NOTES ON FINITE ELEMENT MODELING OF ISOLATED COMPONENTS

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Introduction

Finite element modeling is used to determine the natural frequencies and mode shapes of components mounted via isolators. The modal analysis may be extended to determine the response to a given base input.

The first six modes are usually spring-mass type modes with the component behaving as a rigid point mass. Higher modes involve the elastic response of the main component, along with isolator participation.

The isolators are modeled as translational springs. Each isolator may be represented by three DOF springs, one for each orthogonal axis.

The component and its bracket assembly are typically modeled with plate or solid elements.

The component assembly hardware is almost always significantly stiffer than the isolators. This condition may yield to numerical instability and error in the finite element results. In some cases, the modal analysis may yield one or more frequencies with a value of zero, even though the component has no rigid-body modes. Ironically, the mode shapes corresponding to the zero frequencies may appear to be correct.

The solution is to add rotational springs to the model.[1] Selecting the location of these springs and their respective stiffness values is a matter of engineering judgment, along with some trial-and-error.

As a parallel effort, the component hardware should be modeled as a point mass at the system center-of-gravity. The point mass should include the rotational inertia. The isolators are represented by translational DOF springs for this six-degree-of-freedom spring-mass model. The natural frequencies and mode shapes are then calculated using the method in Reference 1.

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[1] This approach came from discussions with representatives of NE/Nastran.

The finite element model thus has the added rotational springs for numerical stability, although the spring-mass model does not. Each has rotational inertia.

The first six natural frequencies of the finite element model should reasonably match the corresponding frequencies of the six-dof spring-mass model. The finite element model's rotational springs need to be modified accordingly until this condition is met.

In some sense, this process calibrates the finite element model so that it can be used for further modal and frequency response function analysis. This calibration process should be performed as early as possible in the analysis task to avoid losing time with an incorrect finite element model.

Case History 1

An avionics component was mounted via four Lord isolators. Each isolator had a translational stiffness of 80 lbf/in, which was uniform in all three axes.

The bracket was modeled with plate elements. Each node of each plate element has three translational and three rotational degrees-of-freedom. The main component was modeled as a point mass, attached to the bracket via rigid links. The point mass included both mass and rotational inertia. The rotational inertia was estimated.

Three rotational DOF springs were added at each of the four mounting locations for numerical stability. The rotational stiffness value was 100 in-lbf/rad. Whether this stiffness represents the true rotational stiffness of the Lord isolator model is unclear and perhaps irrelevant.

The undeformed model is shown in Figures 1 and 2. The first and second mode shapes are shown in Figures 3 and 4, respectively.

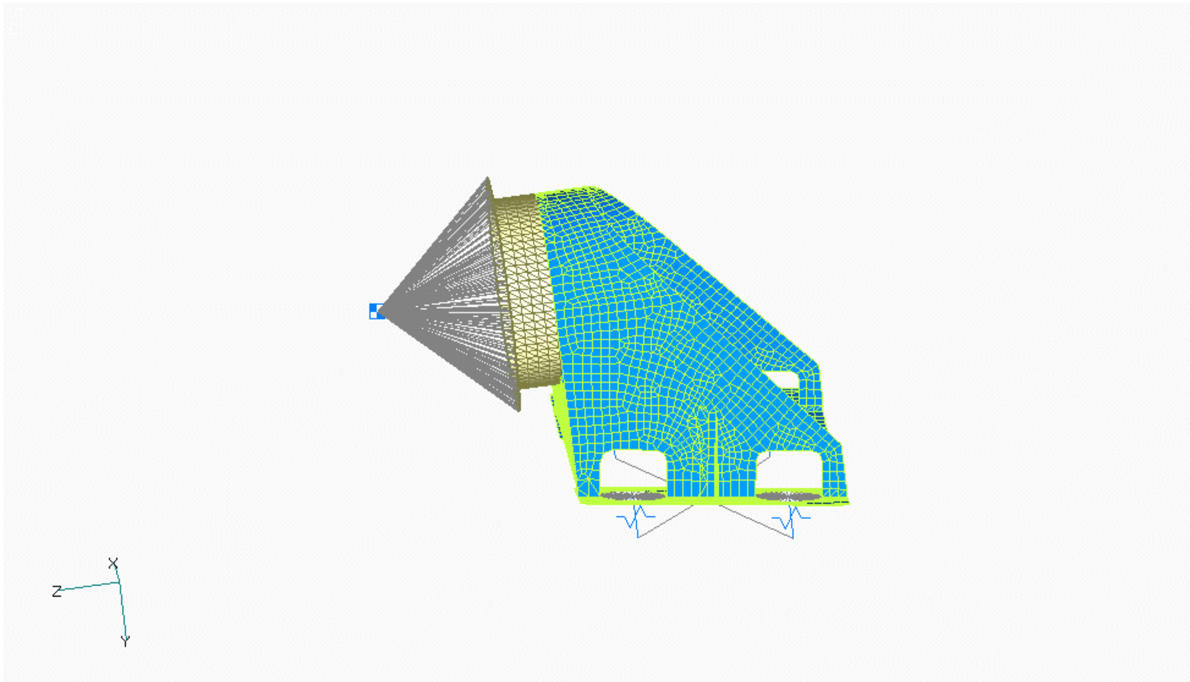

Figure 1. Case 1, Undeformed Model, View 1

Figure 2. Case 1, Undeformed Model, View 2

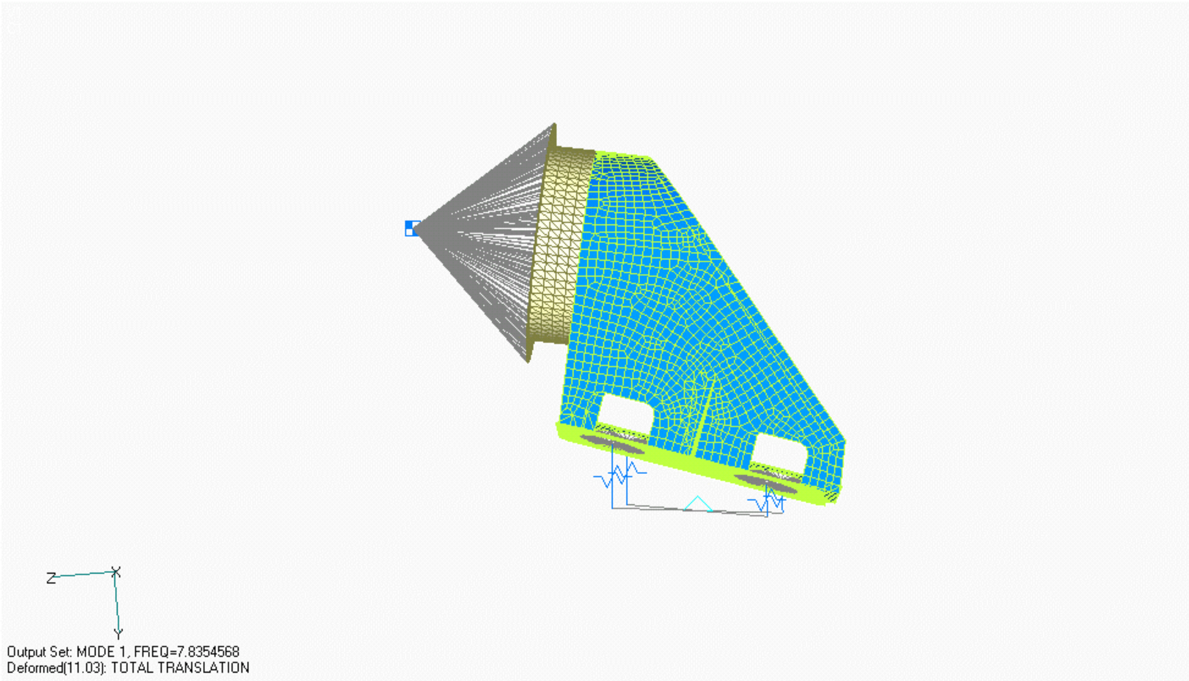

Figure 3. Case 1, Mode 1

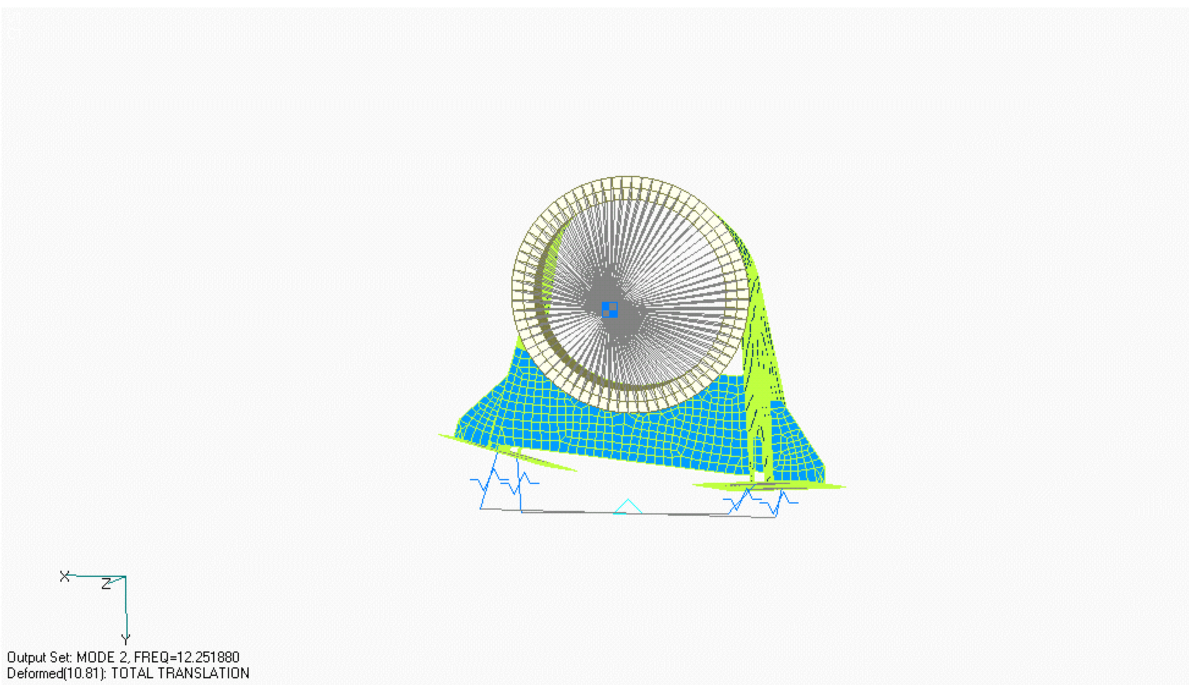

Figure 4. Case 1, Mode 2

Table 1. Case 1, Isolated Model, FEA Natural Frequency Results			
Mode	FEA Frequency (Hz)	Hand Calculation (Hz)	Error
1	7.8	6.7	14.1%
2	12.3	14.1	-14.6%
3	26.3	26.3	0%
4	40.5	36.6	9.6%
5	53.1	51.6	2.8%
6	57.5	66.8	-16.2%

The natural frequency results are given in Table 1. Again, the FEA model included rotational springs for numerical stability. The "Hand Calculation" is made using Reference 1. The Hand Calculation mode shapes were also verified with respect the FEA mode shapes.

Case History 2

Another avionics component was mounted via four Lord isolators. Each isolator had a translational stiffness of 80 lbf/in, which was uniform in all three axes.

The main component was modeled as a point mass, attached to the bracket via rigid links. The point mass included both mass and the estimated rotational inertia.

The bracket was modeled with solid elements. Each node of each solid element had three translational but zero rotational degrees-of-freedom. This is inherent in the element formulation.

Adding rotational springs at the translational spring locations was ineffective given that the solid elements lack rotational degrees-of-freedom. Instead, the point mass was attached to the base via rotational DOF springs for numerical stability. Two springs were used, one for each lateral axis. The rotational stiffness was 100 in lbf / rad.

The undeformed model is shown in Figure 5. The first and second mode shapes are shown in Figure 6 and 7, respectively.

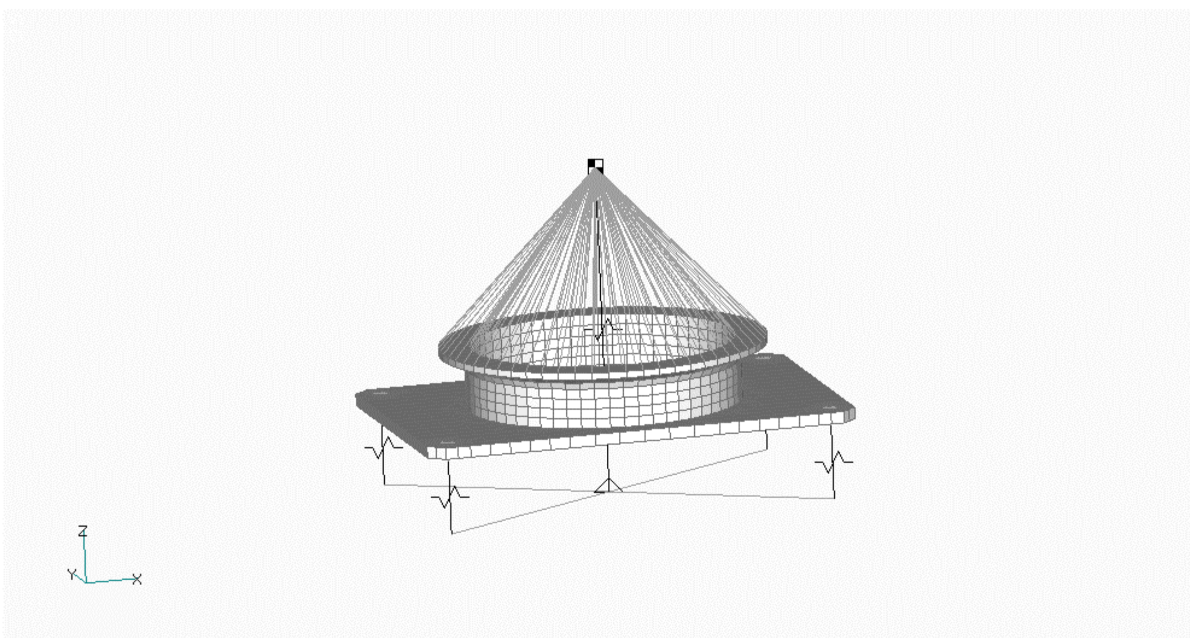

Figure 5. Case 2, Undeformed Model

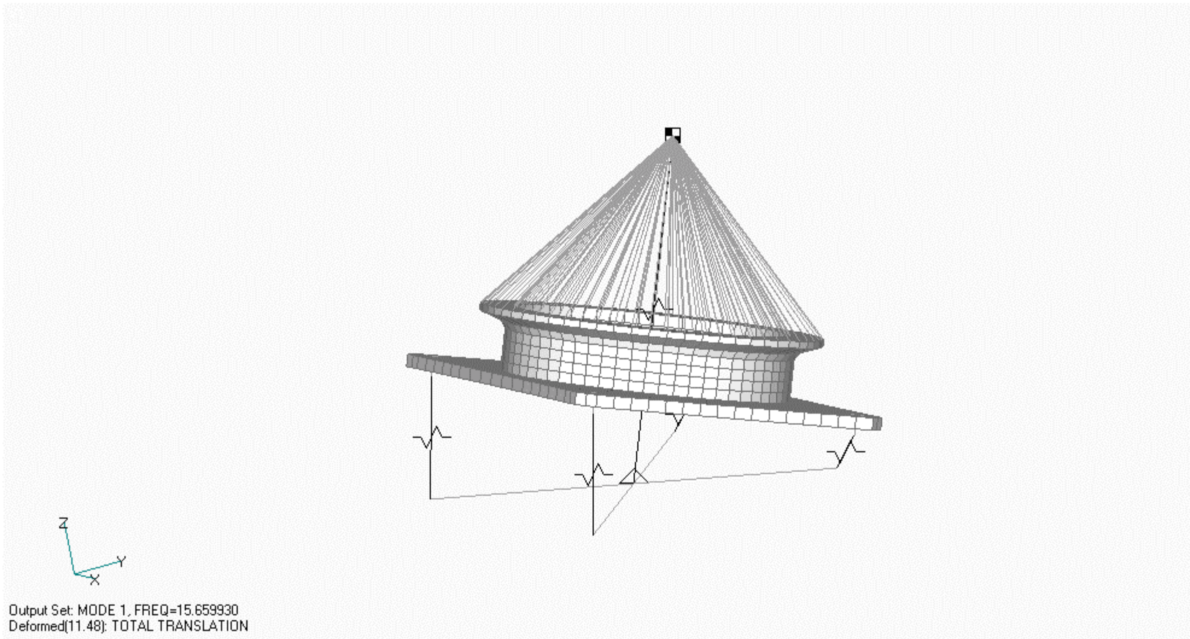

Figure 6. Case 2, Mode 1

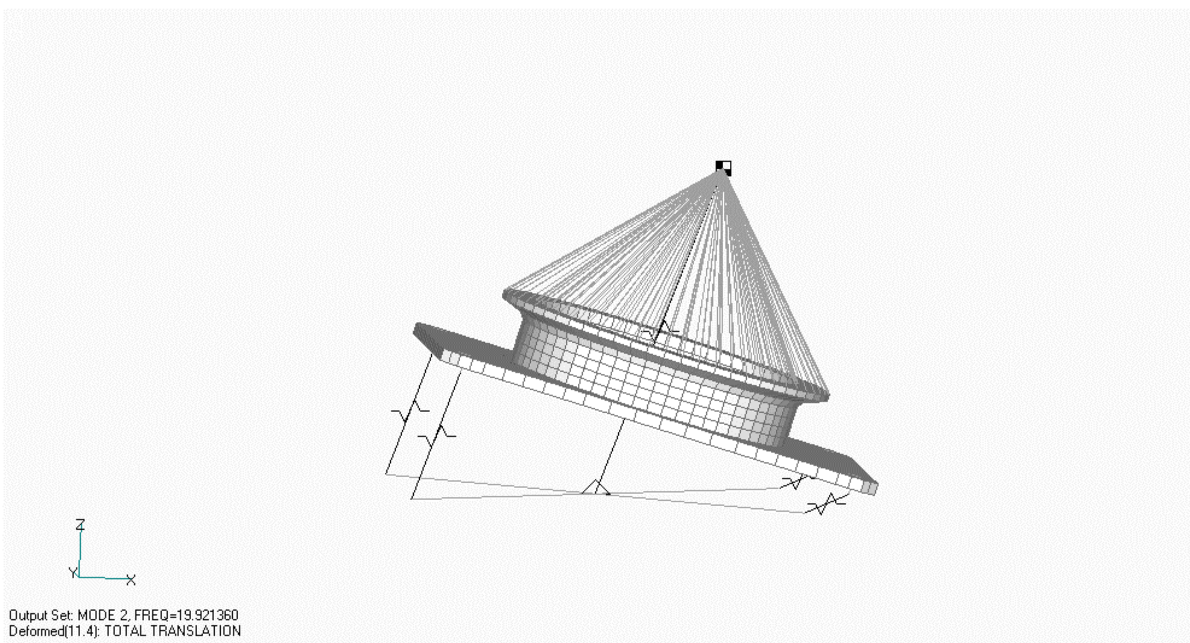

Figure 7. Case 3, Mode 2

Table 2. Case 2, Isolated Model, FEA Natural Frequency Results			
Mode	FEA Frequency (Hz)	Hand Calculation (Hz)	Error
1	15.7	15.7	0.0%
2	19.9	22.0	10.6%
3	30.4	30.5	0.3%
4	72.4	70.2	-3.0%
5	76.8	79.8	3.9%
6	78.2	86.8	11.0%

The natural frequency results are given in Table 2. The Hand Calculation mode shapes were also verified with respect the FEA mode shapes.

References

1. T. Irvine, Vibration Analysis of an Isolated Mass with Six Degrees of Freedom, Revision C, Vibrationdata, 2005.