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Root cause analysis and solution of recurring fatigue failures in the cooler pipe bends of a high pressure LDPE compressor

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ABSTRACT
Machinery vibration is one of the main causes of damage and failures in industrial plants, and reciprocating compressors are particularly subject to vibration-related phenomena as alternating forces and pressure pulsations are inherent in their design.

Vibration problems may also arise after a machine is installed and is set in operation due to inadequate attention to interaction between the compressor and plant as a whole during the design stage.

This paper will examine a case that occurred in a polyethylene plant located in Ragusa, Italy, where high vibration issues resulted in broken pipes and welding failures. In particular, the area that was most affected by vibration was the inter-stage cooler and the piping in the immediate vicinity.

New, more effective supports were studied and the exact positioning of the supports was based on the results of a dynamic forced-response FEM analysis; the vibration measurements were also used to tune the FEM model, and subsequently to identify, through the model, the key points where a stiffer structure was required. As a result, many of the existing supports were reinforced and additional supports were positioned where they would be more effective according to the FEM analysis. High damping rubber pads were also added between the supports and the piping in order to further lessen the impact of the vibration. Finally, also the cooler structure and foundations were reinforced.
1 INTRODUCTION
The production of Low Density Polyethylene requires compressors capable of bringing the gas, ethylene or an ethylene-based mixture, to the polymerization pressure, ranging from 160 to 350 MPa depending on the type of process utilized [1]. Due to the very high pressure, safety and plant operation are key factors. [2]

The compression of the gas is normally made in two steps:

- the first step, up to about 30 MPa, is normally performed with a heavy-duty balanced-opposed reciprocating compressor, called Primary Compressor;
- the second step, to reach polymerization pressure, is fulfilled with very special compressors, still reciprocating type but with many high-technology features to withstand the extreme working conditions, widely called Hyper Compressors or Secondary Compressors.

The reactors where the polymerization occurs are normally of the tubular type for the highest pressure processes and of the autoclave vessel type for the lower pressure ones.

Due to the fact that the piping, valves and other devices of LDPE plants have to operate under very high and variable pressures an accurate acoustic calculation of the pressure pulsations is necessary to avoid problems in operation also in consideration that, volume bottles cannot be used in this environment. The basic law governing pressure pulsation is the following:

$$\frac{\Delta P}{P} = \Delta Q \frac{K}{S} \frac{c}{c}$$

where $\Delta P$ is the pressure pulsation, $P$ is the average pressure, $\Delta Q$ is the flow pulsation, $K$ is the ratio of the specific heats, $S$ is the pipe section and $c$ is the speed of sound.

In the case of Hyper compressors there are very special conditions of the parameters influencing pressure pulsations: the speed of sound is very high (for example in second stage discharge it is about five times that of air at atmospheric pressure) but also the K is very high so that pressure pulsations are largely dependent on the pipe diameters. However a deep knowledge of the thermodynamics of the gas is essential.

As it is known, pipes can originate acoustic resonance in the following conditions:

- **Quarter-Wave ($\lambda/4$)** when the acoustic circuit is closed at one end and open at the other, with resonant frequency $F_R = [2N-1] \frac{c}{4L}$
- **Half-Wave ($\lambda/2$)** in two configurations when the acoustic circuit is closed or open at both ends, with resonant frequency $F_R = N \frac{c}{2L}$

Where: $\lambda$ = Wave length, $N$ = Harmonic number, $L$ = Resonant pipe length, $c$ = speed of sound

In both cases, high harmonic numbers are expected because of the high speed of sound of ethylene at high pressures.

Pressure and flow pulsation must be kept at low level as they are linked with acoustic energy which interacts with the mechanical system of the plant generating mechanical vibrations which require special attention.[3]
The dynamic forces that are the source of the vibration phenomena are due to:

- unbalanced inertia forces and moments of the compressor;
- forces induced on the gas pipes by displacement of the cylinders;
- pressure pulsations;

High vibrations generated by acoustic resonance conditions which, as stated above, can be of rather high frequency. Moreover, if this frequency is close to the mechanical natural frequency of the piping, vibrations can result extremely high and cause fatigue failure phenomena.

In order not to fall into this kind of problems an accurate acoustic analysis followed by a mechanical analysis is necessary taking care of a correct simulation of both the thermodynamic of the gas and the piping arrangement with the relevant supporting system.

The paper will discuss a problem that occurred at a polyethylene plant, where there were high vibration issues which resulted in piping failure recurring almost every 3-4 months over some years. High vibration velocity occurred on the interstage lines and on the intercooler structure (as shown on Tables 1 and 2).

During the survey on site, the piping system was analysed, data collected and the history of failures reconstructed to overcome the encountered problems.

2 \textbf{ISSUES EXPERIENCED ON A HYPERCOMPRESSOR}

The compressor was an old machine, installed in the seventies. About ten years ago it was revamped, and the frame and the driver were replaced, increasing the average piston speed. After these modifications, the customer started to experience frequent piping issues, generally in the form of cracks on the flanges or pipe segments in the intercooler area.

Despite the installation of restriction orifices following an acoustical study in the new conditions, the problems persisted. At that time, it was also suggested that some supports be modified but this was only partially implemented. The compressor is equipped with four 1st stage and four 2nd stage cylinders, and the intercooler has four sections.

The historical failures of the last 8 years were studied and it was found that not all the sections experienced the same behaviour: the area where the failures were most frequent was line B, especially on the pipes at the cooler exit. Subsequently, also lines A and D experienced a comparable number of failures while line C was the most reliable.

A detailed vibration measurement campaign had revealed 17 points with high vibrations on the interstage piping, with displacement values exceeding both the EFRC vibration amplitude limits and the South West Research Institute danger curve (in the frequency spectrum), above which corrections are urgently needed as failures are likely to happen even in a short time. Figure 1 and Table 1 shows some of these points with high vibrations.
Figure 1: Interstage piping layout and Points with high vibrations on 2nd stage suction lines D and B

<table>
<thead>
<tr>
<th>Direction</th>
<th>Overall displacement [µm RMS]</th>
<th>Overall velocity [mm/s RMS]</th>
<th>Frequency of the max Peak [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>P002T</td>
<td>X 185</td>
<td>41</td>
<td>57.3</td>
</tr>
<tr>
<td>P004T</td>
<td>X 76</td>
<td>38</td>
<td>78.8</td>
</tr>
<tr>
<td>P007T</td>
<td>Y 177</td>
<td>55</td>
<td>49.8</td>
</tr>
<tr>
<td>P009T</td>
<td>Y 95</td>
<td>95</td>
<td>41.5</td>
</tr>
<tr>
<td>P011T</td>
<td>Z 116</td>
<td>56</td>
<td>62.3</td>
</tr>
<tr>
<td>P013T</td>
<td>Y 116</td>
<td>56</td>
<td>70.5</td>
</tr>
</tbody>
</table>

Table 1: Points with high vibrations on 2nd stage suction pipelines D and B

In addition, also the cooler was analysed and the vibration amplitudes were found to be in some cases even 10 times over the danger limit. To give an idea of the magnitude of those vibrations, some of the horizontal pipes of the cooler vibrated at more than 300 mm/s rms while the EFRC unacceptable limit (keyzone C/D) is 28.5 mm/s!

Despite the technical measures that were undertaken over the last few years to avoid high levels of vibration (i.e. acoustical and mechanical analysis, restriction orifices and new piping supports) vibration continued to be a problem. In this case, the vibrations were not the result of a poor detailed design of the compressor piping system but were due to the fact that acoustical interaction between the compressor and the plant had not been taken into account with all due attention.

In particular, some of the circumstances that led to this situation were:

- the modifications to the piping supports suggested by the compressor manufacturer were assigned by the end user to another contractor and resulted in a poor and weak design (see Figure 2);

- the coolers structure was not originally supplied by the compressor manufacturer, who therefore did not have the fabrication drawings and related engineering details. When the new acoustical and mechanical analyses were performed, only the acoustical aspect was considered while the cooler was assumed to be a rigid body; however, it was in very bad condition due to rust, repeated repairs and invasive maintenance activity (for example, some beams had been cut several times and re-welded to change the pipes). See Figure 3;

- the vibrations on the cooler foundations were also high above cooler B, and a borescope inspection revealed underground voids due probably to subterranean water erosion.
(cooling pipe joints were continuously leaking a large amount of water). As a result, the foundations, where the shaking forces are transferred in order to limit the vibrations, were not acting properly.

Figure 2: Examples of the ineffective supports

![Figure 2: Examples of the ineffective supports](image)

Figure 3: Cooler structure

The two main areas with high vibrations were identified and a different approach was used for each one, due to the dissimilar nature of the vibration.

In the interstage piping area, between the compressor and the cooler, like the area shown in Figure 2, most of the highest vibration harmonics (peaks) were between 60 and 70 Hz, therefore at a very high frequency, considering that the first exciting harmonic was at just 4.1 Hz. The pulsation amplitudes at this high frequency were very low, especially after introduction of the restriction orifices, therefore a simplified analysis of the piping segment mechanical frequency was performed, according to API 618.

The result was that most of the pipe segments with a length between 1.5 and 1.7m had a mechanical natural frequency in the range of 60-70Hz. In particular, this was due to the fact that these supports should have acted as fixed constrains, whereas in the actual situation, they were simply mono-directional supports.

In the intercoolers section, instead the main criticalities were that:

- the bends shown in Figure 3 had a small bend angle, thus the exiting supporting beams were not suitable to sustain the resulting shaking forces;
the inlet and outlet connections were considered in the mechanical analysis as fixed supports but were instead just fixed with wooden clamps (see Figure 2), so they had a very low stiffness;

- the side pipes were more than 7 m long and were just clamped in the middle with simple U bolts;

- on cooler B also the baseplate vibrations were above the limit, due to voids in the foundation.

3 PROBLEM SOLVING: MODIFICATIONS INTRODUCED

In consideration of the short time available for the intervention (10-day outage), a simplified support design (Figure 4), was studied to minimize the time required for order processing and delivery of materials:

- a uniform clamp design was developed, using a variable support plate to adapt it to each pipe height

- a layer of fabric reinforced, elastomeric pad, 4mm thick, was placed between the clamp and the gas pipe (rolled all-round the pipe).

- the clamp was made from a bent plate and the anchoring was made using U-bolts welded around the outer diameter.

![Figure 4: Typical support design](image)

This type of clamp was adopted to substitute all the weak supports of the interstage piping, together with a reinforcement of the beams with the addition of stiffening plates.

A total of 83 weak supports were selected for substitution and for each one a criticality value was given to indicate the priority to be given to the substitution, considering that only a certain number of supports could be replaced within the scheduled outage.

In order to identify the most efficient and effective upgrades to significantly reduce the vibrations on the cooler structure, it was decided to perform a dynamic FEM (Vibration Analysis). Not having the construction drawings, the cooler structure (both front, bay and cylinder rear side) was
modelled based on dimensional measurements carried out in the field. Once modelled, the analysis of linear dynamic vibration was set, paying particular attention to extrapolation of the forcing factors to be imposed on the existing structure.

Using the results obtained from the acoustic calculation, the pressure pulsation values related to each loop were extracted. After that, the shaking forces were calculated using the API RP 688 [4] formula:

\[ F_{Result} = 2P_{dyn} \frac{\pi}{4} D^2 \cos \left( \frac{\varphi}{2} \right) \]

Where:

- **F** is the shaking force
- **P_{dyn}** is the dynamic pressure
- **D** is the internal diameter
- **F_{Result}** is the resultant shaking force
- **\varphi** is the elbow angle

The shaking force values indicated in Figure 5 were calculated for each harmonic (Shaking Force harmonic cut-off frequency \(\sim 139.7\) Hz). Subsequently, these values (Harmonic Loading) of Shaking Force amplitude [N] were imposed in the FEM model using the principle of superposition.

![Figure 5: Shaking force harmonic amplitude](image)

The modal superposition method represents the vibration response of a structure by using the superposition of responses that characterize Single Degree of Freedom systems (SDOF). The natural frequencies and directions of vibration of these systems correspond to the natural frequencies of the analysed structure. The number of SDOFs contributing to a dynamic response is equal to the number of modes calculated by a pre-requisite modal analysis or frequency analysis.

A harmonic analysis assumes that the load is a function of frequencies rather than being directly dependent on time as is in the case of a time response analysis [5].

\[ [M] \ddot{d} + [C] \dot{d} + [K]d = F(t) \sin \omega t + B \cos \omega t \]

Where:

- **[M]** Mass matrix
- **[C]** Damping matrix
- **[K]** Stiffness matrix
- **F(t)** Vector of nodal loads (this vector is a function of frequency)
- **d, \dot{d}, \ddot{d}** Unknown vector of nodal displacements, speed, acceleration
In this case the analysis shall be performed in the range between 0 to 70 Hz, because the higher frequencies will have a negligible value of shaking force amplitude.

After defining the frequency range on which the vibration dynamic response shall be analysed, a modal Global Damping of 3% was defined. Assigning the same damping ratio to all modes represents a simplified and conservative approach. In most cases damping for higher modes will be higher than that for lower modes.

The results of the vibration analysis confirmed that the model has values very close to those obtained by the measurements in the field: 220 mm/sec rms measured on the front surface of the cooler vs a calculated value of 218 mm/sec rms as a mean of various points.

![Figure 6: Cooler model](image)

After creating a reliable model (Figure 6), four different options were studied, proceeding to check the vibrational response of each of them applying the same excitations (Figure 7):

- solution n°1: lateral “L” profile supports and boxing reinforcement of the vertical support.
- solution n°2: bottom lateral support in lower position.
- solution n°3: bottom lateral support in higher position.
- solution n°4: reinforcing gusset added on the top of vertical support.

On the basis of the structural response obtained through dynamic analysis, (carried out at the front of the cooler subjected to variable loads in the frequency domain), it was decided to select and engineer solution n°4 as it offered the best compromise between the vibration reduction and the activities required in the field.

![Figure 7: Dynamic analysis structural response](image)
During a planned shutdown, it was decided to implement the changes both on the line clamps (about 60 of them) and on the two most critical coolers. In Figure 8 there are two examples of clamps installed in the field.

![New clamp design](image)

Figure 8: New clamp design

The structural stiffening of the cooler realized and installed during the plant shutdown is shown, highlighted in green, in Figure 9.

![New cooler stiffening structure](image)

Figure 9: New cooler stiffening structure

The tightening torque for the bolts of the new clamps and support were chosen as the best compromise between the damping efficiency of the fabric pad and the support stiffness, using proprietary software developed specifically for this purpose. In fact, the manufacturer of the material recommends a specific maximum pressure to be applied in order not to excessively reduce the thickness of the material, with a consequent reduction in its damping effect. The optimum torque to be applied to the bolts was obtained from this value.

![Screenshot of the software used to evaluate the appropriate tightening torque](image)

Figure 10: Screenshot of the software used to evaluate the appropriate tightening torque
4 VIBRATION MEASUREMENTS AFTER MODIFICATION

A few months after the modifications, a diagnostic survey was performed at the plant to evaluate the effectiveness of the new supports by making new measurements and comparing them to the old ones taken before the modifications.

Displacement and velocity were measured and a real-time frequency analysis was conducted.

The survey covered all the 4 interstage lines, each with the related cooler. The measurement points were the same as those of the preliminary survey before the optimization.

The frequency peak of each measured point was compared with "Piping Allowable Vibration Limit Correction Curve" and "Piping Allowable Vibration Limit – Danger Curves" of the SwRI [6] [7]. These curves define the acceptable limits of the vibrations on the pipes associating a maximum displacement value with each frequency.

Regarding the interstage pipeline, Figure 11 shows an example of comparison between frequencies recorded on one point, before and after the optimization. The peak frequency is the most relevant to be considered in the current analysis, because it is the one that, without intervention, will cause pipe failure in the long term.

All the measured points were no longer above the "danger curve", but close to or below the "correction curve" so that one can say that the piping vibration issue may be considered solved.

![Vibration comparison on "point 009" before and after modifications](image)

**Figure 11 Amplitudes of the harmonics at the most relevant point, before and after modifications**

Regarding the coolers, the ones considered most critical due to high vibrations and history failures are those related to lines A and B: on both, the measurements show a clear improvement with respect to the previous situation. Table 2 shows the vibrations in terms of velocity (mm/s rms) "before" and "after" the intervention. The main vibration frequencies of all coolers are the same as the compressor speed (4.3 Hz).

<table>
<thead>
<tr>
<th>Cooler</th>
<th>E303B</th>
<th>X</th>
<th>Y</th>
<th>Z</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(mm/s rms)</td>
<td>before</td>
<td>after</td>
<td>before</td>
</tr>
<tr>
<td>inlet pipe</td>
<td></td>
<td>26</td>
<td>27</td>
<td>14</td>
</tr>
<tr>
<td>mid curves</td>
<td>255</td>
<td>29</td>
<td>200</td>
<td>36</td>
</tr>
<tr>
<td>outlet pipe</td>
<td>40</td>
<td>18</td>
<td>24</td>
<td>8</td>
</tr>
<tr>
<td>lower lateral pipe</td>
<td>170</td>
<td>30</td>
<td>-</td>
<td>9</td>
</tr>
<tr>
<td>upper lateral pipe</td>
<td>16</td>
<td>28</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

**Table 2: Cooler B vibration (mm/s rms): comparison before and after the optimization**
It was noticed that on two parallel sections of the suction line (line A and B) there was a significant difference in vibrations on certain sections of the piping. These lines are symmetrical to the compressor and cooler and have identical supports. The difference in vibration was explained by a different tightening torque being applied to the new clamps during the mounting stage: a test was carried out, observing the shift in the spectrum frequency in real time as tightening of the bolts was decreasing. The effect was the lowering of peak frequencies, moving within the acceptance area of SwRI curve.

**Piping supports**
The supports optimization produced a general improvement in the vibrations throughout the whole interstage system.

According to the frequency analysis, all points with vibration above the "danger curve" of SwRI dropped below it; moreover, the peaks of vibration and the relative frequency peak of each point, are now all below, or in a few cases near the "correction curve" of SwRI. At the moment, no point is believed to be in the danger zone. Regarding the few points that are still next to the "correction curve", all of them had an increment of the vibration frequency, which is more likely due to an over-tightening, which has altered the damping power of the rubber installed between the pipe and support.

**Coolers**
Observing the vibrations measured as velocity rms, a strong improvement was noted for all the coolers, including the items C and D where no structures had been stiffened. This is probably due to the combination of the following actions:

- Reduction of vibrations in the interstage piping, which now transmits less excitation to the cooler.
- Intervention on the cooler B foundation and skid
- Replacement of the wooden blocks of the cooling jackets on the cooler C and D themselves.

Observing the peak displacement, coolers B and C have still some isolated vibrations peaks on the lower side pipes, still below the "danger curve".

Further improvement may be obtained adjusting tightening of the clamps, measuring in real time the effect on vibration, to optimize the clamping to the value that generates a lower vibration. However, in order to be sure of bringing all points below the "Correction curve" it is necessary to completely revise the cooler structure.

It was also recommended that a new measurement campaign be carried out after the winter period to assess whether the vibrations on cooler C and D returned to high values: in which case, the cause should be sought in the loss of damping power of the new wooden dowels.

After 5-6 months of operation a couple of other failures occurred on cooler B, which was the most critical. These failures occurred at points where vibrations and supports were not considered critical at the time of the first survey. Analysis and recommendations are still in progress but the conclusion has been that the cooler structure and its foundations were not responding "ideally" as per the FEM analysis due to the state of preservation of the foundations, the presence of holes in the cooler structure caused by rust and the repeated welding and fixing on the old clamps and beams. An injection of concrete in the foundations and a general replacement of the ruined cooler parts has been scheduled for the next shutdown to better replicate the structure that has been simulated in the FEM analysis.
5 CONCLUSIONS
This case history is a typical one. An old compressor that was revamped with most of the auxiliary equipment remaining the same as at the beginning of its operation. During its life the various parts of the machine and the associated system have gone progressively deteriorating even if in a not evident and dramatic manner. Nevertheless that machine is a part of a plant that can continue to be productive, provided that it is submitted to an expert maintenance to take care of the modified operation, of the deterioration occurred and of the steps made by the technology in the meantime.

In the case of the high pressure LDPE plant reported in this article all of this was present with the addition of the non-availability of original drawings of the cooler. Nevertheless an accurate survey of the real actual situation, an assessment of the performance for the identification of the critical points and the application of new more powerful investigation technologies allowed to strongly improve a vibratory situation that exceeded by far the values considered acceptable. All of that was done also considering the constraint due to the production schedule of the plant which is not a research center but an economic activity and has to fulfil its engagements.

ACKNOWLEDGMENTS
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REFERENCES