DIESEL KNOCK COMBUSTION AND ITS DETECTION USING ACOUSTIC EMISSION

DAVID P. LOWE¹, TIAN RAN LIN¹,², WEILIANG WU¹ and ANDY C. C. TAN¹
¹ School of Engineering Systems, Queensland University of Technology, GPO Box 2434, Brisbane, QLD 4001, Australia, ² CRC for Infrastructure and Engineering Asset Management

Abstract

This paper presents an experimental investigation into the detection of excessive diesel knock using acoustic emission (AE) signals. Three different dual-fuel diesel engine operating regimes were induced into a compression-ignition (diesel) engine operating on both straight diesel fuel and two different mixtures of fumigated ethanol and diesel. The experimentally induced engine operating regimes were: normal, or diesel only operation, acceptable dual-fuel operation and dual-fuel operation with excessive diesel knock. During the excessive diesel-knock operating regime, high rates of ethanol substitution induced potentially damaging levels of diesel knock. AE data was captured along with cylinder pressure, crank-angle encoder, and top-dead centre signals for the different engine operating regimes. Using these signals, it was found that AE signals clearly distinguished between the two acceptable operating regimes and the operating regime experiencing excessive diesel knock. It was also found that AE sensor position is critical. The AE sensor positioned on the block of the engine clearly related information concerning the level of diesel knock occurring in the engine whilst the sensor positioned on the head of the engine gave no indication concerning diesel knock severity levels.

Keywords: Diesel knock, dual-fuel, condition monitoring

1. Introduction

It is the high pressure-rise rates associated with the auto-ignition of fuel during the premixed combustion stage that produces the characteristic “knocking” noise widely associated with diesel engines. This noise is often referred to as diesel knock [1] or combustion roughness [2]. When excessive, diesel knock results in the propagation of high amplitude pressure waves at frequencies governed by combustion chamber resonance and as is the case with spark-ignition knock, can be extremely detrimental to engine life [2, 3]. Diesel knock has not been a fundamentally limiting factor in the same manner as spark-ignition knock in terms of engine design, however, diesel knock is recognized as a considerable problem associated with the use of alternative fuels in dual-fuel type diesel engines [4]. The increasing use of bio-fuels and the performance limitations associated with excessive diesel knock in dual-fuel engines combine to make diesel knock an important parameter to monitor from both, engine performance, and health viewpoints.

Knock detection in spark-ignition engines is undertaken using both piezoelectric accelerometers and spark plug ion-current sensors. Spark plug ion-current sensors detect knock using an applied spark plug gap voltage whilst piezoelectric accelerometers detect knock by sensing engine vibrations excited by the knock phenomena in a frequency range of 4-8 kHz [5]. Although effective for spark-ignition knock detection, the increased vibration levels associated with normal diesel engine operation result in unreliable excessive diesel knock detection using piezoelectric accelerometers.
Being fundamentally related to cylinder pressure, it is no surprise that cylinder pressure sensors are the most effective for detecting diesel knock. However, pressure sensors are not commonly used in production engines as engine designs having extra recesses into individual combustion chambers are considered undesirable [5]. Acoustic emission (AE) sensors, however, have the potential to monitor multiple cylinders, are non-intrusive [6], and are easily installed and AE based monitoring methods have been successfully demonstrated in many diesel engine monitoring applications. For example, Fog et al. [6 - 8] detail the detection of large marine diesel engine exhaust valve faults as well as misfire using AE signals. Other investigations have demonstrated AE based methods for the detection of injector faults [9] and for monitoring the piston ring and cylinder liner interface [10 - 13]. El-Ghamry et al. [14] have also shown that AE signals contain information relating to combustion. This work demonstrates an indirect method of cylinder pressure measurement using AE signals.

Section 2, provides a concise overview of the cause of diesel knock and the closely related phenomena of combustion chamber resonance. In addition, Section 2 describes the methods used to quantify knock severity. Section 3 briefly describes the test rig and data acquisition systems used during the experimental investigation. Section 4 presents and discusses the results from the investigation and finally, Section 5 provides a summary of the main findings from this investigation.

2. The Diesel Knock Phenomena

Knock, pinging or detonation are all terms that have been widely used to describe the characteristic “metallic rattling” noise associated with abnormal combustion in spark-ignition engines. Spark-ignition knock is caused by the spontaneous ignition of gas ahead of the propagating flame front (the end gas) within the combustion chamber. This spontaneous ignition results in a rapid release of chemical energy and an accompanying rapid rise in cylinder pressure [15]. Unlike spark-ignition knock, diesel knock occurs when injected fuel auto-ignites and combusts in the premixed stage of combustion. Whilst this process is a normal part of diesel engine operation, various circumstances can lead to excess quantities of fuel combusting in a premixed fashion. This situation often develops if the parameters governing combustion lead to abnormally long ignition delay periods. As a consequence, excessive diesel knock can often be a symptom of underlying faults such as poor or contaminated fuels, injection system problems or unsuitable rates of alternative fuel substitution.

The rapid pressure increases associated with both spark-ignition and diesel knock result in the propagation of high amplitude pressure waves at frequencies governed by combustion chamber resonance. The combustion-chamber resonance frequencies are in turn determined by the geometry and the velocity of sound within the combustion chamber [3]. Figure 1 highlights the effects of spark-ignition knock on cylinder pressure. As seen, the knock phenomenon causes high frequency pressure fluctuations, which are recorded by the pressure sensor. It is also seen that the amplitude of the high frequency pressure fluctuations increases as the severity of the knock increases [15].

Whilst discussing spark-ignition knock, Heywood [15] points out that due to the highly variable nature of engine knock, fundamental definitions of knock intensity is extremely difficult to make. However, a method whereby cylinder pressure signals are used to calculate an average pressure rise rate (PRR) is described. Another method, also described by Heywood [15], uses
pressure signals, which are filtered to remove the low frequency components. The maximum amplitude of the knock-induced pressure oscillation is then used as the knock intensity measure.

Fig. 1: Cylinder pressure vs. crank angle plots for (a) normal spark-ignition combustion, (b) light knock and (c) intense knock [15].

Whilst discussing diesel knock specifically, Hsu [2] mentions that maximum pressure rise rate (MPRR) has been used to quantify knock intensity when the frequency response of pressure measurement system is such that the system is unable to sense the individual pressure oscillations. Hsu [2] also points out that MPRR measurements can be misleading when modern instrumentation is used as peak MPRR values may not be associated with premixed combustion. Hsu [2] suggests that the best way to quantify diesel knock is to measure the combustion pressure wave energy at the characteristic frequency.

In addition to the previously mentioned measurement techniques, measurements using rates of heat release, such as those shown by Shiga et al. [16], can also provide insight regarding the relationship between the rapid releases of chemical energy, the accompanying rapid rise in cylinder pressure and the knock severity levels.

3. Combustion Chamber Resonance

As discussed, knock phenomena lead to the propagation of high amplitude pressure waves at frequencies governed by combustion chamber resonance. Therefore, the determination of the resonant frequencies is an important step in the quantification of diesel knock.

The pressure fluctuations within the combustion chamber are acoustic in nature as their amplitude is small compared to the mean in-cylinder pressure at the time they occur [17] and the geometry of a direct injection-type combustion chamber can be approximated as cylindrical cavity having plane ends [3]. The combustion chamber resonance frequencies can be calculated using equation (1).

\[ f_R = c \left[ \left( \frac{\alpha_{mn}}{B} \right)^2 + \left( \frac{p}{2L} \right)^2 \right]^{1/2} \]  \hspace{1cm} (1)

where \( m, n, \) and \( p \) denote circumferential, radial and axial mode numbers, respectively, and \( B \) is the bore diameter. \( L \) is the axial length and the value \( \alpha_{mn} \) is determined using the Bessel function. The speed of sound \( (c) \) is determined using the following thermodynamic relationship with \( R \) being the gas constant, \( T \) is the temperature and \( \gamma \) is the ratio of specific heats.

\[ c = \sqrt{\gamma RT} \]  \hspace{1cm} (2)
Although axial, circumferential and radial modes are associated with cylindrical cavity resonance, the axial modes can be excluded in terms of knock intensity as the frequencies associated with this mode are generally above the audible range [18]. This is due to the small axial dimensions when the piston is close to top-dead centre (TDC), and the high speed of sound at combustion temperatures [19]. When the axial term in equation (1) is set to zero \((p = 0)\), equation (1) can be rewritten as equation (3):

\[
f_{m,n} = (c/B)\alpha_{m,n}
\]  

Equation (3) was used to calculate the first four transverse (radial and circumferential) mode frequencies. Based upon other bulk gas values detailed previously [18, 20], a bulk combustion temperature estimate of 2000 K was used. The calculated resonant frequencies are listed in Fig. 2 along with graphical representations of the corresponding modes.

![Diagram of modes](image)

**Mode** | **\(f_{1,0}\)** | **\(f_{2,0}\)** | **\(f_{0,1}\)** | **\(f_{3,0}\)**
--- | --- | --- | --- | ---
**Calculated frequency (Hz)** | 5151.0 | 8544.3 | 10719.5 | 11753.0

Fig. 2: The first four transverse modes with corresponding calculated resonant frequencies.

Fundamental frequency \((f_{1,0})\) values for the test engine were determined from the recorded pressure signals by spectral analysis using fast Fourier transforms. Measured fundamental frequency \((f_{1,0})\) values ranged between 5100 - 5700 Hz.

It was found that the majority of the signal energy associated with cylinder resonance was contained within the lowest \((f_{1,0})\) mode. This finding has also been noted by other researchers such as Eng [20]. Once a fundamental frequency range directly relating to the level of diesel knock was determined, pressure signals were then used to quantify diesel knock levels.

4. Experimental Methodology

The test facility used in this experimental investigation consisted of a 5.9 litre, six-cylinder turbo-charged Cummins diesel engine coupled to a Froude water dynamometer. The engine and dynamometer are controlled electronically via a Dynalog control system. The test facility is shown in Fig. 3 and the relevant engine specifications are listed in Table 1.

A PAC MicroDisp AE system was used to record AE signals from two PAC 15\(\alpha\) sensors mounted on both engine block and head in close proximity to cylinder 1. These signals were amplified using PAC 2/4/6 AE pre-amplifiers. Figure 4 shows the position of the AE sensors on the front face of the engine. In addition to the AE signals, TDC and crank angle encoder
Table 1: Relevant Cummins diesel engine specifications.

<table>
<thead>
<tr>
<th>Engine Cycle</th>
<th>Four Stroke, Turbocharged</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinders</td>
<td>Six (Inline)</td>
</tr>
<tr>
<td>Firing Sequence</td>
<td>1, 5, 3, 6, 2, 4</td>
</tr>
<tr>
<td>Bore</td>
<td>102 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>120 mm</td>
</tr>
<tr>
<td>Displacement</td>
<td>5.9 Litres</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>17.3:1</td>
</tr>
<tr>
<td>Combustion System</td>
<td>Direct Injection</td>
</tr>
</tbody>
</table>

Fig. 3: Photograph showing the test facility.

Fig. 4: AE sensor positions.
signals were recorded simultaneously. A National Instruments data acquisition system was used to record cylinder pressure, TDC and crank angle encoder signals, using LabView software. The cylinder pressure was recorded using a high-temperature Kistler pressure transducer installed on cylinder 1.

Three different engine-running conditions were induced during the experimental investigation. These running conditions were based on the rate of ethanol substitution. The rates used were: 0% ethanol (diesel fuel only), 30% ethanol and 50% ethanol by energy, respectively. These three engine-running conditions were chosen to represent three different operational regimes. The 0% ethanol (e0) running condition is representative of normal diesel engine operation. The 30% ethanol case (e30) represents acceptable dual-fuel diesel engine operation whilst the 50% ethanol (e50) case represents a case of unacceptable dual-fuel diesel knock. The ethanol was introduced into the engine by injecting the ethanol into the incoming air stream. This substitution technique is widely known as ethanol fumigation. The tests were all undertaken using an applied torque of 700 Nm at an engine speed of 2000 rpm. This loading condition represents approximately 90% of the maximum load.

The diesel knock levels induced during this experimental investigation were quantified using two techniques. The first technique involved the calculation of average amplitudes of pressure oscillation directly from recorded pressure signals by averaging the five highest amplitudes of pressure oscillation within an individual combustion cycle. The second technique involved the decomposition of the combustion window portion of the pressure signal using a three-level discrete wavelet decomposition. The wavelet decomposition was performed using a Daubechies (Db2) wavelet. The results from this signal decomposition were used to calculate the energy levels of the frequency bands associated with the different frequency modes. The results from this signal decomposition are detailed in the following section.

4. Results and Discussion

Figure 5 shows typical pressure traces for the three different ethanol substitution rates. These pressure traces are plotted with respect to crank angle. The pressure traces shown correspond to the final 60° of the compression stroke and the first 100° of the power stroke. The window encompassed by the dashed green outline highlights the portion of the signal where premixed combustion occurs and where the resonant pressure fluctuations are at a maximum. This window, referred to as the “combustion window” in the following discussion, is shown in Fig. 6. The combustion window extends from the combustion TDC to 25° after combustion TDC.

The dashed oval shown in Fig. 6 highlights the distinctive high amplitude pressure fluctuations associated with the e50 pressure curve. These pressure fluctuations are characteristic of excessive diesel knock. Acceptable dual-fuel combustion is shown by the e30 pressure curve and although different to the normal diesel (e0) curve in terms of underlying signal pressure form, the high amplitude pressure fluctuations characteristic of excessive diesel knock are absent.

Comparison of the amplitudes of the high frequency pressure fluctuations from each of the three different engine operating regimes was undertaken by calculating average pressure fluctuation amplitudes for each operational regime. The average pressure fluctuation amplitude values were 187, 139 and 587 kPa for e0, e30 and e50 operating regime, respectively. These values are shown graphically in Fig. 7. The average pressure fluctuation amplitude for the e50 case represents over three times increase from the normal regime.
Fig. 5: Cylinder pressure traces for the three different ethanol substitution rates.

Fig. 6: Close-up view of the combustion window portion of the pressure curves for the three different ethanol substitution rates.

Fig. 7: Average cylinder pressure fluctuation amplitudes.
A second diesel knock level technique was also used. This technique involved performing a three level discrete wavelet decomposition of the pressure signals. The results of this decomposition were used to calculate the combustion window energy associated with specific frequency ranges for the different operating regimes. The combustion window energy content for the 3.125 to 6.25 kHz frequency range showed a similar trend in terms of diesel knock severity to that shown in Fig. 7.

Factors such as engine block geometry, cooling system galleries and gaskets all drastically effect the transmission of AE [21]. In order to determine the most effective sensor position for diesel knock detection, AE sensors were attached to both the engine block and head in close proximity to cylinder 1. The AE signals were then truncated using the combustion window described previously. The resultant signals were used to calculate RMS energy (RMS). As shown in Fig. 8, the total combustion window AE energy (RMS) levels from the engine head for the e0, e30 and e50 cases are all similar. It is clear that the total energy (RMS) values calculated from the engine head show no clear indication regarding the unacceptable knock experienced during the e50 operating regime. Figure 9 shows the combustion window AE energy (RMS) calculated from the engine block. A sharp increase in the AE RMS level for the e50 case commences at approximately 10° after TDC. The increase in AE RMS energy during the e50 case corresponds to the start of the high amplitude pressure fluctuations highlighted in Fig. 6. This increase in AE RMS gives a clear indication that excessive diesel knock is occurring.

Figure 10 shows a trend comparison between the two techniques used to quantify the diesel knock level and the total energy (RMS) values for the engine block AE signal. The various values have been normalized in terms of the normal operation (e0) case. This figure shows that the total AE signal energy (RMS) from the engine block sensor gives a good indication regarding the level of diesel knock as indicated by the two diesel knock quantification techniques. The increased AE energy (RMS) detected during the e50 regime appears directly related to the transverse (circumferential and radial) wave propagation modes associated with diesel knock.

5. Conclusion

Three different diesel engine operating regimes were induced into a (diesel) engine operating on both straight diesel fuel and two different mixtures of fumigated ethanol and diesel. Compar-
Fig. 9: Combustion window AE RMS from the engine block.

Fig. 10: A trend comparison between the Diesel knock level quantification techniques and the total AE energy (RMS) values from the engine block.

The detection of the excessive diesel knock operational regime was possible using AE signals recorded from a sensor located on the engine block. A sensor located on the head of the engine, however, gave no indication regarding the unacceptably high levels of diesel knock.

The findings show that AE recorded from the block contains information concerning the propagation of high amplitude pressure waves at the combustion chamber resonant frequencies. This AE activity originates as a result of the interaction between the pressure oscillations and the combustion related components of the engine.
Acknowledgements

This paper was developed within the CRC for Infrastructure and Engineering Asset Management, established and supported under the Australian Government's Cooperative Research Centre Program. The authors gratefully acknowledge the financial support provided by the CRC.

References