Development of combustion instability and noise during starting of a truck turbocharged diesel engine

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Abstract: In the current study, experimental tests were conducted on a truck turbocharged diesel engine to investigate the mechanisms of combustion noise radiation and combustion instability during various starting schedules experienced in daily driving conditions, namely under cold and hot operations. To this aim, a fully instrumented test bed was set up to capture the development of key engine and turbocharger properties. Analytical diagrams are provided to explain the behaviour of combustion instability and noise radiation in conjunction with all relevant parameters, such as cylinder pressure and pressure spectrum, turbocharger and governor/fuel pump response.

Keywords: diesel engine; starting; combustion instability; combustion noise; cylinder pressure spectrum.

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1 Introduction

The turbocharged diesel engine is currently the preferred powertrain system in the transport sector as well as in medium and medium-large unit applications. Moreover, it continuously increases its share in the highly competitive automotive sector, having already ensured a market share comparable to the one for the gasoline engine in the European Union. The most attractive features of the diesel engine are its reliability and its very good fuel efficiency; consequently, diesel-engined vehicles achieve much lower fuel consumption and reduced CO_2 emissions than their similarly rated spark ignition counterparts. Despite its significant advantages, however, the diesel engine remains by far inferior to the gasoline engine during starting, where the engine:

- notoriously suffers from combustion instability phenomena
- produces an increased amount of pollutant and noise emissions (Heywood, 1988).

Starting is distinguished as either cold or hot depending on the prevailing coolant (and oil) temperature. The former case has initiated much more vigorous research owing to the significantly higher discrepancies experienced by the engine until it manages to reach a self-sustained rotational speed, compared with the relatively 'easier' cases of warm or hot starting.

Unlike spark ignition engines, where combustion is initiated by the spark and is aided by the high gasoline volatility, in diesel engines, the auto-ignition process may prove unreliable during cold starting, particularly at temperatures below 0°C. An automotive engine starting event consists of three phases:

- initially the engine accelerates rapidly with the assistance of the electric starter (cranking phase)
- afterwards it continues to accelerate without the need for external assistance, until
- the point where stabilisation of the idling speed is achieved.

An increased amount of soot, hydrocarbon and carbon monoxide is expected during the cold start phase, particularly if misfiring occurs, which is more likely with colder ambient conditions. Under misfiring conditions, combustion cannot supply enough power to drive the engine and overcome the increased heat losses to the cold cylinder walls and the increased friction and blow-by losses that emanate from high lubricant viscosity at low temperatures. Consequently, formation of a combustible air-fuel mixture is prohibited during some cycles, ultimately leading to combustion instability or even starting failure (Kobayashi et al., 1984; Tsunemoto et al., 1985; Henein et al., 1992; Bielaczyc et al., 2001). The term combustion instability is used to characterise the various in-cylinder irregularities during the starting cycles, most notably the cycle to cycle variation of cylinder gas (peak) pressures.

Previous research, mainly on naturally aspirated engines, has identified many interesting parameters as critical for the occurrence of combustion instability (Phatak and Nakamura, 1983; Kobayashi et al., 1984; Henein et al., 1992): ambient temperature (the lower the temperature the greater the possibility of misfire); fuel cetane number (higher values are preferred); compression ratio (it determines the charge-air pressure and temperature at the point of fuel injection, affecting strongly the ignition delay duration); blow-by losses; cranking speed (the lower the cranking speed the more the available time for heat and blow-by losses); fuel injection timing and pattern (retarded injection is generally favoured); amount of residual gas (exhaust gases recirculating from one cycle to the next affect composition and temperature during the next compression stroke); combustion chamber design (among other things, the longer the distance between the fuel injector and the cylinder wall, the better the possibility of the fuel evaporating before the spray impinges on the wall); starting aids, etc.

Another unfavourable demonstration of a compression ignition engine (cold starting) operation is the emission of noise. The three primary sources of noise generation in a diesel engine are: gas-flow, mechanical processes, and combustion (Austen and Priede, 1958; Lilly, 1984). Gas-flow noise, usually low frequency controlled, is associated with the intake and exhaust processes including turbocharging and the cooling fan (Bozza et al., 2009). Mechanical noise comprises both rotating and reciprocating engine components' contribution; it originates from inertia forces causing piston slap, from gears, tappets, valve trains, timing 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 a strong high-frequency pressure spectrum resulting in vibration of the engine block and, ultimately, in noise radiation (the characteristic diesel combustion 'knock').

Combustion chamber design and injection parameters, e.g., timing, amount and rate of fuel injected during pre- and main injections play a principal role in diesel combustion noise emission by defining the exact rate of heat release during steady-state or transient conditions (e.g., Kondo et al., 2001; Goldwine and Sher, 2009). To analyse this source of noise, the cylinder pressure signal is usually examined on the frequency spectrum, for example, using the Fourier transform (Lilly, 1984). Particularly with diesel-engined vehicles, the unpleasant combustion knock is a matter of discomfort for passengers and pedestrians.

Combustion noise development during a transient event differs to a large extent from the respective steady-state operation; this was the result reached by the surprisingly few studies carried out so far (Watanabe et al., 1979; Head and Wake, 1980; Dhaenens et al., 2001, Shu and Wei, 2007), mainly for naturally aspirated diesel engines. During steady-state operation, engine speed and fuelling remain practically constant; under transient conditions, however, the engine speed changes continuously, following the forced changes in fuelling. As a result, the available exhaust gas energy varies, affecting turbine enthalpy drop while through the turbocharger shaft torque balance, the boost pressure and the air supply to the engine cylinders are influenced. However, due to various dynamic, thermal (e.g., cylinder wall temperature) and fluid delays in the system, combustion air-supply is delayed compared to fuelling, affecting the combustion process and consequently, torque build-up and emission of pollutants and noise unfavourably (Rakopoulos and Giakoumis, 2009). The above transient discrepancies have been reported to be even more prominent during cold starting, where the much lower cylinder wall temperatures lead to even longer ignition delay periods and hence, to harsher premixed combustion and higher noise radiation (Alt et al., 2005).

The target of this study is to expand on the experimental investigation of combustion instability and (on the extremely limited investigation of) combustion noise radiation during starting of turbocharged diesel engines, and shed more light on the relevant phenomena and underlying mechanisms. To this aim, an extended set of experimental tests was conducted on a medium-duty, turbocharged and after-cooled, direct injection, truck diesel engine applying a modern combustion noise-meter for accurate cylinder pressure data analysis. An important aspect of the current investigation is that it focuses on both starting cases experienced in daily vehicle driving, i.e., cold and hot. By so doing it is believed that useful overall conclusions on the diesel engine's starting behaviour can be deduced.

2 Description of the experimental installation and procedure

The engine used in this study is a Mercedes-Benz OM 366 LA, turbocharged and after-cooled, direct injection diesel engine. It is widely used to power mini-buses and small/medium trucks; its basic technical data are given in Table 1. The engine is coupled to a hydraulic dynamometer.

Table 2 summarises the main data of the various starting tests conducted at different idling speeds and coolant/oil temperatures; the lubricating oil pressure is given only as an index of its temperature (the cooler the oil the higher its pressure for constant engine idling speed). For each test, the pedal was fixed to a specific position corresponding to the desired engine idling speed and then the starter button was initiated.

Table 1 Engine and turbocharger specifications

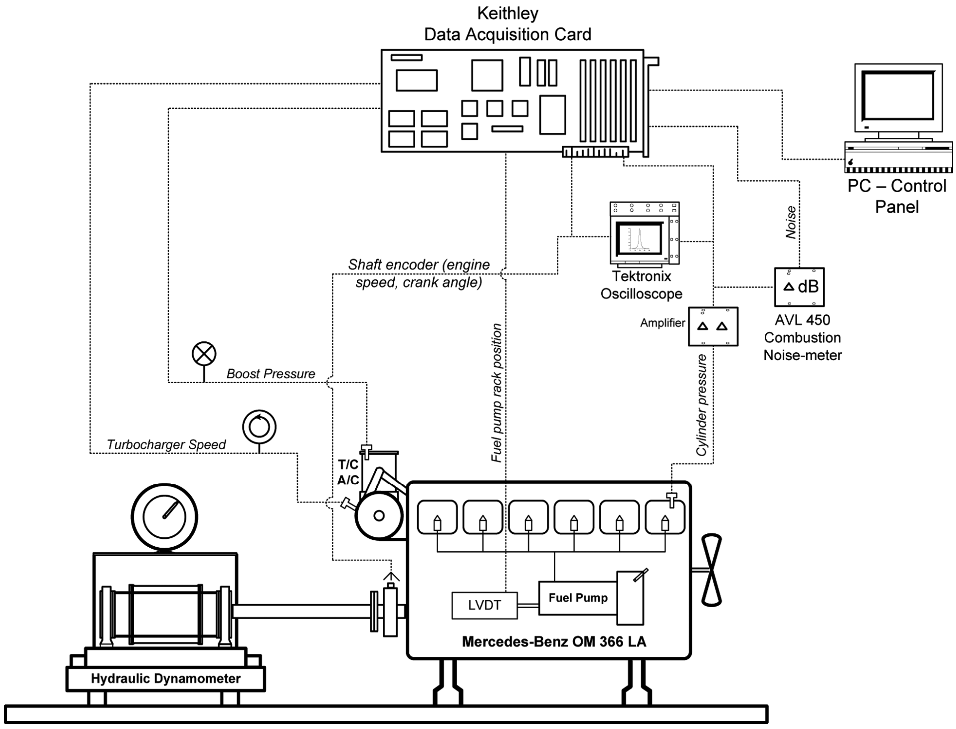
Engine model and type	Mercedes-Benz OM 366 LA, 6 cylinder, in-line, 4-stroke, compression ignition, direct injection, water-cooled, turbocharged, aftercooled, with bowl-in-piston
Speed range	800–2600 rpm
Maximum power	177 kW@2600 rpm
Maximum torque	840 Nm@1250–1500 rpm
Engine total displacement	5.958 L
Bore/Stroke	97.5 mm/133 mm
Compression ratio	18 : 1
Fuel pump	Bosch PE-S series, in-line, 6-cylinder with fuel limiter

The various engine and turbocharger operating parameters measured and recorded continuously were (see also Figure 1): engine speed; cylinder pressure; fuel pump rack position; boost pressure; turbocharger speed, and combustion noise. Particularly as regards the combustion noise measurement, this was achieved using the AVL 450 combustion noise-meter. Its operating principle is based on analysis of the cylinder indicator diagrams on the frequency domain, applying a series of filters to it. Initially, the cylinder pressure signal passes through a U-filter corresponding to the frequency attenuation of the engine block. Finally, the signal is guided through an A-filter; this universally applied signal weighing technique decreases the magnitude of frequencies lower than 1 kHz (the lower the frequency, the stronger the downgrade), and enhances the 1–6 kHz frequencies to adjust the measured signal to the loudness perception of the human ear (AVL, 2000). The produced output signal is further processed by Root Mean Square (RMS) conversion to logarithmic DC values that relate to the aural threshold. The total error of the meter is less than ±1 dB.

Table 2 Tabulation of starting conditions for each test conducted

Test No.	*Conditions*	*Idling speed* (rpm)	*Coolant temperature* (°C)	*Lubricating oil pressure* (bar)
1	'Cold'	900	20	5.9
2	'Hot'	950	80	1.8
3	'Hot'	1215	80	2.5

Figure 1 Schematic presentation of the test bed installation



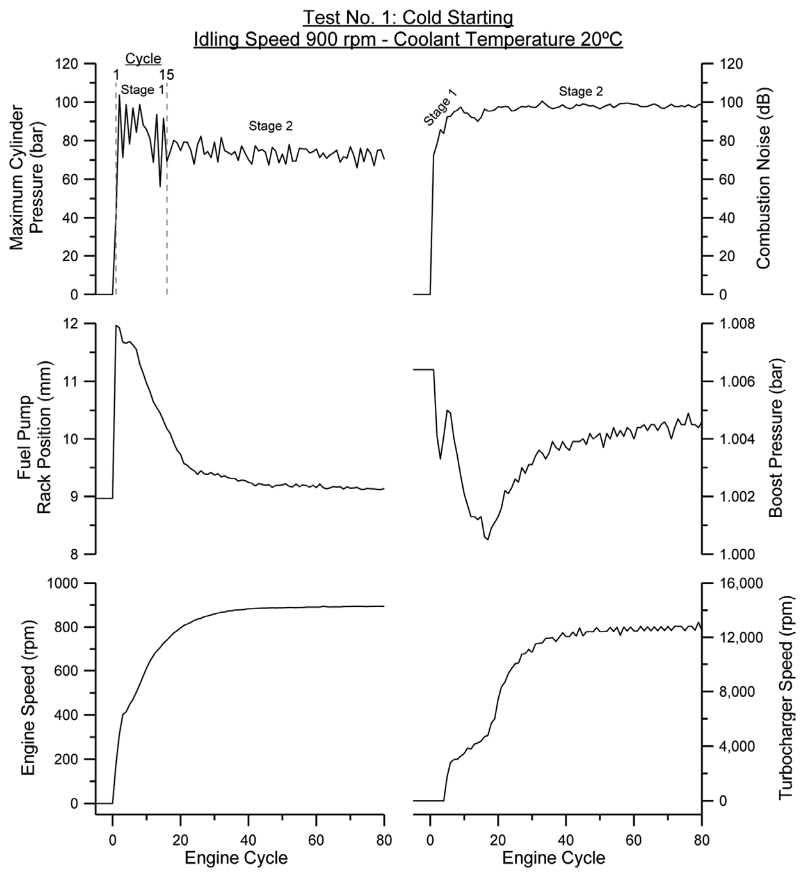
3 Results and discussion

3.1 Cold starting Test No. 1

The first case of starting (Test No. 1) was performed under cold conditions, i.e., the coolant and lubricating oil temperatures were equal to ambient (20°C for the present study) temperature. The development of various engine and turbocharger operating parameters as well as the combustion noise radiation for this test is illustrated in Figure 2.

Before the engine was started, the fuel pump rack was located at its minimum position (middle-left sub-diagram of Figure 2). As soon as the starting event was initiated, the governor sensed the very low cranking speed, which was much lower than the required self-sustained one, and forced the fuel pump rack to shift instantly to its maximum fuelling position. The initial sharp increase of the engine speed noticed in Figure 2 for the first three cycles (lower-left sub-diagram) was supported by the assistance of the electric starter. After the disengagement of the starter (Cycle No. 3 or at $t = 1.4$ s, a rather small number of cycles owing to the moderately low ambient temperature), the engine accelerated by itself at a much slower rate. During this period, since there was clearly a lack of sufficient air-flow due to the still very low engine and turbocharger rotational speeds, locally high fuel-air ratios were experienced, leading to flame quenching (owing to oxygen shortage) and combustion deterioration; the latter has been identified as responsible for the combustion instability phenomena between consecutive cycles (Henein et al., 1992), as will be discussed in the next paragraph. As the engine speed gradually increased, the rack moved progressively to a lower fuel supply position until it ultimately assumed its final steady-state value after the engine had reached its idling, self-sustained speed. It is worthwhile mentioning that, even after the engine speed had stabilised, the whole phenomenon continued to develop from the thermal point of view, since a much longer duration is required for the stabilisation of exhaust gas, coolant and lubricating oil temperatures, as well as for their cylinder and exhaust manifold wall counterparts owing to their high thermal inertia (Rakopoulos and Giakoumis, 2009; Rakopoulos and Mavropoulos, 2009). This thermal transition lasted for at least a few minutes; a relatively long period compared with the duration of the actual starting event.

Figure 2 Development of engine and turbocharger variables and combustion noise radiation during cold starting (Test No. 1)



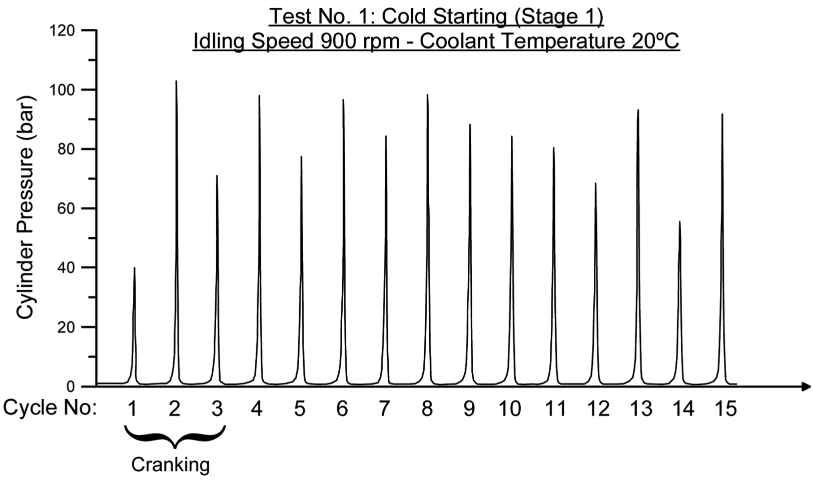
The upper-left sub-diagram of Figure 2 demonstrates in the most explicit way the occurrence of combustion irregularity during the starting event, namely in the form of the highly unstable maximum cylinder pressure traces (stage 1). For a more thorough understanding, a detailed view of the respective pressure diagrams during the 15 engine cycles of stage 1 is provided in Figure 3. Indeed, the cylinder pressure traces exhibit a high degree of variation from cycle to cycle (excluding the first cycle, the maximum difference observed between the peak pressures of two consecutive cycles was almost 38 bar). Due to low wall temperature during cold starting, the air-charge in the cylinder could not always reach temperatures capable of vapourising the injected fuel. Consequently, formation of a combustible air-fuel mixture was prohibited during some cycles, ultimately leading to the combustion instability demonstrated in Figure 3. One key parameter here is the low injection pressure encountered at the low cranking speed (recall from Table 1 that the engine under study is equipped with a mechanical fuel pump) that led to poor spray penetration, atomisation and ultimately fuel evaporation. Furthermore, the synergistic effect of

- the low coolant temperature, which led to aggravated heat loss to the walls
- the low lubricating oil temperature, which caused high frictional losses
- the low engine rotational speed that allowed more time for the above mentioned two losses to develop, and also increased the blow-by losses past the piston rings.

All resulted in low compression pressures (Cheng et al., 2004), leading eventually to incomplete combustion during various cycles.

Confirming the results of previous research for naturally aspirated diesel engine operation (Henein et al., 1992), an important finding from the analysis of the stage 1 cold starting pressure diagrams in Figure 3 is that combustion development from cycle to cycle is not getting gradually more stable and complete as the starting event advances, but rather exhibits an intermittent behaviour, i.e., there exists a series of complete and partial combustion events succeeding one another; the specific in-cylinder conditions during one cycle affecting (positively or negatively) the next one, and leading to a series of firing and misfiring cycles. It has been argued (Arcoumanis and Yao, 1994) that this behaviour is in fact the result of an imbalance between engine dynamics and combustion kinetics. Namely, the high rate of acceleration after a firing cycle reduces the time available for the physical and chemical processes to be completed prior to top dead centre (this can be documented in Figure 4, where it is made obvious that during the early starting cycles, ignition starts later in the cycle, after top dead centre, following the increased ignition delay period imposed by the cold cylinder walls). Likewise, after a misfiring cycle, engine rotational speed generally decreases, allowing more time for pre-ignition chemical reactions around TDC, thus favouring ignition in the next cycles, as is also the case in Figure 3. It has been also proposed (Osuka et al., 1994) that during the ignition cycles, compression of residual fuel from the preceding misfiring cycle(s) causes cold flame reactions, which lead to the formation of an activated environment enabling ignition to occur; this seems to be the case for Cycles No. 2, 4, 6, 8, 13 and 15 in Figure 3 for the current test.

Figure 3 Combustion instability during the first stage of cold starting (Test No. 1)

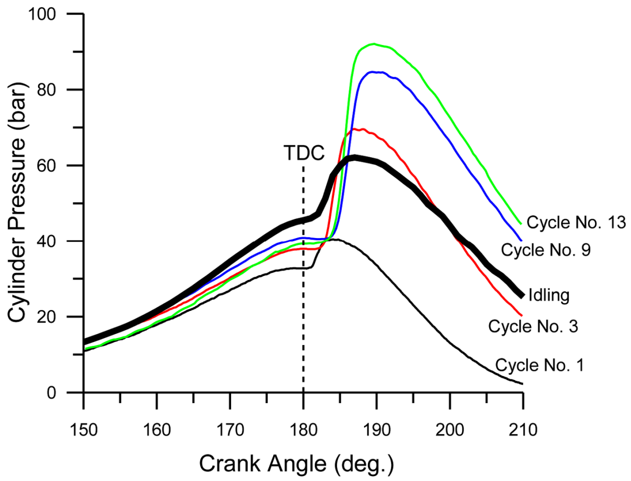


It is also worthwhile mentioning the compression curves trend in the indicator diagrams in Figure 4. During the early starting cycles (stage 1), the compression pressure assumes (much) lower values compared with the stabilisation phase and the final idling conditions (bold line in Figure 4). As discussed earlier, this is directly attributed to the considerably lower engine speeds encountered that allowed more time for heat and blow-by losses, as well as to the higher rate of heat transfer to the cold cylinder walls. Both factors reduce the enthalpy of the working medium, lowering the engine's 'effective' compression ratio and ultimately the gas pressure. A direct comparison between the starting Cycles No. 9 and 13 shows, however, that in the earlier cycle 9, a higher compression pressure is encountered compared with the later cycle 13. This seems, at first, to contradict the previous point but it can be explained by the well established fact that during cold starting, any unburned fuel remaining from previous cycles (of incomplete combustion) affects the fuel-air ratio and accordingly raises the compression ratio of the following cycle(s) (Henein et al., 1992).

The second class of starting discrepancy studied is the evolution of combustion noise. Combustion noise is mainly determined by the cylinder pressure rise rate during the engine cycle (Lilly, 1984), i.e., its gradient with respect to crank angle $dp/d\varphi$. This rate is determined by the exact heat release rate pattern, and is influenced by a variety of parameters including injection timing and ignition delay. Under cold starting conditions, both these parameters behave differently compared with the fully warmed-up engine operation. As discussed previously, during the first cycles of the cold starting test, the rate of mixture preparation was very low. Moreover, the low cylinder wall temperatures promoted a high rate of heat transfer, thus preventing fast and smooth fuel ignition, and led to an intense premixed combustion phase, and hence, steep cylinder pressure gradients (and high maximum cylinder pressures as the upper-left sub-diagram of Figure 2 shows). Consequently, an increasing trend of combustion noise level was also experienced during stage 1 (upper-right sub-diagram of Figure 2) up to the point where the engine achieved the desired idling speed; at that point, combustion noise practically stabilised to its final steady-state value.

The increase in the amount of combustion noise radiation during stage 1 of the cold starting test is further documented in the indicator diagrams of Figure 4. As can be noticed, the cylinder pressure gradient becomes steeper and the absolute values of peak cylinder pressure higher, owing to the previously mentioned harsher premixed combustion phase; the latter behaviour is also 'assisted' by the increasing amount of injected fuel quantity as the starting proceeds and the fuel pump rack moves to positions corresponding to greater fuelling (Shu and Wei, 2007). Figure 5 further supports this argument by illustrating the respective cylinder pressure spectrum (frequency) diagrams; as expected, since the input signal of the cylinder pressure is decomposed into many harmonics through the Fourier analysis, the 'neat' picture of the indicator diagram is transformed into a much 'richer' frequency spectrum. From Figure 5, it is made obvious that the radiation of combustion noise during stage 1 gets higher practically for the whole frequency spectrum. It should be noted that at the same time, the engine speed increases too (from 177 rpm at Cycle No. 1 to 691 rpm at Cycle No. 13), promoting an additional elevation in the overall engine noise radiation as well (not measured) (Lilly, 1984). This combustion noise radiation manifests itself in the domain from a few hundred up to a few thousand Hz; on the other hand, the corresponding engine firing frequency is, during these cycles much lower, namely up to 11.5 Hz.

Figure 4 Cylinder pressure diagrams for selected cycles during the first stage of the cold starting Test No. 1, and comparison with the fully warmed-up conditions (see online version for colours)



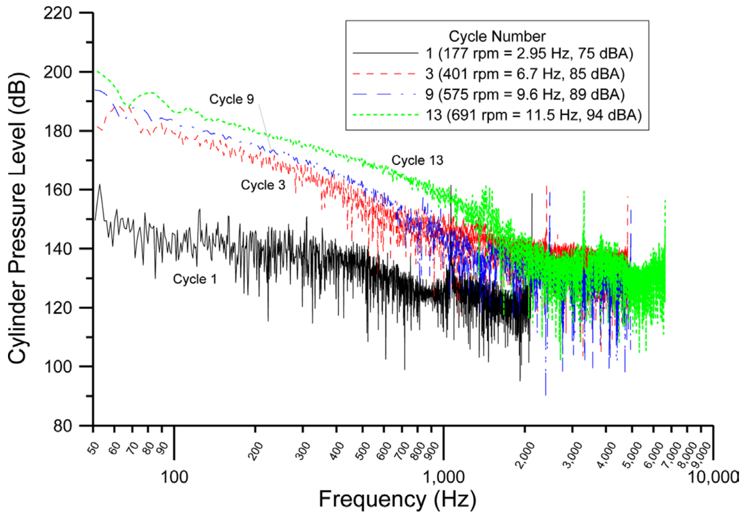
Limitations in the acquisition card sampling frequency prohibited the depiction of pressure level values higher than 8 kHz frequencies; nonetheless, such high frequencies are of negligible importance for the radiated in-cylinder diesel engine combustion noise.

The importance of the engine thermal condition and speed on the amount of emitted combustion noise can be additionally documented by comparing the early starting cycles with the respective idling (i.e., fully-warmed up) operation. As expected, the ignition delay period is shorter during the idling condition (bold line in Figure 4), hence the premixed combustion phase is less steep and the maximum cylinder pressure is lower; the latter are largely attributed to the increase of the cylinder wall temperature, which is adjusting to its steady-state conditions, limiting heat loss and promoting faster mixture preparation (mainly fuel evaporation) and ignition compared with the starting cycles of stage 1. However, the amount of emitted noise during stage 2 does not follow the same decreasing trend with the peak cylinder pressures (upper right and left sub-diagrams in Figure 2). Instead, combustion noise is higher during idling stage 2 compared with the early starting cycles of stage 1. The latter finding can be explained by taking into account:

- the overall higher compression pressure level during the (fully-warmed-up) idling condition
- the operation at a much higher engine speed (of the order of 900 rpm or even more).

Here is one final remark concerning noise development during the starting and the subsequent warm-up stabilisation phase. In general, the irregular behaviour of the maximum cylinder pressure trace (following the combustion, hence in-cylinder pressure irregularity during the whole starting event) is also reflected in a pulsating development of combustion noise radiation around its 'mean' value.

Figure 5 Cylinder pressure frequency spectrum for selected cycles during the first stage of the cold starting Test No. 1 (see online version for colours)

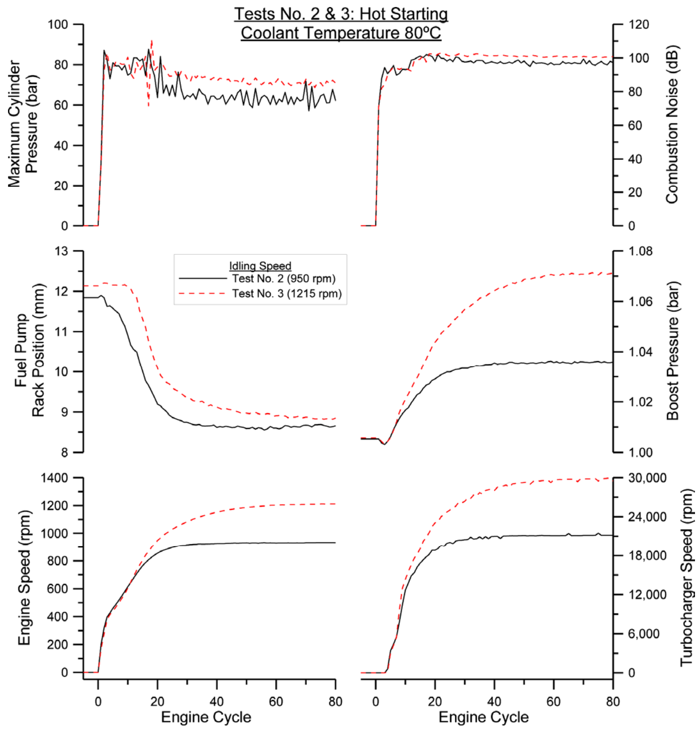


3.2 Hot starting Tests No. 2 and 3

The next two tests were performed under fully warmed-up ('hot') conditions, at two idling speeds (see also Table 2), to also investigate the interesting effect of idling speed on the evolution of combustion instability and noise. Obviously, the much hotter cylinder and exhaust manifold walls were the decisive factors for any differences observed compared with the cold starting Test No. 1. The development of various engine and turbocharger operating parameters as well as the combustion noise radiation for the test cases no. 2 and 3 are presented in Figure 6. Although the final idling speed of Test No. 2 is slightly higher from the cold starting case discussed in the previous sub-section, most of the results are still comparable.

The thermal status of the engine at the fully warmed-up condition as well as the required idling speed affected the turbocharger response, too (middle- and lower-right sub-diagrams in Figures 2 and 6); the latter accelerated much faster compared with the cold starting test, owing to the considerably higher exhaust gas energy content, originating in the (now much) lower heat loss to the already fully warmed-up cylinder and exhaust manifold walls. This can be supported by a direct comparison of the final turbocharger speed reached during Test No. 1 (12,500 rpm) and Test No. 2 (21,000, i.e., difference of the order of 8,500 rpm, or a 68% increase). As a result, turbocharger lag is less decisive now and there exists a smaller discrepancy between fuelling and air-supply that is also mainly responsible for reduced soot emissions (Rakopoulos and Giakoumis, 2009). Finally, and following engineering intuition, both the turbocharger speed and the compressor boost pressure assume higher values for the case of higher engine idling speed (Test No. 3) compared with the lower-idling speed of Test No. 2.

Figure 6 Development of engine and turbocharger variables and combustion noise radiation during hot starting at two idling speeds (Tests No. 2 and 3) (see online version for colours)



The much more favourable in-cylinder conditions during the hot starting tests compared with the cold starting one are also reflected in smaller combustion instability phenomena during the early, decisive starting cycles. This argument is supported by Figure 7 that depicts cylinder pressure development during the first 15 cycles of each hot starting test; moreover, and for better understanding, a direct comparison of the cylinder pressure data between cold and hot starting tests is further illustrated in Figure 8. In this figure, the maximum cylinder pressure and the difference in the peak cylinder pressures between every two consecutive cycles are demonstrated. Clearly, during the hot starting tests:

- the hotter cylinder manifold walls lowered significantly the heat loss from the working medium, allowing faster fuel evaporation and consequently mixture preparation
- the higher lubricant oil temperature reduced the amount of friction losses compared with the cold starting case.

As a result, the risk of combustion failure was significantly reduced too, and any combustion instability phenomena were smoothed; the same holds true for the absolute values of peak cylinder pressures, which were kept at lower levels, too, following the decrease in the ignition delay period. Table 3 summarises the main findings from Figure 8, where the significance of high coolant temperature on cylinder combustion stability is explicitly documented. Closer examination of the results in Figures 7 and 8 and Table 3 reveals also that higher idling speed plays a rather marginal role during the early cycles (comparison between test cases no. 2 and 3). This is not surprising, since it is the cranking system, ambient temperature and compression ratio that are mainly responsible for engine behaviour during the early starting cycles (Henein at al., 1992; Rakopoulos and Giakoumis, 2009).

Figure 7 Combustion instability during the hot starting Tests No. 2 and 3 of Figure 6

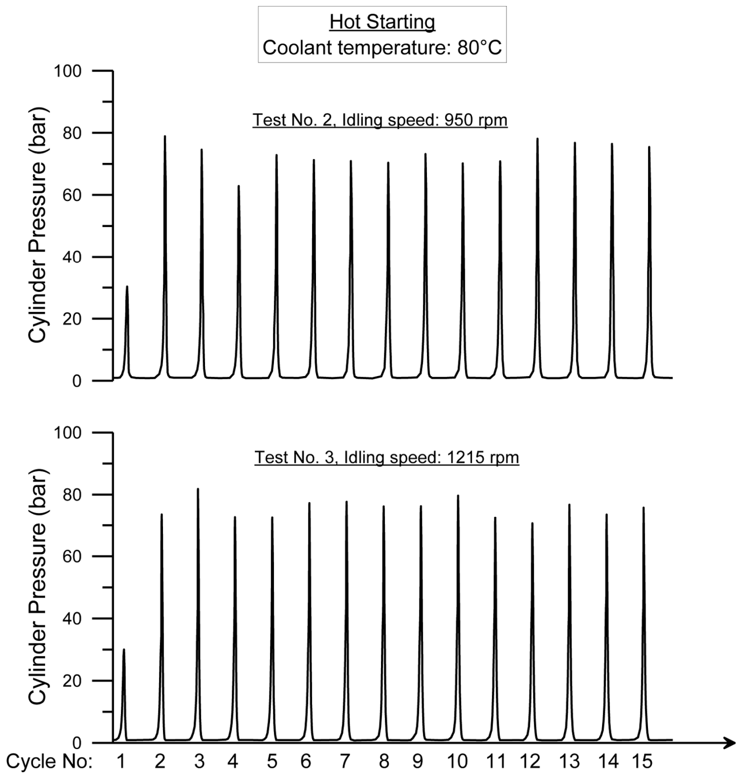


Figure 8 Comparison of cylinder pressure data between the cold starting (No. 1) and the hot starting tests (No. 2 and 3) (see online version for colours)

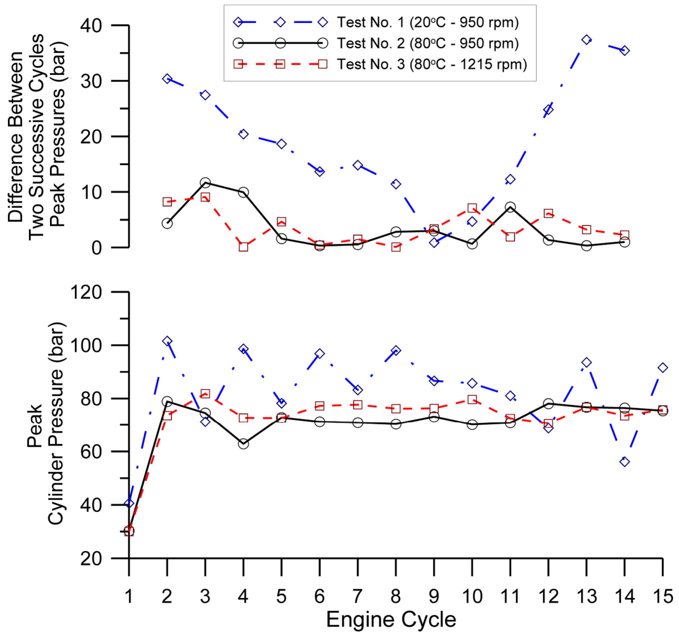
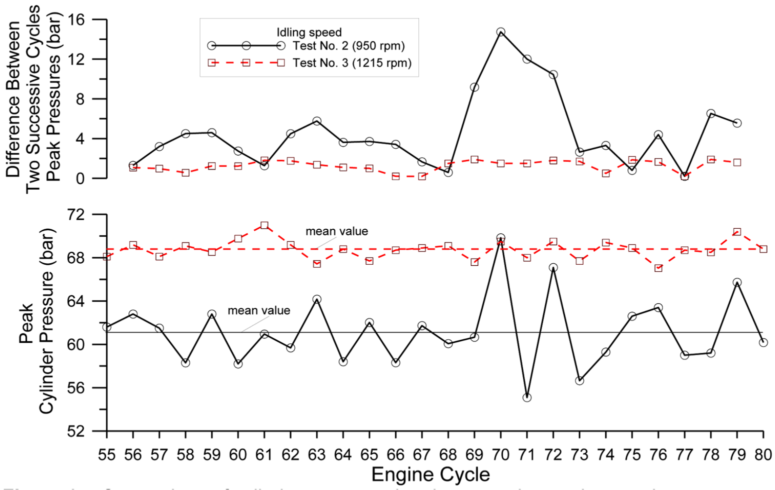


Table 3 Summarisation of statistical cylinder pressure data during the first 15 cycles of each starting test

Test	*Maximum peak cylinder pressure*	*Minimum peak cylinder pressure*	*Mean peak cylinder pressure*	*Peak cylinder pressure standard deviation*	*Maximum difference between two successive cycles peak pressures*	*No of incidents with higher than 20 bar pressure difference between two successive cycles peak pressures*	*No of incidents with higher than 10 bar pressure difference between two successive cycles peak pressures*
1‘cold’	101.6	56.2	85.1	13.14	37.4	5	10
2‘hot’	78.9	62.9	73.3	4.16	11.7	0	1
3‘hot’	81.8	70.6	75.5	3.09	9.1	0	0

To study the effect of idling speed on engine starting behaviour, Figure 9 is provided; it demonstrates elaborated results from 25 successive cylinder pressure diagrams for each hot starting test during the stabilisation phase, i.e., after the desired idling speed had been achieved (from Cycle No. 55 to Cycle No. 80). After the engine speed had stabilised, combustion is clearly more stable during the higher idling speed Test No. 3; this is documented by the almost constant maximum cylinder pressure traces illustrated in the lower sub-diagram of Figure 9 (discontinuous line). The difference in the engine idling speed is the evident reason for this behaviour. The higher engine speed of Test No. 3 produces slightly higher injection pressure and higher compression pressures, while the higher air-supply (documented by the much higher values of turbocharger compressor boost pressure and speed in Figure 6) promotes combustion, at the expense, however, of operation at a higher load and hence, of fuel consumption. On the other hand, the lower idling speed of Test No. 2 causes a relatively higher blow-by loss past the piston rings compared with Test No. 3, and allows more time for heat losses to develop, resulting eventually in the higher (but 'slight' in any case) combustion instability documented in Figure 9, as compared with Test No. 3.

Figure 9 Comparison of cylinder pressure data between the two hot starting tests during idling (see online version for colours)



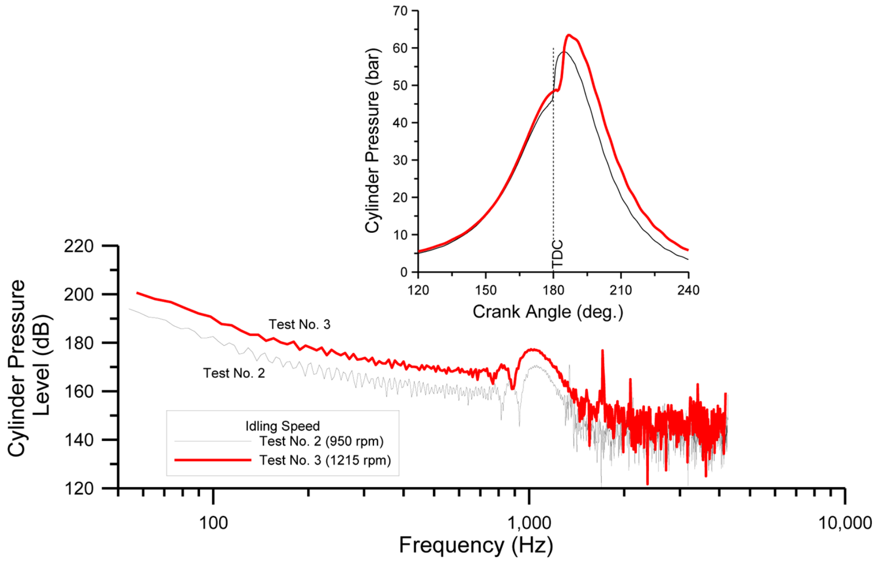
Combustion noise radiation during a hot starting test (upper-right sub-diagram of Figure 6) follows the same general trend discussed previously for the cold starting test, i.e., an initial increase of noise is experienced during the early cycles as the starting event develops and speed and fuelling increase, followed by a stabilisation phase where the noise practically retains the value reached at the end of the previous stage.

Combustion noise seems also to be affected by the desired engine idling speed. The main observation is that the higher idling speed (discontinuous line in Figure 6) induces a higher amount of radiated combustion noise, particularly after the desired idling speed has been achieved (prior to that, it seems that the low engine speed and the starting instability phenomena dominate and suppress any differences). This finding can be explained with reference to Figure 10, where the indicator diagrams for an intermediate-representative cycle during idling of starting Tests No. 2 and 3 are presented together with the respective cylinder pressure spectra. The synergistic effect of

- higher pressures during compression of Test No. 3
- higher fuelling to support the greater frictional losses of the higher load at 1215 rpm speed (see also middle-left sub-diagram of Figure 6)
- higher engine speed of Test No. 3, all contribute to a longer ignition delay period and steeper cylinder pressure rise (i.e., increasing the first derivative of cylinder pressure $dp/d\varphi$).

The latter manifests itself as higher noise radiation, practically for the whole frequency spectrum, but most importantly for the 1–4 kHz region, which is the primarily representative of combustion excitation forces. On the other hand, it is mainly the higher peak pressure of Test No. 3 that is responsible for the higher dB level in the lower frequency range.

Figure 10 Cylinder pressure and pressure frequency spectrum diagrams for two selected cycles during the hot starting tests (see online version for colours)



4 Summary and conclusions

A fully instrumented test bed installation was developed to study the combustion instability and noise emissions of a truck's turbocharged diesel engine during various starting tests conducted at different coolant conditions and idling speeds. A fast response combustion noise-meter was employed for measuring combustion noise radiation. The basic conclusions derived from the current investigation and for the specific engine-hydraulic brake configuration can be summarised as follows:

- Combustion instability was significant mainly during cold starting, with repeatedly great differences in the peak cylinder pressure between successive cycles, reaching up to almost 38 bar.
- The thermal status of the engine and its idling speed played an influencing role on combustion stability and turbocharger response. Specifically, as the engine was operated at hotter coolant temperatures and at higher idling speeds, combustion became more stable, and the turbocharger accelerated faster, producing higher boost pressure.
- The initial low cylinder wall temperature during cold starting prohibited fast mixture preparation (mainly fuel vapourisation) and led to abrupt heat release after the prolonged ignition delay, resulting in steep cylinder pressure rise, high peak

pressures and increased combustion noise. After the achievement of the desired idling speed, and during the subsequent stabilisation phase, it is the higher compression pressures and the higher engine speed that dominate the amount of radiated combustion noise.

- Higher idling speeds induce operation with overall less combustion instability, particularly during the stabilisation phase after achievement of the desired idling speed; however, at the same time, they were found to emit a higher amount of combustion noise.

References

Alt, N., Sonntag, H-D., Heuer, S. and Thiele, R. (2005) *Diesel Engine Cold Start Noise Improvement*, SAE Paper No. 2005-01-2490.

Arcoumanis, C. and Yao, X.G. (1994) 'Transient smoke and unburnt hydrocarbon emissions during cold-start in a turbo-charged DI diesel engine', *Institution of Mechanical Engineers, Seminar on 'Transient Performance of Engines'*, London, UK, pp.43–60.

Austen, A.E.W. and Priede, T. (1958) 'Origins of diesel engine noise', *Institution of Mechanical Engineers, Symposium on 'Engine Noise and Noise Suppression'*, London, UK, pp.19–32.

AVL (2000) *AVL 450 Combustion Noise Meter Operating Manual*, AVL, August.

Bielaczyc, P., Merkisz, J. and Pielecha, J. (2001) *Investigation of Exhaust Emissions From Di Diesel Engine During Cold and Warm Start*, SAE Paper No. 2001-01-1260.

Bozza, F., Costa, M. and Siano, D. (2009) 'Design issues concerning thermofluidynamic and acoustic aspects in a diesel engine suitable for aeronautical applications', *International Journal of Vehicle Design*, Vol. 45, pp.152–181.

Cheng, K.Y., Shayler, P.J. and Murphy, M. (2004) 'The influence of blow-by on indicated work output from a diesel engine under cold start conditions', *Proceedings of the Institution of Mechanical Engineers, Part D, Journal of Automobile Engineering*, Vol. 218, pp.333–340.

Dhaenens, M., van der Linden, G., Nehl, J. and Thiele, R. (2001) *Analysis of Transient Noise Behaviour of a Truck Diesel Engine*, SAE Paper No. 2001-01-1566.

Goldwine, G. and Sher, E. (2009) 'Experimental investigation of the effect of split injection on diesel engine performance', *International Journal of Vehicle Design*, Vol. 50, pp.66–90.

Head, H.E. and Wake, J.D. (1980) *Noise of Diesel Engines Under Transient Conditions*, SAE Paper No. 800404.

Henein, N.A., Zahdeh, A.R., Yassine, M.K. and Bryzik, W. (1992) *Diesel Engine Cold Starting: Combustion Instability*, SAE Paper No. 920005.

Heywood, J.B. (1988) *Internal Combustion Engine Fundamentals*, McGraw-Hill, New York.

Kobayashi, A., Kurashima, A. and Endo, S. (1984) *Analysis of Cold Start Combustion in a Direct Injection Diesel Engine*, SAE Paper No. 840106.

Kondo, M., Kimura, S., Hirano, I., Uraki, Y. and Maeda, R. (2001) 'Development of noise reduction technologies for a small direct-injection diesel engine', *JSAE Review*, Vol. 21, pp.327–333.

Lilly, L.R.C. (1984) *Diesel Engine Reference Book*, Butterworths, London.

Osuka, I., Nishimura, M., Tanaka, Y. and Miyaki, M. (1994) *Benefits of New Fuel Injection Technology on Cold Startability of Diesel Engines – Improvement of Cold Startability and White Smoke Reduction by Means of Multi Injection with Common Rail Fuel System (Ecd-U2)*', SAE paper No. 940586.

Phatak, R. and Nakamura, T. (1983) *Cold Startability of Open Chamber Direct Injection Diesel Engines. Part I: Measurement Technique and Effects of Compression Ratio*, SAE Paper No. 831335.

Rakopoulos, C.D. and Giakoumis, E.G. (2006) 'Review of thermodynamic diesel engine simulations under transient operating conditions', *SAE Transactions, Journal of Engines*, Vol. 115, pp.467–504, also SAE Paper No. 2006-01-0884.

Rakopoulos, C.D. and Giakoumis, E.G. (2009) *Diesel Engine Transient Operation*, Springer, London.

Rakopoulos, C.D. and Mavropoulos, G.C. (2009) 'Effects of transient diesel engine operation on its cyclic heat transfer: an experimental assessment', *Proceedings of the Institution of Mechanical Engineers, Part D, Journal of Automobile Engineering*, Vol. 223, pp.1373–1394.

Shu, G. and Wei, H. (2007) 'Study of combustion noise mechanism under accelerating operation of a naturally aspirated diesel engine', *International Journal of Vehicle Design*, Vol. 45, pp.33–47.

Tsunemoto, H., Yamada, T. and Ishitani, H. (1985) *The Transient Performance During Acceleration in a Passenger Car Diesel Engine at the Lower Temperature Operation*', SAE Paper No. 850113.

Watanabe, Y., Fujisaki, H. and Tsuda, T. (1979) *Di Diesel Engine Becomes Noisier at Acceleration – The Transient Noise Characteristic of Diesel Engine*, SAE Paper No. 790269.

Note

[1]Due to the fact that the engine was coupled to a hydraulic dynamometer (torque rises with respect to speed squared), the idling speed of 1215 rpm of Test No. 3 corresponds to almost 63% higher load compared with the speed of 950 rpm of Test No. 2.