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# MULTI-DIMENSIONAL CFD-TRANSMISSION MATRIX MODELLING OF IC ENGINE INTAKE AND EXHAUST SYSTEMS

O. Chiavola

*Department of Mechanical and Industrial Engineering, University 'Roma Tre', via della Vasca Navale 79, 00146 Rome, Italy. E-mail: chiavola@uniroma3.it*

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A method able to investigate the overall performances of internal combustion engine intake and exhaust systems is proposed. Such a method is based on the combination between a time domain non-linear model (used to perform fluid and thermodynamic analysis in cylinders, valves and manifolds) and a linear acoustic method (used to predict the spectral characteristics of the remainder of the system). The time domain approach is based on the simultaneous use of zero-, one- and three-dimensional fluid dynamic models, applied to different regions of the same geometry. The frequency domain approach is based on acoustic theory and uses the transfer matrix technique. Both the procedures used to couple the different calculation domains analyzed by time domain models and to interface fluid dynamic and linear acoustic models have been developed. In this paper, the comparison between the obtained results and the predictions of another simulation technique and experimental measurements, is shown, proving that the developed method is a reliable tool for the prediction of complete intake and exhaust systems.



## 1. INTRODUCTION

The investigation of fluid dynamic and thermodynamic phenomena characterizing internal combustion engine operation retains practical relevance in the activity devoted to select the configurations able to ensure the optimized matching between noise radiation, pollutant emission and engine performance.

Experimental and computational techniques have been developed aimed at providing a better understanding of the complex phenomena occurring inside intake and exhaust systems in order to analyze and compare various configurations and operating points.

A variety of modelling procedures have been proposed in the literature and are currently applied to predict the propagation of pressure waves in the gas flowing along intake and exhaust systems. These procedures may be collected in two main groups, linear acoustic and non-linear fluid dynamic models, according to the assumptions on which they are based.

One-dimensional linear acoustic models are widely adopted for the evaluation of intake and exhaust systems noise performance, due to their simplicity and low computational requirements in the description of complex geometries. The system is considered as a combination of elements, connected in series or in parallel; each one is described by means of either a matrix representing its acoustical description, in terms of pressure and velocity, or a scattering matrix where the vectors are the wave component amplitudes moving

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towards the source and away from it, or a matrix in which the forward and reverse wave reflection and transmission coefficients are included [1–3].

By characterizing the excitation source (i.e., the engine) by means of assumed or measured impedance values, the acoustic performance are evaluated, expressed in terms of attenuation, insertion and transmission loss spectra.

In order to analyze the acoustic performance of some components in which non-planar higher modes are excited, due to the transverse dimensions that are not much smaller than the wavelength of the excitation and/or to the area discontinuities, some three-dimensional linear studies have been performed by imposing as excitation source a single frequency perturbation [4]. Owing to the high requirements of development time and computational cost, such investigations focus only on particular components of intake/exhaust system, without providing a description of their interaction with the engine.

Since in engine intake and exhaust systems, the pressure fluctuations are characterized by much larger amplitude than that assumed in the linear approach, non-linear models are required to obtain realistic predictions, taking into account for the distortion of high amplitude waveform during the propagation and the interaction between the engine and the ducts system.

Non-linear fluid dynamic models are mainly used to predict the intake and exhaust system performance in terms of pressure, velocity crank angle histories, and are based on lumped parameter (0-D), one-dimensional (1-D) or three-dimensional (3-D) schemes.

0-D schemes are usually adopted coupled with 1-D models, the former one in the investigation of cylinder behaviour and of flow across intake and exhaust valves, the latter one to describe the intake/exhaust system [5–8]. Recently, some 1-D models have been proposed to analyze complex geometries, such as mufflers, in which perforations and adsorptive materials are included [9]. The computed results allow the evaluation of the noise radiation from the tailpipe, by assuming that the open end acts as a monopole source (see, for example, reference [8]).

Sometimes, aimed at evaluating the acoustical performance of some components, 1-D models are used on their own and instead of considering the real excitation source, a single frequency or a white noise perturbation of low amplitude is prescribed, to ensure the linear propagation of pressure waves and, once the stationary conditions are reached, their transfer functions are computed [10, 11].

The quality of the predictions can be improved by introducing some length corrections to 1-D modelling, in this way the complex wave reflection at discontinuities (sudden expansion and contractions in duct cross-section) is taken into account [10, 12].

The geometric characteristics of some components, in which transverse dimensions are not small compared with the perturbation wavelength and/or abrupt discontinuities are included, and the necessity of detailed flow investigation required during the engine design refinement, have led to the development of 3-D non-linear models, but, due to the large computing requirements, the complete engine system cannot be modelled, and the 3-D investigation is focussed only on the analysis of some components, such as junction, fuel injector, intake and exhaust valve regions.

More recently, in order to overcome the limitations of linear acoustic and non-linear fluid dynamic methods, some hybrid approaches, retaining the features of both models, have been developed. In such hybrid approaches, all complex components (silencers), are described in terms of their acoustic model, and the real excitation source (engine) and the interaction between the source and intake/exhaust system are taken into account.

In some hybrid methods the non-linear time-domain analysis is performed to compute the evolution of the thermodynamic condition of the cylinder and associated valve gas flow, while acoustic theory is applied to the intake/exhaust system. Due to the cyclic characteristics of engine operations, the Fourier transformations are used to pass from one domain to the other [13, 14].

Payri, Desantes and Torregrosa have presented an hybrid method in which the non-linear calculation is performed not only upstream of the part analyzed in terms of its acoustic performance, but also downstream of it [15].

In this paper a new procedure is proposed; as in reference [15], a non-linear model is applied upstream and downstream of the linear acoustic calculation domain, and the main difference is due to the coupling of different non-linear models in the fluid dynamic approach.

The method allows to account for the real excitation source that strongly affects the real intake and exhaust performance, the non-linear propagation, the interaction between different components, dealing with the acoustic description of complex elements (mufflers) and with multi-dimensional fluid dynamic modelling of all parts (junctions, abrupt cross-section change, throttle valve) in which complex flow conditions strongly influence the predictions.

Following the introduction, section 2 describes the proposed time–frequency method, sections 3 and 4 consider, respectively, the gas dynamic models and the spectral method; section 5 is concerned with the connection procedure used to couple time and frequency calculation domains and in section 6 an application of the predictive model together with some results are presented. The predictions are compared with results of other numerical technique and experimental measurements.

## 2. TIME–FREQUENCY METHOD

The proposed time–frequency method is based on coupling non-linear thermo-fluid dynamic and linear acoustic models. Since cylinder, valves, manifolds act as time-varying systems, their investigation is best performed in time domain, whereas all parts characterized by complex geometry can be adequately described in frequency through spectral models.

The time-domain method is based on the simultaneous use of 0-D, 1-D and 3-D fluid dynamic models, applied to different regions of the geometry: the first one is used to compute the cylinder behaviour during both opened and closed valve periods, and it is matched with the 1-D model, applied to analyze all parts acting as ducts, therefore which present 1-D features; the 3-D model is adopted in all regions (such as junction, abrupt cross-section change, fuel injector, throttle valve, intake and exhaust valve region), in which a detailed fluid dynamic description is required to obtain more realistic results and a better understanding of the flow pattern compared to usual 1-D prediction. The coupling between different fluid dynamic models allows to describe each engine component with the most appropriate model according to the main features of the phenomena involved, saving computational time and cost requirements. As 1-D model, a numerical scheme based on a refinement of the Riemann method of characteristics has been developed [5, 16].

As 3-D model, the CFD package FIRE [17], based on a finite volume method adopting an implicit scheme to solve the governing equations, has been used.

Of primary importance is both the interface procedure required to model the connections between 1-D and 3-D regions, and the related interpolation technique adopted. Such interface has been developed and tested and in section 3 some details are presented.

All complex geometries always included in intake and exhaust systems, such as commercial silencers, are analyzed in the frequency domain by means of the transfer matrix method, since realistic and detailed time domain calculation schemes for them do not exist or require an excessive computational burden. Such components are regarded as a singularity in the 1-D time domain and their acoustic description is used to prescribe the boundary conditions necessary to perform the 1-D fluid dynamic calculation.

## 3. GAS DYNAMIC MODELS

### 3.1. ONE-DIMENSIONAL MODEL

All quantities are supposed uniform in the cross-section and functions only of the space co-ordinate, $x$, and the time, $t$.

The conservation equations of mass, momentum and energy for one-dimensional unsteady, compressible flow, with wall friction and wall heat transfer effects, and pipe cross-sectional area variable according to a fixed law, are

$$\rho_t + \rho_x u + \rho u_x + \rho u \frac{A_x}{A} = 0, \tag{1}$$

$$\rho u_t + \rho u u_x + p_x = -2\rho u|u| \frac{f}{d}, \tag{2}$$

$$s_t + u s_x = \frac{4Q}{\rho T d} + \frac{2fu^3}{Td}. \tag{3}$$

A relation expressing the nature of the gas has to be added (i.e., ideal gas law). (A list of symbols is given in Appendix A).

The numerical technique used to solve the non-linear hyperbolic system of equations, coupled with boundary conditions, is a refinement of the Riemann characteristics method; it is organized on two-level calculation, characterized by a second order accuracy for both time and space.

The space integration is evaluated according to the sign of the three characteristic lines:

$$\frac{\mathrm{d}x}{\mathrm{d}t} = u \pm a, \qquad \frac{\mathrm{d}x}{\mathrm{d}t} = u. \tag{4, 5}$$

Each pipe is divided into equally spaced interval, $\Delta x$; the time step, $\Delta t$ is imposed by the stability criterion (Courant–Friedrichs–Lewy condition):

$$\Delta t \leqslant \frac{\Delta x}{\max(a + |u|)}. \tag{6}$$

Several boundary conditions can be simulated: junction, open end, capacity, cylinder. In order to consider the development of a hybrid time–frequency model, four types of boundary conditions are taken into account: the connection with the cylinder, therefore the flow from the cylinder to the duct and *vice versa*, through the valve, for both the case of pressure ratio larger than the critical value and of subsonic flow, is simulated (experimental discharge coefficient can be applied); the connection with a 3-D domain; the interaction with a region modelled in frequency domain and the open end termination.

The thermo-fluid dynamic behaviour of the cylinder is computed by means of a lumped parameter model, both during the closed valve and opened valve period.

In the developed and validated calculation procedure [16, 18], the contact discontinuity, caused at the exhaust valve opening by the interaction between hot gas discharged by the cylinder and cold gas inside the duct, is analyzed. It can be considered as a separation front across which some quantities (temperature, entropy, sound speed) are characterized by abrupt variation, while velocity and pressure are continuous. Its propagation through the exhaust system is provided.

### 3.2. THREE-DIMENSIONAL MODEL

The three-dimensional CFD code FIRE is used. It is based on a finite volume method and on an implicit technique to solve the governing equations. Some subroutines have been developed in order to allow the fluid dynamic coupling between 3-D and 1-D calculation domains.

### 3.3. CONNECTION BETWEEN DIFFERENT FLUID DYNAMIC MODELS

The simultaneous use of 1- and 3-D fluid dynamic models applied to different regions of the same geometry, requires the definition of both the interface procedure to be adopted in the connection region and the interpolation technique necessary to pass from a solution procedure to the other which strongly affect the quality of the results.

Since the 1-D code is unable to take into account the spatial flow distribution within the cross-section of the duct, the choice of the position where one locates the connection region 1-D/3-D has to be carefully planned: it has to be placed where no multi-dimensional flow features appear.

An interfacing procedure has been developed and tested by the author [16, 18–20]. It is based on the definition of an overlapping region at each 1-D/3-D link: in the 1-D domain an additional pipe of one cell length is considered, corresponding to the part of 3-D domain which is used for the interface between the different models (Figure 1).

The data transfer from 3-D to 1-D domain is performed at each time step by assigning the averaged FIRE quantities in the overlapping region to the additional pipe that is used to prescribe boundary conditions for 1-D model.

Mass flow and temperature over the interface with FIRE can be obtained by a 1-D calculation scheme and are assigned to the cell nodes contained in the 3-D boundary region. Instead of imposing uniform quantities at FIRE interface, the averaged quantities computed by 1-D code are modified, so that their profiles, determined in the previous 3-D time step, are preserved.

The simulation of 1-D and 3-D regions is performed simultaneously and in synchronism. As the 1-D code uses an explicit scheme to solve the governing equations, the time step depends on the stability criterion and it changes during the simulation; the synchronism between the 1-D and the 3-D code is ensured by controlling the absolute 3-D model time step value. Adopting an implicit scheme, FIRE code is able to accept time step larger than 1-D code; for this reason several iterations are proceeded with the latter code, until the

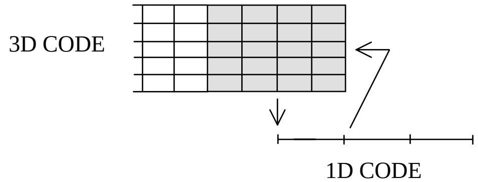


Figure 1. Coupling procedure.

synchronism is reached. In references [16, 18, 19] further details of the developed interface procedure are presented, together with the results of some tests and the comparisons between the predictions and experimental measurements. In references [18, 20] some applications of the method to complex exhaust system are shown.

## 4. SPECTRAL MODEL

### 4.1. TRANSFER MATRIX METHOD

Since usually the intake and exhaust system geometry is characterized by a leading longitudinal dimension, the flow can be analyzed by a one-dimensional approach (plane wave motion), based on the hypothesis that pressure and velocity profiles are uniform in the cross-sectional area and are functions only of the axial position, $x$, along the duct, and the time, $t$ [1, 2, 21].

Plane wave acoustic theory is appropriate below the first cut-off frequency; the hypotheses of non-viscous medium, rigid wall and linear behaviour are valid as long as the wave shape does not alter as it propagates, that means a bounded acoustical pressure value (infinitesimal amplitude of pressure with regard to the mean value). In any case the acoustical theory is appropriate as long as the wavelength exceeds the largest dimension of the cross-section.

The transfer matrix method is used; the intake and exhaust system is considered as the combination of a certain number of acoustic elements, which can be connected in series or in parallel.

Each element can be regarded as excited by the previous one and ended in the following one and can be schematized by means of a four pole matrix that describes the relation between acoustic pressure, $p$, and velocity, $u$, in forward and backward sections of the single element:

$$\begin{bmatrix} p_1 \\ u_1 \end{bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} p_2 \\ u_2 \end{bmatrix}. \tag{7}$$

As each matrix correlates the acoustic quantities at inlet and outlet of the element, it represents the propagation characteristic of a prefixed frequency perturbation in stationary condition across an element of duct of finite or infinitesimal length (abrupt cross-section variation, finite length duct, etc.). See Figure 2.

The matrices of all the elements are combined to obtain the overall intake and exhaust system transfer matrix:

$$[T] = [T]^{(1)} \cdots [T]^{(N)} \tag{8}$$

($N$ stands for the number of component elements); moreover taking into account the matrices representing the evolution of the source and the termination, the acoustical behaviour of the system is computed.

The four terms of the matrix, $T_{ij}$ ($i = 1, 2; j = 1, 2$), are functions of the element geometry, state variable of the medium, mean flow velocity and perturbation frequency; then, at each

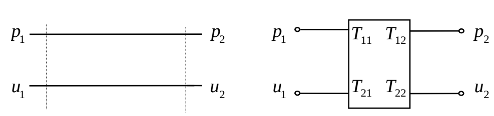


Figure 2. Scheme of the transfer matrix method.

frequency (all other geometrical and physical quantities fixed), a relation between pressure and velocity in initial and final cross-sections is obtained [16].

## 5. CONNECTION BETWEEN TIME- AND FREQUENCY-DOMAIN MODELS

The implementation of the interface procedure necessary to connect time and frequency domains is of primary importance as the effectiveness of the overall methodology deeply depends on it [13, 14, 22].

In the developed hybrid procedure, the case of a region modelled by means of its acoustical performance placed between two domains analyzed by fluid dynamic one-dimensional model formulated in time, is presented (Figure 3).

It is possible to refer this configuration to a scheme where in the region investigated in time domain, a singularity, regarded as a linear system and characterized by its acoustical parameters, is included; the calculation is performed by imposing the acoustic characteristics of the singularity as boundary conditions for the time-domain approach.

To transform time domain results into frequency domain ones and *vice versa*, it has to be considered that the interface represents a boundary where all the quantities are time variant, and, since engine operation, with the associated wave motion, is cyclic, spectral representations are readily inverse Fourier transformed to the time domain, and the converse.

The connection procedure is based on the decomposition of the pressure perturbation into its components, a backward wave, which moves towards the source, and a forward wave, which moves away from the source and on reflection and transmission coefficients used in the transfer matrix method to describe the acoustical behaviour of a system.

The starting point is to transform matrix (7), relating pressure and velocity upstream and downstream of a generic element, into a matrix containing the transmission and reflection coefficients. Pressure and velocity can be expressed in terms of the two pressure components, a forward wave and a backward wave:

$$p = p^+ + p^-, \qquad u = \frac{1}{Z}(p^+ - p^-), \tag{9, 10}$$

where $Z$ represents the impedance: $Z = \rho a$ ($\rho$ and $a$ are, respectively, the undisturbed density and sound speed in the medium).

Substitution of equations (9) and (10) in equation (7) yields the expression

$$\begin{bmatrix} p_1^+ \\ p_1^- \end{bmatrix} = \begin{bmatrix} E_{11} & E_{12} \\ E_{21} & E_{22} \end{bmatrix} \begin{bmatrix} p_2^+ \\ p_2^- \end{bmatrix}, \tag{11}$$

$$E_{11} = \frac{1}{2}\left(T_{11} + \frac{T_{12}}{Z} + ZT_{21} + T_{22}\right), \tag{12}$$

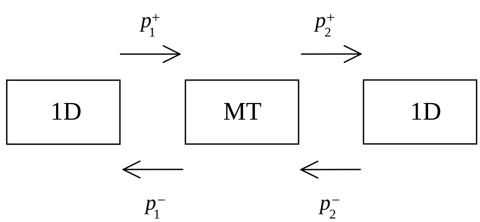


Figure 3. Time–frequency scheme.

$$E_{12} = \frac{1}{2}\left(T_{11} - \frac{T_{12}}{Z} + ZT_{21} - T_{22}\right), \tag{13}$$

$$E_{21} = \frac{1}{2}\left(T_{11} + \frac{T_{12}}{Z} - ZT_{21} - T_{22}\right), \tag{14}$$

$$E_{22} = \frac{1}{2}\left(T_{11} - \frac{T_{12}}{Z} - ZT_{21} + T_{22}\right). \tag{15}$$

$p_1^+$ and $p_2^-$ can be regarded as components carrying information related to the flow upstream and downstream of the discontinuity; $p_2^+$ and $p_1^-$, on the other hand, represent the influence of the singularity on the surrounding domains.

The transmission and reflection coefficients expressing the four parameters that characterize the matrix, can now be obtained ($D$ stands for direct and $I$ for inverse):

$$\begin{bmatrix} p_2^+ \\ p_1^- \end{bmatrix} = \begin{bmatrix} T_D & R_I \\ R_D & T_I \end{bmatrix} \begin{bmatrix} p_1^+ \\ p_2^- \end{bmatrix}, \tag{16}$$

$$T_D = \frac{1}{E_{11}}, \qquad R_I = -\frac{E_{12}}{E_{11}}, \tag{17, 18}$$

$$R_D = \frac{E_{21}}{E_{11}}, \qquad T_I = E_{22} - \frac{E_{12}E_{21}}{E_{11}}. \tag{19, 20}$$

In the left-hand side of equation (16), $p_2^+$ and $p_1^-$ stand for the two wave components that travel from the singularity toward the region analyzed by the time domain method. The inverse Fourier transformation of such spectra provides the description of the corresponding wave in the time domain, allowing one to determine the Riemann variables, $R_1 = a/\delta - u$ and $R_2 = a/\delta + u$, necessary to compute the boundary conditions for 1-D fluid dynamic calculation:

$$R_1 = \frac{2a_0}{\delta u_0}(p_1^-)^{(\gamma-1)/2\gamma} - \frac{a_0}{\delta u_0}, \tag{21}$$

$$R_2 = \frac{2a_0}{\delta u_0}(p_2^+)^{(\gamma-1)/2\gamma} - \frac{a_0}{u_0} \tag{22}$$

($R_1$ and $R_2$ lead to boundary conditions, respectively, for the domain upstream and downstream of the singularity).

In order to establish the data transfer from time to frequency domain, the decomposition of pressure perturbation into forward and backward components has to be performed.

To this end, the scheme depicted in Figure 4 is analyzed.

All quantities in node i at time step $t + \Delta t$ can be computed by making use of the known flow conditions at points $p$ and $q$, at time step $t$:

$$u_i = \frac{a_p - a_q}{2\delta} + \frac{u_p + u_q}{2}, \tag{23}$$

$$a_i = \frac{a_p + a_q}{2} + \frac{\delta}{2}(u_p - u_q), \tag{24}$$

where

$$u_p = \frac{a_p - a_0}{\delta}, \qquad u_q = \frac{a_0 - a_q}{\delta}, \tag{25, 26}$$

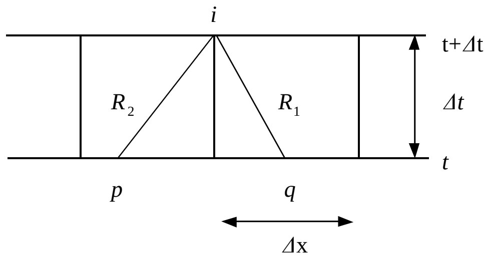


Figure 4. Grid of 1-D non-linear method.

$$\frac{p_p}{p_0} = \left(\frac{a_p}{a_0}\right)^{2\gamma/(\gamma-1)}, \qquad \frac{p_q}{p_0} = \left(\frac{a_q}{a_0}\right)^{2\gamma/(\gamma-1)} \tag{27, 28}$$

(subscript 0 stands for reference conditions).

Sound speed and velocity can be expressed in terms of Riemann variables:

$$a = \frac{\delta}{2}(R_1 + R_2), \qquad u = \frac{R_2 + R_1}{2}. \tag{29, 30}$$

By making use of dimensionless quantities, the expressions of $R_1$ and $R_2$ at points $p$ and $q$ can be obtained:

$$R_{1_p} = \frac{a_p}{\delta u_0} - \frac{u_p}{u_0} = \frac{a_0}{\delta u_0}, \tag{31}$$

$$R_{2_q} = \frac{a_q}{\delta u_0} + \frac{u_q}{u_0} = \frac{a_0}{\delta u_0}. \tag{32}$$

Substitution of Riemann variables into the dimensionless pressure expression yields

$$p = \left(\frac{\delta(R_1 + R_2)a}{2a_0/u_0}\right)^{2\gamma/(\gamma-1)}. \tag{33}$$

The wave components, respectively, at points $p$ and $q$ are defined by

$$p_1^+ = \left(\frac{1}{2} + \frac{\delta R_{2_q}}{2a_0/u_0}\right)^{2\gamma/(\gamma-1)}, \tag{34}$$

$$p_2^- = \left(\frac{1}{2} + \frac{\delta R_{1_p}}{2a_0/u_0}\right)^{2\gamma/(\gamma-1)}. \tag{35}$$

These equations represent the boundary conditions at the interface between the time and frequency domains. In order to obtain the complex spectral amplitudes, the Fourier transformation of these values is performed.

The overall hybrid time–frequency methodology is an iterative procedure, as the response of the singularity in the frequency domain is imposed at the boundary analyzed in the time domain during the period following that one during which it has been computed. The calculation is performed for successive periods, until a suitable convergence criterion is achieved.

The value of the time step used in the 1-D fluid dynamic model calculation has to be such as to satisfy the Courant–Friedrichs–Lewy stability criterion (6); $\Delta t$ cannot be variable (the grid spacing $\Delta x$ is fixed at the beginning of the calculation and velocity and temperature

TABLE 1

*Ruggerini RDM 901 Diesel engine*

| | |
|---|---|
| Bore | 90 mm |
| Stroke | 85 mm |
| Connecting rod length | 142 mm |
| Compression ratio | 18:1 |
| Inlet valve diameter | 37·6 mm |
| Exhaust valve diameter | 36·9 mm |
| EVO | 65° before BDC |
| EVC | 39° after TDC |
| IVO | 42° before TDC |
| IVC | 91° after BDC |

vary from one step to the other); it has to be constant and defined by the number of values in each period, to ensure that time resolution in the time domain is equal to frequency resolution in the frequency domain.

In the linear acoustic description of the singularity, the coupling between different harmonics, which arises during the wave propagation throughout the ducts system, is neglected; then, due to the non-linearity of the propagation process, the different harmonics are not, in general, solutions of the fluid dynamic equations. In the proposed hybrid model, the transfer matrix is representative only of the singularity behaviour, since the remainder of the geometry is analyzed by the 0-D/1-D/3-D non-linear models; by this all regions upstream and downstream the singularity are investigated in time domain, allowing to account for the non-linear effects.

In summary, the hybrid method calculation steps are as follows: choice of an initial solution $p_1^+(t)$ and $p_2^-(t)$ for a whole period; determination of $p_1^+(\omega)$ and $p_2^-(\omega)$ by means of Fourier transformation; calculation of $p_1^-(\omega)$ and $p_2^+(\omega)$ (response of the singularity) by means of the transfer matrix method; inverse Fourier transformation in order to obtain time domain values $p_1^-(t)$ and $p_2^+(t)$; application of the 1-D fluid dynamic non-linear method to compute the new $p_1^+(t)$ and $p_2^-(t)$ to apply during the following calculation period.

## 6. APPLICATION OF THE TIME–FREQUENCY APPROACH

The overall hybrid procedure has been applied to predict the unsteady flow in a one cylinder 4-stroke engine, coupled with a simple exhaust system, and the results have been compared with the experimental measurements and the computed pressure histories presented by Cordiner *et al.* [23, 24].

The investigated system is composed of one cylinder (the main parameters of the engine are reported in Table 1) connected to a straight pipe of constant cross-section. Such a geometry has been chosen as test case since for it numerical data (obtained by using a 1-D scheme coupled with a lumped parameter model for cylinder and engine volumes) and experimental measurements are available, and non-linear fluid-dynamic and linear acoustic models can be easily applied. The schematization has a general validity, since it can be adopted to analyze the most general intake/exhaust system, organized in any complex architecture (in reference [25] the results of an application to a typical multi-dimensional geometry are presented).

The thermodynamic behaviour of the cylinder has been analyzed by means of the developed 0-D time-domain model. The exhaust pipe has been divided in different regions

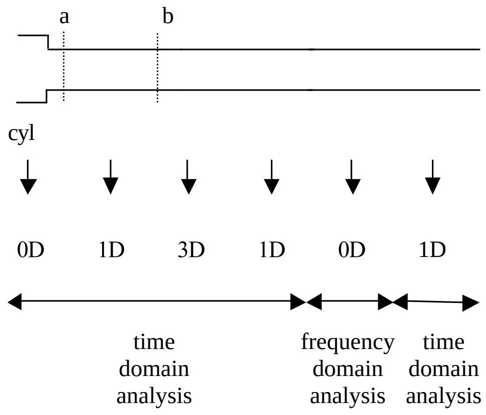


Figure 5. Scheme of the simulated exhaust system.

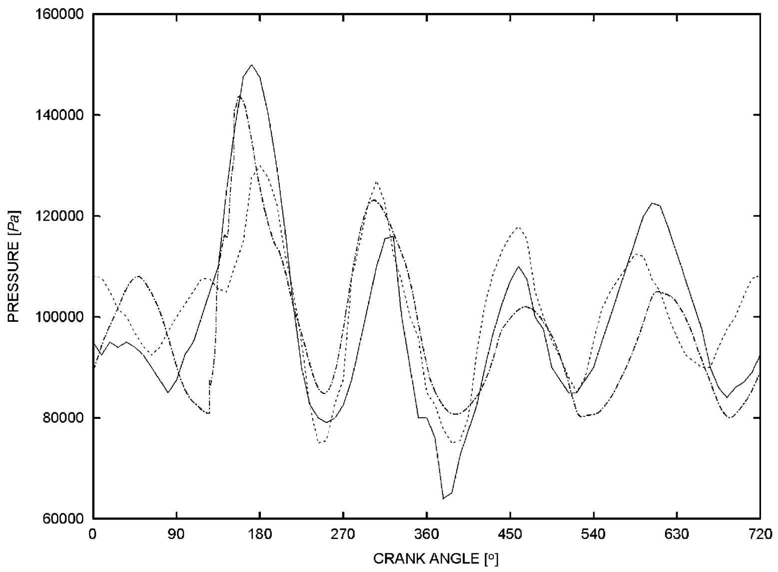


Figure 6. Comparison between predicted and measured pressure histories at location a of Figure 5: computed, —·—·, measured [15] ——, computed [15] ……. .

where the fluid dynamic time domain models and the acoustic frequency domain approach have been applied, following one another, as shown in Figure 5. Close to the exhaust valve, a time domain region is considered, in which 1-D, 3-D and 1-D domains are included; the 3-D domain consists in a essentially 1-D geometry. A frequency-domain region follows. The residual part of the pipe, close to the open end, is investigated by means of the 1-D fluid dynamic model.

The time–frequency approach is an iterative predictive tool, since the time-domain calculation needs time-varying boundary conditions at the interfaces with frequency domain regions for a whole period. The rest condition in all regions has been imposed at the beginning of the overall simulation; then some calculation cycles have been performed until the convergence of the fluid dynamic quantities has been reached. If a first estimate of

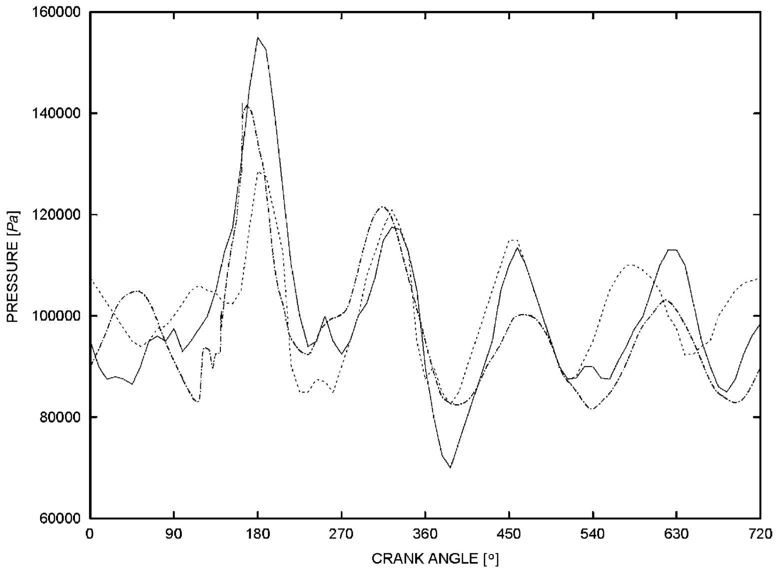


Figure 7. Comparison between predicted and measured pressure histories at location b of Figure 5: computed, —·—· measured [15] ——, computed [15] ....... .

pressure and velocity fields in the geometry is assumed as initial conditions, a faster convergence may be obtained.

The choice of the time step to be used in the calculation has been carried out by considering that the 1-D fluid dynamic code adopts an explicit scheme to solve the governing equations, and for this reason the time step depends on the stability criterion; moreover the time resolution in time domain has to correspond to the frequency domain resolution in the frequency domain, therefore the calculation has been performed with constant time interval, defined by frequency resolution and compatible with stability criterion.

Two different monitoring locations placed along the exhaust pipe have been considered, a, near the cylinder, and b, downstream of it, at 1/3 of the overall length of the duct, as shown in Figure 5. The pressure traces obtained for the last cycle of simulation have been analyzed and compared with those computed by means of another numerical technique (0-D scheme coupled with 1-D fluid dynamic model), and with experimental measurements (Figures 6 and 7). From both figures it can be pointed out that the accuracy of the predictions is quite good, both during the opened and the closed valve periods. The main differences in the pressure histories are located during the combustion process, and affect both numerical approaches; these discrepancies are probably due to a disturbed signal in the experimental rig. The graphs highlight that the time–frequency approach allows one to provide a correct evaluation of the propagation time of the pressure disturbance. From the comparison between predicted results and experimental traces some differences regarding the amplitude value appear; they are due to the influence of different loss coefficients adopted in both 1-D numerical approaches; these coefficients have not been modified in order to ensure a better agreement with measured data, since the primary aim of the present paper was not to reproduce a particular result but to evaluate the capability of an integrated time/frequency approach to catch the main features characterizing the propagation of a pressure perturbation along intake and exhaust systems which affect the engine performance.

## 7. CONCLUSIONS

The proposed hybrid methodology holds practical effectiveness in modelling complete intake and exhaust system of multi-cylinder engine.

In such systems different kinds of elements are always included, some characterized by 1-D behaviour, others typically 3-D, others having a complex shape, such as mufflers.

The time–frequency approach, based on coupling linear and non-linear models, allows one to investigate each component by means of the most appropriate model, according to the main features of the phenomena involved, accounting for the real excitation source and for the interaction between different engine components.

The non-linear approach is based on the simultaneous use of 0-D (to analyze cylinder behaviour), 1-D (to investigate all parts acting as ducts) and 3-D models (to simulate all elements requiring a multi-dimensional analysis to obtain realistic predictions and allow further optimization, such as junctions, abrupt cross-section change, fuel injector, throttle valve, intake and exhaust valve regions).

For all complex components (mufflers), a multi-dimensional analysis does not seem profitable because of its long time and cost requirements for both mesh generation and computation; 1-D models which have been proposed in the literature, are very useful for a detailed comprehension of the phenomena, but do not seem appropriate for a global investigation; conversely, the large availability of experimental data allows an easy schematization of such elements by means of the transfer matrix method.

The method here outlined may be adopted for both fluid dynamic and acoustic predictions, and then to evaluate the engine performance and the noise radiation. Moreover, the hybrid time–frequency approach may be applied to characterize a complex geometry by means of a transfer matrix, thus being very useful in the activity of setting up and tuning the overall intake and exhaust geometry of multi-cylinder engine organized in any complex architectures.

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## APPENDIX A: NOMENCLATURE

| | |
|---|---|
| $A$ | cross-section area |
| $a$ | speed of sound |
| $d$ | duct diameter |
| $f$ | friction factor |
| $p$ | pressure |
| $Q$ | heat flow |
| $R_1$, $R_2$, $R_3$ | Riemann variables |
| $R_D$, $R_I$ | direct and inverse reflection coefficients |
| $s$ | entropy |
| $t$ | time |
| $T$ | temperature |
| $T_{ij}$ | elements of the transfer matrix $[T]$ |
| $T_D$, $T_I$ | direct and inverse transmission coefficients |
| $u$ | fluid velocity |
| $Z$ | impedance $\rho a$ |
| $\delta$ | $\gamma - 1$ |
| $\gamma$ | ratio of the specific heats |
| $\rho$ | density |
| $\Delta t$ | time step |
| $\Delta x$ | mesh length |
| $\omega$ | radian frequency |

*Subscripts*

| | |
|---|---|
| 0 | reference condition |
| $t$ | differentiate with respect to time |
| $x$ | differentiate with respect to space |