A FEM MODAL ON STRUCTURAL DYANAMIC MODELED ROTOR INSTABILITY WITH EXPERIMENTAL VARIFICATION

CHINTAKUNTLA CHENDRUDU

Associate Professor in Mechanical Engineering department, SVR Engineering College, Nandyal, Kurnool District, Andhra Pradesh, India

Dr.Y. VENKAT MOHAN REDDY

Professor And Dean, G.Pulla Reddy Engineering College, Kurnool, Andhra Pradesh, India

Prof.K.HEMA CHANDRA REDDY

Chairman, APSCHE, Guntur, Andhra Pradesh, India

ABSTRACT

Increasing operational speed of machines with rotors supported in fluid film bearings imposes higher demands on rotor dynamics solution. Calculation of critical speeds is important design tool, but methods of ensuring rotor stability attract increasing significance. Despite the standard design procedures of renowned producers of rotating machines, cases of rotor instabilities are still encountered in practice. Types and sources of instability are defined, some cases of rotor instability are documented and methods of their suppression are shown in the paper.

KEY WORDS

Rotor stability, oil whirl, oil whip, journal bearing, tilting-pad bearing, floating ring bearing, lobbed geometry, stability reserve, logarithmic decrement, labyrinth seal, antiswirl breaks

1 INTRODUCTION

Still higher operational speed of machines with rotors supported in fluid film bearings brings growing problems with rotor stability. Apart from calculation of critical speeds and response to residual unbalance it is therefore inevitable to calculate rotor stability limits. This process should proceed already in the phase of machine design, because additional modifications of the rotor or bearings are difficult and not always successful. Though standard design procedures include all above-mentioned steps, cases of rotor instabilities are still encountered in practice [e.g. 1, 2, 7]. Causes of instability are varying and sometimes even surprising. As excessive rotor vibration in unstable region made permanent operation of machines impossible, it is necessary to find method of their suppressing or at least reducing their intensity. Some cases of instability encountered during the author’s practice will be mentioned together with methods of their control.

2 TYPES AND SOURCES OF INSTABILITY

Rotors supported in fluid film bearings exhibit basically two types of instability, both of which are characterized by subharmonic vibration with big amplitudes. Instability of „oil whirl“ type, with frequency dependent on rotational speed, occurs more likely with rigid rotors, paradoxically namely with rotors in gas bearings. For „oil whip“ instability, encountered mainly with elastic rotors, is characteristic constant vibration frequency with some – usually but not always – the lowest eigenfrequency of the system. Quite frequent in practice are cases, when instability of „oil whirl“ type converts in region of the 1st eigenfrequency of the system to „oil whip“ type.

The source of instability is in most cases the bearing support itself. Cross coupled terms of stiffness matrix, which promote journal orbit around bearing centre, prevail at some operational conditions (low load, high speed) over direct terms. Special shape of bearing surface, consisting of several areas with preload relative to the bearing centre (lemon bearing, lobbed geometry), increase stability limit, but it need not be always sufficient. Tilting-pad journal bearings are inherently stable, because cross coupling terms are at least one order (usually more than one order) lower than direct terms. However, in cases of intensive external excitation, e.g. from labyrinth seals, even tilting-pad bearings may not ensure stability of all rotors, as will by shown later. Labyrinth seals of big dimensions with high pressure drops constitute one of the most often encountered sources of rotor instability.

Special type of bearing support, so called floating ring, is used in turbochargers (TCH). The bearing consists of two circular bearings, one of which is rotating with the speed equal to about 0,1 to 0,3 of the rotor speed. There are two oil films in series ensuring strong damping of rotor vibration. The bearings are relatively near one to another and heavy impellers are located at both overhang ends of the rotor. Great moments of impeller inertia produce substantial gyroscopic moments influencing bending critical speeds of the rotor by splitting to branches with co-rotating and counter-rotating precession. Calculation predicts in almost all cases instability of the rotor in floating ring bearings. In reality most rotors run stably with very small amplitudes of vibrations, though in some case instability of outer oil film is indicated by subharmonic component with frequency equal to one half of the ring rotational speed. As will be shown later, cases of fully developed instability, with amplitudes of vibration reaching almost the whole of bearing clearance, were encountered too.

3 SOME CASES OF ROTOR INSTABILITY

3.1 Instability of expansion turbine rotor

Rotors of modern turbines or compressors have often impellers attached directly to pinions of high-speed gearbox. Distance of the impeller centre of gravity from the nearest bearing is frequently bigger than the distance between bearings. If sufficiently strong source of external excitation exists, rotor instability can occur even if pinion support is provided by the most stable type of bearing, i.e. titling-pad bearings. In the case described further, the expansion turbine had two stage and both pinions with speeds of 24.000 and 18.750 rpm were arranged in one gearbox (see scheme in Fig. 1).

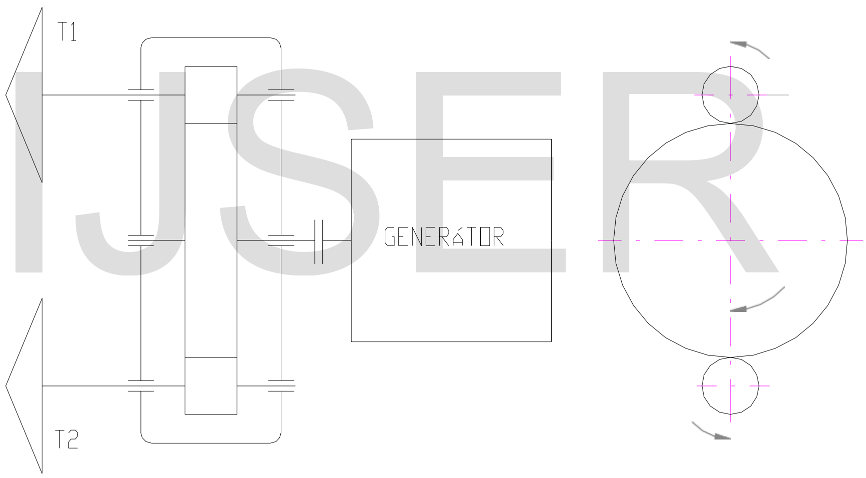


Figure 1: Scheme of two-stage expansion turbine

While the pinion of the 2nd stage with lower speed and greater diameter (higher stiffness) was from dynamic point of view trouble free, pinion of the 1st stage showed in some operational regimes highly increased level of vibration with pronounced subharmonic component. Thorough dynamic analysis [3] of both pinions showed, that both of them have relatively low stability reserve (SR). SR is more often defined as logarithmic decrement (LD); both SR and LD represent ratio of real and imaginary part of eigenvalue. At full output the pinion of the 1st stage had SR of 4,2%, pinion of the 2nd stage had SR even lower - 3,6%. However, stability of the 1st stage pinion was unfavourably influenced by the fact, that the 1st and 2nd eigenfrequencies are very near one to another, factually 130,1 and 138,2 Hz, and that also SR of the 2nd eigenvalue was relatively low – 5,2%. On the other hand, the 1st and 2nd eigenvalues of the 2nd stage pinion were relatively separated – 101,3 and 171,8 Hz respectively – and the 2nd eigenvalue had much higher SR – 11,3%. Vibrations of the 1st stage pinion showed big subharmonic component with amplitude exceeding 40 μm with frequency ranging from 122 to 126 Hz, thus very near to the calculated 1st eigenfrequency of the system. Although vibration level of the 2nd stage pinion was acceptable, dominant subharmonic component with frequency of about 100 Hz was also apparent in the vibration spectra, which was very near to the calculated 1st eigenfrequency of the 2nd stage pinion. As both pinions were already supported in tilting-pad bearings, it was not possible to solve the problem by replacing the bearings for the type with higher resistance to instability. The source of destabilizing forces was labyrinth seal with relatively big diameter and considerable pressure gradient, located at the backside of turbine impeller. Circumferential flow in the seal resulted in generation of strong tangential force component, so that the seal destabilized the rotor in a way similar to circular bearing. For the rotor stabilization it was therefore sufficient to suppress the circumferential flow component in region of labyrinth seal by barriers orientated in radial direction. After the modification the operation of the turbine was without problems in all regimes.

3.2 Instability of gas turbine with compressor

The next case of instability of „oil whip“ type was recorded with the rotor of gas turbine combined with axial compressor (Fig. 2). Two stage gas turbine with the output of 9 MW and operating speed of 6.000 rpm had impellers fixed by means of tooth system and screw connection to the rotor of 14 stage axial compressor. The rotor was supported in lemon bearings 160 mm in diameter.

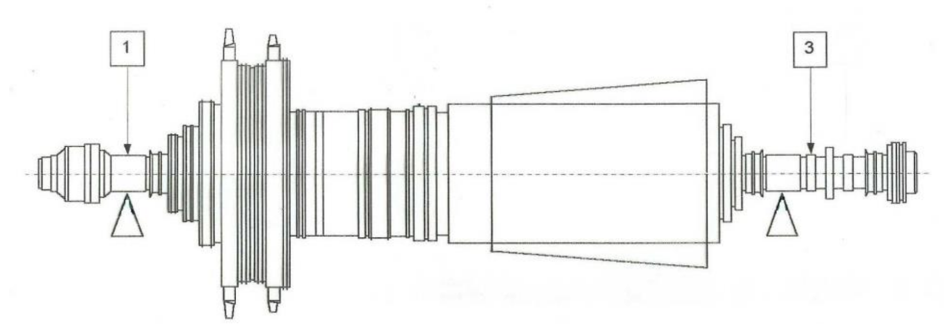


Figure 2: Scheme of gas turbine-compressor rotor

From measured amplitude-frequency characteristics in Fig. 3 it is apparent, that the rotor had the 1st bending critical speed in region around 1.150 rpm, which corresponds to 19 Hz. From about 2.200 rpm strong vibration of the rotor appeared with frequency of about 18 Hz, which is quite close to the measured 1st bending eigenfrequeny.

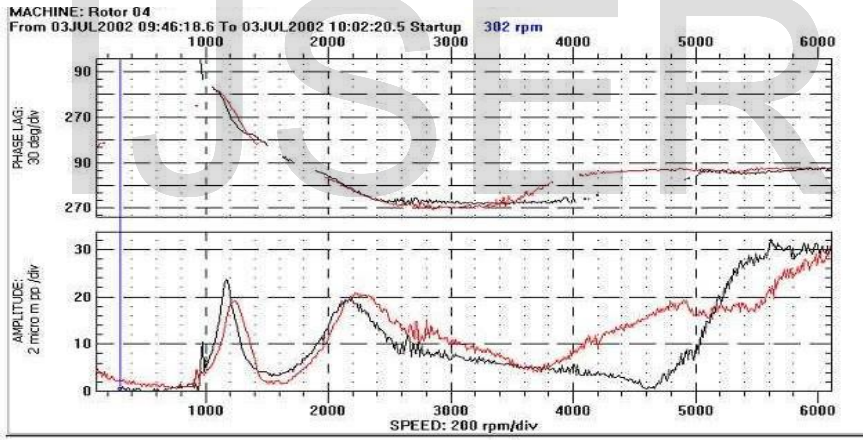


Figure 3: Amplitude-frequency characteristics of the rotor

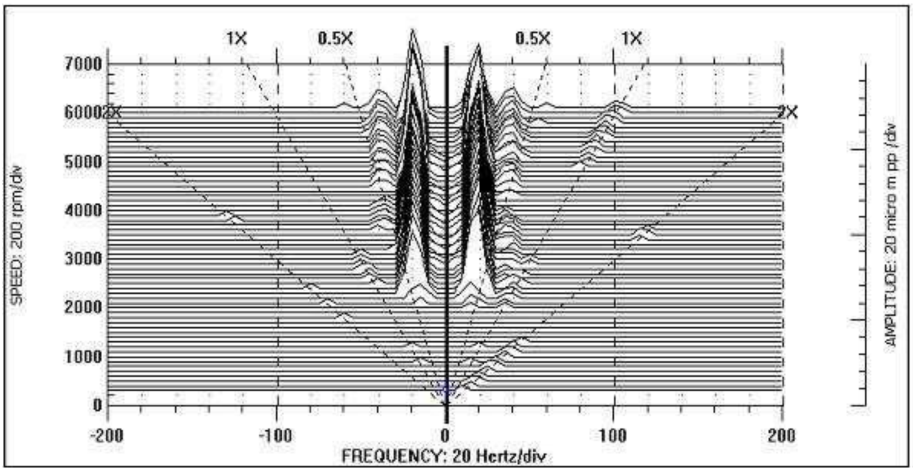


Figure 4: Full spectrum of rotor vibration

Dominating subharmonic vibration component is well apparent from Campbell diagram in Fig. 4. Maximum amplitudes of subharmonic vibrations exceeded 200 μm, which was unacceptable for rotor operation. Calculated 1st eigenfrequency of the rotor on the whole corresponds to measured critical speed. This example testifies generally recognized fact, that rotors with operating speed above double of the 1st eigenfrequency have often problems with stability. As the 1st bending mode had nodes practically in bearings, it was impossible to secure stability even by use of tilting–pad bearings. Solution should be searched in changing the rotor geometry.

3.3 Instability of high pressure turbine rotor

Probably the most complicated and persistent case of instability was encountered with high-pressure (HP) steam turbine rotor having relatively high operating speed for such a big machine - 5.500 rpm. HP turbine was coupled to low-speed low-pressure turbine and generator through reduction gearbox. The machine unit could be operated at full speed up to approximately 50% of its nominal output. With further increasing of turbine output severe vibrations with amplitude over 150 μm set in, resulting at shut-down of the machine due to exceeding the admissible level of vibrations (see Fig. 5, bearing No. 1 ϕ140 mm – left, bearing No. 2 ϕ180 mm - right). Dominant subharmonic component in neighbourhood of 37 Hz was found in rotor frequency spectra, which very well corresponds to the calculated as well as measured 2nd bending eigenfrequency of the rotor 2.250 rpm (see Fig. 6). The rotor was supported in lemon bearings with relatively low value of preload 0,5. At first sight it therefore looked as very simple case, which could be solved by use of bearings with better dynamic properties. Dynamic calculation of the rotor [4] indicated relatively low stability reserve (SR), but other machines with rotors of similar geometry and operating conditions with still lower calculated SR were operated quite without problems. In presented case the instability was caused apparently by effect of labyrinth seals, which were longer than in other trouble-free machines.

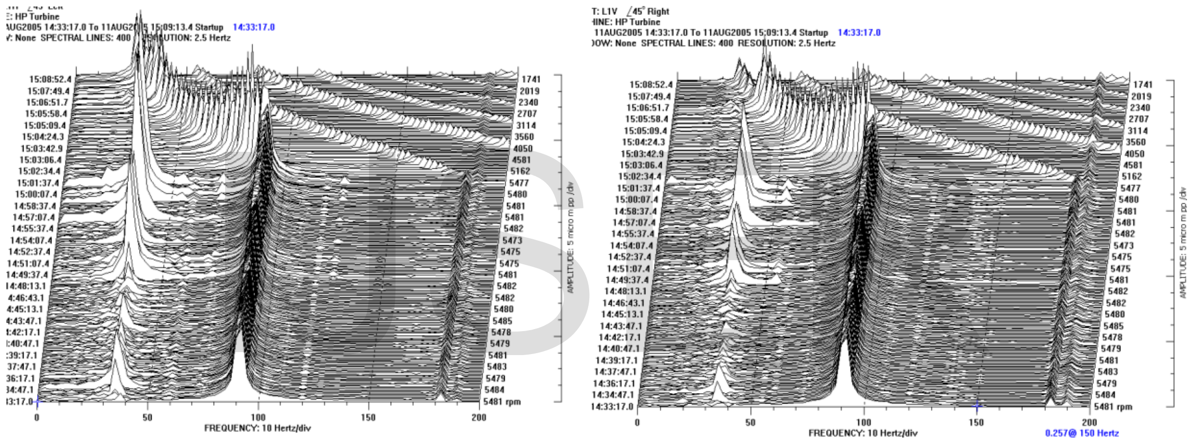


Figure 5: Spectrum of rotor vibration at onset of instability.

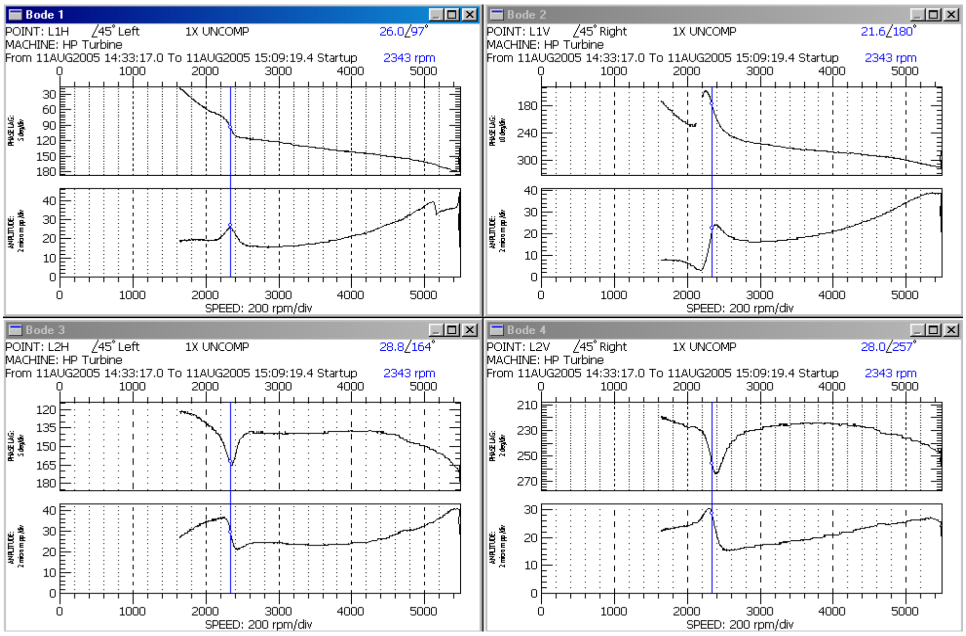


Figure 6: Phase- and amplitude-frequency characteristics of HP turbine rotor

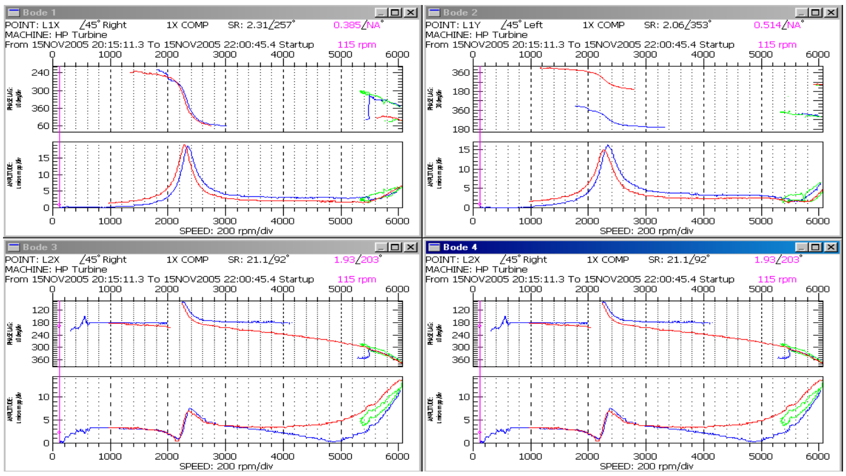
The first calculations carried out showed, however, that solution of this problem would not be as simple as expected. As well as in upper mentioned case of gas turbine with compressor, the nodes of unstable rotor eigenmode were in close vicinity of bearings and the influence of bearings on rotor dynamics was therefore very weak. Practically all existing types of bearings were included into analysis, but stability reserve was not practically affected by change of bearings (see Table 1 – variants 2-4)

Table 1: Stability reserve and logarithmic decrement of the two lowest rotor eigenvalues

variant	bearing type	1st eigenvalue		2nd eigenvalue	
		stability reserve (%) log. decrement	frequency (Hz)	stability reserve (%) log. decrement	frequency (Hz)
1	**lemon 1/2- original**	**6,58 / 0,207**	**28,8**	**7,45 / 0,234**	**35,7**
2	lemon 1/3 – reduced	9,73 / 0,306	26,7	3,88 / 0,122	36,1
3	offset halves	3,78 / 0,119	27,7	5,29 / 0,166	36,1
4	tilt-pad, 5p - load on pad	8,00 / 0,254	34,2	3,35 / 0,105	36,5
	with reduced static load				
5	lemon 1/2- original	unstable from 4000 rpm		13,0 / 0,408	30,2
6	**tilting-pad, 4p - load between pads**	**15,4 / 0,484**	**29,5**	**15,4 / 0,484**	**29,8**

It is apparent from the table, that if SR of one eigenvalue was slightly increased, SR of the other eigenvalue decreased. Apart from already mentioned fact, that bearings were practically in nodes of vibration, the reason of small effect of bearings was also relatively high specific load on bearings and from it resulting high bearing stiffness and consequently low damping. Logical search for improved damping led therefore to decreasing significantly bearing stiffness. This was, however, impossible to achieve only by change of bearing geometry, it had to be realized by reducing the bearing load. The turbine had symmetrically arranged partial admission, so that it was possible to change it to unsymmetrical one and to direct the resulting force against gravitation. By unsymmetrical admission the static load was decreased by more than 30% and bearing stiffness was reduced to less than one half of its original value. With common geometry bearings (e.g. variant 5 in Table 1) SR of the 2nd eigenvalue considerably increased, but that the 1st eigenvalue was even considerably decreased. The problem was at last solved by optimum configuration of tilting-pad bearings. Four-pad bearings with relatively big clearance and low preload brought about increase of SR of both lowest eigenvalus above 15%, i.e. to more than double the original value (logarithmic decrement of 0,48 as compared with original 0,21 and 0,23). Further improvement was achieved by increasing the bearing length, which decreased once more specific load and bearing stiffness, and increased damping by enlarged sliding surface. The final version of bearings with reduced manufacturing tolerances increased SR to about 30% (LD of 0,9), i.e. more than four times in comparison with original values. In spite of definitely positive results of calculation, certain distrust to this solution existed, resulting from persisting opinion, that rotor stability could be on the contrary improved by increasing the static load. This distrust was overcome when all other known means of rotor stabilization, which proved ineffective, were exhausted. One of them was retardation of circumferential component of flow in front of the inlet into labyrinth seals by means of so called „swirl brakes“. Although these modifications were made on all seals with higher pressure level, instability with original lemon bearings set in at practically the same turbine output as before seal modifications.

Figure 7: Phase- and amplitude-frequency characteristics of HP turbine rotor after change of bearings



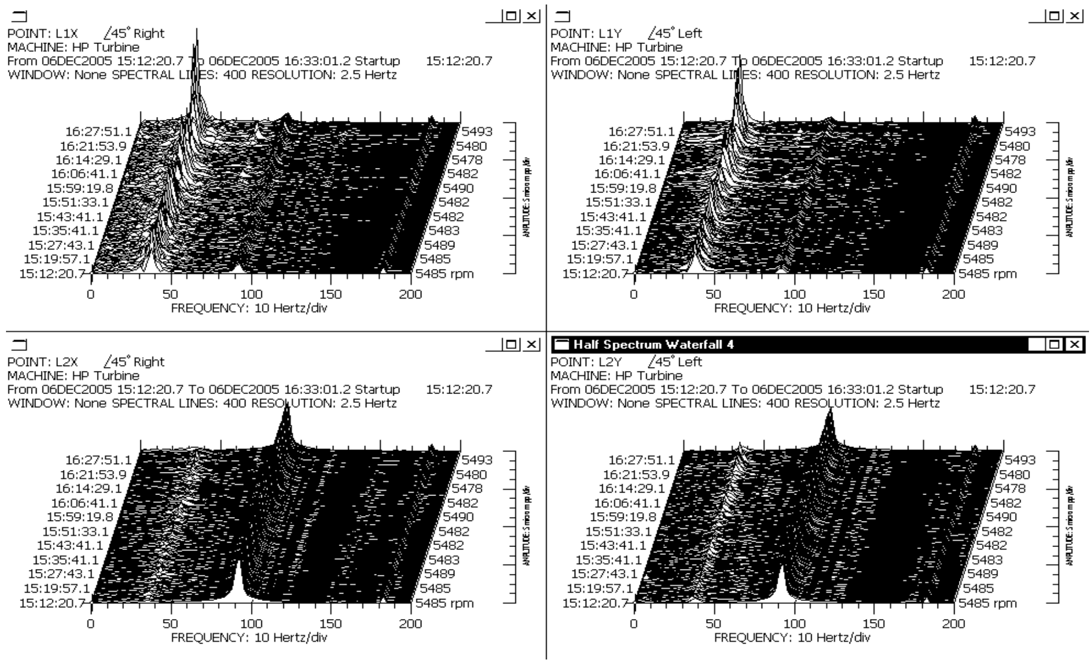


Figure 8: Spectrum of rotor vibration after change of bearings and modification of partial admission

After installation of above described tilting-pad bearings and modification of partial admission, the sub-harmonic frequency component did not quite disappear, but its vibration amplitudes were substantially decreased, so that the machine unit could be operated at full nominal output (see Fig. 8). Substantial difference in intensity of sub-harmonic vibrations in both bearings could be observed, which is evident from comparison of Figs. 5 and 8. While with original lemon bearings the sub-harmonic component was very big in both bearings, after change to tilting-pad bearings it is dominant only in bearing 1 (upper part of Fig. 8, two directions of measurement), while in bearing 2 (lower part of Fig. 8) it is still present, but relatively small. The difference between the bearings is apparently caused by the two factors, i.e. by greater bearing diameter and therefore higher damping in bearing 2, and by thrust bearing located near bearing 2, which naturally contributes to overall damping.

3.4 Instability of turbocharger rotor

Most of turbocharger (TCH) rotors are supported in floating-ring bearings with rotating bushings (Fig. 9, item 4). These bearing are undemanding from manufacturing point of view and at the same time have good dynamic properties resulting from high damping of the two oil film in series.

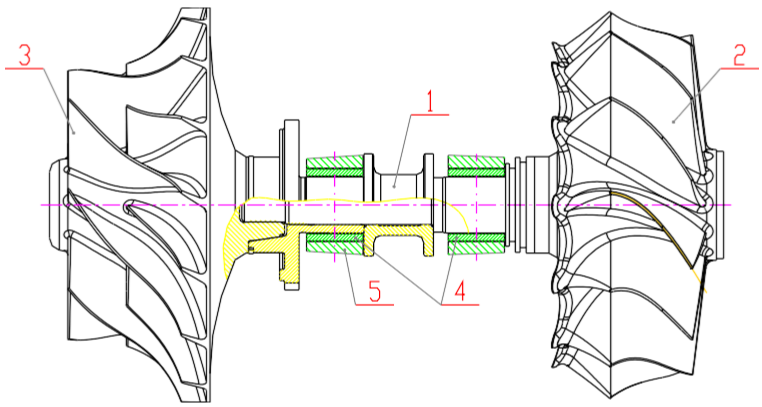


Fig. 9: Rotor of medium sized turbocharger

When we monitor relative vibrations of the rotor, which is necessary for provision of safe operation at least with prototype TCH, we could be observe rising instability of outer oil film. Tendency to outer oil film instability is manifested by vibration with frequency equal to one half of bushing rotational frequency. It is evident from vibration spectra of the rotor and bushing in Fig. 10, where frequency component with one half of bushing speed (about 110 Hz) is quite dominating. In this case the instability would not develop fully and vibration amplitudes remained limited.

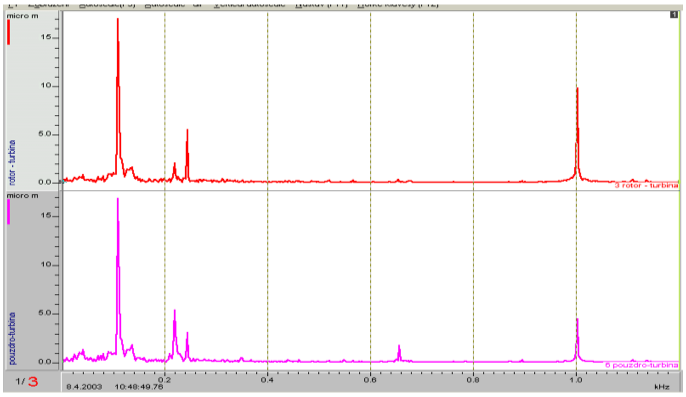


Fig. 10: Vibration spectra of the rotor (upper curve) and floating ring (lower curve) at 60.000 rpm

However, in some cases instability of outer oil film is fully developed and rotor vibrates practically within the whole bearing clearance (see Fig. 11) [5]. Top down are signals from: rotor – compressor side, rotor - turbine side, bushing – compressor side, bushing - turbine side. It is evident from presented vibration records, that both ends of the rotor as well as both floating bushings vibrate in-phase with subharmonic frequency equal to roughly one half of bushing speed (about 60 Hz) and with rotor amplitudes achieving more than 200 μm. Instability sets in already at minimum TCH operational speed of 20.000 rpm and persists trough the whole operating range up to maximum speed of 44.000 rpm. The rotor rotational speed 700 Hz can be hardly distinguished in frequency spectra of Fig. 11.

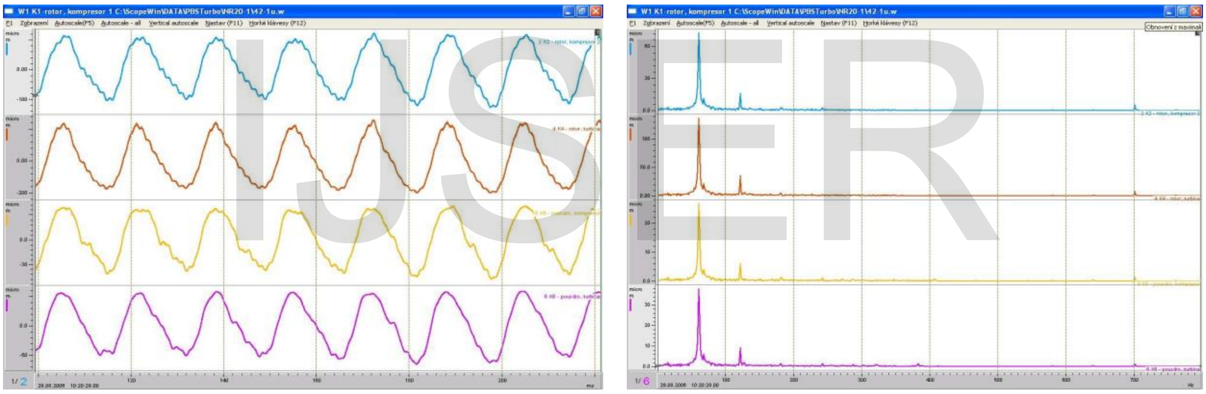


Fig. 11: Vibration signals and frequency spectra of TCH rotor with fully developed instability at 42.000 rpm

TCH operating at regime of outer oil film instability do not break down immediately only due to non-linear properties of the oil film. Stiffness of the film grows substantially at high eccentricity of the bushing, thus preventing contact between sliding surfaces. However, permanent TCH operation with these extremely high amplitudes of vibration is very dangerous, because relatively small change of operating conditions or penetrating of some impurity into the bearing gap can result in extensive damage of bearings and rotor.

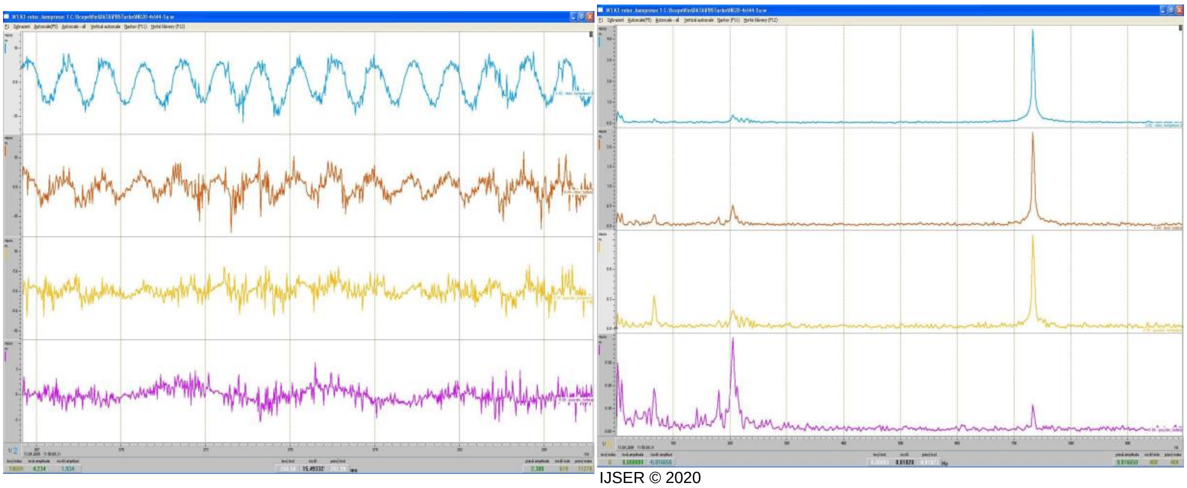


Fig. 12: Stable operation of the rotor with non-rotating bushings – 44.000 rpm

If instead of rotating bushings we use non-rotating bushings with lobbed inner geometry and high preload, TCH of the same type will run in stable regime. This is confirmed by records of relative vibrations and pertinent frequency spectra of the same rotor as in Fig. 11, but supported in stalled bushings with 3lobbed inner geometry and preload about 0,8, which is presented in Fig. 12. Maximum vibration double-amplitude of the rotor decreased from more than 200 μm to about 11 μm. Only small sub-harmonic vibration component is present with the amplitude lower than 0,5 μm.

4 CONCLUSIONS

Rotor instability is a very dangerous phenomenon and permanent operation in region of instability is not permissible. Instability of the rotor in sliding bearings cannot be in most cases detected by vibration measurement at bearing pedestals or machine casing. Some cases were reported, when fatigue failure of bearing lining took place, without recording increased vibration level at the machine casing. Fortunately absolute majority of big rotating machines is nowadays equipped with diagnostic system with relative vibration sensors, which are able to detect onset of instability immediately and to shut down the machine in time. However, there are plenty of small high-speed machines running without any diagnostic and it is therefore necessary to carry out rotor relative vibration measurement at least on their prototypes.

Suppressing the instability at already finished machines is always very troublesome and in some cases even impossible. That is why it is vital to provide calculations of bearing properties and rotor dynamics as accurate as possible. If calculation does not indicate possible instability and in spite of that instability sets in, it is necessary to look for reasons in so far insufficiently analysed effect of other factors, namely labyrinth seals. There is a great number of means for rotor stabilization, starting with change of bearings up to suppressing circumferential flow in seals. Standard interventions, though, need not be always effective, as is apparent from the case described in sect. 3.3. The situation must be considered from all aspects, not only according to simplified theses.

A new research project dealing with possibilities of rotor stabilization through external excitation of bearings was started. Test stand was designed and manufactured and initial analysis indicates possibilities of affecting rotor dynamics [6].

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