American Journal of Vehicle Design, 2013, Vol. 1, No. 1, 9-15
Available online at http://pubs.sciepub.com/ajvd/1/1/2

DOI:10.12691/ajvd-1-1-2

Investigation in Adjusting the Parameters of a Diesel Engine Converted to Forced Aspiration Gas Engine

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Received July 01, 2013; Revised July 29, 2013; Accepted July 31, 2013

Abstract Many cities and countries today have programs to convert older, polluting diesel transit buses and trucks to run on clean, economical natural gas. Properly implemented, this is an excellent way to quickly reduce fuel costs, clean up the air and reduce noise with minimum capital costs. A poorly executed conversion program, however, can lead to higher exhaust emissions, much higher fuel consumption, unacceptable power losses, poor durability and high maintenance costs. This article gives the results of our investigation of a forced aspiration gas engine 6GCHN13/14 converted from diesel. The mathematical model of the combustion process specified for the gas engine uses a variable Wiebe combustion factor. Use of the developed characteristic maps has shown improvement of engine power indicators in comparison with that of a naturally aspirated engine, as well as a decrease in average operational emissions with the use of adjustable forced aspiration.

Keywords: *combustion, wiebe combustion factor, mathematical model, natural gas, engine mapping*

Cite This Article: Maamri, Rachid, Fedor Ivanovitsh Abramtshuk, Alexandre Nikolaevitsh Kabanov, Mekhael Sergeevitsh Lipinsky, Yves Dubé, Lotfi Toubal, and Agbossou Kodjo, "Investigation in Adjusting the Parameters of a Diesel Engine Converted to Forced Aspiration Gas Engine." *American Journal of Vehicle Design* 1, no. 1 (2013): 9-15. doi: 10.12691/ajvd-1-1-2.

1. Introduction

With increasing concerns about energy shortages and environmental protection, research on improving engine fuel economy and reducing exhaust emissions has become the major research aspect in combustion and engine development. Due to limited reserves of crude oil, development of alternative fuel engines has attracted more and more concern. Alternative fuels usually imply fuels clean compared to diesel fuel and gasoline fuel in the engine combustion process. The introduction of these alternative fuels is beneficial to slowing down the fuel shortage and reducing engine exhaust emissions.

Converting diesel engines of trucks into gas spark ignited engines is now a subject of actuality [1]. However the execution of this task faces a number of difficulties. For example, the chamber of combustion of an original diesel engine can't support the thermal load created after conversion to gas spark ignition without further modifications. Also the stringent regulation imposed on automobile internal combustion engines concerning gas emissions requires the use in the converted engines of the concept of "lean burning" [1].

Application of this concept causes a decrease in the power of the engine, especially in modes of maximum loads. To compensate these losses of engine power of the converted diesel engine, it is expedient to use supercharging.

Projects to convert diesel engines to spark-ignited forced-aspiration gas engines using this concept of "lean burning" are found all over the world. In Russia such work on a KAP-740 diesel engine has been reported in [2]. In the USA, a similar program is pursued by companies such as John Deere [3], Cummins [4], Detroit Diesel [5], Caterpillar [6] and others. In Europe such gas fuel conversion activity takes place for public transport, as exemplified by Iveco City Class buses [7]. In Ukraine similar work is carried out in LNTU [8], KNAHU [1] and other centres.

Analysis of the literature has shown that in the majority of studies on gas engines converted from diesel engines, forced-aspiration is either used immediately, or planned for the near future.

2. Experimental Setup

The test bench for carrying out the measurements, and also for verifying the validity of the analytical models, is shown in Figure 1. This work was executed in the Gas Laboratory of the Internal Ignition Engines Section in KNAHU. The bench was created around a gas spark ignited engine with supercharging 6GCHN13/14, converted from a diesel engine YMP-236.

2.1. Engine Test Bench

In order to permit a smooth change of drag torque on the engine shaft, during tests, the bench is equipped with an electric loading device MPB-100 (1 in Figure 1). The power from the engine is delivered to the balancing machine via a mechanical gearbox VAZ2103 2 and a cardan joint 3. This gearbox can work in a wide frequency range of rotation of the engine crankshaft. The bench is equipped with an additional air-fan 4, and an additional heat exchanger 5, with which it is possible to model a temperature regime which approaches real running conditions. For the same purpose, the bench also includes a noise muffler 6 and a resonator 7 from the regular exhaust system of a Daewoo Model Sens. A damper valve 8 is installed to measure the exhaust system resistance.

In order to explore a variety of gases, , the bench is equipped with a fourth-generation gas system which includes: several cylinders with a variety of gases 9, a two-level gas pressure reducer 10, high and low pressure lines, a rail with gas injectors 11 and a control panel 12. The fuel gas is selected by a series of switches 13 on board 14.

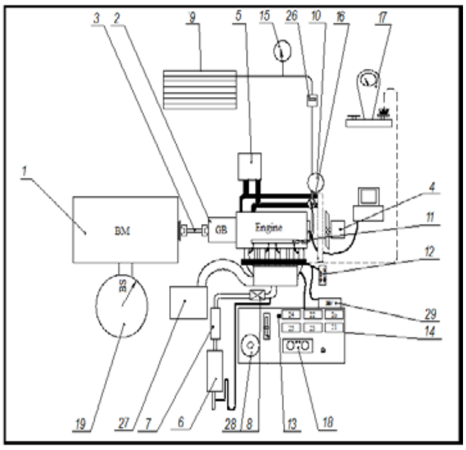


Figure 1. Engine test bench

Investigations in the field of operator workplace ergonomics [9] have shown that the greatest informational content is provided by visual observations of low-frequency processes. Hence a series of registering devices are included, such as damped analogical sensors (15, 16, 17, 18, 19), and a LED digital display (20 - 25).

The bench also includes a gas flow meter (26) and an air flow meter (27) to measure these input parameters.

The stand is also equipped with an oil temperature measuring device (21) and an engine load control system (28).

Finally, an engine electronic control unit 29 is mounted on the test bench.

The stand is equipped with all the necessary equipment to determine actual and indicated indicators of power and fuel economy and toxicity.

3. Mathematical Model

To reduce the number of tests, we used a computational and experimental study. To implement this approach, the test is replaced by a mathematical model. This model is verified and confirmed by a series of tests. Then, using the verified model, we performed the calculations.

The mathematical model chosen is the model of Wiebe [10] with the specified calculation of exponent m in the Wiebe function, as used by Kabanov [11].

As preliminary studies have shown, the charging pressure of the fuel gas influences the form of the characteristic of thermal release. However as [11] did not allow for charging pressure variability, the equation for Wiebe exponent m as used there for calculating the thermal release characteristic cannot be used in its original form in the present work, and must be modified to include the influence of the charging pressure. Besides, forced aspiration also influences burn duration.

As a result of these considerations and on the basis of preliminary experimental studies, the following equations for a variable Wiebe exponent m_{var} and burn durations φ_z are proposed.

$$m_{\text{var}_i} = 10.639 \cdot \bar{\varphi}_i \cdot A \cdot B \cdot C \cdot D \cdot E \cdot F + J \quad (1)$$

where, m_{var_i} is the variable Wiebe exponent of the combustion phase i,

$$A = (\theta + 18) / 40 \quad (2)$$

$$B = \alpha + 0.00025 \quad (3)$$

$$C = (0.005\eta_v + 0.005) / 0.01 \quad (4)$$

$$E = (n + 1100) / 3000 \; (5) \quad (5)$$

$$F = (\pi_c + 0.001) / 0.95 \quad (6)$$

$$J = -28.025\bar{\varphi}_i^2 + 98.045\bar{\varphi}_i^3 - 156.86\bar{\varphi}_i^4 + 86.88\bar{\varphi}_i^5 \quad (7)$$

where $\bar{\varphi}_i$ is the relative burn angle of the combustion phase i, $\bar{\varphi}_i$ = 0…1 (0-100% burn duration); n is the frequency of rotation of the crankshaft of the engine, in min-1; α is the air–fuel excess ratio; η_v is the volumetric efficiency; θ is the spark advance, crank angle before the top dead center (TDC), and π_c is the rate of charging pressure rise. Moreover, the burn duration:

$$\begin{aligned}\varphi_z = 28.25(0.182 \cdot \alpha - 0.045 \cdot \theta + 13.233 \cdot 10^{-4} \cdot n \\ - 0.1258 \bullet \eta_v - 0.576(\pi_c - 1) + 0.152)\end{aligned} \quad (8)$$

where π_c or CPR is total pressure ratio is the ratio of the compressor discharge air pressure to the ambient air pressure (the ratio of the air total pressure exiting the compressor to the air pressure entering the compressor).

$$\pi_c = \frac{p_{02}}{p_{01}} = f_{\pi_c}\left(\frac{N_{tc}}{\sqrt{T_{01}}}, \frac{\dot{m}_c\sqrt{T_{01}}}{p_{01}}\right) = (T_{02} / T_{01})^{\gamma/\gamma - 1} \quad (9)$$

The total temperature ratio T_{02} / T_{01} across the compressor is related to the pressure ratio by the isentropic flow equations.

p_{01} : air pressure entering the compressor.

p_{02} : air total pressure exiting the compressor.

T_{02} : total temperature exiting the compressor.

T_{01} : temperature entering the compressor.

γ : ratio of specific heats.

As the concentration of toxic components in the exhaust gases cannot be calculated with sufficient accuracy with the model of Wiebe because the impossibility of determining the temperature distribution in the cylinder, a two-zone model as described in [12] has been added in our model. The advantage of the two-zone model is that it allows not only the calculation of the concentrations of the nitrogen oxides NOx in the exhaust gases with sufficient accuracy, but also to obtain the data necessary for the calculation of the possibility of detonation occurring, and its quantitative estimation. The compression ratio choice is determined by two factors: the mechanical durability of the engine components and the probability of detonation occurring.

4. Choice of Compression Ratio and the Maximum Charging Pressure

The compression ratio choice is determined by two factors: the mechanical durability of the engine components and the probability of early detonation occurring at maximum loads.

The compression ratio of the original diesel engine is much higher than the modified diesel engine. Which gives the forced aspiration gas engine parts a high reliability. Therefore, the choice of compression ratio is influenced only by the second factor.

Autoignition and knock (the sound of an auto-ignition initiated pressure wave inside the engine cylinder) occurs when end gases release sufficient amount of energy above a specific threshold.

An appropriate criterion for knock in spark ignition engines must be based on the total energy released within the end gas due to auto-ignition reaction activity per unit of the corresponding instantaneous volume. Such a specific energy release needs to be compared for any engine, fuel and operating conditions to the corresponding total energy release that would take place normally due to regular flame propagation. The latter needs to be related to the total cylinder size and volume. A dimensionless knock criterion k_c described in [13] can thus be defined as the ratio of the following two relative energy releases during the course of the combustion process:

$$k_c = \frac{Energy\ released\ by\ gas\ reation / V_t}{Energy\ to\ be\ released\ by\ combustion / V_0} \quad (10)$$

where the numerator represents the total energy released by the unburned end gas pre-ignition chemical reaction activity per unit of the instantaneous charge volume (Vt), while the denominator is the total energy that can be released normally through flame propagation over the whole cycle per unit of cylinder swept volume (V_0). Vt, the instantaneous volume at time t, is the sum of the combustion chamber volume, the volume displaced by the piston (depending on the crank angle) and mechanical deformations produced by the gas pressure and inertial efforts; V_0, the cylinder volume swept by the piston displacement, is the full stroke movement of the piston.

The energy released by the end gas self-reactions is defined as:

$$H_{ut} = \int_{t_{st}}^{t} dE_u = (h_{st} - h_t) m_u \quad (11)$$

where h is the enthalpy of the mixture per unit of mass $(h = \sum c_i h_i)$ and m_u is the mass of the unburnt zone which is the end gas region at any instant of time t. Subscripts st and t indicate the values at the spark discharge and at any instant of time, respectively.

The energy released due to normal combustion can be simplified as:

$$H_o = h_o m_o \quad (12)$$

where h_o is the effective heating value of the fuel and m_o is the initial or total mass.

Since the spark timing is usually set such that much of the combustion process takes place in the vicinity of the top dead center position so as to achieve optimum efficiency and power $V_t \approx V_c (clearance\ volume)$, then the volume ratio VR ,where: $VR = V_0 / V_t$, can be approximated to (ε – 1). The ratio is calculated by the following formula:

$$\varepsilon = \left({V_0}/{V_c} \right) + 1 \approx \left({V_0}/{V_t} \right) + 1 \quad (13)$$

Consequently, equation 10 may be written as:

$$K_c = \frac{(h_{st} - h_t) m_u}{h_o m_o} (\varepsilon - 1) \quad (14)$$

The knock criterion of equation (10) can be used to test for the onset of knock and to represent effectively engine knock intensity in gas-fueled spark ignition engines.

The example described in [13] and other similar results show that the operating condition for the knock limit may be determined theoretically by assuming a value of 1.5 for maximum knock criterion value at borderline knock operation. Excess of this value means possible detonation presence at the end of the on the regime, and its intensity is directly proportional to the value K_{cmax}. The detonation occurring is influenced by two major factors: compression ratio ε and rate of charging pressure rise π_c.

Supercharging is designed to increase the density of the air into the cylinders by increasing its pressure in the intake manifold. Engine power thus increases as the work cycle for a supercharged engine is much superior to that of a naturally aspirated engine. However the compression ratio should not be excessive so as to avoid reaching maximum cycle pressures which may be too high. Research [6] has shown that it is possible to obtain the best engine power if the compression ratio is reduced, thus increasing charging pressure to a maximal value. The design of the turbocharger and the mechanical durability of parts of the combustion chamber can reach this value.

Calculations using the criterion K_c have shown that at the maximum rate of charging pressure rise provided by a TKP-9-12-07 turbo-compressor on regimes of maximum load (π_{c_max} = 1.45), a compression ratio ε = 11.8 allows to operate without premature detonations.

5. Multi-criteria Optimization of Adjusting Parameters

The resolution of the problem of optimizing the adjusting parameters of a 6GCHN13/14 engine was by means of the technique of parameters space using Soble's grids as described in [14].

The resolution of this problem assumes a search for a compromise between power, combustible economy and toxicity of the engine emissions. Therefore we used the following as criteria of quality of the working process: effective power N_e, kW; specific emissions of nitrogen oxides g_{NOx}, gr / (kW·h), and the thrust specific fuel consumption (TSFC) or sometimes simply the specific fuel consumption (SFC) [g_e is the ratio of the fuel consumption [gr/h] to the brake power [kW]]. SFC is an engineering term that is used to describe the fuel efficiency of an engine design with respect to thrust output. It allows the efficiency of different-sized engines to be compared directly.

The engine design assumes that the natural gas will be used in a regime of higher limits of lean mixtures, which can partially implement a quality control of mixture. That is to say, engine regulation is supposed to be carried out not only with the throttle position, but also with α (the air-fuel excess ratio) and the charging pressure.

Therefore the following parameters will be subject to variation in the optimization search: air–fuel excess ratio α, spark advance θ, crank angle (deg) before TDC, crankshaft frequency of rotation n, min^{-1}, throttle position angle φ_{tr}, % and rate of charging pressure rise π_c.

The ranges of change of the varied parameters are shown in Table 1.

Table 1. Range of variation of engine parameters for optimizing engine performance

α	θ	n	π_c	Φ_{tr}
–	crank angle (deg) before TDC	min^{-1}	–	%
1	5	800	1	0
1.5	40	2100	1.45	100

The ranges of change of variable parameters for solving the optimization problems of regulation parameters listed in Table 1 and there are selected from the following considerations.

For α < 1 the fuel burns only partially and, as a consequence, the emissions of CO and CH increase strongly. For α > 1.5, the NOx concentration in the exhaust gases decreases significantly and CH emissions suddenly increase. However because of a sudden increase in cyclic instability of the operating process and an increase in frequency of pre-ignition, the power and the fuel economy of gas engines decrease considerably.

The limits for change of spark advance θ, the crank angle (deg) before TDC, was selected on the basis of the experimental data received as a result of preliminary tests on the gas engine 6GCHN13/14.

The range for change of the frequency of rotation of the crankshaft n, min-1 was decided from considerations of guarantee of stable working conditions of the engine at any combination of other factors.

The throttle positions go from fully closed (φ_{tr} = 0 %) to fully open (φ_{tr} = 100 %). The rate of aspiration pressure rise π_c (due to transporting the fuel-air mixture through the compressor) varies from total absence of forced aspiration (π_c = 1) to the maximum rate of aspiration (π_c = 1.45).

6. Results of the Optimization Search and Construction of the Characteristic Maps

The analytical and experimental study carried out has shown that operating the gas engine without forced aspiration allowed it to reach the nominal power regime of the basic unmodified diesel engine N_e = 130 kW only at $\alpha \approx 1$.

The strength margins of the engine allows an increased charging pressure in a nominal regime to π_c = 1.45, allowing it to reach a theoretical power of 180 kW at α = 1. However this leads to a thermal overstrain of parts of the engine and pre-detonation occurring.

To reduce the influence of these last factors, the engine must be operated at a leaner mixture. Research has shown that to assure the engine operating without pre-detonation in nominal regime, it is necessary to make the mixture lean to $\alpha \approx 1.3$. This operational limit at π_c = 1.45 will produce power N_e = 130 kW at n = 2100 min^{-1} and torque M_e = 670 N·m at n = 1500 min^{-1}.

NO_x emissions in regimes of maximum load at π_c = 1.45 and $\alpha \approx 1.3$ show a decrease of 35-45 % in comparison with operating at π_c = 1.45 and $\alpha \approx 1$.

So, reducing the speed at the maximum relative load (P_e = 100%) is appropriate to carry through the increase of excess air ratio to $\alpha \approx 1.5$. The effectiveness of ignition at high values of α provides high concentration of fuel near the spark plug provided by the forced aspiration.

So, load reduction to $\bar{P}_e \approx 0.6$ is appropriate to carry through the increase of α. When π_c = 1.45, α can be increased to α ≈1.4... 1.6. At further increase in α, appear misfiring.

Further reduction of the load is required to reduce π_c. This requires a small increase in α, proportional to the reduction of π_c Moreover, the increase in α in these conditions required due to the deterioration of the ignition and combustion at small value of $\bar{P}_e$.

So, at the minimum idle speed n = 800 min^{-1}, the value of α is 1.05. This approach can reduce NO_x emissions on medium engine speed on 60…70 %, compared to the naturally aspirated version, operating at low values of α.

Thus, on the engine 6GCHN13/14 the mixed regulation of power is offered to use: by means of change of throttle opening rate and α change. Besides, there is an appearance of third operating factor – rise of pressure charging rate π_c.

Figure 2 to 5 show the characteristic maps for the optimized engine performance, namely the dependence of the respective parameters α, π_c, φ_{tr} and θ respectively at n and $\bar{P}_e$. These data will serve as reference values for a microprocessor control system of the engine.

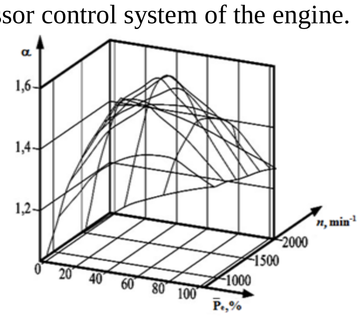


Figure 2. Characteristic map for α

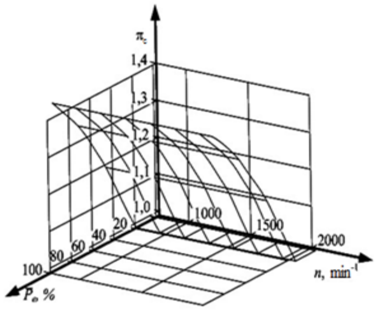


Figure 3. Characteristic map for π_c

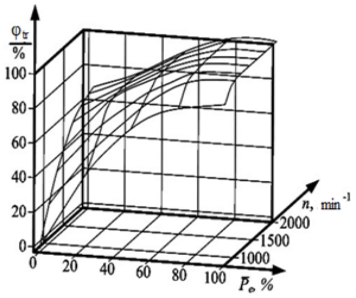


Figure 4. Characteristic map for φ_{tr}

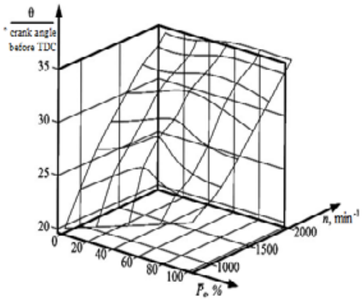


Figure 5. Characteristic map for θ

7. Results of Tests of the Engine 6G4SS13/14

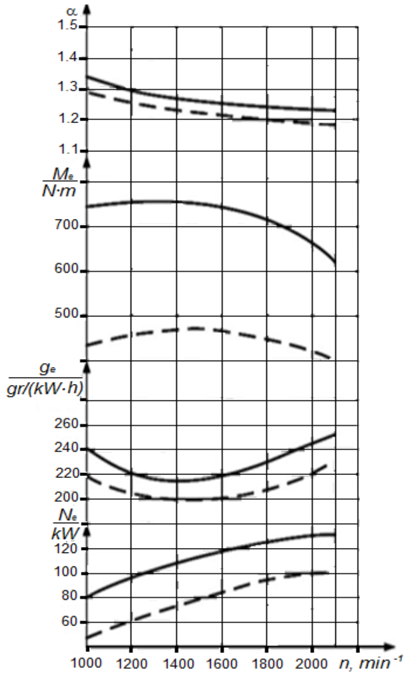
The results of tests on engine 6GCHN13/14 under control of a microprocessor control system embodying the results of Figures 2 to 5 are shown in Figure 6. The comparison of the external high-speed characteristics (EHCE) – as described by the parameters α, N_e, g_e and M_e as a function of n - is between the forced-aspiration engine and the natural-aspiration engine configurations.

Forced aspiration has allowed engine 6GCHN13/14 to reach the power characteristics of a basic naturally-aspired diesel engine. So, on a nominal regime effective power N_e increases by 26 % (about 100 kW to 135 kW). The frequency of rotation corresponding to maximum M_e, have changed from n_{Memax} = 1500 min^{-1} to n_{Memax} = 1400 min^{-1}. The maximum effective torque corresponding to this frequencies, has increased by 36 % (about 480 N·m to 750 N·m).

From Figure 6, it is visible that at use of forced aspiration on regimes of external high-speed characteristics, N_e increases on 26…31 %, M_e: on 35…41 %, although specific effective fuel consumption g_e increases on 7…9%. Despite the rise of power, α increases lightly on 3…4 %.

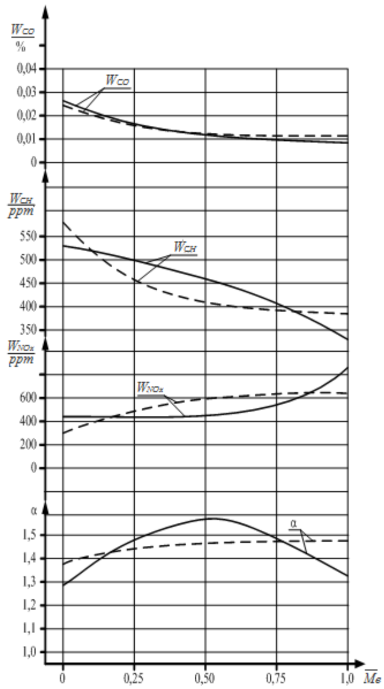
When converting the average power diesel engine to gas engine, adjustable forced aspiration can be used not only for the preservation of the power at the same level as the base engine, but also to reduce emissions of nitrogen oxides.

In Figure. 7 and Figure 8, we compare the emissions from the naturally aspirated engine 6GCH13/14with those of the forced aspiration engine 6GCHN13/14.



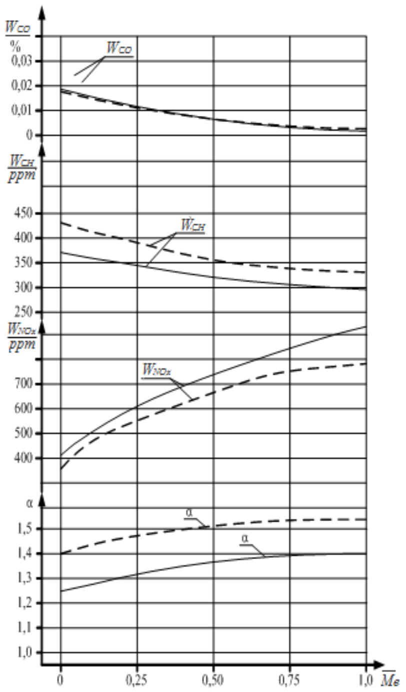
———— – forced aspiration;
– – – – – – natural aspiration.

Figure 6. The external high-speed characteristic of the engine 6GCHN13/14:



– – – – – – naturally aspiration;
———— – forced aspiration

Figure 7. Emissions variation of the engine 6GCHN13/14 by load (n = 1500 min^{-1}):



– – – – naturally aspiration;

———— – forced aspiration

Figure 8. Emissions variation of the 6GCHN13/14 by load (n = 2100 min^{-1}):

The increase of α promotes decrease in in-cylinder temperatures of the engine on 5…10 % on all regimes of EHCE. It allows to lower nitrogen oxides NO_x emissions on EHCE regimes on 8…15 %. Emissions of CO and CH are in both engines practically at identical level.

On partial regimes, charging pressure decreases, in parallel α increases. Besides, the mixed regulation of a mixture on partial regimes modes is carried out. Owing to it, on partial regimes, NO_x emissions decrease on 19…36 %. On regime of idle, the difference in NO_x emissions practically is absent.

8. Determination of the Average Emission of Toxic Compounds, Simulating with the 13-mode, Steady-state ESC Test Cycle

The ESC test cycle [15] has been introduced, for emission certification of heavy-duty diesel engines in Europe. The ESC is a 13-mode, steady-state test.

The ESC test is characterized by high average load factors and very high exhaust gas temperatures.

The choice of speeds to determine the test modes n_A, n_B, n_C is implemented with the help of auxiliary speeds n_{hi} and n_{lo}. The high speed n_{hi} is determined by calculating 70% of the declared maximum net power. The highest engine speed where this power value occurs (i.e. above the rated speed) on the power curve is defined as n_{hi}. Here n_{hi} = 2200 min^{-1}. The low speed n_{lo} is determined by calculating 50% of the declared maximum net power. The lowest engine speed where this power value occurs (i.e. below the rated speed) on the power curve is defined as n_{lo}. Here n_{lo} = 1500 min^{-1}.

The engine speeds to be used during the test are then calculated from the following formulas:

$$n_A = n_{lo} + 0.25(n_{hi} - n_{lo}) \quad (15)$$

$$n_B = n_{lo} + 0.5(n_{hi} - n_{lo}) \quad (16)$$

$$n_C = n_{lo} + 0.75(n_{hi} - n_{lo}) \quad (17)$$

In accordance with the dependencies n_A = 1675 min^{-1}, n_B = 1850 min^{-1}, n_C = 2025 min^{-1}.

Each mode is characterized by the mode significance factor δ_s , demonstrating the conditional relative proportion of time during which, the engine work at this mode.

Tests are conducted at low idle speed (XX) and at 4 values of the relative torque: 0.25, 0.5, 0.75, and 1.0.

The results are shown in Table 2.

Table 2. The results of tests with forced aspiration engine 6GCHN13/14 with the ESC test cycle

№	δз	$\overline{M}_e$	N	NO_x	CO	CH
–	–	%	min^{-1}	g /(kW·h)	g/(kW·h)	g/(kW·h)
1	0.15	0	XX	0.32	0.96	2.1
2	0.08	1.0	n_A	3.05	1.22	1.24
3	0.1	50	n_B	2.52	1.22	1.12
4	0.1	75	n_B	2.74	1.19	1.16
5	0.05	50	n_A	2.44	1.13	1.15
6	0.05	75	n_A	2.66	1.17	1.16
7	0.05	25	n_A	1.86	1.16	1.31
8	0.09	100	n_B	3.8	1.15	1.19
9	0.1	25	n_B	1.92	1.12	1.05
10	0.08	100	n_C	3.55	1.15	1.1
11	0.05	25	n_C	2	1.04	0.94
12	0.05	75	n_C	2.82	1.11	1
13	0.05	50	n_C	2.68	1.2	1.07

Average operational emissions of toxic components, g / (kW · h), defined as the sum of products of emissions of toxic components in 13 modes cycle on significance factor of these regimes.

$$\overline{g} = \sum_{i=1}^{13}(g_i - \delta_{si}) \quad (18)$$

where,

$\overline{g}$ – emissions components on regime;

δ_s – significance factor of the regime.

According to the relation (18), the average operational emissions of toxic components of the engine 6GCHN13/14, g / (kW · h): $\overline{g}_{NO_x}$ = 2.36; $\overline{g}_{CO}$ = 1.13; $\overline{g}_{CH}$ = 1.27.

Table 3. The results of tests naturally aspirated engine 6GCH13/14 with the ESC test cycle

№	δ3	$\overline{M}_e$	n	NO_x	CO	CH
–	–	%	min^{-1}	g/(kW·h)	g/(kW·h)	g/(kW·h)
1	0.15	0	XX	0.44	0.98	2.16
2	0.08	1.0	n_A	4.24	1.24	1.28
3	0.1	50	n_B	3.50	1.24	1.15
4	0.1	75	n_B	3.81	1.21	1.19
5	0.05	50	n_A	3.39	1.15	1.18
6	0.05	75	n_A	3.70	1.19	1.19
7	0.05	25	n_A	2.59	1.18	1.35
8	0.09	100	n_B	5.28	1.17	1.23
9	0.1	25	n_B	2.67	1.14	1.08
10	0.08	100	n_C	4.93	1.17	1.13
11	0.05	25	n_C	2.78	1.06	0.97
12	0.05	75	n_C	3.92	1.13	1.03
13	0.05	50	n_C	3.73	1.22	1.1

According to the relation (18), average operational emissions of toxic components of the engine 6GCH13/14, g / (kW · h): $\overline{g}_{NO_x}$ = 3.28; $\overline{g}_{CO}$ = 1.15; $\overline{g}_{CH}$ = 1.31.

Results of tests of the gas engine 6GCHN13/14 with use of 13-mode cycle ESC have shown following average operational emissions of normalised toxic components, gr/(кW·h): emissions of nitrogen oxides, $\overline{g}_{NO_x}$ = 2.36; emissions of carbon monoxide, $\overline{g}_{CO}$ = 1.13 and hydrocarbon emissions, $\overline{g}_{CH}$ = 1.27. Thus average operational emissions of normalised toxic components the gas naturally aspired engine 6GCH13/14 have show, gr/(кW·h) [9]: $\overline{g}_{NO_x}$ = 3.28; $\overline{g}_{CO}$ = 1.15; $\overline{g}_{CH}$ = 1.31.

Decrease in average operational emissions at use of adjustable charging on the gas engine 6GCH13/14 in comparison with a naturally aspired engine has show: on NOx = 28%, on CO = 10 % and on CH = 18 %.

9. Conclusions

Analysis of the literature showed that the use of adjustable forced aspiration to the gas engine can eliminate deficiencies of the concept of a "poor combustion", retaining its advantages.

Tests on the diesel-to-gas converted engine 6GCHN13/14 with forced aspiration and using the characteristic maps developed in part 6 and summarized in Figures 2 to 5 have shown a definite improvement of the engine power indicators in comparison with the naturally-aspired configuration: that can be summarized as a 7-9% improvement in fuel consumption, 26-31 % in Ne, 35-41 % in Me and a 10-28% reduction in noxious gas emissions.

Tests of the gas engine 6GCHN13/14 on 13-regime cycle ESC have shown that decrease in average operational emissions at use of adjustable charging on the gas engine 6GCHN13/14 in comparison with a naturally aspired variant has give: on NOx: 28 %, on CO: 10 %, on CH: 18 %.

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