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Structural-acoustic interaction of a three-dimensional panel–cavity–duct system with non-uniform boundary restraints

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HIGHLIGHTS

- 3-D drum silencer with non-uniform boundary is modeled via energy principle.
- Structural-acoustic field functions are constructed as improved Fourier series.
- Current model is validated through the comparison with those in literature.
- Non-uniform boundary can improve sound attenuation in certain frequency range.

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ABSTRACT

Plate drum-silencer has received a lot of research attention due to its excellent low frequency noise attenuation and flow-through characteristics. The majority of existing studies are limited to the classical or uniform elastic boundary conditions, while there is little effort devoted to the study of boundary restraint non-uniformity effect on the vibro-acoustic behavior of such silencing system. Motivated by the current limitation, the structural-acoustic coupling model of a three-dimensional panel–cavity–duct silencer is established, in which the elastic boundary restraint can be set as arbitrary distribution function. Energy principle is formulated to describe the vibro-acoustic dynamics of such 3-D panel–cavity–duct silencing system, with the admissible field functions constructed as the superposition of standard Fourier series and the auxiliary edge/interface smoothed terms. All the coupled system response information can be derived in conjunction with Rayleigh–Ritz procedure. Numerical examples are presented to validate the proposed model through the comparison with those from other approaches. The coupling effects of boundary restraining coefficient and plate bending stiffness on sound attenuation performance of such cavity–backed plate–duct silencer are then discussed and analyzed. The results show that the relationship of translational restraints at the duct entrance and exit, k_{x0} and k_{xLx}, can be obtained through property inverse proportional functions to achieve the optimal sound attenuation. There will be significant improvement in some special frequency band compared with the uniform restraints distribution, especially for the plate with lower bending stiffness The experimental study is also performed to verify the theoretical prediction from current model. In reality, the arbitrary non-uniform elastic edge restraints represent the most general class of boundary conditions, and can provide more optimal space for such drum silencer design.

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Nomenclature

B	Bending stiffness of plate
B^*	Non-dimensional bending stiffness
B_s	Complex bending stiffness
c_0	Sound speed
$\mathbf{C_{a\&p}}$	Cavity–plate interaction matrix
$\mathbf{C_{d\&p}}$	Duct–plate interaction matrix
E	Young's modulus
$f(x)$	Distribution function along x
$f(y)$	Distribution function along y
f_1	Initial frequency of stop-band
f_2	Terminal frequency of stop-band
G	Shear modulus
h_c	Cavity depth
h	Duct height
H	Heaviside function
k_0	Plane wave number
k	Stiffness of spring
k_c	Spring coefficient
k_1	Minimum value of distribution function
k_2	Maximum value of distribution function
k_{x0}, k_{xLx}	Translational stiffnesses at, respectively, $x = 0$ and $x = L_x$
k_{y0}, k_{yLy}	Translational stiffnesses at, respectively, $y = 0$ and $y = L_y$
K_{x0}, K_{xLx}	Rotational stiffnesses at, respectively, $x = 0$ and $x = L_x$
K_{y0}, K_{yLy}	Rotational stiffnesses at, respectively, $y = 0$ and $y = L_y$
$\mathbf{K_p}$	Stiffness matrix of plate
$\mathbf{K_a}$	Stiffness matrix of cavity
L_x	Plate length
L_y	Plate width
m^*	Mass ratio
$\mathbf{M_p}$	Mass matrix of plate
$\mathbf{M_a}$	Mass matrix of cavity
p_{cavity}	Acoustic pressure inside the cavity
p_{duct}	Acoustic pressure inside the duct
p_i	Incident plane sound wave
$\mathbf{Q}$	Reactive sound intensity
$\mathbf{S}$	Active sound intensity
V	Particle velocity of plate
w	Flexural displacement of plate
ω	Frequency in radian
μ	Poisson's ratio
ρ_s	Mass density of plate
σ	Plate thickness
ρ_0	Air density
δ_{mn}	Kronecker delta function.
σ_s	Stiffness damping coefficients
ψ_{poly}	Polynomial function
ψ_{tri}	Trigonometric function

1. Introduction

Low frequency noise in duct is often encountered in various engineering occasions, and is widely accepted as a great challenge for the noise control engineers. For many years, a lot of research efforts have been devoted to the development of low frequency duct noise control techniques. Generally, the current control methods can be divided into two categories, based on if there is a secondary external energy introduced to the primary system. For the active control, the electronic control and sensing devices are required, which will further make the system more complicated and expensive. That may be why the extensive application of active control is still not observed in various industries. Compared with the