 Bulletin of JSME, Vol. 28, No. 244, October 1985

Paper No. 84—0018

Cavitation Phenomena and Performance of Oil Hydraulic Poppet Valve*
(2nd Report, Influence of the Chamfer Length of the Seat and the Flow Performance)

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The influences of the chamfer length of the valve seat on the flow performance and the cavitation phenomena of an oil hydraulic poppet valve are studied experimentally using the half cut model. It is made clear that the discharge coefficient becomes maximum when the ratio of the chamfer length S to the restriction height h is a specific value within $1 \leq \bar{S}/h \leq 3$ in the case of a diverging flow, and $S/h \leq 1$ in the case of a converging flow, if the valve lift $\bar{x}$ and Reynolds number R_e^* are fixed. The critical cavitation number Kc also becomes maximum in the same range of S/h as the above. In the case of a diverging flow with chamfer length of $S=(1\sim2.5)h$, a periodic flow turbulence occurs at the restriction under some pressure conditions, and it induces an unusual sound noise like a whistling with resonant vibration of the oil column in the pipe line.

Key Words : Cavitation, Fluid Power Systems, Poppet Valve, Half Cut Model, Flow Performance, Chamfer Length, Discharge Coefficient, Pressure Distribution, Unusual Sound Noise, Experimental Study

1. Introduction

Most oil hydraulic poppet valves have a chamfer on the valve seats, which is sometimes produced by machine tools and also formed by striking with the poppets in the other cases. The length of the chamfer is generally small, but the existence of the chamfer has a large effect on the valve performances. Its effects on the discharge coefficient [1]~[4], thrust coefficient [5][6], instability [7][8] of the valve, etc., have been studied. However, the cavitating conditions have not been directly investigated in those works. There are only a few works by Aoyama [9]~[12] concerning the cavitation in a poppet valve, as far as we know.

In our previous report [13], appearance of cavitation and the effects of cavitation on the flow characteristics of a hydraulic poppet valve with a relative large chamfer have been made clear through direct observations of cavitation and detailed measurements of pressure distributions, by using a half cut model which was manifactured by cutting the original valve in half and covering the cut surface with a transparent perspex plate.

In the present paper, several experiments are made by changing the chamfer length of the half cut model for the purpose of studying the influence of the chamfer length on the flow performance and the cavitation characteristics of a poppet valve. Observation of cavitation is carried out by naked eye and photographic means; and the pressure distribution within the restriction, the limitation for the cavitation occurrence and the sound noise level are measured. It is made clear that the ratio of the chamfer length $\bar{S}$ to the height of the restriction h has a large influence on the valve performance. Within the specified range of $\bar{S}/\bar{h}$, the incipient cavitation number has a maximum due to the jet turbulence. It is also found that the jet turbulence induces an unusual loud noise by resonant vibration with the oil column in the pipe line, under several valve conditions.

2. Test apparatus and method of arrangement of results

Test apparatus and method of measurement are the same as in the previous report [13]. For the purpose of studying the influence of the chamfer length, four valve seats with different S as shown in Fig.1 are prepared. The vertical angle of the tapered surface of the chamfer is the same as the poppet angle $2\phi=90°$ in every case.

Figure 2 shows the hydraulic circuit used for the experiment. Oil is supplied to the test poppet valve through a 10 μm line filter. Inlet and outlet pressures are regulated respectively with relief valves. When a much lower outlet pressure is required, the throttle valve is opened. The flow rate is measured by weighing method and the noise level by the sound level meter set at 10 cm from the test

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valve. Oil temperature is measured with a thermistor thermometer in the upstream pipe line and it is held at $40\pm1°C$. The oil is Daphne Hydraulic Fluid 56, and the density ρ is 851 Kg/m^3 and the viscosity is 4.6×10^{-2} Pa.s at 40°C. The air content and the critical pressure for air separation are approximately the same as the values obtained by Hibi [13]. Measurements are usually started after preparatory running of about 30 minutes. The experimental results are arranged as follows.

The discharge coefficient is defined by Eq.(1).

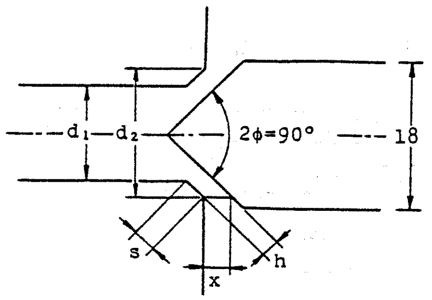
$$C=Q/\{a_1(x)\sqrt{2\Delta P/\rho}\} \qquad \cdots\cdots\cdot (1)$$

where Q is the volumetric flow, $\Delta P(=P_1-P_2)$ the pressure difference and $a_1(x)$ the cross sectional area of the flow passage at one end of the restriction, which is calculated by Eq.(2).

$$a_1(x)=\frac{1}{2}\pi d_1 x\sin\phi\left(1-\frac{x}{d_1}\sin\phi\cos\phi\right) \qquad \cdots (2)$$

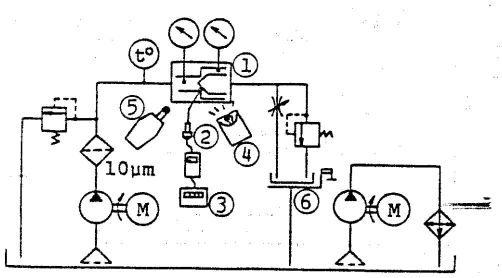
P_1 and P_2 are absolute inlet and outlet pressures, respectively, d_1 the valve seat diameter, x the valve lift and ϕ the half poppet angle.

Critical cavitation number K_c and Reynolds number R_e^* are defined by Eqs.(3) and (4), respectively.



Valve Seat No.	d_1 mm	d_2 mm	s mm
1	12.00	—	0
2	12.00	12.92	0.65
3	12.01	13.87	1.32
4	12.05	15.33	2.32

Fig.1 Dimensions of valve seats



① Test Poppet Valve ② Pressure Transducer
③ Digital Volt Meter ④ Strobo Light
⑤ Sound Level Meter ⑥ Platform Scale

Fig.2 Hydraulic circuit for the experiment

$$K_c=P_2/\Delta P \qquad \cdots\cdots\cdot (3)$$

$$R_e^*=(h/\nu)\sqrt{2\Delta P/\rho} \qquad \cdots\cdots\cdot (4)$$

The ratio S/h is used as a factor indicating the length of the chamfer. Where, S is the actual length of the chamfer and $h(=x\sin\phi)$ the height of the restriction. Even for the same value of S/h, there is a little difference between the following two cases; in one x is variable with S fixed and in the other S is variable with x fixed. Namely, since the rate of alteration of the cross sectional area of the flow passage within the restriction per unit circumference length of the valve seat γ is expressed as Eq.(5), γ changes with x.

$$\gamma=\frac{a_2(x)-a_1(x)}{S}\Big/\left(\frac{\pi d_1}{2}\right)$$
$$=2x\sin^2\phi/d_1 \qquad \cdots\cdots (5)$$

where, $a_1(x)$ is given by Eq.(2) and $a_2(x)$ is obtained by substituting d_2 for d_1 in Eq.(2). S/h is therefore varied by changing S with x fixed, in order to eliminate the effect of alteration of γ.

3. Case of diverging flow

Figure 3 shows the changes in the discharge coefficient C, the noise level N and the representative pressure P_{*-*} in the restriction, when P_2 is gradually reduced with P_1 fixed at a constant pressure of 5 MPa(abs.) and x at 0.8 mm. P_{*-*} is the pressure at the specified location along the surface of a valve seat, where the pressure indicates the most significant change with an increase of ΔP, so that the location differs for each valve seat. For example, the pressure P_{0-0} on the edge of the seat is plotted in the case of valve seat No.1, and the pressure P_{0-1} at the point of 0.1 mm inside of the restriction from the entrance corner is plotted in the case of No.2.

The mark ☛ in Fig.3 indicates cavitation inception, where cavities begin to appear near the surface of the jet in the downstream region. The mark ☞ shows the initiation of the "fixed cavitation" [14], where the cavities fixed to the surface of the seat appear continuously at the entrance corner of the restriction and (☞) shows the occurrence of the "traveling cavitation" [14], where cavities intermittently appear from the entrance corner and flow away along the surface of the jet.

It has been noted in the previous report that the pressure within the restriction rapidly drops and becomes a constant value near 0 MPa and the flow reaches the choking condition in the case of valve with chamfer when cavitation appears at the entrance of the restriction. If the flow pattern under such condition is assumed to be similar to that in the non-chamfered valve and P_2 is 0 MPa, the flow rate may be expressed as $Q=C_0a_1(x)\sqrt{2P_1/\rho}$. This equals Eq.(1), and the following equation is obtained.

$$C = C_0\sqrt{P_1/\Delta P} \quad \cdots\cdots\cdots \quad (6)$$

where C_0 is the discharge coefficient of the non-chamfered valve with P_2=0 MPa and takes a value of 0.766 from the data of valve seat No.1.

Since the flow condition in the case of S/h=0 is approximately similar to that of a sharp edged orifice, the discharge coefficient is relatively small as shown in Fig.3 due to an extreme contraction of stream line. The occurrence of cavitation has no significant effect on the discharge coefficient in this case. On the other hand, in the case of S/h=4.09, the flow separated from the seat surface at the entrance is considered to re-attach to the seat surface within the restriction and prevent the reversal flow to come into the restriction from the downstream side. Hence the pressure falls deeply below P_2 near the entrance of the restriction and the pressure difference which actually regulates the flow becomes large as ΔP increases. This causes an increase of C with ΔP. When the cavitation occurs at the entrance, the pressure within the restriction drops sharply and becomes a constant value of about 0 MPa. Since the flow saturates, C falls along the calculated result by Eq.(6) with an increase of ΔP as shown in Fig.3. The fact mentioned above is undestood from the pressure distributions in Fig.4. In the case of small S/h, the pressure within the restriction does not fall so deeply that it becomes nearly equal to 0 MPa. The value of C is therefore smaller than the calculated result by Eq.(6). In the case of S/h=1.15, the stream line is considered to re-attach to the surface of the seat within the restriction, when ΔP is less than 1 MPa. But, when ΔP exceeds 1 MPa the oil flow does not re-attach, and C becomes constant or decreases slightly with an increase of ΔP in the same manner as S/h=0. However, the value of C is larger for S/h=1.15 than for S/h=0 because the pressure falls below P_2 on the valve seat side and it makes the contraction of stream line weak. The inception of cavitation has no significant influence on the value of C because the cavities do not appear within the restriction but in the downstream region. However, when the travelling cavitation occurs at the entrance of the restriction, C begins to fall and it shows a similar tendency to the case of large S/h; namely the flow rate approaches the choke and C falls along the curve shown by Eq.(6).

It is noticeable that the noise level of S/h=2.33 is higher than the others, the reason for which will be given in the following chapters 5 and 6.

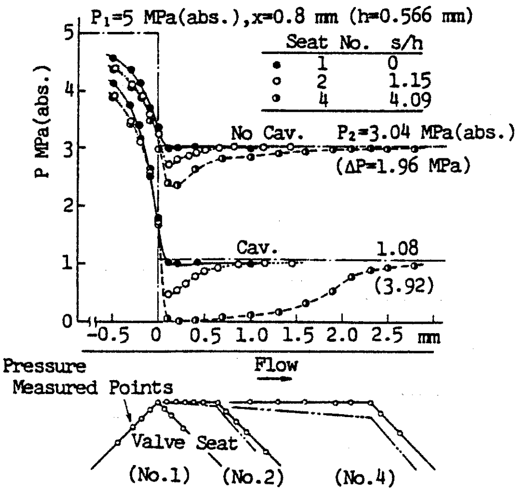


Fig.4 Pressure distribution on the valve seat (diverging flow)

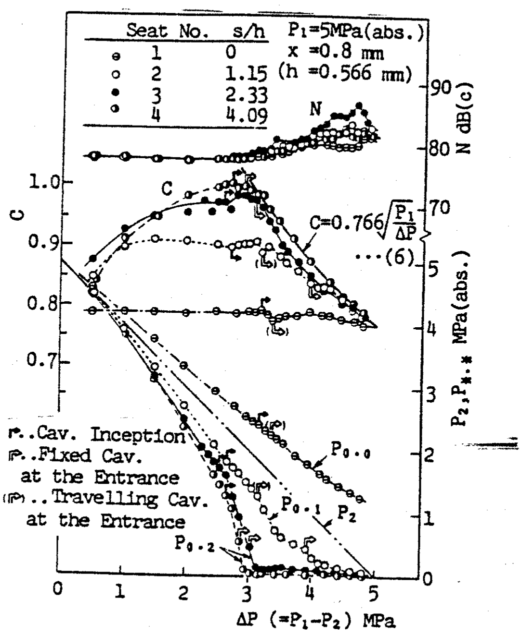


Fig.3 Flow characteristics (diverging flow)

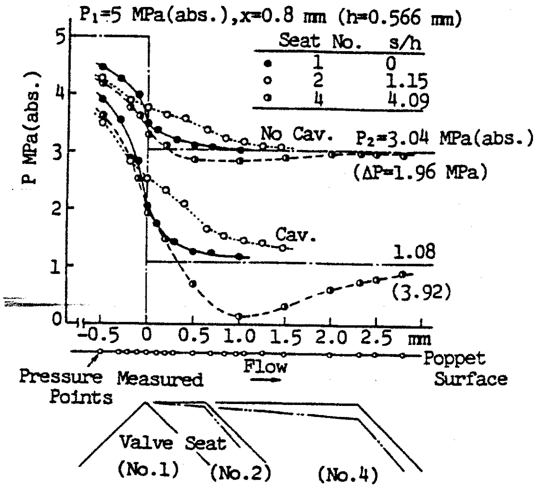


Fig.5 Pressure distribution on the poppet surface (diverging flow)

Figure 4 shows the pressure distributions along the surface of the seat. In the case of S/h=0, the pressure falls and becomes almost equal to P_2 at just behind the edge of the restriction. But in the other two cases with chamfered seat, the pressure falls more deeply below P_2 at just behind the entrance of the restriction. The pressure reduction is larger as S/h is larger.

Figure 5 shows the pressure distributions along the poppet surface. In the case of S/h=4.09, the pressure falls and comes close to 0 MPa at the middle of the restriction which approximately equals to that of the valve seat side. This suggests that the effect of turning the flow disappears at this point, and the flow re-attaches to the surface of the seat and goes straight along the flow passage after that. On the other hand, in the case of S/h=1.15, the pressure on the poppet surface is still higher than that of the valve seat side (see Fig.4) even in the downstream region. It suggests that the flow does not re-attach to the seat surface and turns even after passing the restriction. It is found from the direct observation that the cavities flow along the poppet surface in the case of S/h=4.09, but in the cases of S/h=0 and 1.15, they slightly deflect to the valve seat side. The angle of the flow direction from the central axis is larger than ϕ. This seems to be due to the difference between the pressures on the poppet and on the seat surface.

Figure 6 shows the relation of discharge coefficients measured under non-cavitating condition to S/h, with x and R_e^* fixed. It is found that C becomes maximum at the specified S/h between 1 and 3 when x and R_e^* are small, and the value of S/h generally becomes large with an increase of x and R_e^*. Within the range of S/h less than a certain value at which C becomes maximum, the flow condition becomes close to that of a sharp edged orifice in which the flow does not re-attach. So C becomes small with a decrease of S/h due to the contraction of stream line. On the other hand, in the range of large S/h, the flow becomes apt to re-attach. Hence, C becomes smaller with a increase of S/h due to an increase of viscous resistance. After all, it is likely that C has a maximum for the critical value of S/h at the boundary between no re-attachment and re-attachment of the flow in the restriction.

It is also found that C is a function of S/h and R_e^* independently of x in the case of the diverging flow.

4. Case of converging flow

Figure 7 shows the change of C, N and $P_{s.s}$ with gradual reduction of P_2 under the condition of x=0.8 mm and P_1=5 MPa(abs.).

When cavitation occurs at the entrance of the restriction and the flow is saturated, the flow pattern within the restriction of the chamfered valve is assumed to be similar to that of non-chamfered seat valve of which the diameter is d_2 and P_2 is 0 MPa. On this assumption, the flow rate is expressed as $Q=C_0 a_2(x)\sqrt{2P_1/\rho}$. Equating this to Eq.(1), the following equation is obtained for the discharge coefficient.

$$C=C_0\{a_2(x)/a_1(x)\}\sqrt{P_1/\Delta P} \quad \cdots\cdots \quad (7)$$

As the value of C_0, 0.754 is obtained from the result in the valve seat No.1. The calculated results for the four valve seats are shown by thick lines in Fig. 7.

C is smaller in the case of S/h=0 than in the other cases due to the effect of contraction. It is found from Fig.9 that the pressure falls slightly with the occurrence of cavitation in the restricted part and becomes less than P_2 in the downstream side. This pressure reduction causes a little increase of C.

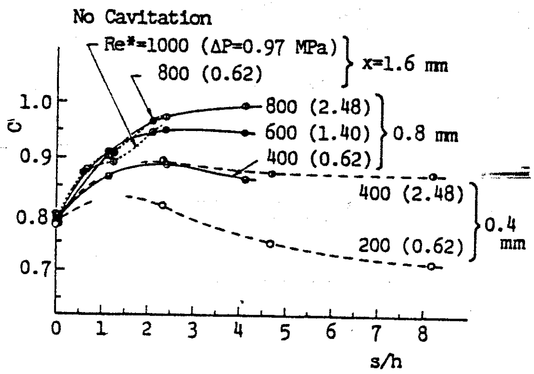


Fig.6 Discharge coefficient versus S/h (diverging flow)

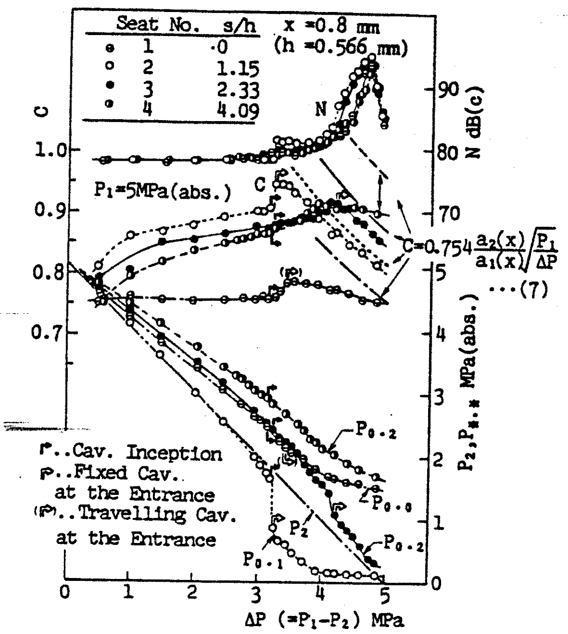


Fig.7 Flow characteristics (converging flow)

In the case of S/h=1.15, the flow is assumed to separate from the surface of the seat at the entrance corner but soon re-attach to the surface within the restriction. Since the pressure falls more deeply near the entrance of the restriction and the contraction of stream line is not stronger than in the case of S/h=0, C in this case shows the largest value within the non-cavitating region. When cavitation occurs at the entrance, the pressure within the restriction drops immediately and the discharge coefficient increases discontinuously. At the same time, the noise becomes larger. A similar discontinuous change of the flow has been reported with a short cylindrical choke [16]. The position of the incipient cavitation is much nearer to the restriction in the cases of S/h=1.15 and 0 than in the other two cases. This is explained as follows, that is, as S/h is small the pressure gradient within the restriction is large, and the flow passes the restriction before the boundary layer develops completely. Hence the shear between the jet surface and the stationary oil occurs violently near the outlet of the restriction in the downstream region. The cavities appear near the outlet of the restriction and occupy the gap between the jet surface and the seat wall. Hence, the pressure in the neighbourhood of the restriction suddenly falls below P_2. It makes the separation of the stream line at the entrance corner extremely severe, and induces a sudden occurrence of cavitation at the entrance. The pressure distributions before and after cavitation occurrence are shown in Fig.8. In Fig.7, the pressure $P_{0\text{-}1}$ still falls with an increase of ΔP even after the occurrence of cavitation, but C begins to go down because the area of the flow passage is contracted by growth of the cavities from the entrance. In the region where $P_{0\text{-}1}$ becomes constant, the seat surface is completely covered with cavities and the saturation of flow happens in the same manner as in the diverging flow. Hence the value of C decreases along the curve by Eq.(7). However, C indicates a little lower values than the curve since the pressure within the restriction is somewhat higher than 0 MPa.

The above mentioned tendency becomes ambiguous in the case of large S/h. As the position of cavitation occurrence is relatively far from the outlet of the restriction, the occurrence of cavitation has little effect on the value of C. Cavitation is not observed at the entrance in the case of S/h=4.09. Therefore, C increases simply with ΔP.

Noise level is higher in the cases of S/h=1.15 and 2.33 than in the other two cases.

Figure 9 shows the pressure distributions along the valve seat. The pressure falls below P_2 in the downstream region as cavitation occurs in the case of S/h=0. In the case of S/h=1.15, the pressure falls

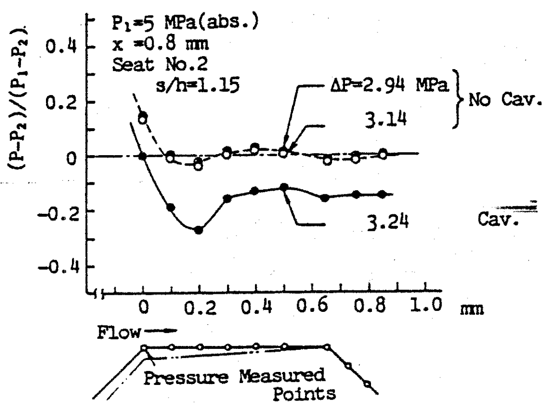


Fig.8 Change of pressure distribution with occurrence of cavitation

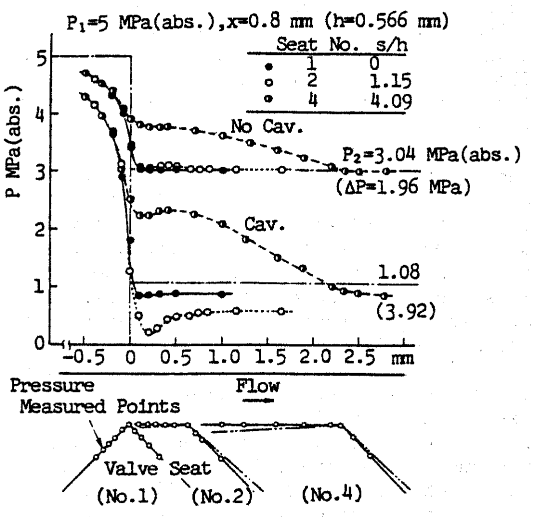


Fig.9 Pressure distribution on the valve seat (converging flow)

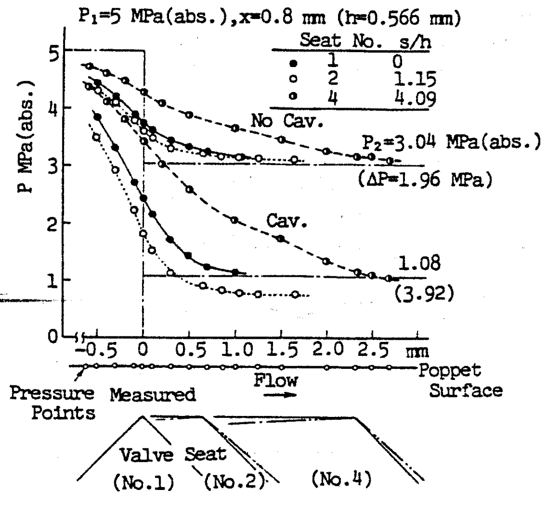


Fig.10 Pressure distribution on the poppet surface (converging flow)

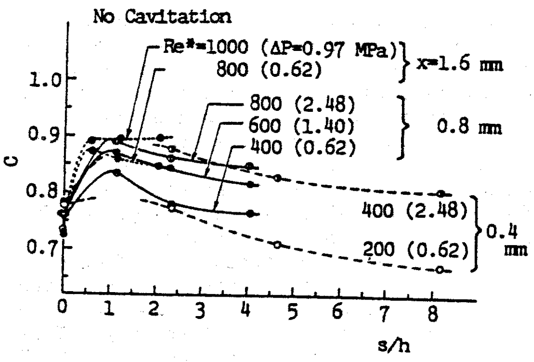


Fig.11 Discharge coefficient versus S/h (converging flow)

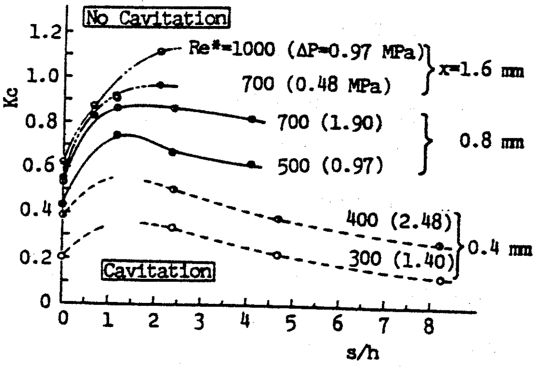


Fig.12 Critical cavitation number for the inception (diverging flow)

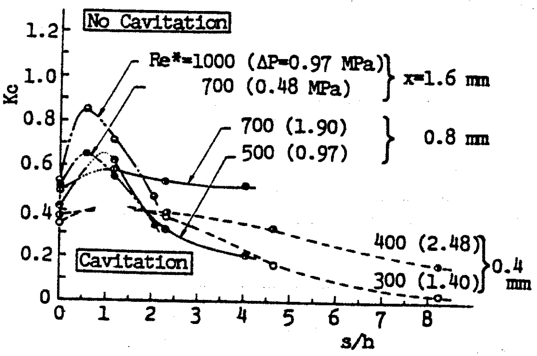


Fig.13 Critical cavitation number for the inception (converging flow)

more sharply and becomes nearly equal to the atmospheric pressure just behind the entrance corner of the restriction and subsequently recovers somewhat, but it is yet approximately 0.4 MPa lower than P_2 even after the flow passes through the restriction. Especially, in the case of a converging flow, the pressure does not easily recover once it falls down, because the cross sectional area of the jet becomes smaller even in the downstream side after passing through the restriction.

Figure 10 shows the pressure distributions along the poppet surface. In the case of S/h=1.15, also the pressure becomes about 0.3 MPa lower than P_2 in the downstream region under the cavitating condition. It is supposed that the fact mentioned above has a major influence on the thrust performance of a poppet valve.

Figure 11 shows the relation of discharge coefficients measured under the non-cavitating condition and S/h, with constant x and R_e^*. The value of S/h at which C has a maximum is nearly equal to unity, and it becomes smaller than unity with an increase of x. This tendency is opposite to the diverging flow case. This seems to be caused by the difference of γ in Eq.(5). That is, in the case of the diverging flow, the flow separates more easily from the valve seat as γ is larger. On the other hand, the tendency is reversed in the case of the converging flow. Hence, the value of S/h at which $\bar{C}$ has a maximum becomes smaller in the case of the converging flow. It is also a different tendency from the diverging flow that C is not expressed as a function of S/h and R_e^* independently of x. C becomes larger as x is smaller even if S/h and R_e^* are constant. This seems to be due to the difference of γ. Namely, since γ becomes smaller as x is smaller, the separation of flow line becomes stronger.

5. Critical condition of cavitation occurrence

The critical cavtation number K_c for cavitation inception is indicated as a function of S/h with constant R_e^* in Figs.12 and 13 for the diverging and the converging flow, respectively. It is clear from these figures that K_c has a maximum for a specified value of S/h. In the case of a diverging flow, the specified S/h is approximately between 1 and 2, and tends to become larger with an increasing x and R_e^*. On the other hand, the specified value of S/h is nearly equal to unity and tends to be less than unity when x and R_e^* become larger in the case of a converging flow. These tendencies are very similar to the relation of C to $\bar{S}/h$ shown in Figs.6 and 11. It is therefore supposed that $\bar{K_c}$ has a maximum for the specified $\bar{S}/h$ where the flow is at the boundary condition between no re-attachment and re-attachment in the restriction. Lichtarowicz [17] has indicated that the flow in the nozzle-and-flapper valve was apt to be unstable when the ratio of the width of land to the gap was between 1 and 5. Thus, it is also supposed in the case of poppet valves that the flow becomes unstable and cavitation comes to occur easily when the ratio of $\bar{S}$ to h is nearly equal to the value as mentioned above.

It has been shown in the first report [13] that K_c became a function of $\bar{R_e^*}$ irrespective of x. But, according to the more detailed examination in the wide range of x and S/h, K_c has the same tendency as result of the previous report in the case of diverging flow, but it shows a large scatter in the case of converging flow. This seems to be due to the difference of γ in the same manner as $\bar{C}$ in Fig.11.

6. Occurrence of unusual sound noise

Figure 14 shows the change of the noise level when P_2 is gradually reduced with P_1 fixed at 5 MPa(abs.) in the case of a diverging flow with the valve seat No.2. In the case of S/h=1.54, an unusual sound noise which is different from the noise emitted by cavitation, occurs suddenly when ΔP reaches 4 MPa; that is, immediately after the cavitation occurrence at the entrance. When ΔP increases and becomes 4.4 MPa, the unusual sound suddenly disappears. Under the microscope, the periodic occurrence of vorticies is observed near the outlet of the restriction when the unusual sound is heard. At the same time, it is also found that the direction of jet and the volume of cavities change every moment. An example of photograph is shown in Fig.15.

On the assumption that the unusual sound noise is caused by the resonant vibration between the periodic turbulence of jet and the oil column in the outlet pipe line, an experiment is carried out changing the pipe length L. As a result, it is found that the unusual sound occurs most significantly for a specified length of L. When L is shorter than the specified length, the frequency becomes higher, and when L is longer it becomes lower. But, in both cases, the unusual sound does not show a clear tendency due to the presence of an other noise with a different frequency component and the noise level falls down. Figure 17 shows the changes of P_1, P_2, N recorded by a digital memory scope with L=0.98 m. The fundamental frequency in this case is approximately 840 Hz, and the outlet pressure P_2 also shows a large variation at the same frequency. This length L is approximately equal to 0.57 times the wavelength, if the sound velocity is 1300 m/s in oil. The same unusual sound occurs under several other conditions, but only when the value of S/h is between 1 and 2.5 in the case of a diverging flow. In addition, a different unusual sound of which the frequency is about 3.2 kHz is heard, like a whistling, accompanied by the occurrence of cavitation in the case of seat No.4 with S/h=2.05. In this case, the vorticies are also observed within the restriction.

Judging from this phenomenon, it is understood that the flow tends to be very unstable when S/h is approximately between 1 and 2. It is interesting to know that the unusual sound in this test valve without moving parts is similar to the high frequency sound caused by the vibration of valve elements in actual valves. It seems necessary to investigate this phenomenon more elaborately in future.

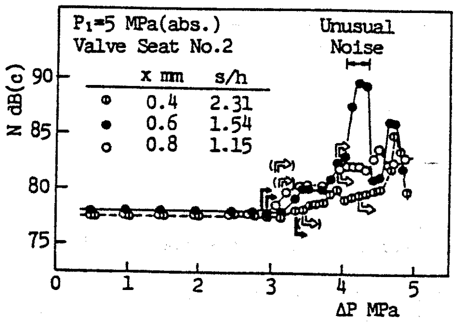
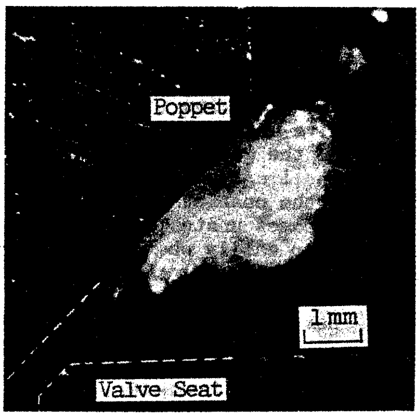


Fig.14 Noise performance (diverging flow, valve seat No.2)



Seat NO.2, x=0.6 mm
P_1=5 MPa(abs.), P_2=0.7 MPa(abs.)

Fig.15 An example of cavitating condition with unusual sound (exposure time 3×10^{-6}s)

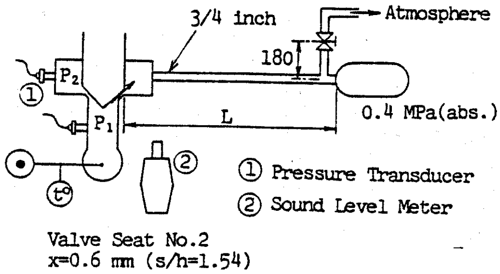
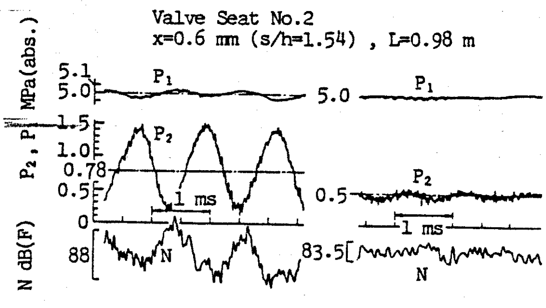


Fig.16 Schematic diagram of apparatus for the examination of unusual sound



(a) with unusual sound (b) without unusual sound

Fig.17 Waves of sound noise and pressure

7. Conclusions

The effects of the chamfer length of the valve seat on the flow performance and cavitation characteristics of an oil hydraulic poppet valve are experimentally investigated. Using the half cut model of a poppet valve, a direct observation of the cavitating flow within the restricted part and measurements of the flow rate, the detailed pressure distributions, the occurrence limits of cavitation and noise level are carried out. The results are summarized as follows:

(1) The valve performances depend strongly on the ratio of the chamfer length S to the height of the restriction h. In the case of a diverging flow, the saturation of flow rate with occurrence of cavitation at the entrance appears more clearly as S/h is larger. On the other hand, in the case of a converging flow, the pressure within the restriction and the flow rate changes discontinuously and then saturates with the occurrence of cavitation at the entrance in the case of a relatively small chamfer as S/h=1.15. When S/h is large, such discontinuity and the flow saturation do not occur clearly.

(2) The discharge coefficient under the non-cavitating condition has a maximum for the specified value of S/h with constant x and R_e^*. In the case of a diverging flow, the specified S/h is almost between 1 and 3, and has a tendency to become larger as x and R_e^* are large. Besides, in the case of a converging flow, the specified S/h is approximately unity or less than unity. It is supposed that, for this specified S/h, the flow is at the boundary condition between no re-attachment and re-attachment in the restriction.

(3) The relation of the critical cavitation number K_c to the value of S/h is very similar to the discharge coefficient C to S/h which is mentioned in conclusion (2). In the case of a diverging flow, the value of S/h at which cavitation occurs most easily is between 1 and 2, and it has a tendency to become larger as x and R_e^* are large. On the other hand, in the case of a converging flow, S/h is equal to or less than unity. It is assumed that the flow condition becomes unstable and the turbulence induces the occurrence of cavitation for this critical value of S/h.

(4) It is found that an unusual high frequency sound occurs when S/h is between 1 and 2.5 in the case of a diverging flow. It is considered to be caused by the resonant vibration between the periodic turbulence of jet and the oil column in the pipe line.

Authors wish to thank A. Hibi; Assistant Professor of Toyohashi University of Technology for his advice on the study, and A. Takeshita for his carrying out the experiments.

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