11th World Congress on Computational Mechanics (WCCM XI)
5th European Conference on Computational Mechanics (ECCM V)
6th European Conference on Computational Fluid Dynamics (ECFD VI)
E. Oñate, J. Oliver and A. Huerta (Eds)

DEVELOPMENT OF EFFICIENT MODELS FOR THE STUDY OF COMPLEX ROTATING MACHINES AND OF THE ROTORS-FOUNDATIONS INTERACTION

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Key words: Rotating machines, Rotordynamics, Fundations, Bearings

Abstract. The accurate three-dimensional (3D) modelling of the dynamical phenomena characterizing rotating machines plays a fundamental role in rotordynamics and turbomachinery to ensure system stability, to obtain safe operating conditions and to minimize the costs (design, testing installation and maintenance).
A generic rotating machine usually comprises four main structures: rotors, bearings, casings and foundation. In the last decades the research mainly focused on the 2D and 3D analysis of the single previous parts without considering the whole mechanical system [1][2]; however this kind of approach could only lead to partial results. In the last years it has become evident that global 3D models, comprising all the system parts, are required to reach the desired goals in terms of accuracy, stability, safety and costs [2] [4]. At the same time high numerical efficiency and memory consumption are needed to get a satisfying compromise between accuracy and numerical efficiency. Currently this is one of the main open problems in rotordynamics and turbomachinery.
In this work the authors present a global 3D finite element (FE) approach to model complex rotating machines by taking into account all the main parts of the system. The authors aim to obtain some improvements with respect to the state-of-the-art of the discipline especially concerning the achievements of a better compromise between accuracy and numerical efficiency in modelling complex rotating systems comprising one or more turbomachines (turbines, compressors, etc.). The authors focus also on the feasibility and the performance of different modal reduction techniques [9] to be applied both to the rotor and to the foundation modelling.
The importance of the elastic support arise because of the complexity of the dynamic behavior of whole system that it isn't a simple sum of the original dynamic of the single

part.
The whole model has been developed and validated in cooperation with GE Oil & Gas which provided both the technical documentation and the experimental measurements related to several experimental campaigns performed in the last years on different operating turbomachines composed by turbines, compressors, motors and generators.

1 Introduction

Considering a generic rotating machinery as mainly partitioned in three components (rotors, bearings and foundation) it is possible to recognize that many research activities have been performed in these years by studying single components without taking into account the whole system assembly.
The importance of a combined analysis of rotors and foundations arises from the continuous development of turbomachinery, in particular with the increasing need of efficiency, request of larger MTBF (mean time between failure) and higher and higher performance of rotors.
The diffusion of turbomachinery plants in very different fields, like off-shore installations, leads to the necessity of a more complete point of view, needed for the optimal design of the whole structure, expecially in off-shore installation where the underconstrained anchoring system plays a fundamental role.
As FEM theory offers numerous solution to represent rotor system (beam and solid model, transfer function..) and many formulations of the problem have been proposed (based on Timoshenkos [6] and Bernoullis relations [7]), in this paper great importance is reserved to the implementation of a reduced order model for the supporting structures with the aid of modal dynamic reduction.
From predesign to verification, there are several ways to model a rotor: simple models made up of *beams and lumped masses* are well suited to assess the global dynamic behavior of the rotor with short computation times; *3D solid* models can be built to reproduce accurate results because they naturally include all details of the rotor. Numerous theoretical approaches exist to consider the complex behaviour of the substructure, characterized by drawbacks and opportunities (state space representation using modal parameters, transfer function to model equivalent spring and damper, different approach of modal reduction). The complex modelling of a bearing, as reported in literature, is mainly split into two big categories: spring-damper, mass and inertia concentrated models or a full 3D model. Every approach has its own main advantages and disadvantages but it is obvious that a model with concentrated characteristic require lower hardware resources while a 3D model guarantee more accurate results.
This research activity aims to show the evidence of how elastic supports generate a deep to coupling among different bearings and wants to demonstrate the unavoidable necessity of the inclusion of this component in the analysis since project preliminary stages.

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

- the response of the system includes the modes of the different components of the assembly;
- the rotor modes shift to lower frequency without interference on the deformed shape;
- the rotor maintain its original modes but deformed shapes change.

There are two important goal to achieve: accuracy and efficiency. The first is related to the confidence of the results. The second target instead consist in the reduction of calculations times and it is closely related to modal reduction advantages.
The general architecture of this study aims to highlight the mutual interaction between the three main components of this model: rotor, bearings and foundation. The flow of local variable Fig.1 (load due to gravity and gyroscopic effects and consequent displacement) shows the primary importance of bearings as filter element interposed between the baseplate and the rotor. Bearings will be modelled as equivalent spring-damper dependent on the running operational velocity.
Two steps are required to accomplish the analysis. *The first step* considers the rotor mounted on foundations and none of them is subject to any sort of model simplification. *The second step* characterized by the simplifying assumptions to represent supporting structure trying to obtain a good compromise between accuracy and efficiency. A graphic representation of the general architecture is visible in Fig.1.
The first level in Fig.1 represents the physical\real system and in the second level the equivalent FEM model is described.
The general architecture diagram shows the flux of variables between the three main components of the assembly: it is possible to recognize that the exchange of information clearly hinge on bearings that transmit a continuous flow of action and reaction forces: the motion of the rotor, disrupted by the unbalance action excites the elastic supporting structure that reacts according to its resonance frequencies, while bearings represent the mean of communication.
As the system correspond to a full 3D model the displacement will be explicitly described in their 3 directions x, y and z (respectively u on the longitudinal axis parallel to the symmetry axis of the rotor, v trasversal and w in the vertical direction). The three rotation are θ_x, θ_y and θ_z.
TheI^{st} *STEP* has no simplification and the II^{nd} *STEP* exploit the modal reduction to describe the supporting structure.

2 The model

In this paragraph equations and hypothesis are explicitly expressed to better understand the FEM model that stand behind results.

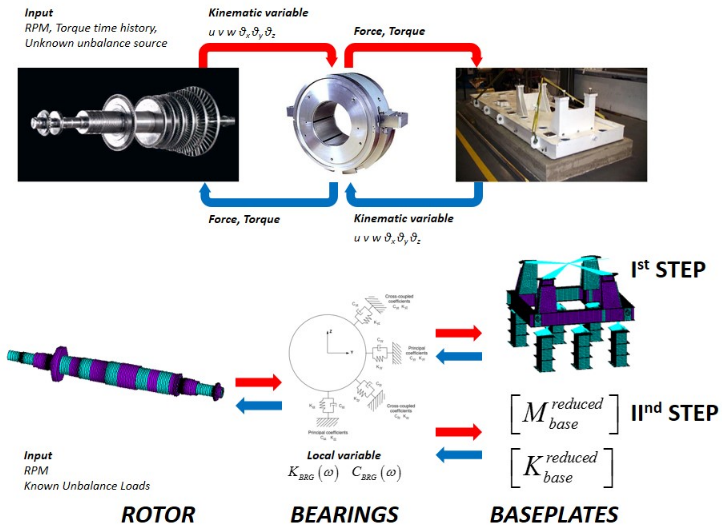

Figure 1: General architecture.

Even though the implemented model is used for a stationary analysis (harmonic response) a transitory simulation is possible and ready, in anticipation of future comparison and studies.

2.1 Ist step

The rotor, that classically shows as presented in Fig.2, is modelled with beam element.

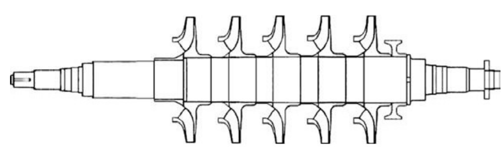

Figure 2: Example of 5 stage centrifugal compressor

The complete rotor shaft is a superposition of different effects as it would be in the real case (Fig.2) if we suppose to key and shrink over the naked shaft: impeller that transform the energy derived from the fluid and element of junction to transmit wrench and get the machine running and other sleeves like seals.
There are two kind of effect to impart to the rotor: the increase in mass due to the extra component on the shaft and the change in the dynamic response due to the inertial properties of the disks. The mass contribution is implemented as concentric sleeves, namely a beam with hollow circular cross sections, laid on the shaft for the length of the real component.

If we consider a two node element and sum the effects of the fitted component the result of this superposition is a system like the one showed in Eq.1

$$\begin{aligned}\{[M_{shaft}]+[M_{sleeve}]+[M_{mi}]\}\{\ddot{q}_{rotor}\}+\\+\left\{[C_{shaft}]+\Omega^2[G_{shaft}]+\Omega^2[G_{mi}]\right\}\{\dot{q}_{rotor}\}+\\+[K_{shaft}]\{q_{rotor}\}=0\end{aligned} \tag{1}$$

where M_{shaft}, K_{shaft}, C_{shaft}, G_{shaft} respectively specifies the naked rotor mass, stiffness, damping and gyroscopic effect matrix, M_{sleeve} indicate the mass contribute of shrink fitted elements (like impeller), M_{mi} and G_{mi} explicit the dynamic contribute (concentrated properties m_x, m_y, m_z, I_p, I_t) of elements modelled as sleeves, then F represent external forces.

It is now possible to write the complete equation that comes from Jeffcott's theory in a summarized array:

$$[M]\,\ddot{q}_{rotor}+([C]+\Omega^2[G])\dot{q}_{rotor}+[K]\,q_{rotor}=F_{external} \tag{2}$$

A sensitivity mesh analysis has been carried out on the beam model to verify the discretization and then the minimum number of node that guarantees the accuracy and numerical efficiency. The F_y and F_z forces that the bearings exert on the rotor in the plane transversal to its axis can be expressed in terms of linearized force coefficients for small perturbations about a stationary equilibrium at a given shaft speed.

A common representation of the reaction forces that a bearing exerts on the shaft allows a direct apprehension of the equations that describe to bearing behaviour:

$$\{F_{brg}\}=\left\{\begin{array}{c}F_y\\F_z\end{array}\right\}=\left[\begin{array}{cc}-K_{yy}&-K_{yz}\\-K_{zy}&-K_{zz}\end{array}\right]\left\{\begin{array}{c}y\\z\end{array}\right\}+\left[\begin{array}{cc}-C_{yy}&-C_{yz}\\-C_{zy}&-C_{zz}\end{array}\right]\left\{\begin{array}{c}\dot{y}\\\dot{z}\end{array}\right\} \tag{3}$$

where K_{yy}, K_{zz} and C_{yy}, C_{zz} are *Direct stiffness and damping coefficients*, while K_{yz}, K_{zy} and C_{yz}, C_{zy} are *Cross-coupling stiffness and damping coefficients*.

The entire baseplate, is bounded to the ground with spring element whose characteristic is extracted from dynamic stiffness simulations to satisfy the results of hammer test on the supporting pedestals. Assembling the equations of the full system, Eq.4, so introducing the effects of supports, the new set of coordinate $\{q\}=\left\{q_{rot}^T,q_{base}^T\right\}^T$ correspond to a rearrangement of the vector containing the coordinate of the node related to the rotor $\{q_{rot}\}$ and to the support structure $\{q_{base}\}$

$$\begin{aligned}\{[M_{rotor}]+[M_{base}]\}\{\ddot{q}\}+\left\{[C_{rotor}]+[C_{base}]+\Omega^2[G_{rotor}]\right\}\{\dot{q}\}+\\+\{[K_{rotor}]+[K_{base}]\}\{q\}=F_{brg}+F^{ext}\end{aligned} \tag{4}$$

In Eq.4 F^{ext} this time contains all the loading forces external to the complete assembly, that is to say unbalance load configuration.

2.1.1 IInd step: Modal Reduction

The second step doesnt introduce any change to the rotor model while the baseplate undergoes modal reduction.
Master DOFs are chosen on the boundary of the system and so the nodes that perfectly

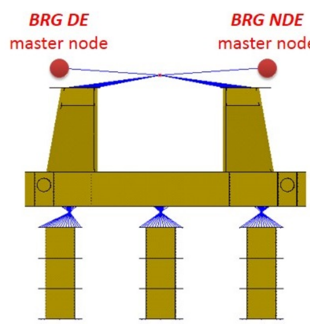

Figure 3: Superelement Master Nodes

suit the circumstances are, as highlighted in the left view of Fig.3, nodes on the extremity of elements that represent the linking between bearings and foundations.
As the goal of this analysis is to describe the flexo-torsional behaviour of the assembly each master node retain all six DOF (three rotations and three translations). The below equation Eq.6 represents the equivalent form of Eq.4 when the baseplate undergoes modal reduction. As in the previous paragraph it is now possible to write the equation that represent the whole system Eq.6. This time the coordinate vector (Eq.5) is in reduced order size since it contains the reduced coordinate of the baseplate

$$\{q_{red}\} = \left\{q_{rotor}, q_{base}^{red}\right\}^T \tag{5}$$

and so Eq.4 turns into

$$\begin{aligned} \left\{[M_{rotor}] + \left[M_{base}^{red}\right]\right\}\{\ddot{q}_{red}\} + \left\{[C_{rotor}] + \left[C_{base}^{red}\right] + \Omega^2 [G_{rotor}]\right\}\{\dot{q}_{red}\} + \\ + \left\{[K_{rotor}] + \left[K_{base}^{red}\right]\right\}\{q_{red}\} = F_{brg} + F^{ext} \end{aligned} \tag{6}$$

3 Test case: *five stage centrifugal compressor*

The system under investigation (Fig.4) is part of a more complex rotor train (Fig.5) composed of a steam turbine driver and a longer commune foundation, constrained to the ground with three anchoring points. The steam turbine and the centrifugal compressor are linked with an elastic coupling that directly transmit the wrench output of the steam expansion.
The original rotor train, in turn, is a section of a plant designed for off-shore applications and the constrain system is able to compensate small thermal expansions and deck deflections. The system is designed to be part of a *FLNG* (Floating Lquified Natural Gas)

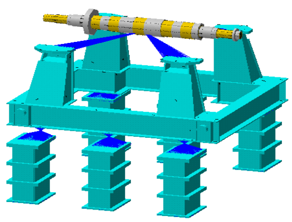

Figure 4: Temporary baseplate

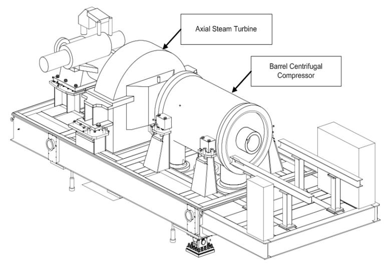

Figure 5: Turbo-compressor rotor train

vessels which extract, process and liquefy natural gas.
The rotor is constrained with two radial tilting pad fluid dynamic bearings each one owning 5 pads. The two radial bearings work coupled with axial tilting pad bearings. The rotor operate in a speed range of about 2000rpm where 4000rpm represent the Minimum allowable speed and 5775rpm is the Trip speed. As result of high shaft vibration issue a transient run-up and run-down test has been accomplished to verify the acceptance criteria and threshold vibrational level.
A mechanical Running test is a standard procedure to verify acceptance vibrational level. The equipment shall be operated at speed increments of approximately 10% from zero to the maximum continuous speed and run at the maximum continuous speed until bearings, lube-oil temperatures and shaft vibrations have stabilized. As it is possible to see in Fig.6 this is a classic transitory analysis with a trapezoid profile of velocity.

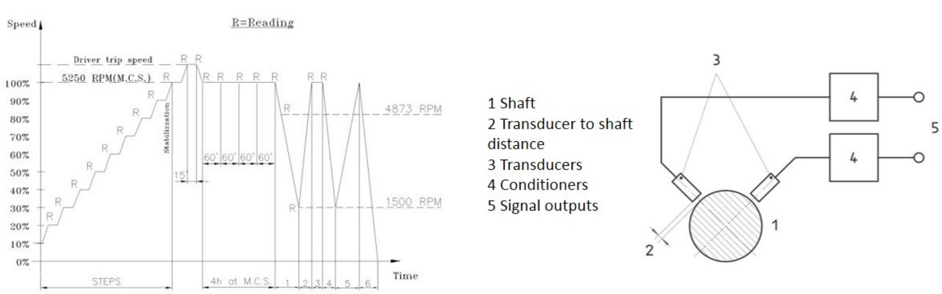

Figure 6: Test sequence and relative displacements transducer

What will be show and discussed in the next paragraph will be the rotor vibration amplitude of the vertical displacements, that is specifically relative displacement at the bearing location between the non rotating supporting structure.

4 Results and major findings

Observing the dynamic behaviour of the rotor it is possible to see the presence of an amplification of vibration with a resonance peak placed at 2500rpm.

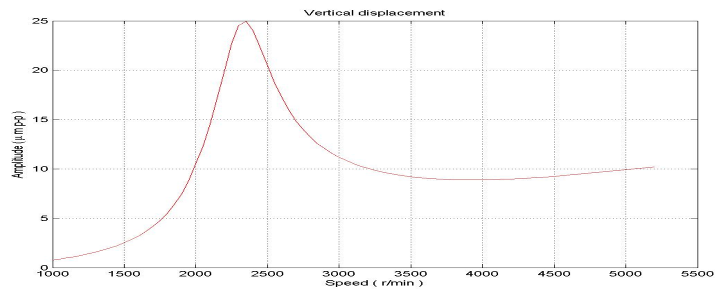

Figure 7: Absolute vertical displacements rotor

The Undamped Critical Speed (UCS) map (Fig.8) shows that in the operative range there are no critical frequency of the rotor considered as a standalone component and this is exactly what the unbalance response analysis (Fig.7) confirm.

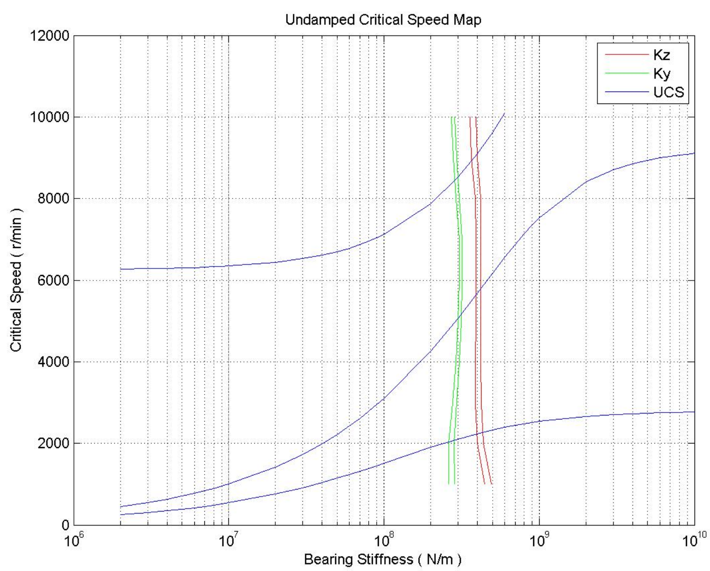

Figure 8: Undamped Critical Speed Map

Introducing the elastic support in the analysis (Fig.9), considering first of all absolute diplacements, it is evident the appearance of a new amplitude peak that stand in the operative range of the compressor.

The experimental results obtained with proximity sensor on the bearing housing shows

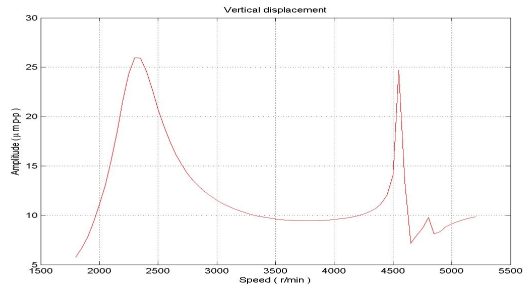

Figure 9: Absolute vertical displacements rotor side

the existence of the second peak. Overlaying the experimental result (Fig.10) with the

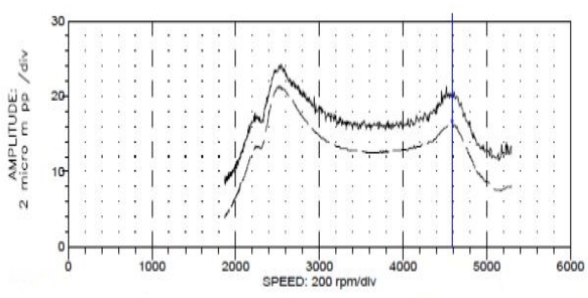

Figure 10: Experimental relative vertical displacements

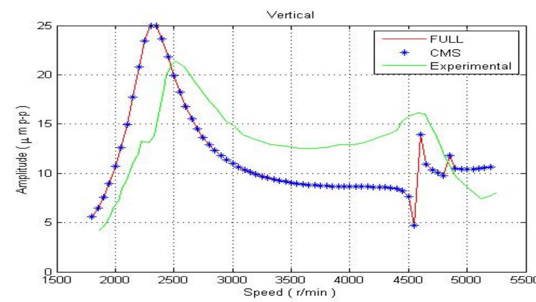

Figure 11: Relative vertical displacements rotor - comparison

simulated response to the unbalance, it is possible to see that the model detects the presence of the resonance. In Fig.11 the comparison among the response of the full model (FULL with no simplifications), the one of the reduced model (CMS) and the experimental response (Experimental) is reported.
Time of calculation take deeply advantage from the introduction of modal reduction: as shown in Tab.1 there is a 80% decrease

Table 1: Time of calculation.

ANALYSIS	*HYPOTHESIS*	*TIME* [hh:mm:ss]
Ist step	No simplification	00:25:16
IInd step	Modal reduction	00:03:32

5 Conclusions

The behavior of a Rotor machine is not simple to predict and many tools and theories need to be compared at the same time to have a complete vision of the real behavior.
It is not sufficient to evaluate the different component of the rotating machine separately, it is important to have a deep knowledge of the entire assembly behavior (rotor, bearing and support structure) to predict the mutual influence of design parameter and allow different parallel working team to elaborate the problem with the right input.
The model, oriented in creating an accurate tool for the rotor dynamic of the whole machine is at the moment a good improvement in terms of time and accuracy.
A future development is the representation of a complete train shaft line with experimental match out of dedicated experimental data.

REFERENCES

[1] D. Childs, Turbochinery Rotordynamics, Wiley & Sons, New York, 1993.

[2] M. Lalanne, G. Ferraris, Rotordynamics prediction in engineering, Wiley & Sons, New York, 1998.

[3] A. Muszynska, Rotordynamics, Dekker Mechanical Engineering, Dordrecht, Netherlands: Taylor & Francis, 2010.

[4] API 617 7th Edition, Axial and Centrifugal Compressor and Expander-compressor for Petroleum, Chemical and Gas Industry Services, July 2002.

[5] D. L. Thomas, Dynamics of rotationally periodic structures, International journal for numerical methods in engineering, VOL. 14, 81-102 (1979)

[6] G. R. Cowper, The shear coefficient in timoshenko's beam theory,Journal of Applied Mechanics 33, 335-340 (1966) (6 pages); doi:10.1115/1.3625046

[7] S K Park and X-L Gao, BernoulliEuler beam model based on a modified couple stress theory, Journal of Micromechanics and Microengineering, 16 2355. doi:10.1088/0960-1317/16/11/015 (2006)

[8] J. C. Wachel and Fred R. Szenasi, Analysis of torsional vibrations in rotating machinery

[9] M. C. C. Bampton and R. R. Craig Jr, Coupling of substructures for Dynamic Analysis, AIAA Journal, vol. 6, no. 7, 1968.