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Off-design operation of an 900 MW-class power plant with utilization of low temperature heat of flue gases

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Abstract

This article presents the off-design operation of a 900 MW-class steam turbine cycle upgraded with utilization of low-temperature waste heat taken from boiler flue gas. The low-temperature heat contributes to increasing the efficiency of power plants without introducing many complex changes to the whole system. The base for investigations was a power unit operating in off-design conditions and supplied with steam from a BB–2400 boiler. Modifications to the model were made using commercially available software and by applying the Stodola equation and the SCC method. Calculations for off-design conditions show that, after making some modifications to the system, both heat and electricity generation could be increased through the addition of a low-temperature heat exchanger.

Keywords: Low grade waste heat utilization, steam turbine, coal fired boiler, off-design operation

1. Introduction

Many industrial processes generate large amounts of waste heat that, in most cases, is discharged to the atmosphere or dissipated by other means. Usually, the heat carrier is a liquid, gas or a mixture which is at a temperature of anywhere from nearly equal to the surrounding temperature to as much as 1,000°C. The temperature of the flue gases leaving the facility determines the amount of waste heat and, therefore, the efficiency of the installation. Where the temperature of flue gases is in the region of 540°C waste heat is one of the main energy losses. At 1,000°C more than half of the fuel energy is converted into waste heat.

The European Union has placed limits on CO_2 emissions by Member States as part of its Emission Trading Scheme [1]. This impacts fossil fuel power plants to a significant degree, as their emissions are governed by the number of emission allowances they receive from the Member State allocation. There are a variety of methods available to remove CO_2 from a fossil fuel power plant system. Almost all of the methods for recovering CO_2 (sequestration) that are currently proposed result in decreased power unit efficiency and demand capital investment. The disadvantage was significantly limited for systems with fuel cell technologies [2–21], in which the Molten Carbonate Fuel Cells [22–25] separate the gas and simultaneously increasing the power and efficiency of the unit. Thus, there is a need to look for a new way to improve the efficiency and power of power plants to reduce this negative effect of Carbon Capture and Storage (CCS) [26]. Recovery (reuse) of waste heat makes it possible to significantly increase the efficiency of an existing facility. This is mainly done by recycling some of the heat to the most appropriate point in the main installation or by generating additional electrical energy [27]. Currently applied technologies for heat conversion utilize gases at higher temperatures than the flue gases of a conventional power unit [28].

Each case of low-temperature heat recovery should be

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analyzed from the thermodynamic, technical and economic points of view [29, 30]. In order to perform such analysis it is necessary to have appropriate mathematical models [31] both of the whole system as well as of its separate elements. The following aspects are central: heat capacity of the flue gases, their flow intensity, required operating parameters, for example of the installation to which the heat should be delivered, etc.

A procedure for designing a set of heat exchangers was developed, e.g. for use in the oil refining industry. Both graphical and mathematical computations were used, which resulted in the creation of an approach that could be used in various networks [32].

Additionally, the size and type of heat exchanger have to be considered. Low-temperature heat should be recovered in a counter-flow heat exchanger, as this is the most efficient way from the exergetic point of view. The other available options are cross-flow and parallel designs [33, 34]. However, since the capital costs involved in a counter-flow heat exchanger are the greatest, it is not advised for facilities operating during short periods of time [35].

Low-temperature heat can be reused to transform a regular CHP plant into a polygeneration plant which produces ethanol from wood. Simulations led to the conclusion that combining a CHP plant with an ethanol production facility delivers an 11% increase in efficiency compared to separate production [36].

In boilers that are currently in use the heat of flue gases is recovered only to a certain extent. This is determined by the condensation temperature of the steam in flue gases. If the flue gases are cooled to a temperature below the dew point, steam will condense, and, for fuels containing sulfur (e.g. hard coal and lignite) it will react with sulfur dioxide and produce sulfuric acid. At present this is the main reason that limits the temperature of cooling flue gases.

In order to use the heat of the flue gases efficiently, a small temperature difference between the cooling and cooled media must be retained. To obtain minimal values for the temperature difference, heat exchange must be intensified (particularly for a gaseous medium with a low convective heat transfer coefficient). The intensification of the heat exchange can be achieved by making its area more complex, i.e. by finning the surface. However, a too densely finned surface can become dusted due to the fact that flue gases may contain a large amount of volatile particles.

For technical reasons the easiest way to reuse low-temperature heat is to heat up industrial water. Unfortunately, its use is limited significantly by the low temperature of returning water in the municipal heating systems. The Polish heating grid is a high-temperature one. In summer the returning water temperature is kept at around 43..46°C, whereas in winter it is 60°C. Therefore, it is feasible to recover and reuse low-temperature heat in summer.

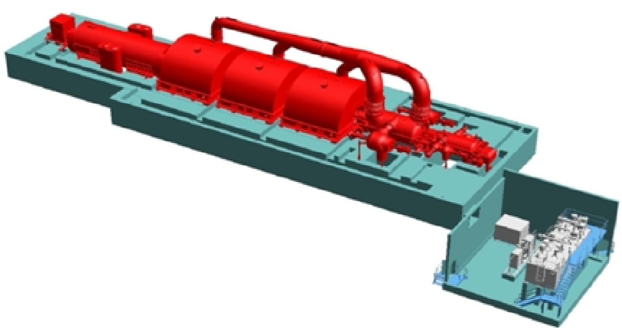

Figure 1: STF-100 steam turbine view

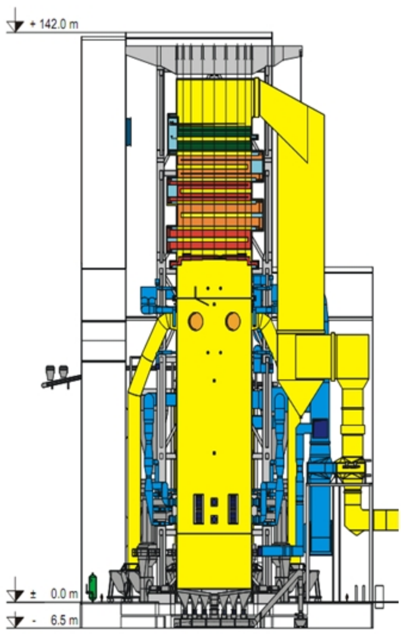

Figure 2: Lignite fueled boiler type BB–2400

2. Mathematical model of a 900 MW-class unit

A model of a STF–100 (see Fig. 1) steam turbine unit was built to specify operating at off-design conditions. The model of the unit and its modifications was made based on GateCycle software [37, 38]. Steam flows through the subsequent stages of the turbine were determined using the Stodola equation and the efficiencies were

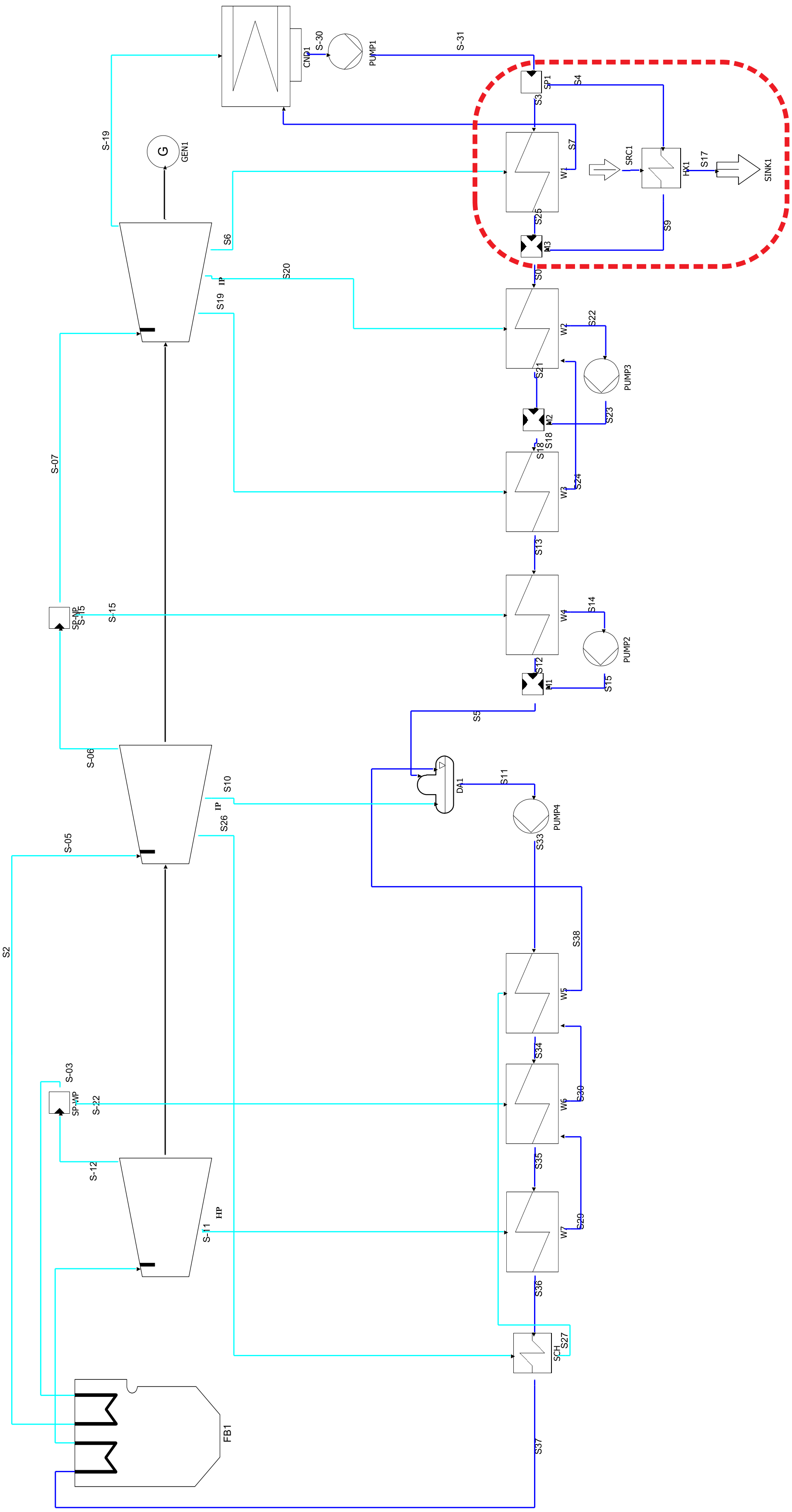

Figure 3: Scheme of 900 MW class power plant, bypassed heat exchanger is indicated by an oval

calculated by the SCC method (from names: Spencer, Cotton, Cannon) [39]. This method is recommended by the American Society of Mechanical Engineering—ASME, to calculate the efficiency of turbines in conventional plants. A modified Stodola equation (otherwise known as an ellipse Stodola) is as follows:

$$W = C\sqrt{\frac{p}{v}}\sqrt{1-\left(\frac{r-r^{*}}{1-r^{*}}\right)^{2}} \tag{1}$$

where: W—steam flow, C—flow coefficient, p—inlet steam pressure, v—specific volume of steam at turbine inlet, r—pressure ratio, r*—critical pressure ratio.

The heat source for STF–100 units is an BB-2400 boiler (see Fig. 2) with the following live steam parameters: 260/54.3 bar and 554/582°C. Steam is decompressed in a condensing turbine, and for the initial configuration the unit generated 900 MW_e during condensation operation. The model of the unit was based on the heat diagram of the 900 MW-class unit presented in Fig. 3.

The power plant is equipped with a steam turbine STF-100 (DKY5-6N41B)—see Fig. 1, axial, five-bodies (single flow high pressure body, double flow medium pressure body, and three double flow low pressure bodies). The DKY5-6N41B turbine is a reaction, condensing unit with inter stage reheating, and seven bleeders. The regeneration sub-system is composed of three low pressure heat exchangers (W1) connected in parallel, and connected to them in series are heat exchangers W2, W3 and W4. Additional heating of the main condensate is done by a heat recovery system from the exhaust. The feed water is heated by a high pressure regeneration system consisting of three series-connected heat exchangers W5, W6, and W7. The following parameters were considered as the nominal operation conditions of the boiler: steam pressure after the boiler: 253/58.3 bar, thermal power: 900 MW, steam temperature after the boiler: 550/580°C.

The regeneration heat exchangers included in the model have the following parameters: minimum temperature difference: 5°C, condensate supercooling: 5°C, convective heat transfer coefficient in the volume where condensation occurs: 3.5 $W/m^2/K$ convective heat transfer coefficient from the condensate: 2 $W/m^2/K$. Utilization of low-temperature waste heat is done by bypasing W1—third turbine bleed is closed, and condensate from a condenser flow to heat exchanger LP1, in which the low temperature heat is utilized. Other system elements remain the same. The results for two systems are presented: the original system (Reference Case) and the system equiped with heat exchanger LP1—Case NP1. All the heat exchangers

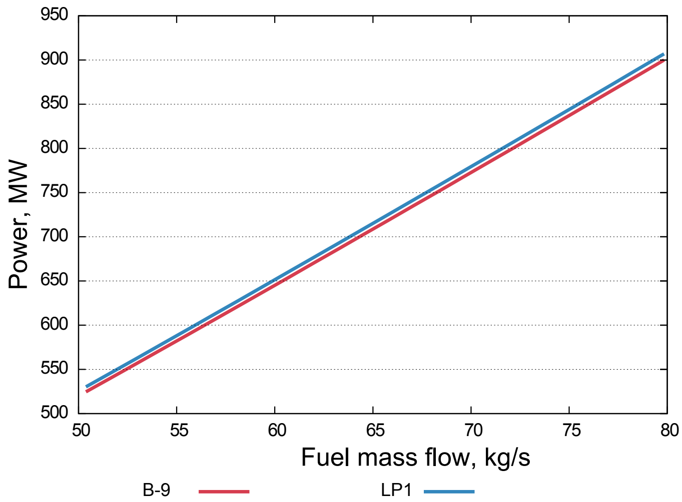

Figure 4: Changes in the power generated, depending on the fuel supply

were modeled for off-design conditions with a constant heat exchange area and given values of heat transfer coefficient.

$$\eta_{el} = \frac{P}{Q_{HHV}} \tag{2}$$

where: P—total power, kW; Q_{HHV}—heat delivered to the system, kW.

Efficiencies of electrical energy generation were calculated using the formula 2 and were compared.

3. Off-design operation of an 900 MW-class power plant with utilization of low temperature heat of flue gases

The most suitable position for installing the low-temperature heat exchanger was determined previously [40]. Based on the analysis made, the power generated by the individual units is calculated as a function of fuel supply. Changes in the power generated are shown in Fig. 4. For better illustration, the difference obtained between the powers generated in both cases are indicated separately. As can be seen, the power generated by unit NP1 is several megawatts higher due to higher efficiency. The changes in efficiency depend on the fuel supply stream, as shown in Fig. 5. The course of changes as a function of the power generated is also plotted in Fig. 6.

The study made it possible to determine the temperatures of the stream which leaves heat exchanger LP1 and the temperature of water which leaves the condenser. Inlet parameters of both fluids are fixed in all variants apart from the mass flow of the condensate leaving the condenser. The external fluid has a constant flow rate of

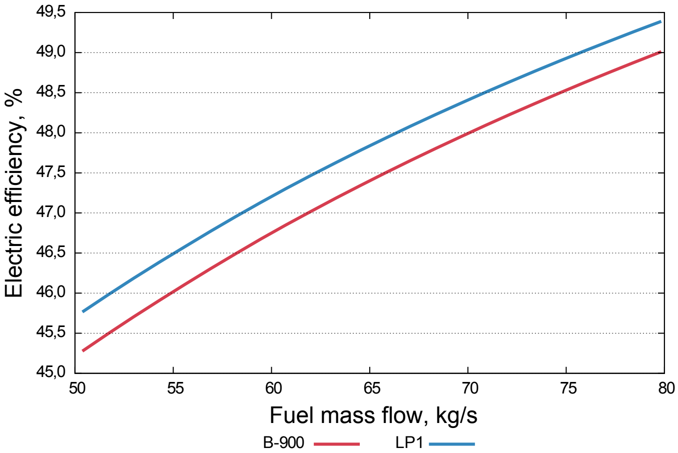

Figure 5: The efficiency of power generation, depending on the fuel supply

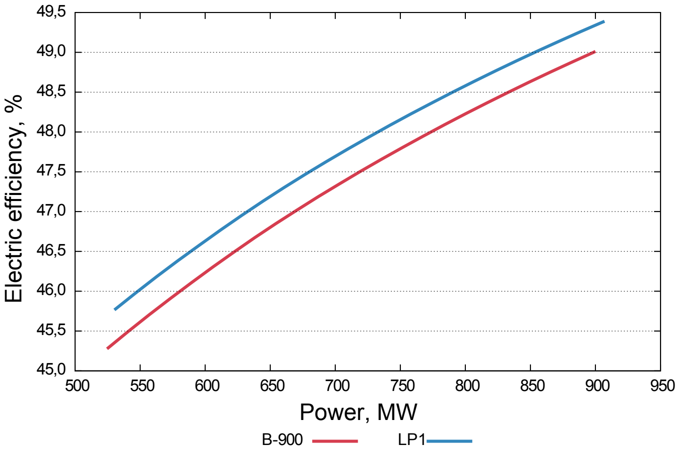

Figure 6: The efficiency of power generation depending on the power generated in both cases

347 kg/s and inlet temperature of 90°C, but the condensate has a temperature of 33°C, and the flow rate varies in a range from 401 kg/s for maximum power to 266 kg/s at power of 530 MW. Outlet temperatures and duty of heat exchanger LP1 are shown in Fig. 7.

It is shown that the external stream can be cooled to 43.4°C for maximum power, but as the power decreases, this temperature rises and at power of 530 MW it is almost 8°C higher. Utilization of low grade heat, in this case is smaller by 11 MW, gives only 84% heat utilization compared to the highest power point.

The important factor here is power division between specific parts of the steam turbine during off-design operation. Fig. 8 presents power generated by the each part of the steam turbine: HP, IP, and LP with and without utilization of low grade heat. All curves are almost identical—thus not included in the paper. The only slight difference is that in the case of low grade heat utilization, the LP part

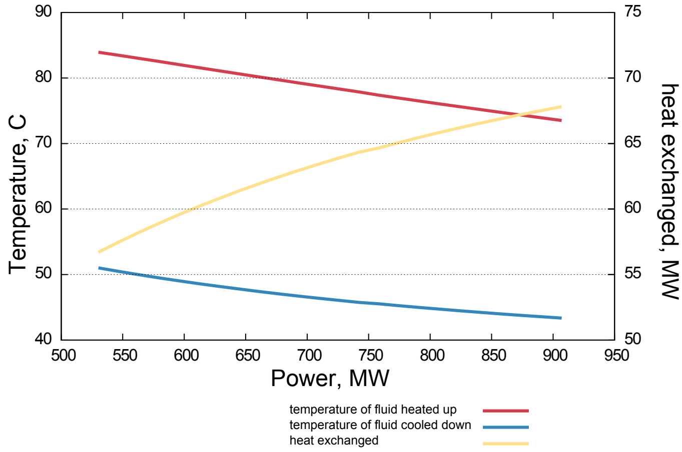

Figure 7: Outlet temperatures of streams which flow through heat exchanger LP1 as a function of power generated

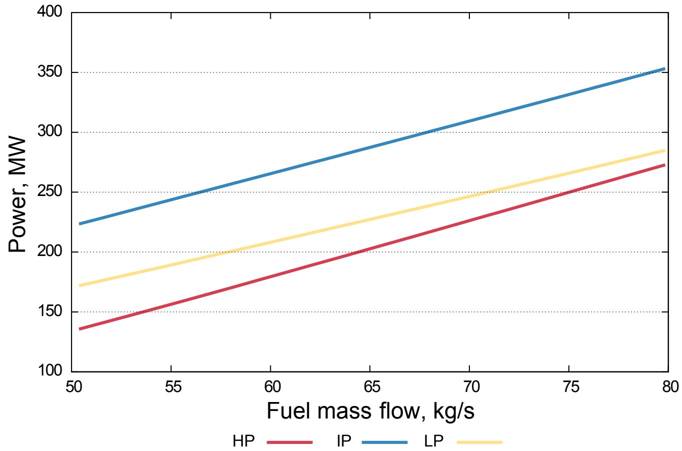

Figure 8: Power generated by the each part of the steam turbine

of the turbine generates higher power. During off-design operation all parts of steam turbine work in almost identical fashion to the original case—see Fig. 9. Achieved exhaust gases temperature is in the range 51–43°C, what means that steam condensation occur here. Thus special attention is needed to avoid negative effect of sulfuric acid which will be created as the result of reaction between sulfur dioxide and water by using acid resistant steels or/and more efficient desulfurizing systems.

It is shown that the influence of the off-design operation of the cycle performance can be neglected.

4. Conclusions

The heat exchanger added to the original system is one of the elements of the heat recuperation system. The start up procedure will predict its operation together with the other heat exchangers placed in the heat cycle to assure the proper temperature of the boiler feedwater. The

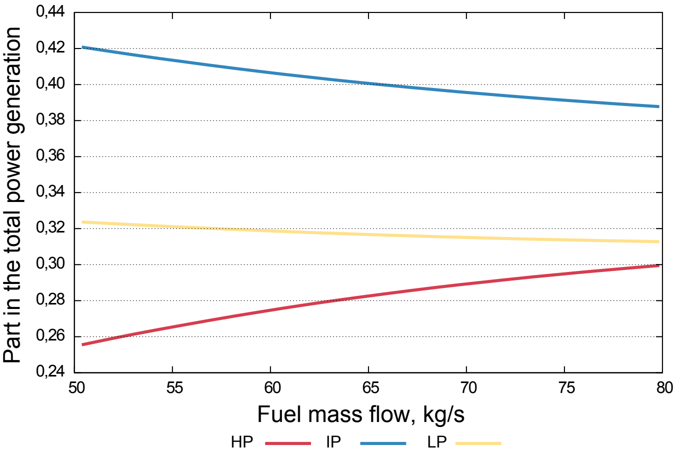

Figure 9: Ratios of each part of the steam turbine in total power generation

added heat exchanger can be started up prior to running the whole cycle due to the fact that usually a boiler starts delivering low grade heat prior to the steam turbine.

Acknowledgments

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