. Komvopoulos [8] found that rough surface friction is the main mechanism of boundary lubrication. Razzaque [9-10] introduced the influence of the groove shape of the friction plate into the study of the ring extrusion film. Liu [11] simplified the friction pair into an axisymmetric model, neglecting the groove and the centrifugal action of oil. Zhu and Wang [12-13] established the finite
element thermal analysis model. But the effect of lubricating oil and the groove on the friction disc are neglected.

In previous studies, the permeability of materials has been neglected, such as references [9-10], and the shape of grooves is seldom involved, such as references [11-13]. In this paper, the mathematical model of Cu-PM material friction pair with radial grooves was established and the boundary conditions considering the permeability of materials are also established. The research in this paper has guiding significance for the design and manufacture of friction pairs of Cu-PM materials.

2. Physical properties of Cu-PM friction materials and their effect on friction pair performance

2.1 Physical properties of Cu-PM friction materials

The friction material used in this study is Cu-PM friction material (figure 2). It is made of copper or copper alloy with strengthening elements, lubrication and friction elements. The strengthening elements are iron, nickel, aluminum, tin, zinc and manganese, the friction elements are silicon dioxide, silicon monoxide and zirconia, and the lubricating elements are graphite and molybdenum disulfide. The specific mass fraction is shown in table 1.

![Figure 2. Copper based powder metallurgy friction disc](image)

**Table 1. Composition and physical properties of Cu based powder metallurgy friction materials**

| Component | Characteristic |
|-----------|---------------|
| Cu        | Sn            | Pb | Fe | C | SiO₂ | Density, kg·m⁻³ | Hardness, HB | Thermal conductivity, W·m⁻¹·K⁻¹ | Ultimate strength, MPa |
| 68~76     | 8~10          | 7~9 | 3~5 | 6~8 | 2~4 | 6.0~6.2 | 25~28 | 15.5 | 24.0 |

The Cu-PM friction material has the advantages of good thermal conductivity and stable friction coefficient. The sliding friction coefficients $f_c$ at different relative sliding velocities $v_{rel}$ are obtained by the sliding experiment of friction discs and steel discs, and then the relationship between the sliding friction coefficient $f_c$ and the relative sliding velocity $v_{rel}$ is established as below.

$$f_c = 0.13 - 0.008\log_{10}(v_{rel})$$

(1)

2.2 Influence of Cu-PM friction material on contact surface slip

The microstructure of the Cu-PM friction material is shown in figure 3[14]. The observation scale is 100 μm. It can be seen from the figure 3 that the internal pores of the Cu-PM friction material are criss-crossed and interpenetrated, and the porosity is high. Under the pressure, the lubricating oil can discharge the gap between the friction disc and the steel disc from the pores.

At the contact interface between the lubricating oil film and the friction material, the velocity $u$ of the lubricating oil in the radial direction is no longer the same as the radial velocity of the friction plate, that is, $u \neq 0$, and the sliding boundary condition of the lubricating oil can be expressed as:
In formula (2), \( \alpha = 0.2 \) is the Beavers-Joseph slip factor, \( \Phi \) is the permeability of the porous friction material, and \( u_B \) is the velocity boundary condition of the lubricating oil in the radial direction of the contact interface with the porous friction material, as shown in figure 4.

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2.3 Influence of Cu-PM friction material surface roughness on friction torque contact surface slip
The friction torque transmitted by the friction pair is composed of the viscous shear torque of the lubricating oil and the friction torque. The friction torque is calculated by the rough surface model (figure 5) and the friction coefficient model.

According to the Hertz theory and statistics, the expression of the real contact area \( A_c \) and the contact pressure \( F_c \) is:

\[
A_c = \pi \eta A_N \beta \sigma F_c \left( \frac{h}{\sigma} \right)
\]

\[
F_c = \frac{4}{3} \eta A_N E' \beta^{1/3} \sigma^{2/3} F_N \left( \frac{h}{\sigma} \right)
\]

where \( \eta \) is the distribution density of the micro-convex peaks on the rough surface, \( \sigma \) is the standard deviation of the height-probability distribution, and \( A_N \) is the rough surface and the smooth surface, \( E' \) is the equivalent elastic modulus.

3. Dynamic simulation of friction pairs of Cu-PM friction materials.
3.1 Dynamic model of friction pair engagement process
In the process of friction pair engagement, under the action of axial loading pressure (figure 6), the friction disc is gradually bonded with the steel disc, and the torque is transmitted by friction torque and viscous shear torque.

The oil film between the friction discs and the steel discs is a ring film. The axial load during the engaging process is less than 5MPa. The differential equations of motion of viscous incompressible fluid are formulated in cylindrical coordinates as follows.

\[
\begin{align*}
\rho \left( \frac{Du}{Dt} \right) &= \rho \cdot f - \frac{\partial p}{\partial r} + \mu \left( \nabla^2 u - \frac{u}{r^2} \frac{\partial v}{\partial \theta} \right) \\
\rho \left( \frac{Dv}{Dt} \right) &= -\frac{u}{r} \frac{\partial p}{\partial \theta} + \frac{1}{r} \frac{\partial p}{\partial \theta} + \mu \left( \nabla^2 v - \frac{v}{r^2} \frac{1}{r} \frac{\partial u}{\partial \theta} \right) \\
\rho \left( \frac{ Dw}{Dt} \right) &= \frac{\partial p}{\partial z} + \mu \nabla^2 w
\end{align*}
\]  

In formula (5), \( u, v \) and \( w \) are the velocity components along the direction of \( r \), \( \theta \) and \( z \), \( \rho \) is the density of lubricating oil, \( \mu \) is the dynamic viscosity of lubricating oil, \( f_r, f_\theta \) and \( f_z \) are the components of unit mass of lubricating oil in the direction of \( r \), \( \theta \) and \( z \), respectively.

\[
D = \frac{\partial}{\partial t} + \frac{\partial}{\partial r} \left( u \frac{\partial}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left( v \frac{\partial}{\partial \theta} \right) + \frac{\partial}{\partial z} \left( w \frac{\partial}{\partial z} \right)
\]

\[
\nabla^2 = \frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} + \frac{1}{r^2} \frac{\partial^2}{\partial \theta^2} + \frac{\partial^2}{\partial z^2}
\]

3.2 Method for solving dynamic model
In this study, the multi-grid method is used to solve the difference equation. The solution of the modified Reynolds equation difference form at each level is solved by the line relaxation method. The expressions (5) and (6) can be reduced to the following form:

\[
\begin{align*}
A_0 p_{i,j}^{n+1} + A_1 p_{i,j+1}^{n+1} + A_2 p_{i+1,j}^{n+1} &= b - A_3 p_{i+1,j}^{n} - A_4 p_{i-1,j}^{n} \\
- A_0 p_{i+1,j+1}^{n+1} - A_2 p_{i+1,j-1}^{n+1} - A_4 p_{i-1,j+1}^{n+1} - A_4 p_{i-1,j-1}^{n+1} &= 0
\end{align*}
\]  

Given \( p_{i,j}^{n} \) is pressure of each node in the n iteration step, the pressure \( p_{i,j}^{n+1} \) of each node in the \( n+1 \) iteration step can be solved according to equation (7).

3.3 Example
The dynamic characteristics of friction pair during the engagement process are obtained by solving the numerical model of wet friction pair with radial groove. The input parameters are shown in table 2.

Through numerical solution, the dynamic load characteristics of the friction pair are obtained. The transmission torques are shown in figure 7. At the beginning of the engaging process \( (t=0s, \ n_1=1500\text{rev/min}) \), the process is in the stage of hydrodynamic lubrication, and the load pressure of oil
film is higher; with the increase of axial load, the thickness of oil film decreases rapidly, and the process enters the stage of boundary lubrication (t=0.25s). When the viscous shear torque $T_h$ of oil film reaches its peak value, it begins to decrease, and the friction torque $T_c$ increases continuously. When the relative angular velocity is 0 (t=1.4s), the viscous shear torque $T_h$ of oil film decreases to 0, and the friction torque $T_c$ increases to the maximum value. The wet friction pair engagement process is completed.

Table 2. Input parameters in the experiment

| Parameter                              | Value       |
|----------------------------------------|-------------|
| Friction disc inner diameter $a$        | 0.051 m     |
| Friction disc outer diameter $b$        | 0.0685 m    |
| Thickness of friction material $d$      | $5.0 \times 10^{-4}$ m |
| Friction disc speed $n_1$               | 1500 r/min  |
| Loading pressure $F_{app}$              | 1.0 MPa     |
| Moment of inertia $I$                   | 0.5 kg·m²   |
| Initial lubricant film thickness $h_0$  | $2.54 \times 10^{-5}$ m |

The pressure distribution of the lubricating oil film is also obtained by the numerical solution. Figure 8 shows the pressure distribution of the oil film with radial groove at $t=0.25$s. It is found that the pressure of the oil film is 0 in the inner and outer diameters, and the pressure of the oil film is very small except around the positions of the groove, and it is periodically distributed in the circumference. The maximum value of oil film pressure appears in the middle of groove. This is because the thickness of oil film changes in the groove position. According to the hydrodynamic pressure effect, the oil film pressure in the groove position increases and the peak value appears.

Figure 7. Engaging torque curve of friction discs with radial groove

Figure 8. Pressure distribution on friction disc (t=0.25s). (a) 1/16 oil film, (b) complete oil film
4. Test bench and experimental results analysis

A test bench (figure 9) was built to test the wet friction pair with Cu-PM friction materials. It includes motor, flywheel, torque sensor, speed sensor and test box. During the test, the starting and stopping of the motor, the speed of the motor, the pressure of the hydraulic system and the oil supply of the lubrication system are controlled by computer. The maximum loading pressure is 10MPa, the maximum fuel flow is 10L/min, and the acquisition card model is MCC USB-2408 which has 12-bit resolution and 500 kS/s sampling rate.

![Figure 9. Dynamic performance test bench for friction pair engaging process](image)

The experimental data of relative rotational speed and transfer torque during the engaging process are compared with the simulation results as shown in figure 10.

![Figure 10. Comparison of experimental and simulation results (a) relative speed; (b) torque.](image)

Figure 10 (a) shows that the experimental engaging time is $t_{exp} = 3.923s$, which is 4.84% larger than the simulation result $t_{simu} = 3.903s$. This is probably because of the pressure system of the test bench has some delay. The experimental peak value of the transmission torque is about $T_{max, exp} = 9.01 N.m$, and the simulation result is $T_{max, simu} = 9.05 N.m$ (figure 10 (b)). The experimental results of torque and relative speed are consistent with the simulation results.

5. Conclusions

In this paper, the physical properties of Cu-PM friction material were studied, and its influence on the friction pair was analyzed. The mathematical model of engaging process of the friction pair with Cu-PM materials was established. Experimental data prove the correctness of this model.

During the engaging process, the viscous shear torque $T_h$ gradually increases and reaches its peak value at $t=0.07s$, then begins to decrease gradually. Meanwhile, the friction torque $T_c$ increases rapidly at the beginning, slows down at $t=0.25s$ and reaches its peak value at $t=1.4s$, when $T_h$ becomes 0.

The pressure of the oil film is zero in the inner and outer diameters, and the pressure of the oil film is very small except around the positions of the groove. The shape of groove plays a significant role in
the oil film thickness variation. Therefore, the influence of different groove shapes, such as spiral groove and square groove, on the friction pair performance of Cu-PM is a promising direction for further research.

References
[1] Matsumoto T, Ohuma M and Kuse T 1998 Drive System Technique 2 p16-23
[2] Matsumoto T 1994 SAE Transactions: Journal of Passengers Cars 103 p1445-56
[3] Burton R A and Gainesburton R. Tribology Transactions 35(4) p756-760
[4] Burton R A 1960 Tribology Transactions 3 (1) p1-10
[5] Burton R A 1980 Wear 59 p501-508
[6] Matsumoto T 1993 SAE Paper 932924
[7] Miyazaki T and Matsumoto T 1998 Tribology 120 (4) p393-398
[8] Komvopoulos K and Suh N P 1985 ASME
[9] Razzaque M M and Takahisa Kato 2001 Tribology Transactions 44 (1) p97-103
[10] Razzaque M M and Kato T 1999 Journal of Tribology 121 (1) p56-61
[11] Liu X C, Zhang Z G and Shi X H 2015 J Chongqing (Natural Science Edition)
[12] Zhu H Q 2012 Sliding Friction Characteristics and Heat Load Characteristics of Wet Clutch Zhejiang University (Doctoral dissertation)
[13] Wang H W, Zhang X Q and Zhang J L 2013 Journal of Beijing University of Technology 33 (1) p47-51
[14] Wang X F, Huang Q Z and Yin X L 2008 Journal of Central South University (Natural Science Edition) 03 p517-521