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ABSTRACT

Building Integrated Photovoltaic/Thermal (BIPV/T) systems transform solar energy into both useful heat and electricity. In open loop air-based BIPV/T systems outdoor air is drawn by a variable speed fan into the channel formed by the PV modules and an insulated envelope layer. The open loop nature of the system is such that large temperature differentials exist between the interior side of the PV module and the bulk temperature of the air flowing in the BIPV/T channel. The experimental outdoor BIPV/T system studied in this paper has a length/hydraulic diameter ratio of 38 which is representative for a BIPV/T roof system. Experimental results show significantly higher Nusselt number as compared to pure forced convection heat transfer in pipes when the air is drawn through the BIPV/T system with a variable speed fan. The calculated Rayleigh and Grashof numbers are high and confirm that the system operates in the mixed convective regime for the tested Reynolds number range.

1. INTRODUCTION

Open loop Building Integrated Photovoltaic/Thermal (BIPV/T) systems draw outdoor air with a variable speed fan into a channel formed by the PV modules array and an insulated envelope layer. The outdoor air functions as the heat transfer fluid to recuperate a portion of the energy that would otherwise be lost from the PV modules to the exterior. The forced convection increases the heat transfer from the PV modules, lowering their temperature and increasing their electrical conversion efficiency. Monocrystalline and polycrystalline silicon based PV modules have typical solar energy conversion efficiencies (η) around 5-18% and have negative maximum power point efficiency temperature coefficients, μ_{mp}, between -0.04 to -0.08 % /°C (Sandberg, 1999; Duffie & Beckman, 2006; Sandia National Laboratories, 2006).

In order to properly evaluate BIPV/T system performance, numerical models must use an appropriate convective heat transfer correlation (Candanedo *et al.*, 2010). Many previous studies have generally relied on Nusselt number correlations developed for pipes and ducts that do not reproduce the actual heat transfer found in a BIPV/T system. Some of the different parameters that affect the performance of BIVP/T systems are: short developing lengths, heating asymmetry (top surface only), high aspect ratios, non-constant heat fluxes, non-uniform wall temperatures and non-uniform cross sections due to structural framing.

An experimental open loop air-based BIPV/T set up was constructed and tested outdoors in order to study the heat transfer characteristics for this system. The first part focused on the determination of Nusselt number correlations for a smooth channel at tilt angles of 45° and 30°. No significant difference was found between the experimental data for the two angles with the exception of Nusselt numbers for Reynolds number below 1600. For Re<1600, the Nusselt numbers at the 45° tilt angle were slightly higher than those for 30°.

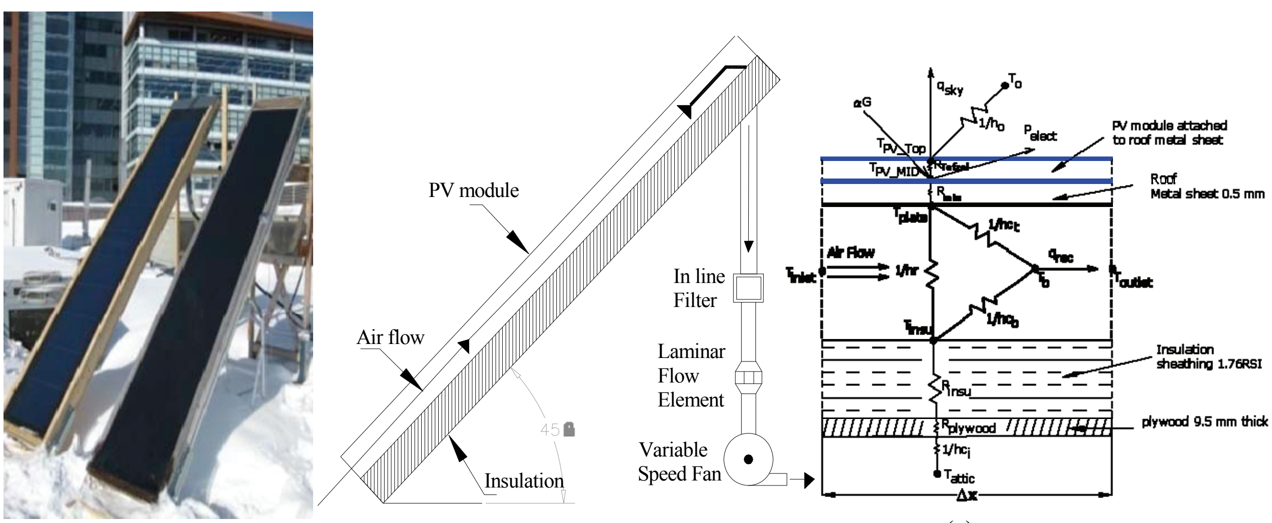
The new correlation shows higher Nusselt numbers compared with correlations for pipes and ducts such as the Gnielinski (Incropera & De Witt, 2002) correlation and the Dittus-Boelter correlation (Dittus & Boelter, 1930). The Dittus-Boelter correlation is used frequently as a reference in mixed convection and in turbulence enhancement effects studies. The use of the Dittus-Boelter equation has been recommended by McAdams (1954) for flow in pipes with flow conditions of: 10,000<Re<120,000, Length/Diameter>60, and moderate ΔT. It is not described by McAdams which values can be considered "moderate" for the ΔT.

2. LITERATURE REVIEW

The maps for the regimes of free, forced and mixed convection for horizontal and vertical pipes by Metais & Eckert (1964) can be employed to determine when mixed convection heat transfer must be taken into account. Previously, Metais (1963) developed a correlation to determine the Nusselt number at mixed regime for a horizontal tube. The equation was obtained using fluids with Prandtl number in the range 2.43<Pr<1200. The correlation has a margin of error of 20%. The same equation is listed in (Metais & Eckert, 1964). They state that the figures are useful for preliminary assessment and that they "may have to be adjusted when more results become available". For this reason the figures should not be used as a final determination of the actual convection regime.
McComas & Eckert (1966) carried out experiments in a horizontal circular tube with air. The tested Reynolds number was in the laminar region (100 to 900) and the Grashof number in the range 0.13 to 1000. It was found that for the Gr = 1000 and Re 220 the measured experimental Nu was around 1.8 the value of 4.36 (48/11).
Mori *et al.* (1966) studied heat transfer in a horizontal pipe using air as the heat transfer fluid. The pipe was uniformly heated; their results show that the experimental Nusselt numbers were higher compared to the value 48/11 which corresponds to Poiseuille flow. The 48/11 is for pipes with uniform surface heat flux and laminar fully developed conditions that is independent of Re, Pr and axial location (Incropera & De Witt, 2002).
Mixed convection heat transfer for opposed flows using water in a vertical pipe was studied in (Fewster & Jackson, 1976; Jackson & Fewster, 1977). Opposed flow conditions are found when the direction of the flow is downwards in a heated pipe or duct. Aided flow conditions occur when the fluid moves upward in a heated pipe or duct. In Fewster & Jackson's work it was found that the Nusselt number increased due to buoyancy. The experimental Nusselt numbers where higher compared to the Nusselt numbers predicted by the Petukhov-Kirillov equation which predicts the Nusselt number for pure forced convection. Fewster & Jackson (1976) also developed a correlation to predict the ratio of the effective Nusselt number to the Nusselt number predicted by the Petukov-Kirillov equation.
Experiments for air at atmospheric pressure were performed by Axcell & Hall (1978) in a vertical pipe for opposing flow conditions. The results showed that the effective Nusselt number were higher than the results as predicted by the Petukhov-Kirrilov equation and were even higher than the correlation developed by (Fewster & Jackson, 1976).
Maughan & Incropera (1987) carried numerical and experimental studies on mixed convection heat transfer for airflow in horizontal and inclined channels. The channel studied was heated from below and Re was in the range of 125 to 500. Nusselt number increased with the influence of higher Grashof number.
Sudo *et al.* (1990) investigated mixed convection heat transfer in a vertical rectangular duct by heating water in aided and opposed flow conditions. In this investigation, most of the time, heat transfer enhancement was found for buoyancy assisted and opposed flow conditions compared to the heat transfer predicted by the Dittus-Boelter equation. Smyth & Salman (1991) carried out experiments of combined free and forced convection heat transfer in a rectangular duct for the laminar region. For Gr_q in the order 10^7, the Nusselt number was found to range between 14-19. In more recent work (Dutta *et al.*, 1998) showed that for aided flow heat transfer for water subjected to asymmetric heating conditions in a square channel the heat transfer was enhanced. The same was also true for opposed flow conditions. The studied Re number was in the range of 400-10,000. Busedra & Soliman (2000) studied laminar mixed convection in an inclined semicircular duct for both assisted and opposed conditions. The experiment was carried out for 3 Reynolds numbers (500, 1000 and 1500). The employed fluid was water. For the tilt angle inclination of 20° from the horizontal (this was the maximum angle tested), the fully developed Nusselt number was found to increase with $Gr_{q,r}$ number (based on heat flux and radius) and were less dependent on Re number.
Chong *et al.*(2008) studied the effect of the inclination angle for an inclined rectangular duct with a heated plate in the middle of the channel. The studied Reynolds numbers were in the range of 420 to 2630 while the Gr was in the range of 6.8×10^3 to 4.1×10^4. The tested angles were -90°, -60°,-30°, 0°, 30°, 60° and 90°. The negative sign is used to signal opposed flow conditions. The authors report that the maximum heat transfer occurred at 30° tilt angle and that for Reynolds numbers above 1800, the influence of the tilt angle on the Nusselt number seems to diminish.

3. EXPERIMENTAL SETUP

The experimental setup used in this study consists of a near full-scale BIPV/T system and a similar system without PV panels (just metal roof) connected to an outdoor test facility fully instrumented for air collector testing. The BIPV/T system is a small scale version of the roof BIPV/T system in the EcoTerra^{TM} demonstration near net-zero energy house (Chen *et al.*, 2007).
The channel is constructed using a wood frame structure. The top surface is formed by an amorphous PV module that is pasted to a metallic roof sheet (referred to as plate). There is a 4 cm gap between the interior surface of the metal roof and the insulated bottom surface. The channel employed has a high aspect ratio (width to height) close to ten. The total fluid flow length is 2.84m.
The BIPV/T system was tested at 30° and 45° tilt angles from the horizontal. The airflow was driven by a variable speed fan. The flow rate was measured with a laminar flow element. The measurement of the temperature distributions in the BIVP/T system is carried out using T-type thermocouples. The data was collected with a data acquisition system; data included temperature distribution, flow rates, solar radiation, exterior wind speed and outdoor air temperature every minute.



(a) (b) (c)

Figure 1: (a) experimental BIPV/T setup replicating 1 strip of the BIPV/T system of the EcoTerra house with amorphous PV modules attached (left) and without PV modules (right); (b) schematic of the setup; (c) thermal network representation for a control volume of the BIPV/T system.

4. RESULTS

The experiment was done at field conditions (solar radiation & wind speed continuously changing) so data for quasi-steady state conditions had to be considered for the determination of the heat transfer coefficients. Clear sky days with low wind speeds were preferred since the convective heat fluxes were more stable under these conditions. It was experimentally determined that once the flow rate was changed, the bulk air temperature distribution reached steady-state after about 8 minutes (with solar radiation, ambient temperature and wind speed approximately constant).
Since there were high temperature gradients from the inlet to the outlet, six control volumes were used to determine local heat transfer coefficients. The total heat added to the flowing air for each control volume was computed with Equation 1. The radiation exchange for each control volume between the interior surface of the metal plate and the insulated surface is given by Equation 2

$$Q_{cv} = \dot{m} \cdot c_p \cdot (T_{outlet} - T_{inlet}) \tag{1}$$

$$\mathrm{Qrad}_{cv} = \frac{F_{plate,insu} \cdot A_{cv} \cdot \sigma \cdot (\overline{T^4{}_{plate}} - \overline{T^4{}_{insu}})}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \tag{2}$$

The measured emissivity of the steel plate was 0.80 while the corresponding value for the insulation board was 0.2. The convective heat transfer coefficients for the top heated surface and for the bottom surface are calculated as follows:

$$hc_{t,cv} = \frac{Q_{cv} - Qrad_{cv}}{A_{cv}\left(\overline{T_{plate}}_{cv} - \overline{T}_{b_{cv}}\right)} \quad (3), \qquad hc_{b,cv} = \frac{Qrad_{cv}}{A_{cv}\left(\overline{T}_{insu,cv} - \overline{T}_{b,cv}\right)} \quad (4)$$

The bottom heat loss can be considered negligible due to the high value of the insulation thermal resistance.
The average Nusselt numbers for the top and bottom surface are in turn given by Equations 5 and 6 respectively.

$$Nu_t = \frac{\overline{hc_t} D_h}{k_{air}} \quad (5), \qquad Nu_b = \frac{\overline{hc_b} D_h}{k_{air}} \quad (6)$$

The average Nusselt number coefficients for the top and the bottom surfaces were calculated from the local convective heat transfer distributions for the six control volumes. Correlations for the average Nusselt number as a function of the Reynolds number have been obtained as follows (the correlation was obtained by means of an optimized version of the Levenberg-Marquardt method for minimization (Parametric Technology Corporation, 2007)):
For the top surface, for $250 \leq Re \leq 7500$

$$Nu_{top} = 0.052 Re^{0.78} Pr^{0.4} \quad (7)$$

and for the bottom surface, for $800 \leq Re \leq 7100$

$$Nu_{bottom} = 1.017 Re^{0.471} Pr^{0.4} \quad (8)$$

Comparison of equations (7) and (8) reveals that the bottom Nusselt numbers are higher compared to the top surface. This is expected because the heating asymmetry produces bulk air temperatures that are very close to the temperature of the insulated surface in the cavity. The actual value of the Nusselt number for the bottom surface is not as important as the coefficients for the top heated surface. The reason is that the major source of heat transfer is from the top surface, which is exposed to the solar radiation.
For laminar convection, the experimental data is very close to the findings by (Chong et al., 2008) for a tilted channel employing air as heat transfer fluid. The results are also close to the findings of Smyth & Salman (1991) who carried out experiments for a rectangular horizontal duct, where the top surface was heated.
The uncertainties for the top Nusselt numbers are smaller than the ones for the bottom coefficient. For the top surface, they range between 4.8% to a maximum of 7% for $1760<Re<7500$. The highest uncertainties occur at the lowest Reynolds numbers. For example, the uncertainty is 66% at Re = 256 and 13% for Re = 802. The same behavior has been reported by Novotny et al. (1964). For the bottom coefficient, the uncertainties are much higher, ranging from 18 to 84%. The uncertainties for the Re numbers are between 3.3 to 8% for $1100<Re<7500$. At low flow rates the uncertainties are higher, e.g. for Re = 250 the uncertainty is 31% and for Re = 800 is 10%.

4.1 Tilt angle effect on average Nusselt numbers

As can be seen from Figure 2, the average Nusselt number is independent of the tilt angles 30° and 45° for Reynolds numbers above 1600. For Reynolds numbers below 1600, the Nusselt numbers for the 30° tilt angle are slightly lower than the ones for 45°.

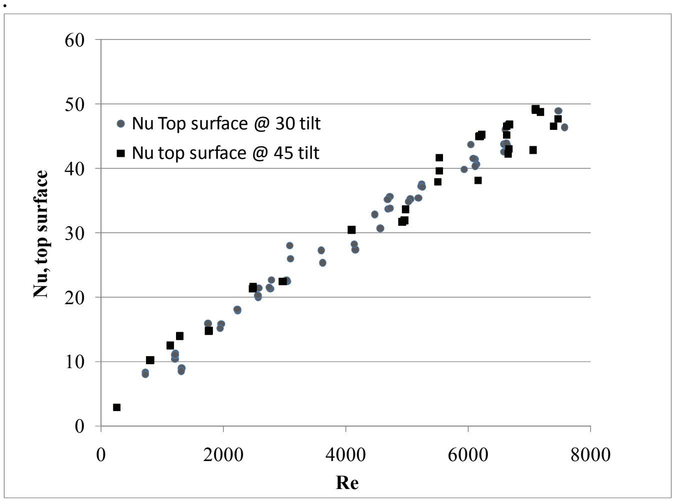


Figure 2: Nusselt numbers for the top surface versus Reynolds number - comparison for 45º and 30º tilt angles.

4.2 Rayleigh numbers

The product RaD_h/L was computed from the experimental data and plotted in Figure 3a. As can be seen, the experimental data falls in the mixed convection regime for both laminar and turbulent flow regime as compared to the map by Metais & Eckert (1964) (Figure 3 b). The Ra numbers are in the range of 1×10^5 to 9×10^5.

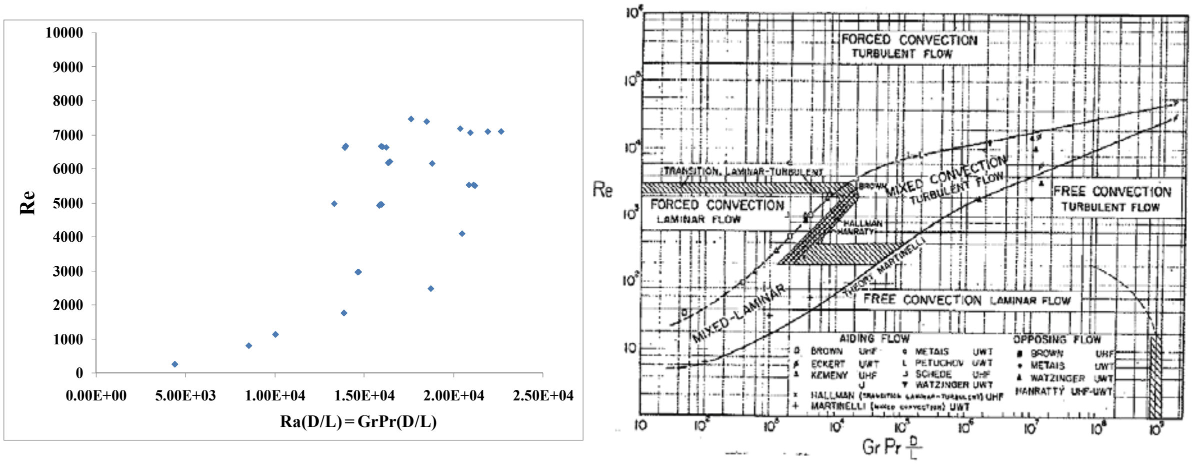


Figure 3: a) Experimental Re and $Ra(D_h/L)$ data. b) Metais & Eckert map (Metais, 1964)

4.3 Nusselt numbers for the laminar convection regime (Re<2300)

Incropera & De Witt (2002) and Kays et al. (2005) have mentioned that a common practice is to correlate mixed convection heat transfer with an expression of the form

$$\mathrm{Nu}^n = \mathrm{Nu_F}^n \pm \mathrm{Nu_N}^n \tag{9}$$

where Nu_F corresponds to correlations for forced convection and Nu_N for natural convection correlations. The + sign is recommended for assisting and transverse flows.

It was found that for the laminar regime the effective Nusselt number could be represented as the simple addition of the forced convection to the natural convection effect (n = 1). For the forced convection, the Dittus-Boelter is employed (Equation 10). For the natural convection a correlation developed by Azevedo and Sparrow using water in an inclined channel is used (Equation 11- see Incropera & De Witt, (2002).

$$Nu_F = 0.023Re^{0.8}\mathrm{Pr}^{0.4} \qquad (10), \qquad\qquad Nu_N = 0.645\left[Ra\frac{D_h}{L}\right]^{1/4} \qquad (11)$$

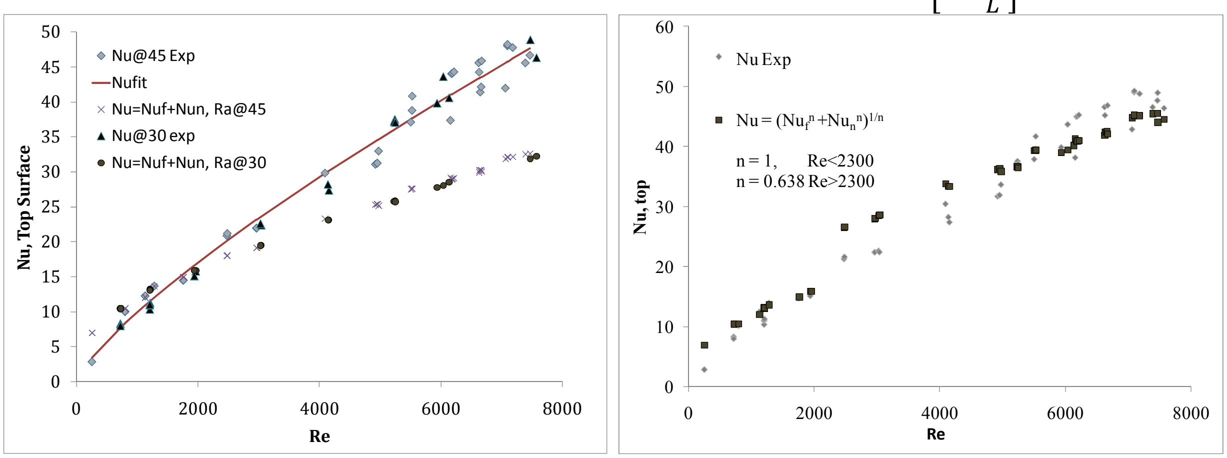


Figure 4: a) Experimental data of measured Nusselt number compared to the addition of the Dittus-Boelter and natural convection components for 30° and 45° tilt angle (n=1 in Equation 9). b) Experimental data comparison of Nusselt numbers to equation 9 (n = 1 for Re < 2300 and n = 0.638 for Re > 2300).

As can be seen in Figure 4-a, the Nusselt number predicted as the simple addition of the two components works relatively well for the laminar region Re<2300. Although, it seems to slightly overestimate the Nusselt number. If the uncertainties in the measurement are taken into account, the difference is negligible. For the turbulent convection regime another exponent n for the forced convection and the natural convection should be found.

4.4 Nusselt numbers for the turbulent convection regime (Re>2300)

A curve-fitting technique was used to determine the n exponent that will give the least square error to the measured data with equation 9. The value found was n=0.638. Figure 4-b compares the results for the two regimes. For 2300<Re<4500 the n = 0.638 fitting over predicts the Nusselt number value. For higher Re number the effect of the natural convection tends to diminish.

4.5 Comparison with Wu, Xu and Jackson correlations

Wu et al. (2002) developed a correlation where the ratio of effective Nusselt number to the Nusselt number for forced convection is given by a function of the Grashof (based on heat flux), Re and Pr numbers. The equation was developed for a vertical annular passage. In this configuration, there was a heated core and a thermally insulated outer casing. The fluid employed was water. The experimental results were represented with the following correlation:

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{F,Wu}}} = \left[1 + 1.25 \times 10^{5}\left(\frac{\mathrm{Gr}_{\mathrm{q}}}{\mathrm{Re}^{3.425}\mathrm{Pr}^{0.8}}\right)\left(\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{F,Wu}}}\right)^{-2}\right]^{0.46} \tag{12}$$

where $\mathrm{Nu}_{\mathrm{F,Wu}}$ is an equation for forced convection developed by Wu et al. (2002). The $\mathrm{Nu}_{\mathrm{F,Wu}}$ predicts values that are about 6% higher than those predicted by the Dittus-Boelter equation.

$$\mathrm{Nu}_{\mathrm{F,Wu}} = 0.042\mathrm{Re}^{0.74}\mathrm{Pr}^{0.4} \tag{13}$$

The Wu et al. equation is valid for 6,000<Re<20,000. Equation 12 was developed for opposed flow conditions. One of the strengths of equation 12 is that it correlates the influences in Grashof, Re and Pr numbers. The experimentally calculated Gr_{q} in the BIPV/T system is in the range of 5.56×10^{6} to 6.13×10^{7}. Equation 12 is plotted together with the experimental data for the BIPV/T system for Re>6000. A least squares optimization technique was employed to find the constant that multiplies the ratio of Grashof to Re and Pr numbers.

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{F,Wu}}} = \left[1 + 1.9 \times 10^{6}\left(\frac{\mathrm{Gr}_{\mathrm{q}}}{\mathrm{Re}^{3.425}\mathrm{Pr}^{0.8}}\right)\left(\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{f,Wu}}}\right)^{-2}\right]^{0.46} \tag{14}$$

which is valid for the BIPV/T system for 6000>Re>7500. Equation 14 is plotted in Figure 5 as well.

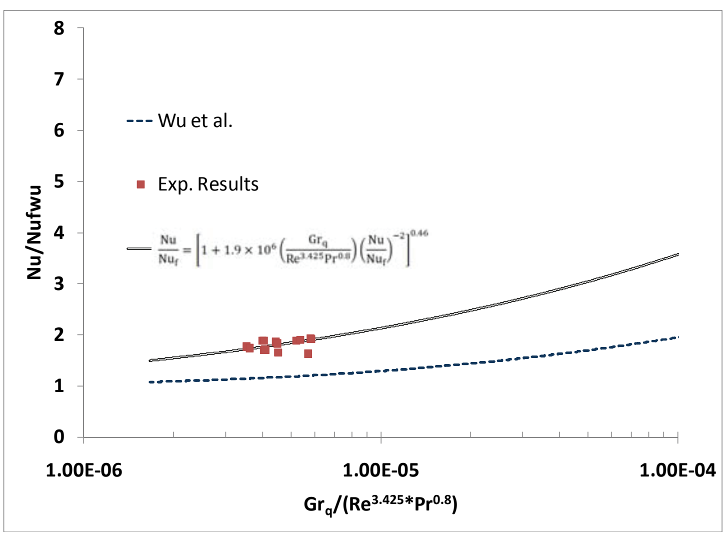


Figure 5: Experimental data comparison to Equations 12 and 14.

5. CONCLUSIONS

In the studied BIPV/T system strong buoyancy effects in the Nusselt number were found, which is consistent with the investigations by (McComas & Eckert, 1966; Mori *et al.*, 1966; Sudo *et al.*, 1990; Dutta *et al.*, 1998). It was

found that most of the data based on the product of RaD_h/L falls under the category of mixed convection heat transfer as compared with the Metais & Eckert map (see Figures 3-a and 3-b). The experimental Nusselt number data is very close to the findings by (Chong et al., 2008) for laminar convection for a tilted channel employing air as heat transfer fluid. The results are also close to the findings of (Smyth & Salman, 1991) for laminar flow.
In the studied BIPV/T system heat transfer enhancement has been found for aided flow conditions. Jackson & Fewster (1977) found heat transfer enhancement for opposed flow conditions in a vertical pipe. In the tilted BIPV/T system, the average direction of the velocity vector corresponds to the tilt angle of the BIPV/T system and therefore the flow is transverse to the buoyancy force and turbulence in enhanced. This explains the similar behavior found by Jackson & Fewster (1997) for opposed flow conditions.
Regarding the tilt angle effect, it was found that for 30° tilt angle the Nusselt number is slightly lower for Re<1600 compared to the Nusselt number at 45°. However, the difference tends to disappear with increasing Re numbers.
It was found that for laminar conditions (Re<2300) the effective Nusselt number correlates well with the simple addition of the forced convection (as predicted by the Dittus-Boelter correlation) and the natural convection components (predicted by the Azevedo and Sparrow correlation (Incropera & De Witt, 2002)). For Re>2300, the use of n = 0.638 in Equation 9 correlates relatively well the effective Nusselt number, if the uncertainties in the measurement are considered. However, a more comprehensive equation is required in order to correlate the influence of Grashof, Reynolds and Prandlt numbers. A preliminary equation (14) that correlates data for Re>6000 with Grashof, Re and Pr numbers has been found. It is recognized that more data for a wider range needs to be collected for adequate determination of the parameters in the equation.
Equations 7 and 8 are recommended to calculate the average Nusselt number in open loop BIPV/T systems. Preliminary results, not discussed here, showed that framing effects in BIPV/T system can account for a 30% increase for the Nusselt taking as a reference Equation 7.

NOMENCLATURE

$Re = \rho V D_h/\mu$	Reynolds number
Nu	Nusselt number
$Ra = g\,\beta(T_w - T_{bulk})D_h^3/\upsilon\alpha$	Rayleigh number
h	Convective heat transfer coefficient, $W/m^2 \cdot K$
$D_h = A_c/U$	Hydraulic diameter, m
k	Thermal conductivity, W/m·K
$Gr = g\rho^2\beta(T_w - T_b)D_h^3/(\mu^2)$	Grashof number
$Gr_q = (g\beta q_w D_h^4)/(\upsilon^2 k)$	Grashof number based on heat flux
$Gr_{q,r} = (g\beta\rho^2 q\, r^3)/(\mu^2 k)$	Grashof number based on r and heat flux
L	Channel length, m
β	Thermal expansion coefficient, 1/K
q_w	heat flux on the wall, W/m^2
υ	kinematic viscosity, m^2/s
μ	dynamic or absolute viscosity, kg/(m·s)
$\mu_{mp} = \frac{(\eta_{mp} - \eta_{mp,ref})}{T_c - T_{c,ref}}$	maximum power point efficiency temperature coefficient,%/ °C
ρ	air density, kg/m^3
Q_{cv}	Total heat transfer rate in control volume, W
Pr	Prandlt number, v/α
A	collector exposed area, m^2
A_c	cross sectional area, m^2
σ	Stefan Boltzmann constant, W/m^2K^4
T_{plate}	interior side temperature of the metal sheet, °C, or in K
T_{insu}	interior side temperature of the insulation, °C, or in K
$\varepsilon_1, \varepsilon_2, \varepsilon_3$	longwave emissivities
U	wetted perimeter, m
V	average air velocity in the channel, m/s

Subscripts

N	natural
F	forced convection
t	top surface
b	bottom surface
h	hydraulic
cv	control volume
c	cross section
mp	maximum power

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