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VIBRATION EXPERIMENTS AND NUMERICAL SIMULATIONS OF PULSATION BEHAVIOR IN ACTUAL SIZE MOCK-UP PIPING

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ABSTRACT

Vibration experiments for pressure pulsation behavior were made using actual size mock-up piping of nuclear power facilities. The mock-up was a closed loop consisting of a three-strand plunger pump, tanks, piping and valves. It was 40 m long to allow interaction of the acoustic resonance frequency of fluid inside with the mechanical natural frequency of the piping. The influence of valve closing and opening operations to change inner pressure during pump operations on the pulsation boundary condition was investigated in this study. A drastic change in the boundary condition of the acoustic resonance behavior by using a slightly different valve opening ratio to set a different inner pressure was shown in the experimental results. The phenomenon was numerically simulated by using the method of characteristics. The simulation results showed that the boundary condition of the acoustic resonance changed from the closing–opening condition to the closing–closing condition when the valve opening ratio was changed slightly from 10 % to 15 %. This indicated that the boundary condition of the acoustic resonance had a pulsed change. Therefore, the boundary condition of the acoustic resonance was sensitive to a slight change of the valve opening ratio.

1. INTRODUCTION

Pumps and compressors can cause pressure pulsation in piping systems in many industrial plants, including nuclear power plants. In both reciprocating compressors and plunger pumps having intermittent suction and discharge especially, the change in flow is so large that pressure pulsation, including high harmonic components of rotation frequency of the compressors and pumps, occurs in the piping system. When the frequency of the pressure pulsation is close to the acoustic resonance frequency, the amplitude of the pressure pulsation increases, and then it induces vibration of piping and malfunction of electronics and control instrumentation. This frequently has a heavy impact on the plant operation [1]. In both compressors and pumps with rotor blades such as turbo machines, the pressure pulsation and fluid noise with the frequency equal to the multiplicative frequency of rotational speed and the number of the blades and the high order harmonic frequencies occur and have further similar impact. Furthermore, because of the resonance between the frequency

of fluid vibration and the natural frequency of the piping, the piping vibration increases and vibration fatigue leads to pipe failure. Many researchers [2–17] have studied pressure pulsation in piping systems in order to prevent these phenomena. Topics have included experiments regarding pulsation characteristics [8,11,13], the formulation and modeling of pulsation behavior [2,6,7,14,17], analysis methods of pulsation [3,4,16], and evaluation methods of pulsation considering coupling with pipe structure or compressors [5,9,10,12,15]. For instance, Hayama et al. [6] discussed damping for pulsation analysis, Fujita and Tanaka [12] proposed a response analysis method considering coupling between pulsation behavior and pipe oscillation, and Kato et al. [15] proposed a pulsation analysis method considering fluid-structure interaction in compressor-piping system. However, it is still difficult to estimate the magnitude of the pressure pulsation in actual plants and to predict the pulsation behavior including acoustic resonance. Therefore, it is necessary to clarify the behavior of the pressure pulsation including acoustic resonance by using actual size mock-up piping. It also is important to verify and validate numerical simulation methods for piping design by using the experimental results.

The purpose of this study was to improve the prediction accuracy of pressure pulsation behavior which has a heavy impact on operation of nuclear power plants. First, the mock-up piping system was made using a pump, pipe and tanks connected in a closed loop. In the piping system, the frequency of pressure pulsation, the acoustic resonant frequency and the mechanical natural frequency of piping system were set to coincide closely. Pressure pulsation including acoustic resonance was examined by vibration experiments. Next, the experimental results were simulated by numerical analysis and the analysis accuracy was discussed.

2. EXPERIMENTS AND SIMULATIONS

2.1 Experimental set-up

Figures 1 and 2 show a photograph and schematic view of the experimental set-up, respectively. The dimensions of the experimental set-up were approximately 4.0 m × 4.5 × m × 1.2m. A three-strand plunger pump (reciprocating pump) was used in the set-up. The main specifications of the pump are shown in Table 1. The pressure regulation valve was installed at the inlet of the tank (one end of the pipe) to set the discharge line pressure between 1 and 3 MPa. A typical angle valve was used for the pressure regulation. The pipe length was designed to be 40 m to generate the pressure pulsation with acoustic resonance. The pipe was made of SUS 304 and the nominal outside diameter was 3/4B (27.2 mm) and the thickness was Sch40 (2.9 mm). As shown in Fig. 3, pipe supports were designed to be movable to allow matching of the natural frequencies of fluid inside with that of the pipe structure in the set-up.

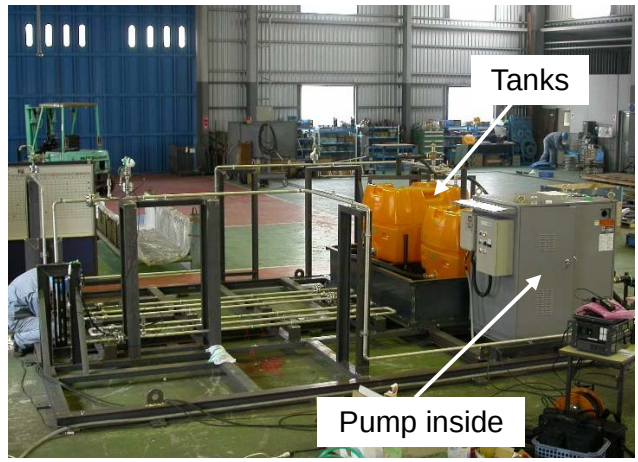


Fig. 1 Photograph of mock-up piping

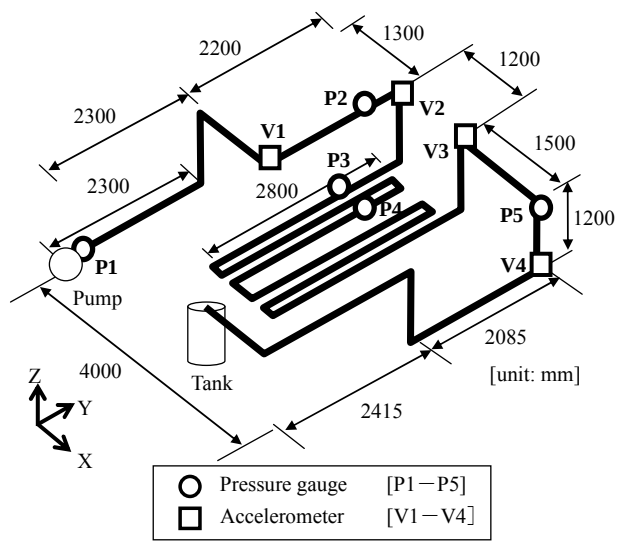


Fig.2 Outline of mock-up piping

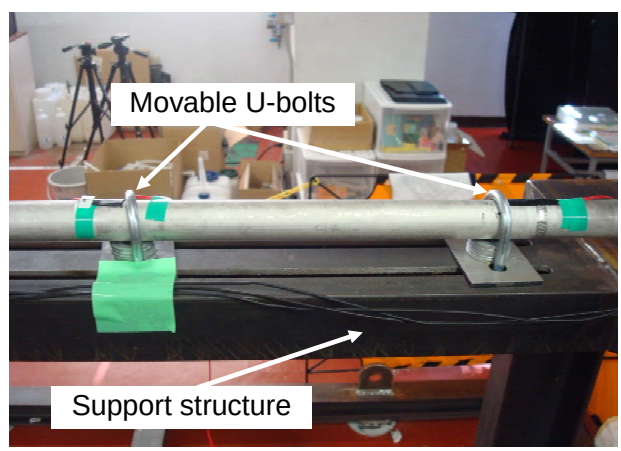


Fig. 3 Movable pipe support

Table 1 Specifications of the pump

Vender	Arimitsu Industry Co., Ltd.
Type	C-2550
Number of cylinders	3
Rotational speed (rpm)	75 – 460
Discharge flow rate (L/min)	24.3 – 149.0
Discharge pressure (MPa)	1 – 3

2.2 Experimental method

After the pressure in the pipe was adjusted by the pressure regulation valve at the tank inlet, vibration experiments were conducted under the sweep operation or steady operation of rotation frequency of the pump. The experimental conditions are summarized in Table 2. The pressure and acceleration generated during the experiment were measured by pressure gauges (P1 to P5) and accelerometers (V1 to V4), located as shown in Fig. 2. The sampling frequency was 512 Hz.

The frequency of acoustic resonance caused by pressure pulsation was calculated by Eq. (1) for the case in which the boundary conditions of the pump outlet and the tank inlet were closing–closing or opening–opening and by Eq. (2) for the case in which the boundary conditions of the pump outlet and the tank inlet were opening–closing or closing–opening. For instance, assuming that acoustic velocity was 1100 to 1200 m/s, 14.9 to 16.2 Hz was calculated for the first order resonance frequency and 29.7 to 32.4 Hz for the second order by using Eq. (1), and 7.4 to 8.1 Hz for the first order, 22.3 to 24.3 Hz for the second order and 37.2 to 40.5 Hz for the third order by using Eq. (2).

$$f_0 = (c/2L)\cdot n \qquad (n = 1,2,3\cdots) \qquad (1)$$

$$f_0 = (c/4L)\cdot(2n-1) \qquad (n = 1,2,3\cdots) \qquad (2)$$

Here f_0 is resonance frequency, c is acoustic velocity, L is pipe length and n is the order number. These results are summarized in Table 3.

In order to match the natural frequencies of fluid inside the piping with that of the pipe structure, the location of the support on both sides of the four pipe elbows near the accelerometers (V1 to V4 in Fig. 2) was adjusted. After the adjustment, the first natural frequencies of the out-of-plane vibration were 19 Hz for the elbows near accelerometers V1 and V3 and 32 Hz for V2 and V4.

Table 2 Experimental conditions

Pump operation modes	Sweep	Steady		
Rotational speed (rpm)	150 to 450	150	342	450
Pulsation frequency (Hz)	7.5 to 22.5	7.5	17.1	22.5
Discharge flow rate (L/min)	51.8 to 147.4	51.8	110.1	147.4
Sweep speed (rpm/s)	1	–		
Pressure (MPa)	1, 2, 3			

Table 3 Calculated frequency of acoustic resonance

Acoustic velocity (m/s)		1100 – 1200		
Order number		1	2	3
Boundary condition and frequency (Hz)	Opening–Opening or Closing–Closing	14.9 – 16.2	29.7 – 32.4	44.6 – 48.6
	Closing–Opening or Opening–Closing	7.4 – 8.1	22.3 – 24.3	37.2 – 40.5

Table 4 Results of hammering test

Location	Natural frequency (Hz)	Damping ratio (–)
V1	19.13	5.83×10^{-3}
V2	32.5	5.95×10^{-3}
V3	19.25	1.42×10^{-2}
V4	32.13	1.33×10^{-2}

2.3 Numerical simulation method

The analysis model is shown in Fig. 4. The pipe line was modeled and the pump and the tank were considered as boundary conditions. The numerical simulation was conducted under the same conditions as the experimental conditions shown in Table 2. The acoustic velocity was set to 1200 m/s and the inner surface roughness of the pipe was set to 0.05 mm. In this study, the one-dimensional network thermal-hydraulic analysis code Flowmaster [18] was used for simulating pressure pulsation behavior. The pressure pulsation behavior was calculated by the method of characteristics. The discharge flow wave of the pump based on the experimental results was used as input. The discharge flow wave was determined by the following equation:

$$y = F_1 + F_2 \cdot \sum_i^n a_i \cdot \sin 2\pi f_i (t - \phi_i) \qquad (3)$$

where y is total discharge flow rate (m^3/h), F_1 is steady flow rate (m^3/h), F_2 is unsteady flow rate (m^3/h), a_i is amplitude ratio of the i-th high harmonic component, f_i ($f_i = i\cdot f_1$) is frequency of the i-th pulsation mode (Hz), t is time (s) and ϕ is phase lag (s). F_1 was determined from specifications of the plunger pump. F_2 was determined based on the ratio between steady flow and unsteady flow in the experiments. For the sweep operation test and the steady operation test, n=1 and 4 were chosen, respectively.

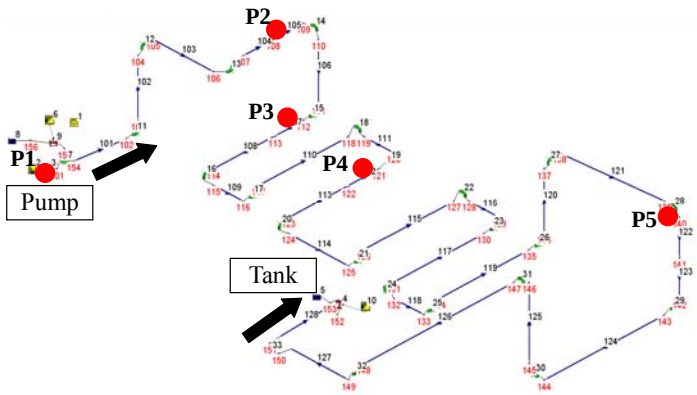


Fig. 4. Simulation model

3. RESULTS

3.1 Experimental results

The hammering test was done to examine the mechanical natural frequency of the pipe structure. The test results are summarized in Table 4. The results indicated the mechanical natural frequency of the pipe structure was set to the target frequency (the acoustic resonance frequency in the piping system) by adjustment of the support locations. Therefore, the mock-up piping was set to an interactive state between the acoustic resonance frequency and the mechanical natural frequency of the pipe structure. This state is one of the causes of pipe failure by vibration fatigue in actual plants. Also, the damping ratio of the pipe structure was very small because of the relatively simple configuration of the piping system.

Table 5 shows damping ratio at the peak frequency of pressure pulsation when the inner pressure was 1 MPa in the sweep test. The damping was calculated by the half power method. As the Table 5 results indicated the damping was large because of the small bore pipe.

Figures 5, 6 and 7 show waterfall plots of pressure pulsation measured at P1 during sweep operation of the pump at inner pressures of 1 MPa, 2 MPa and 3 MPa, respectively. The pulsation behavior at P1 was considered and discussed as typical experimental results in this paper. The location of peak frequency was determined by applying the frequency maximum value analysis in the waterfall plots. There was a peak at about 37 Hz for inner pressures of 1 MPa and 2 MPa as shown in Figs. 5 and 6, and there were two peaks at about 17 Hz and about 32 Hz for inner pressure of 3 MPa as shown in Fig. 7. The 37 Hz peak in Fig. 5 corresponded to the third order frequency of acoustic resonance in the boundary condition closing–opening, while the 17 Hz and 32 Hz peaks in Fig. 7 corresponded to the first and second order frequency of acoustic resonance in the boundary condition closing–closing. This indicated that the boundary condition of acoustic resonance drastically changed from closing to opening when the opening of the pressure regulation valve was slightly changed to adjust the inner pressure from 1 MPa to 3 MPa.

In many actual plants, the inner pressure of pipes is controlled by using pressure regulation valves. Therefore, a slightly different opening of the valve may often significantly affect the boundary condition of the pressure pulsation. This was probably one of the reasons which caused the pulsation estimation to be inaccurate.

Figure 8 shows typical frequency analysis results of the time history of the pressure pulsation. The time history was measured by pressure gauge P1 (located next to the three-strand plunger pump) under the inner pressure of 2 MPa and the rotational speed of 342 rpm. From this figure, the discharge flow from the three-strand plunger pump was found to include a lot of the high harmonics in addition to the fundamental harmonic (17.1 Hz) corresponded to the rotational speed of the pump. Significantly, the secondary component was predominant. This was thought to be due to the approximate rectification of discharge flow to a half wave by opening and closing the discharge flow valve.

Based on these experimental results, the input wave of the discharge flow was considered for the numerical simulation. The high harmonics from secondary to fourth orders were included in the input wave and the amplitude ratio among the harmonics was determined from the intensity ratio in Fig. 9. The phase difference among the high harmonics was determined based on the measured time history.

Table 5 Damping ratio at peak frequency of pulsation

Location	Order number	10 Hz vicinity		20 Hz vicinity		40 Hz vicinity	
		Frequency (Hz)	Damping ratio	Frequency (Hz)	Damping ratio	Frequency (Hz)	Damping ratio
P1	1	8.5	0.279	–	–	–	–
	2	–	–	23.7	0.180	40.0	0.082
	3	–	–	25.0	0.112	40.0	0.102

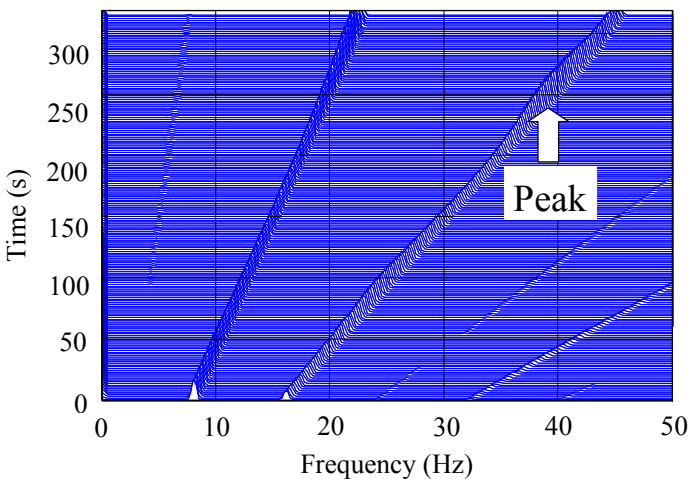


Fig. 5 Experimental results of sweep test
(P1, 1 MPa, Waterfall plot)

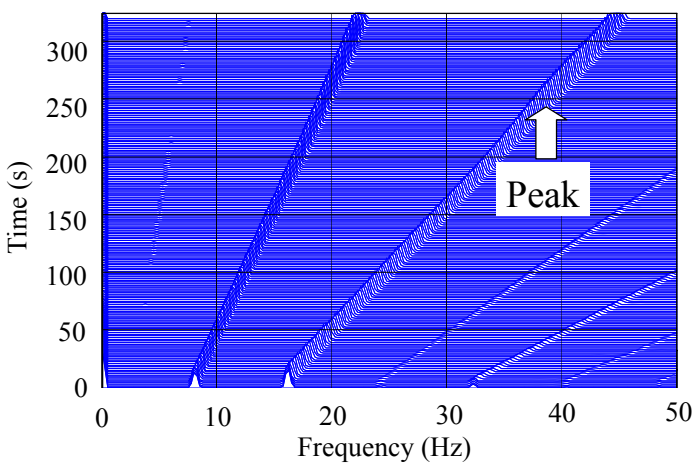


Fig. 6 Experimental results of sweep test
(P1, 2 MPa, Waterfall plot)

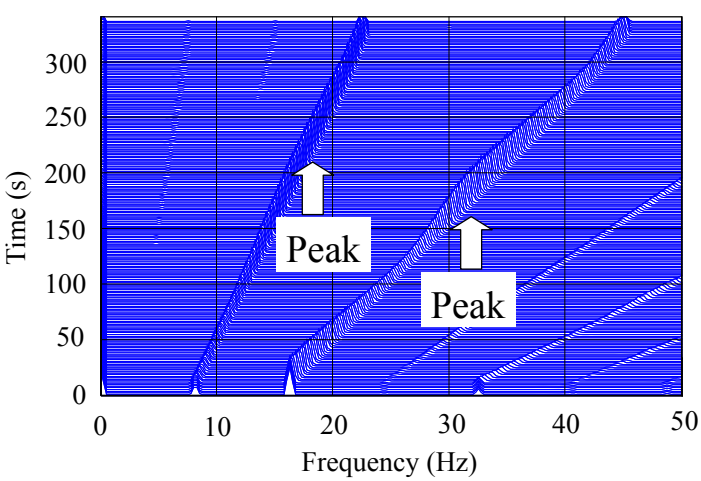


Fig. 7 Experimental results of sweep test
(P1, 3 MPa, Waterfall plot)

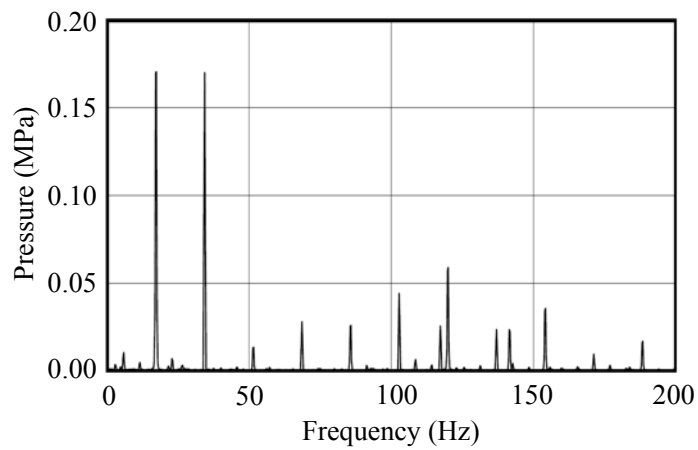


Fig. 8 Typical FFT analysis of steady operation test
(P1, 2 MPa, 342 rpm)

3.2 Simulation results

Figures 9, 10 and 11 show the waterfall plots of pressure pulsation analyzed under the sweep operation of the pump with valve opening ratios of 5 %, 10 % and 15 %, respectively. The acoustic velocity for the analysis was 1200 m/s. The discharge flow wave of the three-strand plunger pump was made and used as input. The input wave was made based on specifications of the pump in Table 1. The wave was also made considering the experimental amplitude ratio of the fundamental and high harmonics from second to fourth orders. The phase differences of high harmonics were assumed to be $\pi/2$. The numerical simulations were conducted under the sweep operation from 0 rpm to 500 rpm at sweep rate of 1 rpm/s. The simulation results in Figs. 9 and 10 showed that there was a peak at 15 Hz for the valve opening ratios of 5 % and 10 %, while the result in Fig. 11 showed that there was a peak at 8 Hz for the opening ratio of 15 %. It was thought that the 15 Hz peak was the first resonance frequency of the acoustic resonance under the boundary condition closing–closing and the 8 Hz peak was the first resonance frequency of acoustic resonance under the boundary condition closing–opening. The simulation results also indicated that the slight change of the valve opening ratio from 10 % to 15 % caused the boundary condition to be drastically changed from closing–closing to closing–opening.

Figure 12 shows the relationship between the valve opening ratio and acoustic resonance frequency from the first to third orders obtained from the simulation results. In this figure, the experimental acoustic resonance frequency regions at the inner pressure conditions of 1, 2 and 3 MPa are also shown to estimate the corresponding valve opening ratio. The comparison of the simulations with the experiments suggests that the inner pressure conditions of 1 and 2 MPa corresponded to more than 15 % of the valve opening ratio and 3 MPa corresponded to less than 10 %.

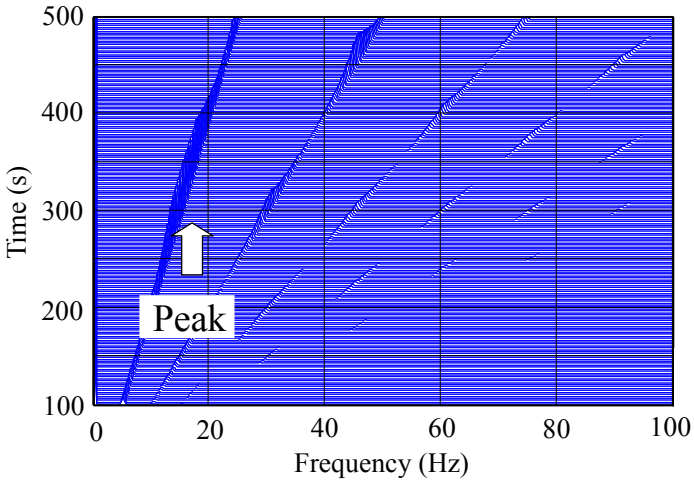


Fig.9 Simulation results of sweep test
(P1, Valve opening ratio 5 %, Waterfall plot)

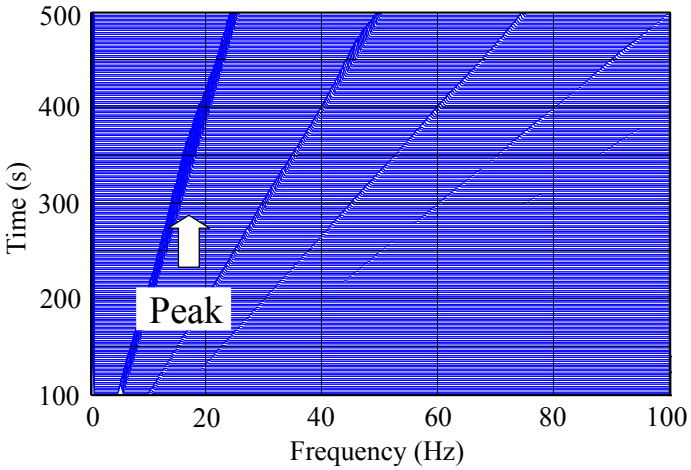


Fig.10 Simulation results of sweep test
(P1, Valve opening ratio 10 %, Waterfall plot)

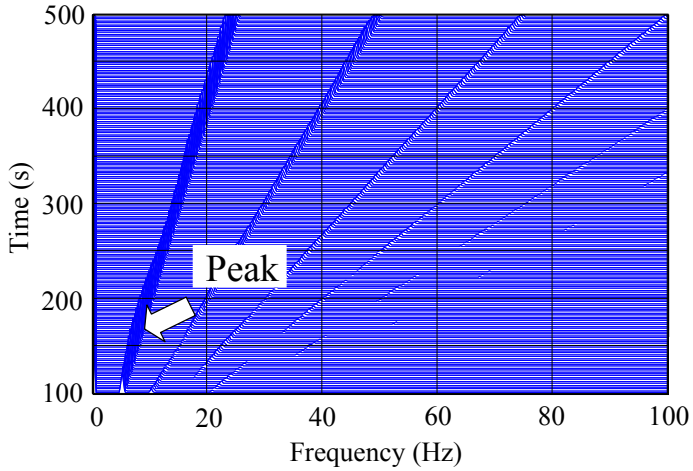


Fig.11 Simulation results of sweep test
(P1, Valve opening ratio 15 %, Waterfall plot)

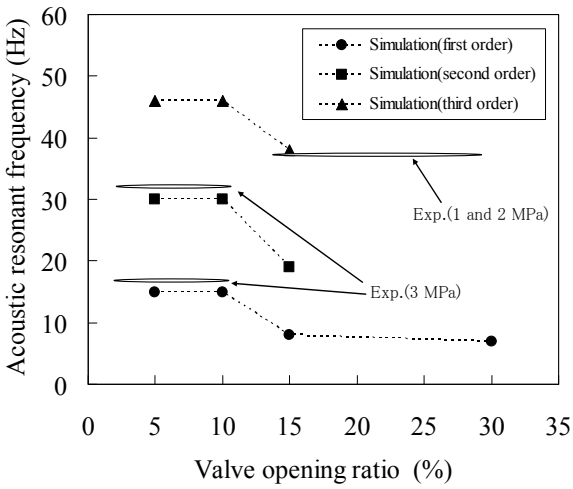


Fig. 12 Relationship between acoustic resonance frequency and valve opening ratio

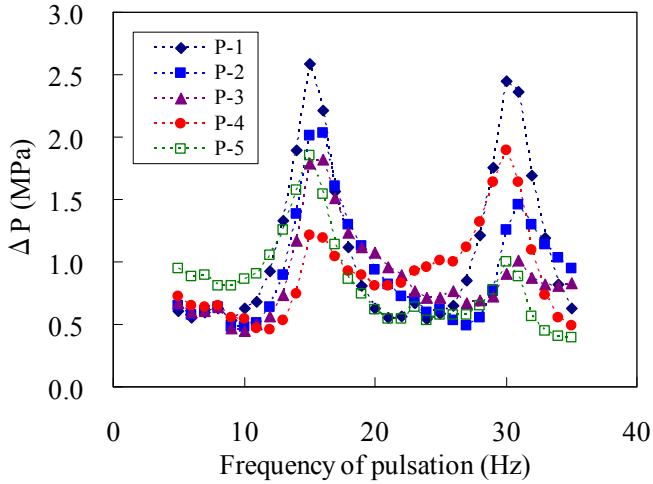


Fig. 13 Sensitivity analysis in steady pump operation
(Valve opening ratio 5 %)

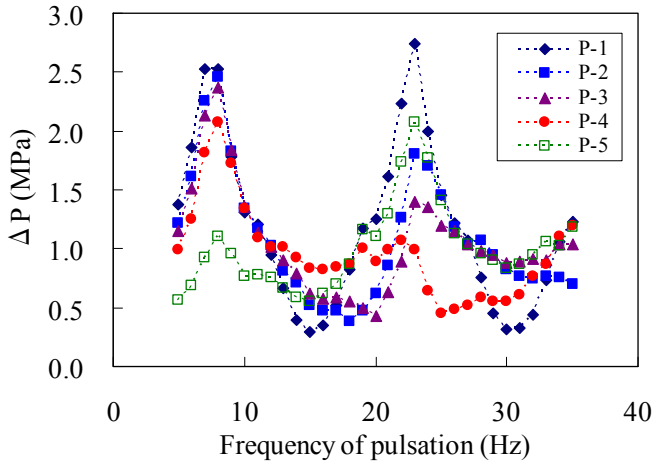
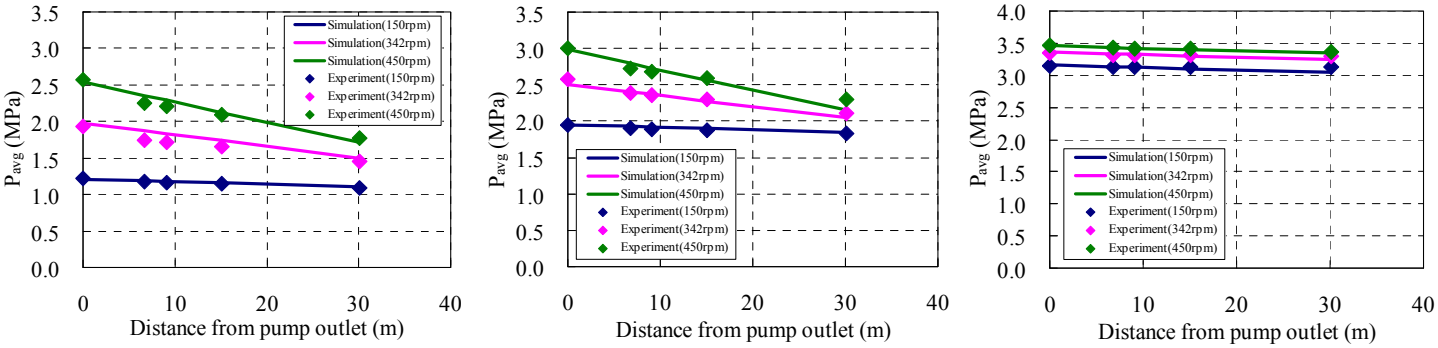
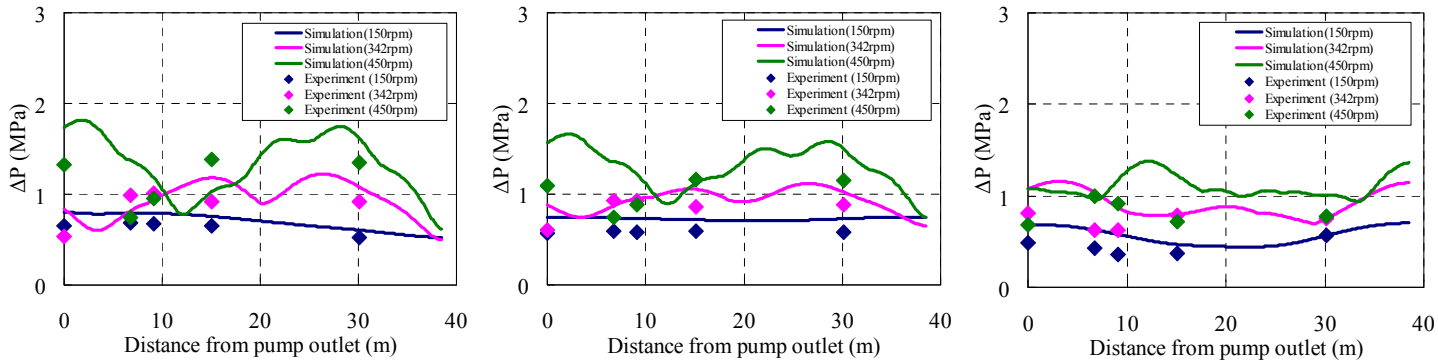


Fig. 14 Sensitivity analysis in steady pump operation
(Valve opening ratio 30 %)



(a) 1 MPa (b) 2 MPa (c) 3 MPa

Fig. 15 Comparison of simulation and experiment concerning steady pressure of pressure pulsation



(a) 1 MPa (b) 2 MPa (c) 3 MPa

Fig. 16 Comparison of simulation and experiment concerning unsteady pressure of pressure pulsation

The sensitivity analysis assuming steady pump operation was conducted next. Figures 13 and 14 show relationships between pressure change and pulsation frequency at the valve opening ratios of 5 % and 30 %, respectively. It was assumed for the analysis that the steady flow rate was 6.1 m^3/h and the unsteady flow rate was 0.9 m^3/h. The pressure response in the piping system was parametrically analyzed by increasing pressure pulsation frequency from 5 to 35 Hz at 1 Hz intervals. In Fig. 13, which was for the 5 % opening ratio, there were two peaks at 16 and 30Hz. These corresponded to the first and second resonance frequencies of acoustic resonance under the closing–closing boundary condition (See Table 3). In Fig. 14, which was for the 30% opening ratio, there were two peaks at 7 and 23 Hz. These corresponded to the first and second resonance frequencies of acoustic resonance under the closing–opening boundary condition (See Table 3). The boundary condition change from closing–closing to closing–opening was found to occur at the valve opening ratio of less than 30 %, which is a small valve opening ratio. The unexpected boundary condition change, which could occur even at the small valve opening ratio, could cause estimation of the pulsation to be inaccurate.

Figures 15 and 16 compare the simulation and experiment with regard to the relationships between pressure and the distance from the outlet of the pump. The experimental pressures in both figures were measured at P1 to P5. Figure 15 shows steady pressure and Fig. 16 shows unsteady pressure. In the simulations, the valve opening ratio was set to be 1, 2 and 3 MPa as the inner average pressure at the end of the pipe. The discharge flow wave was set based on the experimental results at pressure gauge P1, considering the experimental amplitude ratio and phase difference of the fundamental and high harmonic components from second to fourth orders.

The results of the simulations and the experiments agreed well in Fig. 15. However, there was a difference at the high rotational speed (450 rpm) in Fig. 16. The difference was thought to be due to the influence of the high harmonics of more than fifth order which were not considered in the simulations.

4. CONCLUSIONS

The mock-up piping system was built to investigate the pressure pulsation behavior including acoustic resonance. In the vibration experiments, the acoustic resonance frequency was examined, and the experimental results were compared to results of numerical simulations. The following conclusions were obtained.

(1) The boundary condition of acoustic resonance showed a pulsed change. It was drastically changed by having a slightly different valve opening ratio. This probably made accurate estimation of the pulsation to difficult.

(2) The boundary condition change from closing to opening occurred even at the small valve opening ratio, which was less than 30%. This can cause estimation of the pulsation not to be accurate.

(3) The simulations agreed closely with the experimental results. The simulation accuracy was thought to be improved more when considering much higher order harmonic components of pressure pulsation generated by the pump.

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