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ELASTO – HYDRODYNAMIC SIMULATION OF COMPLEX GEOMETRIES IN HYDRAULIC MOTORS

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ABSTRACT

As many other machines, the radial piston hydraulic motor contains a lot of tribological interfaces. One important area is the piston assembly and the journal bearing contact between the piston and cam roller. There exists good models to describe the performance of simpler geometries such as journal bearings, but when put into a system or when having a more complex geometry, the models do not apply very well. To be able to predict the tribological performance of such components, it is important to have a model that is able to include the real geometry and the properties of the system.

A simulation model of the piston assembly in Hägglunds Compact hydraulic motor was built using FE software which made it easy to include the complex geometries. The model includes the deformation of the piston. The hydrodynamics is solved by using Reynolds equation. Density/pressure and viscosity/pressure dependency for the oil are included. The whole model was solved with the built in solvers in the software.

Simulation results such as friction, hydrodynamic pressure and oil leakage was compared with test results. Good agreement between simulation and tests shows that this kind of model can be a useful tool in development and optimization of tribological systems.

1. INTRODUCTION

The performance of highly loaded lubricated contacts in hydraulic motors is often dependent upon the geometrical shape of the machine components. A good example of this is the piston assembly of the Hägglunds Drives Compact motor. Much effort has been put into experimental testing and development of especially the piston design. The

focus has been on minimizing the losses and wear in the sliding interface between the piston and the roller. The piston-roller interface is lubricated by both hydrostatic and hydrodynamic film formation. Losses in a sliding interface like this can be from viscous shearing in full film lubrication or a combination of viscous shearing and boundary friction. Because of the hydrostatic lubrication, losses also come from leakage. The piston has a rather complex geometry and the performance has been found to be sensitive to small changes in the design. To be able to make the development of such components more effective it is important to have a reliable numerical model that can adapt to different geometries.

In this paper a FE model of the piston assembly is presented. The model is built using the software Comsol Multiphysics [1] and includes elastic deformations, hydrodynamic and hydrostatic film formation using Reynolds equation. Using a software like this makes it easy to adopt the model to in principal any geometrical shapes of the sliding interface and bodies. Measurements of friction and leakage from a test rig for the piston assembly [2] has been used for comparison and validation of the numerical model.

2. EXPERIMENTAL MODEL DESCRIPTION

A test rig was built to make specific testing of the piston assembly and especially the interface between the piston and the roller. In contradiction to that in the motor, the piston is not moving with respect to the cylinder in the test rig. The piston and roller are statically pressed against a rotating drive shaft that represents the cam ring. To avoid tilting forces on the drive shaft there are two piston assemblies mounted in opposite directions. The piston assemblies can be moved sideways to test different angles (θ) corresponding to a specific point in the piston-stroke in the motor.

Figure 1: Motor and test rig schematics.

Parameters that are measured in the test rig are mainly friction and leakage but also oil film pressure can be measured in pistons with small holes drilled into the interface. The leakage is measured by metering the oil that is supplied to the interface. This oil is supplied through a small pipe that can be seen in Fig. 2. The supplied oil holds the same pressure as the high pressure underneath the piston but it is sealed to separate it from the leakage between the piston and cylinder wall.

Figure 2: Piston with oil supply pipe attached.

In Fig. 2 the roller is removed so that part of the oil supply groove at the interface can also be seen. Friction is estimated by measuring the applied torque at the drive shaft. The torque needed to overcome friction in the drive shaft bearings and other drive train losses are measured before the tests and is subtracted from the measure. The remaining friction is assumed to come from the piston/roller and roller/drive shaft interface. For further information about the test rig see [2].

3. NUMERICAL MODEL DESCRIPTION

The equations that are used and the assumptions that are made for the simulation model of the piston assembly are described. The model is implemented into the FE software, Comsol Multiphysics and some technical details about that are also presented. The model is built to represent the conditions in the test rig described in chapter 2.

3.1 Oil film equations

Under conventional thin film assumptions, the pressure in the oil film can be described by Reynolds equation:

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{\eta} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{\rho h^3}{\eta} \frac{\partial P}{\partial y} \right) = 6 u \frac{\partial}{\partial x} (\rho h) \quad (1)$$

The viscosity η is considered to be constant across the film and its variation according to pressure is described by Barus [3] expression:

$$\eta = \eta_0 \exp(\alpha P) \quad (2)$$

The Dowson-Higginson [4] density-pressure relationship is used with coefficients D_1 and D_2 from measurements by Larsson et al. [5]:

$$\rho = \rho_0 \left(1 + \frac{D_1 P}{1 + D_2 P} \right) \quad (3)$$

The boundary conditions for the pressure are set to zero at the outer boundary and to P_h in the supply groove. This means that the calculated pressure will be absolute pressure minus the surrounding pressure.

Cavitation may occur near the trailing edge of the piston-roller bearing. In order to take cavitation into account a density modification model is added to avoid large negative pressures:

$$\rho = \rho_0 \exp(\beta P), \quad P < 0 \quad (4)$$

This is the same technique for handling cavitation as used by Almqvist [6], just with a different density expression. By using this technique mass continuity is fulfilled. The expression for density in equation (4) overrides equation (3) whenever the calculated pressure is below zero. The parameter β which is determining the steepness in the density-pressure relationship is modified until a desired lowest pressure level is reached.

3.2 Elastic deformations

Elastic deformations of the piston assembly will affect film thickness and has to be included. The deformations in the piston assembly that dominates the influence on film thickness are those of the piston. This is mainly because of the geometry but also because it has a lower elastic modulus than the cylinder block and roller. Therefore only the deformations of the piston are calculated in the model. The forces acting on the piston and determining its deformations are oil film pressure, hydraulic pressure: P_h under the piston and contact pressure against the cylinder wall. The contact pressure is modelled through the penalty-barrier method [7] and is described by:

$$\begin{cases} P_c = t_n - e_n h_c, & h_c \leq 0 \\ P_c = t_n \exp\left(-\frac{e_n}{t_n} h_c\right), & h_c > 0 \end{cases} \quad (5)$$

Where t_n is the estimated contact pressure and e_n is a constant deciding the stiffness. h_c is the separating distance between the piston and cylinder wall:

$$h_c = h_{c0} - h_{cdef} \quad (6)$$

h_{c0} Contains the non deformed clearance and h_{cdef} contains the normal component of the piston-deformation/movement. A big advantage by modelling the contact this way is that the geometry of the piston can be made perfectly cylindrical and easy to mesh. This is because the small details and curvatures of the piston surface are already taken account for in the expression for h_{c0} . After solving the deformations, the film thickness can be expressed as:

$$h = h_0 - h_{def} \quad (7)$$

where h_0 is determined by the position of the roller in the non deformed piston and h_{def} contains the attribution from the piston-deformation/movement.

3.3 Force balance

The applied load must be balanced by the film pressure and the frictional forces. The applied load can be divided into X and Y composants which leads to the following equations:

$$\begin{cases} \iint ((P + \tau_{zx}) n_x) dx dy - F_x = 0 \\ \iint ((P + \tau_{zy}) n_y) dx dy - F_y = 0 \end{cases} \quad (8)$$

Where the FY composant is automatically taken care of through h_{def} in Eq. (7). This is because the piston movement is not constrained in Y direction and the film thickness will therefore settle at a level where F_y is balanced. The F_x composant is balanced by modifying the position of the roller in the expression for h_0 .

3.4 Implementation into FE software and solving the model

The model equations are solved with the FE software, Comsol Multiphysics. The deformations are solved by using the built in code for structural mechanics. The oil film equations are added to the surface mesh at the interface of the piston and roller

The piston is meshed with tetrahedral mesh and second degree Lagrange elements are used. There is no need for a separate mesh for the oil film equations; they are simply solved on the surface mesh.

The coupled sets of equations are solved simultaneously by the built in non linear solver. Because of the non linearity of the problem, it is important to have a good initial guess of the solution to achieve convergence. This can be obtained by first solving a

simplified problem, for example with iso-viscous oil and with some constraints to the piston movement. The solution of the full problem can be achieved by removing constraints and solving with the previous solution as initial guess.

4. RESULTS AND DISCUSSION

The results in this work are divided into three different sections. Input data for each section are presented in table (1). In each section results from test rig are compared with results from the simulation model. Numerical results are computed on a mesh consisting of approx. 6000 tetrahedral elements for the elastic deformations and 265 triangular elements for the oil film equations. The model is solved for steady state with parametric solver where sliding velocity, hydraulic pressure or other parameters are stepped. Each steady state solution requires approx. one minute computational time on a normal PC (2Gb RAM, 3GHz CPU).

Table 1: Input data

Section	4.1 Pressure	4.2 Friction	4.3 Leakage
P_h	10-25 MPa	10 MPa	10 MPa
u	1 m s ⁻¹	0.1-2 m s ⁻¹	0.1-2 m s ⁻¹
α	20e-9 Pa-1	20e-9 Pa-1	20e-9 Pa-1
D_1	0.6e-9 Pa-1	0.6e-9 Pa-1	0.6e-9 Pa-1
D_2	1.7e-9 Pa-1	1.7e-9 Pa-1	1.7e-9 Pa-1
θ	26.9°	26.9°	26.9°
t_n	1e7 Pa	1e7 Pa	1e7 Pa
e_n	1e14 Pa m-1	1e14 Pa m-1	1e14 Pa m-1
η_0	36e-3 Pa s	36e-3 Pa s	36e-3 Pa s
ρ_0	900 Kg m-3	900 Kg m-3	900 Kg m-3

4.1 Pressure

The point where pressure is measured was chosen as the position where the maximum pressures inside the supply ring groove has been seen in the simulations. The position of maximum pressure can vary with different conditions. In the test rig pressure is only measured at one point so the actual pressure shown in Fig. 3 is not the maximum but the pressure at one fixed position. This position is, at least according to the numerical model very close to the maximum value. The deviations between measured and simulated values are quite small. As can be seen in Fig. 4 the pressure is a lot higher at the skirt of the piston and this would have been more interesting to compare. Because of technical difficulties in doing a measure at this location, the inside of the supply groove was chosen.

Figure 3: Measured and simulated oil film pressure at the center line and a circumferential position of 26 degrees from the bottom of the piston at a sliding speed of 1 m/s.

Figure 4: Film pressure distribution and deformation of the interface (the deformation is greatly exaggerated for visualization).

4.2 Friction

The numerical model presented here only takes account for the friction at the piston/roller interface and is only considering full film lubrication. When comparing friction, a friction coefficient is calculated by dividing the friction force with the applied force:

$$\mu = \frac{\iint \tau_{zx}(z=h) dx dy}{\sqrt{(F_x^2 + F_y^2)}} \quad (9)$$

Where $\tau_{zx}(z=h)$ is the oil film shear stress at the surface of the roller. Good correlation between measured and simulated friction can be seen in Fig. 5. Of course, when moving into lower speeds with mixed lubrication, the numerical model will not give accurate results. A small deviation can also be seen at higher speeds. One possible reason for this is that the friction at the driveshaft/roller interface gets proportionally bigger as the speed increases. Higher sliding speeds also imply higher thermal loads which can affect the friction. Higher temperatures would lead to lower viscosity which generally would lead to lower friction. But not only the viscosity will change, thermal expansion and

also the change of viscosity it self will affect the deformation of the piston assembly. This will alter the load distribution (film pressure) and thereby also the friction of the system. The complexity makes it hard to predict in which way it would affect but it would be interesting to include it in the model.

Figure 5: Friction coefficient from tests and from simulation for piston assembly with zero none-deformed radial clearance, $P_h=10$ MPa and $\theta=26.9^\circ$.

4.3 Leakage

The flow in thin films is given by the sum of couette and poisseuille flow:

$$\begin{cases} q_x = -\frac{h^3}{12\eta} \frac{\partial P}{\partial x} + \frac{u h}{2} \\ q_y = -\frac{h^3}{12\eta} \frac{\partial P}{\partial y} \end{cases} \quad (10)$$

The terms in Eq. (10) describes the oil film flow field (averaged across the film). An example of a typical flow field for the piston assembly can be seen in Fig. 6.

Figure 6: Simulated oil film flow field.

Integrating the normal component of the flow field along the outer edge of the oil supply groove gives the total oil leakage between piston and roller. Large discrepancies can be seen when comparing simulated and measured leakage. This is especially seen when comparing piston assemblies with small radial clearances. An example of this is shown in Fig. 7 where the mismatch is as large as up to 500%.

Figure 7: Oil leakage in piston assembly with zero none-deformed radial clearance, $P_h=10$ MPa and $\theta=26.9^\circ$.

Equation (11) tells that the leakage depends strongly on the film thickness h . This gave the idea to test if a small deviation from a circular form of the bearing bore could affect the result. When the inside of the piston is machined it tends to get a slightly higher surface at the bottom and at the uppermost parts. To simulate this, a sinusoidal deviation from the circular form with amplitude of only $1.5 \mu\text{m}$ was added to the model. Even though the film thickness is approximately one order of magnitude bigger than this, it had a big influence on leakage. In Fig. 8 a big increase in leakage, up to the level of the measured can be seen for the simulated results. This indicates that it is important to include the roundness of the piston bearing bore to get accurate leakage. When looking at friction these small roundness errors have not affected the result significantly (slightly lower simulated friction for the piston with roundness error).

Figure 8: Oil leakage in piston assembly, test1 and test2 same as in Fig. 6. Simulation results with $1.5\mu\text{m}$ sinusoidal roundness error.

5. CONCLUSIONS

Model is suitable for EHL-simulation; the use of FE software makes it easy to implement the model to complex shaped geometries.

Good agreement between simulated and measured friction and pressure for the Hägglunds drives piston assembly. Simulation results for leakage shows the importance of including the roundness of the bearing bore.

The measured friction coefficient shows that piston assembly runs in boundary lubrication regime under low speeds. To be able to predict the performance when running near or in boundary lubrication regime it would be interesting to include the effects of surface roughness.

The importance of the small change in roundness indicates that thermal distortion also could be of importance. When optimizing a system it is often interesting to increase load and speed, raising the power/weight ratio. It is often thermal effects that will set the limit when doing so. This makes it desirable to include thermal effects to the model.

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NOMENCLATURE

X, Y, Z	Cartesian space coordinates	m
x	Spatial coordinate, sliding direction	m
y	Spatial coordinate	m
z	Spatial coordinate, across film	m
n_x, n_y, n_z	Surface normal composants	
P	Pressure	Pa
P_h	Hydraulic pressure, pressure in the cylinder	Pa
α	Pressure-viscosity coefficient	Pa-1
η	Viscosity	Pa s
η_0	Viscosity at ambient pressure	Pa s
ρ	Density	Kg m-3
ρ_0	Density at ambient pressure	Kg m-3
h	Film thickness	m
u	Sliding velocity	m s-1
τ	Shear stress	Pa m-1
t_n	Approximated contact pressure parameter	Pa
e_n	Contact pressure stiffness parameter	Pa m-1
θ	Contact angle against cam ring/drive shaft	°
D_1	Pressure-density coefficient	Pa-1
D_2	Pressure-density coefficient	Pa-1
β	Pressure-density coefficient (cavitation zone)	Pa-1

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