Design and Simulation of Hydro-viscous Drive Experiment

Niu Dexue[1,a]

[1]Department of electromechanical engineering, Binzhou University, Binzhou, 256603, China

[a]niudexue9090@163.com

Keywords: hydro-viscous, drive, testbed, friction plate

Abstract. In order to study the hydro-viscous drive mechanism, on the basis of establishing the oil slick calculation model of hydro-viscous drive among friction pairs, this paper designs the hydro-viscous drive structure and makes a theoretical analysis of the oil slick transmitting torque through taking viscosity-temperature characteristics of operating oil into consideration. The experiment shows that as the incoming frequency is increasing, it needs shorter time to eliminate the gaps among friction pairs, the set pressure and output torque will increase in proportion. The flow of lubricating oil will make changes as the input frequency in the variable frequency motor changes. The increase of the oil slick temperature will decrease its viscosity. The transmitting torque of oil slick is in direct proportion to viscosity. The experimental value is basically consistent with the theoretical value with the torque changing.

Introduction

The hydro-viscous drive is a new fluid transmission form by taking the viscosity of the fluid or the shearing function of the oil slick as the transmitting drive, characterized by stepless speed, synchrodrive and small start shock. It can achieve the speed governing start, widely applied into ribbon conveyers, large-scale fans, water pumps and oil rigs and greatly reducing the energy consumption. The hydro-viscous drive is a controllable gearing with high performance and has a bright future.

Torque calculation of the hydro-viscous drive and analysis of the drive mechanism

Set the basic parameters: the driving and driven friction plate is revolving with the angular velocity of ω_1 and ω_2; the number of friction pairs is n; the gap of single friction pair is δ; the inner diameter and outer diameter of the disc oil silck with useful effect are r_1 and r_2 respectively.

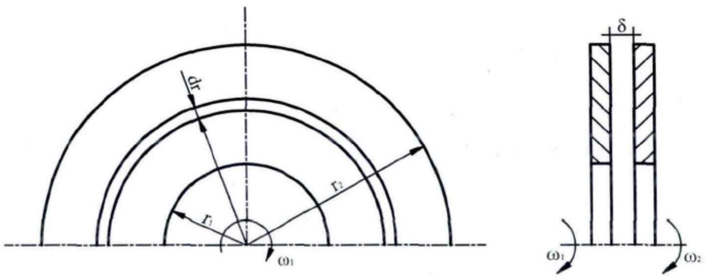


Figure 1 Torque Calculationb of the Dics Oil Slick

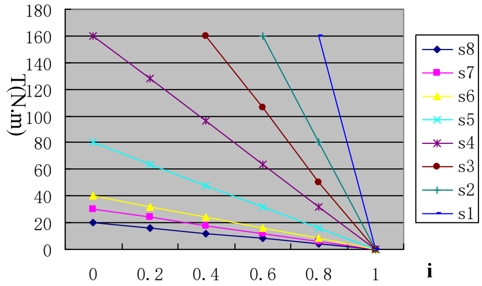
Get a small annulus size= $dA = 2\pi r dr$ at the place of the dics oil slick' diameter r, the shearing stress of the liquid transmission $T = u \cdot (\omega_1 - \omega_2) r / \delta$, and then the transmitting torqur $dT = rdF = r\tau dA$, that is :

$$dT = \frac{2\pi\mu(\omega_1 - \omega_2)}{\delta} r^3 dr \qquad (1)$$

The transmitting torque of the hydro-viscous speed governing clutch to friction pairs is:

$$T = \frac{n\pi\mu(\omega_1 - \omega_2)(r_2^4 - r_1^4)\backslash}{2\delta} \cdots \qquad (2)$$

The torque is in direct proportion to the angular velocity difference of the two friction plates $\Delta\omega = \omega_1 - \omega_2$. Usually, ω_1 and ω_2 are variables. Set that the ratio of the output speed and input speed of hydro-viscous drive clutch is I, and then $\Delta\omega = \omega_1 - \omega_2 = \omega_1(1-i)$. The relation curve of the tortue T and the speed ratio i with different oil slicks is shown in Figure 2.



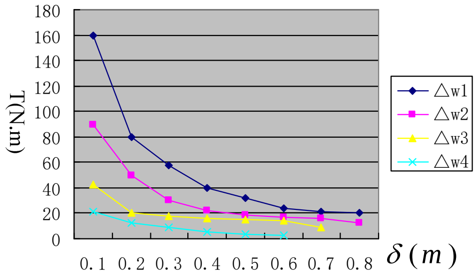


Figure 2 relation curve of the tortue T and the speed ratio i

Figure 3 relation curve of the tortue T and the oil slick δ

The torque is inversely proportional to the oil slick. The adjustment of the oil slick ω_1 can change the torque and speed. Figure 3 shows the relation curve of the tortue T and the oil slick δ with different angular velocity differences. Set a bigger value of the biggest oil slick and then produce a smaller value of the smallest output torque. It can be seen that the increase of the biggest oil slick will widen the speed governing range of the hydro-viscous devices, but will increase the axial dimensions.

In the oil slick shearing transimitting process, the starting torque will increase with the decrease of the oil slick gap, and will decrease with the decrease of the slip frequency. Therefore, in the starting process, reducing the oil slick gap will ensure a certain starting torque. However, the driving firction plates have radial and annular oil sink, the following expression can be got without taking the friction inner and outer splines into consideration and given that the oil slick thickness among different firction plates is the same:

$$M = \frac{n\mu(1-\psi)A\Delta\omega}{2(c+h)}(r_2^2 + r_1^2) + \frac{n\mu\psi A\Delta\omega}{2h}(r_2^2 + r_1^2) \qquad (2)$$

ψ is the contact coefficient of the friction plates, and the paper friction disc ψ is 0.5-0.7; c is the depth of the oil sink, usually within 0.8 mm.

When the liquid is the viscous friction, the surface of the friction plates are completely separated by continuous oil slicks and the gaps among friction plates are bigger. It is the case with a larger slip frequency. In this case, the shapes and material of the friction plates have fewer effects on the torque.

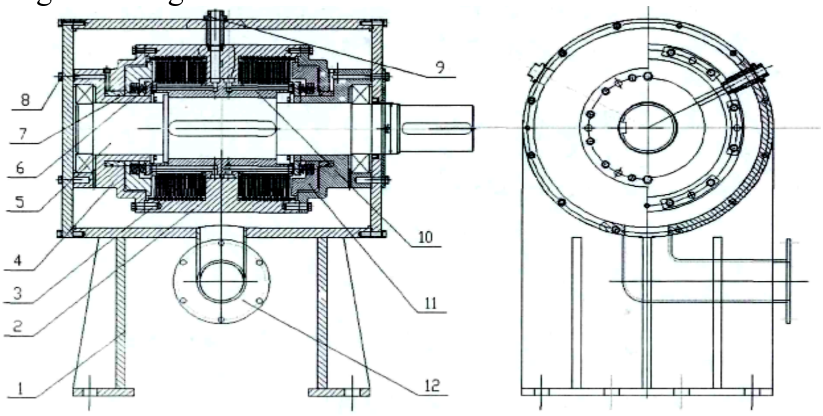
Set up the basic equation of three-dimensional fluid model of the friction pairs' oil sink, and establish simplified model of the friction pairs' oil sink, and then solve the equation of the pressure field by use of the software Anasys. Assume that the simulated calculation of the flow field is the flow of pure flow field, calculate the flow of the solid-fluid coupling among friction pairs, and ignore the influence of the single fluid unit and the pressure changes in the direction of slick thickness, and take the oil slick in the flow filed as axial symmetry. So the N-S equation can be simplified as follows:

$$\begin{cases} -\dfrac{1}{\varphi r}\dfrac{\partial p}{\theta} + \dfrac{\partial^2 V_0}{z^2} = 0 \\ -\dfrac{1}{\varphi}\dfrac{\partial p}{r} + \dfrac{\partial^2 V_r}{z^2} = \dfrac{\rho V_0^2}{\varphi r} \end{cases} \qquad (3)$$

Structure design of the hydro-viscous drive

Engine design of the hydro-viscous drive

According to the requirements for the engine functions of the hydro-viscous drive testbed, the testbed engine is designed as Figure 4.



1 — shell:2 — brake drum:3 — static friction plate group;4 — cylinder:5 — ratation axis;
6 — dynamical friction plate group;
7 — return spring:8 — oil filler;9 — oil-leading port;10 — driving gear:11 — piston;12 — oil return flange

Figure 4 Structure of the testbed's engine

Thestructure of the testbed's engine is similar to that of the hydro-viscous drive speed governing clutch, mainly including driving part, driven part, execution part of controlling the thickness of the oil slick, the sealing and lubricating part, support part and monitoring part. The working conditions of the engine are divided into three types: ①when the control oil pressure is 0, the pistons will be pressed back with the effect of return spring. The gap between driving and driven friction plates is too big, so the shearing force od the oil slick cannot make the driven shaft revolve. At this moment, the engine stay off-line, which mainly happens at the preparation stage before working or in the working process with the motor not stoping and the output stopping. ②when the control cylinder has some control oil pressure, the piston will push the friction plate to change the thickness of the oil slick. With the shearing force of the oil slick, the output shaft will get a certain speed and torque. Now the engine is at the working status of speed governing. ③ when the control oil pressure is up to some value, and the force of return spring is offset, the piston will completely push the firction plates tightly and the oil slick among the firction plates will be squeezed out. Denpending on the static friction among the friction plates transmits the torque. The output and input shafts will have the same speed. Now, the engine stays in a synchrodrive status.

Design of the hydraulic control system and lubrication system

The hydraulic control system of the hydro-viscous drive provides the controlled oil pressure for the engine and controls the thickness of the oil slick among the friction plates. The lubrication system offers the operating oil for the engine, forms the dynamic shearing drive oil slick and lubricates and cool the firction plates. According the experiment's functional requirements, the schematic diagram of the hydraulic control system in Figure 5 is designed.

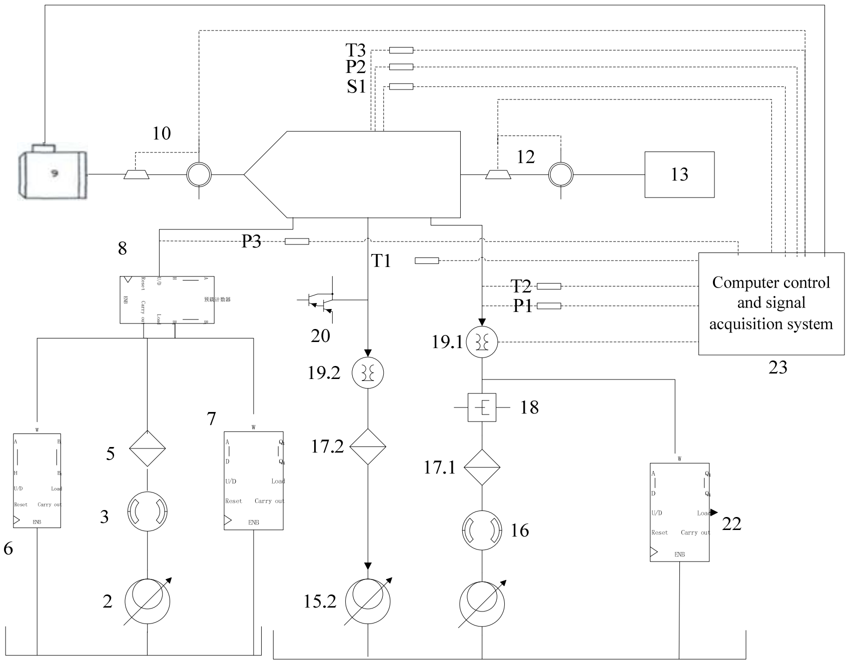


Figure 5 Schematic diagram of the hydraulic control system of the hydro-viscous drive

In order to study the transmission of different hydro-viscous working medium and the heat transfer property, the hydraulic control system and lubrication system adopt the way of individual oil supply. Compared with the working medium of the hydraulic control system, the lubrication system has higher requirements for the function of the working medium. The hydraulic lubrication system adopts independent oil tank for supply for the convinience of changing the operation fluid and studying the performences of different hydro-viscous working liquid. The calculation shows that the proper type of the lubrication oil can be NB-C50F with the output volume of 50ml/r and nominal pressure of 6.3 MPa. The drive motor can choose the type of YT132M with 10 KW.

Design of the control system

As shown in Figure 6, when using the PID controller, we don't need the specific mathematic model of controlled objects. In the pressure system of the control devices, the changes of the oil temperature and the belt elastic elongation will impact the establishment of the transmitting functions, and it is almost impossible to get accurate transmitting functions. PID will directly make adjustments. In the control loop of CST, it needs the collaboration of multi PID. Because PID has high flexibility, it just needs to adjust the diameters on the basis of the requirements to change the control ways and realize those complex control, such as the multichannel control and cross-channel control.

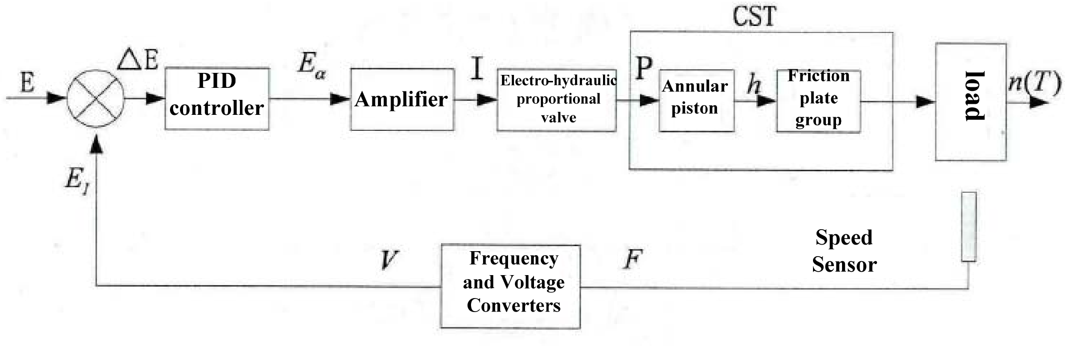
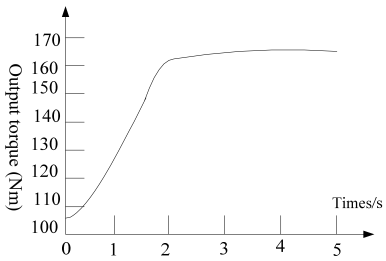


Figure 6 PID control chart

Analysis of the experimental results

This paper uses the hydro-viscous drive with the type of YNRQD 350/1500, and the power source is three-phase asynchronous motor with the type of YZ-355M2-4. The reducer is the parallel shaft spur gear reducer with the type of ZDY315-2.24. the type of the torque speed sensor is NJ3 一 I0000N·m. The power supply frequencies are 30Hz, 35Hz,45Hz and 50Hz. The length of the oil duct is 0.174m. The lubricant pump adopts YB 一 E193, the control pump adopts the CBWL 一 E32O/E3OS parameters of asynchronous motors. The measured motor is 75Kw. The flow of lubricating oil is 300L/min. Figure 7 ,8, 9 and 10show the torque with different powers.



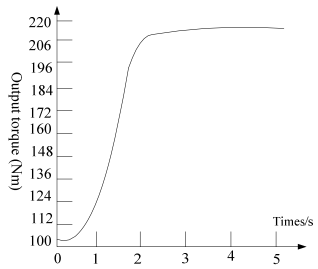
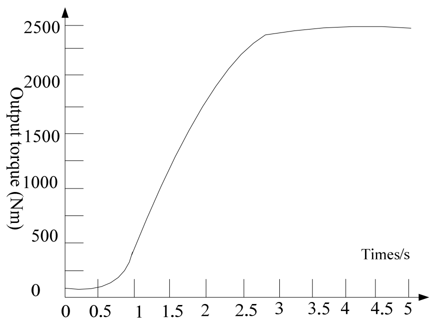


Figure7 Output torque curve with 30HZ   Figure 8 Output torque curve with 35HZ



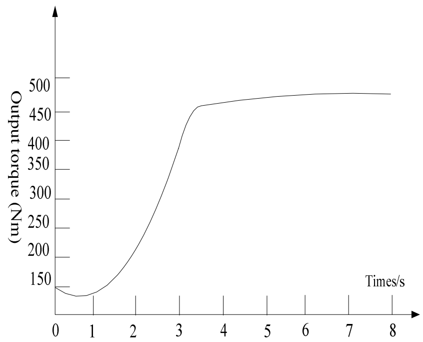


Figure9 Output torque curve with 45HZ   Figure 10 Output torque curve with 50HZ

The dynamometer will produce different torques with different frequencies. Studies are conducted with the different frequencies of 35 Hz, 50 Hz and 30Hz. As the input frequency increases, the set pressure and output torque will increase proportionally. It can be seen from the figures that the increase of the input frequency wil reduce the time of eliminating the gaps among friction pairs.

With fixed flow of operating oil, speed difference and oil slick thickness, it discusses the influence of temperature on the transmitting torque. Keep the fixed oil slick thickness, set the speed difference as 1000r/min, and discuss the change of the transmitting torque with time when the operating oil changes from 20°C to 50°C. From Figure 11, it can be seen that generally speaking, the transmitting torque is reducing with the time increasing. The reason is that the increase of the oil slick's temperature will lead to the decrease of the oil slick's viscosity, while the transmitting torque is in direct proportion to the viscosity. The experimental value of the torque change with time is basically consistent with the theoretical value.

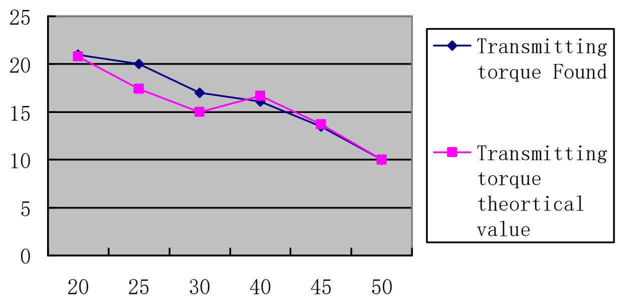


Figure 11 Relationships between Temperature and the Oil Slick Transmitting Torque

Conclusion

On the basis of establishing the oil slick calculation model of hydro-viscous drive among friction pairs, the expression of oil slick transmitting torque is got. The results show that when the oil slick thickness is fixed, the oil slick transmitting torque is in direct proportion to the operation oil viscosity and speed difference. Through taking the viscosity of operation oil into account, get the change rule of the oil slick transmitting torque with time and do the numerical simulation of oil slick among interfaces. The result shows that the oil transmitting torque is bigger in the conditions of constant temperature than that in the varying temperature. Set up the hydro-viscous drive testbed and get the results of the relationship between the control current and control oil pressure in the control system of hydro-viscous drive, the relationship between the output revolving speed and output torque with a fixed control current, and the relationship between the control oil pressure and output torque with a fixed revolving speed. This paper illustrates the application of hydro-viscous drive into the centralized-control system with a fixed torque, but it needs further studies on the application of hydro-viscous drive into various operating conditions of standardized and serialized centralized-control system.

References

[1] Mohammad S，Vahid H, Iraj M．Computer aided analysis of flow in a rotatingsingle disk.World Academy of Science Engineering and Technology, Vol 58，2009,pp 160-163

[2] Aleksander Yevtushenko，Michal Kuciej. Temperature and thermal stresses in a pad/disc during braking. Applied Thermal Engineering, Vol （5），2010，p.p 233-236

[3] Zhang Lijun, Si Yang, and Yu Zhuoping. Studies on the Coupling Characteristics of Inhomogeneous Disc Brake Heat Engine. Automobile Technology, Vol （6），2008, pp45-49.

[4] Hao Hongtao & Ma Yue. Studies on the System Identification of Hydro-viscous Dynamometer. Transaction of Beijing Institute of Technology,Vol 25(1),2005,pp9-12

[5] Gu Zhaojun, Yang Qianming, and Xu Liang. Comparative Analysis of the Advantages and Disadvantages of Hydro-viscous Speed Regulating Clutch and Hydraulic Coupler Speed Regulation . Modern Manufacturing Technology and Equipment, Vol 6,2006, pp28-31

[6] Jiang Jianyong. Studies on the Experimental System of Hydro-viscous Speed Regulating Clutch. Hangzhou, Zhejiang University, 2006, pp 59-64.

[7] Huang Gui, Chen Fei and Li Xiaoya. Friction Vice Transmitting Torque Calculation of Hydro-viscous Speed Regulating Clutch and Heat Balance Check .Machine Tool & Hydraulics, Vol 37（3），2009, pp83-89

[8] Wei Aiguan & Zhao Jiaxiang. Hydro-viscous Drive Technology. Beijing: National Defense Industry Press, 19-21

[9] Jiang Jianyong. Studies on the Experimental System of Hydro-viscous Speed Regulating Clutch. Hangzhou, Zhejiang University,2006, 13-15

[10] Sun Qin, Yang Xinjun and Yang Qianming. Experiment and Studies on the Hydro-viscous Speed Regulating System. Chinese Hydraulics & Pneumatics, Vol 12,2006,pp22-24.