Periodicals of Engineering and Natural Sciences ISSN 2303-4521
Vol. 7, No. 1, June 2019, pp.281-286
Available online at: http://pen.ius.edu.ba

Results of simulation and experimental research of automobile gas diesel engine

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Article Info

Received Dec 23th, 2018

Keyword:

Diesel engine conversion
Gas diesel engine
Dual fuel engine
Gas feed system
Engine control system

ABSTRACT

The paper is dedicated to conversion of a truck diesel engine into a gas diesel engine. Different ways of diesel engine conversion for operation on natural gas were analyzed. The gas diesel working process with minimized portion of igniting diesel fuel supplied by the Common Rail system was selected as the most suitable for such an engine. Modular gas feed and electronic engine control systems were developed in MADI which may be mounted on high- and medium-speed gas diesel engines. The fuel supply system of the base diesel engine was preserved though a new algorithm of fuel injection for the gas diesel engine was developed. The systems were perfected using simulation by computer model developed in MADI and during engine tests. The gas diesel engine demonstrated good fuel efficiency and considerable decrease of NO_x and CO_2 emissions, though its power at low speed decreased compared to the base diesel engine. The ways to improve this drawback during the future work were proposed.

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1. Introduction

Conversion of diesel engines for operation on natural gas is a pressing issue because the discovered reserves of natural gas on our planet are larger than that of oil. Natural gas is almost twice cheaper than diesel fuel in Russia and the difference will grow as more and more oil is produced from hard-to-extract reserves. Conversion of diesel engines of trucks and buses for operation on natural gas would enable to lower considerably operational costs. Transfer from oil to gas fuel results in considerable drop of particles and nitrogene oxide (NO_x) emissions, as well as decrease of hydrocarbon (CO_2) emissions due to lower content of carbon in natural gas and lower gas consumption compared to diesel fuel as natural gas has a higher caloric value especially if a gas diesel working cycle is used.

Many methods IC engines working process organization for operation on natural gas exist depending on engine size, power augmentation, application, operation conditions, etc.

Gas engine with spark ignition using a stoichiometric gas-air mixture requires only minor modifications of the base engine. It ensures reliable engine start and stable combustion at all operating conditions, less problems with knock compared to engines using a lean gas-air mixture. A three-way catalyst may be used to clean efficiently all three toxic emissions: NO_x, HC and CO. Though stoichiometric gas-air mixture results in the increase of fuel consumption, thermal load of the engine and turbocharger, NO_x emissions without catalyst are high. Engine power augmentation is limited by a high temperature of exhaust gases at the turbine inlet. This method is often used for conversion of passenger cars petrol engines and in some cases – of bus engines [1].

Gas engine using a lean gas-air mixture may have much higher power augmentation and its NO_x emissions are significantly lower than that of the stoichiometric engine which makes it possible to comply with the latest NO_x ecological standards without the ammonia catalyst. This method can be successfully used on high- and medium-speed engines if special knock preventing measures are taken. Jenbacher 6th series gas engines for electric power plants using a lean gas-air mixture attain the brake mean effective pressure 2.4-2.6 MPa and have a high rated speed 1500 rpm [2]. For efficient operation on a lean mixture, a prechamber is used. Stoichiometric gas-air mixture in the prechamber is easily ignited and then the sprays of burning gas inflame the lean gas-air mixture in the main combustion chamber. The Miller cycle is realized which prevents knock, increases the effective efficiency, lowers NO_x emissions. A two-stage turbocharging system is mounted to compensate for loss of the filling efficiency caused by the early closure of the intake valves in the Miller cycle. This conversion method is good for engines operating all the time at a high speed because it prevents knock. But for high boosted transport engines operating in a wide range of speeds and loads, as well as at transient modes, this method requires complicated measures to avoid knock.

Gas diesel engine using a traditional diesel fuel feed system. These engines have a relatively low substitution of diesel fuel by gas because the percentage of diesel fuel is 20-30% at full loads, it increases at low loads and becomes 100% at idle [3].

Gas diesel engine with minimized portion of diesel fuel injected by a high pressure Common Rail system. A high injection pressure, multiple injections and injection timing varied in a wide range due to computer control ensure good ignition of natural gas owing to fine atomized diesel fuel sprays which enables to decrease the percentage of diesel fuel to 3-5% at full load and keep it small at low loads and idle [4]. This method makes it possible to have a high substitution of diesel fuel by natural gas and improves significantly engine fuel efficiency and ecological parameters.

Though if a fuel supply system of the base diesel engine is used, the injectors may overheat because when only 3-5% of diesel fuel is injected through them, cooling of injector nozzles by diesel fuel is poor. The problem may be solved by mounting on a big gas diesel engine a fuel supply system of a smaller diesel engine. In this case, the gas diesel engine will not be able to operate on diesel fuel only. In contrast to “gas diesel engine” which can operate equally well in diesel and gas diesel modes, this version is often called “dual fuel engine”.

The diesel fuel supply system of a gas diesel engine should ensure high fuel injection parameters in diesel mode to comply with the latest ecological standards, as wel as high engine power and fuel efficiency, and in gas diesel mode – ensure efficient ignition of the natural gas with the minimized portion of diesel fuel. For this, it should be thoroughly designed to raise the injection pressure [5], ensure a perfect fuel distribution in the combustion chamber [6], multiple injections [7], and control of the injection rate shape [8].

On the basis of the analysis performed, a gas diesel working cycle with minimized portion of diesel fuel injected by a high pressure Common Rail system was selected for the truck engine as it ensures the best fuel efficiency and low emissions, high percentage of diesel fuel substitution. The engine can be high boosted and operate in a wide range of speeds and loads without limitations related to knock.

2. Development of a modular gas fuel supply and electronic control systems

As the research was executed within the framework of the State program of development of systems for high- and medium-speed engines fed with natural gas, a modular gas feed system and electronic engine control system which may be mounted on both the high- and medium-speed gas diesel engines were developed [9, 10].

The gas fuel system has several identical modules linked with each other. Every module reduces pressure and supplies gas to the cylinders. This enables to use the right number of modules for every engine. For example, the high-speed Cummins Kama gas diesel engine needs one module and the medium-speed gas diesel engine having 6 in-line cylinders with D/S=200/280 mm needs three modules. The gas feed system supplies natural gas with operating pressure 1 MPa to the cylinders of the gas diesel engine having external mixture formation. It has gas metering valves with electronic control.

A brand new electronic engine control system was developed that controls supply of diesel and gas fuel. The electronic engine control system for the L6 gas diesel engines produces control impulses for actuators and carries out synchronization and distribution of impulses across the cylinders depending on the engine operating conditions based on the data supplied from many sensors.

The standard Common Rail fuel supply system of the base diesel engine was preserved. For operation in the gas diesel mode, a new control module was developed which ensures an optimal quantity of diesel fuel and its injection timing in a wide range of engine speeds and loads.

3. Simulation method

A one-zone model of diesel, gas diesel and gas engine developed in MADI [11, 12] was used for simulation. The model is based on the first law of thermodynamics in a differential form.

Combustion process is calculated on the basis the heat release rate obtained by the well known formula of I. Vibe and heat transfer – using the empirical formulas of G. Woschni or A. Annand. The working medium properties are calculated taking into account variable composition of the gas fuel. It may have about 20 components. The polynomials for calculation of the specific heat for more than 20 gases as a function of temperature are integrated into the model.

Gas exchange calculation is carried out using the quasi-stationary method. Additional charging during the inlet valve closure delay period and scavenging during the valves overlap period are calculated. Partial substitution of intake air with gas is taken into account. Average temperatures of the in-cylinder surfaces and mean pressure of mechanical losses are calculated by empirical formulas.

Pulsations of pressure and temperature in the intake and exhaust manifolds are calculated on the basis of a simple “Filling-Emptying” method.

Parameters of the engine and turbocharger joint operation are found using iterations method. Initial values of a number of engine operation parameters are assigned which are defined more accurately during iterations. Iterations process stops when the difference between the compressor and turbine power is less than 2%. Experimental compressor and turbine maps are used.

The model was calibrated by the results of the experimental research of truck and bus diesel, gas diesel and gas engines.

Simulation was used for the following:

- Prediction of the gas diesel engine parameters which are required for the development of fuel supply and engine control systems
- Analysis of the gas diesel engine experimental test results for estimation of the quality of its working process and planning the ways of its future improvement.

4. Analysis of experimental tests and simulation results

The load and speed characteristics of the gas diesel engine are shown in Fig. 1 and Fig. 2. Calculations were performed using the optimal ignition advance angle for every operation mode. Real values of the diesel fuel injection rate and natural gas injection rate obtained during the engine tests were used for the calculations.

Pressure losses in the intake and exhaust systems and in the intercooler, as well as the intercooler efficiency determined during the engine tests were used as input data.

As seen from Fig. 1, there is a good agreement of calculated and experimental values of the boost pressure p_s and gas pressure before the turbine p_t at all operation modes. The difference between the values of the air access coefficient α, air consumption G_a, boost air temperature T_s and effective efficiency η_e does not exceed 2% at full load and is a bit higher at low loads. This may be explained by not very accurate description of experimental compressor and turbine maps by the polynomials used in the model in the areas of low engine loads.

In most points of the load characteristic presented in Fig. 1 and the speed characteristic presented in Fig. 2, the effective efficiency η_e of the gas diesel engine is close to the η_e value of the base diesel engine. (Parameters of the base diesel engine are not presented in the diagrams.) Taking into account the fact that the caloric value of methane (50 MJ/kg) is by 15% higher than the caloric value of diesel fuel (42.5 MJ/kg), the brake specific fuel consumption g_e of the gas diesel engine should be approximately by 14% lower at full load when the diesel fuel portion is 5.5% and by 10% lower at idle when the diesel fuel portion is 33%. The g_e value is calculated taking into account the caloric values of methane and diesel fuel and the percentage of methane and diesel fuel injected, correspondingly.

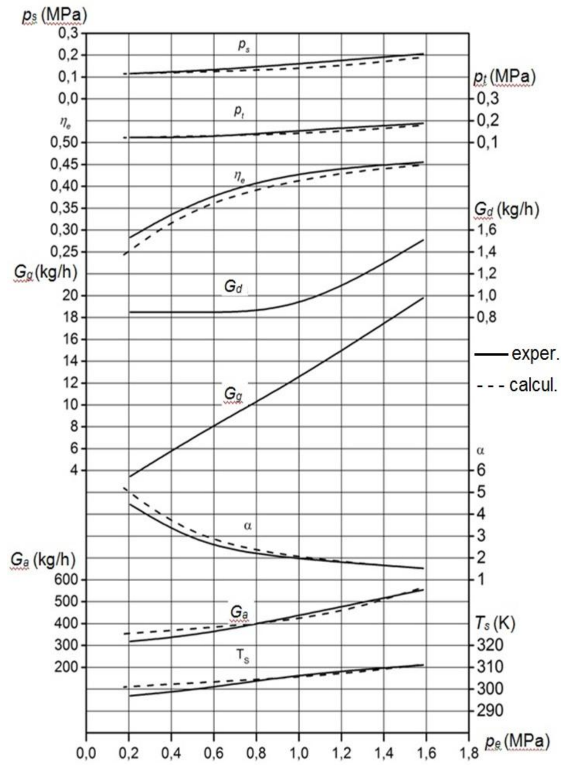

Figure 1. Gas diesel engine load characteristic

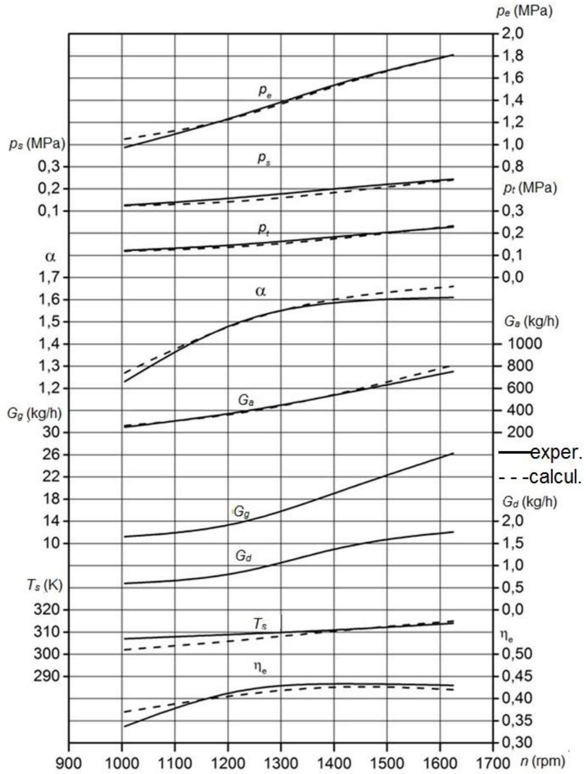

Figure 2. Gas diesel engine speed characteristic

As seen in Fig. 2, at low engine speed, experimental effective efficiency η_e is considerably lower than the calculated one and its maximal difference is about 9% at 1000 rpm. The calculated brake mean effectible pressure p_e is also lower at a low engine speed. This may be explained by two reasons:

As indicated in [13], the igniting ability of the diesel fuel degrades when the air backpressure in the cylinder decreases because the process of fuel droplets breaking is poor. The pressure in the cylinder decreases due to the boost pressure drop in turbocharged engines at low speeds. The problem may be solved using turbocharging control systems.

In diesel engine, the droplets of liquid fuel are located only in the combustion chamber volume and burn perfectly. In gas diesel engine, the gas penetrates into the gaps between the piston and the cylinder head, as well as between the piston and the cylinder wall where it burns incompletely or does not burn at all increasing emissions of hydrocarons. This is especially evident at low engine speeds when the air motion speed is low and the turbulence is poor. In the simulation model used, poor combustion in the gaps between the piston/cylinder head and th episton/cylinder wall is not taken into account, therefore the calculated values of η_e are high. It is proposed in [13] to use a special baffled piston to increase the tangential speed of the gases

Comparison of parameters of diesel and gas diesel versions of the Cummins Kama engine after experimental perfection of the modular fuel supply and engine control systems for three engine speeds (n) and two loads (M_e) at every engine speed is shown in Table 1. A high percentage of diesel fuel substitution by gas was achieved which was on the average: 5.6% at full load, 8.8% at approximately 35% load and 33% at idle. The effective efficiency for both the engines was close and its maximal value was 44% in the speed characteristic. On the average, emissions of NO_x of the gas diesel engine decreased 1.52 times and of CO_2 – 1.18 times compared to the base diesel engine.

Table 1. Comparison of ecological parameters of diesel and gas diesel versions of the gas diesel engine

n (rpm)	*Me* (N·m)	Diesel fuel portion (%)	CO_2 emissions (%)		NO_x emissions (ppm)	
			diesel	gas diesel	diesel	gas diesel
1220	660	4.5	9.0	7.5	2100	1570
	280	8.8	4.9	3.2	1260	162
1420	840	6.2	8.8	7.7	1123	949
	285	8.9	4.8	2.8	766	207
1625	940	6.0	8.5	7.6	1030	504
	260	8.7	4.6	3.9	540	51

5. Conclusions

1. The analysis demonstrated that gas diesel working process with a minimized igniting portion of diesel fuel supplied by the CR system is the most reasonable method of conversion of a high boosted truck diesel engine for operation on natural gas.
2. Modular gas feed and electronic engine control systems were developed using gas diesel engine simulation by the one-zone model developed in MADI. The base Cummins Kama diesel engine was converted into gas diesel engine using the systems developed, and the systems were calibrated during the engine tests.
3. A high degree of diesel fuel substitution by gas was achieved: the average diesel fuel portion amounted to 5.6, 8.8 and 33%, correspondingly, at full load, approximately 35% load and idle. The effective efficiency of diesel and gas diesel engines was close with the maximal value 44% at full load. On the average, for the gas diesel engine, emissions of CO_2 decreased 1.47 and 1.15 times and emissions of NO_x decreased 7.4 and 1.52 times correspondingly, at full and partial loads. The brake specific fuel consumption of the gas diesel engine was by 10-14% lower than of the base diesel engine.
4. At low speed, the torque of the gas diesel engine was lower than of the base diesel engine. Explanation was given and remedial methods were proposed.

6. Legend

Greek symbols
α – air excess coefficient
η_e – effective efficiency
Latin symbols
CH – hydrocarbons (ppm)
CO – carbon oxide (ppm)
CO_2 – carbon dioxide (%)
D – cylinder diameter (mm)
G_a – air consumption (kg/h)
G_d – diesel fuel consumption (kg/h)
G_g – gas fuel consumption (kg/h)
M_e – engine effective torque (N·m)
NO_x – nitrogen oxides (ppm)
n – engine speed (rpm)
p_e – mean effective pressure (MPa)
p_s – boost air pressure (MPa)
p_t – gas pressure before turbine (MPa)
S – cylinder stroke (mm)
T_s – boost air temperature (K)

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