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# Estimation of the possibility of using a simple thermodynamic model to evaluate the benefits of modifying a real spark ignition engine

**Abstract**

Improving engine performance is the general goal of all designers of new cars. Nevertheless, there is and will continue to be a significant number of older cars on the market, where modifications can improve their performance in terms of fuel consumption and emissions, but also in terms of driving properties, giving the user fuel economy and greater driving pleasure. However, before starting the modification, it is worth knowing computationally what final effects we can expect. The paper shows that it can be estimated on the basis of simple thermodynamic calculations based on the basic Otto cycle, where the data for the calculations are very limited: they require geometric dimensions as well as pressure and temperature values provided by the selected compressor. The thermodynamic calculations of the cycle in two versions were shown, and the results were verified on a real engine while driving and related to the results of road tests and the chassis dynamometer.

**Key words**: SI engine, turbo-compressor, model, experimental verification.

# Estymacja możliwości zastosowania prostego modelu termodynamicznego do oceny korzyści wynikających z modyfikacji rzeczywistego silnika z zapłonem iskrowym

**Streszczenie**

Poprawa parametrów pracy silników jest generalnym celem wszystkich konstruktorów nowych samochodów. Niemniej jednak na rynku istnieje i długo jeszcze będzie istniała znacząca ilość samochodów starszych, gdzie modyfikacje mogą poprawić ich parametry pracy pod względem zużycia paliwa i emisji, ale także własności jezdnych, dając użytkownikowi oszczędność paliwa i większą przyjemność z jazdy. Przed przystąpieniem do modyfikacji warto jednak wiedzieć obliczeniowo, jakich efektów finalnych możemy się spodziewać. W pracy pokazano, że można to estymować na podstawie prostych obliczeń termodynamicznych, opartych na podstawowym obiegu Otto, gdzie dane do obliczeń są bardzo ograniczone: wymagają wymiarów geometrycznych oraz wielkości ciśnienia i temperatury podawanych przez dobraną sprężarkę. Pokazane zostały obliczenia termodynamiczne obiegu w dwóch wersjach, a wyniki zostały sprawdzone na rzeczywistym silniku w czasie jazdy oraz odniesione do wyników badań drogowych i na hamowni podwoziowej.

**Słowa kluczowe**: silnik ZI, turbosprężarka, model, weryfikacja eksperymentalna.

## 1. Introduction

The majority of modern scientific papers on the subject of internal combustion engines, such as works of Zu, Yang, Wang and Wang (2019) as well as Mahamoudi, Khazaee and Ghazaikhani (2017), concerns compression ignition engines and is primarily concerned with the issue of reducing emission of substances harmful to atmosphere. All modifications to engine construction and rigging are introduced with the goal of meeting the increasingly rigorous exhaust gases emission standards. The number of studies concerning modeling engines is increasing, such as works of Eriksson, Nielsen, Brugard, Bergstorm and Pettersson (2002). The studies on the influence of exhaust gases' recirculation on the exhaust gases' temperature and fuel consumption are also being conducted by Luján, Climent, Novella and Rivas-Perea (2015). Further studies performed by Wang, Liu and Reitz (2017) are focused on occurrence of engine knocking/pinking. The studies such as those performed by Sendyk, Filipczyk (2008) as well as Rinaldini, Breda, Fontanesi and Savioli (2015), focusing on increasing and improving engine's working parameters, are sparse. However, there is a significant number of vehicles on the market the rolling characteristics of which could be improved by applying better fuel feed systems and incorporating turbocharger compressors into naturally aspirated engines. Such treatment improves flexibility of an engine and its most crucial working parameters. In essence, there are no items in the subject literature which would enable a possible user to assess the results of modernizing a particular engine a priori, on the grounds of a simple model, whereas workshops realizing such modifications act on the basis of own experience. Theoretical parameters of an engine are dependent on the progress of the Otto cycle on the grounds of which we may ascertain efficiency of converting heat energy within the fuel into mechanical work. During conversion of heat energy into mechanical energy a part of this energy is being lost in accordance with the second law of thermodynamics. One of the methods of increasing power of an engine and its actual efficiency is increasing pressure of the injected fuel blend, i.e. charging. Therefore the first attempts to charge piston engines were made as early as in years 1885-1914. The goal of these attempts was to recover the energy from exhaust gases and convert it into the circular motion of the turbocharger compressor which would, in turn, compress the air fed into the engine. Increasing pressure of the initial compression positively influences average effective pressure which, in turn, contributes to increasing power and torque of an engine. The improvement of working parameters of engines for the purposes of motor-sports has been discussed by Bell (1997) and Bell (2003). In this paper the selected elements of the methodology presented therein, in particular the selection of components such as intercooler and type of exhaust manifold, are being used.

The innovative goal of these studies is to demonstrate that the influence of a turbocharger compressor on increase in power and torque as well as possibly other engine parameters can be predicted through calculations and use of simple thermodynamic models. Such calculations and models present a basis for possible desirability of modernizing an SI engine through incorporating a turbocharger compressor. The theoretical calculations have been verified by the studies performed on an engine designated M102.982 with a cubic capacity of 2,299 $cm^3$ and power of 100 kW originating from a Mercedes-Benz W124 car.

## 2. A theoretical calculation model of an SI engine

A theoretical Otto cycle consists of two isochores and two isentropes. Valve flow resistance is being disregarded and thus suction and exhaust conversion are omitted in calculations. Calculating a theoretical combustion temperature is a major goal. In the simplest model combustion temperature is a direct result of energy balance of burning a single unit of fuel within a piston, without taking into consideration the process of high-temperature $CO_2$ dissociation. In reality such combustion is impossible, the achieved temperatures are lower because some of the energy is being absorbed through dissociation. Therefore the calculations presented below represent two approaches: with and without dissociation. Actual values for the modeled engine utilized during calculations are presented in Table 1.

Table 1

Parameters for Otto cycle theoretical model's calculations

| | Symbol | Value | Unit | Description |
|---|---|---|---|---|
| Aspirated engine | $t_{air}$ | 298 | K | Ambient air temperature |
| | $W_u$ | 43 000 | kJ/kg | Caloric value of fuel |
| | $p_1$<br>$m_{air}$<br>$m_{fuel}$ | 100 000<br>0.00076<br>0.000051 | Pa<br>kg<br>kg | Inlet manifold pressure<br>Air mass<br>Fuel mass |
| | A | 0.0072 | $m^2$ | Piston surface |
| | $V_s$ | 0.000574 | $m^3$ | Piston displacement |
| | ks | 1.3996 | - | Isentropic compression exponent |
| | kr | 1.374 | - | Isentropic expansion exponent |
| | $V_k$ | 0.000072 | $m^3$ | Volume of combustion chamber |
| | R | 287.04 | J/kgK | Gas constant |
| | λ | 1 | - | Air-fuel ratio |
| | $Z_t$ | 14.7 | kg/kg | Theoretical fuel combustion air requirement |
| | $cs_{fuel}$ | 0.775 | kJ/kgK | Specific heat of exhaust gases |
| Pressure charged engine | $c_{air}$<br>$p_1$<br>$T_1$<br>$m_{air}$<br>$m_{fuel}$ | 0.716<br>180 000<br>301<br>0.0012<br>0.000082 | kJ/kgK<br>Pa<br>K<br>kg<br>kg | Specific heat of air<br>Inlet manifold pressure<br>Intake air temperature<br>Air mass<br>Fuel mass |

The ideal gas model with the isentrope index values presented in Table 1 has been utilized in calculating the cycle. The calculation formula for the cycle which does not take into consideration thermal dissociation has been presented below. For calculating the model which takes into consideration dissociation the methodology presented in works of Szewczyk and Wojciechowski (2207) has been applied.

Theoretical air mass in the cylinder:

$$\boldsymbol{m_{pow}} = \frac{\boldsymbol{P_1 \cdot V_s}}{\boldsymbol{T_1 \cdot R}}\ [\boldsymbol{kg}] \tag{1}$$

Stoichiometric mass of fuel required for complete combustion of $m_{pow}$:

$$\boldsymbol{m_{pal}} = \frac{\boldsymbol{m_{pow}}}{(\boldsymbol{\lambda \cdot Z_t})}\ [\boldsymbol{kg}] \tag{2}$$

The temperature of compressed blend is:

$$\boldsymbol{T_2 = T_1 \cdot \varepsilon^{ks-1}[K]} \tag{3}$$

Combustion temperature:

$$\boldsymbol{T_3 = T_2 + \frac{W_u \cdot m_{pal}}{cs_{pal} \cdot (m_{pal} + m_{pow}) + c_{pow} \cdot m_{pow} \cdot (\lambda - 1)}\ [K]} \tag{4}$$

Temperature of expanded exhaust gases:

$$\boldsymbol{T_4 = \frac{T_3}{\varepsilon^{kr-1}}[K]} \tag{5}$$

Compression pressure:

$$\boldsymbol{P_2 = P_1 \cdot \varepsilon^{ks}[Pa]} \tag{6}$$

Combustion pressure:

$$\boldsymbol{P_3 = \frac{T_3}{T_2} \cdot P_2[Pa]} \tag{7}$$

Exhaust gases pressure:

$$\boldsymbol{P_4 = \frac{P_3}{\varepsilon^{kr}}\,[Pa]} \tag{8}$$

Final outlet exhaust gases temperature:

$$\boldsymbol{T_5 = T_4 \cdot \left(\frac{P_1}{P_4}\right)^{\frac{kr-1}{kr}}\,[K]} \tag{9}$$

Theoretical efficiency:

$$\boldsymbol{\eta_t = 1 - \frac{1}{\varepsilon^{ks-1}}} \tag{10}$$

Isochoric degree of increase in pressure:

$$\boldsymbol{\alpha_v = \frac{P_3}{P_2}} \tag{11}$$

Average theoretical pressure:

$$\boldsymbol{P_t = P_1 \cdot \frac{\varepsilon^{ks} \cdot \eta_t \cdot (\eta_t - 1)}{(ks-1) \cdot (\varepsilon - 1)}\,[Pa]} \tag{12}$$

The formula for theoretical power of an engine according to Otto cycle is:

$$\boldsymbol{N_t = m_c \cdot T_1 \cdot c_v \cdot \frac{(\varepsilon^{ks-1}) \cdot (\alpha - 1)}{t_{obiegu}}} \tag{13}$$

where: $m_c$ – gross weight of the blend

$\alpha$ – degree of isochoric increase of pressure = $\frac{P_3}{P_2}$

$t_{obiegu}$ – duration of the cycle consisting of two revolutions of a crankshaft

$$\boldsymbol{t_{obiegu} = \frac{2}{n_s}} \tag{14}$$

where: $n_s$ – rotational speed, rpm$s^{-1}$

The results of all calculations necessary for modeling theoretical Otto cycles are contained in Table 2.

Table 2

Results of theoretical calculations

| Symbol | A – a naturally aspirated engine w/o dissociation | B – a turbocharged engine w/o dissociation | C – a naturally aspirated engine with dissociation | D – a turbocharged engine with dissociation | Unit |
|---|---|---|---|---|---|
| $T_1$ | 298 | 301 | 298 | 301 | K |
| $T_2$ | 717 | 724 | 717 | 724 | K |
| $T_3$ | 4 251 | 4 258 | 2 814 | 2910 | K |
| $T_4$ | 1 869 | 1 872 | 1 237 | 1 280 | K |
| $T_5$ | 1 134 | 1 138 | 840 | 863 | K |
| $P_1$ | 100 000 | 180 000 | 100 000 | 180 000 | Pa |
| $P_2$ | 2 165 498 | 3 897 897 | 2 165 498 | 3 897 897 | Pa |
| $P_3$ | 12 828 666 | 22 918 120 | 8 499 316 | 15 663 057 | Pa |
| $P_4$ | 627 178 | 1 119 566 | 415 198 | 765 151 | Pa |
| $\eta_t$ | 0,584 | 0.584 | 0.584 | 0.584 | - |
| $\alpha_v$ | 5.929 | 5.88 | 3.925 | 4.02 | - |
| $P_t$ | 1 951 109 | 3 476 993 | 1 157 854 | 2 150 731 | Pa |
| $Q_d$ | 2.21 | 3.937 | 2.21 | 3.9 | kJ |
| $Q_o$ | 0.982 | 1.75 | 0.59 | 1.1 | kJ |

Figure 1 presents graphs for the theoretical cycles of an aspirated engine and a turbocharged engine for a model with and without dissociation. Increase in suction pressure during charging and significant increase in compression pressure are visible, albeit such pressure can be realistically achieved for the model which takes into consideration the phenomenon of dissociation.

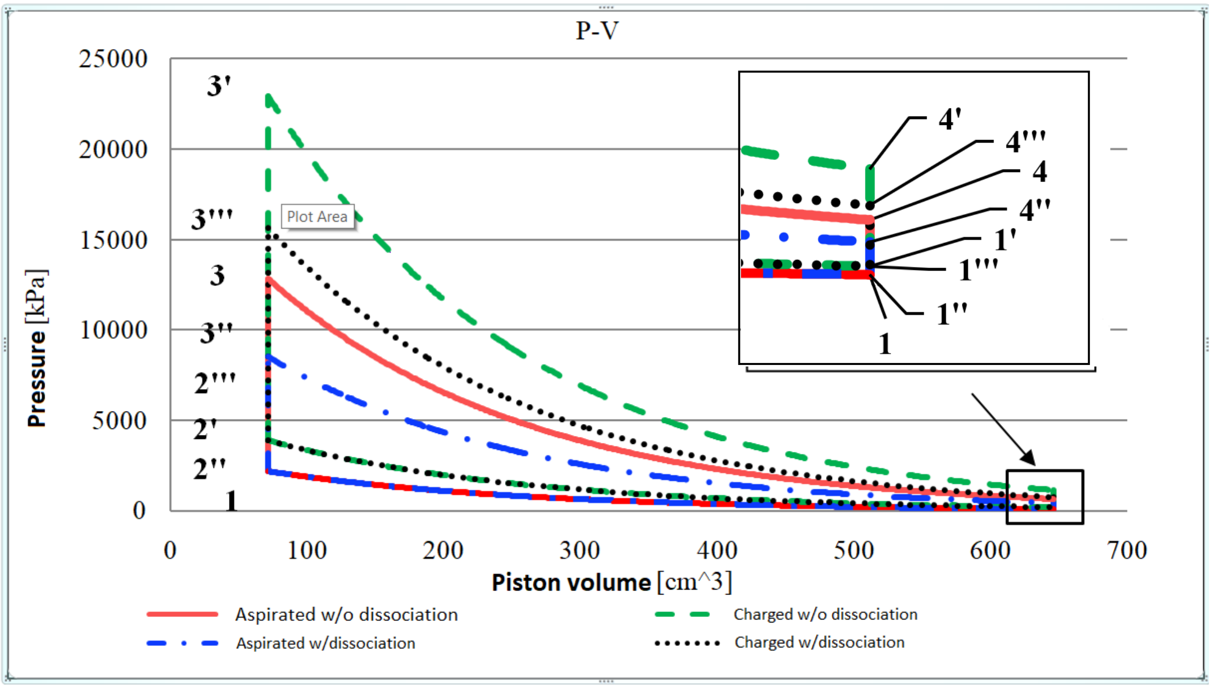


Figure 1. Theoretical p-V Otto cycles for naturally aspirated and turbocharged engines with and without dissociation

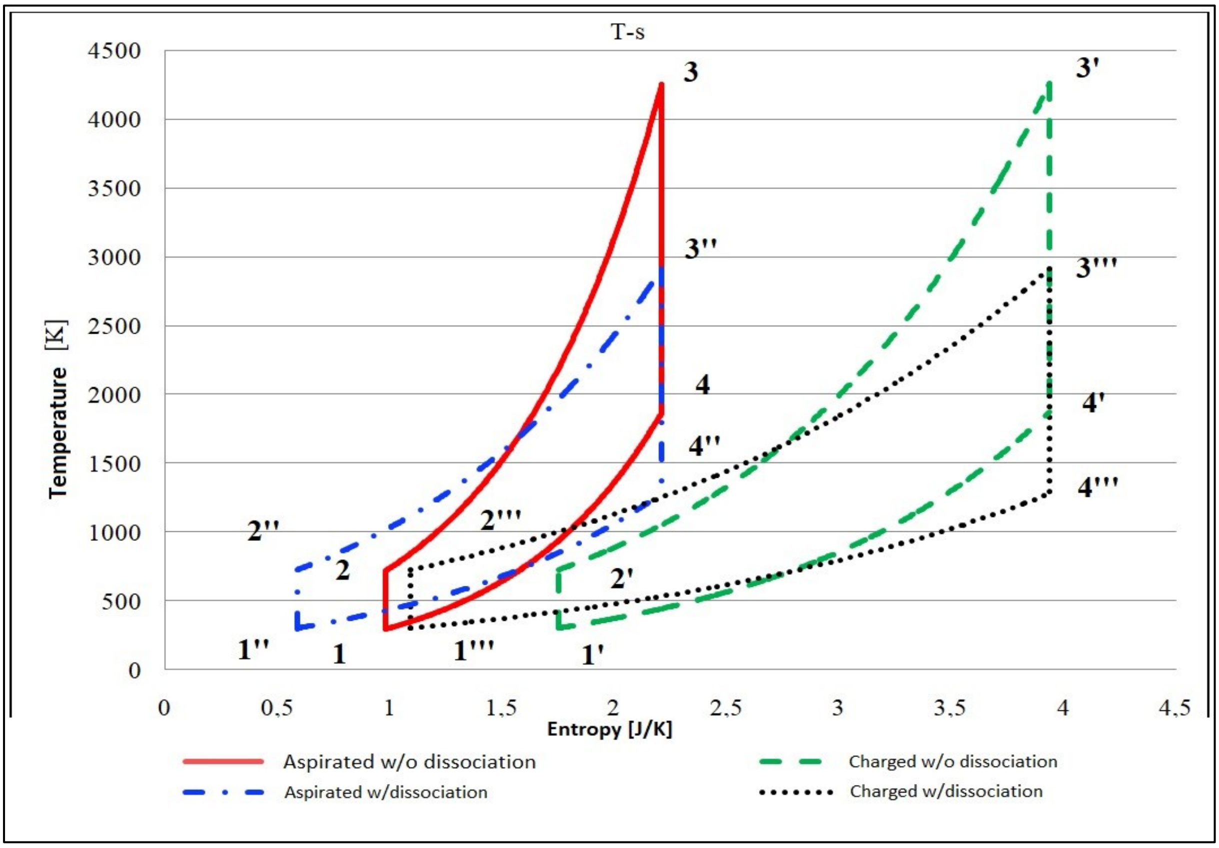


Figure 2. Theoretical T-s Otto cycles for naturally aspirated and turbocharged engines with and without dissociation

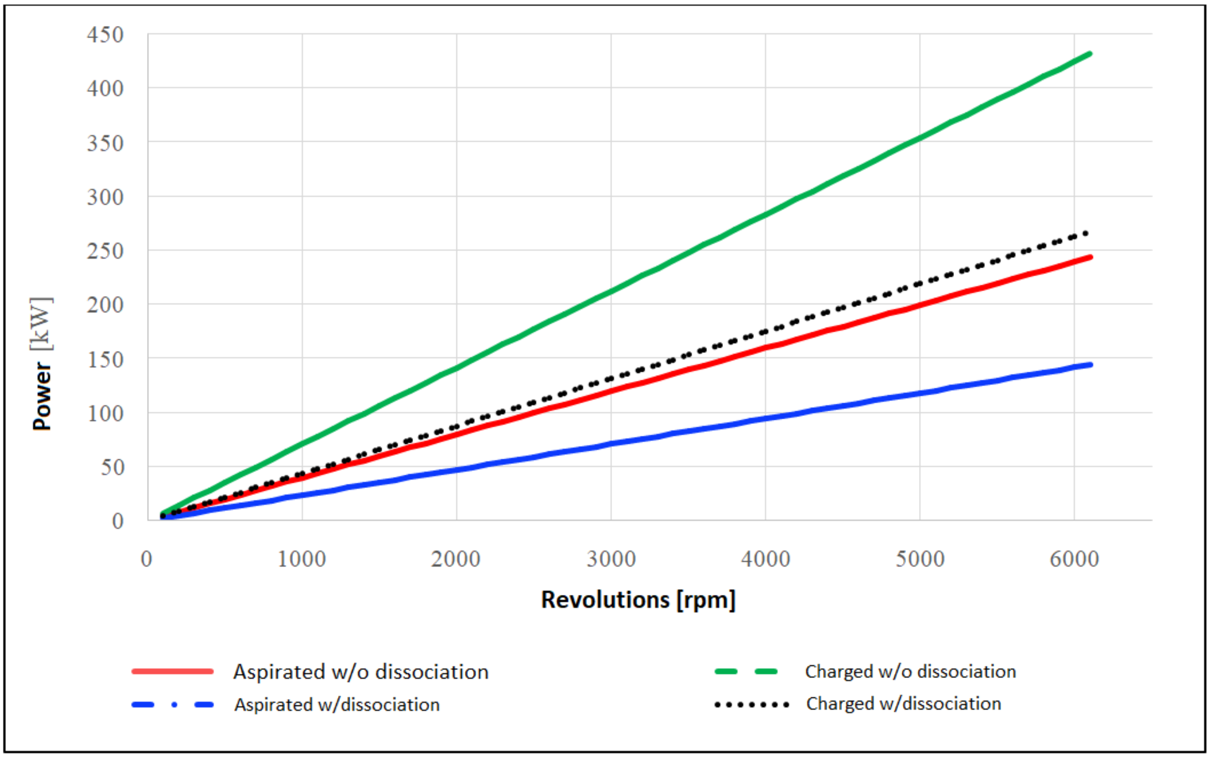


Figure 3. Comparison of theoretical power of naturally aspirated and turbocharged engine with and without dissociation

Figure 2 presents the theoretical T-s graphs. The differences in temperature for models with and without dissociation are visible; the temperatures in case of the simpler model without dissociation are unrealistically high. As presented in Figure 3 the difference in theoretical power is significant when calculated through standard formulas and when taking into consideration the $CO_2$ dissociation. In case of theoretical calculations the power of a turbocharged engine calculated in accordance with the standard formulas not accounting for dissociation is higher by 188.5 kW than the power of a naturally aspirated engine which translates into a 77% increase in power. The comparison of the maximum theoretical power of a naturally aspirated engine with the power of a turbocharged engine which accounts for dissociation displays that the power increased by 123.3 kW, an 86% increase in power.

## 3. Experimental studies

As indicated previously the engine of a Mercedes-Benz W124 car designated M102.982 with a cubic capacity of 2,229 $cm^3$ has been modified. The engine has been inspected prior to modifications and later, after several modifications have been made with the goal of increasing efficiency and pressure of suction. Figure 3 presents characteristics of the TD04H – 13T turbocharger compressor with a high efficiency reaching (73-76%) under the assumed charging pressure of 80 kP selected for the studied engine.

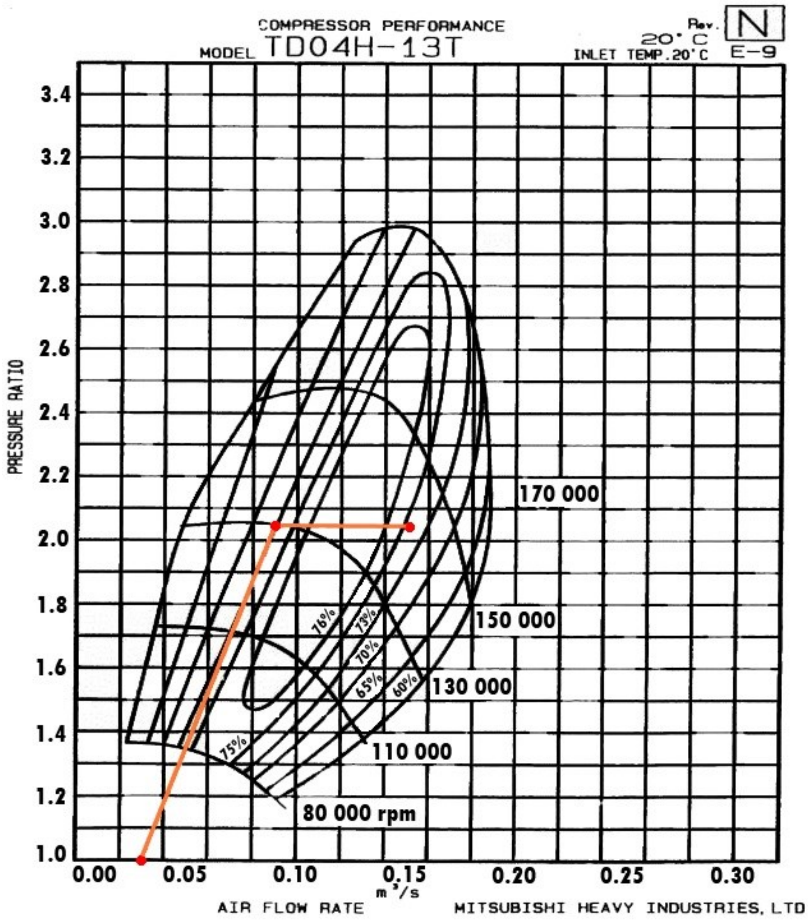


Figure 4. Determining operating range on the turbocharger compressor map

(source: NASIOC, 2006)

The Ke-Jetronic mechanical injection system has been replaced with an electronically controlled multi-point indirect fuel injection system (EFI). The exhaust manifold has been replaced to compensate for exhaust drag of individual pistons. The air intake system was also modified through replacing air filter with a cone filter in order to reduce flow drag during intake. An appropriate radial centrifugal compressor and Intercooler have been selected.


Figure 5. View of the engine with modifications and location of measuring transducers indicated A – turbocharger, B – cooling liquid temperature transducer, C – intake pressure transducer, D – intake air temperature transducer, E – air bleed valve, F – air cooler (Intercooler), G – cone filter

A device for indirect engine control (Digital Ecu Tuner 3) operating in "Fuel Implant" mode has been used for testing. This control unit receives signals from measuring transducers indicated in Figure 5 and then controls opening of the injectors on the basis of the fuel maps generated during engine tuning. The engine was tuned by use of Innovative LM-2 broadband probe controller. The pressure in the manifold was recorded by MAP sensor MPX4250AP with operating range from 20 to 250 kP. The rotational speed was read by use of external inductive sensor and the crank pattern had been mounted on the pulley wheel of the arbour. Original measuring transducers of the engine were used to record cooling liquid temperature and air temperature.

The process of measuring engine power was the same for both the aspirated an turbocharged engine and consisted of driving several times on a straight, dry road without elevations. The ride proceeded with the third gear locked from approximately 1600 rpm to electronic fuel shut-off which occurs when approximately 6500 rpm are reached. Figure 6 presents an example of a single measurement – the measurement includes four parameters read from sensors during measuring. Subsequently power and torque of a wheeled vehicle are calculated on the basis of engine rotational speed/acceleration time graph after inputting basic data such as: mass of the vehicle, rolling resistance coefficient, front surface of the vehicle, revolutions-speed ratio, as well as wheel parameters.

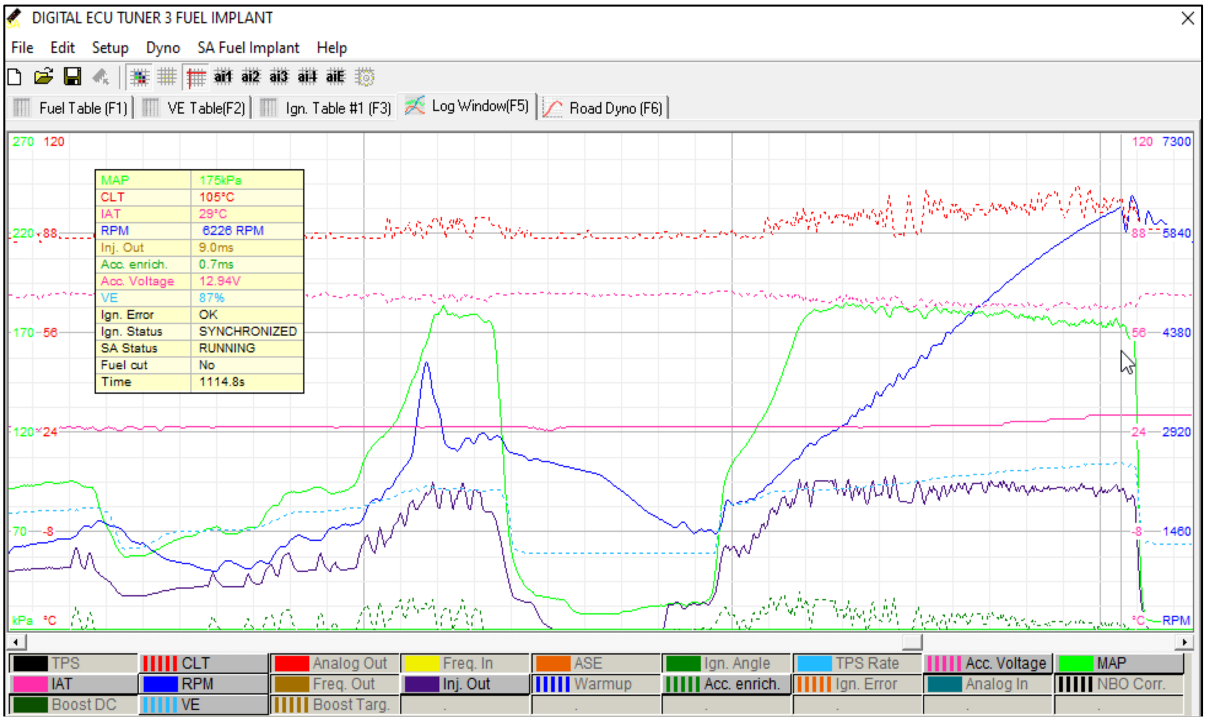

Figure 6. Graph of the engine parameters recorded during vehicle acceleration road trial

Figure 7 presents the compiled graphs created on the basis of measurements of the same model of a car with the same type of engine and gearbox taken prior and post modification on a chassis dynamometer. The results were compiled for the purpose of this analysis on the grounds of the results of measurements made available online by the person taking the measurements.

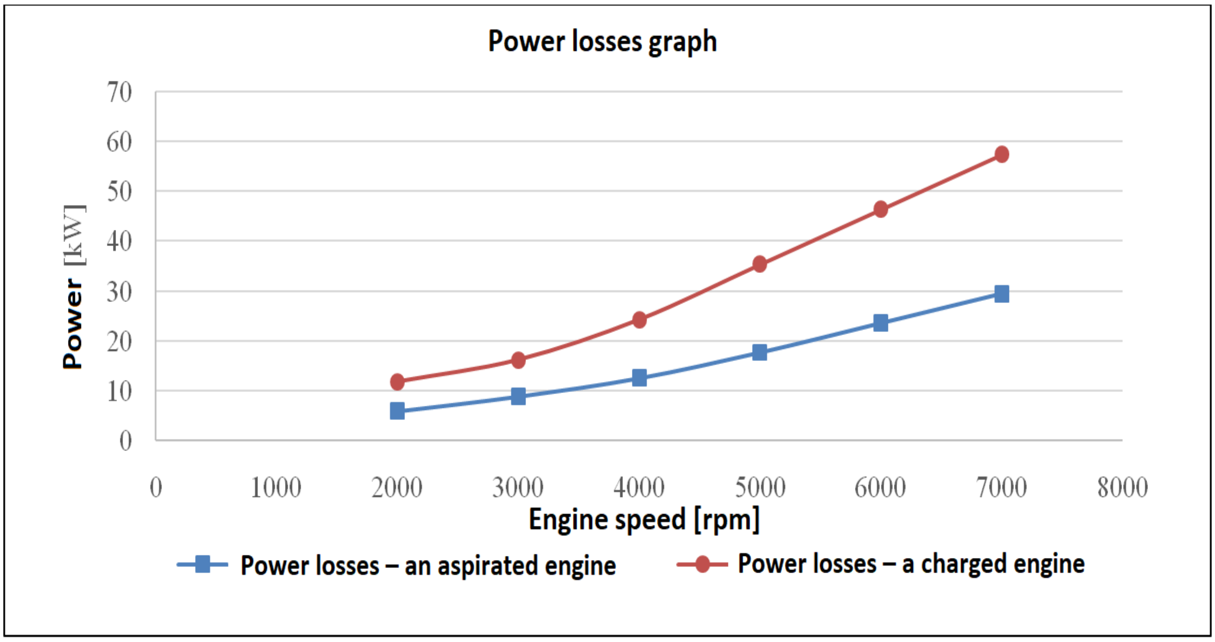

Figure 7. Power losses graph

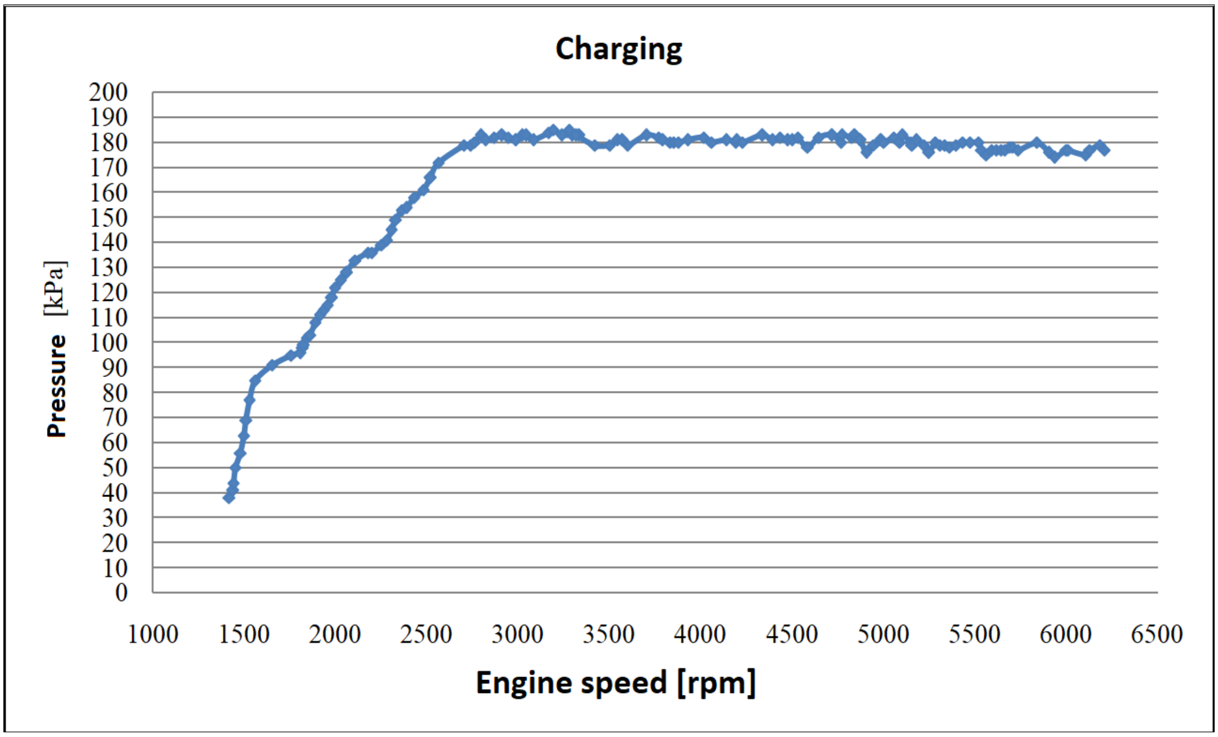


Figure 8. Graph of absolute suction pressure downstream of the turbocharger in relation to engine rotational speed

Figure 8 presents the intended progress of the pressure generated downstream of the turbocharger compressor in relation to the engine rotational speed under full load of the engine in the studied vehicle. The study of the charging pressure has been performed during third gear driving; after reducing engine speed to approx. 1500 rpm the throttle has been fully opened in order to ensure that the engine is operating under the highest possible load. By the time of reaching 2700 rpm the constant charging at the level of 80 kP of relative pressure displays high efficiency of the turbocharger compressor and only a slight increase in the temperature of air achieved through use of a highly efficient air cooler. In case of the naturally aspirated engine the temperature of intake air recorded during the study was 24°C whereas the highest temperature recorded for the turbocharged engine was 29°C, which proves that the compressor was selected correctly.

As it can be observed in Figure 9 the development of torque and power of a turbocharged engine is very similar in terms of shape but displays markedly higher values across the entire range of rotational speed than the graph for a naturally aspirated engine. It is so because the engine remained unaltered in terms of construction and structure and the increase is based on increasing the average effective pressure of the cycle through feeding greater amount of fuel-air blend.

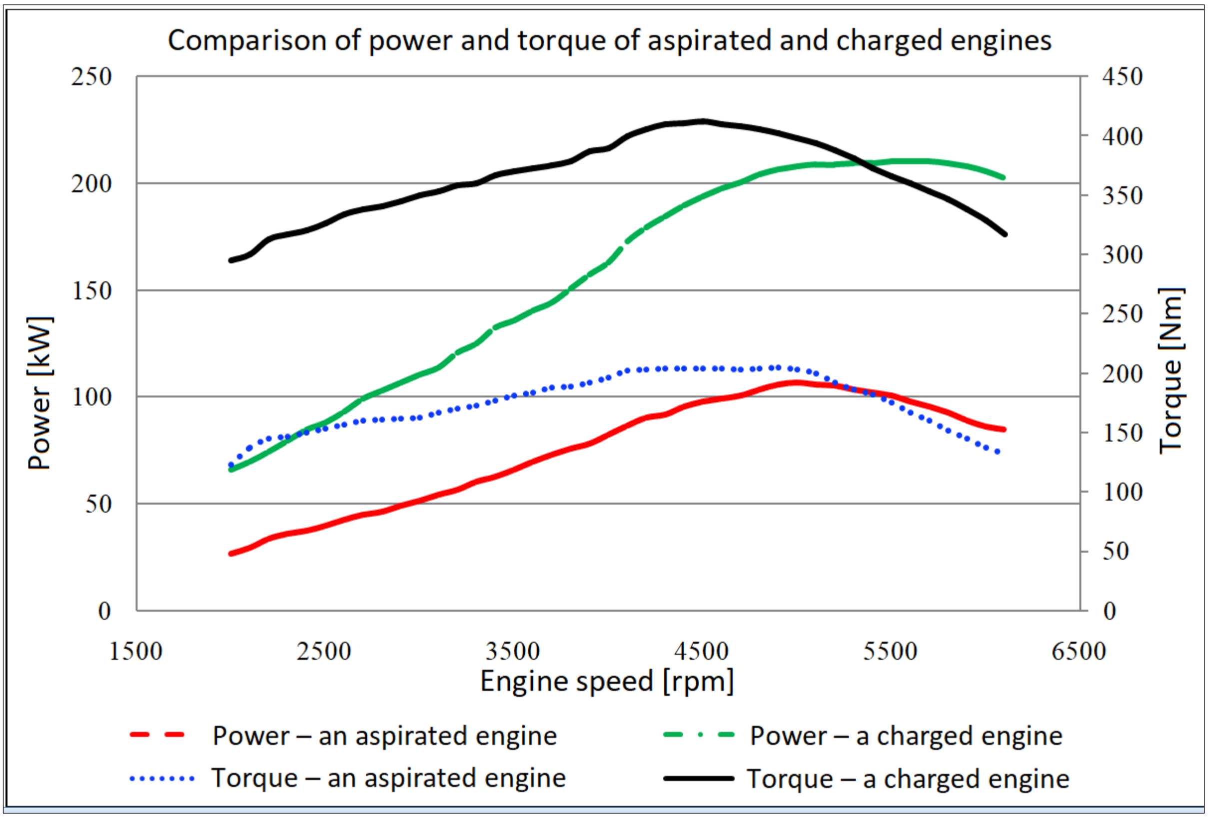


Figure 9. The graph of actual power for a naturally aspirated and a turbocharged engine

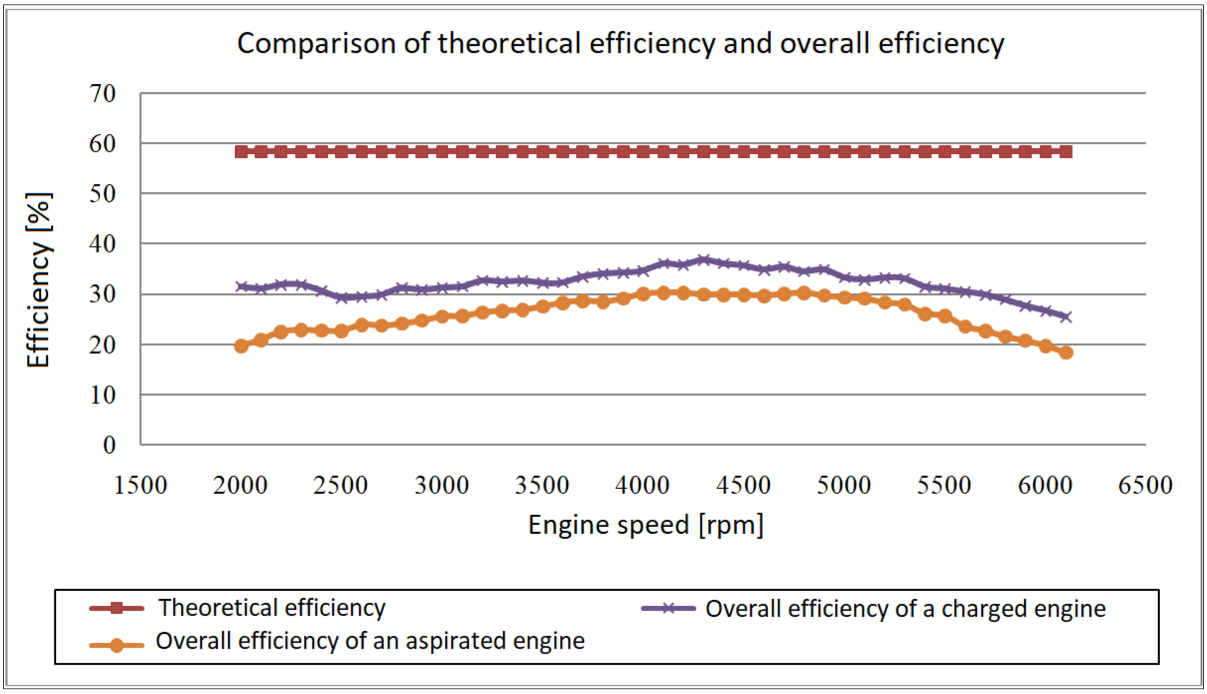


Figure 10. Theoretical efficiency and overall efficiency graph

Improving the overall engine efficiency (Figure 10) is a result of better piston feeding due to application of turbocharging and replacement of the fuel injection system. The theoretical model does not take into account improvement of efficiency as both these elements are treated in the model as perfect. Therefore we may expect greater percentage improvement of power and torque in an actual engine in comparison to the theoretical model, by 10 percentage points at the least.

Figure 11 displays power and torque increase in comparison to a naturally aspirated engine in terms of the measured power and torque as well as the theoretical power with and without dissociation. It is demonstrable that a more accurate model (taking dissociation into consideration) better displays the achieved actual effects but even the ordinary model, very simple to recalculate, enables assessment of the potential of an engine. After incorporation of a turbocharger compressor the theoretical power in the dissociation model increases by 85% and by 108% percent in an actual engine – a difference of 23 percentage points. This enables us to assess potential of the engine through calculations. As it has been ascertained the efficiency of the studied engine has been improved by at least 10 percentage points by use of means other than turbocharging and as a result we were able to achieve greater power and torque increase than indicated by simple theoretical models with constant theoretical efficiency.

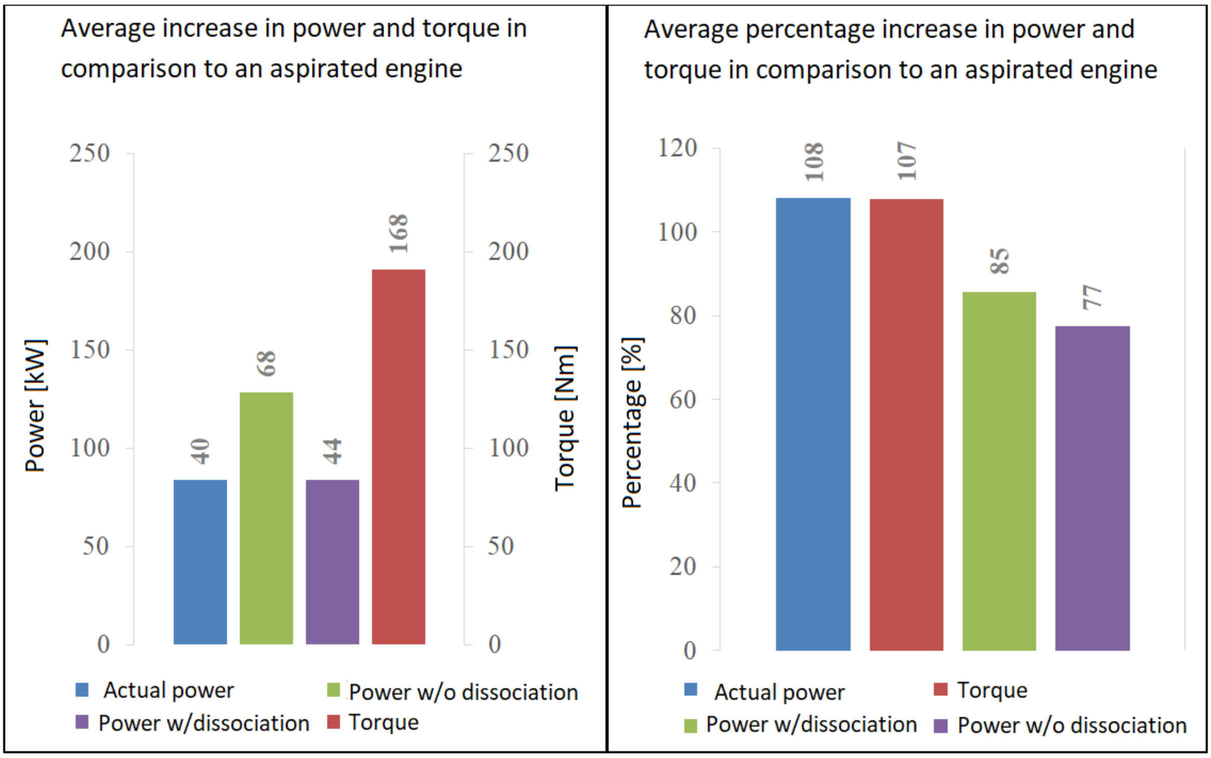


Figure 11. The average absolute increase in power [kW] and torque [N·m] as well as appropriate values of relative increase in power and torque [%]

An additional effect of improving engine operation is reduction of specific fuel consumption for a turbocharged engine by 70 g/kWh on average as a result of increased engine efficiency displayed in Figure 10. The increased flexibility of engine operation has also been observed.

## 4. Conclusions

This paper demonstrates that even the simplest theoretical model enables approximate assessment of the possibility of improving power and torque of a naturally aspirated SI engine after incorporation of a turbocharger compressor. In a perfect model increase in efficiency cannot be assessed as this value remains constant in the Otto cycle and is dependent solely on geometry of an engine. In the experimentally researched engine the efficiency has been further improved and thus influenced reduction of fuel consumption and $CO_2$ emission as well as power and torque and therefore the increase in case of an actual engine is even higher than calculated for a theoretical model. This is, however, not solely the result of increasing pressure of the intake air but also the result of modifications to injection system which enable improving progression of work process in the combustion chamber. Furthermore, increasing suction pressure increases the amount of blend in the piston and limits losses resulting from suction vacuum. The remaining elements of the theoretical model remain "perfect" and thus the efficiency of an actual engine may only attempt to approach the calculated values. The increase in torque and efficiency is dependent on pressure and temperature of the sucked blend which increase pressure and temperature towards the end of the isochoric burning of blend although the blend itself is in reality limited by $CO_2$ dissociation.

On the grounds of the performed studies it can be ascertained that theoretical calculations enable assessment of possibility of improving engine parameters through calculations and that the model accounting for $CO_2$ dissociation ensures better and more accurate assessment than the model without dissociation.

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