

Evaluation of Different Turbocharger Configurations for a Heavy-Duty Partially Premixed Combustion Engine

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ABSTRACT

The engine concept partially premixed combustion (PPC) has proved higher gross indicated efficiency compared to conventional diesel combustion engines. The relatively simple implementation of the concept is an advantage, however, high gas exchange losses has made its use challenging in multi-cylinder heavy duty engines. With high rates of exhaust gas recirculation (EGR) to dilute the charge and hence limit the combustion rate, the resulting exhaust temperatures are low. The selected boost system must therefore be efficient which could lead to large, complex and costly solutions.

In the presented work experiments and modelling were combined to evaluate different turbocharger configurations for the PPC concept. Experiments were performed on a multi-cylinder engine. The engine was modified to incorporate long route EGR and a single-stage turbocharger, however, with compressed air from the building being optionally supplied to the compressor. Experimental combustion heat release rates and boundary conditions were used to validate a simulation model. This model was then used to compare three different turbochargers: two single-stage turbochargers and one two-stage. The whole speed and load range was covered in the simulations to determine the engine performance. The influence of high EGR rates as well as the effect of charge air and EGR cooler gas outlet temperatures were also investigated.

The simulation results showed that the two-stage turbocharger was able to give the highest load over the whole speed range with a brake mean effective pressure of 25.6 bar, whereas the two single-stage turbochargers reached 22.2 and 23.1 bar respectively. The average brake efficiency was 39.3, 39.7 and 40.2 %. It was found that decreasing the inlet temperature is critical for obtaining high loads and system efficiencies. Finally, it was shown that the optimal amount of EGR was dependent on the turbocharger efficiency and cooler temperatures.

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INTRODUCTION

The stricter regulations on emissions and high consumer demands on low fuel consumption are driving engine manufacturers to look for other technology solutions than the standard diesel engines available today.

A great part of the research focus of compression ignition engines has been put on reducing the in-cylinder gas temperature in order to reduce energy loss from heat transfer and thereby increasing the overall engine efficiency. The lower combustion temperature is also leading to a simultaneous reduction of nitrous oxides (NO_x) and particulate matter, [1, 2, 3]. Examples of low temperature combustion concepts are homogeneous charge compression ignition (HCCI) (e.g. [4, 5, 6]), partially premixed combustion (PPC) (e.g. [7, 8, 9]) and reactivity controlled compression ignition (RCCI) (e.g. [10, 11, 12]). Common to all these concepts is that they depend on high rates of exhaust gas recirculation (EGR) and high amounts of excess air.

Researchers use single-cylinder experiments when new engine concepts are developed, e.g. [13, 14, 15]. An external source is most often used for air supply which is convenient because the intake manifold pressure can be regulated independently of exhaust manifold enthalpy. When conducting such an experiment it is likely to find that higher air-fuel-ratio (i.e. higher intake manifold pressure) leads to an increase in gross indicated efficiency, [16]. This might be a problem when trying to transfer the findings from single-cylinder experiments to a multi-cylinder engine located on a vehicle where the elevated intake manifold pressure must be produced by a source that uses the exhaust energy (e.g. a turbocharger). Additionally it is common to use heat exchangers which cool the hot gases before the intake manifold, [17]. There are typically two coolers, one to cool the charge air and one to cool the EGR. A lower temperature increases the gross indicated efficiency and the achievable load as well as it suppresses NO_x emissions, [18]. It is possible to use cold tap water in

the experimental setup because the engine is stationary. In contrast, the charge air cooler on the truck is typically using ambient air and the EGR cooler is using engine coolant water. It is therefore possible to reach significantly lower intake manifold temperatures in the experimental setup. A heat exchanger that is subjected to low temperature can experience fouling due to water condensation. In case of a long route EGR arrangement in which the EGR and air are mixed before the compressor, water condensation will cause significant damage to the impeller and the compressor efficiency will decrease, [19]. In order to prevent condensation the gas temperature must be kept above the dew temperature.

Caton et al. conducted engine cycle simulations to thermodynamically investigate the reasons behind the high efficiencies that are obtained with low temperature combustion concepts, [20, 21, 22, 23, 24]. High amounts of air and EGR significantly decrease the engine heat transfer and increase the ratio of specific heats resulting in high gross indicated efficiencies. By applying a second law analysis, they show that although the destruction of exergy increases as the in-cylinder temperature decreases, the thermal efficiency increases. This emphasizes the importance of low combustion temperatures to produce high gross indicated efficiencies. In addition their results show that the high rates of EGR and air lead to a lower mechanical and gas exchange efficiency¹.

Chadwell et al. combine experimental data from the literature and engine simulations to show the differences in performance of an engine using four different low temperature combustion concepts and conventional diesel combustion, [25]. They show that the low temperature combustion concepts provide higher gross indicated thermal efficiencies but the brake efficiency is as high as with conventional diesel combustion. This is because the exhaust temperatures in the low temperature combustion concepts are so low that there is not enough energy available for the turbocharger and this leads to low gas exchange efficiencies.

In contrast, Tunér et al. showed in a theoretical study that their heavy-duty engine, when using PPC, is able to produce a peak brake thermal efficiency of 48 % and that the brake efficiency is above 42 % in the majority of the operating range, [26]. Instead of using a single-stage turbocharger, they make use of a two-stage turbocharger.

These research findings points to an existence of a trade-off between high gross indicated efficiency and low gas exchange efficiency for the low temperature combustion concepts. Here we have evaluated four different boosting configurations in order to see if the change of turbocharger can increase the gas exchange efficiencies in PPC. Engine experiments were conducted using a multi-cylinder, heavy-duty compression ignition engine operating in PPC mode. An inadequate induction system limited the engine speed and load range. To overcome this limitation the data was extended using a theoretical approach. The engine test rig was modelled in the one-dimensional tool GT suite and validated against the experimental data using

operating conditions that could be run in the experiments. After the validation was completed, the model was used to estimate the performance of the engine and the maximum load and speed range that could be obtained when using the different turbochargers. The findings in this work indicated that it is possible to reach higher engine load and engine efficiency by carefully choosing the turbocharger design.

EXPERIMENTAL PROCEDURE

The engine experiments were run on a multi-cylinder, heavy-duty compression ignition Scania D13 engine. The details of the test cell setup including the control system, the injection XPI common rail system and signal processing are described in recent publications, e.g. [27,28], and will not be reproduced here. A schematic view of the engine is given in Figure 1 and brief specifications are given in Table 1.

Recent studies, e.g. [29], on PPC have shown that a fuel with low reactivity, e.g. gasoline, is beneficial for high gross indicated efficiencies. It should, however, have a lower octane rating than regular service station gasoline. Therefore, a mixture consisting of 80 volume % service station gasoline of research octane number (RON) 95 and 20 volume % n-heptane, with a resulting RON of ~76, was used in this study. The properties of the fuel mixture are shown in Table 2. The test rig was supplied with compressed air from the building at conditions² when the turbocharger could not produce the desired boost pressure. In the engine's single-stage turbocharger, the EGR and air were compressed further to yield the ultimate boost pressure. A total of three coolers were situated on the test rig for charge air and long and short route EGR. Only the long route EGR line was used in the experiments conducted in this work. Tap water was used as the coolant medium in the charge air and EGR coolers. The fast thermal valve (FTM) was controlled in order to keep the gas outlet temperature at 20 °C.

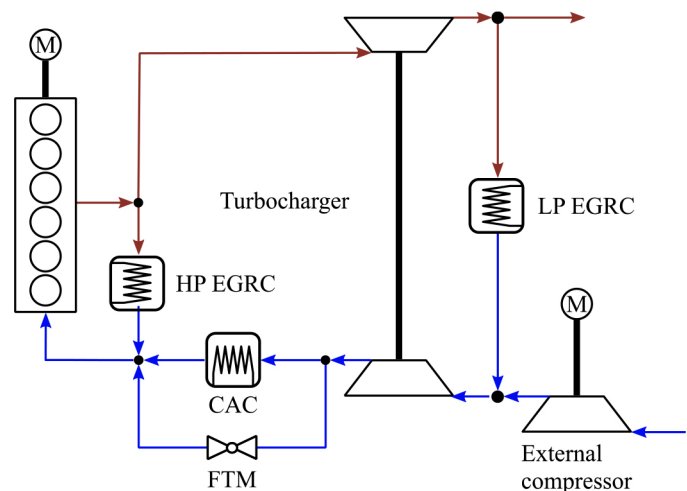


Figure 1. Schematic layout of the multi-cylinder, heavy-duty compression ignition test engine used in the experiments, [30].

1. The engine brake efficiency can be divided into four part efficiencies; combustion efficiency, thermodynamic efficiency, gas exchange efficiency and mechanical efficiency, [37].

2. This only occurred at the higher loads. A higher back pressure after the turbine was then applied in order to drive the EGR.

Table 1. Scania D13 test engine specifications.

Item	Value
Displacement	12.74 l (in-line 6)
Bore × Stroke	130 mm × 160 mm
Connecting rod length	255 mm
Compression ratio	17.3:1
Number of valves	4 per cylinder
Inlet valve closing angle	-151 °CA ATDC
Exhaust valve opening angle	137 °CA ATDC
Injector system	XPI common rail
Injector dimensions	8 × 148° × 0.175 mm
EGR system	Short and long route
Induction system	Single stage waste gate turbocharger + external air source

Table 2. The properties of the fuel mixture used in the experiments.

Item	Value
95 RON service station gasoline	80 volume %
n-heptane	20 volume %
RON	76
Carbon atoms	7.8
Hydrogen atoms	17.63
Oxygen atoms	0
Lower heating value	43.7 MJ/kg
Stoichiometric air fuel ratio	15.67

$$\text{lambda}, \lambda = \frac{\text{AFR}}{\text{AFR}_{\text{stoichiometric}}}$$

(1)

$$\text{gross IMEP (L)} = \frac{\int_{-180}^{180} p dV}{V_d}$$

(2)

$$\text{EGR rate} = \frac{(x_{CO_2})_{\text{intake}}}{(x_{CO_2})_{\text{exhaust}}}$$

(3)

The goal of the experiments was to achieve a high overall gross indicated efficiency using the PPC concept. This was done by optimising the following set of parameters; lambda λ , EGR rate (%), intake manifold temperature T_{im} (°C), EGR cooler gas outlet temperature T_{EGR} (°C), start of injection SOI (°CA ATDC) and the number of injections n_{inj} . The definitions for lambda³, EGR and gross IMEP can be seen in Equations (1), (2), (3). A total of 17 operating points were run and they are listed in Table 3.

Table 3. Engine operating conditions used in the experiments. Engine speed N (rpm), gross IMEP L (bar), EGR rate (%), intake manifold pressure p_{im} (bar), lambda λ (-), intake manifold temperature T_{im} (°C), start of injection SOI (°CA ATDC) and number of injection events n_{inj} .

N	L	EGR	P_{im}	λ	T_{im}	T_{EGR}	SOI	n_{inj}
1300	5	49	1.24	1.94	24	17	-30.4	2
1300	8	46	1.54	1.45	22	16	-24.0	2
1300	10	47	1.77	1.33	23	17	-10.4	1
1300	12	42	2.03	1.52	23	20	-8.8	1
1300	13	44	2.18	1.51	23	17	-9.4	1
1300	15	44	2.42	1.39	22	17	-9.6	1
1300	19	45	2.89	1.22	22	20	-7.6	1
1400	5	53	1.21	1.70	21	16	-34.8	2
1400	7	51	1.30	1.31	22	16	-34.8	2
1400	10	47	1.62	1.31	21	16	-36.8	2
1400	13	47	1.97	1.19	21	16	-37.0	2
1400	19	42	2.52	1.32	22	21	-9.6	1
1600	5	50	1.27	1.79	23	16	-23.8	1
1600	8	49	1.49	1.44	23	16	-17.8	1
1600	10	50	1.81	1.45	23	16	-16.2	1
1600	13	46	2.19	1.35	22	16	-12.4	1
1600	16	47	2.45	1.20	22	21	-12.4	1

MODEL SETUP

Engine cycle simulations were performed in the 1-D modelling tool GT-power [31]. The work was divided into two parts. The first part focused on building and validating a detailed model of the test engine against the experimental data. Applying the model, the latter part focused on evaluating the performance of the engine when using four different turbochargers.

Validation of Simulation Model

The validation of the model followed, principally, the approach outlined in [10,32]. The detailed model included inlet air path, turbocharger, EGR loops (with coolers), charge air cooler, engine crank train, engine combustion cylinders and outlet exhaust path. The valve lift profiles were measured on the engine and valve lashes were calibrated using the model. The combustion was governed by the post-processed rate of heat release from the experiments.

As a first step the engine was validated without the turbocharger to simplify troubleshooting. The same operating conditions were run as in the experiments and the boundary conditions, including for example charge air and EGR cooler gas outlet temperatures, EGR rate, air mass flow, fuel mass flow, injection parameters, rate of heat release and exhaust manifold pressure were imposed. Experimental data, such as in-cylinder pressure, exhaust temperatures and static pressure measurements which were not used as boundary conditions, were used to validate the match between experiments and simulations. In the second step the turbocharger was added, however, without performance maps because they were not available for the experimental turbocharger. Instead simple turbine and compressor models were used with prescribed isentropic efficiencies to impose the correct intake and exhaust manifold pressures.

3. This definition takes into account the air in the EGR and residual gas.

In-cylinder heat transfer was modelled using a correlation from [33]. A convective heat transfer multiplier was used to scale the total heat transfer. Especially the expansion phase was sensitive to heat transfer and the multiplier was calibrated in order to get a match between the simulated in-cylinder pressure traces and the experimental traces. It was found that a value of 0.5 for this multiplier satisfied most of the operating points. Therefore this value was chosen for all cases in the rest of the simulation work.

Simulating Engine Performance

In the second phase of the modelling work, engine simulations were conducted to evaluate the performance of this engine when it is run with PPC. The operating conditions (i.e. load and speed) were selected to attempt to cover the complete range of the engine. They are presented in Table 4 together with the EGR rate (%), lambda, λ , intake manifold temperature, T_{im} (°C), EGR cooler gas outlet temperature, T_{EGR} (°C), SOI (°CA ATDC) and the number of injections, n_{inj} . All values were selected based on the experimental data. The maximum gross IMEP, at each engine speed, was determined using an optimisation approach in which the brake mean effective pressure (BMEP) was maximised while the inlet manifold pressure and start of injection were varied. The constraints for this optimization were values on EGR, lambda and intake manifold temperature. Additional constraints were put on the peak cylinder pressure which could not exceed 225 bar and the start of injection was limited in order to have the combustion phasing (50 % burned) no later than 15 °CA ATDC. At some operating conditions, which will be shown subsequently, the desired load could not be achieved with the selected combination of lambda and EGR. The load was in those cases decreased until a value that could be met was obtained.

Table 4. Engine operating conditions used in the second part of simulation work. Engine speed, N (rpm), gross IMEP, L (bar), EGR rate (%), lambda, λ (-), intake manifold temperature, T_{im} (°C), start of injection, SOI (°CA ATDC), number of injection events, n_{inj} .

N	L	EGR	λ	T_{im}	T_{EGR}	SOI	n_{inj}
800	5	49	1.94	22	18	-30.4	2
800	13	44	1.51	22	18	-9.4	1
800	19	45	1.22	22	18	-7.6	1
800	Max.	45	1.20	22	18	Opt.	1
1300	5	49	1.94	22	18	-30.4	2
1300	13	44	1.51	22	18	-9.4	1
1300	19	45	1.22	22	18	-7.6	1
1300	Max.	44	1.20	22	18	Opt.	1
1600	5	50	1.79	22	18	-24.2	1
1600	13	46	1.36	22	18	-12.4	1
1600	19	47	1.22	22	18	-7.6	1
1600	Max.	45	1.20	22	18	Opt.	1
1900	5	50	1.79	22	18	-23.8	1
1900	13	46	1.36	22	18	-12.4	1
1900	19	47	1.22	22	18	-7.6	1
1900	Max.	45	1.20	22	18	Opt.	1

Higher Charge Air and EGR Cooler Gas Outlet Temperatures

The outlet gas temperatures from the charge air (CA) and EGR coolers were extremely low in the experiments. The implications that this could have on the heat exchangers and compressor were described earlier in the introduction. Therefore a second set of simulations were performed in order to evaluate the effect of a higher cooler temperature on the load range and the brake efficiencies. In these simulations the outlet gas temperature was set according to Equations (4) and (5).

$$T_{EGR} = \begin{cases} T_{dew, EGR} + 20 & \text{if } T_{dew, EGR} + 20 > 90 \\ 90 & \text{if } T_{dew, EGR} + 20 < 90 \end{cases} \quad (4)$$

$$T_{CA} = \begin{cases} T_{dew, CA} + 20 & \text{if } T_{dew, CA} + 20 > 40 \\ 40 & \text{if } T_{dew, CA} + 20 < 40 \end{cases} \quad (5)$$

The dew point temperature determines when condensation occurs. Therefore to prevent condensation in the coolers the temperature cannot go below the dew point. A safety margin of 20 °C was included. It was assumed that the coolant medium in the EGR cooler was the engine coolant fluid and that the minimum temperature that could be reached was 90 °C. The charge air cooler was on the other hand assumed to be using ambient air as a cooling medium and it was assumed that the minimum gas outlet temperature that could be reached in the charge air cooler was 40 °C.

Influence of EGR

The influence of EGR on system performance was investigated in the final part of the simulation work. The effect of turbocharger efficiency and engine boost pressure on engine exhaust pressure was analysed first using a theoretical analysis (see Equations (A1), (A2), (A3), (A4), (A5), (A6), (A7), (A8) and Table A1). Finally, and using the GT simulation model, an EGR sweep was performed at the maximum load of every engine speed while the lambda was kept constant.

Extending the Operating Range

Rate of heat release (RoHR) was needed for the extended operating range for the speeds and loads that were not run in the experiments. While applying the PPC concept, Tunér et al. demonstrated how a RoHR could be extrapolated using RoHRs at other engine speeds and loads, [32]. When varying the engine speed, at constant load, the shape of the RoHR as a function of crank angle degrees does not change. Therefore, the RoHRs at 1300 rpm were used at 800 rpm and the RoHR at 1600 rpm were used at 1900 rpm. When increasing the load, at constant speed, the premixed part of the RoHR decreases while the diffusion part increases. This can be seen in Figure 2 and Figure A1, A2, A3, where the RoHRs at 1300, 800, 1600 and 1900 rpm are shown which were used in this work. In this figure the RoHRs are normalised with the maximum value (J/°CA) of each trace.

Table 5. The specifications of the four different turbochargers used in the modelling work. The diameters of the turbine and compressor impellers are normalized with respect to TC1's turbine diameter.

	HP turbine	HP compressor	LP turbine	LP compressor
TC0	$\eta_{is} = 78 \%$, varying diameter	$\eta_{is} = 78 \%$	(-)	(-)
TC1	VGT, diameter = 1.00 (-)	diameter = 1.10 (-)	(-)	(-)
TC2	VGT, diameter = 1.05 (-)	diameter = 1.21 (-)	(-)	(-)
TC3	VGT + WG, diameter = 1.05 (-)	diameter = 1.57 (-)	WG, diameter = 1.46 (-)	diameter = 1.91 (-)

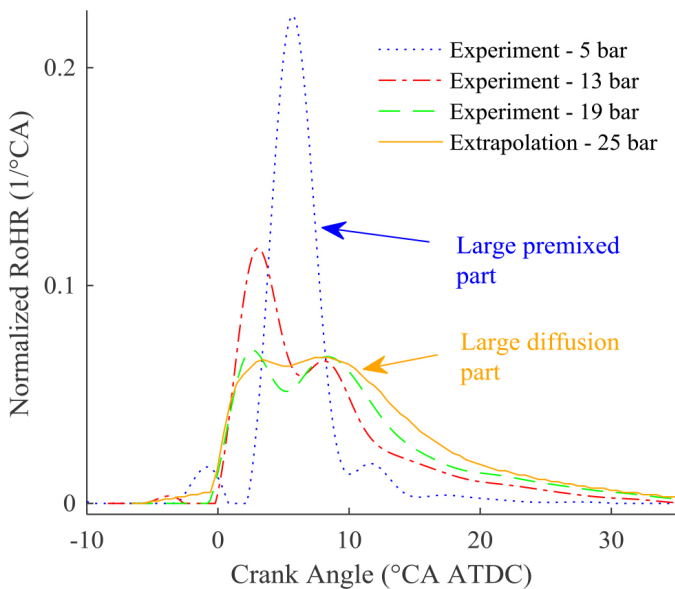


Figure 2. Normalized rate of heat release at 1300 rpm. 5, 13 and 19 bar IMEP gross corresponds to experimental data and 25 bar IMEP gross is extrapolated from the other three curves.

Friction

A model estimating the engine friction was needed in order to predict the brake efficiency. In this work a friction model was used that is outlined in [34]. The model is based on a single equation, presented in Equation (6), which is dependent on the engine load, the mean piston speed (i.e. engine speed), and the peak cylinder pressure.

$$\text{FMEP} = c \cdot 0.0104 \cdot \text{IMEP} + 0.15 \cdot V_p + 1.0955 \cdot 10^{-3} \cdot p_{\max}$$

$$V_p = 2 \cdot S \cdot N$$

$$c = \begin{cases} 1, & \text{at 50 \% load} \\ 2, & \text{at 100 \% load} \end{cases}$$

(6)

The resulting friction mean effective pressure (FMEP) as a function of engine speed and load is shown in Figure 3.

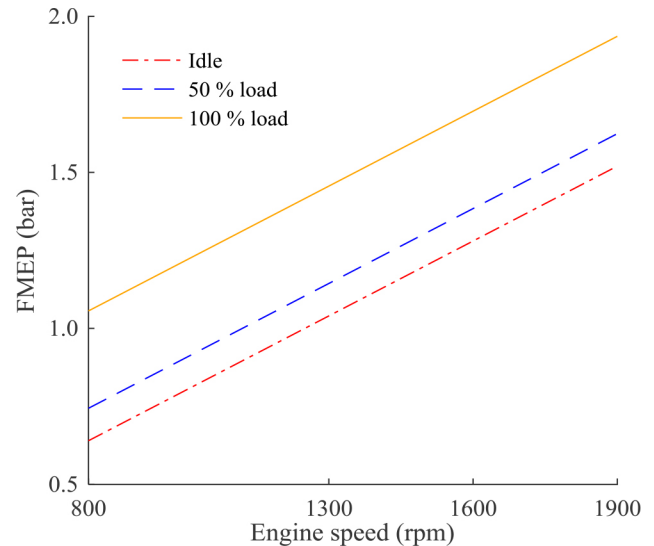


Figure 3. Engine friction model used in the simulations. Taken from [34].

EATS and Air Filters

The test engine did not have an emission after-treatment system or inlet air filter although it is common on production engines. This meant that they could not be included in the validation process. These parts introduce flow losses which are important to include in the modelling if the ultimate performance of the engine is going to be predicted. Calibration data from another similar production engine were provided by the engine manufacturer and used in all simulations succeeding the validation phase.

Turbochargers

Four different turbochargers were used in the modelling work. The first one was a shaft-less, single-stage, turbocharger (hereafter called "TC0"), with a prescribed isentropic efficiency of 61 %⁴. The average pressure before the turbine (resulting from different operating conditions) was controlled by changing the turbine effective diameter. The reason for using a non-real turbocharger was that it could be used for a comparable prediction of the limit of the system (engine plus turbocharger). Real turbochargers do not, however, operate with a constant isentropic efficiency. Instead the efficiency varies depending on operating condition. The second and third turbochargers, "TC1" and "TC2", were off-the-shelf production turbochargers from BorgWarner Inc. They consisted of a single stage and the turbine of these was a so called variable geometry turbine (VGT), which means that the swallowing capacity of the turbine could be altered. Choosing

4. The turbine's and the compressor's respective isentropic efficiencies were 61 % and 70 %, which with (7) equals the total turbocharger isentropic efficiency of 61 % when the mechanical efficiency was assumed to be 100 %.

a VGT instead of a turbine with waste-gate (WG) meant that more of the exhaust energy could be utilized, [35]. However, this would come with additional cost and complexity. The performance of TC1 and TC2 had been experimentally measured on a gas stand where gas was supplied to the turbine inlet under steady state conditions. The results were then compiled into standard SAE performance maps and used in the simulations. It was said earlier that the PPC concept requires high boost levels and that this probably needs to occur at high turbocharger efficiencies across the entire operating range. This might not be possible using only a single-stage turbocharger. There exist many alternatives to the single-stage configuration. Here, we chose a two-stage turbocharger, "TC3", in which the low pressure (LP) and the high pressure (HP) stages operated in series. The turbine of the high pressure stage included a VGT and a WG and the low pressure turbine a WG. The performance maps of TC2 were used for TC3 but were scaled in order to obtain the maximum efficiency at an engine speed of 1300 rpm. The turbine and compressor efficiencies were, however, not changed. The exact size of the turbochargers in terms of impeller tip diameter cannot be revealed due to confidentiality. However, in Table 5, normalized diameters with respect to TC1's turbine diameter are shown.

In two-stage configurations, it is common to use an inter-cooler between the low and high pressure compressor stages to cool the gas before the second compression (in the high pressure stage). This is done to decrease the power required in the high pressure compressor and to decrease the compressor outlet temperature to prevent damage. However, an inter-cooler also increases the size and complexity of the system. In this work we decided not to use an inter-cooler because it was difficult to estimate the pressure losses over the pipes in the cooler. However, it must be realised that this will have an effect on the results and conclusions.

A potential problem of using performance maps to predict the interactions between turbocharger and engine is that the flow at the turbine's inlet is all but in steady state which is the case on the gas stand. The implications that this has on the results of an engine simulation are reviewed in [36]. The conclusions are that although it is possible to model and predict the interactions between engine and turbocharger with one-dimensional tools and performance maps, it is not possible to capture the three-dimensional flow effects in the compressor and turbine. In addition, when simulating a two-stage configuration it is difficult to predict the heat transfer that occur between the hot gas in the turbine and the relatively cool gas in the compressor, however this should have a minor effect on the results simulated in this work.

EXPERIMENTAL RESULTS

The combustion efficiencies η_c , gross indicated efficiencies η_g and coefficient of variance of gross IMEP CoV L from the experiments are presented in Table 6. The average combustion efficiency was 99.7 % and the gross indicated efficiency had an average of 51.3 % with a peak of 53.5 %. The measured in-cylinder pressure of every cylinder was averaged over 300 cycles and the rate of heat release of each cycle was calculated using Equation (8), from e.g. [37].

$$\frac{\partial Q}{\partial CA} = \frac{\gamma(p,T)}{\gamma(p,T) - 1} p \frac{\partial V}{\partial CA} + \frac{1}{\gamma(p,T) - 1} V \frac{\partial p}{\partial CA} + \frac{\partial Q_{ht}}{\partial CA} \quad (8)$$

The six curves were then averaged resulting in a single curve. The results are shown for one operating point in Figure 4 where one standard deviation is included which shows the cylinder-to-cylinder variations. A more detailed representation of the experimental results can be found in [28].

Table 6. Experimental results. Combustion efficiency η_c (%), gross indicated efficiency η_g (%), coefficient of variance of gross IMEP CoV L (%) and 50 % burn duration CA50 (°CA ATDC).

N	L	η_c	η_g	COV L	CA50
1300	5	99.6	50.5	2.7	5.5
1300	8	99.9	51.6	1.5	8.5
1300	10	99.9	51.7	1.2	6.6
1300	12	99.9	52.2	2.5	5.9
1300	13	99.9	53.5	0.9	5.7
1300	15	99.9	53.3	1.0	5.9
1300	19	99.8	52.3	1.1	8.0
1400	5	99.3	48.7	7.2	7.5
1400	7	99.5	49.5	2.4	8.4
1400	10	99.8	50.0	1.6	6.4
1400	13	99.5	48.9	1.5	5.9
1400	19	99.9	51.8	1.2	5.7
1600	5	99.5	50.9	2.0	6.4
1600	8	99.8	51.9	4.7	6.6
1600	10	99.8	51.8	1.5	6.1
1600	13	99.9	52.5	4.2	5.6
1600	16	99.7	50.8	1.3	5.6
Average:		99.7	51.3	2.3	6.5

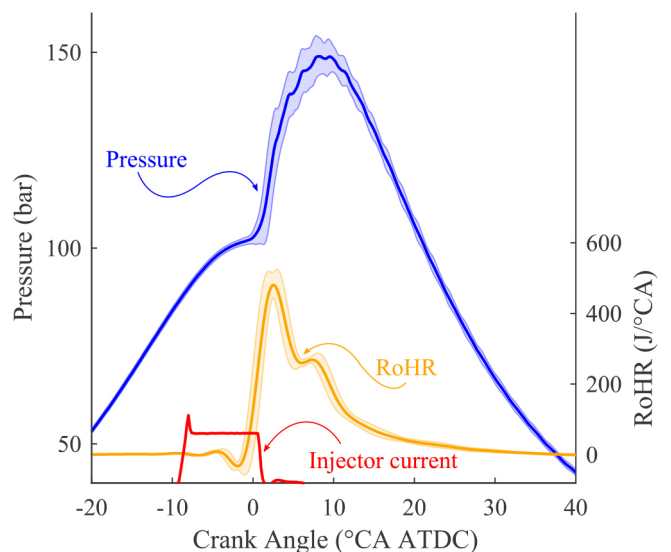


Figure 4. Cycle and cylinder averaged in-cylinder pressure, rate of heat release and injector current (-) of the experimental operating point (13 bar, 1300 rpm) that showed the highest gross indicated efficiency. Solid lines represent the ensemble average value, while shaded regions represent one standard deviation.

MODEL VALIDATION RESULTS

The GT-power model was validated against the experimental data as described earlier. Figure 5 and Figure A4, A5, A6, A7 show the simulated in-cylinder pressure during the gas exchange cycle and a comparison with experimental values for a selection of operating conditions. In Figure 6 the difference in net IMEP between experiments and simulations is presented. The biggest difference was almost 5 % (which translates into approximately 2 percentage points of net indicated efficiency), however, for a parameter as sensitive as the net IMEP, including cylinder-to-cylinder and cycle-to-cycle variations (CoV in Table 6), it was considered well enough for this work.

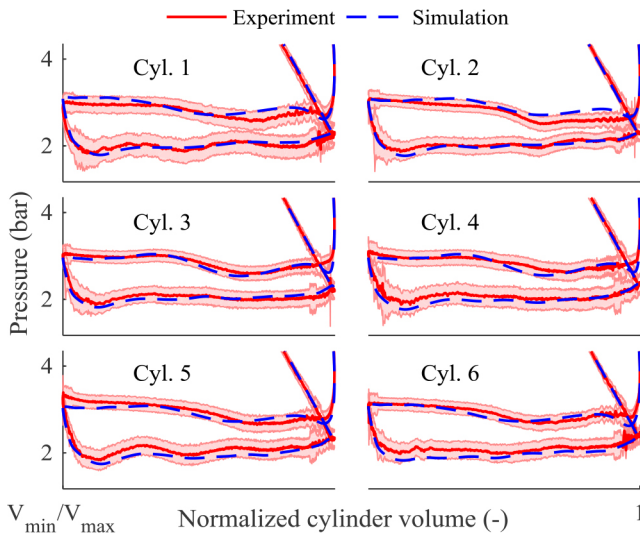


Figure 5. Simulated in-cylinder pressure of the gas exchange cycle and a comparison with the experimental values from the operating condition of 1300 rpm and 13 bar. The red solid lines represent the ensemble average of the experiment, the red shaded regions represent one standard deviation of the experiment and the blue dashed lines represent the simulation.

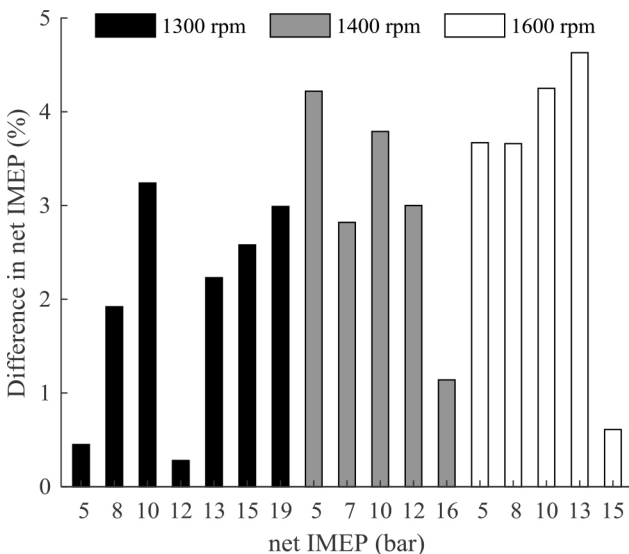


Figure 6. The absolute difference in net IMEP between experiments and simulations. $\frac{IMEP_{exp} - IMEP_{sim}}{IMEP_{exp}}$

SIMULATED ENGINE PERFORMANCE RESULTS

Engine simulations were conducted using the operating conditions in Table 4 as explained previously. It was not possible to achieve all of the operating conditions. At 1900 rpm and 19 bar gross IMEP, TC1 and TC2 failed to give the desired gross IMEP. Instead the load had to be decreased to 17.7 and 16.5 bar respectively. The simulated brake efficiencies of the engine of the different turbocharger configurations are presented in Figure 7 and Table A2.

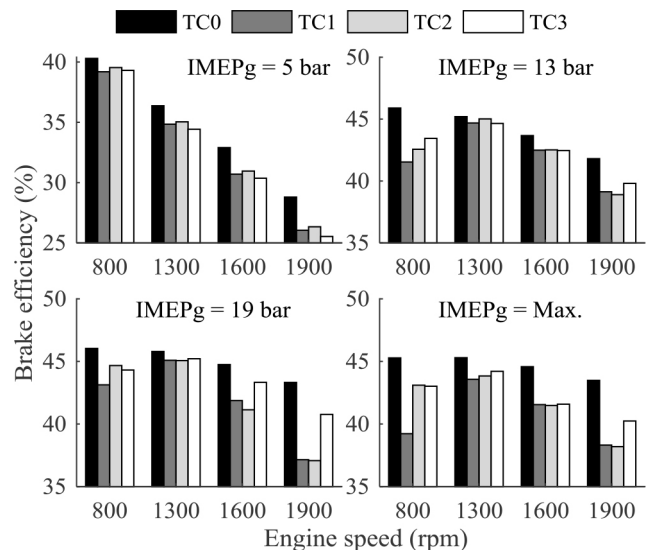


Figure 7. Engine brake efficiency of the complete speed-load range and comparing TC0, TC1, TC2 and TC3. Notice the difference scale on the y-axis. Specific values can be found in Table A2.

The peak brake efficiency was found at the same operating condition for all turbochargers, except TC0, and was located at an engine speed of 1300 rpm and 19 bar gross IMEP. TC0's peak brake efficiency occurred at the same load but at 800 rpm. The minimum efficiency was found at 1900 rpm and 5 bar. Comparing the different turbochargers the average brake efficiency was much higher for TC0 than for the rest (42.1, 39.3, 39.7, 40.2 for TC0, TC1, TC2, TC3) and demonstrated the importance of high overall turbocharger efficiency. TC1, TC2 and TC3 performed equally at the lower engine speeds and lower loads. When increasing the engine speed and load, however, TC3 started to achieve higher brake efficiencies than TC1 and TC2. This was a consequence of the extended range of a two-stage turbocharger.

Compressor Performance

Figure 8, 9, 10, 11 show the compressor pressure ratio as a function of reduced mass flow for TC0, TC1, TC2 and TC3 respectively. The figures are examples of so called compressor maps, where the contours display normalized isentropic efficiencies. Equations (9), (10), (11)