Spark ignition turbocharged twin cylinder engine for vehicles

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Abstract: *This paper is about downsizing for small SI engines. Downsizing is a method of reducing engine displacement keeping the same engine power and speed by increasing inlet manifold density. These engines are generally turbocharged. We can achieve lower fuel consumption and CO2 emissions for downsized engine. The twin cylinder engine is a new concept for small vehicles (e. g. Fiat 500, VW Up). This paper is focused on engine simulation for 0D model and 1D model (software GT Power). Objective for this paper is to optimize turbocharging of twin cylinder engine (displacement: 803 cm*^{3}*). This paper researches intake and exhaust manifold for maximal power and low fuel consumption. The paper shows technical advantages and economic problems of this downsized engine.*

Introduction: European Union established emission norm Euro 1 in 1991. Since that time motor car companies have had to put into praxis exhaust manifold catalyst. Recently, the green-house gases are going to be limited, additionally. European parliament adopted for year 2015 new regulation for CO_2 emissions. The average CO_2 emissions for cars will be 125g/km CO_2. This is big chance for small downsized engines. Other advantages consist in improving engine performance and increasing torque at low speed.

1. New engine model

1.1. Currently manufactured twin cylinder engines

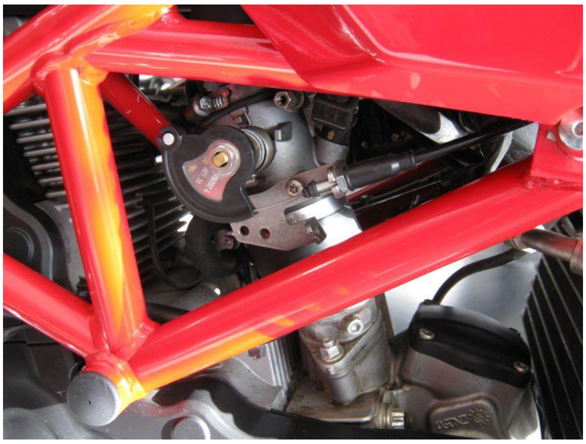
The spark ignition twin cylinder was used in history in Citroen 2 CV, Fiat Panda and Fiat 126. Nowadays, Fiat produces spark ignition twin cylinder engine only for model 500. I compared Fiat engine and many twin cylinder engines for motorcycles. Using these results I set ambitious target for my engine: the power of 80 kW at 6500 min^{-1} and brake mean effective pressure 22 bar inside the range of 1500- 4800 min^{-1}. Fiat has introduced brake mean effective pressure 14 bar at 1900 min^{-1}. The range of higher brake mean effective pressure is limited by knocking, especially at low engine speed. Knocking is a negative phenomenon caused by auto-ignition of unburned fuel/air mixture, which burns in almost shock wise and may cause engine failure.

	Ducati- motocycle	Fiat
Bore [mm]	88	80,5
Stroke [mm]	66	86
Displacement [cm^3]	803	875
Compression ratio [-]	10,4	10
Max. power [kW]	64 @ 8250 min^{-1}	62,5 @ 5500 min^{-1}
Max. torque [Nm]	78 @ 6250 min^{-1}	145 @ 1900 min^{-1}
Intake valve diameter [mm]	43	no data available
Exhaust valve diameter [mm]	38	no data available

1.1.1. SI twin cylinder engines

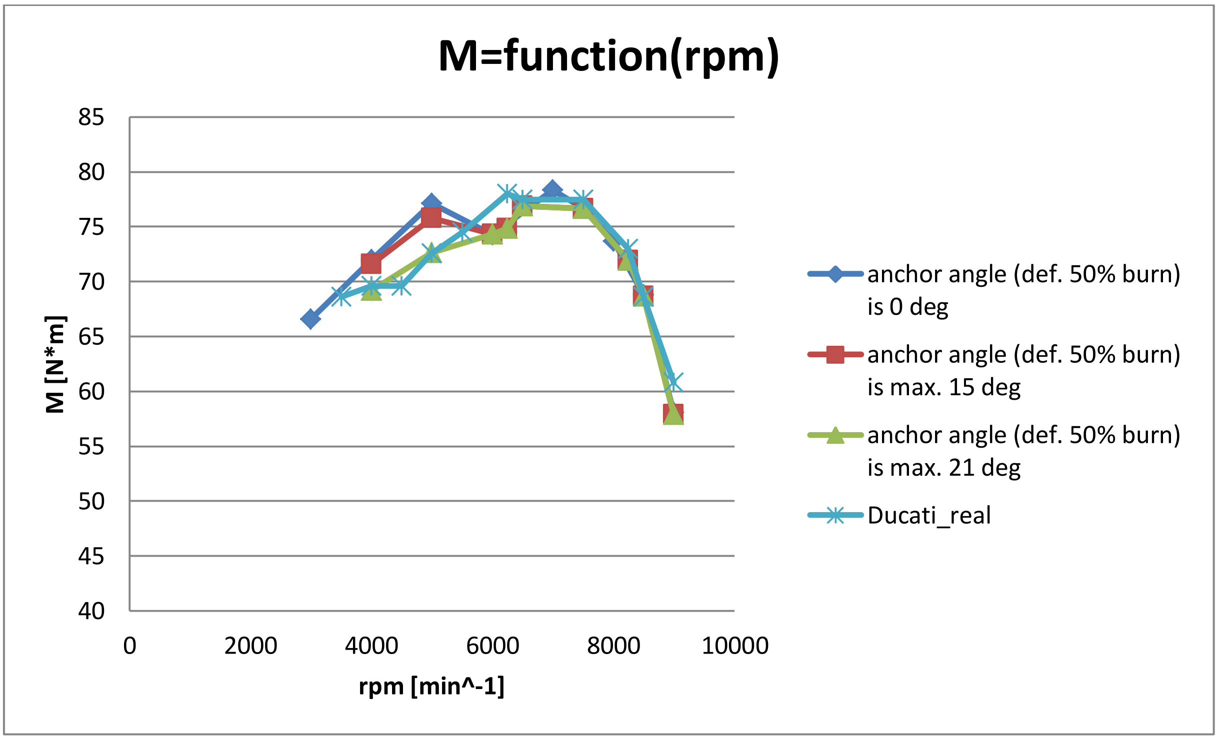
1.2. The engine model

The Ducati engine was selected for modeling engine in GT Power (1 D simulation software for engines). This is an engine for motorcycles, which was remodeled for vehicles. The Fiat engine was not selected because of incomplete technical data. The technical parameters of Ducati engine are based on a service manual but this information is insufficient for complete calibration). I made photographic documentation. I measured exhaust and intake manifold diameter and length. The model has many simplifications (e. g., air box is modeled in GT-Power as a cube).

1.2.1. Real Ducati engine

The engine model results were compared with the measured data using WOT torque in dependence on engine speed. The calibration found approximation of the measured data by finding suitable combination of chosen independent values, i.e., valve timing, ROHR anchor angle - defined by 50% burnt mass. The measured diameter and length of exhaust and intake manifold was changed from 85 to 115% for elimination inaccuracy of manifold simplifications. Large differences between a real engine and the model occurred especially at low speed. ROHR anchor angle was moved to expansion from this reason. This setting decreases knocking and it is probably used at the Ducati engine, as well. The final calibration of the engine is presented in Figure 1.2.2. Maximal torque difference between the real Ducati engine and the model was reduced to 3 Nm at 6250 min^{-1}, which is only 2.3 % torque error.



1.2.2. Comparison between real Ducati engine and GT Power model

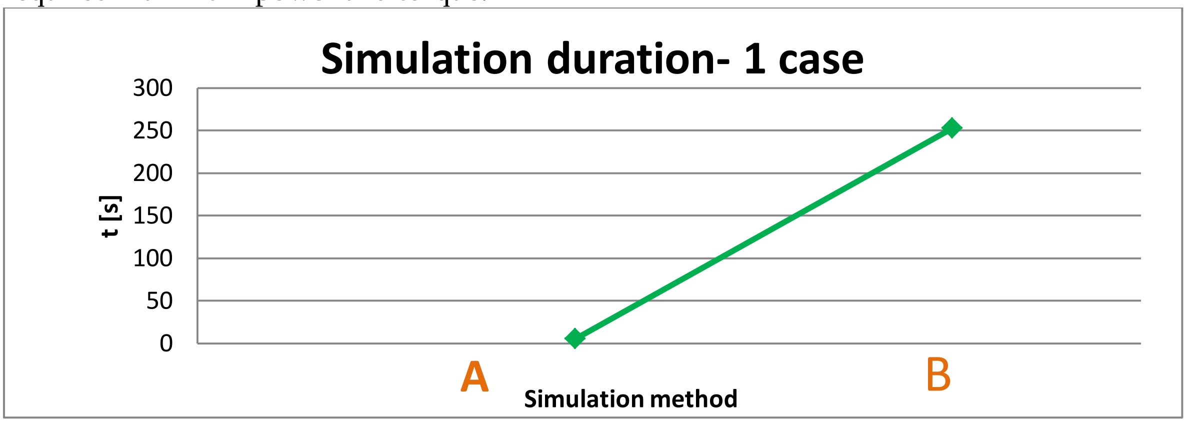
2. Turbocharged SI twin cylinder engine for vehicles

2.1. Engine modeling methods

The simplest method of engine simulation is a 0 D (zero dimension) one. This method solves basic thermodynamic equations (e. g., continuity and energy conservation equations). I used this method for initial matching of a turbocharger to the engine. I programmed this method in MS Excel. The solution is very quick.

The next simulation method is 1 D simulation. This method solves partial differential equations for every part of engine systems in 1D (manifold, cylinder). Last method of engine simulation is 3D. This method is usually applied for certain parts of an engine, because 3D method is very CPU time-demanding.

The goal of 0-D and 1-D simulations was finding a suitable turbocharger covering required maximum power and torque.



2.1.1. A- 0D simulation, B- 1D simulation

2.2.The goal of simulation

The next target was to introduce the boost pressure control of a turbocharger. I had two possibilities, either waste gate by-passing or the application of variable turbine geometry. A waste gate by- passes exhaust gas along a turbine if its mass-flow rate is too big. If the boost pressure exceeds the pre-defined limit the waste-gate valve opens decreasing the turbine power. The variable turbine geometry has movable stator turbine blades making the change of turbine nozzle area controllable. This solution is rather complicated and expensive, especially at high operation temperature, typical for spark ignition engines. This temperature requires special material for a turbine stator blades and shroud. Therefore, I preferred the use of waste gate control in the current cas.

2.3.0D simulation of engine

I programmed this method in MS Excel for these basic equations.

$$bmep = \eta_e \cdot \eta_{pl} \cdot \frac{Hu}{(Z + \lambda \cdot Lt)} \cdot \frac{\varepsilon}{\varepsilon - 1} \cdot \frac{p_{in}}{r_{in*} T_{in}}$$

$$p_{in} = \frac{bmep \cdot (Z + \lambda \cdot Lt) \cdot (\varepsilon - 1) \cdot r_{in*} T_{in}}{\eta_e \cdot \eta_{pl} \cdot Hu \cdot \varepsilon}$$

2.3.1. Equation of brake mean effective pressure

$$\dot{m}_{in} = \eta_{pl} \cdot \frac{p_{in}}{r \cdot T_{in}} \cdot V_Z \cdot i_v \cdot \frac{rpm}{120}$$

2.3.2. Equation for Engine MFR (mass flow rate)

$$\dot{m}_t = A_{ref} \frac{p_{0,T1}}{\sqrt{r_T T_{0,T1}}} \sqrt{\psi(\pi_T)} \quad \pi_T = \frac{p_{0,T1}}{p_{T2}}$$

$$A_{ref} = \frac{\dot{m}_t \cdot \sqrt{r_T T_{0,T1}}}{p_{0,T1} \cdot \sqrt{\psi(\pi_T)}}$$

$$\psi(\pi_T) = \left\langle \begin{matrix} \frac{2\kappa_{out}}{\kappa_{out} - 1}\left(\pi_T^{\frac{-2}{\kappa_{out}}} - \pi_T^{\frac{-1-\kappa_{out}}{\kappa_{out}}}\right) \\ \kappa_{out}\left(\frac{2}{\kappa_{out} + 1}\right)^{\frac{\kappa_{out}+1}{\kappa_{out}-1}} \end{matrix} \right. \quad by \left\langle \begin{matrix} \pi_T \le \left(\frac{\kappa_{out} + 1}{2}\right)^{\frac{\kappa_{out}}{\kappa_{out}-1}} \\ otherwise \end{matrix} \right.$$

2.3.3. Equation for turbine mass flow rate. Moreover, engine MFR equals approximately to both compressor and turbine MFR

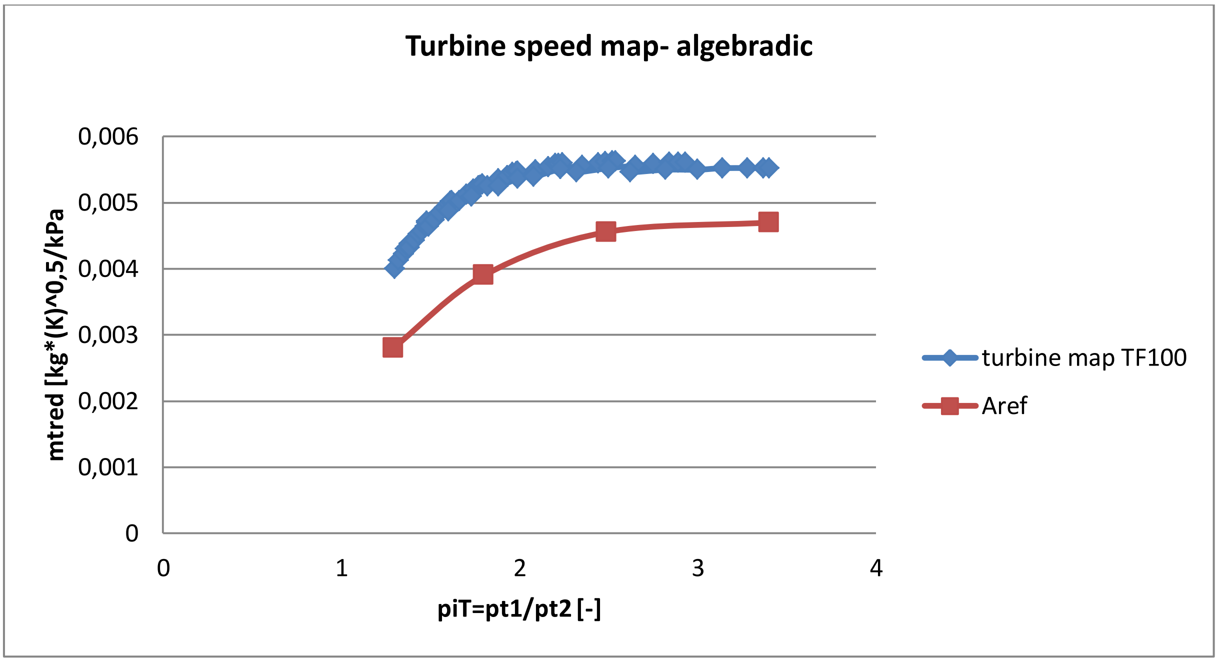
$$P_c = P_T \cdot \eta_{m,TC}$$

$$\dot{m}_c \cdot c_{p,in} \cdot T_{C1} \cdot \left[\left(\frac{p_{C2}}{p_{C1}} \right)^{\frac{\kappa_{in}-1}{\kappa_{in}}} - 1 \right] \cdot \frac{1}{\eta_{cs}} = \dot{m}_T \cdot c_{p,out} \cdot T_{T1} \cdot \left[1 - \left(\frac{p_{T2}}{p_{T1}} \right)^{\frac{\kappa_{out}-1}{\kappa_{out}}} \right] \cdot \eta_{TS} \cdot \eta_{m,TC}$$

$$p_{T1} = \frac{p_{T2} \cdot ((\dot{m}_T \cdot c_{p,out} \cdot T_{T1}) + 1)^{\frac{\kappa_{out}-1}{\kappa_{out}}} \cdot \eta_{TS} \cdot \eta_{m,TC}}{- \dot{m}_c \cdot c_{p,in} \cdot T_{C1} \cdot \left[\left(\frac{p_{C2}}{p_{C1}} \right)^{\frac{\kappa_{in}-1}{\kappa_{in}}} - 1 \right] \cdot \frac{1}{\eta_{cs}}}$$

2.3.4. Rateau equation is a balance between compressor and turbine power (Tt1 can be estimated)

The result is a reference area for turbine (2.3.3). The reference area of turbine is 1,181 cm^2.

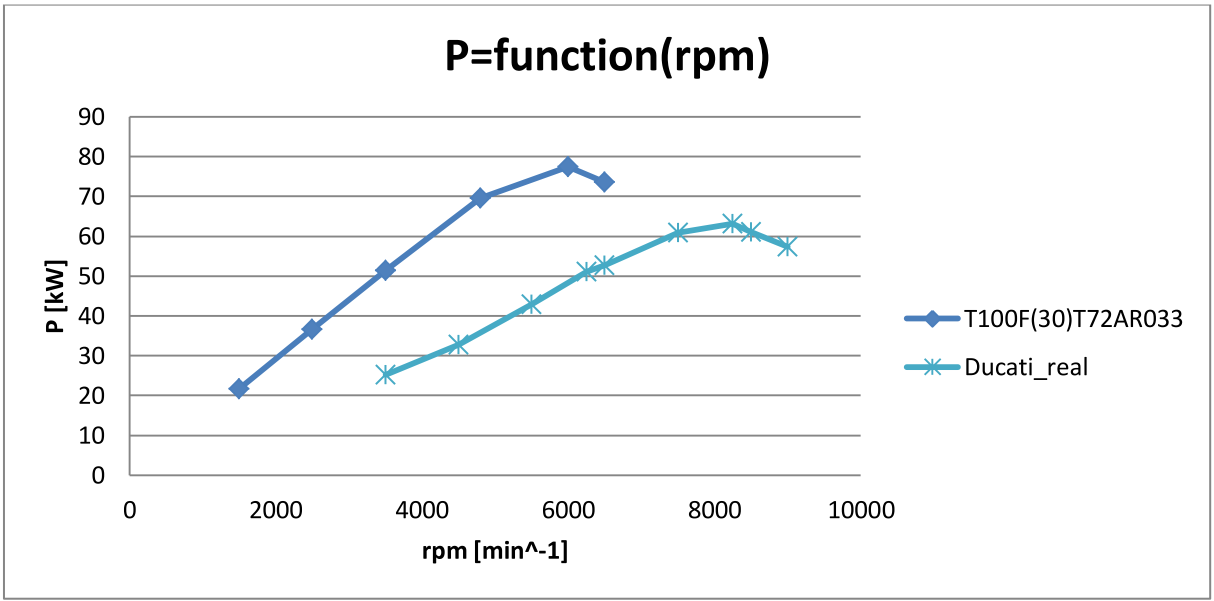


2.3.5. Comparison of maps for the smallest available Garret turbine TF100 and turbine map required according 0 D model simulation

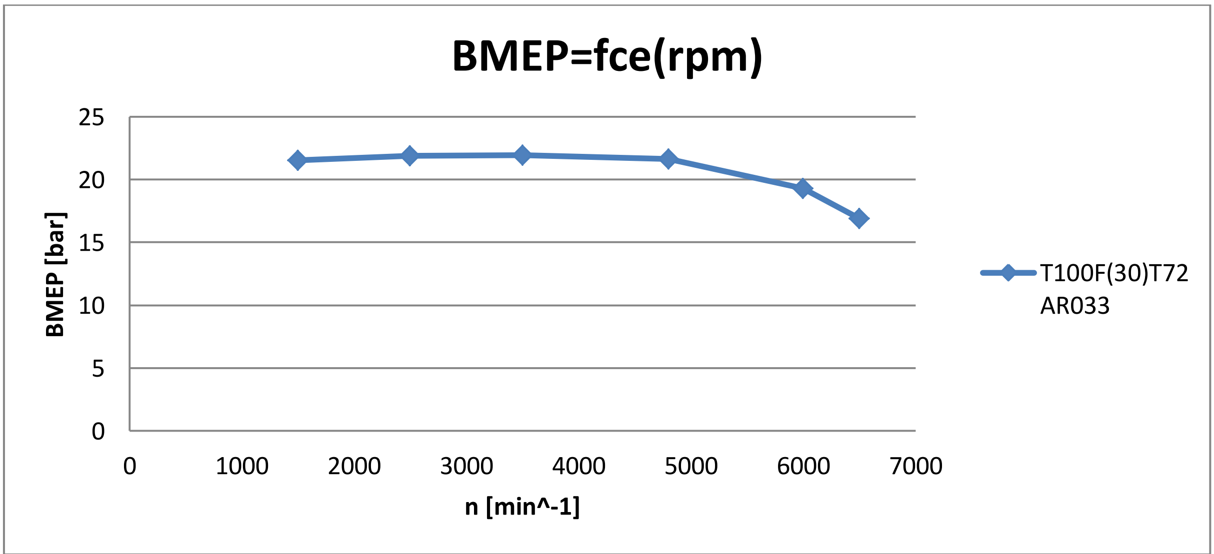
The result (2.3.5.) shows that the real turbine Garret TF100 is too big for this engine. The final matching of a turbine was done according to the results from the 1D model (see below). The TF100 turbine was reduced by -10%, which is in reasonable accordance with the preliminary 0 D estimate.

2.4.1D simulation of engine

I modeled twin cylinder engine in software GT-Power. This software solves 1 D unsteady partial differential equations for manifolds and cylinders. The finite volume method is used for manifolds divided into small sections. Engine cylinder is described by ordinary differential equations for several zones with different temperature and gas composition under the same pressure. The goal of this simulation was power 80 kW @ 6500 min^{-1} and brake mean effective pressure was 22 bar @ 1500- 4800 min^{-1}. I optimized valve timing, manifold dimensions (length, diameter). Due to lack of suitably small turbochargers at the market, I rescaled mass-flow rate coordinate of available compressor and turbine maps. The figure (2.4.1.) shows the comparison of naturally aspirated and boosts versions of the engine.

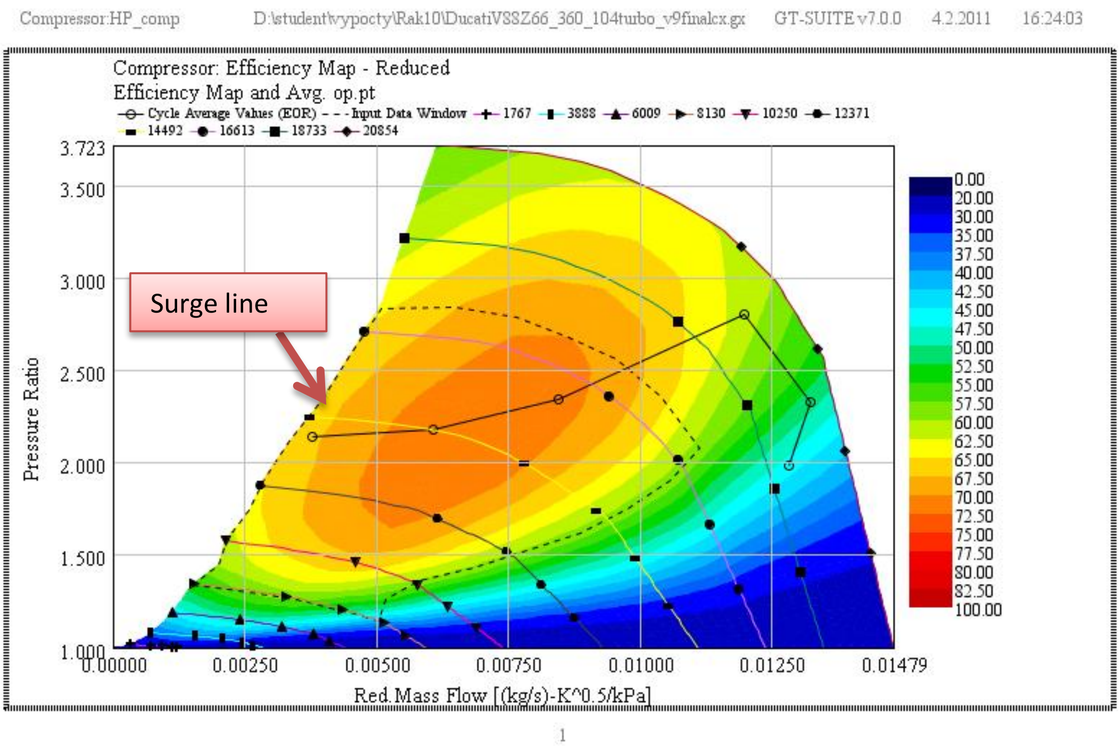


2.4.1. Comparison of naturally aspirated and boost version of engine



2.4.2. bmep- indicated mean effective pressure

I met the target with reasonable peak power difference of 2.5 kW. The bmep achieves at least 22 bar @ 1500- 4800 $\min^{-1}$. This good torque performance is based on the careful optimization of a boosting system for this twin-cylinder engine. The specific attention had to be paid to increasing turbine power reduced by small exhaust mass flow rate at low engine speed. Due to small engine displacement, two cylinders are used only. Then, the exhaust pulse interval is as much as 360° of crank angle, i.e., highly pulsating flow. It decreases turbine mean efficiency and causes significant turbocharger speed non-uniformity, especially at low engine speed. This effect is combined with the general trend to compressor surging at low engine speed.

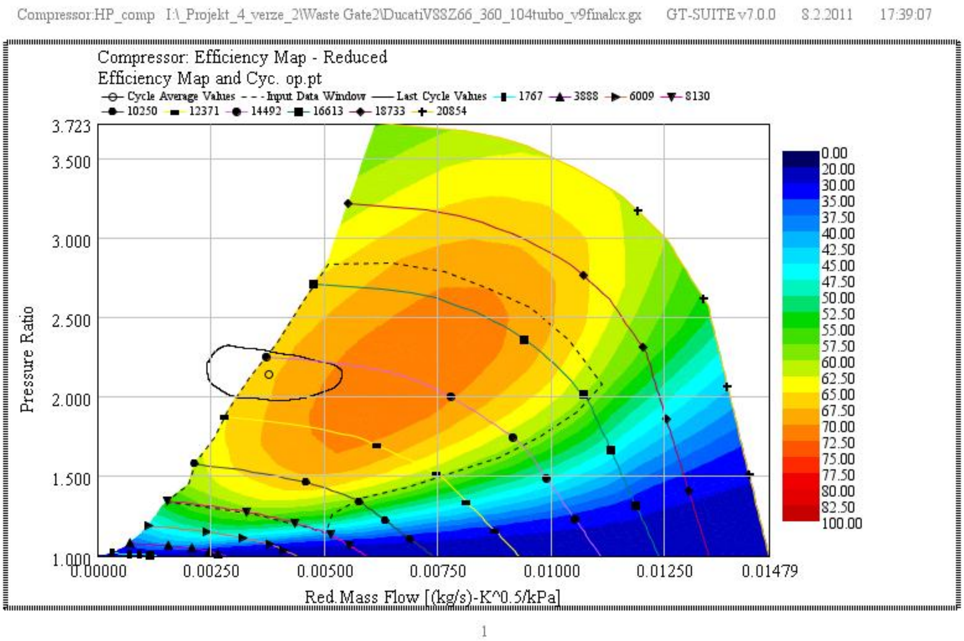


2.4.3. Compressor map

Compressor map (2.4.3.) shows the averaged values for several operating points. The instantaneous values (MFR, speed, pressure ratio) feature pulsations in the vicinity of mean values. The range of required MFR for the brake mean effective pressure of 22 bar is rather broad, causing the both dangers of surging and choking.

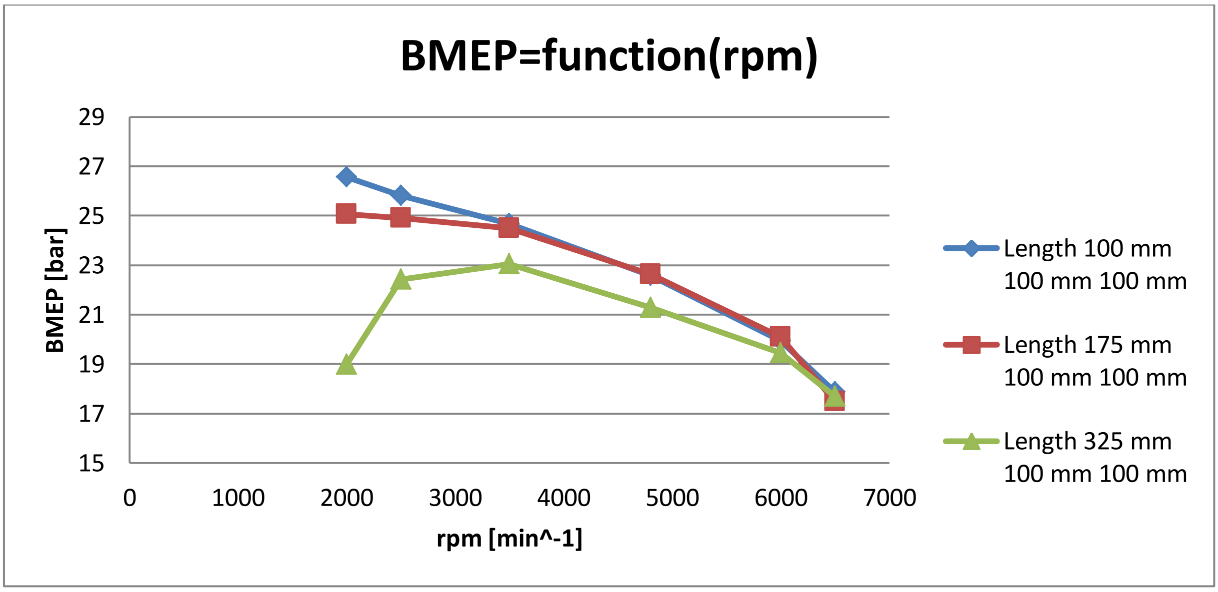
2.4.4. Integrated compact exhaust manifold (*this picture is for four- cylinder engine*)

The length of exhaust manifold is very important for any engine. In this case, the shortest exhaust manifold is proved to be the best one. The disadvantage of this solution is very high turbine inlet temperature, especially in the case of SI engine and stoichiometric A/F ratio. The temperature of gases may reach up to 1400°C. The suitable limit of temperature for the reasonable automotive materials is 1050°C today, which was respected in manifold length design.

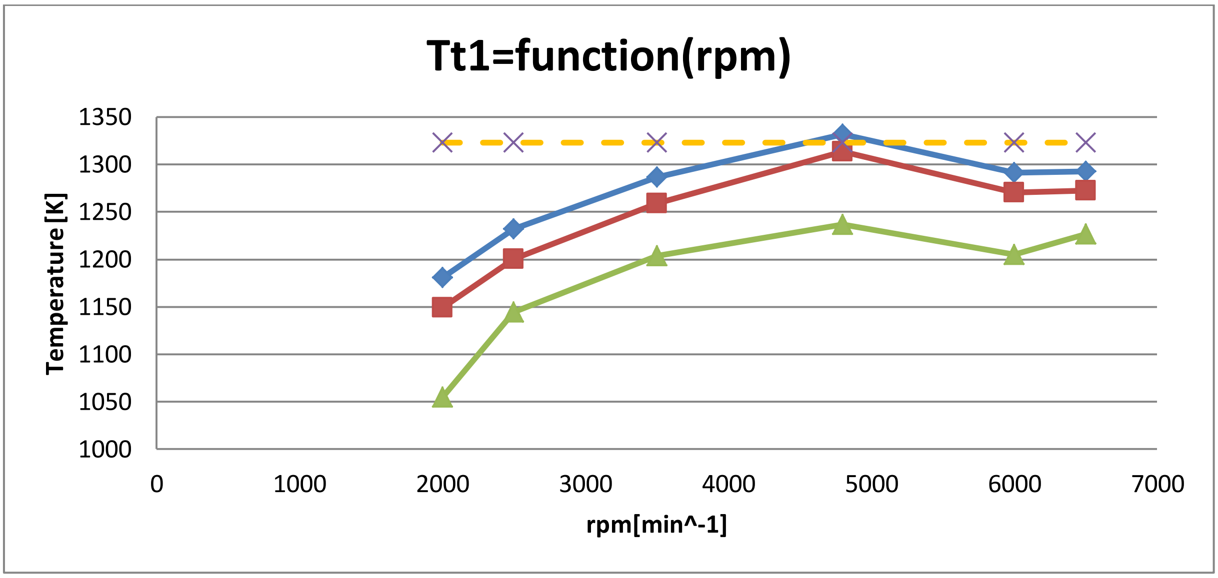


2.4.5. Pulsation of operating point @ 1500 min-1

I investigated three layouts of exhaust manifolds with length of 100 mm, 175 mm and 325 mm. The 100 mm exhaust manifold get over the temperature limit. The 325 mm has not reached required bmep. The best variant is length of 175 mm.

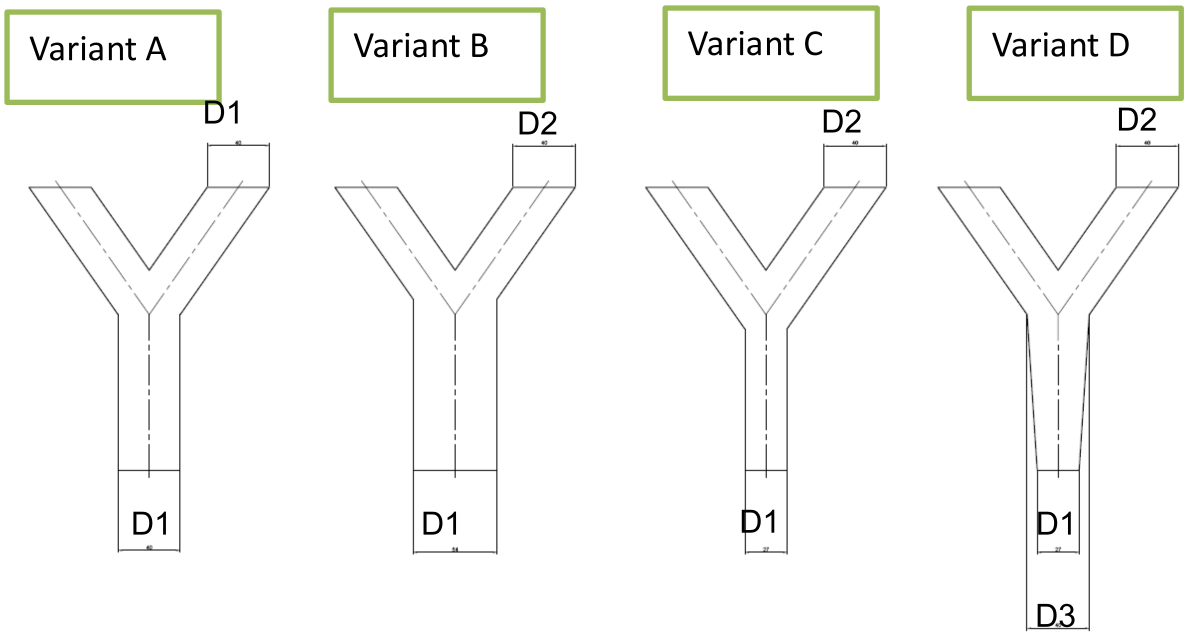


2.4.6. Exhaust manifold length optimization



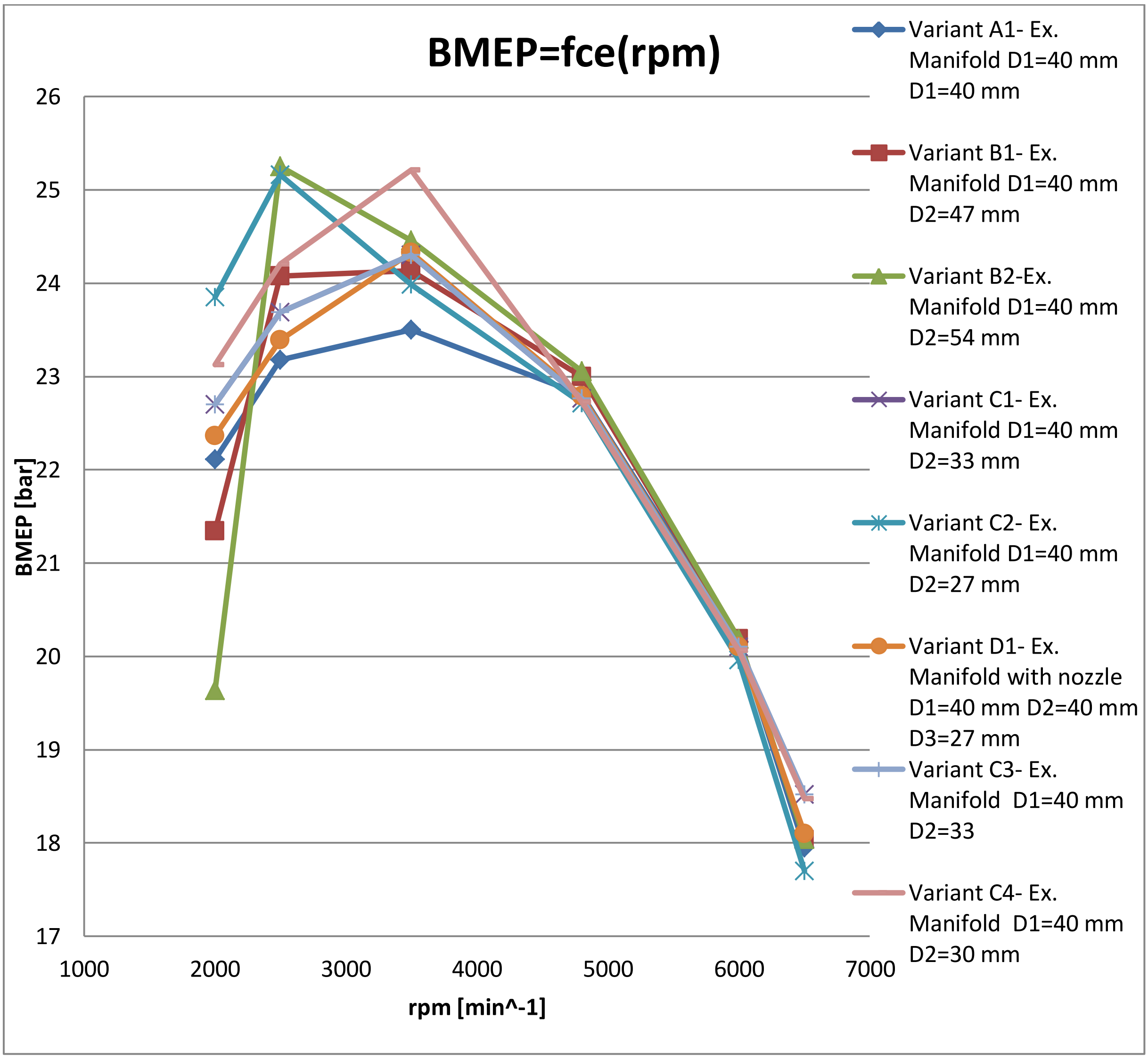
2.4.7. Exhaust manifold length optimization (*yellow line is a temperature limit in Kelvin degrees*)

The diameter of exhaust manifold determines level of pulsations in exhaust manifold. Small diameter produces higher pulsations and better transient and low-speed operation. I compared in 1 D model four designs of exhaust manifold (2.4.8.).



2.4.8. Exhaust manifold- design optimalization

In the variant A was used same diameter for inlet and outlet of exhaust manifold. The variant B acts as a diffusor unlike the variant C accelerating flow upstream of a turbine. The variant D extends the pressure pulse using integrated nozzle upstream of a turbine, which breaks the flow from a cylinder runner.



2.4.9. Exhaust manifold- design optimization

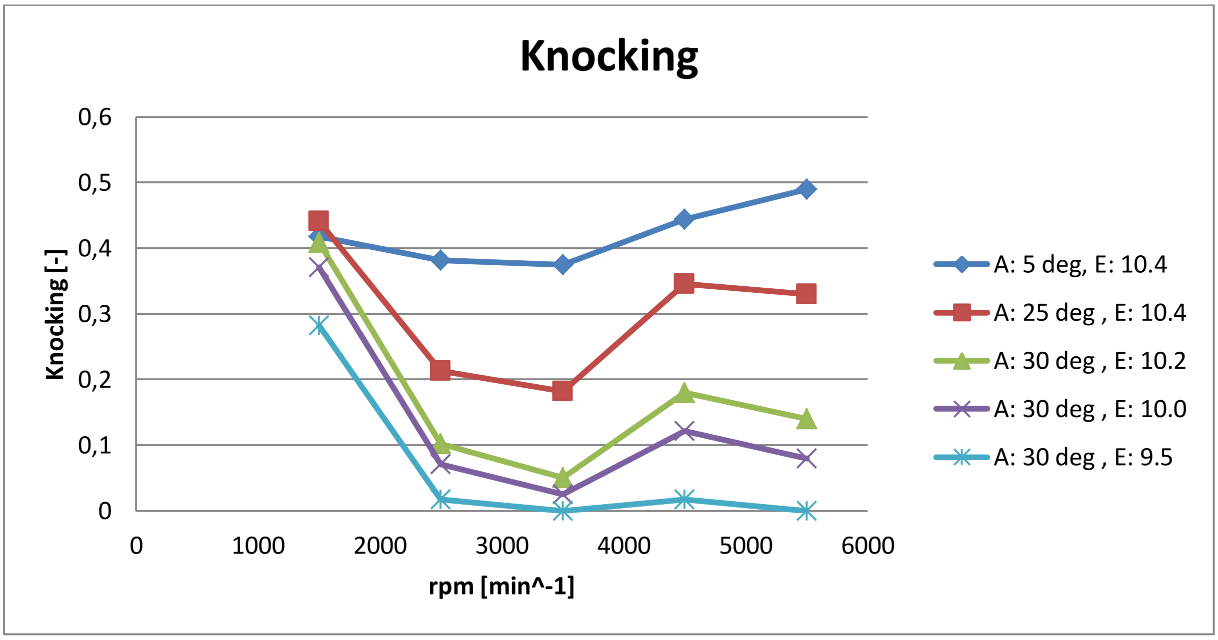
The basic variant of exhaust manifold is A1. The bmep in the variant A1 is not good at low rpm. The variant B is not effective at low rpm, too. The best variant is C4 (nozzle) with diameter 30 mm. The special nozzle (variant D) increases bmep at low rpm. The pressure increase is about 6%.

2.5. Knocking

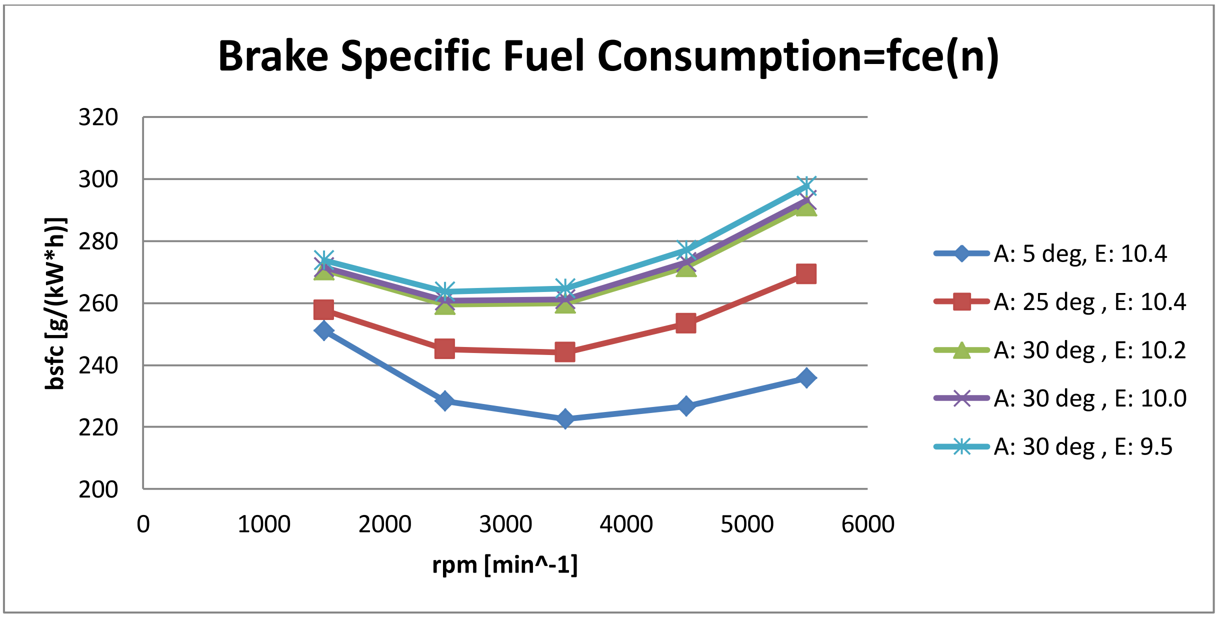
Knocking is a negative phenomenon. The unburned fuel/air mixture burns in short duration, detonation may occur. The detonation breaks a cylinder wall boundary layer, causing surface overheating. The increased temperature and pressure of unburned mixture reduces auto-ignition time. The measures for increasing engine efficiency or power (e.g., compression ratio, boost pressure level) increase danger of shortest exhaust manifold, too.

The simulation predicted the percentage of auto-ignited unburned mixture. The knocking model was derived from real measured data, unfortunately at much greater SI engine. The results are accurate at least qualitatively. As it is well-known, the prediction based on greater engine overshoots the danger of knocking. The model development should be focused on this issue in the future.

Maximum knocking percentage for regular operation of an engine should not exceed 10%. To avoid knocking, the delayed ignition advance (i.e., the shift of 50% ROHR anchor angle) offers the initial remedy but in some cases, compression ratio or even engine torque (i.e., boosting pressure) have to be reduced. Lower compression ratio decreases total efficiency of an engine. Lower total efficiency increases fuel consumption (Figure 2.5.2.).

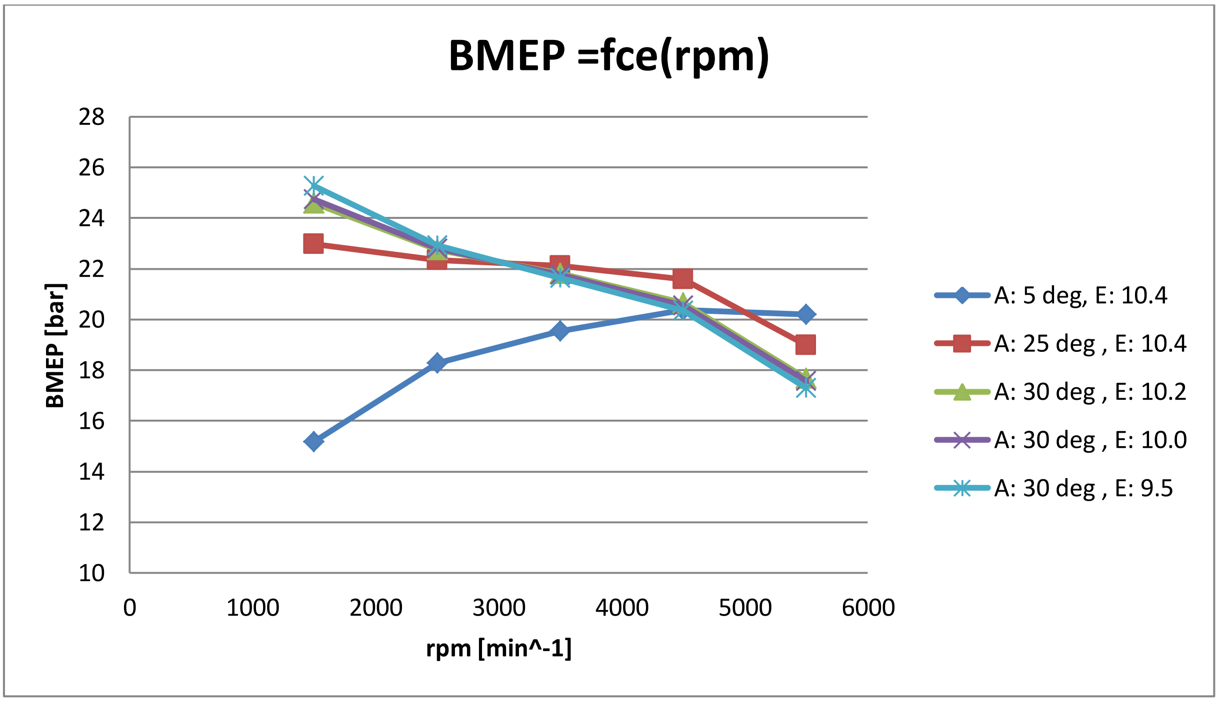


2.5.1. Knocking percentage mass (A: *ROHR anchor angle (defined by 50% burnt mass)*, E: *Compression ratio*)



2.5.2. Fuel consumption

The critical rpm for engine knocking are 1500 min-1. Decreasing compression ratio does not help without additional delay of combustion. It is reflected by bmep. WOT curve, as well (Figure 2.5.3.).

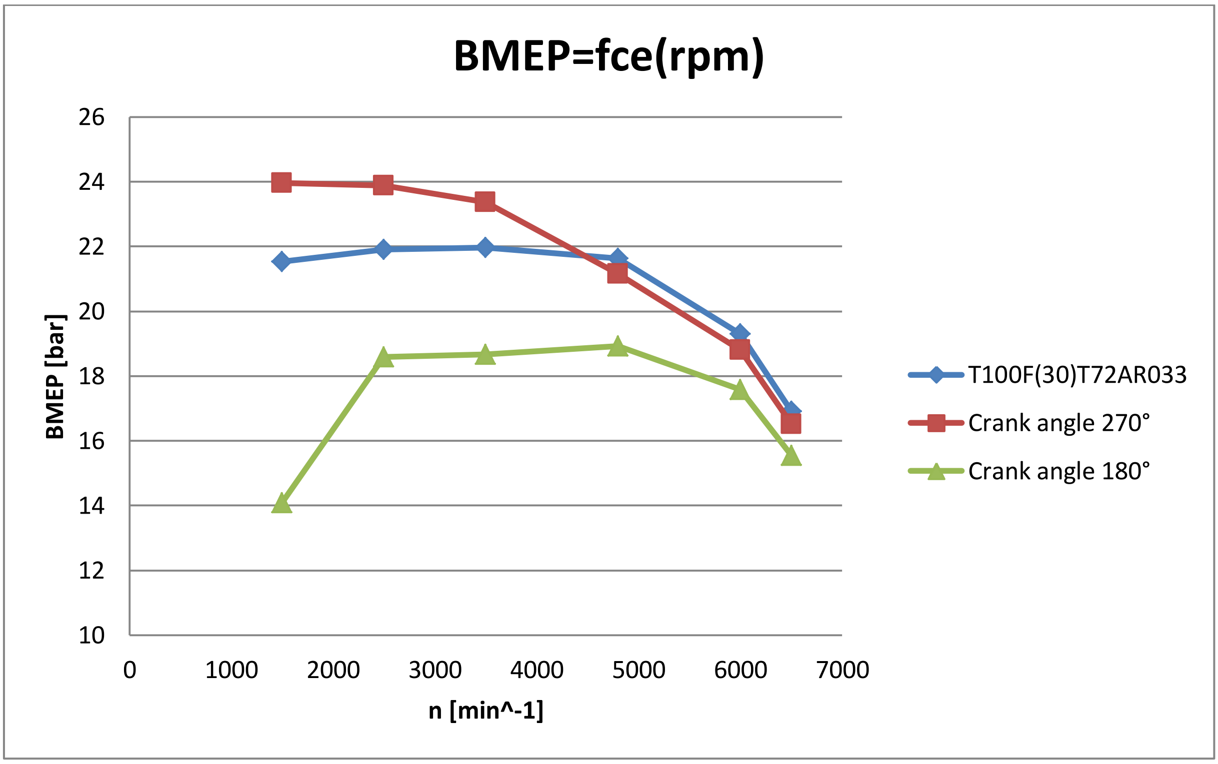


2.5.3. bmep= function rpm (A: *ROHR anchor angle (defined by 50% burnt mass)*, E: *Compression ratio*)

2.6. Investigation of crankshaft alternative layouts

The regular interval between pressure and torque pulses at single crank (360° crank angle) for twin cylinder in-line engine features some disadvantages, as well. The main issue is the balancing of reciprocating inertial forces, which are just coupled in this case. The addition of balancing shaft(s) poses the same problems as in the case of single cylinder engine.

The competitive solutions are either 270° crank and/or ignition angle or flat twin engine with 180° ignition angle. The flat boxer with 360° would be ideal lay-out (combination of regular pulses with balanced inertial forces) but still too expensive. The pioneer of 270° design was Yamaha TRX 850 in the field of motorcycles. This solution was used for reduction of vibrations and sound of the 90° V-twin engine (this sound imitation is not favorable for vehicles). The crank angle of 270° has some advantages considering balancing, as well. Generally, the 270°/450° spacing of exhaust pulses yields advantages for boosting at low speed. The IMEP was increased by 11 %. Unlike it, the pulse spacing of 180°/540° has only disadvantages for boosting and it does not reduce vibrations.



2.6.1. Comparing between crank angle 360° (*blue line*), crank angle 270° (*red line*), crank angle 180° (*green line*)

2.7. Conclusions

The model used for further twin cylinder engine optimization was successfully calibrated. Between measured and calibrated Ducati engine WOT speed curves is 3,1 Nm (inaccuracy of 1D GT Power model) maximum torque difference. The twin cylinder engine reaches power range of 35 – 80 kW, which makes it applicable in naturally aspired or turbocharged version for small cars or for hybrid vehicles as range extender. Boosted version of 60 kW seems to be an optimum version of twin cylinder engine for vehicles. 60 kW version does not suffer from knocking and compressor surging at low rpm. The boosted version of 80 kW features some problems with knocking and compressor surging at low rpm, which can be mitigated by the careful engine optimization. This engine rated parameters can be supported by using twin turbo system or variable geometry turbine. Still the 80 kW boosting version has higher cost and higher fuel consumption (lower compression ratio). For boosting versions is very important correct design of exhaust manifold. The best solution is small length and nozzle system. For boosting at low rpm is better 270° crank angle. Crank angle 360° is compromise for boosting and mechanical balancing of engine.

The market for this engine are small cars (e.g. Fiat Panda, VW Up)

bmep	[Pa]	Brake mean effective pressure
IMEP	[Pa]	Indicated mean effective pressure
rpm	[min-1]	Engine speed
WOT	[-]	Wide open throttle
A/F	[-]	Air to fuel ratio
bsfc	[g/(kW*h)]	Brake specific fuel consumption
$\dot{m}_{in}$	[kg/s]	Input mass flow rate
$\dot{m}_c$	[kg/s]	Compressor mass flow rate
$\dot{m}_t$	[kg/s]	Turbine mass flow rate
Hu	[kJ/kg]	Calorific value of fuel
ε	[-]	Compression ratio
λ	[-]	Relative air-to-fuel ratio
L_t	[-]	Stoichiometric air-to-fuel ratio
V_z	[cm^3]	Displacement of cylinder
i_v	[-]	Number of cylinders
η_e	[-]	Brake efficiency
η_{pl}	[-]	Volumetric efficiency
Z	[-]	Constant=1 for multi point injection
T_{in}	[K]	Air temperature – inlet manifold
T_c	[K]	Air temperature- compressor inlet
p_{in}	[Pa]	Inlet pressure
p_{C1}	[Pa]	Compressor inlet pressure
p_{C2}	[Pa]	Compressor outlet pressure
π_C	[-]	Compressor pressure ratio

η_C	[-]	Compressor isentropic efficiency
p_{T1}	[Pa]	Turbine inlet pressure
p_{T2}	[Pa]	Turbine outlet pressure
π_T	[-]	Turbine pressure ratio
η_T	[-]	Turbine efficiency
η_m	[-]	Mechanical efficiency of turbocharger
A_{ref}	[cm2]	Reference surface of turbine
c_p	[J/(kg*K)]	Specific heat capacity at constant pressure
c_v	[J/(kg*K)]	Specific heat at constant volume
κ_{out}	[-]	Poisson constant- exhaust
κ_{in}	[-]	Poisson constant- intake
r_{in}	[J/(kg*K)]	Gas constant for air