TRANSONIC FLOW VIBRATION IN A STEAM CONTROL VALVE FOR A POWER PLANT

Koichi YONEZAWA, Yoshinobu TSUJIMOTO
Osaka University, 1-3 Machikaneyama, Toyonaka, Osaka, 560-8531, Japan
Kenichi TEZUKA, Michitsugu MORI
Tokyo Electric Power Company, 4-1, Egasaki, Tsurumi-ku, Yokohama, Kanagawa, 230-8510, Japan
Ryo MORITA and Fumio INADA
Central Research Institute of Electric Power Industry, 2-11-1, Iwatokita, Komae, Tokyo, 201-8511, Japan

ABSTRACT

Transonic flow vibrations in a main steam control valve for nuclear and thermal power plants are investigated. In the steam control valve, flow becomes supersonic around the valve head because of the large pressure difference between upstream and downstream of the valve. Shock waves occur and cause flow separations. The separated jet can become unsteady under a certain condition. As a result, serious pressure fluctuations occur because the shock wave fluctuation follows the unsteady motion of the separated jet. In the present paper, experimental and numerical studies are carried out to clarify the characteristics of the flow fluctuation. As a first step, unsteady pressure measurements and unsteady simulations are conducted for the case that the valve head is supported rigidly. Results show that several modes of pressure fluctuations appear around the valve head and its downstream pipe. As a second step, a flow induced vibration is investigated with a flexible support of the valve head. Results show that a diverging oscillation of the valve head is caused by the flow fluctuation.

1. INTRODUCTION

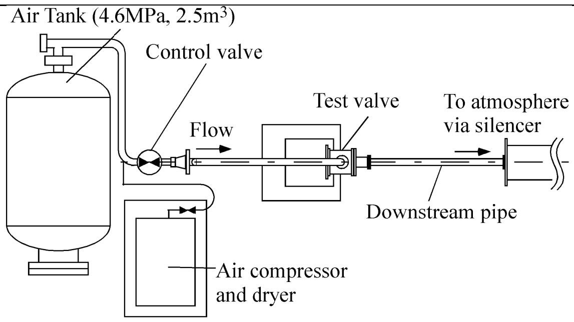
In nuclear and thermal power plants, a lot of flow control valves are used. A main steam control valve is one of the most important valves and used to control an output of the plant by means of controlling the main steam flow from a steam generator or a nuclear reactor to a steam turbine. Under partial load operating conditions of the plant, the main steam control valve limits the flow by reducing the clearance between the valve head and the valve seat. In such a situation, the pressure difference between upstream and downstream of the valve becomes so large that the flow velocity around the valve head becomes higher than the speed of sound. Shock waves occur in the supersonic jet and cause flow separations due to a shock-boundary layer interaction. The separated supersonic jet induces complex and unsteady flow patterns downstream of the valve and results in serious pressure fluctuations. The high level pressure fluctuation may cause an unexpected shutdown of the power plant.

Problems originated from the unsteady flow in the steam control valve have been studied for many years. Araki et al. (1981) and Jibiki (2000) have reported experimental and numerical investigations of a flow field around the valve. In these papers, it is reported that the pressure fluctuation is caused by asymmetric and unsteady flow patterns at a certain condition. Shioyama et al. (1995) and Michaud et al. (2001) have reported their studies about a coupled oscillation of an acoustic resonance and a pipe wall natural fluctuation in a pipe system downstream of the valve. Recently, Morita et al. (2004) have reported experimental investigation and three-dimensional numerical simulations. By means of 3-D simulations, detailed characteristics of flow patterns were clarified. Morita et al. (2005) also examined the valve head fluctuation induced by the flow fluctuation.

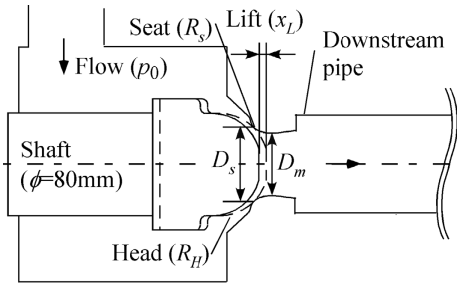
In the present paper, experimental and numerical studies are carried out in order to understand following two problems. The first problem is a relation between the unsteady flow around the valve head and the acoustic resonance in the downstream pipe. The flow characteristics of the pressure fluctuation in the pipe are shown. The second problem is detailed investigation of the valve head oscillation caused by the flow fluctuation. Although Morita et al. (2005) have reported about this problem, the detailed mechanisms and numerical analysis has not been reported yet. By means of the experimental and numerical studies, mechanisms of the flow induced vibration are examined.

2. EXPERIMENTAL EQUIPMENT

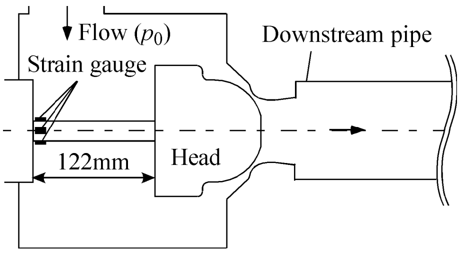
Figure 1 shows a cold flow test facility. Dry air has been used as the working fluid instead of steam. Although physical properties are different from steam, it is considered that we can investigate fundamental characteristics of the flow fluctuation qualitatively. Figure 1(a) shows an overview of the facility. The dry high pressure air is stored in a high



(a) *Overview of test facility*



(b) *Rigidly supported valve head configulation*



(c) Flexibly supported *valve head configulation*

Figure 1: Experimental apparatus

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

Table 1: Valve geometries

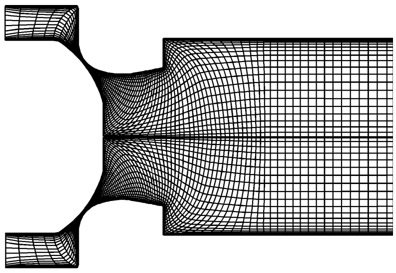


Figure 2: Computational mesh around valve

pressure tank. The inlet pressure of a test valve is controlled by an electric-magnetic control valve. Air discharged to atmosphere through the test valve, a downstream pipe and a silencer. Figure 1(b) shows the schematic of rigidly supported valve head configuration. The valve is supported by a shaft with a diameter of 80mm made from stainless steel. Dimensions of the valve head, the seat and the pipe are shown in Table 1. Pressure transducers (KULITE CT-190-700A) are located at the radius of 0.3 D_s (D_s: Seat diameter) on the head, at 0.1D_s downstream of the seat and at 5D_s down stream of the seat (on pipe wall). At each location, four pressure transducers are located at each 90 degrees in circumferential direction. The position of the valve head is measured by a digital linear gauge sensor with a minimum resolution of 0.01mm. Figure 1 (c) shows the flexibly supported valve head configuration. The valve head is supported by low rigidity slender pipe made from steel. The head oscillation is measured by four strain gauges put on the slender shaft. Unsteady pressures are measured on the head after confirming negligible effects of vibration.

Two parameters are used to define operating conditions of the valve. The first is a valve opening ratio.

$$\varepsilon_L = \frac{x_L}{D_s} \tag{1}$$

Here, x_L is a lift of the valve head. The second parameter is a pressure ratio.

$$\psi_p = \frac{p_b}{p_0} \tag{2}$$

Here, p_b is a back pressure and p_0 is a inlet pressure. In the present study, atmospheric pressure is used as p_b because a pressure loss in the silencer is negligible.

3. NUMERICAL METHODS

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. AUSMDV with 4th order compact MUSCL TVD interpolation scheme are adopted as an upwind scheme of convective terms. A second order central difference scheme is used to calculate viscous terms. Spalart-Allmaras model is adopted as a turbulence model. As a time integration scheme, a second order three point backward implicit difference scheme with LU-SGS is applied.

Figure 2 shows a computational mesh around the valve head. The computational region is from

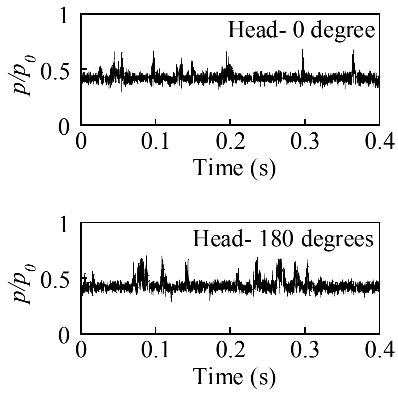


Figure 3: Pressure fluctuation on valve head (Experiment, ε_L=0.046, ψ_p=0.51)

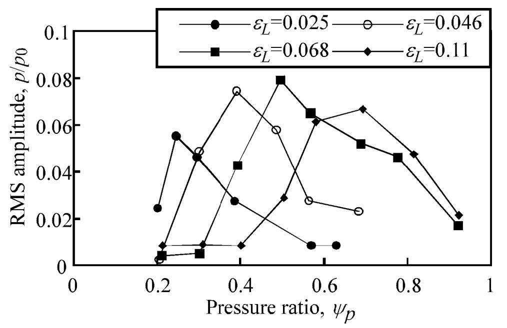
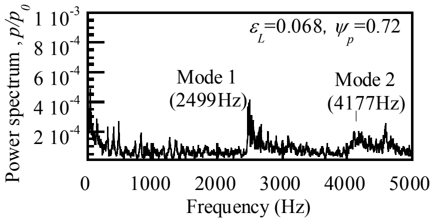
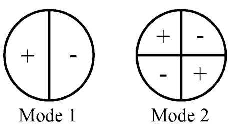


Figure 4: Amplitude of pressure fluctuation on head versus pressure ratio



(a) Spectrum



(b) Mode shape in cross section

Figure 5: Pressure fluctuation in downstream pipe

upstream of the throat between the head and seat to an outlet of the downstream pipe. The mesh consists of 714 points in axial direction, 70 points in radial direction and 69 points in circumferential direction. Following Boundary conditions are applied. At the inlet, a total pressure and a total temperature is fixed by assuming subsonic inflow. At the outlet, the static pressure is fixed at p_b. Walls of the valve and the pipe are treated as an adiabatic and non-slip wall.

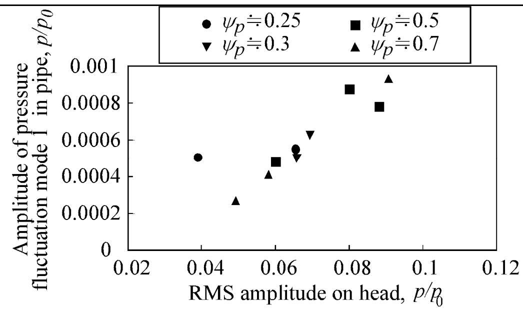


Figure 6: Amplitude of pressure fluctuation mode 1 in downstream pipe versus RMS amplitude of pressure fluctuation on valve head

4. RESULTS AND DISCUSSIONS

4.1 Experimental results with rigidly supported valve head

In this section, experimental results with rigidly supported valve head are shown in order to understand fundamental characteristics of flow fluctuations.

Figure 3 shows typical wave forms of pressure fluctuations on the rigidly supported valve head. In these figures, random pressure fluctuations with large amplitudes are observed.

Figure 4 shows the RMS amplitude versus the pressure ratio (ψ_p) for four valve opening ratio (ε_L). In each condition, maximum amplitude appears at a certain ψ_p. By comparing four cases, ψ_p with the maximum amplitude becomes larger as ε_L becomes larger. Similar pressure fluctuation has been reported by Morita et al (2004) in their studies with a simplified geometry model.

Figure 5 shows a spectrum of the pressure fluctuation on downstream pipe wall. There are two peaks at about 2500 Hz (Mode 1) and about 4200 Hz (Mode 2). According to phase differences between four measuring points in each 90 degrees, the mode shape is identified as shown in Figure 5 (b). It has been reported that such pressure fluctuations are caused by a coupled oscillation of a natural vibration of the pipe wall and an acoustic resonance in a cross section of the pipe. Murakami et al. (1989) has introduced a method to calculate the frequency of such a coupled oscillation. By this method, we can obtain frequencies of 2516Hz for the strong pressure fluctuation mode 1 and 4310 Hz for the weak pressure fluctuation mode 2.

Figure 6 shows the amplitude of the pressure fluctuation mode 1 in the downstream pipe versus RMS amplitude of the random pressure fluctuation on the valve head. In this figures, the amplitude of the mode 1 is proportional to the RMS amplitude of the random pressure fluctuation on the valve head.

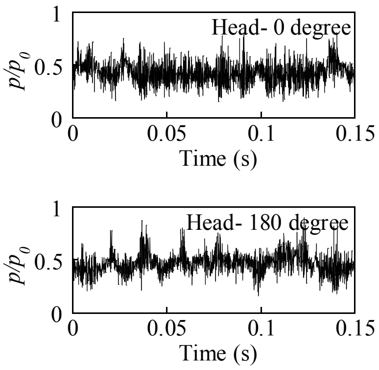


Figure 7: Pressure fluctuation on valve head (Numerical simulation, ε_L=0.046, ψ_p=0.48)

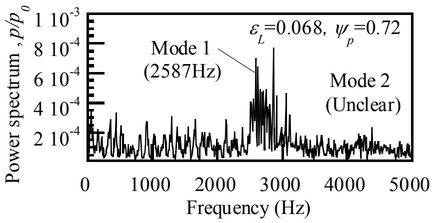


Figure8: Spectral analysis of pressure fluctuation in downstream pipe (Numerical simulation)

4.2 Numerical result for rigid valve

In this subsection, comparisons between the experimental and the numerical results are shown in order to validate our simulation software. After that, the flow field is investigated in detail.

Figure 7 shows the pressure fluctuation on head obtained by numerical simulation. The random pressure fluctuation with large amplitude is observed as is the case with experimental results.

Figure 8 shows the pressure fluctuation in the downstream pipe. Although the influence of the pipe wall fluctuation is not taken account, the pressure fluctuation with the frequency of about 2500Hz is captured. This is caused by acoustic resonance. However, the weak pressure fluctuation with a frequency of 4200Hz is unclear.

According to these result, it is considered that the numerical results can predicts the flow pattern in the valve and the pipe with acceptable accuracy.

Figure 9 shows instantaneous Mach number and pressure distributions around the head. From the Mach number distribution, it can be observed that supersonic jets are asymmetric and highly disturbed near the valve head. The pressure distribution shows that there is a high pressure region on the head surface. This high pressure region moves at random and causes the pressure fluctuations as shown in Figure 3 and 7.

Figure 10 shows an instantaneous pressure distribution of the pressure fluctuation mode 1 component on the downstream pipe surface. As shown in Figure 5 (b), there is one period of the wave in circumferential direction. These results suggest that the acoustic resonance fluctuation can be predicted without consideration of the pipe wall oscillation and external disturbances.

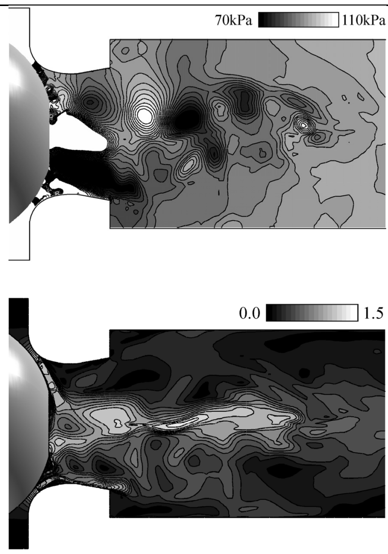


Figure9: Instantaneous pressure distribution (Top) and Mach number distribution (Bottom) (Numerical simulation, ε_L=0.046, ψ_p=0.48)

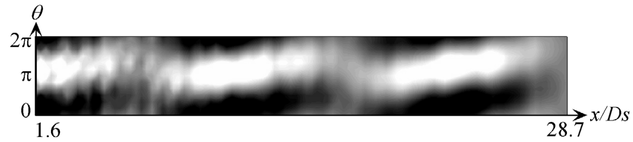


Figure10: Instantaneous pressure distribution of mode 1 component on downstream pipe surface (Numerical simulation, ε_L=0.025, ψ_p=0.25)

	Inner diameter	Outer diameter
Shaft-A (Steel)	10mm	14mm
Shaft-B (Steel)	20mm	24mm

Table 2: Specification of head support shafts

4.3 Experimental results with flexibly supported valve head

In the present study, two kinds of pipes with the same length of 122mm and different diameter are used to support two kinds of head with the same contour and different mass (Heavy head (stainless steel): 2.6kg and light head (Acrylonitrile butadiene styrene (ABS)): 0.34kg). Specifications of shafts are shown in Table2.

Figure 11 shows growth of oscillation with ψ_p=0.046 and ε_L=0.50. The displacement of the head is measured by four strain gauges put on the

	f_n (Hz)	f (Hz)
Shaft A/ heavy head	29.3	31.5
Shaft A/ light head	67.4	72.5
Shaft B/ heavy head	60.8	63.7
Shaft B/ light head	140.6	146.48

Table 3: Natural frequencies (f_n) of shaft head system by impact test and observed frequency of head oscillation (f)with flow in experiments

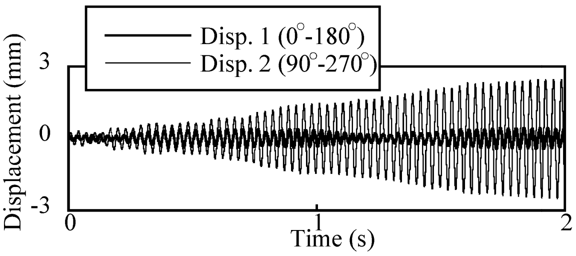


Figure11: Time history of head displacement (With shaft A and heavy head, ε_L=0.046, ψ_p=0.50)

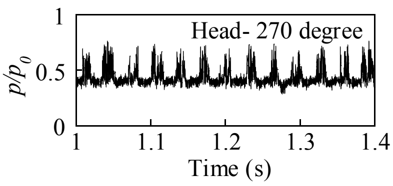


Figure12: Pressure fluctuation on head (With shaft A and heavy head, ε_L=0.046, ψ_p=0.50)

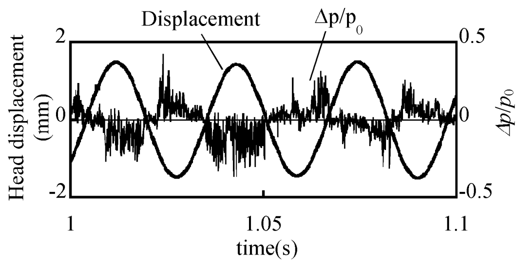


Figure13: Head displacement and pressure difference fluctuation on head as fluid force (With shaft A and heavy head, ε_L=0.046, ψ_p=0.50)

shaft surface at each 90 degrees as shown in Figure 1 (c). Figure 11 shows diverging oscillation of the head. When the head oscillation is large, the motion of the head is almost reciprocation.

Frequencies for the cases of all combinations of the shaft and the head are shown in Table 3. Measured frequencies with flow are slightly higher than natural frequencies obtained by impact test. Additionally, the frequencies are independent from operating conditions of the valve (ψ_p and ε_L).

Figure 12 shows pressure fluctuation measured on the head. When the amplitude of the head oscillation is large, the pressure fluctuation becomes periodic. This result is contrast to the result with the rigidly supported case.

Directions of fluctuating fluid force are evaluated from pressure difference between two measuring points which are located face to face. Figure 13 shows time histories of the head displacement and the pressure difference in the direction of the head displacement. A quasi static measurement showed that the pressure force is in the direction opposite to the displacement. In Figure 13, however, $\Delta p/p_0$ delays behind the displacement by about 206°. This means that the phase of the pressure fluctuation delays behind the quasi static response by about 26°. The work done by the fluid force per period is given as follows.

Head displacement: $y = A\sin 2\pi ft$

Fluid force: $F = -F_0 \sin\left(2\pi ft - \phi\right)$

Work: $$W = \int_0^{2\pi} F \cdot y dt = AF_0 \cdot 2\pi^2 f \sin\phi \qquad (3)$$

According to equation (3), the work becomes positive when the phase delay of the pressure difference fluctuation, ϕ is between 0° and 180°. In the experimental result shown in Figure 11-13 therefore, the work done by the fluid force is positive. This means that the diverging oscillation of the head is a kind of self excited oscillation caused by the fluid force.

4.4 Numerical result for oscillatory head

As mentioned in subsection 4.3, the coupled oscillation of the flow and the head shows many aspects of self excited oscillation. In this subsection, the mechanism of the oscillation is examined by means of the numerical simulation. The simulation is carried out under the prescribed head oscillation with a frequency of 32Hz. The head oscillation is simulated by deforming the computational mesh. By considering the flow response, it is possible to evaluate the aerodynamic damping force without carrying out the direct simulation of flow-structure coupled oscillation.

Figure 15 shows the time history of head displacement and the pressure difference in the direction of the head displacement. The phase delay of the unsteady fluid force (pressure difference in this figure) behind the quasi static response is 46°. Although this value is larger than that of experimental result, the flow response can cause diverging oscillation.

Figure 16 shows the Mach number distributions around the valve and the pressure distributions on head surface at each phase angle of 90°of head motion. From this figure, it can be observed that the flow response delays behind the head motion. For example, in Figure 16 (c), the flow reattaches upper

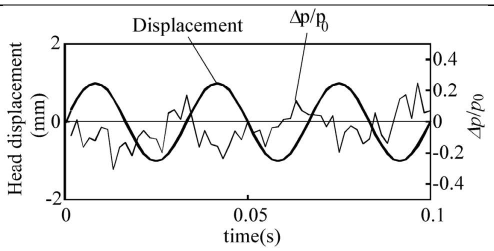


Figure14: Head displacement and pressure difference fluctuation on head as fluid force (Numerical result, f=32Hz, ε_L=0.046, ψ_p=0.50)

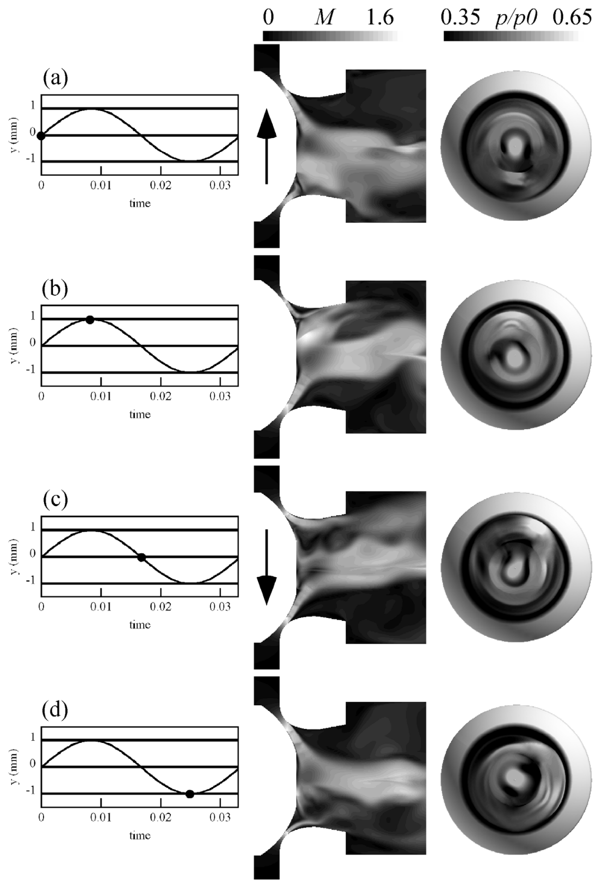


Figure 16 Head displacement (left), Mach number distribution (center) and pressure distribution on head surface (right) (Numerical result, f=32Hz, ε_L=0.046, ψ_p=0.50)

wall even though the head reaches the neutral position. Such a delayed response influences on the pressure distribution on the valve head surface and results in the excitation force.

5. CONCLUSIONS

In the present study experimental and numerical investigations are conducted in order to understand characteristics of a transonic flow fluctuation in a steam control valve. Results are summarized as follows.

- Random pressure fluctuations with the large amplitude occur around the valve head under a certain operating condition.
- When the amplitude of the pressure fluctuation on the valve head is large, the amplitude of the pressure fluctuation originated from the acoustic resonance in the downstream pipe becomes large.
- The pressure fluctuation on the head can cause the diverging oscillation of the head if the rigidity of the head support is low.

6. REFERENCES

Araki, T., et al., 1981, Fluid-Induced Vibration of Steam Control Valves, *Toshiba review (In Japanese)*, **36-7**: 648-656.

Jibiki, K., 2000, Fluctuation of Steam Control Valve, *Turbomachinery (In Japanese)*, **28-4**: 225-229.

Michaud, S., et al., 2001, Acoustic Fatigue of a Steam Dump Pipe System Excited by Valve Noise, *Journal of Pressure Vessel Technology*, **123**: 461-468

Murakami, Y., et al., 1989, Coupling Vibration between Pipe Wall and Acoustic Wave , *Turbomachinery (In Japanese)*, **17-8**: 484-489.

Morita, R., et al., 2004, CFD Calculation and Experiments of Unsteady Flow on Control Valve, *ASME Paper,* **HT-FED2004-56017**.

Morita, R., et al., 2005, Flow Induced Vibration of a Steam Control Valve in Middle Opening Condition, *ASME Paper*, **PVP2005-71334**.

Shioyama, T., et al., 1995, Coupling Vibrations between Pipe Wall and Internal Higher-Order Acoustic Wave Caused by Valve Choking, *Trans. JSME, Series C (In Japanese)*, **61-589**: 3450-3455.