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SILENCING OF EXHAUST ENGINE NOISE

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INTRODUCTION

Since several countries have decided more severe regulations about noise produced by internal combustion engines, new techniques have been carried out to detect and reduce the strength of different noise sources: exhaust systems, cooling equipments, mechanical vibrations, etc.. Particularly, in this paper it is described a theoretical and experimental approach to the investigation of the gas-dynamic noise characteristics of a spark ignition engine exhaust system. The radiated noise was experimentally analysed by means of an adequate testing equipment, that allowed contemporary power dynamometer tests and noise level measurements in a special test room simulating the open field condition. In order to considerably reduce the experimental work we found very useful to support practical tests with a theoretical model of the analyzed system. Particularly, a computer program has been developed which predicts the velocity-time history at the open end of the exhaust system, using the concept of "characteristics".
The radiated exhaust noise was then calculated by assuming monopole radiation from the tailpipe outlet. The pressure-time histories and noise levels calculations have given some useful criteria for the exhaust system design.

THEORETICAL INVESTIGATION

The flow in the exhaust pipe was considered to be unsteady, viscous and with heat transfer at the wall. The change of the pipe cross-sectional area was taken into account, but the flow was analysed in a one-dimensional manner to save computer time. It means that all quantities were considered constant on a cross-section and functions of the only position coordinate x (along the pipe axis) and time t. Assuming as unknown quantities: velocity of medium $u=u(x,t)$, pressure $p=p(x,t)$ and density $\rho=\rho(x,t)$, the equations of continuity, momentum and energy written

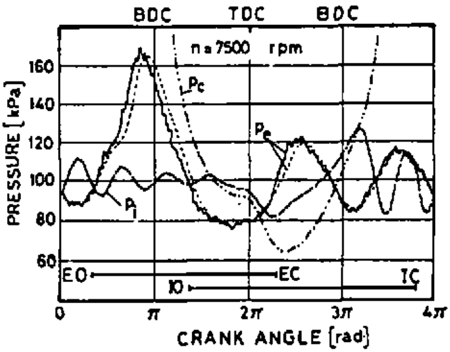

Fig.1. Pressure values measured (bold line) and computed (dotted line) in the exhaust (p_e) and inlet (p_i) pipes on the sections near the valves. (p_c = pressure in the cylinder; EO = Exhaust Open; EC = Exhaust Closed; IO = Inlet Open; IC = Inlet Closed).

along the "characteristic lines" gave the following compatibility conditions:

$$dp - \frac{2k}{k-1}\frac{p}{a}\,da + \rho(q+uF)dt = 0$$

$$dp + \rho\, a\, du - \rho(k-1)(q+uF)dt + \rho\, a\, F\, dt + a^2\rho\,\frac{u}{S}\frac{dS}{dx}\,dt = 0$$

$$dp + \rho\, a\, du - \rho(k-1)(q+uF)dt - \rho\, a\, F\, dt + a^2\rho\,\frac{u}{S}\frac{dS}{dx}\,dt = 0 \qquad (1)$$

where: S = Area of cross-section; F = friction force/unit mass.
$\dot{q}$ = rate of heat input/unit mass.

Numerical solutions of system (1) gave pressure-time and velocity-time histories in any cross-section and particularly at the open end of the exhaust system. We considered then this last section as a "source" of gas of strength $Q = S\cdot u_o$. The problem of the noise emission from the exhaust pipe became thus equivalent to that of the emission by a pulsating body [1], with its typical solution for sound pressure level at a distance: $r \gg \lambda$ (wavelength):

$$p(t_o) = \frac{\rho_o S}{4\pi r}\, u_o\left(t - \frac{r}{a_o}\right) \equiv \dot{u}_o\left(t - \frac{r}{a_o}\right) \qquad (2)$$

where: u_o = medium velocity at the open end of the pipe;
ρ_o, a_o = density and sound velocity in the space surrounding the pipe.

According to equation (2), we considered the derivative respect the time of the velocity at the open end of the pipe ($\dot{u}_o$) and expanded it into its frequency components. The mean square value of each component

could give a reasonable prediction of sound pressure levels in the frequency domain produced by simple engine exhaust systems. The accuracy of the prediction obviously depends on the accuracy of the velocity-time calculations and the simplifications assumed by the theory, namely:
1) the gas velocity was supposed to be constant across the section of the open end of the pipe (one-dimensional flow);
2) it was assumed that no sound waves were reflected from the ground.

DISCUSSION OF RESULTS

The ability of the above theoretical model to predict the engine performance and exhaust noise levels was tested on a monocylinder 4 stroke engine supplied with different simple shapes of the exhaust systems. In fact in a special test room, isolated from the outside noise and with deadening panels on the walls to reach the semianechoic room condition, the exhaust systems influence was tested at the same time both on engine performances and on noise radiated. Particularly, the engine was closed into a special box to remove the effect of other noise sources (namely: mechanical vibrations, cooling equipments, etc..). The actual sound pressure level emitted by the exhaust system was recorded at 1 m from the end of the pipe by means of a microphone. Meanwhile some phisical quantities (temperature along the pipe, pressures in the cylinder and in some sections of the exhaust system, etc..) were measured in order to collect input data for the model and to control the accuracy of computations. Fig. 1 shows a tipycal comparison between measured (bold line) and computed (dotted line) pressure values in the exhaust and inlet pipes. The accuracy of the prediction was quite good both for amplitudes and principal frequencies of pressure waves (that is: for the influence of the tested system on engine performances). The high frequency signal found in the measured value of exhaust pressure (and not predicted by the model) was at least partly considered as

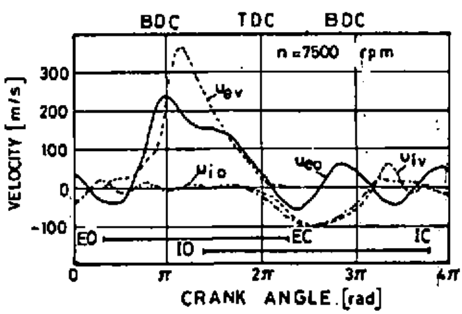

Fig.2. Velocity values computed in the exhaust (u_e) and inlet (u_i) pipe. (index: o = open end of the pipe; v = section of the pipe near the valve).

a noise due to the transducer. In fact we thought that some mechanical vibrations were transmitted by the tube to the quartz transducer, because after inserting an elastic connection between them, the signal collected was cleaner. We cannot exclude however, that a part of these high frequency components have a gas dynamic origin which are not considered by the model.

In Fig. 2 appear the computed velocity values from which, according to equation (2), were calculated the pressure sound levels at a distance $r = 1$ m from the open end of the exhaust system, and compared in Fig. 3 with measured ones in the frequency domain. The agreement is satisfactory only for frequencies lower that 1 KHz, while the measured high frequency noise is not predicted by the model. This is probably due to the model approximations: the flow is in fact pluridimensional, there are some local turbulences, cross oscillations, etc...

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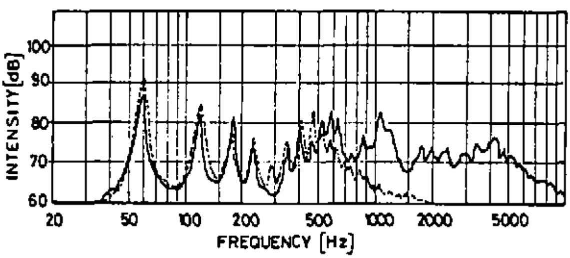

Fig.3. Frequency domain of computed (dotted line) and measured (bold line) exhaust noise in a semianechoic room.