Characteristics of Displacement Control Mechanisms in Axial Piston Pumps*

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Pump control with a variable displacement type piston pump is mainly adopted for hydrostatic transmission of a relative large power. In this paper, the characteristics of servo type displacement control mechanisms used in a swash plate type and a bent-axis type piston pump are analyzed theoretically and experimentally.

The calculated values by the non-linear theory coincide well with the test data. Effects of the parameters on the stability, the speed of response and so on are also discussed for general utility to design purposes.

1. Introduction

The "pump control" of hydrostatic power transmissions is desirable from the standpoint of energy efficiency especially for a relative large power.

With piston pumps, S. Ishihara⁽¹⁾⁽²⁾ has investigated steady state moments acting around a swivel yoke or a swash plate. On the dynamic characteristics of displacement control mechanisms, however, there have been only a few papers.⁽³⁾⁽⁴⁾ It is difficult to estimate the characteristics of the pump control, for some of them have treated pump characteristics rather simply and the others have not been verified experimentally.

In this paper the characteristics of the basic hydraulic servo type control mechanisms have been investigated both theoretically and experimentally. The functional build-ups of the tested control mechanisms are shown in Fig. 1. The mechanism (a) is used in a swash plate type piston pump and (b) in a bent-axis type. These mechanisms are in principle available for either type.

The hydraulic servo type control mechanisms are the mechanisms that change the output, i.e., the angle of inclination of a swash plate or of a swivel yoke corresponding to the input, i.e., the displacement of the pilot spool.

2. Nomenclature

- A: area
- A_c : sectional area of pump cylinder
- A_s : sectional area of servo piston
- A_{r1} : sectional area of flow path in restrictor
- C_{d1} : discharge coefficient
- K: fluid bulk modulus
- m: mass

M: moment acting around swash plate or swivel yoke

M_s : M due to servo piston

n: rotational speed

N: number of pistons

p: pressure

p_1 : inlet pressure (for $\alpha > 0$)

p_2 : outlet pressure (for $\alpha > 0$)

p_c : pressure in pump cylinder

p_s : pressure in servo-cylinder

$\Delta p = p_2 - p_1$

q: flow rate

q_c : pump delivery

q_{r1} : flow rate in restrictor

t: time

V: pump displacement; fluid volume

V_s : fluid volume in servo-cylinder

x: position of pilot spool

y: position of servo piston

α : angle of inclination of swash plate or swivel yoke

θ : rotational angle measured from bottom dead center for $\alpha > 0$

(a) 2 port valve type (swash plate pump)

(b) 3 port valve type (bent-axis pump)

Fig.1 Graphical diagram of displacement control mechanism

* Received 3rd December, 1976.

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ρ : fluid density
 ω : frequency of input, that is, pilot spool displacement

Suffix:

' : derivative with time
 $\bar{}$: time mean value
 $\cdot$: value concerning motor or load
 $\circ$: steady state value

3. Theory

3.1 Moment acting around swash plate or swivel yoke

3.1.1 Moment due to pump characteristics

The moment due to p_c is now derived in the case of swash plate type pumps. The moment M_i through a piston which is at angle θ is given by

$$M_i = A_c p_c (R \cos \theta - a \sin \alpha - e) / \cos^2 \alpha \quad (1)$$

where nomenclature shown in Fig. 2 is used. For the whole piston, we obtain

$$M = \sum_{i=1}^N M_i \quad (2)$$

In this paper, let M be positive when it increases α .

In order to express p_c , $\theta_1 \sim \theta_4$ are defined as follows: θ_1 is the rotational angle of the cylinder-bore center corresponding to the instant when the flow path between the outlet port of the valve plate and the cylinder port of the cylinder block opens after the cylinder bore center passes the bottom dead center, θ_2 when the path between the both ports closes, θ_3 when the path between the inlet port of the valve plate and the cylinder port opens after the cylinder bore center passes the top dead center, and θ_4 when the path closes.

In first approximation, we assume that $p_c = p_2$ at $\theta = \theta_1 \sim \theta_2$ and $p_c = p_1$ at $\theta = \theta_3 \sim \theta_4$. And considering that, in general, N is an odd number, we can say as follows: when one of the center of cylinder bores passes through the angle θ_1 , there are $(N+1)/2$ cylinders on the outlet side. And then, when one of the center of cylinder bores passes through the angle θ_3 , the number of pistons on the outlet side decreases to $(N-1)/2$. Considering this periodical characteristics in the pump construction, we obtain the following from Eq.(2).

$$\left. \begin{aligned} \theta_1 \leq \theta < \theta_3 - (N-1)\pi/N : \\ M_1 &= -\frac{A_c}{\cos^2 \alpha} \left[\frac{1}{2} \Delta p R \frac{\sin(\theta - \pi/2N)}{\sin(\pi/2N)} \right. \\ &\quad \left. + \frac{1}{2} \{ (N+1)p_2 + (N-1)p_1 \} (a \sin \alpha + e) \right] \\ \theta_3 - (N-1)\pi/N \leq \theta < \theta_1 + (2\pi/N) : \\ M_1 &= -\frac{A_c}{\cos^2 \alpha} \left[\frac{1}{2} \Delta p R \frac{\sin(\theta - 3\pi/2N)}{\sin(\pi/2N)} \right. \\ &\quad \left. + \frac{1}{2} \{ (N-1)p_2 + (N+1)p_1 \} (a \sin \alpha + e) \right] \end{aligned} \right\} \quad (3)$$

where M_i is M corresponding to the case where p_c changes in rectangular wave. In general, the frequency of fluctuation in M_i given by Eq.(3) is much higher than that of α . Then, it may be sufficient to take the average value. That is,

$$\bar{M}_1 = (N/2\pi) (A_c / \cos^2 \alpha) [\Delta p R (\sin \theta_3 - \sin \theta_1) - (a \sin \alpha + e) \{ (\theta_3 - \theta_1) \Delta p + 2\pi p_1 \}] \quad (4)$$

In second approximation, it may be necessary to consider the trapping phenomena.⁽⁵⁾

Neglecting leakage flows during the trapping, p_c is given by

$$p_c - p_c^* = -K \ln (V_c / V_c^*) \quad (5)$$

where $V_c(\theta)$: fluid volume of chamber, and * means the value at the beginning of the trapping. Then, the supplementary moment M_2 due to the trapping is given by

$$\left. \begin{aligned} \theta_1 - 2\pi \leq \theta < \theta_1 : \\ M_2 &= A_c (p_c - p_1) (R \cos \theta - a \sin \alpha - e) / \cos^2 \alpha \\ \theta_2 \leq \theta < \theta_3 : \\ M_2 &= A_c (p_c - p_2) (R \cos \theta - a \sin \alpha - e) / \cos^2 \alpha \end{aligned} \right\} \quad (6)$$

Only one cylinder comes usually in the range of angle given in Eq.(6). The time average is given by

$$\bar{M}_2 = \frac{N}{2\pi} \left[\int_{\theta_1 - 2\pi}^{\theta_1} M_2 d\theta + \int_{\theta_2}^{\theta_3} M_2 d\theta \right] \quad (6')$$

There are miscellaneous forces which contribute to M , e.g., the return spring force in cylinders, the spring force which pushes the cylinder block to the valve plate, the frictional force between pistons and cylinder bores and so on. The moments due to these forces are usually small enough compared with M_1 .

Next, let us consider the case of a bent-axis type. The moment acting around a swivel yoke axis is given by Eq.(7) based on the same treatment as the swash plate type.

$$\bar{M}_1 = (N/2\pi) A_c \Delta p R (\sin \theta_3 - \sin \theta_1) \quad (7)$$

$$\bar{M}_2 = \frac{N}{2\pi} \left[\int_{\theta_1 - 2\pi}^{\theta_1} A_c (p_c - p_1) R \cos \theta d\theta + \int_{\theta_2}^{\theta_3} A_c (p_c - p_2) R \cos \theta d\theta \right] \quad (7')$$

where R : pitch circle radius of spherical bearings in pump driving flange. And it is assumed that the center of the above mentioned bearings is on the same plane as the axis of the yoke pintle.

3.1.2 Moment due to servo piston

The servo mechanism used in the tested swash plate pump is shown in Fig.3. The outlet fluid enters the cylinder chamber by way of a fixed restrictor, and then returns to the suction side by way of a restrictor composed of the pilot spool and the servo piston. In an equilibrium condition, the force acting on the servo piston due to the servo-cylinder pressure p_c which is controlled by the restrictors is equal to the force acting on it by the swash plate mechanism. By changing the pilot spool position, the area of the variable area restrictor A_{v2} changes and then p_c changes from an equilibrium value i.e., the equilibrium state is lost. However, the piston moves so as to compensate for the change, then a new equilibrium state is attained.

Fig.2 Schematic representation of pump mechanism

From the continuity condition we obtain

$$q_{v1} = q_{v2} + A_v \dot{y} + (V_v/K) \dot{p}_v \quad (8)$$

where

$$\left. \begin{aligned} q_{v1} &= \text{sign}(p_2 - p_v) C_{v1} A_{v1} \sqrt{2|p_2 - p_v|/\rho} \\ q_{v2} &= \text{sign}(p_v - p_1) C_{v2} A_{v2} \sqrt{2|p_v - p_1|/\rho} \\ A_{v2} &= r^2(2\delta - \sin 2\delta), \quad y - x = r(1 - \cos \delta) \end{aligned} \right\} \quad (9)$$

q_{v1}, q_{v2} : flow rates passing through restrictor, C_{v1}, C_{v2} : discharge coefficients, A_{v1}, A_{v2} : areas of restrictors (A_{v2} is composed of two circular ports), A_v : sectional area of servo piston, V_v : fluid volume of chamber.

The equation of motion of the servo piston is

$$(m_v + \rho V_v) \ddot{y} = A_v(p_v - p_1) - F_v$$

where m_v : mass of servo piston, F_v : force acting on swash plate through linkage. The frictional force is neglected here. The characteristics of the linkage is generally given by $y = y(\alpha)$. The following is a good approximation for the present case.

$$y = f_0 \sin(\alpha - \gamma_0) + f_1 \quad (10)$$

where f_0, f_1, γ_0 : constants. The moment acting on the swash plate due to F_v is approximately given by

$$M_v = f_0 \cos(\alpha - \gamma_0) [A_v(p_v - p_1) - (m_v + \rho V_v) \ddot{y}] \quad (11)$$

The servo mechanism used in the tested bent-axis pump is shown in Fig.4. In this case, a constant pressure source of pressure p_s is adopted. From the continuity condition

$$q_{v2} = q_{v3} - A_v \dot{y} + (V_v/K) \dot{p}_v \quad (12)$$

where

$$\left. \begin{aligned} q_{v2} &= \text{sign}(p_s - p_v) C_{v2} A_{v2} \sqrt{2|p_s - p_v|/\rho} \\ q_{v3} &= \text{sign}(p_v - p_1) C_{v3} A_{v3} \sqrt{2|p_v - p_1|/\rho} \end{aligned} \right\} \quad (13)$$

Both A_{v2} and A_{v3} are composed of two circular ports, and an overlap condition is adopted, i.e., the length of spool land l_v is greater than the diameter of port $2r$. Then

$$\left. \begin{aligned} A_{v2} &= r^2(2\delta_2 - \sin 2\delta_2), \\ y - x &= r(1 - \cos \delta_2) + (l_v - 2r)/2 \\ A_{v3} &= r^2(2\delta_3 - \sin 2\delta_3), \\ y - x &= -r(1 - \cos \delta_3) - (l_v - 2r)/2 \end{aligned} \right\}$$

where $A_{v2} = 0$ for $y + r < x + (l_v/2)$ and $A_{v3} = 0$ for $y - r > x - (l_v/2)$. In the above equations, x and y are measured from the position corresponding to $\alpha = 0$. If the leakage between the piston (spool) and the sleeve is taken into account, a term proportional to $(p_s - p_v)$ is added to the right side of Eq.(12) and terms proportional to $(p_s - p_v)$ and p_v respectively are added to the right side of Eq.(13).

M_v is given by

$$M_v = f_0 [A_v p_v - A_v p_s - (m_v + \rho V_v) \ddot{y}] \quad (14)$$

and we obtain the approximate characteristics of the linkage as follows:

$$y = f_0 \tan \alpha \quad (15)$$

3.2 Fundamental Equations

The equation of rotational motion of the swash plate and the swivel yoke is given by

$$I \ddot{\alpha} + T_f(\dot{\alpha}/|\dot{\alpha}|) = \bar{M}_1 + \bar{M}_2 + M_v \quad (16)$$

where I : equivalent moment of inertia, T_f : moment due to Coulomb friction. The gyroscopic effect of a cylinder block of bent-axis pumps works perpendicular to the axis

of inclination of α and it does not appear directly in Eq.(16).

The simultaneous equations, i.e., Eqs. (16) and (8) or (12), are the fundamental equations of the displacement control mechanisms.

We can express the characteristics of pump flow rates, fluid lines and loads in various ways. Here, let us use the following for the swash plate pump, considering the experimental equipment and conditions. For the pump flow rate,

$$q_2 = nV - c_l \Delta p V_{\max} / \mu \quad (17)$$

where c_l : leakage coefficient, μ : fluid viscosity. For the delivery line, only the compressibility is taken into account, then

$$q_1' = q_2 - V_v \dot{p}_2 / K \quad (18)$$

where q_1' : flow rate at outlet of pipe line, V_v : fluid volume. The characteristics of the load connected to the pipe line is

$$p_2 = p_2(q_1') \quad (19)$$

In the case of the bent-axis pump, we use

$$\left. \begin{aligned} (V_v/2\pi) \eta_v' \Delta \dot{p} &= nV \eta_v \eta_v' - n'V' \\ (2\pi)^2 I' \ddot{\alpha}' / \eta_v' + \Delta p_0 V' &= \Delta p V' \end{aligned} \right\} \quad (20)$$

where n' : rotational speed of motor, I' : moment of inertia of motor system, η_v' : motor torque efficiency, η_v, η_v' : volumetric efficiencies of pump and motor respectively, Δp_0 : Δp measured at steady condition, V_v : one half of fluid volume in pipe line between pump and motor.

4. Tests and Discussions

4.1 Test results

In the swash plate pump the pilot spool was driven sinusoidally and the angle of inclination of the swash plate was measured. A

Fig.3 Schematic representation of servo mechanism (2 port valve type)

Fig.4 Schematic representation of servo mechanism (3 port valve type)

petroleum base hydraulic fluid was used. The tests were carried out for a constant p_2 controlled by a relief valve and for a p_2 varied by an orifice at the end of the pipe line. Both the position of the pilot spool and the angle of inclination were measured by potentiometers, and pressures by strain gauge type transducers. The maximum displacement of the test pump V_{max} was $57.0 \text{ cm}^3/\text{rev}$ at $\alpha_{max}=16.7^\circ$ and the range of α was 5.0° to 16.7° . The test results for $p_2 = \text{const.}$ are shown in Fig. 5. G on the vertical axis means

$$G = (\Delta\alpha/\alpha_{max})/[\Delta x/(x_{max}-x_{a=0})]$$

where $x_{a=0}$ is the value of x corresponding to $\alpha=0$. And $\Delta\alpha_0$ is the amplitude of α at $\omega \rightarrow 0$. The upper limit of ω is given by a stopper which limits the magnitude of $x-y$. That is, at a large ω the spool drives directly the servo piston. The curves in the figure are the calculated values based on the linearized equations. From the steady test results C_{v1} and C_{v2} are equal to 0.6 and 0.7 respectively. In the theoretical calculation, it is assumed that $T_f = 0$ and there is no stopper, unless otherwise mentioned.

To clarify the effect of non-linearity, the calculated values of the fundamental equations by Runge-Kutta-Gill method are shown in Fig. 6. The solid lines correspond to $T_f = 0$, the chain line to $T_f(0.3)$ and the broken line to $T_f(0.7)$ are plotted. It is clear that the non-linearity has a significant effect on the characteristics for a large ω . The main cause for this is the non-linearity (including the characteristics as a limiter) of the area of the restrictors. On these figures both the amplitude ratio and the phase angle are formally depicted against ω . The amplitude ratio is defined as (maximum value - minimum value)/2 and the phase angle is the mean of the phase angles corresponding to the maximum and minimum values. This expression is adopted in both the test results and the theoretical calculations.

In Fig. 7 the results for an orifice load are shown and $V_t = 230 \text{ cm}^3$. The chain line corresponds to the case where T_f is included, and the broken line to the linear theory.

In these figures it is seen that the non-linear theory including Coulomb friction can explain the experimental results. The test pump has no relief-groove and theoretically the effect of M_2 on the characteristics is negligible.

For the bent-axis pump, the rotational speed of the motor making up a closed loop hydrostatic transmission was measured when the pilot spool was manually moved in such a way that its position changed at a constant speed. The maximum displacement V_{max} was $2.73 \text{ cm}^3/\text{rev}$ at $\alpha_{max}=8.1^\circ$ and the range of α was -7.9° to 8.1° . The load characteristics of the pump are shown in Fig. 8. The test results are shown in Figs. 9 and 10. The theoretical results are obtained by solving Eqs. (16) and (20) numerically. In this calculation, the load characteristics near $n = 0 \text{ rpm}$ are assumed as the broken line in Fig. 8. The test conditions are as follows: $x_{max} = 0.761 \text{ cm}$, $\Delta p_{o,max} = 27.0 \text{ kgf/cm}^2$, $p_s = 5.0 \text{ kgf/cm}^2$, $n_{o,max} = 575 \text{ rpm}$, $V' = 9.40 \text{ cm}^3/\text{rev}$ and $\eta_{p'} = 0.90$. It is assumed that $\eta_{p'} = \eta_T = 1.0$

for simplicity. T_s in Figs. 9 and 10 is the time required to change the pilot spool position, and the minimum value of T_s is limited by a stopper in the servo mechanism. T_s given in the figures is close to this limit. The calculated values coincide fairly well with the test results.

4.2 Discussion

4.2.1 Stability

We can easily obtain the stability conditions of the mechanisms applying Routh-Hurwitz criterion. In the basic case of constant p_2 the stability condition of the swash plate pump composed of a two-port valve is given by

Fig. 5 Comparison of theory with test (swash plate pump, $p_2 = \text{const.}$)

Fig. 6 Effects of input amplitude and Coulomb friction (swash plate pump, $p_2 = \text{const.}$)

Fig. 7 Comparison of theory with test (swash plate pump, orifice load)

$$\frac{c_{v1}A_{v1}}{[(p_2-p_{v0})(p_{v0}-p_1)]^{1/2}} + \frac{c_{v2}A_{v20}}{p_{v0}-p_1} - 8c_{v2}\frac{V_{v0}}{A_vK}[(2r-y_0+x_0)(y_0-x_0)]^{1/2} > 0 \quad (21)$$

where $T_f = 0$ is assumed.

For the bent-axis pump composed of a three-port valve, we obtain

$$k_{22} + k_{32} - (k_{21} + k_{31})(V_{v0}/KA_v) > 0 \quad (22)$$

where

$$\left. \begin{aligned} \Delta q_{v2} &= k_{21}\Delta(y-x) - k_{22}\Delta p_v \\ \Delta q_{v3} &= -k_{31}\Delta(y-x) + k_{32}\Delta p_v \end{aligned} \right\}$$

and the constants k_{21} , k_{22} , k_{31} and k_{32} are positive. In this case an overlap valve is used, then it is practical to choose a finite value as $\Delta(y-x)$ to estimate the constants.

It is reasonably considered that the characteristics are close to those of an underlap valve, because there is some leakage between the spool land and the sleeve, and M_1 has a fluctuating part whose frequency is decided by both N and n .

Equations (21) and (22) give the limit of the fluid volume in the mechanism. For example, from Eq. (22) we obtain

$$\frac{k_{21} + k_{31}}{k_{22} + k_{32}} \left(= \frac{(\Delta p)_v}{\Delta(y-x)} \right) < \frac{KA_v}{V_{v0}} \left(= \frac{(\Delta p)_c}{\Delta(y-x)} \right)$$

where $(\Delta p)_v$ means the pressure change due to the valve characteristics caused by a small change $\Delta(y-x)$ under the condition that $\Delta q_{v2} = \Delta q_{v3}$, and $(\Delta p)_c$ means the pressure change due to the change in fluid volume of the chamber $A_v\Delta(y-x)$ under the same condition. Then, the above expression in parenthesis means that if the stiffness defined by the valve characteristics is smaller than that by the fluid compressibility, the equilibrium point is stable because the force applied to the servo piston is determined by the valve characteristics.

The displacement control mechanisms should operate stably under various conditions. The stability conditions of such oil hydraulic systems are derived by taking the pump load characteristics into account. Even in these cases, Eq. (21) or (22) may be one of the main conditions.

4.2.2 Speed of response

Speed of response is a criterion of the evaluation of the mechanisms. The relationship between Δx and $\Delta \alpha$ may be basically expressed by a first order lag element. In the case of $p_2 = \text{const.}$, time constant T will be:

2 port valve,

$$T = A_v / 4c_{v2} \sqrt{2/p_{v0} - p_1} \sqrt{(2r - y_0 + x_0)(y_0 - x_0)} \quad (23)$$

3 port valve,

$$T = A_v / (k_{21} + k_{31}) \quad (24)$$

Physically, T corresponds to the time required to fill up the space caused by the change of position of the pilot spool with the fluid passing through the variable area restrictor. In order to obtain the fast response, in the case of Eq. (23), it is effective to decrease the area A_v , to increase the flow area of restrictors (either or both of fixed and variable restrictors), and to have a large p_{v0} .

From the calculated results shown in Fig. 6, the angular frequency corresponding to

the amplitude ratio -3dB is 165rad/s (phase angle -45°) for small fluctuations and 29rad/s (-36°) for $\Delta \alpha_0 / \alpha_{\max} = 0.119$. In the case of a three ports zerolap valve system, the angular frequency is 55rad/s (phase angle -46°) for $\Delta \alpha_0 / \alpha_{\max} = 0.013$ and 195rad/s (-44°) for 0.394 under the conditions $p_1 = p_s = 5.0 \text{ kgf/cm}^2 \text{ g}$ and $p_2 = 150 \text{ kgf/cm}^2 \text{ g}$.

Fig. 8 Characteristics of pump load (bent-axis pump)

Fig. 9 Comparison of theory with test (bent-axis pump - 1)

Fig. 10 Comparison of theory with test (bent-axis pump - 2)

4.2.3 Supply pressure of servo

For the bent-axis type, the fluid of constant pressure is supplied to the mechanism by means of an integrated gear pump and relief valve. Since $\bar{M}_2$ is smaller than $\bar{M}_1$, we obtain the following steady state equation from Eq.(16).

$$p_s = p_{v0} \frac{A_v}{A_{v1}} - \frac{N}{2\pi} \frac{A_z}{A_{v1}} \frac{R}{f_0} \Delta p_0 (\sin \theta_3 - \sin \theta_1)$$

Let the minimum value of p_{v0} be $0 \text{ kgf/cm}^2 \text{ g}$, and we obtain

$$p_{s \min} = - \frac{N}{2\pi} \frac{A_z}{A_{v1}} \frac{R}{f_0} \Delta p_0 (\sin \theta_3 - \sin \theta_1) \quad \dots\dots\dots(25)$$

In general, $\sin \theta_3 - \sin \theta_1 < 0$ and $\Delta p_0 < 0$ for $\alpha < 0$, then, p_s should be larger than $p_{s \min}$ in the case of $\alpha > 0$. Under unsteady condition, p_s should be larger than the steady state value so as to overcome the inertia force. If it is insufficient, cavitation may result.

For the swash plate type, the fluid of pressure p_2 is supplied to the servo. Then, a discussion on $\Delta \alpha / \Delta p_2$ for a constant χ will be presented here. By linearizing Eqs. (16) and (18) and omitting the small terms in low frequency range, we can derive Eq.(26) with Eq.(23).

$$\mathcal{L} \left[\frac{\Delta \alpha}{\Delta p_2} \right] = \frac{[c_{v1} A_{v1} \sqrt{2/\rho(p_{20} - p_{v0})} (\partial \bar{M}_1 / \partial p_2)_0 + f_0 A_{v1}] + c_{v2} A_{v2} \sqrt{2/\rho(p_{20} - p_1)} (\partial \bar{M}_1 / \partial p_2)_0}{2 A_{v1}^2 f_0^2 \cos(\alpha_0 - \gamma_0) (1/T + s)} \quad \dots\dots\dots(26)$$

where $\mathcal{L}[\]$: Laplace transform, s : Laplace's parameter. In the case of $\alpha_0 = 10.7^\circ$ and $p_{20} = 70 \text{ kgf/cm}^2 \text{ g}$, we obtain $1.5 \times 10^{-6} \text{ deg cm}^2/\text{kgf}$ as the amplitude ratio for the test pump, that is, α scarcely changes even when p_2 changes.

5. Conclusion

In this paper, the characteristics of displacement control mechanisms which are used in a swash plate type and a bent axis type pumps are presented. The calculated values coincide fairly well with the tested results. In order to give design instructions, parameters which have effects on the speed of response and stability of the mechanisms are discussed, and the supply pressure of the mechanisms are also discussed.

6. Acknowledgement

The authors are grateful to the Yuken Kogyo Co., Ltd. for providing the tested swash plate pump. The test of the bent-axis pump was carried out at the University of Sheffield, U.K. by one of the authors. The authors would like to thank Prof. J. K. Royle and Mr. J. D. Stringer for their kindness.

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