Study on Dynamic Output Characteristics of Liquid - viscous Speed Regulating Clutch and Simulation Analysis of Dil Film Flow Field Between Friction Plates Based on Computer Simulation Technology

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Abstract. To improve the drive torque without increasing the installed power of the system, a drive system of “motor + fly wheel + hydroviscous variable speed clutch” is designed. The mathematical model for each energy transmission link of the drive system, the oil film bearing capacity model and the Computer simulation model for the drive system are established to reveal the influence law of the factors such as the fly wheel moment of inertia (MOI) and oil film thickness control curve on the dynamic engagement features of hydroviscous variable speed clutch, obtain the change curve of torque, speed and impact, and build the corresponding experimental bench. The results of the simulation experiment suggest that through reasonable control of the oil film thickness of the hydroviscous variable speed clutch, the output of twice the rated torque that lasts up to 50 s can be achieved to meet the engineering demand of large and medium-sized mechanical equipment for large starting torque and small impact.

Keywords: Hydroviscous Variable Speed Clutch, Oil Film Flow Field between Friction Plates, Dynamic Engagement Features, Computer Simulation Technology

1. Introduction

In general, the starting torque of large construction machinery is higher than its rated torque. In the design\cite{1,2}, it is often necessary to increase the installed power of the system to meet the starting demand, where soft start technology is required to avoid the impact of starting, HVC) has been successfully applied in this field due to its flexible transmission features. Since the hydroviscous variable speed clutch is mainly used in various large-scale fans, pumps and other occasions for speed regulation and energy saving, and the requirements for the dynamic features of the hydroviscous variable speed clutch in these occasions are not high, researchers focus mostly on its steady-state features to carry out studies\cite{3,4}. Studies on the dynamic engagement features of the clutch mainly focus on the wet clutch. The structure of the hydroviscous variable speed clutch is similar to that of the wet clutch, and the research on the dynamic engagement performance of the wet clutch has important reference significance for the research on the hydroviscous variable speed clutch, but there is a great
difference between the two. The engagement process of the wet clutch is almost instantaneous, and it is challenging to control the dynamic process of engagement effectively. The process is accompanied by a large amount of sliding friction work, and the direct friction heating between the metals is easy to cause the friction plate to fail [5-6]. The dynamic engagement process of the hydroviscous variable speed clutch is longer, because there is a buffer transition between the active and passive friction plates, the speed of the passive friction plate can be accelerated from zero to synchronous speed, There is no direct friction and wear between metals, and the heat generated is also taken away by a special lubricating oil circulation system, thereby avoiding the impact brought about by the startup process effectively. The dynamic torque can also be adjusted in real time through the control of oil film thickness to control the dynamic process of speed rise. Hence, the hydroviscous variable speed clutch is more applicable to the soft start application of large and medium-sized power equipment.

In this paper, a drive scheme of “motor + inertia fly wheel + hydroviscous variable speed clutch” is designed, which aims to achieve large torque output and reduce impact on the system by using mechanical energy storage function of fly wheel and reasonable control of hydroviscous variable speed clutch without increasing the installed power of the system. To verify the effectiveness of the scheme, AMESim is used to establish a simulation model of the scheme, On this basis, the influence of fly wheel inertia and clutch oil film thickness control on dynamic engagement process is studied and verified by experiments.

2. Mathematical model for the energy transmission link of the drive system

From the perspective of energy transmission, when the HVC friction plate is in the condition of complete separation, the prime mover with the fly wheel accelerates to store energy; when the HVC is in the condition of oil film shearing, the fly wheel end is subject to the resistance of the HVC link to slow down and release energy; according to the principle of force and reaction force, the load end is accelerated from zero under the dynamic action of the HVC link at this time; When the speed of HVC active end drops to synchronous speed, that is, when the speed of active end and passive end is equal, the system is in synchronous operation condition. It is assumed that JA is the moment of inertia of fly wheel, and all the slip loss of HVC is converted into heat. Based on the law of energy conservation, the following can be obtained.

\[
\int T_a \omega_a dt + \frac{1}{2} J_a \omega_a^2 - \frac{1}{2} J_s \omega_s^2 + \frac{1}{2} J_e \omega_e^2 + \int T_f \omega_b dt + \int T_c (\omega_b - \omega_h) dt
\]

(1)

When the clutch is fully engaged, the fly wheel, clutch and load can be regarded as an inertia body.

\[
(J_a + J_b) \dot{\omega}_h = T_c - T_f
\]

(2)

The above analysis process suggests that the fly wheel moment of inertia and HVC control are the key factors affecting the starting features of the drive system, the fly wheel moment of inertia affects the energy release process of the fly wheel, while the HVC oil film thickness control affects the HVC engagement and torque transmission process.

In this scheme, the supporting load simulation motor is a dynamometer used to achieve the reliable load torque at zero speed. At a target speed of zero, it works in the torque control mode. The corresponding drive motor needs to work in the speed control mode, that is, to set an upper limit value for the torque of the drive motor, when the load torque is less than the upper limit value of the drive motor torque. The output torque of the drive motor is equal to the load torque. while, the drive motor always outputs the upper limit value of the torque. Hence, the following motor output features are established.

\[
T_a = \begin{cases} 
T_{max} & (T_e < T_f) \\
T_f & (T_e \geq T_f) 
\end{cases}
\]

(3)
Where $T_s$ represents motor output torque, $n \cdot M$.

$T_{max}$ represents upper limit value of set torque, $n \cdot M$.

$T_f$ represents load torque, $n \cdot M$.

The electro-hydraulic proportional relief valve is generally used to regulate the pressure of the hydroviscous variable speed clutch, so as to control the piston displacement, change the oil film thickness and adjust the rotating speed torque. The response time of the electro-hydraulic proportional relief valve selected in this system is about 10ms, which is at least three orders of magnitude different from the system response time, so the input voltage and output pressure of the relief valve can be equivalent to the first order inertia link. Hence, its transfer function can be obtained as follows:

$$
\Delta p_f = \frac{K_r \Delta u}{Ts + 1}
$$

(4)

Where $K_r$ represents the proportion coefficient.

$\tau$ represents time constant.

$\Delta u$ represents input voltage.

When the torque of the drive motor is higher than the load starting torque, the load starting torque (mainly static friction torque) is still maintained for a certain period, then reduced to normal torque (can be regarded as dynamic friction torque), and rises with the increase of speed. Hence, the simulation load is close to the actual situation. Although the theoretical model established is not sufficiently sophisticated, the research approach has reference value.

3. Simulation analysis of system dynamics

AMESim has been extensively used in the dynamic performance simulation of the hydraulic system. However, the hydroviscous variable speed clutch model is not provided in AMESim. There is only a simple model of multi-plate wet clutch. The biggest difference between hydroviscous variable speed clutch and wet clutch is that the former has oil film shearing condition, i.e. it maintains a certain thickness of oil film when oil film bearing capacity and piston force are balanced, and it depends on this oil film to transmit torque. This condition is very important for soft start, Because if there is no controllable and buffered transition of oil film, the sudden engagement of active and passive friction plates will bring great impact to the system. Therefore, it is necessary to reconstruct the simulation model of the hydroviscous variable speed clutch according to the oil film bearing capacity model and the wet clutch model of AMESim itself. The mathematical formula of the bearing capacity of each oil film is directly given.

Some critical parameters in the model are shown in Table 1. Set the initial speed of the fly wheel as 800r / min, and also set the moment of inertia of the fly wheel as 5, 10, 15, 20kg \cdot M2 by using the curve of oil pressure and oil film thickness change shown in Figure 1, to get the HVC transmission torque. The figure shows that under the same oil film thickness control curve, the greater the moment of inertia of the fly wheel, the faster the HVC torque rises, and the higher the peak torque, The slower the fly wheel speed drops, the faster the corresponding load end speed rises, the greater the synchronous speed, and the greater the impact on the system. When the fly wheel moment of inertia is 5kg \cdot m2, the maximum impact is 59M / S3, which still exceeds the standard (in the automotive field, the impact recommended by Germany is 10m / S3, and the impact recommended by China is 17.64m / S3). Therefore, when the HVC torque meets the starting demand, The moment of inertia of the fly wheel should be reduced to the minimum.
### Table 1. Key simulation parameters

| Parameter                                                                 | Numerical value |
|---------------------------------------------------------------------------|-----------------|
| Maximum output torque of the motor (n · m)                                | 100             |
| Number of friction discs                                                 | 8               |
| Outer diameter of the friction plate / mm                                  | 314             |
| Inner diameter of the friction plate / mm                                  | 225             |
| Initial total friction plate clearance / mm                               | 3               |
| Piston mass / kg                                                          | 16              |
| Clutch spring stiffness / (n · m⁻¹)                                       | 200000          |
| Load moment of inertia / (kg · m²)                                        | 200             |
| Torque component under normal load operation / (n · m)                    | 50              |
| Viscosity friction coefficient / (n · m · s · rad⁻¹)                      | 0.1             |
| Load starting torque / (n · m)                                            | 200             |

#### Figure 1. Change curve of the oil pressure and oil film thickness

To further verify the simulation results, Experimental research has been carried out. The experimental platform is mainly composed of drive device, hydroviscous variable speed clutch, analog load device, speed and torque sensor, fly wheel, supporting hydraulic system and electric control system. The pci1723 and LabVIEW program of Advantech realize the collection and storage of various pressure, torque, speed and other signal data in the experiment, including drive motor, hydroviscous variable speed clutch, load. The parameters such as fly wheel inertia and oil pressure control are completely consistent with the simulation settings. The HVC joint characteristic curves under different moment of inertia, 1 series of oil film control laws and 2 series of oil film control laws are respectively obtained as shown in Figure 2. The results show that the change trend of the experimental curve is consistent with the simulation results, with good coincidence, This shows that the drive system of “motor + fly wheel + HVC” makes full use of the kinetic energy of the fly wheel and the features that the hydroviscous variable speed clutch can control the gradual release of energy, improves the energy utilization rate, and has practical engineering application significance.
Figure 2. Torque curve of HVC under the different moments of inertia

4. Conclusion
In this paper, a drive scheme of “motor + fly wheel + hydroviscous variable speed clutch” is designed, where the mechanical energy storage function of fly wheel can be used with the reasonable control of hydroviscous variable speed clutch without increasing the installed power of the system to implement large torque startup while avoiding the impact during startup effectively at the same time. The established simulation model of the drive system can be used to analyze the influence of fly wheel inertia and oil film thickness control curve on the starting features of the system effectively. The larger the fly wheel inertia, the faster the decrease of oil film thickness, the faster the rise of the starting torque, the higher the peak value, and the greater the impact. Compared with the speed control law for single oil film thickness decrease, the speed control law for variable oil film thickness decrease can produce sustained torque more easily, which is conducive to implementing load startup. The drive system can achieve the output capacity twice the load torque for a duration up to 50 s by controlling the thickness of the variable oil film of the hydroviscous variable speed clutch at a maximum impact of 15.7 m/s³ only.

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