EFFECT OF EXHAUST SYSTEMS ON DIESEL ENGINE PERFORMANCE

by

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تأثير مجموعات العادم على كفاءة محركات الديزل

بقلم
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خلاصة المقالة :

ان الغرض من وضع كاتم الصوت لمحركات الاحتراق الداخلي هو تخفيض الضوضاء الصادرة منه الى حدود يمكن للاذن البشرية تحملها بدون تعريضها للاصابة باضرار او لفقد حاسة السمع اذا ما تعرضت لهذه الضوضاء مددا طويلة. ولكن وجود كاتم الصوت بمواسير العادم اللازمة له يتسبب في رفع ضغط العادم نتيجة للموانع التي تعوق سريان غازات العادم الى الخارج والتي من شأنها ان تخفض الضوضاء.

وهذا البحث عبارة عن موازنة بين الفائدة الناتجة من تخفيض الضوضاء وبين القدرة المفقودة من المحرك. وهو يغطي الجانب النظري والعملي لكواتم الصوت وقد قسمت الى انواع حسب طريقة خفض الصوت المتبعة في كل نوع مع ذكر المعادلات اللازمة لتصميم كل منها – وهذه المعادلات مشتقة من نظرية (المرشحات الصوتية) مع تحليل لاسباب زيادة الضغط على مجموعة العادم.

وقد تم قياس كل من خواص الاداء لكل من هذه الانواء سواء من الناحية الصوتية او الميكانيكية باستعمال اجهزة قياس شدة الصوت مع اجهزة القدرة والاداء بالنسبة لمحرك ديزل (بركنز) ٤ سلندرات قدرة ٤٥ حصان عند سرعة ٢٠٠٠ لفة/دقيقة.

وهذا البحث يسهل على المصمم اختيار النوع الملائم من كواتم الصوت للمحرك واختيار ابعاده وتركيبه بدون فقد طاقة كبيرة نتيجة لخفض الضوضاء.

Abstract: The function of the exhaust silencers is to reduce the noise level to an acceptable value to avoid the risk of ear damage when exposed to such noises for long periods. The drawback of the presence of asilencer is the high back pressure created due to restricting the gas flow or according to the pressure drop in the attenuating elements within the silencer. This investigation covers the theoretical formulae that guide the designer of a silencer, and which are derived from the theory of sound filtration. The silencers are classified according to the principle of muffling. The pressure drop along the exhaust system is presented also in the theory and experiment. The experimental results are obtained for both the acoustical and mechanical performance of each type of silencers with a comparison between them for the suitable choice of the silencer type. Obviously, the problem must be a compromise between the high noise levels and the high values of power loss due to engine quitening.

NOMENCLATURE

C	Velocity of sound
C_o	Acoustic conductivity.
d	Diameter of exhaust pipe.
d_c	Diameter of connecting orifices.
D	Diameter of muffler.
F_m	Fanning friction factor for turbulent flow in pipes.
f	Frequency of sound wave.
f_r	Resonant frequency.
k	Wave number $= \frac{2\pi}{\lambda}$
L_e	Effective length of muffler.
L_t	Tailpipe length.
L_c	Length of connecting orifices.
m	Expansion ratio.
n	Number of holes.
S	Area of exhaust pipe.
S_c	Area of the connecting orifices.
V	Reonator volume.
v	instantaneous particle velocity.
W_i	Power input.
W_o	Power output.
ρ	Average gas density.
λ	Wave length.

1. INTRODUCTION:

Extensive researches and studies were done to determine the effects of the variations of the back pressure on the performance of internal combustion engines. These studies payed no but little attention to the factors causing the rise in the back pressure of the engine and to eliminate them. It is the object of this investigation to determine the effect of the exhaust systems on the back pressure and engine power, with comments on the silencing properties of different mufflers.

2. CAUSES OF BACK PRESSURE:

Back pressure is created in the exhaust pipe due to many factors. Among of these factors rises the effect of the installation of unsuitably designed silencer. The diameter, length, and the presence of large bends in the exhaust pipes have also pronounced effects.

The pressure drop due to silencer existence is different from one type of silencer to another, according to the method of silencing used. Obviously, a straight-through type will cause the minimum back pressure to be created while the silencer which depends, in attenuating the exhaust noise, on the reversing of gas directions, with many restrictions to the flow, will result in attaining high values of back pressure. The single expansion-chamber silencer causes a back pressure somewhat higher than that caused by the straight-through type.

The pressure drop through the expansion chamber silencer, with a finite tailpipe length, is made of three principal compartemnts as shown in figure 1, from which:

1-A pressure drop (ΔP_c) caused by the sudden change in diameter at inlet and outlet of the expansion chamber.

2-A pressure drom (ΔP_1) caused by the turbulent flow in the tail pipe.

3-A pressure drop (ΔP_x) caused by the expansion of gas flowing at the end of the tailpipe to the atmosphere.

ΔP_c depends on the chamber dimensions and the relative size of the inlet and the exhaust pipes. This can be minimised by avoiding sharp square corners, and by making the diameter of the tailpipe somewhat larger than the inlet duct diameter to allow for the expansion of the gas flowing.

The pressure drop due to turbulent flow in the tailpipe (ΔP_1) may be expressed, as shown in (1), by:

$$\Delta P_1 = 4\frac{F_m L_t}{d} \times \tfrac{1}{2}\rho v^2$$

where:

L_t = tailpipe length.

d = tailpipe diameter.

ρ = average gas density.

F_m = friction factor for tailpipe.

v = instantaneous linear velocity in the tailpipe.

F_m is approximately 8×10^{-3}, for turbulent flow in commercial steel pipes or other pipes of comparable roughness as shown by Knudsen and Katz (4).

The pressure drop due to end expansion is depeding on $\frac{1}{2}\rho v^2$. It must be noted that the instantaneous values of ΔP_x and ΔP_1 dempend on the instantaneous particle velocity. This means that the presense of any alternating particle velocity will tend to increase the values of ΔP_x and ΔP_1.

So, if the silencer is designed to attenuate considerably the sound wave at the firing frequency and its harmonics, it will cause a reduction in the instantaneous particle velocity and cosequently a reduction in ΔP_1 and ΔP_x hence an improve in the mechanical performance of the engine.

3. NECESSITY OF SILENCING:

Exhaust silencers are installed in the exhaust pipe to impede the transmitted sound waves. The sound output must be at levels lower than which are safe for the ear if a person is exposed to noise for a long period. LImiting sound curves are shown by the Commitee in reference (6), in which the maximum sound pressure levels in decibles versus frequency in cycles per second are drawn. These curves give the maximum sound levels that are acceptable for 8-hours of daily exposure.

The engine exhaust noise exceeds these limits considerably at certain frequencies which are the firing frequency and its harmonics. These levels must be reduced to lower, accepted values, which can be done by a suitable silencer.

4. MUFFLERS:

Practicaly, an infinite number of silencer designs can be obtained, but they can be classified princ ipaly as:

a) Expansion chamber type, Fig. (1).

(b) Resonator types, Fig. (2).

c) A combination of (a) and (b).

The single expansion chamber with finite tailpipe of length L_t can be constructed of a circular or elliptical cross-section. The shape is immaterial, but the expansion ratio is the main factor affecting both the pressure drop and the attenuating characteristics. This can be observed from the cuves presented in figure (3), which represents the attenuation of a single expansion chamber type with a tailpipe of length L_t, reference (7).

The equation of the expansion chamber without a tailpipe is given in references (1,2,5) as:

$$\text{Attenuation} = 10\,\text{Log}_{10}\left[1 + \frac{1}{4}\left(m - \frac{1}{m}\right)^2 \sin^2 kL_c\right] \quad \text{db}$$

where:

$$m = \frac{\text{muffler cross} - \text{section area}}{\text{duct cross-section area}}$$

k = wave number.

L_c = effective length of muffler.

It is clear that the maximum attenuation depends only on the expansion ratio, while the length determines the frequencies at which this maximum and minimum occur. The theoretical attenuation in decibles against the frequency for the resonant type, follows the equation:

$$\text{Attenuation} = 10\,\text{Log}_{10}\left[1 + \left|\frac{\frac{\sqrt{C_0 V}}{25}}{\frac{f}{f_r} - \frac{f_r}{f}}\right|^2\right] \text{db}$$

The resonant frequency f_r is given by $\frac{C}{2\pi} \cdot \sqrt{\frac{C_0}{V}}$

where:

C_0 = acoustic conductivity of the holes

$$= \frac{n S_c}{L_c + 0.8\sqrt{S_c}} \quad \text{Reference (3).}$$

The parameter $\frac{\sqrt{C_0 V}}{2S}$

is called the attanuation parameter because it affects the attenuation value, while

$$\sqrt{\frac{C_0}{V}}$$

is called the resonance parameter. The representation of the above formula is shown in figure 4. The increase of the attenuation parameter causes a wider range of attenuation with higher values to be attenuated.

5. EQUIPMENTS AND TESTS:

The engine used in this investigation is a 4-stroke siesel engine, delivering 45 horse power at full load speed of 2000 RPM. A Froud dynamometer is used to indicate the speeds and loads at which the engine was running. A U-tube manometer, filled with mercury, is used to determine the back pressure in the exhaust pipe. A sound level meter and frequency analyser are used to determine the noise levels at the various running conditions.

The performance curves of the engine are presented without silencer and then with the different types of silencers installed to show a comparison between them. The total noise levels in decibles are presented also for each condition. An exhaust pipe with a butterfiy valve is installed in the exhaust system to enable the gtadual rise in the back pressure to be obtained by gradually closing the valve by a small handle. The set angle is read on a circular graduated disc (Protractor).

6. RESULTS AND DISCUSSION:

The power loss due to the insertion of silencers (A) and (B) are shown in figure 5. It can be seen that the loss due to installation of the straight-through resonator is less than that caused by the installation of the expansion chamber type. The effect of pressure rise in the exhaust pipe. on the maximum power delivered by the engine. is shown in figure 6.

Experimentally. it is found that. for each one $1b/in^2$ rise in back pressure. the power of the engine will be decreased by 1.5 %. This is due to the mixing of the fresh charge by the exhaust gases; and also due to the increase of work to be done by the piston during the exhaust stroke.

The actual measurements of the total noise level for the unmuffled engine and also for the engine caryying either of the silencer A or B are shown in figure 7. From this figure. it can be noted that the sound level increases with the engine speed.

The silencer shown in figure 8 is a special type which depends on resisting the flow of gases. This type reduces the noise considerably but the resistance to the gas flow increases and the pressure rise increases also by the same action. But. the increase in back pressure is too high a price to pay for engine silencing. This gype can be theoretically analysed by the electric analogy as shown in figure 9. This shows that the orifices are analogous to the electric resistances R_1 and R_2 while the surrounding volume is analgous to the electric capacitance C. The atenuation is given by

$$10 \ \mathrm{Log}_{10} \frac{W_i}{W_o}$$

where W_i, W_o are the input and output powers respectively.

7. CONCLUSION:

The single expansion chamber of a sufficient colume is of a great value in silencing of small engines and where the space is not a premium. While resonators are recommended for noises having a wide range of frequencies with high sound levels.

The round section straight through resonator is the most economical type with the least power loss. But as the engines have become larger and the spaces arranged for them are also limited. the reverse flow elliptical cross section silencers proved to be a necessity regardless of its increased cost. So far. it has been possible to get more attenuation per cubic inch from a reverse flow than from a straight through muffler.

References:

1- Beranek, L.L. "Noise Redudtion", Chapter 16, McGraw-Hill Co. Inc. ,1960.

2- Davis, D.D. ,Stokes, M.G., Moore, D. and George, L., N A C A Report 1192, 1954.

3- Harris, G.M., "Handbook of Noise Control", Chapter 21, McGraw-Hill Co. INc. 1957.

4- Knudsen, I.G. and Katz, D.T., "Fluid Dynamics and Heat Transfer", University of Michigan Press. An Arbor, 1954.

5- Muller, L.E., Exhaust Systems, fundamentals and gesign considerations. Buick Motors Division. G.M.C., S.A.E. Trans., 1954.

6- Rosenblith, W.A., The Relation of Hearing to Noise Exposure. A report by Sub-Commitee Z 24-X-2, A.S.A. Inc., 1954.

7- Elsharkawy, A.I., Studies on the exhaust silencers. M.Sc. Thesis, Alexandria University, 1969.

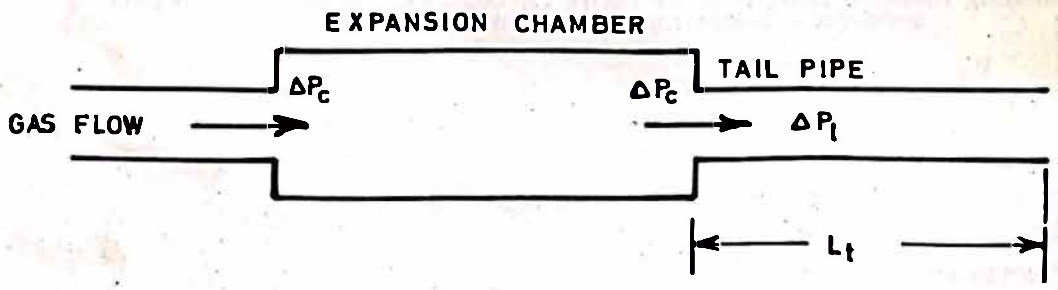

FIG. 1. EXPANSION CHAMBER MUFFLER SILENCER 'A'

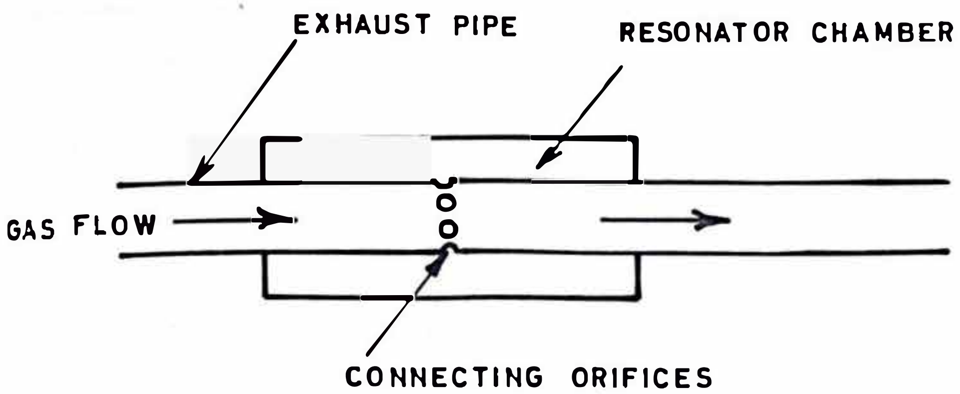

FIG. 2. SINGLE CHAMBER RESONATOR SILENCER 'B'

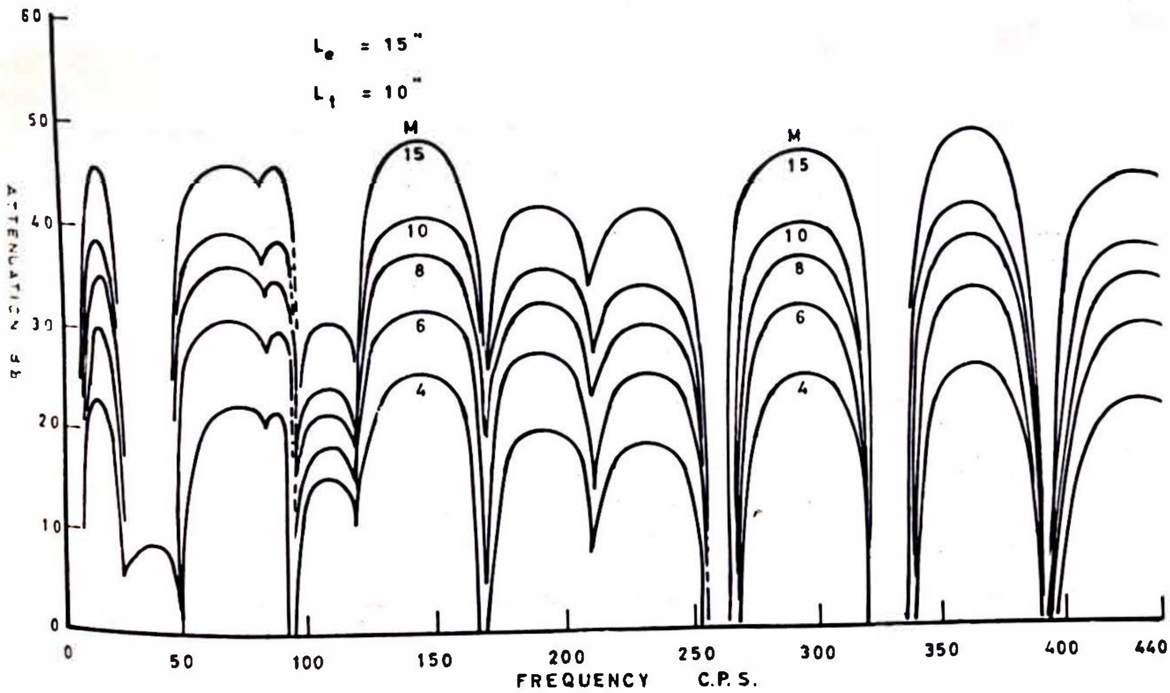

FIG. 3. ATTENUATION OF AN EXPANSION CHAMBER MUFFLER.

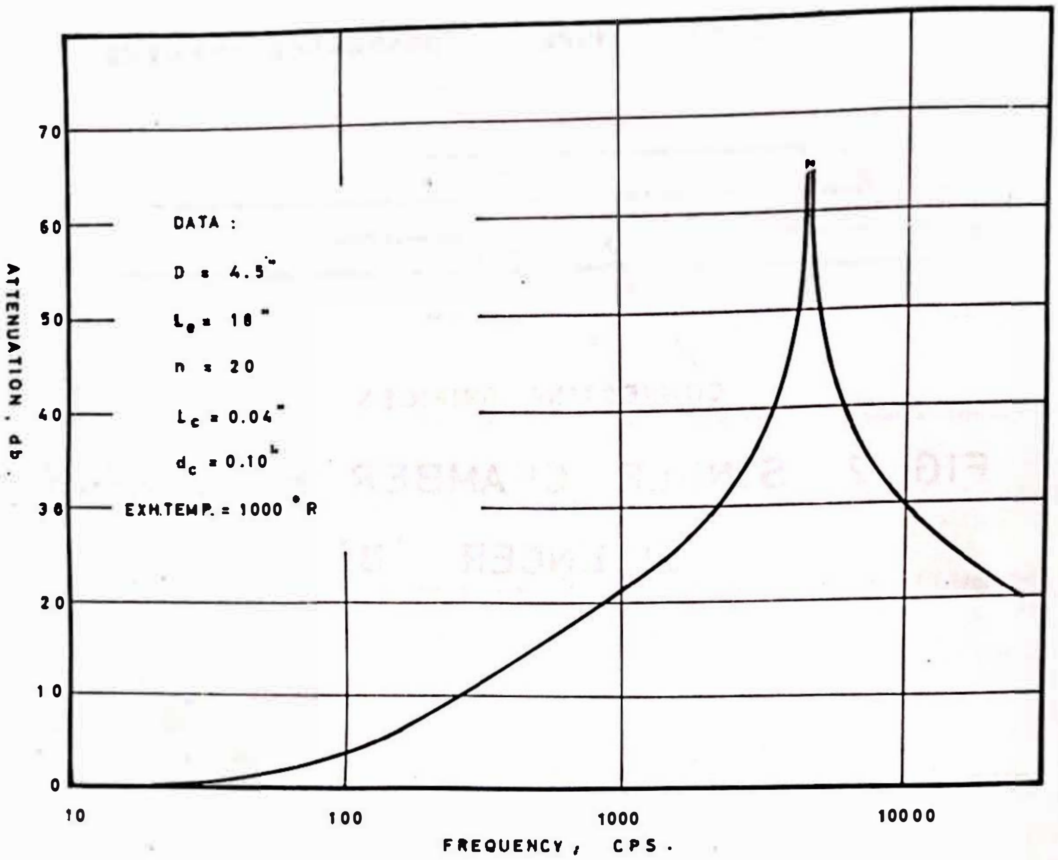

FIG. 4. ATTENUATION OF THE RESONATOR SILENCER.

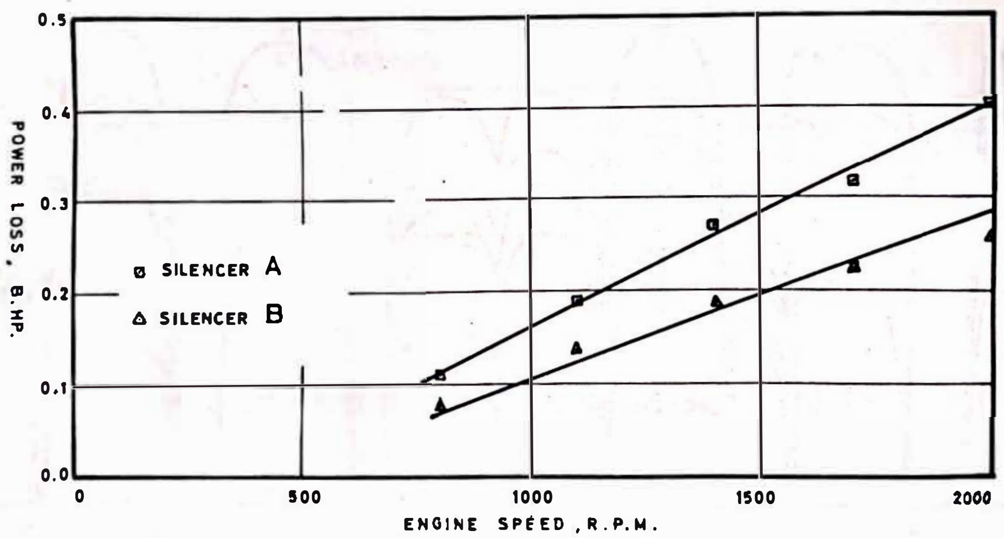

FIG. 5. POWER LOSS DUE TO INSTALLATION OF SILENCERS 'A' and 'B'.

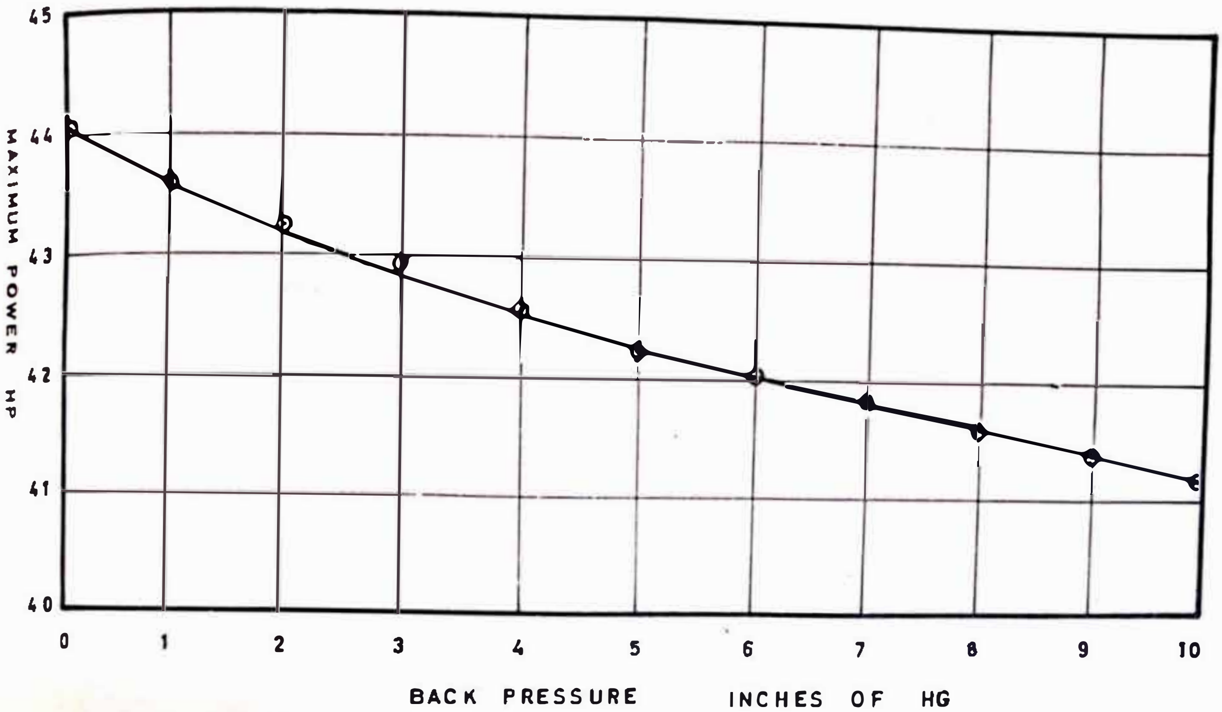

FIG. 6. EFFECT OF BACK PRESSURE ON POWER

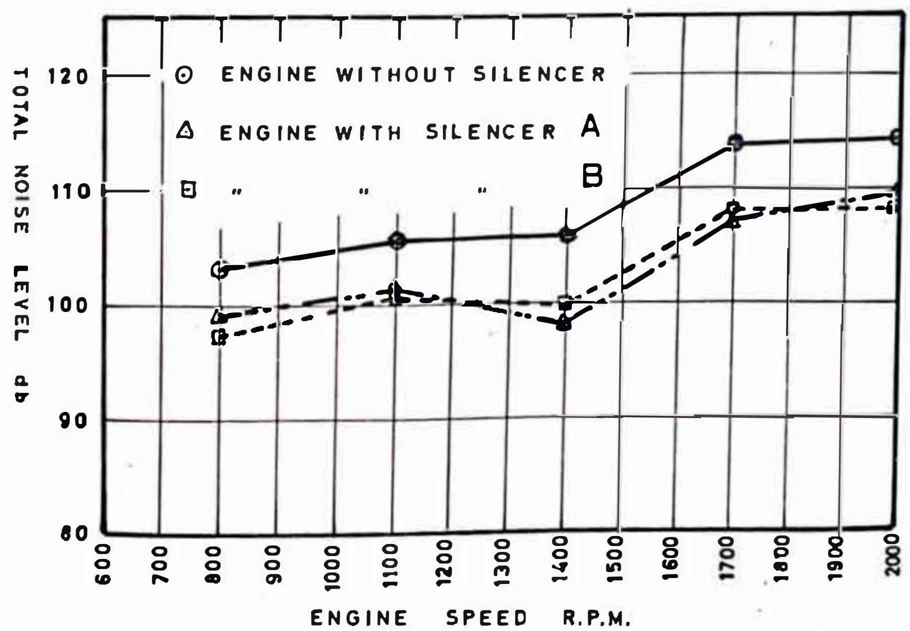

FIG. 7. EFFECT OF SILENCERS 'A' and 'B' ON THE TOTAL NOISE LEVEL