ACUSCOMP and *ACUSYS* – A powerful hybrid linear/non linear simulation suite to analyse pressure pulsations in piping

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Abstract
The aim of this paper is to explain the theoretical and practical aspects of ***ACUSYS*** and ***ACUSCOMP***, a simulation suite for the analysis of pressure pulsations in piping and the prediction of compressor performances under realistic piping dynamics. ***ACUSYS*** is a tool to analyse the response of a generic piping system to predefined input signals generated at any interface with the gas flow. ***ACUSYS*** alone allows a preliminary linear analysis to focus the intrinsic piping response and flow resonance attitude in the frequency and time domain. When the interaction between the machine thermodynamics, the valves mechanics and the fluid exchange process are to be taken into account, together with the piping flow dynamics, ***ACUSCOMP***, in cooperation with ***ACUSYS***, allows the time domain non linear modelling of the entire system. Both these programmes are applications developed by the leading author and his collaborators for the MATLAB-Simulink® environment of The Mathworks Inc. (Natick, Mass., U.S.A.).

1 Introduction

ACUSYS, developed formerly by the leading author ([1], [2], [3], [5]) is a tool to analyse the response of a generic piping system, such as a plant or muffler, to predefined disturbance input signals generated at any interface with the gas flow, e.g. at the flanges connecting one or more compressors or engine cylinders to the piping suction and/or discharge side, or at piping obstructions or blind side branches exciting flow vortexes at specific Strouhal number conditions [8].

These disturbing signals (either in terms of pressure or mass flow rate) can be either calculated or estimated in advance by the user based on background or previous analyses. ***ACUSYS*** alone allows a preliminary linear analysis of the complex response to these sources by the piping. Also, ***ACUSYS*** allows transfer function and forces analysis to focus the intrinsic piping response and flow resonance attitude in the frequency and time domain.

However in some cases, particularly in the compressors industry field, this preliminary analysis focused only on the piping is not sufficient, because of the difficulty in making such a signals estimate which depend not only on the generating source, i.e. compressors or engines, but also on the response characteristics of the plant when coupled together. Widely used stringent rules such as API 618 [10] require indeed that such interaction between the machine thermodynamics, the valves mechanics, the fluid exchange process and the piping flow dynamics are taken into account since the design phase.

ACUSCOMP fulfils this further requirement and, in cooperation with ***ACUSYS***, allows the time domain non-linear modelling of the entire system including suction and discharge piping connected to a fluid-handling machine. In this paper a typical reciprocating compressor example is presented.

2 Main features of the suite

The ***ACUSYS / ACUSCOMP*** simulation suite is one of the few commercial software products capable to analyse unsteady flow in pipe systems integrated with the internal machine thermodynamics. Outstanding features are:

- Both programmes run on PCs;
- Extremely simple and user-friendly graphical interface;
- Fully integrated with the MATLAB-Simulink® environment and MS Excel®[1];
- Full exploitation of the MATLAB-

[1] MS Excel is a registered product of Microsoft Corp.

Simulink® functionalities, that means the possibility of inserting additional blocks, for example to analyse the rotary shaft mechanics or to integrate the model with control routines, either of the machine speed or of the valves (e.g. to implement variable flow capacity control).

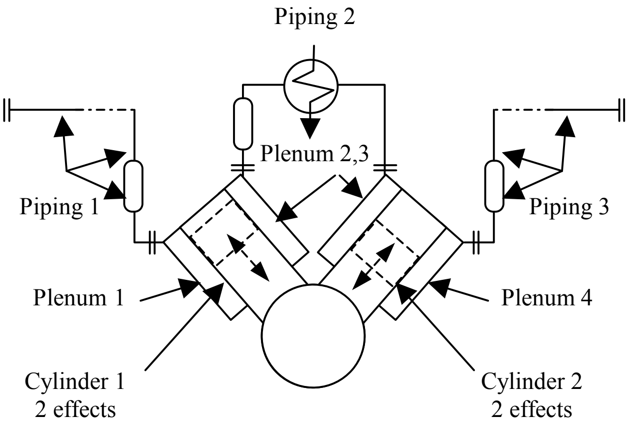

Fig. 1 Example of system suited for ***ACUSYS*** - ***ACUSCOMP*** simulations.

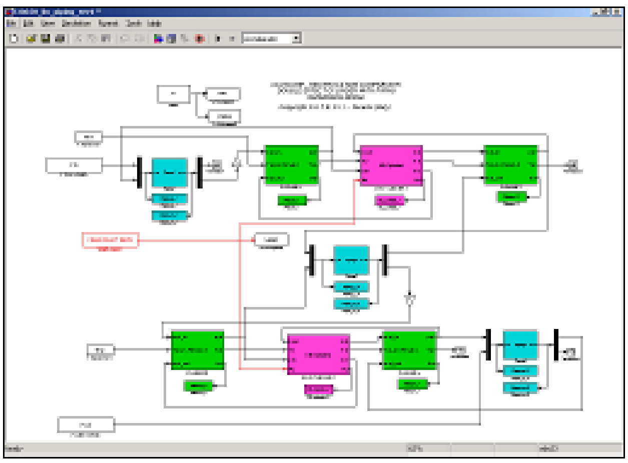

Fig. 2 ***ACUSCOMP*** model in the Simulink® environment of the system of Fig. 1.

The user can build up the ***ACUSCOMP*** and ***ACUSYS*** Simulink® models by easily assembling by very few vectorised lines the various component blocks (Fig. 2), or choosing one in a library of pre-assembled configurations. The blocks can be:

- *Piping state space matrixes* (Blue in Fig. 2), i.e. the overall dynamic characteristics of each piping side (suction, discharge or intermediate) connected to one or more cylinders. These state space matrixes are calculated by ***ACUSYS***.
- *Single or double effect cylinders* (Magenta in Fig. 2), which include, nested inside:
  - *Suction and discharge valve dynamics*, as function of either mechanical commands (i.e. cam shaft) or differential pressure drive, between the cylinder and the plant side (both dynamically calculated by ***ACUSCOMP***)
  - *Cylinder thermodynamics*, which include both the open phase (exchange with the piping) and closed phase (closed valves).
- *Plena* (Green in Fig. 2), 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 blocks to analyse the behaviour of the machine without the multimodal dynamic behaviour of the piping

In the data files only dimensional and engineering parameters must be defined.

3 The software architecture for a complete pulsation analysis

The operative sequence to perform a complete analysis and the input-output relationship between the two programmes can be summarised as in the scheme of Fig. 3.

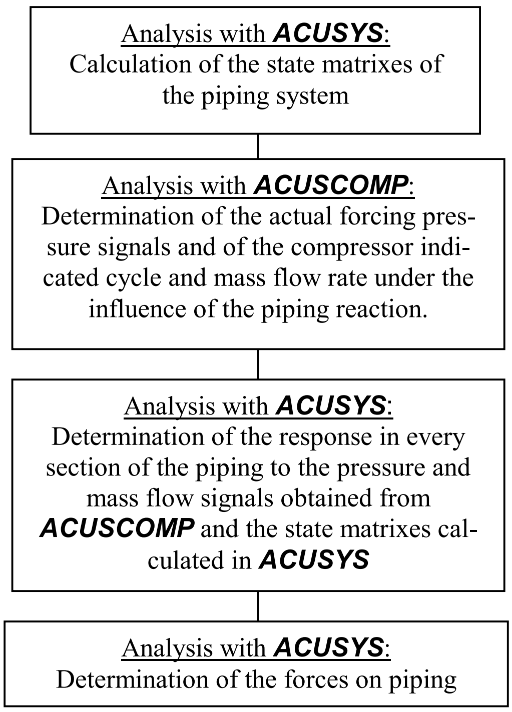

Fig. 3 Scheme for a complete pulsation analysis

Data are transferred to/from one program to the other using the potentialities of the MATLAB environment or through worksheet files, automatically edited and filled in by ***ACUSCOMP*** and ***ACUSYS***. This helps the user also in managing the data and editing freely their representation.

4 Theoretical approach of the model

In the following sections the mathematical approach, which was followed in the implementation of the model components, is described

4.1 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 waves 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:

$$f < 0.6\frac{c}{D} \tag{1}$$

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. The higher frequencies are dampened by the viscous effects and the sound wave scattering.

The fluid medium is considered bounded in pipes and containers, rigid in the axial and bending mode, with even variable diameter and arbitrary shape, such as pulsation dampers, heat exchangers, etc.

Besides the plane wave propagation model, associated to the range of frequencies studied, ***ACUSYS*** runs under the hypothesis of linear relationship between amplitudes of pressure variations and local fluid velocities, valid until the latter and the mean fluid velocity are small enough compared to the sound speed. Such hypothesis is better known as the condition for the electro-acoustic analogy.

The electro acoustic analogy allows a simplification of the continuity and momentum equations of the unsteady compressible flow in an elementary tube [7], as if they were written for an electric circuit like that of Fig. 4, taking into account also the friction losses effects.

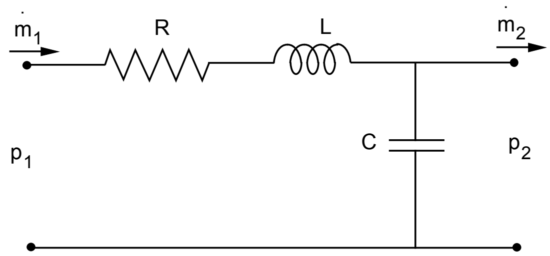

Fig. 4 Electro-acoustic analogy adopted in ***ACUSYS***.

$$\begin{aligned} &R\dot{m}_1 + L\frac{d\dot{m}_1}{dt} = p_1 - p_2 \ ; \\ &L = \frac{\Delta x}{A} \quad ; \quad R = \frac{\lambda}{2}\frac{\Delta x}{D}\frac{\dot{m}_0}{\rho_0 A^2} \end{aligned} \tag{2}$$

$$\begin{aligned} &C\frac{dp_2}{dt} = \dot{m}_1 - \dot{m}_2 \\ &C = \frac{A\Delta x}{c^2} \end{aligned} \tag{3}$$

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 linearization of the turbulent friction relationship as discussed below (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 1st harmonic period)

In linearising the friction losses attention must be paid to the criterion by which this is performed, as it must comply with the steady state relationship:

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

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

Indeed the correct usual approach for dynamic damping linearisation would be by the first derivative of Eq. (4) in the point of average operating flow (Fig. 5), yielding an expression for R, double compared to the one in Eq. (2).

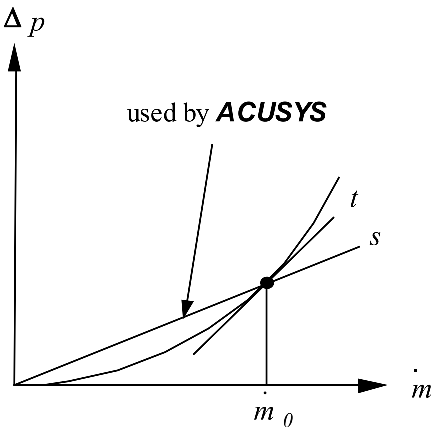

Fig. 5 Graphical representation of the two flow friction linearisation options (t = tangent or derivative, s = secant or average)

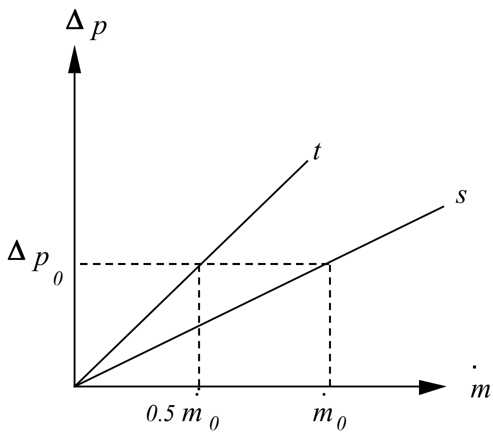

Fig. 6 Error in the calculation of the mean flow when using the derivative (tangent) friction resistance factor (Ref. to Fig. 5).

This would well approximate the damping of each harmonic component of the dynamic response, but under steady state condition, e.g. constant pressure difference signals between the two ends, it would yield a 50% error in the mass flow rate (Fig. 6). To match this condition it is instead convenient to use the expression for R given in Eq. (2), which means to linearise Eq. (4) by the "secant" approach rather than the "tangent" one (Fig. 5) and to underestimate the damping of the pressure pulsations in the piping.

It must be highlighted, however, that this approximation does not impair the validity of the dynamic analysis. Indeed the dynamic response of a very low damping system is affected by the damping factor only within a very narrow frequency bandwidth around the natural frequencies. Outside this it is well known that the response is mostly determined by the reacting component of the system impedance, i.e. capacitance and inductance, which are not affected by the above criterion for the damping definition. On the other hand, when such a system resonates, the pressure and flow amplitudes would be anyway so high that the whole electro-acoustic analogy would loose its validity in quantifying the actual response amplitudes. Nevertheless under these conditions the output of the model would fulfil anyway the need for highlighting critical situations.

4.2 Compressors

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 Fig. 7. For high flow rates relative to the volume or, more precisely, for short residence time of the gas in the volumes, they can be considered as adiabatic.

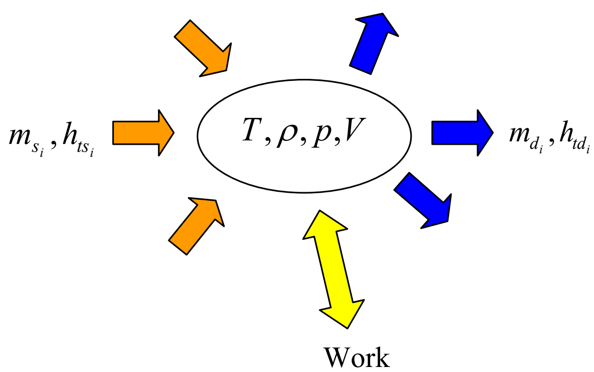

Fig. 7 General mass-energy balance representation of an element with variable volume

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.

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

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.

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

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 expression (11) of the energy balance inside V can be obtained, where $\gamma = {}^{c_p}\!/\!{}_{c_v}$.

$$u = \frac{U}{\rho V} = c_v T \tag{7}$$

$$h = c_p T \tag{8}$$

$$dL = pdV \tag{9}$$

$$\frac{p}{\rho} = \bar{z}RT \tag{10}$$

When considering a polytropic of exponent m instead of the reversible adiabatic transformation, it can be demonstrated that Eq. (11) modifies into Eq. (12).

The Eqs. (5) and (12) can be further adapted to the specific component to be modelled, i.e. cylinders or plena, as per the following subsections.

4.2.1 Cylinder thermodynamics

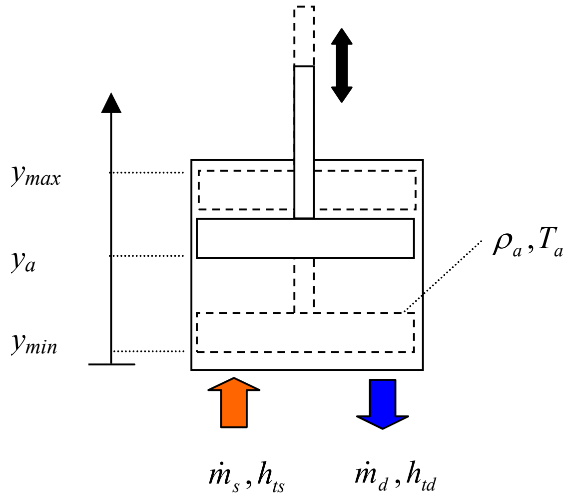

Fig. 8 Cylinder scheme

For a cylinder, as represented in Fig. 8, only one input and one output must be considered, while the variation of 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. (5) and (12) can be rewritten as (13) and (14).

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

$$\frac{dT}{dt} = \frac{m-1}{\gamma-1}\left[\gamma\frac{1}{\rho V}\sum_{i=1}^{n_s}\dot{m}_{s_i}T_{ts_i} - T\left(\gamma\frac{1}{\rho V}\sum_{i=1}^{n_d}\dot{m}_{d_i} + \gamma\frac{1}{V}\frac{dV}{dt} + \frac{1}{\rho}\frac{d\rho}{dt}\right)\right] \tag{12}$$

$$\frac{d\rho_a}{dt} = \frac{1}{A_a y_a}\left(\dot{m}_s - \dot{m}_d - A_a\rho_a\frac{dy_a}{dt}\right) \tag{13}$$

$$\frac{dT_a}{dt} = \frac{m-1}{\gamma-1}\left[\frac{\gamma}{A_a}\frac{\dot{m}_s T_{ts}}{\rho_a y_a} - T_a\left(\frac{\gamma}{A_a}\frac{\dot{m}_d}{\rho_a y_a} + \gamma\frac{1}{y_a}\frac{dy_a}{dt} + \frac{1}{\rho_a}\frac{d\rho_a}{dt}\right)\right] \tag{14}$$

4.2.2 Plena

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 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. (15) and (16) are derived from (5) and (12) deleting the derivative of V as this is constant.

$$\frac{d\rho}{dt} = \frac{1}{V}\left(\sum_{i=1}^{n_s} \dot{m}_{s_i} - \sum_{i=1}^{n_d} \dot{m}_{d_i}\right) \tag{15}$$

$$\frac{dT}{dt} = \frac{m-1}{\gamma-1}\left[\gamma\frac{1}{\rho V}\sum_{i=1}^{n_s}\dot{m}_{s_i}T_{ts_i} - T\left(\gamma\frac{1}{\rho V}\sum_{i=1}^{n_d}\dot{m}_{d_i} + \frac{1}{\rho}\frac{d\rho}{dt}\right)\right] \tag{16}$$

4.2.3 Valve dynamics

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 of relevance.

Based on the above balance the valves open only when a positive pressure difference exceeds the spring preload and the friction forces, like in the real compressors.

The mass flow through the valves orifices is calculated as function of the instantaneous valve opening, the pressure upstream and downstream and the temperature upstream the valve, i.e. in the cylinder or plenum.

Based on well-known literature [9] the valve reference flow section is calculated by Eq. (17).

$$A_r = 24.4\cdot 10^{-3}\cdot X_v A_v C_1\left[1+(\gamma-1)\frac{M_1^2}{2}\right]^{-\frac{\gamma+1}{2(\gamma-1)}} \tag{17}$$

In the above X_v is the valve opening, derived by the above shutter motion equation, A_v is the equivalent valve, full opening, flow coefficient provided by the valves manufacturers for water flow tests, C_1 is the gas correction factor and M_1 is the Mach number upstream the valve, usually negligible, compared to the vena contracta condition.

Since in the flow through an obstruction all pressure losses are localised downstream the vena contracta, the total pressure in the latter equals that of the upstream flow. Knowing this pressure, from the adjacent volume balances described above, back-solving Eq. (18) allows the calculation of the Mach number in any section, in particular the vena contracta, while the other Eqs. (19), (20), (21) and (22) define all the static and total thermodynamic variables in the same section.

Under choking conditions, i.e. critical flow at Mach 1, the mass flow reaches its maximum value, depending on the upstream conditions.

$$\frac{\dot{m}(RT_t)^{0.5}}{Ap_t} = \frac{\gamma^{0.5}M}{\left[1+(\gamma-1)\frac{M^2}{2}\right]^{\frac{\gamma+1}{2(\gamma-1)}}} = F(M) \tag{18}$$

$$T = \frac{T_t}{1+\frac{\gamma-1}{2}M^2} \tag{19}$$

$$p = p_t\left(1+\frac{\gamma-1}{2}M^2\right)^{\frac{\gamma}{1-\gamma}} \tag{20}$$

$$\rho_t = \frac{p_t}{zRT_t} \tag{21}$$

$$\rho = \frac{p}{zRT} \tag{22}$$

The gas compressibility factor z used in the above equations is assumed as the mean value between those calculated at suction and discharge conditions.

The above generalised orifice model allows calculating both the flow through compressors valves, which is normally subcritical, and that through leaking seals or valve seats, which instead can be chocked, as function of the instantaneous gas conditions on the two sides of the obstruction (e.g. the pressure in the two volumes of a double effect cylinder).

4.2.4 Restrictions

The generic orifice model, which was presented in the previous section, can also be used to model the flow through any other restrictions, e.g. equivalent obstructions or loss elements that can be used in place of an entire piping. This would allow to preliminarily analyse the behaviour and cycle of a compressor disregarding the multimodal dynamic behaviour of the piping discussed in section 4.1, as it is described in sec. 5.

4.3 Forces on piping

The integrated ***ACUSYS-ACUSCOMP*** suite allows also, as the final step, to calculate and analyse the forces generated by the pressure pulsations at the pipe bends or junctions, as the input for a dynamic structural analysis, to verify possible structural resonant vibrations, as foreseen for instance by the API 618 rules [10].

The elementary node of Fig. 9 represents the bend between two intersecting pipe elements.

Under the hypothesis that the bend or joint extension is small compared to the sound wave lengths in the frequency bandwidth considered, the pressure p is constant along the node at each time; the total force acting on it can then be calculated as the sum of two contributions, defined by Eqs. (23) and (24). The former is the vector sum of the pressure forces; the latter is the change of momentum of the flow, between the two node ends at time t.

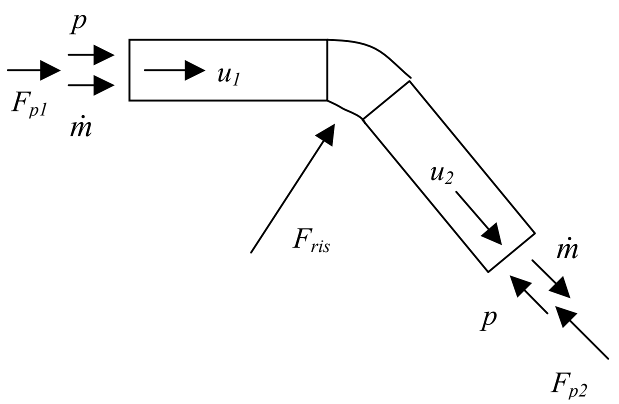

Fig. 9 Elementary node representation

The resultant is directed along the bisecting line of the angle formed by the two piping elements, as represented in Fig. 9. In Eq. (25) the pressure p and the mass flow $\dot{m}$ are functions of time, while ρ and A are assumed to be constant in the node.

$$\overrightarrow{F_p} = pA \cdot (\overrightarrow{u_1} - \overrightarrow{u_2}) \qquad (23)$$

$$\overrightarrow{F_{dev}} = \frac{\dot{m}^2}{A\rho_0} \cdot (\overrightarrow{u_1} - \overrightarrow{u_2}) \qquad (24)$$

$$\overrightarrow{F_{ris}} = \left(pA + \frac{\dot{m}^2}{A\rho_0} \right) \cdot (\overrightarrow{u_1} - \overrightarrow{u_2}) \qquad (25)$$

5 Results

In this chapter the main typical results, which can be obtained by means of ***ACUSYS*** and ***ACUSCOMP***, are presented.

An example two-stage/two-effects air compressor is considered, interfaced with suction, interstage and discharge piping (Fig. 1), with the following nominal characteristics:

- Mass flow rate 0.1169 kg/s
- Inlet pressure 8 bar
- Intercooler pressure 20 bar
- Outlet pressure 46 bar
- Revolution speed 710 rpm

As explained above the time domain analysis was performed in ***ACUSCOMP***, after calculating by ***ACUSYS*** the three sets of the piping state space matrixes. After loading the data file and running the ***ACUSCOMP*** model, the following cylinder variables, referred to each effect, can be plotted:

- Piston stroke;
- Piston velocity;
- Cylinder pressure;
- Cylinder temperature;
- Suction and discharge valves boundary pressures;
- Suction and discharge valves differential pressure;
- Suction and discharge valves flow rates;
- Suction and discharge valves opening.

For each plenum the following plots are available:

- Pressure;
- Temperature;
- Input mass flow rate;
- Output mass flow rate.

The piping end pressures and flow rates, or equivalent restrictions ones, can be plotted as well.

Parameter	**D.P.**	**Restr.**
Indicated work (kJ)	0.566	0.606
Indicated torque (Nm)	90.0	96.5
Indicated power (kW)	6.70	7.17
Volumetric efficiency	0.854	0.844
Compression ratio	2.49	2.46
Ave. mass flow rate (kg/s)	0.1294	0.1290

Table 1 Global performance results: 1st stage 1st effect cylinder (D.P. = Dynamic Piping, Restr. = Piping equivalent restriction)

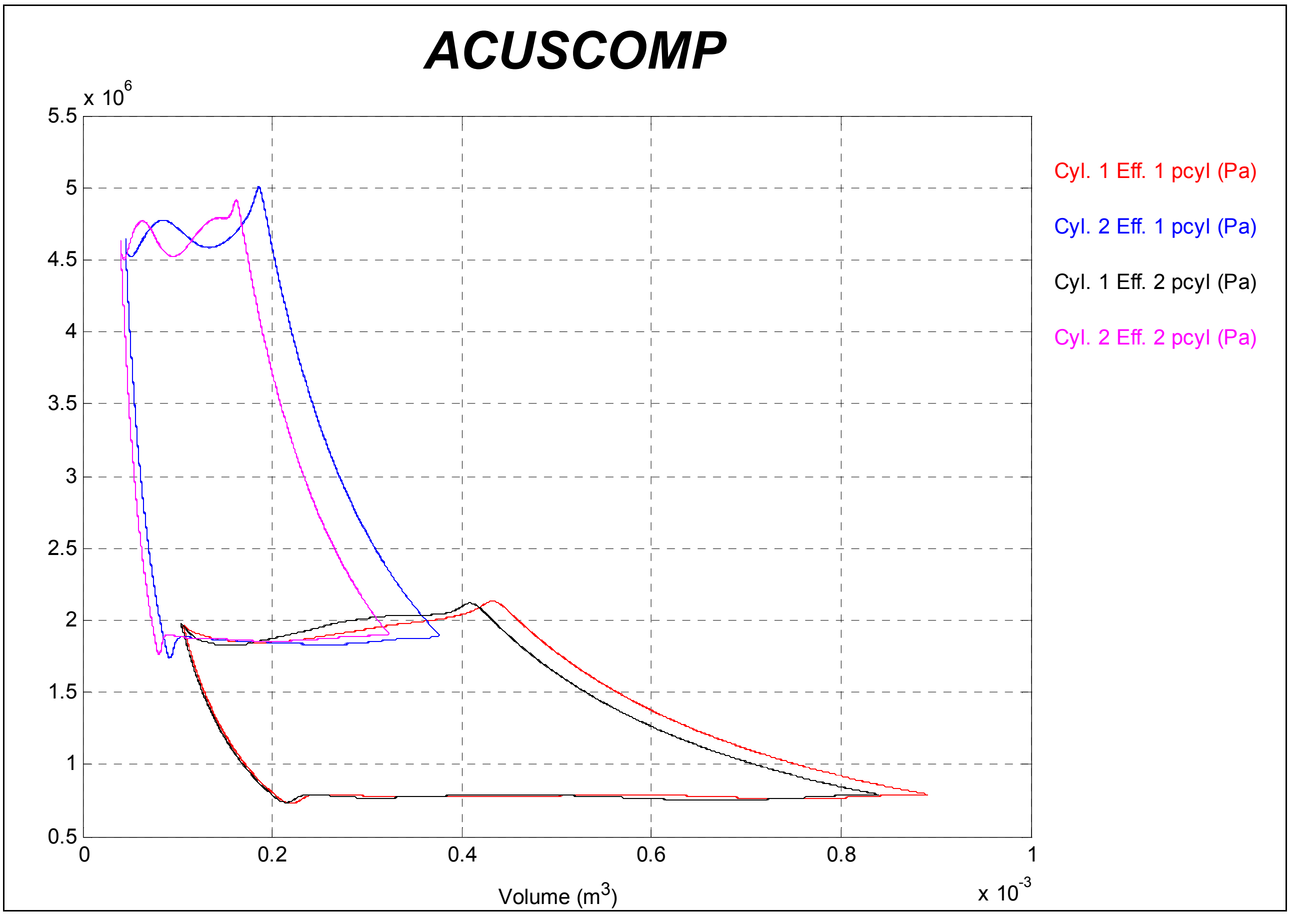

Fig. 10 Indicated cycles calculated with ***ACUSCOMP*** and dynamic piping, for the example case.

Each of the variables listed above can be compared in the same plot with all the others and can be displayed versus four different abscissas:

- Time;
- Cylinder volume
- Crank angle;
- Piston position.

For the example considered, the results for a stabilised cycle of the first effect of the first stage cylinder are shown in Table 1. The indicated cycles for all the stages and effects are instead represented in Fig. 10. This case shows the influence of the dynamic piping characteristic as a wavy pressure signal above all during the discharge phase of the 2nd stage. The two effects show also some obvious signal phase shifts.

Besides the in-cylinder variables, the most important outcomings for the pressure pulsation analyses are the input signals to be provided to the third step of the analysis (Fig. 3), namely the pressure signals at the input sections of the piping, i.e. the pressure in the plena. These can be exported to ***ACUSYS***, for the time and frequency analysis in all piping sections, yielding the response in terms of pressure (Fig. 11) or mass flow rate.

To highlight the influence of the piping dynamics on the compressor performance and signals generated at the interface with the piping, a similar analysis is shown for the same compressor, with pure restrictions instead of the complete dynamic piping models, as mentioned in Sec. 4.2.4.

The other characteristics of the system, such as cylinders, valves and plena are kept unvaried.

The global performances do not vary appreciably in this case (Table 1) compared to the one previously described, as no great pulsations occur: with the piping dynamic model the performances predicted are slightly better (lower indicated power). This relative invariance of performances can be explained by the lack of great pressure pulsations in the piping and plena. Indeed from the subsequent ***ACUSYS*** analysis it results that the response at the intermediate piping sections are lower than the input signals (Fig. 11). The slightly lower torque and power with the dynamic piping model can be explained by the favorable depression waves at the end of the gas discharge (Fig. 12 and Fig. 13) particularly in the 1st stage.

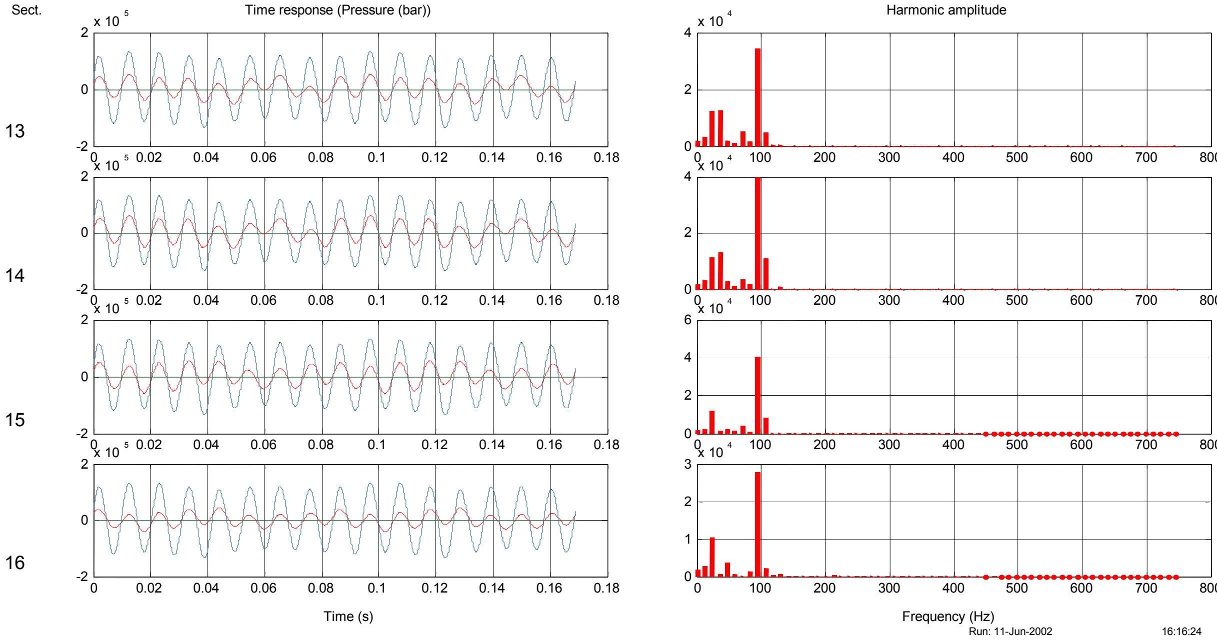

Fig. 11 ***ACUSYS*** pressure pulsations in the suction piping of the test case (the red line indicates the response, while the blue and green lines indicate the input signals)

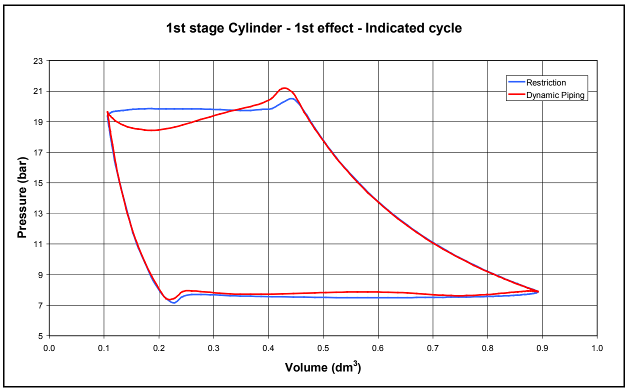

Fig. 12 1st stage P-V diagram; dynamic vs. restriction piping.

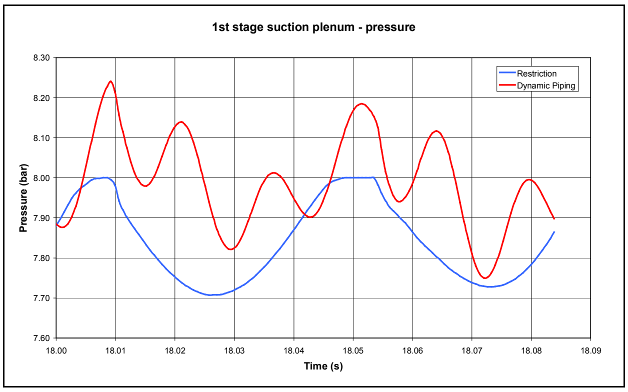

Fig. 14 1st stage suction plenum pressure; dynamic vs. restriction piping.

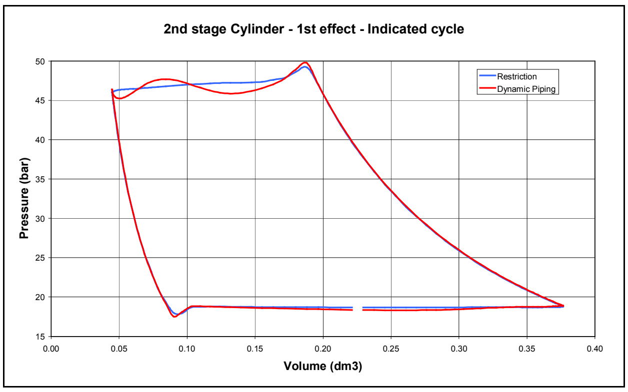

Fig. 13 2nd stage P-V diagram; dynamic vs. restriction piping.

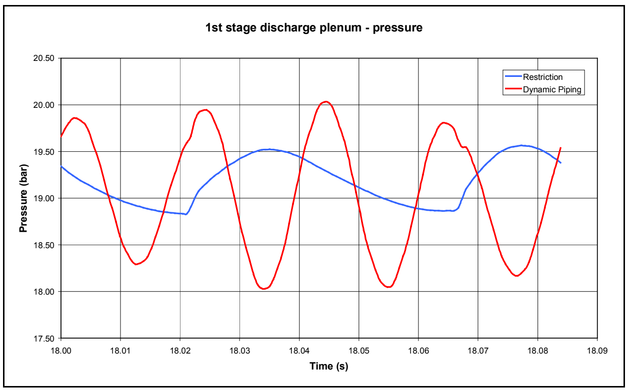

Fig. 15 1st stage discharge plenum pressure; dynamic vs. restriction piping.

Despite the apparent equivalence of the two models when looking at the overall performance, important differences between them are evident when looking at the pressures signals in the plena

(Fig. 14 and Fig. 15), which imply important differences in the results from the ***ACUSYS*** analysis of the pulsations in the whole piping.

The final step of a typical pressure pulsation analysis is then the calculation of the forces generated by the fluid at the nodes of the piping. The layout of the latter can be visualised in ***ACUSYS***, as shown in Fig. 16, as a 3D coloured pipe showing also the pressure or flow rate instantaneous values and the nodes of forces reduction.

As an example of the output, Fig. 17 represents the forces along x, y and z in four nodes of the inlet piping of the compressor model.

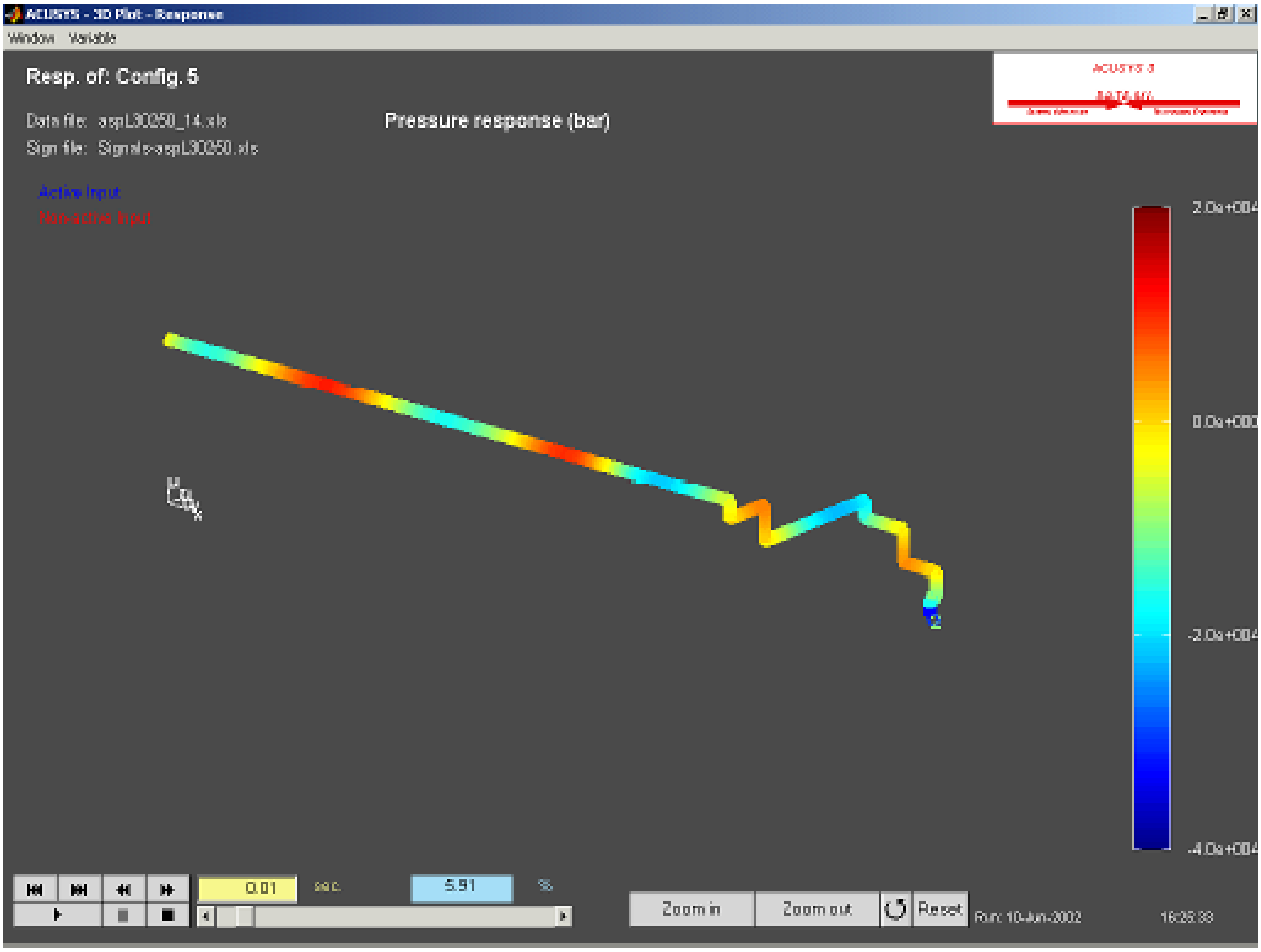

Fig. 16 ***ACUSYS*** - 3D representation with colored pressure levels of the 1st stage suction piping of the example compressor; the buttons allow scrolling of pressure patterns along with time.

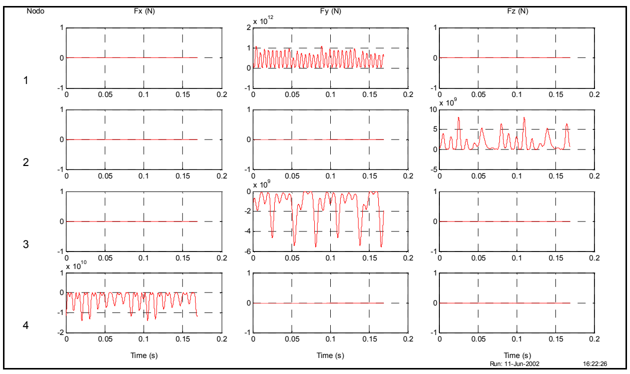

Fig. 17 ***ACUSYS*** - Forces at four nodes of the inlet piping of the example compressor.

6 Conclusions

The work presented in this paper is the result of a multiyear development aimed at providing engineers a flexible and effective tool for the analysis of pressure pulsations in plants. ***ACUSYS*** and ***ACUSCOMP***, despite their apparent simplicity embed a sophisticated modelling approach that allows reliable simulations of reciprocating compressors performance, as well as the analysis of other fluid piping systems, when transients or other sources of pressure pulsations are of concern. The past authors' experience in using these tools also for their services in a variety of situations showed the effectiveness and usefulness of the simulations to prevent dangerous situations or to understand previously unexplained pipe vibration phenomena found in the real plants operations.

The work on this subject is in continuous progress and will encompass also new tests and comparison among models and experimental data.

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