Validation of Pressure Pulsation and Vibration Analyses in a Reciprocating Compressor for Design Optimization

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Abstract:

Pressure pulsations are one of the main issues in reciprocating compressors applications. They can generate high pulsation induced shaking forces leading to high vibrations on compression plant elements such as piping, dampers, coolers, small branches and shaking forces, which can even lead to fatigue failures. In addition, pulsations can affect the reliability and life of compressor valves. For this reason, pressure pulsation prediction according to API 618 standard is a fundamental step in reciprocating compressor plant design.
The present work shows the comparison between design simulation, carried out through a dedicated proprietary multi-physics software, and the results of a dedicated test on the plant (2 stages, 800 kW

compressor for Oil&Gas application) performed together with the OEM. The test was fitted with dynamic pressure and vibration measurements in a series of key points on the cylinders and dampers.
The simulation is solved in the time domain, using a one-dimensional CFD model for piping, dampers and plenums, and a lumped model for the cylinders. The flow dynamics in the ducts relies on conservation equations (mass, momentum and energy) using the Lax-Wendroff method to obtain numerical solutions (based on finite differences, second-order accuracy in space and time) for internal points and the method of characteristics for external points.
If compared with simulations based on acoustic theory, this method provides a better accuracy because it can take into account cylinder-plant mutual interaction, the non-linear effects of viscosity, the heat exchange and the real behaviour of valves.
On the other hand, this method requires a large amount of input data. Therefore, a comprehensive consistency between test and simulation can be difficult to achieve, due to possible measurement interferences and to lack of information regarding the actual setup. In this paper, the methods to overcome possible discrepancies are shown in detail.
Furthermore, a study to reduce pulsations in the most critical condition was carried out and the effect on pulsations of alternative capacity control techniques (reverse flow and additional clearance) was also investigated. Finally, a tuning with vibration measurement was used to show the correct setting of the model.
The work constitutes an important lesson learned that allowed the OEM to better evaluate and optimize the effectiveness of efficient countermeasures to reduce pulsations, as pulsation dampers and pipe and support geometries.

1 Introduction

Reciprocating compressors are cyclic machines and the generation of pressure pulsations is one of their main issues. Each cylinder is equipped with suction and discharge valves, acting like check valves; they stay open for a small part of the cycle and thus generate flow and pressure pulsations propagating in the lines through the piping and the auxiliary equipment at the speed of sound.
Unless carefully analyzed, pressure pulsations can have several negative effects on a compressor: they can cause change in performance, reduction of valve availability, and errors in flow measurement. But primarily, designers have to take care of pulsations because they cause shaking forces in the plant, at all points where there is a discontinuity, with consequent vibration of piping and equipment and potential fatigue failures.
In order to reduce pulsations amplitude, damping volumes and orifices are commonly used. General advices for sizing these devices can be found in [1].
API 618 standard provides procedures to predict and control pressure pulsations and vibrations under certain limits, by means of Pulsation Analysis and Mechanical Analysis [1].
The Pulsation Analysis can be performed by means of the Acoustic Wave Equation or by means of numerical methods for Navier-Stokes equations. For more detailed review on fluid equations see [3].
The Acoustic Wave Equation consists in linear equations, which can be solved in frequency domain with a relatively low computational effort but can introduce uncertainty in the results, since they neglect the non-linear terms of the Navier-Stokes equations.
Numerical methods for Navier-Stokes equations can be solved in time domain, taking into account viscosity, heat exchange, mean flow and gas compressibility [4].
After the Pulsation Analysis a Mechanical Analysis is performed, having as input the gas and inertia forces of the compressor and the shaking forces acting on the piping and on the pulsation dampers, obtained with the former analysis.
Furthermore, a study to reduce pulsations in the most critical condition was carried out and the effect on pulsations of alternative capacity control techniques (reverse flow and additional clearance) was also investigated.

2 Case Study: Measurements & Simulation

2.1 Measurements

The compressor (see Figure 1) is a horizontal balanced-opposed model with four double-acting cylinders; compression takes place in two intercooled stages. Two equal cylinders (90° phased) are dedicated to each stage with common suction and discharge dampers. Table 1 shows the main compressor data.

Table 1: Main Compressor Data

Shaft speed	490	RPM	Suction Pressure	2.89	bar-a
Motor nominal power	800	kW	Discharge Pressure	19.89	bar-a
Cylinders phasing	90	°	1° Stage compression ratio	2.56	-
Processed gas	Air	-	2° Stage compression ratio	3.06	-

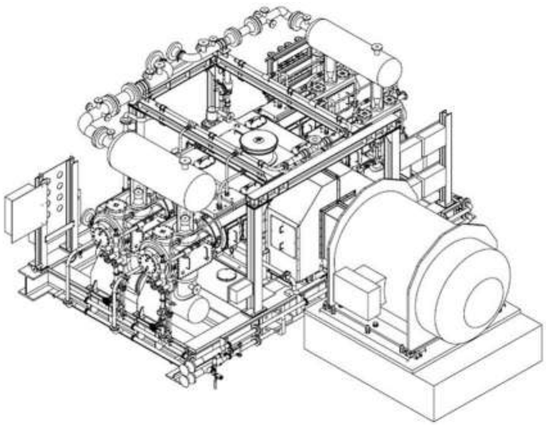

Figure 1: Compression skid

2.1.1 Pulsation Measurements

There are twenty pressure measurement points: eight in the cylinders (two for each cylinder) and twelve in the piping (one for each cylinder flange and one for each line side volume bottle connection). These points allow API pulsation requirements check.
The measurement chain includes a piezoresistive transducer, probe assembly and analyser.
The pulsation analysis performed during the design phase was repeated for the actual test conditions after the measurements, in order to get a good comparison between the model and the measurements. Figure 2 shows the analyser used for the test campaign.

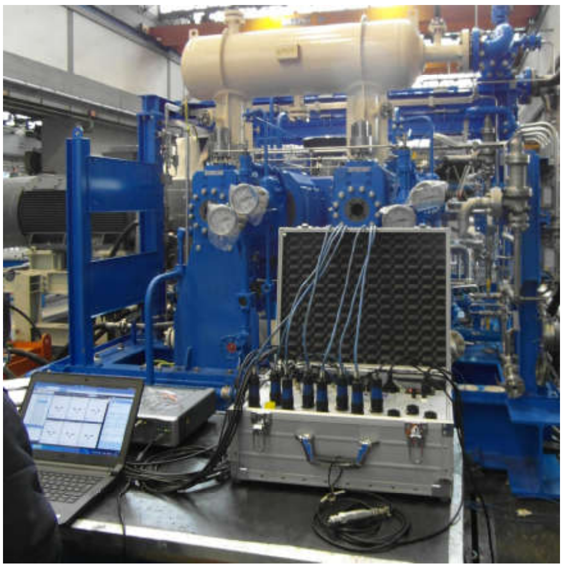

Figure 2: Analyser

2.1.2 Vibration Measurements

Vibration measurements have been carried out on cylinders, dampers and piping most significant points by means of an accelerometer (Figure 3).

Figure 3: Accelerometer for vibration diagnostic

2.2 Modelling and simulation

2.2.1 Acoustical analysis

The model used to simulate the plant was developed by CST, based on the multi-physics software AmeSim.

Figs. 4, 5, 6 and 7 show the main elements of the model and the related connections; respectively:

- the double-acting cylinder, with the piston chamber and piston, driven by the crank-mechanism (to be connected to suction and discharge valves),
- the suction/discharge cylinder valves: check valves connected to the plenum and to the cylinder, comprehensive of the dynamic of the shutter (the moving element of the valve) considering springs, mass, and damping.
- the suction and discharge cylinder plena, connecting the valves of the head and crank ends (HE and CE) to the related volume bottle (in the case shown, each cylinder has just one suction and one discharge valve for each end, therefore only two suction and two discharge valves in total),
- the pulsation damper (which is commonly a volume bottle) modelled like a header connecting the cylinder HE and CE plenum and the pipeline.

Plenum and damper complex geometries are modelled as a sequence of mono-dimensional pipes.

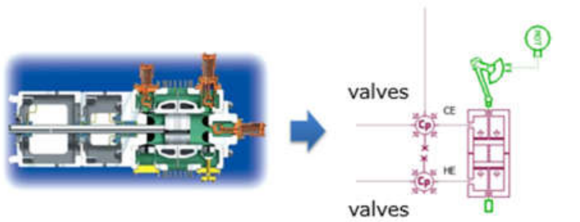

Figure 4: Double Acting Cylinder

Figure 5: Cylinder Valve

Figure 6: Cylinder Plenum

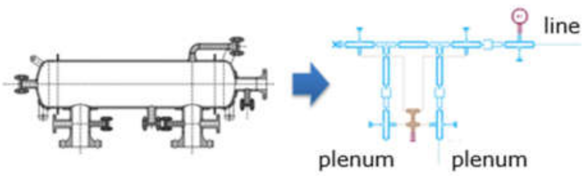

Figure 7: Pulsation Damper

The simulation is solved in the time domain; several cycles are computed starting from initial conditions up to steady-state: only the last 80 cycles were considered for the post-process.
The test was performed on a closed circuit equipped with a throttling valve and an additional off-skid damper between final discharge and first stage suction.
The boundaries of the model consist in two anechoic endings, positioned up- and down-stream the throttling valve.
The model consists of mono-dimensional CFD models for piping, dampers and plenums, and a lumped parameters model for cylinders (variable volume chambers generating the flow pulses). The flow dynamics in the ducts relies on conservation equations (mass, momentum and energy) solved with finite differences using the Lax-Wendroff Method (second-order accuracy in space and time) for internal points and the Method of Characteristics for external points. If compared with simulations based on acoustic wave equation, this method provides a better accuracy because it can take into account cylinder-plant mutual interaction, the non-linear effects of viscosity, the heat exchange and the real behavior of valves.

2.2.2 Mechanical analysis

The finite elements model of the compressor unit includes:

- 1st and 2nd stages suction and discharge cylinders,
- Distance pieces,
- Compressor frame including crossheads slides,
- 1st and 2nd stages suction and discharge dampers,
- Suction and discharge piping,
- Support structure.

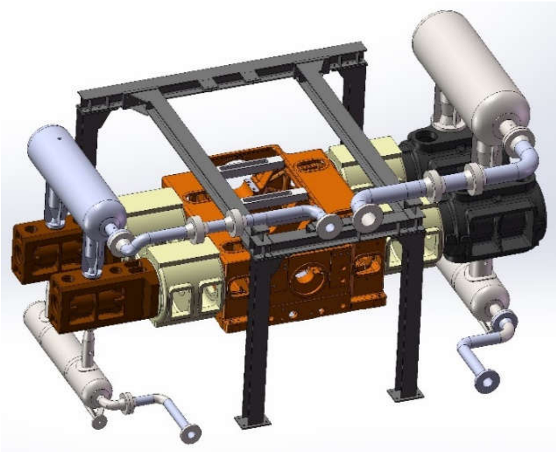

Figure 8: 3D model

The forced mechanical response analysis has been set up imposing dynamic forces on the 3d model. Dynamic loads have been considered on cylinders, crossheads slides and main bearings houses, in order to take into account gas forces and inertia, whilst the shaking forces (direction, amplitude, frequency and phase) computed with the previous acoustical simulation have been imposed on piping and dampers. Spectra of dynamic gas and inertia loads can be seen in Figures 9 and 10.

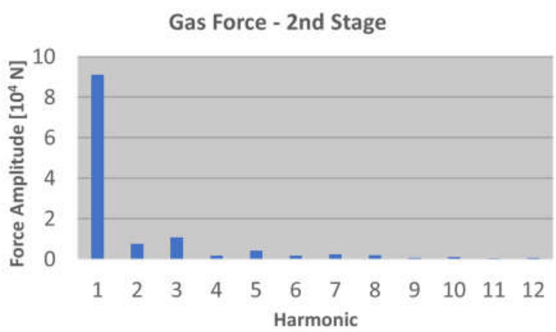

Figure 9: 2nd stage cylinder gas dynamic loads

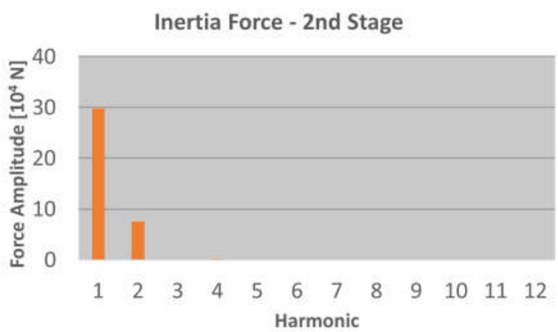

Figure 10: 2nd stage inertia dynamic loads

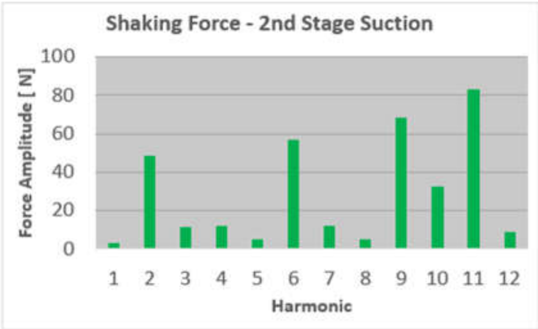

Figure 11: 2nd Stage Suction Damper shaking forces

3 Analysis of results

3.1 Results of Pulsation Analysis

A correct calculation should consider many precautions. Here we want to point out that a good analysis of the field data could require reduction techniques to take into account phenomena affecting the measurement.

3.1.1 Resonance in pressure transmitter connection

Figure 12 shows the PV-cycle of the CE of one of the 2nd stage cylinders, with the calculated and measured (unfiltered) PV-cycle.

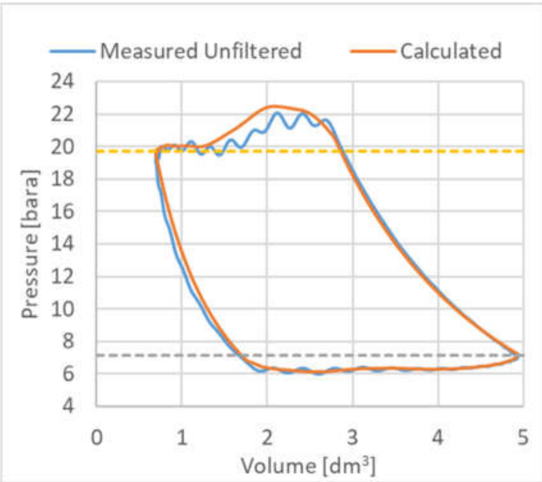

Figure 12: PV-cycle 2nd Stage Cylinder CE – measured unfiltered vs. calculated

Anomalous oscillations at high frequency can be noticed in the measured cycle, but not in the calculated one. The cause of these oscillations was found in the acoustic resonance occurring in the internal connection of the probes. In fact, according to the acoustic theory, this connection can be schematized as an Open-Closed pipe, the length of which is related to a 1st natural frequency of approx. 350 Hz (as shown in Table 2).

Table 2: Acoustic resonance in probe connection

Harmonic order	Frequency Hz	O-C pipe resonant length m
37	302.17	0.331
…	…	…
46	375.76	0.266

After filtering the measured pressure in order to eliminate the above-mentioned effect, the PV-cycle shown in Figure 13 is obtained, showing better consistency with the calculated one.

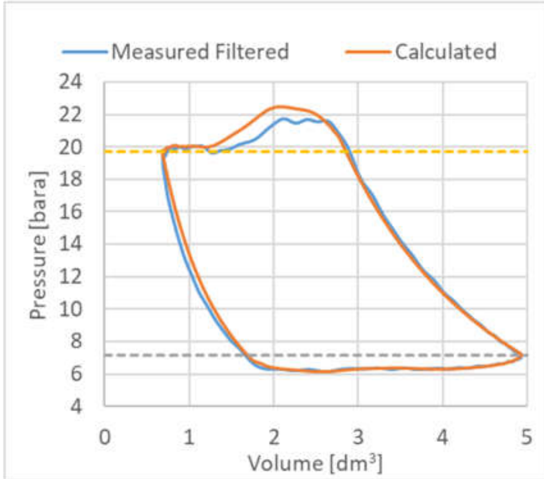

Figure 13: PV-cycle 2nd Stage Cylinder CE – measured filtered vs. calculated

3.1.2 Electrical noise (50 Hz)

The bar chart in Figure 14 shows the calculated and measured peak-to-peak pulsation amplitude for the 6th harmonic at the twelve piping measuring points. The blue bars refer to the measured values obtained with a first post-process method, performing the Fast Fourier Transformation (FFT) on each cycle and then making the average, with a frequency resolution of 8.16 Hz. The green bars show the measured values obtained with a second post-process method, performing the FFT on the entire data sample with a frequency resolution of 0.1 Hz.

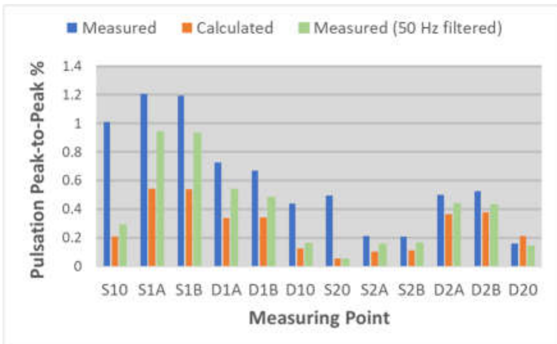

Figure 14: Pulsation amplitude in the measuring points: 6th Harmonic

The difference between measured values post-processed with the first method and the calculated ones is due to the presence of interference coming from the electrical measurement apparatus (50 Hz). The noise was not distinguished from the 6th harmonic in the first method, but clearly identified with the second one (see also Figure 15 showing the spectrum obtained from the measurements post-processed with higher resolution).

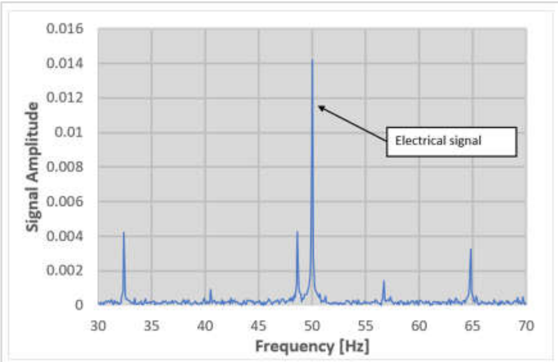

Figure 15: Measured pulsation amplitude spectrum

As can be seen the higher resolution, which makes it possible to distinguish the electrical noise contribution from the pulsation, allows us to get results much closer to the calculated values.

3.1.3 Heat exchanger modelling improvement

The heat exchanger response, in terms of pulsation transmission, is strongly dependent on its pressure drop (other than its internal geometry); so, it must be correctly taken into account. The bar chart in Figure 16 shows a comparison between the spectrum of the pulsation downstream the heat exchanger, calculated on the basis of the actual pressure loss (green bars) and of the value resulting from data sheets (red bars). It can be noted the big difference between the two simulations.

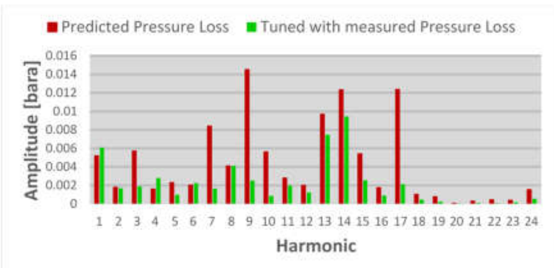

Figure 16: Effect of heat exchanger model on pulsation

Concluding the analysis of the results of the case study, we can see bar charts of calculated and measured pulsation values (peak-to peak percentage for each harmonic) at the 1st Stage Suction Damper nozzle line side (Figure 17). The measured values are in accordance with the calculated ones.

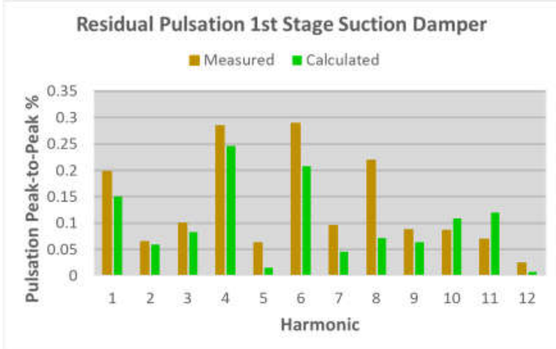

Figure 17: Residual Pulsation @ 1st Stage Suction Damper Nozzle (line side)

3.2 Results of Mechanical Analysis

The mechanical simulation results in terms of vibration velocity have been compared with the vibration measurements. The results of this benchmark have been summarized in Table 3.

Table 3: Vibration velocity values benchmark

VIBRATION VELOCITY BENCHMARK		
Component	Measurements	Simulation
	Vres [mm/s]	
1st Stage Cylinder	3.9	3.1
2nd Stage Cylinder	9.0	6.9
1st Stage Suction Damper	6.7	6.2
1st Stage Discharge Damper	3.6	2.5
2nd Stage Suction Damper	10.8	8.2
2nd Stage Discharge Damper	3.8	4.1

As it can be seen, there is a good match between the vibration velocity values found with the dynamic simulation and the measured values. More specifically, the estimated values are found to be on average lower than those found during measurements. This trend can be due to relative movement which may occur in actual foundation bolts tightening. In the simulation the two components are forced not to have relative movement instead, as it would happen for an ideal bolting. Spectra of measured and calculated vibration velocity for 1st stage cylinder are shown in figures 18, 19, 20.

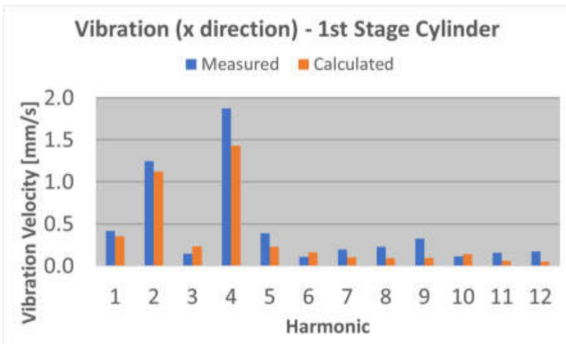

Figure 18: 1st stage cylinder vibration in compressor shaft axis direction (x)

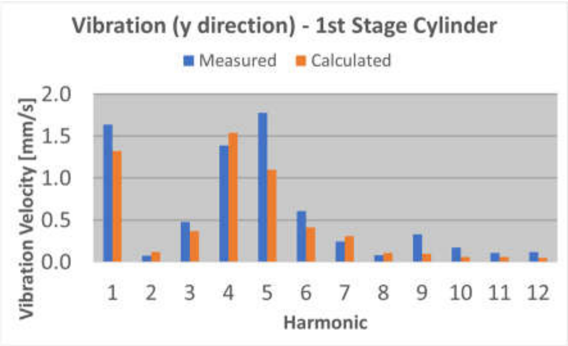

Figure 19: 1st stage cylinder vibration in vertical direction (y)

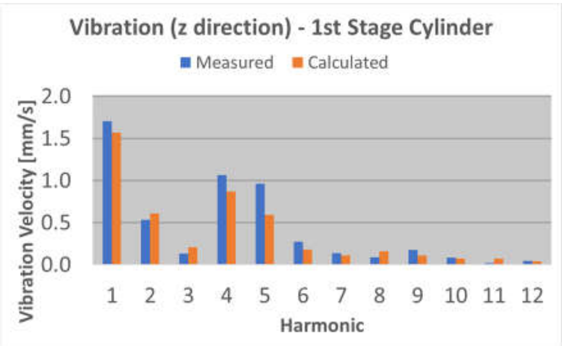

Figure 20: 1st stage cylinder vibration in cylinder axis direction (z)

4 Comparison of Capacity Control Techniques

The capacity control system has a significant effect on the pulsation generated by the compressor, since reduced capacity operation very often generates more critical conditions than at full load, producing a higher pulsation amplitude and/or changing the damper sizing harmonic order. The bar chart in Figure 21 compares the residual pulsation calculated at line side nozzle of the 1st stage suction damper at full load (100% capacity with all four cylinder ends acting – grey bars) and at 50% load obtained in two different ways:

- 50% with unloaded cylinder ends (only the two CE acting – red bars)
- 50% with reverse flow devices acting on all suction cylinder valves (yellow bars).

With unloaded ends, the 1st harmonic is dramatically higher than at full load and represents the sizing condition for the suction damper. In fact, considering API 618 Design Approach 2 pulsation level limit, the full load condition would not require a damper, while 50% load (by means of unloading effects) requires a 200-litre damper.

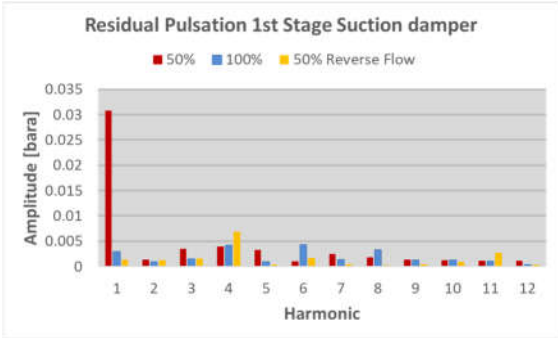

Figure 21: Residual Pulsation @ 1° Stage Suction Damper Nozzle line side

With a reverse flow control system, the harmonic content of the significant frequencies is similar to the full load one, and the 1st harmonic is much smaller; consequently, the damper volume can be much smaller, thus resulting in less costs and less oscillating mass, therefore higher vibration frequency for the cylinder header.

Another way to control the capacity is to use additional clearance pockets on the HE of the cylinders. The diagrams in figs. 22 and 23 show respectively the volume required for 1st stage suction and discharge volume bottles vs load condition with the three different systems. Unloading ends is the most critical situation for both suction and discharge volume bottles.

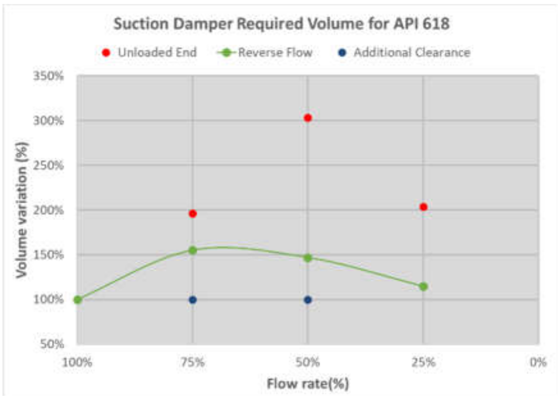

Figure 22: Volume required for suction damper according to API 618 preliminary sizing

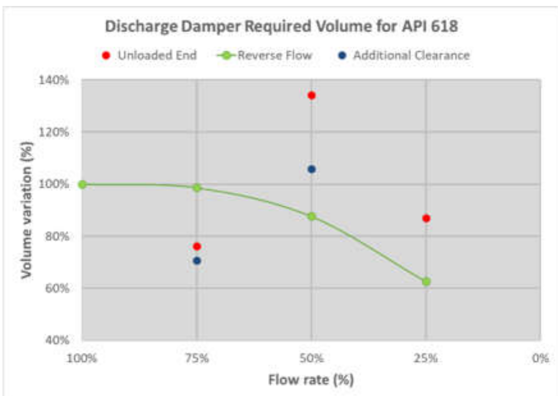

Figure 23: Volume required for discharge damper according to API 618 preliminary sizing

5 Conclusions

The model used for Pulsation Analysis can get the best accuracy, but on the other hand, it is elaborate and requires a large amount of input data. Therefore, the information regarding the actual setup has to be clear and complete. Moreover, API acceptance criteria provide limit for pulsation amplitude at each pulsation damper connection line side and the limit is function of the frequency, so several measurement points are to be taken and all the values have to be carefully post-processed with FFT to get the pulsation spectra. All these values can be affected by interferences which complicate the possibilities to achieve a comprehensive consistency between test and simulation.

The work constitutes an important lesson learned allowing the OEM to correctly analyze the pulsation values measured on field, to better evaluate and optimize the effectiveness of the countermeasures to be taken in order to reduce pulsations and consequent vibrations (i.e. pulsation dampers, pipe geometries and supports).

In particular, some reduction techniques are shown, which must be used in Pulsation Analysis to take into account phenomena affecting the model/measurement:

- resonances in the sensor duct
- electrical noise superimposed to the pulsation signal

It has also been shown the big influence of the capacity control system selected on the volume of dampers required.

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