Study of Prospects of Two-Phase Gravity Thermosiphons Used in Waste Heat Boilers of Cogeneration Units

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Abstract. The significance of this work is justified by the lack of experimental data on the operation of thermosiphons as part of the waste heat boilers (WHB) with gas turbine engines (GTE), whose capacity is from 2 to 10 MW. The aim of the work was achieved by physical modeling of the heat transfer process in the thermosiphon cavity in the range of heat loads from 0.5 to 17 kW/m^2. The study of the internal temperature difference of two-phase gravity thermosiphons at thermal loads up to 17 kW/ m^2 was performed experimentally. The paper shows a scheme of the experimental research stand. The graphical dependence of the temperature difference in the thermosiphon cavity on the heat flux density is presented. The root-mean-square error of experimental results was calculated, being up to 5.7%. The significance of the obtained results lies in that the existing calculation method was improved due to the mathematical dependences obtained for calculations of the internal temperature difference, and became applicable in the calculation of heat exchangers based on two-phase thermosiphons operating in the heat load range of up to 17 kW/m^2. The experiments performed confirm the competitiveness and high thermal efficiency of the two-phase gravitational thermosiphons under the regime conditions typical for the WHB plants. This makes promising the use of the two-phase gravitational thermosiphons compared to traditional coil heating surfaces.

Keywords: two-phase gravitational thermosiphon, waste heat boiler, heat flow density, thermal resistance.
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Studiul perspectivelor utilizării termosifoanelor gravitaționale în două faze în cazanele termice utilizatore ale instalațiilor de cogenerare

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Rezumat. Lipsa datelor experimentale privind funcționarea termosifoanilor în componența cazanelor termice utilizatoare ale centralelor electrice cu cu turbine cu gaz (GTE) cu o capacitate de la 2 până la 10 MW determină relevanța acestei lucrări. Scopul lucrării în obținerea relațiilor pentru calcularea diferenței de temperatură internă a termosifoanilor la sarcini termice de la 0,5 la 17 kW / m^2. Obiectul cercetării îl reprezintă parametrii și caracteristicile procesului de recuperare a căldurii în elementele cazanelor de recuperare a termosifonului, ca parte a circuitelor de recuperare a căldurii din centralele de cogenerare cu gaz-abur (ITGA). A fost realizat experimental studiul diferenței de temperatură internă a termosifoanelor gravitaționale în două faze la sarcini termice de până la 17 kW / m^2. Lucrarea prezintă o diagramă a unui stand de cercetare experimentală. Rezultatele cele mai semnificative ale lucrării se prezintă relațiil de calcul a diferenței interne de temperatură în intervalul de sarcină termică de la 0,5 la 17 kW / m^2. Semnificația rezultatelor obținute constă în faptul, că metoda de calcul existentă a fost îmbunătățită datorită dependențelor matematice obținute ale calculului diferenței de temperatură internă şi a devenit aplicabilă pentru calculul schimbătoarilor de căldură pe baza termosifoanilor cu două faze, care operează în gama de sarcini termice până la 17 kW / m^2. Experimentele confirmă competitivitatea și eficiența termică ridicată a termosifoanelor gravitaționale în două faze în condiții de funcționare, tipice pentru cazanele termice utilizatoare ale instalațiilor de cogenerare, ceea ce le face mai promițătoare în comparație cu suprafețele tradiționale de încălzire de tip spirală.

Cuvinte-cheie: sifon gravitațional bifazic, cazan de căldură utilizator, densitate de flux de căldură, rezistență termică.

Исследование перспектив использования двухфазных гравитационных термосифонов в котлах-утилизаторах когенерационных установок

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Аннотация. Отсутствие экспериментальных данных по работе термосифонов в составе котлов-утилизаторов энергетических установок с газотурбинными двигателями (ГТД) мощностью от 2 до 10

МВт обусловливает актуальность данной работы. Целью работы является получение зависимостей для расчета внутреннего температурного перепада термосифонов при тепловых нагрузках от 0.5 до 17 кВт/м^2. Поставленная задача достигается путем физического моделирования процесса теплообмена в полости термосифона в диапазоне тепловых нагрузок от 0.5 до 17 кВт/м^2. Предметом исследования являются параметры и характеристики процесса утилизации теплоты в элементах термосифонных котлов-утилизаторов в составе теплоутилизационных контуров когенерационных газопаротурбинных установок (ГПТУ). Исследование внутреннего температурного перепада двухфазных гравитационных термосифонов при тепловых нагрузках до 17 кВт/м^2 проводились экспериментальным путем. В работе представлена схема экспериментального исследовательского стенда. Наиболее существенными результатами работы являются полученные зависимости, которые позволяют осуществлять расчет внутреннего температурного перепада в диапазоне тепловых нагрузок от 0.5 до 17 кВт/м^2. Представлена графическая зависимость температурного перепада в полости термосифона от плотности теплового потока. Рассчитана среднеквадратичная погрешность результатов экспериментальных исследований, которая не превышает 5.7 %. Значимость полученных результатов состоит в том, что существующая методика расчета была усовершенствована за счет полученных математических зависимостей расчета внутреннего температурного перепада, и стала применима для расчетов теплообменных аппаратов на базе двухфазных термосифонов, работающих в диапазоне тепловых нагрузок до 17 кВт/м^2. Проведенные опыты подтверждают конкурентоспособность и высокую тепловую эффективность двухфазных гравитационных термосифонов в режимных условиях, характерных для котлов-утилизаторов когенерационных установок, что обусловливает перспективность их использования по сравнению с традиционными змеевиковыми поверхностями нагрева.
Ключевые слова: двухфазный гравитационный термосифон, котел-утилизатор, плотность теплового потока, термическое сопротивление.

Introduction

The development of power engineering based on gas turbine and gas-and-steam turbine technologies provides increasing the thermal power plants efficiency and reducing their negative impact on the environment. Against the backdrop of energy systems decentralization, cogeneration GSTP based on the GTE with capacity from 2 to 10 MW is promising for the application in the power engineering.

An important element of the GSTP heat recovery circuit (HRC) is a WHB. On the one hand the use of the coil heating surfaces and forced circulation in the evaporating circuit of the WHB vertical arrangement provides an increase in the compactness of the HRC, and on the other it reduces the reliability of operation because of the failure of the coils [1-5]. Horizontal waste heat boiler (WHB) design with a natural circulation in the evaporator circuit has a specific quantity of metal structure, as well as greater aerodynamic resistance under equal conditions of design and operation, which decreases the whole plant capacity [6-9]. The thermosiphon heating surfaces with a natural circulation in horizontal WHB can increase the efficiency of heat recovery and the reliability of the whole plant [10-15].

Heat utilization is studied by the leading educational institutions, academic institutes and heat exchange equipment manufacturers, using the heat exchangers based on two-phase thermosiphons. However, the knowledge and recommendations concerning thermosiphon WHB use in gas steam turbine plant design are absent.

The existing methods for thermosiphon WHB calculation have a large number of assumptions, like not to take into account the internal temperature difference in thermosiphons and the rate of natural circulation in the evaporative circuit, that leads to the WHB heat transfer coefficient and power coefficient errors [13-17].

Recent studies describe the two-phase thermosiphons and contain information on their operation as to heat transfer agents, such as, deionized water, the water based on titanium dioxide and gold nanofluids with different concentrations as working fluids [18], water nanofluids based on Al_2O_3 [19], and water with iron oxide nanoparticles [20].

The authors of [18] note that the long-term experiments carried out with 0.3% of nanoparticles indicate a massive aging of the porous layer on nanoparticles on the evaporator surface. This fact makes the practical use of nanofluids inexpedient. Thus, it is advisable to use traditional coolants in high-capacity power plants, which contain thermosiphons in their structure.

In [21], the designs of heat exchangers and the peculiarities of their operation as being part of two-phase thermosiphons under heat load on thermosiphon up to 3 kW/m^2 are considered in sufficient detail.

At present, there are no data on the features of the thermosiphons in the WHB structure for the high-capacity power plants from 2 to 10 MW, where the thermal load value per thermosiphon is above 3 kW/m^2.

Thus, the researches aimed at obtaining the knowledge and regulations on the thermosiphons used in the WHB of the cogeneration GSTP are relevant.

A number of dependences and recommendations for the calculation of internal thermal resistance are given in the works of scientists, who study the internal processes in heat pipes, thermopile and thermosiphons. In particular, in [21] for thermosiphons with natural circulation, the thermal resistance of the steam flow is presented as following:

$$R_{\text{int}} = \frac{h \cdot t_{s.ts} \cdot q_{ts} (\rho_w - \rho_s)}{r \cdot \rho_s \cdot q_F \cdot d_{ts}}, \tag{1}$$

where h is the height difference between the liquid level in the condenser and evaporator, m; q_{ts} is the heat flux density related to the surface area of the heat supply, W/m^2.

However, the calculations using formula (1) give a significant error at thermal loads more than 3 kW/m^2 due to neglecting the hydrodynamic friction of the vapor stream on the condensate film, flowing into the evaporation zone and simplifying the length difference between the evaporation and condensation zones.

In [22], the following dependence for determining the internal heat differential of the actuating medium for the steam thermopile is offered:

$$\Delta t_{ts} = \frac{g \cdot t_{s.ts} (l_{ev} - l_c)}{2r} + 16 \frac{l_{ev}{}^2 \cdot q_F \cdot \mu_s \cdot t_{s.ts}}{d_{ts}^3 \cdot r^2 \cdot \rho_s{}^2}, \tag{2}$$

Equation (2) was obtained for the condition of laminar fluid and vapor motion with thermal loads less than 300 W/m^2 and was recommended for the use heat carriers such as kerosene and freon-12.

Another dependence was recommended in [12] for calculations of the internal thermal resistance of thermosiphons in the range of thermal loads up to 10 kW/m^2:

$$R_{\text{int}} = \frac{q_{ts}^{0.333}}{A^{1.333}}, \tag{3}$$

where A is the coefficient with taking into account geometric parameters of thermosiphon and the physical properties of the heat carrier.

$$A = 0.56 \cdot \pi \cdot d_{ts} l^{0.75} \left[\frac{r_w \cdot \rho_w^2 \cdot \lambda_w^3 \cdot g}{\nu_w} \right]^{0.25} \cdot \left[\frac{2l_{ev}}{l_{ev} + l_c} \left(\frac{l_{ev} + l_c}{l_{ev}} - \frac{l_c}{l_{ev}} \right)^{0.5} \right] \tag{4}$$

None of the above mentioned dependencies can’t give reliable results and therefore cannot be used for thermal calculation of heat exchangers for the cogeneration GSTP waste heat boilers with $q_{\text{ts}} = 0.5\ldots17$ kW/m^2.

In the existing methods of thermosiphon heat exchangers calculation [21, 23], one of the assumptions is to determine the thermal power by the average temperature in the cavity of the thermosiphon, which introduces a significant error.

Problem setting

The main purpose of the research is to obtain dependences for the thermosiphons internal temperature difference calculation at thermal loads up to 17 kW/m^2.

The subject of the research is the parameters and characteristics of the heat recovery processes in the elements of the WHB thermosiphon as part of heat recovery circuits of the cogeneration GSTP.

Results and Discussion

This research is based on the physical modeling method. It was used in experimental studies of the internal temperature difference of two-phase gravity thermosiphons at thermal loads of $q_{\text{ts}} = 0.5\ldots17$ kW/m^2.

The technique of experimental research includes the followings: experiment planning, analysis of the obtained results and error estimation, verification of the results acceptability and their interpretation, presentation of the obtained data in an orderly visual form.

The most important characteristic of experimental studies is the error of direct and indirect measurements. The error of the basic measurements error was evaluated at the planning stage of the experiment. This allowed choosing the most rational technique for conducting the experiment. In the literature [24, 25] the principle of choosing the required number of measurements is given. A total measurement error excludes systematic Δ_s and random error Δ_{re}. For the experimental research of the internal temperature difference of thermosiphons, the condition $2\Delta_s \approx \Delta_{re}$ was fulfilled. The number of measurements taken for each conditionally constant mode was equal 5.

The technique of the experiment and the errors determination are presented in Fig. 1.

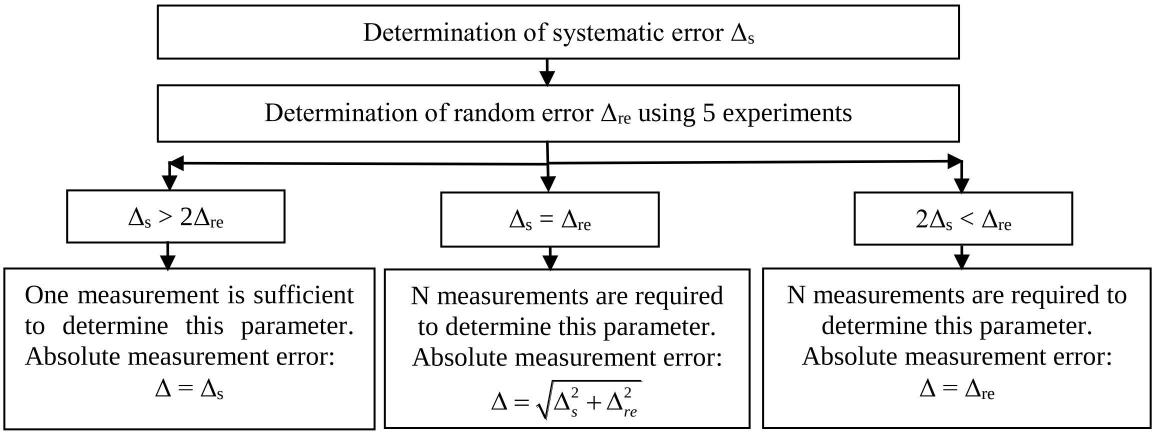


Fig.1. The technique of the experiment and errors determination.

The experimental stand was designed for carrying out the working process in two-phase gravity thermosiphons. This stand is shown in Fig. 2. It allows direct monitoring the processes occurring in all zones of the thermosiphon, as well as simulating the operating conditions, which are typical for the WHB of the GSTP.

The working area of this stand is made of tube of transparent quartz glass with an internal diameter 25 mm and the glass thickness of 1.5 mm. The length of the evaporation zone was 1.0 m and the length of the condensation zone varied from 0.2 to 0.5 m.

A three-way valve with a vacuum manometer for measuring pressure inside the thermosiphon and the Maevskii air-purge valve for vacuuming the thermosiphon were connected to the upper part of the evaporation zone.

The thermosiphon filling-in was carried out with the help of heat carrier through the top valve. The heat carrier removal from the thermosiphon occurred through the valve at the bottom of the thermosiphon.

An important step in the design of the heat transfer systems with thermosiphons is the correct choice of the heat carrier and case material, because it affects the service life and metal consumption of the device as a whole.

The choice of the heat carrier was performed based on the physical properties analysis.

Water is the best heat carrier by many parameters (heat transfer, cost, availability, fire and explosive risk) according to the analysis of thermophysical properties [21, 23].

That is why the distilled water was used as the heat carrier.

During creation of the working area, special attention was paid to the removal of non-condensing gases (air) from the cavity of the thermosiphon.

The incomplete air removal forms additional thermal resistance, which reduces the intensity of heat transfer.

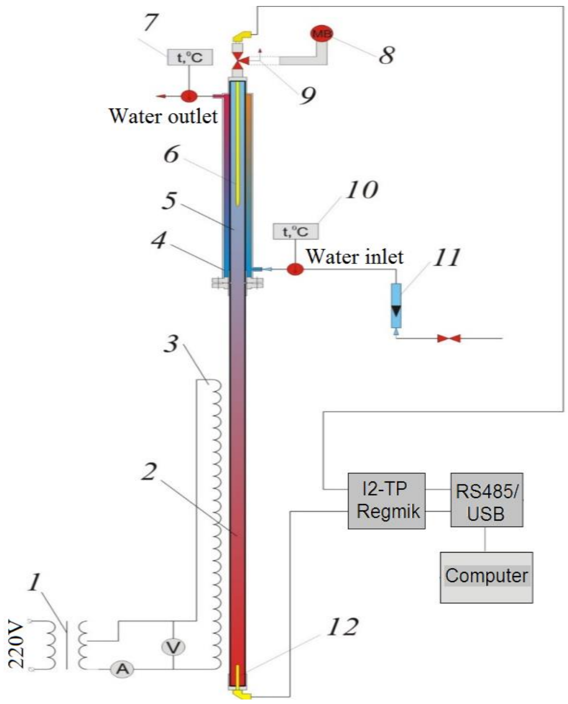
The impact of air increases with its volume and is especially pronounced in the reduced pressure and low thermal loads areas. Vacuuming of the thermosiphon was carried out by evaporation of the heat carrier.

As a result, the air was deflated by the formed steam through the Maevskii air-purge valve.

The heat carrier heating in the thermosiphon was performed using a 0.5-diameter nichrome wire electrical heater.

The wire was wound over the entire height of the evaporation zone with a pitch of 10-15 mm, so it was possible to observe the processes in the pipe.

The heater power was controlled using an autotransformer and measured by ammeter and voltmeter.



1 – autotransformer; 2 – evaporation zone; 3 – heater; 4 – water case; 5 – condensation zone; 6, 12 – thermocouples; 7, 10 – numeral thermometers; 8 – vacuum manometer; 9 – Maevskii air-purge valve; 11 – rotameter.

Fig.2. Scheme of the experimental installation for the thermosiphon analysis.

The condensation zone of the thermosiphon was cooled by water, which was pumped through the circular space between the thermosiphon tube and the water case. The valve was used to control water consumption. It was measured by the PC-5 type rotameter. The water temperature was measured by the digital thermometers with a graduation mark of 0.1 °C at the inlet and outlet of the water case.

The working area was equipped with thermocouples for internal working processes and thermosiphon thermal modes study. The temperature of the heat carrier in the evaporation zone was measured by the THK-002 L100 temperature sensor (chromel-copel thermocouple with a 100-mm working section length), which was set from the lower end of the pipe and sealed by the fitting. Temperature of the heat carrier in the condensation zone was measured in two positions: first, the THK-002 L100 sensor was used to determine the total heat transfer at the thermosiphon. Secondly, the THK-002 L350 sensor was used to determine the average temperature in the thermosiphon condensation zone in three operating positions (at distances 1150, 1250 and 1350 mm from the thermosiphon bottom). The cable terminals of the THK-002 sensors were connected to the I2-TP device designed to receive and convert input signals to the physical values and display the received value on a built-in digital display. The I2-TP device was connected to the computer via the RS485/USB interface converter, where special software made it possible to obtain the information from the sensors, which was viewed and documented [26].

The heat flux density was defined as the ratio of thermal capacity of the thermosiphon to the cross-section area of the evaporation zone. The thermal capacity of the thermosiphon was determined by the cooling water flow rate and change in its temperature. The saturation pressure varied in the range from 0.05 to 0.8 MPa. The cooling water consumption corresponded to the flow regimes in thermosiphon heat exchangers designed to produce the hot water and steam, and varied in the range G_v = (7,8… 14,4)·10^{-3} kg/s. The water temperature at the inlet of the condensation zone remained unchanged and was 12 °C. The degree of evaporation zone filling was ε=35%. The relative length of the thermosiphon under study was *l/d* = 42…53. The length of the evaporation zone was 1 m, and the length of the condensation zone varied from 0.25 to 0.5 m. The internal diameter of the thermosiphon was 0.025 m and the outlet diameter was 0.028 m.

The reliability of the experimental data is ensured by using modern measuring complex "RegMik" I2-TP (the measurement error according to the manufacturer's data is up to $\Delta t_{pr} = \pm\ 0.25\ ^0C$ [26]) and high accuracy of measuring devices.

The accuracy of the indirect measurement result was estimated by the root-mean-square error of the following formula:

$$\delta_f = \pm\sqrt{\sum_{i=1}^{n}\left(\frac{df}{dx_i}\right)^2 \cdot \delta_{xi}^2}\ , \qquad (5)$$

where δ_{xi} is the root-mean-square error of the direct measurements.

The accuracy of the measurement results of the internal temperature difference was estimated by the error:

$$\delta_{\Delta t_{ts}} = \pm\left[\left(\frac{d\Delta t_{ts}}{dt_{ev}}\right)^2 \cdot \Delta t_{ev}^2 + \left(\frac{d\Delta t_{ts}}{dt_c}\right)^2 \cdot \Delta t_c^2\right]^{0.5} \qquad (6)$$

where $\Delta t_v = \Delta t_k = \pm 0.1\ ^0C$ is the absolute errors of thermocouples measurements in the evaporation and condensation zones, respectively; t_v, t_k are the measured values of temperatures in evaporation and condensation zones, respectively, ^0C.

The error values that were determined during the experimental research are shown in table 1.

Visual observations were carried out on the experimental stand in the range of thermal loads $q_{ts} = 0.5…17.0$ kW/m^2, which usually take place during the operation of thermosiphons as part of the WHB at cogeneration plants.

Table 1

Limit values of quantities and its measurement errors

№	Parameter	Marking	Numeric value in modes		Absolute error in modes		Relative error in modes,%	
			min	max	min	max	min	max
1	The temperature of the thermosiphon heat carrier in the condensation zone, °C	t_c	82	99.5	0.1		0.1	0.12
2	The temperature of the thermosiphon heat carrier in the evaporation zone, °C	t_{ev}	87	110	0.1		0.09	0.11
3	Temperature difference in the cavity of the thermosiphon, °C	Δt_{ts}	0.5	4.8	0.023	0.25	4.6	5.7

The pulsation regime of bubble boiling was observed in the thermosiphon evaporation zone at the low heat flow densities up to 3.0 kW/m^2 (Fig.3, a). The fluid level in the thermosiphon is slightly increased due to the appearance of the vapor phase in the liquid column. Bubbles collision and fusion don't occur because bubbles concentration in the boiling liquid was too small at such heat flux densities.

There was a transition to a pulsating boiling mode with increasing thermal power by more than 3.0 kW/m^2 (Fig.3, b). The limited size of the evaporation mirror (cross section of the thermosiphon) is caused by the reason that not all bubbles have time to reach the free surface and collapse.

a b c

Fig.3. Visualization of boiling process in thermosiphon:
a – bubble (non-pulsating) boiling; b – the fusion of steam bubbles; c – emission of coolant

There was a fusion of vapor bubbles into large bubbles in the liquid fluidized bed. The size of these bubbles was comparable to the inner diameter of the thermosiphon. The coolant emission was observed, when the bubbles were destroyed (Fig.3). When the coolant returns the part of the evaporator wall is covered with the evaporating film. The ejection was repeated periodically, so the boiling process became pulsating. Emissions alternated with periods of "silence". The nature of the condensate movement in the evaporation zone did not change with heat load increasing, and the condensate flowed into the fluidized fluid column in the form of separate droplets. After some time, the ejection was repeated, that is, the boiling process became pulsating.

Thus, based on the conducted studies, there are two boiling modes in the evaporation zone of the thermosiphon depending on the thermal load: at $q_{ts} \leq 3.0$ kW/m^2 (the non-pulsating mode of boiling in the liquid column) and at $q_{ts} > 3.0$ kW/m^2 (the pulsating mode of the active bubble boiling in a column of liquid).

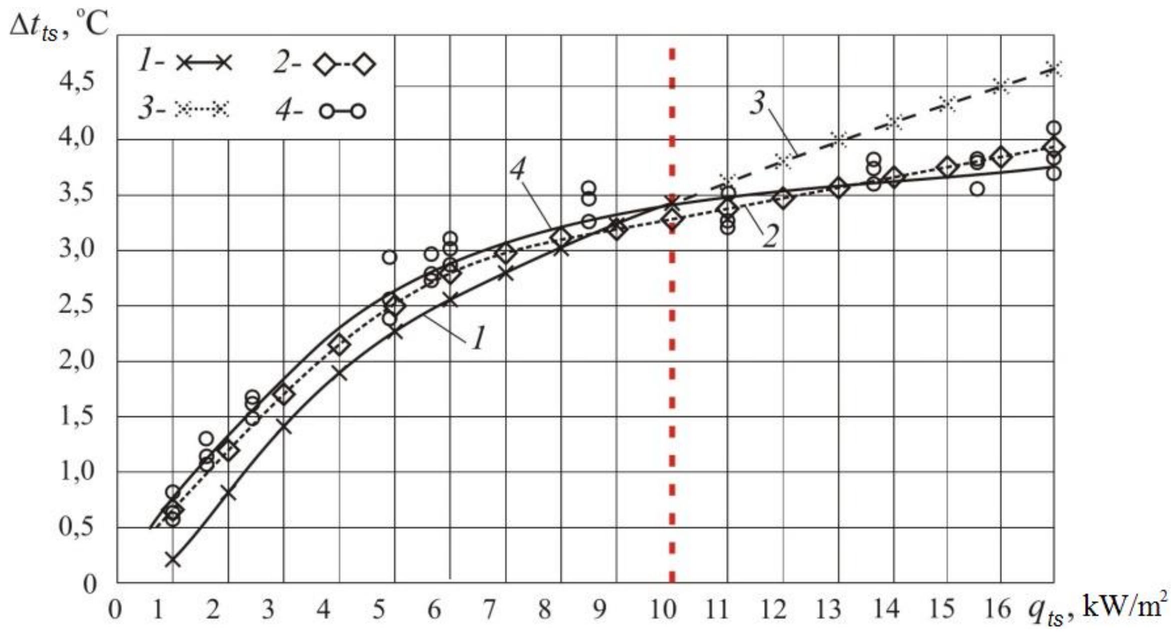
Certain recommendations can be given concerning the evaporation zone filling degree, taking into account the results of the research and qualitative picture of the processes in the evaporation zone. The coolant release into the condensation zone is found undesirable, because it reduces the intensity of heat exchange due to the film thickness increase. The degree of evaporator filling in the range from 20% to 35% is advisable at thermal loads of up to 3.5 kW/m^2 and from 15% to 20% at thermal loads over 3,5 kW/m^2.

The results of the studies showed, that the dependence (3) in the range of up to 10 kW/m^2 gives the difference with the experimental data up to 50%. The maximum error reached 22%, at the thermal load above 10 kW/m^2.

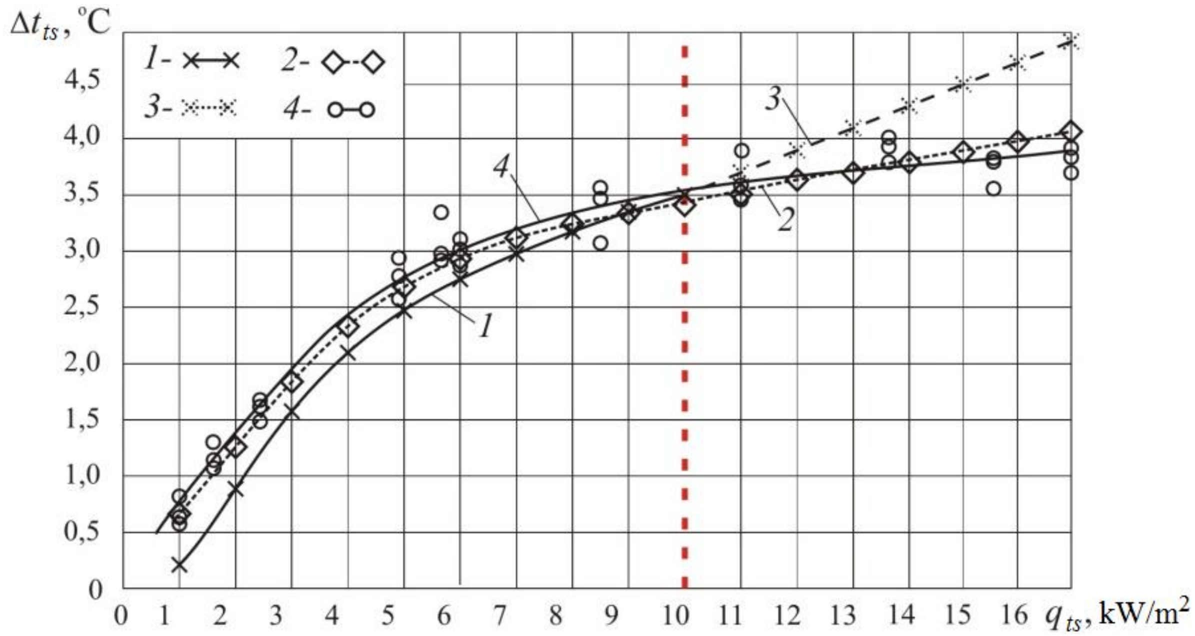
The dependence for the calculation of the internal temperature difference in the cavity of thermosiphons was obtained based on physical modeling of the internal processes in the thermosiphons. Using the basic parameters and considering the degree of their influence on the internal temperature difference, introducing the coefficients of temperature dependence on pressure in a given range of thermal loads, the following dependence was obtained:

$$\begin{cases} \Delta p_{ts} = \dfrac{\left(\xi_{hf} + \xi_{ha}\right)\cdot \omega^2}{2\upsilon''} \\ \omega = \dfrac{G''\upsilon''}{F} \\ G'' = \dfrac{Q_{rs}}{r} \\ t_{ts} = \dfrac{q_{ts}^{1.13}\left(\dfrac{\lambda_{fr}\cdot t_{s.ts}}{d_{ts}} + \xi_{ha}\right)}{2r} \end{cases} \tag{7}$$

where ξ_{hf} is the coefficient of hydrodynamic friction; ξ_{ha} is the hydrodynamic acceleration coefficient.



a



b

1 - the results of calculations according to the formula (3) at $q_{ts} < 10$ kW/m^2, *2* - the results of calculations according to the formula (7); *3* - the results of calculations according to the formula (3) at $10 < q_{ts} < 17$ kW/m^2; *4* - the experimental results.

Fig.4. Dependence of the temperature difference in cavities of the thermosiphon on heat flux density: a – l_{ts} = 1.2 m; b – l_{ts} = 1.5 m

The limits of using the obtained dependences (7):

1) $0,5 < q_{ts} < 17$ kW/m^2;
2) intermediate coolant - water;
3) $42 < l/d_{ts} < 53$.

The results of the studies are presented in Fig. 4 as the dependence of the temperature difference in the cavity of the thermosiphon on the density of the heat flux transmitting by thermosiphon. The analysis of the results showed that in the studied range of values of the heat flux densities, the internal temperature difference increases with the increase in the value of the heat flux density.

Comparison of the experiment results and the those of calculation by the dependence (7) for thermal loads of up to 10 kW/m^2 showed a slight discrepancy up 0.45 ºC. This can be explained by the imperfection of thermosiphon vacuumizing by the evaporation method and the insignificant residue of non-condensing gases in the thermosiphon, which affect the internal temperature difference within the limits of low pressures and thermal loads.

After the analysis, it can be concluded that dependence (7) can be recommended for improving the method of calculation of thermosiphon heat exchangers operating in this range of thermal loads up to 17 kW/m^2.

The experiments have shown that in the studied range of thermal loads the difference in the saturation temperatures of the coolant in the lower part of the evaporation zone and the average saturation temperatures over the condensation zone are up to 4.1 °C (Fig.4). It was established that an increase in the transmitted heat flux density from 0.5 to 17.0 kW/m^2 increases the temperature difference at the height of the thermosiphon from 0.5 to 4.1 °C, which is 5% less than the average calculated value of the saturation temperature.

Conclusions

The conducted experiments confirmed the high thermal efficiency of the two-phase gravity thermosiphons under the mode conditions, which are typical for the WHB of the cogeneration plants. The obtained experimental data on the internal temperature difference confirm the competitiveness of thermosiphon heating surfaces compared to the traditional coil and allow their using in the WHB designs.

On the basis of the conducted researches, in order to prevent the release of the coolant into the condensation zone, it is advisable to keep the degree of the evaporator filling in the range from 20% to 35% at thermal loads up to 3.5 kW/m^2

and from 15% to 20% at thermal loads more than 3.5 kW/m^2.

As a result of the experimental studies of the internal temperature difference of the two-phase gravity thermosiphons by approximation, an analytical dependence was obtained for the calculation of the internal temperature difference of thermosiphons Δt_{ts} = f (q_{ts}), in the range of 0,5 ≤ q_{ts} <17 kW/m^2 and 42 < l/d_{ts} < 53 using water as an intermediate coolant. The accuracy of the experimental results was estimated by the root-mean-square error, which is up to 5.7%.

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