

Research of Hydrostatic CVT for Passenger Vehicles

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ABSTRACT

Hydraulic Continuously Variable Transmissions (“CVT”) have a large capacity for power transmission and have achieved proven results in industrial machines. Research was conducted on hydraulic CVT for use in passenger cars with large engine displacement, particularly in the areas of fuel economy performance, high-speed performance and quietness where passenger cars have higher demands compared to industrial machines. An HMT (Hydro Mechanical Transmission), which has the highest efficiency of all hydraulic CVT, was adopted to improve efficiency at cruising which greatly affects fuel economy when cruising at high speeds. In addition, the lock-up brake mechanism and the hydraulic pump and motor portions were improved to reduce agitation, resulting in a 6% improvement in efficiency when cruising at 50 mph. The hydraulic motor rotates at high speed during high-speed cruising, but centrifugal force has little effect and prospects for durability were obtained by adopting a bowl type slipper and improving the cooling method. A method for reducing noise was also derived by focusing on fluctuations of swash plate movement, which are correlated with noise.

1. Introduction

Continuously variable transmissions (CVT) have gained attention as an attractive technology for increasing fuel efficiency. Since Honda began mass production of a CVT for the Civic in 1995, its use has been expanded into many other models. The basic structure of this CVT is a metallic belt running between two pulleys to provide a continuously variable set of speed ratios. At present, however, there are limits to the torque which can be carried by this transmission, and it has been a challenge to apply it to large-displacement engines.

Oil hydraulics CVT consisting of hydraulic pumps and motors are widely used in industrial machinery. These provide high torque transmission, but they have several issues preventing their application to passenger automobiles (cars): their efficiency is low, they cannot withstand high-speed operation, and they are very noisy.

Honda has had an interest in overcoming these issues and has continued development of oil hydraulics CVT. Products have been developed for motorcycles and other small vehicles. Honda introduced the JUNO scooter in 1962, a lawn mower in 1989⁽¹⁾ and, more recently, in 2000, a series of all-terrain vehicles⁽²⁾. These technologies were used as a basis for the present research on automotive oil hydraulics CVT.

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2. Aim of Research

The aim for this research was application of a CVT for cars with large-displacement engines. Thus, it was assumed that the transmission would be mounted on cars of the currently dominant front-wheel drive (FFD) design featuring transverse-mounted engines. Also, in addition to the requirement of fuel efficiency, which grows with engine size, higher operating speed and lower noise level are also essential; therefore, the following points were emphasized in this research:

- (1) Technologies for higher efficiency;
- (2) Technologies for higher rotational speed;
- (3) Technologies for less noise.

3. Technologies for Higher Efficiency

CVT have been anticipated to offer higher fuel economy than staged automatic transmissions (AT). Due to its wide range of speed ratios, the CVT can be set to allow the engine to operate continually in its range of peak thermodynamic efficiency, despite the high frequency of acceleration and deceleration and the low speeds usually encountered in city driving. In suburban and freeway driving, with their lower frequency of acceleration and deceleration but high speeds, the CVT is again at an advantage due to its higher top-end speed ratio, which allows lower engine speeds. Both features are expected to improve fuel economy. However, AT have recently been equipped with a greater number of speed ratios; some cars with large-displacement engines now have 6-speed AT. Although CVT still offer the capability of operating the engine in the range of peak thermodynamic efficiency, their advantage in ratio range has been eroded. Therefore, it is important to improve the cruise efficiency at the top ratio in order to improve fuel mileage during high-speed driving.

3.1. HMT

Hydrostatic transmissions (HST), which convert all the power of the driving engine to hydraulic power, are commonly used for industrial equipment due to their simplicity of operation and installation. Honda's lawn mower used this type⁽¹⁾. However, HST have a low power transmission efficiency, so are unsuited to automotive use.

Hydro-mechanical transmissions (HMT) are one method of raising the efficiency of oil hydraulics CVT (see Fig. 1). Power from the engine is split; an efficient mechanical connection is used to transmit some of the power from the engine, while the rest of the power is carried by an inefficient but continuously variable hydraulic transmission, and the two power streams are combined at the output for a high-efficiency, continuously variable speed output. HMT were used in the JUNO scooter and ATVs mentioned above to obtain higher efficiency⁽²⁾.

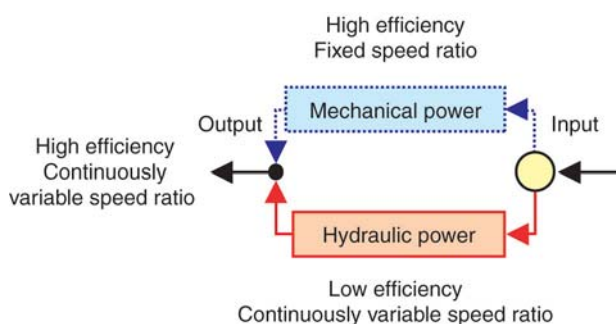


Fig. 1 Concept of HMT

3.1.1. Structure

The highly efficient HMT was used as the CVT in this study since it would provide the high fuel efficiency demanded of cars. There are many configurations of HMT; the configuration in Fig. 2 was chosen for its adaptability to the transverse engine in a FFD car. Here, the oil hydraulic pump lies parallel to the oil hydraulic motor. Both the pump and the motor are the variable displacement, axial-piston swash plate type; the angle of the swash plate can be adjusted and set to change the suction and discharge volumes. The power divider mechanism uses double sun planetary gears, which allow for a wide range of ratios.

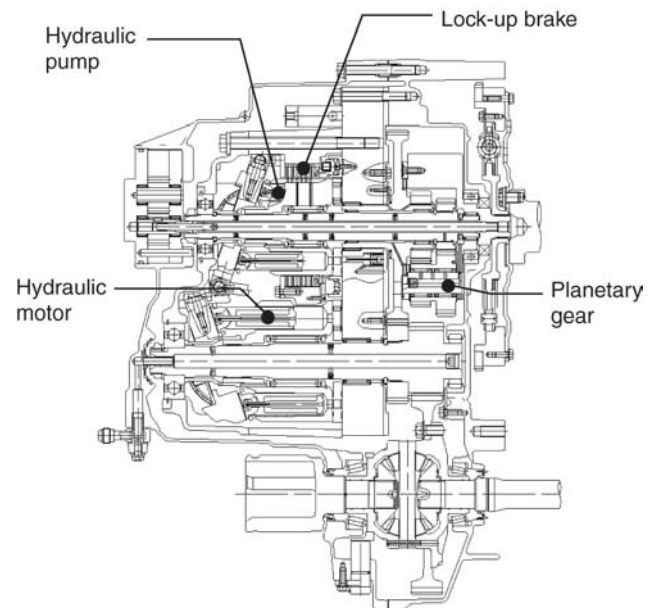


Fig. 2 Section of HMT

3.1.2. Power flow path

Fig. 3 shows the path by which power is transmitted through the HMT. Power enters from the engine to the carrier of a planetary gear, and is divided by two sets of pinion gears and sun gears into hydraulic power and mechanical power. Each path is shown below.

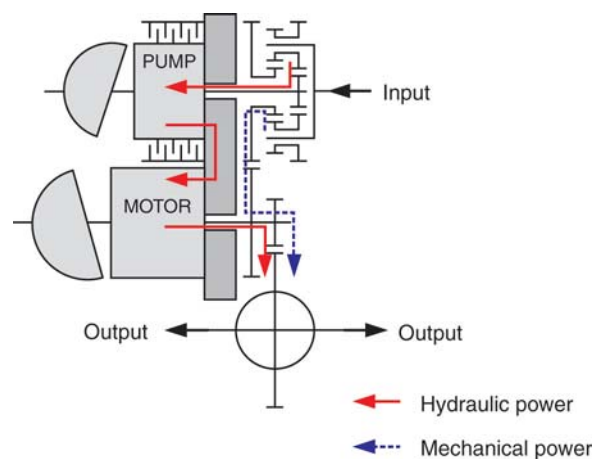


Fig. 3 Power transfer path of HMT

(1) Hydraulic power

Power passes through a large pinion gear and a small sun gear to an oil pump. Here, it is converted to oil pressure and flow, which is transmitted through passages to the hydraulic motor, and from that to the motor shaft.

(2) Mechanical power

Power passes through a small pinion gear, a large sun gear and a pair of gears to the motor shaft.

Thus, the two separate power flows are re-united at the motor shaft and are output from there through the differential to the car's wheels.

3.1.3. Principles of operation

The variation in the swash plate angle ratio of the pump and the motor with respect to the output:input speed ratio of the HMT (I/O speed ratio) is shown in Fig. 4, and the ratios of motor and pump speeds with respect to the same parameter are given in Fig. 5. The 'swash plate angle ratio' is defined as the dimensionless ratio of the maximum angles of each swash plate. The 'rotation speed ratio of pump and motor' is defined thus: for the pump, it is the dimensionless pump speed when the I/O speed ratio is zero; for the motor, it is the dimensionless maximum motor speed. Fig. 5 shows the case when the input speed of the HMT is held constant. The operating principles of the HMT will be explained by relating these figures to vehicle motion.

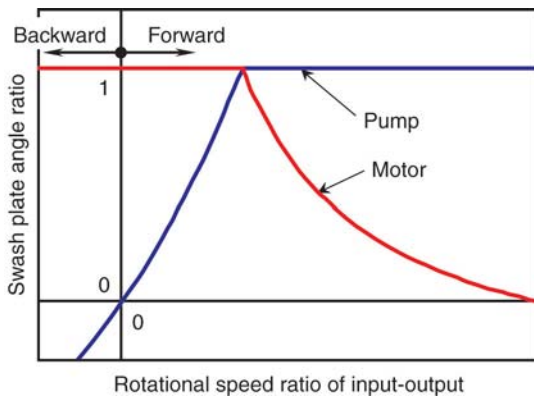


Fig. 4 Swash plate angle ratio of pump and motor

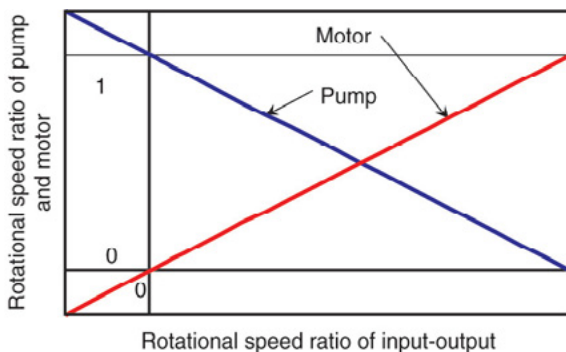


Fig. 5 Rotational speed ratio of pump and motor

(1) Vehicle is stopped

When the I/O speed ratio between input and output speeds is zero, the motor swash plate angle is set at its maximum value and the suction volume is maximized. Meanwhile, the pump swash plate is set at 0° and the discharge volume is 0. Thus, even when the pump is turning, there is no discharge, and the motor does not turn, so the vehicle wheels that are connected to the motor do not turn.

(2) Vehicle starts

As the I/O speed ratio is increased from zero, the motor swash plate angle remains at a maximum and the pump swash plate angle increases from 0° , so that oil is discharged from the pump. This oil causes the hydraulic motor to turn, causing the wheels to turn and setting the vehicle in motion. Then, the pump rotational speed begins to diminish. Once the pump swash plate angle reaches its maximum value, the motor swash plate angle is reduced. As the pump rotational speed diminishes, the motor rotational speed increases, increasing the vehicle speed. Once the I/O speed ratio reaches a maximum, the motor swash plate angle is 0° ; since the motor suction volume is now zero, the pump is no longer able to discharge fluid, and it stops rotating. Here, the fluid power transfer is zero, and all power transfer is mechanical.

(3) Reverse

As the I/O speed ratio is decreased from zero, the motor swash plate angle remains at a maximum and the pump swash plate angle is set from 0° to a negative angle. In this situation, the discharge direction is the opposite of that of the forward direction, causing the motor to revolve in the reverse direction, and the vehicle moves in reverse.

3.2. Lock-up Brake

The condition where the I/O speed ratio is maximum (i.e., at the top gear ratio) is the most commonly used condition in cars, so the efficiency at this ratio is critical to obtaining good fuel mileage. The pump is stopped and power is only transmitted mechanically, so this system is more efficient than a HST. However, in this condition, even though the pump is stopped and the oil is not flowing, the oil is still pressurized. There is oil leakage and frictional losses at sliding parts of the motor (between the slipper and swash plate, between the cylinders and valve plate, etc.) Thus, this system incorporated the lock-up brake shown in Fig. 2 in order to further improve efficiency.

Once the pump has been stopped after the I/O speed ratio is maximized, the lock-up brake is activated, fixing the pump cylinders and case. Then, the torque which was acting on the pump shaft is borne by the case. This stops the pressurizing of the oil and further improves efficiency. Fig. 6 shows the variation of efficiency with input torque at a speed of 2,000 rpm. The effect of the lock-up brake improves with increasing torque, resulting in a maximum 8% increase in efficiency.

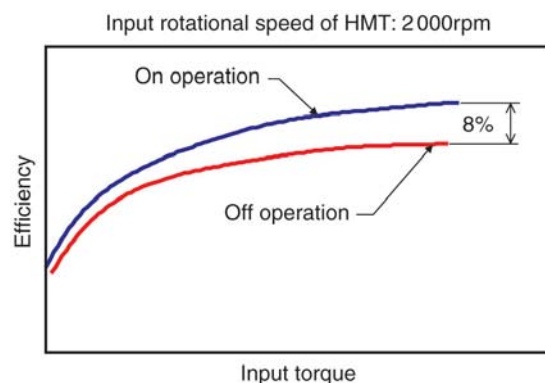


Fig. 6 Effect of lock-up brake

3.3. Part Lubrication

Most oil hydraulic pumps and motors are operated inside a case filled with oil in order to use the oil itself for lubrication and cooling. This design results in considerable agitation loss, however, so a new design was developed which supplies oil only to the necessary locations. The pump and motor are the same structure. As shown in Fig. 7, oil was supplied from the shaft directly to the sliding parts (between the slipper and swash plate, between the cylinders and valve plate, etc.)

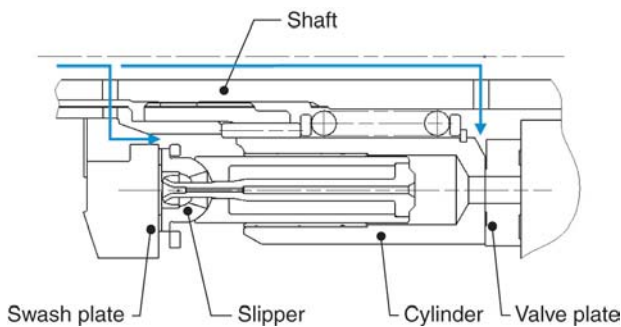


Fig. 7 Part lubrication

Fig. 8 shows the benefit of part lubrication for oil agitation loss in the motor alone. The higher the speed, the greater the benefit.

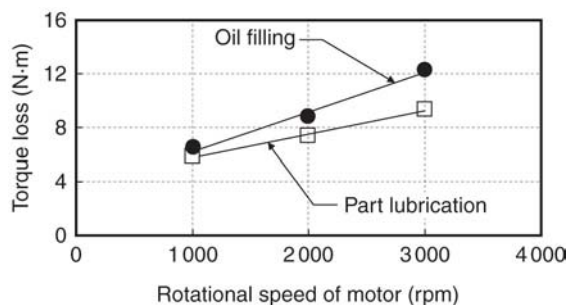


Fig. 8 Agitation loss

3.4. Cruise Effect

The effects of the lock-up brake and part lubrication on cruise efficiency are shown in Fig. 9. At 50 mph, the cruise efficiency was improved by 6% due to the two features.

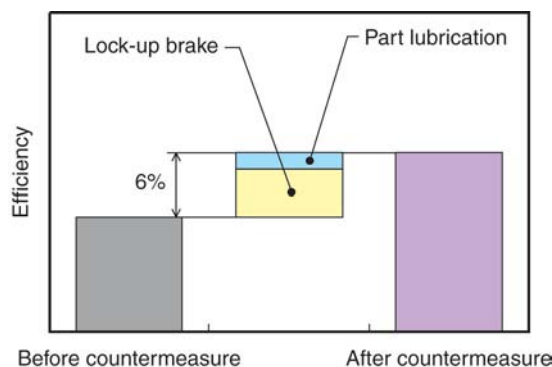


Fig. 9 Efficiency when cruising (50 mph)

4. High-speed Technology

In Fig. 10, the rotational speed of the pump and motor are shown when the accelerator is opened all the way. Initially, the pump speed rises with vehicle speed, then reaches a maximum and falls off. Meanwhile, the motor speed rises in direct proportion to vehicle speed and is maximum at top speed. The actual top motor speed depends on the engine driving the HMT and the vehicle itself. In the sample used for this study, the top pump speed was on a par with that in industrial equipment, 3,500 rpm, but the top motor speed was 7,000 rpm. Thus, technology is needed for handling high rotational speeds.

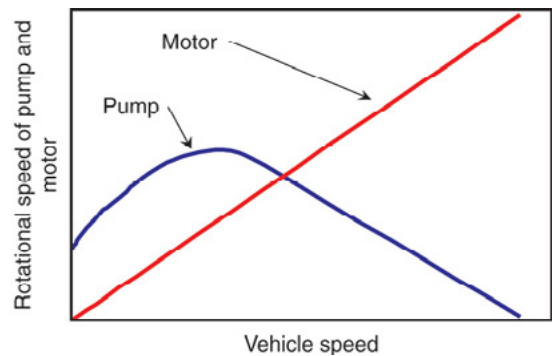


Fig. 10 Rotational speed of pump and motor

4.1. Bowl Type Slippers

At the tips of the pistons in axial swash plate piston pumps and motors are hydrostatic pressure bearings called 'slippers'. The joint between a slipper and a piston has a spherical shape in order to allow the piston to follow the swash plate. There are many configurations for this joint⁽³⁾, but the caulking type and with ball type shown in Fig. 11 are common.

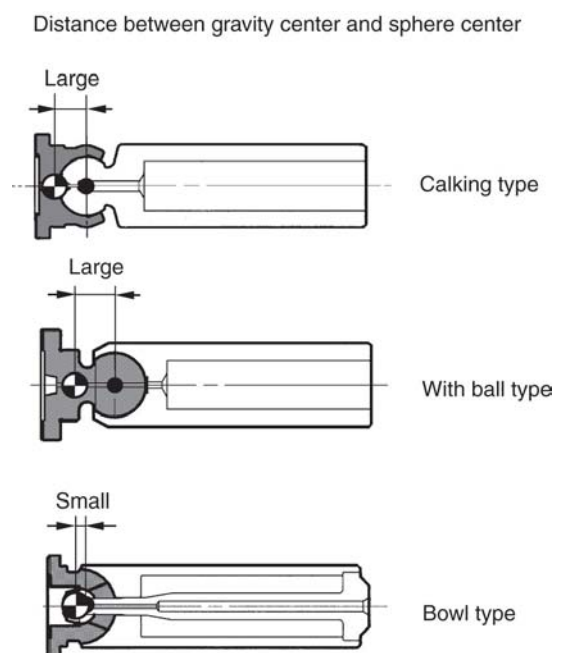


Fig. 11 Type of slipper

However, in the upper two designs in Fig. 11, the center of mass of the slipper is far from the center of the sphere in the joint. A large moment is developed about the center of the sphere when the motor is operated at high speeds, due to the centrifugal force on the mass center. This moment tends to tip the sliding face of the slipper out of contact with the swash plate. This causes an abrupt increase in oil leakage in conventional slippers when they are operated above a certain speed, as shown in Fig. 12. Not only is leakage increased by tipping of the sliding part, the uneven contact gives rise to wear and seizing, greatly lowering durability.

The larger the displacement of the motor, the greater the distance between the motor axis and the slipper, and the greater of the mass of the slipper, magnifying the centrifugal force on the slipper and the concerns with durability. Adopting a 'bowl type slipper' resolved these issues. The center of mass is very close to the center of gravity in this slipper.

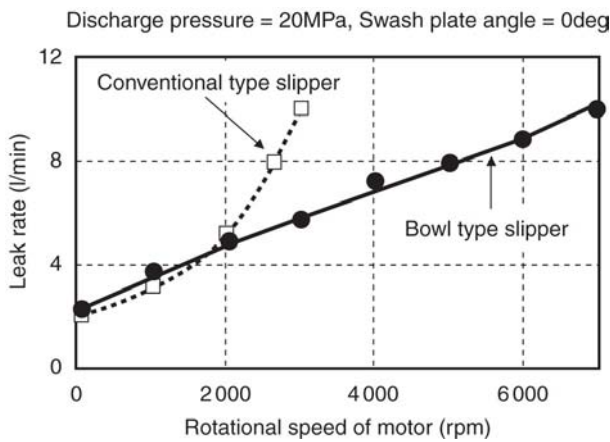


Fig. 12 Leak rate of motor

4.2. Improving Part Lubrication

Part lubrication, mentioned in Section 3.3 as lessening agitation loss, was further refined in order to augment durability at high operating speeds. As shown in Fig. 13, the sliding part is divided from the swash plate. Oil is introduced via the swash plate (not shown) through the back of the sliding component in order to cool them, and then, to the slipper. An oil filling area was created using an oil guide, which ensures that the volume between the slippers and the sliding component is always full of oil. An oil guide was also designed into the cylinder-valve plate region to create an oil filling area there.

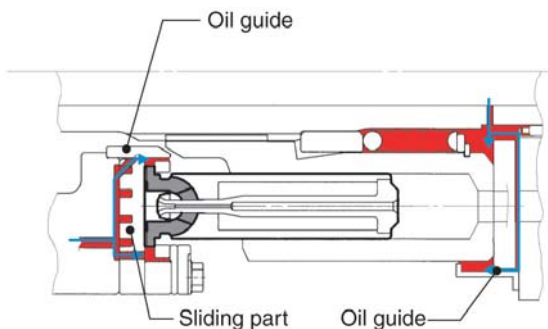


Fig. 13 Improvement of part lubrication

4.3. Results in Durability

The high-speed technologies described above were used in a 160 cc/rev displacement motor to provide high durability at 7,000 rpm (swash plate angle 0°). These results point to application of these technologies in cars with 4-L class engines.

5. Low Noise Technologies

One of the issues in adopting oil hydraulic CVT is the noise from piston pumps and motors. There has been vigorous research into reducing noise from piston pumps for many years⁽⁴⁾. It has been reported that swash plate vibration due to axial-direction thrust loads from the pistons on the swash plate accounts for a large part of the noise from pumps and motors⁽⁵⁾. Experiments by the authors have shown that transfer of swash plate vibration into the cabin is the main factor in noise. However, progress in reducing noise has been thwarted by the nearly unpredictable variations in pressure and flow rate that arise during pumping. Therefore, the authors estimated the swash plate vibration level at the design stage and used those results to modify the specifications in order to reduce swash plate vibration. Experiments were then performed to verify the results.

5.1. Noise Generation Mechanisms

The main cause of swash plate vibration is the violent fluctuations of the pressure in the cylinders (cylinder pressure) in pumps and motors due to pumping, which bring about violent fluctuations in the moment and thrust load on the swash plate. As shown in Fig. 14, the vibrations transfer from the case to the mounts to the vehicle frame, appearing eventually as noise in the cabin.

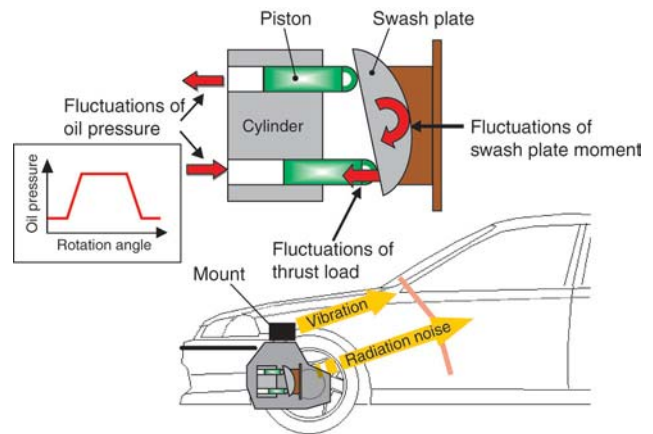


Fig. 14 Mechanism of noise

It is well known that reducing the displacement at the transmission mounting point (hereafter "Mt point") lowers the noise in the cabin. These experiments established a correlation between displacement at the Mt point and fluctuations in moment on the swash plate, as shown in Fig. 15. On the basis of this finding, research was continued with the objective of reducing the fluctuations of the moment on the swash plate.

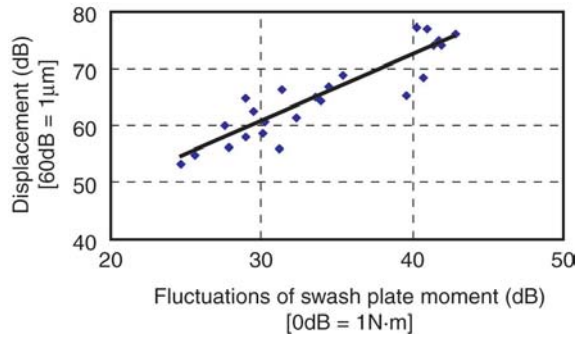


Fig. 15 Relation between fluctuations of swash plate moment and mounting point displacement

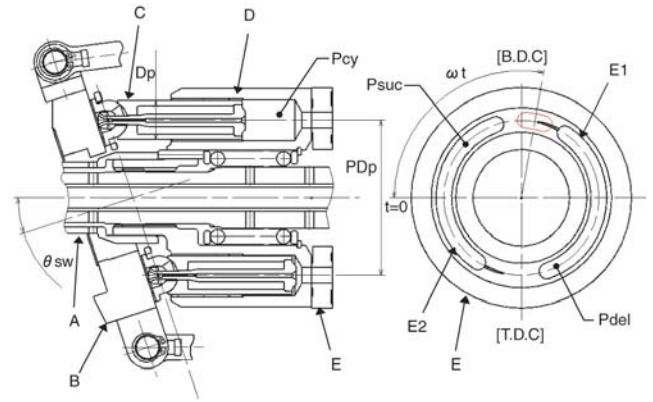


Fig. 16 Pump structure

5.2. Calculation of Cylinder Pressure and Moment on Swash Plate

Because the pressures developed in variable displacement pumps vary with the discharge volume even if the input torque is constant, it has been more difficult to establish an effective method for reducing noise from these motors under the full range of operating conditions than in the case of motors whose pressures do not vary. There is an established method of calculating the cylinder pressure and the moment on the swash plate from the specifications of the pump⁽⁶⁾. That method is explained in detail in the following section.

5.2.1. Nomenclature and structure

Fig. 16 shows the structure of the pump. Here, when axis A turns, the swash plate B vibrates, causing a reciprocating motion of piston C and its cylinder D, and, in turn, pumping the oil through the mechanism. The valve plate E consists of two parts, the discharge port E1 and the suction port E2. During the discharge process of the pump, the cylinder ports are aligned with the discharge port E1, and during the suction process, they are aligned with the suction port E2.

The symbols have the following meanings:

- Pcy: Cylinder pressure
- Pdel: Discharge pressure
- Psuc: Suction pressure
- V: Volume of oil in cylinder
- V0: Volume of oil in cylinder at t=0
- V1: Fluctuation in oil volume due to piston stroke
- V2: Volume of oil entering cylinder at discharge port
- V3: Volume of oil entering cylinder at suction port
- V4: Volume of oil leaking from cylinder
- Dp: Piston diameter
- PDp: Piston pitch circle diameter
- θsw: Swash plate angle
- Cdel: Flow coefficient for passage connecting discharge port and cylinder
- Csuc: Flow coefficient for passage connecting suction port and cylinder
- Adel: Area of restrictor in passage connecting discharge port and cylinder
- Asuc: Area of restrictor in passage connecting suction port and cylinder
- g: Gravitational acceleration
- γ: Oil specific gravity
- β: Oil compressibility
- ω: Angular speed of pistons
- t: Time

- Fcyk: Thrust of kth piston
- αk: Angle of kth piston
- Pcyk: Cylinder pressure of kth piston
- α0: Angle of piston #1 at t=0
- n: Number of pistons

5.2.2 Equations

The change in cylinder pressure dPcy is found from the following equations:

$$dP_{cy} = \frac{1}{V\beta} (-dV1 + dV2 + dV3 - dV4)$$

$$dV1 = - \frac{Dp^2 \pi \cdot PDp \cdot \omega \cdot \tan \theta_{sw} \cdot \cos \omega t}{8} dt$$

$$dV2 = \pm C_{del} \cdot A_{del} \cdot \sqrt{\frac{2g}{\gamma} |P_{del} - P_{cy}|} \cdot dt$$

(Right side is positive when $P_{del} \geq P_{cy}$, negative otherwise)

$$dV3 = \pm C_{suc} \cdot A_{suc} \cdot \sqrt{\frac{2g}{\gamma} |P_{suc} - P_{cy}|} \cdot dt$$

(Right side is positive when $P_{suc} \geq P_{cy}$, negative otherwise)

$$V = V0 + \frac{Dp^2 \cdot \pi \cdot PDp \cdot \tan \theta_{sw} \cdot \sin \omega t}{8}$$

The moment (M) acting on the swash plate around its rotational axis is given by:

$$M = \sum_{k=1}^n \frac{F_{cyk} \cdot PDp \cdot \sin \alpha_k \cdot \sec^2 \theta_{sw}}{2}$$

$$F_{cyk} = \frac{PDp^2 \cdot \pi \cdot P_{cyk}}{4}$$

$$\alpha_k = \frac{2\pi(k-1)}{n} + \alpha_k \cdot t + \alpha_0$$

Fig. 17 provides a comparison of the calculated pressures and the actual measured pressures in a cylinder of a pump with the basic specifications; the two curves matched quite well. There was also a good match of the pressure variations during the θ_1 range, over which the pressure was building to the maximum, and during the θ_2 range, over which it was falling to the minimum. Fig. 18 shows the calculated fluctuations of moment on the swash plate using calculated cylinder pressure and actually measured values of the moment. The peaks at each order frequency matched well, allowing an estimate of the noise level of improved pump designs.

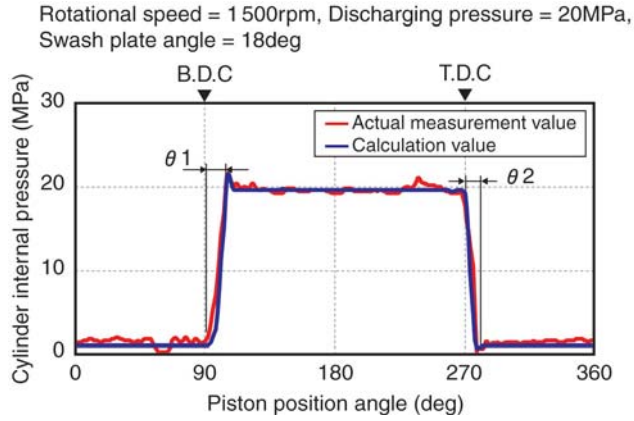


Fig. 17 Cylinder internal pressure

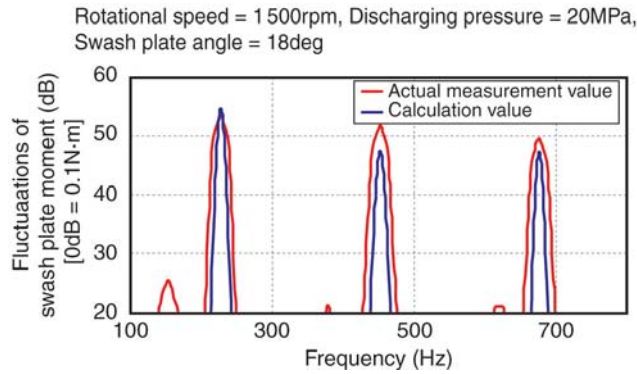


Fig. 18 Fluctuations of swash plate moment

5.3. Methods for Reducing Noise

Methods for reducing pump noise were derived using the equations just explained above and the results of experiments with those designs are described here. Fig. 19 shows the experimental device. The cylinder pressure was measured with a pressure sensor attached to the cylinder. The moment applied to the swash plate was calculated by measuring the loads on the two piston rods and converting these values.

5.3.1. Optimization of cylinder pressure

Fig. 20 shows the relation of the θ of the cylinder pressure θ ($= \theta_1 = \theta_2$) and fluctuations of moment on the swash plate in a 9-piston pump. This figure makes it plain that, if θ is 20° or 40° , the swash plate moment fluctuations will be small. There are two particular values of θ_1 and θ_2 , which result in low swash plate moment fluctuation. These angles were optimized to take advantage of this tendency of the swash plate moment fluctuation.

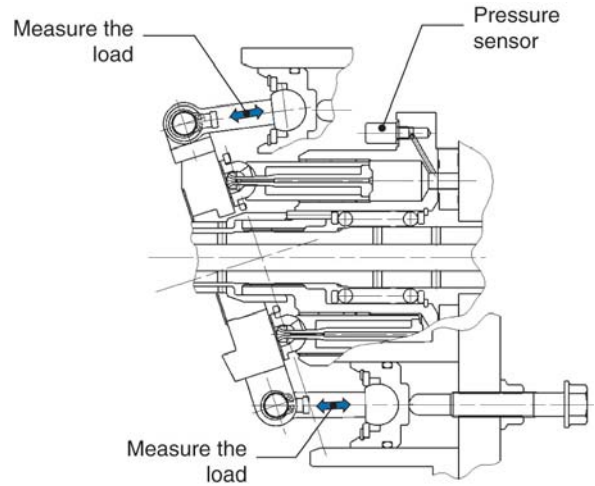


Fig. 19 Experimental equipment

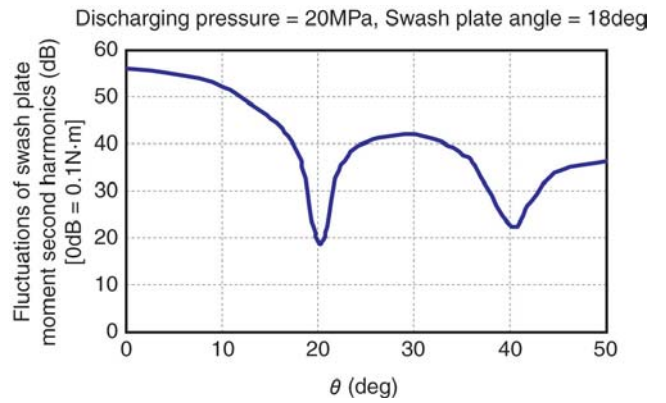


Fig. 20 Relation between θ and swash plate moment

Fig. 21 displays an improved valve plate for a pump with the basic specifications (9 pistons) whose port timing has been adjusted to provide θ_1 and θ_2 near 20° . The cylinder port is connected with the discharge port 20° before bottom dead center (B.D.C.). By introducing the high pressure of the discharge port into the cylinder while the cylinder volume is still in the process of being increased, there results a smoother pressure transition. Again, the cylinder port is connected with the suction port at 20° before top dead center (T.D.C.), while the cylinder volume is still being decreased, by introducing the cylinder pressure into the suction port to allow a smooth pressure transition.

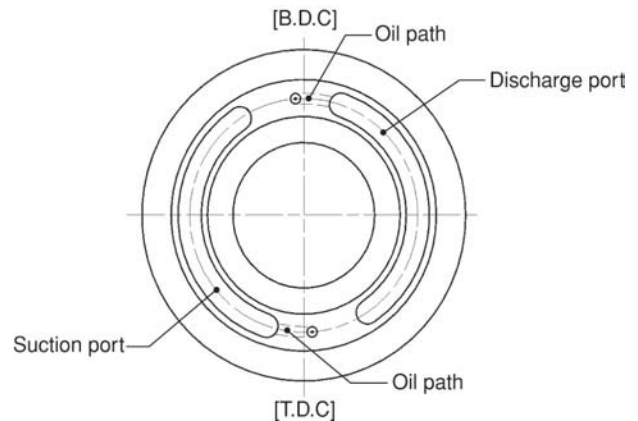


Fig. 21 Port timing change valve plate

Figs. 22 and 23 show experimental measurements of the cylinder pressure and of the fluctuations of moment on the swash plate, respectively. The specifications of the valve plate for the new port timing enable control of both θ_1 and θ_2 at about 20° . The fluctuations of the moment on the swash plate have been reduced and compared with the basic model, showing a beneficial effect of adjusting θ_1 and θ_2 to 20° .

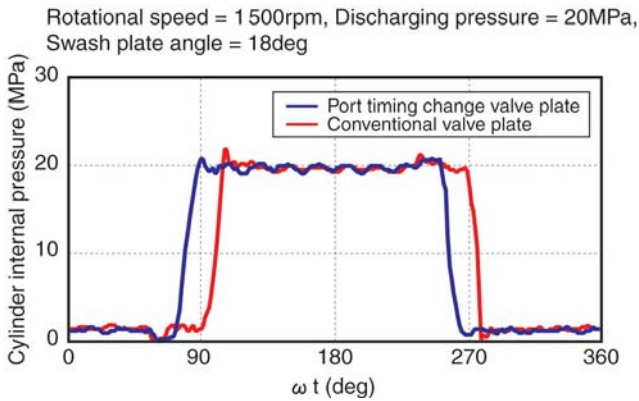


Fig. 22 Cylinder internal pressure with port timing change valve plate

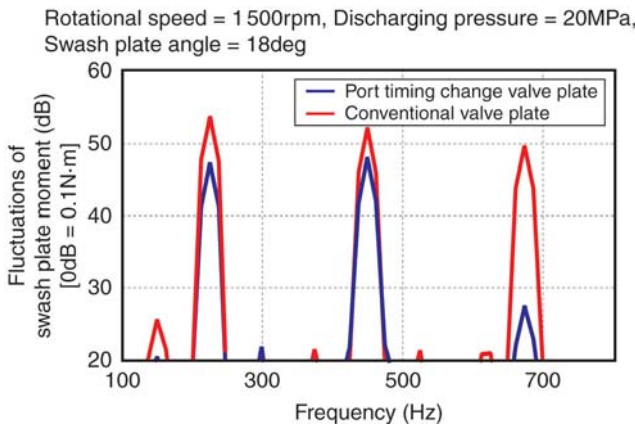


Fig. 23 Fluctuations of swash plate moment with port timing change valve plate

5.3.2. Number of pistons

This section describes the reduction of noise by increasing the number of pistons. Fig. 24 shows the relation of altered piston number with θ ($= \theta_1 = \theta_2$) and swash plate moment fluctuation. Increasing the number of pistons from 9 to 11 lowers the swash plate moment fluctuation.

Fig. 25 shows test results for the displacement at the Mt point. The peak displacement has been reduced, indicating a benefit from increasing the number of pistons to 11.

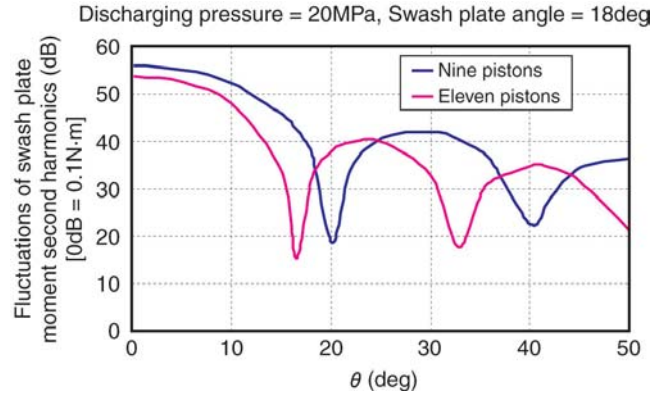


Fig. 24 Relation between number of pistons and fluctuations of swash plate moment

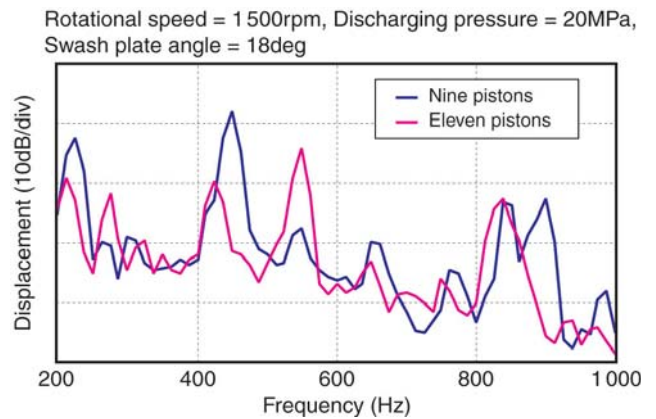


Fig. 25 Relation between number of pistons and displacement of mounting point

5.3.3. Mechanism for variable port timing

Fig. 26 shows a proposed mechanism for variable port timing in order to optimize the cylinder pressure for pumping conditions (pressure, swash plate angle, pump speed). This controls the timing of connections of the cylinder ports with the discharge and suction ports in order to maintain the θ_1 and θ_2 of the cylinder pressure in the neighborhood of 20° under differing operating conditions.

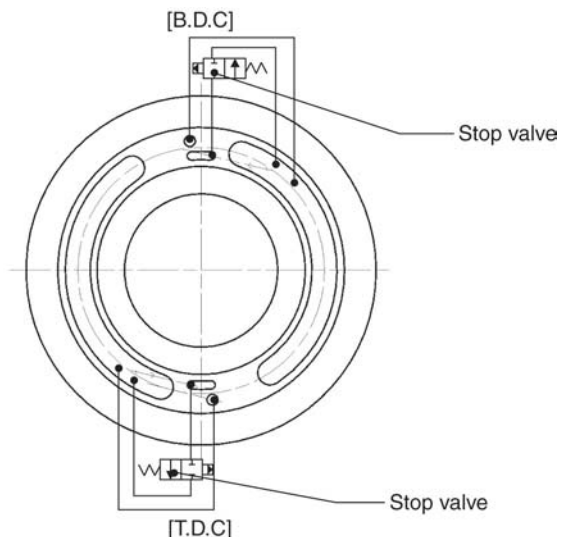


Fig. 26 Variable port timing system

Fig. 27 shows an estimate of mount displacement when the vehicle is traveling at a low speed and these conditions: swash plate angle, 6°; pump speed, 700 rpm; and pressure, 10 MPa. Displacement is predicted to be lower if the variable ports are closed. Fig. 28 shows an estimate of mount displacement while the vehicle is being accelerated, under the following conditions: swash plate angle 18°; pump speed 1,500 rpm; and pressure, 20 MPa. Here, mount displacement is lower if the variable ports are open. Thus, the closed or open condition of these ports may be selected on the basis of the operating conditions in order to minimize noise.

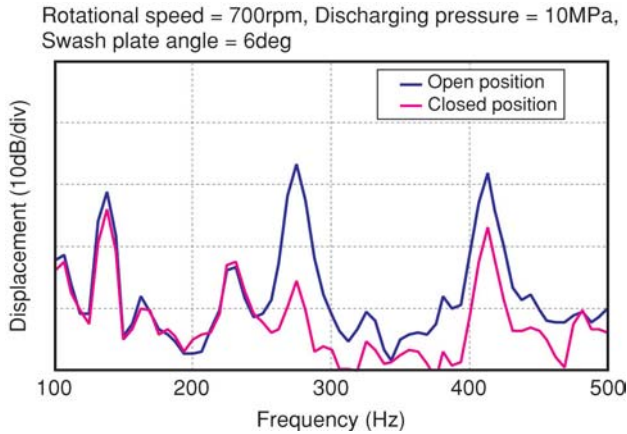


Fig. 27 Displacement of mounting point at low vehicle speed

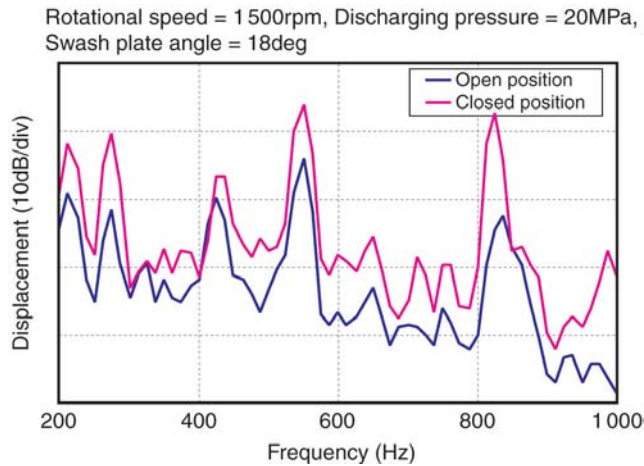


Fig. 28 Displacement of mounting point when accelerating

5.3.4. Cylinder ports with unequal pitch

Cylinder ports with unequal pitch were employed with the goal of reducing the prominence of both the fundamental order and the harmonic component of pump vibration (Fig. 29). By leaving the piston pitch even while introducing variation in the port pitch, variations in the cylinder pressure timing were introduced. Specifications were chosen on the basis of calculations in order to avoid large increases in the fluctuations of moment on the swash plate or in the fluctuations in specific frequencies.

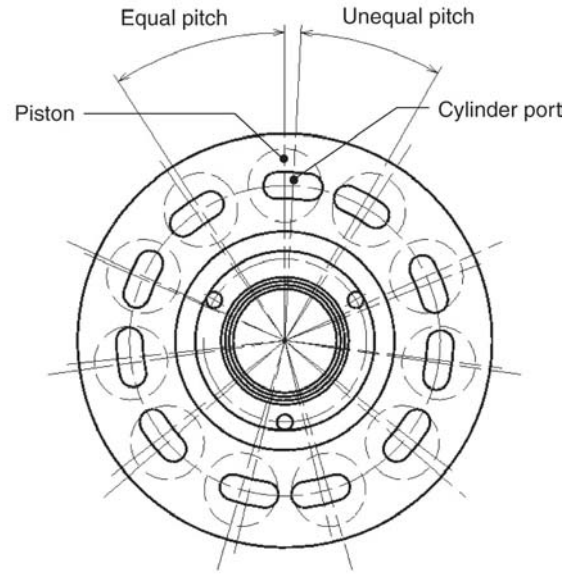


Fig. 29 Unequal pitch cylinder port

Fig. 30 shows a comparison of mount displacement in pumps with even-pitch cylinder port spacing and uneven-pitch cylinder port spacing. Not only did the change to uneven pitch result in a lower peak mount displacement, in the high-frequency region, no large displacement of specific frequencies was observed. These effects mean that the sound quality becomes much less annoying. Thus, the anticipated pump noise in an actual cabin has been reduced more than expected from the data.

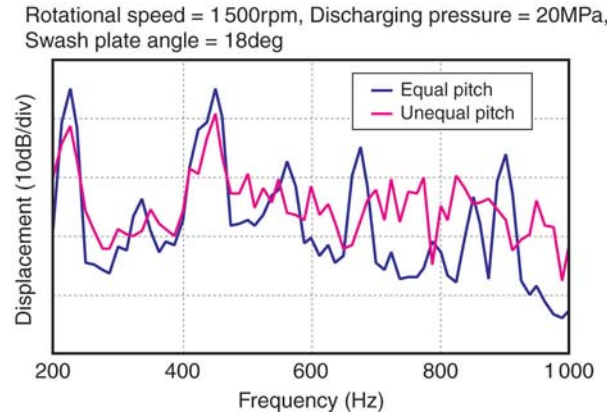


Fig. 30 Displacement of mounting point when unequal pitch cylinder port

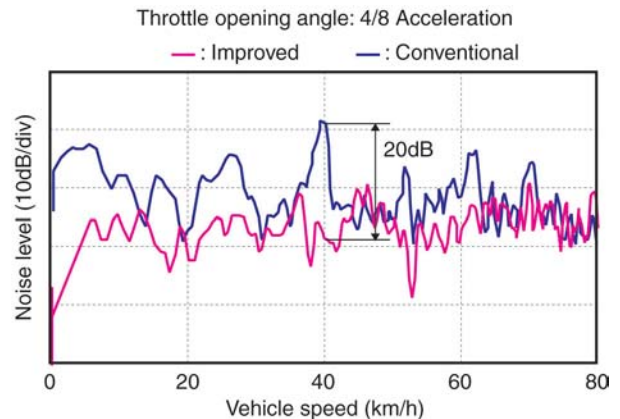


Fig. 31 Pump noise level in the cabin

As shown in Fig. 31, adoption of the above measures against noise has resulted in a reduction of pump noise in the cabin of, at a maximum, 20 dB below the noise from the current pumps. These measures can also be expected to reduce hydraulic motor noise.

6. Summary

Research was carried out on automotive oil hydraulics CVT for adoption in large-displacement, transverse engines for FFD vehicles. The following results were obtained:

- (1) A high cruise efficiency was obtained in an oil hydraulics CVT by using a high-efficiency HMT equipped with a lock-up brake and with part lubrication in the pump and motor to reduce the agitation loss.
- (2) The speed of the motor reaches 7,000 rpm at the peak speed of the vehicle, but by using bowl type slippers, which are exposed to lower centrifugal force, and by adding a sliding part to the slipper to aid cooling, better prospects for durability were realized.
- (3) It was shown that there is a correlation between the noise in a real vehicle and the fluctuations in moment on the swash plates in pumps and motors. The pump was used as an example to verify that a calculated value for fluctuations in moment on the swash plate closely matched observed fluctuations. Then, a design was developed for varying port timing to adjust cylinder pressure in accordance with driving conditions, and the number of pistons was increased to 11; these measures reduced swash plate moment fluctuations. Also, a cylinder port design with unequal cylinder pitch was tested; no large displacement of characteristic frequencies was observed and pump noise was reduced, so that the quality of pump noise was no longer annoying.

7. Acknowledgements

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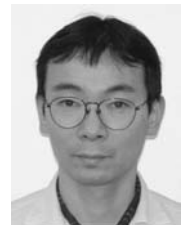
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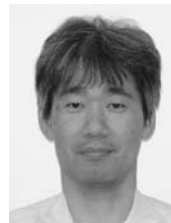
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