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Numerical optimization of heat recovery steam cycles: Mathematical model, two-stage algorithm and applications

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A B S T R A C T

Most of the advanced integrated energy systems need a heat recovery steam cycle (HRSC), either fired or unfired, that recovers the waste heat from gas turbines and process units in order to generate electric power and supply mechanical power to compressors, heat to endothermic processes, and steam to external users. The key feature of such HRSCs is the integration between the heat recovery steam generator (HRSG) and the external heat exchangers. This paper presents a rigorous mathematical programming model, a linear approximation, and a two-stage algorithm for optimizing the design of integrated HRSGs and HRSCs, simultaneously considering the HRSG together with the heat recovery steam network and the intensive steam cycle variables. A detailed application of the methodology is described for an integrated gasification combined cycle plant with CO_2 capture and results for other interesting plants are reported. A significant efficiency gain is obtained with respect to usual practice designs.

1. Introduction

Complex energy conversion systems and chemical processes like coal and biomass to fuels or integrated gasification combined cycles (IGCCs) degrade a large part (larger than 50%) of the fuel chemical power into thermal power due to a number of exothermic reactions (combustion, water–gas shift, methanation, Fischer-Tropsch synthesis, etc.). This large quantity of thermal power can be efficiently recovered by a heat recovery steam cycle (HRSC). Such a steam cycle is substantially different from those adopted in conventional combined cycles, because the integration with the chemical process calls for the recovery of thermal power from either a heat recovery steam generator (HRSG) and multiple "heat sources" like syngas coolers, the supply of thermal power to multiple "heat users" like endothermic reactors, the supply of steam to multiple "steam users", and the supply of mechanical power to multiple "mechanical users" like compressors. In some cases these external heat sources/users supply/absorb a significant portion of the total recoverable thermal power, even more than that supplied by the steam generator. For example, the HRSC of the Shell IGCC with Carbon Capture and Storage (CCS) reported in Fig. 1 (and described in Section 10) can recover 689.5 MW from the HRSG and 657.5 MW from the syngas coolers, must supply about 42.7 MW to the heat users (reboiler of the Selexol unit, syngas heaters and sour water stripper), and must deliver 488.5 MW of superheated medium pressure (MP) steam for gasification and water gas shift reaction. The recoverable heat is 73% of the coal LHV power and syngas coolers provide 49% of the total recoverable power (the remainder is provided by the gas turbine flue gases). While the HRSG is capable of economizing, evaporating and superheating steam, the external heat sources may not contain all the steam generation phases because they operate on a narrow range of temperatures. For instance, the second syngas cooler downstream of the gasifier operate in the temperature range from 350 °C to 250 °C, and then can economize but not evaporate or superheat high pressure (e.g., ≥120 bar) steam. On the other hand, it is not thermodynamically advantageous to use its thermal power to economize low pressure water. This syngas cooler can be conveniently used to evaporate a fraction of the MP level (e.g., <30 bar) steam mass flow rate. It is sufficient to connect in parallel the syngas cooler with the MP evaporator of the HRSG. Another advantageous option could be to use the syngas cooler to economize high pressure (HP) water and then connect it in parallel with a proper section of the HP economizer of the HRSG. In a flowsheet with several heat/steam sources/users and multiple pressure levels, a large number of integration options are feasible and it is necessary to determine the best set. Furthermore, regarding the syngas cooler mentioned above, what is the optimal mass flow rate of MP and HP water to be

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Nomenclature

CCS	carbon capture and sequestration
CPSO	constrained particle swarm optimizer
DEA	deareator
ECO	economizer
EVA	evaporator
FW	feedwater
HE	heat exchanger
HEN	heat exchanger network
HHV	higher heating value
HP	high pressure
HRSC	heat recovery steam cycle
HRSG	heat recovery steam generator
IGCC	integrated gasification combined cycle
LHV	lower heating value
LLP	low low pressure
LP	low pressure
MILP	mixed integer linear program
MINLP	mixed integer non linear program
MP	medium pressure
NLP	non linear program
PSO	particle swarm optimizer
SH	superheat
RH	reheat
TIT	turbine inlet temperature
TOT	turbine outlet temperature

Symbols

c_p	specific heat capacity at constant pressure (kJ/(kg K))
h	specific enthalpy (kJ/kg)
h_{LHV}	lower heating value of a fuel (kJ/kg)
$\boldsymbol{h}$	vector of enthalpies of steam/water streams (kJ/kg)
$h_{satS}(p)$	specific enthalpy of saturated steam at pressure p (kJ/kg)
$h_{satL}(p)$	specific enthalpy of saturated liquid water at pressure p (kJ/kg)
N	integer value
n	rotational speed of a turbine section (Hz)
n_s	non-dimensional rotational speed of a turbine section
p	absolute static pressure (bar)
$\boldsymbol{p}$	vector of pressures of steam/water streams (bar)
$p_{sat}(T)$	saturation pressure of water at temperature T (bar)
P	thermal, mechanical or electrical power (W)
q	mass flow rate (kg/s)
q_s	non-dimensional mass flow rate of a turbine section
$\boldsymbol{q}_{fuel}$	vector of mass flow rates of fuel for supplementary firing (kg/s)
R	specific gas constant (J/(kg K))
T	temperature (°C)
$\boldsymbol{T}$	vector of temperatures of steam/water streams (°C)
$T_{eva}(p)$	saturation temperature of water at pressure p (°C)
T_{evaLLP}, T_{evaLP}, T_{evaMP}, T_{evaHP}	saturation temperature of respectively LLP, LP, MP and HP steam (°C)
ΔT	temperature difference (K)
v	specific volume of liquid water (m^3/kg)
$\boldsymbol{x}$	vector of variables of the linear program
$\boldsymbol{X}$	vector of molar fractions defining the molar composition of a gas
$\boldsymbol{y}$	vector of variables of the linear program in standard form
w	set-up parameters of the CPSO algorithm
η	mechanical or electric efficiency of components
β	pressure ratio of a turbine sections
η_{iso}	isentropic efficiency of a turbine sections
δ	seal loss coefficient of a turbine sections
γ	vector of variables of the upper level problem

economized, and MP steam to be evaporated? Are there other (additional) attractive heat recovery options? A common design rule is to maximize the production of steam/water which minimizes the heat transfer irreversibility (exergy loss) with the hot process stream (e.g., syngas), i.e., to maximize the production of higher pressure steam. In Section 10 we show that this heuristic rule may not be sufficient to design a HRSC and can lead to sub-optimal solutions because does not consider simultaneously the overall system.

Moreover the integration between the HRSG and the process units may have a large effect on the design of the steam generator and on the optimal intensive variables of the steam cycle (i.e., pressures of evaporators, headers and turbine extractions, and superheat, reheat and feedwater temperatures). For instance, the Shell IGCC of Fig. 1 does not have the same optimal pressures of levels of a conventional triple pressure level combined cycle because a large quantity of thermal power is recovered from the process. Therefore the integration HRSG – external HEs and the intensive variables of the steam cycle must be optimized simultaneously.

In addition, the optimization of the steam cycle can be heavily influenced by the limitations and performance maps of the steam turbines. Generally large size steam turbines can be only partially customized (i.e., may not be designed or completely modified for the specific application) because either they are selected from the vendor catalogue or they are already given (e.g., retrofit of an old steam power plant). As a consequence, the design of the integrated HRSC must take into account also the constraints due to the matching of the HRSC with the selected steam turbines: if the size and geometry of the steam turbines are already given, the steam mass flow rates and the isentropic efficiencies of such machines vary significantly with the rotational speed, the inlet pressure and temperature, and the outlet pressure (see Section 3.4). Therefore it is not correct to assume a constant isentropic efficiency and neglect the effect of the steam mass flow rate on the turbine performance and outlet condition (i.e., enthalpy and temperature). Instead, it is necessary to consider the nondimensional performance maps which are generally provided by the producer of the turbines. These maps allow to find the optimal trade-off between the turbines performance and the HRSC with respect to steam mass flow rates, pressures and temperatures.

On the other hand, if some steam turbines are designed specifically for the plant, a detailed turbine model is needed to predict the expansion efficiency, the labyrinth seal losses and the mechanical and electrical efficiency. As shown by the work of Lozza (1990), it is extremely incorrect from a thermodynamic point of view to assume a constant isentropic or polytropic turbine efficiency. For instance, an increase of the turbine inlet pressure requires an additional high pressure stage with low blade height and then a much lower (isentropic or polytropic) efficiency then the average. On the other hand, when the expansion line enters the two-phase zone, the presence of water droplets penalizes significantly the efficiency of the last stages. Such considerations can heavily affect the optimization of the cycle parameters (limiting the admission pressure in order to have high pressure stages with larger blade heights, and high steam qualities at turbine outlet). Also an accurate estimation of the labyrinth seal losses is important to compute the power generated by the turbine and to quantify the outlet steam mass flow rate. For instance, the steam mass flow rate entering a reheater, and