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REDUCTION OF NOISE FROM A 125 CC MOTO-CROSS MOTORCYCLE

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INTRODUCTION

During recent years the performance of motor racing has become more popular in Denmark. Combined with the growing environmental consciousness among the population and the relatively high population density of the country, this has resulted in a number of environmental noise problems in connection with motor racing tracks.

DESCRIPTION OF VEHICLE

A Suzuki RM 125 moto-cross motorcycle of the year 1980 was used in the investigation. The standard vehicle is equipped with a 125 cc air-cooled two-stroke petrol engine, which yields a maximum of 19.5 kW at 10,500 RPM. The air intake into the carburettor takes place through an air filter of polyurethane, which is placed in a plastic box with a volumen of app. 6 litres. The exhaust is led through a 1200 milimetres long exhaust pipe, which is designed as an expansion chamber. After the exhaust pipe a 620 cc absorption exhaust silencer is mounted. The cylinder and cylinder head are made of light-alloy metal and equipped with cooling fins for air cooling of the engine.

NOISE SOURCE IDENTIFICATION

The noise source identification was performed in order to determine the relative contribution of the separate partial sources of noise to the total noise emission of the vehicle. The contribution was determined from the following partial noise sources:

1) Exhaust opening (standard exhaust silencer).
2) Air intake (standard intake filter).
3) Cylinder and cylinder head.
4) Exhaust pipe (surface radiation).

Rubber pieces stuck between the cooling fins were used as a means of reducing the noise emitted from the cylinder and cylinder head. However, this arrangement only had a limited effect (2-3 dB reduction of the sound power level for frequencies above 2 kHz). Therefore, supplementary tests were carried out with a Suzuki RM 125 with a water-cooled engine (full-speed passing measurements). The results of these tests indicate that the cylinder emission of a water-cooled engine is substantially lower than that of an air-cooled engine in the frequency range 1 kHz to 4 kHz.

CONCLUSION

Based on the present investigation it can be concluded that a replacement of the original 620 cc absorption exhaust silencer with a 1850 cc absorption silencer, will reduce the A-weighted sound power level of the vehicle by 4-5 dB (measured at 8000 rpm and full load). A further reduction of 2 dB can be achieved by vibration damping of the cooling fins of the engine with rubber pieces, vibration damping of the exhaust pipe, and slightly modifications of the air intake filter.

Further reduction of the noise emission from the vehicle by app. 3-4 dB should be possible, if the noise from the cylinder and the cylinder head could be reduced efficiently, especially in the frequency range 1-2 kHz. This could implicate use of water-cooled engines, but needs further investigation.

ACKNOWLEDGEMENTS

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The applied measurement method was the so-called selective wrapping. During the measurements the vehicle was driven on a car dynamometer. The noise source identification was performed at 8,000 rpm, and full load, which corresponds reasonably well to realistic driving conditions. The car dynamometer was placed in a workshop and the contributions of the partial noise sources were determined as sound power level per 1/3 octave.
The main results of the noise source identification are shown in Figure 1. The figure shows the sound power level per 1/3-octave for standard exhaust silencer, standard intake filter, cylinder and cylinder head, and exhaust pipe, respectively. It is seen that the noise from the exhaust silencer is predominant in the frequency range from 500 Hz to 3 kHz, and that the cylinder emission can be expected to yield a substantial contribution in the frequency range from 1 kHz to 2 kHz, while the intake noise is the most powerful for frequencies lower than 500 Hz.

NOISE REDUCING MEASURES

In order to reduce the exhaust noise various exhaust silencers were tested. Some of them were standard absorption silencers with a cubic content of 800-1000 cc and some were test models of a 1850 cc absorption silencer and a combined reflection/absorption silencer. The latter, which turned out to be the most efficient, consisted of a 1850 cc double-chamber resonator, tuned to approx. 600 Hz, followed by a 925 cc absorption silencer. The test model of a 1850 cc absorption silencer had a slightly inferior insertion loss at approx. 600 Hz but will probably be sufficiently effective, reducing the A-weighted sound power level of the exhaust noise to 108 dB. The standard absorption silencers had a poor insertion loss in the frequency range from 400 Hz to 1.6 kHz. Furthermore, one of the standard silencers had a poor insertion loss above 1.6 kHz, as the applied absorption material had a too high flow resistance. None of the tested exhaust silencers reduced the mechanical power of the vehicle.

After having applied the improved exhaust silencer, the engine itself was the dominant noise source. An investigation of the noise radiation from the engine was performed using acoustic intensity technique (Brüel & Kjær 2-microphone system type 3360). The system was connected to a HP 1000 computer for data processing.

The acoustic intensity was measured in 35 positions 50 mm from the left side of the engine. The measurement positions were placed in a grid, and spaced 50 mm from each other. In each position the sound intensity was measured in 3 orthogonal directions. The resulting intensity vector could then be calculated and displayed. Figures 2-4 show the intensity vector for the 1/3-octave band at 1600 Hz. As can be seen, the acoustic intensity in this frequency region is primarily emitted from the upper part of the cylinder.

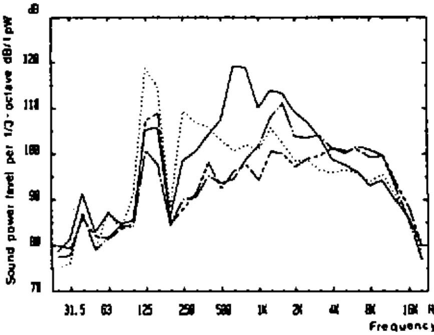

Figure 1.
Noise source identification for standard RM 125, 8000 rpm, full load.

———— Exhaust , L_{WA} = 123 dB
········· Intake , L_{WA} = 111 dB
——·—— Cylinder, L_{WA} = 115 dB
- - - - - Exh.pipe, L_{WA} = 106 dB
Residual sources, L_{WA} = 110 dB
Total emission , L_{WA} = 124 dB

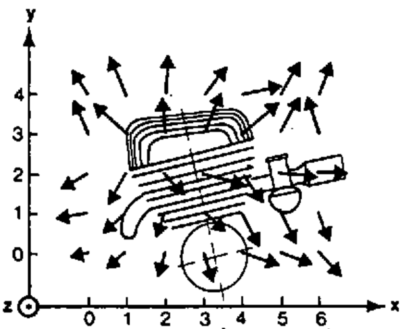

Figure 2.
Acoustic intensity vector in the x-y plane. 1/3-octave centre frequency f_c = 1600 Hz.
Left side of engine. 8000 rpm, full load.
Zero level 90 dB re. 1 pW
Scale: ⟼ = 20 dB

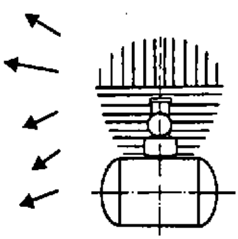

Figure 3.
Acoustic intensity vector in the y-z plane at x = 3.

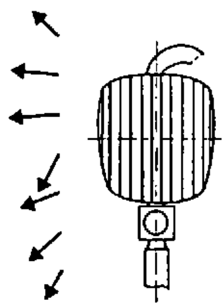

Figure 4.
Acoustic intensity vector in the x-z plane at y = 2.