Performance of a Direct Contact Heat Exchanger with Meshes for a Solar Thermal Energy System

Nam-Jin Kim
Graduate School, Inha University
Chong-Bo Kim*, Tae-Beom Seo
Department of Mechanical Engineering, Inha University
Byung-Ki Hur
Department of Biological Engineering, Inha University

In order to improve the efficiency of a direct contact heat exchanger for a solar thermal energy system, the working fluid should be dispersed into small and uniform droplets, and stay within a heat exchanger for a long time. Therefore, installation of meshes in a direct contact heat exchanger is suggested in the present study, and the performance of the direct contact heat exchanger with several layers of meshes is experimentally investigated. Diethyl phthalate is used as the working fluid, and the performance of the heat exchanger is tested for several different operating conditions and compared to that of the heat exchanger without meshes. The results of this investigation show that meshes make droplets uniform and small when the flow rate is low. The relationship between the Peclet number and the Nusselt number becomes linear if it is steady. And, the Nusselt number for the direct contact heat exchanger with meshes becomes greater than that without meshes as the Peclet number increases.

Key Words : Direct Contact Heat Exchanger, Solar Thermal Energy System, Mesh

Nomenclature

- a : Interfacial area per unit volume, cm^{-1}
- A : Area through which heat transfer occurs, cm^2
- c_d : Specific heat of the working fluid, J/g · ℃
- d : Droplet diameter, cm
- F : Flow rate of the working fluid, cm^3/sec
- h_c : Convection heat transfer coefficient outside droplets, J/sec · cm^2 · ℃
- h_d : Convection heat transfer coefficient inside droplets, J/sec · cm^2 · ℃
- H : Fraction of the effective volume of the heat exchanger occupied by the working fluid (holdup), dimensionless
- k_d : Thermal conductivity of the working fluid, J/sec · cm^2 · ℃
- S : Horizontal cross sectional area of the heat exchanger, cm^2
- U : Overall area heat transfer coefficient, J/sec · cm^2 · ℃
- U_v : Volumetric heat transfer coefficient, J/sec · cm^3 · ℃
- V : Effective volume of the heat exchanger, cm^3
- V_s : Slip velocity between the working fluid and the continuous phase fluid, cm/sec
- μ_c : Viscosity of the continuous phase fluid, g/cm sec
- ρ_d : Mass density of the working fluid, g/cm^3

1. Introduction

The use of water as the working fluid for a solar thermal energy system causes several problems. In the winter, the water can be frozen so that it may break the system, and it would be very inconvenient to drain the system every night.

* Corresponding Author,
E-mail : cbkim@inha.ac.kr
TEL : +82-32-860-7313 ; **FAX** : +82-32-868-1716
Mechanical Engineering, Inha University, Inchon 402-751, Korea. (Manuscript **Received** July 24, 2000; **Revised** November 27, 2000)

Also, it is difficult to prevent corrosion if water is used as a working fluid. In order to avoid these problems, ethylene glycol, which is the well-known antifreeze, used to be utilized as a working fluid for a solar thermal energy system. Because ethylene glycol is miscible with water, an indirect contact heat exchanger should be installed inside a storage tank. However, the heat transfer rate of an indirect contact heat exchanger is relatively low compared to that of a direct contact heat exchanger, so that the heat transfer area should be large to obtain the desired heat transfer rate. This means that bigger heat exchangers and storage tanks are required. In addition, it is easy to get dirt so that cleaning is regularly required. Finally, the toxicity of ethylene glycol creates problems if it leaks.

On the other hand, the direct contact heat exchanger is widely used as the thermal storage tank for a solar thermal energy system to avoid the disadvantages of an indirect contact heat exchanger. The most important characteristic of the direct contact heat exchanger is that it can operate with a very small temperature difference unlike an indirect contact heat exchanger. Oil or hydrocarbons whose densities are lower than that of water are generally used as the working fluid in most direct contact heat exchangers. In this case, the lighter working fluid is injected into the heat exchanger from a perforated plate located at the bottom of the heat exchanger. One of the major difficulties with this arrangement is the control of the interface at the top of the heat exchanger. It is preferred that the interface remains fixed when the water is introduced into the heat exchanger below the interface. The other is the control of the rate of coalescence of the droplets, which affects the location of the interface at the top of the column. The rate of coalescence can be catalyzed by installing a honeycomb structure at the desired interface location and its material is preferentially wetted by the working fluid (Ward et al., 1977). Jacobs and Golafshani (1985) investigated a model using the assumption of no drop internal resistance to heat transfer and another where the heat transfer was governed by diffusion within the drop. Stamp et al. (1986) developed a heat transfer model for a liquid-liquid spray column employing a one-dimensional dispersion model. Kim et al. (1995) showed that the volumetric heat transfer coefficients of dimethyl phthalate and diethyl phthalate were similar in a spray column.

In order to solve the technical difficulties associated with the previous arrangement, it is proposed to use immiscible liquids that are heavier than water, with low freezing and high boiling temperatures. With this arrangement, it becomes possible to eliminate the internal parts of the heat exchanger to obtain a technically and economically effective design, and the problems caused by light working fluids can be eliminated. Therefore, the proposed system can bring a significant improvement of the performance with low temperature energy sources.

Ward et al. (1977) carried out an experiment to investigate the technical feasibility and economic practicality of direct contact heat exchangers. It was shown that diethyl phthalate and butylbenzyl phthalate would be promising working fluids for direct contact heat exchangers in the future. Moresco and Marschall (1980) showed that the droplets produced by multiple nozzles were not perfectly uniform, and individual droplet diameter and their distribution were dependent on the method of droplet formation. Steiner and Hartland (1983) showed that the size of the droplet became large and uniform if the velocity at the nozzle was low. On the other hand, all the droplets have jetting break-up formation and became small and uniform if the velocity of the working fluid is high at the nozzle. In addition, Jacobs and Eden (1985) found that small and uniform droplets improved the performance of a direct contact heat exchanger.

In order to make the size of droplets small and uniform as well as to keep the droplets inside the continuous phase fluid longer, installation of meshes in a direct contact heat exchanger is suggested in the present study. The performance of the suggested heat exchanger is experimentally investigated to verify the technical feasibility of the system. Diethyl phthalate is used as a working fluid whose properties are 1.059 g/cm^3 of density at 85℃, −45℃ freezing point, and 298℃ boil-

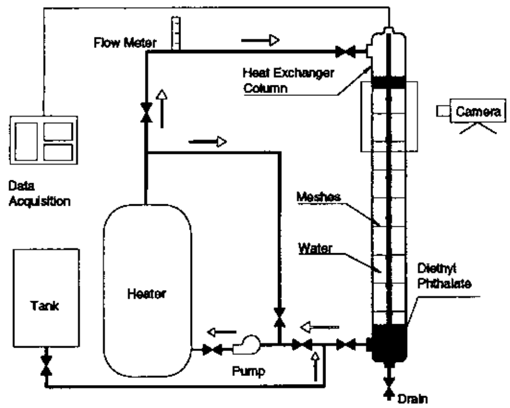

Fig. 1 Schematic of the experimental apparatus

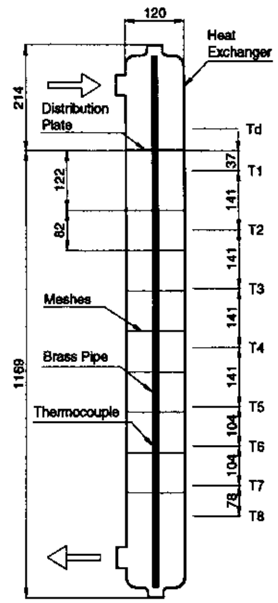

Fig. 2 Sketch of the test section

ing point. In addition, the experimental results of the heat exchanger with meshes are compared to those for an equivalent direct contact heat exchanger without meshes, in order to investigate the effect of meshes on the performance.

2. Experiment

Figure 1 is the schematic of the experimental apparatus. The working fluid and the continuous phase fluid were diethyl phthalate and water at the room temperature (22℃ ~24℃), respectively. The heat exchanger column consisted of a Pyrex glass pipe, 120 mm in diameter, 1, 383 mm in total length and 3 mm in thickness, in order to enable the inside of the heat exchanger column to be observed. A brass pipe, in which eight thermocouples were placed, was put into the heat exchanger column to measure the temperature of the continuous phase fluid, as shown in Fig. 2. The brass pipe had holes at the thermocouple locations so that the water could flow into the pipe through the holes and the thermocouples could have clear contact with the water. Stainless steel meshes (80 mesh) were installed at the holes so that drops of the working fluid could not get into the pipe. The distributing plate was made of 3 mm-thick copper plate with 44 holes. The diameter of each hole was 1 mm, and the distance between the holes was 15 mm, as shown in Fig. 3(a). Eight layers of the meshes were installed for the experiment. If the layers of the meshes were close, the size of the droplet became larger because the droplets coalesced. Therefore, the number of layers of mesh was selected to effectively make the droplets small. The stainless steel meshes shown in Fig. 3(b) were used. The distance between the layers of mesh was 82 mm and the top layer was installed 122 mm below the distributing plate.

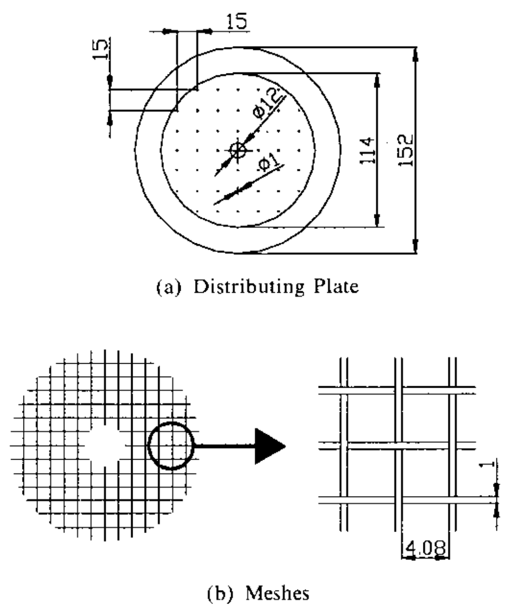

Fig. 3 Distributing plate and meshes installed in the heat exchanger for the experiment

The working fluid was circulated by a pump and heated up to the desired temperature (85℃) in the heater, and it went to the top of the heat exchanger column after passing through the flow meter. Diethyl phthalate was dispersed into the water by the distributing plate and discharged from the bottom of the heat exchanger column. The heat exchanger column was filled with water and the volume of the water in the heat exchanger did not change during the experiment. The experiments were performed with and without meshes for the flow rates of 10, 20, and 30 cm^3/sec. 20 mm and 40 mm air spaces between the free surface of the water and the distributing plate were considered to investigate the effect of the air space on the performance of the heat exchanger. The temperatures of the inlet and the outlet of the dispersed working fluid and water in the heat exchanger were measured by the thermocouples inserted into the brass pipe. The behavior and the size of the droplets were studied by investigating the pictures taken by the camera. In order to reduce the refracting effect of the curved wall of the heat exchanger column, a transparent acrylic box that was filled with water was installed outside the heat exchanger column. Once the temperature of the water stopped increasing, the experiment was terminated and the holdup was measured.

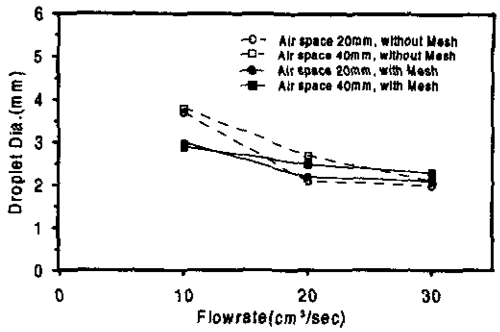

Fig. 4 The comparison of the average droplet diameter for different air space with and without installed meshes

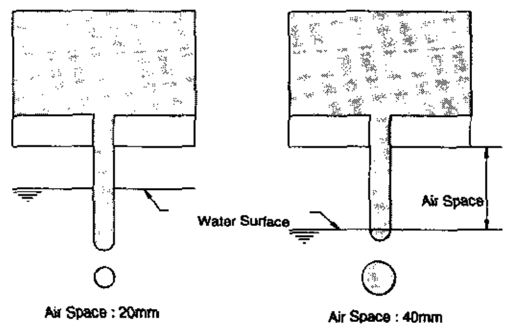

Fig. 5 Sketch of droplet formation

3. Results and Discussion

3.1 Droplet size of the working fluid

Figure 4 shows the average droplet size for different operating setups. When the meshes are not installed, the droplet size and its distribution for the two different air spaces of 20 mm and 40 mm are dependent on the flow rate of the working fluid. The average droplet sizes for two different air spaces are almost the same if the flow rate is 10 cm^3/sec. However, it is found that the sizes are not uniform and are distributed from 2 mm to 7 mm because the shape of the working fluid is dropwise when leaving the distributing plate. If the flow rate is 20 cm^3/sec, the average droplet size for the air space of 20 mm is 2 mm, which is smaller than that for the air space of 40 mm. This is because the jet of the working fluid leaving the distributing plate keeps the jet-cylinder shape for a while after contacting the water surface, as shown in Fig. 5. On the other hand, when the air space is 40 mm, the jet of the working fluid turns into droplets once it touches the water surface. The droplet sizes are not uniform and are distributed from 1 mm to 5 mm. When the flow rate is 30 cm^3/sec, the average droplet sizes for the two air spaces are 2 mm and almost uniform. In addition, the effect of the air space on the average droplet size is not significant. It is found that if the working fluid keeps the jet-cylinder shape after entering the water, the average droplet size becomes small and uniform, and this trend becomes strong as the flow rate increases and the air space decreases.

If the flow rate is 10 cm^3/sec, the average droplet diameters for the two air spaces of 20 mm and 40 mm decrease by 19% from 3.7 mm to 3.0 mm after installing meshes in the heat exchanger column. If the flow rate is 20 cm^3/sec and the air

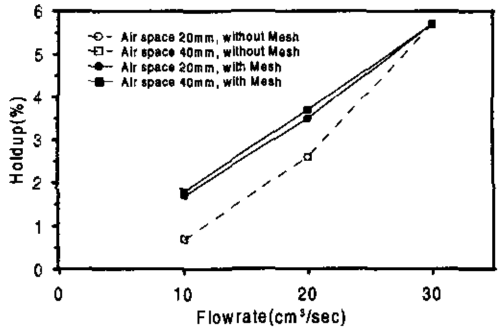

Fig. 6 The comparison of the holdup diameter for different air space with and without installed meshes

space is 40 mm, it is decreased from 2.7 mm to 2. 5 mm by installing meshes. However, the effect of meshes on the size of the droplets is not significant if the flow rate is greater than 20 cm³/sec. The difference becomes even smaller if the air space is 20 mm. This means that the droplets become small and uniform if the flow rate is high and the air space is narrow.

3.2 Holdup

The holdups, which are the ratio of the volume of the working fluid to the volume of the continuous phase fluid, for three different flow rates are shown in Fig. 6. It can be easily found that the air space does not make any difference to the results for the holdup. On the other hand, when the air space is 20 mm and the flow rate is 10 cm³/sec, the holdup increases by a factor of 2.4 from 0.7% to 1.7% after installing meshes. For 20 cm³/sec of the flow rate, the holdup increases by a factor of 1.34 after installing meshes. And, when the flow rate is 30 cm³/sec, the difference is hardly seen. Therefore, it can be summarized that the holdup can be increased by meshes if the flow rate is low. On the other hand, the holdup cannot be changed by meshes if the flow rate is high. And, we can guess that meshes can lower the descending velocity of the working fluid within the continuous phase fluid if its flow rate is low.

3.3 Distribution of temperature

In Figs. 7 and 8, the temperature variations with time at seven different axial locations in the heat exchanger column are shown. The temperature variations for the heat exchanger without meshes and with meshes are presented side by side to show the effect of meshes on the temperature distribution of the continuous phase fluid. Figure

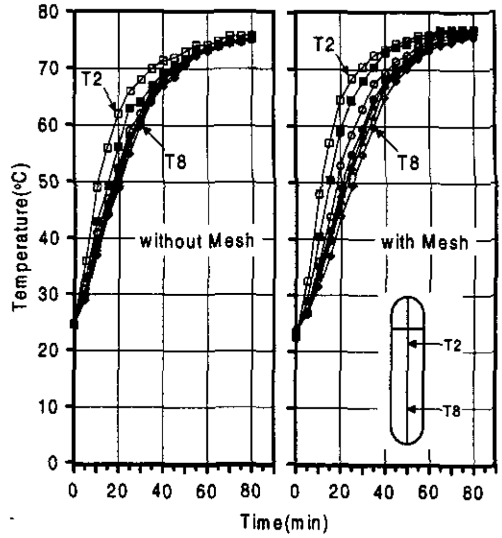

Fig. 7 Temperature variations in the heat exchanger(□ : T2, ■ : T3, ○ : T4, ● : T5, △ : T6, ▲ : T7, ◇ : T8; flowrate: 10cm³/s, air-space: 20 mm)

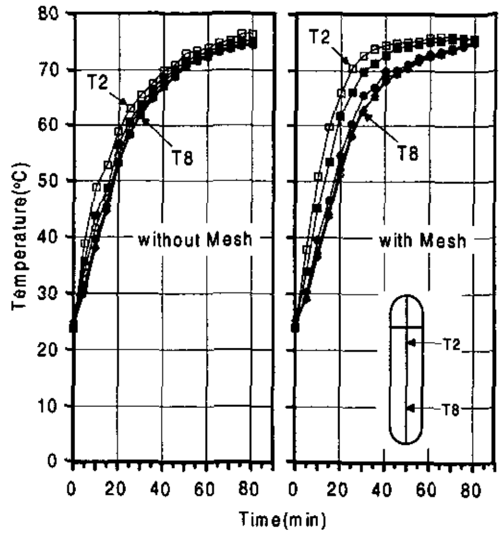

Fig. 8 Temperature variations in the heat exchanger(□ : T2, ■ : T3, ○ : T4, ● : T5, △ : T6, ▲ : T7, ◇ : T8; flowrate: 10cm³/s, air-space: 40 mm)

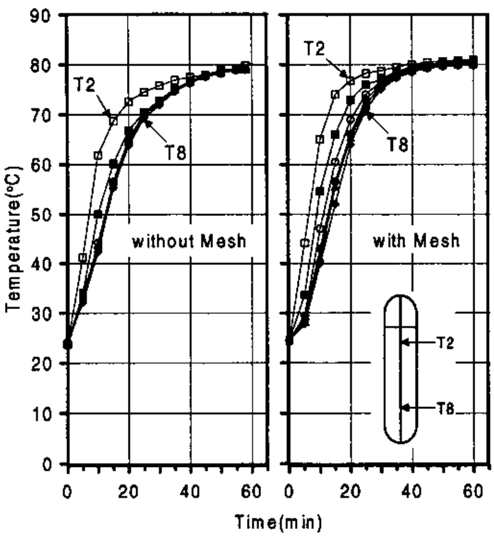

Fig. 9 Temperature variations in the heat exchanger(□ : T2, ■ : T3, ○ ; T4, ● : T5, △ : T6, ▲ : T7, ◇ : T8; flowrate: 20cm^3/s, airspace: 20 mm)

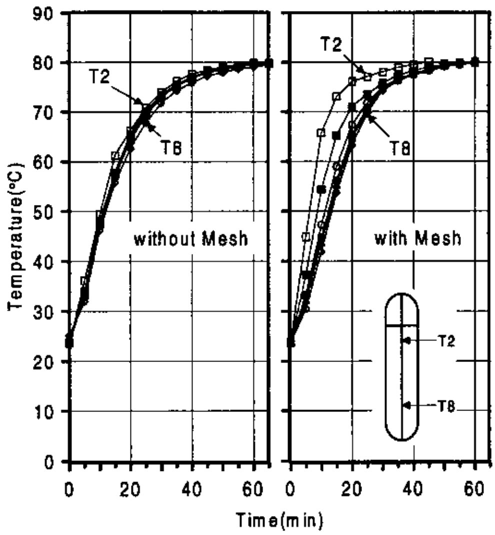

Fig. 10 Temperature variations in the heat exchanger(□ : T2, ■ : T3, ○ ; T4, ● : T5, △ : T6, ▲ : T7, ◇ : T8; flowrate: 20cm^3/s, airspace: 40 mm)

7 shows the results for a flow rate of 10 cm^3/sec and an air space of 20 mm. The temperature variation for the same flow rate and an air space of 40 mm is presented in Fig. 9. Comparing the figures, the effect of meshes on the temperature distribution can be easily discerned. In addition, temperature stratification can be seen in all the figures, but it is enhanced when meshes are installed and the air space is 40 mm. For example, the temperature differences between T2 and T8 for the heat exchanger column with meshes are greater than those of the heat exchanger column without meshes. If meshes are not installed, temperatures measured at the different vertical locations are relatively close to each other compared to those of the heat exchanger with meshes. To raise the temperature T2 up to 70C when the air space is 20 mm, it takes 28 minutes and 35 minutes with and without meshes, respectively. This means that the heat transfer rate of the upper part of the heat exchanger with meshes is greater than that without meshes. When the air space is 40 mm, it takes 25 and 45 minutes with and without meshes, respectively. Therefore, it is known that the effect of meshes on heat transfer augmentation becomes greater if the air space is 40 mm. It takes about 50 minutes to get to 70℃ at the location of T8 for all the operating setups. In addition, it takes about 80～90 minutes to complete one set of experiments.

The temperature variations with time at seven different axial locations for a 20 cm^3/sec flow rate are shown in Figs. 9 and 10. The air space is 20 mm for Fig. 9 and 40 mm for Fig. 10. The overall trend is very similar to those shown in Fig. 7 and Fig. 8, but temperature stratification is weak if meshes are not installed. The temperature differences between T2 and T8 with meshes are much greater than those without meshes when the air space is 40 mm. If the air space is 20 mm, it takes about 13 minutes with meshes and 17 minutes without meshes to get to 70℃ at the location of T2. If the air space is 40 mm, it takes 13 minutes and 25 minutes with and without meshes, respectively, for T2 to rise to 70℃.

In Figs. 11 and 12, temperatures at the vertical locations in the heat exchanger column are presented with time when the flow rate is 30 cm^3/sec. These figures show that the temperature distributions are clearly stratified by installing meshes, and this effect for the air space of 40 mm is greater than that for the air space of 20 mm, as

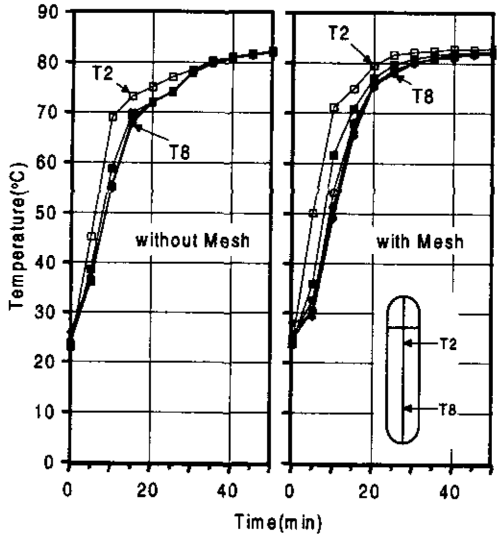

Fig. 11 Temperature variations in the heat exchanger(□ : T2, ■ : T3, ○ ; T4, ● : T5, △ : T6, ▲ : T7, ◇ : T8; flowrate: 30cm³/s, airspace: 20 mm)

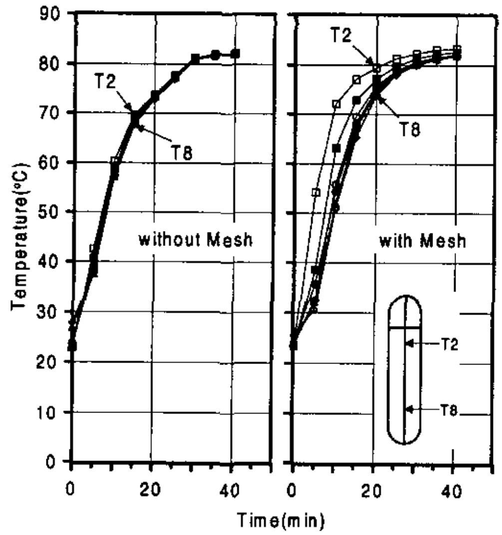

Fig. 12 Temperature variations in the heat exchanger(□ : T2, ■ : T3, ○ ; T4, ● : T5, △ : T6, ▲ : T7, ◇ : T8; flowrate: 30cm³/s, airspace: 40 mm)

in the previous results. When the air space is 20 mm, it takes about 9 minutes for T2 to rise up to 70C with meshes while it takes about 12 minutes without meshes. When the air space is 40 mm, it takes about 9 minutes with meshes and about 15 minutes without meshes, respectively. The experiment for this setup was terminated after 40~50 minutes.

3.4 Heat transfer result

The overall heat transfer coefficient U for a direct contact heat exchanger can be described as follows, because there is no heat transfer medium between the fluids:

$$\frac{1}{U}=\frac{1}{h_d}+\frac{1}{h_c} \tag{1}$$

where h_d is the convection heat transfer coefficient in the droplets of the working fluid and hc is the convection heat transfer coefficient of the continuous phase fluid. Because the droplets are descending in the continuous phase fluid, forced convection heat transfer occurs outside the droplets. On the other hand, the major heat transfer mode in the droplets is natural convection. Therefore, h_c is much greater than h_d, so the overall heat transfer coefficient U is approximately equal to the convective heat transfer coefficient of the working fluid h_d.

The temperature difference between the continuous phase fluid and the working fluid leaving the heat exchanger is almost zero so it is assumed that the log mean temperature difference($\overline{\Delta T}$) is half the difference ΔT between the inlet and outlet temperatures of the working fluid (Ward et al., 1977). The heat transfer rate Q is, therefore, given as follows:

$$Q=c_d\rho_d F\Delta T=UA\overline{\Delta T} \tag{2}$$

From Eq. (2), the overall heat transfer coefficient can be obtained, and the volumetric heat transfer coefficient U_v, which is defined as the heat transfer rate per unit volume and per unit temperature difference, is calculated from

$$U_v=U_a=U\frac{A}{V}=\frac{2Q}{V\Delta T}=\frac{2c_d\rho_d F}{V} \tag{3}$$

As shown in Eq. (3), the volumetric heat transfer coefficient is dependent on the flow rate of the working fluid, once the geometry of the heat exchanger and the properties of the dispersed working fluid are fixed. The important dimensionless parameters are defined as follows:

Table 1 Heat transfer results

	Airspace	F	ΔT	d	H	Vs	U	Uv	Nu	Re	Pe
Without mesh	20	10	7.2	0.37	0.007	14.85	0.0412	0.00467	11.76	242.68	6710
		20	5.61	0.21	0.026	7.51	0.0118	0.00877	1.92	71.94	1989
		30	2.14	0.2	0.057	5.17	0.0077	0.01317	1.20	45.65	1262
	40	10	7.84	0.38	0.007	14.85	0.0423	0.00468	12.40	249.24	6891
		20	5.74	0.27	0.026	7.46	0.0151	0.00872	3.16	91.79	2538
		30	2.74	0.21	0.057	5.17	0.0081	0.01319	1.32	47.93	1325
With mesh	20	10	8.56	0.3	0.017	5.73	0.0128	0.00435	2.97	73.62	2098
		20	3.49	0.22	0.035	5.60	0.0092	0.00878	1.57	54.42	1504
		30	1.88	0.21	0.057	5.16	0.0081	0.01319	1.32	47.84	1322
	40	10	8.98	0.29	0.018	5.33	0.0116	0.00732	2.59	68.23	1886
		20	5	0.25	0.037	5.30	0.0099	0.00879	1.92	58.50	1617
		30	1.56	0.23	0.057	5.16	0.0089	0.01323	1.58	52.40	1448

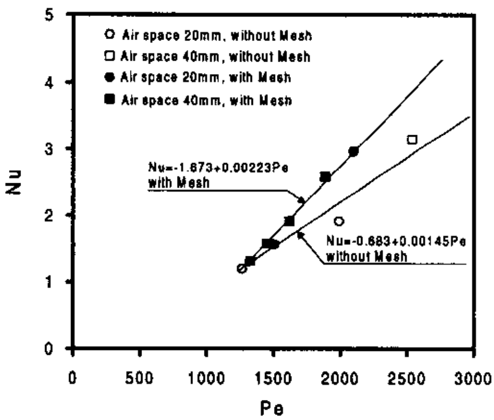

Fig. 13 The comparison of the Nusselt numbers with and without installing meshes

$$Pe=\frac{c_d\rho_d V_s d}{k_d} \tag{4}$$

$$Nu=\frac{U_d}{k_d} \tag{5}$$

$$Re=\frac{\rho_d V_s d}{\mu_c} \tag{6}$$

where V_s is the relative velocity between the two fluids and is defined as:

$$V_s=\frac{F/S}{H} \tag{7}$$

The summary of the experimental results is shown in Table 1, and is graphically displayed in Fig. 13. From the figure, it can be found that the Nusselt number for the system with meshes is greater than that for the system without meshes and the difference between these two Nusselt numbers grows as the Peclet number increases. This means that meshes make the heat transfer rate increase and the effect of the meshes on the heat transfer becomes more important as the Peclet number increases. The Peclet number becomes large when the flow rate of the working fluid is low. As mentioned before, the sizes of the droplets become large and nonuniform in the heat exchanger without meshes if the flow rate is low. Even though large and nonuniform droplets are not good for heat transfer, meshes make the droplets small and uniform so that both the heat transfer area and the heat transfer rate increase.

4. Conclusions

In the present study, the performance of a direct contact heat exchanger for a solar thermal energy system is experimentally investigated. Installation of meshes inside the heat exchanger is suggested to improve the performance of the heat exchanger, and diethyl phthalate is used as the working fluid. From the results, several conclusions ware drawn as follows:

(1) When the flow rate is low, the working fluid leaving the distributing plate becomes dropwise. On the other hand, the shape of the working fluid turns jetwise as the flow rate

increases. It is easy for the jetwise droplets to penetrate the interface between the air and water if the air space is narrow, so we can easily get small and uniform droplets in the water.

(2) It is found that large and non-uniform droplets are broken into small and uniform droplets by meshes if the flow rate is low. The effect of meshes on the formation of droplets is not important for jetwise droplets because those become uniform and small even when meshes are absent.

(3) If meshes are installed and the flow rate is low, the temperature of the upper part of the heat exchanger becomes high. This means that meshes enhance the temperature stratification in the heat exchanger.

(4) The relationship between the Peclet number and the Nusselt number is linear if it is steady. The Nusselt numbers with meshes are greater than those without meshes.

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