Determining Parameters of Double-Wiebe Function for Simulation of Combustion Process in Overload Diesel Engine with Common Rail Fuel Feed System

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Abstract. To analyze the peculiarities of the combustion process in an overload diesel engine with the system of Common Rail type with one-stage injection, the indicator diagram was registered. The parameters of the combustion process simulated by the double-Wiebe function were calculated as satisfactorily reconstructing the law of burning rate variation. The main parameters of the operating cycle obtained through the indicator diagram processing and the double-Wiebe function calculation differed insignificantly. And the calculated curve of the cylinder pressure differed notably only in the end of the expansion stroke. To improve the performance of the diesel engine, a two-stage fuel injection was recommended.

1. Introduction
With an ever increasing frequency, models on the two I.I. Wiebe Combustion Laws are used to simulate the operating cycles of diesel engines [1-6 et al]. These models allow one to reveal the duration of the characteristic periods and peculiarities of the variation of fuel burning rates in them. In case of high fuel injection pressure in Common Rail systems (over 150 MPa), fuel is sprayed more finely, and the injection duration reduces.

The research objective included determining the parameters of the double-Wiebe function for simulation of the combustion process in an overloaded diesel engine with Common Rail type system, and namely estimating the share of fuel involved in the initial period of combustion, as well as the duration and indicators of the combustion characteristics in the initial and main periods of the combustion process. It is convenient to determine the parameters of the double-Wiebe function using the results of analyzing the indicator diagram obtained during an actual engine testing. The knowledge of these parameters will allow one to reduce the scope of experiment research and to determine the trends of variations in fuel-injection equipment regulations in order to improve the vehicle diesel engines’ performance in case of overload.

2. Experimental Setup and Methodology
The calculation and experiment research was performed with regard to an overloaded vehicle diesel engine with the Common Rail type of fuel feed system with one-stage injection. This system had preliminarily been studied using a unique “Injection” research installation created at South Ural State University [7]. Main parameters of the diesel engine are given in Table 1.
Table 1. Parameters of Overloaded Vehicle Diesel Engine.

| Parameter                          | Value  |
|-----------------------------------|--------|
| Diesel engine capacity, kW        | 380    |
| Crankshaft speed, min⁻¹           | 1.500  |
| Number of cylinders               | 6      |
| Cylinder diameter D, m            | 0.15   |
| Piston stroke S, m                | 0.16   |
| Degree of compression             | 13     |
| Diameter of indicating canal, m   | 0.006  |
| Length of indicating canal, m     | 0.06   |
| Fuel pressure in battery, MPa     | 165    |

The experimental research on the diesel engine was performed with registering of the indicator diagram by the identification system FEV GmbH (Germany) at the pace of 1.0 degree of crank angle. The angular reamer was made by means of an angle sensor (encoder) installed on a free tailshaft of the engine. The obtained pressure diagrams were processed to reveal characteristics of heat release and parameters of the combustion process.

The method of analyzing the combustion process against the indicator diagram of its operation is based on the First Law of Thermodynamics and comprises an equation on calculating the amount of heat \( \Delta Q_{\text{comb}} \) released by fuel burning over the calculation pace from point 1 to point 2 with the crankshaft turn and relevant change of the volume of working medium in the cylinder. The method of calculating heat release in the engine’s cylinder allows one, starting with end-of-injection parameters and against elementary segments from point 1 to point 2 with the pace of \( \Delta \phi_{1-2} \) as per the crank angle, to calculate the current values of heat from burning, temperature, velocity of burnout, and share of burnout fuel at the calculation pace in compression and expansion strokes. While transiting from one elementary segment to the next one, the calculated parameters of the end of the previous segment (indexed 2) successively become the initial ones for the current segment (indexed 1).

\[
\Delta Q_{\text{comb}} = \frac{m_1 + m_2}{4} \left[ p_2 \left( v_2 - v_1 \right) \frac{k+1}{k-1} - p_1 \left( v_1 - v_2 \right) \frac{k+1}{k-1} \right] + \Delta Q_{W1-2},
\]

where \( m_1, m_2, p_1, p_2, v_1, v_2 \) are mass, pressure and specific volume of working medium in the beginning (point 1) and the end (point 2) of the calculation pace, respectively. The volume values are calculated against the known dependences of the kinematics of the crank-and-rod mechanism of an internal combustion engine (ICE);

\( k \) is an adiabatic parameter at the calculation pace; \( k \) is calculated against heat capacity of the mixture of gases composing the working medium;

\( \Delta Q_{W1-2} \) is heat rejected from the working medium to the walls of the above-piston volume over the calculation pace; it is calculated by the known Newton-Richmann law

\[
\Delta Q_{W1-2} = \frac{\alpha_i F_i (T_1 - T_w) \Delta \phi_{1-2}}{6n}.
\]

Here \( \alpha_i \) is the coefficient of release of heat from the working medium to the walls of the above-piston volume for medium-speed diesel engines with volume mixture generation, normally calculated by the known Voshni expression; \( F_i, T_i, T_w \) are area and temperature of the surface of the above-piston volume in contact with the working medium in the beginning of the calculation pace; at the first approximation, temperatures of all internal surfaces of the above-piston volume are taken as constant and equal to \( T_w \) due to high heat capacity and heat conductivity of structure materials; \( T_i \) is the working medium temperature at the beginning of the calculation pace; \( n \) is crankshaft speed, min⁻¹.

The heat release rate in the engine’s cylinder over the calculation pace is determined by ratio
Further on, the amount of fuel burnt over the calculation pace is determined:

\[ \Delta m_{\text{comb},1-2} = \frac{\Delta Q_{\text{comb},1-2}}{\xi H_u}, \]

where \( \xi \) is the combustion efficiency coefficient; \( H_u \) is the fuel’s lower calorific value, MJ/kg.

This amount of fuel related to calculation pace \( \Delta \varphi \) gives us the burnout velocity:

\[ \frac{dm_{\text{comb}}}{d\varphi} = \frac{\Delta m_{\text{comb},1-2}}{\Delta \varphi}. \]

The total amount of fuel burnt over a cycle is calculated by the following expression:

\[ m_{\text{comb}} = \sum_{i=1}^{n} \Delta m_{\text{comb},i-(i+1)}. \]

The share of the burnout fuel at the \( n \) pace of calculation is determined by expression

\[ x = \frac{m_{\text{comb}}}{m_f}, \]

where \( m_f \) is cycle fuel feed, kg.

Next, the average temperature of the working medium in point 2 is calculated by the gas equation:

\[ T_2 = \frac{p_2 v_2}{R}. \]

While processing the indicator diagram, the main indicator parameters of the operating cycle are calculated by the known formulas.

Indicator cycle operation: \( l_i = \sum_{i=1}^{n} l_{i-2} \), where \( l_{i,2} \) is work performed by gas in the engine’s cylinder over the calculation pace, \( l_{i-2} = \frac{(p_2 + p_1) \cdot (v_2 - v_1)}{2} \).

Average indicator cycle pressure: \( p_i = \frac{4 \cdot l_i}{\pi \cdot D^2 \cdot S}. \)

Indicator capacity: \( N_i = \frac{p_i \cdot n \cdot \pi \cdot D^2 \cdot S}{480}. \)

Indicator efficiency rate: \( \eta_i = \frac{l_i}{H_u \cdot m_f}. \)

Indicator specific fuel consumption: \( g_i = \frac{3600}{\eta_i \cdot H_u}. \)

Simulation of the combustion process was performed using the new model and program of calculating the operation cycle of the cylinder by the two I.I. Wiebe Combustion Laws [8] developed at South Ural State University and based on the following expression:

\[ x = \Phi \times \left\{ 1 - \exp \left[ -c \left( \frac{\varphi_{\text{fir}}}{\varphi_{z,\text{fir}}} \right)^{m_{\text{fir}}+1} \right] \right\} + (1 - \Phi) \times \left\{ 1 - \exp \left[ -c \left( \frac{\varphi_{\text{sec}}}{\varphi_{z,\text{sec}}} \right)^{m_{\text{sec}}+1} \right] \right\}, \]

where \( \Phi \) is the share of fuel of the total cycle feed which burns at the initial period; \( \varphi_{\text{fir}}, \varphi_{\text{sec}} \) are the current values of the angles of the initial and main periods; \( \varphi_{z,\text{fir}}, \varphi_{z,\text{sec}}, m_{\text{fir}}, m_{\text{sec}}, m_{\text{sec}} \) are the durations and characteristic indicators of the initial and main periods of the combustion process, respectively.
3. Results and Discussion

As per results of indicating the cylinder pressure at \( n = 1500 \text{ min}^{-1} \), the analysis was performed regarding the processes of mixture generation and combustion during operation of the diesel engine with the Common Rail type of the fuel feed system. The results of processing the indicator diagram are given as graphs in Figure 1. Numerical values are given in Table 2.

![Figure 1. Variations of pressure and temperature of the working medium in the engine’s cylinder in compression and expansion strokes.](image)

Table 2. Main Parameters of the Operation Cycle.

| Parameters                          | Indicator Diagram Processing | Calculation at 2 Wiebe max |
|-------------------------------------|------------------------------|---------------------------|
| Indicator cycle operation \( l_i, J \) | 5.779                        | 5.784                     |
| Average indicator cycle pressure \( p_i, \text{MPa} \) | 2.044                        | 2.046                     |
| Indicator capacity \( N_i, \text{kW} \) | 72.24                        | 72.3                      |
| Indicator efficiency rate \( \eta_i \) | 0.3807                       | 0.3811                    |
| Indicator specific fuel consumption \( g_i, \text{g/kW*h} \) | 223.04                       | 222.8                     |
| Maximum combustion pressure \( p_{\text{max}}, \text{MPa} \) | 16.89                        | 16.78                     |
| Maximum gas temperature \( T_{\text{max}}, \text{K} \) | 1.810                        | 1.798                     |
| Fuel cycle feed \( m_f, \text{g} \) | 0.358                        | 0.358                     |

Obviously, these oscillations are caused by the indicating canal’s geometry influencing the pressure sensor readings. A long canal causes acoustic oscillations with the speed of sound depending on gas temperature in the cylinder. The graph on variations of the average rate of heat release is also given in Figure 2.
Figure 2. Change in the heat release rate in the engine’s cylinder with consideration of (blue line) and without consideration of (orange line) harmonic oscillations, provided that the pure compression process is $\Delta Q_{\text{comb.1-2}}=0$.

The experimental temperature curve (Figure 1) drops in the range from 25 to 15 degrees of the crank angle before the upper dead center (UDC). This is caused by excessive heat release from the working medium in the range from 36 to 15 degrees of the crank angle before UDC, which reveals itself as a negative heat release rate shown in Figure 2. It is obvious that this is caused by consumption of heat for fuel heating and evaporation not taken into account in the method of the combustion process analyzing.

The process of combustion starts at 20 degrees of crank angle after the beginning of injection. The period of firing delay amounts to 2.2 ms. Such big delay may be caused by big energy consumption for heating and evaporation of fuel injected at high speed during one-stage injection [9]. This results in significant overcooling of the fresh charge in the fuel flare and slowing down of the combustion process. The maximum rate of heat release of 300 J/degrees of the crank angle is achieved within 28-30 degrees of the crank angle. At 130 degrees of crank angle discharge valves open, what manifests itself as negative heat release rate.

The shape of the heat release rate curve (Figure 2) differs from the standard one typical for other diesel engines shown in Figure 3. One may state that during the fuel combustion process in the studied diesel engine during one-stage injection it is difficult to differentiate kinetic period of combustion which is characterized with high heat release rate within 5–10 degrees of crank angle, as well as the diffusion period with the second maximum of heat release rate.

Figure 3. Characteristics of heat release in an overloaded motor vehicle diesel engine [10].
The results of simulation of the operating cycle for selecting parameters of each of the two combustion stages of the overloaded vehicle diesel engine in the same mode as during the experimental testing are given in Table 2 and in graphs in Figures 1 and 4. These demonstrate that the main indicators of the operating cycle obtained during the indicator diagram processing and calculation by the double-Wiebe function differ insignificantly. And the calculated cylinder pressure curve notably differs only in the end of the expansion stroke when rejection from the cylinder starts not taken into account by the calculation program. The calculated parameters of the combustion process simulation are given in Table 3. The curves in Figures 2 and 4, when compared, reveal that the selected Wiebe simulation parameters satisfactorily reconstruct the law of burning rate variation in this overloaded diesel engine with Common Rail system during one-stage injection.

| Table 3. Parameters of Double-Wiebe Function Simulation of the Combustion Process. |
|-----------------|-----------------|
| Parameter        | Initial Stage   | Main Period |
| Fuel share       | 0.65            | 0.35        |
| Initial stage angle, degrees of crank angle | 9 before UDC | 25 after UDC |
| Period duration, degrees of crank angle     | 70              | 130         |
| Combustion character parameter               | 1               | 1.1         |

![Figure 4. Variation of the rate of heat release in the engine’s cylinder during double-Wiebe function simulation of the combustion process.](image)

The obtained share of burnout fuel at the initial period significantly differs from value of 0.38 recommended as per results of the optimization calculations [8], though the indicators of the combustion character comply with the ones recommended earlier [8].

4. Conclusion

1. The processing of the indicator diagram of the overloaded vehicle diesel engine with the fuel feed system of Common Rail type during one-stage injection resulted in the indicator parameters and characteristics of the cylinder heat release with practically no kinetic period of combustion with high heat release rate within 5–10 degrees of the crank angle, typical of other diesel engines.
2. The following parameters of the combustion process during the double-Wiebe function simulation were calculated: share of burnout fuel at the initial period amounts to 0.65; durations of the initial and main periods equal 70 and 130 degrees of crank angle; indicators of the character of the
initial and main periods are 1 and 1.1; angles of the beginning of combustion at the initial and main periods equal 9 degrees of the crank angle before UDC and 25 degrees of the crank angle after UDC. They satisfactorily reconstruct the law of burning rate variation in this overloaded diesel engine during one-stage injection, but differ from the recommended ones as per results of the optimization calculations [8].

3. The main indicators of the operating cycle obtained during processing of the indicator diagram and calculation by the double-Wiebe function differ insignificantly. And the calculated cylinder pressure curve notably differs only in the end of the expansion stroke.

4. To improve the performance of the vehicle diesel engine with the system of Common Rail type, it is reasonable to use two-stage fuel injection.

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