

Research on Friction and Thermal Load Performance of Hydro-viscous Clutch

Li-gang Ma, Chang-le Xiang, Ming-gang Du and Liu-yang Guo

Abstract In this paper, the theory of friction coefficient and thermal loads in slip state of a new hydro-viscous clutch has been studied and the influencing factors are determined. Furthermore, the friction characteristics test is designed. The test results show the variation tendency of friction coefficient when the hydro-viscous clutch operates at the difference temperature, difference normal pressure and difference rotating speed. The formula of different rotating speed on the impact of the coefficient of friction is fitted. The thermal load theory calculation reflects the temperature rising trend under different thermal loads of hydro-viscous clutch. Comparing the theoretical calculation and experimental results and analyzing the heat dissipation way of hydro-viscous clutch, it shows that the equation to calculating the thermal load that described the frictional state of frictional disk of hydro-viscous needs to be developed.

Keywords Hydro-viscous clutch · Friction coefficient · Thermal load · Testing by experiment

1 Introduction

Hydro-viscous clutch has the advantages of stepless speed regulation, small volume, high transmission power, and high reliability. Hydro-viscous clutch has been widely used in civil industry, mainly used in pumps, fans, and other large mechanical equipments in domestic. Its application fields are usually industries like coal and oil. Nowadays, it is widely used in the speed control device of tracked vehicles cooling fan in the military industry.

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The hydro-viscous clutch applies internal lubricant of transmission as a medium whose temperature is up to 120 °C. Therefore, the copper-based friction plates with high temperature resistance are chose as the friction element. In view of the characteristics of lubricating medium and friction element, the friction characteristics and thermal load of the hydraulic clutch are studied in this paper.

The hydro-viscous clutch works in the friction state over long term. The friction state of active and passive plates, as Stribeck curves shown can be divided into three states—boundary lubrication, mixed lubrication, and hydrodynamic lubrication—as shown in Fig. 1.

The processes from difference speed to combining of the hydro-viscous clutch are mainly in hydraulic lubrication and mixed lubrication region. According to the different lubrication states corresponding to the typical value of the friction coefficient, as shown in Fig. 2, the friction coefficient of the hydro-viscous clutch should be less than 0.1, especially when the hydro-viscous clutch is completely working in the phase of the hydraulic lubrication and the friction coefficient is less than 0.05.

Due to the long-term friction of the friction plate, a large amount of heat is generated in the hydraulic and mixed lubrication region. If there is no enough oil to take away the heat, the heat of the clutch will gradually accumulate. Once reaching a certain limit value, the friction will stick together or even burn out. Therefore, it is necessary to study the thermal load bearing capacity of the clutch under the limited lubrication. In this paper, the friction and thermal load characteristics of the clutch are studied.

Fig. 1 Stribeck curve

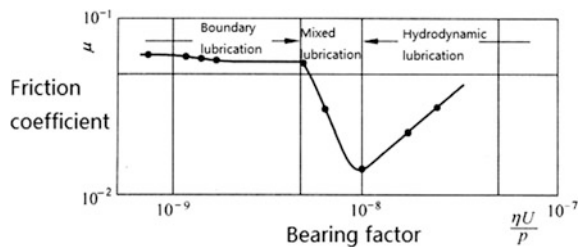
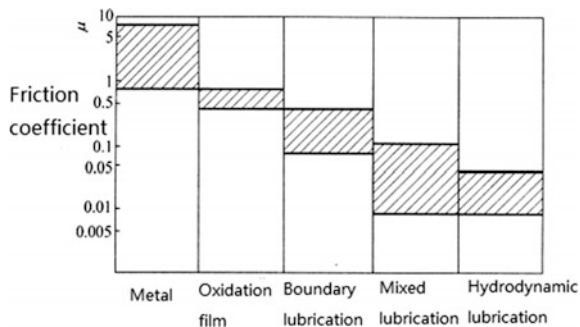


Fig. 2 Friction coefficient in various friction regions



2 The Theoretical Analysis and Experimental Study on Friction Characteristics

2.1 Theoretical Analysis

Because of the certain friction state of the clutch, the friction coefficient of the hydro-viscous clutch is analyzed by using the fluid mechanics and heat transfer. According to the working mechanism of the clutch, the torque of the transmission is

$$M = Kn2\pi\mu p \int_{R_1}^{R_2} R^2 dR = Kn\mu F \frac{2(R_2^3 - R_1^3)}{3(R_2^2 - R_1^2)} \quad (1)$$

$$M = Kn\mu F \frac{2(R_2^3 - R_1^3)}{3(R_2^2 - R_1^2)} \quad (2)$$

where the friction coefficient is calculated as following:

$$R = \frac{2(R_2^3 - R_1^3)}{3(R_2^2 - R_1^2)} \mu = \frac{M}{FRnK} \quad (3)$$

where μ is the friction coefficient of the hydro-viscous clutch plate, M is friction torque, F is positive pressure, n is number of friction pairs, K is the pressing force coefficient, take 0.9, R_2 and R_1 . Respectively, is the outer ring radius and the inner radius of the friction surface, R is the equivalent radius of the friction surface.

2.2 Test Method

According to the results of theoretical analysis, the friction coefficient can be calculated by the torque and positive pressure through the test. According to the Stribeck curve, the friction coefficient is related to the viscosity, the rotational speed, and the thickness of the oil film. Therefore, the load will be locked. Testing the friction coefficients of the different initial entry temperature and the same inlet temperature by adjust the speed difference.

2.3 Test Device

The test device of hydro-viscous clutch is shown in Fig. 3. Changing the input speed by DC motor and maintaining input speed in a constant during the test while locking the output shaft of the clutch to guarantee output speed is zero. Adjusting

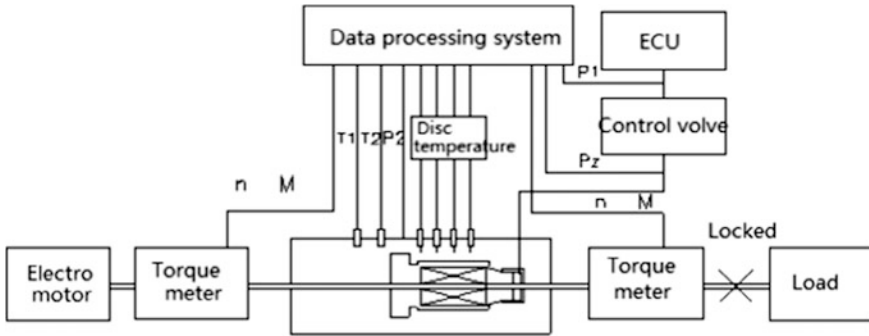


Fig. 3 Test bench sketch

control pressure the clutch until the motor's input torque to achieve the corresponding torque value.

2.4 Test Result

2.4.1 Effect on Friction Coefficient by Difference Speed

As shown in Fig. 4, the variation of friction coefficient with the difference speed under control oil pressure is 0.66 MPa and oil temperature is 90 °C. Assuming the effect of temperature is ignored, the friction coefficient curve is fitted at a given

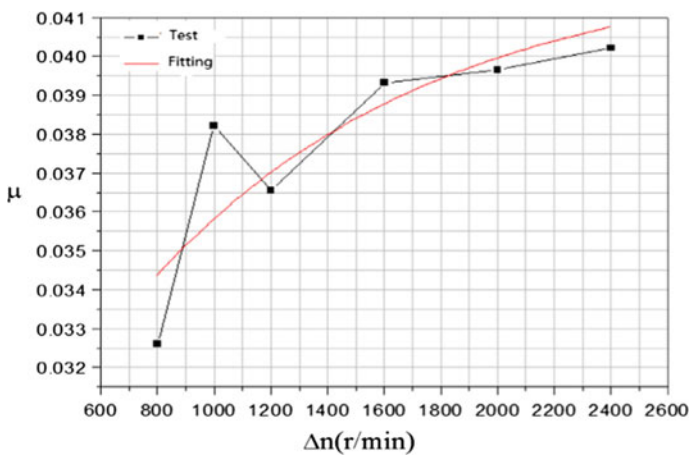


Fig. 4 The curve of friction coefficient under speed difference by experimental data and fitting

temperature. The red line in Fig. 4 is the friction coefficient fitting curve, where the fitting formula is

$$\mu = \mu_0 + ae^{\frac{\Delta u}{b}} \tag{4}$$

as seen from the fitting curve, the friction coefficient is a function of the difference speed, the formula is

$$\mu = \mu_0 + ae^{\frac{\Delta v}{cR}} \tag{5}$$

where $c = \frac{\pi b}{30}$, R is the equivalent radius of the friction surface.

Because of the relationship between the friction coefficient and friction materials, lubricants, and test temperature, the formula is not general only for the hydro-viscous clutch and oils in this test.

2.4.2 Effect on the Friction Coefficient by Positive Pressure

The positive pressure does not affect the friction coefficient. In fact, the friction gap is small, carrying capacity is relatively large and the friction coefficient is changing with positive pressure. As shown in Fig. 5, the changes of the friction coefficient are due to the changes of friction pair clearance, heat, and temperature caused by the positive pressure changes. Therefore, although positive pressure has nothing to do with the friction coefficient in theory, the actual friction coefficient is affected by positive pressure in certain extent.

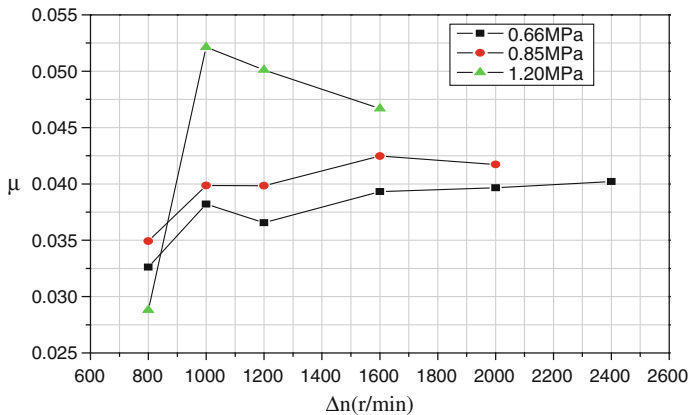


Fig. 5 The curve of friction coefficient under speed difference and different control oil pressures

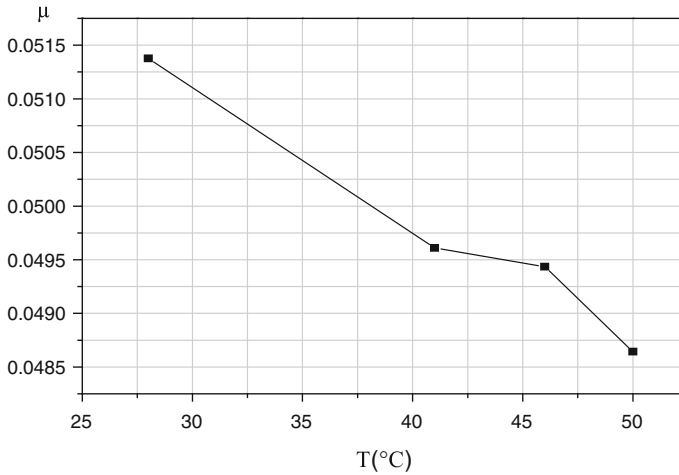


Fig. 6 The curve of friction coefficient under the same speed difference and different oil temperature

2.4.3 Effect on Friction Coefficient by Initial Oil Temperature

As shown in Fig. 6, it is the curve of the friction coefficient with temperature that varies at a given speed. It can be seen that the temperature is an important factor affecting the friction coefficient. Because the temperature in friction process will rise rapidly, it is difficult to eliminate friction effect of heat on the initial temperature in the test. The friction heat temperature rises reduce the effect of initial temperature on the friction coefficient to some extent. So the change of friction coefficient is very small under the initial temperature change form 20 to 50 °C. However, the trend of friction coefficient changing is reasonable, that is, when the temperature is higher, the viscosity, the shear resistance, and the friction coefficient are smaller.

3 Theoretical Analysis and Experimental Study on Thermal Load Characteristics

3.1 Theoretical Calculations Assume

1. Assuming that the friction power of clutch completely converted into heat;
2. Assuming that the heat in the operating process of hydro-viscous clutch is completely taken away by the flow of oil.

3.2 Frictional Power Calculation

According to the assumptions above, the friction energy of the clutch can be calculated:

$$W_f = \int_0^t N_H dt = \int_0^t M_1(\omega_1 - \omega_2) dt \tag{6}$$

The theoretical loss power of the clutch is

$$N_H = (1 - i) \cdot n_1 \cdot M_1 / 9549 \tag{7}$$

where i is the speed ratio, namely the ratio of the output speed and input speed, n_1 is the input speed, ω_1 and ω_2 is, respectively, the input angular velocity and output angular velocity, M_1 is the input torque, N_H is the loss power, W_f is the friction energy.

As shown in Fig. 7, assuming that all the loss power is converted to heat, the theoretical power consumption of the hydro-viscous clutch is calculated according to the load. The largest loss power is about 28 kW when the ratio is 0.7. The limited value should be covered in the friction characteristic test of hydro-viscous. The inertia of load and the power transfer gear pair can also cause the loss power. Therefore, the maximum test load is 35 kW, which is 1.25 times larger than the theoretical power of the clutch.

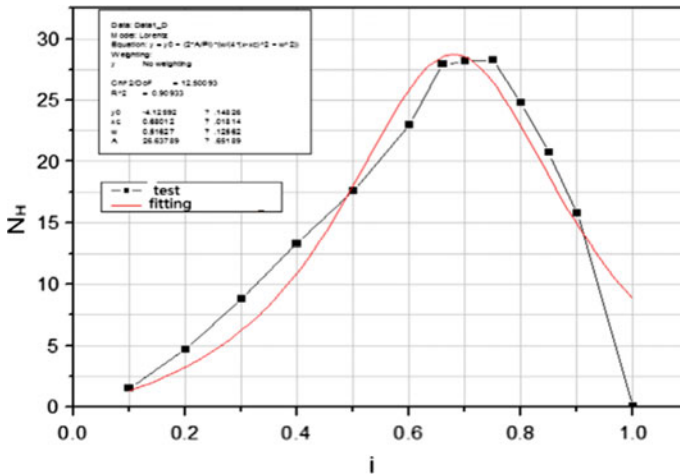


Fig. 7 Theoretical power loss of viscous clutch

3.3 Test Methods

According to the maximum thermal load of the calculation, the loss power of clutch is divided into four points, that is, 35, 20, 30, and 10 kW. Maintaining the friction work of clutch for some time at each point in the test, then getting the heat energy of hydro-viscous clutch through monitoring and measuring the temperature of the friction plate export, which is the thermal load characteristics.

3.4 Test Results

3.4.1 The Comparative Analysis Between the Calculation and Test Results

Assuming that all the heat is taken away by the oil, the temperature rise of the oil is based on the following formula,

$$\Delta T = \frac{N_H}{\rho \cdot Q \cdot C} \quad (8)$$

where ΔT is the average temperature of lubricating oil, °C, C is the specific heat of lubricating oil, 1161 J/kg °C, Q is the flow of lubricating oil, m³/s, ρ is the density of lubricating oil, 862 kg/m³.

As shown in Fig. 8, the temperature of lubricating oil is less than the calculated values. It is about half of the calculated value. It is indicated the heat generated in the clutch is not only taken away by lubricating oil, but also through the friction plate itself and the heat conduction of the box, as well as air heat convection and heat radiation in a variety of ways to take away a lot of heat.

3.4.2 The Test of Friction Plate Outlet Oil Temperature

The Fig. 9 shows the installation location of temperature sensor for the friction plate outlet. As shown in Fig. 10, though comparing the temperature rise of lubricating oil on the friction plate export, the heat is the largest where far from the piston of clutch. The temperature rises up to 45 °C on TO3 position. The following is the TO2 and TO1 position, the minimum heat is the friction pair which is the closest to the piston and the maximum temperature rise is less than or equal to 25 °C. This phenomenon reflects the real situation of friction heat. Due to large number of friction pair, the accumulation heat is more. Especially the heat of friction will be more concentrated in the central of axial. At the same time, the installation position of the oil temperature sensor has a certain influence on the accuracy of temperature measurement. The outer ring of the clutch outer hub will block the splashing of oil,

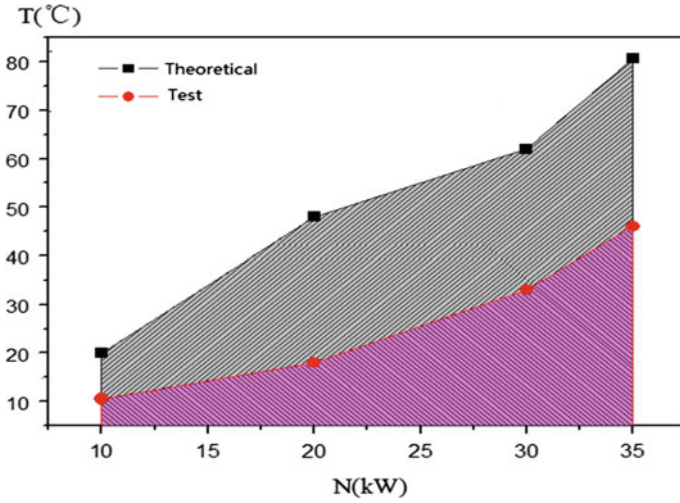


Fig. 8 Temperature rise by test and calculation

Fig. 9 Diagram for installation location of temperature sensor in friction plate outlet

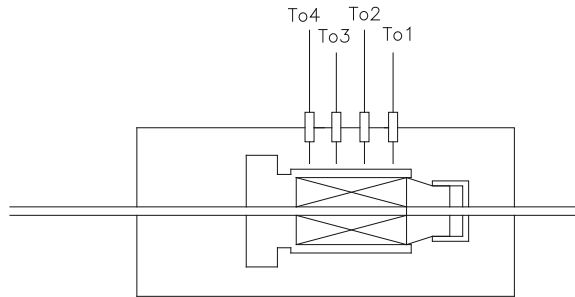
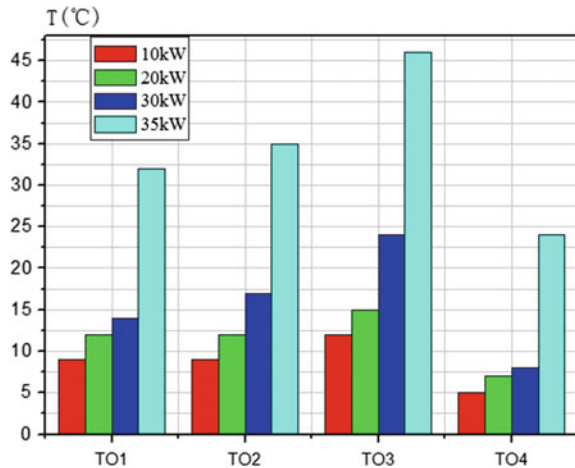


Fig. 10 Temperature rise of lubricant at thermal load of 10, 20, 30, and 35 kW



thus affecting the temperature of the oil temperature sensor. Therefore, during the structural design, it is necessary to change the position of the outer ring. At the same time, it is also necessary to increase the number and density distribution of lubrication holes in the axial center. To optimize the lubrication flow matching, it will reduce the number of lubrication hole in intermediate shaft at TO4 and TO1 which close to the ends of the friction plate.

4 Conclusions

In the paper, according to the friction principle, the friction state of the clutch and the range of the friction coefficient are estimated.

In the friction characteristic test of hydro-viscous clutch, it can be concluded the dynamic coefficient of friction of the clutch. The friction coefficient formula is fitted out at given temperature. It is a technical foundation for the further research in the future.

According to the test results, the friction coefficient of the clutch is not only affected by the speed difference, but also by the temperature and positive pressure. Due to the friction condition is more complex, the temperature, normal pressure, and difference rotating speed are related. It is difficult to obtain a formula to represent the friction coefficient.

Comparing the theoretical calculation and experimental results, it shows that the thermal load theory calculation reflects the temperature trend under different thermal loads of hydro-viscous clutch. But, it is not appropriate to calculate the heat convection only by a simple formula. It is necessary to introduce the coefficient or update the formulas.

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