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BASIC POSITIONS OF THE ENTHALPY-ENTROPY METHODOLOGY OF THERMODYNAMIC ANALYSIS OF GAS TURBINE POWER PLANTS

ОСНОВНЫЕ ПОЛОЖЕНИЯ ЭНТАЛЬПИЙНО-ЭНТРОПИЙНОЙ МЕТОДИКИ ТЕРМОДИНАМИЧЕСКОГО АНАЛИЗА ГАЗОТУРБИННЫХ ЭНЕРГОУСТАНОВОК

Summary. *The basic positions of the enthalpy-entropy methodology of thermodynamic modeling of processes in gas turbine units (GTUs) and combined power plants on basis GTUs are presented. The main requirements and conditions of this methodology are formulated, they allows the construction of a sequential (without iterations) algorithm for the computational diagnostics of the thermodynamic parameters of the GTU cycle, which includes the calculation blocks for the compressor, combustion chamber, turbine, and exhaust tube of the GTU. The obtained regression equations are presented. The use of these equations simplifies of the procedure for evaluating the thermodynamic parameters of the components at the nodal points of the cycle. The advantages of the proposed methodology in comparison with the traditional thermal-entropy methodology are indicated.*

Key words*: gas turbine units, thermodynamic modeling, regression equations.*

Аннотация. *Изложены основные положения энтальпийно-энтропийной методики термодинамического моделирования процессов в газотурбинных (ГТУ) и комбинированных на их основе энергоустановках Сформулированы основополагающие требования и условия этой методики, позволяющие построение последовательного (без итераций) алгоритма расчетной диагностики термодинамических параметров цикла ГТУ, включая блоки расчета компрессора, камеры сгорания, турбины и выхлопного патрубка ГТУ. Приведены полученные уравнения регрессии, с использованием которых достигается упрощение процедуры оценки термодинамических параметров компонент в узловых точках цикла. Указаны преимущества предлагаемой методики по сравнению с традиционной термическо-энтропийной методикой.*

Ключевые слова: *газотурбинные установки, термодинамическое моделирование, уравнения регрессии.*

Designations. h – enthalpy, kJ/kg; S – entropy, kJ/kg; $\pi_к$ – compression ratio in the compressor; $\theta = T_3/T_1$ – dimensionless temperature; T_1 – temperature at the compressor inlet, K; T_3 – temperature at the turbine inlet, K; GTU – gas turbine unit.

One of the important directions in the development of the world energy industry is the use of cogeneration technologies based on gas turbine engines. A number of studies have been devoted to solving the problem of increasing the energy efficiency of these technologies, for example [1-3]. The multivariance of the solution to the problem of increasing the energy efficiency and unit capacity of gas turbine power units, the need to obtain a reliable assessment of the competitiveness of their various options requires the creation of scientifically grounded methods of thermodynamic modeling of the processes occurring in such power units.

Application of the traditional method of approximate thermodynamic analysis of a gas turbine plant using T – S diagrams [3-7] in the case of production of hot water or steam turns out to be very difficult when considering power plants combined on the basis of gas turbine plants.

Under these conditions, it is advisable to use h–ΔS diagrams, which are universal, since they allow comparing the GTU cycle with processes combined with it in cogeneration schemes. In other words, the use of h–ΔS diagrams makes it possible to use a unified form of analysis of the energy characteristics of a gas turbine plant and power plants combined on its basis.

The proposed enthalpy-entropy method of thermodynamic modeling of a gas turbine plant and units combined on its basis is grounded on the such requirements and conditions:

- consideration of the working fluid as a one-component gas (air) in the compressor and a two-component gas (air + fuel) or three-component gas (air + fuel + steam) in the ducts of the combustion chamber, turbine and at the exhaust tube of the plant;

- taking into account pressure losses in the paths of the plant when assessing its technical and economic indicators and thermodynamic parameters at the nodal points of the cycle;
- construction of a sequential (without iterations) algorithm for the computational diagnostics of the thermodynamic parameters of the GTU cycle, including the blocks of calculation for the compressor, combustion chamber, turbine and GTU exhaust tube.
- ensuring the possibility of realizing thermal-entropy and enthalpy-entropy versions of thermodynamic analysis of gas turbine plants and the possibility of constructing cycles in [T-Δs] and [h-Δs] coordinates;
- simplification of the procedure for evaluating the thermodynamic parameters of the components at the nodal points of the cycle based on the construction and use of the corresponding regression equations.

This article presents the necessary regression equations, determined from the results of statistical processing of the data available in the literature, provided that the minimum dispersion $2\sigma \rightarrow \min$ is achieved. The value of the variance 2σ corresponded to the confidence level of 0.96 with the degree of correlation R close to unity. Table 1 shows the corresponding data for air in two ranges of variation of the defining parameters (zone I and II), which ensures the achievement of the minimum error in calculating the parameters.

Table 1

Regression equations for air according to S.L. Rivkin [8]

Zone	**Regression equations**	**Dispersion, ± 2σ**	**Range of variation of thermodynamic parameters**
I	$h = \exp[5{,}67+0{,}282\cdot\ln(\pi_к)]$	2,56	$1{,}49 \le \pi_к \le 125{,}6$
	$s = 5{,}9 + 3{,}15\cdot10^{-3}\cdot h - 1{,}12\cdot10^{-6}\cdot h^2$	0,015	h = 323-1130 *кДж/кг*
	$\theta = -0{,}039 + 3{,}69\cdot10^{-3}\cdot h - 3{,}2\cdot10^{-7}\cdot h^2$	$1{,}2\cdot10^{-3}$	h = 323-1130 *кДж/кг*

	$h = 16,36 + 263,8 \cdot \theta + 9,58 \cdot \theta^2$	0,56	$\theta = 1,12 - 3,72$
	$h = \exp[-0,887 + 0,984 \cdot s]$	2,3	$s = 6.78 - 8,052$, *кДж / кг.К*
II	$h = 5,5 + 276 \cdot \theta + 7,1 \cdot \theta^2$	2,63	$\theta = 3,72 - 6,15$
	$s = 6,78 + 1,36 \cdot 10^{-3} \cdot h - 2,1 \cdot 10^{-7} \cdot h^2$	0,001	$h = 1130 - 1970$, *кДж / кг*
	$\theta = 0,0715 + 3,42 \cdot 10^{-3} \cdot h - 1,7 \cdot 10^{-7} \cdot h^2$	0,0083	$h = 1130 - 1970$, *кДж / кг*
	$h = \exp[-9,08 + 7,724 \cdot \ln(s)]$	0,73	$s = 8,052 - 8,653$, *кДж / кг.К*

The corresponding regression equations for the design fuel (methane) and water vapor are shown in Tables 2 and 3.

Within the framework of the proposed technique, when performing the procedure of thermodynamic modeling of the characteristics of the installation, in addition to the regression equations (Tables 1-3), equations are also used to determine the entropy at various nodal points of the cycles, as well as the equations, heat and energy balances.

Table 2

Regression equations for the design fuel (methane) according to the tabular data of V.P. Glushko [9]

Zone	**Regression equations**	**Dispersion, ± 2σ**	**Range of variation of thermodynamic parameters**
I	$s = \exp[1,29 + 0,18 \cdot \ln(h)]$	0,019	$h = 603 - 3006$, *кДж / кг*
	$\theta = 0,123 + 1,57 \cdot 10^{-3} \cdot h - 1,5 \cdot 10^{-7} \cdot h^2$	0,018	$h = 603 - 3006$, *кДж / кг*
	$h = 114,2 + 347,7 \cdot \theta + 140,3 \cdot \theta^2$	2,4	$\theta = 1,0 - 3,47$
	$h = \exp[-7,18 + 5,55 \cdot \ln(s)]$	16,6	$s = 11,54 - 15,436$ *кДж/кг·К*
II	$h = -844,9 + 890,5 \cdot \theta + 62,9 \cdot \theta^2$	4,35	$\theta = 3,47 - 6,246$

	$s = 12,21 + 1,2 \cdot 10^{-3} \cdot h - 5,0 \cdot 10^{-8} \cdot h^2$	0,009	$h = 3006 - 7170$, *кДж / кг*
	$\theta = 1,079 + 8,53 \cdot 10^{-4} \cdot h - 2,0 \cdot 10^{-8} \cdot h^2$	0,0043	$h = 3006 - 7170$, *кДж / кг*
	$h = \exp[13,3 - 81,83 / s]$	16,0	*s* = 15,436 – 18,46 *кДж/кг·К*

The proposed enthalpy-entropy method of thermodynamic modeling of processes in gas turbine and combined thermal power plants on their basis, in comparison with the method of approximate calculation traditionally used for gas turbine plants, based on the use of $T - S$ diagrams, is characterized by the following advantage:

Table 3

Regression equations for water vapor according to S.L. Rivkin [10]

Zone	**Regression equations**	**Dispersion, ± 2σ**	**Range of variation of thermodynamic parameters**
I	$s = 8,976 + 3,08 \cdot 10^{-3} \cdot h - 5,4 \cdot 10^{-7} \cdot h^2$	0,005	$h = 532 - 2466$, *кДж / кг*
	$\theta = -0,0246 + 1,99 \cdot 10^{-3} \cdot h - 1,2 \cdot 10^{-7} \cdot h^2$	0,0017	$h = 532 - 2466$, *кДж / кг*
	$h = 34,4 + 470 \cdot \theta + 27,19 \cdot \theta^2$	1,31	$\theta = 1,0 - 4,17$
	$h = \exp[0,79 + 0,527 \cdot s]$	12,6	$s = 10,41 - 13,34$, *кДж / кг* × *К*
II	$h = -73,24 + 515 \cdot \theta + 22,74 \cdot \theta^2$	0,45	$\theta = 3,125 - 5,2$
	$s = 10,77 + 1,268 \cdot 10^{-3} \cdot h - 9 \cdot 10^{-8} \cdot h^2$	0,002	$h = 2400 - 3957$, *кДж / кг*
	$\theta = 0,336 + 1,69 \cdot 10^{-3} \cdot h - 6,0 \cdot 10^{-8} \cdot h^2$	0,001	$h = 2400 - 3957$, *кДж / кг*
	$h = \exp[-8,94 + 6,46 \cdot \ln(s)]$	0,85	*s* = 15,288 – 14,35 *кДж/кг·К*

1. Versatility in terms of the possibility of using a unified form of analysis of the energy characteristics of gas turbines and combined power plants on their basis.

2. Greater accuracy in determining the main energy characteristics of installations (specific work of its elements, efficiency of installations, etc.) due to the use of a number of simplifying prerequisites in the approximate technique, such as the constancy of the heat capacity of working bodies, etc.

3. Higher efficiency of the computational algorithm and the corresponding software product due to:

a) the absence of an interactive procedure for determining the thermodynamic parameters, typical of the traditional method;

b) simplifying the determination of the thermodynamic characteristics of the components of the working fluid at the nodal points of the cycle based on the construction of the corresponding regression equations for the thermodynamic properties of these components.

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