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Acoustic Measurements in Diesel Engine Condition Monitoring

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Abstract— ***Diesel engine acoustic signals are rich in information about operating parameters and physical condition. Unfortunately, due to the number of acoustic sources and the environmental effects, these signatures are very complex and may be highly corrupted, making fault detection and diagnosis difficult. This research focuses on the extraction of useful information about diesel engine operating conditions and health from the acoustic signals emitted in a normal, acoustically untreated laboratory environment, without any sound measurement precautions. Signal processing techniques were applied to real data collected from the engine test rig operating under normal and abnormal conditions for the purpose of condition monitoring.***

Key words: **Diesel engine, Acoustic, Diesel engine noise, Condition monitoring, internal combustion.**

I. INTRODUCTION

Diesel engine failure can have severe cost consequences for the plant served; therefore, care must be taken to protect such engines from failure. Condition monitoring systems can detect faults in the early stages before failure occurs. Acoustic monitoring of diesel engines can deliver useful information, so it is important to study these acoustic signals.

In most cases machines produce not only vibration but also noise, otherwise known as acoustic signals. These are often picked up by microphones, offering several advantages over vibration monitoring: microphones are easy to install and have greater frequency response ranges, thus delivering more detailed information. This remote, non-intrusive data collection by microphones is an attractive option for online condition monitoring. Another advantage of acoustic monitoring is that when the problem of noise is the subject of study, this technique provides a more direct interpretation of noise sources and generation mechanisms. One of the main problems with acoustic monitoring is the contamination of acoustic signals by background noise. The study of diesel engine noise in this research work employs this relatively new monitoring technique, as diesel engines have a number of noise sources, each contributing to some extent to the total noise level. In short, acoustic monitoring is preferable to vibration monitoring in that it allows noise from all possible sources to be recorded over a wider frequency range.

II. LITERATURE REVIEW

Chan and Anderton studied the relationship between the acoustic signals emitted by engines and other parameters such as engine size and type, speed, load and combustion system [1]. A variety of engine noise level prediction formulas were obtained, based on past experience and experimental results [2]. Schmillen and Wolshchendorf [3] used cyclic combustion noise variation in fault diagnosis. Aouichi and Herrmann [4] investigated the internal mechanisms causing diesel engine noise and described how noise level changed with fault condition. Izumi compared combustion noise and vibration information [5]. Chaudhri showed that sound measurement is more useful in the detection of faults which cause considerable changes in noise output but little or no change in vibration signals [6]. Gu et al. managed to eliminate background noise by digital filtration [7]. Albarbar et al. found that the lubrication condition of the engine had a noticeable influence on structure-borne acoustic and airborne acoustic signals [8]. In [9] a simple approach was used to predict the engine health using statistical parameters such as RMS and kurtosis. Alhussain et al. outline the diesel engine acoustic sources and their fundamental characteristics are studied in time domain, frequency domain, mean value, kurtosis and RMS values [10]. Members of the same team extracted useful information about the diesel engine lubricating oil quality and condition by analyzing the measured vibration and airborne acoustic signals caused by piston slap and investigated the effects of load, speed variation, and temperature [11].

III. TEST RIG DESCRIPTION AND SPECIFICATIONS

This research used a Ford FSD 425 four-cylinder four-stroke direct injection diesel engine, a model widely used as a power supply in land and marine transportation vehicles [12]. Figure 1 shows the diesel engine test rig, with a dynamometer fitted to vary the external load. Table 1 lists some of its technical specifications.

Table 1. Test rig specifications

Bore	93.67 mm
Stroke	90.54mm
Cubic capacity	2496 mm^3
Power output	52 kW @ 2700 rpm (BS)
Torque	145 Nm @ 2700 rpm (BS)
Injection sequence	1,2,4,3
Compression ratio	19:1
Inlet valve timing	Opens 13° before TDC, closes 39° after TDC
Exhaust valve timing	Opens 51° before BDC, closes 13° after TDC
Compression pressure	3.38 MPa @ starter motor speed

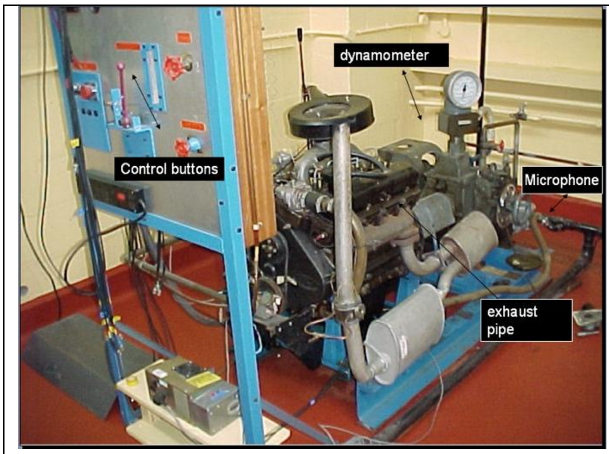

Fig 1. The test rig

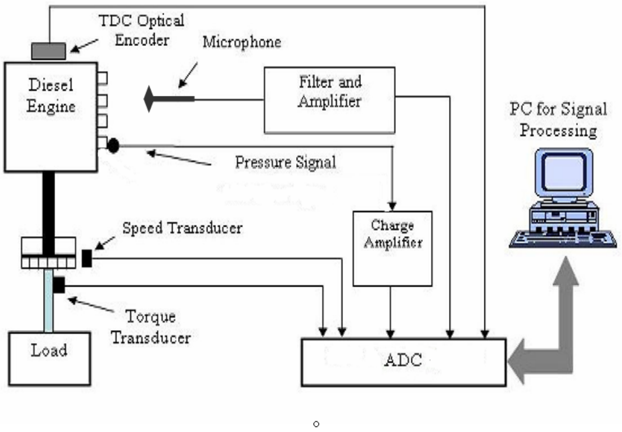

Fig 2. Schematic diagram of engine test system

IV. TEST PROCEDURE AND FAULT SIMULATION

The research objective was to acquire real data from the engine test rig operating under normal and abnormal conditions and to apply signal processing techniques for the purpose of condition monitoring. The engine was tested under loads of 0, 20, 40, and 60 Nm at speeds of 1000 and 2000 rpm. Due to time constraints, only exhaust valve clearance faults were investigated. As shown in Figure 3, cylinder #1 exhaust valve clearance was altered from 0.4 mm (healthy clearance) to 0 mm, 0.25 mm, and 0.6 mm, to simulate to some extent faults of leakage, exhaust valve timing and in-cylinder pressure.

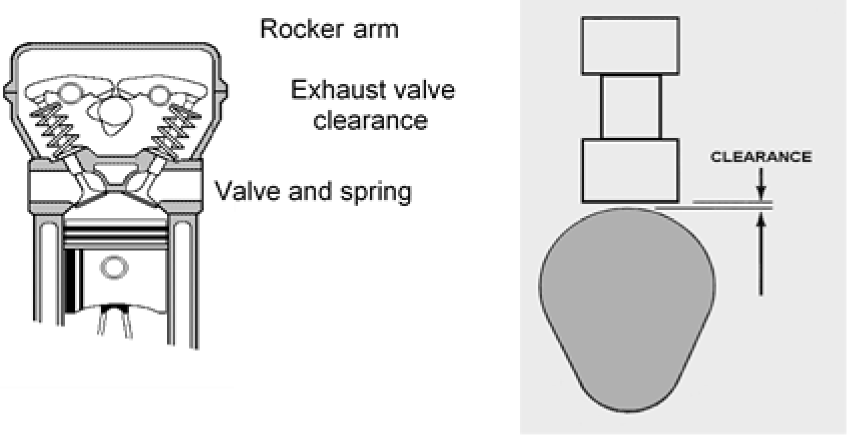

Fig 3 Exhaust valve clearance

V. DIESEL ENGINE ACOUSTIC WAVEFORM AND IN-CYLINDER PRESSURE

Figure 4 shows the amplitude of the acoustic signal in the time domain, with pressure in cylinder #1, recorded at 1000 rpm under no load.

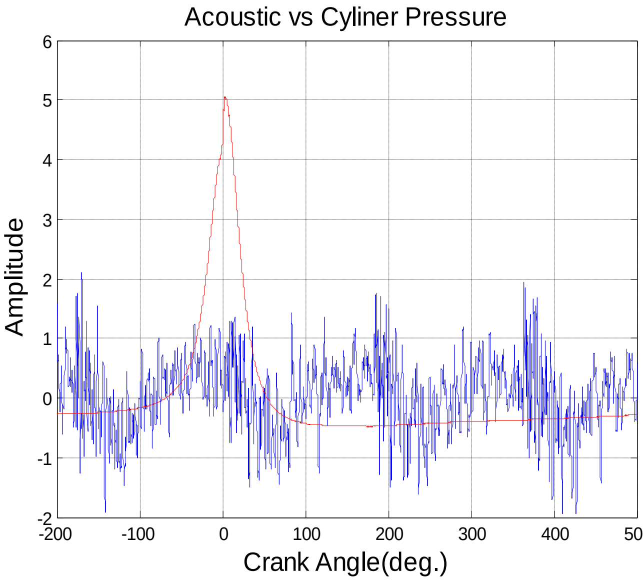

Fig 4. Engine acoustic waveform (blue) and pressure in cylinder #1 (red)

The main features to be observed from the acoustic waveform shown in Figure 4 are the four peaks corresponding to the engine firing sequence and representing combustion events in cylinders #3, 1, 2, and 4 sequentially. The pressure trace of cylinder #1 is overlaid on the engine acoustic waveform to confirm the combustion peak of cylinder# 1, which is relatively indistinct in the acoustic trace.

In the associated power spectrum, shown in Figure 5, four peaks can be seen distinctly: the first at twice the frequency of revolution (33.4 Hz), the second at four times the frequency of revolution (66.8 Hz), the third at 100 Hz, and the fourth at 470 Hz. The amplitudes of any higher harmonics can be ignored because they contain considerably less energy than the first four leading terms.

Each cylinder experiences fuel injection and combustion once for every two complete revolutions of the crankshaft. Thus, the number of combustions per single revolution of the camshaft will be equal to (number of cylinders)/2. Here there are four cylinders, so there will be two combustion events during each complete revolution of the camshaft and the corresponding noise peak will occur at twice the fundamental frequency (2 × 16.7 ≈ 33 Hz). As would be expected, the amplitude of this peak is highly dependent on the combustion conditions. The second peak (67 Hz) is probably due to the closing knock of the valves, which occurs twice per crankshaft revolution, i.e. every second revolution in each cylinder.

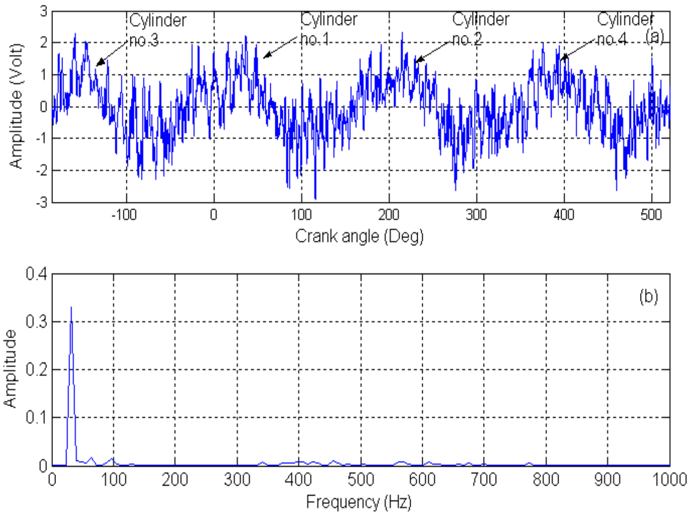

Fig 5. Acoustic waveform power spectrum at 1000 rpm and no load

VI. EFFECTS OF LOAD AND SPEED ON THE ACOUSTIC SIGNAL

To examine the effects of load and speed on the acoustic waveforms, the engine load was varied between 20, 40, and 60 Nm and the speed was increased from 1000 to 1500 and then to 2000 rpm. The measured acoustic waveforms and their associated power spectra are presented in Figures 6 and 7. As engine load was increased, the amplitude of the first peak increased slightly, while the increase in the amplitude of the second was more pronounced (Figure 6). The third peak at 100 Hz, corresponding to the third harmonic of the firing frequency, also increased in amplitude at higher loads.

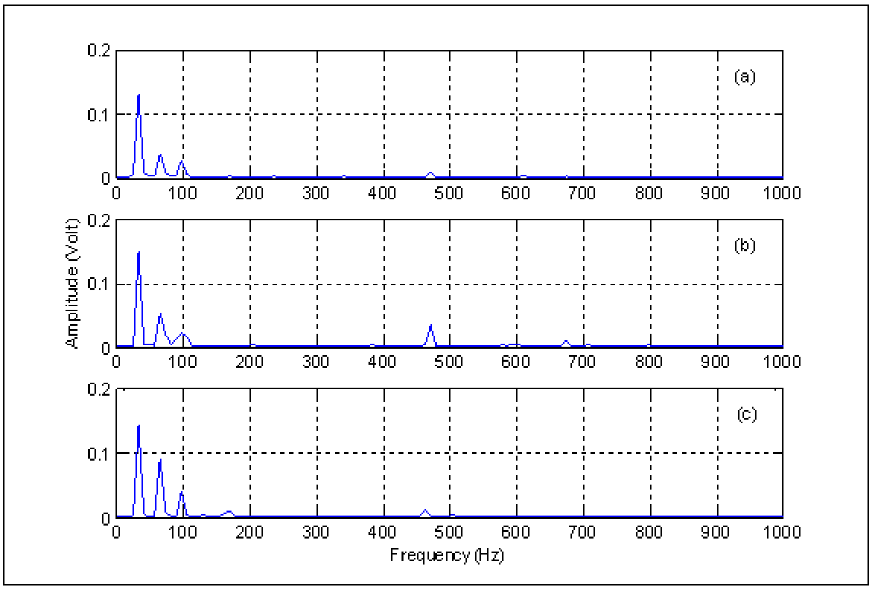

Fig 6. Acoustic waveform power spectra of diesel engine at loads of (a) 20 Nm, (b) 40 Nm, (c) 60 Nm

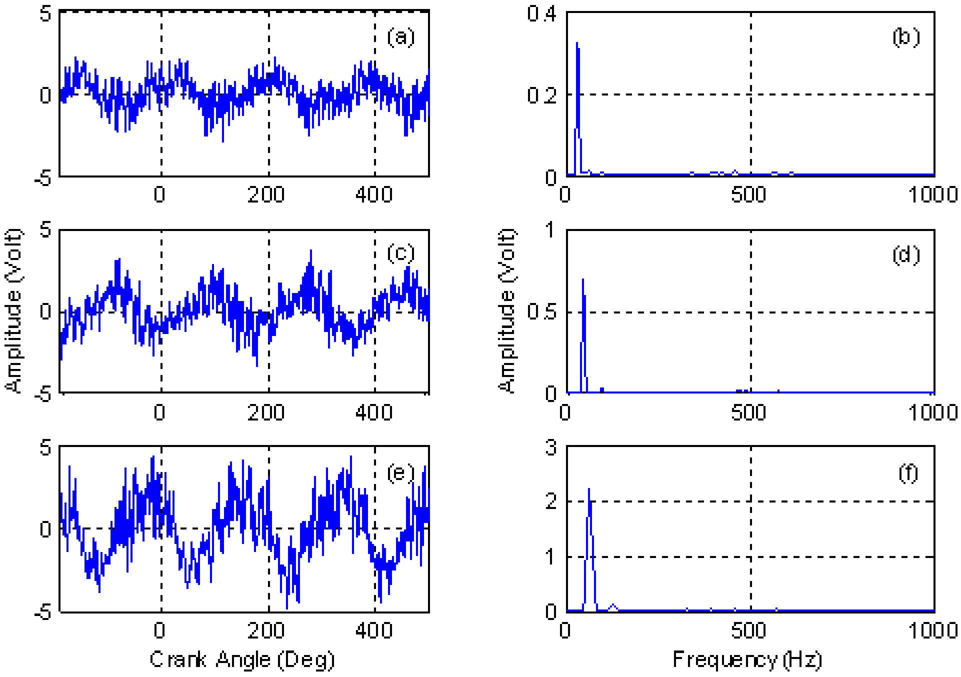

Fig 7. (a, c, e) Acoustic spectra in time domain; (b, d, f) power spectra in frequency domain at engine speeds of (a, b) 1000 rpm, (c, d) 1500 rpm, (e, f) 2000 rpm

Figure 7 shows the time domain acoustic spectrum at each of the three speeds with the associated power spectrum in the frequency domain. It can be observed that the amplitudes of the time domain waveforms increased as the speed increased. In the frequency domain, the amplitude of the peaks increased with increasing speed, and the frequency at which the peak occurred also increased in direct proportion to the increase in speed. To gain a better understanding of the speed effects, Figure 8(a) shows the superimposed signals for two speeds, 1000 and 2000 rpm. The corresponding acoustic frequency domain representations are shown in Figures 8(b) and (c) respectively.

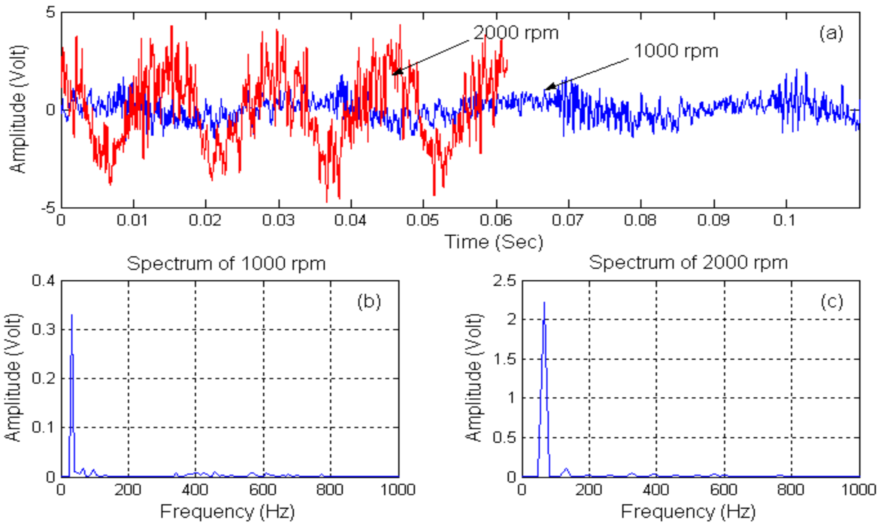

Fig 8. Time and frequency domain analysis of acoustic signals from engine running at 1000 rpm and 2000 rpm.

Figure 8(a) shows the time domain analysis of the acoustic signals at the two speeds, for two complete revolutions (720o) of the crankshaft: about 0.11 seconds and 0.55 seconds duration respectively. The difference in waveform amplitude is obvious: At 2000 rpm it was more than double that at 1000 rpm. This confirms that engine speed had a substantial effect on the measured sound levels. It can also be seen from Figure 8 that varying the speed did not affect the distinctive cyclical character of the acoustic waveform.

VII. FAULT DETECTION IN THE FREQUENCY DOMAIN

The time domain method is useful for reciprocating machines such as diesel engines, because it shows particular timed events such as firing order, but it cannot provide good condition monitoring information. Frequency domain analysis was therefore conducted on the acoustic data collected under two operational load conditions: at 0 and 60 Nm. The engine was operated at a constant speed of 1000 rpm and the exhaust valve clearance was changed from 0.4 mm (healthy condition) to 0.0 mm (faulty condition). Figure 9 shows acoustic signals in the frequency domain for healthy (a, c) and faulty (b, d) conditions, at 0 Nm (a, b) and 60 Nm (c, d).

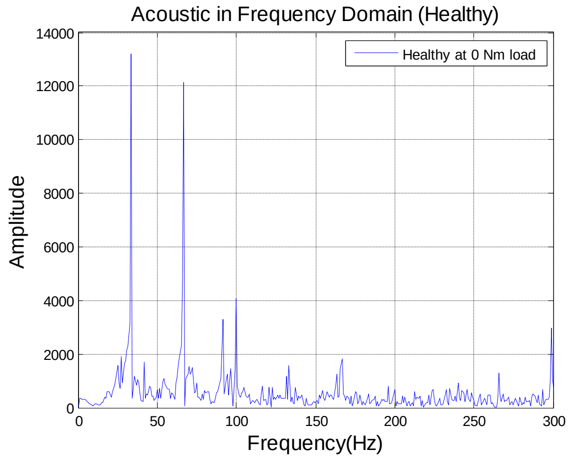

Fig 9(a). Acoustic signals in the frequency domain for the healthy condition at 0 Nm

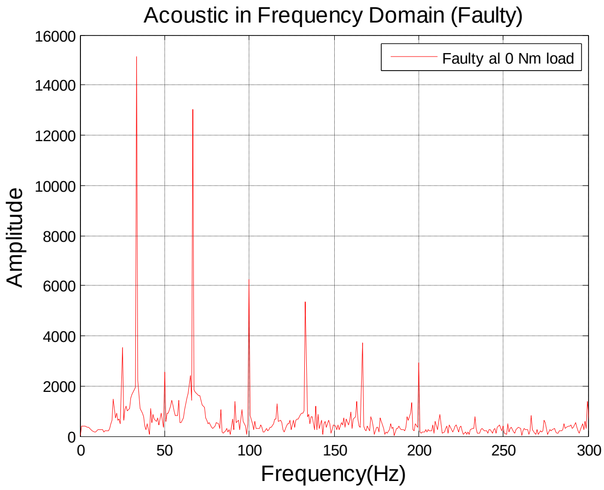

Fig 9(b). Acoustic signals in the frequency domain for the faulty condition at 0 Nm

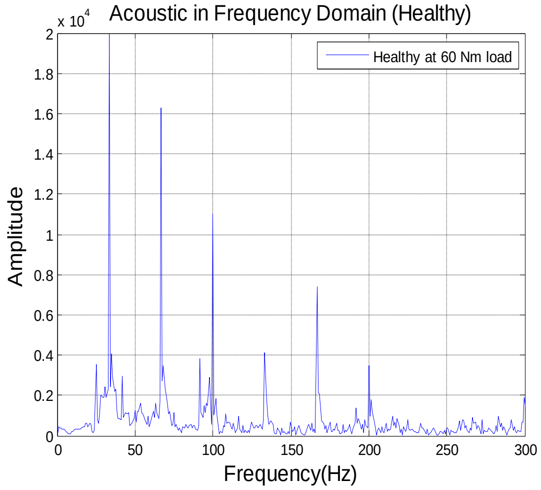

Fig 9(c). Acoustic signals in the frequency domain for the healthy condition at 60 Nm

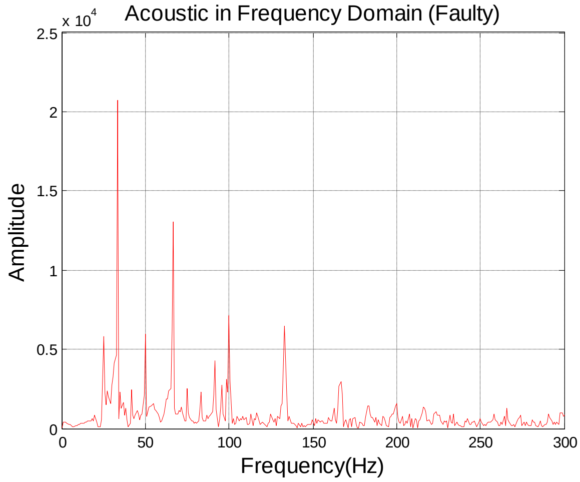

Fig 9(d). Acoustic signals in the frequency domain for the faulty condition at 60 Nm

Figure 9 shows that at zero load the amplitudes of the first and second harmonics were 13000 and 12000 Pa respectively in the healthy condition, but 15000 and 13000 Pa respectively in the faulty condition, while under load the corresponding values were 20000 and 16000 Pa in the healthy condition, but 21000 and 13500 Pa in the faulty case. These differences result from changing the engine load and the valve clearance settings. The appearance of the sidebands is due to the incorrect valve clearance in cylinder #1, which caused abnormal combustion because of the advanced opening and late closing of the exhaust valve.

VIII. FAULT DETECTION BASED ON STATISTICAL PARAMETERS

RMS values of the amplitude of acoustic data signals were also analyzed to give an indication of faults. The engine was tested under four loads (0, 20, 40, and 60 Nm) and four valve clearances: 0.4 mm (healthy condition), 0.0 mm, 0.25 mm, and 0.6 mm. The RMS values of the measured acoustic signals under each condition are shown in Figure 10.

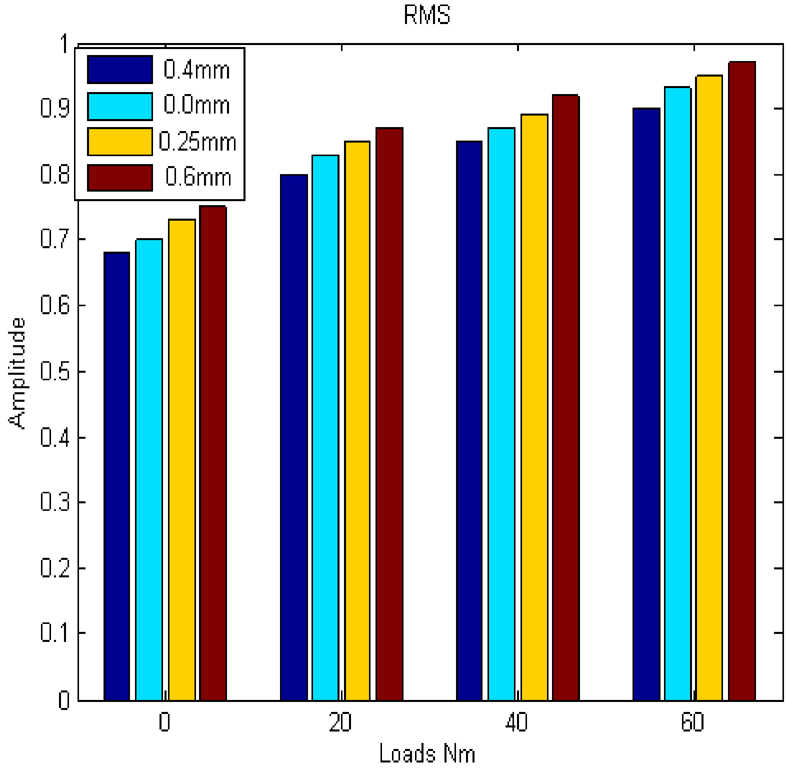

Fig 10. RMS amplitudes of the acoustic signal in four load and four valve clearance conditions

Figure 10 shows that the RMS values gave a direct indication of exhaust valve timing and clearance setting at all loads. It was noted that any deviation from the healthy valve clearance setting would cause an increase in amplitude of engine noise, because changing the valve clearance disrupts the valve timing, thus impairing the combustion process and affecting the acoustic signals emitted.

The kurtosis and skewness of the acoustic signals measured under the same load conditions and exhaust valve clearances were also calculated, as plotted in Figures 11(a) and 11(b). It was noted that as load increased, the kurtosis increased and the skewness decreased; however, it is difficult to infer fault information from these parameters, as in all cases the healthy values lie within range of unhealthy values.

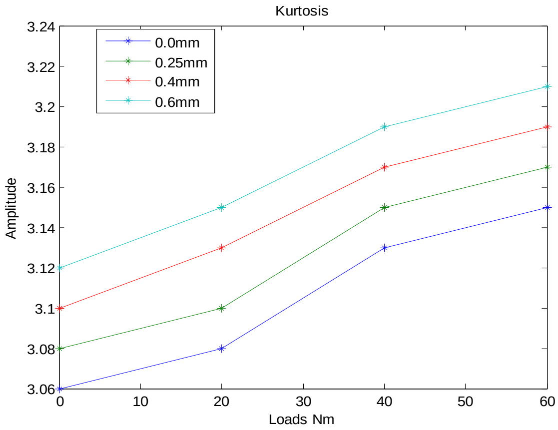

Fig 11(a). Kurtosis of the acoustic signal in four load and four valve clearance conditions

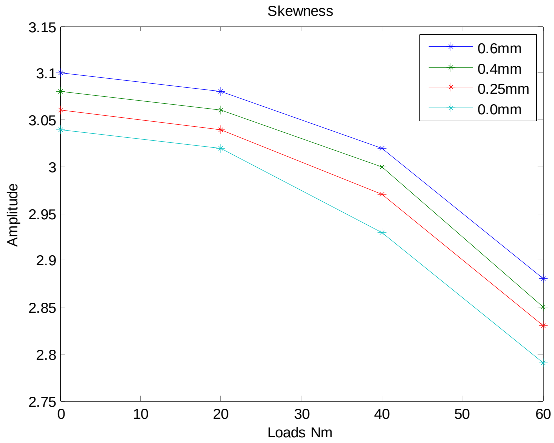

Fig 11(b). Skewness of the acoustic signal in four load and four valve clearance conditions

IX. CONCLUSION

This paper has reported the main time and frequency domain characteristics of diesel engine acoustic signals and has demonstrated a basis for fault detection using the RMS values of these signals. The main drawback of the time domain analysis is that it gives limited information. The frequency domain analysis gives only some information about the frequency components of the measured signals and in the case of diesel engine acoustic condition monitoring these frequency components are dominated by the firing frequencies of the engine. Calculating the RMS values for the measured acoustic signals may give a quick and accurate indication of the presence of abnormalities or deviations from the healthy condition.

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