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EXHAUST SILENCERS WITH MINIMUM POWER LOSS

F. B. SHENODA

Acoustic Laboratory, National Institute for Standards (Dokki, Cairo, Egypt)

Based on the principle of reflection free duct termination, silencers were designed. These silencers were applied to reduce the exhaust noise of internal combustion engines.

Acoustical and mechanical measurements have been taken as a function of engine speed for both full and partial load. For comparison, the same measurements have also been carried out on a conventional dissipative silencer of equal length, and also, for purposes of reference, on the exhaust pipe without a silencer.

A high level of acoustic absorption could be realized over a wide frequency range, without changing both the electric power and the fuel consumption of the applied engine, with respect to those for an untreated pipe.

Introduction

It is well known that internal combustion engines have uncomfortable noise. With increasing stringent noise legislation, manufacturs of internal combustion engines are faced with considerable difficulties in meeting these enacted requirements. Extensive investigations [1]–[3] have shown, that exhaust noise is one of the predominant noise sources of internal combustion engines.

In reducting exhaust noise, it seems, that the design of a suitable silencer is prefered to an attempt of lowering the noise generated at the source. The insertion of a silencer in the exhaust duct can attenuate the pressure pulsation, before they reach the surrounding atmosphere.

According to the mechanism of attenuation, silencers are divided into reactive silencers, dissipative silencers and acoustic resonators. A reactive silencer [4] provides an impedance mismatch for the acoustic energy travelling along the exhaust duct. This silencer type causes high backpressure and high pressure loss for the gas flow, which passes through it. As a result, a reactive silencer has a negative effect on the mechanical performance of the engine, on which it is mounted. The acoustical performance of a dissipative silencer is determined mainly by the presence of a sound absorbing material, which does an efficient job of filtering high frequencies. An acousting resonator is incorporated to provide a relatively narrow band of attenuation [5]. Usually a low frequency resonator has an irregular shape and large volume, which is not always available.

Different types of silencers mentioned above may próduce good noise results for a given system or application and produce poor results for another. Although many investigations [6]–[8] of exhaust silencing have been made, little was done to account for the behaviour of exhaust silencing with minimum power losses.

Exhaust silencers presented in this paper have been designed to be applied on internal combustion engines. The basic concept of these silencers is based on an earlier design made by the author [9]. This design is a reflection free duct termination, which gives a good matching impedance for an acoustic wave travelling along a pipe without change in its cross-section.

The designed silencers were mounted on a turbo – charged diesel engine. Acoustical and mechanical measurements have been assumed as a function of engine speed for both full and partial load. For comparison, the same measurements have been carried out on a conventional dissipative silencer of equal length, and also; for purposes of reference, on the exhaust pipe without a silencer.

Description of silencers

The basic concept of the construction uses a reflection free duct termination [10]–[13] by means of a gradually changing flow resistance along the walls of the exhaust pipe. This is realized by a suitable perforation of the pipe walls in combination with a sound absorbing material. The treatment covers 2 meters of pipe length.

Fig. 1 shows the construction principle of silencers designed in the present work, which are briefly described as follows:

Silencer S1

It consits of a straight pipe with a length ($l = 2$ m) and diameter ($d = 7$ cm) equal to the diameter of the common exhaust pipe of the applied engine. This pipe has a single slit at the pipe walls showing an exponential change in width $b(x)$ according to the relation

$$b(x) = b_0 e^{\delta x} \quad [\text{mm}], \tag{1}$$

where $b_0 = 1$ mm is the slit width at $x = 0$ and δ was chosen to be equal $= 0.00207$ to give a suitable width $b(x) = 63$ mm at the end of the treated pipe ($x = l = 2000$ mm). From the acoustics point of view, this variable slit can be replaced practically by a variable perforated area (more than $35^0/_0$ of the slit area is perforated), which follows the above relation (1). An absorptive material is packed (in a half cylinder — shaped closed chamber of a 15 cm radius) with a constant density, so that it gives a flow resistance of about ϱc (the air characteristic impendance).

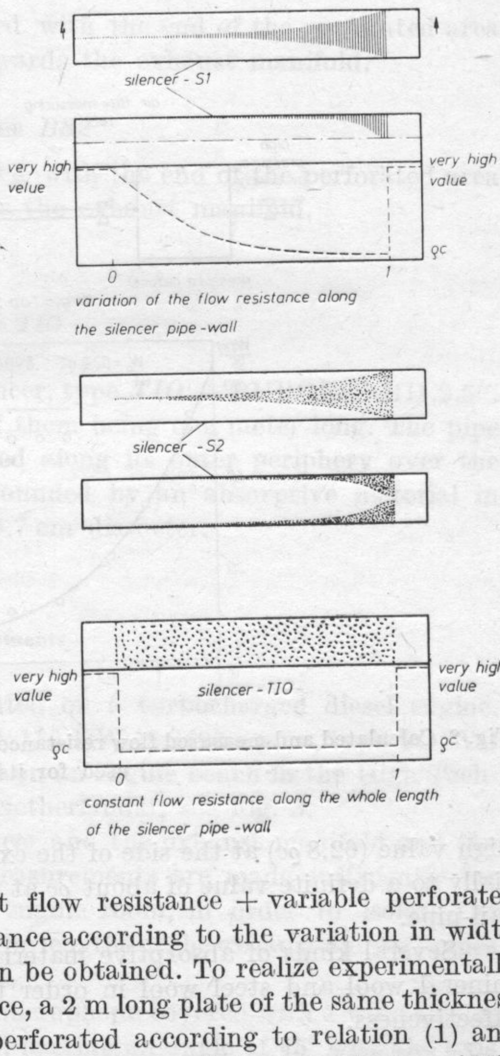

Fig. 1. Principal of construction of the measured silencers

Using this method (i.e.; constant flow resistance + variable perforated area), a gradually changing flow resistance according to the variation in width of the perforated area (relation (1)), can be obtained. To realize experimentally such a gradually changing flow resistance, a 2 m long plate of the same thickness as that of the pipe wall is variably perforated according to relation (1) and divided into ten equal parts (each 20 cm long) with the holes lying in the middle of an area equal to the cross section of a suitable holder ($20 \times 23.5\ cm^2$), made for a flow resistance measuring apparatus. Each perforated part acts as a face to the absorptive material which is packed at a depth of 15 cm, equal to that of the closed chamber of the silencer. The density of the absorptive material is chosen, so that it gives approximately the mean value of the flow resistance, which lies at the same length of a calculated curved. These measurements showed that the densities of the absorptive material used with the different perforated parts are slightly different from each other. Consequently, a mean density was chosen to be used in the designed silencers. Fig. 2 shows the x-dependance of the calculated and measured flow resistance. The flow resistance has a very

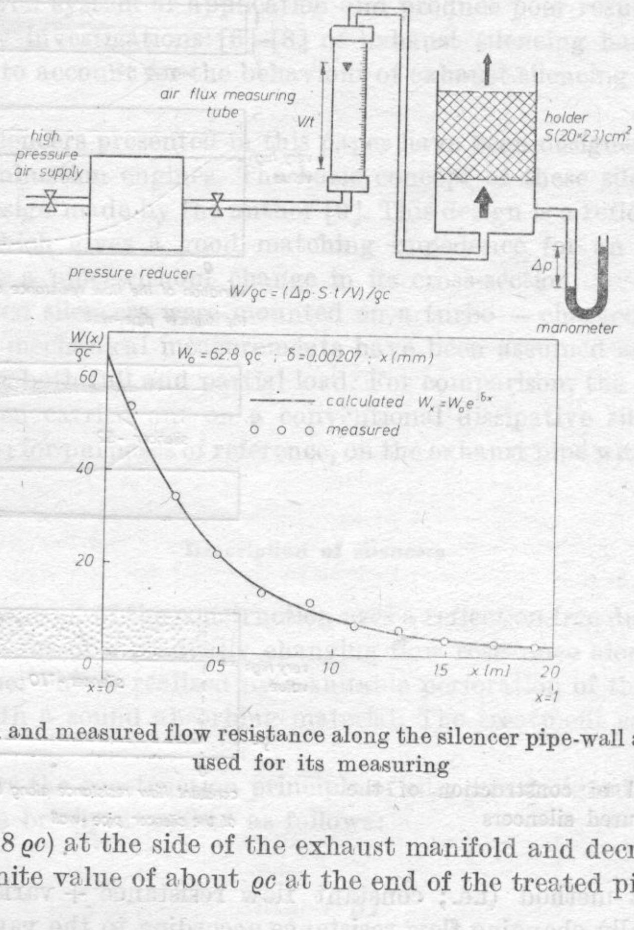

Fig. 2. Calculated and measured flow resistance along the silencer pipe-wall and the apparatus used for its measuring

high value (62.8 ϱc) at the side of the exhaust manifold and decreases exponentially to a definite value of about ϱc at the end of the treated pipe towards the tail-pipe.

Several kinds of absorptive materials have been used such as glass wool, mineral wool and steel wool in order to choose a suitable one with optimal effectiveness.

Silencer S2

It consists of two equally variable perforated areas at two opposite sides of the pipe walls (each perforated area is typical to hat of the silencer — *S1*). Each is covered with an absorbtive material packed in a half cylinder shaped closed chamber (of 15 cm radius) with a constant density, so that it gives the same variation of the flow resistance along the pipe wall as that of *S1*.

The silencer *S2* is applied in two forms, namely:

1 Silencer *FS2*

In this case, the silencer *S2* was used with the end of the perforated area being smaller in width ($b_0 = 1$ mm) towards the exhaust manifold.

2 Silencer *BS2*

In this case, the silencer *S2* was used with the end of the perforated area being greater in width (63 mm) towards the exhaust manifold.

Silencer TIO

It is a conventional dissipative silencer, type *TIO* – BURGESS-HD 2.5″. It consists of two treated pipes, each of them being one meter long. The pipe is uniformly and continuously perforated along its outer periphery over the whole length. The treated pipe is surrounded by an absorptive material in a cylinder-shaped closed chamber of 15.7 cm diameter.

Measurements

The described silencers were mounted on a turbocharged diesel engine, 6 cylinder in line, cubic capacity 6.7 dm^3, 130 KW at 2400 r.p.m., type VOLVO TD70-B. The measurements were carried on an engine bench in the High Technical School Autotechnik in Apeldora (Netherlands); see Fig. 3.

The exhaust pipe between the silencer and the exhaust monifold and the tail-pipe is long enough, so that the measurements are made sufficiently far away from the engine and outside the engine room, in order to isolate the structural noise from the exhaust noise. The lengths of the exhaust and tail-pipes were kept constant in all measurements.

A BRUEL and KJAER (B & K) 1/2″-microphone covered with a wind shield, was located 40 cm from the exhaust exit, making an angle of 45° with the axis the tail-pipe, and about 3 m above the ground for normal incidence. Before each series of measurements, the sound level amplification was checked and adjusted using a standard noise source to ensure accurate measurements. The complete acoustic system was calibrated by using a piston-phone. The calibration signal was recorded on magnetic tape at the beginning and end of each series to provide a reference for the analysing system and in order to check the system stability.

The ambient temperature was controlled to keep it nearly the same in all measurements. In the present case, the microphone is very near the exit of the tail-pipe and accordingly the ambient temperature does not affect the recorded sound level. To avoid the effect of wind, the measurements were carried out

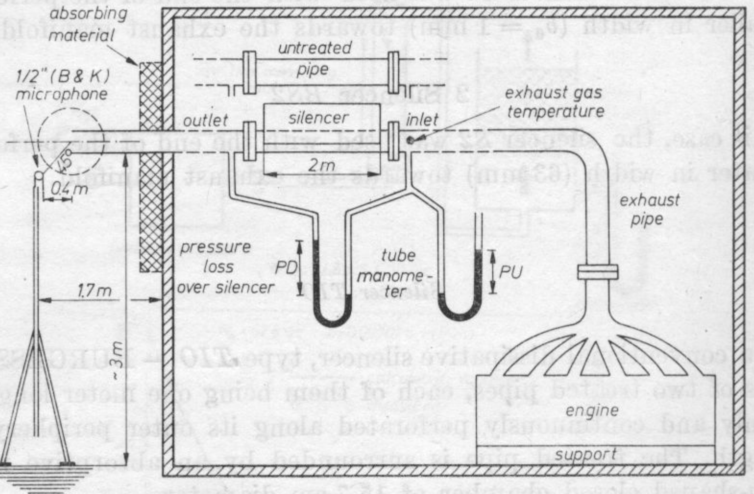

Fig. 3. The measuring arrangement and block diagram of an engine bench at the H.T.S. Autotechnik in Apeldoorn, Netherlands

when the wind speed was below 12 miles/hour. Humidity conditions and barometric pressure were also controlled during measurements. The sound absorption by the surface of the ground and the effect reflections from any surrounding objects were very small and could be neglected.

Prior to every test, the engine was warmed to an operating temperature. The engine speed n varied from 1400 to 2400 r.p.m. and was kept at ± 5 r.p.m. at each test. Measurements were carried out for partial load and full load of the engine every 200 r.p.m. increments. The exhaust gas temperature T °C at the front of the silencers under was measured by means of Cr—Ni thermocouple and thermoelectric-Digitime unit.

Acoustic measurement were carried out to determine the A-weighted sound pressure level L_A [dB (A)] at the microphone position and the insertion loss I_A, defined by the difference in sound level between the measurements with the untreated pipe and the pipe with silencer.

To study the influence of silencers on engine performance, several engine parameters have been measured as a function of engine speed and engine load. For purpose of reference, the same measurement have also been carried out on an untreated pipe with equivalent length (Fig. 3) to silencers.

The exhaust gas volume flow V [m³/s] in the silencer has been computed from the temperature T [K] and the pressure ($PU+1$ atm) values at the front of the silencer. The velocity of the exhaust gas G_v is determined from

$$G_v[\mathrm{m/s}] = \frac{4\dot{V}}{\pi d^2}, \tag{2}$$

where d is the diameter of the exhaust pipe.

The exhaust gas pressure difference PD over the silencer was also measured. The difference ΔPD with respect to the untreated pipe is the back pressure due to the silencer. The dissipated power W_d has been calculated from the relation:

$$W_d = \dot{V}PD \tag{3}$$

and accordingly the power loss due to the silencer ΔW_d can be determined from

$$\Delta W_d = W_d(\text{over silencer}) - W_d(\text{over untreated pipe}).$$

The effective electric power EP of the test engine was also measured

$$EP = \frac{F_m n \cdot 0.736}{1000} \quad [\mathrm{kW}], \tag{4}$$

where F_m is the mechanical power in Newtons and n is the rotational engine speed. Accordingly, EP with respect to the untreated pipe is computed:

$$EP = EP \text{ with silencer} - EP \text{ with untreated pipe}.$$

The fuel consumption F_c was determined from the relation:

$$F_c = \frac{\dot{V}_f \varrho}{t} \quad [\mathrm{g/s}], \tag{5}$$

where t is time in seconds, in which a volume V_f of the fuel is consumed and ϱ is the density of the level (for gasolin $\varrho = 0.84$ kg/m³). The change in fuel consumption $\Delta F_c(\%)$ due to the silencer with respect to that for the untreated pipe also determined,

$$\Delta F_c(\%) = 100\,\frac{F_c \text{ with silencer} - F_c \text{ with untreated pipe}}{F_c \text{ with untreated pipe}}.$$

Results and discussion

Analysis of the exhaust noise of the applied engine was made at different engine speeds and for both partial load and full load, in order to show the characteristic frequency patterns of the unsilenced noise level. For example, Fig. 4 shows the unsilenced and silenced A-weighted third octave noise level at 2000 r. p.m. for partial load. At low frequency below 315 Hz the exhaust noise level is a complex blend of its fundamental firing frequency and its higher harmonics as well as the resonance of all components of the exhaust system. From 315 Hz to about 630 Hz, the unsilected noise level increases slowly and afterwards a steady noise level can be stated till 5 kHz. In general, it was found, that the unsilected noise level depends on engine speed. At low engine speed, it depends strongly on engine load. The silenced noise levels obtained with both *FS2* and *BS2* silencers are similar and equal over most of the frequency range. From 250 Hz to 1000 Hz,

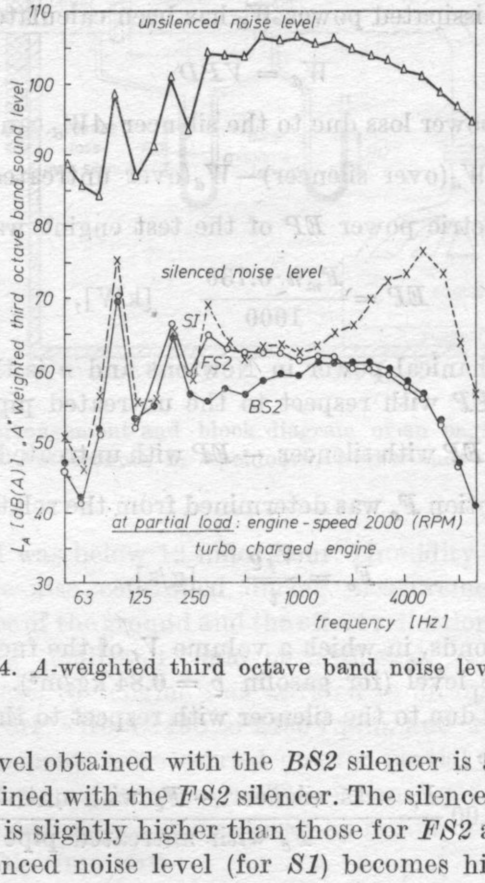

Fig. 4. *A*-weighted third octave band noise level

the silenced noise level obtained with the *BS2* silencer is 5 dB (*A*) in the mean lower than that obtained with the *FS2* silencer. The silenced noise level obtained with the *S1* silencer is slightly higher than those for *FS2* and *BS2* till 2000 Hz. Afterwards, the silenced noise level (for *S1*) becomes higher and this is due to the radiation effect, when the sound wave passes through the silencer unaffected by the absorbing material.

The effect of absorbing materials on the insertion loss I_A has been measured. For this purpose, absorptive materials such as glass wool, mineral wool and steel wool were used. Using these materials, the insertion loss I_A of the silencers *S1*, *FS2* and *BS2* were measured as a function of engine speed for both partial load and full load and compared with I_A measured with the conventional silencer *TIO* in the same conditions. Obtained results are represented in Fig. 5

from which the following results have been achieved:

1. In general, I_A depends on both engine speed and engine load and this may be due to different velocities of the exhaust gas at different operating conditions.

2. Glass wool, which is used only in silencer *S1*, is more effective than both

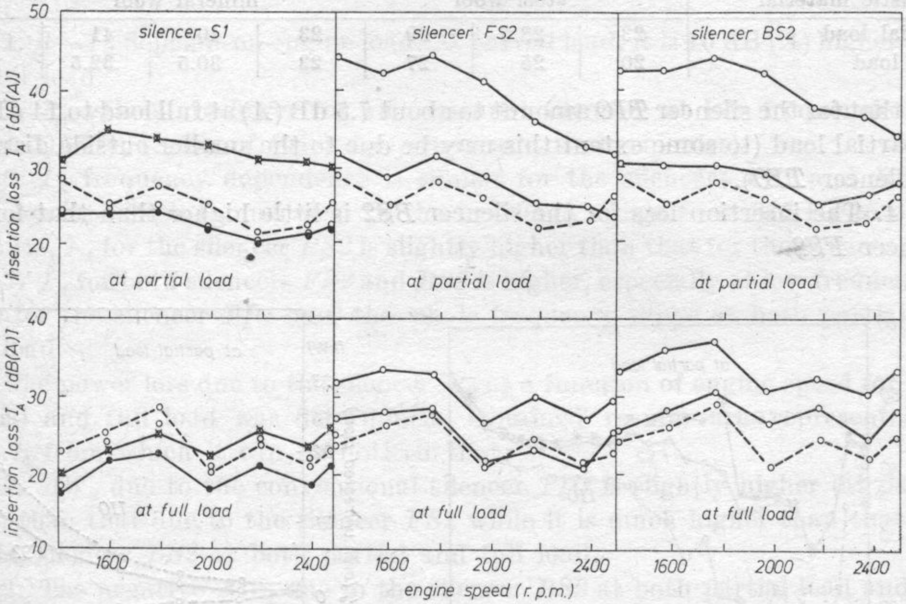

Fig. 5. *A*-weighted insertion loss I_A dB (*A*) against engine speed *n* (r.p.m.). x ——— x (glass wool), ○ ——— ○ (mineral wool), • ——— • (steel wool), ○ - - - - - - ○ Silencer *TIO* (conventional silencer)

mineral wool and steel wool at partial load. At full load, I_A obtained with glass wool becomes lower and this is due to the strong variation of its acoustic performance as a function of gas temperature and local velocity effects. The little difference in I_A – values obtained with glass wool and mineral wool at full load seems to confirm this. For this reason glass wool is not preferable to be used in exhaust silencers.

3. The figure clearly shows, that mineral wool is more effective then steel wool.

To compare the effectiveness of these materials, the I_A – mean values (at different engine speeds) are determined and represented in Table 1.

From Table 1, it can be seen that:

1. The difference in insertion loss I_A between the single slit silencer *S1* and the double slit silencer *FS2* or (*BS2*) is very clear.

2. Mineral wool as absorptive material has a 10 dB (*A*) at partial load and a 5 dB (*A*) at full load higher insertion loss compared with that for steel wool.

3. If mineral wool is preferred as absorptive material for both silencers *FS2* and *BS2*, the difference in I_A – mean values for silencers *FS2* and *BS2*

Table 1

	I_A – mean value dB (A)						
Silencer	*S1*	*FS2*	*BS2*	*S1*	*FS2*	*BS2*	*TIO*
Acoustic material	steel wool			mineral wool			
Partial load	23	28.5	29	23	40	41	26
Full load	20	25	27	23	30.5	32.5	24

and that for the silencer *TIO* amount to about 7.5 dB (A) at full load to 14 dB (A) at partial load (to some extent this may be due to the smaller outside diameter of silencer *TIO*).

4. The insertion loss for the silencer *BS2* is little higher than that for the silencer *FS2*.

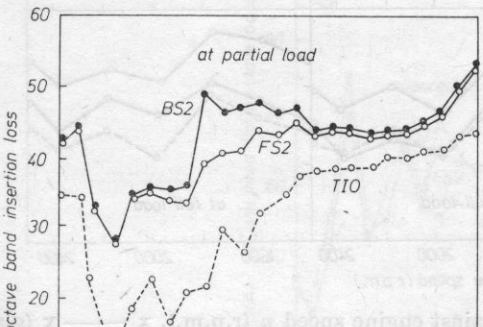

Fig. 6. A-weighted insertion loss I_A dB (A) against frequency f [Hz] mineral wool used as absorptive material

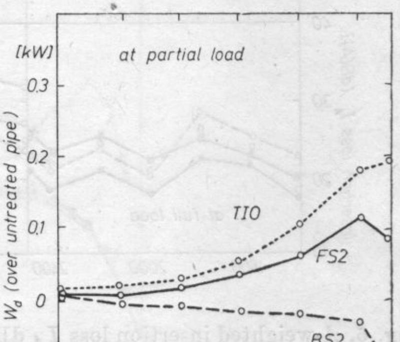

Fig. 7. Dissipative power loss ΔW_d [kW] against engine speed n (r.p.m.)

The frequency dependence of the A-weighted third octave band insertion loss I_A was measured for the silencers *FS2, BS2* (mineral wool as absorptive material) and silencer *TIO* at different engine speeds for both partial and full load. For example results obtained at engine speed 2000 r.p.m. for both partial load and full load are represented in Fig. 6. From the figure, it can be noticed, that:

1. $I-I_A$ depends on engine load. At partial load, it is 10 dB (A) higher than at full load.

2. I_A values are lower for all measured silencers at low frequencies (till about 315 Hz, region of firing frequency and its harmonics).

3. I_A frequency dependence is similar for the silencers *FS2* and *BS2* at both low and high frequencies. In the frequency range from 315 Hz to about 1000 Hz, I_A for the silencer *BS2* is slightly higher than that for the silencer *FS2*.

4. I_A for both silencers *FS2* and *BS2* is higher, especially at low frequencies, than for the silencer *TIO* over the whole frequency range at both partial and full load.

The power loss due to the silencer W_d as a function of engine speed for both partial and full load was determined. Obtained results are represented in Fig. 7, from which it can be noticed that:

1. ΔW_d due to the conventional silencer *TIO* is slightly higher (at partial load) than that due to the silencer *FS2* while it is much higher than that due to the silencer *BS2* at both partial and full load.

2. The negative ΔW_d due to the silencer *BS2* at both partial load and full load means that the power loss due to the silencer *BS2* is smaller than that due to the untreated pipe.

The measured and computed mechanical parameters obtained for the untreated pipe (2 m length and 7 cm diameter) as a reference are represented in Table 2 for both partial and full load. Table 3 gives the differences between these parameters for each silencer and for the untreated pipe, at different engine speeds for both partial and full load.

Table 2

	n	*PD*	*PU*	*Gv*	*EP*
	(r.p.m.)	(cm H_2O)	(cm H_2O)	[m/s]	[kW]
At partial load	1600	3.72	8.61	36.78	31.9
	1800	5.18	10.4	44.64	40.4
	2000	7.82	14.9	55.76	55.0
	2200	10.1	20.3	72.32	73.0
	2400	12.9	27.0	91.08	93.9
At full load	1600	7.82	15.8	66.08	98.3
	1800	11.2	21.9	77.75	109.1
	2000	14.7	25.4	89.89	119.7
	2200	16.9	30.5	99.83	125.6
	2400	20.2	39.9	110.3	129.6

In Table 3: Δ = measured parameter with silencer — measured parameter with untreated pipe.

Data represented in Table 3 leads to the following results:

1. The back pressure due to the *PD* silencers depends on both engine speed and engine load. At partial load, it increases for both silencers *FS2* and *TIO*, at increasing engine speed. At full load, it is slightly lower than at partial load at the same engine speed. The back pressure due to the silencer *BS2* is approximately independent from engine speed at partial load, while it decreases with engine speed at full load. The negative ΔPD due for the silencer *BS2* means that the pressure difference over the silencer *BS2* is lower than over an untreated pipe with equivalent length; what is at this moment unexplained. The back pressure ΔPD due to the conventional silencer *TIO* is clearly higher than that due to both silencers *FS2* and *BS2* at all engine speeds, and for both partial and full load.

Table 3

	n (r.p.m)	ΔPD [cm H_2O]			ΔPU [cm H_2O]			ΔGv [m/s]			ΔEP [kW]			ΔF_c [%]		
		FS2	BS2	TIO	FS2	BS2	TIO	FS2	BS2	TIO	FS2	BS2	TIO	FS2	BS2	TIO
at partial load	1600	0.9	-0.1	1.2	-0.2	-1.4	0.2	-1.8	-1.3	-1.9	-0.2	-0.1	-0.1	-0.3	0.0	0.1
	1800	1.4	-0.2	1.6	1.3	-0.6	0.0	1.6	-0.8	-1.9	-0.1	-0.1	00.1	0.7	0.1	-0.2
	2000	1.8	-0.8	2.8	0.9	-1.6	2.1	-0.4	-0.3	-1.0	00.0	00.0	00.2	0.2	0.8	1.8
	2200	2.4	-0.5	4.2	0.2	-1.3	2.4	-1.5	-1.8	-2.5	00.2	00.0	00.2	0.5	-1.3	1.3
	2400	3.6	-0.5	6.0	0.2	-3.5	3.1	-1.1	-1.3	-4.5	-2.3	-2.3	-1.9	-0.8	0.2	0.4
	mean	1.7	-0.4	3.2	0.5	-1.7	1.6	-0.6	-1.1	-2.4	-0.5	-0.5	-0.3	0.06	0.04	0.7
at full load	1600	2.0	-0.9	1.0	0.2	-1.1	-0.3	0.9	-0.3	-2.0	-0.8	-2.4	-1.7	0.9	-1.0	0.6
	1800	1.5	-1.5	1.7	-0.4	-2.7	-0.4	2.5	2.3	-0.7	-0.2	-0.6	-0.6	-0.2	0.0	0.2
	2000	1.2	-2.7	2.2	1.6	-1.5	2.0	3.4	1.7	-1.5	-1.1	-2.1	-2.3	0.9	0.7	0.7
	2200	2.4	-2.4	3.9	1.4	-1.6	2.8	3.7	4.0	-1.2	-0.8	-1.4	-1.7	-0.4	1.4	0.9
	2400	1.5	-3.5	4.2	-3.7	-7.4	-0.4	5.3	4.4	-1.2	-1.4	-2.3	-2.7	1.4	1.1	1.2
	mean	1.7	-2.2	2.6	-0.2	-2.9	0.7	3.2	2.4	-1.3	-0.9	-1.8	-1.8	0.5	0.4	0.7

2. The pressure of the exhaust gas at the front of the silencers ($PU + 1$ atm) with respect to that for the untreated pipe ΔPU behaves differently at different engine speeds and different loads. The negative sign means that the pressure of the exhaust gas at the front of the silencer is lower than that at the front of the untreated pipe.

3. At partial load, the velocity of the exhaust gas *GV* is lower for all silencers than for the untreated pipe. The lowest *GV* values noticed are those for the silencer *TIO*. At full load, it is lower than for the untreated pipe in case of the silencer *TIO*, but the difference is somewhat smaller than at partial load. In case of the silencers *FS2* and *BS2*, it is higher than for the untreated pipe. It increases with increasing engine speed.

4. The electric power *EP* [kW] of the engine is slightly lower with the silencers than for the untreated pipe. It is lower than for an untreated pipe by about 0.7% in the mean at partial load and about 1.5% for the *TIO* and *BS2* silencers while it is only 0.77% lower for the *FS2* silencer in the mean.

5. The fuel consumption F_c is only silightly different from that of the untreated pipe.

Conclusions

Measurements and results lead to the following conclusions:

1. At a constant flow resistance, mineral wool as absorptive material is more effective than steel wool.

2. The insertion loss of a dissipative silencer becomes lower at both high engine speed and full load. This may be due to higher velocity at these two conditions.

3. A high level of acoustic absorption could be realized over a wide frequency range without

a) changing the electric power of the applied engine

b) changing the fuel consumption

with respect to an untreated pipe.

4. The measured low back-pressure due to the designed silencers *FS2* and *BS2* compared to those measured for the conventional *TIO* and the untreated pipe, leads to a minimum power loss of the exhaust gas. In case of silencer *BS2*, a power gain noticed over the whole range of engine speed and for both partial and full load. As a result, the characteristic of the engine, on which those silencers are mounted does not alter and the combustion process can show improvement.

Acknowledgements

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