Head loss analysis of thermosyphon heat exchanger with complex geometry

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Abstract: - The focus of this study is to determine the pressure drop across the two tracts of a system using a thermosyphon type heat-exchanger. For this purpose, experimental studies were carried out on the introduction of the air heater, and various modes (loads) for Thermal Power plants were tested. Experimental studies were conducted at two different fuel bases - coal and natural gas. It was found that the pressure drop on the two tracts is the smallest at the lowest load of the boiler, respectively at the lowest flue gas and fresh air velocity. Experimentally, it was found that the pressure drop on coal fuel for the flue gas tract was 147.15 Pa, and for fresh air - 118.70 Pa. At nominal boiler load, this pressure drop is 205.03 and 247.21 Pa respectively. Experimental studies indicate that the pressure drop in the two tracts of natural gas fuel is about 10-15% higher than that of coal, which is dictated by the specifics of the combustion process. Numerous studies have been carried out, and the results are compared with experimental ones. The error of the numerical solution is not more than 9%, which is within the admissible for this type of research.

Key-Words: Thermosyphon, air-heater, thermal power plant, numerical study, head loss

1. Introduction

There has been a significant increase in energy consumption in the industrial sector over the last few decades. On one hand, this is due to economic growth and on the other hand due to the depreciation or use of obsolete production equipment. This is one of the reasons to seek ways and means for reducing energy consumption in end users and systems for manufacturing and supply of energy. In heat production systems, a significant amount of heat is lost with the exhaust gases. For this purpose, thermosiphon air heaters are often used, the efficiency of which depends on the system parameters and the type of the fuel base.

In a number of works [1-5], the thermal performance of two phase thermosyphons has been experimentally studied. The total heat transfer coefficients are determined based on the operating parameters of the thermal power plant. However, the research concerns thermosiphon tubes of the regular shape. The influence of the thermosyphon geometric characteristics (angle of inclination, degree of filling, etc.) on the thermo-technical characteristics is investigated in [6]. Cases where pressure drops have been investigated in the introduction of such systems are rare, and conducting experimental research is a difficult task. In [7-9], a pressure drop test using two-phasic thermosyphons in refrigeration technology is presented.

2. General information

The general application of the termosyphons is to utilize energy from the exhaust gases. Current study focuses on heat exchanger termosyphon type with complex geometry. The heat pipes are hermetically sealed and filled with 30% chemically treated water.

Two zones in the pipe are clearly identifiable – evaporation and condensation zones. Such types of facilities are installed behind the thermal power plants in order to increase their efficiency.

A project for the needs of "Toplofikacija – Pernik" EAD was prepared. The project includes installation of air heater with "thermosyphons" (AH-TS) for steam generators №1 and №2 in the Thermal power plant (TPP). The concerned steam generators are very old running on coals with the following technical characteristics:

- Moisture content $W^r = 16\%$;
- Ash content $A^r = 43\%$;
- Volatile matter $V^r = 51.5\%$;
- Lower Heating Value $Q_r^i = 10500 kJ / kg$.

It turns out that in the course of time, the calorific value of the fuel used has decreased significantly, and for the period 1951-2001 it has decreased by about 22% reaching 8 164 kJ / kg. In one of the planned modernizations of the plant in year 2000, the management of the facility had decided to introduce two thermosyphons in order to reduce the temperatures of the exhaust gases, respectively increase the efficiency. The complex shape of the facility is dictated by both the location of the main equipment on the site and the requirements for a low pressure drop when passing the flue gases, respectively fresh air through the heat exchanger. The aim was to minimize the load on the ventilation system.

2.1 Installation set-up

Fig. 1 shows a diagram of a two-phase thermosyphon with complex geometry. Evaporation and condensation zones are clearly identifiable. The points for measuring the temperatures of the two streams are also shown.

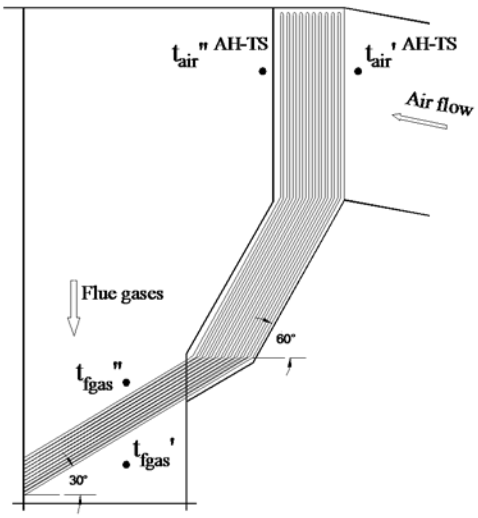

Figure 1. Location of the parameters measured.

Sometime later, a change of the fuel base was made, as coal was replaced by natural gas. Due to the specifics of the two fuels, a change in the characteristics of the thermosyphons is expected both from the thermal and hydraulic point of view.

Experimental studies were conducted on different boiler operating modes in order to assess the efficiency of thermosyphons and their hydrodynamic behavior.

Flue and air gas temperatures are measured with 8-channel microprocessor thermometer, Pt 100 resistance thermometers.

Pitot tube probe and hot wire anemometers are used to measure the velocity and pressure for both flows (flue gases and fresh air). Summarized technical data of the probes are presented in Table 1.

Table 1. Summary technical specification of a multifunction meter - Testo 435-4

Probe	Parameter	Measuring Range	Accuracy, %
Humidity probe	Humidity	0 to +100 [%rH]	0.1[%rH]
Pitot tube	Differential Pressure	0 to 25 [hPa]	0.01 [hPa]
Hot wire anemometer	Velocity	0 to 20 [m/s]	0.01 [m/s]
Hot wire anemometer	Temperature	-50 to +125 [°C]	± 0.4 [°C]

2.2 Experimental study

Table 2 is a summary of the results from the experimental studies carried out at the two fuel bases and different operating modes of the boilers. It shows the flue gas temperatures before and after the two-phase thermosyphon, the realized heat transfer coefficients, the speeds along the path of the two flows (in the inter-tubular space) as well as the pressure drop.

Table 2. Experimental data

Coal		D_1, t/h	D_2, t/h	D_3, t/h
		124	110	96
t'_{gf}	^0C	222	216	230
t''_{gf}	^0C	183	181	195
α_{fg}	W/(m^2.K)	114.3	105.9	100.3
α_{air}	W/(m^2.K)	97.2	90.1	84.2
Δp_{gf}	Pa	247.21	187.37	147.15
Δp_{air}	Pa	205.03	155.0	118.70
w_{fg}	m/s	13.7	12.4	11.7
w_{air}	m/s	10.4	9.4	8.6
Natural Gas		**D_1, t/h**	**D_2, t/h**	**D_3, t/h**
		125	112	98

t'_{gf}	^0C	220	213	221
t''_{gf}	^0C	187	182	188
α_{fg}	$W/(m^2.K)$	94.1	108	100.1
α_{air}	$W/(m^2.K)$	82.2	94.9	87.5
Δp_{gf}	Pa	296.26	220.73	180.50
Δp_{air}	Pa	276.64	206.99	168.73
W_{fg}	m/s	13.8	12.2	10.9
w_{air}	m/s	10.8	9.6	8.4

where: D is the mode of the thermal power plant; t'_{gf} is the temperature of the flue gases before the air-heater and t''_{gf} is the temperature of flue gases after the air-heater; α_{fg} and α_{air} are the overall heat transfer coefficients for flue gases and air; Δp is the pressure loss of the air heater; w_{fg} and w_{air} are the velocities of flue gases and air.

3. Numerical study

A detailed numerical study of the head loss in the air heater has been carried out. Because of the size of the air-heater, only a single tube bundle was thoroughly analysed in order to reduce the time taken for the calculation procedures. The focus of the current work is the pressure loss of the air heater via a thermosyphon.

3.1 3D model of the air-heater

For the purpose of the study a 3D model has been created (figure 2).

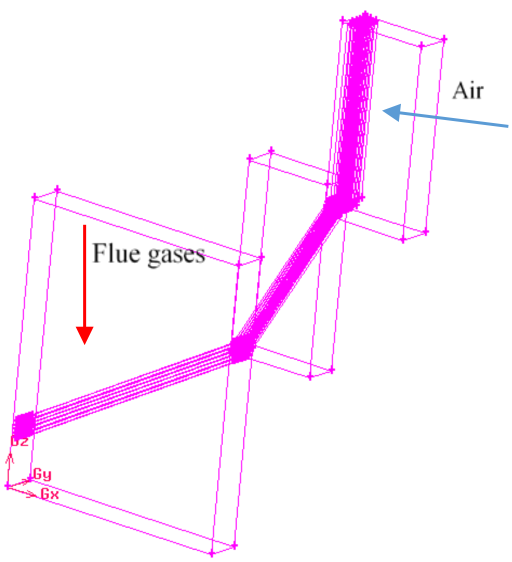

Figure 2. Model of the air heater

The concerned air heater consists of 95 tubes in a row, placed on two sides, and 9 tubes in height, arranged in a square tube layout. The total number of the thermosyphons is 1 710. Each tube's diameter is $d = 32/4mm$, the longitudinal tube pitch $s_1 = 42mm$; the transverse tube pitch between the tubes $s_1 = 45mm$. The evaporation zone of the thermosyphon is $3m$, the adiabatic one $2m$, and the condensation zone $2m$.

3.2 Mathematical modeling

In determining the pressure drop through the air heater, the roughness of the chosen pipe walls is taken into consideration, their geometry, as well as the dynamics of the flue gases and the fresh air. Numerical study was performed based on the continuity equation, momentum equation and energy equation combined with the appropriate turbulent model. Modelling of the turbulence is based on the Reynold Averaged Navier-Stokes Equations (RANS). Finite Volume Method (FVM) was used for numerical solution of RANS. The fundamental equations were derived in FVM using integral approach.

The numerical procedures are performed at different boiler modes respectively different velocities of flue gases and fresh air. The input data are according Table 2.

4. Results

Below are the results from the numerical solution. The velocity and pressure distribution fields before and after the air heater in the various boiler operating modes were presented. Initially, the numerical solution was carried out at the coal-fired combustion plant.

In Fig. 3 shows information about the distribution of the speed in the pipe bundle at the lowest load of the boiler in the course of the flue gases. The average speed in the tube bundle is $11.7m/s$ (Table 2).

Fig. 4 shows the total pressure field in the vicinity of the tube bundle of the air heater. The results of the numerical study show that the pressure drop is 152.3 Pa or about 5.2% higher than the experimentally measured.

Undoubtedly, the pressure drop is due to both the geometry of the preheater's tube bundle and the roughness of the pipe wall, all of which has an impact on the nature of the flow, as it is highly turbulent in the field of the tube bundle. This can also be seen on Fig. 5. It is clear that in the area of the preheater, the turbulent intensity increases with 290%, which is a serious prerequisite for the intensification of the heat transfer. The turbulent viscosity value reaches 0.007 kg / (m.s).

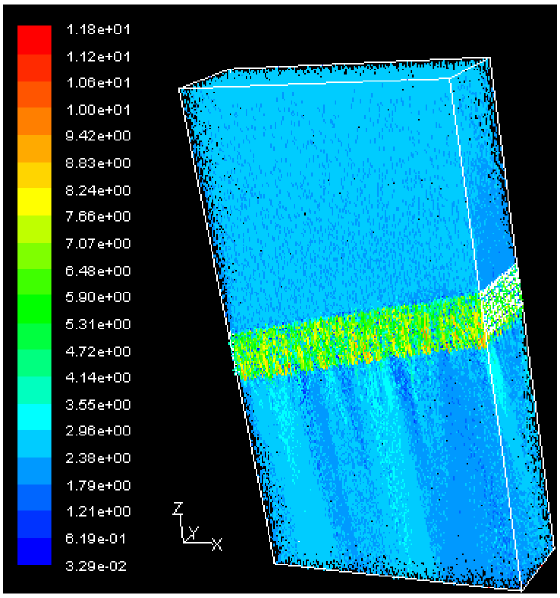

Figure 3. Flue gases velocity filed distribution.

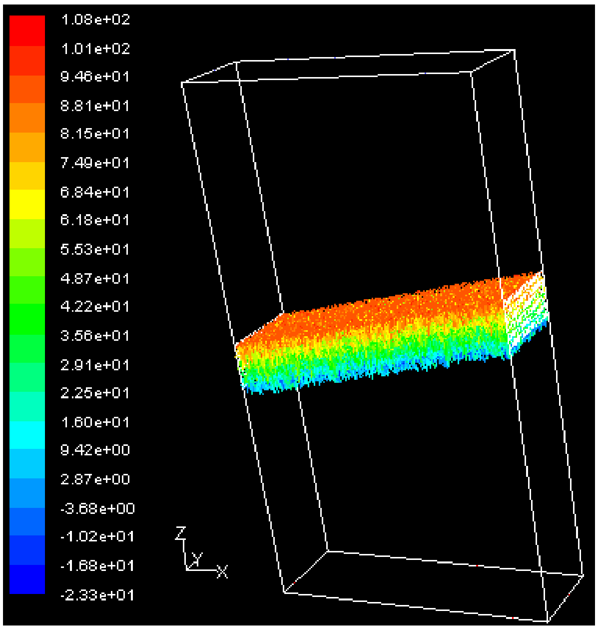

Figure 4. Velocity vectors colored by total pressure, Pa

It should also be noted that the initial turbulence, according to the conducted experimental studies, is about 10%.

Table 3 presents a comparison between numerical and experimental studies with respect to the pressure drop across the two flows. From the table, it is evident that the difference between experimental and numerical studies is between 3.5 and 9.5%, which is within the tolerance of this type of research.

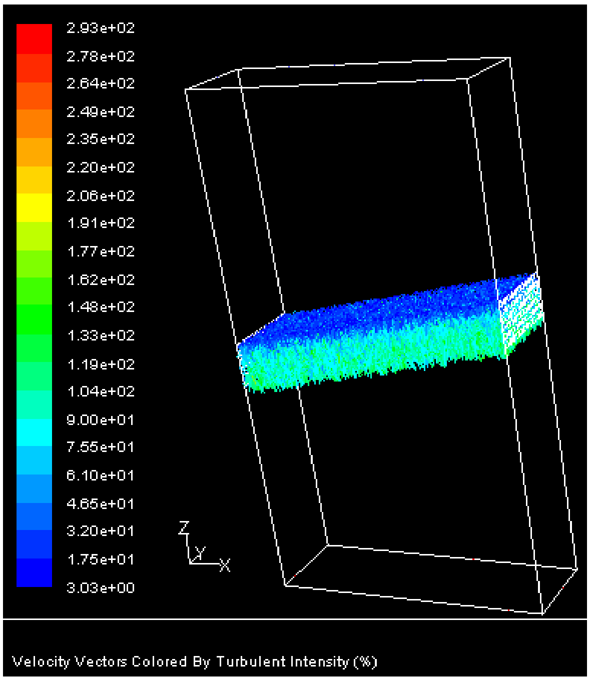

Figure 5. Turbulent intensity distribution, %

Table 3. Experimental data

Coal	Experimental study		Numerical study	
D, t/h	Δp_{gf}, Pa	Δp_{air}, Pa	Δp_{gf}, Pa	Δp_{air}, Pa
124	247.21	205.03	257.1	215.0
110	187.37	155.0	195.8	164.3
96	147.15	118.7	155.3	122.5
Natural Gas	**Experimental study**		**Numerical study**	
D, t/h	Δp_{gf}, Pa	Δp_{air}, Pa	Δp_{gf}, Pa	Δp_{air}, Pa
125	296.26	276.26	310.15	295.20
112	220.73	206.99	242.0	219.38
98	180.50	168.73	199.85	210.45

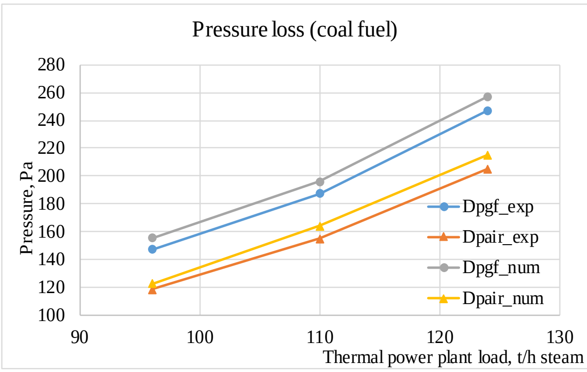

Figure 6. Pressure drop in the tract for coal running thermal power plant

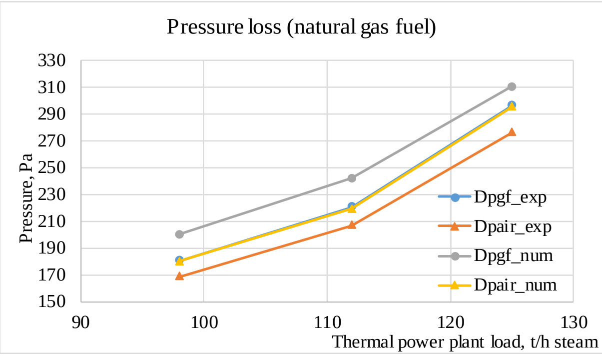

Figure 7. Pressure drop in the tract for natural gas running thermal power plant

Fig. 6 and 7 show the variation of the pressure drop across the two tracts for the two boiler fuel bases. The impression is that with a natural gas fuel base, the pressure drop is higher than that of a coal fuel base. This is due to the fact that the amount of excess air differs in the combustion process, resulting in higher speeds and greater pressure drop.

4 Conclusion

The use of an air heater for heat recovery from exhaust flue gases is a common solution in order to increase the efficiency of the system. However, the installation of this type of equipment can introduce significant resistance to the system, which can significantly change the dynamics of the main facility. With the experimental tests, the pressure drop was measured for air and flue gases path through the thermosyphon air heater for both fuel bases - boilers working on coal and natural gas.

Experimental studies were conducted at various boiler loads, and it was found that the natural gas fuel bas has in about 15% higher pressure loss than coal. This is due to the specifics of the combustion process, requiring a higher excess of air, higher flue gas velocities, more fresh air. A numerical solution performed demonstrate that it can be successfully used in modelling the pressure drop in such systems.

The error between the experimental and the numerical results in the pressure drop is not more than 5% for the boiler's incomplete load and up to 9% at full load.

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