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Thermal analysis of plate condensers in presence of flow maldistribution

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Abstract

Flow maldistribution in plate heat exchangers causes deterioration of both thermal and hydraulic performance. The situation becomes more complicated for two-phase flows during condensation where uneven distribution of the liquid to the channels reduces heat transfer due to high liquid flooding. The present study evaluates the thermal performance of falling film plate condensers with flow maldistribution from port to channel considering the heat transfer coefficient inside the channels as a function of channel flow rate. A generalized mathematical model has been developed to investigate the effect of maldistribution on the thermal performance as well as the exit quality of vapor. A wide range of parametric study is presented, which shows the effects of the mass flow rate ratio of cold fluid and two-phase fluid, flow configuration, number of channels and correlation for the heat transfer coefficient. The analysis presented here also suggests an improved method for heat transfer data analysis for plate condensers.

Keywords: Condensation; Heat transfer; Mathematical modelling; Phase change; Transport processes

1. Introduction

Plate heat exchangers (PHEs) have been widely used in food processing, chemical reaction processes and many other industrial applications due to their high effectiveness, compactness, flexibility, and cost competitiveness. Generally speaking, PHEs can serve as an alternative to shell and tube heat exchangers for most applications at low and medium pressures. In recent times, plate heat exchangers have been also widely used in application of two-phase processes, both condensation and evaporation [1–5]. Note that PHEs are often called plate condensers when they are used for vapor condensation. Some examples of industrial applications of vapor condensation were described by Wang and Sunden [6]. However, the knowledge about their performance is very limited due to the relative short experience. In order to further expand this application, work must be carried out to improve the understanding and prediction of thermal and hydraulic performance for PHE condensers.

Flow maldistribution in heat exchangers causes deterioration of both thermal and hydraulic performance. According to Mueller and Chiou [7], there are many factors influencing maldistribution in exchangers: mechanical issues (design, tolerances), self-induced maldistribution due to the heat transfer process itself, fouling and/or corrosion, or the use of predisposed flows such as two-phase flows. The flow distribution from a header (manifold) to parallel channels has become of interest in predicting the heat transfer performance of compact heat exchangers especially in PHEs [8–11]. Often, flow rates through the channels are not uniform and in extreme cases, there is almost no flow through some of them, which result in a poor heat exchange performance. The situation becomes more complicated especially with two-phase flows. In a condenser, uneven distribution of the liquid will create zones of reduced heat transfer due to high liquid flooding. Thus, for the design and optimization of compact heat exchangers based on parallel flow technology, the understanding of two-phase flow distribution is of great importance.

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Nomenclature

A	heat transfer area for effective plate, m^2
A_c	cross-sectional area of the channel, m^2
A_p	cross-sectional area of the port, m^2
a	exponent of Re in Eq. (11)
a_1, a_2	exponents in Eq. (11)
c_p	isobaric specific heat of the fluid, $J\ kg^{-1}\ K^{-1}$
H	dimensionless parameter, $H = \frac{c_p \Delta T}{h_{fg}}$
h	heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
h_f	enthalpy of the liquid, $J\ kg^{-1}$
h_{fg}	latent heat of vaporization, $J\ kg^{-1}$
Ja	modified Jakob number
L	fluid flow length in a channel, m
$\dot{m}$	mass flow rate, $kg\ s^{-1}$
m^2	flow distribution parameter
n	number of channels per fluid
N	total number of channels for the both fluids
NTU	number of transfer units
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
r_h	ratio of heat transfer coefficients
rv	ratio of velocities
rh_f	ratio of the enthalpy of liquid to latent heat of evaporation
T	fluid temperature, K
$T_{1,in}$	inlet temperature of the fluid (1), K
$T_{2,in}$	inlet temperature of the fluid (2), K
$r\Delta T$	dimensionless parameter, $r\Delta T(i) = \frac{\Delta T_i}{\Delta T_{uniform}}$
ΔT	temperature difference, $\Delta T = T_{sat} - T_{c,i}$, K
t	non-dimensional temperature $= \frac{T - T_{1,in}}{T_{2,in} - T_{1,in}}$
u	dimensionless volume flow rate in the channels
X	dryness fraction of two-phase fluid
y	space coordinate for the fluid flow, m
Y	non-dimensional flow path coordinate $= x/L$
z	dimensionless flow path coordinate in the port
ε	effectiveness of plate heat exchanger $= \frac{(\dot{m}c_p)_1 (T_{1,out} - T_{1,in})}{(\dot{m}c_p)_{min} (T_{2,in} - T_{1,in})}$

Subscripts

i	ith channel
in	inlet
out	outlet
min	minimum
uniform	the case of uniform flow distribution amongst channels
w	plate
w_i	ith plate
1	the fluid in odd channel
2	the fluid in even channel

Most of the previous research work in PHEs was conducted with the assumption of plug (one-dimensional) flow inside channels and an equal distribution of fluid into the channels from the port. However, it is a well known fact [12] that the flow maldistribution within the plate heat exchanger and particularly from channel to channel plays a major role for the deviation of the performance compared to the idealised case with flow equally distributed among the channels. The effect of unequal distribution of fluid inside the channels was analysed using a numerical technique by Datta and Majumdar [13] and in more detail using an analytical technique by Bassiouny and Martin [14,15]. The flow maldistribution due to port to channel flow has a tremendous effect on heat exchanger performance which has been investigated analytically and experimentally by Rao et al. [16] and Rao and Das [17]. These investigations have been carried out based on the analytical models for single phase flow distribution from port to channel in a PHE conduit by Bassiouny and Martin [14,15]. From the discussion, it is clear that a proper heat transfer model of the plate heat exchanger based on the unequal flow distribution in channels is required. This is not only to analyse the heat exchangers but also to obtain proper heat transfer data from experiments on heat exchangers with multiple channels, which will be stripped by the influence of number of channels or flow configuration used in the experiment. However, all these studies have been carried out for single phase flow in PHE conduits. So far no study has been reported concerning maldistribution effects on the thermal performance for two-phase plate heat exchangers. This is the main inspiration of the present work. In the present work the plate heat exchanger is thermally modelled with unequal flow in the channels taking the distribution suggested by Baussiouny and Martin [14,15]. While carrying out this investigation the heat transfer coefficients inside the channels are assumed to vary according to general correlations for the Nu for both the cooling and two-phase channels.

For two-phase flow distribution in manifolds, authors like Seeger et al. [18] and Lahey [19] have investigated two-phase (vapor and liquid) distribution in single T-junctions. Only a few studies on the topic of two-phase distribution in manifolds have been presented. Nagata et al. [20] conducted experiments on a horizontal manifold with four vertical upward tubes. Recently, Bernoux et al. [21] experimentally studied the distribution of vapor and liquid phases at the inlet manifold of a compact heat exchanger. Two-phase distribution in the channels are obtained by the use of mass flow rate and mass quality measurements in each channel using transparent windows allowing the observation of two-phase flow pattern of refrigerant 113 at different operating conditions. Vist and Pertersen [22] have conducted an experimental investigation of two-phase flow distribution in compact heat exchanger manifolds consisting of 10 round parallel tubes. Lee and Lee [23] have examined the distribution of two-phase annular flow at header-channels junctions for compact heat exchangers. They focused on the effect of the intrusion depth of the channels to find an opportunity to achieve a uniform flow distribution of the liquid-phase that may be useful in designing evaporators. Two-phase flow structures in compact heat exchangers have been studied by Rong et al. [24] and Watanabe et al. [25]. Watanabe et al. [25] investigated how the heat load on the branches affected the two-phase distribution. However, none of these works studied the effect of flow maldistribution on the thermal performance of plate condensers or evaporators. The present study investigates the flow maldistribution effect on the performance of plate condensers.

There are a limited number of studies dealing with condensation of process fluids in plate heat exchangers. A few classical studies to obtain the heat transfer data for condensation in plate heat exchangers are illustrated by Tovazhnyanski and Kapustenko [26] for steam, Uehara and Nakaoka [27] and Arman and Rabas [28] for ammonia and Kumar [29] and Chopard et al. [30] for refrigerants. A wide range of Reynolds number and different corrugation angles ranging from 30° to 60° have been considered. In the shear-dominated regime, the Akers [31] method is recommended by Kumar [29] for calculating the condensation heat transfer coefficient. The single-phase flow relationships (for corrugated geometry) are then used to calculate the heat transfer coefficient corresponding to the effective liquid mass flux. In the gravity-controlled regime, Thonon and Chopard [32] recommended the use of the Nusselt equation for a plain surface. Therefore, in order to account for the effect of plate corrugations, the Nusselt number is multiplied with an enhancement factor that is obtained from single-phase data (a ratio of heat transfer coefficients for the corrugated surface to a plain surface at the same *Re*). Wang and Zhao [33] have conducted an analysis of the heat transfer and pressure drop for steam condensation and discussed the main factors affecting the average condensation heat transfer coefficient in a plate condenser (PC). These are: total mass velocity, steam content at the outlet of the PC, average condensation temperature difference, average condensation pressure, and Prandtl number of the liquid. Thonon and Bontemps [34] report results on condensation of pure fluids and mixtures of hydrocarbons in a compact welded heat exchanger. Wang and Sunden [6] pointed out that the theoretical prediction of plate condensers is very difficult at this stage and hence most of the investigators have concentrated on experimental investigations. However, most of the literature available for two-phase flow plate condenser concerns prediction of the heat transfer coefficient and friction factor on the two-phase side.

The aim of the present study is to evaluate the thermal performance of falling film plate condensers considering the flow maldistribution from port to channel. A generalized mathematical model has been developed to investigate the effect of flow maldistribution on the thermal performance as well as the exit quality of the vapor of the two-phase flow. To apply the present model, process saturated steam has been considered as the two-phase fluid and water as the cooling fluid. The obtained governing equations have been solved numerically by a semi implicit procedure.

2. The mathematical model

To model the heat transfer process between two-phase and single phase channels in a plate heat exchanger, the following assumptions are considered:

1. The thermophysical properties of the fluids are considered to depend on the average saturation temperature and pressure for the two-phase fluid but no variation has been considered for the single phase fluid.
2. The steam entering the condenser is saturated vapor ($X_{\text{in}} = 1.0$) at the given pressure. This assumption has been incorporated to avoid desuperheating at the entry to the channels which affects the location for the onset of condensation.
3. It is assumed that complete condensation does not take place inside the plate which is often the practical case in the process industry. This is also to avoid the possibility of a liquid filled part near the exit of the channels.
4. Heat transfer is assumed to take place only between the channels and not between the channel and the ports or through the seals and gaskets.
5. The heat exchanger is assumed to be insulated from the surroundings.
6. The flow distribution inside a channel is taken to be uniform giving a ‘plug flow’ of fluid inside each channel.
7. The flow maldistribution from channel to channel has been taken into account through the Baussiouny model [14] but in addition the heat transfer coefficient for the two-phase fluid is considered as a function of flow rate and the ratio of liquid to vapor while that for single phase fluid is assumed to be a function of flow rate only for a given plate geometry.
8. The plates are considered to be thin enough to neglect conduction in them.

Based on the above assumptions, a small control volume of fluid inside the channel and a control volume of solid plate are considered as shown in Fig. 1. For the first and last channels, heat transfer on one side has been considered while the other side has been treated as adiabatic. Energy balances over these control volumes using the above assumptions yield the following fluid and plate equations:

For single phase, fluid 1

$$(\dot{m}c_p)_1 \frac{\mathrm{d}T_i}{\mathrm{d}y} = \frac{h_i A}{2L}(T_{wi} - T_i) + \frac{h_i A}{2L}(T_{wi+1} - T_i) \qquad (1)$$

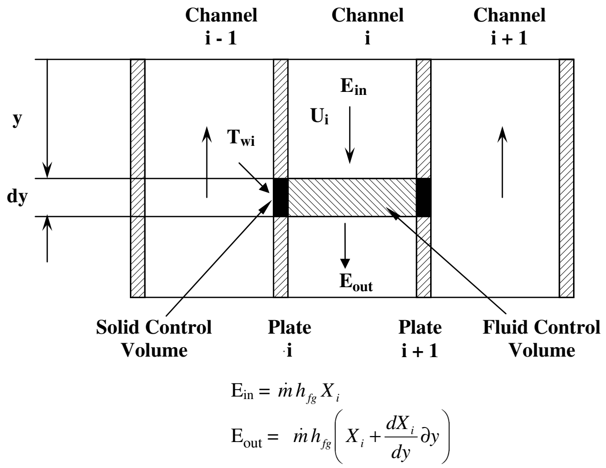

Fig. 1. Control volume for the channels.

For two-phase, fluid 2

$$(\dot{m}h_{\mathrm{fg}})_2 \frac{\mathrm{d}X_i}{\mathrm{d}y} = (-1)^{i-1}\left[\frac{h_i A}{2L}(T_{wi} - T_i) + \frac{h_i A}{2L}(T_{wi+1} - T_i)\right] \tag{2}$$

For the plate

$$\frac{h_{i-1}A}{2L}(T_{i-1} - T_{wi}) + \frac{h_i A}{2L}(T_i - T_{wi}) = 0 \tag{3}$$

These equations can be non-dimensionalised as

$$\frac{\mathrm{d}t_i}{\mathrm{d}Y} = (-1)^{i-1}\mathrm{NTU}(t_{wi} + t_{wi+1} - 2t_i) \tag{4}$$

$$\frac{\mathrm{d}X_i}{\mathrm{d}Y} = (-1)^{i-1}\mathrm{NTU}_i Ja_i(t_{wi} + t_{wi+1} - 2t_i) \tag{5}$$

$$t_{i-1} - t_{wi} = -\frac{h_i}{h_{i-1}}(t_i - t_{wi}) \tag{6}$$

Substituting t_{wi} from Eq. (6) into Eqs. (4) and (5) gives

For single-phase flow channels,

$$\frac{\mathrm{d}t_i}{\mathrm{d}Y} = (-1)^{i-1}\mathrm{NTU}_i\left[\left(\frac{h_{i-1}}{h_{i-1}+h_i}\right)t_{i-1} + \left(\frac{h_i}{h_i+h_{i-1}} + \frac{h_i}{h_i+h_{i+1}} - 2\right)t_i + \frac{h_{i+1}}{h_i+h_{i+1}}t_{i+1}\right] \tag{7}$$

For two-phase flow channels,

$$\frac{\mathrm{d}X_i}{\mathrm{d}Y} = (-1)^{i-1}\mathrm{NTU}_i Ja_i\left[\left(\frac{h_{i-1}}{h_{i-1}+h_i}\right)t_{i-1} + \left(\frac{h_i}{h_i+h_{i-1}} + \frac{h_i}{h_i+h_{i+1}} - 2\right)t_i + \frac{h_{i+1}}{h_i+h_{i+1}}t_{i+1}\right] \tag{8}$$

The advantage of the present approach is that, while maldistribution of the flow is taken into consideration, the flow inside the channel remains one-dimensional. This allows us to incorporate experimental data for condensation inside the plate channels which makes the model applicable for a wide range of commercially available plates for which heat transfer data in the phase change regime are available. In order to enable comparison with the case of uniform flow in all the channels these equations may be written as

$$\frac{\mathrm{d}t_i}{\mathrm{d}Y} = (-1)^{i-1}\mathrm{NTU}rh(i)rv(i)\left[\left(\frac{h_{i-1}}{h_{i-1}+h_i}\right)t_{i-1} + \left(\frac{h_i}{h_i+h_{i+1}} + \frac{h_i}{h_i+h_{i-1}} - 2\right)t_i + \frac{h_{i+1}}{h_i+h_{i+1}}t_{i+1}\right] \tag{9}$$

$$\frac{\mathrm{d}X_i}{\mathrm{d}Y} = (-1)^{i-1}\mathrm{NTU}_u H\left(\frac{\dot{m}_1}{\dot{m}_2}\right)r\Delta t(i)rh_{\mathrm{fg}}(i)rv(i)rh(i) \times\left[\left(\frac{h_{i-1}}{h_{i-1}+h_i}\right)t_{i-1} + \left(\frac{h_i}{h_i+h_{i+1}} + \frac{h_i}{h_i+h_{i-1}} - 2\right)t_i + \frac{h_{i+1}}{h_i+h_{i+1}}t_{i+1}\right] \tag{10}$$

The non-dimensional variables in the equations above are

$$\mathrm{NTU}_u = \frac{hA}{2(\dot{m}c_p)_{\min}};\quad Y = \frac{y}{L};\quad t = \frac{T - T_{1,\mathrm{in}}}{T_{2,\mathrm{in}} - T_{1,\mathrm{in}}}$$

$$rv(i) = \frac{u_{\mathrm{uniform}}}{u_i};\quad rh(i) = \frac{h_i}{h_{\mathrm{uniform}}};\quad Ja(i) = \frac{\dot{m}_c c_p \Delta T}{\dot{m}_s h_{\mathrm{fg}}}$$

$$H = \frac{c_p\Delta T}{h_{\mathrm{fg}}};\quad r\Delta T(i) = \frac{\Delta T_i}{\Delta T_{\mathrm{uniform}}};\quad rh_{\mathrm{fg}}(i) = \frac{(h_{\mathrm{fg}})_{\mathrm{uniform}}}{(h_{\mathrm{fg}})_i}$$

Considering Eq. (10), the flow rate dependence of the heat transfer coefficient considering constant fluid properties $rh(i)$ can be expressed as

$$rh(i) = \frac{h_i}{h_{\mathrm{uniform}}} = \left(\frac{u_i}{u_{\mathrm{uniform}}}\right)^a \quad \text{for single phase fluid}$$
$$= \left(\frac{u_i}{u_{\mathrm{uniform}}}\right)^{a_1}\left(\frac{\rho_{\mathrm{uniform}}}{\rho_i}\right)^{a_2} \quad \text{for two-phase fluid.} \tag{11}$$

From Eq. (11), one can observe from the modified Boyko–Kruzhilin correlation [3,32,33,35], that the heat transfer coefficient of the two-phase flow (assuming all the mass is flowing as liquid) may depend on the flow rate and the density correction factor (the ratio of the fluid liquid density to mean density of the liquid/vapor phase) of the two-phase fluid in the shear-dominated regime of film condensation.

It should be mentioned here that the dependence on Prandtl number does not change h because the property variation for the single phase fluid with temperature is not considered in the present model. Thus Eqs. (9) and (10) are reduced to the following form:

for $i = 2, 3, \ldots, N-1$

$$\frac{\mathrm{d}t_i}{\mathrm{d}Y} = (-1)^{i-1}\mathrm{NTU}'rh(i)rv(i)\left[\left(\frac{h_{i-1}}{h_{i-1}+h_i}\right)t_{i-1} + \left(\frac{h_i}{h_i+h_{i+1}} + \frac{h_i}{h_i+h_{i-1}} - 2\right)t_i + \frac{h_{i+1}}{h_i+h_{i+1}}t_{i+1}\right] \tag{12}$$

However, the first and the last channels transfer heat only to one fluid; hence the equations for these two channels must be written as

$$\frac{dt_1}{dY}=\mathrm{NTU}'rv(1)rh(1)\left[\left(\frac{h_1}{h_1+h_2}-1\right)t_1+\left(\frac{h_2}{h_1+h_2}\right)t_2\right] \quad (13)$$

$$\frac{dX_N}{dY}=\mathrm{NTU}'rv(N)rh(N)\left[\left(\frac{h_{N-1}}{h_{N-1}+h_N}\right)t_{N-1}+\left(\frac{h_N}{h_{N-1}+h_N}-1\right)t_N\right] \quad (14)$$

where

$$\begin{aligned}\mathrm{NTU}' &= \mathrm{NTU}_{\mathrm{uniform}} \quad H\left(\tfrac{\dot m_2}{\dot m_1}\right)r\Delta t(i)rh_{\mathrm{fg}}(i) && \text{for two-phase flow channels}\\ &= \mathrm{NTU}_{\mathrm{uniform}} && \text{for single-phase flow channels}\end{aligned}$$

Eqs. (12)–(14) are the governing differential equations, for which the boundary conditions can be set as,

$$\left.\begin{aligned} &\text{at } Y=0, && t_i=0 && \text{for } i=1,3,5,\ldots,N\\ & && t_i=1 && \text{for } i=2,4,6,\ldots,N-1\\ &\text{at } Y=1, && t_i=1 && \text{for } i=2,4,6,\ldots,N-1\\ & && X_i=X_{\mathrm{in}} && \text{for } i=2,4,6,\ldots,N-1\end{aligned}\right\} \quad (15)$$

The total number of channels is assumed to be odd which is often the practical case in applications in order to minimize heat losses from end channels by circulating cold fluid in both of them ($i = 1$ and N). This complete mathematical model is a set of coupled differential equations. For solution it requires a precise distribution of the flow from channel to channel which is described below.

3. Solution procedure

Using a numerical technique, the system of first order ordinary differential equations (12)–(14) can be solved. The finite difference technique with a semi implicit procedure is used here. The plate heat exchanger channels are divided into M elements of equal size as shown in Fig. 2. The convergence criterion chosen for the termination of computation was 1×10^{-5} in the non-dimensional temperature. The numerically convergent solution was checked for physical consistency through an energy balance between the two fluids. The grid refinement was continued until the error in the energy balance was less than 0.1%.

The most important requirement for this solution is a proper distribution of fluid in the channels from the port. Bassiouny and Martin [14,15] presented the flow channelling formulation for normal geometries where the volumetric flow rate decreases along the flow direction in the entrance port as:

For the case of U type plate heat exchangers

$$u=\left(\frac{A_{\mathrm{p}}}{nA_{\mathrm{c}}}\right)m\frac{\cosh m(1-z)}{\sinh m} \quad (16)$$

The distribution parameter m given in the above expression is dependent on the exchanger geometry. The m value can be computed from the following equation which was suggested by Bassiouny and Martin [14,15] for single phase fluid flow. For identical inlet and exit port dimension, this value reduces to

$$m^2=\left(\frac{nA_{\mathrm{c}}}{A_{\mathrm{p}}}\right)^2\frac{1}{\zeta_c} \quad (17)$$

Here ζ_{c} is the overall frictional resistance of the channel. In this equation, m^2 serves as the parameter quantifying the flow maldistribution. The value of m^2 approaches zero when the flow is uniformly distributed among the channels. The more flow maldistribution, the higher is the value of m^2.

Cooper [35] and Tovazhnyanski and Kapustenko [26] have investigated steam condensation in plate heat exchangers. To calculate the heat transfer coefficient for both two phase flow and single phase flow the following equations have been adopted.

For two-phase flow channels, the modified form of the Boyko and Kruzhilin [36] correlation, suggested for vapor condensation in the PHE channels by Wang et al. [3],

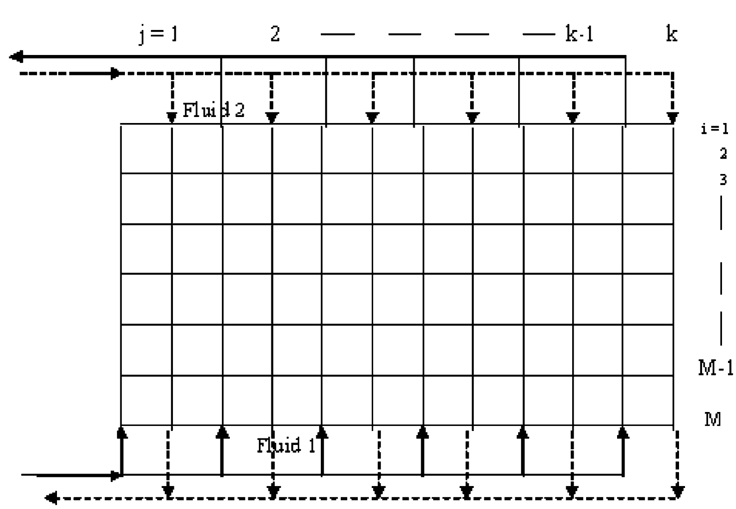

Fig. 2. The grid structure employed in the finite difference analysis of a single pass PHE.

Tovazhyanski and Kapustenko [26] and Cooper [35], for the heat transfer coefficient is

$$h = h_{\mathrm{l}}\left(\frac{\rho_{\mathrm{l}}}{\rho_{\mathrm{m}}}\right)^{0.5} \tag{18}$$

where h_{l} is the heat transfer coefficient assuming all the mass is flowing as liquid flow, ρ_{l} is the density of the liquid phase and ρ_{m} is the mean density of the liquid/vapor phase.

For single phase flow, Manglik and Muley [37] have suggested a Nusselt number correlation as

$$Nu = 0.159 Re^{0.7} Pr^{0.33} \tag{19}$$

Eq. (19) has been used for calculating the heat transfer coefficient for the single phase flow channels. Eq. (17) can be modified for negligible variation in the saturation pressure of the two-phase flow from channel to channel for evaluating the heat transfer coefficient as $h = f(Re)$ and hence Eq. (14) can be simplified as

$$\frac{\mathrm{d}X_N}{\mathrm{d}Y} = \mathrm{NTU}' rv(N) rh(N)\left[\left(\frac{h_{N-1}}{h_{N-1}+h_N}\right)t_{N-1} + \left(\frac{h_N}{h_{N-1}+h_N} - 1\right)t_N\right] \tag{20}$$

where $\mathrm{NTU}' = \mathrm{NTU}_{\mathrm{uniform}} H\left(\frac{\dot{m}_2}{\dot{m}_1}\right)$ for two-phase flow channels.

An iteration process has been continued after calculating the dryness fraction of the two-phase flow. The dryness fraction of the two-phase flow has been calculated by taking an energy balance at the end of each two-phase flow channel.

$$\dot{m}_{\mathrm{total}}(h_{\mathrm{f}} + X_{\mathrm{out}} h_{\mathrm{fg}}) = \sum (h_{\mathrm{f},i} + X_i h_{\mathrm{fg},i})\dot{m}_i \tag{21}$$

where $\dot{m}_{\mathrm{total}} = \dot{m}_1 + \dot{m}_3 + \dot{m}_3 + \cdots + \dot{m}_N$.

$$X_{\mathrm{out}} = \frac{\sum \dot{X}_i rh_{\mathrm{fg},i} u_i}{n} + \left[\frac{\sum u_i rh_{\mathrm{f},i}}{n}\right] - \frac{h_{\mathrm{f}}}{h_{\mathrm{fg}}} \tag{22}$$

where $rh_{\mathrm{fg},i} = \frac{h_{\mathrm{fg},i}}{h_{\mathrm{fg}}}$; $rh_{\mathrm{f},i} = \frac{h_{\mathrm{f},i}}{h_{\mathrm{fg}}}$.

By neglecting the pressure variation in each channel from the first channel to the last channel, i.e., $h_{\mathrm{f}} = h_{\mathrm{f},i}$ and $h_{\mathrm{fg}} = h_{\mathrm{fg},i}$ one has

$$X_{\mathrm{out}} = \frac{\sum \dot{X}_i u_i}{n} + \frac{h_{\mathrm{f}}}{h_{\mathrm{fg}}}\left[\frac{\sum u_i}{n} - 1\right] \tag{23}$$

Similarly for single phase flow channels, the outlet temperature of each channel has been calculated as following:

$$\dot{m}_{\mathrm{total}} c_p T_{\mathrm{out}} = \dot{m}_1 c_p T_1 + \dot{m}_3 c_p T_3 + \dot{m}_5 c_p T_5 + \cdots + \dot{m}_{N-1} c_p T_{N-1} \tag{24}$$

$$T_{\mathrm{out}} = \sum \frac{u_i T_i}{n} \quad \text{for } i = 1, 3, 5, \ldots, N-1 \tag{25}$$

4. Results and discussion

The prediction of the performance of plate heat exchangers operating as condensers for both two-phase (process steam) and single-phase (cooling fluid) have been considered for counter-current flow. A case study has been carried out for downflow falling film partial saturated steam condensation at exit on a vertical plate heat exchanger with water as cooling fluid. The latent heat of the two-phase mixture is a function of the saturation pressure and it affects the overall heat transfer coefficient. Under the conditions of flow maldistribution in plate condensers, the pressure drop will vary from the first channel to the last channel of the parallel plate package and it will affect the available heat energy to be transferred to the cooling fluid. A thermal analysis has been made to predict the performance of a plate condenser under these circumstances. The present results are more useful to predict the behavior of two-phase fluid in a parallel vertical plate condenser under partial condensation.

The overall flow distribution in plate heat exchangers can be mainly classified into two categories, U type and Z type configurations. In the present study, the U type configuration has been chosen to analyse the flow distribution in plate condensers due to the practical importance. Here, fluid 1 has been taken as cold fluid and fluid 2 as the two-phase fluid. Generally, for two-phase flow heat exchangers, the effectiveness of an ideal heat exchanger (with uniform distribution) depends on the number of transfer units (NTU). According to the definition of $\mathrm{NTU}\left(= \frac{UA}{(\dot{m}C_p)_{\min}}\right)$, the minimum heat capacity of the two fluids has been taken from the single phase flow. However, the emphasis of the present work is to bring forward the effect of flow maldistribution on the performance of the plate condenser and hence in addition to NTU, the maldistribution parameter m^2 is taken as the other important parameter here.

The flow distribution from the first channel to the last channel has been computed using the Bassiouny and Martin [14,15] flow maldistribution model. This model was actually developed for single phase flow in the channels and the port friction in the both inlet and outlet ports of the plate heat exchanger were neglected. In the present analysis, the flow distribution has been computed by assuming negligible property variation. In the two-phase flow Nusselt number correlation, the vapor and the liquid density variation from the first channel to the last channel has been neglected and but flow rate variation in the channels considered. It should be mentioned here that due to flow maldistribution, the channels have different heat transfer coefficients, and hence it is difficult to define the NTU for the entire equipment. However, in order to facilitate a comparison with the uniform distribution model, NTU is defined on the basis of the heat transfer coefficient for the hypothetical case of equal flow distribution ($m^2 = 0$).

The main operating parameters of the present model have been taken as the flow rate, the saturation pressure, P_{sat}, the saturation temperature, T_{sat}, the inlet vapor quality, X_{in} and the latent heat of vaporization, h_{fg} for the two-phase fluid. For the single phase fluid, the inlet temperature, $T_{c,\mathrm{in}}$ and the mass flow rate are the main operating parameters. In this analysis, the term, msr, is the mass flux ratio of the cooling fluid (water) and the process stream. In practice, the process vapor is used with negligible superheat before entering the plate condenser. To demonstrate the present model, the inlet condition of the process vapor has been considered as a two-phase fluid with the dryness fraction of 0.90 at 1 bar. The present results were validated against the $\varepsilon - \mathrm{NTU}$ value of an ideal two-phase flow heat exchanger, i.e., $\varepsilon = 1 - e^{-\mathrm{NTU}}$ when the heat capacity flow rate ratio of the two fluids is zero ($R = 0$), as shown in Fig. 3. This has been carried out by setting the maldistribution parameter, $m^2 = 0$ for a total number of the channels equal to 98, the saturation pressure of the two-phase fluid at 1 bar, the inlet vapor dryness fraction, $X_{\mathrm{in}} = 0.90$ and the mass flux ratio of the two fluids as 1.0. The figure shows that the present model predicts the analytical data excellently.

Parametric study has been carried out to observe the behavior of the heat exchanger for the different conditions. Even though the effectiveness is an indicator of the temperature rise of the cooling fluid, in practical applications, the value of the actual dimensional temperature is also important. It gives an idea about the practical ranges of temperatures over which the heat exchanger should operate which in turn decides about the secondary effects such as axial conduction and heat loss from the heat exchanger.

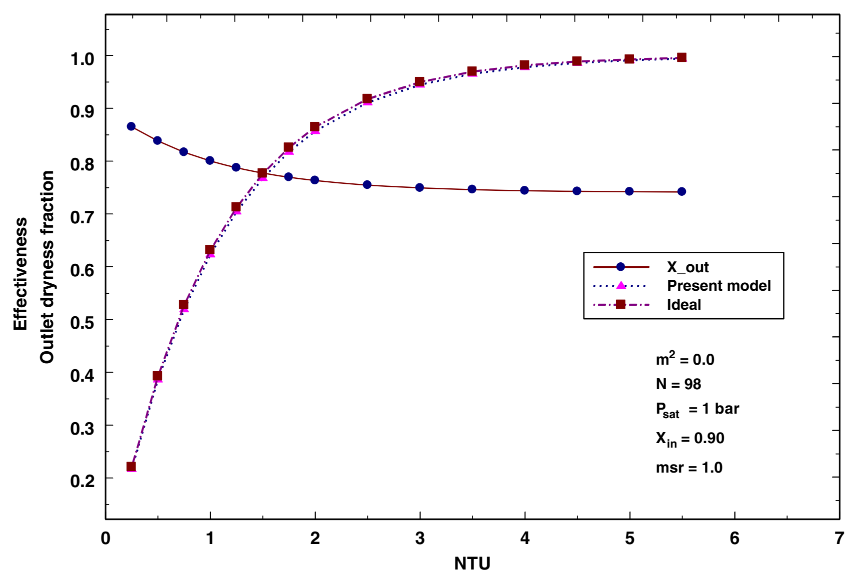

Fig. 3. Validation of the present model against the thermal performance of an ideal two-phase flow heat exchanger.

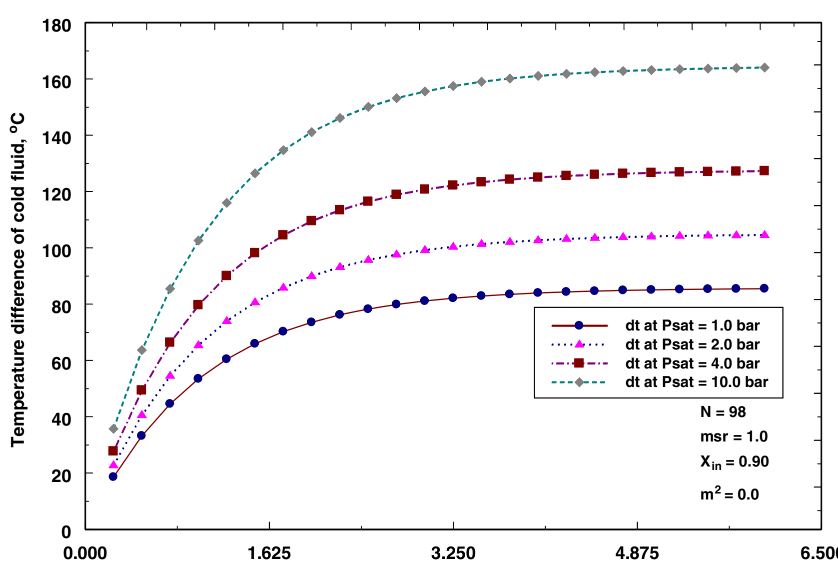

Fig. 4. Effect of the number of transfer units (NTU) on the cold fluid temperature rise (inlet to outlet) for a given plate condenser at different steam saturation pressure.

It also plays a major role in deciding the ranges of dry wall and de-superheating in practical applications where the superheated vapor enters the condenser. For this reason, the dimensional temperature rise of the coolant versus NTU for the range of steam pressures from 1 to 10 bar has been separately plotted in Fig. 4. This figure shows that as the saturation pressure increases the temperature difference from inlet to outlet of the cold fluid increases asymptotically with NTU. Higher saturation pressure gives higher heat transfer and thus greater performance indicated by higher temperature rise of the coolant. At the same conditions, results are plotted to observe the outlet dryness fraction of the steam as a function of the NTU. Fig. 5 confirms that the exit vapor quality decreases with increasing NTU and with increasing saturation pressure of the steam. It has been observed that the variation of steam quality in the condensation process is very small beyond NTU = 3.0. This is due to formation of a thick water film which reduces the heat transfer rate. Further to be observed is the behavior of the condensation of the vapor, due to the mass flux ratio of the fluids (msr) for the different saturation pressures of the steam and these results are depicted in Fig. 6. As the mass flux ratio of the two fluids increases the condensation process increases steeply. This is also true for increasing the saturation pressure of the steam. Hence, these results reveal that in

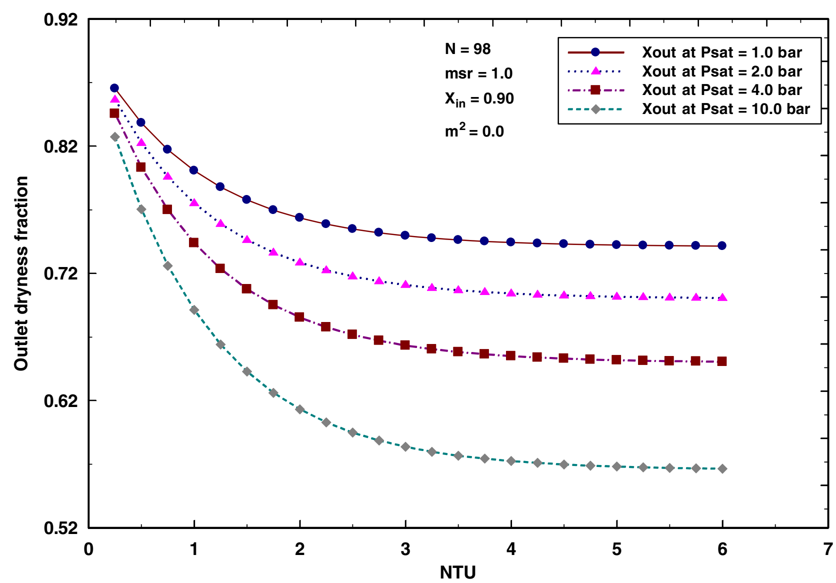

Fig. 5. Effect of the number of transfer units (NTU) on the condensation process for a given plate condenser at different steam saturation pressures.

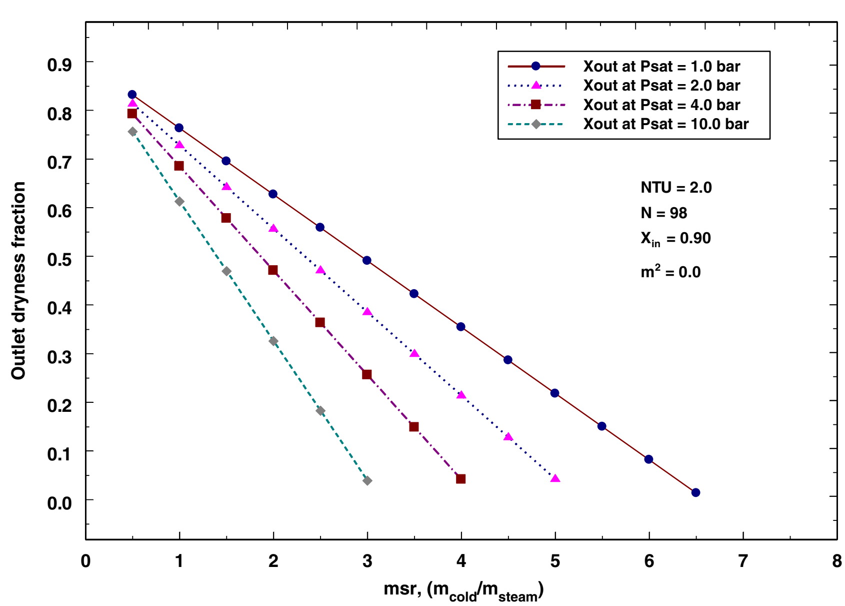

Fig. 6. Effect of the mass flux ratio on the condensation process for a given plate condenser at different steam saturation pressures.

absence of maldistribution ($m^2 = 0$) the condensation increases with increasing NTU, saturation pressure of the two-phase fluid and mass flux ratio of the two fluids.

The above results have been obtained for uniform flow distribution ($m^2 = 0$) to observe how the present model behaves against the parameters of which the influence can be initiatively guessed. The effect of flow maldistribution on the outlet vapor guessed quality of the process condensing steam with variation of the mass flow rate ratio of the cold water and steam has been analysed for a given inlet conditions of the process steam at the saturated pressure, $P_{sat} = 10$ bar and 1 and inlet vapor quality, Xin = 0.90, NTU = 2.0 (based on cold fluid) and the inlet temperature of cold fluid is 15 °C. It has been observed that there is a very small difference in outlet quality of steam both at high and low pressure when there is maldistribution from the port to the channels as compared with the uniform flow in each channel as shown in Fig. 7. This small difference justifies neglecting the properties variation when maldistribution is taking place from port to the channels. Fig. 8 show the effect of flow maldistribution on the temperature difference of the cold fluid against the variation of NTU for a given plate condenser of having mass flow rate ratio of 3.0 and the same inlet conditions as mentioned earlier. The results indicate that due to the non-uniform flow in each channel, the net enthalpy drop of steam in the plate

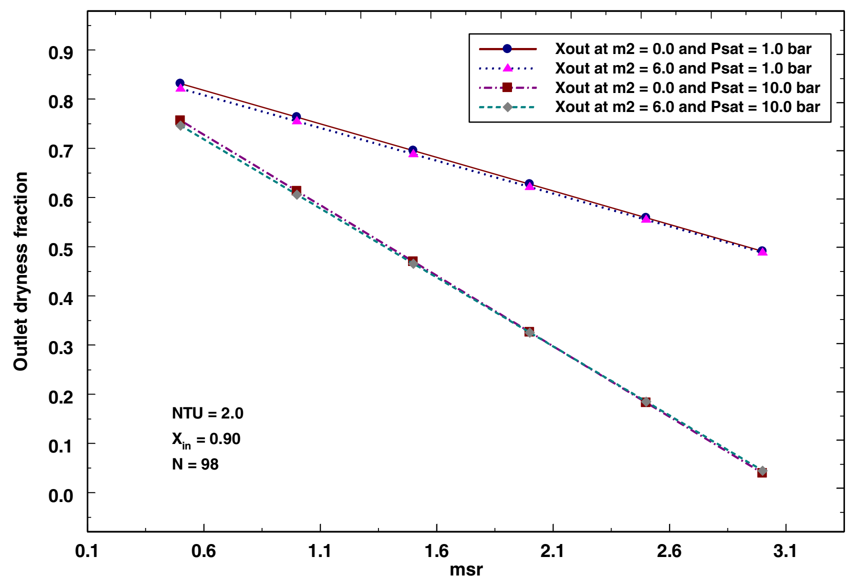

Fig. 7. Effect of msr on the condensation process with flow maldistribution in a plate condenser.

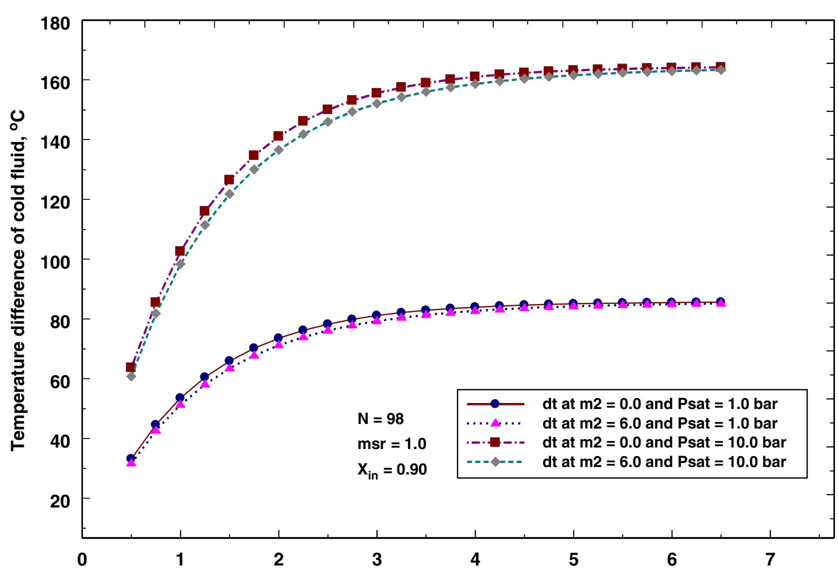

Fig. 8. Effect of NTU on the condensation process with flow maldistribution in a plate condenser.

package will be decreasing with increase of flow maldistribution and results in the decrease in temperature difference of the cold fluid due to less heat transfer between the process and the cooling fluid. It is interesting to observe that this effect increases with pressure and is significant in the usual operating range of NTU for plate condenser (2–4). This effect will be even higher when the changes of the properties are taken into the consideration. Fig. 9 compares the effectiveness of the plate condenser with and without flow maldistribution from the port to the channel, i.e., $m^2 > 0$. The flow maldistribution parameter, m^2 has been varied from zero to 6.0 and pressure from 1 to 10 bar. It has been observed from Fig. 9 that when the maldistribution increases the effectiveness of the condensation process decreases with increasing NTU. However, the figure indicates that saturation pressure has only a small effect on the change of effectiveness with maldistribution. As the pressure changes, the densities of the vapor and liquid also change as well as the latent heat of the vaporization. Thus the net heat transfer between the two fluids change. However, due to these changes the heat transfer potential also changes and it changes in similar proportions. Hence, the thermal effectiveness of the two-phase fluid does not depend on the variation of the operating pressure.

Wang and Zhao [33] have carried out an analysis of the performance of steam condensation in plate heat

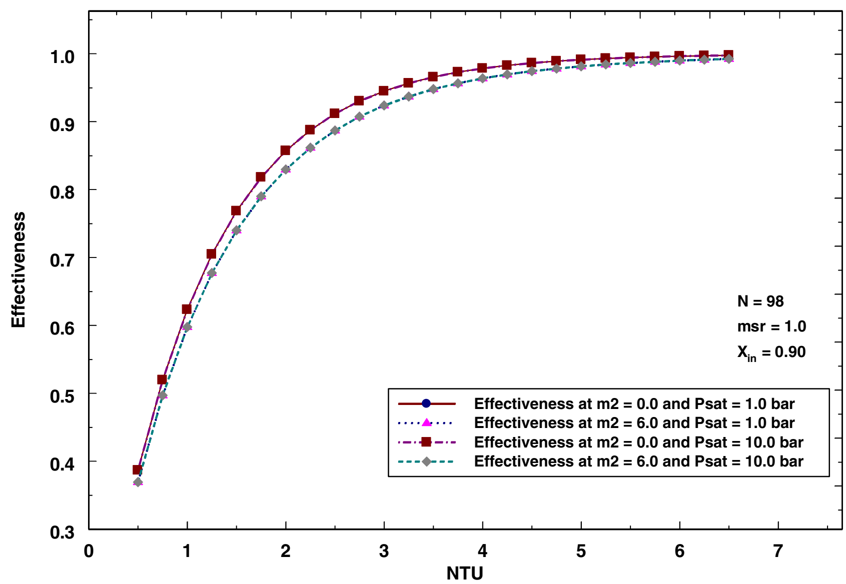

Fig. 9. Effect of NTU on the thermal effectiveness of a plate condenser with flow maldistribution.

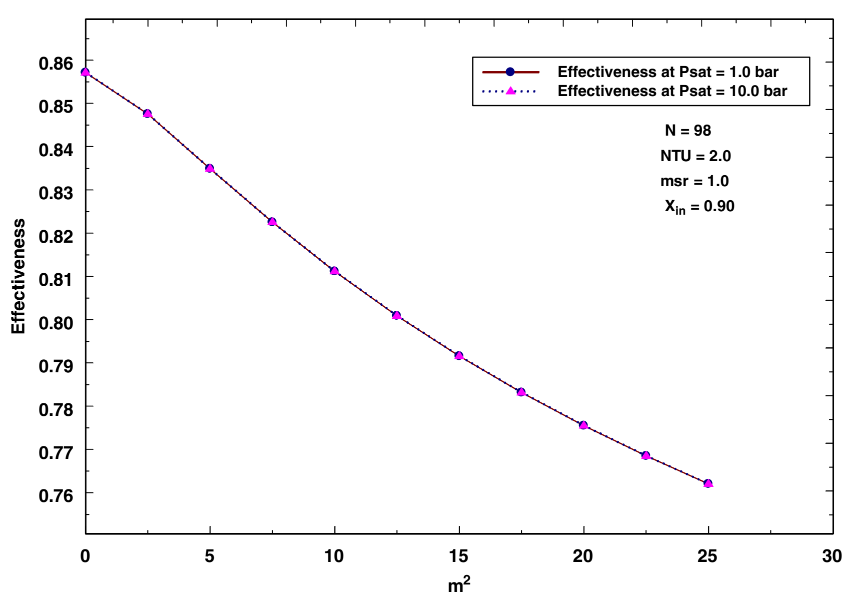

Fig. 10. Effect of flow maldistribution on the thermal performance of a plate condenser.

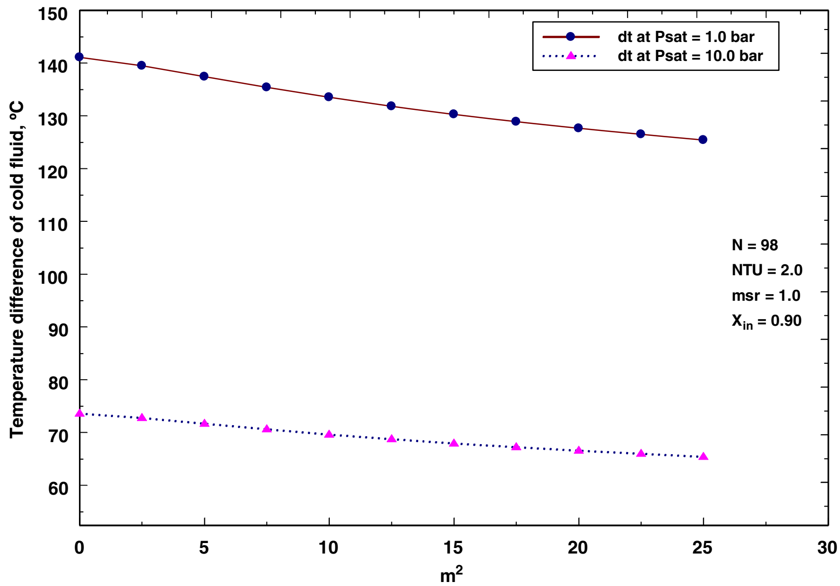

Fig. 11. Effect of flow maldistribution on the temperature rise of the cold fluid for a plate condenser.

exchangers. They have suggested a correlation for evaluating the heat transfer coefficient for two-phase fluid channels as

$$Nu = 0.00115\left(\frac{Re_1}{H}\right)^{0.983} Pr_1^{0.33}\left(\frac{\rho_1}{\rho_v}\right)^{0.248} \tag{26}$$

In Eq. (26), the exponent of the Reynolds number term, $\left(\frac{Re_1}{H}\right)$ is 0.983. It gives an impression that the heat transfer coefficient does not vary much even though the channel flow rate varies when neglecting the density variation in the channel and hence the thermal performance may be affected for variation of the channel flow rate in the plate heat exchanger. Therefore, the present results (Fig. 9) indicate that the changes in thermal performance in a plate condenser are small due to maldistribution from channel to channel as the variation of saturation pressure between the channels is neglected. The same result can be depicted by directly plotting effectiveness against m^2 which qualifies the decrease of effectiveness due to flow maldistribution (Fig. 10).

Similarly, a plot of the temperature rise in the coolant against m^2 gives an idea of the effect of maldistribution on the condensation process, see Fig. 11. This plot indicates that even though pressure has little effect on the change of effectiveness of the condenser with maldistribution it has significant effect on the amount of condensation as evident from the different slopes of the two curves.

5. Conclusions

A generalized mathematical model has been presented to study the effect of maldistribution on the thermal performance as well as the exit quality of vapor of a plate heat exchanger. The present model was validated against the ε – NTU value of a two-phase flow plate heat exchanger with no flow maldistribution. A wide range of the parameters has been studied to show the effects of the mass flow rate ratio of the cold to the two-phase fluid, operating pressure and flow distribution index. The results show that the exit vapor quality decreases and cooling water temperature rise increases with increasing NTU and with increasing operating pressure of the steam indicating a higher rate of condensation. Also as the mass flux ratio of the two fluids increases the condensation rate increases steeply. With increasing maldistribution, the effectiveness and condensation rate of the vapor decreases but it remains virtually unaffected by change in operating pressure. However, the effects of maldistribution and operating pressure are significant on the temperature rise of the cooling water but they are less prominent on outlet steam quality. Generally speaking, the performance and the condensation will be deteriorated as the maldistribution increases. The analysis presented here also suggests an improved method for heat transfer data analysis for plate condensers. The results confirm that this model predicts the behavior of the plate condenser in a physically consistent way under the flow maldistribution from the port to the channel.

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