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# The Control Elements of Pulsation in Pressure Gas Systems

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## Abstract

*To avoid vibration problems and to optimize the dynamic behavior, it has to realize a pulsation analysis during the design stage of an installation. In the 5th edition of the API Standard 618 this analysis is mandatory in a design approach 3 analysis. The article concerns of the possibility of using the numerical methods in the design of an attenuator in pressure gas systems. We use program Comsol, acoustic section, to generate and analyze the model. The results are presented comparatively with the analytical method of calculus, based on matrix calculus, using for this purpose the Matlab program.*

**Key words:** *noise, vibration, compressor, attenuator, numerical simulation.*

## Introduction

Problems with reciprocating compressors generally fall into one of three categories: loss of capacity; noise and vibration; failure to run. Possible causes of compressor vibrations there are: inadequate foundation; the compressor is not properly grouted; the attenuators not functioning correctly; incorrect piping design [1,2].

There is always some noise associated with a compressor in operation. Sources could include clicking of valves, vibrations of transmission from engine, air noise from motor, gas flow in piping, or slight belt slap. In compressors reports of knocking noise are more common [9,10]. Most knocks will be found in the compressor cylinder, caused by: loose pistons; insufficient head clearance; too great a piston-to-cylinder bore clearance; broken piston rings; loose or broken valves; air moisture, etc. Possible causes of knocking in compressor running gear include the following: loose flywheel or sheave; loose or worn bearings; crosshead pin-to-bushing clearance too large; mechanical packing loose in gland; excessive crosshead-to-guide clearance; connecting rod hitting end of piston rod in crosshead; belts misaligned. Possible causes of squealing noise in compressors include the following: motor or compressor bearing too tight; lack of oil; belts slipping; leaking gasket or joint.

The intermittent functioning of the reciprocating compressor determines the pulsations of the pressure. Because discharge flow is interrupted while the piston is on the suction stroke, pressure pulses are superimposed on the discharge system's mean pressure. At the suction side of the system, the same type of interruption is going on, causing the suction pressure to take on a non-steady component. The frequency of the pulses is constant when the speed is constant, which is the most normal condition. When a forcing phenomenon is superimposed on a system with elastic and inertial properties (a second order system), a resonant response is likely to occur. This is particularly true when the band of exciting frequencies is as broad as the type of system under consideration. The gas system meets the criteria of the second order system, as gas is compressible (elastic) and has inertia (mass). If left unchecked and a resonant response were to occur, the pressure peaks could easily reach a dangerous level.

While a single, low pressure compressor may require little or no treatment for pulsation control, the same machine with an increased gas density, pressure, or operational changes may develop a problem with pressure pulses. Especially for larger compressors, compressor manifold vibrations are important as the mass of compressor parts and pulsation dampers increases, leading to low-resonance frequencies, which may be excited by pulsation forces, by gas loads in the compressor cylinder, and by unbalanced forces and moments of the compressor[3,7,8].

To avoid vibration problems and to optimize the dynamic behavior, it is common practice to carry out a so-called pulsation and mechanical analysis during the design stage of an installation. This analysis includes the investigation of compressor manifold vibrations, which are the vibrations of compressor cylinders, distance pieces, crosshead guides, pulsation dampers and piping near the compressor. In the 5th edition of the API Standard 618 this analysis is mandatory in a design approach 3 analysis [6]. Pulsation and vibrations in reciprocating compressors may disturb safe and reliable operation. API 618 contains a chart that recommends the type of analysis that should be performed, based on horsepower and pressure. The pulsation control elements can have several forms, such as plain volume bottles, volume bottles with baffle plates and pipe with orifices, and acoustical filters. Regardless of which device or element is selected, a pressure loss evaluation must be made before the selection is finalized because each of these devices causes a pressure drop. For those installations where a detailed pulsation analysis, API 618 Design Approach 2 or 3, is required, is necessary a study performed by specialized companies. The procedures used in planning a new installation include frequently the pulsation study in the contract with the compressor vendor.

Numerical models have to be used to analyze the compressor manifold vibrations accurately. The compressor manifold analysis is based on finite element models of compressor parts, derived from CAD models. The method is efficient and easy to use and more accurate than usual analytical methods [2,4,10].

In the first step of a compressor manifold analysis procedure according the API Standard 618 is to generate the mechanical model of the pipe system [6]. The second step is to generate mechanical models of the compressor parts. For those cases where acoustical and mechanical natural frequencies coincide (critical situations), a forced response analysis must be carried out to calculate vibration levels of compressor and piping, and cyclic stress levels in piping and pulsation dampers. The vibration and cyclic stress levels must then be compared with allowable levels. In case they exceed allowable levels, modifications have to be investigated to finally achieve acceptable levels.

## A simple procedure

The objective of this approach is to improve the reliability of the system without having to design acoustical filters. For many systems, a plain volume bottle is all that is needed. The pulsation level for API 618 at Design Approach 1, the outlet side of any pulsation control device regardless of type, should be no larger than 2% peak-to-peak of the line pressure, or the value given by the following equation, whichever is less:

$$p = \frac{10}{p_{line}^{1/3}} \quad , \tag{1}$$

where: $p$ is maximum allowable peak-to-peak pulsation level at any discrete frequency, as a percentage of average absolute pressure; $p_{line}$ – average absolute line pressure. We'll express following the calculus relations for the size of a suction and a discharge volume bottle for a single-stage, single-acting, lubricated, reciprocating compressor. The gas being compressed is considered at the following conditions: suction temperature $t_s$; discharge temperature $t_d$; suction pressure $p_s$; discharge pressure $p_d$; isentropic exponent $k$; specific gravity $\gamma$ used to establish ccoefficients $Z_1$ , $Z_2$ , relation (3); percent clearance $c$. The geometrical elements of the compressor there are: cylinder bore $D$; cylinder stroke $S$; rod diameter $d$. We'll find first the suction (2) and discharge (5) volumetric efficiencies using:

$$\eta_{vs}=1.00-[(1/f)r_p^{1/k}-1]c-L \qquad (2)$$

where: $f$ is ratio of discharge compressibility factor to inlet compressibility factor (expressing the alteration of ideal conditions):

$$f=Z_2/Z_1 \qquad (3)$$

$r_p$ – pressure ratio;

$$r_p=p_d/p_s \qquad (4)$$

$L$ – considers the effect of gas slipage past the piston rings in the various types of construction; $L$= 0.03 for lubricated compressors and 0.07 in nonlubricated machines.

$$\eta_{vd}=\frac{\eta_{vs}}{r_p^{1/k}}f \qquad (5)$$

Find the piston displacement using equation (6) for a single-acting compressor and (7) for double acting compressor:

$$V_s=S\cdot\frac{\pi D^2}{4} \qquad (6)$$

$$V_d=S\cdot\frac{\pi(2D^2-d^2)}{4} \qquad (7)$$

For the purpose of quick estimates, we consider the curve from figure 1, [7]. This curve is not meant to supersede a comprehensive analysis. It should be used in checking vendor proposals or in revising existing installations where a single cylinder is connected to a header without the interaction of multiple cylinders. While not a hard rule, the curve should be conservative for compressors under 7 MPa and 350kW [8]. The volume bottle is simple, unbaffled type. Using the volumetric efficiencies found previously, find the size multiplier from the volume bottle sizing chart, figure 1, for suction multiplier $SM$ and discharge multiplier $DM$. Find the required bottle volume from the displacement and the multiplier, suction bottle volume $SBV$ and discharge bottle volume $DBV$:

$$SBV=V_s\cdot SM \qquad (8)$$

$$DBV=V_d\cdot DM \qquad (9)$$

To complete the solution for the volume bottle dimensions, assume 2:1 elliptical heads and use the following relationships for the bottle diameter $d_b$ and respectively $L_b$ volume bottle length:

$$d_b=0.86(SVB)^{1/3} \text{ or} \qquad (10)$$

$$d_b=0.86(DVB)^{1/3} \qquad (11)$$

$$L_b=2\,d_b \qquad (12)$$

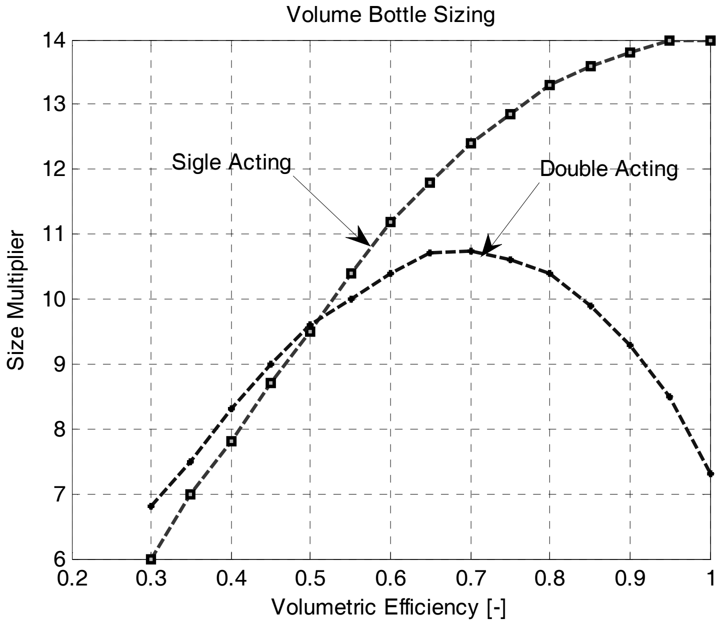



**Fig. 1.** Selection of coefficient for volume multiplication.

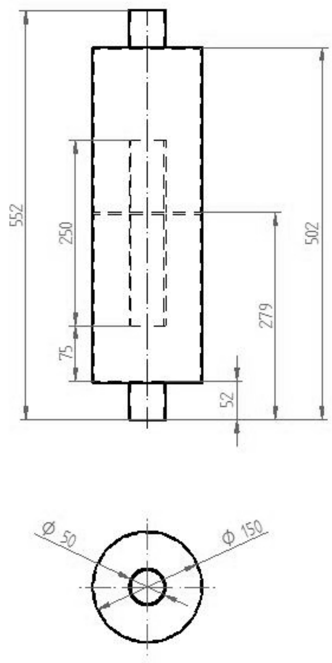



**Fig. 2.** A model for a reactive attenuator.

## Design of an acoustic attenuator

If we intend to use a special device to eliminate a known frequency wave, we'll use a tridimensional model. For example, we'll construct the device from figure 2, a reactive attenuator, with the dimensions expressed in mm. Reactive mufflers use a number of complex passages (or lumped elements) to reduce the amount of acoustic energy transmitted [2]. This is accomplished by a change in impedance at the intersections, which gives rise to reflected waves (and effectively reduces the amount of transmitted acoustic energy). Since the amount of energy transmitted is minimized, the reflected energy back to the source is quite high. Opposite to absorptive mufflers, which dissipate the acoustic energy, reactive mufflers keep the energy contained within the system. Reactive mufflers are very efficient in low frequency applications [8,9]. Other application areas include: harsh environments (high temperature/velocity engines, turbines, etc), specific frequency attenuation (using a Helmholtz like device, a specific frequency can be toned to give total attenuation of radiated sound power), and a need for low radiated sound power (car mufflers, air conditioners, etc). The performance of the attenuator can be expressed with the value of transmission loss, *TL*. It is defined as the difference between the sound power level of the incident wave to the muffler system and the transmitted sound power:

$$TL = 10 lg \frac{I_i}{I_t} \tag{13}$$

where $I_t$ and $I_i$ there are the transmitted and incident wave power, respectively s. figure 3. From this expression, it is obvious the problem with measure *TL* is decomposing the sound field into incident and transmitted waves which can be difficult to do for complex systems (analytically).

$$TL = 10lg\left[\frac{1}{4}\frac{S_2}{S_1}\left|A + \rho_0 cC + \frac{B}{\rho_0 c} + \frac{S_1}{S_2}D\right|^2\right] \tag{14}$$

The matrix which represents the mathematical model of the attenuator is a product of matrices. Each of them is characteristic for an element that modifies the aspect of pressure wave. For example in the case of attenuator presented into the figure 2, the characteristic matrix is:

$$[T] = [T_1][T_2][T_3][T_4][T_5][T_6][T_7] = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \tag{15}$$

where:

$$T_1 = \begin{bmatrix} 1 & 0 \\ 0 & S/s_1 \end{bmatrix} \tag{16}$$

$$T_2 = \begin{bmatrix} \cos kL_1 & i \cdot \rho_0 \cdot c \cdot \sin(kL_1) \\ i \cdot \sin(kL_1)/(\rho_0 \cdot c) & \cos kL_1 \end{bmatrix} \tag{17}$$

$$T_3 = \begin{bmatrix} 1 & 0 \\ 0 & s_2/S \end{bmatrix} \tag{18}$$

$$T_4 = \begin{bmatrix} \cos kL_2 & i \cdot \rho_0 \cdot c \cdot \sin(kL_2) \\ i \cdot \sin(kL_2)/(\rho_0 \cdot c) & \cos kL_2 \end{bmatrix} \tag{19}$$

$$T_5 = \begin{bmatrix} 1 & 0 \\ 0 & S/s_2 \end{bmatrix} \tag{20}$$

$$T_6 = \begin{bmatrix} \cos kL_3 & i \cdot \rho_0 \cdot c \cdot \sin(kL_3) \\ i \cdot \sin(kL_3)/(\rho_0 \cdot c) & \cos kL_3 \end{bmatrix} \tag{21}$$

$$T_7 = \begin{bmatrix} 1 & 0 \\ 0 & S/s_3 \end{bmatrix} \tag{22}$$

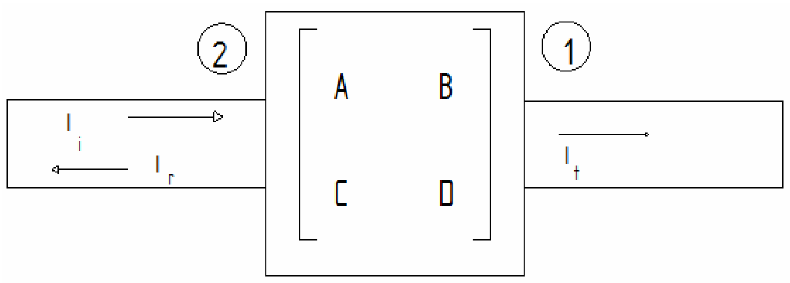



**Fig. 3.** Elements used to define transmission loss.

where: $S$ is the aria of the section of attenuator chambers, supposed equal; $s_i$ – aria of the section of the pipe ( index 1 input pipe; index 2 inner pipe; index 3 output pipe); $L_i$– length of chambers (index 1 first chamber, index 3 second chamber) and of the inner pipe (index 2); $\rho_0$– density of the gas; $c$ – velocity of sound through gas; $k$– wave number expressed by relation (23) ($\omega$ angular frequency, $F$ frequency of the wave):

$$k = \frac{\omega}{c} = \frac{2\pi F}{c} \tag{23}$$

With relations (14-23) we calculated the value of transmission loss, represented into the figure 4. From figure 4 we can see that a significant effect is obtained at frequencies of 475 Hz and 950 Hz. The analyze of the attenuator can be made also into specialized programs like Comsol [4], figure 5. Comsol program uses differential equation theory to model engineering problems. In our case we realize a time-harmonic analysis (or frequency-domain formulation) uses a Helmholtz equation:

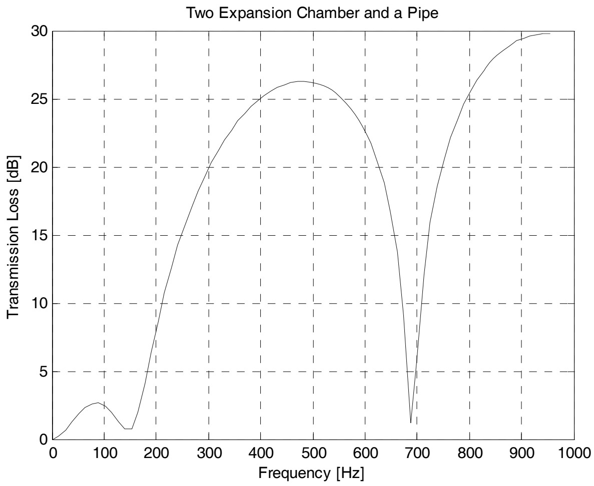



**Fig. 4.**Transmisson loss, relation 14, for the attenuator represented in figure 2.

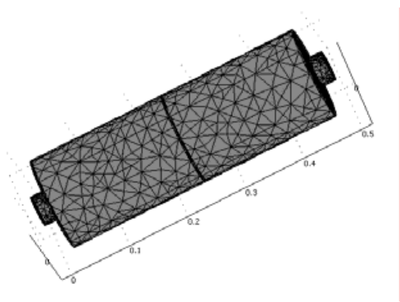


**Fig.5.** Using finite method element to analyze the model.

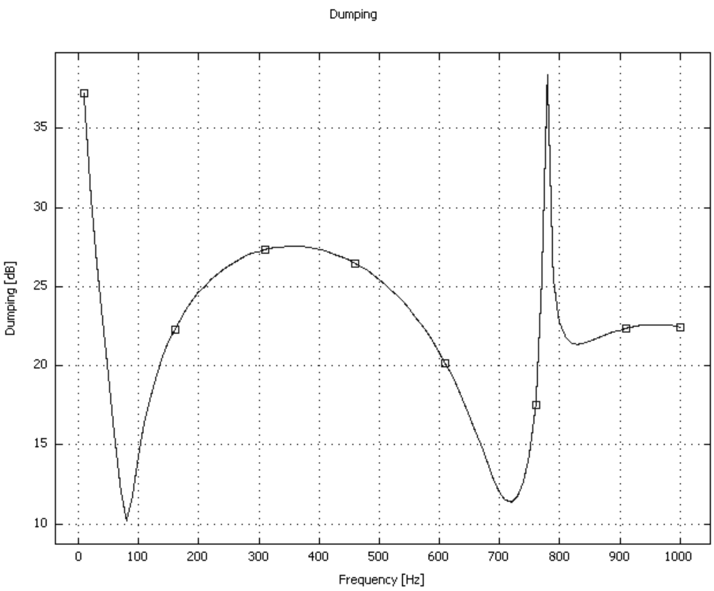
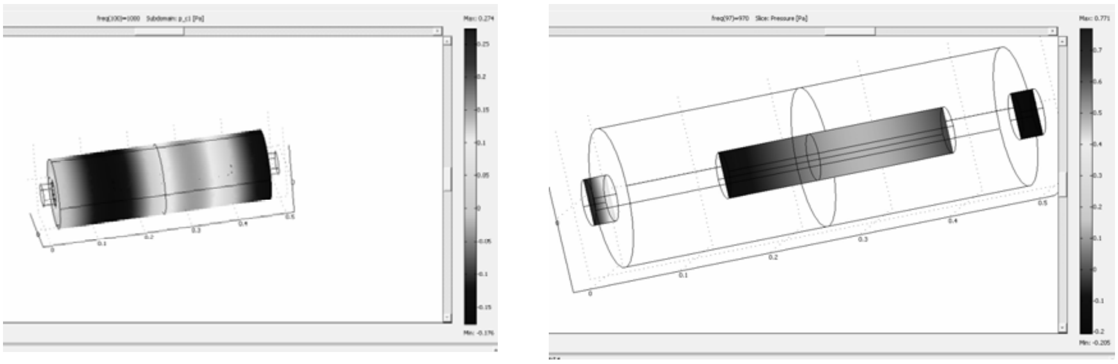



**Fig.6.** The values of transmission loss obtained with Comsol program.

$$\nabla\left(\frac{-\nabla p}{\rho_0}+q\right)+\frac{\omega^2 p}{\rho_0 c^2}\,. \tag{24}$$

The $q$ term is a dipole source with the same units as acceleration. We model first the attenuator as a 3D element. Following we introduce the boundary conditions for the tree pipes and two chambers which constitute the attenuator. The input pressure is 1 and the output pressure is 0. We study the model for a domain of frequencies between 10 and 1000 Hz, with a step of 10 units. We observe that the favorable frequency is around 375 Hz, and in the domain of high frequencies the amortization effect is lower as the effect predicted by relation (14).



a b

**Fig. 7.** Distribution of acoustic pressure in the elements of attenuator.

We consider that the favorable effect in a very short domain (near 10 Hz and 775 Hz) couldn't be used in practical applications. As we can see there are differences between the results obtained in Comsol and formula (14) , figures 6 and 4. Probably the results of a tridimensional study, realized in Comsol are better as relations (14), but an experimental study is imposed.

## Conclusions

With the advent of modern workstations and faster PC computers, the solution of the differential equations of motion for acoustical waves in piping system on a digital computer has become feasible [2,10]. As we see the analyze with Comsol is a powerful instrument in detecting the shape of attenuator for different input conditions, figure 7. In current practice, pulsation design studies using digital computer technology can produce the same results as obtained with a dynamic simulation on the analogue system [5]. The results from digital simulation satisfy the requirements of API 618. The digital computer has the advantage of data file storage. With storage capability and the ability to readily manipulate the data, it is not as necessary to have immediate decisions made. Piping changes that are recommended for acoustical control can be evaluated in a more comprehensive manner taking into

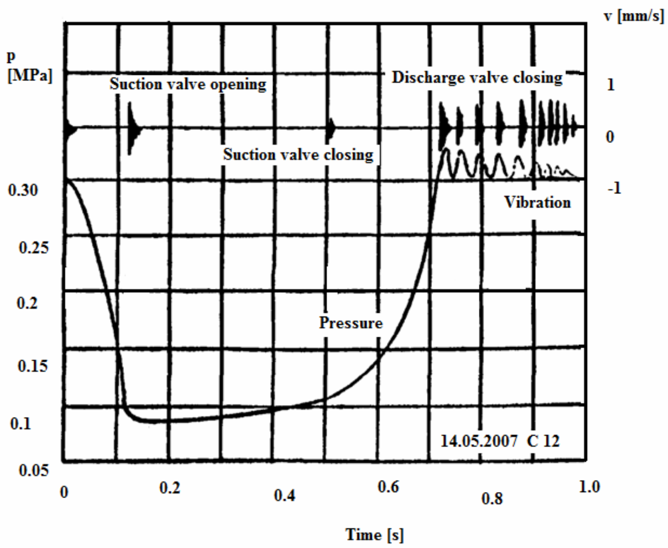



**Fig. 8.** A source of vibration induced by springs of discharge valve.

account safety, cost, maintenance, and operational considerations. An additional benefit is realized if system changes are anticipated at a later time. The data files can be retrieved and the system rerun with the changes to the thermo physical properties or in the piping system itself without the need to remodel the entire system. The use of vibration and ultrasonic patterns in combination with cylinder *pV* patterns is used currently in diagnosing the condition of valves, rings, and packing [5,10]. We can use the measuring of vibrations in establishment of the different situation, in which the compressor and the attenuator will work together. For example, when valves open and close, they produce vibrations that are detected with an accelerometer. Figure 8 shows the variation of pressure a cylinder of a compressor and the vibration patterns of valves are also shown. Springs in discharge valve are too heavy so repeated closures are shown on vibration trace. Using these experimental results we could establish the alterations in the shape of the attenuator to eliminate (reduce) the noise, in new conditions.

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# Utilizarea amortizoarelor în sistemele cu gaz sub presiune

## Rezumat

*Pentru a evita problemele legate de vibrații şi pentru optimizarea comportării dinamice, este o procedură obişnuită să executăm o analiză mecanică şi a pulsațiilor în timpul fazei de proiectare a instalației. In a cincea ediție a standardului API numărul 618, această analiză este obligatorie pentru o proiectare spațială. Acest articol studiază posibilitatea folosirii metodelor numerice în proiectarea unui amortizor pentru sistemele cu gaz sub presiune. Utilizăm programul Comsol, secțiunea acustică, pentru generarea şi analiza modelului. Rezultatele sunt prezentate comparativ cu metodele analitice de calcul, ce folosesc calculul matricial, utilizând pentru acesta programul Matlab.*