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THERMODYNAMIC ANALYSIS OF THE CONROD-FREE ENGINE

Summary. The paper considers the problems related with the thermodynamic analysis of the conrod-free engine. A mathematical model of the intake system of the conrod-free engine is proposed, which takes into account the kinematics of the slider-crank mechanism. The computational model of the intake system of the engine with the slider-crank mechanism is presented. The model allowed determining dependencies of the mixture temperature in the intake pipeline on the rotation frequency of crankshaft and on the angle of opening of the throttle gate. The total pressure losses in the engine intake system are determined taking into account the kinematics of the slider-crank mechanism. The mathematical model of the intake system of engine allows evaluating the measures aimed at optimizing the engine design and increasing its fuel efficiency.

1. INTRODUCTION

The main trend in modern automotive industry is the improvement of environmental and economic performance of piston engines.

The existing engines with the classic power mechanism are not economical enough. The design of the engines becomes much more difficult while improving the fuel economy.

Above listed requirements correspond to engines of a non-traditional design. We can consider an engine with a conrod-free slider-crank mechanism (SCM) with variable compression ratio (VCR).

Partial conditions constitute a significant part of the total running time of the engines. Therefore, it is impossible to predict fuel consumption in the engine operation at partial conditions. The operation of engines with an unconventional power mechanism on partial conditions has not been practically investigated. An integral part of the calculation is the modeling of the engine intake system with an unconventional power mechanism in partial conditions.

The foregoing points to the relevance of the topic. There is a need to investigate the economic effectiveness of a conrod-free engine (CFE) in terms of its application in road transport.

The constructions of engines with VCR were engineered by constructors of companies such as Ford, Mercedes-Benz, Nissan, Peugeot and Volkswagen. VCR is a technology to adjust the compression ratio of an internal combustion engine while the engine is in operation. This is done to increase fuel efficiency while under varying loads. The engineers of research institutes and companies have received thousands of patents [1-5].

The engines with original kinematics are being explored now [6-8]. An author [6] has analyzed the equilibrium of the engine with VCR and working volume. It is shown that the equilibrium of traversing mechanisms at moderate values of the number of cylinders of an engine is complicated on comparison with other known constructions.

The research and the engineering works on the creation of an engine with a SCM are being conducted in many countries [9]. From the analysis of works of some authors [10-16], we can conclude that the VCR is easier to implement on a CFE with an SCM.

The method of modelling the working processes in the combustion chamber of the internal combustion engine (ICE) had been developed. The method includes the description of the preparation of the 3D pattern of the engine combustion chamber, the putting of a finite elemental grid, and the problem of boundary conditions [17-19].

The method of simulation of thermal processes in a low-power piston engine was developed [20]. Modern design packages were used for simulation of this process.

A diverse numerical simulation of the scavenging of the intake valve system of the ICE with using a new implicit modification of the method of large particles was conducted [21-23]. The recommendations for improving the consumed characteristics of the gas exchange system of the classical engine were obtained.

However, in these works, there are no methods and results of the study taking into account the kinematics of an unconventional power mechanism of the engine with VCR.

The examples of an unconventional power mechanism are the Stiller-Smith engine [24], the engine with the hypocycloid power mechanism [25], the Balandin engine [26], slider-crank four-bar straight line guide mechanism Scott-Russell and the Bourke engine. These engines have the following characteristics:

- dynamic balance;
- a high uniformity of stroke;
- high mechanical efficiency;
- high operating life of a cylinder-piston group; and
- low noise and vibration.

Consequently, a CFE has great prospects for use in cars.

For reaching this topic, the next task was to simulate the engine intake process and determine the total pressure loss in the engine intake system while taking into account the kinematics of an unconventional power mechanism.

Physics-mathematical modelling of motion of air flow at the intake is performed for the flow of gases in stationary conditions. It is difficult to judge about the results of theoretical studies the reliability and degree of applicability of the data [27-28]. The modelling is researched based on the processes in the existing design of the air-gas system of engine for remedial action.

The main difference between the traditional model and the proposed model for calculating the intake system is that the mixture temperature in the intake pipeline is calculated from the obtained dependences on the basis of experimental data, taking into account the load and rotation frequency of crankshaft.

2. THERMODYNAMIC ANALYSIS OF THE INTAKE PROCESS OF ENGINE

2.1. Refinement of the mathematical model of the intake process of the engine

The engine model is made up of three sub-models:

- intake system;
- cylinder; and
- exhaust system.

The model of the intake system has a good look. The model consists of an air cleaner, a throttle gate, an intake pipeline, a cylinder head and an intake valve.

The assumptions were made to compile the mathematical model. The pressure before the intake valve is constant and equal to the average conditional pressure in the intake pipeline. The thermal condition of fresh charge is stable. Temperature and charge density are constant. The flow of gas in the intake pipeline of the engine is considered to be constant. In the intake pipeline on partial loads of the engine, the flow after the throttle gate is not reinstated. The pressure before the intake valve is equal to the gas pressure in the flow passage of the throttle gate. It takes into account the loss of pressure in the air cleaner, the cylinder head and friction along the length of the pipeline. With a small pressure difference, the compressibility of gas is negligible.

Taking into account the assumptions, the simulation scheme of the intake system of engine was compiled. The difference between the calculation process and the actual one is taken into account by the corresponding coefficients.

2.2. The calculation model the intake system of engine

The calculation model of the intake system of the engine is shown in Fig. 1.

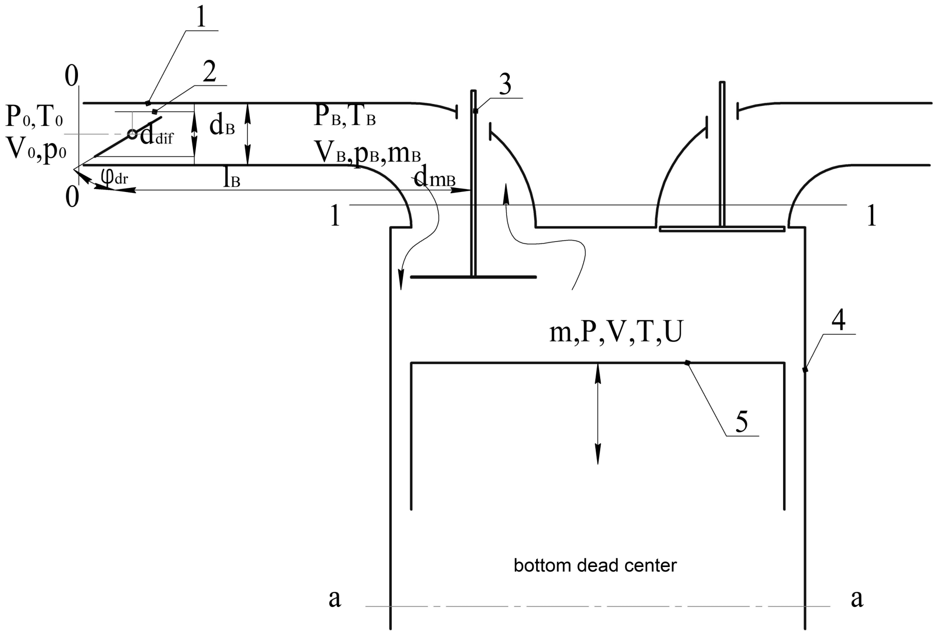

Fig. 1. Computational model of the intake system of the engine: 1 – intake pipeline; 2 – throttle gate; 3 – intake valve; 4 – cylinder; 5 – piston

In the proposed intake system, there are three sections:

- the cross section 0-0 (the air velocity is taken equal to zero ($v_0 = 0$) and the pressure is equal to the atmospheric pressure p_0);
- the cross section 1-1 is taken before the intake valve; and
- the cross-section *a*-*a* is located at the end of the inlet.

The process of intake system is considered in the following formulation: the parameters of the fresh charge stream [p_B, T_B] before the intake valve are determined, taking into account the load φ_{dr} and rotation frequency of crankshaft *n*.

Thermodynamic parameters of the mass of the mixture in the engine cylinder are determined using the equation of conservation of mass, the heat exchange, the first law of thermodynamics and the state equation of gas according to known the flow parameters in front of the intake valve.

The calculation of the parameters of the fresh charge in the intake system is performed using the known hydraulics formulas.

2.3. The determination of mixture temperature in the intake pipeline

Taking into account the speed and loading conditions of the engine, dependence of the temperature change of the mixture before the intake valve is obtained using experimental data of the temperature of the mixture in the intake pipeline [29], which are approximated for modern engines:

$$T_B = T_B^n \frac{T_B^{dr}}{T_B^n} \tag{1}$$

where T_B^n – mixture temperature [K] depending on rotation frequency of crankshaft *n,* min^{-1}; T_B^{dr} – mixture temperature [K] depending on the opening angle of the throttle gate φ_{th} , %.

Mixture temperature from a rotation frequency of crankshaft *n* is determined:

$$t_B^n = 77 - \left(77 - t_B^N\right) \cdot \left(\frac{1600 + n}{1600 + n_N}\right)^{\frac{1}{h}} \tag{2}$$

where t_B^N – mixture temperature at nominal condition of the engine, ºC; n_N – nominal rotation frequency of crankshaft, min^{-1}; *h* – index of power, *h* = 4.

Mixture temperature [K] is determined as follows:

$$T_B^n = t_B^n + 273.15. \tag{3}$$

Depending on the size of the opening of the throttle gate [%], the mixture temperature is determined by the system of equations:

$$\begin{cases} t_B^{dr} = 85 - \left(85 - t_B^N\right) \cdot \left(\dfrac{\varphi_{dr}}{100}\right)^{\frac{1}{h_1}} \text{when } \varphi_{dr} \le \varphi_{dr}^k \le 100\% \\ t_B^{dr} = 55 - \left(55 - t_B^k\right) \cdot \left(\dfrac{\varphi_{dr}}{\varphi_{dr}^k}\right)^{\frac{1}{h_2}} \text{when } 0 \le \varphi_{вк} \le \varphi_{dr}^k \end{cases} \tag{4}$$

where φ_{dr}^k = 25 % – intermediate angle of opening of the throttle valve; h_1 = 3.5; h_2 = 2 – index of power; t_B^k – mixture temperature at φ_{dr}^k, ºC;

$t_B^k = t_B^{dr}\left(\varphi_{dr}^k\right)$.

Temperature [K] is determined as follows:

$$T_B^{dr} = t_B^{dr} + 273.15 \quad . \tag{5}$$

Lowering the temperature T_B owing to fuel evaporation in the intake pipeline is not taken into account.

Averaged temperature T_p from the piston motion is determined by following equation:

$$T_p = \frac{1}{\Delta\varphi_i} \int_0^{\varphi} T_c(\varphi) d\varphi \quad , \tag{6}$$

where φ – crank angle; $\Delta\varphi_i$ – cycle time (degree); T_c – temperature over the height of a cylinder bearing surface, K.

A value T_p represents the average integral temperature of the cylinder bearing surface at the current working volume of the cylinder.

Temperature T_c over the height of a cylinder bearing surface changes from 60 to 190 K. In the model, temperature-height graph of a cylinder (over piston stroke) is taken into account. Temperature T_c is determined by the empirical formula obtained by approximating the experimental data:

$$T_c = T_{TDC} - \left(T_{TDC} - T_{BDC}\right) \cdot \left(\bar{s}(\varphi)\right)^{\frac{1}{z}} \quad , \tag{7}$$

where T_{TDC} – temperature of cylinder wall near top dead center, K; T_{BDC} – temperature of cylinder wall near bottom dead center, K; *z* – index of power (*z* = 2.8); $\bar{s}(\varphi)$ – relative movement of piston.

2.4. The determination pressure of the mixture in the intake pipeline

Pressure losses at sections 0-0 and 1-1 consisted of loss from hydraulic resistance along the length of the flow and loss from local resistances.

Total pressure loss in the intake engine is determined as follows:

$$\Delta p_B = \Delta p_{tr} + \Delta p_{dr} + \Delta p_{no} + \Delta p_{rc} \quad , \tag{8}$$

where Δp_{tr} – pressure loss from hydraulic resistance in a pipeline, Pa; Δp_{dr} – pressure loss from hydraulic resistance in a throttle gate, Pa; Δp_{no} – pressure loss from hydraulic resistance in an air cleaner, Pa; Δp_{rc} – pressure loss from hydraulic resistance in a cylinder head, Pa.

The friction pressure losses in the intake pipeline between the throttle gate and the section 1-1 are calculated by the Darcy formula [30]:

$$\Delta p_{tr} = \lambda_{in} \cdot \frac{l_B}{d_B} \cdot \rho_{In} \cdot \frac{v_B^2}{2} \quad , \tag{9}$$

where λ_{in} – The Darcy coefficient, which characterizes the resistance of flow along length of the intake pipeline; l_B – length of the considered section of the flow in the intake pipeline; d_B – internal diameter of the intake pipeline; v_B – average per-section speed of the motion of charge in the intake pipeline; ρ_{In} – pre-determined average charge density at the considered section of the intake pipeline.

The relative loading λ_N depending on a value of opening of a throttle gate φ_{dr} and rotation frequency of crankshaft n is calculated by the following formula:

$$\lambda_N = 1 - \left(\frac{100 - \varphi_{dr}}{100 - \varphi_{drxx}} \right)^m \quad , \tag{10}$$

where m – index of power, which is determined by the dependence:

$$m = (m_{\max} - m_{\min}) \cdot \left(\frac{n_{\max} - n}{n_{\max} - n_{\min}} \right)^a + m_{\min} \quad . \tag{11}$$

Accepted the following: $m_{max} = 4.5$; $m_{min} = 2.041$; $n_{max} = n_{dis}$;
$n_{min} = 0.25 \cdot n_N$; $a = 4.5$,
where n_{dis} – distinct rotation frequency of crankshaft, min^{-1}.

The value of the local resistance ξ_{dr} is determined by the Weissbach formula [31], which is the function of the angle of opening of the throttle gate:

$$\xi_{dr} = \frac{2 \cdot \Delta p_{dr}}{\rho_0 \cdot v_B^2} \quad , \tag{12}$$

where pressure loss in the throttle gate is determined as follows:

$$\Delta p_{dr} = \Delta p_k - \Delta p_{no} - \Delta p_{dif} \quad , \tag{13}$$

where Δp_k – dilution in a intake pipeline, Pa.

2.5. The determination of pressure loss in the air cleaner

Pressure loss in the air cleaner in partial conditions is determined by the formula:

$$\Delta p_{no} = \xi_{no} \cdot \rho_0 \cdot \frac{v_{no}^2}{2} \quad , \tag{14}$$

where ξ_{no} – coefficient of local resistance, which is determined:

$$\xi_{no} = \frac{2 \cdot \Delta p_{no}^N}{\rho_0 \cdot v_{noN}^2} \quad , \tag{15}$$

where Δp_{no}^N – pressure loss in the air cleaner at the nominal mode [31], Pa. This is determined as follows:

$$\Delta p_{no}^N = (0.05...0.1) \cdot \Delta p_{tr} \quad , \tag{16}$$

where Δp_{tr} – total resistance of the intake line at nominal condition, Pa. This is determined according to the data [31]:

$$\Delta p_{tr} = (11.5...20.4) \cdot 10^3 \quad . \tag{17}$$

Accepted the following: $\Delta p_{tr} = 18 \cdot 10^3$ Pa.

2.6. The determination pressure loss in the cylinder head

Pressure loss from hydraulic resistance in a cylinder head is accepted on the basis of results of experimental studies of hydraulic resistance of elements of the intake system of different ICE [6, 9-11]:

$$\Delta p_{rc} = \Delta p_{trub} \quad . \tag{18}$$

This analysis shows that losses in local resistance in all cases are owing to more losses on friction. At the same time, the pressure loss in the throttle gate is determining at partial operating conditions of the engine.

3. THE RESULTS OF THE MATHEMATICAL MODELING OF THE INTAKE PROCESS OF ENGINE

Experimental studies were carried out on a single-cylinder gasoline ICE both conrod-free SCM and classic power mechanism.

Comparative evaluation is qualitative and quantitative comparisons of parameters of intake system of the CFE with classic engine.

The change in the mixture temperature of the intake pipeline before the intake valve is dependent on the following:

- the rotation frequency of crankshaft n according to the Formula (2), which is shown on Fig. 2, and
- the value of the opening angle of the throttle gate φ_{th} by the Formula (4), which is shown on Fig. 3.

Experimental data of the mixture temperature of the intake pipeline before the intake valve were approximated for ICE depending on the extent of throttling, which is shown below for Tab. 1.

Table 1

Dependence of mixture temperature of the intake pipeline t_B^{dr} on opening angle of the throttle gate φ_{th}

Opening angle of the throttle gate φ_{th} [%]	0	5	10	15	25	50	75
Mixture temperature t_B^{dr} [ºC]	53	52.8	52.2	51.2	48	40	34

Dependence of mixture temperature of the intake pipeline on the rotation frequency of crankshaft n is shown below for Tab. 2.

Table 2

Dependence of mixture temperature of the intake pipeline t_B^{n} on rotation frequency of crankshaft n

Rotation frequency of crankshaft n [min^{-1}]	800	1000	2000	3000	4000	5000	5400
Mixture temperature t_B^{n} [ºC]	41	40	36.5	35.2	32	30.5	30

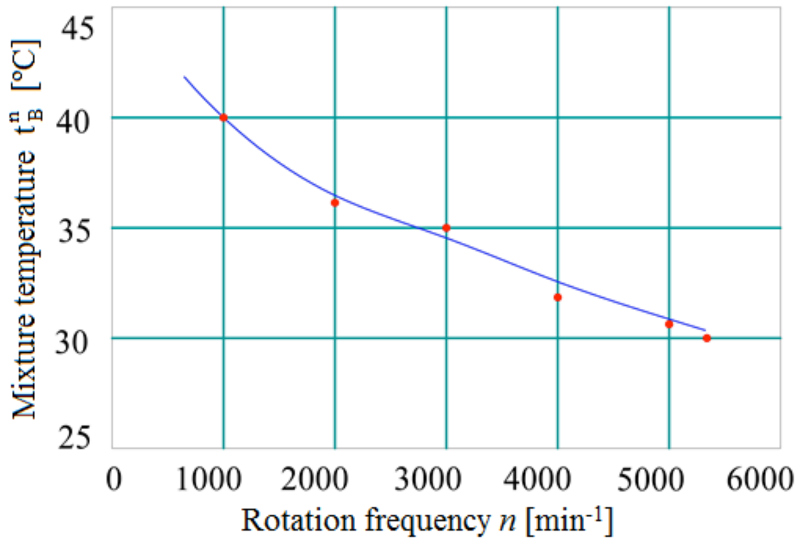

Fig. 2. Dependency of the mixture temperature t_B^n [ºC] on the rotation frequency n [min^{-1}]: ··· - experiment; — - calculation

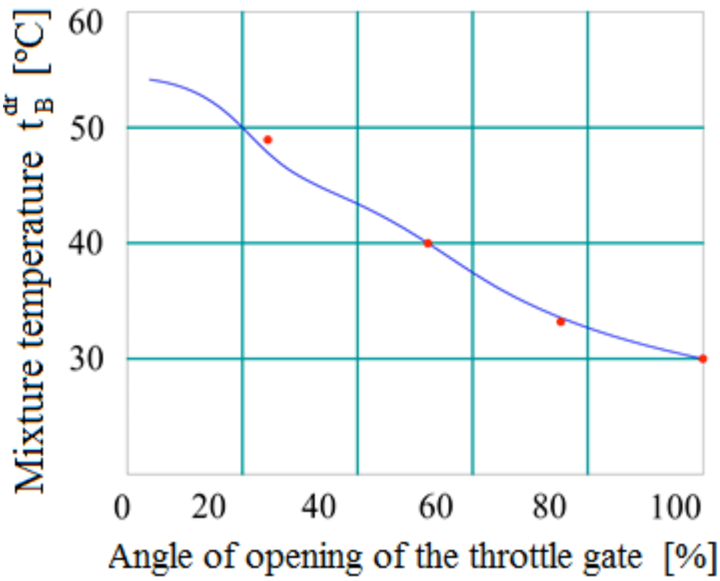

Fig. 3. Dependency of the temperature of the mixture t_B^{dr} [ºC] on the angle of opening of the throttle gate φ_{th} [%]: ··· - experiment; —— - calculation

The model of the intake system of engine is based on the equations of pressure loss in the elements of the intake system. Parameters of the intake system are used for further calculations of processes of compression, combustion, expansion and exhaust out.

The mathematical model has an advantage over known programs regarding the following:

- less labor-intensive in the calculation;
- less machine time in the calculation; and
- estimating the measures, which aimed for optimizing the design of an engine and increasing its fuel efficiency.

The adequacy of the mathematical model was determined by quantitative comparison of model and experimental results. The criterion of adequacy was determined in the variant, when the comparison is carried out on variable parameters, but for fixed points (experiments) by varying independent factors in terms of speed and loading characteristics.

Comparative evaluation of efficiency of conrod-free SCM (effective specific fuel consumption g_e; indicated specific fuel consumption g_i) and classic power mechanism [$g_{e_{cl}}$; $g_{i_{cl}}$] is shown on Fig. 4.

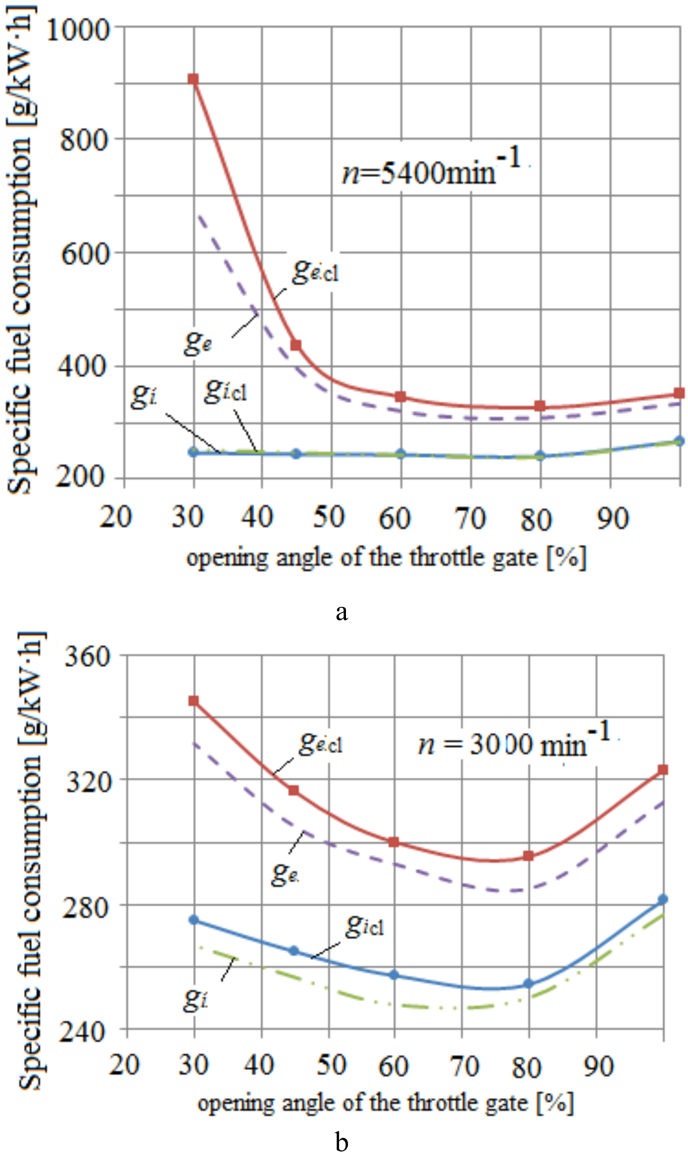

Fig. 4. Load characteristic of engine of conrod-free SCM [g_e ; g_i] and classic power mechanism [$g_{e_{cl}}$; $g_{i_{cl}}$]: a) n = 5400 min^{-1}; b) n = 3000 min^{-1}

Engine of conrod-free SCM has greater indicated efficiency at the average of 2-3%. This indicates the best behavior of the thermodynamic cycle in a CFE as compared with classic ICE (see Fig. 4).

The values of change of effective specific fuel consumption Δg_e correspond to engine behavior, which is shown in Tab. 3.

Table 3

Change of effective specific fuel consumption $\Delta\, g_e$
corresponds to engine behavior [φ_{th} ; n]

Change of effective specific fuel consumption $\Delta\, g_e$ [%]	Opening angle of the throttle gate φ_{th} [%]	Rotation frequency of crankshaft n [min^{-1}]
3.1	100	5400
3.2	100	3000
7.8	60	5400
3.7	60	3000
16.7	30	5400
4.5	30	3000

CFE has an advantage over classic ICE when the average value $\Delta\, g_e$ = 10-20%.

CONCLUSIONS

1. The analysis of engines was conducted with different kinematics of the power mechanism and VCR.
2. The mathematical model of the intake system of CFE is devised. The change in the mixture temperature of the intake pipeline before the intake valve is obtained.
   The temperature distribution along the height of the cylinder (along the piston stroke) is taken into account in the model under consideration; this is the averaged temperature along the piston stroke. The temperature *Tp* is determined from the empirical formula obtained by approximating the experimental data.
3. Determination of the total pressure losses in the intake system of the engine was done. The resulting model allows evaluation of the measures aimed at optimizing the design of the engine and improving its fuel economy.

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