An Experimental Study of the Direct Contact Heat Exchanger for Ice Slurry Production

Raksit Thitipatanapong and Bundit Limmeechokchai*
Sirindhorn International Institute of Technology, Thammasat University
P.O. Box 22 Thammasat Rangsit Post Office, Pathum Thani 12121, Thailand
**Corresponding Author Email: bundit@siit.tu.ac.th*

Abstract

An experimental pilot scale direct contact heat exchanger (DCHE) for ice slurry production was fabricated and evaluated. This study investigates the DCHE of Ø114mm and 1000 mm height using evaporated refrigerant as the disperse phase and solidified water as the continuous phase. The heat transfer rate across the DCHE was varied between 3.0 and 6.5 kW while the water flow rate across a heat exchanger was varied, and its inlet temperatures was controlled at 13°C and 15°C. As a result, the freezing point deviation of the ice-water solution was found to be 7.95°C. Moreover, the first law of thermodynamics indicated that the DCHE could be considered as a theoretical adiabatic mixing chamber. The selected performance parameters reveal that the stream conditions of water and refrigerant flow across the direct contact heat exchanger has a high influence on the system performance.

Keywords: Ice slurry production, direct contact heat transfer, cool storage

1. Introduction

Latent heat thermal storage has a high energy intensity relating to the volume of the storage tank. Ice is used as a storage medium utilizing latent heat of fusion as 333 kJ kg^{-1} at the freezing point of 0°C. Although ice has advantages compared to other storage media, both in energy intensity and cost, it has disadvantages that result in low energy efficiency during ice formation. Alternatively, direct contact heat transfer in ice formation has been proven as an efficient ice production method for thermal storage whereby the heat transmittance is maintained throughout the freezing process. The water/ice is the continuous phase and the refrigerant is the disperse phase. The cold refrigerant is fed in at the bottom, absorbing the heat from the water until the water, is frozen and the refrigerant becomes superheated vapor.

The direct contact package latent heat energy storage was proposed by Utaka et al. [1]. In addition, the achievement of this device stated that the refrigerant must: (1) be insoluble or barely soluble with water, (2) have a larger liquid density than water, and (3) have a low boiling point and sufficiently high vapor pressure compared to that of water. Moreover, the experimental investigation on a Ø261 mm and 400 mm long pipe with CFC-12 and CFC-114 showed results in faster charging rate (7MJ in 5400 seconds) than conventional methods of growing ice from the cooled wall because the thermal transmittance was maintained at high levels throughout the entire storage process. Thus the ice produced by this method was porous. Furthermore, Utaka et al. [2] re-investigated using a larger experimental rig size Ø314 mm and 1030 mm long with HCFC-142b that stored about 7MJ energy in 12000 seconds. The results correlated with the previous experiments, but the ice formation temperature of water deviated from 0.0°C to 13.1°C reported. Y.P. Lee et al. [3] made the comparative experimental study between the direct contact package latent heat storage and conventional ice-on-coil system using the HCFC-141b as the refrigerant. The heat transfer coefficient was found to be about 3.25 times higher than the theoretical heat transfer coefficient for a ice-on-coil system. In addition, a freezing point deviation of water from 0°C to 2°C was experienced.

In 1998, Kiatsiriroat et al. [4] proposed another concept that the direct contact heat exchanger for ice production be integrated into the evaporator of the refrigeration unit. It was an ordinary direct contact heat exchanger with a size of Ø300 mm and 1100 mm height. It contains 40 kg of water as the phase change material and a 3.5 ton refrigeration unit. Furthermore, the investigation of the entire system performance using CFC-12 showed results of a relatively higher COP of the refrigeration unit (about 3.4 to 3.6). The performance of the direct contact evaporator, in terms of heat transfer coefficient depended on the mass flow rate of refrigerant. Moreover, the thermal analysis of the direct contact evaporator applied the lump model, the first law of thermodynamics (the energy equation), to the experiment [5]. Therefore the experimental results were correlated. In addition, the volumetric heat transfer coefficients were investigated, and they were found to be in the range from 2 to 16 kW $m^{-3}K^{-1}$ and from 2 to 52 kW $m^{-3}K^{-1}$ for CFC-12 and HCFC-22 respectively. Consequently, the dimensionless heat transfer correlation was within acceptable errors of 30%. Previously, Thitipatanapong et al. [6-7] designed a ice slurry production system using the direct contact heat transfer concept acquiring the parameter called average volumetric heat transfer coefficient ($U_{v,avg}$) over freezing process to analyze its efficiency. The system was an ordinary direct contact heat exchanger with a size of ∅114 mm and 1000 mm height. It included a 50-liter water storage tank and cooling capacity of 2 refrigeration tons. Moreover, the comparative study between the selected refrigerants which are HFC-134a and CFC-12, was conducted. The experimental results revealed that ice formation temperatures were found to be 8.8°C for HFC-134a and 4.4°C for CFC-12. Furthermore, the stream conditions inside the DCHE dominated overall performance of the system as the $U_{v,avg}$ tended to increase with increasing either water feed for ice slurry production or refrigerant feed (ranged from 55 to 250 kW/m^3K). The investigations from the experiments also showed that the COP of the cooling system was related directly to $U_{v,avg}$. However, the exact relation was not determined because the experimental fluid streams did not combine together and the other parameters were not controlled precisely.

In this study, an experimental DCHE was analyzed using the adiabatic mixing chamber mode as the reference to the ideal condition. Furthermore, the working fluids used in the DCHE are evaporated HFC-134a as the disperse phase and solidified water as the continuous phase, and the water feed to the DCHE was controlled at 13°C and 15°C.

2. Apparatus and Procedure

The heart of the system is the DCHE, where the water and the refrigerant were mixed and heat is transferred to each other is transferred to produce ice, which was 114 mm (4 inches) nominal diameter stainless steel pipe with 1100 mm height, 11 liters of volume, with flange connection, and the refrigerant mass flux inside varied from 1.50 to 3.20 kg $m^{-2}s^{-1}$ which reflects the heat transfer rate from 3.0 to 6.5 kW. Moreover, the water mass flux varied from 8.00 to 14.50 kg $m^{-2}s^{-1}$. Three sight glasses were mounted longitudinally along the DCHE for observing inside phenomenon. The sight glasses were 12 mm thick tempered-glass with 80 mm diameter. At the bottom, there are two inlet ports; the refrigerant passes through the distributor from the bottom while the water inlet to the DCHE is at the side above the distributor as shown in Figure 1.

In addition, the auxiliary devices were installed in the experimental facility as shown in Figure 2 which consists of 3.7 kW_e vapor compressor refrigeration unit, two sets of 4 kW_e electric heaters, a positive displacement water pump and measurement devices.

To prepare the experiments, first of all, reverse-osmosis purified water is charged into the receiver. Then, the total system is evacuated by the vacuum pump. As it reaches vacuum (approximately 2.33 kPa_{abs}), the water become degassed water. If the water does not reach the degassed state, the condensation process in the intermediate heat exchanger will stop after operating for a while. Finally, the refrigerant vapor is charged at the inlet of the intermediate heat exchanger. To start the experiments, the refrigerator is turned on to develop circulation of the refrigerant in the system and to reduce the pressure of the system. Then, the positive displacement pump is activated causing the water to circulate in the system. After developing the flow of both streams, electricity is supplied for simulated a heat load which is regulating for controlling the water inlet temperature at 13°C and 15°C. Finally, the system is stabilized within 30 to 45 minutes; after that the data is logged for 10 to 15 minutes.

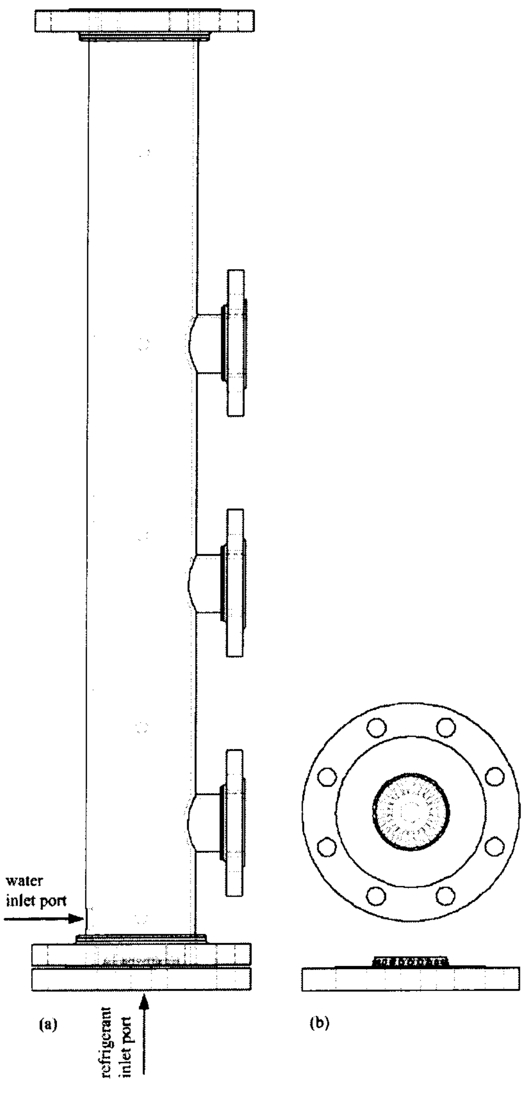


Figure 1 drawings of (a) the direct contact heat exchanger and (b) distributor.

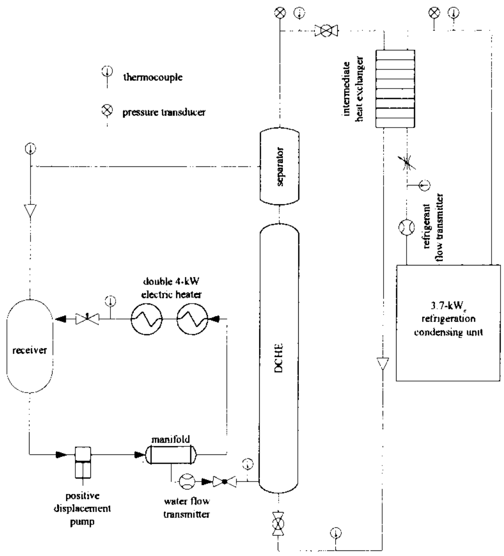


Figure 2 Schematic diagram of the experimental facility including auxiliary devices.

3. Data Analyses

The analysis models for evaluating the performance of the DCHE are described. The DCHE can be considered as an adiabatic mixing chamber. Moreover, the adiabatic mixing chamber model, as a theoretical performance, is used to compare the theoretical results with the experimental results as illustrated in (1) that is assumed to have equalized temperatures at the outlet in both streams.

$$\dot{m}_w h_{w,in} + \dot{m}_r h_{r,in} = \dot{m}_w h_{w,out} + \dot{m}_r h_{r,out} \quad (1)$$

Where $h_{w,out}$ and $h_{r,out}$ have the same temperature.

Furthermore, the mean temperature difference (*MTD*) as a significant key parameter of the heat exchanger is used to represent the average temperature difference between hot and cold fluids especially for latent heat transfer in this heat exchanger. The cold stream in this application is evaporating which is an isothermal heat transfer process, so the mean temperature of the cold stream is equal to the saturation temperature of refrigeration at the operating pressure ($T_{r,sat@pressure}$) as shown in (2).

$$MTD = \left| \frac{(T_{w,out} + T_{w,in})}{2} - T_{r,sat@pressure} \right| \quad (2)$$

However, the *MTD* does not include the size of the DCHE; so, the key performance is revealed as the volumetric heat transfer coefficient (U_v) that is shown in (3) which originated from Newton's law of cooling, where V and $\dot{Q}$ are volume and heat transfer rate, respectively, of the DCHE.

$$U_v = \frac{V(MTD)}{\dot{Q}} \tag{3}$$

Another utilized parameter is called the latent heat factor (*LHF*) which is the ratio of energy transfer to the water/ice solution in latent heat and total heat transfer shown in (4). It is used to explain the criteria in designing the effective DCHE.

$$LHF = \frac{\dot{Q} - \dot{m}_w C_{p,w}(T_{w,in} - T_{w,out})}{\dot{Q}} \tag{4}$$

Moreover, the fluid streams, both water and refrigerant, across the DCHE are defined as mass flux ($\ddot{m}$) which is the ratio between mass transfer rate ($\dot{m}$) and cross-sectional area of DCHE (A_{DCHE}) and shown in (5)

$$\ddot{m} = \frac{\dot{m}}{A_{DCHE}} \tag{5}$$

4. Results and Discussion

The ice slurry formation temperature was expected to deviate from the natural freezing point of pure water at 0°C. There had been ice formation temperature deviation because of water contamination with the contact refrigerant changing its thermo-physical properties. The temperature of the ice-water solution at the outlet of the DCHE were examined using statistical analysis from 113 experimental tests that included controlled temperatures at 13°C and 15°C.

As a result, the average temperature of the ice-water solution was 7.95°C with a standard deviation (SD) of 0.785 and the statistical distribution curve is illustrated in Figure 4. Consequently, this value would be utilized to represent the ice formation temperature in the DCHE in the following analyses.

Figure 3 Ice slurry produced from the system.

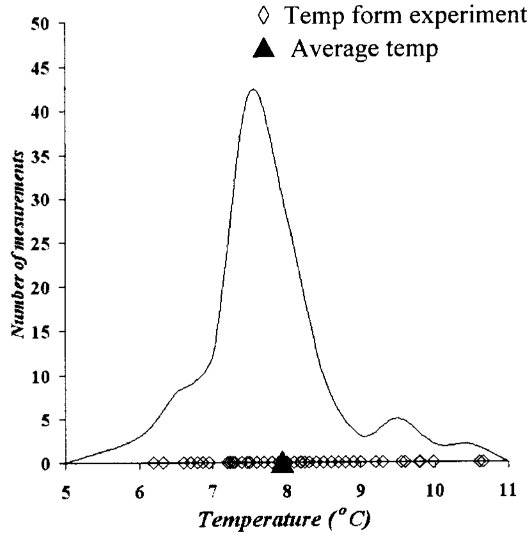


Figure 4 Distribution curve of ice-water solution temperature.

Next, an adiabatic mixing chamber was assumed for the DCHE to be metered so that the system operates close to a theoretical process which should be the same temperature at the exit of the DCHE. The comparison of the experimental results is illustrated in Figure 5, and most of experimental results lay within ±10% of error. It could be concluded that the heat exchanger operates close to the ideal condition which sometimes is considered as uneconomically excessive due to the construction of the heat exchanger.

Moreover, referring to the adiabatic mixing assumption, the *MTD* can be calculated theoretically as the refrigerant evaporation temperature, $T_{r,sat@pressure}$, and ice-water solution at outlet, $T_{w,out}$, were assumed to be the same 7.95°C, so the theoretical estimation of *MTD* were 2.5°C and 3.5°C for the controlled water inlet temperature, $T_{w,in}$, at 13°C and 15°C, respectively as shown in Figure 6(a) and 6(b).

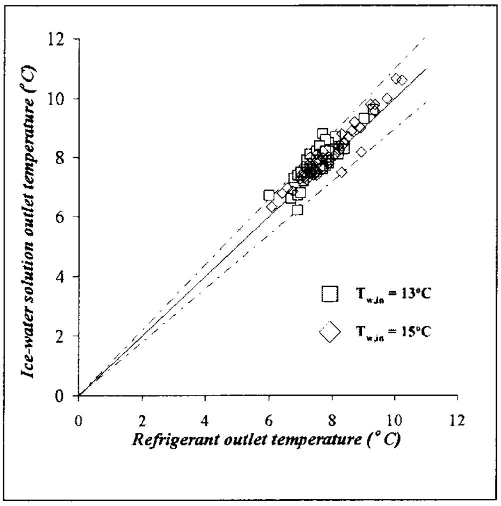
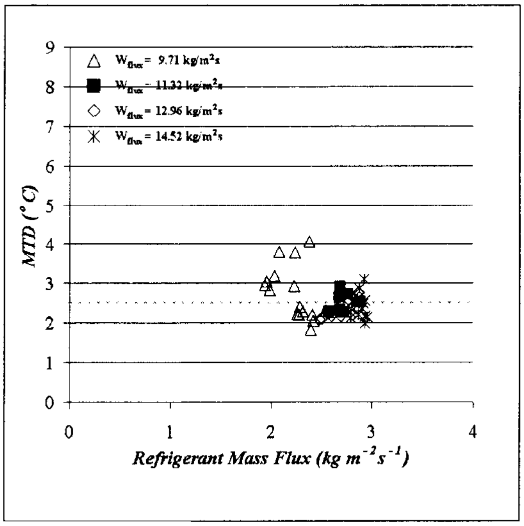


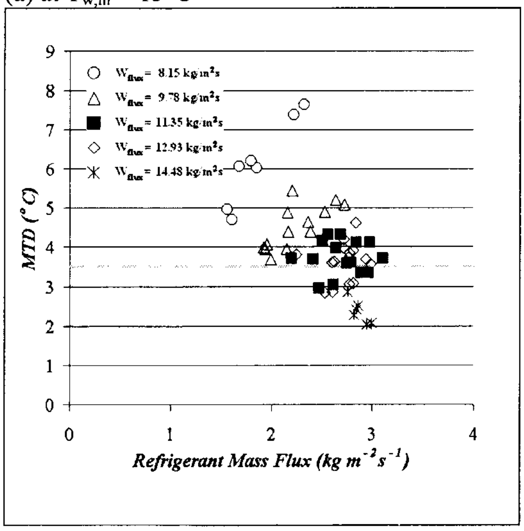
Figure 5 Comparison of temperature at outlet between both streams.

According to Figure 6(a), the *MTD* reached theoretical condition at 2.5°C when the refrigerant mass flux across the DCHE was higher than 2.00 kg $m^{-2}s^{-1}$, and the water mass flux was higher than 11.32 kg $m^{-2}s^{-1}$. Similarly, at another controlled temperature, the *MTD* approaches ideal condition at 3.5°C with the refrigerant mass flux and water mass flux greater than 2.00 kg $m^{-2}s^{-1}$ and 11.35 kg $m^{-2}s^{-1}$, respectively. In summary, it is obvious that fluid streams, both the refrigerant and the water, have a strong influence on the *MTD* as the refrigerant mass flux was increased, causing the *MTD* to be close to the theoretical *MTD*, also, the *MTD* approached the theoretical value with increasing water mass flux.

However, the *MTD* did not show the performance, including the capacity and size of the DCHE. To evaluate the overall performance, the volumetric heat transfer coefficient (U_v) is introduced. Figure 7 shows that U_v tended to increase with the increasing fluid stream conditions, which were the refrigerant and water mass flux. These results correlated with the *MTD*, a high value of U_v could be reached when the *MTD* approached the theoretical value.



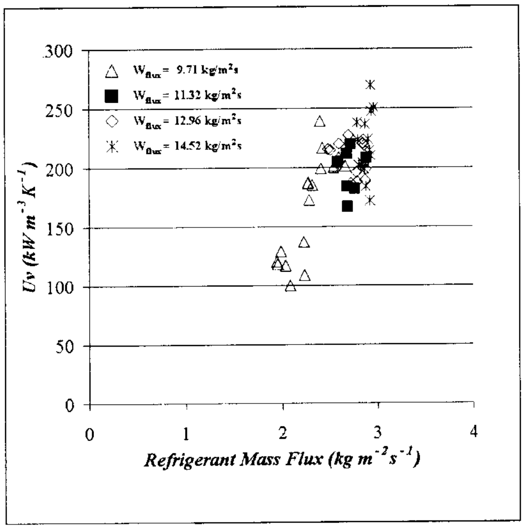
(a) at $T_{w,in}$ = 13°C



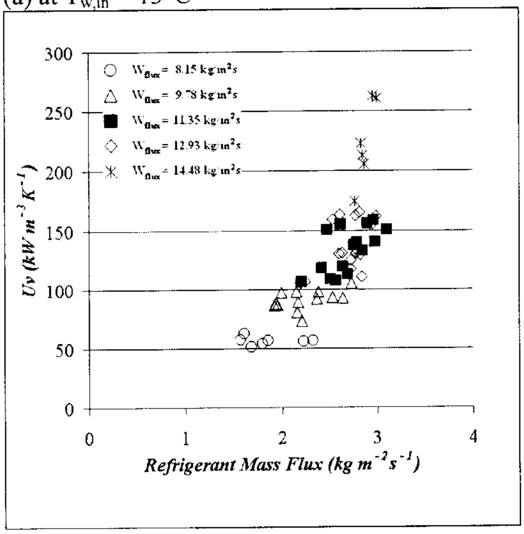
(b) at $T_{w,in}$ = 15°C

Figure 6 Mean temperatures (*MTD*) vs. refrigerant mass flux at various water mass flux.

Furthermore, it is obvious that the U_v in Figure 7(a) was shifted to be higher than Figure 7(b) since the former operated closer to ice formation temperature than the latter and had a lower theoretical *MTD*.



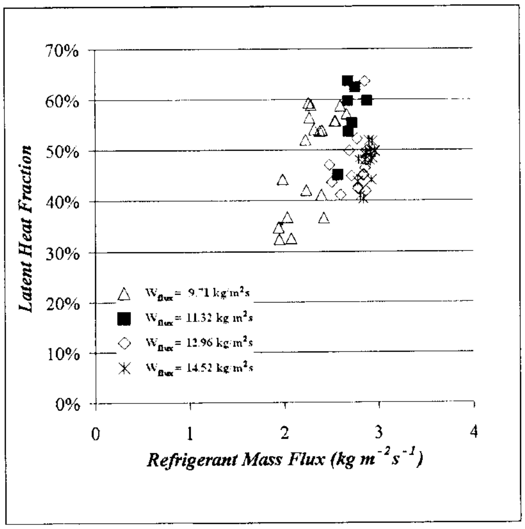
(a) at $T_{w,in}$ = 13°C



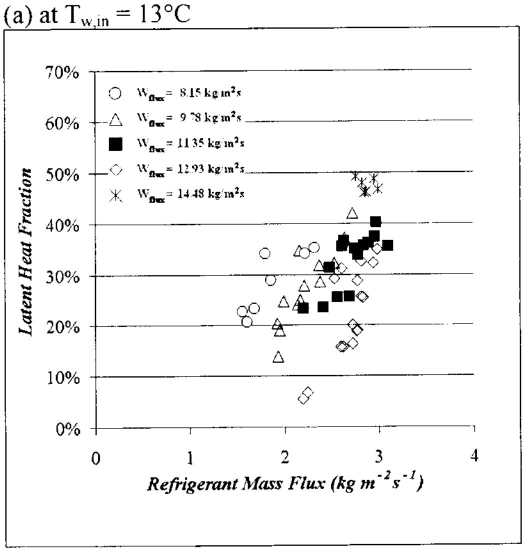
(b) at $T_{w,in}$ = 15°C

Figure 7 Refrigerant mass flux vs. the performance of DCHE in various water mass flux.

Another parameter was latent heat fraction (*LHF*) that is the amount of energy usage for changing phase from water to ice per total energy transfer in the DCHE. Figure 8 shows them in various fluid stream conditions. It is obvious that the *LHF* increases with refrigerant mass flux while accelerating water mass flux reduced gradually the percentage of the *LHF* for both controlled temperatures. In addition, the lower controlled temperature in Figure 8(a) had obviously higher *LHF* than the lower controlled temperature in Figure 8(b).



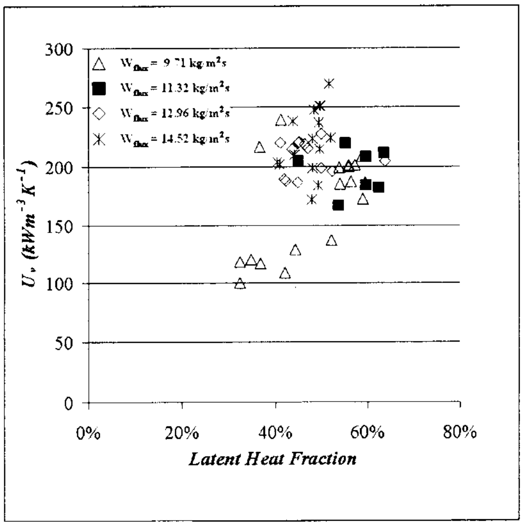
(a) at $T_{w,in}$ = 13°C



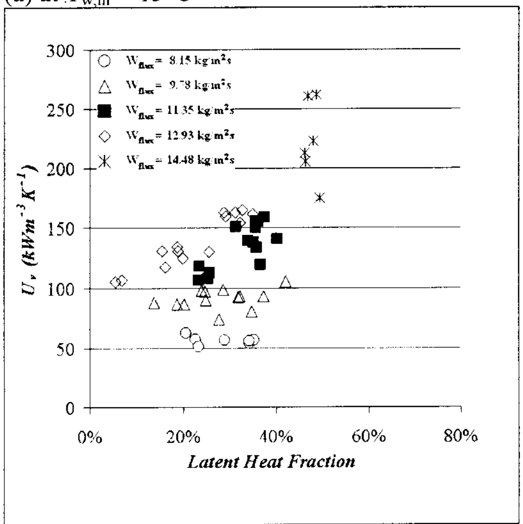
(b) at $T_{w,in}$ = 15°C

Figure 8 Refrigerant mass flux vs. latent heat fraction in various water mass flux.

The *LHF* can be utilized to advise the effective, U_v, criteria of the DCHE as illustrated in Figure 9(a) and 9(b). Obviously, the *LHF* and U_v were related to each other. Moreover, for both controlled temperatures, the U_v tended to be higher for a higher latent heat fraction.



(a) at $T_{w,in}$ = 13°C



(b) at $T_{w,in}$ = 15°C

Figure 9 Volumetric heat transfer coefficient compared to Latent heat fraction at various stream conditions.

Therefore the effective hess of the DCHE can be established at high *LHF*. However, the U_v dropped for a reduction of water mass flux but, theoretically, rising water mass flux means reduction of the *LHF* so the stream conditions limits the performance of the DCHE.

In conclusion, the experimental results obviously agreed with the previous experiments that the U_v tended to increase with increasing of the refrigerant mass flux at the same series of water mass flux. In addition, the multiplication of water mass flux also accelerated the U_v. Ultimately, from the analyses, the criterion to establish ideal operating condition is highlighted that the *LHF* should be kept at the maximum possible in order to receive high efficiency while the deceleration of the fluid stream conditions caused the deficiency that resulted in the system not operating at the ideal operating conditions. As suggested for this DCHE, the refrigerant mass flux should be maintained higher than 2.00 kg $m^{-2}s^{-1}$, and the water mass flux is suggested to be maintained higher than 11.30 kg $m^{-2}s^{-1}$.

5. Conclusion and remarks

The experimental investigation of direct contact heat exchanger for ice slurry production using evaporated HFC-134a as the disperse phase and solidified water as the continuous phase was performed. The results obtained in this research are summarized as follows. The freezing point deviation from the natural freezing point of water was found as expected. Although the first law of thermodynamics indicated that the system for every fluid stream condition operated close to the theoretical conditions, the further parameters, mean temperature difference (*MTD*) and volumetric heat transfer coefficient (U_v), revealed that the DCHE performances depended on the fluid stream condition across the heat exchanger. Ultimately, the minimum operating conditions to make an effective DCHE have been proposed.

6. Acknowledgements

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7. References

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