IOP Conf. Series: Earth and Environmental Science 208 (2018) 012009 doi:10.1088/1755-1315/208/1/012009

Research on Power Enhancement of TCD2015V08 Diesel Engine Based on One-Dimensional Simulation and Analysis

Zhenghu Chen [1], Lian Xie [2], Haikun Shang [2], Jianfeng He [2], Jie Zheng [3, *], Quanshuai Li [2], Shuainan Shi [2]

[1] Baoji Military Representative Office, Baoji, China.
[2] Hebei Huabei Diesel Engine Plant, Shijiazhuang, China.
[3] Zhonghua Communication System Co., Ltd., Shijiazhuang, China.

*Corresponding author e-mail:421618013@qq.com

Abstract. One-dimensional machine modeling of German TCD2015V08 diesel engine is carried out, and the diesel engine is subjected to parameter calculation and one-dimensional simulation analysis. The supercharged air parameters, fuel supply advance angle and compression ratio are analysed in detail under the diesel combustion mode. The impact on performance, combined with the analysis conclusions, proposes a power-enhanced design.

1. Introduction

Improving the power of the engine is a goal that engine researchers have been pursuing. The results of previous studies show that increasing the intake air volume, improving the air flow and combustion in the cylinder are the main ways to improve the engine power. The power that the diesel engine can emit is, in the final analysis, the influence of the amount of air to be inhaled and the air-fuel ratio. The optimal design and matching of the diesel engine boosting system can improve the characteristics of the intake system in a certain range of speed of the diesel engine, thereby increasing the intake of the diesel engine. Gas volume, increase power, improve torque characteristics, reduce fuel consumption and smoke. On the other hand, further optimization and matching design of the fuel supply system can also improve the in-cylinder combustion process and improve engine power. Therefore, optimization of diesel engine intake and injection characteristics is of great significance for improving diesel engine performance.

2. The Introduction of TCD2015V08 Diesel Engine Power Enhancement

The main technical parameters of the TCD2015V08 diesel engine produced by DEUTZ in Germany are as follows.

a) Type: exhaust gas turbocharged water-cooled four-stroke electronically controlled
b) Calibration power / speed: 500 kW/2100 r/min
c) Maximum torque / speed: 2780 N•m/1400 r/min
d) Maximum no-load stable speed: 2310 r/min
e). idle speed: 600 r/min
f) Calibration point fuel consumption rate: ≤245 g/kW•h

The main content of this diesel engine power intensification study, after one-dimensional performance simulation calculation of diesel engine, based on the calibration point intake and fuel supply parameters, considering the structural strength and material characteristics of the whole machine, the highest combustion pressure is 17.5 MPa, the highest vortex Under the limitation of the front exhaust temperature of 750 ° C, the diesel engine was intensively studied to strengthen the calibration power to more than 560 kW.

3. The Establishment of TCD2015V08 Diesel Engine Simulation Model

The working process in the diesel engine cylinder is complicated, and it is a comprehensive process including physical, chemical, flow, heat transfer and mass transfer. In order to obtain the comprehensive performance parameters of the whole machine, the matching of the diesel engine and the turbocharger, the problem is simplified, and the following basic assumptions are made:

1) It is considered that the working state parameter is only a single-valued function that changes with time, and the instantaneous spatial distribution is assumed to be uniform, that is, the pressure, temperature and concentration of each point in the same instantaneous cylinder are equal. It is also assumed that during intake, the air entering the cylinder through the boundary of the system and the residual exhaust gas in the cylinder are instantaneously completely mixed;

2) The working fluid is an ideal gas, which conforms to the ideal gas state equation. Its specific heat, internal energy and helium parameters are only related to gas temperature and gas composition: the flow process of the working fluid flowing into or out of the cylinder is regarded as quasi-stable flow, ie Consider a steady flow within a sufficiently small calculation step;

3) The axial geometry of the pipe is much larger than the radial dimension of the pipe. The axial flow in the pipe should be much larger than the radial flow effect. Therefore, the radial flow effect is omitted, and the flow in the pipe is considered to be one-dimensional. For each flow parameter, it should be understood as the average value of the parameter on the corresponding pipe section.

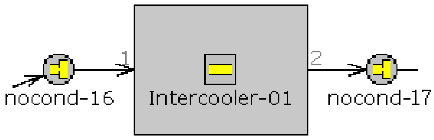
3.1. Parameter Setting and Model Building

According to some main structural parameters of the supercharged diesel engine (see Table 1), the diesel engine is simplified into an intake system, a combustion system (cylinder), an exhaust system, a booster system, an intercooler, and environmental boundaries and corresponding connecting lines. Etc., the calculation model composed of the model, the calculated model of the supercharged diesel engine is shown in Figure 1.

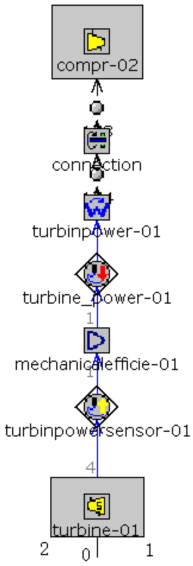
Table 1. Main parameters of diesel engine

Item		Value
Engine characteristic parameters	Compression ratio	17.5
	Fire order	5-4-8-1-3-7-2-6 *
	Cylinder arrangement	V-shaped 90° angle
	Stroke	4
	Number of cylinders	8
Cylinder geometry	Cylinder diameter (mm)	132
	Travel (mm)	145
	Connecting rod length (mm)	262
Fuel injection	number of injectors	8
	Number of orifices	8
	Spray hole diameter (mm)	0.21
Intake and exhaust valve parameters	Intake valve diameter (mm)	46
	Exhaust valve diameter (mm)	42
	Gas distribution phase	IVO: 55.5°CA before top dead center; IVC: 66.5 °CA after the bottom dead center; EVO: 86°CA before the bottom dead center; EVC: 54°CA after top dead center.

*Note: The cylinder number in the firing sequence is shown in Figure 1.



(a) Intercooler model



(b) Supercharger model

Figure 1. GT-power based diesel engine one-dimensional simulation model

The intake and exhaust systems mainly include a booster intake line, an intake volume chamber, an intake manifold, an intake passage, an exhaust manifold, an exhaust manifold, and the like, according to the intake and exhaust pipes of the diesel engine under study. Engineering drawings and 3D structural drawings to determine their structural parameters. GT-Power provides a variety of combustion models including Weber and Double Weber models. The fuel injection amount and the combustion data obtained from the test can be directly input into the heat release law. The compressor and turbine models of the supercharger have simple models and complex models. The complex models must be separately input into the map of the compressor. These parameters are provided by the turbocharger manufacturer. The user can adjust the flow rate, pressure ratio and efficiency of the supercharger as needed. The coefficient meets the requirements of diesel engine performance.

3.2. Combustion model

In-cylinder combustion uses an approximate Weber model. According to the test data of the whole bench test cylinder, the heat release rate curve of the prototype is shown in Figure 2. When modeling, refer to the heat release rate of the same series of diesel engines and adjust the Weber parameters when the model is checked. The parameters of the Weber model are shown in Table 2, and the heat release rate curve is fitted.

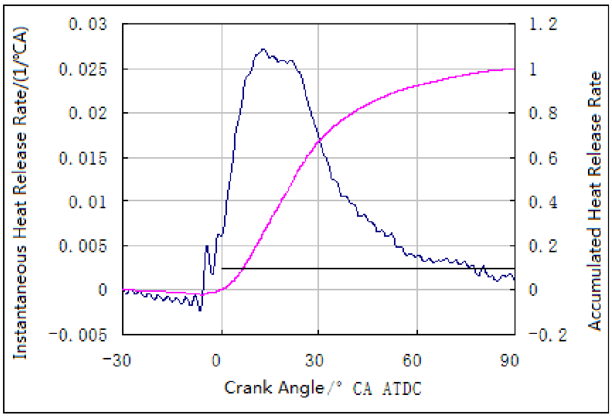


Figure 2. Prototype heat release rate

Table 2. Prototype Weber model parameters

Rotating speed (r/min)	2100	1900	1700	1500	1400	1300	1200	1000
Delayed combustion period (°CA)	14	10	9	8	7.5	8	8	2
Injection time (°CA)	-12.5	-12.1	-11.6	-10	-8.7	-11.9	-11.1	-5.5
Pre-combustion duration (°CA)	3	3	3	5	5	3	3	4
Main combustion duration (°CA)	39	38	36	35	32	30	28	30
Post-firing duration (°CA)	50	50	50	45	45	50	50	52
Pre-combustion ratio	0.001	0.003	0.008	0.01	0.01	0.009	0.01	0.01
Afterburning ratio	0.3	0.3	0.25	0.2	0.2	0.2	0.2	0.2
Pre-combustion index	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
Main combustion index	1	1.2	1.2	1.2	1.3	1.4	1.4	1.4
Afterburning index	1.5	1.5	1.6	2	2	2	2	2

4. Pressurization system enhanced matching research

4.1. Calculation of basic parameters of combustion air

4.1.1. Recirculating oil supply. When the engine power Pe and the rated working speed ne are set, the circulating oil supply amount mf can be calculated by the formula (1):

$$m_f = \frac{\tau \cdot b_e \cdot P_e}{120 \cdot n_e \cdot i} \tag{1}$$

Where: τ is the number of strokes, four-stroke diesel engine τ=4

i is the number of cylinders in the diesel engine be is the specific fuel consumption of the diesel engine, the unit is g/kW.h.

It can be calculated from the calculation that the calibration point be is calculated according to 245 g/kW.h, the maximum torque point is calculated according to the maximum value of 213 g/kW.h, and the maximum value of the circulating oil supply at the calibration point is 0.272 g, and the maximum torque point is cyclically supplied. The maximum amount of oil is 0.275 g.

4.2. Calculation of combustion air flow and boost pressure ratio

The boost pressure ratio is determined according to the required intake pressure, and the intake pressure depends on the circulating oil supply amount mf and the excess air coefficient, and the air-fuel ratio is 14.3, and the expression is (3):

$$\mathrm{P_{in}} = \frac{\alpha \cdot 14.3 \cdot \mathrm{m_f} \cdot \mathrm{T_{in}}}{273 \cdot \rho_0 \cdot \mathrm{V_h} \cdot \eta_v} \approx \frac{0.04 \cdot \alpha \cdot \mathrm{m_f} \cdot \mathrm{T_{in}}}{\mathrm{V_h} \cdot \eta_v} \tag{2}$$

$$\pi_0 = \frac{\mathrm{P_{in}}}{\mathrm{P_0}} \tag{3}$$

Where: Tin is the intake air temperature, unit K; is 273K, air density in 1bar state;

Vh is a single cylinder displacement in m3;

ηv is the efficiency of inflation;

Pin is the pressure of the gas in the intake pipe;

P0 is the pressure before the compressor;

Generally, the civil air diesel engine rated point excess air coefficient is ⩾ 2, the maximum torque point rotation speed is low, the fuel has sufficient time and air mixing, so it can take a lower excess air coefficient, generally = 1.7 ~ 1.9. Under the current combustion theory and technical conditions, the excess air coefficient of the rated point of the diesel engine is generally about 1.6 to 1.8, and the excess air coefficient of the maximum torque point can be reduced to 1.5 to 1.7.

According to this, the required intake pressure of the diesel engine at the rated point and the maximum torque point can be calculated. Considering the resistance in the intake pipe, air cleaner and intercooler of the whole vehicle, the design pressure ratio is increased by 0.015 based on the pressure ratio calculated by the above formula(3), so that:

$$\pi = \pi_0 + 0.15$$

After the boost ratio is determined, the air flow Qair of the corresponding working point can be calculated according to the parameter relationship such as displacement and rotation speed, and the calculation formula is(4):

$$Q_{air} \approx \frac{\pi_0 \cdot n \cdot i \cdot V_h \cdot 1.29 \cdot 273}{T_{in}} \cdot \frac{\eta_v}{120} = \frac{2.935 \cdot \pi \cdot n \cdot i \cdot V_h \cdot \eta_v}{T_{in}} \quad (4)$$

From this, it can be calculated that the calibration point boost pressure ratio is 3.72, the maximum torque point boost pressure ratio is 3.07; the calibration point air flow rate is 3434kg/h, and the maximum torque point air flow rate is 2206kg/h.

4.3. Performance analysis after initial reinforcement

According to the calculation and matching results, the calculation model is used to simulate the performance of the whole machine. The required supercharger compressor and turbine flow parameters are input into the simulation model to simulate the calculated combustion air parameters, and the cycle injection quantity is increased to obtain the performance parameters after initial strengthening of the whole machine. See Table 3.

Table 3. Performance parameters after initial strengthening of the whole machine

Rotating speed (r/min)	2100	1900	1700	1500	1400	1300	1100
Power (kW)	561	546	531	480	431	328	227
Air flow (kg/h)	3346	3003	2616	2159	1839	1396	1027
Cycle fuel injection(g)	0.266	0.277	0.297	0.300	0.291	0.246	0.206
Fuel consumption(g/kW.h)	239	231	228	225	227	234	240
Maximum combustion pressure (MPa)	18.7	18.2	17.7	16.3	15.0	11.9	9.4
Pressure ratio	3.59	3.41	3.19	2.89	2.61	2.09	1.69

The joint operation curve is shown in Figure 3.

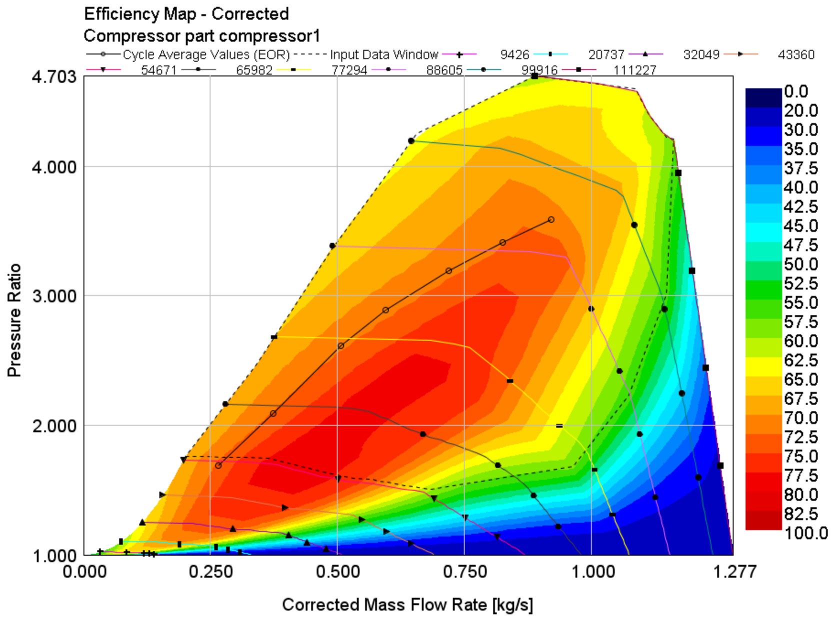


Figure 3. Joint running curve

It can be seen from the simulation calculation that the main performance after matching the enhanced supercharger basically meets the strengthening requirements, but the maximum torque is relatively small compared with the prototype, and the low speed zone is lower than the prototype.

5. Oil supply advance angle enhancement study

Combined with the calculation result of the supercharging system, based on the turbocharger characteristic parameters, comprehensive adjustment can be made, and five kinds of supercharged intake parameter states can be obtained, and then the injection advance angle is changed under each of the supercharged intake parameters. The data of 25 sets of calibration points and maximum torque points are obtained, and the power, the maximum combustion pressure and the fuel consumption rate are comprehensively compared. The calculation results are as follows.
See Table 4 for the comparison of power in different states.

Table 4. Power comparison

Power(kW)	2100 (r/min)				
Advance angle(°CA)	-8.5	-10.5	-12.5	-14.5	-16.5
Status1	528	539	554	565	573
Status 2	535	547	558	572	582
Status 3	536	551	562	574	583
Status4	538	552	565	575	584
Status 5	539	552	566	578	585
Power(kW)	1400 (r/min)				
Advance angle(°CA)	-8.5	-10.5	-12.5	-14.5	-16.5
Status 1	472	480	487	493	497
Status 2	468	477	478	457	443
Status 3	461	457	430	415	399
Status 4	418	403	391	373	376
Status 5	356	325	313	297	281

The comparison of the highest combustion pressures in different states is shown in Table 5.

Table 5. Comparison of the highest combustion pressure

Maximum combustion pressure (bar)	2100 (r/min)				
Advance angle (°CA)	-8.5	-10.5	-12.5	-14.5	-16.5
Status 1	200	202	205	207	212
Status 2	183	187	192	201	205
Status 3	180	184	187	193	201
Status 4	175	177	183	190	196
Status 5	167	170	175	181	187
Maximum combustion pressure (bar)	1400 (r/min)				
Advance angle (°CA)	-8.5	-10.5	-12.5	-14.5	-16.5
Status 1	171	174	180	187	195
Status 2	155	163	165	167	168
Status 3	148	149	150	150	151
Status 4	133	136	138	139	141
Status 5	115	114	112	110	110

See Table 6 for a comparison of fuel consumption rates for different states.

Table 6. Comparison of fuel consumption rate

Fuel consumption rate (g/kW.h)	2100 (r/min)				
Advance angle (°CA)	-8.5	-10.5	-12.5	-14.5	-16.5
Status 1	250	245	242	238	232
Status 2	248	244	240	236	231
Status 3	247	243	239	235	230
Status 4	245	242	238	234	229
Status 5	244	242	238	234	229
Fuel consumption rate (g/kW.h)	1400 (r/min)				
Advance angle (°CA)	-8.5	-10.5	-12.5	-14.5	-16.5
Status 1	233	228	224	221	219
Status 2	234	230	226	224	221
Status 3	236	231	227	225	223
Status 4	237	233	229	226	224
Status 5	239	236	232	230	229

It can be seen from the simulation calculation that the maximum combustion pressure exceeds the limit under the premise of meeting the strengthening requirements, limited to the calibration point and the maximum torque point performance index. Therefore, by selecting a state with a relatively high maximum combustion pressure, an ideal two-group matching combination can be obtained, which is the first state, the advance angle is 14.5° CA and the second state, and the advance angle is 14.5° CA.

6. The study of compression ratio optimization

The main goal of compression ratio optimization is to seek the best combination of each parameter balance under the constraints of design boundary conditions.

Using the one-dimensional performance simulation model, under the above-mentioned two selected ideal combination conditions, taking the calibration point as the research object and changing the compression ratio respectively, the performance analysis can be performed through simulation to obtain the main parameters of the diesel engine under different compression ratios.

The results of the simulation analysis under the first combination are shown in Table 7.

Table 7. Simulation analysis results under the first combination

Rotating speed (r/min)	2100				
Compression ratio	17.5	16.5	15.5	15	14.5
Power (kW)	565	557	555	550	547
Torque (N.m)	2569	2533	2524	2501	2488
Average effective pressure (bar)	22.0	21.7	21.6	21.5	21.4
Intake flow (kg/h)	3507	3540	3549	3569	3580
Cycle fuel injection (g)	0.267	0.265	0.265	0.264	0.264
Fuel consumption rate (g/kW.h)	238	240	241	242	243
Pressure ratio	3.91	3.95	3.96	3.98	3.99
Maximum combustion pressure (bar)	213	192	186	179	171
Vortex front exhaust temperature (℃)	672	675	679	688	693

The results of the simulation analysis under the second combination are shown in Table 8.

Table 8. Simulation analysis results under the second combination

Rotating speed (r/min)	2100				
Compression ratio	17.5	17	16.5	15.5	15
Power (kW)	572	568	565	562	550
Torque (N.m)	2601	2583	2569	2556	2528
Average effective pressure (bar)	22.2	22.0	21.9	21.8	21.7
Intake flow (kg/h)	3369	3386	3393	3412	3430
Cycle fuel injection (g)	0.268	0.268	0.268	0.269	0.264
Fuel consumption rate (g/kW.h)	236	238	239	241	242
Pressure ratio	3.65	3.67	3.68	3.70	3.71
Maximum combustion pressure (bar)	202	187	182	173	168
Vortex front exhaust temperature (℃)	682	689	692	701	715

It can be seen from the calculation results that as the compression ratio decreases, the maximum combustion pressure also decreases. At the same time, the decrease in the combustion rate also causes the exhaust gas temperature to rise, and the increase in the exhaust temperature increases the thermal load of the diesel engine. Therefore, the exhaust temperature is also an important control parameter. As the compression ratio decreases, the power also decreases. The first combined calibration point power quickly drops below 560 kW, but the maximum combustion pressure still exceeds the design limit, so the first combination does not meet the optimization requirements; the two combinations are limited to the calibration point power and the highest combustion pressure, and a compression ratio that satisfies the optimization model can be selected. When the compression ratio is reduced to 15.5, the parameters are within the requirements of the optimization model.

7. Post-enhancement performance simulation analysis

Based on the selected matching technology state, the diesel engine external characteristic parameters are obtained through calculation, see Table 9, the external characteristic curve is shown in Figure 4, and the combined operation curve is shown in Figure 5.

Table 9. Simulation calculation results

Rotating speed (r/min)	2100	1900	1700	1500	1400	1300	1100
Power (kW)	562	547	521	473	443	392	259
Torque (N.m)	2551	2751	2927	3011	3022	2880	2251
Air flow (kg/h)	3412	3090	2701	2239	1979	1605	1131
Cycle fuel injection(g)	0.269	0.282	0.292	0.293	0.290	0.285	0.238
Fuel consumption (g/kW.h)	241	235	229	223	220	227	242
Maximum combustion pressure (MPa)	17.3	17.1	16.7	15.6	14.9	12.8	10.0
Pressure ratio	3.7	3.5	3.3	3.0	2.8	2.4	1.9
Vortex front exhaust temperature (°C)	701	695	679	671	662	676	692

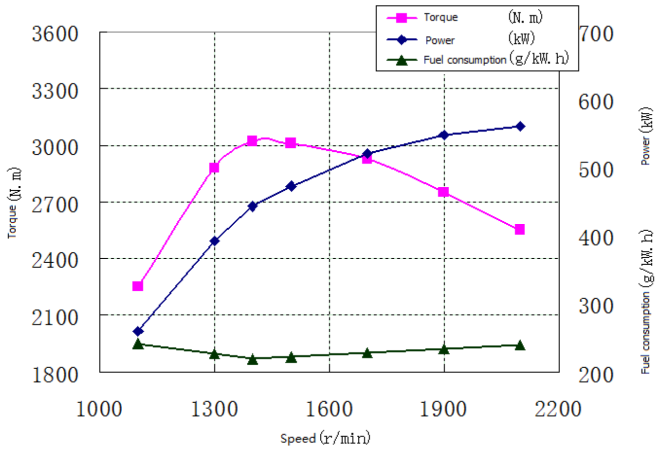


Figure 4. External characteristic curve

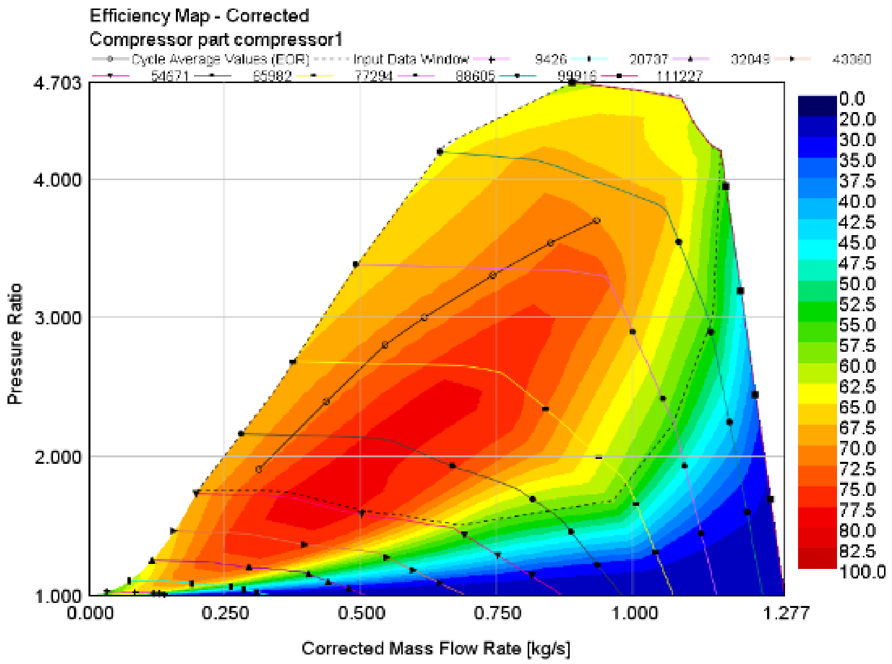


Figure 5. Joint running curve

8. Conclusion

Through the use of GT-power software to complete a one-dimensional simulation model establishment and performance enhancement simulation analysis of a supercharged eight-cylinder diesel engine (TCD2015V08 model), the different turbocharger technical parameters, different fuel supply advance angles and different are analyzed. The compression ratio is selected for the diesel engine. A combined state that meets the research requirements is selected, and the one-dimensional performance simulation analysis is completed. The final calibration power reaches 562 kW, the maximum combustion pressure is 173 bar, and its calibration power, maximum combustion pressure, The vortex front exhaust temperature, compression ratio, and boost pressure (pressure ratio) can meet the requirements of the optimization model, and all the parameters have reached the expected effect of strengthening the research.

References

[1] Cui Y. Pan W. Leylek J H. Cylinder-to-cylinder variation of losses in intake regions of IC engines.SAE 981025 1998.
[2] Wu Feng, Wang Zengquan, Hou xinrong et al. The Capability Research of 8V150 Turbocharged Intercooled Diesel Engine. Chinese Internal Combustion Engine Engineering, 2003 (1): 62-69.
[3] Ugur Kesgin, Study on the design of inlet and exhaust system of a stationary internal combustion engine, Energy Conversion and Management 46 (2005) 2258–2287.
[4] Gao Deming, Gan Haiyan,Wang Feng and so on. Total Design Of G32 Series Diesel Engine. The links of The Article: http://d.g.wanfangdata.com.cn/Conference_3204509. aspx.
[5] Li Xiangrong, Wei Rong, Sun Bogang and so on. Combustion Science and Technology For Internal Combustion Engines[M]. Beijing: Beijing University of Aeronautics and Astronautics Press, 2012
[6] Wang Enbo. Simulation and Experimental study on working process of JX493ZLQ Diesel Engine. University Master degree thesis of Tian jin.2006:1-5.
[7] Ma Yi, Li Jinlong, Hao Baoyu and so on. Diesel Engine Simulation and Optimization Based on GT_Power [J]. Autiomotive Engineer, 2012(12): 27-30.