Some cases of specific behaviour of rotors – instability of operation

With growing speed of modern rotating machines we more and more often encounter cases of rotor instability. Rotors supported in sliding bearings exhibit two types of instability, which manifest itself by vibration with subharmonic frequency and big amplitude. Instability of the type „oil whirl“, with frequency dependent on rotational speed, is more likely encountered with rigid rotors, paradoxically namely with rotors in aerodynamic bearings. For instability of the type „oil whip“ (called also “shaft whip”), encountered with elastic rotors, is characteristic constant frequency of vibration with some – usually the lowest – system eigenfrequency. In practice can be relatively often seen cases, when instability of „oil whirl“ type in region of the 1\textsuperscript{st} system eigenfrequency passes over into „oil whip“ type (see Fig. 1).

![Fig. 1 Transition from “oil whirl” to “oil whip” instability](image)

Based on experience, the occurrence „oil whip“ instability can be expected in machines, which operate above double of the 1\textsuperscript{st} rotor eigenfrequency. It does not mean, that these machines will exhibit instability always, but if the machine is designed in such a way, it is necessary to pay great attention to dynamic rotor calculation, namely as regards stability. Influence of labyrinth seals and other potential destabilizing forces should be included into dynamic analysis.

Instability of high-speed rotors occurs due to big cross-coupling term of stiffness matrix. The exception present tilting pad journal bearings, the cross-coupling terms of which are at least two orders lower than principal stiffness terms. However, even rotors supported in tilting pad bearings can exhibit instability due to external destabilizing forces, occurring in gaps between stationary and rotating blades of turbines and axial compressors or in labyrinth seals.

Resistivity of the rotor against onset of instability is determined by magnitude of stability reserve, or logarithmic decrement; both these values express ratio of real to imaginary part of eigenvalue. Mutual relation of both quantities is given by relations:

\[ \text{stability reserve } \chi = -2 \frac{\Re(\lambda)}{\Im(\lambda)}.100 = \frac{(\log \delta).100}{\pi} \% , \]

where $Re(\lambda)$, and $Im(\lambda)$ ... real and imaginary part of eigenvalue,

**logarithmic decrement** $\log\delta = -2\pi \cdot \frac{Re(\lambda)}{Im(\lambda)} = \chi \cdot \pi/100$.

As values sufficient for ensuring rotor stability we usually consider $\chi \geq 15\%$, or $\log\delta \geq 0.3$. With substantial destabilizing effects of external forces even these values need not be sufficient. Rotor instability occurs usually with the lowest rotor eigenmode (with the lowest eigenfrequency). However, there was a case of instability with the 2nd eigenmode (further reported case 7), when it was very troublesome to suppress instability and when even values of $\chi \cong 30\%$ ($\log\delta \cong 0.9$) were not sufficient for full suppression of subharmonic vibration component.

Permanent operation in region of instability is unacceptable with regard to big vibration amplitudes. Onset of instability can be prevented by suitable bearing design and qualified dynamic analysis of the rotor. In spite of that, cases of instability of machines just put into operation or even machines operated for a long time with changed of operating conditions, are occasionally found in practice. Then it is necessary to suppress instability or at least reduce its exhibition to degree acceptable for long-term operation, while using the most economic means.

Further text contains some cases of instability, which the author encountered during his more then 40 years long practice and in its solution took part.
1. Radial compressor driven by high-speed gearbox:

Rotor of seven stage radial compressor driven by high-speed gearbox, supported in tilting pad journal bearings, had operating speed of 20,000 rpm. Strong vibration of gearbox pinion with frequency from 109.6 to 119.2 Hz and amplitude exceeding 300 µm occurred during tests above roughly 17,500 rpm (Fig. 1.1 and 1.2). Gearbox pinion was supported in lemon bearings and increased vibrations were not reported during test of separate gearbox.

As was ascertained by calculation, at 10% of input (it corresponds to idle run of compressor) the 1st pinion eigenfrequency is 105.6 Hz (6,338 rpm), thus very near to measured vibration frequency and calculated stability limit of the pinion was about 16,000 rpm. The problem was solved by replacing the lemon bearings of the pinion by tilting pad bearings.

Fig. 1.1 Spectrum cascade of gearbox pinion
Fig. 1.2 Amplitude-frequency characteristics of gearbox pinion

Resume: Apparently it was instability of the type “oil-whip“, characterized by vibrations with frequency equal to the lowest eigenfrequency of the system. However, combination “oil whirl“ – “oil whip“ cannot be quite excluded, because certain dependence on speed is apparent in Fig. 1.1.
2. Gas turbine integrated with compressor

Two-stage gas turbine with the output of 9 MW constitutes one rotor with axial compressor. Turbine impellers are fixed to the rotor by Hirt tooth system and central screw (Fig. 2.1). Rotor with operation speed of 6,000 rpm is supported in lemon bearings 160 mm in diameter. According to measured amplitude-frequency characteristics (Fig. 2.4), the 1st eigenfrequency of the rotor was around 16.7 Hz (1000 rpm), which was in quite good agreement with calculated results. From about 2,200 rpm (Fig. 2.2) severe vibration of the rotor with dominating subharmonic component occurred, the amplitude of which exceeded 200 µm and which died away at speed decreased to about 2500 rpm (Fig. 2.3). Dominating vibration component had constant frequency of about 18 Hz, thus very near to calculated 1st system eigenfrequency. Although amplitude-frequency characteristics were recorded from the signal filtrated to synchronous frequency, vibration with subsynchronous frequency is apparent even on this record (Fig. 2.4). Results of calculation show, that the rotor is stable, but with very small stability reserve. In given case it was impossible to secure stability even with tilting pad bearings, because the bearings affect the stability only marginally due to their vicinity to nodes of vibration. Each of the series of several rotors of this type had somewhat different behaviour, but instability developed with all of them. Different behaviour can be explained by difference in central screw preload.

Fig. 2.1 Rotor of gas turbine with axial compressor

Fig. 2.2 Spectrum cascade in left bearing – run-up
Some cases of specific rotor behaviour

Resume: It is apparently instability of the type „oil-whip“, characterized by vibrations with constant frequency equal to the lowest eigenfrequency of the system. For suppression of instability, however, is not sufficient to replace bearings for „more stable“ type, because bearing are located practically at the nodes of vibration. To suppress instability it would be necessary to reconstruct the whole rotor.
3. Two-stage expansion turbine

Modern high-speed turbines have impellers mounted directly on pinions of high-speed gearbox (see scheme in Fig. 3.1). Rotors of analyzed two-stage expansion turbine had the following operation speeds:

1\textsuperscript{st} stage – 24.000 rpm,
2\textsuperscript{nd} stage – 18.750 rpm.

Both pinions were supported in tilting pad bearings with injection of oil in front of the pad inlet edge. Strong vibration appeared at some operational regimes on the 1\textsuperscript{st} stage pinion (with amplitude over 40 µm) with frequency ranging from 122 to 126 Hz (Fig. 3.2). \textbf{Eigenfrequencies of the 1\textsuperscript{st} stage pinion}, determined by dynamic calculation, \textbf{were 130.1 and 138.2 Hz}, thus very near one to another and \textbf{both with very small stability reserve}. Pinion of the 2\textsuperscript{nd} stage has the \textbf{lowest eigenfrequencies 101.3 and 171.8 Hz}, thus at greater distance one from the other. However, stability reserve of the 1\textsuperscript{st} eigenvalue was even lower, than that of the 1\textsuperscript{st} stage pinion. Vibrations of the 2\textsuperscript{nd} stage pinion were in allowed limits, nevertheless frequency component of about 100 Hz could be found in frequency spectrum, which was close to the 1\textsuperscript{st} pinion eigenfrequency. The reason of increased vibration of the 1\textsuperscript{st} stage was destabilizing influence of labyrinth seal at rear side of impeller. The seal with high pressure gradient and big diameter generated great destabilizing forces, which together with low stability reserve and nearness of both lowest eigenvalues led to pinion instability. The problem was solved by installation of barriers braking circumferential flow at the rear side of the impeller. Other possibility of suppressing or reducing the excitation effect of labyrinth seal is so called antiswirl blowing, i.e. introducing the flow of gas directed against rotor rotation, which also disturbs peripheral flow component in the seal.

![Fig. 3.1 Scheme of two-stage expansion turbine](image-url)
Fig. 3.2 Vibration spectra of the 1st stage of expansion turbine – original conditions

Fig. 3.3 Vibration spectra after installation of barriers at the rear of impeller
As can be seen from frequency spectra in Fig. 3.3, subharmonic frequency around 130 Hz did not disappear after installation of barriers, but its amplitude was reduced to about 14 µm, which was acceptable for turbine operation.

Above mentioned case lead up to reconsideration of this type of turbine design, which have bigger impeller overhang than bearing distance (see Fig. 3.4).

![Fig. 3.4 Typical rotor of “gear-box” turbine](image)

Optimizing study [1] has shown, that stability reserve can be much influenced by geometry of the rotor. As can be seen from Tab. 3.1, significantly higher stability reserve can be reached even with very high operation speed (much higher than was encountered with problematic two stage expansion turbine).

Tab. 3.1 Stability reserve of the 1st and 2nd eigenvalue of different types of realized turbines and optimized rotors

<table>
<thead>
<tr>
<th></th>
<th>Stability reserve of 1st / 2nd eigenvalue (%)</th>
<th>Operation speed (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>two stage expansion turbine – the 1st stage</td>
<td>4,2 / 5,2</td>
<td>24,000</td>
</tr>
<tr>
<td>two stage expansion turbine – the 2nd stage</td>
<td>3,6 / 11,3</td>
<td>18,750</td>
</tr>
<tr>
<td>1 stage expansion turbine</td>
<td>4,9 / 8,3</td>
<td>22,580</td>
</tr>
<tr>
<td>1 stage steam turbine</td>
<td>7,4 / 7,5</td>
<td>26,270</td>
</tr>
<tr>
<td>optimized rotor – worst variant</td>
<td>2,9 / 4,2</td>
<td>33,000</td>
</tr>
<tr>
<td>optimized rotor – best variant</td>
<td>9,0 / 11,2</td>
<td>33,000</td>
</tr>
</tbody>
</table>

Resume: This case belongs also to instabilities of “oil-whip“ type. Due to strong destabilizing influence of labyrinth seal at overhang end of rotor, stability could not be ensured even by use of tilting pad journal bearings.

4. High-speed steam turbine

The machine design is similar to that in preceding case, i.e. turbine with the impeller mounted on pinion of high-speed gearbox. The difference is in the fact, that it was steam turbine with only one impeller. Turbine, which has operation speed about 19,000 rpm, exhibited during run-up to operation speed substantially increased vibration level with marked subharmonic frequency component. By rotor dynamic calculation, considering stiffness and damping of the oil film, it was established, that subharmonic frequency component around 150 Hz is close to the 1\textsuperscript{st} bending critical speed of the rotor, the eigenvalue of which has very low stability reserve. Stability reserve rapidly decreased with growing turbine output and with full output results of calculation already indicated rotor instability.

Fig. 4.1 Frequency spectra of the rotor with increasing speed

Fig. 4.1 documents presence of subharmonic frequency component in vicinity of operation speed. Consistency with results of calculation, indicating strong dependence of stability reserve
on turbine output, is documented in Figs. 4.2 and 4.3, where record from the bearing at turbine side is on the left side. In idle operation subharmonic component was not present in vibration spectra even at maximum speed of 19,140 min\(^{-1}\) (Fig. 4.2). However, in conditions nearing full load this component became in the bearing at turbine impeller side quite dominant and arisen vibration level led to shut down of the machine by vibro-diagnostic system (Fig. 4.3).

Fig. 4.2 Frequency spectra of the rotor at operation speed – idle run

Fig. 4.3 Frequency spectra of the rotor at operation speed with load

The reason of increased vibration level with dominant subharmonic frequency component was, as well as in preceding case, labyrinth seal at the back of impeller. The seal was again located at relatively big diameter and had high pressure gradient, which increased with growing machine output. Labyrinth seal therefore affected rotor by destabilizing effects, similarly as cylindrical bearing, which has at low load conditions cross-coupling stiffness term one order higher than principal ones.

The pinion was supported in tilting pad journal bearings, so that it was not possible to suppress instability by use of bearings with better dynamic properties and it was necessary to reduce external excitation, by which the instability was evoked. Problem was solved by inserting the barriers between impeller backside and turbine casing; the barriers prevented circumferential flow in this area, so that the steam entered labyrinth seal with low circumferential velocity component.

Resume: As well as in preceding case the instability was of „oil-whip“ type. Due to destabilizing effect of labyrinth seal at overhang end of rotor, stability could not be ensured even by use of tilting pad journal bearings. The reason of instability was low stability reserve of the 1\(^{st}\) eigenvalue together with destabilizing forces originating in labyrinth seal.
5. Test rotor

Test rotor (Fig. 5.1) was used to demonstrate better stability properties of offset-halves bearings in comparison with lemon bearings. According to calculation, with bearings located outside rotor discs, the rotor should be unstable in both bearing types, while the difference between both types was not significant. Location of bearings between discs, which would accentuate positive influence of offset halves bearings, could not be realized due to impossibility to pass critical speed. Greater anisotropy of lemon bearings was manifested by two nearby resonance peaks, as is apparent from amplitude-frequency characteristics of the rotor in lemon (Fig. 5.2) and offset (Fig. 5.3) bearings.

With the rotor in lemon bearings instability of „oil whip“ type set in at the speed of 5.160 rpm (Fig. 5.4) and at run-down it persevered to 3.200 rpm (Fig. 5.6). The same rotor supported in offset halves bearings began to show symptoms of instability at speed of 5.400 rpm (Fig. 5.5), but during run-down instability disappeared already at about 4.600 rpm (Fig. 5.7), thus much earlier. Frequency of subharmonic vibrations of 36.7 Hz (2.200 rpm) quite well corresponds to calculated 1st eigenfrequency of the system.

Advantages of offset halves bearings in comparison with lemon ones, consisting in better stability properties, were demonstrated by somewhat higher limit of occurrence of instability, but first of all by much earlier disappearance of instability.

![Fig. 5.1 Test rotor](image-url)
Some cases of specific rotor behaviour

Fig. 5.2 Amplitude-frequency characteristics of rotor in lemon bearings

Fig. 5.3 Amplitude-frequency characteristics of rotor in offset halves bearings
Some cases of specific rotor behaviour

Fig. 5.4 Run-up of test rotor in lemon bearings

Fig. 5.5 Run-up of test rotor in offset halves bearings
Some cases of specific rotor behaviour

Fig. 5.6 Run-down of test rotor in lemon bearings

Fig. 5.7 Run-down of test rotor in offset halves bearings
Some cases of specific rotor behaviour

Rotor in lemon bearings exhibited in region around 2,300 rpm **counter-rotating precession** in one measuring plane. It is evident from Fig. 4.8, which shows also the situation preceding origin of counter-rotating precession. Counter-rotating precession, which was predicted also by calculation, is relatively rare phenomenon with rotors excited only by unbalance. It occurs only in specific conditions, when there is a big difference between major and minor axes of journal trajectory ellipse.

![Journal trajectories at 2.120 rpm (co-rotating precession) and at 2.300 rpm (counter-rotating precession).](image)

**Resume:** Test rotor suffers from instability of „oil whip“ type, because rotor vibrates with constant frequency. Much earlier disappearance of instability at run-down proved better dynamic properties of offset halves bearings in comparison with lemon ones.
6. High-speed steam turbine

Steam turbine manufactured in 1972, with operating speed of 10,000 rpm, run until repair in 2003 without any problems. Rotor was supported in 4 lobbed symmetrical bearings with static load directed to oil groove. After repair in 2003 high level of vibrations appeared in region of operation speed (from about 9,600 rpm) and distinct frequency component with one half of rotational speed appeared in relative vibration spectra (Fig. 6.1). This frequency component is well apparent also on journal centre trajectory (Fig. 6.2). Sudden grow of vibration amplitude is apparent from the record of the diagnostic system (Fig. 6.3).

Both journal bearings were replaced during repair, but instability occured henceforth from about 9,600 rpm. Explanation can be looked for in the fact, that the system worked from beginning of its operation near stability limit and due to small change of parameters (e.g. wear of bearing journals, or bearing diameter at upper tolerance limit) the stability limit was somewhat lowered. The machine operation can be also affected by operation of a new turbine with the same speed, which is located nearby. The single turbine runs stable, instability appears when it is coupled to gearbox, which reduces speed to 1,500 rpm; the gearbox pinion vibrations, however, were quite acceptable. With increasing output the vibration amplitude decreased, which can be explained by partial admission of the turbine; with increasing output therefore stabilizing force of constant direction grows. Situation was made worse by decreasing misalignment of turbine rotor and gearbox pinion from 0,5 to 0,2 mm (probably the bearing load on coupling side was somewhat decreased). According to manufacturer calculations and also on the basis of measured run-down, the 1st eigenfrequency of the system is around 4,700 rpm (in the bearing at rotor free end). The machine thus operates above double of its 1st eigenfrequency and belongs therefore between potentially problematic ones.

Fig. 6.1 Spectrum cascade of machine run-up
Some cases of specific rotor behaviour

Fig. 6.2 Journal centre trajectory

Fig. 6.3 Record from diagnostic system during run-up to operating speed (top down: speed, vibration of bearing at coupling side, vibration of bearing at rotor free end)

Resume: With the most probability it is instability of „oil-whirl“ type, though dependence on speed is indistinct (change of speed is small). The rotor could be apparently stabilized by replacing 4lobbed bearings by tilting pad ones, which should be confirmed by rotor dynamic calculation.
7. Instability of high pressure steam turbine

Steam turbine with the output of 90 MW consisted from high pressure (HP) section with operating speed of 5,500 rpm and combined medium and low pressure section with speed of 3,600 rpm, which were connected by elastic coupling and gearbox. Rotor of HP section was supported in lemon bearing 140 and 160 mm in diameter. At the output above 40 MW the rotor lost stability, which was manifested by marked subharmonic component with frequency about 38 Hz (Fig 7.1, left – record from left bearing 140 mm diameter, right– record from right bearing 160 mm dia).

Fig. 7.1 Onset of instability with increasing output and its immediate disappearance after turbine shut-down by vibro-diagnostic system

Trajectory of journal centre and signals of relative vibrations in time domain are presented in Fig. 7.2

Fig. 7.2 Journal centre trajectories at the regime of instability
Some cases of specific rotor behaviour

Fig. 7.3 Amplitude-frequency characteristics of the rotor with original lemon bearings

Frequency component 38 Hz is close to the calculated 2nd bending eigenfrequency of the rotor, with nodes very close to bearings. Damping effect of bearings and their influence on rotor stability is therefore relatively insignificant. Stability reserve (SR) and logarithmic decrement (LD) of the 1st and 2nd eigenvalue were relatively small. It is apparent from Table 7.1, which compares calculated values of SR and LD for different bearing types.

Tab. 7.1 Stability reserve of lowest rotor eigenvalues with different bearing support types

<table>
<thead>
<tr>
<th>var.</th>
<th>bearing type</th>
<th>the 1st eigenvalue</th>
<th>the 2nd eigenvalue</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>SR (%) / LD</td>
<td>frequency (Hz)</td>
</tr>
<tr>
<td>1</td>
<td>lemon 1/2 – original</td>
<td>6,58 / 0,207</td>
<td>28,8</td>
</tr>
<tr>
<td>2</td>
<td>lemon 1/3 – narrow</td>
<td>9,73 / 0,306</td>
<td>26,7</td>
</tr>
<tr>
<td>3</td>
<td>offset halves</td>
<td>3,78 / 0,119</td>
<td>27,7</td>
</tr>
<tr>
<td>4</td>
<td>tilting pad 5pLOP</td>
<td>8,00 / 0,251</td>
<td>34,2</td>
</tr>
<tr>
<td></td>
<td>after reduction of load</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>tilting pad 4pLBP</td>
<td>15,4 / 0,363</td>
<td>29,5</td>
</tr>
<tr>
<td>6</td>
<td>tilting pad – greater l/D</td>
<td>28,8 / 0,907</td>
<td>28,6</td>
</tr>
</tbody>
</table>

5pLOP … classic bearings with 5 pads, load oriented to lower pad
4pLBP … 4 pad bearing, load oriented between pads, bearing support optimized for asymmetrical partial admission oriented upwards

Table 7.1 presents values for static load due to rotor own weight, i.e. for symmetrical partial admission, for variants 1 to 4. It can be seen from the table, that usually efficient methods of increasing stability, as increased preload, increased specific load or use of bearing with higher resistance to instability, have only marginal significance and for the 2nd eigenvalue they bring in
all cases even worse results in comparison with original state. To maximize damping effect of bearings it was necessary to decrease significantly their stiffness. Reduction of stiffness, however, could not be achieved by change of bearing geometry. It was necessary to decrease static load by replacing the symmetrical partial admission for asymmetrical one, with resultant force oriented against gravitational forces. Change of partial admission was considered with all bearing types, but with common bearing geometries it did not bring the required effect. The only bearing type, which increased stability reserve, was 4pad bearing with small value of preload and great clearance. It increased stability reserve to more than double the original value (var. 5 in Tab. 7.1). Further improvement was achieved by increasing the bearing length and reducing manufacturing tolerances. In the final phase the logarithmic decrement was increased to about 0.9 (RS around 28%), thus to more than quadruple of the original value (var. 6 in Tab. 7.1). Modification of labyrinth seals was carried out simultaneously with change of admission and bearings, in order to minimize their destabilizing influence.

Modifications carried out did not succeed in full suppression of instability, but enabled operation of machine unit. As is apparent from Fig. 7.4, with the output increased above 50% of nominal value, dominant frequency component 38 Hz appears in frequency spectrum of left bearing (record in upper part of Fig. 7.4), which is close to the 1st bending eigenfrequency of the rotor. Tilting pad bearings are almost isotropic, so that the original 1st and 2nd eigenfrequency merge – see amplitude-frequency characteristics in Fig. 7.5. However, amplitude of subharmonic component is much lower than with original configuration of the rotor and with further increasing of output it even decreases. As is apparent from records in lower part of Fig. 7.4, subharmonic component in right bearing is much smaller than in left one. It can be explained by bigger bearing dimension resulting in higher damping, and by the effect of thrust bearing, located nearby right journal bearing, which contributes to damping of vibrations.

Fig. 7.4 Spectrum cascade after redirecting of admission and change of bearings
Some cases of specific rotor behaviour

Fig. 7.5 Amplitude-frequency characteristics of the rotor after redirecting of admission and change of bearings

Resume: Also in this case it is instability of „oil-whip“ type with dominating subharmonic frequency equal to the 2\textsuperscript{nd} rotor eigenfrequency. Due to strong destabilizing influence of labyrinth seals and nodes of vibrations near to bearing locations it was not possible to suppress instability by any common method. The instability was suppressed by decreasing the bearing stiffness through significant reduction of static load and using special geometry of tilting pad bearings.
8. Radial compressor

Rotor of the six-stage radial compressor with operating speed of 13,600 rpm was supported in tilting pad journal bearings. The rotor was trouble-free from the standpoint of critical speeds, however, the 1st two rotor eigenvalues exhibited low stability reserve, namely 4.5 % and 3.7 %. Previously delivered machines with similar values of damping ratio operated more or less without problems, but in given case strong vibrations with subharmonic frequency of about 42 Hz appeared at the speed around 11,900 rpm. Amplitude of the rotor relative vibrations exceeded 80 µm (Fig. 1), so that it was not possible to achieve operation speed. Subharmonic frequency of 42 Hz corresponds to the calculated and actual 1st eigenfrequency of the rotor, which is evident from amplitude-frequency characteristics in Fig. 8.1.

![Fig. 8.1 Amplitude-frequency characteristics of the rotor](image)

![Fig. 8.2 Spectrum cascade showing onset of instability indicated by subharmonic frequency](image)
Some cases of specific rotor behaviour

Fig. 8.3 Frequency spectra of the rotor at 11.200 rpm (left) and 11.900 rpm (right)

Figs. 8.2 and 8.3 illustrate sudden increase of vibration amplitude of subharmonic frequency component at around 11.900 rpm, which led to instant shut-down of the machine. Quite similar were further attempts to achieve operation speed. The machine worked in test conditions practically without any load. Due to relatively high pressures in labyrinth seals further increase of destabilizing forces and intensifying of instability symptoms could be expected with real operating conditions.

As the rotor was already supported in tilting pad bearings, it was not possible to use standard means, i.e. utilize bearings with better stability properties. The only viable solution was therefore to optimize tilting pad bearing properties in order to achieve substantial increase of stability reserve. Vibration nodes of the 1st bending mode are practically in bearing locations, therefore it was necessary to decrease as much as possible bearing stiffness, in order to increase journal excursions in bearings and by that also damping of vibrations. After checking up of several variants as optimum proved again bearing geometry with four pads and static load oriented between pads (LBP). In bearings with above-standard clearance and minimum preload the stability reserve of the 2nd eigenvalue increased from original 3.7 % (log. decrement 0.12) to 14.1 % (log. decrement 0.44). Rotor stability is thus preserved even when destabilizing effect of labyrinth seals is included into rotor dynamic analysis.

Resume: Classical case of „oil-whip“ type instability with subharmonic frequency equal to the lowest eigenfrequency of the rotor, resulting from low stability reserve of the two lowest eigenfrequencies. Substantial increase of stability reserve was achieved by use of 4pad bearings with above-standard clearance and minimum preload, analogous to previous case of high-pressure turbine section.
9. Turbocharger rotor supported in floating ring bearings

Most of turbocharger (TCH) rotors are supported in rotating floating bushings (Fig. 9.1, detail 4).

These bearings are undemanding from manufacturing point of view and have favourable dynamic properties, resulting from high damping of two oil films in series. When measuring relative rotor vibrations, certain tendency to instability of outer oil film is apparent in some cases, which is indicated by vibration frequency equal to one half of the bushing rotational frequency. This is apparent from frequency spectra of the rotor and bushing in Fig. 9.2, where frequency component equal to one half of bushing speed (about 110 Hz) dominates in frequency spectra of the rotor and bushing. Fully developed instability in this case does not arise due to already mentioned high damping and also because exciting frequency from the rotor does not support this instability.

However, in some machines the instability is fully developed and the rotor vibrates in the range of whole bearing clearance (Fig. 9.3). Immediate failure does not take place only due to non-linear properties of oil film, the stiffness of which significantly grows at high eccentricities of journal and bushing. However, permanent operation of TCH with this vibration level is dangerous, because even relatively small change of conditions or penetration of some impurity into bearing can result in extensive damage of the rotor and bearings.
Some cases of specific rotor behaviour

Fig. 9.3 Fully developed instability of outer oil film at 42.000 rpm
(rotating bushings – maximum vibration amplitude 140 µm)

Top down in Figs. 9.3 and 9.4 are shown signals: rotor –compressor side
rotor –turbine side
bushing - compressor side
bushing - turbine side

It is apparent from the record in Fig. 9.3, that both ends of the rotor and both floating bushings vibrate in-phase by subharmonic frequency equal roughly to one half of bushing speed (about 60 Hz) with amplitude reaching practically the whole bearing clearance. Rotor frequency of rotation (700 Hz) is hardly discernable in frequency spectra.

To stabilize the rotor it was necessary to replace rotating bushings by non-rotating (stalled) bushings with lobbed (non-circular) inner geometry. With stalled bushings the damping in outer film is generated solely by squeeze effect of the oil film. If the bushing does not rotate, the instability of outer film cannot occur and vibration amplitudes are reduced by one order. It is demonstrated in Fig. 9.4, which shows records of relative vibrations of the same rotor as in Fig. 9.3, but with stalled bushings with three-lobbed inner geometry. Maximum amplitude was reduced by the change of bearings from 140 to about 11 µm.

Fig. 9.4 Stable operation of the rotor with stalled bushings at 44.000 rpm
(maximum vibration amplitude at turbine side 10.7 µm)

Resume: Rotors in rotating floating bushings exhibit either indication of potential instability or in rare cases fully developed instability of outer oil film. Due to strong damping and prevailing excitation with frequency not corresponding to this type of instability, the instability could, but need not develop (the excursions will be limited). Permanent operation with fully developed instability is dangerous and should be avoided.
Conclusions

Rotor instability is a very dangerous phenomenon and permanent operation in region of instability in unacceptable. In most cases instability of rotors in sliding bearings cannot be disclosed by measurement of vibrations at bearing pedestals or machine casing. Cases of bearing lining fatigue damage due to rotor instability were reported without recording of increased vibration level at machine casing. Luckily the most big rotating machines are now equipped with diagnostic systems with relative rotor vibration sensors, which are able to disclose immediately onset of instability and shut down the machine. However, a whole lot of small high-speed machines with sliding bearings, which are not equipped with any diagnostics, is still in operation. In these cases it is unconditionally necessary to ensure measurement of rotor vibration at least on prototype machines.

Suppressing the instability on already finished machine is always very complicated and in some cases even impossible. The most accurate calculation of bearing properties and rotor dynamic analysis are therefore needed. If the calculation does not indicate instability and in spite of that the instability appears, it is necessary to look for other factors, which were unfortunately so far little investigated, e.g. influence of labyrinth seals. A number of possibilities for rotor stabilization exist, starting with change of bearings and ending with restricting circumferential flow in seals. However, standard procedures need not be always effective and the whole situation should be viewed in a complex manner, not only on the basis of simplified thesis.