TROUBLE SHOOTING BY A COMPREHENSIVE VIBRATION STUDY DURING THE START-UP OF A TURBO COMPRESSOR

Dr.-Ing. Johann Lenz

KÖTTER Consulting Engineers KG, Rheine, Germany
lenz@koetter-consulting.com

Abstract:
During the start phase of a new turbo compressor repeated damages of the mechanical seal system occurred. In order to detect the cause for this, a comprehensive investigation was carried out. Three main topics were checked: vibrations of the pinion and bull gear shaft, axial shaft displacement and leakage flow of the seal systems. The measured shaft vibrations didn't exceed the allowed limits. The axial shaft displacement during ramp up of the compressor confirmed the allowance by the pre-specified clearance. The measurements at nine different operating points showed no change of sign in axial load at the pinion shaft. Additionally, a high axial vibration of the pinion shaft and also of the mechanical seal in the rotation frequency of the bull gear shaft was detected. Furthermore, the casing vibration in the horizontal direction was pretty high. The analysis of the measured signals and the phase of the casing vibrations showed an unbalance at the slow speed shaft. For this reason the coupling flange at the bull gear shaft was balanced and the casing vibrations as well as the high axial vibrations of the pinion shaft could be reduced strongly. After the realisation of this measure the compressor went into operation without any further problems.

Key words:
Axial load; balancing; gas seals; leakage flow; multichannel measurement; shaft vibration; turbo compressor.

Situation and task assignment:

The start-up of a plant section in a newly established chemical factory was delayed for several months. The reason was a repeated case of a defective mechanical seal system on the pinion shaft of a gear type centrifugal compressor (fig. 1).

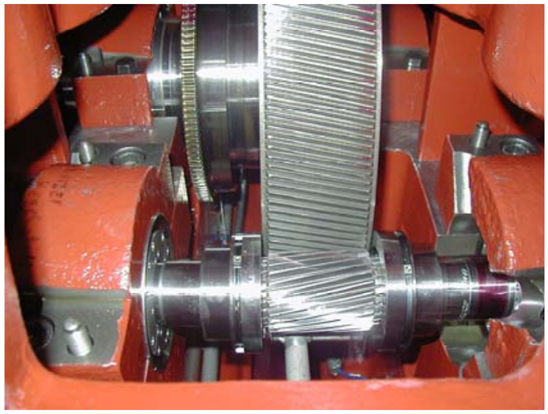
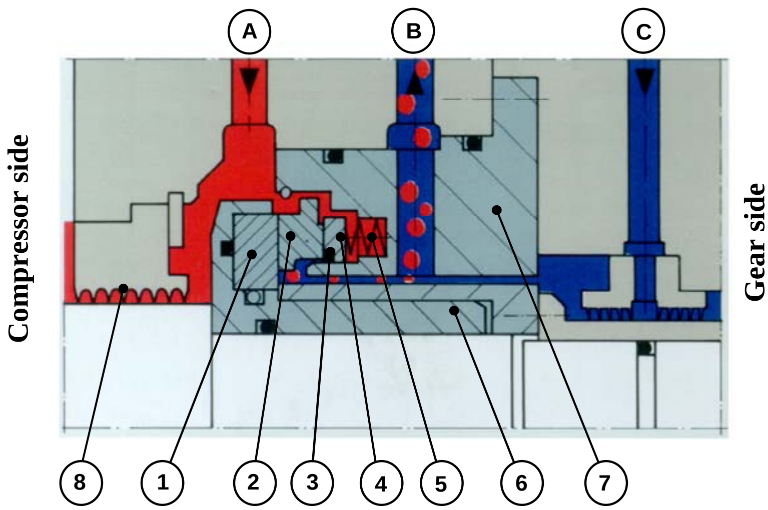


Figure 1: View on the pinion shaft (foreground) and bull gear of a gear type centrifugal compressor.

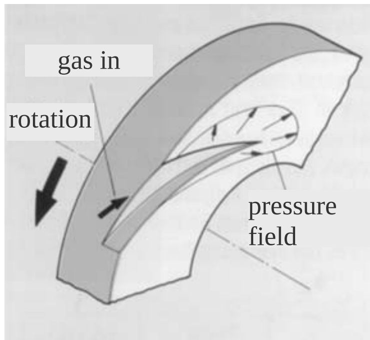
The seal system which seals off the pinion shaft rotating at approx. 35,000 rpm to the stationary case is based on a dry gas seal. Compared with conventional mechanical seals the non-contact operation decisively reduces the moment of friction and the wear of the seal in the application range that is characterised by high speeds.



A buffer gas
B flare gas
C separating gas
1 rotating seat
2 stationary face
3 O-ring
4 thrust ring
5 thrust spring
6 cartridge
7 stationary casing
8 additional labyrinth seal

Figure 2: Functional arrangement of a dry gas face seal for compressors [1, 2].

The seal consists of a stationary, spring-loaded slide ring and a rotating seat (fig. 2). The sealing surfaces of these two rings slide on each other without a surface contact. In the outer zone of the rotating seat the compression gas is applied and compressed by means of the grooves machined into the ring surface. A difference is made in the groove geometry between grooves dependent on the direction of rotation (unidirectional) and those that are not dependent on the direction of rotation (bidirectional) (fig. 3).



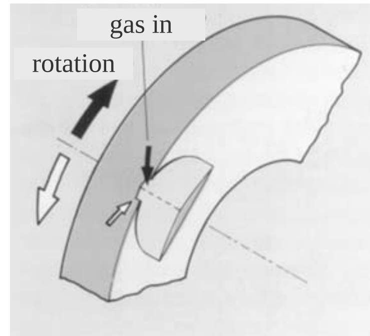


Figure 3: Groove geometry, V-groove dependent on direction of rotation (left) and U-groove independent of direction of rotation (right) of the rotating seat [1, 2].

The gas flows through the sealing gap where a very low leakage flow is achieved because of the narrow gap width. The inner zone of the counter ring has no particular surface characterisation and subsequently leads to a throttle effect of the leakage.

The gap that is formed in operation between the rotating and the stationary ring has a width of only 4 µm. The gas film that is established at that location has a spring-type action with a progressive characteristic curve between both sliding surfaces. This leads to a stable sealing gap so that, in principle, even vibrations can be transmitted without a contact of both sealing surfaces.

In addition to this mechanical seal and in the case presented here, a labyrinth seal is arranged towards the compressor side and a sealing system with spring-loaded carbon rings is arranged towards the opposite machine side. Based on an excess pressure of the buffer gas A and of the separating gas C, a kind of outer sealing system is established.

Almost all cases of damage were detected shortly after a trip alarm of the monitored leakage gas volume or the vibrations. Various precautionary measures already carried out, such as the change of the sealing system by that of another supplier, were not successful. The plant constructor assumed a location instability of the pinion shaft. After six cases of mechanical contact seal damage had occurred within one year, KÖTTER Consulting Engineers was commissioned to perform a comprehensive analysis on the causes involved.

The way of analysing the cause:

A metrological examination was carried out in order to investigate the physical causes for the damage to the mechanical contact seal. Due to the damage history, the main focus of attention was placed on the dynamics of the pinion shaft and the mechanical seal. In order to accomplish this comprehensive task, the following simultaneously measured signals were necessary: axial displacement and vibrations of the pinion and the bull gear shaft, radial vibrations of the pinion shaft, absolute casing vibrations, static and dynamic torque of the drive shaft, static and dynamic pressures and leakage flow of the seal system (fig. 4 and 5).

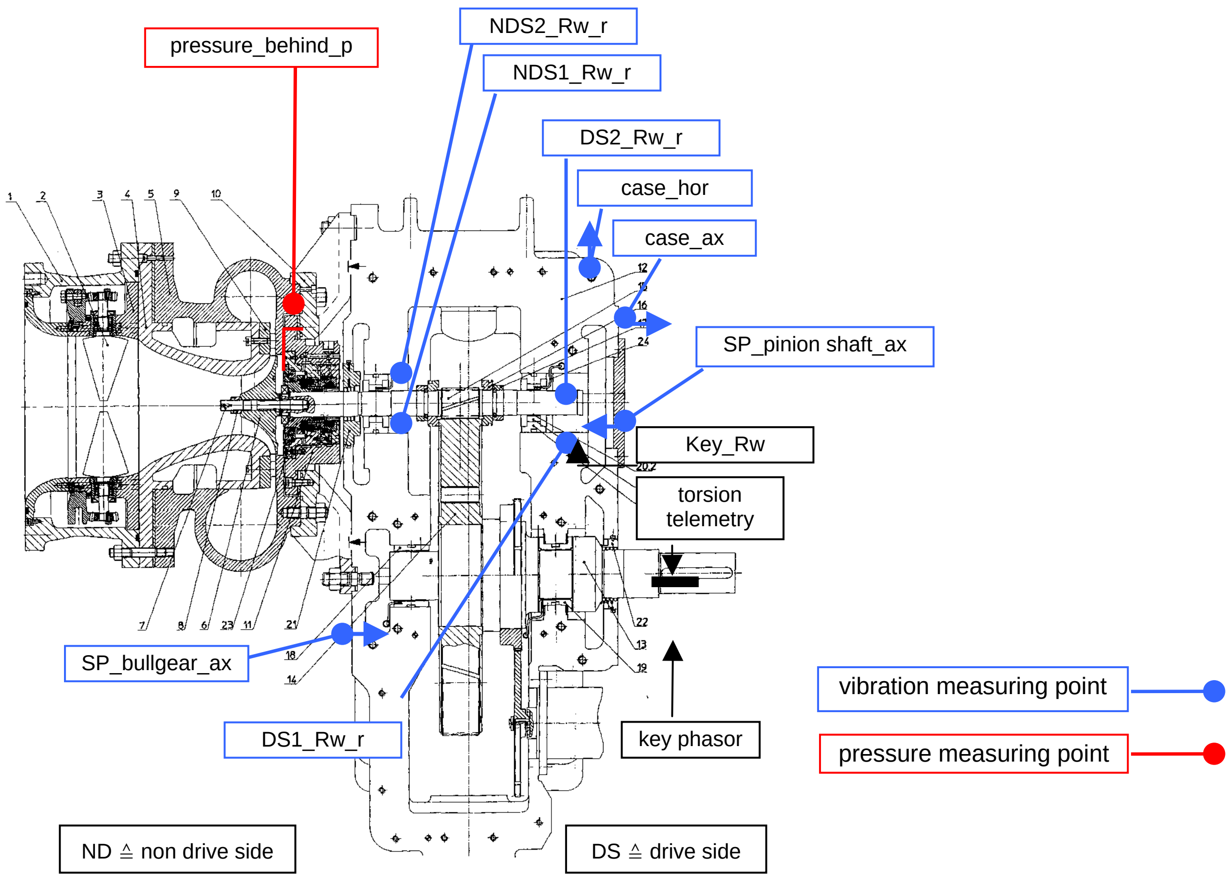


Figure 4: Measuring points at the turbo compressor.

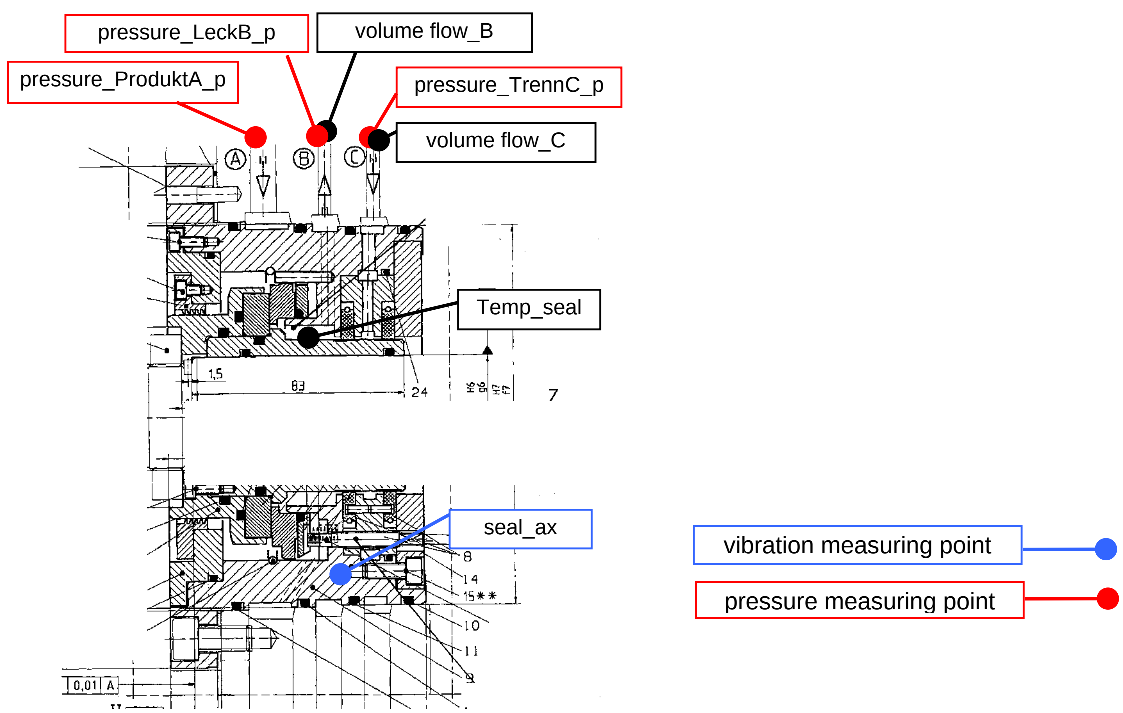


Figure 5: Measuring points in the direct zone of the seal.

The examination was performed after deinstallation on-site with nitrogen in a close-loop mode of the compressor at a test rig of the manufacturer. In a first step, the crossing of the first natural bending frequency of the pinion shaft was examined during capacity increase and coast down of the compressor, respectively.

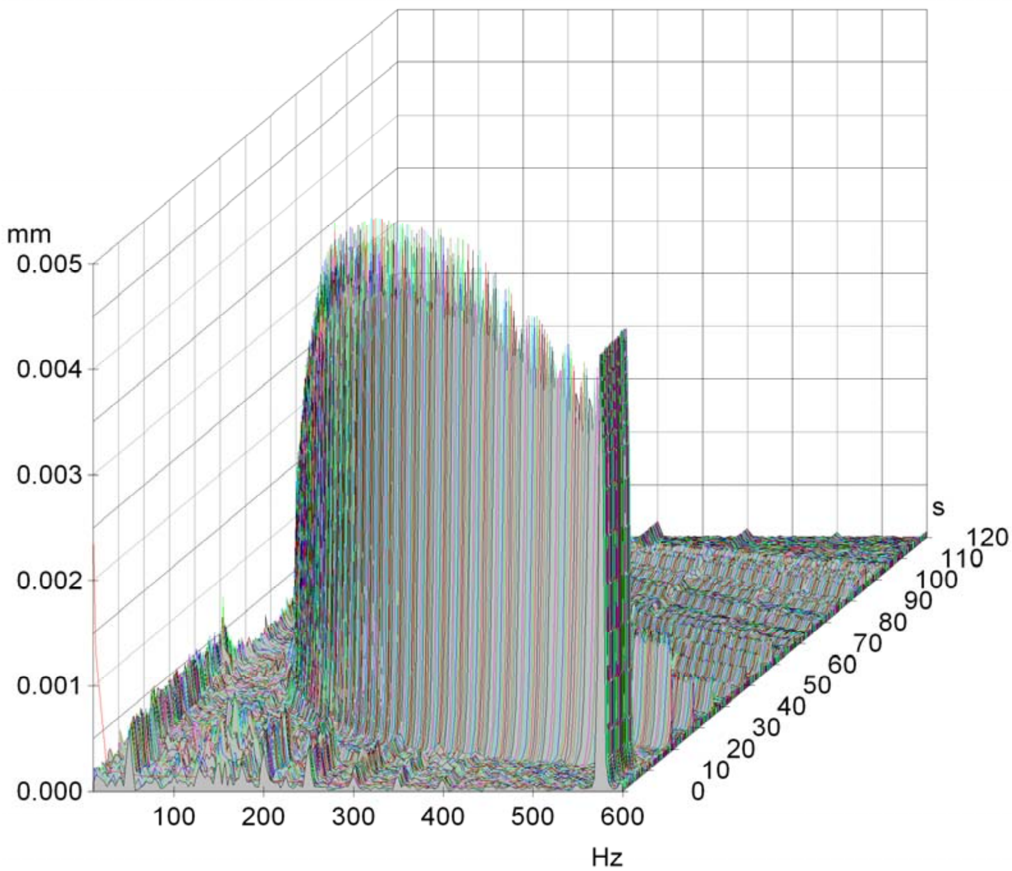


Figure 6: Waterfall chart of the amplitude spectra of the radial shaft vibrations on the non-drive side of the pinion shaft (measuring point NDS1_Rw_r) during coast down (operating speed approx. 570 Hz) of the compressor.

The amplitude of the shaft vibration did not indicate any conspicuously resonant increased values, neither during capacity increase nor during coast down of the compressor (1. bending frequency was in the range of 200 Hz, fig. 6). In the stationary operation the allowable limit values of the radial shaft vibration were also not exceeded.

In the next step, the following points were examined and compared with the conspicuous results of the calculations performed by the manufacturer:

1. During ramp up of the compressor a change of the axial pinion shaft loading took place. The calculations of the manufacturer showed a change of direction shortly before reaching the final speed of the compressor.

2. In the stationary nitrogen operation of the compressor an axial pinion shaft force of approx. 1 kN occurred in the direction of the compressor side. Because of this low axial load a location instability of the pinion shaft was quite conceivable.

With regard to the first point, the axial displacement and vibration of the pinion and wheel shafts were recorded during the start phase of the compressor (fig. 7). The pinion shaft was displaced by approx. 0.3 mm and the bull gear shaft by approx. 0.05 mm in the direction of the compressor side (refer to the measuring point location in fig. 4). A comparison with the pre-specified clearance of the pressure comb (0.25 mm) as well as of the axial bearing (0.1 mm) confirmed the allowance of the displacements determined in operation.

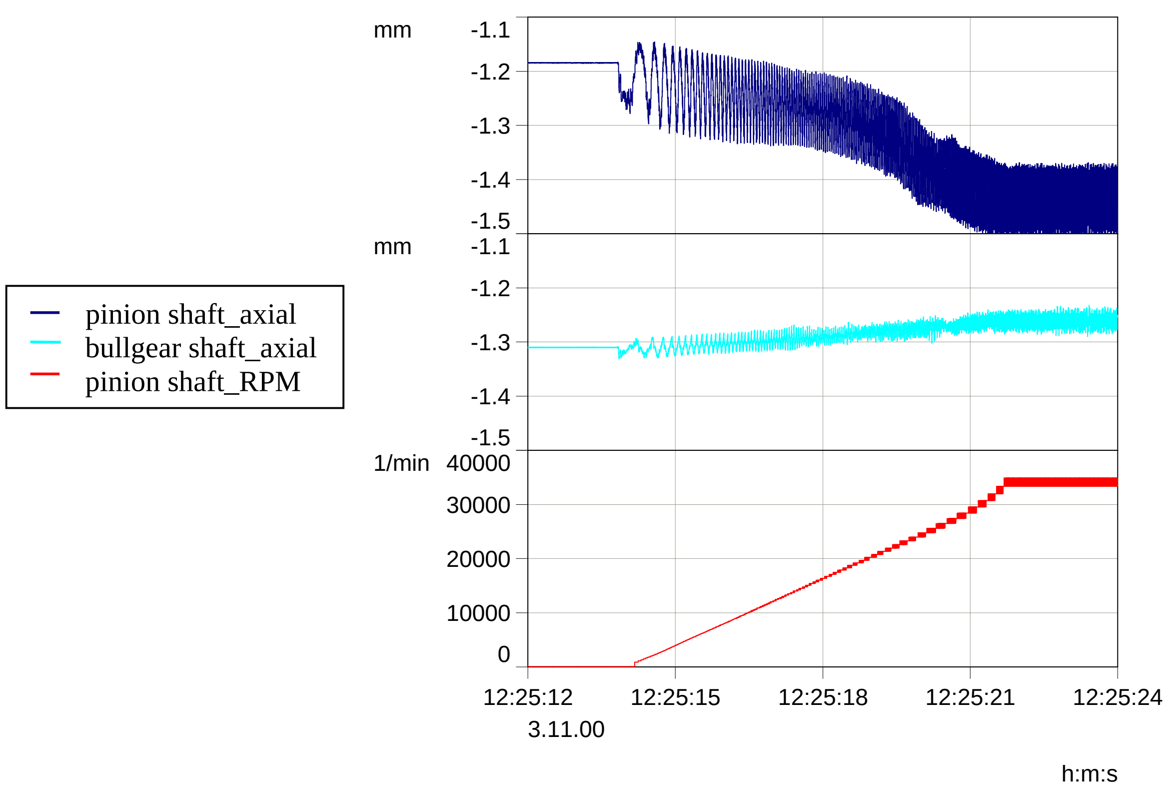


Figure 7: Axial displacement of the pinion and bull gear shaft during capacity increase of the compressor.

The second point shows that - from a calculation viewpoint - relatively low axial forces determine the position of the pinion shaft. For this reason, the axial load of the pinion shaft was determined theoretically combined with measured pressures for various operating conditions. A total of nine stationary operating points (table I) were targeted and entered by means of a variation of the pressure and the volume flow of the compressor.

Table I: Boundary conditions of the nine different operating points.

test	guide wheel position [°]	plant-		pressure_ ProductA_p [bar a]	pressure_ hiLauf_p [bar a]
		suction pressure p_S [bar a]	discharge pressure p_{FD} [bar a]		
A	70	3.07	3.39	3.65	3.71
B	60	3.07	3.41	3.86	3.92
C	60	3.16	4.88	4.16	4.25
D	60	3.13	5.47	4.43	4.53
E	60	3.12	5.79	4.68	4.79
F	0	3.13	4.56	4.32	4.42
G	0	3.11	5.17	4.42	4.53
H	0	3.08	5.60	4.55	4.67
I	0	3.07	5.86	4.73	4.84

The absolute and dynamic pressures upstream and downstream of the impeller as well as in the various sealing sections were measured simultaneously with the vibrations and displacements of both shafts. In order to receive the total axial load acting within the nine operating points at the pinion shaft the relevant surfaces and related pressure loads as well as the impulse equation were used for the calculation (fig. 8).

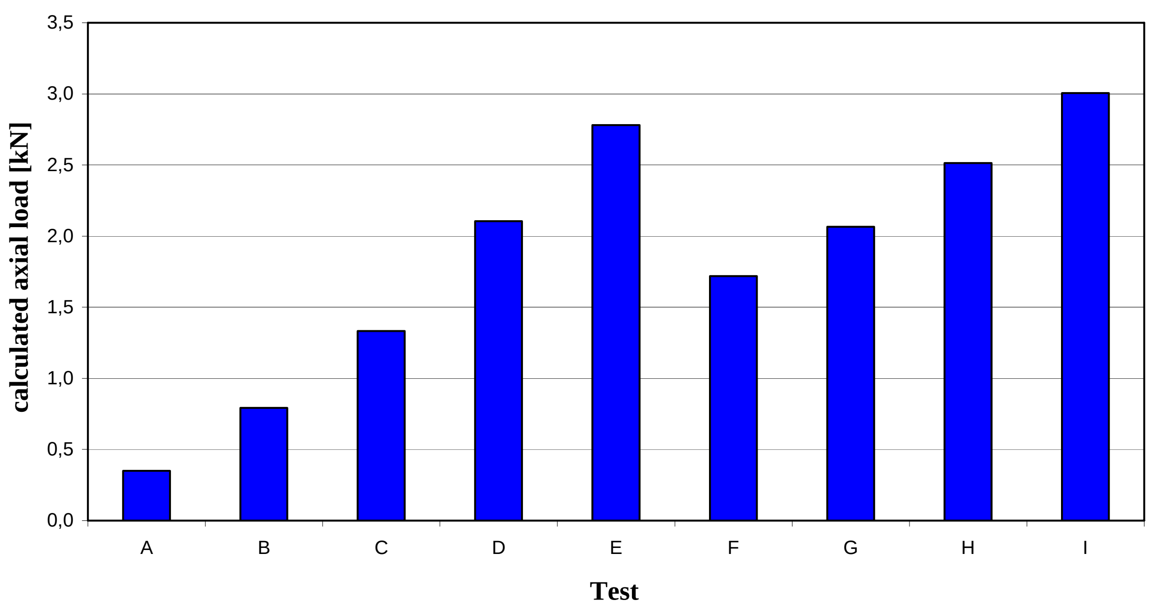


Figure 8: The axial load of the pinion shaft calculated from the measured pressure for the nine different operating points.

Figure 9 shows these calculated axial loads together with the measured axial pinion shaft position at the various operating conditions.

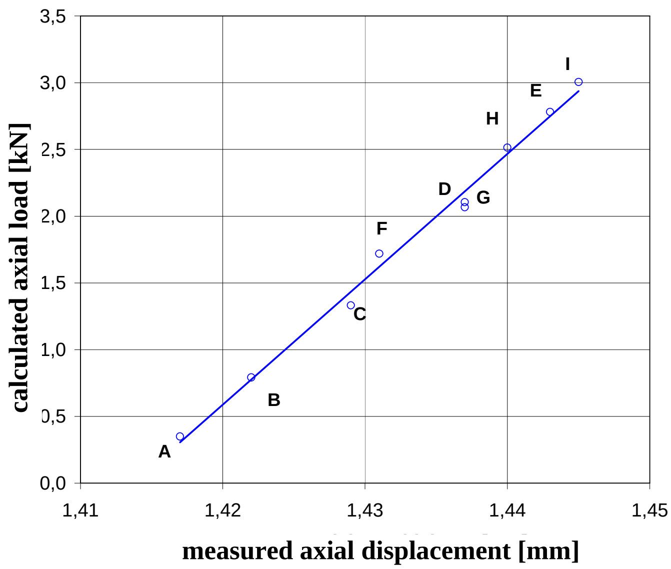


Figure 9: Comparison of the calculated axial loads at the pinion shaft and measured axial displacement for the nine different operating points.

Proceeding from the operating point A and by means of the straight line entered in the diagram, there is a very good linear dependence between the measured axial displacement and the calculated axial load so that an axial instability of the pinion shaft in the stationary condition can be ruled out.

Figure 10 shows the axial pinion shaft vibrations as measured at the different operating points. No functional correlation to the acting forces at the pinion shaft can be determined here.

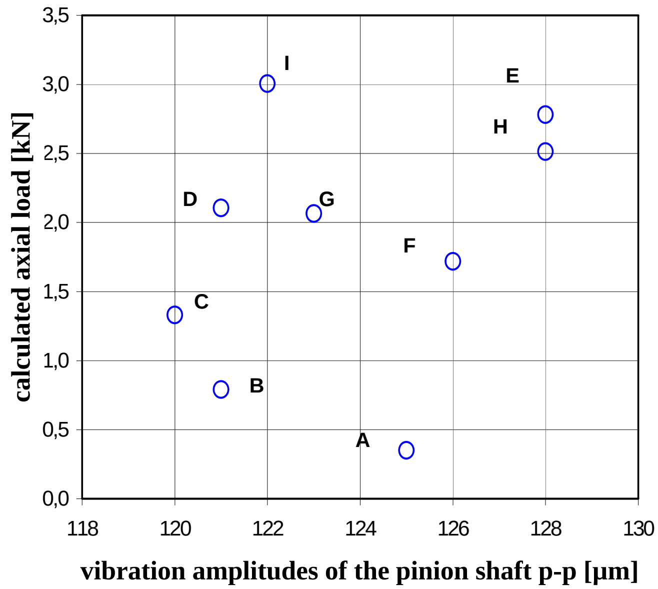


Figure 10: Comparison of measured axial vibrations and calculated axial forces at the pinion shaft for the different nine operating points.

However, in addition to these facts which have remained inconspicuous up to now, an increased level of the axial shaft vibrations of the pinion shaft (fig. 10) was detected. The measured spectrum, as shown in figure 11, shows for the axial vibration of the pinion shaft an amplitude of approx. 55 µm for the rotation frequency of the bull gear shaft of 49.5 Hz.

The limit value specified by the manufacturer for the axial vibration of the seal was exceeded so that a non-allowable loading of the seal can certainly occur here in this case. A remarkable fact was that the axial vibration of the driven wheel shaft at this rotation frequency only had an amplitude of approx. 18 µm (fig. 11).

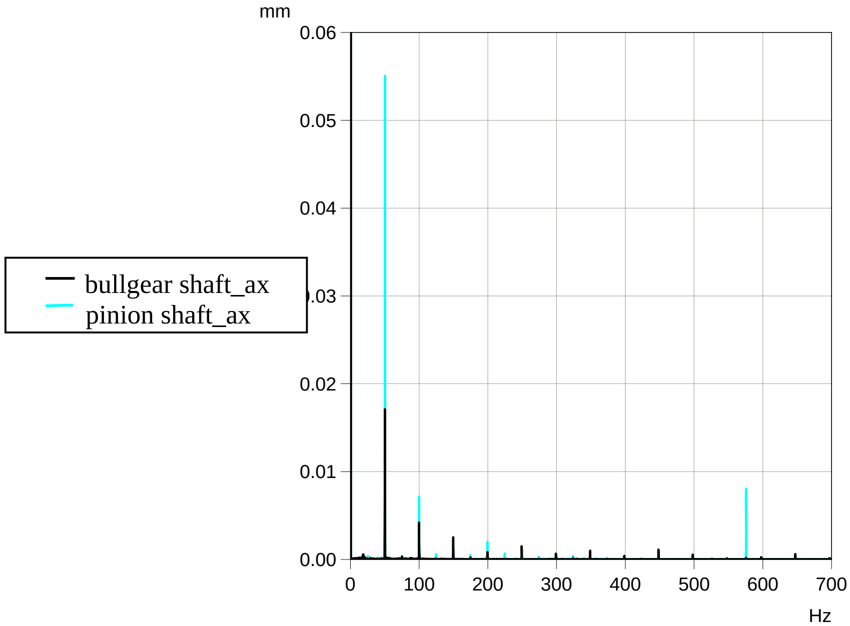


Figure 11: Amplitude spectra of the measured axial vibrations of pinion shaft and bull gear shaft.

Subsequently, the task was to analyse as to how this axial vibration is amplified from the bull gear shaft to the pinion shaft. A simultaneous measurement of the horizontal casing vibration offset at 90° to the shaft location showed an increased vibration velocity of v_{eff} = 5.6 mm/s with a distinctive amplitude of 7 mm/s, likewise in the rotation frequency of the bull gear shaft. The concentricity accuracy of the slow-running bull gear shaft had been inspected several times during the introductory phase. For this reason, the bull gear shaft was balanced at the coupling flange. In this way, it was possible to reduce the horizontal case vibration to an effective value of 1.2 mm/s. At the same time and in this way, the increased axial vibration of the pinion shaft was diminished to an amplitude of approx. 23 µm (fig. 12).

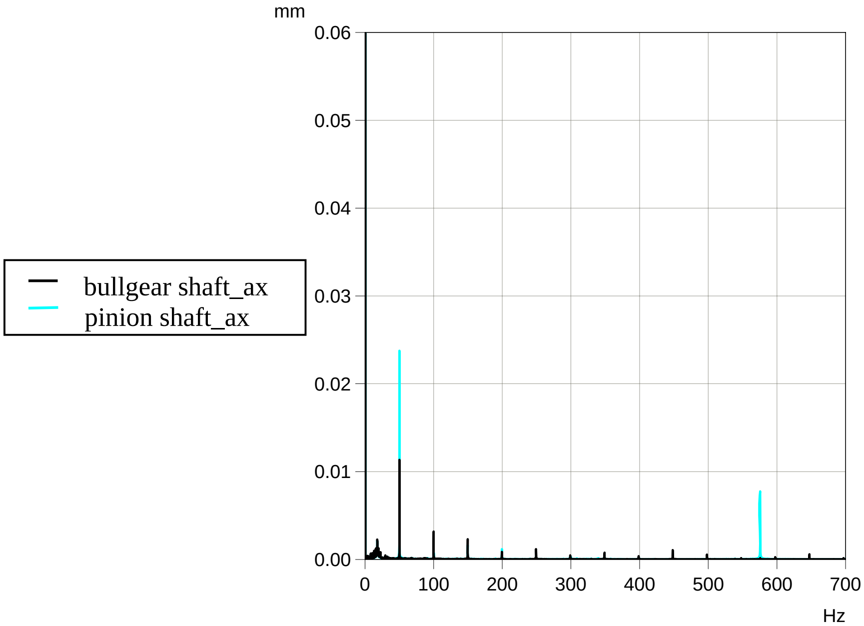


Figure 12: Amplitude spectra of the measured axial vibrations of pinion shaft and bull gear shaft after balancing.

Result:

The cause of the increased axial vibrations of the pinion shaft and presumably also for the occurring damage to the mechanical contact seal was unbalance in the slow-running drive line. The casing vibrations were decisively reduced by precise dynamic balancing at the coupling between the gear and the engine. In parallel to this, the axial vibrations of the pinion shaft and of the mechanical contact seal also were reduced by more than half. In this type of gear configuration, it was evident that the axial vibrations of the high-speed pinion shaft react very sensitively to an unbalance of the slow-running bull gear shaft.

Moreover, the bidirectional mechanical contact seal was replaced by a more vibration-insensitive unidirectional seal that had been once originally deployed in the compressor. Following implementation of the precautionary measures, the compressor went into operation without any problems and has been running for more than 20,000 hours without any further disturbances.

Bibliography:

[1] Burgmann, F.: Dry gas face mechanical contact seal, contributions of staff members of Feodor Burgmann Dichtungswerke GmbH & Co., Wolfratshausen, ISBN 3-929682-15x, 1997.

[2] Tietze, W.: Manual for Seal Practice, 2. edition, Vulkan Publishers, Essen, ISBN 3-8027-2192-6, 2000.