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Comparison of power plant steam condenser heat transfer models for on-line condition monitoring

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HIGHLIGHTS

- Models are based on effectiveness-NTU calculation as single heat exchanger, or in many elements.
- The multi-element condenser model is significantly more accurate at low pressures and close temperature approach.
- With experimental correction factor the difference in accuracy is considerably reduced.
- The HEI standards could not be adjusted by a simple correction to accurately predict heat transfer coefficient.

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ABSTRACT

In this paper heat transfer models for large power plant condenser were examined. The goal was to develop a model capable of predicting not only the condenser pressure but the overall heat transfer coefficient. Such a model can be used for condenser condition monitoring. The results of a two-dimensional (2-D) condenser heat transfer model and single-point, zero-dimensional (0-D) model are presented together with the results from Heat Exchanger Institute (HEI) standards curves. Both 0-D and 2-D models can account for the effects of steam-side pressure drop and in a simplified manner also some effects of tube bundle geometry. For all models an experimental correction as a function of cooling water temperature was implemented to improve their accuracy. The results are presented in comparison with the measured plant data for three different tube bundle geometries, with and without the experimental correction factor. The 2-D model proved to be the most consistently accurate of the models both without the correction, and at varying steam and coolant flow with the correction applied. The results indicate significant local variation of pressure drop related effects, which the 0-D model failed to accurately predict particularly in cases of close temperature approach. In predicting the heat transfer coefficient the HEI model was the least accurate, significantly overestimating the impact of coolant flow rate change, and failing to match the measurements even with a correction applied.

1. Introduction

This paper concerns the development of a heat transfer model for seawater condensers of a large steam power plant, providing a reasonable compromise between computational time and accuracy of the results. The model should provide results fast enough to be used as a part of an on-line condition monitoring system, while also accurate enough for determining foulant layer development inside the seawater tubes, as well as showing likely changes in condenser performance if the plant operating parameters are slightly varied.

Predicting the heat transfer in a large condenser is challenging. Depending on the required accuracy and maximum acceptable computation time, different approaches are possible. The fastest but least accurate option are correlations provided by Heat Exchanger Institute (HEI) or British Electrotechnical and Allied Manufacturers Association (BEAMA), giving the overall heat transfer coefficient U as a function of cooling water flow and inlet temperature, and various tabulated correction factors [1]. While simple, this method fails to account for several phenomena affecting heat transfer and is unlikely to yield accurate results for heat transfer coefficients, though condenser pressure prediction is satisfactory. A calculation based on an average U determined from heat transfer coefficients at average flow conditions was also considered questionable given the vast local variations in flow conditions in a large condenser.

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Nomenclature

A	area [m^2]
B	width [m]
C	experimental correction factor for condensation heat transfer coefficient: $C = h_{\text{adjusted}}/h_{\text{correlation}}$ [−]
C_f	friction factor [−]
c_p	specific heat in isobaric process [J kg^{-1} K^{-1}]
d	diameter [m]
f	friction factor (Darcy) [−]
F_x	correction term for U in HEI standards [−]
G	mass velocity [kg s^{-1} m^{-2}]
g	gravitational acceleration [m s^{-2}]
h	1. heat transfer coefficient [W m^{-2} K^{-1}]2. specific enthalpy [kJ kg^{-1}]
h_{fg}	latent heat of condensation [kJ kg^{-1}]
i_{max}	number of calculation segments in tube axis direction [−]
j_{max}	number of tube rows in steam flow direction [−]
k	thermal conductivity [W m^{-1} K^{-1}]
L	length [m]
$\dot{m}$	fluid mass flow rate [kg s^{-1}]
N	number of tubes [−]
n	index of calculation elements [−]
NTU	number of transfer units (dimensionless conductance) [−]
Nu	Nusselt number [−]
P	tube pitch [m]
p	pressure [Pa]
Pr	Prandtl number [−]
R	thermal resistance [K W^{-1}]
Re	Reynolds number [−]
R''	thermal resistance per surface area [m^2 K W^{-1}]
s	tube wall thickness [m]
T	temperature [°C]
w	velocity [m s^{-1}]
x	steam quality [−]
U	overall heat transfer coefficient [W m^{-2} K^{-1}]

Greek symbols

Δp	pressure drop [Pa]
ε	heat exchanger effectiveness [−]
Φ	heat transfer rate [W]
μ	dynamic viscosity [Pa s]
ρ	density [kg m^{-3}]

Subscripts

C	cumulative
c	cold (sea water) side
cl	clean
gr	gravity
h	hot (steam) side
i	1. tube inside2. calculation element index in tube axis direction
in	inlet
j	calculation element index in tube row direction
L	liquid phase
o	tube outside
out	outlet
s	tube surface
sh	1. shell2. shear
T	transverse to steam flow direction
tb	tube
TOT	total
TRU	true value (according to measurements)
V	vapour phase

At the other extreme, detailed 2-D and 3-D numerical models have been developed to model the behaviour of both large power plant condensers and laboratory-scale test equipment. Al-Sanea et al. used a single-phase 2-D model [2]. Later Al-Sanea et al. [3] and Bush et al. [4] implemented two-phase 2-D models. A quasi-3-D method was used to model power station condensers by Zhang et al. [5], and laboratory-scale test condenser by Zhang and Bokil [6]. Malin used a 3-D model [7] to model a marine condenser. Ramon and Gonzalez developed a 3-D model of a church window type condenser [8], and Prieto et al. compared the results of similar model to both HEI correlations and a 2-D simplification of the 3-D model [9]. Hu and Zhang developed improvements to turbulence [10] and inundation [11] modelling for numerical condenser simulations. Zeng et al. [12] developed 3-D models of three power plant condenser configurations, and compared the results to HEI correlations. While the 3-D models are likely the most accurate option in the absence of extensive proprietary data available to condenser manufacturers, the difficulties of modelling two-phase flow, phase change and interaction of the two phases will still produce significant uncertainties in the results. For on-line condition monitoring purposes or a use as a component module of a larger power plant model, the computational complexity of such numerical models would also be excessive.

The approaches studied in this paper are a 2-D model based on a geometrical simplification broadly similar to that presented by Prieto et al. in Ref. [9] calculating the condenser as a heat exchanger network of smaller condensers, and a 0-D model based on an average U calculated at average flow conditions. The possibility of an even simpler implementation was investigated by comparing these results to calculation with an average U obtained from the HEI standards for steam surface condensers. All three methods were implemented for three separate condenser types, one of which is similar to the church window type analysed by Prieto et al. All condensers considered are of two-pass configuration with seawater in horizontal tubes.

The 2-D method described in this paper differs from that of Prieto et al. mainly in the treatment of condensation heat transfer, and in the inclusion of an experimental parameter to fit the model to measured performance. In Ref. [9] the vapour phase heat transfer coefficient was determined according to Taborek [13] and the phase change and heat and mass transfer were modelled according to film theory by Colburn and Hougen [14], corrected by Ackermann's factor according to [15]. The condensate film heat transfer coefficient was obtained from Nusselt's correlation for single horizontal tube without vapour shear, originally presented in Ref. [16], and modified by a shear correction from Ref. [17].

In this work it was assumed that given the simplification of actual flow patterns into 2-D or 0-D models and the difficulties of modelling condensate behaviour and the possible formation and effects of inert gas pockets in the tube bundle, a purely theoretical model could not achieve sufficient accuracy. An experimental correction factor C was introduced to fit the model results to measurements by adjusting the condensation heat transfer coefficient obtained from heat transfer correlations (i.e. $C = h_{\text{adjusted}}/h_{\text{correlation}}$). The unadjusted $h_{\text{correlation}}$ is based on correlations of Nusselt number Nu for gravity- and shear-dominated cases. With an experimental correction applied to account for the uncertainties in condensation modelling, it appeared unlikely that the more