

# Small High-Efficiency Vane Pump Based on Vane Pump Theory

Yoshiharu Inaguma and Naohito Yoshida  
JTEKT Corp.

## ABSTRACT

This article describes a design method for improving the mechanical efficiency in a small vane pump, which is driven by an electric motor with a battery for a conventional vehicle. An aim of designing is a decrease in friction torque without decreasing the volumetric efficiency. The influence of two important factors concerning the friction torque, i.e. the dimensions of pump parts such as a cam ring, a rotor and a vane, and the coefficient of friction between the cam contour and the vane tip on the mechanical efficiency is clarified in two stages. In the experimental analysis, the coefficient of friction between the vane and the cam contour in the actual pumps is estimated and the influence of the pump-operating conditions such as oil temperature is also investigated. Furthermore, the relationship between the coefficient of friction and the mechanical efficiency is demonstrated. Ultimately, through the two improvements, the newly designed small-vane pump with a displacement of 1.25 cm<sup>3</sup>/rev. for the electric motor drive has good volumetric and mechanical efficiencies.

**CITATION:** Inaguma, Y. and Yoshida, N., "Small High-Efficiency Vane Pump Based on Vane Pump Theory," *SAE Int. J. Passeng. Cars - Mech. Syst.* 8(2):2015, doi:10.4271/2015-01-1496.

## INTRODUCTION

An electric motor-driven oil pump was proposed for a vehicle having no space to install an engine-driven oil pump, a hybrid vehicle with an intermittent engine start and an electric vehicle. It has been already developed and used in a hydraulic power steering system [1,2,3,4,5,6] and in a drive-train system [7,8]. Also, for improving fuel-economy in a vehicle installing a Continuously Variable Transmission (CVT) or an Automatic Transmission (AT), a gerotor pump intermittently driven by the electric motor has been adopted to deliver oil during engine-stopping and keep an adequate oil pressure to engage a clutch for smooth starting [9,10,11].

The electric motor-driven pump system is required to be compact and usually operated using 12V battery similarly to other auxiliary parts. Therefore, its supply power is insufficient for a high hydraulic output and the hydraulic pump should be operated with great efficiency. Especially, a reduction of the friction torque in the pump is desired.

For the use at a low flow rate and high pressure condition such a power steering system, a balanced vane pump [1,2,3,5] or an external gear pump [4,6] is suitable to keep a high volumetric efficiency. Although the balanced vane pump is excellent in light-weight, low noise and vibration derived from pressure ripples in comparison with the external gear pump, it has a disadvantage in the mechanical efficiency due to its higher friction torque. Then, it would be very important to improve the mechanical efficiency in the vane pump through a reduction in the friction torque without aggravating the volumetric efficiency.

The friction torque of the vane pump caused by the friction between a cam contour and vane tip occupies a large part of the total friction torque because the vanes are always pushed to the cam contour by the delivery pressure. The vane theory for the mechanical efficiency introduced by Inaguma and Hibi, which was established through the work using vane pumps of 10.5 cm<sup>3</sup>/rev., clarified the essence of the mechanical efficiency using the parameter  $\varepsilon$  defined as a ratio of the cam lift to the vane thickness [12]. That is, the mechanical efficiency in the vane pump is greatly affected by  $\varepsilon$  and the friction between the cam contour and the vane tip. This theory was experimentally verified [12,13] and the coefficient of friction at the vane tip was investigated in detail using cylindrical rings without the cam lift as well [14]. This article describes a design method for improving the mechanical efficiency in a small vane pump with a displacement of 1.25 cm<sup>3</sup>/rev. according to the vane pump theory.

## FUNDAMENTAL FEATURE OF MOTOR-DRIVEN PUMP

### Construction of Motor-Driven Vane Pump

Figure 1 shows a cross-sectional view of an electric motor-driven oil pump. A balanced vane pump, composed of a cam ring with an elliptic inner bore, a rotor with a series of radially disposed vanes, two side plates with suction and delivery ports, is used in it. The side plates have vane back pressure grooves at the sides facing the rotor, and the delivery oil is introduced to all the bottoms of the rotor vane slot. Then, the vane tip always slides with the loads imposed by the delivery pressure on the cam contour.

The balanced vane pump is designed so that both suction and delivery ports are diametrically opposed to provide complete balance of all internal radial forces on a pump shaft. Even if the pump is operated at an extremely high pressure condition, no bending force acts on the shaft, resulting in only the consideration of the torsional strength with a small diameter of the pump shaft. Due to balancing the radial forces, the bearing or bushing can be eliminated in the vane pump and it contributes to compactness. The rotor of the vane pump can be directly driven by an electric motor shaft through a very loose-fitting spline without a coupling between the pump and the motor. This connecting formation is adopted in the motor-driven gerotor pump [10], in which an inner rotor is directly driven by the motor shaft, for the engine stopping system for a vehicle being stationary. In this case, the pump operates under an extremely low pressure less than 1 MPa condition. When the pump operates at a high pressure condition, the pumps other than the balanced vane pump could not adopt this driving configuration.

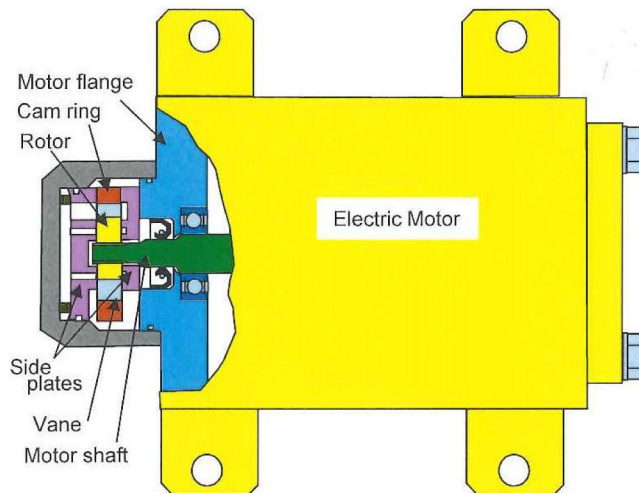


Figure 1. Structure of electric motor-driven vane pump

### Fundamental Characteristics of Electric Motor and Hydraulic Pump

Figure 2 shows fundamental characteristics of a DC electric motor and a hydraulic pump. Their characteristics are simply expressed by linear equations in this figure [15]. In Figure 2(a), the motor speed  $N$  decreases from  $N_0$  and the motor current  $I_m$  increases from  $I_0$  with an increase in the torque  $T$ . As shown in Figure 2(b), the pump speed at no load  $N_0$  increases with an increase in the motor input voltage  $E_m$  with the X-intercept  $E_0$ . In the hydraulic pump, the flow rate  $Q$  decreases from  $Q_0$  and the pump driving torque  $T$  increases from  $T_0$  with an increase in the pressure differential across the pump  $\Delta p$ , almost equal to the pump delivery pressure, as shown in Figure 2(c).

Hereupon, a significant peculiarity in an electric motor driven pump unit having the characteristics shown in Figure 2 should be brought into focus. Supposing that the improvement of the mechanical efficiency in the pump decreases  $T$  by  $dT$  as the dotted line in Figure 2(c) shows, the reduction by  $dT$  brings the increase in  $N$  by  $dN$  and the decrease in  $I_m$  by  $dI_m$  in Figure 2(a). When  $I_m$  decreases, the electric power loss in the electric wires decreases, preventing  $E_m$  at the electric motor input from falling. Because  $N$  depends also on  $E_m$ ,

the decrease in  $T$  in the pump leads to the increase in  $N$ , and  $Q$  increases at a high pressure condition without improving the volumetric efficiency in the pump. Thus the improvement in the mechanical efficiency, i.e. the reduction in the friction torque in the hydraulic pump is important because it increases the electric motor speed, resulting in the improving the volumetric efficiency in the pump and motor assembly.

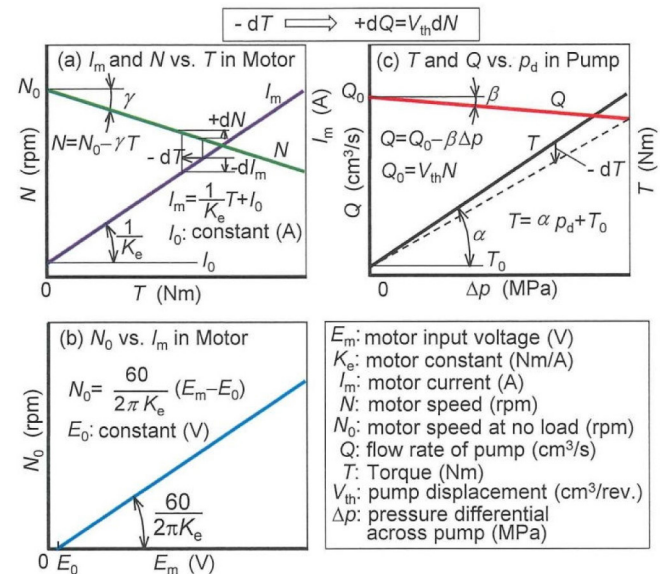


Figure 2. Fundamental characteristics of electric motor and hydraulic pump

### IMPROVEMENT IN MECHANICAL EFFICIENCY IN VANE PUMP

#### Dimensions of Vane Pump Parts for High Mechanical Efficiency

Improvement in the mechanical efficiency in a small displacement vane pump was conducted in two stages. At first, on the basis of the experimental results using various dimensions for basic vane pump parts such a cam ring, rotor and vane, it was investigated how the dimensions of these parts affected the friction torque characteristics. Figure 3 shows a cross-sectional view of the test vane pump having a pump shaft to drive the rotor. In this test, the pump was driven by an electric motor (22 kW) via pulleys and a belt. Considering the pump rotor is driven by an electric motor shaft in a pump and motor assembly shown in Figure 1, the friction torque concerning the pump shaft should be eliminated. Then, it will be eliminated in the experimental analysis of the pump friction torque.

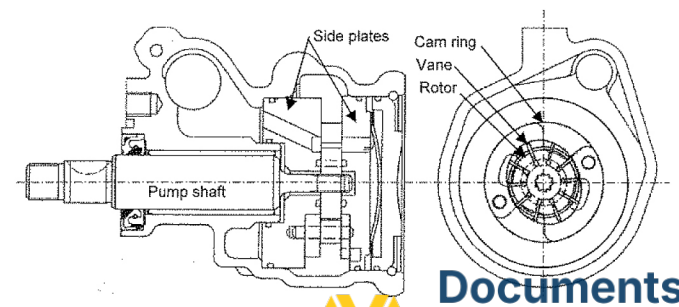


Figure 3. Cross-sectional view of test vane pump

In this test, four vane pumps A-D, each in a combination using two kinds of cam ring which differ in the size, four kinds of vane which differ in the thickness  $w$  and four rotors to match the vanes were prepared, and dimensions of these pumps are presented in Table 1. The cam rings, the rotors and the vanes were made of carburized and quenched ferro-sinterd alloy, quenched steel and quenched high-speed tool steel, respectively. The value of the surface roughness of the cam contour becomes important at the second stage. The inner surface of the two cam rings used in this test was finished by normal grinding to roughness of 1.097–1.603 and 1.061–1.528  $\mu\text{mRz}$ . The vanes were finished by barrel polishing after grinding to about 0.3  $\mu\text{mRz}$ . The rotor and the vanes rotate with running clearances, respectively. These clearances affect leakage flows, i.e. the volumetric efficiency. The values of these clearances are presented in Table 1. The hydraulic fluid used is commercial mineral oil and its properties are given in Table 2.

Table 1. Dimensions of pumps A-D

| Pump                                                                                    | A     | B     | C     | D     |
|-----------------------------------------------------------------------------------------|-------|-------|-------|-------|
| Cam ring small radius $R_1$ (mm)                                                        | 21.31 | 13.90 | 13.90 | 13.90 |
| Cam ring large radius $R_2$ (mm)                                                        | 20.00 | 12.30 | 12.30 | 12.30 |
| Rotor radius $R_r$ (mm)                                                                 | 19.50 | 11.80 | 11.80 | 11.80 |
| Rotor width $b$ (mm)                                                                    | 5.0   | 7.0   | 7.0   | 7.0   |
| Vane thickness $w$ (mm)                                                                 | 1.8   | 1.4   | 0.9   | 0.4   |
| Vane tip radius $R_v$ (mm)                                                              | 2.0   | 2.0   | 1.0   | 0.5   |
| Number of vanes $z$                                                                     | 10    | 10    | 10    | 10    |
| Pump displacement $V_{th}$ ( $\text{cm}^3/\text{rev}$ )                                 | 1.50  | 1.50  | 1.61  | 1.73  |
| Suction port angle (deg)                                                                | 51.0  | 52.0  | 53.0  | 54.0  |
| Delivery port angle (deg)                                                               | 51.0  | 52.0  | 53.0  | 54.0  |
| Rotor side clearance (difference of width between cam ring and rotor): 15 $\mu\text{m}$ |       |       |       |       |
| Rotor side clearance (difference of width between cam ring and rotor): 19 $\mu\text{m}$ |       |       |       |       |

Table 2. Properties of oil

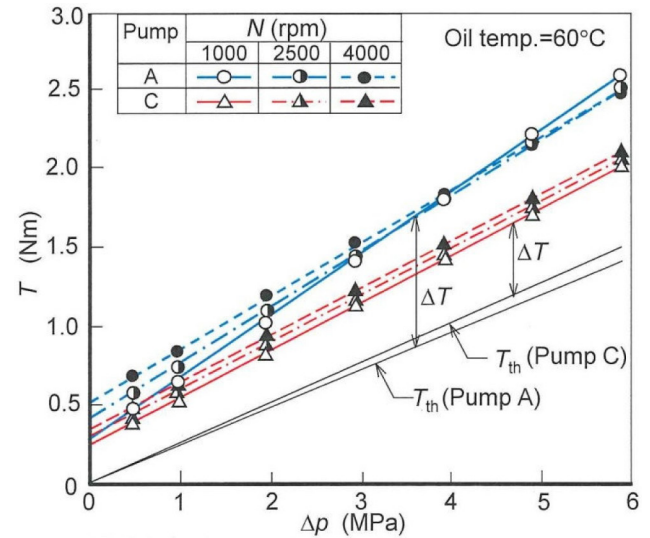
| Temperature ( $^{\circ}\text{C}$ )           | 40     | 60     | 80     |
|----------------------------------------------|--------|--------|--------|
| Density $\rho$ ( $\text{kg}/\text{m}^3$ )    | 855    | 842    | 829    |
| Viscosity $\mu$ ( $\text{Pa}\cdot\text{s}$ ) | 0.0293 | 0.0153 | 0.0089 |

Next, the experimental results are explained. Figure 4 shows the torque characteristics against the pressure differential  $\Delta p$  for pump A with a larger cam ring/rotor and pump C with a smaller cam ring/rotor as an example. Although pump A has a slightly smaller pump displacement  $V_{th}$  in comparison with pump C, its driving torque  $T$  is much higher. In Figure 4(a), the ideal torque  $T_{th}$  denoted by the following equation is indicated for each pump.

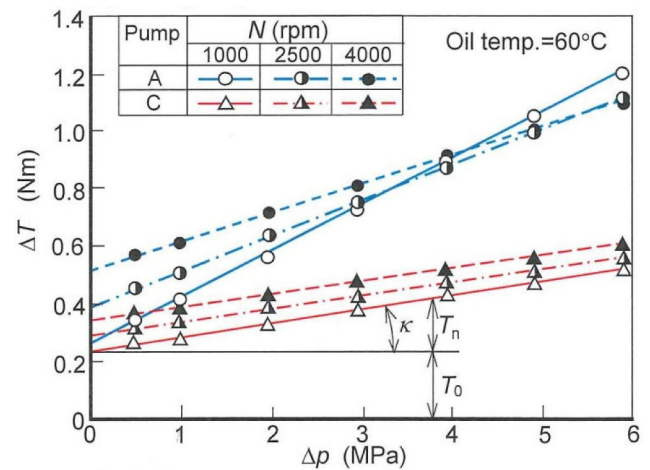
$$T_{th} = \frac{V_{th}}{2\pi} \Delta p \quad (1)$$

Comparing the friction torque  $\Delta T (=T - T_{th})$  as shown in Figure 4(b), the difference of the torque characteristics between pumps A and C can be intelligible. In both the pumps,  $\Delta T$  increases linearly with an increase in  $\Delta p$ , and  $\Delta T$  can be divided into  $T_0$  independent of  $\Delta p$  and  $T_n$  dependent on  $\Delta p$ . The values at  $\Delta p=0$ , i.e. the Y-intercept  $T_0$  is considered to be caused by the friction of the pump shaft and the viscous friction

between the rotor and the side plate and so on. Pump A has a distinct increase in  $T_0$  with increasing  $N$  due to the greater viscous friction using the cam ring and rotor with a larger diameter. This fact suggests that the friction torque dependent on  $N$  could be decreased by downsizing radially the inner radius of the cam ring and the rotor.



(a). Driving torque characteristics



(b). Friction torque characteristics

Figure 4. Torque characteristics of pumps A and C

In contrast,  $T_n$  has a slope  $\kappa$  against  $\Delta p$  which is smaller with an increase in the pump speed  $N$ . The component  $T_n$  is considered to be caused by the friction between the cam contour and the vane tip because the friction force at the vane tip is proportional to  $\Delta p$ . In the case of these test pumps, because the angle of the suction part is equal to that of the delivery part, the average vane force caused by the pressure difference becomes  $bw\Delta p/2$  per vane. Assuming that the average cam radius is  $(R_2 + R_1)/2$  and considering the number of vanes  $z$  and the coefficient of friction  $\lambda$  between the cam contour and the vane tip,  $\kappa$  is denoted as  $z\lambda wb(R_2 + R_1)/4$  [12]. In pump A having thicker vanes, the values of  $\kappa$  are much greater than those in pump C, and  $T_n$  occupies a large part of  $\Delta T$  at a high pressure condition.