Finite Element Modelling of Machinery Vibration Isolation Systems

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Introduction

Vibration isolation of ship machinery is needed to achieve low radiated and habitable space noise levels. Shipboard isolation systems can be more difficult to design than similar systems on land due to space restrictions, weight limitations and ship motion considerations. Supporting foundation structures are often lightweight and relatively flexible and can influence the isolation system effectiveness. DREA is enhancing an in-house finite element vibration and strength analysis code VAST [1] to facilitate the analysis of passive vibration isolation systems. This paper describes the application of VAST to two problems. The first examines a simple machine decoupled from a flexible foundation structure using sixteen resilient isolators. The second considers the prediction of the natural frequencies of vibration of a helical compression spring isolator.

Modelling capabilities

The amplitudes of machinery vibration are normally small and it is generally sufficient to consider the isolation system dynamics using three-dimensional linear elastic models. Finite element analysis (FEA) methods can be used to predict isolation characteristics in the lower frequency range and are well suited for modelling the complex structural geometry. Upper frequency limits are dependent on the size of the structure (whether modelling an entire ship or a smaller component) and the degree of discretization that can be achieved with available computer resources. Machine, raft and supporting structure flexibility can be modelled as well as multiple degrees of freedom (DOF) at resilient isolator interfaces. Rotations and moments may in some cases be as important as translations and forces for transmission of vibrational energy. Wave effects within isolation mounts can be included using detailed finite element models of the resilient components.

The VAST code is presently being enhanced to include frequency dependent complex stiffness properties. This should improve capabilities to model 'rubberlike' materials and viscoelastic damping materials which are often used in vibration isolation systems.

Machine on multiple isolators with beam foundation

This example considers the application of VAST to the analysis of a simple machine consisting of a rectangular steel block (2489mm long, 508mm wide and 348mm tall) mounted on 16 isolation mounts (each with a vertical stiffness of 840N/mm and a horizontal stiffness of 420N/mm) modelled by massless beam elements of solid circular cross-section which were clamped to the support beams and pinned to the machine. The two steel support beams were 2845mm long with a 51mm wide by 152mm deep solid rectangular section. The material properties used for the machine and support beams were: a Young's modulus of 2.07×10^5MPa; a Poisson's ratio of 0.3; and, a density of 7874kg/m^3. The ends of the support beams were rigidly clamped. A harmonically varying machine excitation force was applied to the centre of the top face of the machine mass. The effect of a non-vertical excitation force was

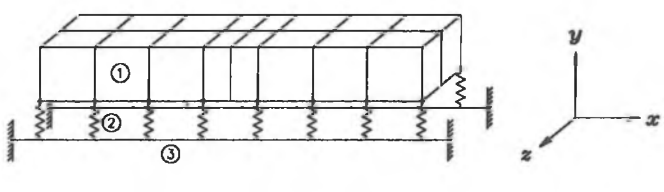


Figure 1: FE Model of machine on beam foundation and multiple isolators (1 = machine, 2 = isolator, 3 = support beam)

considered by applying the force obliquely with equal components in the x, y, and z directions. The finite element model is shown Figure 1. The machine mass consisted of sixteen brick elements (8-noded). One 2-noded beam element was used for each mount and nine 2-noded beam elements were used for each support beam.

The first 50 undamped natural frequencies and mode shapes of the system were predicted and used in a forced response analysis over the frequency range of 1 to 1600 Hz. The decoupling of the machine mass and the support beams caused localization of vibration modes. Some modes involved mainly motion of the machine mass and others involved mainly motion of the support beams. The lowest six modes were essentially machine rigid body modes (5.4 Hz to 15 Hz). The next twelve modes (39 Hz to 298 Hz) were vertical and horizontal support beam bending modes which occurred in pairs. The remaining modes (300 Hz to 1650 Hz) included machine bending modes, higher order beam bending modes, and torsional and longitudinal deformation modes in the beams and machine.

This problem was originally considered by Snowdon [2] (for vertical motion only and a rigid machine). He specified complex stiffnesses for mounts and beams. The present VAST program cannot handle damping of this type and, instead, a viscous damping was assumed (with a damping ratio of 0.05 for the first mode and 0.01 for higher modes). The vertical force transmissibility at one end of the support beams is plotted as a function of the machine excitation forcing frequency in Figure 2 (a different transmissibility was obtained for each support due to the unsymmetric loading; only one is shown). Peaks in transmissibility occurred near natural frequencies of both machine and beam vibration modes, giving significantly greater force transmission to the supports at higher frequencies compared to a case which modelled the machine and beams as rigid structures (shown by the dashed line in Figure 2).

Modelling of a spring isolator

The previous problem considered a system in which each isolator was assumed to be massless. Resonances within mounts can affect the isolation characteristics and this second example considers the prediction of the natural frequencies of a helical compression spring with closed ground ends, compressed between two rigid surfaces. This type of spring was modelled by

Table 1: VAST natural frequency predictions for the helical spring (Hz)

Mode:	1	2	3	4	5	6	7	8	9	10
Beam model	1089	1206	1475	1495	1898	2097	2189	2734	3020	4889
Solid model	1086	1193	1450	1475	1850	2088	2194	2713	2810	4661
Dev.(%)	0.3	1.1	1.7	1.4	2.6	0.4	-0.2	0.8	7.5	4.9

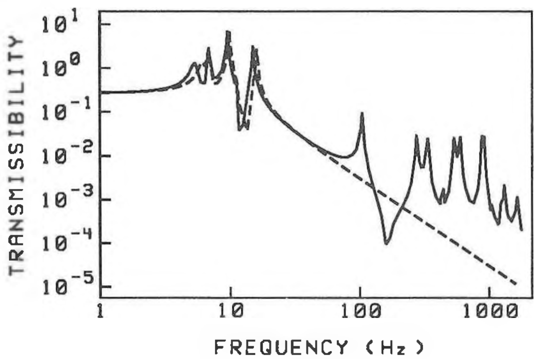
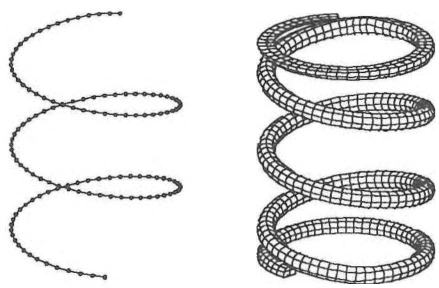


Figure 2: Vertical force transmissibility to the clamped end of one support beam



a) Beam (126 el.) b) Solid (944 elements)

Figure 3: FE Models of helical compression spring

Pearson [3] using a beam element transfer matrix method and assuming clamped ends and pinned points where the tip of the spring wire contacts the adjacent coil. A finite element model was constructed using 2-node general beam elements along the spring wire centre-line (shown in Figure 3a). A solid model was constructed using 20-noded brick elements (shown in Figure 3b) to provide an improved model of the end coil boundary conditions.

The details of the springs and the comparison to Pearson's results can be found in Reference [4] where it is shown that the beam element models were reasonably accurate in predicting the first eight to ten modes. The beam and solid FE model predictions are given in Table 1. The springs considered by Pearson and modelled in this example were much smaller (19 to 25mm long, 14mm O.D) and have much higher natural frequencies than those likely to be used in isolation systems.

Although the solid element models can provide more accurate high frequency predictions, the resulting finite element models become too large to include in overall isolation system models which involve many isolators, machinery rafts and the supporting structure.

Concluding Remarks

This paper has demonstrated that the VAST finite element program, and FEA methods in general, provide a useful tool for the analysis of machinery vibration isolation systems; however, the enhancement of capabilities for modelling of damping is desirable. Damping properties can vary significantly between different components of the isolation system and can include viscous, structural and frequency dependent damping found in 'rubberlike' materials which can be included using a complex stiffness. A direct frequency response method is being implemented in VAST [5] which will allow modelling of frequency dependent complex stiffness properties for materials.

Component mode synthesis methods [6] are also being implemented in VAST which may allow analysis of larger system models and efficient reanalysis for the evaluation of structural modifications such as changing some of the natural frequencies of the isolation system components to avoid coincidence with machine excitation frequencies. Power flow finite element methods [7] are in the early stages of development and potentially offer a useful finite element based technique for higher frequency analysis.

References

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