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ON THE PHENOMENON OF TWO-PHASE FLOW MALDISTRIBUTION IN A HEAT EXCHANGER UNDERGOING CONDENSATION

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Abstract

The non-uniformity of two-phase flow rates among the circuits in a heat exchanger reduces its thermal performance. In this work, the effects of a maldistributed condensing two-phase flow profile in an arbitrary cross-flow heat exchanger has been investigated. The results of a discretization numerical analysis shows that the trend of the degradation effect is similar to that found for single phase flows. The thermal performance degradation factor, *D*, is dependent on the standard deviation and skew of the flow profile and the change of vapour quality along the flow circuits. The magnitude of *D* varies as the square of normalized standard deviation and liquid Reynolds number, and linearly with the normalized skew. However, the effect of vapour quality is not as significant as compared to that caused by the statistical moments of probability function of the flow maldistribution profile. Flows with low standard deviation and positive skew are preferred to give low magnitudes of *D*

Keywords: Heat exchangers, Thermal performance deterioration, Statistical moments, two-phase flow, vapour quality, condensation.

1. Introduction

The adverse effect of flow maldistribution on the thermal performance of a heat exchanger has been extensively deliberated in many research works. The research done covers various types of heat exchangers for different applications. For example, Chiou [1] and Ranganayakulu et al. [2] investigated the effect of flow maldistribution in cross-flow heat exchangers. Bassiouny and Martin [3, 4] and Rao et al. [5] characterized the maldistribution phenomenon in plate heat exchangers while Lee et al. [6] and Ryan and Timoney [7] studied maldistribution of refrigerant flow in fin-tube heat exchangers used for air-conditioning equipment.

Nomenclatures

A	Area, m^2
C_{max}	Maximum heat capacity rate, W/K
C_{min}	Minimum heat capacity rate, W/K
D	Thermal performance degradation factor
d	Diameter, m
h	Condensation heat transfer coefficient, $W/m^2.K$
i	Specific enthalpy, J/kg
k	Thermal conductivity, W/m.K, or constant
m	Exponent index
N, n	Number of discrete elements and exponent index, respectively
p_r	Reduced pressure
Pr	Prandtl number
q, Q	Elemental and total heat capacity, respectively, W
Re	Reynolds number
r	Thermal resistance, K/W
v	Velocity, m/s
x	Vapour quality

Subscripts

fg	Change of phase
int	Internal
I, J	Element number along x and y axis, respectively
L	Liquid
r	Refrigerant
sp	Single phase
s	Surface
u	Uniform

Greek Symbols

ε	Heat exchanger effectiveness
ΔT	Temperature difference, °C
σ'	Standard deviation (normalized)

Kim et al. [8] examined gas flow maldistribution in shell-and-tube heat exchangers.

In all these research, the thermal and hydraulic performances of the heat exchanger have been observed to deteriorate, i.e., the flow maldistribution causes reduced heat transfer rates and increased pressure drops. In the work done by Chin and Raghavan [9, 10], the first three moments of probability density function of the maldistribution profile, i.e., mean, standard deviation and skew, have been shown to give significant effect on the heat exchanger performance. The fourth moment, kurtosis, has insignificant effect on the heat exchanger performance. Flow maldistribution with large standard deviation and negative skews causes more severe degradation effects.

A convenient index used to quantify the magnitude of thermal performance degradation is the thermal performance deterioration factor, D, which was first defined by Chiou [1], as:

$$D = \frac{\varepsilon_{uniform} - \varepsilon_{maldistributed}}{\varepsilon_{maldistributed}} \times 100\% \qquad (1)$$

Where, ε is the heat exchanger effectiveness. Eq. (1) can also be expressed as the following [9]:

$$D = \left(1 - \frac{Q_{maldistributed}}{Q_{uniform}}\right) \times 100\% \qquad (2)$$

where Q is the heating capacity of the heat exchanger.

The maldistributed fluid through the heat exchanger could either be in single- or two-phase. Examples of single-phase fluids include air and water. This has been extensively researched in many papers, e.g. Fagan [11] and Rabas [12] who studied non-uniform air flow in air-conditioning fin-tube coils and air-cooled condensers, respectively, and Shaji and Das [13] who examined maldistributed water flow rate in the channels of plate heat exchangers. Raul et al. [14] investigated the effect of a modified inlet header with double baffle design to improve the single-phase flow distribution into a plate fin heat exchanger. However, not much work has been done to analyse the effect of non-uniform two-phase flow. Most of the research that has been done is related to refrigerant distribution in air-conditioning fin-tube heat exchangers. In the experimental work by Choi et al. [15], maldistributed R22 refrigerant flow in a 3-circuit evaporator fin-tube coil has caused degradation of heat transfer capacity by up to 30% as the superheat at the evaporator outlet was measured at 16.7^{o}C. Capacity recovery of 4% was achieved as the superheat was controlled at 5.6^{o}C by regulating the refrigerant flow rate in each circuit. In the companion report of this paper authored by Payne and Domanski [16], the potential benefits of using smart distributors to control the two-phase refrigerant maldistribution have been identified. Hwang et al. [17] examined the impact of the geometry of the inlet horizontal manifold on the R410A refrigerant distribution in a vertical micro-channel heat exchanger. However, Lee and Domanski [18] have observed that the influence of maldistributed air flow on the heat exchanger coil is more significant than the maldistributed refrigerant flow on the capacity degradation. In actual situations, the non-uniform air flow affects the refrigerant distribution in the coil itself, which then leads to further performance degradation. Back et al. [19] has also investigated the use of interleave circuitry in an evaporator to mitigate the effects of maldistribution. Wang et al. [20] has developed a numerical model of a multi-pass condenser with vertical header where it was found that the aspect ratio and pass arrangement gave significant impact on the flow distribution and pressure drop.

In general, two-phase flows in heat exchangers undergo either the process of evaporation or condensation. During these processes, the proportion of vapour and liquid in the fluid changes as heat is exchanged with the surroundings. The vapour quality ratio is commonly used to describe the relative quantities of these two phases. It is defined as the ratio of mass (or mass flow rate) of vapour to total mass (or mass flow rate) of the mixture fluid [21]. During condensation, the vapour quality, x, decreases from 1.00 at saturated vapour condition to saturated liquid at 0.00. The reverse takes place during flow boiling. Nevertheless, sub-cooling could also occur at the exit of condensers and superheating at the exit of evaporators.

The work presented in this paper attempts to analyse and quantify the contribution of vapour quality to the thermal degradation of heat transfer performance of a condenser due to two-phase flow maldistribution. A review of the literature indicates that most of the mathematical models developed to analyse the effects of flow maldistribution are applicable for single-phase fluids. Examples of these include the finite element approach used by Chiou [1], Kou and Yuan [22] and Ranganayakulu and Seetharamu [23]. The research on two-phase flow maldistribution is mainly done experimentally [15-17]. However, there are several works which analyses the problem through numerical calculations which takes the vapour quality into consideration. In the work by Bobbili et al. [24] on falling film plate condensers, the vapour quality, or dryness fraction, of steam is used to evaluate the inlet and outlet energy of a control volume in the two-phase flow channel. The results of the work indicate a reduction in heat exchanger effectiveness as the flow distribution parameter, m^2, increases for a given inlet vapour quality. This also corresponds to a reduction in the outlet vapour quality as m^2 increases. But this work does not demonstrate the relationship between the inlet vapour quality and the heat exchanger thermal performance.

This problem has also been examined by Koern et al. [25] in their work on an A-coil of a residential air-conditioning unit. The distribution of vapour and liquid between the two circuits of the A-coil is quantified with the phase distribution parameter, F_x which is the ratio of the inlet vapour quality in circuit 2 to the inlet vapour quality before entering the distributor. The results of the numerical simulation show that as F_x decreases, which corresponds to more vapour phase flowing into the second circuit, the cooling capacity and COP of the system reduces.

With the lack of a suitable model to describe the influence of two-phase flow and vapour quality maldistribution on the heat exchanger performance, the objective of this work is to derive an analytical equation which would allow prediction of the magnitude of thermal performance degradation due to condensing two-phase maldistribution. This would be an extension of the works done by the author [9, 10] which are applicable for single-phase flows.

2. Mathematical Model

To examine the effect of vapour quality on the thermal performance deterioration factor, D, the same discretization technique described in [9] is applied on an arbitrary cross-flow heat exchanger which has two fluid streams A and B flowing through it. Fluid Stream A is the maldistributed stream of two-phase flow undergoing condensation, while the single-phase Fluid Stream B on the other side has uniform flow and temperature distributions. In general, the maldistributed two-phase flow is the C_{max} stream while the C_{min} stream is on the other side (Fluid Stream B). This is illustrated in the following Fig. 1.

In this model, the two-phase Fluid A stream (C_{max}) flows through the heat exchanger in several parallel circuits. These circuits do not cross each other, i.e., unmixed. In the same way, the uniform Fluid B stream (C_{min}) on the other side of the heat exchanger can also be considered as flowing through several parallel channels. With these in mind, the circuits of both fluid streams could be configured according to the number of passes each circuit makes through the heat exchanger. Some examples are shown in Fig. 2. A nomenclature for the number of passes along the three principal *x*, *y* and *z* axis, i.e., (*x*, *y*, *z*), is established, as shown in Fig. 2.

For this analysis, the mass flow rates of Fluid A through these parallel circuits are non-uniform. And, the fluid undergoes condensation as it flows through the circuits, starting from the inlet where saturated vapour enters, i.e., inlet vapour quality = 1.00. As a result, the condensing temperature in the circuits remains constant. On the other side, the inlet temperature of the uniform Fluid B stream is also invariant. For the purpose of simplification, the configuration of single pass for both fluid streams are considered for the analysis, i.e., corresponding to Fig. 2(a).

Therefore, the (1, 0, 1) heat exchanger shown in Fig. 1 is discretized into *a* x *b* number of elements, i.e., with *a* number of circuits (*y*-direction) and *b* number of cell elements along the heat exchanger length (*x*-direction). The total number of discrete cells, $N = a \times b$. This is further illustrated in the following Fig. 3.

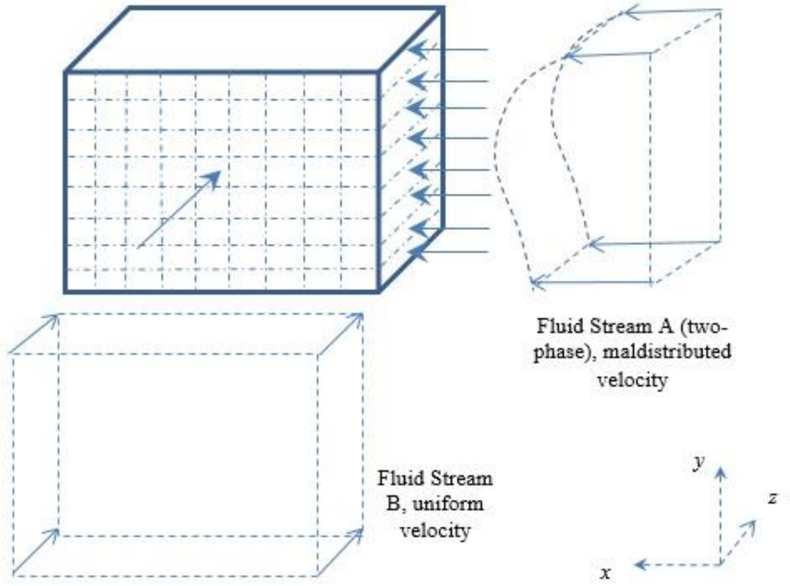


Fig. 1. Representation of cross-flow heat exchanger with maldistributed two-phase flow.

Every element along each circuit has the same refrigerant mass flow rate (I = 1 to *b*). Therefore, the statistical moments of probability density function (PDF) of the maldistribution are defined by the flow velocity field at the circuit inlets (J = 1 to *a*).

Since Fluid B is uniform, every element has the same fluid velocity flowing in cross-flow direction to the refrigerant flow. By analysing the heat transfer occurring on the internal refrigerant side of each discrete cross-flow element, the elemental heat transfer rate, q_{IJ}, is given as:

$$q_{IJ} = h_{IJ} A_{\text{int}} \Delta T_{r-s} \tag{3}$$

where h_{IJ} is the condensation heat transfer coefficient on the refrigerant side, A_{int} is the element heat transfer surface area on the refrigerant side, ΔT_{r-s} is the temperature difference between the refrigerant and internal wall surface.

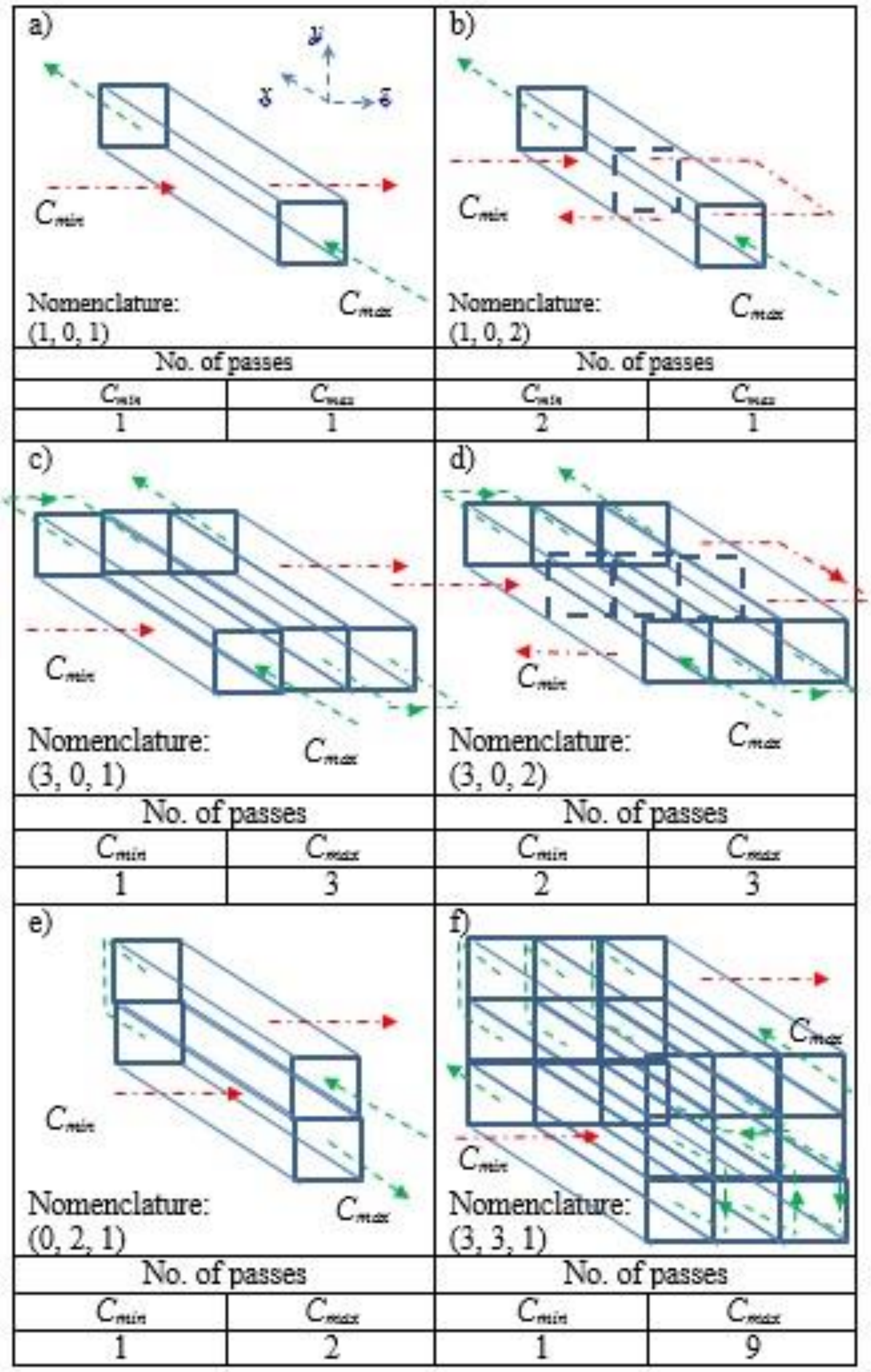


Fig. 2. Examples of pass configurations through heat exchanger.

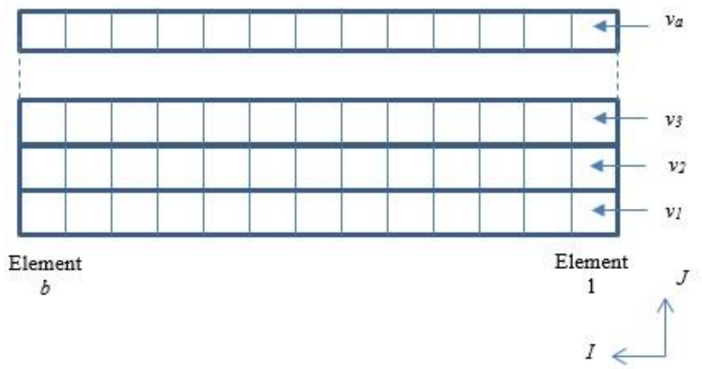


Fig. 3. Discretization of circuits.

With Fluid Stream B having uniform temperature distribution, and with a constant condensing temperature, the temperature difference ΔT_{r-s} is taken as constant.

The condensation heat transfer coefficient in the flow circuits can be calculated with suitable correlations, for example, the Shah correlation [26] for annular refrigerant flow condensation in round tubes, which is given as:

$$h_{IJ} = h_L \left[(1 - x_{IJ})^{0.8} + \frac{3.8 x_{ij}^{0.76} (1 - x_{IJ})^{0.04}}{p_r^{0.38}} \right] \quad (4)$$

where

$$h_L = 0.023 \mathrm{Re}_L^{0.8} \mathrm{Pr}^{0.4} k_L / d \quad (5)$$

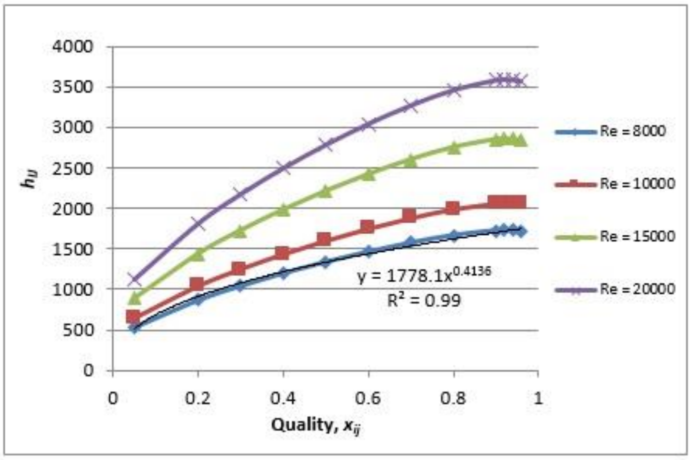
and x_{IJ} is the refrigerant quality of element (I, J), p_r is reduced pressure, defined as the ratio of actual refrigerant pressure to the critical pressure, d is the diameter of the flow channel. The subscript L in Eq. (4) and (5) refers to the condition where all the refrigerant mass is assumed flowing as liquid.

It is seen from Eq. (4) that for a constant refrigerant temperature and pressure, the heat transfer coefficient is dependent on refrigerant velocity, v_L and vapour quality, x_{IJ}. A plot of Eq. (4) as the quality changes is show in Fig. 4(a). The data is also re-plotted to illustrate the relationship with the liquid Reynolds number, Re_L (Fig. 4(b)). An example of the best-fit regression equation, and the corresponding coefficient of determination, R^2, demonstrating the supra-linear trend is shown in both Fig. 4(a) and 4(b). Similar trends of the dependence on velocity and vapour quality can be obtained from other published flow condensation correlations, for example Akers et al. [27] and Cavallini and Zecchin [28].

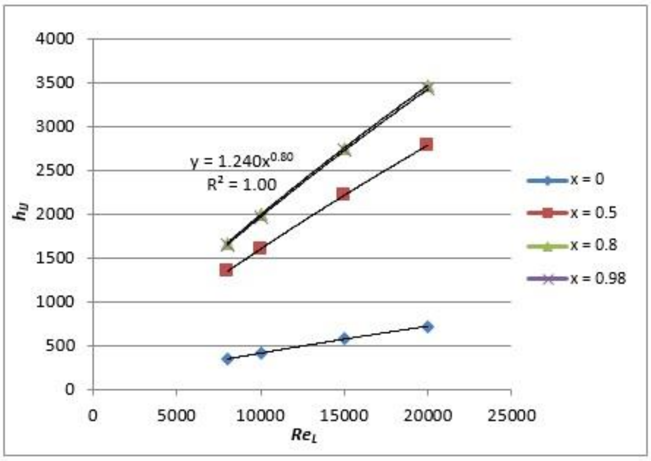
Within the range of $0.1 \le x_{IJ} \le 0.95$, and $8{,}000 \le \mathrm{Re}_L \le 20{,}000$, the condensation heat transfer coefficient can be estimated with the following relationship:

$$h_{IJ} = f(\mathrm{Re}_L, x_{IJ}) = k v_{L,J}^{\,n} x_{IJ}^{\,m} \quad (6)$$

where k is a constant and m and n are indices.



(a)



(b)

Fig. 4. Plot of heat transfer coefficient from Shah's correlation with respect to (a) vapour quality, x_{IJ}, and (b) Reynolds number, Re_L, for refrigerant R22 [Pr = 2.4, p_r = 0.34, k_L/d = 8 W/m^{2}K] [23].

In [9], it has been established that $n < 1.0$. The trend line plots shown in Fig. 3(a) indicate that m is also < 1.0. However, the magnitude of this index is dependent on the heat flux and mass flux in the heat exchanger itself, as can be seen from the work done by Honda et al. [29]. It is also dependent on the flow regime of the fluid, as pointed out by Cavallini et al. [30].

Combining Eqs. (3) and (6) gives:

$$q_{IJ} = K v_{L,J}{}^{n} x_{IJ}{}^{m} \tag{7}$$

where $K = kA_{\text{int}}\Delta T_{r-s}$

Therefore, the total condensing heat transfer capacity, under maldistributed refrigerant flow, is given as:

$$Q_{cond,m} = K\sum_{I=1}^{b}\sum_{J=1}^{a} v_{L,J}{}^{n} x_{IJ}{}^{m} \tag{8}$$

For the case of uniform velocity distribution, $\bar{v}_L$, in Fluid Stream A, Eq. (8) gives:

$$Q_{cond,u} = K\bar{v}_L{}^{n} a\sum_{I=1}^{b} x_{I,u}{}^{m} = K\bar{v}_L{}^{n} x_{sum} \tag{9}$$

where $x_{sum} = a\sum_{I=1}^{b} x_{I,u}{}^{m}$

The thermal performance degradation factor, *D*, is then obtained from (2) as:

$$D = 1 - \frac{Q_{cond,m}}{Q_{cond,u}} = 1 - \sum_{I=1}^{b}\sum_{J=1}^{a} v'_{L,J}{}^{n}\; x'_{IJ} \tag{10}$$

where

$$v'_{L,J} = \frac{v_{L,J}}{\bar{v}_L}$$

$$x'_{IJ} = \frac{x_{IJ}{}^{m}}{x_{sum}}$$

The derivation of Eq. (10) demonstrates that the degradation factor, *D*, of a heat exchanger with two-phase condensing flow, is dependent not only on the flow velocity, but also on the change of vapour quality of the fluid as it flows along the circuit.

It is interesting to note that the form of Eq. (10) for the thermal degradation factor, *D*, is similar to that for single phase fluid, as can be seen from [10], i.e.,

$$D_{sp} = 1 - \sum_{i=1}^{N} v'_i{}^{n} \tag{11}$$

With the ratio x'_{IJ} being less than one, it is deduced from Eq. (10) that larger variation of vapour quality along the two-phase fluid stream in the heat exchanger will increase the magnitude of the thermal degradation factor, *D*. The magnitude of *D* also becomes larger as the standard deviation of two-phase flow rates in the heat exchanger increases which results in larger variation of vapour quality among the circuits.

Even though Eq. (10) has been derived with the (1, 0, 1) pass configuration, it is also equally applicable for other configurations, i.e., (3, 0, 1), (0, 2, 1) and (3, 3, 1), as long as condensation is taking place in the C_{max} flow channels, and that C_{min} makes a single pass through the heat exchanger. With multiple C_{min} passes, e.g. (1, 0, 2) and (3, 0, 2), the degradation factor for each pass can be determined with respect to the corresponding entering fluid temperature of each pass. The summation of these degradation factors would then give an indication of the total degradation of the whole heat exchanger.

The derivation described above is applicable for two-phase flow within the range of saturated liquid and saturated vapour, i.e., $0.00 < x < 1.00$. However, in most practical applications, the heat exchanger has superheated vapour and/or sub-cooled liquid at the inlet and outlet. This gives rise to an added temperature maldistribution of the fluid which further complicates the problem. The effect of this phenomenon in a micro-channel heat exchanger has been investigated by Chng [31].

3. Numerical calculation

To demonstrate the theoretical derivation shown in the previous section, a numerical calculation is performed on a discretized fin-tube heat exchanger which has a flow of refrigerant condensing in the tubes (C_{max} fluid stream). The air passing through the fins (C_{min} fluid stream) gets heated up. The geometry of the heat exchanger is given in Table 1.

Table 1. Geometry of fin-tube heat exchanger used in calculation.

Number of rows	1, 2 and 3
Length of heat exchanger	2000 mm
Height of heat exchanger	254 mm
Fin pitch	1.411 mm
Tube pitch	25.4 mm
Row pitch	22.0 mm
Tube outer diameter	9.52 mm
Fin pattern	Wavy

The heat exchanger is discretized in the same manner as described in Fig. 2. The total number of discrete elements is 100, i.e., in a 10×10 grid. The calculation procedure used in this work is similar to that used in [10] where each element is analysed as an individual cross-flow heat exchanger. This is done by using Microsoft EXCEL with the aid of Visual Basic macro programming.

In the simulation, refrigerant R22 is used as the medium flowing in the tubes. The condensing temperature of the refrigerant is set at 50.0^{o}C while air flowing over the external fins has an inlet temperature of 35.0^{o}C. The refrigerant nominal mass flow rate is set at 20 kg/h, which enters the heat exchanger at specific vapour qualities, while the air volume flow rate is 3000 m^3/h. The thermodynamic properties of the refrigerant are obtained from the software RefProp 7.0 developed by NIST [32].

The air flow rate distribution on the fins is kept uniform, i.e., every discrete element has the same air velocity on the fins. However, the refrigerant mass flow rate distribution in the tube circuits are non-uniform.

The internal heat transfer coefficient in the tube is calculated with Eq. (5) while the external heat transfer coefficient on the fin surfaces are calculated by using the *j*-factor correlation develop by Wang et al. [33]. With these known, the overall heat transfer coefficient, U_o, can be obtained from the thermal resistance network for each discrete element. The heating capacity of each element, q_{IJ}, can then be determined by using the ε-*NTU* method with the known fluid inlet temperatures. The summation over all the elements will then give the total heating capacity of the condenser under maldistributed refrigerant flow, $Q_{cond,m}$.

For each discrete element, the heating capacity can be expressed as the enthalpy difference between the inlet and outlet refrigerant streams, i.e.,

$$q_{IJ} = \dot{m}_r (i_{r,in} - i_{r.out}) \tag{12}$$

With the inlet conditions and elemental heating capacity known, the refrigerant outlet enthalpy can be determined. Hence, the outlet vapour quality can be calculated from the following equation:

$$i_{r,out} = i_L + x_{IJ} i_{fg} \tag{13}$$

This vapour quality will then be the input for the analysis of the next discrete element. Consequently, the change of vapour quality along the refrigerant flow circuit can be obtained.

The calculation procedure is repeated for the case of uniform refrigerant flow rate distribution to arrive at $Q_{cond,u}$. With this, the value of D is computed. This numerical analysis is done to cover a range of statistical moments for the refrigerant flow distribution among the 10 circuits of the fin-tube heat exchanger, i.e., with normalized standard deviation, σ': 0.10 to 0.60, and normalized skew, γ': -1.40 to +1.00. The refrigerant mean flow rate is also allowed to vary between 20 kg/hr and 80 kg/hr. But, the kurtosis is not considered due to its insignificant effect on heat exchanger performance. The definitions of these moments are given in [10].

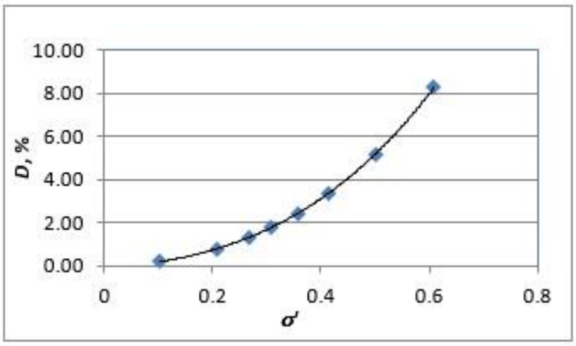
3.1. Results of simulation with uniform inlet vapour quality

The simulation is firstly conducted by having the refrigerant entering the heat exchanger as saturated vapour, i.e., x = 1.00. This inlet quality is uniform among all the 10 circuits.

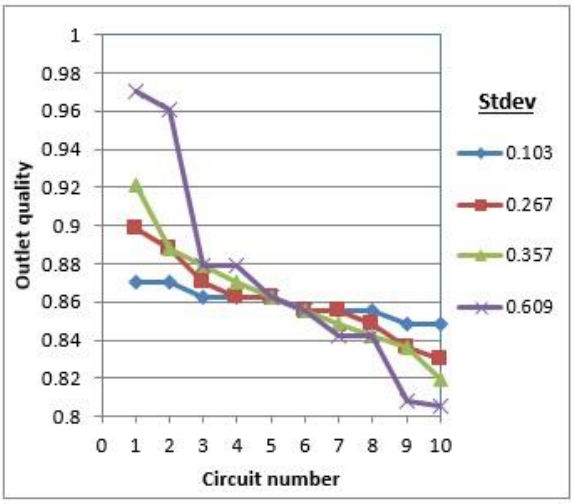
The results of the simulation with respect to the changes in standard deviation of the refrigerant flow rate is presented in Fig. 5(a). It is observed that the thermal degradation factor, D, increases as the refrigerant flow distribution standard deviation increases. A fitting of the data indicates that D varies as the square of standard deviation. This trend is the same as that obtained for single phase flows, as reported in [9, 10].

At the same time as the standard deviation changes, the refrigerant quality along the 10 parallel circuits also changes. Fig. 5(b) shows that the vapour quality at the outlet of the circuits changes more as the standard deviation of the refrigerant flow rate increases.

Figure 6 shows the results of the simulation when the skew of the refrigerant flow rate changes for a known standard deviation. The trend of Fig. 6(a) indicates a linear relationship between D and skew. Higher skews give lower magnitudes of thermal performance degradation. Again, this is similar with the findings reported in [9, 10] for single phase flows. Consequently, the variation of vapour quality along the circuits is larger when the skew is more negative, as shown in Fig. 6(b).

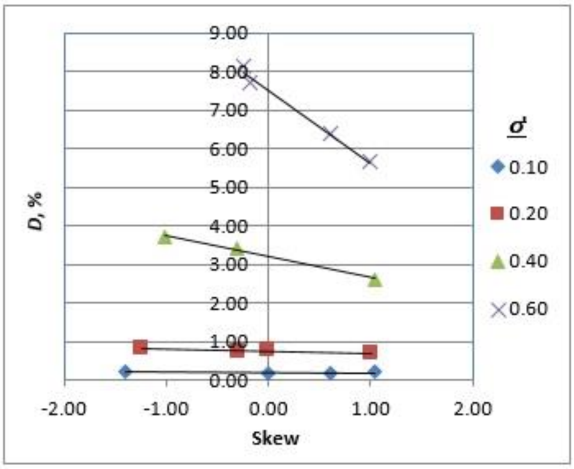


(a)

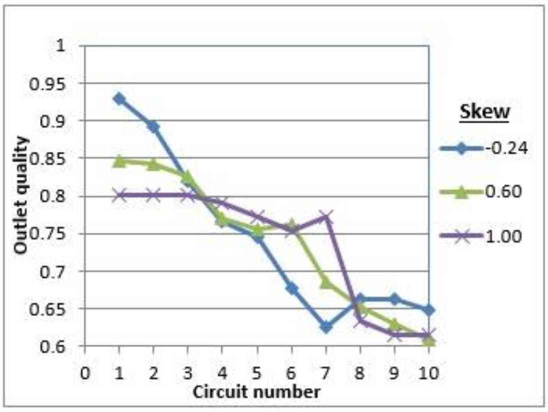


(b)

Fig. 5. (a) Trend of *D* vs. flow rate normalized standard deviation, and (b) Changes in outlet vapour quality as normalized standard deviation changes [Mean = 1.00, skew = 0.00].



(a)



(b)

Fig. 6. (a) Trend of *D* vs. normalized skew, and (b) Changes in outlet vapour quality as skew changes at normalized standard deviation = 0.60 [Mean = 1.00].

In Fig. 7, the results of the trend of *D* as the magnitude of refrigerant flow rate, i.e., the mean of the flow, changes. This is represented with the liquid Reynolds number of the flow. As the flow rate increases, *D* describes a parabolic trend line, i.e., it increases until a critical Re_L is reached where it starts to decrease. A similar phenomenon has also been reported in [9, 10] for single phase fluids.

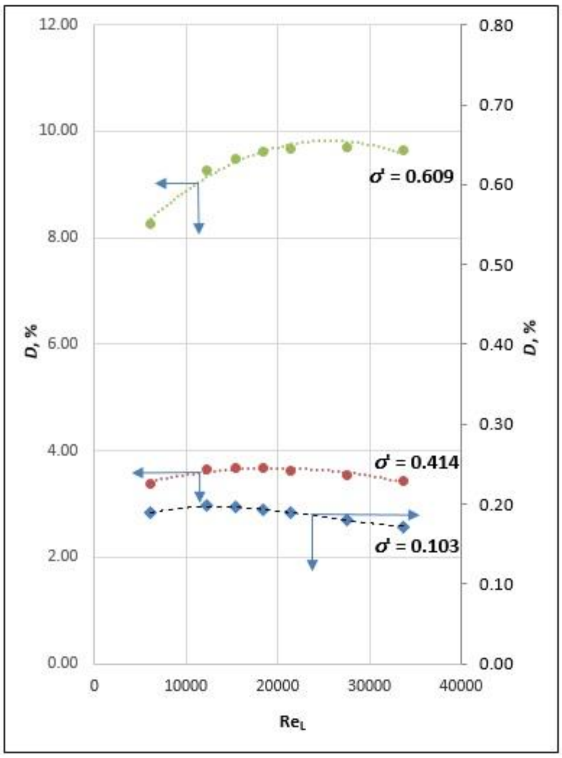


Fig. 7. Trends of *D* vs. liquid Reynolds number at specific flow distribution normalized standard deviation (Skew = 0.00).

3.2. Results of simulation with different magnitudes of uniform inlet vapour quality

The simulation is then repeated with different magnitudes of uniform inlet vapour qualities in the inlet pipes of the heat exchanger. For this simulation, the vapour quality is varied between 0.50 and 1.00. The refrigerant flow distribution statistical moments are maintained at normalized standard deviations of 0.357 and 0.609, with skew of 0.00. The heating capacities for the case of uniform and maldistributed refrigerant flow rates are calculated at the same inlet vapour quality to determine the thermal degradation factor, *D*. The results of the simulation are presented in Fig. 8. It is seen from the results that the thermal degradation factor does not changes significantly as the uniform inlet vapour quality is varied.

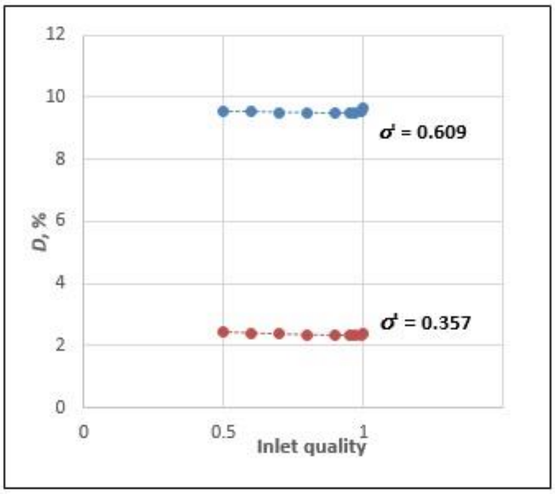


Fig. 8. Trend of *D* vs. uniform inlet vapour quality (Mean = 1.00, skew = 0.00).

3.3. Results of simulation with non-uniform inlet vapour qualities

For the next simulation, the inlet vapour qualities among the circuits into the heat exchanger are non-uniform. Two scenarios are examined, i.e., (a) the refrigerant flow rates are uniform, and (b) the refrigerant flows rates are also non-uniform (i.e., normalized standard deviation = 0.357, 0.501 and 0.609, with skew = 0.00).

For both cases, the inlet vapour quality distribution is varied to give a range of standard deviation between 0.07 and 0.20 with skew = 0.00. The mean inlet vapour quality is set at 0.65. The thermal degradation factor, *D*, is calculated with reference to the case of uniform inlet vapour quality distribution at the same mean inlet quality.

The results of the simulation are shown in the following Fig. 9. It can be seen that the inlet vapour quality distribution has an effect on *D* where higher vapour quality standard deviation gives larger magnitudes of *D*. It is observed that this effect becomes more insignificant as the flow rate distribution standard deviation increases.

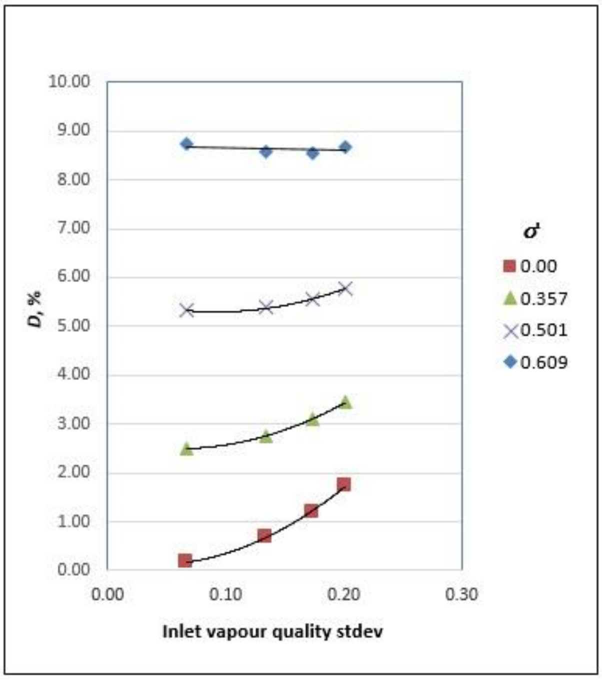


Fig. 9. Trend of *D* vs. inlet vapour quality standard deviation (Mean inlet quality = 0.65, skew = 0.00) for the case of uniform and non-uniform (Normalized standard deviation = 0.357, 0.501, 0.609 and skew = 0.00) flow distribution.

4. Discussion

The results of the numerical study indicate that the trend of the thermal performance degradation factor, *D*, with respect to the statistical moments of probability density function of the two-phase mass flow rate maldistribution is similar with that obtained for single-phase fluids, as reported in the previous work [9, 10]. In summary, it is observed that:

a) *D* varies as the square of flow rate distribution normalized standard deviation
b) *D* varies linearly with the flow rate distribution normalized skew
c) *D* varies as the square of the liquid Reynolds number, Re_L

Larger standard deviations of flow give larger spread of velocities among the tube circuits. The detrimental effect of lower velocities in the tube circuits, which reduces the internal heat transfer coefficient, on reducing the heating capacity outweighs the effect of higher velocities to increase the capacity, which therefore increases *D*. And, flow distribution with positive skews has more tubes with higher fluid velocities which tend to increase the heating capacity and reduces *D*. As for the mean flow rate, as represented by the liquid Reynolds number, the observed critical value of Re_L at a specific flow standard deviation indicates the transition of significance of the internal thermal resistance, r_i, with respect to the external thermal resistance, r_o, which is fixed in this simulation. As the tube flow velocity increases up to the peak of the parabolic curve shown in Fig. 7, the internal heat transfer coefficient increases which causes r_i to decrease, which in turn causes the tube surfaces more sensitive to the degradation effects of maldistribution, i.e., increasing *D*. With further increase of velocities after the peak, the resistance r_i becomes even lower, and the external resistance, r_o, becomes more dominant in determining the effects of maldistribution. As the ratio r_i/r_o decreases, the fixed external thermal resistance acts to dampen the thermal degradation effects, i.e., decreasing *D*.

However, this similarity of trend with single-phase fluids is unexpected in view of the additional vapour quality term x_{IJ}' in Eq. (11). Hence, it is deduced that the contribution of this term in Eq. (11) to the magnitude of *D* is not so significant. The data shown in Fig. 7 supports this hypothesis where it is seen that the magnitude of *D* remains about the same as the inlet vapour quality changes. Furthermore, Fig. 8 shows that the variation of *D*, with both non-uniform flow and inlet vapour quality distribution present, is small. Within the range of standard deviation simulated, the variation is observed to be less than 2%. This also explains the observed phenomenon that *D* becomes insensitive to inlet vapour quality maldistribution as the flow rate standard deviation increases. The degradation effects of flow maldistribution is more significant that inlet quality maldistribution.

5. Conclusions

The thermal degradation factor, *D*, of the heat exchanger due to maldistributed two-phase condensation flow rates is dependent on the statistical moments of the flow profile and the change of vapour quality along the flow direction. The derivation of *D* for two-phase condensation flow shows resemblance to that of single-phase flow.

A discretized numerical calculation of the heat exchanger has shown that the effect on *D* due to the change of vapour quality at the inlet is not as significant as the standard deviation and skew of the flow profile. Flows with low standard deviation and positive skew are preferred to give low magnitudes of *D*.

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