Modeling of filling process of marine diesel engines

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Abstract. The article deals with the efficiency of marine diesel engines produced by JSC «RUMO», Russia, Nizhny Novgorod. The parameters of the working processes determining the specific power of the engine are determined. The basic mathematical dependences that determine the efficiency of the engine are given. The article analyses the influence of individual factors and based on the results, the indicator surface. The influence of the charge air pressure, the delay angle of the inlet closure and the geometric compression ratio of the individual cylinder are considered separately.

1. Introduction

Combined eight-cylinder four-stroke engines with a supercharger, the size of the cylinder 22/28 with a capacity of 1250 kW, specific fuel consumption 195 g/kW/hour and the speed of the crankshaft 1000 min⁻¹, produced by JSC “RUMO”, are used in various industries and transport. In a combined piston diesel engine, part of the chemical energy of the fuel burning in the working chamber is converted into mechanical energy. The energy conversion efficiency is estimated by the efficiency coefficient which is determined by the formulas [1, 2]:

\[ \eta_e = \frac{L}{Q_1} = 1 - \frac{Q_2 + Q_T + Q_M}{Q_1}, \]  

\[ \eta_e = \eta_t \cdot \eta_g \cdot \eta_M. \]  

Thus, the effective efficiency is the product of the thermal efficiency \( \eta_t \), the relative efficiency \( \eta_g \) and the mechanical efficiency \( \eta_M \). These efficiencies characterize, respectively, the degree of perfection of the ideal theoretical cycle on the principally irremoveable by the second law of thermodynamics heat loss \( Q_2 \), the degree of approximation of the design cycle to the ideal by the principally eliminated heat loss \( Q_T \), the degree of perfection of the design and manufacture of the engine on the principle of minimized heat loss \( Q_M \), spent on mechanical losses and consisting of friction losses between moving parts, friction losses on the drive of auxiliary units and the work of pumping moves [1, 2, 3]. Different diesel engines have \( \eta_e = 0.30…0.54 \).

2. Modeling review

The organization of mass filling of diesel engines with an oxidizer with stoichiometric composition of the fuel mixture and complete combustion of fuel vapors is a complex problem. The solution to this problem can be partially realized through the indicators of tension and the limits of the engine forcing. These include:

1. A detailed formulation of the effective power of the engine:
\[
Ne = \frac{\pi \cdot D^2}{4} \cdot S \cdot \eta_n \cdot \rho_a \cdot \frac{1}{\alpha \cdot l_0} \cdot Q_n \cdot \eta_e \cdot n \cdot \frac{1}{60} \cdot \frac{1}{m} \cdot i \cdot W_t.
\] (3)

2. System of equations of gas exchange process for open thermodynamic system (differential equation of energy conservation for system with variable mass, differential equation of mass conservation and equation of state of ideal gas mixture):

\[
\begin{align*}
\frac{du}{dt} &= i^* \cdot dM_i - i^*_E \cdot dM_E - dQ_H - p \cdot dV \\
\frac{dM}{dt} &= M \cdot R \cdot T
\end{align*}
\]

where \(i^*, i^*_E\) — the enthalpy of the inhibited flow in the boundary section of the inlet and outlet, respectively:

\[
\begin{align*}
i^* &= C_{p, air} \cdot T_i + \frac{\omega_i^2}{2} \\
i^*_E &= C_{p, waste} \cdot T_E + \frac{\omega_E^2}{2}
\end{align*}
\]

3. Equation of litre power:

\[
Ne_i = \frac{Ne}{i \cdot V_s}, \text{ W / l.} \quad (5)
\]

4. Piston power equation:

\[
Ne_p = \frac{Ne}{i \cdot \pi \cdot D^2}, \text{ W / m}^2. \quad (6)
\]

One of the most important means of reducing the weight and size of the engine is to boost its specific power. Methods of upgrading engines running on liquid fuel can be outlined from the expressions (1-6). From these expressions, it follows that the specific power depends on the following factors:

- the working volume of the cylinder taking into account under pistons seal in the workflow cycle;
- organization of mass filling of the engine cylinder with air, determined by the coefficient of purge, residual gases, flow, narrowing and filling;
- the use of gas-dynamic phenomena in the intake and exhaust gas exchange subsystem, characterized by hydraulic diameter and air density;
- application of stoichiometric composition of the fuel mixture determined by the speed of the fuel torch in the compression chamber;
- the use of light fuels with an increased cetane number, estimated by the chemical composition of the fuel, theoretically necessary for the complete combustion of one kilogram of fuel by the amount of air and the lowest calorific value of the fuel;
- the effective efficiency of the working cycle of the engine, the evaluated thermal, mechanical and relative coefficients;
- the number of strokes in the cycle, the number of cylinders and the speed of the engine crankshaft, estimated design and operational capabilities.

From these factors used to increase the specific power, it follows that the limits of forcing power with increasing fresh charge pressure require organization of the mass filling of the engine cylinder. At the same time, the increase in mechanical and thermal tension of the engine is the main reason limiting the mass filling of air entering the cylinder. It follows that, increasing the mass of the charge to increase the specific power, it is necessary to limit the maximum gas pressure in the cylinder and the rate of its increase, temperature and temperature gradients in the walls of the parts [4, 5, 6, 7]. This is achieved by the introduction of design and technological improvements, as well as the rational organization of the working process of the combined engine. There are several ways to reduce the
mechanical and thermal tension of the engine: reducing the compression ratio, reducing the angle of the injection timing advance, the choice of the appropriate fuel injection characteristics and the method of mixture formation.

Lowering the compression ratio of the diesel to 10-11 with an increase in the mass filling and limiting the maximum pressure in the cylinder allows keeping the degree of pressure increase approximately constant and thus avoiding a sharp deterioration in efficiency.

Figure 1 shows the indicator diagrams of diesels with different mass air filling of cylinders and compression ratios of 9.5 and 13.5, provided that $p_z=const$. Comparison of the indicator diagrams shows that with the increase in mass filling due to $p_k$, with the restriction of the value of $p_z$, duration of the combustion process increases; with $p_k=0.38$ MPa, the main amount of heat is supplied at $p=const$. This section is about 35 of angle of rotation of the crankshaft ($^\circ$arc). Such combustion process leads to a significant increase in the thermal intensity of the engine and the deterioration of its efficiency: the indicator specific fuel consumption at $p_k=0.38$ MPa is 224 g / kW/hour instead of 202 g / kW/hour at $p_k=0.175$ MPa and the same degree of compression 9.5.

3. Conclusion
The reduction of the lead angle of the injection timing of fuel to reduce the pressure $p_z$ leads to a prolongation of the combustion process. In this case, the combustion of fuel vapors mainly occurs during the expansion process, and this worsens the efficiency and increases the heat stress of the parts forming the combustion chamber, exhaust bodies and turbines.

Improvement of mixture formation and characteristics of fuel injection to reduce the maximum of the values $p_z$ and $\Delta p/\Delta \phi$ is possible to make by the factor of dynamic cycle speed and fuel injection [8, 9]. This makes it possible to increase the average effective pressure up to 30% without increasing the maximum cylinder pressure and combustion stiffness [10, 11].

Depending on the required efficiency of the engine, the rational mass filling and pressure of the $p_k$, and hence the level of forcing of the combined engine, are changed. Thus, the most advantageous mass filling and pressure of the $p_k$ for a four-stroke engine with a specific effective fuel consumption of 210 g / kW/hour varies from 0.165 to 0.32 MPa with increasing $p_z$ from 12 to 18 MPa and $\alpha=1.45$. 

![Figure 1. Indicator surface](image-url)
Analyzing the indicators characterizing the intensity and limits of forcing, it should be noted that the choice of a method of forcing depends on the type, design and purpose of the engine. The values of litre and piston power for some types of engines are given in table 1.

| Manufacturer | Engines | Power liter $Ne_{l}, \text{kW/l}$ | Piston power $Ne_{p}, \text{kW/dm}^2$ |
|--------------|---------|----------------------------------|----------------------------------|
| RUMO         | 8-cylinder 4-stroke supercharged with cylinder dimensions 22/28 | 14.7 | 41.1 |
| MAK          | 8-cylinder 4-stroke supercharged with cylinder dimensions 20/30 | 20.2 | 60.5 |
| Zulzer       | 8-cylinder 4-stroke supercharged with cylinder dimensions 20/30 | 13.8 | 41.4 |
| Wartsila     | 8-cylinder 4-stroke supercharged with cylinder dimensions 22/26 | 17.97 | 46.7 |

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