Optimisation of Acoustic Silencer for the Screw Compressor System

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Abstract: In one of the screw compressor system, designed silencer was not optimal. A great challenge was the large variation in operating conditions, especially the variation of the molecular weight of the gas. There was need to optimize the silencer. This paper describes the acoustic modelling tools to optimize the acoustic performance of the silencer. Optimization steps carried out are based on the process parameters (molecular weight and speed of sound of the gas) and geometric parameters (size of silencer, thickness and length of the absorption core). For this acoustic modelling COMSOL has been used extensively. A 3-D model of the silencer has been used in the present investigation. Transmission losses and pressure pulsation reduction ratios were calculated for each case. Results are presented below.

Keywords: Acoustic silencer, screw compressor, mineral wool, transmission loss, pressure reduction factor.

1. Introduction

Screw compressors are becoming more common in installations of the process industries due to increased needs for gas compression applications. Normally these compressors generate significant pulsations at the lobe passing frequencies and its multiples. The frequency of the source, in the range of 100-2000 Hz, affects a strong coupling of this acoustical power to the mechanical structure and it can cause vibrations that are difficult to control. These high pulsations can transmit into the upstream or downstream piping of the system and affects the integrity of the system. Often, one badly designed silencer can underscore the importance of reliable design and the need for a complete system acoustic/mechanical evaluation.

A careful analysis of the system design is mandatory to avoid issues due to excessive pulsations and pulsation-induced vibrations. Industrial standards such as the API 619 standard (for screw compressors) stipulate a design approach to identify and resolve pulsation issues in the design phase, as will be discussed in section 2. A careful specification and evaluation of the full operating envelope is essential. This is especially true for systems with an extreme operating envelope such as collective vent and flare gas compressor systems. The variance in the gas composition has an important impact on the pulsation behavior for several reasons. In one case it appeared that an original design of pulsation silencer was not acceptable due to the high pressure pulsations that occurred at certain process conditions. It appeared that the performance of the silencers was insufficient to cover the full range of process conditions, specifically molecular weights. Therefore the analysis of the silencer performance over the entire range of molecular weights. As especially the performance of the absorption material changes with molecular weight, the focus of this paper is on the acoustic damping.

2. Pulsation analysis of screw compressor systems

2.1 Pulsation sources

For the waste gas application a screw compressor appeared to be the best choice. For screw compressor systems the pulsation problem is hardly recognized, which has proven to lead to vibration problems especially for the larger screw compressors. In spite of the fact that the pulsation phenomenon is more complex in screw compressors, the API 619 standard recommends only a relatively simple criterion for allowable pulsations.

The generated pulsations have a much higher frequency than pulsations generated by reciprocating compressors. The high frequency causes that, especially for the larger compressors, in the propagation of pulsations three dimensional (3D) modes play a role and prohibit the commonly used one dimensional

(1D) modeling. Moreover also the vibration response is more complicated, because in addition to the axial bending modes also circumferential modes are excited.

2.2 Pulsations generated by screw compressors

Screw compressors are sources of pulsations. This can be explained by considering the mechanisms of the compression. The pressure of the gas is increased by transfer of mechanical energy from the rotation of 2 helical rotors (female and male lobes) to the gas. At the suction side, a pocket of gas is inhaled and enclosed within a cavity between the lobes. By the design of the rotor geometry, the volume of the cavity is reduced while the gas pocket travels toward the discharge side, thus increasing the pressure. Upon opening of the pocket, the compressed gas is exhausted into the discharge piping. This occurs on an intermittent basis, and thus leads to a pulsating flow. In many screw compressor designs, 4 pockets per revolution (4 male lobes) are transported to the discharge side. In that case, the pulsation source spectrum is dominated by the integer multiples of the pocket-passing frequency (PPF). For example at the 4th harmonic of the compressor speed (1*PPF), the 8th harmonic (2*PPF), the 12th harmonic (3*PPF) etcetera.

The typical pulse shapes are illustrated in Figure 2. Note that these time signals are associated with 1 pocket contained by the rotors. In reality, the signal will be repeated, 4 times each crank revolution, with a phase shift of 90 degrees.

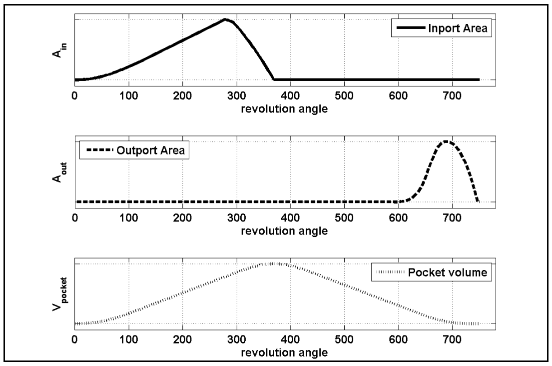

Figure 1. Typical inport area , outport area and pocket volume time functions.

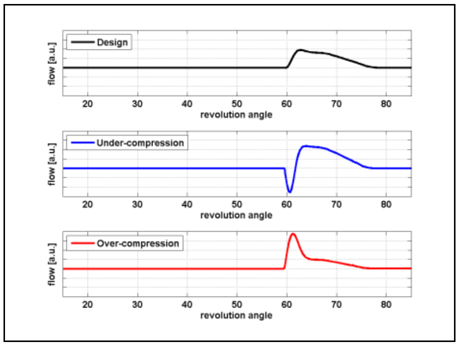

Figure 2. Typical discharge pulse shapes generated by one lobe, for ideal design, under-compression case and over-compression case.

The propagation of the pockets from suction to discharge side takes more than 1 revolution of the compressor axis; in the example of Figure 1 nearly 700 degrees. It must be noted that, due to the geometrical layout of the rotors, multiple pockets can be exposed to the suction and discharge side. See for example in Figure 1, the opening of a pocket at the suction side is nearly 360 degrees. Per revolution, 4 pockets (each pocket is shifted 90 degrees relative to the previous one) are exposed at the same time. This has an important effect on the amplitude of the pulsations. When multiple pockets are exposed to the inlet our outlet piping, the unsteady effect will tend to be more 'averaged'. This is one of the reasons why pulsations at the suction side are generally less strong (multiple pockets are exposed, in contrast to the discharge side where usually only one or 2 pockets are exposed simultaneously). Another reason is the absence of a pressure mismatch at the suction side, in contrast to the discharge side. However, also the suction flow will have an unsteady character, and thus the risk of pulsations shall be carefully evaluated in the design!

2.3 Pulsation analysis according to the API 619 standard

Since positive displacement machinery can be significant sources of pulsations, industry standards are developed that stipulate the design evaluation of these systems. for screw compressors, the API 619 standard [1] addresses some design considerations with respect to pulsations. The description is rather global, and focusing on the pulsation suppressors/silencers.

Exact details on the approach are not included in the standard.

It is stated explicitly that the entire operating range shall be considered. It is clear that for vent gas screw compressors this is an important and challenging statement.

The API 619 standard provides the following allowable level for pressure pulsations in the system:

$$P_{allowable} = \frac{28.6}{P^{1/3}} \text{, or 2\%, whichever is smaller} \qquad (1)$$

$P_{allowable}$ is the allowable pulsation level in % peak-to-peak of the mean pressure. P is the line pressure in kilopascal absolute. The criterion is illustrated in Figure 3.

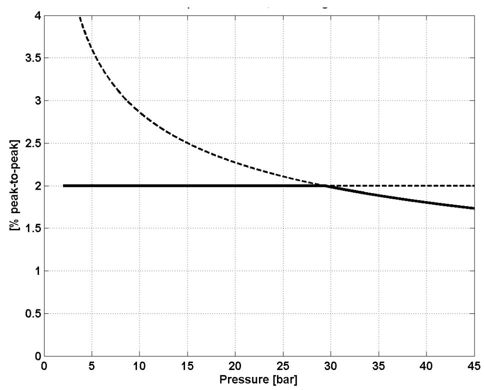

Figure 3. Allowable pulsation level, according to API 619 standard.

Furthermore, the standard addresses an important point. Due to the relatively high rotation speed, the acoustic modes associated with the pulsations, are three-dimensional (3D). The propagation of low-frequency pulsations can be conveniently described with plane-wave (one-dimensional) theory. This is a simplified approach that is valid for low-frequency pulsation sources, such as reciprocating compressors.

With increasing frequency, more and more higher-order acoustic modes will contribute to the propagation of the acoustic waves. The onset for the occurrence of 3D modes is the so-called cut-on frequency fc,i. For circular pipes, the cut-on frequency for the ith acoustic mode is described by:

$$f_{c,i} = k_i \frac{c}{\pi D_i} \qquad (2)$$

k_i is the cut-on coefficient for mode i, c is the speed of sound of the gas and Di is the inner diameter of the pipe. The first higher order (3D) mode has k_i=1.84. For example for a natural gas system with a speed of sound of 400 m/s, **Error! Reference source not found.** below illustrates the cut-on frequencies of the first higher order (3D) mode, for different pipe sizes.

Table 1. Frequency limits for plane-wave propagation of pulsations.

Pipe size	Inner diameter [mm]	Plane wave limit frequency
1”	26.6	8794 Hz
2”	52.5	4463 Hz
4”	102.3	2291 Hz
8”	202.7	1156 Hz
16”	387.4	605 Hz

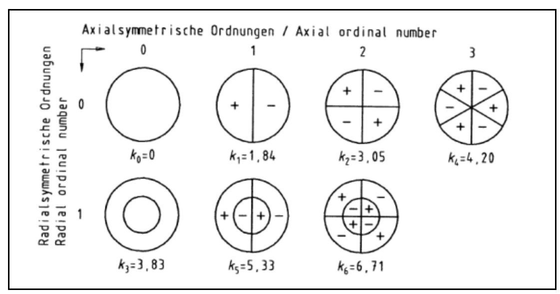

Figure 4. Illustration of plane wave (k0) and higher order acoustic modes, from [2].

Screw compressor systems with high rotational speed, large pipe sizes and heavy gases will readily violate the plane-wave assumption shown in Table 1. Often, the lowest order(s) of the Pocket Pass Frequency (PPF) are in the 1D-range, but the higher orders, will trigger 3D acoustic effects. This is particularly true inside the silencer, where in general larger geometrical dimensions are found, compared to the piping. Including 3D effects into the simulation model, requires an excessive increase in modelling effort and calculation time.

2.4 Pulsation dampers used in screw compressor system

Generally, absorptive or a combination of absorptive and reactive type of silencers are used in screw compressor systems. Absorption material core placed in the middle of the silencer. Normally, glass wool, rock wool, mineral wool or polymer foam is used as acoustic absorption material. They perform very well at the higher frequencies and dissipate the acoustic power. Sometimes, an additional absorptive material layer along the inner wall of the silencer shell is placed. The absorption inner core is made up of four layers, absorption material, thin fibre layer, wire-mesh around and finally packed into the perforated sheet. This combination gives a good mechanical integrity. If the acoustic attenuation is not sufficient, an additional resonator, which is effective in the lower frequency range, can be installed in combination of the absorptive silencer. It is very important to note that API 619 [1] also stipulates the allowable pressure losses across the silencers. Therefore the absorption material should still allow a sufficient flow area.

3. COMSOL Multiphysics modeling

The 3D modeling of the complete silencer has been done using COMSOL version 4.3. The acoustic pressure frequency domain module of the COMSOL has been used in the present investigation. Boundary conditions used are:

Inlet: plane wave with incident pressure field (1 Pa)
Outlet: Reflection free boundary
Walls: sound hard wall boundary

Mesh dependency test has been carried out to see the mesh adequacy on the results. Typical geometry of design case silencer is shown in Figure 5. Typical mesh of the geometry is shown in Figure 6. It can be seen that there is a central core mineral wool core and gas flows through the annulus (between core and the silencer wall). Fluid in the open flow path is modelled as linear elastic model.

Absorption material core is modelled in COMSOL through macroscopic empirical porous models which mimics the bulk losses in certain porous/fibrous materials. The model represents a porous medium with the following complex propagation constants:

$$kc = \frac{w}{c}\left[1 + C_1\left(\frac{\rho_0 f}{R_f}\right)^{-C_2} - iC_3\left(\frac{\rho_0 f}{R_f}\right)^{-C_4}\right] \quad (3)$$

$$Z_c = \rho_0 c\left[1 + C_5\left(\frac{\rho_0 f}{R_f}\right)^{-C_6} - iC_7\left(\frac{\rho_0 f}{R_f}\right)^{-C8}\right] \quad (4)$$

Where k_c is the complex wave number, Z_c is the complex impedance, ρ_0 is the fluid density (kg/m^3), f is frequency (Hz), c is the speed of sound (m/s), R_f is the flow resistivity (Pa.s/m^2) in the porous medium and C_1-C_8 are constants. These constants are different for different absorption material. For the mineral wool these values are not available in the COMSOL tool so we used these constant from text book by Beranek [2].
The constants used are:
$C_1 = 0.136$, $C_2 = 0.641$, $C_3 = 0.322$, $C_4 = 0.502$
$C_5 = 0.081$, $C_6 = 0.699$, $C_7 = 0.191$, $C_8 = 0.556$

Flow resistivity of 35000 (Pa.s/m^2) is used for the calculations. Nevertheless, a sensitivity of the flow resistivity (varied from 5000 – 35000 Pa.s/m^2) has been carried out and it has a significant impact on the performance.

Eigen mode analysis has been carried out for a design case to understand and gain insight on the type of 3-D acoustic modes present in the silencer.

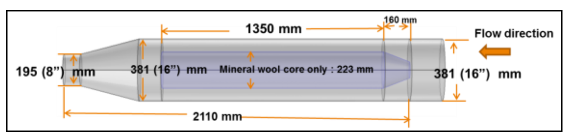

Figure 5. Simulation model of the design case silencer .

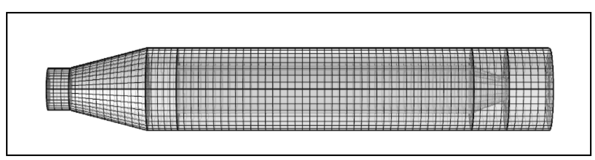

Figure 6. Mesh of the Simulation model of the design case silencer .

4. Results and discussions

The performance of absorptive silencer depends on the geometry of the silencer and absorption material. Geometrical aspects of the silencer are:

- Length of the silencer
- Diameter of the silencer

The performance of the absorption material depends on the gas composition and on the following aspects:

- Wave length versus thickness of the material (inner core and/or layer around the inner wall of the silencer)
- Operating pressure and density.
- Porosity of the material, usually classified as flow resistance.
- Viscosity of the gas.

Performance of the silencer is measured by Transmission losses and pressure reduction ratio calculated for a the frequency range of 10 – 3000 Hz.

The transmission loss of a silencer is defined as

$$TL = \log_{10}\left(\frac{W_{in}}{W_{out}}\right) \qquad (5)$$

Where Win is the incoming power at the inlet and Wout is the outgoing power at the outlet.
The incoming power at the inlet is

$$W_{in} = \int_{Ain} \frac{P_{in}^2}{2\rho c_s} dA_{in} \qquad (6)$$

Outgoing power at the outlet is given by

$$W_{out} = \int_{Aout} \frac{P_{out}^2}{2\rho c_s} dA_{out} \qquad (7)$$

Where Pin is the average pressure at the inlet, Pout is the average pressure at the outlet, ρ is the gas density, c_s is the speed of sound and A is the surface area.
Pressure reduction ratio is defined as

$$P_{red} = \frac{P_{out}}{P_{in}} \qquad (8)$$

The transmission loss for the low molecular weight gas with is shown in Figure 7. This gas has mol weight of 9.5 kg/kg-mol and speed of sound of 620 m/s. It can be observed that silencer performs well at the higher frequencies (> 1200 Hz). Reduction in pressure pulsation ratios is shown in Figure 8. It can be observed that only 10 % of the pressure amplitude is reduced at the 400 Hz (1st order of the pocket-passing frequency: compressor speed of 6000 rpm with 4/6 lobes layout).

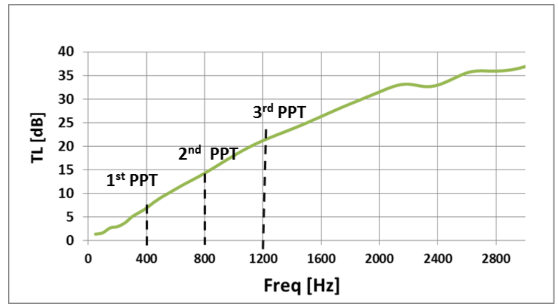

Figure 7. Transmission loss of the reference geometry .

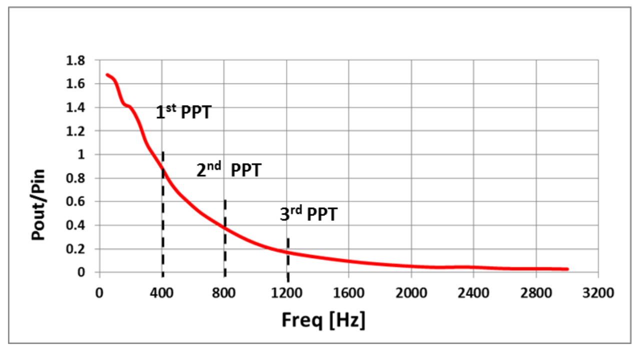

Figure 8. Pressure pulsation ratio of the reference geometry .

The results of Eigen mode analysis are shown in Figure 9 – Figure 10. Higher order acoustic modes can be observed.

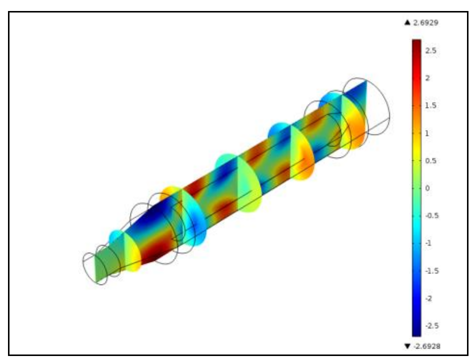

Figure 9. Typical result of a COMSOL eigen mode calculation at 1250 Hz.

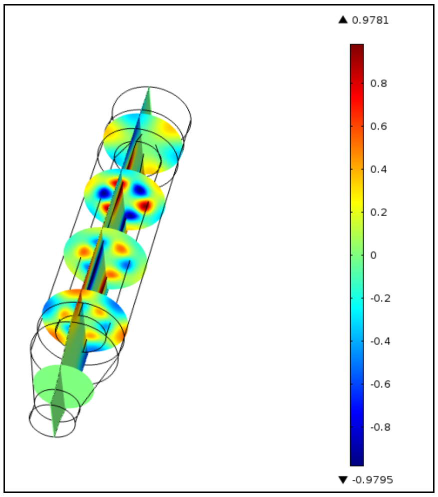

Figure 10. Typical result of a COMSOL Eigen mode calculation at 2100 Hz.

Optimization results are presented in the sections below.

3.1 Effect of process conditions

Well-designed silencer need to show good performance for all the operating gas compositions. Results of three different gas compositions are calculated for the reference geometry. The pressure pulsation reduction as function of frequency is shown in Figure 11. The heaviest gas has molecular weight of 40.4 kg/kg-mol and speed of sound of 285 m/s whereas lightest gas has molecular weight of 9.7 kg/kg-mol and speed of sound of 620 m/s respectively. The performance of the silencer is best at the heavy gases (high Molecular Weight, low speed of sound c) while worst at the light gases. The performance improves with increasing frequency.

Heavier gases can absorb the pulsation and dissipate the power better than the lighter gas in frequency range of 400-2000 Hz. In screw compressor applications, the dominating pressure pulsations are at one and two times the pocket-passing frequencies.

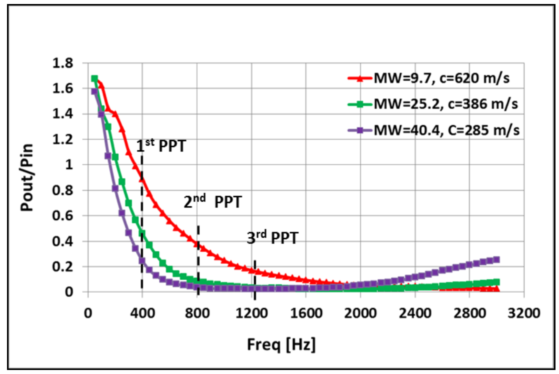

Figure 11. Effect of gas composition on the performance.

3.2 Effect of core geometry

The effect of mineral wool core thickness has been investigated. Core thickness varies from 223 mm (reference geometry) to 303 mm (see Figure 12). An additional layer of 50 mm around the mineral wool core consists of wiremesh, knitted stainless steel and perforated plate. The rest of the silencer geometry is kept same. This means that for the larger core thicknesses the flow path is narrower and thus the pressure loss higher.

The performance of the silencer for three different mineral wool cores is shown in Figure 13. Results indicates that higher the core diameter better is the performance. This better performance is at the cost of pressure loss. There is a trade-off between the core thickness and flow gap which is available for the gas to flow, i.e. a trade-off between acoustic damping and pressure loss. There is a API 619 guideline that specifies the allowable pressure loss across the silencer.

It has been observed that increasing the silencer size (also core length increases) has a positive impact on the silencer performance.

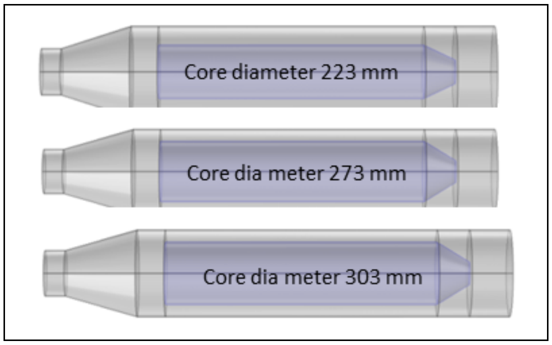

Figure 12. Variation of mineral wool core thickness for the same silencer diameter.

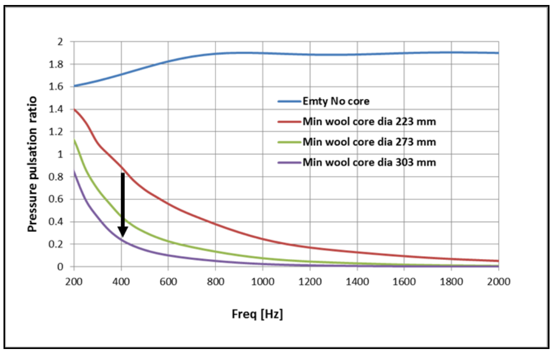

Figure 13. Effect of core thickness on the performance.

3.2 Optimized geometry

The silencer geometry is optimized based on the feasibility and the cost. Length of the silencer is increased by 20 % (effective core length from 1350 mm -> 1600 mm), cross-sectional area of the silencer by of the silencer is reduced by 12 % (inner diameter form 381 mm -> 362 mm). Mineral wool core thickness is kept the same (223 mm (reference geometry).

Results are shown in Figure 14. It can be seen that for the lighter gas there is 30 % reduction in pressure pulsations (amplitude) compared to the designed silencer (Pressure reduction ratio from 0.9 to 0.6). same holds good for the other gas compositions. This optimized results and model has been in the further pulsation analysis in the field piping.

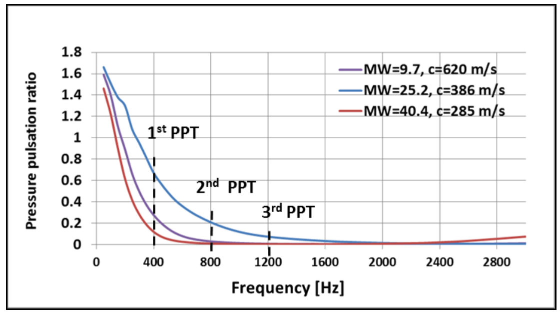

Figure 14. Results of optimized geometry

4. Conclusions

Numerical modeling has been performed to optimize the performance of the absorptive silencer for screw compressor system. 3D modeling is necessary to capture higher order acoustic modes in the silencer.

The COMSOL's pressure acoustics frequency domain module has been an excellent tool for the optimization. The silencer has been optimized with respect to geometry of silencer, mineral wool core thickness and different operating gases. These results were used in the pulsation analysis of the field piping.

5. References

[1] API 619 standard, "Rotary-type, Positive-displacement compressors for Petroleum, Petrochemical and Natural Gas Industries", 4th edition, 2004.

[2] VDI 3733 standard, "Noise at pipes", 1996.

[3] Leo L. Beranek (ed.), "Noise and Vibration Control", McGraw-Hill Inc. (1971), ISBN 0-07-004841-X