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# Combustion noise radiation during the acceleration of a turbocharged diesel engine operating with biodiesel or *n*-butanol diesel fuel blends

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**Abstract**

In the current study, experimental tests were conducted on a turbocharged truck diesel engine in order to investigate the mechanism of combustion noise radiation during various accelerations and for various fuel blends. With this aim, a fully instrumented test bed was set up in order to capture the development of key engine and turbocharger parameters. Apart from the baseline diesel fuel, the engine was operated with a blend of diesel with either 30 vol % biodiesel or 25 vol % *n*-butanol. Analytical diagrams are provided to explain the behaviour of combustion noise radiation in conjunction with the cylinder pressure, the pressure rise rates, the frequency spectrum and the turbocharger and governor–fuel pump responses. The blend of diesel fuel with *n*-butanol exhibited the highest noise emissions throughout each of the transient tests examined, with differences up to 4 dBA from those with neat diesel operation. On the other hand, the biodiesel blend was found to behave marginally noisier than neat diesel oil but without a clear trend established throughout the transient events.

**Keywords**

Turbocharged diesel engine, combustion noise, acceleration, biodiesel, *n*-butanol

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## Introduction

During the last few decades, great effort has been placed on the research into the development of the diesel engine, with considerable success being achieved; the compression ignition engine is nowadays the preferred prime mover in the transport sector and in medium and medium–large industrial and marine applications, as well as an important competitor in the automotive market. The most attractive features of the diesel engine are its reliability, its very good fuel efficiency as well as its ability to operate as a turbocharged engine, a fact that further enhances its torque and its fuel efficiency. Consequently, vehicles with diesel engines achieve much lower fuel consumption and reduced carbon dioxide ($CO_2$) emissions than their similarly rated spark ignition counterparts over the entire engine operating range and vehicle lifetime. From an acoustic point of view, however, as the diesel engine is a very complex system consisting of various dynamic forces acting on a structure of varying stiffness, damping and response characteristics, it remains far inferior to the gasoline engine.

The three primary sources of noise generation in a diesel engine are as follows: gas flow, mechanical processes and combustion.[1–3] Gas-flow noise, usually low frequency controlled, is associated with the intake and exhaust processes, including turbocharging and the cooling fan. Mechanical noise consists of contributions from both rotating and reciprocating engine components; it originates from the inertia forces causing piston slap, and from gears, tappets, valvetrains, timing

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drives, fuel injection equipment and bearings. The mechanism behind the third source of noise, namely combustion noise, lies in the (high) rate of cylinder pressure rise, mainly after the ignition delay period, which causes discontinuity in the cylinder pressure frequency spectrum and an increase in the level of the high-frequency region, resulting in vibration of the engine block and, ultimately, in combustion noise radiation (the characteristic diesel combustion 'knock'). This combustion noise radiation manifests itself in the domain from a few hundred hertz up to a few thousand hertz. Although noise originating from mechanical and gas-flow sources is encountered during gasoline engine operation too, problems associated with combustion are primarily restricted to diesel engines, with their spark ignition counterparts behaving in a noisy way only when abnormal combustion (detonation) is experienced.

It has long been acknowledged that combustion chamber design and injection parameters, e.g. the timing and the amount and rate of fuel injected, play decisive roles in combustion noise emission by defining the exact rate of heat release.[1,4] In order to analyse this source of noise, the cylinder pressure signal is usually examined in the frequency spectrum, e.g. using Fourier transform.[1,5]

Anderton[6] and co-workers pioneered diesel engine noise research by investigating the effect of various parameters on the noise generated during steady-state conditions such as two-stroke versus four-stroke operation, naturally aspirated versus turbocharged engines and cylinder configuration. Recent studies on steady-state (combustion) noise emissions have focused on the effects of the injection pressure, profile and rate. Electronically controlled common-rail injection systems that split the injection event into one pilot injection and one main injection can prove beneficial by facilitating the physical preparation of the air–fuel mixture, thus reducing premixed combustion and limiting (combustion) noise radiation.[7,8] On the other hand, the emergence of new promising diesel combustion technologies, such as low-temperature combustion and premixed charge compression ignition, are based on lower cycle temperatures, e.g. using very high exhaust gas recirculation (EGR) rates, in order to limit soot and nitrogen oxide ($NO_x$) emissions simultaneously;[9,10] these technologies are expected to have a detrimental effect on combustion noise owing to the higher portion of premixed combustion that the lower cycle temperatures induce.[11]

Combustion noise development during a transient event differs to a large extent from that in the respective steady-state operation; this was the result reached by the surprisingly few studies carried out so far.[12–17] The main finding made by Head and Wake[13] was that combustion noise is generally higher during acceleration, typically of the order of 4–7 dBA, compared with the respective steady-state conditions of a naturally aspirated engine. They argued that this increase was mainly due to the lower cylinder wall temperature during the first few cycles of the transient event, which lowered the evaporation rate and increased the ignition delay period. Similar results were obtained by Rust and Thien,[14] who also extended the analysis to load acceptance transients, again for naturally aspirated diesel engine operation. The work by Dhaenens et al.[15] is one of the two known references, apart from the current research group, to have focused on noise development from a *turbocharged* diesel engine acceleration; their investigation revealed that transient overall engine noise exceeded steady-state levels by 5 dBA maximum (measured at 1 m distance from the engine surface), while it was also characterized by a broadband level increase combined with amplified resonance effects. The present research group has recently published an investigation on a turbocharged truck diesel engine focusing on all three major transient cases (acceleration, load increase and cold starting) experienced in daily driving conditions, confirming the previous results.[17] On the other hand, Payri et al.[16] proposed and validated a new method to assess the noise level in direct-injection (DI) diesel engines. Based on the selection of operation parameters that indirectly affect the combustion process and combustion components that characterize the in-cylinder noise source, two significant indicators that allow prediction of the overall noise level were identified. The engine speed was the operation parameter that was found to correlate best with noise, whereas the maximum in-cylinder pressure derivative was identified as the most relevant parameter for the combustion process.

In parallel, during the last few decades, considerable attention has been paid to alternative fuel sources, mainly initiated by the depleting crude oil reserves and their growing prices; the main emphasis is on biofuels that possess the added advantage of being renewable, thus showing an ad hoc advantage in reducing $CO_2$ emissions.[18] Biofuels made from agricultural products (oxygenated by nature) reduce the world's dependence on oil imports, support local agricultural industries and enhance farming incomes. Among those, vegetable oils and mainly their derived biodiesels (methyl esters or ethyl esters) are considered to be very promising fuels.

Experimental studies of pollutant emission measurements, during steady-state[19–22] and transient[23–28] operation, when using biodiesel blends, have appeared in numerous published papers during the last years. The vast majority of these investigations have reported decreases in smoke, particulate matter, carbon monoxide and unburned hydrocarbons and (moderate) increases in $NO_x$ emissions with increasing biodiesel percentage in the fuel blend.

On the other hand, only a handful of studies are available regarding the biodiesel effects on noise radiation, all limited to steady-state engine operation and with contradicting results. Nabi et al.[29] reported *overall* noise reduction for a naturally aspirated engine operating at high loads, ranging from 1 dBA for a B10 blend up to 2.5 dBA for neat karanja biodiesel (B100). Although the reduction was, in a rather simplistic manner, attributed to the oxygen content of the biofuel, it was most probably the higher cetane number of biodiesel that was responsible for a shortening of the ignition delay period that ultimately reduced the cylinder pressure rise rate and the combustion noise contribution to the total emitted noise. On the other hand, Haik et al.[30] found that algae oil methyl ester was responsible for higher cylinder pressure rise rates $dp/d\varphi$ (a typical 'surrogate' property of combustion noise) compared with neat diesel fuel or raw algae oil for their indirect-injection diesel engine. Bunce et al.[31] proposed that a re-adjustment of the engine calibration is required in order to attain decreased pollutant and noise emissions simultaneously with the addition of soy biodiesel in the fuel blend. Although Anand et al.[32] did not specifically measure combustion noise, they too reported on the cylinder pressure rise rate for neat karanja biodiesel fuel blends and concluded that the trend versus neat diesel fuel was not clear; in fact, the contributing factors acted in favour or against each fuel blend depending on the speed or load conditions. In all the previous studies, the different fuel properties (the cetane number, the calorific value, the molecular composition and structure and the heat of vaporization) of diesel fuel and biodiesel were identified as key contributors for any differences observed.

Apart from biodiesel, a very challenging competitor for use as fuel in compression ignition engines is ethanol and, better still, *n*-butanol; surprisingly, neither has been extensively tested in diesel engines. *n*-butanol is of particular interest as a renewable biofuel as it is less hydrophilic and it possesses a higher heating value, higher cetane number, lower vapour pressure and higher miscibility than ethanol does; this makes it preferable to ethanol for blending with conventional diesel fuel.

The literature concerning the use of diesel–*n*-butanol fuel blends in diesel engines and their effects on the performance and (exhaust) emissions is extremely limited, a fact that constitutes another major aspect of the originality of the present work. An early study by Yoshimoto and Onodera[33] dealt with the performance and exhaust emission characteristics of a diesel engine fuelled with vegetable oils blended with oxygenated organic compounds, including ethanol and *n*-butanol. Miers et al.[34] extended the investigation to a driving cycle analysis for a light-duty turbocharged diesel-engined vehicle. The present research group recently published the results of an experimental investigation on the same engine as that under study in this work during steady-state operation;[35] these revealed the beneficial effects of using various blends of *n*-butanol with normal diesel fuel on the smoke and CO emissions at the expense of hydrocarbon and $NO_x$ emissions. Similar results were obtained by Yao et al.[36] and Lujaji et al.[37] In any case, none of the above-mentioned investigations has studied the effect of oxygenated fuels on combustion noise radiation even for steady-state operation, let alone during transient conditions.

The targets of this study are to investigate the (almost unexplored so far) field of transient combustion noise radiation of turbocharged diesel engines when running on blends of diesel fuel with various biofuels and to shed light into the relevant phenomena and underlying mechanisms. Two challenging biofuels were chosen for the analysis (no modifications were made on the engine side when using these biofuels) as follows:

(a) diesel with 30 vol % biodiesel;
(b) diesel with 25 vol % *n*-butanol (normal butanol).

In order to accomplish our task, experimental tests were conducted on a medium-duty turbocharged and after-cooled DI truck diesel engine, applying a modern combustion noise meter for accurate cylinder pressure data analysis. Since the engine is of the automotive type, only acceleration transients were chosen for the investigation. It is believed that useful overall conclusions on transient combustion noise radiation can be deduced on the basis of the individual engine transient characteristics in conjunction with the composition and properties of the fuel blends. Moreover, the differing physical and chemical properties of biodiesel and *n*-butanol among themselves and against those of the diesel fuel form valuable data for the analysis and interpretation of the experimental findings.

## Description of the experimental installation

A general layout of the test bed installation, the instrumentation used and the data acquisition system is illustrated in Figure 1. A brief description of the individual components is provided in the following sections; a more detailed description of the experimental set-up has been given by Rakopoulos et al.[38]

### *Engine under study*

The engine used in this study is a Mercedes–Benz OM 366 LA turbocharged and after-cooled DI diesel engine. It is widely used to power mini-buses and small–medium trucks; its basic technical data are given in Table 1. Two notable features of the engine are, on the one hand, its retarded fuel injection timing in order to achieve low $NO_x$ emissions (as will be discussed later in the text, this has a strong influence on combustion noise radiation too) and, on the other hand, the fuel-limiter (cut-off) function in order to limit the exhaust smoke level during demanding conditions such as transients or low-speed high-load steady-state operation. The engine was coupled to a hydraulic dynamometer.

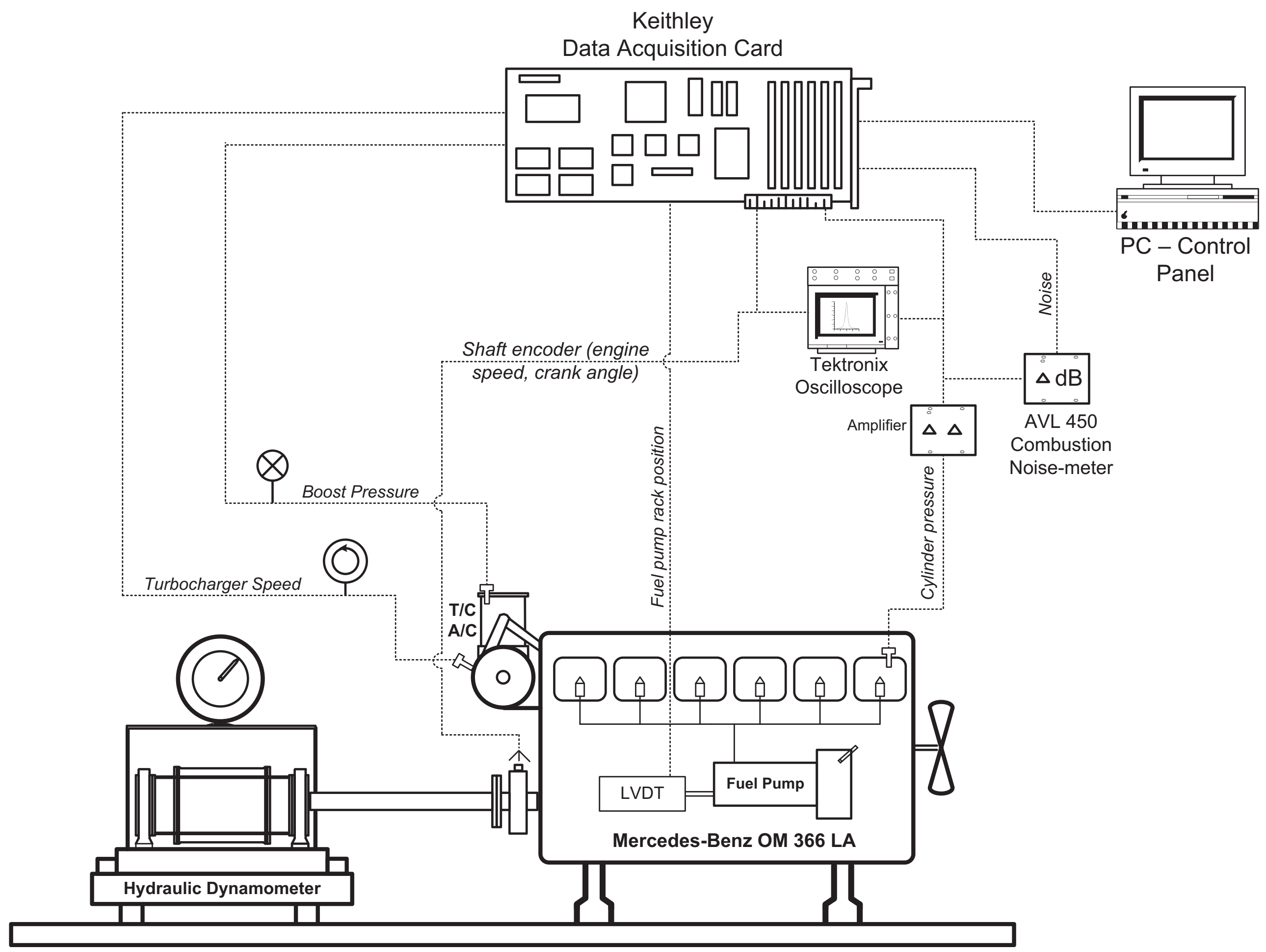


**Figure 1.** Schematic layout of the test bench installation.
PC: personal computer; T/C: turbocharger; A/C: after-cooling; LVDT: linear variable-differential transducer.

**Table 1.** Engine and turbocharger specifications.

| | |
|---|---|
| Engine model and type | Mercedes–Benz, OM 366 LA, six-cylinder, in-line, four-stroke, compression ignition, direct-injection, water -cooled, turbocharged, after-cooled, with bowl-in-piston |
| Speed range | 800–2600 rpm |
| Maximum power | 177 kW at 2600 rpm |
| Maximum torque | 840 Nm at 1250–1500 rpm |
| Engine total displacement | 5958 $cm^3$ |
| Bore | 97.5 mm |
| Stroke | 133 mm |
| Compression ratio | 18:1 |
| Fuel pump | Bosch PE-S series, in-line, six-cylinder with fuel limiter |
| Static injection timing | $5\pm1°$ crank angle before top dead centre (at full load) |
| Turbocharger model | Garrett TBP 418-1 with internal wastegate |
| Aftercooler | Air-to-air |

## *Combustion noise estimation*

Combustion noise estimation was achieved using the AVL 450 combustion noise meter. Its operating principle is based on the analysis of the cylinder indicator diagrams in the frequency domain, applying a series of filters to it.[39] Initially, the cylinder pressure signal passes through a U-filter, corresponding to the frequency attenuation of the engine block. This filter has been derived from the measurement of a large number of (mainly) truck diesel engines such as that under study; thus it is believed that it is representative of (combustion) noise characteristics for this type of engine. Afterwards, there is a possibility of filtering the combustion chamber resonance with selectable low-pass filters. Finally, the signal is guided through an A-filter that matches a standard value correction in

**Table 2.** Tabulation of the acceleration tests conducted.

| Test | Initial conditions | | Final conditions | |
|---|---|---|---|---|
| | Speed (rpm) | Load (%) | Speed (rpm) | Load (%) |
| 1 | 1000 | 10 | 1800 | 15 |
| 2 | 1500 | 10 | 2000 | 20 |
| 3 | 1500 | 40 | 2000 | 75 |

acoustics to the audible characteristics of the human ear (dBA). The produced output signal is further processed by r.m.s. (root mean square) conversion to logarithmic d.c. values that relate to the aural threshold. The total error of the meter is less than $\pm 1$ dBA.

### *Measurement of operating parameters and processing of the data*

The various engine and turbocharger operating parameters measured and recorded continuously were as follows: the engine speed; the cylinder pressure; the fuel pump rack position; the boost pressure; the turbocharger speed. The location of each measuring device on the experimental test bed installation is shown in Figure 1. A custom-made 'stop' with various adjustable positions, each corresponding to a specific engine speed, was fitted on the (accelerator) pedal in order to ensure a constant pedal position at the end of each acceleration test as well as repeatability of the accelerations.

All the above-mentioned signals from the measuring devices and instruments were fed to the input of the data acquisition module, which is a Keithley KUSB 3102 analogue-to-digital converter card connected to a personal computer via a universal serial bus interface. Following the storage of the recorded measurements into files, the data were processed using an in-house-developed computer code.

### *Experimental procedure*

The first task of the test bed installation was the investigation of the steady-state performance and combustion noise behaviour of the examined engine in order to gain a first insight into the engine's operating strategy and calibration. With this aim, an extended series of steady-state trials was conducted (using only neat diesel fuel), covering the whole engine rotational speed and load operating ranges.

The main task of the experimental procedure was to study the development of engine combustion noise during various acceleration schedules. Since the engine was coupled to a hydraulic dynamometer, during all accelerations, the brake load (resistance) increased accordingly. The acceleration tests were performed for various combinations of initial engine rotational speeds and loads, and for various demanded speed changes, mimicking the real acceleration of a vehicle under different (vehicle) speeds and gears; the details of all test cases reported in this work are summarized in Table 2.

Initially, all acceleration tests were conducted with the engine running on neat diesel fuel, which constituted the 'baseline' with which the corresponding cases are compared when using the blends of diesel with the two biofuels. The same tests were performed for each fuel blend assuring the best possible repeatability through the custom-made 'stop' fitted on the accelerator pedal. For every fuel change, the fuel lines were cleaned and the engine was left to run for a sufficient period of time to stabilize at its new condition.

## Properties of fuels tested and their blending

The conventional diesel fuel used was supplied by the Aspropyrgos Refineries of Hellenic Petroleum SA and represents a typical Greek automotive low-sulphur (0.035 wt %) diesel fuel; it formed the baseline fuel of the present study. The biodiesel used was a mixture of 50 vol % methyl ester originating from sunflower oil and 50 vol % methyl ester originating from cottonseed oil. A blend of 30 vol % of this was prepared by mixing with conventional diesel fuel.

The methyl esters were supplied from and produced at the Chemical Process Laboratory of the School of Chemical Engineering, of the National Technical University of Athens, via transesterification with methanol of the corresponding feedstock raw material at a pilot plant using potassium hydroxide as the catalyst; they were tested according to the EN 14214 standard.[40] The original vegetable oils (sunflower and cottonseed) were produced from Greek feedstock and obtained from Greek commercial processing facilities.

As regards butanol ($C_4H_9OH$), its isomer *n*-butanol (otherwise called 1-butanol), having a straight-chain structure and the hydroxyl group (–OH) at the terminal carbon atom, was used in the present study; it was of 99.9% purity (analytical grade). A blend of 25 vol % of this was prepared by mixing with conventional diesel fuel. Preliminary evaluation tests on the solubility of *n*-butanol in the diesel fuel with blending ratios up to 50 vol % *n*-butanol–50 vol % diesel proved, in fact, that the mixing was excellent with no phase separation for a period of several days.

It was decided to use a rather high blending ratio of both alternative fuels with conventional diesel oil in order for the differences to be more prominent and the underlying mechanisms to be better understood. However, a smaller blending ratio was applied for *n*-butanol than for biodiesel owing to its much lower cetane number and much higher oxygen content relative to biodiesel and diesel fuel. Table 3 summarizes the

**Table 3.** Properties of diesel fuel, methyl esters of sunflower and cottonseed oils, and *n*-butanol (normal butanol).

| Property (units) | Value for the following fuels | | | |
|---|---|---|---|---|
| | Diesel fuel | Sunflower methyl ester | Cottonseed methyl ester | *n*-butanol |
| Density at 20 °C (kg/m$^3$) | 837 | 880 | 885 | 810 |
| Cetane number | 50 | 50 | 52 | ≈25 |
| Lower calorific value (kJ/kg) | 43,000 | 37,500 | 37,500 | 33,100 |
| Kinematic viscosity at 40 °C (mm$^2$/s) | 2.6 | 4.4 | 4 | 3.6[a] |
| Boiling point (°C) | 180–360 | 345 | 345 | 118 |
| Latent heat of evaporation (kJ/kg) | 250 | 230 | 230 | 585 |
| Bulk modulus of elasticity (bar) | 16,000 | 17,500 | 17,500 | 15,000 |
| Oxygen content (wt %) | 0 | 10.9 | 10.9 | 21.6 |
| Stoichiometric air-to-fuel ratio | 15.0 | 12.5 | 12.5 | 11.2 |
| Molecular weight (kg/kmol) | 170 | 284 | 284 | 74 |

[a]Measured at 20 °C.

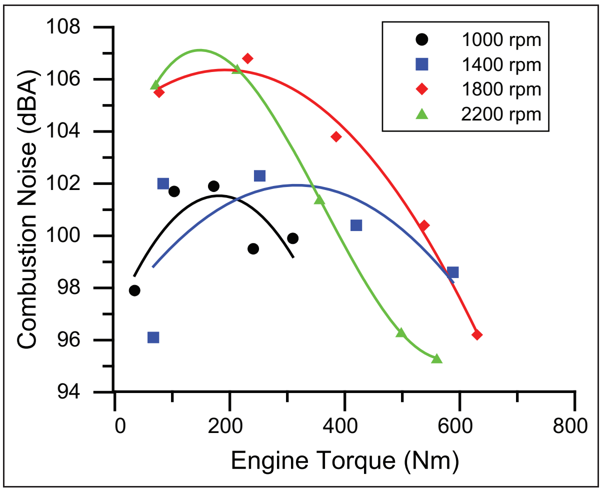


**Figure 2.** Steady-state map of engine combustion noise for neat diesel fuel operation.

properties of the diesel fuel, the two methyl esters constituting the biodiesel and the *n*-butanol.

## Experimental results and discussion

### *Steady-state results*

Figure 2 illustrates the steady-state combustion noise map of the engine in hand when operating with neat diesel fuel. This diagram is given in order to highlight an important feature of the specific engine calibration, i.e. with increasing torque (or load) a decreasing trend in the combustion noise is experienced; the peak is observed roughly at 30–40% load for each rotational speed. For example, when the engine operates at 1800 rpm, a significant decrease, of the order of 11 dBA (from 107 to 96 dBA), is experienced when moving from 230 Nm (30% load) to 630 Nm (80% load). This trend is largely attributed to the injection strategy of the engine, which with increasing torque or load shifts the injection timing closer to (or even after) the top dead centre (TDC) in order to limit $NO_x$ emissions; since combustion starts later in the cycle in a more favourable air environment, the ignition delay and therefore the rate of change in the in-cylinder pressure decrease[1,4] (plots on the left in Figure 3), despite the higher value of the peak cylinder pressure. As a result, the radiation of combustion noise is noticeably limited, most particularly in influencing the high-frequency spectrum (close to and above 1 kHz) as the plots on the right in Figure 3 illustrate. It will be argued later in the text that this injection calibration also influences strongly the whole pattern of combustion noise during transients and for every fuel blend.

### *Acceleration test results*

The discussion of the results and their interpretation in this section deals primarily with the effects of the fuel properties on the engine operating characteristics (injection, fuel spray development and combustion processes), performance and combustion noise developed during acceleration. The analysis of the phenomena experienced during transient engine operation (most notably the turbocharger lag), which affect decisively the performance and exhaust emissions, have been detailed recently in previous publications by the present research group[17,38] (for the engine in hand) and will only be outlined here for brevity.

*Test 1.* Figure 4 illustrates the engine and turbocharger response during the first acceleration (test 1 in Table 2). In particular the development of the engine speed, the fuel pump rack position, the boost pressure, the turbocharger speed, the peak cylinder pressure and estimation of combustion noise radiation are detailed for a high-speed change (approximately 800 rpm), commencing from a low engine rotational speed and low load. The specific test is particularly demanding for both the engine and the turbocharger, since the latter accelerates from almost zero boost.

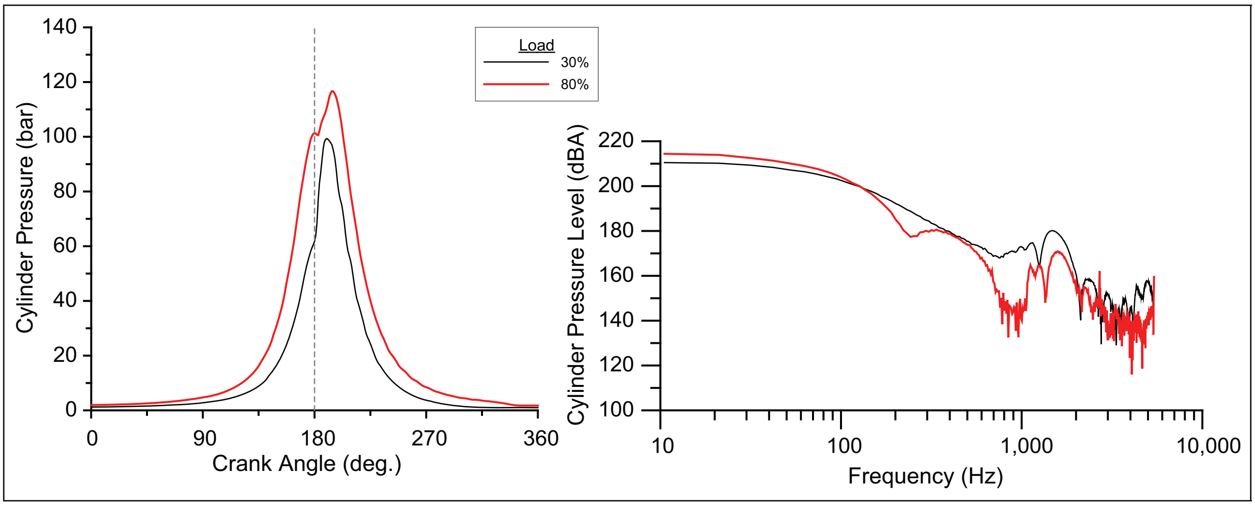


**Figure 3.** Two representative steady-state cylinder pressure and sound level diagrams at 1800 rpm; note that, despite the higher peak pressure for the 80% load, the respective pressure gradient is lower, resulting also in a lower amount of radiated noise.

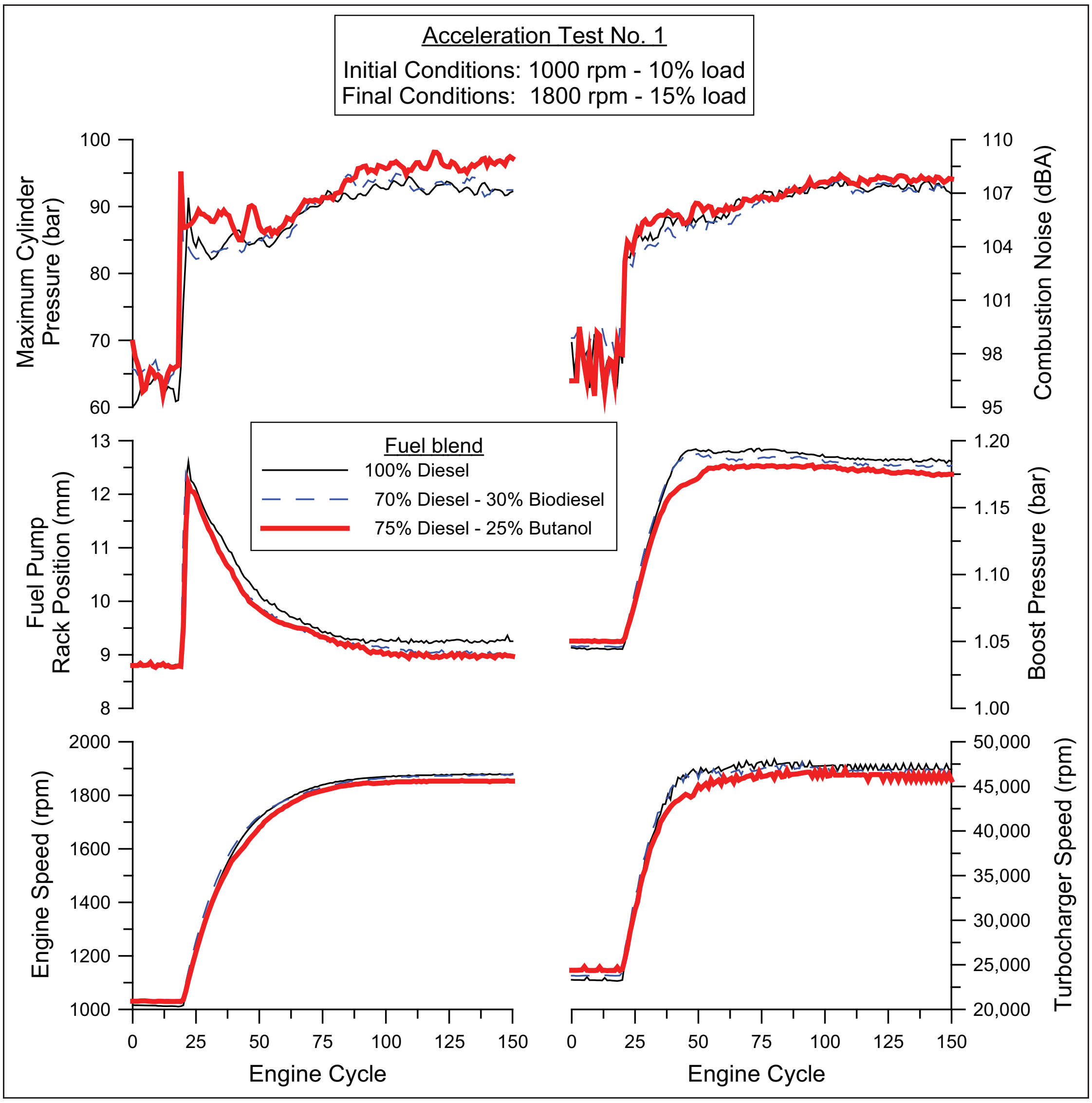


**Figure 4.** Development of the engine and turbocharger variables and combustion noise radiation during acceleration commencing from a low speed and a low load.

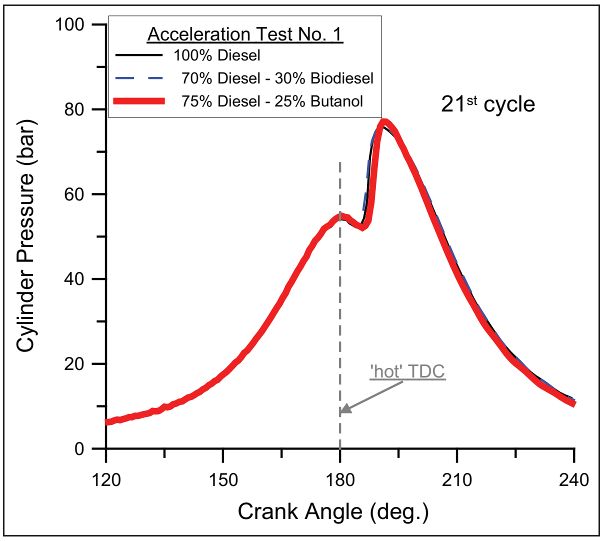


**Figure 5.** Three representative cylinder pressure diagrams during the 21st cycle of acceleration test 1 in Figure 4.
TDC: top dead centre.

For all three blends, the fuel pump rack responds almost instantly to the fuelling increase command and shifts to its maximum position within three engine cycles. Then, it gradually moves backwards to its final steady-state position; these positions are almost the same for all fuel blends. The fuel pump rack movement provokes, in general, similar acceleration rates for the three fuels, with a small differentiation observed only as regards the diesel–*n*-butanol blend. The same findings hold true for the turbocharger speed and boost pressure development. It must be noted here that such slight deviations are unavoidable in a non-electronically controlled test bed. However, the deviations observed here are rather modest and it is believed that they will not affect the general trends for each fuel blend examined.

The development of the peak cylinder pressure traces in Figure 4 is indicative of combustion evolution; the latter is well known to be influenced by the injection timing, spray development and ignition delay. These three individual processes are strongly dependent on the specific physical and chemical properties of each fuel. Closer examination of the peak cylinder pressure curves (upper left plots in Figure 4) reveals that for both neat diesel and the diesel–biodiesel blend the maximum cylinder pressure responses are essentially the same, whereas an increase is experienced for the diesel–*n*-butanol blend. This is further documented in Figure 5, which gives three representative cylinder pressure diagrams during the 21st cycle of the acceleration test for all fuels tested, as well as in the upper plots in Figure 6, which depict the cylinder pressure rise rate during the same cycle. Confirming the results of previous research during steady-state engine operation (see, for example, the papers by Pucher and Sperling[41] and Karabektas

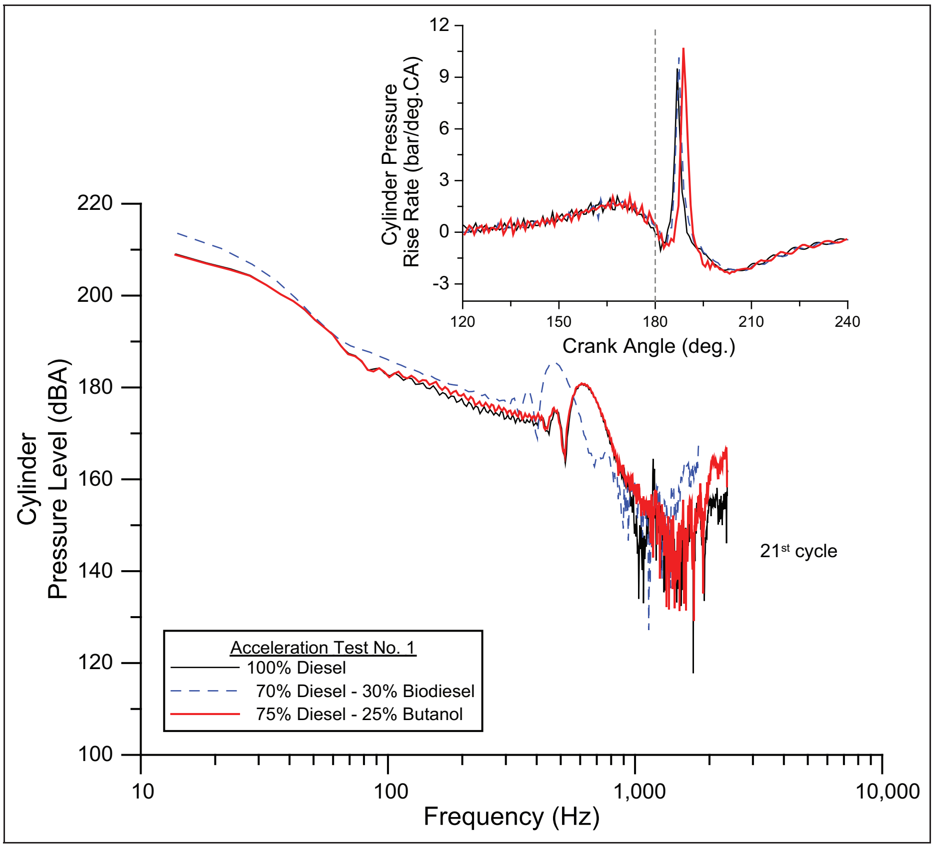


**Figure 6.** Cylinder pressure rise rate and sound level during the 21st cycle of acceleration test 1 in Figure 4.
CA: crank angle.

and Hosoz[42]), a slight increase in the ignition delay period for the diesel–*n*-butanol blend is noticed compared with the other two blends, which results in a higher cylinder pressure increase rate and ultimately higher noise radiation (upper right plots in Figure 4). This higher noise radiation can be further established in the lower plots in Figure 6, which illustrate the respective cylinder pressure spectra for the three examined indicator diagrams in Figure 5. (Noise emission is dependent on the first derivative $\mathrm{d}p/\mathrm{d}t$ of pressure with respect to time. In Figures 5 and 6 (as well as in the respective figures for the next test cases examined), the term $\mathrm{d}p/\mathrm{d}\varphi$ is used for noise estimation, since indicator diagrams from the same engine cycle are chosen for all three examined fuel blends; these cycles have the same engine rotational speeds (considered to be constant throughout each cycle).) The higher amount of emitted noise for the diesel–*n*-butanol blend is apparent in Figure 6 during the critical (for the combustion excitation forces) region between 500 Hz and a few kilohertz, as well as in the higher values of the cylinder pressure rise rate.

In order to interpret these results and to understand the slightly noisier behaviour of the diesel–*n*-butanol blend, the following (sometimes contradicting) factors should be considered.

The most apparent reason for the increase in the ignition delay period (Figure 5), which is well known to be responsible for higher noise radiation[1,4] when using the diesel–*n*-butanol blend relative to the other two fuels, is the significantly lower cetane number of *n*-butanol (25) compared with those of neat diesel fuel (50) or biodiesel (52 for the current study). Other factors that aid the previous trend are, first, the higher kinematic viscosity of *n*-butanol, which leads to the formation of larger-diameter droplets the evaporation of which is more difficult, and, second, its higher latent heat of vaporization that cools the surrounding air charge during injection.

In parallel to the previous three factors, there are two other aspects during the combustion process that act contrarily, namely in the direction of ignition delay reduction. The first has to do with the dynamic injection timing, which is affected by the speed of sound through which the pressure waves are transmitted in the (mechanical) injection pump. The lower values of the density and bulk modulus of elasticity (Table 3) of *n*-butanol relative to neat diesel fuel result in an increase in its dynamic injection timing (the static injection timings are the same for all the cases examined).[43] Consequently, for the diesel–*n*-butanol blend, injection is expected to start slightly later in the cycle, in a possibly more favourable environment (the opposite is true for the diesel–biodiesel blend). Moreover, the lower boiling point of *n*-butanol is known to accelerate evaporation of the fuel droplets compared with neat diesel operation.

It is the synergistic effect of the above-mentioned factors that is responsible for the behaviour illustrated in Figures 5 and 6 (i.e. slightly longer ignition delay and higher cylinder pressure rise rate, and hence higher combustion noise, for the diesel–*n*-butanol blend). In any case, the much lower cetane number of *n*-butanol is believed to play the decisive role here, as also in the results obtained by Strahle et al.[44] during steady-state engine operation. Likewise, for the diesel–biodiesel blend the cetane number is almost the same as that of neat diesel oil; hence for the biodiesel–diesel blends the difference in the estimated emitted noise relative to that of neat diesel fuel is smaller than that of the diesel–*n*-butanol blend (upper right plots in Figure 4).

Returning to the cylinder pressure frequency spectrum in Figure 6, it can be stated that, in general, there are three features of the combustion process that can be identified by such a frequency spectrum: the peak cylinder pressure is indicated by the pressure level at low frequencies (e.g. 10–30 Hz); the mean rate of pressure rise is generally reflected in the 1–4 kHz region, whereas the high frequencies (above 5 kHz) are usually indicative of the pressure rise change rates at the actual start of combustion.[1] Confirming the results of previous research,[45,46] as the premixed combustion was extended for the diesel–*n*-butanol blend compared with neat diesel oil, the high-frequency components of the cylinder pressure level were also enhanced, ultimately resulting in higher amounts of radiated noise. It was the limitations of the acquisition card sampling frequency that prohibited the depiction of sound level values for even higher frequencies in Figure 6; the latter are, in any case, of minor importance for the amount of emitted in-cylinder diesel engine combustion noise.

*Test 2.* Figure 7 demonstrates the results for the transient test 2 (Table 2). Here, acceleration from an initial speed of 1500 rpm to a final speed of 2000 rpm is performed under a low load. The brake loading increased accordingly during the acceleration, namely from 10% to 20%. As was also the case with the previous test, the responses of five representative engine and turbocharger operating parameters as well as the development of the estimated combustion noise are depicted in this figure. This test, as well as the next test, is more indicative of the general noisier trend established for the diesel–biofuel blends.

As with test 1, the fuel pump rack movements are generally the same for all three fuel blends during test 2, instantly shifting to the maximum position before gradually settling to the final steady-state value. Thus, the engine speeds are satisfactorily comparable for all three cases, with the turbocharger speed and boost pressure development following accordingly, ensuring in this way a reliable comparison of the results.

The upper left plots in Figure 7 illustrate the peak cylinder pressure response, which is a useful indication of the emitted combustion noise. A close examination of these plots reveals that the higher peak cylinder pressures are (again) experienced for the diesel–*n*-butanol

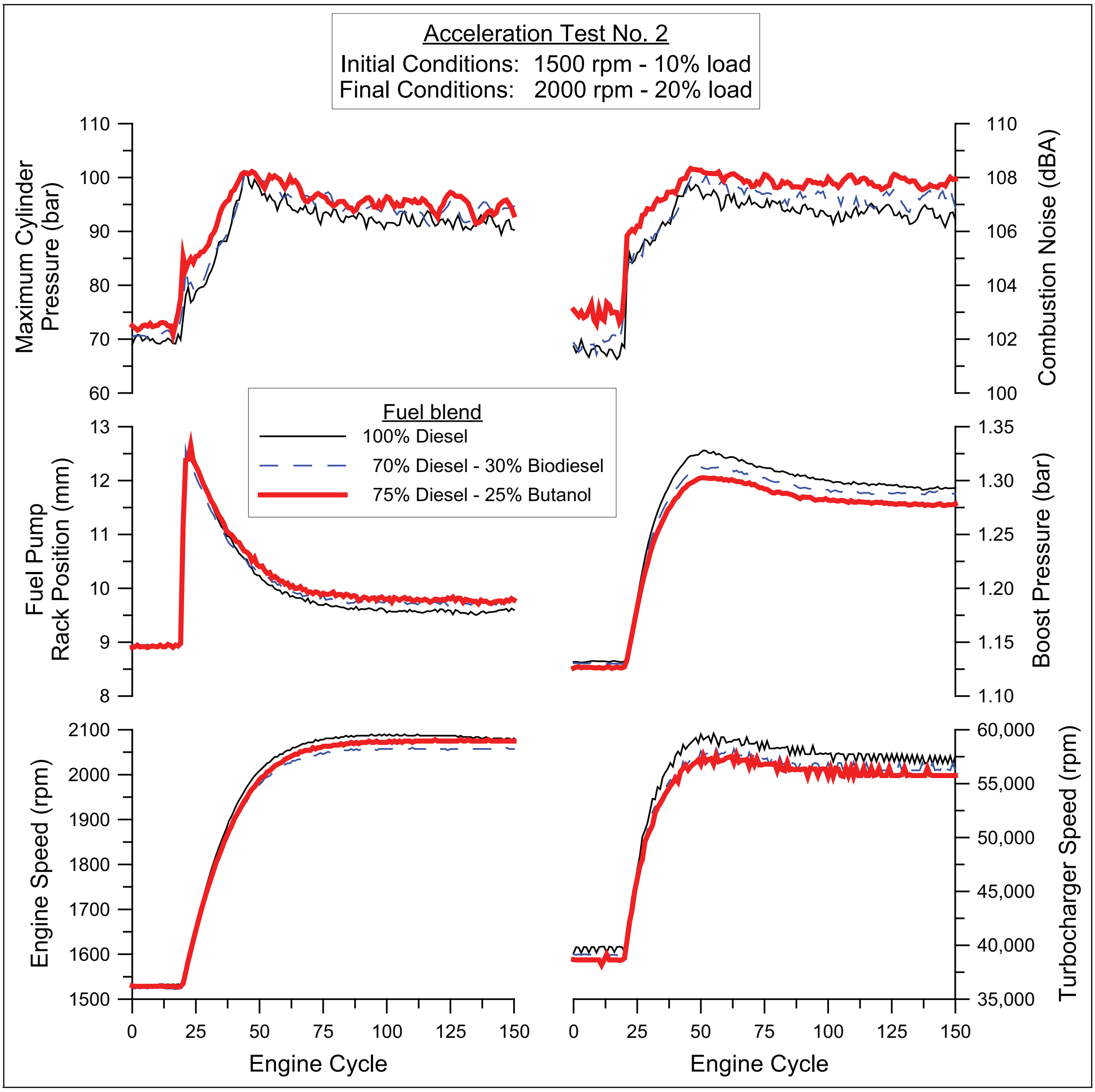


**Figure 7.** Development of the engine and turbocharger variables and combustion noise radiation during acceleration commencing from a medium speed and a low load.

blends, with the diesel–biodiesel blend exhibiting lower values but still higher than for neat diesel oil operation. Nonetheless, the main arguments are provided by Figure 8, which represents three typical indicator diagrams during the 46th cycle of the acceleration event. From Figure 8 it is also observed that the diesel–biodiesel blend experiences a slightly shorter ignition delay period, as past research has already established. This is largely attributed to the higher cetane number of biodiesel than that of the diesel fuel; however, a higher cylinder pressure rate and peak pressure are encountered. As was also the case during the previous acceleration, the diesel–*n*-butanol blend exhibits the longest ignition delay.

Consistent with these comments, the radiation of the estimated combustion noise (upper right plots in Figure 7) is clearly higher for the diesel–*n*-butanol blend (as a result of the longer ignition delay, which increased the cylinder pressure rise rate accordingly; the latter is further demonstrated in the upper plots in Figure 9). This higher noise emission is up to 2.2 dBA during the transient event compared with neat diesel operation and is experienced during the whole acceleration event. The lower plots in Figure 9 further enhance these findings by demonstrating the respective cylinder pressure spectra for the indicator diagrams in Figure 8, where the overall higher values for the diesel–*n*-butanol blend are obvious during the critical region above 500 Hz, which

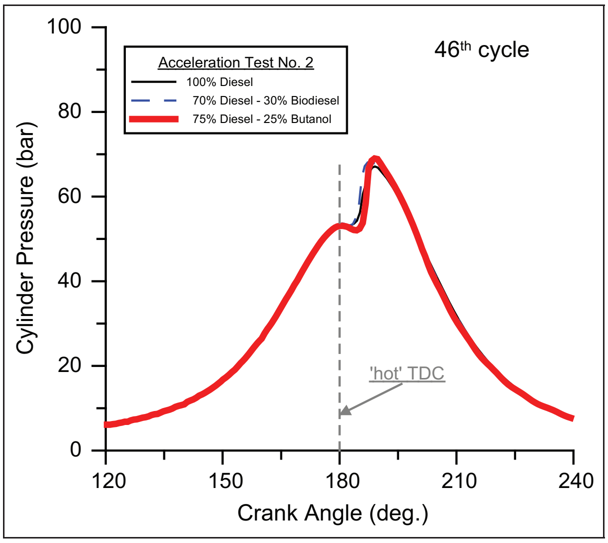


**Figure 8.** Three representative cylinder pressure diagrams during the 46th cycle of acceleration test 2 in Figure 7.
TDC: top dead centre.

represents the contribution of the combustion excitation forces to the total amount of radiated noise.

An explanation is due regarding the slightly higher combustion noise (and the higher peak pressures and rate of pressure change as the upper plots in Figure 9 demonstrate) of the diesel–biodiesel blend relative to neat diesel operation despite the shorter ignition delay period observed in Figure 8. The main factor responsible for this behaviour is the mechanical fuel injection pump, which operates on a volumetric basis rather than a gravimetric basis. Consequently, the higher density of the biodiesel results in larger amounts of injected fuel than for neat diesel operation. Another contributing factor is the reduced fuel leakage losses in the fuel pump owing to the higher kinematic viscosity of biodiesel than that of neat diesel fuel, which leads to a higher injection pressure[32] and a larger mass of injected fuel. All in all, although the initial preparation period is actually shorter because of the marginally higher cetane number of biodiesel, the fact that during the premixed combustion phase a greater mass of fuel is burned results in higher peak pressures and $dp/d\varphi$ rates; the latter are also reflected in higher emission of estimated combustion noise (up to 1.5 dBA compared with neat diesel operation, which nonetheless is very close to the accuracy of the noise meters), as in the upper right plots in Figure 7, as also the corresponding frequency spectrum diagram in Figure 9 corroborates. The latter finding seems to confirm the steady-state results found by Selim et al.;[47] although Selim et al. worked on an indirect-injection diesel engine, one of their conclusions was that with late injection timings and low loads (such as those experienced during the current acceleration) their jojoba methyl ester produced fairly high pressure rise rates (the latter was used as an indication of the emitted combustion noise). The same finding of higher (soybean) biodiesel peak pressures and pressure rise rates than with neat diesel operation was reached by Qi et al.,[48] again for low-load engine operation, but this time for a DI diesel engine such as that under study in this work.

*Test 3.* The last test conducted test (test 3) is almost the same as the previous test in terms of the engine

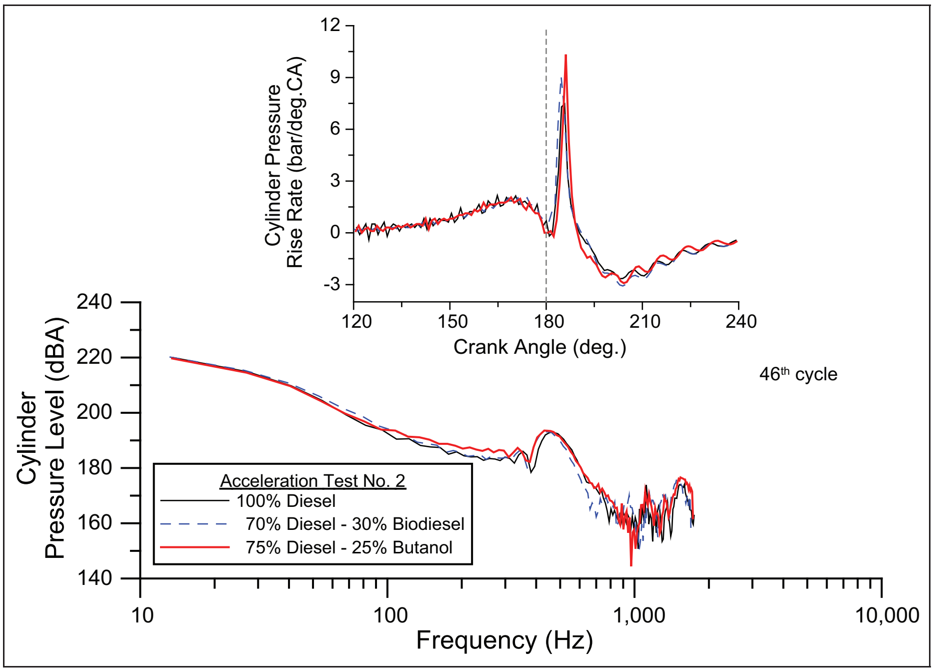


**Figure 9.** Cylinder pressure rise rate and sound level during the 46th cycle of acceleration test 2 in Figure 7.
CA: crank angle.

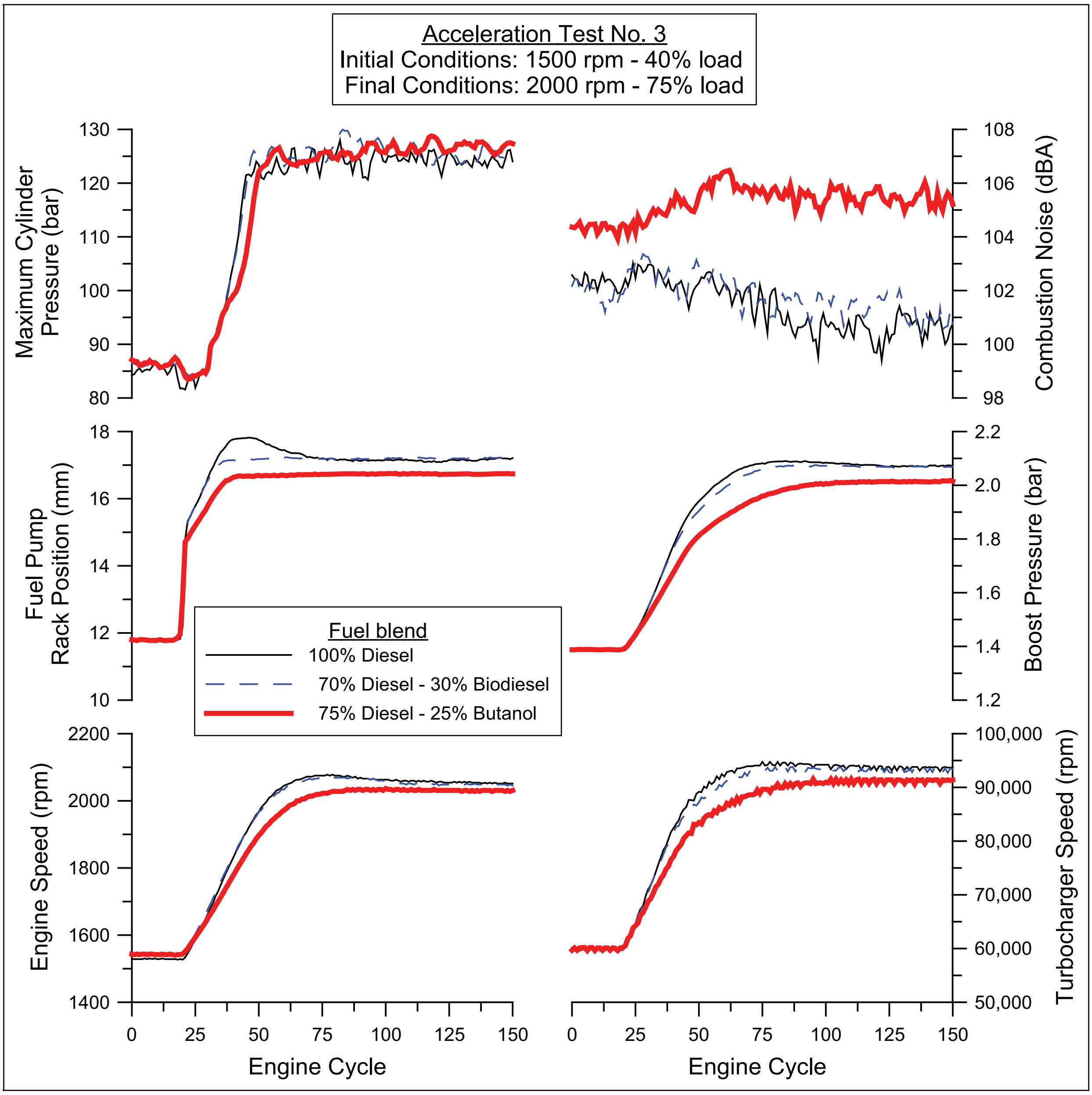


**Figure 10.** Development of the engine and turbocharger variables and combustion noise radiation during acceleration from a medium speed and a medium load.

acceleration (same initial and final speed), differing only in the initial (and final) engine loading. As with the previous tests, Figure 10 illustrates the responses of the five interesting engine and turbocharger operating parameters and the combustion noise emission estimation.

In all three cases, the fuel pump rack responds in a similar manner (middle left plots in Figure 10), shifting initially to a first peak position and then followed by a smoother movement to the final steady-state position. The latter behaviour highlights in an explicit way the fuel limiter operating principle that does not allow sharp fuelling increases when the boost pressure is still low. As can be seen in Figure 10, differentiation was experienced in the operation of the fuel pump rack for the case of the *n*-butanol blend, which affected the engine speed and the turbocharger response accordingly. Unfortunately, such (moderate) deviations are unavoidable in a non-electronically controlled test bed.

In general, the current acceleration test is characterized by the same remarks and explanations that have already been detailed in the previous section, i.e. a longer ignition delay period and higher cylinder pressures (Figure 11) and pressure rise rates (upper plots in Figure 12) for the diesel–*n*-butanol blend that are also reflected in (even) higher (up to 4 dBA) differences in the amount of radiated combustion noise estimation

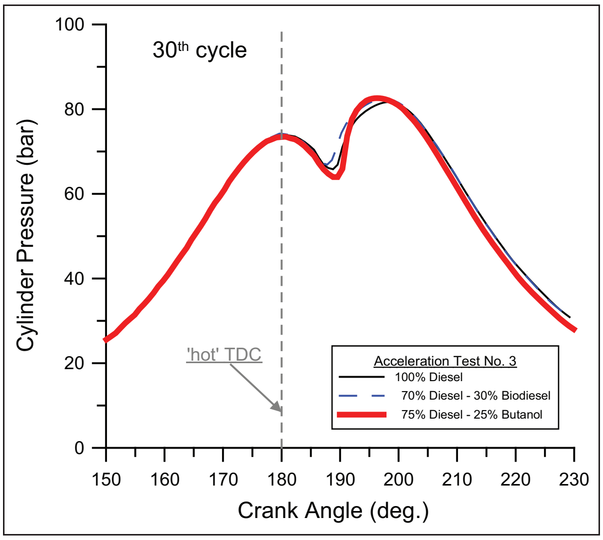


**Figure 11.** Three representative cylinder pressure diagrams during the 30th cycle of acceleration test 3 in Figure 10.
TDC: top dead centre.

compared with that in neat diesel operation (illustrated in the upper right plots in Figure 10, as well as in the lower plots in Figure 12 for the frequency spectra of the representative cylinder pressure diagrams given in Figure 11). The diesel–biodiesel blend seems to exhibit marginally higher amounts of emitted noise than neat diesel oil does; however, and in accordance with the results reached by previous research during steady-state engine operation,[32,47,48] this trend is not unambiguous throughout the whole transient event, as was also the case with tests 1 and 2.

There are two important aspects of the current acceleration test that are intriguing and will be discussed in more detail. The first is the absolute values of the estimated emitted noise compared with the previous test, and the second is the different trend in the combustion noise development of the diesel–*n*-butanol blend (upper right plots in Figure 10) from those of the other two fuels studied during this acceleration event.

Regarding the absolute values of the combustion noise radiation, it is observed that the engine behaves in a less noisy manner during this acceleration test, compared with test 2 irrespective of the fuel blend used; specifically the reduction is up to 2.5 dBA for the diesel–*n*-butanol blend and more than 4 dBA for the other two blends. Recall that these two tests are indeed comparable since the initial and final engine speeds are the same. This less noisy behaviour during test 3 compared with test 2 is attributed to the following two factors.

1. The higher initial turbocharger operating point rendered test 3 an overall 'easier' task than test 2; the effect of turbocharger operation on the estimated engine combustion noise development can be explained as follows. The main mechanism behind the increase in combustion noise radiation during transients compared with the respective steady-state operation lies in the operating principles of a transient event.[4,17] In the first few cycles after acceleration, such as those demonstrated in Figures 7 and 10, the injected fuel quantity has already increased substantially, cooling the charge-air temperature; however, the cylinder wall temperature is still low (up to 100 °C lower than the corresponding steady-state conditions[13]), as the

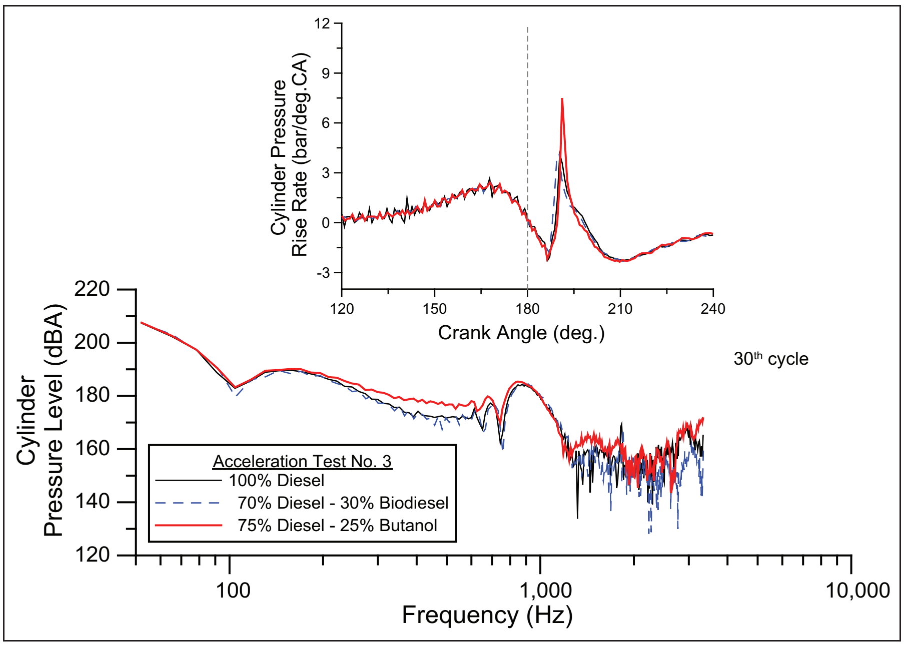


**Figure 12.** Cylinder pressure rise rate and sound level during the 30th cycle of acceleration test 3 in Figure 10.
CA: crank angle.

thermal transient proceeds at a much slower rate owing to the (generally) high cylinder wall thermal inertia. Moreover, the rapid increase in the fuel injection pressure upon the onset of each instantaneous transient causes penetration of the liquid fuel jet within the combustion chamber to increase. Since the initial higher-pressure fuel jets are injected into an air environment that is almost unchanged from the previous steady-state conditions, the higher-momentum fuel jet is not accompanied by equally enhanced gas motion. Thus, liquid fuel impingement on the still cool combustion walls increases, lowering the rate of mixture preparation.[49] The combination of increased fuelling with the 'cooled' charge-air temperatures cause the mixture preparation process to deteriorate, resulting in a longer ignition delay, and hence more intense premixed combustion periods with faster, more abrupt initial heat release; the latter manifests itself as steeper cylinder pressure gradients $dp/d\varphi$ and, consequently, higher combustion noise levels than for the respective steady-state operation. The above phenomena are more prominent during acceleration test 2 than during test 3, since for the former acceleration the turbocharger accelerated from a much lower initial point (38,000 rpm instead of 60,000 rpm for test 3), and the turbocharger lag was more pronounced.

2. The specific engine calibration, which with increase in the engine load (higher during test 3) shifts the injection timing closer to or even after the TDC to limit $NO_x$ emissions, is the second decisive parameter (recall also the comments regarding the steady-state engine noise map in Figure 2). As a result of this later injection, the fuel is now injected in a more favourable (hotter) air environment, which reduces the ignition delay period; hence the $dp/d\varphi$ rate and the estimated combustion noise radiation are lower too during test 3 than during test 2. This trend is obvious if the indicator diagrams in Figures 8 and 11 are compared (although the corresponding cycles do not correspond to exactly the same points in each cycle, the general trend is indicative).

The second aspect of the current acceleration has to do with the general trend of combustion noise radiation during the whole acceleration process. For neat diesel and the diesel–biodiesel blend, a clear tendency is established, namely a decreasing trend of combustion noise as the acceleration event develops owing to the three reasons detailed previously. However, this trend is not that clear when acceleration using the diesel–*n*-butanol blend is studied. In fact, an initial increasing trend is noticed (up to the 50th cycle), followed then by the decreasing trend that is also experienced for the other two fuel blends. Although the engine injection strategy was the same, regardless of the fuel tested, it is strongly felt that the very low *n*-butanol cetane number was again the contributing factor for this behaviour, by outperforming the positive (as regards noise radiation) effect of the later injection timing and hotter surrounding environment.

Owing to the higher noise differences experienced during these tests between neat diesel and the diesel–*n*-butanol blend, a clear difference in the cylinder frequency spectrum is also established (Figure 12). Again, as the premixed combustion was intensified, the medium-frequency and high-frequency (above 500 Hz) components of the cylinder pressure level were also enhanced, increasing accordingly the amount of the radiated noise.

## Summary and conclusions

A fully instrumented test bed installation was developed in order to study the combustion noise radiation of a turbocharged diesel engine when fuelled with various diesel–biofuel blends. A fast-response noise meter was employed for combustion noise estimation. The tested fuel blends were 100 vol % (neat) diesel oil, 70 vol % diesel–30 vol % biodiesel and 75 vol % diesel–25 vol % *n*-butanol. As the engine is intended for automotive applications, a variety of acceleration events were performed from various initial speeds and loads; the investigation was supported by representative cylinder pressure and pressure gradient diagrams, as well as frequency spectra analysis.

The basic conclusions derived from the current investigation and for the specific engine–hydraulic brake configuration and the fuel blends used are summarized as follows.

1. The blend of diesel with *n*-butanol was found to emit higher combustion noise (up to 4 dBA more than neat diesel; see also the comparative results in Figure 13) for all the accelerations examined, and with a clear noisier trend throughout each transient event; this was mostly attributed to the low cetane number of *n*-butanol, which increased the ignition delay period and led to a harsher premixed combustion phase.
2. Although biodiesel is characterized by a slightly higher cetane number, its acceleration response compared with that for neat diesel operation was proven to be marginally noisier (up to 1.5 dBA for the cases examined in this work), and this was attributed to its higher density which increased the total injected mass and the fuel mass burned during the influencing premixed combustion phase. However, the trend was not clear throughout each acceleration (summarized in Figure 13); it is suspected that higher blends might be required in order to establish a possibly unambiguous trend.
3. Irrespective of the fuel used, the fuel limiter action governing the turbocharger initial operating point and, particularly, the engine injection timing

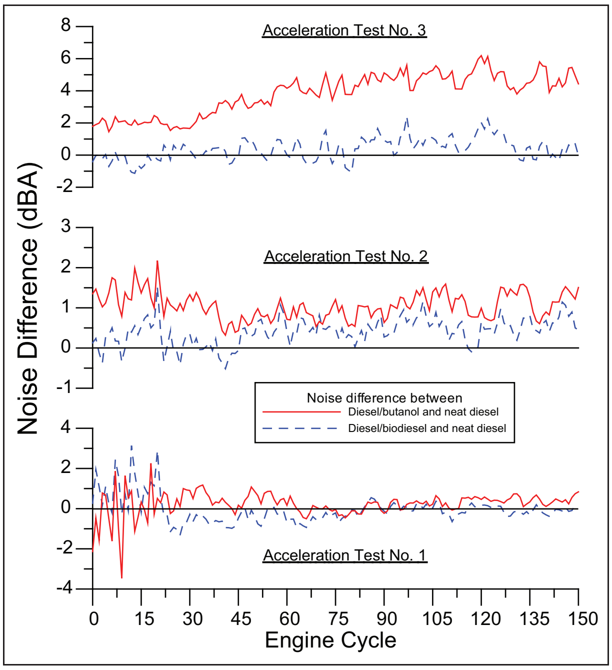


**Figure 13.** Comparative results in the combustion noise difference between the examined fuel blends throughout each of the three acceleration tests.

calibration was found to play a decisive role in the combustion noise emission by determining the cylinder pressure gradient and frequency spectrum; it also influenced the engine speed and turbocharger responses.

4. Irrespective of the fuel blend tried, transient noise radiation became smoother, the higher the initial engine loading (and hence the turbocharger operating point).
5. None of the examined blends seemed to affect the transient performance of the engine (or the turbocharger), both of which retained almost the same response characteristics, a fact that is believed to render the noise comparisons even more reliable.

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