Noise and Vibration Induced by Throttling of High Pressure Compressible Fluid*

(Part 1 - Characteristics of Noise and Vibration Generated
by Cage-guided Control Valve)

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Features of noise and vibration generated by a cage-guided control valve are discussed. In this type of valve, kinetic energy of the throttled jets is dissipated by mutual collision of the jets themselves. The pressure reduction process is made considerably smooth, and the noise level becomes lower than that of a freely expanding jet. However, due to a resonance between the jets and the valve cavity, a transitional increase of noise appears remarkably in a two-jets configuration within a pressure range of low supercritical flow. For a six-jets configuration, such a noise increase is suppressed, and, instead, stem vibration of high level is induced. Finally, the configuration of two-jets makes the static thrust increase with an increasing valve opening. This feature may cause a vibration of an elastically positioned stem of the valve.

Key Words : Control Valve, Noise and Vibration, Compressible Flow, Pressure Reduction, Multiple Colliding Jet, Cage-guided Valve, Blow-down Experiment.

1. Introduction

Managing and controlling of high pressure fluid flow are required in most of modern industrial fluid energy systems. The fluid pressure may be converted into high momentum flux in some local throttling devices. The high speed flows induce various phenomena which disturb the smooth control of the system with additional influences on the surroundings. Especially around control valves, dynamic disturbances such as acoustic noise, mechanical vibration and downstream flow fluctuation are generated at significant levels in their intensities. Prediction, control and

reduction of the intensities constitute an important subject in the fluid-transport and energy-control engineerings.^{(1), (2)}

The present research purports to investigate such dynamic problems arising in the control of compressible fluid flow by valves. In the operation at high pressure ratio, the flow is choked at the throat of the valve and various types of supersonic jet flow are formed, see Fig. 1. The flows take complicated configurations with multiple shock waves and are frequently unstable or oscillatory. Heavy vibration of the valve stem is sometimes induced. And intense noise is generated due not only to turbulent mixing but also to the interaction between turbulent eddies and

Fig. 1 Dynamic problems arising in throttling process of high pressure gas flow.

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shock waves in the process of diffusing to a subsonic flow. The noise and vibration are closely related to the local structures of the flow as well as to their transmission mechanisms. They are also affected by the geometry of the throttle and planning of the valve installation.

One of the authors investigated supersonic flow in a contoured type valve, and found that the noise has a particular relation with the flow pattern which shows a staged change with pressure ratio.^{(3)~(5)} Many related investigations have been made by different authors. Generation mechanisms of noise and vibration are investigated by two dimensional flow visualizations.^{(6)~(7)} The noise transmission through pipe wall is studied using propagating pressure waves inside a duct.⁽⁸⁾ A pressure surge induced by a flow instability within a relief valve is discussed in relation with the leading length of the piping.⁽⁹⁾ In industries, many types of so called "low noise and/or vibration valves" have been developed and utilized, and various rules have been presented to predict the noise and vibration levels. Nevertheless, the essential understandings and the effective treatments about the dynamic problems of valve flow are not yet established.

The first step of the research should be to collect and analyze the noise and vibration data using various types of industrial valves. In the present report, noise and stem force generated by cage-guided valves are discussed. The dissipation of the fluid energy in this type of valve is processed by colliding and mixing of multiple counter jets which are issued from throttling ports located around the cage, see Fig.1(c). The characteristics are expected to be considerably different from the characteristics induced by valves where wall shear stress has a major participation in the diffusion. The generation mechanisms of the particular phenomena found in the report, such as discrete tone noise generation, are to be made clear by the subsequent and closer investigation.

2. Experimental Facility and Method

2-1 Experimental facility⁽¹⁰⁾

A blow-down facility has been designed and constructed to make the research of the valve flows. The system is illustrated in Fig. 2. High-pressure air stored in the reservoirs is intermittently supplied to the test section. To keep the flow stationary at specified conditions and to suppress the upstream turbulence, a collecting chamber is installed at the entrance of the test section. Controlling the pressure and the temperature in the chamber, the stagnation pressure P_T and the stagnation temperature T_T of the flow are set at a steady state within the range up to 2.5MPa and 373°K respectively. A silencer is equipped to the collecting chamber to absorb acoustic disturbances from the upstream. A noise reduction by more than 20 dB is realized at the frequency of 1kHz.

The test section of 4.4 m total length constructed with schedule 60 steel pipes of 3 in. diameter is installed inside of an anechoic testing room of 64 m³ in volume. The test valve is mounted at the middle of

Fig. 3 Choked flow duration versus stagnation pressure P_T in the 3" valve testing.

Fig. 2 Blow-down facility used in the research of valve flows.

Table 1 Specifications of measuring instruments.

Instrument	Specifications
Condenser Microphone	Type :4133(B&K) Diaphragm dia. :1/2 in. Frequency range:4Hz~40kHz
Sound Level Meter	Type :2607(B&K) Frequency range:2Hz~200kHz Filter:A,B,C,D,22.5,22.5K
Load Cell	Type :9021A(KISTLER) Max.load: ±10kN
Charge Amplifier	Type:504(KISTLER) Frequency range:<100kHz
Data Recorder	Type:R-510(TEAC) Frequency range:DC~20kHz(FM)
Fourier Analyzer	Type:5451C(HP) Frequency range:DC~50kHz
Semi-conductor Microphone	Type:MIC-052 (KULITE) Sensitivity:-144.0dB (0dB/1v/μbar)

Fig. 4 Test valve and cages.

the section. Noise measurements within a distance of 1.5 m from the valve could be made under the condition of an acoustic-free field at more than 1 kHz.

A typical performance of the facility is presented in Fig.3., which shows the maximum duration to maintain the flow at supercritical for a respective opening of 3 in. valve using 0.7 MPa reservoirs. It is found necessary to anticipate a time loss of 15~20% from the value estimated by assuming a Fanno flow. The loss is due to a precharging of the collecting chamber at the beginning of blow-down operation. By utilizing the full reservoirs of 3 MPa storage pressure, it is possible to keep the stagnation pressure at 1 MPa over a time duration more than 30 sec in the test of valves at a C_v -value less than 50.

2-2 Test valve and experimental method

The geometry of the cage-guided valve is shown in Fig.4. The throttling ports of the cage, seen in the photography, are partially masked by a coaxially moving plug, so that the through-flow area is varied to increase by lifting up the plug. The jets from each of the ports collide with each other at the cage center. Most of the kinetic energy is dissipated by the collision, and the high speed jets would never impinge directly on the valve wall.

All of the cages tested have nearly equal opening area at the same lift. The stem of valve is fixed rigidly by a lift-

adjusting device at a required opening position. The series of the testing are made under steady condition which is determined by both of readings of pressures P_1 at the valve inlet and P_2 at the outlet.

The pressures P_1 and P_2 are measured by transducers with accuracy of 0.25% F.S. at sections of $2D$ and $4D$ respectively in the upstream and in the downstream of the valve; where D is the pipe diameter. Temperature is measured by a C-C thermocouple of 2 mm diameter at $2D$ upstream of the valve. A condenser microphone is located at 1 m away from the pipe surface and at 1 m downstream of the valve.⁽¹¹⁾ A quartz load cell is installed to the stem to measure the thrust force. All of the data are recorded on magnetic tapes, and analyzed by a off-lined FFT processor. Specifications of the instruments are listed in Table 1.

3. Throttling Characteristics of Cage-guided Valve

Figure 5 shows a typical throttling characteristics of mass flow rate G versus pressure reduction ratio $\sqrt{\Delta P/P_1}$. The mass flow rate shows a monotonous increase until the mass flow is choked. The throttling performance is represented in terms of a factor of pressure recovery $K_g \equiv \Delta P/\Delta P_{vc}$; where ΔP is the pressure drop across the valve and the ΔP_{vc} is the drop between

Fig. 5 Throttling characteristics of a cage-guided valve.

Fig. 6 Comparison of the throttling performance between cage-guided valves and other types of valves.

the inlet and the vena contracta. This factor means a degree of dissipation of the kinetic energy downstream of the throat; i.e. as the factor approaches one, the pressure recovery by the kinetic energy conversion reduces to zero.

The recovery factor K_g is evaluated at the critical pressure ratio by a method of Filippis.⁽¹²⁾ A classical Wantzel-Saint Venant formula applied for orifice is reduced to express the critical mass flow rate G_{cr} as,

$$G_{cr} = 1.41 \sigma_r C_v \sqrt{P_1 / \sqrt{Z T_1}} \quad (1)$$

and, for air,

$$f = 1.46 (\Delta P / P_1)_{cr} + 0.307 \quad (2)$$

where σ_r : a constant depending on the isentropic exponent (0.484, air), C_v : incompressible valve flow coefficient, Z : compressibility factor, T_1 : temperature in °K, and the subscript cr denotes the critical condition.

The broken line in Fig. 5 is obtained by Eq. (1), and the intersection with the measured flow rate curve determines the value of $\sqrt{\Delta P / P_1}$ at the critical condition. Then, introducing the critical pressure ratio upstream of the vena contracta, K_g is evaluated.

In Fig. 6, the factor K_g of various valves are compared. The values of cage valves are considerably high in a wide range of the opening. This indicates that momentum mixing is achieved almost within the cage, and that the collision of the jets is more effective to the energy loss than streamwise velocity shear such as expected in the other type of valve.

4. Noise Characteristics of Cage-guided Valve

4-1 Sound pressure level SPL

A typical pattern of the change of the over-all noise level is shown in Fig. 7. The measurement is made in two different ways of pressure setting, i.e., case (1): variable outlet pressure P_2 at a constant inlet pressure P_1 , and case (2): variable P_1 at a constant P_2 .

In the case (1), the noise level SPL increases monotonously with the valve pressure ratio before the critical condition, denoted by c.p., is reached. At

a pressure ratio of three slightly higher than the critical, a level increase of almost 5 dB takes place and the highest level is attained. The noise radiation at this level continues to exist in a range of $P_1/P_2 < 5$. Then, the level is reduced to approach another constant value which is lower by 3~4 dB than the highest. In the range of the highest level, a fluctuation or an intermittency is significantly noticed in the reading of the sound meter.

In the case (2) of variable P_1 where P_2 is at atmospheric, the level coincides with the locus of the rightmost data measured in the case (1). Then the data indicate the lower limit of the levels measured in the series.

The noise level changes in a complicated and irregular manner with pressure ratio as described above. The characteristics are influenced significantly by the number of the ports as well as by the valve opening. The mechanism by which those particular characteristics are induced must be related with the behavior of the colliding jets. The collision may be unstable or oscillatory in some ranges of the Mach number and diameter-to-length ratio of the jet, especially in the 2-Ports valve. Large scale eddies traveling along the jets will also be formed to generate a noise of discrete frequency.

Fig. 7 Typical pattern showing the change of the noise level induced by a cage-guided valve.

Fig. 8 Influence of the valve opening on the acoustic efficiency of 4-Ports valve. Efficiency of the free jet is cited from ref. (14).

Fig. 9 Influence of the number of ports on the acoustic efficiency. Transitional increases are shown at the critical in 2- and 3-Ports.

Fig. 10 The frequency spectra corresponding to the data in Fig. 9.

Fig. 11 Transitional increase of noise level at the critical is attributed to the resonant modes originated in the plug cavity.

4-2 Acoustic efficiency

For the convenience of comparing the noise level data with each other, an acoustic efficiency is introduced. The efficiency is a factor with which a part of the kinetic energy W_k of the fluid is converted into the acoustic energy W_a , and is expressed as,

$$\eta = W_a / W_k \quad (3)$$

In control valve application, the noise is mainly radiated from downstream pipe wall as an acoustic line source. Using a mathematical model⁽¹³⁾ of pipe-transmission loss which depends on the pipe geometry and the fluid condition, and expressing the kinetic energy of fluid in terms of the valve flow coefficient C_v , the noise level at a distance of 1 m from the pipe surface is correlated as,

$$SPL = 10 \log \left(\frac{2.6 \times 10^5 C_v C_f P_1 P_2 D^2 T_0}{t^3} \right) \text{ in dB}, \quad (4)$$

where $C_f = \sqrt{K_g}$, t : pipe-wall thickness in mm, and D : pipe diameter in mm.

The efficiency η derived from the measured data is compared in Figs. 8 and 9 with the theoretical efficiency of a free isentropic jet expressed as $\eta_f \approx 10^{-4} M^5$.⁽¹⁴⁾ If the port number and the valve opening are fixed, the acoustic efficiency is determined by the pressure ratio, or the jet Mach number M with a slight participation of the inlet pressure P_1 . Transitional increase of the efficiency, as mentioned previously, appears in a range of relatively low valve opening. It should be noticed that the acoustic efficiency of a colliding jet of small diameter is significantly low, and that the efficiency increases as the jet diameter becomes large and almost coincides with the efficiency of a free isentropic jet.

In Fig. 9, the acoustic efficiency at an intermediate valve opening is compared. The fluctuations of noise level are the most remarkable for valves of 2-Ports and 3-Ports within a narrow range of pressure ratio around the critical. The amplitude of the fluctuation is about 10 dB in the present case. In 4-Ports and 6-Ports cages,

the fluctuation disappears, and the efficiency shows a smooth increase with P_1/P_2 . An interest is that the 6-Ports cage shows a very gradual increase in the efficiency at pressure ratio higher than three.

4-3 Frequency analysis and dominant frequency components.

The characteristics of the acoustic noise are analyzed in more detail by a FFT analyzer. The number of the ensemble average is 80, and the frequency resolution is 100 Hz in the band width of 20 kHz.

Figure 10 shows the noise spectra corresponding to the typical situations denoted in Fig. 9 by ① and ②. Various components of discrete frequency appear in the spectra. Particular components denoted by $P_{n,m}$ and $D_{n,m}$ change in their respective levels versus pressure ratio as shown in Fig. 11; where $P_{n,m}$ and $D_{n,m}$ represent acoustic modes in cylindrical volumes respectively within the plug cavity and the downstream pipe.

It is frequently recognized in applications of valves in a piping system that a component in resonance with the pipe volume dominates in the noise, and that the frequency is determined by the pipe diameter. In the cage-guided valve, an acoustic resonance with the plug cavity must be as well taken into account; see the flow configuration in Fig. 1.

The frequency of an acoustic mode with the cylindrical configuration is obtained by solving the following equation⁽¹⁵⁾,

$$\frac{d}{dx}\{J_n(x)\} = 0, \dots \dots \dots (5)$$

where $J_n(x_m)$ is a Bessel function of the first kind of n -th order, and the successive solution x_m describes a particular mode. The mode (n,m) expresses a mode of m nodal circles with $2n$ angular sections. The resonant frequency in the present case takes the following value,

$$\begin{aligned} D_{1,1} &\approx 2.5\text{kHz}, D_{2,1} \approx 4.4\text{kHz}, \\ D_{0,3} &\approx 13.5\text{kHz}, \\ P_{2,1} &\approx 5.2\text{kHz}, P_{0,3} \approx 18\text{kHz}. \end{aligned}$$

It is clearly recognized in Fig. 11 that the resonant frequency components $P_{2,1}$ and $P_{0,3}$ contribute to the transitional increase of noise level at the near critical pressure ratio; i.e.,

(1) In case of the 2-Ports valve, the resonant components with plug modes $P_{2,1}$ and $P_{0,3}$ exhibit very singular behavior at the near critical pressure ratio. However, the resonant component with the pipe mode $D_{1,1}$ grows up beyond the component $P_{2,1}$ at pressure ratio higher than 5, and takes the highest level.

(2) In 3-Ports valve, $P_{2,1}$ is the only source of the transitional noise increase.

(3) In 4- and 6-Ports valve, the plug mode components diminish to low intensities, and the smooth increase of the pipe mode components almost determine the smooth increase of the over all level.

Concludingly, the transitional and fluctuational increase of acoustic efficiency is attributed to the oscillating jets in resonance with the acoustic oscillation of the plug cavity. The

acoustic modes in the downstream pipe almost determine the monotonous characteristics in the noise level increase over the entire range of pressure ratio.

An experimental result which is closely related to the generation mechanism of acoustic mode $P_{2,1}$ is shown in Fig. 13. Since the mode $P_{2,1}$ describes a transverse oscillation in cross-sectional plane of the plug cavity, the mode would be influenced by the position of pressure balance holes drilled at the top of plug cavity, see Fig. 12. At a relative position to the jets as illustrated in Fig. 13(a), the mode $P_{2,1}$ is recorded with a level of almost 90 dB. On the other hand, the intensity of the mode is reduced to a level less than 80 dB at a hole position of (b), and the pressure ratio where the mode appears is restricted in a significantly narrow range. This result shows a possibility of suppressing the transitional increase of noise by locating the balance hole at appropriate positions relative to the jets.

Figure 14 indicates the region of the transitional resonance in terms of the valve opening and the pressure ratio. At a large valve opening, such resonant phenomena are hardly caused because the flow does not take a sharp jet configuration.

Fig. 12 Geometry of plug and pressure balance hole.

(3-Ports, Lift = 48%, $C_v = 15.4$)
 $P_1 = 1.1 \text{ MPa}, P_1/P_2 = 2.11$

Fig. 13 Influence of the position of balance hole on the mode $P_{2,1}$

Fig. 14 Region showing the generation of transitional noise increase

It is concluded that the transitional increase originated in a resonance with plug cavity grows up under the limited condition of low valve opening and small number of jets.

The understanding of the mechanism of the resonance requires further research focused on the unsteady behavior of the multiple jets colliding in the plug cavity. The authors expect the following feed back mechanism; i.e. the shear layer instability of the jet is excited by pressure waves in the plug cavity, and large scale eddies are formed in each of the jets. Their collision at the center produces a pressure wave system, whose mode must coincide with one of the acoustic mode possible to appear in the cavity.

4-4 Comparison with free expansion jets noise

Logarithmic proportions of the acoustic efficiency to the efficiency of the free jet are shown in Fig. 15. The characteristics at high Mach number deviate from the subsonic characteristics. Nevertheless, it is found that, in case of high C_v -value or of thick jets collision, the acoustic efficiency coincides well with the efficiency of the free jet. While, in case of low C_v -value, the jet diameter is

Fig. 15 Logarithmic proportions of acoustic efficiency η to the efficiency η_f of free jet.

Fig. 16 Plot of velocity exponent n for subsonic and supersonic valve flow

small, and the efficiency is significantly lowered. This result means that a low-noise valve should be designed to utilize collisions of multiple concentrating jets with small diameter.

Jet noise level is frequently correlated with jet velocity V by

$$\eta \propto V^n \quad \dots \dots \dots (6)$$

Using the isentropic value as the velocity V , exponent n of the valve noise is plotted in Fig. 16. The abscissa $D_c/2d$ represents the jet length divided by a effective jet diameter d , which is determined by replacing the geometrical port with a circular port of equivalent area.

In subsonic flow range, the valve noise exponent almost agree with the well known exponent of a free jet estimated by a quadrupole model. However, the supercritical data are scattered out of range of expecting any meaningful rules.

Fig. 17 Pressure fluctuation measured inside of pipe and acoustic noise measured outside.

4-5 Pressure fluctuation inside of the pipe downstream of valve.

A direct source of the far field noise is the fluctuating pressure force acting to the inner wall of the pipe. The pressure is measured using a semi-conductor microphone at 10 times diameter downstream the valve, and the typical spectrum is shown in Fig.17(a). The corresponding noise spectrum at the outer surface is also shown in the figure (b), which is estimated by a simple addition of the distance attenuation to the far field noise data.

In both of the figures, participations of the air supply and the exhaust pipes are indicated by broken lines, which is obtained by removing the test valve. The upstream influence from the exhaust system is isolated in a fluid mechanical sense, because the back pressure controller is choked. The participation is only scant so long as the frequency component higher than 5 kHz is concerned.

The difference in the magnitudes in each of the spectrum means the so-called transmission loss, and it is estimated as 60 dB in the mean. This is higher by more than 10 dB than a value estimated by an empirical rule.⁽¹⁶⁾ The only component of discrete frequency in the inside data is related to the mode $P_{2,1}$ and it is extremely dominated in the spectrum, whereas a numerous number of spikes including the components of mode $P_{2,1}$ and $D_{0,3}$ appear in the spectrum of the outside noise.

Thus, the major interest among the dynamic problems of the valve flow is to find the mechanism, how and why the mode $P_{2,1}$ is excited. Due to the transmission characteristics of the pipe wall, the outside noise data show considerably different features. The component of mode $P_{2,1}$ is too much reduced in compared with the component of mode $D_{0,3}$ and components due to the structural vibration of pipings.

5 Characteristics of Thrust Acting to the Stem.

5-1 Static thrust

The thrust forced by the flow to the stem is plotted versus the pressure drop ΔP in Fig. 18, where the inlet pressure P_1 is held at a constant value. The continued lines indicate estimated values by a simple relation as,

$$F = A_s(P_2 - P_a) + B(L)t(P_2 - P_2), \dots \dots (7)$$

Fig. 18 Characteristics of static thrust acting to the stem

Fig. 19 Levels of thrust fluctuation and the frequency spectra.

where A_s : cross-sectional area of the stem, P_a : atmospheric pressure, $B(L)$: partial length of the plug brim in contact with the jets, P_p : static pressure of the jet at the port and t : thickness of the plug brim. The first term of the right-hand side of Eq.(7) indicates the force by the pressure in the plug cavity to the stem, and the second term expresses the pressure force due to jets to the plug brim. P_p is taken approximately as the outlet pressure in the subsonic flow range, and is assumed equal to the critical pressure $0.528P_1$ for the supersonic flow.

The estimation shows the fundamental of the characteristics of the stem thrust. In 6-Ports valve, the measured values agree well with the estimated values at a high valve opening. While, the measured data show significant discrepancies with the estimations as both the valve opening and the number of ports decrease. The influence of leak flow through the gap between the cage and plug should be also taken into account. However, it should be noticed that the thrust of 2-Ports valve increases with an increasing lift. By the

peculiarity, a self-sustained oscillation may be possibly induced for the valve if the stem is supported with a soft spring.

5-2 Fluctuation of thrust

Figure 19 indicates that the thrust fluctuation takes the maximum value at the near critical pressure ratio. The r.m.s. value of the fluctuation of 6-Ports valve surpasses that of 4-Ports valve, and the value increases to a comparable or surpassing level to the static thrust. The ratio of the thrust fluctuation $\bar{F}_{rms}$ to the static thrust F at the critical pressure ratio is shown in Fig. 20. In the 6-Ports valve, in spite of the acoustic efficiency is the lowest among the test cages as described in paragraph 4-4, the relative fluctuation level of the stem is the highest among them.

The natural frequency of the valve-stem support system, which is about 1 kHz, dominates in the spectrum of the thrust fluctuation. Corresponding to the transitional noise increase, the resonant mode $P_{2,1}$ appears in the spectra of the stem vibration. While, the amplitude is not so high. In contrast with this, the natural frequency of the support system appears in the noise spectra with a considerably low amplitude, as shown in Fig. 10.

Fig. 20 Relative levels of thrust fluctuation.

6 Conclusions

The characteristics of noise and vibration induced by a cage-guided valve are investigated. Inserted in a practical valve, the test cages of 2-, 3-, 4- and 6-Ports are used in the investigation. The results are summarized as follows;

- (1) The pressure-recovery factor of cage-guided valve takes a considerably high value over the wide range of valve opening. This means that the energy dissipation by the collision of multiple jets is the major part of the total dissipation by the valve.
- (2) The acoustic efficiency at high valve lift increases with pressure ratio in a similar manner to the efficiency of a free jet. At a low valve lift, the acoustic efficiency is lowered below the efficiency of the free jet.

- (3) The noise reduction is significant as the number of ports increases. However, in 2- and 3-Ports cages, a transitional increase of noise level appears remarkably at the near critical pressure ratio. This is due to an acoustic resonance with the plug cavity. For a 6-Ports cage, such transitional noise increase is suppressed.

- (4) The static stem thrust of 2-Ports valve increases as the valve opens. This characteristics may be a cause of a self-sustained vibration of the stem if it is positioned by a soft spring. The amplitude of the fluctuation of stem thrust increases remarkably as the number of jets increases.

Among various phenomena stated here, the coupled oscillation between the supersonic colliding jets and the cavity flow within the plug is the most interesting to be studied. On this event, further investigation is continued by means of visualization of the flow structures.

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References

- (1) Arant, J. B., Instrum. Technol., Vol. 20, No.3 (1973-3), p.37.
- (2) Hynes, K. M., ISA Trans., Vol.10, No. 4 (1971), p.416.
- (3) Nakano, M., et al., Trans. Japan Soc. Mech. Engrs. (in Japanese), Vol.45, No.391, Ser.B (1979-3), p.350.
- (4) Nakano, M., et al., Trans. Japan Soc. Mech. Engrs. (in Japanese), Vol.48, No.431, Ser.B (1982-7), p.1266.
- (5) Nakano, M. and Tajima, K., Trans. Japan Soc. Mech. Engrs. (in Japanese), Vol.49, No.444, Ser.B (1983-8), p.1658.
- (6) Nuding, W., Regelungstechnische Praxis, No.7 (1978-7), p.194.
- (7) Araki, T., et al., TOSHIBA Review, Vol.36, No.7 (1981-6), p.648.
- (8) Kerschen, E. J. and Johnson, J. P., J. Sound Vib., Vol.76, No.1 (1981), p.95.
- (9) Föllmer, B. and Zeller, H., 3rd Int. Conf. Pressure Surges (1980), p.429.
- (10) Okutsu, R., et al., Report of Researches, Nippon Institute of Technology, (in Japanese), Vol.83, No.1, (1983-1), p.1.
- (11) ISA Draft Standard, dSP75.07 (1979)
- (12) Filippis, L., ISA Trans., Vol.13, No.4 (1974), p.347.
- (13) Reed, C., ISA Trans., Vol.17, No.1 (1978), p.105.
- (14) Lighthill, M. J., AIAA J., Vol.1, No.7 (1963), p.1057.
- (15) Reed, C., ISA Paper, 76-836, No.10 (1976).
- (16) Fagerlund, A. C., ASME Publ., 73-WA/PID-4 (1973).