The Influence of Refrigerant (R32) Flow Maldistribution on The Thermal Performance of A Microchannel Heat Exchanger Without Considering Superheat and Sub-Cool

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ABSTRACT

The research work reported in this paper investigates the numerical study on the influence of statistical moments of probability density function of R32 tube-side flow maldistribution on the thermal performance of microchannel heat exchanger. A model without considering superheat and sub-cool was developed to analyze the effect of mean, standard deviation and skew of the refrigerant maldistribution profile on the heating capacity degradation of the heat exchanger. Next, the performance degradation of microchannel heat exchanger due to refrigerant (R32) maldistribution was quantified and analyzed. After that, the model was validated by doing experiment. Finally, a performance deterioration correlation related to refrigerant maldistribution without considering superheat and sub-cool was developed to offer a faster and simpler method to analyze the maldistribution problem. It is found that mean and standard deviation have the highest impact on the performance deterioration. Maldistribution with high standard deviation and high negative skew gives a large magnitude of performance deterioration. For standard deviation more than 0.3, the deterioration can up to 1%. Besides

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that, the deterioration reaches 1% for skew below -0.5. Furthermore, the deterioration is as high as 4% when the mean is less than 0.9. Hence, the first three moments such as mean, standard deviation and skew should be optimized in the effort of improve the thermal performance of the heat exchanger. The proposed correlation in this work offers a faster and simpler method for a quick estimate of the degradation factor.

Keywords: *Refrigerant Maldistribution; Microchannel Heat Exchangers; R32.*

Introduction

The world is now concerned about saving energy. In order to satisfy the needs of the population, it is vital to move forward and improve the energy efficiency of house appliances, such as air-conditioning unit. Hence, various kinds of energy efficiency heat exchangers and refrigerants have been developed and introduced into the market. In this present study, microchannel heat exchanger (MCHE) and R32 refrigerant have been employed due to their superior thermal performance and efficiency.

The MCHE is able to maximize the thermal contact area between the fins and tubes, which increases heat transfer performance when compared to conventional fin-tube heat exchangers. Li et al. [1] have concluded that the MCHE is a highly efficient air-cooled heat exchanger. Thus, the refrigerant maldistribution problem in the MCHE must be analyzed in greater detail to arrive at an optimum refrigeration cycle design which gives excellent power savings to the air-conditioner [2].

The CFCs such as R12 have been phased out in developed countries since 1996 [3]. An alternative refrigerant without ozone depletion and low Global Warming Potential GWP is encouraged to be used in air conditioner system application. Many researchers have recommended using R32 refrigerant, which is also known as difluoromethane, as the heat transfer medium for air-conditioninng applications. This is due to R32 has lower (GWP) as compared to other refrigerants, such as R410A. Xu et al. [4] also found that the cooling capacity and coefficient of performance (COP) with R32 was higher by 10% and 9%, respectively, as compared to R410A. They also stated that R32 had better thermo-physical properties and more environmental friendly [3].

In order to have a comprehensive analysis on the performance of the air conditioning system, the R32 flow maldistribution problem in microchannel heat exchangers must be analyzed. Marchitto et al. [5] have indicated that uneven distribution in heat exchangers is a cause of reduction in thermal performances. The refrigerant flow maldistribution problem becomes even worse in MCHE which has small multiple tube channels and their spacing may be manufactured with a relatively large uncertainty [6]. Chin and Raghavan [2] have indicated that any effort to analyze and predict the detrimental effects due to flow maldistribution must take into consideration the effects of the first three distribution statistical moments. Without consideration of these effects, it would be difficult to analyze properly the tube side maldistribution problem. Chin and Raghavan [7] also concluded that distributions with low standard deviations and positive skews tend to have lower performance degradation. Byun and Kim [8] found that R410A maldistribution in Parallel Flow Condenser caused the heat transfer performance reduced by 13.4% compared to the uniform distribution.

Moreover, a suitable model which investigates the influence of the statistical moments of probability density on refrigerant (R32) flow maldistribution in MCHE should be developed. Cho, Choi, Yoon and Kim [9] developed a model on the mass flow distribution in microchannel heat sink while Kaern and Elmegaard [10] developed a model of a fin and tube evaporator in the object-oriented modeling language and used R410a as heat transfer medium. Chin and Raghavan (2011a) had developed a model which investigated the influence of the higher statistical moments of probability density function of the flow maldistribution profiles on the performance degradation of Heat Exchangers. However, they did not consider the tube-side maldisbution in MCHE and they only analyzed the air side maldistribution in fin tube heat exchanger. Nielsen et al. [11] used water as heat transfer medium in their model. There is none of the researcher develops a model which is able to investigate the impact of performance deterioration for MCHE due to refrigerant (R32) flow maldistribution in term of statistical moments such as standard deviation, skew and mean which is supported by Chng, Chin and Tang [12].

Furthermore, performance deterioration correlation related to refrigerant maldistribution is yet remained a challenge. Ryu and Lee [13] generated new friction factor and Colburn factor for corrugated louvered fins. Villanueva and Mello [14] generated the heat transfer and pressure drop correlations for one finned plate heat exchanger evaluated using CFD (Computational fluid dynamics) simulations. None of them develop the performance deterioration correlation related to refrigerant maldistribution.

With comprehensive analysis on the influence of refrigerant (r32) flow maldistribution on the thermal performance of a MCHE, performance deterioration correlations are generated to estimate the performance deterioration and maximum thermal performance of a heat exchanger.

Model

In this work, a model which considered the influence of the statistical moments of probability density on refrigerant (R32) flow maldistribution in Microchannel Heat Exchanger without considering superheat and sub-cool effect was developed. A microchannel heat exchanger was modeled where the fin pattern, number of passes, number of rows, air temperature and air velocity remained constant. Table 1 shows the initial geometry configuration of condenser.

Table 1: Initial geometry configuration of MCHE

Tubes			**Fins**		
Height	1.3	mm	Pitch	1.2	mm
Width	16	mm	Height	8.2	mm
Number of channels	16		Thickness	0.1	mm
Number of tubes	10				
Number of passes	1		**Louver**		
Hydraulic diameter	0.83	mm	Length	7	mm
			Pitch	1.3	mm
Condenser			Height	0.5	mm
Length	300	mm	Angle	28	°C
Height	103	mm			
Width	16	mm			

The effects of tube-side maldistribution will be investigated while air-side distribution was kept uniform. The performance deterioration due to tube-side maldistribution in term of standard deviation, skew and mean was analyzed. The equations of statistical moments used in this research were shown in below:

$$\text{Mean: } \mu = \frac{\sum_{i=1}^{N} u_i}{N} \tag{1}$$

where u is sample data while N is sample size

$$\text{Standard deviation: } \sigma = \frac{\sum_{i=1}^{N} (u_i - \mu)^2}{N-1} \tag{2}$$

$$\text{Skew: } \gamma = \frac{E(u_i - \mu)^3}{\sigma'^3} \tag{3}$$

where E is the expectation function

The methodology done by this work is similar to that used by Chin and Raghavan [7]. Firstly, a flow maldistribution profile which reflected the three statistical moments (mean, standard deviation and skew) were generated. In order to do this, the face area of the heat exchanger was discretized into 100 elements, as shown in Figure 1. Chin and Raghavan [7] also used the similar mesh size in their studied.

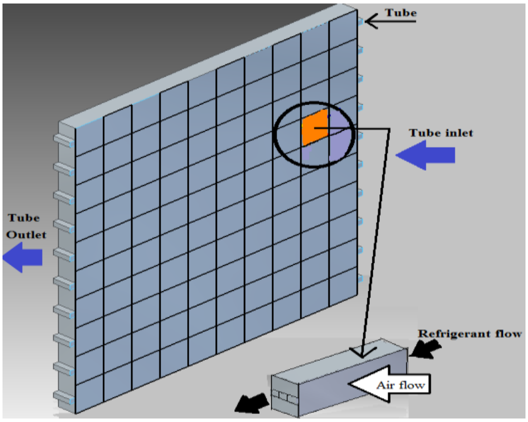

Figure 1: Schematic diagram of heat exchanger with 10 x 10 grids

Each cell element in the coil had its own refrigerant mass flow rate value. Continuous probability density functions (PDF) were used to generate the magnitude of these mass flow rates and their required quantities that constitute the distribution. These PDF functions can be found in Sheskin [15].

The analysis in this research will be performed by using a non-dimensional method where the local mass flow rate for each cell was normalized with the mean value in the range of 0.1 to 2.0, at intervals of 0.1, as suggested by Chin and Raghavan [7]. This practice was also used by Chiou [16] and other researchers. The total area under the PDF curve was equal to 1. Therefore, the quantity of mass flow rates required in the distribution can be obtained by multiplying the area under the PDF curve for any interval with the number of discrete elements. Besides that, the locations of the discrete mass flow rates were placed randomly along the 10 inlet tube of the heat exchanger.

Next, the model without superheat and sub cooling effect was applied in the flow maldistribution profile. The heat duty, q, for each element was calculated by using the ε-NTU method with the known fluid inlet temperatures and mass flow rates. The sum of heat duty for all the elements yield the total exchanger heat duty. The inlet parameters and conditions for the next element were obtained according to the following sequence, which recommended by Copetti et al. [17]:

a) First, calculate external area of tube and fin
b) Next, obtain the global heat transfer coefficient, U as a function of heat transfer coefficient of air and refrigerant using the formula below:

$$1/ UA = 1/(h_{air}A_{fins}) + 1/(h_{Ref} A_{tube}) \tag{4}$$

where A_{fins} and A_{tube} are the area of fin and tube, respectively.

c) After that, calculate the number of transfer units, NTU, and effectiveness, $\mathcal{E}$, according to the two phase flow in the element:

$$NTU = UA/ C_{min} \tag{5}$$

where C_{min} is the minimum heat capacity rate while A is the effective heat exchange area. The effectiveness is given by the following equation which is used by Shah and Seulic [18]:

$$\mathcal{E} = 1 - \exp(-NTU) \tag{6}$$

d) Finally, the heat transfer rate for each element, q, is obtained by:

$$q = \mathcal{E}C_{min} (Ti_{Ref} - Ti_{air}) \tag{7}$$

where Ti_{Ref} and Ti_{air} are inlet temperatures of refrigerant and air, respectively.

The outlet enthalpy, i_o, and dryness fraction, x, of the element can be obtained by following equations:

$$q = \dot{m} (i_o - i_i)$$
$$x = (i_o - i_f) /(i_{fg}) \tag{8}$$

where $\dot{m}$ is mass flow rate, i_i is inlet enthalpy and i_{fg} the condensation enthalpy.

Heat Transfer Correlation

The air side heat transfer coefficient is calculated using the equation as below:

$$j = (h_{air}\, P_r^{\,2/3})/c_p G \tag{9}$$

where G is mass flux, c_p is specific heat and P_r is Prandtl number of air.
The Colburn factor, j, recommended by Chang and Wang [19] for louvered fins is shown as follows:

$$j = (Re_{Lp})^{-0.49}(\Theta/90)^{0.27}(F_p/L_p)^{-0.14}(F_l/L_p)^{-0.29}(T_d/L_p)^{-0.23} (L_1/L_p)^{0.68}(T_p/L_p)^{-0.28}(\delta_f/L_p)^{-0.05} \tag{10}$$

where Re_{Lp} is Reynolds number based on louver pitch, Θ is louver angle, F_p is fin pitch, L_p is louver pitch, F_l is fin length, F_d is fin depth, L_l is louver length, T_p is tube pitch and δ_f is fin thickness.

The refrigerant side heat transfer coefficient is calculated by using the correlation recommended by Shah [20]. The equations used are shown as below:

$$h_{ref} = h_L\{(1-x)^{0.8} + [3.8x^{0.76}(1-x)^{0.04}]/P_{rl}^{0.38}\} \tag{11}$$

$$h_L = (0.023 Re_L^{\,0.8}\, P_{rl}^{\,0.4}\, k_l)/D_h \tag{12}$$

where h_L, is the heat-transfer coefficient assuming all the mass flowing as liquid, P_{rl} is Prandtl number of liquid, Re_L is the Reynolds number assuming all mass flowing as liquid, k_l is the thermal conductivity of liquid and D_h is the hydraulic diameter of the microchannel tube.

Lastly, the deterioration factor, D, which is shown in equation (13) is calculated. This is presented by Chin and Raghavan [2].

$$D = (Qu - Qm)/Qu \ \times 100\% \tag{13}$$

where Qu is the heat transfer capacity for uniform refrigerant flow while Qm is the heat transfer capacity for non-uniform refrigerant flow.

Pressure Drop Correlation

The refrigerant side pressure drop coefficient is calculated using the two phase pressure drop correlation introduced by Mishima and Hibiki [21]. The equations used in this work are shown as follows:

$$\Delta P_{tp} = \Delta P_{tp,f} + \Delta P_{tp,a}$$

where ΔP_{tp} is the total two-phase pressure drop, $\Delta P_{tp,f}$ is the pressure drop due to frictional loss and $\Delta P_{tp,a}$ is the pressure drop due to acceleration.

$$\Delta P_{tp,f} = \frac{L}{x_{out}} \int_{x_{in}}^{x_{out}} \frac{2 f_f G^2 (1-x)^2 v_f}{d_h} \emptyset_f^2 \, dx \, ;$$

$$\emptyset_f^2 = 1 + \frac{C}{X_{vv}} + \frac{1}{X_{vv}^2}; \; C = 21[1-\exp(-0.319 \text{ x } 10^3 d_h)];$$

$$X_{vv} = \left(\frac{\mu_f}{\mu_g}\right)^{0.5} \left(\frac{(1-x_{out})}{x_{out}}\right)^{0.5} \left(\frac{v_f}{v_g}\right)^{0.5} \tag{14}$$

$$\Delta P_{tp,f} = G^2 v_f \left[\frac{x_{out}^2}{\alpha_{out}}\left(\frac{v_f}{v_g}\right) + \frac{(1-x_{out})^2}{1-\alpha_{out}} - 1\right]];$$

$$\alpha_{out} = \frac{1}{1 + \left(\frac{(1-x_{out})}{x_{out}}\right)\left(\frac{v_f}{v_g}\right)^{2/3}}$$

(15)

where L is the length of micro-channel, f_f is the friction factor based on local liquid flow rate, v_f is the specific volume of liquid, v_g is the specific volume of gas, dh is the hydraulic diameter, μ_f is the viscosity of liquid and μ_g is the viscosity of gas.

Figure 2 shows the summary of model development without superheat and sub-cool effect. Firstly, the flow maldistribution profile with varied statistical moments was generated by continuous PDF. Then, the heat thermal performance of the specific maldistribution profile (based on the magnitudes and quantities of mass flow rate generated by continuous PDF) and uniform distribution profile (based on average value of the maldistribution profile) using ε-NTU method. Finally, the thermal deterioration factor of the specific maldistribution profile was calculated.

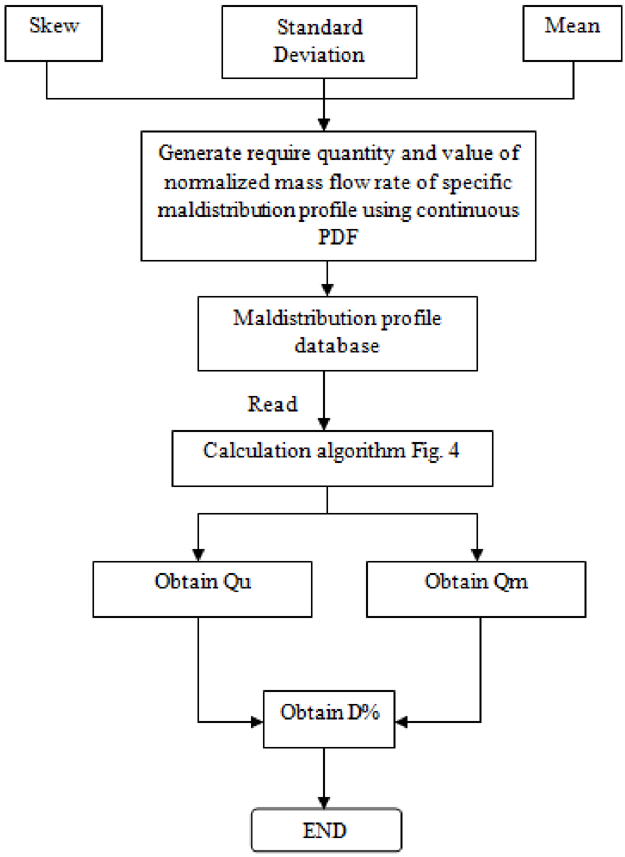

Figure 2: Summary of model development

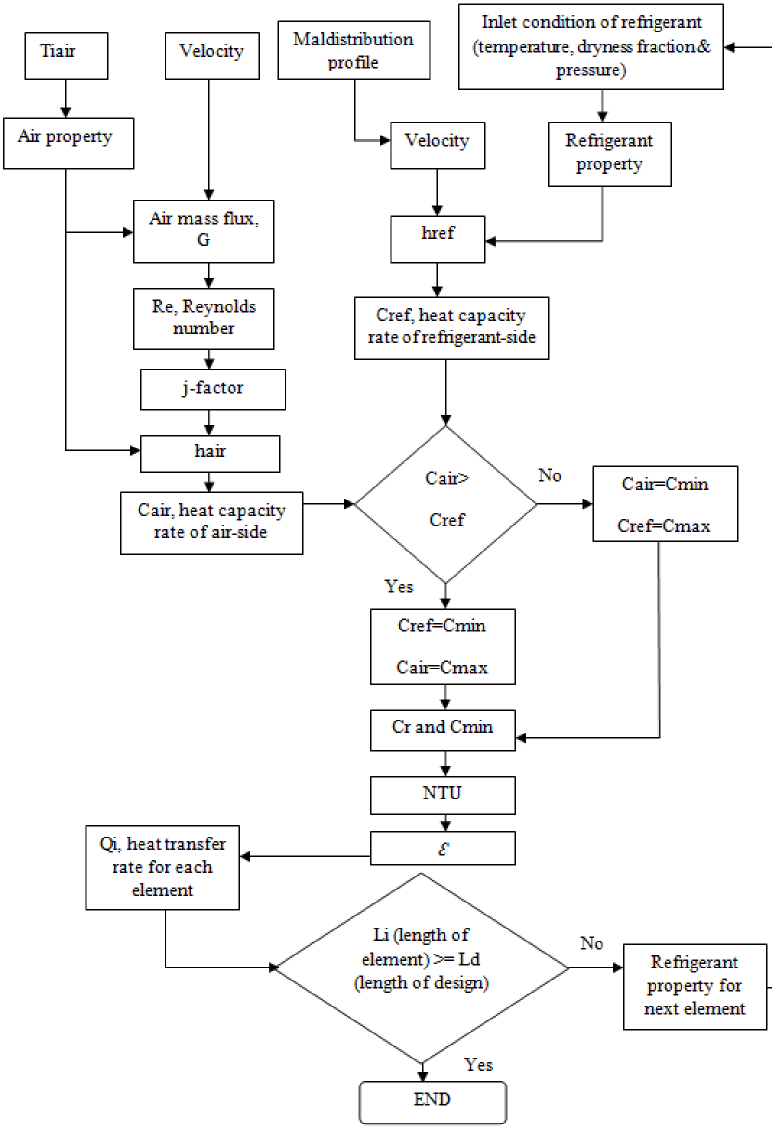

Figure 3: Calculation algorithm to obtain thermal performance

Observation and Discussion

Figure 4 shows the performance deterioration factor, D versus standard deviation. From the graph, it is shown that higher tube-side standard deviation has significant impact on the D. This is due to the larger difference between the highest and lowest velocities in the profiles with high standard deviation. Hence, the heat transfer capacity of the whole heat exchange performance drops due to the lower internal heat transfer coefficients arising from the lower refrigerant mass flow rate [7].

When the standard deviation approaches 0.1, the deterioration factor is almost equal to zero. Besides that, the magnitude of D calculated is as low as 2% although the standard deviation is high at 0.50. It is observed that the trend of deterioration versus standard deviation corresponds well with the findings of the prior work [7] where the D increases to the square of standard deviation. Nielsen et al. [11] also stated that the effective thermal performance of heat exchanger decreased as the standard deviation increased.

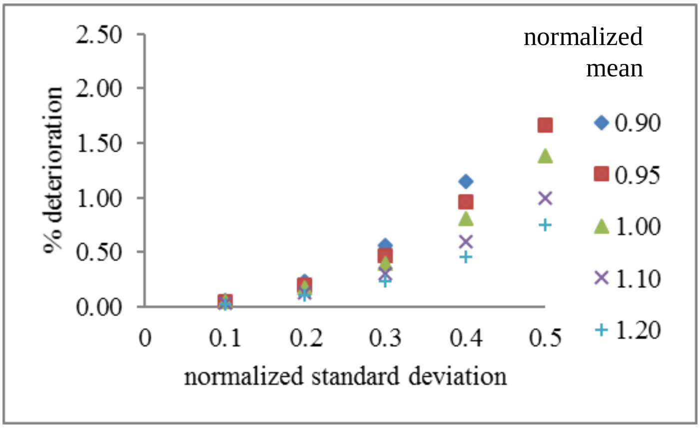

Figure 4: Effect of standard deviation on D (skew=0)

Figure 5 shows the performance deterioration versus skew. It is observed that skew has no significant effect when the standard deviation is lower than 0.10. However, D varies proportionally with skew when the standard deviations is higher than 0.1. It is found that the D becomes smaller at higher positive skews. This is due to the flow distribution having a larger portion of higher velocities which counteract the negative effects of the lower velocities [7].

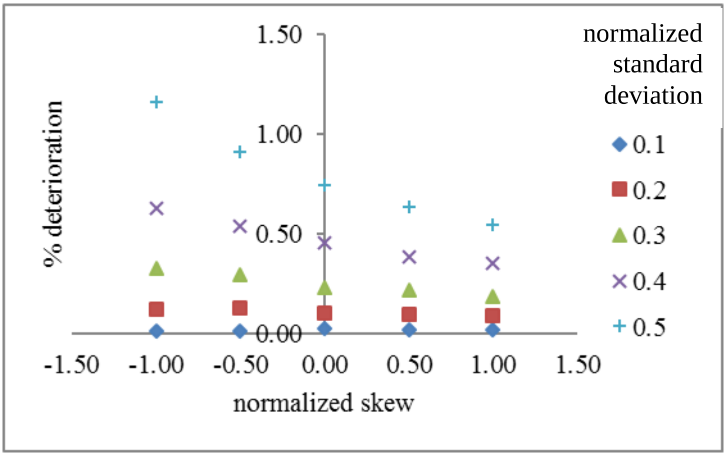

Figure 5: Effect of skew on D (mean=1.2, ṁ=0.013kg/s)

Figure 6 show the effect of mean on the performance deterioration while figure 7 shows the effect of mean on the heat transfer capacity. From the graph, it is obviously showed that lower mean have a higher impact on the D. Besides, the D is greater at a lower mean and larger standard deviation. At larger mean, the internal heat transfer coefficient becomes higher and increase the heat transfer capacity. However, it is found that the heat transfer capacity increases logarithmically when mean increases which is showed in figure 7. This is due to the higher heat transfer rate through the heat exchanger causing it to be less susceptible to the degradation effect caused by maldistribution [7]. In this situation, the external resistance (air side) becomes more significance as the internal resistance (refrigerant side) approaches zero due to high internal heat transfer coefficient which is supported by Chin and Raghavan [7]. Hence, the impact of maldistribution has been minimized and the magnitude of D has been reduced significantly. Nielsen et al. [11] also stated that large thermal conductivity did not show a major impact on the flow maldistribution.

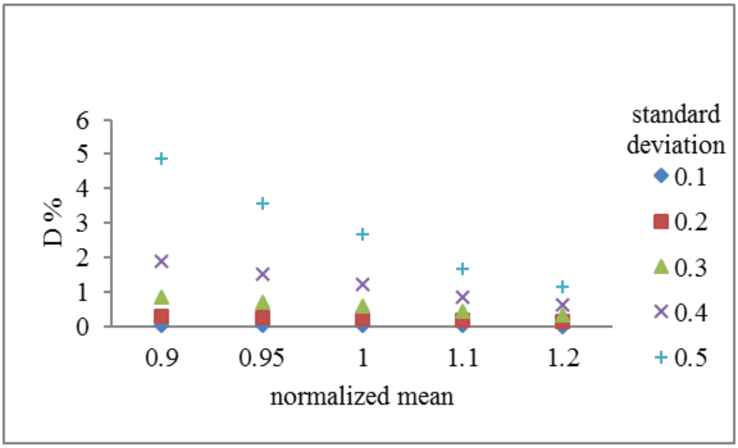

Figure 6: Effect of mean on D with different standard deviation (skew=-1, $\dot{m}$=0.013kg/s)

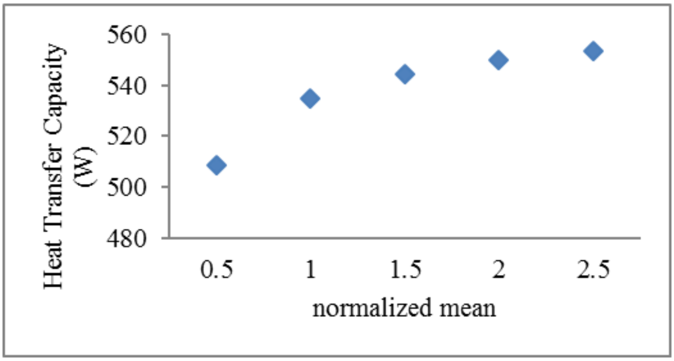

Figure 7: Effect of mean on the heat transfer capacity (uniform flow)

Correlation Development

Next, performance deterioration correlation due to refrigerant maldistribution was developed by determine the relationship between the statistical moments and performance deterioration of MCHE.

From the results, it is observed that D is a function of the normalized standard deviation, skew and mass flow rate. It is observed that the 2nd order of polynomial equation fits well with the trend of normalized standard deviation versus D. The similar trend is observed for normalized skew versus D and normalized mass flow rate versus D. By applying the observed trends in the preceding sections for these parameters, the form of the correlation to predict the magnitude of D is proposed to be the following:

$$D=(a\sigma'^2+b\sigma')(c\,\gamma'^2+d\,\gamma'+e)(f\mu'^2+g\,\mu'+h) \tag{16}$$

where σ' is normalized standard deviation, γ' is normalized skew, μ' is normalized mass flow rate, a, b, c, d, e, f, g, h are constants.

Finally, the constants were then solved by non-linear regression analysis. The constants calculated by using the Datafit software [22] and the solutions are given in Table 2. However, the correlations are only valid for even air flow distribution and without considering superheat and sub-cooling.

Table 2: Constant for Performance Deterioration Correlation

Constant	Value
a	3.36
b	-0.37
c	0.28
d	-0.65
e	1.26
f	0.28
g	-1.90
h	4.52

The proposed correlation in this work offers a faster and simpler method to analyze the maldistribution problem. The designers of HVAC (heating, ventilating, and air conditioning) system only need to insert the value of statistical moments and mass flow rate into the algebraic correlation equation for a quick estimate of the degradation factor. This will shorten the development of HVAC system and save lots of cost to achieve the target performance of HVAC products.

Verification by Simulation

A simulation was carried up by using Coil Designer software to verify the mathematical model which developed by the author. Jiang et al. [23] stated, "An overall agreement of 10% is found between the simulation results and the experimental data" (p. 607). This has proven that the results from Coil Designer is accurate and can be trusted. Table 3 shows the correlation used in Coil Designer software [23].

Table 3: Correlation used in Coil Designer [23]

Heat Transfer Correlation	
Air Side	Chang and Wang (1997)
Refrigerant Side (single phase)	Gnielinski (1976)
Refrigerant Side (two phase)	Shah (1979)
Pressure Drop Correlation	
Air Side	Chang and et al. (2000)
Refrigerant Side (single phase)	Churchill (1977)
Refrigerant Side (two phase)	Mishima and Hibiki (1996)

In the software simulation, the refrigerant distribution was remained uniform while the mass flow rate was varied. The results from Coil Designer were compared with the predicted result from the mathematical model developed by the author. Figure 8 shows the simulated results versus the predicted results from mathematical model. From the result, it is found that most of the predicted data agrees well within 5% of the simulated data from Coil Designer. According to ISO 5151 [24], the maximum tolerance of capacity is 10% and the results from the mathematical model developed by the author shows a very good promising accuracy.

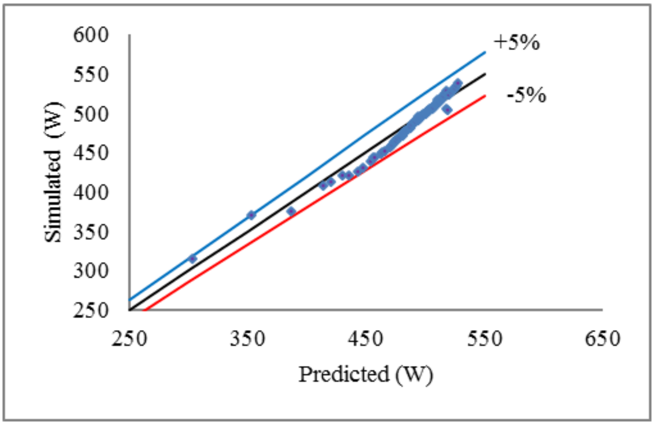

Figure 8: Simulated result versus predicted result from mathematical model

Conclusion

The objective of this research is to investigate the influence of statistical moments of probability density function of R32 tube-side flow maldistribution on the thermal performance of microchannel heat exchanger. By analyzing the thermal degradation effects, the distribution of refrigerant flow rate in the tubes of the heat exchanger can be optimized. Hence, the heating capacity can be increased which leads to higher energy efficiency rating of the air-conditioning unit.

In order to have a comprehensive analysis of the refrigerant maldistribution, it is recommended that the statistical moments of refrigerant flow distribution in the tubes such as mean, standard deviation and skew, to be optimized. From the simulation result, it is found that D is greater at lower mass flow rates. Besides that, it is observed that a maldistribution profile with low standard deviation and high positive skew is preferred in order to minimize the deterioration effect. For normalized standard deviation more than 0.3, D can up to 1%. The D can up to 1% for normalized skew below -0.5. Furthermore, D is as high as 4% when the normalized mean is less than 0.9. Finally, a performance deterioration correlation related to refrigerant maldistribution without considering superheat and sub-cool was developed to provide a faster solution to analyze refrigerant maldistribution problem and enhance the process of developing new HVAC system.

However, there are some limitations for the performance deterioration correlation due to refrigerant flow maldistribution. The biggest disadvantage for this correlation is the correlation is not applicable to those condensers which have superheat and sub-cool. In reality, most of the condensers in HVAC products have superheat and sub-cool. Thus, it is recommended that a comprehensive performance deterioration correlation considering superheat and sub-cool effect is developed.

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