*Journal of Mechanical Engineering Research and Developments*
*ISSN: 1024-1752*
*CODEN: JERDFO*
*Vol. 43, No. 1, pp. 197-203*
*Published Year 2020*

# Study On Resonance Control By Locally Expanding The Outer Tube In A Coaxial Thermoacoustic System

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ABSTRACT: Toward the enhancement of the cooling ability of a coaxial-type thermoacoustic system, the control of the sound field in the tube, in particular that of the resonance mode, is investigated. The coaxial type thermoacoustic system consisting of two tubes with different radii is capable of forming a sound field of traveling wave with a high energy conversion efficiency similar to a loop-tube type thermoacoustic system. In addition, because of the linear shape, this system is advantageous for downsizing. However, the setting position of the prime mover (PM) is different from that for the loop-tube type. When the setting position is moved to closer to the tube end, the sound pressure of the fundamental mode is enhanced and the cooling ability increases in the range not too close to the tube end. However, when the position is too close to the end, the sound pressure of the second mode increases instead of the fundamental mode and the cooling ability is degraded. Therefore, assuming the shift to the second mode is the cause to lower the cooling ability, in order to make the input heat concentrate to the fundamental mode the radius of the tube is locally enlarged at the position of the antinode in the second mode. As the result, the decrease of the second mode sound pressure and the increase of the fundamental mode sound pressure are confirmed. The lowered temperature at the high temperature end and the promotion of the energy conversion from sound to heat are also confirmed.

## INTRODUCTION

Recent years, the environmental issue such as global warming and air pollution has been serious. A thermoacioustic system is then getting all the attention. A thermoacoustic system is the system to apply the thermoacoustic phenomenon that is a mutual conversion between heat and sound. [1-6] By allocating a device called stack to have numerous narrow tubes and producing steep temperature gradient between both ends, a sound wave oscillates. Since this system is an external-combustion engine, exhaust heat from factories and solar light can be used for the driving source. [7-9] Furthermore, because of no mechanical moving parts, it is maintenance-free.

A thermoacoustic system is categorized to standing-wave type and traveling-wave type. Since the energy conversion in the traveling-wave type is performed in the traveling-wave phase, the energy conversion efficiency is high. As the system of the traveling-wave phase, a loop-tube type thermoacoustic system is currently mostly studied.[10] However, since the loop-tube type is so large that the total length is several meters and has an empty space at the center, it is disadvantageous for downsizing. To solve this subject, the present research group proposes a coaxial-type thermoacoustic system. The coaxial-type consists of two tubes with different diameter arranged coaxially. Due to its acoustical structure, this type is capable of forming a traveling-wave sound field similar to the loop tube. Owing to the linear shape, it is assumed to be advantageous for downsizing comparing with the loop-tube type. [11-15]

In the previous study, the influence of the setting position for the prime mover (PM) of the coaxial type on the cooling ability was experimentally investigated.[15] Although the cooling effect was expected to be enhanced as the positional relation between PM and the heat pump (HP) approaches the positional relation in the loop-tube type of the same traveling-wave type, the enhancement was never confirmed. As the contributing factor, the shift of the resonance mode to the secondary mode was found. From this result, it was surmised that the cooling ability turned down because the thermal input was dispersed to the fundamental mode and the second mode.

Therefore the present paper attempts to provide a sound field condition to suppress the energy dispersion to the secondary mode. Namely, the outer tube is locally expanded at the position of a node of the fundamental mode that is an antinode of the secondary mode. The suppression of the shift of the resonance mode and the increase of the fundamental-mode sound pressure that is roughly proportional to the cooling ability are experimentally investigated.

## EXPERIMENTAL METHOD

The experimental system is illustrated in Figure 1. A stainless steel tube with a 42 mm inner diameter and a 2100 mm total length is used for the outer tube closed at both ends. A coordinate whose origin ($x$ = 0) is set at the left end of the outer tube is defined. Another stainless tube with a 27 mm outer diameter, a 2000 mm total length and a 1 mm thickness is used for the inner tube open at both ends. The inner tube is set in the range 50 mm $\leq x \leq$ 2050 mm coaxially with the outer tube. Air of atmospheric pressure is encapsulated in the tube. A stack made from a honeycomb ceramics with a 50 mm length and a 0.65 mm flow-path radius is used for PM. PM is set so that the position of the normal temperature end where 20°C water is circulated is $x_{PM}$ = 1700 mm. On the other hand, an electric heater is installed at the high temperature end of PM and a 330 W electric power is supplied. In order to suppress the shift of the resonance mode, attention is paid to the distribution of the sound pressure formed in an annular flow path (the flow path in the outer tube). The sound pressure distribution at the secondary mode formed in the annular flow path is illustrated in Figure 2. By expanding the radius of the outer tube in the range 990 mm $\leq x \leq$ 1040 mm for the acoustical boundary condition for suppressing the secondary mode, the condition that the node easily appears there is furnished. This position is the location for the antinode of the secondary mode. Since the condition that the node easily appears at the position of the antinode is provided, the reduction of the secondary mode is expected. The inner diameter $x_{ID}$ of the expanded outer tube is changed from 42 mm (before the expansion) to 60, 80 and 100 mm. At each size, the sound pressure in the annular flow path is measured with a pressure sensor (product of PCB Co.) after making sure the temperatures at both ends of PM have been reached at the steady state. [16]

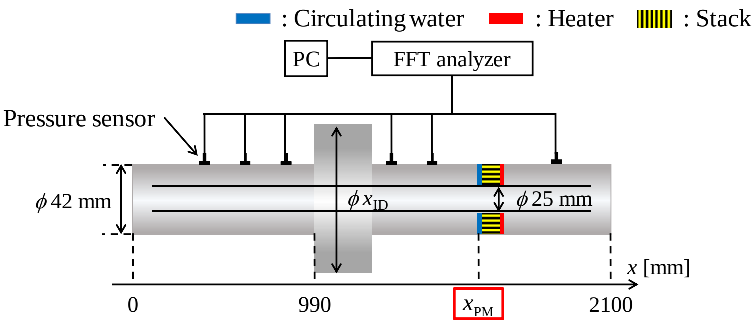


**Figure 1.** Experimental setup.

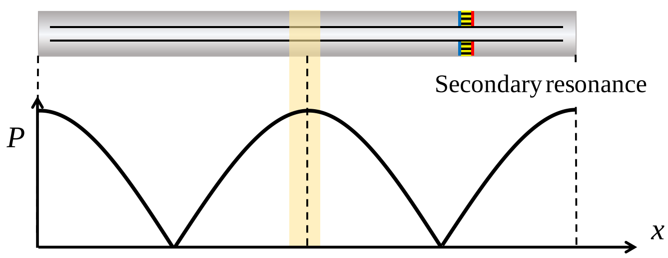


**Figure 2.** Schematic of the sound pressure distribution in the annular area.

## EXPERIMENTAL RESULTS AND DISCUSSION

In each value of $x_{ID}$, the sound wave with a wavelength of total flow path length is confirmed to be excited. Examples of the spectrum observed at $x$ = 120 mm with $x_{ID}$ = 42 mm (before the expansion) is shown in Figure 3 and with $x_{ID}$ = 100 mm is shown in Figure 4. Increasing the level difference between the fundamental mode and secondary mode is confirmed with the expanded tube. To confirm the suppression effect of the second mode by increasing $x_{ID}$, the level difference between the fundamental mode and the secondary mode observed at $x$ = 120 mm is shown in Figure 5. Since the sound level difference between the fundamental mode and the secondary mode increases with $x_{ID}$, it is seen that the expansion of the outer radius of the annular flow path is effective for the suppression of the second mode. The reason is discussed from the view point of the sound pressure distribution in the annular flow path. When the radius of the outer tube is expanded, the boundary condition to have the node of each mode easily appear at the expanded position is provided. Since the vicinity of the center of the tube is expanded in this experiment, it is the boundary condition that the center of the tube easily becomes the node. The schematic of the sound pressure distribution formed in the annular flow path is shown in Figure 6. Figure 6 (a) shows the pressure distribution of the fundamental mode and Figure 6 (b) shows the secondary mode whose suppression is attempted here. When the boundary condition to make the center of the tube ready to set the node is provided, the condition that the secondary mode as shown in Figure 6 (c) is easily excited is obtained. However, since the gradient of this sound pressure distribution does not agree with the gradient of the pressure distribution of the fundamental mode at the PM setting position, the oscillation of this second mode is hardly excited. Then the resonance shift to the secondary mode turns to be suppressed. An example of the sound pressure distribution in the fundamental mode formed in the annular flow path of $x_{ID}$ = 60 mm is shown in Figure 7. From this figure, the fundamental mode can be confirmed to have the sound pressure distribution with a node at the center of the tube. Since the boundary condition to facilitate the appearance of the sound pressure node is provided at the position of the node of the sound pressure of the fundamental mode, the fundamental mode is scarcely affected by the boundary condition, and then it is supposed that the difference in the sound pressure level between the fundamental frequency component and the second harmonic frequency component is enhanced.

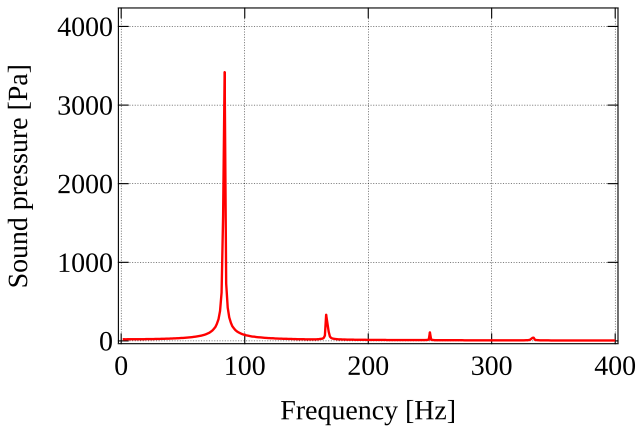


**Figure 3.** Frequency spectrum with $x_{ID}$ =42 mm

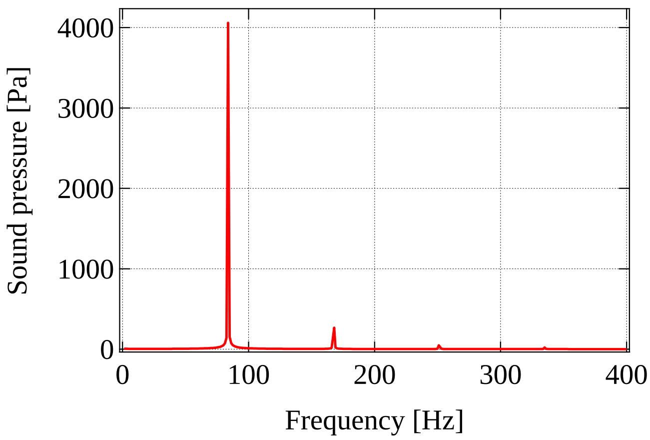


**Figure 4.** Frequency spectrum with $x_{ID}$ =100 mm

The sound pressure at $x$ = 50 mm (the assumed setting position of HP) for various $x_{ID}$ is shown in Figure 6. Comparing with the system with a uniform outer-tube radius of $x_{ID}$ = 42 mm, the enhanced sound pressure is confirmed in the system of $x_{ID}$ = 60 and 80 mm. However, in the system of $x_{ID}$ = 100 mm, the sound pressure turns down comparing with the system of a uniform outer-tube radius. To investigate this reason, the attention is paid to the thickness of the viscous boundary layer of the high temperature end of PM. The thickness of the viscous boundary layer $\delta_v$ is defined as Eq. (1).

$$\delta_\nu = \sqrt{\frac{2\nu}{\omega}} \qquad (1)$$

Working fluid in the viscous boundary layer does not contribute to the conversion between heat and sound. Using the temperature of the high temperature end of PM for various $x_{ID}$, the thickness of the viscous boundary layer $\delta_v$ is shown in Figure 7. The thickness of the viscous boundary layer $\delta_v$ is confirmed to be reduced in the cases of $x_{ID}$ = 60 and 80 mm where the sound pressure is enhanced. This thickness of the viscous boundary layer $\delta_v$ reducing is assumed to take place because the energy conversion from heat to sound is more activated. On the other hand, in the case of $x_{ID}$ = 100 mm, the thickness of the viscous boundary layer $\delta_v$ at the high temperature end is elevated compared to the system of uniform radius. Although the thermal input is concentrated to the fundamental mode because the secondary mode decreases, since the boundary condition designed in this experiment turns to be the condition that the working fluid is hard to exchange energy from heat to sound, the conversion efficiency from heat to sound lowers. Therefore it is seen that the optimum expanded radius exists between $x_{ID}$ = 80 mm and $x_{ID}$ = 100 mm.

## SUMMARY

In the present paper, to suppress the shift of the fundamental resonance mode to the second mode that caused the degradation of the cooling effect, the control of the resonance mode was investigated by the technique to provide additional boundary condition. As the result, the suppression of the second mode generation was successfully performed and the promotion of the energy conversion from heat to sound resulted in the enhanced sound pressure of the fundamental mode. However, it was suggested that extreme boundary conditions could suppress not only the second mode but also the fundamental mode.

## ACKNOWLEDGMENTS

This work was partly supported by JSPS grants-in-aid for young scientists (A) and (B), JSPS grant-in-aid for challenging exploratory research, grant-in-aid for scientific research (C), program for fostering regional innovation and JST super cluster program.

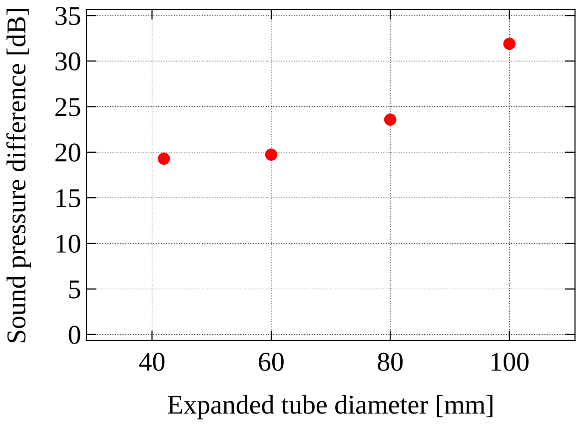


**Figure 5.** Difference of sound pressure level between fundamental and second mode components.

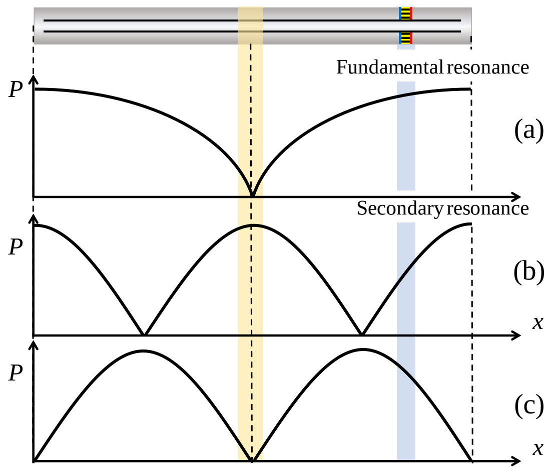


**Figure 6.** Schematic of fundamental and second mode sound pressure distributions in the annular area.

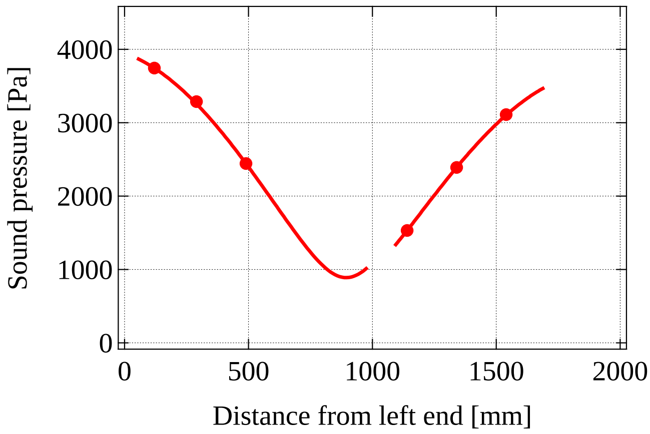


**Figure 7.** Distribution of sound pressure.

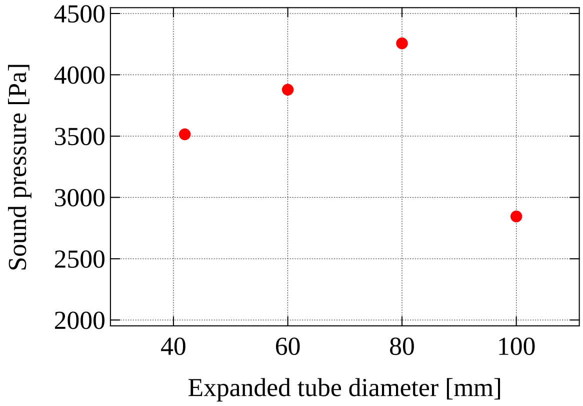


**Figure 8.** Relationship between expanded tube diameter and sound pressure.

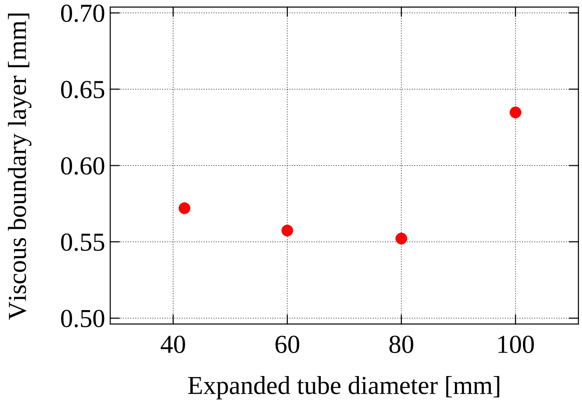


**Figure 9.** Relationship between expanded tube diameter and viscous boundary layer.

## NOMENCLATURE

$\delta_v$ thickness of the viscous boundary layer, mm

$v$ dynamic coefficient of viscosity, $m^2/s$

$\omega$ angular frequency of vibration, rad/s

*Subscripts*

$x$ distance from left end

$x_{PM}$ position of prime mover

$x_{ID}$ expanded tube diameter

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