Development and preliminary validation of a new strategy to model the interaction between rotating machines and elastic supporting structure

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Abstract

The accurate modelling of the complicated dynamic phenomena characterizing rotors and support structures represents a critical issue in rotor dynamics field. A correct prediction of the whole system behavior is fundamental to identify safe operating conditions and to avoid instabilities that may lead to erroneous project solutions or possible unwanted consequences for the plant.

Although a generic rotating machine is mainly composed by four components (rotors, bearings, stators and supporting structures), many research activities are often more focused on single components rather than on the whole system.

The importance of a combined analysis of rotors and elastic supporting structures arises with the continuous development of turbo machinery applications, in particular in the Oil & Gas field where a wide variety of structurally optimized solutions with reduced weight on off-shore installations or modularized turbo-compression and turbo-generator trains, requires a more complete analysis not only limited to the rotor-bearing system.

Complex elastic systems, in some cases, strongly affect the entire shaft line rotor dynamic response such as mode shapes, resonance frequencies, unbalance response and critical speeds.

adfa, p. 1, 2011.

The aim of the study is a development of a new efficient methodology based on FEM approach to model the complete rotating machinery systems (rotors, bearings, stators and supporting structures), by means of appropriate transfer functions matrix.

Taking advantage from the matrix of transfer functions H(ω) obtained through PSD analysis, the baseplate dynamic behavior can be timely and CPU efficiently computed, avoiding computationally expensive harmonic sweeps.

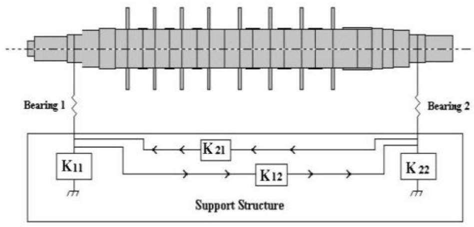
The appropriate usage of undocumented ANSYS command 'TFUN' has been pursued in order to extract the required components of the transfer functions matrix at the bearing location. With such a solution the full dynamic interaction between the system components was accurately accounted.

The outcome of the new methodology was successfully tested in a real field issue where evidences of structure to rotor interaction emerged at the proximity probe measurement during machine start-up.

Keywords: Rotating machines, rotors, supporting structures, bearings

1 Introduction

This research activity aims to study how baseplates cause coupling transfer functions [1] between different bearings and the unavoidable need of the inclusion of this component in the analysis since project preliminary stages. There is clearly a limitation in the simplification of the supporting structure model and its effect on the dynamic of the system. As visible in Eq.1, in the stiffness transfer function Tf, the cross talking terms (K_{12}, K_{21}) link the DOFs of different bearings, as reported on Fig. 1, effecting the final dynamic response of the system.



$$Tf = \begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix} \qquad (1)$$

Fig. 1. Direct and cross talk terms

The appearance of cross talk terms deeply modify the response of the rotor [2], if compared to the classical approaches where $K_{12} \equiv K_{21} \equiv 0$ and the direct terms K_{11} and K_{22} act as a series of springs with the characteristic of bearings. In particular the influence of cross talk terms become more interesting in the second rocking or flexible mode of the rotor typical of the off-shore installation due to the isostatic anchoring system and to the deck flexibility.

The integration of the supporting structure dynamics may lead to the following:

- The system response highlights the modes of each individual component of the assembly;

- The rotor modes shift in frequency without modification of the deformed shapes;
- The rotor keeps almost constant modes frequency but with different shapes.

As mentioned in [3], when the stiffness ratio $K_{support}/K_{brg} \leq 3.5$, the support flexibility begins to have a significant influence on the system's critical speeds and response characteristics. Such ratio is clearly a guideline to guarantee a good behavior in the operating range; however the structural dynamic response is way from being constant in the machine speed range and a more deep characterization may be needed.
The general architecture (see Fig. 2) of this study highlights the mutual interaction between the three main components: rotor, bearings and support. The flow of local variables (forces, torques and kinematic variables) shows the primary importance of bearings as filter element interposed between the baseplate and the rotor.
A scheme of the general architecture is visible in Fig. 2.

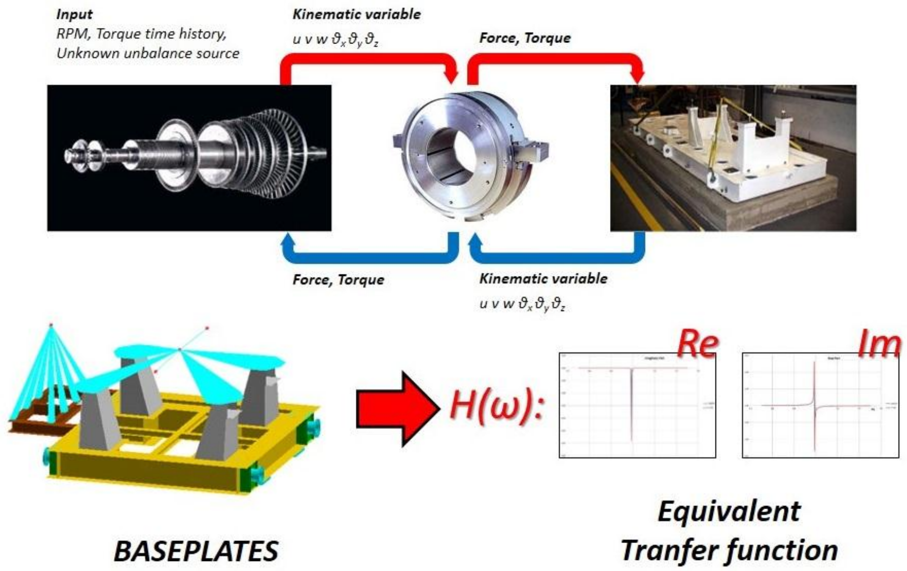


Fig. 2. General architecture

The main interesting point of discussion is the substitution of the complete FEM assembly that represent the elastic support structure, with an equivalent frequency dependent transfer function able to accurately represent the original system.
The model is implemented using one of the more common FEM software, Ansys.

2 The model

Following what specified in [1], the dynamic behavior of a coupled system (made up of several submodels) can be evaluated by calculating the dynamic contribution of each submodel and then recoupling them through the congruence equations. In particular, the FBR method (FRF based substructuring), predicts a coupled system's dynamic behavior on the base of transfer functions (matrix [H]) of the free boundary surfaces of uncoupled components, and the possible stiffness combination between the two subsystems. Therefore the dynamic stiffness of the base support can be evaluated without modeling the rotor, keeping the interface nodes free between the base and the rotor and evaluating the transfer function among excitation and coupling points.

The standard methodology involves a set of frequency sweeps (a series of harmonic analysis with full or mode super-position solver) in order to evaluate the transfer function matrix and then get the dynamic stiffness of the frame for a linear and time-independent system; in particular, indicating as 1 and 2 the interface nodes between the rotor and the base support (Fig. 3), a sinusoidal unit force is applied for every DOF of the interface nodes, evaluating in each case the displacements of all the DOF of nodes 1 and 2; the procedure is repeated varying the frequency (ω), obtaining in this way the matrix of transfer functions, function of ω.

For this reason, in order to build the matrix of the transfer functions, this procedure needs to run six harmonic sweeps, which involve extremely long computational times.

2.1 Transfer Function model

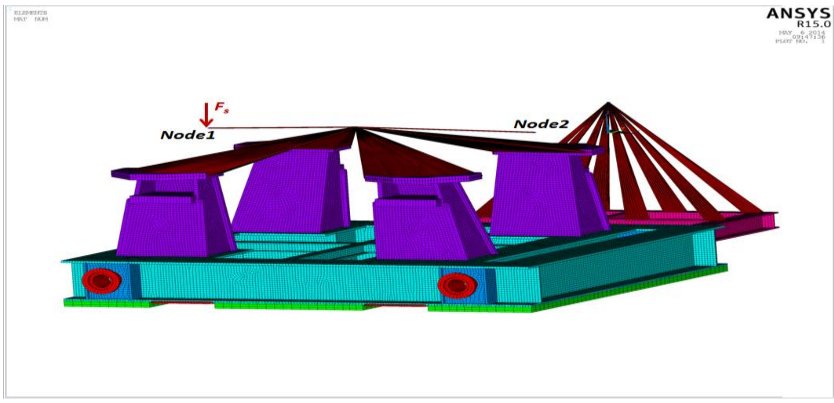


Fig. 3. Baseplate system

The base support system (Fig. 3), discretized through a FEM model, is represented by a linear and time-independent system with N degrees of freedom; applying a sinusoidal force (F) on the degree of freedom s, the characteristic dynamic equation at a given frequency is Eq.2

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}=\{F\}e^{j\omega t} \tag{2}$$

where [M] is the mass matrix, [C] is the damping matrix and [K] is the stiffness one.

The displacement of the DOFs for a linear and time-independent system, under a sinusoidal load, can be written as in Eq.3

$$\left\{u(t)\right\} = \sum \left\{\phi_i\right\} \alpha_i(t) = [\phi]\{\alpha\} = [\phi]\{\alpha\} = [\phi] \tag{3}$$

where u(t) represents the nodal displacement vector, while z_0, in general, is a complex number containing the information regarding amplitude and phase of every modal oscillator excited at ω frequency, and ϕ represents the generic eigenvector.

Replacing Eq.3 in Eq. 2 the result is Eq.4

$$-\omega^2 [M][\phi]\{z_0\} + j\omega[C][\phi]\{z_0\} + [K][\phi]\{z_0\} = \{F\}\,. \tag{4}$$

Multiplying every part of the previous equation for $[\phi]^T$, and assuming that the damping [C] is expressed in the Rayleigh form Eq.5, which allows to uncouple the equations (Eq.6), the results is

$$[C] = \alpha[M] + \beta[K] \tag{5}$$

$$\left(\left[-\omega^2\right] + j\omega\left[2\xi_n\omega_n\right] + \left[\omega_n^2\right]\right)\{z_0\} = [\phi]^T \{F\} \tag{6}$$

where

$$\left[-\omega^2\right] = \begin{bmatrix} -\omega^2 & 0 & 0 \\ 0 & -\omega^2 & 0 \\ \cdots & \cdots & \ddots \end{bmatrix} \tag{7}$$

$$\left[2\xi_n\omega_n\right] = \begin{bmatrix} 2\xi_n\omega_n & 0 & 0 \\ 0 & 2\xi_n\omega_n & 0 \\ \cdots & \cdots & \ddots \end{bmatrix} \tag{8}$$

$$\left[\omega_n^2\right] = \begin{bmatrix} \omega_1^2 & 0 & 0 \\ 0 & \omega_2^2 & 0 \\ \cdots & \cdots & \ddots \end{bmatrix}. \tag{9}$$

The generic component n-th of $\{z_0\}$ is

$$\left(z_0\right)_n = \frac{\sum_{k=1}^{N} \phi_{nk}^T F_k}{-\omega^2 + \omega_n^2 + j2\xi_n\omega_n\omega} = \frac{\sum_{k=1}^{N} \phi_{kn} F_k}{\omega^2 - \omega_n^2 + j2\xi_n\omega_n\omega}\,. \tag{10}$$

If only one load component is imposed on the degree of freedom s it becomes

$$z_{0ns} = \frac{\phi_{sn} F_s}{\omega_n^2 - \omega^2 + i2\xi_n\omega_n\omega}\,. \tag{11}$$

As a consequence, the transfer function of the r-th DOF, applying only one load component on the s-th DOF at a given frequency ω, is equal to

$$\begin{aligned} u_{rs}(t) &= \frac{\phi_{ri}\phi_{si}F_s e^{j\omega t}}{\omega_i^2 - \omega^2 + j2\xi_i\omega_i\omega} = \\ &= \left(\sum_{i=1}^{N} \frac{\phi_{ri}\phi_{si}F_s e^{j\omega t}}{\omega_i^2 - \omega^2 + j2\xi_i\omega_i\omega} \right) F_s e^{j\omega t} \end{aligned} \quad . \tag{12}$$

From this result the transfer function for the r-th DOF is extracted under the action of a load component applied on the s-th DOF

$$H_{rs}(\omega) = \sum_{i=1}^{N} \frac{\phi_{ri}\phi_{si}}{\omega_i^2 - \omega^2 + j2\xi_i\omega_i\omega} \tag{13}$$

where $H_{rs}(\omega)$ is matrix of transfer functions varying the frequencies, ϕ_i is the generic eigenvector at i-th frequency and ω the i-th resonance frequency.

At the end of these considerations, according with the Power Spectral Density (PSD) analysis theory, an innovative procedure has been implemented in order to quickly extrapolate the transfer functions' matrix.

2.2 Use of PSD analysis and TFUN

As shown in Eq.14, in the PSD analyses the software automatically couples the transfer functions matrix (as in Eq.13) following the results obtained from the previously done modal analysis

$$RPSD = \left[H(\omega)\right] PSD \left[H(\omega)\right]^{*T} \tag{14}$$

where $H(\omega)$ is the matrix of the transfer functions, PSD the input of the Power Spectral Density analysis, and RPSD the system response due to the excitation.

It appears therefore clear that the transfer functions matrix is known once the ω_i (system resonance frequencies) and the ϕ_i (eigen vector i-th at ω_i frequency) are calculated, once the modal analysis has been solved. As the $H(\omega)$ matrix is assembled during the run of a power spectral density simulation, an in-depth study has been conducted in order to find a procedure to extract the above-mentioned matrix directly inside ANSYS environment.

In particular a large number of numerical tests regarding the not documented TFUN (transfer function) ANSYS command have been executed, in order to verify first of all if this command allows to extract correctly the components of stiffness matrix, comparing them with those obtained through sweeps in frequency and, secondly, to verify if the use of this strategy can lead to a drastic reduction of computational costs.

3 Test and results

Several tests have been conducted on a typical centrifugal compressor steel structure support for on shore installation with the aim of validating the described procedure. A detailed ANSYS FEM model was built using SHELL63 elements (3D Elastic Shell; ~82000 nodes) and SOLID45 (3D-structural solid; ~18000 nodes), for a total of approximately 100000 nodes and 260000 DOF (Fig.4).

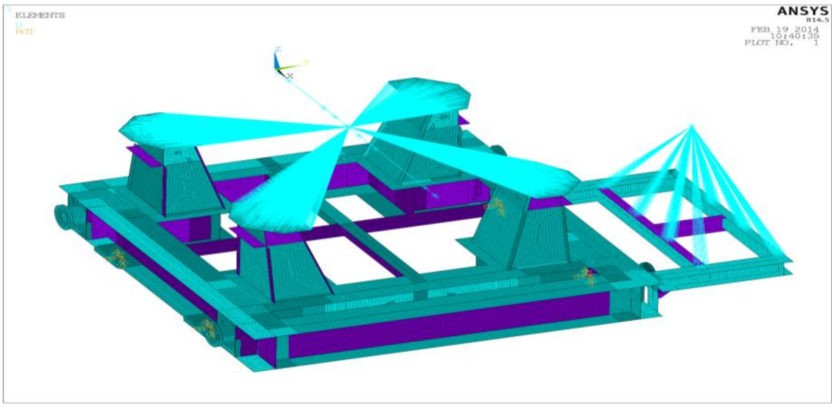


Fig. 4. Baseplate

As critical high shaft vibrations were detected during field operation during the compressor start up phase, there has been the opportunity to collect a great number of vibration data both on the structure and the shaft proximity probes to validate the computational results. One can see in **Fig. 5** how the dynamic response of the rotor only, as considered not linked to the supporting structure, can change when the steel structure is added and even more if compared with the coupled analysis of the whole system before and after a concrete reinforcement is poured under the baseplate to strenghten the anchoring system (No Grouting / Grouting).

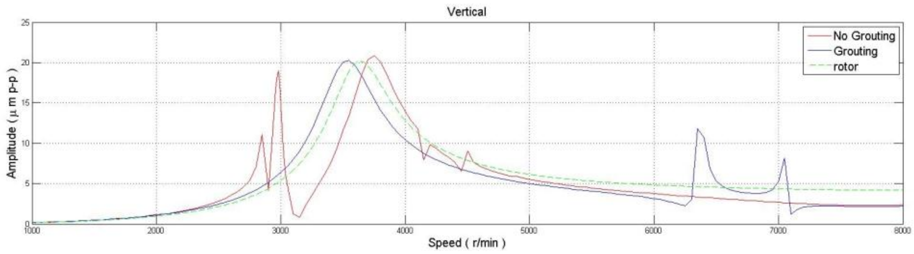


Fig. 5. Unbalance response: relative displacement, amplitude µm p-p vertical direction

The analisys on the rotor (Fig. 5) without the influence of the support system has been developed on a rotordynamic tool and the behaviour of the whole system including rotor and baseplate have been directly modeled in Ansys.

Six frequency sweeps with mode superposition method, have been executed on the examined model applying a unit load on nodes 1 and 2 along their respective DOFs. For each and every sweep analysis the displacements were extracted on the above mentioned DOFs. The resulting transfer functions matrix H(ω) is a [6x6] matrix that varies with frequency.

The same procedure has been applied through the TFUN command; in particular six PSD simulations were executed to obtain the same transfer functions matrix. From the comparison as shown in Fig. 6, it is clear that the results coming from the use of TFUN command perfectly correspond to those generated by the typical frequency sweep.

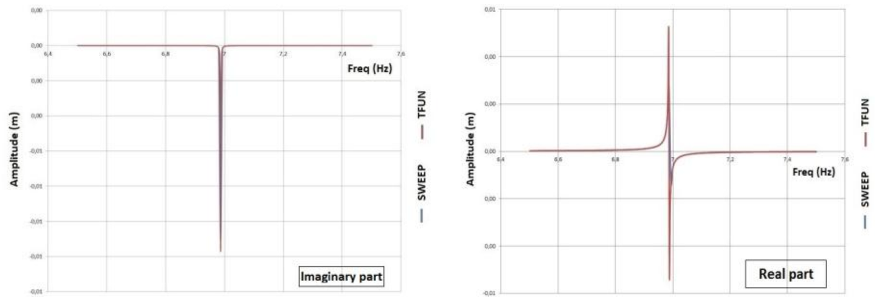


Fig. 6. Real and Imaginary part of a component of H

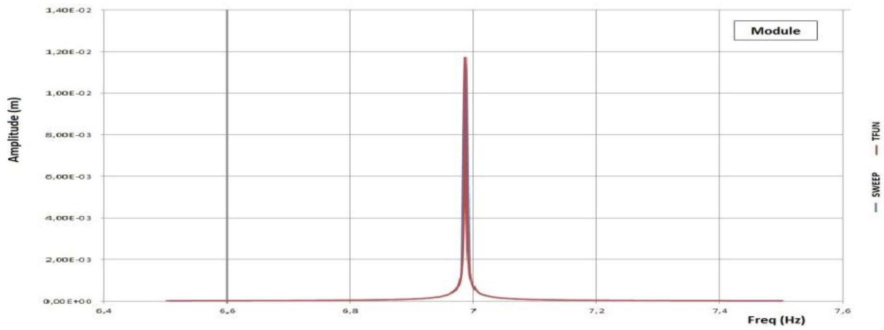


Fig. 7. Module of a component of H

In addition to the excellent correspondence of the two results, it is important to highlight the accuracy with which the transfer functions matrix is extracted in frequency through TFUN command with extremely low computational cost. The software simply evaluates the transfer function in the required nodes leveraging the modal results. To obtain the same accuracy with the sweep method, a large number of samples are required, thus drammatically increasing the computation time. This can be clearly seen comparing the results of the two procedures with refence to the above mentioned model (~260000 DOFs), solved on a WIN64x machine with 2 CPU and 32Gb RAM, solved with harmonic analysis in mode superposition method in the range between 1 and 56 Hz with a total of 5000 substeps, each of them with a running time of approximately 34 sec.

The comparison among the computational times of each simulation methods is reported in the following diagram. The TFUN procedure results to be 48 times faster than the classic frequency sweep. Such ratio increases as the frequencies range and the frequency resolution grows and it will be even more pronounced as the number of model DOF increases. This peculiarity makes the TFUN metodology optimal for extremely complex structures, such as offshore baseplates.

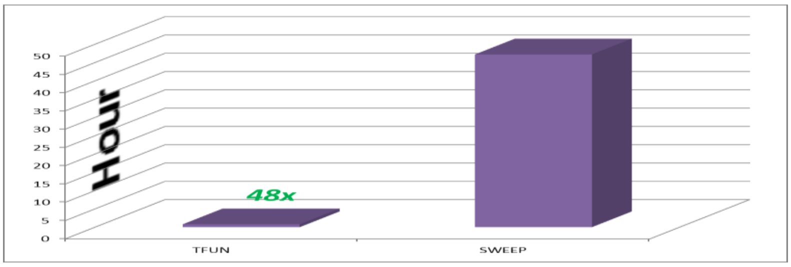


Fig. 8. Computational Time Comparison Limitations

Besides the impossibility to consider any non-linearity (contacts, large deflections and non-linear material models) both for PSD and sweep methods, the TFUN command has some intrinsic restrictions as it needs to use symmetrical solvers for the modal analysis. It doesn't allow the definition of asymmetric elements as well as the use of asymmetric reals and the possibility to consider the Coriolis effect.

Moreover, as the PSD is based on the modal results previously obtained, the procedure can't be applied if the vibration modes vary with ω, a negligible aspect for the study of the base support. However, in case it is required the software to take into account the dependence of the modes with ω, a special procedure could be implemented setting up a frequency sweep with a "full" solver. For any harmonic it is possible to consider the variation of the stiffness and the damping matrix which would cause the modes shifting, the simulation time will however consequently increase.

An alternative procedure can be explored introducing the possibility to evaluate several modal analyses with different ω and to analytically assembly the transfer functions' matrix.

According to [1], considering two coupled systems, the matrices of transfer functions of the two systems can be independently evaluated and subsequently coupled using the equations of continuity (equal displacement of the connection DOFs) and continuity (interfaces forces in the connections DOFs should cancel).

For this reason, the the baseplate transfer functions matrix obtained as previous described, can be coupled to the rotor system through the appropriate equations of compatibility and continuity and used inside rotordynamic software. In the rotodynamic software only the rotor is modeled and the matrix of the transfer functions allow to consider both the contribution of the baseplate and the coupling with the rotor.

4 Conclusions

A proper representation of the dynamic behaviour of complex elastic supporting structures reveals to be extremely important to correctly develop reliable rotordynamic models.

When evaluating the whole rotordynamic response of a complex assembly where rotor, bearings and support structures are dynamically interconnected, the outcoming informations can give a greater insight on the system behaviour often helping the resolution of critical field issues.

All the current methodologies in use in the industries can now be further expanded introducing hidden tool capabilities with the objective to improve accuracy and efficiency on rotordynamic assessments.

The new strategy as described in the current paper will set down the basis for a new code development capable to move accuracy and calculation speed ahead at minimum computational efforts.

Since the computational time gain on the very simple baseplate is in the order of ~50 and the scalability becomes much more pronunced with increasing model size, the methodology is of a great interest for complex off shore shaftlines often extended over 20 meter long including 3 or more rotating machines in one shaft

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