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INVESTIGATION INTO A DUAL FUEL TWO STAGE COMBUSTION CONCEPT ENGINE

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ABSTRACT

The paper presents the investigation of the Dual Fuel Two Stage Combustion Engine using 1D simulation software Ricardo WAVE. The study is carried out against a typical baseline single cylinder direct injection (DI) naturally aspirated diesel engine of 330cc and a compression ratio of 17.6. The compression ratio of the concept engine is reduced to 12 whilst the capacity is kept the same by increasing the clearance volume. It is postulated that at the end of the compression process the first stage of combustion is achieved by directly injecting petrol fuel inside the cylinder producing a stratified charge which would be ignited with the aid of a spark plug. The quantity of petrol injected should produce an amount of heat just enough to bring the cylinder pressure at the end of its combustion to be equal to that of the DI diesel engine at the end of its compression process. The second stage of combustion will then start by directly injecting diesel fuel inside the cylinder, the amount is such that the total heat added in both stages are equivalent to that added during the baseline diesel engine.

Key words: Dual fuel, two stage combustion, concept engine, gasoline and diesel combustion, 1D engine simulation,

1. INTRODUCTION

In recent years there were a lot of developments in the field of internal combustion engines. The awareness of air pollution introduced strict emission regulations on engine there by limiting the optimization of engine performance thus demanding alternate ways to increase the efficiency. Recently more research works has been carried out on alternate fuels and methods than the usual fuels and engine type. The technologies like Homogeneous charge compression ignition engine [1-,3], variable compression ratio engine [4-7], Gasoline direct injection engine [8-11], hydrogen engine [12-15] etc found their way on the development. One of the areas which researchers are more focused on is duel fuels engines because of their high efficiencies and capability of handling multiple fuels [16,17].

Most of the duel fuel engine designs use natural gas/diesel engine technologies. One of the papers describes the benefits and drawbacks of using natural gas as main fuel and diesel fuel as pilot injection for initiating the combustion [18].

In the work it is given that the combustion is similar to that of the Otto cycle engine, but the combustion of the premixed charge is not initiated by the spark plug but because of the compression ignition and combustion of the diesel pilot fuel.

Another research work carried out in the same field mentions about a new dual fuel engine concept [19]. This engine combines the principles of Otto and Sabathe (mixed/Dual) theoretical process in one engine. In lower engine load, the Otto cycle is employed and in the higher engine load Sabathe process with changing share of heat supplied at constant volume is applied.

This paper describes a concept of duel fuel engine in which its performance is compared against a baseline diesel engine of similar capacity by simulating it on a 1D simulation software Ricardo 'WAVE'.

2. DFTSC ENGINE CONCEPT

Dual fuel engines have their own advantages and disadvantages. In spite of the difficulties this

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concept provides the possibility of burning alternative fuels including heavy fuels with increased efficiencies over the diesel engine. Theoretically the DFTSC engine uses a high volatile and a low volatile fuel. The low volatile fuel is burned due to the high temperature and pressure produced as the combined result of compression and first stage combustion, there for the compression ratio is reduced significantly to a lower value compared to the diesel equivalents.

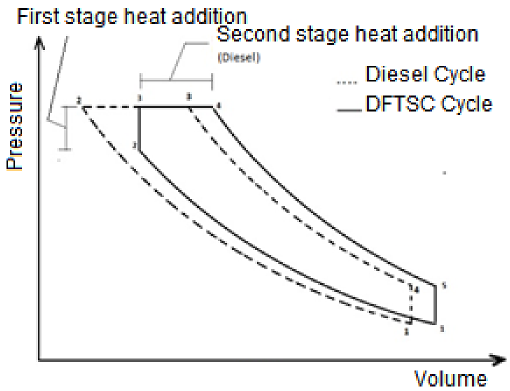

Figure 1. Ideal pressure – volume diagram representing Diesel cycle and DFTSC cycle

In this paper, Gasoline is used to represent the high volatile fuel and Diesel for low volatile fuel. DFTSC Engine uses two direct injectors for injecting these fuels separately into the combustion chamber. It is also equipped with a spark plug which is used to ignite the directly injected petrol in the combustion chamber. The petrol is injected so that it forms a stratified charge around the spark plug for the efficient burning. At the Top dead centre of the piston, the stratified charge is ignited with the spark plug. It is assumed that the petrol is burned at constant volume which produces the required pressure and temperature for the efficient burning of diesel fuel. The first stage combustion produces huge turbulences which makes the combustion of the diesel fuel much effortless. At the next instant of the ignition of the gasoline fuel, but with a definite time gap the diesel fuel is injected. The atmosphere inside the combustion chamber is well suited for the efficient burning of diesel fuel there by reducing it ignition delay. This gives more work output per cycle when compared to the diesel engine.

3. SIMULATION OF DUAL FUEL TWO STAGE COMBUSTION (DFTSC) ENGINE

The performance of the simulated DFTSC Engine is compared with a similarly analyzed Diesel engine for a meaning fuel comparison. For simplifying the simulation, a single cylinder engine is chosen and only engine performance and emission characteristics are studied. Standard temperature of 298K and pressure of 1 bar were taken for the inlet conditions. Average surface temperature for the components like piston, cylinder liner, intake valve and exhaust valve were taken as the default values from the Ricardo WAVE guidelines. The engine specifications for the baseline diesel engine were taken for a typical engine.

Bore	-	69.6 mm
Stroke	-	82 mm
Compression ratio	-	17.6
Clearance height	-	1.2 mm

The multi component Wiebe combustion sub-model was used to simulate both engines with the use of fuel cumulative burned mass fraction profile obtained from separate WAVE simulations [20]. 2-zone combustion model was used in all the analysis to capture the details of the processes taking place during the combustion period. The simulations were carried out assuming the working fluid to be an ideal gas with values for the molecular weight of CO, NO_2 and unburned hydrocarbon were taken accordingly from the Ridardo WAVE guidlines for the determination of the emission characteristics for the engine. The baseline diesel engine was modeled and simulated using a fuel/air ratio of 0.055 which is 20% less than that of stoichiometric fuel/air ratio for the combustion of diesel fuel [21] and the start of the injections were selected such that the maximum pressures were obtained near to TDC for minimum advance for best torque. Fuel mass flow rates obtained from the baseline engine simulation were used as reference for the simulation of the DFTSC engine.

Simulation of the DFTSC engine required two different combustion processes in the same cycle. WAVE does not have the capability of simulating dual-fuel systems or two different combustion sub-models simultaneously, thus a compromised approach was required, utilizing WAVE's multi-component Wiebe combustion sub-model to superimpose multiple Wiebe curves which separately represent the pre-injection and

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late injection combustion events, thus providing a complete burn profile.

The first stage combustion for the gasoline fuel was modeled using SI Wiebe combustion sub-model, an equivalence ratio of 1.1 around the spark plug was assumed to represent a stratified rich mixture and a Wiebe exponent of 1.5, to represent the fast burning in the initial stage and slowing towards the end of the combustion. The first stage combustion determines the amount of fuel required to raise the pressure of the DFTSC engine to that of the baseline diesel engine as shown in Figure 2. This also gives the cumulative fuel burn mass fraction profile which is necessary for the DFTSC engine simulation. Full details of the simulation are given in reference [20].

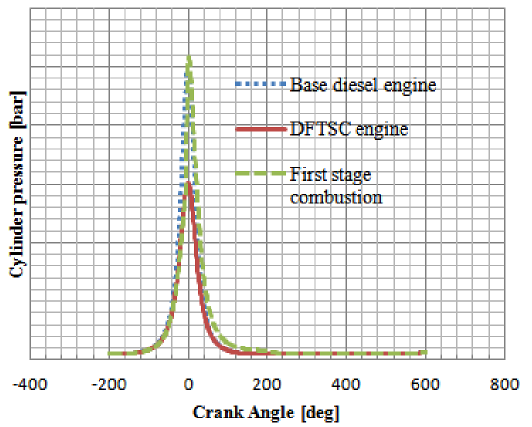

Figure 2. Cylinder pressure at the end of compression of baseline diesel engine and end of first stage combustion Vs crank angle

Figure 3 shows the pressure after the primary combustion process of the DFTSC engine is similar to that of the peak compression pressure of the baseline diesel engine.

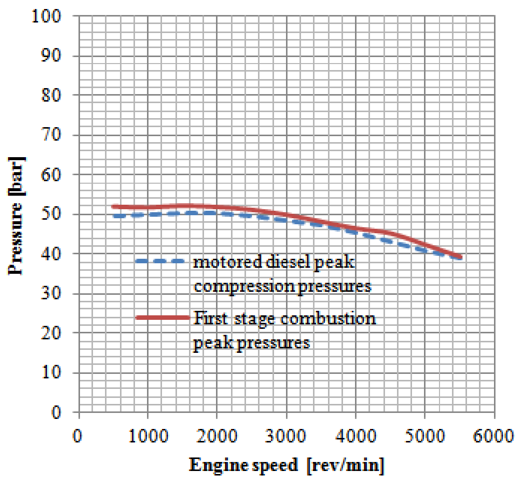

Figure 3. Peak compression pressure of the baseline diesel engine and the pressures at the end of first stage combustion of DFTSC engine Vs engine speed

The second stage of the combustion for diesel fuel was modeled using the diesel Wiebe combustion sub-model, similar to the baseline diesel engine. The amount of diesel fuel used in this stage was calculated by converting the amount of gasoline fuel used in the first stage into equivalent diesel fuel on the energy basis. This was then deducted from the amount of diesel fuel used in the baseline diesel engine. This was necessary in order to keep the total amount of energy used by both engine the same. The simulation gave the cumulative fuel burn mass fraction profile which was required for the full simulation of DFTSC engine

The simulation of DFTSC engine was carried out by superimposing the cumulative fuel mass burn fraction profile obtained from the first and second stage combustion simulation with their respective fuel flow rate using multi-Wiebe combustion sub-model. The super imposed fuel cumulative burned mass fraction and the burn rate are shown in Figure 4.

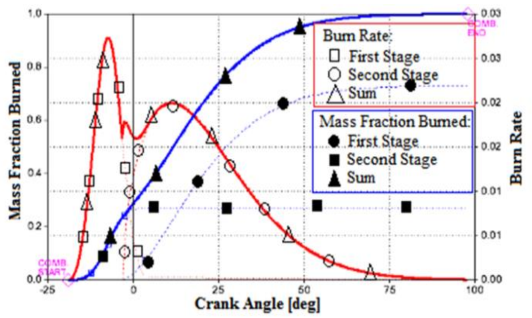

Figure 4. Mass fraction burned and the burn rate Vs crank angle for DFTSC engine

4. RESULTS AND DISCUSSIONS

4.1 Engine Performance

The brake torque and the brake power of both the baseline diesel and DFTSC engines against the engine speed range are shown in Figure 5. The figure clearly shows that both engines have similar torque and brake power values up to a speed of 1500 for the torque and 2000 rev/min for the power. The DFTSC engine shows an increase of up to 10.4% at the maximum speed of 5500 rev/min above that of the baseline diesel engine. At the rated engine speed of 4000 rev/min the increase was 6.5%.

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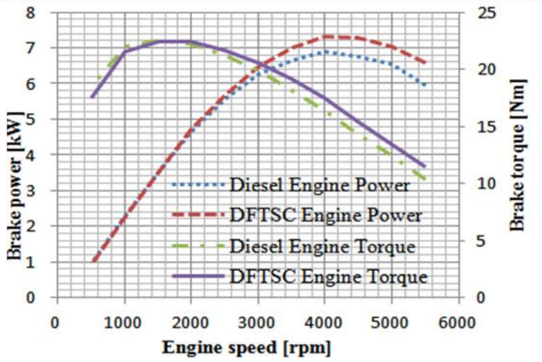

Figure 5. Comparison of brake power and torque Vs engine speed for both engines

Figure 6 shows the brake specific fuel consumption (bsfc) and the brake thermal efficiency, for both engines, at various engine speeds. Both engines gave similar bsfc and brake thermal efficiency up to a speed of 1500 rev/min and the DFTSC engine showed a reduction in bsfc of 9.4% at the maximum engine speed and 6.1% at the rated engine speed. Also the DFTSC engine showed an increase in the brake thermal efficiency of 10.4% at the maximum engine speed with an increase of 6.5% at the rated engine speed.

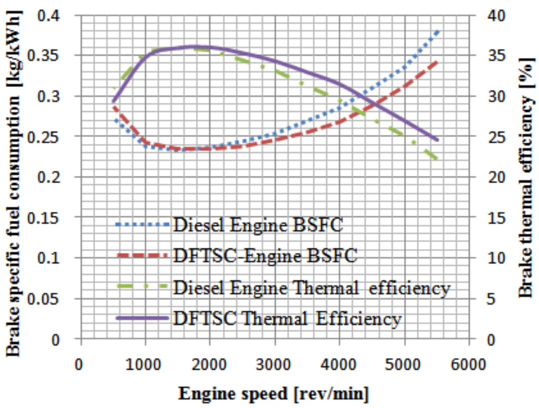

Figure 6. Brake specific fuel consumption and brake thermal efficiency Vs engine speed for both engines.

The improvement of the thermal efficiency for the DFTSC engine above that of the baseline diesel engine can be explained from figure 7, showing the pressure crank angle diagram. It is clear that the compression negative work is lower due to the lower compression ratio used which leads to increase in the net work.

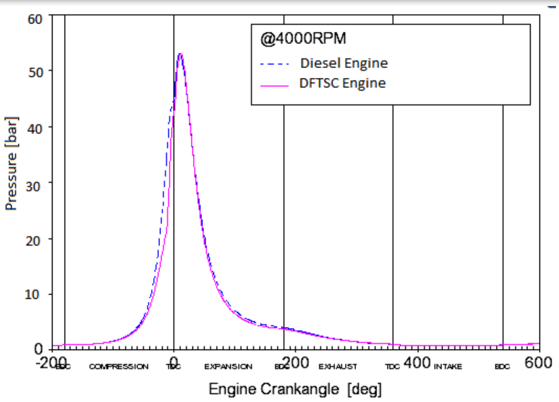

Figure 7. Cylinder pressure Vs Crank angle Diagram for both the engines at the rated engine speed.

The improvement of the thermal efficiency for the DFTSC engine above that of the baseline diesel engine can also be explained from figure 8 which the DFTSC engine produces a higher heat release rate with a maximum of 23 J/deg compared to 17 J/deg for the base diesel engine. This could be explained by the higher temperature achieved towards the end of the first stage combustion. At the top dead centre the cylinder temperature of the DFTSC engine is 1337 K when compared to 951 K for the baseline diesel engine.

Additionally figure 9 shows that the total duration of the DFTSC engine is much shorter, 42^0 compared to the 61^0 of the baseline diesel engine.

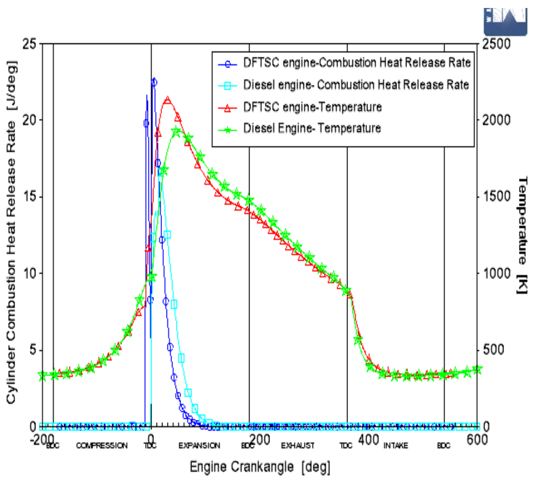

Figure 8. Cylinder combustion heat release rate Vs Engine crank angle for both engines at rated engine speed.

The increase in the efficiency of the DFTSC engine for the higher engine speed could also be explained by the examining the change of the timing of 10% and 90% fuel mass burned and combustion duration as shown in Figure 8. The completion of the 10% fuel mass burned for the

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DFTSC engine is 7^0 BTDC compared to 5^0 ATDC for the baseline diesel engine at the rated engine speed. Simulations clearly showed that the 10% fuel mass burned is completed during the first stage gasoline combustion [20]. This will lead to a faster combustion compared to the baseline diesel fuel combustion. Figure 8 also shows that the completion of 90% fuel mass burn of DFTSC engine has reduced from 56^0 ATDC compared to 35^0 ATDC for the baseline diesel engine at the rated engine speed.

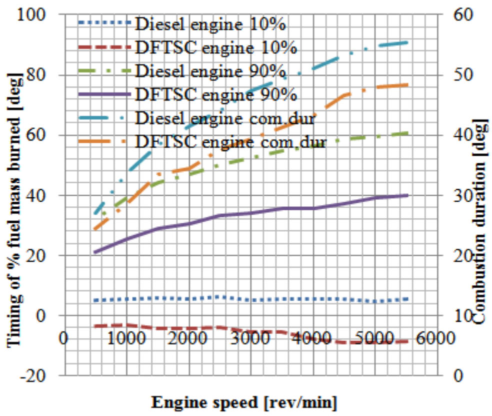

Figure 9. Timing of 10% and 90% fuel burned and combustion duration Vs engine speed for both engines.

The amount of heat released by burning gasoline fuel during the first stage combustion will lead to a high increase in temperature of the combustion chamber contents about 50% over the corresponding maximum compression temperature for the baseline engine at rated engine speed, as shown in Figure 10.

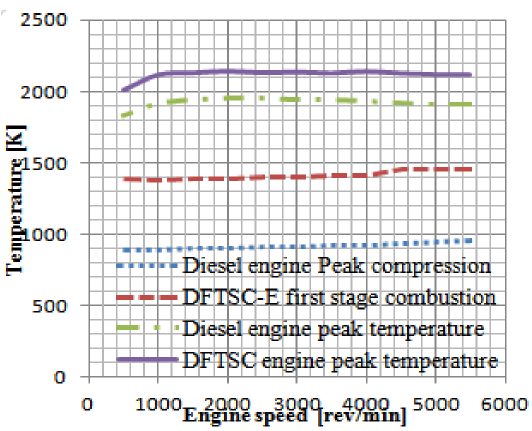

Figure 10. Peak compression and end of first stage combustion temperatures Vs engine speed for both engines

The increase in temperature will also lead to an improvement of the vaporization process for the diesel fuel which in turn will increase the rate of combustion of the diesel fuel during the second stage of combustion process. The simulation showed that the total combustion for the DFTSC engine is much lower than that of the baseline diesel engine by around 15% at rated engine speed [20]. Reduction in the total combustion duration for the DFTSC engine increases the work output per cycle and hence the brake thermal efficiency as show in Figure 6.

4.2 Engine Emissions

The WAVE simulation has used the unburned HC sub-model which is based on the observation of the uncontrollable fuel trapped within the injector sac and hole volume to be the major source of unburned HC. The model assumes that the engine out unburned HC is proportional to the injector sac volume. A recommended value of 0.2 of sac volume fuel has been used [22]. Figure 11 shows that for engine speeds lower than 1500 rev/min, the DFTSC engine produce slightly higher HC emissions but engine speeds above 1500 rev/min, emissions are reduced to a value of 7.2% at rated engine speed and 11.21 % at maximum engine speed.

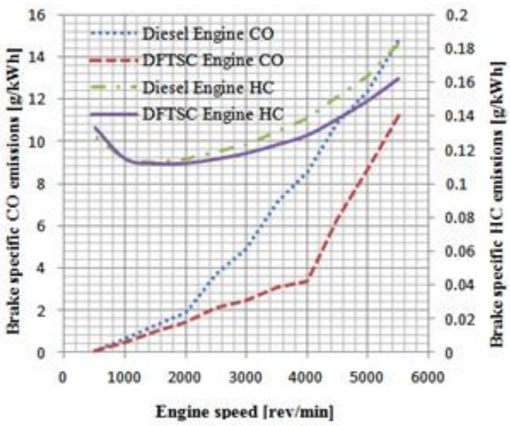

Figure 11. Brake specific CO emissions and brake specific HC emissions Vs engine speed for both engines

The CO emission sub-model in WAVE uses the chemical reaction originally used in the WAVE's gas property calculation which gives good results for the rich mixture combustion [22] and uses the chemical reaction suggested by Newhall [23] which gives good results for lean combustion. Figure 10 shows the variation of the brake specific CO and the brake specific HC emissions. The brake specific CO emissions for DFTSC engine much lower than for the baseline diesel engine at all engine speeds. At the rated engine speed the amount of CO emissions is 3.4 g/kWh for DFTSC engine whilst 8.5 g/kWh for baseline

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diesel engine with a reduction of 24.2%. At maximum engine speed it has the emission of 14.8 g/kWh for baseline diesel engine and 11.22 g/kWh for DFTSC engine. The first stage combustion process for the DFTSC engine and the increase in temperature by at the end of this process as explained previously in Figure 9 will lead to more efficient combustion and hence explains the observed reduction in the emissions of CO for the DFTSC engine.

The details of the NOx emission sub-model are given in Ref.22 in which the concentration of NO is obtained from correlation of the data reported by Finimore [24] and using the extended Zeldovich mechanisms for the NOx formation. Figure 11 shows the brake specific emissions of nitrogen dioxide (NO_2) to be higher for the DFTSC engine at all engine speeds, with a value of 19.3 g/kWh compared to 16.3 g/kWh for the baseline diesel engine at the rated engine speed. This higher increase could be explained by the high localized temperature formed after the combustion of the first stage gasoline fuel as shown in Figure 9. At the rated engine speed a higher value of 47% above that for the peak compression temperature of the baseline diesel engine is achieved. Such high values of temperature and NO_2 emissions can be explained by the absence of turbulence in the chosen version of WAVE simulation software. The lack of turbulance model in the simulaiton have the negative effect on the NO_2 prediction. Allowing addition of turbulence effects in the simulation should lead to a decrease in the temperature and hence a reduction of NO_2 emissions [25, 26]. This should be the subject of future research.

The simulation of DFTSC engine is carried out using a single injection process as Ricardo WAVE software does not support the multiple injectors in a single cylinder. This necessitates that the whole amount of fuel to be injected in a single process. So while burning the first stage of combustion, the second stage of fuel is being mixed with the air to form a readily available reacting mixture for the combustion process during the second stage. This will lead to the combustion process during the second stage releasing the heat at a higher rate with a higher prediction of NO_2 emissions [22].

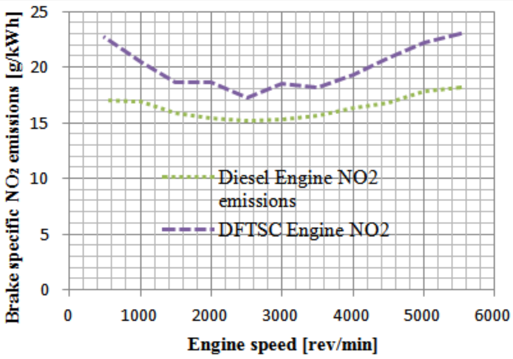

Figure 12. Brake specific NO_2 emissions Vs engine speed for Diesel engine and DFTSC Engine.

5. CONCLUSIONS

A concept engine based on dual fuel two stage combustion processes has been presented and studied using the Ricardo WAVE one-dimensional software. The concept engine was compared with a baseline diesel engine of the same engine capacity but at a much lower compression ratio. Gasoline and diesel fuels were directly injected in the concept single cylinder engine whilst the diesel fuel is directly injected into the baseline diesel engine. The total amount of heat added was kept the same during the combustion process for both engines.

Engine performance characteristics such as brake torque, brake power, brake specific fuel consumption, brake thermal efficiency, were compared for both engines. The results of the simulation showed that the performance of the Dual Fuel Two Stage Combustion is much higher than the baseline direct diesel engine. The improvements have been shown to be more significant with increasing engine speed. This has been explained by the lower negative work, higher cylinder combustion heat release rate, higher temperature towards the first stage combustion and lower total combustion duration.

Emission characteristics for both engines have been compared and the results showed that the DFTSC engine producing much less CO and HC emissions. The brake specific emission of NO_2 has shown to be higher for the DFTSC engine at all engine speeds in comparison with the baseline diesel engine. This increase has been explained by the high localized temperature formed after the combustion of the first stage gasoline fuel. These high temperatures can be explained by the absence of the turbulence model in the WAVE software version used in the simulation. Adding turbulence effects in the simulation would lead to a decrease in the temperature and hence a reduction of NO_2 emissions.

It is recommended that further study should be done incorporating 'Dynamic In-Cylinder Swirl' and 'Dynamic In-Cylinder Tumble' using MCC 2-zone or multi-zone combustion models in AVL Boost to predict with higher accuracy the combustion and hence the NOx formation.

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