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TURBOCHARGING OF HIGH PERFORMANCE COMPRESSED NATURAL GAS SI ENGINE FOR LIGHT DUTY VEHICLE

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ABSTRACT
Natural gas as an automotive fuel has many benefits in comparison with traditional fossil fuels. Favorable anti-knock properties of methane allow us to utilize higher boost levels and the engine power than that of gasoline engines. High level of intake boosting make possible to achieve loads, comparable to the state-of-the-art diesel of engines without soot and PM emission. Stoichiometric operation within the full range of the complete engine map enables the use of a relatively simple exhaust gas aftertreatment, based on a three-way catalyst.

The paper describes a chosen 1-D thermodynamic modelling studies, calibrated and validated by experimental data. The investigations were performed on a spark ignition, direct injection, four-cylinder engine with 1.6 L displacement. The engine was optimized for mono fuel operation with compressed natural gas.

Due to complexity of gaseous fuel infrastructure in vehicles, compared to the traditional fuels, it is desirable to keep the turbocharging system as simple as possible. Traditional variable geometry turbine systems were tested. Practical design constraints as peak cylinder pressure, turbine inlet temperature, compressor outlet temperature and others were met. Various strategies on how to achieve high load at low engine speed were investigated.

The authors propose a single stage turbocharger to cover the demand for a high torque at low engine speed and high power at full speed, with boost levels comparable to a dual stage turbocharging. It was concluded that the single stage turbocharging enables the engine to operate with maximum BME of 3 MPa between 1500 and 2750 rpm. Maximum engine speed had to be limited to a similar value that is usually applied in a diesel engine due to limited control range of turbocharging.

KEYWORDS: CNG ENGINE, MILLER CYCLE, VALVE TIMING, TURBOCHARGER

SHRNUTÍ
Zemní plyn jakožto automobilní palivo má mnoho výhod ve srovnání s tradičními fosilními palivy. Příznivé antidetonační vlastnosti metanu umožňují využít vyšší plnicí tlak a dosáhnout vyššího výkonu motoru než u benzínových motorů. Vysoká úroveň plnicího tlaku umožňuje dosáhnout zatížení, které je srovnatelné se vznětovými motory bez emisí sazí a pevných částic. Stechiometrická koncepce v celé pracovní oblasti motoru umožňuje použití relativně jednoduché následné úpravy výfukových plynů, založené na trojcestném katalyzátoru.

Článek popisuje vybrané studie termodynamického 1-D modelování, kalibrované a ověřené experimentálními daty. Výpočty byly provedeny na zážehovém čtyřválcovém motoru s přímým vstřikem paliva o objemu 1,6 l. Motor byl optimalizován pro provoz se stlačeným zemním plynem.

Vzhledem ke složitosti infrastruktury plynných paliv ve vozidle ve srovnání s tradičními palivy je žádoucí, aby systém přeplňování byl co nejjednodušší. Byly testovány tradiční turbíny s variabilní geometrií. Byly dodrženy praktická konstrukční omezení jako tlak ve válci, vstupní teplota výfukových plynů do turbíny, výstupní teplota stlačeného vzduchu z kompresoru a další. Byly zkoumány různé strategie, jak dosáhnout vysokého zatížení při nízkých otáčkách motoru.

Autoři navrhují jednostupňové turbodmychadlo, které pokryje poptávku po vysokém točivém momentu při nízkých otáčkách motoru a po vysokém výkonu ve vysokých otáčkách motoru, při úrovni plnicího tlaku srovnatelné s dvojstupňovým turbodmychadlem. Výsledkem je návrh jednostupňového turbodmychadla umožňujícího motoru pracovat s maximálním BMEP (středním efektivním tlakem) 3 MPa v rozsahu otáček 1500 a 2750 min^{-1}. Maximální otáčky motoru musely být sníženy na podobnou hodnotu jako u naftového motoru kvůli omezenému regulačnímu rozsahu turbodmychadla.

KLÍČOVÁ SLOVA: PLYNOVÝ CNG MOTOR, MILLERŮV CYKLUS, ČASOVÁNÍ VENTILŮ, TURBOKOMPRESOR

1. INTRODUCTION

Compressed natural gas as a fuel for spark ignition internal combustion engines, besides the advantages as a high resistance to knock, has favorable hydrogen to carbon ratio [1] and also the benefit of possibility to direct injection into the combustion chamber [2, 3, 4]. From the efficiency point of view, high resistance to knock allows the exploitation of high compression ratio and also the possibility to achieve high power density. For this, the DI concept, compared with conventional port fuel injection [5] significantly improves volumetric efficiency and reduces demands for turbocharging as an alternative to two stage turbocharging [6, 7]. Combination of the direct injection and a single stage turbocharger enables us to achieve high performance of gas SI engine. Previous studies show maximum achievable performance of the conventional ICE at the level of 3.5 MPa of brake mean effective pressure [8, 9, 10, 11], when uncompromising, unconstrained optimization of engine parameters (would be) implemented.

In comparison of natural gas direct injection to a Diesel direct injection and gasoline direct injection engines the particle matter emission issue can be eliminated. CO_2 emission index can be reduced by advantageous ratio of hydrogen to carbon in methane as a main component of natural gas. Full operational map stoichiometric engine operation enables to usage of a relatively simple aftertreatment system, based on a three-way catalyst.

Presented article describes selected studies from the thermodynamic modelling performed within the framework of subprojects WP04 and WP06 of Horizon 2020 EU project GasOn [12]. The subproject aims at the development of the VAN with the mono fuel SI turbocharged engine of cylinder displacement of 1.6 liter, with a direct injection of CNG and with a full map stoichiometric operation. Authors were responsible for the simulation support of the engine development. As a main tool a 1-D GT Suite [13] software was used. Target performance parameters are following:

Engine torque 180 Nm (BMEP 1.4 MPa) at 1000 rpm
Peak engine torque 380 Nm (BMEP 3 MPa) at 2000 rpm
Maximum power 125 kW (78 kW/l) at 5500 rpm.

The speed range for the maximum engine power was not a fixed value. According to the initial estimate the speed range of the peak power might lie within 3500 – 5500 rpm.

Large number of simulations have been carried out within the framework of this project. The main activity was the matching of the single stage turbocharger for fulfilling the demanded engine performance parameters, including the selection of the turbocharger control system using the variable turbine geometry (VTG) or the exhaust gas bypass of the turbine, so

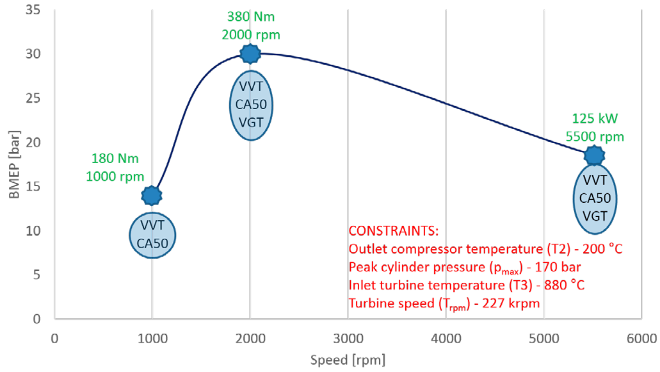

FIGURE 1: Main targets
OBRÁZEK 1: Hlavní cíle

called waste gate (WG). In all investigated areas, the variable valve timing and two cam profile switching [14] were taken into account with optimization of combustion phasing and optimum rate of exhaust gas recirculation (EGR). The design of appropriate EGR loop was a part of this process. All of the simulations were performed with respect to the practical design and operational limits. The authors would like to point out several specific challenges and the approach on how to handle them.

2. ENGINE MODEL

For the task of turbocharger matching, valve lift and timing design an engine thermodynamic model was developed in a GT Power software. Model was created by merging of the two different models. Firstly, a model of a diesel four-cylinder engine with the similar geometry parameters (piston stroke, cylinder bore and intake and exhaust manifolds) and secondly, the model of a single cylinder engine with a direct injection of gaseous fuel into the cylinder [7], however with different cylinder bore.

Following features and components have been modified for natural gas engine performance simulations:

- Intake manifold was arranged to the final design of the GASON engine.
- Intercooler efficiency was adjusted by empirical equation provided by industrial partner.
- The valve lift profiles and a base valve timing was derived from the original Diesel engine model.
- Short duration cam profile, suggested for a full load at low engine RPM, was newly designed to fulfill the GASON project targets.
- Cylinder heat transfer was adjusted based on a sensitivity study and author's experience from the former R&D activity. Finally, unmodified Woschni correlation was used for a heat transfer coefficient determination.

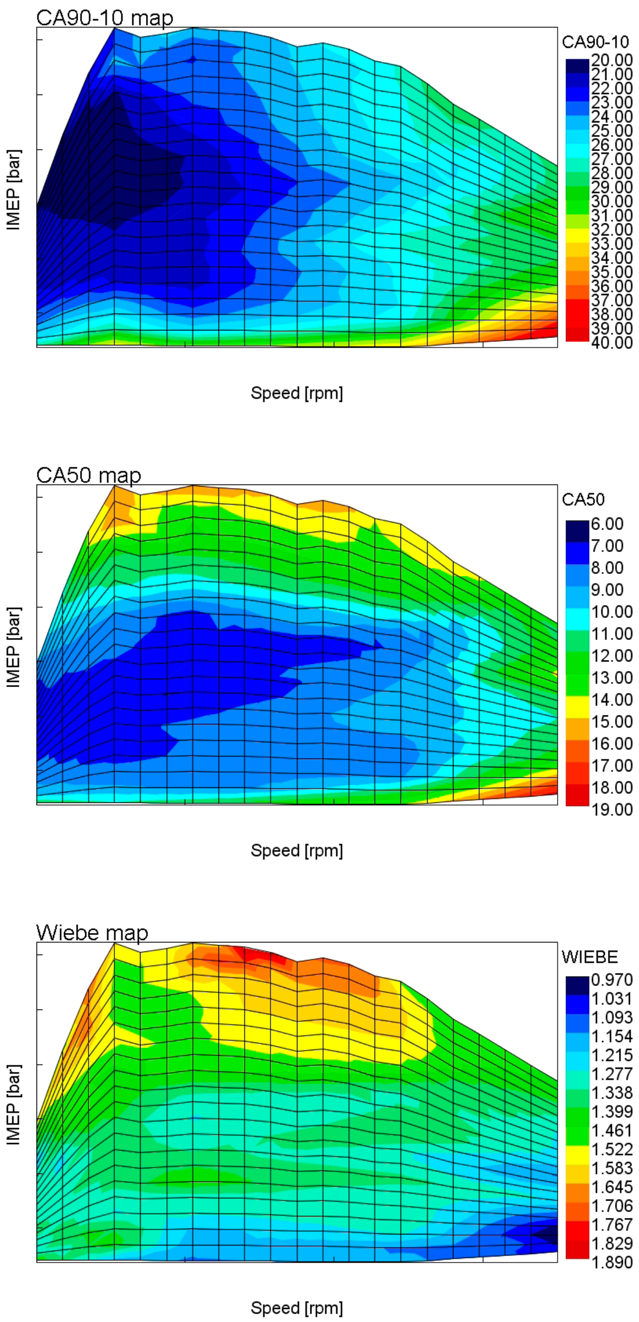

FIGURE 2: Maps of combustion parameters
OBRÁZEK 2: Mapy parametrů hoření

- Back pressure of a real exhaust gas aftertreatment system and muffler was emulated using a friction multiplier of individual components of the exhaust system.
- EGR cooler efficiency was adjusted based on empirical data obtained from the industrial partner.
- A simple GT Power direct natural gas injector model was designed according to measured injection mass in dependence of injection pulse width characteristic provided by industrial partner;
- Model of closed loop lambda=1 control was designed.
- Demanded power control strategy was prepared as a combination of
  – Closed loop turbocharger variable turbine geometry (VTG) control
  – Closed loop throttle valve control
- Variable valve timing (VVT) for minimum BSFC

Parameters of intake and exhaust ports were adopted from the model of the single cylinder engine and the data from the combustion models that were calibrated based on the results of a single cylinder engine combustion analysis [7, 15]. A GT Power EngCylCombSIWiebe model was selected for this task and the maps of the parameters, covering the full engine characteristics are displayed in Figure 2.

The top Figure shows a map of a combustion duration (CA90-10), the middle Figure shows a map of the combustion phasing expressed as an angle of a 50 percent burn mass (CA50) and at the bottom figure, a Wiebe exponent map is presented.

To ensure an appropriate fraction of EGR rate, the model was coupled with a low pressure cooled EGR system, including its control [16]. The model also comprises a switchable system for the control of the peak cylinder pressure by means of the variation of the spark advance control.

All the control systems were tuned in respect to the simulation stability. All simulations at this stage were performed in steady state conditions. The engine performance targets had to comply with practical constraints given by the engine and TC manufacturers:

- Compressor outlet temperature T2
- Turbine inlet temperature T3
- Turbocharger speed T_{RPMMAX}
- Peak cylinder pressure p_{MAX}
- Compressor surge limit according to the steady state compressor map

Due to lack of experimental data the knock occurrence was completely neglected in this study.

3. METHODOLOGY

Three sensitivity studies were selected for this article.

1. Comparison of real and unconstrained turbocharger
2. Usable engine speed range with the single stage turbocharger
3. Achievable low speed torque based on intake cam profile

All studies were performed for important points of the engine full load curve.

4. RESULTS

4.1 REAL VS. UNCONSTRAINED TURBOCHARGER

The main goal of this study was assessment of the possibility of using of Miller cycle [14] within the region of the maximum achievable torque of 380 Nm at 2000 rpm. The top left part of the Figure 3 shows the contours of the brake mean effective pressure (BMEP) in the coordinate system given by intake valve timing in case of the real turbocharger. The horizontal axis represents intake valve opening (IVO) and the vertical axis represents the intake valve closing (IVC). The constant valve lift is assumed, it means that by changing of the mentioned parameters the cam profile is shifted, stretched or narrowed along the crank angle coordinate. Original valve timing is marked in the figure. VTG rack position was chosen with respect to maintain demanded BMEP of 30 bar. It can be observed that retarding of the IVC (so called Millerization) has the limits, given by the parameters of selected turbine in a limit position of the VTG rack (with corresponding value of its efficiency). With further retardation of the IVC the BMEP drops under the demanded limit. The top right Figure displays the same situation when the so-called unconstrained TC is used. It means that for this case the relevant efficiencies were constant within the whole operation region with the values of compressor efficiency $C_{eff} = 74\,\%$, and turbine efficiency $T_{eff} \approx 70\,\%$. By using the unconstrained turbocharger, the region with demanded BMEP is enlarged and the drop under the limit value does not occur until the very early closing of the intake valves.

Bottom part of the Figure 3 shows a brake specific fuel consumption (BSFC) dependency on intake valve timing for both turbocharger variants. In case of real turbocharger, the IVC retarding causes the increase in the BSFC and it is evident that it is beneficial to keep the original timing of the intake valves. In this case, the benefits of Millerization do not balance the shift of the compressor and turbine characteristics into the region with lower efficiency. Therefore, the resulting BSFC is worsened. On the other hand, in case of the unconstrained turbocharger, the reduction in BSFC is clearly visible, partly due to gradual Millerization and partly due to shift of IVO with retarded values, which for the constant valve timing means reduction of the valve overlap angle. Numerically, this BSFC reduction by means of Millerization can reach 1.5 %.

4.2 USABLE SPEED RANGE GIVEN BY A SINGLE STAGE TURBOCHARGING

The next study deals with a determination of the optimum engine speed for a maximum power output. Small turbine can help to achieve of target BMEP in low and medium engine

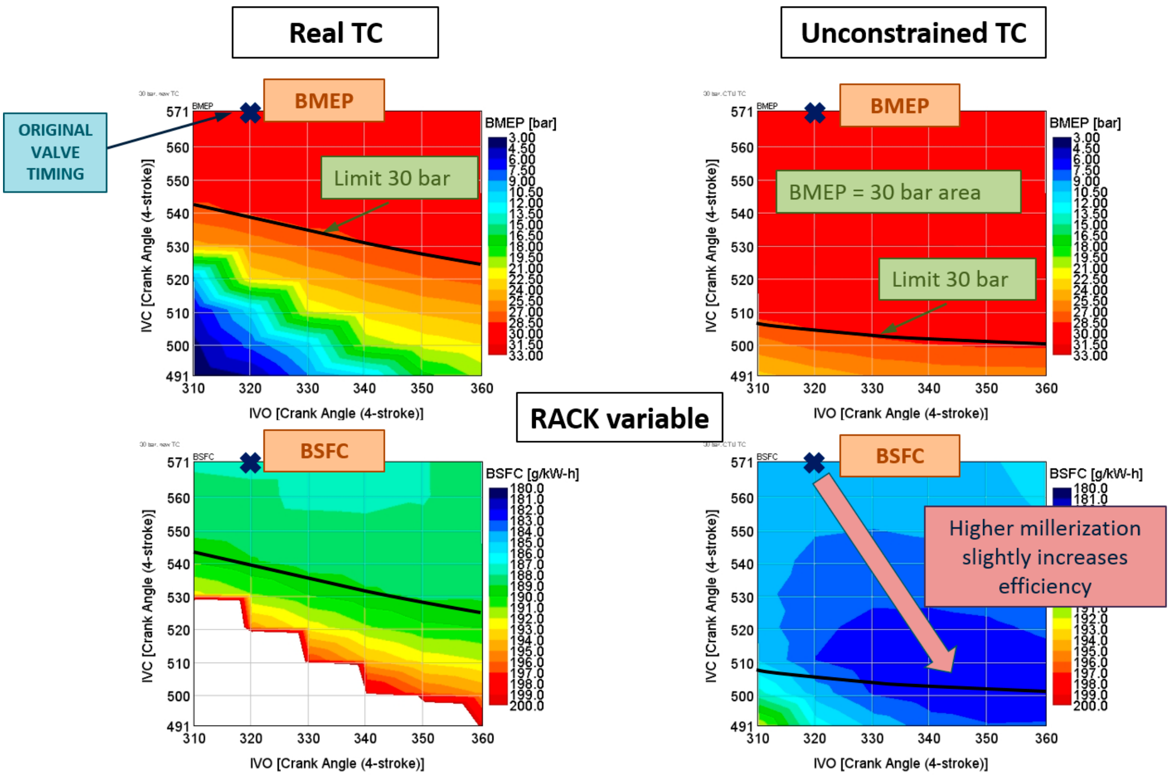

FIGURE 3: Comparison of turbochargers
OBRÁZEK 3: Porovnání turbodmychadel

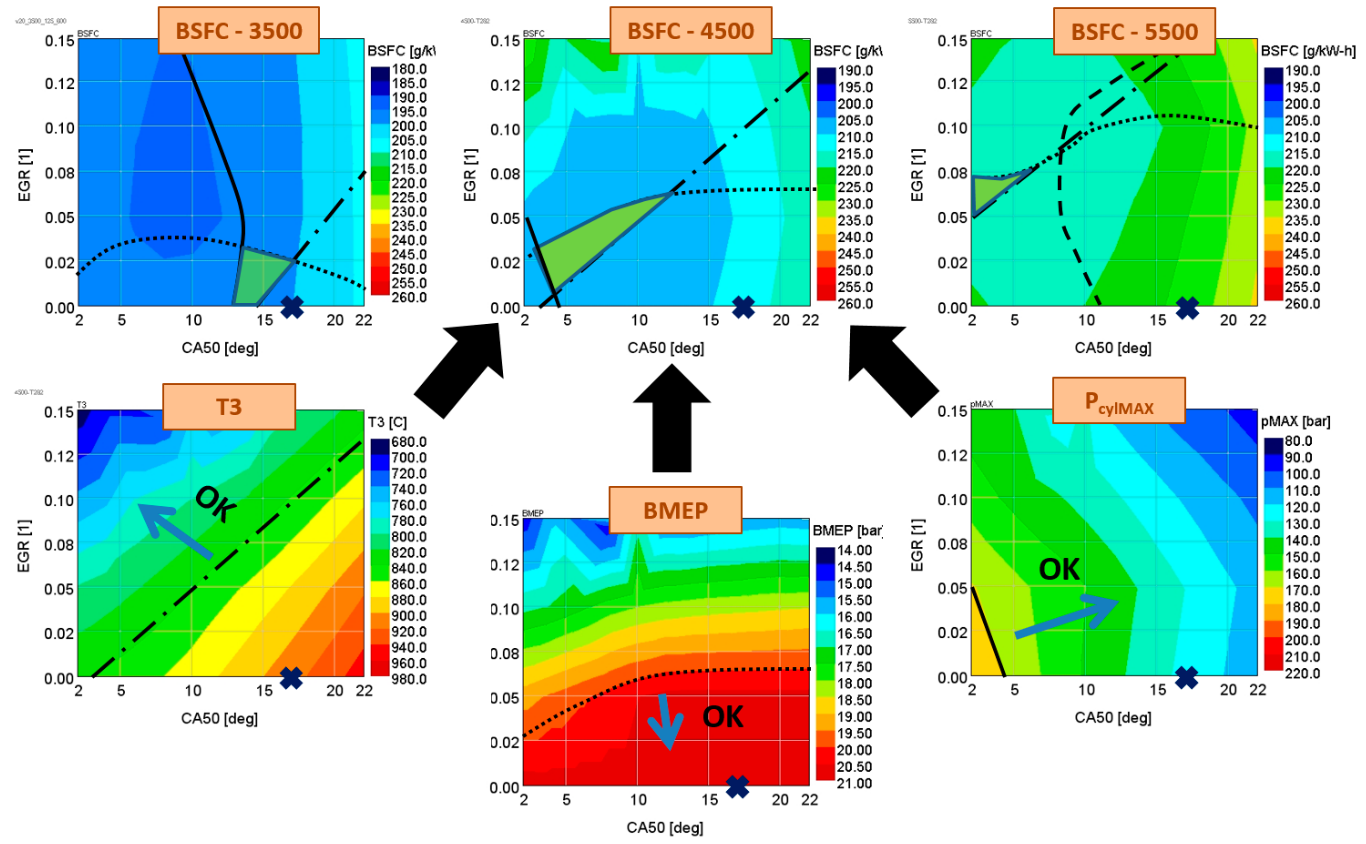

FIGURE 4: Full load speed comparison
OBRÁZEK 4: Porovnání v oblasti maximálního výkonu

speed. However, there is no mass flow reserve for a low pressure (LP) EGR. On the other hand, a larger turbine gives a sufficient flow reserve in the full power regime, but it cannot give a sufficient level of the boost level in low engine speed. Lowering the speed for the maximum power eliminates this compromise and the gas engine characteristics approaches the characteristics of a diesel engine.

The top part of the Figure 4 presents a dependency of the BSFC in the coordinates of the CA50 and ratio of the LP EGR for three engine speed values of 3500, 4500 and 5500 rpm at the power output of 125 kW. The black cross marks a base adjustment of the CA50 and LP EGR. The green region corresponds to area in which all the practical constraints are fulfilled. Bottom figures present in the specific case of 4500 rpm the example of the creation of the working regions. The middle Figure at the bottom of the Figure 4 the BMEP value is depicted by colored contours. The dotted black line limits the region of the complying with the criteria of the target BMEP and the grey arrow shows the direction inside the region. At the bottom right graph, the regions of maximum cycle pressure is displayed by the color contours. The black continuous line limits the region of the fulfilling the criteria with lower peak cylinder pressure than the limit one (p_{MAX}) and again the arrow shows the direction of the appropriate region. At the bottom left graph, the contours show the exhaust temperature upstream of the turbine T3. The dashed line limits the usable region with temperature lower than the limit one.

It is evident from the comparison of the top figure, that the reduction of the BSFC can be achieved by shifting of the speed of the maximum engine power output towards lower values. Reduction or a complete elimination of the need for the demanded LP EGR is another benefit of this shift. Therefore, the speed of 3500 rpm was selected as the maximum power speed.

4.3 LOW SPEED TORQUE CAM PROFILE DEPENDENCE

The last study is related to optimization of the intake cam profiles for the speed of 1000 rpm. This low speed proved to be the key to the TC matching, because of the difficulty with the achievement of the desired BMEP at the base intake cam profiles and valve timing. For practical engine operation the use of the two different intake cam profiles and the mechanism of their continuous phasing are assumed. The first profile is the one adopted from the single cylinder engine being marked as 169CAD. The seeking for an appropriate second profile was the subject of performed optimization. The profile was designed the way that a maximum lift and opening angle was

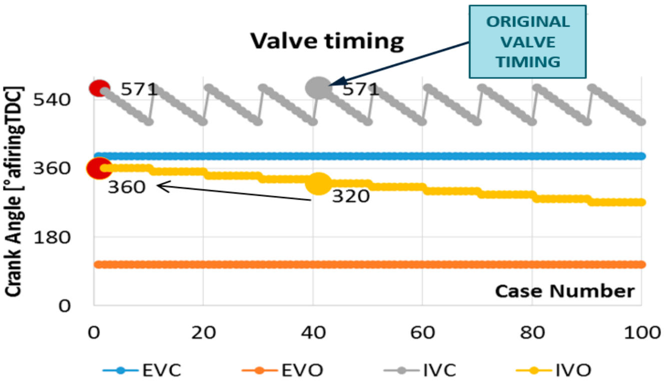

FIGURE 5: Valve timing sweep strategy
OBRÁZEK 5: Strategie vyhledávání časování ventilů

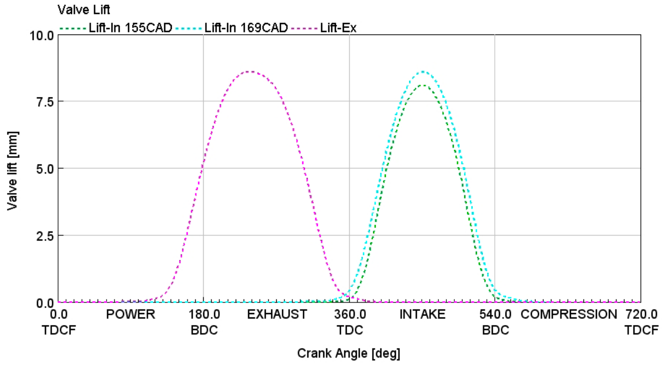

FIGURE 7: Valve profiles
OBRÁZEK 7: Profily ventilů

determined to maintain constant dynamical properties of the valve (the acceleration profile). Parametric study of the intake valve timing was carried out for each speed and TC variant. As a rule, the IVO was kept constant and the IVC was varied simultaneously with the maximum valve lift. Exhaust valve lift profile and timing was kept constant. This strategy is plotted in Figure 5.

One example of the result of this study is plotted in Figure 6. All results are displayed in the coordinate system with IVO as a function of IVC. The black cross marks again the base intake valve timing. The color contours in the top left Figure plots the achieved engine BMEP, the black continuous line represents the limit line of the BMEP. It is evident that the retarded IVO is necessary for achievement of the limit BMEP. This narrows the cam profile and also reduces the valve overlap. The top right Figure shows that BMEP reduces the engine BSFC accordingly, indeed, the increased BMEP causes an increase in temperature T3 and peak cylinder pressure p_{MAX}, which can be observed from the bottom graphs in Figure 6.

After the optimization the resulting variant of the intake cam profile marked as 155CAD was designed.

Both variants of cam profiles are displayed in Figure 7, the 169CAD with the blue dotted line and the 155CAD with a green dotted line. The violet curve describes a profile of the exhaust cam. Further pictures will use the same color coding and line types of the profiles.

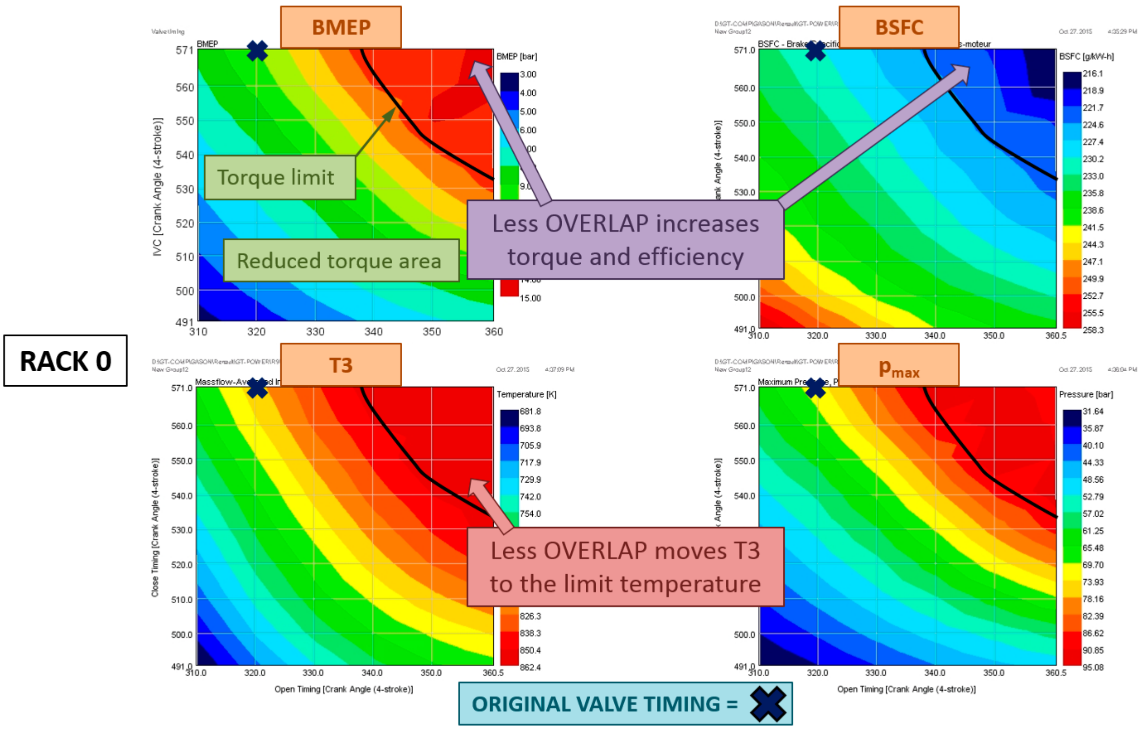

FIGURE 6: Results of valve timing sweep
OBRÁZEK 6: Výsledky vyhledávání časování ventilů

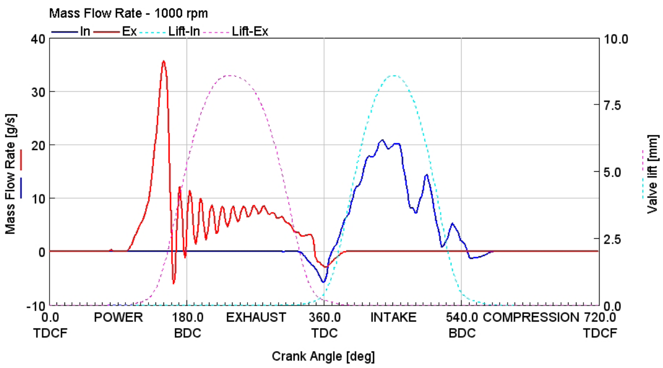

FIGURE 8: Mass flow rate through valves– 169CAD
OBRÁZEK 8: Hmotnostní tok ventily– 169CAD

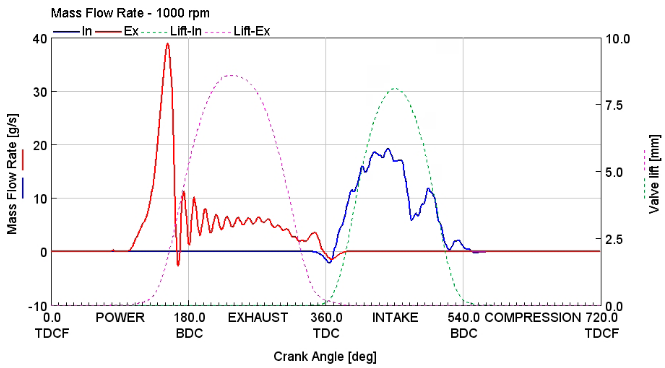

FIGURE 9: Mass flow rates through valves– 155CAD
OBRÁZEK 9: Hmotnostní toky ventily – 155CAD

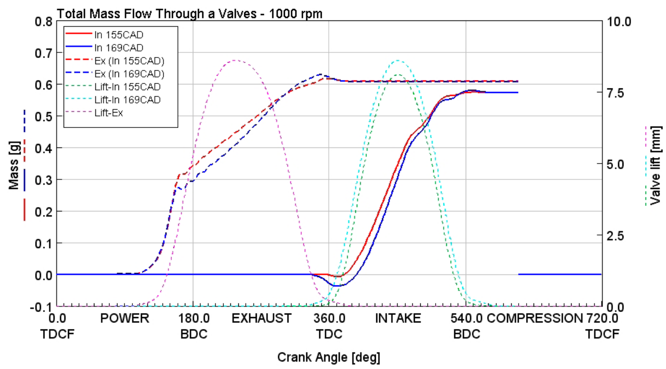

FIGURE 10: Total mass flow through valves
OBRÁZEK 10: Integrál hmotnostních toků pracovní látky proteklé sacími ventily

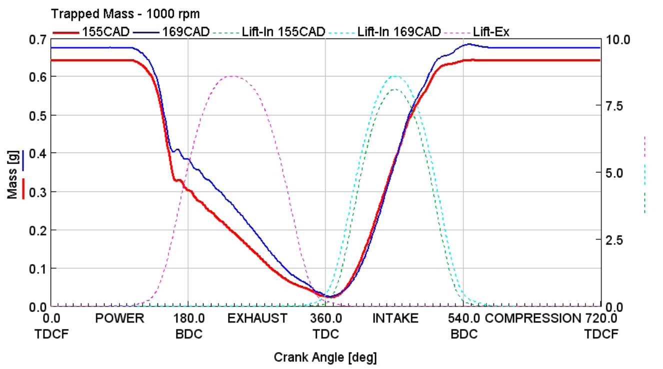

FIGURE 11: Total mass flow through valves
OBRÁZEK 11: Integrál hmotnostních toků pracovní látky proteklé sacími ventily

Figure 8 and Figure 9 show crank angle-based profiles of the instantaneous mass flow through the intake and exhaust valves. Blue line represents the intake air mass flow and the red line plots the instantaneous mass flow of the working substance through the exhaust valve. Dotted line represents the exhaust and intake valve lift profiles according to the color code defined in the previous paragraph.

By comparison of mass flow through the intake valves, it is evident that higher back flow of the cylinder charge occurs during the valve overlap period for the cam with the profile of 169CAD (Figure 8) over the 155CAD (Figure 9) cam profile.

The wider valve overlap for the case 169CAD also increases back flow through the exhaust valves. Figure 10 shows the total integral of the mass flows and confirms previous statements. The red continuous line represents total mass inflow through the intake valve for the 155CAD cam, blue continuous line plots the total mass inflow through the intake valve for the 169CAD case. The exhaust total outflows are plotted with the dashed curves. Dotted lines show all of the valve lift profiles. The difference between inflow and outflow is caused by the quantity of the fuel directly injected into the cylinder.

The higher back flow through the valves affects significantly the residual gas content in the cylinder.

Even though the total mass inflow through the intake valves for both cams after the closing of the intake valves is equal. Figure 11 shows a visible difference in the trapped cylinder charge mass after the intake valve closing. From the composition of the cylinder charge after the intake valve closing the model shows residual gas content for the 155CAD cam of 5.4 % compared to 10.2 % for the 169CAD cam. Higher residual gas content for the 169CAD cam causes the higher demand for the boost pressure to reach equal BMEP as the 155CAD cam. The sufficient boost pressure for the 155CAD cam is 1.63 bar while the 169CAD cam demands 1.82 bar. To achieve higher boost pressure with the same turbocharger, it is necessary to increase turbocharger speed and shift the VGT Rack position toward lower limit value that means to the region of the turbine map, where the mass flow and efficiency values are reduced. This increases the turbine inlet pressure from the value of 1.67 bar for the 155CAD cam to the level of 2.33 bar for the 169CAD cam. Increased turbine inlet pressure results in a higher backflow of the exhaust gas to the cylinder, which amplifies above mentioned phenomenon.

The main effect to reach target level of BMEP by using the narrower cam lies in the reduction of the angle of the valve overlap. This will result in a lower backflow of the air through the intake valves to the intake manifold and the backflow of the exhaust gas to the cylinder from the exhaust manifold through the exhaust valves. The same amount of the fresh charge in the cylinder can be achieved with lower losses and hence with the lower boost pressure. The demand for the lower boost pressure at engine low speed is essential for the achievement of the demanded engine torque.

5. CONCLUSION

This article presents three selected sub-studies, which were performed during a complex development of the turbocharged CNG engine with high specific power output and with low brake specific fuel consumption.

Possibilities of using a Miller cycle were assessed in the first study. When the use of a real single stage turbocharger is considered, though with the cutting edge performance parameters, at the region of the engine maximum torque, with the gradual Millerization of the cycle the BMEP reduces and the fuel consumption increases as a result of prevailing losses given by the shift of the turbocharger working point towards the regions with lower efficiency. Theoretically, the full advantage of the Millerization can be taken by the use of turbochargers with flat efficiencies of the compressor and turbine within the whole working area.

The selection of the right speed range for maximum engine power is performed in the second study. Assuming the use of a single stage turbocharger for an engine with a high maximum specific torque and power, it is appropriate to adapt the speed range of the gas-fueled engine to the speed range corresponding to the current turbocharged diesel engines. By shifting the speed of maximum power to the lower values, it is possible to achieve the lower engine specific fuel consumption and completely eliminate required EGR.

The use of a single-stage turbocharger to achieve high maximum power at high speed and high torque at low speed on the other hand, may not occur without the change of the valve timing. The third study deals with the design of the valve timing parameters of the intake cam to achieve the high torque at low engine speed. The greatest effect of how the desired torque at low speed can be achieved is to reduce the valve overlap. This can be done by narrowing of the intake cams. By reducing the valve overlap, both the backflow of charge from the cylinder to the intake manifold and the backflow of the exhaust gas through the exhaust valves to the cylinder can be reduced. The reduction of the backflows will result in reduction of the required intake manifold pressure and allows seamless achievement of the desired torque.

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LIST OF NOTATIONS AND ABBREVIATIONS

BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
CA50	combustion anchor angle
CA90-10	combustion duration
C_{eff}	compressor efficiency
CNG	compressed natural gas
EGR	exhaust gas recirculation
EVC	exhaust valve closing
EVO	exhaust valve opening
IVC	intake valve closing
IVO	intake valve opening
LP	low pressure
P2	compressor outlet pressure
RACK	turbine rack position
p_{MAX}	maximum in-cylinder pressure
SI-BDUR	combustion duration
SI-THB50	combustion anchor angle
SI-WIEBE	Wiebe exponent
T2	compressor outlet temperature
T3	turbine inlet temperature
T_{eff}	turbine efficiency
T_{RPMMAX}	turbine maximum speed
VTG	variable turbine geometry
VVT	variable valve timing

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