SOLUTION OF SUBSYNCHRONOUS VIBRATION PROBLEMS IN RADIAL FLOW HIGH-SPEED TURBINES

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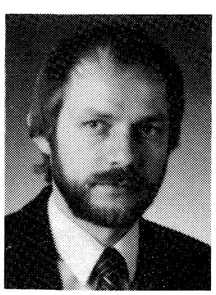

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ABSTRACT

Experience, theoretical and practical, investigations are presented on two radial flow high-speed turbines, which showed unacceptable high subsynchronous vibrations during operation onsite.

The first example deals with a two-stage integrally geared turbine for energy recovery, with generator brake installed in a chemical plant. Measurements and calculations ruled out that the subsynchronous vibrations were caused by the bearings. Measurements of pressure at the inlet and outlet of both stages and in the inter-stage pipe hinted at an acoustic excitation in the annular space between impeller and adjustable nozzle ring. The acoustic excitation and resulting subsynchronous vibrations could be reduced by installing a few fixed nozzles, thus interrupting the vaneless space to the extent that safe operation of the gear turbine is possible adhering to the relevant guidelines.

The second example is a compressor-loaded turbine with bilaterally overhung radial impellers employed in an air separation plant. Non-reproducible, subsynchronous vibrations occurred, due to variations of manufacturing tolerances of bearing geometry, labyrinth seals and gap geometries representing the main causes. The results of parameter studies in the bearing and sealing zones showed that the damping effect of the bearings can be eliminated by the influence of the labyrinths. Thus, the rotor becomes unstable. The rotor was stabilized by the installation of swirl brakes and antiswirl sleeves. After that, subsynchronous vibrations ceased to exist resulting in stable and safe operation.

INTRODUCTION

Although, today, rotordynamic design of turbomachines has become standard, directed in particular at optimizing the damping effect of journal sleeve bearings, subsynchronous vibrations are experienced in production turbomachines. Sometimes, these vibrations lead to unacceptable high and fluctuating vibration levels, so that it is imperative to search for associate reasons and introduce appropriate action for correction.

Subsynchronous vibrations for example are,

due to self-excitations

- instability caused by bearings
- aerodynamic swirl excitation
- destabilizing forces in seals
- external elasticity
- internal friction

due to forced-excitations

- seal rubs

• cracked shafts

• dynamic forces coming from the process gas or from the gears

The subsynchronous vibrations experienced in two different turbomachines, one integrally geared turbine and one booster-turbine, were represented and analyzed.

GEAR TURBINE

The integrally geared turbine (Figure 1) expands natural gas in two stages.

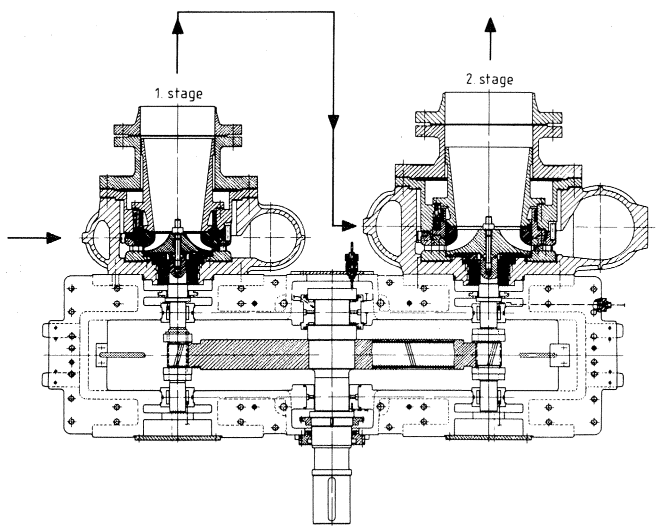

Figure 1. Gear Turbine.

The design conditions of this gear-turbine are

• maximum power	2.5	MW	(3,400	hp)
• bull gear speed	1800	rpm	(30	Hz)
• first stage pinion speed	27000	rpm	(450	Hz)
• second stage pinion speed	21975	rpm	(366	Hz)
• inlet pressure	23.4	bar	(340	psia)
• outlet pressure	5.1	bar	(74	psia)
• inlet temperature	422	K	(300	°F)
• outlet temperature	374	K	(215	°F)

The pinion shafts of each stage have an overhung radial impeller and are supported in tilting pad journal bearings with five pads. One of the two lobed bearings of the bull gear shaft is a combined journal thrust bearing with fixed thrust sliding faces. The thrust forces are transferred from the pinion shafts to the bull gear by thrust collars. The power of the turbine is controlled by adjustable inlet nozzles on both stages.

COMPRESSOR TURBINE

The compressor-loaded turbines are used in air separation plants. The compressor is driven directly by the turbine. The speed is automatically adjusted according to the power balance between the turbine and the compressor and adapts itself to power changes.

The cross section of the compressor turbine (Figure 2) shows the turbine at the left hand side. The process gas streams through the turbine impeller from outside to inside and leaves the impeller in axial direction. The turbine is fully admitted. The adjustable nozzles are located around the impeller and control the mass flow and the pressure drop in the turbine near optimum for production.

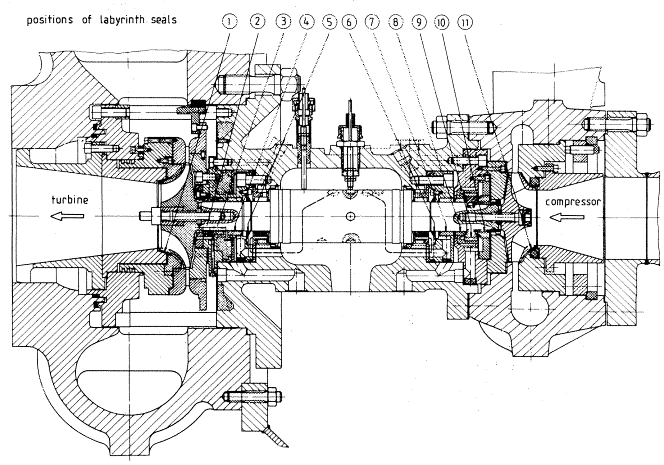

Figure 2. Compressor Turbine.

The compressor is arranged on the right hand side of the cross section. The compressor impeller is overhung, too. The gas flows from inside to outside. The bearing housing with the sliding bearings, the seals and the thrust load balancing system, are located between the turbine and compressor. The seals consist of labyrinths with carbon bushes. On the turbine side, buffer gas is fed in, the pressure of which is a little higher than that on the back side of the turbine impeller. In the direction of the bearing, there is an intermediate pressure chamber, via which the leakage gas is vented with a pressure of about 6.0 bar (85 psia). The sliding bearing area has atmospheric pressure.

On the compressor side, there is a balance piston, by means of which the thrust load onto the thrust bearing can be reduced. The balance piston is pressurized on one side, with the compressor inlet pressure and on the other side with the turbine outlet pressure.

The sliding bearings are oil lubricated. The loaded thrust bearing is located at the turbine side. The thrust bearings are tapered land fixed geometry bearings, whereas the journal bearings are tilting pad bearings with five pads. The oil is fed to the sliding faces via an annular channel and bores in the bearing bush. The oil is throttled at the outlet so that the bearing is flooded.

Typical design conditions of the compressor loaded turbine for the given frame size are

• power	up to 1200 kW	(1,600	hp)
• speed	up to 34000 1/min	(570	Hz)
• impeller diameter	about 190 mm	(7,5	inch)

for the turbine

• inlet pressure	about 55 bar	(800	psia)
• outlet pressure	about 6 bar	(85	psia)
• inlet temperature	about 285 K	(54	°F)
• outlet temperature	about 175 K	(−145	°F)

for the compressor

• inlet pressure	about 36 bar	(520	psia)
• outlet pressure	about 55 bar	(800	psia)

With other frame sizes other operating ranges are obtained.

LATERAL VIBRATION ANALYSIS AND HIGH SPEED ROTOR DESIGN

For high-speed rotors, an extensive lateral vibration analysis is carried out in most cases. For modelling of the rotor and bearing systems carefully tested computer programs are in use.

Substitute System of the Rotor and Dynamic Bearing Coefficients

For the substitute system, the rotor is divided into shaft sections with a constant cross section [1]. Further masses and their mass moments of inertia can be added. Bearing forces, other additional forces, stiffness, damping and forced exciting forces can act at the end of each section. At the journal bearing positions, the properties of the sliding bearings are linearly approximated with the stiffnes and damping coefficients [2]. Additional external elasticity, damping and masses can be added in the bearing positions, which are modelled as complex substitute coefficients for the whole bearing.

The substitute systems of both rotors, with the overhung masses of the impellers, the journal bearing and the assumed unbalance positions are shown in Figures 3 and 4.

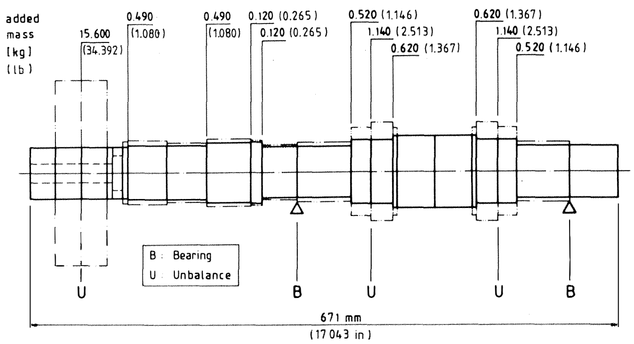

Figure 3. Substitute System of Pinion Shaft of Gear Turbine (second stage).

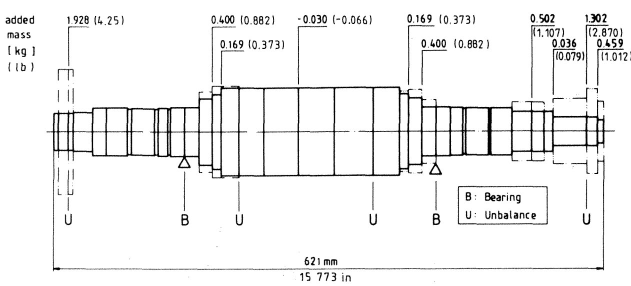

Figure 4. Substitute System of Compressor Turbine Rotor.

Critical Speed Map

The critical speed map allows a quick overview on the position of the critical speeds in relation to the rotational speed (Figures 5 and 6). This map shows the natural frequencies as a function of a wide range of the isotropic bearing stiffness without damping influence. The bearing stiffnesses are influenced, for example, by manufacturing tolerances or different operating conditions. With this map it can easily be shown how this stiffness change influences the critical speeds.

The pinion shaft of the gear-turbine operates between the first and second critical speed (Figure 5), which is typical for pinions

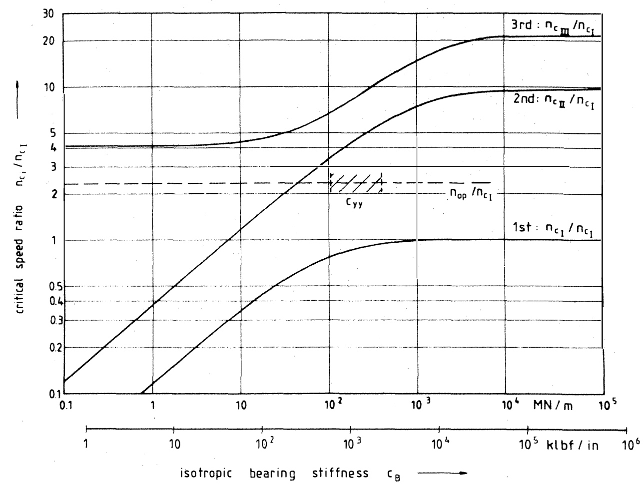

Figure 5. Critical Speed Map of Pinion Shaft (second stage); Stiffness Range According to Tables 1 and 3.

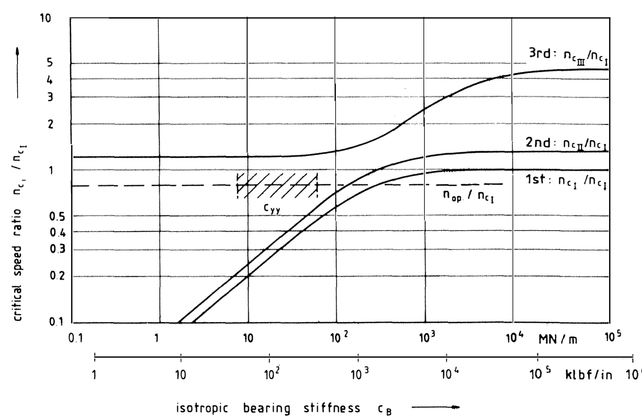

Figure 6. Critical Speed Map of Compressor Turbine Rotor; Stiffness Range According to Table 2.

with one overhung wheel. The rotor of the compressor turbine runs between the second and third critical speeds, because with a stiff symmetric rotor, the first and second natural modes are close together (Figure 6).

System Damping and Natural Frequencies

The natural modes of the totally coupled rotor-bearing system are calculated with the unisotropic and antimetric bearing stiffnesses and dampings, so that generally one natural mode is split in two eigenvalues which are composed of the natural frequencies and the system dampings.

The logarithmic decrement δ also often used can be calculated from the system damping D* as follows

$$\delta = 2\pi D^*$$

Positive values mean stable conditions. The natural frequencies of the gear turbine pinion are shown in Figure 7 and the related system damping values in Figure 8. As the dynamic coefficients of the tilting pad bearing are calculated for idealized conditions, the rotorbearing system has no stability limit. The damping reserve at operating speed is high enough for these idealized pre-assumptions.

For the rotor of the compressor-turbine the natural frequencies (Figure 9) are only slightly different, because the influence

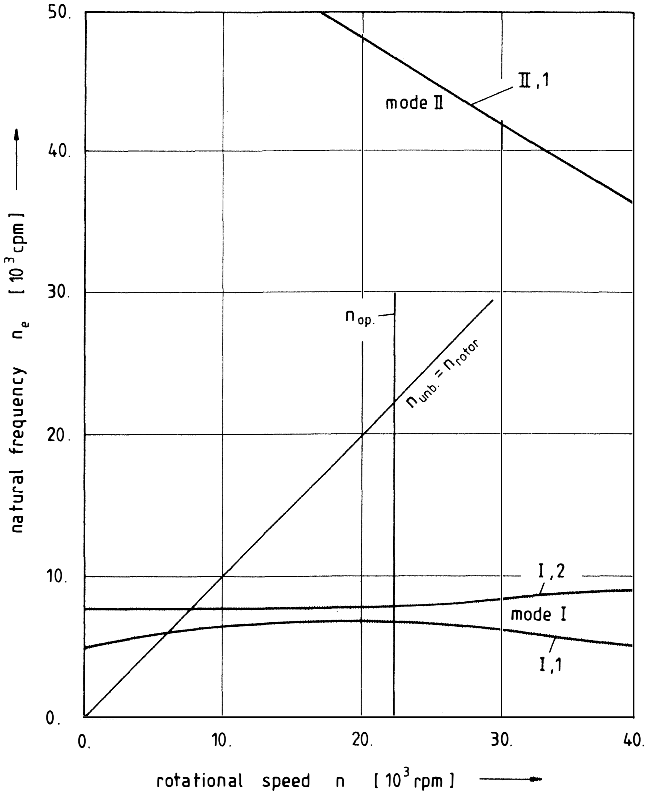

Figure 7. Natural Frequencies of Pinion Shaft (second stage).

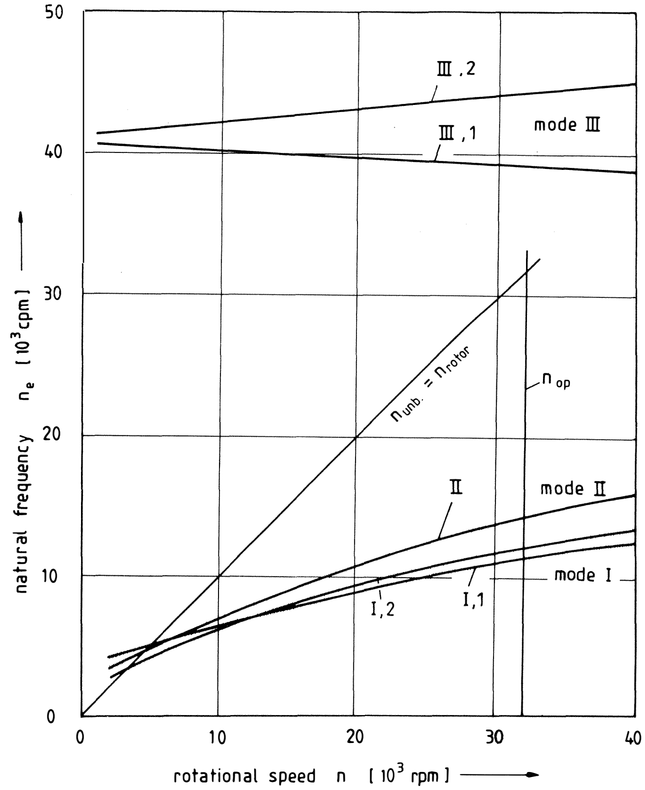

Figure 9. Natural Frequencies of Compressor Turbine Rotor.

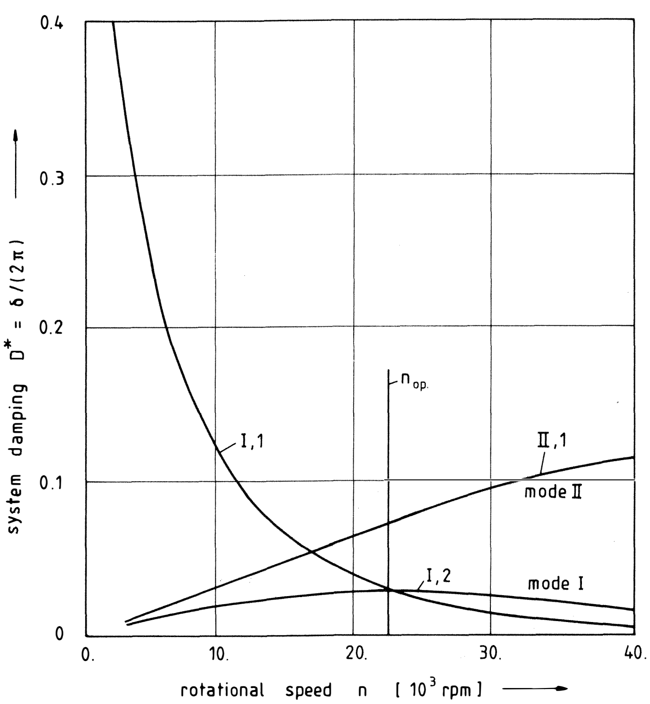

Figure 8. System Damping of Pinion Shaft (second stage).

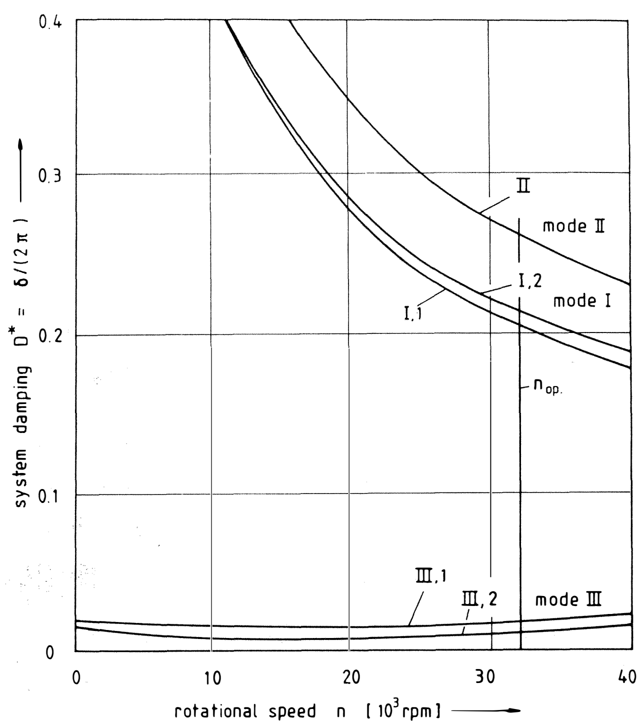

Figure 10. System Damping of Compressor Turbine Rotor.

of mass moments of inertia and anisotropy of the low loaded bearings is very small. The damping of the first two natural modes is relatively high while that of the third natural mode is very small (Figure 10).

Influence of Bearing Clearance and Preload Factor

Parameters which are essentially influencing the dynamic coefficients of tilting pad journal bearings are bearing clearance and preload factor. It is necessary to vary these parameters for optimizing the bearings and for estimating the influence of changes due to manufacturing tolerances or operating conditions.

For the pinion shaft of the gear turbine, the change of bearing clearance has only a small influence. The system damping of the first natural mode is reduced a little with increased bearing clearance. On the other hand, the system damping increases remarkably by reducing the preload factor (Table 1).

Table 1. Influence of Bearing Clearance and Preload Factor on Eigenvalues of Pinion Shaft.

bearing clearance $\Delta D/D$ [‰]	preload factor m	natural frequency n_e [cpm] system damping $D^* = \delta/(2\cdot\pi)$ I,1 n_e	I,1 D^*	I,2 n_e	I,2 D^*	II,1 n_e	II,1 D^*
1.2	0.50	7260	0.015	9170	0.023	47800	0.029
1.5	0.50	7120	0.018	8960	0.028	47600	0.046
1.8	0.50	7020	0.020	8800	0.029	47400	0.065
2.1	0.50	6950	0.020	8700	0.028	47300	0.081
1.8	0.33	7000	0.027	8780	0.038	47700	0.055
1.8	0.50	7020	0.020	8800	0.029	47400	0.065
1.8	0.67	7030	0.014	8800	0.021	46830	0.081

For the rotor of the compressor turbine, the system damping of the first three natural modes increases with decreasing bearing clearance and with decreasing preload factor (Table 2).

Influence of an Additional Static Load

The static bearing loads of the pinion shafts vary during operation by controlling the power of the gear turbine. During no load mechanical tests (sometimes also carried out without the impellers), even the weight of the pinion shaft itself can dominate. For all these operating conditions, a stable running behavior has to be guaranteed.

For the pinion shaft supported in tilting pad bearings, the system damping of the first natural mode decreases strongly with increasing additional static load (Table 3).

Influence of External Elasticity

The external elasticity being in series to the bearing lubricant film can often not be avoided by the design. For example, the pivot point of a tilting pad has only a limited stiffness sometimes similar to bearing stiffness. Additional elasticities are also possible around the bearing pedestal.

The external elasticity influences the overall stiffness and also the overall damping of the bearing structure. The system damping of all three natural modes of the compressor turbine is remarkably reduced by increasing external elasticity that means decreasing of the external stiffness (Table 4).

Table 3. Influence of Additional Static Loads on the Eigenvalues of Pinion Shaft.

power [%]	bearing load	I,1 n_e	I,1 D^*	I,2 n_e	I,2 D^*	II,1 n_e	II,1 D^*
100	6970	7020	0.020	8800	0.029	47400	0.065
60	4180	6750	0.034	8380	0.048	47900	0.089
0	460/74	6230	0.062	7570	0.090	49000	0.115

Sealing Influence

The sealing influence on the rotor vibration can be easily explained. With a small deflection of the rotor in the sealing area, a pressure force component perpendicular to the deflection comes about. This effect is amplified by gas flowing through the seal, with a velocity component in the direction of the rotor rotation.

The sealing influence can be explained mathematically by an antimetrical stiffness matrix which consists of cross coupling coefficients.

To calculate the cross coupling coefficients of the labyrinth seals, a simplified model was made especially for the swirl velocity, in accordance to Benckert and Wachter [3]. With this computer program, the cross coupling coefficients were calculated for each of the 11 sealing parts of the compressor turbine (Figure 2), depending on the geometry and gas parameters and on the assumed swirl velocity (Table 5). The influence on the rotor system damping was dependent on the value of the coefficient and also on the related amplitude of the natural mode. The reduction of the system damping due to the influence of each labyrinth seal is also shown in Table 5. The labyrinth seals located near the cover plates of the impeller shrouds (Figure 2, positions 1 and 11) and near the balance piston (positions 8 to 10) influence the system damping the most.

Table 2. Influence of Bearing Clearance and Preload Factor on Eigenvalues of Compressor Turbine Rotor.

bearing clearance $\Delta D/D$ [‰]	preload factor m	natural frequency n_e [cpm] ; system damping $D^* = \delta/(2\cdot\pi)$ I,1 n_e	I,1 D^*	I,2 n_e	I,2 D^*	II,1 n_e	II,1 D^*	II,2 n_e	II,2 D^*	III,1 n_e	III,1 D^*	III,2 n_e	III,2 D^*
1.5	0.5	17913	0.291	18712	0.311	21273	0.413	21398	0.406	39043	0.061	44477	0.054
2.0	0.5	12105	0.214	12740	0.222	14403	0.270	14473	0.270	39338	0.021	44626	0.020
3.0	0.5	6867	0.124	7418	0.126	8178	0.148	8372	0.148	39126	0.006	44415	0.005
2.0	0.3	9580	0.330	10136	0.339	11165	0.412	11265	0.411	39103	0.022	44402	0.021
2.0	0.5	12105	0.214	12740	0.222	14403	0.270	14473	0.270	39338	0.021	44626	0.020
2.0	0.7	13960	0.137	14673	0.143	16839	0.174	16889	0.175	39608	0.019	44872	0.017

Table 4. Influence of External Elasticity on Eigenvalues of Compressor Turbine Rotor.

external stiffness [MN/m] (klbf/in)	natural frequency n_e [cpm]; system damping $D^* = \delta/(2\cdot\pi)$											
	I,1		I,2		II,1		II,2		III,1		III,2	
	n_e	D^*	n_e	D^*	n_e	D^*	n_e	D^*	n_e	D^*	n_e	D^*
∞	12105	0.214	12740	0.222	14403	0.270	14473	0.270	39338	0.021	44626	0.020
170(970)	11366	0.170	11999	0.175	13650	0.212	13711	0.212	39479	0.015	44765	0.014
50(285)	9900	0.108	10504	0.110	11958	0.131	11997	0.131	39432	0.007	44712	0.006
30(171)	8876	0.077	9454	0.079	10723	0.093	10751	0.092	39336	0.003	44610	0.003

Table 5. Cross Coupling Stiffness Coefficients of Labyrinth Seals and the Influence on System Damping of the Compressor Turbine Rotor.

seal position	cross - coupling coefficient c_{xy} [MN/m]	[klbf/in]	reduction of system damping ΔD^*
1	0.456	2.603	0.026
2	0.351	2.004	0.011
3	0.720	4.110	0.019
4	0.244	1.393	0.005
5	0.002	0.011	0.0
6	0.003	0.017	0.0
7	0.244	1.393	0.004
8	0.783	4.469	0.018
9	1.100	6.279	0.039
10	1.050	5.993	0.037
11	0.445	2.540	0.029

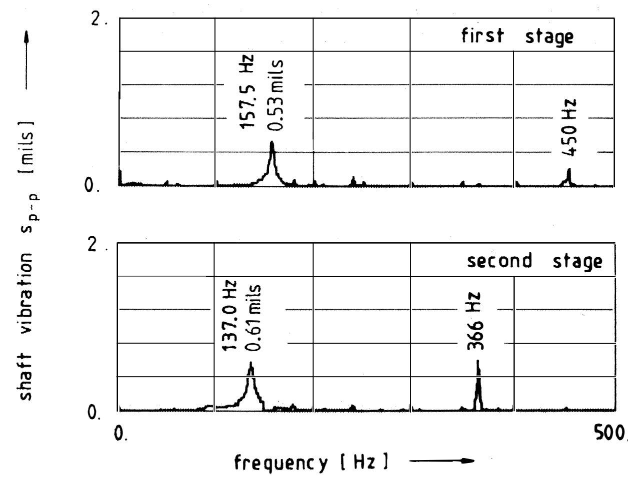

Figure 11. Vibration Spectra of First and Second Stage Pinion. Shaft of Gear Turbine.

VIBRATION EXPERIENCE WITH GEAR TURBINE

When operating the gear turbine at part load, high subsynchronous vibration levels occurred in the shaft vibration spectra of both turbine stages (Figure 11). The vibrations led to shutdown of the turbine unit in some cases. With increasing load, the subsynchronous vibrations were reduced.

The frequency spectra measured in the course of the testing at the manufacturer did not demonstrate such instabilities. To find out what caused the vibrations, different test runs were made onsite. The following observations were made:

• The subsynchronous vibrations are always present at certain part load operations and strongly depend on the nozzle position.

• The subsynchronous frequencies of the vibrations of both rotors were between 30 to 40 percent of the synchronous constant running speeds.

• Depending on the load the thermodynamic and aerodynamic conditions inside the turbines change, i.e., pressure, temperature, flow direction and velocity. The disturbing frequencies were dominating, when the pressure ratio across the nozzles was low and the circumferential component of the absolute flow velocity was high.

The situations was complicated by the fact that the frequencies observed at the rotors were close to the first natural lateral bending frequency and therefore the rotors showed an increased sensitivity due to exciting frequencies of this value. An alteration sufficient to come away from the excitation range would have required a completely new design of the rotors.

The earlier tests to improve the damping behavior of the bearings by changing preload factor and clearance ratio also showed no positive results to ensure stable performance. To determine the exciting mechanism, not observed so far, pressure pulsation measurements were carried out. Frequency analyses of the pressure pulsations were made at different locations, namely at the entry and exit of both stages, at the back of the wheels and in the gap between stator (nozzles) and wheel. The measurements showed a relation between shaft vibrations and dynamic pressure changes at the entry and back of the wheel, and thus, demonstrated an aerodynamic excitation effect. The existence of pressure pulsations in the annular space between nozzles and turbine wheel could be made evident. The nozzle position and aerodynamic and thermodynamic conditions in the annular space were dependent on each other in all cases.

The adjustable nozzle ring and the change of geometrical dimensions of the annular space between nozzles and turbine wheel are shown in Figure 12. With closed nozzles the cross sec-

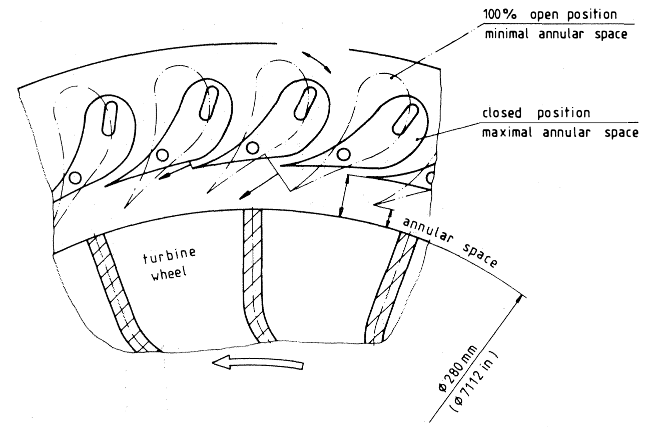

Figure 12. Change of Annular Space between Nozzles and Turbine Wheel with Nozzle Position.

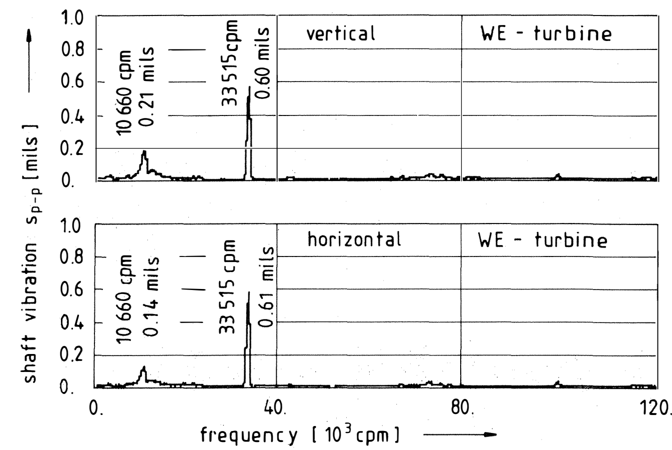

Figure 13. Vibration Spectrum of Compressor Turbine.

tion of the annular space and its equivalent peripheral length is maximal. When the nozzles are opened, the cross section and the peripheral length decrease until there is only a small gap between nozzles and turbine wheel.

The velocity of sound "a" depending on the gas conditions in the annular space, relates to the equivalent peripheral length l_{eq} and results in a frequency $f = a/[4 * l_{eq}]$ which corresponds to the measured excitation frequency of the subsynchronous vibrations. The relation shown refers to a $\lambda/4$-resonance of a standing wave, in a pipe open at one end, and closed at the other.

If the acoustic resonance really existed in the annular space, it could be eliminated by changing the peripheral length of the annular space. Therefore, three of the 27 nozzles were fixed in a 100 percent open position, distributed evenly around the circumference. Consequently, the peripheral length was divided in the complete part load operation range. The efficiency loss created by the uneven flow entering the wheel at part load was small and acceptable.

After the modification with fixed nozzles, the frequency spectra showed almost no subsynchronous vibration. A stable operation within the vibration specification was possible. Up to now, it could not be clarified for certain if the pulsations in kind of a "standing wave" were exclusively involved or if other effects contributed to the excitation process. Additional measurements will reveal if there is a locally fixed pressure maximum or minimum in the annular space, or if there are rotating pressure pulsations. Since the measurements are not yet completed, the complex interrelations cannot be completely interpreted now.

VIBRATION EXPERIENCE WITH COMPRESSOR TURBINE

In compressor loaded turbines, subsynchronous vibrations were observed at full load in some cases. During the test runs in the test bed of the manufacturer, however, the machines operated at part load without any problems. Troubles connected with subsynchronous vibration was on the one hand dependent on the operation conditions such as load, speed, seal gas pressure, oil supply, and on the other hand, dependent on accidental scatter in tolerance bands such as balancing status, geometry of bearings, bearing clearance, clearance in labyrinth seals and gaps and clearances adjusted during assembly. The frequencies of the subsynchronous vibrations were between about 20 and 65 percent of the speed.

Vibrations before Modifications

A typical vibration spectrum of a compressor-turbine is shown in Figure 13. The speed of the rotor is 33515 rpm. The synchronous vibrations of 0.6 mils (15 μm) are the main part of the total vibrations. The subsynchronous vibrations occur at 10660 rpm and have a peak-to-peak level of 0.14 to 0.21 mils (3.5 to 5 μm). The synchronous vibration is limited to a narrow frequency band, whereas the subsynchronous vibration can be observed in a broader band.

In another compressor turbine, the subsynchronous vibration was higher than the synchronous vibration (Figure 14). The level of subsynchronous vibration is oscillating, the synchronous and ultraharmonic vibrations, however, are constant. For this turbine, the vibration behavior during coastdown is shown in a cascade plot (Figure 15). The frequency of the subsynchronous vibration depends only very slightly on speed, while the amplitude of subsynchronous vibration changes remarkably with speed.

At that stage of the investigation, the rotordynamics were checked several times with various parameters, i.e., preload factors and clearance ratio of the journal tilting pad bearings.

Other influences such as piping forces, stiffness, alignment tolerances, and oil were discussed and investigated. Uneven oil distribution and flows inside the bearing, creating uneven temperatures and temperature increases were supposed, but could be ruled out by calculations. Detailed measurements of temperatures and pressures inside the bearing and of the oil flows confirmed the calculated results.

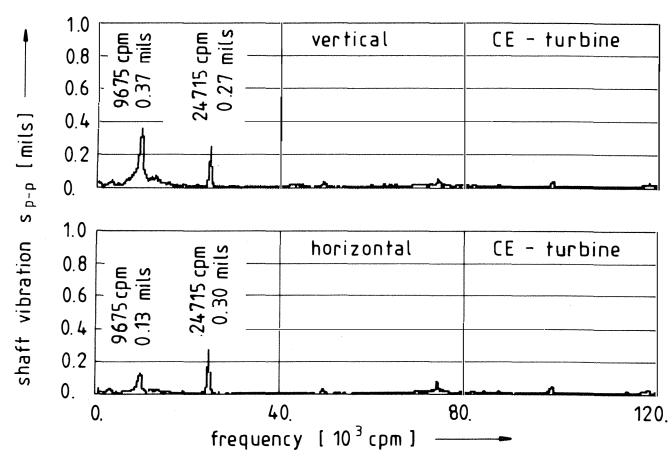

Figure 14. Vibration Spectrum with High Subsynchronous Vibrations of Compressor Turbine.

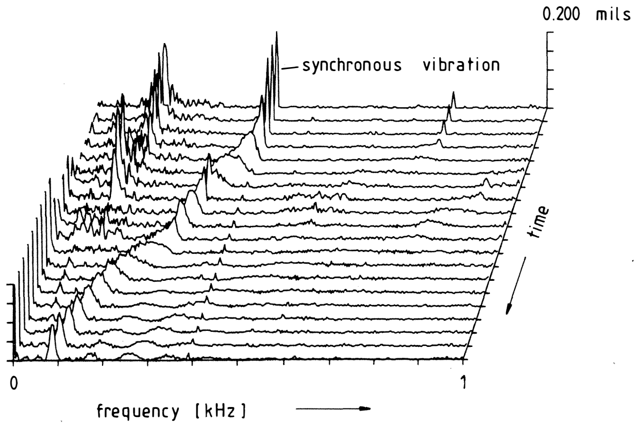

Figure 15. Cascade Plot of Vibrations During Coast Down of Compressor Turbine.

The change of only a single parameter in a realistic range was not sufficient to calculate an instability of the rotor in theory.

Vibration after Modifications

The theoretical parameter studies showed that the only important destabilizing effects are the cross coupling coefficients of the labyrinth seals and the external elasticity of the journal bearings. When both effects are considered together, negative damping, i.e., instability is calculated for the second eigenvalue (Figure 16). Damping values for the other natural frequencies are all in the high positive stable range. The main influence of the destabilizing effects is also in the labyrinth seals as is demonstrated in Figure 16.

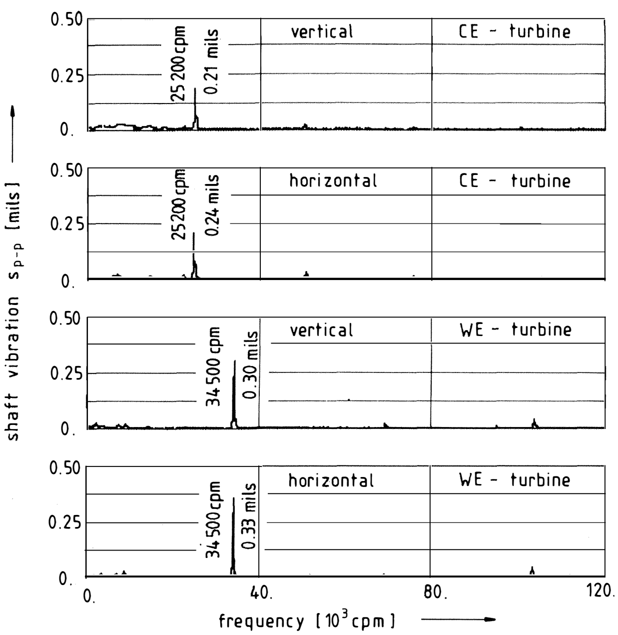

Figure 16. Effect of External Influences on System Damping of Compressor Turbine Rotor.

Based on the knowledge gained from the investigations and literature [4], [5] a catalogue of action was arranged.

• Swirl brakes at the gap cover plate located at the turbine wheel shroud according to Figure 17. The 24 grooves located at the circumference shall interrupt the rotating gas flow in the gap and produce a flow with a low circumferential component admitted to the labyrinths.

• Swirl brakes at the cover plate located at the turbine wheel rear. Grooves cut into the cover plate shall reduce the rotational flow of the gas in the gap.

• Swirl brakes at the gap cover plate located at the compressor impeller shroud reduce the rotational flow between the impeller and housing and at the entry to the labyrinth. The braking of the swirls is obtained by grooves similar to those shown in Figure 17.

• Admission of gas to the balance piston in an counter-rotational direction by bores drilled tangentially into the labyrinth seal ring of the piston (Figure 18).

• Seal gas supply into the labyrinth ring of the turbine side shaft seals via tangential bores similar to those in Figure 18.

• Mounting of the bearing into the casing with a very tight fit to obtain a higher external stiffness of the bearings. The positive fit was produced by machining the outer diameter of the bearing housing according to the actual bore diameter of the casing.

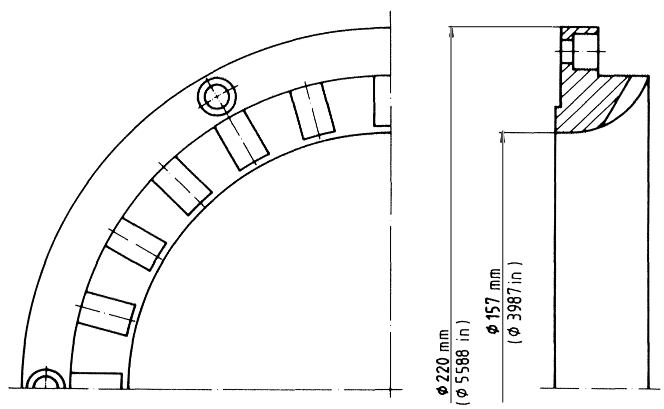

Figure 17. Gap Cover with Swirl Brakes.

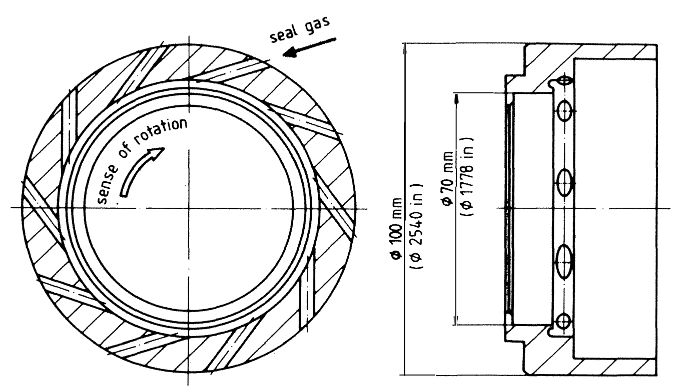

Figure 18. Labyrinth Ring with Tangential Gas Supply Bores.

Unfortunately, all actions had to be carried out in parallel because of production reasons, so the individual effects of each action could not be affirmed by field measurements.

The tight fit of the bearing in the casing proved to be bad because of the special temperature conditions in the turbine. In spite of a clearance ratio of 0.0026 during assembly, the bearing failed after about 20 hr of operating time, due to shrinking of the casing by low temperature, thus reducing the clearance. Subsynchronous vibrations did not appear during this time. The bearing damage clearly showed inferior clearances at operating conditions. Therefore, the same bearing was used again which had run already in the machine for several months with subsynchronous vibrations.

The vibration spectra after the next start are shown in Figure 19. The vibration level is far below the allowable limit. Remarkable subsynchronous vibration amplitudes do not occur anymore.

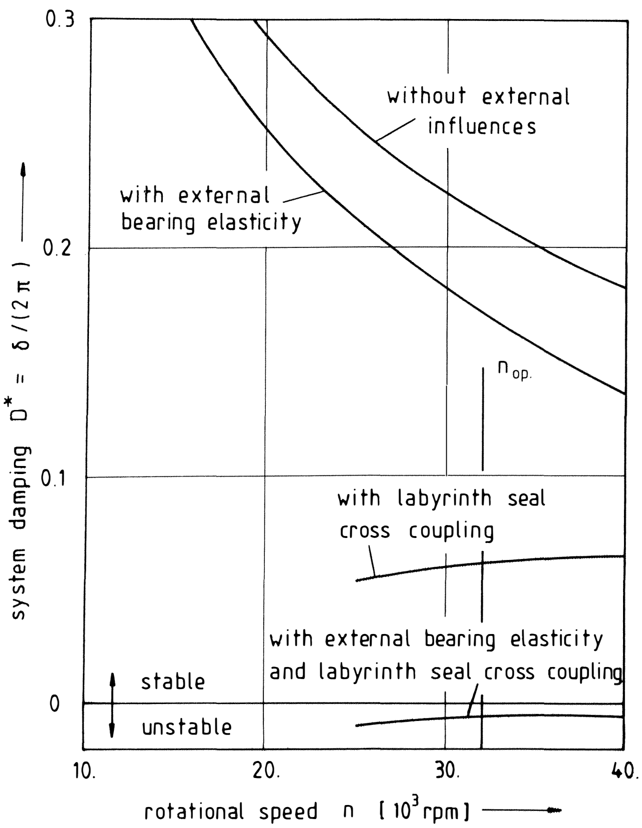

Figure 19. Vibration Spectra of Compressor Turbine after Modification.

SUMMARY AND CONCLUSION

The subsynchronous vibration problems were described and analyzed as observed at two different turbine installations, a two-stage turbine with gear box and a single-stage compressor loaded turbine.

Vibrations in the gear turbine were caused by an acoustic resonance in the annular space between nozzles and turbine wheel. The acoustic phenomenon has not been mentioned before in literature as an excitation force of subsynchronous vibration in turbomachines. The available measurements and data are not sufficient to back a theoretical approach. Therefore, further investigations and measurements will be necessary for final clarification.

The acoustic resonance in the gear turbine could be avoided by dividing the annular space. Subsynchronous vibrations were reduced to an acceptable level. The subsynchronous vibrations in the compressor turbine were caused by a combined effect of destabilizing influences. The main influence came from the labyrinth seals in combination with an external elasticity in the bearing.

By installing modified gap cover plates and seal rings into the compressor-turbine to interrupt the rotational flow in the seals, the subsynchronous vibration was eliminated.

The theories and computer codes available now enable manufacturers to find the causes for subsynchronous vibration and to work out measures to avoid the vibrations in future designs.

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