

# Mathematical Analysis of Influence of Oil Temperature on Efficiencies in Hydraulic Pumps for Automatic Transmissions

Yoshiharu Inaguma and Naohito Yoshida  
JTEKT Corp

## ABSTRACT

This paper deals with a mathematical analysis of pump efficiencies in an internal gear and a balanced vane pumps. They are commonly used in current automatic transmissions including a continuously variable transmission. For these pumps, the influence of oil temperature as well as pump-operating pressures and pump speeds is clarified by using mathematical models taking oil temperature into consideration, which are constructed on the basis of actual measured flow and torque data under various pump-operating conditions. For fuel economy in a vehicle, because the pump should be operated under conditions to obtain higher pump efficiencies, it is very important to understand the relationship between the pump efficiencies and the pump-operating conditions. This paper reveals that the mathematical models representing the leakage flow and friction torque characteristics accurately is helpful to investigate the pump efficiencies as well as the flow and torque performance under various pump-operating conditions. As a result, the overall efficiency decreases at extremely low and high oil temperatures. In the internal gear and the balanced vane pumps tested in this work, the region with a higher overall efficiency lies around an oil temperature of 80°C.

**CITATION:** Inaguma, Y. and Yoshida, N., "Mathematical Analysis of Influence of Oil Temperature on Efficiencies in Hydraulic Pumps for Automatic Transmissions," *SAE Int. J. Passeng. Cars - Mech. Syst.* 6(2):2013, doi: 10.4271/2013-01-0820.

## INTRODUCTION

In an automatic transmission (AT) and a continuously variable transmission (CVT), an internal gear pump has conventionally been used [1,2,3,4,5]. These days, a balanced vane pump with a fixed-displacement is also used for CVT operated at the maximum pressure of 5 MPa because of its good volumetric efficiency [6,7,8]. Although a hydraulic pump is necessary as a hydraulic power source for controlling the pressure to change rotational speed of the output shaft and for lubrication, energy consumption of the pump becomes a significant loss factor in these transmissions [1,9]. Actually, the pump is operated across a broad range of oil temperatures as well as pressures and pump speeds in the vehicle. Because such pump-operating conditions affect significantly the pump performance, i.e. the flow and torque characteristics, the pump efficiencies also depend on the pump-operating conditions. Therefore, it is very important to understand how the pump efficiencies depend on the pump-operating conditions.

Although the volumetric, the mechanical and the overall efficiencies as well as the flow and the pump-driving torque characteristics for various pumps for AT and CVT of vehicles have already been investigated [10,11], they were

insufficiently analyzed. For the analysis of the pump efficiencies, a mathematical model representing the leakage flow or the friction torque characteristics of the pump is effective. Since Wilson constructed a mathematical model for the flow and the torque characteristics in hydraulic pumps and motors [12, 13], Schlösser [14] and Thoma [15] proposed their models based on the Wilson's model. However, their models were conceptual and have an insufficiency to represent accurately the actual characteristics of pump flow and torque. Hibi and Ichikawa [16], then, proposed an improved model for the friction torque characteristics with a non-linear change against pump speed, and the author proposed a model for the friction torque, which was based on the Hibi-Ichikawa model and took oil temperature into consideration [17,18,19]. Because the model for the pump flow proposed by Wilson, Schlösser or Thoma represented inaccurately the actual pump flow, the author proposed also a model for the leakage flow which took into consideration that the thickness of clearances changes according to pressures. And it was revealed that the model represents the flow characteristics accurately in various pumps [20]. In order to improve the efficiency of the transmission for fuel economy,

it would be one of the good methods for that the pump operates under a good condition of high pump efficiencies.

In this study, the effectiveness of calculation using the mathematical model is validated by comparing the calculated values with the experimental ones. And through the analysis, it is analyzed how the pump efficiencies depend on the pump-operation conditions, especially oil temperature.

## TEST PUMPS

In this study, two types of pumps, which are currently used for AT and CVT, were investigated. Pump A is an internal gear pump widely used for AT and CVT. [Figure 1](#) shows a cross-sectional view of an internal gear pump used for a transmission of AT or CVT, composed of an inner rotor with external gear teeth, an outer rotor with internal gear teeth, a body holding both the rotors, a cover and a stator shaft supporting the inner rotor via a bush bearing. The stator shaft is fixed to the cover and does not rotate. The bush bearing is inserted in the inner rotor and makes the rotation of the inner rotor on the stator shaft smooth without seizure under high pressure conditions. This construction contributes to an improvement in the volumetric efficiency by setting a small clearance between the tooth tips of the inner and outer rotors. The tooth forms of the inner and outer rotors are designed by combining two hypocycloidal curves. The inner rotor has one tooth less than the outer rotor and this pump has no crescent separator between tips of the inner rotor and the outer rotor. In comparison to the gerotor pump, the internal gear pump with this tooth profile has a larger displacement volume per gear tooth cavity and the radial dimensions of both the rotors in this pump can be smaller than those in the gerotor pump when the pump displacement and the width of both the rotors are the same. The reduction in the radial dimensions of the rotors in the internal gear pump leads to an improvement in the mechanical efficiency with a decrease in the viscous friction caused by shearing oil at the sides of the rotors. The dimensions of pump A are presented in [Figure 2](#).

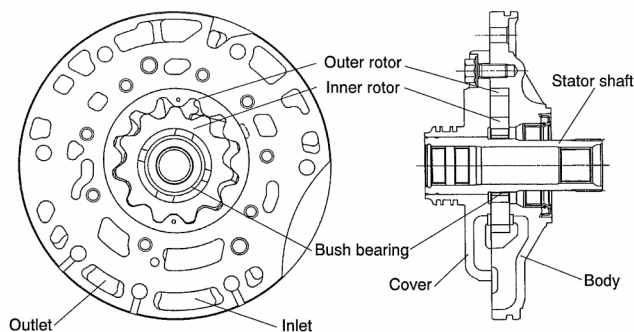


**Figure 1. Cross-sectional view of pump A (internal gear)**

Pump B is a balanced vane pump with a fixed-displacement. [Figure 3](#) shows a cross-sectional view of pump B, composed of a cam ring with an elliptic inner bore, a rotor with a series of radially disposed vanes, two side plates located on both sides of the rotor and a shaft. The rotor is

driven by the shaft through a very loose-fitting spline and rotates counterclockwise in [Figure 4](#). The rotor and the vanes rotate between the two side plates with running clearance. The balanced vane pump is designed so that both suction and delivery ports are oppositely located in the pump to balance radial forces acting on the shaft diametrically. In this pump, the side plates have vane back pressure grooves at the sides facing the rotor to introduce the delivery pressure to all the bottoms of the rotor vane slot. During the pump operation, the vanes are always pushed from the bottom by the delivery pressure and rotate on the cam contour with the loads imposed by the vane tip. The balanced vane pump has a double displacement process, with two cycles of suction/discharge per revolution. The dimensions of pump B are presented in [Figure 4](#).

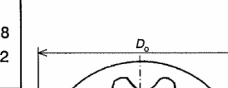
|                                                             |      |                                                                                     |
|-------------------------------------------------------------|------|-------------------------------------------------------------------------------------|
| Inner rotor with external gear                              |      |  |
| External diameter of external gear $D_{e1}$ (mm)            | 69.8 |                                                                                     |
| Internal diameter of external gear $D_{i1}$ (mm)            | 57.2 |                                                                                     |
| Number of teeth $z_1$                                       | 11   |                                                                                     |
| Outer rotor with internal gear                              |      |                                                                                     |
| External diameter of internal gear $D_{e2}$ (mm)            | 63.7 |                                                                                     |
| Internal diameter of internal gear $D_{i2}$ (mm)            | 76.3 |                                                                                     |
| Number of teeth $z_2$                                       | 12   |                                                                                     |
| Outer diameter $D_o$ (mm)                                   | 85.2 |                                                                                     |
| Width of rotors $b$ (mm)                                    | 11.0 |                                                                                     |
| Inner diameter of rotor bush $D_b$ (mm)                     | 31.5 |                                                                                     |
| Theoretical pump displacement $V_m$ (cm <sup>3</sup> /rev.) | 13.0 |                                                                                     |

**Figure 2. Dimensions of pump A (internal gear)**

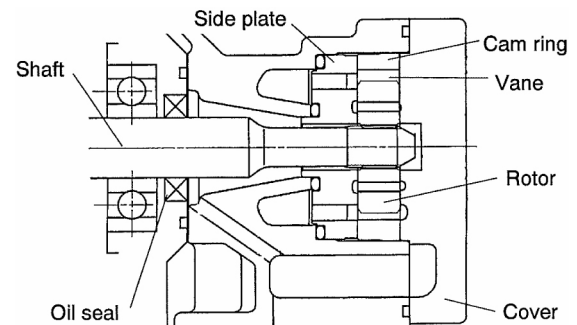


**Figure 3. Cross-sectional view of pump B (balanced vane)**

|                                                                |       |  |
|----------------------------------------------------------------|-------|--|
| Cam ring small radius $R_1$ (mm)                               | 18.85 |  |
| Cam ring large radius $R_2$ (mm)                               | 22.32 |  |
| Rotor radius $R_r$ (mm)                                        | 18.55 |  |
| Rotor width $b$ (mm)                                           | 12.0  |  |
| Vane thickness $w$ (mm)                                        | 1.40  |  |
| Number of vanes $z$                                            | 10    |  |
| Theoretical pump displacement $V_{in}$ (cm <sup>3</sup> /rev.) | 9.44  |  |

**Figure 4. Dimensions of pump B (balanced vane)**

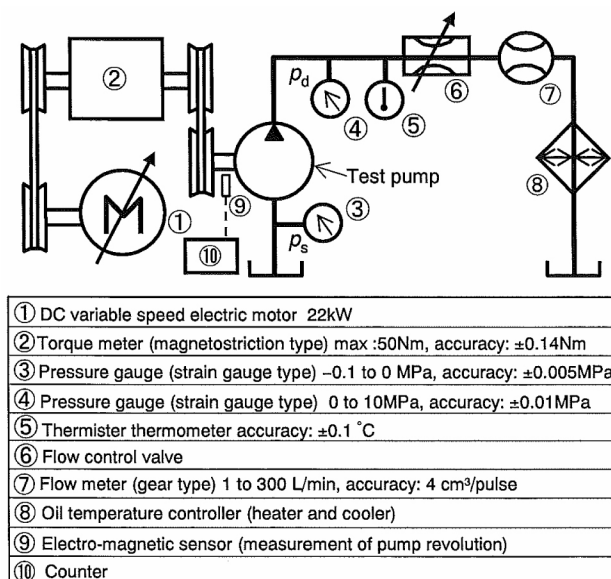


Figure 5. Experimental system

## EXPERIMENTAL APPARATUS

Figure 5 shows a schematic diagram of the experimental system for measuring flow rate out of the pump and pump driving torque. The test pump was driven by a variable speed electric motor via pulleys and a belt. In the experimental system, oil delivered from the test pump returned through a flow-control valve to control the delivery pressure, a flow meter and then an oil temperature controller to an oil reservoir. The delivery pressure,  $p_d$ , and suction pressure,  $p_s$ , were measured at the outlet and inlet of the pump, respectively. The pressure differential across the pump  $\Delta p$  was defined by subtracting  $p_s$  from  $p_d$ . The oil temperature  $\theta$  was measured at the pump outlet. The pump flow rate  $Q$  was measured twice with a gear type flow meter and the pump driving torque  $T$  was also measured twice with a magnetostriction type torque meter. In this study, by setting pump speed  $N$  and oil temperature  $\theta$  at a measuring point within  $\pm 10$  rpm and  $\pm 3^\circ\text{C}$ ,  $Q$  and  $T$  were measured in the process to increase  $p_d$  and to decrease it again. During the measurement of  $Q$ ,  $p_d$  was maintained within  $\pm 0.02$  MPa. After the measurement of  $Q$  for all pump speeds was completed,  $\theta$  was changed. The mean values of the two measured  $Q$  and  $T$  values were adopted as  $Q$  and  $T$  in this study. The hydraulic fluid is commercial mineral oil for CVT transmission, and its properties are given in Table 1.

Table 1. Properties of oil

|                           |        |        |        |        |        |        |        |
|---------------------------|--------|--------|--------|--------|--------|--------|--------|
| Temperature $\theta$ (°C) | 20     | 40     | 60     | 80     | 100    | 120    | 140    |
| Density $\rho$ (kg/m³)    | 843    | 834    | 822    | 810    | 798    | 786    | 774    |
| Viscosity $\mu$ (Pa·s)    | 0.0550 | 0.0254 | 0.0140 | 0.0085 | 0.0056 | 0.0040 | 0.0029 |

## EXPERIMENTAL RESULTS AND DISCUSSION

### Flow Characteristics and its Mathematical Model

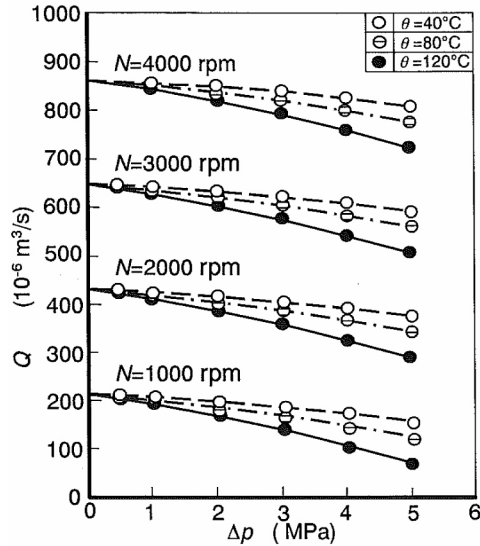
First, relationships between the pressure differential  $\Delta p$  and the flow rate  $Q$  for various pump speeds  $N$  and oil temperatures  $\theta$  in tested pumps are shown in Figure 6 (a) and (b). Throughout this study, the pump performance was investigated at three pressure differential conditions of 1, 2 and 5 MPa. In AT or CVT, the pump is operated at a delivery pressure of around 1MPa under usual driving conditions, and the maximum operating pressures of AT and CVT are 2 and 5 MPa, respectively. In both the pumps,  $Q$  decreases with an increase in  $\Delta p$  and the decreasing slope of  $Q$  against  $\Delta p$  becomes greater with rising of oil temperature  $\theta$ . In addition, for all the tested oil temperatures,  $Q$  increases by the product of the theoretical pump volume and the difference in  $N$  with an increase in  $N$ . Comparing both the pumps, the change in  $Q$  against  $\Delta p$  is linear in pump B shown in Figure 6(b), and is non-linear in pump A shown in Figure 6(a).

To investigate in detail, the leakage flow characteristics for these pumps are effective. The leakage flow  $\Delta Q$  is obtained by subtracting the actual pump flow rate  $Q$  from the ideal flow rate  $Q_{th}(=V_{th}N/60)$  as follows.

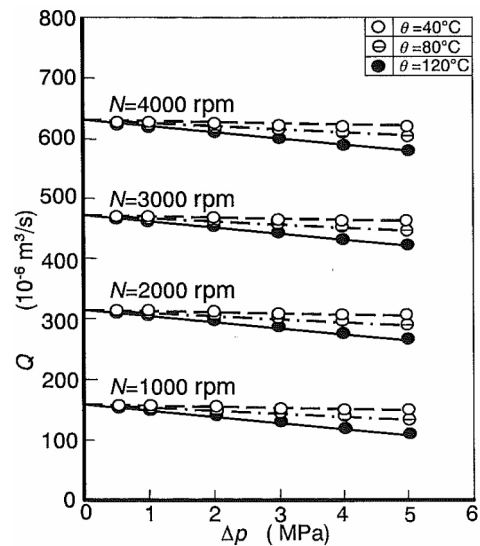
$$\Delta Q = Q_{th} - Q \quad (1)$$

As seen from Figure 7,  $N$  does not affect  $\Delta p$ - $\Delta Q$  characteristic for each temperature in both the pumps. Namely,  $\Delta Q$  is independent of  $N$  and depends on oil temperature  $\theta$  (the viscosity of oil  $\mu$ ) and  $\Delta p$ . In the case of pump A, the increasing rate of  $Q$  becomes greater with an increase in  $\Delta p$ . On the basis of the experimental data, a

mathematical model for the leakage flow characteristics will be introduced.



(a). Pump A (internal gear)



(b). Pump B (balanced vane)

Figure 6. Flow characteristics of test pumps

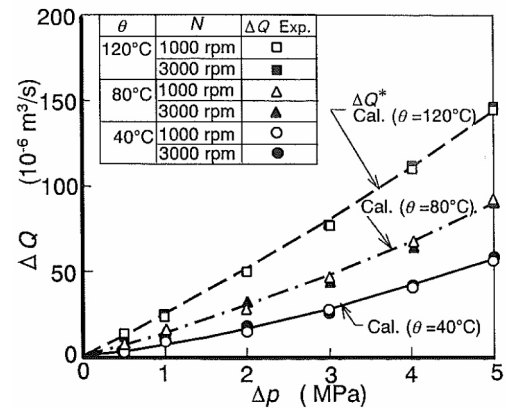
The leakage flow  $\Delta Q^*$  in the mathematical model for leakage flow proposed by Schlösser [14] is expressed by the following equation. In this study, because a loss flow due to cavitation is not taken into account, this effect was removed from the Schlösser's model.

$$\Delta Q^* = \frac{C_1}{\mu} \Delta p + \frac{C_c}{\sqrt{\rho}} \Delta p^{1/2} \quad (2)$$

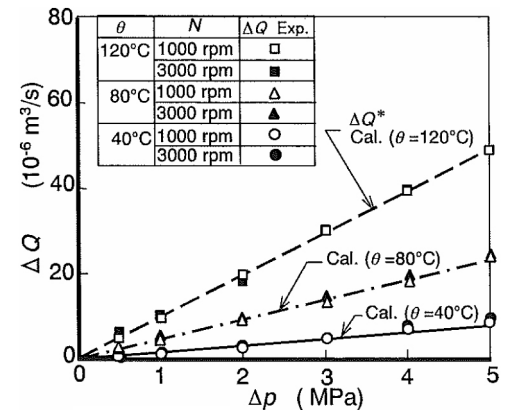
Removing the second term proportional to  $\Delta p^{1/2}$ , the equation becomes the model of Wilson [12,13] or Thoma [15]. In the model proposed by Wilson [12,13] and that by Thoma [15], it is considered that the leakage flow  $\Delta Q$  is

proportional to the pressure differential  $\Delta p$ . This concept cannot explain the  $\Delta p$ - $\Delta Q$  characteristics in pump A shown in Figure 7(a).

The Schlösser's model [14] added the component of the orifice flow leakage proportional to the square root of  $\Delta p$  to the Wilson's model, but cannot also explain the  $\Delta p$ - $\Delta Q$  characteristics of pump A, in which the increasing rate of the leakage flow against  $\Delta p$  becomes greater and greater with an increase in  $\Delta p$ .



(a). Pump A (internal gear)



(b). Pump B (balanced vane)

Figure 7. Leakage flow characteristics in test pumps

Therefore, the author proposed a new model for the leakage flow  $\Delta Q$ , in which the thickness of the clearance causing the leakage changed according to the pressure differential  $\Delta p$ . As a result, the model was able to represent the  $\Delta p$ - $\Delta Q$  characteristics under various pump-operating conditions, such as pressures, pump speeds and oil temperatures for various types of pumps. The model used is as follows:

$$\Delta Q^* = \frac{C_a}{\mu} \Delta p + \frac{C_b}{\mu} \Delta p^2 + \frac{C_c}{\sqrt{\rho}} \Delta p^{1/2} + \frac{C_d}{\sqrt{\rho}} \Delta p^{3/2} \quad (3)$$

where  $C_a$  to  $C_d$  are pump constants according to the exponent of  $\Delta p$ ,  $\mu$  is the viscosity of oil and  $\rho$  is the density of oil. The derivation of equation (3) is described in the reference [20].