Vibration Analysis of a Centrally Clamped Disk

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BIOGRAPHICAL NOTES

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KEY WORDS

Ansys, Polytec PSV400, FEA, EMA, Frequency volume source

ABSTRACT

This is the abstract section. In the study of modal analysis is it very important to understand the vibrational behavior of parts in order to reduce their contribution to radiated noise. Predicting vibration using today's simulation methods is still restricted by limited knowledge of boundary conditions and material parameters. Using an experimental modal analysis, modal parameters such as mode shapes, natural frequencies, modal damping can be measured. This measured data can be used as input data for numerical simulation. With correlation analysis, the results from finite element analysis (FEA) and experimental modal analysis (EMA) can be compared.

1. Experimental Modal Analysis

An experimental analysis was carried out in order to measure disk reference mode shapes, natural frequencies and damping. To be able to compare the results of a measurement with results of simulation, all measured data were exported to LMS Test lab. to post-processing.

1.1 Setup of the experiment

Setup of experimental modal analysis is shown in Figure 1. LMS MID frequency volume source (excitation range 200 Hz – 8000 Hz) with real time accurate volume acceleration signal was used for excitation. Distance between measured disk and excitation nozzle was 10mm (recommended distance range is from 9mm – to 30mm). Non-contact vibration measurements, visualization of mode shapes were performed by Polytec PSV 400 scanning vibrometer. Measured surface was scanned and probed automatically using measurement grid. Non-contact measurement eliminates influence of mass added to the disk.

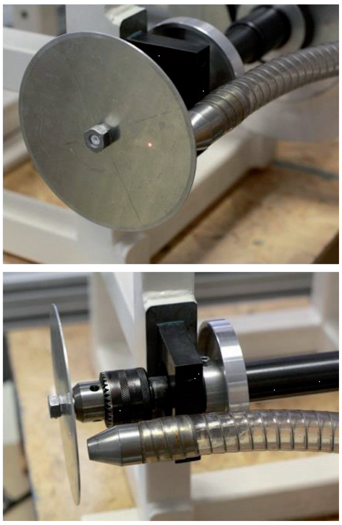

Fig. 1: *Setup of experimental modal analysis*

1.2 Results of EMA

All experimental modal parameters are obtained from measured operational deflection shapes (ODS). ODS was measured with Polytec PSV 400 scanning vibrometer. Each mode is defined by a natural (modal) frequency, modal damping, and a mode shape. Figure 2 shows the plots of mode shapes and natural frequencies.

2. Numerical Simulation of Disk in Stationary Condition

A parallel study using finite element analysis was carried out in order to provide a numerical check of clamped disk. The finite element program Ansys v.14 was used to compute the natural frequencies and modes. To improve FEM model, results

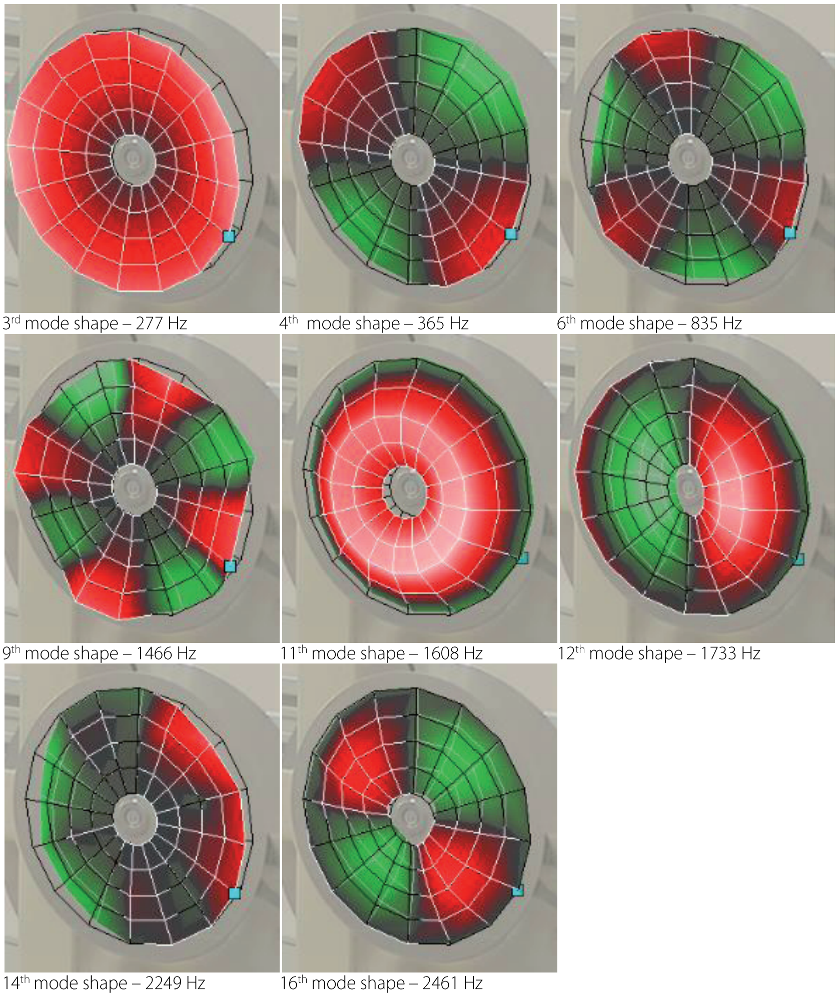

Fig. 2: *Operational deflection shapes and natural frequencies.*

from experimental modal analysis were used. The disk used in the experimental modal analysis was constrained by an internal hub (Figure 3). To eliminate model stiffening in finite element analysis, the hub was added. With fixed inner side of disk, the model was very "stiff". As a result of stiffening, the contact between hub and disk was added. In the finite element model, this contact was considered as bonded with normal stiffness factor 0.2. This setting eliminates "stiff" effect.

2.1 Boundary conditions and mesh used in analysis

Boundary conditions applied on the model: Fixed support (both surfaces of hub to eliminate very "stiff" model), Material of hub – structural steel, material of disk – aluminum alloy, Bonded contact with normal stiffness factor 0.2, Frequency range 5 – 2500 Hz.

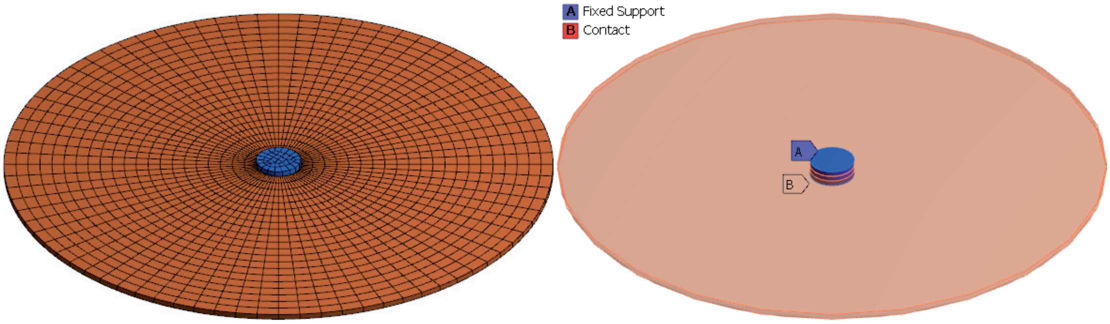

Fig. 3: *Mesh and boundary conditions*

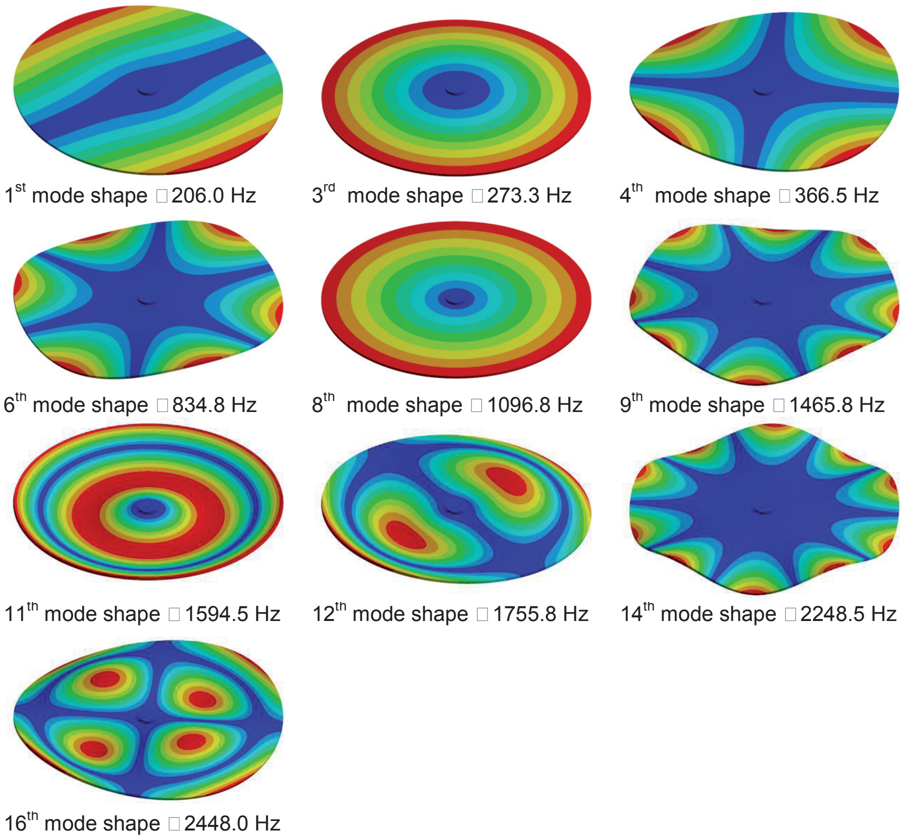

Fig. 4: *Mode shapes and natural frequencies.*

2.2 Results of FEA

Ten calculated mode shapes and frequencies of the centrally clamped disk of the stationary plate rigidly clamped to a pin are shown in Figure 4. The 1st and 2nd mode shape is characterized by a one nodal diameter through which the relative phase of displacement changes about 180º. The 3rd mode shape has nodal points at the center, which is rigidly fixed (umbrella mode). The 4th and 5th mode shapes have four point of maximum curvature with 90º phase displacement and two nodal diameters. In the case of 6th mode and 7th mode, there are three nodal diameters with six points of maximum. The 8th shape has nodal point at the center which is rigidly fixed and maximum displacement at outer diameter (radial mode). The 9th and 10th shape are characterized by a four nodal diameters with eight points of maximum. The 11th mode shape has nodal point at center. Maximum curvature region consist of two circular rings along. The 12th and 13th shape is the first buckled shape with one nodal diameter and two point of maximum. In case of 14th mode and 15th mode there are five nodal diameters with ten points of maximum. The 16th and 17th mode is the second buckled shape with two nodal diameters and four points of maximum.

3. Correlation Analysis

The modal assurance criteria (MAC) were used in selecting measured points. MAC evaluates the correlation between two different mode shapes. Experimental measured mode shapes and numerically calculated mode shapes . The following equation is used to evaluate MAC values.

$$MAC\{\psi_1,\psi_2\} = \frac{\left|\{\psi_1\}^T\{\psi_2\}\right|^2}{\left(\{\psi_1\}^T\{\psi_1\}\right)^T\left(\{\psi_2\}^T\{\psi_2\}\right)^T} \qquad (1)$$

In general a higher MAC value indicates better correlation between modes. Two mode shapes with 100% correlation represents perfect match. The measured mode shapes are compared with the numerically simulated mode shapes. The correlation results in MAC matrix form is presented in figure 5.

The following interpretation has been suggested for the MAC: a value less than 5% indicates non-correlated mode shapes and a value higher than 90% correlated mode shapes. In real life situation a MAC values below 50% may indicate poor correlation and values above 70% good correlation. Image 6 shows deformed mapped meshes (right side is measured, left side is numerically simulated) from MAC table.

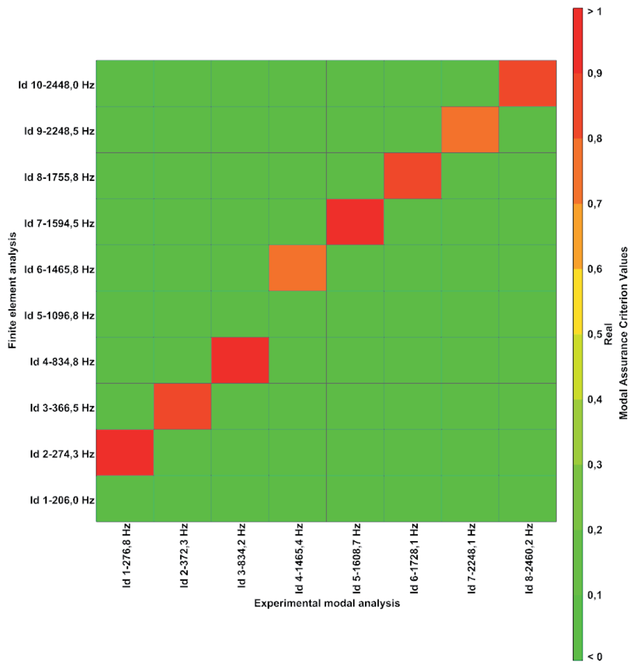

Fig. 5: *MAC matrix of measurement and simulation.*

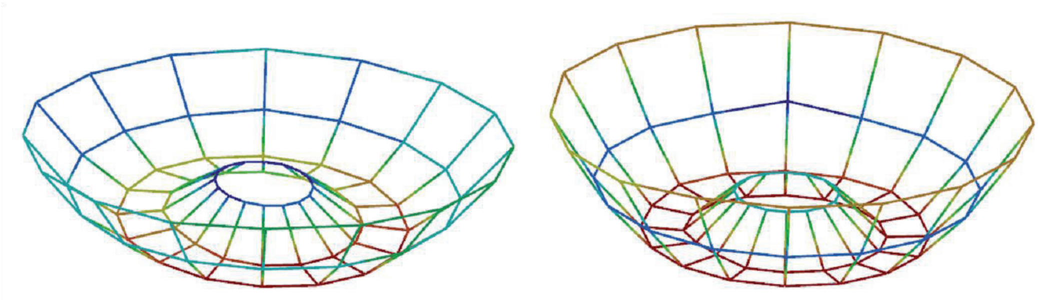

Fig. 6: *MAC Mode pair table results of 1594Hz and 1608 Hz*

4. Conclusion

Combination of non-contact scanning vibrometry and the numerical simulation is powerful method to predict vibrational behavior of structures. The measurement was made with zero influence of mass under operating conditions. Two mode shapes and frequencies which wouldn't have been visible with measurement were visible in numerical simulation. Contact stiffness between hub and disk changed only lower shapes of the disk. Modal shapes, where nodal diameter crosses the hub, were influenced by frequency. Summarized results of analysis are shown in Table 1. Errors of natural frequencies numerically computed and measured is under 2% and can be caused by non-precise disk production. The lowest value of correlation is 74%. This procedure can be used with complicated parts of machines to eliminate errors in numerical simulations.

Table 1: *Values of FEA, EMA, Correlation and Damping*

Mode	FEA [Hz]	EMA [Hz]	Error [%]	MAC [%]	EMA Damping [%]
1	206	-	-	-	-
2	273,3	276,8	1,3%	99%	0,2642%
3	366,5	372,3	1,6%	80%	0,0754%
4	834,8	834,2	-0,1%	97%	0,2370%
5	1096,8	-	-	-	-
6	1465,8	1465,4	0,0%	74%	0,0217%
7	1594,5	1608,7	0,9%	92%	0,5741%
8	1755,8	1728,1	-1,6%	88%	0,5390%
9	2248,5	2248,1	0,0%	75%	0,0153%
10	2448	2460,2	0,5%	80%	0,0412%

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