Introduction of Rotor Dynamics using Implicit Method in LS-DYNA®

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Abstract

Rotor dynamics is commonly used to analyze the behavior of structures ranging from jet engines and steam turbines to auto engines and computer disk drives. This paper introduces the implementation of rotor dynamics in LS-DYNA. The structural transient analysis and mode analysis and other applications, like brake squeal analysis, are presented and validated with other finite element software.

1. Introduction

Rotor dynamics is a study of vibration of rotating parts found in a wide range of equipment including engines, turbines, aircrafts, hard disk drives and more. The rotation of rigid bodies, mainly those with regular shapes, like shafts, disks and cylinders, has been well understood for several decades. However, for flexible bodies, especially those with irregular shapes, their rotational behavior requires more modern tools such as finite element method. Quite a lot studies ([1] and [2]) have presented the theoretical foundation of rotor dynamics. The finite element formulation has also been investigated by some researchers, such as J. Barlow [3].

On the other hand, automotive disk brake squeal has been a challenging issue for many years. Many researches have been done in this area in order to understand the brake squeal mechanisms. There are a lot of literature reviews on brake squeal ([4], [5], [6], [7], [8] and [9]) providing a comprehensive source of information. According to the mechanism of generation, brake squeal noise has a dominant frequency over 1kHz and is the result of friction-induced vibration. In numerical methods, there are two major approaches used in the industry: the complex eigenvalue analysis and the transient analysis.

This paper introduces the implementation of rotor dynamics in LS-DYNA. The structural vibration response and Campbell diagram, which shows the natural frequency changes with the change of rotating speed, are presented. An application using rotor dynamics is the brake squeal problem. By solving the complex eigenvalue problem, it can tell whether the friction instability occurs or not and the results are validated with another finite element software.

2. Implementation in LS-DYNA

Rotor dynamics can be analyzed in two coordinate systems: the rotating and fixed coordinate systems. The dynamic equation in rotating coordinate system can be written as:

$$[\mathrm{M}]\{\ddot{\mathrm{u}}\} + 2\Omega[\mathrm{C_c}]\{\dot{\mathrm{u}}\} + ([\mathrm{K}] - \Omega^2[\mathrm{K_c}])\{\mathrm{u}\} = \{\mathrm{F}\} \quad (1)$$

In fixed coordinate system, it is:

$$[\mathrm{M}]\{\ddot{\mathrm{u}}\} + \Omega[\mathrm{C_g}]\{\dot{\mathrm{u}}\} + [\mathrm{K}]\{\mathrm{u}\} = \{\mathrm{F}\} \quad (2)$$

where $[\mathrm{M}]$ and $[\mathrm{K}]$ are the system mass and stiffness matrices, respectively; Ω is the rotating speed. $2\Omega[\mathrm{C_c}]$ is the Coriolis matrix and will only show up in rotating coordinate system; $\Omega[\mathrm{C_g}]$ is the Gyroscopic matrix. $\Omega^2[\mathrm{K_c}]$ is the centrifugal stiffness matrix. The choice of the two coordinate systems can refer to Table 1.

Table 1 Choice of coordinate system

	Symmetric rotor	Unsymmetric rotor
Symmetric support	Both rotating and fixed	rotating
Unsysmmtric support	fixed	N.A.

A new keyword card *CONTROL_IMPLICIT_ROTATIONAL_DYNAMICS had been added to LS-DYNA. The parameters are listed in the following table:

Table 2 keyword card *CONTROL_IMPLICIT_ROTATIONAL_DYNAMICS

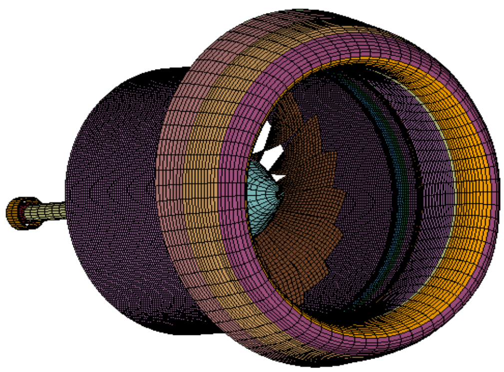
Card	1	2	3	4	5	6	7	8
Variable	SID	STYPE	OMEGA	VID	NOMEGA	IREF		
Type	I	I	F	I	I	I		
Variable	OMEG1	OMEG2	OMEG3	OMEG4	OMEG5	OMEG6	OMEG7	OMEG8
Type	F	F	F	F	F	F	F	F

In which, SID is the part or part set ID depending on set type (STYPE); OMEGA is the rotating speed; VID is the vector ID used to define the rotating axis, which can be defined by *DEFINE_VECTOR and *DEFINE_VECTOR_NODES; NOMEGA is the number of rotating speeds, which is used to automatically get eigenvalues with different rotating speeds, which are defined in second line with OMEGn; IREF indicates the choice of coordinate system.

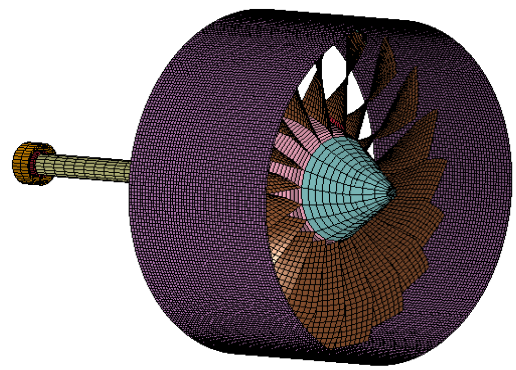
3. Rotor Dynamics Study in LS-DYNA

3.1 Transient study case: Fan Blade Off (FBO)

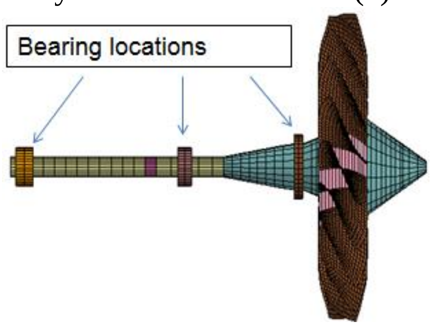
The rotor dynamics can be used to study the FBO problem in engine. At the initial stage, it is a very complicated high speed crush problem as one blade is off and contacting the containment. It is usually simulated by LS-DYNA explicit. After the blade is off, the rotating shaft will move away from the center due to the unbalanced centrifugal load. This procedure can take a long time and implicit method is more applicable. In this primary study, the FBO problem is analyzed in two stages: LS-DYNA explicit and implicit, and the models are given in Figure 1. It can be seen that the model in implicit is much simpler than that in explicit as the main focus in implicit is the shaft vibration under the unbalance load. Three bearings are used and their locations are indicated in Figure 1(3). The shaft center movement after one blade off using LS-DYNA explicit is given in Figure 2(1) and the shaft had rotated for 30 ms. After one blade is off, the bearing closest to the fan will fail due to a big bearing force. So in the implicit study, the first bearing is not included. The shaft center movement using implicit is given in Figure 2(2) and the shaft had rotated for 54 ms. It can be seen that the shaft center orbits are very similar, so it is applicable to use implicit rotor dynamics to study the shaft movement in FBO second stage.



(1) model in explicit analysis

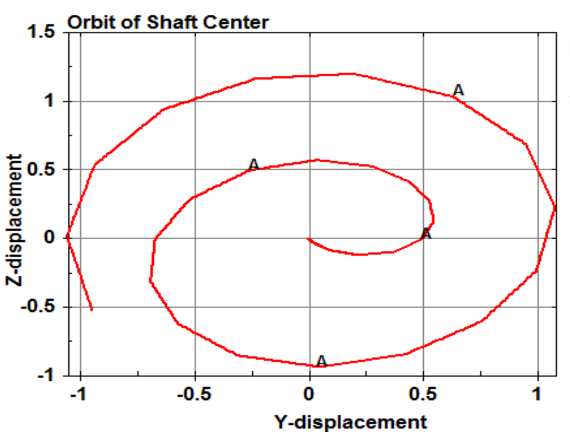


(2) model in implicit analysis

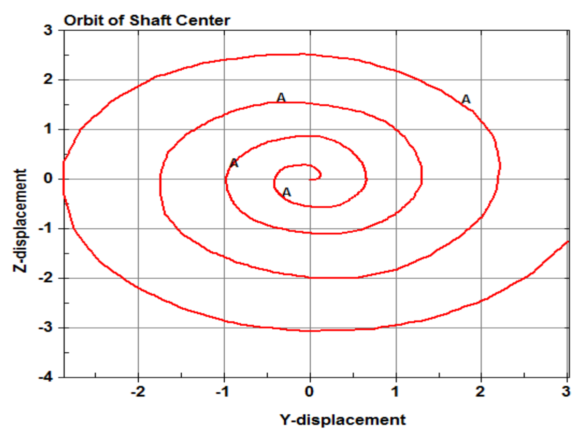


(3) bearings in the model

Figure 1 FBO model in explicit and implicit



(1) shaft center orbit in explicit



(2) shaft center orbit in implicit

Figure 2 Shaft center orbit in FBO problem

3.2 Modal Analysis

In rotor dynamics study, the mode frequencies often depend on the rotating speeds due to the gyroscopic damping and softening stiffness effects. The steady mode in static state may become unstable when the rotor begins to rotate at a certain speed, so it is very important to know how the frequency changes with the change of rotating speed (Campbell diagram). Figure 4 shows the Campbell diagrams of the first five mode frequencies using LS-DYNA and ANSYS for the disk model in Figure 3. The magnitudes of the frequencies and the mode splitting phenomenon are almost the same using these two software.

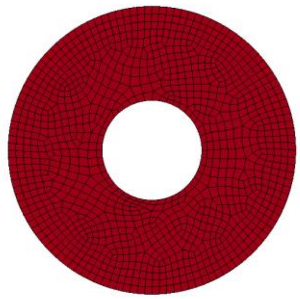
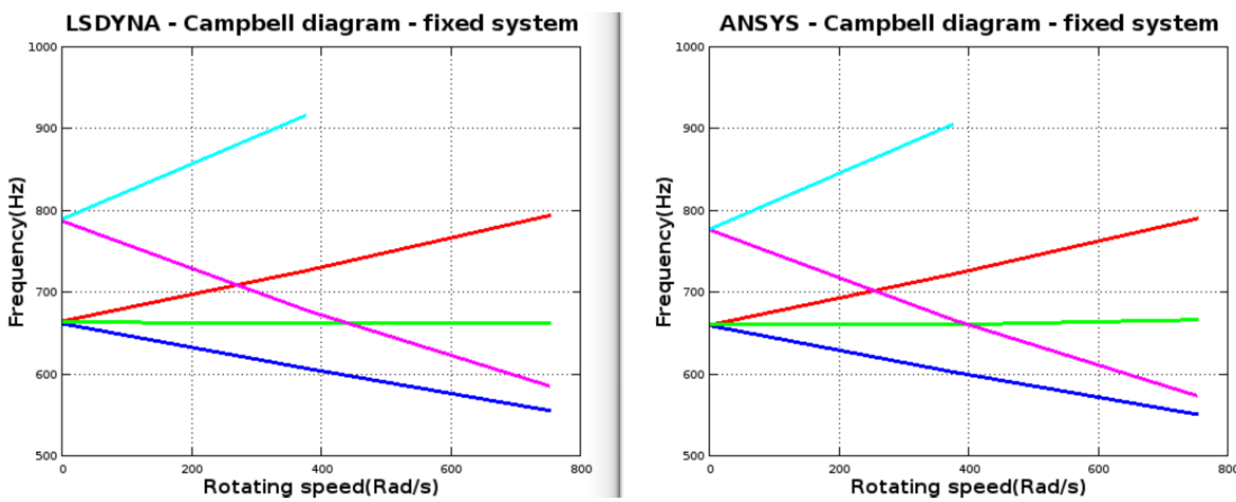


Figure 3 Disk model



(1) LS-DYNA (2) ANSYS

Figure 4 Campbell diagram for disk model (LS-DYNA and ANSYS)

3.3 One application: Brake Squeal Anslysis

Brake squeal noise is the result of friction-induced vibration. The complex eigenvalue analysis and the transient analysis are the two major numerical methods used in the automobile industry. The keyword *CONTROL_IMPLICIT_ROTAITONAL_DYNAMICS is used to define the brake disk. The contact between the pad and the disk is defined in *CONTACT_..._MOTOR, which introduces non-symmetry to the system stiffness matrix. In LS-DYNA, we use the intermittent eigenvalue analysis to combine the complex eigenvalue analysis and transient analysis.

Figure 5 gives the disk brake model in this study. The disk is constrained at its bolt holes. For the pads, the leading and trailing edges are constrained in both the circumferential and radial directions. The brake-line pressure is applied onto the back plates. The time history of the brake-line pressure and the disk rotating speed are used to describe the operating conditions of the disk brake model, as illustrated in Figure 6. At first stage, a pressure is applied gradually until time 0.01 and then it becomes constant, while the disk starts to rotate at time 0.01 and increases until time 0.02. Using the LS-DYNA intermittent eigenvalue analysis, eigenvalues can be extracted during the transient simulation, for example at time 0.02. By solving the complex eigenvalue problem, the real and image part of the eigenvalues are obtained. If the real part of the eigenvalue is positive, which indicates displacement explosion, we can say the disk brake instability appears. We define a parameter, damping ratio, as $-2*Re(\lambda) / |Im(\lambda)|$, where λ is the eigenvalue. When the real part of λ is positive, the damping ratio becomes negative. Therefore the disk brake instability can be told by the damping ratio. Figure 7 gives the damping ratio for the disk brake model in Figure 5. It indicates an instability at around 2 kHz from both LS-DYNA and another finite software.

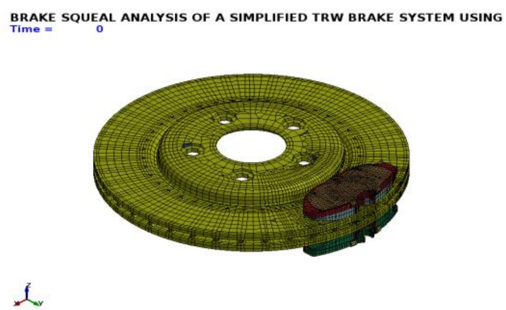


Figure 5 Disk brake model

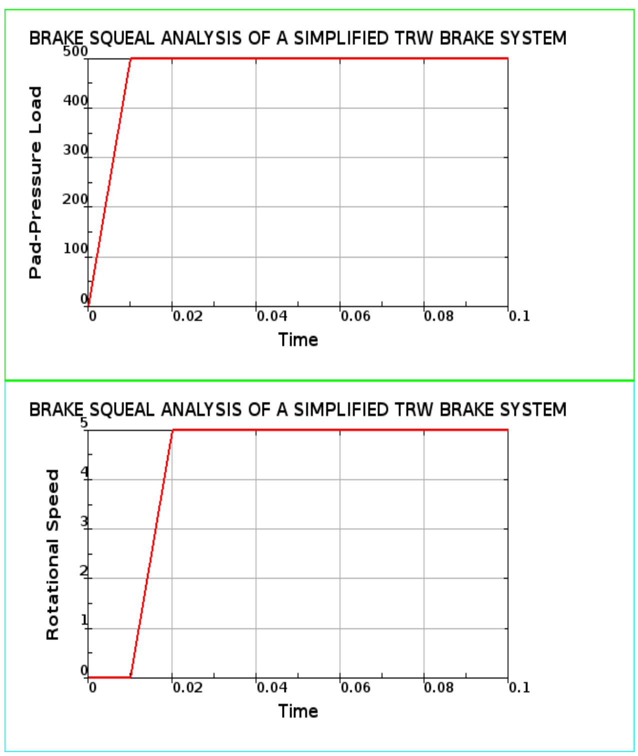
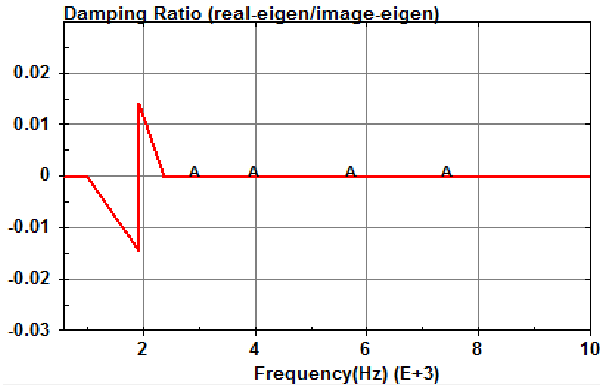
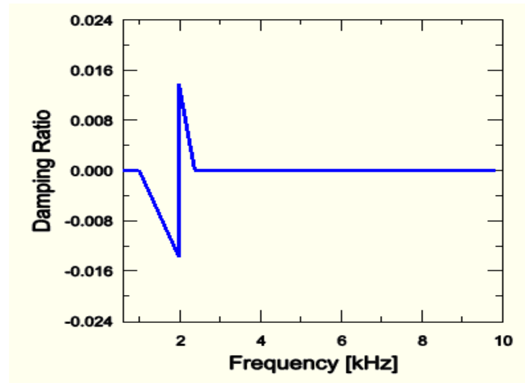


Figure 6 brake squeal transient analysis simulation procedure



(1) LS-DYNA



(2) another software

Figure 7 Damping ratio

4. Conclusion

A new keyword card *CONTROL_IMPLICIT_ROTATIONAL_DYNAMICS is added to LS-DYNA to do rotor dynamics analysis. In order to test this new feature, both the transient and mode analysis are taken and are verified with LS-DYNA explicit results or other finite element software results. Brake squeal problem is also analyzed by combining the transient and mode analysis and the result is compared with another finite element software result.

There are still many other studies need to be considered in LS-DYNA rotor dynamics, such as the bearings between the rotor and the stator, rotor dynamics analysis in frequency domain, etc.

5. References

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