Journal of Energy & Environment 2 (2002) 83-91

Natural Convection Heat Transfer in Inclined Isothermal Parallel Plate Enclosure

C. M. Feroz, A. M. M. S. Alam & M. A. R. Akhanda
Mechanical Engineering Department
Bangladesh University of Engineering & Technology (BUET) Dhaka-1000, Bangladesh
Email: feroz@me.buet.ac.bd

Abstract
Experimental study of natural convection heat transfer across inclined air layers placed inside a rectangular enclosure with adiabatic side walls, a semicircular corrugated bottom wall heated from below and an upper cold flat wall has been carried out. Effects of inclination angle, aspect ratio, temperature potential and Rayliegh number on average heat transfer coefficients are investigated within the range of $45° \le \theta \le 75°$, $3.5 \le A \le 9.5$, $10°C \le \Delta T \le 35°C$, and $3.36x10^4 \le Ra_l \le 2.06x10^6$. The developed correlation agrees with the experimental data with a scatter of ±25%. The results are compared with the available information on natural convection and found to be about 25% and 15% higher than those for sinusoidal and vee corrugations, respectively; at low Rayleigh numbers $3.36x10^4 \le Ra_l \le 1.0x10^6$.

Nomenclature

A	aspect ratio, L/H
A_{cor}	projected area of the test section of the experimental corrugated plate, m^2
Cp	specific heat at constant pressure, kJ/kg K
g	acceleration due to gravity, m/s^2
H	amplitude of corrugation, m
h	average natural convective heat transfer coefficient, W/m^2 K
I	measured current passing through main heater, A
k	thermal conductivity, W/mK
L	mean plate spacing, m
Nu_l	average Nusselt number, hL/k
P	pitch of corrugation, m
Pr	Prandlt number
q	heat flux, W/m^2
q_r	radiative heat flux, W/m^2
Ra_l	Rayleigh number based on mean plate spacing, $g\beta\Delta T L^3/\nu\alpha$
S	width of corrugation, m
T	temperature, K
T_f	fluid film temperature, $(T_{cor} + T_{cold}) / 2$, K
V	voltage drop across the main heater, V
W	width of the test section, m
α	thermal diffusivity, $k/(\rho C_p)$, m^2/s
β	volumetric thermal expansion coefficient, $1/T_f$, $(1/K)$
ΔT	temperature difference between the hot corrugated plate and the cold flat plate, K
ε	emissivity
θ	tilt angle of inclination of the enclosed air layers with the horizontal, degree
σ	Stephan-Boltzmann constant

v	kinematic viscosity of the convecting fluid (air), m^2/s
cor	hot corrugated plate
cold	cold flat plate
exp	experimental
pred	predicted
room	room condition

Introduction

Convection motion and heat transfer occurring in a horizontal or an inclined fluid layers heated from below have been the subject matter of a large number of research works. Recent interest in solar collector applications has led to refined correlations, most of which envisage air as the working fluid. Specifically, a reduction in heat loss from the absorber plate of a solar collector through the cover plates improves collector efficiency. Therefore, the natural convection heat loss across air layers bounded by two parallel plates is of special interest to the designers of solar collectors.

In a confined space, most of the investigations on heat transfer have been carried out with parallel plates in horizontal and inclined positions. Globe and Dropkin [1] investigated natural convection heat transfer in liquids confined between two horizontal plates and recommended the following correlation for convective heat transfer in the range of Ralyleigh number from $3x10^5$ to $7x10^9$:

$$Nu_l = 0.069 Ra_l^{1/3} \, \mathrm{Pr}^{0.074} \tag{1}$$

Many researchers have reported about the enhancement of heat transfer for fluids with corrugated plates. Elsherbiney *et al.* [2] conducted experimental investigation of natural convection heat transfer for air layers bounded by a lower vee corrugated plate and an upper cold flat plate for the range of $10<Ra_l<4x10^6$, $0°\leq\theta\leq 60°$, and $1.0\leq A\leq 4.0$. They reported about 50% enhancement of heat transfer for vee corrugated and flat plates than those for two flat plates. Akhanda and Latifa [3] also experimentally investigated the vee corrugation, having amplitudes of 10 to 25 mm for the range of $3.29x10^4 \leq Ra_l \leq 1.88x10^6$, $1.4 \leq A \leq 9.5$ and $0^o \leq \theta \leq 75^o$. They found that at a particular Rayleigh number, a rise in amplitude of corrugation increases the convective heat transfer coefficients. Heat transfer coefficients for 25 mm amplitude of corrugation are about 15% and 8% higher than that of 10 mm, and 15 mm amplitude of corrugation, respectively. Finally, they proposed the following correlating equation:

$$Nu_l = 0.276 (Ra_l \cos\theta)^{0.294} A^{-0.31} \tag{2}$$

Chowdhury and Akanda [4] investigated natural convection heat transfer across air layers inside an enclosure having a bottom trapezoidal corrugated plate heated from below and an upper cold flat plate for the range of $9.8x10^4 \leq Ra_l \leq 2.29x10^6$, $2.60\leq A\leq 5.22$ and $0^o \leq \theta \leq 75^o$. To correlate the measured heat transfer coefficients they proposed the following equation:

$$Nu_l = 0.0112 (Ra_l \cos\theta)^{0.52} A^{-0.46} \tag{3}$$

They also reported that the measured heat transfer coefficients for trapezoidal corrugation are about 15% lower than those for sinusoidal corrugation of Akhanda and Kabir [5]. Akahanda and Chowdhury [6] performed experimental investigation for natural convection heat transfer across air layers between hot rectangular and square corrugated plates with a cold flat plate at the top for the range of $3.29x10^4 \leq Ra_l \leq 2.29x10^6$, $2.33 \leq A \leq 6.33$ and $0^o \leq \theta \leq 75^o$. They reported that for the same aspect ratio, the measured heat transfer coefficients for rectangular corrugation is about 4% higher than those for square ones and the difference between them becomes smaller with increasing inclination. Afterwards, they proposed the following equation to correlate the measured heat transfer coefficients for both rectangular and square corrugations:

$$Nu_t = 0.295(Ra_l \cos\theta)^{0.265} A^{-0.42} \quad (4)$$

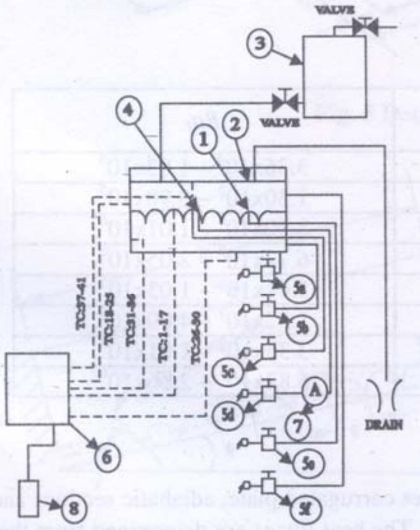
An extensive experimental study by Jularia and Gebhart [7] consisted, in part, of measuring heat transfer coefficients from an upright hemisphere on a base in water with Rayleigh number ranging from $2x10^8$ to $2x10^9$. Stewart-Johnson [8] also experimentally investigated natural convection heat transfer from isothermal spherical and hemispherical zones of 60^o, 90^o, and 120^o in the range of $2.8x10^5 \le Ra \le 2.8x10^7$. They reported that the data of 90^o zones (hemisphere) show a slightly smaller dependence of Nusselt number on the Rayleigh number. Finally, they proposed the following equation to correlate their measured values and the data of Jularia and Gebhart:

$$Nu = 0.49 Ra^{0.25} \quad (5)$$

To the authors' knowledge, no experimental investigation has been carried out for semicircular corrugation in an enclosed space between two planes. Therefore, the purpose herein is to study natural convection heat transfer across inclined air layers inside semicircular corrugated and flat plates at atmospheric pressure condition.

Experimental Apparatus and Test Procedure

Schematic diagram of the experimental set-up and the test section are shown in Fig.. 1 and 2, respectively. The inclination angle of the test section is varied by the alignment plate, providing the change in inclination angle from 0^o to 90^o at a step of 15^o. Changing the position of the cold plate assembly relative to the hot plate from 35 mm to 95 mm varies the air gap. The experimental hot plate assembly is heated by passing electric current through the main heater sandwiched between the hot corrugated plates. Details of the hot plate assembly are shown in Fig. 3. Guard heaters fixed on the guard heater assemblies are also heated by electric current. The schematic design of the inclined air layer bounded by a corrugated plate at the bottom and on upper cold flat plate is displayed in Fig. 4. The cold flat plate is placed above the hot corrugated plate assembly by four vertical clamps, which are fixed on the upper guard heater assembly. Passing a steady flow of cooling water from a reserve overhead tank cools this cold plate assembly. A digital thermometer is used to measure the temperature of the test section of the hot plate and the cold plate by 14 (36 SWG) chromel-alumel thermocouples.



1. Experimental Hot F
2. The Cold Plate
3. The Reserve Water
4. The Test Section
5a. Variac for heater-5
5b. Variac for heater-6
5c. Variac for heater-3
5d. Variac for heater-1
5e. Variac for Heater-2
5f. Variac for heater-4

Fig. 1 Chematic diagram of the

For monitoring surface temperatures of guard heater assemblies, 27 (36 SWG) chromel-alumel thermocouples are used. To obtain the total heat input of the experimental test section of the hot plate, the input current to the corresponding heater (main heater) is measured by a precision ammeter. The voltage across the main heater is measured by a precision digital voltmeter. Before starting the experiment, room temperature is recorded for selecting T_{cor} to maintain a steady $\Delta T = T_{cor} - T_{cold} = 10^{o}$C, where $T_{cold} \approx T_{room}$. The spacing between the hot corrugated plate and the cold flat plate is measured as the distance between the bottom surface of the cold plate and the mid section of corrugation of the corrugated plate. The spacing L, i.e., an air gap of 95 mm is maintained by fixing the upper outer guard heater assembly. Then the rig is aligned to the horizontal position by setting the inclination angle of the air layer, $\theta = 45^{o}$. All heaters, driven through the respective variac as shown in Fig. 1, are switched on and the water line of the cold plate opened simultaneously. By regulating the variac of each heater, the temperatures monitored by thermocouples 1-36 connected to the test section and the guard heater sections are made equal. The temperature of the test section (T_{cor}) is kept constant by controlling heat input to it. Moreover, there are four guard heaters installed around the test section. Consequently, there is virtually no heat loss from back and sides of the test section. The cold plate is also maintained at the room temperature by controlling water flow through it to maintain ΔT constant during a particular test run. Temperatures of the hot plate and the cold plate are recorded. Readings of the ammeter and the voltmeter connected to the test section of the hot plate are also recorded. All operations so far performed are repeated for an inclination angle, $\theta = 75^{o}$, keeping air gap, L =95 mm throughout. By setting the air gaps to 75 mm, 55 mm, and 35 mm, respectively, these operations are repeated. These operations are also repeated for other values of ΔT, i.e., for 18°C, 26°C and 35°C, respectively. The convective heat transfer coefficient, h is obtained from the convective heat flux as follows:

$$h = (q - q_r)/(T_{cor} - T_{cold}) \tag{6a}$$

where,

$$q = VI / A_{cor} \tag{6b}$$

$$q_r = \sigma\varepsilon^{-}_{cor}(T_{cor}{}^{4} - T_{cold}{}^{4}) \tag{6c}$$

Experimental parameter and their ranges are given in Table 1.

Table 1. Experimental parameter and their ranges

θ (degree)	ΔT (°C)	L (mm)	$A = L/H$	Ra_l
45	10 ~ 35	35	3.5	$3.36\text{x}10^4 \sim 1.03\text{x}10^5$
		55	5.5	$1.30\text{x}10^5 \sim 3.98\text{x}10^5$
		75	7.5	$3.30\text{x}10^5 \sim 1.01\text{x}10^6$
		95	9.5	$6.72\text{x}10^5 \sim 2.05\text{x}10^6$
75	10 ~ 35	35	3.5	$3.41\text{x}10^4 \sim 1.03\text{x}10^5$
		55	5.5	$1.32\text{x}10^5 \sim 4.00\text{x}10^5$
		75	7.5	$3.35\text{x}10^5 \sim 1.01\text{x}10^6$
		95	9.5	$6.82\text{x}10^5 \sim 2.06\text{x}10^6$

Experimental Uncertainty

Thermocouples used for measuring the temperatures of the hot corrugated plate, adiabatic sections and cold flat plate are estimated to have uncertainties smaller than 0.2 °C. The heat fluxes are determined from the electrical power supplied to the test section through the main heater sandwiched between the hot corrugated plates, measured with an uncertainty of less than 2 percent. In particular, heat losses are estimated to be less than 2 percent of the measured flux. Therefore, no adjustment for losses is made in calculating the heat flux.

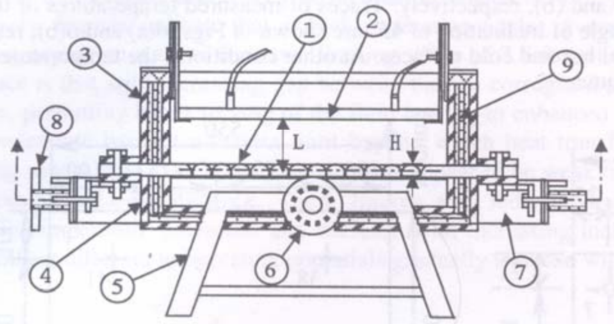


Fig. 2 Details of test section

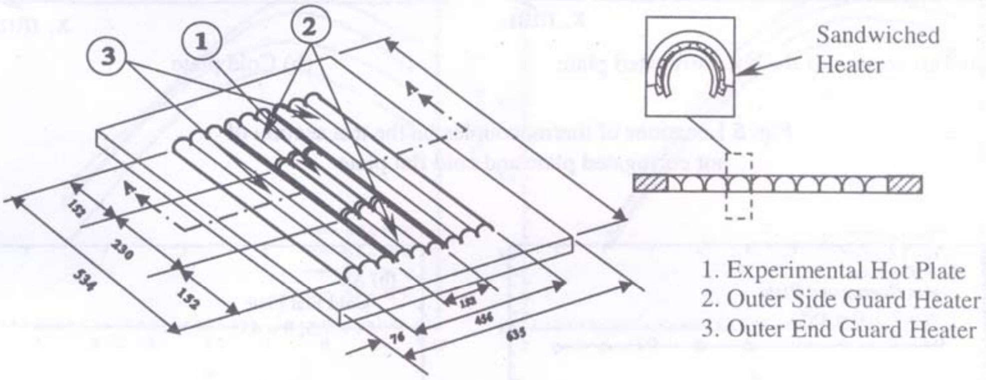


Fig. 3 Details of the hot plate assembly

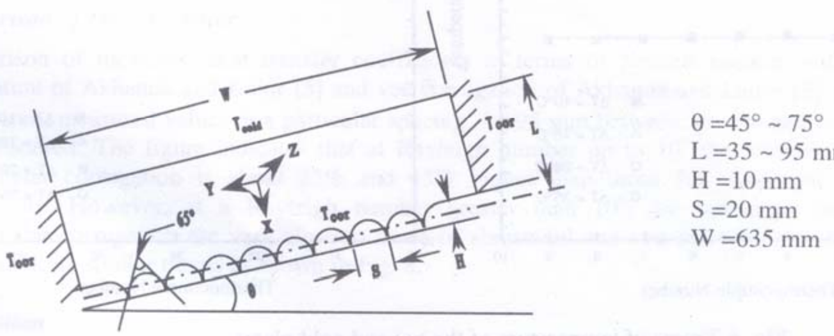


Fig. 4 Schematic design of the inclined air layer bounded by hot and cold plates

Results and Discussion

Traces of Temperature

Temperatures of the test section of hot corrugated plate and cold flat plate are measured at nine and five different locations as shown in Figs. 5(a) and (b), respectively. Traces of measured temperatures of the hot and cold plates at an aspect ratio of 7.5 and angle of inclination of 45° are shown in Figs. 6(a) and 6(b), respectively. Traces of temperatures indicate isothermal hot and cold surfaces. At other conditions, the temperature traces are almost similar to those shown in these figures.

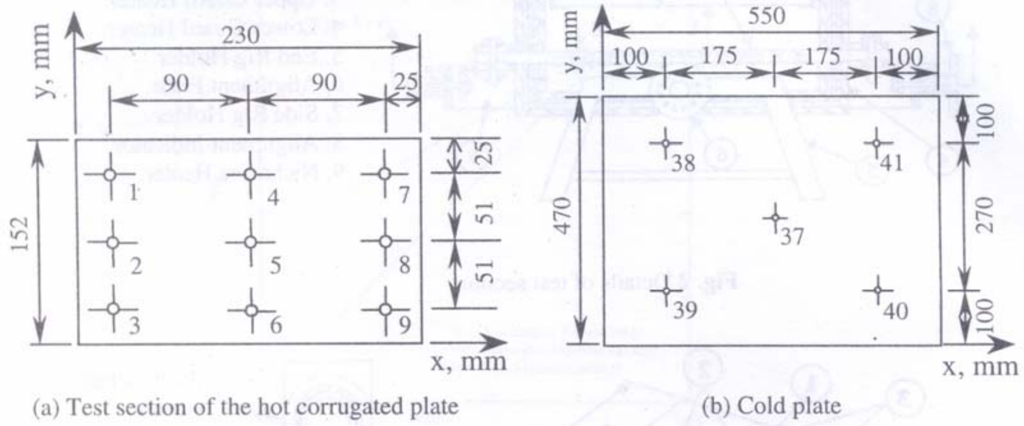


Fig. 5 Locations of thermocouples on the test section of hot corrugated plate and cold flat plate

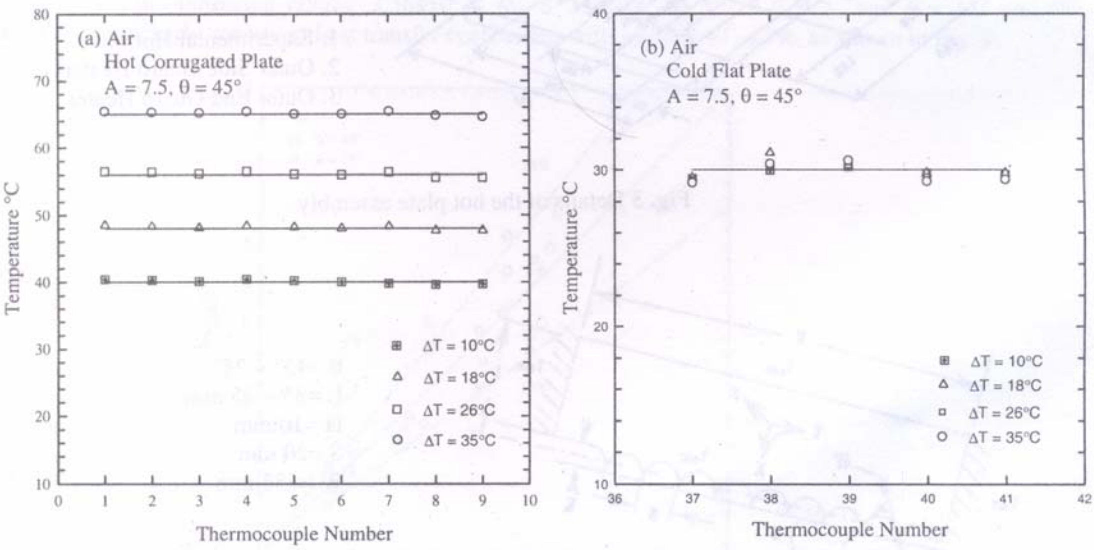


Fig. 6 Traces of temperature of the hot and cold plates

Effect of Aspect Ratio on Heat Transfer

Effect of aspect ratio on average heat transfer coefficients across inclined air layers in a rectangular enclosure for various temperature differences at inclination angles of 45 ° and 75 ° is shown in Fig.. 7(a) and 7(b), respectively. Figures indicate that at both inclination angles, average heat transfer coefficients increase with increasing aspect ratio up to a certain limiting value of 7.5 for all temperature potentials. The reason of such occurrence is that with increasing gap between the hot corrugated plate and the cold flat plate the aspect ratio increases, permitting better mixing of the fluid leading to enhanced heat transfer rate. But this augmentation in heat transfer rate has got a certain limit beyond which heat transfer coefficients are found to decrease with increasing aspect ratio. The convective currents appear to be weak to reach the cold plate, transferring less heat after crossing a larger distance. These figures also indicate that heat transfer coefficients increase with increasing temperature potentials and decrease with increasing inclination angle. Differences in heat transfer coefficients at different temperature potentials gradually increase with increasing aspect ratios.

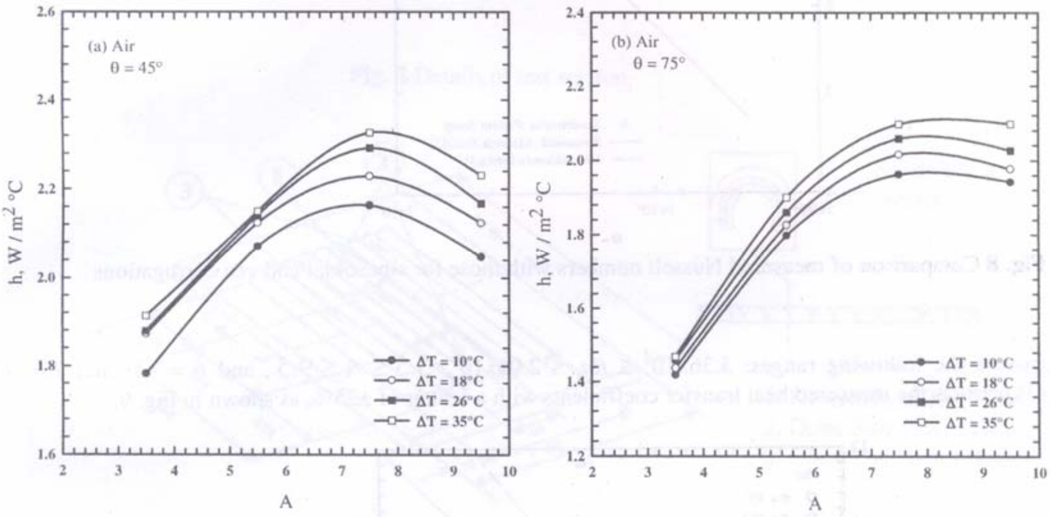


Fig. 7 Effect of aspect ratio on average heat transfer coefficient

Comparison of Heat Transfer

Comparison of measured heat transfer coefficients in terms of Nusselt number with those for sinusoidal corrugation of Akhanda and Kabir [5] and vee corrugation of Akhanda and Latifa [3] is shown in Fig. 8. For comparison, measured values at a particular spacing, L =95 mm between the hot and cold plates with θ = 45 ° are considered. The figure indicates that at Rayleigh number up to 10^6 the measured heat transfer rate for semicircular corrugation is about 25% and 15% higher than those for sinusoidal and vee corrugations, respectively. However, at a Rayleigh number greater than 10^6, the measured values corresponding to semicircular corrugation are very close to those of sinusoidal and vee corrugations. At other conditions, the results indicate similar trends as shown in Fig. 8.

Correlation

From the analysis of the problem of heat transfer by natural convection through air layers in enclosures, it can be concluded that Nusselt number depends on Rayleigh number. The aspect ratio describing the geometry of the confined space is also found to have influence on Nusselt number. Resorting to dimensional analysis, the following correlation is proposed to predict measured heat transfer coefficients for semicircular corrugation:

$$Nu_l = 0.0257(Ra_l \cos\theta)^{0.5}(A)^{-0.48} \tag{3}$$

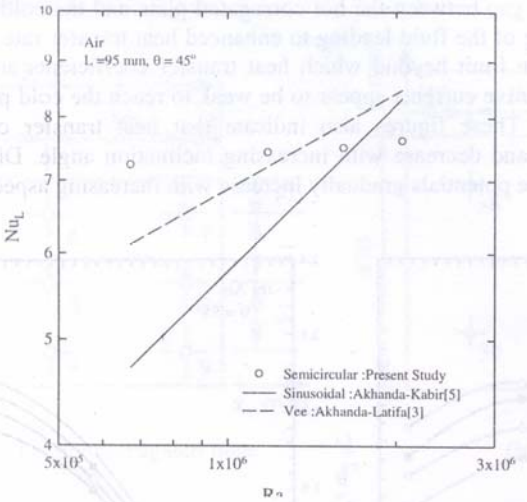


Fig. 8 Comparison of measured Nusselt numbers with those for sinusoidal and vee corrugations

The data covers the following ranges: $3.36\text{x}10^4 \le Ra_l \le 2.06\text{x}10^6$, $3.5 \le A \le 9.5$, and $\theta = 45°$ and $75°$. Equation (3) predicts the measured heat transfer coefficients with a scatter of ±25%, as shown in Fig. 9.

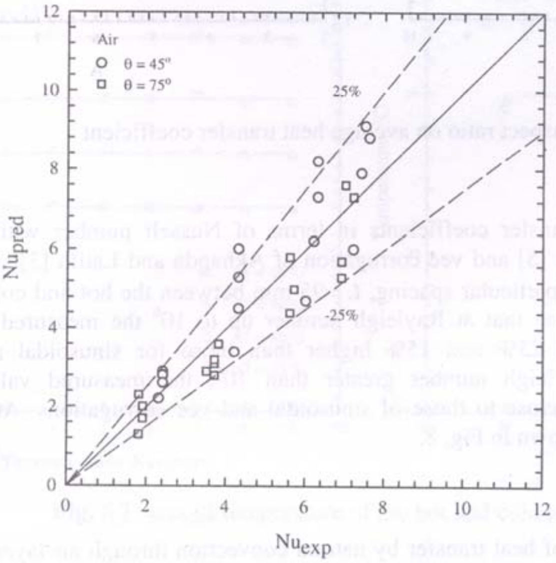


Fig. 9 Comparison of measured heat transfer with the prediction

Conclusions

Experimental study for natural convection heat transfer adhering to the proposed model gives the following results:

- Natural convection heat transfer coefficient is found to be dependent on Rayleigh number, aspect ratio, and inclination angle.
- The average heat transfer coefficient increases with increasing aspect ratio up to a value of 7.5. Beyond this critical value, a further increase in aspect ratio results in a decrease in heat transfer coefficient.
- Comparison of the measured heat transfer coefficients for semicircular corrugation with the relevant works available in the literature indicates that for the same plate spacing, the measured values at a low Rayleigh number are about 25% and 15% higher than those for sinusoidal and vee corrugations, respectively. Nevertheless, at a higher Rayleigh number, the values for semicircular, sinusoidal and vee corrugations are almost coincident.
- The proposed correlation predicts the measured data inside semicircular corrugated and flat plates with a scatter of ±25%.

References

[1] Globe, S., and Dropkin, D., Natural convection heat transfer in liquids confined between two horizontal plates, Journal of Heat Transfer, Vol. 81, p.24, (1959).

[2] Elsherbing, S. M., Hollands, K. G. T., and Raithby, G. D., Free convection across inclined air layers with one surface vee corrugated, Heat Transfer Solar Energy Systems, Howell, J. R., and Min., T., (eds.), American Society of Mechanical Engineers, New York, (1977).

[3] Akhanda, M. A. R., and Latifa, B., Natural convection heat transfer from hot vee corrugated plates to a cold flat plate, Proceedings of the 8th Asian Congress of Fluid Mechanics, Shenzher, China, pp. 607 - 610, (1999).

[4] Chowdhury F. M., and Akhanda, M. A. R., Natural convection heat transfer from hot trapezoidal corrugated plate to a cold flat plate, Institution of Engineers, India, Journal-M6, Vol. 74, pp. 133 - 138, (1994)

[5] Akhanda, M. A. R., and Kabir, H., An experimental investigation of natural convection heat transfer from hot corrugated plate to a flat plate, Proceedings of the 1st ISHMT-ASME Heat and Mass Transfer Conference and the 12th National Heat and Mass Transfer Conference, Bhabes Atomic Research Centre, Bombay, India, 1994, pp. 337-342, (1994).

[6] Akhanda, M. A. R., and Chowdhury F. M., Natural convection heat transfer from hot a rectangular and a square corrugated plates to a cold flat plate, International Journal of Thermal and Fluid Sciences, Vol. 9, No. 3, pp. 243 - 248, (2000).

[7] Jaluria, Y., and Gebhart, B., On the buoyancy-induced flow arising from a heated hemisphere, Journal of Heat Transfer, Vol. 18, pp. 415 - 431, (1975).

[8] Stewart, W. T., and Johnson, J. C., Experimental natural convection heat transfer from spherical natural zones, ASME Journal, Heat Transfer, Vol. 107, pp. 463 - 465, (1985).