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PRESSURE PULSATIONS IN A DRUM EXCITED BY A CENTRIFUGAL COMPRESSOR CONNECTED TO A PIPING SYSTEM

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ABSTRACT

The characteristics of the pressure pulsations in a drum connected to a piping system excited by a centrifugal compressor or a blower operated at blade-passing frequencies were investigated. In this study, the equivalent resistance of a compressor and that of a piping system were introduced and linked to the three dimensional calculation model, so that the non-linear damping proportional to velocity squared in the system is properly incorporated. The experiment was performed in order to validate the proposed simulation model.

As a result, the three dimensional pressure response in the drum as well as the pipe can be well evaluated by this model. Furthermore, the effect of the acoustic dynamic absorber on the pressure pulsations in the pipe and drum is evaluated. When the resonant frequency of the pipe coincides with that of the drum, two peaks appear in the frequency response curve around the resonant frequency of the pipe, because the drum acts as an acoustic dynamic absorber. It is shown that the maximum pressure amplitude in the drum is obtained when the resonant frequency of the pipe is slightly shifted from the resonant frequency of the drum under the small damping conditions. The effect of the damping in the drum and the mode shape of the drum on the maximum pressure amplitude in the drum is discussed in detail.

NOMENCLATURE

A	:	cross-sectional area of a pipe [m^2]
c	:	speed of sound [m/s]
D	:	pipe diameter [m]
F	:	excitation force [N]
f	:	frequency [Hz]
f_n	:	resonant frequency [Hz]
$f_{n,pip}^*$	:	non-dimensional resonant frequency in a pipe [-] (ratio to the resonant frequency in a drum)
f_{peak}	:	peak frequency [Hz]
ℓ	:	length of a piping element [m]
$\overline{m}$	:	average mass flow rate [kg/s]
m	:	fluctuation component of a mass flow rate [kg/s]
$\hat{m}$	:	equivalent mass flow rate [kg/s]
p	:	fluctuation component of pressure [Pa]
p_a^*	:	non-dimensional pressure pulsation amplitude [-]
R	:	equivalent resistance [kg/m^{2}s]
$\overline{u}$	:	average fluid velocity [m/s]
u	:	fluctuation component of a fluid velocity [m/s]
Δx	:	length of an excitation part in a compressor [m]
γ	:	linear damping coefficient in a drum [Ns/kgm]
λ	:	friction factor for a pipe flow [-]
ρ	:	fluid density [kg/m^3]
ω	:	angular frequency [rad/s]
ξ	:	damping ratio [-]
ζ	:	damping coefficient of concentrated resistance [-]

SUFFIX

c	:	excitation part in a compressor
d	:	discharge side of an excitation part in a compressor
$drum$	:	in a drum
in	:	pipe inlet
out	:	pipe outlet
p	:	piping element
pip	:	piping system
r	:	concentrated resistance
s	:	suction side of an excitation part in a compressor

t : total system

INTRODUCTION

Severe noise and vibrations in piping systems are caused by pressure pulsations from a compressor or a blower. In particular, the reduction of pressure pulsations at blade passing frequency is required to be studied at the design stage of a centrifugal compressor piping system. Although, research [1-3] was reported regarding blade passing sound/pulsation generated in a centrifugal fan/blower, the characteristics of pressure pulsations under resonant conditions and the damping characteristics of the systems were not clear. Therefore, authors have proposed one dimensional model of a compressor piping system [4-6], so that the non-linear damping proportional to velocity squared in the system is properly incorporated. As a result, the pressure amplitude in piping systems under the wide operating range can be well evaluated by taking into account the response characteristics of a propagating passage.

On the other hand, when the large diameter drum is connected to the piping systems, three dimensional resonance mode of the drum can be excited [7,8] resulting in fatigue failure of the drums[7]. Therefore, three dimensional pressure distribution should be evaluated for such an application. In this study, the one dimensional model for pressure pulsations in the piping [4-6] was developed to the three dimensional model. As a result, three dimensional pressure response in a drum as well as that in a pipe can be well evaluated by using the proposed model. Furthermore, the effect of the acoustic dynamic absorber on the pressure pulsations in the pipe and drum was evaluated. The characteristics of dynamic absorber effect in a simple acoustic system are studied [9], and it can be applied as a noise control device for the duct [10]. Recently, the characteristics of the dynamic absorber effect in an acoustic structural coupled system were investigated [11] as well using simple mathematical model. In this study, the influence of the damping of the drum and acoustic mode shape in the drum on the pressure response is discussed in order to grasp the dominant parameter to be considered which is necessary when we apply such an idea to the practical engineering problem.

SIMULATION METHOD OF PULSATIONS

SIMULATION MODEL

A three dimensional FEM model for the pressure pulsations in compressor piping system is shown in Fig.1. As shown in Fig.2, the piping model was divided into two models at the plane close to the excitation source point, so that the non-linear damping proportional to velocity squared in the system can be properly incorporated without adding any specific function on the acoustic simulation code. The method proposed in this paper makes it possible to make use of the general simulation code such as SYSNOISE usually unable to treat the non-linear damping.

EXCITATION SOURCE MODEL IN A COMPRESSOR

The pressure excitation force was loaded to the fluid by Eq.(1) based on the assumption that the pressure pulsation generated when blades are passing by a volute tongue can be treated as a one dimensional plane wave [2-4]. The equivalent damping coefficient ζ_t presenting the total damping in the compressor and the piping except for the drum was introduced by applying Eq.(1) – Eq.(4) to the excitation part in the compressor, in order to use the general acoustic simulation code for the calculation of the pulsation in this system.

$$\frac{\partial m_d}{\partial t} + \bar{u}\frac{\partial m_d}{\partial x} + A_c\frac{p_d - p_s}{\Delta x} + \frac{R_t}{\Delta x}u_d = \frac{F}{\Delta x} \tag{1}$$

$$m_d = m_s - \frac{\partial p_d}{\partial t}\cdot\frac{A_c \Delta x}{c^2} \tag{2}$$

$$R_t = \frac{\zeta_t}{2}\hat{m} \tag{3}$$

$$\begin{aligned}\zeta_t = \zeta_c &+ \zeta_{rin}\frac{(\rho_c A_c)^2}{(\rho_{rin}A_{rin})^2}\frac{\overline{m}_{rin}m_{rain}{}^2}{\overline{m}_c m_{ca}{}^2} \\ &+ \zeta_{rout}\frac{(\rho_c A_c)^2}{(\rho_{rout}A_{rout})^2}\frac{\overline{m}_{rout}m_{raout}{}^2}{\overline{m}_c m_{ca}{}^2} \\ &+ \sum_{i=1}^{N}\lambda\frac{\Delta\ell_i}{D_i}\frac{(\rho_c A_c)^2}{(\rho_p A_p)^2}\frac{\overline{m}_{pi}m_{pai}{}^2}{\overline{m}_c m_{ca}{}^2}\end{aligned} \tag{4}$$

DAMPING IN A DRUM

A volume of the drum connected to the piping is so large that the damping caused by the pressure loss based on the mean velocity in the drum is negligible. Therefore, the damping of the drum was evaluated by applying the damping coefficient γ in Eq.(5) at each element in the drum, based on the assumption that the acoustic attenuation in the static fluid dominates the damping characteristics in the drum.

$$\rho\frac{\partial \mathbf{u}}{\partial t} + \nabla\mathbf{p} + \gamma\rho\mathbf{u} = 0 \tag{5}$$

CALCULATION FLOW

The calculation flow of this study is shown in Fig.3. First of all, pressure boundary condition was set at the excitation part, so that the velocity distribution in the pipe was calculated without damping in the pipe. Next, mass flow rate at the excitation part was changed so that Eq.(2) was satisfied. Then, equivalent mass flow rate and equivalent resistance were calculated by Eq.(4) by using the velocity distribution. This process was repeated by changing the mass flow rate at the excitation part until Eq.(1) was satisfied. Finally, the pressure distribution in the pipe was determined. In this proposed method of calculation, velocity distribution will be calculated only once for the initial condition, because this step is linear calculation and is out of the iterative process. Therefore, three dimensional acoustic FEM calculation by SYSNOISE is to be performed only once, resulting in low calculation cost. In this study, the following two calculation methods were applied;

(1) Separated model

The model of the drum is separated from the piping model. The boundary condition of the pipe end where the pipe is connected to the drum is treated as an open boundary, and then the velocity fluctuation at the boundary is obtained as the input data to the drum.

(2) Integrated model

The drum is included in the model as is shown in Fig.1. The pressure pulsation in the drum and the pipe is obtained simultaneously.

When the separated model is applied, the piping system can be modeled by one dimensional element. Therefore, it is possible to reduce the calculation cost by applying this model. However, the effect of the pressure pulsation in the drum on the pressure response in the pipe is not incorporated. In this study, the calculation results obtained by the separated model were compared with those obtained by the integrated model, so that the influence of the pressure pulsation in the drum on the pressure pulsation in the whole piping systems could be discussed.

EXPERIMENTAL CONDITION

Pressure pulsation amplitude in piping systems was measured experimentally for the validation of the simulation model. Figure 4 shows the experimental setup for the compressor piping system. The experimental setup was the same as that used in the previous study [4,5]. The drum of height 0.6m and diameter 1.0m was attached to the discharge pipe end. The acoustic damping in the drum was obtained by loading the impact noise from the nozzle of the drum after removing the pipe from the drum. The excitation force and the resistance coefficient in the compressor were obtained experimentally, by calculating the transfer matrix in the compressor under the acoustic loading by a loud speaker from the pipe end [4,5].

RESULTS AND DISCUSSION

PRESSURE PULSATION IN A PIPE

Figure 5 and Figure 6 show the frequency response of pressure pulsations in the discharge pipe obtained by experiment and calculation. Pressure amplitude was non-dimensionalized by dynamic pressure based on the tip velocity of the impeller. It is shown that the pressure pulsation amplitude in the pipe can be well evaluated by introducing the equivalent damping in the compressor regardless of the type of models i.e. both the separated model and the integrated model can be used. However, when the acoustic diametral mode was excited in the drum as shown in Fig.7, the pressure response in the pipe was changed. Figure 8 shows the frequency response of pressure pulsations in the discharge pipe around the resonant frequency of the drum. The frequency of the third order acoustic mode of the pipe coincides with that of the diametral mode of the drum at 206Hz. Although, the pressure amplitude in the pipe at this frequency was reduced in the experimental result, the calculation result obtained by the separated model shows different aspect. On the other hand, the pressure response obtained by the integrated model shows the reduction of the pressure amplitude when the resonant frequency of the pipe coincides with that of the drum (Fig.9). As a result, the calculated pressure response by the integrated model shows two peaks and this aspect agrees with that of experiment. This reduction of the pressure amplitude in the pipe is caused by the acoustic dynamic absorber effect which can be seen when the resonance occurs in the subsystem attached to the main system. The pressure response in the pipe is influenced by the change of the ratio of the pressure amplitude to the velocity amplitude (impedance) at the connected part with the drum. Figure 10 shows impedance at the drum inlet nozzle. From Fig.10, it is shown that the impedance around the resonant frequency in the drum was drastically increased, thus the acoustic mode in the pipe could not be excited, because an open boundary condition at the drum inlet could not be applied. This phenomenon causes the reduction of pressure amplitude in the pipe around the resonant frequencies.

The frequency response of pressure pulsations in the drum obtained by the separated model and the integrated model are shown in Fig.11 and Fig.12 respectively. Although the calculation result obtained by the separated model was ten times larger than that obtained by experiment, the calculated result obtained by the integrated model agrees well with the experimental results.

INFLUENCE OF ACOUSTIC RESONANCE IN A PIPE ON PRESSURE RESPONSE IN A DRUM

In order to investigate the influence of the acoustic resonance in the pipe on the pressure response in the drum, the frequency response of pressure pulsations are shown in Fig.13 and Fig.14 for the different frequency ratio $f_{n,pip}^*$ which is defined as the ratio of the peak frequency of the drum to the third order resonant frequency of the pipe. The resonant frequency of the pipe was controlled by changing the sound speed of the gas. Figure 13 shows that the maximum pressure amplitude in the pipe was almost constant regardless of the resonant frequency of the pipe. On the other hand, Figure 14 shows that the maximum pressure amplitude in the drum was changed depending on the frequency ratio $f_{n,pip}^*$.

Next, in order to clarify the cause of change in the maximum pressure amplitude in the drum, the relation between the maximum pressure amplitude in the drum and the resonant frequency of the pipe are shown in Fig.15. In Fig.15, γ is the liner damping coefficient in the drum. The value of 2.216[Ns/kgm] was obtained for γ by experiment. From Fig.15, it is shown that the maximum pressure amplitude in the drum for γ =2.216[Ns/kgm] was observed when the resonant frequency in the drum was slightly shifted from the resonant frequency of the pipe. The cause of this phenomenon was discussed by investigating the relation between the peak frequency of the drum and the resonant frequency of the pipe as shown in Fig.16. It is shown from Fig.16 that the peak frequency of the drum was shifted from the resonant frequency of the drum, when the frequency ratio $f_{n,pip}^*$ was close to the unity. Therefore, the amplification ratio which is defined as the ratio of the pressure response to the velocity excitation was decreased when the frequency ratio $f_{n,pip}^*$ was close to the unity as shown in Fig.17(γ =2.216). On the other hand, the pressure amplitude in the discharge pipe which represents the excitation force to the drum was increased, when the frequency ratio $f_{n,pip}^*$ was close to the unity as shown in Fig.17. Thus, since the pressure response in the drum is obtained by multiplying the excitation force by the amplification ratio, the maximum pressure amplitude in the drum shows local minimal value when the frequency ratio $f_{n,pip}^*$ =1.0 as shown in Fig.15.

INFLUENCE OF ACOUSTIC DAMPING IN A DRUM ON PRESSURE RESPONSE IN A DRUM

The acoustic damping ratio to the critical damping coefficient in the drum was obtained experimentally by Eq.(6), by using the results of the impact noise test. The liner damping coefficient was calculated so that the pressure pulsation amplitude obtained by the mode superposition technique was in accordance with that obtained by direct analysis in the frequency domain with applying the uniform linier damping coefficient for the fluid in the drum. The damping ratio obtained by experiment was 0.07% and the equivalent linear damping coefficient was 2.216[Ns/kgm].

$$p_a(t) = p_a(0)e^{-\xi\omega t} \tag{6}$$

. Acoustic damping in the drum is affected by the properties of the fluid. Therefore, the influence of the acoustic damping in the drum on the pressure response in the drum was investigated by changing the damping ratio in the drum from 0.007% (γ = 0.2216[Ns/kgm]) to 0.5% (γ = 15.83[Ns/kgm]). The relation between the maximum pressure amplitude in the drum and the resonant frequency in the pipe for different acoustic damping are shown in Fig.15. It is shown that the maximum pressure response in the drum decreased remarkably as the frequency ratio $f_{n,pip}^*$ was close to the unity if the acoustic damping in the drum was small (γ =0.2216). In contrast, the maximum pressure response in the drum increased as the frequency ratio $f_{n,pip}^*$ was close to the unity, when the acoustic damping was large (γ =15.83). This trend that the acoustic damping affects on the maximum pressure response in the drum is related to the amplification ratio as shown in Fig.17. The amplification ratio decreased as frequency ratio $f_{n,pip}^*$ was close to the unity, when the acoustic damping was small (γ =0.0216) in Fig.17. On the other hand, when the acoustic damping was large (γ =15.83), the amplification ratio was almost constant independent of the frequency ratio $f_{n,pip}^*$, and then, the pressure response in the drum became maximum at the frequency ratio $f_{n,pip}^*$ =1.0, according to the frequency response of the pressure pulsation in the pipe.

INFLUENCE OF ACOUSTIC MODE OF A DRUM ON PRESSURE RESPONSE IN A DRUM

In this study, the effect of the diametral acoustic mode excited in the drum on the pressure response was investigated. However, the different type of acoustic mode can be excited if the dimension of the drum, sound speed of the fluid and/or concerned frequency range is different. Therefore, the pressure response in the piping systems was investigated by changing the type of the acoustic mode of the drum as shown in Fig.19. The vertical acoustic mode was excited in the drum at 206Hz, by changing the speed of sound in the drum.

The relation between the maximum pressure amplitude in the drum and the resonant frequency in the pipe at the frequency around the vertical resonant mode of the drum are shown in Fig.20. It is shown from Fig.20 that the pressure response in the drum became maximum when the frequency ratio $f_{n,pip}^*$ =1.0, independent of the acoustic damping. This is attributed by the fact that the drum inlet nozzle is located on the node of the pressure pulsation of the vertical mode of the drum. That is, the acoustic dynamic absorber effect is not observed because the pressure pulsation at the connected part of the pipe and the drum remains small, even though the resonance in the drum occurs at this vertical mode. This phenomenon can be explained by investigating the impedance at the drum inlet nozzle as shown in Fig.10. It is shown from Fig.10 that the impedance at the nozzle was not changed in the case that the vertical mode was excited in the drum, thus the acoustic resonance in the drum does not affect on the pressure response in the pipe. Therefore, in the case that the connected part of the pipe to the drum is located on the node of the pressure pulsations in the drum, the maximum pressure amplitude can be obtained when the resonant frequency of the pipe coincides with that of the drum.

CONCLUSION

The characteristics of pressure pulsations in a drum connected to a piping system by a centrifugal compressor at blade passing frequency were investigated by introducing the equivalent resistance of a compressor and that of a piping system. As a result, the following conclusions are drawn:

1. The three dimensional pressure response in the drum as well as that in the pipe can be well evaluated by applying the proposed model.
2. The dynamic absorber effect which is observed when the resonant frequency of the pipe coincides with that of the drum can be evaluated by applying the proposed model.
3. The maximum pressure response in the drum is observed when the resonant frequency of the pipe is slightly shifted from the resonant frequency of the drum, if the pressure amplitude at the connected part of the pipe to the drum is large.
4. In contrast, the maximum pressure response in the drum is observed when the resonant frequency of the pipe coincides with that of the drum, if the pressure amplitude at the connected part of the pipe to the drum is small.

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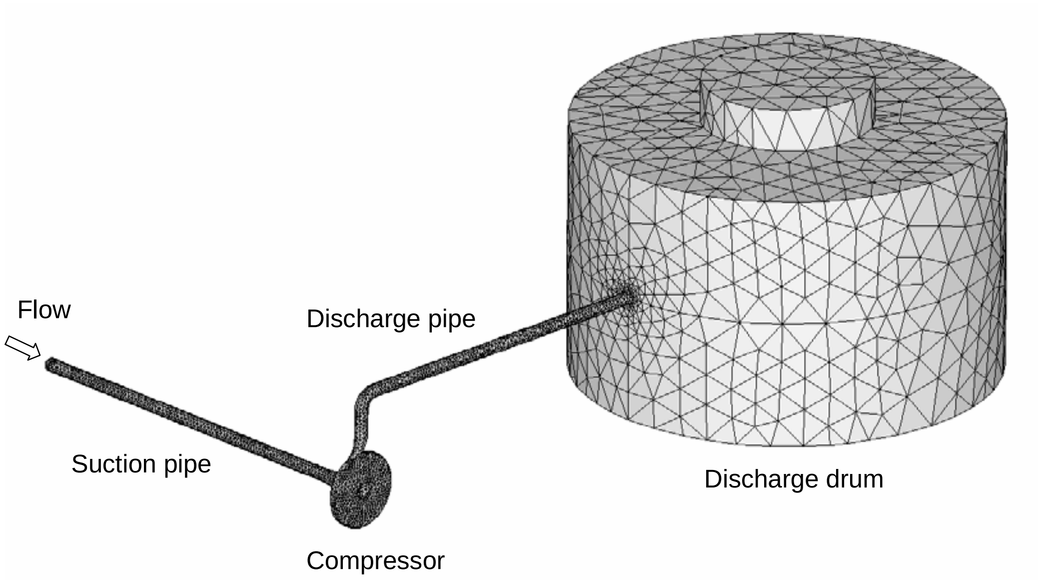

Figure 1 Three dimensional simulation model for finite element method.

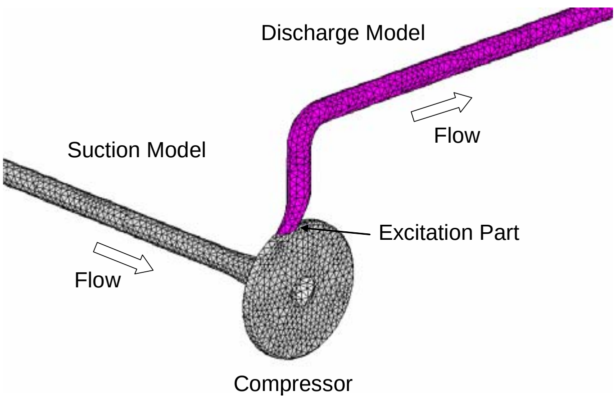

Figure 2 Connection part of the suction and discharge model

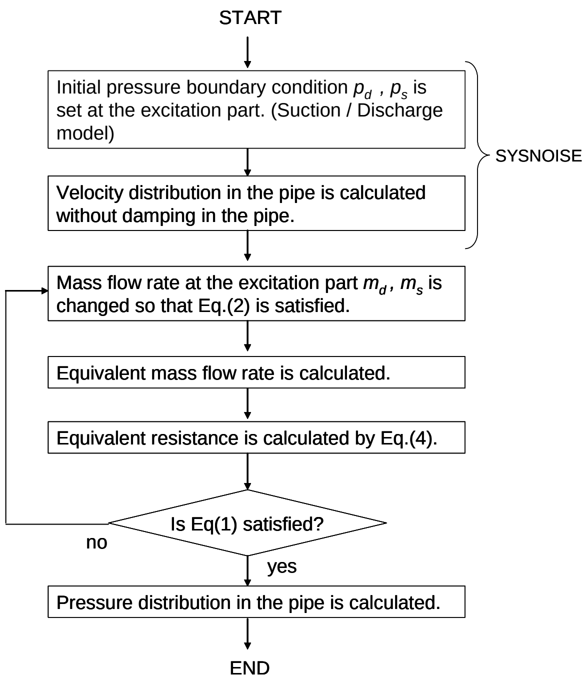

Figure 3 Calculation flow

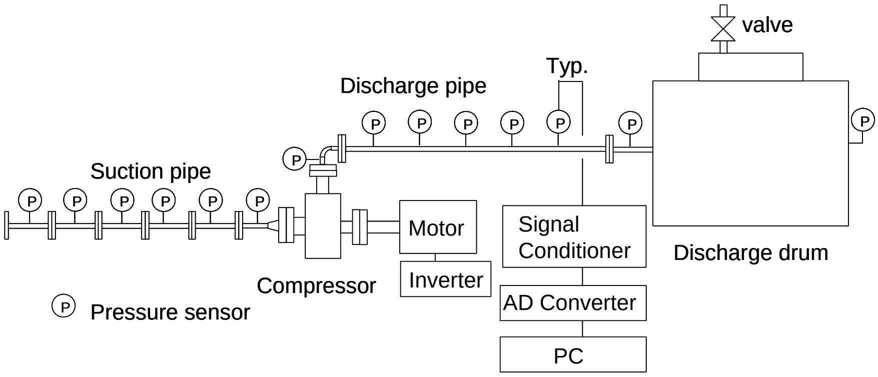

Figure4 Experimental setup

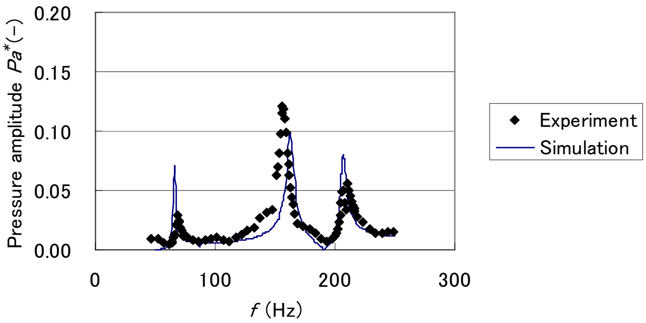

Figure 5 The frequency response of pressure pulsations in the discharge pipe. Separated model is applied.

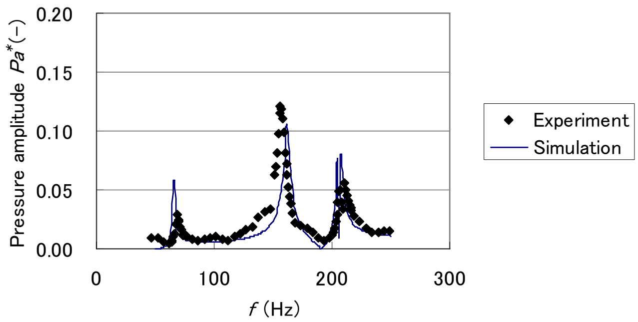

Figure 6 The frequency response of pressure pulsations in the discharge pipe. Integrated model is applied.

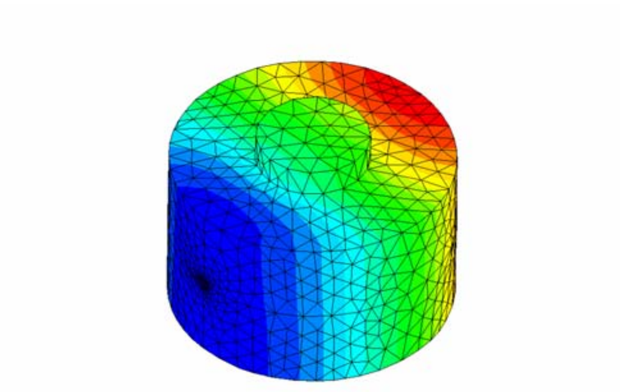

Figure 7 Pressure distribution of the diametral resonant mode of the drum at 206Hz. Red and blue color show high pressure amplitude.

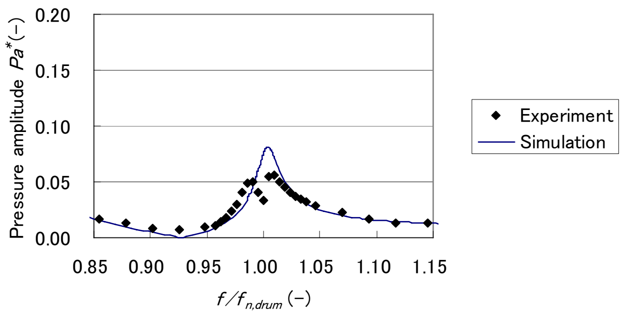

Figure 8 The frequency response of pressure pulsations in the discharge pipe around the resonant frequency of the drum. Separated model is applied.

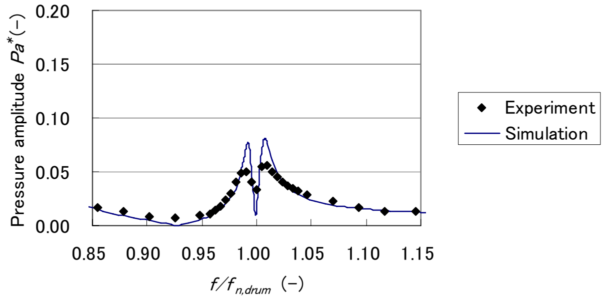

Figure.9 The frequency response of pressure pulsations in the discharge pipe around the resonant frequency of the drum. Integrated model is applied.

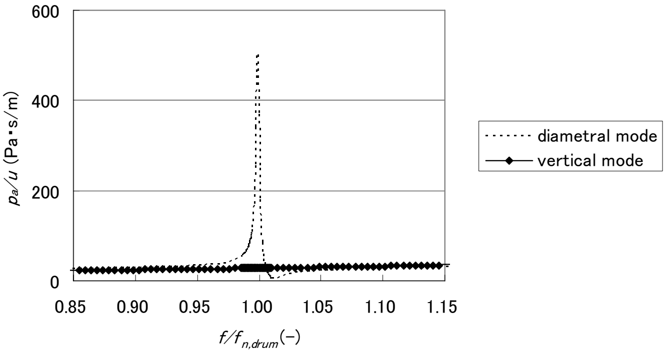

Figure 10 Impedance at the drum inlet nozzle. Integrated model is applied.

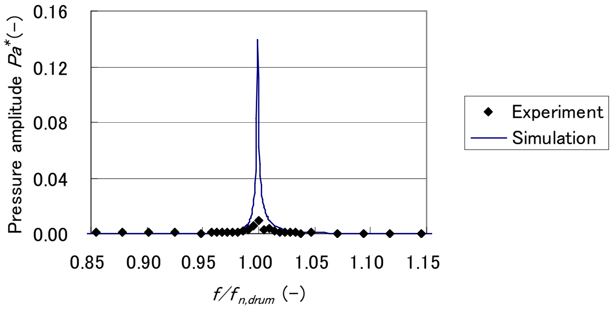

Figure 11 The frequency response of pressure pulsations in the drum around the resonant frequency of the drum. Separated model is applied.

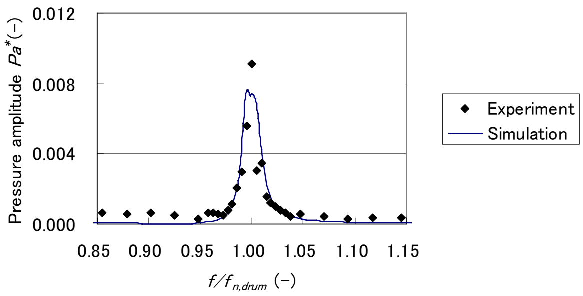

Figure 12 The frequency response of pressure pulsations in the drum around the resonant frequency of the drum. Integrated model is applied.

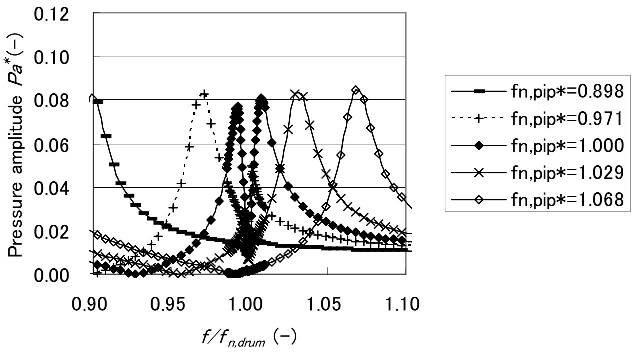

Figure 13 The frequency response of pressure pulsations in the discharge pipe. The resonant frequency of the pipe is changed.

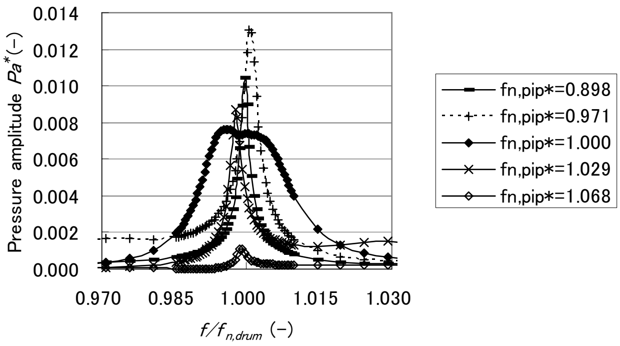

Figure 14 The frequency response of pressure pulsations in the drum. The resonant frequency of the pipe is changed.

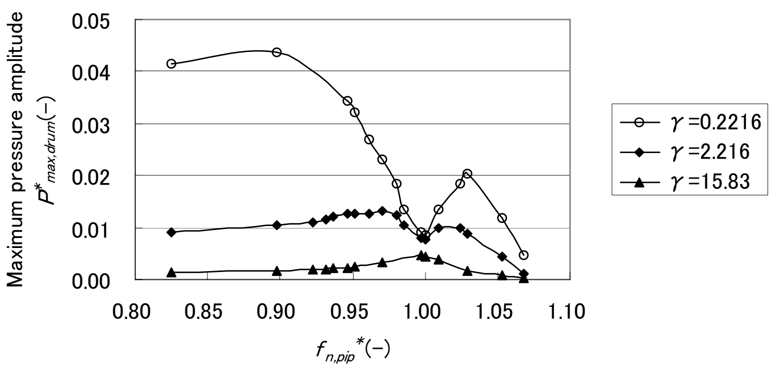

Figure 15 The relation between the maximum pressure amplitude in the drum and the resonant frequency of the pipe.

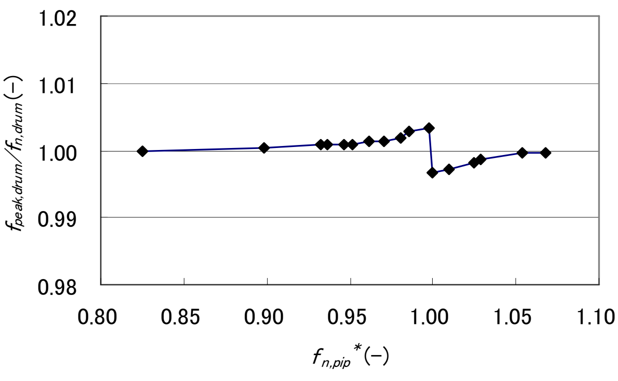

Figure 16 The relation between the peak frequency in the drum and the resonant frequency of the pipe. γ =2.216[Ns/kgm].

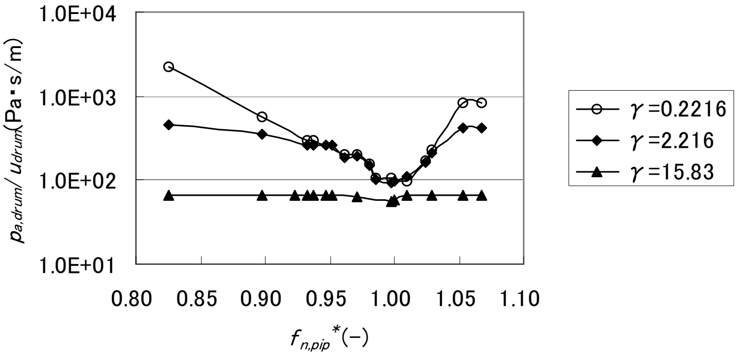

Figure 17 The relation between the ratio of the pressure response to the velocity excitation and the resonant frequency of the pipe.

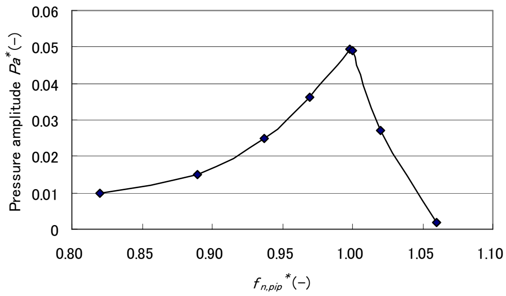

Figure 18 The relation between the pressure amplitude in the discharge pipe at the peak frequency in the drum and the resonant frequency of the pipe. γ =2.216[Ns/kgm].

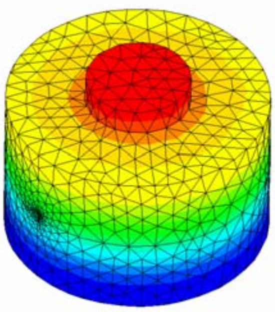

Figure 19 Pressure distribution of the vertical resonant mode of the drum at 206Hz. Red and blue color show high pressure amplitude.

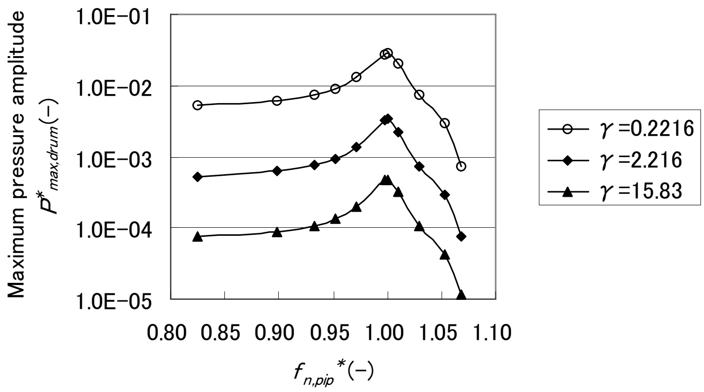

Figure 20 The relation between the maximum pressure amplitude in the drum and the resonant frequency of the pipe. Vertical resonant mode is excited in the drum.