STUDYING THE CHARACTERISTICS OF OPERATIONAL PROCESS OF DIESEL ENGINE AND GAS DIESEL ENGINE CYCLES

ИССЛЕДОВАНИЕ ХАРАКТЕРИСТИК ЦИКЛОВ РАБОЧЕГО ПРОЦЕССА ДИЗЕЛЕЙ И ГАЗОДИЗЕЛЕЙ

Doctoral candidate Shotadze Z.,Assoc.Prof. Dr. KamladzeA.,Prof. Dr. Kochadze T.
AkakiTsereteli State University, Kutaisi, Georgia
E-mail: ZURAB.SHOTADZE@gmail.com , kamladze@mail.ru , Temko1954@mail.ru;

Abstract: *Energy efficient improvement and bringing to conformity with long-term environmental safety requirements are considered as an important stage to improve ship engines. The values of the mentioned characteristics are mostly determined by the type of consumed fuel and organization of operational process that in turn conditions the intensity of the process of combustion and heat release in the cylinder. And these parameters are the determined factors in the view of improving the technical and economic performance of diesel engines. From this point of view, at this stage conversion some types from the ship diesel engine cycle into the gas diesel engine cycle.*

The paper aims to determine the heat release characteristics on the basis of theoretical and experimental analysis of ship engine performance when implementing the operational process of both diesel engine and gas diesel engine cycles. The given paper dwells on comparative characteristic and efficiency analysis of diesel engine and gas diesel engine cycles, on the basis of experimental modeling.
KEY WORDS: *GAZ DIESEL, ENERGY EFFICIENCY, GAZ FUEL, OPERATIONAL PROCESS*

1.Introduction

Energy efficient improvement and bringing to conformity with long-term environmental safety requirements are considered as an important stage to improve the modern ship engines. The values of the mentioned characteristics are mostly determined by the type of consumed fuel and organization of operational process that in turn conditions the intensity of the process of combustion and heat release in the cylinder. And these parameters are the determined factors in the view of improving the technical and economic performance of diesel engines. From this point of view, at this stage, conversion of some types from the ship diesel engine cycle into the gas diesel engine cycle.

Gas diesel engine is an engine running on two types of fuel, on which, in contrast to the basic diesel type, there is mounted the additional equipment in the form of gas-fitting. Such engines operate in a diesel cycle without any engineering renovations, but they cannot run on a gas fuel. During the process of charging in the mentioned engines, there is supplied into the cylinder the gas-air mixture, which undergoes compression. Ignition of mixture, in other words the creation of the combustion sites is carried out at the end of compression by a diesel fuel injected into the cylinder, which in ship engines, when operating on diesel cycle, makes up 5-10% of consumed fuel.

The paper aims to determine the heat release characteristics on the basis of theoretical and experimental analysis of ship engine performance when implementing the operational process of both diesel engine and gas diesel engine cycles. The given paper dwells on comparative characteristic and efficiency analysis of diesel engine and gas diesel engine cycles, on the basis of experimental modeling.

2. Preconditions and means for resolving the problem

Diesel engine and gas diesel engine performance indicators depend significantly on the improvement of combustion process in the cylinder. The combustion process occurring in the engine cylinder involves very complex physical and chemical phenomena, due to which it improvement is associated with a number of difficulties, though the great experience from the experimental studies of the combustion process allows for judging correctly the nature of process behavior by the indicator-diagram shape, since it is well-known what must be the indicator-diagram shape, when the engine generates maximum power and fuel efficiency. In addition to the experimental indicator-diagram, for analysis of the combustion process there are also widely used the mathematical methods, which are based on the values obtained as a result of solving the equation of a temperature curve.

The combustion process in diesel and gas diesel engines can be broken down into four phases or periods: spontaneous combustion prevention period (induction period); constant pressure combustion (delayed combustion) and after-combustion. Fig.1 illustrates the process indicator-diagram in the $p - \varphi$ coordinates.

By the end of the compression process, when the piston is about to reach the upper dead center, fuel injection is started. This means that the first phase begins when lifting up the injector valve (point 1) and lasts until completion of fuel injection (point 2). During this period, the fuel injected into the cylinder is contacted with the air heated from compression (air and methane mixture in gas diesel engine) and undergoing physical and chemical changes. Physical change is manifested in the fuel evaporation, but chemical one – in fact that the fuel component carbohydrates are oxidized by aerial oxygen and unstable compounds are formed. This means that in the first phase, the fuel injected into the cylinder is preparing for combustion.

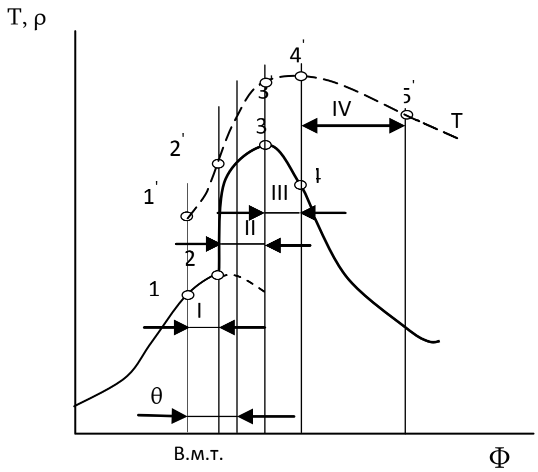

Fig.1. *Diesel engine combustion process pressure and temperature variation diagram.*

The second phase is a period of demonstrative injection of fuel, in other words, rapid combustion begins from the point (2) and ends at the point (3). Most of fuel injected in the first phase and that part of fuel, which is injected in a current phase, are burned during this phase. The process is characterized by a sharp rise of pressure and approached to the isochoric combustion process.

During this period, the rate of pressure rise depends on the first phase's value. The longer is this period, the more fuel is accumulated in the cylinder by the beginning of combustion, and accordingly, the more fuel will be burned during the second period. Hence it is clear that the longer the period of delayed combustion is, the sharper is pressure rise, but excessive pressure rise causes the engine to rough run.

The third phase – the delayed combustion period, begins from the point (3) and ends at the point (4). This period is characterized by a delayed rise of pressure that is caused by fact that in this phase, combustion process is occurred in the conditions of significant rise of volume, at almost constant pressure.

The fourth phase – after-combustion period, lasts after the pint (4), shifts to an expansion phase and ends at the point (5). At the point (4), pressure of gases in the cylinder reaches the maximum, the gases start to expand and pressure sharply goes down. A part of fuel previously injected into the cylinder is continuing to burn in the combustion process.

If we analyze the above mentioned considerations, then we shall see that the spontaneous combustion prevention period has great influence on diesel engine running that, of course, depends on the beginning of fuel injection, and, in fact, is a different process in case of a gas diesel engine. Investigations have shown that the dose of igniter of fuel injected into the cylinder, mixture-formation process and the fuel injection advance angle have great influence on a gas diesel engine characteristics. As a rule, gas diesel engine cycle is characterized by two peaks of heat liberation, first one of which corresponds with the combustion process of a liquid fuel igniter dose, but another peak corresponds with gaseous fuel combustion. When possible, reduction of a liquid fuel dose and increasing gaseous fuel dose result in first peak drain and second peak magnification.

But after reaching a certain ratio, the first peak may be eliminated completely.

In order to implement the mentioned work process, it is necessary to determine the compulsory manipulated variables as follows: the value of fuel cyclic delivery; liquid fuel injection advanced angle; the quality of air and natural gas homogenization, etc. Also, it is possible to consider the engine formation by inflation.

As an on object of studying the diesel engine and gas diesel engine work process characteristics, there have been taken the four-cylinder diesel engine 1.7 DTL with dimensions 82,5/79,5, as well as its gas diesel analog. Study and assessment of characteristics were carried out on the basis of comparative analysis of mathematical modeling of the experimental indicator-diagram and combustion process.

The initial data used during simulation of gas diesel and diesel engines envisage the following items:

- Engine design features; number and stroke capacity of cylinders; stroke of piston and cylinder diameter; the effective cross section of the valves; compression quality; valve timing and dimensions of heat collecting surfaces.
- Boundary conditions: piston bottom surface, temperatures of the walls of inlet and outlet pipes, and of the plates of inlet and outlet valves.
- Computation conditions determining parameters: integration step; number of computation cycles; crankshaft rotation angle by the beginning of computation; exit-from-computation conditions.
- Operating regime determining parameters: fuel injection advance angle; number of crankshaft revolutions; excess air factor; cyclic air delivery; cyclic fuel delivery.
- Thermal and physical characteristics of working medium: environmental parameters; fuel composition and lowest combustion heat; temperature dependence of air-and-fuel mixture heating capacity.

The experimental test-bench is equipped with the engine behavior parameters – oil pressure, temperature of oil and cooling liquid; the number of crankshaft revolutions – with the controlling systems. The liquid fuel igniter dose rate was determined by weighting method, but the fuel gas consumption was measured in an altitude test chamber by using volumetric measurer, with allowance for pressure and temperature. With purpose of studying processes occurring in a cylinder, the indication works have been carried out by the pore-developed procedure, and Picture 2 illustrates the advanced indicating-diagram in cases of both diesel cycle and gas diesel cycle.

Analysis of the indicating-diagrams has shown that the cycle maximum pressure (P_{max}) and temperature (T_{max}) have been achieved in diesel engines earlier, than in gas diesel engines, and their values are also different significantly. In addition, the combustion process changes into the expansion process that is more apparent in case of cyclic operation of gas diesel engine, although in accordance with the indicating-diagram we can conclude that gas diesel engine, in comparison with diesel engine, is characterized by lower rigidity ($dP / d\varphi$).

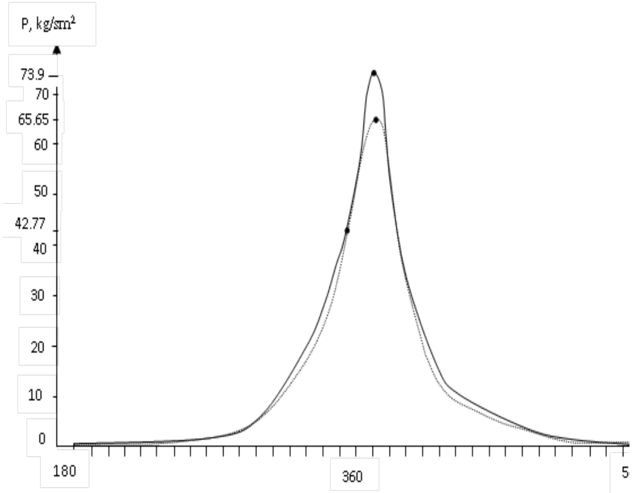

Fig.2 *The advanced indicating-diagram*

——— diesel, ----- gas diesel.

The purpose of the study is to determine heat release characteristics and comparative analysis of operating process in case of gas and gas diesel cycles. It is well-known that according to the combustion process characterizing equations and the indicating-diagram data, it is possible to determine with specified degree of accuracy the active heat release coefficient in a cylinder, which implies the relation of internal energy and thermal energy (Q) consumed for the performance of work to the overall amount of released heat (Q_c) during one cycle of fuel combustion in a cylinder $\xi = Q/Q_c$.

The equation for calculation of heat release coefficient is mainly based on the first law of thermodynamics and generally is written down in the following form:

$$(1) \qquad Q = \xi Q_c = \Delta U + \int PdV = U_x - U_0 + \int_{V_0}^{V_x} PdV,$$

where, ΔU – a charge internal energy variation, but, $\int PdV$ – a charge operation, U_x - a charge internal energy at a considering moment, but U_0 – is the same one by the beginning of combustion, V_0 – is a charge volume by the beginning of combustion, but V_x – is a volume at a considering moment.

With the purpose of improving the calculation process and obtaining a real picture, there have been considered four sections at the diagram (the sections between the charactizing points, Pic. 1):

1. $\xi = 0$ – is a section of the combustion process curve from a separation point of the compression process curve to a top dead center;
2. A section from a top dead center until reaching a maximum pressure (P_{max}) in a cylinder;
3. A section from a maximum pressure point to the appropriate maximum temperature (T_{max}) point;
4. A section from the appropriate maximum temperature (T_{max}) point before starting the valve opening (b point).

The equation (1) appropriate to a maximum pressure point in a cylinder is written down in the following form:

$$\xi_{P_{max}} \cdot Q_c = U_{P_{max}} \cdot \left\{1 + (K_P - 1)\left[\frac{\pi}{4}\left(\frac{\varepsilon_T}{\varepsilon_z} - \frac{\varepsilon_T}{\varepsilon_\Gamma}\right)\lambda_T\right]\right\} + U_c\left\{1 + (K_c - 1)\left[\frac{3+\lambda_c}{4}\right]\left(1 - \frac{\varepsilon_c}{\varepsilon_z}\right) - 0{,}215\lambda_c\left(\frac{\varepsilon_c}{\varepsilon_z} - \frac{\varepsilon_c}{\varepsilon_\Gamma}\right)\right\},$$

where: $\lambda_c = \frac{P_{\text{ъ.д.ђ.}}}{P_c}$, $\varepsilon_z = \frac{V_a}{V_{P_{max}}}$, $\varepsilon_\Gamma = \frac{V_a}{V_{\text{ъ.д.ђ.}}}$, $\lambda_T = \frac{P_{max}}{P_{T_{max}}}$, $\varepsilon_c = \frac{V_a}{V_c}$, $\varepsilon_T = \frac{V_a}{V_{T_{max}}}$.

The given magnitudes are taken for a specified engine.

$$K_P = 1 + \frac{4{,}187}{m \cdot C_V\big|_0^{T_c}} U_{P_{max}} = P_{max} \cdot V_{P_{max}} \cdot \frac{1}{(K_P - 1)}$$

The equation (1) appropriate to a maximum temperature point T_{max} in a cylinder is written down in the following form:

$$\xi_{T_{max}} \cdot Q_c = U_{T_{max}} \cdot \left\{1 + (K_T - 1)\left[\frac{1+2\lambda_T}{3}\left(1 - \frac{\varepsilon_T}{\varepsilon_z}\right) - \frac{\pi}{4}\left(\frac{\varepsilon_T}{\varepsilon_z} - \frac{\varepsilon_T}{\varepsilon}\right)\lambda_T\right]\right\} - U_c\left\{1 + (K_c - 1)\left[\frac{3+\lambda_c}{4}\right]\left(1 - \frac{\varepsilon_c}{\varepsilon_z}\right) - 0{,}215\lambda_c\left(\frac{\varepsilon_c}{\varepsilon_z} - \frac{\varepsilon_c}{\varepsilon_\Gamma}\right)\right\},$$

where, $U_{T_{max}} = P_T V_T \frac{m \cdot C_V|_0^{T_{max}}}{4{,}187}$, $C_V|_0^{T_{max}}$ – is a variation of thermal capacity within a specified temperature interval.

At the moment of starting the release process in a cylinder (point 1), the equation (1) is written down in the following form:

$$\xi_b = \xi_{T_{max}} + \frac{U_b - U_{T_{max}}}{Q_c} + \frac{P_{T_{max}} \cdot U_{T_{max}} - P_b V_b}{(n_z - 1)Q_c}$$

$U_b = P_b V_b \frac{1}{n_2 - 1}$, n_2 – an expansion polytropy indicator.

The values of the coefficients characterizing the active heat release process, which have been obtained as a result of calculations are shown in Table 1.

Table 1.

Type of engine	Separation point	Point P_{max}	Point T_{max}	Point b
Diesel	0	0,68	0,75	0,966
Gas diesel	0	0,61	0,712	0,918

3. Conclusion

Analysis of performed calculations has shown that the available factor of the released heat at the points characterizing the indicating-diagram in gas diesel engine is lower comparatively to a base diesel engine. Similarly, a maximum pressure in a cylinder is low, as well as the rated power, highlighting the need to improve this process. Rom this point of view, the problem can be solved on the basis theoretical and experimental studies by determining the regulating parameters as follows: cyclic fuel delivery; liquid fuel injection equilibrium angle; air and gas fuel mixture homogenicity, etc.

The implementation of the mentioned activities is possible by using novel fuel-injection equipment, or it is possible to consider the timing performance by blowdown.

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