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Thermodynamic Analysis of a Thermal Cycle of Supercritical Power Plant

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The study presented in this paper deals with the analysis of operating conditions of a modern supercritical power plant. The 460 MW reference thermal cycle, which is based on the Łagisza supercritical, coal fired power plant was selected for this study. The thermodynamic analysis was performed with the use of the industrial software package IPSEpro, designed for power plant engineering. The main objective was to demonstrate the role of supercritical parameters in enhancing the efficiency of the thermodynamic process. It was done among the others by the comparative analysis of two thermal cycles, one working with standard and the other with supercritical parameters. Apart from nominal operating conditions part load operation was analyzed.

Keywords: Power plant, steam cycle, system analysis, IPSEpro

1. Introduction

The increasing energy consumption is the reason that the coal is still one of the main sources of electricity all around the world and cannot be instantaneously replaced by renewable energy sources. However due to CO_2 emission limits and corresponding penalties, the conventional coal–fired power plants with the efficiency lower than 40% (with operating parameters below temperature $T_{kr} = 374.16^o$C and pressure $P_{kr} = 221.15$ bar) become less cost–effective [5]. Such situation appears to be a very serious problem for the power energy industry. The emission of the carbon dioxide can be avoided or mitigated either by development of technology to remove the CO_2 from the combustion process or from the flue gasses or the improvement of the overall efficiency of the power unit. One of the most popular trends in thermal efficiency improvement is raising the key parameters of the steam cycle i.e. a temperature and pressure of the live steam. In this paper the latter approach is considered and analysed.

Successive progress in material technology and increasing demand for power plant efficiency creates the supercritical unit as a new best worldwide solution [1]. According to [6] in the early 1990s Japanese as the first run the supercritical plant with operating temperatures near to 600^oC. By the late 1990s the concept of ultrasupercritical (USC) was introduced and steam parameters raised to 290 bar under the temperature 600^oC. Bugge et al. [2] compared various ultrasupercritical power plants being in service and under construction. The efficiency reported under this investigation varies from 40% to almost 50%. It is expected that if the operating temperature reaches 700^oC, the efficiency of the power plant will exceed 50% [1].

The primary objective of this work was to demonstrate the effectiveness of supercritical steam cycle based on the reference 460 MW unit [4]. It was done by the comparison of the performance of steam cycle with the one working with standard parameters (T = 540^{o}C, P = 183 bar). The secondary aim was to analyse the variation of process parameters under the part load conditions. The analysis presented in this paper was done with the heat– and mass–balance commercial software IPSEpro ver. 4.0.

2. Characteristic of the reference cycle

The scheme of supercritical 460 MW thermal cycle prepared in IPSEpro environment is presented in Fig. 1. It was developed on the basis of steam cycle discussed by Chmielniak and Zięblik [4]. Shaded areas show the location of a boiler, particular groups of turbine stages and regeneration systems. Superheated steam leaves the boiler with the temperature of 560^oC under the pressure 275 bar and 361kg/s mass flux and enters the high pressure (HP) turbine, where it is expanded to the pressure equal 50.26 bar. At HP section part of steam is extracted to the regenerative feed water heating system, so at its exit the mass flow rate is reduced to 305.6kg/s. Before the steam enters the intermediate pressure (IP) section it is reheated from 315^0C to 580^oC. In the IP section the steam is expanded to 266.74^{o}C at a pressure 5.55 bar and than to temperature 35.9^{o}C and pressure 0.059 bar in low pressure (LP) section. The IP section has two extractions to the regeneration system, so the mass flux leaving this section is reduced to 258.5 kg/s, while LP section has three extractions and the mass flow leaving this section equals 206.5 kg/s. In the condenser the water is subcooled to 32.5^{o}C and then is pumped under 22 bar pressure into LP–regeneration section. Here the feed water is re–heated to 158.8^{o}C and its flux rises to 277.4 kg/s due to outcomming fluxes from the LP turbine extractions. The flow with such parameters enters a deaerator, where undesired steam ingredients are removed (mainly oxygen and carbon dioxide). The water leaving the deaerator is once more pumped up to the 329 bar and enters the HP–regeneration section, where it is re–heated to 289.6^{o}C and feeds the boiler under the pressure 319 bar. The structure of the given thermal cycle includes the steam turbine, which drives the main supply pump and the regeneration system expanded by section, which uses the heat recovered from exhaust gases.

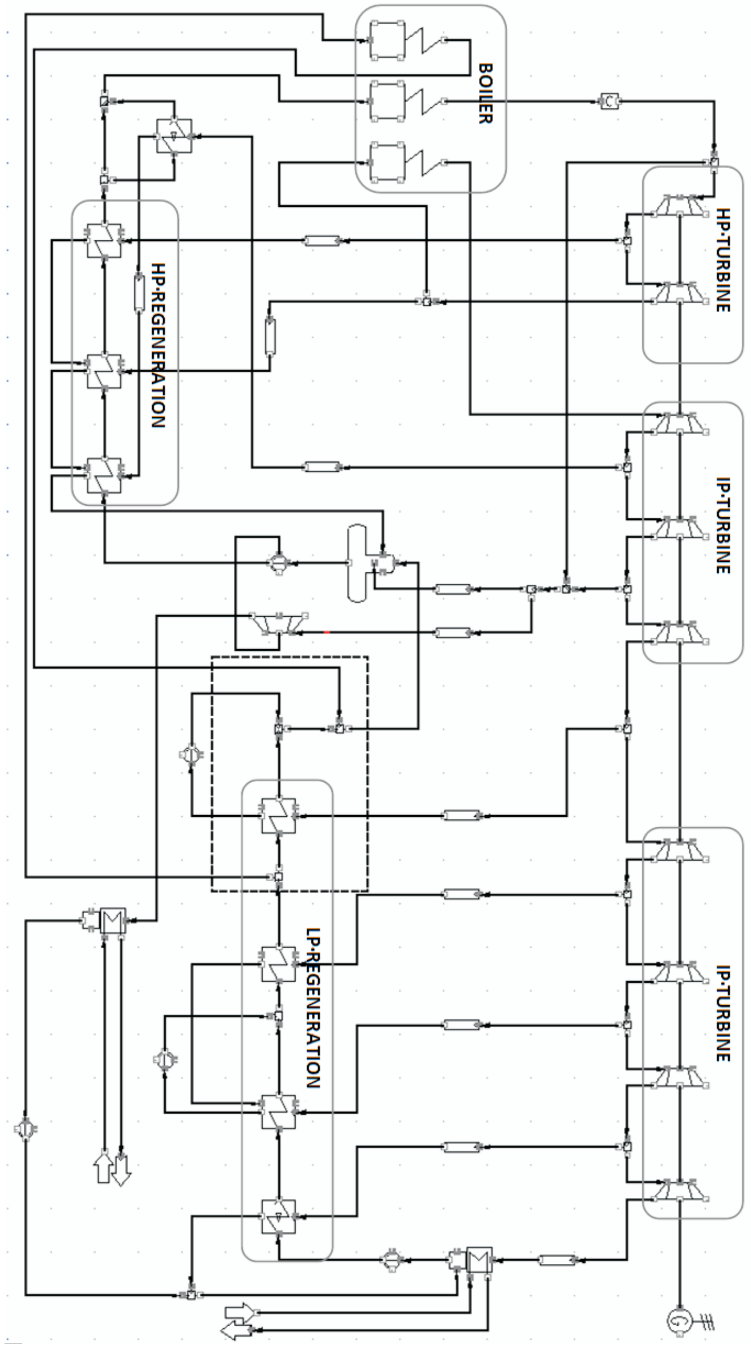

Figure 1 Supercritical thermal cycle for 460 MW unit

3. Results

3.1. *Full load operation with different configurations of reference supercritical steam cycle*

In the first stage the thermodynamic cycle was modelled for nominal operating conditions and for this purpose the so called design type calculations were performed. Efficiencies assumed for the main power plant components are summarised in Tab. 1.

Table 1 Efficiency of power plant components

Efficiency [%]	Turbines			Pumps	Generator
	HP	IP	LP		
Isentropic	88.9	91.9	87.2	85.25	
Mechanical	99.9	99.9	99.9	90	99.6
Electrical					99.2

The main goal was to evaluate, the impact of the thermal circuit structure on the efficiency and the power output. The particular attention was given to the assessment how an electrical power and a cycle efficiency is influenced by the drive type of a main supply pump (steam turbine / electric motor), the inclusion in the regeneration system of the heat recovered from the exhaust gases as well as leaks in the HP turbine stages. The reference steam cycle abbreviated as RSC was calculated with the assumption of electric drive of the main pump. In the next stage the auxiliary steam turbine was included in circuit (MP&ST – Main Pump and Steam Turbine). At the third stage the cycle was extended with waste heat regeneration (CWHR – Cycle with Waste Heat Regeneration). Finally, the leaks (CL) that occur in the high pressure turbine part, due to high differential pressure, were taken into account.

The comparative analysis was done based on fundamental parameters for the entire cycle such as, net and gross output power and net and gross efficiency. Net output power P_N is a power produced by generator, while gross output power P_G is defined as:

$$P_G = P_N + P_T \tag{1}$$

where P_T is a power generated by steam turbine driving the main pump. So, now efficiencies of the thermal cycle were calculated respectively:

$$\eta_N = \frac{P_N}{Q_I} \qquad \text{net efficiency} \tag{2}$$

$$\eta_B = \frac{P_G}{Q_I} \qquad \text{gross efficiency} \tag{3}$$

where Q_I heat input supplied to the boiler and kept constant for all cases.

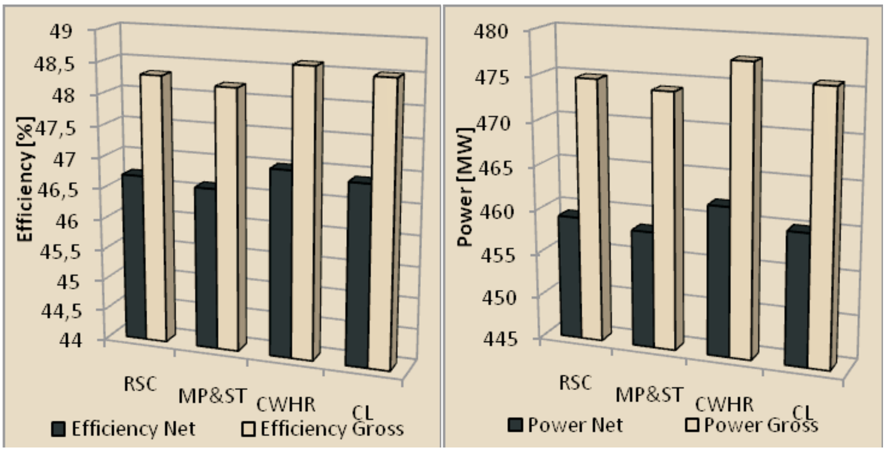

Figure 2 Influence of steam cycle complexity on basic parameters

Additionally, to check the impact of the thermal circuit structure on the regeneration system it was decided to analyse the heat balance of one given heat exchanger. For this purpose the specific control volume was defined around the last heat exchanger of low pressure regeneration system, shown by dashed line in Fig. 1. The results of calculations are shown in Fig. 2. Application of the steam turbine instead off electrical motor to drive the main pump must lower the parameters as some portion of steam is taken from the main stream and directed to auxiliary turbine. This drop is however, not so crucial and in addition one should remember that the output power drop by 0.8 MW is supplemented by lower demand for the power needed for power plant own needs. In the next stage, the thermal scheme was complemented by the CWHR system. The amount of additional heat recovered from exhaust gases and brought to the cycle equals 13.420 MW. This amount of heat was not taken into account for η calculation. One may notice (Fig. 2) that it caused an evident rise of both efficiency as well as of output power. An increase of efficiency by 0.3% which is accompanied by the growth of output power of 4 MW confirms that it is one of the best methods to improve power plant operation. One must take into consideration that in the real installation the unavoidable leakage in HP turbine is present. As the result, some mass flux does not perform the work. In the considered circuit the leakage was modelled by extraction of 2.568 kg/s of steam from the before HP turbine, which is supplied than to a deaerator inlet. Such a modification causes the drop of efficiency by 0.1% and drop of the power output by 2 MW. It is an important decrease of global parameters and should be taken in to account in the course of modelling and design of the realistic object.

The analysis of heat balance of the heat exchanger shows, that noticeable impact on its operation has only the addition of the waste heat to the regeneration system (Fig. 3). The stream of 47 kg/s of feed water is taken before the last low–pressure regeneration heat exchanger, then it is heated in an additional heat exchanger by flue gases and then supplied to the main circuit before the deaerator. The additional heat flux causes the decrease of heat demand from steam extracted from turbine and supplied to the considered heat exchanger and causes the increase of the output power. In Fig. 3 is seen that with the almost constant value of feed temperature one may also observe a rise of drain temperature of the condensate. The change of the main pump drive and the inclusion of the leakage in HP turbine do not have significant influence on the operation of the heat exchanger.

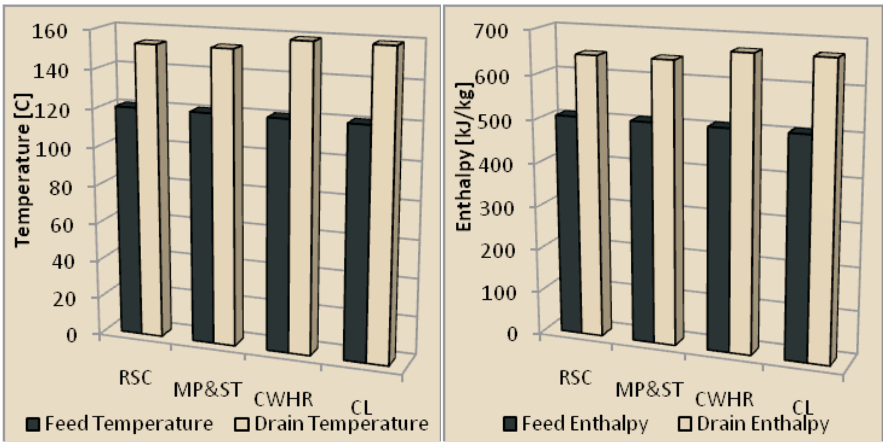

Figure 3 Influence of steam cycle extension on basic parameters of heat exchanger

3.2. *Comparison of supercritical with under–critical stem cycle*

According to Chmielniak and others [3] the supercritical steam cycles provide higher efficiency and higher output power, when compared to under–critical steam cycles. To demonstrate such an effect, based on final structure of the steam cycle (SC) presented in the Fig. 1 the under–critical steam cycle was created. It is assumed the net output is constant. The calculations were realized in two stages. In the first one (UC1 case), the superheated live steam leaves the boiler with the temperature of 540^{o}C under the pressure 183 bar. Then it is expanded to the same level like in the reference case 54.76 bar. Reheated temperature was equal to 540^{0}C in both cases. It is assumed the net output power is constant. In the second stage (UC2 case) the pressure of the reheated steam was reduced to 42 bar and the boiler efficiency was lowered from 94.5% to 88%. The parameters of the live and the reheated steam were taken from the RAFAKO data sheet [7]. The comparative analysis was conducted for net efficiency and output power. The results are shown in Fig. 4. It is seen that the application of supercritical parameters significantly increases the efficiency, which is a result of the increase in the average temperature of heat supplied to the circuit. From the view point of efficiency it is also beneficial to increase the boiler feed water temperature at the boiler input. It is also clear (see Fig. 4) that

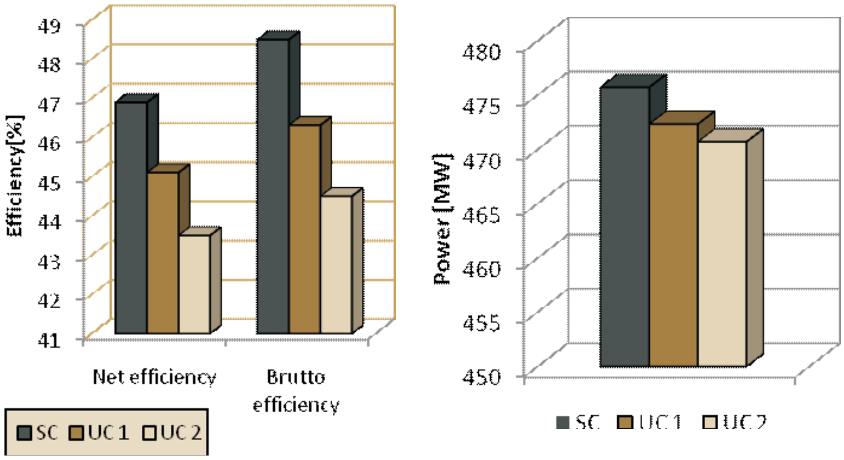

Figure 4 Comparison of supercritical and under–critical steam cycle

the steam cycle performance is very sensitive not only to parameters of the live steam but also to the efficiency of the boiler. It is worth to note, the lower demand for power of the main pump in case of under–critical steam cycles which is equal 12.3 MW for UC1 and 10.7 MW for UC2 in comparison with 15.7 MW for supercritical cycle. It results from the lower pressure rise required.

3.3. *Part load operation*

Apart from the work on the nominal loading it is important to know the response of the system to partial loading. In the course of calculations the load was changed from 100% to 45% by every 5%. For this purpose the off–design simulation of the steam cycle presented in Fig. 1 were carried out. The IPSEpro default characteristic describing the change of the turbine efficiency versus the mass flow rate for part load were used. In Fig. 5 the net power and the efficiency are given as a function of part load. With a drop of load, the power decreases from 460 MW to 207 MW and its distribution can be described by the linear relationship. The efficiency drops from 48.5% to 42.6% and its variability is clearly non–linear. It is seen that the steeper decline starts from 55% of loading. It means that the steam cycle can operate with reasonable efficiency in the partial load range between 55% and 100%

With the decrease of output load, the required heat input supplied to a boiler decreases as well. As it is seen in Fig. 6 the relation between two above parameter is linear. At the same time the thermal power, which is extracted in the condenser and HTEX (all heat exchangers) drops as well. For 45% part load power, the cycle request 488 MW less heat input than for full load. Consequently the condenser and HTEX heat exchangers transfer respectively 216 MW and 165 MW less heat.

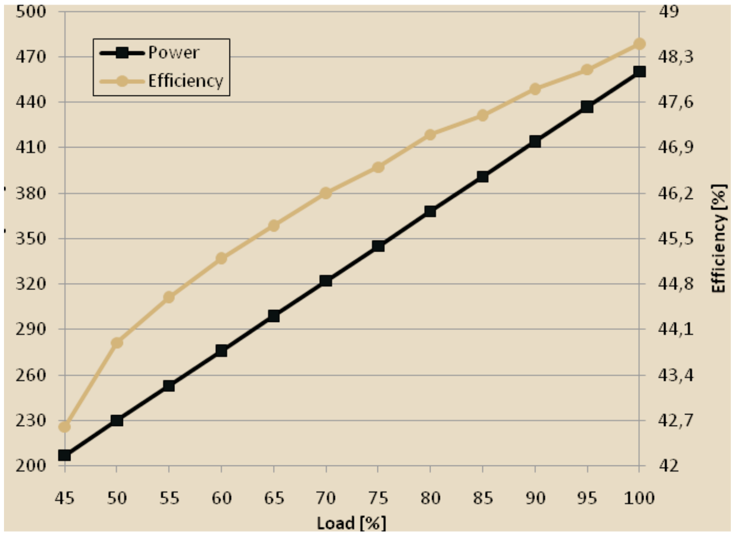

Figure 5 Output power and efficiency at part load

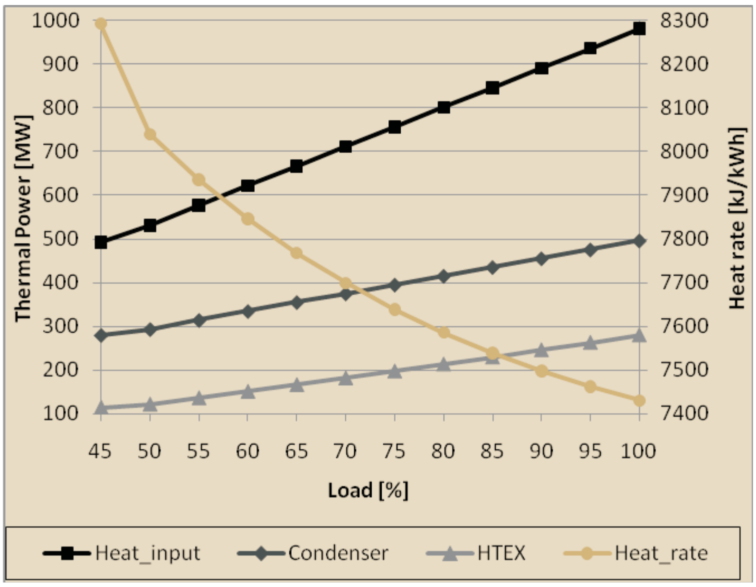

Figure 6 Influence of power heat inlet and heat rate for part load calculation

Once of the most useful parameters to determine the quality of the steam cycle is the heat rate, which is defined as:

$$q = 360^o \frac{Q_I}{P_N} \tag{4}$$

The heat rate, similarly to efficiency, has non–linear characteristic. During the part load calculation it increases by ∼12% from 7410 kJ/kWh to 8300 kJ/kWh.

4. Conclusions

The results presented in this paper show the variation of global parameters in the course of development of supercritical 460 MW thermal cycle. Expanding the reference thermal cycle with such an additional elements as MP&ST, CWHR and CL allows to demonstrate how this expansion affects the output power and efficiency. Comparative analysis of under–critical and supercritical steam cycles demonstrates almost 3.4% rise off efficiency in case of super–critical concept. The analysis of part loading shows that that steam cycle can operate between 55% and 100% load with reasonable efficiency.

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