I.C. ENGINE COMBUSTION NOISE AND METHODS FOR ITS REDUCTION

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INTRODUCTION

It is now well established that the type of combustion system used in an engine design has a very considerable effect on the noise produced by the engine. Only in the high speed petrol engine and the turbocharged diesel at full load is noise of a mechanical origin a controlling feature. As combustion development continues we approach the time when mechanical noise will become a major consideration, but with increasing emphasis on increased efficiency and the imminent changes in fuel quality and type this time may well be some way off.

RELATION BETWEEN OVERALL ENGINE NOISE AND COMBUSTION NOISE

The overall noise at full load of some 44 different I.C. engines, measured at the ISVR over the last 12 years, is shown in Figure 1 as a function of engine speed. The engines tested include two-stroke diesels, normally aspirated and turbocharged four-stroke diesels, indirect injection (swirl chamber) diesels and petrol engines. For all engines at rated speed the variation in overall noise

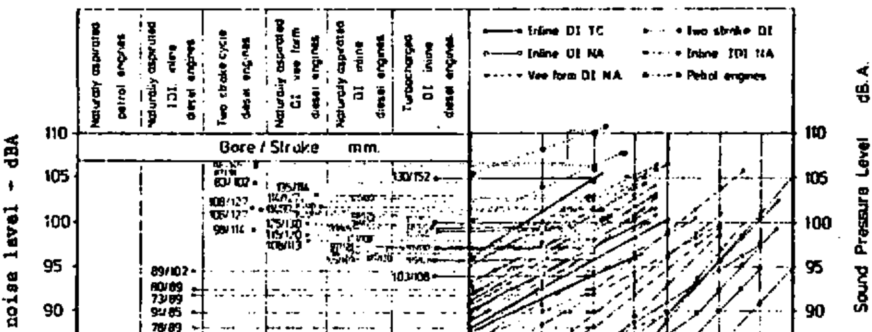

Fig. 1 - Noise versus speed at full load for a range of engine types

measured 1m from the engine surface is from 95 to 110 dBA. The importance of engine speed in controlling engine noise is clear from the figure, but even at a constant engine speed of 2000 rev/min, for example, there is a 35 dBA spread in overall noise level according to engine type. It is well known that combustion noise in normally aspirated direct injection diesels is the predominant noise source. From Figure 1 the average noise level for this type of engine at 2000 rev/min is about 100 dBA. For the level of a petrol engine to be 75 dBA at 2000 rev/min (the lowest noise level measured) it therefore follows that the combustion noise in this petrol engine must be at least 25 dBA lower than the diesel. It is therefore clear from Figure 1 that very large differences in combustion excitation level should be measured in the various I.C. engine combustion systems. The engine structure is loaded by a force down the connecting rod determined by the product of cylinder pressure level and piston area. This applied force is shown in Figure 2. For the diesel engine the root mean square oscillating force applied to the structure at cycle firing frequency (16.6 Hz) is some 2830 lbs (12,589 N), whilst at 1000 Hz the applied loading is some 20.6 lbs (91.6N). For the petrol engine at cycle firing frequency the rms oscillating force applied is some 533 lbs (2371 N) whilst at 1000 Hz it is only 0.53 lbs (2.22 N). Thus the square of the force input to the petrol engine structure at 1000 Hz is some 33 dB lower than that of the diesel. For a linear system the noise due to combustion at 1000 Hz would also be lower by 33 dB, provided that no other noise sources were present and thus the major effect of the form of cylinder pressure development on overall engine noise is clear.

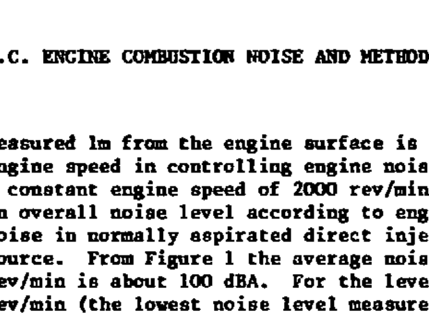

Fig. 2 - Combustion force spectra for D.I. diesel and gasoline engine

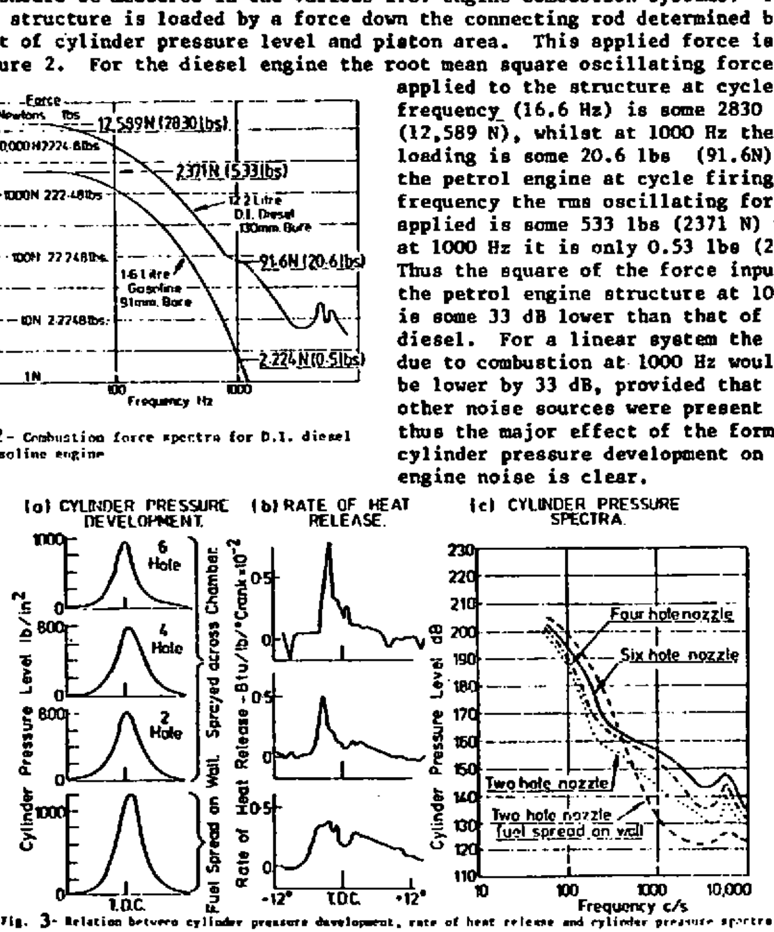

Fig. 3 - Relation between cylinder pressure development, rate of heat release and cylinder pressure spectra

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Priede (1) showed, some 20 years ago, that the noise of diesel engines could be related to changes in the level of the high frequency components of the frequency spectrum of the cylinder pressure development. Making sure to use flush mounted pressure transducers such measurements have been found to be remarkably repeatable. Figure 3 shows the relationship between cylinder pressure development, rate of heat release and cylinder pressure spectra. When combustion is abrupt, the rate of pressure rise and initial peak rate of heat release are large and the level of the high frequency (above 500 Hz) cylinder pressure spectrum components are also high (2). There is a direct relationship between initial peak of heat release and rate of pressure rise when combustion is abrupt and under these conditions combustion noise can be related to either (3). However, when the combustion is smooth, as in petrol combustion, well matched full load turbocharged diesel or high speed two-stroke, the combustion noise can only be related to the form of cylinder pressure development (4). At present the only fully reliable method of combustion noise assessment is by the full frequency analysis of the exact cylinder pressure development. Priede (5) also showed that the noise of engines (and machines) increased rapidly with speed because of the rapidly sloping frequency spectra which shifted upwards in frequency as the cycle fundamental frequency increased, giving rise to higher force inputs to the structure in the high frequency range.

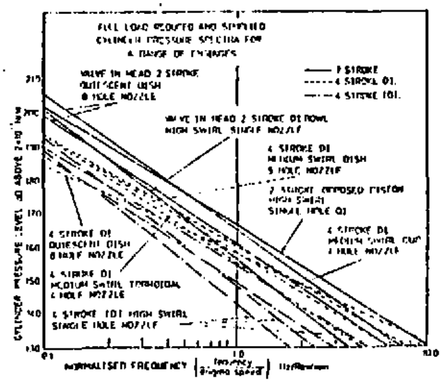

Fig.4. Full load normalised and simplified cylinder pressure spectra for a range of n.a. engines.

In order to compare the various combustion systems it is necessary to reduce them to a standard form which is independent of speed. This can be achieved by normalising the spectra by dividing by speed and the spectra can be further simplified by a straight line approximation over the frequency range of interest - 500-3500 Hz.

In Figure 4 the simplified and normalised spectra for 13 different normally aspirated diesel engines are shown. The variation in combustion excitation level at 1000 Hz ranges from 27 dB at 1000 rev/min to about 20 dB at 3000 rev/min depending on the combustion system used. Two-stroke diesel engines have the highest excitation levels with Direct Injection four-strokes slightly lower and Indirect Injection four-strokes lower still. In general it can be seen that the multiple hole nozzle low swirl chamber configuration gives rise to higher levels of combustion excitation than does the high swirl single hole nozzle chamber configuration. The normalised spectra can be grouped according to the form of combustion and then be averaged to give a single line representation of the average combustion excitation level in that group. This comparison of combustion systems is shown in Figure 5. In broad terms the combustion excitation for two-stroke diesels is a little higher than that for normally aspirated four-stroke diesels (however for the same swept volume the two-stroke will be producing twice the power output). Indirect injection diesels and turbocharged diesels (two and four stroke) both have about the same combustion excitation at about 8-12 dB lower than the normally aspirated four-stroke and petrol engines have an even lower excitation at about 18-23 dB lower than the normally aspirated four-stroke. Using these average values, measured directly from the engine combustion system, in combination with an experimentally derived 'Standard Structure Attenuation' which is assumed to represent all engines, the average noise levels likely to be associated with each combustion system can be calculated and the results compared to the actual measured noise of the engines (6).

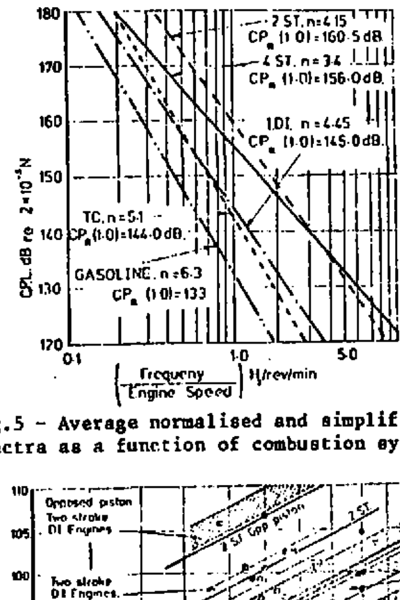

Fig.5 - Average normalised and simplified spectra as a function of combustion system

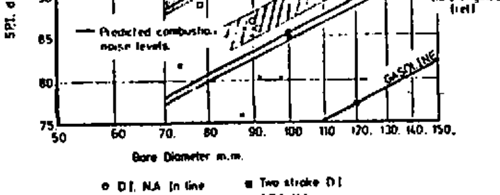

Fig.6 - Measured and predicted overall noise versus bore size for engines in the various groups at 2000 rev/min

To illustrate this the noise data of Figure 1 for an engine speed of 2000 rev/min are plotted as a function of bore size (Figure 6). For the diesel engines the variation of noise with (bore)^5 can be clearly seen as also can the grouping of the noise levels according to the form of combustion.

It can be seen that the measured overall noise levels of the engines do fall into specific groups:-

(a) All normally aspirated DI engines fit within a 3 dB band of slope (bore)^5. It is clear that there are no differences between the overall noise of vee-form and in-line engines. Some of the IDI engines also fall within this same band but these generally have 'abrupt' or 'advanced' pressure diagrams.

(b) The turbocharged engines occupy a band about 3 dBA below.

(c) The remaining IDI engines fall within a band some 8 dBA below the DI engines. These engines generally have smooth or retarded type pressure diagrams.

(d) Two-stroke cycle engines fall within a band some 3 dBA higher than the DI engines.

(e) Opposed piston two-stroke cycle engines (separate crank-shaft arrangements) fall in a band of 12 dBA higher than DI engines.

(f) Petrol engines show considerable scatter but are about 15 dBA below the DI engines.

It is clear that the effect of engine structural configuration (for current production engines) on noise is small. A comparison between the measured noise levels and those calculated to be due to combustion induced noise shows that for conventional and opposed piston two-stroke diesels the calculated combustion noise levels correlate very well with the measured data. However, the levels calculated for the IDI petrol and turbocharged engines lie below those measured, particularly in the case of the turbocharged engines (full load). This supports the generally held view that in petrol engines and in turbocharged diesel engines (full load) mechanical noise is important.

REDUCTION OF COMBUSTION NOISE

The noise produced by an IC engine combustion system is determined (for a given structure) by the detail design of the system used (swirl level, geometry, speed, fuel used, etc.) in conjunction with the detail injection characteristics and the use of turbocharging or any other form of supercharging which determines the form of the cylinder pressure diagram.

It is given generally by

$$\text{Overall Noise dBA} \propto \text{Intensity (I)} \propto N^n B^5 \qquad (1)$$

N = engine speed
B = engine bore
n = combustion index

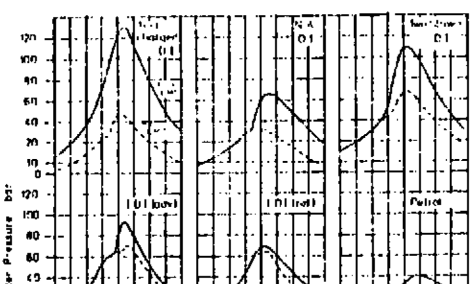

Fig. 7 - Typical cylinder pressure diagrams for various combustion systems.

Figure 7 illustrates the general form of cylinder pressure development for the main types of IC engine. In the absence of a complete frequency analysis of the development it is usual to try to characterise the noise as being determined by either peak pressure, rate of pressure rise, or acceleration of pressure rise.

For cylinder pressure developments where the initial stages of burning are abrupt (normally aspirated two and four stroke, I.D.I. and part load turbocharged) the major influencing parameter has been shown to be rate of pressure rise and a relationship for predicting the overall noise at any speed and load is (7):-

Overall noise dBA at 1m:

$$\text{dBA} = 30\log_{10} N + 50\log_{10} B + (2.1\ \text{RPR}-13)\log_{10}\left(\frac{5455}{N}\right) - 103.0 \qquad (2)$$

N = engine speed
B = engine bore mm
RPR = maximum rate of pressure rise in bar /^{o}CA

Reduction of combustion noise can therefore be achieved by reduced running speed, reduced bore size or reduced rates of pressure rise in these engines, the rate of pressure rise having more effect at low speed. In new engines a choice of speed and bore is possible but in existing engines only rate of pressure rise can be controlled. Equation (2) can be used to estimate, from the average measured noise levels of the various engine types, the average rate of pressure rise associated with those combustion systems. The results are shown in Table 1.

TABLE 1. RATES OF PRESSURE RISE FOR VARIOUS COMBUSTION SYSTEMS

Combustion Combustion System	Average Maximum Rate of Pressure Rise Estimated from Measured Noise
N.A. vee-form four-stroke	8.0 bar/^{o}CA (117psi/^{o}CA)
N.A. inline four stroke	7.2 " (105psi/^{o}CA)
Inline and vee turbocharged four-stroke	5.0 " (72psi/^{o}CA)
I.D.I. four-stroke (N.A.)	3.2 " (47psi/^{o}CA)
Roots blown two-stroke	5.0 " (72psi/^{o}CA)

It is clear therefore that the means of obtaining substantially different rates of pressure rise lies in the selection of a complete combustion system, only small changes being obtained by modifications to an existing one.

When premixed burning phase becomes smoothly blended to the compression phase - as in the turbocharged diesel at full load or the petrol engine - the controlling features for noise become the diagram half width and the rate of pressure rise but in this case the rate of pressure rise affects the noise to a larger extent, the noise intensity I:

$$I \propto dBA \propto 20 \log_{10} \left(\frac{1}{W}\right) + 35 \log_{10} (RPR)$$

RPR = maximum rate of pressure rise
W = diagram half width

Figure 8 shows a comparison of two petrol engine cylinder pressure diagrams in which the level of high frequency excitation is identical, but in which the peak pressure varies by a factor of four. Also shown (dotted) is a fast burn cycle and its analysis showing the influence of rate of pressure rise.

Turbocharging is a very effective means of reducing combustion noise at full load and has become generally used to create higher specific power without an increase in noise (which would result if speed or engine size were varied). The turbocharger, when producing reasonable boost, pre-heats the inlet air, and this results in a reduced delay period and consequently smoother pressure development (6) Figure 9. The result is to produce a substantial reduction of

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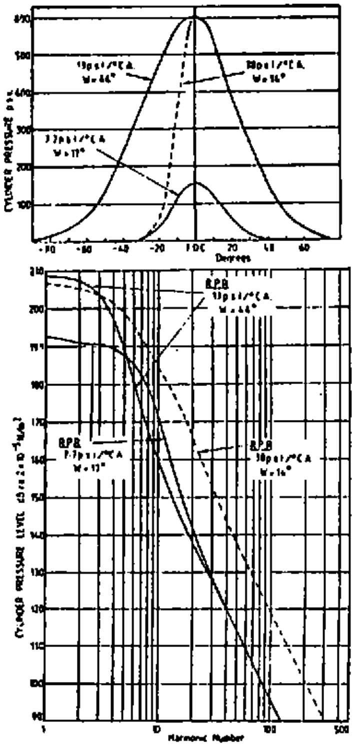


Fig.8 - Relation between form and harmonic series for smooth cylinder pressure developments

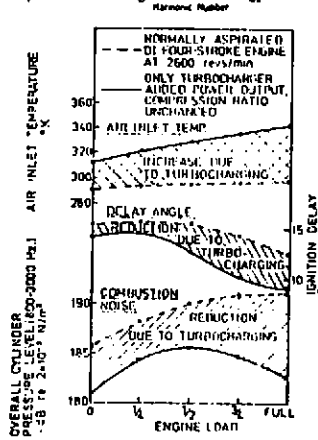
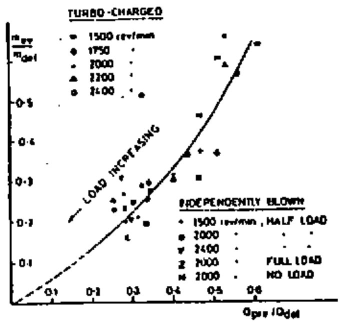


Fig.9 - Mechanism of noise reduction and effect of load due to turbocharging

noise at full load but to produce higher noise levels at part load and during acceleration. Higher chamber densities during combustion tend to increase the delay period again and consequently the control of combustion noise offered by turbocharging is rather limited - the combustion chamber and fuel injection equipment design being a controlling factor. Figure 10 illustrates this for an eight litre inline six four-stroke engine in which measurements were taken with a turbocharger fitted and also with an independent air supply. The relation between the mass of fuel evaporated during the ignition delay and mass burned during the pre-mixed phase is similar whether the turbocharger is fitted or not and varies between 0.2 and 0.6 depending on the rate of injection and rate of fuel evaporation - features which are controlled by general chamber and injection system design. Retard of injection is well known to produce small reductions in noise at about ½ dB per crankshaft degree and perhaps 2 to 4 dBA total reduction possible.

The considerable effect that other injection characteristics can have on combustion noise is illustrated in Figure 11 for a 2½ litre I.D.I. diesel running at 3600 rev/min. Two different fuel pumps were fitted to the engine, with substantially different injection rate characteristics, pump A being the standard pump. The measured rates of injection are shown together with the total fuel delivery. For the standard pump A almost all the fuel was injected during the delay period (90%) whilst for pump B the initial rate of injection was very much lower with about 25% injected during the delay period. The rates of heat release illustrate the severity of combustion for pump A and a comparison of the cylinder pressure shows the very marked effect on rate of pressure rise which results in a marked reduction in combustion induced noise. For pump B the engine power output was unchanged and both smoke and s.f.c. were reduced.

FIG 10. CORRELATION BETWEEN MASS of FUEL EVAPORATED DURING the IGNITION DELAY and MASS BURNING IN the PRE-MIXED PHASE : ENGINE A



CONCLUSIONS

Historically, combustion induced noise has been controlled by controlling the rate of pressure rise and this has been achieved by the adoption of markedly different combustion systems. Turbocharging offers good noise reductions at full load but has some drawbacks at part load and under transient conditions. Even with turbocharging the correct matching of combustion geometry and fuel injection characteristics for low noise is essential for success.

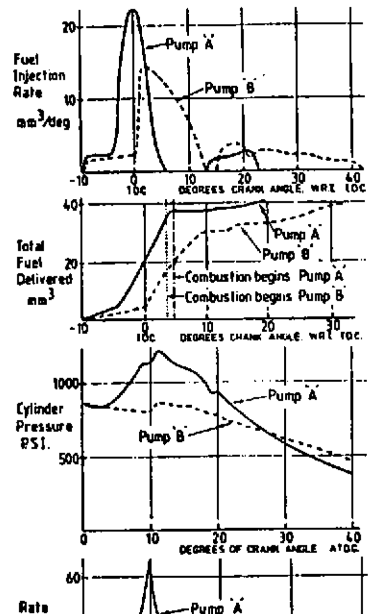


Fig.11 - Effect of fuel injection characteristics on combustion noise

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