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SIMULATING HEAT-TRANSFER PROCESSES IN TWO-PHASE HEAT-UTILIZER AT HEAT PIPES

Abstract

The choice of rational heat-utilizer heating scheme is a very current problem of energy saving when using boiler units. This article proposes the heat computation technique of heat exchanger at heat pipes with deep combustion - product cooling in boilers and water steam condensation. In addition, the construction scheme of heat exchanger made with two units in which the heat pipes are filled with different working medium is described. The numerical study results of heat carrier temperature and pressure distribution, namely: combustion products and water being heated according to counterflow - heat exchanger length are presented. The efficiency of two-phase heat-utilizer is determined.

Keywords: boiler unit, heat carrier, heat pipes, heat utilizer

1. Introduction

At present about 35–40% of fuel-power resources in Ukraine is used for heat supply of enterprises and inhabited localities, 70% of heat being generated at centralized and individual boiler-houses. Expenses on building heating are not less than 50% of the overall housing-municipal sector.

One of the perspective directions in energy saving is combustion-product heat utilization of boiler units at the expense of their cooling below the dew point (50÷55°C) and emitting heat of water-vapour condensation. At the same time condensate will include water-solute gases and weighted particles. That is why the deep combustion product cooling of boiler units in the condensation mode provides not only the increase of the fuel use ratio but also the increase of their ecological efficiency.

Heat losses with waste gases of boiler units reach 5÷6%. When the capacity of DE-type gas and fuel-oil boilers is nominal, the temperature of waste gases behind the economizer is 140÷160°C using gas and from 170 to 190°C using oil as fuel.

Different types of heat utilizers are used in the heat utilization systems. The choice of heat utilizer type and capacity required is determined by the available users of the heat being utilized rather than the fixed boiler unit capacity: heating the chemically pure water, heating the blowing air, hot water supply system, heating the return delivery water, technological demands of an industrial enterprise, heating the water for heat supply systems of seed-bed and greenhouse economy and etc.

Contact heat-exchangers in which heat-exchange takes place between smoke gases and cooling water are widely used. Such heat exchangers with suitable sizes, moderate metal consumption for their fabrication and comparatively small power consumption while operating provide the deep cooling of smoke gases up to 40÷45°C and 60–90% water-vapour condensation contained in gases.

In contact heat-exchangers water heating is possible only up to the temperature of “wet thermometer” that is 50÷60°C and depends on moisture content of smoke gases and air excess coefficient. When the moisture content increases, the partial pressure of water vapour in smoke gases grows and the temperature of the “wet thermometer” increases as well. The efficiency of contact heat exchangers decreases with the increase of water (air) temperature and their application is recommended when the water temperature at the outlet is not more than 35°C .

Water being heated contains carbon dioxide ranging from 0.008 to 0.1 g/m^3 that increases its corrosion activity when pH = 2.8÷4.9.

The absorption of NO_x, O_2, CO_2 gives the condensate the properties of weak carbonic and nitric acids, having the increased corrosive qualities but at the same time nitric oxide content is decreased by 28–31% that improves ecological characteristics of the boiler unit.

Condensing surface heat utilizers-economizers are applied in The Netherlands, France, Germany, the USA, Canada, Italy and other countries.

Russia produces bimetallic (steel-aluminum) heat exchangers (calorifers) – KC_k-4-11).

The comparative analysis concerning the characteristics of contact – and surface-type-condensing boilers is given in [14, 16].

The comparison of contact-and-surface-heat exchangers parameters shows that surface apparatuses are characterized by lower capital expenses, less power consumption. Their cost price and compensation period are considerably lower. Combined heat utilizers in which the first stage is the contact apparatus, the second stage is the surface one are being worked out [24].

The usage of condensation heat utilizers at the heating boiler houses allows to increase the efficiency by 6–8% and decrease the cost price of the heat energy being generated by 9–11%.

2. Analysis of publication, materials, methods

The heat of waste gases in heat power and technological units is the main kind of secondary power resources (SPK) in industry [11].

Significant reserves of secondary power resource saving consist not only of heat utilization of high temperature gases but also of low temperature ones (up to 300°C).

In Japan [14] an original design of the heat exchanger with 75 MW heat capacity for utilizing the heat of waste gases leaving the boiler was developed and used in industry. Air is allowed to be heated from 50°C up to 340°C at the initial gas temperature of 370°C; the temperature of waste gases decreases up to 130°C .

Russian industry manufactures ЭК-БМ1-1 and ЭК-БМ1-2 water economizers in series [12] which are characterized by the following parameters: heat capacity from 0.87 to 1.22 MW, gas rate from 3600 to 14400 kg/h, water rate from 8 to 40 t/h being heated up to 42–55°C, waste gases enter at 140°C and cool up to 30–40°C.

Devices having intermediate liquid heat-carriers (KTAN – Contact Heat Exchanger with an active nozzle) are being designed [20] characteristics being presented in Table 1.

Table 1. KTANs characteristics

Characteristics of KTANs-utilizers КТАН – 1.5 УГ	
Heat capacity, MW	1.5
Gas rate, nm^3/s	4.5
Temperature of water being heated, ºC	
at the inlet	5
at the outlet	50
Temperature of smoke gases, ºC	
at the inlet	140
at the outlet	40
Heat exchange surface, m^2	52.4
dimensions, m	
length	2.66
width	1.75
altitude	4.16
Aerodynamic resistance, Pa	493

The usage of condensing heat exchangers is more effective for low-capacity boilers than for devices with KTANs [20].

BAXI, Viessmann, Ferroli, Beretta, BOSH, Buderus condensing boiler and others with the 200 kW capacity are equipped with a condensing surface heat exchanger in which the deep cooling of combustion products occurs while condensing water vapour.

Heat losses with waste gases are lowered up to 2%. Condensing utilizers of КСК- and ТП-Т1РК types were developed by the institutes (Gorky State Research Institute – Santechproject).

Table 2. The TP46-T1RK03 characteristics

ТП46-Т1РК03 Characteristics	
Air rate, thon, m^3/h	16
Surface heat exchange area, m^2	68.0
Dimensions, mm	
length	1727
width	180
altitude	1075

Effective compact heat-exchangers-utilizer at the ribbed thermosyphons for low-capacity steam boilers (with steam capacity of 1 t/h) were developed in Kiev Polytechnic Institute (NTUU "KPI") [2].

The scheme of the simplest utilizer at the heat pipes is shown in Fig. 1

Heat-exchanger is made up of two gas passages 1 and 2, separated by gas-dense baffle 3. Closed heat-transferring elements in a kind of heat pipes 4 having free ends placed in gas passages with "hot" and "cold" heat-carriers are fixed in the baffle. The intermediate heat carrier by means of which the heat in vapourizing – condensating cycle transfers from "hot" to "cold" heat carrier is put inside the closed elements (thermosyphons).

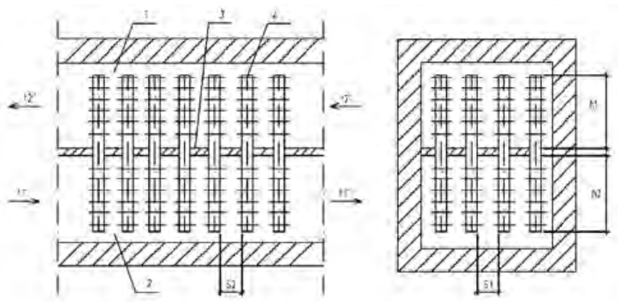

Fig. 1. The scheme of the simplest utilizer at the heat pipes (1, 2 – gas passages, 3 – gas-dense baffle, 4 – heat pipe)

Cooling the smoke gases in the condensing heat-exchanger lowers the moisture content by 70–80% that does not exclude the possibility of water vapour condensation in gas passages and smoke pipe. The absence of reliable dependencies determining the process of moisture content change does not permit (operational staff) to make correct decision concerning the possibility of increasing technical and economical parameters and plant reliability.

Setting the condensing heat utilizer into the gas passage causes the danger of breaking an expensive and crucial plant element – a smoke pipe by an aggressive condensate. That is why while operating the boiler unit one should define the part of smoke gases to be taken when there is maximum heat emission in the heat utilizer providing dew point occurs in the smoke pipe mouth.

Thermodynamical processes when cooling combustion products and condensing water vapour are shown in the i-d diagram [7] (Fig. 2).

While cooling relative air moisture will increase until it reaches 100% at the temperature called dew point in the i-d diagram – cross section x = const, φ = 1. Smoke gas cooling below dew point is accompanied by moisture condensation, i.e. gas drying (process 1, Fig. 2).

When installing surface heat utilizers behind the boilers waste gases are cooled and partly dried. Gas moisture content decreases up to 0.03÷0.06 kg/kg d.g., and dew point up to 30÷40°C , that is a positive factor because the condensate falling out is prevented in the smoke pipe.

In the contact heat utilizer one can observe adiabatic cooling (variant 3, Fig. 2). Fluid can be heated up to the temperature of wet thermometer.

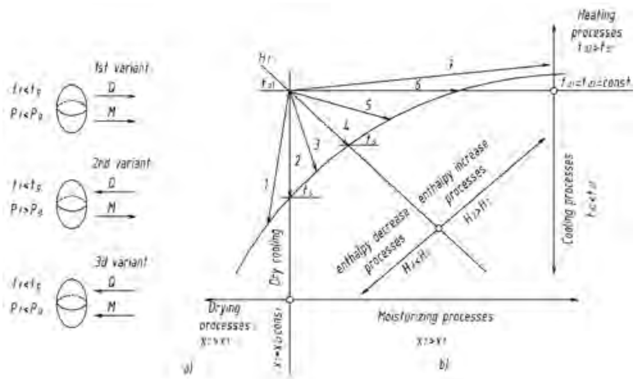

Fig. 2. Main processes occurring in the contact of drops and gas: a) direction variants of heat and moisture transfer; b) process presentation on H-x diagram: 1 – gas cooling and drying; 2 – dry gas cooling; 3 – cooling and moisturizing with the resulting gas temperature decrease; 4 – cooling with moisturizing without enthalpy change; 5 – cooling with enthalpy increase and change; 6-isothermal moisturizing; 7 – gas heating with moisturizing

Variants 1 and 2 (Fig. 2) – vapourizing water cooling and drop heating with simultaneous vapourizing the moisture from their surface respectively – do not practically occur in contact heat utilizers.

Steam condensation from the smoke gas mixtures in the presence of noncondensing gases (CO_2, NO_x and others) on the surface of a solid (or liquid) is less intensive compared to pure steam condensation. If steam is immobile, even negligible gas content in it results in the abrupt decrease of condensation intensity. With the increase of velocity gas effect upon process intensity decreases. The dependence of dew point on the value of air excess ratio and CO_2, NO_x gas concentration (that are changed when smoke gases move along the gas passage) making the heat-calculation technique of the heat utilizer rather complicated. Figures 3, 4 illustrates the effect of the parameters mentioned on dew point.

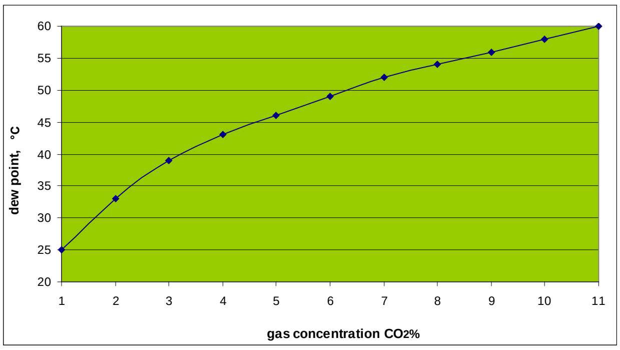

Fig. 3. Plot of dew point vs. CO_2 gas concentration

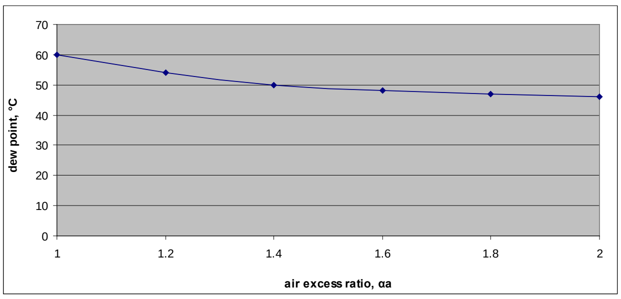

Fig. 4. Plot of dew point vs. air excess

Moist-combustion-product parameters of the boiler unit capacity are presented below (Tab. 3)

Table 3. Combustion product parameters of КВГМ-100 boiler

Parameters	Boiler capacity mode		
	mini-mal	ave-rage	maximal
Air rate ratio α	1.76	1.23	1.14
Moisture content, d, kg/kg	0.08	0.11	0.12
Dew point, °C	49.3	55.3	56.6

Thus, the technical restriction preventing the implementation of the given heat utilization technology are the following:

– complexity concerning heat utilization process calculation products;
– the necessity to maintain the given values of waste gas temperature and moisture by means of bypassing a part of waste gas rate.

Heat-calculation techniques known in literature relating to condensing surface heat-exchangers are either presented not completely or their data are quite different. References [9, 11] give information on heat transfer ratio. If in [9] the value of heat transfer ratio is 300÷400 W/(m²K), in [11] it is – 45÷88 W/(m²K).

The most valid calculating dependencies used in practice are obtained experimentally [2–8, 11–14, 18, 17–22]. However, known calculating dependencies for heat utilizers with deep smoke gas cooling at vertical heat pipes, filled with low temperature working medium should be defined more precisely.

Get us analyze some of them.

The value of heat rejection coefficient from smoke gases towards the ribbed surface is estimated by the equation [11]:

$$Nu_d = 4.55\ \mathrm{Re}^{0.315}\ K^{0.388}\ \mathrm{Pr}^{2/3} \qquad (1)$$

Where Nu, Re, Pr, K – Nusselt number, Reynolds, Prandtl and irrigation criteria, respectively.

The presented heat rejection coefficient from smoke gases forwards the outer heat-utilizer surface is estimated by the equation (1) in terms of irrigation (density) with the condensate. The criterion equation is valid for $КС_к$-4-11-02 ХЛ3-type condensing heat utilizer-calorifer (the producer – JS "Calorifer plant", Kostroma).

Figure 5 shows the plot of heat transfer coefficient vs. combustion product speed and irrigation density.

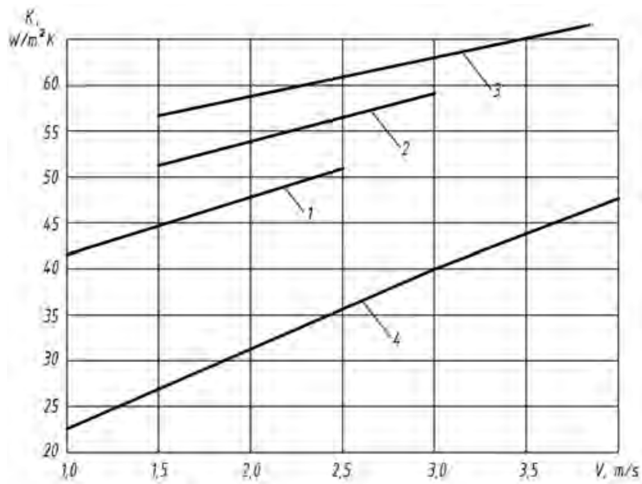

Fig. 5. The plot of heat transfer coefficient Km W/(m²K) vs. gas rate V m/s and irrigation density W (1 – W = 2.28 ÷ 3.57; 2 – W=2.83 ÷ 3.67; 3 – W=3.19 ÷ 4.03; 4 – for air ("dry" heat exchange))

Criterion dependence is presented in [24]

$$\overline{Nu} = 0.24\,\mathrm{Re}^{0.62}\ K^{0.25}\ \mathrm{Pr}^{0.33} \qquad (2)$$
$$\left(K = wD/\mu\right)$$

Irrigation criterion K allows the effect of irrigation density W (kg/m²h), connected with the cooling rate of smoke gases to be evaluated.

Intensity (density) of irrigation W (kg/m²h) is determined by the rate of condensate dropping out and surface heat exchange area.

References [13, 21] presents the plot $Nu = A\mathrm{Re}^{0.6}\ exe(m\Theta)$ that defines additional increase of Nusselt criterion while condensing water vapour from combustion products. The heat rejection ratio is shown to be increased by 25–40 W/m²K (30 – 50%).

It is experimentally found that the value of heat transfer coefficient under condition of water vapour condensation is 1.5 and 1.8 times larger than the K value in case of dry heat exchange for water industrial economizers respectively with the irrigation density 3.19–4.03 kg/m^{2}h and gas rate 1.89–3.78 m/s.

When estimating the heat transfer coefficient the most difficult thing is to define α_g, α_f heat rejection ratios from steam and gas mixtures towards the condensate film surface and from condensate film towards the cooling fluids (surface).

Heat rejection ratio α_g is determined by the known plots [3].

Heat rejection ratio α_f for horizontally placed pipe bunches is estimated by the equation:

$$\alpha_f = c_w \alpha_н, \tag{3}$$

where C_w – the ratio taking into account motion rate of gas and steam mixture; $\alpha_н$ – heat rejection ratio while condensing slowly moving steam:

$$\alpha_н = 0.728 \sqrt{\frac{g \rho_c^2 \lambda_c^3 r}{\mu_c d \Delta t_{sw}}}, \tag{4}$$

where ρ_c, λ_c, μ_c – density, heat transfer ratio, dynamic condensate toughness ratio; d – pipe diameter; Δt_{sw} $t_{гр} - t_{cг}$ – temperature head "steam-wall", $\Delta t_{sw} \approx 2°C$, respectively.

Mass rejection ratio β can be defined based on analogy between heat exchange and mass exchange by familiar plots [2, 3, 4, 13] using the plot of the $\beta = f(\alpha_g)$ type and criterion equations.

Reference [3] presents the equation:

$$\beta = \frac{\alpha_g}{C_{pav} P_{mf}} \cdot \frac{\mu_n}{\mu_{av}} \cdot \left(\frac{\Pr}{\Pr_d}\right)^{\frac{2}{3}}, \tag{5}$$

where μ_{av} – average molecular mass (weight) of gas and steam mixture:

$$\mu_{av} = \mu_g \xi_g + \mu_n (1 - \zeta), \tag{6}$$

Pr – Prandtl's heat criterion; $\Pr_d$ – Prandtl's diffusion criterion; P_{mf} – motive force of mass exchange, determined by the equation:

$$P_{mf} = \frac{P_s - P_b}{\ln \frac{P - P_g}{P - P_s}}, \tag{7}$$

The density of cross mass flow is defined by the equation:

$$g = \beta (P_s - P_b), \tag{8}$$

Theoretical heat rejection ratio from the steam and gas mixture towards the heat-exchange surface wall is estimated by the equation:

$$\alpha_1 = \left\{ \frac{1}{\beta \left[r + c_k \left(\vartheta_{am}^{in} - \vartheta_{am}^{ou} \right) \frac{P_s - P_b}{\vartheta_{ami} - t_{rpi}} + \alpha_g \right]} + \frac{1}{\alpha_f} \right\}^{-1} \tag{9}$$

Common theoretical heat-transfer ratio (coefficient) of heat transfer from the steam and gas mixture towards the heat-carrier being heated is determined in the following way:

$$K = \frac{\phi}{\frac{1}{\alpha_x} + \sum R_i + \frac{1}{\alpha_в}}, \tag{10}$$

where φ – ribbing coefficient.

The given amount of the condensate being emitted from the combustion products while they are cooled is estimated by the equation:

$$\Delta g^g = \left[g_{da}^g + g_{com}^g (\alpha_{ou} - 1) \right] \left(x_{in}' - x_{out}'' \right), \tag{11}$$

where g_{da} – the given theoretical rate of dry blowing air, g_{da}=1.415; g_{com} – the given theoretical rate of dry combustion products, g_{com} = 1.333; α_{in} – air excess ratio in front of the heat utilizer; x'_{in}, x''_{out} – initial moisture content of combustion products at heat utilizer inlet and its outlet, kg/kg.d.g., respectively.

The characteristics given are determined in relation to the lowest combustion heat Q^p_l, Mcal/m^3. Absolute condensate amount is defined in the following way:

$$\Delta g = \Delta g^g Q_L^p, \tag{12}$$

Values x'_{in}, x''_{out} are determined by the equations:

$$x_{in}' = (0.13 + x_m \alpha_{in}) / (\alpha_{in} - 0.058)$$

$$x_{out}'' = \frac{0.0006382 + 0.004 \alpha_{out}}{0.199 + \alpha_{out}} \exp\left(0.062 t_{out}''\right),$$

where x_m – moisture content of blowing air, kg/kg.d.a., t_{out} – waste combustion product at the outlet from the heat utilizer, °C .

Thus, the amount of the condensate being emitted depends on blowing air moisture-content, air excess ratio and combustion product temperature at the heat-utilizer outlet.

3. The aim and setting the research problem

The object of the work is working out the heat calculation technique of two-stage heat utilizer using heat pipes when having the deep cooling of combustion products with water vapour condensation.

Zone by zone calculation method [9, 13], being the most effective one, is widely used when making the heat calculation of condensing heat utilizers.

Combustion products of gaseous fuel in the boiler units constitute steam and gas mixture, making up of inert mixtures (CO_2, CO, N_2, NO, H_2O, air and so on), their water vapour content being about 15÷17%. Water vapour condensation is observed when having the deep cooling of combustion products. Cross-flow density of the condensate mass towards the cooling heat utilizer surfaces is 0.1 kg/(m·s) that is much lower as compared to pure water vapour condensation. The film condensation mode is observed, condensation surface geometry greatly influencing the intensity of heat mass exchange processes. The heat rejection ratio for horizontal surfaces is two times more than for vertical ones $\alpha_{vert} \approx 0.55\alpha_{hor}$. When installing the ribs the heat rejection intensity (when condensing water vapour) can be greatly increased and approach the value α_{hor} [3–5].

4. The main section

The given work proposes the heat calculation technique of two-stage heat utilizer using heat pipes (thermosyphons) in terms of water vapour condensation with the deep cooling of the boiler unit combustion products.

This calculation technique suggests zone-by-zone calculation of parameters (temperature, pressure, thermosyphons capacity, condensate dropping out, etc) along the heat exchanger.

The scheme of heat carrier motion is the following: water-cross-traverse multi-stroke motion; combustion products-traverse-countercurrent one. Thermosyphon design features are: condensation zone and evaporation zone – ribbed ones with cross ribs, thermosyphon zone length being different, $h_2>h_1$. "Gas-liquid"-type heat utilizer is used for obtained hot water with 50 ÷55°C temperature. Pipe diameter is equal to 32÷57 mm that provides making restrictions according to maximum density of heat flow in the gas passage [2].

Heat transfer coefficient of the heat utilizer is determined by the equation:

$$K_i = \frac{1}{\dfrac{1}{\phi_1 \alpha_{in1}} + \dfrac{1}{\alpha_v} + \sum_i \dfrac{\delta_w}{\lambda_w} + \dfrac{1}{\alpha_c} + \dfrac{1}{\alpha_{in2}\phi_2}} \qquad (13)$$

where α_{in1}, α_{in2} – heat exchange ratio between heat exchangers (combustion products, water) and inner thermosyphon surface in the vaporation and condensation zones; α_v, α_c – heat rejection ratios in the evaporation and condensation zones, respectively; $\sum\delta_w/\lambda_w$ – thermal resistance of thermosyphon walls, φ_1, φ_2 – ribbing coefficients.

Heat output of the heat utilizer is defined by the equation:

$$Q = KF\Delta\bar{t} \qquad (14)$$

where F – surface area; Δt – mean temperature head.

Heat rejection ratios in different heat utilizer zones are determined according to the following plots (Tab. 6–10).

Suitable accuracy of computation and simplicity of dependence that is required in the optimization problem were taken as the criterion of selecting computation plots.

The ratio of outer heat exchange between combustion products and ribbed thermosyphons in the "dry" zone and in the zone of water vapour condensation was determined using different plots.

Dependence comparison results are shown in Tables 4, 5.

Table 4. The comparison of heat rejection ratio calculation-dependencies for ribbed chess-like pipe bunches in the "dry" heat utilizer zone

Reference	Dependence	Value of α, W/m²K
[29]	$Nu = 0.192(a/b)^{0.2}(s/d)^{0.18}(h/d)^{-0.14} \times \mathrm{Re}^{0.65}\,\mathrm{Pr}_{ж}^{0.36}(\mathrm{Pr}_{ж}/\mathrm{Pr}_{сп})^{0.25}$	11.40
[10]	$Nu = 0.41\,\mathrm{Re}^{0.6}\cdot \mathrm{Pr}_{ж}^{0.33}\cdot\left(\dfrac{\mathrm{Pr}_{ж}}{\mathrm{Pr}_{ст}}\right)^{0.25} \varepsilon_i\varepsilon_s$	13.70
[22]	$Nu = 0.36\,\mathrm{Re}^{n}\cdot \mathrm{Pr}^{0.3}\,c_z c_s\cdot\phi^{-0.5} \quad n = 0.6\cdot\phi^{0.7}$ with Re>10³	16.70

Table 5. The comparison of heat rejection ratio calculation-dependencies for ribbed chess-like pipe bunches in the zone of water vapour condensation

Reference	Dependence	Value of α, W/m²K
[11]	$Nu = 4.55\,\mathrm{Re}^{0.315}\,\mathrm{Pr}^{\frac{2}{3}}\,K^{0.388}$	50÷60
[23]	$\alpha_{\Sigma} = \alpha_{конв}\left[1+\frac{r_{t_s}}{c_p}\cdot\frac{R}{R_0}\cdot\frac{r_n - r_{no}}{t_z - t_o}\right]$	30÷40
[21]	$Nu_{\Sigma} = Nu_{конв} + Nu^{доп} = Nu_{конв} + A\,\mathrm{Re}_{ж}^{0.6}\exp(-14\theta),\quad A = 0.001\exp(87X)+0.3/X$	40÷60

Heat exchange coefficients inside the thermosyphon in the evaporation zone were determined according to the dependencies presented in Table 6.

Table 6. Calculation dependencies for heat rejection ratio in the evaporation zone of the thermosyphon

Reference	Dependence	Value of α, W/m²K
[18]	$Nu_* = 0.0096\cdot Pe^{0.7}\cdot\mathrm{Pr}^{-1}\cdot K_p^{0.6}\quad Nu_* = \frac{\alpha\cdot\ell_*}{\lambda'};\quad \ell_*\left(=\frac{\sigma}{g(p'-p'')}\right)^{0.5}$	5228.33
[18]	$Nu_* = 0.0123\cdot\mathrm{Re}_*^{0.5}\cdot\mathrm{Pr}^{0.35}\cdot K_p^{0.54}\left(\frac{d_{вн}}{\ell_*}\right)^{0.17}$	6025.00
[18]	$Nu_* = 4.05\cdot\mathrm{Re}^{0.5}\cdot\left(\frac{d_{вн}}{\ell_u}\right)^{0.17}$	6870.00
[30]	$\alpha_{1u} = B\left(\frac{\lambda_s}{v_s\sigma_s T_s}\right)^{\frac{1}{3}} q^{\frac{2}{3}}$	4697.15

Calculation dependencies for heat rejection ratio in the thermosyphon condensation zone

Table 7. Heat exchange coefficient inside the thermosyphon at the condensation

Reference	Dependence	Value of α, W/m²K
Nusselt's theory	$\bar{\alpha} = 0.943\cdot\sqrt[4]{\frac{r\rho'^2 g\lambda'^3}{u'(T_s - T_{cm})\ell_{к}}}$	1100
[2]	$\frac{N\bar{u}^*}{\mathrm{Pr}^{0.54}} = 0.21 Fr^{*0.24}$	3240
	$Nu^*\frac{\alpha_{к}}{\lambda}\left[\frac{v^2}{g(1-S''/S')}\right]$	
	$Fr^* = \frac{W_0''^2}{g\delta}\cdot\frac{\rho''}{\rho'}$	
[28]	$\alpha_{1к} = 400\cdot\frac{r\mu_s}{(l_u\cdot\Delta t)}\cdot\left\{1+0.625\cdot\mathrm{Pr}^{0.5}\left[\frac{(l_u\Delta t)}{(l_u\Delta t)_{kp}} - 1\right]\right\}^{\frac{4}{3}}$	982.13

Table 8. The coefficient of outer heat exchange between water being heated (in the thermal siphon condensation zone) and smooth and ribbed thermosyphon surface

Reference	Dependence	Value of α, W/m²K
Smooth pipe bunches		
[30]	$\overline{Nu} = 0.22\ \mathrm{Re}^{0.65} \cdot \mathrm{Pr}_{жс}^{0.36} \cdot \left(\frac{\mathrm{Pr}_{жс}}{\mathrm{Pr}_{cm}}\right)^{0.25}$	728
[10]	$\overline{Nu} = 0.26\ \mathrm{Re}^{0.65} \cdot \mathrm{Pr}_{жс}^{0.33} \cdot \left(\frac{\mathrm{Pr}_{жс}}{\mathrm{Pr}_{cm}}\right)^{0.25} \varepsilon_i \cdot \varepsilon_s$	766
[29]	$\overline{Nu} = 0.27\ \mathrm{Re}^{0.63} \cdot \mathrm{Pr}^{0.36}$	760
[22]	$\overline{Nu} = 0.20\ \mathrm{Re}^{0.65} \cdot \mathrm{Pr}^{0.33}$	680
Ribbed pipe bunches		
[22]	$\overline{Nu} = 0.2\ \mathrm{Re}^{n} \cdot \mathrm{Pr}^{0.33} \cdot c_z \cdot c_s \cdot \phi^{0.7} \quad n = 0.65 \cdot \phi^{0.7}$ with $R=10^4 \div 10^5$	2976
[29]	$\overline{Nu} = 0.303\ \mathrm{Re}^{0.625} \cdot \mathrm{Pr}^{0.36} \cdot \left(\frac{\mathrm{Pr}_{жс}}{\mathrm{Pr}_{cm}}\right)^{0.25} \cdot \phi^{-0.375}$	3112

Equations [2, 22, 23, 28, 30] were used in the program while calculating the heat rejection ratio.
The results of numerical calculations are the following:

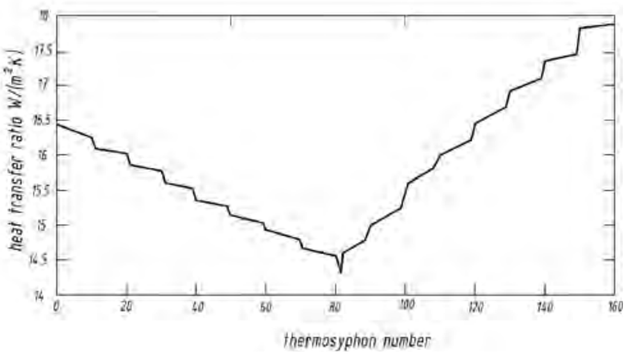

Fig. 6 Shows heat transfer ratio change along heat utilizer

5. Conclusions

The results of numerical study show the possibility of increasing the efficiency of heat utilizers, made according to two-stage scheme with various heat carriers in thermosyphons.

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