Pulsation Suppression Device Design for Reciprocating Compressor

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Abstract**—**Design and evaluation of reciprocating compressors should include a pulsation study. The object is to ensure that predicted pulsation levels meet guidelines to limit vibration, shaking forces, noise, associated pressure drops, horsepower losses and fabrication cost and time to acceptable levels. This paper explains procedures and recommendations to select and size pulsation suppression devices to obtain optimum arrangement in terms of pulsation, vibration, shaking forces, performance, reliability, safety, operation, maintenance and commercial conditions. Model and advanced formulations for pulsation study are presented. The effect of the full fluid dynamic model on the prediction of pulsation waves and resulting frequency spectrum distributions are discussed. Advanced and optimum methods of controlling pulsations are highlighted. Useful recommendations and guidelines for pulsation control, piping pulsation analysis, pulsation vessel design, shaking forces, low pressure drop orifices, pulsation study report and devices to mitigate pulsation and shaking problems are discussed.

Keywords**—**Pulsation, Reciprocating Compressor.

I. Introduction

RECIPROCATING compressors need a suitable design of pulsation control devices in order to limit the pulsation, vibration and shaking forces in compressor package as well as plant piping and facilities in downstream and upstream of machine [1].

Reciprocating compressors are the most common type of compressors found in industrial applications [2]–[4]. Worldwide installed reciprocating compressor horsepower is approximately two times that of centrifugal compressors and maintenance costs of reciprocating compressors are approximately three times greater than those for centrifugal compressors [5]. Reciprocating compressors are most efficient and most flexible available compressor type [2]–[4].

Pulsation suppression device for reciprocating compressor have to comply with optimum selection and arrangement as a function of application to minimize initial investment, operating costs, reliability of reciprocating compressor package and whole system connected to it.

Previous approaches for pulsation study mainly using transient acoustic equations with numerical solver. Previous reports and papers generally did not attempt to solve the actual fluid dynamic equations of pulsation [6]. Previous approaches mainly solved a linear and simplified set of equations derived from the original fluid dynamic equations. These equations were greatly simplified to require no accurate flow, infinite small pressure pulsations, no viscosity and quasi-constant gas properties in order to obtain a fast solution for pulsation problem. These assumptions are severely limit the previous method ability to correctly predict pulsation behavior of reciprocating machines and associated facilities [6].

This paper presents an effective method for solution of most advanced and general form of fluid dynamic equations to study of pulsation and shaking force in reciprocating compressor systems. Detailed modeling, simulations and investigation of pulsation and vibration of all facilities under pulsation excitation shall be performed to make sure about reliability of reciprocating compressor package and all related facilities and piping.

II. Definition of Problem

Pressure pulsation comes from the discontinued nature of gas flow in reciprocating compressor. Pulsating actions of piston to compress gas generates fundamental pulsation that its frequency corresponds to the compressor speed of rotation. Cylinder valves and other accessories generate various harmonics of fundamental pulsation (frequencies are multiple of fundamental frequency). Machine generated pulsations propagate in the gas as waves. These pulsations interact (for example due to reflection) with plant facilities and piping in upstream and downstream of compressor. The level of harmonic components can be increased considerably due to resonance with plant installations (resonance between harmonic frequency and natural frequency of piping or facilities in plant).

Pulsation studies and reviews shall be performed for all operating conditions as well as all transient operating cases and combinations of pressures, speeds and load steps. Pulsation can also alter the timing of the valve motion and decrease efficiency and reliability [7].

Pulsation study and pulsation device design have great effects on reciprocating compressor efficiency, operation and reliability. Improving efficiency and reducing the total cost (investment and operation costs) of reciprocating compressor package can be accomplished by optimum pulsation device arrangement. Areas that can result in significant savings and improvements are: 1- Pulsation control devices introduce pressure drop into the system. Optimum pulsation design

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results in system lower pressure drop. It can realize a significant financial reward by increase capacity and reduce power consumption. 2- Avoid overly conservative pulsation control design. Optimum design can reduce initial investment, fabrication time and cost. 3- Evaluating the pulsation control system performance at current and future operating conditions.

The standing wave pattern of the pressure pulsation carries from the cylinder valves through the cylinder gas pressures and the cylinder nozzle into the pulsation vessels (bottles), on each side of the cylinder. Shaking forces resulted form pulsation effects are sufficient to cause the problems such as valve problem, piping vibration, fatigue failure, break cylinder support, break bottle support, break anchor bolt and grout especially under crosshead guide [8], [9]. Pulsation study is a necessary step to assure reliability and safety of reciprocating compressor and all piping and facilities upstream and downstream of compressor.

III. Model

Distinguishing pulsation generated by reciprocating compressor as distinctly different from acoustic waves will result in a more accurate depiction of the pulsation amplitudes and the response of the piping.

As noted, previous approaches solved a linear set of equations derived from the original fluid dynamic equations. These equations were greatly simplified. These assumptions are severely limit the previous methods ability to correctly predict pulsation behavior. In this paper full fluid dynamic model is presented. This general form model and formulations are used to predict pulsation pressures, pulsation wave form and resulting frequency spectrum distributions in upstream and downstream network connected to reciprocating compressor.

In order to overcome the limitations of the acoustic solution and more accurately describe the pulsation flow field, a full one-dimensional time domain flow equations and solutions, applicable to any complex flow network (piping, manifold, facilities, etc) are presented. This formulation respects all terms of the governing equations including fluid inertia, diffusion, viscosity and energy dissipation. Presented method is used to solve pulsating flow in a heavy duty high power special purpose compressor system.

The state of gas in reciprocating compressor and attached piping in upstream and downstream depends on two factors 1- The kinematics of reciprocating compressor machine in terms of actual pulses generated by the piston and release through cylinder valves. It provides pulsation pressure forcing to system. 2- The fluid dynamic behavior (response) of the piping, manifolds and facilities and passive outlet conditions.

Presented model and algorithms provide superior accuracy over the older methods. This is necessary to assess overall pressure drop and performance throughout the system. Parameters used in pulsation model and formulations are as follows:

c Speed of Sound in Compressed Fluid.
D_e Equivalent Section Diameter (Pipe or Equipment).
f_μ Friction Factor (Viscosity Friction).
p Instantaneous Pressure.
t Time.
v Instantaneous Speed.
ρ Density of Fluid.
x Coordinate Along Axis.
μ_s Viscosity.
$[Z]$ Pulsation Impedance Matrix.
$\{p\}$ Pulsation Pressure Vector.
$\{q\}$ Flow Vector.

IV. Formulations

In order to calculate the velocity and the pressure at every point in time in system the transient one dimensional ´´Navier-Stokes´´ equations are used. Based on ´´Navier Stokes Fluid Model´´ [6], fluid equations can be expressed as (1) and (2) respectively.

$$\rho\left(\frac{\partial v}{\partial t}+v\frac{\partial v}{\partial x}\right)=-\frac{\partial p}{\partial x}+\mu_s\frac{\partial^2 v}{\partial x^2} \tag{1}$$

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho v_x)}{\partial x}=0 \tag{2}$$

The viscosity "μ_s" is the combined viscosity and turbulent eddy viscosity which is usually determined using a second order Reynolds number based on turbulent model [6]. Fluid equation motion and continuity equation can be introduced as (3) and (4).

$$\frac{1}{\rho}\frac{\delta p}{\delta x}+v\frac{\delta v}{\delta x}+\frac{\delta v}{\delta t}+f_\mu\frac{v|v|}{D_e}=0 \tag{3}$$

$$v\frac{\delta p}{\delta x}+\frac{\delta v}{\delta t}+\rho c^2\frac{\delta v}{\delta x}=0 \tag{4}$$

With respect to presented formulations (3) and (4), pulsation in gas network (compressor and related facilities) can be analyzed using "Transfer Matrix Method". For this purpose pulsation impedance (matrix $[Z]$) is introduced as complex ratio between pulsation pressures (vector $\{p\}$) and flow (vector $\{q\}$) as (5) and (6) respectively. Subscripts "u" and "d" are for upstream and downstream of machine. All plant facilities and piping can be described with presented method using matrix transfer. In this way an overall transfer matrix can be developed for the whole unit under study (including reciprocating compressor package and upstream and downstream facilities and piping). These equations can be solved for each frequency (harmonic) of pulsation excitations from machine. Pressure pulsations at any point of plant can be computed using these harmonics.

$$[Z]\{q\} = \{p\} \tag{5}$$

$$\begin{bmatrix} Z_{11} & Z_{12} \\ Z_{21} & Z_{22} \end{bmatrix} \begin{Bmatrix} q_u \\ q_d \end{Bmatrix} = \begin{Bmatrix} p_u \\ p_d \end{Bmatrix} \tag{6}$$

Obtained numerical results can be presented and studied in following two forms: 1- Pulsation pressure vs. time. 2- Spectra Analysis (mainly in form of harmonic composition). First form has global information and total pulsation effects of all harmonics. Second form gives information about single harmonic.

V. Pulsation Dampener Vessel

Pulsation vessels are the most efficient way to reduce the pressure pulsation, shaking force and vibration. At the same time these pulsation vessels are in the vicinity of reciprocating compressors and can be subjected to high vibrations and shaking forces. API 618 requires top to bottom flow to cylinder to avoid liquid accumulation. Discharge vessels are generally at bottom side of cylinder and can be better supported and clamped. Suction dampeners are located at top and need more attention for support design [1].

The first step in pulsation study is to determine pulsation vessel sizes and type. Three common pulsation dampener types are: 1- Volume bottle without internal. 2- Dampener with choke tube (one chamber). 3- Dampener with choke tube and baffle (two chambers) [1].

To select each type, optimization process shall be done with respect to operating conditions, performance (such as pressure drop, etc) and commercial terms (vessel size, fabrication, available space, etc). Optimum solution shall be found considering compromised pulsation limit, fabrication, space limitations, and all other operation and delivery factors.

Good indication for selection of dampener vessel type is ratio between frequency of compressor pressure pulsation and natural frequency of dampener. Frequency of compressor generated pressure pulsation depends on compressor machine characteristics mainly compressor speed, single or double acting mode and crankshaft arrangement. Natural frequency of dampener depends on gas sound velocity, size and length of vessel, choke tube, volume, etc. When this ratio (frequency of compressor pressure pulsation / natural frequency of dampener) is below "1.4", dampener without internal seems more efficient. For higher ratios, dampener with choke tube may become more efficient. Based on experience most effective arrangement is often obtained using vessels without internals (without choke tube or baffle or similar). This solution (vessels without internals) has minimum pressure drops and avoid possible mechanical problems associated with internals (such as choke tube or baffle or similar) [1]. Generally low pass acoustic filters only may be used when operating conditions are expected to be fixed for many years. In case of compressor package revamping or change of operating condition (for example change of process gas), the efficiency of these devices will drastically decrease. In those situations low pass acoustic filters are not effective, just producing excessive pressure drops and it is necessary to replace these devices [1]. Low pass acoustic filter devices may be effective for specific cases but they are less flexible than the bottles without internals to cover operating condition changes.

Geometry of pulsation vessel should also be taken into account. Lower length to diameter (*L/D*) ratio results in more effective pulsation control effects. However a reasonable (*L/D*) ratio should be selected for fabrication and arrangement reasons. Average (*L/D*) ratio 3 to 4 is recommended. Ratio (*L/D*) lower than 5 is necessary to avoid drastically efficiency reduction [1].

Specific code(s) have to be applied (such as ASME, PED, etc) for pulsation vessels with respect to final destination country where compressor package will be installed. For all applications, it is strongly recommended to follow ASME Section VIII Division 1 for design of vessels. For equipment which will be installed in Europe Union, CE marking is also mandatory. ASME U-stamp is strongly recommended for all applications with design pressure more than 100 Barg. However this requirement (ASME U-stamp) may be mandatory for all pressures and vessels with respect to Client specification and destination country standards.

Followings are some engineering practices for pulsation vessels: 1- For all connection WN (Welding Neck) shall be used. Especially for small connection, integrally forged LWN (Long Welding Neck) shall be used. 2- All pressure and integral parts shall be full penetration welds. 3- For critical applications, nozzle connection with counter suitable for butt weld and RT/XR test shall be used.

The low cycle fatigue analysis can be carried out following the procedure described by ASME VIII Division 2 Appendix 5. Adequate stress concentration factors and proper safety factors shall be respected. These factors can decrease cyclic stress limit to around 25 Mpa peak to peak (around eight times less than API cyclic stress limit).

VI. Piping Pulsation and Vibration

To avoid resonance and excessive vibration, pulsation and vibration analysis of the compressor package with upstream and downstream piping is needed to verify design. Modifications of the piping system may be necessary based on this analysis. The force response is required if the pulsation and mechanical design do not meet the required guidelines.

During the early stage of piping layout design, following criteria are recommended to minimize the pulsation and shaking forces: 1- Knock out drum / Separator close to compressor. 2- Minimizing point of shaking forces generation in equipment and piping such as elbow, tee, size variation, etc. 3- Locating normally closed valve on branch lines close to the branch point. 4- Control maximum gas velocity (below 30 m/s [1], common speed range is 10 to 20 m/s, speeds for suction are higher, around 15 to 20 m/s, and for discharge lower around 10 to 15 m/s) to avoid gas turbulence / vortex, high pressure drop, noise, etc.

The maximum piping span between two consecutive supports can be calculated to control resonances. In the other words, it is necessary to achieve minimum piping natural frequencies and to keep mechanical natural frequencies out of the range of excitation components. In addition support structure design shall be according to calculated maximum shaking forces obtained from pulsation studies. Piping shall be design as close to ground as possible. Extra supports shall be provided near concentrated masses such as elbows, tees, valves, etc. For fix equipment such as filters, separators, etc, skirt support is strongly preferred. Leg type supports [1] or other relatively flexible supports shall be avoided.

Support modification (generally addition or shift of support and sometimes removal of support) may be required to achieve separation of frequency and excessive vibration and cyclic stress level. The results of pulsation study indicate the support modifications and reaction forces at supports. These reaction forces shall be used to check and verify the design of support.

VII. Pulsation Shaking Forces

Reduction of pressure pulsation (for example using larger pulsation vessels) can be accompanied by an increase in shaking forces (or unbalanced forces) [8], [9]. Shaking forces shall be determined and controlled. Piping, pulsation vessels and all other facilities upstream and downstream of reciprocating compressor shall properly be supported. The margin of separation between the mechanical natural frequency (MNF) of system (including piping and bottles) and excitation frequency is 20%. Also MNF shall be greater than 2.4 times of maximum run speed [8], [10]. If not meet limits, the force response (including stress analysis) is required. The cylinder gas forces (also called frame stretch or cylinder stretch force) can be significant source of excitation (can cause high frequency vibration on the bottles and piping close to the compressor) and lead to excessive pulsation bottle and piping vibration even if the pulsation shaking forces meet limits. Flow induced pulsation is rarely seen [9]. API 618 Design Approach 3 and less rigorous analysis, to control pulsation and shaking (unbalanced) force levels and avoiding mechanical resonance can result in an optimized design [9].

As noted, a force response analysis may require if the results of the pulsation analysis and the mechanical analysis (MNF model) do not meet guidelines. Based on experience, mechanical forced response study is also required for medium to high speed units when power per cylinder is more than ′550 KW′, or rod load exceed ′80%′ rated rod load, wide speed range operation (more than ′25%′ of rated), compression ratio below ′1.7′ or critical application (remote location, high availability required, etc). Piping system forced response study is usually not required for standard compressor package because pulsation design will typically reduce pulsation forces to low levels and mechanical analysis will avoid resonance at the first and second order of compressor speed where the highest pulsation energy typically occurs. It is required if optimizing the system design. For example determining optimum bottle sizes for multi unit, trade off in piping support and piping layout design around coolers to meet MNF guideline and thermal expansion, or existing facilities where making changes is costly [8].

As explained, pressure pulsation of gas produces vibration and shaking forces in vessels, coolers, piping and other facilities. For dampeners (pulsation vessels) best practice to reduce vibration is to make symmetric inlet and outlet connections in order to minimize the shaking forces. For example cylinder connection can be routed to center of pulsation bottle. Furthermore the length of piping between cylinder and bottle has to be limited to the minimum possible, because longer it is, more harmonic components may resonate. Generally there may be fewer problems for very light gases (such as hydrogen). It is because gas sound velocity is relatively high and no resonance is foreseen for relatively short connections. However as some of light gas compressors (such as hydrogen compressors) are designed for alternative gas (nitrogen) for alternative operation mode (such as start up), special care must be taken for this issue. In some cases, due to space limitation or compact design, connection to centre of vessel can not be routed outside of vessel. In such cases internal piping shall be used. This internal piping has no effect on the damping behavior. This should not be confused with internals such as baffle, choke tube, etc which have completely different damping effects. This internal piping is rigidly connected to the vessel (welded on shell periphery) and consequently is not critical from mechanical design point of view [1].

The design to reduce shaking forces should meet two key parameters. 1- Separation margin between MNF and the shaking force or excitation frequency. Predicted MNF shall be separated from significant excitation frequencies by 20%. 2- Minimum MNF of any element in the system shall be bigger than 2.4 times of maximum run speed. Acoustic shaking forces shall not exceed the limits based upon the calculated effective static stiffness and the design vibration guidelines. API 618 [10] defines a method for estimating the effective stiffness of the piping and the bottle. As noted if the separation margin or shaking force criteria cannot be met, a force response analysis of the compressor mechanical model must be conducted. The analysis is to include the pulsation shaking forces and cylinder gas forces. The design must meet the allowable cyclic stress criteria [8].

Piping system review may include all piping in the pulsation analysis but is generally limited to specific areas where the forces exceed the guideline or the mechanical design criteria cannot be met. This piping detailed analysis is recommended for new installations, correction of existing installations, design optimization studies where analyses have been competing requirements such as elevated piping around air coolers, hot discharge piping, etc [8].

Moving the MNF above 2.4 times of run speed will avoid resonance problems. For slow and medium speed units, the minimum MNF guideline is not a difficult design problem. For high speed machines (1200-1800 rpm), however, the design becomes much more challenging. In general, piping or vessels must be much stiffer than in slower machine units. Early communication between all parties is necessary to ensure this requirement.

In case of cylinder stretch force excitation calculation, first step is to calculate these forces. The second step is evaluating the mode shapes calculated in the mechanical design. It may be very difficult to meet the 20% separation margin criterion in some cases, such as variable speed compressors. The cylinder stretch forces are constant for each order of compressor speed, that is, the forces have fixed amplitude over a wide frequency range. A mechanical modal analysis is required in these cases. A risk analysis can be done in early of design stage to identify when these studies are likely required [8].

Fabrication practice such as installation and mounting details to the skid (or foundation) are very important to provide the required stiffness of components. Skid (or package) drawings and mounting details must be available at the time of the mechanical analysis. As vessel design (length vs. diameter) vs. mounting design can dramatically affect MNFs, vessel fabrication drawings must also be available at this time. Accurate Finite Element Analysis (FEA) modeling techniques that closely match the real MNFs must be used. Shortcuts in modeling create high risk. Modeling techniques must be field verified (for example in site performance test). In some cases, it may be impractical to reach these high frequencies. Inter-tuning can be studied as last option. Early discussion on mechanical design options are recommended with respect to amount of vibration risk and to avoid costly changes later [8]. For vertical vessels the most important factor for vibration and shaking forces is the supporting assumption (between the vessel and the skid or foundation). Model shall be included base details, mounting plate, bolts, beams, and local skid construction. Simplistic models those assume a rigid base or generic estimate of stiffness, may be very risky. Beware of models with rigid support or anchor or assumed stiffness. These have been proven to be inaccurate and shall be avoided. Case studies are available illustrating that more than 15% error is associated with simplistic models [8]. High error can mean high vibration and failure, or excessive costs (conservative mechanical design).

For specific high pressure and high power application, it is recommended to investigate a hypothetic plant arrangement including resonating lengths to evaluate the consequent shaking forces and assure appropriate damping of design. This simulation can assure proper damping and acceptable pulsation and vibration even in case of resonance. It can drastically reduce the impact of problem and piping changes for shaking forces limits during final pulsation study.

VIII. Low Pressure Drop Strategically Located Orifice

Optimum pulsation reduction techniques trend to dissipate less energy than reliance on special solutions such as orifices to control pulsation levels [11]–[13]. However low pressure drop orifices may be necessary in some applications. These orifices shall be located in strategic points to have highest pulsation reduction effects and lowest pressure drops. In some cases the most effective solution is low pressure drop orifice located at strategically selected positions (points with maximum velocity in the standing wave field of resonance and optimum effects of damping – maximum pulsation reduction effects with minimum pressure drops). In this design the efficiency of the damping is amplified by the instantaneous flow pulsation. The position can be obtained from trial and error methods to damp as many harmonic components as possible.

For some applications, insertion of orifices allows to keep required pulsation limit and pulsation vessel (dampener) fabrication schedule without need to wait for the accurate pulsation study performed. In addition this approach allows meeting contract delivery without need for precise piping plant (upstream and downstream of compressor package) in early stage of project. For some other cases damping of the resonance is necessary to avoid excessive shaking forces [14] inside the vessel and avoid damage in cylinder valves. Orifice plates in the throat of flanged inlet nozzle connections are one of effective tools to reduce the amplitude of acoustic resonance presented between the cylinder and the volume bottles. For most effective operation these orifices must be located exactly at the outlet for suction (outlet flange of suction pulsation vessel connected to piping of cylinder suction port) and at inlet for discharge of the volume bottles (inlet point of discharge flow to discharge pulsation vessel). It is because the length of section in which resonance is generated is very short and pulsations can vary suddenly in these sections. At explained strategically locations, for example throat flanged inlet nozzle, orifice results in a relatively low pressure drop and effective control of resonances and shaking forces.

Preliminary and rough predicted pressure drop values for pulsation and inter-stage facilities are as follow: pulsation devices (total pressure drop for pulsation vessels, internals, orifices, etc): around 1% pressure, intercooler: around 0.70 bar. The use of orifice plates, especially on high-speed single-act machines, can contribute to significant pressure drops [2]. This kind of high pressure drop orifices shall be avoided. Based on experience, correct design and proper location selection for low pressure drop orifices will result in around 0.05%-0.2% pressure drop (commonly 0.1% pressure drop). This figure can be acceptable. Based on experience using orifice requires "Orifice Justification Report" to make sure about necessity of orifice application and proper design of orifice type and location.

IX. Results and Discussions

Analytical results are presented for heavy duty special purpose four throw three cylinder (double acting cylinder) hydrogen reciprocating compressors. These compressors are heart of hydro-cracker unit. Train speed is 327 rpm. Each train has three stages. Four step capacity controls, 0-50-75-100%, using fixed volume pocket at cylinder head end and suction valve unloaders are provided. Total motor power of each train is 6.5 MW (around 2.1 MW per working throw-cylinder, it is a high risk and high power machine). Driver is direct coupled eighteen (18) poles induction electric motor.

Table I shows selected optimum pressure pulsation limit values at vessel outlet flange (plant side) as percentage of ´API 618 – Approach 3´ limit values for design stage. These values are selected based on detailed calculation and optimization processes performed based on presented pulsation study method with respect to performance, reliability, safety and commercial conditions. Generally for special purpose units it can be recommended to select pulsation limits in design stage around 95%-85% of API 618 (Approach 3) limits to have some margin (around 5-15%) to mitigate risk during construction and installation periods as well as unpredicted deviations and problems. Generally first stage suction and last stage discharge need lower limits compare to inter-stage pulsation limits.

TABLE I
SELECTED OPTIMUM PRESSURE PULSATION LIMIT VALUES (PLANT SIDE) AS PERCENTAGE OF API 618 – APPROACH 3 LIMIT VALUES

	Suction Pulsation Limit as % of API618 Appr.3 Limit	Discharge Pulsation Limit as % of API618 Appr.3 Limit
Stage 1	85%	91%
Stage 2	88%	95%
Stage 3	97%	90%

Table II shows pressure pulsation at discharge of reciprocating compressor package in three different operating situations. Full flow, half flow (50% flow) and operation with nitrogen during start up. This table presents that different operating situations have completely different pulsation values and behaviors. Simulation results show how pulsation vessels reduce pulsation values (compare pulsation pressures at cylinder flange with values at vessel outlet flange). Case of half flow (50% flow) show more pressure pulsation values both at cylinder flange and vessel flange compare to full flow operation. This case (half flow or 50% flow) represents highest pressure pulsation value at pulsation vessel outlet flange among all operating cases and induce more pulsation to plant side. Presented table shows high pulsation value at cylinder flange for operation with nitrogen during start up. This high pulsation (6.1%) is around API limit, however, final pulsation value after vessel is relatively low. It presents a challenge since this case requires extra pulsation control device (reducing cylinder flange pulsation limit to safe margin value for this operating case requires low pressure orifice). With respect to this case (operation with nitrogen during start up) it is necessary to insert low pressure drop orifice in vessel flange. This table shows that it is necessary to consider all operating conditions, case by case, in pulsation modeling and simulation. All possible operating cases such as unloading steps, alternative gas (nitrogen and different process gases for start up and normal operation), spill back flow regulation, speed variation, compressors running in parallel, etc shall be respected and simulated.

Table II shows any attempt to design a system to meet line side pulsation guidelines and values in the cylinder flange would be impractical. The line side pulsation guideline should not be used between the bottle and the cylinder. A shaking force guideline (based on API 618 [10]) makes more sense for this portion of the system, combined with the current compressor side guideline [9]. It is not common practice of engineering companies or operation and maintenance teams to ask and evaluate pulsation at compressor cylinder flange during bidding stage. Main concern is always line side pulsation. Usually it will be left for detailed design review. It is a good recommendation to communicate this matter (pulsation at compressor cylinder) from early stage of compressor design and evaluation. A maximum allowable unfiltered (overall) pressure pulsation level at the compressor cylinder flange, to be defined in Purchase Order [8] - [10]. Purpose of this guideline is to protect the compressor valve and other compressor components from excessive pulsation. In addition shaking force study shall be done to protect the system from excessive shaking forces at discrete frequencies.

TABLE II
PRESSURE PULSATION IN RECIPROCATING COMPRESSOR PACKAGE IN VARIOUS OPERATING CONDITIONS

	Pulsation at Cylinder Flange (%)	Pulsation at Vessel Flange (%)
Full Flow	3.3%	0.8%
Half Flow	4.3%	0.9%
Alternative Gas (Nitrogen)	6.1%	0.6%

Fig. 1 shows pressure pulsation values at pulsation vessel flange (pressure pulsation detected values in plant side) vs. harmonic component for three different process operating conditions. This plot presents pressure pulsation components (as percentage of total value) vs. harmonic component (1^{st} to 20^{th} harmonic components). With respect to reciprocating machine speed, basic (1st) harmonic component is 5.45Hz (327 RPM) and presented plot shows harmonic components from 5.45 Hz (1st) to 109 Hz (20th). This figure shows that different process conditions lead to different pulsation behavior of machine and network. In case of full flow, major pulsation component is in 2^{nd} harmonic (10.9 Hz). Half flow operation has peak pulsation in 1th harmonic (5.45 Hz) since it is achieved by unloader (unloading one side of double acting cylinder). For alternative gas operation (Nitrogen in start up), pulsation simulation shows relatively considerable pulsation in 19^{th} harmonic (103.55 Hz). This is because of resonance with mechanical system. It requires low pressure orifice to mitigate risk. Based on simulation results, significant effects and problems are expected with higher molecular weight and pressures (in this case Nitrogen).

This plot shows importance of simulation of pulsation for all possible process conditions. All pulsation harmonic components equal or below 20th harmonic (20 times of machine operating speed) shall be respected in simulation. There is a difference between the amplitude and phase of the pulsation at different points on the standing wave. Because of the different phases for the various harmonics, their single contributions to the pulsation can not be completely added up. This shows how useful it is to study harmonic analysis and harmonic component curves.

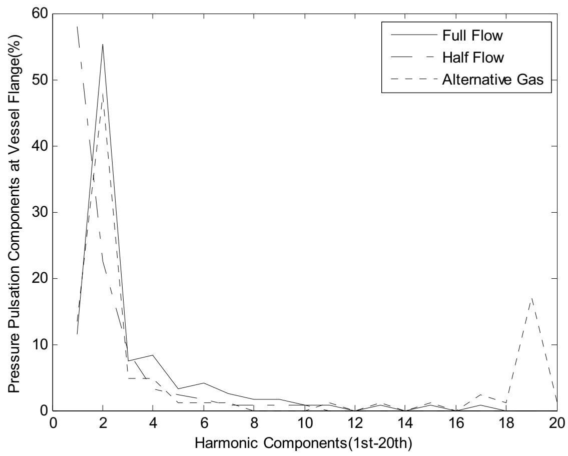

Fig. 1 Pressure pulsation values at pulsation vessel flange vs. harmonic component

Simulations and detailed studies show that for presented 6.5 MW reciprocating compressors in refinery service, an optimized pulsation analysis introduces significantly less pressure drop compare to conservative methods, result to around 2-5% less power consumption and around 7% more delivered gas, which translates to around 2 million USD of incremental yearly production. In addition reduction in pulsation vessel fabrication cost (around 7-10%) compare to conservative designs and reduction of maintenance costs shall be respected due to proper pulsation study and optimum pulsation design.

X. Practical Recommendations

In design stage for special purpose machine, pulsation limits are recommended around 95%-85% of API 618 (Approach 3) limits to have some margin (around 5-15%) to mitigate risk during construction and installation periods as well as unpredicted deviations and problems.

As a rule of thumb rough optimum values for pressure pulsation at flange of pulsation vessel to piping may be around 0.8 - 1.1% peak to peak of absolute pressure. It is just for rough check. Final optimum values need presented pulsation study method, iterations and optimization process.

Locating suction pulsation vessel presents a challenge to engineers and operators. Overhead horizontal vessel design leads to shorter length of piping between vessel and cylinder, less risk for harmonic resonance with vessel-cylinder connecting pipe and less required footprint. However, this design requires much more support, may introduce more risk for vessel-support shaking forces and difficulties in maintenance. This design (overhead horizontal vessel design) is common and traditional design however it is difficult to implement for big packages due to difficulties in support and vibration of overhead extra large size horizontal pulsation vessels. Vertical vessel design has taken considerable attention in recent years especially for heavy duty big reciprocating compressors. This selection and design shall be done very carefully and with respect to simulation results, reliability, performance, maintenance, available space, commercial conditions, time schedule, etc. Even details shall be selected very carefully. As noted, length of piping between cylinder and bottle has to be limited to reduce possibility of resonance. Fig. 2 shows an example of piping modification for this purpose in a vertical pulsation vessel design.

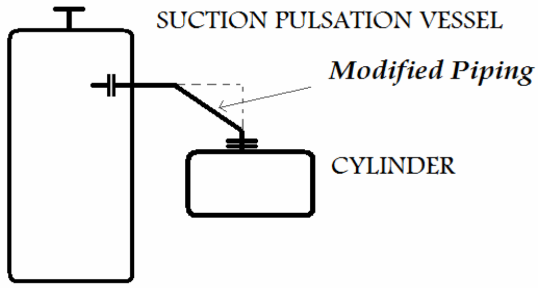

Fig. 2 Modified Suction Piping from Suction Pulsation Vessel to Compressor Cylinder

For special cases, performance of pulsation bottles can be adjusted by removable orifices and in very special situations by additional volume located near existing one. Pulsation vessels (Bottles) usually are ordered before piping system (upstream and downstream of reciprocating compressor) is defined and finalized. The drawback is that there may be difficulty for optimizing the pulsation control solution once the final piping system is determined. Pressure drop may be higher, bottle or piping may need to be modified.

It is expected by Client that compressor to deliver the required capacity with defined power consumption and pulsation limits. However the accurate system performance can not be predicted until final piping configuration is defined and the final pulsation study performed. Once the final configuration is defined, the compressor performance can then be re-evaluated. Variance between actual system performance and assumed one can be up to 5%-10%. It may have significant impact on plan performance and operation. It shows us importance of proper arrangement for piping design, vessel fabrication and pulsation study in project.

Following options are available to arrange pulsation study and pulsation vessel fabrication. This arrangement shall be done with respect to project technical, commercial, time schedule and operation conditions:

1) Final piping isometric drawings and facility data are available for final pulsation study before start of the fabrication of pulsation vessels. In this case the final pulsation study is performed and necessary modifications will be applied on vessels.

2) Final data are not available for final pulsation before the start of the fabrication of vessel. Based on experience it is possible to minimize the risk with a proper "Preliminary Pulsation Study" to be performed immediately before the fabrication of vessels.

2-1) The preliminary analysis to be done considering plant with a no-reflective line. With this option the maximum allowable pressure pulsation levels at the pulsation suppression device line side flange must be selected 80% of the allowable values of defined pulsation limits (70% of the allowable values when two or more pulsation suppression devices are connected to a common piping system). This analysis must also calculate suppression device shaking force amplitudes in order to reach acceptable amplitudes. When the piping system is finalized, final pulsation study can be performed. In this stage there is around 20% pulsation value margin to mitigate risks however if not enough or in case of resonance the only remaining system adjustment methods will be the installation of orifices, piping modifications and increasing the stiffness of the piping system (modification of supports).

2-2) The preliminary analysis must consider the configuration of the vessels with an hypothetic plant built up on the basis of the preliminary information (such as pipe size, line connections, equipment data, etc.) indicated in preliminary available documents (for example P&ID - Piping and Instrument Diagrams, Equipment Data Sheet and every document else available). The length among the various components (piping, equipment, etc) where tuned to find possible plant resonance. At this point, an iterative study, which could include changes on volume bottle size and geometry have to be made to control pressure pulsation in defined limit. This analysis evaluates pulsation resonances and shaking force amplitudes among the various operating conditions and variable lengths of hypothetic plant (for example piping length between elbows, etc). It is possible to determine range of shaking forces variations to establish their acceptability. After this vast and parametric study, the pulsation suppression vessels can be fabricated with very limited risk. In this method, exceptions (possible risks) are just limited and local cases such as closed branch lines for example by-pass or lines going to safety valve where local acoustic resonances may be detuned by specific change of piping size. In some rare case where the real acoustic resonances exceed the hypothetic ones in preliminary study, it may be necessary to accept slightly higher pulsation levels or insert proper orifice (slightly higher pressure drop).

Pulsation and vibration analysis report shall include Time Domain (TD) and Frequency Domain (FD) simulation results, Time Domain (TD) plots of key forces and pressure pulsation, dynamic pressure drop, models including mounting details (mounting plate, bolts, localized skid, etc) and shell flexibility (nozzle connection flexibility), calculated cylinder stretch forces, mode shape of bottles and piping and compressor stiffness assumption (compressor frame modeled as flexible support). Stress analysis of system and stress analysis of bottle internals are also recommended.

Inter-stage facilities including cooler, piping, etc and downstream facilities especially after coolers shall be sized carefully. Undersized facilities can cause excessive pressure drop and power loss. Pulsation and shaking force studies are necessary to avoid vibration problem in related facilities especially piping and cooler. Increased facilities and coolers cross section area to decrease pressure drop can cause significant increase of shaking force and vibration. Secondary volumes may be studied to reduce this vibration however in some cases this solution can not reduce vibration and modifications of recycle line are required to significantly lower shaking forces.

XI. Conclusion

Pulsation study and analysis of shaking forces and vibration are necessary for safety and reliability of reciprocating compressor packages and upstream and downstream facilities in plant. In this paper a comprehensive model and formulation are presented for pulsation and shaking force study of equipment and piping. This method can present optimum configuration with respect to performance, safety, reliability and operation. Numerical results are presented for optimum pulsation limits and pulsation simulations. Practical guideline regarding pulsation limits, optimum pulsation device arrangement, pulsation vessel designs, piping design, control of shaking forces, low cycle fatigue, orifices and pulsation control are also presented. Information obtained from presented pulsation simulation method will enable design engineer or operator to decide which pulsation solution is optimum.

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