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An improved methodology for thermal calculation and design optimization of a thermosiphon waste heat boiler for CCPP☆

Iurii Dolganov[a,*], Alexander Epifanov[b], Pavel Patsurkovskyi[b], Bohdan Lychko[b]

[a] Faculty of Science, Technology and Medicine, Department of Engineering, University of Luxembourg, Luxembourg
[b] Department of Technical Thermal Physics and Steam-Generating Units, Admiral Makarov National University of Shipbuilding, Mykolaiv, Ukraine

HIGHLIGHTS

- Thermosyphon waste heat boiler is a reliable and easily manufacturable device.
- Ignoring the thermosiphon temperature gradient overestimates the boiler power.
- The circulation rate in the evaporator determines the boiling HTC.
- A modified calculation method refines the boiler thermal output up to 6 %.

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ABSTRACT

The study is dedicated to enhancing design and computational methodologies for thermosiphon waste heat boilers (WHB) within combined cycle power plants (CCPP). The optimized WHB design and an improved thermal calculation approach are shown. The efficiency of the proposed WHB has been improved by optimizing the area and arrangement of the surface. The novel method incorporates considerations for the internal thermal resistance within thermosyphons and the graded velocity of natural circulation within the evaporation circuit. To refine the calculation method, empirical and experimental data concerning internal temperature gradient in thermosiphon functioning within a heat load spectrum of up to 17 kW/m^2, were used. Important findings include three categories. Firstly, increasing the number of sections will slightly increase capital costs and decrease WHB gas temperature, which could be neglected by the unification of thermosiphon sections. Secondly, by applying this modified methodology to a power plant facility featuring a 6700 kW gas turbine engine (GTE), notable adjustments in thermal power were realized, amounting to 157 kW (approximately 6 %), along with corresponding electrical power adjustments of 149 kW (approximately 2 %). Third, the new method is limited to a GTE power of up to 10 MW due to the experimental data used for validation.

1. Introduction

The advancement of the energy sector centered around the gas turbine and combined gas-steam turbine technologies is instrumental in augmenting the efficiency of thermal power plants while concurrently mitigating their environmental footprint [1]. Moreover, with the prevailing global trend towards decentralization of energy systems, gas-steam turbine units with gas turbines ranging from 2

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* Corresponding author.
E-mail address: iurii.dolganov@uni.lu (I. Dolganov).

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to 10 MW in capacity have emerged as a promising option for implementation. These units can be used in a range of applications, including distributed generation, combined heat and power, and industrial processes. The use of gas-steam turbine units has several advantages over traditional power generation methods. For example, they offer higher efficiency, lower emissions, and increased flexibility in terms of fuel options, which can help reduce costs and improve performance. Additionally, the smaller size of these units allows for more localized power generation, reducing the need for long-distance transmission lines and improving grid stability [2].

An important component of the heat recovery system of a combined cycle power plant (CCPP) is the waste heat boiler (WHB). The selection and justification of the appropriate boiler type for each specific installation are fraught with various challenges [3]. The incorporation of water-tube heating surfaces and forced circulation within the evaporator circuit of vertically oriented WHBs enhances the compactness of the heat recovery system. However, this approach compromises operational reliability due to tube failures, resulting in diminished electrical and thermal output of the unit [4]. Horizontally oriented WHBs that have natural circulation in the evaporator circuit require a higher initial investment and result in greater aerodynamic resistance compared to other designs. This ultimately reduces the power output of the GTE and the overall installation. To optimize the heat generated by burning fuel in a CCPP, it is necessary to design the steam-generating system considering operating conditions. The design and performance evaluation process involves selecting several design factors, each of which independently affects the conversion process (for example, steam outlet temperature and superheater pressure) [5]. Some studies have focused on finding possible locations for waste heat recovery inside a power plant using thermodynamic and exergy analysis, which in turn allows for more efficient use of energy resources [6].

The use of such an approach allows for better heat transfer between the heating surfaces and the working fluid, resulting in improved energy conversion and reduced risk of equipment failure. Overall, this technique can help to optimize the performance of CCPPs, making them a more cost-effective and sustainable source of energy [7].

Leading academic institutes and organizations around the world are extensively researching the use of two-phase thermosiphon heat exchangers for heat recovery in power plants, in the chemical and petrochemical industry. However, there is a lack of both theoretical and empirical data regarding their application in CCPP, which prevents practical implementation recommendations. Therefore, the proposed case study aims to gain new insights into the use of thermosiphon WHBs in CCPPs and improve their heat recovery efficiency. This study is highly relevant and significant for practical purposes.

Heat transfer devices - heat pipes, and their most effective type - two-phase gravity thermosyphons, have been widely used in various fields of technology: industrial energy reuse [8], geothermal district heating systems [9], solar water heating systems [10], innovative energy storage [11], pillow plate thermosiphon reboilers [12]. A distinctive aspect of thermosyphon operation within WHBs lies in the absence of an adiabatic zone (Fig. 1).

This deviation arises because the heating region is entirely situated within the gas duct, whereas the condensation region is within the cooling zone, and the thermosiphon attachment point in the tube plate is negligible. This particular feature, along with the considerable length of the thermosiphon, which is over 1 m, results in the internal thermal resistance having a significant impact on the heat transfer coefficient. However, current calculation methods do not consider the internal temperature gradient of the thermosiphon [13]. Failure to consider the internal temperature gradient may result in an inaccurate calculation of the heat transfer coefficient, leading to suboptimal performance of the thermosyphons in WHBs. By accounting for the internal temperature gradient, the efficiency of these heat transfer devices can be significantly improved, resulting in better performance and energy savings.

According to Ref. [13], the most optimal heat utilization scheme using heat recovery boilers involves the obtaining of steam with energy parameters. In this scenario, the heat energy from waste gases is efficiently harnessed, being converted into electrical or mechanical energy while concurrently generating steam with specified technological parameters or hot water for industrial, communal, and economic requirements.

The study [14] introduces a new explosion-proof construction for a waste heat boiler characterized by an extended operational lifespan, ensuring consistent benefits through the incorporation of a dual-circuit heat conducting system. The study [15] describes an air-gas heat pipe heat exchanger for waste heat recovery in the ceramics industry. This system leverages closed two-phase thermosiphons within the primary circuit, interfacing with the heated gas flow. The works [16,17] present thermosyphon heat exchangers for air-water heat recovery systems.

Different designs of thermosiphon heat recovery boilers have been developed by the Department of Thermal Physics, National University of Shipbuilding, Ukraine (Fig. 2) [18,19].

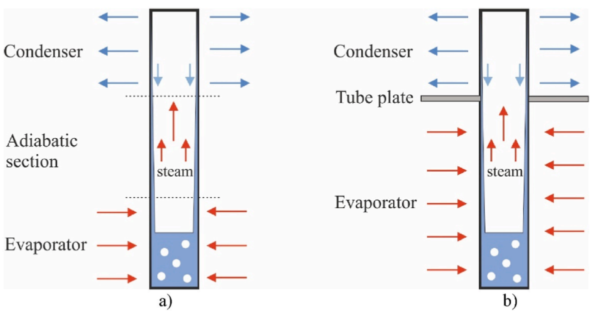

Fig. 1. Thermosiphon operating principles: a) general case; b) heating surface of WHB.

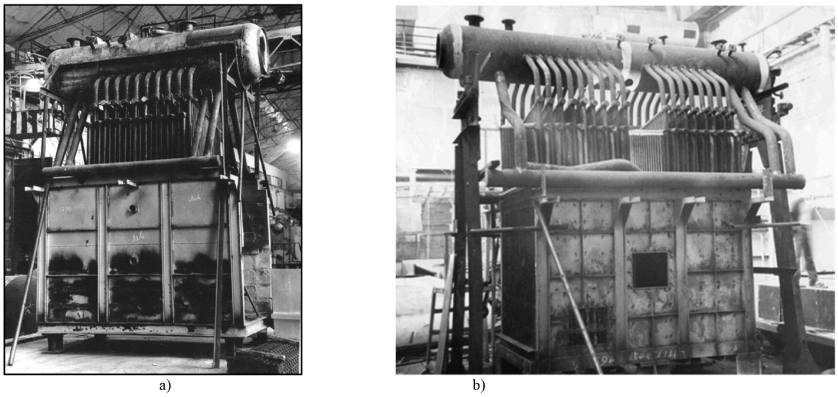

Fig. 2. General view of thermosiphon sectional WHBs. Development of the Department of Thermal Physics, National University of Shipbuilding: a) – thermosiphon steam generator for waste-fire plant [18]; b) - thermosiphon steam generator for glass furnace [19].

Splitting the evaporation circuit into sections helps improve heat transfer and reduce the size and weight of the boiler. However, this can result in varying thermosyphon lengths due to uneven heat absorption across the thermosyphons in each section. An alternative approach involves placing sections of thermosyphons perpendicular to the gas duct, with vertical pipes having decreasing lengths corresponding to the condensation sections of the thermosyphons. This design strategy reduces the metal consumption of the boiler. Nevertheless, it is worth noting that this design prevents the standardization of heating surface elements, as each subsequent row along the gas flow features shorter condensation sections.

Based on a literature review and patent search, as well as an analysis of the aforementioned design limitations, our study highlights the problems associated with engineering calculations and determination of the thermal power of the sections and the reliability of the evaporator circulation.

- unification of thermosiphon sections can lessen the impact of increasing heating surface on capital costs.
- neglecting the internal thermal resistance of thermosiphons and averaging the circulation rate in the evaporator can lead to incorrect assessment of heat transfer coefficients in the boiler.
- developing a new algorithm requires experimental data on the internal temperature gradient in thermosiphons within the corresponding GTE power range. This will impose restrictions on the use of the new method.

2. Design optimization

The analysis of thermosiphon heat exchanger designs yields the following conclusions.

- Using thermosyphons as the heating surface in WHBs creates a reliable, durable, and easily manufacturable device characterized by high maintainability.
- Thermosyphons function independently. Failure of one or more does not shut down the device.
- The utilization of an intermediate heat transfer medium allows for an elevation in the thermosiphon wall temperature and a reduction in the extent of corrosive degradation, primarily attributed to low-temperature corrosion.
- The installation of thermosyphons within the tube plate using detachable connections facilitates straightforward replacement.
- Cantilever mounting of thermosyphons at a single point eliminates thermal stresses caused by temperature-induced expansion, unlike conventional heat exchangers with two tube sheets.

Based on analysis of previous experience and prototypes, the authors have developed a WHB design (Fig. 3). The distinguishing characteristic of this development lies in the incorporation of an economizer, a gas condensate heater, and a gas water heater, all positioned along the gas flow. These thermosyphons are interconnected to form a parallel system, enabling the additional provision of hot water for district heating purposes. Detailed design specifications are elucidated in Ref. [20].

In contrast to water tube WHBs and preceding prototypes of thermosiphon WHBs, the utilization of thermosyphons amalgamated into integrated sections yields a waste heat recovery boiler imbued with the following advantages.

- Thermosyphons operate autonomously, preventing complete boiler failure due to partial malfunction caused by wall deterioration.
- Redesigned boiler-utilizer eliminates pipe bends, reducing complexity and increasing reliability.
- Uniform heating surface sections simplify boiler design and maintenance while reducing costs.
- Thermosyphon cantilever installation reduces pipe stress from temperature changes and aids in removing contaminants via vibration cleaning in heating zones.
- The absence of a circulation pump simplifies both installation and operational aspects.

Based on these advantages, the developed WHB is recommended for the utilization of waste heat from low-capacity gas turbine engines.

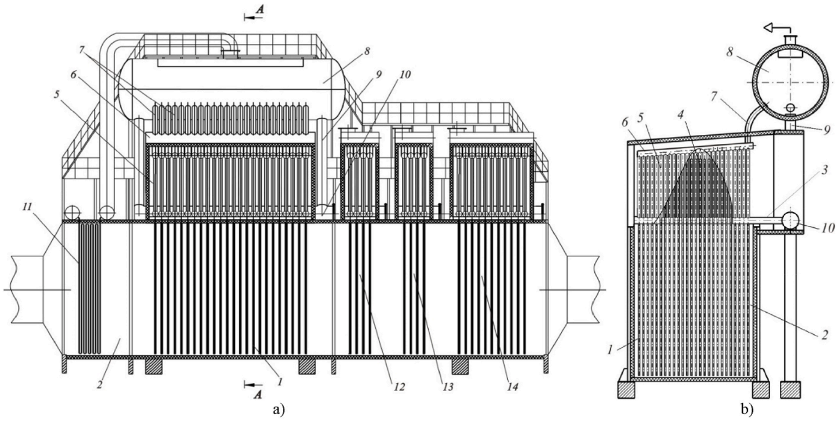

Fig. 3. Sectional thermosiphon WHB for a CCPP (not to scale) [20]: a) longitudinal section; b) cross-section; 1 – evaporation zone of thermosiphons; 2 – gas duct; 3 – feeder pipes; 4 – condensation zone of thermosiphons; 5 – evaporator riser pipes; 6 – steam collection pipes; 7 – steam outlet pipes; 8 – steam drum; 9 – downcomer pipes; 10 – water collector; 11 – coil superheater; 12 – thermosiphon economizer; 13 – thermosiphon gas condensate heater; 14 – thermosiphon water heater.

3. Improvement of the thermal calculation methodology

This section presents the main aspects of the new calculating methodology for thermosiphon WHB. Detailed mathematical models and experimental results that became the basis of this method are presented in our previous scientific publications, links to which are indicated in the appropriate places.

3.1. Thermosiphon internal temperature gradient

To improve heat exchange calculations in thermosiphon heating surfaces of a sectional WHB, it's necessary to derive accurate relationships for determining the internal temperature difference within two-phase gravity thermosyphons. Well-known methods rely on estimating thermal power based on the cavity's average temperature. The saturation temperature in a thermosyphon is calculated without considering internal thermal resistance, which is especially significant for thermosyphons longer than 1 m, introducing some degree of error [13].

The equation for computing the internal temperature difference within the thermosyphon cavity was derived through a combination of physical modeling of internal thermosyphon processes and subsequent experimental verification [21]. By considering the principal parameters and their impact on the internal temperature difference, along with the introduction of coefficients that account for temperature-pressure relationships within a specified range of thermal loads, the following relationship was established:

$$\begin{cases} \Delta P_{ts} = \dfrac{\left(\xi_{hf} + \xi_{ha}\right) W^2}{2 \upsilon_{st}} \\ W = \dfrac{G_{st} \upsilon_{st}}{F} \\ G_{st} = \dfrac{Q_{ts}}{r} \\ \Delta T_{ts} = \dfrac{{q_{ts}}^{1.13} \left(\frac{\lambda_f T_{s.ts}}{d_{ts}} + \xi_{ha}\right)}{2r} \end{cases} \tag{1}$$

3.2. Natural circulation velocity in evaporation circuit

The primary objective of hydraulic calculation for the evaporating surface is to determine the circulation velocity in the evaporator riser pipes and evaluate the circulation's reliability. However, the traditional approach is not appropriate for circulation system configurations that include thermosyphon sections. The flow rate varies not only between different sections due to the decreasing temperature of flue gases but also as these gases move away from the steam drum. We have modified the method for calculating the boiler evaporator using the analytical model data obtained in Ref. [22].

$$W_a = W_{circ.min}, W_b = W_{circ.max} \tag{2}$$

$$f\left(W_{circ}\right) = S_{circ}\left(W_{circ}\right) - \Delta P_{st}\left(W_{circ}\right) \tag{3}$$

$$W'_{circ} = root\left(f\left(W_{circ}\right), W_{circ}, W_a, W_b\right) \tag{4}$$

The model developed is different from traditional approaches as it includes a ranked variable that allows for the calculation of circulation rates not only for boiler sections but also for individual riser pipes. This graded variable velocity helps to determine the hydraulic properties of the evaporator tubes and the steam content inside the heating zone. As a result, the testing process for boiling and circulation problems has been made more efficient. In addition, the mass flow rates were quantified concerning specific boiler sections.

3.3. Algorithm of the modified calculation methodology

Thermal calculation determines the standardized section count and heating surface parameters of a sectional thermosyphon WHB for optimal steam production. The algorithm has three stages: pre-calculation, hydraulic calculation, and refinement calculation. Fig. 4 presents a detailed algorithm for thermal calculation.

During the Precalculation stage, the heat balance, saturation temperature in the thermosyphons, and geometry of the section are determined. Once this stage is complete, the pipes' diameters, pitches, and construction are adopted, and the calculation of the evaporating surface sections is performed sequentially according to the gas flow. Starting with the first section, a verification calculation is

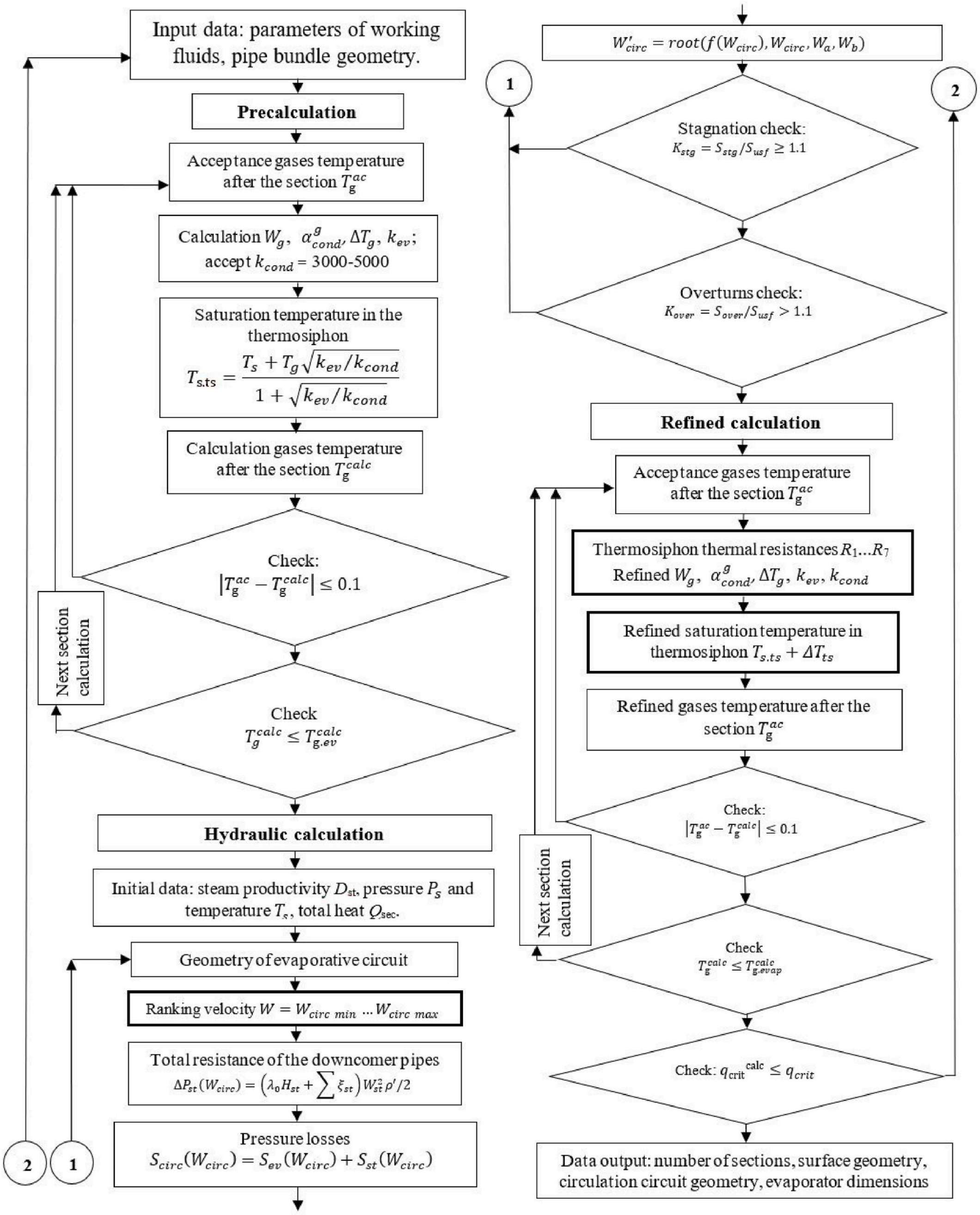

Fig. 4. Algorithm for the thermal calculation of a sectional thermosiphon WHB.

carried out based on the gases' temperature in each section. The difference between the section heat power from the heat balance and heat transfer equations should not exceed 1 %.

$$D_{stj,i}\left(h_{st\ i}-h_{ec\ j}\right)-Q_{st\ j,i}=k_{ev}F_{ev}\Delta T_g \tag{5}$$

To calculate the average temperature of the intermediate heat carrier, it is recommended to use a formula based on the minimum total surface area of the heating and condensation zones of thermosiphons [13].

$$T_{\mathrm{s.ts}}=\frac{T_s+T_g\sqrt{k_{ev}/k_{cond}}}{1+\sqrt{k_{ev}/k_{cond}}} \tag{6}$$

The Precalculation stage concludes when the temperature difference between the gases behind the boiler and the intermediate coolant in the thermosyphons of the last section matches the accepted temperature pinch point at the cold side.

The Hydraulic Calculation stage yields the circulation rates for both the sections of the boiler and the individual riser pipes. These circulation velocities help us determine the heat transfer coefficients in the riser tubes $\alpha_{conv.ev}$, $\alpha_{boil.ev}$ of each section and in the heating and condensation zones of thermosiphons [13].

$$\alpha_{conv.ev}=\frac{Nu\lambda_w}{d} \tag{7}$$

$$Nu=0.021Re^{0.8}\ Pr^{0.43}\left(\frac{Pr}{Pr_w}\right)^{0,25}\varepsilon_1 \tag{8}$$

$$Re=\frac{W_{circ}d_{ev}}{\upsilon} \tag{9}$$

$$\alpha_{boil.ev}=38,7P_{ts}^{0,5}\left[\left(\frac{T_{s.ts}^{ref}-T_{s.ev}}{2}\right)-T_{s.ev}\right] \tag{10}$$

The heat transfer coefficient in the heating zone of thermosyphons is calculated according to the [13], W/(m^2·K),

$$\alpha_{boil.ev}=2,35\alpha_{pool}\left(\frac{P_s}{P_{crit}}\right)^{0,14} \tag{11}$$

Pool-boiling heat transfer coefficient by Labuntsov correlation [23].

$$\alpha_{pool}=0.075\left[1+10\left(\frac{\rho_{st}}{\rho-\rho_{st}}\right)^{0.67}\right]\left(\frac{\lambda^2}{\sigma\upsilon T_{sat}}\right)^{0.33}q_F^{\ 0.67} \tag{12}$$

or

$$Nu=0,125Re^{0.65}\ Pr^{0.33} \tag{13}$$

$$Nu=2,63^{-2}\left(\frac{\lambda\Delta T}{r\rho_{st}\upsilon}\right)^{1.86}\bullet Pr^{0.952} \tag{14}$$

An internal temperature gradient is added in thermosiphons at the Refinement calculation stage, which directly affects the heat transfer coefficient in the evaporator. The refined saturation temperature in the thermosiphon is:

$$T_{s.ts}^{ref}=T_{s.ts}+\Delta T_{ts} \tag{15}$$

The heat transfer coefficient for condensation of the intermediate heat carrier is determined by formula (16). The wall temperature is set as the average of the temperatures of the intermediate heat carrier $t_{s.ts}^{ref}$ at the reference point and the boiling temperature in the riser tubes $t_{s.drum}$ [21].

$$\alpha_{cond}=0.943\frac{\sqrt[4]{\frac{\lambda^3 rg(\rho-\rho_{st})}{\upsilon}}}{\sqrt[4]{L_{cond}\left(T_{s.ts}^{ref}-T_{s.ev}\right)}} \tag{16}$$

The result of the calculation is the number of sections, surface geometry, circulation circuit geometry, and evaporator dimensions. The obtained data can be used for design calculations of the WHB.

Table 1
Initial data.

GTE Power, kW,	6700
GTE specific fuel consumption, $m^3/(kWh)$,	0.317
GTE combustion gases volume, m^3/m^3,	41.23
WHB optimum steam pressure, MPa	1.5
Saturation temperature ($t_{s.drum}$ = 1.65 MPa) °C,	203
Enthalpy of gases behind the superheater, kJ/m^3	21880
Gas temperature behind the superheater, °C,	391
The temperature of the gases behind the evaporator, °C	223
Pipe bundle geometry	
Type of pipe bundle and structure	finned/struggled
Type of ribbing	spiral tape
The outer diameter of thermosiphon pipes, m	0.032
Pipe thickness, m	0.003
Rib height, m	0.014
Rib step, m	0.005
Rib thickness, m	0.001
Height of the gas duct, m	2.0
Width of gas duct, m	2.5
Transverse pitch of pipes, m	0.08
Longitudinal pipe step, m	0.06

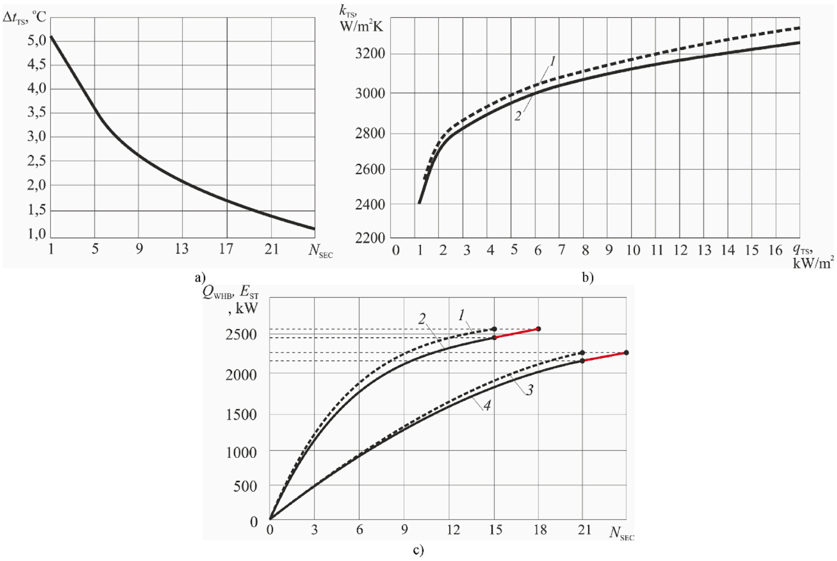

Fig. 5. Calculation results a) the internal temperature gradient of thermosiphons by sections; b) total heat transfer coefficient as a function of the heat flux density: 1 - old calculation method; 2 - improved calculation method; c) the influence of the internal temperature gradient of thermosiphons on thermal power of the WHB Q_{WHB} and electric power of steam turbine E_{ST}: 1 – Q_{WHB} by old calculation method; 2 – Q_{WHB} by improved calculation method; 3 – E_{ST} by old calculation method; 4 – E_{ST} by improved calculation method.

4. Results

The improved calculation method was verified using data from a CCPP with a UGT6000 gas turbine engine [24]. The initial data are shown in Table 1. When determining the thermal power of the WHB sections, the temperature gradient inside thermosiphons is taken into account (Fig. 5a). The internal temperature gradient, which is calculated using formula (1), is added to the saturation temperature of the thermosyphon. This results in a reduction of the total heat transfer coefficient (Fig. 5b). The thermal calculation method that does not include the internal thermal resistance of the thermosiphon gives underestimated values for the area of the heating surfaces and the number of sections. As a result, it overestimates the useful thermal power of the boiler and the electrical power of the steam turbine (Fig. 5c).

Table 2 displays the output data from the program. The improved method of thermal calculation revealed that there is a 2 % and 6 % lack of electrical and thermal energy respectively due to overestimation of the heat transfer coefficients in the heating surfaces. By introducing in the calculation method for the CCPP with a capacity of GTE 6700 kW, we were able to increase the thermal power by 157 kW (approximately 6 % of the total) and the electric power by 149 kW (approximately 2 % of the total) due to the increase of the corresponding heating surfaces. The necessary electric power of the steam turbine can be achieved by installing 24 sections of thermosyphons instead of 21, and the required thermal power is achieved by installing 18 sections of thermosyphons instead of 15.

Table 2
The calculation results of the evaporative heating surface of the thermosiphon WHB for a GTE 6700 kW.

Flue gas mass flow, kg/s	30,5
Steam pressure in the separator, MPa	1,65
Steam productivity, t/h	10,1
Temperature of gases at the inlet to the evaporator, °C	391
Temperature of gases at the outlet of the evaporator, °C	223
Heating surface area, m^2	1365
Number of sections	23
The number of thermosyphons in the section	32
Diameter of thermosyphons, m	0,032
The length of the heating zone, m	1,9
The length of the condensation zone, m	0,55
Diameter of the riser pipe, m	0.063
Height of the riser pipe, м	0.6
Diameter of the downcomer pipe, m	0.15
Height of the downcomer pipe, m	1.0
Diameter of the steam outlet pipe, m	0.1
Height of the steam outlet pipe, m	0.5
Diameter of the steam outlet collection pipe, m	0.1
Inclination of the steam outlet collection pipe, °	5

Obviously, an increase in the number of sections leads to an increase in capital costs, however, a unified thermosyphon design minimizes these costs concerning their impact.

5. Conclusion

The developed sectional thermosyphon WHB for use in CCPP represents a promising avenue for enhancing energy efficiency and environmental sustainability in the power generation sector. A novel approach, considering internal temperature gradients of thermosyphons and natural circulation velocities in the evaporator, has refined heat transfer coefficients and increased thermal and electrical power.

The study reveals several significant findings.

- Following the inclusion of the internal temperature gradient of thermosiphons in the calculation methodology, for the CCPP featuring a GTE with a capacity of 6700 kW, it became feasible to achieve a boost in thermal output by 157 kW (roughly equivalent to 6 % of the overall total) and electrical output by 149 kW (approximately 2 % of the total), attributed to the expansion of the relevant heat exchange surfaces. Obviously, for more powerful power plants the effect will be more significant, since the length of the thermosiphon, and accordingly, the internal temperature gradient will increase.
- The new method is only applicable for GTE power up to 10 MW due to the limited experimental and numerical data used for validation. The formula for the internal temperature difference was obtained within the thermal load range of the thermosyphon up to 17 kW/m2. Furthermore, numerical modeling of the circulation rate was carried out within the same limits. To expand the range of this method, a series of experimental studies of the thermal characteristics of thermosiphons are required, which also includes increasing their length for more powerful WHBs.
- An increase in the number of thermosiphon sections leads to an increase in capital costs. This influence can be reduced through the proposed design optimization and unification of thermosiphon sections.

While this study has provided promising results and a foundation for further research and practical implementation, it is essential to recognize the ongoing need for innovation and validation in the field of thermosyphon. Continued exploration of these technologies holds the potential to not only increase energy efficiency but also reduce environmental impacts, strengthen energy security, and contribute to the development of sustainable energy solutions.

The method's development has significant implications for the efficiency and competitiveness of thermosyphon heat recovery systems, not just in CCPP, but also in other sectors of the energy industry. We assume that this boiler design, along with the calculation methodology, is highly flexible and can be used without significant modifications for different waste heat sources, as long as they fall within the specified limits.

In summary, this research underscores the importance of advancing heat recovery technology and highlights the potential benefits of sectional thermosyphon WHB in improving the overall efficiency and competitiveness of energy systems.

CRediT authorship contribution statement

Iurii Dolganov: Writing – review & editing, Writing – original draft, Software, Investigation, Conceptualization. **Alexander Epifanov:** Supervision, Project administration, Methodology. **Pavel Patsurkovskyi:** Validation, Resources. **Bohdan Lychko:** Formal analysis, Data curation.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

Nomenclature

D	steam productivity, kg/s
d	diameter, mm
E	electrical power, W
F	surface area, m^2
G	mass flow, kg/s
h	enthalpy, J/m^3
g	gravity acceleration, m/s^2
L	length, m
N	number of sections
P	pressure, Pa
Q, q	total heat, W; heat flux, W/m^2
r	specific heat of vaporization, J/kg
S	pressure loss, Pa
T	temperature, K
W	velocity, m/s

Greek symbols

α	heat transfer coefficient, W/m^2K
λ	thermal conductivity, W/m·K
ν	cinematic viscosity, m^2/s
ρ	density, kg/m^3
σ	surface tension, $N \cdot m^{-1}$
ζ	pressure loss coefficient

Subscripts and superscripts

ac	accepted
boil	boiling
crit	critical
calc	calculated
circ	circulation
cond	condensation
conv	convection
ec	economizer
ev	evaporator
g	flue gases
ha	hydrodynamic acceleration
hf	hydrodynamic friction
pool	nucleate pool-boiling
ref	refined
s	saturation
sec	section
st	steam
ts	thermosiphon
w	wall

Abbreviations

CCPP	Combined cycles power plant
WHB	Waste heat boiler

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