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Daniele Bortoluzzi, Vittore Cossalter, Alberto Doria

Department of Mechanical Engineering, University of Padua, Italy

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THE EFFECT OF TUNABLE RESONATORS ON THE VOLUMETRIC EFFICIENCY OF AN ENGINE

Daniele Bortoluzzi, Vittore Cossalter, Alberto Doria
Department of Mechanical Engineering, University of Padua, Italy

ABSTRACT

The acoustic phenomena which take place inside an intake system greatly influence the volumetric efficiency of an engine. This paper deals with the problem of improving the volumetric efficiency of a single cylinder engine for a wide range of frequencies. The proposed solution is the addition of tunable resonators to the intake system. The variation of resonator tuning makes it possible to exploit the acoustic vibrations for a wide range of piston frequencies in the best way. Several intake systems equipped with resonators are studied using analytical methods. The best results are obtained when an in-series resonator with a variable volume is placed near the cylinder and when a side-branch resonator with a variable cross section is connected to a constant volume resonator. Several results are presented which deal with the behavior of the intake system during the induction stroke and which show the effectiveness of tunable resonators.

INTRODUCTION

The volumetric efficiency of an engine depends both on quasi-static phenomena, like charge heating and choking, and on acoustic phenomena, which are related to the inertia and compressibility of the charge inside the intake system. The acoustic phenomena are important especially in sport car and motorcycle engines, since the piston frequency is high and high power is required. When the dimensions of the intake system are smaller than the wavelength of sound, the acoustic study can be carried out by means of a lumped element approach, in which the volumes (e.g. the cylinders) are considered to be lumped acoustic springs and the ducts are considered to be lumped acoustic masses [1] [2] [3].

For a single cylinder engine equipped with an intake pipe, the volumetric efficiency reaches the maximum value when the natural Helmholtz frequency of the cylinder and pipe is about double the piston frequency. Multi-cylinder engines sometimes show two peaks of the volumetric efficiency, which are related to the two natural frequencies of the system [1], [4], [5], [6].

This paper focuses on the improvement of the volumetric efficiency of single-cylinder engines for a wide range of piston frequencies. The lumped element approach gives some indications for achieving this goal. The simpler solution is the variation of the intake duct length, but it is impracticable in many cases.

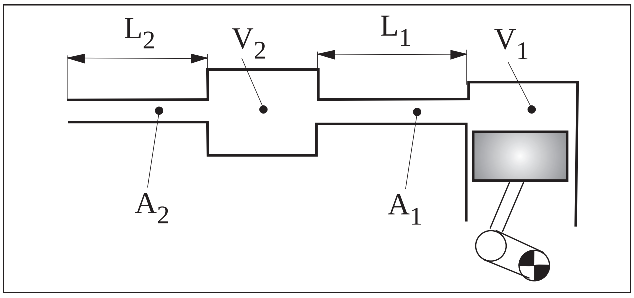

Fig. 1 Intake system with in-series resonator

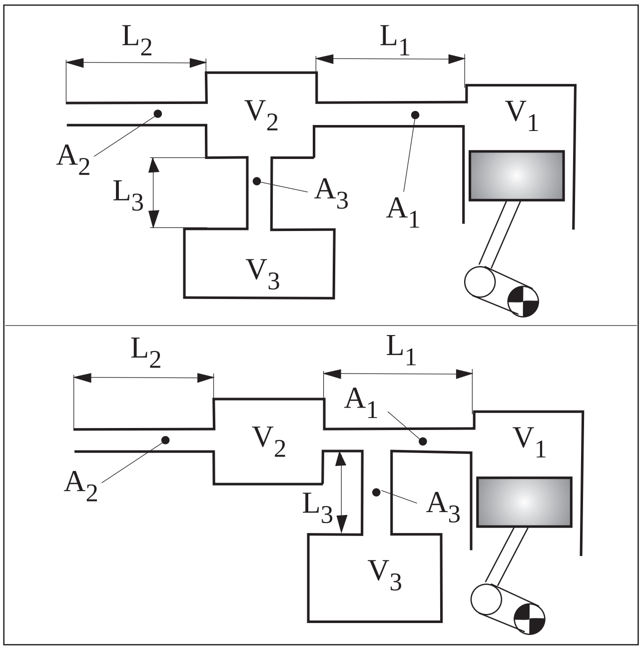

Fig.2 Intake systems with side-branch resonators

The solution proposed here is the addition of one or more tunable resonators to the intake system. Figure 1 shows an intake system equipped with an in-series resonator, while figure 2 shows two intake systems equipped with both an in series-resonator and a side-branch resonator. With this solution the resonance frequencies of the intake system depend on the volumes and ducts of the resonators and can be tuned to different piston frequencies by regulating the volumes of the resonators or the ducts' sections of the side-branch resonators.

A simple mathematical model for the time domain acoustic analysis of intake systems equipped with in-series and side-branch resonators is presented. It is based on the lumped element approach and makes the analytical calculation of the intake system response possible. This solution makes it easier to carry out a parametric study of the influence of resonator dimensions on the intake system's behavior. A filling index (λ_V) is introduced; it is defined as the ratio between the volume of the fluid which enters the cylinder and the volume generated by the piston stroke and it takes into account only the acoustic phenomena in the intake system. Then, in order to carry out the analysis in the field of the high frequencies, an improved mathematical model is introduced; it assumes a linear distribution of the velocities inside the ducts and the volumes.

Several intake systems equipped with resonators are studied and in particular the configurations which are more sensitive to the variation of resonator dimensions are found.

Finally the fluid motion inside the intake systems is studied in more detail using a finite difference code for the fluid-dynamic analysis: hence the validity and the limits of the lumped element models are pointed out and discussed.

MATHEMATICAL MODEL

Strictly speaking a non-linear model is needed to study the dynamics of the intake systems because the geometry changes owing to the motion of the valves and the piston. The aim of this section is the development of straightforward methods for the analysis and the comparison of intake systems equipped with resonators; therefore, some simplifications are carried out in order to avoid developing a non-linear model.

Only the suction phase is considered, when the outlet valve is closed, the inlet valve is open and the cylinder is connected to the intake manifolds. The wave effects which take place in the intake system when the inlet valve is closed are not taken into account, because they cause only small ripples in the volumetric efficiency curve [6].

The volume of fluid which enters the cylinder depends on the timing of the inlet valve. The inlet valve is assumed to have an ideal behavior: it opens instantaneously at the top dead center (t.d.c.) and closes instantaneously at the bottom dead center (b.d.c.). This assumption is precautionary, since it is impossible to completely exploit the inertia of the column of fluid with this valve timing. The pressure wave and the reverse flow (from the cylinder to the inlet duct) that take place if the inlet valve begins opening during the later phase of the exhaust stroke are not taken into account, because usually valve overlap is small in four stroke engines [5] and valve opening timing point does not influence significantly engine performance [7]. The volume of the cylinder is considered constant and equal to the average value V_1:

$$V_1 = \frac{V_{gen}}{2}\frac{R+1}{R-1} \qquad (1)$$

where V_{gen} is the volume generated by the piston stroke and R the compression ratio [5].

The actual system is excited by the motion of the piston. If the crank is much shorter than the connecting rod, the position of the piston, with respect to the top dead center, is given by:

$$x_p \approx r\left(1-\cos(\omega_m t)\right) \qquad (2)$$

where r is the crank length and ω_m the piston frequency. The acoustic model is excited by the pressure fluctuations caused by the piston motion. They are calculated by means of the following expression:

$$\Delta p = \rho c^2 \frac{A_p}{V_1} r\left(1-\cos(\omega_m t)\right) \qquad (3)$$

where A_p is the piston cross-section, ρ the density, c the sound speed; the cylinder volume is assumed constant and equal to the average value V_1.

SIMPLE LUMPED ELEMENT MODEL (S.L.E.M.) - The intake system equipped with both an in-series resonator and a side branch resonator (figure 2) can be studied by means of a three degree of freedom simple lumped element model. The equations of the acoustic vibrations are the following:

$$[M]\begin{Bmatrix}\ddot{x}_1\\ \ddot{x}_2\\ \ddot{x}_3\end{Bmatrix} + [C]\begin{Bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\end{Bmatrix} + [K]\begin{Bmatrix}x_1\\ x_2\\ x_3\end{Bmatrix} = \begin{Bmatrix}1\\ 0\\ 0\end{Bmatrix}\rho c^2 \frac{A_1 A_p}{V_1} r\left(1-\cos\omega_m t\right) \qquad (4)$$

$$[M] = \begin{bmatrix}\rho L_1 A_1 & 0 & 0\\ 0 & \rho L_2 A_2 & 0\\ 0 & 0 & \rho L_3 A_3\end{bmatrix} \qquad (5)$$

$$[C] = \alpha[M] + \beta[K] \qquad (6)$$

$$[K] = \rho c^2 \begin{bmatrix}\frac{A_1^2}{V_1}+\frac{A_1^2}{V_2} & -\frac{A_1 A_2}{V_2} & \frac{A_1 A_3}{V_2}\\ -\frac{A_1 A_2}{V_2} & \frac{A_2^2}{V_2} & -\frac{A_2 A_3}{V_2}\\ \frac{A_1 A_3}{V_2} & -\frac{A_2 A_3}{V_2} & \frac{A_3^2}{V_3}+\frac{A_3^2}{V_2}\end{bmatrix} \qquad (7)$$

where x_1, x_2 and x_3 are the displacements of the fluid in the ducts, L_1, L_2, L_3 the duct lengths, A_1, A_2, A_3 the duct cross-sections, V_2, V_3 the volumes. The lengths of the ducts can be corrected taking into account that a portion of the fluid contained in the volumes participates in the motion [8].

[*M*] is the mass matrix which takes into account the inertia of the fluid contained in the ducts, [*K*] is the stiffness matrix which takes into account the compressibility of the fluid contained in the volumes. The estimate of the damping coefficients of the system is difficult to make because the motion in the ducts is transient and several different phenomena contribute to the dissipation of the energy: viscous losses, turbulent losses, radiation losses and area discontinuities. Hence, the damping matrix [*C*] is assumed to be a linear combination of the mass and stiffness matrices (α and β are two coefficients). This assumption makes it possible to analyze the acoustic vibrations using the modal approach and the introduction of modal dampings, which can be varied in a parametric way.

Equations 4 hold even when the system has an in-series resonator only. In fact, if V_3 tends to zero, the stiffness of the side-branch resonator tends to infinite and hence the fluid contained in the third duct does not move. In these conditions, the first and second equations describe the acoustic vibrations of the system equipped with the in-series resonator only (which has two degree of freedom).

The response of the intake system to the transient excitation caused by the piston stroke is calculated taking into account the forced response and the free response and setting the initial conditions. The modal approach is followed. First the three natural frequencies and the modal vectors are calculated. Then, by means of the modal matrix [*U*] (which is normalized with respect to the mass matrix), equations 4 are transformed into a set of three independent equations in modal coordinates; their solution is the following:

$$\ddot{\eta}_i = e^{-\zeta_i\omega_i t}\left(A_i \cos\sqrt{1-\zeta_i^2}\,\omega_i t + B_i \sin\sqrt{1-\zeta_i^2}\,\omega_i t\right) + \frac{N_i^0}{\omega_i^2} + \frac{N_i}{\omega_i^2}\frac{1}{\sqrt{\left(1-\left(\frac{\omega_m}{\omega_i}\right)^2\right)^2 + \left(2\zeta_i\frac{\omega_m}{\omega_i}\right)^2}}\cos(\omega_m t + \varphi_i)$$

$$\varphi_i = \arctan 2\,\frac{-2\zeta_i\,\frac{\omega_m}{\omega_i}}{1-\left(\frac{\omega_m}{\omega_i}\right)^2} \quad i = 1,3 \qquad (8)$$

where η_i are the modal coordinates, ω_i the natural frequencies, ζ_i the modal dampings, φ_i the phases, $N°_i$ the modal forces produced by the constant term of the excitation, N_i the modal forces produced by the harmonic term of the excitation and A_i and B_i constants.

The initial condition in modal coordinates are introduced and finally the motion of the physical coordinates is calculated using the modal matrix:

$$\begin{Bmatrix} x_1 \\ x_2 \\ x_3 \end{Bmatrix} = [U]\begin{Bmatrix} \eta_1 \\ \eta_2 \\ \eta_3 \end{Bmatrix} \qquad (9)$$

The forced response of the intake system is a function of the piston frequency and depends on the engine parameters (r, A_p, R) and on the geometric parameters of the intake system. The analysis of equations 4 shows that the influence of the geometry can be expressed by means of A_1, the ratio L_1/A_1 (the impedance of the first duct [1]) and the following non dimensional ratios: the area ratio $A=A_2/A_1$, the length ratio $L=L_2/L_1$, the volume ratio $V=V_2/V_1$, the side resonator area ratio $A_r=A_3/A_1$, the side resonator length ratio $L_r=L_3/L_1$ and the side resonator volume ratio $V_r = V_3/V_1$.

The volume of charge which enters the cylinder is calculated by integrating the velocity in section A_1, hence the filling index is:

$$\lambda_v = \frac{A_1\int_0^{t^*}\dot{x}_1\,dt}{2rA_p} \qquad (10)$$

where t* is the instant when the valve closes. The filling index is related to the volumetric efficiency η_v by the relation:

$$\eta_v = \lambda_v\,\frac{\tilde{\rho}}{\rho_{a.i}} \qquad (11)$$

where $\tilde{\rho}$ is the average density of the charge which enters the cylinder and $\rho_{a.i}$ the inlet air density. Since the filling index does not consider the variations in density it is appropriate for the analysis of inertia and acoustic phenomena only.

IMPROVED LUMPED ELEMENT MODEL (I.L.E.M.) - The simple lumped element model holds in a range of piston frequencies where the geometric parameters satisfy the following conditions:

$$V_i^{1/3} \lhd\lhd \frac{2\pi c}{\omega_m} \quad A_i^{1/2} \lhd\lhd \frac{2\pi c}{\omega_m} \quad L_i \lhd\lhd \frac{2\pi c}{\omega_m} \quad i = 1,2,3 \qquad (12)$$

Hence, if a duct dimension is rather large, the simple model gives poor results in the field of the high piston frequencies.

To overcome this problem an improved acoustic model has been developed in which both the inertia and the compressibility of the fluid are taken into account in every element of the intake system (ducts, resonators, cylinders). In this model the velocity of the fluid which is contained within the element changes in linear way from the value at the inlet of the element ($\dot{x}_i$) to the value at the outlet of the element ($\dot{x}_{i+1}$).

The kinetic energy of the fluid inside an element of the intake system is given by:

$$E_k = \frac{1}{6}\rho A_i L_i\left(\dot{x}_i^2 + \dot{x}_{i+1}^2 + \dot{x}_i\dot{x}_{i+1}\right) \qquad (13)$$

The potential energy, which is caused by the compression of the fluid inside an element of the intake system, is given by:

$$E_p = \frac{1}{2}\rho c^2 \frac{A_i^2}{V_i}\left(x_i - x_{i+1}\right)^2 \qquad (14)$$

where x_i and x_{i+1} are the displacements of the fluid at the two edges of the element. The total kinetic and potential energy are calculated by adding the terms of the various elements.

Then the continuity equation is used to correlate the velocity at the outlet of one element with the velocity at the inlet of the following element:

$$\dot{x}_{i+1}A_i = \dot{x}_j A_j \qquad (15)$$

The acoustic equations are written in the matrix form (4) by means of the Lagrange's method; the displacements of the fluid at the inlet of the elements are the generalized coordinates. The matrices of the intake system equipped with an in-series resonator have dimensions 4x4 because four generalized coordinates are needed to model the system.

Finally the acoustic equations are solved with the modal approach described in the previous section.

COMPARISON WITH EXPERIMENTAL RESULTS

Engelman [1] showed that three-cylinder and four-cylinder intake manifolds can be modeled as two degree of freedom systems. Hence the scheme of figure 1 can be the model of a multi-cylinder engine as well. In this case the first volume (V_1) is the cylinder undergoing its intake stroke, the first duct (A_1, L_1) is the runner to the branch point, the second volume (V_2) represents the volume of the idle pipes and the second duct (A_2, L_2) is the feeder pipe. Hence, in order to have an experimental confirmation, a comparison is made between the filling index predicted by the lumped element models and the torque generated by an actual four-cylinder engine.

In particular the intake system described in reference [4] is studied and the results are compared with the torque curve presented in the same reference.

The torque which is generated by the engine is related to the filling index by the following equation:

$$T = \frac{\eta_f \lambda_v \frac{\tilde{\rho}}{\rho_{a.i}} V_{gen} Q_{HV} \rho_{a.i} (F/A)}{4\pi} \qquad (16)$$

where: η_f is the fuel conversion efficiency, Q_{HV} the fuel heating value, *F/A* the fuel/air ratio.

The results are summarized in figure 3, where the filling indexes predicted by lumped element models and the experimental values of the generated torque are plotted.

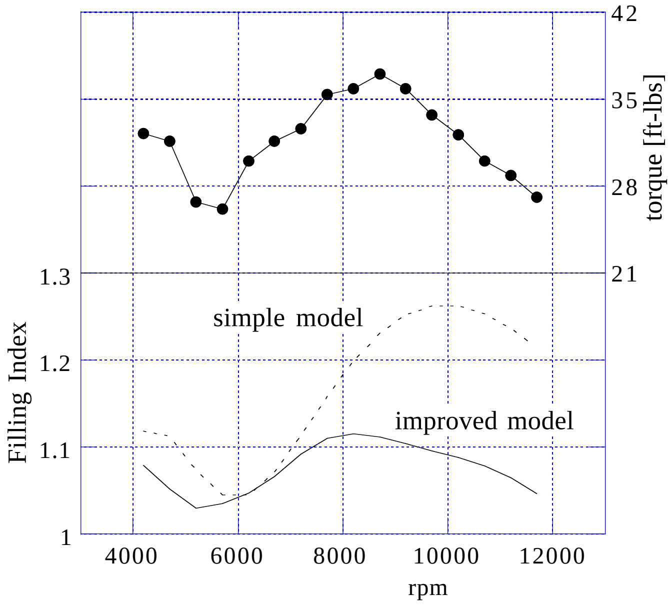

Fig.3 Comparison with experimental results

The simple lumped element model predicts two peaks of the filling index and there is a quite good agreement between their frequencies and the torque peak frequencies. The improved mathematical model predicts peak frequencies of the filling index which are closer to the peak frequencies of the torque. It is worth pointing out that there is a qualitative good agreement between the analytical results and the experimental data, even if the filling index is calculated assuming an ideal behavior of the valves.

ANALYSIS OF INTAKE MANIFOLDS WITH IN-SERIES RESONATORS

First the intake systems equipped with acoustic resonators in series with the cylinder are considered (see figure 1). If the dimensions of the ducts and volumes are much smaller than the sound wavelength they behave like two degree of freedom systems and can be studied by means of the S.L.E.M.. In practice, these systems can be built by exploiting the air-box, the manifold or any other cavity which is present along the intake line as a resonator.

Three configurations which differ in the length ratio are analyzed, in the following they are named short-long, long-short and short-short configurations. Their geometric parameters are summarized in table 1, actually each configuration can be realized with different lengths and cross sections. The three configurations are tested considering a 500cc engine having *r*=0.046m, A_p= 0.005411m^2, *R*=9.

Table 1: Configurations

Configuration	L_1/A_1	L_2/L_1	A_2/A_1	V_2/V_1
short -long	147	7.000	1.	variable
long-short	1031	0.143	1.	variable

short -short	147	0.908	1.	variable

Figure 4 shows the filling index of the short-long configuration against the piston frequency; the parameter of the curves is the volume ratio (*V*). When *V*=0 (no resonator) the filling index shows only one peak at about 2700 rpm. The ratio between the natural frequency and the piston frequency of the maximum filling is 1.83.

When the in-series resonator is introduced, the filling index shows two peaks. The first peak is in the range of the low frequencies and is lower than the peak of the system without resonator. The second peak is in the range of the high frequencies; it becomes higher when the resonator volume increases and causes a very significant improvement of the filling index. With *V*=10, the filling index is about 1 from 0 to 10000 rpm and the ratio between the first natural frequency and the first frequency of the maximum filling is 1.80, whereas the ratio between the second natural frequency and the second frequency of the maximum filling is 1.83.

The calculations show that, when the *V* ratio is high, the second natural frequency is rather different from that of the system without a resonator. This happens because, when the volume of the in-series resonator is very large, the system behaves like the cylinder connected to the atmosphere by the short duct only.

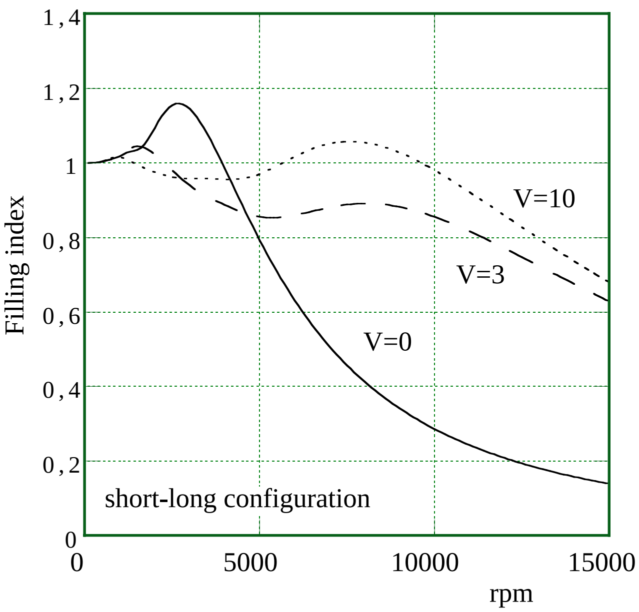

Fig.4 Filling index of the short-long configuration

The optimum filling of the cylinder in the range of 0÷15000 rpm can be achieved with an on-off control of the resonator volume. In fact, in the range of 0÷4000 rpm the resonator volume has to be minimized in order to exploit the high peak of the system without a resonator, whereas above 4000 rpm, the resonator volume has to be maximized to exploit the peak related to the second natural frequency. This configuration has the drawback of requiring a lot of space in series with the cylinder.

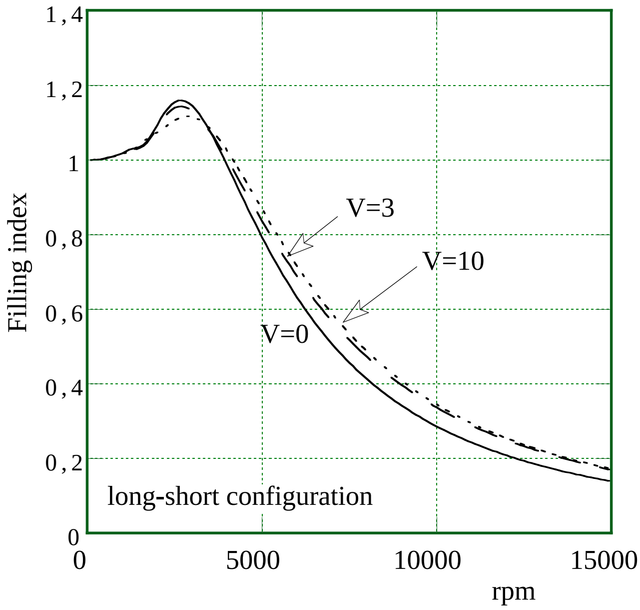

Fig.5 Filling index of the long-short configuration

The filling index of the long-short configuration is represented in figure 5. With this configuration, even if the volume of the in-series resonator is large, the filling index shows only one peak at about 2700 rpm and is very low at the high piston frequencies. This happens because this system, when equipped with a large resonator, behaves like the cylinder connected to the atmosphere by a duct a bit shorter than that of the system with *V*=0. The second natural frequency, which is introduced by the resonator, is close to the first and does not influence the filling index in the range of the high piston frequencies.

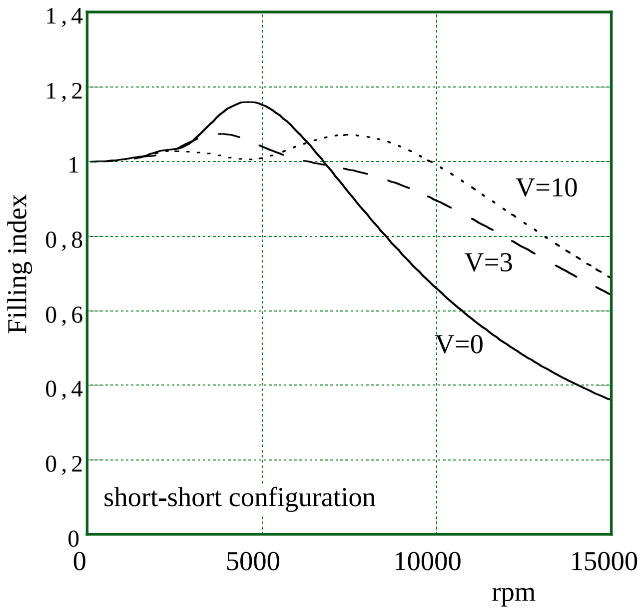

Fig.6 Filling index of the short-short configuration

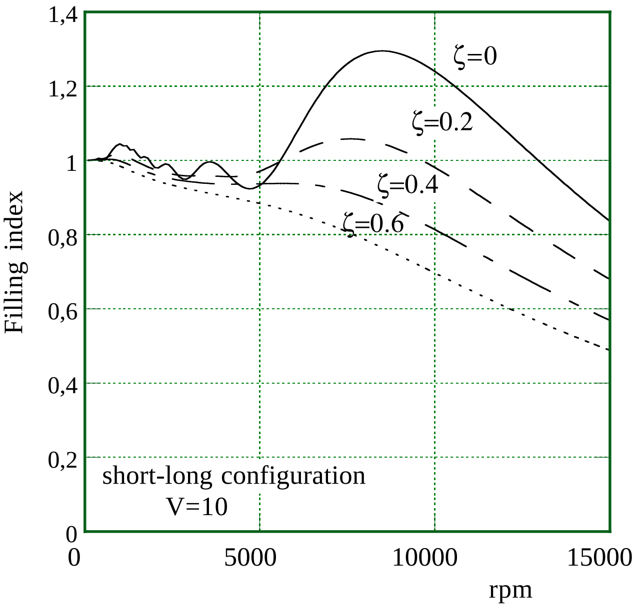

Fig.7 Effect of the damping on the filling index

Figure 6 shows the filling index of the short-short system. If *V*=0 the filling index is >1 up to 6750 rpm and there is only one peak at 4600 rpm, the ratio between the natural frequency and the frequency of the maximum filling is 1.84. When *V* increases, the filling index increases in the field of the high frequencies. With *V*=10 there are two peaks of the filling index; the ratios between the natural frequencies and the frequencies of maximum filling are 1.72 and 1.90 respectively. The filling index in the range 0÷15000 rpm can be optimized by means of an on-off control like the one described for the short-long system.

The above mentioned analyses have been carried out considering the modal dampings of the system equal to 0.2. Figure 7, which refers to the short-long configuration with *V*=10, highlights the effect of damping: the increase of the modal damping decreases the peaks of the filling index (especially the second). Nevertheless, the resonator improves the filling index in the high frequency range.

NUMERICAL SIMULATIONS

At the present time there are several codes which make it possible to perform the numerical analysis of the intake and exhaust systems of the engines. Some codes solve the differential equations of one-dimensional unsteady compressible flow by means of the mesh method of characteristics [9]. Other one-dimensional codes are based on the Lax-Wendroff method [10] [5] or on the finite volume approach [11].

In this paper the numerical simulations of the fluid-dynamic behavior of the intake systems are carried out by means of the Fluent code [12]. Fluent is a powerful finite-differences code which has been already used to model flows in turbomachines and engines. It has the capability of modelling compressible and incompressible flows in 2D and 3D geometries taking into account turbulence, the phenomena which take place where there are abrupt area changes, heat exchange and chemical reactions too.

The aims of numerical simulations are to obtain a confirmation of the analytical results and to analyze the behavior of the intake systems when the duct dimensions are not much smaller than the sound wavelength.

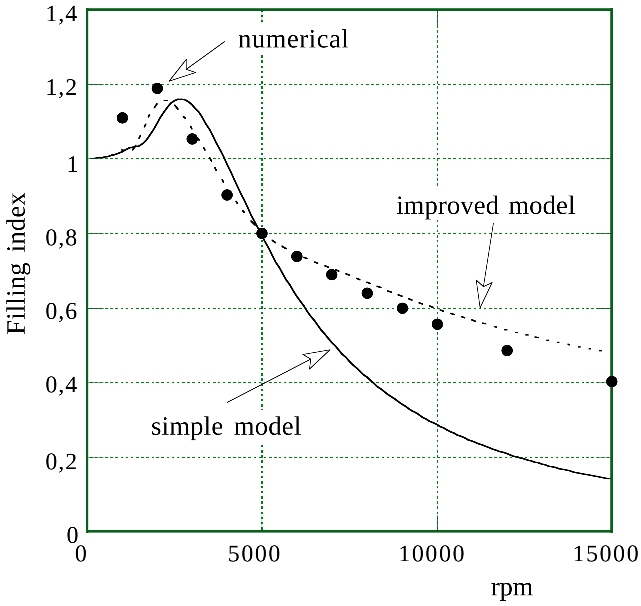

Fig.8 Intake system with a long duct: comparison between the numerical and the analytical values of the filling index

Some of the hypotheses of the analytical model are maintained to simplify the problem. The geometry is assumed to be axisymmetric and fixed, the volume of the cylinder is considered constant and equal to the average value V_1. The fluid field is assumed to be bidimensional because the swirl motion is neglected. The turbulence is taken into account by means of the k-ε model and the physical properties of the fluid are those of air in normal conditions. The valve is assumed to have an ideal behavior. The numerical model is excited by a flux entering the constant volume cylinder, which is equal to the flux caused by the actual piston stroke.

Firstly a simple intake system with L_1=0.9m and A_1= 0.0006789m^2 is simulated. Since the duct is rather long, this case is useful to point out the limits of the lumped element models in the field of the high frequencies.

In figure 8 the values of the filling index which are calculated by means of the numerical method are compared with the values of the filling index which are calculated by means of the lumped element models.

The simple lumped element model predicts a peak frequency slightly higher than the numerical one and in the field of the high frequencies predicts values of the filling index which are lower than the numerical values. The differences at high frequencies are caused by the high order acoustic modes, which have an important effect when sound wavelength becomes comparable with duct dimensions (at 15000 rpm sound wavelength is 1.33 m).

The analytical results obtained by means of the improved lumped element model are very close to the numerical results both in the field of the low frequencies and in the field of the high frequencies. In fact four acoustic modes are taken into account by this model, which has four degree of freedom.

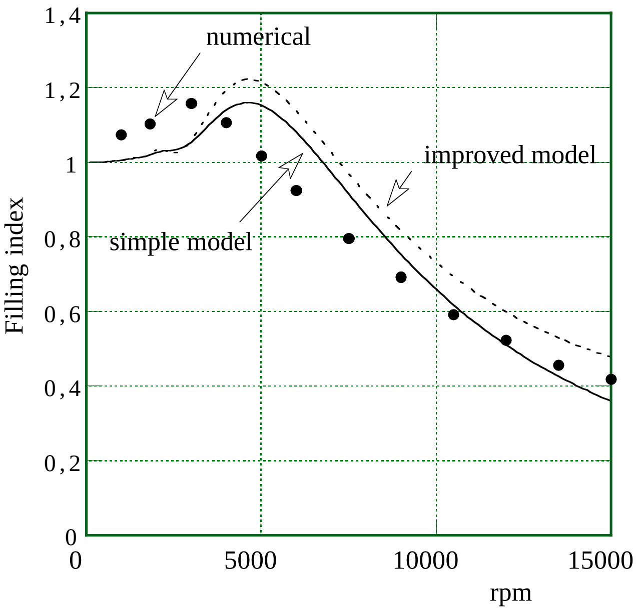

Fig.9 Intake system with a short duct: comparison between the numerical and the analytical values of the filling index

Then a short intake system without resonator (L_1=0.287m and A_1= 0.0006789m^2) is simulated. Figure 9 shows that there is a good agreement among the values of the filling index calculated with the three methods over the full range of frequencies because, even when the piston frequency is high, the duct length is smaller than sound wavelength.

The numerical code make it possible to compare the velocity field inside an intake system equipped with a resonator with the velocity field inside an intake system without a resonator.

The short-long configuration is considered and in figures 10 and 11 the velocity fields at half stroke and at the end of the stroke are presented. The piston frequency is 4000 rpm; the darker tones of gray represent higher velocities. At half stroke, the velocity of the charge which is entering the cylinder is higher in the system with the resonator (V=10) than in the system without a resonator (V=0). At the end of the stroke, in the system with the resonator, the velocity is very low in the entire intake system, because the filling is almost over, whereas in the system without a resonator, the velocity is still high because the fluid moves with a lag.

Finally the hypothesis of inlet valve closure at the b.d.c is disregarded and the effect of the actual instant when the inlet valve closes is studied by means of the numerical model. The inertia of the fluid column is completely exploited if the inlet valve closes when the inlet velocity is zero. Hence the optimum timing of the inlet valve is determined for each value of the piston frequency by looking for the instant when the inlet velocity is zero. Then, since the piston frequency is constant, the optimum timing is expressed in terms of the crank rotation after the b.d.c..

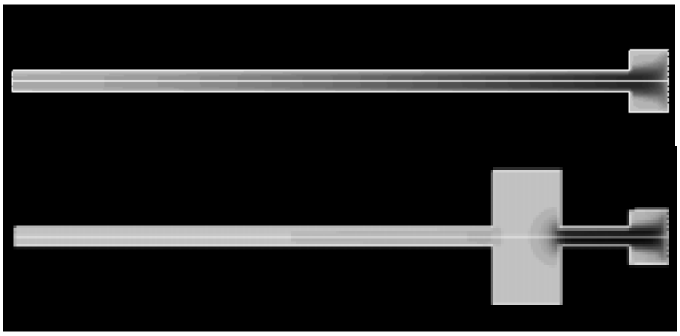

Fig.10 Example of velocity field at half stroke

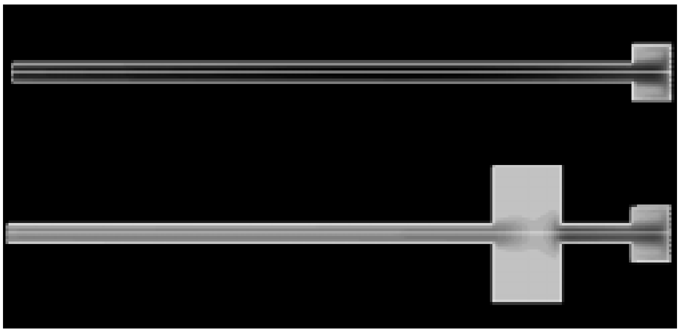

Fig.11 Example of velocity field at the end of the stroke

Figure 12 shows the optimum timing for a simple intake system with a long duct (L_1=0.9m and A_1=0.0006789m^2). The delay of the inlet valve's closing increases with the piston frequency but does not vary much above 5000 rpm; hence the optimum filling in the range of 5000÷15000 rpm can be achieved by means of a constant valve timing.

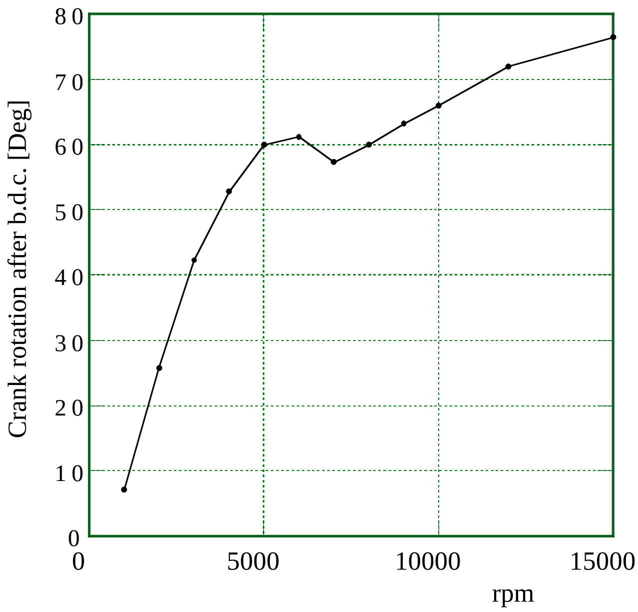

Fig.12 Optimum inlet valve timing for an intake system with a long duct

ANALYSIS OF INTAKE MANIFOLDS WITH SIDE-BRANCH RESONATORS

The intake systems equipped with side-branch resonators are represented in figure 2. If the dimensions of the ducts and volumes are much smaller than the sound wavelength the intake systems behave like three degree of freedom systems and can be studied by means of the S.L.E.M..

The presence of the third natural frequency increases the possibility of tuning. The system may be tuned by modifying not only ratios V and V_r but also by varying the cross section of the duct of the side-branch resonator (ratio A_r) because this adjustment does not cause pressure drops in the main flux which takes place in ducts 1 and 2.

Figure 13 shows the filling index of the intake system with a side-branch resonator which derives from the short-long configuration with V=3. The volume and duct length of the side-branch resonator are held constant (V_r=10 and L_r=1) whereas the duct cross section is regulated in the range A_r =0÷2. When the side duct is partially open (A_r=0.25), a new peak of the filling index occurs whose frequency is close to the first peak's. A further increase of the opening of the side duct (A_r=0.5÷1) moves the new peak towards the higher frequencies. When the side duct cross section is larger than that of the first duct (A_r=2) the new peak merges with the high frequency peak.

This behavior can be explained by analyzing the natural frequencies and the system's modes of vibration. Figure 14 shows that the first natural frequency is influenced by A_r only when A_r is very small. For this reason the filling index always shows a small peak in the field of low frequencies. The third natural frequency increases a bit with A_r, but is always in the range of 16000÷17000 rpm; therefore, there is always a peak of the filling index in the range of 7000÷8000 rpm. On the other hand, the second natural frequency increases markedly with A_r and hence the plots of the filling index show that the central peak moves towards the higher frequencies when A_r increases.

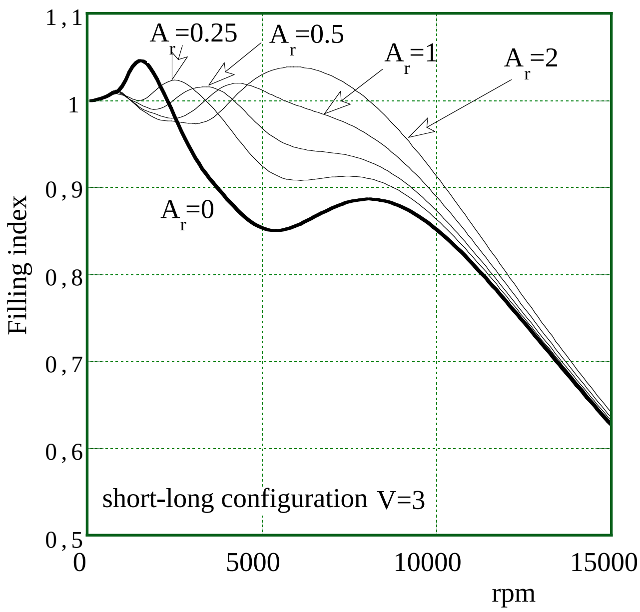

Fig.13 Filling index of the short-long configuration with the side-branch resonator

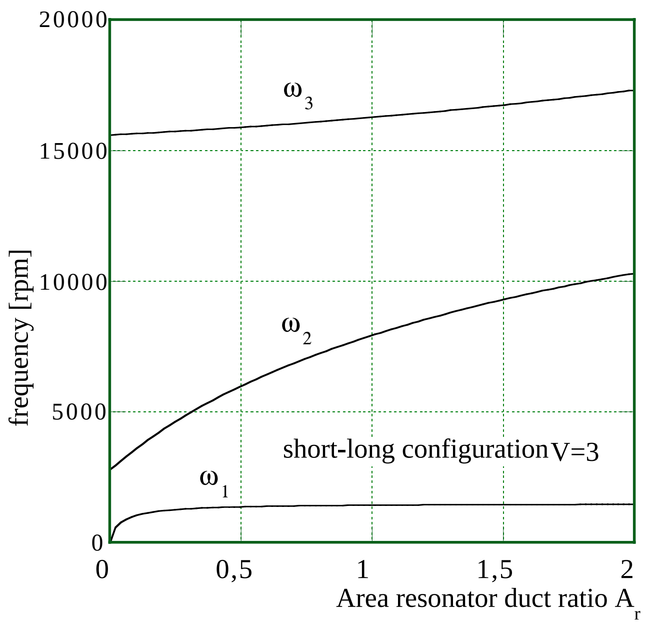

Fig.14 Influence of A_r on the natural frequencies

The elements of the modal vectors are proportional to the velocities inside the three ducts and, if A_r=A=1, they are proportional to the fluxes too (otherwise the different cross sections of the ducts have to be taken into account). Figure 15 shows the modes of the system with A_r=1, where the modal velocities are normalized by taking the velocity in the first duct as unity.

The resonator gives a large contribution to the filling of the cylinder when the second mode of vibration is excited, because a large flux leaves the side-branch resonator and enters the cylinder.

The performance of this intake system can be optimized with a continuous adjustment of the cross section of the side duct. In fact, this adjustment makes it possible to exploit the peak of the filling index in the range of low frequencies with A_r=0, the peaks in the range of 2500÷5000 rpm with A_r=0.25÷1 and finally the peak above 5000 rpm with A_r=2.

This configuration requires some space, but it is easier to fit a large volume resonator sideways than is series with the intake line.

The last figure (16) shows the effect of the side-branch resonator on the filling index of the short-long system with V=10. The volume and duct length of the side branch resonator are held constant (V_r=10 and L_r=1) and the duct cross section is adjusted in the range of A_r =0÷2. In this case, the system without a side-branch resonator still has a good filling index over a wide range of piston frequencies and the resonator improves the filling index especially in the range of 2000÷6000 rpm. The best performance of the system can be achieved by keeping A_r=1÷2 over the full range of piston frequencies.

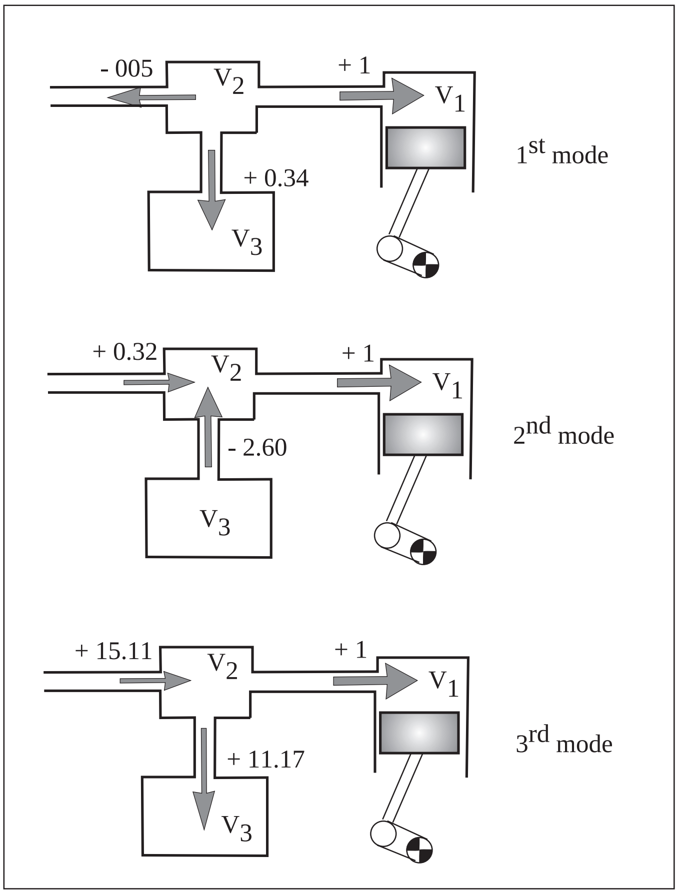

Fig.15 Acoustical modes

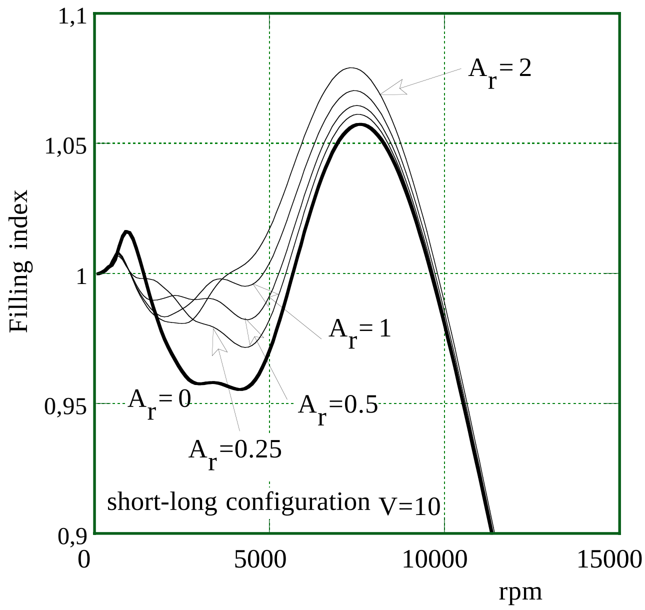

Fig.16 Filling index of the short-long configuration with the side-branch resonator

CONCLUSIONS

The effect of tunable resonators on the volumetric efficiency of single-cylinder engines was studied. The primary aim was to investigate the possibility of varying the optimum filling frequency by adjusting the volume or the duct of the resonator.

The analyses were carried out by means of a simple lumped element model, an improved lumped element model and by means of a numerical code for fluid-dynamics analysis. The indications given by the simple lumped element model were confirmed by the numerical code in the range of frequencies where sound wavelength was much grater than the intake system dimensions. The predictions of the improved lumped element model were in good agreement with the numerical results over a wider range of frequencies.

It is worth pointing out that it takes only a few minutes to calculate a curve of the filling index against frequency using the lumped element methods whereas it takes about two hours to calculate just a point using the numerical code.

The results showed that the intake system equipped with an in-series resonator could be regulated when the duct between the cylinder and the resonator is short; i.e. the Helmholtz frequency of the cylinder and the first duct alone occurs in the field of high frequencies. In these cases, an on-off regulation with a large increase in the resonator volume in the field of the high frequencies optimizes the filling index in the range of 0÷15000 rpm.

The intake systems equipped with a side-branch resonator offer several possibilities of adjustments. Good results were obtained by varying the cross section of the side-branch resonator duct. In this case, a continuous increase in the opening of the duct makes it possible to move the peak of the filling index from low frequencies (≈2500 rpm) to high frequencies (≈6000 rpm).

Future developments of this research will be the development of experimental tests on an engine equipped with a variable volume resonator, the improvement of the lumped element model in order to take into account the non linear effects which are caused by the motion of the piston and the valves.

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