

# Elucidation of Coupled Vibrations in Highly Damped Systems Using Modal Energy Contribution Analysis

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Automotive noise, vibration and harshness phenomena occur through dynamic coupling among multiple subsystems, necessitating clear specification of required characteristics for each subsystem. Metrics based on *kernel* compliance analysis (*kCA*) have been proposed to analyze subsystems' coupling and quantify the modal contributions of individual subsystems to whole structure resonances. However, these conventional metrics targeted undamped or lightly damped systems, leaving challenges in applying them to heavily damped systems.

This paper proposes a method applicable to heavily damped systems by directly utilizing the transmitted forces between subsystems derived from the Frequency-based substructuring. For steady-state responses at arbitrary excitation frequencies in heavily damped systems, the proposed method evaluates modal energy contribution of each subsystem, while decomposing the modal contribution into two factors: a mode stimulation factor and a standalone subsystem's modal property factor. This enables independent, quantitative evaluation of whether the cause of whole structure resonance lies in the proximity of a standalone subsystem's eigenfrequency or in mode stimulation, thereby facilitating the clarification of subsystem requirements and the derivation of efficient structural modification guidelines.

The validity of the proposed method was demonstrated by applying it to a finite element model (Fig. 1) that simulates the vibrations of a tire-suspension system. This model consists of two rectangular plates (subsystem A and B) coupled by a scalar spring with five degrees of freedom at the hub, excluding the tire rotational direction. An oblique harmonic external force is applied to region *i* of subsystem A. Additionally, a structural damping of loss factor  $\eta = 0.04$  is applied. The energy contributions  $E_{Kr}$  and  $E_{Pr}$  (top two rows of Fig. 2) revealed that the 4th mode is dominant in both subsystems. Analysis of the mode stimulation factor  $x_r$  (third row of Fig. 2) identified a peak around 120 Hz in both subsystems. On the other hand, the standalone modal property factors  $y_r$  and  $z_r$  (fourth and fifth rows of Fig. 2) show that the 4th mode of subsystem A has an eigenfrequency peak around 110 Hz, with its skirt contributing to the whole structure resonance at 120 Hz. In contrast, the peak of the 4th mode of subsystem B is distant from 120 Hz, resulting in a small contribution from its standalone properties.

These results indicate that the contribution of subsystem A is dominant in the whole structure resonance near 120 Hz, driven by both high modal stimulation ( $x_r$ ) and close eigenfrequency proximity ( $y_r, z_r$ ). Consequently, structural modification guidelines were obtained: to control the whole structure resonance near 120 Hz, it is necessary to either adjust the separation of the standalone eigenfrequency of subsystem A or suppress the modal stimulation level.

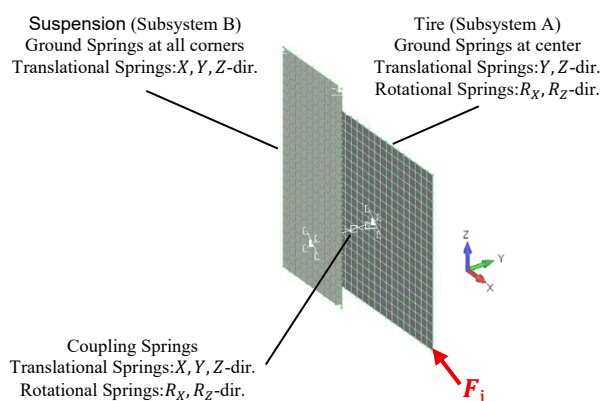


Fig.1 Finite element model capable of simulating a tire-suspension system.

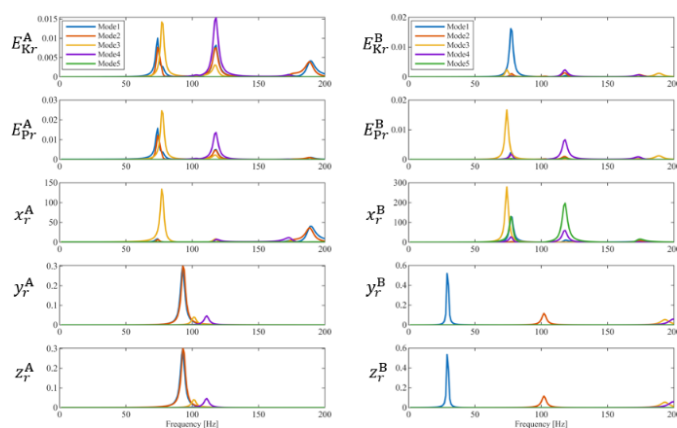


Fig. 2 Frequency responses of decomposed component modal energies ( $E_{Kr}, E_{Pr}$ ), force input factors ( $x_r$ ), and modal property factors ( $y_r, z_r$ ) for subsystem A (left) and subsystem B (right).