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A New Modeling of Air Dynamics Inside ICE Intake Manifolds

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Abstract

Higher performances, lower fuel consumption, emission reduction and better derivability are strictly related to the Air to Fuel ratio control. So, the new technology challenges in this area need new control strategies able to satisfy near and far future demands: an increasing role of the mathematical modeling is expected to occur, mainly for on board applications (A/F model based control).

This paper presents a new modeling for the air dynamics able to describe the transient effects inside engine intake manifolds. The model can be considered as a further evolution of the Mean Value Engine Models, widely used in literature, that compensates for transient effects. Through a suitable formulation of the mass, momentum and energy equations the model predicts mass flow rates during engine transients without renouncing to simplicity and possibility to be used in the on board applications (compatible with the near future electronic capabilities). In this sense, the model appears to be an intermediate stage between lumped and distributed parameter models.

The model considers the engine intake manifold ducts as a collection of small interconnected capacities, and each capacity interacts with the surrounding capacities as boundary condition. In this way, it becomes more simple to model the complicated duct geometry, which characterize the practical designs of the internal combustion engine's intake manifolds. The application of the conservation equations at the capacity under consideration gives a set of three first-order differential equations, whose solution describe the pressure and the temperature in the capacity, in addition to the air velocity between adjacent capacities. Transient processes can be therefore described and their effects on the air mass trapped into the cylinder.

1. Introduction

As emission regulations become more stringent throughout the last two decades, model-based SI engines fueling control becomes one of the most favorable alternatives. Over the last few years, many researchers [1,2,3,4,5,6] have discerned this possibility. The interest increases as much as the microprocessors become more powerful and less expensive to let foresee a massive on board application. Model based fuel injection control has the feature to calculate, during engine running, variables that are difficult to be measured with the required accuracy. Among these variables, the intake manifold pressure which is the most important and suitable property to predict the instantaneous air mass trapped into the engine cylinders. This, of course, has a crucial importance because emissions are sensitive to the instantaneous air/fuel ratio as well as the effectiveness of the three way catalytic converter.

Both for direct and indirect fuel injection, the air estimation is the key point and the most important step toward a model based control. The computer resources available on the electronic boards limit the precision of the prediction in the engine transients and produce a poor A/F control.

Different models of increasing complexity to estimate the entering mass to engine cylinder are available in the literature. Among them, the Mean Value Engine Models [1,7,8,9,10], that give acceptable results in the steady state and small transient engine operations and poor results in more severe transient operation as well as in highly tuned manifolds. Model based nonlinear observers [2,3,4,5,9] are characterized by higher accuracy but still below the sought level of accuracy in all engine operation phases and intake manifold designs. A recent model has been proposed to satisfy the model-based control demands [11,12]: the authors called it Quasi-Propagatory to enlighten its capability to match a lumped parameter approach and a description of a wave effects. This model applies to more critical situations than those in the intake engine manifolds; due to this, the computing resources it needs hardly satisfies the near future on board capabilities.

It is well known in fact that a model based control requires precision in predicting the air trapped into the cylinder with a computing resources that lie within the on board electronic limits.

This paper proposes a new modeling that can describe air dynamics inside intake manifolds with an effective compromise between precision and computing requirements. In this sense, the model rises its potential possibility to be a favorable alternative in the model-based A/F control.

This new modeling can be considered as a further evolution of the Mean Value Engine Models that compensates for transient effects which are not considered in the literature. Through a suitable formulation of the continuity, momentum and energy equations, the model gives predictions for the mass flow rates during engine transients maintaining the possibility to be applied on board.

The model allows a representation of the real intake duct geometry, often complex in practical situations. This is done by a collection of a small interconnected capacities to which the conservation equations are applied. This gives a physical consistency to the description in terms of temperature and pressure in each capacity and air velocity between adjacent capacities.

This new modeling has been labeled as a Method Of Interconnected Capacities (MOIC) and appears to be an alternative formulation to the full one dimensional transient approach.

2. MOIC Derivation

The model considers the engine intake manifold ducts as a collection of small interconnected capacities. Each capacity interacts with those adjacent exchanging mass, momentum and energy. In this way, complex geometries which characterize real intake systems can be described without the need of simplification, particularly critical in multi-cylinder engines.

An essential part of an intake system is represented in figure 1, where a generic duct has at its ends an upstream and downstream flow dynamic boundary conditions. To the j capacity the conservation equations are written as follows:

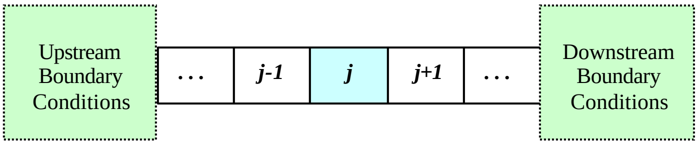

Figure 1: A generic intake manifold duct as it is divided into *n* capacities.

Continuity Equation

The rate of change of the mass inside the capacity m_j is given by

$$\frac{dm_j}{dt} = \dot{m}_i - \dot{m}_e \tag{1}$$

being $\dot{m}_i$ and $\dot{m}_e$ are the inflow and the outflow mass rates evaluated as

$$\dot{m} = rAu \tag{2}$$

where r is the density, A is the cross-section of the flow area, and u is the flow velocity.
The left side of Eq. 1 can be expressed by the derivative of state equation as

$$\frac{dm_j}{dt} = \left(\frac{V_j}{RT_j}\right)\left[\frac{dP_j}{dt} - \left(\frac{P_j}{T_j}\right)\frac{dT_j}{dt}\right] \tag{3}$$

where P_j is the pressure, V_j is the capacity volume, R is the gas constant of air, and T_j is the temperature.

Energy Conservation Equation

The application of the first law of thermodynamics to the j capacity gives the following

$$\frac{dE_j}{dt} = \dot{m}_i\, h_i - \dot{m}_e\, h_e + \dot{Q} \tag{4}$$

where dE_j/dt is the time rate of change of internal energy in the capacity, h_i and h_e are the specific stagnation enthalpy of the entering and exiting masses, and $\dot{Q}$ is the net heat transfer rate into the capacity.

The right side of Eq. 4 can be derived as

$$\frac{dE_j}{dt} = \left(\frac{g}{g-1}RT_i + \frac{u_i^2}{2}\right)\dot{m}_i - \left(\frac{g}{g-1}RT_e + \frac{u_e^2}{2}\right)\dot{m}_e + \left(\frac{P_jV_j}{RT_j}\right)\left(\frac{2\,fC_p|\bar{u}_j|}{D}\right)\left(T_w - T_j\right) \tag{5}$$

in terms of g ratio of the specific heats, f friction coefficient, C_p constant pressure specific heat, D mean duct diameter, and T_w mean wall temperature.

The rate of change of internal energy in the capacity is given by

$$\frac{dE_j}{dt} = \frac{d}{dt}\left(m_j e_j\right) \tag{6}$$

being

$$e_j = \left(\frac{1}{g-1}\right)\frac{P_j}{r_j} + \frac{\bar{u}_j^2}{2} \tag{7}$$

where $\bar{u}$ is the mean velocity inside the capacity. Eq. 6 is therefore

$$\frac{dE_j}{dt} = \left(\frac{V_j}{R}\right)\frac{d}{dt}\left(\frac{P_j}{T_j}\left\{\frac{1}{g-1}RT_j + \frac{\bar{u}_j^2}{2}\right\}\right) \tag{8}$$

A first-order differential equation in terms of pressure, temperature and velocity $\bar{u}$ inside the capacity under consideration can be obtained as shown in Eq.9.

$$\left(\frac{R}{V_j}\right)\frac{dE_j}{dt} = \left\{\left(\frac{R}{g-1}\right)+\left(\frac{\bar{u}_j^2}{2T_j}\right)\right\}\frac{dP_j}{dt} - \left(\frac{\bar{u}_j^2 P_j}{2T_j^2}\right)\frac{dT_j}{dt} + \left(\frac{\bar{u}_j P_j}{T_j}\right)\frac{d\bar{u}_j}{dt} \tag{9}$$

Momentum Conservation Equation

Applying the conservation of momentum to the capacity *j* gives a first-order differential equation for the mean velocity inside the capacity $\bar{u}$. Considering only the pressure and friction forces, the equation is

$$\frac{d\bar{u}_j}{dt} = \left(\frac{RT_j}{P_j V_j}\right)\left\{P_i A_i - P_e A_e - \bar{u}_j \frac{dm_j}{dt}\right\} - sign(\bar{u}_j)\, f \frac{\bar{u}_j^2}{2}\frac{4}{D} \tag{10}$$

Equations 3, 9 & 10 can be rewritten in the form of matrix equation ***AB*** = ***C*** as

$$\begin{bmatrix} 1 & -\left(\frac{P_j}{T_j}\right) & 0 \\ \left\{\left(\frac{R}{g-1}\right)+\left(\frac{\bar{u}_j^2}{2T_j}\right)\right\} & -\left(\frac{\bar{u}_j^2 P_j}{2T_j^2}\right) & \left(\frac{\bar{u}_j P_j}{T_j}\right) \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \frac{dP_j}{dt} \\ \frac{dT_j}{dt} \\ \frac{d\bar{u}_j}{dt} \end{bmatrix} = \begin{bmatrix} \left(\frac{RT_j}{V_j}\right)\frac{dm_j}{dt} \\ \left(\frac{R}{V_j}\right)\frac{dE_j}{dt} \\ \left(\frac{RT_j}{P_j V_j}\right)\Phi \end{bmatrix} \tag{11}$$

whose solution is

$$\begin{bmatrix} \frac{dP_j}{dt} \\ \frac{dT_j}{dt} \\ \frac{d\bar{u}_j}{dt} \end{bmatrix} = (g-1)\left(\frac{T_j}{P_j V_j}\right) \begin{bmatrix} -\left(\frac{\bar{u}_j^2 P_j}{2T_j}\right)\frac{dm_j}{dt} + \left(\frac{P_j}{T_j}\right)\frac{dE_j}{dt} - \left(\frac{\bar{u}_j P_j}{T_j}\right)\Phi \\ -\left\{\left(\frac{R}{g-1}\right)+\left(\frac{\bar{u}_j^2}{2T_j}\right)\right\}T_j\frac{dm_j}{dt} + \frac{dE_j}{dt} - \bar{u}_j\Phi \\ \left(\frac{R}{g-1}\right)\Phi \end{bmatrix} \tag{12}$$

being the terms dm_j/dt, dE_j/dt given by Eq. 1 and Eq. 5, respectively, while Φ results as

$$\Phi = P_i A_i - P_e A_e - \bar{u}_j \frac{dm_j}{dt} - sign(\bar{u}_j)\, f \frac{\bar{u}_j^2}{2}\frac{4}{D}\left(\frac{P_j V_j}{RT_j}\right) \tag{13}$$

In the calculation of Φ a further simplification can be done, by which a sufficiently accurate solution can be obtained. In that, P_i has been replaced by P_j and $\bar{u}_j$ can be considered equal to u_e. By this way, the modeling of the capacities at the extreme boundaries becomes very simple.

3. MOIC Boundary

Defining boundary capacities conditions needs high attention. Several different situations can happen (subsonic and sonic, direct and reverse flow). All these situations must represent the behavior of flow through sudden retractions or sudden openings according to the direction of the flow. The modeling of the boundary condition has been specified according to the following situations:

Intake Pipe Section: The modeling is different according to the flow direction. In case of direct flow, $sign(P_o - P) > 0$, the application of the energy conservation for isentropic flow gives for subsonic flow

$$\dot{m}_t = C_{d_t} A_t P_o b \Big/ \sqrt{RT_o} \tag{14}$$

and for sonic flow

$$\dot{m}_t = C_{d_t} A_t P_o a \Big/ \sqrt{RT_o} \tag{15}$$

being

$$b = \left\{ \frac{2g}{g-1} \right\}^{1/2} P_r^{1/g} \sqrt{1 - P_r^{(g-1)/g}} \tag{16}$$

$$a = g^{1/2} \left\{ \frac{2g}{g+1} \right\}^{(g+1)/(2(g-1))} \tag{17}$$

$$P_r = P/P_o \tag{18}$$

In case of reverse flow the pressure can not be fully recovered through a sudden enlargement; considering all the kinetic energy lost, the following conditions applies

$$P = P_o \tag{19}$$

Intake Valve Section: The modeling of this section is deeply discussed in many references like [11,12,14]. And in the same way as it was done in the previous section, according to the flow direction. In case of direct flow, from pipe to cylinder through the valve, $sign(P - P_c) > 0$. For subsonic flow

$$\dot{m}_v = r C_{d_v} A_P \left\{ \frac{2gRT_c}{g-1} \left[P_r^{(g-1)/g} - 1 \right] \Big/ \left[\frac{1}{f^2} P_r^{2/g} - 1 \right] \right\}^{1/2} \tag{20}$$

being

$$P_r = P/P_c \tag{21}$$

$$f = A_v / A_P \tag{22}$$

while for sonic flow

$$\dot{m}_v = r_o C_{d_v} A_v f \sqrt{gRT_o} \Big/ K^{2/(g-1)} \tag{23}$$

where coefficient *K* is given by the solution of

$$f^2 = \left\{ \frac{g+1}{g-1} - \frac{2}{g-1} K^2 \right\} K^{4/(g-1)} \tag{24}$$

In case of reverse flow, from cylinder to pipe through the valve, it is similar to direct flow in the intake pipe section, previously discussed, considering cylinder pressure and temperature conditions as the stagnation conditions.

Throttle Body: The modeling of the throttle body has been done for a fixed flow direction according to the previously discussed two cases; for example : in case of direct flow the behavior of the flow in the first part of the throttle valve can be described as the direct flow in the intake pipe section, while the following part, the behavior is similar to that of reverse flow in the intake pipe section.

By-Junction: The by-junctions can be easily represented with or without a pressure loss according to the precision required. The application of the conservation equations between the all adjacent capacities which represent the branch is sufficient to describe the flow conditions.

4. Model Validation

The MOIC has been validated according to theoretical and experimental data. In the first case, the solution given by Method of Characteristics has been considered as a reference: Simple situations have been considered in order to evaluate the ability of the MOIC of predicting basic unsteady situations. In the second case, comparisons have been done with respect to pressures and masses inducted measured on an experimental engine test bench.

Theoretical Comparison:

Two situations have been studied representing :

A. Single pipe (one meter long) with an inlet plenum conditions at the upstream and nozzle conditions downstream.
B. Cylinder emptying through a valve .

In the case A the following two cases have been considered:

1. Constant pressure difference at pipe ends was set to 0.5 bar.
2. Variable pressure difference at pipe ends given by a sinusoidal law 0.15(1+sin(250t)).

Figure 2 shows the pressure and the velocity evaluated on three points as a function of time in case of fully open pipe ends. The agreement between the two is satisfactory in spite of the reduced number of capacities considered (11) and the important unsteady effects are present.

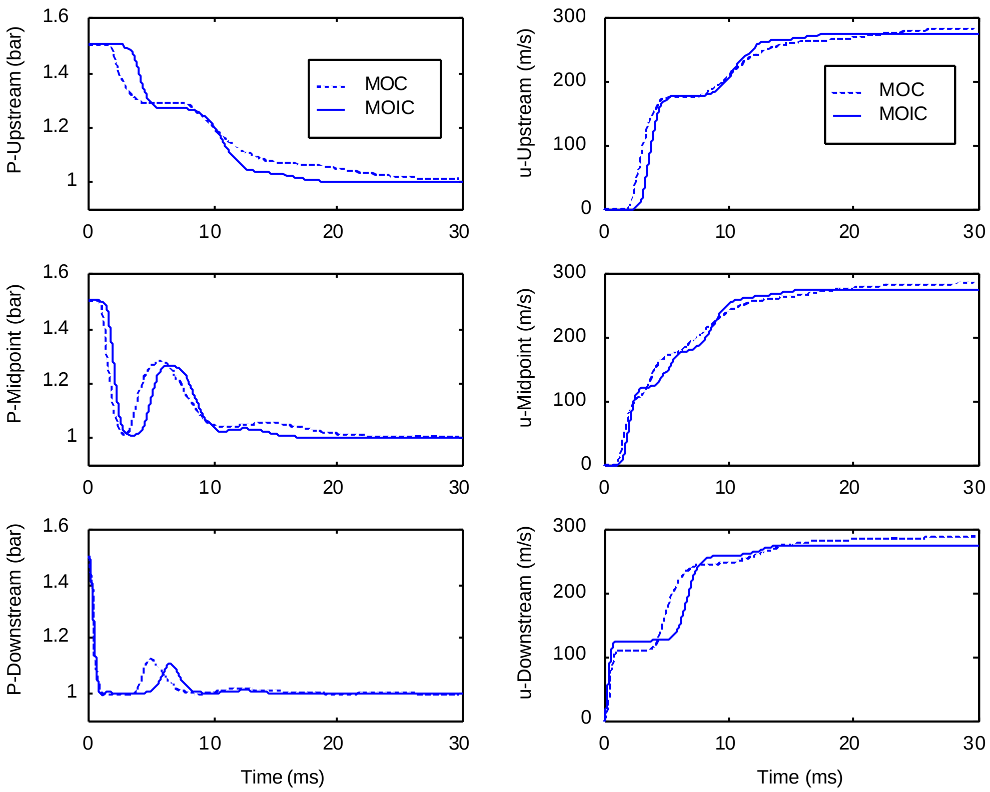

Figure 2: Up and down stream wide open end under constant pressure difference.

Figure 3 witnesses the same degree of accuracy when the pressure difference at pipe ends is variable. In figure 3.a the throttle ratio has been set to 0.5 while in figure 3.b equals 0.15.

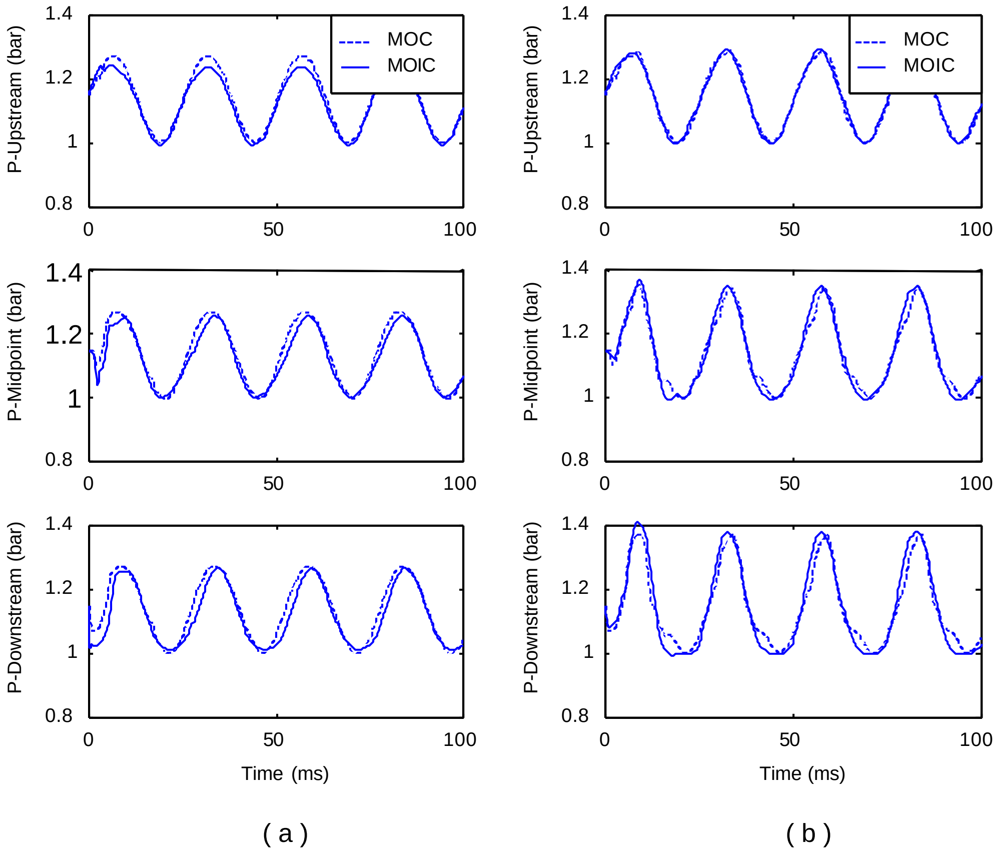

Figure 3: Upstream open end and downstream partial open end with variable pressure difference.

Along all the pipe, pressure transients are represented with right sequence in time and amplitude. The root mean square error is of the order of 1-3 percent and varies as a function of number of capacities as shown in figure 4. In which the study case given by figure 2 was considered, and the reference was the solution recommend by Benson [14] to this study case using MOC. The errors have been determined for the first 25 ms in which the transient behavior take place. Considering the results given by figure 4, the rms. (root mean square) error is higher if small number of capacities is used, then tends to reach the minimum but uneven values in the three points considered when 6-9 capacities is used, after that tends to be constant as the number of capacities increase.

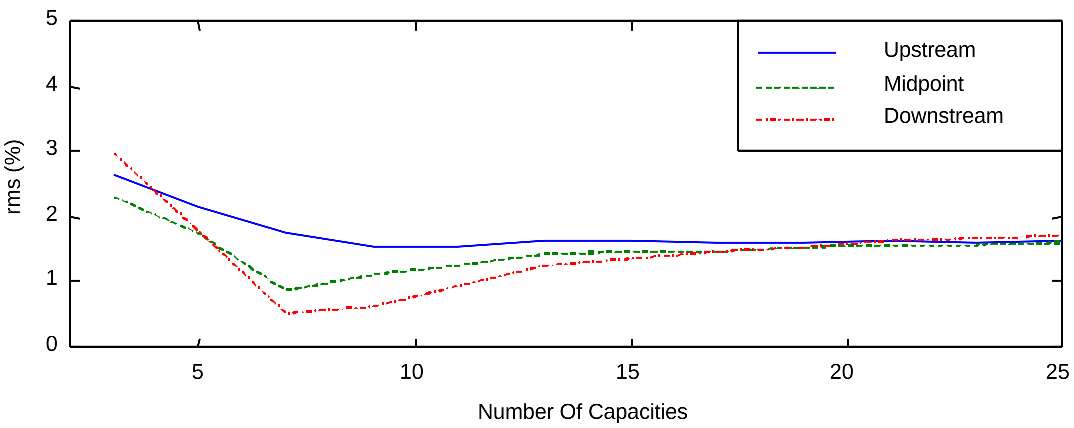

Figure 4: Root Mean Square (rms.) error in three different points as a function of number of capacities.

Figure 5 gives a comparisons between the computing time required by the MOC and MOIC. The strong reduction in the computing time is clearly obvious.

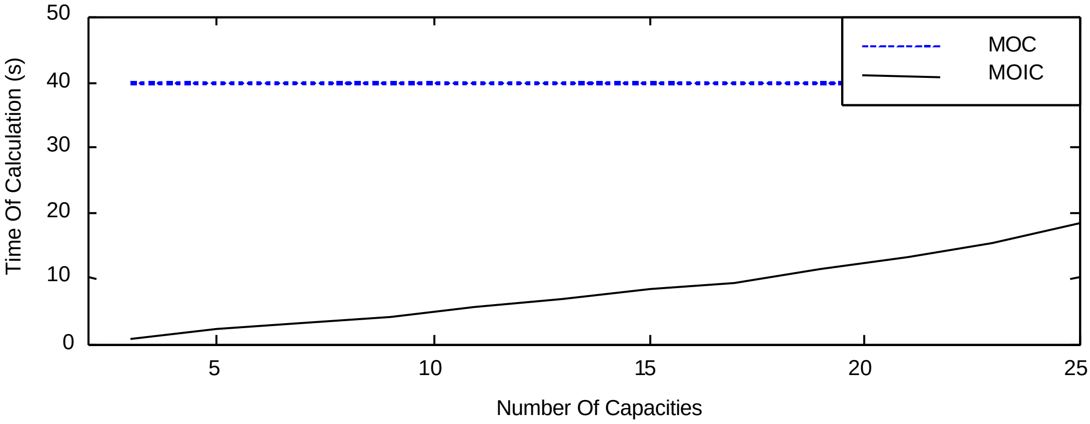

Figure 5: Comparison between MOC and MOIC calculation times for the 50 ms period.

In case B, the cylinder has been represented as a tube of 250 mm long with a valve opening represented in figure 6. The initial pressure difference has been set to 5.0 bar. Figure 7 shows the pressure variations inside cylinder in three

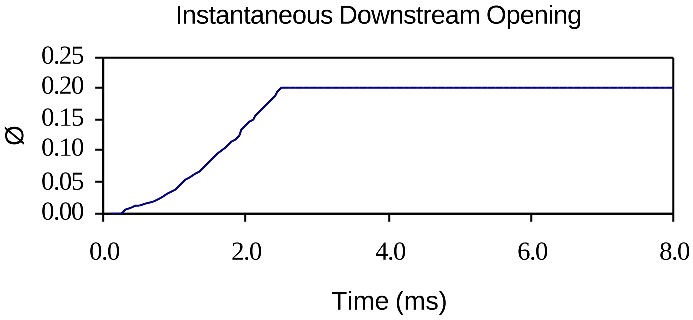

Figure 6: Downstream opening as a function of time.

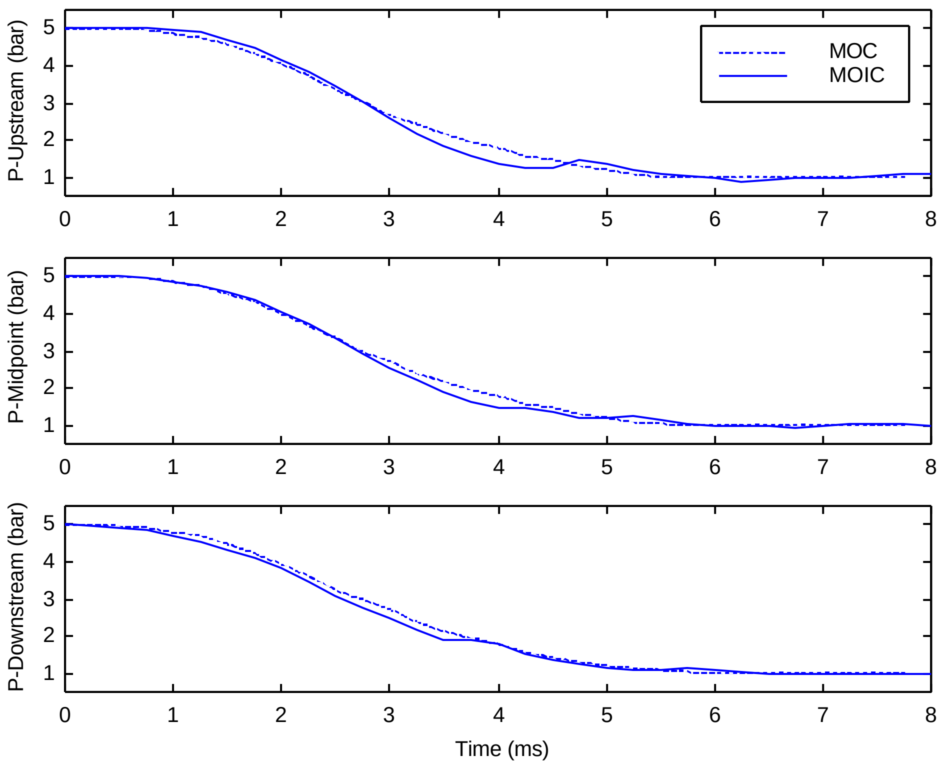

Figure 7: Upstream closed end and downstream time dependent open end (Cylinder Emptying).

locations during the voiding process. In spite of severe flow dynamic transients (sonic flow occurs downstream), the MOIC appears able to describe with a good accuracy such situations.

Experimental Validation:

The proposed method has been validated by simulating a number of situations measured in internal combustion engines intake manifolds. Experimental data have been obtained from an unsteady test bench at C.R.F (Fiat Research Center). A constant 60 mbar pressure difference has been applied to a real intake geometry of a 2500 cc high-speed five-cylinder engine. The camshaft was electrically actuated at various speed to reproduce real situations. Air mass flow rate was measured by laminar flow meter Cussons P7205 and instantaneous pressures by piezoelectric transducer (Endevco 8520A-50). Figure 8 gives a sketch of the intake manifold, runner-only configuration. The experimental pressure values refer to point located in the injection set are 121.8 mm from cylinder edge .

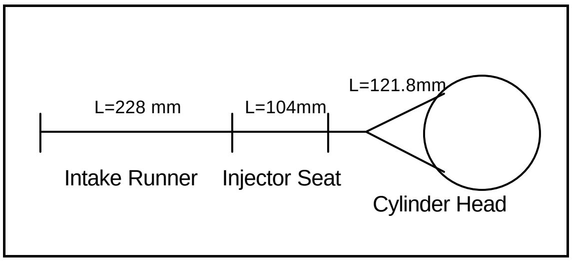

Figure 8: Intake Manifold -Runner Only Configuration.

Figure 9 shows a comparison, at various engine speeds (2000, 3000, 4000, 5000 rpm), between theoretical and experimental results. Five capacities has been chosen to reproduce the intake geometry of figure 8. Agreement appears to be quite satisfactory at all engine speeds and during the over all engine cycle the wave phenomena are sufficiently reproduced as a sequence in time and absolute amplitude values. Differences are more evident at medium speeds where engine tuning more effective: in this case only a fully one dimensional transient modeling is able to reproduce the behavior.

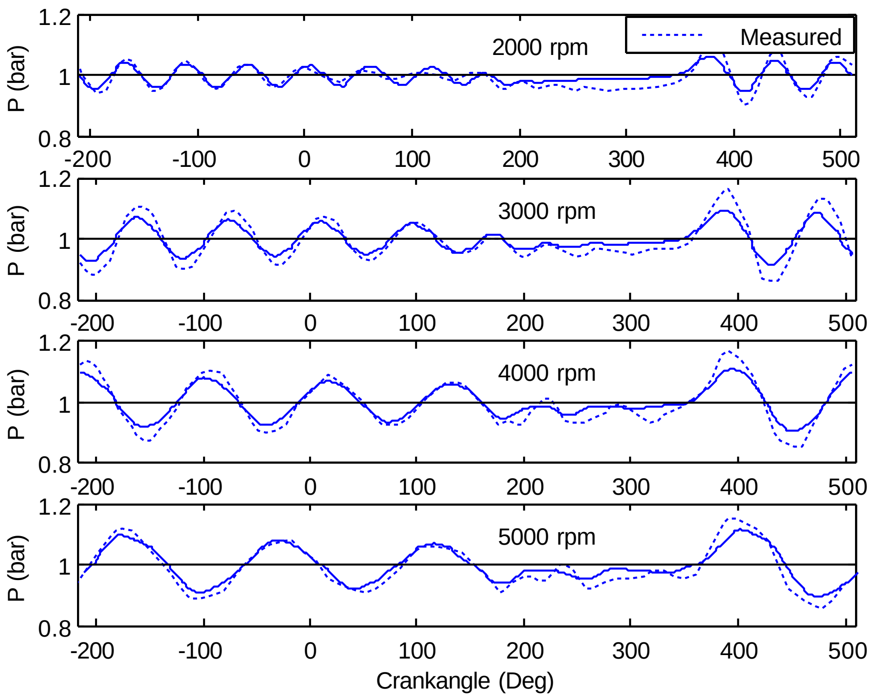

Figure 9: Comparison between calculated and measured pressure values.
(where 0 is the BDC of the exhaust stroke).

The proposed MOIC appears suitable to predict the air mass inducted into the cylinder which is the key point for A/F model based control.

Figure 10 shows the air mass inducted per cycle as a function of engine speed. The values have been normalized with respect to the value of 2000 rpm. The agreement between measured and calculated results is satisfactory accurate, which demonstrates the ability of the method to predict wave effects inside the tested intake system. The maximum error is of the order of 12 percent as shown in figure 10, while the root mean square error of 5-6 percent results.

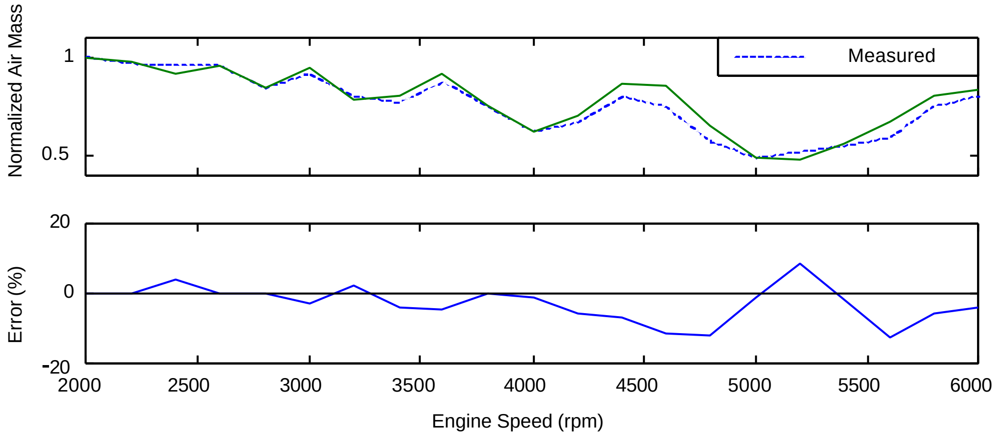

Figure 10: Comparison between measured and calculated normalized mass flow rates as a function of engine speed.

5. Conclusions

A new method for modeling air dynamics inside SI engines intake manifolds has been proposed. Its main application is in the A/F model based control where the compromise between the accuracy and the minimal computing time is strongly recommended.

The method can be considered as a further evolution of the mean value engine models, widely used in the literature, with respect to the wave propagation effects that can be described keeping a lumped parameter approach. Complex intake system geometries can be represented easily which give a room to the application in the practical situations.

The method does not require empirical data, being based only on the application of the conservation equations suitably formulated to simplify the solution.

The proposed method has been validated using solutions obtained by Method of Characteristics and comparing its predictions with experimental data in terms of instantaneous pressure and air mass inducted. In all the cases the prediction of the degree of accuracy appears to be suitable for A/F control.

Acknowledgments

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Nomenclature

P Pressure
T Temperature
u Flow velocity
m Air mass
$\dot{m}$ Air mass flow rate

A Cross-sectional area
V Capacity volume
f Friction factor
g Ratio of specific heats
R Air gas constant

f Restriction coefficient
t Time

Suffixes:
i,e,o Inlet, Exit, Upstream
t,v,c Throttle, Valve, Cylinder