EGR Support Investigation on a Diesel Engine

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Next generation emission standards (Euro 6 and US EPA 10) include significant limitations, especially in NOx and soot concentration of the exhaust gases. These cannot be fulfilled without exhaust gas recirculation (EGR). In several operation points of the engine, the naturally developed EGR-rate with the fully opened EGR valve cannot reduce the NOx emission to meet the requirements. In these cases the rate can be further increased with additional measures at special locations of intake or exhaust system. In this paper, these various EGR support possibilities will be investigated with through validated engine simulation model. The effect on the soot formation was taken into consideration as well.

A legújabb emissziós törvényi előírások (Euro 6 és US EPA 10) jelentős szigorításokat tartalmaznak, különösképpen a kipufogógázok NO_x és koromtartalmára vonatkozóan. Ezek az előírások nem teljesíthetőek kipufogógáz-visszavezetés nélkül (EGR). A motor számos munkapontjában, az EGR-szelep teljes nyitása esetén kialakuló EGR-arány nem elegendő a nitrogén-oxid-kibocsátás határérték alatt tartásához. Ilyen esetekben a kipufogógázok aránya a töltetben tovább növelhető a szívó- és kipufogórendszer megfelelő helyein elhelyezett beavatkozókkal. Ezen cikkben a különböző EGR-arány-növelési lehetőségeket vizsgáljuk meg validált motorszimulációs modell segítségével, a koromkibocsátásra gyakorolt hatásuk figyelembevételével.

INTRODUCTION

Legal framework for internal combustion engines are getting more and more rigorous and it is not different in case of commercial vehicle diesel engines. The Euro III (approved in October 2000) emission standards allowed 5 g/kWh maximum NOx production. From January of 2013 in the Euro VI it will be reduced to 0,4 g/kWh (measured in both cases during the European Stationery Cycle, ESC) [1]. It is less than ten times it was ten years ago. The NOx specification is the most challenging limitation in the new directives. These requirements can not be fulfilled only with additional set of equipments which engine manufacturers could to install in their ex-haust path.

There are basically two ways to reduce nitrogen-oxides. The first one is the exhaust gas aftertreatment. In this case the formation of raw nitrogen oxides are not prevented along the combustion process but it is eliminated using catalytic reductions (e.g.: selective catalytic reduction – SCR, or storage catalytic converter).

The other way is the raw nitrogen-oxide formation reduction during the combustion process. There are five main ways of NOx production during the combustion: the thermal or Zeldovich-mechanism, prompt or Fenimore-mechanism, NOx via N2O, NOx via NNH and by the fuel-bounded nitrogen [2]. The largest amount of nitrogen oxides in diesel engines is produced along the Zeldovich mechanism, which needs the following conditions: high temperature (T>1900 K, the activation energy of its first reaction is very high) and high oxygen concentration (that is why it forms in the post-flame region). So arrangements for lower cylinder temperature and lower oxygen con-centration are effective in the NOx concentration reduction. For in-stance these arrangements can be lower compression ratio, retarded fuel injection, boost pressure reduction and exhaust gas recirculation (EGR) etc. In this paper this last, in the future indispensable solution will be investigated with help of an engine simulation model. The exhaust gases will be recirculated into the intake system, mixed with the fresh air in a specially designed EGR-mixer. After this the engine cycle takes place with this more or less homogenous exhaust gas and fresh air mixture. The exhaust gases are inert thus the burning phase will extended, the heat releases slower. Moreover the inert 3-atomic gas in the exhaust gas increases the heat capacity and all this indicates lover peak temperatures in the cylinder. The oxygen concentration of the mixture is less than in the fresh air, so the thermal-NOx production will be decreased. Moreover the EGR have some additional effect for turbocharged engines: the exhaust gas is by-passed upstream to the turbine, the turbine pressure ratio decreases, therefore the turbine shaft power also decreased. It means lower charger speed and boost pressure.

In this paper the setup of a detailed diesel engine simulation model will be described with cooled high pressure EGR-system. For the validation of the model measurements were performed on an en-gine test bench. The measurement system and data used for model validation also will be introduced. These above mentioned interven-tions for combustion temperature reduction are generally increasing the fuel consumption but there are special conditions and engine op-erations where the brake specific fuel consumption (BSFC) could also be improved, especially in case of a turbocharged diesel engine with EGR. There are engine operation points where the attainable EGR-rate with a fully opened EGR-valve is not high enough. Here alternative options are investigated to the further increase EGR rate with measures at special locations of the exhaust or intake system.

ENGINE SIMULATION MODEL

Bore [mm]	102
Stroke [mm]	120
Number of cylinders	4
Engine displacement [cm3]	3922
Number of valves	4/cyl.
Compression ratio	17,3:1
Rated effective torque [Nm]	600 (1200 to 1600 RPM)
Rated effective power [kW]	123 at 2500 RPM

Table 1. Engine parameters

The engine simulated and used for the validation meas-urements, is a medium duty turbocharged and intercooled, commercial vehicle, common rail diesel engine. The main parameters are given in Tab.1.

In this section the engine model implemented in GT-Suite environment [3] will be described. The accuracy of a wave action engine model depends heavily on the discretization of the intake and exhaust domains. To achieve high accuracy level, the complete intake and exhaust system (pipes, intercooler, manifolds, ports etc.) were modeled in three-dimensional CAD software and then the 3D CAD model was transformed to 1D GT-Power flow objects by the GEM3D tool of GT-Suite. In this way the intake and exhaust system could be transformed to one-dimensional form as accurate as possible. The 3D geometry of the intake manifold and its intake ports are depicted in Fig.1.

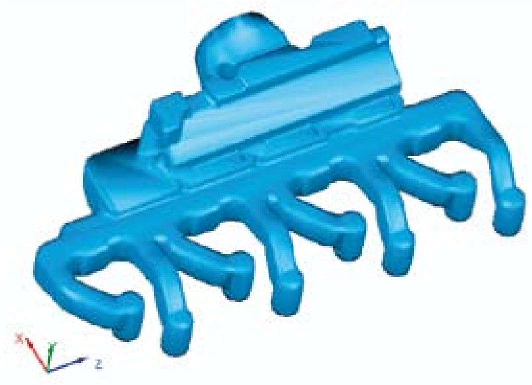

Figure 1. 3D CAD model of the intake manifold and ports

The 1D flow domain discretization length was chosen to the re-com-mended 0.4xbore diameter, namely 40 mm at the intake side, and tp 0.55xbore diameter, namely 55 mm at the exhaust side. The difference is due to the higher speed of sound in the exhaust system [3]. The intercooler was decomposed at the end as straight parallel pipes. The outlet air temperature and the pressure drop were adjusted to the measurements. It has essential influence to the volumetric efficiency (by intake air temperature) and also to the transients (by its volume). In the EGR loop a check valve were applied downstream the EGR cooler. The above described method of the one-dimensional intake and exhaust system modeling provides realistic flow system volumes, cross-sectional areas and pipe lengths.

- The compressor and the turbine of the turbocharger were modelled with performance maps. The compressor performance map can be seen in Fig.2.

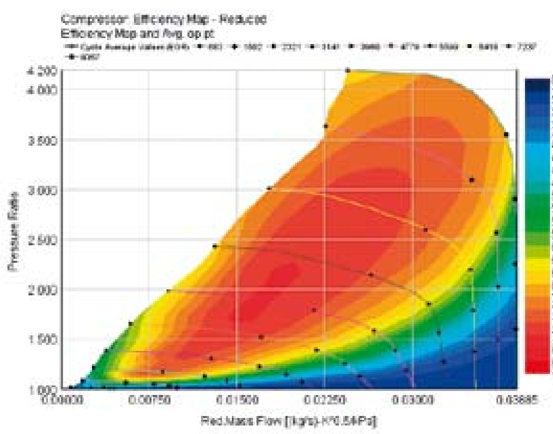

Figure 2. Compressor performance map

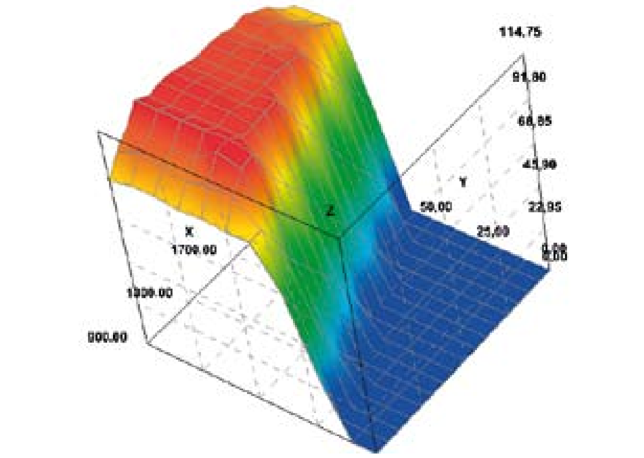

Figure 3. Fuel rate as function of the accelerator pedal position

The turbocharger is wastegate controlled actuated by a membrane chamber. In case of higher engine speeds the boost pressure does not exceed a desired value (the turbine will be bypassed). This boost pressure limitation was implemented in the model as a boost pressure sensor and a PID controller.

After completing the intake and exhaust system, and turbocharger modelling the appropriate air pressure and temperature in the intake manifold, and the exhaust back pressure were calibrated. Since the cylinder volumetric efficiency depends only on the intake and exhaust valve parameters, the valve lift curves were measured. Discharge coefficients were enrolled based on typical values and tuned taking into account the measured mass flow rates.

Injection modelling started with the fuel rate map definition. Map values come from fuel consumption measurements. The fuel rate map a function of the accelerator pedal position can be seen in Fig.3.

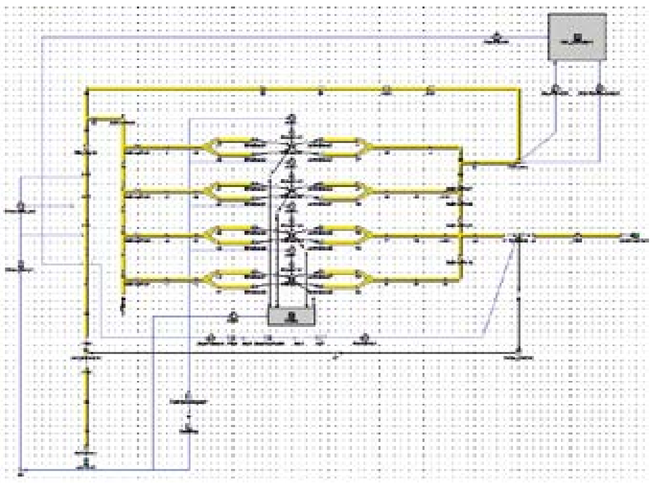

Figure 4. Simulation model layout with EGR loop

The injection process can be specified by the injector geometry, rail pressure, injection timing and duration. The rail pressure map was measured with the pressure sensor of the engine controller unit (EDC). The injector geometry was measured by a measurement mi-croscope. The injection duration and timing was measured with a current clamp installed on the solenoid controller wire. This way the pilot injections could be measured as well.

- Since the main goal was to predict emissions and cylinder pressures even at high EGR rates, a quasi-dimensional DI-Jet com-bustion model was chosen [4, 5]. It was calibrated to measured in-cylinder pressure and heat release curves.

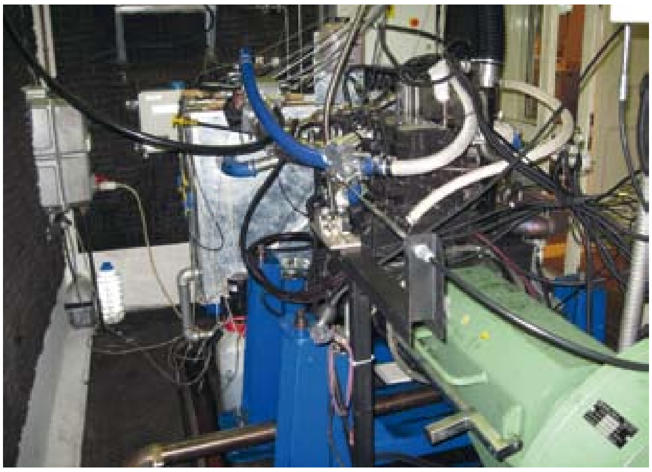

Figure 5. Engine test cell

– Then the friction mean effective pressure (FMEP) values were the only missing data to reach accurate effective brake torques. FMEP values were measured in the whole engine map with cylinder pressure indication.

ENGINE TEST CELL AND MEASUREMENT SYSTEM DESCRIPTION

The engine bench is based on an eddy current brake dynamometer. The torque is measured by the load cell of the brake. The accelerator pedal position is actuated and measured by a linear motor. The fuel consumption is metered based on the gravimetric fuel mass.

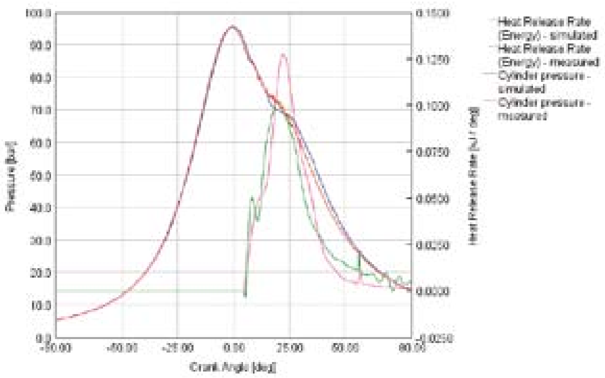

Figure 6. Cylinder pressure and heat release rate validation

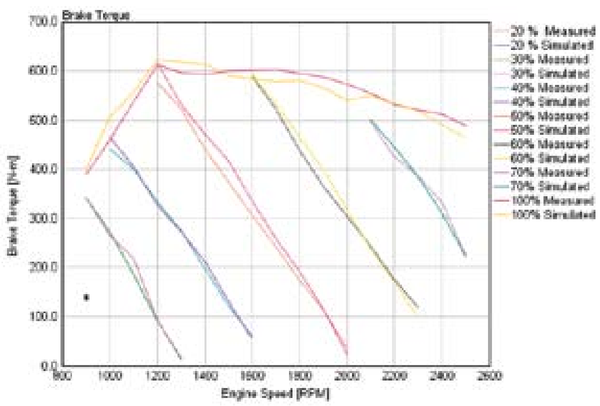

Figure 7. Effective engine torque validation

To validate the flow conditions pressure and temperature sensors were used in pars at different locations of the intake and exhaust system.

In the intake system:

– upstream to the compressor
– between compressor and intercooler
– downstream to the intercooler, at the intake manifold inlet

In the exhaust system:

– upstream to the turbine, at the exhaust manifold outlet
– downstream the turbine, before the electro-mechanic exhaust brake
– downstream the exhaust brake

A cylinder pressure indicating system was installed in each cylinder. Low pressure indicating sensors are installed in the intake and exhaust manifold. The crankshaft angle is measured by an optoelectronic angle encoder.

MODEL VALIDATION

The pressures and temperatures in the intake and exhaust system were verified at the above mentioned locations and their de-viations were below 5% to between calculated and measured values.

The injection and combustion model was validated based on indicated cylinder pressure curves and heat release rates. The fit of the measurements and the simulation can be seen in Fig.6.

Finally, the validation of the effective engine torques was performed. The simulation results are depicted along the measure-ments in the Fig. 7. The difference here is also below 5 %, so the model can be accepted for the control investigations.

EGR SYSTEM INVESTIGATION

For emission certification of medium-duty diesel engines in stationary mode the ESC (European Stationery Cycle) test is used. In this paper three operation points were isolated in which the various support measures of exhaust gas recirculation were investigated. The engine speed and load diagram of the ESC test are seen in Fig. 8.

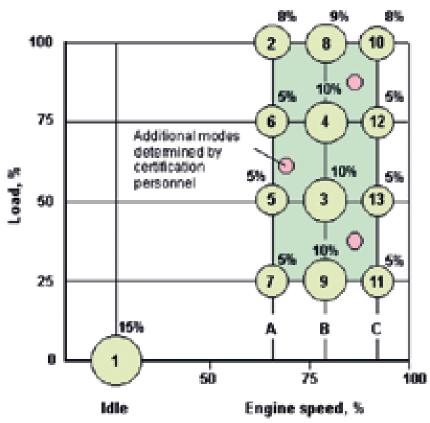

Figure 8. European Stationary Cycle (ESC) [1]

The selected operation points are: the 'A' speed at 25% load, the 'B' speed at 50% load and the 'C' speed at 75% load.

For the particular engine, the speeds and the effective torques are: A25=1500 RPM and 150 Nm, B50=1900 RPM and 290 Nm, C75=2300 RPM and 375 Nm.

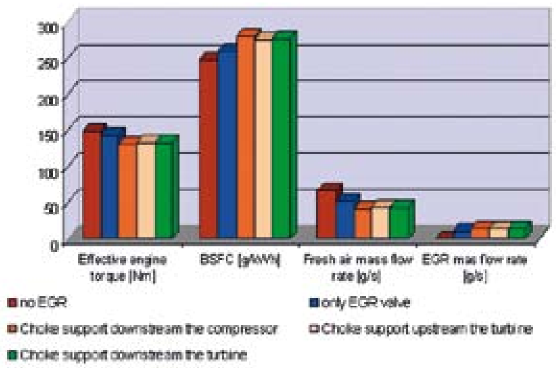

Figure 9. Torque, brake specific fuel consumption, gas mass flow rates in the A25 operation point

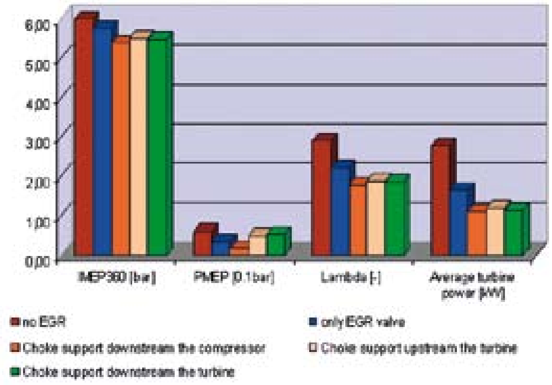

Figure 10. IMEP360, PMEP, lambda and turbine power in the A25 operation point

To satisfy the low NOx production limitation of the Euro VI emission norm (0,4 g/kWh in ESC or ETC test) without exhaust gas after treatment it is essential to reach high EGR-rates. For these high rates in several engine operation mode it is not enough to fully open the EGR valve. To increase the EGR rate additional interventions are needed. These can be carried out with flaps at different locations of the intake and exhaust system. The investigated placements are the following:

- downstream the compressor
- upstream the turbine
- downstream the turbine

At each operation point the different EGR supports were investigated. First, as reference, the EGR valve was fully closed. In a second simulation the EGR valve was fully opened and no other support was used. In a third case the support downstream to the compressor was applied and increased the EGR rate to higher level. In fourth and fifth cases the same EGR rate was target like in the third case but with supports at upstream and downstream locations to the turbine respectively. The injected fuel rate, injection timing, duration and pressure profiles are identical to the initial no EGR case in each operation point. The goal is to compare the different exhaust gas re-circulation choices.

RESULTS IN A25 OPERATION POINT

In the initial case the EGR valve was closed. It was depicted with brown colour in the diagrams. In the second case only the EGR valve was opened (the choke flaps were fully opened), which is illustrated with blue colour. It resulted in 16.4 % EGR rate, which is the maximum achievable without any support. The EGR rate can be calculated with the following equation:

$$EGR_{rate} = \frac{\dot{m}_{EGR}}{\dot{m}_{EGR} + \dot{m}_{AIR}} \quad \rightarrow \quad (1)$$

In the last three cases the desired EGR rate was increased to 25 % by the support measures. The throttle position was set by a controller to achieve the desired 25 % EGR rate.

The effective torque is maximal at the no EGR case and decreases with higher EGR rates. Higher EGR rates result prolonged heat release and hence less gross indicated mean effective pressure (IMEP360). The turbine power depends strongly on the mass flow rate, so it correlates to the fresh air mass flow rate.

Pumping mean effective pressures (PMEP) are relative low values since the intake and exhaust pressure is nearly identical to the initial case (positive PMEP represents losses).

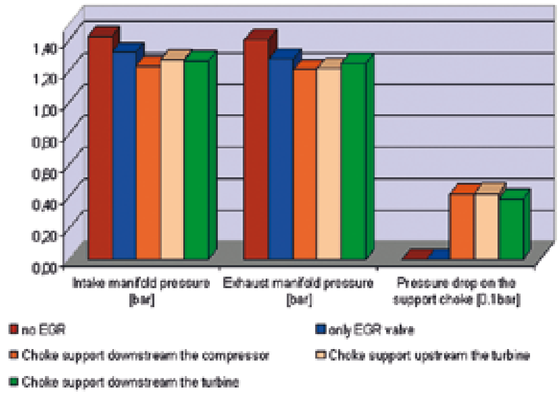

Figure 11. Manifold pressures and pressure drop values on the chokes in the A25 operation point

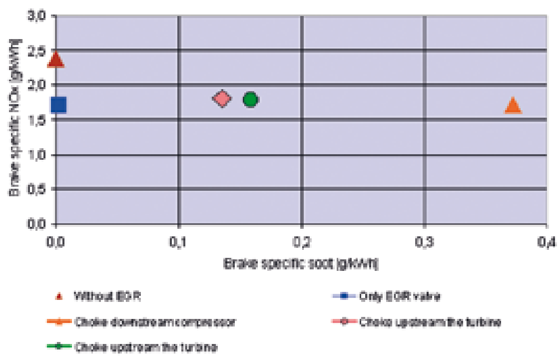

Figure 12. NO_x-soot trade-off in the A25 operation point

The pressure drops on the chokes are similar values in each case. The maximal intake manifold pressure with the 25% EGR rate was found when the pressure drop was applied upstream the turbine.

In the initial case the amount of the soot is minimal due to high air-fuel ratio. Applying 16.4 % EGR rate the NOx reduced by 38%. The further 10 % EGR-rate increase could not decrease the NOx produc-tion but promoted the soot formation.

RESULT IN B50 OPERATION POINT

With fully open EGR valve 14.5% EGR-rate was developed. It was increased to 25 % in the last three cases with support. The torque and fuel consumption shape shows a similar trend to the A25 case:

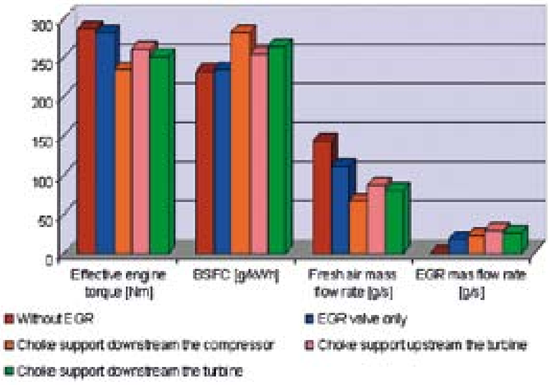

Figure 13. Torque, fuel consumption, gas mass flow rates in the B50 operation point

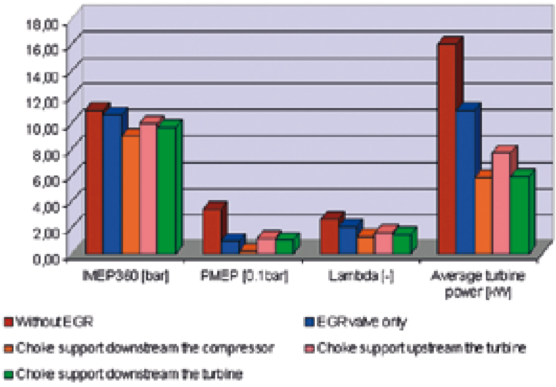

Figure 14. IMEP360, PMEP, lambda and turbine power in the B50 operation point

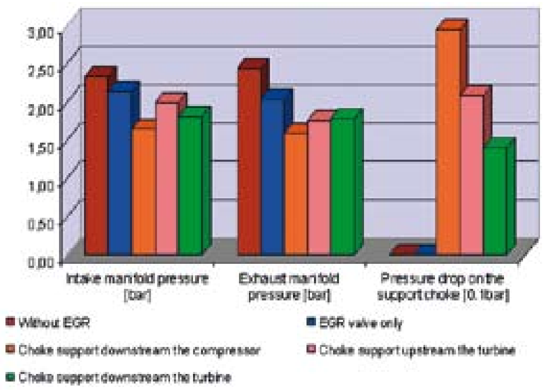

Figure 15. Manifold pressures and pressure drop values on the chokes in the B50 operation point

the initial torque is the highest and the support downstream to the compressor results the least effective support.

The highest overall intake mass flow rate of the 25 % EGR cases is when the support is applied upstream to the turbine. The highest lambda and with this the maximum indicated mean pressure is de-veloped here from the last three high EGR cases.

The difference between the intake and exhaust manifold pres-sure is maximal in the first case without EGR. If the EGR valve is open, the pressure will be equalized through the EGR system. It means less charge exchange losses and less fuel consumption.

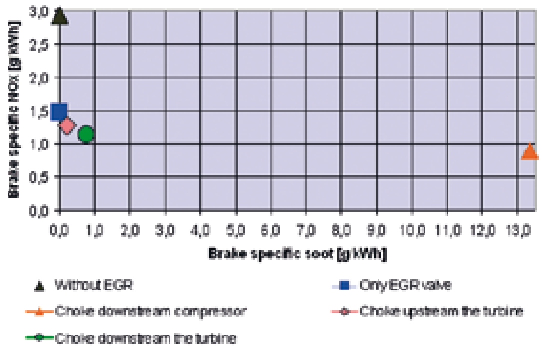

Figure 16. NO_x -soot trade-off in the B50 case operation point

The NOx concentration decreases significantly (by 70%) in the higher EGR cases. The minimum value results with the intake support but due to low oxygen concentration the soot emission is high.

RESULTS IN THE C75 OPERATION POINT

The attainable EGR rate without any support was 19.9 %. The per-formance results are shown in Fig.17.

The brake torque values are here the most balanced. With fully open EGR valve but no support (blue case) the simulation results higher brake power.

As seen in Fig. 18. the indicated mean pressure is less in the second case than the initial was due to the recirculated inert ga-ses. Despite this fact the net power can be increased due to the minimized charge ex-change losses (PMEP).

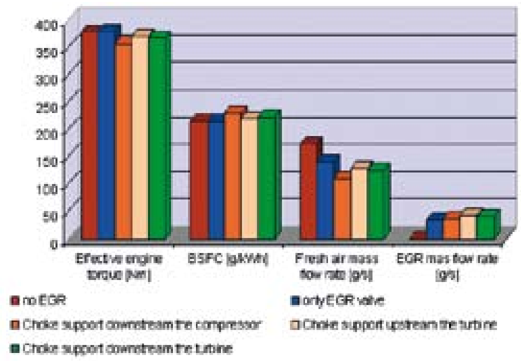

Figure 17. Torque, fuel consumption, gas mass flow rates in the C75 operation point

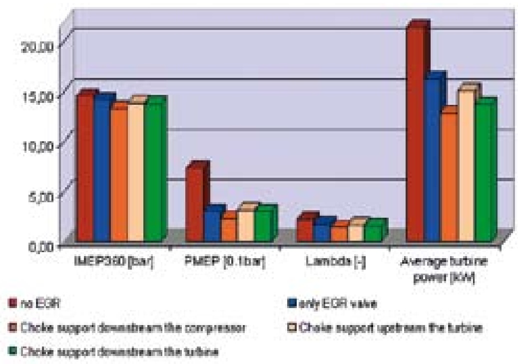

Figure 18. IMEP360, PMEP, lambda and turbine power in the C75 operation point

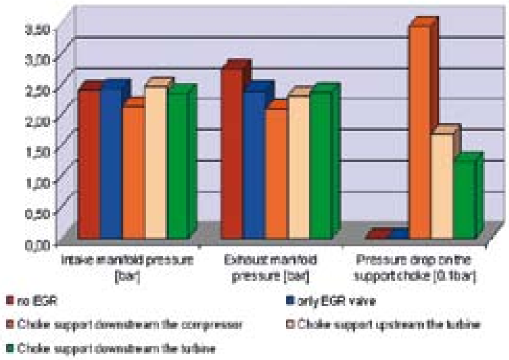

Figure 19. Manifold pressures and pressure drop values on the chokes in the C75 operation point

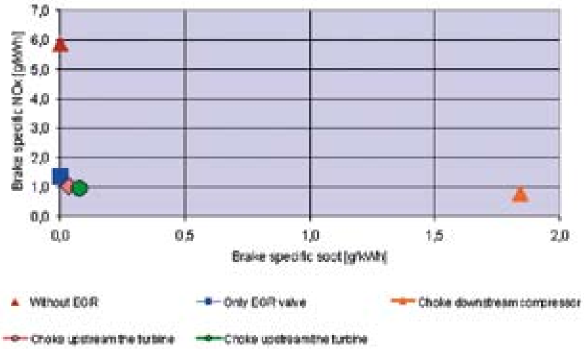

Figure 20. NO_x-soot trade-off in the C75 operation point

The reason for the lower PMEP values in EGR cases can be traced in the Figure 19. With opening the EGR valve the intake and exhaust manifold will be connected to each other and the pressures can be balanced through the EGR loop.

The NOx-soot trade-off shaped similarly as in the previous simula-tions. The NOx reduction is even more pronounced with EGR.

CONCLUSIONS

This paper investigated the EGR rate support alternatives in terms of engine performance specific fuel consumption and NOx and soot emission levels in selected operation points of the ESC map.

In A25 operation point the NOx formation was moderately reducted by the exhaust gases and a further EGR-rate increase was not effective: only soot emission increased. The lowest nitrogen-oxide concentration resulted with the support downstream the compressor but with an inc-reased PM emission. Turbine up and downstream support was similarly effective, but with much lower PM. It was interesting to see that an EGR rate increase is how far not the only factor for low NOx emission. On the other hand other effects of higher EGR rate realization makes severe consequences to the engine performance and soot emission!

The B50 operation point results show similar effects, but the EGR-rate increase with the support continued to decline the nitro-gen-oxides. The best value occurred again with the flap in the inta-ke system but the soot is unacceptable. Turbine supports are the same way preferred with low PM emissions and similar NOx.

In C75 operation point an improvement of the effective torque can be noticed with the fully open EGR valve! It is casued by the balanced intake and exhaust manifold pressure. The pumping losses are minimized this way. The NOx-soot trade-off diagram looks similar to the previous cases.

As summarization it can be assessed that the support downst-ream to the compressor is the best EGR-rate increase measure if the goal were only to minimize the nitrogen-oxide emissions. However it is disappointing in PM emission and fuel consumption. In perspective of the fuel consumption, particulate matter and NOx the best support option could be upstream to the turbine. A drawback of this solution at the realisation is the hard environ-ment due to the high exhaust gas temperatures. Considering the design difficulties a reasonable compromise could be achieved with controlled flap downstream the turbine.

ACKNOWLEDGEMENTS

The work is connected to the scientific program of the “Development of quality-oriented and harmonized R+D+I strategy and functional mo-del at BME” project. This project is supported by the New Széchenyi Plan (Project ID: TÁMOP-4.2.1/B-09/1/KMR-2010-0002).

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