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# EFFECT OF CAVITATION IN WATER HYDRAULIC POPPET VALVES

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**Abstract**

In this paper, cavitation in water hydraulic poppet valves is investigated by an experimental method with a half cut test model. The situation of cavitation appearance, the effects of cavitation on the characteristics of flow rate, noise level, pressure distributions and the boundary of inception of the cavitation are investigated. Comparison between a poppet valve with sharp edged seat and another, which has a length on the seat, is made. The effects of change in the shape of the seat are discussed as well as the effect of cavitation appearance. As a result, it is revealed that the sharp edged seat valve is less influenced by the cavitation on its characteristics.

**Keywords:** hydraulics, water, cavitation, poppet valve, shape of valve seat, half cut model, experimental study

## 1 Introduction

Poppet valve is one of seat valves which have advantages especially when used in water hydraulic systems where tap water is used as a fluid medium. There is possibility in water hydraulics to increased leakage due to the low viscosity of water and to raised cost of components because some special materials are used to avoid erosion and corrosion. Therefore, simpler and less leaky construction is more important than in other fluid power systems (Mauer, 1995).

Water hydraulics has many advantages; environmental friendliness, non fire risk, non-toxic, easy availability and others (Varandili, 1999). The applications have become more common e.g. in food processing industries and a trend seems to be upwards. However, there are still problems to be solved for the water hydraulic systems so that they could be used commonly in industries. One of the problems is to get components with longer lifetime. Cavitation is one of major factors to shorten the life of water hydraulic components in addition to the less lubricant ability and corrosive properties of water (Backé, 1999).

Generally, cavitation causes undesirable problems in the fluid power systems and components - for example; efficiency reduction, increase of vibrations and noise, unexpected change in the characteristics of flow rate and flow force (Oshima and Ichikawa, 1985, 1986) and in the most violent case the erosion of components. Many studies on cavitation in oil hydraulic valves have been done up to the present, but not yet so many studies in water hydraulic valves. Only erosion due to impingement of cavitating water jet has been studied actively (Yamaguchi, Kazama and Wang, 2000).

It is important to know fundamentally the cavitation phenomenon and the effects on the characteristics of water hydraulic valves in order to research and develop the water hydraulic valves with longer life and higher quality performances. In this paper, the cavitation phenomenon in two water hydraulic poppet valves - one with a sharp edged seat and another with a chamfered edge of the seat - is investigated experimentally with a half cut test model. The situation of cavitation appearance, the effects of cavitation on the characteristics of flow rate, noise level, pressure distributions and the boundary of inception of cavitation are investigated. The results are discussed comparing between the both different shaped valves and the effect of change in the shape of valve seat is revealed. The flow situation in the half cut valve is a little different than in the real poppet valve because there is a zero-velocity condition on the cut plane. This may have an effect to the discharge coefficient of the valve. It has been confirmed experimentally with oil hydraulic poppet valves that there is little difference between the flow rates with actual valve shaped model and a half cut model (Oshima and Ichikawa, 1985).

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## 2 Experimental Method

Figure 1 shows the structure of the test apparatus of half cut test model of a poppet valve and Fig. 2 shows the overview of the main part and the important dimensions. Two different shaped valves are used for the tests; the valve No. 1 has sharp edged seat and No. 2 has 1.2 mm length on the seat.

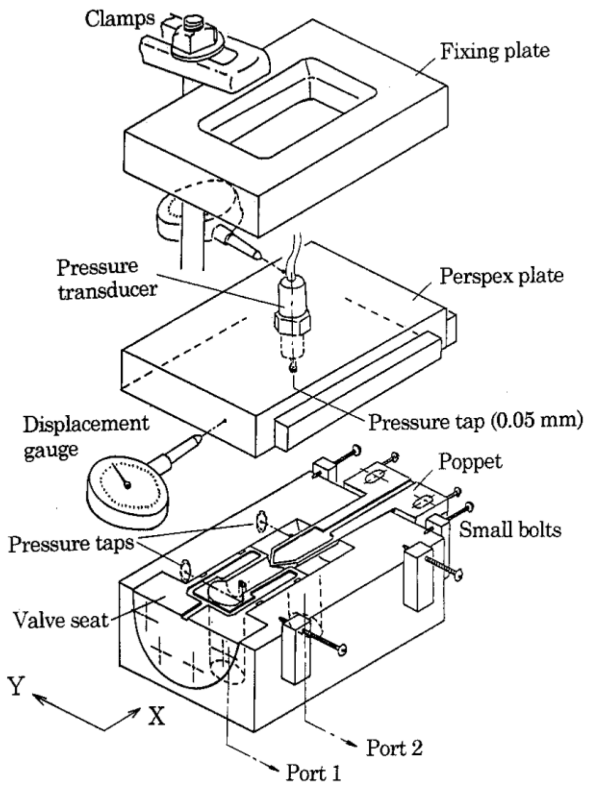
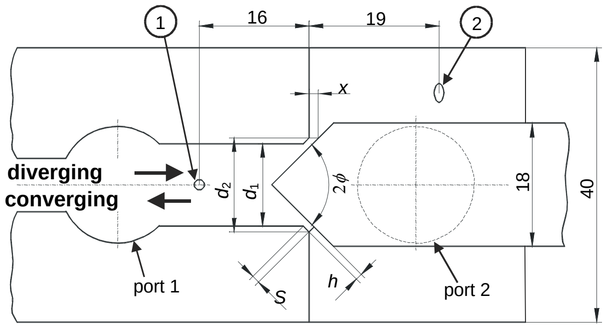



**Fig. 1:** *Structure of half cut test model*




| Valve No. | $d_1$ (mm) | $d_2$ (mm) | $S$ (mm) | $2\phi$ |
|---|---|---|---|---|
| 1 | 12 | 12 | 0 | 90° |
| 2 | 12 | 13.7 | 1.2 | 90° |

**Fig. 2:** *Overview of main part and the important dimensions*

The valve is cut along its longitudinal axis and the cut surface is covered with a transparent Perspex plate of 20 mm thickness. On the Perspex plate is piled a fixing plate of stainless steel with a square window and they are fastened against the cut surface of the valve body with 8 clamping arms and bolts. The body, the seat and the poppet are all made of stainless steel. To measure the pressure inside the valve, a pressure transducer is attached on the Perspex plate through the square window, as shown in Fig. 1.

The pressure is detected through a pressure tap of 0.05 mm diameter made by laser cutting on a stainless steel chip of 2 mm diameter that is fixed on the surface of Perspex plate. The pressure tap is able to be located at any position on the cut surface inside the valve because the Perspex plate is movable when the clamps are not tightened. It can be precisely located in X- and Y-directions using small bolts and its displacement can be measured with two dial displacement gauges.

When cavitation is observed, another Perspex plate without the pressure tap is used. The cavitation bubbles are observed through the square window on the Perspex plate using a repeatedly flashing stroboscopic light to illuminate the inside of the valve. To take photos, the shutter speed of a camera and the frequency of the stroboscopic light are adjusted so that only one flash can be caught during the shutter is open. The duration of one flash of the stroboscopic light is a few microseconds.

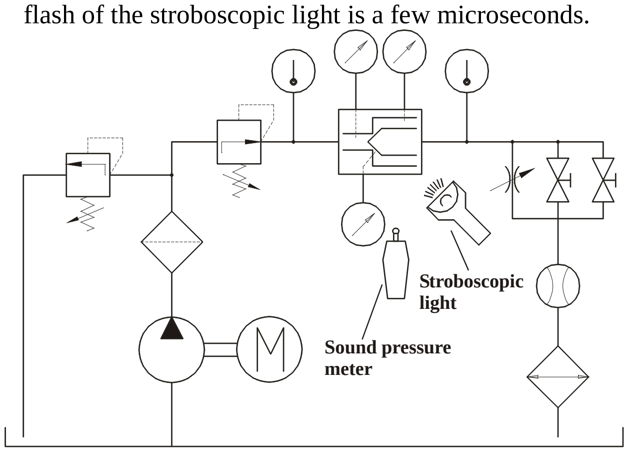



**Fig. 3:** *Hydraulic circuit used for tests*

Figure 3 shows the hydraulic circuit used for the tests. The inlet pressure is adjusted with a pressure reducing valve and the outlet pressure with throttle valves. The inlet and outlet pressures are detected with pressure transducers at the pressure taps indicated with circled numbers 1 and 2 in Fig. 2. The tests are carried out in two different flow directions, “diverging flow” and “converging flow”. The port-1 is the inlet and port-2 is the outlet in the case of diverging flow and vice versa in the case of converging flow. The water temperature is also measured at the inlet and outlet ports. Flow rate is measured on the downstream line with a magnetic flow meter. The data of pressures and flow rate are acquired with a computer and a data acquisition card. Sound pressure level is detected by a microphone located at 10 cm far from the valve body. Most tests are carried out under a constant inlet pressure of 5 MPa (abs.) and the water temperature from 25 to 30 °C. Water is circulated except a slight external drain during the tests. The absolute sound pressure of back-ground noise during the tests is approximated to be about 83-84 dB(A). However, the change of sound pressure is essential, not the absolute value.

The transducers used to measure the pressures at the inlet and outlet are Trafag NA100.0 V (0-10 MPa) and the absolute pressure transducer used to measure the pressure distribution is Kristal RAG25A50BV1H (0-5

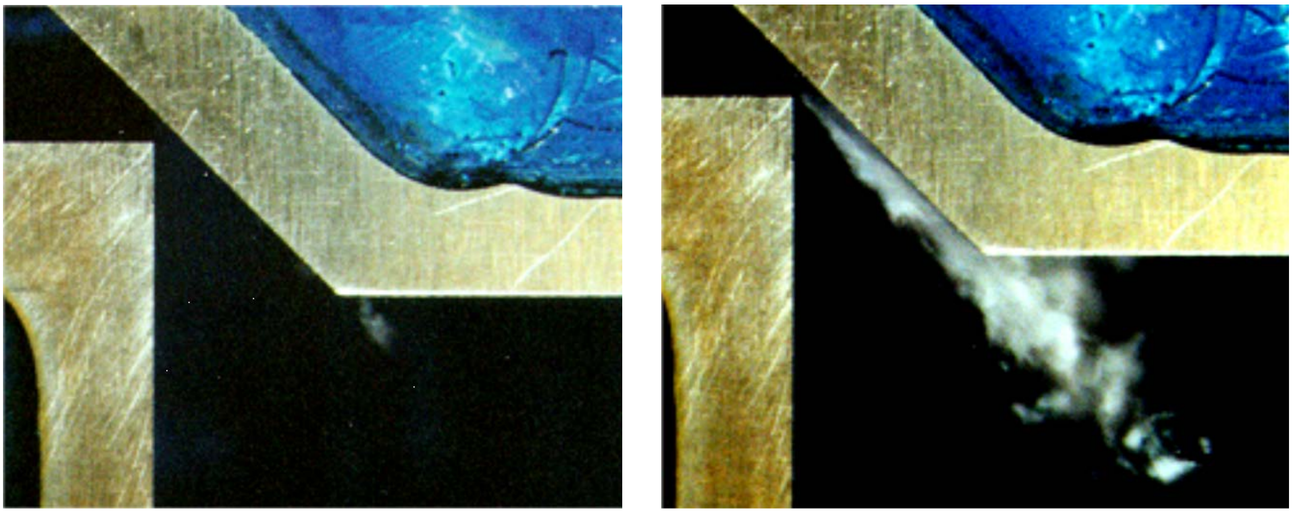
MPa abs.). The flow rate is measured with Heinrichs Messtechnik PIK 4.3000E DN 25 electromagnetic flowmeter. The measured signals mentioned above are read to Data Translation DT5743 acquisition card and current-voltage conversion is used when needed. Temperatures are measured and observed with Pt100-thermocouples and Nokeval 538-8 Multipoint Indicator. Sound pressure level is measured with Brüel & Kjær Impulse Precision Sound Pressure Meter model 2204 using Condenser Microphone Cartridge type 4133.

## 3 Process of Cavitation Appearance

### 3.1 Case of Diverging Flow

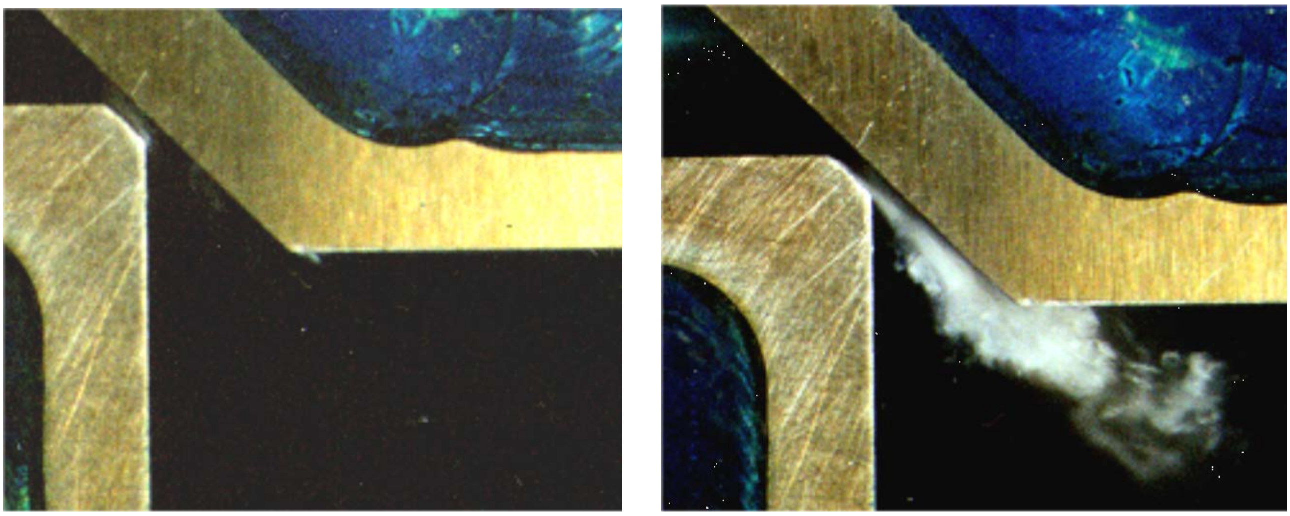
Figure 4 shows the situation of cavitation appearance in the case of diverging flow when reducing the outlet pressure and keeping the inlet pressure constant at 5 Mpa (abs.) and using the valve opening of 0.6 mm. In the case of valve No.1 (Fig. 4 (a)) when the downstream pressure $P_2$ is reduced and reaches 1.3 MPa (abs.), cavitation starts to appear in the downstream side slightly far from the sharp edged corner of the seat. Small bubbles appear intermittently for a moment at the location between the sharp edged corner of the seat and the downstream side corner on the poppet. They are rapidly carried by the flow along the surface of the poppet and disappear close to the downstream side corner of the poppet. In addition, some bubbles appear also just behind the downstream corner on the poppet like shown in the left photo in Fig. 4 (a).

Noise by cavitation starts to be heard slightly and intermittently in this situation. As the outlet pressure $P_2$ is reduced, amount of bubbles increases and the starting location of cavitation becomes closer to the sharp edged corner of the seat. When $P_2$ is reduced and reaches 0.8 MPa (abs.), bubbles appear continuously at the corner of the seat and travel along the poppet surface like shown in the right photo in Fig. 4 (a).



$P_2$=1 MPa (abs.) $P_2$=0.5 MPa (abs.)

**Fig. 4(a):** *Situation of cavitation appearance in diverging flow (Valve No. 1)*



$P_2$=2 MPa (abs.) $P_2$=0.6 MPa (abs.)

**Fig. 4(b):** *Situation of cavitation appearance in diverging flow (Valve No. 2)*

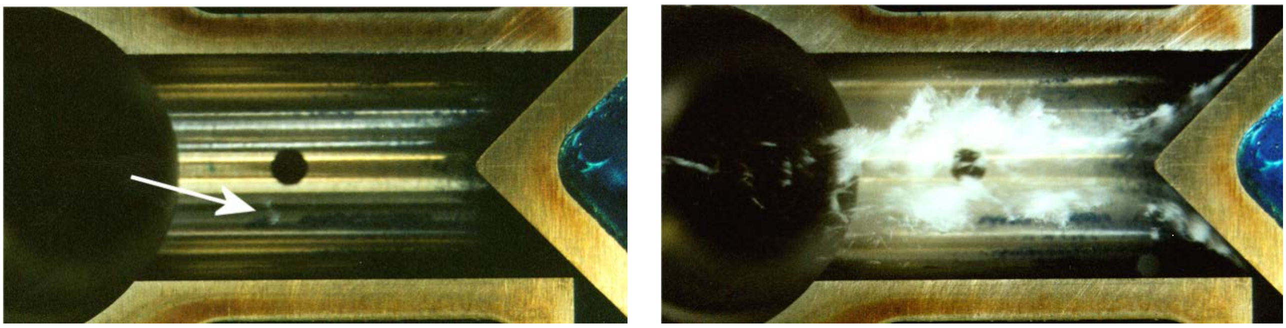
The amount of bubbles is looked as if fluctuating in high frequency. Much bubble appear also just behind the downstream side corner on the poppet. Most bubbles disappear rapidly near the downstream side of the corner on the poppet. Noise also becomes continuous and louder as $P_2$ is reduced. Figure 4 (a) shows the situation of cavitation when $P_2$ is 1 and 0.5 MPa (abs.).

In the case of valve No. 2 (Fig. 4 (b)), cavitation starts to appear at two positions - in the entrance corner of the orifice and just behind the downstream side of the corner on the poppet when $P_2$ reaches 2.6 MPa (abs). In this situation, intermittent weak noise is heard irregularly. When $P_2$ is reduced below 2.3 MPa (abs.), the amount of bubbles becomes larger and some bubbles start to appear intermittently also on the way to the downstream corner of the poppet like shown in the left photo in Fig. 4 (b). These bubbles disappear rapidly near the downstream corner of the poppet. Noise starts to become larger and continuous in this condition. The amount of bubbles increases and noise becomes louder as $P_2$ is reduced further. Figure 4 (b) shows the situation of cavitation when $P_2$ is 2 and 0.6 MPa (abs.).

### 3.2 Case of Converging Flow

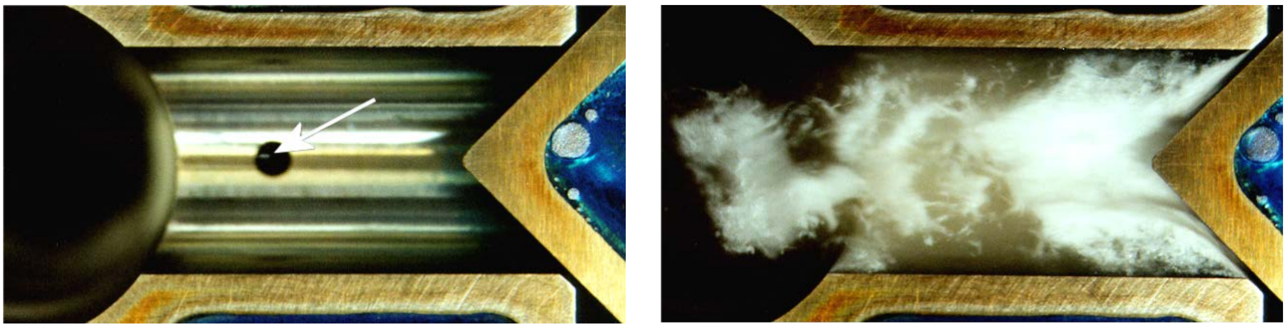
Figure 5 shows the situation of cavitation in the case of converging flow. The conditions of inlet pressure and valve opening are same as in the case of diverging flow in Fig. 4. In the case of valve No. 1 (Fig. 5 (a)), when $P_2$ is reduced and reaches 1.7 MPa (abs.), cavitation starts to appear in the down stream area relatively far from the sharp edged corner of the seat, but the location is not fixed. The bubbles appear intermittently for a moment, flow rapidly and disappear on the way to the outlet port. Small bubbles can be seen near the pressure tap in the downstream chamber in the left side photo in Fig. 5 (a). As $P_2$ is reduced, amount of bubbles increases and the starting location of cavitation closes on the corner of the seat. Most bubbles flow near the outlet port and disappear there rapidly. When $P_2$ reaches 0.8 MPa (abs.), bubbles start to appear at the corner of the seat and they are seen as travelling continuously along the surface of water jet like shown in the right photo in Fig. 5 (a). Noise also becomes more continuous and extremely louder as $P_2$ is reduced. The amount of bubbles fluctuates in high frequency. Most bubbles disappear rapidly near the entrance of outlet port. Figure 5 (a) shows the situation of cavitation when $P_2$ is 1.5 and 0.5 MPa (abs.).

In the case of valve No. 2 (Fig. 5 (b)), small bubbles start to appear and disappear intermittently along the surface of water jet in the downstream chamber when $P_2$ reaches 2.2 MPa (abs.). Weak noise is intermittently heard. As $P_2$ is reduced below 1.7 MPa (abs.), bubbles start to appear also at the inside of the orifice. They are seen to travel toward the downstream in short distance and disappear rapidly near the outlet of the orifice. Noise starts to be larger and continuous at this situation. When $P_2$ is reduced further, the amount of bubbles increases and the travelling distance becomes longer like shown in the right photo in Fig. 5 (b). A great amount of bubbles disappear rapidly near the outlet port and noise becomes extremely large. Fig. 5 (b) shows the situation when $P_2$ is 2 and 0.4 MPa (abs.).



$P_2$=1.5 MPa (abs.) $P_2$=0.5 MPa (abs.)

**Fig. 5(a):** *Situation of cavitation appearance in converging flow (Valve No. 1)*



$P_2$=2 MPa (abs.) $P_2$=0.4 MPa (abs.)

**Fig. 5(b):** *Situation of cavitation appearance in converging flow (Valve No. 2)*

## 4 Effects on Flow Rate, Noise and Pressure Distributions

### 4.1 Case of Diverging Flow

Figure 6 shows the discharge coefficients of valve No. 1 and No. 2 when $P_2$ is reduced while $P_1$ is constantly 5 MPa (abs.). Arrow marks in the figure show the inception points of cavitation. In the case of valve No. 1, the value of $C$ is almost constant independently of change in $\Delta P$. It does not change even if cavitation appears. On the other hand, in the case of valve No. 2, the value of $C$ is considerably larger than in the case of valve No. 1 and it decreases sharply with increase of $\Delta P$ when cavitation appears inside the orifice. Before the cavitation appearance, the value of $C$ is almost constant. When cavitation appears inside the orifice, so called "choking" phenomenon appears, i.e. the flow rate does not more rise in spite of the increase in pressure difference.

The pressure distributions in valves No. 1 and No. 2, measured along the surfaces of the valve seat and the poppet, are shown in Fig. 7. Although, in the case of valve No. 1, the pressure on the seat surface becomes almost equal to the outlet pressure $P_2$ just behind the corner of the seat, the pressure inside the orifice of valve No. 2 becomes lower than $P_2$. At a specific area inside the orifice, it becomes lower than the atmospheric pressure when the choking phenomenon appears. Since the minimum pressure inside the orifice becomes less than atmospheric pressure and is almost constant while $P_2$ is reduced further, the pressure difference, which defines the flow rate through the orifice, becomes almost constant. The flow rate, therefore, becomes constant in spite of increase of $\Delta P$. This phenomenon is not appearing with the sharp edged valve because there is not a real "inside" of the orifice.

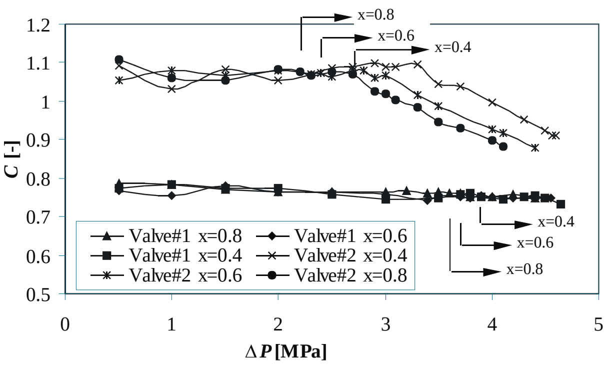



**Fig. 6:** *Discharge coefficient in diverging flow*

It is considered that the contraction of streamlines inside the orifice of valve No. 2 becomes less severe due to the presence of solid wall of the seat (Oshima and Ichikawa, 1986). The difference in the pressure distributions shown in Fig. 7 and the difference in the degree of contraction of streamlines inside the orifice are the major reasons for the difference of discharge coefficients between valve No. 1 and No. 2 shown in Fig. 6.

Sound pressure level of noise is shown in Fig. 8. It is found that the noise in valve No. 2 is considerably larger than in No. 1. The reason is that cavitation occurs more violently in the case of valve No. 2 due to the pressure reduction and higher flow velocity inside the orifice compared to the case of valve No. 1 under the same outlet pressure condition.

The locations of coordinate systems in both flow directions and both valve types while measuring the pressure distribution are presented in Fig. 10.

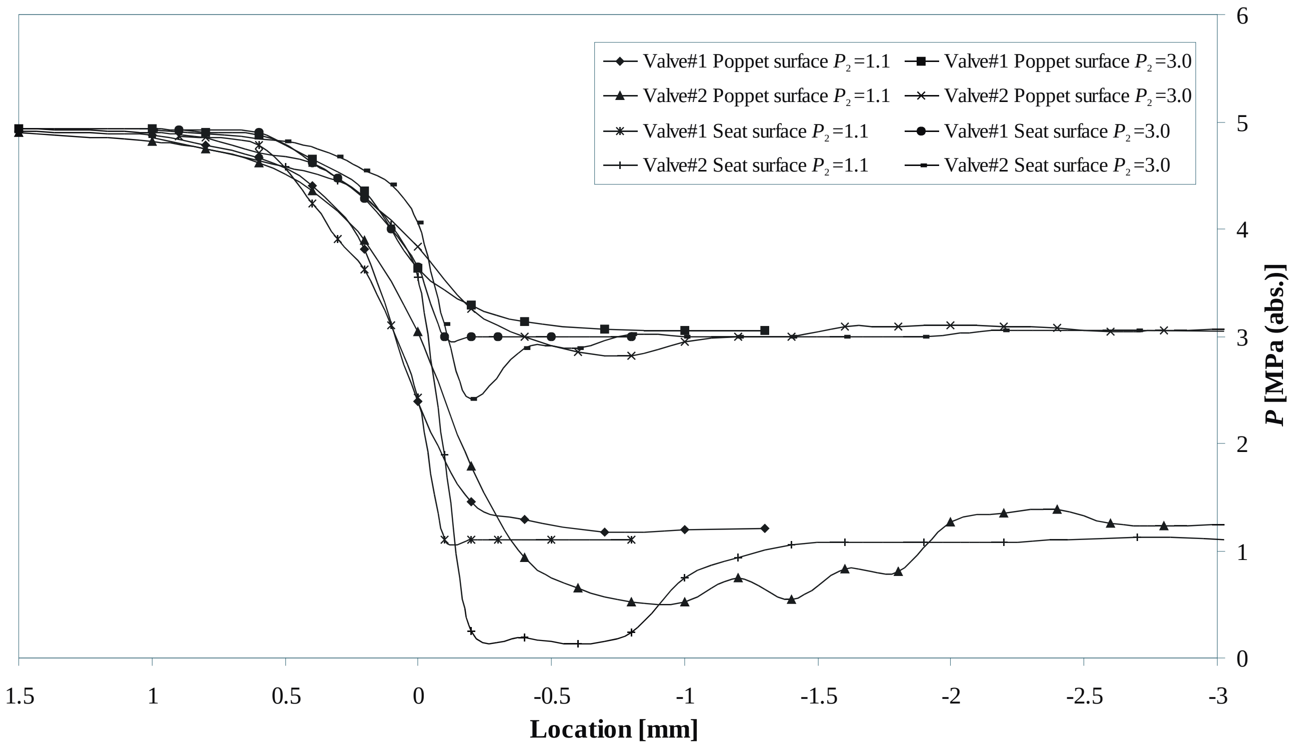



**Fig. 7:** *Pressure distribution in diverging flow*

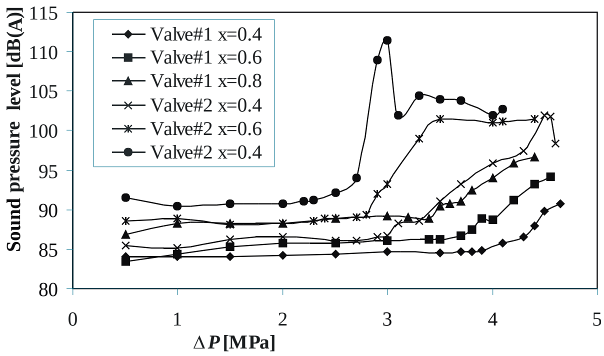



**Fig. 8:** *Sound pressure level of noise in diverging flow*

## 4.2 Case of Converging Flow

Figure 9 shows the discharge coefficients of valve No. 1 and No. 2 when $P_2$ is reduced while $P_1$ is constantly 5 MPa (abs.). There is no data of $x$ = 0.8 mm of valve No. 2 due to the limit of pump capacity.

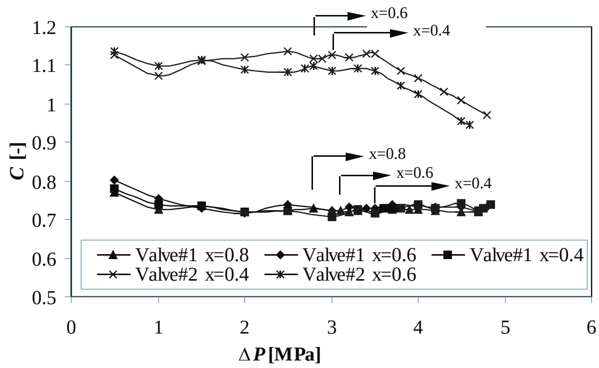



**Fig. 9:** *Discharge coefficients in converging flow*

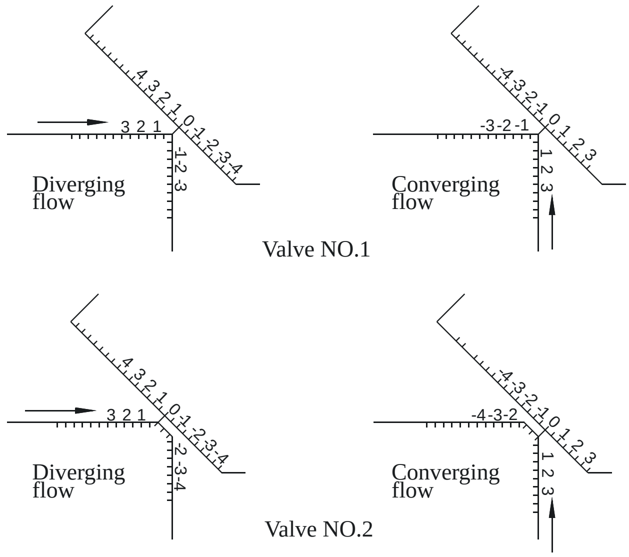



**Fig. 10:** *Systems of coordinates in pressure distribution measurements*

The results are almost similar as in the case of diverging flow. The value of $C$ in the case of valve No. 2 is considerably larger than in the case of valve No. 1. When cavitation appears inside the orifice, the choking appears also in this case.

The pressure distributions along the surfaces of the valve seat and poppet of valve No. 1 and No. 2 are shown in Fig. 11. In the case of valve No. 2, the pressure inside the orifice is below the atmospheric pressure at a specific position on the surface of the valve seat when the choking appears. On the other hand, in the case of valve No. 1, the pressure on the surface of seat becomes almost equal to the downstream pressure $P_2$ behind the corner of the seat similarly as in the case of diverging flow.

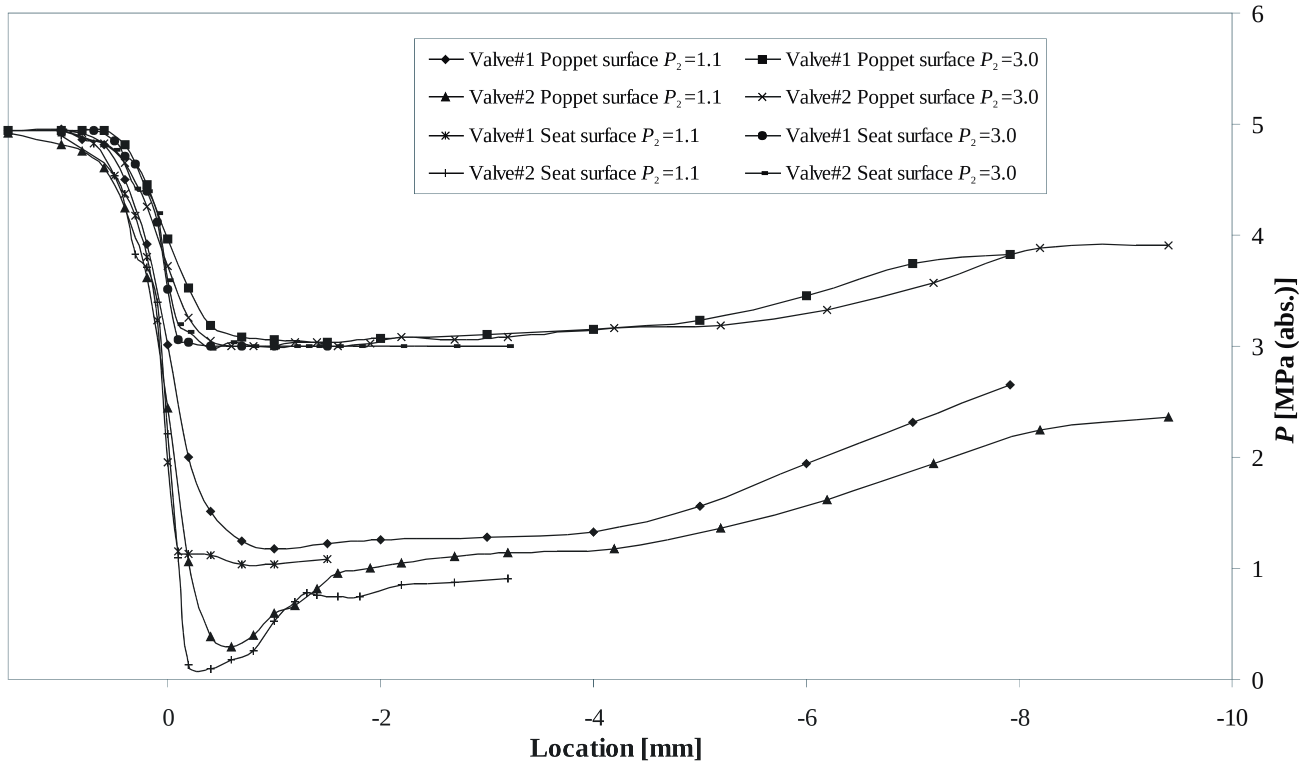



**Fig. 11:** *Pressure distribution in converging flow*

The pressure on the poppet surface goes up above the outlet pressure $P_2$ when the location closes to the top of the poppet. It becomes considerably higher than $P_2$ near the top of the poppet in the both cases of valve No. 1 and No. 2. Therefore, the cavitation bubbles are not seen near the poppet surface as shown in Fig. 5. This pressure rising is caused by the change in the momentum of water jet.

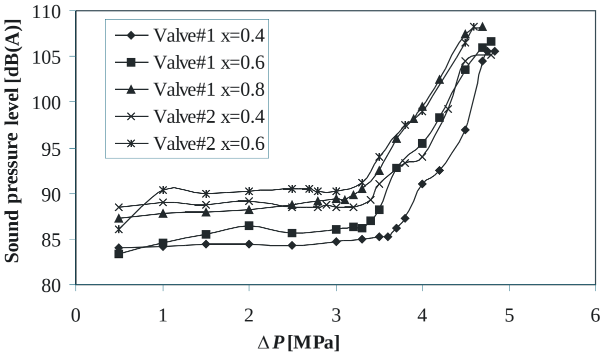



**Fig. 12:** *Sound pressure level of noise in converging flow*

The noise is shown in Fig. 12. Even in the case of valve No. 1, the noise is considerably large. The maximum value is 10 to 15 dB (A) larger than in the case of diverging flow. The difference between the cases of valve No. 1 and No. 2 is not so large as in the case of diverging flow. The reason is considered to be that most of the bubbles appear in the downstream chamber and collapse in the area relatively far from the outlet of the orifice. Bubbles grow during travelling to the outlet port and then collapse there violently. Therefore, it is considered that the existence of a length on the seat has not so big effect on the situation of collapse of cavitation bubbles.

The values of measured sound pressure levels are quite high in the whole pressure drop range for the both cases of diverging flow and converging flow. There is an effect by the reflection of noise from the shield wall which surrounds the test apparatus on the three sides. Therefore, the absolute values are not interested here.

## 5 Boundary of Cavitation Inception

Figure 13 shows the boundaries of cavitation inception in the case of diverging flow and converging flow of Valve No. 1 and No. 2. It is found that there is considerably large difference between the cases of valve No. 1 and No. 2 in the case of diverging flow (a), and the difference becomes considerably larger when the Reynolds' number increases. On the other hand, in the case of converging flow (b), there is not so large difference between the cases of valve No. 1 and No. 2.

When the results are compared between the diverging flow and converging flow, it is found that the value of $K$ for the inception of cavitation in diverging flow is smaller than in converging flow in the case of valve No. 1, but the tendency is inverse in the case of valve No. 2. The major reason in the case of valve No. 1 is considered to be that the reduction of flow velocity after going through the orifice in the case of diverging

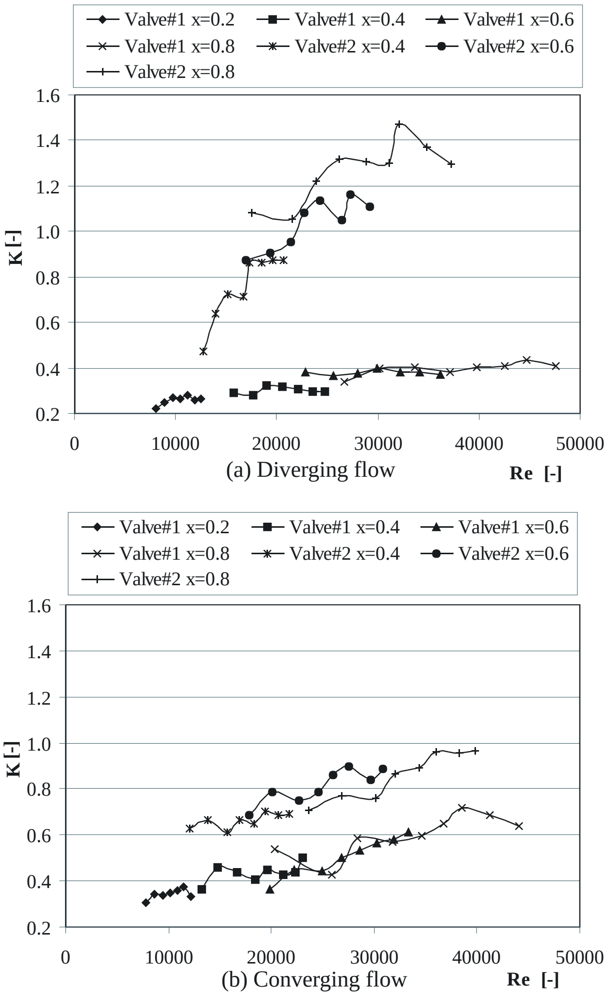



**Fig. 13:** *Boundary of cavitation inception*

flow is larger than in the converging flow. The major reason in the case of valve No. 2 is considered to be that the pressure reduction inside the orifice in the diverging flow is larger than in the converging flow under the same outlet pressure condition. However, in the both cases, it is found that existence of length on the seat has an effect to make the cavitation occurrence more easy.

## 6 Uncertainty of Measurements

The uncertainty estimates are defined for the plotted data. In most cases, the estimates should be defined for each measured and calculated point, because the accuracy of $P_1$, $P_2$ and $Q$ measurement depends on the measured value.

In Fig. 6 and 9 the maximum error in the value of discharge coefficient, when the pressure drop ($\Delta P$) is 2 MPa, is ±0.075. The uncertainty decreases when the pressure drop value increases and vice versa. The value given above is the maximum value in the whole flow rate range and the real error is mostly smaller than that.

In Fig. 7 and 11 the uncertainty comes directly from the measurement of the pressure including the maximum errors of pressure transducer and the acquisition card. It gets value ±0.0313 MPa on the pressure level of

5 MPa and ±0.0156 MPa near zero level. The estimated error of the location is ±0.03 mm.

In Fig. 13 the uncertainty of the cavitation number value is smaller with higher upstream and downstream pressure values being between ±0.016 and ±0.128 but mostly smaller than ±0.071.

The maximum error for the sound pressure meter was not found. However, the change of the sound pressure level is more interesting rather than the absolute value.

## 7 Conclusion

The process of cavitation appearance, the effects of cavitation on the characteristics of flow rate, noise and pressure distributions and boundary of inception were studied using two different shaped poppet valves - one had a sharp edged seat and another had a chamfer on the seat. It was found that the existence of length on the seat made the cavitation occurrence more easy, noise louder and induced choking phenomenon in the flow rate characteristics. At the viewpoint to reduce the undesired effects caused by the cavitation, except erosion, it is concluded that the sharp edged seat valve is better than the valve with a length on the seat.

In this research, erosion by cavitation is not discussed. During our tests, the erosion was not found on the surfaces of valve seat and poppet, but it was found on the Perspex plate. The experimental method with a half cut test model used here is considered to be available for the research on erosion by cavitation inside the valve. It is considered to be important at the next stage to reveal the effect of change of the valve shape on the erosion by the cavitation.

## Nomenclature

| | |
|---|---|
| $a$ | minimum sectional area of orifice $a = \frac{1}{2}\pi d_1 h \left(1 - \frac{h}{d_1}\cos\phi\right)$ |
| $C$ | discharge coefficient, $C = Q/\left(a\sqrt{2\Delta P/\rho}\right)$ |
| $d_1, d_2$ | diameters of valve seat |
| $h$ | effective opening of orifice, $h = x\sin\phi$ |
| $K$ | cavitation number, $K = P_2/\Delta P$ |
| $P_1$, $P_2$ | inlet and outlet pressures |
| $\Delta P$ | pressure difference, $\Delta P = P_1 - P_2$ |
| $Q$ | flow rate |
| Re | Reynolds' number, Re $= \frac{h}{\nu}\sqrt{2\Delta P/\rho}$ |
| $S$ | length of the seat |
| $x$ | valve opening |
| $\nu$ | kinematic viscosity |
| $\rho$ | density |
| $2\phi$ | top angle of poppet |

## Acknowledgements

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