Investigation on the influence of pressure pulsations on multistage reciprocating compressors - Comparison between test and simulation results

by:

Ing Attilio Brighenti,
Ing. Andrea Pavan
S.A.T.E. S.r.l.
Venezia
Italy

Ing Massimo Maffeis

SIAD Macchine Impianti S.p.A.
Bergamo
Italy

Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
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Abstract:

The response of a reciprocating compressor piping system to predefined disturbance signals, generated at any interface with the gas flow, can be analysed by a digital computer model, which includes all the components of the compressor and piping system.

This allows the intrinsic piping response and flow resonance attitude in the frequency and time domain to be analysed, as a function of pressure or mass flow rate signals that can be calculated when the interaction between the machine thermodynamics, the valves mechanics and the fluid exchange process are taken into account together with the piping flow dynamics.

This paper, after a brief summary of the mathematical modelling approach, concentrates on the presentation of the results and the interpretation of experimental and simulation work, applied to a real industrial high-pressure compressor.

1 Introduction

Mathematical modelling of reciprocating compressors is an activity in increasing demand in industrial projects for the verification of pressure pulsations in piping, e.g. to verify the compliance of an installation with the API 618[1] rules. In particular approach 2 therein, requires the interaction between the machine and the piping dynamics with which the same is connected to be considered; in particular the cylinder gas thermodynamics, the valves mechanics, the compressible fluid exchange process and the piping flow dynamics.

This requirement stems from the fact that, while the piping dynamics have a linear behaviour in typical correct design conditions (Mach lower than 0.2 and pressure pulsations amplitudes close to the acceptable limits), which allow the application of the well-known electro acoustic analogy, the disturbing signals at the interface with the piping are determined by several non linear phenomena and by the afore mentioned interactions.

The objective of creating a set of practical design and verification tools suitable to fulfil all the necessary steps for a complete dynamic analysis required the following stages of development, started in 1994:

- Developing and analytically/numerically validating ***ACUSYS***[2,3,4], the linear analysis tool and piping subsystems block builder.

- Developing and analytically/numerically validating ***ACUSCOMP***[5,6], a non-linear model of multi-cylinder reciprocating compressors incorporating plena, non-linear flow restrictions, valves and linear piping subsystems.

- Enhancing the modelling capability of ***ACUSCOMP*** to incorporate models of leakages through suction and discharge valves, piston ring and rod seals of the compressor.

- Setting up portable instrumentation for in-cylinder measurements and comparing the results of simulations with test data on a real commercial compressor.

The paper illustrates the overall theoretical basis of the tools developed showing the preliminary results from the last of the above step, still in progress, and the potential applicability of these tools also for on/off-line diagnostics of reciprocating compressor installations.

2 Mathematical model

The machine elements considered in the overall compressor plant model are:

- *Piping state space matrixes*, i.e. the overall dynamic characteristics of each piping side (suction, discharge or interstage) connected to one or more cylinders.

- *Single or double effect cylinders*, which include:

  - *Cylinder thermodynamics*, which include both the open phase (exchange with the piping) and closed phase (closed valves).

  - *Suction and discharge valve shutter dynamics*, as a function of either mechanical commands (e.g. cam shaft or rotary actuators) or differential pressure drive, between the cylinder and the plant side (both dynamically calculated).

  - *Suction and discharge valve gas flow dynamics*, as a function of the instantaneous valve opening and gas conditions at their inlet and outlet sections.

  - *Flow dynamics through narrow gap leakages*, i.e. through valves seats, piston rings and rod seals.

- *Plena*, representing the buffer volumes that are at the interface of the machine, between valves and piping.

- *Restrictions,* representing equivalent pressure loss elements that can be used in place of the entire piping state space blocks to analyse the behaviour of a machine without the multimodal dynamic behaviour of the piping (this can be a preliminary step during a new machine or installation design)

The following sections summarise the mathematical basis of the above model components.

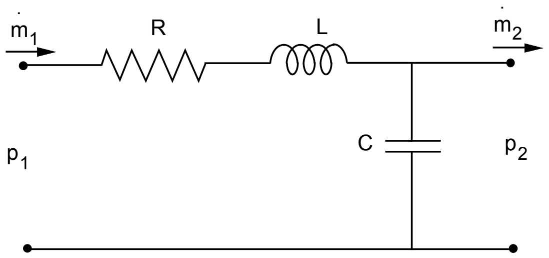
2.1 Piping model and pulsation propagation

The model of a plant, implemented in ***ACUSYS***, is essentially a finite elements model of a mono-dimensional field of propagation.

This hypothesis of plane wave propagation is valid when the wavelength is sufficiently greater than the tube diameter or than the characteristic dimension orthogonal to the propagation direction, namely when:

$$f < 0.6\frac{c}{D} \qquad (1)$$

In Eq. 1 f is the frequency (Hz) of a generic signal component, c the sound speed inside the fluid pipe (m/s), corrected to account for the pipe hoop compliance and D the maximum pipe cross section dimension, orthogonal to the propagation direction of the sound waves (m). This condition is usually verified inside compressors piping. For example for a piping having a diameter of 240 mm and a sound speed of 450 m/s this cut-off frequency is of 135 Hz, while the compressors speed fundamental typically ranges from 10 to 20 Hz. Viscous effects and sound wave scattering dampen the higher frequencies.



Picture 1: Electro-acoustic analogy adopted in ***ACUSYS***

Besides the plane wave propagation model, associated to the range of frequencies studied, ***ACUSYS*** runs under the hypothesis of a linear relationship between amplitudes of pressure variations and local fluid velocities, valid as long as the latter and the mean fluid velocity are small enough compared to the sound speed. Such hypothesis leads to the electro-acoustic analogy[7], represented in *Picture 1*.

$$R\dot{m}_1 + L\frac{d\dot{m}_1}{dt} = p_1 - p_2 \;; \qquad (2)$$
$$L = \frac{\Delta x}{A} \quad ; \quad R = \frac{\lambda}{2}\frac{\Delta x}{D}\frac{\dot{m}_0}{\rho_0 A^2}$$

$$C\frac{dp_2}{dt} = \dot{m}_1 - \dot{m}_2 \qquad (3)$$
$$C = \frac{A\Delta x}{c^2}$$

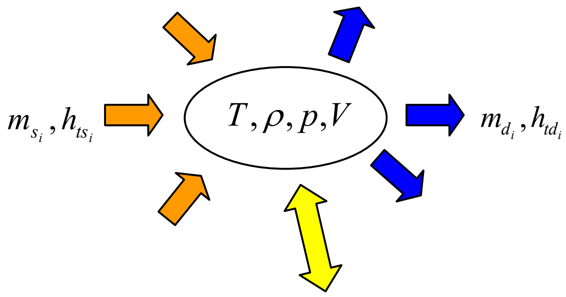
In Eqs. (2) and (3), which respectively represent the continuity and momentum balance after the above-mentioned approximation, the following notations and definitions are applied (in S.I. units):

- R: equivalent resistance derived from the linearisation of the turbulent friction relationship (Eq. 4),
- L, C: equivalent inductance and capacity of the element respectively
- $\dot{m}$, p: flow rate and pressure of the fluid at sections 1 and 2
- t: time
- A: pipe cross section area
- Δx: element length
- c: sound speed of the fluid in the element
- λ: friction head loss factor per unit length
- D: pipe diameter
- ρ: mass density of the fluid
- $_0$: subscript indicating the mean value in a period of time (e.g. the period of the 1st harmonic component)

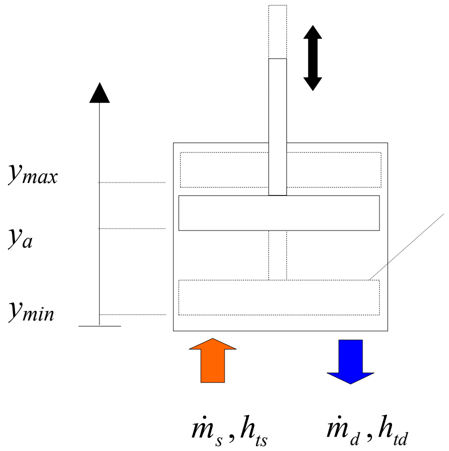
$$\Delta p_0 = (p_1 - p_2)_0 = \lambda\frac{\Delta x}{D}\frac{\dot{m}_0^2}{2\rho_0 A^2} \qquad (4)$$

where: $\dot{m}_{10} = \dot{m}_{20} = \dot{m}_0$

2.2 Cylinders and plena



Picture 2: General mass-energy balance of an element with fixed or variable volume



Picture 3: Mass-energy balance representation for a cylinder

In the ***ACUSCOMP*** model of a compressor, elements with fixed volume, like plena, and with variable volume, like cylinders, are both considered, as represented in *Picture 2*.

In general the mass balance for a single element can be written as in Eq. (5), where ρ is the fluid density in the volume V and the subscripts s and d refer to the suction and discharge ports respectively.

$$\frac{d\rho}{dt} = \frac{1}{V}\left(\sum_{i=1}^{n_s} \dot{m}_{s_i} - \sum_{i=1}^{n_d} \dot{m}_{d_i} - \rho \frac{dV}{dt}\right) \quad (5)$$

Eq. (6) is instead the general expression of the energy balance, where U is the total internal energy, h is the specific enthalpy of the gas flows and L the work performed by the gas in the adiabatic volume.

$$\frac{dU}{dt} = \sum_{i=1}^{n_s} \dot{m}_{s_i} h_{ts_i} - \sum_{i=1}^{n_d} \dot{m}_{d_i} h_{td_i} - \frac{dL}{dt} \quad (6)$$

In the case of a real gas with compressibility factor z, assumed as constant for the gas conditions at the temperature and pressure in the volume V, the expressions of the specific internal energy u and enthalpy h can be written like for an ideal gas, as in Eqs. (7) and (8) respectively.

Considering the work performed by the gas in the volume V, see Eq. (9), and the real gas state equation (10) and combining together Eqs. (6), (7), (8), (9) and (10), the final Eq. (11) of the energy balance inside V can be obtained; in this equation $\gamma = {}^{c_p}/_{c_v}$ and m are respectively the isentropic and the polytropic exponent. The latter is defined in terms of a polytropic efficiency that can take into account internal thermodynamic losses and heat exchange with the boundary surface (e.g. cylinder wall).

$$u = \frac{U}{\rho V} = c_v T \quad (7)$$

$$h = c_p T \quad (8)$$

$$dL = p dV \quad (9)$$

$$\frac{p}{\rho} = \bar{z} RT \quad (10)$$

$$\frac{dT}{dt} = \frac{m-1}{\gamma-1}\left[\gamma \frac{1}{\rho V}\sum_{i=1}^{n_s} \dot{m}_{s_i} T_{ts_i} - T\left(\gamma \frac{1}{\rho V}\sum_{i=1}^{n_d} \dot{m}_{d_i} + \gamma \frac{1}{V}\frac{dV}{dt} + \frac{1}{\rho}\frac{d\rho}{dt}\right)\right] \quad (11)$$

For a cylinder, as represented in *Picture 3*, the volume V depends only on the piston stroke. Thus, calling A_a the piston area, y_a the piston position, T_a the temperature and ρ_a the density inside the cylinder, the Eqs. (6) and (11) can be rewritten as (12) and (13).

$$\frac{d\rho_a}{dt} = \frac{1}{A_a y_a}\left(\dot{m}_s - \dot{m}_d - A_a \rho_a \frac{dy_a}{dt}\right) \quad (12)$$

$$\frac{dT_a}{dt} = \frac{m-1}{\gamma-1}\left[\frac{\gamma}{A_a}\frac{\dot{m}_s T_{ts}}{\rho_a y_a} - T_a\left(\frac{\gamma}{A_a}\frac{\dot{m}_d}{\rho_a y_a} + \gamma \frac{1}{y_a}\frac{dy_a}{dt} + \frac{1}{\rho_a}\frac{d\rho_a}{dt}\right)\right] \quad (13)$$

A plenum is an element of fixed volume interfaced with the piping (or restriction) on one side and the valves (suction or discharge), on the other, which is put in communication with the cylinder volumes when the valves open. In double effect cylinders a single plenum can surround a pair of valves of the same type (suction or discharge). Eqs. (14) and (15) are derived from (6) and (11) deleting the derivative term of V, which is constant.

$$\frac{d\rho}{dt} = \frac{1}{V}\left(\sum_{i=1}^{n_s} \dot{m}_{s_i} - \sum_{i=1}^{n_d} \dot{m}_{d_i}\right) \quad (14)$$

$$\frac{dT}{dt} = \frac{m-1}{\gamma-1}\left[\gamma \frac{1}{\rho V}\sum_{i=1}^{n_s} \dot{m}_{s_i} T_{ts_i} - T\left(\gamma \frac{1}{\rho V}\sum_{i=1}^{n_d} \dot{m}_{d_i} + \frac{1}{\rho}\frac{d\rho}{dt}\right)\right] \quad (15)$$

2.3 Valves and restrictions

ACUSCOMP calculates the opening of each compressor valve solving the single degree of freedom equation of the shutter position. The forces considered are those due to the pressure difference between the cylinder and the adjacent plenum, acting on the effective shutter area, the restoring force by non-linear springs and friction damping. End stroke shock constraints are modelled too, which yield bouncing behaviour to be visible, when occurring, in the resulting plots. Based on the above balance the valves open only when a positive pressure difference exceeds the spring preload and the friction forces.

The mass flow through the valves orifices is calculated as a function of the instantaneous valve opening, the pressure upstream and downstream and the temperature upstream of the valve, i.e. in the cylinder or plenum, based on well-known orifice gas dynamic relationships[8]: Eqs. (16) through (19). This calculation procedure is based on the commonly accepted assumption that the head losses along the fluid accelerated stream, from the intake (subs. "1") to the vena contracta (subs. "a"), are negligible compared to the expansion losses in the decelerated stream, i.e. from the vena contracta to the outlet section (subs. "2") and that the intake Mach number is also low enough (less than 0.2)

$$\dot{m} = X_v A_v \Psi \sqrt{p_{t1} \rho_{t1}} \qquad (16)$$

$$A_v = \sqrt{2} \cdot \Phi = \sqrt{2} \alpha A = A \sqrt{\frac{2}{\zeta}} \qquad (17)$$

$$p_a = p_{t1} - \frac{p_{t1} - p_{t2}}{1.2 \cdot 10^{-3} C_1^2} \qquad (18)$$

$$\Psi = 0.0244 C_1 \sqrt{\frac{\frac{2\gamma}{\gamma - 1}\left[\left(p_{t1}/p_a\right)^{\gamma-1/\gamma} - 1\right]}{\left(p_{t1}/p_a\right)^{\gamma+1/\gamma}}} \qquad (19)$$

The following other notations and definitions are applied in the above equations (S.I. units):

- ρ: gas density (kg/m^3),
- X_v: instantaneous valve opening, derived by the shutter motion equation (not shown here for brevity),
- A: valve flow area (m^2 in S.I. units, at full opening) provided by valve manufacturers.
- A_v: valve flow coefficient (m^2 in S.I. units, at full opening) provided by valve manufacturers after incompressible flow tests,
- α: flow area contraction ratio (dimensionless),
- ζ: head loss factor (dimensionless),
- C_1: gas correction factor (dimensionless)[8].
- Subscript "*t*" indicates the *total* or stagnation quantities as opposed to the *static* ones, following the common convention in gas dynamics.

The generalised orifice model above allows the calculation of both the flow through compressors valves, which is normally sub critical, and that through any other restrictions, e.g. equivalent obstructions or loss elements that can be used in place of an entire piping. This allows the analysis of complex plants dynamics with process control and choke valves whose flow is strongly non linear or also to approximate pipelines by equivalent lumped restrictions (quasi-steady non linear model), in order to make rapid preliminary analyses of the behaviour and indicated cycle of a compressor, checking the pre-sizing of the compressor valves, while neglecting pressure pulsations.

2.4 Leakage sources

The same orifice compressible flow model is the basis for the dynamic calculation of leaking flows through seals or valve seats, which can be chocked, as a function of the instantaneous gas conditions on the two sides of the orifice (e.g. the pressure in the two volumes of a double effect cylinder).

The modelled leaking pseudo-orifices considered in each ***ACUSCOMP*** two effect cylinder model are the following four pairs:

- piston ring seal (leaking to/from either effects)
- piston rod seal (leaking to/from crankcase)
- suction valve (leaking from each effect to the suction plenum)
- discharge valve (leaking from the discharge plenum into each effect)

Implementing these modelling features proveessential to better approximate the indicated (*p-V*) diagram of real compressors, avoiding the rough approach of using equivalent polytropic exponents, which instead should be calculated from appropriate realistic values of the polytropic efficiency during compression and expansion, regardless of leaks.

The leakages influence the pressure inside the cylinders, at any given position along the piston stroke. The inter-stage pressure settlement is consequently affected depending on the input/output flow at the two ends, i.e. on the shape and duration of the valves flow duty cycle. All these factors impact on the consequent pressure spectra at the ends of the interconnecting piping and the piping dynamics, which ultimately provide a dynamic feedback to the valves, influencing the actual flow rate to/from the cylinder. All these complex interactions, which ultimately do affect the compressor performance, are difficult to evaluate unless by fine numerical modelling.

Another reason for separating the modelling of leaking flows from the thermodynamics is that by this method it is possible to evaluate the effects of increasing leakages on the *p-V* diagram and plena pressure spectra. Having this knowledge in advance allows the support of monitoring and diagnostic activities during the compressors lifetime, which could rely on robust algorithms for the evaluation of trends of specific signal features, associated to pre-analysed fault levels and faults/signals cause-effects relationships.

3 Experimental apparatus

3.1 Compressor

A three-stage air compressor produced by SIAD Machine Impianti S.p.A. has been used in the subject development programme *(Picture 4)*, in order to gain experience in data acquisition in a real industrial environment. Its nominal characteristics are the following:

• Mass flow rate	0.3843	kg/s
• Suction pressure	1.013	bar a
• 1st interstage pressure	3.6	bar a
• 2nd interstage pressure	12.5	bar a
• Discharge pressure	41.0	bar a
• Revolution speed	1185	rpm

The compressor is a three, double effect cylinder type. In the test installation it delivers air to a discharge pipe ending with a choked valve set manually to stabilise the flow at the desired test pressure.

Downstream of each stage the compressed air passes through a pulsation damper and a shell and tube water cooler which brings its temperature down to +35°C.

3.2 Data acquisition system

The measurements recorded during the tests are the following:

- Average gauge pressure at the discharge plenum of each stage.

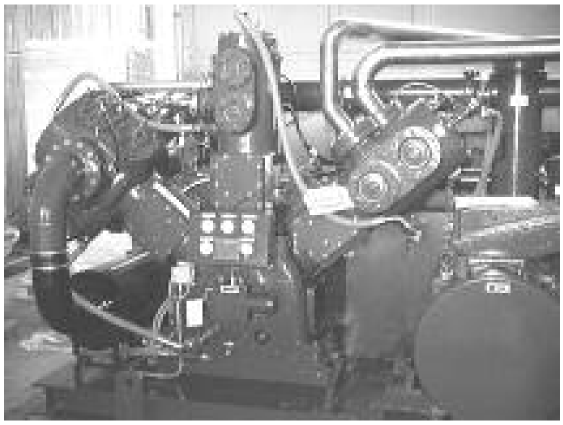
- Dynamic pressure by means of a piezoelectric transducer having the following characteristics:

o Resolution:	0.028	kPa
o Dynamic range	34500	kPa
o Max. Static pressure	69000	kPa
o Frequency range	5-50000	Hz

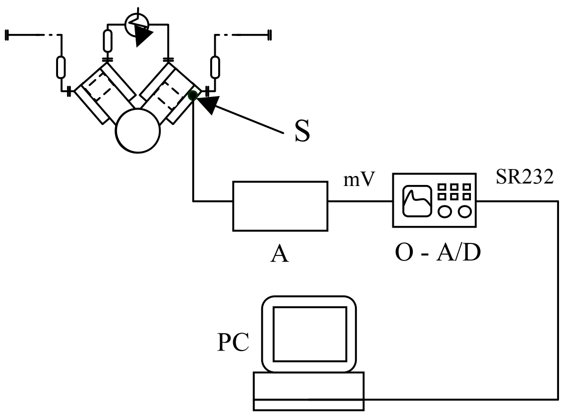
- Temperature at six points (suction, discharge inlet/outlet of the interstage coolers).

- Flow rate at the compressor discharge.

The TDC is triggered by means of an encoder, having an accuracy of ± 1°; data are sampled at 20 kHz frequency.



Picture 4: Three-stage air compressor used for test



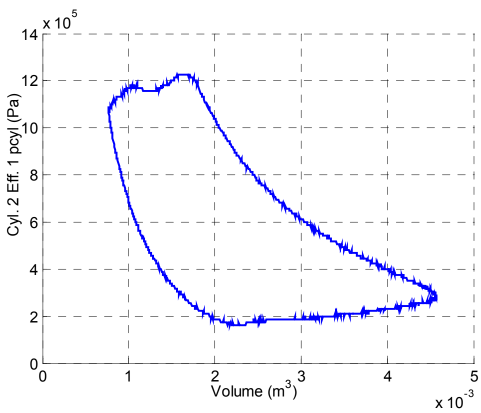
Picture 5 - Scheme of the data acquisition system.

The data acquisition system is shown in Picture 5, with particular reference to the measurement of the dynamic pressure. The dynamic pressure, whose average value is zero, is converted into absolute pressure values during data post-processing, after having accounted for average pressure bias and the sensor gain sensitivity to gas temperature. Simulations on the sensor thermal dynamics, based on ACUSCOMP results, proved that this gain can be considered constant throughout a cycle, as the thermal inertia and the small convective surface of the sensor prevents it from following the ca. 20 Hz frequency of the gas temperature changes. Therefore the gain of the sensor can be calculated based on the average in-cylinder gas temperature, calculated on its own by means of cycle simulations or by other measurements. In the future the in-cylinder temperature could also be added to the experimental setup, to enhance the accuracy. The pressure sensor is mounted centrally at the top of the cylinder head, with no modifications of the standard components, exploiting the existing plugged bore, foreseen to access the piston head for mounting. Therefore the transducer senses without delay the pressure changes in the real compressor cylinder. During this first phase of the development programme, which has the purpose of exploring the applicability and reliability of the test method, only one piezoelectric sensor was available. Therefore the tests had to be performed by shutting-down/restarting the compressor from/to the same conditions, to allow pressure readings in each cylinder.

4 Results

4.1 Experimental data

To date the measurements recorded during the test sessions were sufficiently indicative only for the second stage cylinder, which shall be illustrated here below (Picture 6). It should be noted that the measurements recorded were affected by earthing noise at 50 Hz, generated by surrounding industrial installations. This noise frequency is within the range of the phenomena of interest, which spans from the fundamental harmonic at 19.75 Hz (corresponding to the shaft speed) up to ca 300 Hz, to appropriately include the valves dynamics.



Picture 6 – Experimental *p-V* diagram of the 2nd stage – 1st effect cylinder of the tested compressor.

It may clearly be seen that both the suction and discharge pressures are lower than the nominal values of respectively 3.6 and 12.5 bar or even to the 3.4 and 11.7 bar that corresponds to the test delivery pressure of 40 bar absolute. This difference can be due either to an actual different settlement of the inter-stage pressures or due to measurement errors that are being investigated.

Despite this discrepancy this plot is taken as the reference background (dashed line in all the pictures), for the comparisons made in the following section with the simulation results. This helps illustrate the sensitivity to various physical phenomena.

4.2 Simulation results

Many simulation runs have been performed during this phase of the work, in order to gain sensitivity to the importance of the many phenomena and parameters that affect the performance of a real compressor. Among others the following model set ups are presented:

a) Model with quasi-steady piping, i.e. with non linear restrictions (see sec. 2.3)

b) Model with dynamic linear piping, i.e. with full state-space piping subsystems (see sec. 2.1)

c) Model like (a) but with leakages (see sec. 2.4) through:

   1. Piston rings (0.1 % of piston bore)

   2. Suction valves (50 μm equivalent lift for the 1st and 2nd stage, 20 μm for the 3rd stage, out of a lift of 2.2 and 1.6 mm respectively)

d) Model like (b), but with the same added leakage sources as model (c).

All the other parameters were maintained constant throughout the above simulations, complying with the nominal data of the compressor and the reference test, performed at 40 bar discharge. Other parameters of interest are:

- Polytropic efficiency on both compression and expansion: 90 %

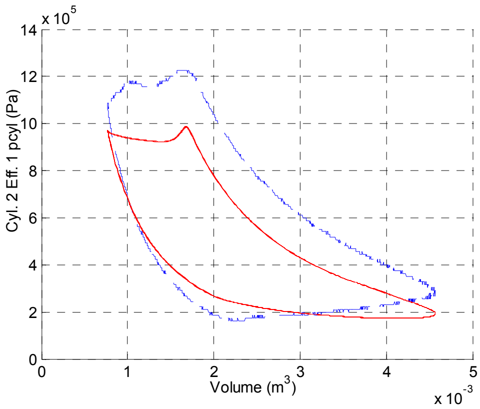
  yielding:

- Polytropic exponent on compression: 1.47

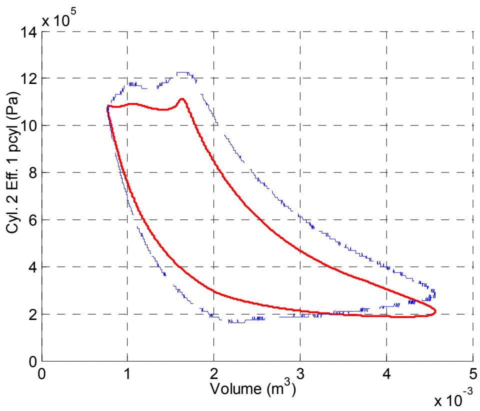
- Polytropic exponent on expansion: 1.35

4.3 Discussion

The comparison of the simulation results from models (a) and (b) from Picture 7 and Picture 8 shows the great importance of dynamic pressure pulsations in the piping, as to the settlement of the inter-stage pressures, even if the inlet and outlet average values are as expected (Picture 9 vs. Picture 10). But another influencing factor that must be accounted for in compressors modelling is the leakage, e.g. through the piston rings and the valves. Despite the relatively low values used for the examples provided in this paper, it is clear that adding these factors to the base models (a and b) approaches reality more closely.

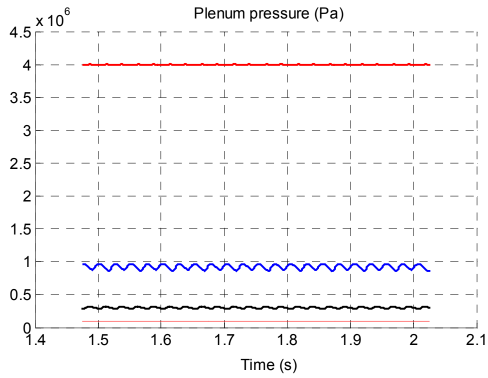


Picture 7 - Simulated *p-V* diagram of the 2nd stage – 1st effect cylinder of the tested compressor – model (a): quasi-steady non linear piping - no leakages.

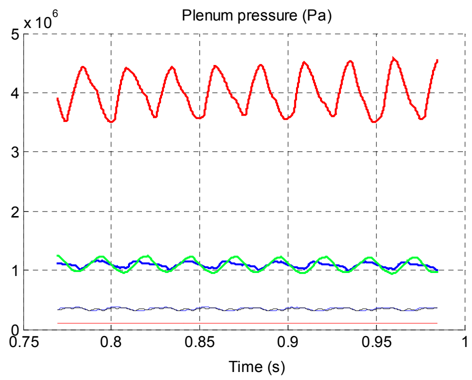


Picture 8 - Simulated *p-V* diagram of the 2nd stage – 1st effect cylinder of the tested compressor – model (b): dynamic linear piping – no leakages.

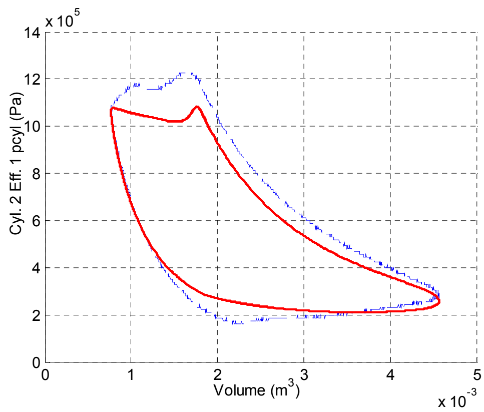
A direct comparison between Picture 11 and Picture 12 shows that simulation results present a better match with the experimental ones when the leakage factors are added, respectively to the model on quasi-steady piping and that on the dynamic piping. In both cases the effect is to increase the inter-stage pressure, as could be expected, to compensate for the reduced net flow rate. In the case of the dynamic piping the discharge pressure tends to be overestimated, with the given parameters. Other leaking elements have yet to be simulated at the time of writing this paper, as the work is still ongoing. However it is evident from these examples that exploring, by modelling the main factors shown herewith helps the understanding of the influence of some components on the overall performance.



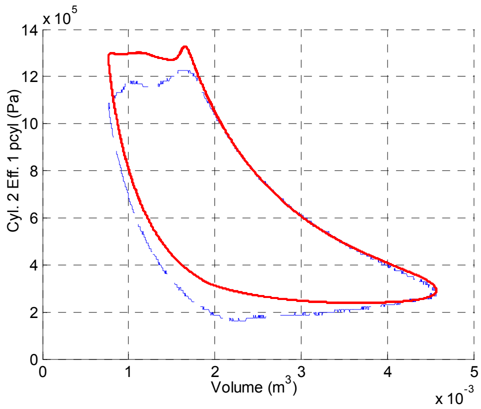
Picture 9 – Simulated plena pressures – model (a): quasi-steady non linear piping - no leakages.



Picture 10 – Simulated plena pressures – model (b): dynamic linear piping – no leakages.



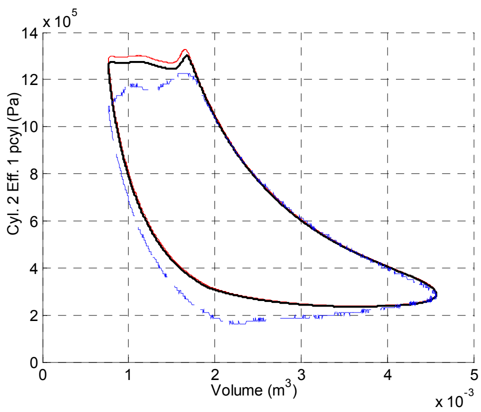
Picture 11 - Simulated *p-V* diagram of the 2nd stage – 1st effect cylinder of the tested compressor – model (c): quasi-steady non linear piping – with leaking piston rings and suction valves.



Picture 12 - Simulated *p-V* diagram of the 2nd stage – 1st effect cylinder of the tested compressor – model (d): dynamic linear piping – with leaking piston rings and suction valves.

4.4 Potential for diagnostic applications

The last result worthy of mention, from the work done so far, is the possibility to detect failures from clear signals changes. If the simulation with the model (d) is repeated with a leakage area increased by just 30% through the suction valve of the 1st effect of the 2nd stage, the *p-V* diagram changes again, showing a decreased discharge pressure on this effect, as shown in Picture 13, where the black line indicates the failed compressor cycle, overlapped to that of Picture 12 (solid line) and the experimental one (dashed line). This is just an example as previous work done on a different compressor demonstrated that the changes spread over all the signals, i.e. even plenum pressures and cylinders not affected directly by the failure.



Picture 13 - Simulated *p-V* diagram of the 2nd stage – 1st effect cylinder of the tested compressor – model (d): dynamic linear piping – with leaking piston rings and suction valves. Increased leakage through the suction valve of the 1st effect of 2nd stage cylinder (black line).

5 Conclusion

The work presented in this paper shows the possibility to predict and evaluate in quantitative terms the magnitude of pressure pulsations and their importance on the performance of reciprocating compressors, making simulations a valuable tool not only to assess a plant according to API 618 rules, but also to ensure a correct and efficient performance throughout its lifetime.

The influence of these dynamic factors and engineering parameters goes through various direct and indirect phenomena modifying the *p-V* diagram and valves flows of all the cylinders of a multistage compressor. Performances are also greatly influenced by leakages through e.g. the valves shutters, the piston rings and, though not explored in this work so far, through the rod seals.

The parallel activity on a real industrial compressor shows a reasonable match between simulated and test data, considering the uncertainties that are still pending on the measurements. The experimental work is also providing valuable specific experience about the problems to be encountered in the industrial environment, in the management of dynamic signals. A thorough critical analysis and new tests are pending at the time of writing this paper.

Finally this work shows the great potential of simulations in view of diagnostic applications. Indeed generating signal patterns from models of the normal operating conditions of a compressor, to be stored in a database together with other patterns generated on simulated faulty conditions, would allow faster symptoms detection and early warning on potential breakout. When a monitoring system is already installed on a facility the addition of diagnostic tools based on models would add only offline decision support software functionalities with no intrusion into the equipment hardware or controllers operation.

6 Thanks

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