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Evaluating gas-diesel cycle efficiency in machine-tractor power plants

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Abstract. To date, the gas-diesel cycle of operation of diesel internal combustion engines is one of the promising methods of improving the efficiency of existing designs in transport and technological machines. This engine conversion allows to significantly improve the environmental performance of the engine with a significant reduction in operating costs. For estimation of influence of engine conversion to gas-diesel cycle the principles of dynamic systems modelling are used. Solutions of first or second order differential equations with constant coefficients are used as indicators of operation in unsteady modes. The engine torque is considered as a two-parameter quantity. This approach makes it possible to estimate, among other things, such engine parameters as hourly and specific fuel consumption and effective power. The paper discusses the influence of changing the ratio of the reserve fuel portion on the dynamic characteristics of the engine. The obtained optimum values of the diesel fuel reserve dose correlate with the results of other authors' research in this area.

1 Status analysis of the issue

We are currently facing the challenge of eliminating or significantly reducing the use of fossil fuels for energy production and transport. Many years of uncontrolled burning of fossil fuels have led to far-reaching, unfavorable changes in the Earth's climate. In many places on Earth, especially in heavily industrialized areas, humanity faces a significant air pollution problem, leading to human health problems. Prolonged exposure to highly polluted air contributes to irreversible changes in the bodies of not only humans but also animals. Gas emissions associated with the extraction, processing and subsequent combustion of hydrocarbon fuels have led to unfavorable changes in the earth's atmosphere [1].

Diesel engines have been an integral part of our economy and industrial growth for a long time. They have dominated a variety of applications including heavy transport, small power generation, construction and agricultural machinery and passenger cars. They have evolved over the last hundred years through extensive research and development efforts around the world. Sophisticated electronically controlled fuel injection systems have replaced previous generation mechanical fuel injection equipment for many applications. However, some

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applications require robust and reliable systems, such as agricultural and construction machinery engines, where mechanical fuel injection systems are still used and preferred. The superior energy conversion efficiency, high power output and torque characteristics of diesel engines are unrivalled by other technological alternatives. Consequently, diesel engines will undoubtedly exist in the future. Although increasing levels of pollutants in the environment have created a negative image of diesel engines, engine researchers and automotive engineers are making significant efforts to control engine emissions to make them cleaner by using alternative fuels, low temperature combustion and exhaust aftertreatment systems [2, 3].

Emission standards are becoming increasingly stringent to control these harmful emissions that harm the environment and people. To meet the stringent emission standards for diesel engines, many researchers have focused on efforts to develop engine or fuel related technologies such as gaseous fuels or alternative oxygenated fuels derived from biomass [4, 5].

The most common diesel engine related technologies include the use of Common Rail fuel systems with different fuel injection control strategies, high- or low-pressure exhaust gas recirculation (HP/LP EGR), exhaust gas aftertreatment (e.g., particulate filters, oxidation catalysts, selective catalytic reduction, depleted NOx catalysts) and advanced engine combustion techniques (e.g., homogeneous charge compression ignition, low temperature combustion). However, due to the PM/NOx trade-off, simultaneous reduction of PM and NOx emissions is quite difficult and requires huge additional costs for installation of engine components and aftertreatment systems. On the other hand, the use of gaseous fuels using an ignition portion of conventional fuels in diesel engines is a realistic strategy to replace large amounts of petroleum-based fuels and reduce emissions [6, 7].

2 Proposed solution

One of the promising options for solving this problem is the use of gas-diesel cycle, in which the main type of fuel is gaseous fuel (methane), and combustion in the cylinder occurs due to the supply of an ignition portion of fuel (up to 30%) [8, 9]

The modern design of the gas-diesel engine system can be considered on the example of gas-diesel GD-243. Gas-diesel GD-243 is designed for agricultural tractors, municipal services, road and construction machinery [10]. It is equipped with a modern fuel system (Fig. 1), which consists of two fuel systems for gas and diesel fuel supply.

The gas-diesel engine is capable of operating both on the gas-diesel cycle and on pure diesel fuel. In the gas-diesel mode, compressed natural gas (methane) and a reserve dose of diesel fuel are used as gas fuel. The substitution of diesel fuel with gas reaches up to 70 per cent in the rated power mode.

Compressed natural gas is stored in cylinders mounted on the vehicle at a pressure of up to 20 MPa. Cylinders are equipped with valves with automatic fire protection valve. Gas from the cylinders at a pressure of 20 MPa enters a two-stage high-pressure reducer. The reducer reduces the gas pressure from 20 MPa to 2-4 bar.

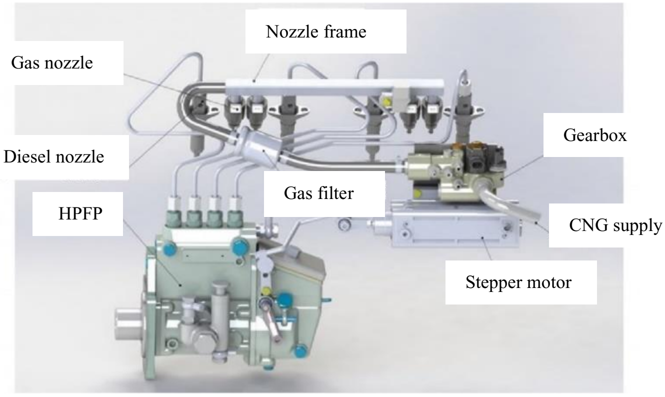

Fig. 1. Fuel equipment of gas-diesel engine GD-243 [10]

From the reducer, the gas flows through the gas filter to the gas ramp. The high-pressure reducer maintains a constant pressure of up to 4 bar in the ramp.

The HPFP fuel supply control lever is equipped with an HPFP rail position sensor via a special mechanism, which monitors the position of the rail when the driver (mechanic) sets the desired crankshaft speed.

The fuel supply is limited by the stepper motor by acting on the HPFP rail through the stop lever. Control is accomplished by limiting the travel of the HPFP rail.

The gas-diesel is started in diesel mode only. Conversion to gas-diesel mode is performed by the driver (mechanic) using a special fuel switch button installed in the tractor (vehicle) cab. The diesel engine is switched to gas-diesel mode only after the coolant temperature reaches 38-40°C. For this purpose, the high-pressure reducer is equipped with a coolant temperature sensor and is connected to the gas-diesel cooling system to avoid freezing of the reducer due to gas pressure drop.

The gas supply is controlled by the electronic control unit (ECU) by means of the opening time of the gas nozzle.

For precise dosing of the gas supply quantity, the gas-diesel is equipped with:

- crankshaft speed sensor;
- camshaft position sensor.

A modified HPFP-004 is installed on the gas-diesel engine, which differs from the standard HPFP by the presence of igniting dose setter (IDS). IDS is designed to limit the diesel fuel supply when the engine is operating in the gas-diesel mode, as well as to automatically regulate the gas supply to the engine using the standard speed regulator.

In the gas-diesel mode, as well as in the diesel mode, the regulator performs all-mode speed control.

To compare the efficiency of the gas-diesel cycle we will use the methodology proposed in [11, 12]. In it, the author applies the method of modeling dynamic systems, according to which the change of parameters in non-steady-state modes is described by solutions of differential equations. Using the assumptions of stationarity and linearity of characteristics and applying the principle of superposition it is possible to extend these solutions to any types of load modes of power plants.

First order differential equations are used for the corrector branch of the regulatory characteristic of a power plant:

$$T_{1i}\frac{d\Delta l_i}{dt} + \Delta l_i = K_{1i}\,\Delta M_c \quad (1)$$

for the regulatory branch of the second:

$$T_{2i}\frac{d^2\Delta l_i}{dt^2} + T_{1i}\frac{d\Delta l_i}{dt} + \Delta l_i = K_{2i}\,\Delta M_c \tag{2}$$

here T_{1i}, T_{2i} – coefficients of differential equations determined from experimental data on test loads;

K_{1i} – torque gain of the i-th parameter;

ΔM_c – increment of resistance torque on motor shaft;

Δl_i – is the increment of i-th parameter at change of load on the motor shaft.

It should be noted that if the condition is met, equation (2.2) corresponds to the aperiodic link of the second order, i.e., with sufficient accuracy it is also applicable to the correction branch.

Let us represent the torque of the power plant as a function of two variables, namely crankshaft speed n and cycle fuel supply gc:

$$M_{tor} = K_{Mp}\cdot n + K_{Mgc}\cdot g_c \tag{3}$$

here K_{Mp}– coefficient of proportionality of torque to crankshaft speed, $\left[\frac{n\cdot m}{\min^{-1}}\right]$,

K_{Mgc}– torque proportionality coefficient by cycle fuel supply, $\left[\frac{n\cdot m}{g/c}\right]$.

In unsteady operating modes:

$$M_{tor.dyn} = M_{tor0} + K_{Mp}\cdot \Delta n + K_{Mgc}\cdot \Delta g_c \tag{4}$$

here M_{tor0}– torque of the power plant in steady-state mode at the initial moment of time, $[n\cdot m]$.

To determine the gain factors K_{Mp} and K_{Mgc} , the following dependencies need to be considered: $M_{tor} = f_1(n)$ and $g_c = f_2(n)$ for two arbitrary points 1(n1, gc1, Mtor1) and 2 (n2, gc2, Mtor2). And solving the resulting system of equations

$$\begin{cases} M_{tor1} = K_{Mp}\cdot n_1 + K_{Mgc}\cdot g_{c1} \\ M_{tor2} = K_{Mp}\cdot n_2 + K_{Mgc}\cdot g_{c2} \end{cases} \tag{5}$$

we obtain the desired coefficients:

$$K_{Mgc} = \frac{\frac{M_{tor1}}{n_1} - \frac{M_{tor2}}{n_2}}{\frac{g_{c1}}{n_1} - \frac{g_{c2}}{n_2}} \tag{6}$$

$$K_{Mn} = \frac{M_{tor1}}{n_1} - \frac{\frac{M_{tor1}}{n_1} - \frac{M_{tor2}}{n_2}}{\frac{g_{c1}}{n_1} - \frac{g_{c2}}{n_2}}\cdot\frac{g_{c2}}{n_2} \tag{7}$$

For the D-243 engine this characteristic is shown in Fig. 2 (combined characteristic).

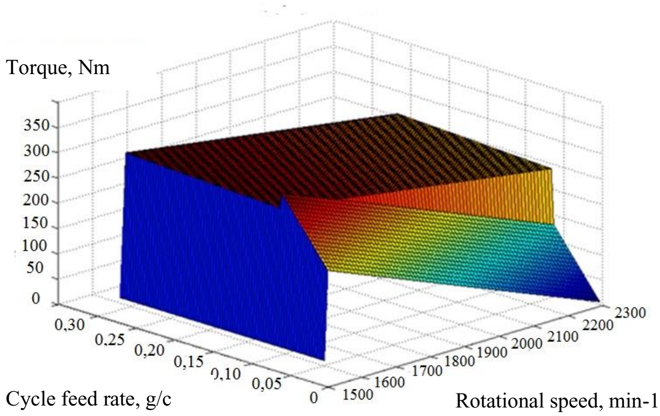

Fig. 2. Combined characterization

If the surface is approximated by a polynomial of the second degree, the obtained characteristic is more attractive (Fig.3).

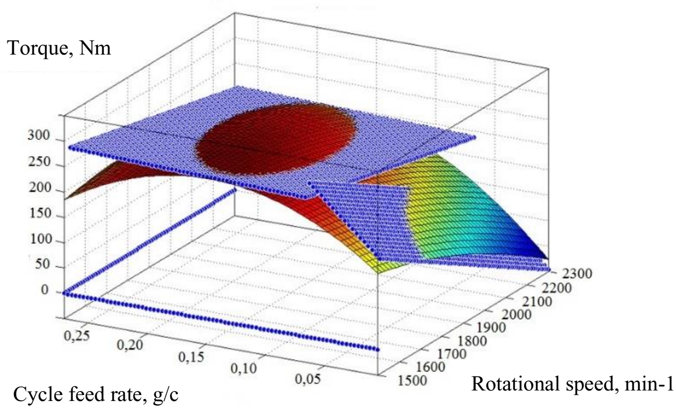

Fig. 3. Setting the surface by a polynomial of degree 2

Consider the influence of the dose of the fuel portion on the transient time for our case. We will change the ignition dose in the range of 25-35% for nominal operation mode [13, 14, 15].

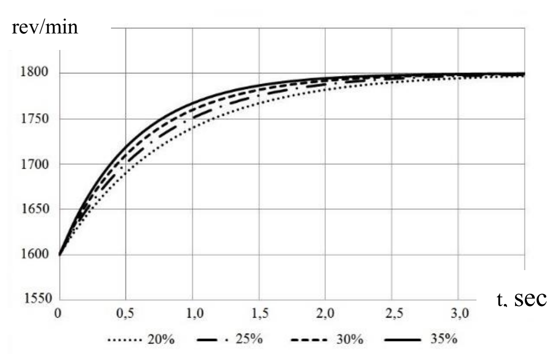

Fig. 4. Crankshaft n speed transient function:

Further increase in the percentage of the reserve fuel dosage reduces the economic effect of using the gas-diesel cycle and was not considered in our studies.

As a result, the values of time constants by rotation speed for different values of the reserve fuel dosage were obtained, which are shown in Table 1.

Ignition dose, %	20	25	30	35
Time constant T, sec	2.5-2.7	2.1-2.3	1.9-1.7	1.5-1.7

3 Discussion

The experience of research of gas-diesel engine operation cycle allows us to make a preliminary conclusion that this proposal has an economically justified effect, both in terms of environmental indicators of exhaust gases and the efficiency of engine utilization as part of transport and technological machines. The correct choice of the reserve dose of traditional fuel depending on the engine operation modes is important. Changing its value from 25% to 35% for the nominal mode of operation can improve the dynamic characteristics of the engine: so, the time constant at crankshaft speed is reduced from 2.5 sec at 25% to 1.5 sec at 35%. Using the above formulas (1-4), the results of these studies can be extended to the effective indicators of engine utilization on the gas-diesel cycle.

The proposed methodology for determining the optimal values of the reserve dose of fuel is of practical interest for engineering and technical workers engaged in improving the efficiency of the parameters of the use of power plants of transport and transport-technological machines in the agro-industrial complex conditions [16].

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