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Recovery and energy use of flue gas from a coal power plant

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Abstract

The paper presents a technology for heat recovery from flue gas generated in power units of coal-fired power plants. The technology for heat recovery from flue gas is based on bringing the water vapor contained in flue gas to condensation. A thorough analysis was performed of the potential of recovering heat from gas formed through the combustion of coal and lignite. The results were confirmed experimentally at the laboratory scale and in a pilot-scale real facility. The test results were used in the formulation of conclusions and guidelines that will provide valuable insight to the design of a heat recovery system. The analysis and tests performed demonstrated that the recovery of waste heat from flue gas through the condensation of water vapor is justified only if the flue gas contains a lot of moisture, e.g. flue gas formed through the combustion of lignite.

Keywords: flue gas, heat recovery, waste heat, power plant unit

1. Introduction

In the conventional thermal power industry, one of the biggest drag factors affecting power plant efficiency is chimney loss. The flue gas from a power unit is cooled down to a temperature not lower than the dew point, due to the moisture and sulfur compounds contained in it. The combination of these two chemicals causes corrosion of flue gas ducts. Heat allocation and recovery systems have been used in coal-fired power plants with a traditional chimney system, but due to the above-mentioned temperature limits the heat flux recovered from the gas was small. The introduction of new technologies in coal-fired power plants, which involved the integration of the chimney system with a cooling tower bypassed this limitation and helped lower the flue gas temperature to below the dew point, thus increasing the flux of heat recovered from the flue gas.

The waste heat recovery systems used to date [1–5] can be divided into three groups: heat allocation systems, recovery systems that use waste heat and recovery systems involving the condensation of water vapor. Heat allocation systems are designed to reduce the temperature of the flue gas upstream of the desulfurization system, and then to increase it before the chimney. Lowering the temperature of the flue gas upstream of the desulfurization system is aimed at increasing the safety of operation and efficiency of this system [2]. The second group includes heat recovery systems, the purpose of which is not only to reduce the flue gas temperature upstream of the desulfurization system, but also to direct some heat to the energy generation process. The waste heat recovered is used generally in power boilers to heat the intake air, as well as to heat the condensate in the regeneration, district heating and ORC systems [6]. The last group includes heat recovery systems involving the condensation of water vapor contained in the flue gas. Such systems, in addition to recovering large heat fluxes, also significantly contribute to cleaning the flue gas [7]. An increase in the recovery of heat flux is associated with the latent heat of the phase transition of water vapor contained in the flue gas.

The purpose of this paper is to present the results of extensive studies on the use of a condensing heat exchanger for recovery of waste heat from flue gas in a power unit. The heat recovery technology presented here definitely increases the flux of the recovered waste heat, which can be used both in the power unit and elsewhere.

2. Balance mathematical model of a condensing heat exchanger

2.1. A model of the process of condensation of water vapor from flue gas

In order to investigate the impact of the type of coal burned on the recovery of waste heat from the flue gas, a balance mathematical model of a condensing heat exchanger was developed. This model takes into account the process of condensation of water vapor in the presence of inert gas [8–10].

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As is known, apart from air, the most common gas containing water vapor is wet flue gas. In such a mixture, the portion of the gas which is not subject to phase transition is called dry gas or inert gas. The composition of the mixture is calculated based on the reaction of the combustion of coal with a given composition. The total pressure of the flue gas is the sum of the inert gas partial pressure and the water vapor partial pressure, which in the range of low pressures is described by Dalton's law (1).

$$p = p_i + p_v, \tag{1}$$

where: p—pressure of wet gas, p_i—partial pressure of inert gas, p_v—partial pressure of water vapor.

Partial pressure of the water vapor p_v in flue gas is used to determine the saturation temperature at which the process of its condensation begins. This pressure is determined on the basis of the molar fraction of water vapor r_{H2O} in the gas in accordance with equation (2)

$$p_v = r_{H2O}p. \tag{2}$$

Based on the partial pressure of water vapor, the saturation temperature was calculated. It was adopted as the point at which the condensation of water vapor contained in the flue gas begins.

In order to determine the amount of moisture in the flue gas, the moisture content was defined as the ratio of the mass of water vapor contained in the inert gas to the mass of the inert gas

$$X = \frac{m_v}{m_i}, \tag{3}$$

m_v—mass of water vapor, m_i—mass of inert gas.

After taking into account the partial pressure of water vapor and the molar mass of water vapor M_v and inert gas M_i, the moisture content X was expressed as:

$$X = \frac{p_v}{p - p_v}\frac{M_v}{M_i}. \tag{4}$$

The process of condensation of water vapor in the presence of inert gas is a more complex phenomenon than the condensation of pure water vapor. The course of this process is shown in blue in the Mollier diagram (Fig. 1).

As a result of cooling the flue gas, the enthalpy decreases (A-B line). This process takes place at a constant mixing ratio of flue gas and continues until the flue gas achieves the saturation line (marked in red). From point B, further cooling of flue gas is accompanied by the water vapor condensation process. Condensation causes the water vapor content in the flue gas to decrease (in accordance with the direction of the B-C green line), bringing with it a reduction in partial pressure. The moisture content in the gas X changes.

The mass flow rate of the condensate being formed depends on the difference between the initial and final mixing ratios of the flue gas and is expressed by the following relation

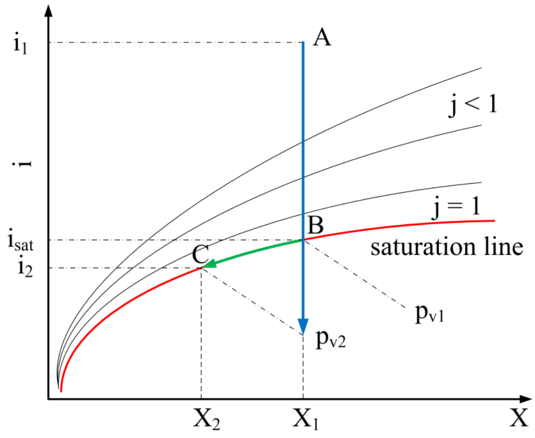

Figure 1: Isobaric cooling of flue gas

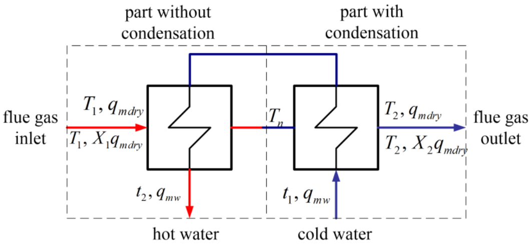

Figure 2: The division of the condensing heat exchanger into the part without condensation and the part with condensation of water vapor

$$q_{mc} = q_{mdry}\left(X_1 - X_2\right). \tag{5}$$

In turn, the heat flux from the flue gas is calculated as the difference between the initial and final enthalpies of the flue gas, using the following equation

$$Q_f = q_{mdry}\left(i_{f1} - i_{f2}\right). \tag{6}$$

The enthalpy of the flue gas is calculated as the sum of the enthalpy of inert gas and the enthalpy of water vapor

$$i_f = i_i + Xi_v. \tag{7}$$

In the process of water vapor condensation described thus, the only variable parameter is the inert gas enthalpy (after the saturation temperature is exceeded, the mixing ratio of the flue gas changes too). Thus, reference to the mass flow rate of inert gas in the mathematical model is convenient, because its value remains constant in the condensation process.

2.2. Heat balance of a condensing heat exchanger

In order to achieve a heat balance, the condensing heat exchanger was divided into two parts: the part without condensation and the part with condensation of water vapor. Fig. 2 presents a flowchart of the process of waste heat recovery from flue gas in a condensing heat exchanger.

Flue gas at temperature T_1 flows into the heat exchanger. The stream of flue gas was divided into two parts with the mass flow of inert gas q_{mf} and mass flow of water vapor $X_1 q_{mdry}$. Flue gas is cooled to the final temperature T_2 and then flows into the flue gas desulfurization system. If flue gas temperature T_2 is higher than the saturation temperature, the mass flow rates of both gases do not change. In turn, if flue gas temperature T_2 is lower than the saturation temperature, condensation of water vapor occurs in the heat exchanger. The condensation of water vapor changes its partial pressure, which in turn changes the mass flow rate of water vapor to the value of $X_2 q_{mdry}$.

The heat exchanger is cooled by water with mass flow rate q_{mw} and temperature t_1, which absorbs heat from flue gas. The cooling water leaving the condensing heat exchanger heats up to the temperature t_2. Thus, the heat flux transferred to cooling water is expressed by the equation

$$Q_w = (i_{w2} - i_{w1})\, q_{mw}. \tag{8}$$

where i_{w1}, i_{w2} are initial and final enthalpies of water, respectively.

Hence, the total heat flux transferred from flue gas to water is

$$Q_f = \left(i_{f1} - i_{f2} + X_1 i_{v1} - X_2 i_{v2}\right) q_{mdry} \tag{9}$$

If condensation of water vapor occurs, the predominant component of the heat flux transferred to the cooling water will be the heat associated with phase transition of water vapor, as described by the following equation [4, 7]

$$Q_{con} = (X_1 - X_2)\, q_{mdry} r_v, \tag{10}$$

where r_v is heat of evaporation.

The thermal power of the condensing heat exchanger is the sum of the sensible heat associated with the difference in enthalpies of flue gas at the inlet and at the outlet of the heat exchanger and the latent heat resulting from the phase transition occurring in water vapor. Thus, heat exchangers of this type are characterized by a much greater difference in temperature between the flue gas inlet and outlet as compared to heat exchangers without condensation of water vapor. This leads to an increase in the amount of heat recovered from flue gas.

The process of condensation of water vapor contained in flue gas begins at the saturation temperature. During this process the temperature is not constant as in the case of condensation of pure water vapor. This is caused by the presence of inert gas in the flue gas and by the decrease in the volumetric fraction of moisture due to condensation, which causes a change in the partial pressure of water vapor. Latent heat flux is proportional to the flow rate of the liquefied condensate, i.e., the difference between the moisture content in flue gas at the inlet and outlet of the heat exchanger. Thus, an important parameter in the calculations is the moisture content in flue gas, which primarily depends on the type of coal used. The moisture content for hard coal is approx. 9%, while for lignite it is approx. 50%. After the coal has been burned, this translates into the moisture content in flue gas, which in the case of hard coal is approx. 80 g/kg, while with lignite is approx. 240 g/kg. Therefore, in the case of flue gas from the combustion of lignite it is possible to obtain much larger differences in mixing ratio of flue gas between the inlet and outlet of the heat exchanger.

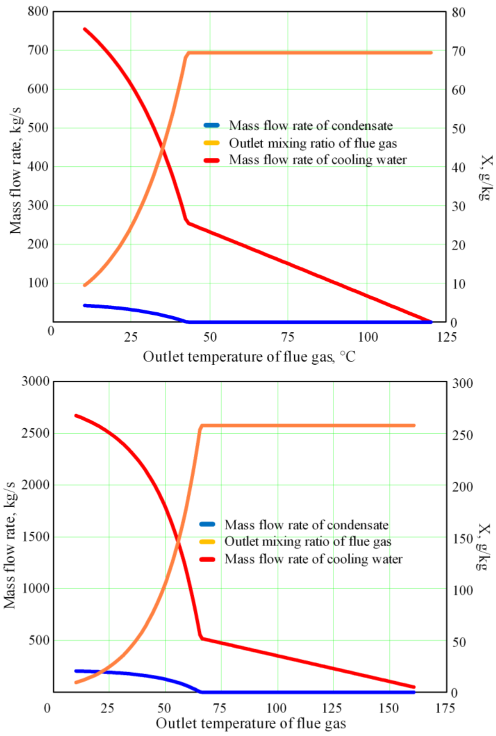

Figure 3: Dependence of the heat flux on the final temperature of flue gas from the combustion of coal and lignite

3. Analysis of waste heat recovery from flue gas from the combustion of hard coal and from the combustion of lignite

The purpose of this analysis was to demonstrate how the flux of recovered waste heat changes depending on the degree of cooling of flue gas.

The range of variability of the final temperature of flue gas for hard coal was assumed to be from 110°C to 10°C, while for lignite—from 160°C to 10°C. Based on the mathematical model of the condensing heat exchanger and the effect of condensation of water vapor from flue gas in the presence of inert gas, the heat flux recovered from flue gas for hard coal-fired and lignite-fired boilers was calculated.

The sensible and latent heat fluxes were calculated depending on the temperature to which the flue gas is cooled. The results are presented in the form of diagrams in Fig. 3. In both diagrams, the change in sensible heat is a linear dependence resulting from the change in enthalpy of only inert gas.

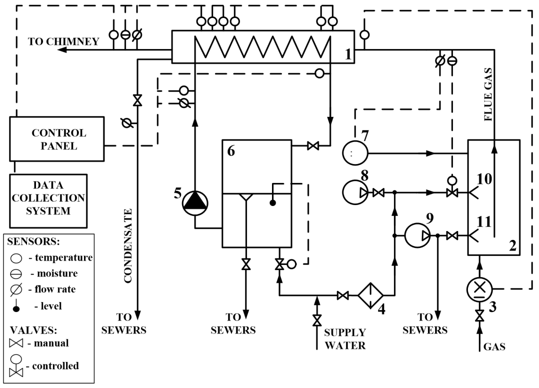

Figure 4: Schematic diagram of the test rig for testing the condensing heat exchanger; 1—condensing heat exchanger, 2—combustion chamber, 3—gas burner, 4—water filter, 5—water supply pump, 6—water tank with level control, 7—centrifugal fan, 8—air compressor, 9—high-pressure pump, 10—spray nozzle, 11—high-pressure nozzle

The enthalpy of water vapor changes similarly in a linear way, until the saturation temperature is reached, at which point the condensation process begins. Cooling of water vapor is accompanied by a phase transition and release of latent heat. The latent heat flux changes non-linearly along with the change in temperature, while slight increases in temperature are accompanied by large increases in heat flux.

The process of cooling flue gas to the saturation temperature is accompanied by a linear increase in the mass flow rate of cooling water, while the mixing ratio of flue gas is maintained at a constant level equal to that at the inlet of the flue gas heat exchanger. After the saturation temperature has been reached, the process of condensation begins, which is demonstrated by a non-linear increase in the mass flow rate of cooling water and in the mass flow rate of the liquefied condensate. The mixing ratio of flue gas at the outlet of the heat exchanger is reduced as a result of liquefaction of water from flue gas. As in the case of heat fluxes, slight changes in the temperature below the saturation temperature are accompanied by large changes in the mass flow rates of cooling water and condensate, as well as in the mixing ratio of flue gas at the outlet.

With hard coal, the process of condensation begins at approx. 42°C, while for lignite—at approx. 65°C. This is caused by the different mole fractions of moisture in flue gas and thus different values of partial pressures of water vapor. Hence, to bring about the condensation of water vapor contained in flue gas from the combustion of hard coal, the flue gas must be cooled much more than it does when lignite is burnt. However, it should be noted that the inlet temperature of flue gas from the combustion of hard coal is approx. 120°C and is lower than the inlet temperature of flue gas from the combustion of lignite, which is approx. 170°C.

4. Results of the laboratory tests

4.1. Description of the test rig

Laboratory tests on the recovery of waste heat from flue gas were performed on a test rig designed and built especially for this purpose. This test rig was used to simulate the flow of flue gas from the combustion of coal or lignite by setting an appropriate moisture content in the flue gas. The flue gas was obtained by mixing hot air with cold air and water vapor. It has been assumed that the test rig should be used only for tests concerning the process of condensation of water vapor in the presence of inert gas and thus other components present in flue gas (e.g. sulfur compounds) were omitted, as they have no significant impact on the condensation process [11]. Fig. 4 shows a diagram of the test rig with a condensing heat exchanger.

The main elements of the test rig include a condensing heat exchanger with cross-counter flow and a capacity of 46.4 kW (1), a combustion chamber (2) the purpose of which is to supply the condensing heat exchanger (1) with flue gas, a cooling system of the heat exchanger, and measuring and control instrumentation. The combustion chamber is equipped with a modulated gas burner (3). Simulation of the flue-gas composition was carried out by mixing the flue gas stream from the combustion chamber with cold air supplied by the side fan (7). The desired humidity of the air was obtained by injecting water into the combustion chamber via two spray nozzles (10) and one high-pressure nozzle (11). The flue gas prepared in this way is directed to the condensing heat exchanger, where it is cooled down and then discharged via a flue gas duct to the atmosphere. The condensing heat exchanger is cooled by water in an open-closed system. The constant temperature of water in the tank (6) is maintained by mixing hot water from the heat exchanger with cold water from the water supply system. Excess water is discharged through the overflow. The test rig is controlled from a control cabinet equipped with control systems for maintaining a constant temperature, volumetric flow rate of flue gas, mixing ratio of flue gas and temperature of cooling water.

The shell of the heat exchanger is made of stainless steel, while the pipes through which the cooling water flows are made of polytetrafluoroethylene. They form a system of 10 parallel coil pipes.

4.2. The results of the tests of the waste heat recovery performed on the laboratory test rig

An extensive test program concerning the process of waste heat recovery was completed on the test rig. The impact of flue gas inlet temperature on heat recovery was selected as a test example. In this test, the volumetric flow rate of flue gas and the volumetric flow of water were kept at a constant level, while the temperature of flue gas at the inlet of the heat exchanger was changed.

Based on the results of measurements from the balance model, a balance of heat fluxes in the condensing heat exchanger was prepared (Fig. 5).

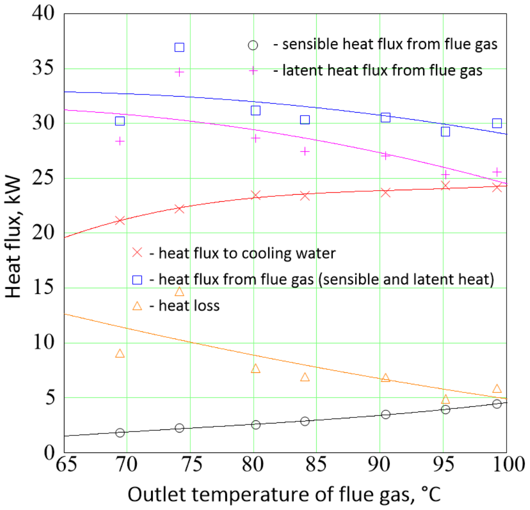

Figure 5: Balance of heat fluxes in the condensing heat exchanger depending on the inlet temperature of flue gas

The blue line indicates the total heat flux transferred by flue gas. It is composed of sensible heat flux (black line) resulting from cooling the flue gas from the initial temperature to the final temperature and the latent heat flux (pink line) resulting from the process of condensation of water vapor contained in the flue gas. The blue line indicates the heat flux transferred to the cooling water. The difference between the total heat flux transferred from flue gas and the flux transferred to water was defined as the flux of heat losses. However, the heat losses include not only the losses to the environment, but also all uncertainties of measurement, inaccuracy in calculations and averaging of the physical properties of fluids and materials.

An increase in the temperature of inlet flue gas causes a slight increase in sensible heat and a decrease in latent heat. An increase in the heat transferred to water takes place. At higher temperatures of flue gas, it stabilizes at a constant level. The value of the heat transferred to the cooling water corresponds to the waste heat recovered from flue gas. It appears from Fig. 7 that the latent heat has a greater share than the sensible heat. Hence the conclusion that the use of a condensing heat exchanger for the recovery of waste heat from flue gas is justified. The heat losses in the general meaning happen at the level of 5 to 10 kW. Only at one point described above was a higher value found. In relation to the capacity of the heat exchanger, they account for 20–25%.

5. Tests of the heat recovery from flue gas on a pilot-scale plant

5.1. Pilot-scale plant for investigating the recovery of waste heat from flue gas

The recovery of waste heat from flue gas was investigated at a pilot plant located in PGE Górnictwo i Energetyka Konwencjonalna S.A., Bełchatów Power Plant Branch. The pilot-scale plant for investigating waste heat recovery from flue gas was supplied with real flue gas from the combustion of lignite. In the flue gas duct, downstream of the main induced draft fan and upstream of the desulfurization system, a bypass supplying the condensing heat exchanger was made. Flue gas from the outlet of the condensing heat exchanger was directed back to the flue gas duct upstream of the main induced draft fan and downstream of the system for removal of dust from flue gas. Fig. 6 shows a general diagram of the pilot-scale plant.

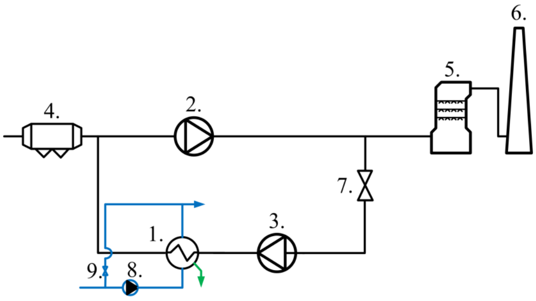

Figure 6: Diagram of the pilot-scale plant for investigating waste heat recovery from flue gas. 1– condensing heat exchanger, 2—main induced draft fan in the flue gas duct, 3—supporting fan, 4—system for removal of ash from flue gas, 5—flue gas desulfurization system, 6—chimney, 7—shut-off gate valve, 8—cooling water pump, 9—control solenoid valve

In order to adjust the mass flow rate of flue gas passing through the condensing heat exchanger, a supporting fan was installed upstream of the heat exchanger. The flue gas flowing through the condensing heat exchanger was cooled down to a temperature below the dew point with the use of a system of pipes with cooling water. The water cooling system of the heat exchanger was an open system, i.e., heated water was directed to the sewage system. As a result of cooling the flue gas to a temperature below the dew point, there occurred condensation of the water vapor contained in the flue gas and liquefaction of the condensate. The condensate collected by the tray was also directed to the sewage system. The pilot-scale plant was equipped with measurement collection and recording systems as well as a system controlling the main thermal and flow parameters. The main parameters of flue gas measured in the plant included the flue gas volumetric flow rate upstream of the heat exchanger, the flue gas temperature on the inlet and outlet stub pipes of the heat exchanger, as well as the flue gas temperature distribution along the heat exchanger and in a selected cross-section. Also measured were: parameters of the cooling water, such as the volumetric flow rate as well as the temperature on the inlet and outlet stub pipes. Thanks to the use of inverters for controlling the rotational speed of the supporting fan and water pump, it was possible to change the volumetric flow rate of flue gas and cooling water. In addition, the temperature of the cooling water supplying the heat exchanger was changed by mixing, in a predetermined proportion, cold raw water with the hot water flowing out of the heat exchanger.

Table 1: Measurements of the thermal and flow parameters of the heat exchanger in the pilot-scale plant

Item	q_{vfwet}	T_1	T_2	q_{vw}	t_1	t_2
	m^3/s	°C	°C	dm^3/s	°C	°C
1.	0.843	154.0	58.2	0.67	20.5	87.3
2.	0.944	156.1	59.0	0.67	20.5	90.9
3.	1.048	158.2	60.1	0.57	19.6	94.1

Thus, the pilot-scale plant made it possible to investigate the impact of the volumetric flow rate of flue gas, the volumetric flow rate of cooling water and the temperature of feed water on the operating point of the heat exchanger.

The main element of the pilot-scale plant for waste heat recovery was a condensing heat exchanger with a capacity of 312 kW. The purpose of the condensing heat exchanger was to cool the flue gas from the power unit to below the dew point and to recover waste heat from the flue gas in the form of hot water. The heat exchanger was placed on a steel supporting structure anchored to the base. It consisted of two units connected in series: a condensing part and a non-condensing part. This was a shell and tube heat exchanger with cross-counter flow. The shell of the heat exchanger was made of carbon steel protected with paint coatings, while the pipes with cooling water were made of Teflon (PTFE). Flue gas flowed in the shell of the heat exchanger, while cooling water flowed counter-currently through a system of 40 parallel coil pipes.

5.2. The results of tests concerning waste heat recovery performed at the pilot-scale plant

Extensive studies concerning the process of waste heat recovery were carried out at the pilot-scale plant. As an example, a study was presented on the impact of volumetric flow rate of flue gas on waste heat recovery. Its purpose was to determine the impact of the volumetric flow rate of flue gas on the operating point of the heat exchanger and on the amount of waste heat recovered. This study consisted in changing the volumetric flow rate of flue gas while maintaining constant values of other parameters such as temperature, moisture content in inlet flue gas, inlet temperature, and volumetric flow rate of cooling water. Three values of the volumetric flow rate of flue gas were set (measured upstream of the heat exchanger), and then all thermal and flow parameters were measured in the steady state. Table (1) shows the heat and flow parameters measured at the pilot-scale plant.

For the operating points of the heat exchanger presented in Table (1), the thermal balance and the moisture content in flue gas at the inlet and outlet of the heat exchanger were determined based on the balance mathematical model of the heat exchanger. The moisture content in flue gas at the inlet of the heat exchanger was calculated on the basis of the mole fraction of moisture in flue gas given in Table (1). The partial pressure of water vapor was determined and then the moisture content in flue gas was calculated using equation (4). The mixing ratio of flue gas was calculated in a similar way by substituting the saturation pressure at the temperature of the flue gas to equation (4). Based on the moisture content in flue gas at the inlet and outlet of the heat exchanger, the mass flow of the condensate formed was calculated.

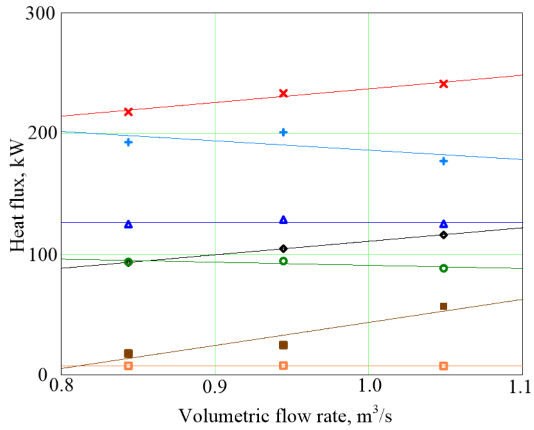

Figure 7: Thermal balance of the condensing heat exchanger at a variable volumetric flow rate of flue gas

Fig. (7) shows a balance of heat fluxes in the condensing heat exchanger at a variable volumetric flow rate of flue gas. Heat fluxes were calculated from equations (8-10). Also calculated were: the heat losses resulting from losses to the environment, the inaccuracy of calculations, averaging the physical properties of fluids and the uncertainty of measurements.

The total heat flux transferred from flue gas is shown in red (×). It consists of heat flux transferred by inert gas (⋄) and heat flux transferred by water vapor contained in flue gas (△). In turn, the latter is the sum of the sensible heat resulting from the difference of the inlet and outlet enthalpy of water vapor and the latent heat associated with the phase transition (○). Heat flux transferred to the cooling water is shown in blue (+) and is proportional to the difference between the enthalpy of water at the inlet and at the outlet of the heat exchanger. At the same time, this flux is the waste heat recovered from flue gas coming from the power unit. In addition, a slight heat flux is carried along with the condensate flowing out. This flux is marked in orange (□). In order to calculate the heat losses, the heat flux transferred to cooling water and the heat flux carried away with the condensate were subtracted from the total heat flux of flue gas. An increase in heat losses accompanying the increase in the volumetric flow rate of flue gas was observed. However, in relation to the actual capacity of the heat exchanger, these losses did not exceed the value of 60 kW, i.e. approx. 23%. In the investigated range of flow rates, an increase in the volumetric flow rate of flue gas caused a slight change in the sensible heat flux transferred by inert gas and thus in the total heat from flue gas. The heat flux transferred to cooling water remained practically at a constant level due to small changes in the velocity of the flowing flue gas and thus constant heat exchange conditions.

6. Summary/Conclusions

The paper describes the application of technology for waste heat recovery from flue gas generated in coal-fired power units, with the use of the condensation of water vapor. This technology was verified using a mathematical analysis performed on the basis of the mathematical model of the process of waste heat recovery, with the condensation of water vapor, which was verified experimentally in tests performed at a laboratory and pilot scale. The test rigs built for this purpose were described and examples of the test results were presented.

The most important conclusions to be drawn from the tests are as follows:

1. the technology described here is dedicated to power units with the flue gas outlet going through a cooling tower, thanks to which it is possible to use the phase transition of the water vapor contained in the flue gas to significantly increase the flux of the waste heat recovered;
2. the recovery of waste heat from flue gas with the condensation of water vapor is justified only with respect to flue gas from the combustion of lignite;
3. the location of a condensing heat exchanger for recovery of waste heat from flue gas upstream of the flue gas desulfurization system protects the latter against a sudden rise in temperature [7].

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