in hydraulic conveyor through a horizontal straight pipe, and obtain the experimental formula by applying the method of the pneumatic conveyor by T. Uematsu and T. Kano.

The experimental formula obtained by this investigation will be very useful in practice.

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Thrust of the Disc Valves*

By Toshio TAKENAKA**, Ryuichiro YAMANE***,
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Clear knowledge on the thrust of valves is necessary to discuss the stability of valves which plays an important role concerning the stability of a pipe line. In order to have a general idea of relations between the thrust and the lift, and between the discharge coefficient and the lift of common valves, we studied the characteristics of the most simplified valve, that is, a circular disc valve.

The flow between the valve and the valve seat is very complex. The pressure distribution over the valve surface and the lifting-force were calculated analytically for various types of the flow, and examined experimentally for three kinds of disc valve. Experimental results show the correlation between the lifting-force and the discharge coefficient.

1. Introduction

In order to have a general idea of the relations between the thrust and the lift, and between the discharge coefficient and the lift of common valves, we studied the characteristics of the most simplified valve, that is, a circular disc valve. Uzawa⁽¹⁾ and Tasaka⁽²⁾ calculated the thrust of a circular disc valve when the flow between the valve and the valve seat was laminar and turbulent respectively, but we have a different opinion from them about the application of momentum theory. We showed the relations of the thrust and the lift using the flow rate as a parameter for these flow types, and also calculated the thrust using the analytical result for the radial laminar flow between two parallel planes by Murata⁽³⁾, and considered the curve of the flow at the entrance

of the space between the valve and the valve seat applying the analytical method by Miyazu⁽⁴⁾.

The flow between the valve and the valve seat is very complex, and its type may change extremely with the lift. So we analyzed the thrust for various types of the flow.

2. Analysis of the thrust

It is necessary to make clear the condition of the flow between the valve and the valve seat, that is, the deflection of the entrance, separation, growth of the boundary layer and pressure distribution to discuss the thrust of the disc valves. We calculated the pressure distribution assuming that the flow was laminar or turbulent and the velocity distribution was parabolic or 7th-root respectively, and derived the thrust of a disc valve.

2.1 Case when the flow is laminar

(a) Pressure loss due to laminar friction

When the valve and the valve seat overlap each other widely and the lift is small, the viscous friction is not negligible. For the disc valve shown

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in Fig. 1, pressure loss p_f due to viscous friction between r and r_2 is

$$p_f = \frac{6\mu Q}{\pi l^3} \log_e \frac{r_2}{r} \quad (1)$$

where μ : coefficient of viscosity, Q : flow rate, l : lift, u : mean radial velocity at radius r .

(b) Discharged energy loss Kinetic energy p_v discharged at r_2 is calculated from the velocity distribution at the valve end as

$$p_v = 1.543 \gamma \frac{u_2^2}{2g} \quad (2)$$

where u_2 : mean radial velocity at r_2 .

(c) Pressure loss of the flow between two parallel planes Pressure loss p_L is approximately the sum of the pressure loss due to viscous friction p_f and the discharged kinetic energy p_v . Then

$$p_L = \frac{6\mu Q}{\pi l^3} \log_e \frac{r_2}{r} + 1.543 \gamma \frac{u_2^2}{2g} \quad (3)$$

Substituting $Q = 2\pi r_2 l u_2$, Eq. (3) is rewritten as

$$p_L = \frac{6\mu Q}{\pi l^3} \log_e \frac{r_2}{r} + 0.772 \gamma \frac{1}{g} \frac{1}{r_2^2} \left(\frac{Q}{2\pi l} \right)^2 \quad (4)$$

(d) Thrust of the disc valve According to Bernoulli's equation, the pressure p at r is

$$p = p_2 + p_L - \gamma \frac{u^2}{2g} \quad (5)$$

Assume the pressure behind the valve p_2 is equal to atmospheric pressure, then the thrust on the valve F is

$$F = \pi r_0^2 p_0 + 2\pi \int_{r_0}^{r_2} p r dr \quad (6)$$

where r_0 : radius of the hole of the valve seat, p_0 : supply pressure. From Eqs. (4), (5), (6)

$$F = \pi r_0^2 p_0 + 3\mu \left(r_2^2 - r_0^2 + 2r_0^2 \log_e \frac{r_0}{r_2} \right) \frac{Q}{l^3} + \frac{\gamma}{4\pi g} \left(0.772 \frac{r_2^2 - r_0^2}{r_2^2} - \log_e \frac{r_2}{r_0} \right) \frac{Q^2}{l^2} \quad (7)$$

2.2 Case when the inertia term is considered

The flow is assumed to be symmetrical concerning the z -axis of the cylindrical coordinate (r, θ, z) or the axis of the disc valve. Let u, w be radial and axial velocity respectively, stream function $\psi = \psi_l \psi^*$, $r = l\alpha$, $z = l\beta$, $\psi_l = -Q/2\pi$ and eliminate the pressure terms of Navier-Stokes equations in non-dimensional forms, then the solution may be written as

$$\psi^* = \psi_2 \alpha^2 + \psi_0 + \psi_{-2} \alpha^{-2} + \psi_{-4} \alpha^{-4} + \dots \quad (8)$$

Changing u, w, p in non-dimensional forms $V_u = u/u_l$, $V_w = w/u_l$, $P = p/\rho u_l^2$ (where u_l : velocity at $r=l$) and

substituting them into the equation of motion and integrating this, we get the pressure as

$$P = -\frac{1}{2} \left(P_2 \alpha^2 + P_0 \log_e \alpha - \frac{1}{2} P_{-2} \alpha^{-2} - \frac{1}{4} P_{-4} \alpha^{-4} + \dots \right) + C \quad (9)$$

where C : integration constant. Adopting the boundary conditions

$$u = w = 0, \psi = 0 \text{ when } z = 0, \\ \text{or } V_u = V_w = 0, \psi^* = 0 \text{ when } \beta = 0$$

$$u = w = 0, \psi = \psi_l \text{ when } z = l$$

$$\text{or } V_u = V_w = 0, \psi^* = 1 \text{ when } \beta = 1$$

$$p = p_2 \text{ when } r = r_2 \text{ or } P = P_2 \text{ when } \alpha = \alpha_2$$

we get

$$P_2 = 0$$

$$P_0 = -12/R_1$$

$$P_{-2} = -1.54284$$

$$P_{-4} = 0.011429 + 0.0072357 R_1$$

where $R_1 = -u_l l / \nu$

P is shown approximately with four terms obtained above.

Finally p is written as

$$p = p_2 + \frac{6\mu Q}{\pi l^3} \log_e \frac{r_2}{r} + 0.192855 \frac{\rho Q^2}{\pi^2 l^2} \left(\frac{1}{r_2^2} - \frac{1}{r^2} \right) + \left(0.002857 \frac{\rho}{\pi^2} Q^2 + 0.0009045 \frac{\rho^2 Q^3}{\pi^3 \mu l} \right) \left(\frac{1}{r_2^4} - \frac{1}{r^4} \right) \quad (10)$$

and when p_2 is equal to atmospheric pressure,

$$F = \pi r_0^2 p_0 + 3\mu \left(r_2^2 - r_0^2 + 2r_0^2 \log_e \frac{r_0}{r_2} \right) \frac{Q}{l^3} + 0.38571 \frac{\rho}{\pi} \left(\frac{r_2^2 - r_0^2}{2r_2^2} + \log_e \frac{r_0}{r_2} \right) \frac{Q^2}{l^2} + \left(0.002857 \frac{\rho}{\pi} Q^2 + 0.0009045 \frac{\rho^2 Q^3}{\pi^2 \mu l} \right) \left(\frac{r_2^2 - r_0^2}{r_2^4} - \frac{1}{r_0^2} \right) \quad (11)$$

2.3 Case when the flow is turbulent

Applying the momentum theory in the direction of r to the small element of the fluid shown in Fig. 2, we get an equation

Fig. 1

Fig. 2

$$\frac{dp}{dr} + \frac{2\tau_w}{l} + \frac{\gamma}{g} \frac{1}{r} \frac{d(ru_m^2)}{dr} = 0 \quad (12)$$

where τ_w : shear stress on the surface, u_m : mean radial velocity at r .

Assume that the velocity distribution of the flow between the valve and the valve seat follows the Kármán's 7th-root law, then τ_w is shown as

$$\tau_w = 0.03325 \frac{\gamma}{g} \frac{\nu^{1/4}}{l^2} \left(\frac{Q}{2\pi} \right)^{7/4} \left(\frac{1}{r} \right)^{7/4} \quad (13)$$

Equation of continuity is

$$\frac{d}{dr}(ru_m) = 0 \quad (14)$$

Substituting Eqs. (13), (14) and the boundary conditions $p=0$ (assume), $u_m=u_2=Q/2\pi r_2 l$ at $r=r_2$ into Eq. (12), we get the pressure distribution and the thrust of the valve in the form

$$p = \frac{\gamma}{2g} \left[0.1773 \frac{\nu^{1/4}}{l^3} \left(\frac{Q}{2\pi r_2} \right)^{7/4} r_2^2 \left\{ \left(\frac{r_2}{r} \right)^{3/4} - 1 \right\} + \left(\frac{Q}{2\pi r_2 l} \right)^2 \left\{ 1 - \left(\frac{r_2}{r} \right)^2 \right\} \right] \quad (15)$$

$$F = \pi r_0^2 p_0 + \frac{0.3546}{(2\pi)^{3/4}} \frac{\gamma}{g} \nu^{1/4} \left(\frac{r_2^{5/4} - r_0^{5/4}}{5} - \frac{r_2^2 - r_0^2}{8r_2^{3/4}} \right) \frac{Q^{7/4}}{l^3} + \frac{\gamma}{8\pi g} \left(\frac{r_2^2 - r_0^2}{r_2^2} - 2 \log_e \frac{r_2}{r_0} \right) \left(\frac{Q}{l} \right)^2 \quad (16)$$

2.4 Case when the flow is laminar and the curve of the flow at the entrance is considered

Curve of the flow at the entrance of the valve may not be negligible when the overlap of the valve and the valve seat is small as in practical valves.

Inside the hole of the seat u and w , the velocities of r -, z - direction respectively of the cylindrical coordinate shown in Fig. 3, are written as

$$w = -\left(\frac{W}{l} \right) z + W \quad (17)$$

$$u = \left(\frac{1}{2\mu} \frac{\partial p}{\partial r} z - \frac{U}{l} \right) (z-l) \quad (18)$$

Fig. 3

when the inertia terms and the pressure change in the direction of z are neglected. U and W are velocities of r -, z - direction respectively at the exit AB (See Fig. 3). Equation of continuity about a small element of fluid whose thickness is l shown in Fig. 4 is written as

$$\frac{1}{r} \frac{d}{dr} \left(r \frac{\partial p}{\partial r} \right) - \frac{6\mu}{l^2} \frac{1}{r} \frac{d(ru)}{dr} - \frac{12\mu W}{l^3} = 0 \quad (19)$$

Integrating this, we get the pressure loss due to the viscous friction p_f between r' and r'' in the form

$$p_f = \frac{6\mu}{l^2} \int_{r'}^{r''} U dr - \frac{12\mu}{l^3} \int_{r'}^{r''} \left\{ \frac{1}{r} \int (rW) dr \right\} dr + C \log_e \frac{r''}{r'} \quad (20)$$

where C is integration constant.

(a) Assume that the velocity distribution of z - direction at the exit AB of the hole is parabolic and is written in the form (See Fig. 5)

$$W = u_{M_0} \left\{ 1 - \left(\frac{r}{r_0} \right)^2 \right\} = 2u_0 \left\{ 1 - \left(\frac{r}{r_0} \right)^2 \right\} \quad (21)$$

where u_{M_0} : maximum velocity, u_0 : mean velocity. And assume

$$U = 0 \quad (22)$$

Substituting Eqs. (21), (22) into Eq. (20), we get the friction loss

$$p_f = \frac{3\mu u_0}{2l^3} r^2 \left(4 - \frac{r^2}{r_0^2} \right) \quad (23)$$

(b) Assuming that the velocity distribution at the exit AB of the hole is uniform

$$W = u_0 \quad (24)$$

and integrating Eq. (20) under the same condition as (a), we get

$$p_f = \frac{3\mu u_0}{l^3} r^2 \left\{ 1 - \left(\frac{r}{r_0} \right)^2 \right\} \quad (25)$$

(c) Eqs. (23) and (25) can be applied in the

Fig. 4

Fig. 5

region $0 \leq r \leq r_0$. For the region $r_0 \leq r \leq r_2$ considering $W=0$ the same result as derived in 2.1 is obtained.

Combining the friction loss obtained above and the effect of kinetic energy, we get the static pressure distribution as follows:

$$r_0 \leq r \leq r_2$$

$$p = p_2 - \frac{\gamma}{2g} u^2 + 0.772 \frac{\gamma}{g} u_0^2 + \frac{6\mu Q}{\pi l^3} \log_e \frac{r_2}{r} \dots (26)$$

$0 \leq r \leq r_0$ for parabolic velocity distribution

$$p = p_2 + 0.772 \frac{\gamma}{g} u_0^2 - \frac{\gamma}{2g} \frac{u_0^2}{l^2} r^2 \left\{ 1 - \frac{1}{2} \left(\frac{r}{r_0} \right)^2 \right\} + \frac{3\mu u_0 r_0^2}{2l^3} \left[4 \log_e \frac{r_2}{r_0} + \left\{ 1 - \left(\frac{r}{r_0} \right)^2 \right\} \left\{ 3 - \left(\frac{r}{r_0} \right)^2 \right\} \right] \dots (27)$$

for uniform velocity distribution

$$p = p_2 + 0.772 \frac{\gamma}{g} u_0^2 - \frac{\gamma}{8g} \frac{u_0^2}{l^2} r^2 + \frac{3\mu u_0 r_0^2}{l^3} \left\{ 2 \log_e \frac{r_2}{r_0} + 1 - \left(\frac{r}{r_0} \right)^2 \right\} \dots (28)$$

Assuming the pressure behind the valve and p_2 are atmospheric and substituting $u_0 = Q/\pi r_0^2$, $u_1 = Q/2\pi r_0 l$, $u_2 = Q/2\pi r_2 l$, the thrust F is written as:

for parabolic distribution

$$F = 3\mu r_0^2 \left\{ \left(\frac{r_2}{r_0} \right)^2 - \frac{1}{3} \right\} \frac{Q}{l^3} + \frac{\gamma}{4\pi g} \left(0.314 - \log_e \frac{r_2}{r_0} \right) \left(\frac{Q}{l} \right)^2 \dots (29)$$

for uniform distribution

$$F = 3\mu r_0^2 \left\{ \left(\frac{r_2}{r_0} \right)^2 - \frac{1}{2} \right\} \frac{Q}{l^3} + \frac{\gamma}{4\pi g} \left(0.552 - \log_e \frac{r_2}{r_0} \right) \left(\frac{Q}{l} \right)^2 \dots (30)$$

2.5 Case when the flow is turbulent and the curve of the flow at the entrance is considered

Assume that the velocity distribution between the valve and the seat follows the Kármán's 7th-root law and the pressure loss is due only to viscous friction, and $W=u_0$.

Fig. 6

Applying the momentum theory in the direction of r to the small element shown in Fig. 6, we get

$$\frac{dp}{dr} + \frac{\tau_w}{l} + \frac{\gamma}{g} \frac{1}{r} \frac{d(ru_m^2)}{dr} = 0 \dots (31)$$

τ_w is written in a similar form to Eq. (13)

$$\tau_w = 0.03325 \frac{\gamma}{g} u_m^{7/4} \left(\frac{\nu}{l} \right)^{1/4} \dots (32)$$

The mean velocity u_m is written as

$$u_m = \frac{r}{2l} u_0 \dots (33)$$

Substituting Eqs. (32), (33) into Eq. (31) and integrating, we get the pressure distribution in the region $0 \leq r \leq r_0$

$$p - p_1 = 0.03325 \times \frac{2^{1/4} \gamma \nu^{1/4}}{11 g l^3} \left(\frac{Q}{\pi r_0^2} \right)^{7/4} (r_0^{11/4} - r^{11/4}) + \frac{3}{8} \frac{\gamma}{g} \left(\frac{Q}{\pi r_0^2 l} \right)^2 (r_0^2 - r^2) \dots (34)$$

The pressure distribution in the region $r_0 \leq r \leq r_2$ is the same as Eq. (15) and p_1 can be calculated from this. Then Eq. (34) becomes

$$p = 0.03325 \frac{\gamma \nu^{1/4}}{2g l^3} \left(\frac{Q}{2\pi} \right)^{7/4} \left\{ \frac{16}{3} \left(\frac{1}{r_0^{3/4}} - \frac{1}{r^{3/4}} \right) - \frac{8}{11} \frac{1}{r_0^{7/2}} (r^{11/4} - r_0^{11/4}) + \frac{\gamma}{2g} \left(\frac{Q}{2\pi l} \right)^2 \left(\frac{1}{r_2^2} + \frac{2}{r_0^2} - 3 \frac{r^2}{r_0^2} \right) \right\} \dots (35)$$

The thrust can be calculated from Eqs. (15) and (35) as

$$F = \frac{\gamma}{8\pi g} \left(1.5 - 2 \log_e \frac{r_2}{r_0} \right) \left(\frac{Q}{l} \right)^2 - 0.03325 \frac{\gamma}{2g} \frac{\nu^{1/4}}{(2\pi)^{3/4}} \left(\frac{8}{5} r_2^{5/4} + \frac{132}{95} r_0^{5/4} \right) \frac{Q^{7/4}}{l^3} \dots (36)$$

3. Measurement of the thrust

3.1 Apparatus and method of experiment

Apparatus is shown in Fig. 7. Water was sent to the test valve from the overflow tank where the head of water was held constant. The arrangement of the test valve, the valve seat and the

Fig. 7 Experimental apparatus

casing is shown in Fig. 8. Diameters of the valves d_2 were 30 mm, 50 mm and 60 mm, and the diameter of the hole of the seat d_0 was 20 mm. On the valve surface there were some pairs of holes of 1.5 mm diameter for pressure measurement (Fig. 9 and Table 1). Each pair was symmetrical concerning the valve center and connected to the same tube so that the mean pressure might be measured. These pressures were measured with water column and mercury manometers. The microscope having a scale of 1/100 mm accuracy was used for the measurement of the lift of the valves. The water level in the casing was held constant by controlling the rate of the flow discharged.

3.2 Arrangement of the data The total pressure on the valves and the thrust can be calculated from the pressure distribution on the valve surface measured above.

The total pressure P is

Fig. 8 Construction of the test valve

Fig. 9 Position of the pressure holes

$$P = \int_0^{r_2} p \cdot 2\pi r dr = \int_0^{\pi r_2^2} p d(\pi r^2)$$

where p : pressure at r , r_2 : radius of the valve. Then P can be derived by surface integration of the curves of p against πr^2 , and F is

$$F = P - (\text{weight of the water over the valve}) + (\text{buoyancy})$$

F and the lift l are written in non-dimensional forms F/F_0 and l/d_0 respectively. F_0 is the total pressure when the valve was closed and $F_0 = \pi r_0^2 p_0$.

Discharge coefficient c is written as

$$c = \frac{Q}{2\pi r_0 l \sqrt{2gH_e}}$$

Where H_e : net head.

3.3 Experimental result Relations between the thrust and the lift for three kinds of valve are shown in Fig. 10. The thrust F decreases suddenly and becomes the lowest and then gradually

Table 1 Positions of the pressure holes

Unit : mm

d_2/d_0	1.5	2.5	3.0
p			
2	0	0	0
3	7.5	8.0	10.0
4	12.5	13.0	15.0
5	—	18.0	20.0
6	—	22.5	25.0

Fig. 10 Relations between the lift and the thrust in non-dimensional forms

increases with the increase of the valve lift l . The values of the lift corresponding to the lowest thrust are $l/d_0 = 0.048, 0.066, 0.070$ for three kinds of valve respectively and changes a little with d_2/d_0 .

The relations between the flow rate Q and the lift l , and between the discharge coefficient c and the lift l are shown in Figs. 11 and 12 respectively. Though the change of Q with l is not large, that of c with l/d_0 is very large. This curve shows that the flow may be supposed to be laminar when l is small and c increases suddenly and becomes the highest and then gradually decreases with the increase of the valve lift.

4. Discussion

According to the analysis in Chapter 2 the friction loss and the pressure decrease due to kinetic energy have the main effects on the thrust of valves. The former is proportional to Q/l^3 when the flow is laminar and to $Q^{7/4}/l^3$ when turbulent, and the latter is proportional to Q^2/l^2 .

Fig. 11 Relations between the lift and the flow rate

Fig. 12 Relations between the lift and the discharge coefficient

The thrust is the sum of these terms and is written for laminar flow

$$F = A\mu \frac{Q}{l^3} - B \frac{Q^2}{l^2} + \pi r_0^2 p_0$$

where A, B : constants.

The relation between l and Q is similar to that shown in Fig. 11, and changes of the viscous term Q/l^3 and the term of the kinetic energy Q^2/l^2 can be calculated. Fig. 13 shows changes of these terms with l and their effects on the thrust of the valve. Similar result can be derived for the turbulent flow.

Analytical result in Chapter 2 and experimental results are compared in Figs. 14, 15 and 16.

According to the analysis of the flow between two parallel planes the parabolic velocity distribution is completed when $4\nu x/l^2 u_{m0} \geq 0.1$ (where x :

Fig. 13 Relation between the lift and the thrust

Fig. 14 Comparison of calculated thrust with measured one

distance from the entrance, u_{m0} : mean velocity)⁽⁵⁾
Applying this, we get the ranges of the lift within which the flow between the valve and the seat is laminar as follows:

$$l/d_0 \leq 0.006 \text{ for } d_2/d_0 = 1.5$$

$$l/d_0 \leq 0.009 \text{ for } d_2/d_0 = 2.5$$

$$l/d_0 \leq 0.016 \text{ for } d_2/d_0 = 3.0$$

These results show that the flow is laminar only when the lift is very small. From this point of view we can see that the analytical results for the valves with wide overlap of the valve and the seat (or large d_2/d_0) especially for $d_2/d_0 = 3.0$ agree with the experimental results well. For a large l/d_0 analytical results for a turbulent flow especially when the curve at the entrance is considered and those for a laminar flow when the inertia term is considered agree with experimental results in their inclination.

Experimental and analytical results of the pressure distribution in the r -direction are shown in Fig. 17. The diameter d_2 and the lift l of the valve are 50 mm and 2.12 mm respectively and this corresponds to the point $l/d_0 = 0.106$ when the thrust is gradually increasing in Fig. 15. These

curves show the inclination of the pressure distribution on the valve surface. At the entrance of the space between the valve and the valve seat the flow area is the smallest and the velocity is the fastest and the pressure becomes the lowest and negative. Then the static pressure increases with the increase of r to the pressure in the casing. The pressure distribution inside the hole of the seat is not uniform and even negative when $r=r_0$ though this was assumed in some analyses. For a considerably large valve opening the flow is supposed to be partial and the condition may be different from that assumed in the analyses. The analytical results for a turbulent flow when the curve is considered and for a laminar flow when the inertia term is considered agree pretty well.

This pressure distribution curve is the typical example of the case when the thrust is negative. For a very small lift of the valve the pressure in the region $r_0 \leq r \leq r_2$ is negative and its absolute value is small, and the distribution is approximately uniform. With the increase of the lift the pressure between the valve and the seat decreases, especially the pressure at the entrance decreases

Fig. 15 Comparison of calculated thrust with measured one

Fig. 16 Comparison of calculated thrust with measured one

extremely and the pressure inside the hole of the valve seat becomes non-uniform. Near the point of the lowest thrust the pressure at the entrance is very low and the pressure between the valve and the seat increases rapidly toward the valve end. For a large lift the pressure between the valve and the seat increases and the point of the lowest pressure drifts toward outside. This is supposed to suggest

Fig. 17 Pressure distribution on the valve surface

Fig. 18 Relation between the lift and the pressure on the valve surface

the effect of the partiality of the flow. For valves with $d_2/d_0=1.5$ the absolute value of the negative pressure between the valve and the seat is small and the pressure becomes positive at once with the valve opening. This pressure change has not so large effect on the thrust as the supply pressure. These pressure changes are shown in Figs. 18, 19 and 20.

As the result of the discussion made above we can suppose that the flow between the valve and the valve seat may be treated as a laminar flow between the two parallel discs for a very low lift of the valves, but this range is very small and the flow readily becomes turbulent. In practical

Fig. 19 Relation between the lift and the pressure on the valve surface

Fig. 20 Relation between the lift and the pressure on the valve surface

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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