RIGA TECHNICAL UNIVERSITY

INSTITUTE OF HEAT, GAS AND WATER TECHNOLOGY

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MATHEMATICAL MODELS FOR DESIGN OF GAS TURBINE HEAT EXCHANGERS

Summary
of the Thesis for Scientific Degree
of the Doctor of Engineering Sciences

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RTU Publishing House
Riga -2004

GENERAL DESCRIPTION OF THE THESIS

Importance of the Thesis

The operation of gas turbine plants involves frequent changes to operating conditions within the plant and its components. These changes are caused by variations in electrical and thermal loads as well as seasonal and daily changes of outside air parameters. This establishes the need to design thermal equipment of gas turbine plants for operation at variable conditions, considering a wide range of changes in initial conditions.

The thermal design of heat exchangers involves the use of heat transfer equations, which contains the sought quantities (heating media temperature) in implicit and transcendental form. Therefore, thermal design of a heat exchanger must be carried out using the method of successive approximations. Time and man-hours required for calculation drastically increase in the event that variable operating conditions are to be considered. In this connection it would be vital to develop the calculation method for variable operating conditions of gas turbine plant heat exchangers, which would allow necessary calculations to be made promptly and with sufficient accuracy during both the design and operation stages within the turbine plant.

The purpose of the thesis is to develop analytical mathematical models of the thermal state of gas turbine plant heat exchangers and implement such models in the form of corresponding software systems, which would enable sufficiently effective, quick and non-laborious multivariate calculations for design optimization during the design and operation of turbine gas plants.

Thesis Statements to be Defended:

1. PC-based mathematical model for thermal state of heat exchanger.
2. PC-based analytical solution for thermal design of heat exchangers under variable operating conditions within gas turbine plant.
3. Method for monitoring the state of the heat exchange surface of heat exchangers under operating conditions.
4. Calculation method for variable operating conditions of heat exchangers, which accounts for forced change of the heat exchange surface area under operating conditions.

5. Calculation method for variable operating conditions of heat exchangers, which accounts for forced change of the heating media flow pattern under operating conditions.

Scientific Innovation

1. Analytical solutions for thermal design of heat exchangers under nominal and variable operating conditions of gas turbine plants.
2. Set of methods which can be used under operating conditions and would enable calculations to be made to monitor the state of heat exchange surface of heat exchangers, taking into consideration the forced changes in the heat exchange surface area (caused, for example, by plugging emergency drains) and account of forced changes in the heating media flow pattern.

Supportabilitv and Reliability of Results are confirmed by quite a reasonable degree of coincidence (£2.0%) between the results obtained by using the standard calculation method, from use of the suggested methods and results.

Practical Importance of the Thesis is evident in the fact that developed mathematical models implemented by the author in software systems have practical value and have been implemented by SIEMENS for the design of advanced gas turbine plants.

Materials of this thesis are used in the academic process for the course entitled *Heat Exchangers on Gas Turbine Plants.*

Testing

The main results of the thesis were reported and discussed at scientific and technical conferences in Munich (1999), Erlangen (2000), Moscow (2001), St. Petersburg (2002), and Riga (2003 and 2004).

Publications

The main results of the thesis are published in scientific articles and study guides. The author published the total of 9 works on the topic of the thesis, including 1 monograph and 2 study guides.

Personal Contribution of the Author

This document summarizes the results of research, which was carried out by the author both independently and together with scientific teams headed by the author. The author's contribution included; identifying the problem and setting and summarizing the objectives of the research; developing mathematical models of the thermal state of heat exchangers within gas turbine plants implemented in corresponding software systems; developing the set of calculation methods which allow the control of the state of the heat exchange surface within the heat exchangers, and accounting for the forced change of the heat exchange surface area and heating media flow pattern.

Structure of the Thesis

The thesis consists of introduction, five chapters, and conclusion. It contains 108 pages of text, 22 illustrations, 16 pages of appendices (tables), and a bibliography including 106 entries.

CONTENTS OF THE THESIS

Introduction shows the urgency of the topic and formulates the main objective of the thesis.

Chapter One describes the role of heat exchangers considering the increase of technical and economic performance of gas turbine plants operating in stand-alone mode and as part of a combined cycle plant.

The literature on research, calculation, design and operation of gas turbine plant heat exchangers was reviewed and analyzed.

Two primarily used methods for the thermal design of heat exchangers were analyzed. One of the methods is based on the average temperature head At^, while the other is based on the correlation between heat efficiency of the heat exchanger i_r and heat transfer parameter N. References were made to the works of such scientists as S.S. Berman, L.G. Gelfenbein, M.D. Gryaznov, Y.M. Dedusenko, E.K. Koshkin, V.M. Kalinin, A.L. London, E.Y. Sokolov, V.M. Keis, Helmut Hausen, V.V. Uvarov, et al.

Existing calculation methods for variable operating conditions of gas turbine plant heat exchangers are laborious and time-consuming.

Our analysis allowed not only a critical review of the existing methods for the thermal design of heat exchangers, which were published in literature, but also a suggestion with regards to a new calculation method for variable operating conditions within gas turbine plants. This new method is effective during both the design and operation stages.

Chapter Two reviews the mathematical model of the thermal state of heat exchangers for the common pattern of heating media flow presented in Fig. 1.

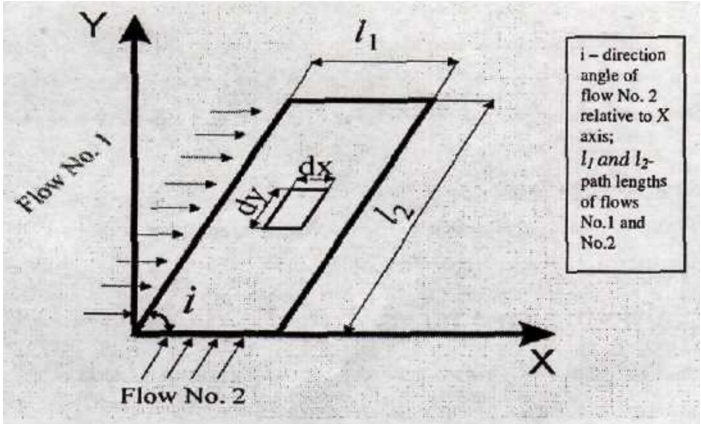

Fig. 1. General Pattern of Heating Media Flow in Heat Exchanger

Parallel flow is used when $i = 0$, when $i = \frac{\pi}{2}$ cross flow is used, and when $i = \pi$ counter flow is used, $\frac{\pi}{2} > i > 0$ and $\frac{\pi}{2} < i < \pi$ are cases when the heating medium flow is angled. The solution of the above task is limited to a review of a simplified case based on the assumption that there is no longitudinal thermal conductivity in heating media flows and matrix, the flow of heating media in any point of flow is one-dimensional, thermal resistance of the matrix in the direction perpendicular to the flow of heating media is negligible, and that thermal capacity of heating media and matrix as well as density, heat transfer coefficients, velocities and flows do not depend on temperature, time and coordinates.

After transformations the system of differential energy equations for heating media and matrix may be presented as follows:

$$-\left(\frac{\partial t_1}{\partial \varphi} + \frac{\partial t_1}{\partial \xi}\right) = N_1(t_1 - t_м) \; ; \qquad (1)$$

$$\bar{\tau}\frac{\partial t_2}{\partial \varphi} + \frac{l_2}{l_1}\frac{\partial t_2}{\partial \xi}\cos i + \frac{\partial t_2}{\partial \psi}\sin i = N_2(t_м - t_2) \; ; \qquad (2)$$

$$\overline{C}_м \frac{dt_м}{d\varphi} = N_1(t_1 - t_м) - \frac{W_2}{W_1} N_2(t_м - t_2) \; , \qquad (3)$$

$\zeta = x / l_1$, $\Psi = y / l_2$, $\varphi > = \tau / \tau_1$ are dimensionless quantities of coordinates and time; t is temperature of the medium, °C; $\tau = \tau / \tau_2$; $W = GC_p$; $N_1 = \alpha_1 F_1 / W1$; $N_2 = \alpha_2 F_2 / W_2$; $C_M = m_M C_M / (m_1 C_{p1})$; G is the flow of heating medium, kg/sec; C_p is specific thermal capacity at constant pressure; F is heat exchange surface, m^2; a is heat transfer coefficient, $kJ/(m^2h\text{-}K)$. Indices «1», «2» and «M» indicate primary heating medium, secondary heating medium and heat exchanger matrix respectively.

Equations (1), (2), (3) represent a closed system because their number equals the number of unknown quantities t_1, t_2 and t_M. This system of equations is solved for the cases when $F_1 = F_2$ and $i = \pi ; \pi\pi ; 0$ under stationary conditions.

On the basis of this mathematical model the following expressions were derived to determinethe area of the heat exchange surface.

For all heating media flow patterns when $W_M / W_6 = 0$

$$F = A \ln \frac{1}{1 - \eta_\tau} \qquad (4)$$

where $A = W_M / K = const$; η_τ is thermal efficiency of heat exchanger; K is heat transfer coefficient, $kJ/m^2h\text{-}K$; and indices «M» and «6» indicate the lowest and the highest values of W respectively.

For all heating media flow patterns when $W_M / W_6 = 1$, the following expressions can be derived:

for parallel flow:

$$F = 0{,}5 A \ln \frac{1}{1 - 2\eta_\tau}, \qquad (5)$$

for counter flow:

$$F = A \frac{\eta_\tau}{1 - \eta_\tau}, \qquad (6)$$

Expression (6) was first derived in 1935 by V.V. Uvarov for counter flow air heater $\eta_{\tau r}$ was used to represent the degree of regeneration.

for cross flow with mixing one of the heating media:

$$F = A\ln(1 - \ln\frac{1}{1-\eta_\tau})^{-1} \quad , \tag{7}$$

for cross flow with mixing both of the heating media:

$$F = A\ln\frac{1}{1-2\eta_\tau}, \tag{8}$$

Results of calculation of the heat exchange surface using formulae (4) - (8) when $W_M/W_6=Q$ and $W_M fW_s = 1$ for various patterns of heating media flow and heat efficiency values

η_τ are summarized in Table 1 and illustrated in Fig. 2.

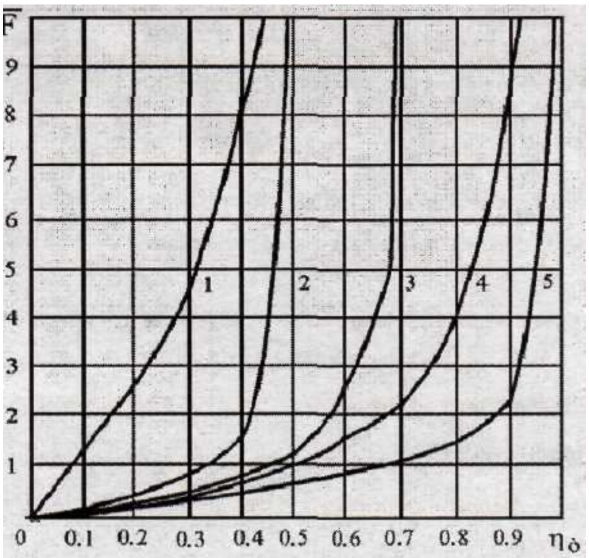

Fig. 2. Curve of $\overline{F}$ Versus η_T in Case of Simple Heating Media Flow Patterns

1 -parallel flow when $(W_M/W_6)= 1$; 2 - cross flow with mixing of both heating media when $(W_M/W_6)= 1$; 3-cross flow with mixing of one heating medium; 4 -counter flow when $(W_M/W_6)= 1$; 5 - for all patterns when $(W_M/W_6)= 0$

We developed the method for determining the heat transfer coefficient, taking into consideration the heat exchange surface fouling.

We also developed the method, which can be used during the design stage to determine the period between the cleaning of the heat exchangers under operating conditions.

Table 1

Results of Calculation of the Heat Exchange Surface for Various Heating Media Flow Patterns Calculated Using Formulae (4) - (8)

η_T	Heat Exchange Surface F/A at $A = W_{м}/K = const$				
	$W_{м}/W_{б}=0$	$W_{м}/W_{б}=1$			
	for all patterns	parallel flow	counter flow	cross flow: with mixing one flow	cross flow: with mixing both flows
0	0	0	0	0	0
0.1	0.104	0.111	0.111	0.122	0.223
0.2	0.223	0.256	0.250	0.253	0.512
0.3	0.357	0.471	0.429	0.441	0.942
0.4	0.513	0.805	0.668	0.716	1.610
0.5	0.692	∞	1.000	1.176	∞
0.6	0.914	-	1.500	2.450	-
0.7	1.205	-	2.333	-	-
0.8	1.609	-	4.000	-	-
0.9	2.300	-	9.000	-	-
1.0	∞	-	∞	-	-

This mathematical model was implemented on PC in Windows 98 environment within TOANOM software system.

Chapter Three presents our method for determining dimensionless curves for gas turbine plant heat exchangers implemented on a PC in a Windows 98 environment. The method is based on the system of differential equations of heat transfer and heat balance, which can be reduced to the following differential equation of energy:

$$dQ = -G_r C_{pr} dt_r = -G_x C_{px} dt_x = K(t_r - t_x) dF, \text{ W}, \qquad (9)$$

where dQ is thermal flow that passes through the element of heat exchange surface dF of the heat exchanger; t, G, C_p, K stand for temperature, mass flow, specific isobaric thermal capacity of the heating medium, heat transfer coefficient of the heat exchanger respectively; indices «r» and «x» indicate hot (primary) heating medium and cold (secondary) heating medium, while indices «1» and «2» represent heating media parameters at the inlet and outlet of the heat exchanger respectively.

The following expressions were derived for the counter flow heat exchanger, the design model of which is shown in Fig. 3.

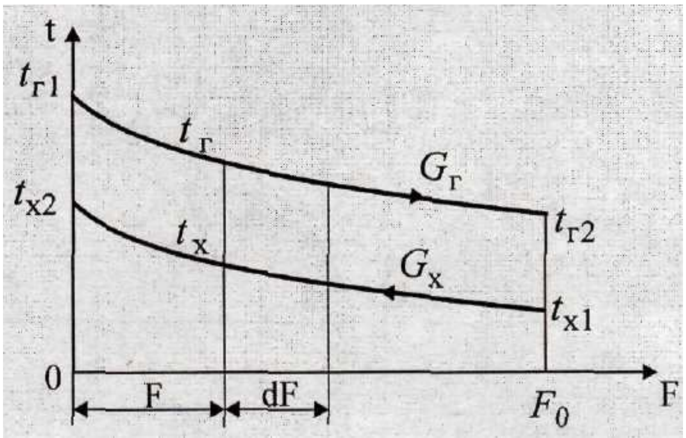

Fig. 3. Design Model of a Counter Flow Heat Exchanger on Gas Turbine Plant

F and F_o - current and full heat exchange surfaces respectively; G_r and G_x, - mass flows of hoit (primary) and cold (secondary) heating media; t_{r1} and t_{x1}- temperatures of hot and cold heating media at the inlet of the heat exchanger; t_{r2} and t_{x1} - temperatures of hot and cold heating media at the outlet of the heat exchanger; t_r and t_x -current temperatures of hot and cold heating media in the heat exchanger.

Expression of relative ultimate thermal power:

$$\overline{Q} = \frac{e^{kkF} - 1}{e^{mkF} - \overline{W}} \quad ; \tag{10}$$

Expression of final temperature of heating media:

$$t_{r2} = t_{r1} - \overline{Q}.(t_{r1} - t_{x1}), °C \tag{11}$$

$$t_{x2} = t_{x1} - \overline{Q}.\overline{W}(t_{r1} - t_{x1}), °C \tag{12}$$

where F is full heat exchange surface of the heat exchanger, m^2 ;

$$m = \frac{1}{G_r C_{pr}} - \frac{1}{G_x C_{px}}; \frac{h \cdot K}{kJ};$$

$\overline{W}$ = $G_r C_{pr} / (G_x\ C_{px})$ is relative water equivalent of hot and cold heating media. Full heat exchange surface is determined using the following formula:

$$F = \ln[\frac{(t_{r1} - t_{x1}) - (t_{r1-} t_{r2})\overline{W}}{t_{r2} - t_{x1}}]/(mk),\ \text{м}^2 \tag{13}$$

The design of the heat exchanger is selected on the basis of the calculation of nominal operating conditions. Therefore, the values characterizing the nominal operating conditions of the exchanger are known. When the heat exchanger performance curve is constructed for other non-nominal operating conditions, the nominal values are recalculated. The values representing nominal operating conditions of the exchanger are indexed with «G». Besides, it is assumed that C_{pr} = *const* and C_{px} = *const*.

Then

$$\overline{W} = \overline{W}_0 (\overline{G_r} / \overline{G}_x) \quad ; \tag{1 4}$$

$$m = \frac{1}{W_{r0}\overline{\overline{G_r}}} - \frac{1}{W_{x0}\overline{\overline{G_x}}} \tag{15}$$

where $\overline{G}_r = G_r / G_{r0}; \overline{G}_x = G_x / G_{x0}; \overline{W}_0 = \frac{G_{r0} C_{pr}}{G_{x0} C_{pr}}; W_{r0} = G_{r0} C_{pr};$

$$W_{x0} = G_{x0} C_{px}$$

Relative values of heat transfer coefficients Ct_T and Ct_x on heating media side are determined by the following expressions:

$$\overline{\alpha}_r = f_r(t)\left(\overline{G_r}\right)^{nr};$$

$$\overline{\alpha}_x = f_x(t)\left(\overline{G_x}\right)^{nx}$$

where $f_r(t) = \left(\overline{\mu}\right)^n \left(\overline{\Pr_r}\right)^m \overline{\lambda_r}$ and $f_x(t) = \left(\overline{\mu_x}\right)^n \left(\overline{Pr_x}\right)^m \overline{\lambda_x}$ are temperature correction functions for hot and cold heating media respectively; $\overline{\mu}$, $\overline{\Pr}$, $\overline{\lambda}$ are relative values of dynamic viscosity, Prandtl number and heat transfer coefficient of the heating medium respectively.

The following expression for heat transfer coefficient was derived;

$$K = \frac{B}{\dfrac{1}{\alpha_{ro} f_r(t)\left(\overline{G}_r\right)^{n_r}} + \dfrac{1}{\alpha_{xo} f_x(t)\left(\overline{G}_x\right)^{n_x}} + R}, \qquad \frac{kJ}{m^2 \cdot h \cdot K}, \tag{18}$$

where B is the factor of heat exchange area fouling during operation; and R is thermal resistance of heat conductivity of heat exchange surface.

Thus, dimensionless performance curve of the heat exchanger at coordinates $\overline{Q}$ - $\overline{W}$ may be constructed using equations (10), (14), (15) and (18). Type of functions $f_r(t)$ and $f_x(t)$ are determined when parameters of specific heat exchangers are calculated.

For the design of parallel flow or cross flow heat exchangers we developed methodical recommendations based on utilizing dimensionless curves constructed for counter flow heat exchangers.

This method was used to construct dimensionless performance curves (See Fig. 5 and 6) for two-section intermediate air cooler within a gas turbine plant (See Fig. 4 for the general layout of the cooler).

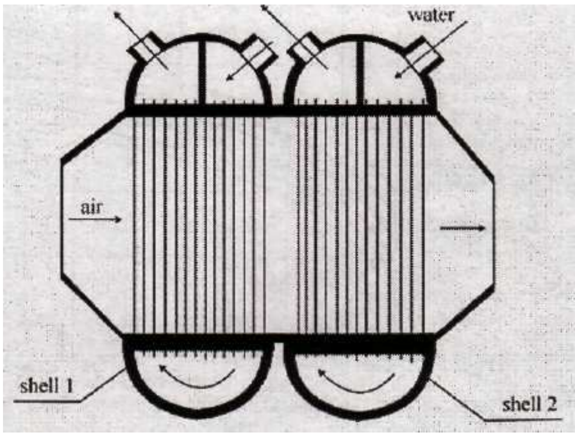

Fig. 4. General Layout of Two-Section Intermediate Air Cooler of Gas Turbine Plant.

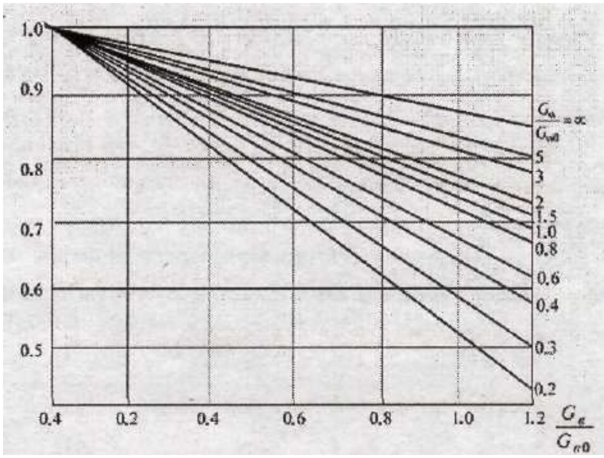

Fig. 5. Curves of Two-Section Intermediate Air Cooler of Gas Turbine Plant

at Coordinate $\overline{Q} - \frac{G_\theta}{G_{\theta 0}}$ when G_{b0}=*225,7kg/s,* G_{w0} =750t/h.

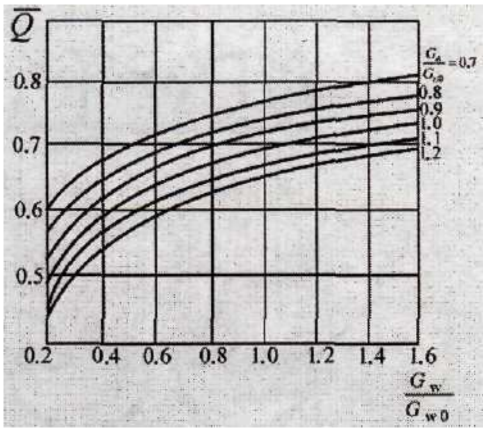

Fig. 6. Curves of Two-Section Intermediate Air Cooler of Gas Turbine Plant at Coordinate $\overline{Q}$ - $\frac{G_w}{G_{w0}}$ when G_{b0}= *225,7 kg/s*, G_{w0}=750 t/h

Chapter Four contains methodological data designed to facilitate the completion of mportant checking tasks during the design and operation stages within gas turbine plant heat xchangers.

It describes the method for checking the design of a heat exchanger under variable iperating conditions using dimensioniess curves. The method is reviewed using a 2-section air ooler as an example.

It also proposes the method for diagnosing the state of the heat exchange surface under iperating conditions, based on the dependence of the state of the heat exchange surface on the hange of minimum relative difference of heating media temperature values:

for separate section

$$\frac{\Delta t_{\min}}{t_{r1} - t_{x1}} = 1 - \overline{Q} \tag{19}$$

for two-section heat exchanger

$$\frac{\Delta t_{\min}}{t_{r1} - t_{x1}} = 1 - \overline{Q_1} - \overline{Q_2} - \overline{Q_1}\,\overline{Q_2} \tag{20}$$

The calculations for equations (20) and (21) can be done relatively quickly using dimensionless curves. The water supply conditions at the station (e.g. limited capacity of cooling towers) may require lower water flow. In this case water supply sections of the air cooler are connected not in parallel but in series. Therefore we developed a method which uses dimensionless curves to account for changes in water flow direction in the air cooler at the time of calculation of its variable operating conditions.

Heating media temperature at the outlet of the heat exchanger is determined using the following formulae:

$$t_{r2} = \frac{\left(1-\overline{Q}\right)^2 t_{r1} + \overline{Q}\left[2-\overline{Q}\left(1+\overline{W}\right)\right] t_{x1}}{1-\overline{Q}^2\overline{W}}, \qquad (21)$$

$$t_{x2} = \frac{\overline{Q}\,\overline{W}\left[2-\overline{Q}\left(1+\overline{W}\right)\right] t_{r1} + \left(1+\overline{Q}\,\overline{W}\right) t_{x1}}{1-\overline{Q}^2\overline{W}} \qquad (22)$$

Equations (21) and (22) allow for easy and quick calculation of variable operating conditions of the cooler, using a dimensionless curve when the water flow direction changes in the cooler.

<u>Chapter Five</u> reviews the limits for using dimensionless curves of heat exchangers and analyzes reliability of the results of analytical solutions.

The method of determining the limits for using dimensionless curves of heat exchangers was developed on the basis of confidence interval of criterion heat transfer equations. These limits are determined by the interval of changing relative flow of the heating medium.

$$\left(\overline{G}\right)_{min} = \left(\overline{\mathrm{Re}}\right)_{min} / \left(\overline{\mu}\right)_{min}$$

$$\left(\overline{G}\right)_{max} = \left(\overline{\mathrm{Re}}\right)_{max} / \left(\overline{\mu}\right)_{max}$$

For example, the following flow change intervals were established for both sections of the two-section air cooler of a gas turbine plant with developed turbulence of heating media flow in both sections (See Fig. 7).

for air

$0.09 \le \overline{G_\theta} \le 5$;

for water

$0.4 \le \overline{G_w} \le 20.$

The actual interval of turbulent water flow will be greater. When water flow is low, average water temperature will be higher than the design temperature as a result of increased water heating. Thus, the turbulent water flow interval will expand to $0.3 \le \overline{G_w} \le 20$. The $\overline{G_w} = 0.2$ curve reflects operation of the section in cogeneration mode, i.e. when average hot water temperature is 6O...7O°C. In this case: $\overline{G_w} = 0{,}15 < 0{,}2$.

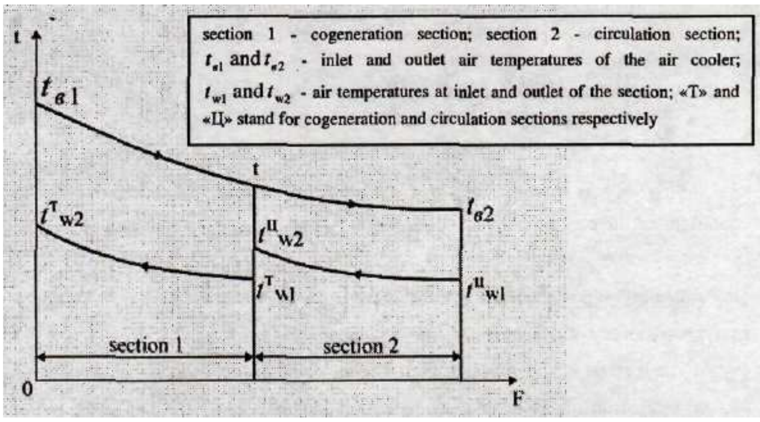

Fig. 7. Temperature Curves of Air Cooler. Sections in Series (air) and in Parallel (water)

The results obtained allow us to conclude that the turbulent flow of both heating media is present in the entire area of the family of dimensionless curves of heat exchangers (See Fig. 5 and . Fig. 6).

The reliability of calculation results obtained is determined by comparing them with the standard method results. The air temperature at the outlet of the section (or shell) of the cooler is determined on the basis of dimensionless curves using equation (23). After differentiation of this equation, several transformations and the acquisition of Finite differences, we will have

$$\frac{\Delta t_{r2}}{t_{r1} - t_{x1}} = -\frac{mFe^{mFk}(1-\overline{W})}{(e^{mFk} - \overline{W})^2}\Delta K \quad (23)$$

The analysis of the structure of equation (23) in terms of gas turbine plant air cooler shows that air temperature deviation t_{r2} is determined by the difference of the initial temperatures of air and water $(t_{r1} - t_{x1})$and $\overline{Q}$. $(t_{r1}$-$t_{xl})$ is a given value and therefore does not contain any calculation errors. On the contrary, deviation of $\overline{Q}$ is dependent on the calculation error of the values included.

Equation (23) is used as a basis for establishing correlation between Δt_{r2}^{τ} for cogeneration shell and Δt_{r2}^{u} for circulation shell.

$$\frac{\Delta t_{r2}^{\tau}}{\Delta t_{r2}^{u}} = \frac{e^{mkF} - \overline{W}}{2(1-\overline{W})} = \frac{1}{2(1-\overline{Q})} \quad (24)$$

From equation (24) we can derive that:

when $\overline{Q} \to 1{,}0, \frac{\Delta t_{r2}^{\tau}}{\Delta t_{r2}^{u}} \to \infty$ hence, $\Delta t_{r2}^{u} \to 0$ i.e. if the degree of cooling $\overline{Q}$ is high, the error of determining final air temperature becomes infinitesimal;

when $\overline{Q} = 0{,}5, \frac{\Delta t_{r2}^{\tau}}{\Delta t_{r2}^{u}} = 1{,}0$ i .e the errors after both sections are equal;

when $\overline{Q} = 0, \frac{\Delta t_{r2}^{\tau}}{\Delta t_{r2}^{u}} = \frac{1}{2}$ i.e. calculation error of air temperature behind the cooler is two times greater than the error of air temperature behind the cogeneration section.

Thus, deviation of the thermal power of cogeneration section in 0.5 ≤ $\overline{Q}$ ≤ 1.0 interval is compensated by a corresponding change of thermal power in the next circulation section, so the final air temperature fluctuation behind the cooler is cancelled out.

This does not happen when $\overline{Q} \leq$ 0.5 Residual imbalance during thermal design based on standard method is 1% of nominal thermal power, i.e. $\frac{\Delta q}{q} = -\frac{\Delta t_{\Gamma 2}}{t_{\Gamma 1} - t_{\Gamma 2}} = 0{,}01.$ i which means that absolute error in determining outlet air temperature depends not only on the accuracy of the determination of related values, but also on the extent of the decrease in air temperature. When the latter is decreased by $t_{r1} - t_{r2} = 150$ °*C*, absolute error of the accurate correction method will be $\Delta t_{r2} = 1.5°C$. t_{r2} results in the greatest absolute error, which is equal to 1-1.5 °*C*. This error is substantially lower for final water temperature.

MAIN RESULTS OF THE THESIS

The purpose of the thesis is to resolve important scientific and technical problems of developing analytical mathematical models of the thermal state of gas turbine plant heat exchangers and implementing such models in the form of corresponding software systems, which would enable the performance of sufficiently effective, quick and non-laborious multivariate calculations for design optimization during the design and operation stages within gas turbine plants.

The following scientific and practical results were developed in this thesis.

1. We developed a software system based analytical solution for the thermal design of gas turbine plant heat exchangers for the calculation of variable operating conditions.
2. We developed the calculation method for monitoring the change of the state of heat exchange surface of heat exchangers under operating conditions.
3. We developed the method accounting for forced changing of the heat exchange surface area of heat exchangers under operating conditions (caused by the need to plug theemergency drains) during the calculation of variable operating conditions.
4. We developed the method accounting for forced changing of the heating media flow pattern under operating conditions during the calculation of variable operating conditions of gas turbine plant heat exchangers.
5. We developed the method for determining limits of application of dimensionless curves of gas turbine plant heat exchangers.

REFERENCES

1. Куландин А.А., Толмачев В.В., Гольдберг Л. Комбинированные парогазовые установки. - СПб.: Изд-во Санкт - Петербургского института машиностроения, 2003. -106с.
2. Суханов В.А., Гольдберг Л., Безухов А.П., Бодров А.И. Теоретические основы расчета и проектирования теплообменных аппаратов газотурбинных установок. Учебное пособие. - СПб.: Изд-во Санкт - Петербургского института машиностроения, 2003. - 177с.
3. Куландин А.А., Толмачев В.В., Гольдберг Л., Бодров А.И. Термодинамический анализ циклов газотурбинных установок. Учебное пособие. - СПб.: Изд-во Санкт-Петербургского института машиностроения, 2003. - с.
4. Бодров А.И., Суханов В.А., Гольдберг Л. Основные уравнения для расчёта безразмерных характеристик и переменных режимов работы теплообменных аппаратов ГТУ при противоточном течении теплоносителей. //Современное турбостроение: Сб. научн. тр. - Вып.5. - СПб.: МАН ВШ. СПб отд-ние. СПб институт машиностроения (ЛМЗ-ВТУЗ).-2003. -с. 82-92.
5. Бодров А.И. Суханов В. А., Гольдберг Л. К вопросу о расчёте безразмерных характеристик и переменных режимов теплообменных аппаратов ГТУ при прямоточном и перекрестноточных схемах течения теплоносителей. //Современное турбостроение: Сб. научн. тр. - Вып.5.- СПб.: МАН ВШ. СПб отд-ние. СПб институт машиностроения (ЛМЗ- ВТУЗ).-2003.-с. 93-96.
6. Бодров А.Й. Суханов В.А., Гольдберг Л. Определение доверительного интервала использования безразмерных характеристик теплообменных аппаратов ГТУ. //Современное турбостроение: Сб. научн. тр. - Вып.5.- СПб.: МАН ВШ. СПб отд-ние. СПб институт машиностроения (ЛМЗ-ВТУЗ). - 2003- с. 97-103.
7. Goldberg L., Krēsliņš A., Bodrov A.I., Sukhanov V.A. Methods for calculation of dimensionless responses and operating modes of heat exchangers. - *Latvian Journal of Physics and Technical Sciences,* 2004, Nr 2, p. 26-35.
8. Goldberg L., Kresling A., Bodrov A.I., Sukhanov V.A. Dimensionless responses of exchangers: Restrictions on the use. - *Latvian Journal of Physics and Technical Sciences,* 2004, Nr 3, p. 11-16.
9. Goldberg L., Krēsliņš A., Bodrov A.I., Sukhanov V.A. Dimensionless responses and operating modes of heat exchangers: Calculation of countercurrent schemes. – *Latvian Journal of Physics and Technical Sciences,* 2004, Nr 3, p. 17-20.