International Journal of Heat and Mass Transfer 67 (2013) 153–158

Contents lists available at SciVerse ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Power from a hot gas stream with multiple superheaters and reheaters

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ARTICLE INFO

Article history:
Received 24 May 2013
Accepted 18 July 2013
Available online 30 August 2013

Keywords:
Constructal design
Superheater
Reheater
Steam turbine
Power plant

ABSTRACT

In this paper we explore the opportunity to maximize the production of power in a steam-turbine power plant by properly configuring the hardware at the interface between the stream of hot gas produced by the furnace and the steam that circulates through the power producing cycle. The interface consists of four heat exchangers, superheaters and reheaters, in parallel flow and counter flow. The search for better configurations is based on constructal design, and consists of searching for the distribution of heat exchanger surface (number of heat exchangers, types, sizes) such that the total power output of the turbines is maximum, subject to fixed total size for the heat transfer surface, and fixed maximum allowable steam heat exchanger wall temperature. The emergence of the more effective configurations is documented along with the migration of the spot of maximum temperature along the flow path.

1. Introduction

The most basic question in the design of steam power plants is how to obtain more work per unit of fuel consumed. This is why the thermodynamic optimization of heat and fluid flow processes and devices is a permanently active field, which is reviewed regularly in research meetings and books (e.g., Refs. [1–16]).

The key aspect of this challenge is that in the thermodynamics of power cycles it is routinely assumed that the energy input is already available as heat transfer. In almost every application, however, the energy input is initially carried into the power plant by a hot stream-e.g., a mixture of fuel and oxidant, hot products of combustion, exhaust gases, or hot geothermal steam. To convert the energy of the hot stream into a heat input for the cycle executed by the working fluid of the power plant is the function of one or more heat exchangers. The interface between the power cycle, which rests on thermodynamics, and the stream of hot gas requires a heat transfer perspective, which stresses the role played by the physical configuration of the flow system, the design of the hardware.

In this paper, we propose to focus on the physical configuration of the interface between the stream of hot gas and the working fluid (the steam) that circulates through the power plant. The approach is based on constructal design [17], which is the search for progressively better flow configurations that lead to greater performance at the level of the power plant. At the interface between the hot gas and the steam, this is achieved by properly distributing the heat transfer surface along the gas stream, that is by allocating the finite heat transfer surface to the various heat exchangers that constitute the interface.

2. Thermodynamic model

Consider an example of industrial tower type boiler [13,14], as depicted in Fig. 1. The objective of the analysis is to determine the best allocation of the heat exchange surfaces to the superheaters (SH) and reheaters (RH) such that the total power output at the high pressure and low pressure turbines is maximum. The sequence in which the different heat exchangers are assembled is maintained, in accord with Fig. 1.

Fig. 2 is a sketch of the two fluid paths, the hot gas C_g and the steam C_s. The heat exchanger literature [18] shows that a cross flow heat exchanger with many passes behaves as either a counter flow heat exchanger, or a parallel flow heat exchanger, depending on the stream direction. Note the temperature nomenclature defined in Fig. 2. The gas temperature decreases from T_{in}, which is fixed, to T_{out}, which depends on the heat exchanges with the superheaters and reheaters. The gas temperatures T_{g1}, T_{g2}, and T_{g3} occur respectively between the superheater in parallel flow, the reheater in parallel flow, the superheater in counter flow, and the reheater in counter flow. The steam enters at the known temperature T_1 and reaches T_9 at the exit from the low pressure turbine.

There are two superheaters: the first is arranged in parallel flow, and the second in counterflow. The two reheaters are inserted between the two superheaters. The first in counterflow, and the

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2013.07.048

Nomenclature

A	heat transfer area, m^2
C	heat capacity rate, $W\,K^{-1}$
c_P	specific heat at constant pressure, $J\,kg^{-1}\,K^{-1}$
m	mass, kg
N	number of heat transfer units, Eqs. (7)–(10)
P	pressure, Pa
Q	heat transfer rate, W
R	ideal gas constant, $J\,kg^{-1}\,K^{-1}$
T	temperature, K
U	overall heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
W	power, W
x	thermal conductance allocation ratio between superheater and reheater
y	thermal conductance allocation ratio between parallel flow and counter flow heat exchangers

Greek symbols

α	temperature ratio, T_{5is}/T_5
β	temperature ratio, T_{9is}/T_9
ΔT	temperature differences, K
ε	effectiveness, Eqs. (17) and (20)

Subscripts

g	gas
in	inlet
is	isentropic expansion
max	maximum
min	minimum
out	outlet
R	reheater
s	steam
S	superheater
0	ambient

Superscripts

$(\tilde{\ })$	dimensionless
$(\dot{\ })$	per unit time

second in parallel flow. The model that we propose here consists of summing up and maximizing the power output of the two turbines by varying the complex configuration of heat exchangers.

The important physics is that the steam temperatures are *variables* that depend on the sizes $(UA)_S$ and $(UA)_R$. In other words, they cannot be fixed. One constraint is the total size

$$(UA)_S + (UA)_R = UA, \text{ constant} \tag{1}$$

or

$$(UA)_S = xUA, \quad (UA)_R = (1-x)UA \tag{2}$$

where x varies between 0 and 1. In addition, we write for the superheater

$$(UA)_{Spf} + (UA)_{Scf} = (UA)_S \tag{3}$$

$$(UA)_{Spf} = y_S(UA)_S, \quad (UA)_{Scf} = (1-y_S)(UA)_S \tag{4}$$

where y_S is the fraction that accounts for the allocation of size between the two superheaters. For the reheater we write

$$(UA)_{Rpf} + (UA)_{Rcf} = (UA)_R \tag{5}$$

$$(UA)_{Rpf} = y_R(UA)_R, \quad (UA)_{Rcf} = (1-y_R)(UA)_R \tag{6}$$

where y_R is the size allocation fraction of the reheaters. The number of heat transfer units for the superheater is defined as

$$N_S = \frac{(UA)_S}{C_{min}} \tag{7}$$

The number of heat transfer units for the superheater in parallel flow and in counter flow, N_{Spf} and N_{Scf},

$$N_{Spf} = \frac{(UA)_{Spf}}{C_{min}}, \quad N_{Scf} = \frac{(UA)_{Scf}}{C_{min}} \tag{8}$$

Similarly, for the reheater we write

$$N_R = \frac{(UA)_R}{C_{min}} \tag{9}$$

$$N_{Rpf} = \frac{(UA)_{Rpf}}{C_{min}}, \quad N_{Rcf} = \frac{(UA)_{Rcf}}{C_{min}} \tag{10}$$

Alternatively, the overall size constraint is expressed as

$$N_S + N_R = N, \text{ constant} \tag{11}$$

or

$$N_S = xN, \quad N_R = (1-x)N \tag{12}$$

For the superheater and reheater we write similarly

$$N_{Spf} = y_S \cdot N_S \tag{13}$$

$$N_{Scf} = (1-y_S) \cdot N_S \tag{14}$$

$$N_{Rpf} = y_R \cdot N_R \tag{15}$$

$$N_{Rcf} = (1-y_R) \cdot N_R \tag{16}$$

The effectiveness relations for the four heat exchangers are [18]

$$\varepsilon_{Scf} = \frac{1-\exp\left[-N_{Scf}\cdot(1-r)\right]}{1-r\cdot\exp\left[-N_{Scf}\cdot(1-r)\right]} \tag{17}$$

$$\varepsilon_{Rcf} = \frac{1-\exp\left[-N_{Rcf}\cdot(1-r)\right]}{1-r\cdot\exp\left[-N_{Rcf}\cdot(1-r)\right]} \tag{18}$$

$$\varepsilon_{Spf} = \frac{1-\exp\left[-N_{Spf}\cdot(1+r)\right]}{1+r} \tag{19}$$

$$\varepsilon_{Rpf} = \frac{1-\exp\left[-N_{Rpf}\cdot(1+r)\right]}{1+r} \tag{20}$$

where

$$r = \frac{C_{min}}{C_{max}} = \frac{C_g}{C_s} \tag{21}$$

We follow the steam path, and we write sequentially.

2.1. Superheater in parallel flow (S_{pf})

The heat transfer continuity equation reads

$$T_2 - T_1 = r\cdot\left(T_{in} - T_{g1}\right) \tag{22}$$

and