Development and application of vibration improvements

by:

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Abstract

Piston compressors are used in very different industrial sectors. Big advantages of these compressors are the variable operating conditions, the good efficiency and the manifold possibilities of flow regulation. The dynamic behaviour of the piston compressors and the connected piping depends strongly on these basic conditions. Therefore, this influence must not be neglected in vibrational investigations. Compared to other compressor systems, at piston compressors the influence of acoustic induced vibration has to be considered together with the connected pipe work. Increased structural vibrations can be caused by mechanical and often by acoustic resonances, too. Accordingly, measures for an effective vibration reduction have to be adjusted. In this article an approach for a vibration investigation is described on the basis of two case-studies with adjusted measures applicable at acoustic and mechanical resonances.

1 Case study A: Mechanical resonance problem

In a cavern storage system natural gas is fed into underground storage using reciprocating compressors at pressures of up to 160 bar to cover consumption peaks. The examined reciprocating compressor is equipped with 4 cylinders in Boxer design and is operated at speeds varying between 270 and 350 rpm in 2 pressure stages. High pipeline vibrations were observed at the pipeline between the two pulsation vessels on the suction side (figure 1).

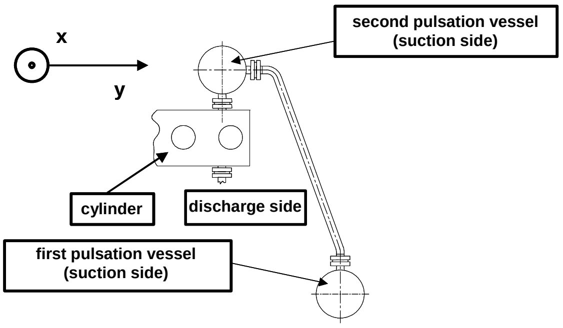

Figure 1: Pipeline between the first stage pulsation vessel on the suction side.

At first, the current situation was analyzed by taking measurements. The speeds of oscillation were recorded at different operating speeds at three positions with the compressor at a constant load (figure 2). Figure 3 shows an excerpt of the effective vibration velocity in one direction.

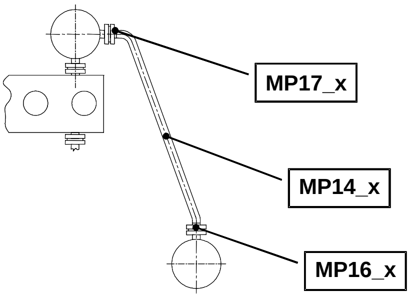

Figure 2: Measuring positions to record the pipeline vibrations.

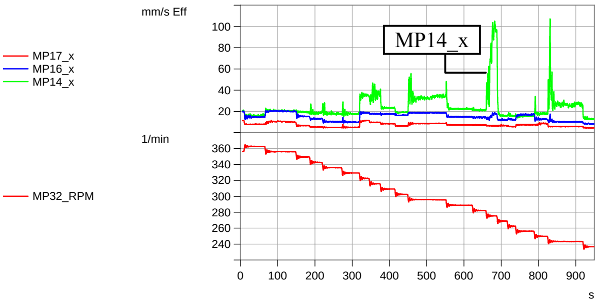

Figure 3: Current pipeline vibration situation correlated with the compressor speed.

It can be seen that the vibrations in the middle of the pipeline increase strongly. To analyze this in more detail, figure 4 shows the amplitude spectra for measuring point MP14_x (middle of the pipe) generated over time as a color chart.

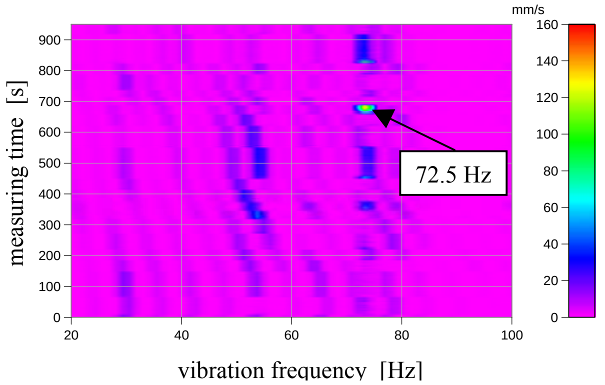

Figure 4: Amplitude spectra of the measured operational vibration from figure 3 at measuring point MP14_x as a color chart.

The maximum vibration level arising during operation (figure 4, time T = 680 s) occurred at a frequency of 72.5 Hz (16 x operation speed). To examine this further, bump tests were performed on the pipeline section when the compressor was shut down (figure 5).

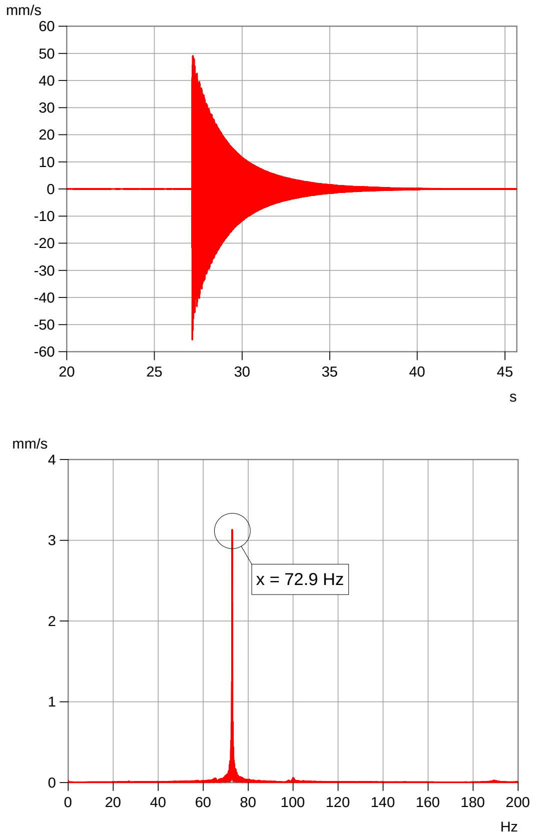

Figure 5: Time signal (above) and spectrum (below) of the bump tests at measuring point MP14_x (response).

From the bump test it can be seen that the pipeline has a weakly dampened resonant mechanical frequency of approx. 73 Hz. Additional pressure pulsation measurements during operation of the plant indicated that the excitation level of the pulsations was of a negligible magnitude. It was deduced that the mechanical excitation of the 16^{th} order frequency of rotation was the actual cause of the high vibrations. Reduction measures such as detuning the system (by stiffening, additional mass) were not promising because the speed of the compressor was variable. Additional dampening using a pipeline damper was also not recommended because of the low efficiency of such dampers at high frequencies. Therefore, a dampened vibration absorber tuned to the frequency of the pipeline vibrations was suggested as an alternative. Furthermore, the absorber has the advantage that the damping forces for example do not have to be transmitted to a supporting device.

1.1 Theory of the vibration absorber

Vibration absorbers compensate the excitation forces using mass forces so that at certain frequencies individual points in the structure remain at rest or at least vibrate much less. There are two main types of vibration absorbers:

- Standard vibration absorbers with a relative small damping effect only work at a fixed or only slightly varying excitation frequency.

- Dampened vibration absorbers with a relative large damping effect work in a wide range of excitation frequencies.

In the following, the theory of the vibration absorber will be explained by using a simple model and without a detailed explanation of the corresponding mathematics. An oscillator with one degree of freedom describes the basic vibration system.

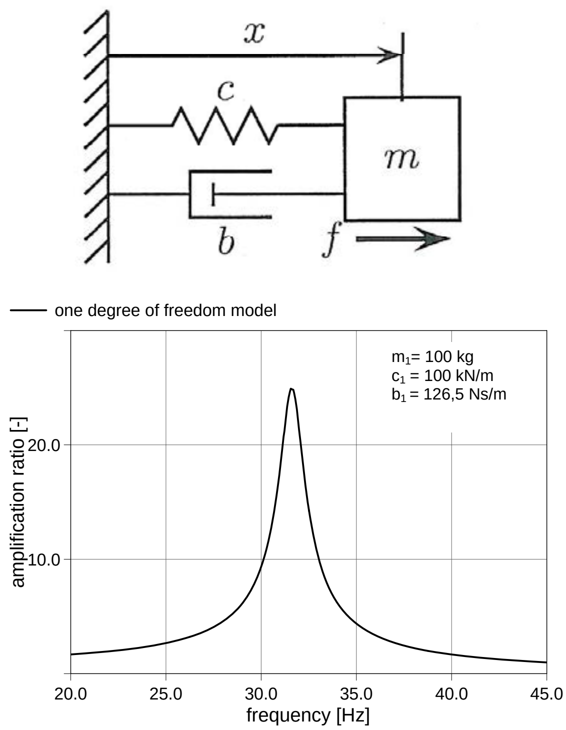

Figure 6: Frequency response function (RFF) of the one degree of freedom oscillator with the corresponding amplitude response.

The model consists of mass m, spring with stiffness c and damper with damping constant b. When the excitation frequency Ω matches the natural frequency of the system $\omega = \sqrt{\frac{c}{m}}$, the result is a high amplitude (resonance case).

To describe the dynamic behavior of the system, figure 6 shows the frequency response (FRF) for the specified parameter values (m, c, b). With the help of this simple model a lot of vibration problems can be described. In the following the absorber will be explained based on this model. The one degree of freedom system (figure 7) is expanded by a second oscillator which represents the absorber.

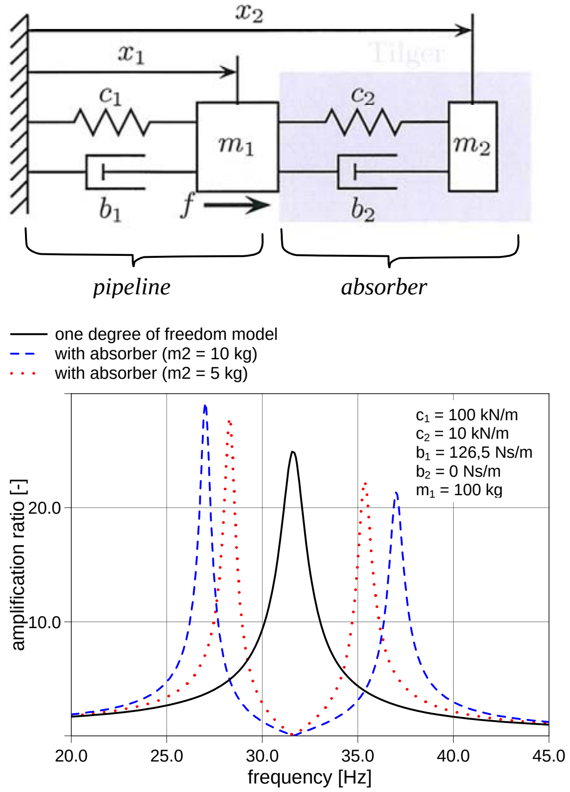

Figure 7: Extended vibration model with an absorber (above) and the correlated frequency response functions (below).

This absorber model consists of the mass m_2, the spring stiffness c_2 and the additional damping described using the damping constant b_2. To absorb the vibrations, the natural frequency $\omega_2 = \sqrt{\frac{c_2}{m_2}}$ has to be adjusted so that it matches the resonant frequency of the original system. Figure 7 shows the frequency response for a system without damping ($b_2 = 0$) and different mass ratios $\frac{m_1}{m_2}$.

The amplitude response in case the absorber mass m_2 is 10 % ($m_2 = m_1 \cdot 0.1$) of the oscillating mass m_1 is compared to the response of the one degree of freedom oscillator (thick line) in figure 7.

It can be seen that two new resonance peaks were generated to the left and right of the original peak in the amplitude response. The actual absorption occurs at the original resonance point $\Omega = \omega$, where the amplitude of the vibration has been reduced to 0.

This also has the disadvantage of an undampened absorber. The absorber system detunes the initial system so that two new resonance peaks arise directly next to the old resonance peak. If the excitation frequency is not constant vibration problems can arise in the neighboring frequency ranges.

To avoid this the extra damper b_2 (see figure 8) is adjusted. This damper reduces the amplitudes at both resonant frequencies.

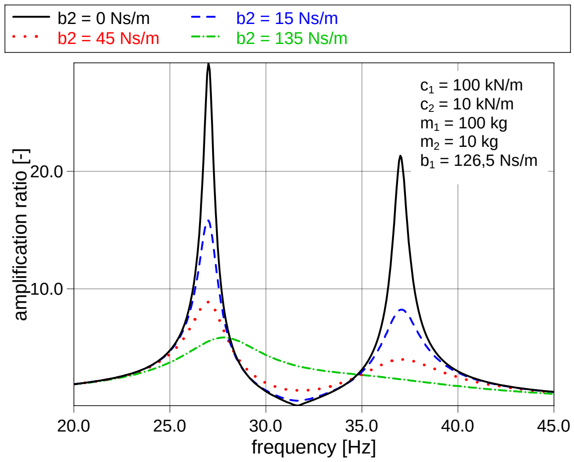

Figure 8: Frequency response and the effect of different absorber damping values on the vibration response.

The higher the damping is selected, the lower are the neighboring resonance amplitude peaks, which means equally poorer vibration absorption at the old location resonance. The damping has to be adjusted for optimization purposes depending on the situation.

1.2 Realization and implementation

For the absorption of the pipeline vibrations at the often existing neighboring resonant frequencies on rotation-symmetric cross-sections, a three-dimensional vibration absorber (figure 9) has been developed.

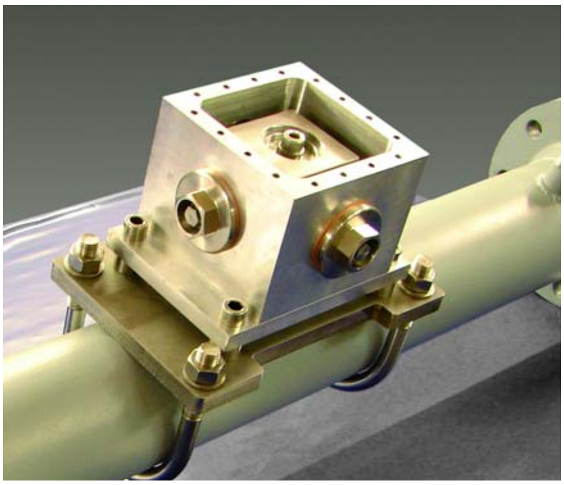

Figure 9: Three-dimensional dampened vibration absorber mounted on pipeline section DN = 100 mm (without housing cover).

The vibration absorber consists of an aluminium housing with a tuned absorber mass and a total of 6 tuned springs. Damping is achieved by filling with a silicone oil with a constant viscosity at temperatures between -40 °C and +150 °C. Figure 10 shows the tuned vibration absorber mounted on the critical section of the pipeline. The absorber was designed in that way that vibrations in both horizontal planes are absorbed.

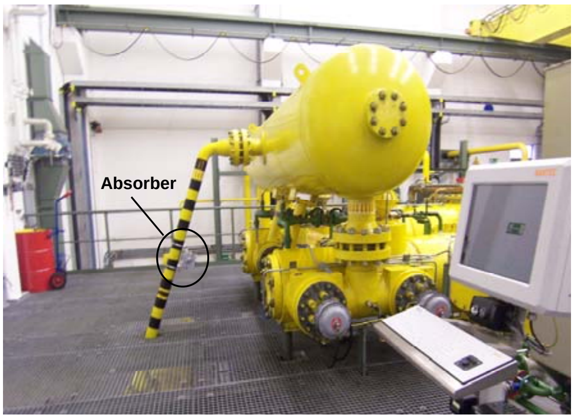

Figure 10: Photograph of the installed three-dimensional vibration absorber.

For the purpose of comparison, figures 11 and 12 show the pipeline vibrations in both horizontal planes with and without the absorber at different compressor speeds.

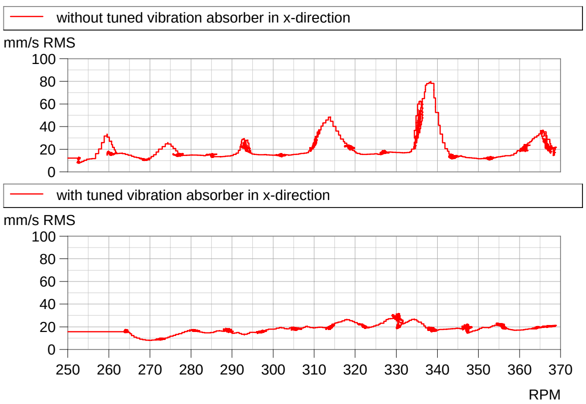

Figure 11: Measured pipeline vibrations with and without the tuned absorber in x-direction.

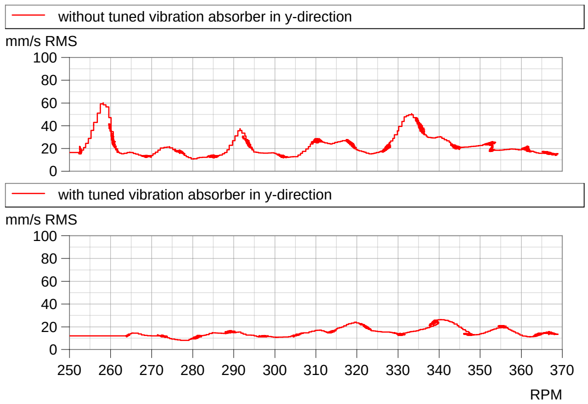

Figure 12: Measured pipeline vibrations with and without the tuned absorber in y-direction.

It can be seen clearly that the individual resonant oscillation peaks are significantly reduced independently from the compressor speed or the direction. The absorber has therefore proven to be an effective measure for reducing pipeline vibrations without any external support when specifically designed for the problem.

2 Case study B: Combination of acoustic and mechanical resonance

As extension of a natural gas compressor plant two VFD driven 6-cylinder reciprocating compressors in Boxer design have been installed. The 2-stage compressors (figure 13) operate between 500 1/min and 1,000 1/min, the e-motor has a maximum power of 4.5 MW.

Figure 13: 1st stage of the 6-cylinder reciprocating compressor plant with VFD-driven e-motor drive.

In order to meet the higher demand with respect to remaining pulsation level within the pipeline system, the pulsation dampers have been designed as 4-chamber-damper. Three of these chambers are directly assigned to the 3 cylinders.

The 4th chamber is implemented as acoustic filter. In turn, this allows a quite compact design of the pulsation damper and simultaneously a good reduction of pulsations towards the connected piping system.

During the commissioning of the compressor plant increased pipeline vibrations were observed. To conduct the root cause analysis and to develop adequate measures, KÖTTER Consulting Engineers have been assigned.

2.1 Measurement based investigation

The conducted measurement based investigation of the pipeline system showed higher vertical vibrations, particularly in the area of the 2nd stage at the compressor cylinder. This phenomenon was detected at both compressors which are designed identically. For a detailed investigation the vibrations (z-direction) in that area were recorded at the cylinders and at the pulsation damper for different speeds of the compressor. The measuring positions are shown in figure 14.

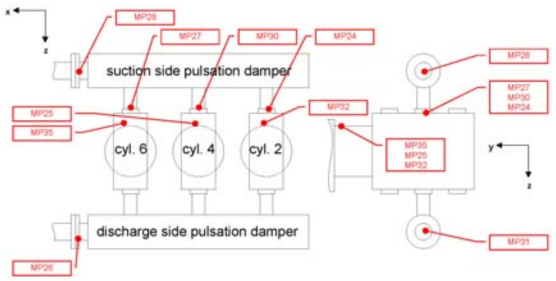

Figure 14: Vibration measuring positions at the 2nd stage of the compressor in z-direction.

The following diagram (figure 15) shows the progression of the measured vibration velocities at the compressor as well as the RPM during the start-up from 500 1/min to 1,000 1/min.

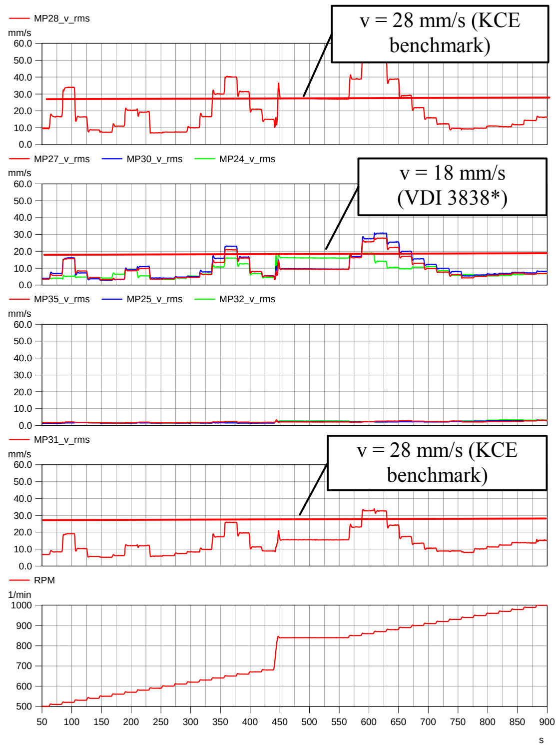

*Figure 15: RMS-vibration velocities (z-direction) and guideline at the 2nd stage of the compressor as well as the RPM (*VDI: Verein Deutscher Ingenieur: Association of German Engineers).*

Beside the increased vibrations of the pulsation dampers at the suction and discharge side considerably higher vibration velocities of up to 31 mm/s appeared at the cylinders (measuring points MP24, MP27, MP30). The cylinder vibrations increased during the run-up, particularly in the RPM-range 520 1/min, 640 1/min and 860 1/min.

The vibrations at the cylinders exceeded the guideline value of 18 mm/s rms clearly. Simultaneously to the increased vibrations at the cylinders, the vibrations at the pulsation dampeners (MP28 and MP31) increased as well. For a detailed frequency analysis figure 16 shows the determined amplitude spectrum of measuring point MP28 as color chart generated over the measurement time (T > 800 s).

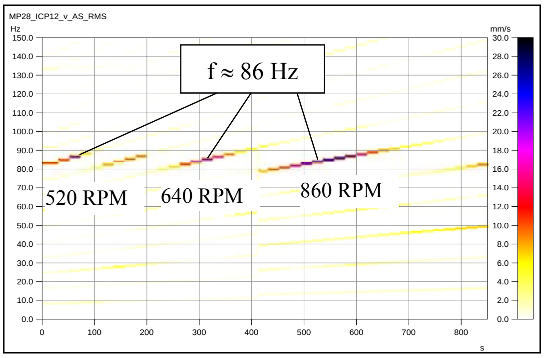

Figure 16: Color chart of the amplitude spectrum during run-up at MP28, RPM-range 500 1/min – 1,000 1/min.

A comparison of figure 15 and figure 16 shows the appearance of increased vibrations at different points of time (T1 = 70 s, T2 = 310 s, T3 = 600 s) always at a frequency range of about 86 Hz at MP28. This was ascertained for all three measuring points.

To identify the responsible cause and effect mechanism, the synchronously recorded pressure pulsations upstream to the suction side - and downstream to the discharge side pulsation dampers were analyzed. It resulted that the pressure pulsations do not correlate to the recorded vibrations.

Subsequently conducted impact hammer investigations during standstill of the compressor could not detect any prominent mechanical resonance frequency at 86 Hz. It has to be considered that impact hammer tests during standstill in the area of the cylinders do not show definite results due to mass oscillation (piston etc.) of the running compressor.

Out of the operation vibration analysis it showed that the phase of the vibration signal turned during the increasing of the amplitude changed. The phase turned to 90° in the maximum and to 180° after reaching the next vibration minimum. This indication defines the position of an additional mechanical resonance frequency at 86 Hz. Figure 17 shows the measured vibration mode in the following three points of displacement.

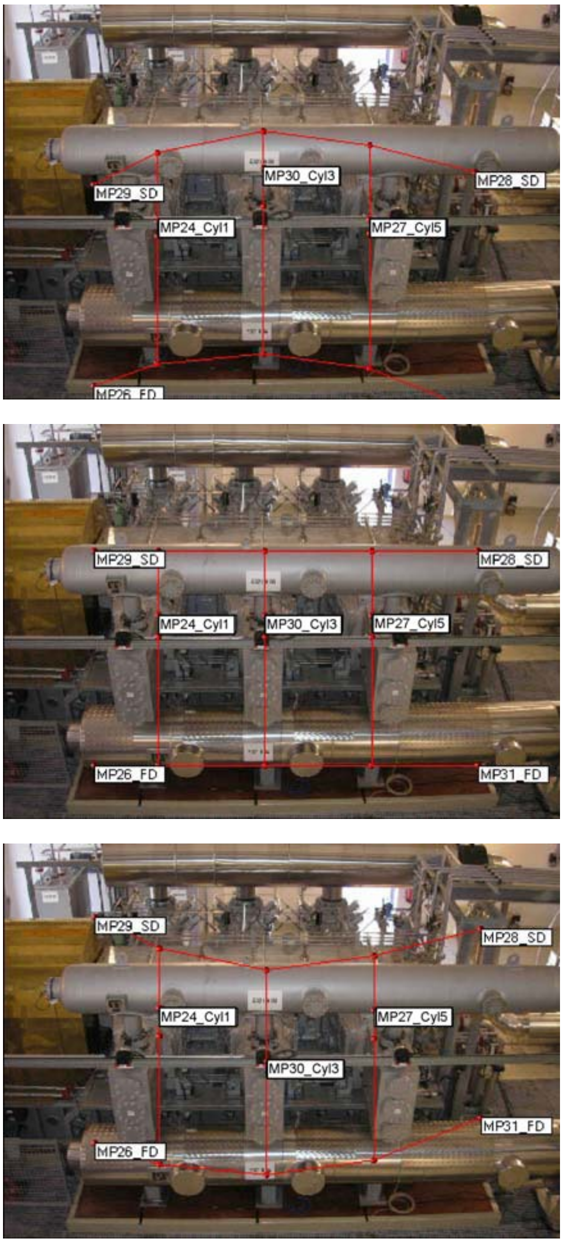

Figure 17: Description of the vibration mode of the 2nd stage of the compressor at 86 Hz.

This vibration mode always turned up at the compressor speeds of 520, 640 and 860 1/min. For a deeper investigation of the cause and effect mechanism, one cylinder-internal pressure (head end) was recorded synchronously.

The analysis of the cylinder pressure trend showed that at the moment of increased vibrations - during the exhaust stroke of the gas - pressure pulsation of approx. 86 Hz occured (see figure 18).

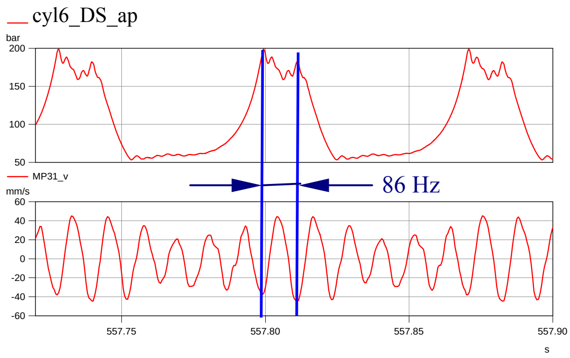

Figure 18: Trend of internal cylinder pressure (head end) and vibration velocity at measuring point MP31.

Based on the typical operating conditions, an additional acoustical calculation of the 2nd stage was conducted to verify the pressure pulsation situation. The check confirmed that an acoustic resonance appears between the cylinders and the pulsation damper of the discharge side at around 85 Hz.

With this result the cause and effect mechanism, which led to the higher pipeline and cylinder vibrations, was clearly worked out. Due to the acoustic resonance at the cylinder outlet of the 2nd stage combined with a mechanical resonance highly increased vibrations occured.

2.2 Realisation of measures

For an effective reduction of the vibrations different possibilities were checked with calculations. Figure 19 shows the design and the used acoustic model of the discharge pulsation damper (2nd stage).

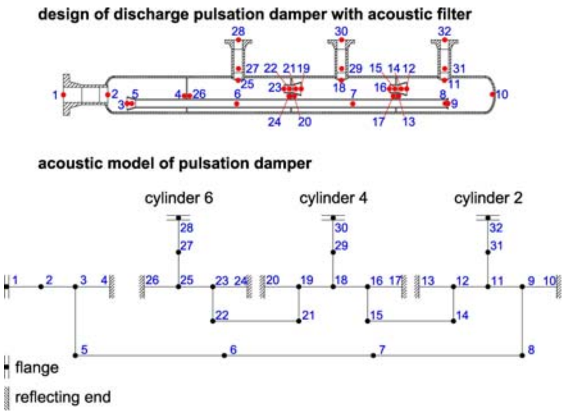

Figure 19: Design and acoustic model of the discharge pulsation damper.

Finally, modified taper-lok-plates in KÖTTER design (see figure 20) were installed between the cylinders of the 2nd stage and the pulsation damper of the pressure side. The advantage of this patented design is a good ratio between pressure drop and pulsation reduction at higher frequencies.

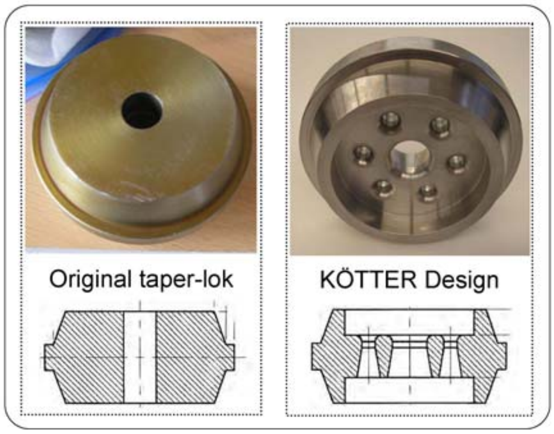

Figure 20: Original and modified taper-lok-plates.

After installation of the modified taper-lok-plates the measurements were repeated under comparable operating conditions (figure 21).

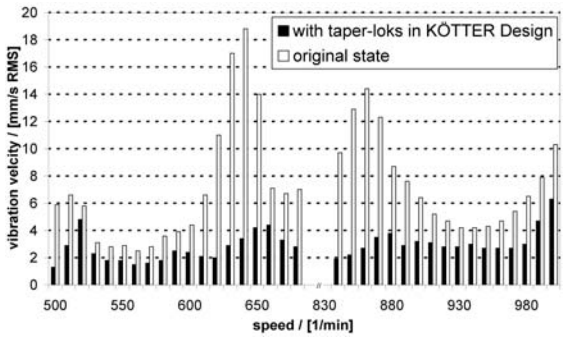

Figure 21: Cylinder vibrations before and after installation of modified taper-loks.

The measurement results confirmed a considerable reduction of the cylinder vibrations below the guideline values. It has been shown that by combining a theoretical and a measurement based investigation at a reciprocating compressor effective measures have been worked out.

3 Summary

This paper shows typical procedures for investigations of vibration problems at reciprocating compressors. The ascertained vibration phenomena have been analyzed in detail, the cause and effect mechanism has been revealed.

In the first example, the higher pipeline vibrations could be traced back onto a weakly dampened mechanical natural frequency. As a measure a three-dimensional vibration absorber has been introduced and explained. Following to the installation of the absorber a measurement confirmed the broad-band vibration reduction at the pipeline of variable speed driven reciprocating compressor.

In the second example, an acoustic resonance between cylinder and succeeding pulsation damper was detected as critical cause and main effect mechanism. As a measure the available taper-lok-plates were modified in KÖTTER design. Again, the subsequently conducted measurement confirmed the effectiveness of the measure.