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SIMULATION OF THERMOPHYSICAL PROCESSES IN A ROTOR-BLADE ENGINE WITH EXTERNAL HEAT SUPPLY BY NODAL ANALYSIS

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Abstract. The well-known algorithm of mathematical modeling in nodal values has been used to analyze the operation of a rotary-blade engine with external heat supply. The calculation procedure allows obtaining the required combination of exchange processes between the introduced elements using the formation of a graph in the form of a connection matrix when the switching of nodes depends on time. The values of thermodynamic functions found as a result of calculation by a known algorithm were compared with their values taken from indicator diagram characteristics. The influence of the finite rate of heat and mass transfer processes on the pressure and temperature values within a repetitive cyclic process was demonstrated.

Keywords: rotary-blade engine, nodal analysis method, cyclic process, numerical simulation

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МАТЕМАТИЧЕСКОЕ МОДЕЛИРОВАНИЕ МЕТОДОМ УЗЛОВОГО АНАЛИЗА ТЕПЛОФИЗИЧЕСКИХ ПРОЦЕССОВ В РОТОРНО-ЛОПАСТНОМ ДВИГАТЕЛЕ С ВНЕШНИМ ПОДВОДОМ ТЕПЛА

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Аннотация. Известный алгоритм математического моделирования в узловых значениях применен для анализа работы роторно-лопастного двигателя с внешним подводом тепла. Методика расчета позволяет при помощи формирования графа в виде матрицы связей, когда коммутация узлов зависит от времени, получать требуемое сочетание обменных процессов между введенными элементами. Значения термодинамических функций, найденные в результате расчета по известному алгоритму, сопоставлены с их значениями, взятыми из индикаторных диаграмм. Продемонстрировано влияние конечной скорости протекания процессов тепло- и массообмена на значения давления и температуры в рамках повторяющегося циклического процесса.

Ключевые слова: роторно-лопастной двигатель, метод узлового анализа, циклический процесс, численное моделирование

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Introduction

The ultimate answer to the rising demand for environmentally friendly engines is the design incorporating an external heat supply. In this case, it is assumed that the thermodynamic process with the working fluid occurs in a closed cycle. Several types of engines have been designed to implement a cyclic process including heat transfer accompanied by heat dissipation from the working fluid, compression, heat transfer with heat addition, and expansion (power cycle).

If the transformation in the state of the working fluid is described by two isotherms and two adiabats, such a cycle is considered perfect (Carnot cycle). Its efficiency is determined by the temperature level of the working fluid in the heater and cooler. Machines operating based on the Stirling cycle provide efficiencies on a par with those based on the Carnot cycle. The Stirling cycle is described using two isotherms and two isochores. In general, the energy conversion efficiency of the process is lower than the ideal, but using a heat regenerator in the cycle also allows to achieve high efficiency values.

Notably, the work output from a Stirling cycle is greater than from a Carnot cycle for the same temperatures on the isotherm lines. For this reason, there is an ongoing search for a viable engine design with the power cycle more or less approaching the Carnot or Stirling process.

There are three main modifications of the Stirling piston engine: α-, β- and γ-type. They differ in the number of cylinders used and the organization of the heat transfer process [1]. However, all modifications have a common disadvantage: the working fluid undergoes oscillatory motion between the heater and the heat sink. In most cases, this motion of the working fluid leads to additional losses of energy spent on acceleration and deceleration. This drawback can be partially eliminated by organizing circulatory motion of the working fluid between the heat exchangers. The mechanism for maintaining inertial motion of the working fluid is implemented in a number of designs, but a rotor blade engine with an external heat supply has certain advantages [2].

Compared to the Wankel rotary piston engine and the Balandin conrod-free engine, a rotor blade engine can operate at very high speeds. All parts are well-balanced, gas distribution is regulated by valves.

The cycle of an ideal rotor blade engine is organized as two isochores and two adiabats. Heat is supplied and removed in isochoric processes, adiabatic processes are used to perform work and compress gas. The work performed and the efficiency of such a cycle can be estimated from the temperature differences under heating and cooling of the working fluid.

Introducing the quantities T^{heat}_s and T^{heat}_e (K) as the initial and final temperatures of the working fluid in the heater, and the quantities T^{cool}_s and T^{cool}_e (K) as the initial and final temperatures in the cooler, we can determine the specific values of supplied and extracted heat q^{heat} and q^{cool} (J/kg), as well as the specific work A (J/kg) by the expressions

$$\begin{gathered} q^{heat} = c_V\left(T_e^{heat} - T_s^{heat}\right), \\ q^{cool} = c_V\left(T_e^{cool} - T_s^{cool}\right), \\ A = q^{heat} + q^{cool}, \end{gathered} \tag{1}$$

where c_V, J/(kg·K), is the specific heat capacity at a constant volume.

The efficiency for the process under consideration is

$$\eta = \frac{A}{q^{heat}} = 1 - \frac{T_s^{cool} - T_e^{cool}}{T_e^{heat} - T_s^{heat}}. \tag{2}$$

Since the pairs of nodal temperature values in the cycle belong to the same adiabate, expressed in terms of variables $T-V$ by the dependence $TV^{\gamma-1} = const$, we can introduce the quantity n as the ratio of the maximum to the minimum volume in the given cycle, and determine the efficiency in terms of the volume variation and the adiabatic index:

$$\eta = 1 - n^{1-\gamma}. \quad (3)$$

It follows from the above formula (3) that a high efficiency of the process can be obtained if the volume ratio is as high as possible, while the preferable working fluid is monatomic gas with the highest value of the adiabatic index γ. An illustration of the ideal cycle of a rotor blade engine in the $p-V$ diagram is shown in Fig. 1.

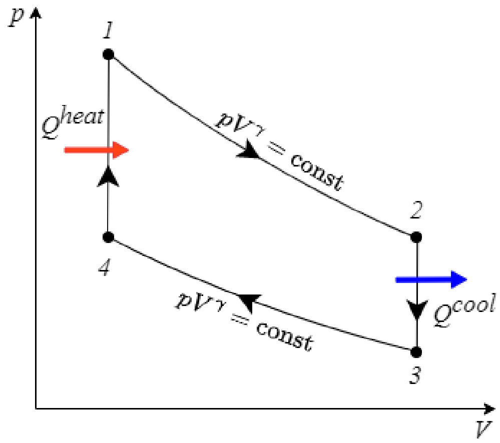

Fig. 1. $p-V$ diagram for ideal thermodynamic cycle of a rotor blade engine. The figure shows isochores (lines $2 \to 3$, $4 \to 1$) and adiabates (lines $1 \to 2$, $3 \to 4$)

In addition to the above-mentioned processes of adiabatic compression and expansion, isochoric heat addition/removal, the real cycle includes the flow of the working fluid to and from heat exchangers. The outflow of gas from the chamber or, vice versa, into the chamber occurs at a finite speed, limited at least by the speed of sound. Heat transfer processes between the working fluid and the bounding surfaces in the heater and cooler also occur at finite rates.

Thus, mathematical modeling accounting for the time dependence of thermodynamic functions should be performed to correctly describe the overall processes in the rotor blade engine.

Several notable studies considered the the characteristics and operating modes of the rotary-blade engine. General assessment of calculation methods and approaches to mathematical modeling for rotor blade engines are discussed in [3]. A particular conclusion is that the paradigm of nodal analysis is the most practical for primary analysis within the framework of mathematical modeling. Gas dynamic processes in containers with variable volume are considered in [4] in the phases of compression or expansion, taking into account heat transfer on the bounding surfaces. The effect of leakage in seals on the performance characteristics of the engine was assessed in [5]. Calculation formulas were obtained in [6, 7] for the problem statement based on the laws of conservation of mass and energy. A special case of gas filling a container with variable volume was considered.

Summarizing the above studies, we should stress two major points. Firstly, heat exchangers were excluded from mathematical modeling. Secondly (this follows from the first point), no calculations are available for the complete cycle of the working fluid, allowing to coordinate all heat and mass transfer processes in a rotor blade engine.

After analyzing the current state of the problem, we set out to conduct first-level mathematical modeling within the framework of the nodal analysis method.

To achieve this, we need to detect the structural elements performing specific functions in a rotor blade engine.

Consider the design of a four-bladed engine (Fig. 2). Diametrically opposite blades *7*, *9* and *8*, *10* are connected in pairs. As the shaft rotates, one pair of blades makes oscillatory motions relative to the other pair using a cam mechanism. This assembly is placed in a drum, making up four chambers with variable volume. The volume in each of the chambers varies following the same pattern, but it is shifted in phase in adjacent chambers by an angle $\pi/2$ during the cyclic process. Fig. 2 shows a configuration where chambers *1* and *3* have a minimum volume, and chambers *2* and *4* have a maximum volume (the chambers are numbered counterclockwise, in the rotation direction of the motor shaft). In addition to chambers *1*–*4*, there are chambers of hot and cold heat exchangers *5* and *6*. The volume of these two chambers does not depend on time. Partitioned this way, the rotor blade engine can be represented by six elements.

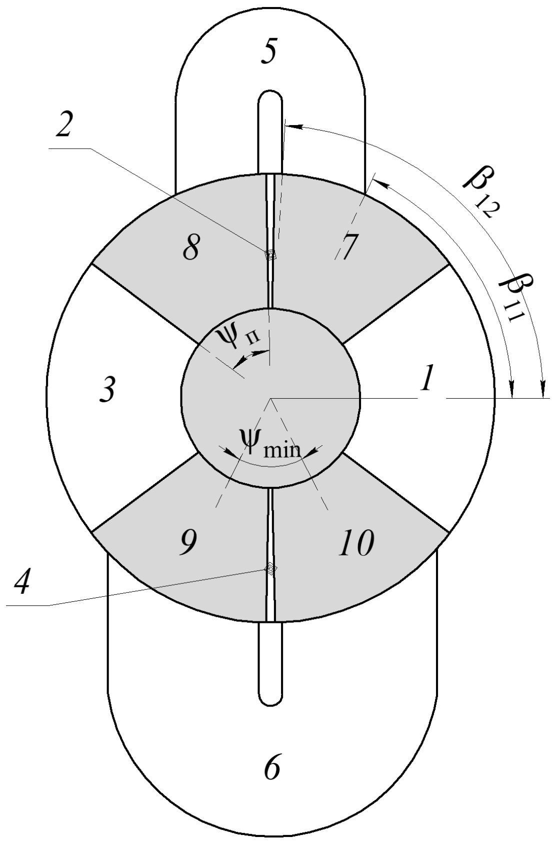

Fig. 2. Schematic of rotor blade engine: working chambers *1–4*; heater *5*; cooler *6*engine blades; *7–10*; Ψ_{II} is the blade size with respect to the angular coordinate; Ψ_{min} is the minimum distance between the midpoints of adjacent blades along the angular coordinate;β_{11}, β_{12} are the positions of the inlet valve to the heater in angular coordinates

Gas exchange between chambers *1–4* and heat exchangers occurs through switching valves. As the shaft rotates, the chamber is positioned with an inlet or outlet to the heat exchange chamber of the heater or cooler. The size of the area where the valve and the chamber overlap, as well as the volume of chambers *1–4* depend on the rotation angle and, ultimately, on time. Fig. 2 shows the position of the inlet valve to the heater in angular coordinates β_{11} and β_{12}.

Let us consider the parameters of the problem that determine the operating mode of a rotor blade engine. To formulate and solve the Cauchy problem, we should set the geometric characteristics, m^3:

$$V_1(t) - V_4(t),\ V_5,\ V_6;$$

angular frequency ω (rad/s); composition of the working fluid, which determines the thermodynamic properties and the adiabatic indexγ; mass M (kg) of the working fluid in all six chambers. In addition, we need the surface temperatures $T^w{}_{heat}$ and $T^w{}_{cool}$ (K) in the heater and cooler, as well as the volumetric heat transfer coefficient h (W/m^3).

Setting these parameters for the problem statement is sufficient to determine such important characteristics as the amount of added heat Q^{heat} and removed heat Q^{cool} (J), and, consequently, the amount of work performed by the engine on the shaft.

Two ordinary differential equations expressing the balance of mass and energy were used to describe the behavior of thermodynamic functions in each part of the system under consideration. The system of equations was closed by the gas equation of state, expressed as the Mendeleev–Clapeyron formula:

$$\begin{cases} \dfrac{d}{dt}\left(V_j\rho_j\right) = \sum\limits_{i=1}^{N} \dot{m}_{ji}, \\ \dfrac{d}{dt}\left(c_V V_j \rho_j T_j\right) = \sum\limits_{i=1}^{N} \dot{m}_{ji} c_P T_{ji} - p_j \dfrac{dV_j}{dt} + h\left(T_j^w - T_j\right)V_j, \\ p_j = R_{gas}\rho_j T_j, \end{cases} \tag{4}$$

where j is the number of the element in the engine; N is the number of elements; m_{ji} is the gas flow rate between the elements with subscripts j and i; p_j, ρ_j, T_j are the pressure, density and temperature; c_P, R_{gas} are the specific heat capacity at constant pressure and the gas constant, respectively.

The variations in mass and energy are described by unsteady terms, the transfer processes between the engine elements appear in the right-hand sides of system of equations (4). Formally, each element j can be associated with another element i, which is expressed by summation over all N objects. The specific communication of the elements at the given time instant is determined by the coupling matrix acting as a filter excluding unnecessary mass fluxes. Depending on the rotation angle of the shaft, the actual values of the overlap area S_{ji} are calculated simultaneously when chambers and valves of heat exchangers are switched. Mass transfer between the elements occurs with the choice of the flux direction. It is assumed that mass inflow into the given element j is described by a positive flow rate ($\dot{m}_{ji} > 0$), while the quantity T_{ji}, expressing the temperature of the gas incoming into the jth element, has the value $T_{ji} = T_i$. If $\dot{m}_{ji} < 0$, then $T_{ji} = T_j$. The second term in the second equation of the system takes into account the influence of the volume variation rate on the temperature variation, and the third term describes the heat transfer processes with elements of heat addition or extraction.

The direction and magnitude of mass transfer processes is determined by comparing the pressure level in the objects. The maximum and minimum pressures are selected by the formulas

$$p_{\max} = \max(p_j, p_i) \text{ and } p_{\min} = \min(p_j, p_i).$$

This technique allows to identify an element with a high pressure value and label all other functions in this element as maximum. The absolute value of the gas flow rate is determined by an expression that is based on isentropic formulas and takes the form

$$\begin{cases} \dot{m}_{ji} = \dot{m}^*_{ji} = \left(\dfrac{2}{\gamma+1}\right)^{\frac{(\gamma+1)}{2(\gamma-1)}} S_{ji}\rho_{\max}\sqrt{\gamma R_{gas} T_{\max}}, \; p_{\min} < \dfrac{2}{\gamma+1} p_{\max}, \\ \dot{m}_{ji} = \dot{m}^*_{ji}\left(\dfrac{2}{\gamma-1}\left(\dfrac{\gamma+1}{2}\right)^{\frac{\gamma+1}{\gamma-1}}\left(\dfrac{p_{\min}}{p_{\max}}\right)^{\frac{2}{\gamma}}\left(1-\left(\dfrac{p_{\min}}{p_{\max}}\right)^{\frac{\gamma-1}{\gamma}}\right)\right)^{1/2}, \; p_{\min} > \dfrac{2}{\gamma+1} p_{\max}. \end{cases} \tag{5}$$

The first formula in system (5) reflects the case of choked supersonic flow at a pressure ratio in the chambers above the critical value. The second formula describes the modes of subsonic gas flow.

System of equations (4), (5) was integrated numerically by the explicit two-layer Euler scheme. The calculation algorithm was validated with a series of solutions to the simplest problems. In particular, the time in which a constant-volume container is filled is estimated in [8], the problem on filling of a container is solved in [9]. The behavior of the gas temperature in a closed volume due to variation of this volume or of the heat transfer is described by simple analytical formulas.

The data for analysis of the engine performance were taken from [2]. The parameters required for the calculations are summarized in Table.

The kinematic displacements of the blades are determined by two angular velocities

$$\omega_{7,9} = \omega\left(1-\left(\frac{\pi}{2}-\Psi_{\min}\right)\sin(2\varphi)\right), \; \omega_{8,10} = \omega\left(1+\left(\frac{\pi}{2}-\Psi_{\min}\right)\sin(2\varphi)\right), \tag{6}$$

while the positions of their faces can be found from the solution of an ordinary differential equation

$$\frac{d\varphi_j}{dt} = \omega_j. \tag{7}$$

Knowing the angular coordinates of the positions of the chamber's front and rear, we can determine its size with respect to the angular coordinate as the difference of the corresponding angles. In addition, comparing the angular coordinates of the the chamber's front and rear with the angles of heat exchanger valvues allows to determine the overlap areas.

The estimated values of the functions in the heater and cooler taken from the indicator diagram were used as an initial approximation for thermodynamic functions. Calculations consisting of several hundred cycles should be performed to obtain the results for the steady-state mode.

The transient process of the engine warming up from the given initial state to steady operation is illustrated by the dependence of temperature and pressure in the heater and cooler (Fig. 3).

Table

Calculated parameters used for rotor blade engine and their values

Parameter, unit	Notation	Parameter value calculated by method	
		IA	NV
Temperature, K			
heater	T_h	603.15	605.183
cooler	T_c	300.15	276.217
Pressure, MPa			
heater	p_h	–	0.408
cooler	p_c	0.100	0.100
Mean temperature of working fluid, K			
heater	$<T_{gh}>$	$T_g = T_{he}$	603.15
cooler	$<T_{gc}>$	$T_g = T_{ce}$	300.15
Volume, m^3			
heater	V_h	$V_{he} >> V_{w.ch}$	1.0
cooler	V_c	$V_{ce} >> V_{w.ch}$	1.0
Angular valve size, degrees	β_{11}	–	64.00
	β_{12}	–	88.90
	β_{21}	–	91.10
	β_{22}	–	116.00
	β_{31}	–	216.95
	β_{32}	–	268.90
	β_{41}	–	271.10
	β_{42}	–	323.05
Angular blade size, degrees	Ψ_{II}	52.0	
Minimum distance between adjacent blades, degrees	Ψ_{min}	54.1	
Blade width, m	W_b	0.12	
Rotor radius, mm	R_2	50	
Radius of head gasket, mm	R_1	125	
Total volume of chambers, m^3	V_{ch}	$2 \cdot 10^{-3}$	
Angular velocity of rotor, rad/s	ω	10π	

Notations: IA, NV are indicator analysis and nodal values, respectively; T_g is the temperature of the working fluid (gas–air); T_{he}, V_{he} are the temperature and the volume of the hot heat exchanger, respectively; $V_{w.ch}$ is the volume of the working chamber.

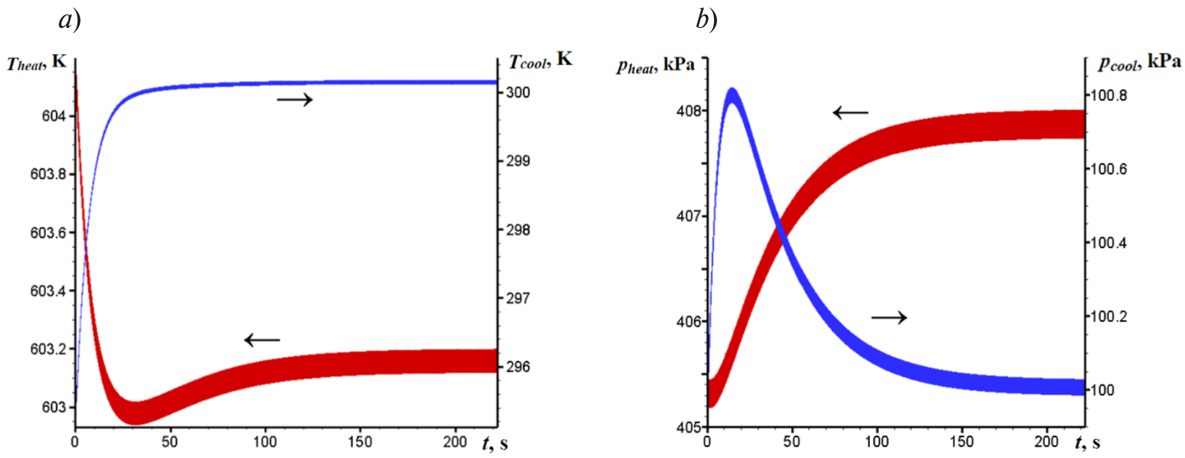

Fig. 3. Time dependences of temperature (*a*) and pressure (*b*)
in the cooler and heater (blue and red curves, respectively)

A characteristic of the steady-state operating mode is that the gas mass supplied to the heater during the cycle coincides with the gas mass supplied to the cooler. Fig. 4 shows the time dependences of pressure and temperature for two consecutive cycles at steady-state operation in chamber *1*.

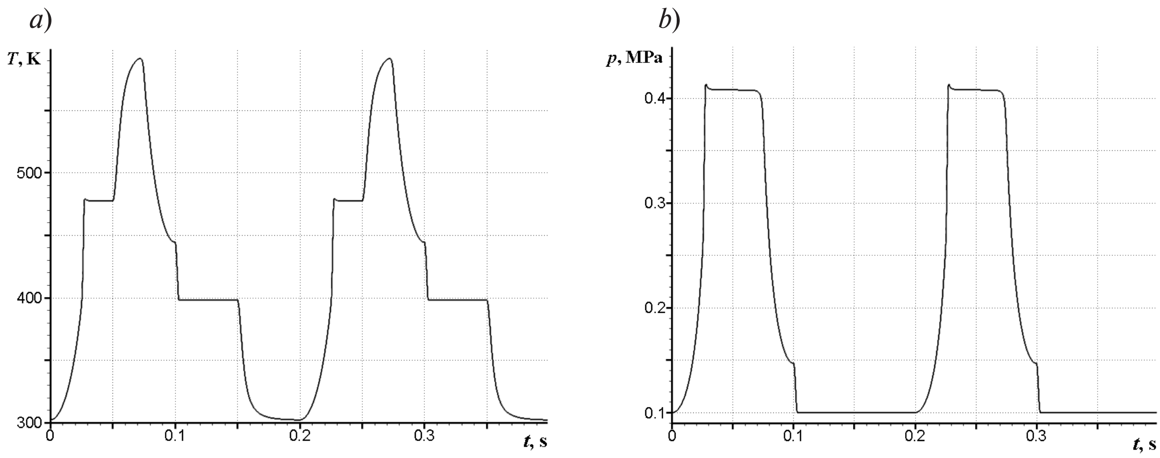

Fig. 4. Dependences of temperature (*a*) and pressure (*b*) on time in the first chamber

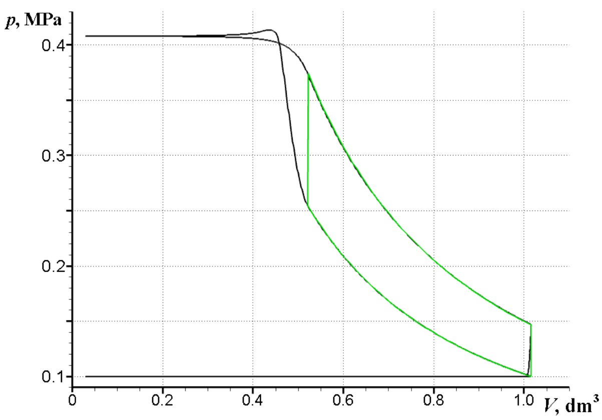

Fig. 5. Cyclic dependences of pressure on instantaneous volume value; the results were obtained by indicator dependences and by the nodal analysis method (green and black curves, respectively)

Fig. 5 allows to visually compare the simulation results for the operation of a rotor blade engine obtained by indicator analysis and by the nodal value method. The representation of the functions in the $p-V$ diagram confirms the hypothesis that the assumption of infinitely large heat exchangers is indeed equivalent to considering an isochoric process. The final dimensions of the heat exchangers produce «loops» in the $p - V$ diagram. In addition, this diagram shows the processes of gas exchange between the chamber and the heat exchanger, which are expressed both in the final rate of pressure variation and in the deviation of the pressure in the chamber from the pressure in the heat exchanger.

Conclusion

The nodal analysis method was applied in this paper to determine the performance characteristics of a rotor blade engine with external heat supply. Taking into account the processes in heat exchangers helped us gain an understanding of the behavior of thermodynamic functions in a cyclic process. Changing the state of the working fluid in a complete cycle allows to correctly determine the real values of heat fluxes in heat exchangers and, consequently, work output of the engine shaft.

The formulated mathematical model takes into account the time scales, including those associated with the finite rates of heat and mass transfer processes. We established that the indicator diagram satisfactorily describes the behavior of thermodynamic functions for case with the rotor blade engine shaft rotating at low rpm.

The nodal values method requires little resources and has clear advantages over the indicator analysis method, so calculations by the former method yield a good preliminary approximation for subsequent comprehensive mathematical modeling in a two- or three-dimensional statement of the problem. The approach used in the study to describe the characteristics and operation of the rotor blade engine has ample opportunities for development and modification.

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