Development of a Low-Torque Electric Oil Pump for Cooling of BEV Motors

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To reduce the torque of EOPs (JTEKT's electric pumps equipped with motors and internal gear pumps), we invented an internal gear pump with a new structure featuring additional grooves on the inner surface of its housing. Through performance testing, we confirmed this internal gear pump demonstrated notably lower torque than conventional pumps. This is especially suitable for high-flow-rate EOPs such as those used in cooling systems for BEV (Battery Electric Vehicle) motors. This report discusses mechanism analysis and approximate mathematical expression, as well as the method and results of contribution by factor in relation to the pump's torque loss. This report also details the structure of the developed product conceived based on the aforementioned analysis, etc., as well as test results.

Key Words: *internal gear pump, torque, mechanism analysis, contribution by factor, electric pump*

1. Introduction

The first hydraulic pumps (oil pumps) driven by motors instead of engines for use in automotive hydraulic systems such as transmission clutches, power steering systems, and various lubrication systems have been available for quite a long time. JTEKT has been producing and supplying EOPs (JTEKT's electric oil pumps equipped with motors and internal gears) with a unique structure shown in **Fig. 1** since 2011 to respond to recent trends toward energy saving and electrification.

An EOP is a motor and pump unit where a motor and its controller (PCB are integrated with a pump. They are smaller and require fewer components than conventional alternatives.

Initially, EOPs were mostly used in engine start-stop systems for petrol vehicles, and their fluid flow rates were generally 4 l/min or less. In recent years, market demand for electric oil pumps to, for example, cool BEV (Battery Electric Vehicle) motors has been rapidly growing. In line with this demand, pumps are more and more often expected to deliver higher flow rates of up to 15 l/min. This trend is driven by such factors as the expanding range of BEVs and the growing popularity of systems that use the oil as an electrically insulated coolant circulating over wider areas and directly cooling internal electric parts to help reduce their size and increase their efficiency.

JTEKT sees great potential in high flow rate EOPs and is developing new products. This paper discusses a low-torque internal gear pump which is an indispensable feature when designing a high energy efficiency and small EOP.

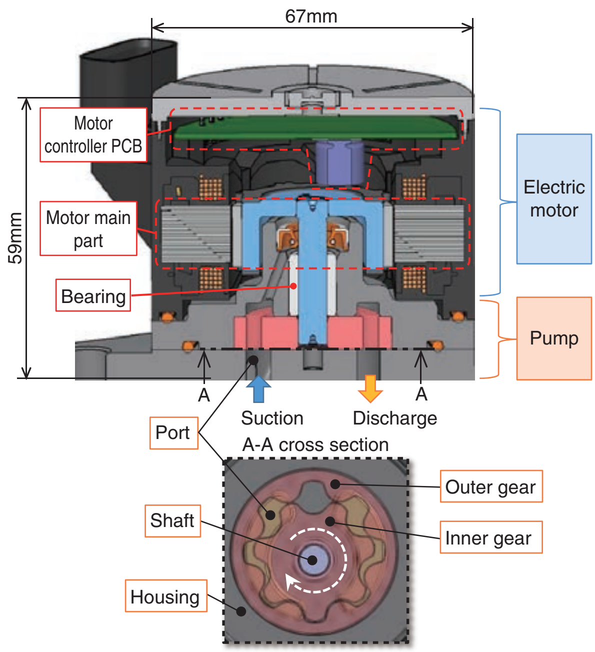


Fig. 1 JTEKT's EOP (example) and internal gear pump structure

2. Internal Gear Pump for EOP

2. 1 Structure

Figure 1 shows the internal gear pump used in the EOP discussed in this paper. As shown in the figure, the pump has a structure where a pair of external and internal gears rotate inside a housing filled with hydraulic fluid (oil). One of the gears, called the inner gear, has a shaft driven by a motor, and the other, called the outer gear, is meshed with and driven by the inner gear. This is a positive displacement pump, which suctions and discharges fluid from and to the outside through the ports created on the housing as the volume of one of the chambers located between the gears expands while the volume of the other chamber shrinks.

As they rotate, the gears slide with the housing in two types of locations; one is on the front and back of the gears, the other is on the outer circumference of the outer gear. Although clearances are provided in these locations (**Fig. 3**), they are designed to be very small so that the reduction in the pump discharge flow rate due to internal leakage is minimized. The clearance is only a few dozen micrometers on the front and back of the gears, while it is several times greater that on the outer circumference of the outer gear.

2. 2 Impact of Pump Torque, Current Situations

The motor in an EOP is affected by the torque of the pump. The effects are as follows:

- Roughly in proportion to the torque, either the electric current, the axial length or diameter of the motor main part (**Fig. 1**), the magnetic flux density of the magnets, or the number of turns of the coils increases.
- Heat is generated roughly in proportion to the square of the torque, and in order to suppress the temperature rise due to this heat to an acceptable level, the motor must have components made of expensive high-quality materials and be made larger so that it has sufficient area for heat dissipation.
- The electronic components on the motor controller PCB must be upgraded to expensive ones to meet the increase in the electric current in proportion to the torque.

These factors make it evident that reducing the torque of an EOP reduces the electric consumption as well as the size and cost of the EOP. These reductions are significant especially if it is a high-power product with a pump that produces a high flow rate. Moreover, this minimizes the necessary changes in the structure and motor of an EOP already proven to be reliable in the field when implementing renewed output specifications to it. This makes pump torque reduction a valuable means to maintain the reliability of EOP.

The internal gear pump suitable for the target EOP has a pump pressure specification of as small as about 0.1 to 0.2 MPa. As indicated in the results obtained by testing with conventional structure pumps in **Fig. 6**, torque loss accounts for more than half of the pump torque.

In contrast, the engine-driven internal gear pumps for vehicle transmission produced by JTEKT deliver pressures between about 2 to 6 MPa, where the effective torque, which is proportional to the product of pump displacement per revolution and pressure, comprises the dominant fraction[1]. Here, torque loss accounts for only a minor fraction of the pump torque, and therefore reducing it has not been as important an issue[2) 3)].

We set ourselves the goal of reducing the torque of the internal gear pump for EOP, which is especially beneficial for high flow rate EOP. To that end, we analyzed the mechanism of how pump torque loss occurs, determined the factors affecting the mechanism, and identified the degrees of contribution of these factors. Based on this analysis, we invented a structure that notably reduced the torque of the pump.

3. Torque Loss Generation Mechanism, Factor Analysis

3. 1 Theoretical Analysis

This section discusses in detail the mechanism that causes the loss of torque in internal gear pumps and a mathematical model that represents the mechanism. **Figure 2** shows the physical phenomena that are considered to be the major factors that affect the mechanism, the cause-effect relationships between them, and the calculated rates of contribution of each factor to the overall torque loss.

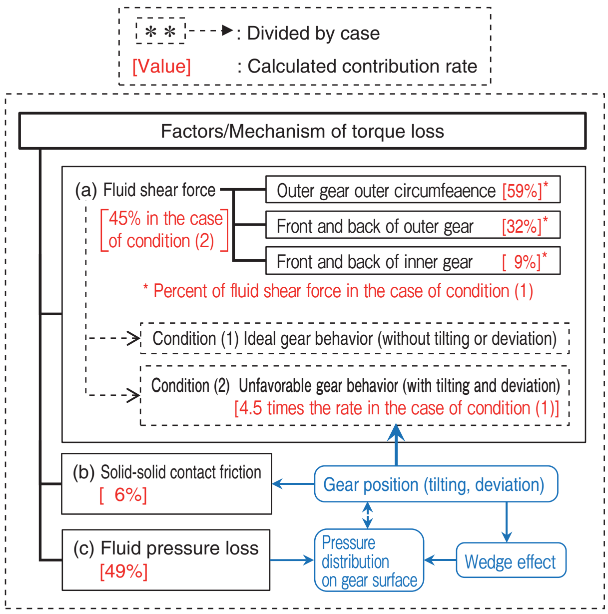
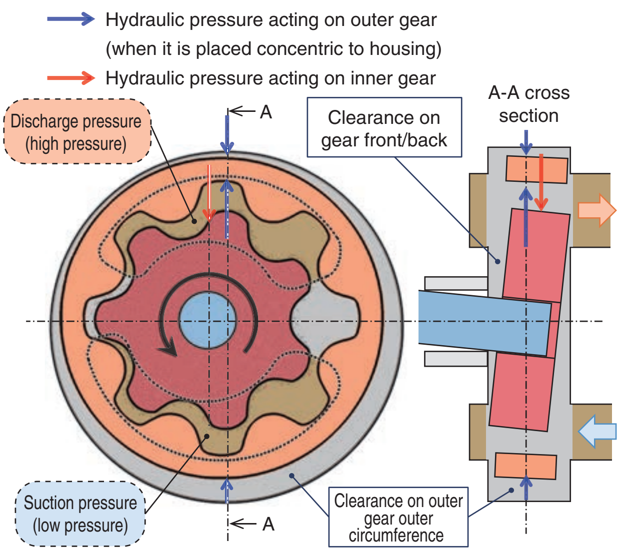


Fig. 2 Mechanism for pump torque loss generation (relationship of each factor's influence and contribution rate calculation results)

3. 1. 1 Gear Behavior

When the outer gear of the pump discussed in this paper is placed concentric to the housing, hydraulic pressure acts on the gear in the direction radial to it during pump operation, as indicated by the arrows in **Fig. 3**, if we take the qualitative distribution of the pressure on the gear's surfaces into account. This means that, as in **Fig. 3**, the outer gear rotates in a radially displaced position inside the housing, and a narrow part is created in the clearance between the outer gear outer circumference and the housing inner circumference. In this narrowed part, the wedge effect[4] arises and generates a hydraulic reaction, and as a result the pressure on the gear is balanced.

The inner gear of the pump is supported by its cantilever shaft. It is therefore tilted against the housing due to hydraulic pressure, again as shown in **Fig. 3**. The outer gear is also tilted against the housing because of its contact with the inner gear, although to a lesser degree than the inner gear.



The clearances are greatly exaggerated in this figure.

Fig. 3 Gear behavior due to hydraulic actuation

3. 1. 2 Force Generated by Contact with Fluid in the Clearance

In each clearance, fluid shear force acts on the gear in the direction opposite to the direction of the gear's motion because of the viscosity of the fluid. The force is negligible in the region in between the gears because both the area of sliding and the relative velocity are small.

The magnitude of the fluid shear force per unit area at an arbitrary point on the gear surface is expressed by the following equation:

$$\tau = \mu \cdot \frac{du}{dy} \fallingdotseq \mu \cdot \frac{\omega r}{h} \qquad (1)$$

where:

μ = static viscosity of the fluid,
u = component of fluid flow velocity in the direction of motion of the gear,
y = height above the gear surface (vertically toward clearance),
ω = angular speed of rotation of the gear,
r = distance of an arbitrary point on the gear surface from the center of rotation, and
h = height of clearance.

In equation (1), the fluid shear force increases to infinity as the height of clearance h approaches zero. Its magnitude can be quite significant when the clearance is partially narrowed due to the gear behavior discussed in the earlier section. Meanwhile, the actual force becomes smaller than the force calculated in equation (1) as the clearance is narrowed. One of the reasons for this is that the fluid is less viscous due to heat by friction when the clearance is extremely small. Another reason its the tiny unevenness on the surfaces of the gear and the housing, which results in a very small area of true contact even when they are in contact with each other.

The gears are tilted and radially displaced by the hydraulic pressure acting on them, as we discussed earlier. This does not, however, necessarily mean that the gears directly contact the housing. Rather, the fluid can be present in between these solids due to the fluid wedge effect, which arises when the clearance is small and causes an increase in the fluid pressure due to the kinetic energy of the solids on their surfaces. The wedge effect is complicated, as it is affected by various factors including the tilting of the gears and the surface roughness of the solids concerned.

3. 1. 3 Pressure Loss of Fluid Flowing in the Pump

Fluid loses some of its pressure energy when it flows. The pressure acting on the gears in the pump is therefore larger than the nominal pressure of the pump, which is the difference between the discharge and suction pressures detected in the piping that is outside of the pump. This pump pressure loss should not be ignored in the case of low pressure high flow rate pumps like the one discussed in this paper.

The factors contributing to the pressure loss are as follows: the flow in the chambers that either expand or shrink between the gears, the abrupt change in the cross-section of the flow when it enters one chamber from the inlet port and leaves the other chamber from the outlet port, and the resistance of the fluid flowing through the ports until it reaches the piping.

As the fluid flows in the abovementioned manner, intense swirls of various sizes appear in it. Therefore, the dominant factor of the pressure loss is not the viscous

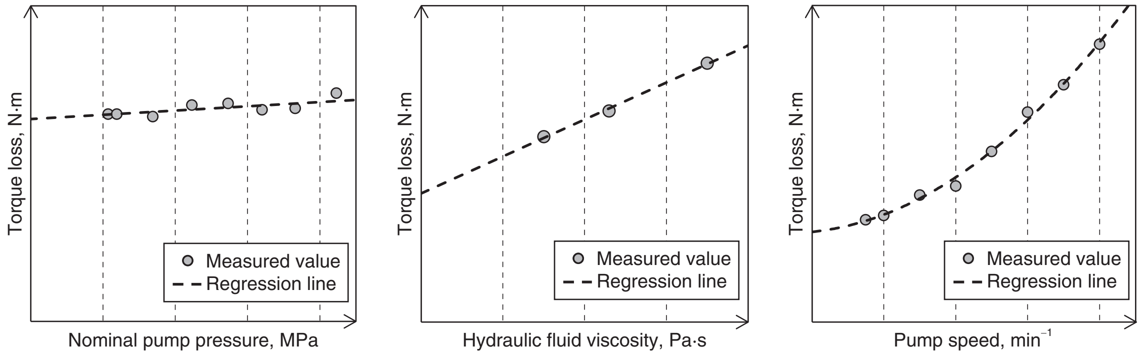


Fig. 4 Parameter characteristics of torque loss measurement values

friction of the fluid but rather the dissipation of kinetic energy, which is expressed by the following equation[5)]:

$$\Delta p = \zeta \cdot \frac{1}{2} \rho \cdot v^2 \tag{2}$$

where:
ζ = pressure loss coefficient,
ρ = fluid density, and
v = representative speed of the fluid flow.

3. 1. 4 Approximate Theoretical Equation of Torque Loss

The torque loss of the pump discussed in this paper results from the abovementioned factors that mutually affect each other as summarized in **Fig. 2**. It is therefore not easy to achieve an analytical solution of the torque loss through a detailed mathematical formula, or even to model the loss using a CAE simulation. The wedge effect that affects and is affected by gear behavior and the sliding between solids across fluid (with possible mixture of solid-solid contact within a tiny clearance) are the subjects of tribology and are of extraordinary complexity.

In consideration of the above, we propose the following simple theoretical equation that models the torque loss ΔT using pump operating parameters μ, ω, and p as well as coefficients a, b, and c obtained by approximation.

$$\Delta T = \frac{V}{2\pi} \Delta p + b \cdot \mu \cdot \omega + a \cdot (p + \Delta p) \tag{3}$$

where:
$\Delta p = c \cdot \omega^2$,
V = pump displacement per revolution,
p = nominal pump pressure,
Δp = pressure loss of the fluid,
μ = static viscosity of the fluid, which is dependent on temperature, and
ω = pump speed.

In the above equation (3), the first term on the right side denotes the fluid pressure loss, the second term the fluid shear force, and the third term the torque caused by friction due to solid-solid contact.

3. 2 Analysis of Experimentally Measured Values, Identification of the Rates of Contribution by Factor

Figure 4 shows the experimentally measured characteristics of the torque loss by parameter, which indicates the following:

- Torque loss linearly increases with regard to the fluid pressure and viscosity.
- Torque loss increases in a non-linear monotonic manner in relation to the pump speed, where the increment rate grows in line with the growth of the speed.
- All three characteristic curves have intercepts.

All of them are qualitatively consistent with equation (3).

To quantify the magnitude or rate of contribution of each torque loss factor incorporated in equation (3), we estimated the regression functions of the experimentally measured characteristics for each parameter. The degrees of functions were made to be the same as those of the theoretical equation, i.e., the degrees were one in the cases of fluid pressure and viscosity and two in the case of pump speed. Based on the above, we identified the constants of the functions and applied them to the coefficients in equation (3).

As a result, as shown in **Fig. 2**, it was determined that the fluid shear force accounted for almost half of the total contribution to the torque loss, and the fluid pressure loss was responsible for the other half.

Meanwhile, the torque loss by fluid shear force can be calculated by integrating the product of equation (1) and the distance of a point on the gear from the rotational center (r) over the clearance, if the tilting and radial displacement of the gears are ignored. As in **Fig. 2**, the fluid shear force on the outer gear outer circumference contributed the most (59%) among the three sliding surfaces, i.e., on the outer gear outer circumference, on the front and back of the outer gear, and on the front and back of the inner gear. It was also determined that the fluid shear force was about 4.5 times greater when gear tilting or radial displacement was present or when both were present. Here, the increase in the fluid shear force due to this accounted for 35% of the torque loss.

3. 3 Further Discussions of the Analysis in the Previous Section

The accuracy of the rates of contribution to torque loss by factors calculated as above depends on the accuracy of approximation in the modeling of the proposed theoretical equation (3).

In particular, there could be other factors that account for the nonlinearity observed in the measured characteristics between the torque loss and the pump speed in addition to the nonlinearity of the fluid pressure loss, which was incorporated into equation (3). The wedge effect, which arises when the clearance is narrowed under a slow pump speed and increases the fluid shear force, could be one such factor.

Therefore, the fluid shear force should be a little larger and the fluid pressure loss a little smaller than their contribution rates as calculated in the previous section.

4. Low-torque Pump with the Invented Structure

4. 1 Invented Structure

The analysis discussed in the earlier sections indicates that the fluid shear force is increased by certain gear behaviors and this is one of the major factors contributing to the torque loss of the pump. Now, we have invented a pump structure that reduces the outer gear's radial displacement, one of the identified behaviors.

Figure 5 shows the newly invented structure, which is patent pending. Two kinds of grooves are added on the inner surface of the housing. In **Fig. 5**, (1) marks the grooves added on the housing's inner circumference facing the outer gear outer circumference across a clearance, located on the suction and discharge sides. The grooves marked as (2) in **Fig. 5** are added on the housing facing the front and back of the gears across the clearances, and they are connected to the suction and discharge chambers. The fluid in the grooves therefore has the same pressure as the fluid in the chambers to which they are connected.

The circumferential lengths of the grooves (1) are designed so that the pressure on the outer gear outer circumference acts in the direction opposite to the direction of the pressure acting on the outer gear inner circumference, whereas the areas of pressurization are the same between the outer and inner circumferences of the gear. Through this design, the pressures acting on the gear are balanced in the radial direction and therefore radial displacement of the gear toward the housing's inner circumference during pump operation is greatly reduced.

Meanwhile, the grooves (1) are not created throughout the housing in the axial direction. Rather, a certain area of sliding surface is maintained in the middle of each groove. The aim of this design is to maintain the robustness against torque increase due to contact between one of the corner edges of the grooves and the outer gear outer circumference even if the gear is pressed against the housing inner circumference as a result of an incomplete balance between the radial pressures on the gear or an external disturbance such as the application of vibration.

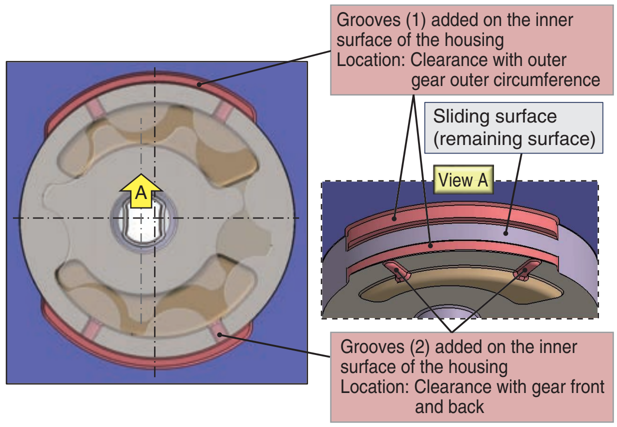


Fig.5 Structure of developed pump with additional grooves

4. 2 Test Results

For the target EOP with a higher flow rate, we selected a pump with suitable capacity and tested three samples with the same specifications and dimensions except for the presence or absence of the additional grooves under high fluid temperature and high pump speed conditions. **Table 1** summarizes the structures of the three samples, one of which has the newly invented structure. Through the test, we investigated the difference in the performance, i.e., the discharge flow rate and torque, between the pumps with the three structures.

As shown in **Fig. 6**, the test revealed that the torque of the pump with the newly invented structure was 28% less than the torque of the pump with the conventional structure without any additional grooves. This adequately fits the result of the aforementioned analysis and a calculation made based on it. The reduction in the discharge flow rate of the invented structure pump was about 3% compared to the flow rate delivered by the conventional structure pump, which was due to an increase in the pump's internal leakage as a result of the addition of the grooves.

The torque of the pump with the conventional structure with additional grooves (1) in **Table 1** was about 5% less than the torque of the conventional structure pump without any additional grooves. Here, the torque reduction was probably only due to the reduction in the area of sliding.

The above results suggest that the notable torque reduction observed with the invented structure pump was mostly because of the two kinds of additional grooves, which greatly prevented the outer gear from being radially displaced.

Table 1 Types of test samples

Sample	Structure (presence and type of the additional grooves)
Conventional structure	No additional groove on the inner surface of the housing (Has the same structure as the pump shown in **Figs. 1** and **2**)
Conventional structure with additional grooves (1)	Has only the additional grooves (1) of **Fig. 5** (on the housing's surface facing the outer gear outer circumference)
Invented structure	Has the additional grooves (1) and (2) of **Fig. 5**

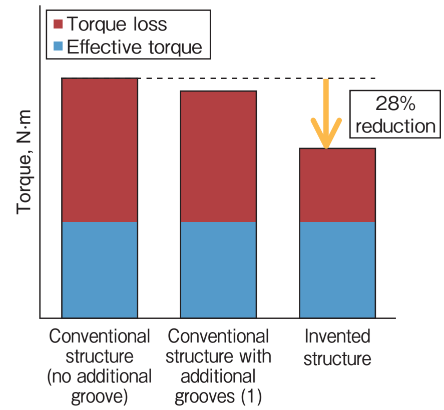


Fig. 6 Comparison of actual torque values for conventional and developed pumps

5. Conclusion

We invented a low-torque internal gear pump in order to reduce the electric consumption, size, and cost of EOPs. We analyzed the mechanism that caused the torque loss of the pump, proposed a simplified theoretical equation, analyzed experimental results, and successfully determined the hitherto unknown mechanism that explains the loss of pump torque as well as the degrees of contribution of each factor affecting the mechanism.

The results suggested that preventing radial displacement of the pump outer gear in the housing due to imbalances between the pressures acting on the gear was beneficial because this displacement increased the fluid shear force in the clearance on the gear's outer circumference. Based on this, we invented a structure where grooves were added in certain places on the inner surface of the housing (**Fig. 5**). We tested the structure under high fluid temperature and high pump speed conditions, and confirmed that the torque of the invented structure pump was notably reduced by 28% compared to that of a conventional pump.

Our next challenge is to apply the pump with the newly invented structure to EOPs, other vehicle parts, and other commercial products to reduce their size, weight, and energy consumption, and in so doing enhance the satisfaction of our customers and contribute to a decarbonized society.

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