International Journal of Heat and Mass Transfer 122 (2018) 765–774

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Comparison of empirical models with an experimental database for condensation on banks of tubes

Mudather Ibrahim Mudather Zeinelabdeen [a,*], Muhammad Sajid Kamran [b], Adrian Briggs [a]

[a] School of Engineering and Materials Science, Queen Mary University of London, Mile End Road, London E1 4NS, UK
[b] Department of Mechanical Engineering, University of Engineering and Technology, Lahore, Pakistan

ARTICLE INFO

Article history:
Received 20 November 2017
Received in revised form 24 January 2018
Accepted 25 January 2018

Keywords:
Condensation
Bank of plain tubes
Empirical models
Vapour shears stress
Film condensate
Inundation
Presence of incondensable air

ABSTRACT

Empirical models for calculating the heat-transfer coefficients for condensation on banks of tubes were compared with the experimental data set obtained by the various previous investigators consisting more than 4000 data points for 6 different condensing fluids and 13 different tube bank configurations. All the banks considered in this study, involved the condensation of the pure vapours, the exceptions are that of the Briggs and Sabaratnam (2003) and Shah (1978, 1981), since their pure vapours consisted of incondensable air. For the forced convection flow region ($F < 3.5$), it was observed that some of the data were underpredicted by the recent models, recommending that the effect of the shear stress due to high vapor velocity overcomes the effect of the inundation on the heat transfer rate while vice versa is the case for the models overpredicting results. Similarly, for the free convection flow region ($F > 3.5$), it is suggested that the data overprediction by some of the models was due to the boundary layer separation and inundation effects, whereas the data underprediction was due to the generation of the turbulence within the condensate film due to high velocity on the condensate film. The inclusion of the inundation effect to a pure forced convection model of Shekriladze and Gomelauri (1966) as recommended by Cipollone et al. (1983) lead the model of Cavallini et al. (1985) to be the most accurate model compared to the other models for steam only. It was found that the Fujii and Oda model (1986) is the most accurate model among the empirical models been demonstrated in this paper, giving an agreement with the experimental data base to within an average absolute of the errors of about 21.5%. It accounts for the effects of the shear stress on the surface of the film condensate and the inundation within the bank.

1. Introduction

Condensers are important and costly pieces of equipment, which play a key role in the power, air conditioning and refrigeration industries. Continuous research in the area of the condensation heat-transfer has aimed to minimise this capital cost while maximising efficiency.

Condensation occurs on a surface when the saturation temperature of a vapour is higher than the temperature of the surface. Generally, there are two types of the condensation: drop-wise or film-wise. Drop-wise condensation takes place when the condensate does not wet the surface but rather accumulates to form the droplets, whereas film-wise condensation occurs when the solid surface is wetted by the condensate and forms a continuous film. Drop-wise condensation produces vapour-side, heat-transfer coefficients up to 20 times higher than for film-wise [1], but it has only ever been maintained consistently under laboratory conditions and even then only for high surface tension fluids such as water. Because of this, in practical condensers it is always assumed film-wise condensation will occur as this produces a conservative design. For film-wise condensation of a saturated pure vapour, it is safe to assume that the vapour-side heat-transfer coefficient is controlled by the thermal resistance of the condensate film, and hence the thickness of this film is critical to the heat-transfer resistance and a thinner film will result in higher vapour-side coefficients.

In 1916, Nusselt [2] obtained theories for determining the heat-transfer coefficients in the case of film-wise condensations of a pure vapour on single tubes. Over the years, developments have been made to the Nusselt [2] theory to account for effects which were neglected in the original model. In the case of a single tube, increasing the vapour velocity leads to vapour shear stress at the surface of the condensate film which can result in a decrease of the average condensate film thickness, which in turn causes the heat-transfer rate to the surface to be increased.

* Corresponding author.
E-mail address: mzeinelabdeen@yahoo.com (M.I.M. Zeinelabdeen).

https://doi.org/10.1016/j.ijheatmasstransfer.2018.01.109
0017-9310/

Nomenclature

c_P	specific isobaric heat capacity
d	outside diameter of plain tube
F	dimensionless quantity, $\frac{\mu h_{fg} g d}{k \Delta T U_{mv}^2}$
G	dimensionlessquantity, $\frac{k\Delta T}{\mu h_{fg}}\left[\frac{\mu\rho}{\rho_v \mu_v}\right]^{\frac{1}{2}}$
g	specific force of gravity
h_{fg}	specific enthalpy of evaporation
k	thermal conductivity
l	length of the tube
M	mass velocity, $\rho_v U_\infty$
N	number of tube rows in a bank
Nu	Nusselt number, $\frac{\alpha d}{k}$
Nu_{gr}	gravity-controlledNusselt number
Nu_{sh}	shear-controlledNusselt number P-dimensionless number, $\frac{\rho_v h_{fg} \mu}{\rho k \Delta T}$
P_t	horizontal tube pitch
P_v	vertical tube pitch
Pr	Prandtl Number, $\frac{c_P \mu}{k}$
$Re_{f\text{-}gr}$	Reynolds number of film based on gravity drained flow
$Re_{f\text{-}u}$	Reynolds number of film based on uniformly distributed flow
$Re_{l\text{-}max}$	Reynolds number of liquid flowing alone through the bundle and based on the maximum velocity (i.e. through minimum cross-sectional area between the tubes)
Re_v	vapour Reynolds number for a horizontal tube, $\frac{\rho_v U_\infty d}{\mu_v}$
$Re_{v,max}$	maximum Vapour Reynolds number for a horizontal tube, $\frac{\rho_v U_{max} d}{\mu_v}$
Re_{tp}	two-phase Reynolds number for a horizontal tube based on the minimum velocity, $\frac{\rho U_\infty d}{\mu}$
$Re_{tp,max}$	two-phase Reynolds number for a horizontal tube based on the maximum void velocity, $\frac{\rho U_{max} d}{\mu}$
$Re_{tp,mv}$	two-phase Reynolds number for a horizontal tube based on the mean void velocity, $\frac{\rho U_{mv} d}{\mu}$
T_v	vapour temperature
T_w	average wall temperature
U_∞	free stream vapour velocity
U_{max}	maximum free stream vapour velocity
U_{mv}	mean void stream vapour velocity
x	vapour mass quality
X_{tt}	Lockhart-Martinelli parameter

Greek symbols

α	mean vapour-side heat-transfer coefficient
Γ_N	mass flow rate of the condensate per unit length drained from Nth tube, (or tube row)
γ_N	condensation rate per unit length on the Nth tube
ΔT	average temperature difference across the condensate film, $T_v - T_W$
ε	void fraction (free volume divided by total volume)
μ	dynamic viscosity
ν	kinematic viscosity
ρ	density

Subscripts

None	property of condensate
N	Nth tube of the bank
gr	gravity-controlled region
tp	two phase
sh	shear-controlled region
v	property of vapour
w	wall surface

Shell and tube condensers, which consist of banks of plain tubes are usually arranged in the in-line or staggered form. An accurate estimation of the vapour-side heat transfer coefficient during condensation within a bank requires an understanding of the physical processes involved, which are more complex than for single tubes, since they involve complex interactions between the vapour and the condensate film and inundation (condensate from higher tubes impinging on lower tubes). These interactions usually result in lower condensation rates lower down the tube bank. This reduction is due to the drop-off in the vapour shear effect as the vapour mass flux decreases and an increase in condensate inundation, both of which lead to an increase of the condensate film thickness. Many researchers have conducted theoretical and experimental studies, and recommended models and empirical correlations to determine the heat-transfer coefficient for tube banks, but there are still uncertainties in estimating the heat-transfer coefficient for tube banks. In this study, several such models will be evaluated against an extensive data base of experimental studies.

2. Experimental studies

Fig. 1 shows the schematics of the test banks [3], and Table 1 summarises experimental database obtained by the various investigators respectively. It can be seen in Fig. 1 that the majority of the test sections are staggered. In some cases, the whole test bank is active (with the exception of dummy half tubes on the walls of the triangular test banks) while in others a single active tube is positioned in a bank of dummy tubes. In both cases artificial condensate inundation is sometimes employed to simulate conditions near the bottom of a large tube bank.

Michael [4] determined the heat-transfer coefficients by measuring the wall temperatures of the instrumented tubes using four thermocouples situated in rows one, two, four, six, eight and ten mid-way along the tube length, while the heat-transfer coefficient for the non-instrumented tube rows were obtained indirectly by subtracting the coolant and the tube's thermal resistances from the overall thermal resistance. Cooling water was delivered individually to each row through the ten turbine flow meters. The heat flux was determined from the coolant flow rate and the temperature rise (measured using the thermocouples located in mixers at the inlets and outlets of the rows). The velocity of an upstream steam was obtained from the steam mass flow rate that was measured by an orifice plate.

Beech [5], using the same apparatus as Michael [4], obtained the experimental vapour-side heat-transfer coefficients for three different test banks at near atmospheric conditions by estimating the tube wall temperature using thermocouples, with four inserted in each of the active tubes within each of the banks. The heat flux of the active condensing rows within each of these banks were calculated using the measured coolant mass flow rate and the temperatures at the inlet and outlet for each of these rows individually. Upstream vapour velocity was again measured using an orifice plate. Beech [5] also carried out experiments using a dummy staggered bank that consisted of 7 rows, where the tube wall temperature of a single active tube, which was located in the fourth row, was measured using six thermocouples embedded in its surface.