Large industrial screw compressors for refrigeration units Experiences out of operation and maintenance

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Abstract

Over the time the cooling capacity of refrigeration units has been increased steadily. Therefore also the compressors themselves became larger and larger. Now we are operating oil injected screw compressors with power ratings of 1.5 MW and more. These compressors – and this is typical for all screw compressors – are the source of vibrations.

The first part of the presentation describes the vibration measurement procedure at such screw compressor units and the evaluation of the vibration amplitudes according to the VDI 3836. It gives also an overview about the measuring points and some practical hints how to do the vibration analysis. In the next chapter the different sources of dynamic forces are discussed. Out of practical experiences, a correlation between dynamical fluid and mechanical forces and the measured vibration amplitudes together with rotordynamic calculations is presented. Here the influence of the type of foundation is also discussed. In the third part of the presentation failure modes which were found in the last years during maintenance and repair of these compressors are presented and correlated with measured vibration analysis. Out of this the authors are raising the question whether the existing standards for the evaluation of the vibration and with this the reliability of such screw compressor units are still valid.

Part 1: Vibration measurement procedure for screw compressor units

Vibration measurements are a common tool for the assessment of machine conditions. For the evaluation of machines it is essential to perform the measurements and analysis on the basis of standards which are accepted by the parties involved in the project. In Germany the standard DIN ISO 10816 is a well-established paper which describes the procedure for vibration measurements and the evaluation of the measured data. Depending on the type of machine the operator can find guideline values as basis for the decision if the machine is in a good or bad condition.

The different parts of this standard [1-7] consider different operating and mounting conditions, different power classes and different functional principles of the diverse machines.

Although this ISO standard deals with a lot of different types of machines, there are still some machines which are not represented by this standard. An example for these types of machines is screw compressors, which run rougher than turbo compressors but not as rough as reciprocating compressors. In order to give a help to those people who have to perform such measurements at screw compressors and roots blowers, the German association VDI initiated a working group with several experts for this kind of machines. As a result of this working group the VDI standard 3836 “Measurement and evaluation of mechanical vibration of screw-type compressors and roots blowers, Addition to DIN ISO 10816 - 3” [8] was published in August 2006. Besides of the measurement position, there are different frequency ranges described with dedicated values for housing vibrations and relative rotor vibrations.

Table 1: Recommended limits for housing vibrations as RMS-values of the vibration velocity in mm/s, VDI 3836 [8]

Machine group	Type of installation	Zone limit	Frequency range A	Frequency range B
Group 1 Screw compressors with sleeve bearings and timing gears *P* ≥ 55 kW	rigid	I/II II/III	8,0 12,0	3,0 4,5
	resilient	I/II II/III	10,0 15,0	4,5 7,0
Group 2 Screw compressor with roller bearings or roller and sleeve bearings and timing gears *P* ≥ 37 kW	rigid	I/II II/III	10,0 15,0	3,0 4,5
	resilient	I/II II/III	12,0 18,0	4,5 7,0
Group 3 Screw compressors without timing gears *P* ≥ 55 kW	rigid	I/II II/III	8,0 12,0	3,0 4,5
	resilient	I/II II/III	10,0 15,0	4,5 7,0
Group 4 Roots blowers with roller bearings *P* ≥ 22 kW	rigid	I/II II/III	10,0 15,0	4,5 7,0
	resilient	I/II II/III	12,0 18,0	7,0 11,0

Note 1: These values apply to measurements taken in the radial direction at the housings of the compressor and to measurements taken in the axial direction at thrust bearings during steady-state operation at nominal speed or within the intended speed range. They must not be used if the compressor passes through transient states (for example, changing speeds or loads).

Note 2: Different and/or higher values can be permitted in the case of special compressors or special subframe and operating conditions. In such cases, this should be agreed between manufacturer and customer. Experience shows that there are compressors, particularly high-speed and lightweight compressors, whose vibrations lie above the values given here but which, nevertheless, do not incur damage over years of continuous operation.

The frequency ranges are defined as follows:

Frequency range A:

10 Hz to 1,000 Hz (at least the 3rd harmonic of the output frequency is registered)

Frequency range B:

10 Hz to 2.2 times of the rotational frequency

For the evaluation zones in table 1 the following definitions are registered [8]:

Zone I: Compressors whose vibrations fall within this zone are regarded as suitable for running without restrictions in continuous operation.

Zone II: Compressors in this zone have high vibration values. A check must be made in each individual case to see whether the measured values are permissible for unlimited continuous operation taking into account the design and operating conditions of the unit in question. In general the compressor may, however, be run for a limited time in this state until a suitable opportunity arises for remedial measures.

Zone III: Compressors whose vibration values fall within this zone are usually shut down for examination and repair since their vibration values are regarded as so high that damage could be caused to the machine.

Besides this, there is another standard called EN ISO 10440 - 1 “Petroleum, petrochemical and natural gas industries - Rotary-type positive-displacement compressors - Part 1: Process compressors” [9] which deals with vibration measurements at screw compressors. This standard bases on the API standard 619 [10] and gives vibration limits for dry and oil-flooded screw compressors. These limits differ from those given by the VDI 3836, so it is very important that the involved parties determine the standard which has to be used for the vibration measurements.

Part 2: Sources of dynamic forces and their correlation with measured vibrations

Every user of screw compressors knows that they are noisy, that they have vibrations and are often the reason for problems at the compressor system like cracks at pipe works small bore nozzles or problems at the compressor itself like bearing failures or other machine parts which are wearing. Almost every time the gas pulsation due to the unsteady flow is expected to be the main source for the vibrations. Fluid dynamical resonances in the pipes can cause very high pulsation amplitudes and out of this excite the whole system. Also a bad adjustment of the internal compression ratio to the operation conditions can cause a so called under- / or over – compression, which leads to high pulsation levels. These phenomena’s are well known and out of this, in case of troubles at such systems the experts are mainly focused on fluid dynamical problems and how to solve these. There are a lot of papers describing case histories of successfully solved vibration problems where the pulsations and with this the vibration amplitudes could be reduced by installing orifices, multi hole plates, changing of discharge pipe wall thickness because of a resonance of the natural frequency of the pipe shell wall with the pocket pass frequency and stiffening small bore nozzles with additional brackets.
The common strategy to analyse screw compressor systems with high vibrations is to:

- measure and analyse bearing housing and pipe work vibration

- if possible measure the gas pulsations
- do bump tests to identify natural frequencies
- do fluid dynamic calculations to check for resonances

Because the vibration situation can change significantly with the operation conditions of the compressors, it is necessary to work out a test procedure for the measurements which cover all the operation conditions to get assessable data's.

In our case history the described measurements were done consecutively. The results showed:

- Very high vibration levels at the compressor and at the pipe work up to 40 mm/s rms and more
- Very high acoustical noise
- Dominant frequencies are pocket pass and multiples
- Pressure pulsations in the discharge pipe of about +/- 0.4 bar
- Pressure pulsation levels and bearing housing vibrations did not correlate

With these first results it was clear that the pressure pulsations could not be the root cause for the very high vibrations. So other sources of dynamic forces must play a major role in this case. Possible sources were identified:

- Weak frame with natural frequencies in the near of pocket pass
- Exciting the torsional natural frequency of the rotor system
- Exciting the natural frequency of the compressor rotors

FEM calculations were done for the rotor system and the frame and the measurements were extended. Additional measurement points at the frame should give a clearer understanding of the vibration mode shape of the frame. A special sensor was installed to measure the dynamical torsional stresses at the coupling spacer and eddy current probes (non contactless) were installed to measure the relative shaft displacement at the male rotor between the compressor and the coupling hub.

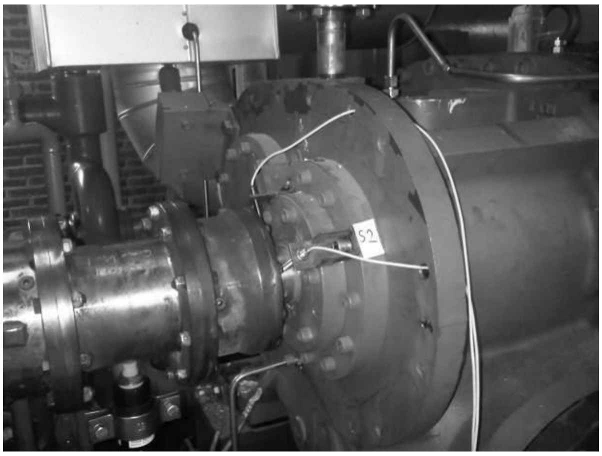


Fig. 1: Installation of eddy current probes and torque sensor at male rotor

FEM calculations

It was a little surprise that the rotor calculations showed that the natural frequencies of the rotors are in the near of the pocket pass frequency as the major excitation. The torsional natural frequency was also calculated and found well beyond dominant excitation frequencies. The frame showed also resonance situations. To solve the vibration problems it was decided to stiffen the frame and to mount the frame with anchor bolts directly to the concrete instead of mounting on rubber plates.

Further investigations and measurements were done for the rotors. A rotor dynamic calculation was done by the user BASF SE to double check the modifications which were done by the manufacturer.

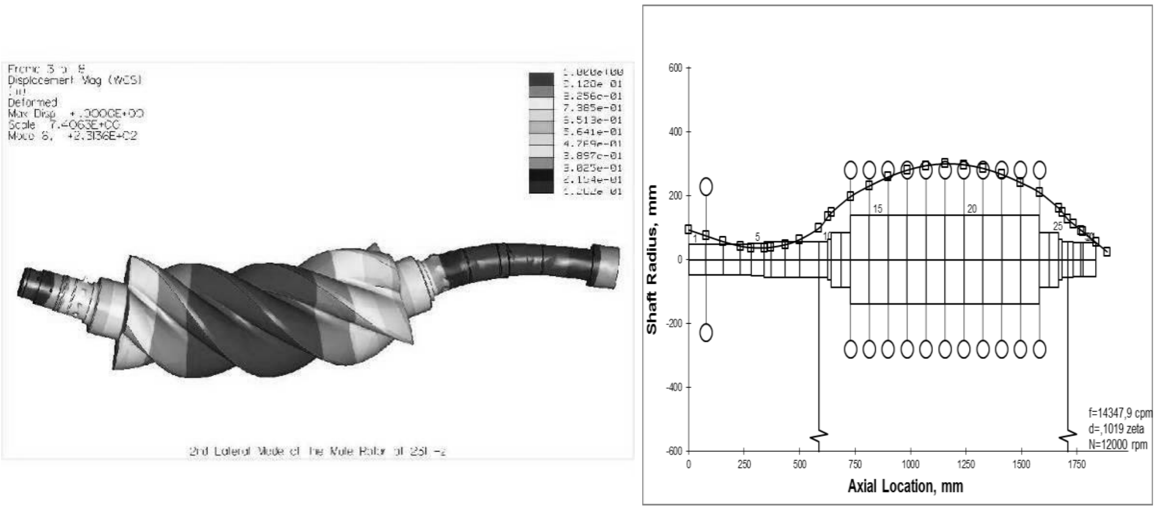


Fig. 2: FEM and rotor dynamic model

The rotor dynamic model was adjusted to match the calculated Eigen frequency values of the FEM calculation with the same bearing stiffness.

Now it was possible to apply a dynamic force to match the relative shaft displacements which were measured during the test with the additional eddy current probes.

After this procedure it was possible to modify the model according the proposed changes and to run the rotor response calculations.

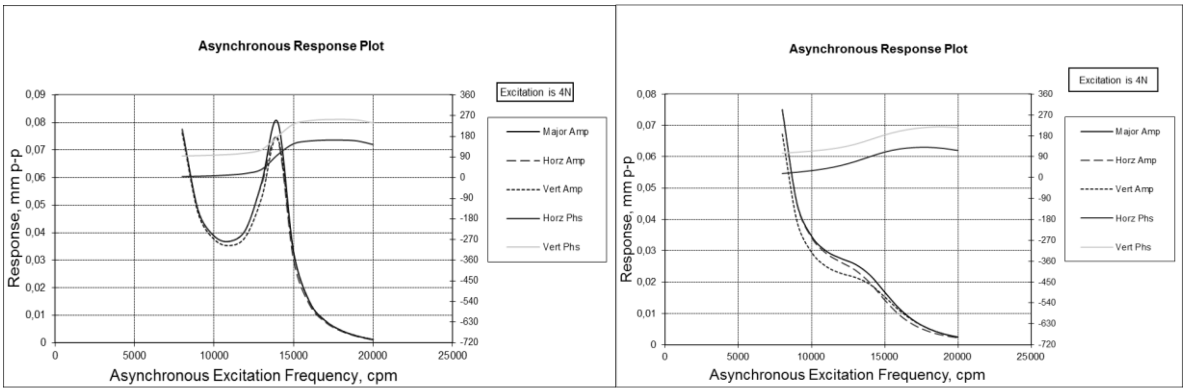


Fig. 3: Rotor response calculations original and modified rotor – bearing system

The response plots show that with the modifications at the rotor – bearing system the vibration amplitudes of the rotor could be lowered by a factor of nearly 4 at the pocket pass frequency.

Measurement with the modified compressor showed also that the relative shaft vibrations and the bearing housing vibration were lowered by a factor of about 4.

Compressor foundation

During the first test runs of the compressor mounted originally on rubber plates no significant vibrations at the concrete were noticed. After mounting the frame with anchor bolts together with the concrete the vibrations travels through the whole concrete base plate with significant levels of about 1 to 2 mm/s rms at points which are quite far away from the compressor unit. With these vibration levels bearing failures at stand by machines can be expected.

The reason for this modification was to stiffen significantly the frame and as a consequence to reduce the vibration amplitudes at the unit. The measurements showed that the difference of the vibration amplitudes at the compressor itself did not change significantly if the frame is mounted to the concrete or not.

A short calculation shows that this result was foreseeable.

Assuming that we have an anchor bolt of 16 mm diameter and a length of 300 mm and expecting vibration amplitude at 200 Hz of 5 mm/s rms. This gives peak displacement amplitude of 5.6 µm and needed dynamic force acting at the bolt of about 790 N.
The example shows that due to the elasticity of the steel it is nearly impossible to reduce the vibration amplitudes at these high frequencies.

The case history shows that mechanical, lateral rotor vibration has a significant influence at the vibration levels of the whole compressor unit and can be a major source. Good compressor design is needed to reduce this excitation to get a sound compressor unit.

Part 3: Failure modes and their expected root causes

Oil injected screw compressors are used in industrial refrigeration for compression of ammonia and other refrigerating gases. The schematic diagram of a screw compressor is shown in Fig. 4

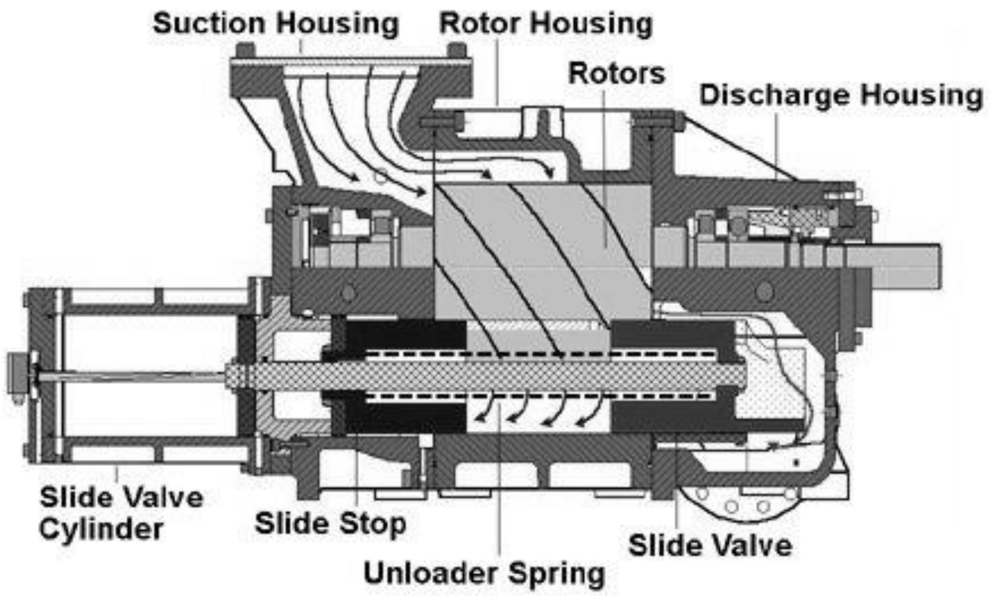


Fig. 4: Screw Compressor with capacity control Slide Stop, Slide Valve [11]

High reliability with continuous operating period of five years, low maintenance costs and a very wide overall operating range are some expected specific features of oil injected screw compressors installed in industrial refrigeration for compression of ammonia and other refrigerating gases from an end user point of view.

Experience shows, however, that continuous operating period of five years cannot, or can only be achieved with additional expenditure. There is nothing really new that dirt in the pipe system and liquid carry over will cause problems for years (see Fig.5/6/7/8). It is highly recommended to keep dirt out of the system during fabrication.

Fig. 5: Marks (male rotor) due to dirt

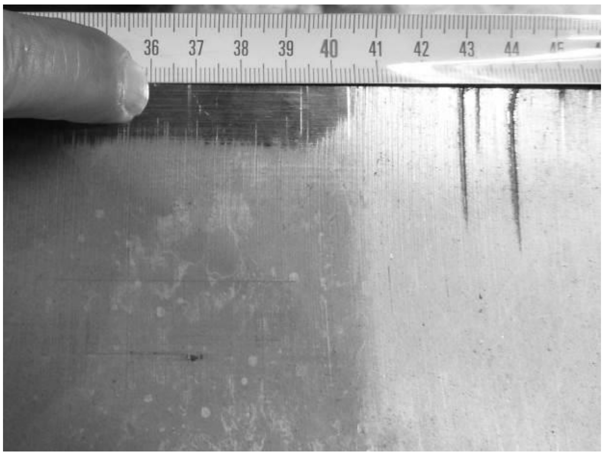


Fig. 6: Marks (slide valve) due to dirt

Fig. 7 Wear (male rotor) due to liquid carryover

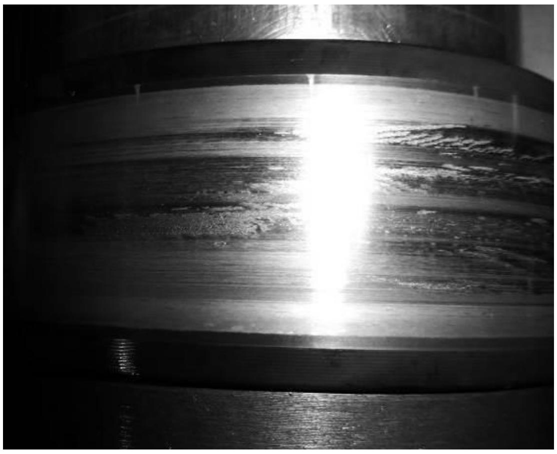


Fig. 8 Wear (inner ring) due to liquid

Slide valves controlling the discarge port are a very common type of capacity control device used in screw compressors. The position of the slide valve can be controlled automatically or manually to provide an infinite number of positions. This gives the machine the ability to flow the amount of gas required by the process. In some cases the machine control system is not aligned with the design of the entire compresssor unit. Thus leading to intense movements of the slide valve with distinct wear at the sliding surface (see Fig.9/10/11).

Fig. 9: Detail Slide valve scuffing

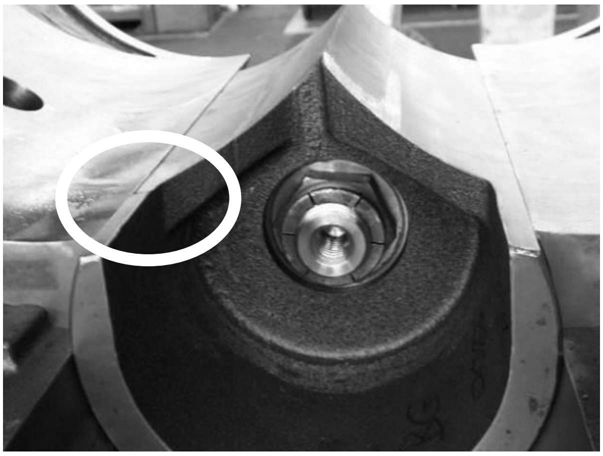


Fig. 10: Slide Valve scuffing

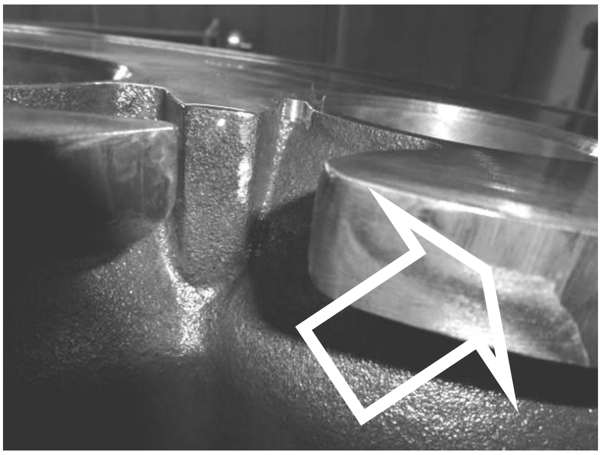


Fig. 11: Discharge housing scuffing

The male and female part of the slide valve is different in the size of their surfaces. In combination with the internal pressure an additional torsional moment is generated and the slide valve touches the male and female rotor. Depending on the machine design and the load condition heavy wear on the slide valve and the rotors appear (see Fig. 13/14/15/16).

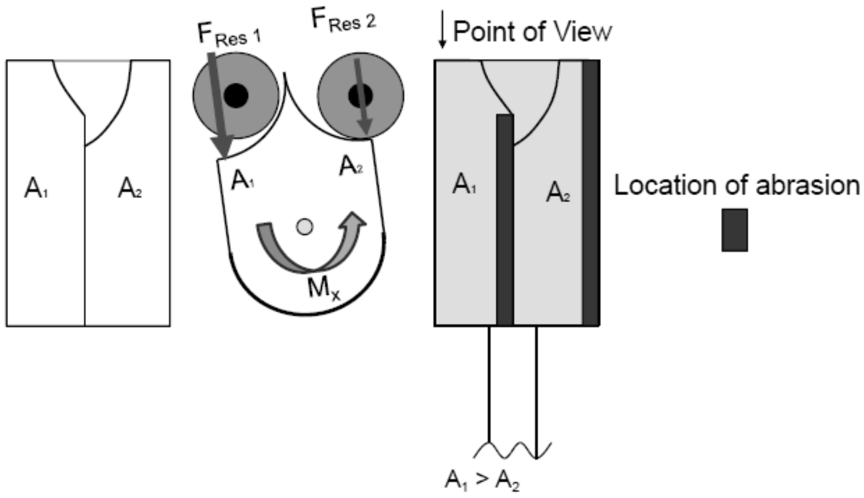


Fig. 12: Wear on slide valve due to resulting moment

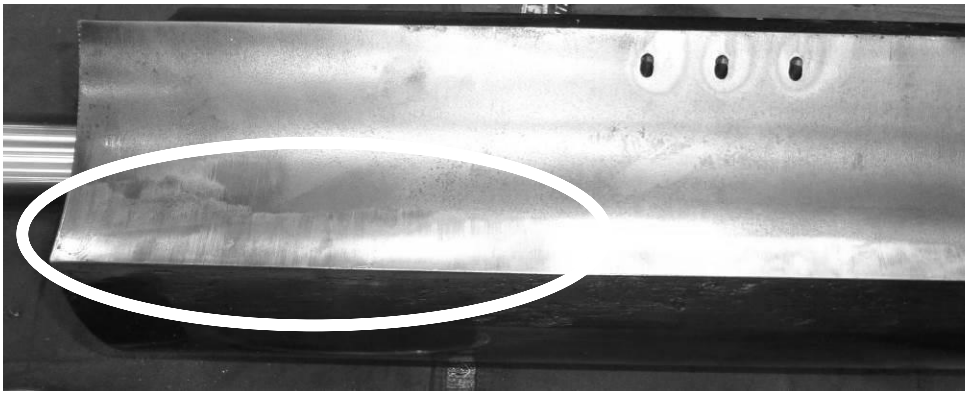
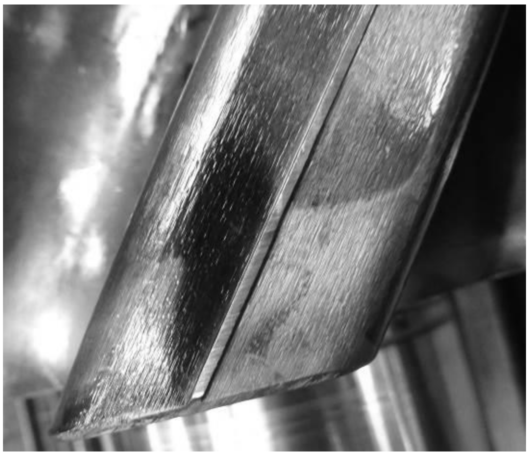


Fig. 13: Wear on slide valve due to contact with male and female rotor

Fig. 14: Wear on male and female rotor due to contact with slide valve



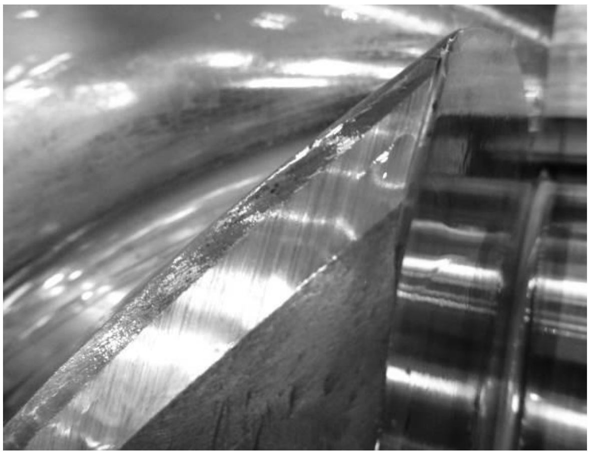


Fig. 15:Sealing lips in a good shape Fig. 16:Sealing lips heavily worn out

As the rotors are fully lubricated they act like a cycloidal gear. In normal operation the male rotor drives the female rotor and the rotors contact each other on their driving sides. For rotors in abnormal operation, however, the screw rotor makes contact on both their driving sides (see Fig.16/17). Abnormal vibration behavior and intensive wear appears. Only proper design and a conservative selected L/D ratio will lead to reliable compressor units.

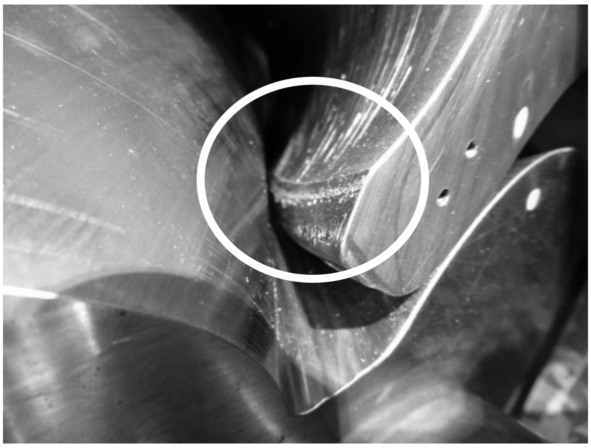


Fig. 16: Contact on both driving sides Fig. 17: Contact on both driving sides

The failure mode analysis shows that the basis for reliable screw compressor operation has its origin in proper machine design. A close relationship between original equipment manufacturer (OEM), universities and research centers, end user and maintenance is required to discuss the occurrence of events and to develop reliable solutions.

Part 4: Correlation of failure modes and vibration measurements

The measurements during the test runs at the compressor of the case history in part 2 of this paper, showed at different operating conditions (slide valve position etc) very different vibration behaviour. At some conditions the vibration amplitudes and also the noise was very unstable while at other conditions the trend shows a stable situation. The absolute vibration level is therefore nearly constant.

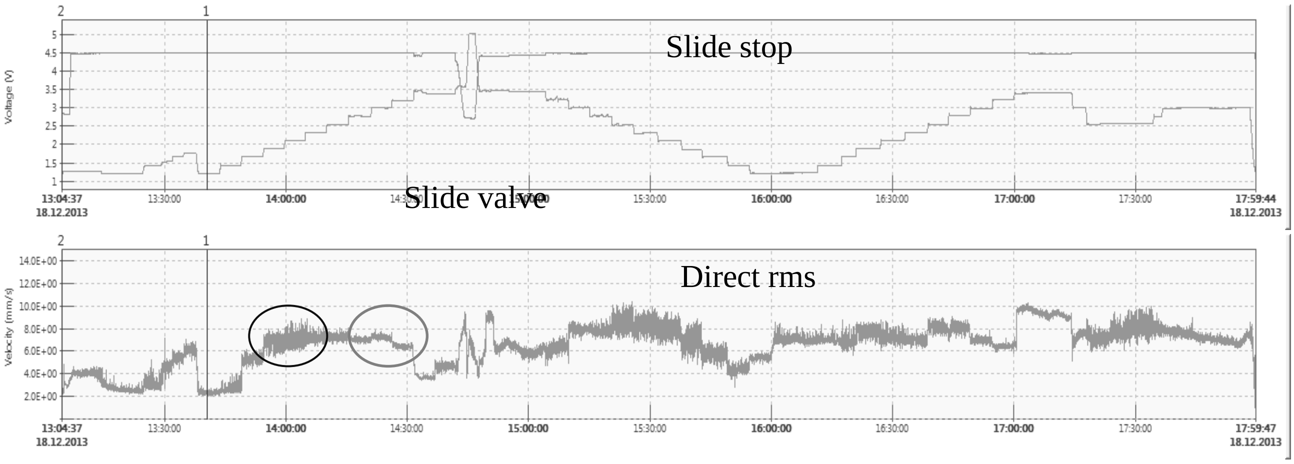


Fig. 17: Trend overall vibration amplitudes black= unstable; grey= stable

If these unstable situations are the root cause for the wear, rotor and bearing failures and the reduced lifetime of compressors, the question raises whether not only the vibration amplitude but also the stability of the values over time during a constant operation condition should be taken into account to qualify a compressor system.

Also as shown in the part 3 of the presentation wear at the slide valve and the rotors are affected by the compressor control system. Permanent adjusting of the slide valve due to a sensitive control algorithm has a negative effect for these parts. Out of this the question from the user side comes up, not only to design the control algorithms for best efficiency but also for longer lifetime and reliability of the compressor.

References

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[10]	API Standard 619 2010-12	Rotary-type positive displacement compressors for petroleum, petrochemical and natural gas industries, 5th edition
[11]	Training information	York by Johnson Controls, www.johnsoncontrols.com