Velocity analysis of oil film between friction pairs in hydro-viscous drive clutch

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Abstract. A mathematical model describing the flow field and velocity distribution between rotational friction pairs has been proposed based on viscosity-temperature properties and centrifugal force of working oil. The effect of inlet pressure, rotation speed and inertia on the flow are theoretically analysed. It is found that inlet pressure has a linear effect on the radial direction and the flow field is the Poiseuille flow. However, inertia makes the flow more complex to conserve the continuity. With the effect of inertia, the velocity is negative at entrance and positive at exit. Relative to inertia and inlet pressure, the rotate speed is supreme dominate the flow field. And also, the variety of viscosity has a great effect on the radial velocity. The proposed model may provide technical support for the design of Hydro-viscous drive systems.

1. Introduction
Hydro-Viscous Drive (HVD) clutch contains lots of friction pairs which is filled with working oil. It can transmit power by the sheer force of working oil as friction pairs rotate and realize step-less speed change [1, 2] by adjusting the distance between friction pairs. And is widely used in energy transmission equipment, especially for belt conveyer.

Over the past years, researchers have conducted enormous studies on mechanism system of HVD. Jianzhong Cui [3, 4] et al. simulated two types of warping deformation and obtained the temperature distributions of multi-disk friction pairs with a numerical investigation. Kimura [5] and Gao [6, 7] took surface roughness into account in order to investigate the engagement process of HVD clutch. Zhao [8] and Aphale [9] established a 3D computational fluid dynamics model to simulate the pressure and torque distribution of the friction pairs. Jen [10] and Meng [11] studied speed regulating start in engagement process and found oil film squeezing had a significant effect for system operation. Marklund [12] obtained a torque transfer model of a wet clutch working in boundary lubrication regime. Xie Fangwei [13, 14] developed a simulated pressure and temperature distribution in radial and circumferential direction and obtained influence of grooves on temperature and pressure distribution. Wu Wei [15] and Huang JiaHai [16, 17] studied and analyzed the influence of radial grooves on pressure, load carrying capacity and torque transmission. In a word, their research shows that the flow field of oil film varies significantly during the high-speed operation.

Although great progress is made in the experimental and simulation study of the flow field, it is a lack of analytic solution of the simplified N-S equations. In addition, the viscosity of working oil was
generally treated as a constant in the previous research, generating a big discrepancy between simulation and experimental results. Therefore, the aim of this paper is to theoretically analyze the flow field of oil film by obtaining the analytic solution of the simplified N-S equations and develop a model based on viscosity-temperature formula by Taylor expansion.

2. Mathematical model

Due to the influence of pressure difference and centrifugal force, the oil film moves in a spiral form when the system is set in motion as shown in Fig.1. It is assumed that the oil film is an incompressible Newton fluid and the flow is laminar and steady, hence \( \partial / \partial t = 0 \) and the dynamic viscosity \( \mu \) is unrelated to the pressure; the thickness of oil film is far less than radial distance \( \delta / r = 0 \), hence axial velocity \( w=0 \).

\[
\begin{align*}
\rho \left( \frac{u^2}{r} + \frac{v^2}{r} + r \omega^2 \right) &= \frac{\partial p}{\partial r} - \mu \frac{\partial^2 u}{\partial z^2} \\
\rho \left( \frac{\partial v}{\partial r} + \frac{uv}{r} \right) &= \mu \frac{\partial^2 v}{\partial z^2} \\
\frac{\partial p}{\partial z} &= 0 \\
\frac{\partial u}{\partial r} + \frac{u}{r} &= 0
\end{align*}
\]

Fig 1. Model of friction pairs.

Then the simplified N-S equations and continuity equation can be expressed as:
And the boundary conditions can be described as
\[
\begin{cases}
  z = 0 & u = 0, \\
  z = \delta & u = 0,
\end{cases}
\]
\[v = r\omega_2,\]
where \(\omega = \frac{\omega_1 - \omega_2}{\delta} z + \omega_2,\) \(u\) is radial velocity, \(v\) is circumferential velocity, \(\omega_1\) and \(\omega_2\) are the angular velocity of friction dicks and separator disks, respectively.

As the momentum equations are non-homogeneous, it is difficult to solve them directly. By the constant variation method, the momentum equations be solved in two steps: the first step is to transform the original equations to homogeneous.

\[\frac{\partial p}{\partial r} = \mu \frac{\partial^2 u_1}{\partial z^2},\] (5)

\[\mu \frac{\partial^2 v_1}{\partial z^2} = 0,\] (6)

then substituting the solutions from first step into original equations.

Integrating Eq.5 and Eq.6 along \(z\) direction with the boundary conditions, \(u_1\) and \(v_1\) can be written as:

\[u_1 = \frac{z(z - \delta)}{2\mu} \frac{\partial p}{\partial r},\] (7)

\[v_1 = \frac{r(\omega_1 - \omega_2)}{\delta} z + r\omega_2,\] (8)

Then radial flow can be written as:

\[Q_1 = 2\pi r \int_0^\delta u_1 dz = -\frac{\pi r \delta^3}{6\mu} \frac{\partial p}{\partial r},\] (9)

Integrating Eq.5 with \(p_{r_1} = p_1,\) \(p_{r_2} = 0\) yields:

\[u_1 = -\frac{z(z - \delta)p_1}{2\mu r \ln \frac{r_2}{r_1}}\] (10)

Substituting Eq.7 and Eq.10 into Eq.1 and Eq.2, the momentum equations can be written as
\[ \mu \frac{d^2 u}{dz^2} = \frac{\partial p}{\partial r} - \frac{\rho p_1^2 (z-\delta)^2 z^2}{4 \mu^2 r^3 (\ln \frac{r_2}{r_1})^2} - 2 \frac{\rho r}{r_1} (\frac{\omega_1 - \omega_2}{\delta}) z + \omega_2 \]  

(11)

\[ \mu \frac{d^2 v}{dz^2} = \frac{\rho(\delta - z) z p_1}{\mu r \ln \frac{r^2}{r_1}} \left[ (\frac{\omega_1 - \omega_2}{\delta}) z + \omega_2 \right] \frac{\partial p}{\partial r} \]  

(12)

Integrating Eq.11 and Eq.12, yields the velocity:

\[ u = \frac{z(z-\delta) \frac{\partial p}{\partial r}}{2 \mu} - \frac{\rho p_1}{240 \mu^3 r^3 (\ln \frac{r_2}{r_1})^2} \left[ 2 z^6 - 6 \delta z^5 + 5 \delta^2 z^4 - \delta^5 z \right] 
- \frac{\rho r}{20} \left[ (\omega_1 - \omega_2)^2 z^4 + 4(\omega_1 - \omega_2) \omega_2 \delta z^3 + 6 \omega_2^2 \delta^2 z^2 - (\omega_1^2 + 2 \omega_1 \omega_2 + 3 \omega_2^2) \delta^3 z \right] \frac{\partial p}{\partial r} \]  

(13)

\[ v = -\frac{\rho p_1}{\mu^2 r \ln \frac{r^2}{r_1}} \left[ \frac{\omega_1 - \omega_2}{20} z^5 + \frac{2 \omega_1 - \omega_2}{12} z^4 - \frac{\omega_2 \delta}{6} z^3 + \frac{2 \omega_1 + 3 \omega_2}{60} \delta^3 z \right] 
+ \frac{r(\omega_1 - \omega_2)}{\delta} z + r \omega_2 \]  

(14)

Then radial flow becomes

\[ Q = -\frac{\pi r \delta^3}{6 \mu} \frac{\partial p}{\partial r} + \frac{\pi \rho p_1^2 \delta^3}{560 \mu^3 r^2 (\ln \frac{r_2}{r_1})^2} + \frac{\pi \rho r^2 \delta^3}{30 \mu} (3 \omega_1^2 + 4 \omega_1 \omega_2 + 3 \omega_2^2) \]  

(15)

Integrating Eq.15 with \( p_{r_1} = p_1, p_{r_2} = 0 \) yields:

\[ Q = \frac{\pi \delta^3 p_{r_1}}{6 \mu \ln \frac{r_2}{r_1}} + \frac{\pi \rho p_1^2 \delta^3}{1120 \mu^3 (\ln \frac{r_2}{r_1})^3} \left( \frac{1}{r_1^3} - \frac{1}{r_2^3} \right) - \frac{\pi \rho \delta^3 (r_1^2 - r_2^2)}{60 \mu \ln \frac{r_2}{r_1}} (3 \omega_1^2 + 4 \omega_1 \omega_2 + 3 \omega_2^2) \]  

(16)

Combining Eq.15 and Eq.16, radial pressure gradient can be written as
\[
\frac{\partial p}{\partial r} = -\frac{p_i}{r \ln \frac{r_2}{r_1}} + \frac{3 \rho p_i^3 \delta^4}{560 \mu^2 r (\ln \frac{r_2}{r_1})^2} \left[ \frac{2}{r^2} - \frac{1}{\ln \frac{r_2}{r_1}} \right] (\frac{1}{r_2} - \frac{1}{r_1}) \\
+ \frac{\rho}{10r} (2r^2 + \frac{r_1^2 - r_2^2}{\ln \frac{r_2}{r_1}})(3\omega_1^2 + 4\omega_1\omega_2 + 3\omega_2^2)
\]

(17)

Then the radial velocity and pressure can be expressed as

\[
u = \frac{z(\delta - z) p_i}{2 \mu r \ln \frac{r_2}{r_1}} + \frac{3 \rho p_i^3 \delta^4 z(\delta - z)}{1120 \mu^3 r (\ln \frac{r_2}{r_1})^2} \left[ \frac{2}{r^2} - \frac{1}{\ln \frac{r_2}{r_1}} \right] (\frac{1}{r_2} - \frac{1}{r_1}) \\
- \frac{\rho}{4 \mu^3} (2z^6 - 6\delta z^5 + 5\delta^2 z^4 - \delta^3 z)
\]

\[
\frac{4 \mu^3 r^3 (\ln \frac{r_2}{r_1})^2}{60} (\frac{2}{r^2} + \frac{r_1^2 - r_2^2}{\ln \frac{r_2}{r_1}})(3\omega_1^2 + 4\omega_1\omega_2 + 3\omega_2^2)
\]

(18)

According to Eq.18, three factors affect the radial velocity of oil film: inlet pressure, rotation speed and inertia. Their formulas are shown as follows:

\[
u_{\text{inlet}} = \frac{z(\delta - z) p_i}{2 \mu r \ln \frac{r_2}{r_1}}
\]

(19)

\[
u_{\text{rotate}} = \frac{\rho r^2 z}{6 \delta^2 \mu} [(\omega_1 - \omega_2)^2 z^3 + 4(\omega_1 - \omega_2) \omega_2 \delta z^2 + 6\omega_2^2 \delta^2 z - (\delta^2 + 2 \omega_1 \omega_2 + 3 \omega_2^2)\delta^3]
\]

\[
u_{\text{inertia}} = \nu - \nu_{\text{inlet}} - \nu_{\text{rotate}}
\]

(20)

(21)
According to Xie [18] considering the influence of temperature, the viscosity can be written as

\[ \mu = 0.2007e^{-10.98r} \]  

3. Result analysis

3.1. Effect of inlet pressure

Eq.19 shows the inlet pressure has a linear effect on radial velocity of oil film. When working oil flows into the space between friction pairs, it is a Poiseuille flow due to the viscosity of oil film.

3.2. Effect of rotate speed

Because of the viscosity, working oil rotates with friction pairs. As friction disks rotate faster than separate disks, the position with the maximum velocity shifts to friction disks in axial direction. Fig.4 shows the velocity distribution between the friction pairs. In radial direction, the velocity is linearly increasing. In axial direction the velocity is asymmetric. While in Fig.5 it can be seen that as speed of friction disks increase from 1000rpm to 3000rpm, the position with the maximum velocity does not change.
3.3. Effect of inertia

Fig. 6 shows that the shape of velocity distribution influenced by the inertia is point symmetry. In radial direction, the velocity is linearly increasing, being negative at entrance and positive at exit. It indicates that the inertia hinders working oil flowing in the inner region and promotes in the outer region to ensure the continuity of the flow. Eq. 21 shows that the inertia consists of inlet pressure and rotation speed. From Fig. 7 and Fig. 8, it can be seen that the effect of inlet pressure is much smaller than that of the rotation speed on radial velocity with a magnitude of $10^{-10} \text{ m/s}$. In axial direction, the shape of velocity graph is a parabola as shown in Fig. 9.

Fig 4. Velocity distribution under rotation speed

Fig 5. Curve of radial velocity under rotation speed in axial direction

Fig 6. Velocity distribution under inertia

Fig 7. Curve of radial velocity under inertia without rotation speed

Fig 8. Curve of radial velocity under inertia without inlet pressure

Fig 9. Curve of radial velocity under inertia in axial direction
3.4. Effect of viscosity
From Eq. 18 and Eq. 24, it can be seen that in radial direction the viscosity gets smaller and is negatively correlated with the radial velocity. Fig. 10 and Fig. 11 show the velocity distribution under constant viscosity and variable viscosity. At the location \( r = 0.14m \), comparison of velocity under these two conditions is shown in fig. 12. The velocity under variable viscosity is 30\% larger than that under constant viscosity. That means more oil is needed to ensure the operation of the HVD clutch.

![Fig 10. Velocity distribution under constant viscosity; Fig 11. Velocity distribution under variable viscosity](image)

![Fig 12. Comparison of radial velocity under constant and variable viscosity](image)

4. Conclusion
A new model was proposed using Taylor series to describe HVD system, taking the viscosity-temperature characteristics into consideration. The flow field of working oil was analyzed in the aspects of inlet pressure, rotation speed and inertia. Several conclusions can be drawn from the study.

(1) Inlet pressure has a linear effect on the radial velocity. In radial direction, velocity is linearly decreasing and is symmetrical in axial direction.

(2) When the friction pairs rotate, the flow is offset to the friction disks while as the speed increases, the offset does not change.

(3) Inertia plays different roles at different position in radial direction. The in inner region, it hinders working oil flowing and promotes oil flow to ensure the continuity of the flow.

(4) Viscosity of working oil has a good effect on the radial velocity which under variable viscosity is 30\% larger than that under constant viscosity.

We are convinced that this work will be a contributor tool in the design and optimization for HVD clutch.
Acknowledgments
The present work was supported by the National Natural Science Foundation of China (51575245, 51675234). The corresponding content is sponsored by Natural Science, Talent Foundation (2014-23) of Jiangsu Province and Key Project of Jiangsu Province (BE2015134), Postgraduate Research & Practice Innovation Program of Jiangsu Province KYCX17_1764. Open Foundation of the State Key Laboratory of Fluid Power and Mechatronic Systems of Zhejiang University (GZKF-201717).

References
[1] C. Ning, Theoretical and Application Researches on Hydroviscous Drive, Zhejiang University, 2003.
[2] J. Cho, A multi-physics model for wet clutch dynamics, University of Michigan, 2012.
[3] J. Cui, C. Wang, F. Xie, R. Xuan, and G. Shen, Numerical investigation on transient thermal behavior of multidisk friction pairs in hydro-viscous drive, Applied Thermal Engineering, 2014, pp. 409-422.
[4] J. Cui, F. Xie, and C. Wang, Numerical investigation on thermal deformation of friction pair in hydro-viscous drive, Applied Thermal Engineering, 2015, pp. 460-470.
[5] Y. Kimura, and C. Otani, Contact and wear of paper-based friction materials for oil-immersed clutches—wear model for composite materials, Tribology International, 2006, pp. 943-950.
[6] H. Gao, G. Barber, and M. Shillor, Numerical simulation of engagement of a wet clutch with skewed surface roughness, Journal of Tribology, 2002, pp. 305-312.
[7] H. Gao, and G. C. Barber, Engagement of a rough, lubricated and grooved disk clutch ith a porous deformable paper-based friction material, Tribology transactions, 2002, pp. 464-470.
[8] S. Zhao, G. E. Hilmas, and L. R. Dharani, Behavior of a composite multidisk clutch subjected to mechanical and frictionally excited thermal load, Wear, 2008, pp. 1059-1068.
[9] C. R. Aphale, J. Cho, W. W. Schultz, S. L. Ceccio, T. Yoshioka, and H. Hiraki, “Modeling and parametric study of torque in open clutch plates,” Journal of tribology, 2006, pp. 422-430.
[10] T.-C. Jen, and D. J. Nemecek, Thermal analysis of a wet-disk clutch subjected to a constant energy engagement, International journal of heat and mass transfer, 2008, pp. 1757-1769.
[11] M. Qing-Rui, and H. You-Fu, Effect of oil film squeezing on hydro-viscous drive speed regulating start, Tribology International, 2010, pp. 2134-2138.
[12] P. Marklund, R. Mäki, R. Larsson, E. Höglund, M. M. Khonsari, and J. Jang, Thermal influence on torque transfer of wet clutches in limited slip differential applications, Tribology international, 2007, pp. 876-884.
[13] F. Xie, Y. Tong, D. Wu, B. Zhang, and K. Dai, Numerical simulation research on effect of oil groove forms on thermal behavior of friction pair in hydro-viscous clutch, Industrial Lubrication and Tribology, 2016, pp. 287-298.
[14] F. Xie, F. Xie, D. Wu, D. Wu, Y. Tong, Y. Tong, B. Zhang, B. Zhang, J. Zhu, and J. Zhu, Effects of structural parameters of oil groove on transmission characteristics of hydro-viscous clutch based on viscosity-temperature property of oil film, Industrial Lubrication and Tribology, 2017, pp. 690-700.
[15] W. Wu, Z. Xiong, J. Hu, and S. Yuan, Application of CFD to model oil–air flow in a grooved two-disc system, International Journal of Heat and Mass Transfer, 2015, pp. 293-301.
[16] H. J. Q. M. L. Lingling, and F. Linjian, Numerical simulation of flow field between frictional pairs in hydroviscous drive surface, Chinese Journal of Mechanical Engineering, 2008, pp. 72-75.
[17] J.-h. Huang, Y.-r. Fan, M.-x. Qiu, and W.-m. Fang, Effects of groove on behavior of flow between hydro-viscous drive plates, Journal of Central South University, 2012, pp. 347-356.