Trouble-Shooting in a natural gas compressor plant

by:

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Reliability and economics of compression systems - recent trends in the market of reciprocating compressors
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Abstract:

Inadmissibly high vibrations had occurred in the main pipelines between the discharge-side pulsation dampers of a newly installed natural gas compressor. A duly conducted measurement analysis of the reason for these vibrations indicated that they were caused by strong pressure pulsations in the cylinder rooms and in the downstream connection pieces to the pulsation damper vessels. A direct link between these pressure pulsations and the mechanical vibrations was found in the region of the double-acting cylinders. The mechanical natural frequency of the connected pipe bends also added quite considerably to the amplitude of the vibrations. Two recommendations were adopted in parallel in order to reduce the vibrations. First pulsation damper plates were installed directly on the gas-outlet flange of the cylinders to reduce the pressure pulsations. The influence of the pulsation damper plates on the compression process in the cylinder room and the unsteady flow were modelled theoretically. Then the mechanical natural frequency of the pipeline was displaced upwards out of the critical frequency range. Proposed designs for a reinforcing system were examined and checked using the Finite Element method. Finally, the manufacturer carried out a measurement investigation confirming the effectiveness of the measures adopted and the safety of the plant in operation.

1 Introduction and problem definition

Germany operates over forty underground natural gas storages to cover demand peaks in the consumption of natural gas. Reciprocating compressors are often used to charge the gas into these storage sites, because they are highly efficient and can cater for the variety of different operating conditions encountered.

Despite having conducted acoustical and mechanical pulsation studies of the compressor and the connected pipelines, greatly increased vibrations occurred in the connection lines between the discharge-side pulsation damper vessels, when a two-stage natural gas reciprocating compressor having a total power output of 7 MW was put into operation. The cause of these vibrations could not be subsequently explained by the studies.

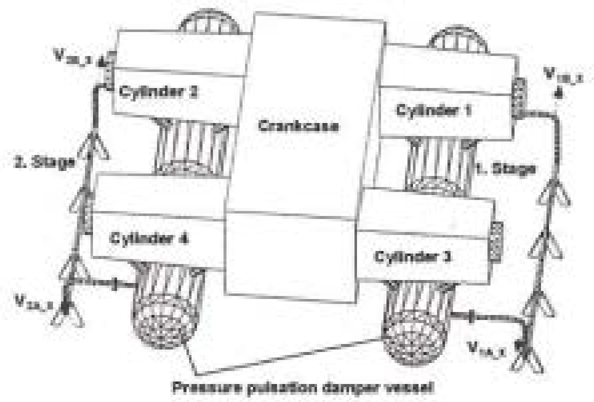

Fig. 1: Schematic diagram of reciprocating compressor, discharge pressure pulsation damper vessels, connection lines and the four points, at which vibrations were measured.

2 Measurement investigation

The first measurement of vibration rates made at the pipe bends in the modified version of the system shown in Fig. 1 exceeded the manufacturer's admissible standard levels several times over.

Therefore at the same time pressure pulsations and vibration rates at cylinder no. 4, at the pressure-pulsation damper vessel connected to it and in the pipe bends were recorded selectively for further analysis (Fig. 2).

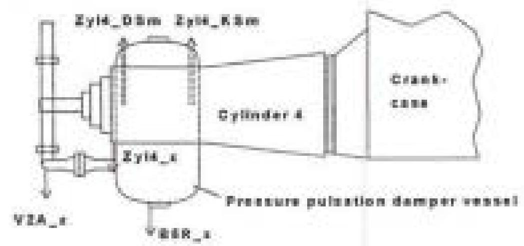

Fig. 2: Position of measurement points for detailed analysis of vibrations at cylinder 4.

Fig. 3 shows portions of the signals measured at cylinder 4 when running up and when operating the compressor plant. The cylinder room (Zyl4_KSm) at the crankshaft end was shut off by controlled valve-suppression, when this measurement was taken.

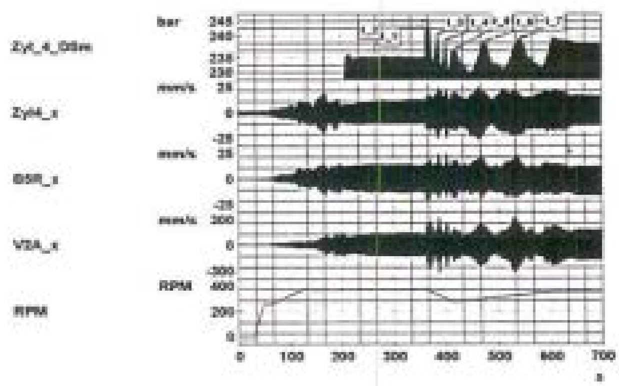

Fig. 3: Simultaneously recorded vibrations and pressure developments at cylinder 4.

The chart shows the development of cylinder room pressure on the cylinder end, vibration rates at different measurement points and rotational speed extending over a period of about 11 minutes from plant start-up. Especially at pipe bend V_{2A_X}, there is no recognisable increase in vibration at the measurement points, when the plant is running up ($t = 40 - 60$ s), so the likelihood of vibration being excited solely by the inertia force of the compressor can be ruled out. Once pressure has built up in the cylinder room ($t > 200$ s), however, continuously increased vibrations of about $V_{2A\_X} = 100$ mm/s become evident at the pipe bend measurement point. The compressor is run at its rotational speed in the period $t > 360$ s (other operating conditions remaining constant), this being accompanied by a marked instance of short-time increases (t_1 to t_7) in cylinder-space pressure and in vibration rate at the measurement points shown.

Colour charts of the amplitude spectra of the vibrations were produced to allow more precise examination of the relationship between the increased pressure pulsations and the vibrations. To do this, the recorded time-signal was divided into small, overlapping time sections (Δt = 1.6 s), the amplitude spectrum (FFT) for each section then being duly calculated and represented as colour chart (amplitude level identified by colour) along the time-axis. The colour chart in Fig. 4 clearly shows that, at cylinder 4, two separate amplitude increases (58 Hz; 63 Hz) occur at different times along the x-axis. Comparing the times of the maximum increases in cylinder room pressure in the function envelopes in Fig. 3, the 63 Hz amplitude increase is the only one occurring at the same time in each case (Fig. 4). While the order of magnitude of the vibration rates measured at the cylinder is not a cause for concern, those at the pipe bend (measurement point V_{2A_x}) are inadmissibly high, especially the 63 Hz recording (Fig. 5).

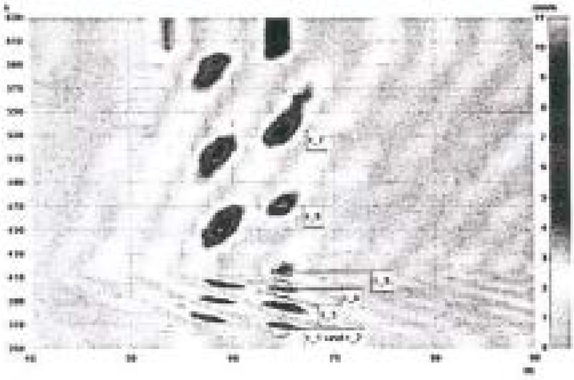

Fig. 4: Colour chart of amplitude spectra for measurement point Zyl4_x at the cylinder within the time range t = 350 to 630 s (time signal from Fig. 3).

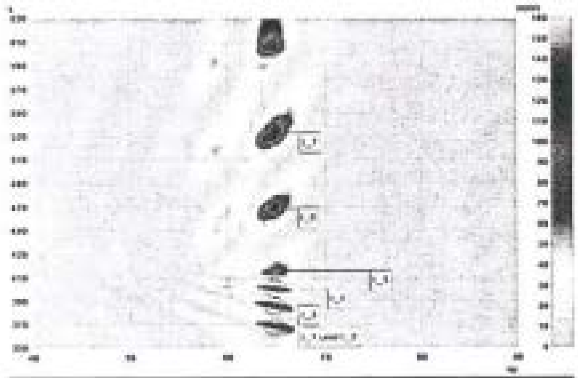

Fig. 5: Colour chart of amplitude spectra for measurement point V_{2A_x} at the pipe bend within the time range t = 350 – 630 s (time signal from Fig. 3).

There is a striking increase in vibration between cylinder and pipe bend (compare scale of Fig. 4 and Fig. 5).

To arrive at an explanation of this increase in amplitude, the natural frequency of the pipe bend was analysed with the compressor plant at a standstill (Fig. 6).

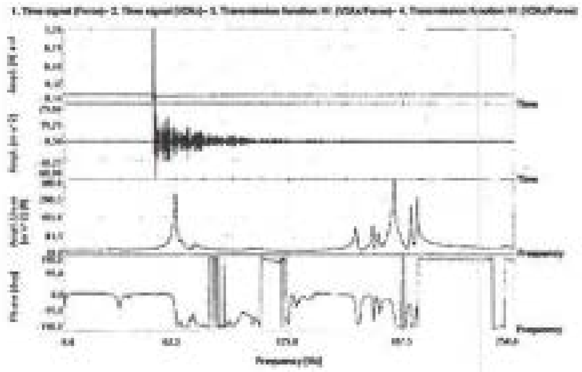

Fig. 6: Transfer function of acceleration signals of an force impulse excitation.

Excitation was produced by means of an impulse hammer at measurement point V_{2A_x}, at which the acceleration response was also recorded along the x-axis in each case. An amplitude increase occurs at about 62 Hz, accompanied by a 180° phase-shift between excitation and response signal. Accordingly, the increase in operational vibration between cylinder and pipe bend is caused by a mechanical natural frequency, thus disclosing the transmission path of vibrations emanating from the cylinder. To ascertain the cause of the cylinder vibrations, the measured development in cylinder room pressure from Fig. 3 at the maximum point (t_2 = 368.2 s) and at the succeeding minimum (t_2^* = 375.6 s) are plotted over time in Fig. 7.

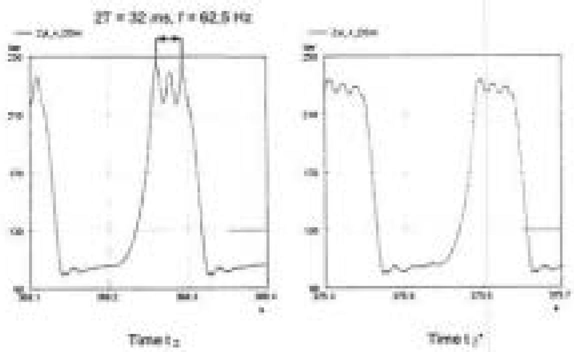

Fig. 7: Measured pressure pulsations at 330 rpm (t_2) and 300 rpm (t_2^*).

Pressure pulsations of up to 25 bar peak-to-peak (depending on rotational speed) occur at a frequency of f = 62.5 Hz in the process of exhausting the natural gas (Fig. 7). These pulsations are the actual cause of the increased vibrations, which are transmitted mechanically from the cylinder, through the pulsation damper vessel, to the pipeline. The pulsation is excited by an acoustic natural frequency in the intermediate pipe (the connection between cylinder and pressure pulsation damper vessel), when it coincides with higher-harmonic frequencies of the rotational speed. The same effect was also found to exist in the other cylinders of the compressor.

The acoustic natural frequency in the intermediate pipe can be subjected to a simplified theoretical examination in isolation. The acoustic marginal conditions (open - closed) produce an natural (resonant) frequency at ¼ the wavelength. When the temperature of the natural gas and the real gas factor are taken into account, the result is a calculated frequency of about 68 Hz, which correlates with the effects ascertained through measurement.

3 Recommendations

Reduction measures were necessary for two reasons. In the first instance, the measured pipeline vibrations of over 100 mm/s eff. caused additional dynamic stress capable of resulting in damage to the pipeline. Secondly, the pressure pulsation in the cylinder room resulted in not inconsiderable dynamic strain on the drive unit. These additional loads correspond to a weight of about 10 tonnes, applied to the piston rods at a frequency of 63 Hz.

Accordingly, a combined approach to an effective reduction of the vibrations was proposed and implemented to make sure the plant would operate safely and reliably. Pulsation damper plates were to be installed with the intention of dampening the exciting pressure pulsations, and at the same time, the natural frequency of the pipeline was to be moved out of the excitation range by providing a dynamically stable support. Operational requirements made it essential to rectify the critical situation in the course of a once-off modification.

3.1 Installation of patented KÖTTER pulsation damper plates

The crucial need was to introduce a means of reducing the pulsations inside the cylinders and in the connection pipes to the discharge-side pulsation dampers that, while producing the desired acoustic effect, would result in hardly any loss of pressure. The plant compression levels guaranteed by the manufacturer would otherwise cease to be maintainable.

The decision was therefore taken to replace the existing orifice plates fitted on the discharge-side cylinder flanges with patented "pulsation damper plates based on the KÖTTER principle". The unsteady, compressible, viscous compression and gas flow in the compression room and in the connection pipes was simulated numerically with a view to ensuring that the modifications would duly achieve their purpose. The simulation was based on the one-dimensional Navier-Stokes equation, the equation of continuity, the energy equation and the equation of state.

This system of partial non-linear differential equations can be converted by a transformation of coordinates to lines of indeterminate cross-derivation - so-called characteristics - to produce conventional differential equations.

The flow field is then numerically integrated along the characteristics by means of a space-time discretisation. Marginal conditions used in the simulation were the known speed of the piston and a constant pressure in the discharge-side pulsation damper vessel, in which (as substantiated by the measurements) only faint traces of interfering pulsations at a frequency of 63 Hz were still to be found. Allowance was also made for the opening and closing of pressure valves as a function of flow ratios.

To begin with, the original system including its orifice plates was modelled for the purpose of validation. Fig. 8 illustrates the measured indicator diagram together with the calculated, time depending pressure at the discharge-side cylinder flange.

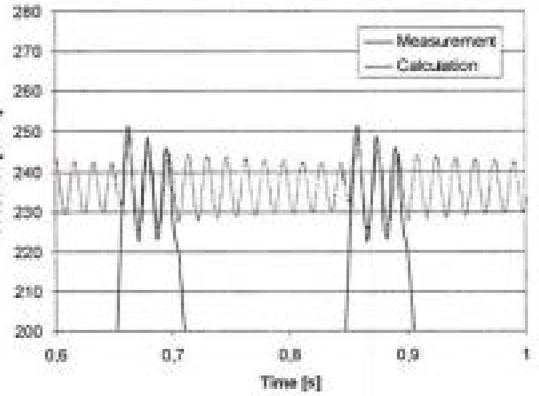

Fig. 8: Measured indicator diagram and calculated pressure inside the connecting pipe in original state.

It can clearly be seen that each time the pressure valve is opened (accompanied by the associated, abruptly beginning mass flow), a feebly dampened acoustic natural frequency is excited in the cylinder room and in the connection pipe. This acoustic resonance is responsible both for the increased strain on the drive unit and for the inadmissibly high pipe vibrations. The amplitude of the gas vibration is very decisively determined here by favourable or unfavourable coincidence between the opening of the valve and the phase angle of the acoustic resonance (see Fig. 7).

The pulsation damper plate (Fig. 9) was then incorporated into the design model as the second stage of the simulation.

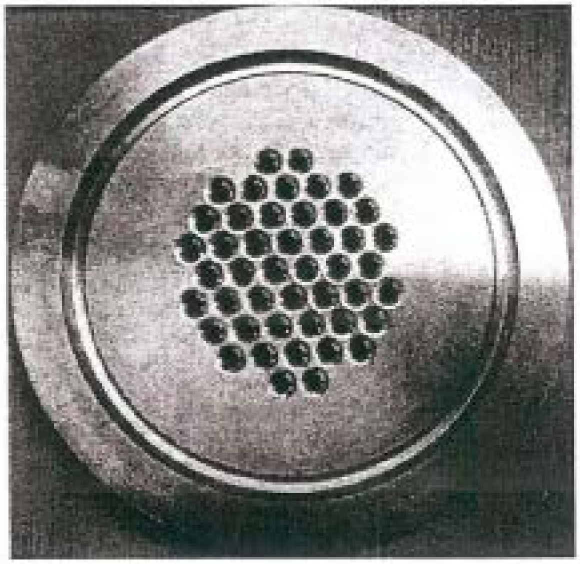

Fig. 9: Gas inlet side of patented "pulsation damper plate based on the KÖTTER principle" fitted to cylinder output flange of second stage.

While only one parameter can be varied in an orifice plate (namely the diameter of the plate), the pulsation damper plate offers many adjustment options. Apart from providing a choice of number of bore holes, hole pattern and bore-hole diameters, the venturi-like openings of this plate can be formed "at will", so the pulsation damper plate permits optimisation of acoustic or flow-conducting characteristics without additional loss of pressure.

Fig. 10 illustrates the outcome of this design process with the calculated time depending pressure at the discharge-side cylinder flange with pulsation damper plates installed. The measured indicator diagram after implementation of the modification is included in the figure for comparison purposes.

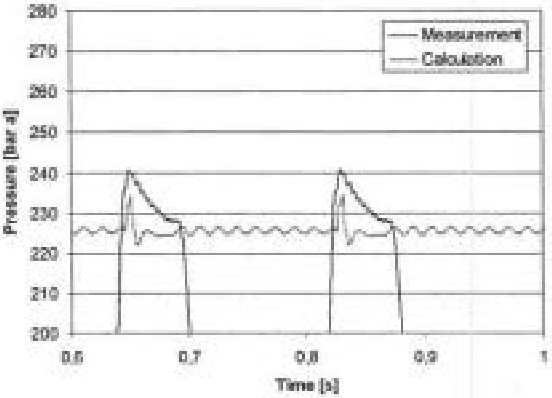

Fig. 10: Measured indicator diagram and calculated pressure inside the connecting pipe with pulsation damper plate installed.

The theoretically ascertained satisfactory effect of the pulsation damper plates is confirmed by subsequent measurement. There are no more significant gas vibrations to be found in the indicator diagram either. The above-mentioned additional strain placed on the drive unit by alternating pressure loads on the piston has been eliminated by this means, and the alternating pressure amplitudes in the connection pieces have also been distinctly reduced. However, it can still be seen that the measures adopted have only slightly increased the degree of acoustic damping provided, when the pressure valve is closed. The reason for this is the predetermined installation position of the pulsation damper plate.

The only usable flange was located directly at the cylinder, which means that the pulsation damper plate is sited at the acoustically closed end, when the pressure valve is closed.

3.2 Additional stiffness and damping in the region of the pipe bend

Furthermore, to make sure that the critical vibrations would no longer be excited by the pulsations still present in the connection pieces, the natural bending frequency of the pipe bends and pipe sections was displaced to a higher frequency (> 70 Hz) by means of an additional 'A' support, and broadband damping was incorporated. Because of the high temperature of the gas and the pipeline, a special means of connecting the support (Fig. 11) was chosen for the region of the pipe bend (at the positions of the measurement points in Fig. 1).

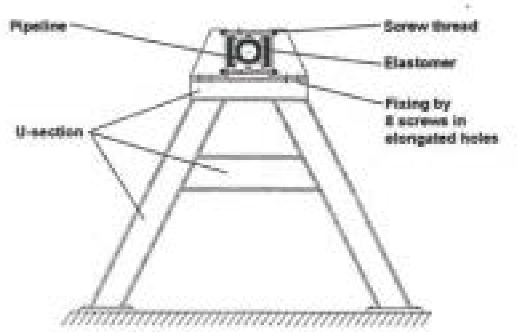

Fig. 11: Additional 'A' support as modification of pipeline section.

The elastomer coating shown in Fig. 11 is arranged without direct structure-borne noise bridges in such a way that the actual surface pressure of the elastomer (a decisive dynamic parameter) can be adjusted and optimised in situ by altering the tightening torque of the four bolts. Compared with conventional elastomer inserts, which are inserted directly between pipe and pipe clamp, the heat convection inherent in this design tolerates higher temperatures at the outside of the pipe.

Since the elastomer coating employed can only be used up to about 90°C, the design in question was essential in the present case to provide for additional dissipation of heat.

A Finite Element calculation was performed in advance to optimise the dynamic characteristics of the intended pipe connection. The Finite Element model (FE) of the original pipeline section was adjusted with reference to the measured characteristic shape, thus making it possible to allow for the influence of the actual marginal conditions of the pipeline in a simplified FE model (bar model).

Fig. 12 shows the calculated mode of the first bend in the pipeline with the optimised 'A' support and the elastomer connection. Then the natural frequency is displaced upwards by 30% as a result of this modification. The critical factor determining the position of the first natural bending frequency is the dynamic stiffness or rigidity of the elastomer.

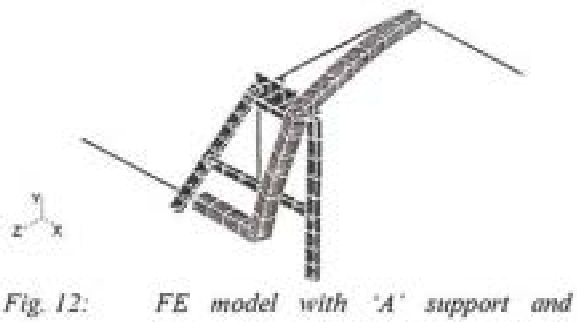

Fig. 12: FE model with 'A' support and elastomer, first natural bending frequency 79.4 Hz.

4 Checking the modifications

In conclusion, the work of modification was followed by an examination of the measures implemented. To do this, the manufacturer arranged a detailed measurement of vibrations within the whole system at different operating points of the compressor plant, so as to rule out the possibility of any localised displacement of the pipeline vibration problem to other points. At an effective value of 15.5 mm/s, the maximum vibration rate measured in the process was well below the admissible standard value of 28 mm/s eff. The measurements also failed to disclose any relocation of the vibration problem.

Renewed measurement of the natural frequency of the first characteristic bending form or shape indicated a frequency of 83 Hz in the pipeline with 'A' support. In the light of the simplified FE model and the normal deviation of elastomer characteristics from manufacturer's specifications, this indicates good compatibility and confirms the validity of the procedure described above. The results of an additionally conducted measurement of operating vibrations at measurement point V_{2Ax} in the modified system are shown in Fig. 13 for the purpose of comparison with the measured vibrations of the original system.

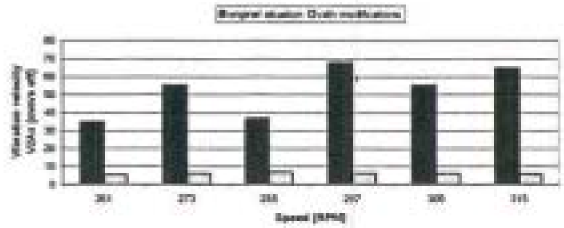

Fig. 13: Measured pipe bend vibrations in original situation and following modification.

The comparison of the vibration situation before and after modifications impressively confirms the effectiveness of the measures adopted. A final examination of power loss attributable to the pulsation damper plates also demonstrated the precision employed in designing for pressure loss at the compressor output.