Forecasting of a boosted locomotive gas diesel engine parameters with one- and two-stage charging systems

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ABSTRACT

Conversion of locomotive engines for operation on natural gas lowers considerably expenses for fuel and reduces exhaust emissions which makes it possible to omit large and expensive aftertreatment systems. The permanent need to raise the engine power requires a considerable increase of the boost pressure. This can be realized by using a high pressure turbocharger or a two-stage charging system. In the research, parameters of a high boosted D200 6-cylinder locomotive engine having D/S=200/280 mm are forecasted using a one-zone model developed in MADI. An analysis was carried out to explain why the 1st stage compressor of the two-stage charging system should be specially profiled to have its map tilted to the right. Calculations were performed for the gas diesel engine having a break mean effective pressure (BMEP) 2.7 MPa with one and two-stage charging systems. In both cases, close fuel efficiency was obtained, though for the two-stage charging system, the boost air pressure was higher. The engine with one turbocharger had no reserves for further power augmentation while the two-stage charging system enabled to increase the boost air pressure further. Therefore, parameters of the engine having a higher BMEP 3.2 MPa were calculated. In that case, not to exceed the peak combustion pressure, a retarded fuel injection was used which resulted in fuel efficiency drop by approximately 1.5%.

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1. Introduction

Conversion of diesel engines for operation on natural gas is important because there are more explored reserves of natural gas than oil on our planet, the price of natural gas is twice lower in Russia and the price difference will grow as the oil reserves are depleted. Transfer from diesel to gas fuel results in considerable reduction or zero emissions of particles and decrease of nitrogen oxides NO_x emissions up to 50%. Emission of carbon dioxide CO_2 decreases by 15% due to lower content of carbon in methane and may be lowered by additional 15% because methane has approximately 15% higher caloric efficiency than diesel fuel. Saving expenses for fuel for locomotive engines fed by natural gas is especially high because, in contrast to automobile transport, locomotives often operate 24 hours a day seven days a week.

High boosting of locomotive engines is a relevant issue because the space in the engine compartment and load of the locomotive carriage are limited while the requirements for power are growing permanently. In addition, high boosting improves engine fuel efficiency due to its mechanical efficiency growth.

High boosting of engines requires high charging pressure. Today, more and more diesel and gas diesel engines use Miller cycle, first of all, to comply with stringent emissions standards because it reduces NO_x emissions up to 50%. As the Miller cycled is realized by the early closure of the intake valve, the filling efficiency of the engine decreases and therefore, charging systems with high boost pressure are required for engines with Miller cycle even if they have not very high power augmentation.

Different methods of organization of the working process of locomotive engines fed with natural gas were analyzed in detail in (Shatrov et al., 2018). It was demonstrated that a gas diesel cycle with the minimal portion of igniting diesel fuel has many advantages over a spark ignition gas engine cycle for locomotive engines mainly because it is not sensitive to knock origination and therefore can be used on highly boosted engines with large cylinder size.

Highly boosted diesel and gas diesel engines require advanced diesel fuel supply systems. As the air density in the cylinder increases, the fuel supply systems should ensure higher injection pressure, injection rate front shape control to decrease NO_x emissions and one should take into account fuel pressure oscillations in the high pressure line having a considerable impact on the fuel injection process in case of a high fuel injection pressure and multistage injection (Shatrov et al., 2020).

Maximum values of the brake mean effective pressure of medium speed diesel and gas diesel engines including locomotive engines which are in development in Russia and abroad attain 2.7-2.8 MPa. This requires the use of charging systems ensuring the compressor pressure ratio of 5.5 and higher. Reaching such a high pressure ratio in a conventionally compressor results in a drop of efficiency and overheating of the compressor wheel which needs more expensive materials. The scheme of a two-stage charging system is presented in Figure 1.

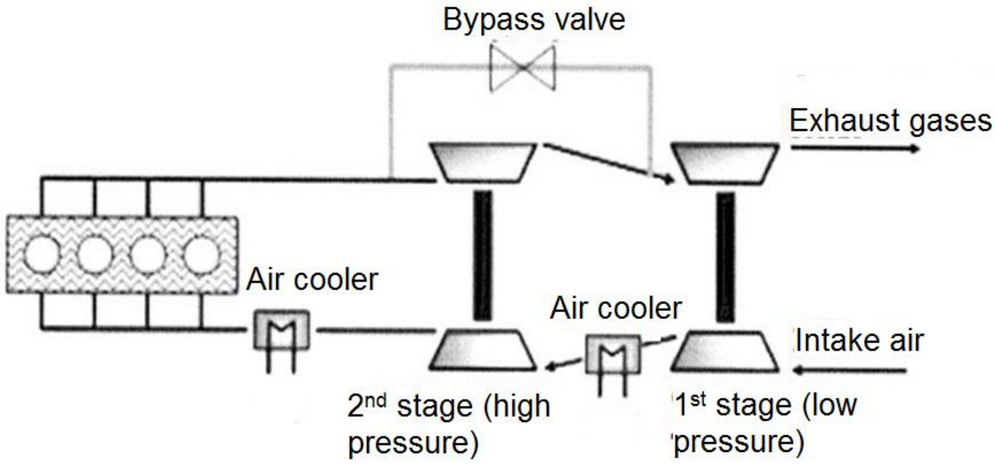

Figure 1. Two-stage charging system with a two-stage compressed air intercooling

The ABB Turbosystems Company manufactures high pressure turbochargers with pressure ratio in the range of 5.5….6.5. The compressors have high thermal loads and their wheels are cooled by the charging air from the intercooler after reaching the pressure ratio of about 4.0. Production of these turbochargers requires the use of high-precision manufacturing equipment, advanced materials and technologies. Such turbochargers make high requirements for operation conditions and service level. It may cause problems in Russia, where the temperature varies from -50 to +50 Celsius and locomotives often work in remote regions. A high pressure turbocharger was developed by the Penzadieselmash company. It has one axial and one radial flow wheels, can reach a high compression ratio 5.5, has a pretty simple design and ordinary technology can be used for its manufacturing (Shatrov et al., 2015).

On the other hand, it is possible to get the required pressure ratio of 5.5 and higher in two-stage turbocharging systems using ordinary turbochargers having the pressure ratio of 2.5-3.5. Such turbochargers have been long manufactured in Russia and they have proven high durability in the most different operating conditions. According to Trapp et al. (2011), Grigorov (2016) and Behr et al., (2013), one- and two-stage charging systems have close equivalent efficiency at compressor pressure ratio up to 4.0-5.0. At higher pressure ratio, the two-stage system is more efficient. The equivalent efficiency of two-stage charging systems is few percent higher compared with one-stage systems because the air is cooled twice after each compressor (see Figure 1). A lower boost air temperature results in a better filling efficiency, lower thermal loads of the engine parts and lower NO_x emission. The disadvantages are larger weight and dimensions and longer "turbo-lag" due to a higher inertia of two rotors.

2. Problem Statement

General compressor maps of the 1st and 2nd stages of a two-stage charging system are presented in Figure 2. It is clearly seen that the compressor map for the 2nd stage looks in a quite ordinary way while the compressor map of the 1st stage is tilted to the right side to work efficiently in the area of higher air flows. This is explained by formulas used for calculation of the reduced values of the compressor air flow $G_{a.red}$ and turbocharger rotor speed $n_{r.red}$.

Reduced rotor speed

$$n_{\mathrm{r.red}} = n_{\mathrm{r}} \times \sqrt{\frac{\mathrm{T}_0^*}{\mathrm{T}_1^*}}. \tag{1}$$

Reduced compressor air flow

$$G_{\mathrm{a.red}} = G_{\mathrm{a}} \times \sqrt{\frac{\mathrm{T}_0^*}{\mathrm{T}_1^*}} \times \frac{\mathrm{p}_0^*}{\mathrm{p}_1^*}. \tag{2}$$

Here: n_r – rotor speed (rpm); G_a – compressor air flow (K); T^*_0=288 – compressor map reduction temperature (K); p^*_0=0.1 – compressor map reduction pressure (MPa); T^*_1 – temperature at compressor inlet (K); p^*_1 – pressure at compressor inlet (MPa).

As seen from equation (1), the reduced rotor speeds of the 1st and 2nd stages do not differ much from the real values because the temperatures T^*_0 and T^*_1 are very close for the 1st stage compressor and do not differ much for the 2nd stage compressor due to the presence of an efficient intercooler between the compressors. Additionally, the temperature ratio is under the square root.

For the reduced air flow, the influence of the temperature ratio is the same, but the influence of the pressure is quite different.

In case of the two-stage turbocharging, the air flow G_a passing through each compressor which is in the numerator of equation (2) grows 2-3 times compared with the air flow passing through the compressor when it is used in a one-stage charging system.

- 2nd stage compressor. The pressure p^*_1 at the compressor inlet which is in the denominator grows 2-3 times approximately proportionally to the airflow increase. Therefore, the reduced airflow of the 2nd stage compressor remains close to the airflow of the compressor used for one-stage turbocharging and compressor has a conventional map as shown in Figure 2,b.
- 1st stage compressor. The pressure at the compressor inlet p^*_1 is close to the reduction pressure p^*_0 while the airflow grows 2-3 times as for the 2nd stage. Therefore, the reduced air flow also grows 2-3 times. If we use an ordinary compressor designed for one-stage turbocharging, the working point will move to the the right, to the area of very low compressor efficiency. Therefore, using a special compressor having its map tilted to the right as shown in Figure 2,a will be the best solution.

If one does not have a possibility to manufacture such a compressor, it is possible to take just a larger size turbocharger which has a high efficiency at larger air flows. In this case, the turbocharger will be oversized which is not beneficial in terms of its weight, size and price.

On the bases of the analysis performed, the following goals of the paper may be indicated:

- Analysis of the features of one- and two-stage high-boost charging systems
- Selection of turbochargers for a two-stage charging system of the D200 gas diesel engine
- Calculation of parameters of the D200 gas diesel engine boosted to the brake mean effective pressure p_e=2.7 MPa, with one- and two-stage charging systems
- Prediction of the maximal possible brake mean effective pressure and performance parameters of the D200 gas diesel engine with the two-stage charging system selected

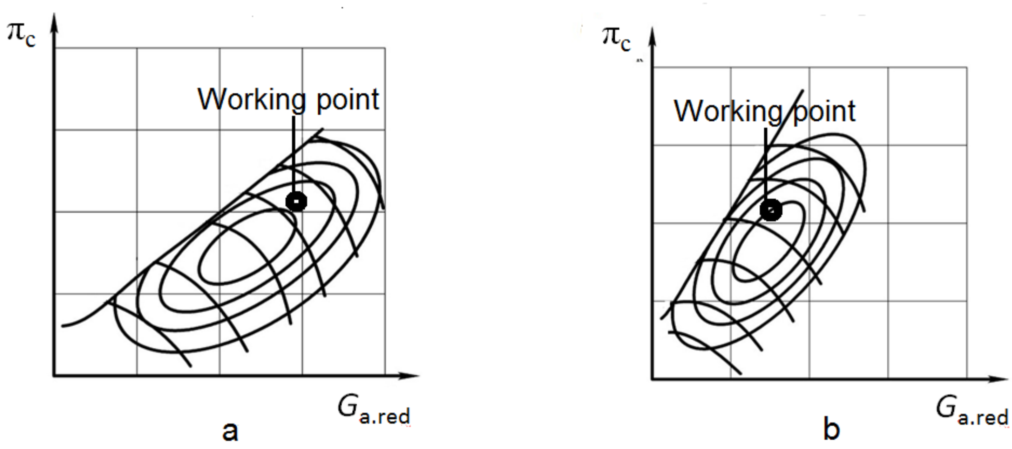

Figure 2. Compressor maps of the 1st (a) and 2nd (b) stage turbochargers

3. Simulation Model

Parameters of joint operation of a gas diesel engine with one-stage and two-stage turbocharging systems were calculated by a model of diesel/gas diesel/gas engine developed in MADI (Khatchiyan et al., 2015). The one-zone model is based on the first law of thermodynamics. Well known empirical formulas of I. Vibe and G. Woschni are used for calculation of heat release and heat losses into the cylinder walls. The temperatures of the cylinder surfaces (fixed cylinder head and piston crown areas and variable cylinder liner area) and mechanical losses are also calculated by empirical formulas. Variations of pressure and temperature in the intake and exhaust manifolds are calculated on the base of the ”filling-emptying” method. Filling efficiency decrease due to the replacement of a portion of intake air by gaseous fuel is calculated by a method offered by Matyukhin et al. (2020). Parameters of joint operation of engines with one-stage and two-stage turbocharging systems are calculated using iteration method. Initially, approximate values of turbocharger rotor speed n_r, air excess coefficient α, charging pressure p_c, exhaust gases back pressure at the turbine inlet p_t and some other input parameters are entered and then redefined using iterations. Iterations are stopped when the difference between the compressor and turbine power is less than the predetermined value (2%).

The research object was a D200 gas diesel engine, which has 6 in-line cylinders with cylinder bore 200 mm and stroke 280 mm, compression ratio 14.0:1, rated speed 1000 rpm and rated brake mean effective pressure p_e=2.7 MPa. The ignition diesel fuel portion is 5%. Limitations were used: air excess coefficient α>2.0, peak combustion pressure p_z<22 MPa, exhaust gases temperature T_t< 873 K, turbocharger rotor reduced speed $n_{r.red}$ lower than the maximal speed shown on the compressor map, the surge margin >10%. For every calculation point, the optimal in terms of fuel consumption fuel injection advance angle was found. If the peak combustion pressure p_z was higher than the maximal admitted value for this engine (22 MPa), the fuel injection advance angle was retarded.

For one-stage charging system, compressor and turbine maps of the high pressure TK-2202 turbocharger manufactured by the Penzadieselmash company were used. For the two-stage charging system, standard maps of turbochargers for the medium speed engines of the Russian manufacturers were used (Turbochargers TK23-TK40, 2020). A TK30H-17 turbocharger was selected for the 1st stage and TK23C-011 – for the 2nd stage (Figure 3).

4. Results and analysis

For the gas diesel engine boosted to the brake mean effective pressure p_e=2.7 MPa, position of the working point on compressor map for the one-stage charging system is presented in Figure 3 and for the two-stage charging system – in Figure 4. In all three compressor maps, the working point is located in the area of high compressor efficiency and far from the surge line. As seen from Figure 3, there are no reserves for further engine power augmentation because the reduced rotor speed $n_{r.red}$ is close to maximum. With the two-stage charging system, there are reserves for increasing the rotor speed (Figure 4). Therefore the gas diesel engine operation with higher power augmentation corresponding to p_e=3.2 MPa was calculated.

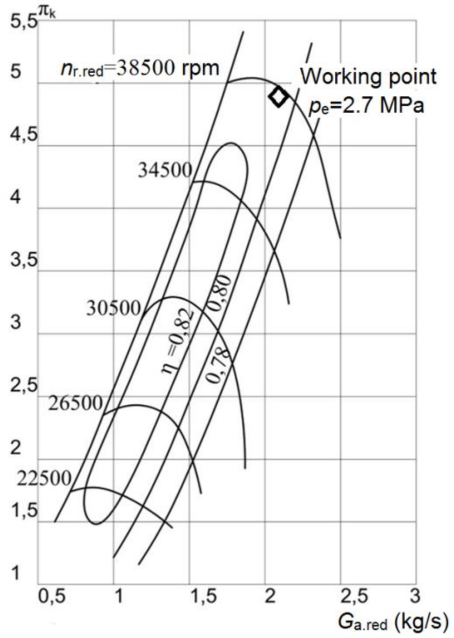

Figure 3. Compressor map of the TK2202 turbocharger

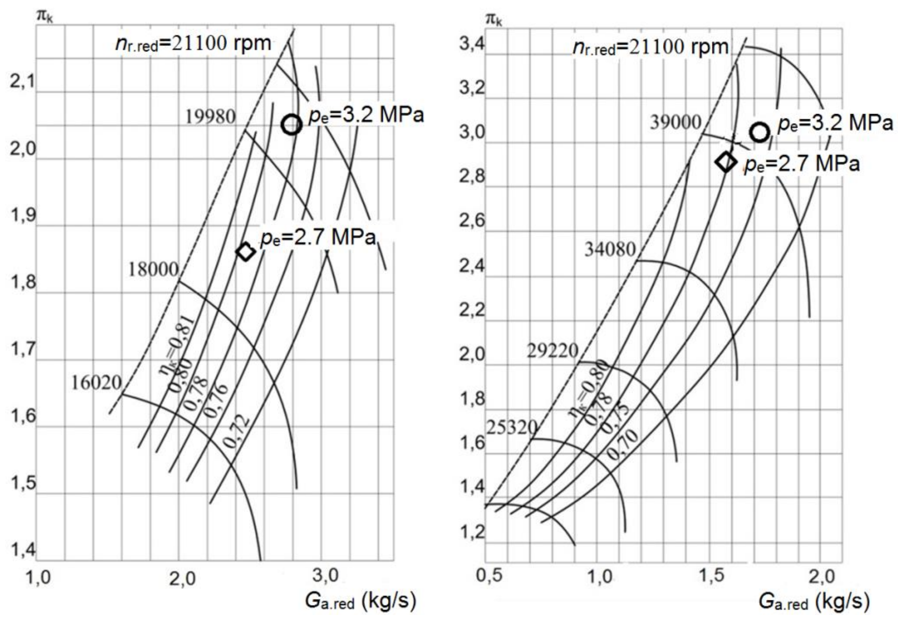

Figure 4. Compressor map of the 1st stage turbocharger TK30-H17 (a) and 2nd stage turbocharger TK23-C011 (b)

The peak cylinder pressure p_z exceeded the critical value of 22 MPa, and to correct the situation, a fuel injection start angle retarded by 8 crank angle degrees was used. This resulted in a small drop of fuel efficiency - 1.5%.

Tables 1 and 2 show the parameters of the D200 gas diesel engine and its turbochargers with one- and two-stage charging systems boosted to p_e=2.7 MPa, as well as for the engine with the two-stage charging system boosted to p_e=3.2 MPa

Table 1. Parameters of the gas diesel engine D200

Charging system and engine boosting	p_c (MPa)	p_t (MPa)	η_i	α	p_z (MPa)	g_e (g/kWh)	T_c (K)	T_t (K)
One-stage, p_e=2.7 MPa	0.468	0.431	0.493	2.49	21.9	183	350	784
Two-stage, p_e=2.7 MPa	0.495	0.405	0.494	2.53	21.8	184.5	328	800
Two-stage, p_e=3.2 MPa	0.572	0.432	0.484	2.10	22.0*	187.6	330	953

* To reduce p_z, the fuel injection advance angle was retarded

Table 2. Parameters of the turbochargers

Charging system and engine boosting		π_c	π_t	$n_{r.red}$	$G_{a.red}$	η_c
One stage, p_e=2.7 MPa		4.92	4.10	38060	2.05	0.81
Two stage, p_e=2.7 MPa	1st stage	1.85	1.58	19080	2.44	0.794
	2nd stage	2.90	2.49	38280	1.57	0.786
Two-stage, p_e=3.2 MPa	1st stage	2.05	1.72	20990	2.76	0.780
	2nd stage	3.01	2.43	41240	1.72	0.756

As seen from Tables 1 and 2, performance parameters of the gas diesel engine with o charging systems at p_e=2.7 MPa are close. Accordingly, we got a low brake specific effective fuel consumption (g_e=183 and 184.5 g/kWh), high boost pressure (p_c=0.468 and 0.495 MPa), indicated efficiency (η_i=0.493 and 0.494) and excess air ratio (α=2.49 and 2.53), though the values of p_c and α for the two-stage system are higher. It should be noted that the charge air temperature is lower in the case of a two-stage charging system (T_c = 328 K compared with 350 K) due to double air cooling. Compressor pressure ratio was quite lower for the two-stage system turbochargers (π_c=1.85 and 2.90) compared with the one-stage system (π_c=4.92). The ratio of compressor pressure ratio to the degree of the turbine pressure reduction π_t was higher for the one-stage turbocharging than for the two-stage (π_c/π_t=1,207 compared with 1.17 and 1.16) due to a higher π_c value and turbocharger efficiency of the one-stage system. The higher is the π_c/π_t ratio and correspondingly the ratio of the boost pressure p_c to the turbine inlet pressure p_t, the lower is gas-exchange work which improves engine power and efficiency. The values of the peak combustion pressure p_z and gas temperature at the turbine inlet T_T are safe for the durability of the engine and turbocharger. Also safe reduced rotor speed $n_{r.red}$ and surging margin are obtained, as can be seen from Fig. 4, 5, where the points of joint operation with the engine are plotted on the compressor maps.

For compressors of a two-stage supercharging system, lower values of the compressor adiabatic efficiency (η_c = 0.794 and 0.786) were obtained compared with the one-stage system compressor (η_c = 0.81). This is due to the fact that a modern TK2202 turbocharger designed for the base diesel engine D200 was used for the one-stage charging system. The turbochargers for the two-stage system were developed long ago. If turbochargers of the same level were used for all three charging systems, performance of the gas-diesel engine with the two-stage turbocharging would be better.

In case of the gas diesel engine boosted to p_e=3.2 MPa with the 2-stage charging system, initially, p_z was 27 MPa (this calculation is not presented in the tables), and to reduce it to 22 MPa, the fuel injection advance angle was retarded by 8°, which increased g_e by 4 g/kWh. Therefore we got g_e=187.6 g/kWh compared with 184.5 g/kWh (by 1.5% higher) for the engine boosted to p_e=2.7 MPa. All charging system parameters increased, but no limitation parameters (air excess coefficient α, turbine inlet temperature T_t, maximum rotor speed $n_{r.red}$, surge margin) were exceeded.

Any further boosting of the engine with these turbochargers seems unreasonable because the rotor speeds of both the turbochargers are close to maximum and any further increase of the peak cylinder pressure would require the retard of the fuel injection advance angle which would result in degradation of fuel efficiency.

5. Conclusions

1. Compared with one-stage turbocharging, two-stage turbocharging provides a higher boost pressure, lower boost air temperature and higher charging system efficiency. To obtain the best parameters of the two-stage charging system, a special first-stage compressor with a characteristic tilted to the right is required to ensure a high compressor efficiency in the area of high air flow.

2. From the model line of standard Russian turbochargers, a model TK30-H17 was selected for the 1st stage and model TK23-C011 – for the 2nd stage of the two-stage charging system which enabled to obtain high parameters of the D200 gas diesel engine.

3. The D200 gas diesel engine can be boosted to have a brake mean effective pressure p_e=2.7 MPa with one high pressure turbocharger and with two-stage charging system. In both cases, the engine performance parameters are close, the brake specific fuel consumption g_e was correspondingly 183.0 and 184.5 g/kWh. In the two-stage system, the boost pressure is slightly higher. The critical values of maximal cylinder pressure p_z, exhaust gases temperature T_t and reduced rotor speed n_r were not exceeded, as well as a safe surging margin was obtained.

4. Parameters of the maximally boosted D200 gas diesel engine with the two-stage charging system using the selected turbochargers were forecasted. The maximum value of p_e=3.2 MPa was obtained. Limitation of the value of p_z=22 MPa were fulfilled by retarding the fuel injection advance angle by 8° which resulted in a small drop of the g_e value by 1.5% and finally g_e=187.6 g/kWh was obtained.

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