

Turbomachinery High Speed Modal Balancing: Modeling and Testing of Scale Rotors

Matteo Bertoneri and Paola Forte

Abstract The object of this work is the application of a modal balancing approach to turbomachinery rotors. The investigation was carried out on a small commercial rotor dynamic test bench suitably modified to represent in scale a typical turbomachinery balancing problem. Software based on literature theoretical formulations was developed for both the more widespread influence coefficient method and modal balancing. The modal balancing software was coupled with a finite element rotor dynamic code in order to determine the modal shapes necessary to impose the orthogonality conditions and calculate the modal calibration sets. Results of the two approaches were compared both in terms of balancing quality and in terms of time to balance the rotor. The test results highlighted essential information for an eventual true scale industrial application of the modal method and also interesting new aspects such as the possibility of balancing modes out of the operating speed range.

Keywords Modal balancing · Influence coefficients · Scale testing · Turbomachinery · Centrifugal compressors

1 Introduction

Since rotor unbalance and misalignment are one of the main causes of malfunctioning and failure of rotating machinery, during rotor manufacturing and assembly processes different balancing steps are currently provided for. In general, turbomachinery rotors are balanced first at low speed, then after any major component installation, and finally at high speed (HSB) [1, 2].

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HSB methods are based on two basic approaches: the influence coefficients (ICM), and the modal one (MBM). The main advantage of ICM that has made it so appealing and widespread in industry is that it is an entirely empirical method while MBM requires an accurate rotor dynamic model that introduces unavoidable modelling errors in the final result. However ICM may take a long time due to the fact that the number of calibrations is equal to the number of balancing planes (for centrifugal compressors it can also reach 5) and sometimes it is necessary to accept some compromise at the expense of final balancing quality. MBM instead requires less time because the number of calibrations is equal to the number of modes that influence the rotor response in the operating speed range (e.g. centrifugal compressors usually work between the first and the second critical speeds, and are affected at most by the third mode). It has a greater physical meaning compared to the ICM black-box approach, which is reflected in a higher quality final balancing, in particular at high and low speeds. Finally, one aspect that motivated this study promoted by GE Oil & Gas Nuovo Pignone, leading company in turbomachinery design, construction and installation, is the still limited application of MBM to turbomachinery rotors.

This work presents a preliminary experimental activity carried out on a reduced scale balancing bench aimed at:

- testing MBM on a configuration similar to a GE turbomachinery rotor;
- comparing MBM with ICM both in terms of effectiveness (balance quality) and efficiency (time needed to equilibrate the rotor);
- investigating the limits and potentiality of MBM for future industrial application.

2 Outline of Influence Coefficients and Modal Balancing Methods

In this section the two basic HSB methods will be briefly outlined in order to introduce our work. Excellent reviews have been published in the past years and can be referred to [3–5].

ICM has the only hypothesis of linearity of the rotor system. At a certain rotating speed, the dynamics of the rotor is considered as a black-box, where a transfer matrix $\mathbf{A}$, the influence coefficient matrix, relates the input vector of rotor unbalances ($\vec{u}$) to the output system response vector ($\vec{z}$),

$$\vec{z} = \mathbf{A}\vec{u} \quad (1)$$

The matrix $\mathbf{A}$ is identified by the calibration runs. The j -th run consists in adding a known mass c_j on the j -th balancing plane and in analyzing the output signal variation with respect to the original or baseline signal (Run 0). In this way one can calculate the j -th column of the influence matrix as

$$a_{ij} = \frac{z_{ij} - z_{i0}}{c_j} \quad (2)$$

where $z_{ij} - z_{i0}$ is the vibration phasor variation with respect to Run 0. Therefore the method requires as many calibrations as many are the balancing planes. Once the influence coefficient matrix is known, the balanced solution can be found by calculating the correction to be applied $\vec{b}$ to neutralize the initial unknown unbalance $\vec{u}$

$$\vec{u} + \vec{b} = \vec{0} \xrightarrow{\text{yields}} \vec{b} = -\vec{u} = -\mathbf{A}^{-1}\vec{z} \quad (3)$$

Since the response must lie within a tolerance limit over the entire speed range of interest, the transfer matrix depends on the rotating speed as well as the response vector, while the unknown initial unbalance vector does not. Therefore the problem is highly over constrained and is solved by the usual weighted minimization techniques:

$$\vec{b} = -\left(\hat{\mathbf{A}}^T \mathbf{W} \hat{\mathbf{A}}\right)^{-1} \hat{\mathbf{A}}^T \mathbf{W} \vec{Z} \quad (4)$$

where $\vec{Z}$ is a column vector collecting the responses at the different speed values $\vec{z}(\Omega_1)$, $\vec{z}(\Omega_2)$... while $\hat{\mathbf{A}}$ is a matrix composed by the matrices of the influence coefficients $\mathbf{A}(\Omega_1)$, $\mathbf{A}(\Omega_2)$... arranged in a column and $\mathbf{W}$ is a matrix of weighting factors.

One can say that the ICM aims at lowering the unbalance response by means of a process of identification and minimization of the transfer matrix for different rotating speeds. This approach can be quite long in the case of many balancing planes and also it can happen that a satisfactory balancing quality is not achieved in the whole speed range of interest (a solution obtained at high operating speeds could be unacceptable at low speeds).

On the other hand the modal approach is based on the modal superposition principle: the rotor response can be considered as the sum of a certain number of modal responses. The MBM is based on the idea of balancing each single modal response in order to reduce the overall response within tolerance. To do that it is necessary to excite the system with sets of masses (not just one per plane) that excite selectively each mode of interest in the operating range. To calculate the modal calibration sets it is necessary to impose the orthogonality condition with respect to the rotor modal shapes so that the h -th set will excite only the h -th mode. Due to the experimental difficulty in having sufficient measurement points, the mode shapes are calculated using a finite element (FE) model of the rotor and the supports. After each modal calibration one can correct the relative modal unbalance, using a sort of modal influence coefficient, getting the h -th modal correction $\vec{b}_h$. The modal correction should be calculated in the vicinity of the h -th critical speed, where the deformation of the rotor takes the corresponding modal shape and therefore maximizes the Lagrangian component of the active forces generated by

the modal calibration set. Once the solutions for the single modes $\vec{b}_h$ are calculated, the final correction will be obtained as the sum of the different $\vec{b}_h$, extending the sum to the modes with significant modal participation in the Run 0 response.

Referring to the theory described above, a balancing software was developed in MATLAB environment to calculate the correction vector on the basis of the rotor response variations following calibrations. In particular, the modal balancing software was interfaced with the FE rotordynamic code DYNROT [6] that was employed to export the mode shapes, necessary to impose the orthogonality conditions and calculate the modal calibration set.

3 Experimental Setup

The experimental activity was carried out at the University of Pisa on Bently Nevada Rotor Kit test bench (Fig. 1) that can simulate several rotating machinery malfunctions, including unbalance response. It is supplied with a slender stainless steel shaft (10 mm diameter h6 tolerance, 560 mm long, 0.001" TIR straightness tolerance). Two discs of 0.8 kg each (562.5 kg mm² polar moment of inertia, 323 kg mm² moment of inertia about diameter) are provided to be fixed on the shaft. For this work an additional disc was included in the set in order to show more modes in the operating speed range. These discs have threaded holes at a radius of

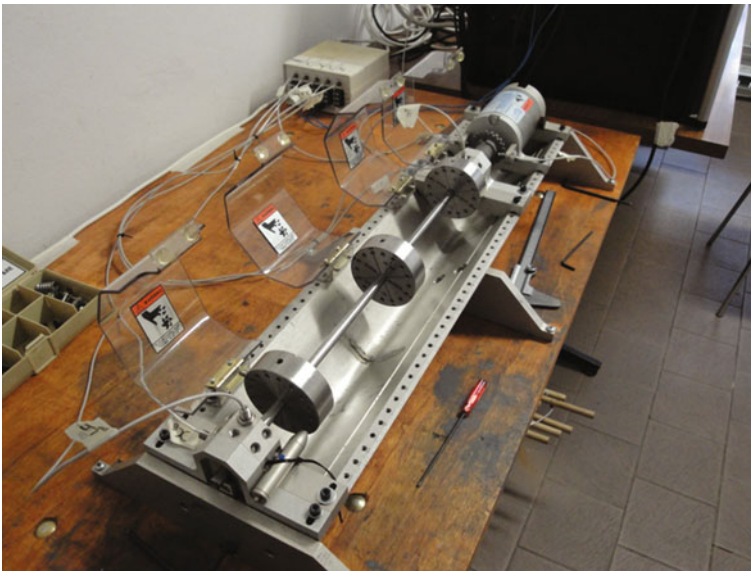


Fig. 1 The Rotor Kit test bench

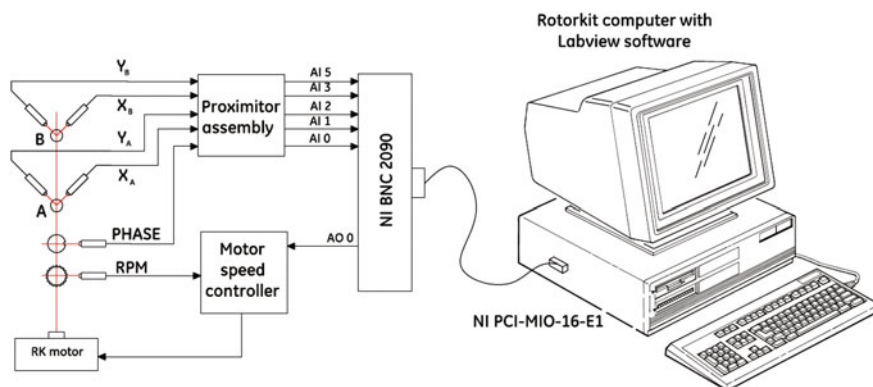


Fig. 2 Schematic drawing of the data acquisition and control system

30 mm in order to create the desired unbalance by inserting screw weights, ranging from 0.1 to 2 g.

The shaft, supported by two oilite bearings and driven by a speed controlled DC motor, can reach a maximum rotational speed of 10,000 rpm. O-rings are interposed between the bearings and the pedestals. Since their flexibility affects the rotor response, they were characterized by tuning the calculated values of the critical speed with the experimental results.

Figure 2 shows the block diagram of the data acquisition and control system. The bench is equipped with six proximity eddy current probes, four of which measure shaft displacement in two locations near the supports and two orthogonal directions (as in the real industrial balancing facility) while the other two probes measure the rotor speed and give a trigger reference respectively. The signal is sampled, digitized and collected by a PC data acquisition board used also to control the engine speed, through a LabView code. The raw data are averaged, filtered and transformed in frequency, using the complex Fourier transform. Only the positive synchronous harmonic of the frequency spectrum, that is the $1\times$ forward component, is exported to a text file since it is the only component that balancing is able to correct. The text file is then post-processed in MATLAB obtaining the displacement amplitude and phase as functions of the rotational speed.

An issue that has to be faced when operating on scale tests is the extendibility of the results to true scale, in this case GE turbomachine rotors. In particular, reference was made to some centrifugal compressors with the third mode very close to the operating range, trying to reproduce their rotordynamic behavior in the Rotor Kit scale tests. Since the masses and the support stiffness are given, the only design variables that influence the rotordynamic behavior are the positions of discs and supports along the shaft. An array $\vec{x} \in N^6$ of the geometric variables, directly related to the FEM model nodes, was defined as and an iterative process of constrained optimization was set according to the diagram of Fig. 3.

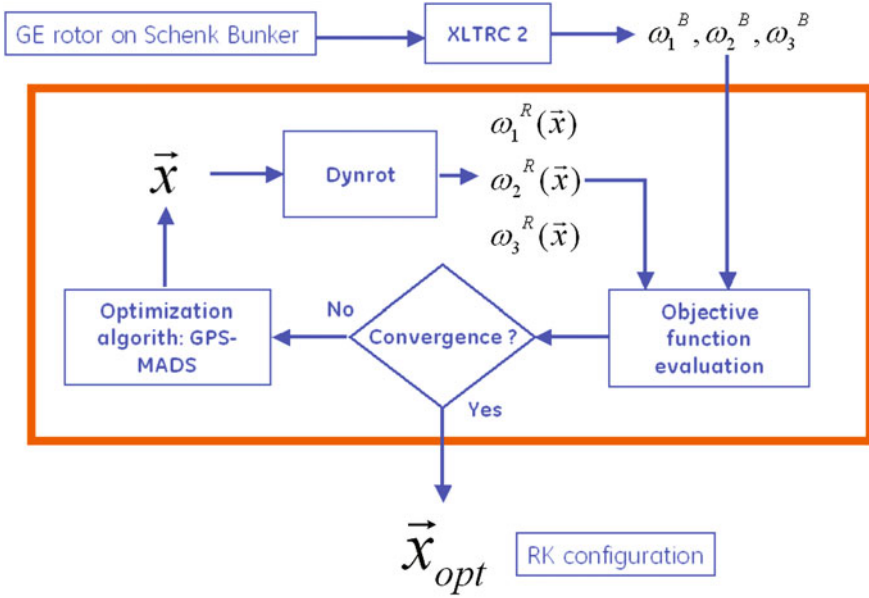


Fig. 3 Flow-chart of the optimization process for Rotor Kit optimum configuration

Given the first three critical speeds of a real GE rotor to be balanced ω^B and a geometrical configuration of the Rotor Kit $\vec{x}$, the FEM software DYNROT calculates the Rotor Kit critical speeds $\omega^R(\vec{x})$. Then the objective function, defined as a weighted average difference of the critical speeds reduced to the maximum corresponding rotor speed in order to maintain a mutual similarity of the rotordynamic influence of the various modes,

$$f(\vec{x}) = \frac{1}{3} \left(\epsilon_1 \left| \frac{\omega_1^R(\vec{x})}{\omega_{max}^R} - \frac{\omega_1^B}{\omega_{max}^B} \right| + \epsilon_2 \left| \frac{\omega_2^R(\vec{x})}{\omega_{max}^R} - \frac{\omega_2^B}{\omega_{max}^B} \right| + \epsilon_3 \left| \frac{\omega_3^R(\vec{x})}{\omega_{max}^R} - \frac{\omega_3^B}{\omega_{max}^B} \right| \right) \quad (5)$$

is evaluated.

If this difference is large a MADS algorithm for constrained optimization, available in a MATLAB toolbox, is applied to calculate the new configuration $\vec{x}$, until convergence is reached. The constraints are related to conditions of non-interference between rotor components and the avoidance of cantilever configurations. This method of optimization gave excellent results, with an error of 7 %. Figure 4 shows the Rotor Kit configuration obtained by optimization and used for balancing testing, which represents the scaled rotor of a multistage centrifugal compressor.

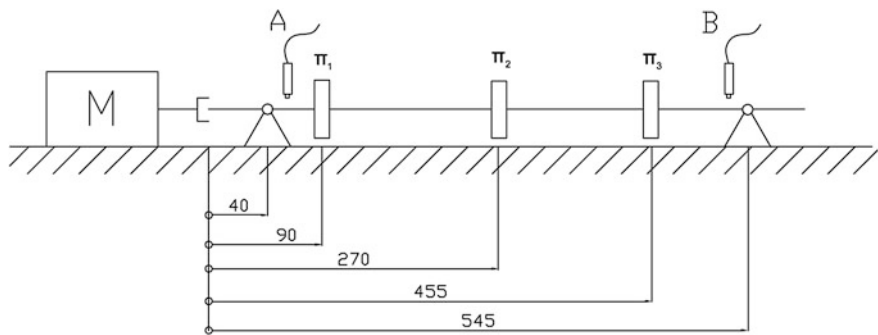


Fig. 4 Rotor Kit configuration obtained by optimization for rotordynamic similarity

4 Experimental Results

Some of the more significant experimental results are reported in Figs. 5, 6, 7 and 8. For the optimum Rotor Kit configuration, Run 0 results indicate that critical speeds are at 1600, 4400 and 6600 rpm.

First of all, classical ICM was tested. Table 1 shows the applied calibration sets. According to the ICM theory, calibration runs follow the one weight-one plane approach. In the Rotor Kit configurations, balancing planes are coincident with

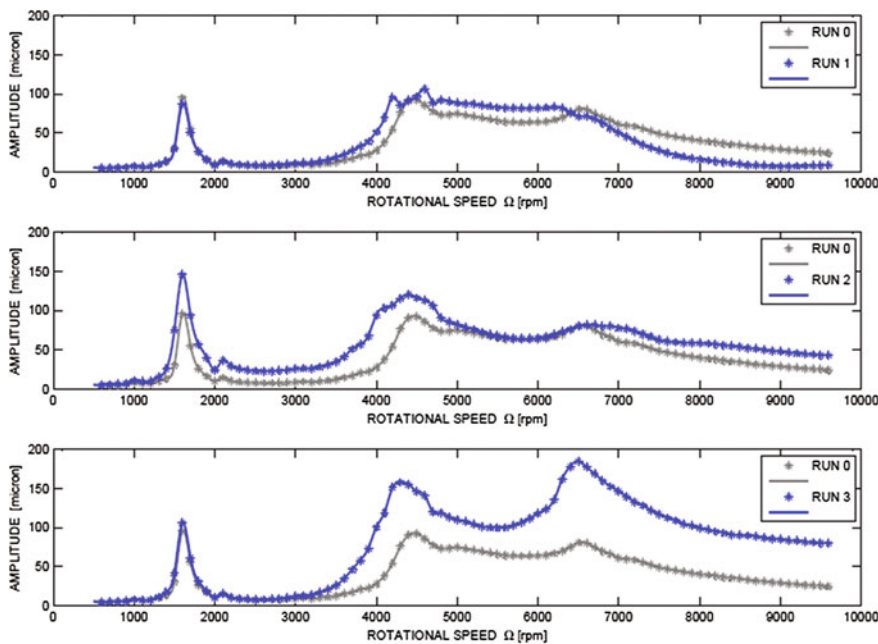


Fig. 5 Calibration effect of ICM compared to Run 0

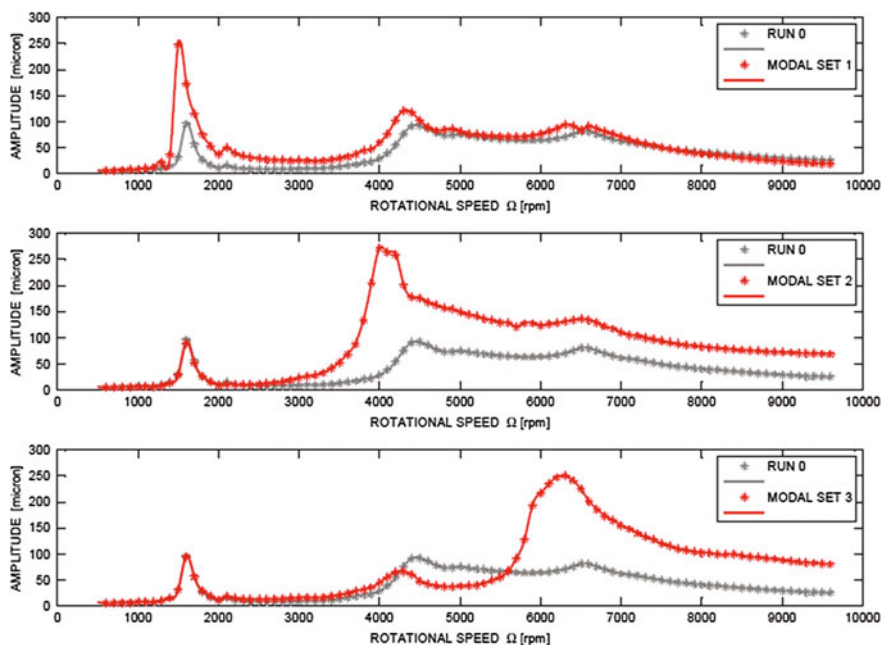


Fig. 6 Calibration effect of MBM compared to Run 0

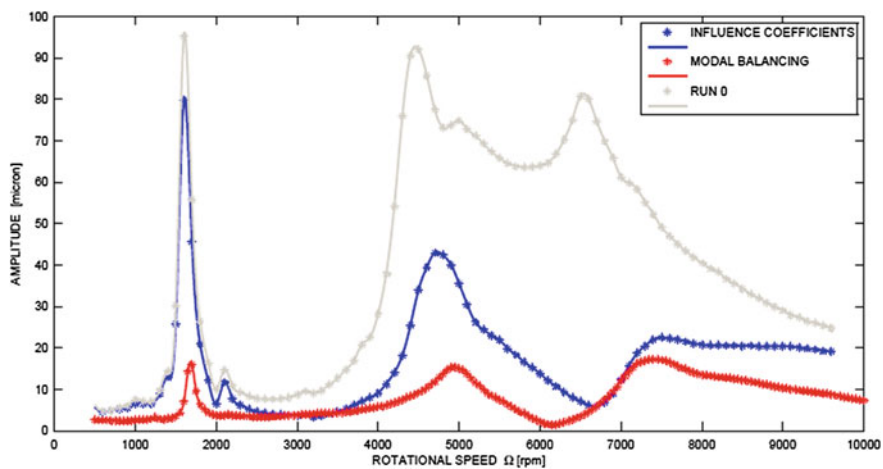


Fig. 7 ICM and MBM solutions compared to Run 0

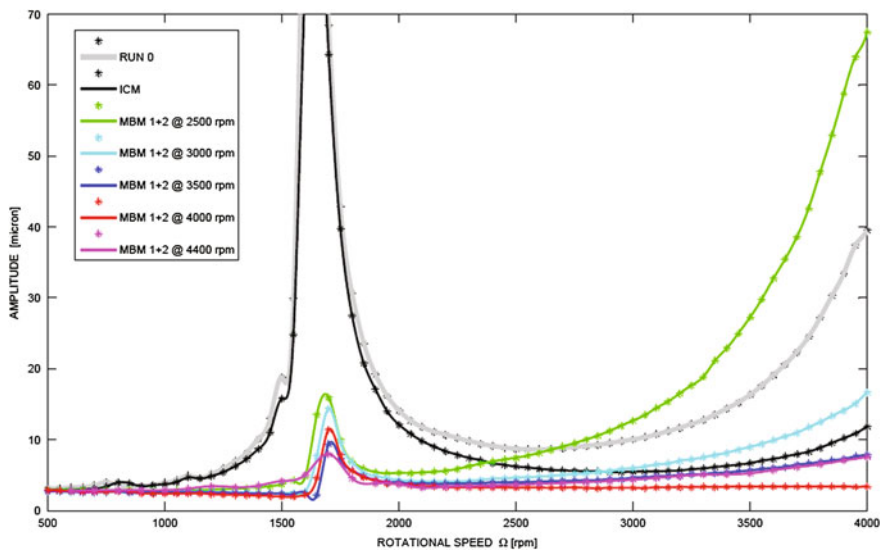


Fig. 8 ICM and MBM solutions compared to Run 0

Table 1 ICM calibration weight sets

Calibration	π_1	π_2	π_3
Run 1	1.0 g @ 45°	–	–
Run 2	–	1.0 g @ 90°	–
Run 3	–	–	1.0 g @ 270°

mass locations, and this is an intrinsic limitation because correction planes and the biggest sources of unbalance are axially coincident. Nevertheless, this is also true in centrifugal compressors with impellers.

Figure 5 shows the effects of ICM calibrations, compared to Run 0. It is noteworthy that the ICM calibration excites simultaneously all modes and that the amplification factor for the first mode is greater in the second calibration because the corresponding plane is at midspan.

Table 2 lists the modal calibration runs, calculated with the MATLAB balancing software developed respecting orthogonality conditions to the first three modal shapes. Figure 6 shows the clear effects of single mode excitation of MBM calibrations compared to Run 0.

Table 2 MBM calibration weight sets

Calibration	π_1	π_2	π_3
Mode 1	0.40 g @ 112.5°	1.00 g @ 112.5°	0.50 g @ 112.5°
Mode 2	1.00 g @ 292.5°	0.20 g @ 292.5°	1.00 g @ 112.5°
Mode 3	0.80 g @ 45°	0.74 g @ 225°	0.78 g @ 45°

Table 3 MBM and ICM solution weight sets

Solution	π_1	π_2	π_3
MBM 1 + 2 + 3	0.88 g @ 280°	0.59 g @ 262.5°	0.39 g @ 33.8°
ICM	0.49 g @ 265.6°	0.30 g @ 135°	0.20 g @ 123.8°

After modal calibrations, three modal solutions were calculated considering only one mode at a time. Each mode was balanced very well with negligible effect on the others, especially as the mode was less damped. Finally MBM complete solution was calculated and compared with ICM solution. Table 3 lists the solution weight sets.

Figure 7 presents a comparison between the two methods showing the superiority of the modal method, especially in balancing lightly damped modes. Furthermore, with the modal method, it is possible to select the modes to consider in the calculation, getting a further gain in terms of time. For example, if the curve of the initial response shows a strong modal participation in two of the three modes, it is possible to calibrate only these two modes, saving the calibration of the second. Therefore, MBM is potentially faster than traditional ICM. The former conclusion is quite important for the industrialization of the method because HSB is a time consuming operation in the rotor assembly phase of turbomachinery manufacturers' shops.

Another aspect that has emerged from these tests is the possibility, with the modal method, of balancing modes out of the operating range (according to theory it would be impossible to calculate the solution) with a greater residual unbalance the farther the critical speed of the mode is with respect to the maximum speed at which the solution is calculated. This last aspect, in addition to being innovative compared to the results known in the literature, is of particular importance for the industrial application of the balancing method to centrifugal compressors. In fact most centrifugal compressor rotors work between the first and the second mode, therefore the second mode is outside their speed range. As an example Fig. 8 shows the effect of the rotational speed used to balance the second mode: it is clear that an acceptable solution could be reached using values well below the second critical speed.

5 Conclusions

This paper has presented the experimental comparison between Modal Balancing and classical Influence Coefficients methods carried out on a scaled turbomachinery rotor. MBM showed its selective mode excitation and lower residual unbalances compared to ICM's, especially for lightly damped modes. Moreover it showed the capability to take into account out of speed range modes and to balance a mode using experimental points that are not near a critical speed. Therefore, the main conclusion of the Rotor Kit experimental campaign is that the industrial application of MBM in order to balance centrifugal compressors is feasible and could be useful also in those cases where the maximum steady speed is below the second critical speed.

Rotor Kit experimental results were consistent with preliminary tests carried out on GE high speed balancing facility to screen the variables of MBM (such as number of planes, number of modes to be taken into account, maximum speed). MBM has been successfully introduced in GE Oil & Gas, and it was recently helpful to balance some centrifugal compressor rotors operating above the second critical speed that had shown problems with the standard ICM method.

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