A STUDY OF THE CENTRIFUGAL COMPRESSOR DISCHARGE PIPELINE CONSTRAINED OSCILLATION

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Abstract: The paper presents a solution of a pipeline constrained oscillation that appeared on a natural gas compression plant. The station operator complained about a problem after installation of new higher efficiency impellers. Series of pipeline modifications were suggested for the problem solution. The structural modifications were simulated by the Finite Elements Method in ANSYS Workbench v.13.0. The numerical calculation results were compared to the each single modification and were verified by experimental measurements of the discharge pipeline shell oscillation.

Keywords: oscillation, pulsation, discharge pipeline, blade pass frequency, harmonic analysis

1 Introduction

Vibration of the mechanical systems is a well-known technical term and a widely analyzed phenomenon. Vibration problems appear in the whole range of machines with reciprocating and rotation parts movement. In addition, the mechanical vibration problem can be observed in machines which are exposed to external or internal excitation pulses. Coherent process technology influencing can be a good example of an external excitation force. The forces from unbalanced rotor parts, flowing medium turbulence or operating environment pulsation are typical internal excitation forces.

Mechanical vibrations are common for compressor machinery applications. In certain circumstances vibration complicates a safe operation. An agreement between the natural frequency of the system and an excitation frequency can lead to a resonance state of a mechanical system and cause an incident or failure.

ČKD Kompresory, a.s. is a traditional manufacturer of industrial compressors of both turbo- and reciprocating design. Three centrifugal compressors were renovated on a natural gas compression plant in 2011. Retrofit mainly covered modernization of the stator inserts and impellers with higher nominal parameters. Increased vibration of a discharge piping was reported after the first compressor proof test. Moreover, break-downs of measurement sensors, like temperature and pressure gages, were detected after the next short-time operating. Although compressor pipelines are commonly designed for particular compressor machines and not vice versa, compressor manufacturer offered a qualified technical assistance in vibration eliminating solution.

2 Vibration operating measurements

2.1 Technical parameters of the structure

With regard to the subject of retrofit, it is supposed, that higher discharge pressure pulsation can cause higher pipeline vibration. Moreover, the phenomenon of the system resonance state must be taken into account.
Compressor discharge pipeline design has following parameters:
- outer diameter 1020 mm,
- wall thickness 22 mm,
- pipeline material L415QB steel.

Centrifugal compressor has following nominal parameters:
- nominal rotation speed 3600 rpm,
- number of impeller vanes 14 pcs,
- pressure on the suction 4.85 MPa,
- pressure on the discharge 7.45 MPa,
- temperature on the suction 23 °C
- temperature on the discharge 60 °C

The centrifugal compressor is driven by a synchronous electric motor with variable speed range between 2700 rpm and 3900 rpm.

2.2 Operating measurements

Two peak frequencies of the compressor discharge pipeline shell were identified experimentally after vibration operating measurements were provided by an external company. The first pulsation peak is located near the frequency 1640 Hz. The second one is located near the frequency 2450 Hz. The amplitude-frequency response of the measurements is shown in the Diagram 1. These peak frequencies are located almost exactly in 2x and 3x blade pass frequency. The blade pass frequency (BPF) equals the product of the running rotor speed and the number of impeller vanes. This confirms the suspicion about compressor discharging pipeline shell resonance state. In other words, a clear system constrained oscillation phenomenon appears.

Diagram 1 - The amplitude-frequency measurement response
3 Numerical simulation of current discharge pipeline

3.1 Model for modal and harmonic analysis

Numerical simulation was done in several steps. First of all, it was necessary to confirm the peak pulsation frequency values and compressor discharge pipeline resonance state. Geometric model of the compressor set was simulated for the numerical calculations. Model consisted of compressor casing part and discharge pipeline part. Compressor casing part was derived from inner surface of the casing. The discharge pipeline was defined from compressor discharge branch to the second ell included. The model is in Image 1.

![Geometry](Image 1 – Geometric model for analysis)

Mathematical model for modal and harmonic analysis was modelled by SHELL181 element type [1] in this step of simulation. Numerical model consisted of approximately 35,000 elements and 35,000 calculating nodes (Image 2). Compressor casing part was bound by Fixed Support on the side edges. Boundary conditions were the same for modal and harmonic analysis.

Numerical model for harmonic analysis was loaded by pressure pulsation, medium weight and deformations from omitted discharge pipeline part behind the second ell. Pressure pulsation was applied on an inner cylindrical surface of the discharge pipeline on 1 m length from compressor discharge branch. Loaded pipeline length and pressure pulsation value were determined from previous transient CFD simulation, which is not a subject of this paper. Deformations from omitted discharge pipeline were applied on the second ell cross-section. Values of directional deformations and rotation angles applied on the cross-section area were the output data from an external static analysis of discharge main and are not a subject of this paper. Constant damping ratio value was considered equal to 1%.

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3.2 Results of the current discharge pipeline harmonic analysis

Results of the modal analysis showed that a great number of shell mode piping natural frequencies is between 1500 Hz and 3000 Hz. But it doesn’t mean that all of these modes can be excited. Resonance states of discharge pipeline and their shapes can be found out using the harmonic analysis. The Diagram 2 represents the harmonic response of discharge pipeline along 2.5 m length from compressor discharge branch. Analyzed length was determined with regards to temperature and pressure gages disposition on the pipeline. Harmonic amplitude-frequency response was calculated in frequency range between 0 Hz and 3000 Hz. Since exciting force value is unknown, calculated results might be used only for consequent comparison.
The first peak of frequency resonance state lies around the frequency 1630 Hz. Corresponding discharge pipeline shell shape mode is in Image 3. The second peak of frequency resonance state range was not determined in FEM simulation. It can be explained in several ways. Firstly, value of the constant damping ratio may be inappropriate. And secondly, it can be due to an unknown value and exciting force behavior. However, next adjusting of the numerical model was not carried out. It wasn’t done because the maximum amplitude response at frequency around 1630 Hz is much higher than amplitude response at frequency 2450 Hz. It was supposed that damping of the first resonance response range would solve the problem of discharge pipeline constrained oscillation.

![Image 3 – Mode shape of the current discharge pipeline at frequency 1630 Hz](image)

4 Damping using an external system

4.1 External damping system design

According to operating measurements, discharge pipeline oscillation decreases significantly with the distance along pipeline. Therefore, discharge pipeline damping drafts were aimed at the first pipe section along 2.5 m length between discharge compressor branch and the first ell.

The first modification draft included damping by an external vibration absorber. Vibration absorber consisted of elastic metal cushions and supporting structure for the positioning and fixing on the discharge pipeline shell. The damping draft drawing is in Image 4. The cushions are formed from knitted wire mesh with elastic properties. Damping by cushions is caused by friction between the individual wire windings. More information about metal cushions can be found on the manufacturer’s web-site [4].

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4.2 Harmonic analysis of the first damping draft

Geometrical model form previous analysis was complemented by damping absorber model. Damping metal cushions were modelled by finite element COMBIN14 [1]. Stiffness and damping values were specified in compliance with manufacturer’s data. Damping assisting system construction was modelled by finite element SOLID186. The mathematical model of pipeline, boundary conditions and loads remained the same as in previous analysis.

Results of the first damping draft numerical simulation showed a significant vibration damping of the first pipeline section. Oscillation amplitude decreased to 50%.

4.3 Realization and operating measurement of the first damping draft

External damping absorber was manufactured and installed on the compressor discharge pipeline. After that, the vibration operating measurement was provided. Unfortunately, vibration declination of the discharge pipeline shell was low. The Diagram 3 shows comparison between the operating measurement results for pipeline with and without external damping absorber.

Mismatch of calculated and measured results can be explained by several circumstances. Firstly, metal cushions damping efficiency depends on the normal to
cushion displacement magnitude. The value of the normal displacement of discharge pipeline shell is approximately 20 μm. Probably, this value is too low to develop appropriate friction between the individual wire windings. Secondly, damping absorber has six metal cushions. It is very difficult to set the same assembling conditions for each cushion during installation in practice. Possibly, the initial compression of each cushion can be different. And finally, the contact surface between metal cushions and pipeline shell was insufficient for effective oscillation damping.

5 Damping using a wall thickness increasing

5.1 Stiffness increasing design

The second vibration damping draft included stiffness increasing of the pipeline shell. The simplest way how to do this is increase the pipeline wall thickness. It was supposed that the first section pipeline with 22 mm wall thickness is replaced by another with 45 mm wall thickness. The substituting pipeline length is approximately 2.5 m long. Decreasing of the resonance state shell amplitude maximum values was expected. Additionally, it is possible to shift the resonance frequency of pipeline shell out of the resonance frequency range by stiffness changing.

5.2 Harmonic analysis of the second damping draft

Harmonic analysis of the pipeline design with increased stiffness was simulated for the efficiency and operating safety confirmation. Geometrical model of current pipeline was modified by the first section wall thickness increasing from 22 mm to 45 mm. Pipeline geometry was also complemented by detailed models of the welds between the new pipeline section and discharge compressor branch or the first ell respectively. The rest of geometrical model didn’t change. Detailed welded connections were modelled by SOLID186 and are shown in the Image 5a and Image 5b. Boundary conditions and loads remained the same as in harmonic analysis of original pipeline.

According to the analysis results, the maximum value of amplitude-frequency response decreased by 70%. The Diagram 4 shows comparison of the calculated results for original pipeline and discharge pipeline with increased stiffness. Image 6 shows the calculated shell mode shape of discharge pipeline at 1636 Hz frequency value. The mode

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shape of the pipeline with increased stiffness is nearly the same as for current pipeline shape form.

![Diagram 4](image_url)  
**Diagram 4** – Amplitude-frequency calculated response comparison for pipeline with 22 mm and 45 mm wall thicknesses

![Image 6](image_url)  
**Image 6** – Mode shape of the discharge pipeline with 45 mm wall thickness at frequency 1636 Hz

The maximum equivalent stress values were calculated in discharge pipeline harmonic analysis. Their locations are on welds between the new pipeline section and discharge compressor branch or the first ell respectively. The first maximum equivalent stress value is equal to 4 MPa and is located on the welded connection between discharge compressor branch and new pipeline section. The second maximum equivalent stress value 2 MPa is on the welded connection between new pipeline section and first ell.

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These values are considered as amplitudes superposed on nominal stress distribution. Image 7 and Image 8 show the equivalent stress distribution from harmonic loading for both welded connections.

![Image 7](image7.png)  
**Image 7 – Equivalent stress distribution in the first welded connection**

![Image 8](image8.png)  
**Image 8 - Equivalent stress distribution in the second welded connection**

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5.3 Strength analysis of the second damping draft

Strength analysis was done in order to confirm the safety of long-term operation of compressor discharge pipeline. Moreover, it was important to analyze the static stress from operation loads in new welded connections. Two critical zones with maximum equivalent stress values on the new welded connections were chosen. Image 9a and Image 9b show the stress distribution in the new welded connections from the static loading. The maximum equivalent stress value in the first welded connection is 171 MPa, in the second welded connection is 234 MPa.

![Image 9a](attachment:9a.png)  ![Image 9b](attachment:9b.png)

Image 9 – Equivalent stress distribution in the new pipeline section welded connections
a – on compressor discharge branch side
b – on first pipeline ell side

The stress analysis of the discharge pipeline shell was done according to the European Standard [2]. The resultant stress \( \sigma_v \) shall be compared with allowable stress \( \sigma_d \). The standard allows using HMH stress for the analysis. The HMH stress can be calculated in accordance with equation (1), or can be taken directly from the FEM analysis.

\[
\sigma_{\text{HMH}} = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2 - \sigma_x \cdot \sigma_y - \sigma_y \cdot \sigma_z - \sigma_z \cdot \sigma_x + 3 \cdot (\tau_x^2 + \tau_y^2 + \tau_z^2)}
\]  \hspace{1cm} (1)

where \( \sigma_x, \sigma_y, \sigma_z \) – principal normal stresses, [MPa];
\( \tau_x, \tau_y, \tau_z \) – principal shear stresses, [MPa].

The allowable stress \( \sigma_d \) is calculated in accordance with equation (2):

\[
\sigma_d = 0.72 \cdot R_{0.5}
\]  \hspace{1cm} (2)

where \( R_{0.5} \) – minimum 0.5% proof strength, [MPa].

The resultant stress \( \sigma_v \) is determined by the sum of stress values from static and dynamic loadings in accordance with equation (3).
\[ \sigma_v = \sigma_{st} + \sigma_{dyn} \]  

(3)

where \( \sigma_{st} \) – stress from static loading [MPa];

\( \sigma_{dyn} \) – stress from dynamic loading [MPa].

Mechanical properties of discharge pipeline material are in the Table 1. The mechanical properties are given in accordance with the European Standard [3].

<table>
<thead>
<tr>
<th>( \rho ) [kg·m(^{-3})]</th>
<th>( \nu )</th>
<th>( E ) [MPa]</th>
<th>( R_{0.2} ) [MPa]</th>
<th>( R_m ) [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>7 850</td>
<td>0.3</td>
<td>210 000</td>
<td>415</td>
<td>520</td>
</tr>
</tbody>
</table>

Resultant stresses for the first and the second critical places are:

\[ \sigma_v^I = 171 + 4 = 175 \text{ [MPa]} \]  

(4)

\[ \sigma_v^{II} = 234 + 2 = 236 \text{ [MPa]} \]  

(5)

The allowable stress is:

\[ \sigma_d = 0.72 \cdot 415 = 298.8 \text{ [MPa]} \]  

(6)

By comparing result stresses for the first and second welded connections with allowable stress the following can be received:

\[ \sigma_v^I = 175 \text{ MPa} < 298.8 \text{ MPa} = \sigma_d \quad \text{– satisfactory} \]  

(7)

\[ \sigma_v^{II} = 236 \text{ MPa} < 298.8 \text{ MPa} = \sigma_d \quad \text{– satisfactory} \]  

(8)

The design with increased stiffness of the compressor discharge pipeline satisfies the European Standard requirements.

6 Conclusion

The constrained oscillation of the centrifugal compressor discharge pipeline was analyzed by FEM and was compared to operating measurements. Two design improvements were offered to solve the resonance state of the compressor discharge pipeline.

The first improvement was damping by the external vibration absorber with elastic metal cushions. This draft was analyzed and experimentally validated. The first draft was rejected after the operation measurement because of low efficiency.
Damping by pipeline wall thickness increasing was the second improvement draft. It was supposed that stiffness is increased by replacing the first discharge pipeline section 2.5 m long with the new one. The new pipeline section has 45 mm wall thickness compared to the 22 mm wall thickness of the original one. The second draft was analyzed by FEM and evaluated according to European Standards. Practical realization of the second improvement will have been done by the end of 2012.

Acknowledgement

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References

[1] ANSYS 13.0 Help
[4] [http://www.stop-choc.de]

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