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Novel method for determining optimal heat-exchanger layout for heat recovery steam generators

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ABSTRACT

A new method for determining the optimal heat-exchanger layout in a heat recovery steam generator and its operating parameters are presented in this paper. A robust mathematical model is developed, where arbitrary steam-pressure levels and steam-reheating levels can be set. The method considers all the possible heat-exchanger layouts, in both serial and parallel arrangements of steam pressure levels or steam reheating levels. The maximum thermodynamic efficiency of the steam-turbine cycle is set as the objective function. The results show that the optimal high pressure in heat recovery steam generator without reheating is in the region of subcritical pressures, whereas that for a heat recovery steam generator with reheating is in the region of supercritical pressures. In the case of similar water or steam temperature profiles in the heat exchangers of different steam pressure levels or reheating level, from a thermodynamic viewpoint, it is justified to use a parallel heat-exchanger arrangement.

1. Introduction

The electrical energy consumption in the world increases each year, i.e., the consumption of fossil fuels increases yearly. The use of fossil fuels causes greenhouse gas emissions such as CO_2 emissions, which, according to many scientists, are one of the main causes of climate change. To reduce CO_2 emissions, the European Union (EU) imposed the "2020 Climate & Energy Package," which is a set of binding legislations for ensuring that the EU meets its climate and energy targets by the year 2020. The main goals of this legislation are a 20% reduction in greenhouse-gas emissions, a 20% improvement in energy efficiency, and obtaining 20% of total produced energy from renewables [1]. The need to improve the thermodynamic efficiency of the combined-cycle power plant (CCPP) is emerging as one of the measures of the proposed EU 2020 package for increasing energy efficiency.

The best modern CCPPs achieve a thermodynamic efficiency of above 60%. Examples are CCPP Irsching 4 in Germany with a thermodynamic efficiency of 60.4% [2] and CCPP Bouchain in France with a thermodynamic efficiency of 62.22% [3]. The thermodynamic efficiency of the CCPP (η_{CCPP}) can be increased in two ways: by increasing the thermodynamic efficiency of the gas-turbine part of the power plant (η_{GT}) or by increasing the thermodynamic efficiency of the steam-turbine part of the power plant (η_{ST}). However, not every increase in η_{GT} or η_{ST} results in a corresponding increase of η_{CCPP}, because increasing η_{GT} does not necessarily increase η_{ST} . The same is valid for an increase in η_{ST}. There is an optimal change in η_{GT}/η_{ST} that results in an increase in η_{CCPP} [4]. Contemporary CCPPs have an almost continuous expansion curve in the gas-turbine part of the cycle, that ranges from approximately 1500 to 600 °C and continue in the steam-turbine part of the cycle, from 600 to 25 °C. Minimal disruption exists, only because of the necessary temperature difference between the flue gas and the fresh steam at the steam-generator outlet. In a heat recovery steam generator (HRSG), unlike in a conventional steam generator, the temperature difference between the inlet flue gas and the fresh steam is relatively small; thus, it comes to pinch-point occurrence [5]. The diagram in Fig. 1 shows the relationship between the temperature profile of the flue gas and working fluid and the exchanged heat flux. In this case, the working fluid enters an HRSG with a temperature of 25 °C, and the flue gas enters the HRSG with a temperature of 600 °C. The flue-gas temperature decreases as it transfers heat to the working fluid, whose temperature increases. At the saturation temperature, the temperature profile of the working fluid (subcritical pressures) remains constant, i.e., the working fluid evaporates. During the evaporation, the specific heat capacity of the working fluid becomes infinite. Because of this phenomenon, which affects the working-fluid temperature profile, it can be said that the pinch point is the result of the increase in the specific heat capacity of water heated in the economizer and

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Nomenclature

g_{rad}	HRSG radiation heat losses to the environment	TP	triple pressure
DP	double pressure	wf	working fluid
fg	flue gas	x_i	proportion of mass flow of lower pressure levels compared to mass flow of HP pressure level
FP	feed pump	Δh	enthalpy increment [J/kg]
h	enthalpy [J/kg]	ΔT_{PP}	pinch point
HP	high pressure	η_{CCPP}	thermodynamic efficiency of CCPP
i,k	pressure-level index	η_{GT}	thermodynamic efficiency of gas-turbine part of power plant
IP	intermediate pressure	η_{HRSG}	thermodynamic efficiency of HRSG
j	reference to the HRSG element within individual pressure level	η_{SC}	thermodynamic efficiency of steam-turbine cycle
LP	low pressure	η_{ST}	thermodynamic efficiency of steam-turbine part of power plant
P_{FP}	feed-pump electrical power [W]	Φ_{GT}	outlet gas turbine heat flux [W]
P_{ST}	steam-turbine electrical power [W]	$\Phi_{HRSG,ex}$	heat flux exchanged in HRSG [W]
q_m	mass flow, [kg/s]	$\Phi_{HRSG,in}$	inlet heat flux to the HRSG [W]
RH	reheating		
SH	superheating		
SP	single pressure		

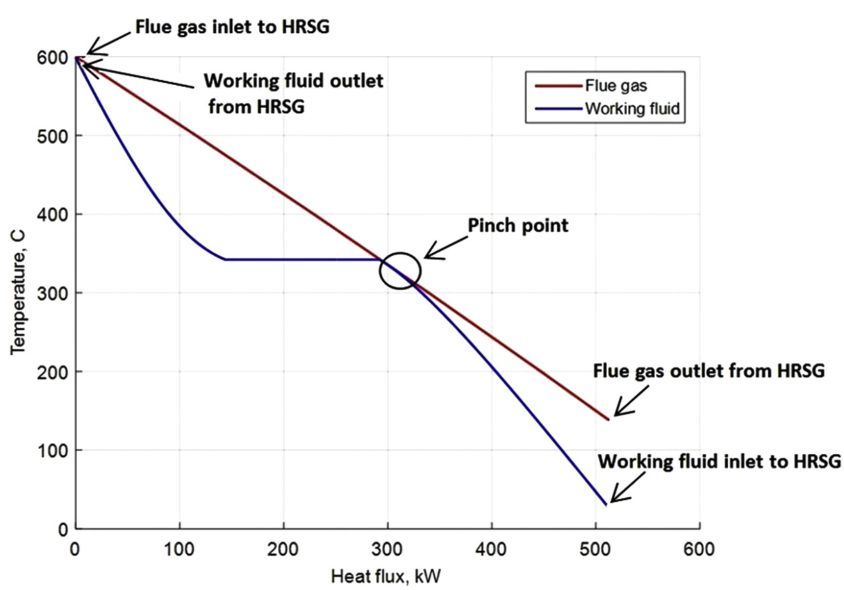

Fig. 1. Flue gas and working fluid temperature profiles inside an HRSG.

its infinite value during evaporation on one hand and a result of the small temperature difference between the inlet flue gas and the fresh steam on the other hand. The increase of the aforementioned temperature difference always reduces the pinch-point effect. A consequence of the pinch-point effect is the inability to cool the flue gas to temperatures close to the inlet feed water temperature.

This reduces the thermodynamic efficiency of the HRSG (η_{HRSG}). Thus, to achieve a high η_{ST}, it is necessary for the η_{HRSG} and the thermodynamic efficiency of the steam cycle (η_{SC}) to be high, which are mutually opposing parameters in HRSGs. Therefore, it is necessary to determine the optimal relationship between these two values. This optimal relationship between η_{HRSG} and η_{SC} can be achieved only via the simultaneous optimization of the heat-exchanger layout of the HRSG and the working-fluid parameters (pressure, temperature, mass flow). Modern HRSGs have more than one pressure level. Additional pressure levels offer lower pressures and lower superheating temperatures. They allow further utilization of the flue-gas waste heat in contrast to systems with only one pressure level [6]. Unfortunately, the η_{SC} of these additional pressure levels is lower than the η_{SC} of the first pressure level. Many studies have been performed on the optimization of a CCPP. Some studies use a thermoeconomic approach, which is a compromise between improving the thermodynamic efficiency and reducing investment costs. Valdes et al. [7] optimized the combined cycle with HRSGs having one and more pressure levels, using the production cost per unit of generated electricity and the annual cash flow as objective functions. Optimizing the heat-exchanger layout was not an aim of their work. They optimized only the working parameters, such as the pressure and temperature. Katovicz and Bartela [8] optimized a HRSG with triple-pressure (TP) steam and a reheater, analyzing the influence of the fuel price on the optimum operating parameters. The objective function was the net present value of investment. They did not optimize the heat-exchanger layout. Rahim [9] performed a sensitivity analysis for single, double, and TP HRSGs in a CCPP. They performed a parametric analysis of the influence of the working parameters on the