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Finite element analysis of the standing wave field in a tube with lossy wall for thermoacoustic system
熱音響システムのための管壁損失を考慮した定在波音場の有限要素解析

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1. Introduction

In order to put the thermoacoustic system to the practical use, it is necessary to improve the efficiency of the energy conversion between sound and heat in thermoacoustic phenomenon [1]. For the efficiency improvement, it is important to analyze the sound field in a thermoacoustic tube. In the prior investigation, the sound field is numerically analyzed by the FDTD method [2]. It is, however, difficult to analyze the steady-state standing wave field in the thermoacoustic tube because the FDTD method is a time-domain method.

In this paper, the steady-state sound field in a thermoacoustic tube is numerically analyzed in frequency domain using the finite element method (FEM) [3]. Although the thermoacoustic field is nonlinear and isothermal, a linear and adiabectic model is proposed in this paper, in which the sound energy flow is only considered. In the model, the energy conversion from heat to sound is modeled as sound source and one from sound to heat as loss related with tube wall. Some numerical demonstrations are made for two-dimensional standing wave field in a linear tube with lossy wall.

2. Finite element model

Figure 1 shows the two-dimensional finite element model for the steady-state field in the thermoacoustic tube. The sound field Ω is assumed to be linear without loss. The boundary conditions are given as follows

$$\frac{\partial p}{\partial n} = \hat{q} \quad (\text{on} \ \ \Gamma_d) \qquad (1)$$

$$\frac{p}{u} = Z_w \quad (\text{on} \ \ \Gamma_w) \qquad (2)$$

where p is sound pressure, u is particle velocity, $\partial / \partial n$ indicates the derivative with respect to the normal direction. $\hat{q}$ is flux corresponding to the vibration velocity of the sound source, and Z_w is acoustic surface impedance of wall. The energy flows in sound field through source Γ_d and flows out through Γ_w as a loss.

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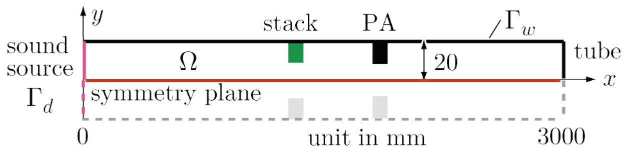

Fig.1 Finite element model.

An acoustic tube of 3 m in length and 4 cm in width is assumed in which the upper half domain is only considered for the analytical region. The sound field is divided into 1500 × 80 triangular elements of the first order. The left end of the tube is driven with uniform velocity of 2.45 mm/s and another end is rigidly terminated. The sound speed of air is c_0=340 m/s and density is ρ_0=1.2 kg/m^3. The normalized acoustic surface impedance of the tube wall is assumed to be $\overline{Z}_w = Z_w/(\rho_0 c_0)$=400. **Figure 2** (a) shows a stack of honeycomb like structure with many channels, and (b) shows a finite element model of the stack in which single channel is assumed with low acoustic impedance of $\overline{Z}_w = 38.5$. The model of phase adjuster (PA) is same as the stack.

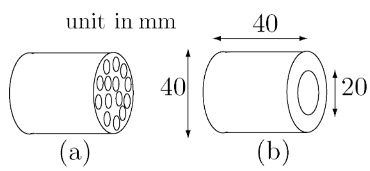

Fig.2 Model of stack and phase adjuster.

4. Numerical experiments

4.1 Standing wave field with stack

Figure 3 shows distributions of sound pressure and particle velocity at 114.1 Hz of the second resonant frequency. In the figure, the solid lines indicate the results with stack, and the broken lines without stack. The resonant frequency hardly changes in the presence of the stack. The amplitude decreases, and the node in front of the stack changes in the presence of the stack. **Figure 4** shows the sound intensity distributions. The sound intensity monotonically decreases without the stack, while it rapidly decreases between the front and the rear of the stack. The difference of sound intensity ΔI is equivalent to the outflow of the acoustic energy from the stack. The difference ΔI is converted into the thermal energy if the tube acts as a thermoacoustic system. The conversion efficiency $\eta = \Delta I / I_0$ in this case is 43.4%, where I_0 is the input intensity at the source.

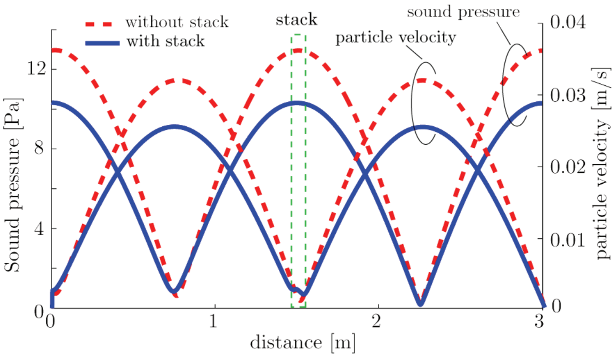

Fig. 3 Distributions of sound pressure and particle velocity.

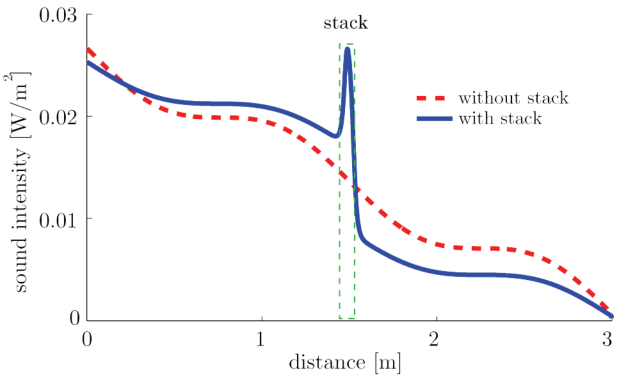

Fig. 4 Distributions of sound intensity.

Figure 5 shows the relation between the position of the stack and the efficiency. The maximum efficiency is obtained at the position of the maximum sound pressure.

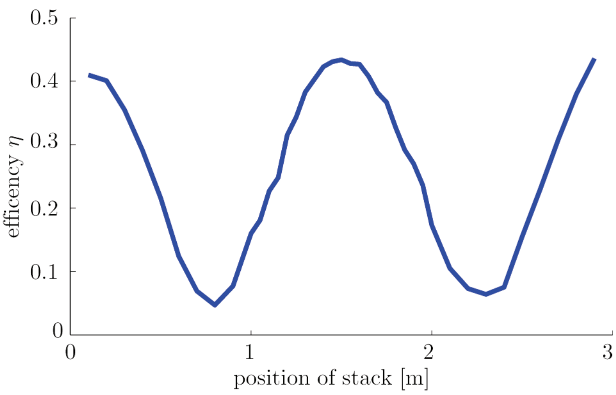

Fig. 5 Relation between efficiency and the position of the stack

4.2 Effect of phase adjuster

Figure 6 shows the distributions of sound intensity with the phase adjuster (PA), which is located behind 0.3m from the stack. The intensity is slightly changed in the presence of PA. **Figure 7** shows the relation between the position of PA and the improvement ratio of the energy conversion efficiency η / η_0, where η_0 is the efficiency without PA. In the figure, the internal diameter of PA has been changed with 10 mm, 20 mm, and 30 mm, respectively. The efficiency is improved when PA is located near the stack, and PA with narrower internal diameter is located. The efficiency improvement of several ten % is reported in the experiments of the loop tube system [4], while the effect of PA is slight in this numerical result. This is because the effect of PA repeatedly applied in the loop tube.

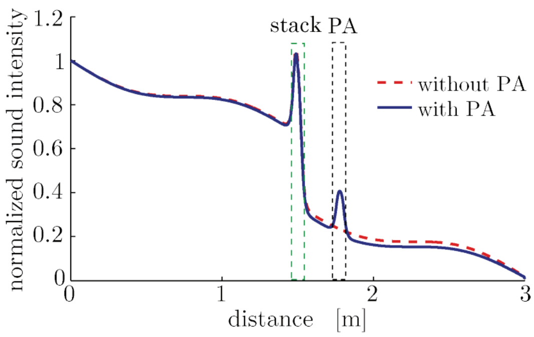

Fig. 6 Distribution of sound intensity with PA.

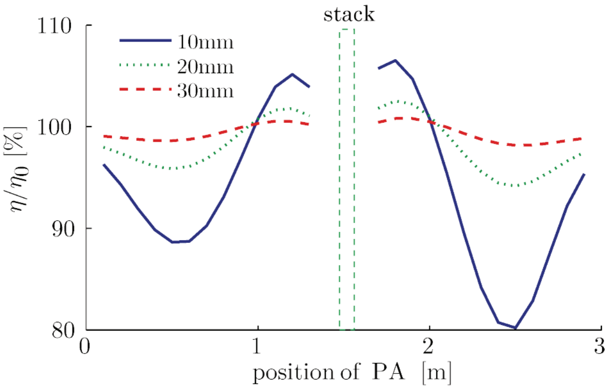

Fig. 7 Relation between efficiency improvement and position of PA.

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