Mitigation Method for Pressure Fluctuations Induced by Acoustic Resonance

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Abstract. Multi-stage pumps used for boiler water supply of thermal power plants have large capacity and pressure. Fluctuations may cause vibration and noise in pumps, piping systems, and surrounding structures. Lots of studies have been conducted on the pressure fluctuation phenomenon of turbopump piping systems. Some of the factors are the rotor-stator interaction, a resonance between the fluid in the pipe and, excitation source and resonance between structure and pump operation. Besides, studies on mitigation method for pressure fluctuations have been conducted. Inserting orifice to add damping, adjusting the length of pipe to change resonance frequency and, installing resonator to mitigate a specific frequency. However, these studies have not quantified the solution to the pressure fluctuations, so further research is needed to elucidate the causes and establish countermeasures quantitatively. In this study, pressure fluctuations due to acoustic resonance are generated by using a speaker. An orifice, a bent tube, and a branch pipe are installed at the node and antinode of the secondary pressure standing wave in the pipe, and the effects on the standing wave are observed. Moreover, the mitigation method for the pressure standing wave is established by performing a sweep test. Numerical analysis is conducted by AMESim to validate the results of experiment. It is clarified that by inserting the orifice at the appropriate position of a standing wave, pressure fluctuations are mitigated. In the case of branch pipe, pressure fluctuations are effectively mitigated by inserting it having appropriate pipe length and boundary condition.

Keywords: resonance, standing wave, turbopump

1. Introduction

The pressure fluctuations generated by the operation of turbopump induce unstable phenomena such as vibration and noise in the piping system and the surrounding structures. When pressure fluctuates periodically if the frequency coincides with the eigenfrequency of the air column/liquid column in the pipe, a resonance occurs, and the vibration and noise become larger.

Sano studied the relationship between various parameters such as rotation speed and pipe length and liquid length and eigenfrequency using a double suction centrifugal pump [1-4]. As a result, the first report concluded that the eigenfrequency depends on the pipeline and the influence of the discharge rate, and the rotation speed is low. In the second report, the results are obtained that it becomes maximum when the pulsation source of the pump coincides with the node of the pressure fluctuations, and it becomes minimum when it coincides with antinode of them. In the third report, the eigenfrequency is calculated for the composite pipeline using the rigid matrix method and the transfer matrix method, and the effectiveness of replacing the pump with an equivalent pipeline is confirmed. In the fourth report, the result has found the relationship between the ratio of the equivalent pump length to the typical pump length without fluctuations, and the dimensionless frequency does not depend on the specific speed.
In this study, a pressure standing wave is generated in a pipe, and piping structures such as an orifice, a bent tube, and a branch pipe are installed at various positions, and the effects on the pressure standing wave are examined.

2. Nomenclature

- $d_i$ Dimensionless distance of each measurement point from speaker [-]
- $L_i$ Distance of each measurement point from speaker [m]
- $f$ Friction factor [-]
- $L$ Pipe length [m]
- $\rho$ Fluid density [kg/m$^3$]
- $D_{eqv}$ Equivalent diameter [m]
- $Q$ Volumetric flow rate [m$^3$/s]
- $A$ Cross sectional area of pipe [m$^2$]
- $K$ Bulk modulus [-]
- $\dot{m}$ Mass flow [kg/s]
- $C$ Sound velocity [m/s]
- $r$ Pipe radius [m]
- $k$ Specific heat ratio [-]
- $M$ Molecular weight [kg/mol]
- $R$ Gas constant [J/K·mol]
- $t$ Temperature [℃]

3. Experiment

An orifice, a bent tube, and branch pipe is installed to the main pipe separately, and the effect on pressure standing wave is investigated. Fig.1 and Table 1 shows the details of pressure sensor used for the measurements.

![Fig.1 Pressure sensor](image)

| Table 1: Details of the pressure sensor |
|--------------------------------------------------|
| Maker | ACO |
| Type | TYPE 4153N(TYPE2) |
| Release voltage | -32dB(25.1mV/Pa) |
| Pressure sensitivity | -33dB±3dB re 1V/Pa (22.4mV/Pa) |
| Frequency characteristics | 20Hz~10kHz |
| Maximum sound pressure level | 140dB |
| Self-noise level | 18dB(A) |
| Temperature coefficient | 0.01dB /°C or less |

3.1. Orifice insertion test

One orifice is inserted separately at the node, antinode, and midpoint between the node and antinode of secondary pressure standing wave, while changing the orifice opening to 25%, 50%, and 75%. Fig.2 shows the experiment apparatus and cross-sectional drawing of the piping structure used in this test. Table 2 shows the positions of the pressure sensors. The positions of pressure sensors are described as dimensionless distance from speaker, which is defined as $d_i = L_i / L$ ($i = 1,2,\ldots$). The locations for pressure sensors are same in three cases.

![Fig.2-(a) Experimental apparatus](image)
Fig. 2-(b) Experiment model (orifice insertion test)

Table 2: Measurement points (orifice insertion test)

| Point | Dimensionless distance from speaker $d_i$ |
|-------|------------------------------------------|
| 1     | 0.149                                    |
| 2     | 0.243                                    |
| 3     | 0.338                                    |
| 4     | 0.432                                    |
| 5     | 0.527                                    |
| 6     | 0.622                                    |
| 7     | 0.716                                    |
| 8     | 0.811                                    |
| 9     | 0.905                                    |

3.2. Bent tube insertion test
A bent tube is inserted into the node and antinode of the secondary pressure standing wave. Fig. 3 shows a model of this test. The total length of pipe is conserved during both cases as 1.654m. Table 3 shows the locations of the pressure sensors.

Fig. 3-(a) Experiment model (bent tube insertion test, node)
Fig. 3-(b) Experiment model (bent tube insertion test, antinode)
Table 3: Measurement points (bend insertion test)

| Point | Dimensionless distance from speaker $d_i$ | Node | Antinode |
|-------|------------------------------------------|------|----------|
| 1     | 0.1667                                   | 0.0948 |
| 2     | 0.3333                                   | 0.2159 |
| 3     | 0.4282                                   | 0.3364 |
| 4     | 0.6033                                   | 0.3967 |
| 5     | 0.6636                                   | 0.5175 |
| 6     | 0.7844                                   | 0.6667 |
| 7     | 0.9052                                   | 0.8333 |
| 8     | -                                         | 0.9964 |

3.3. Branch pipe insertion test

A closed-end and open-end branch pipe ($\lambda/4$, $\lambda/2$, $3\lambda/4$) is inserted into the secondary pressure standing wave. Fig. 4 shows the model of the piping structure of this test. And Table 4 shows the positions of the pressure sensors.

![Fig. 4 Experiment model (branch pipe insertion test)](image)

Table 4: Measurement points (branch pipe insertion test)

| Point | Dimensionless distance from speaker $d_i$ | Dimensionless distance from junction |
|-------|------------------------------------------|-------------------------------------|
|       | Main pipe | Branch pipe ($\lambda/4$) | Branch pipe ($\lambda/2$) | Branch pipe ($3\lambda/4$) |
| 1     | 0.0930    | 0.2000                  | 0.1000                  | 0.0667                  |
| 2     | 0.1523    | 0.4000                  | 0.2000                  | 0.1333                  |
| 3     | 0.2116    | 0.6000                  | 0.3000                  | 0.2000                  |
| 4     | 0.2709    | 0.9060                  | 0.4000                  | 0.2667                  |
| 5     | 0.3302    | -                       | 0.5000                  | 0.3333                  |
| 6     | 0.3845    | -                       | 0.6000                  | 0.4000                  |
| 7     | 0.4487    | -                       | 0.7000                  | 0.4667                  |
| 8     | 0.5080    | -                       | 0.8000                  | 0.5333                  |
| 9     | 0.5673    | -                       | 0.9000                  | 0.6000                  |
| 10    | 0.6443    | -                       | -                      | 0.7333                  |
| 11    | 0.7771    | -                       | -                      | 0.8000                  |
| 12    | -         | -                       | -                      | 0.8667                  |
| 13    | -         | -                       | -                      | 0.9687                  |
4. Numerical analysis
One-dimensional numerical analysis is performed using AMESim, which is a general-purpose one-dimensional simulation software. A sub-model considering the fluid inertia, the pipe resistance, and the fluid volume is used for the main pipe and branch pipe. A sub-model considering only the pipe resistance and fluid volume is used for the bent tube. An orifice and a T-pipe is expressed as a sub-model with flow coefficient and variable friction factor respectively. Pipe resistance and fluid volume is calculated by Eq. 1 based on Darcy-Weisbach Equation and Eq. 2 based on polytropic law respectively.

\[ \Delta P = f f \frac{L}{D_{eqv}} \frac{\rho Q^2}{2A^2} \]  
\[ \frac{dP}{P} + K \frac{dV}{V} = 0 \]  

The fluid inertia in the component is described as Eq. 3 considering Newton’s equation.

\[ \Delta P = \frac{L}{A} \frac{d}{dt} m \]  

In numerical analysis, frequency response analysis feature is used to calculate the frequency of acoustic resonance. Input signal consists of sine wave, and the frequency is changed continuously from 0Hz to 1000Hz to calculate the frequency response. Fig. 5 shows the circuit for AMESim.

5. Result and discussion

5.1. Frequency of pipe resonance
The theoretical frequency of acoustic resonance in a pipe with closed end pipe is expressed by the following equation.

![Simulation circuits](image-url)
The sound velocity $C$ is represented by the following equation.

$$ f_{2n-1} = \frac{C}{4L} (2n - 1) $$

(4)

The primary resonance frequency calculated by Eq.4 are 81.5Hz and the gain shows the peak at these frequencies in Fig.6 and Fig.7. According to these results, the possibility of evaluating resonance frequencies qualitatively is clarified.

5.2. Effect of orifice

Fig.8 shows the gain when an orifice is inserted into the antinode, the node, and the midpoint between the antinode and the node of the secondary standing wave. The dashed line in Fig.8 shows the location where orifice is installed.
As shown in Fig. 8-(a), when an orifice is inserted into the antinode, the effect of mitigation does not appear regardless of the orifice opening. According to Fig. 8-(c), when the orifice is inserted into the node, the pressure fluctuations are further mitigated as the orifice opening decreases. Fig. 8-(b) shows that mitigation effects appear only when orifice opening is 25% (minimum opening).

Fig. 8-(c) indicates that the gain at the position where the orifice is inserted has the almost same value regardless of the opening of the orifice. Besides, the gain at the measurement point 4 (\(d_i = 0.432\)) tends to be minimum regardless of the orifice opening, compared with no orifice result. Therefore, it could be that the resonance frequency of acoustic resonance has changed rather than the pressure fluctuations mitigated. Fig. 9 shows the results of a sweep test when an orifice is inserted into the node to clarify the mechanism of mitigation method. The dashed line, chain line, and two-dot chain line in Fig. 9 represent the primary, secondary, and tertiary frequencies of acoustic resonance calculated by Eq. 4 respectively. Fig. 10 shows the frequency response calculated by AMESim when the orifice is inserted into the node and antinode.
According to Fig.9, the more orifice opening becomes narrower, the acoustic resonant frequencies move lower side. However, in Fig.10-(a) the resonant frequencies move higher side, and the frequency response of three cases (orifice opening is 25%, 50%, 75%) is same. The tendency appears in Fig.10-(b), which shows the analysis result of orifice insertion to antinode. Fig.8-(a) indicates that there is no effect to inserting orifice to antinode, but the analysis indicates the different result. In AMESim, orifice component deals information based on flow coefficient. As a result, the acoustic resonance frequencies are calculated by using the length between signal input point and orifice, then the frequencies become higher ones. Considering, the result of experiment and analysis result, the mechanism of mitigation is assumed to be changing the equivalent length of the pipe. The more orifice opening becomes narrower, the more equivalent pipe length longer, then resonant frequency becomes lower.

5.3. Effect of bent tube

Fig.11 shows the gain when a bent tube is inserted at the node and the antinode of the pressure standing wave. Fig.11 also shows the result of the experiment using a straight pipe model that includes the bent tube center length. The dotted line and dashed dot line in the figure indicate the node and antinode respectively. Fig.12 shows the frequency response calculated by AMESim.

Fig.11 shows that the gain does not change when a bent tube is inserted into either the node or the antinode of the pressure standing wave. Fig.12 shows that there is no change in peak of frequency response for primary and secondary mode. Therefore, it is considered that a bent tube has no pressure mitigation effects for these two modes.
5.4. Effect of branch pipe

Fig. 12 and Fig. 13 show the gain when a branch pipe with closed end and open end is installed at the antinode of the pressure standing wave. The dashed line in Fig. 12-(a) and Fig. 13-(a) indicates the location where T-pipe is installed.

As shown in Fig. 12, when a closed end branch pipe is installed at the antinode of the main pipe, pressure fluctuations are mitigated when the piping length is \( \lambda/4 \) and \( 3\lambda/4 \). According to Fig. 13, in the case of an open pipe, the mitigation effect appears when the pipe length is \( \lambda/2 \). The reason is the formation of a standing wave in the branch pipe whose phase is opposite to that of the main pipe, which mitigated the pressure fluctuations effectively.

Fig. 14 shows the frequency response calculated by AMESim. In numerical analysis, the result of no branch pipe condition is also simulated. The results of numerical analysis show the same results with experiment in the respect of secondary pressure standing wave.
6. Conclusion

The following knowledge are obtained regarding the mitigation method for pressure standing wave generated in pipes.

1) When an orifice is inserted at the antinode of the pressure standing wave, there is no mitigation effects regardless of orifice opening.

2) When an orifice is inserted at the node of the pressure standing wave, the pressure fluctuations could be mitigated more as the orifice opening becomes smaller. This effect is caused by shifting the resonance frequency to a lower frequency.

3) When a bent tube is installed at the node and antinode of the pressure standing wave, there are no mitigation effects.

4) When a branch pipe is installed at the antinode of the pressure standing wave, the pressure fluctuations are mitigated in the case where the standing wave whose phase is opposite to the main pipe is formed in the branch pipe.

7. Reference

[1] Sano M 1983 Study on pressure fluctuations in turbo pump piping system: 1st Report, experiment on eigenfrequency of liquid column in centrifugal pump piping system (in Japanese) *JSME paper B* vol 49-440 pp828-36

[2] Sano M 1984 Study on pressure fluctuations in turbo pump piping system: 2nd Report, Effect of pump position on fluctuation amplitude at resonance (in Japanese) *JSME paper B* vol 51-461 pp2316-2324

[3] Sano M 1985 Study on pressure fluctuations in turbo pump piping system: 3rd Report, Liquid column resonance in complex pipeline (in Japanese) *JSME paper B* vol 51-461 pp115-124

[4] Sano M 1986 Study on pressure fluctuations in turbo pump piping system: 4th Report, Influence of pump geometry on equivalent pipeline of centrifugal pump and fluctuation amplitude at resonance (in Japanese) *JSME paper B* vol 52-474 pp578-584

[5] Tanaka S and Okamoto M 2002 Highly accurate pipe length measurement for straight pipes with open ends using stationary waves *SICE paper* vol 38 pp821-828

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