Pulsation study for two 2,250 bar hyper compressors – measurement, theory, measures –

by:

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Abstract:

In a LDPE-plant two hyper compressors are operating in parallel. In order to improve the blend performance, it was planned to increase the static pressure on the discharge side of the compressors up to 2,400 bar. By means of suitable measures the release of the safety valves caused by pulsation related pressure peaks had to be prevented for the further operation. Additionally, a safe and reliable operation of the plant from vibration technical aspects had to be ensured. For processing a pulsation study in conjunction with pressure and pipe vibration measurements was conducted. It turned out that in order to avoid the release of the safety valves due to pulsations a synchronisation of both compressors at a fixed phase relationship is the favourable solution. Additionally, high pipe vibrations were reduced by installation of orifices and new pipe supports. Finally, after realisation of the measures the discharge pressure has been raised up to 2,400 bar without any pulsation or vibration problems.

1 Introduction

For the production of polyethylene (LDPE) two 2-stage, 4-crank, maximum pressure reciprocating compressors (hyper compressors) are applied, which produce in a present parallel operation a reactor mean pressure (discharge pressure 2nd stage) of 2,250 bar. In order to increase the blend performance, the reactor mean pressure of the plant has to be increased to 2,400 bar. The new static mean discharge pressure of the reciprocating compressors does not exceed the original design range of the plant. Therefore, no rebuilding measures are planned for the compressors and the piping system.

Based on previous examinations and the experience of the operator it is however known that maximum pressures in the piping system are temporary significantly above the static mean value. The pulsations excited by the reciprocating compressors are responsible for this. In case the temporary pressure peaks lie above the release pressure of the safety valves, a blow-off occurs and this leads to an immediate shut-down of the plant.

The first target of the in the following presented pulsation study therefore is to design measures, which reduce the maximum pressure peaks of the 2nd stage to a level possibly below 2,500 bar. Generally, two approaches are to be examined for this purpose. One possibility is the favourable synchronisation of both compressors (phase relationship). The other possibility is the installation of pulsation dampers (e.g. λ/4-resonators). The second target of the pulsation study is to ensure a safe and reliable operation of the plant from vibration technical aspects for the planned discharge pressure. In case of expected critical vibrations, effective measures have to be proposed.

2 General approach

The processing of the project is carried out in two phases.

In phase I the vibration technical actual-status-situation of the plant in the zone of the maximum pressure compressors is measurement-technically registered, proceeding from the suction side of the 1st stage to the interstage and then to the discharge side of the 2nd stage up to the beginning of the reactor. Here, the pressure pulsations of both compressors at various positions are recorded simultaneously together with the piping vibrations as well as the top dead center signals of the 8th cylinder of each compressor. The measurements are performed in the steady operation of the plant. In order to detect the influence, which the mean static pressure has on the pulsations and vibrations, the discharge pressure is reduced for a brief period by approximately 150 bar and then raised again. The measured data are analysed with regard to conspicuous vibration levels and compared to the valid guideline values.

In phase II the two compressors as well as the piping system between the inlet flanges of the pulsation dampers, 1st stage, and the reactors are digitised. Subsequently, the unsteady viscous compressible flow in these sections is numerically calculated. The results are compared to the measurements and the numerical system as well as the acoustical characteristics of the reactor is adapted accordingly.

Using the verified numerical model the optimal phase relationship between both compressors with reference to the pulsations is calculated for the increased discharge pressure of 2,400 bar. Furthermore, various pulsation dampers are simulated and their effects are compared. Based on the changes in the pulsations – and therefore in the exciting shaking forces – the changes in the piping vibrations are forecasted. In case of expected critical pipe vibrations, effective measures are designed. Finally, the recommended phase relationship between both compressors as well as the necessary further measures and procedures for the assurance of a safe and reliable operation of the plant are submitted to the operator.

3 Basics of assessment and guideline values

As initial orientation values for the evaluation of stationary flexural vibrations on piping the vibrations [1] as shown in figure 1 are adopted and these were also incorporated into the VDI guideline 3842 [2]. These are experience values, which were determined statistically over a period of over 25 years by means of measurements taken on pipework with geometries and holder spacings as normally applied in petrochemical engineering. They are not suitable for the assessment of shell vibrations, brief-period vibrations or vibrations on short piping attachments (e.g. nozzles).

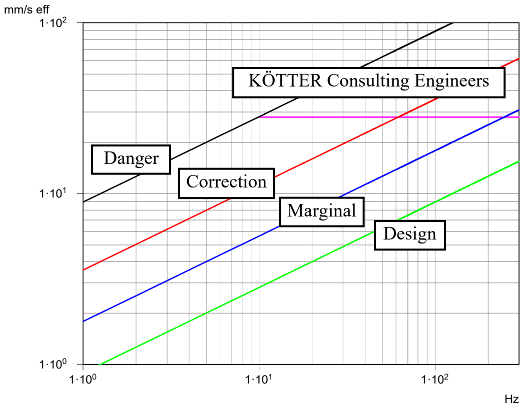

Figure 1: *Orientation values of allowable piping vibrations with a stationary flexural stress and strain.*

Determined vibration velocities above the line "danger" are normally regarded as being so dangerous that damage to the plant can occur. Additionally, the flexural vibrations of the piping above a frequency of 10 Hz, which exceed a value of 28 mm/s rms are subject to a closer analysis. This procedure is based on our own experience and the standard of Neuman & Esser [3], which envisages this limit value for flexural vibrations on piping and represents an extension of the former VDI guideline 2056 [4].

Generally, attention is drawn to the fact that the values stated are merely orientation values. The decisive factor for allowing vibrations is ultimately the material stress within the pipe wall, which is detected / recorded at critical locations by means of a mobile strain gauge measurement.

4 Basics of numerical calculations

4.1 Compressor data

Both hyper compressors are of the same type and manufactured by N.P.O. Frunze, Sumy, Ukraine in the year 1977.

Type:	4-crank, 2-stage ethylene hyper compressor
Cylinders:	8 single-acting (2/2 A-flow and 2/2 B-flow)
Speed:	250 1/min (fixed)
Suction pressure:	250 bar
Interstage pressure:	approx. 1,100 bar
Discharge pressure:	2,250 bar (condition during measurements)
	2,400 bar (aspired condition)
Driver:	synchronous electric motor
Drive power:	5 MW

4.2 Numerical flow simulation

The process consists of two separate flows, the A-flow and B-flow. Both flows are established at the suction side, 1st stage, of both compressors and are merged together inside the reactor. The numerical flow simulation is carried out separately for the A-flow and B-flow. For example the piping sections of the interstage of compressor A and the discharge side of the 2nd stage of both compressors (B-flow) are shown in the figures 2 and 3. The digitised piping section starts at the inlet flange of the pulsation damper, 1st stage, suction side and ends at the beginning of the reactor. The acoustical model includes the relevant cylinders and valves of the compressors. Because there will be no pressure change at the suction side, 1st stage, it is not necessary to extend the flow simulation at this side.

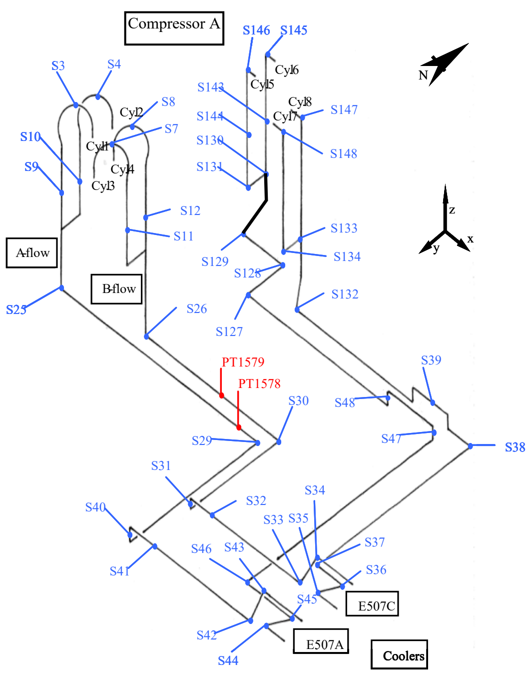

Figure 2*: Sketch of the interstage of compressor A (A- and B-flow) and the pipe vibration (e.g. S48) and pressure pulsation (e.g. PT1578) measuring points.*

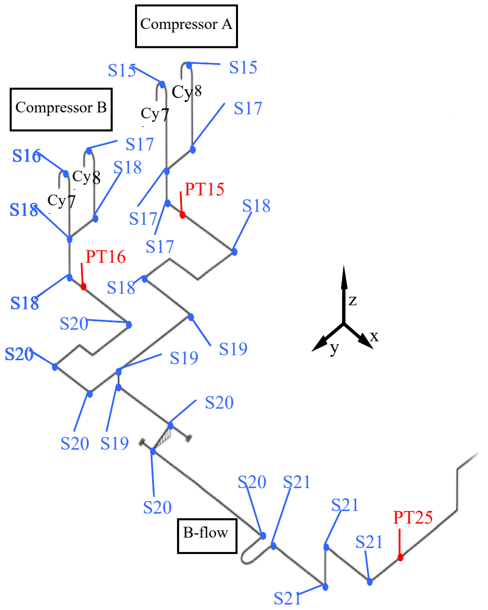

***Figure 3:** Sketch of the discharge piping, 2nd stage, (B-flow) of both compressors and the pipe vibrations (e.g. S170) and pressure pulsations (e.g. PT1604) measuring points.*

The physical boundary condition at the beginning of the digitised piping section is an anechoic end. At the end of the digitised piping section (reactor) the boundary condition is adapted to the measuring data. It turned out that an anechoic end is a good choice, too.

The calculation of the flows in the areas as mentioned is performed on the basis of one-dimensional, unsteady, viscous, compressible Navier-Stokes-equation, the continuity equation and the conservation of energy by means of the method of characteristics in the time domain. In this case, the plunger movement of each cylinder – and with this, the time-dependent volume in the cylinder area – is given full consideration by means of an adaptive network. The opening and closing of the suction and discharge valves is effected in accordance with the actual differential pressures over the individual valve. The state variables of the fluid in the cylinder area and the overall piping system are continually calculated using the dependencies ρ (p,T), c_p (p,T), c_v (p,T) [5] according to the actual pressures p and temperatures T. In this case, the real gas behaviour based on the equations of state mentioned above as well as the wall influence of the piping on the sound velocity of the fluid are taken into consideration.

As the result for any point of the truncated system, the calculation provides the time-dependent shape of the pressure, the density and the flow velocity. Based on this information the shaking forces acting on the piping induced by the pulsations are calculated.

4.3 Numerical mechanical vibration simulation

Due to the unchanged exciting frequencies and unchanged piping resonance frequencies a numerical calculation of the mechanical pipe vibrations is not required for the entire pipe system. Whereas the changing shaking forces due to the changing pulsations caused by the increased static mean pressure are important in terms of the expected pipe vibrations.

In case of required modifications to the pipe support in areas of conspicuous structure vibrations the piping system with and without modifications is modelled using the program ANSYS.

5 Measured data and assessment (original situation)

From the signals of the proximity sensors for the top dead centers the phase relationship of $\varphi = 285°$ results between the compressors A and B during the entire examination of the pipe vibrations. Accordingly, cylinder no. 8 of compressor A is 285° (190 ms) earlier in the top dead center than cylinder no. 8 of compressor B. The cylinder sequence with reference to the top dead center is in timing sequence:

cyl. 1 + 6 – cyl. 4 + 7 – cyl. 2 + 5 – cyl. 3 + 8,

where the phase offset is 90° in each case. Therefore, even with single acting cylinders the main pulsation frequency (8.3 Hz) is twice the rotational speed of the compressor (250 1/min).

5.1 Mechanical vibrations

The vibration measurements have been carried out at more than 300 measuring points at the piping and the compressors. At each measuring point the vibrations have been measured in three directions:

x… shaft direction of the compressor
y… horizontal
z… vertical.

It turned out that there are high vibration levels even in the unchanged system, which will be discussed in the following. For this purpose - as an extract out of the total system – the measured vibrations at the piping of the interstage of the compressor A (A- and B-flow) and at the discharge side, 2nd stage, of both compressors (B-flow) are shown in figures 4 and 5. The corresponding measuring points are described in figures 2 and 3.

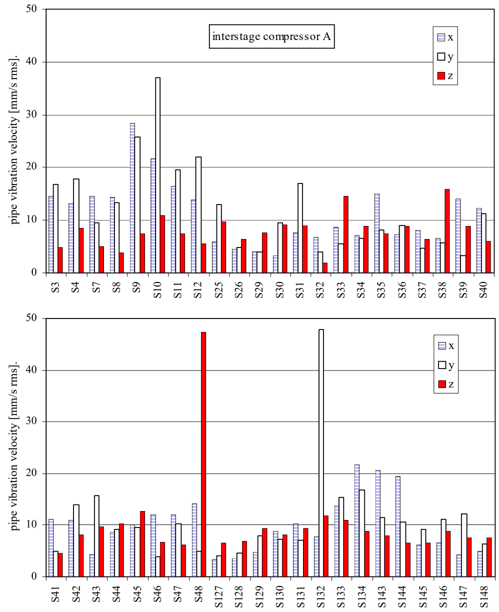

Figure 4: *RMS vibration velocities (2 - 1,000 Hz) of the piping on the interstage of the compressor A.*

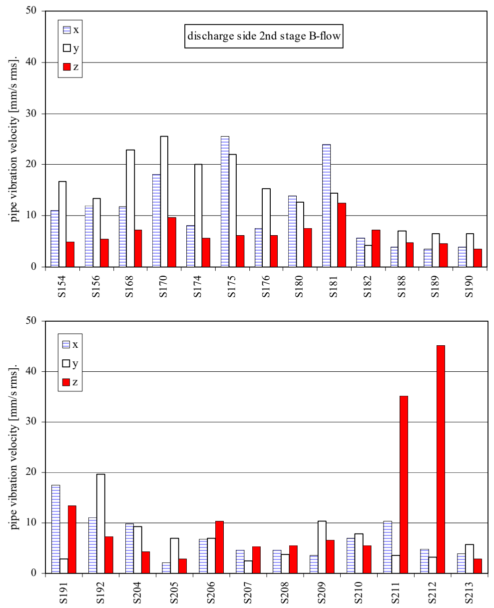

Figure 5: *RMS vibration velocities (2 - 1,000 Hz) of the piping of the B-flow on the discharge side, 2nd stage.*

The measured vibration frequencies are generally above 20 Hz and therefore the orientation value of allowable pipe vibrations is 28 mm/s rms. The measured vibrations are in the range or above this value in several areas. High vibrations at the piping of figures 2 and 3 are measured:

- in the vicinity of the cylinders (e.g. S9, S10, S132, S170, S175, S181)
- at the A-flow of the interstage (e.g. S48)
- at the B-flow, 2nd stage, discharge side (e.g. S211, S212).

These areas are selected from the total system because of their typical excitation and amplification mechanisms subsequently discussed.

5.2 Pressure pulsations

The pressure pulsations have been measured synchronously at several points using the installed absolute pressure sensors of the process control system. Together with the installed carrier frequency amplifier these transducers are able to measure pulsations up to a frequency of about 200 Hz.

As an example the measured pulsations at the interstage of the compressor A and at the B-flow, 2nd stage, discharge side are shown in figures 6 and 7. It is obvious that the dominant pulsation frequency is twice of the rotational speed of the compressor.

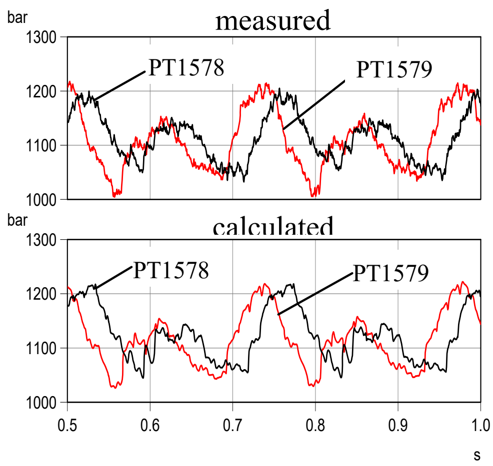

Figure 6: *Measured and calculated pressure pulsations at the interstage of compressor A (original situation).*

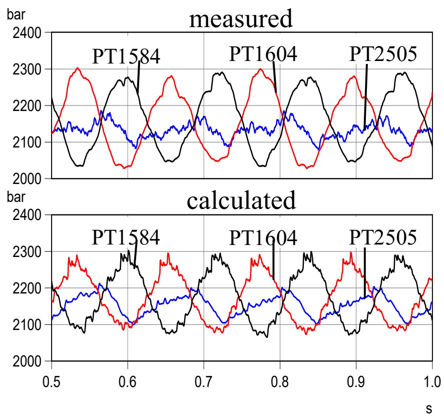

Figure 7: *Measured and calculated pressure pulsations at the discharge side of the 2nd stage B-flow (original situation).*

6 Prognosis and recommended remedial actions

6.1 Comparison between measured and calculated pulsations

Besides the measured the numerically calculated pressure pulsations are shown in figures 6 and 7. Generally, there is a good agreement between the measurement and the numerical simulation. The numerical model therefore can be regarded as being verified. In the following it will be used for:

- understanding the cause-and-effect chain for conspicuous pipe vibrations
- calculation of the optimal phase relationship between both compressors and the design of required pulsation dampers as necessary
- calculation of the pulsation induced shaking forces acting on the piping

for the planned operation condition (increased mean static pressure).

6.2 Variation of the phase relationship between both compressors

In the present case there is the possibility of reducing pressure pulsations at the 2nd stage, discharge side, by optimising the phase relationship φ between both compressors. The mean static pressure in the zone of the compressors for the following calculations is assumed to be 2,400 bar for the A-flow and 2,330 bar for the B-flow (future situation).

For the analysis the calculated amplitudes of the pressure pulsations at 8.3 Hz (twice the rotational speed of the compressors) are plotted versus the phase angle φ (figure 8). It is evident that the level of the pulsations depends on the phase relationship φ. The curves show a periodic behaviour with reference to a phase shifting of 180° (e.g. φ = 15° corresponds to φ = 195°). The physical reason for this lies in the immediate join-up of the gas flows of the two cylinders of one compressor (e.g. cylinders 5 and 6) downstream the discharge valves. Concerning the pulsations it is irrelevant, whether for example the cylinder 5 of the compressor A runs with a phase shift of φ to the cylinder 5 of compressor B, or with the same phase shifting to cylinder 6 of the compressor B.

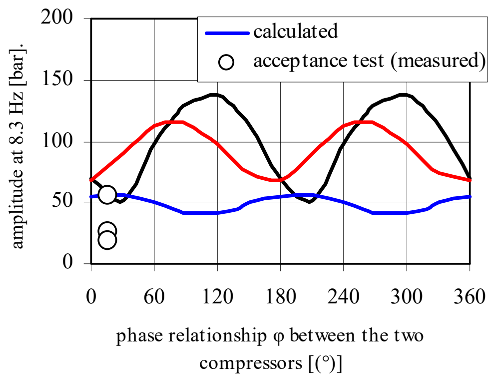

Figure 8: *Pressure pulsation amplitudes of the B-flow for the frequency of 8.3 Hz in the planned situation with increased mean static discharge pressure versus the phase relationship φ between the two compressors.*

It can be figured out that the phase relationship φ = 15° is favourable, because in this situation the B-flow has relatively low pulsation amplitudes (A-flow too, but not presented). In order to verify that higher pressure fluctuations do not locally occur in the piping system, additionally the calculated pressure pulsations for a large number of points along the pipe axis have been checked. It appeared that at a mean static discharge pressure of 2,400 bar and a fixed phase relationship of φ = 15° (between both compressors) the maximum temporary pressure peaks in the A-flow and B-flow at the discharge side, 2nd stage, are below the acceptable maximum pressure of 2,500 bar.

Instead of the described synchronisation of the two compressors (fixed phase shift) the pressure fluctuations in this case can also be reduced effectively with branch-off resonators (e.g. λ/4-resonator, Helmholtz-resonator). These resonators are piping pieces, which are closed at one end. The open side is coupled to the line containing the flow by a T-piece. Using this damping devices the numerical flow simulation shows residual pressure pulsations downstream the resonators – independently of the phase relationship φ – below 50 bar (0-peak). The effect of the resonators is therefore comparable to the synchronisation of the two compressors so that it was possible to make a selection from the view of operational and / or commercial aspects.

Finally, it was decided to realise the synchronisation of both compressors with the fixed phase relationship of φ = 15°.

6.3 Reduction of conspicuous pipe vibrations

In order to reduce conspicuous pipe vibrations, it is important to understand the cause-and-effect chain of the vibrations. In the present case it turned out that there are two different amplification mechanisms. In the vicinity of the cylinders (e.g. S9, S10, S132, S170, S175, S181) the primary cause for high vibrations lies in a strong flow resonance between neighbouring cylinders (e.g. cyl. 1 + 2) and their junction. At locations more away from the cylinders (e.g. S48, S211, S212) the exciting pulsations in the frequency range above 20 Hz are pretty low. In this region it is more common that structural resonances are the main cause for high vibration.

In figure 9 is - as an example for pulsation induced pipe vibrations - the situation at the measuring point S10 (interstage) shown. The measured dominant vibration frequency is about 33 Hz. The calculated pressure pulsations at S10 - original situation - show this dominating frequency, too. Based on the knowledge of the physical dependencies the best opportunity to reduce the exciting pulsations and therefore the high pipe vibrations in this case is to install orifices at the junction of the separate cylinder flows. By this solution the flow resonance between neighbouring cylinders is almost vanished (figure 9).

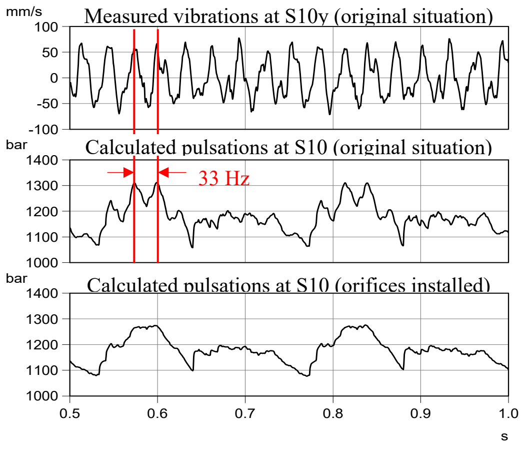

Figure 9: *Measured pipe vibrations at the measuring point S10 (top) and calculated pressure pulsations (centre) in the original situation as well as calculated pressure pulsations (bottom) with installed orifices.*

In addition to a vibration reduction it is to note in particular that the orifices have a protective effect on the compressor valves as well as on the dynamic engine loads, because they also reduce the pressure fluctuations in the cylinder area and the compression zone. Furthermore, due to their installation positions these orifices are not comparable with orifices that are located in a uniform flow. In the case dealt with here, the orifices primarily lead to a reduction of the pulsations and not to a significant change of the static pressure upstream and downstream of the orifice. In this way for example the temporary maximum absolute pressure is reduced by the proposed orifices – also between cylinder and orifice – and is not raised – like in a steady flow (figure 9).

Concerning pipe vibrations due to mechanical resonances the piping section at measuring point S212 is analysed in the following. Based on a frequency response analysis the natural frequency and the mode shape of the piping at the area of this point (discharge side, 2nd stage) is calculated (figure 10). It turned out that the calculated natural frequency (36 Hz) and the corresponding mode shape correlates with the measured vibration frequency (33 Hz) and mode shape (dominant vibrations in the vertical direction). The best solution to avoid the high vibrations therefore is to install an additional support, which is stiff especially in the vertical direction. In order to avoid new resonance vibrations, the used piping clamp should contain damping material.

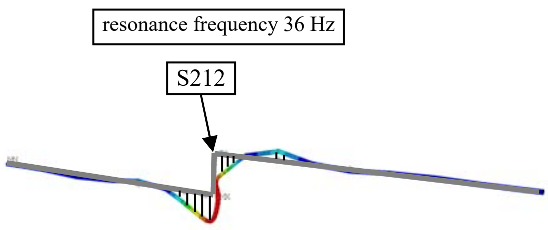

Figure 10: *Calculated mode shape of the piping at the resonance frequency of 36 Hz in the area of measuring point S212.*

Altogether, with the objective to reduce and avoid conspicuous pipe vibrations at each cylinder at the suction side and at the discharge side an orifice has been installed. The additional power consumption because of these 16 orifices at each compressor is about 1.4 % of the drive power (5 MW). At locations of mechanical resonance vibrations additional supports have been installed.

7 Results of acceptance test

After realisation of the recommended measures

- fixed phase relationship between both compressors (φ = 15°)
- installation of 16 orifices at each compressor
- installation of additional pipe supports at locations of structural resonance vibrations

the residual pressure pulsations and pipe vibrations have been checked. The measured amplitudes of the pressure pulsations are shown in figure 8. The pipe vibrations are presented in figure 11. It appears that all values are in an acceptable range.

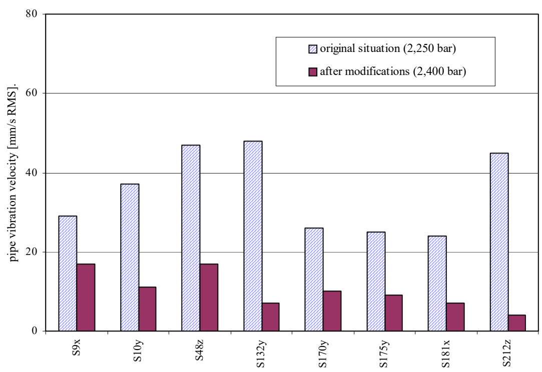

***Figure 11** Measured pipe vibrations in the original situation and after realisation of measures for the increased discharge pressure of 2,400 bar.*

Altogether, the whole project of increasing the discharge pressure up to 2,400 bar could be realised without a release of the safety valves or vibration problems. Additionally, the operating company observed an improved behaviour of the compressors concerning the required maintenance, which might be an outcome of the reduced pressure pulsations in the area of the compressors. Ultimately, this presumption must still be confirmed, however, by time.

8 Conclusion

In a LDPE-plant two hyper compressors are operating in parallel. In order to improve the blend performance, it was planned to increase the static pressure on the discharge side from a level of 2,250 bar up to 2,400 bar. By means of suitable measures the release of the safety valves caused by pulsation related pressure peaks had to be prevented for the future operation. Additionally, a safe and reliable operation of the plant from the view of vibration technical aspects had to be ensured.

For processing a pulsation study was established with a numerical model of unsteady flows of both compressors. The model was verified using measured pressure signals. Subsequently, the optimal phase relationship between the running compressors with regard to the pressure pulsations was determined. It turned out that the residual pressure pulsations are below the acceptable limit of 2,500 bar in case of a fixed synchronisation of both compressors to a phase relationship of φ = 15°.

The causes for conspicuous pipe vibrations were analysed based on flow and structural simulations as well as measurements. Close to the compressor high pipe vibrations were caused by acoustical resonances. In contrast to that primary mechanical natural frequencies were responsible for conspicuous vibrations of the piping system outside of the compressor environment. In order to avoid high pipe vibrations, the flow resonances were attenuated through orifices and the mechanical resonance vibrations were reduced through additional pipe supports.

After realisation of the recommended measures the mean static pressure on the discharge side of both compressors has been increased up to 2,400 bar without any pulsation or vibration problems as confirmed by an acceptance test.

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