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ANALYSIS OF CALCULATED CYCLES PARAMETERS IN CASE OF NATURAL GAS SUPPLY AND DIESEL ENGINE

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The paper is dedicated to comparative analysis of cycles of engines supplied with natural gas having diesel compression ratio and quality power control with the basic diesel engine cycle. Two types of gas engines were investigated: with internal mixture formation and glow plug air-gas mixture ignition; with external mixture formation and air-gas mixture ignition with the help of pilot diesel fuel injection. Calculation results by external speed characteristics demonstrated that the gas engines were at least as good as the base diesel engines as regards to power and fuel efficiency. At the same time they have lower mechanical and thermal loads and considerably lower CO_2 emissions.

Natural gas powered engine, diesel engine, analysis of calculated cycles, fuel efficiency, CO_2 emissions.

Introduction

At present time, research works aimed at the conversion of diesel engines to natural gas are very important. This is due to the fact that, first of all, the proved resources of natural oil on our planet are depleted faster than that of natural gas, and consequently the price of the gas fuel is lower compared to the price of the oil fuel (in Russia – approximately twofold). Moreover, a lower content of carbon in the gas fuel ensures CO_2 emission reduction. There are many ways of organizing a working process of gas supplied engines each having its advantages and disadvantages. Therefore it is very important for designers to know what they can expect from implementation of one or another working process in their engine used for its specific application.

The goal of calculations

The goal of calculated comparative analysis of cycle parameters of a diesel engine and engines supplied with natural gas having diesel compression ratio and internal and external gas-air mixture formation was to evaluate the possibility of achieving at least not lower fuel efficiency in engines supplied with natural gas than in the base diesel engine.

For cycle parameters calculations, we used three versions of the four-stroke V8 turbocharged KAMAZ engine having cylinder diameter D=120 mm, piston stroke S=120 mm and compression ratio 17.0:

1. The base diesel engine;
2. Engine supplied with natural gas, external mixture formation and gas-air mixture ignition by a glow plug;
3. Engine supplied with natural gas with internal mixture formation and gas-air mixture ignition with the help of pilot fine atomized diesel fuel injection.

Method and conditions of making calculations

Cycle parameters of the engines were calculated according to the method indicated in (Khatchiyan *et al.*, 2001) based on the first law of thermodynamics. Empirical formula of I.Vibe (Vibe, 1962) was used for heat release calculation and empirical formulas of G.Woschni (Woschni, 1967) and A.Annand (Annand, 1963) were used for heat losses calculation into the cylinder walls. The temperatures of the cylinder surfaces (fixed cylinder head and piston crown surfaces and variable cylinder liner surface) were also calculated by empirical formulas. The calculations were made using the following conditions: boost pressure and exhaust backpressure p_k=$p_т$=0.25 MPa, engine speed n=2200 rpm, boost air temperature T_k=330 K, air excess coefficient α=1.55, peak combustion pressure $P_{z\ max}$ less than 15 MPa. The computer model was searching for the optimal ignition advance angle $\Theta_{ig\ adv}$ automatically. If the peak combustion pressure exceeded the critical value, the ignition advance angle was retarded.

Results of calculations of cycles

Table 1 shows the results of calculations for the base diesel engine (line 1), for the gas engine with external mixture formation and air-gas mixture ignition with the help of pilot portion of a fine atomized diesel fuel (lines 2 and 3) and for the gas engine with internal mixture formation and air-gas mixture ignition by a glow plug (line 4).

In case of the gas engine with external mixture formation and air-gas mixture ignition by a pilot portion of a fine atomized diesel fuel injection, the cycle calculations were made using two heat transfer functions – A.Annand (line 2) and G.Woschni (line 3).

For all the engines investigated, heat release was calculated by the equation of I.Vibe. For the base diesel engine, we used the two-stage heat release formula and accepted the heat release duration in the initial phase $\varphi_{z\ in}$ = 6 crankshaft rotation degrees and in the main phase – φ_z = 60 crankshaft rotation degrees, the heat release character parameter by I.Vibe in the initial phase of heat release m_{in} = 0.5 and in the main phase – m = 1, a portion of heat during the first phase of the heat release A_1= 0.15. For natural gas engines, we used a one-stage heat release formula and accepted the heat release duration φ_z = 45 crankshaft rotation degrees and the heat release character parameter by I.Vibe m = 2. In Table 1, the following parameters are also indicated: $\Theta_{ig\ adv}$ – ignition advance angle, G_{inj} – cycle fuel delivery, T_{fix} – average temperature of fixed cylinder head and piston surfaces facing the working medium, T_{lin} – average temperature of the liner surface, γ_{res} – residual gases coefficient, η_v – volumetric gas supply efficiency, Q_w – heat losses into the cylinder walls, Q_{cyc} – heat released into during the cycle, p_i – mean indicator pressure, $(dp/d\varphi)_{max}$ – maximum pressure rise speed, $\alpha_{ht\ avr}$ – average heat transfer coefficient, p_z – peak combustion pressure, T_{res} – heat transfer resultant mean temperature of the charge.

Analysis of the cycle parameters calculation

Comparing the calculation results of the diesel engine and gas engine cycles (see Table 1, lines 1, 2 and 3), we can see that using the formula of A.Annand for heat exchange modelling gives considerably lower heat losses into cooling media (Q_w). The average value of the heat transfer coefficient ($\alpha_{ht\ avr}$) was lower and heat transfer resultant mean charge temperature (T_{res}) was higher. The influence of the average value of the heat transfer coefficient prevailed and heat losses into the cooling medium were higher when calculating using the G.Woschni formula. This was the reason of getting lower indicator efficiency – η_i (approximately by 2%). The difference in the mean indicated pressure (p_i) was even smaller. This was due to a high value of the volumetric efficiency (η_v) and the quantity of heat inputted into the cycle (Q_{cyc}). It is quite possible that when calculating by the G.Woschni formula, the incoming charge heating during the intake process is lower.

Comparison of the results of calculations of the gas engine cycles in case of external mixture formation with the results of calculations of the diesel engine cycles (see Table 1, line 1) demonstrates quite close values of indicated efficiency (η_i): 0.4758 – in case of the diesel engine and 0.4757 – in case of the gas engine. The average value of the indicated efficiency selected by two calculations (lines 2 and 3) was chosen. At the same time, the mean indicated pressure (p_i) was considerably higher in case of the diesel engine (almost by 12%). This difference can be explained by two reasons:

The first reason is a lower filling of cylinders with air when gas is supplied through the intake system. One should bear in mind that the differences in the volumetric efficiency (η_v) do not characterize the differences in filling with air because in case of the gas engine, volumetric efficiency is calculated by the mixture quantity rather than by the air quantity. The second reason of differences in p_i is lower caloric value of the gas-air mixture (Hu/l_0) in case of using natural gas – 2.88, while for diesel engine Hu/l_0 is 2.923.

You can see the results of calculations of the gas engine cycle with internal mixture formation in line 4 of Table 1. In this case, heat transfer in the gas engine, as well as in the base diesel engine, was calculated by the G.Woschni formula. Heat losses into the cooling medium (Q_w) turned out to be close. The relative heat losses when supplying natural gas into the cylinders were slightly higher (approximately by 0.3%). A bit higher value of indicated efficiency (η_i) when using natural gas (approximately by 1%) was caused evidently by the prevailing influence of a lower heat release duration. The mean cycle indicated pressure was nevertheless higher when running on diesel fuel as a result of a higher value of Hu/l_0 (by 1.5%).

Table 1. Results of engine cycles calculations

	Engine type	Heat release calculation	Heat exchange calculation	Mixing	φ_z (CRA)		A_1	$\Theta_{ig\ adv}$ (CRA to TDC)	G_{inj} (mg/cyc)	T_{fix} (K)
					m_{in}	m				T_{lin}, (K)
1	Diesel engine	Two-phase by Vibe	Woshni	Internal	60		0.15	3	149.4	495.2
					0.5	1				430.0
2	Gas engine	One-phase by Vibe	Annand	External	45		0	5	113.1	495.2
					-	2				430.0
3	Gas engine	One-phase by Vibe	Woshni	External	45		0	5	114.8	495.2
					-	2				430.0
4	Gas engine	One-phase by Vibe	Woshni	Internal	45		0	9	123.8	495.2
					-	2				430.0

Table 2. Results of engine cycles calculations (continued)

	γ_{res}	η_v	Q_w (J)	Q_{cyc} (J)	p_i (MPa)	η_i	$(dp/d\varphi)_{max}$ (MPa/^{o})	$\alpha_{ht\ avr}$, (W/m^2.K)	p_z (MPa)	T_{res} (K)
1	0.038	0.943	772	6352	2.2268	0.4758	1.64	722	15.760	1122.8
2	0.048	0.916	504	5604	1.9848	0.4802	0.23	475	13.057	1105.7
3	0.05	0.929	656	5686	1.9763	0.4713	0.25	643	12.980	1013
4	0.039	0.941	769	6185	2.1918	0.4809	0.45	718	16.535	1193.5

Calculations of a joint operation of an engine with a turbocharger

Calculations of external speed characteristics of the base diesel engine and gas engine having the diesel compression ratio, quantity power level control and internal mixture formation using variable geometry turbocharger were carried out by a computer program modeling a joint operation of a diesel engine and a gas engine with a turbocharger described in (Lukanin *et al.*, 2007). In this model, polynomial expressions for calculation of indicated efficiency (η_i), peak combustion pressure (p_z), volumetric efficiency (η_v), average exhaust gases temperature at the inlet of the turbine (T_t) and mean gas exchange losses pressure (p_m) versus six factors: engine speed (n), air excess coefficient (α), boost air pressure (p_k), boost air temperature (T_k), exhaust back pressure (p_t), ignition advance angle ($\Theta_{ig\ adv}$) were used. The aforementioned polynomial expressions were obtained earlier by calculations of the gas engines and diesel engines cycles using the experiment planning methods. The mean mechanical losses pressure (p_m) is calculated using an empirical formula. Turbocharger is presented in the model in the form of its compressor and turbine maps described by polynomial expressions.

The model uses iterations method. Initially, the approximate values of the turbocharger rotor speed (n_r), air excess coefficient (α), boost air pressure (p_k), exhaust back pressure (p_t), mean mechanical losses pressure (p_m) and some other parameters are assigned, and further these values are defined more precisely by iteration processes. The iterations are stopped when the difference between the turbine power and the compressor power becomes less than a predetermined value (2 percent). The model also has a function of searching for an optimal fuel efficiency ignition advance angle ($\Theta_{ig\ adv}$) on condition that the peak combustion pressure (p_z) does not exceed a predetermined critical value.

Figure 1 shows the calculated values of variations of the effective efficiency (η_e), peak combustion pressure (p_z), air excess coefficient (α) and mean effective pressure (p_e) at external speed characteristics of the gas engine and diesel engine having the same dimensions and arrangement of cylinders: V8, D=120 mm, S=120 mm.

Calculations made in case of equal mean effective pressure p_e (when the torque backup amounted to 29%) demonstrated that the average values of the effective efficiency (η_e) of both the engines were close to each other.

At high engine speeds (n), the effective efficiency values (η_e) of the engine powered with natural gas were higher (maximum difference 2.4%) and at low engine speeds, the diesel engine ensured a little better fuel efficiency. Maximum difference did not exceed 5.3%.

The differences in peak cycle pressures (p_z) (the values are favorable for the gas engine) can be evidently explained by a considerable difference in heat release nature.

A slightly lower air access coefficient (α) at high engine speeds in case of the engine supplied by natural gas is explained by an earlier termination of heat release and as a consequence, lower values of the average exhaust gas temperature at the inlet of the turbocharger turbine.

The results of the comparative calculated analysis of the cycle parameters in case of the quality power level control demonstrated that if the engine supplied with natural gas is perfected properly in the whole engine speed range, it will be at least as good as the base diesel engine regarding specific fuel efficiency and energy parameters. At the same time, the gas engine exceeds the base diesel engine by parameters influencing the performance reliability (lower value of the peak combustion pressure (p_z)) and ensuring a considerable drop of carbon dioxide emissions – the base greenhouse gas.

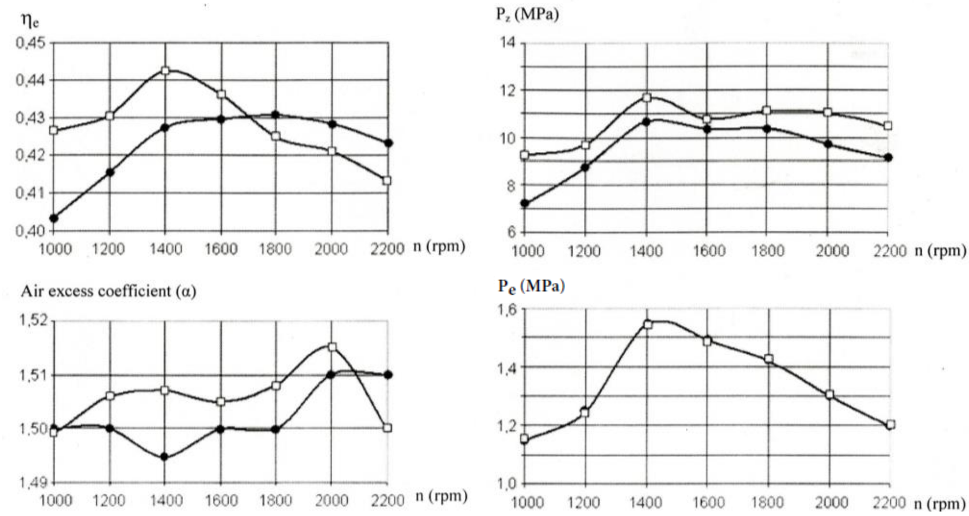

Fig. 1. Comparison of calculated cycle parameters of the gas engine and basic diesel engine by full external speed characteristic -□- Diesel engine; -●- Gas diesel engine

Conclusions

Pe (MPa)

Summarizing the results of the comparative calculation analysis, it should be noted that if one succeeds to optimize the working process using natural gas and achieve the complete combustion of natural gas, the indicated efficiency (η_i) would be at least not less than that in the diesel engine. In practice, this result may mean the equal fuel efficiency of both types of engines because there are no reasons to expect higher mechanical losses in the gas engine.

It should be also noted that the mean indicated pressure of a real cycle will be also close for both the engines if natural gas is supplied directly into the cylinders. At the same time, in case of external mixture formation, a considerable drop of the mean indicated pressure is inevitable. Also we should mention a lower maximum pressure rise rates when using natural gas due to one-phase heat-release process which will ensue lower noise emission.

Calculations of external speed characteristics of the turbocharged base diesel engine and turbocharged gas engine having the diesel compression ratio, quantity power level control and internal mixture formation confirmed that if the natural gas engine is perfected properly, it will not yield the base diesel engine in fuel efficiency and power parameters in the whole engine speed range. It will have lower values of parameters influencing the performance reliability and ensuring a considerable drop of carbon dioxide emissions – the base greenhouse gas.

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