



**VEHICLE CENTER OF
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MOBILITY**

Simulation of Biogas Engine Concept with Separated Air and Gas Compression

Report for PBS Turbo s.r.o.

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1. Introduction

This report describes development of numerical model of turbocharged industrial biogas engines. The engine layout reflects demands on formation of mixture in high pressure intake manifold downstream of compressors. Separate compressors are used for compressing air and biogas. Two engine variants were considered. First variant contains high pressure mixer and second variant contains high pressure solenoid operated gas admission valves in intake ports directly upstream of engine head. These models were used for specification of optimal turbochargers sizes for electric generator power demand.

1.1. Engine variants

Demanded engine power was 1100 kW / 1500 min⁻¹. Engine cylinder size was based on the Jenbacher engine type 4 and 6. Brake mean effective pressure was 18 bar. The rest of parameters were based on experience with similar engines. Mixture formation in high pressure manifold requirement is to prevent intake manifold explosion in case of backfire. For low pressure mixture formation is necessary to install flame arresters which negatively influence engine total efficiency by its pressure drops. Standard biogas stations combine biogas production line with several biogas cogeneration units running on biogas with relatively small compensation reservoir. The biogas is not pressurized enough for high pressure mixing devices (pressure should be higher than 5 bar). This is one of the reasons why biogas cogeneration units are almost solely equipped by low pressure mixer upstream of compressor. External compressor for biogas would negatively influence biogas station total efficiency by its power input. The selected solution uses two parallel turbochargers with separated compression of pure air and pure biogas.

There are two variants of high pressure mixture formation described in this report, high pressure mixer and high pressure gas admission valves.

2. Numerical simulation of engine thermodynamic cycle

2.1. Engine model simulation

Numerical simulation model of combustion engine was used for turbochargers specification and for optimization of control algorithms. Engine model was created in software GT-Power v7.4 from Gamma Technologies. This software combines standard 0-D description of processes in engine cylinder, turbine and compressor with 1-D description of processes in pipes. Basic engine parameters are listed in *Tab.1*.

Nominal power [kW]	1000	1100
Nominal engine speed [min^{-1}]	1500	1500
Engine layout	R 6	V 16
Bore [mm]	190	145
Stroke [mm]	220	185
ε [1]	14	14
Intake system	boosted, with intercooler	
Brake mean effective pressure [bar]	21.38	18
Fuel	biogas (60% CH_4 , 40% CO_2)	
Number of intake valves	2	
Number of exhaust valves	2	

Tab.1 – Basic engine parameters

The engine uses one a standard turbocharger. The engine intake and exhaust manifolds were modified to use two parallel turbochargers. Turbines were connected in parallel but compressors have separated pipes for pure air and pure biogas. Turbochargers will be designated as air turbocharger and fuel turbocharger. Intercoolers are used also two separated for air and fuel. The mixture formats in high pressure mixer or by gas admission valves directly upstream of engine head. Engine layouts are displayed in *Fig. 1* and *Fig. 2*.

Dimensions of engine parts, intake and exhaust manifolds were based on the authors' experience with similar engines. Valvetrain system was derived from an engine with similar dimensions and the same type of fuel. Valvetrain timing was set to have almost no overlap to have no scavenging and no fuel leakage to exhaust. Valvetrain uses Miller-type of valve timing with early intake valve closing and prolonged effective expansion stroke.

Air and fuel intercooler was modeled as a set of small pipes with constant wall temperature. Intercooler cooling efficiency and pressure loss was set with respect to real intercoolers characteristics.

The engine model was not calibrated to measured data because the engine exists only in concept phase. Nevertheless, the engine model was calibrated on the basis of authors' experience with similar engines. Engine model gives good results for turbocharger specification and boosting concept evaluation.

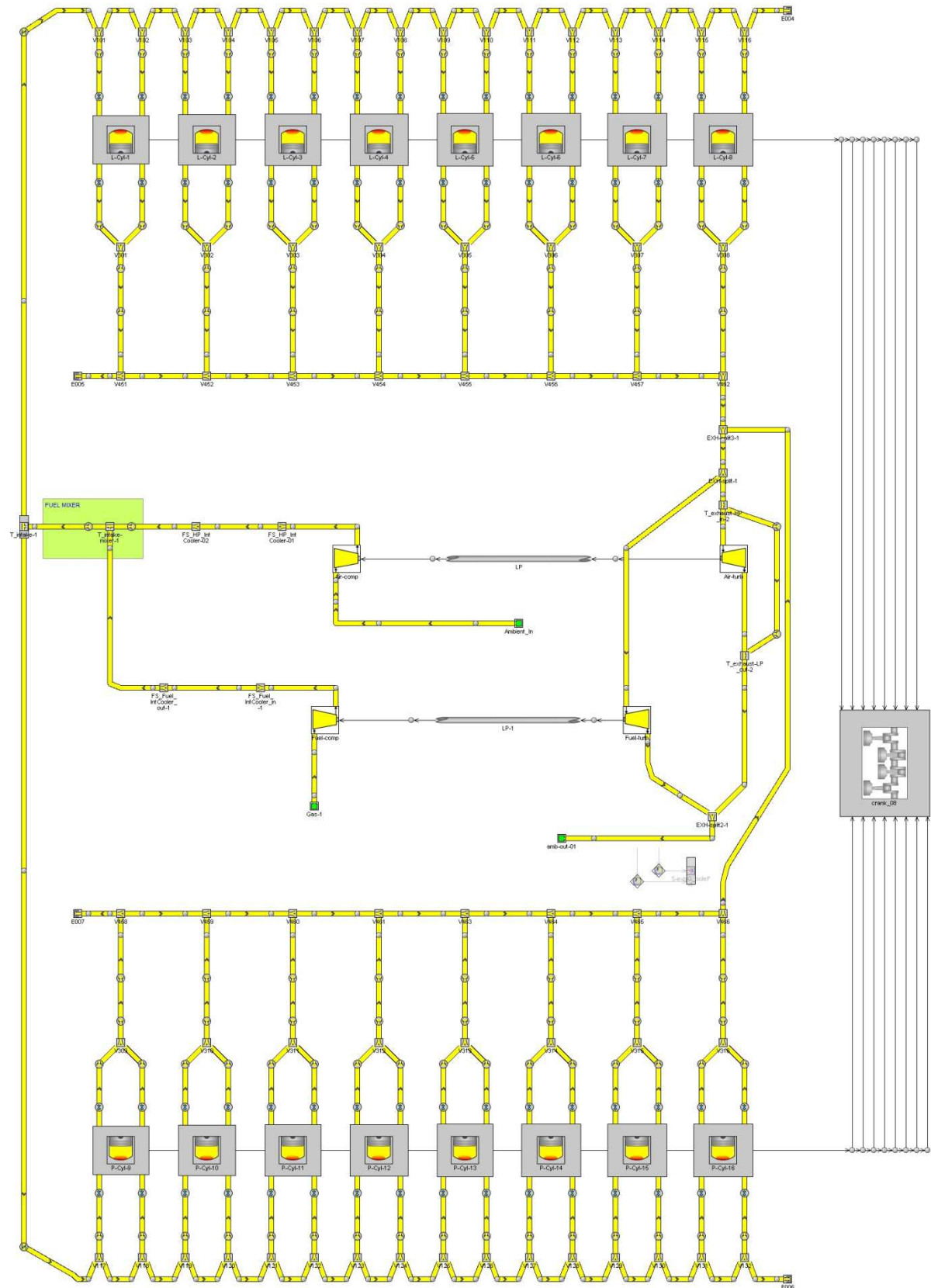


Fig. 1 – Engine model with high pressure mixer in software GT-Power

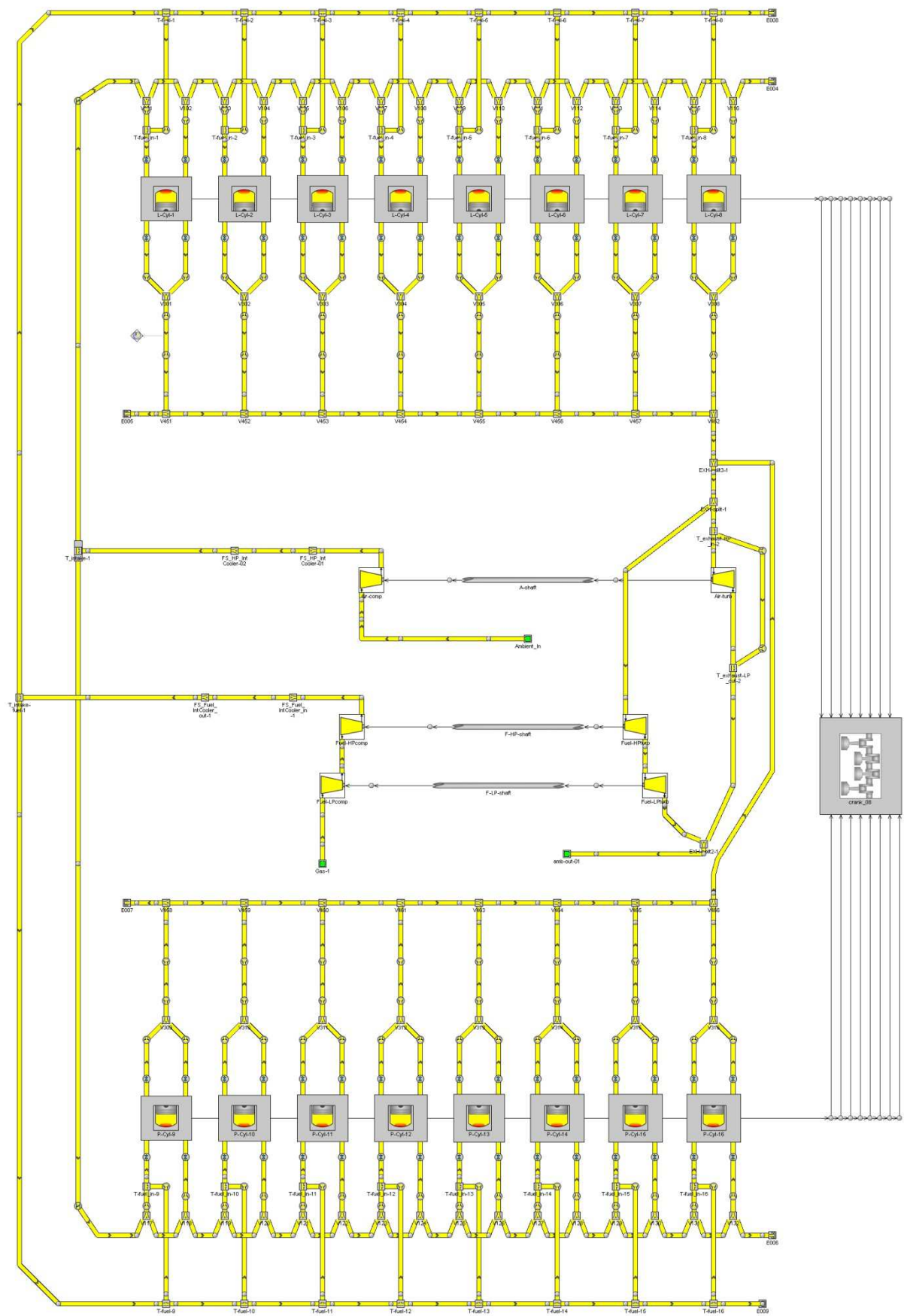


Fig. 2 – Engine model with gas admission valves in software GT-Power

2.1.1. Engine valvetrain

Design of valves and their flow characteristics were based on similar engine with approximately the same brake mean effective pressure. There are four valves in the engine head. Their discharge coefficients were the same for intake and exhaust valves. There are displayed in *Fig. 3*.

Valve lift profiles were also used from similar engine. The timing of valves was adjusted to keep valve overlap at minimal level since mixture leakage to exhaust manifold is undesirable due to safety and economic reasons. The Miller-type cycle with prolonged expansion was used. Intake valve closing angle was set to keep volumetric efficiency approximately at 70% related to manifold conditions at nominal engine speed and power.

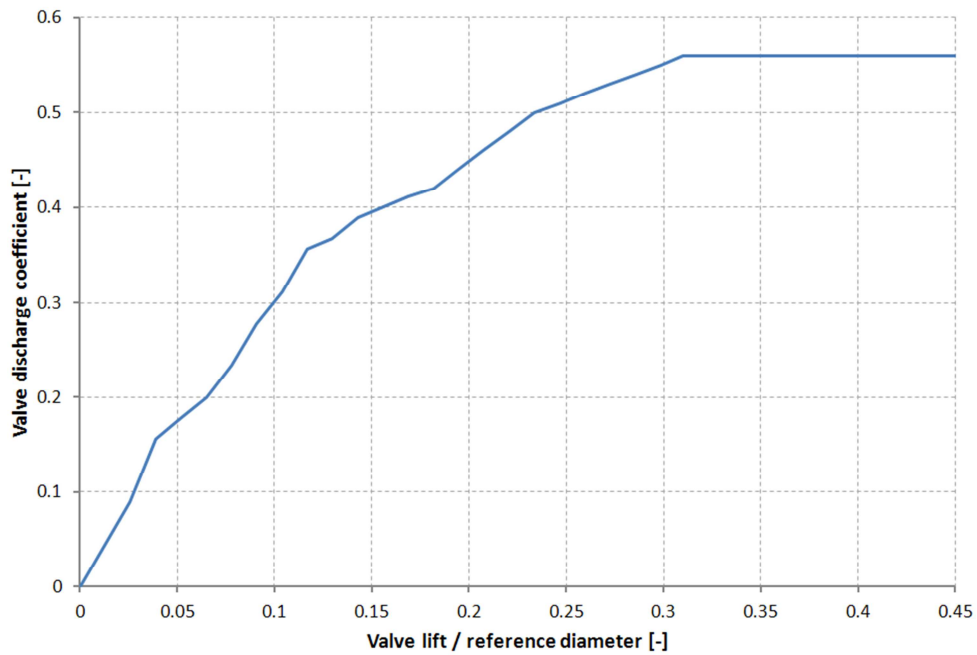


Fig. 3 – Intake and Exhaust valves discharge coefficient in dependence on valve lift over reference lift diameter

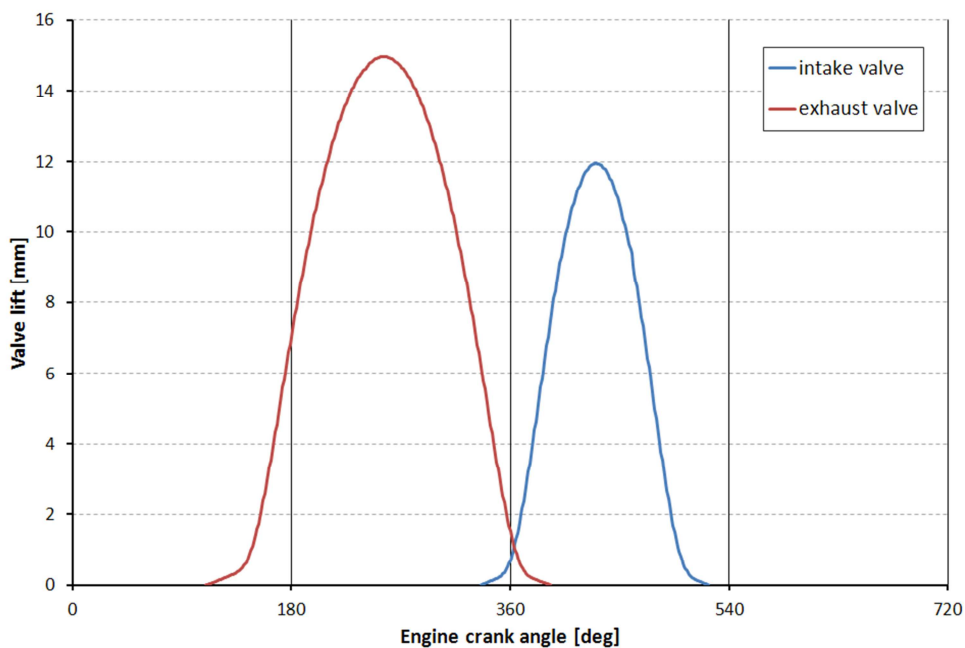


Fig. 4 – Intake and Exhaust valve lift profiles

2.1.2. Biogas

The biogas composition depends on various parameters and it is not constant in time. The main fractions are methane (CH_4) and carbon dioxide (CO_2). The constant biogas composition was chosen for the simulation purposes. Fuel composition is listed in *Tab.2*.

	Methane CH_4	Carb. Dioxide CO_2
Volumetric fraction [-]	0.6	0.4
Mass fraction [-]	0.3535	0.6465
Stoichiometric A/F ratio [-]	6.0519	

Tab.2 – Biogas composition

2.1.3. Requirement on engine power

Engine power output demand is stated in *Tab.1* and nominal speed is given by the electric generator for site frequency 50 Hz , which is 1500 min^{-1} .

2.1.4. PBS turbocharger model

PBS Turbo is a manufacturer of turbochargers for large industrial engines with high turbocharger efficiency. PBS Turbo produces turbochargers for wide range of engines used for example in trains or ships. Turbochargers maps were provided to authors for series TCR 10, TCR12, TCR 14 and TCR 16 to cover wide range of mass flow rates. All compressors and turbines could be modified in defined range of mass flow rates.

2.1.5. Control System

Engine contains several control valves actuated by control system. Control system main task is to keep air excess ratio at prescribed level. Engine output power or generator electric power is also controlled.

Turbochargers employed in parallel configuration should be chosen by their sizes since the exhaust mass flow should be divided in approximately the same ratio as required by the air to fuel ratio of mixture. Nevertheless, the configuration with gas admission valves requires fuel overpressurized above air pressure to be able to transfer fuel only during the intake stroke of each cylinder. Exhaust throttle valves installed downstream of turbines provide capability of change to turbine pressure ratio and consequently to turbine power and both exhaust gas mass flow distribution between turbines.

2.1.6. Air-biogas mixture

Pollutant production of industrial engines is limited by several regulations, for example TU Luft in Germany. There are two basic solutions. It is possible to purify the exhaust gas by applicable catalyst. Other and more common system is low pollutant production engine setting without processing of exhaust gases. The main pollutants are nitrous oxides in case of gas and biogas engines. Their production can be effectively decreased by combustion temperature reduction. It can be arranged by several modifications of engine cycle. The mixture can be diluted by the pure air or exhaust gases. Usage of lean mixture is usually sufficient precaution. It is simple and inexpensive solution limited only by ability to ignite the mixture by the ignition system. Usage of exhaust gas recirculation (EGR) could further improve nitrous oxides production.

Only lean mixture without EGR was used in our case. The air excess ratio was set to approximately 1.6 .

2.2. Preliminary turbocharger specification

The first target of simulation was to specify turbocharger sizes for required engine power. Turbocharger size needed to be specified for air and fuel sides separately according to specific engine layout. The first version of engine with nominal engine power of 1000 kW was used for this preliminary study. This engine is displayed in *Fig. 1*. High pressure mixer was chosen for its design simplicity. Control system was also relatively simple. One throttle valve was placed upstream of mixer on fuel side. PI controller was used to keep constant mixing ratio.

As a base turbocharger type was used one from the middle of product type series TCR 14 from PBS Turbo and its characteristic was extrapolated to higher speed and higher pressure ratios for the needs of this preliminary study. Turbocharger size could be adjusted to fulfil all demands for engine operation by mass flow multipliers. Turbine rack could be used to change their sizes as well. Engine should be able to reach target power from idle operation. Exhaust throttle valve downstream of turbine of fuel turbocharger was applied to set the pressure ratio of air and fuel turbines. This control ability simultaneously helps to keep sufficient turbine power of air turbocharger. If both turbocharger sizes are set properly, there will be no need for throttling downstream of turbines.

Turbocharger approximate size specification can be obtained from base engine parameters. On the other hand, finding a turbocharger size, which is optimal for given purpose, is difficult task. Universal optimization software Mode Frontier from Esteco was used to find the combination of mass flow multipliers for both turbochargers. This software enables to use several optimization algorithms and optimization procedures. Genetic optimization algorithm was used for its robustness. The optimization objective was minimal brake specific fuel consumption of designs which were able to reach defined target power. Moment of inertia of turbocharger of this size is quite high. Level of brake mean effective pressure, the use of Miller-type cycle and fuel mixture with high air excess ratio lead to the need of high boost pressure. It brings high requirements on turbochargers. Transient ability of the engine was verified by setting simulation model intake and exhaust manifold initial conditions to atmospheric condition. The engine simulation is, in fact, transient load from idle to nominal power at constant engine speed, as it is displayed in *Fig. 6*. Layout of the optimization procedure in Mode Frontier is displayed in *Fig. 5*.

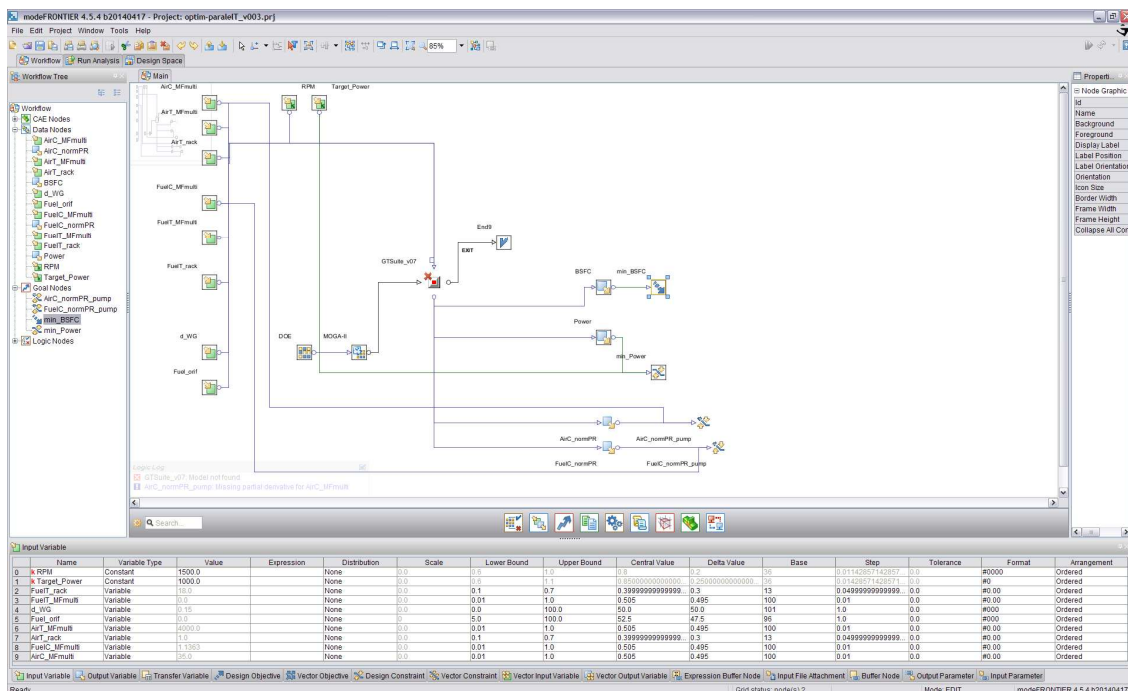


Fig. 5 – Optimization process programmed in software Mode Frontier

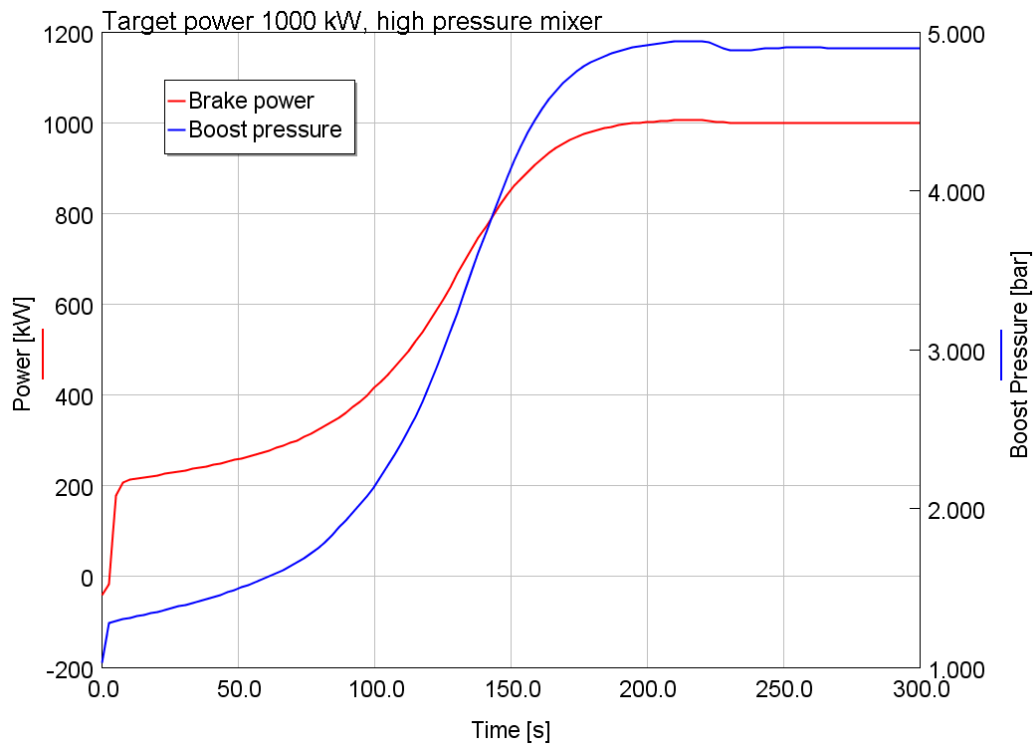


Fig. 6 – Engine power and boost pressure during transient from idle to nominal power in constant engine speed 1500 RPM

2.2.1. Optimization results – turbocharger sizes specification

The optimal turbocharger size was chosen from a Pareto set of optimization results. Base turbochargers modification by mass flow multipliers is defined by *Tab.3* while turbochargers working points are defined in *Tab.4*. All turbochargers working points are displayed in their maps presented in *Fig. 7*, *Fig. 8*, *Fig. 9* and *Fig. 10*.

Compressor	<i>Air turbocharger</i>	<i>Fuel turbocharger</i>
type	TCR14 minRCQ	TCR14 minRCQ
mass flow multiplier	0.72	0.07
Turbine	<i>Air turbocharger</i>	<i>Fuel turbocharger</i>
type	TCR14	TCR14
rack position	0.45	0.4
mass flow multiplier	0.63	0.08

Tab.3 – Turbocharger type and size specification

		<i>Air turbocharger</i>		<i>Fuel turbocharger</i>	
		<i>Compressor</i>	<i>Turbine</i>	<i>Compressor</i>	<i>Turbine</i>
Pressure ratio	-	5.08	4.59	5.02	4.00
Reduced mass flow rate	(kg/s)-K ^{0.5} /kPa	0.2218	0.0836	0.0241	0.0103
Mass-averaged efficiency	%	74.7	75.6	66.0	77.6

Tab.4 – Turbocharger working points specification

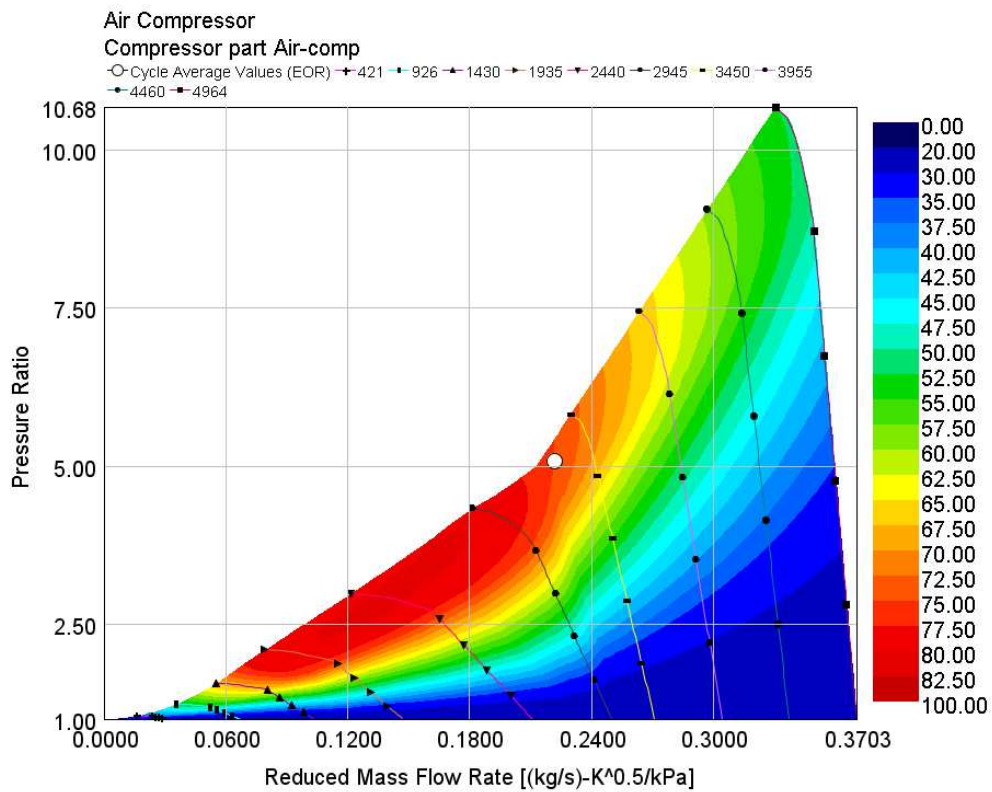


Fig. 7 – Working point (white circle) displayed in compressor map of air turbocharger

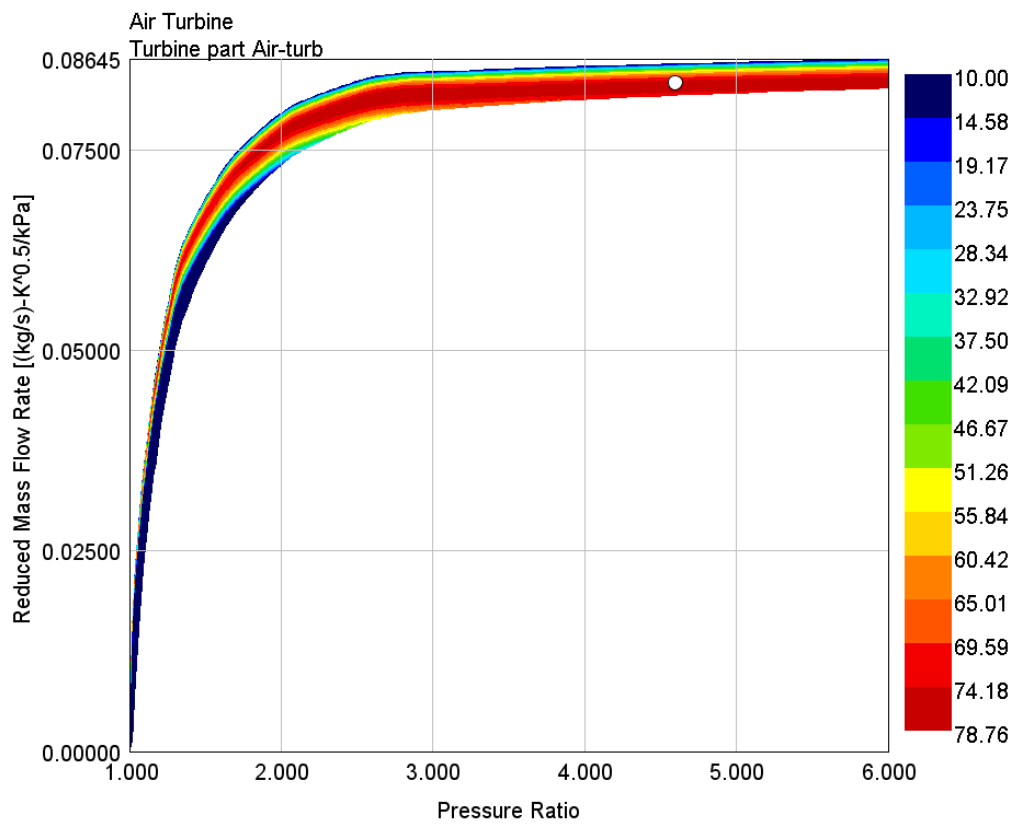


Fig. 8 – Working point (white circle) displayed in turbine map of air turbocharger

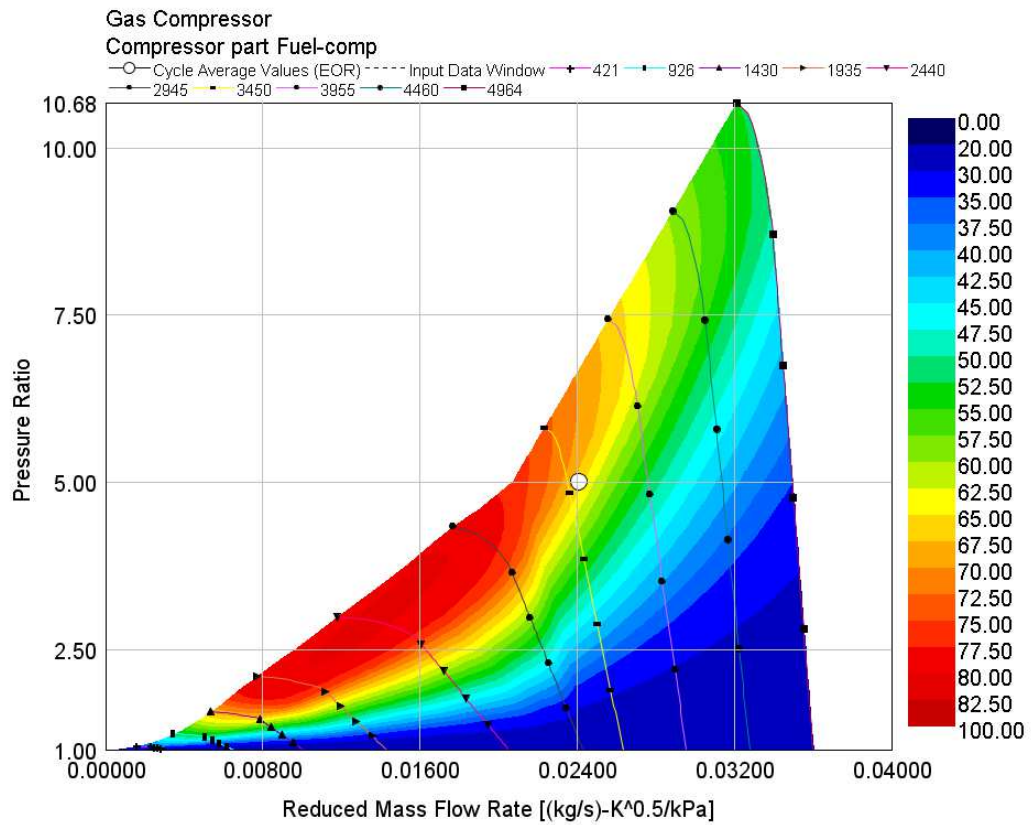


Fig. 9 – Working point (white circle) displayed in compressor map of fuel turbocharger

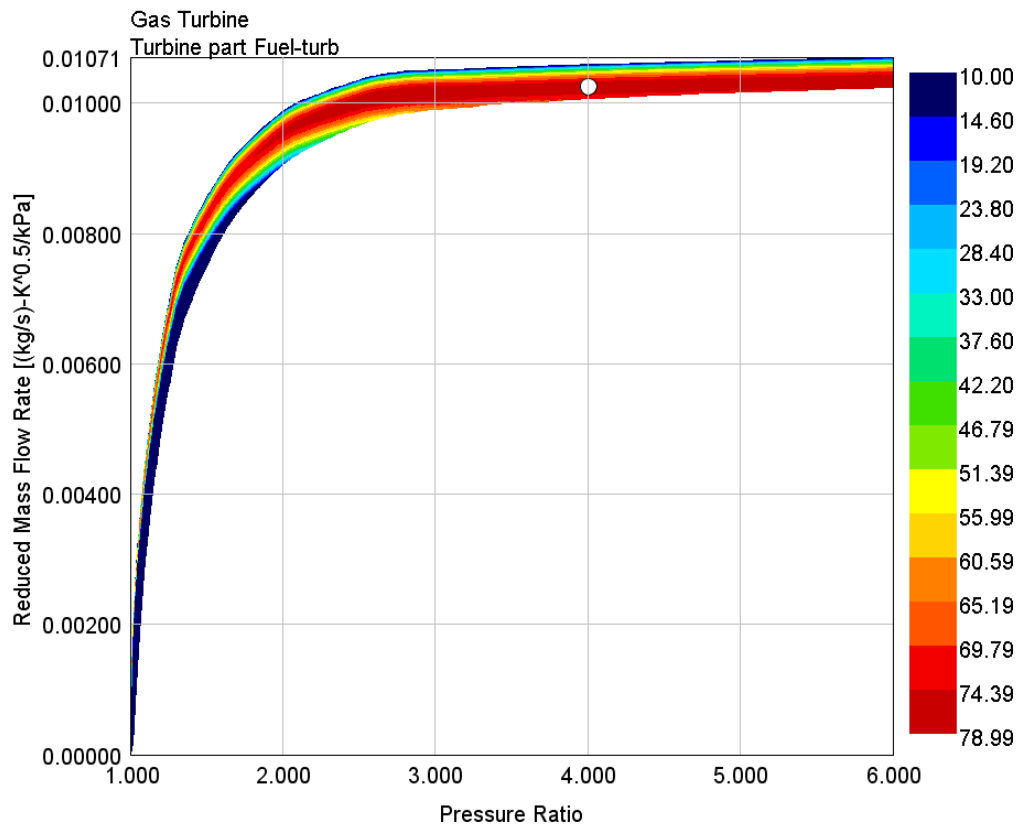


Fig. 10 – Working point (white circle) displayed in turbine map of fuel turbocharger

2.3. Turbocharger production type specification

The next step in turbocharger size specification is to choose compressor and turbine maps which will have working points specified in *Tab.4* placed in a zone of best efficiency.

Matching of working point with existing PBS Turbo turbocharger maps showed the following results. Air turbocharger mass flow rate corresponds to one member of type series TCR 14. Air turbocharger pressure ratio is above the capabilities of PBS Turbo turbochargers. Fuel turbocharger mass flow rate is too low to match with any of PBS Turbo turbochargers. Smaller type of turbocharger has to be used as a fuel turbocharger. Consequently, the turbocharger from ČZ Strakonice a.s. was selected. Turbochargers for automotive application do not reach such high pressure ratios. Therefore, it is necessary to couple two turbochargers in series.

Required boost pressure of the first engine configuration is at the limit of one stage boosting unit even without separated air and fuel compression. The decision was made to change the engine for this study. It was necessary to decrease brake mean effective pressure of the engine to reach lower boost pressure level. Another engine model was based on engine manufactured by Jenbacher company. Main parameters of the second engine are specified in *Tab.1*. This engine has larger displacement. Nevertheless, despite of increased engine power, the BMEP is lower. The V16 engine uses also one boosting unit for all cylinders.

2.3.1. Turbocharger specification for the second engine

Main geometrical and output parameters of both engines are similar. Therefore, turbocharger working points from *Tab.4* were used to choose proper turbocharger type. Several variants of PBS Turbo turbochargers were tested as an air turbocharger.

ČZ turbochargers were used in serial connection with even distribution of their pressure ratios. Simulation of hot gas closed loop stand was used to choose suitable compressor for turbine. This model is displayed in *Fig. 11*. Normalized turbine blade speed ratio was kept in vicinity of one to keep proper turbine load by compressor from the efficiency point of view. Turbocharger moment of inertia was set to values typical for turbocharger of its size. Moment of inertia of fuel turbochargers was several times lower in comparison to air turbocharger. This fact helps to control the air excess ratio because fuel pressure is increasing faster than air pressure. Several variants of combination of ČZ turbochargers were tested. They belong to C12 and C13 production type for high pressure stage and K27 for low pressure stage.

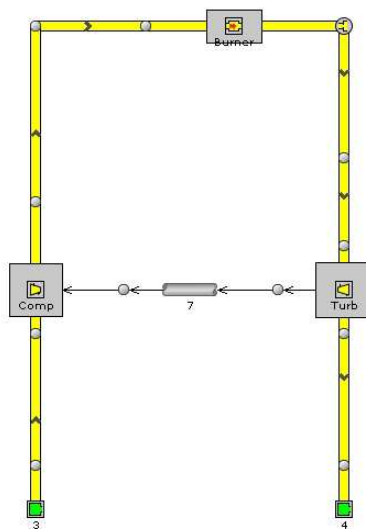


Fig. 11 – Hot gas closed loop stand simulation model

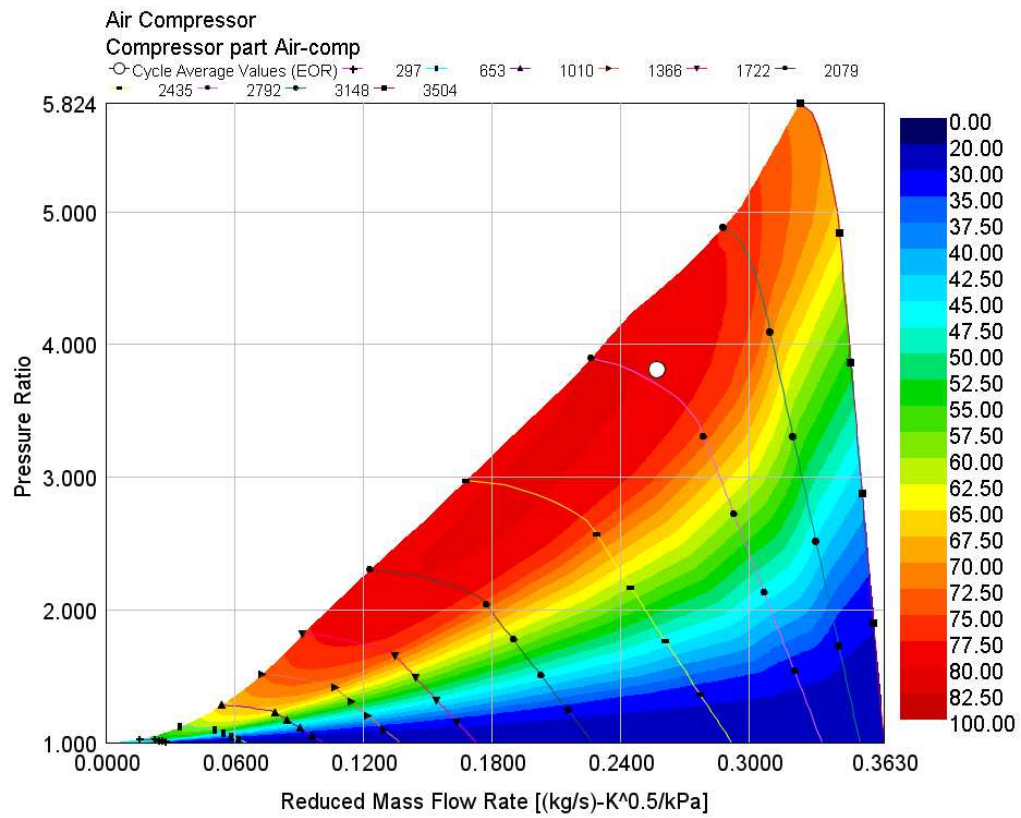


Fig. 12 – Working point (white circle) displayed in compressor map of air turbocharger

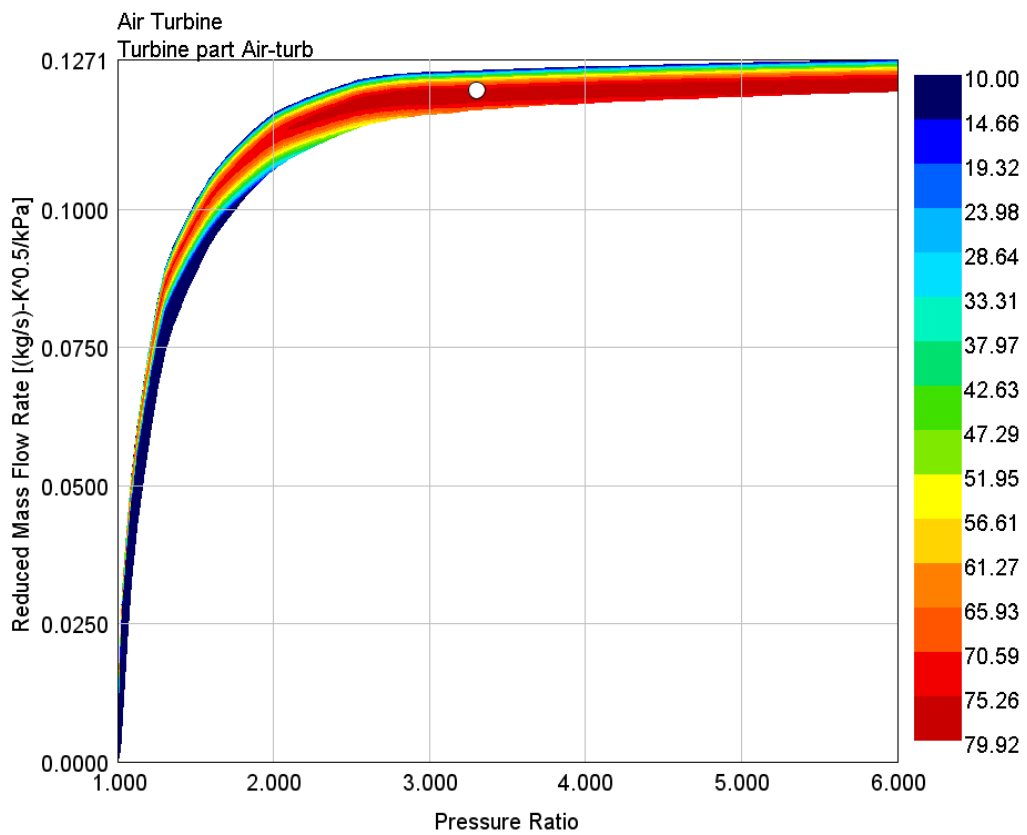


Fig. 13 – Working point (white circle) displayed in turbine map of air turbocharger

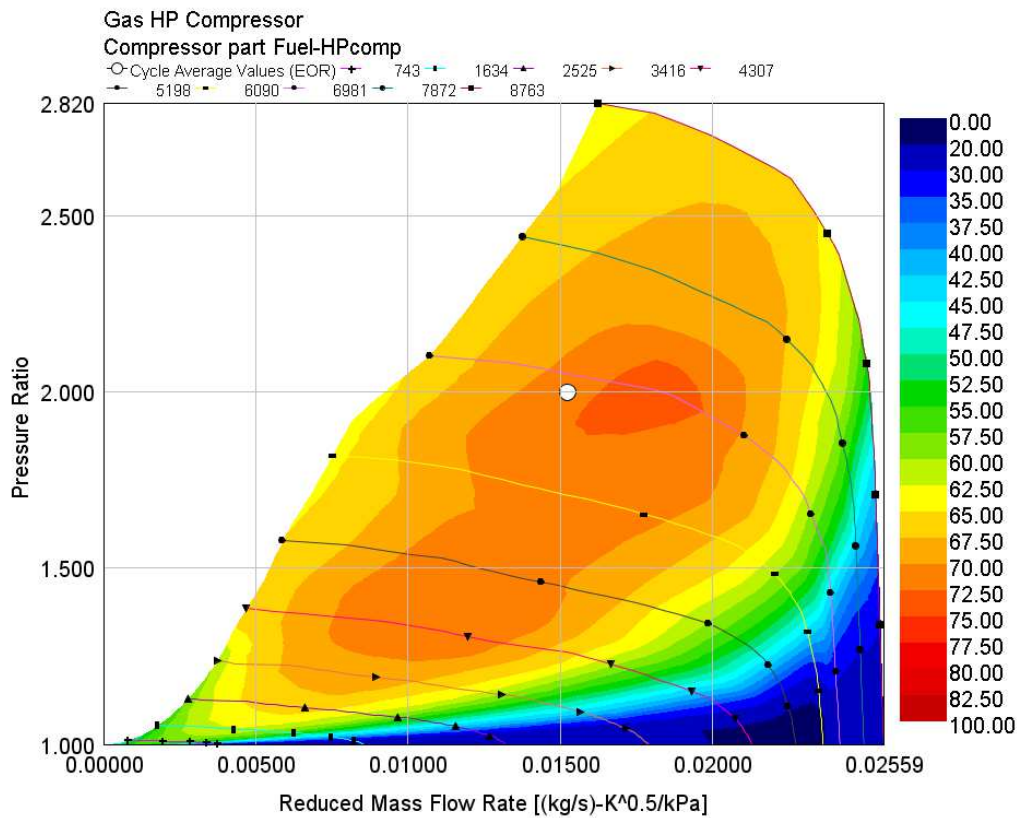


Fig. 14 – Working point (white circle) displayed in compressor map of fuel turbocharger in high pressure stage

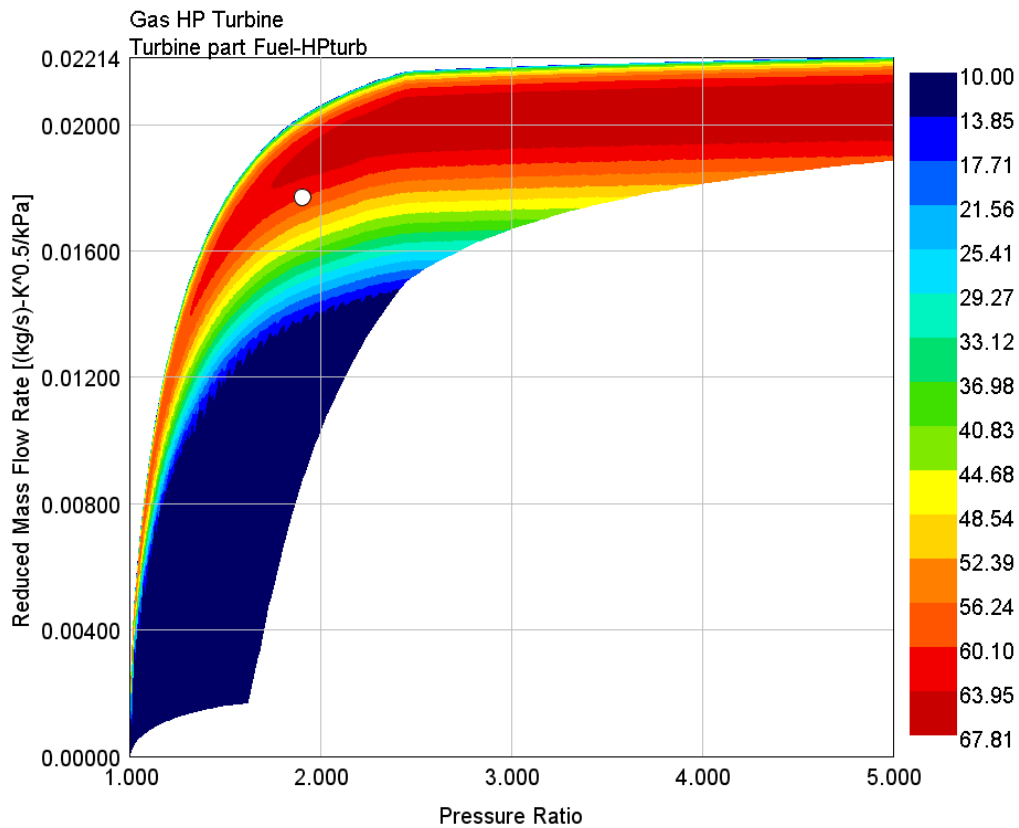


Fig. 15 – Working point (white circle) displayed in turbine map of fuel turbocharger in high pressure stage

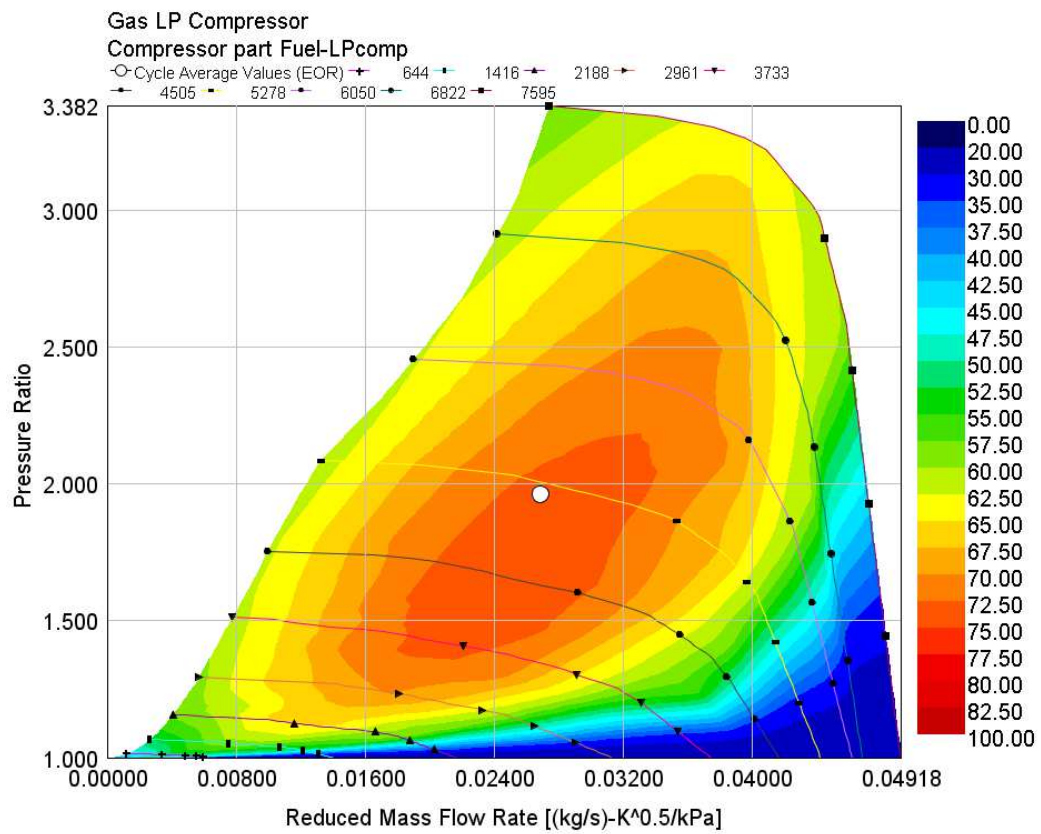


Fig. 16 – Working point (white circle) displayed in compressor map of fuel turbocharger in low pressure stage

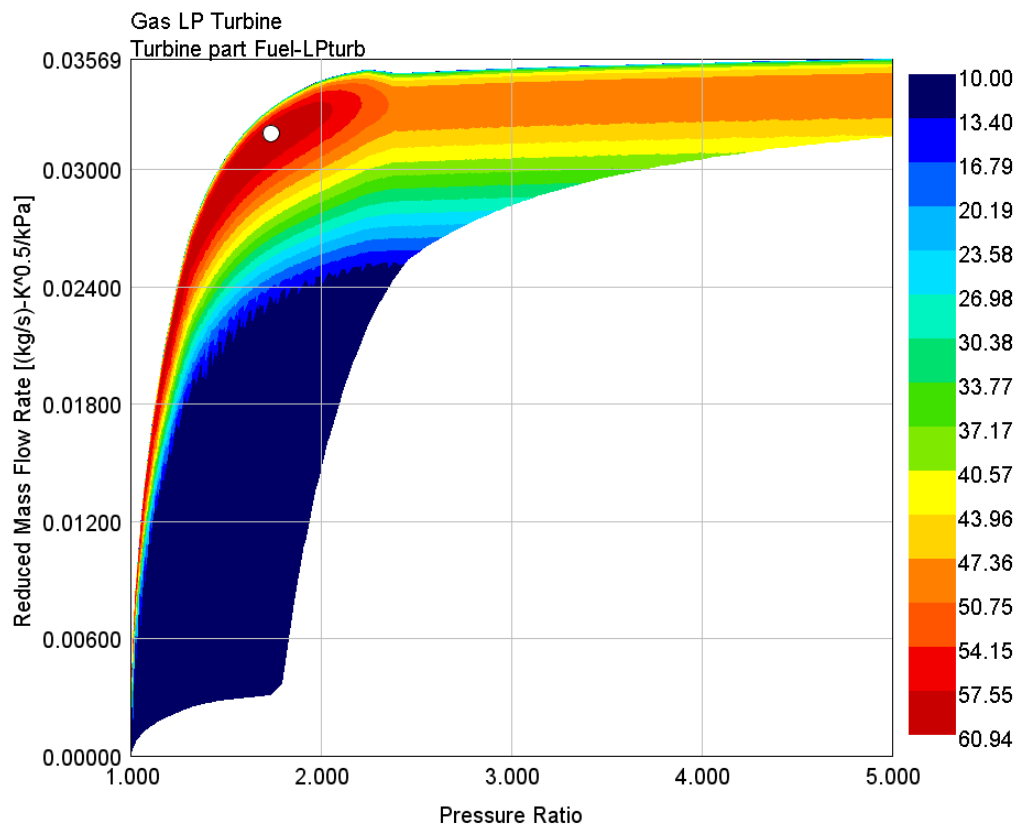


Fig. 17 – Working point (white circle) displayed in turbine map of fuel turbocharger in low pressure stage

The final combination of turbocharger types was the result of manual optimization of boosting unit with several combinations of turbochargers. They are specified in *Tab.5*. The final version was able to reach required engine power without need of additional control action to exhaust throttle flaps downstream of turbines. Air turbocharger compressor and turbine working points are displayed in *Fig. 12* and *Fig. 13*. High pressure compressor and turbine working points of fuel turbocharger are displayed in *Fig. 14* and *Fig. 15*. Low pressure compressor and turbine working points of gas turbocharger are displayed in *Fig. 16* and *Fig. 17*. All working points are defined in *Tab.6*.

	<i>Air turbocharger</i>	<i>Fuel high pressure turbocharger</i>	<i>Fuel low pressure turbocharger</i>
Compressor type	TCR14 minRCQ	C14 – 1457B	K27 – 3057G
Turbine type	TCR14	C13 – 7.12	K27 – 13.21
turbine rack position	0.4		

Tab.5 – Final turbochargers type and size specification

		<i>Air turbocharger</i>		<i>Fuel turbocharger</i>			
		<i>Compressor</i>	<i>Turbine</i>	<i>High press. Compressor</i>	<i>High press. Turbine</i>	<i>Low press. Compressor</i>	<i>Low press. Turbine</i>
Pressure ratio	-	3.81	3.30	2.00	1.90	1.97	1.73
Reduced mass flow rate	(kg/s)- $K^{0.5}/kPa$	0.2570	0.1215	0.0152	0.0177	0.0269	0.0319
Mass-averaged efficiency	%	79.6	77.3	71.5	60.1	73.6	60.2

Tab.6 – Turbochargers working points specification

Control system of engine operated with separated turbochargers is more complicated in comparison to the engine with low pressure mixer. Control system has to ensure several tasks. The first and the most important one is control of mixture air excess ratio. This is mainly done by valve in mixer. Higher or at least the same fuel pressure than air pressure is necessary for maintaining proper mixer functionality. This is ensured by exhaust gas distribution to both turbines by throttle flap downstream of turbines. Throttle flap is located downstream of fuel turbocharger. Power control of fuel turbine is convenient since only exhaust gas flow fraction, necessary for given fuel overpressure, is used. The rest of the exhaust gas flow is routed to air turbine and it can be used for boost pressure increase. Engine power was controlled by two throttles positioned in intake manifolds in both rows of engine cylinders.

The simulation started from the same initial conditions as the engine simulation described in Chapter 2.2. All important engine parameters during transient part of engine simulation are displayed in *Fig. 18*, *Fig. 19* and *Fig. 20*. The target mixture air excess ratio was kept lower for the transient cycle beginning for faster transient time.

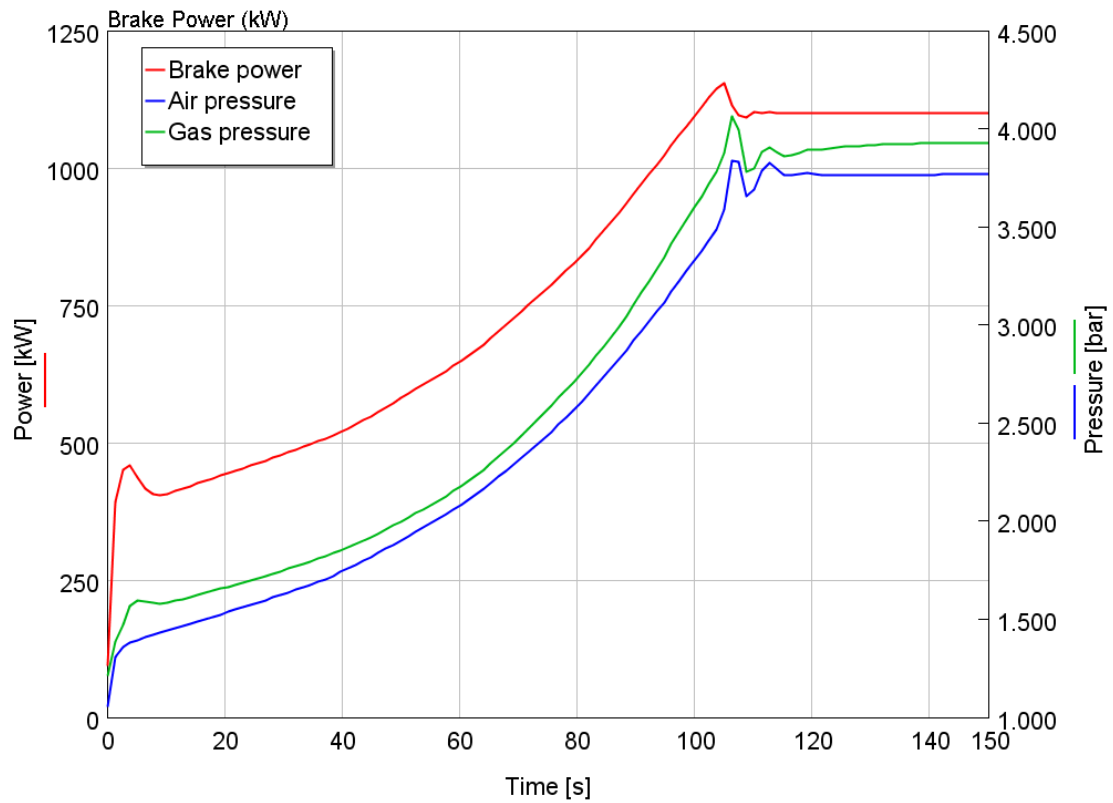


Fig. 18 – Engine power, air and fuel pressure during transient from idle to nominal power in constant engine speed 1500 RPM of engine with high pressure mixer

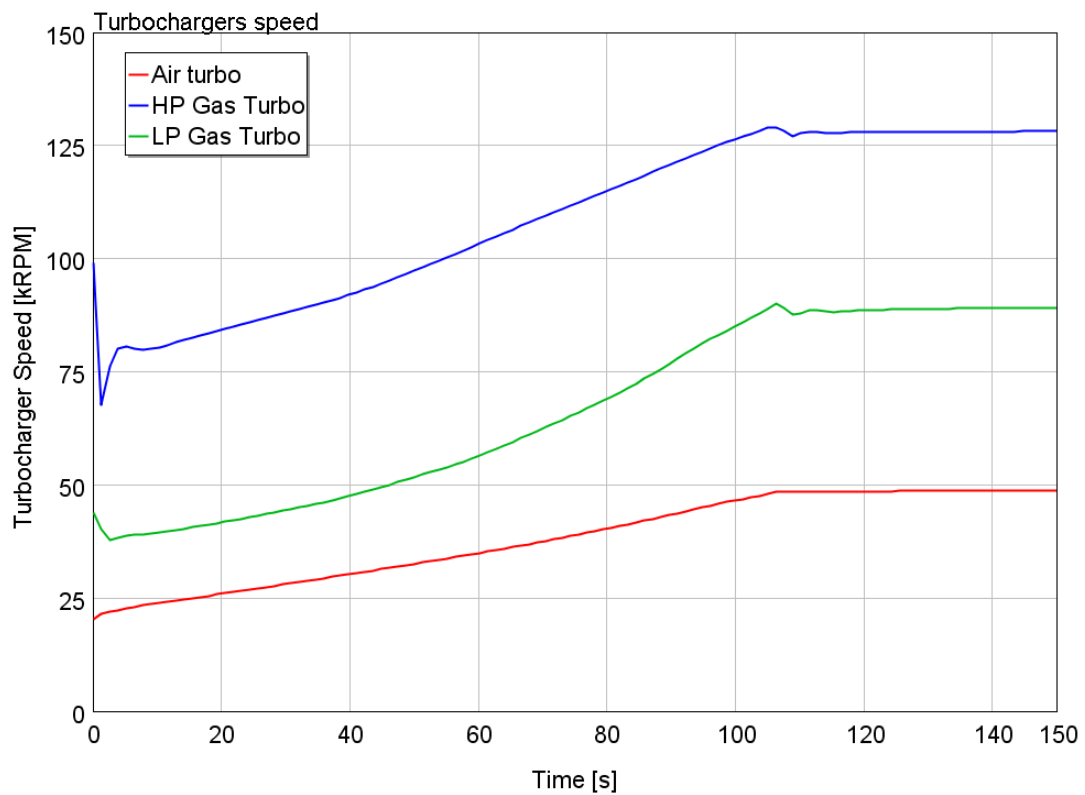


Fig. 19 – Air turbocharger, high pressure fuel turbocharger and low pressure fuel turbocharger speed during transient from idle to nominal power in constant engine speed 1500 RPM of engine with high pressure mixer

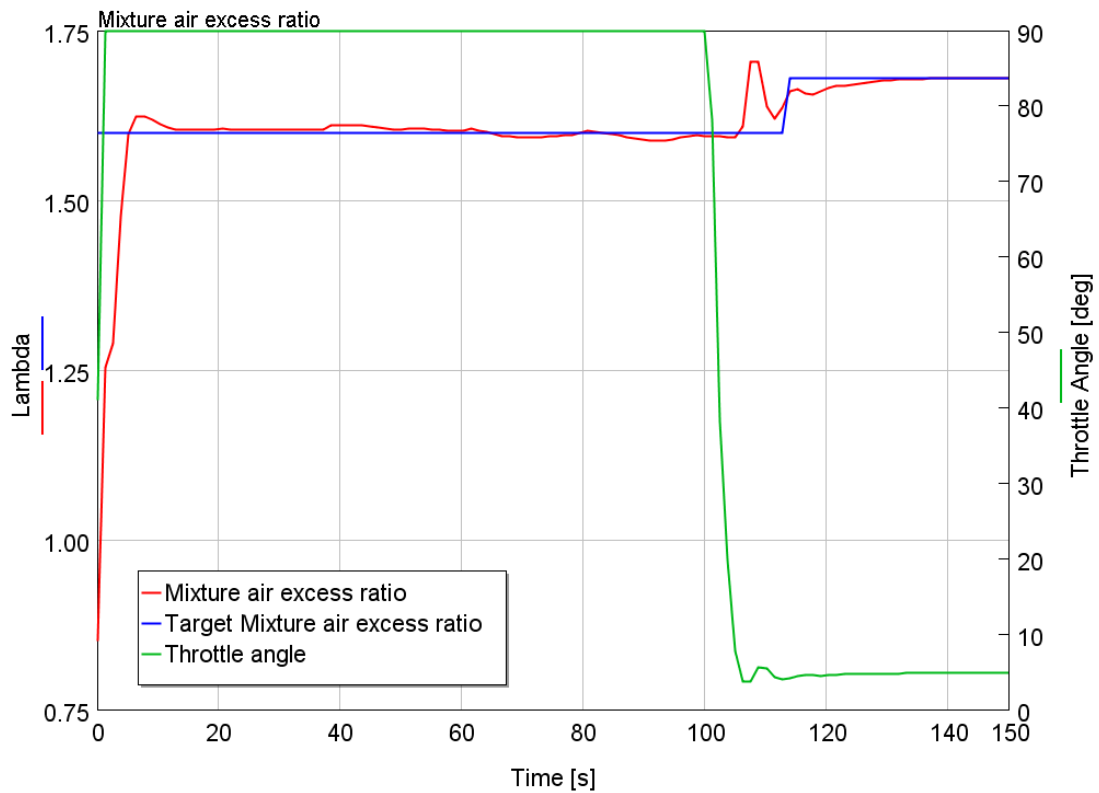


Fig. 20 – Mixture air excess ratio, target mixture air excess ratio and intake throttle angle during transient from idle to nominal power in constant engine speed 1500 RPM of engine with high pressure mixer

2.4. Engine with gas admission valves

The last modification was application of gas admission valves for the second variant of engine with power output of 1100 kW. Engine simulation model is displayed in Fig. 2. It can be seen that high pressure fuel led to the gas admission valves directly from fuel compressor. Gas admission valves are connected to the intake ports and each of them is synchronized with intake valve opening phase. If the fuel overpressure is not sufficient, the gas admission valve opens before intake opening phase. The mixture formation takes place in intake ports and cylinders. Nevertheless, pressure pulsation in intake manifold is causing induction of portion of the mixture from intake ports by neighboring cylinders which leads to unequal mixture air excess ratio of all cylinders. The earlier gas admission valve opens, the stronger is this effect. Only individual feedback control system could fully resolve this phenomenon.

2.4.1. Gas admission valves type specification

SOGAV gas admission valves from Woodward manufacturer were chosen for this application. These valves are designed for use on four-stroke, turbocharged, natural gas or dual-fuel engines. These valves are produced also in pressure-balanced version which does not need gas overpressure for their opening. Gas overpressure is necessary only for sufficiently short gas admission time available during intake stroke period. Two largest types were taken into account – SOGAV 105 (displayed in Fig. 21) and SOGAV 250. The valve mass flow rate is defined by Saint-Venant-Wantzel equation [1], where constant Z is gas admission valve constant (SOGAV 105 – $Z=105$). Value of sg is ratio of gas and air densities, p_1 is gas pressure and p_2 is air pressure, κ is ratio of gas specific heats at constant volume and constant pressure and T_g is gas temperature.

$$\dot{m} = Z \sqrt{\frac{2\kappa}{\kappa-1} \cdot sg \cdot p_1 \cdot \left(\frac{293,15}{273,15 + T_g} \right) \cdot \left(\left(\frac{p_2}{p_1} \right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_1} \right)^{\frac{\kappa+1}{\kappa}} \right)} \quad [1]$$

This equation was used for calculation of theoretical mass flow rate. The equivalent orifice diameter was set by a gas admission valve steady flow simulation. This diameter was set to 19,5 mm for SOGAV 105 and 30 mm for SOGAV 250.

Mass flow rate calculated by equation [1] shows that both valves are sufficient from the mass flow rate point of view. SOGAV 250 is almost two and half times more expensive than SOGAV 105. Therefore, SOGAV 105 was chosen for the use with the second engine variant. SOGAV 250 could be used place of SOGAV 105 if its mass flow capacity would not be sufficient.



Fig. 21 – SOGAV 105 – solenoid-operated gas admission valve

2.4.2. Engine control system setting

Previous control system had to be extended and reorganized for use with engine with gas admission valves. Control system has the same functions but they have been arranged in a different way. Moreover, modified control system is quite complex and all control loops influence one another. It is quite tricky to tune this complicated system which is based only on PID controllers. Nevertheless, in this case, a stable solution for engine simulation was found. More ingenious type of regulators should be use for practical application on real engine, for example some physical based control system.

The most important task of control system is gas admission control. Pulse-width modulated signal synchronized with relevant cylinder was used for opening gas admission valves as described in Chapter 2.4. It is displayed in *Fig. 22*. The in-cylinder mixture composition after intake phase was used as a feedback. This information is accessible in simulation case. However, it or would be quite challenging to obtain similar information on real engine. Nevertheless, individual lambda sensors could be used.

Individual control of each gas admission valve is one possibility. Another one is usage of main control loop and individual correction of opening phase of remaining gas admission valves. The main issue is stability of control system. It was solved by setting PI controllers to rather slow reaction time in the simulation case when target mixture air excess ratio was not always fulfilled. Another possibility is use of fuel super-critical pressure ratio. The fuel flow would be dependent only on the length of the opening period and not on the intake air pressure. With this approach, the control system could be simplified but there will be greater demand on all boosting units.

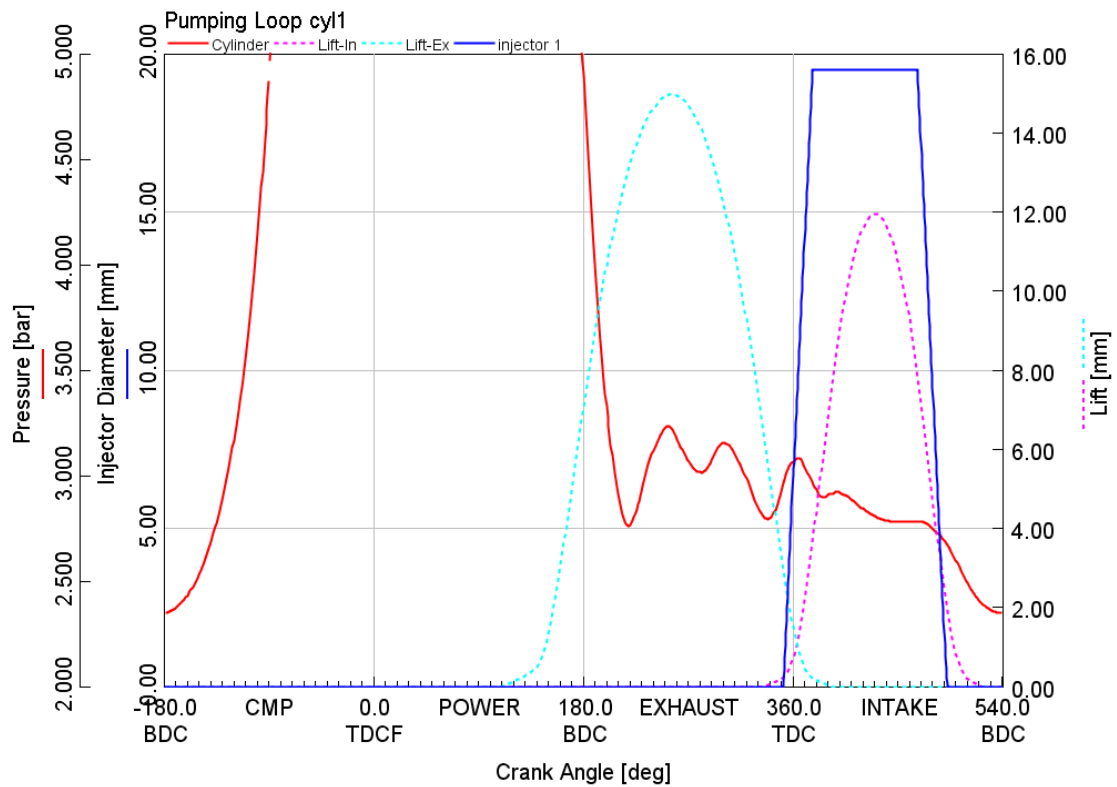


Fig. 22 – Pressure in cylinder, intake and exhaust valve lifts and injector valve diameter in dependence on crank angle

The second part of control system sets fuel overpressure by exhaust throttle flap. This system works in the same way as described in Chapter 2.3.1. Gas admission system with valves needs at least low fuel overpressure to keep valve-open-period short. Optimal valve open period is only during intake valves opening phase which keeps fuel suction to other cylinders at low level. This is main difference with high pressure mixer which uses Venturi pipe and can work on equal or even at slightly lower fuel pressure than air.

The third part of control system sets engine power. Several variants of control system were investigated. Throttling of intake air caused immediate decrease of mixture air excess ratio. Fuel throttle valve was employed to compensate for change of air and fuel pressure difference during engine throttling. Unfortunately, this control system was not stable. It was caused mainly by unequal pressure loss in air and fuel throttle flap and there was still an issue with change of mixture air excess ratio as a side effect of engine throttling.

In contrast to throttling of engine intake is a use of waste-gate parallel to turbine. Waste-gate bypassed both turbines since both are connected in parallel. Waste-gate decreases pressure ratio on both turbines simultaneously. Thanks to that, there is no influence on mixture air excess ratio which is good for control system stability.

2.4.3. Simulation results from engine with gas admission valves

Simulation of engine with gas admission valves was set similarly to engine with high pressure mixer. Initial conditions in pipes were set to atmospheric condition. Consequently, engine simulation was actually transient simulation until engine reached target power and steady state. Fig. 23 shows time history of engine power, and air and fuel pressure. Target power is reached after approximately 140 seconds. Time history of speed for all turbochargers is displayed in Fig. 24. Lambda control behavior can be observed in Fig. 25. Blue line represents CH_4 mass concentration in the first cylinder mixture which is set by the length of gas admission valve opening period, displayed by red line in Fig. 26. Last two control actuators are exhaust flap and waste-gate. They are displayed in Fig. 26. Control system, despite of slight oscillation, reached steady state after approximately 330 seconds.

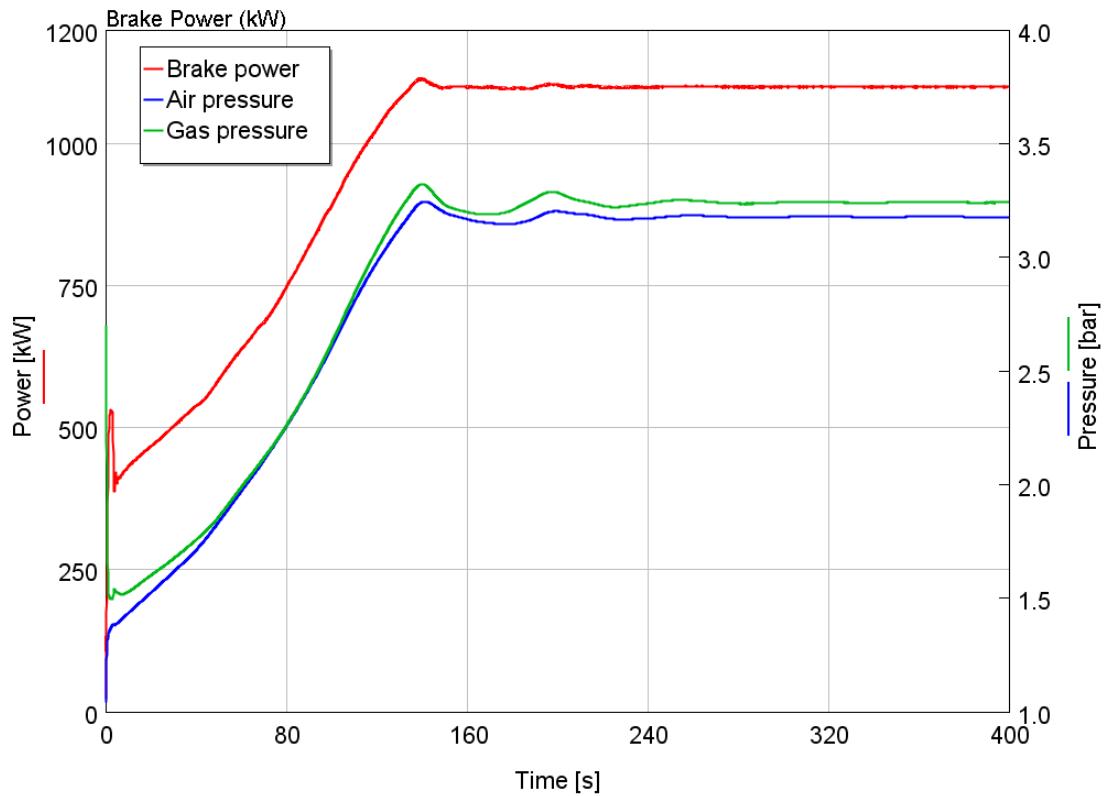


Fig. 23 – Engine power, air and fuel pressure during transient from idle to nominal power at constant engine speed 1500 RPM of engine with gas admission valves

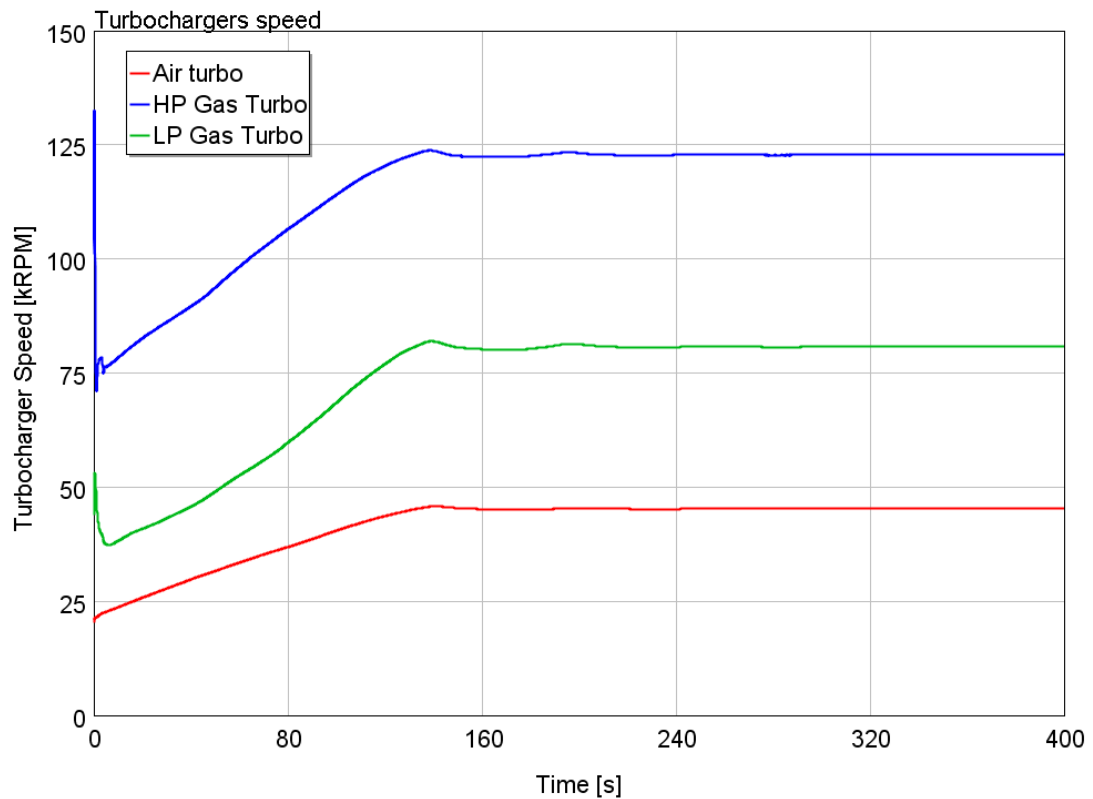


Fig. 24 – Air turbocharger, high pressure fuel turbocharger and low pressure fuel turbocharger speed during transient from idle to nominal power at constant engine speed 1500 RPM of engine with gas admission valves

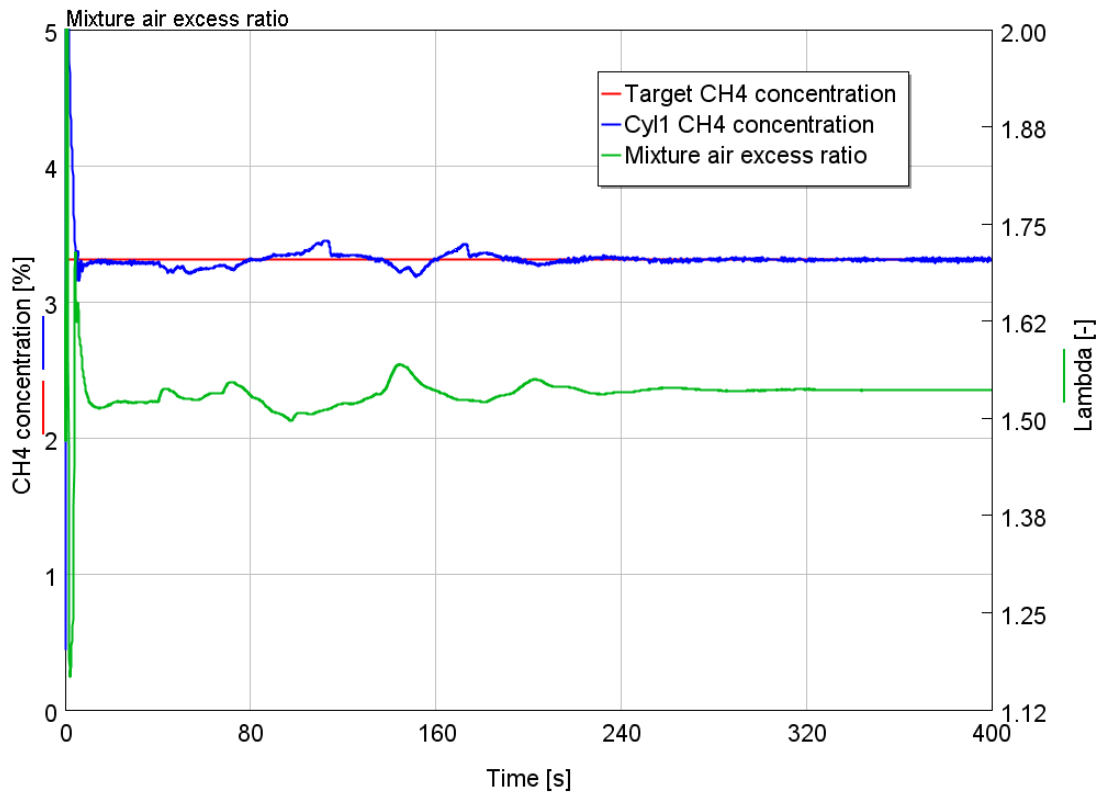


Fig. 25 – Mixture air excess ratio, target mixture air excess ratio and intake throttle angle during transient from idle to nominal power at constant engine speed 1500 RPM of engine with gas admission valves

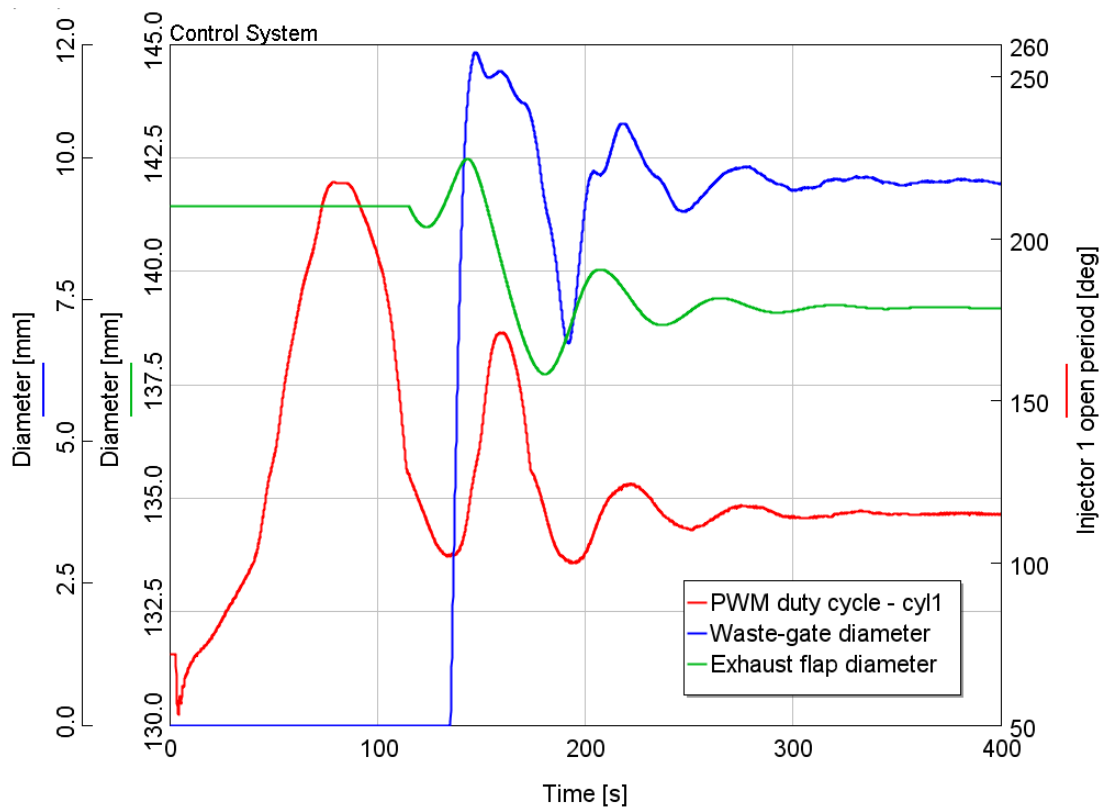


Fig. 26 – Pressure in cylinder, intake and exhaust valve lifts and injector valve diameter in dependence on crank angle of engine with gas admission valves

3. Conclusion

The simulation models of two biogas engines were developed. Both of these models have modified layout of intake and exhaust manifolds with separated compression of air and gaseous fuel. The first model was used for preliminary specification of turbocharger sizes. The ratio of mass flow capacity of air and fuel turbochargers should be approximately the same as mixture air to fuel ratio of mass fraction. The second engine model was used to specification of existing turbochargers. Consequently, this model was used also for control system tuning. Stability of such complicated system is crucial for possible future development.

Two configuration of mixture formation were investigated, the high pressure mixer and gas admission by valves close to engine cylinders. Gas admission system is more difficult to control, but it is also the safest solution of mixture formation used for gas engines. The Engine control system is quite complex and all actuators influence each controlled variables. Nevertheless, stable solution of control system was found during simulation. Special care should be taken of control system used on real engines.

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