

FEDSM-ICNMM2010-30676

**EXPERIMENTAL AND NUMERICAL INVESTIGATION OF FLOW INDUCED
VIBRATION OF STEAM CONTROL VALVE**

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ABSTRACT

Control valves of the main steam flow for power plants are operated under wide range of valve openings and pressure ratios. In the present paper, experimental and numerical investigations are described in order to clarify mechanisms of the valve head vibration. Experiments are conducted with two types of the valve head support. One is a flexible support and the other one is with an exciter. Results show that valve head vibrations with large amplitude appear with the flexibly supported valve head under certain range of valve openings and the pressure ratio. With the valve head exciter, dynamic fluid forces are measured. Results show that the added damping force becomes negative around the condition where the valve head oscillation is observed with flexible support. Numerical analyses are carried out in order to observe the flow field. In the simulations, forced vibrations of valve head are assumed. Results show that the pressure distribution on the valve head surface changes depending on the excitation frequencies, and as a result, the negative damping force occurs.

INTRODUCTION

In nuclear and thermal power plants, a main control valve is located between a steam generator and a turbine in order to control the steam flow. The pressure difference between upstream and downstream of the valve becomes large

especially during startup and shutdown transients of the plant. Therefore, the flow around the valve head becomes transonic and separates from the valve head and the seat. This flow separation is associated with a shock-boundary layer interaction [1]-[4]. As a result, the flow field becomes complex and large unsteady forces act on the valve head and pipe systems. Consequently, such unsteady flows can cause serious vibrations [5]-[7]. These problems make stable operation of the plants problematic.

In the present study, experimental and numerical investigations of the self-excited vibrations of the valve head are conducted with subscale valve models. In order to observe the appearance of the valve head vibration, a test valve with the valve head supported by flexible shaft is adopted. Dynamic fluid forces are also measured by exciting the valve head with an exciter. The influence of the valve head vibration on the flow pattern is examined with numerical simulations.

EXPERIMENTAL METHODS

Experiments are conducted with a blow down type wind tunnel shown in Fig. 1. Dry compressed air is stored in a high pressure tank. The inlet pressure is controlled by an electric-magnetic control valve. The working fluid from the test valve

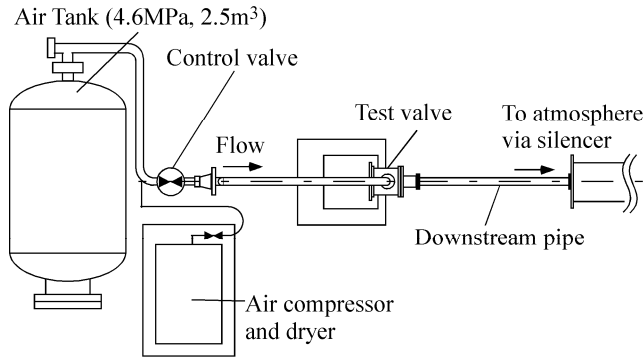


Fig. 1 Experimental apparatus

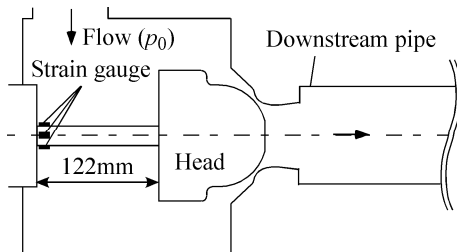


Fig. 2 Schematic of test valve with flexibly supported valve head

Table 1 Dimensions of test valve

Seat diameter, D_s	62.4 mm
Minimum diameter downstream of valve, D_m	52 mm
Diameter of downstream pipe	81.9 mm
Length of downstream pipe	1800mm
Curvature radius of head surface	43.7 mm
Curvature radius of seat surface	12 mm

is discharged to atmosphere through a downstream pipe and a silencer.

Figure 2 shows a schematic of the test valve with the flexibly supported valve head. The valve head is supported by steel pipe with the outer diameter of 14mm and the thickness of 1mm. Four strain gauges are set in order to monitor the displacement of the valve head. The natural frequency measured by a tapping test without flow is 29.3Hz.

Main dimensions of the test valve are shown in Table 1. The contour of the valve head is similar to the real valve in the plant [7]. There is a step downstream of the valve seat.

Figure 3 shows schematics of the valve head exciter. The valve head is axially supported by the shaft with a pivot and laterally connected to a crank shaft driven by an AC servo motor to provide lateral vibration. Therefore, the valve head vibrates alternatively in one direction. The displacement of the valve head is measured with a laser displacement sensor (Keyence, LK-G35, Detecting distance is 30 ± 5 mm and maximum error is 0.02%).

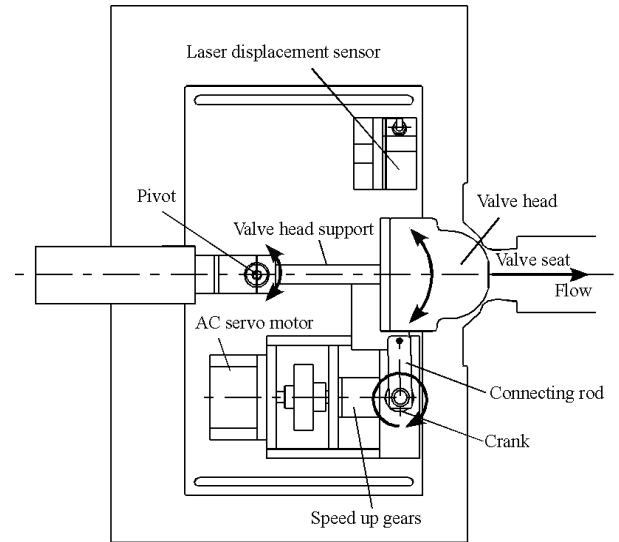


Fig. 3 Schematic of test valve with exciter

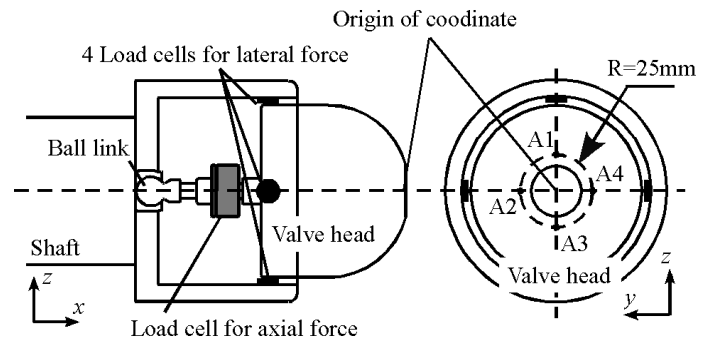


Fig. 4 Schematic of fluid force measurement system in valve head and location of pressure taps (A1-A4)

Under the forced vibration, dynamic fluid forces are measured using loadcells as shown in Fig. 4. The valve head is supported by a load cells (KYOWA Electric Instrument, LUR-A-2KN-SA1, rated capacity is 2kN and maximum error is 0.5%) in axial direction and four loadcells (KYOWA Electric Instrument, LMA-A-50N, rated capacity is 50N and

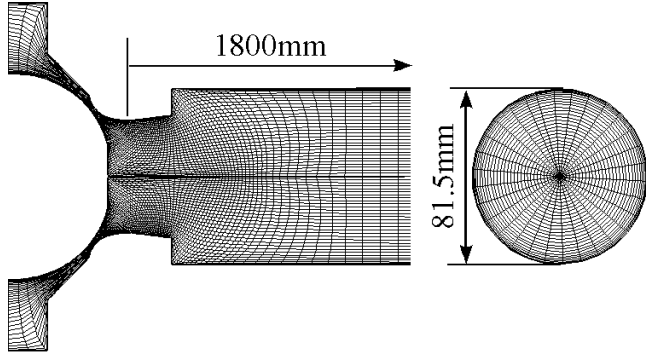
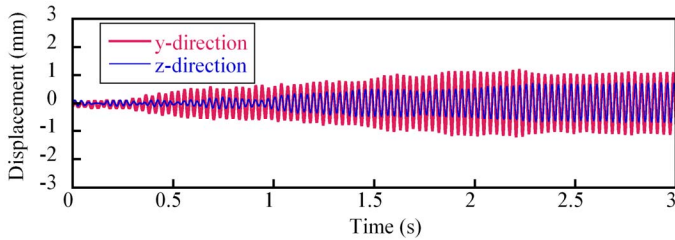
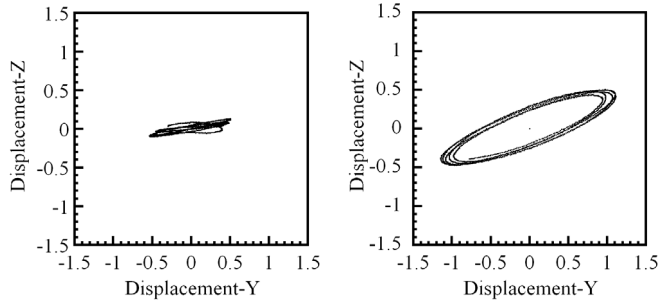


Fig. 5 Computational grid (Every 4 lines are shown)



(a) Displacement of valve head in y and z direction



**(b) Lissajous curve of valve head
(left: 0.16s-0.32s, right: 0.80s-0.96s)
Fig. 6 Vibration of the valve head
($\varepsilon_L=0.046$, $\psi_p=0.45$)**

maximum error is 1.0%) in lateral directions. The root of the load cell in axial direction is freely supported in lateral directions with a ball link. Wall pressures are also measured on the valve head surface with four pressure transducers (KULITE, CT-190-100A, rated capacity 689.5kPa and maximum error is 0.1%). In the experiment with the flexibly supported valve shown in Fig. 2, only wall pressures are measured at the same locations shown in Fig. 4. In the present study, the coordinate axes are defined as shown in Fig. 4. The origin of the coordinates is at the center of the valve head cusp as shown in Fig. 4.

Operation conditions of the valve are defined by the valve opening ratio and the pressure ratio as follows.

The opening ratio is expressed as:

$$\varepsilon_L = x_L / D_s \quad \dots(1)$$

here, x_L is the lift of the valve head and D_s is the valve seat diameter.

The pressure ratio is

$$\psi_p = p_0 / p_b \quad \dots(2)$$

here, p_0 is the inlet pressure measured at the valve case wall and p_b is the back pressure. In the present facility, atmospheric pressure is used as the back pressure because the pressure loss in the silencer is negligible.

NUMERICAL PROCEDURES

Unsteady numerical simulations are carried out in order to understand flow patterns. Governing equations are three-dimensional Navier-Stokes equations. Numerical schemes of the simulation are based on a finite difference method. SHUS [8] with fifth order Weighted Compact Nonlinear Scheme (WCNS-E-5) [9] are adopted for convective terms. A sixth order compact scheme [10] is used to calculate viscous terms. DES model based on Spalart-Allmaras model [11] is adopted as a turbulence model. As a time integration scheme, a second order three point backward implicit difference scheme with LU-SGS method [12] are applied. The time step is 3.125×10^{-6} seconds.

Figure 5 shows computational grid around the valve. The number of grid nodes is 500 points in axial direction, 100 points in radial direction, and 90 points in circumferential direction. Therefore the total number of grid nodes is 5,220,000 points. Computations are carried out with HITACH, SR11000 in Supercomputing Division, Information Technology Center, The University of Tokyo.

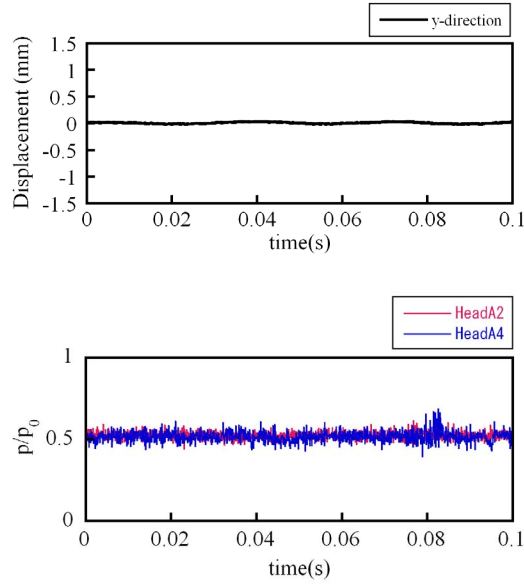
In the present paper, the flow around the fluctuating valve is examined. The valve head vibrates with the frequencies of 16Hz and 32Hz, and the amplitude of 1.0 mm in one direction alternatively to simulate the experimental conditions. At the inlet boundary, the inlet total pressure is fixed at designated value and the stagnation temperature is fixed at 300K and Riemann invariants are extrapolated from inner nodes. At the outlet, the static pressure is fixed at 100kPa and the density and each velocity components are extrapolated. The walls are treated as non-slip adiabatic walls.

RESULTS AND DISCUSSIONS

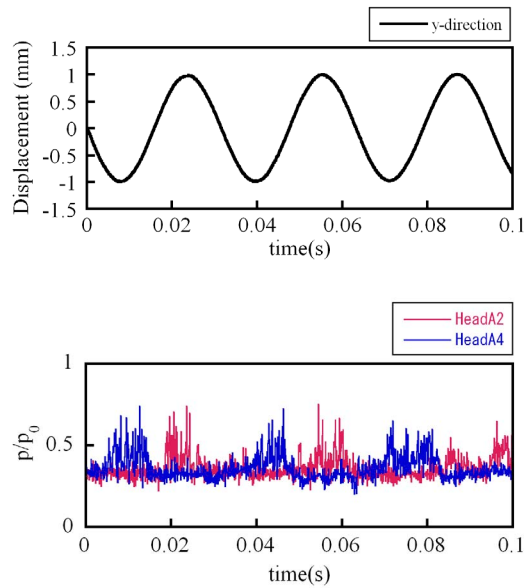
OBSERVATION OF THE VALVE OSCILLATION

As a first step, the valve head vibration is observed under various valve opening ratios and pressure ratios with the test valve with the flexibly supported valve head.

Figure 6 shows time histories of the valve head oscillation observed with $\varepsilon_L=0.046$ and $\psi_p=0.45$. As shown in Fig. 6 (a),



(a) $\varepsilon_L=0.046$, $\psi_p=0.60$



(b) $\varepsilon_L=0.046$, $\psi_p=0.45$

Fig. 7 Pressure oscillation on the valve head surface

the amplitude of the valve head vibration becomes large gradually, and finally, becomes constant. Fig. 6 (b) shows Lissajous curves of the valve head trace during the amplitude increases. At the beginning of the valve head vibration, the valve head vibrates in one direction. However, as the amplitude becomes larger, the Lissajous curve becomes elliptic.

The pressure oscillation shows different behavior depending on the amplitude of the valve head oscillation. Figure 7 shows the valve head displacement and the pressure oscillation on the valve head surface. Figure 7 (a) shows the case when the amplitude of valve head vibration is small. The

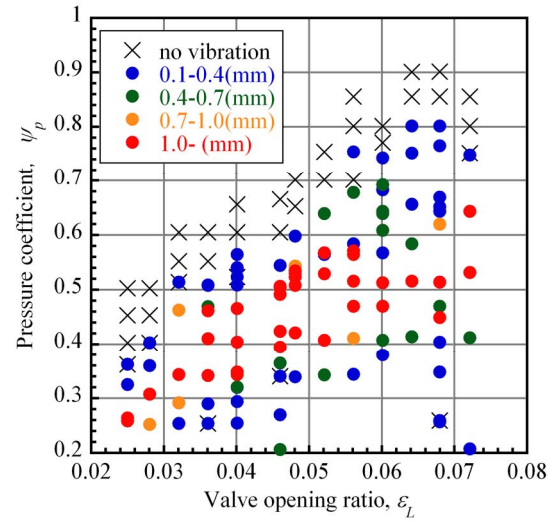
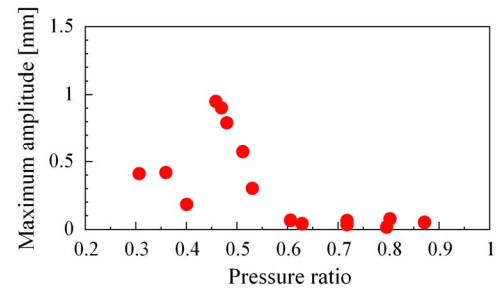
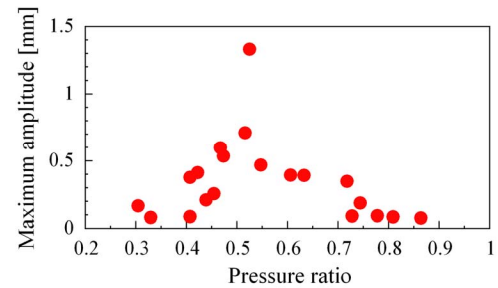


Fig. 8 Amplitude of the valve head vibration plotted in ε_L - ψ_p plane



(a) $\varepsilon_L=0.046$



(a) $\varepsilon_L=0.068$

Fig. 9 Amplitude of the valve head vibration versus pressure ratio

pressure oscillation is at random in time. On the other hand, when the amplitude of the valve head vibration is larger, the pressure oscillation becomes periodic as shown in Fig. 7 (b). The fundamental frequency of the pressure oscillation, 31.5Hz is the same as the valve head vibration frequency. The frequency of the valve head is 31.5Hz at all cases with the present experimental setup. This is slightly higher than the above mentioned natural frequency of 29.3 Hz determined from the tapping test result.

Figure 8 shows amplitudes of the valve head vibration plotted in $\varepsilon_L - \psi_p$ plane. The valve head vibration appears in a limited region in $\varepsilon_L - \psi_p$ plane. At the smaller valve opening ratio, the valve head vibration appears around small pressure ratios. As the valve opening ratio becomes larger, the region spread to large pressure ratio side.

Figure 9 shows the amplitude of valve head vibration versus the pressure ratio for $\varepsilon_L=0.046$ and $\varepsilon_L=0.068$. The amplitude distributions become maximum at a certain pressure ratio as can be seen from Fig. 9.

MEASUREMENT OF FLUID FORCES

As mentioned above, the valve head vibration occurs in a certain range of the operating conditions. The valve motion also affects the flow characteristics. In the present section, dynamic fluid forces are measured by exciting the valve head with the exciter. Dynamic fluid forces are examined as follows.

With the exciter, the valve head vibration is in y-direction shown in Fig. 4 and the equation of motion of the valve head is expressed as follows;

$$m\ddot{y} + c\dot{y} + ky = F \quad \dots(3)$$

$$y = A \sin \omega t \quad \dots(4)$$

here, m is the mass of the valve head, c is the structural damping coefficient, k is the stiffness coefficient of the valve head support, A is the amplitude, ω is the angular frequency, and t is time. We represented fluid force by

$$\begin{aligned} F &= F_0 \sin(\omega t - \phi) \\ &= F_0 \cos \phi \sin \omega t - F_0 \sin \phi \cos \omega t \\ &= \frac{F_0}{A} \cos \phi \cdot y - \frac{F_0}{A\omega} \sin \phi \cdot \dot{y} \quad \dots(5) \\ &= -(Ky - C\dot{y}) \end{aligned}$$

where, ϕ is the phase difference, $K = -F_0 / A \cos \phi$ is an added stiffness coefficient and $C = F_0 / (A\omega) \cdot \sin \phi$ is an added damping coefficient. When C is negative, the flow destabilizes the valve motion.

In the present experiment with the exciter, the measured fluid forces include inertial force. The measured data is examined after removing the inertial force measured without flow. To calculate fluid force components, the FFT analysis was applied to the data obtained in one second with sampling rate of 1/8192 are carried out. The following results are obtained by averaging 5 sets of the FFT analyses at different times. The maximum and minimum of 5 sets of measured data are also indicated by error bars.

Figure 10 shows measured damping coefficients for the

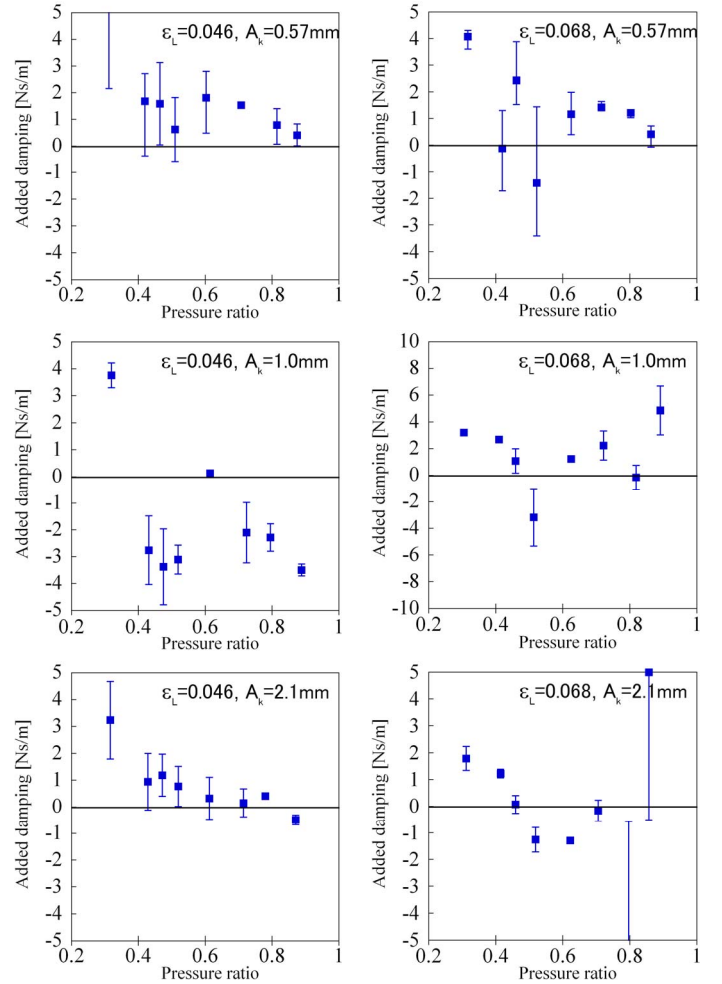


Fig. 10 Measured added damping coefficients ($f_k=32\text{Hz}$)

cases of two opening ratios ($\varepsilon_L=0.046$ and 0.068) and three amplitudes ($A=0.57\text{mm}$, 1.0mm and 2.1mm) at the excitation with the frequency of 32 Hz . The valve lifts are 2.87mm at $\varepsilon_L=0.046$ and 4.24mm at $\varepsilon_L=0.068$. For both cases, the relationship between the excitation amplitude and the added damping coefficient are nonlinear as can be seen in Fig. 10. For the case of $\varepsilon_L=0.046$ shown in Fig. 10 (a) with the amplitude of 0.57mm , the averaged added damping coefficient is positive at all pressure ratios. However, minimum values indicated by error bars are negative between the pressure ratios of 0.4 and 0.5 . With the amplitude of 1.0mm , the averaged damping coefficient becomes negative in wide range of the pressure ratios. With the amplitude of 2.1mm , the damping coefficient becomes larger than for the amplitude of 1.0mm . Fig. 10 (b), the influence of the excitation amplitude is smaller than that in Fig. 10 (a). At the amplitudes 0.57mm and 1.0mm , however, the regions where the damping coefficient becomes negative are smaller than that for 2.1mm . Additionally, the variability of each data becomes smaller as

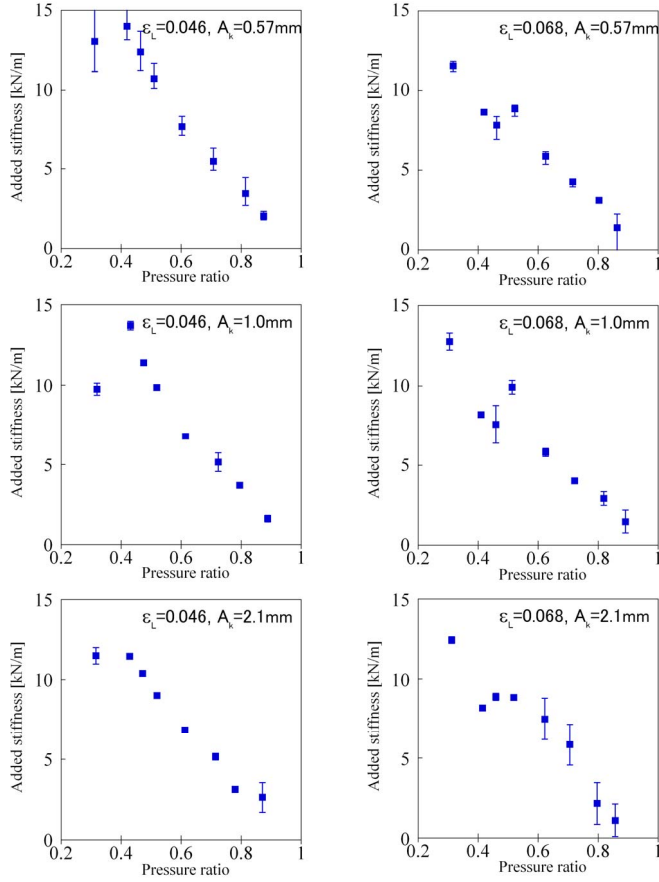
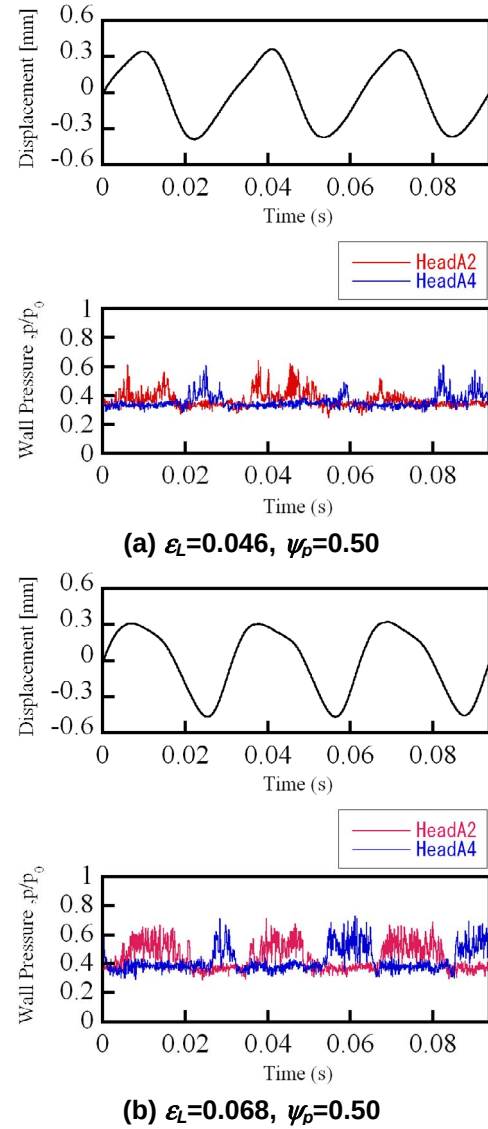


Fig. 11 Measured added stiffness coefficients ($f_k=32\text{Hz}$)

the amplitude becomes larger at the pressure ratios between 0.4 and 0.7.

Although the valve head vibration is considered to be caused by the negative damping fluid force, there are some points showing the disagreement of the relation between the amplitude of valve head vibration shown in Fig. 9 and the added damping coefficient shown in Fig. 10. At some conditions, added fluid forces shown in Fig. 10 are positive around pressure ratios where the amplitude of valve head vibration is large in Fig. 9. However error bars are long around such pressure ratio cases. At these conditions, it is observed that the added damping varies between positive and negative during the measurement. Such a trend was observed from the pressure oscillation on the flexibly supported valve head. This suggests that the coupling of the flow and valve head motion is not stable and as a result, the amplitude is limited. At $\varepsilon_L=0.046$ and 4.24mm, the added damping coefficient is negative at high pressure ratio although the valve head oscillation is not observed in Fig. 9. The reason of this disagreement has not been cleared yet.

Figure 11 shows added stiffness coefficient. As the pressure ratio increases, the added stiffness coefficient becomes smaller. However, the influence of the opening ratio is small.



(a) $\varepsilon_L=0.046$, $\psi_p=0.50$
Fig. 12 Displacement of valve head and pressure oscillation ($f_k=32\text{Hz}$)

The frequency of the valve head is constant at 31.5 Hz and higher than the natural frequency measured without the flow in the experiment with flexibly supported valve head. This means the added stiffness is almost constant but the measured stiffness coefficient depends on the pressure ratio. From the result of experiments with the flexibly supported valve head, the added stiffness coefficient is estimated to be 26.3kN/m. Although measured values are less than half of this value, it explains the increase of the frequency under operation condition.

Figure 12 shows displacement of the valve head and pressure oscillation. The pressure transducers A2 and A4 are placed in the direction of the vibration but at the opposite locations. As observed in Fig. 7, the pressure oscillation is periodic and has the same frequency with the valve head

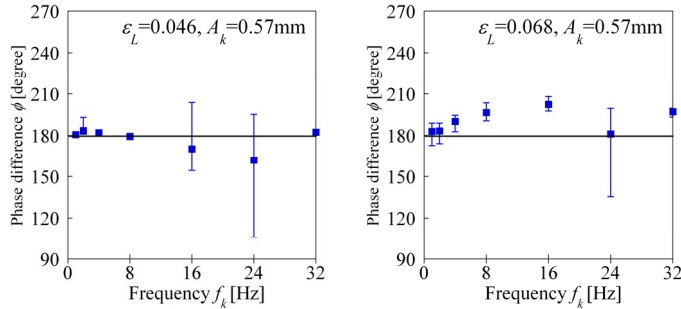
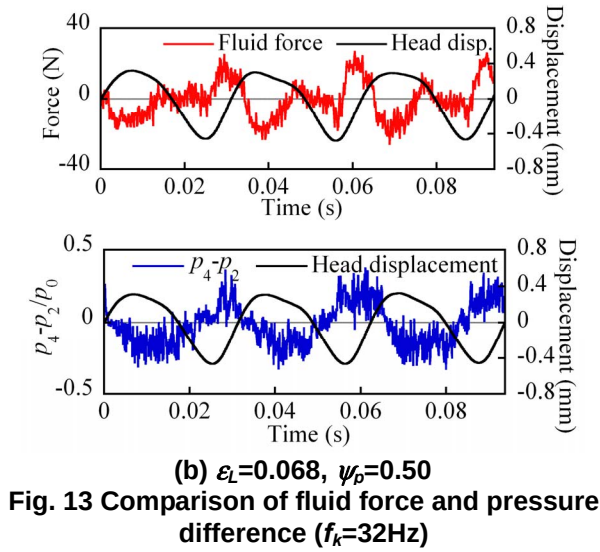
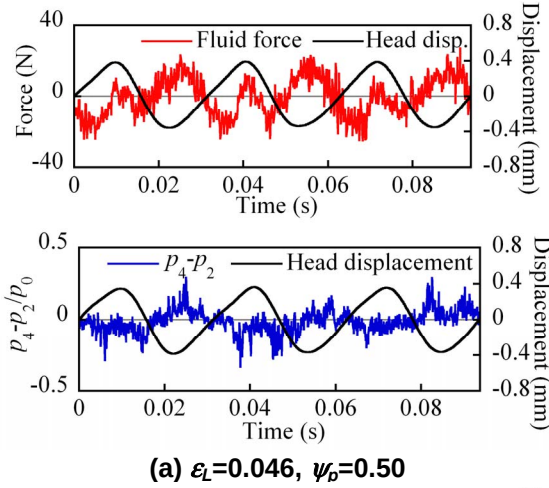


Fig. 14 Phase difference between valve head vibration and fluid force oscillation for various excitation frequencies

excitation.

Figure 13 shows the fluid force and pressure difference in comparison with the displacement. The wave forms are similar to each other.

Figure 14 shows the phase difference between the valve

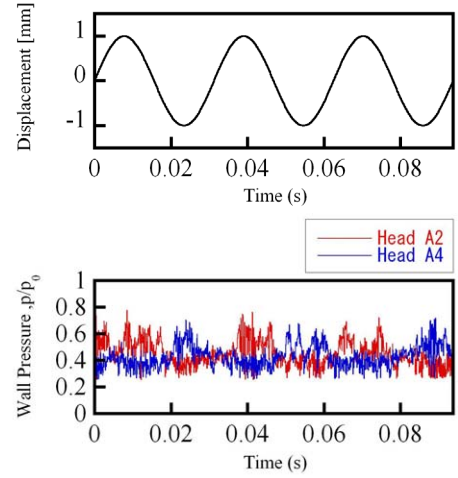


Fig. 15 Displacement of valve head and pressure oscillation (CFD, $\varepsilon_L=0.068$, $\psi_p=0.45$, $f_k=32\text{Hz}$)

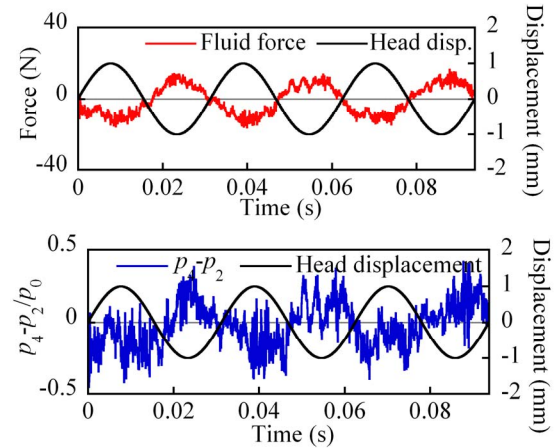


Fig. 16 Comparison of the fluid force and the pressure difference (CFD, $\varepsilon_L=0.068$, $\psi_p=0.45$, $f_k=32\text{Hz}$)

head displacement and the fluid force oscillation versus the excitation frequencies. In these figures, if the phase difference is larger than the 180° , the added damping coefficient becomes negative. At low excitation frequencies, the phase difference approaches 180° . As the frequency becomes larger, the phase difference becomes larger although there is large variability.

NUMERICAL RESULT

Figure 15 shows the pressure oscillation at A2 and A4 under the assumed lateral valve head vibration shown in the upper figure. Comparing with the experimental result shown in Fig. 12, the numerical result shows acceptable agreement in averaged pressures and the amplitude of the pressure oscillation.

Figure 16 shows the fluid force and the pressure difference in comparison with the valve head displacement. Comparing with experimental results shown in Fig. 13, they show

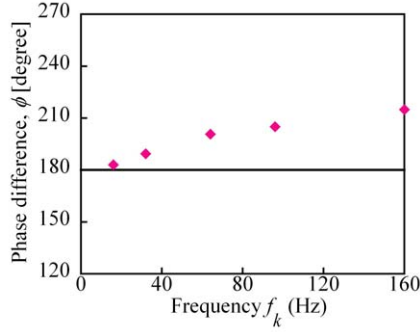


Fig. 17 Phase difference between the valve head vibration and the fluid force oscillation for various excitation frequencies (CFD, $\varepsilon_L=0.068$, $\psi_p=0.45$)

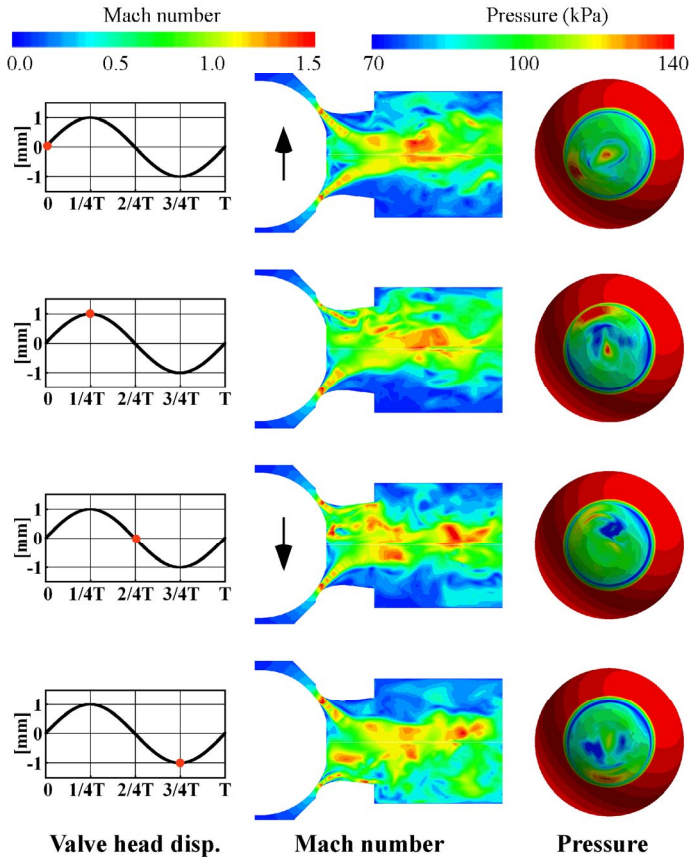


Fig. 18 Valve head displacement, Mach number distribution in meridian cross section and pressure distribution on valve head surface (CFD, $\varepsilon_L=0.068$, $\psi_p=0.45$, $f_k=16\text{Hz}$)

acceptable agreement in the amplitude of the fluid force and the pressure oscillation. The phase difference between the valve head vibration and the fluid force oscillation is 189.5° . In the experimental result at the same condition, the phase difference is 216° . Although the values are different, the damping coefficients are negative in both results.

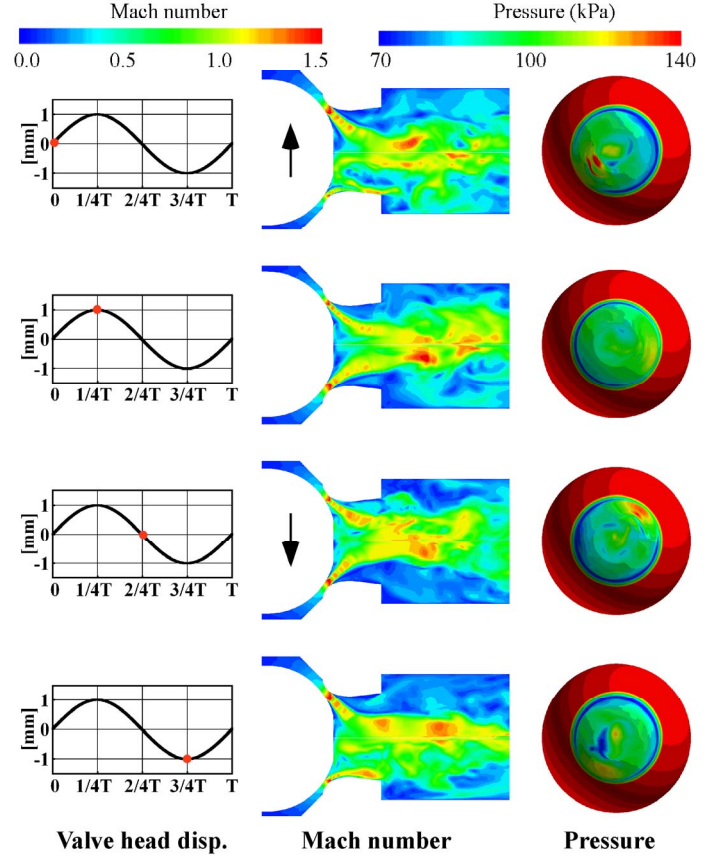


Fig. 19 Valve head displacement, Mach number distribution in meridian cross section and pressure distribution on valve head surface (CFD, $\varepsilon_L=0.068$, $\psi_p=0.45$, $f_k=32\text{Hz}$)

Figure 17 shows the phase difference between the valve head vibration and the fluid force oscillation versus excitation frequencies. The phase difference approaches 180° at low excitation frequencies. As the frequency becomes larger, the phase difference becomes larger. Comparing with the experimental result shown in Fig. 14, smaller phase difference is predicted. Although a dip at 24Hz is found in the experiment, the phase difference in numerical simulation increases monotonically beyond the maximum frequency 32Hz of the experiment.

Figure 18 shows the displacement of the valve head, the Mach number distribution in the meridian cross section and the pressure distribution on valve head surface for every 1/4 period of the valve head vibration with the excitation frequency of 16Hz. At this frequency, Fig. 17 shows that no phase delay of the fluid force occurs. The influence of the valve head vibration is not clear from the Mach number distribution. On the valve head surface, a high pressure region appears because of the asymmetric flow downstream of the valve head. When the displacement of the valve head is maximum at $t=1/4 T$ (1mm) and $3/4 T$, (-1mm) the high pressure region is at the same side of the valve head displacement. Therefore, the fluid force acts

in the opposite direction to the displacement of the valve head and it results in the resilience. The added damping force is small because the phase difference between the valve head vibration and the fluid force oscillation is almost 180° as can be seen in Fig. 17.

Figure 19 shows the case of the excitation frequency of 32Hz. In this figure, the high pressure region is at the opposite side of the valve head velocity direction at $t=0$ and $2/4T$. Therefore, the direction of the fluid force is the same with the valve head velocity. The similar phenomenon is observed in the results with higher excitation frequencies.

CONCLUSIONS

In the present study, experimental and numerical studies have been carried out in order to clarify the mechanisms of the serious vibration of the valve head.

Experimental results with the valve head supported by flexible shaft show that the valve head vibration appears with the large amplitude in a certain range of the operation conditions of the valve. The pressure oscillation on the valve head is at random when the amplitude of the valve head vibration is small. When the valve head vibrates with a large amplitude, however, the pressure oscillation becomes periodic with the frequency of the valve head vibration. Forced vibration experiments were also carried out. The fluid forces are decomposed into the added stiffness and the added damping force. At the condition where the valve head vibration occurs in the experiment with the flexible shaft, the added damping force becomes negative. Numerical results show that the unsteady behavior of the pressure distribution on the valve head surface changes depending on the excitation frequencies.

The fluid force treated in the present study occurs in response to the valve head vibration. Therefore, it is important to increase the stiffness of the valve head support system.

NOMENCLATURE

A	: Amplitude of valve head vibration
c	: Damping coefficient
C	: Added damping coefficient
ε_L	: Valve opening ratio
f	: Frequency
f_k	: Excitation frequency
p_0	: Inlet pressure
p_b	: Back pressure
ω	: Angular frequency
x_L	: Valve lift
ψ_p	: Pressure ratio

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