STUDY OF INTEGRAL CHARACTERISTICS AND EFFICIENCY OF A HEAT EXCHANGER OF THERMOSYPHON TYPE WITH FINNED TUBES

by

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The experimental research aims at the analysis of the thermal performance of a gas-liquid heat exchanger in a pilot plant. Results of the conducted experiment with a finned tubes thermosyphon heat exchanger using natural gas are presented. The installation was mounted at the exit of a flue gas from an existing steam generator "PK-4" with total power of 2.88 MW in the boiler room of Vini, Sliven, Bulgaria. Different experiments were carried out at different loads of the steam generator in order to determine the efficiency of the heat exchanger. Based on these results the coefficient of heat transfer of flue gas to the finned tubes was determined, based on different modes of operation with crossed and straight pipe bundles. The effectiveness-number of transfer units method was used.

Key words: *ε-NTU method, finned tubes thermosyphon, heat transfer, waste heat recovery*

Introduction

Fossil fuel combustion emits large amounts of flue gas and results in considerable heat release, which can be utilized. A number of different heat exchangers were developed for transferring the waste heat to other fluids or support other burning processes. One such heat exchanger is the thermosyphon heat exchanger with finned tubes from the side of flue gas. It has a simple design and is quite compact compared to other heat exchanges.

Many scientists have been working on these heat pipe heat exchangers (HPHE). Some experimental examples are given.

The two main methods used for design and experimental investigation of HPHE are: (1) the log-mean temperature difference model (LMTD), and (2) the effectiveness-number of transfer units model (ε-NTU) [1, 2]. The ε-NTU method is preferred for calculating the HPHE performance. Azad and Geoola [3] applied the ε-NTU model on air-to-air heat pipe heat exchanger. They discovered a new correlation for condensing water vapour on vertical pipes and determined that the external thermal resistances in those cases limit the performance of HPHE. Liu *et al.* [4, 5] investigated heat transfer characteristics of HPHE with latent heat storage and suggested a new thermal storage system and a heat pipe heat exchanger with la-

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tent heat storage. The new system can operate in three basic operation modes: the charging only, the discharging only, and the simultaneous charging/discharging modes. The main factors affecting the performance and determine the operation mode of the system were also identified. The authors presented broad experimental results for these three modes and also effects of the inlet temperature and the flow rate of the cold/hot water.

Shah and Giovannelli [6] analysed heat pipe heat exchanger performance by modelling a HPHE using both LMTD and the ε-NTU method. Many existing correlations and results were used to determine the thermal resistance. Tan *et al.* [7] analysed an air-to-air heat pipe heat exchanger using the ε-NTU method. They determined the optimum position separating a heat pipe into an evaporator and condenser, where regions in the heat pipe heat exchanger were formulated by minimizing the total thermal resistance of the heat path.

Wadowski *et al.* [8], studied performance of an air to air thermosyphon-based heat exchanger using R-22 as a working fluid and its thermal performance. They analysed the influence of supply and exhaust air stream mass flow rates, stream temperatures and exhaust stream moisture content on the effectiveness of the heat exchangers.

Yang *et al.* [9], Noie and Majidian [10], Noie [11], and Aliabadi *et al.* [1] used ε-NTU method for experimental study of performance of HPHE in industrial plants, hospital, laboratories, *etc.* Lin *et al.* [12] used CFD simulation for modelling the processes in heat pipe heat exchanger. Detailed analysis of various tubes and fin geometries were presented by Kays and London [13], and Rohsenow [14].

In this study the ε-NTU is used for the analysis of a gas-liquid heat pipe heat exchanger installed in a boiler power station of Vini, in Sliven, Bulgaria.

Theoretical formulation

The ε-NTU method

The ε-NTU method is based on the heat exchanger effectiveness, ε, which is defined as the ratio of the actual heat transfer in a heat exchanger to the heat transfer occurred in a heat exchanger with an infinite surface. The exit temperature of the low-temperature fluid is equal the inlet temperature of the high-temperature fluid. Therefore, the effectiveness can be defined as follows [12]:

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_{\text{hot}}(t'_{\text{hot}} - t''_{\text{hot}})}{C_{\min}(t'_{\text{hot}} - t'_{\text{cold}})} = \frac{C_{\text{cold}}(t'_{\text{cold}} - t''_{\text{cold}})}{C_{\min}(t'_{\text{hot}} - t'_{\text{cold}})} \tag{1}$$

where the heat capacities of fluid in the evaporator and condenser sections of a heat pipe heat exchanger are C_{evap}, C_{cond}, respectively.

Due to phase change, the maximum heat capacity is several orders of magnitude larger than the minimum heat capacity. Therefore, the heat capacity ratio between minimum and maximum heat capacities are equal to zero ($C_{\min}/C_{\max} \approx 0$), and the expressions for effectiveness are presented:

$$\varepsilon = 1 - \exp(-\text{NTU}) \tag{2}$$

Determination of U-value

Heat transfer coefficient from hot to the cool coolant can be defined:

$$U = \frac{1}{\dfrac{1}{h_{\mathrm{evap}}^{\mathrm{out}}} + \dfrac{\delta_{\mathrm{w}}}{k_{\mathrm{w}}} + \dfrac{1}{h_{\mathrm{evap}}^{b}} + \dfrac{1}{h_{\mathrm{c}}^{\mathrm{c}}} + \dfrac{\delta_{\mathrm{w}}}{k_{\mathrm{w}}} + \dfrac{1}{h_{\mathrm{c}}^{\mathrm{out}}}} \tag{3}$$

The coefficient of heat transfer from the hot flow towards the outer surface of the thermosyphon in the evaporator is based on the following equation [15]:

$$h = \frac{\mathrm{Nu}_1 k_1}{d_{\mathrm{out}}} \tag{4}$$

The coefficient of heat transfer for the inner surface of the thermosyphon with an intermediate heat transfer medium can be calculated [15]:

$$h_2 = 780 \frac{k^{1.3} \rho^{0.5} \rho_{\mathrm{vap}}^{0.06}}{\gamma^{0.5} l_{\mathrm{vap}}^{0.6} \rho_{\mathrm{vap},o}^{0.66} c_p^{0.33} \nu^{0.3}} q^{0.6} \tag{5}$$

The calculation of the coefficient for heat transfer of steam in a vertical pipe is determined in relation with the Nusselt theory for layered condensation and is defined from the following equation [14]:

$$h_{\mathrm{c}} = 0.943 \left[\frac{l_{\mathrm{vap}} \rho_l^2 \mathrm{g} k_l^3}{\mu_l L_{\mathrm{c}} (t_{\mathrm{s}} - t_{\mathrm{w}})} \right] \tag{6}$$

Heat transfer coefficient from the outer surface of thermosyphon tube to the cold coolant can be calculated [15]:

$$h_{\mathrm{c}}^{\mathrm{out}} = \frac{\mathrm{Nu}_{\mathrm{cold}} \, k_{\mathrm{cold}}}{d_{\mathrm{out}}} \tag{7}$$

Experimental equipment

The flow diagram for the utilization of heat from exhaust gases, resulted from the natural gas combustion with the use of thermosyphon heat exchanger, is shown in fig. 1. This has been designed in 2014 at Vini, in Sliven, using a steam generator PK-4. This is type of industrial steam boiler with steam production of 4 tons per hour saturated steam at pressure 13 bar and temperature 194 °C. Steam boiler is made from Kotlostroene JSCo**,** Sofia, Bulgaria [16].

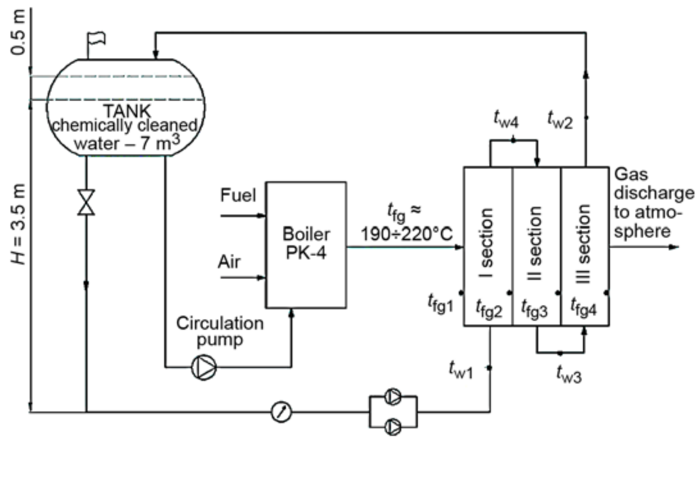


Figure 1. Schematics of heat source connected with heat pipes

The temperature of the exhaust gases of the steam generator varies between 150 °C and 220 °C depending on its working mode, whereas the temperature of the feeding water stream is around 15 °C. The steam generator works at a coefficient of excess air $\alpha \leq 1.15$. The water heater is designed for water, which passed a chemical process of desalination. The remote water heater with thermosyphon heat exchanger is not included in the loop of high pressure. The connection between the water heater and the heat pipes for water and gas circuits is illustrated in fig. 1.

The developed experimental model of the separate water heater consists of three sections. First and second sections are identical, made of two rows of 18 corridor pieces thermosyphon finned tubes. The evaporator zone is located in the middle of flue gases where the pipes are finned aluminium slats in order to intensify the heat exchange. Condensation zone is located in an aqueous environment, where the pipes of the thermosyphon are smooth.

Table 1. Main specifications of the water heater

Location of the pipes	Staggered	Parallel
Outside diameter of tube, D, [mm]	25	18
Outside diameter of fins, D_{fin}, [mm]	55	37
Fin height, H_{fin}, [mm]	15	9.5
Fin thickness, Δ_{fin}, [mm]	0.2	0.6
Fin size, S_{fin}, [mm]	3	4.5
Transverse pitch, S_1, [mm]	56	38
Relative pitch, σ_1, [–]	2.24	2.11
Longitudinal pitch, S_2, [mm]	49	38
Relative pitch, σ_2, [–]	1.96	2.11
Diagonal pitch, S_2, [mm]	56	
Number of pipes in a row, n_1, [–]	12	18
Number or rows, z, [–]	3	2
Length of the pipes, L, [mm]	928	928
Hydraulic diameter, D_{h}, [mm]	1.52	0.73

The third section of the tank is made of 36 pieces thermosyphon finned tubes, staggered in three rows. The heat exchanger is designed to heat chemically purified water received from the reservoir at a temperature ranging from 15 °C to 65 °C. The geometrical dimensions of the three sections of the developed boiler are shown in tab. 1. The diagram of the heater is presented in fig. 2.

The cut of the tank is consistent with the desired speed for optimal heat transfer fluid. Hydraulic resistances are low and do not affect the work of smoke extracting fan and chimney. Flue gas is coming from the left side of the tank in the first section through the transition with flange connections. Second and third sections are arranged sequentially in the path of the flue gases.

The temperature of the feed water prior to the heat exchanger is 15 °C. Its circulation through the heat exchanger is provided by a circulating pump with a frequency control. Water is supplied from the chemically treated water reservoir. A possibility was provided at the time of the experiment so that the heated water could returned to the tank or to be disposed of. The water flow can be adjusted over a wide range, which favours conducting tests in various operating modes.

Conducting the experiment

Experiments were carried out by equipment operating at three different loads of the steam generator. The chemically treated water flow, coming into the tank for heating, was also changed. It was possible to change the heating surface by removal of the first and second section, in which the thermosiphons are places in parallel. The location of the pipes can be clearly determined from fig. 2.

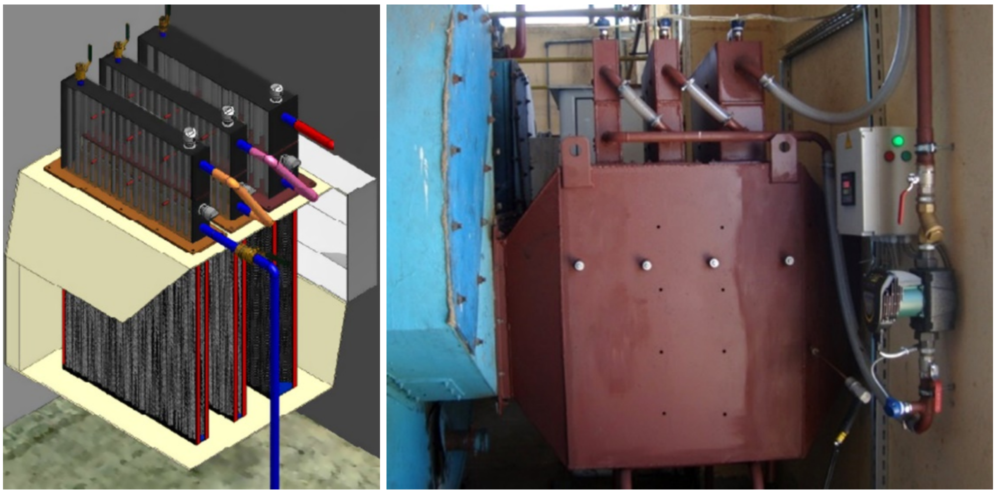


Figure 2. Example of a water heater with finned thermosyphon pipes [1]

During the experiments, the parameters given in tab. 2 were measured.

Table 2. Technical specification of measuring equipment

Parameter	Type of the meter	Range	Accuracy
Water flow rate	Ultrasonic liquid and fluid flowmeter LRF2000H	Ø = 15-6000 mm V = 32 m/s	1%
Volume fuel flow rate	Rotational gas meter G160	0.6-160 Nm^3/h	1%
Natural gas temperature (before the burner)	Microprocessor 8 channel thermometer with thermistors Pt 100 type PRT Scanner HR2562		
Feed water temperature and flue gases temperature		–200 °C to 850 °C	0.014 °C
Flue gases velocity difference between the full and static pressure in flue gas	Pitot tube	+1 to + 100 m/s	0.1 m/s
Oxygen content in the flue gases	Gas analyser TESTO 340	O_2	0.1%
Nitrous oxide content in the flue gases		0-4000 NO	1 ppm
Carbon monoxide content in the flue gases		0-10 000 ppm CO	1 ppm

The difference between the full and static pressure in flue gas is measured using a Pipot tube. For maximum accuracy in the calculations the velocity was calculated based on the combustion process and sections [4].

The results of some of the experiments are presented in tab. 2. Three modes of operation of the boiler are discussed, in which the volume and the temperature of the flue gas vary.

Temperatures of the flue gases and the water along the heat exchanger are also presented. The designated temperature at the inlet of the first section are shown with index 1, index 2 – the input of the second section, index 3 – at the entrance to the third section, and index 4 – at the outlet of the heat exchanger.

Taking into account the volume of flue gas and the smallest diameter in the pipe, the speed of the gases is calculated for the case of parallel and staggered distribution of pipes. The results can be seen in tab. 2.

In order to make the calculations for the heat exchanger the following characteristics are taken into consideration: type fluid and saturation temperature of the intermediate heat transfer medium, diameter of the pipes and ratio between the condenser and evaporator. It is important to choose the optimal temperature for the heat transfer medium saturation as this allows to decrease the heated surface of the heat exchanger.

The following assumptions are made for the thermosyphon calculations:

– the process of creating steam is achieved by surface evaporation of the condenser,
– the temperature of the steam does not change throughout the pipes,
– layered condensation according to Nusselt theory is present in the condenser,
– the vapour impact on the movement of condensate on the wall of thermosyphon is ignored, and
– the thermosyphon is placed vertically.

The calculations are made under the following conditions: high temperature exhaust gases from natural gas combustion are chosen as the hot heat carrier, and water as the cold heat carrier.

Results and discussion

The efficiency of the chosen heat exchanger is determined using the ε-NTU method in two different ways:

– using eq. (1) and the experimental data from tab. 2, and
– by determining the number NTU in eq. (2) after calculating the U-value using eqs. (3)-(7).

Results were calculated for two modes of operation of the boiler and presented in tab. 3:

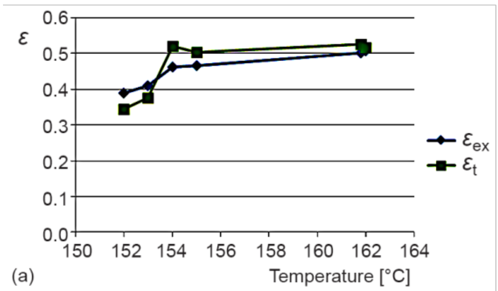
– mode 1 – consumption of natural gas as fuel is 32.5 Nm^3/h, and
– mode 2 – c onsumption of natural gas as fuel is 75 Nm^3/h.

The effect of the inlet temperature of the hot heat carrier (flue gases) on the efficiency of the heat ebxchanger was examined. Figure 3 presents the dependencies of the experimental and theoretical efficiency of the heat exchanger from the fluid inlet temperature. The comparison of experimental and theoretical values of efficiency show a very good match for both modes.

Under the first mode of operation, increased efficiency is observed when temperature is increased: 12 °C of the temperature increase in 0.117 increase of the ε value. Under the second mode of operations, where much higher temperatures of flue gasses are observed (180-210 °C), the increase in temperature results in a decrease in the efficiency, which is most noticeable when temperatures increase above 210 °C. In this mode of operation, the flue gas flow rate, and therefore, its speed is higher than those for mode 1 (tab. 3), and it reduces the efficiency of the heat exchanger. With the increase of the temperature and the velocity of the flue gases at the entry point of the heat exchanger, the latter’s temperature at the exit point of the apparatus also increases. It leads to an increase in the useful temperature difference between the temperatures of the flue gases and the water (the hot and the cold fluids, respectively), which furthermore influences the heat exchanger efficiency, eq. (1). The good match between theoretical and experimental results is illustrated by tab. 2 and fig. 3(b).

Table 3. Experimental and theoretical results

V_a [m^{3}s^{-1}]	T_{inlet} [°C]	U [Wm^{-2}K^{-1}]	$C_{\text{hot}}/C_{\text{cold}}$	ε_{exp}	ε_{t}
	Mode 1				
0.606	162	49.919	0.117	0.507	0.457
0.606	161.8	49.941	0.086	0.501	0.458
0.594	155	48.931	0.101	0.465	0.460
0.592	154	48.882	0.091	0.462	0.461
0.597	156.3	48.931	0.135	0.412	0.376
0.593	153	49.094	0.140	0.410	0.373
0.591	152	48.340	0.138	0.410	0.298
	Mode 2				
1.456	180.3	84.119	0.083	0.507	0.513
1.459	181.2	84.115	0.086	0.501	0.511
1.530	203.5	84.240	0.089	0.462	0.450
1.553	212.3	84.545	0.101	0.465	0.432
1.555	212.8	84.635	0.091	0.462	0.431
1.574	218.7	85.333	0.135	0.412	0.343
1.581	219	85.541	0.138	0.410	0.244



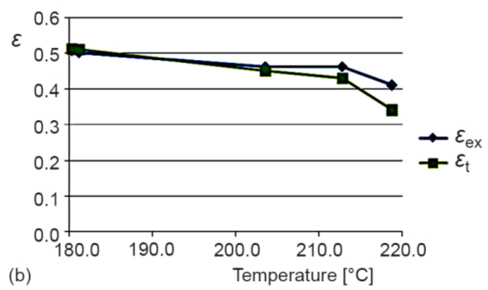


Figure 3. Dependence between effectiveness ε of the heat exchanger and the temperature of inlet hot gasses; (a) regime 1, (b) regime 2

Figure 4 shows the correlation between the efficiency, *ω*, of heat exchangers and heat capacities ratio $C_{\text{hot}}/C_{\text{cold}}$. The $C_{\text{hot}}/C_{\text{cold}}$ ratio is one of the most important factors influencing the heat exchanger efficiency. This heat exchanger uses flue gases as a hot fluid and water for the cold one. Mass flow meters are selected so that all modes comply with $C_{\text{hot}}/C_{\text{cold}} < 1$. For all the modes $C_{\text{hot}} < C_{\text{cold}}$, the efficiency decreases by increasing the ratio of $C_{\text{hot}}/C_{\text{cold}}$, because the sensible heat of the high temperature fluid stream is less than the low temperature fluid stream. In the tested modes of operation, the value of C_{hot}/C_{cold} changes from 0.086 to 0.140, mode 1, fig. 4(a), in which the experimental efficiency, ω, of the heat exchanger decreases from 0.501 to 0.410. In mode 2, fig. 4(b), C_{hot}/C_{cold} changes from 0.083 to 0.140, whereby the efficiency, ε, decreases from 0.507 to 0.410. From the results obtained it follows that the volume flow rate, and the velocity of the hot fluid accordingly, do not practically influence the efficiency at equivalent values of C_{hot}/C_{cold}, presented the results of the study. In this, Zare *et al.* [1] study on the influence of the ratio C_{hot}/C_{cold} on a hot air – water heat exchanger, the efficiency also goes down if C_{hot}/C_{cold} decreases, due to the exchanged heat declining.

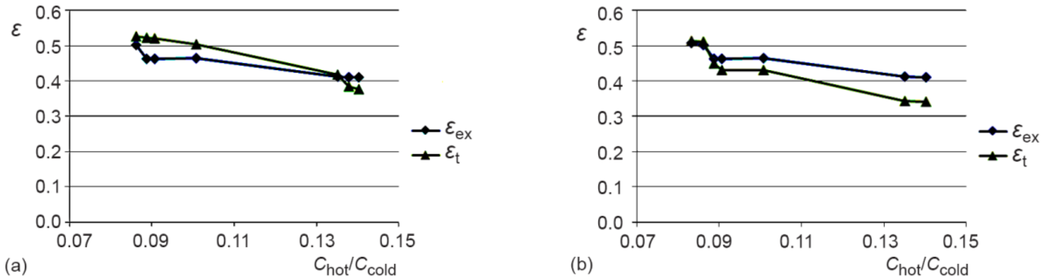


Figure 4. Dependence between effectiveness ε of the heat exchanger and the heat capacities ratio C_{hot}/C_{cold}; (a) regime 1, (b) regime 2

Conclusions

The presented experiments were conducted on HPHE developed at an industrial site. The flue gas flow rate and the temperatures of hot and cold fluid were measured, and the analysis of the results led to the following conclusions:

- The efficiency of the heat exchanger has been determined according to experimental data and ε-NTU method.
- There is a strong correlation between experimental and theoretical results.
- The effect of the inlet temperature of the hot fluid on the efficiency of the HPHE was also analysed. It was discovered that in the range of 150-165 °C temperature increase leads to an efficiency increase, while at higher temperatures in the range of 180-200 °C, the increase in the inlet temperature reduces the efficiency of HPHE.

The influence of the heat capacities ratio C_{hot}/C_{cold} on the efficiency, ω, of HPHE is studied. Upon increase of C_{hot}/C_{cold} the efficiency, ω, decreases.

Nomenclature

c_p – specific heat capacity, at p = constant, [$\mathrm{Jkg^{-1}K^{-1}}$]
C – heat capacity, [$\mathrm{JK^{-1}}$]
D – diameter of tube, [m]
g – gravitational acceleration, [$\mathrm{ms^{-1}}$]
k – thermal conductivity, [$\mathrm{Wm^{-1}K}$]
L – height of thermosyphon tube, [m]
l – latent heat, [$\mathrm{kJkg^{-1}}$]
Nu – Nusselt number
Q – heat flux density, [$\mathrm{Wm^{-2}}$]
T – temperature, [°C]
t_{gf} – flue gas temperature, [°C]

Greek symbols

γ – surface tension of the intermediate coolant, [$\mathrm{Nm^{-1}}$]
μ – dynamic viscosity of the liquid phase in, [Pa·s]
ρ – density of the liquid phase of the intermediate coolant, [$\mathrm{kgm^{-3}}$]
δ – thickness, [m]
ε – effectiveness of heat exchanger, [–]

Subscripts

c – condensing zone
cold – cool fluid

fg – flue gas
hot – hot fluid
out – outer
s – saturation temperature
vap – vapour phase intermediate coolant
vap,o – vapour phase intermediate coolant at atmospheric pressure
w – wall (of tubes)

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