Research Article

Zero-dimensional modelling of a four-cylinder turbocharged diesel engine with variable compression ratio and its effects on emissions

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Abstract
With emission legislation becoming ever more stringent, declining fossil resources and an increase in greenhouse effect caused by CO_2 emissions, manufacturers are exploring new ways to match the emissions regulations without compromising on the performance of the engine. This study included development of zero-dimensional model of a 2.0 L turbocharged diesel engine and then study the effects of changing its compression ratio in the numerical model. This paper gave a framework in determining the effect of compression ratios in different operational conditions of the engine. Implementation of variable compression ratio technology on a numerical model proved to be very cost-effective, time saving and validated the fact that numerical models can be used to study different parameters of the engines during the development stage. The main effect of an increase in compression ratio, was found to be as expected, a decrease in brake specific fuel consumption and an increase in thermal efficiency with a greater impact at low rpm-low load regions. Assuming, that the variable compression ratio technology can be utilized in the engine, this work found the best combination of compression ratios around the engine map, giving a best fit of trade-offs between the emissions and performance of the engine. This study also validates the fact that variable compression ratio technology, if implemented in the engine could not only reduce emissions of the engine but can be given as an option to the end-user to switch to more economic fuel consumption values during idling or cruising at long distant journeys.

Keywords Internal combustion engine · Variable compression ratio · Numerical modelling · Combustion · Emissions · Zero dimensional

Abbreviations

BDC	Bottom dead centre	HC	Hydrocarbons
BMEP	Brake mean effective pressure	HCCI	Homogeneous charge compression ignition
BSFC	Brake specific fuel consumption	HiL	Hardware in the loop
CA	Crank angle	HP	High pressure
CAC	Charge air cooler	IMEP	Indicated mean effective pressure
CO	Carbon monoxide	LNT	Lean NOx trap
CO_2	Carbon dioxide	MD	Torque
DAC	Digital analogue convertor	ME	Mechanical efficiency
DI	Direct injection	MF_Fuel	Mass flow of fuel
DPF	Diesel particulate filter	MF_IA	Mass fuel of intake air
EAS	Exhaust aftertreatment system	MFB50%	Mass fuel burnt 50%
ECU	Electronic control unit	MiL	Model in the loop
EGR	Exhaust gas recirculation	MoBEO	Model based engine optimization
		N	Engine speed

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SN Applied Sciences (2019) 1:1162 | https://doi.org/10.1007/s42452-019-1185-x

Received: 21 May 2019 / Accepted: 28 August 2019 / Published online: 5 September 2019

NO	Nitrogen oxide
NOx	Oxides of nitrogen
NTE	Not-to-exceed
P_EO	Pressure at engine out
PFP	Peak firing pressure
PID	Proportional–integral–derivative
QWHT	Wall heat transfer
RDE	Real driving emissions
RPM	Revolutions per minute
ROHR	Rate of heat release
SCR	Selective catalytic reduction
SFC	Specific fuel consumption
SiL	Software in the loop
SOC	Start of combustion
SVC	SAAB variable compression
T_EO	Temperature at engine out
TDC	Top dead centre
THC	Total hydrocarbons
VCR	Variable compression ratio
VGT	Variable geometry turbo
VNT	Variable nozzle turbine
VVT	Variable valve timing

1 Introduction

With emission legislation becoming ever more stringent, declining fossil resources and an increase in greenhouse effect, manufacturers are exploring new ways to match the emissions regulations without compromising on the performance of the engine. A "Not-To-Exceed" zone is also being introduced as a part of these legislations, where tail pipe emissions must not exceed a specific value. Legislations like these break the stronghold of keeping these engines economical under the same performance circle. Although new technologies like the control methods, exhaust gas recirculation (EGR), and variable valve timing (VVT), have helped to reduce the NOx and soot emissions considerably [1–8], further measures are to be taken to meet the increasingly restrictive emission standards.

It is of no contradiction that diesel engines are in the top list to face emission limitations in near future and the trade-offs between NOx–CO_2–Soot emissions must be handled without compromising on the performance outputs [9, 10].

Several techniques and methodologies have been introduced in making combustion leaner and lower the emission levels. One way to reduce these pollutant emissions is using the technology called homogenous charge compression ignition (HCCI) combustion. It consists of preparing a highly homogenous air/fuel mixture that will result in a homogenous and efficient in-cylinder combustion. Thus, a complete and controlled combustion would mean better performance, lower emissions and above all no soot is produced [11].

Among many other factors, compression ratio is a direct influencer on the performance of the engine. In engines, a fixed compression ratio over the whole load and rpm range caps a limit on its operational efficiency and emissions over different regions. In conventional engines, a fixed compression ratio is selected which successfully operates it at variable rpms and loads meanwhile achieving a reliable self-ignition when starting the engine at cold-start.

The real problem is the fact that a fixed compression ratio of internal combustion engine is not optimal for all operating conditions. It is normally set as a compromise between partial and full-load requirements. A very high compression ratio could improve high part-load efficiency but will result in a knock (gasoline engines) and peak pressure limitations in diesel engines. An engine a fixed with low compression ratio provides performance with low maximal cylinder pressure which in turn means low frictional losses, costs and weight of the engine but limits the maximum achievable torque. An engine like this requires a glow plug which gives it a cold start ability resulting in a more stable idle operation and less smoke [11–14]. Therefore, the new technology of variable compression ratio (VCR) offers more flexibility in terms of controlling power and emissions at different operating load conditions.

Turbocharged diesel engines are usually also restricted by the stress levels in moving mechanical components. With an increase in boost pressure, maximum pressure, thermal loading and friction losses also increase proportionally, unless engine design and operating conditions are changed. Practically, compression ratio is often reduced in turbocharged engines to maintain peak pressures and thermal loading at acceptable levels [15].

VCR technology could not provide a cold-start ability but also could change the engine's combustion capability in such a way to get the optimal trade-offs of emissions, performance and fuel consumption at desired rpm and loads [16]. Different VCR methodologies implemented in the industry are also critically being examined to understand its applications and scope and a thorough study in this context can be found in [17].

A study comparing the experimental results with numerical modelling results was performed before in [18]. But as per our knowledge, no previous study has been done comprising of a detailed model especially implementing VCR directly in the numerical model and comparing the performance parameters in terms of engine maps.

According to theoretical studies, the overall efficiency of internal combustion engines can be improved: up to 40% for otto and up to 50% for diesel engines. Efficiency of over 50% is achieved only by two-stroke marine diesel engines [19, 20].

Early stage calibration is a way to assure meeting of the emission requirements at various ambient conditions while still delivering the desired engine performance and fuel consumption. In this process the major concern always remains to find the optimal compromise between emission, performance and fuel consumption targets under all ambient conditions and to achieve this with very limited access to non-standard engine tests beds.

Modern numerical models and tools help to pre-calibrate these engine correction functions at a virtual test-bed embedded with numerical models of the engine. This delivers an identical environment as on real testbeds, while saving time and money. With emission standards becoming narrower every day, these simulation tools can prove to be very effective in development process to meet the standard called real driving emission (RDE). The virtual test-bed is setup with high precision empirical and physical models/components to simulate the real working of the engine under different scenarios as accurately as possible. This widely helps in combustion refinement and pre-optimizing the thermodynamic cycle of the engine.

'Concept-to-Road' calibration is therefore considered an important part of the pre-prototyping phase. A baseline requirement of numerical validation is based on data of the base model of the engine from real testbed. Therefore, numerical modeling tools play a crucial part in the optimization and validation of the powertrain development process providing a very swift and cost-effective way at the early development stages, before even the prototype becomes available.

2 Objectives

The aim of this work is to develop a zero-dimensional numerical model of a 2.0 L four-cylinder turbocharged diesel engine in AVL Cruise-M (MoBEO-model based engine optimization) [21] which will then be simulated with VCR technology to understand the effect of this technology on the performance and emissions of the engine.

Modelling of the thermodynamic cycles of the engine is very easy and understandable. However, these cycles become more complex when combustion refinement is being performed by tuning various parameters of the engine. Thus, there are limits to which these parameters can be altered in the thermodynamic processes. The influence of compression ratios on the changes in thermodynamic combustion process in an engine has been studied in [10] and is considered while modelling the engine.

The first part of this study will focus on the methodologies for correlating dynamic simulations with any performance parameter in the dynamic engine operation with insights on the emissions data. This correlation can be performed using simulated test runs within AVL's mean value engine model, MoBEO. Using this approach, the main objective will be to explain how the correlated engine values and the small differences between measure and simulated results can be sourced by specific dynamic phenomena and how they impact the obtained results. It will also help to understand that having a correlation between the model and measurement data, can help in achieving front-load calibration with a very high level of confidence. This study will also summarize the factors which are influenced by changing compression ratio in diesel engine, which may be used in developing a strategy towards the concept of 'One Engine for all fuels or One Fuel for all engines' [22].

This work can help in developing an industrial approach towards zero-dimensional numerical model of a real engine and could add to the confidence on relying at such models. One of the main aims of the work is also to show the load conditions of the engine terms of various regions on engine maps. This would help in framing the zones in the engine map where VCR technology has the maximum and minimum effect on the emissions and performance of the engine. An optimal compression ratio specific to each region can also be suggested through this process.

Effects of several operating parameters will also be studied using the obtained results which can act as a solid rung towards showing the strategies to be utilized for further studies.

2.1 Zero-dimensional modelling approach

Cruise-M was used to create the numerical model of the engine. MoBEO is a tool developed by AVL GmbH which integrates empirical and physical models. This means that the airpath of the modelled engine was fully physical whereas, the cylinder model was based on semi-physical modeling approach as can also be seen in Fig. 5. Instead of complex cylinder configurations, MoBEO enables 'concept model capability'—tune the cylinder using fit-parameters based on ten thousand of engine test bed measurements. Thus, it demands 25 basic cylinder and fueling parameters measured from the testbed in order to perform the combustion refinement based on previously embedded data. Emission models are a part of this semi-physical approach. More details regarding the empirical, semi-physical and physical based models used by MoBEO can be accessed from [21].

A numerical diesel engine generally deals with mean engine values of performance and emission parameters (pressures, flow rates, temperatures, boundary conditions etc.). Moreover, the aftertreatment system can also be

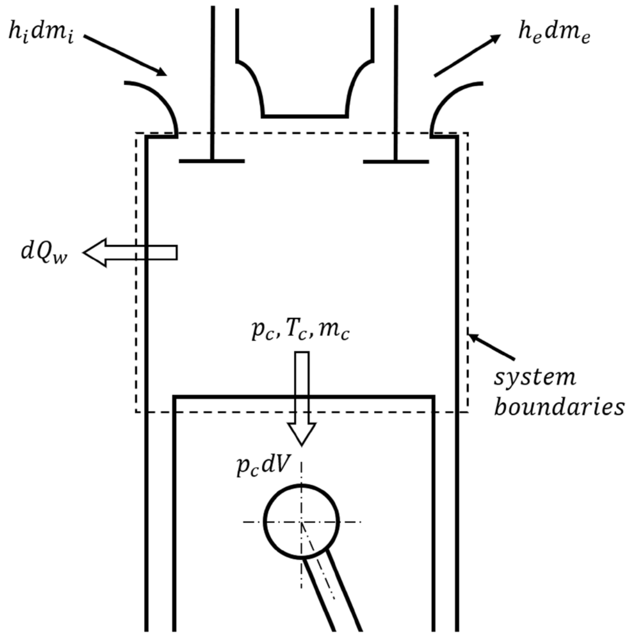

Fig. 1 Energy balance of the combustion chamber

Table 1 Engine specifications

Type	Four-stroke diesel
Displacement	2.0 L (1996 cc)
Bore length	84 mm
Stroke length	90 mm
Number of turbochargers	Single stage turbocharged
Type of turbocharger	Variable geometry turbocharger
Number of cylinders	In-line four cylinders
Fuel injection type	Direct in-cylinder
Injections	One pilot and one main
Compression ratio	16.5
rated power	125 kW @ 3500 rpm
Max torque/speed range	405 Nm @ 1750–2500 rpm
Idle engine speed	840 rpm
Maximum engine speed	5100 rpm

integrated and dealt in detail within the model. Since the complexity of the engine increases when testing change of compression ratios on physical testbeds, a numerical approach to study the trending technology of VCR can be useful in validating the circumstances which give the optimal trade-off of performance and emissions. One of the most vital advantage of engine modelling is the cut-off in both costs and time. Virtual test beds with engine models can save a lot of expenses that could otherwise incur during modification and testing process on real testbeds.

Numerical models cannot always replace the test-bed engine testing but could produce very close estimates of changes in the operational conditions, emissions and performance. This vastly helps in predicting the relevant parameters and selecting the best case during the engine development stage.

Basis of powertrain engineering is to enable a rapid setup of numerical model and to use it in a wide variety of applications. The engine domain is the heart of the entire multi-disciplinary model-based development solution. This becomes possible because of dedicated numerical solvers that work at explicit constant time step and implicit adaptive time step integration using transient and steady-state zero-dimensional component models. A steady-state approach is normally applied in modelling the basic flow characteristics. Numerical models are not only capable of being faster than real-time but also support a very high amount of cycle simulations with different hardware variants. They are also capable of transient system operation

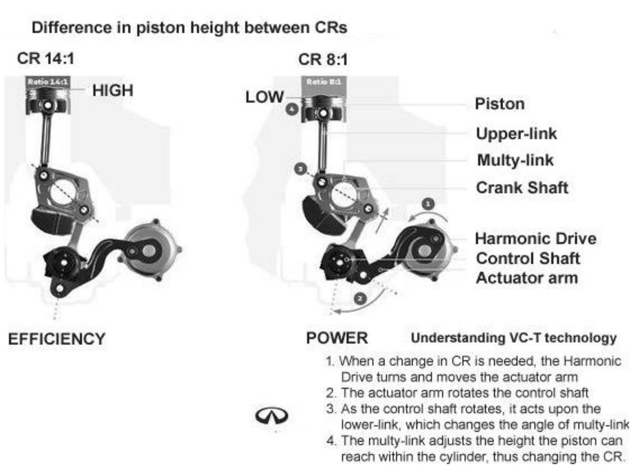

Fig. 2 Infiniti VC-T technology [20]

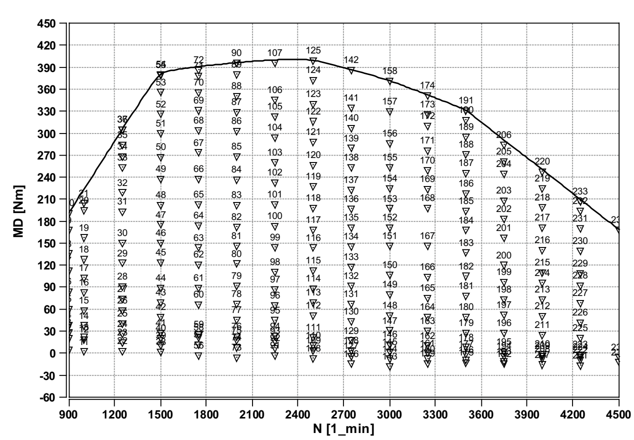

Fig. 3 Log points of measurement data collected from testbed

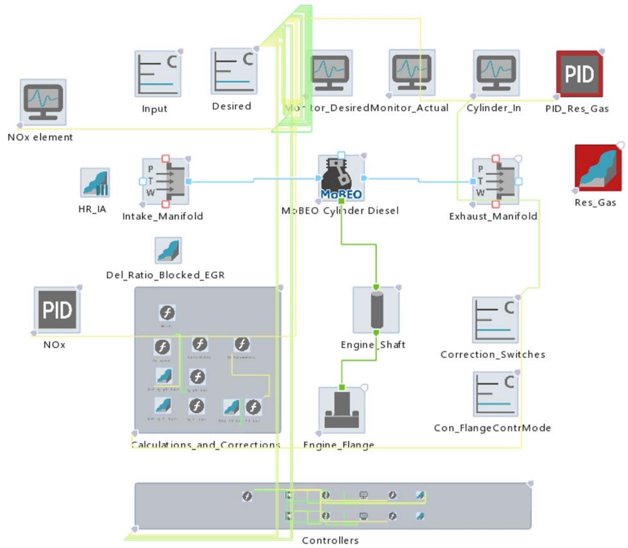

Fig. 4 Standalone model in AVL Cruise-M

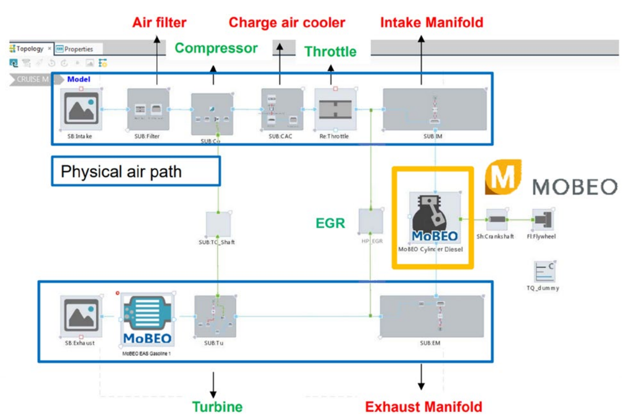

Fig. 5 Full engine model in AVL Cruise-M

which makes it easier to study stochastic changes of the engine and exhaust gas aftertreatment system.

2.2 Single zone combustion model

There are several textbooks that explain the main principles of operations of internal combustion engine. One book which discusses most aspects of internal combustion engines is [23]. The conservation equations of all the models basically comprise of mass, energy and species balances. The basic idea of the energy split in the combustion chamber is sketched in Fig. 1.

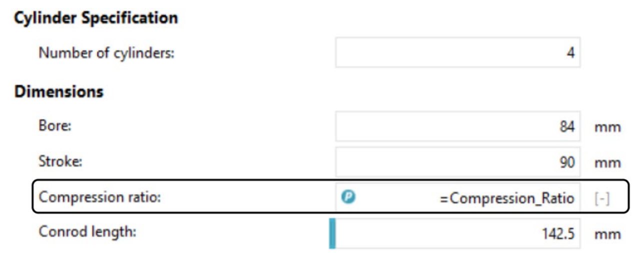

Fig. 6 VCR simulation setup in MoBEO

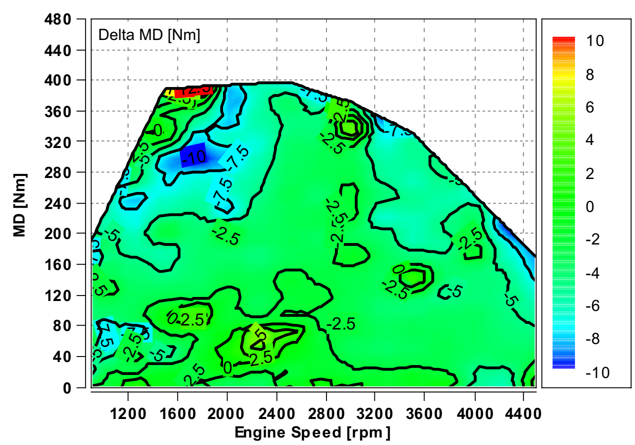

Fig. 7 Difference in brake torque

The calculation of the thermodynamic state of the cylinder is based on the first law of thermodynamics:

$$\frac{d(m_c \cdot u)}{d\alpha} = -p_c \cdot \frac{dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_w}{d\alpha} + \sum \frac{dm_i}{d\alpha} \cdot h_i - \sum \frac{dm_e}{d\alpha} \cdot h_e - q_{ev} \cdot f \cdot \frac{dm_{ev}}{d\alpha} \tag{1}$$

where m_c is mass in the cylinder, u is specific internal energy, p_c is cylinder pressure, V is cylinder volume, α is crank angle, Q_F is fuel energy, Q_w is wall heat loss, dm_i is mass element flowing into the cylinder, h_i is specific enthalpy of the in-flowing mass, dm_e is mass element flowing out of the cylinder, h_e is enthalpy of the mass leaving the cylinder, q_{ev} is evaporation heat of the fuel, f is fraction of evaporation heat from the cylinder charge and m_{ev} is evaporating fuel.

2.3 Variable compression ratio

There are three direct methodologies through which an efficient combustion can be achieved in the engine: downsizing, supercharging and varying the compression ratio [24].

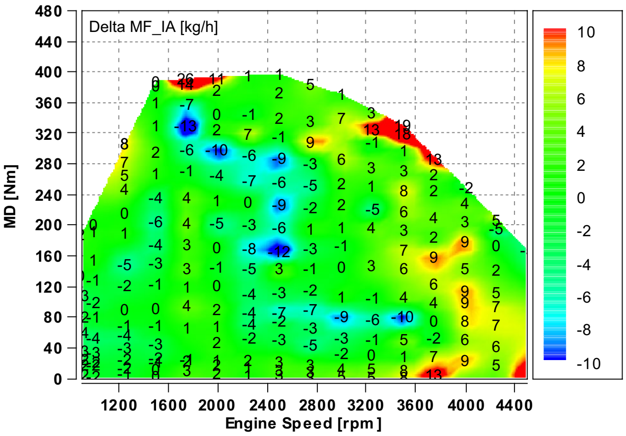

Fig. 8 Difference in mass flow of intake air

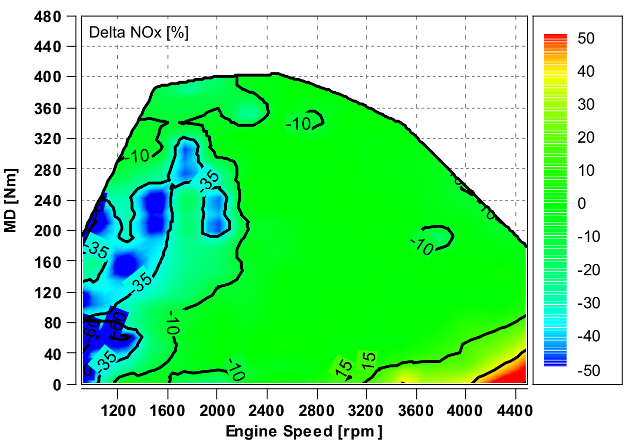

Fig. 9 Difference in engine out NOx

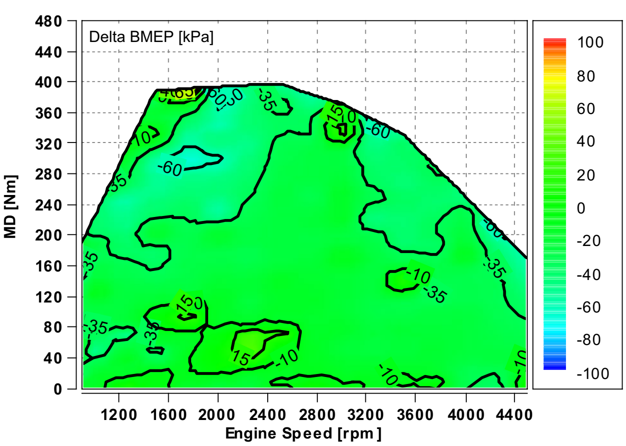

Fig. 10 Difference in brake mean effective pressure

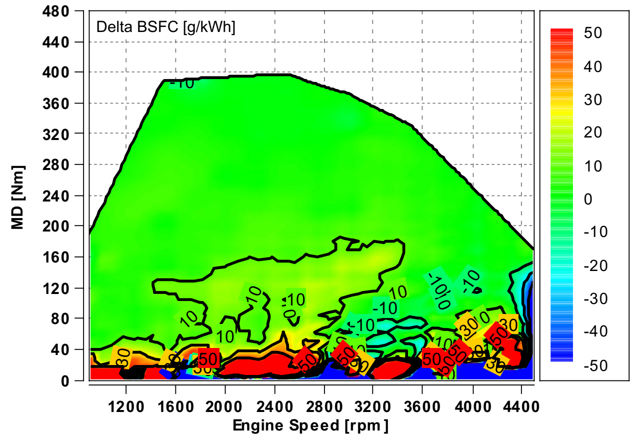

Fig. 11 Difference in brake specific fuel consumption

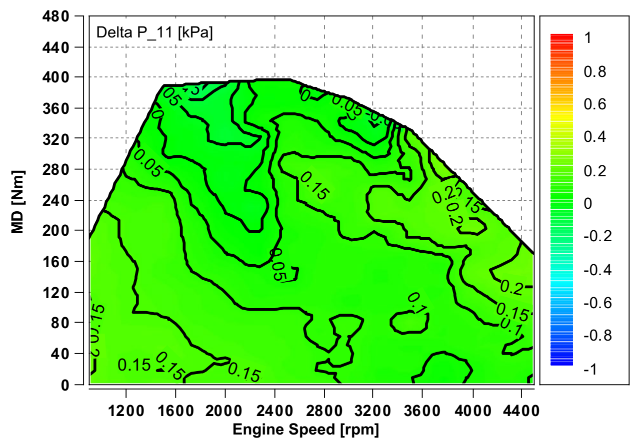

Fig. 12 Difference in intake pressure

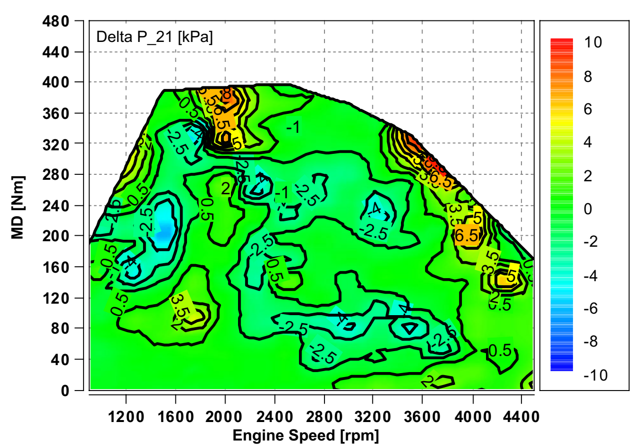

Fig. 13 Difference in boost pressure

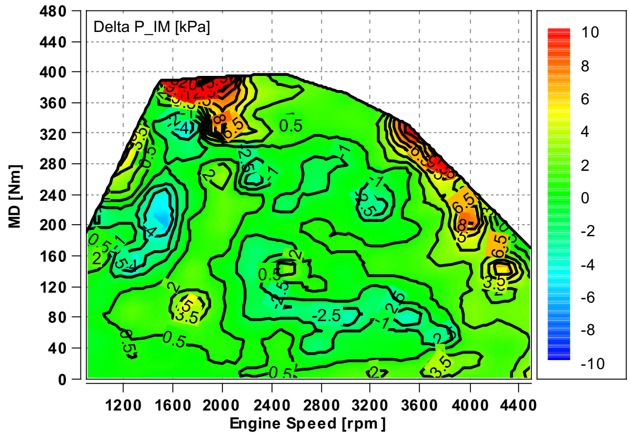

Fig. 14 Difference in intake manifold pressure

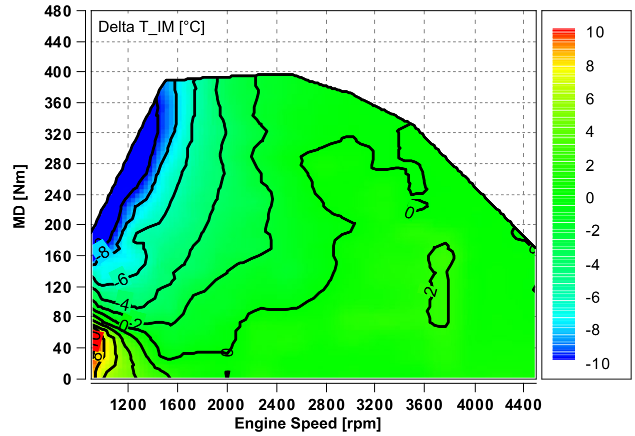

Fig. 17 Difference in temperature at intake manifold

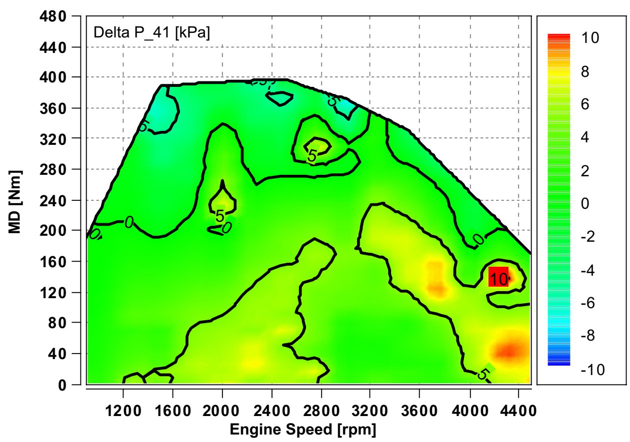

Fig. 15 Difference in engine out pressure

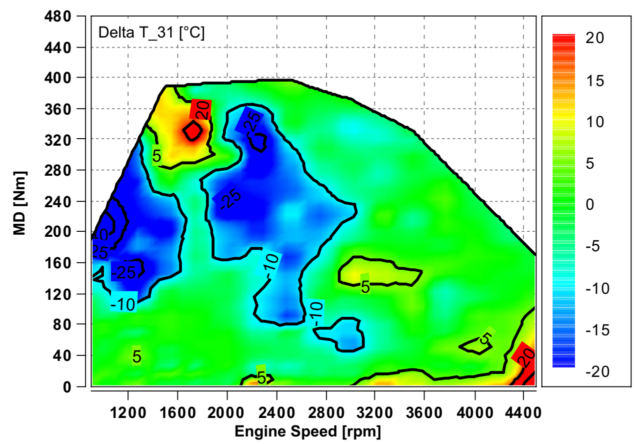

Fig. 18 Difference in temperature at engine exhaust manifold

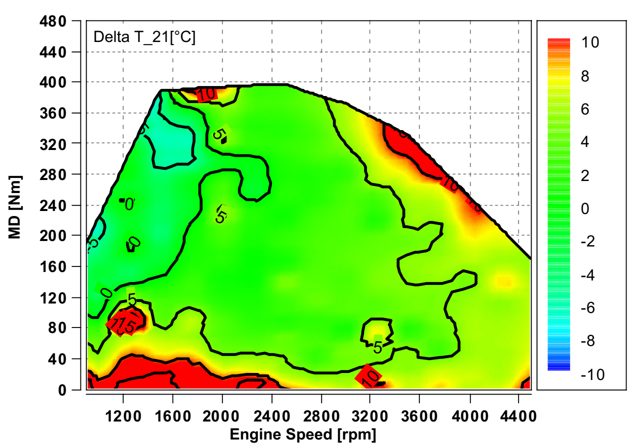

Fig. 16 Difference in boost temperature

Varying the compression ratio in a running engine is a new technology that has been developed called variable compression ratio, which radically improves the fuel consumption without impairing engine performance at certain regions of the engine map. VCR engines have been categorized into several classes in [25].

Compression ratio, r_c, is generally defined as:

$$r_c = \frac{V_d + V_c}{V_c} \tag{2}$$

where V_d is the displaced volume and V_c is the clearance volume. In other words, the amount by which the fuel/air mixture is compressed in the cylinder before it is ignited. Among the other factors, compression ratio of an engine is one of the most important factors that determine how efficiency an engine could utilize the energy in the fuel.

VCR is a technology which directly changes the compression ratio of the engine by changing the stroke under the same operating conditions. This phenomenon diversifies the engine to adjust its compression ratio directly according the load and performance need. Diesel engines have high thermal efficiencies as compared to gasoline engines mainly because of their higher compression ratios. As compression ratio decreases, both the gas pressure and temperature decrease at the end of compression stroke,

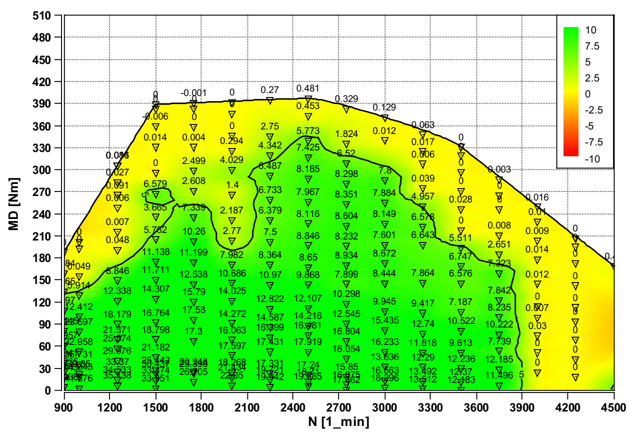

Fig. 19 Percentage massflow of EGR as compared with total massflow

thus it has a very critical influence on the combustion process and ultimately the performance of the engine.

For an ideal Diesel cycle the theoretical efficiency is:

$$\eta_{th} = 1 - \frac{r_v^{1-\gamma}\left(r_c^{\gamma} - 1\right)}{\gamma\left(r_c - 1\right)} \tag{3}$$

where γ is ratio of specific heats (1.4 at ambient temperature), r_v is compression ratio of the engine and r_c is the cut-off ratio.

Diesel engines normally have compression ratios between 14:1 to 25:1. Higher compression ratios mean a higher thermal efficiency, which means that theoretically maximum efficiency can be achieved at an infinite compression ratio. However, a high compression ratio results in higher cylinder pressures and temperatures which limit it to a certain value.

A higher compression ratio also results in an increase in frictional losses and lowers down the mechanical efficiency [10]. Furthermore, it also increases heat transfer to the combustion chamber walls due to small clearance volume resulting in a very high temperature gradient. High compression ratio also comes with shortened ignition

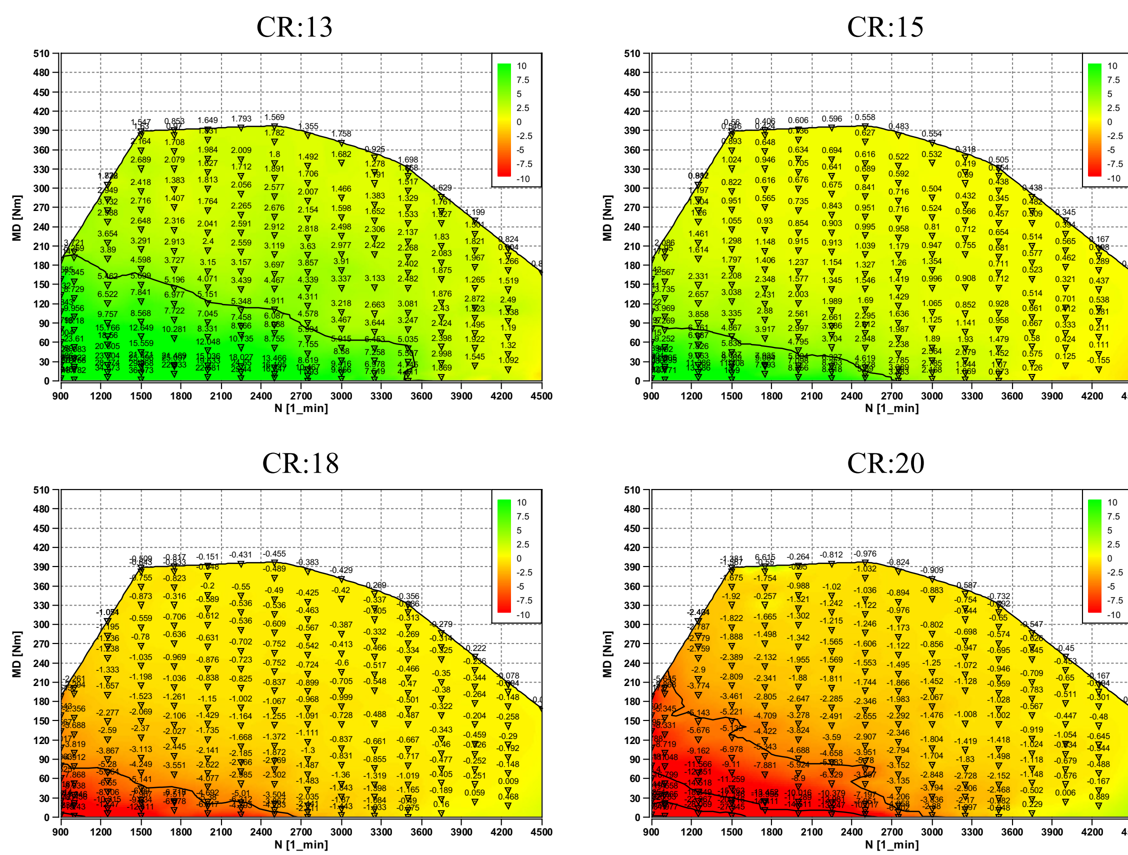

Fig. 20 Mass flow of fuel as compared with CR16.5

delays as both pressure and temperature in the cylinder increase during fuel injection. Moreover, not only the weight and cost of the engine increases but it also introduces more complexity, frictional losses and unburnt hydrocarbons. Theoretically, the brake thermal efficiency increases with an increase in compression ratio, reaches an optimal maximum value and does not show any further increase after a certain compression ratio value.

In conventional engines, the minimum compression ratio is calculated such that it supports the start of engine with little amount of fuel (very low load operation) or cold start (when temperatures are below 0 °C). Starting from that minimum compression ratio, a single compression ratio value is then determined which gives a perfect trade-off between the emissions and performance at different operational conditions. One of the important facts is that diesel engines do not run at the same loads. For instance, at a long-distance motorway journey, the engine will operate at low load/idling condition while during acceleration, it will work under full load conditions.

With a VCR engine, compression ratio can be increased at start-up and low power operations to get a stable ignition and operation, while the compression ratio can be lowered during high load operations to get more power by burning more fuel. Therefore, the VCR concept can be considered as a very reliable solution to the problems being faced by engineers in varying the combustion conditions in the engine during different load conditions. An engine with VCR makes it possible to increase its overall efficiency, by choosing high compression ratios at low loads to maximize the efficiency, and low compression ratios at high loads to avoid knock (in case of gasoline engines).

There are various patents which all correspond to realizing the VCR technology in the engines [9]. Moreover, a VCR engine can use fuels with any cetane number under normal operation [22].

Majority of mechanical methodologies vary the volume of the combustion chamber to vary the compression ratio. For instance, Infiniti has a harmonic drive embedded with the crank shaft that varies the compression ratio between 8:1 and 14:1 as shown in Fig. 2 [20].

Volvo/Alvar engine varies the volume of the combustion chamber through a secondary piston in the cylinder head [26]. The constrain to this design was that since it had four valves per cylinder, supercharging was also necessary to achieve the effect completely.

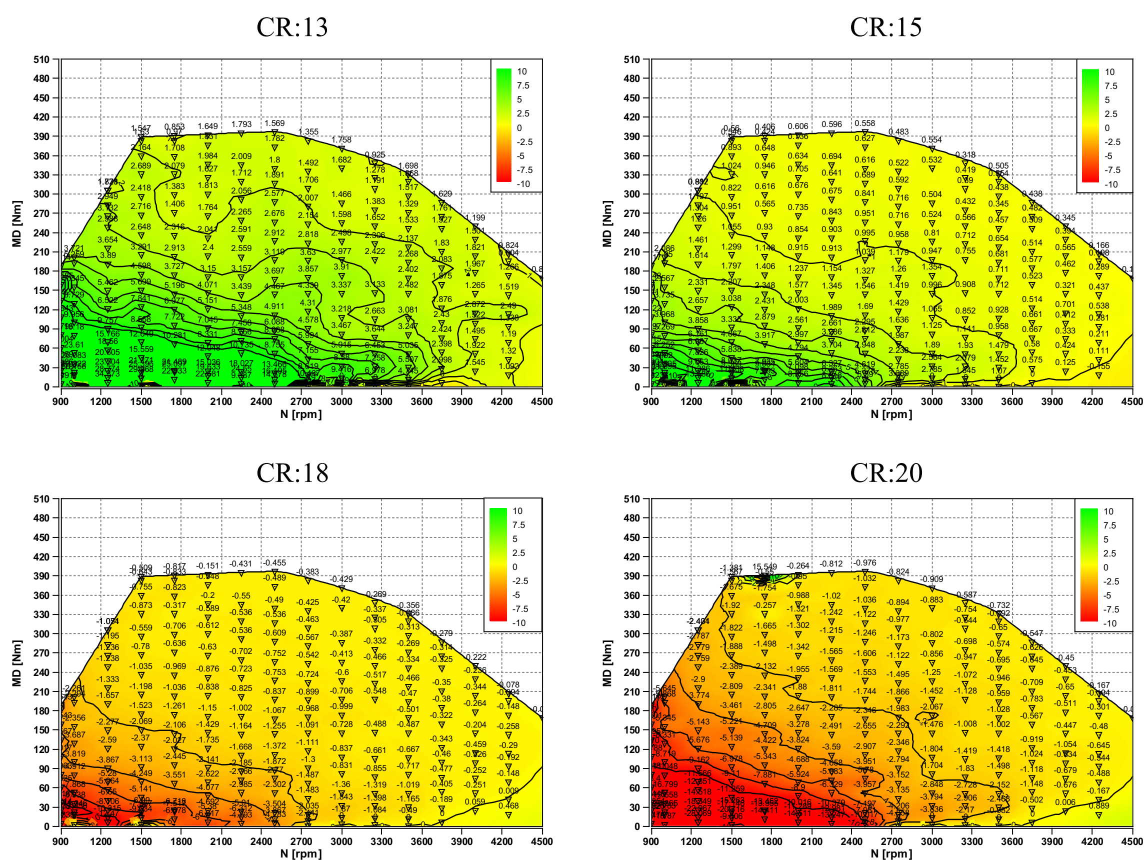

Fig. 21 Brake specific fuel consumption as compared with CR16.5

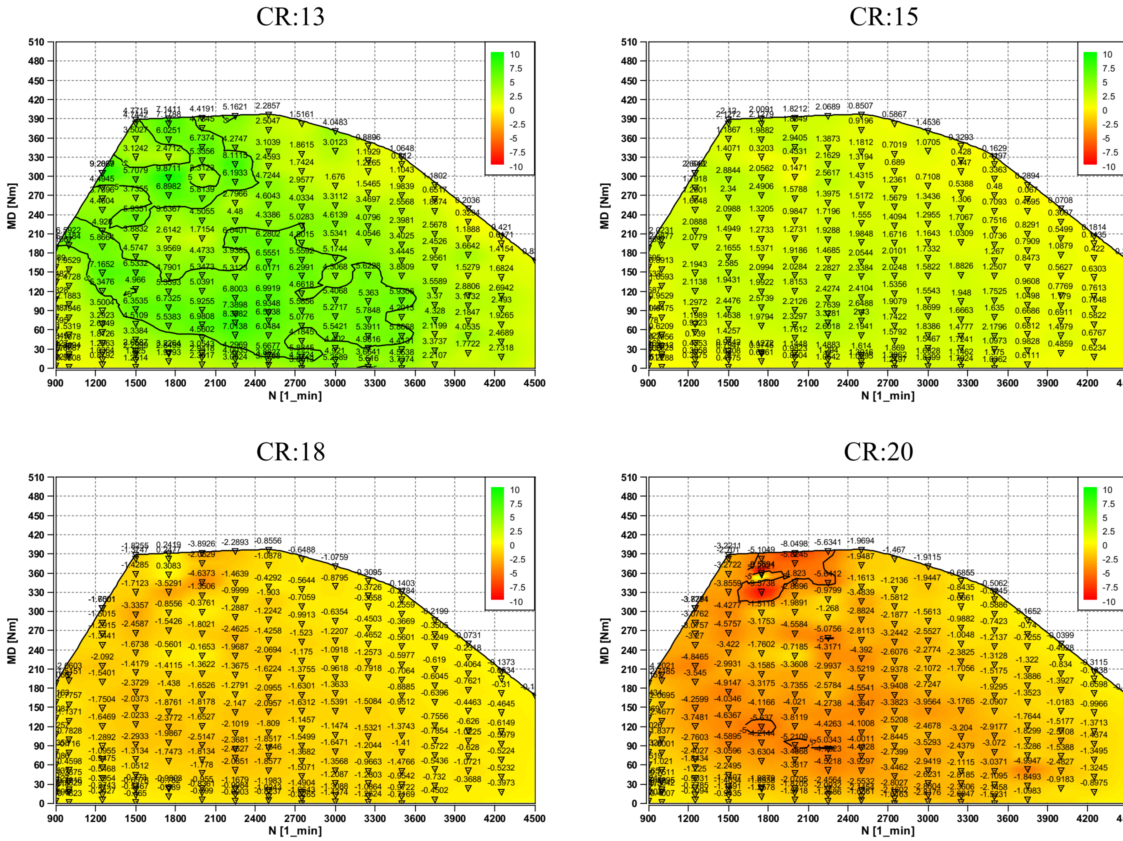

Fig. 22 Mass flow of intake air as compared with CR16.5

SAAB practiced variable compression ratio technology through its SVC Engine. The technology was implemented by repositioning of the cylinder or cylinder head [24].

FEV is also realized this technology by working on different methods. One of them was to use a connecting rod with variable length [27]. While the other one used eccentric bearings that change the TDC and BDC positions of the piston [28].

3 Specifications of engine

The study includes numerical modelling of a four-cylinder turbocharged 2.0 L direct injection diesel engine with the specifications shown in Table 1.

4 Model setup

Numerical Modelling was performed using AVL Cruise-MoBEO tool with measurement data of actual engine obtained by dynamometer tests.

A standard AVL engine development testbed with 220 kW dyno and inhouse sensors was used in the collection of different parameters. The sensors were mounted on the test bed upstream and downstream of all the engine components in order to record all the maximum possible pressure, temperature, emissions readings of a running engine on the required test points. The data was collected using AVL PUMA software interface.

Steady-state data collection requires a waiting time for the engine to reach that desired steady state and the readings were then recorded for a certain amount of time. Several readings of the same point were taken, and log-point average of the obtained readings was used.

An intensive *data check* of the raw data from sensors was performed using post-processing software, AVL Concerto, in order to make sure that the collected data correlated with each other as well as general trends expected to be observed. Out of 300 logged points, only 236 points were filtered to be usable for steady-state characterization of the numerical model.

Figure 3 shows how each log-point of data was collected at different rpms and torques. Data was collected

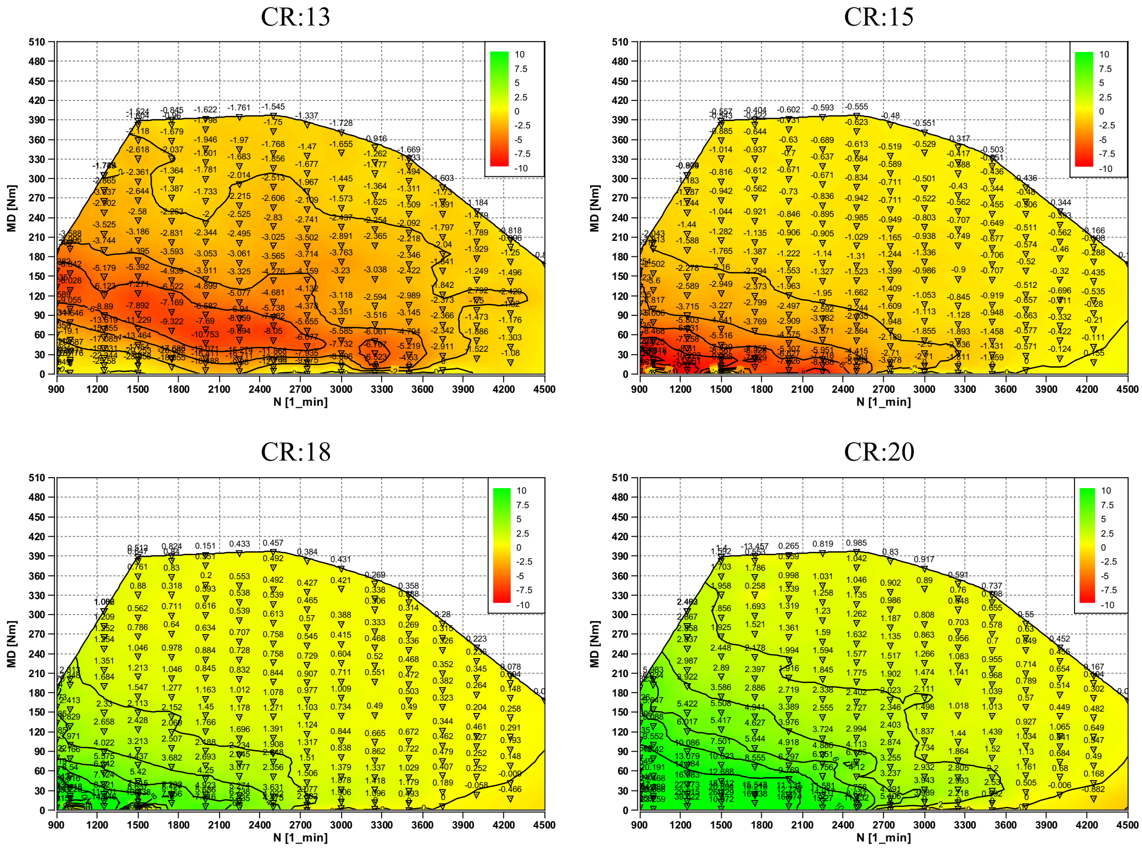

Fig. 23 Thermal efficiency as compared with CR16.5

from the engine starting from 900 rpm all the way till 4500 rpm making steps uptil maximum achievable torque at each rpm step. Figure 3 shows the numbering of log-points in the sequence tests were conducted. A total of 236 log-points covering all the scenarios of rpms and torques were recorded starting from low torque for each rpm, going up to the maximum torque achievable for that rpm. Figure 3 indicates that log-points number 90, 107 and 125 have recorded the maximum torque of approximately 405 Nm @ 2000, 2250 and 2500 rpms respectively.

4.1 Standalone model simulation

The basic purpose of standalone model is to study the working of semi-physical characterized in-cylinder parametrization before moving on the full physical airpath. The in-cylinder combustion refinement is performed at this stage to check if the semi-physical model works in correlation with the physical engine. Standalone model as shown in Fig. 4, includes MoBEO cylinder, intake and exhaust manifolds and system boundaries.

Standalone model approach is used to check if MoBEO model configuration fits the combustion parameters of the engine.

4.2 Full model characterization and simulation

In Fig. 5, a full model of a 2.0 L, 4-cylinder turbocharged diesel engine can be seen. The engine has one single stage turbo charger with variable geometry turbine (VGT), single-step high pressure exhaust gas recirculation (EGR), diesel particle filter (DPF), selective catalytic reduction (SCR), lean NOx trap (LNT) and muffler.

After the creation of general engine model, sizing of components including their volumes, dimensions, heat transfer coefficients and other parameters are set to concede with the physical components. The model consists of plenums, restrictions, solid walls, turbo components, heat transfers and other components like functions and constant table blocks. PID controllers are used to characterize the required components.

After the whole model is characterized, a full open-loop (without PID control) simulation is given to see the proximity of the model with the actual measurement data. Some tunings/refinements are also performed after this stage to eliminate minor characterization errors. The obtained 236 steady-state log points at different load conditions were validated at steady-state and transient

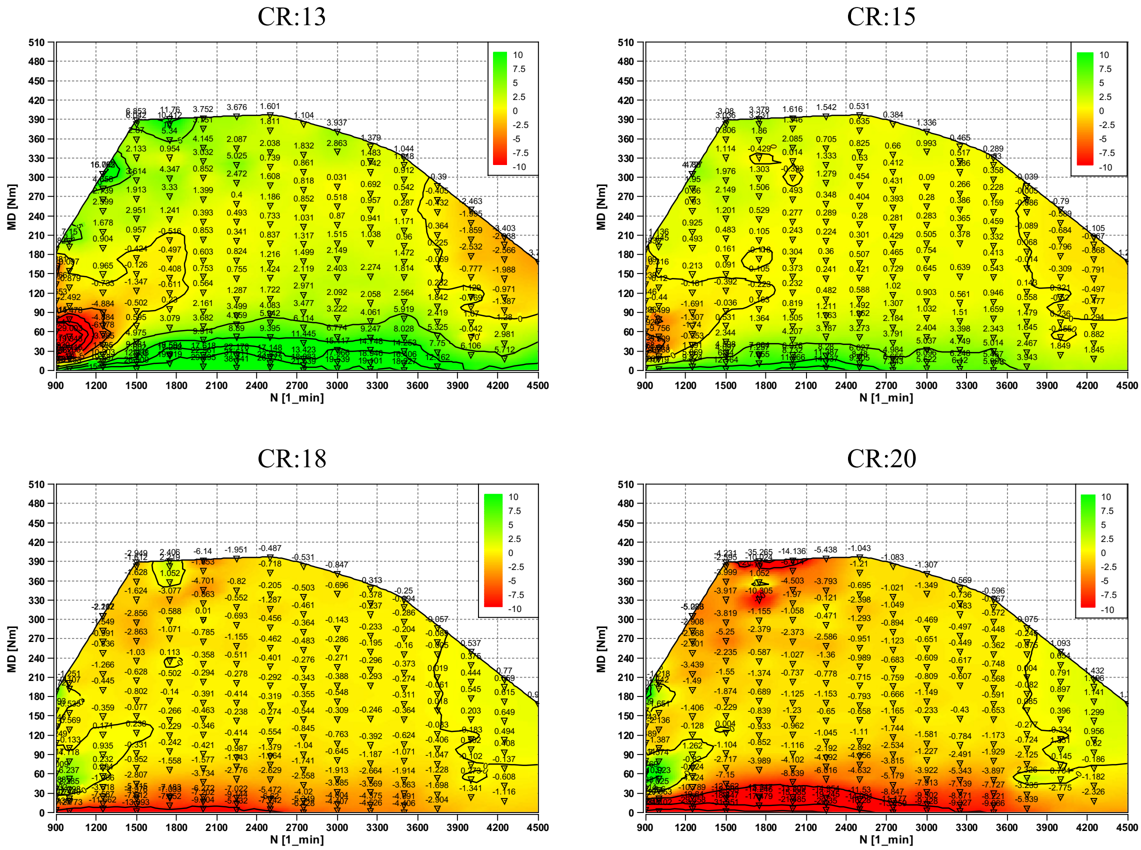

Fig. 24 Engine out NOx as compared with CR16.5

conditions to make sure they pass the quality gates of the characterized parameters of the engine model.

4.3 VCR simulation setup

The numerical model inputs tab allowed the compression ratio to be connected as a parameter of input and thus change it according to the case to be studied. After the full model was setup and characterized, studies of VCR were conducted by altering this factor as shown in Fig. 6.

5 Results and discussion

5.1 Results from steady state simulation

The steady-state results are shown as below. The map shows the difference between the simulation and measurement data (measurement-simulation) where the green colour dominates the model to be working accurately:

z-axis = Measurement Result − Simulation Result

As can be seen in Figs. 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17 and 18, the difference between the results obtained from steady-state simulations of the model and the measurement data is negligible for most of the regions of the maps. This indicates that the characterised numerical model is working very similar to the actual engine.

VCR Simulation study was conducted after the validation of the correct working of the open loop numerical model of the engine.

5.2 Results from numerical model with VCR

5.2.1 Strategy of VCR implementation in numerical model

All the maps of the engine are given in terms of rpm on x-axis, torque on y-axis and percentage change on z-axis. The main strategy applied to the model was to keep the torque and rpm constant with the operational map of CR16.5. At different compression ratios, torque was expected to increase or decrease for the same fuel quantity, thus a shift in the whole map can occur and could make the comparison of the graphs complicated and

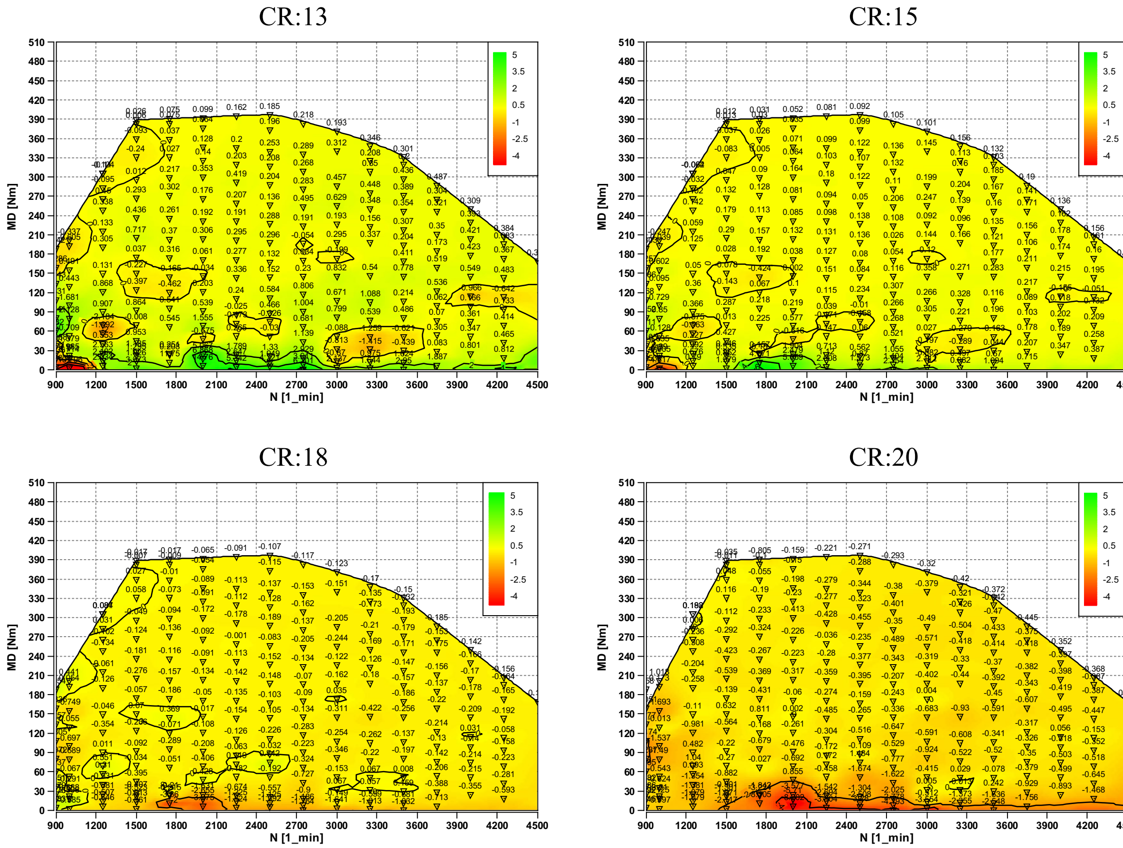

Fig. 25 Mechanical efficiency as compared with CR16.5

more difficult. Therefore, the same set-point torque was achieved by connecting a PID to control the fuel quantity.

For different compression ratios, as the fuel quantity changes to achieve the same operational conditions, mass-flow of intake air also changes, while EGR valve position remains the same causing the Intake Air:EGR ratio to change for the particular set-point. To study changes in NOx more accurately, the percentage of mass flow of EGR at each log point was kept the same with the original data of CR16.5 through the following formula:

$$MF_{EGR} = \frac{MF_{EGR}}{MF_{IA} + MF_{EGR}} \times 100\% \tag{4}$$

The graph of the percentage of EGR in the total mass flow taken from CR16.5 is shown in Fig. 19.

The results are calculated as percentages of increase or decrease as compared to CR16.5:

$$Result = \frac{NewCR - CR16.5}{CR16.5} \tag{5}$$

5.2.2 Mass flow of fuel

As explained earlier, the torque was kept constant at every set-point by changing the fuel quantity using a PID.

As the compression ratio increases, the torque produced for the same fuel quantity also increased, thus PID decreased the torque to the original point by decreasing the fuel quantity.

The opposite can be seen when the compression ratio was decreased, which was due to more fuel was required to produce the same set-point torque. Figure 20 illustrates a decrease in fuel quantity as compression ratio is increased and an increase in fuel quantity as compression ratio is decreased.

5.2.3 Brake specific fuel consumption

The figures shown in Fig. 21, shows the percentage change in the brake specific fuel consumption values at different compression ratios as compared with compression ratio 16.5. It can be clearly observed that the BSFC increases as compression ratio decreases and decreases as the compression ratio increases.

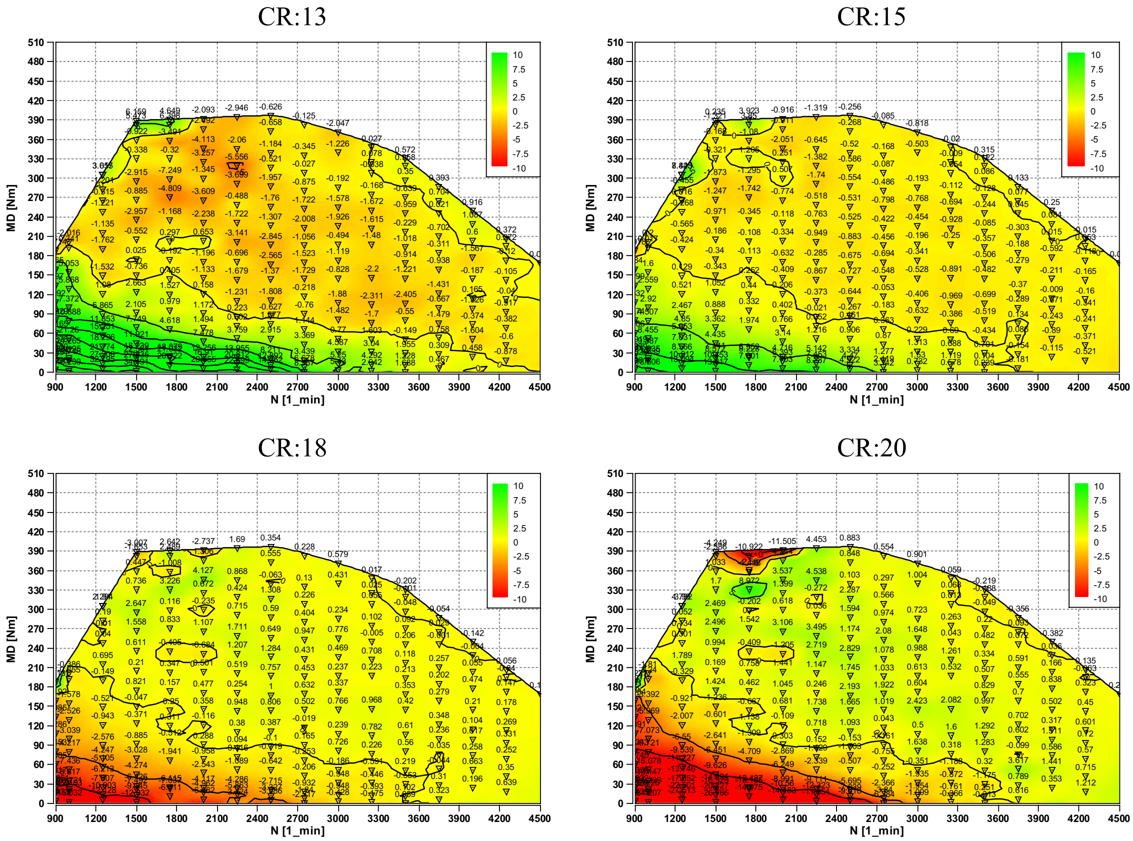

Fig. 26 Engine out CO_2 as compared with CR16.5

The effect of this change is seen to be more at low load-low rpm conditions and this trend decreases at high load-high rpm conditions.

5.2.4 Mass flow of intake air

As shown in Fig. 20, a decrease in compression ratio compels an increase in mass flow of fuel to achieve the same set-point torque. Since the mass flow of fuel is increased, this also increases the mass flow of intake air as seen in Fig. 22, when the compression ratio decreases. Vice versa phenomenon is observed when compression ratio is increased.

5.2.5 Thermal efficiency

The figures shown in Fig. 23 shows the percentage change in the thermal efficiency values at different compression ratios as compared with compression ratio 16.5. Thermal efficiency is:

$$\eta_t = \frac{W_c}{m_f Q_{HV}} = \frac{P_s}{\mu_f Q_{HV}} \tag{6}$$

where W_c is work per cycle, P_s is power output, m_f is mass of fuel per cycle, QHV is heating value of fuel and μ_f is Fuel mass flow rate.

Q_{HV} for diesel is 43.5 MJ/kg so Eq. (6) can be written as:

$$\eta_t = \frac{1}{Sfc \cdot Q_{HV}} = \frac{3600}{Sfc\,(\mathrm{g/kW\,h})Q_{HV}\,(\mathrm{MJ/kg})} = \frac{82.76}{Sfc} \tag{7}$$

Matching the inverse proportionality of the formula shown above and trend shown in BSFC values in Fig. 23, it can be clearly observed that the thermal efficiency increases as compression ratio increases and decreases as the compression ratio decreases.

This change is seen to be more prominent at low load-low rpm conditions and becomes less prominent towards high load-high rpm conditions.

5.2.6 NOx

Figure 24 shows that with an increase in compression ratio there is a decrease in NOx and vice versa.

This phenomenon is based on the fact that a higher compression ratio means a more complete combustion and combining it with the observation from Fig. 20, less

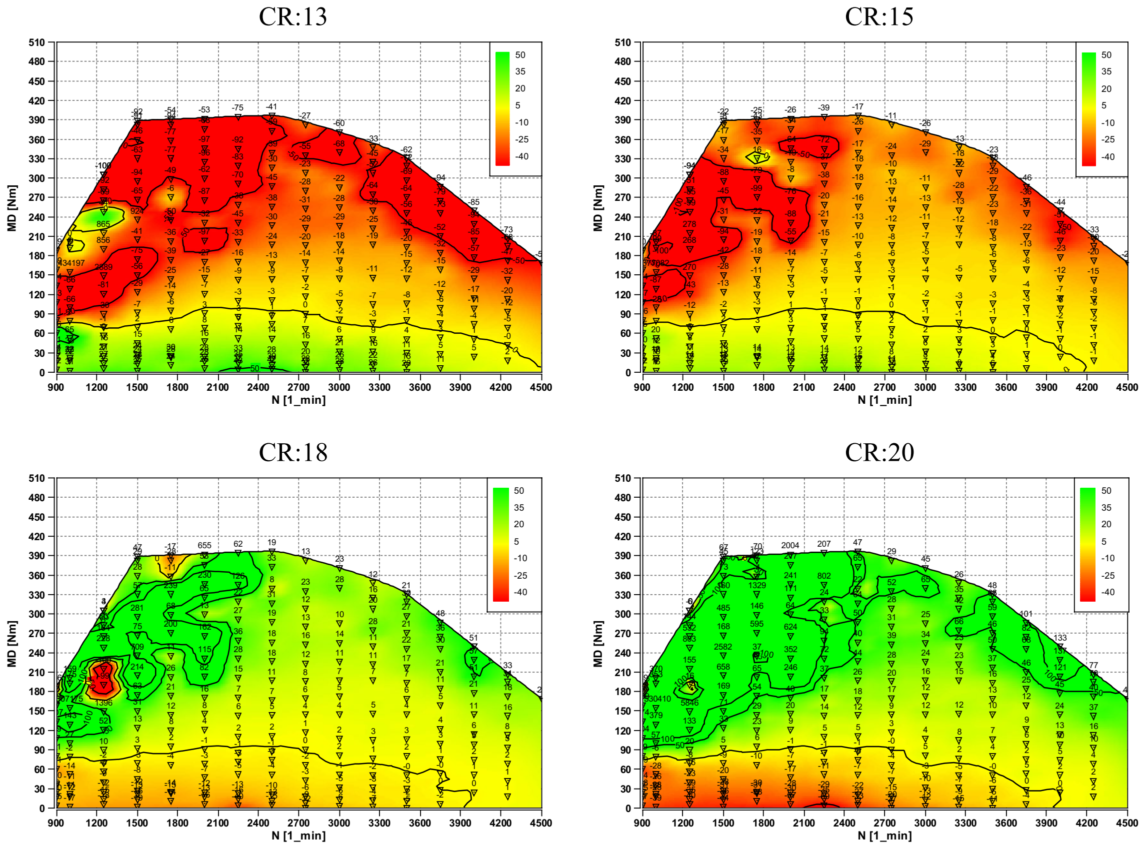

Fig. 27 Engine out soot as compared with CR16.5

fuel is needed to achieve the same torque, means less heat energy will be produced which will ultimately result in a lower temperature; reducing the production of NOx.

The effect is more prominent at low load-low rpm conditions while some of its prominence can also be seen at high torque-low rpm values.

5.2.7 Mechanical efficiency

Figure 25 shows that with an increase in compression ratio, there is a slight decrease in the mechanical efficiency of the engine. The opposite is observed when the compression ratio is decreased. Although the percentage change is very small, ME decreases because as CR increases, frictional losses of the mechanical components along with some pumping losses also increase (especially in the low torque operational areas). This ultimately results in slightly affecting the overall mechanical efficiency of the engine.

5.2.8 Co$_2$

It can be clearly observed from Fig. 26, that the amount of CO_2 produced is inversely proportional to compression ratio. The effect of this change of compression ratio on this emission is greatly at low torque-low rpm values. This is because at low torque-low rpm the air/fuel mixture (low in quantity as compared to high torque-high rpm) in the cylinder is not homogenously compressed and burnt in the cylinder resulting in a poor combustion.

5.2.9 Soot

Figure 27 shows the effect of compression ratio on the percentage change in Soot. With an increase in compression ratio, soot increased at a very sharp rate throughout the map except from low torque-low rpm operation conditions (where it was observed to decreased). This is because at high torque-high rpm conditions the amount of air needed for a more complete combustion was less than the amount ingested by the engine. Thus, a supercharger can be a suggestive solution to increase the amount of air in the cylinder during high load operational conditions to decrease soot production.

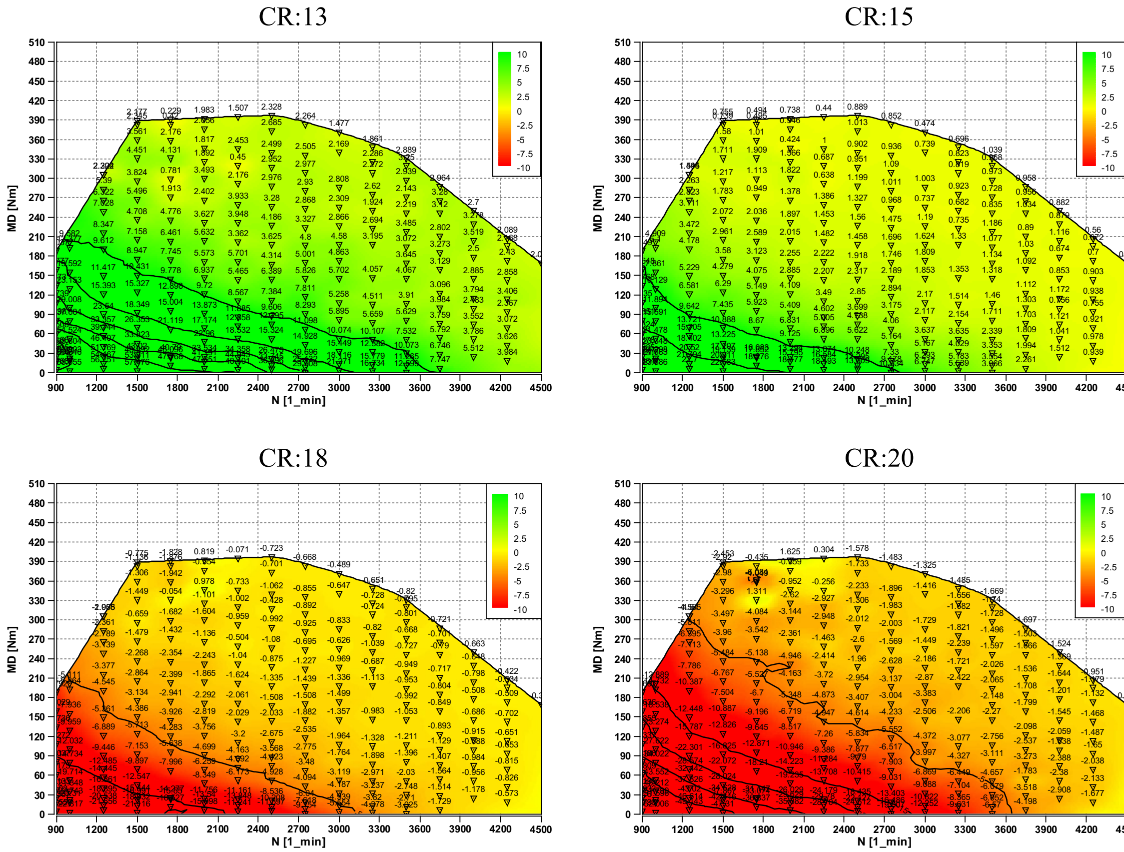

Fig. 28 Engine out temperature as compared with CR16.5

5.2.10 Engine out temperature

Exhaust temperature of the cylinder is directly proportional to the amount of combustion. Combustion in turn is directly related to the amount of injected fuel; more fuel would mean more combustion leading to more release of heat energy. Therefore, as shown in Fig. 20, an increase in compression ratio decreases the fuel consumption to achieve the same torque, thus it results in a decrease exhaust temperature as illustrated in Fig. 28.

5.2.11 Engine out pressure

In the early stage design of the engine, the pressure difference between the intake and exhaust manifolds of the engine is always considered and studied with importance, to make sure it remains within an acceptable threshold.

A wide difference could result in malfunctioning of various engine components. For instance, if the difference is too high, a slight opening of EGR valve could result in massive flow of air to pass by it and could ultimately lead to a malfunction of airflow control.

Figure 29 shows that an increase in compression ratio decreases exhaust pressure because of the phenomenon that less fuel is burnt with an increasing compression ratio to achieve the same operational conditions throughout the map.

5.3 Summary of results based on zones

The engine map was categorized in three zones (as shown in Fig. 30):

Zone 1-covering low load area from low until normal operating rpm values;
Zone 2-covering middle load area from normal until high operating rpm values;
Zone 3-covering high load area (including rated power) with very high rpm values

As can be seen from Fig. 31, thermal efficiency at high load regions of Zone 2 and Zone 3 is approximately around 40%.

A high compression ratio in these zones only increases it by 0.71% and 0.30% with CR18 and 1.57% and 0.40%

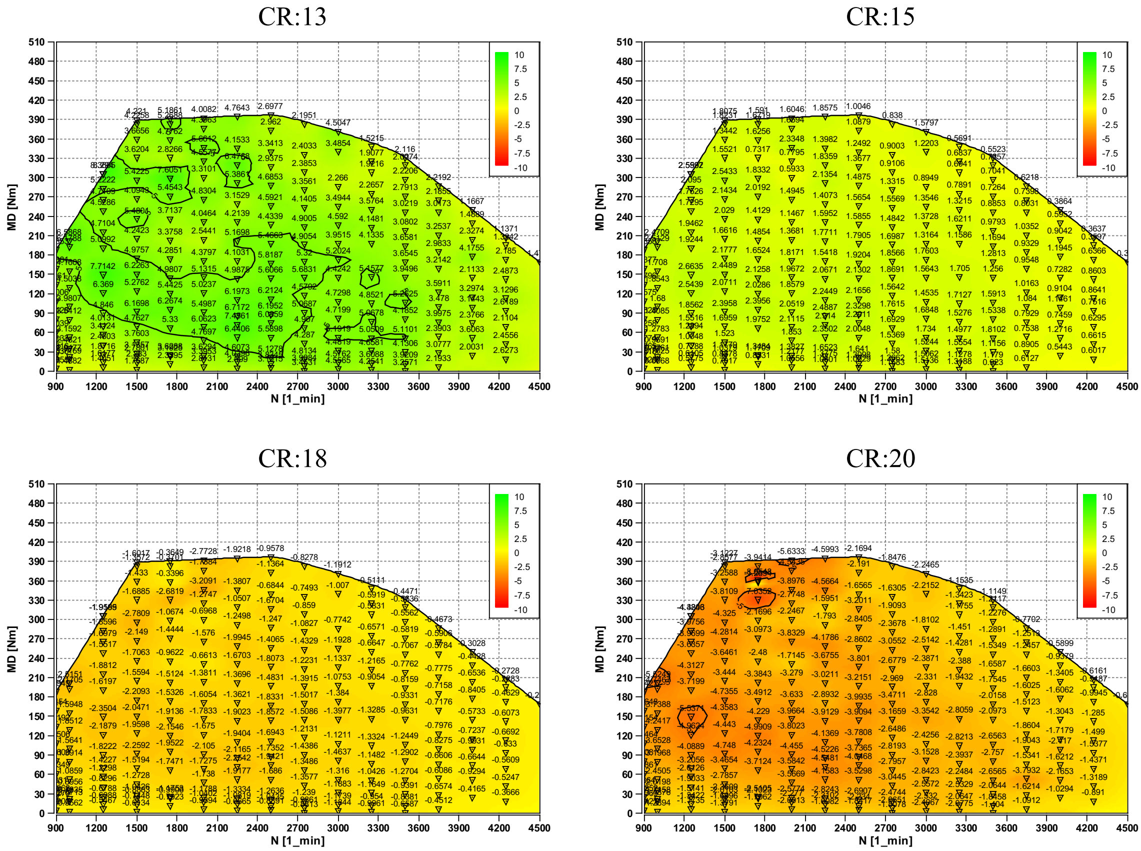

Fig. 29 Engine out pressure as compared with CR16.5

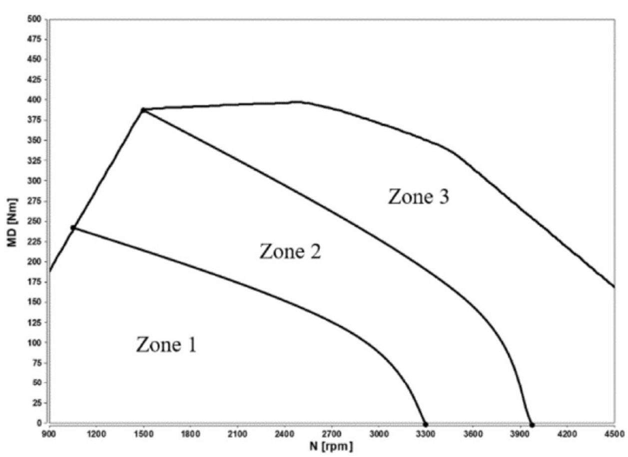

Fig. 30 Zones in engine map

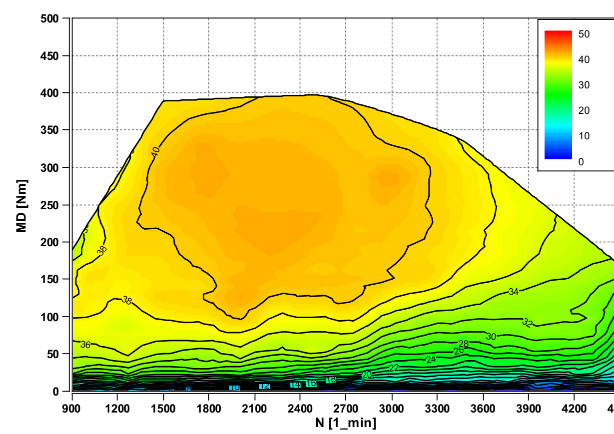

Fig. 31 Thermal efficiency at CR16.5

with CR20. Thus, this change is not as much as can be observed at low load conditions, where normal thermal efficiency was around 25% and was observed to increase by 9.70% at CR20.

The summary of the percentage changes in the values in the three different zones can be seen in Figs. 32, 33 and 34.

BSFC decreases by 8.14%, 1.54% and 0.37% in Zone 1, 2 and 3 respectively when compression ratio is 20. Observing changes in emissions at the same set point, it

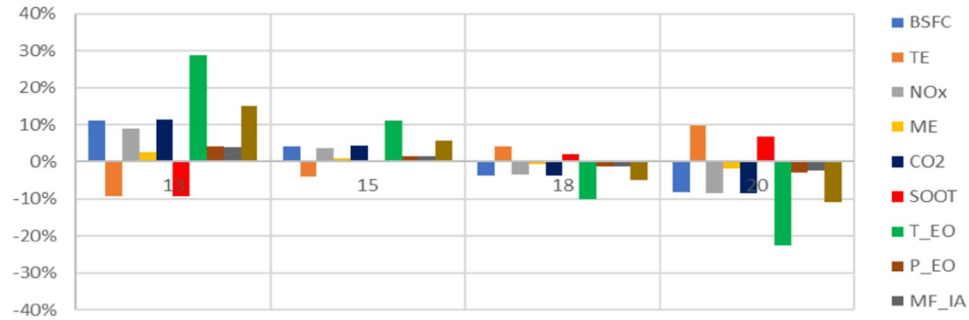

Fig. 32 Average percentage change in Zone 1 as compared to CR16.5

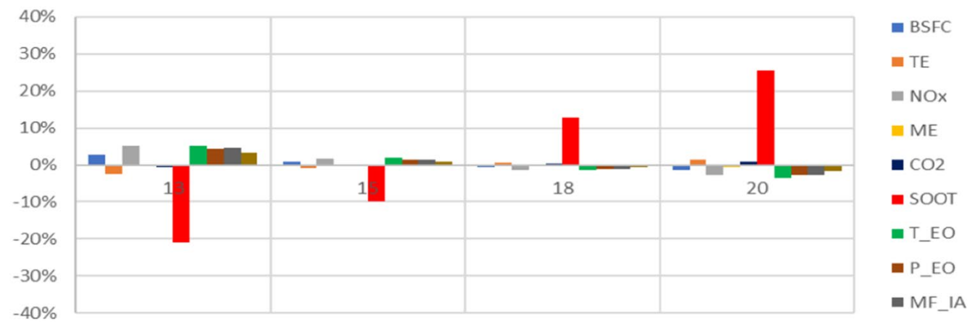

Fig. 33 Average percentage change in Zone 2 as compared to CR16.5

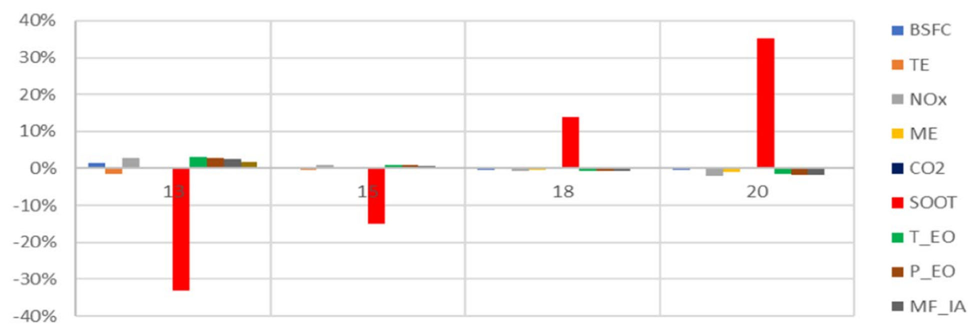

Fig. 34 Average percentage change in Zone 3 as compared to CR16.5

is seen that NOx decreases by 8.33%, CO_2 by 8.50% while Soot increases by 6.91%. This clearly suggests that a higher compression ratio should be preferable in Zone 1.

Observing soot and BSFC values with CR20 in Zone 2 and 3 clearly suggest that it is not preferable as a good trade-off. Soot values clearly suggest that a low compression ratio is preferable for Zone 2 and 3. For a smooth transition between Zone 1 and 2, a compression ratio lower than 18 for Zone 2 may result in a sudden jerk produced by the engine. Keeping this in mind, a compression ratio of 18 is preferred for Zone 2.

The transition of compression ratios between the zones is always kept as smooth as possible to avoid any sudden unwanted behaviour from the engine.

Finally, to keep up with the smoothness trend and to avoid any further decrease in performance, the default compression ratio of 16.5 is preferred in Zone 3.

6 Summary/conclusions

The main effect of an increase in compression ratio, was found to be, as expected, a decrease in brake specific fuel consumption (BSFC) and increase in thermal efficiency. However, this trend comprised on the best efficiency-emissions trade-off for low rpm-low load region of the map (Region 1). With an increase in load and rpm

(Region 2 to Region 3), the trade-off gap between efficiency and emissions becomes wide off since soot starts to become a prominent factor leading to uncompromisable level.

Assuming, that the instantaneous VCR changing technology can be utilized in the engine, this work found the best combination of compression ratios around the engine load map, giving a best fit of trade-offs of the engine.

The developed numerical model of the engine proved to be a great help in studying VCR concept on an existing engine and proved that numerical modelling can save a lot of expenses in studying concepts like these in the development phase of the vehicle. However, there are some limitations to the numerical model approach such as; not having a complete insight of all the emissions, as improving one factor of emission can prove to vastly effect the emissions which are not given by the numerical model. Moreover, the aging factor of the engine also could not be taken into account using this approach.

The following generalized conclusions can be written from the study performed during experimental work and presented modelling:

- Numerical modelling can greatly aid in getting an idea on the engine being developed at initial stages before the prototype becomes available. The numerical model can also be used in calibration of the engine if soft-ECU is developed in the model;
- An engine equipped with VCR technology can increase the efficiency and decrease the emissions at different operational conditions;
- Increasing compression ratio at low load operations highly increases the thermal efficiency and decreases NOx. At the same time, soot emissions increase in the same operational conditions;
- Future vehicles with the concept of 'one engine all fuels–one fuel all engines' will be key option in development of future transport in the world.

Acknowledgements First, I would like to thank my family for supporting me at all times. I would also like to thank everyone at Marmara University and AVL Research and Engineering Turkey for a nice and stimulating time, especially to my supervisor, Prof. Dr. M. Zafer Gül and Department Manager, Mr. Caner Bayburtlu for their encouragement and valuable ideas. Further, I would like to thank Ramazan Şener for all help in the lab, and Oğuz Ulutürk for his precious time and aid in modelling of the engine. Furthermore, I would like to thank BAPKO (Bilimsel Araştirma Proje Komisyonu) for their support through Project Number: FEN-C-YLP-200318-0113.

Compliance with ethical standards

Conflict of interest On behalf of all authors, the corresponding author states that there is no conflict of interest.

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