Dynamic Stability of Rotating Systems in Turbomachinery Under API Standards

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Abstract

The stability of rotating systems in turbomachinery has to account for the structure design, the actual geometry of bearings and all of the remaining forces, including operational data as well as temperature of the bearing lubricating oil. Thus, a study of the dynamic stability of rotating machines is complex and time consuming in terms of modelling and calculations. Even though it is recognized that other methods of analysis and acceptance criteria have been used to evaluate stability, API standardization procedures ensure appropriate notification and participation in the development process. According to these procedures, firstly we prepare a rotor technical documentation based on geometry measurements, frequently with optical scanning. Secondly, a theoretical analysis consisting of calculations with the finite element method based on the program that allows us to build a numerical model of rotor dynamics, is carried out. Then, the so-called "bump test" is performed to measure natural frequencies of a freely suspended rotor, which makes it possible to "tune" the theoretical model, making it compatible with the real object not only in terms of geometric dimensions and mass, but also from the point of view of the form and frequency of free vibrations. Thus, we obtain an experimentally verified numerical model which can be used for future machine diagnostics and other needs.

Keywords: turbomachinery, rotating system stability, model tuning, API standards

1. Introduction

In all types of rotating machines, a rotating force is periodically applied to the rotor of the machine during its operation. Its presence follows from the way the machine works and the unbalance. A periodically variable force acting on the rotor results usually in a periodic shaft movement in different directions, i.e., vibrations. Under certain conditions, phenomena such as resonance or instability of vibrations can lead to much larger displacements of the shaft and acting forces than during the normal operation. Excessive forces can lead to faster fatigue wear of some parts. Too large displacements can cause rotating parts to come into contact with fixed parts, which, consequently, can be followed by damage of the machine. Other effects of excessive vibrations may yield a reduced operational precision, e.g., in machine tools and medical devices, or an increased clearance in seals and a deterioration of the efficiency in turbomachines (turbines, pumps or compressors and fans).

Therefore, it is necessary to design machines in a way that guarantees operation at a sufficiently low level of vibrations. A meticulous and reliable theoretical dynamic analysis of the structure is required for newly built or modernized machines to confirm their compliance with the requirements of standards. In relation to industrial turbomachines, these are usually API 617, API 612 and API 610 standards describing in detail the methodology and methods for such an analysis as well as specifying the requirements the designed structure has to meet in terms of dynamics during its operation.

The present article describes an example of a theoretical model and its tuning for large industrial compressors and shows the usefulness of this procedure and its impact on the reliability of the dynamic analysis results.The ultimate purpose was to develop a numerical model of the free-hanging rotor of the compressor and tune it on the basis of the experiment to allow the users to conduct stability calculations according to their needs.

In modern calculations of machine dynamics, the finite element method (FEM) is most commonly used. It requires a theoretical model of the structure to be developed by dividing it into a number of elements (finite elements) in order to generate a computational grid. Then, the equations of displacement and motion are determined in every possible direction (depending on the number of degrees of freedom), for each of mesh nodes, depending on node forces. The acting forces and the number of degrees of freedom are known due to the given boundary conditions and the relationships existing between individual nodes.

According to API 617, it is necessary to perform vibration analysis of the compressor rotating system, which should include, among others, values of all critical speeds, from 0 to as high as 2.2 times the maximal rotational speed during continuous operation, determined at the rotational speeds from 25% to 125% of this speed. If the bearing foundation is not rigid enough and it is necessary to model it, the machine body can be modelled as a beam system placed on the foundation with certain stiffness and damping. The described stages of works are connected to preparation and tuning of the rotor model and to selected stability issues of the machine dynamical analysis.

2. Stability analysis of the machine rotating system

The majority of failures of overcritical rotors are caused by a local increase in the synchronous or asynchronous vibration amplitude and by exceeding the clearances when the critical frequency or the stability threshold of the rotating system is exceeded. It is evident that dynamic characteristics of the machine rotating system depends on rotor-support dynamic properties. It means that the external forces generated during machine operation can alter the dynamic properties of the whole rotating system. However, this effect depends strongly on flexibility of the shaft and dynamic properties of the rotating system components (bearings, seals and centrifugal or axial flow stages).

Let us consider a generalized two-degree-of-freedom linear rotor system presented in Fig. 1.

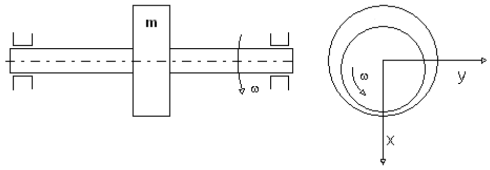


Figure 1. Two-degree-of-freedom linear rotor system

A simplified equation of the lateral movement of the rotating shaft mass $\boldsymbol{m}$ at a constant speed $\boldsymbol{\Omega}$ can be written in the following way:

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} = \begin{Bmatrix} m e \Omega^2 \cos(\Omega t) \\ m e \Omega^2 \sin(\Omega t) \end{Bmatrix} \tag{1}$$

The generalized damping and stiffness matrix of equation (1) contains all forces acting on the flexible shaft, i.e., bending stiffness, internal and external damping, bearing stiffness and damping, fluid cross-coupling forces, auxiliary active support damping and stiffness, as well as other forces acting on the mass $\boldsymbol{m}$. The generalized stiffness $[k_{ij}]$ and damping $[c_{ij}]$ matrices can be decomposed as follows:

$$\begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix} = K_{el} + K_{rot} \qquad \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix} = C_{diss} + C_{gir} \tag{2}$$

where:

$$K_{el} = \begin{bmatrix} k_{xx} & \frac{1}{2}\left(k_{xy} + k_{yx}\right) \\ \frac{1}{2}\left(k_{xy} + k_{yx}\right) & k_{yy} \end{bmatrix} \quad K_{rot} = \begin{bmatrix} 0 & \frac{1}{2}\left(k_{xy} - k_{yx}\right) \\ -\frac{1}{2}\left(k_{xy} - k_{yx}\right) & 0 \end{bmatrix} \tag{3}$$

$$C_{diss} = \begin{bmatrix} c_{xx} & \frac{1}{2}\left(c_{xy} + c_{yx}\right) \\ \frac{1}{2}\left(c_{xy} + c_{yx}\right) & c_{yy} \end{bmatrix} \quad C_{gir} = \begin{bmatrix} 0 & \frac{1}{2}\left(c_{xy} - c_{yx}\right) \\ -\frac{1}{2}\left(c_{xy} - c_{yx}\right) & 0 \end{bmatrix} \tag{4}$$

From the point of view of the rotating shaft vibration control, this decomposition into conservative (K_{el} and C_{gir}) and non-conservative (C_{diss} and K_{rot}) parts allows us to analyze an influence of the additional cross-coupled forces furnished by the bearing/seals system as well as an effect of the flow stages on the shaft energy level in each cycle of motion. The total work of conservative forces per one cycle of motion is still equal to zero but the total work of non-conservative forces can disturb the energy balance of the rotating system even under steady-state conditions. These energy considerations show the following requirements for an external additional active support of the flexible rotating shaft in order to improve the reliability of the machine rotating system without risk of its destabilization:

1. During one cycle, the work generated by non-conservative damping forces expressed by the matrix C_{diss} has always a negative value and decreases the energy balance of the rotating system regardless of the precession direction. Therefore, an influence of external support damping forces is absolutely beneficial and always stabilizes the rotating system, lowering the energy of the total precession.
2. During one cycle, the sign of the work performed by non-conservative forces of coupled stiffness depends on the direction of the total precession. This indicates that the sign of the term (k_{xy}-k_{yx}) becomes decisive for stabilization of the forward and backward modes of the shaft elliptical movement. For (k_{xy}-k_{yx}) > 0, energy increases for the forward precession and decreases for the backward precession and, consequently, for (k_{xy}-k_{yx}) < 0, energy decreases for the forward precession and increases for the backward one.

From the point of view of the stability margin of the flexible rotating shaft, let us consider an influence of the residual unbalance on the energy balance in the elliptical motion. The above-mentioned considerations related to the energy balance in the elliptic motion of the flexible shaft allow us to formulate general recommendations for the forces of the rotating shaft that should act without a risk of destabilization of the machine rotating system. These recommendations are as follows:

- possibly high values of external damping forces that absolutely stabilize the rotating system,
- in the case of permissible alternations in the dynamic characteristics of the system from the viewpoint of its operation conditions, the forces that increase direct stiffness (k_{xx}, k_{yy}) are acceptable,
- the forces that are related to the forces altering the cross-coupled stiffness terms (k_{xy} , k_{yx}) should be seriously analyzed and avoided as far as possible.

The latest editions of the API standards draw special attention to a theoretical stability analysis taking into consideration anticipated cross-coupling terms of the rotating system. A simple and very useful technical indicator of the machine rotating system stability estimations used in the API standards methodology is a logarithmic decrement of damped natural frequencies of the whole rotating system. A negative logarithmic decrement or a damping factor indicates system instability. A stability analysis shall be performed for those machines, where the maximal speed is greater than the first undamped critical speed on rigid supports, especially when all interactions that are related to the forces altering the cross-coupled stiffness terms (k_{xy} , k_{yx}) are considered. These forces are connected to dynamic properties of the rotating system support components (bearings, seals) as well as to interactions related to the flow in machine centrifugal or axial stages.

3. Model of the rotor dynamics

The dynamic model built to determine, among others, the eigenfrequencies of the considered object have to be composed of an appropriate number of elements with specific material properties. These can be elements of the shaft-type, which affect the inertia and rigidity of the model, or of the disc-type, which influence the rigidity of the model only. Disc-type elements represent components of the shaft (half-couplings, rotors, balance disks, bushes, etc.). In specific sections (corresponding to the centres of gravity of these elements), their mass and moments of inertia (necessary to take into account the gyroscopic effect) are introduced. An influence of bearings and seals is taken into account by determining the centre position of this type of elements and specifying the matrix of the values of stiffness and damping coefficients as boundary conditions.

In fact, shield-type components also add extra rigidity to the shaft, especially if they fit tightly in the shaft. In this case, their absence in the model may lead to an underestimation or overestimation of the rigidity. In engineering practice, an impact of this additional stiffness is difficult to determine and, therefore, usually not considered at the design stage. However, if the experimental results differ from the model predictions by more than 5%, the model tuning is required in accordance with the standard.

Figure 2 shows a scheme of the rotor structure with its corresponding dynamic model developed on the basis of optical scan measurements. The calculated parameters of the rotor are given in Table 1.

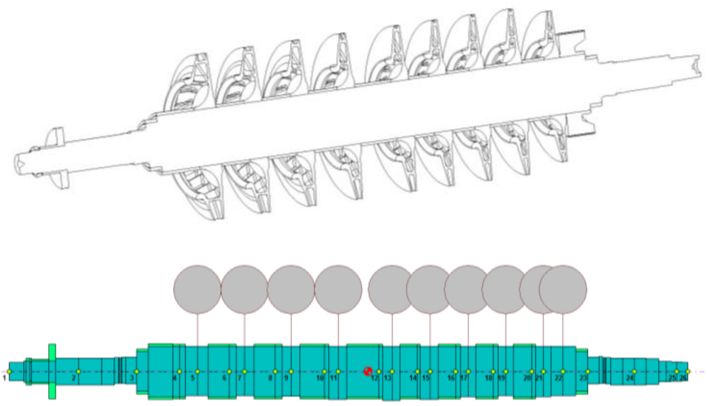


Figure 2. Scheme and visualization of the numerical model of the rotor structure

Table 1. Calculated parameters of the rotor

total length [mm]	total mass [kg]	position of the centre of mass (from suction side) [mm]	moment of inertia (diametral) [kg m^2]	moment of inertia (polar), [kg m^2]
2570	831.8	1362	249.9	14.658

Rotor wheels and a relieving piston were modelled on the basis of the developed technical documentation and in the dynamic model they were symbolically represented as disks with a concentrated mass in nodes (stations) of the rotor numerical model. Physical parameters of these elements were calculated and listed in Table 2.

Table 2. Physical parameters of model elements

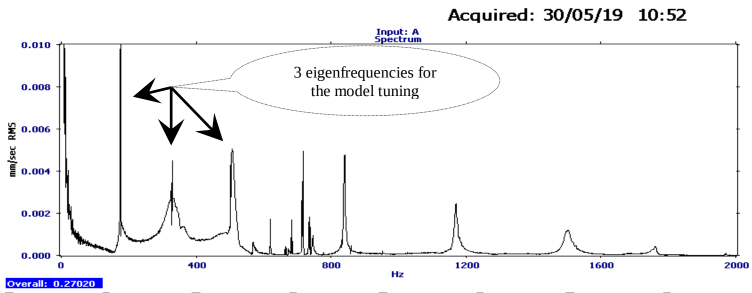
element	node	element mass [kg]	moment of inertia (diametral) [kg m^2]	moment of inertia (polar) [kg m^2]
disc 1	5	40.660	0.80078	1.5309
disc 2	7	37.750	0.74076	1.4356
disc 3	9	39.460	0.75184	1.4597
disc 4	11	36.700	0.67967	1.3302
disc 5	13	34.760	0.65008	1.2752
disc 6	15	34.290	0.63731	1.2527
disc 7	17	33.050	0.60466	1.1892
disc 8	19	35.680	0.66280	1.3057
disc 9	21	28.280	0.44547	0.87582
piston	22	40.000	0.44938	0.83794

4. Bump test and model adjustment

In order to verify and adjust the model, a bump test needs to be carried out to identify eigenfrequencies of the analysed rotor. Figure 3 shows the free-hanging compressor rotor used to record the spectrum of frequencies. The aim of the bump test is to verify and tune the numerical model developed. Below, we can see how vibration sensors were positioned. In addition, optical markers, which were used for measurements of the rotor geometry with a scanner – a 3D measuring system operating on the principle of streak projection, ensuring a high precision and a detailed resolution at high speeds, are visible.

Figure 3. Free-hanging rotor of the compressor during natural vibration measurements for numerical model tuning

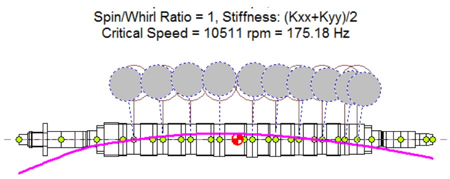
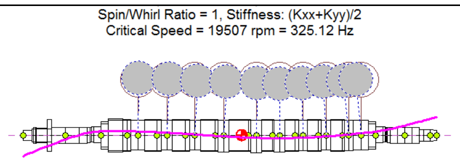
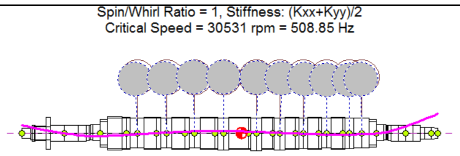
Figure 4 shows test results. During the tuning of the theoretical model, the first three eigenfrequencies of the object under analysis, pointed with arrows, were taken into account. Table 3 summarizes their values and shapes of free vibrations from the rotor measurements along with those calculated theoretically after model tuning.



Frequency Data		Frequency Data		Frequency Data		Frequency Data	
2.72	0.09353	4.18	0.04033	5.95	0.0212	7.92	0.01109
9.47	0.00983	11.62	0.00963	13.58	0.0063	15.2	0.00485
16.5	0.00446	17.25	0.00426	18.54	0.00383	19.99	0.00401
175.35	0.01152	325.97	0.0039	328.27	0.00466	500.92	0.0049
506.56	0.0059	713.26	0.0036	715.88	0.00551	839.07	0.00557

Figure 4. Example of the recorded vibration spectrum of the excited free-hanging rotor with the 3 first eigenfrequencies pointed

Table 3. Eigenmodes and eigenfrequencies from the calculations and from the measurements

Rotor eigenmode shape and frequency from the calculations and the measurements			
1st eigenmode	175.18 Hz	175.13 Hz	
2nd eigenmode	325.12 Hz	325.97 Hz	
3rd eigenmode	508.85 Hz	506.56 Hz	
Rotor mass	831 kg	830 ± 2 kg	

As follows from these results, the model after tuning shows a very good conformity with the real rotor dynamic behaviour and, thus, it can be used in further investigations for designing or diagnostic purposes.

5. Conclusions

API standards related to technical conditions of turbomachines, increasingly used in industry, require a reliable theoretical analysis of the dynamic state of the machine rotating system. The latest editions of these standards draw special interest to ensuring a high stability margin and an elimination of sources of self-excited vibrations in the rotating system of modern (modernized) machines.

Currently available calculation methods allow for reliable modelling and a theoretical dynamic analysis of turbomachinery, both for designing or modernization purposes. However, such an analysis should be conducted on the basis of detailed data as regards the machine design and operating requirements. An experimentally verified (tuned) model of the real rotor shaft allows us to determine an impact of structural changes on machine dynamics as well as to simulate some symptoms of incorrect operation long after the model was built.

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