Study of heat transfer characteristics of wire-and-tube heat exchangers

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Free convection heat transfer from a number of wire-and-tube heat exchangers was experimentally investigated. The experiments were carried out at different inclination angles that ranged from 0.0 (horizontal test) to 90 (vertical test). The spacing-to-diameter ratio of wire (s_w) was varied from 2.0 to 5.7, while the spacing-to-diameter ratio of tubes (s_t) was varied from 5.0 to 12.1. The height of the exchangers were in the range 0.83 m to 1.105 m. The effect of a vertical flat plate near to a vertical heat exchanger on a heat transfer rate was also studied. The distance between the flat plate and the exchangers (x) was varied from 1.67d to 6.67 d (where d is the tube diameter). It was found that the heat transfer rate decreases upon increasing the inclination angle regardless of the spacing-to-diameter ratios. The heat transfer rate decreases as the spacing-to-diameter ratio of the wire s_w decreases. Also results show that the Nusselt number increases with the distance x between the vertical flat plate and the vertical exchanger. A general relationship that relates the heat transfer rate to various parameters was developed.

تناقش هذه الدراسة انتقال الحرارة بالحمل الطبيعي لعدد من المبادلات الحرارية ذو الأنابيب والأسلاك. وقد أجريت التجارب على عدد من المبادلات الحرارية بزوايا ميل مختلفة تتراوح بين زاوية صفر عند الاختبار الأفقي وزاوية ٩٠ عند الاختبار الرأسي. وكانت النسبة بين المسافة آلي القطر بين الأسلاك تتراوح من ٢ آلي ٥,٧ بينما كانت النسبة بين المسافة آلي القطر بين الأنابيب تتراوح من ٥ آلي ١٢,١ . وكذلك استخدمت مبادلات حرارته تتراوح ارتفاعها بين ٠,٨٣ – ١,١٥م. وقد ناقشت هذه الدراسة تأثير وجود مستوى رأسي قريب من مبادل حراري رأسي بحيث كانت المسافة بين المستوى والمبادل تتراوح بين ١,٦٧ق-٧,٦٧ ق (حيث ق هي قطر الأنبوب). وقد وجد أن معدل انتقال الحرارة يقل مع زيادة زاوية الميل بغض النظر عن نسبة المسافة إلى القطر. وكذلك وجد ان معدل انتقال الحرارة يقل مع نقصان نسبه المسافة آلي قطر الأسلاك. وقد بينت أن معامل ناسلت يزيد مع زيادة المسافة بين المسوي والمبادل الراسي. وقد تم استنتاج علاقة تربط بين معدل انتقال الحرارة والعوامل المختلفة

Keywords: Free convection, Heat transfer, Wire-and-tube heat exchanger

1. Introduction

Extended surface heat exchangers consisting of wires attached to tubes are widely used, especially in small, air-cooled refrigeration appliances (e.g., domestic refrigerators and supermarket food display cabinets). In these fields, the device is used to condense the refrigerant flowing inside the tubes.

Witzell and Fontaine [1, 2] were among the first to investigate heat transfer from wire-and-tube heat exchangers. Further contributions were made by Witzell et al. [3], Cyphers et al. [4], Collicott et al. [5], and Cavallini and Trapances [6].

Despite the widespread use of these exchangers, only few early works dealing with heat transfer characteristics are available in the literature. Tanda and Tagliafico [7] studied free convection from vertical wire-and-tube heat exchangers. A Nusselt number correlation to predict free convection heat transfer from a vertical wire-and-tube heat exchanger to ambient air was presented. To the authors knowledge, a heat transfer relationship including the effects of the inclination angle, the numerous geometrical and operating parameters affecting the performance of such heat exchangers has not yet been developed.

The purpose of this study is to investigate the effects of the inclination angle, operating parameters, and different geometry of wire-and-tube heat exchangers on heat transfer characteristics. Also, effect of presence of a vertical flat plate in the vicinity of the exchanger on heat transfer was studied. Finally, a correlation giving the natural-convection heat transfer rate from the external surface of the heat exchanger to the ambient air was obtained.

2. Analysis

The wire-and-tube heat exchangers considered in this study consist of a tube, wound as a serpentine, with wires welded to both sides in a direction normal to the tubes, as shown in fig. 1. The wires act as extended surface that increases the heat transfer from the tube to the external environment. The geometry is defined by the overall height and width (H) and (B) of the exchanger respectively, the external diameters of the tube and wire (d_t) and (d_w) respectively, the tube pitch (p_t), and the wire pitch (p_w). Further geometric parameters could be introduced, such as the angle of inclination (α) of the exchanger with horizontal and the distance between the vertical flat plate (metal sheet) and the exchanger (x). In this study, the heat exchanger has different inclination angles, 0.0° (horizontal), 30°, 45°, 60°, and 90° (vertical) without confining walls (free-standing configuration).

When a hot fluid circulates inside the tube, heat transfer between the fluid and external environment takes place. The heat is transferred from the tube wall to the external environment by natural convection and radiation.

The total heat transfer rate (q_{tot}) from the external walls of the exchanger to the surroundings is given by

$$q_{tot} = q_c + q_r, \tag{1}$$

where subscripts "c" and "r" denote convective and radiative components, respectively. The convective heat transfer rate may be expressed as;

$$q_c = h\, S_{tot}\, (T_{ex} - T_\infty), \tag{2}$$

where h is the average heat transfer coefficients, S_{tot} is the total heat transfer surface area, T_∞ is the ambient air temperature, and T_{ex} represents the weighted average of the temperature on the external side of the exchanger, which can be evaluated as follows:

$$T_{ex} = (S_t T_t + S_w T_w) / S_{tot}, \tag{3}$$

where T_t, S_t and T_w, S_w are the mean external surface temperature and the heat transfer surface area of the tube and wires respectively. If the fin efficiency η_w is introduced under the assumption of uniform heat transfer coefficient along the wire surface, one obtains

$$\eta_w = (T_w - T_\infty)/(T_t - T_\infty). \tag{4}$$

Eqs. (2-4) lead to

$$q_c = h\,(T_t - T_\infty)(S_t + \eta_w\, S_w). \tag{5}$$

3. Experimental apparatus and procedure

The experimental arrangement, shown schematically in fig. 2, consisted of a water tank, water heater, stirrer motor, water pump, flow meter, and heat exchanger. A 5 kW water heater is used to heat the water inside the water tank. A thermostat immersed in the heater allowed the water temperature in the tank to be varied from 40 °C to 80 °C. The water tank was insulated by glass wool to reduce heat loss to environment. A stirrer motor fixed in the cover of the tank was used to get homogenous temperature of the circulating water. A regulating valve in the line allowed the volume flow rate to the exchangers to be varied from approximately 0.2 lit/min to 2 lit/min.

In this study different heat exchangers (serpentines) were used. The spacing-to-diameter ratio of wire (s_w) was varied from 2.0 to 5.7, while the spacing-to-diameter ratio of tubes (s_t) was varied from 5.0 to 12.1. The heights of the exchangers were in the range 0.83 m to 1.105 m.

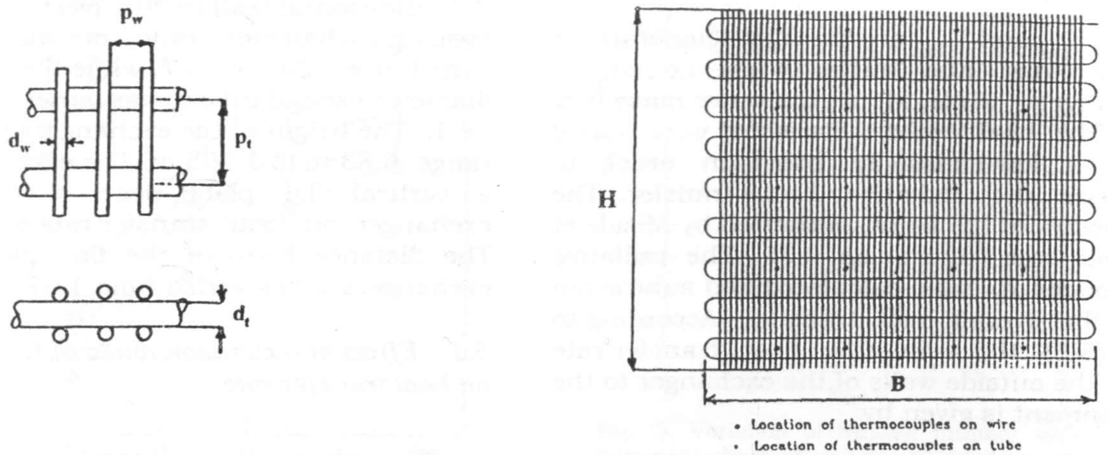

Fig. 1. Sketch of the wire-and-tube heat exchanger and distribution of thermocouples on wires and tubes.

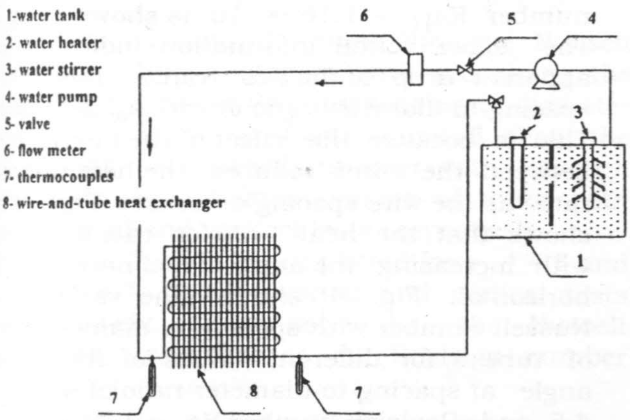

Fig. 2. Schematic drawing of the experimental apparatus.

The effect of a vertical flat plate (metal sheet) placed in the vicinity of the heat exchanger on heat transfer rate was studied. The distance (x) between the plate and serpentine was varied from 1,67d to 6.67d (where d is the tube diameter).

The tubes and wires of the exchanger were instrumented with copper-constantan thermocouples at locations shown in fig. 1. One thermocouple was attached to the outer surface of each tube at the midplane. Twenty thermocouples were attached to the wires at five locations along the height of the exchanger (four thermocouples in each location). Inlet and outlet water temperatures from the exchanger were measured by means of two thermocouples. All thermocouples were calibrated using a constant-temperature water bath. The temperature of the ambient air T_∞ was evaluated as the average of the values measured by five thermometers located at different elevations. The volume flow rate was measured by means of a pre-calibrated flow meter (within ±1 percent).

For each set of experiments, the exchanger was adjusted at the desired inclination angle. The water temperature was raised and varied and a hot water flow coming from the water tank circulated steadily inside the heat exchanger tube. The volume flow rate was controlled by means of a regulating valve in the line. A digital thermometer was employed for all the temperature measurements. The temperatures were recorded after steady-state conditions had been reached. The steady-state temperatures of tubes and wires was taken as the average of the readings of the thermocouples installed on the surface of the tubes and the wires,

4. Data reduction

All individual temperature measurements were first corrected using their corresponding calibration curves. The temperature readings were averaged to given average surface temperature for the wire and tube. From an overall energy balance for the heat exchanger, it follows that

$$q_{tot} = M\, c_{p,f}\, (T_{f,in} - T_{f,out}), \quad (6)$$

where $T_{f,in}$ and $T_{f,out}$ are the fluid temperatures at the inlet and outlet of the exchanger, respectively, while M is the water mass flow rate. The heat exchangers tested were coated with a low-emissivity paint in order to minimize the radiative heat transfer. The emissivity of the paint, measured by Misale et al. [8], was taken as 0.35. The radiative component q_r was calculated and subtracted from the overall heat transfer. According to Eqs. (5-6), the convective heat transfer rate from the outside walls of the exchanger to the environment is given by:

$$q_c = Mc_{p,f}\, (T_{f,in} - T_{f,out}) - q_r = h\, (T_t - T_\infty)(S_t + \eta_w\, S_w). \quad (7)$$

Since the thermal resistance associated with the forced convection of water inside the tube and with the conduction through the tube wall are typically two to three orders of magnitude lower than the air-side thermal resistance, the tube temperature can be assumed equal to the water temperature. With reference to the entire exchanger, the mean external surface temperature of the tube T_t can be estimated as the mean of $T_{f,in}$ and $T_{f,out}$. In conclusion, the average heat transfer coefficient of the exchanger can be expressed as:

$$h = [Mc_{p,f}\, (T_{f,in} - T_{f,out}) - q_r\,] \;/\; [((T_{f,in} + T_{f,out})/2) - T_\infty)(S_t + \eta_w\, S_w\,)]. \quad (8)$$

Eq. (8) has been used to evaluate the average heat transfer coefficient for each experimental run.

The experimental uncertainty in h values, evaluated according to the procedure outlined by Moffat [9], turned out to be ±8 percent. All air physical properties are to be evaluated at the reference temperature here defined as $(T_t + T_\infty)/2$.

5. Results and discussion

Free convection heat transfer from wire-and-tube heat exchangers was experimentally investigated. The experiments were carried out for different inclination angles ranged from 0.0° (horizontal test) to 90° (vertical test). The spacing-to-diameter ratio of wire s_w was varied from 2.0 to 5.7, while the spacing-to-diameter ratio of tube s_t was varied from 5.0 to 12.1. The height of the exchangers were in the range 0.83 m to 1.105 m. The effect of holding a vertical flat plate near a vertical heat exchanger on heat transfer rate was studied. The distance between the flat plate and the exchangers x was varied from 1.67d to 6.67d.

5.1. Effect of inclination angle of the exchanger on heat transfer rate

The variation of Nusselt number with spacing-to-diameter ratio of wire s_w for various values of inclination angle at spacing-to-diameter ratio of tube $s_t = 9.74$ and Rayleigh number $Ra_H = 1.16 \times 10^9$ is shown in fig.3. The experimental information indicates the apparent drop of the heat transfer rate as the spacing-to-diameter ratio of wire s_w decreases. This is because the effect of the interference between the wires reduces the heat transfer rate as the wire spacing decreases. Fig. 3 also shows that the heat transfer rate decreases with increasing the angle of inclination to the horizontal. Fig. 4 shows the variation of Nusselt number with spacing-to-diameter ratio of tube s_t for different values of inclination angle at spacing-to-diameter ratio of wire s_w = 4.5 and Rayleigh number $Ra_H = 1.74 \times 10^9$. It is seen that the Nusselt number Nu_H increases with increasing the spacing-to-diameter ratio of tube s_t. Also fig. 4 shows that the horizontal test (α=0.0°) has a heat transfer rate higher than the vertical test (α=90°) and the other testes (α=30°, 45°, and 60°).

5.2. Effect of operating parameters and exchangers geometry on heat transfer rate

The variation of Nusselt number with Rayleigh number for different values of s_w for both horizontal and vertical tests are shown in figs. 5 and 6, respectively. The results for horizontal test (α=0.0°) are presented for spacing-to-diameter ratio of tube s_t = 7.1, while the results for vertical test (α=90°) are presented for spacing-to-diameter ratio of tube s_t =9.74.

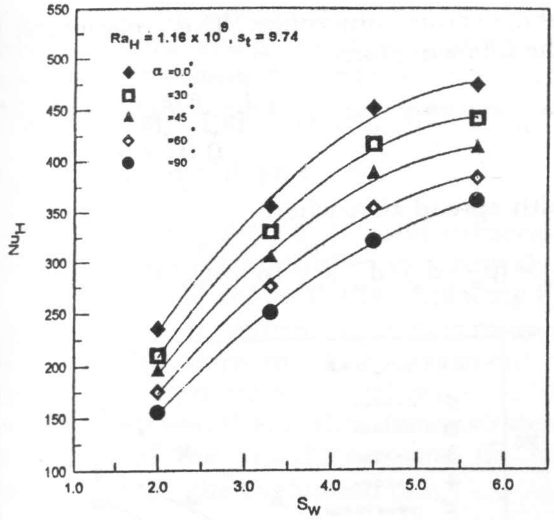

Fig.3. Variation of Nusselt number with spacing-to-diameter ratio of wire for various values of inclination angle (RaH=1.16 ×10, st=9.74).

Figs. 5 and 6 show that the Nusselt number Nu_H increases with increasing the Rayleigh number Ra_H and spacing-to-diameter ratio of wire s_w. Figs. 7 and 8 show the variation of Nusselt number with Rayleigh number for different values of s_t for both horizontal and vertical tests respectively.

Figs. 7 and 8 are plotted for s_w = 4.5 and s_w = 3.3 for horizontal and vertical tests respectively. It is seen that the Nusselt number increases as the Rayleigh number increases.

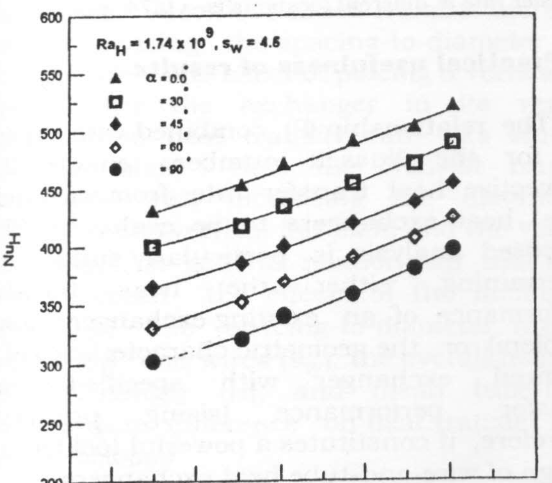

Fig. 4. Variation of Nusselt number with spacing-to-diameter ratio of tube for various values of inclination angle (Ra_H =1.47 ×10^9, s_w=4.5).

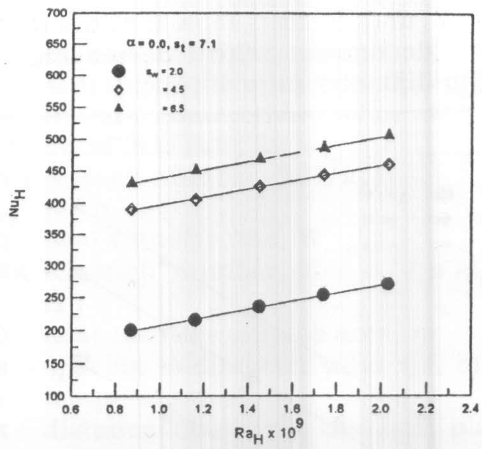

Fig. 5. Variation of Nusselt number with Rayleigh for different values of spacing-to-diameter ratio of wire for a horizontal test (s_t=7.1).

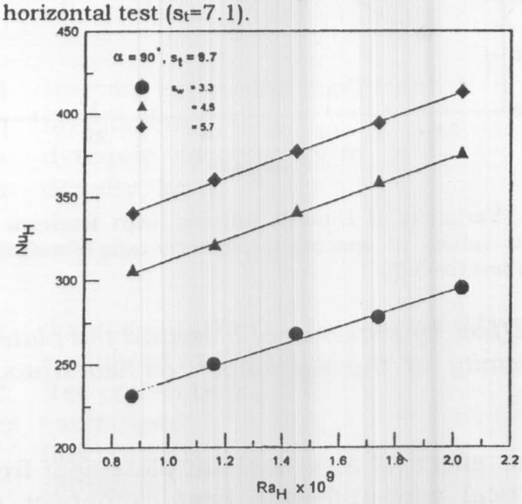

Fig. 6. Variation of Nusselt number with Rayleigh for different values of spacing-to-diameter ratio of wire for a horizontal test (s_t=9.74).

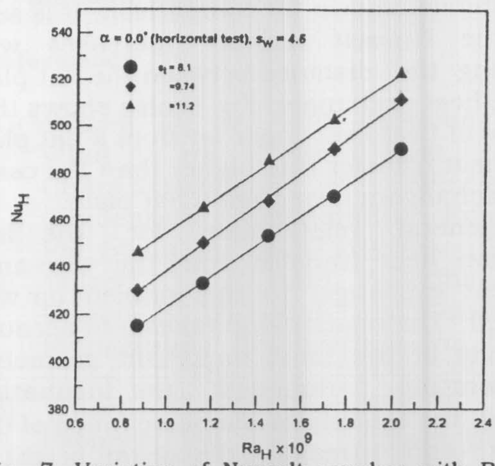

Fig. 7. Variation of Nusselt number with Rayleigh for different values of spacing-to-diameter ratio of wire for a horizontal test (s_t=4.5).

Figs. 7 and 8 also show that the Nusselt number decreases with decreasing the spacing-to-diameter ratio of tube s_t.

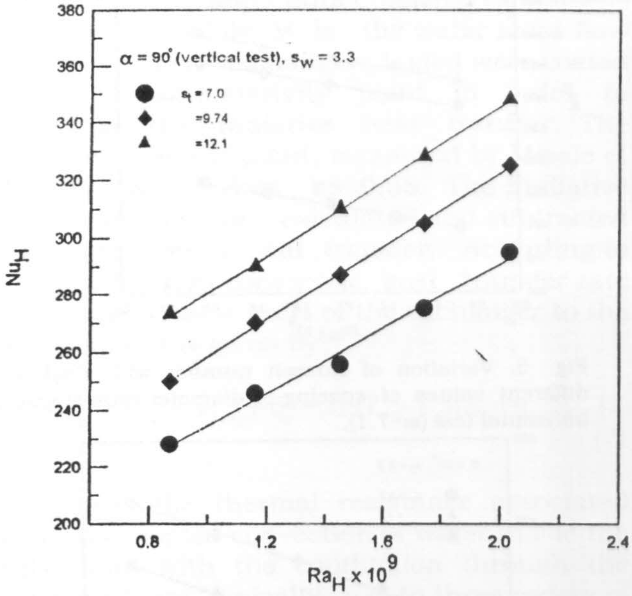

Fig. 8. Variation of Nusselt number with Rayleigh for different values of spacing-to-diameter ratio of wire for a vertical test (s_t=3.3).

5.3. *Effect of presence of a vertical flat plate in the vicinity of the exchanger on heat transfer rate*

The effect of a vertical flat plate near from a vertical wire-and-tube heat exchanger on heat transfer rate at different locations is shown in fig. 9. The case of wire-and-tube heat exchanger far from a flat plate (without flat plate) is shown for comparison. It is seen that the Nusselt number increases with increasing the distance between the flat plate and the heat exchanger. Fig. 9 also shows that the case of heat exchanger far from a flat plate has a heat transfer rate higher than the cases of the exchangers near from a flat plate.

A general relationship for the free convection heat transfer from the wire-and-tube heat exchanger to the ambient air was developed. The relationship takes into account the effects of the most important geometric and operating parameters: the inclination angle (α), the spacing-to-diameter ratios of the tubes (s_t) and wires (s_w), the overall height of the exchanger (H), and the mean tube-to-air temperature difference. The relationship has the following form:

$$Nu_H = 0.158 (Ra_H H/d_t)^{0.232} (s_w)^{0.78} (s_t)^{0.32} (\cos\alpha)^{0.4}$$
$$0.0^{o} \le \alpha < 90, \qquad (9)$$

with spread ±8% where

$s_w = (p_w - d_w)/d_w$, $s_t = (p_t - d_t)/d_t$,

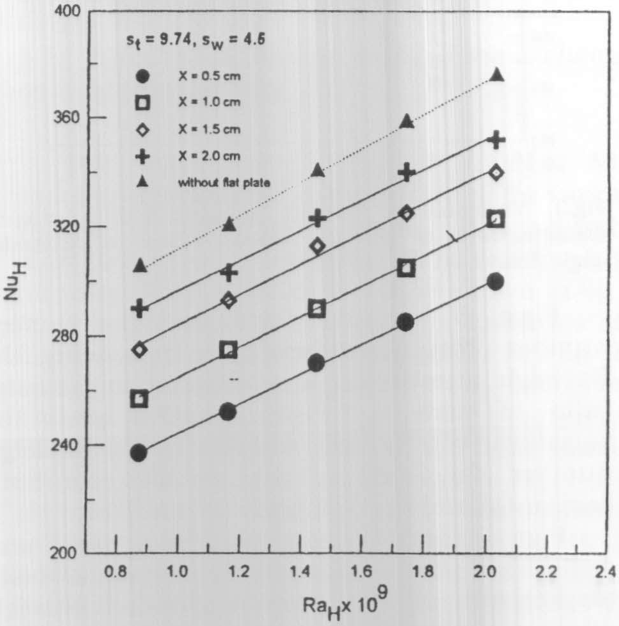

Fig. 9. Effect of a nearby vertical flat plate on heat transfer rate at different locations (s_t = 9.74, s_w = 4.5).

6. Practical usefulness of results

The relationship (9), combined with the eq. (5) for the Nusselt number, enables the convective heat transfer rate from wire-and-tube heat exchangers to be evaluated. The proposed analysis is particularly suitable for determining either the heat transfer performance of an existing exchanger (rating problem) or the geometric characteristics of a required exchanger with specified heat transfer performance (sizing problem). Therefore, it constitutes a powerful tool for the design of wire-and-tube heat exchangers.

It should be pointed out that the use of eq. (5) requires the knowledge of η_W, which depends on the temperature distribution along the wire and the tube. A simplified approach can be employed to evaluate η_w without the availability of detailed local temperature measurements along the wire and the tube surfaces. According to the conventional one-dimensional fin model, η_w can be expressed as:

$$\eta_w = [\tanh (m\, p_t/2)]/(m\, p_t/2)\ , \qquad (10)$$

where $m = [4h_w / (k_w d_w)]^{1/2}$ and subscripts w and t refer to wires and tubes, respectively.

Experiments showed that replacing h_w by the estimated h values, η_w values calculated from eq. (10) were in close agreement with those estimated from eq. (4) with within 4 percent between them. This demonstrates the suitability of the one-dimensional fin model assumption for the evaluation of η_w.

7. Conclusions

Free convection heat transfer from wire-and-tube heat exchanger to ambient air was experimentally investigated. The effects of inclination angle, the spacing-to-diameter ratios of wires and tubes, the overall height of the exchanger, and the mean tube-to-air temperature difference on heat transfer rates were studied. It was found that the heat transfer rate decreases with increasing the inclination angle to horizontal for all spacing-to-diameter ratios of the tube and the wires. The heat transfer rate decreases as the spacing-to-diameter ratio of the wire decreases. Also the Nusselt number increases with increasing the spacing-to-diameter ratio of the tube. The effect of placing a vertical flat plate near the exchanger in its vertical position on heat transfer rate was studied. Results show that the Nusselt number increases with increasing the distance x between the vertical flat plate and the vertical exchanger. A general relationship that takes into account the effects of the inclination angle (α), the spacing-to-diameter ratio of tubes (s_t) and wires (s_w), the overall height of the exchanger (H), and mean tube-to-air temperature difference on heat transfer rates was developed.

Nomenclature

c_p constant-pressure specific heat, J kg^{-1} K^{-1}
d outer diameter, m
g acceleration of gravity, m s^{-2}
H exchanger height, m
h average heat transfer coefficient, W m^{-2} K^{-1}
k thermal conductivity, W m K
M mass flow rate, kg s
Nu Nusselt number, (hH/k_a)
p pitch, m
q heat transfer rate, W
Ra Rayleigh number, $[(\beta\rho^2 c_p / \mu k)_a\, g\, (T_t - T_\infty) H^3]$
S heat transfer surface area, m^2
s spacing-to-diameter ratio, $(p - d)/d$
T temperature, K
x distance between the flat plate and the heat exchanger, m

Greek symbols

β thermal expansion coefficient, K^{-1}
η fin efficiency
μ dynamic viscosity, kg m^{-1} s^{-1}
ρ density, kg m^{-3}

Subscripts

a denoting physical properties of air, to be evaluated at $(T_t + T_\infty)/2$
c free convection
ex exchanger
f fluid (water)
r radiation
t tube
tot total (tubes + wires)
w wire
∞ ambient air

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