valves with the small overlaps between the valves and the valve seats the entrance region becomes comparable with the length of the overlaps and the flow becomes very complex with the separation and partiality, so these effects should be considered in analysis.

5. Conclusion

The flow between the valve and the valve seat is very complex and cannot be easily discussed without precise observation and investigation. We divided the flow in various types and derived the thrust from the pressure distribution of each flow type, while we obtained the thrust of the three kinds of disc valve from the pressure distribution of the valve surface measured experimentally. These results show that the thrust is large owing to the viscosity of the fluid when the lift is very small, but decreases suddenly and becomes the lowest and then increases gradually with the increase of the lift.

The discharge coefficient is approximately in inverse relation with the thrust. This fact suggests that the stability of the valves can be discussed with the discharge coefficient instead of the lift of the valves which is difficult for measurement with practical valves. In closing we wish to acknowledge the helpful advices given by Prof. Dr. Tatu Itaya in the course of the present research.

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Performances of Oil Hydraulic Control Valves*

By Toshio TAKENAKA**

Oil hydraulic control valves are used in an oil hydraulic system with an oil hydraulic pump or an oil hydraulic motor, etc. From the view-point of their functions they are roughly classified into three types, that is, pressure, flow and directional control.

The author made experiments on some typical hydraulic control valves, and compared the results with those of analyzing all performances of the valve to make their performances clear; they are the performances of the direct and pilot valve type relief valve, the reducing valve, the second pressure of which is constant, the flow control valve with pressure compensator and the spool valve.

1. Introduction

Oil hydraulic control valves are used in an oil hydraulic system with an oil hydraulic pump or an oil hydraulic motor, etc. From the view-point of their functions they are roughly classified into three types, that is, pressure, flow and directional control type. Many detailed explanations have been made^{(1)~(4)} on the kind, construction and treatment of these control valves, but studies of

the performance are few. In this report, the author has analyzed the performance of typical control valves such as relief valve, reducing valve and flow control valve, and compared the analytical results with the experimental ones. The performance of the valves naturally depends on the over-all characteristics including the material, the construction, dimensions, work accuracy and hydraulic characteristics, but in this report, only the hydraulic characteristics are treated.

2. Pressure control valve

The pressure control valve is used to control

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** Assistant Professor. Tokyo Institute of Technology, Meguro-ku, Tokyo.

the pressure in the oil hydraulic circuit, and there are a relief valve, a reducing valve, a sequence valve, a force counter valve, an unloader valve, etc., in this valve. The performances of the relief valve and reducing valve are as follows.

2.1 Relief valve The operation of the direct relief valve in Fig. 1 is such that the valve opens at the balance of the fluid pressure and the spring force, and the fluid flows through the annular path between the valve and the valve seat. So, in an ideal relief valve, such discharge-pressure curve should be as shown in Fig. 2, A, but the actual relief valve has such a performance as shown in B. This is called pressure override performance, which is one of the problems of the relief valve. This performance occurs through the variation of the spring force due to the variation of the opening of the valve according to the fluid quantity through the valve, and the pressure in the passage of the valve. Thus the valve is opened at the pressure lower than the setting pressure in the circuit. Figs. 6 and 17 show the relief performance of the direct and pilot valve type relief valve with the setting pressure taken on the abscissa and the discharge on the ordinate.

2.1.1 Static performance of direct relief valve The following is the explanation on the static performance of the direct relief valve shown

Fig. 1 Conical valve (symbols)

Fig. 2 Relief performance

in Fig. 3, consisting of a conical valve ($d_0=14.85$ mm ϕ , $d_2=24.83$ mm ϕ , $2\phi=90^\circ$) and a regulating spring. Fig. 4 shows the experimental apparatus; the fluid is to be returned into the reservoir through the conduit, direct relief valve and hand adjusting valve with the vane pump directly connected with the motor. The experimental method is as follows: first, the setting pressure is decided by tightening the regulating spring. Then, by closing the adjusting valve gradually, the pressure in the conduit rises and the fluid begins to relieve. Fitting length of the regulating spring was measured with a reading microscope, the pressure with the Bourdon tube pressure gauge corrected beforehand, the relief discharge with a weighing balance and stop-watch, and the opening of the valve with a dial gauge which could be read to 1/1 000 mm. The oil was Swalube RO-150.

When the cross sectional area of flow of the fluid including the center of the valve seat is a , the discharge coefficient of the valve is given by

$$c = Q / a \sqrt{2g(p_1 - p_2) / \gamma} \quad \dots\dots\dots (1)$$

where Q is the discharge, p_1 and p_2 are the pressures of the input and the inside of the valve, and

Fig. 3 Direct relief valve

Fig. 4 Experimental apparatus

γ is the specific weight of the working fluid. Then, when the average velocity of the fluid in a is $v=Q/a$, Reynolds' number is

$$Re = v\delta_0/\nu \dots\dots\dots (2)$$

where δ_0 is the clearance between the valve and the valve seat, and when the opening of the valve is l , $\delta_0=l\sin\phi$ is given (2ϕ : conical angle of the valve), and ν is the kinematic viscosity of the working fluid.

The result is, as shown in Fig. 5, that the values of c become the functions of Reynolds' number, and they converge on a curve, regardless of the opening of the valve and the pressure and in the range of $40 < Re < 200$, the following experimental formula is given:

$$c = 0.0556 Re^{1/2} \dots\dots\dots (3)$$

Next, changing the setting pressure of the relief valve variously, and taking p on the abscissa and Q on the ordinate, Fig. 6 is given.

Then, using the symbols given in Fig. 1, the performance is analyzed. First, when the relief discharge in the case when the opening of the valve is l is given by

$$Q = ca(l)\sqrt{2g(p_1 - p_2)/\gamma} \dots\dots\dots (4)$$

c is, from Eq. (3),

Fig. 5 Re - c (conical valve)

Fig. 6 Relief performance (direct type)

$$\left. \begin{aligned} c &= A \cdot Re^{1/2} \\ A &= 0.0556, Re = Q/\pi d_m \nu, d_m = (d_0 + d_2)/2 \end{aligned} \right\} \dots\dots (3)'$$

Moreover, the cross sectional area of the flow including the center of the valve seat is

$$a(l) = \pi d_m l \sin \phi \dots\dots\dots (5)$$

So, from Eq. (4)

$$Q = \frac{2\pi A^2 g d_m \sin^2 \phi}{\gamma \nu} l^2 (p_1 - p_2) = B l^2 p \dots\dots\dots (6)$$

is obtained, where $B = 2\pi A^2 g d_m \sin^2 \phi / \gamma \nu$, $p = p_1 - p_2$. The flow of the clearance between the valve and the valve seat is laminar, and Q is proportional to p . Fig. 7 gives the experimental value found by Eq. (6), and it shows that B is a constant value.

Next, the thrust acting on the valve is, according to the momentum theory,

$$F = \frac{\pi}{4} d_0^2 p + \frac{\gamma}{g} Q (V_1 - V_2 \cos \phi) \dots\dots\dots (7)$$

where $V_1 = Q/\frac{\pi}{4} d_0^2$, and $V_2 = Q/\pi d_2 l \sin \phi$, so the following equation is given:

$$F = \frac{\pi}{4} d_0^2 p + \frac{\gamma}{g} \left(\frac{4}{\pi d_0^2} - \frac{\cot \phi}{\pi d_2 l} \right) Q^2 \dots\dots\dots (8)$$

Then, considering the equilibrium of the force acting on the valve, when the shrinkage of the spring from the free length when $l=0$ is l_1 , the following equation holds:

$$k(l_1 + l) + W = F \dots\dots\dots (9)$$

where W is the sum of weights of the valve and the spring. Substituting Eqs. (6) and (8) into Eq. (9), the following is obtained:

$$\begin{aligned} B' l^3 \left(\cot \phi - \frac{4d_2}{d_0^2 l} \right) p^2 - \frac{\pi}{4} d_0^2 p \\ + \{k(l_1 + l) + W\} = 0 \dots\dots\dots (10) \end{aligned}$$

where $B' = \frac{\gamma}{g} \frac{B^2}{\pi d_2}$. Accordingly,

$$p = p' + \frac{4B' l^3 \left\{ \cot \phi - \frac{4d_2}{d_0^2 l} \right\}}{\pi d_0^2} p'^2 + \dots\dots\dots (11)$$

is given, where $p' = 4\{k(l_1 + l) + W\} / \pi d_0^2$

(value of p when inertia term is neglected) Accordingly, with parameter l , the value of p is given from Eq. (11) and Q from Eq. (6), and by combining them, a p - Q curve is plotted. The

Fig. 7 p - B

calculation result is shown in Fig. 6, including the case without inertia term. In this figure, the influence of the inertia term on the calculated value increases with the increase of the setting length l_1 of the spring, but this is not so large as the case without inertia term and there is a pretty difference from the experimental value. This means that it is not proper to assume simply that the equilibrium of the force to regulate the opening of the valve l is the equilibrium between the static pressure of the fluid and the spring force; that is, when an overlap of the valve on the valve seat is larger than the diameter of the seat hole, the pressure distribution of the fluid flowing through the clearance of the valve changes as shown in Fig. 8, and the thrust in this part seems to have influence on that of the valve; that is, the thrust F_s due to the pressure distribution of the clearance of the valve is given by the equation,

$$F_s = 2\pi \sin \phi \int_{s_1}^{s_2} p_s ds \quad (12)$$

and if $p_s = p_s(s)$ holds, F_s shall be determined. However, p_s is indistinct. So, as a trial, the effective area receiving the pressure A_s ,

$$A_s = F/(p_1 - p_2) \quad (13)$$

is obtained in Eq. (9) from the experimental values with reference to values of l/d_0 , as shown in Fig. 9; in the range of $l/d_0 = 0 \sim 0.015$, A_s/A_m is constant, in which A_m is the area including the center of the valve seat $\{(\pi/4)d_m^2\}$. Thus in the range where the opening of the valve is small, the

Fig. 8 Flow in clearance of valve

Fig. 9 l/d_0 - A_s/A_m

thrust,

$$F = A_m p \quad (14)$$

acts, while in the opposite range,

$$F = F_0 + F_s = A_0 p + \frac{\pi}{4} (d_2^2 - d_0^2) p_m \quad (15)$$

where $A_0 = \frac{\pi}{4} d_0^2$.

Next, in Fig. 10, showing the value of p with the parameter l_1 , there are break points between $l = 0.25 \sim 0.30$ mm, and in the range of $l = 0 \sim 0.25$ mm and more than $l = 0.25$ mm, respectively (oil temperature $\theta = 30^\circ\text{C}$)

$$\left. \begin{aligned} p &= C e^{0.8l} \\ p &= D e^{2.4l} \end{aligned} \right\} \quad (16)$$

C and D are given, as the functions of l_1 , as follows:

$$\left. \begin{aligned} C &= 2.02 l_1 + 0.520 \\ D &= 1.29 l_1 + 0.283 \end{aligned} \right\} \quad (17)$$

Then, changing the oil temperature to 20° and 50°C and seeking the relation of p and l in the same way as the above case, in the range more than $l = 0.25$ mm, we obtain the equation $p = D e^{E l}$, and between E and the kinematic viscosity of the fluid ν , as shown in Fig. 11, the equation

$$E = 18 \nu^{-2/3} \quad (18)$$

From the above relations, between p and l ,

$$p = C_1 e^{C_2 l} \quad (19)$$

usually holds, and C_1 is the function of l_1 , and C_2 is the constant number or the function of ν .

Fig. 10 l - p Fig. 11 ν - E

2.1.2 Static performance of relief valve with pilot valve Construction of the relief valve (3/4 in) is given in Fig. 12. This valve is the so-called balanced piston type, and in the piston a hydraulic equilibrium is kept when the pilot valve is not opened; and the valve part on the bottom of the piston rod tightly contacts with the valve seat, and the valve closes. When the fluid pressure surpasses the setting pressure, the pilot valve opens, and the fluid in the upper part of the piston escapes through the pilot valve. Thus, as the equilibrium of the piston is broken, the main valve opens and the fluid escapes to keep the oil pressure in conduit constant.

Fig. 12 Relief valve with pilot valve

Fig. 13 Opening areas of valves

Fig. 14 Relation between R_e and c (for pilot valve)

The areas of the opening a of the pilot valve and the main valve are, respectively,

$$\left. \begin{aligned} a &= \pi \left(d_0 - \frac{l}{2} \sin 2\phi \right) l \sin \phi \\ a &= \frac{\pi}{2\sqrt{2}} l (2d_0 - l) \quad (\phi = 45^\circ) \end{aligned} \right\} \dots\dots\dots (20)$$

and, as shown in Fig. 13, variation of a for the opening of the valve is almost linear.

The discharge coefficients given by the experiment for Reynolds' number and shown in Fig. 14, in the case of the pilot valve, in spite of the pressure variation of 10, 20, 30, 40, 50 kg/cm², fall on a line, and as the experimental formula, in the range of $0.2 < R_e < 15$,

$$c = 0.066 R_e^{1/2} \dots\dots\dots (21)$$

is given. In the case of the main valve also, as shown in Fig. 15, regardless of the pressure variation, they fall on a curve, and the experimental formulas are, in the range of $1 < R_e < 90$,

$$c = 0.080 R_e^{1/2} \dots\dots\dots (22)$$

and in $90 < R_e < 200$,

$$c = 0.042 R_e^{1/3} \dots\dots\dots (23)$$

Next, to analyze the static and dynamic characteristics of the valve, pressure drop through small holes of the piston should be known. The following are the results of experiments about the piston hole of the inner diameter $d = 0.115$ cm and the length $L = 1.19$ cm: in Fig. 16, data are arranged with $L/d \cdot R_e$ taken on the abscissa and

Fig. 15 Relation between R_e and c (for main valve)

Fig. 16 Pressure loss of small holes

$\Delta p/\gamma \frac{v^2}{2g}$ on the ordinate. Moreover, in this figure, curves of pressure loss of Poiseuille and experimental results of Schiller and Hagen are shown, too. At first, assuming that in the range of less than $R_e = 2320$ the flow is laminar, the value of $L/d \cdot R_e$ is 0.00446, and in the range more than this value, the flow is laminar. Next, as there is a break point at $L/d \cdot R_e = 0.012$, the flow in this range may be one of inlet length. In the range over the above range, the following experimental formula is given :

$$\Delta p/\gamma \frac{v^2}{2g} = 28 \left(\frac{L}{d R_e} \right)^{0.5} \dots\dots\dots (24)$$

Fig. 17 gives an example of the experimental results on the static characteristic of the relief valve, taking the pressure in the conduit and vent part on the abscissa, and the relief discharge and drain discharge from the pilot valve on the ordinate.

Fig. 18 shows the principle of the relief valve with its symbols. At first, for the pilot valve, the discharge coefficient is given from the experimental results, by

$$C_A = 0.066 R_e^{1/2} = C_{A1} R_e^{1/2} \dots\dots\dots (21)'$$

and the discharge, by

$$q = C_{A2} x^2 p_3$$

where

$$C_{A2} = \frac{2g C_{A1}^2 \pi d_A \sin^2 \phi}{\gamma \nu} \dots\dots\dots (25)$$

x : opening of the pilot valve

d_A : hole diameter of the pilot valve seat

Secondary, for a small hole of the piston, the following equation is given from Eq. (24) :

$$q = C_{B3} (p_1 - p_2)^{2/3}$$

where

$$C_{B3} = \left(\frac{g^2 d_B^2 a_B^3}{14^2 \nu l_B \gamma^2} \right)^{1/3} \dots\dots\dots (26)$$

d_B : inner diameter of the small hole of the piston

$$a_B = \frac{\pi}{4} d_B^2$$

l_B : length of the small hole of the piston

Thirdly, for the main valve,

$$C_C = 0.080 R_e^{1/2} = C_{C1} R_e^{1/2} \dots\dots\dots (22)'$$

$$Q = C_{C2} y^2 p_1$$

where

$$C_{C2} = \frac{2g \pi C_{C1}^2 d_C \sin^2 \phi}{\gamma \nu} \dots\dots\dots (27)$$

y : opening of the main valve

d_C : diameter of the hole of the main valve seat

Now, the working condition of the relief valve should be considered for the following three cases :

(i) Case when the pilot and the main valves are closed

$$p_1 = p_2 = p_3$$

For the pilot valve,

$$x=0, q=0, p_3 A_A < k_A x_0 \dots\dots\dots (28)$$

holds, in which A_A : hole area of the seat of the pilot valve $= (\pi/4) d_A^2$, k_A : spring constant of the spring of the pilot valve, x_0 : shrinkage of the spring when $x=0$.

For the main valve,

$$y=0, Q=0, p_1 A_{p1} < p_2 A_{p2} + k_B y_0 + W_p \dots\dots\dots (29)$$

holds, where A_{p1} : lower area of the piston in the upper part of the main valve, A_{p2} : upper area of the same piston, k_B : spring constant of the spring of the main valve, y_0 : shrinkage of the spring when $y=0$, W_p : sum of weights of the main valve and piston.

(ii) Case when the pilot valve opens and the main valve closes

When $p_1 \rightarrow p_1'$, $p_2 \rightarrow p_2'$, $p_3 \rightarrow p_3'$, as the hole leading to the pilot valve is large enough, $p_2' = p_3'$

Fig. 17 Relief performance (with pilot valve)

Fig. 18 Symbols (relief valve)

is given.

For the pilot valve, with Bernoulli's force F_x ,
 $p_3' A_A = k_A(x_0 + x) + F_x$ (30)

$$q = C_{A_2} x^2 p_3' \text{ (31)}$$

for the small hole of the piston,

$$q = C_{B_3} (p_1' - p_3')^{2/3} \text{ (32)}$$

and for the main valve, when it is about to open,

$$Q = 0, p_1' A_{p_1} = p_3' A_{p_2} + k_B y_0 + W_p \text{ (33)}$$

holds. From Eqs. (30) and (31),

$$q = q(p_3', x) \text{ (34)}$$

is obtained. So, with parameter x , a p_3' - q curve is given. Moreover, from Eqs. (32), (33) and (34) the value of p_1' when the main valve is about to open is given.

(iii) Case when both the main and the pilot valves open

When $p_1 \rightarrow p_1''$, $p_2 \rightarrow p_2''$, $p_3 \rightarrow p_3''$, $p_2'' = p_3''$, for the pilot valve and the small hole of the piston, equation is given by substituting p_1'' for p_1' , and p_3'' for p_3' in Eqs. (30)~(32), and for the main valve, with Bernoulli's force F_y ,

$$p_1'' A_{p_1} = p_3'' A_{p_2} + k_B(y_0 + y) + W_p + F_y \text{ (35)}$$

$$Q = C_{C_2} y^2 p_1'' \text{ (36)}$$

holds, and in the same way as in the case of (ii),

$$Q = Q(p_1'', y) \text{ (37)}$$

is obtained, and with parameter y , the relation of p_1'' - Q is given.

As Eq. (37) is generally complicated, the following method can be taken to examine the relief performance :

$$\text{If when } \begin{cases} p_1'' \rightarrow p_1'' + \Delta p_1 \\ p_3'' \rightarrow p_3'' + \Delta p_3, \\ \begin{cases} x \rightarrow x + \Delta x \\ y \rightarrow y + \Delta y \end{cases} \text{ and } \begin{cases} q \rightarrow q + \Delta q \\ Q \rightarrow Q + \Delta Q \end{cases} \end{cases}$$

hold, for the pilot valve, piston hole and for the main valve, respectively the following equations are given :

$$\left. \begin{aligned} \Delta x &= C_A' \Delta p_3 \\ \text{where } C_A' &= A_A/k_A \end{aligned} \right\} \text{ (38)}$$

$$\left. \begin{aligned} \Delta q &= C_A'' \Delta p_3 + C_A''' \Delta x \\ \text{where } C_A'' &= C_{A_2} x^2, C_A''' = 2C_{A_2} x p_3'' \end{aligned} \right\} \text{ (39)}$$

Fig. 19 Reducing valve

$$\left. \begin{aligned} \Delta q &= C_B'' (\Delta p_1 - \Delta p_3) \\ \text{where } C_B'' &= \frac{2}{3} C_{B_3} (p_1'' - p_3'')^{-1/3} \end{aligned} \right\} \text{ (40)}$$

$$\left. \begin{aligned} \Delta y &= C_{C_1}' \Delta p_1 - C_{C_2}' \Delta p_3 \\ \text{where } C_{C_1}' &= A_{p_1}/k_B, C_{C_2}' = A_{p_2}/k_B \end{aligned} \right\} \text{ (41)}$$

$$\left. \begin{aligned} \Delta Q &= C_C'' \Delta p_1 + C_C''' \Delta y \\ \text{where } C_C'' &= C_{C_2} y^2, C_C''' = 2C_{C_2} y p_1'' \end{aligned} \right\} \text{ (42)}$$

From Eqs. (38), (39) and (40),

$$\left. \begin{aligned} \Delta p_3 &= C_1 \Delta p_1 \\ \text{where } C_1 &= C_B'' / (C_A'' + C_B'' + C_A''' C_A') \end{aligned} \right\} \text{ (43)}$$

Moreover, from Eqs. (41), (42) and (43),

$$\left. \begin{aligned} \Delta Q &= C_2 \Delta p_1 \\ \text{where } C_2 &= C_C'' + C_C''' C_C' - C_1 C_C''' C_{C_2}' \end{aligned} \right\} \text{ (44)}$$

By substituting concrete values to seek the value of C_2 , the slope of the relief performance is well explained.

2.2 Reducing valve Fig. 19 gives the construction of the reducing valve used in the experiments (3/4 in). This valve is one of the type in which the second pressure is constant and consists of a main piston, a body and a pilot valve, with which the pressure of the second side is set. When the pressure of the second side is lower than the set value, the chief piston goes down, and the first and the second sides are communicated. When the second pressure rises to the set value, the equilibrium of the piston is broken, and the piston goes up, throttling the passage between the first and the second sides to limit the discharge to the second side. Thus the pressure rise of the second side is prevented. There are chokes of necessary dimensions in the chief piston and the body to give resistance between the second side and the pilot valve.

Fig. 20 shows the performance of the reducing valve for the pressure variation of the first side, given by variously changing the set pressure and discharge at every set pressure under a constant

Fig. 20 Reducing performance

fluid temperature. From this figure, it is learned that the value of p_2 is nearly constant independently of p_1 . The performance analysis of the reducing valve can be carried out in the same way as that of the relief valve. The symbols are given in Fig. 21.

With the pilot valve open, $p_1 \rightarrow p_1'$, $p_2 \rightarrow p_2'$, $p_3 \rightarrow p_3'$ are given. [For symbols, refer to Eqs. (25)~(29)].

For the pressure, the following is given

$$p_2' = p_3' + \left(\frac{C_{A_2}}{C_{B_3}} \right)^{3/2} \left\{ \frac{p_3' A_A}{k_A} - x_0 \right\}^3 p_3'^{3/2} \dots (45)$$

and for discharge

$$Q_1 = C_{C_2} (p_1' - p_2') \left[\delta c_0 - \left\{ \frac{(p_2' - p_3') A_{p_2} - W_p}{k_B} - y_0 \right\}^2 + C_{A_2} \left\{ \frac{p_3' A_A}{k_A} - x_0 \right\}^2 p_3' \right] \dots (46)$$

where $\delta c_0 = \delta c + y$
 δc : closing of the main valve
 δc_0 : distance from the valve seat when $\delta c = 0$
 y : opening of the main valve from the valve seat

So, with parameter p_3' , the relation of $p_1' - p_2'$ can be determined for a certain Q .

Moreover, considering the minimum variation, at first for the pilot valve, we have

$$\Delta p_3 = C_A \Delta x \quad \text{where } C_A = k_A / A_A \dots (47)$$

$$\Delta q = C_A' \Delta p_3 + C_A'' \Delta x \quad \text{where } C_A' = C_{A_2} x^2, C_A'' = 2 C_{A_2} x p_3' \dots (48)$$

for the choke

$$\Delta q = C_{B'} (\Delta p_2 - \Delta p_3) \quad \text{where } C_{B'} = \frac{2}{3} C_{B_3} (p_2' - p_3')^{-1/3} \dots (49)$$

for the main valve

$$\Delta p_2 - \Delta p_3 = C_B \Delta y \quad \text{where } C_B = k_B / A_{p_2} \dots (50)$$

$$\Delta Q_2 = C_{C'} (\Delta p_1 - \Delta p_2) - C_{C''} \Delta y \quad \text{where } C_{C'} = C_{C_2} (\delta c - y)^2, C_{C''} = 2 C_{C_2} (\delta c - y) (p_1' - p_2') \dots (51)$$

Fig. 21 Symbols (for reducing valve)]

Moreover, from the equation of continuity,

$$\Delta Q_1 = \Delta Q_2 + \Delta q \dots (52)$$

is given. Then from Eqs. (47), (48) and (49), we obtain

$$\left. \begin{aligned} \Delta p_3 &= C_5 \Delta p_2 \\ \text{where } C_5 &= C_2 / (C_1 + C_2) \\ C_1 &= C_A' + C_A'' / C_A \\ C_2 &= C_{B'} \end{aligned} \right\} \dots (53)$$

and from Eqs. (50) and (51),

$$\left. \begin{aligned} \Delta Q_2 &= C_3 (\Delta p_1 - \Delta p_2) - C_4 (\Delta p_2 - \Delta p_3) \\ \text{where } C_3 &= C_{C'}, C_4 = C_{C''} / C_B \end{aligned} \right\} \dots (54)$$

Furthermore, from Eqs. (52), (53) and (54), we obtain

$$\left. \begin{aligned} \Delta p_2 &= C_6 \Delta p_1 - C_7 \Delta Q_1 \\ \text{where } C_6 &= C_3 C_7 \\ C_7 &= 1 / \{ C_3 + C_4 - (C_1 + C_4) C_5 \} \end{aligned} \right\} \dots (55)$$

Examining the order by substituting the actual values, $C_{A_2} = 5.78 \times 10^3$, $A_A = 0.126 \text{ cm}^2$, $k_A = 26.3 \text{ kg/cm}$, $C_{B_3} = 4.1$, $A_p = 2.84 \text{ cm}^2$, $k_B = 4.43 \text{ kg/cm}$ and $C_1 \approx 10$, $C_2 \approx 10^2$, $C_3 \approx 10^2$, $C_4 \approx 10^3$, $C_5 \approx 10^2$, $C_6 \approx 10^{-1}$, $C_7 \approx 10^{-3}$, so from Eq. (55), the following equation is given:

$$\Delta p_2 \approx 10^{-1} \Delta p_1 - 10^{-3} \Delta Q_1 \dots (56)$$

3. Flow control valve

Flow control valve with the simplest construction is a needle valve. As shown in Fig. 22, the discharge through the valve increases with an increase of the pressure drop before and after the valve. The flow control valve of a common oil hydraulic circuit is, as in Fig. 22 showing its performance equipped with a pressure compensator by which the discharge is kept constant for pressure variation before and after the valve.

3.1 Performance of flow control valve with pressure compensator Fig. 23 shows the construction of the flow control valve used in the experiment. This valve is equipped with a pressure compensator, consisting of a control piston which is a pressure compensator, and a throttle valve to

Fig. 22 Performance for discharge

limit the discharge. The opening of the throttle valve is varied with a handle, and to keep pressure difference before and after the throttle valve always constant, the pressure compensator works.

Fig. 24 shows the performance of a 30 l/min valve, when the fluid temperatures are 30, 45, 60°C and the first pressure $p_1=10, 40, 70$ kg/cm², $p_3=0$, with the opening of the throttle valve taken on the abscissa and the discharge on the ordinate.

It is unavoidable that the fluid quantity varies with the change of the fluid temperature, but the discharge varies with the change of p_1 , too. This means that the pressure compensator is not working satisfactorily.

To show this matter more clearly, the discharge was measured for each pressure of $p_3=0, 5$ kg/cm², when $p_1=10$ kg/cm², $p_3=0, 10, 20, 30$ kg/cm² when $p_1=40$ kg/cm², $p_3=0, 10, 30, 50$ kg/cm² when $p_1=70$ kg/cm², at the fluid temperature 30, 45, 60°C, at each opening. Fig. 25 is an example of the above experimental results; it shows the measuring value at each opening of the throttle valve, when the temperature is 45°C, with the value of p_1-p_3 taken on the abscissa.

Next, the performance of this valve is to be analyzed. In Fig. 26, at first the fluid force of the valve port of the pressure compensator is considered. According to the momentum theory, this

force F_x is given by,

$$F_x = (\gamma/g) Q V_1 \cos \phi \quad (57)$$

and it tends to move the spool to the left hand to close the port.

Then, with $a(x)$: area of the port, n : number of the port, c : discharge coefficient of the port, we obtain

$$V_1 = Q / na(x) \sin \phi$$

$$Q = cna(x) \sqrt{2g(p_1 - p_2)/\gamma} \quad (58)$$

and substituting them for Eq. (57), is shown as

$$F_x = 2c^2na(x) (p_1 - p_2) \cot \phi \quad (59)$$

According to Von Mises⁽⁵⁾, when the port is rectangular orifice $\phi=69^\circ$, and the deviation of the spool x is smaller than other dimensions, Eq. (59) becomes

$$F_x = 0.768 c^2 na(x) (p_1 - p_2) \quad (60)$$

and F_x can be given by the pressure difference $(p_1 - p_2)$. c is explained in Chapter 4.

Next, the pressure equilibrium of the spool is to be considered. Assuming that the fluid pressure behind the throttle valve is p_3 , which acts on the back of the piston on the left side of the spool, and when $x=0$, shrinkage of the spring from free

Fig. 23 Flow control valve

Fig. 24 Discharge performance (Opening-Q)

Fig. 25 Discharge performance (p-Q)

Fig. 26 Symbols (flow control valve)

length is x_1 , the force of x direction is

$$k(x_1 - x) + p_3 A_p + p_2 A_2$$

and that of $-x$ direction is

$$p_2(A_p - A_1) + p_2 A_2 + p_2 A_1 + F_x$$

and assuming these two forces are equal because they are equilibrated, the following equation is given :

$$p_2 - p_3 = (1/A_p) \{k(x_1 - x) - F_x\} \quad (61)$$

Next, when the area of the opening of the throttle valve at a certain opening is a' and the discharge coefficient is c' , the discharge Q can be expressed by

$$Q = c' a' \sqrt{2g(p_2 - p_3)/\gamma} \quad (62)$$

Then, as the outlet of the pressure compensator and the passage in front of the throttle valve are large and short,

$$p_2' = p_2 \quad (63)$$

is assumed. Using Eqs. (62) and (63), F_x is expressed as

$$F_x = 0.768 c'^2 a'^2 (p_2 - p_3) / na(x) \quad (64)$$

and from Eq. (61),

$$p_2 - p_3 = \frac{k(x_1 - x)}{A_p + \frac{0.768 c'^2 a'^2}{na(x)}} \quad (65)$$

is obtained.

Moreover, from Eqs. (62), (63) and (65), Q becomes as follows :

$$Q = c' a' \sqrt{\frac{2g}{\gamma} \frac{k(x_1 - x)}{A_p + \frac{0.768 c'^2 a'^2}{na(x)}}} \quad (66)$$

That is, if the opening of the throttle valve is determined, and the deviation of the spool valve is given, Q can be calculated.

Next, $(p_1 - p_3)$ is separated as follows :

Fig. 27 Discharge performance ($p-Q$)

$$p_1 - p_3 = (p_1 - p_2) + (p_2 - p_3) \quad (67)$$

Substituting Eq. (66) for Eq. (58), the first item of the right hand of Eq. (67) becomes

$$p_1 - p_2 = \frac{c'^2 a'^2}{c^2 n^2 a^2(x)} \frac{k(x_1 - x)}{A_p + \frac{0.768 c'^2 a'^2}{na(x)}} \quad (68)$$

So, from Eq. (67),

$$p_1 - p_3 = \frac{k(x_1 - x)}{A_p + \frac{0.768 c'^2 a'^2}{na(x)}} \left\{ 1 + \frac{c'^2 a'^2}{c^2 n^2 a^2(x)} \right\} \quad (69)$$

is given. So, calculating Q and $p_1 - p_3$ in Eqs. (66) and (69) with deviation of the spool valve x , and plotting the results at every opening of each throttle valve, we obtain Fig. 27. (The calculations were for a 10 l/min flow control valve at $k=5.29$ kg/cm, $x_1=1.52$ cm, $A_p=4.52$ cm², $n=4$, $x=0.03 \sim 0.07$ cm).

Next when the thrust in the spool is not considered, Eq. (66) becomes

$$Q = c' a' \sqrt{\frac{2g}{\gamma} \frac{k(x_1 - x)}{A_p}} \quad (70)$$

and Eq. (69) becomes

$$p_1 - p_3 = \frac{k(x_1 - x)}{A_p} \left\{ 1 + \frac{c'^2 a'^2}{c^2 n^2 a^2(x)} \right\} \quad (71)$$

The combination of the above results is represented as dotted lines in Fig. 27. As shown clearly in the figure, the influence of the thrust of the spool is small, and in every case Q shows a constant value for $p_1 - p_3$, except the range where $p_1 - p_3$ is very small. Moreover, to compare them with the experimental values, the cases when the openings of the throttle valve are (1), (2), (6) and (10) are given in this figure; when the opening is small they comparatively well coincide, but with the opening becoming large, they separate.

In the valve with a comparatively large discharge, as shown in Fig. 28, a maximum value of Q often appears in the range where $p_1 - p_3$ is small, then Q gradually diminishes with the increase of $p_1 - p_3$. The cause of this tendency may be that Bernoulli's force in the pressure compensator becomes comparatively large in the range where $p_1 - p_3$ is large, so the static equilibrium position of the

Fig. 28 $p-Q$

Fig. 29 Measurement of deviation of spool

spool shifts, and Q ceases to be constant. Accordingly, to avoid such variation of Q , it is necessary to design and manufacture the pressure compensator which will be able to diminish Bernoulli's force in the valve port. For example, as shown in the studies of S.-Y. Lee and others⁽⁶⁾⁽⁷⁾, the forms and constructions of the port and land should be improved.

4. Direction control valve

Among many types of the direction control valve, the typical one is the spool valve. The relation between pressure difference and discharge in the port of the spool valve is nonlinear, which becomes a question when the performances of the valve and the hydraulic mechanism are analyzed. The results given by calculation are shown in many books⁽⁸⁾⁽⁹⁾, but for the actual use, they should be compared and examined with those by experiments.

The following is the experimental result of the spool valve which is the pressure compensator of the flow control valve. As shown in Fig. 29, a deviation of the spool was measured with a dial gauge by thrusting a rod sticking on the spool. Inner diameters of the spool valve are 16 and 9.5 mm ϕ , and four circular ports are set on a circle, inner diameters of which are 9 and 5 mm ϕ .

Next, Reynolds' number of the flow through the port is given, using hydraulic mean depth, by

$$Re = 4Q/\nu s \quad \dots\dots\dots (72)$$

where s : length of circumference of the port
For the spool valve of 16 mm ϕ , the value of the discharge coefficient c shown in Fig. 30, when the deviation of the spool is varied to 0.65 mm, in the range of $p_1 = 70 \text{ kg/cm}^2$ and $p_2 = 0 \sim 60 \text{ kg/cm}^2$, for Reynolds' number, the deviation of the spool can be arranged as a parameter, and when the deviation is more than 0.45 mm, c is almost constant independently of Reynolds' number, and at less than 0.45 mm c decreases with the increase of the Reynolds' number.

Fig. 30 Re - c (spool valve)

Commonly the discharge coefficient of the valve can be regarded as the function of the area of the opening of the throttle part and Reynolds' number of the flow passing that part.

5. Conclusion

The author made experiments on some typical hydraulic control valves, and compared the results with those of analyzing all performances of the valve to make clear their performances.

Previously, almost all problems of the performance, design, treatment, etc. of these valves have been empirically settled, but for the solution of all performance problems and the collection of the rational design data, a future study on the components of the control valves should be necessary.

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