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System Optimization for a 2-Stroke Diesel Engine with a Turbo Super Configuration Supporting Fuel Economy Improvement of Next Generation Engines

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Abstract

The objective of this paper is to present the results of the GT Power calibration with engine test results of the air loop system technology down selection described in the SAE Paper No. 2012-01-0831.Two specific boosting systems were identified as the preferred path forward: (1) Super-turbo with two speed Roots type supercharger, (2) Super-turbo with centrifugal mechanical compressor and CVT transmission both downstream a Fixed Geometry Turbine. The initial performance validation of the boosting hardware in the gas stand and the calibration of the GT Power model developed is described. The calibration leverages data coming from the tests on 2 cylinder 2-stroke 0.73L diesel engine. The initial flow bench results suggested the need for a revision of the turbo matching due to the big gap in performance between predicted maps and real data. This activity was performed using Honeywell turbocharger solutions spacing from fixed geometry waste gate to variable nozzle turbo (VNT). New simulations results recommend VNT as it offers a higher potential to reduce BSFC with increase power and low end torque output than the original matching. For the high pressure stage the mechanical Roots type and the CVT superchargers have been assessed and the latter one has been identified having higher power adsorption than traditional positive displacement supercharger. This has allowed the supplier to work on an optimization of the units. Ultimately the VNT with CVT supercharger has been assessed on engine and it allowed confirming the validity and accuracy of the GT Power model after its calibration.

Introduction

The framework of this scientific research falls within the need to assess alternative combustion systems to the four-stroke single turbo Diesel engine with the intent to sustain the need to reduce below 95g/km the carbon emission by 2020. Two-stroke engine promises natural internal gas recirculation supporting PCCI-like, low-temperature combustion with limited emissions of soot and nitrogen oxides. In this context the activity has revealed the air system as a key element to enable achieving ultra low emission levels with an ultra downsized two-stroke diesel engine for mini and sub-mini vehicle segments.

Low exhaust gas temperature of two-stroke diesel and the need for both high scavenging ratio (i.e., large positive difference between boost pressure and turbine inlet pressure) and high boost pressure excluded the use of single turbocharger only. High boost pressure was needed due to limited quality of cross-scavenging with reversed tumble for applied 2+2 valve cylinder head . Instead of single turbocharger layout, the combinations of a supercharger – turbocharger group or electrically assisted turbocharger have to be used. The requirements on high turbocharger efficiency, which can reduce the power input to a supercharger, are clear. Since the torque of engine should ensure good drivability over a wide range of engine speed, boost pressure control is necessary. On the other hand, any turbocharger control reduces the resulting overall efficiency of the whole charging group – more but in reduced operation range in the case of waste-gating the turbine, less but in all the range of operation in the case of variable nozzle turbine. Unlike in the case four-stroke engine, the driving power of a supercharger cannot be recuperated back to engine crankshaft, since there is no pumping loop in the indicator diagram and all pressure difference between inlet and exhaust manifold is devoted to overcoming pressure losses in the pipe system.

Due to those reasons, the careful optimization of the air loop has to be done for real conditions at an engine, including (1) all pressure losses (all pipe splits or joints, inlet filter, intercoolers, EGR pipe splits or venturies – if used, DPF upstream or downstream a turbine, exhaust muffler, etc.) and (2) unsteadiness – pressure waves influence both gas exchange and turbine/compressor performance. Unlike in the four-stroke engine case, operation conditions of all components are closely interconnected. Scavenging, e.g., needed for reasonable fresh gas purity in a cylinder, dilutes burnt gas transferred to a turbine and reduces significantly its temperature, i.e., enthalpy usable at a turbine. Simulation may be used for an early stage of design, if appropriately calibrated by experimental data, of all pressure boosting devices together with intercoolers, aftertreatment equipment and pipe elements, supported by 3-D CFD modelling.

The workflow of the whole air-loop optimization has consisted in

* experimental, steady-flow tests of all potentially useable turbochargers, superchargers and intercoolers
* prediction of pressure losses of other devices in air-loop using CFD steady flow simulations
* building and calibrating 1–D (GT Power) model of the whole system (combustion model was based on single cylinder research engine tests)
* optimization of different air-loop system lay-outs by GT Power and the selection of the best solution
* designing and manufacturing the selected system
* Tests of the engine, validation of results and feed-back to the simulation tools used for detailed optimization.

The current paper describes the main results achieved during system optimization and the features of VNT turbocharger compared to a WG high-efficiency turbine.

Summary of Previous Work

Main Issues of Two-Stroke Engine Pressure Charging

Scavenging of two stroke engine requires the positive pressure difference between inlet and exhaust systems. The excess of scavenging air – the greater, the poorer scavenging efficiency is – dilutes burnt gas, decreasing its temperature. The inlet-exhaust pressure difference has to insure the necessary flow-rate during scavenging, depending significantly on available flow area of engine valves during scavenging period.

The fast assessment of the system may be done using algebraic model of steady flow through engine with overlapped valve opening, turbocharger turbine-compressor power balance and exhaust gas temperature calculated from energy balance of a reciprocating two-stroke engine. All parameters describing gas exchange quality have to be estimated from experience or found by more deep 1-D or 3-D simulations combined with specific scavenging experiments.

The SAE definition of gas exchange parameters – [1] has been used, namely delivery ratio ***λd***(inlet port mass flow rate /perfect-scavenging trapped mass flow rate based on engine displacement volume ***Vs***, speed ***nE*** and density in inlet manifold at pressure ***pim*** and temperature ***Tim***), charging ratio ***λch*** (trapped mass/perfect-scavenging trapped mass) and scavenging efficiency ***ηscav*** (fresh charge mass reduced to overall oxygen contents including rest gas/trapped mass). Moreover, turbocharger and supercharger efficiencies (both isentropic + mechanical) ***ηTC***, ***ηSC*** are taken into account together with engine indicated efficiency  and relative amount of heat transferred to walls by cooling ***Kcool***. Pressure losses in all connecting pipes, intercoolers and exhaust gas aftertreatment devices are respected.

The pressure difference between inlet ***pim*** and exhaust ***pex*** needed for pre-defined scavenging is calculated from averaged reduced flow area  of engine valves during scavenging period (reversely proportional to the engine flow resistance) and required delivery ratio 

 (1)

Power balance of a turbocharger with a low-pressure compressor outlet pressure ***pC2***, used as supercharger inlet pressure, atmospheric pressure ***pa*** and appropriate pressure differences between machines, with constant pressure thermal capacities ***cp*** and appropriate exponents connecting pressure and temperature ratios during isentropic change with exponent ***e*** yields

 (2)

Finally, energy balance of an engine with fuel mass-flow rate and lower calorific value ***Hu***determined at reference temperature***Tref*** , with the share of cooling losses ***Kcool***and indicated efficiency  yields for exhaust gas temperature

 (3)

Fuel mass flow rate can be linked to air mass-flow rate by air excess ***λ***, engine charging efficiency  and stoichiometric air-to-fuel ratio ***Lt***.

Combining those relations, the dependence of achievable turbocharger compressor pressure ratio on inlet manifold pressure  with iterated ratios of pressure losses (which do not vary too much during iteration) and, as well, iterated temperature in inlet manifold (dependent on intercooler efficiency) can be found from the equation



(4)

Especially in the case of poppet valve gear, the limited accelerations during valve motion limit the averaged area during scavenging. The power of a turbocharger turbine, reduced by low exhaust gas temperature, has to be supported by other energy source, mostly a supercharger coupled in series with a turbocharger compressor and covering the pressure difference between ***pC2*** and ***pim***. Unlike the pumping loop of a four-stroke engine the work needed for reaching the pressure difference cannot be recuperated back to engine crankshaft but the supercharger power input has to be added to friction losses, changing thus gross brake power to the final net brake power. The described original procedure makes fast assessment of two-stroke brake efficiency possible while the main parameters of engine are changed in rated mode of operation.

The influence of rated air excess and turbocharger efficiency while other parameters were fixed to values found from experiments and simulations of an extremely downsized two-cylinder diesel engine under development is presented in Figure 1 and Figure 2. The net brake power and net brake fuel consumption is calculated after supercharger power input (dependent on scavenging mass flow-rate, supercharger pressure ratio , supercharger inlet temperature after intercooling the air from a turbocharger compressor and supercharger isentropic efficiency) is subtracted from the engine gross brake power, dependent on assumed indicated efficiency and gross mechanical efficiency.

The former one shows the influence of a boost pressure on a single cylinder power at fixed trapped air excess (denoted as *lam* in Figure 1 and Figure 1) and turbocharger overall efficiency for delivery ratio, charging ratio and scavenging efficiency typical for a tested engine with reversed tumble scavenging. In the current case the charging ratio was approx. 70% and scavenging efficiency approx. 60%, ensuring the charging efficiency close to 40% at a reasonable level of delivery ratio. While indicated and gross brake power at engine speed of 1500 rpm are turbocharger efficiency independent, the net brake power and net brake efficiency strongly depend on turbocharger efficiency, since the missing pressure level needed for scavenging has to be covered by an engine driven supercharger. Moreover, the mixture strength (air excess), which is decisive for pollution level, should be compromised. If set too high, it calls for high inlet manifold pressure for reaching rated power, reducing simultaneously the exhaust gas temperature and decreasing the achievable power of turbine, which is reflected by too low supercharger inlet pressure. Then the difference between gross and net engine parameters is increased.

Figure 1: Single cylinder power (indicated, gross brake and net brake - including a supercharger drive), share of supercharger power input on engine net brake power and net brake efficiency in dependence on required boost pressure for different levels of air excess and turbocharger overall efficiency



The share of supercharger power of the engine total shows clearly that there might be no need for a supercharger in some range of power if the efficiency of a turbocharger is high enough and if the excess air is fixed at reasonable mixture strength level (1.6 for the current case). On the other side, the need of high supercharger power input increases if air excess is too high. For this engine operation point the computed exhaust temperature was 480degC.

Figure 2: Required averaged pressures at a supercharger and in exhaust manifold and net brake specific fuel consumption in dependence on parameters from the Figure 1



Those results are confirmed by the Figure 2 in which the pressures close to a supercharger are drawn for two levels of air excess (causing different engine power at the same inlet manifold pressure level, of course). The bsfc is strongly influenced by the air excess chosen and turbocharger efficiency available. The combustion quality and indicated specific fuel consumption was not influenced by air excess too much in this range of mixture richness.

Air System Concept Analysis

Initial step for the air system definition has been to analyse similar existing solutions. Two architectures have been identified with the desired characteristics of this study. A 2-stroke engine prototype from AVL (1.0l, 3-cylinder) and a Daihatsu (1.2l, 2-cylinder). Both adopt a serial sequential boosting architecture with a mechanical supercharger and turbocharger. The AVL engine lay out considers the mechanical supercharger arranged downstream of the turbocharger while the Daihatsu adopted a lay out with the mechanical supercharger upstream of the turbocharger [2]. What are the main differences we should expect from those two different solutions? The installation of the mechanical supercharger in the high pressure (HP) stage enables selecting a smaller supercharger than the one required in the low pressure (LP) stage. This facilitates the packaging and reduces the overall engine weight and also allows working at lower supercharger speeds. One drawback we need to consider when working with the supercharger in the HP stage is that it will be working at higher compressor inlet temperatures and this will limit the operating range of the mechanical supercharger and have a higher power demand from the engine crankshaft.

The down selection of the concepts was done keeping all of those aspects in consideration trying to achieve the project performance targets. To facilitate the down selection a concept tree was created as described **Table 4**.

Table 4: Overview of the investigated air system concepts



Initial Boosting System Down Selection

Table 5: Assessed air system configurations



The concept tree analysis led to discard the single stage boosting as a viable option due the compression ratio needs and the scavenging characteristic of the 2 stroke engine architecture. Ultimately an initial set of 8 options was identified for the initial assessment in GT power (see **Table 5**)

All concepts have a two stage boosting layout with a turbocharger either in the low pressure (LP) or high pressure (HP) stage arranged in a “serial sequential” lay out, with mechanically driven positive displacement or centrifugal supercharger configurations with variable, single or dual supercharger or continuous variable speed, (C1 to C4bis), e-booster, e-turbo (C6 – C8)

During the partial load operation it was assessed the possibility to have Low pressure (LP-egr) and mid (MP) pressure EGR loops (blended EGR mode).

With reference to the nomenclature in **Table 5**, in the below **Figure 4**, the more conventional lay outs are sketched. Supercharger in LP stage C2 (left) and supercharger in HP stage C3 (right) both with middle pressure (MP) EGR loop where 1-turbocharger with bypass, 2-LP intercooler, 3-supercharger by-pass, 4-supercharger, 5- supercharger transmission, 6-HP intercooler, 7-engine, 8-EGR loop with EGR valve and intercooler.



Figure 4: Comparison of LP vs. HP supercharger configuration

The first step for the concept down selection has been to simulate the fuel consumption of the 8 concepts in the NEDC partial load points summarized in **Table 3**. The results are plotted **Figure 5**, where the x-axis is the time weighted fuel consumption and the z-axis is the technical feasibility index. The feasibility index reflects the parts availability for hardware testing, packaging constrains and performance achievable potential.



Figure 5: Fuel Consumption comparison of the 8 simulated configurations

Table 6: Best options summary



The results allow making few general considerations regarding the difference in Fuel Consumption between the HP and LP layout of mechanical supercharger. The supercharger downstream of the turbocharger (HP) enables lower fuel consumption than the LP lay out. LP layout in fact requires the supercharger to spin faster to provide the requested boost as a consequence of the presence of the low pressure loop EGR that increases the compressor inlet temperature. The two speed drive of the supercharger is preferred over the one speed gear box. On the other hand the variable supercharger drive does not allow any significant fuel benefit in partial load mainly due to the higher mechanical losses.

Based on this consideration and on the data in **Figure 5** it has been possible to identify the configuration **C3-d,** (waste gate fixed geometry turbocharger coupled with positive displacement supercharger downstream and with two speed control, (**Figure 6**), as the one with highest potential to reach the project objectives in terms of power and fuel consumption reduction. Another interesting option is the **C4bis-c** with centrifugal charger set downstream of the waste gate turbocharger and the variable supercharger drive (CVT). This one is considered the backup solution by the authors as the hardware is in a less mature stage than the 2 speed supercharger option.

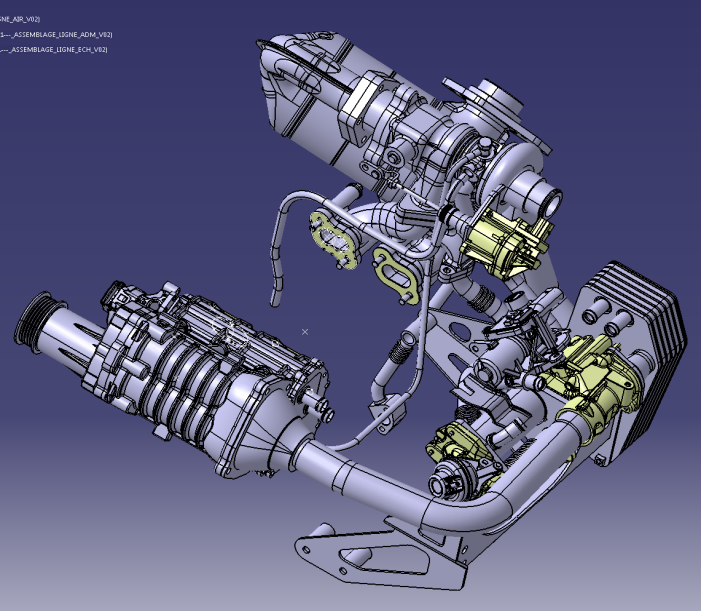


Figure 6 Lay out of the preferred air loop system C3-d. Supercharger downstream of the waste gate turbocharger

Hardware Assessment

Roots – Type Supercharger

Roots supercharger is a positive displacement supercharger with compression ratio depending on pressure in the outlet piping. It uses a shock wave compression to compress the ingested air.

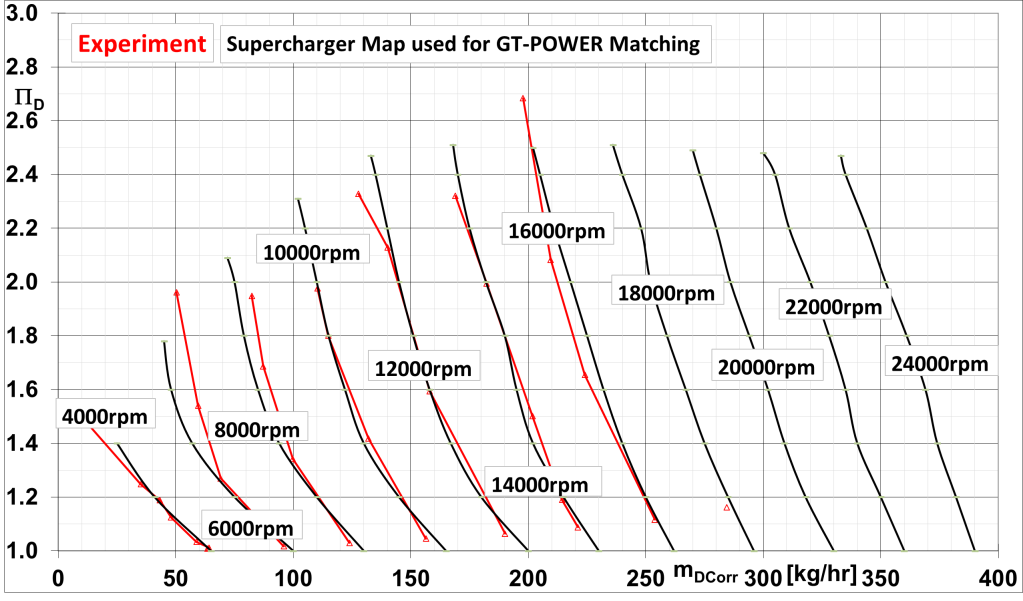


Figure 16: Pressure ratio vs. mass flow rate characteristic of Roots type supercharger

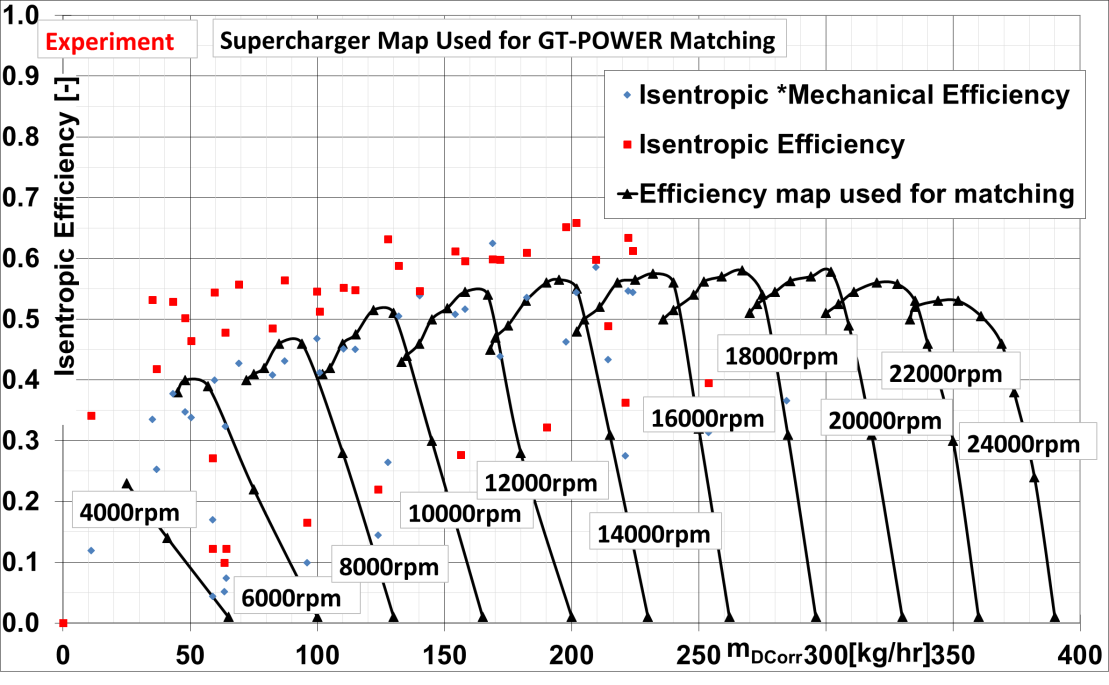


Figure 17: Isentropic efficiency vs. mass flow rate characteristic of Roots type supercharger

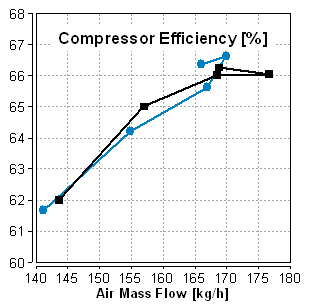
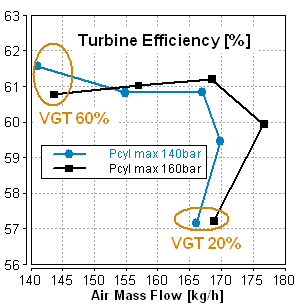


Figure29: Efficiency of Honeywell turbine and compressor at 1500rpm Full Load

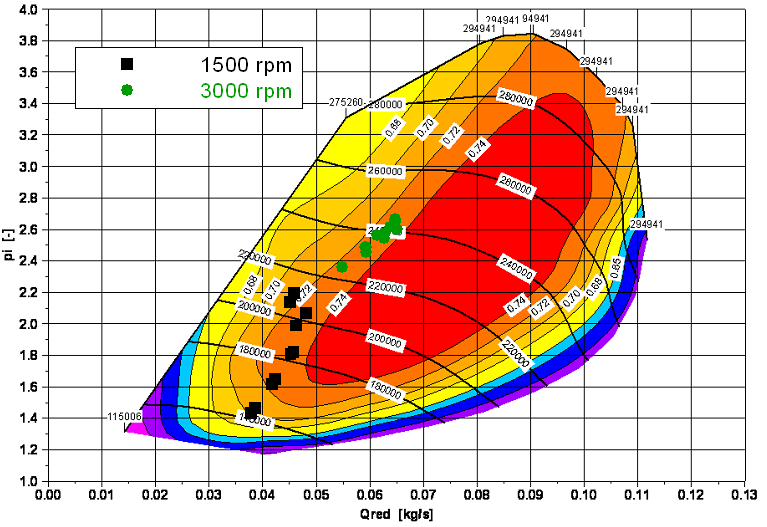


Figure 30: Honeywell compressor efficiency map

Summary/Conclusions

In this paper we have reviewed the results of a GT Power engine model calibration leveraging experimental data from single cylinder combustion test rig, real 2 cylinder engine measurements and turbocharger gas stand performances. The newly calibrated model was used to refine the matching reconsidering the Variable Nozzle Turbine option for the low pressure stage of the boosting system. The simulation results showed that the VNT option is the best solution to deliver the requested performance when coupled either with a Roots supercharger or with the CVT-supercharger.

Additionally an analysis on gas stand of the two possible supercharging solutions has allowed identifying the limits of these machines for this kind of applications. In the case of the Roots supercharger there is the limit in range due to outlet compressor temperatures above certain compression ratios, and in the case of the CVT-supercharger the high power absorption from the engine.

Ultimately a final assessment on fuel consumption and CO2 reduction potential is now planned for the middle of 2014 given the quite good performances of the powertain system on the test bench.

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Definitions/Abbreviations

|  |  |
| --- | --- |
| 1 | inlet |
| 2 | outlet |
| C | compressor |
| CVT | continuously variable transmission |
| DPF | diesel particulate filter |
| EGR | exhaust gas recirculation |
| FC | frequency convertor |
| IGR | internal gas recirculation |
| M | electric motor |
| MBV50 | mean burn value |
| PCCI | premixed charge compression ignition |
| T | turbine |
| VGT | variable geometry turbocharger |
| VNT | variable nozzle turbocharger |
| VVT | variable valve train |
| BMEP [bar] | break mean effective pressure |
| BSFC [g/kW/h] | break specific fuel consumption |
| *cp* [J/kg/K] | thermal capacity |
| *e* [-] | exponents of isentropic change |
| FMEP [bar] | friction mean effective pressure |
| FSN [-] | filter smoke number |
| Hu [MJ/kg] | lower calorific value |
| IMEP [bar] | indicated mean effective pressure |
| Kcool [-] | cooling loss coefficient |
| Lt [-] | stoichiometric air-to-fuel ratio |
| [kg/s] | fuel mass-flow rate |
| [kg/s] | inlet port mass flow rate |
| nE [rpm] | engine speed |
| pa  [kPa] | atmospheric pressure |
| pC2 [kPa] | compressor outlet pressure |
| pex [kPa] | engine exhaust pressure |
| pim [kPa] | inlet manifold pressure |
| Tim [K] | inlet manifold temperature |
| Tref [K] | reference temperature |
| Vs [m3] | engine displacement volume |
| *λ* [-] | trapped air excess |
| *λch* [-] | charging ratio |
| *λd*[-] | delivery ratio |
| [m2] | averaged engine reduced flow area |
| *ηb* [-] | engine brake efficiency |
| *ηi* [-] | engine indicated efficiency |
| *ηm* [-] | engine mechanical efficiency |
| *ηSC* [-] | total supercharger efficiency |
| *ηscav* [-] | scavenging efficiency |
| *ηTC* [-] | total turbocharger efficiency |