NUMERICAL AND EXPERIMENTAL STUDY ON NOVEL HYDRAULIC PUMP CONCEPT

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ABSTRACT

This paper discusses an investigation on a novel hydraulic pump concept. The idea aims on a pump principle to be directly connected to high-speed electric motors to build compact high-pressure drives. The pump can theoretically work without solid contact and has no kinematic pulsation. The composition of the pump is basically similar to cylindrical journal bearings. In general, a journal bearing is used to support external loads on a rotating shaft. When the shaft rotates with the loads, hydrodynamic fluid force is generated in the fluid film, which compensates the external force and reduces an eccentricity of the shaft. The difference of the introduced pump concept from the bearing is that the eccentricity is fixed and it has an outlet port at the high pressure area, through which the fluid can be transferred. To examine the functionality of the pump, a test bench is constructed and experiments are performed to investigate the pressure build-up and discharging flow of the pump concept. The experimental results are numerically analyzed by using elasto-hydrodynamic lubrication (EHL) simulation, based on the 2D Reynold's equation. From these results, the functionality of the pump concept is confirmed. Moreover, several variables that influence the characteristic curve of the pump are studied. Based on these results, geometric parameters of the test pump are redesigned to discharge sufficient flow rate for usage as commercial pumps.

Keywords: Hydrodynamic Pump, High-Speed Drive, Journal Bearing, EHL Simulation

1. INTRODUCTION

Hydraulic pumps are widely used in many industry fields to supply hydraulic energy. The hydraulic energy is transformed from mechanical energy via the pumps, which is often connected to an electric motor for rotation. Thus, the pump can transform the energy with the permissible speed and torque of the connected motor, which determines the operating range of the configured hydraulic system. The further development of electric motor technology has increased operating speed limitations, as well as the miniaturization of the motor, even if it has the same power capacity [1].

The downsized electric motor is one of many ways to reduce weight and enhance compactness in the combined units. The combined unit has still leeway to be reduced by the weight of the pump. High-speed technology is the key to downsizing hydraulic pumps, which can significantly save weight and installation space [2]. This is because, even if the displacement volume is small, the

same flow rate as that of the conventional pump can be discharged at high rotational speed.

Accordingly, studies on high-speed pump are in progress. In case of hydrostatic pumps, such as piston pumps, gear pumps and vane pumps, solid contacts are inevitably generated to form the displacement volume, which shortens the life of the components due to wear. Especially under the high-speed operating conditions, the components are prone to wear and deformation, which significantly reduces pump efficiency. One of ways to improve the tribological characteristic of the material, laser surface texturing is used [3]. In addition, high-speed piston pumps are experimentally proven to have large shear losses in the cylinder block due to the viscosity of the fluid and to increase leakage due to temperature rise [4]. Moreover, components, which create a displacement volume in the pump, result in dynamic unbalance and a slower response to a speed change due to moment of inertia [5].

Unlike the hydrostatic pumps, hydrodynamic pumps, such as impeller pumps, are more suitable

for high-speed drives because the pressure and flow pulsation are quite low and there is no contacting surfaces in the pump [6]. However, the maximum working pressure is normally lower compared to the hydrostatic pumps. Moreover, the impeller pump could generate oscillations with large dynamic pressures force at high operating speed, which could cause cracks in the impeller due to large stress [7].

In a previous research [8], two novel hydrodynamic pump concepts are introduced. The first concept is numerically studied, which is similar to radial journal bearings. The introduced pump concepts theoretically have no solid friction and no kinematic pulsation. It also has an advantage over the conventional hydrodynamic pumps in that it can reach a higher operating pressure. The research also describes a sophisticated concept that both walls are driven to avoid shear loss, based on methods developed with the first concept.

In this paper, the first pump concept is experimentally studied and analysed numerically. An elasto-hydrodynamic lubrication (EHL) simulation is used to analyse pressure distribution and discharging flow of the pump. Influence of several operating conditions on the discharging flow is examined. From experimental results, the functionality of the pump concept is confirmed, and the pump geometry is redesigned for larger volume flow by using the simulation.

2. WORKING PRINCIPLE OF THE NEW PUMP CONCEPT

Many mechanical elements inevitably generate friction during relative movement. To reduce the friction and wear, a lubricated fluid film is used on the conformal elements surface and it provides smooth motion and enhanced operating life for the machine elements. When the surface is moved, shear flow, so called Couette flow, is generated because of the viscosity of the fluid. A hydrodynamic pressure is developed, when the shear flow occurs on converged surfaces. This hydrodynamic pressure implies that it can support external forces and reduce the contact force between the surfaces. This phenomenon is called hydrodynamic lubrication [9].

The conformal surfaces can be shaped infinitely like a journal bearing (**Figure 1a**), and the flow and pressure are developed on the surfaces just by rotating the shaft. Standard

journal bearings have one inlet port to supply fluid for the lubrication. However, installation variety is limited because the inlet port can be blocked, depending on the rotational direction of the shaft and the external load direction.

Unlike the standard bearings, two port journal bearings (**Figure 1b**) are used to supply fluid regardless of the operating condition [10]. However, when one port is located in the high pressure region, a negative flow occurs in which the fluid flows in the opposite direction due to the pressure difference from the port pressure [11]. This causes a pressure loss and reduces load carrying capacity of the journal bearing. To

Figure 1: Two journal bearing types and a novel hydraulic pump concept

prevent the negative flow, check valves are used to restrict the flow direction of the fluid [12].

The idea of the novel pump concept originated from the lubrication theory and the two ports journal bearing points out the possibility of the idea. The negative flow in the two ports journal bearing can be a discharging flow as a hydraulic pump. As shown in the **Figure 1c**, the first pump concept is designed with the same shape as the two ports journal bearing. Differences are that the inlet port is used as an outlet port, through which the fluid can be delivered, and the eccentricity is fixed to allow pressure build-up without the external force creating an eccentricity. The functionality of the hydraulic pump was numerically examined in the previous study [8].

Discharging flow through the outlet port is composed of shear flow (Couette flow) and pressure flow (Poiseuille flow). **Figure 2** shows the postulated discharging flow in θ -direction. Under the assumption that there is a non-turbulent flow at the outlet, the discharging shear flow is calculated as the difference between inflow and outflow. The shear flow is proportional to the perpendicular length in flow direction (B in **Figure 2a**) and the difference of the gap height at the border of the outlet port. That is, the shear flow is significantly affected by the shape of the outlet port.

The pressure flow is generated by the difference between the outlet pressure and the fluid film pressure. The pressure flow is proportional to the pressure gradient at the outlet border. This flow becomes zero when the outlet pressure and pressure near the outlet port are equal, in other words, when the pressure gradient at the outlet boundary is zero. Due to the characteristic that the discharge flow is inversely

Figure 2: Composition of the discharging flow

proportional to the discharge pressure, this pump can be classified as a hydrodynamic pump.

3. TEST BENCH

The test bench, shown in **Figure 3**, is constructed for functionality examination of the pump concept. A rotating shaft, connected to the motor using an elastomer coupling, is supported with one ball bearing and two DIN 504 standard journal bearings [13]. The ball bearing is used to apply the external load to the rotating shaft to generate the eccentricity. The external force is applied by screwing and the load direction is adjustable. The first journal bearing has an outlet port and this works as the novel hydrodynamic pump. **Figure 4** represents the geometry of the pump and the parameters are given in **Table 1**. The second journal bearing is for force balance against the external force. The journal bearing used as the pump, has an outlet port, so it may have different pressure field from other journal bearings depending on the pressure of the outlet port even with the same eccentricity. However,

Figure 3: Test bench of the novel pump concept

Figure 4: Geometry of the test pump

since the eccentricity of both journal bearings cannot be measured in this experimental setup, it is simply assumed that half of the external force is applied to the pump. HLP 46 mineral oil is used for the fluid, which is supplied from the tank at the top and the supply pressure is about 1bar tank pressure. A throttle valve is used to control the outlet port and the mass flow is measured with a platform scale. **Table 2** shows specifications of the test bench.

Table 1: Geometry parameter of the test pump

Parameter	Value [mm]
Journal diameter D	25
Journal width B	60
Nominal gap height h_G	15×10^{-3}
Inlet port diameter d_{in}	15
Outlet port diameter d_{out}	0.7

Table 2: Specification of the test bench

Parameter	Value	Unit
Max. Motor speed	1000	rpm
Max. External load	1000	N
Lubricant	HLP 46	-
Oil Temperature	25 ± 1	$^{\circ}\text{C}$

4. SIMULATION MODEL

In this paper, an elasto-hydrodynamic lubrication (EHL) simulation, set up using MATLAB, is used to interpret the pressure distribution and validate the discharging flow of the hydrodynamic pump concept in a steady state.

First, a fluid film mesh is generated from the geometry of the pump (shown in **Figure 4**) and journal bearing and the parameters (**Table 1**) are

used. The gap height of the fluid film is calculated geometrically, as shown in Equation 1:

$$h(\theta) = h_G + e_x \cos \theta + e_y \sin \theta \quad (1)$$

Then, the pressure boundary conditions, such as inlet pressure, outlet pressure and environment pressure, are defined. The pressure boundary conditions are constrained to the nodes where the geometry is located. For the unconstrained nodes, the pressure is calculated using the Reynolds' equation in two dimensions, which has been discretized for the mesh in the form of finite volumes. In addition, Jakobsson-Floberg-Olsson (JFO) cavitation model [14, 15] is used to avoid negative pressure for more accurate calculation. The Reynolds' equation with the JFO cavitation model is shown in Equation 2:

$$\nabla \cdot \left(\frac{\rho h(1-\xi)}{2} U \right) - \nabla \cdot \left(\frac{\rho h^3(1-\xi)}{12\eta} \nabla p \right) = 0 \quad (2)$$

Fischer-Burmeister function is applied as a mathematical model to constrain the pressure (p) and the cavitation factor (ξ) [16].

The output flow of the pump can be simply calculated at the border of the outlet hole with the assumptions that the pressure at the objects is constant and that there is no turbulent flow in the outlet hole. Therefore, the output flow is calculated by summation of discretized volume flow at the nodes, which are defined as the border of the outlet hole. In the same way, the leakage to both sides of the journal can be calculated.

In this simulation, MATLAB PDE Tool is used to calculate the deformation of the solid structure [17]. Thereby, interfacing work between different programs is avoided and overall computation time is reduced. **Figure 5** shows a FEA model of the test bench. The FEA model is simplified to save calculation time. Material properties of steel (**Table 3**) are applied in each bushing and shaft. Then, as force

Figure 5: Simplified FEA model of the test bench

boundary conditions, the pressure fields, calculated from the Reynold's equation, are applied in the fluid area, and the distributed external load is applied to the center of the shaft.

Table 3: Material properties of the components

Parameter	Value	Unit
Young's modulus E	2×10^{11}	Pa
Poisson's ratio ν	0.3	-
Mass density ρ_{mass}	7850	kg/m ³

The shaft in the bushing is set to be the eccentricity position where the total force is balanced, because the simulation is performed in a steady state. Since the stiffness of the elastomer coupling is sufficiently small compared to the shaft, it is assumed that the effect on the deformation of the shaft is negligible. Therefore, both sides of the shaft are constrained in an eccentric position where the total force is balanced. Under the FEA boundary conditions, the deformation of the shaft is calculated.

The deformation result is again applied to the fluid film simulation to calculate a new pressure field in the deformed surfaces. The iteration process is ended, when the results are converged. **Figure 6** shows the iteration process in the EHL simulation.

Figure 6: Iteration process of EHL simulation

5. RESULTS AND DISCUSSION

Several variables that could affect the discharging flow of the pump are classified into the cases in **Table 4**, and the discharging flow is

measured by changing the outlet port pressure. The EHL simulation is used to analyze the measured results and validated under the same operating conditions.

Table 4: Cases of measurement and simulation

Case	Load [N]	Load angle [deg]	Rotational Speed [rpm]
1	500	90	500
2	500	90	1000
3	500	80	500
4	500	100	500
5	250	80	500

5.1. Deformation effects

In order to confirm the effect of deformation in the simulation, the experimental results for Case 1 are compared with the results with or without deformation. **Figure 7** represents the deformation of the shaft and test pump bushing under the same operating condition at 2 bar outlet pressure. In case of the shaft, the maximum deformation is 6.8 μm in the middle of the shaft, and the deformation appeared almost symmetrically through out the external load. In

Figure 7: Deformation of the pump components

the case of the pump bushing, the pressure distribution of the fluid film with a maximum pressure about 7 bar is applied to the surface. The maximum deformation is $0.06 \mu\text{m}$, which is very small compared to the shaft, and the effect on the gap height is negligible.

Figure 8 shows the characteristic curve of the pump under the operating conditions of Case 1. The discharging mass flow is measured about 3.5 g/min at an outlet pressure of 0 bar (relative pressure) and it decreases with increasing outlet pressure, which is the characteristic of a hydrodynamic pump. The simulation results with deformation are shown to be more similar to the measured results and less error than the other simulation results, which do not consider the deformation.

From the results above, it can be seen that under the current operating conditions, the effect of the pressure on the shaft and bushing deformation is very small. However, it is shown that deformations, caused by external forces and fluid forces, influence the change in gap height and should be taken in to account. Therefore, this deformation is considered in subsequent simulations.

Figure 8: Comparison of measurement and simulation results in Case 1.1 operating condition

5.2. Influence of rotational speed

Figure 9 shows measured results under Case 1 and 2, which are under same load condition with different rotational speed. In addition, since the eccentricity cannot be measured in the test bench, the variation of the eccentricity is estimated through the simulation, as shown in **Figure 10**.

When the eccentricity is fixed, the pump has a wider pressure range because of the increased dynamic pressure at high rotational speeds. In this experiment, however, the eccentricity is not fixed. Therefore, the maximum pressure range of the pump is similar because a pressure against to the external force is formed in the fluid film.

In addition, the discharging flow is measured slightly higher at 1000 rpm in the overall range. Under current experimental conditions, the pressure field appears similar because pressure is generated in response to external forces. However, different eccentricities are created to form the same pressure field. Therefore, in case of 1000 rpm, there is a larger gap height at the outlet port position and has a slightly higher discharge flow. The shear flow is proportional to the rotational speed, but the outlet port is very small, so the portion of the total discharging flow is very small [12]. This explains the tendency that the discharging flow of the two cases becomes similar as the outlet pressure increases.

Figure 9: Measurements under Case 1 condition

Figure 10: Simulation result of the eccentricity

5.3. Influence of load direction

Figure 11 shows measured results at the same rotational speed under the different load direction with 500 N (Case 1, 3 and 4). The graph represents almost the same characteristic curve when the load directions are 80 ° and 100 °. In the case of 90 °, the operating pressure range is until about 6.5 bar.

Figure 12 shows the simulated pressure field with each load direction at the 2 bar outlet pressure. When the external force is applied in

Figure 11: Measurements under Case 1, 3 and 4

Figure 12: Simulation results for pressure distribution under different load directions

90 °, it can be seen that the pressure distribution is nearly symmetrical around the outlet port. However, when the direction of the external force changes, the pressure distribution centering on the outlet port is biased to one side, explaining that the maximum pressure of the outlet port is lowered.

5.4. Influence of load magnitude

In order to confirm the effect of the eccentricity on the characteristic curve, the discharging flow is measured by varying the magnitude of the load force. **Figure 13** shows the experimental results under the test conditions of Case 3 and 5.

When 500 N force is applied on the pump side, the maximum outlet pressure is measured about 5.5 bar. In the other case, the maximum outlet pressure is about 3.5 bar.

Figure 14 shows the simulated results of changing the outlet pressure with 1 bar intervals under the same test conditions to confirm the change of eccentricity. The results represent that the eccentricity is increased when a larger external force is applied. In addition, as the outlet pressure is decreasing, the eccentricity is increased to compensate for the pressure loss in the pressure field, caused by the discharging flow through the outlet port.

Figure 15 shows the pressure distribution at the 2 bar outlet port pressure. Under the 250 N and 500 N load condition, the maximum pressure in the pressure field is calculated to be each around 4.5 bar and around 6.5 bar. Although the eccentricity is always large with the 500 N load, the mass flow varies, depending on the outlet pressure. This is because the gap height at the outlet port position is changed by the eccentricity variation. As mentioned previously, the discharging flow is also affected by the gap

Figure 13: Measurements under Case 3 and 5

Figure 14: Simulation result of the eccentricity under different load

Figure 15: Simulation result for pressure distribution under different load

height. As the eccentricity increases, the gap height at the outlet port position is decreasing relatively. Therefore, although the pressure field has a sharp gradient, the discharging flow could be smaller, depending on the gap height at the outlet port location, like at 0 bar in **Figure 13**.

6. REDESIGN

In the previous experimental conditions, very small mass flows are measured. The reason is that it is difficult to apply a large load, generating eccentricity. In addition, the motor speed is not fast enough to generate a high hydrodynamic pressure. Moreover, it was numerically examined in the previous study that the pump can also generate more mass flow depending on the size of the outlet port [8]. Taken together, the factors affecting the magnitude of the discharging flow are as follows:

- Large gap height at outlet port position
- Large outlet port size
- High rotational speed

Therefore, to verify if it is possible to produce a discharge capacity suitable for use as a conventional pump, the pump is redesigned with different outlet shapes and larger eccentricity in the same journal size. The simulations are carried out under high-speed driving conditions, to the purpose of the pump concept.

Table 5 shows the redesigned geometry parameters. For the large gap height at the outlet port position, the eccentricity and nominal gap height are increased. The diameter of the outlet port is also increased to 15mm diameter. The inlet port shape is changed to rectangular. This is to reduce the cavitation area by widening the oil supply area, especially at high-speed area. These parameters are not yet numerically optimized, but are selected empirically for large outlet flow. In addition, FEA boundary conditions are symmetrically set up, as shown in **Figure 16**. The size of bushing and journal is the same as before.

Table 5: Redesigned geometry parameter

Parameter	Value	Unit
Journal diameter D	25	mm
Journal width B	60	mm
Eccentricity in x-direction e_x	25	μm
Eccentricity in y-direction e_y	-3	μm
Nominal gap height h_G	50	μm
Inlet port width a	30	mm
Inlet port angle b	68	deg
Outlet port diameter d_{out}	15	mm

Figure 16: Redesigned pump model and FEA boundary conditions

Figure 17 represents the simulations results with the redesigned parameters. Not only the mass flow but also the pressure range is increased by increasing the rotational speed, due to the fixed eccentricity. These simulation results suggest that the magnitude of the mass flow can be significantly increased with optimized geometry parameter, compared to the experimental values.

Figure 17: Simulation results of characteristic curve with the redesigned parameters

Figure 18 shows the pressure distribution at 10000 rpm and 1 bar outlet pressure, and **Figure 19** shows the deformation under the operating condition. Maximum pressure is about 50 bar, and the pressure is distributed symmetrically. In the high-pressure region, the gap height increased due to the deformation due to pressure, and it is confirmed that the gap height of the low-pressure side is decreased, as the journal is bent.

7. CONCLUSION AND OUTLOOK

In this paper, a novel hydrodynamic pump concept is introduced. The pump concept is inspired by hydrodynamic lubrication theory and has a design based on conventional journal bearings. The concept is studied experimentally to confirm its functionality as a hydrodynamic pump and analysed numerically using EHL simulation. The experiments and simulations have proven the feasibility of the pump concept. The following conclusions are obtained from the experiment and simulation results.

- The pump concept has the characteristic curve of a hydrodynamic pump, that the discharge flow is decreased with increasing the discharging pressure.
- The pressure of the fluid film is increased by increasing the eccentricity. This means that

Figure 18: Simulation result of pressure distribution at 10000 rpm and 1 bar outlet pressure

Figure 19: Deformed and undeformed gap height in low and high-pressure area

the pressure range of the pump can be controlled by adjusting the eccentricity.

- However, the discharge flow can be increased, when the gap height at the outlet port position is increased.
- The outlet port position on the pressure field affects to the pressure range and discharging flow, because it is related to the gap height.

Based on the conclusions from the results, the geometry parameters of the pump are designed for higher discharging flow and simulations are performed. From the simulation results, it is estimated that the discharge flow can reach the capacity of a conventional small pump with optimized pump geometry.

As a general observation, unlike conventional hydrodynamic pumps (e.g., impeller pumps), the introduced pump concept has a high maximum pressure range. The operating range of the pump (i.e., the maximum output flow and pressure) is influenced by many operating conditions and also pump geometry. However, the introduced pump

has a great advantage, which is that the components are very simple and easy to manufacture. Moreover, the pump can be operated without solid contact. This means that the pump has less friction and wear, which has long service times for higher durability. Therefore, it is expected to be a compact hydrodynamic pump with high reliability.

For future research work, the pump geometry will be optimized by using the EHL simulation. The optimized geometry is evaluated in a new test bench. The simulation will be updated to consider thermal effects under in unsteady state for calculating an efficiency. Also, an analytical equation for the output flow will be derived.

NOMENCLATURE

a	Redesigned inlet port width
B	Journal width
b	Redesigned inlet port angle
D	Journal diameter
d_{in}	Inlet port diameter
d_{out}	Outlet port diameter
E	Young's modulus
EHL	Elasto-hydrodynamic lubrication
$ e $	Absolute value of eccentricity
e_x	Eccentricity in x-direction
e_y	Eccentricity in y-direction
F_{load}	Load force
F_{fluid}	Fluid force
h	Gap height
h_G	Nominal gap height
$\dot{m}$	Mass flow rate
p	Pressure
U	Relative velocity of surface
η	Dynamic viscosity
θ	Radial position
ν	Poisson's ratio
ξ	Cavitation factor
ρ	Fluid density
ρ_{mass}	Mass density

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