Experimental validation data for CFD of heat transfer processes in a heat exchanger thermosyphon type with complex geometry

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Abstract. The paper explores the validation of experimental studies by means of numerical modelling of heat transfer processes occurring in a two-phase thermosyphon air heater. The object analysed is an air heater, used for the utilisation of the heat from flue gases emitted by a steam generator running on solid fuel with nominal load of 125 t/h, and temperature of the flue gases at nominal load 220°C. The experimental study performed shows heat transfer coefficients of the evaporation and condensation zones of $h_{fgas}=104.9$ and $h_{fgas}=84.9$ W(m²·K) respectively, and a reduction of the exhaust gas temperature down to 184°C. The numerical study pertains to a thermosyphon tube with complex geometry. Three zones can be distinguished for the air heater in question. In the evaporation zone the tube is inclined at 30 degrees in the direction of the flue gasses, in addition to the inclines of the adiabatic zone – 30 degrees, and the evaporation zone of the thermosyphon – 90 degrees to the horizon. The commercial software Ansys Fluent was used to perform the numerical calculations.

1. Introduction

Waste heat utilisation from the exit flue gases of steam generators is the most direct way of increasing their energy efficiency. In most cases, however, the lowering of the gases’ temperature leads to the creation of conditions whereby corrosive processes in the exit heating surfaces of steam boilers take place. The application of a technology of heat pipes (the term “heat exchanger with intermediate heat carrier” is used in practice, however it does not correspond to what is used in contemporary literature on the topic - “heat pipe” or more precisely “thermosyphon”) or thermosyphons is suitable for the ‘cold’ part of the first step of air heaters; this has the added benefit of protecting the primary air heater from said corrosion. This paper focuses on the achieved improvement of steam generators № 1 and № 2 of “Republica” TPP (Thermal Power Plant), where air heaters thermosyphons type have been mounted.

A literature review performed shows that interest in two-phase thermosyphons is great. Different studies are conducted both experimentally and from an analytical point of view. In [1-4] experimental studies with the performance of two phase thermosyphons are presented. The focus of the research is the behavior and efficiency of thermosyphons in different modes of operation. The main conclusion is that the geometry and the composition of the fluid in the two-phase thermosiphon have a significant effect on its efficiency.
Validation of results between analytical and experimental studies on the heat transfer coefficients is presented in [5]. Good correlations are observed, giving recommendations on model studies of two-phase thermosyphons.

The processes of heat transfer through the surrounding surface of a two-phase thermosyphon are described in [6]. It is shown both numerically and experimentally how the diameter, evaporation and condenser lengths, in addition to the vapour temperature all affect the operation envelope. Given the huge area of application of thermosyphons, in recent years there has also been an increase in numerical modeling. In [7] numerical study of two-phase thermosyphon is performed where mass flow rate, saturation pressure and also the temperatures have been evaluated. A similar numerical study is presented in [8] where the difference in modeling of single-phase and two-phase thermosyphons are discussed.

2. General information
The operational process of the thermosyphon comes down to the following: hermetically sealed tubes are filled 30% with chemically treated water. This part of the tubes, also called the zone of evaporation is under the effect of flue gas flow. The steam resulting from the gases’ heat condenses at the other end of the tube – the zone of condensation, which is affected by the air stream. The heat from the condensation heats the air, with the condensate trickling down into the evaporation zone owing to the gravitational forces.

In this type of air heaters, the heat taken off from the gases is released to the air by means of an “intermediate heat carrier” (in our case water-steam), due to which the section of the tubes located in the gas duct is characterized by a constant temperature along its length. The main advantages of using air heaters with thermosyphons are:

- their application makes the use of steam calorifiers for preheating the cold air in order to prevent the low temperature corrosion of the heated surfaces redundant;
- they have very high surface temperatures in the gas part, making their exploitation safer with regard to corrosive activity;
- they ensure the heating of outdoor air to a certain temperature, thereby ensuring a non-corrosive regime of the air heater of first degree in all exploitation regimes of the steam boiler;
- they have low aerodynamic resistance in flue gas and air media and do not require the replacement of existing ventilators;
- air leakages are practically absent in them, the eventual malfunction of separate thermosyphons does not lead to an increase in leakages.

All aforementioned benefits serve to make air heaters with thermosyphons highly competitive (and sometimes without alternative) to the air heaters used as exit heating surfaces, in particular when the price of the former is reasonable.

The performed on-site experimental tests show that the outlet flue gas temperature is high and there is a possibility of utilizing certain amount of heat. The plant's management has taken the decision to install additional thermosyphon type heat exchangers to the steam generators to increase their efficiency.

The steam generators object of the study are very old and run on solid fuel “a mix from the local mine” with the following characteristics:

- Moisture content: $W' = 16\%$
- Ash content: $A' = 43\%$
- Volatile matter: $V' = 51.5\%$
- Lower Heating Value: $Q'_l = 10500 \text{ kJ/kg}$

During the years the calorific value of the fuel decreasing as at the end of 2001 its value reaches 8164kJ/kg or 22.6% lower than to the first coal used. The proposed thermosyphons serving the operation of the steam generators were implemented at the end of 2010. A schematic representation of the realized and put into operation air heater with thermosyphons is shown in Figure 1. It is evident from the scheme that the implementation of the thermosyphons differs from the standard one. Their form has been altered.
in order to reduce the aerodynamic resistance of the heat exchanger and respectively to increase the surface temperature of the thermosyphon, in turn making the heat exchanger compact and safe from corrosion. Using this technical solution does not require any displacements/alterations nor reconstructions of the existing ventilators, and thus reduces the ultimate cost of the overall reconstruction. The technical and economic parameters for which the additional Air heater thermosyphon (AH-TS) was designed together with the nominal parameters of №1 and №2 can be found in Table 1 below.

**Table 1. Operational parameters of the steam boiler and thermosyphons.**

| Name                                                      | Nom | Dimenson | Options. Custom sizing of AH-TS |
|-----------------------------------------------------------|-----|----------|---------------------------------|
| Elementary composition of the design fuel for a work mass for AH-TS | -   | %        | C’ H’ S’ O’ N’                |
| Moisture and ash content of the design fuel of AH-TS       | W’% | A’dry%   | until 2000 year after 2000 year |
| LHV by operational data                                   | Q’/kg | 8 540/100 | 8 164/100                     |
| Combustion air per kg fuel                                | V’/kg | 2.269    |                                 |
| Fuel gas volume per kg fuel                                | Vg/ kg | 3.508    |                                 |
| Excess of air coefficient in the combustion chamber       | α  | -        | 1.2                             |
| Volume ratio of air per kg fuel to AH-1                   | V’a/ kg | 2.831    |                                 |
| Dew point temperature of the flue gases                   | t’/ C | 80       |                                 |
| Nominal load of the steam generator (SG) №1               | D/ t/h | 125      |                                 |
| Pressure of preheated steam                               | p’/MPa | 7.7      |                                 |
| Temperature of preheated steam                            | t’/ C | 500      |                                 |
| Pressure of feed water                                    | p’/MPa | 13.2     |                                 |
| Temperature of feed water                                 | t’/ C | 130      |                                 |
| Enthalpy of preheated steam                               | I’/kJ kg | 813.5   |                                 |
| Enthalpy of feed water                                    | I’/kJ kg | 132.5   |                                 |
| Heat losses                                               | q’/ % |          |                                 |
| Heat loss due to flue gas                                 | q_2/ % |          |                                 |
| Heat loss due to carbon remained                           | q_3/ % |          | 0 (according to data from the operation) |
| Heat loss due to unburnt fuel in residual                  | q_4/ % |          | 1.5 (according to data from the operation) |
| Heat loss due to radiation                                | q_5/ % |          | 0.67 (according to data from the operation) |
| Heat loss due to fly ash and slag                          | q_6/ % |          | 0.18                           |
| Enthalpy of outgoing gases for the designed fuel of AH-TS  | I’/kJ kg |          |                                 |

Based on the average load of the boiler

\[
q_i = \left[ \left( I_{\text{theo}}^{\text{fuel}} - \alpha'^\prime T_{\text{fuel}}^{\text{theo}} / Q' \right) \left( 100 - q_i \right) / 100 \right]
\]

(balance equation)

where: \( I_{\text{theo}}^{\text{fuel}} \) - theoretical enthalpy of combustion air, \( kJ / kg \); \( \alpha' \) - excess air rate, -;

For steam generator №1:

- Heat loss due to flue gas: \( q_2 \)
- Enthalpy of outgoing gases for the designed fuel of AH-TS: \( I_{\text{fuel}}' \)
gas, $kJ/kg$; $I'_{ah}$ - enthalpy of ash, $kJ/kg$;

Temperature of gases at nominal load of the boiler $t'_{fwa}$ °C 220 (data from boiler workshop)
Temperature of gases after modernization $t'_{fwa}$ °C 184 (data from boiler workshop)
Air temperature at the inlet of AH-TS $t_{air}^{AH-TS}$ °C 25 (data from boiler workshop)
Air temperature at the outlet of AH-TS $t_{air}^{AH-TS}$ °C 84 (data from boiler workshop)

Enthalpy of outgoing gases
At $t'_{fwa} = 220^\circ C$ $t_{fwa}^{220}$ $kJ/kg$ 975.2

Enthalpy of outgoing gases
At $t'_{fwa} = 184^\circ C$ $t_{fwa}^{184}$ $kJ/kg$ 179.7

Enthalpy of cold air at the entrance of the AH-TS $I'_{in}$ $kJ/kg$ 74.9

Excess of air in the flue gases $\alpha_{fwa}^*$ - 1.37 (data from boiler workshop)

Flue gas losses $q'_2$ at $t'_{fwa} = 184^\circ C$ and $Q' = 8164kJ/kg$

Flue gas losses $q'_2$ at $t'_{fwa} = 220^\circ C$ and $Q' = 8164kJ/kg$

Boiler efficiency at $t'_{fwa} = 220^\circ C$; $\alpha_{fwa}^* = 1.37$;
$t_{aw} = 25^\circ C$ and $Q' = 8164kJ/kg$

Boiler efficiency at $t'_{fwa} = 184^\circ C$; $\alpha_{fwa}^* = 1.37$;
$t_{aw} = 25^\circ C$ and $Q' = 8164kJ/kg$

Difference in boiler efficiency at $t'_{fwa} = 184^\circ C$ and $t'_{fwa} = 220^\circ C$

Amendment in losses $q_2$ at change of $t'_{fwa}$ by $10^\circ C$

Determination of fuel consumption at $t'_{fwa} = 184^\circ C$

Determination of fuel consumption at $t'_{fwa} = 220^\circ C$

Fuel savings at nominal load

Annual usage of the boiler #1 $\tau$ hr/yr 2000 year Current year 4030 4818

Annual fuel savings $B$ hr/yr $B = \Delta B \tau = 6604$

Fuel price $Pr_{fuel}$ EUR/ton 12.61

Price of electrical energy $Pr_e$ EUR/kWh 0.09

Annual cash savings $L_{fuel}$ EUR/yr $L_{fuel} = Pr_{fuel} \cdot B = 83276$

Amount of electricity for coal milling $E$ kWh/ton 25

Energy savings achieved for coal milling $L_e$ kWh/yr $L_e = BE = 165100$

Annual net incomes from electricity savings, $L_n$ EUR/yr $L_n = L_{fuel} + L_e = 98135$
Total heating surface of AH-TS $A_{total} \, m^2$ 860 
Total mass of AH-TS $G \, kg$ 26 144 
Investment costs for installation of AH-TS $L_{total} \, EUR$ 160 350 
Payback period $PB \, years$ 1.63

3. Experimental study
A number of experimental tests at different operating modes of boilers have been carried as the impact of the air-heaters on the boiler efficiency is established.

Figure 1. Location of the parameters measured.

Experimental studies include measurement of airflow and flue gas temperatures before and after thermosyphons. On fig. 1 the measurement points are displayed. The results from the on-site measurements performed are presented in Table 2.

There are two zones in which heat transfer processes occur in the thermosyphone heat exchanger - the evaporation and condensation zone. The heat transfer coefficient is determined using the normative method [9], based on the temperatures measured for both flows (air and flue gases).

Based on the methodology applied the following values for the heat transfer coefficients of for the evaporation and condensation zones respectively are obtained: $h_{gas} = 104.9 \, W/(m^2\cdot K)$ and $h_{air} = 84.9 \, W/(m^2\cdot K)$.

Table 2. Experimental data.

| Parameter                        | Label     | 1     | 2... | Trial.. | 23    | 24    | Average |
|----------------------------------|-----------|-------|------|---------|-------|-------|---------|
| Air temperature in front of AH-TS, °C | $t'_{air}^{AH-TS}$ | 25   | 23.6 | 24.8    | 26.2  |       | 25.0    |
| Air temperature behind AH-TS, °C  | $t''_{air}^{AH-TS}$ | 84.8 | 81.7 | 73.0    | 80.2  |       | 81.2    |
| Flue gas temperature in front of AH-TS, °C | $t'_{gas}$ | 219.5 | 214  | 218     | 217.5 |       | 218.4   |
| Flue gas temperature behind AH-TS, °C | $t''_{gas}$ | 184.3 | 179  | 189     | 185   |       | 184.9   |
| Steam generator load, t/h        | $D_s$     | 122.5 | 118  | 121     | 121   |       | 121.0   |

From the data in Table 1 it is evident that after the installation of air-heater the heat losses due to wasted heat in flue gases are 2.72% lower. It must also be emphasised that lowering the exit gases’ temperature by 10°C (given $\alpha_{gas} = 1.37$ and $Q'' = 8164 \, kJ/kg$) leads to a reduction in the losses $q_2$ by 0.77%. When burning fuel with lower heating value equal to the project one ($Q' = 10550 \, kJ/kg$), heat
losses due to wasted heat in flue gases are 9.96%, and at typical lower heating value (LHV) for the burnt fuel after the year 2000 $Q'_1 = 8164 \text{kJ/kg}$, $q_2 = 12.86\%$, that is, there is a noticeable increase in the losses as a result of the worsened fuel quality.

Steam generator efficiency ($\eta_{SG}$) which is the primary criterion for thermodynamic excellency of the steam generator increased by 2.72% as a result of the reconstruction realizing the air heater with thermosyphons. This does not take into account the fact that in operational conditions such an optimal coefficient of air excess for the exit gases ($\alpha_{f\text{gas}}^* = 1.37$) by lower heating value of the fuel ($Q'_1 = 8164 \text{kJ/kg}$) is hard to obtain; this in turn means that the increase in boiler efficiency is higher than 2.72% in reality.

When determining the economic effect of the realisation of the AH-TS (air heater with thermosyphons) in addition to the direct fuel economy of 1.37 t/h the economy of electricity consumption amounting to 165 MWh/year, necessary for the fuel milling savings, was also taken into consideration.

4. Numerical study
A detailed numerical study of the heat exchange in the air heater has been carried out. Given that the existing air heater is composed of a few assembled parts, only a single tube bundle was thoroughly analysed in order to reduce the time taken for the calculation procedures. Two different processes can be established – the heat transfer between air flow and exhaust flue gases, and the two-phase flow behaviour in the thermosyphon. The focus of the current work is the heat transfer between flue gases and fresh air via a thermosyphon.

4.1. Geometric model of the waste heat recovery system
Numerical procedures are performed on a real size 3D model of the air-heater (Figure 2). The thermosyphon pipes are arranged in a square tube layout. The air-heater consists of 1710 pipes. The distances between pipes in horizontal and vertical directions are 42 mm and 45 mm respectively. Three working zones of the thermosyphon are identifiable:
- evaporation zone – 3 m;
- adiabatic zone 2 m;
- condensation zone 2 m.

The middle zone connects evaporation and condensation zones and it is used to prevent the heat losses as in terms of the above is well isolated. The total head loss is calculated for both tracts: flue gases section 303 Pa; fresh air section 219 Pa.

![Figure 2. 3D Model of the proposed unit.](image)

The experimental studies show that the average velocity of the flue gases in the space between the tubes is $w_{fg} = 14.4 \text{ m/s}$, compared to that of the air $w_{f} = 10.1 \text{ m/s}$. Taking into account that the object analyzed is a single section of the suggested air heater, then the heat transfer processes concern the
flue gases' temperatures before and after the first air heater, as well as the air temperature before and after the second air heater (bearing in mind the opposite direction of movement of both flows).

The numerical procedures are performed on a CFD software. The model used for the study is three dimensional. Because of the size only part of the air-heater (30 pipes) is modeled. Triangle type of meshing elements have been used during the meshing procedure as the total number of cells amounted to 1250000 pcs.

4.2. Mathematical modeling
It was noted above that heat transfer processes are running only at evaporation and condensation zones. The middle zone is adiabatic. Also, it is accepted the steady state process into the two phase thermosyphon.

The numerical procedure block-scheme.

The CFD study concerns the heat transfer processes taking place both between the air heater and flue gases and air-heater and fresh air. The continuity, momentum and energy equations are applied during the numerical procedure. Reynold Averaged Navier-Stokes Equations (RANS) is accepted as an appropriate turbulent model. Different methods are available for discretization of RANS as here is perceived Finite Volume Method (FVM) The fundamental equations were derived in FVM using integral approach. The solution procedure is according Figure 3. The accepted mathematical model for the numerical procedures is according [10].

4.3. Initial conditions
The results from the numerical procedure will show the amount of energy transferred between air heater and air and flue gases. During the numerical study the impact of the middle zone (adiabatic) is also considered. Input parameters for the study are velocities, and temperatures of both media. Also, different turbulent intensities are considered. Based on the on-site measurements the velocity of the flue gases ($w_{fgas}$) before the thermosyphon is 7.8 m/s, and the fresh air velocity - 5.6 m/s. The analysis is also geared to study the impact of turbulent intensity (TI) and heat transfer coefficient ($h$) on the improvement of heat transfer processes. The numerical study is performed with the following input data (Table 3).

| Parameter                        | Label          | Mode 1 | Mode 2 | Mode 3 |
|----------------------------------|----------------|--------|--------|--------|
| Air temperature before AH-TS, °C | $t_{airAH-TS}$ | 25.0   | 25.0   | 25.0   |
| Air temperature after AH-TS, °C  | $t'_{airAH-TS}$| 81.2   | 81.2   | 81.2   |
| Flue gas temperature before AH-TS, °C | $t_{fgas}$    | 218.4  | 218.4  | 218.4  |
5. Results

The results obtained during the numerical study are presented below. On Figure 4a the temperature field of flue gases passing through the thermosyphon air heater is presented (Mode 2, Table 3). It is quite clear that flue gases passing through the air heater were cooled down to 185°C (assuming initial turbulent intensity of 10% for flue gases). Velocity distribution for the current case is presented in Figure 4b. The average velocity in the vicinity of the air heater is 7.8 m/s while the average velocity between pipes reaches up to 14.1 m/s.

![Figure 4a. Flue gases temperature filed distribution.](image1)

![Figure 4b. Flue gases velocity filed distribution.](image2)

Perpendicular and parallel temperature contours of the flue gases in the vicinity of thermosyphon heat exchanger are presented on Figure 5a. The flue gas specific zones (cooling zones around air heater) are clearly identifiable. The initial turbulent intensity (TI) is important for heat transfer exchange. Figure 5b presents the turbulence distribution via the path of flue gases. TI in the inter-tubular space reaches values of 220%, which is one of the reasons for the intensive heat exchange in the proposed type of inclined thermosyphon air heater.

The results of the numerical solution give a clear picture for the distribution of the gas phase and flue gas parameters before, after and in the vicinity of the thermosyphon. Summarized results for main parameters of the flows are presented in Table 4. It is important to note that the heat exchange is enhanced by increasing the turbulent intensity or at higher velocity of the two flows.
Figure 5a. Flue gases temperature contours.

Figure 5b. Flue gases turbulent intensity.

Table 4 summarizes the results of the numerical procedure.

| Numerical procedure case number | $T_f$ | $w_f$ | $T_I$ | $T_a$ | $w_a$ | $T_I$ |
|--------------------------------|-------|-------|-------|-------|-------|-------|
| $T_f$ - 5%, $\alpha_{gas} = 92$, $\alpha_{air} = 74$ W(m$^2$K) | 177   | 13.9  | 220   | 79.0  | 9.5   | 295   |
| $T_f$ - 10%, $\alpha_{gas} = 92$, $\alpha_{air} = 74$ W(m$^2$K) | 179   | 14.1  | 225   | 79.5  | 9.56  | 304   |
| $T_f$ - 20%, $\alpha_{gas} = 116$, $\alpha_{air} = 104$ W(m$^2$K) | 186   | 14.9  | 234   | 82.0  | 9.8   | 315   |
| $T_f$ - 10%, $\alpha_{gas} = 105$, $\alpha_{air} = 85$ W(m$^2$K) | 183   | 14.2  | 224   | 82.0  | 10.0  | 301   |
| $T_f$ - 10%, $\alpha_{gas} = 116$, $\alpha_{air} = 104$ W(m$^2$K) | 192   | 14.1  | 225   | 84.0  | 10.5  | 303   |

6. Conclusion

The paper has studied the possibility of increasing the efficiency of a steam boiler by installing a two-phase thermosiphon. As a result of the measure, a reduction of the flue gas temperature from 218 to 185°C was achieved, resulting in an increasing the efficiency with 2.72%. With the amount of heat transferred, fresh air is heated to 81°C, which eliminate the corrosion of the air path.

The performed numerical study gives a detailed picture for the distribution of the both flows parameters. The impact of turbulent intensity ($T_I$) and heat transfer coefficient ($h$) on the heat transfer processes is clearly presented. Increasing the $T_I$ or velocities of two flows improve the heat transfer processes.

The middle (adiabatic) zone certainly deteriorates the heat transfer processes and also increase the total head loss of the system. Despite the above, due to the constructional features, its avoidance is impossible. In addition, the 15 years of exploitation of the thermosyphon is expressed through leakages (mixing between flue gases and fresh air) between the perlitoconcrete and pipe because of the significant thermal expansion and contraction.

The numerical results obtained by using the initial data presented in row 4 of Table 4 show the best correlation with experimental data. This is the case where the initial turbulence of the two flows does not exceed 10%, as the heat transfer coefficients in both zones (evaporation and condensation) are 105 and 85 W(m$^2$K) respectively.

Validation of the numerical data with one obtained during the experimental studies will allow an express evaluation of the efficiency of a two-phase thermosyphon air heater with complex geometry, all whilst being in accordance with the proposed initial and boundary conditions.
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