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EVALUATION OF EFFICIENCY OF HEAT EXCHANGING DEVICES BUILT INTO THE THERMAL CYCLIC MACHINES

The focus of this work is a method of exergy-economizing calculation for the recuperative and regenerative heat-exchangers that are built-in the thermal cyclic machines. The method is intended for the analysis of efficiency of a prospective design and material for the heat-exchanger that is incorporated into the closed thermodynamic cycle of thermal machines at the initial stage of designing.

Key words*: heat-exchanging machines, loss of exergy, recuperative and regenerative heat-exchangers.*

Introduction. Currently, mathematical models are used for the optimization of thermal cyclic machines that are being developed. These models describe the elements and their connections in the form of systems of equations, of variable and of system of logic conditions [1], [2], [3], [4]. However, process of development and designe of thermal cyclic machines has a number of stages: preliminary outline sketch, outline, technical and then working projects. For each of the aforementioned stages of an optimization task for thermal cyclic machines, it is expedient to distinguish on depth and detail of study of questions. First developmental and designing stages of thermal cyclic machines are characterized by insufficiency of initial data for a choice of a construction design and sizes, for example, of heat-exchangers. Therefore, alongside with mathematical models that are obtained on the basis of detailed and relatively trustworthy information and that are suitable at stages of technical and working projections for the processes in the thermal cyclic machines the simple engineering methods are needed for choosing fundamental sizes and construction of elements for thermal cyclic machines on first stages of designing. Method of Schmidt-Kirkley [5] is currently used as a model for such preliminary calculation of effective fundamental parameters of thermal cyclic machines and optimum values of relative magnitudes of temperatures and volumes of hot and cold cavities. However, as a result of an assumption about ideality of processes in heat-exchangers in the above method, it becomes impossible to define the fundamental sizes of heat-exchangers, the process in which efficiency of a thermodynamic cycle of the thermal cyclic machine is substantially defined.

Objects and problems. Objects of research in the present work are thermo-exchanging devices of regenerative and recuperative type of the thermal machines working on the closed thermodynamic cycle in which cyclic processes of compression and expansion occur at various levels of temperatures, and management of a flow of a working body is carried out by change of its volume.

The purpose of the present work is the evaluation of efficiency of processes in heat-exchangers using exergy-economizing method. Preference of the exergy-economizing method of the thermodynamic analysis due to two reasons: the heat-exchangers in thermal cyclic machines have substantially different temperatures, therefore the heat transferred has different energetic values; additivity of exergy allows to carry out the separate analysis of various kinds of losses independently from each other, which in the end enables to plan concrete actions to increase efficiency of thermal cyclic machines. Another important feature of the exergy method is a direct correlation of exergy parameters with technical and economic characteristics of machines. Such a direct correlation is impossible when energy which cannot be neither "created" nor "destroyed" is used. On the contrary, exergy, as well as cost, can be both created and destroyed in various processes. Despite of some disagreements concerning what exact form correlation between exergic and economic categories has, researchers agree that application of exergy in economic research is extremely fruitful.

Main section. Exergy coming into a heat-exchanger is transferred to doing useful work and to compensating for losses that are results of irreversible processes, and also the exergy is transferred to cooling heat-carrier. Reduction in total losses of exergy in the cycle will allow to increase useful work of a thermodynamic cycle according to exergic balance.

Losses exergy in heat-exchangers of cyclic thermal machines can be presented in the form of the sum

$$\delta E = \Sigma \delta E_{ij} \qquad (1)$$

where the considered heat exchanger is indicated by index i, and the considered type of a loss is indicated by index *j*.

Following assumptions were made during development of the basics for the exergy-economizing method for the evaluation of efficacy of heat-exchanging devices that are built in a closed cycle of heat machines:

– intensity of heat exchange in the heat-exchanger, axial thermal streams, hydraulic resistance of the heat-exchangers are factors that are independent from each other;

– surface temperatures of the heat-exchangers and of a working body of the thermal machine are constant during an input and an output from the heat-exchangers;

– working body (heat-carrier) of the thermal machine is an ideal gas;

– processes of compression and expansion in the thermal machine are isothermal;

– flowing processes of the working body through the heat-exchangers are adiabatic;

– leakages of a working body from the heat-exchangers are absent.

– Considered kinds of exergy losses in the heat-exchangers in thermal cyclic gas machines are:

– losses due to the pressure drop that are caused by hydraulic resistance of channels of the heat-exchangers;

– losses due to non-ideality of the heat exchange that are caused by final difference of temperatures during heat exchange;

– losses due to longitudinal (axial) heat conductivity of a material of the heat-exchanger;

– losses due to presence of free volume of the heat-exchanger;

– losses due to dispersion of heat in an environment.

The fundamental equations of the exergic balance method were used while determining main components of exergy losses:

– exergy heat equation:

$$dE = (T - T_a)dS; \qquad (2)$$

– exergy flow equation:

$$dE = di - T_a dS = c_p dT - T_a(c_p \frac{dT}{T} - R\frac{dP}{P}); \tag{3}$$

– equation of exergy of a substance in an enclosed volume:

$$dE = dU - T_a dS - PdV. \tag{4}$$

The following notations are used in the equations (2, 3, 4):
dE – change in exergy; dS – change in entropy; di – change in enthalpy; dU – change in internal energy; c_p – thermal capacity of a working body of the thermal cyclic machine at constant pressure; T, P – temperature and pressure accordingly; V – cavity volume of the thermodynamic process; T_a – temperature of the surrounding; R – gas constant of a working body of the thermal cyclic machine.

The basis for calculation of separate constituents to losses are is experimental data that is widely presented in the literature in the form of criteria equations of a kind: $Nu = B\,\mathrm{Re}^m$ – for the description of the convection heat exchange; $\xi = \frac{A}{\mathrm{Re}^n}$ – for the description of the hydrodynamic resistance of the heat-exchangers, where ξ – is a coefficient of the hydrodynamic resistance channels of the corresponding heat-exchangers; Re, Nu – are criteria of Reynolds and Nusselt correspondingly; A, B, n, m – are empirical coefficients in criteria equations.

On the basis of the made assumptions and the exergy equations, we receive analytical expressions of each kind of losses in the heat-exchangers.

Exergy loss due to the pressure drop that is caused by hydraulic resistance of the heat-exchangers' channels, during the adiabatic process with a constant enthalpy is ($c_p dT = 0,\ c_p \frac{dT}{T} = 0$):

$$\delta E_{ip} = \frac{30 A_i h_i T_a T_i R^2 \mu^n G^{3-n}}{P_{\max}^2 n_0 \Delta d_{эi}^{n+1} F_i^{2-n}}. \tag{5}$$

According to the exergy heat equation (2), averages for a cycle of exergy loss (due to the presence of unoccupied heat-exchangers' volume that was forcefully included in the volume of the working body that participates in realization of a thermodynamic cycle) are:

$$\delta E_{iM} = \frac{1}{2} F_i h_i P_{\max} \frac{(1-\Delta)^2}{\sqrt{\Delta}} (1 - \frac{T_a}{T_i}). \tag{6}$$

Exergy loss due to non-ideality of the heat exchange, and caused by final difference of temperatures of a working body and a nozzle of a regenerator, is:

$$dE_{pm} = \mu^m \frac{60 F^{m-1}}{n_0} \left(\frac{G d_э}{\Pi}\right)^{2-m} \frac{2 T_a c_p^2 (1-t)^2 \Pi \varphi}{\lambda B h_p t}. \tag{7}$$

Exergy loss due to final temperature difference of walls of recuperative heat-exchangers and the working body:

$$dE_{im}=\frac{T_a d_э^{1-m_i}\mu^{m_i}F_i^{m_i}}{2F_i^* B_i G^{m_i}\lambda_i}\left(\frac{Q_i}{T_i}\right)^2\frac{n_0}{60}. \quad (8)$$

Exergy loss due to presence of a thermal stream on the frame and on the nozzle of a regenerator:

$$\delta E_{pT}=\frac{\lambda_э(1-t)^2 T_a F_p 60}{h_p t \Pi n_0} \text{ and } \delta E_{pTK}=\frac{\lambda_k \pi h_k (D_p+2h_p)(1-t)^2 T_a F_p 60}{h_p t n_0}. \quad (9)$$

Loss due to heat dissipation into surroundings:

$$\delta E_{p\Pi}=\frac{\lambda_к (T_p-T_a)^2 \pi D_p h_p 60}{h_k T_p n_0}. \quad (10)$$

Following notation is used in expressions (5)-(12):

T_a, T_i – temperatures of the surroundings and the working body accordingly in the heat-exchanger, К; $d_{эi}$ – equivalent diameter of channel openings (pores of a porous material of a nozzle, internal diameter of tubes of the recuperative heat-exchangers) of the corresponding the heat-exchanger, m; F_i – area of effective cross-sections of the corresponding the heat-exchanger, m^2; D_p, h_p – diameter and length of the regenerator, m; h_i – length of channels of the corresponding the heat-exchanger, m; G – bulk flow of a working body through the corresponding the heat-exchanger, kg/s; $P_{\min}, P_{\max}, \Delta$ – minimal, maximal pressure, N/m^2 of the thermodynamic cycle and their ratio; n_0 – rotation frequency of a crank of the cyclic thermal, min^{-1}; μ, λ – coefficients of dynamic viscosity, Ns/m^2 and thermal conductivity, W/sК of the working body of the thermal cyclic machine; $t=\frac{T_X}{T_Г}$ – ratio of the temperatures on the entrance and exit from the regenerative heat-exchanger; Q_i – thermal flow through the recuperative heat-exchangers; $\lambda_э, \lambda_к$ – coefficients of heat conductivity, W/sК of regenerator nozzle material and regenerator thermal insulation accordingly; $h_к$ – width, m of thermal insulation of the regenerator; П – porosity of the regenerator nozzle.

In the dependencies related to recuperative heat-exchangers, the values of effective cross-sections F_i and F_i^* of heat exchange area are determined by the sizes of heat-exchangers depending on their types. In this case tube-type heat exchanger:

$$F_i=\pi d_i^2\frac{n_t}{4}, \quad (11)$$

$$F_i^*=\pi d_i n_t h_{ti}, \quad (12)$$

where n_t, d_i, h_{ti} – number, internal diameter and tube length of recuperative heat exchanger.

Exergic analysis demonstrated above allows evaluating the amount of exergy loss during the processes of heat regeneration and recuperation from a cyclic heat machine as a function of their base dimensions. Exergy is directly obtained from naturally existent sources – fuel resources, fissionable matter, etc. Exergic supply constantly demands financial resources, human work hours and energy, which are reflected in the final cost of a unit of fuel. Therefore, for approximate determination of the amount of exergy entering a thermal cyclic machine, a dependency can be utilized, proposed in work [6] for liquid fuels:

$$\frac{E_T}{H_u} = 1{,}0374 + 0{,}0159\frac{H}{C} + 0{,}0567\frac{O}{C} + 0{,}5985\frac{S}{C}(1 - 0{,}1737), \qquad (13)$$

where E_T – exergy of a fuel unit mass; H_u – Lowest heat of burning a unit of mass of fuel; $\frac{H}{C}, \frac{O}{C}, \frac{S}{C}, \frac{N}{C}$ – ratios of atomic numbers of hydrogen, oxygen, sulphur, nitrogen and carbon to a molecule of fuel.

In that case, additional expenditures for fuel during realization of regeneration and recuperation processes in a thermal cyclic machine are determined as follows:

$$\mathrm{S_T} = \mathrm{C_T}\frac{\delta \mathrm{E}}{\mathrm{E_T}}, \qquad (14)$$

where $\delta \mathrm{E}_\Sigma$ – total exergy loss during realization of regeneration and recuperation processes, determined by formulas (5-10); $\mathrm{C_T}$ – cost of a unit mass of fuel.

However, the real losses in a thermal cyclic machine are higher. The reason is that other parts and machine layout pieces took part during exergy conversions and transfers, which also carried human work expenditures. The objective of exergic analysis is to determine financial resource expenditures for execution of thermodynamic cycle of thermal cyclic machine. A particular objective of this analysis is the determination of above-mentioned expenditures for regeneration and recuperation processes to occur. Those expenditures are added from expenditures to manufacture S_N heat-exchangers and their utilization S_{T}:

$$S = S_N + S_T + S_P + S_Y, \qquad (15)$$

where $S_N = K_1 + K_2$ – expenditures to manufacture the heat-exchanger; K_1 – cost of unit of mass of source material during manufacturing heat-exchanger material; K_2 – cost of assembly of a unit of mass of heat-exchanger; S_P – expenditures for planned maintenance of heat-exchanger elements; S_Y – expenditures for utilization of heat-exchanger.

The cost of a unit of mass of a heat-exchanger material, depending on its type, is presented in a table of data, which makes sense to be represented as analytical dependencies like $K_1 = \frac{a}{d_э^b}$. For example, table data on the grid of phosphorous bronze BrOF 6,5-0,4 normal precision according to standard GOST 6613-73 [7], used as regenerators' plantings, could be approximated with this dependency: $K_1 = \frac{52404{,}13}{d^{1{,}834}}$. Analogous dependencies can also be obtained for tube-type heat-exchangers. Manufacturing expenditures for those are also dependent on channel diameter and assembly technology (soldering or welding for example).

Analysis of the obtained dependencies shows that with channel diameter increase of heat-exchangers, expenditures rise sharply for executing thermal exchange process as a consequence of final temperature difference of the walls in recuperative heat-exchangers and working body and decreasing expenditures for pushing through the working body through the channels of the heat-exchangers.

Conclusion. Correlations between exergy and economic parameters of regenerative and recuperative heat-exchangers in thermal cyclic machines are established. The correlations allow to perform comparative characterizations of various types of heat-exchangers and designs for heat-exchangers at the first stages of their development and design, which is performed on the basis of the initial information on their thermal and hydrolytic characteristics. Simplicity of the received correlations allows to find optimum values of the most important measures of heat-exchangers of the thermal cyclic machine and to minimize the cost method of their manufacturing.

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ОЦІНКА ЕФЕКТИВНОСТІ ТЕПЛОВИХ ПРИЛАДІВ, ВИРОБЛЕНИХ В ТЕРМАЛЬНІ ЦИКЛІЧНІ МАШИНИ

Основна увага в цій роботі присвячується методу розрахунку енергоефективності для рекупераційних та регенеративних теплообмінників, що вбудовуються в теплових циклічних машинах. Метод призначений для аналізу ефективності перспективної конструкції та матеріалу для теплообмінника, який включений у закритий термодинамічний цикл теплових машин на початковому етапі проектування. Перевага ексергійно-економічного методу термодинамічного аналізу полягає в наступному: теплообмінники в теплових циклічних машинах мають істотно різні температури, тому передане тепло має різні енергетичні значення; аддиктивність ексергій дозволяє проводити окремий аналіз різного роду втрат незалежно один від одного, що врешті-решт дає змогу планувати конкретні дії для підвищення ефективності теплових циклічних машин. Ще однією важливою особливістю методу ексергії є пряме співвідношення параметрів ексергії з техніко-економічними характеристиками машин. Ексергія, що надходить в теплообмінник, передається для виконання корисної роботи і компенсації втрат, що є результатом незворотних процесів, а також переноситься на охолоджуючий теплоносій. Розглядаються наступні співвідношення: відповідно до рівняння теплової ексергії, середні значення для циклу втрат ексергії (через наявність об'єму незайнятих теплообмінників, який був включений до об'єму робочого органу, який бере участь у реалізації термодинамічного циклу); втрати від ексергії внаслідок неідеальності теплообміну, спричиненої остаточною різницею температур робочого тіла і сопла регенератора; втрати напруги через кінцеву різницю температур стінок рекуперативних теплообмінників та робочого органу; втрати від напруги через наявності теплового потоку на рамі та на соплі регенератора та втрати внаслідок відведення тепла в зовнішнє середовище. Ексергічний аналіз дозволяє оцінити величину втрат енергії під час процесів регенерації тепла та відновлення від циклічної теплової машини як функцію від їх базових розмірів. Екзергічне постачання постійно вимагає фінансових ресурсів, робочого часу людини та енергії, що відображається на кінцевій вартості одиниці палива.

Ключові слова: теплообмінні машини, втрата напруги, відновлювальні та регенераційні теплообмінники