REDUCTION OF FOUNDATION OSCILLATIONS WITH PISTON COMPRESSORS

With Measured Vibration Data, Causes are Determined and Measures are Created for Lowering Vibration Levels Below Prescribed Limits

By Dr.-Ing. Johann Lenz

Piston compressors are widely used to compress a variety of gases in chemical plants, refineries and natural gas transportation and storage. By their nature, as is characteristic of all reciprocating machines, they produce vibration that is transmitted to the whole compression plant. If not properly controlled and dampened, these vibrations can cause severe damage to the plant, with consequent loss of production.

These vibrations are caused, e.g., by unbalanced mass forces or mass moments and also by gas pulsations in the piping systems receiving compressed gas. The industry developed guidelines indicating the maximum acceptable vibration levels (VDI Guidelines 3838, VDI 3842, API Standard 618). When these guidelines are exceeded, taking vibration measurements under actual operating conditions is important. They often do not correspond to the design parameters of the plant. With operating vibration data on hand, causes can be found. Then plant implementation can be directed toward lowering vibration levels under prescribed limits.

A practical example will better describe the procedure followed by Kötter Consulting Engineers to solve the problem. Two single-stage piston compressors, among other items, are used to compress gas mixture in a chemical plant. These horizontal machines are provided with single-acting cylinders. The compressors are operated at a constant speed of 530 rpm by a belt-connected electric motor (Figure 1). Capacity control is achieved through a bypass and a backflow intake valve lifter. Under certain operating conditions, increased vibration amplitudes were detected in the foundation block near the piston compressor and in the piping downstream of the compressor.

Because damage occurred repeatedly, a specific case analysis of the compressor plant was performed. Based on this, effective vibration-reduction measures were worked out.

The following goals were targeted: 1. determination of the causes of the increased vibration; 2. assessment of the vibration situation and determination of existing hazards; 3. prevention of mechanical damage; 4. suggestions for eliminating the causes present and a reduction of vibrations, even after the planned expansion of the speed range.

Measurements were taken in three directions on the compressor itself, at the cylinder head and on both ends of the crankshaft. Six different points were selected on the foundation block. Locations of the measuring points are shown in Figure 2. An example of the amplitude of the vibrations detected on the foundation block shown in Figure 3 are compared to the orienting values for horizontal foundation vibration (solid line). The value is exceeded everywhere in the direction of the compressor cylinder (Y-axis in Figure 2).

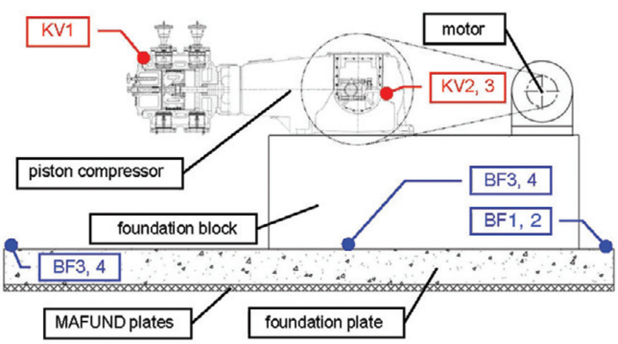


Figure 1. Layout of an electric motor-driven horizontal piston compressor.

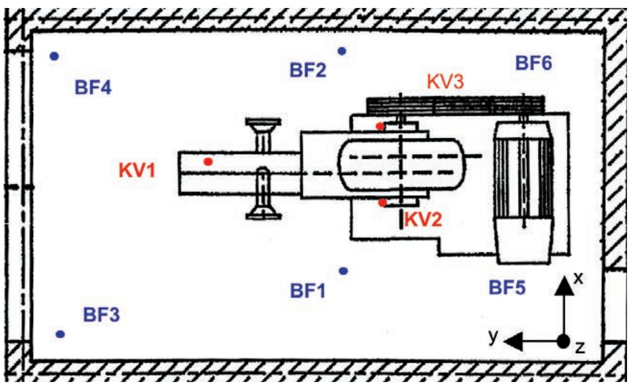


Figure 2. Position of the vibration measuring points on the compressor and on the foundation block.

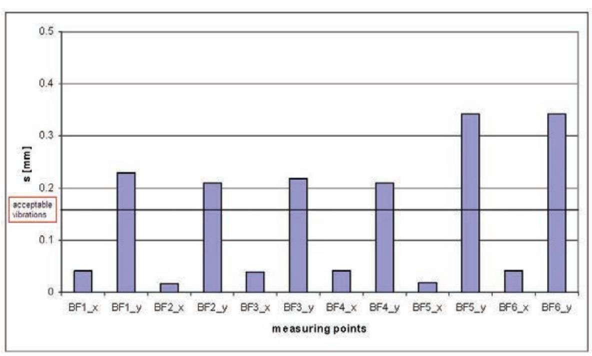


Figure 3. Comparison of guideline values (solid line) to the horizontal vibration amplitudes measured on the foundation block.

Johann Lenz is head of the Structure Dynamic department at Kötter Consulting Engineering KG, Rheine, Germany.

The time signals of measured vibrations on the compressor and concrete foundation block are shown in Figure 4. The compressor was operated at a rotating frequency of f = 8.9 Hz (approximately 530 rpm). The highest vibrations were encountered on the cylinder head, as expected. The additional vibration speeds on the concrete foundation slab (BF1 to BF4) are only slightly reduced by comparison to the vibration speeds on the compressor foundation block (KV1 to KV3).

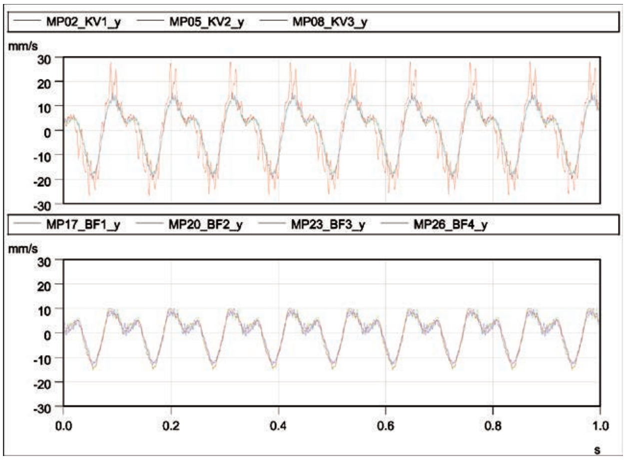


Figure 4. Chronology of the vibration speed measured on the compressor and the foundation block.

Figure 5 shows the measured vibration along the center axis of the piston (Y-direction) as a frequency spectrum to distinguish the frequency components. The main frequency corresponds to the first speed harmonic at f = 8.9 Hz. The second harmonic is located at a frequency of f = 17.8 Hz.

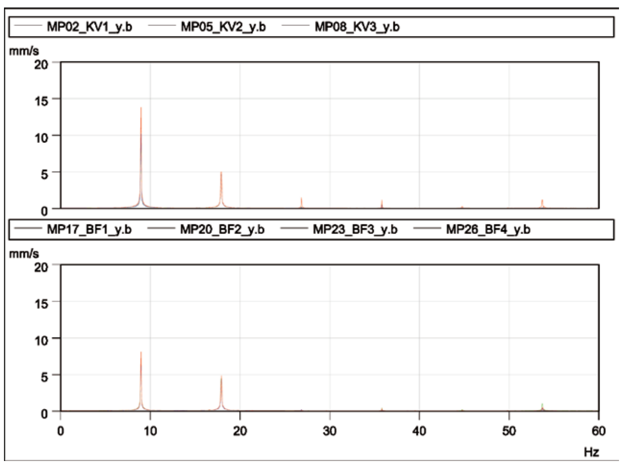


Figure 5. Amplitude spectrum of vibration measured in the compressor and foundation block.

To detect if a resonance response occurs, the first and second order free mass forces of the crank drive were roughly calculated first. The parameters are shown schematically and listed in Figure 6.

The following first and second order free mass forces were calculated with equation 1 and 2:

$F_{1st\ ord} = m\ r\ \omega^{2} \cos\ \alpha$ (1) $F_{1st\ ord} = 49.300$ N

$F_{2nd\ ord} = \lambda\ m\ r\ \omega^{2} \cos 2\alpha$ (2) $F_{2nd\ ord} = 8.800$ N

The Mafund slabs used as vibration isolators have a directionally dependent overall horizontal rigidity of 58.08 kN/mm (direction 1) and 95.04 kN/mm (direction 2) according to the manufacturer. If only the static displacement is based on the free mass forces, then the values listed in Table 1 result.

Free Mass forces	Vibrational displacement in direction 1	Vibrational displacement in direction 2
1st order	0.85 mm	0.52 mm
2nd order	0.15 mm	0.09 mm

Table 1. Calculated vibration displacement from "static" excitation by the free mass forces for both direction-dependent horizontal rigidity values.

The main difference between equation 1 and 2 and the first and second orders, respectively, is the connecting rod ratio, λ. When compared to the measured vibration amplitude spectrum in Figure 7, sufficient agreement occurs with the simplified calculation in direction 2 (specifications of the actual installation direction of the Mafund slabs were not available). The effects of the pipeline connection and the side insulation were not taken into account any further.

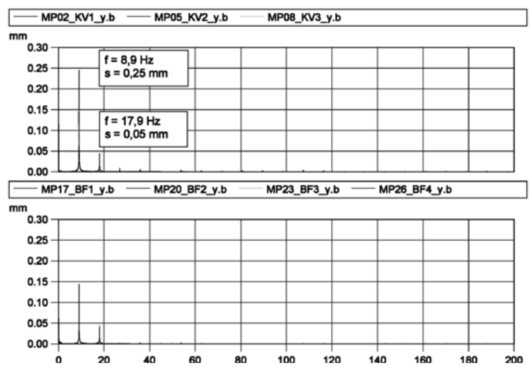
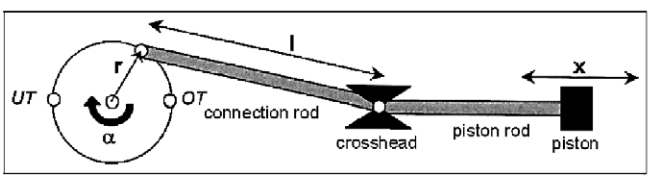


Figure 7. Amplitude spectra of the measured and integrated vibration displacement on the compressor and foundation block.

Based on these results and the boundary conditions of the piston compressor such as the low pressure ratio, any additional forces possible that could possibly excite the foundation were considered negligible.

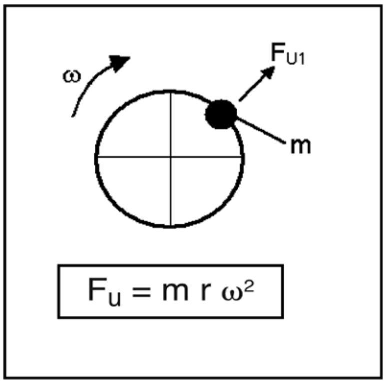
To reduce the vibrations detected in the foundation, the high excitation forces should be reduced or the principal design of the foundation should be changed. Since reconstruction of the foundation and modification of the piston compressor to include an additional complete balancing of masses would have been very expensive, the existence of additional vibration reduction measures was investigated.

In principle, the generation of a counteracting unbalance force, in addition to fully balanced masses, can at least reduce the first-order mass forces (Figure 8).



Connecting rod length	l = 0.45 m
Crank radius	r = 0.08 m
Pulshrod ration	$\lambda = r/l = 0.178$
Angular speed	$\omega = 55.5\ s^{-1}$
Oscillating mass (piston, piston rod, crosshead, connecting rod section)	m = 200 kg

Figure 6. Schematic of the thrust crank drive and existing crank drive parameters.



(m = unbalance mass, r = unbalance radius, ω = angular frequency)

Figure 8. Example of a simple unbalance F_U as a rotating force vector.

Because the resulting force vector F_U rotates, additional, unwanted forces — for example, vertical force components — are generated in addition to the horizontal balancing forces needed. To suppress these forces, a horizontal force F_{Res} can be generated by carefully adjusting two balance disks accordingly. Figure 9 shows the position of the two balance disks rotating in opposite direction at two different angular positions.

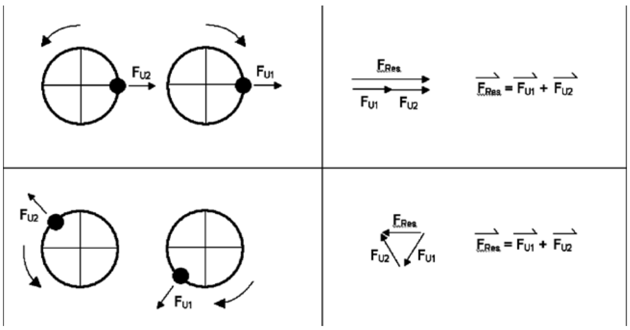


Figure 9. Horizontal force F_{Res} generated by a second, counter-rotating balance force.

The force F_{Res} resulting from the two unbalance forces, F_{U1} and F_{U2}, has only horizontal components regardless of the particular angular position. This fixed relationship and the opposite direction of rotation of the two balancing disks can be implemented using a special unbalance drive unit.

The mass forces generated by the unbalance drive unit must counteract the mass forces developed by the compressor. The magnitude of the necessary mass forces was adjusted for the first-order mass forces of the piston compressor. The position and drive layout of the suggested solution is shown in Figure 10.

Because of the finally adjusted unbalance drive unit, the first-order amplitudes can be completely compensated for in theory. The residual vibrations, belonging to the second-order mass forces are not reduced by the unbalance drive unit.

To avoid possible belt slip, the solution was implemented by connecting the compressor crankshaft to the unbalance unit through two bevel gear units (Figure 11).

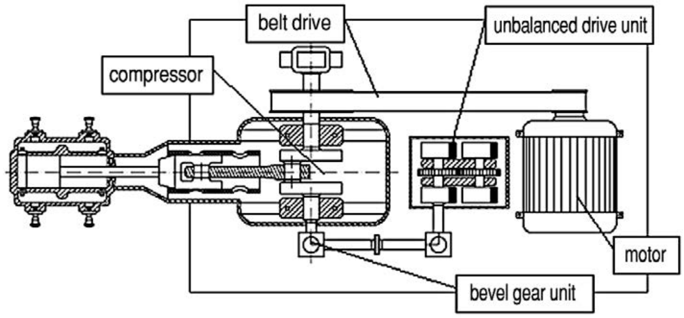


Figure 11. Detailed layout of the unbalance drive unit.

After implementing this system, new measures of the vibration in the foundation were taken during compressor operation. A comparison of the vibration amplitudes before and after implementing the measures is shown in Figure 12.

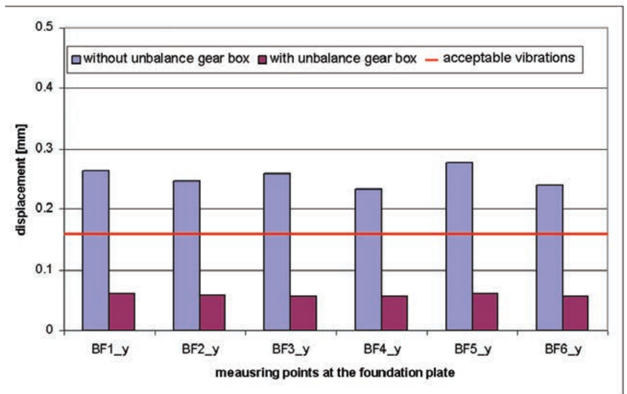


Figure 12. Comparison of vibration in the foundation block before and after installing the unbalance drive unit.

The diagram clearly shows that vibration in the foundation was significantly reduced because of the installation of the unbalance drive. The measured values of the horizontal foundation vibrations were significantly lower than the maximum of 160 μm specified by the guidelines. A problem-free operation of the piston compressor could be guaranteed after drastically reducing the vibration levels.

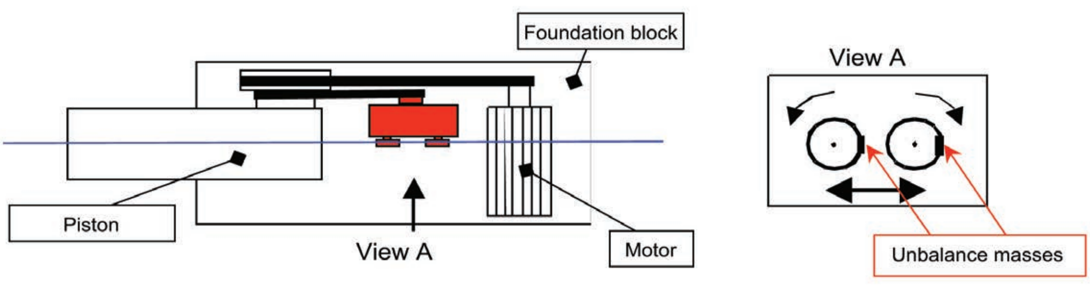


Figure 10. Schematic showing the position of the unbalance drive unit between the piston compressor and motor (left) as well as schematic of the unbalance masses and direction of rotation in the unbalance drive unit (right).