PRELIMINARY VALIDATION OF *ACUSYS*® TO ANALYSE INSTABILITIES IN THE NON LINEAR COMBUSTION ZONE - PLANT ACOUSTIC INTERACTION

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ABSTRACT

Aim of this paper is to demonstrate that the MATLAB/SIMULINK® application ***ACUSYS***, developed formerly by the leading author to analyse pressure pulsations in plants, with the addition of a non linear combustion zone block, based on the κ - τ formulation reported by McManus & others, is suited for the study of instabilities dominated by acoustics and their control. ***ACUSYS***, which models the medium by linear monodimensional finite elements, was applied to the simple case of a ducted premixed flame solved analytically in said previous works. The results confirm by great accuracy the theoretical predictions of the time lag ranges for non linear instability, duly considering the shift of the natural frequencies implied by the medium discretisation. The automatic setting of the pipe finite elements length, based on eigenvalues accuracy criteria, proved also appropriate to predicting modal instabilities over all the range of validity of the plane wave (monodimensional) and linear propagation approach. ***ACUSYS***'s attitude to handle complex piping geometries and parameters variations along tubes, can now be fully exploited in the preliminary analysis of active combustion control.

1. INTRODUCTION

Combustion instabilities and the techniques for their control are the subject of many undergoing R&D activities for a variety of applications. These include utility boilers, furnaces, gas turbine and rocket combustors.

Instabilities create large pressure fluctuations and can increase the heat transfer rate to the combustor surfaces, these two effects having opposite consequences, as they can be detrimental or even cause of failure on one side or beneficial, as to combustion efficiency and pollutants emissions, on the other.

Control and reduction of instability effects cannot be conveniently obtained by a trial and error approach on the hardware, due to the various complex and non linear phenomena which are intervening. However the lack of predictive tools to help the design of stable combustors yielded so far a reduction of their operating range to prevent instability onset.

McManus & others (Ref. 7), reporting a vast work by other authors and himself, discusses in a very comprehensive way the classes of instabilities and techniques proposed to control them; he highlights that acoustic coupling between the combustion zone and the fluid medium is one major and very common mechanism by which instabilities occur.

Other important mechanisms are fluid dynamic instabilities, i.e. turbulence induced periodical structures (vortexes), intrinsic flame thermo-chemical instabilities and, though limited to very special and intense heat rates, shock wave instabilities. The techniques proposed to control these act mostly on the acoustic or on the fluid dynamics of the combustor, the former being most common, e.g. by loudspeakers or variable flow choking devices fed back by microphones and/or flame emission optical sensors.

Despite the above mentioned non exclusive role of acoustic instabilities, there is a great number of cases where this mechanism proves predominant or synergistic with the others.

Benelli & others (Ref. 1) studied the non linear thermo-acoustic vibrations in a pulsating combustor by a general purpose CFD code on a fairly simple axisimmetric geometry. This approach, useful to analyse the combination and interaction of such phenomena, however requires large computing power to handle the acoustic behaviour of complex plants. The authors deem that a combined approach having said sophisticated analysis as the final or intermediate step, but starting and iterating from a thorough comprehension of the acoustic instabilities would be convenient in most cases of practical interest.

Therefore an important step towards understanding, predicting and controlling combustion instabilities goes anyway through the accurate modelling of the acoustic characteristics of plants, mostly linear even with complex geometries, and their interaction with the non linear combustion phenomena.

This paper deals indeed with the demonstration that the MATLAB/SIMULINK application referred to as ***ACUSYS***, developed formerly by the leading author (Refs. 2, 3, 4) to analyse pressure pulsations in compression plants, with the addition of one non linear block, describing the combustion zone, is suited for the study of acoustically dominated instabilities and their control. On this purpose ***ACUSYS*** was applied to the simple case reported and solved analytically by McManus & others (Ref. 7). The two results are compared in the following sections.

2. DESCRIPTION OF *ACUSYS*

ACUSYS (Refs. 2, 3, 4) is an application software, developed in the numerical MATLAB/SIMULINK environment, simulating in the time and frequency domain the dynamic response of the fluid medium inside a plant, induced by pressure or flow pulsations applied to discrete points.

The basic version of ***ACUSYS*** simulates the stationary response of the system to arbitrary periodic inputs, but the extended version used for this work allows the analysis of transients due to non periodic actions (e.g. start up or stop manoeuvres, water hammers, etc.).

ACUSYS library models can be modified to include non linear blocks suited to describe the behaviour of specific zones of the medium, such as a combustion zone.

2.1. *ACUSYS*' mathematical basis

The model of a plant implemented in ***ACUSYS*** with MATLAB and SIMULINK is essentially a finite elements model of a monodimensional 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, in formula:

$$f < 0.6\frac{c}{D} \qquad \text{Eq. 1)}$$

where f is the frequency of a generic signal component, c the sound speed inside the fluid and D the maximum dimension orthogonal to the propagation direction. This condition is usually verified inside combustion plants. For example for a combustor having a diameter of 60 cm and a sound speed of 740 m/s this cut-off frequency is of 740 Hz. Instabilities usually are associated to an even lower frequency bandwidth, the higher frequencies being dampened by the viscous effects and the wave scattering itself.

The fluid medium is considered bounded in structurally rigid pipes and containers with even variable diameter and arbitrary shape, such as pulsation dampers, heat exchangers, etc. However, pipe wall elasticity, if significant, can be accounted for by duly lowering the sound speed value assigned to each pipe segment, as common in water-hammer calculations in hydraulics.

Plant architecture is defined by configuration schemes within an expandable library, with dimensional parameters definable by the user.

Besides the plane wave propagation model, associated to the range of frequencies studied, the base version of ***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.

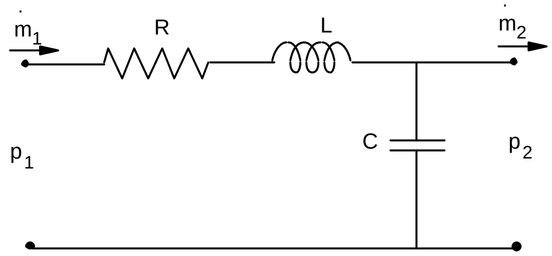


Fig. 1 - Electro-acoustic analogy adopted in ***ACUSYS***.

The electro acoustic analogy allows a description of the continuity and momentum equations of the fluid in an elementary tube as if they were written for an electric circuit like that of Fig. 1, taking into account also the friction effects:

Continuity: (a)

$$R\dot{m}_1 + L\frac{d\dot{m}_1}{dt} = p_1 - p_2$$

$$L = \frac{\Delta x}{A} \qquad R = \lambda\frac{\Delta x}{D}\frac{\dot{m}_0}{\rho_0 A^2}$$

Momentum: (b)

$$C\frac{dp_2}{dt} = \dot{m}_1 - \dot{m}_2$$

$$C = \frac{A\,\Delta x}{c^2}$$

Eqs. 2)

In Eqs. 2 the following notations and definitions are used:

R, L, C	equivalent resistance, inductance, capacity of the element respectively
m, p	flow rate and pressure of the fluid at sections 1, 2
t	time
A	area of the cross section of the tube
Δx	length of element
c	sound speed of fluid inside the element
λ	friction loss factor per unit length

D	diameter of the tube
ρ	mass density of fluid
0	subscript indicates mean value

By combining in series many identical elements like that of Fig. 1, a multi degree of freedom, discrete tube model is built up (Fig. 2). The number of elements necessary to accurately describe the dynamics of a continuous tube within a given frequency bandwidth is determined automatically by the program algorithms (Ref. 4), such that the most significant dynamic modes, in the frequency range considered, are well approximated.

In such a multi element set, the input-output relationship among pressures and flow rates are those highlighted in Fig. 2, by the thick arrows. Different combinations (e.g. pressure input at both sides, flow rate output at both sides) are obtained by eliminating one capacity or inductance at either the tube ends.

The mathematical model here briefly described has been implemented suitably combining SIMULINK, as a flexible tool for the creation of plant configurations models, with MATLAB, as a versatile fourth generation language effective in matrices analysis, for the calculation of the usually large state matrices of basic components, such as tubes (Fig. 3), capacities, manifolds, etc., available in ***ACUSYS***' library.

Should the above linearity assumption be defective, the SIMULINK model of ***ACUSYS*** can be modified using non linear tube elements, such as in a further extended version of the program, now at prototype level.

A deeper insight in ***ACUSYS*** functional architecture and user-friendly graphical interface was reported in earlier works by the leading author (Refs. 2, 3, 4) documenting this product development and its use.

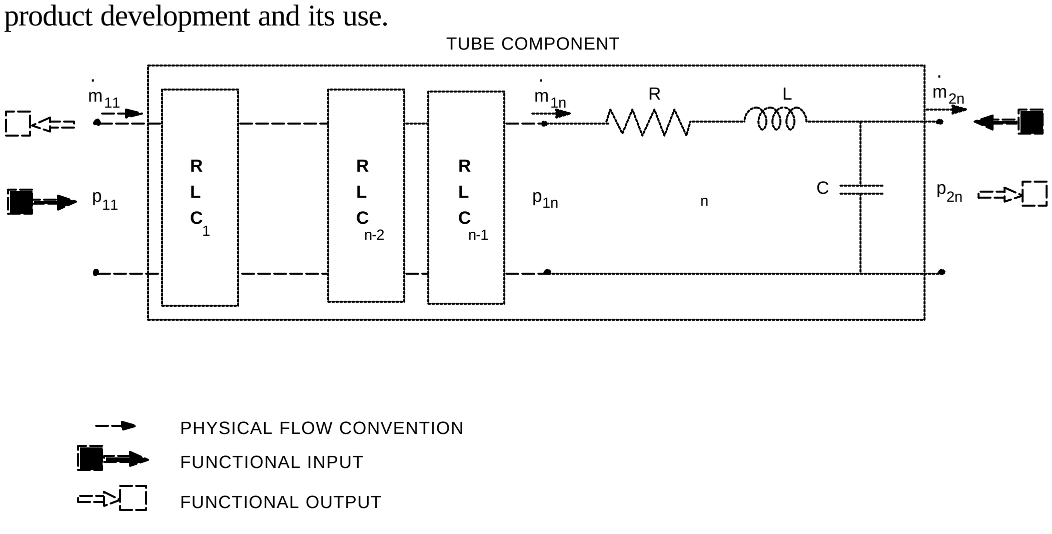


Fig. 2 - Build up of a tube component by RLC serial elements

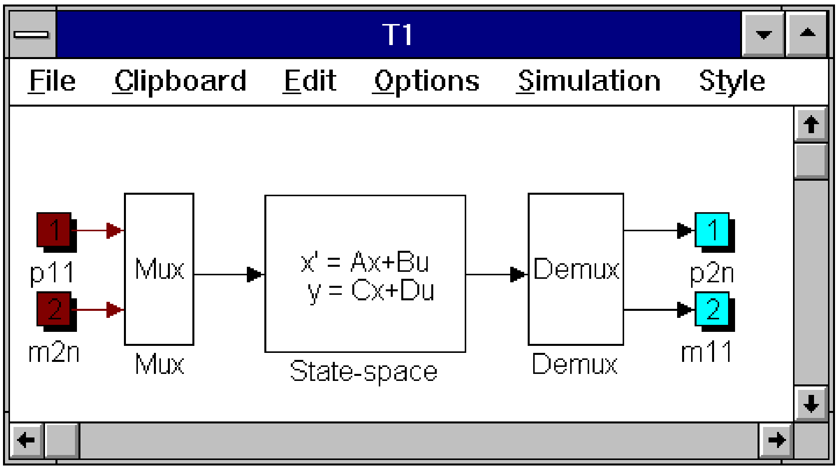


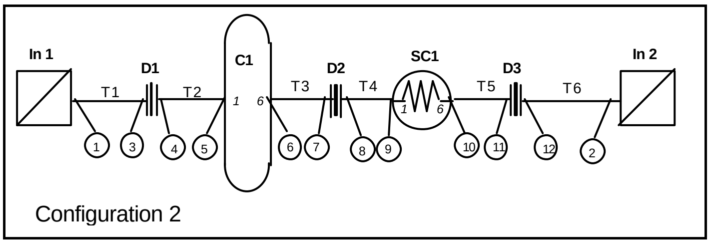
Fig. 3 - Example of the inner structure of a component block used in SIMULINK S-files of ***ACUSYS*** (tube element with pressure and flow rate inputs)

2.2. Typical plant configurations

In Fig. 3 are shown two examples of plant configurations, of different complexity, which are provided in the ***ACUSYS*** systems library. These examples refer to compressors piping systems, common in medium or large multi-stage units, for which the program was first applied.

Each numbered section corresponds to a point of the plant for which transfer functions and responses are calculated, under a wide set of input signals.

These sections, according to the lengths of tubes connecting them, can represent any point in the plant branch they belong to. For instance, to calculate the optimal position of an orifice or a capacitive filter in a branch, the user can run the program repeatedly, each time assigning different and complementary lengths to the two tubes adjacent to said element.



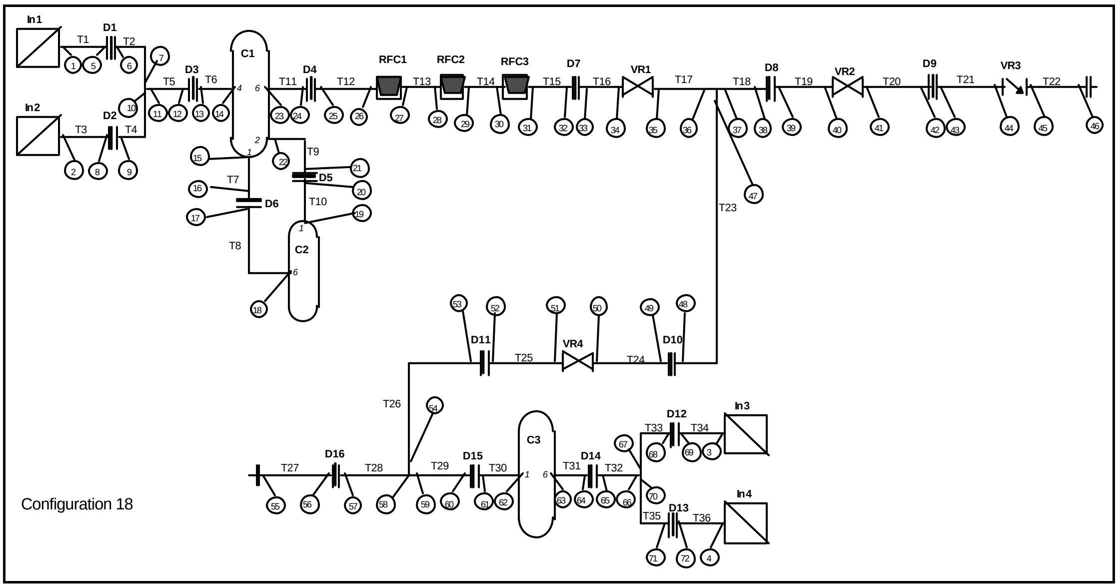


Fig. 3 - Examples of plant configurations in the ***ACUSYS*** library.

3. TEST CASE DESCRIPTION

ACUSYS algorithms were thoroughly compared with analytical results available in the literature for linear acoustic cases.

Since the basic acoustic instabilities mostly involve the frequency range and amplitudes where the electro-acoustic approximation is applicable, the authors wished to verify the results of transients simulations obtained by ***ACUSYS***, when duly upgraded with a non linear model block describing a discontinuity combustion surface, as per the κ - τ formulation reported by McManus & others (Ref. 7) and Crocco (Ref. 6), which is useful to predicting the quantitative behaviour of combustion oscillations in many cases. This does not bounds ***ACUSYS*** application to this formulation as the SIMULINK environment allows an unlimited choice of functions to describe the discontinuity. Rather, after this validation, other combustion zone models could be analysed under the same assumptions on the fluid medium, and compared to each other.

3.1. Test case hypotheses and model

The case analysed by McManus & others is shown in Fig. 4 and refers to a ducted premixed flame. The assumptions applied for its analytical solution were the following (Ref. 7):

(i) The frequencies of the acoustic waves considered are low compared to the duct cut-off frequency and these waves correspond to plane waves propagating along the longitudinal duct axis.

(ii) The dissipation of acoustic waves throughout the duct is negligible.

(iii) The open end of the duct corresponds to a pressure node and the close end to a pressure antinode.

(iv) The mean flow Mach number is small, thus the flow velocity is negligible when compared to the sound speed.

(v) The flame length is small compared to the acoustic wavelength so that the region of heat release may be approximated as a thin sheet located at a given axial location $x=X_{1\_2}$ (Fig. 4).

It can be easily seen that hypotheses (i) to (iv) are identical to those of the ***ACUSYS*** base modelling algorithms summarised above.

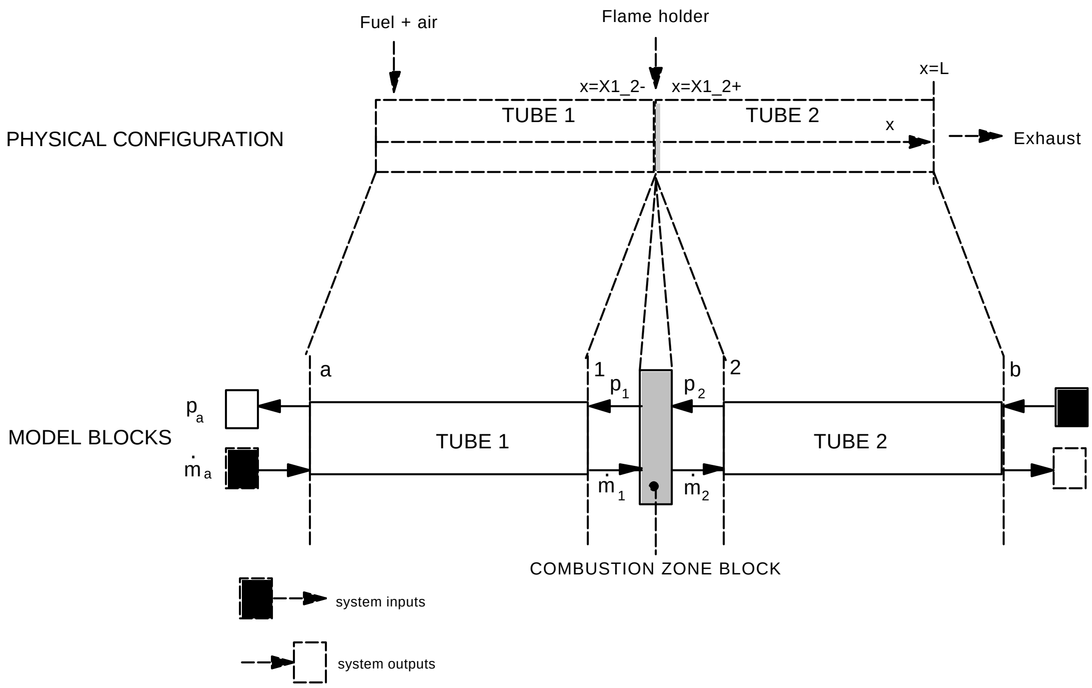


Fig. 4 - Test case physical configuration and model

The combustion zone equations of the "κ-τ" model to be implemented and combined with the acoustic field equations are the following:

$$p_1(t) = p_2(t) = p_{1\_2} \qquad \text{Eq. 3a)}$$

$$\dot{m}_2(t) = \dot{m}_1(t) + \kappa \dot{m}_1(t-\tau) \qquad \text{Eq. 3b)}$$

where $0 < \kappa < 1$ (typically $\kappa \approx 0.2 - 0.3$)
and $\tau \approx 10 - 100$ ms

The former equation represents the equilibrium of pressures across the discontinuity surface, whereas the latter represents the effect of the heat release on the fluid velocity and mass flow rate, which is empirically associated to a gain and a time lag. The two quantities κ and τ are constants, the former, dimensionless and called "interaction index", describes the intensity of the coupling between the flow velocity (or mass flow rate) and the heat release oscillations, while τ, in seconds, is the time lag of such interaction.

It is not the scope of this paper to discuss this assumed formulation, for which the reader is addressed to the cited works, but rather to verify whether ***ACUSYS*** determines correctly the conditions, namely the values of κ and τ, under which instabilities occur, as foreseen by McManus & others in this simple case (Ref. 7).

On this purpose the test configuration implemented, including two tubes in series, with the combustion zone model in between (Fig. 4), included the functional block of Fig. 5. This block allows applying the "κ - τ" model pattern only to the frequencies falling below the threshold frequency for the monodimensional propagation hypothesis, for the reasons which will be clearer in the following.

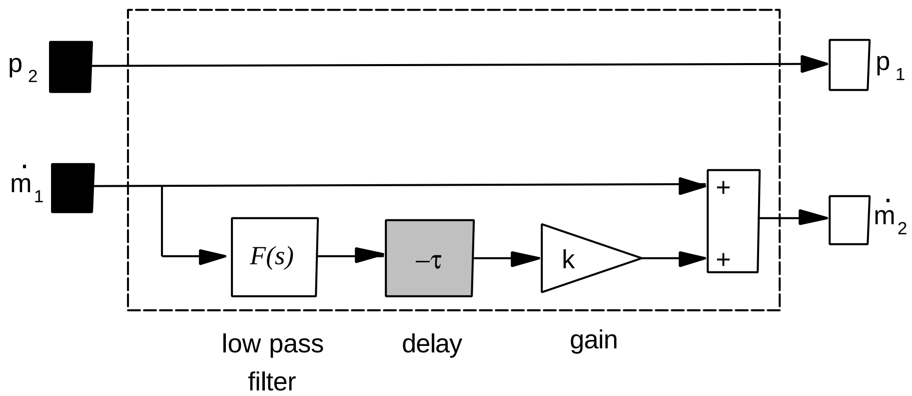


Fig. 5 - ***ACUSYS***' Combustion zone block, following the "κ-τ" model

3.2. Theoretical results

McManus & others (Ref. 7) showed that the instability of the system under discussion is solely associated to the time lag τ. The parameter κ is instead proportional to the rate of change of oscillations.

Instability is shown to depend on the ratio between τ and the natural period T_z of each eigenmode of the whole tube (z = integer). In formulas the intervals of τ values where instability occurs are defined for the first four modes by the following inequalities:

$$T_1 = 1/f_1 \quad ; \quad \left(j+\frac{1}{2}\right)\frac{1}{f_1} < t < (j+1)\frac{1}{f_1}$$
$$T_2 = 1/3f_1 \quad ; \quad j\frac{1}{3f_1} < t < \left(j+\frac{1}{2}\right)\frac{1}{3f_1}$$
$$T_3 = 1/5f_1 \quad ; \quad \left(j+\frac{1}{2}\right)\frac{1}{5f_1} < t < (j+1)\frac{1}{5f_1} \qquad \text{Eqs. 4)}$$
$$T_4 = 1/7f_1 \quad ; \quad j\frac{1}{7f_1} < t < \left(j+\frac{1}{2}\right)\frac{1}{7f_1}$$

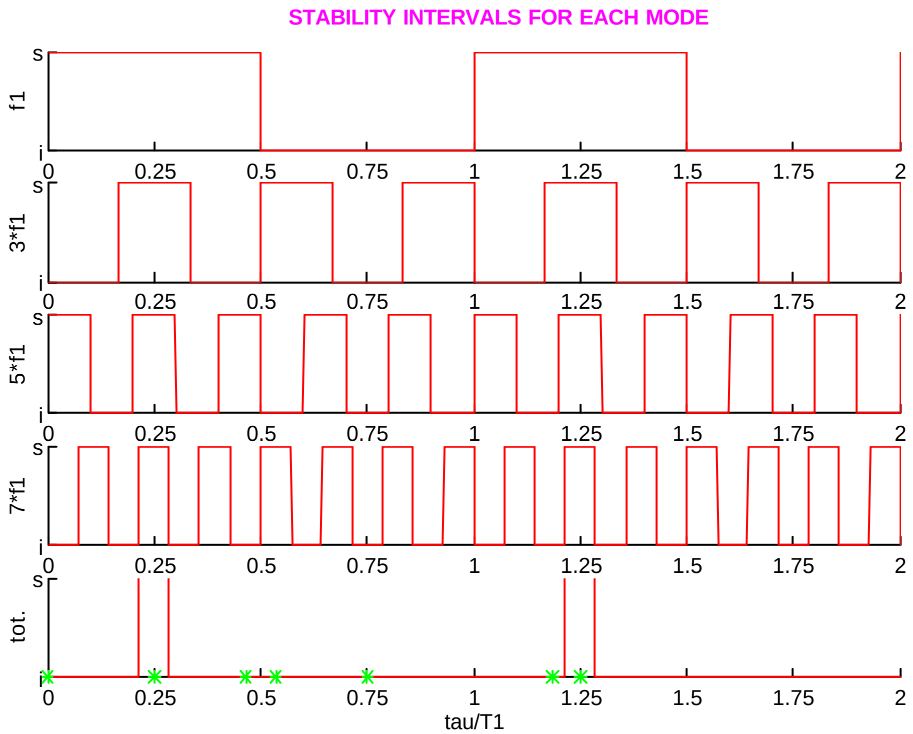
where:

$$f_1 = \frac{c}{4L} = \text{quarter wave mode frequency}$$

j = 0, 1, 2, 3, ...

The extension to higher order modes can be easily obtained by formal analogy with the above, as also explained later.

Since in general the dynamics of a system, characterised by a given τ (or range of τ values), includes a spectrum of frequencies, as function also of the input signal, the response will be stable only if τ is within the intersection of the complementary stability intervals of all modes to be considered. This is shown for example for the first four modes in Fig. 6.



CASES→ 1 7 4 5 2 3 6

Fig. 6 - Stability intervals for the system of Fig. 4 and for the first four modes, as predicted by the "κ-τ" model. In the plots "s" stands for "stability", "i" for "instability"; the lowest plot gives the intersection of these four modes stability ranges. Bottom numbers from 1 to 7, correspond to the (*) character in the lowest axis and identify the main validation cases.

It is evident that the overall stability intervals, having asymptotic points in $\tau/T_1 = 0.25$, 1.25, 2.25, etc. become the narrower the higher modes order is considered. In theory, considering an infinite bandwidth, any value of τ would cause certain modes to fall in the instability field. In practice high frequencies are dampened by wave scattering and friction, so that instabilities associated to the relevant high order modes would not occur in practice. To take this into account the ***ACUSYS*** model shown in Fig. 5 includes a second order low pass filter.

For the same reasons, in order to limit the high frequency content of the signal, an identical low pass filter is put just downstream the input ports. Otherwise simulations would show apparent, slowly rising, instabilities at high frequency. These are not numerical instabilities as could be seen by zooming in the signals, but rather intrinsic instabilities of high order modes, caused by the eigenfrequencies shift of the discrete model, as discussed later on.

3.3. Numerical results with *ACUSYS*

This preliminary verification of the model was limited to check the above stability and instability domains, by exciting the above monotube system with a step flow rate transition suitable to induce multimodal response. The system input and derived characteristic data used for the simulations are the following:

• Length of the two tubes	L_1=L_2=X_{1_2}	5	m	
• Total tube length	L	10	m	
• Tubes diameter	D	0.6	m	
• Tube area	A	0.28	m^2	
• Friction factor	λ	0.02		
• Sound speed	c	740	m/s	([1])
• Reference max. frequency considered	f_{max}	256	Hz	
• Length of the discrete tube elements	Δx	92	mm	([2])
• Total number of discrete tube elements	N	108		
• First natural frequency (quarter wave mode)	$f_1=c/(4*L)$	18.5	Hz	
• Period of the first natural frequency	$T_1=1/f_1$	0.054	s	
• Filters cut-off frequency	f_c	256	Hz	
• Input signals (see Fig. 4):				
Pressure in section 'b'	p_b	0	kg/s $\forall$ t	([3])
Flow rate entering section 'a'	$\dot{m}_a$	0	kg/s t = 0^-	
		2.82	kg/s t > 0^+	
corresponding to				
gas input velocity	v_a	10	m/s	
gas density	ρ	1	kg/m^3	
• Initial system state	p, $\dot{m}_a$	0	at any section	

The basic linear system response, namely with both κ and τ null, is shown in Fig. 7, representing the transient response typical of the "water hammer on valve opening" case in hydraulics. This case is referred to as Case 1 in Fig. 6.

The plots show the pressure response signal at sections 'a' (pipe closed end) and '1_2' (combustion surface) of the system of Fig. 4. The slight decrease of the amplitudes is due to the yet small damping caused by wall friction ($\lambda \neq 0$). The response signal phase distortion appearing as long as the simulation proceeds is instead due to the intrinsic dispersive nature of a discrete medium, like that made up by ***ACUSYS***, which makes the higher harmonics to propagate with slower speed (Ref. 5). The frequency analysis of the signals, not reported here, however proves the presence of the sole odd harmonic components with amplitudes corresponding to square wave signals'.

The second case discussed here is referred to as Case 2 and refers to τ/T_1 = 0.75 (see Fig. 6), which corresponds to the instability of all modes. Since κ influences only the rate of change of the signal amplitude (Ref. 7) it is input identical for all simulations (κ = 0.2). The result provided by ***ACUSYS*** confirms the onset of this instability (Fig. 8), with a dominant role of the fundamental harmonic.

Another unstable case is nr. 3 (τ/T_1 = 1.1833), relative to the 3rd, 4th, 5th and 6th modes (Fig. 9). The shape of the time response makes evident the disappearance of the two first modes from the signal after the first two reflections of the pressure step wave at the two pipe ends. The frequency analysis of the signal (Fig. 10), proves that the response sampled after 0.85 s is dominated by the 92.5, 129.5, 166.5, 203.5 Hz harmonics, that are indeed the above

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[1] A constant sound speed has been assumed to allow a comparison with theoretical results, but ***ACUSYS*** allows different values for each tube to be considered.

[2] This length is automatically set by the program in order to obtain a max 5% error in the calculation of the eigenmode frequencies below the cut-off one (other error tolerances are allowed to user).

[3] The absolute value of the pressure does not affect the dynamic response in linear medium acoustics.

mentioned odd multiples of the fundamental, falling in the instability range given by Eqs. 4. It could be shown that, according to the same criteria, the three next modal components are stable for this case and that stable and unstable modes alternate in groups of three to four.

The last case shown in this brief review refers to Case 6 in Fig. 6 (τ/T_1 = 1.25), for which the theory, dealing with a continuum, foresees the stability of all modal components and the same rate of damping because the imaginary part of the wave number variation is the same for all modes[4] (Ref. 7). In the numerical, discrete medium basing ***ACUSYS***, this is not exactly true, due to the gradual downwards shift of the highest eigenmode frequencies, with respect to the corresponding continuum theory values.

This causes two effects:

1. the imaginary part of the wave number variation changes among the various modes; particularly it has a periodic pattern, initially decreasing in absolute value from the fundamental mode value.
2. some modes beyond a certain order shift to instability domain, again by alternate groups

Fig. 11 shows these two effects in the comparison between the continuum case (theory) and the discrete one, for the first 65 eigenmode frequency and for the stability condition based on a value of τ/T_1 = 1.25 in both cases. The stability criterion for the discrete system is the generalisation of the criterion demonstrated by McManus & others (Ref. 7), which brings to Eqs. 4, or to the following more general form of the instability condition:

$$\mathrm{Im}(k_z^{'}) = (-1)^z \frac{\kappa}{4c} \sin(2\pi f_z \tau) < 0 \qquad \text{Eq. 5)}$$

where $k_z^{'}$ is the oscillating part of the wave number as per the definitions (Ref. 7):

$$k_z = \frac{\omega_z}{c} = k_z^0 + k_z^{'} \quad ; \quad k_z^0 = \frac{\omega_z^0}{c} \quad ; \quad k_z^{'} = \frac{\omega_z^{'}}{c} \qquad \text{Eqs. 6)}$$

Eq. 5 and Fig. 11, provide not only the conditions for instability, but also the rate of change of each component z to the overall response. Stable components with low absolute values of $\mathrm{Im}(k_z^{'})$ are slowly damped. On the opposite unstable ones with high absolute values of $\mathrm{Im}(k_z^{'})$ rise fast.

The first of the two effects above mentioned is visible in Figs. 12 and 13 where the response signal features a stable trend, as predicted by theory, but with high frequency residuals being dampened more slowly[5].

The second of the two effects is visible in Figs. 14 and 15, obtained with a simulation prolonged four times the others, where modes having frequencies beyond the cut-off defined show instability, though very slowly rising. Indeed, despite the low pass filtering, the eigenmode having frequencies around 700 Hz, which are shown in Fig. 11 to be unstable, are those which prevail in the long term response of the discrete system. These frequency components, beyond the cut-off, however were said above not to be of interest as they are actually dampened by other effects in the real system. It is worth moreover highlighting that these instabilities can be identified in advance and duly considered to evaluate the results. In particular, when instabilities in the significant bandwidth occur, these higher order ones have much smaller amplitudes than the preceding ones, thus do not affect comprehension of the results. This discussion allows, for future works, to use a sharper, discrete filtering cut-off of frequencies beyond the bandwidth of physical significance for instability analysis in more general and complex geometry cases.

[4]The modal frequencies, in the continuum, are all odd multiples of the same quarter wave fundamental f_1

[5]It is remembered that square wave harmonics have amplitudes rapidly decreasing with frequency.

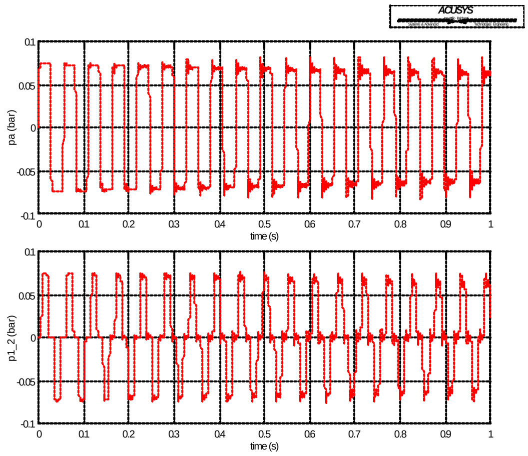

Fig. 7 - Case 1 - Response of the basic linear system (κ = τ = 0) to step flow rate variation.

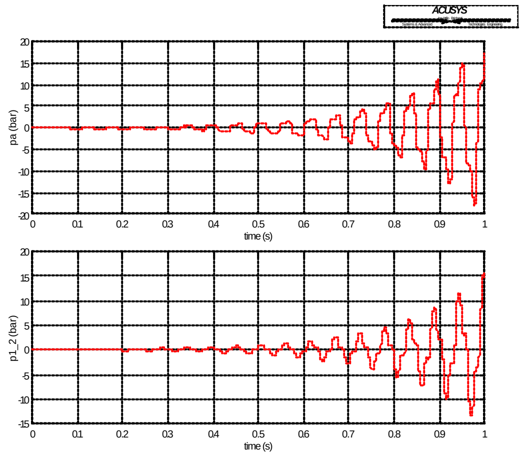

Fig. 8 - Case 2 - Instability of all modes, where 1st mode prevails (τ/T_1 = 0.75, κ = 0.2)

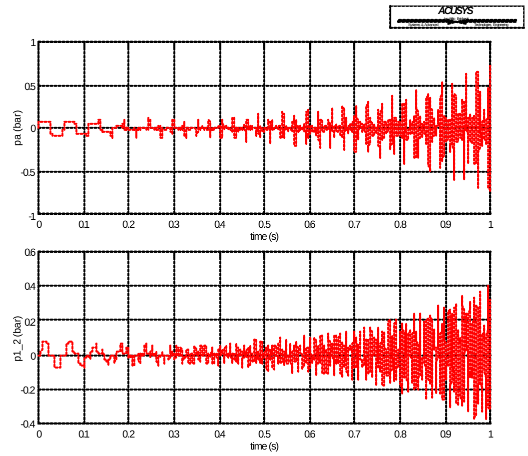

Fig. 9 - Case 3 - Instability of the 3rd, 4th, 5th and 6th modes (τ/T_1 = 0.75 , κ = 0.2).

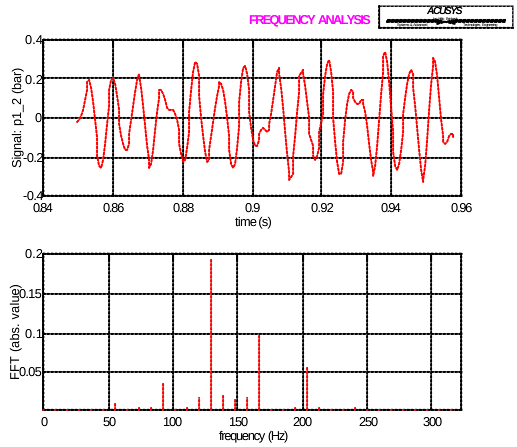

Fig. 10 - Case 3 - Same as in Fig. 9. Sampled signal and its FFT

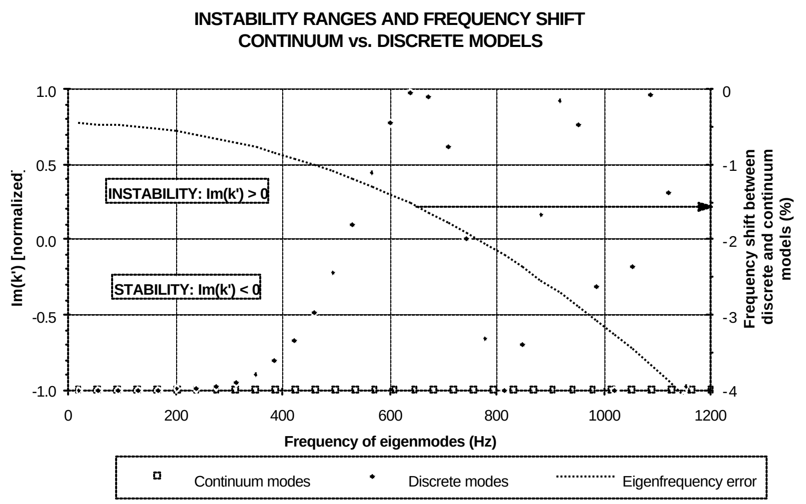

Fig. 11 - Plot of the instability parameter for τ/T_1 = 1.25 (Eq. 5, left axis) and of the eigenmodes frequencies shift (right axis), between continuum and discrete models.

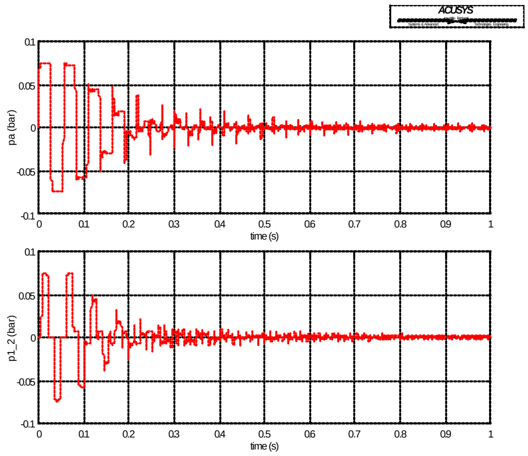

Fig. 12 - Case 6 - Theoretical asymptotic stability of all modes (τ/T_1 = 1.25 , κ = 0.2).

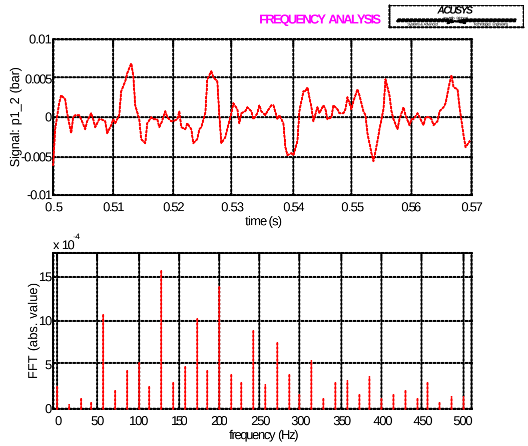

Fig. 13 - Case 6 - Same as in Fig. 12. Sampled signal and its FFT.

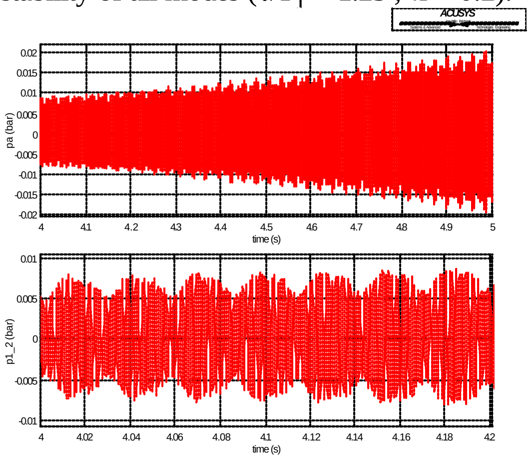

Fig. 14 - Case 6 - Same as in Fig. 12. Long simulation: upper signal from 4 to 5 s; lower signal zoomed from 4 to 4.2 s.

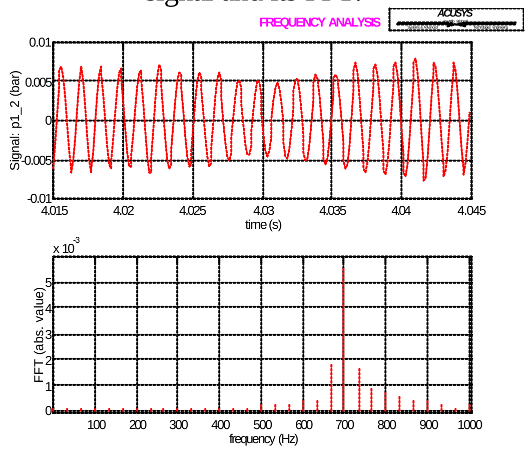

Fig. 15 - Case 6 - Same as in Fig. 12. Long simulation. Sampled signal and its FFT

4. CONCLUSIONS

The above discussion allows drawing positive conclusions on the applicability of ***ACUSYS*** to analyse combustion acoustic instabilities.

Indeed the modelling technique, with due care to the definition of the frequency range of physical interest, obtains the same result of the theory for the simple geometry case where analytical approach is possible. The behaviour of discrete models, somewhat differing to continuum models, can be handled by providing a sufficient degree of discretisation such to limit the shift of the eigenvalues falling in the bandwidth of interest, on one side, and by providing a cut-off digital filter both on the input signal and inside the non linear combustion block to cancel harmonic components of no interest.

The use of ***ACUSYS*** will allow a deeper analysis of more complex cases with benefit for the comprehension and design of pulsating combustion and its control by active means. A measure of the benefits can be drawn by just considering that the mere change of the sound speed between the two tubes, which actually occurs as consequence of temperature change by combustion, would be difficult to handle by analytical approaches and numerical solutions would be mostly necessary. But a much greater benefit will come from the possibility to use this approach with complex and interacting pipe loops, where acoustic characteristics are not immediately identifiable.

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[6]ENEL SpA is the National Electric Energy Company; CRA stands for Research Centre on Automation.