Study of the Effects of some critical Engine Parameters on Performance and Emitted Pollutants of a Light-Duty, Dual Fuel Compression Ignition Engine - a Theoretical Approach

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Abstract. Reduction of exhaust emissions is a major research task in diesel engine development in view of increasing concern regarding environmental protection and stringent exhaust gas regulations. Simultaneous reduction of NOx emissions and particulate matter is quite difficult due to the soot/NOx trade-off and is often accompanied by fuel consumption penalties. Towards this aim, automotive engineers have proposed various solutions, one of which is the use of alternative gaseous fuels as a supplement for the commercial liquid diesel fuel. This type of engine, which operates fuelled simultaneously with conventional diesel oil and gaseous fuel, is called “dual fuel” diesel engine. The majority of the dual fuel diesel engines operate on the fumigation principle where the liquid diesel fuel is partially replaced by gaseous one fumigated into the intake air. One of the gaseous fuels used commonly in compression ignition engine is natural gas, which has a relatively high auto- ignition temperature and moreover is an economical and clean burning fuel. The high auto – ignition temperature of natural gas is a serious advantage against other gaseous fuels since the compression ratio of most conventional DI diesel engines can be maintained. Moreover the combustion of natural gas produces practically no particulates since natural gas contains less dissolved impurities (e.g. sulfur compounds). Previous research studies have shown that the natural gas-diesel fuel dual fuel combustion in a CI engine environment, compared to conventional diesel fuel operation, suffers from high specific fuel consumption, and high carbon monoxide (CO) and unburned hydrocarbons (HC) emissions. Compression ratio and diesel fuel injection timing are two engine parameters, which can influence considerably the combustion process inside the combustion chamber of a dual fuel CI engine. Hence, in the present study, a theoretical investigation is conducted using an engine simulation model to examine the effect of the aforementioned parameters on performance and exhaust emissions of a natural gas/diesel engine. Predictions are produced for a light-duty, medium-speed, dual fuel (natural gas/diesel) compression ignition engine performance characteristics and NO, CO and Soot emissions at specific engine operating point (load & engine speed), by using a comprehensive two-zone combustion model. The main objective of this comparative assessment is to elaborate the relative impact of each one of the above mentioned parameters on engine performance characteristics and exhaust emissions. The conclusions of this study may be proven to be considerably valuable for the application of this technology on existing DI diesel engines.

Keywords: Dual fuel diesel engine; Compression ratio; Injection timing; Performance; Emissions.
PACS: 88.85.mf

INTRODUCTION

With the increasing public interest in energy supply and the environment, attention has focused on the development of ecological and efficient combustion technologies [1-6]. One of these technologies could be the use of natural gas as supplement for the conventional diesel fuel (dual fuel natural gas diesel engine), owing to its inherent clean nature of combustion combined with the high availability at attractive prices [1-6]. For the majority of dual fuel natural gas diesel engines, natural gas, which is the primary fuel, is most usually inducted with the air during the induction stroke (fumigated dual fuel natural gas-diesel engine). Fumigated dual fuel natural gas-diesel engines feature essentially a homogeneous natural gas–air mixture compressed rapidly below its auto-ignition condition and ignited by the injection of a pilot amount of liquid diesel fuel, which represents less than 20% of the total energy released at full engine load operation, around top dead center position [1-3, 6]. The specific type of engine is referred to as pilot ignited natural gas diesel engine. Under constant engine speed, the power output of the specific type of engine is controlled by changing only the amount of the primary gaseous fuel added to the inducted air which results to a change of the combustion air composition since the total amount of the inducted mixture is kept constant.

A substantial research (experimental and theoretical) on pilot ignited natural gas diesel engines has concentrated on the extent of dual-fuelling and its effect on emissions and performance [7-10]. According to the literature the pilot ignited natural gas-diesel engine operation, compared to the conventional diesel one, is found to be an effective way in the simultaneous reduction of nitrogen oxide emissions and particulate matter as well, but the specific engine operation suffers from high unburned hydrocarbon (HC) and carbon monoxide (CO) emissions and also from poor performance, especially at low and intermediate load operating points [1-3, 6, 7-10]. At high load, the improvement of the natural gas utilization leads to a relevant improvement of the specific engine operating characteristics.

According to the literature, alterations in various engine operating parameters, such as the pilot fuel quantity [11-14], the pilot diesel fuel injection timing [15-18], the use of exhaust gas recirculation [19-21], the engine compression ratio [22] etc, may be used to improve the engine efficiency and to restrain the increase of CO and HC emitted from a compression ignition engine running under pilot ignited natural gas diesel operating mode.

Thus, the primary objective of the present work is to examine, on a theoretical basis, the effect of engine compression ratio and liquid fuel injection timing on the performance and emissions of a single cylinder, medium-speed, direct injection compression ignition dual fuel engine. The evaluation of the calculations revealed the applicability of each technique on an existing DI diesel engine operating under dual fuel mode. The theoretical results have been produced by using a two-zone phenomenological combustion model, which predicts in-cylinder pressure and heat release rate histories as well as soot and NO concentration profiles. Theoretical results are validated against experimental values, which were obtained at fixed engine operating point (load & engine speed), from a single cylinder DI diesel engine operating under dual fuel mode at fixed liquid fuel quantity and normal injection advance. Moreover, model predictions are contrasted with additional data obtained from the international literature [15-21] to ensure that the two-zone model predicts with reasonable accuracy the effect of both parameters on engine performance characteristics, soot and NO emissions. This comparison revealed that the developed model captures more in a qualitative rather than in quantitative manner the influence of both parameters on engine performance and exhaust emissions.

From the theoretical findings, important information is derived revealing the effect of both parameters on engine performance and pollutant emissions. This is accomplished through the comparison of the calculated maximum cylinder pressure and total brake specific fuel consumption for various combinations of diesel fuel injection timings and engine compression ratios, at fixed engine operating point. Furthermore, the effect of both parameters is revealed on the formation of pollutant emissions, by comparing the related values to the corresponding ones obtained under normal compression ratio (i.e. CR = 14.54) and normal diesel fuel injection timing (i.e. DIT = 180 deg CA). This information is extremely valuable, if one wishes to determine the proper combination of both engine parameters to improve the engine behavior at specific operating conditions (speed and load) under dual fuel mode.

BRIEF OUTLINE OF THE MODEL

The engine simulation used in the present study is based on a phenomenological two-zone model, which has been successfully used in the past to simulate the performance and emissions characteristics of CI engines, operating under natural gas-diesel dual fuel conditions [12-16, 22]. The model examines the closed part of the engine cycle and its basic philosophy is given in the flow chart diagram presented in Figure 1. More details about the proposed simulation model are given in the references [12-16, 22]

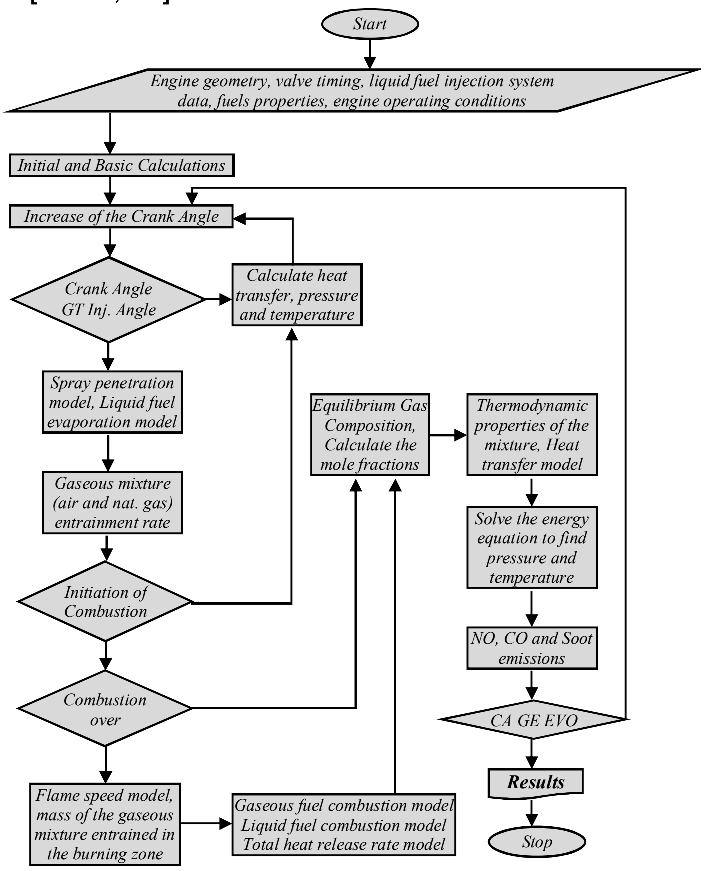


FIGURE 1. The flow chart diagram of the simulation model..

MODEL VALIDATION

To verify the ability of the model to predict, apart from overall performance characteristics, the NO and soot emissions of a diesel-natural gas dual fuel compression ignition engine, an extended theoretical and experimental investigation has been conducted in the past [9-10, 12-16, 22].

TABLE (1). Specifications for "ENG - A" and "ENG - B" research test engines.

	ENG - A	**ENG - B**
Engine Type	CI, Turbocharged	CI, Turbocharged
Number of cylinders	1	1
Bore	0.128 m	0.137 m
Stroke	0.142 m	0.171 m
Compression ratio	17:1	14.54
Number of injector holes	8	5
Injector hole diameter	0.197 mm	0.215 mm
Injection Pressure	500 bar	200 bar
Inlet Valve Opening	243 deg BTDC	30 deg BTDC
Inlet Valve Closure	168 deg BTDC	155 deg BTDC
Exhaust Valve Opening	186 deg ATDC	114 deg ATDC
Exhaust Valve Closure	367 deg ATDC	29 deg ATDC
Pressure at IVC	1.55 bar	2.7 bar

To further evaluate the predictive ability of the proposed model, results are used from an experimental investigation of diesel-natural gas dual fuel combustion conducted on two single cylinder research test engines (i.e. Eng-A and Eng-B). The main geometrical and operational data of these engines are given in Table 1. These results are also used as a basis to evaluate the model predictions concerning the effect of compression ratio and diesel fuel injection timing, on the performance and pollutant emissions characteristics of the diesel-natural gas research test engine "ENG-B".

For the engine "ENG-A", all measurements were taken at constant engine speed of 1500 rpm and part load conditions (i.e. 70% of full load), under various percentages of natural gas mass ratios (i.e. "x(%)"), which represents the quotient of the mass flow rate of the gaseous fuel divided by the total fuel (i.e. diesel fuel and natural gas) mass flow rate and it is given by the formula:

$$x(\%) = \frac{\dot{m}_g}{\dot{m}_g + \dot{m}_d} \cdot 100\% \quad .$$

Moreover, for "ENG-A" all experimental measurements involved in this study refer to the operation of the specific engine with constant compression ratio (i.e. CR = 17) and constant liquid diesel fuel injection timing (i.e. DIT = 175 deg CA).

Furthermore, for the engine "ENG-B", all measurements were taken at constant engine speed of 1800 rpm and full load conditions (i.e. 100% of full load), under constant natural gas mass ratio (i.e. x = 90%), while the specific engine was running with constant compression ratio (i.e. CR = 14.54) and constant liquid diesel fuel injection timing (i.e. DIT = 180 deg CA).

For both engines (i.e. "ENG-A" and "ENG-B"), the experimental measurements included engine speed, brake torque, air mass flow rate, diesel fuel mass flow rate, gaseous fuel mass flow rate, crank angle-resolved cylinder pressure histories and exhaust emissions (i.e. NO and soot). Post-processing of the experimental measurements yielded indicated mean effective pressure, indicated power, net apparent rate of heat release, indicated specific energy consumption, and indicated specific NO and soot concentrations.

For validating the ability of the developed model to predict engine performance, a comparison between experimental and calculated cylinder pressure traces for both engines used (i.e. "ENG-A" and "ENG-B"), is given in Figs. 2-3.

For "ENG-A" the comparison is given at 1500 rpm engine speed and 70% of full engine load conditions, for x = 88% natural gas mass flow ratio under constant DIT = 175 deg CA, while for the "ENG-B" the respective comparison is given at 1800 rpm engine speed and full engine load conditions, for x = 90% natural gas mass flow ratio under constant DIT = 180 deg CA.

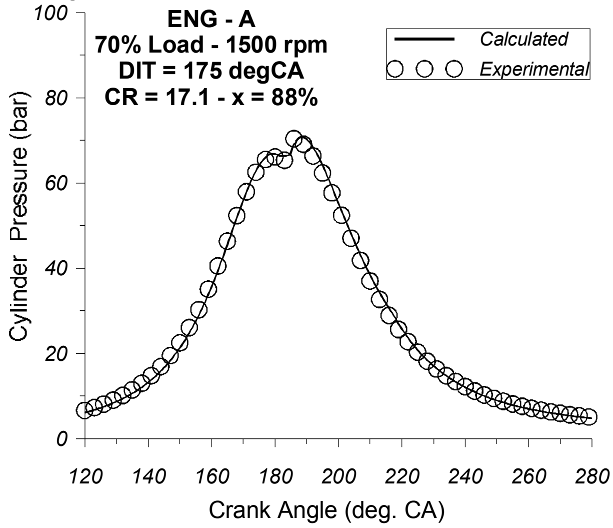


FIGURE 2. Comparison between experimental and computed cylinder pressure traces of the test engine "ENG-A", operating at 1500 rpm engine speed and part load, for x = 88%.

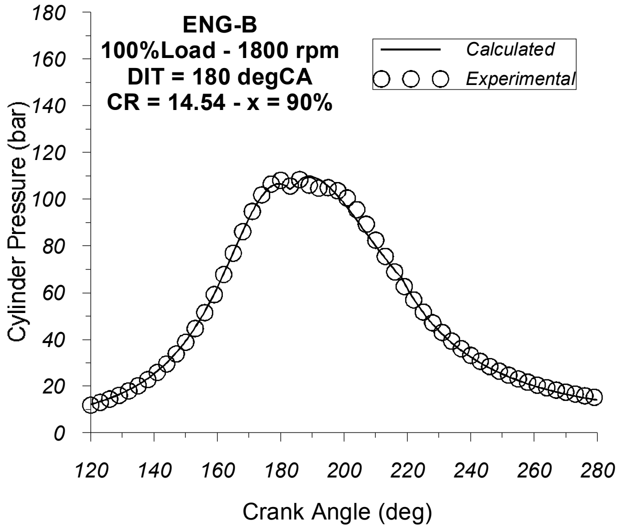


FIGURE 3. Comparison between experimental and computed cylinder pressure traces of the test engine "ENG-B", operating at 1800 rpm engine speed and full load, for x = 90%.

Comparing the experimental and the theoretical cylinder pressure diagrams shown in Figs. 2-3, it is observed that the agreement in both cases is relatively good, revealing the ability of the current model to predict adequately the engine performance for CI engines operating under dual fuel conditions. Model's ability to predict engine performance is quite encouraging, taking into account the complicated combustion process of natural gas in the presence of diesel in a compression ignition environment.

The comparison between experimental and calculated values of some of the most important engine performance characteristics of the "ENG-A", i.e. indicated mean effective pressure and the indicated total specific fuel consumption, is given in Figs. 4-5. The experimental total indicated specific fuel consumption is estimated from the measured indicated power output and the measured mass flow rates of both types of fuel used.

ISSN:1791-4469

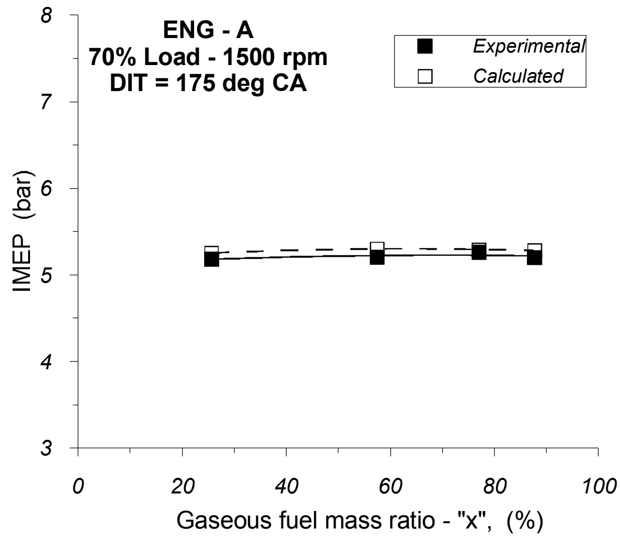


FIGURE 4. Comparison between calculated and experimental indicated mean effective pressure of "ENG-A", operating at 1500 rpm and part load, for various "x" values

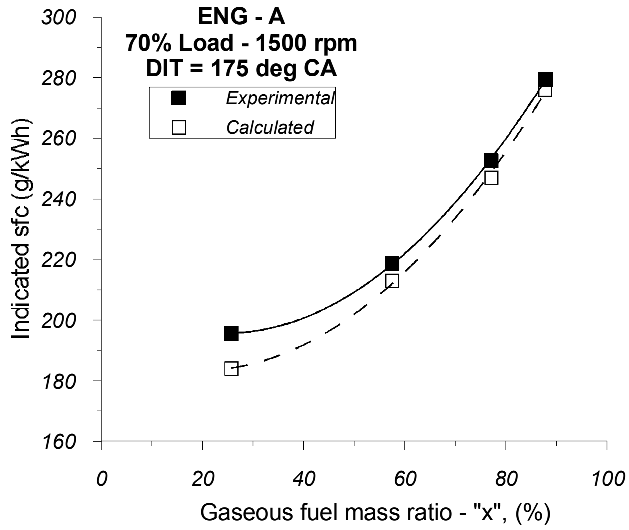


FIGURE 5. Comparison between calculated and experimental indicated specific fuel consumption of "ENG-A", operating at 1500 rpm and part load, for various "x" values

Observing the results given in Figs. 4-5, a very good coincidence is shown between experimental and calculated values, thus revealing the ability of the model to predict adequately the cylinder pressure histories for dual fuel combustion for various gaseous fuel mass flow ratios. Moreover, by observing Fig. 5, it is seen that for all gaseous fuel mass ratios examined, the proposed model predicts with relatively good accuracy the experimental indicated engine efficiency.

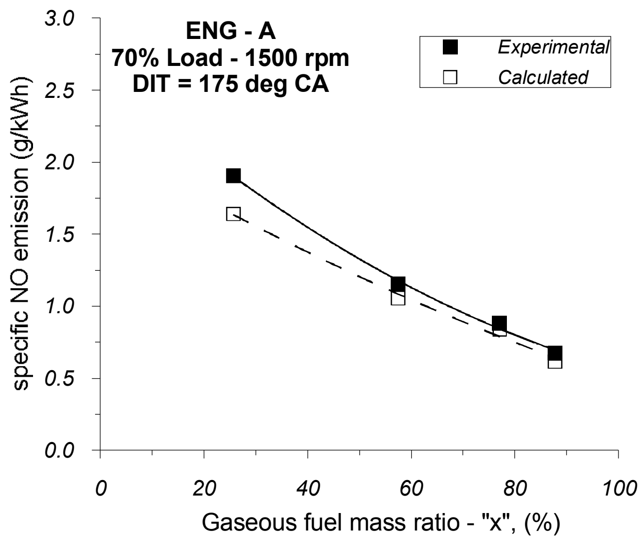


FIGURE 5. Comparison between calculated and experimental indicated specific NO emissions of "ENG-A", operating at 1500 rpm and part load, for various "x" values

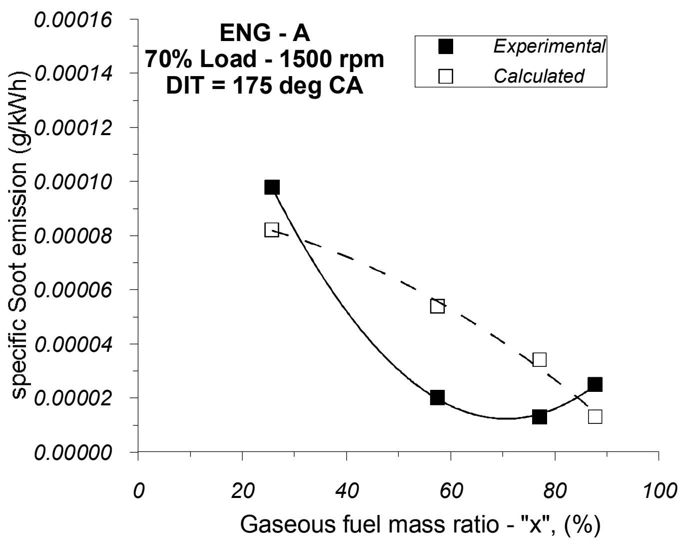


FIGURE 6. Comparison between calculated and experimental indicated specific soot emissions of "ENG-A", operating at 1500 rpm and part load, for various "x" values

The variation of the measured and calculated NO and soot emissions of the engine "ENG-A" with the gaseous fuel mass ratio, is given in Figs. 5-6, respectively. Examining these figures, it is revealed that the model predicts with adequate accuracy the trends of specific NO and soot emissions with gaseous fuel mass ratio. Specifically, for the NO emissions, the theoretical values are slightly slower compared to the experimental ones, which is an expected fact by a two-zone model as it under-predicts the burning zone temperature, which, as known, affects seriously the formation mechanism of NO emissions. By examining Figs. 5, it is observed that the increase of the amount of the gaseous fuel, results in a decrease of NO concentration. This is the result of the lower concentration of oxygen inside the burning zone due to the increased amounts of the gaseous fuel and, also, to the lower burning zone temperatures.

Observing Fig. 6, it is revealed that the model predicts with relatively good accuracy the variation of the measured values of soot emissions with the gaseous fuel mass ratio. Despite the fact that the model does not manage to predict accurately the shape of the experimental soot curve, it is especially encouraging that the effect of supplement ratio is predicted in a

ISSN:1791-4469

correct manner. Moreover, it is revealed that under part loading conditions, the increase of the gaseous fuel mass ratio leads to a decrease of soot emissions. This is, probably, because less amount of liquid fuel is injected.

Consequently, from the comparison between the experimental and theoretical results, it is evident that despite the differences observed between the measured and calculated absolute values, which is primarily attributed to the inherent limitations of the phenomenological two-zone model, the simulation model manages to predict with adequate accuracy the trend of engine performance characteristics and pollutant emissions with the gaseous fuel mass ratio. Hence, the specific phenomenological model can be used to perform a parametric study concerning the effects of the compression ratio and the liquid fuel injection timing on the performance and pollutant emissions characteristics of the engine "ENG-B", operating under dual fuel (natural gas - diesel) mode, at 1800 rpm engine speed and full load conditions (i.e. 100% Load), for fixed gaseous fuel mass ratio (x = 90%).

TEST CASES EXAMINED

After the successful calibration and validation of the developed model by the existing experimental data, it was used for investigating theoretically the relative impact of compression ratio (CR) and diesel fuel injection timing (DIT) on the performance and exhaust emissions of a single cylinder, compression ignition, dual fuel (diesel-natural gas) research test engine (i.e. "ENG-B"). Thus, at fixed gaseous fuel mass ratio (i.e. x = 90%), for constant engine speed (i.e. 1800 rpm) and full load conditions, the effect of compression ratio was initially investigated under the normal diesel fuel injection timing (i.e., DIT = 180 deg CA). On this basis, it was selected to investigate the effect that will have the use of a lower (CR = 14.00) and a higher value of the compression ratio (CR = 15.50), in comparison to the normal one (CR = 14.54), on the performance and exhaust emissions of the "ENG-B" operating under dual fuel mode. Subsequently, the effect of diesel fuel injection timing was examined under all the three CR ratios examined. Thus, diesel fuel injection timing was changed by two, four, six, eight and ten degrees crank angle before the normal diesel injection timing (i.e. DIT = 178, 176, 174, 172 and 180 deg CA).

RESULTS AND DISCUSSION

In this section, the predicted effects of the compression ratio and the diesel fuel injection timing, are explored, on performance characteristics and pollutant emissions of the single research test engine "ENG-B" operating under dual fuel (i.e. natural gas – diesel) mode, at 1800 rpm engine speed and full load conditions, for fixed gaseous fuel mass ratio (i.e. x = 90%).

Effects of CR and DIT on Engine Performance

Effect on cylinder pressure and heat release rate histories

Figs. 7a-b and 8a-b depict the effect of CR on predicted cylinder pressure and heat release rate traces, for DIT = 180 deg CA and DIT = 176 deg CA, respectively.

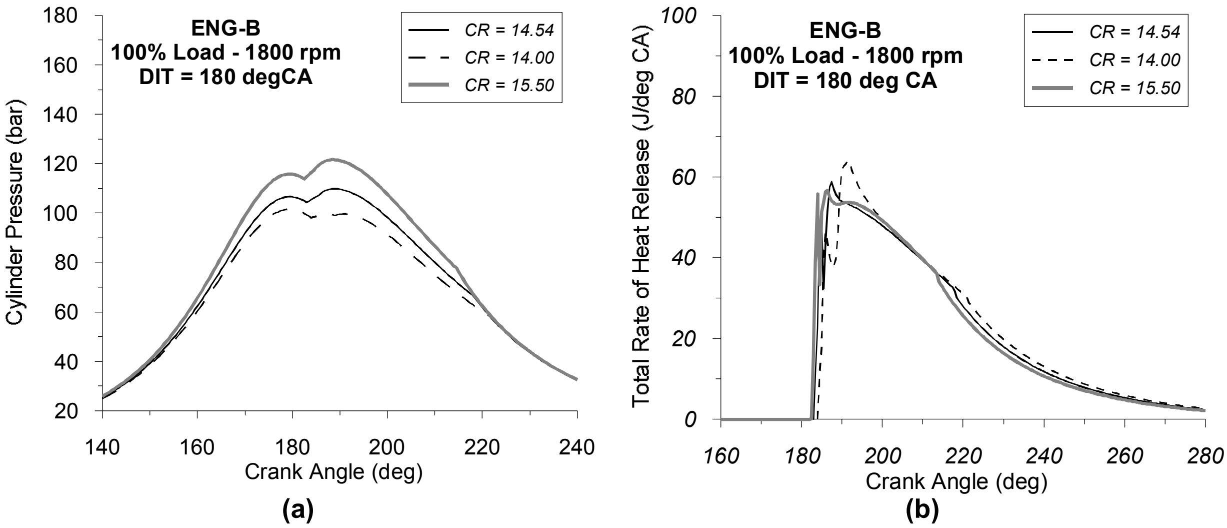


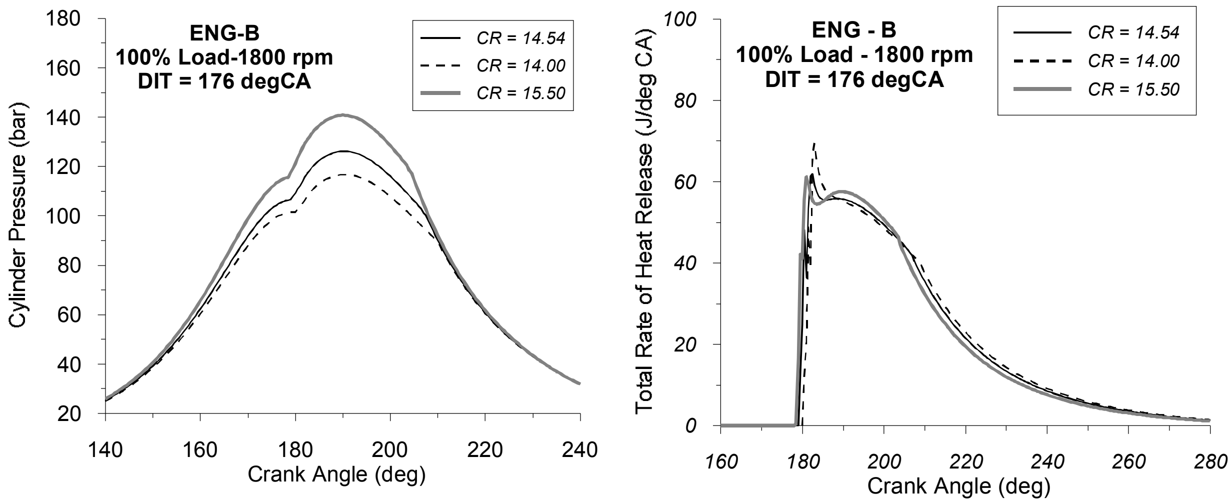
FIGURE 7. Calculated cylinder pressure and heat release rate diagrams of "ENG-B", for various compression ratios with DIT = 180 deg CA, at 1800 rpm and full load, for x = 90%

Observing Figs. 7a and 8a, it is revealed that the compression ratio affects the cylinder pressure histories. Thus, for the same engine operating conditions, during the compression stroke the cylinder pressure corresponding to the lower value of the compression ratio diverges from the respective values observed with increased compression ratio. The difference becomes more evident during the last stages of the compression stroke. After the initiation of combustion, the rate of cylinder pressure rise with increased compression ratio during the initial stage of the combustion process becomes higher, while the peak of the cylinder pressure occurs slightly earlier compared to the respective values observed for CR = 14. This could be attributed to the advance of the initiation of combustion, higher local temperatures, and faster overall combustion.

By examining Figs. 7b and 8b, it is clear that the compression ratio does not seem to affect considerably the shape of the total rate of the heat release. However, at the same time, by increasing the compression ratio, combustion initiates slightly earlier since higher cylinder charge temperatures lead to shorter ignition delay periods, without significantly prolonging or shortening the duration of combustion. It must be stated here, that the duration of combustion is estimated as the crank angle interval between the initiation of combustion and the crank angle where combustion has been completed approximately by 98% (i.e. 98% of cumulative heat release).

Furthermore, it is revealed that the initial peak in the heat release curve corresponding to the higher value of the compression ratio (i.e. CR = 15.5) is lower and appears slightly earlier compared to the respective one corresponding to CR = 14. It results from the lower amount of diesel fuel burned during the premixed controlled combustion phase, probably due to the shorter ignition delay period, and also to the fact that the combustion of gaseous fuel has not yet progressed sufficiently, since the cylinder charge conditions do not favor the spread of the flame front. As far as the second phase of the combustion process is concerned, it is revealed that the total burning rate during this specific phase is higher compared to that observed under CR = 14. This is the result of improvement of the gaseous fuel combustion quality that is caused by the improvement of the cylinder charge conditions (i.e. cylinder gas temperature, etc.), which contributes significantly to the existence and fast spread of the flame front surrounding the burning zone. However, this does not have any significant effect on the cylinder pressure, since the major part of the gaseous fuel combustion occurs during the expansion stroke.

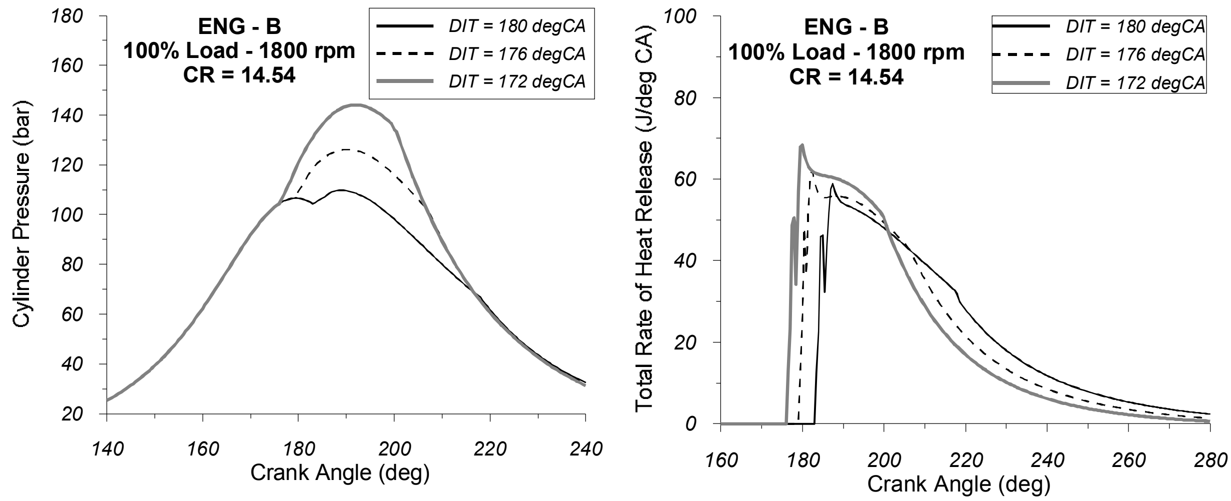
ISSN:1791-4469



(a) (b)

FIGURE 8. Calculated cylinder pressure and heat release rate diagrams of "ENG-B", for various compression ratios with DIT = 176 deg CA, at 1800 rpm and full load, for x = 90%

Figs. 9a-b and 10a-b provide the predicted cylinder pressure and the total heat release rate, for two compression ratios (i.e. CR = 14.54 and CR = 15.5) and three values of the diesel injection timing, i.e. DIT = 180, 176 and 172 deg CA, at 1800 rpm engine speed and full load conditions, for gaseous fuel mass ratio x = 90%, respectively.



(a) (b)

FIGURE 9. Calculated cylinder pressure and heat release rate diagrams of "ENG-B", for various diesel fuel injection timings with CR = 14.54, at 1800 rpm and full load, for x = 90%

By observing Figs. 9a and 10a, it is revealed that the diesel injection timing affects the cylinder pressure history. Thus, by advancing the diesel injection timing, the rate of cylinder pressure rise during the initial stage of the combustion process becomes higher, while the peak cylinder pressure occurs slightly earlier. This is the result of both the earlier initiation of combustion and the higher premixed controlled combustion rate of the liquid fuel, which occurred due to the longer diesel fuel ignition delay period, a fact that affects positively the amount of the liquid fuel prepared for burning.

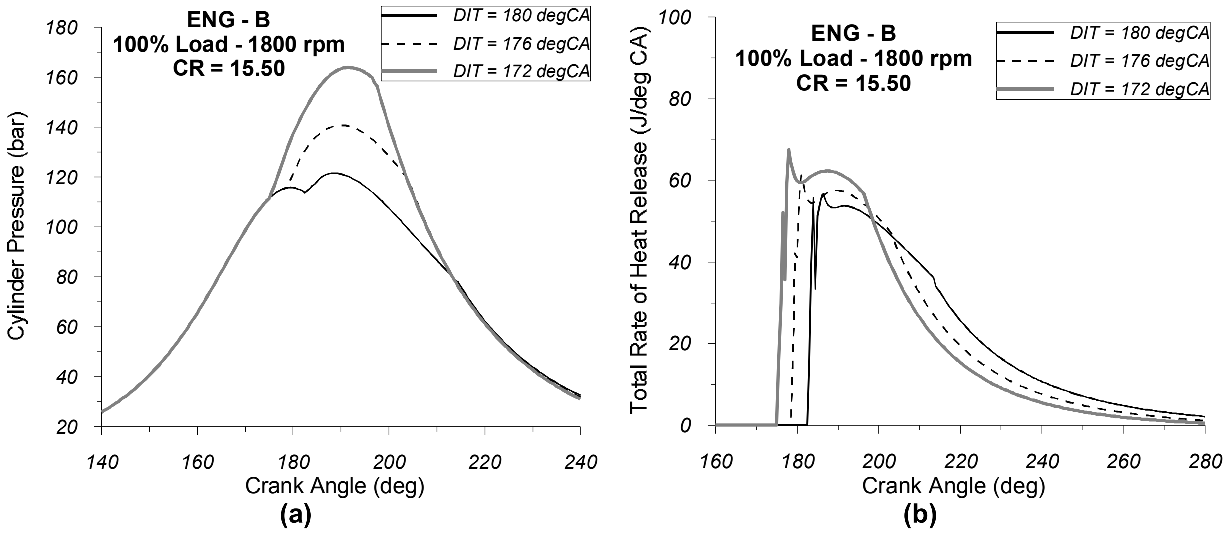


FIGURE 10. Calculated cylinder pressure and heat release rate diagrams of "ENG-B", for various diesel fuel injection timings with CR = 15.5, at 1800 rpm and full load, for x = 90%

Examining Figs. 9b and 10b, it is shown that the injection timing does not affect seriously the shape of the heat release rate. However at the same time, by advancing the injection timing combustion initiates slightly earlier, while the rate of the heat release rise during the initial stages of combustion becomes higher. The latter is the result of the increased liquid fuel mass burned during the premixed controlled combustion phase (i.e. before top dead center), since the advance of the liquid fuel injection timing leads to longer ignition delay periods, a fact that leads to higher liquid fuel amounts prepared for combustion. This affects also positively the existence and the fast spread of the flame front surrounding the burning zone.

Effect on indicated power output

In Fig. 11 results are given concerning the variation of the indicated power versus diesel fuel injection timing for all three values of the compression ratio (i.e. CR = 14, 14.54 and 15.5).

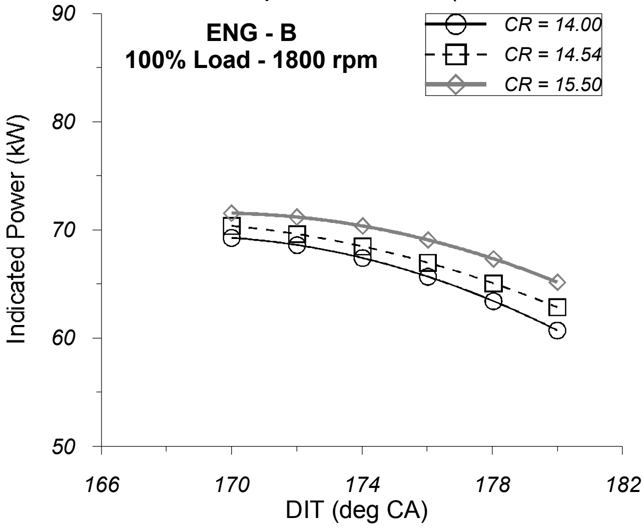


FIGURE 21. Calculated indicated power for "ENG-B", operating at 1800 rpm engine speed and full load conditions, for various compression ratios.

ISSN:1791-4469

All predictions are given at 1800 rpm engine speed and full load and for 90% gaseous fuel mass ratio. By examining this figure, it is revealed that, the advance of the diesel fuel injection timing seems to affect more strongly the indicated power output compared to the relative impact of the compression ratio. Thus, engine power output could be further improved by advancing the diesel fuel injection timing.

Effect on maximum cylinder pressure

The variation of the maximum cylinder pressure versus diesel fuel injection timing, for all three values of the compression ratio, is shown in Figure 12. The study of the effect of compression ratio and diesel fuel injection timing on maximum cylinder pressure is of particular interest, since it is a critical parameter affecting the mechanical integrity of the engine structure. By examining Fig. 12, it is revealed that by keeping constant the normal diesel injection timing, the increase of compression ratio leads to an increase of the maximum cylinder pressure. This is attributed to both the higher pressures achieved during the compression process before the onset of combustion and the enhancement of the gaseous fuel combustion rate.

On the other hand, by advancing diesel injection timing, keeping constant the compression ratio, a considerable increase of the maximum cylinder pressure is also observed. This is primarily attributed to the fact that the advance of the diesel fuel injection timing leads to an increase of the liquid fuel amount burned before TDC. Finally, it is revealed that the aforementioned change of both parameters could lead to a significant problem with regard to the mechanical strength of the engine, since a significant increase in maximum cylinder pressure is then observed.

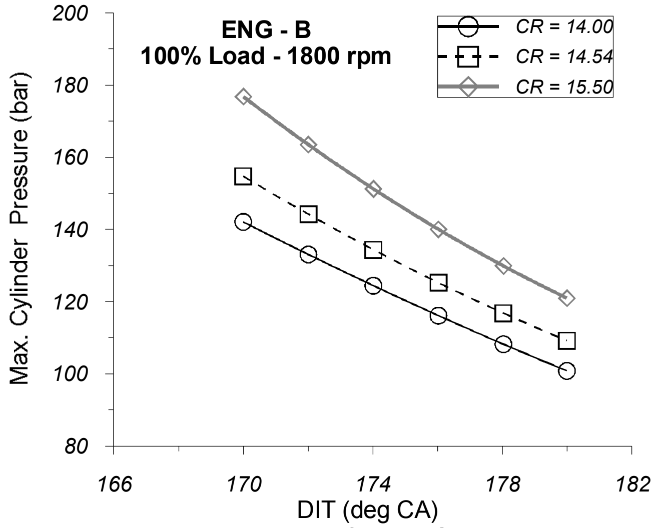


FIGURE 32. Calculated maximum cylinder pressure for "ENG-B", operating at 1800 rpm engine speed and full load conditions, for various compression ratios..

Effect on indicated total specific fuel consumption (ISFC)

Figure 13 illustrate the variation of the calculated indicated total specific fuel consumption versus diesel fuel injection timing for all three values of the compression ratios, examined. All predictions are given at 1800 rpm engine speed and full load conditions, for 90% gaseous fuel mass ratio. For each test case examined, the total indicated specific fuel consumption is estimated from the calculated indicated power output and the measured mass flow rates of diesel and natural gas. The indicated power output was calculated by the simulation model using the predicted indicated mean effective pressure, as computed from the integration of the predicted cylinder pressure diagram.

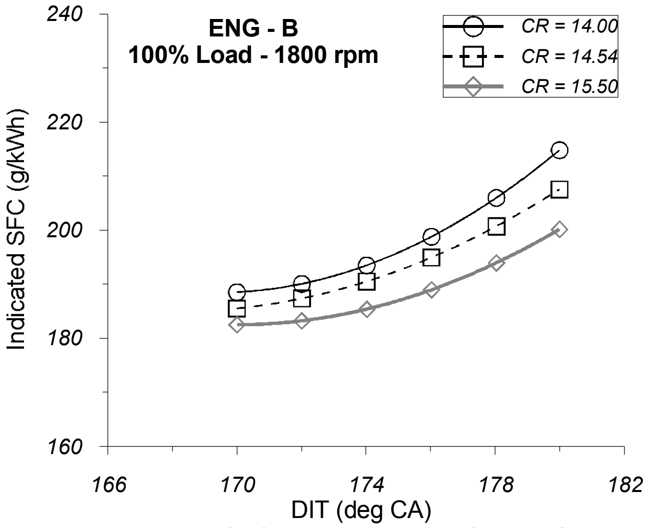


FIGURE 43. Calculated indicated specific fuel consumption for "ENG-B", operating at 1800 rpm engine speed and full load conditions, for various compression ratios.

Observing Fig. 13, it is evident that each one of the examined parameters affects the indicated engine efficiency. Specifically, by examining Fig. 13, it is observed that the increase of compression ratio leads to an improvement of the indicated thermal engine efficiency. This improvement may be attributed both to the increase of the indicated power output and also to the amelioration of the gaseous fuel burning rate. The latter can be attributed to the increase of the cylinder charge conditions (i.e. pressure and temperature) at the end of the compression phase and during the initial stages of combustion, which favor the flame propagation mechanism. Furthermore, for the same compression ratio, the increase of diesel injection advance leads also to an improvement of the indicated engine efficiency, since the increase of diesel fuel injection advance affects positively (i.e. increases) the ignition delay period, which leads to a sharper increase of the total heat release rate curve during the premixed controlled combustion phase. This fact improves the fuel conversion efficiency, since the duration of combustion becomes slightly shorter and more of the fuel is burned before TDC. Finally, by examining Fig. 13, it is shown that the increased compression ratio in conjunction with advanced injection timing seems to contribute to the improvement of the engine efficiency.

Effects of CR and DIT on Exhaust Emissions

Effect on specific NO emissions

In Figure 14, results are given concerning the variation of the calculated indicated specific NO emission versus diesel fuel injection timing for all three values of compression ratio, examined herein. As known [23], NO formation mechanism is predominantly controlled by the cylinder charge temperature and the local oxygen excess ratio.

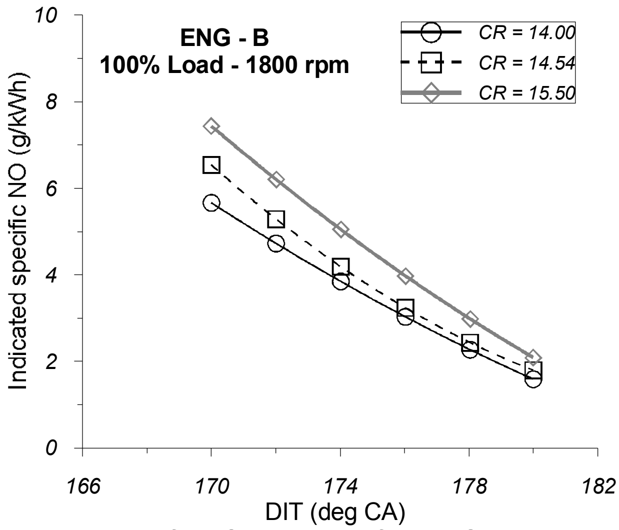


FIGURE 54. Calculated indicated specific NO emissions for "ENG-B", operating at 1800 rpm engine speed and full load conditions, for various compression ratios.

By observing Fig. 14, it is revealed that, keeping constant the diesel fuel injection timing, the increase of compression ratio leads to an increase of the emitted NO concentration. Taking into account the fact that the change of compression ratio does not affect the oxygen availability in the cylinder charge, the increase of NO concentration observed with the increase of CR may be ascribed to the higher cylinder charge temperatures caused by the increased compression ratio (Figure 15a). As far as the effect of diesel fuel injection timing on specific NO emissions is concerned, the increase of injection advance leads to a considerable increase of NO emissions. This may be primarily attributed to the fact that the advance of diesel fuel injection timing results to an earlier start of combustion relative to top dead center (TDC), a fact favoring significantly the NO formation mechanism (Figure 15b), since the charge temperatures inside the combustion chamber increase then significantly. In summary, both strategies examined herein promote the NO formation rate. Comparing the results shown in Fig. 14, it is observed that the specific NO emissions seem to be more sensitive to the advance of the diesel fuel injection timing rather to the increase of the compression ratio.

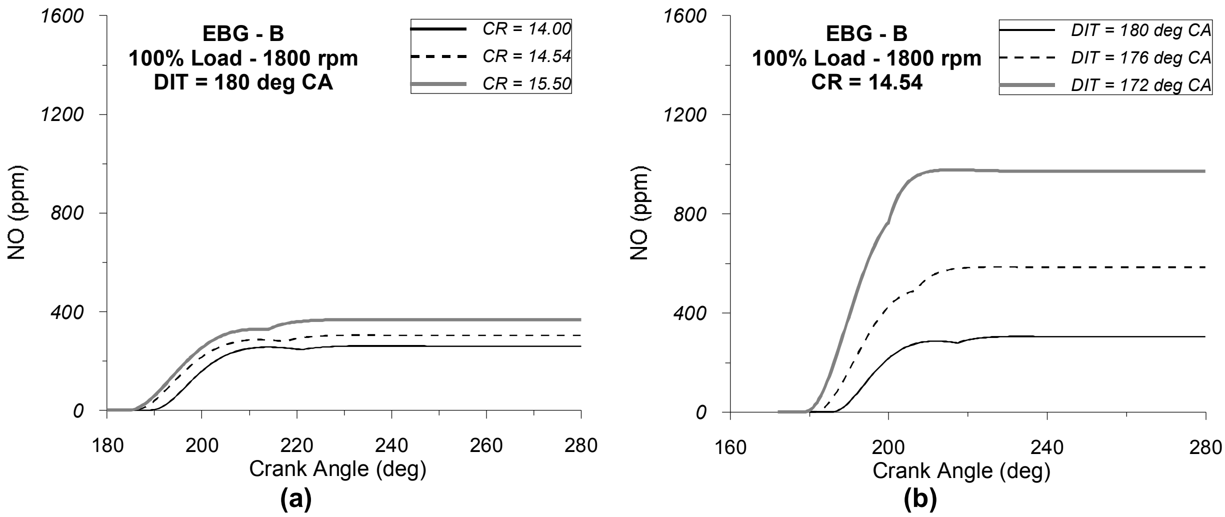


FIGURE 15. Calculated NO formation rate histories of "ENG-B" operating at 1800 rpm engine speed and full load conditions, for **(a)** various compression ratios, and **(b)** various diesel fuel injection timings.

Effect on soot concentration

In Figure 16 the variation of the calculated specific soot emissions is given as a function of diesel fuel injection timing for all three values of the compression ratio, examined herein.

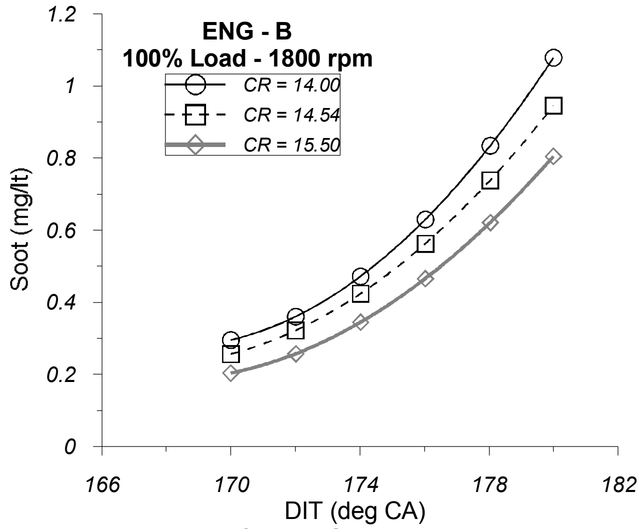


FIGURE 66. Calculated soot concentration for "ENG-B", operating at 1800 rpm engine speed and full load conditions, for various compression ratios.

By examining this Figure, it is observed that the increase of compression ratio keeping constant the diesel fuel injection timing does not seem to affect seriously the specific soot emissions, since the increase of compression ratio does not seem to affect seriously the soot formation mechanism (Figure 17a).

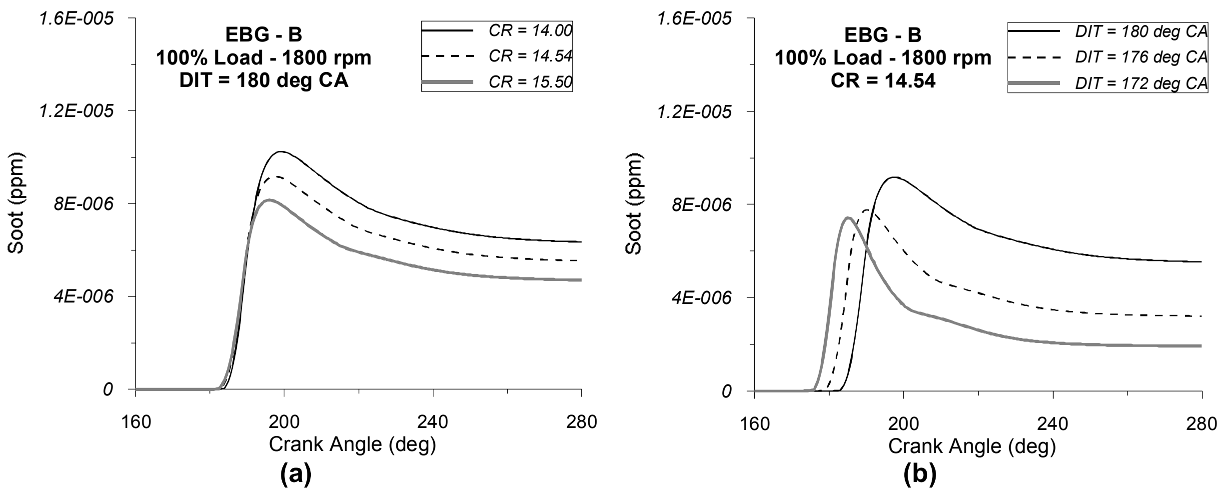


FIGURE 77. Calculated Soot formation rate histories of "ENG-B" operating at 1800 rpm engine speed and full load conditions, for **(a)** various compression ratios, and **(b)** various diesel fuel injection timings.

At the same time, it is revealed that the increase of the diesel fuel injection advance affects positively (decreases) the concentration of soot emission. This is probably due to the higher cylinder charge temperature that restrains the soot formation mechanism (Figure 17b), hence

ISSN:1791-4469

contributing to the reduction of soot emissions. Finally, observing the results depicted in Fig. 16, it is revealed that the effect of diesel fuel injection advance on specific soot emissions seems to be more intense, as compared to the respective effect caused by the increase of compression ratio.

CONCLUSIONS

In the present work, an existing two-zone phenomenological model, after validation with experimental data, was used to examine the effects of the compression ratio and the diesel fuel injection timing on the performance characteristics and pollutant emissions of a turbocharged, dual fuel (i.e. natural gas-diesel) compression ignition single cylinder research engine, operating at 1800 rpm engine speed and full load conditions, for x = 90% of gaseous fuel mass ratio.

Good agreement was observed between calculated and measured performance and emissions parameters. Specifically, the model predicts with reasonable accuracy the absolute values and, most importantly, predicts well the qualitative trends of the most significant engine performance characteristics (cylinder pressure and heat release histories, IMEP, and ISFC), and specific NO and soot emissions, with gaseous fuel mass ratio "x".

After validating the model, it was used to examine, on a theoretical basis, the effects of diesel fuel injection timing and compression ratio on the cylinder pressure and heat release traces, indicated power output, maximum cylinder pressure, total indicated specific fuel consumption, and specific NO and soot. For the specific parametric study, cylinder charge conditions corresponding to IVC, the fuels mass flow rates and the engine speed were kept constant. From the evaluation of the theoretical findings, the following conclusions can be drawn:

For the same diesel fuel injection timing, the increase of compression ratio could lead to an increase of the indicated power output, which could be accompanied with an improvement of the indicated engine efficiency. However, at the same time, it leads to an increase of maximum cylinder pressure. The latter may have detrimental repercussions on the constructional endurance of the engine. Furthermore, the increase of the compression ratio could lead to an increase of the specific NO emissions, while the effect on soot emissions seems to be almost negligible.

For the same compression ratio, the advance of diesel fuel injection timing could lead to an increase of the indicated power output, which could be accompanied with a sensible improvement of the total indicated specific fuel consumption. However, at the same time, advancing the injection timing affects negatively (i.e. increases) the maximum cylinder pressure. Moreover, the advance of diesel fuel injection timing results to an advancement of the combustion initiation, which leads to an increment of the specific NO emissions. However, the negative impact of injection timing on specific NO emissions could be curtailed by decreasing the compression ratio. At the same time, the increase of the diesel fuel injection advance leads to a reduction of the soot concentration, hence leading to lower specific soot concentrations.

In general, the increase of compression ratio accompanied with advanced diesel fuel injection timing could be a promising solution for improving engine power output and efficiency. However, the simultaneous increase of both parameters could cause problems to the engine structure, since the maximum cylinder pressure becomes considerably higher compared to that observed under normal engine operating mode, which may be prove to be harmful for the engine’s structural integrity.

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