Modernisation of an oxygen compressor plant – field report

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The conversion and expansion of existing compressor plants is a challenge for the operating company, planner and staff, especially with the sensitive medium of oxygen. Using the modernisation of oxygen compression in an air separation plant as an example, this article reports on experiences with project engineering and commissioning. Here one of the focal points is on the approach to the vibration engineering design for the machine installation.

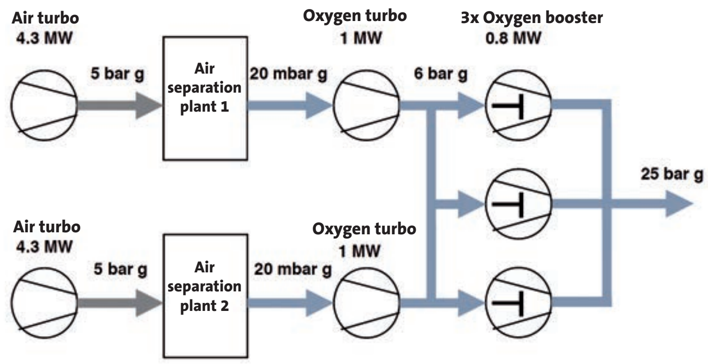

Fig. 2: Compression process schematic (from 1962)

History

The first blast furnace at the Dortmund Hörde (Phoenix) location was put into operation in 1854. As the steelworks expanded over the decades, the demand for oxygen to produce pig iron and steel continued to increase. In 1962 the Dortmunder Hörder Hütten Union decided to construct two new air separation plants. They were constructed on land in the Phoenix-West plant section.

The oxygen capacity of the two plants was 16,000 Nm³/h. After production, the oxygen was compressed and fed into the 25 bar network to supply the blast furnaces and converters.

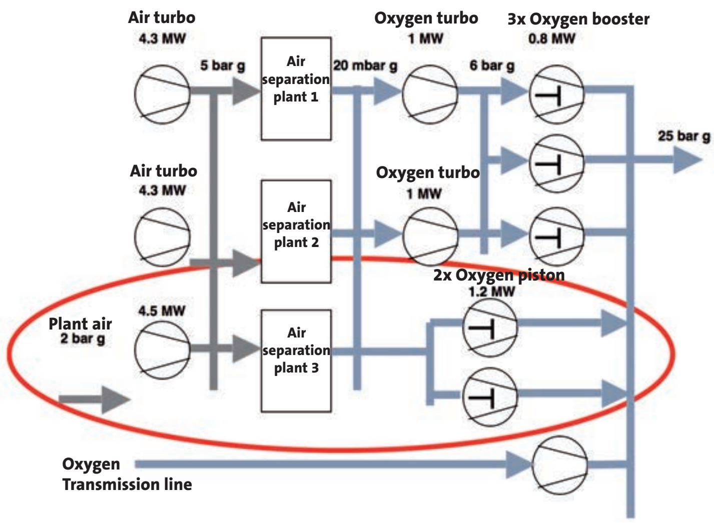

Fig. 3: Compression process schematic (from 1989)

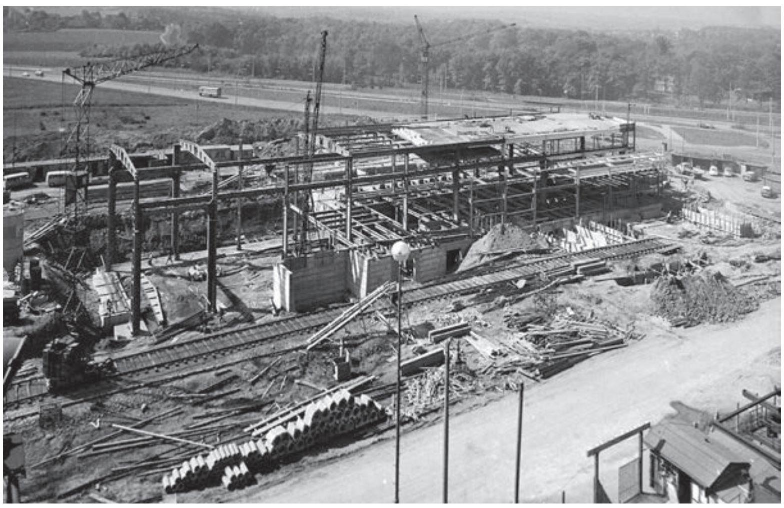

Fig. 1: Construction of the first two plants in 1960

To meet the rising demand for oxygen, the plant was connected to the Rhine-Ruhr oxygen network in 1976. A third air separation plant was constructed in 1989 by the leading producer of technical gases, doubling the existing oxygen capacity at the site.

Due to a preference for the Duisburg location near the Rhine River, the end of steel production in Dortmund was initiated and production ceased in 1998. Oxygen gas therefore became an unnecessary product in this region with this change in circumstances. But since the demand for nitrogen from the roller mills located in Dortmund was starting to increase back in

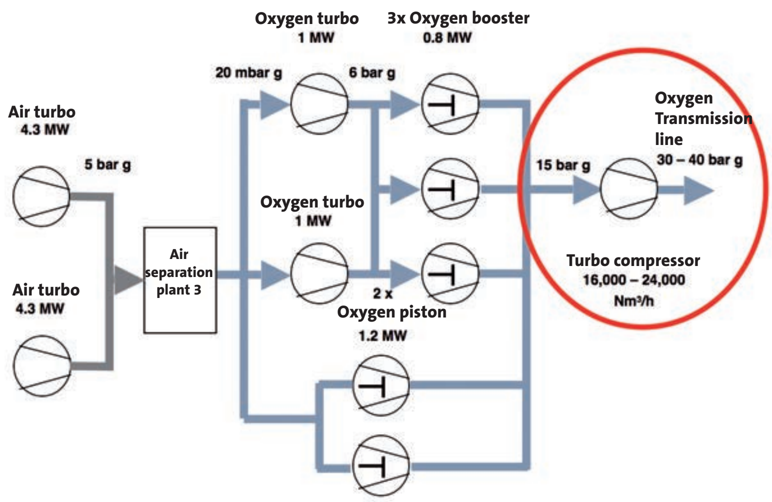

Fig. 4: Compression process schematic (from 1998-2015)

– The compressors have to be installed in the existing machine hall (confined space).
– Operation of the plant has to continue during installation.
– The compressor plant has to comply with the API 618 standard.

Time line and project schedule

The total project term was 18 months. A market analysis was prepared on the basis of technical and commercial aspects. Here the selection of certain parameters was soft on purpose (for instance the maximum flow rate, turn-down and final pressure) in order to obtain the largest possible selection of suitable machines.

the 90s, the liquid products (cryogenic liquefied air gases) were complemented by a nitrogen transmission line to a nearby roller mill.

In order to operating economically, the two old air separation plants were torn down and the transmission line import booster was converted to an export booster (from oxygen importer to exporter).

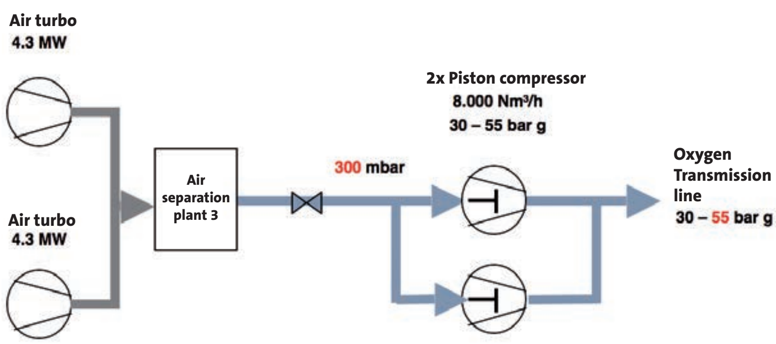

Fig. 5: Process schematic for new O_2 compression concept (from 2014)

Project engineering for the new oxygen compressor plant

Motivation and concept

The existing plants were taken over by the supplier in 2014. Numerous future concepts, retrofitting measures and their investment costs were examined for the takeover. Aside from electrical engineering, the greatest difficulties were encountered in oxygen compression. As previously outlined in the introduction, the plant and compressor constellation changed numerous times over the decades. The equipment itself is mainly from the year the plant was founded (1962).

Due to availability deficits and since the oxygen system was not state-of-the-art, a solution optimised in regards to safe, economical operation with high availability had to be found.

This assessment led to the concept illustrated in Figure 5:

– Installation of two new oxygen piston compressors matching the system pressure, with a delivery volume corresponding to the oxygen equivalent of an air compressor.
– The compressors must have a good turn-down (partial load operation) to avoid losses from blowing off or production adjustments.
– Replacement of the entire pipework system between the air separation discharge and transmission line intake.
– Increase of the maximum final pressure to 55 bar g so the required volumes can be delivered through the transmission line system in a backup scenario.

The following constraints must be met:

– Compliance with the state of the art for oxygen compression (EIGA IGC 10/09, various corporate guidelines, M034 of the Employer's Liability Insurance Association (BG)...).

A highly detailed specification was prepared for the new compressors, listing all relevant machine data (performance, allowable forces, test procedures, quality aspects). This specification was adapted accordingly after choosing the supplier and machine type. Since this was not a classic EPC project, a streamlined organisation was chosen with the most important positions (project management, electrical engineering/instrumentation/control, construction, piping) filled by internal and external personnel. Limiting the interfaces to a minimum was a major objective. The second compressor was initially omitted to reduce the investment costs to a minimum.

Initial rough construction and pipework planning was conducted by means of 3D laser scanning methods. Before choosing the final machine installation location, test drilling was performed at various points on the possible construction site to obtain information about the required foundations. Based on the soil samples, the foundation plans provided by the supplier and the compressor forces, initial dynamic calculations were performed to support preliminary statements about the required foundations. This knowledge is of fundamental importance, especially for construction of the foundations in the existing machine hall.

After establishing the installation of the machines in the existing machine hall, the compressors and foundations were adapted so that two identical machines could be installed. This mainly consisted of altering the pipework and positions of cooling units and tanks. Constructing a continuous rectangular foundation was not possible due to the proximity to the existing main foundations of the machine hall. A special eccentric shape was realised here in order to ensure good isolation from the existing structures.

Various process lines were rerouted and the connection points for the new compressors were prepared during a regular plant shut-down. A new blow-off stack for air and oxygen was also constructed from prefabricated concrete components.

The safety barrier (concrete housing) for the new oxygen compressor was constructed from prefabricated concrete components and set up on three sides. One side was initially left open so as not to impede installation of the pipework and machine components – see Figure 6. The oxygen purity standards require a high level of inspections and quality assurance. Electrical engineering, instrumentation and control were installed after the pipework was complete. All instruments were installed outside the safety barrier to exclude the need to enter the compressor room during operation.

Verification of the vibration engineering layout of the new oxygen compressor plant

Scope of work

Three points in particular (A, B, C) were examined during the planning phase of the new oxygen reciprocating compressor plant for vibration engineering verification. The compressor manufacturer (OEM) conducted a mathematical pulsation and vibration study (A) for the plant as well as a torsion analysis (B) of the drive train consisting of an electric motor, elastic coupling and compressor crankshaft. Calculations were also performed by an external structural engineering firm for the dynamic layout of the compressor foundations (C).

Since the vibration engineering planning of a reciprocating compressor plant is of special importance, the builder also consulted a specialised firm (technical consultant) that performed an independent review of the vibration engineering layout and the corresponding calculations. After putting the compressor into operation, a metrological check of the actual pulsation and vibration situation also had to be documented and a final assessment had to be prepared.

Manufacturer	Burckhardt Compression AG
Model	4D300B-3AB-1
Type	Vertical
Medium	Oxygen
Number of cylinders	4
Number of stages	3
Mode of operation	Double action
Piston rod length	725 mm
Capacity	1,450 kW
Volume control	Variable rotational speed
Rotational speed	225 min^{-1} – 450 min^{-1}
Piston stroke	300 mm
Suction pressure, 1^{st} stage	1.35 bar a – 1.45 bar a
Final pressure, 3^{rd} stage	31.8 bar a – 52.0 bar a
Suction temp., 1^{st} stage	5°C
Final temp., 3^{rd} stage	94°C – 137 °C

Table 1: Technical data and operating conditions for the new oxygen compressors

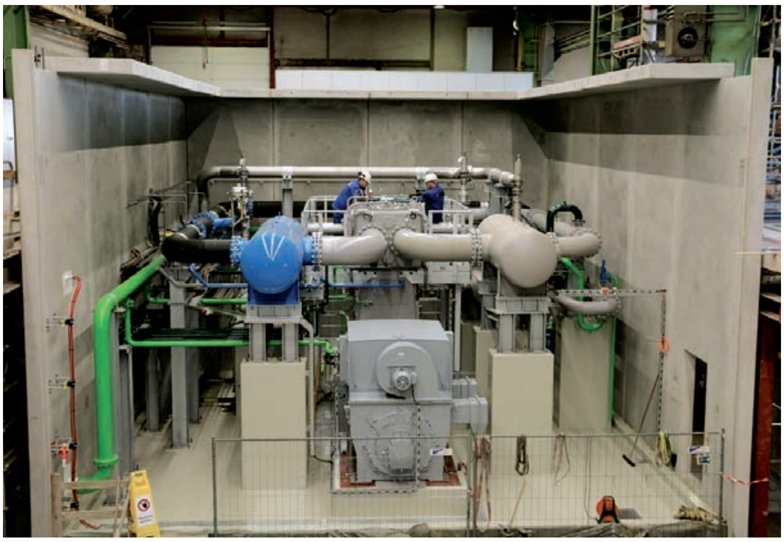

Fig. 6: Compressor M15 shortly before commissioning

Review of the pulsation and vibration study

The pulsation dampers are crucial for operating a reciprocating compressor with minimum pulsation. That is why the acoustic layout of the pulsation dampers (without internal installations) was reviewed by own simulation calculations of the advising specialist firm. The calculation results showed that the pulsation dampers are a good match for compressor operation overall.

The review of the pulsation and vibration study conducted by the manufacturer was based on the submitted reports and documents. Single operation (M15) as well as parallel

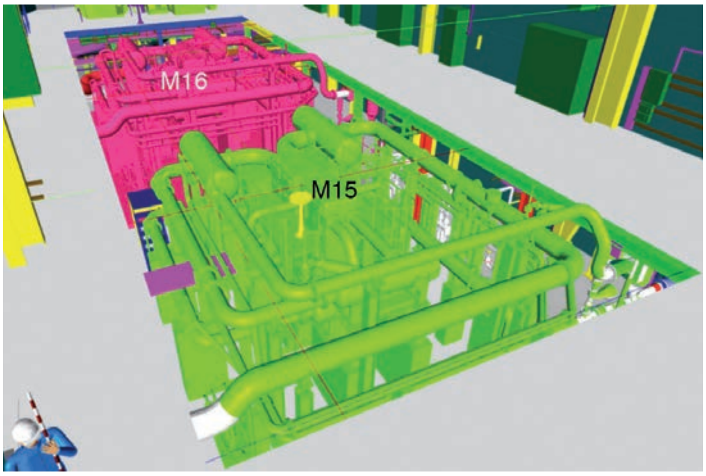

Fig. 7: Interior view of the hall (CAD model) from above onto the two new oxygen piston compressors M15 and M16 with the foundations in the basement, shown without the concrete walls (safety barriers) around the compressors.

operation of both compressors (M15 and M16) was examined in the pulsation calculations for the overall system.

The results that were presented showed that the permissible pulsation level could be significantly exceeded in part according to the original state of planning. Since this is due mainly to acoustic resonance, the use of orifices installed at the pulsation dampener connections was recommended. This is a common method that does not contradict a good acoustic layout of the pulsation dampers.

With the installation of the orifices, the pulsation level allowable according to API was exceeded only slightly and solely on the pressure side downstream of the second compressor M16. The permissibility of the resulting vibration level caused by the gas forces of the piping system was verified in the structural dynamics part of the study. This approach is permissible and compliant with the API 618 standard.

Overall the review and discussion of the results showed that no additional measures beyond the manufacturer's recommendations were required.

Review of the torsional analysis

The purpose of the drive train torsional analysis – consisting of these main components: electric motor, elastic coupling and piston compressor crankshaft – is

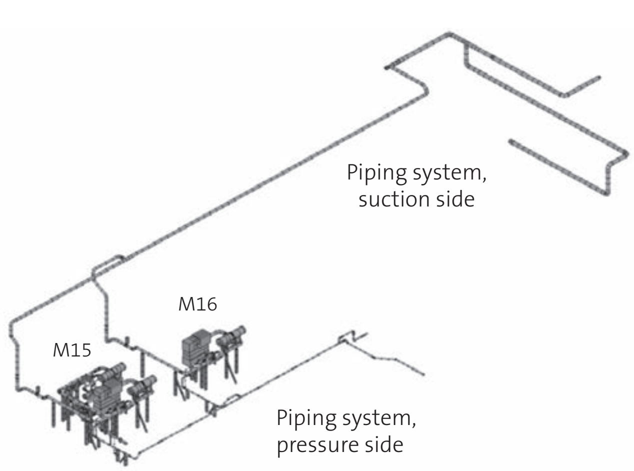

Fig. 8: Sketch of the piping system model space for the pulsation and vibration analysis (source: Burckhardt Compression AG).

to avoid excessive resonant rotational vibrations across the compressor's entire RPM range.

In an initial step, a simplified model of the highly elastic coupling as a single torsion spring was prepared by the compressor manufacturer – see Figure 9 above. However, the structural design of the coupling in detail consists of four individual elastic elements. That is why the consultant created a model for the verification of the torsional analysis (subtask B, see Section 7) that takes the segmented structure of the coupling into account – see Figure 9). This was used to verify the position of the calculated resonance frequencies.

As expected, the lowest resonance frequency corresponded to the mode shape (eigenmode), wherein the motor shaft on one side of the coupling and the compressor crankshaft (with flywheel) on the other side vibrate in phase opposition to each other – see Figure 10, no. 1. Only the coupling elements are deformed, the motor shaft and compressor shaft act as rigid bodies.

The following resonance frequencies and eigenmodes, no. 2, 3 and 4 (see Figure 10), correspond to the movement patterns where only the coupling elements are primarily involved. The shaft sections before and after the coupling exhibit next to no deformation. When the calculated resonance frequencies are examined in a resonance diagram, one can see that these coupling resonance frequencies can be initiated by certain (multiples) of the rotational speed (resonance cases), a. Fig. 11.

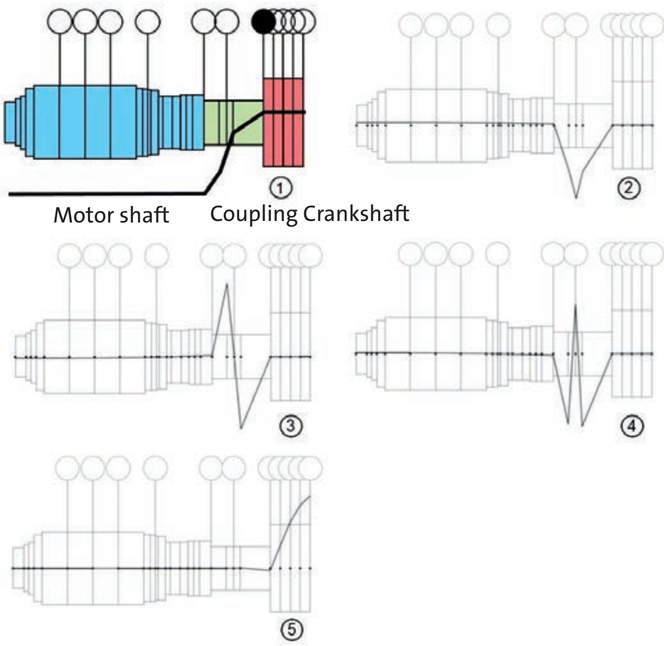

Fig. 10: Calculated torsion eigenmodes (heavy line) - no. 1 (3.1 Hz), no. 2 (23.3 Hz), no. 3 (50.8 Hz), no. 4 (59.0 Hz), bottom: no. 5 (97.5 Hz).

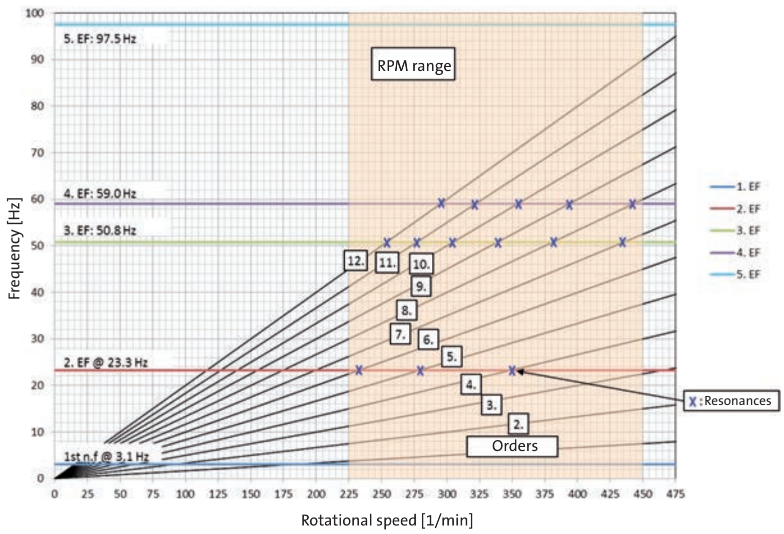

Fig. 11: Resonance diagram (Campbell diagram) for the torsional resonance frequencies of the oxygen compressor up to the 5th eigenmode (EF) and 12th order of the rotational speed, red field: operating speed range of the compressor.

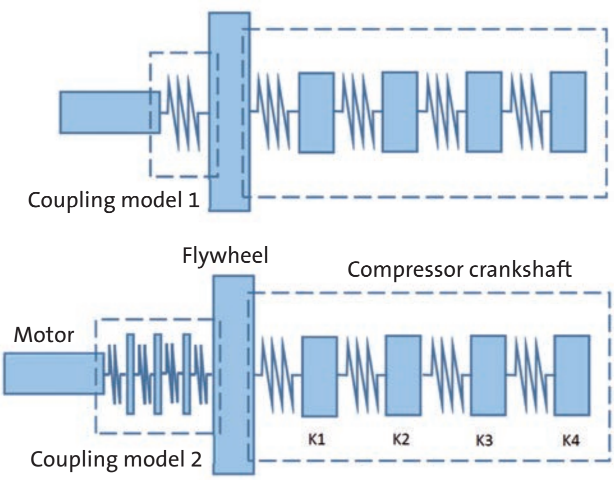

Fig. 9: Schematic layout of the drive train model for the torsional analysis. Top: simplified connection, bottom: extended coupling connection.

Therefore the manufacturer amended the model according to the segmented coupling structure for the mathematical examination of the torsion loads within the coupling. It turned out that the resulting torque values in the coupling are below the permissible values, so that no excessive torsional vibration loads are expected.

Review of the dynamic layout of the foundation

The main purpose of the dynamic foundation layout calculations is to avoid resonance cases for the relevant lowest resonance frequencies of the machine installation. As a rule, these are the possible 6 rigid body eigenmodes of the rigidly installed machine on the foundation block, which is founded on the elastic substrate – also see Figure 12.

The frequency ranges according to Table 2 for the exciting free mass forces are derived according to the RPM range of the oxygen compressors.

Rotational speed		Order		
			1.	2.
n_{min}	225 min^{-1}	f_{min}	3.8 Hz	7.5 Hz
n_{max}	450 min^{-1}	f_{max}	7.5 Hz	15.0 Hz

Table 2: Rotational speed and frequency range of excitation by mass forces of the reciprocating compressor.

Information from the machine manufacturer showed that vertical forces in both the 1st and 2nd order act on the foundation – see Table 3. However, only the 1st order frequency range, that is up to 7.5 Hz, was examined in the calculations being reviewed.

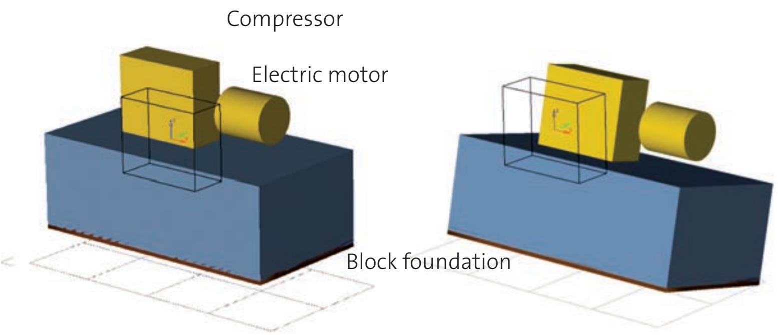

Fig. 12: Two chosen examples of the 6 possible basic eigenmodes of the machine installation (schematic) on the elastic floor. Lattice model of the compressor and the lower level of the block foundation: non deflected position, left: eigenmode with vertical deflection only (lift eigenmode), right: tilting around the longitudinal axis (rocking mode).

Mass forces and moments			
		1st order	2nd order
	[kN]	0	0
	[kN]	12.3	10.4
	[kNm]	75.6	0
	[kNm]	221	83.4

Table 3: Amplitudes of the free mass forces (Rf_{or}) and mass moments (R_{mom}, around the respective axis) during compressor operation according to information from the manufacturer.

The foundation was taken into account as a rectangular concrete block in the calculation model, and the masses of the compressor and electric motor were included as lumped masses. The equivalent stiffness of the floor was determined through the dynamic bedding modulus according to the soil expertise. A damping ratio of 15 % was applied for the operating vibration calculation, which can be considered conservative.

The stated goal of the dynamic foundation layout was to achieve a resonance frequency for the compressor installation (concrete block with machine units on elastic floor) according to DIN 4024 so that it lies 25 % above the relevant excitation frequency during machine operation.

The six calculated resonance frequencies were in the range of approximately 11.0 Hz to 20.0 Hz. The resonance frequencies and eigenmodes (mode shapes) are shown in Table 4 with the assignment of a possible excitation by the mass forces and moments of the 1st and 2nd order. The 4th resonance frequency for the vertical bounce at 16.2 Hz stands out. It is only 8 % above 15 Hz, that is the 2nd order of 450 1/min.

This means the requirement to adjust the resonance frequencies of the foundation to 25 % above the relevant excitation frequencies (above 1.25 x 15 Hz = 18.75 Hz) was initially not met. However, rough calculations performed by the consultant allowed the resonance frequency for the lift mode of the compressor foundation in the vicinity of 16 Hz to be simulated in principle.

Differences in the soil composition were noted in the course of construction work, that is after demolition of the foundations and base slab, so that the foundation concept had to be modified again due to statics considerations. A shaft foundation able to carry the loads over to solid rock at the lower end was now planned. This resulted in a corresponding soil stiffness, which should also have a more favourable effect on the dynamic properties. However, the results even with the stiffer soil showed that the critical eigenmode was still below the required resonance frequency of 18.75 Hz. Nevertheless the operating vibration analysis shows that the expected vibration amplitudes on the foundation were within the allowable range according to the manufacturer's information.

In a review of the more recent results, approximate calculations for the new specific soil values in the upper confidence range showed that the vertical resonance frequency of approximately 18.9 Hz can be reached.

Commissioning

"Breaking in" the labyrinth compressor and test operation with nitrogen

After successfully completing the loop checks, functional tests, leak tightness inspection and rotational direction test, the mechanical trial run was performed. Here the compressor was operated without suction and pressure valves.

Subsequently the individual pistons were broken in. Here the final process stage (crank 3) was first fit-

No.	Resonance frequency [Hz]			Mode	Resonance	Notes
	1st order	2nd order	>2nd order			
1	-	11.1	-	Thrust y	no	No excitation
2	-	11.6	-	Thrust x	no	No excitation
3	-	14.1	-	Rotation x	no	No excitation in the 2nd order
4	-	16.2	-	Lift	yes	Excitation by R_{for} vertical
5	-	-	18.6	Tilt x	no	No excitation
6	-	-	20	Tilt y	no	Excitation is 33 % above 15 Hz

Table 4: Assignment of the calculated resonance frequencies and eigenmodes for the machine installation to a possible excitation of the free forces and moments of the 1st and 2nd order according to Table 3.

ted with suction and pressure valves. The compressor was filled with nitrogen through the process lines and the stage temperature was regulated by the controlled closing of the bypass valve and by increasing the rotational speed. Here the piston was heated to approximately 200 °C, expanding the piston so the labyrinths of the respective cylinder were broken in. This procedure was repeated for all process stages and subsequently the machine was operating for a few hours with nitrogen at nominal pressure.

Commissioning with oxygen

After a further inspection and cleaning of the compressor, the machine was started with nitrogen. Oxygen from the process was gradually added through a manual flap in the suction line until an O_2 concentration of 99.5 % was reached. After reviewing all operating parameters and flushing various line sections, the machine was operated in feed mode.

Metrological check of the vibration and pulsation situation

After the M15 oxygen compressor was put into operation, a metrological check of the actual vibration and pulsation situation was performed in February of 2016. The pipework, cylinder and foundation vibrations as well as pressure pulsation in the pipework were simultaneously recorded at selected points for this purpose.

A comparatively low vibration level was recorded on the compressor plant. The vibration level on the pipework was significantly below the value specified by the manufacturer (30 mm/s RMS) and below the frequency-dependent orientation values of VDI Directive 3842. The cylinder and foundation vibrations were also far below zone limit A/B (suitable for long-term safe operation) according to DIN ISO 10816 – Part 8.

The measured pressure pulsations reached the still allowable level according to API standard 618 on only one measuring point, and were lower – for the most part significantly – at all other measuring points.

Based on the measurement results, it was determined that the new compressor plant as a whole is suitable for unrestricted long-term operation from a vibration engineering point of view.

Conclusion

Project risks in the implementation of what are known as “brownfield” projects can be largely minimised through a systematic approach, by putting together a good team, and by choosing competent service providers, especially for engineering. In this project the time and budget requirements were reduced thanks to good planning, the reduction of interfaces and a high level of decision-making authority when problems arose. The M15 oxygen piston compressor was put into operation on budget and 3 months before the planned date.

The risks related to the rotational speed control of a reciprocating compressor were reduced to a tolerable minimum through compliance with rules, standards and good engineering practices.

This is also reflected by the measurements taken after commissioning. Additional positive effects can be seen from the perspective of the operating company, especially in terms of efficiency and machine control.

Literature

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