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PROPAGATION OF A PULSIVE FLOW WITH FINITE AMPLITUDE IN A STRAIGHT PIPE

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INTRODUCTION

Since D.D.Davis et. al. analyzed the characteristics of the muffler by means of TL in 1954[1], many papers were presented about TL of mufflers. But about the muffler for the exhaust system of internal combustion engines, the tail-pipe, the pipe connecting the element (expansion chamber, resonator and so on) to the engine and the pipes between each element can not be ignored.
One of the authers analyzed the exhaust muffler by means of F parameter (transfer matrix or 4 terminal network constants)[2]. The results are obtained analytically and easily, and we can understand the effect of each element and estimate the total noise reduction of muffler system. But the analysis on the base of the acoustic theory is done under various assumptions, so the estimation of noise reduction is not good agreement with the experimental results on the practical exhaust mufflers. On the other hand, the estimation by the method of characteristics is fairly good agreement with experiment[3][4]. But it does not have the advantage which the method of acoustic theory has.
It is useful that we understand the influence of each assumption in the method of acoustic theory. For the first step, we compute the propagation of a pulsive flow with finite amplitude in a straight pipe by means of the method of characteristics and compare it with the estimation by the method of acoustic theory.

METHOD OF ACOUSTIC THEORY

In application of the acoustic theory, assumptions are as follows
1. Infinite small amplitude sound. 2. One dimensional sound wave.
3. Homogeneous media (temperature and static pressure are constant).
4. Isentropic (no loss and no heat conduction).
5 Mean flow is zero. 6. Constant velocity source.
When the length of each pipe contains the equivalent length due to end

reactance, the expression by means of F parameter is as follow

$$\begin{vmatrix} p_1 \\ u_1 \end{vmatrix} = [F]\begin{vmatrix} p_2 \\ u_2 \end{vmatrix}, \quad [F] = \begin{vmatrix} A & B \\ C & D \end{vmatrix} \quad (1) \qquad p_1 = p_0 - u_1 z_0, \quad p_2 = u_2 R_r \quad (2)$$

where

p_0, z_0 = sound pressure and internal impedance of source.
p_1 u_1 = input sound pressure and volume velocity.
p_2 u_2 = output sound pressure and volume velocity.
R_r = radiation resistance of tail-pipe.
As R_r is very small as compare with the characteristic impedance of the tail-pipe, u_2 and the radiating sound power W_r are about

$$u_2 = \frac{u_1}{D} \quad (3) \qquad W_r = |u_2|^2 R \quad (4)$$

D is the short-circuit transfer constant and it is measured easily as it is equal to the open-circuit (the pipe-end is closed) transfer constant A (ratio of sound pressures) of inverse direction. So the insertion loss becomes

$$IL = 10\log W_r' - 10\log W_r = 20\log |D| - 20\log |D'| \quad \text{dB} \quad (5)$$

W_r' and D' mean the values before attaching the muffler.
For a straight pipe of length ℓ

$$D = \cos k\ell, \quad k = \text{wave number.} \quad (6)$$

When there is a temperature gradient in the pipe, the average wave number k of both end of the pipe may be used.

METHOD OF CHARACTERISTICS

In application of the method of characteristics, the condition of the flow in the pipe is expressed by the 3 variables λ, β, A_0[3][4],

$$\lambda = A_0 P + \frac{\kappa-1}{2} M \quad (7) \qquad \beta = A_0 P - \frac{\kappa-1}{2} M \quad (8)$$

where

$$A_0 = c_0'/c_0, \quad P = (p/p_0)^{(\kappa-1)/2\kappa}, \quad M = u/c_0 \quad (9)$$

c_0 = sound velocity in atmosphere.
c_0' = sound velocity in adiabatic expansion from p to p_0.
p = absolute pressure. p_0 = atmospheric pressure.
κ = specific heat ratio. u = flow velocity.
We assume that there is a constant velocity source at the inlet of the pipe. Fig.1 shows the flow model of the exhaust of 1 cylinder 4 stroke engine. The period of one cycle is 40 ms (3000 rpm), so the fundamental frequency is 25 Hz. Fig.2 shows the flow model of the exhaust of 4 cylinders engine and the fundamental is 100 Hz.
The boundary conditions at the outlet of the equivalent length are as follows,
for outflow, the pressure is eqaul to the atmospheric pressure, so

$$\beta_0 = 2A_0 - \lambda_0 \quad (10)$$

for inflow, from the energy equation of steady flow

$$\beta_0 = \frac{1}{\kappa+1}\{(3-\kappa)\lambda_0 + 2\sqrt{(\kappa-1)(2\lambda_0^2+\kappa+1)}\} \tag{11}$$

and assuming that it is the adiabatic flow

$$A_{00} = 1 \tag{12}$$

RESULTS AND CONCLUSION

Some results computed are shown in Fig.3 -Fig.8. The estimation by the acoustic theory is generally lower in the amplitude of the harmonics than one by the method of characteristics. When there is no temperature gradient, the higher the amplitude of input flow is and the longer the length of the pipe is, the larger the error is in general. But in the case when the magnitude of $|\cos k\ell|$ of any harmonics is extremely small, as shown in the case of ℓ=2.00 m (Fig.3), the error is large in spite of low amplitude and short pipe. For 4 cylinders model, the mean flow is large relatively and the wave number seems to become small (Fig.7). As shown in Fig.8, the influence of temperature gradient is very large.

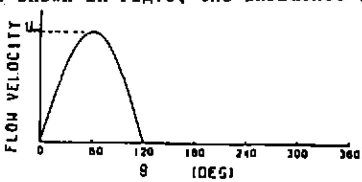

Fig.1 Flow model for 1 cylinder engine.

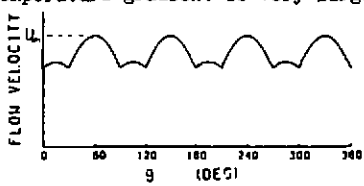

Fig.2 Flow model for 4 cylinders engine.

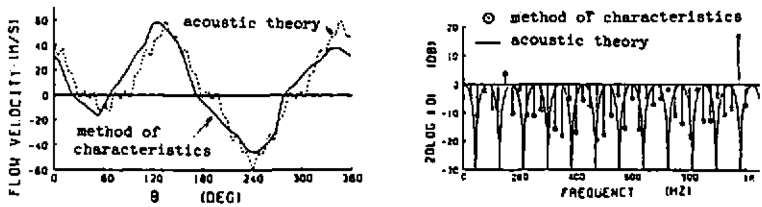

Fig.3 1 cylinder model, U_m=34 m/s, ℓ=2.00 m, 20°C - 20°C.

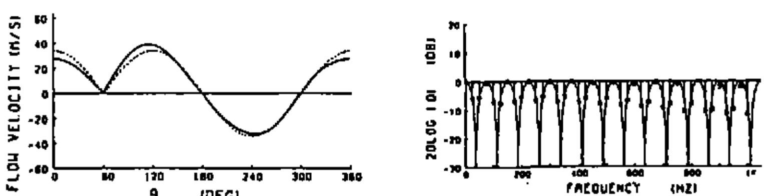

Fig.4 1 cylinder model, U_m=34 m/s, ℓ=2.28 m, 20°C - 20°C.

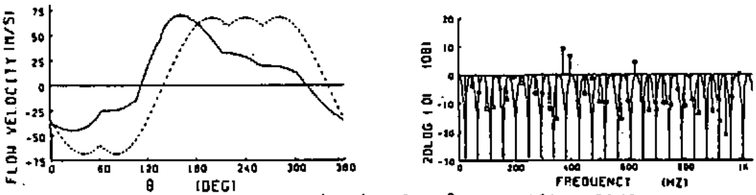

Fig.5 1 cylinder model, U_∞=34 m/s, ℓ=3.80 m, 20°C - 20°C.

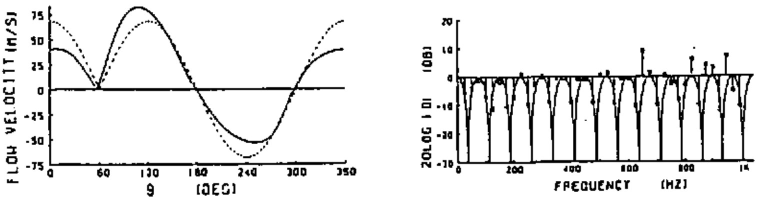

Fig.6 1 cylinder model, U_∞=68 m/s, ℓ=2.28 m, 20°C - 20°C.

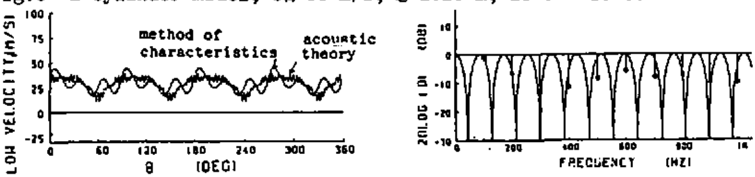

Fig.7 4 cylinders model, U_∞=34 m/s, ℓ=2.00 m, 20°C -20°C.

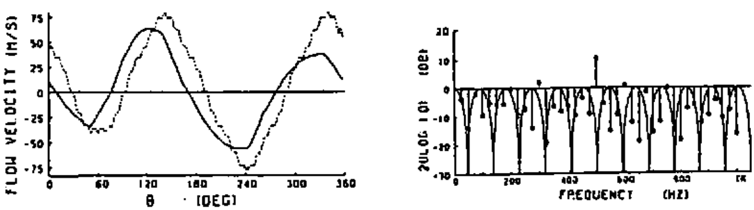

Fig.8 1 cylinder model, U_∞=34 m/s, ℓ=2.28 m, 300°C -20°C.

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