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ENERGY SAVING AND ABATEMENT OF GAS EMISSION IN THE WOOD INDUSTRY

The analysis of heat losses and thermal efficiency of the steam system used for plywood dryers before and after its modernization has been presented. In the wood industry, mainly pressure-free condensate return systems are used. An existing open condensate return system has been improved by the application of a closed condensate tank in order to eliminate secondary steaming. The tested system consisted of three basic parts. One of them was a boiler plant producing superheated steam. Steam and condensate transfer grids connecting the boiler plant with the dryer constituted the next part of the tested steam system. The latter element was a processing device for plywood drying. The application of the closed condensate return system has increased the efficiency of the tested steam system by 15.5%. This resulted in lower emissions of contaminations to the atmosphere. The results of the test were also used to define basic economic indicators of the system in order to determine profitability of its installation.

1. INTRODUCTION

Steam and condensate systems are used most often in industrial plants where saturated or superheated steam is used in various technological processes. Such systems consist of three basic elements: output element, transfer element and receiving element. Each one generates heat losses which consequently decrease thermal efficiency of the whole system [1–3].

Over the last years, numerous papers on heat and energetic efficiency of steam systems and their components have been published. Haung [4] presented a model of a fire tube boiler to demonstrate its thermal efficiency. The model takes into account heat exchange between fuel gases and boiling water as well as heat exchange between flue gases and boiling water, and heat exchange between the external surface of the boiler and its surroundings. This model can be used to simulate efficiency of the boiler

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at various heat loads. Gürüz [5] described a mathematical model of heat exchange in a furnace taking into consideration its soot contamination. Niu and Wong [6] presented a mathematical model of a combustion chamber based on the example of a steam boiler G_{SB} = 2.614 t/h. Also Bueters [7], Richter and Payne [8] presented mathematical models of furnaces and tests on their efficiency. Bujak and Bałdyga [9] described how desalination of a steam boiler may influence its thermal efficiency in the heat load function with various salinity of feed water. One variant concerned flame and smoke tube boilers producing saturated stem, the other one – superheated steam.

Rusinowski and Stanek [10] presented neural modelling of steam boilers. Neural modelling analyses impact of heat lost as flue gases, unburned combustibles in slag, unburned combustibles in flue dust, or through the external boiler surface to the environment on thermal performance of the boiler. Bujak [11] analyzed the influence of heat losses generated through flushing of the combustion chamber of a flame and smoke tube boiler on its thermal efficiency. He presented a simple formula for determination of such losses and explained the methods of their elimination. The author demonstrated as well relation between heat load of a steam boiler and the saturated steam operating present in the boiler, and generating heat losses. Bhatt [12] presented an analytical and diagnostic tool for energy audit of steam systems which is applied to a few industrial causes.

The paper presents the analysis of heat losses and thermal efficiency of the steam system working for plywood dryers before and after its modernization. At present, in the wood industry there are mainly pressure-free condensate return systems used in plywood production. Important objective of this paper is to present improved energy management in a steam system cooperating with a plywood dryer through the application of a closed condensate tank in order to eliminate secondary steaming. This is the best solution, since elimination of secondary steaming prevents from energy and heat losses. Such systems have been implemented by the author in a few plants in Poland. Tests in the plywood production plant were carried out with the use of an analytical and diagnostic tool [13] for steam system thermal efficiency analysis. This optimization modelling meets the objectives of the efficiency targets set by the European Union. The EU wastes more than 20% of its energy due to inefficiency. MEPs call for an energy savings of 9% by 2016. Countries of the EU would also like to meet the target of improving energy efficiency by at least 20% by 2020.

2. ENERGY BALANCE AND THERMAL EFFICIENCY OF STEAM SYSTEMS

2.1. ENERGY BALANCE

The steam system shown in Fig. 1, together with its fittings and connected to other installations, can be treated as an open thermodynamic system exchanging mass, energy and heat with the environment. The illustrated system consists of three basic elements:

- output element – a boiler plant producing superheated steam,
- transfer element – steam and condensate pipes connecting the boiler plant with a plywood dryer,
- receiving element – the plywood dryer.

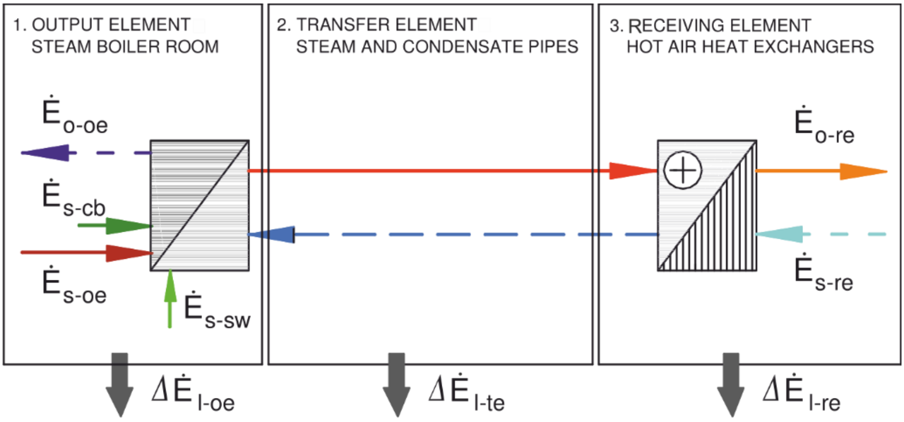

Fig. 1. Steam system with a plywood dryer

The equation of energy balance of the steam system where the output element is a boiler plant producing superheated steam, can be written as:

$$\dot{E}_{s-oe} + \dot{E}_{s-cb} - \dot{E}_{o-oe} = \dot{E}_{o-re} - \dot{E}_{s-re} - \dot{E}_{s-sw} + \Delta\dot{E}_{l-oe} + \Delta\dot{E}_{l-te} + \Delta\dot{E}_{l-re} \tag{1}$$

The enthalpy flux of flue gases carried away from the output element can be considered irrecoverable

$$\dot{E}_{o-oe} = \dot{E}_{l-chl} \tag{2}$$

and

$$\Delta\dot{E}^{1}_{l-oe} = \Delta\dot{E}_{l-oe} + \Delta\dot{E}_{l-ch} \tag{3}$$

Taking into consideration Eq. (3), the energy balance of the steam system can be rewritten as follows:

$$\dot{E}_{s-oe} + \dot{E}_{s-cb} = \dot{E}_{o-re} - \dot{E}_{s-re} - \dot{E}_{s-sw} + \Delta\dot{E}^{1}_{l-oe} + \Delta\dot{E}_{l-te} + \Delta\dot{E}_{l-re} \tag{4}$$

However, enthalpy and heat losses of the output element, fired biomass, can be determined as:

$$\Delta\dot{E}^{1}_{l-oe} = \dot{E}_{l-chl} + \dot{E}_{l-ic} + \dot{E}_{l-bo} + \dot{E}_{l-bd} + \dot{E}_{l-esb} + \dot{E}_{l-esfp} + \dot{E}_{l-es(d+f)} + \dot{E}_{l-sv2} + \dot{E}_{l-cg} \tag{5}$$

where the component E_{l-sv2} denotes the enthalpy loss in the feed water tank due to the steam evaporation through the degassing heater or condensate tank (if it there is in the boiler plant). Separating the component responsible for enthalpy loss due to secondary steaming in the feed and condensate tank, Eq. (5) can be written as:

$$\Delta\dot{E}^{2}_{l-oe} = \dot{E}_{l-chl} + \dot{E}_{l-ic} + \dot{E}_{l-bo} + \dot{E}_{l-bd} + \dot{E}_{l-esb} + \dot{E}_{l-esfp} + \dot{E}_{l-es(d+f)} + \dot{E}_{l-cg} \tag{6}$$

After transposing (6) into (4), the energy balance of the investigated system has the form:

$$\dot{E}_{s-oe} + \dot{E}_{s-cb} = \dot{E}_{o-re} - \dot{E}_{s-re} - \dot{E}_{s-sw} + \Delta\dot{E}^{2}_{l-oe} + \dot{E}_{l-sv2} + \Delta\dot{E}_{l-te} + \Delta\dot{E}_{l-re} \tag{7}$$

Considering the following equation

$$\Delta\dot{E}_{l-re} = \Delta\dot{E}^{1}_{l-re} + \dot{E}_{l-svl} + \dot{E}_{l-ct-re} \tag{8}$$

and transposing the element of Eq. (7) which is responsible for the enthalpy flux of makeup water supplied to the steam system onto the left-hand side, Eq. (7) takes the following form:

$$\begin{aligned}\dot{E}_{s-oe} + \dot{E}_{s-cb} + \dot{E}_{s-sw} = \dot{E}_{o-re} - \dot{E}_{s-re} + \Delta\dot{E}^{2}_{l-oe} \\ + \dot{E}_{l-sv2} + \Delta\dot{E}_{l-te} + \Delta\dot{E}^{1}_{l-re} + \dot{E}_{l-sv1} + \dot{E}_{l-ct-re}\end{aligned} \tag{9}$$

2.2. ENTHALPY FLUX AND HEAT LOSSES OF RESPECTIVE COMPONENTS OF THE SYSTEM

- *Enthalpy flux of flue gases* takes the form of

$$\dot{E}_{l-chl} = \dot{m}_{fg} c_{fg} \left(t_{fg} - t_{ref} \right) \tag{10}$$

Physical enthalpy flux of flue gases constitutes one of the most significant losses indicated in energy balance. It is due to the fact that temperature of flue gases coming out of the boiler is relatively high as compared to the ambient temperature.

- *Chemical enthalpy flux of flue gases* (incomplete combustion) is generated by combustion gases which have not been totally burnt in flue gases. This chemical enthalpy flux is treated as a loss of energy and received by multiplying the mass content of individual combustible components by their calorific value and total mass of flue gasses. For wood-fired boilers, one can assume that carbon oxide makes up the largest part of combustible components. The other components can be ignored:

$$\dot{E}_{l-ic} = \dot{m}_{fg} X_{\mathrm{CO}} H_{\mathrm{CO}} \tag{11}$$

• *Enthalpy flux of water from the boiler blow-off*. During boiler operations, salts in the boiler water become denser. Hence, water alkalinity and density also increase. This process can lead to significant operational difficulties depending on salt concentrations, and in some cases may even lead to boiler destruction. Thus, steam boilers are equipped with blow-off units. A blow-off connection is placed at the height of the water surface, i.e., on the boundary between the liquid and steam zones. The highest concentration of salt occurs here, and it is visible as characteristic salt foam. Therefore, the temperature of the blow-off matter is equal to the water boiling temperature at a specific steam pressure. After exceeding the pre-set (maximal) salinity threshold of the boiler water, the relief valve disposes of the excess blow-off matter

$$\dot{E}_{l-bo} = \frac{\dot{m}_{ss} s_{s-fw} h_{bo}}{s_{s-\max} - s_{s-fw}} \tag{12}$$

The procedure of determining the admissible (maximal) values of salt and silica in boiler water was assumed according to the European Union Regulations [14].

• *Enthalpy flux of water from the boiler blow-down* is

$$\dot{E}_{l-bd} = \dot{m}_{bdw} h_{bdw} \tag{13}$$

In practice, the blow-down process is carried out periodically, with the use of a time controller (blow-down timer) and a blow-down relief valve. The valve opening time is preset on the controller, as are the intervals between particular opening periods. The standard time period for valve opening is from 1 s to 3 s, and the time period between the openings is between 1 h and 5 h.

• *Heat flux lost to the atmosphere through the external surface of the steam boiler, the fittings and pipelines in the boiler plant or the transfer element, deaerating heater column and the feed water tank*. Energy losses connected to heat flowing through the external surfaces of steam boiler, devices and pipelines within the generating, transferring and receiving system constitute a significant part of the whole system losses. Their size depends on the heat exchange surface, thermal insulation quality and heating medium temperature:

$$\dot{E}_{l-esb} = UA(t_i - t_e) \tag{14}$$

$\dot{E}_{l-esfp}$, $\dot{E}_{l-es(d+f)}$ – the heat flux of this loss was determined from Eq. (14), taking into account suitable surfaces and temperatures.

- *Enthalpy flux of cooling water the grid*. Grates in steam furnaces must be cooled with water. This causes energy flux losses which can be determined with the below formula:

$$\dot{E}_{l-cg} = \dot{m}_{w-cg} c_{w-cg} \left(t_{w-cg1} - t_{w-cg2} \right) \tag{15}$$

- *Enthalpy flux by the steam lost to the atmosphere due to saturated water vapour steaming through the deaerating heater column*. Water supplying steam boilers must be properly treated. It requires oxide and carbon dioxide to be removed. The most popular method is thermal degassing for which saturated steam is necessary. Part of the steam is lost in the degassing column:

$$\dot{E}_{l-sv2} = \dot{m}_{ss-c} h_{ss-c} \tag{16}$$

$$\dot{m}_{ss-c} = \alpha A_0 \varepsilon \sqrt{2 \Delta p_{sed} \rho_{sed}} \tag{17}$$

2.3. THERMAL EFFICIENCY OF THE STEAM SYSTEM

The thermal efficiency is defined in the following way:

$$\eta_h = \frac{\dot{E}_u}{\dot{E}_{th}} \tag{18}$$

In the case of the examined system, and assuming that the thermal efficiency is determined by the direct method, Eq. (18) takes the following form:

$$\eta_h = \frac{\dot{E}_u}{\dot{E}_{s-oe} + \dot{E}_{s-cb} + \dot{E}_{s-sw}} \tag{19}$$

After applying right side of Eq. (9) to the denominator of (19), we obtain the thermal efficiency of the discussed system determined by the indirect method:

$$\eta_h = \frac{\dot{E}_u}{\dot{E}_{o-re} - \dot{E}_{s-re} + \Delta\dot{E}^2_{l-oe} + \dot{E}_{l-sv2} + \Delta\dot{E}_{l-te} + \Delta\dot{E}^1_{l-re} + \dot{E}_{l-sv1} + \dot{E}_{l-ct-re}} \tag{20}$$

In the case of the tested system, the usable energy flux can be written:

$$\dot{E}_u = \dot{E}_{o-re} - \dot{E}_{s-re} \tag{21}$$

3. DESCRIPTION OF THE STEAM SYSTEM USED IN PLYWOOD PRODUCTION PLANTS

Plywood is a wood-based material manufactured from thin (ca. 1.5 mm) layers of wood veneer which are glued together. Applied manufacturing processes require significant amounts of heat. Saturated or superheated steam is a typical heat carrier. Dryers are processing devices which consume over 90% of total heat used by such plants. Depending on manufacturing volumes, a demand for saturated steam in the drying process ranges from 4 to 20 t/h, and its pressure from 1.2 to 1.8 MPa. For plywood production, mainly pressure-free condensate return system are used. Condensate returning from the dryer is characterized by very high temperatures. This causes generation of large amounts of secondary steam in the condensate tank which leads to significant heat losses.

Therefore, the task of all currently applied solutions of steam systems cooperating with dryers is to utilize heat contained in secondary steam. Various types of systems for recovering heat from returning condensate have been used, e.g. for the purposes of central heating or warm utility water. However, they do not bring satisfactory results in such plants. In summer, there is no demand for heat necessary for heating purposes, and in winter heat gains from machines producing plywood limit the need for heating production halls to a large extent. Consumption of warm utility water in such plants is too low to obtain substantial reduction of the temperature of returning condensate.

The analysis of a typical steam system with pressure-free (atmospheric) return of condensate was reviewed on the basis of the example of a steam system consisting of:

- output element: biomass-fired (wood scraps) boiler plant with the power of Q_{sb} = 11,750 kW, with superheated steam; operating pressure p_1 = 1.85 MPa, maximum pressure p_{max} = 2.35 MPa,
- transfer element: steam and condensate grid with the total length L = 500 m,
- receiving element: plywood drying equipment, Q_{pd} = 10 800 kW, p_2 = 1.8 MPa, 100% of condensate return.

A steam boiler (SB) (Fig. 2) produces superheated steam with the overpressure p_1 = 1.85 MPa and temperature t_1 = 230 °C which, through a 250 m long steam grid, is fed to the technological equipment system known as a dryer. Steam pipelines are insulated with 12 cm thick mineral wool. Heat load of such steam systems is 100%. Air delivered to the system of 29 heaters at the temperature t_{a1} is heated to t_{a2} = 190 °C. Water steam heated after condensation of the heater flows into the open condensate tank (CT). Atmospheric pressure inside the tank causes secondary steaming of some part of the condensate. This is due to the fact that the temperature of the condensate after the heater greatly exceeds 100 °C. Its average value, measured directly after the system of hot air heaters, amounts to t_{2m} = 197 °C, whereas the average temperature of the condensate before the feed water tank is t_{3m} = 98.0 °C.

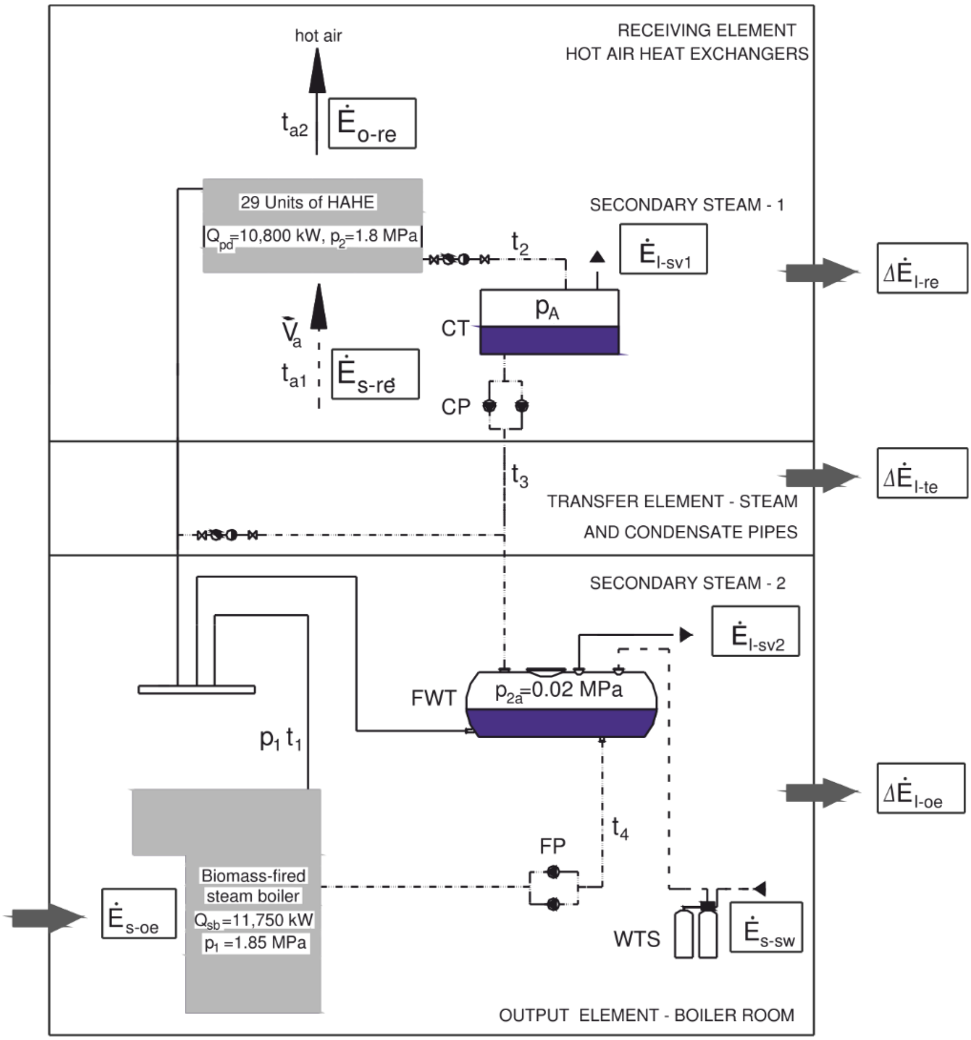

Fig. 2. Diagram of the steam system before modernization – open system

The balance of energy in the condensate tank can be expressed using the formula:

$$\dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{E}_{l-sv1} + \dot{E}_{l-ct-re} \tag{22}$$

where:

$$\dot{E}_{l-sv1} = \dot{m}_{sv1-re} h_{sv1-re} \tag{23}$$

Thus, condensate tank energy losses are the sum of losses of the enthalpy flux contained in secondary steam and the heat flux through the external surface of the condensate to the surroundings.

Taking into account heat flux losses through external surfaces of steam and condensate pipelines in the receiving element and the hot air heater:

$$\Delta\dot{E}^1_{l-re} = \dot{E}_{l-esp-re} + \dot{E}_{l-eshe-re} \tag{24}$$

and condensate tank energy flux losses:

$$\Delta\dot{E}^2_{l-re} = \dot{E}_{l-sv1} + \dot{E}_{l-ct-re} \tag{25}$$

losses of the energy flux of the whole receiving element can be expressed using the below formula:

$$\Delta\dot{E}_{l-re} = \Delta\dot{E}^1_{l-re} + \Delta\dot{E}^2_{l-re} \tag{26}$$

Table 1 shows the energy flux losses in the steam system before modernization broken down into the following elements: output element, transfer element and receiving element as per Fig. 1. Energy flux losses were determined for a full heat load (100%) of the hot air heaters (HAHE). These systems were working at a constant and maximum efficiency. Enthalpy flux losses were isolated in the output element due to steaming through the deaerating heater column ($\dot{E}_{l-sv2}$). In the receiving element, enthalpy flux losses of saturated steam generated through vaporization of the secondary condensate in the open tank ($\Delta\dot{E}^2_{l-re}$) were isolated. Under the full thermal power of the system, the summary loss of all energy fluxes amounted to $\Delta\dot{E}_{l-t}$ = 4931.1 kW and the efficiency of the whole steam system η_h = 64.3%. The actual thermal power of the steam boiler Q_{sb-r} = 11 142.2 kW was a bit lower than the nominal (operating) power Q_{sb} = 11 750 kW. However, when applying a pressure-free condensate return system, the actual thermal power of the dryer Q_{pd-r} = 8893.9 kW did not reach the assumed parameters Q_{pd} = 10 800 kW. This was caused by too much heat loss in the form of secondary steaming in the open condensate tank $\Delta\dot{E}^2_{l-re}$ = 1567.0 kW. It constituted 31.8% of total energy losses of the whole system. The largest loss of energy was found in the output element and it reached the value exceeding 55% of all the energy loss.

Table 1

Losses of the energy flux of the steam system before modernization

Heat load, %			100
Output element – boiler room	$\Delta\dot{E}^{2}_{l-oe}$	kW	2740.3
	$\dot{E}_{l-sv2}$		105.6
Transfer element – pipes, kW	$\Delta\dot{E}_{l-te}$		406.4
Receiving element – hot air heat exchanger	$\Delta\dot{E}^{1}_{l-re}$		111.8
	$\Delta\dot{E}^{2}_{l-re}$		1567.0
Total losses	$\Delta\dot{E}_{l-t}$		4931.1
Actual thermal power of the steam boiler	Q_{sb-r}		11 142.2
Actual thermal power of the hot air heaters	Q_{pd-r}		8893.9
Thermal efficiency	η_h	%	64.3

4. DESCRIPTION OF THE STEAM SYSTEM WITH PRESSURE CONDENSATE RETURN

Figure 3 shows the steam system after modernization. It was carried out within the receiving and output element. The previous open condensate tank and the feed water tank were replaced with a new, pressure tank This was installed in the boiler plant. Saturated stream was delivered to the new condensate tank through a pressure regulator valve in order to ensure appropriate minimum pressure. The pressure in the tank is higher than the pressure responsible for condensate boiling which eliminates secondary steaming. The pressure in the tank should also guarantee an appropriate difference of pressures to provide the flow of steam and condensate in the steam system.

For the analysed case, according to the tests, the maximum condensate temperature t_{3max}, measured directly before the pressure tank, amounted to 187 °C. Therefore, a steam cushion at the level of p_{2b} = 1.4 MPa (absolute 1.5 MPa) was set. Thus, the available pressure of the system was 0.45 MPa. Due to reduction of the available pressure of the whole steam system, a thorough check of the capacity of all dehydrators was performed, especially within the dryer. Some of them needed to be replaced with new ones.

The steam boiler (SB) produces superheated steam with the overpressure p_1 = 1.85 MPa and temperature t_1 = 230 °C which, through a 250 m long steam grid, is fed to the technological equipment system (dryer). Condensate returning from the equipment at ca. t_2 = 191 °C is fed to the condensate pressure tank (PT). Saturated steam under the pressure within p_{2b} = (1.35–1.45) MPa is fed to the upper part of the tank. The range of pressures inside the closed pressure tank can be determined with the pressure regulator (PR).

Energy flux losses in the steam system after modernization broken down into the following elements: outer element, transfer element and receiving element are given in Table 2. Like in the open system, losses of the enthalpy flux were isolated in the output element due to secondary steaming in the pressure tank ($\dot{E}_{l\text{–}sv2}$). However, enthalpy flux losses of saturated steam generated through vaporization of the secondary condensate in the open tank ($\Delta\dot{E}^{2}_{l\text{–}re}$) were not isolated since after modernization of the steam system the tank was removed.

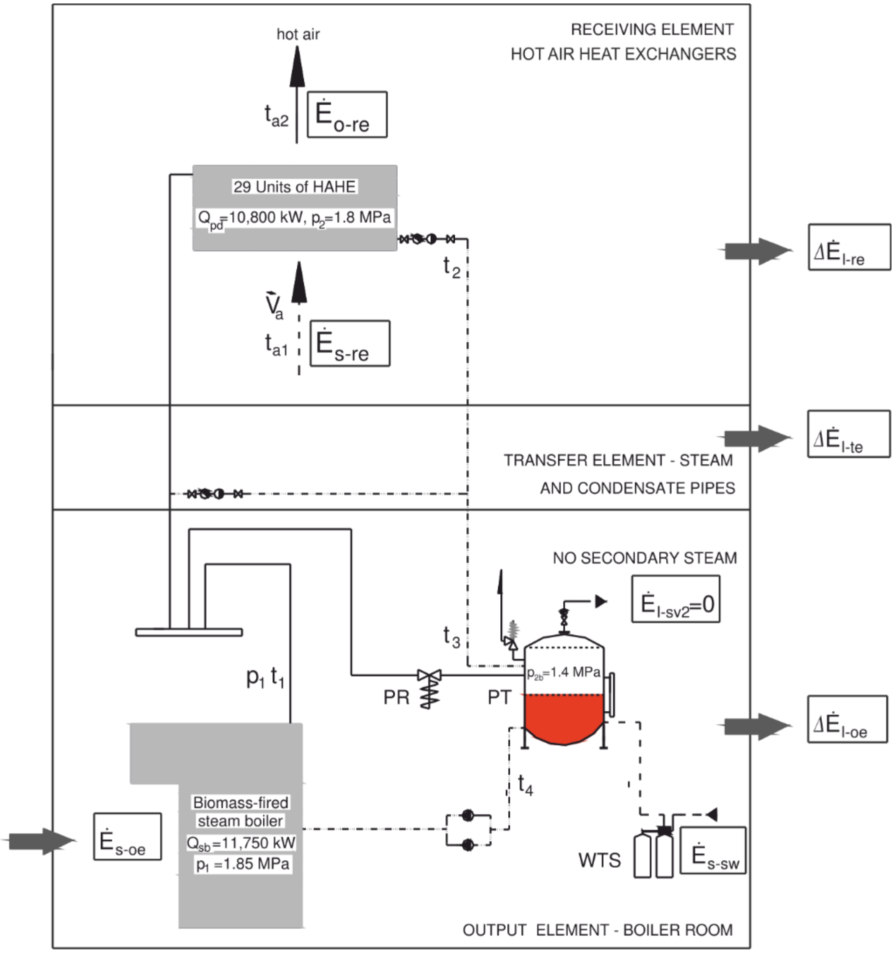

Fig. 3. Diagram of the steam system after modernization – pressure condensate return system

In the analyzed case with full thermal power of the system, the summary loss of all energy fluxes amounted to $\Delta\dot{E}_{l\text{–}t}$ = 2792.7 kW and was over 43% lower than the corresponding loss in the open system. This was caused by the removal of elements responsible for secondary steaming. The efficiency of the whole steam system was increased by 15.5% to the η_h value equal to 79.8%. The actual thermal power of the steam boiler Q_{sb-r} = 11 612.5 kW was very close to the nominal (operating) power Q_{sb} = 11 750 kW. The actual thermal power of the dryer Q_{pd-r} = 11 032.3 kW exceeded design parameters Q_{pd} = 10 800 kW.

Table 2

Energy flux losses of the steam system after modernization

Heat load		%	100
Output element – boiler room	$\Delta\dot{E}^{2}_{l-oe}$	kW	2225.3
	$\dot{E}_{l-sv2}$		0
Transfer element – pipes	$\Delta\dot{E}_{l-te}$		455.6
Receiving element – hot air heat exchanger	$\Delta\dot{E}^{1}_{l-re}$		111.8
Total losses	$\Delta\dot{E}_{l-t}$		2792.7
Actual thermal power of the steam boiler	Q_{sb-r}		11 612.5
Actual thermal power of the hot air heaters	Q_{pd-r}		11 032.3
Thermal efficiency	η_h	%	79.8

5. ECOLOGICAL AND ECONOMIC ASPECTS OF THE PERFORMED MODERNIZATION

5.1. EFFECT OF AVOIDED EMISSIONS

Assuming that the nominal thermal power of hot air heaters is Q_{pd-n} = 11 000 kW (Table 3), the steam boiler before the modernization consumed 6843 kg/h of biomass (wood scraps). An increase of thermal efficiency of the steam system after its modernization resulted in reduced consumption of biomass (wood) by 1329.2 kg/h. This also led to lower emissions of contaminations to the atmosphere, known as the effect of reduced emissions. A significant reduction of contaminations, in particular of suspended dusts, was achieved in the analyzed case – by 13.9 t/year. As far as nitric monoxides are concerned, it was 12.3 t/year. Dusts contain heavy metals, polycyclic aromatic hydrocarbons, dioxins and furans (PCDD and PCDF). Their PM10 and PM2.5 sub-fractions cause respiratory and cardiovascular diseases, as well as various types of allergies. What should be emphasized is the significant reduction of volatile organic compounds by 3.3 t/year. This is so important since wood combustion studies show the presence of more than 350 organic compounds in exhaust gases. The most important ones are: aliphatic hydrocarbons, aromatic hydrocarbons, aldehydes and alcohols. These compounds may pose a serious threat to the environment and human health. The lowest reduction of contaminations emitted to the atmosphere was achieved for a sulfate dioxide – 0.8 t/year and for carbon monoxide – 1.6 t/year. As a result of the application of the pressure condensate return system, the total reduction of emissions of contaminations to the atmosphere was 31.9 t/year which constitutes 19.8% of total emission before modernization.

Table 3

Effect of reduced emissions due to modernization of the steam system

Parameter	Before modernization	After modernization	Difference
Nominal thermal power of heaters, kW	11 000.0	11 000.0	0.0
Thermal efficiency of the steam system, %	64.3	79.8	15.5
Calorific value, raw fuel of wood as received, MJ/kg	9.0	9.0	0.0
Mass flux of burned wood, kg/h	6843.0	5513.8	1329.2
Heater operating time per year, h/year	8200.0	8200.0	0.0
Weight of burned wood per year, t/year	56 112.6	45 213.2	10 899.4
Mass flux of carbon monoxide emitted to the atmosphere, kg/h	1.1	0.9	0.2
Mass flux of sulphur dioxide emitted to the atmosphere, kg/h	0.5	0.4	0.1
Mass flux of nitric monoxide emitted to the atmosphere, kg/h	7.5	6.0	1.5
Mass flux of suspended dusts emitted to the atmosphere, kg/h	8.8	7.1	1.7
Mass flux of volatile organic compounds emitted to the atmosphere, kg/h	1.8	1.4	0.4
Annual emission of carbon monoxide to the atmosphere, t/year	9.0	7.4	1.6
Annual emission of sulphur dioxide to the atmosphere, t/year	4.1	3.3	0.8
Annual emission of nitric monoxides to the atmosphere, t/year	61.5	49.2	12.3
Annual emission of volatile dusts to the atmosphere, t/year	72.1	58.2	13.9
Annual emission of volatile organic compounds to the atmosphere, t/year	14.8	11.5	3.3
Total annual emission of contaminations to the atmosphere, t/year	161.5	129.6	31.9
Effect of reduced emissions – contamination emission reduction, %			19.8

5.2. PROJECT PROFITABILITY

The basic purpose of each business activity is its profitability. It is therefore required to determine if a specific solution which is technically justified, is also economically beneficial. One of the most popular methods of valuation of static group investment projects is SPB – Simple Payback Period. This indicates a period after which the investment project will ensure at least refund of investment expenditures. Thus SPB determines the minimum number of time units (usually years) for which a non-discounted cumulative sum of net cash flows will achieve a value equal to zero. The economic analysis of the tested steam system after modernization and input data taken for the analysis are given in Table 4.

Table 4

Input data. Technical and price indicators and calculation of profitability indicators

Average hourly enthalpy flux of additional superheated steam, kW	2138.4
Average hourly mass flux of additional superheated steam, kg/h	3250.0
Average system load, %	100.0
Price for 1 ton of superheated steam, $	30.0
Operating time, h/year	8000.0
Capital expenditures (CAPEX), $	390 000.0
Simple payback (SPB), years	0.5

Analyzing the economic aspect of the modernization, it has been proved that taking into consideration the simple payback period (SPB) this project is very profitable. Costs incurred for building of the closed system will pay for themselves within six months.

6. CONCLUSIONS

The system consisted three elements: heat source (biomass-fired steam boiler plant), steam and condensate grids and steam receiving equipment in the form of a dryer. The open condensate return system was replaced with a pressure one in order to eliminate secondary steaming (heat losses).

The application of the closed condensate return system increased the efficiency of the tested steam system by 15.5% to η_h = 79.8%. Higher efficiency resulted in an increase of the dryer thermal power by ΔQ_{pd-r} = 2138.4 kW. It means that using the same flux of fuel (wood), over 24% more heat was produced.

As a result of higher efficiency of the steam system, also emissions of contaminations to the atmosphere were reduced. Lowered emissions of such contaminations as carbon and nitric oxides, sulfur dioxide, volatile dust (TSP) and volatile organic compounds were obtained in the analyzed case. Particularly high reductions were observed in volatile dusts (13.9 t/year) and nitric monoxides (12.3 t/year). The total reduction of emissions of contaminations to the atmosphere due to the application of the pressure condensate return system was 19.8% of total emission before modernization.

Additional benefits obtained after modernization of the system were:

- lowered consumption of electric power due to elimination of condensate pumps and application of pumps feeding a steam boiler with lower lifting height,
- limitation of use of chemical agents for makeup water treatment.

The analysis of profitability of the tested modernization indicated that the SPB of the investment expenditures incurred for upgrading the open system to the closed one is 0.5 year. As far as economic aspects are concerned, this is a very profitable project.

SYMBOLS

A – surface of heat exchange, m^2
A_0 – area of diaphragm, m^2
c_{fg} – specific heat of flue gas, kJ/(kg·K)
c_{w-cg} – specific heat of water cooling the grave, kJ/(kg·K)
$\dot{E}_{l-bd}$ – enthalpy flux of water –from the boiler blow-down, kW
$\dot{E}_{l-bo}$ – enthalpy flux of water from the boiler blow-off, kW
$\dot{E}_{l-cg}$ – enthalpy flux of water used for cooling the grid, kW
$\dot{E}_{l-chl}$ – enthalpy flux of flue gases, kW
$\dot{E}_{l-ct-re}$ – heat flux lost to the atmosphere through the condensate tank external surface, kW
$\dot{E}_{l-esb}$ – heat flux lost to the atmosphere through the boiler external surface, kW
$\dot{E}_{l-es(d+f)}$ – heat flux lost to the atmosphere through the external surface of the degassing heater and the feed water tank, kW
$\dot{E}_{l-esfp}$ – heat flux lost to the atmosphere through the external surface of fittings and output element pipelines, in a boiler plant and transfer element, kW
$\dot{E}_{l-eshe-re}$ – heat flux lost to the atmosphere through the external surface of hot air heat exchanger in receiving element, kW
$\dot{E}_{l-esp-re}$ – heat flux lost to the atmosphere through the external surface of fittings, steam and condensate pipelines without condensate tank in receiving element, kW
$\dot{E}_{l-ic}$ – chemical enthalpy flux of flue gases – incomplete combustion, kW
$\dot{E}_{l-sv1}$ – enthalpy flux of steam lost to the atmosphere due to secondary steaming-1 in the condensate tank, kW
$\dot{E}_{l-sv2}$ – enthalpy flux of steam lost to the atmosphere due to secondary steaming-2 in the feed tank, steam evaporation through the degassing heater, kW
$\dot{E}_{o-oe}$ – enthalpy and heat flux carried away from the steam output element, primary side, kW
$\dot{E}_{o-re}$ – enthalpy and heat flux carried away from the receiving element, kW
$\dot{E}_{s-cb}$ – enthalpy flux of air used for combustion of biomass, kW
$\dot{E}_{s-oe}$ – enthalpy flux of biomass supplied to the steam output element, primary side, kW
$\dot{E}_{s-re}$ – enthalpy and heat flux supplied the receiving element, kW
$\dot{E}_{s-sw}$ – enthalpy flux of makeup water supplied to the steam system, kW
$\dot{E}_{th}$ – total flux of energy supplied to the optional system, kW
$\dot{E}_{u}$ – usable energy flux carried away from the optional system, kW
G_{SB} steam boiler capacity, t/h
h_{bdw} – water enthalpy carried away from the steam boiler due to its blow-down, kJ/kg
h_{bo} – water enthalpy carried away from the steam boiler due to its blow-off, kJ/kg
h_{ss-c} – enthalpy of saturated steam carried away to the environment due to the steaming of the deaerating heater column, kJ/kg
h_{sv1-re} – enthalpy of saturated steam carried away to the environment due to the steaming of the open condensate tank, kJ/kg
h_2 – enthalpy of condensate behind the hot air heat exchanger, kJ/kg
h_3 – enthalpy of condensate behind the open condensate tank, kJ/kg
H_{CO} – calorific value of carbon monoxide, kJ/kg
L – length of steam and condensate pipes, m
$\dot{m}_{bdw}$ – flux of water mass carried away from the boiler due to its blow-down, kg/s
$\dot{m}_{fg}$ – flux of flue gases mass, kg/s

$\dot{m}_{ss}$ – flux of superheated steam mass produced by the steam boiler, kg/s
$\dot{m}_{ss-c}$ – flux of saturated steam mass carried away to the environment due to deaerating heater column steaming, kg/s
$\dot{m}_{svl-re}$ – flux of saturated steam mass carried away to the environment due to due to secondary steaming-1 in the condensate tank, kg/s
$\dot{m}_{w-cg}$ – enthalpy flux of grate cooling water, kg/s
$\dot{m}_2$ – flux of condensate mass behind the hot air heat exchanger, kg/s
$\dot{m}_3$ – flux of condensate mass behind the open condensate tank, kg/s
p_1 – operating pressure of superheated steam in a steam boiler, MPa
p_2 – operating pressure of superheated steam in a dryer, MPa
p_{max} – maximum permissible pressure of a steam boiler, MPa
s_{s-fw} – concentration of salt or silica in water feeding a steam boiler, mg/dm^3
s_{s-max} – maximum concentration of salt or silica in water feeding a steam boiler, mg/dm^3
Q_{pd} – thermal power of the dryer, kW
Q_{pd-r} – actual thermal power of the dryer, kW
Q_{sb} – usable power of steam boiler, kW
Q_{sb-r} – real usable power of steam boiler, kW
t_e – temperature outside the pipeline or tank –, reference level 25 °C
t_{fg} – flue gas temperature, °C
t_{ref} – reference temperature –, reference level 25°C
t_{w-cg1} – grate output water temperature, °C
t_{w-cg2} – grate input water temperature, °C
t_1 – superheated steam temperature at the boiler output, °C
t_2 – condensate temperature after the hot air heat exchanger, °C
t_{2m} – mean condensate temperature after the hot air heat exchanger, °C
t_3 – condensate temperature before the feed water or the pressure tank, °C
t_{3m} – mean condensate temperature before the feed water or the pressure tank, °C
t_{3max} – maximal condensate temperature before the feed water or the pressure tank, °C
t_4 – condensate temperature after the feed water or the pressure tank, °C
U – coefficient of heat transmission, W/m^{2}K
X_{CO} – weight in weight concentration of carbon monoxide in flue gases, kg/kg

GREEK SYMBOLS

α – coefficient of discharge
$\Delta\dot{E}_{l-oe}$ – enthalpy and heat flux lost in the output element, kW
$\Delta\dot{E}^1_{l-oe}$ – enthalpy and heat flux lost in the output element considering losses of enthalpy flux of flue gases, kW
$\Delta\dot{E}^2_{l-oe}$ – enthalpy and heat flux lost in the output element without considering losses due to secondary steaming in the feed water tank, kW
$\Delta\dot{E}_{l-re}$ – energy flux lost in the receiving element, kW
$\Delta\dot{E}^1_{l-re}$ – energy flux lost in the receiving element without considering losses through the condensate tank external surface and due to secondary steaming-1 in the condensate tank, kW
$\Delta\dot{E}^2_{l-re}$ – energy flux lost in the receiving element through the condensate tank external surface and due to secondary steaming-1 in the condensate tank, kW

$\Delta\dot{E}_{l-t}$ – total energy flux lost in the steam system, kW
$\Delta\dot{E}_{l-te}$ – energy flux lost in the transfer element, kW
Δp_{sed} – differential pressure before and after the diaphragm, N/m^2
ΔQ_{pd-r} – actual difference between thermal powers of the dryer, kW
ε – coefficient of expansion
η_h – thermal efficiency, %
ρ_{sed} – density of saturated steam before the diaphragm, kg/m^3

SUBSCRIPTS

CT – condensate tank
CP – condensate pump
FP – feed pump
FWT – feed water tank
HAHE – hot air heat exchanger
p_A – atmospheric pressure
p_{2a} – operating pressure in the feed water tank
p_{2b} – operating pressure in the closed tank
PR – pressure regulator
PT – pressure tank
SB – steam boiler
t_{a1} – air temperature before the hot air heat exchanger
t_{a2} – air temperature after the hot air heat exchanger
V_a – air flux flowing through the heat exchanger
WTS – water treatment station

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