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A scaling procedure for the response of an isolated system with high modal overlap factor

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Abstract

The paper deals with a numerical approach that reduces some physical sizes of the solution domain to compute the dynamic response of an isolated system: it has been named Asymptotical Scaled Modal Analysis (ASMA).

The proposed numerical procedure alters the input data needed to obtain the classic modal responses to increase the frequency band of validity of the discrete or continuous coordinates model through the definition of a proper scaling coefficient. It is demonstrated that the computational cost remains acceptable while the frequency range of analysis increases.

Moreover, with reference to the flexural vibrations of a rectangular plate, the paper discusses the ASMA vs. the statistical energy analysis and the energy distribution approach. Some insights are also given about the limits of the scaling coefficient.

Finally it is shown that the linear dynamic response, predicted with the scaling procedure, has the same quality and characteristics of the statistical energy analysis, but it can be useful when the system cannot be solved appropriately by the standard Statistical Energy Analysis (SEA).
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Keywords: Linear dynamic response; High frequency; Asymptotical modal solution; Computational method

1. Introduction

The search for predictive vibroacoustic methodologies relies on the need of increasing the frequency range of analysis of a generic structural and/or acoustic operator at acceptable computational costs.

The deterministic techniques, such as the Finite Element Analysis (FEA), work well for predicting the *local* response in the *low frequency* range. Better saying, they are able to predict the response at a specific location and at specific time (frequency) when a given load is defined, and when the mode shapes are well resonating. The modal overlap factor, μ (the product among the excitation frequency, the modal density and the damping

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Nomenclature

a, a_I, a_{II}	plate lengths
$Area$	plate area
A_{ss}	diagonal term of the energy influence coefficient matrix
b, b_I, b_{II}	plate widths
E	Young's modulus
$E[\cdots]$	statistical expectation
f	excitation frequency
F	mechanical excitation amplitude
h	plate thickness
H	excitation spectrum
i	imaginary unit
j_x and j_y	(integer) numbers of half wavelengths in x and y directions, respectively
m	plate mass
m_{gen}	generalised mass
NT	number of modes retained for the expansion series
N_x	number of half waves along x-axis
N_y	number of half waves along y-axis
S_f	spectrum of the mechanical excitation
T	kinetic energy
v	out-of-plane plate velocity
v^2_{local}	square velocity according to Eq. (1)
v^2_{acq}	mean square velocity according to Eq. (5)
v^2_{mean}	mean square velocity according to Eq. (6)
v^2_{SEA}	mean square velocity according to Eq. (7)
v^2_{Δ}	mean square velocity according to Eq. (8)
x, x_A, x_B	reference axis and coordinates of the generic points A and B
y, y_A, y_B	reference axis and coordinates of the generic points A and B

Greek symbols

γ	damping function
Γ_{jj}	frequency integrals
Δf	frequency band
ε	scaling coefficient
η, η_I, η_{II}	structural damping loss factors
μ, μ_I, μ_{II}	modal overlap factors
υ	Poisson's ratio
Π_{in}	input power
ρ	material density
σ	scaling coefficient
ϕ_i	ith analytical mode shape of the plate
ψ_j	space integral of the jth mode shape of the plate
ω	circular excitation frequency ($= 2\pi f$)
ω_j	natural circular frequency of the jth mode
Ω	frequency interval (rad/s)

loss factor) is the key parameter: it has to be low, $\mu \ll 1$, for fulfilling FEA conditions. In absence of any analytical development, the deterministic techniques model directly the wavelength, by assigning at least five solution points (four elements) for each complete wave [1–3].

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