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Characteristics of vibration isolator model in transfer path analysis

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ABSTRACT

Two types of FEA model of vibration isolator are established to investigating the vibration transfer feature of each model for finding an economical and acceptable way to modeling vibration isolator in rotor-machine induced vibration system transfer path analysis. The FEA model includes: spring-mass-damper model, static stiffness and damping equivalent beam model. Based on the models above, the vibration transfer related parameters that of impedance and stiffness in wide frequency range are obtained utilizing FEA computer code under various system parameters and fully discussed by comparing analysis. Then simulation analysis is performed for a real system which have reference vibration parameters at various location by test using spring-mass-damper and beam model for vibration isolator, the simulation and test results are fully compared and studied. Generally specking for modeling vibration isolator in system vibration simulation analysis, the spring-damper model is a good choice if the vibration isolator so what like a single degree of freedom system or the interested frequency range is a narrow band near to the system natural frequency for the dynamic stiffness and impedance of this model only fit well with reality in narrow frequency band,, the beam model is more suitable than spring model if the vibration isolator have more than one dominant frequency or especially in high frequency interested application for this model have more degree of freedom to capture the dominant frequencies, so the beam model is the better choice for it is easy to model and has acceptable transfer feature compare to real in most cases.

Keywords: vibration isolator, fea model, transfer path analysis
I-INCE Classification of Subject Number: 76

1. INTRODUCTION

Vibration isolator is widely used to isolate or reduce vibration transferring from source to receiver. Normally vibration isolator is designed as a composite structure or superelastic structure for maximizing vibration energy dissipation. In the view of mathematics, this kind of structure leads to discontinuous impedance which is helpful to reduce vibration transmission, and the vibration isolator is commonly use rubber or rubber-made air bag as base elastic part with steel frame for mounting, and it is hard to model directly in vibration transfer path analysis application due to its special composition in structure. In engineering application, the usual approach is to obtain the dynamic stiffness(stiffness matrix) in low frequency range or impedance in high frequency range of the vibration isolator by test[1] or by calculation via the numeric model[2], and in a comprehensive model, such as full-vehicle or rotor-machine concerned pump-pipeline-vessel system, the vibration isolator is normally modeled by a sping-mass-damper system[3].

In this paper, two types of numeric model of vibration isolator are established to investigating the vibration transfer feature of each model for finding an economical and acceptable way to modeling vibration isolator in rotor-machine induced vibration system transfer path analysis. The numeric model include: spring-mass-damper model(spring model), stiffness and damping equivalent beam model(beam model).

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2. METHOD TO DEVELOP BEAM MODEL

For 3-D beam element, the stiffness matrix[4] in element coordinates is expressed below:

$$[K_\ell]=\begin{bmatrix}
AE/L & & & & & & & & & & & \\
0 & a_z & & & & & & & & & & \\
0 & 0 & a_y & & & & & & & & & \\
0 & 0 & 0 & GJ/L & & & & & \text{Symmetric} & & & \\
0 & 0 & -c_y & 0 & e_y & & & & & & & \\
0 & c_z & 0 & 0 & 0 & e_z & & & & & & \\
-AE/L & 0 & 0 & 0 & 0 & 0 & AE/L & & & & & \\
0 & -a_z & 0 & 0 & 0 & -c_z & 0 & a_z & & & & \\
0 & 0 & -a_y & 0 & c_y & 0 & 0 & 0 & a_y & & & \\
0 & 0 & 0 & -GJ/L & 0 & 0 & 0 & 0 & 0 & GJ/L & & \\
0 & 0 & -c_y & 0 & f_y & 0 & 0 & 0 & c_y & 0 & e_y & \\
0 & c_z & 0 & 0 & 0 & f_z & 0 & -c_z & 0 & 0 & 0 & e_z
\end{bmatrix}$$

Where:

A=cross-section area, E=Young’s modulus, L=element length, G=shear modulus

J=torsional moment of inertia, I_x=input torsional moment of inertia

J_x=polar moment of inertia= $I_y + I_z$, $a_z = a(I_z,\varphi_y)$, $a_y = a(I_y,\varphi_z)$, $b_z = b(I_z,\varphi_y)$

⋮

$f_z = f(I_z,\varphi_y)$, $f_y = f(I_y,\varphi_z)$, $a(I,\varphi) = \frac{12EI}{L^3(1+\varphi)}$, $c(I,\varphi) = \frac{6EI}{L^2(1+\varphi)}$, $e(I,\varphi) = \frac{(4+\varphi)EI}{L(1+\varphi)}$

$$f(I,\varphi)=\frac{(2-\varphi)EI}{L(1+\varphi)},\ \varphi_y=\frac{12EI_z}{GA_z^sL^2},\ \varphi_z=\frac{12EI_y}{GA_y^sL^2}$$

I_i=moment of inertia normal to direction i

A_i^s=shear area normal to direction i=A/F_i^s

F_i^s=shear coefficient

If the translational and rotational stiffness at the three orthogonal axes direction of vibration isolator are known, we can use the formula above to obtain parameters of beam element, such as cross section area, moment of inertia, shear coefficient, etc. The beam has the same stiffness with the vibration isolator, and is so-called stiffness equivalent beam model.

Alternatively, if the stiffness of the vibration isolator is obtained by means of a cantilever beam, then the parameters of beam element can be calculated by formulas below:

$$A=\frac{K_xL}{E}\qquad I_y=\frac{K_zL^3}{3E}\qquad I_z=\frac{K_yL^3}{3E}$$

3. FEA MODELS

3.1 Spring model

For the purpose of deduce, a typical rubber made vibration isolator is used and its vertical stiffness is $3\times10^6\ N/m$, its mass is $56kg$, and its damping ratio is 4%. We can use two nodes spring-mass-damper element to simulate the vibration isolator at one direction such as vertical direction for spring element has only one DOF(degree of freedom), and using finite element analysis software, such as ANSYS, with one node fixed while the other node applying white noise force, by performing harmonic response calculation to obtain the dynamic stiffness and impedance, this is an usual approach applied in test. The vibration isolator is modeled using linear spring-damper element named COMBIN14 of ANSYS. The stiffness and impedance curve of the vibration isolator in wide frequency range(from 1Hz to 1000Hz) are depicted in figure 1, from the figure, we can see the stiffness and impedance have similar trends with frequency as that the stiffness and impedance are decreasing while the frequency increasing firstly till reach the valley value at the nature frequency of vibration isolator, then increasing together with the frequency. For the vibration isolator is modeled with one degree of freedom system, and there is only one valley in the curves, so if the frequency interested is much beyond the nature frequency, using spring to model vibration isolator may be have big difference with reality, then the spring model is acceptable under the condition that the interested frequency near the nature frequency or the vibration isolator is almost a one degree system. It should be noted that the nature frequency is not merely be of the vibration isolator and it is be of the system it applied in.

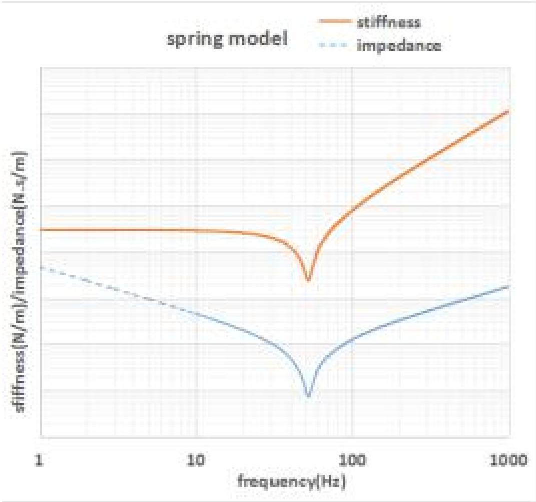


Figure 1: stiffness and impedance of spring model

3.2 Beam model

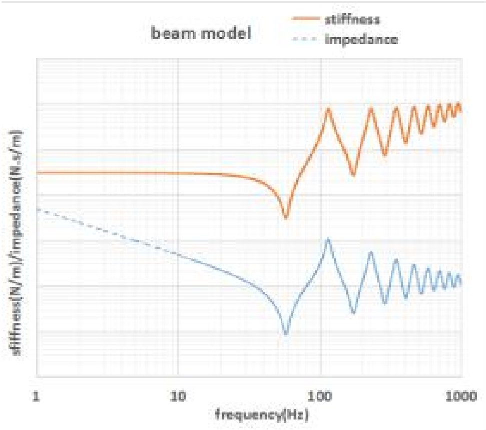
Similarly using the stiffness, mass and damping ratio, an equivalent beam model can be constructed use 3D linear finite strain beam named BEAM188 of ANSYS, and the shape of vibration isolator is assumed as solid cylinder for simplify model. The real constant of beam element is presented in table 1.

Table 1: real constants of beam element

parameter	value
element type	beam188
section type	csolid
section radius	294 mm
density	1145.7 kg/m^3
length	180 mm
young modulus	1.988 MPa
Element number	30

It is to noted that the rubber is hyper-elastic material and its material parameters should be described by hyper elastic constitutive equation in theory, however the relative displacement induced by vibration is very small(normally less than 1mm), so it is belong to small-deformation problem and it is also suitable using elastic constitutive equation to modeling.

By applying the same boundary as spring model, the dynamic stiffness and impedance can be obtained as depicted in figure 2.



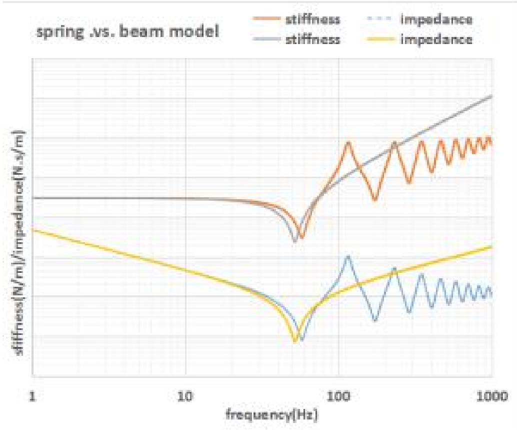


Figure 2: stiffness and impedance of beam mode

Figure 3: difference of spring and beam models

By comparing figure 2 with figure 1, it is clearly to see:

(i). The beam model have nearly the same stiffness and impedance with spring model before about 52Hz which correspond to the first nature frequency.

(ii). After the first nature frequency, the stiffness and impedance of the two model have big difference either in shape or in value. which increase monotonically with frequency increasing for spring model while there are many peaks and valleys for beam model which is due to the fact that the beam model has more DOF(30 active nodes, 180 dof) than spring model, the frequencies of peak and valley respond to the first nature frequency times odd and even serially, and the final value of stiffness and impedance of spring model is bigger than beam model through their numeric relation will changing constantly before.

For better understanding the changing trend with frequency of stiffness and impedance of the two models, all the data are figured in figure 3.

From figure 3, the frequencies of first valley of the two model have slight difference, that is caused by distribution of the mass of numeric models. For spring model, the mass of vibration isolator is lumped at one active node, however there are more than one active mass node for beam mode, so the spring model has lower first nature frequency.

Next the effects of the number of element of beam model to stiffness and impedance will be discussed by manner of increasing the nodes to 50 and decreasing the nodes to 10. Figure 4(1/2) show the stiffness and impedance for 30 nodes and 10 nodes models which have nearly the same stiffness and impedance while the frequency smaller than 232Hz(correspond to the 2nd peak), and beyond that frequency range, the difference become more and more significant with the increasing of frequency. Figure 4(2/2) show the stiffness and impedance for 30 nodes and 50 nodes models which have nearly the same stiffness and impedance at the full frequency range that this calculation concerned, that is to say, the 30 nodes model is enough for this situation, in other word, there may occur unacceptable difference at frequency range higher than 1000Hz, then it is a matter how many nodes is enough for a specified problem, as a general guideline, one can increase model's nodes gradually till the mode frequency have slight change in interested frequency range, and the last model is acceptable.

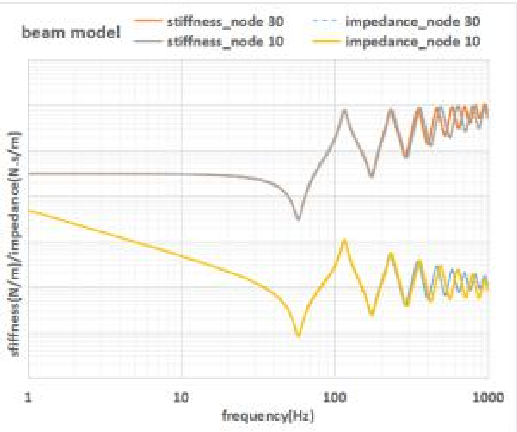


Figure 4(1/2): effects of element number

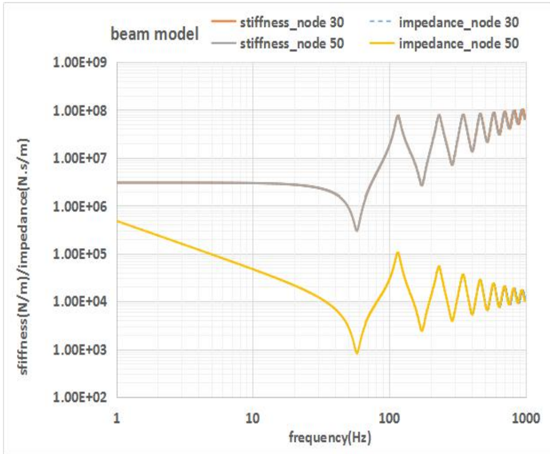


Figure 4(2/2): effects of element number

4. APPLICATION

Hereafter the spring and beam model will be used to model a real rubber-made airbag vibration isolator which has the impedance curve in wide frequency range(5Hz-1000hz) obtained by test, and the impedance is presented in figure 5. By reading from figure 5, it can be found that the real impedance has big difference to the ideal model discussed above, they lie in:

(i). The difference of frequency of adjacent peak and valley is not equal to the first nature frequency as the ideal beam and solid presented earlier .

(ii). There are limit peaks and valleys in somewhat a wide frequency range though the airbag vibration isolator is infinite degree of freedom, this phenomena can be interpreted by the fact that the stiffness of isolator is governed by pressurized air and its mass is very small than steel-made frame and mount interface.

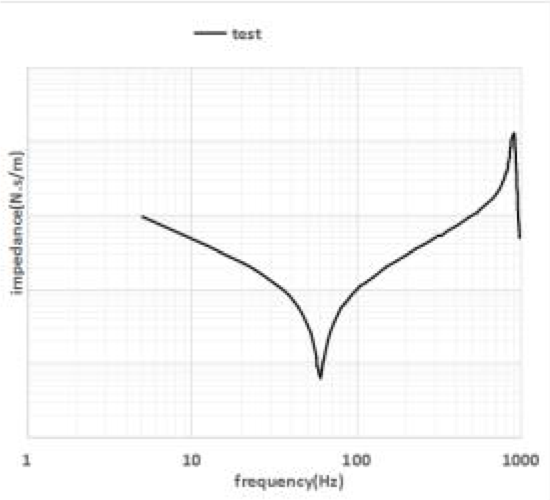


Figure 5: impedance from test

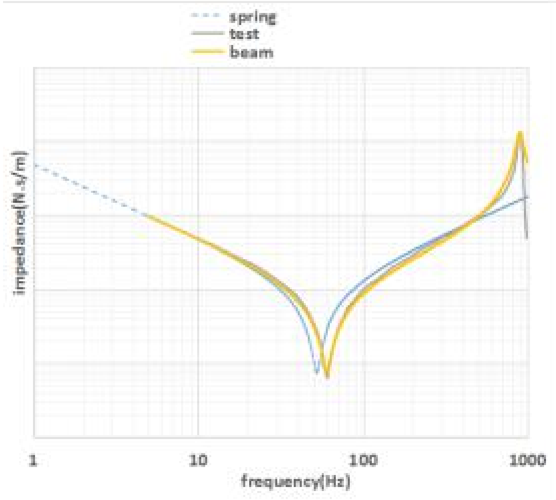


Figure 6: impedance from test and numeric models

In figure 9, the impedance of airbag vibration isolator from test and numeric modes are presented, numeric models include spring model and beam model constructed via its static stiffness which can be calculated by the first impedance divide the corresponding frequency. In the figure, we can see that the impedance of beam model fit that from test very well in large majority frequency point and captured the peak and valley successfully though they have clear difference after the peak frequency. In the other side, the impedance from spring model only fit well with which from test at frequencies smaller than valley frequency and the crossing zone between peak and valley.

So it is seemed that the beam model is better than the spring model, for proving it, the two models are introduced into CAE-NVH model which consist of one centrifugal pump which is the vibration source, the pump is mounted to the support platform via airbag vibration isolator, some pipeline conveying fluid connect to the the pump and vessels which mount on the same support platform of pump. After performing calculation analysis, the difference of the vibration acceleration level in dB of vibration isolator modeled by spring and beam are 31.29dB and 21.46dB respectively, the value calculated according to test is about 20dB, so the more real result is obtained by beam model.

5. CONCLUSIONS

In vibration transfer path analysis, the vibration isolator can be modeled by means of spring, beam and etc, the spring model have the limit that it is only suitable for single degree of freedom or similar system, otherwise the calculation may have significant difference with reality. The beam model can be constructed to fit the transfer feature very well with which of the real vibration isoloator in wide frequency range, and is suitable for most application and so the beam is the economical and acceptable way to model vibration isolator in most case.

7. REFERENCES

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