Natural Convection Heat Transfer in Façade Integrated Solar Concentrators

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Abstract

The use of facade (wall) integrated concentrating solar collectors is an area of developing interest within the field of energy-efficient building technology. One way of achieving such a device could be through the use of a static flat or parabolic reflector used in conjunction with a photovoltaic/thermal absorber and a vertical glazed aperture. However, to precisely predict the performance of such facade integrated collectors, it is essential to understand the heat losses from them. In general the thermal losses from these collectors can be calculated using existing relationships for flat plate solar collectors and fundamental heat transfer concepts.

However there is no relationship in the literature to describe the natural convection heat transfer in the asymmetric enclosed air gap formed by a façade integrated concentrator as described. Hence, in this study, a relationship to describe the natural convection heat transfer in such enclosures was developed using an experimentally validated computational fluid dynamics analysis. The relationship shows that the heat transfer, expressed in terms on the Nusselt number, is strongly dependent on the Rayleigh number and the aspect ratio (A/H), and can be expressed in the form $Nu=a\,Ra^b\,(A/H)^c$.

Keywords: natural convection, triangular enclosure, façade, solar concentrator

1. Introduction

In an urban environment with a limited supply of roof space, façade integration of photovoltaics provide an innovative way of harnessing solar energy and therefore is an area of developing interest in the field of sustainable building practices. Unlike standalone solar power systems there is no additional requirement of land, the cost of the solar façade or roof can offset the cost of the building structure it replaces, and power generated on site replaces the electricity otherwise purchased at commercial rates. The shortcoming of building integrated photovoltaic modules though is their relative low efficiency, as they convert only 10-20% of the solar irradiance to useful energy. This problem is exacerbated when planar modules are integrated into vertical facades, as the often large incidence angles deliver poor optical performance.

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In response to this Karlsson and Wilson (2000) proposed the Maximum Reflector Collector (MaReCo) design that utilised a parabolic reflector to increase the radiation falling on a photovoltaic module. Systems of a similar premise to this, with photovoltaic absorbers, have been discussed by a number researchers including: (Brogren 2004; Brogren and Karlsson 2002; Gajbert, et al. 2007; Mallick, et al. 2004; Zacharopoulos, et al. 2000; Zanesco and Lorenzo 2002).

However, in a recent study Piratheepan and Anderson (2013) compared the illumination profiles of a façade integrated collectors with parabolic and flat reflectors. They showed that using a low concentration flat reflector for façade integration, instead of a parabolic reflector as proposed in the MaReCo design, would reduce the non-uniform illumination of the photovoltaic absorber that could reduce the electrical efficiency and durability of the cells. Furthermore, a flat reflector would allow low cost crystalline silicon solar cells to be used in low concentration ratio systems, instead of CIGS solar cells (Nilsson 2005).

In an ideal scenario the majority of the energy that reaches the photovoltaic collector should be absorbed and utilised. One promising way of achieving this is by using hybrid photovoltaic/thermal absorbers that deliver both thermal and electrical energy simultaneously. Here the thermal energy can be carried away by a transport medium as useful energy; however, thermal losses from the collector to the environment are inevitable. Interestingly, though two of the studies discussed earlier (Brogren and Karlsson 2002; Gajbert 2008) used water to cool the photovoltaic absorber in order to improve their electrical performance, neither study took steps to determine the heat balance of the photovoltaic module. On this basis, Piratheepan and Anderson (2014) proposed a design for a low concentration ratio façade-integrated photovoltaic/thermal concentrator, as shown in Figure 1. Conceptually, this concentrator is similar to that proposed by previous researchers in that it includes a static reflector and a photovoltaic absorber. However, the proposed system includes a vertical glazing, thus forming an enclosed triangular air gap to reduce heat loss from the photovoltaic absorber.

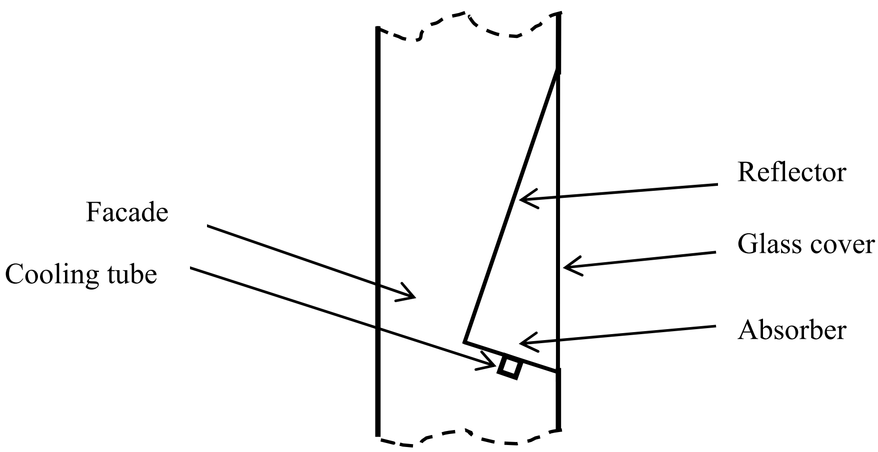

Figure 1 Façade Integrated Solar Concentrator

In considering the natural convection heat transfer in such an enclosure, a large number of computational studies have been performed (Asan and Namli 2001; Holtzman, et al. 2000; Saha 2011; Salmun 1995; Varol, et al. 2007). However, the majority of the studies are related to horizontal right or isosceles triangular spaces (attic shaped enclosures) with one heated and one cooled side or enclosures heated from below (Flack 1980; Holtzman, et al. 2000) and amongst these studies there is a lack of generalised correlations to predict the heat transfer in triangular enclosures that are applicable to the proposed solar concentrator.

That said, there have been a number of studies that examine natural convective heat transfer in concentrating solar collectors. Anderson (2013) developed a relationship to describe the natural convection heat transfer in an enclosed V-trough solar concentrator, essentially a two reflector version of the proposed system. Furthermore, a study by Singh and Eames (2011) reviewed the significance of the shape, aspect ratio, and the orientation of the solar concentrator enclosures in terms of their effect on convective heat transfer coefficients and thermal performances.

Despite the numerous studies on static solar concentrating systems, and the heat transfer from concentrating cavity enclosures, there are no existing relationships that can confidently be used to describe the natural convection heat transfer between the absorber and glazing for an enclosure formed by a façade integrated solar concentrator as shown in Figure 1, or any other façade integrated truncated parabolic reflector enclosures similar to that of Brogren (2004). In light of this lack of relationships, this study evaluates the convection heat transfer coefficient in the proposed collector, and other possible variations, in order to facilitate accurate prediction of façade integrated solar concentrator performance.

3. Computational fluid dynamics analysis

In order to develop a correlation that describes the natural convection heat transfer in the enclosures formed by a façade integrated solar concentrator it was decided to undertake a computational fluid dynamics analysis of the system without radiative heat transfer. Therefore several models were created and analysed using Solidworks Flow Simulation, a commercial computational fluid dynamics (CFD) solver based on the finite volume method. The Solidworks Flow Simulation solver is capable of analysing both laminar flows and turbulent flows using the Reynolds averaged Navier-Stokes equations. In doing this, it uses the same transport equations for both laminar and turbulent flows which gives the flexibility to use them in transitional flows as well (Solidworks 2014). The turbulence in the flow is treated using the turbulent kinetic energy (k) and turbulence dissipation rate (ε) using a standard k- ε turbulence model. In doing this, the standard values for the constants in the transport equations are used, those being $C_m = 0.09$, $C_{e1} = 1.44$, $C_{e2} = 1.92$, $\sigma_e = 1.3$ and $\sigma_k = 1$ (Versteeg and Malalasekera 2007). Despite the inability to modify these coefficients, Anderson et al (2009) showed that this did not unduly affect the solvers ability to accurately model natural convection heat transfer in an enclosure.

In order to reduce the complexity and computational time for the simulations, the geometry and simulation were performed using a two-dimensional simulation. This assumption was made on the basis that the enclosure for such a system would be relatively long and thus the majority of the buoyancy driven flow would be two-dimensional in nature. In defining the boundary conditions for the simulation it was assumed that the inclined wall (the reflector) was an adiabatic surface and the absorber was kept at a constant high temperature (T_h), while the vertical glazing (cold wall) was set at a lower constant temperature (T_c). In order to change the Rayleigh number, the cold wall was kept at a constant temperature while the temperature of the absorber was changed in order to achieve ΔT values in the range of 20-60K. This corresponded to the Rayleigh numbers between 1×10^8 and 1×10^9.

Additionally, three triangular enclosures were tested with angles between the reflector and absorber ranging from 90 - 120 degrees, as shown in Figure 2 (i-iii). This was achieved by changing the length of the adiabatic wall without changing the length of either the hot or cold wall. This was done in order to determine the change in heat transfer for various potential absorber-reflector configurations. This variation is subsequently presented as the aspect ratio, which for this study was defined as the ratio of the length of the absorber (A) and the length of the reflector (H). Hence, the variation of aspect ratio ranged from 0.324-0.381. Furthermore, as previous studies had utilised parabolic rather than flat reflectors a fourth enclosure was modelled, where the flat reflector was replaced by a truncated parabola as shown in Figure 3 (iv). This was done in order to examine the possibility of using the deduced correlation to predict the value of heat transfer coefficients in enclosures where a truncated parabolic reflector might be utilised.

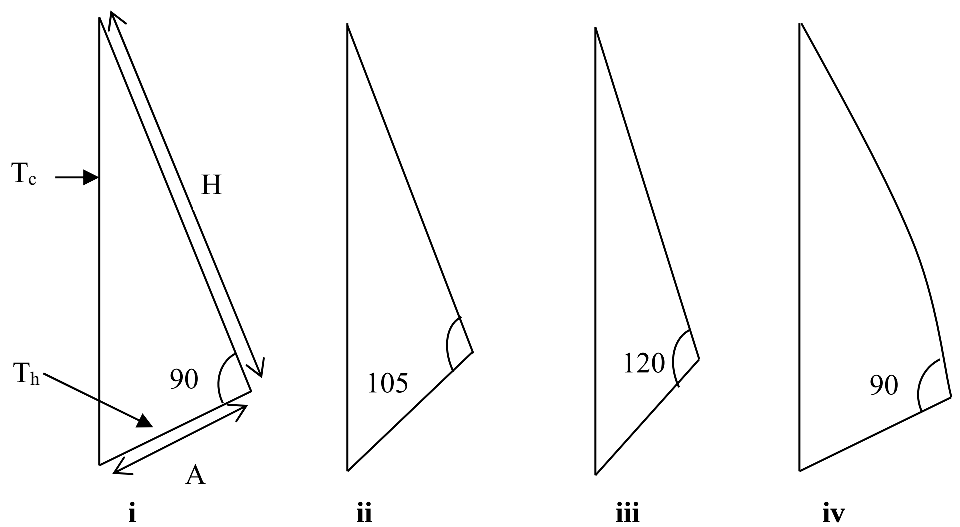

Figure 2 Enclosures modelled in CFD

4. Experimental validation method

In order to validate the results of the computational fluid dynamics analysis an analogue of the collector shown in Figure 3i, with a 90 degree angle, was fabricated. In constructing this analogue, a heating element was used to replicate the photovoltaic absorber plate in the proposed collector and a 1.2 m long enclosure with cross sectional dimensions shown in Figure 3 was built.

To achieve a temperature gradient between the cold plate and the hot plate, the hot plate was made by bonding a flexible resistance heater to a 2 mm thick 1200 mm x 200 mm polished aluminium plate. Aluminium plate was used to ensure a uniform temperature along the length of the enclosure thus providing an isothermal surface and also for its low emissivity, to reduce radiation heat transfer.

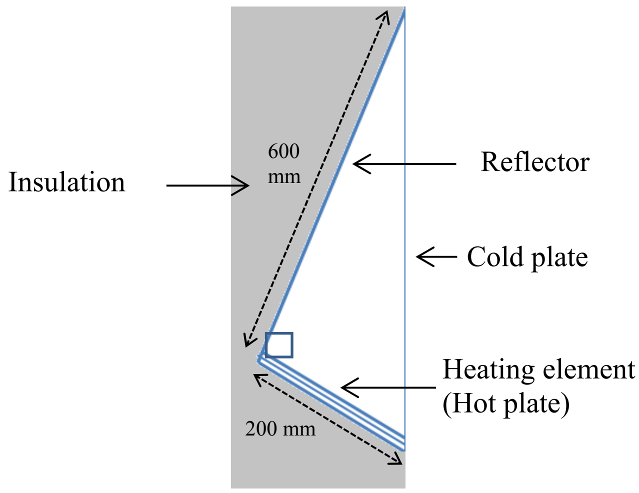

Figure 3: Module to be used for the heat transfer experiment

In an effort to minimize the heat losses through the rear and sides of the heater, the rear side of the heater and the reflector plate were insulated with 100 mm thick (or more) mineral wool fibre with a nominal R-value of 2.8 and enclosed by 18mm plywood, while the ends of the enclosure were fabricated from 18 mm plywood. The inclined "reflector" surface was also constructed of 2 mm thick polished aluminium plate, with a low emissivity, to minimize radiation heat transfer between it and the other surfaces in the enclosure. In addition, the vertical "glazing" (cold plate) consisted of a polished 2 mm thick aluminium sheet in order to keep the temperature uniform over the entire area. Furthermore, the external surface of the "glazing" was cooled by two fans to ensure the majority of the heat from the enclosures was taken away through this surface. Finally all the edges and joints of the enclosure were sealed with high temperature aluminium duct tape to reduce any undesirable heat losses by air leakage.

In order to vary the Rayleigh number, either the temperature difference between the plates or the characteristic length has to be varied. To vary the temperature difference, the power supplied to the electrical heater was varied by a variable transformer (Variac) and the amount of power supplied was measured by a single-phase MS6115 power meter.

The mean temperature of the hotplate was measured by six T-type thermocouples (±0.3K) attached uniformly along the length of the hotplate. Nine more T-type thermocouples were used over the surface of the cold plate to measure its mean temperature. The ambient temperature was measured by another T-type thermocouple. Subsequently all of these thermocouples were connected to two Picolog TC-08 data loggers connected to a computer through a USB interface. A schematic of the experimental configuration is shown in Figure 4.

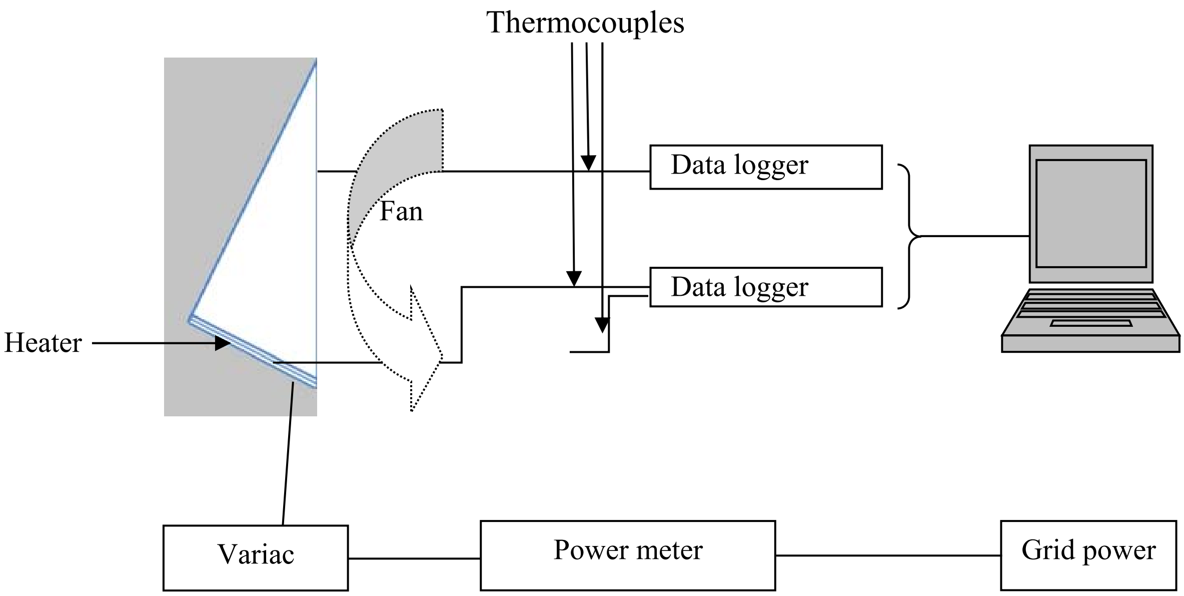

Figure 4 Schematic of the experimental setup

The difference between the hot and cold plate temperatures (ΔT) was calculated using the mean hot and cold plate temperatures. The experiment was allowed to run until it reached steady state, where the thermocouple readings were used to identify the region of steady state. Here the steady state condition was determined by taking the region where the difference between the mean temperature of the heater and the cooled surface did not vary by more than 0.6K over a 30 minute time period.

5. Experimental Analysis

From the experimental results, the overall heat transfer coefficient (U) can be derived from Newton's cooling law. Here the heat transfer coefficient under steady state conditions is given as a function of the input electrical power (Q_T), the temperature difference between the plates (ΔT) and the area of the hotplate (A_h) as shown in Equation 1.

$$U = Q_T/(A_h\, \Delta T) \quad (1)$$

However, this provides only an approximate indication of the convective heat transfer coefficient inside the enclosure. In order to obtain the heat that is being transferred by the convection ($Q_{convection}$) a complete heat balance should be undertaken, as given by Equation 2. Thus, the $Q_{convection}$ can be expressed in terms of total electrical energy input and the other means of heat losses.

$$Q_T = Q_{convection} + Q_{conduction} + Q_{radiation} \quad (2)$$

Where, $Q_{conduction}$ is the sum of rear, base and end conduction losses while the $Q_{radiation}$ is the amount of heat loss by radiation.

The rear, bottom and end heat losses by conduction ($Q_{conduction}$) can be calculated by applying Fourier's law over the areas of interest (A_i) across the thickness (L) where (k) is the thermal conductivity of the wall and temperature difference across the wall is (T_i-T_a) as given in Equation 3. In this equation T_a is the ambient air temperature. In the case of rear and bottom heat losses T_i was the temperature of the hot plate, while the end loss was calculated using the bulk temperature of the air inside the enclosure as the value of T_i.

$$Q_{conduction} = \frac{kA_i(T_i - T_a)}{L} \quad (3)$$

Now, the radiation heat transfer between the hot and cold plate ($Q_{radiation}$) is another portion of the thermal losses from the hotplate. However as the inclined reflector surface of the enclosure was heavily insulated, it can be assumed to be an adiabatic surface under steady state conditions, and the radiation losses can be discounted. Similarly, given that both hot and cold plates were fabricated using low emissivity aluminium plate ($\varepsilon_p \approx 0.06$), one could assume that radiation between these surfaces is also negligible.

Based on this assumption, Equation 2 can be reduced to Equation 4 without the radiation component.

$$Q_{convection} = Q_T - Q_{conduction} \quad (4)$$

By combining Equations 1 and 4, the convection heat transfer coefficient (h_c) can be expressed in terms of $Q_{convection}$, area of the hotplate (A_h) and the temperature difference (ΔT) between the hot and cold plate as given by Equation 5.

$$h_c = (Q_T - Q_{conduction})/(A_h \Delta T) \quad (5)$$

6. Results

6.1 Computational fluid dynamics analysis

In order to determine the natural convection heat transfer coefficient of the proposed enclosures, the temperature of the isothermal heater was varied to achieve a 26, 46 and 63K temperature difference between the heater and the isothermal cold wall.

As expected, Figure 5 shows that there is an increase in the heat transfer coefficient as the temperature difference increases. This is because as the temperature difference increases the degree of turbulence in the fluid increases, and hence there is increased heat transfer from the heater (Cengel 2007).

Furthermore the increase in aspect ratio defined as a ratio between the length of the absorber plate (A) and reflector plate (H)) from 0.324-0.381, achieved by varying the angle between the reflector and absorber, also increased the heat transfer coefficient at a particular temperature difference. This suggests that the heat transfer coefficient is also a function of the aspect ratio.

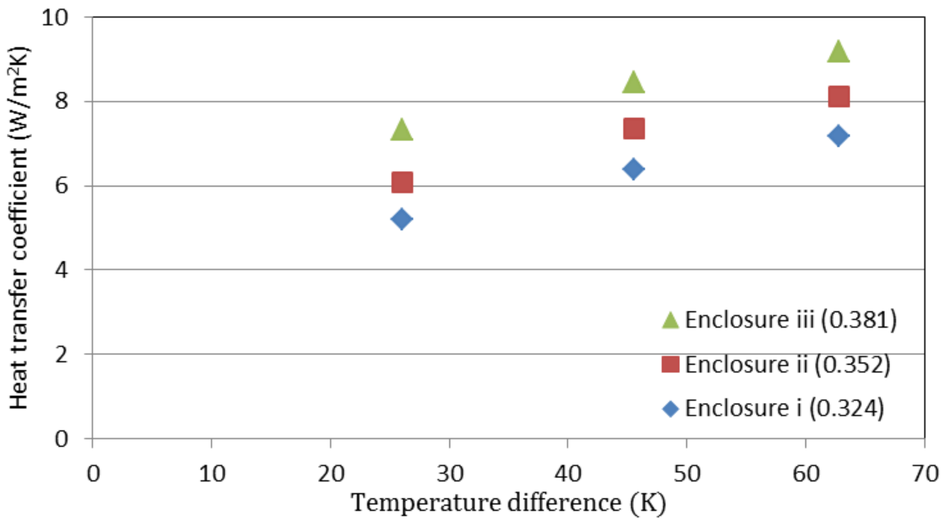

Figure 5. Natural convection heat transfer coefficient v Temperature difference for enclosures with varying aspect ratios

Although the results given in Figure 5 relate the temperature gradient to the heat transfer coefficient, they cannot be used directly to find the heat transfer coefficient in different sized enclosures with different angles. As such, Figure 6 shows the relationship between Nusselt number and Rayleigh number for enclosures of varying aspect ratios. From these it is possible to develop a generalised relationship between these parameters that can be used to obtain the heat transfer coefficient of geometrically similar enclosures of various sizes.

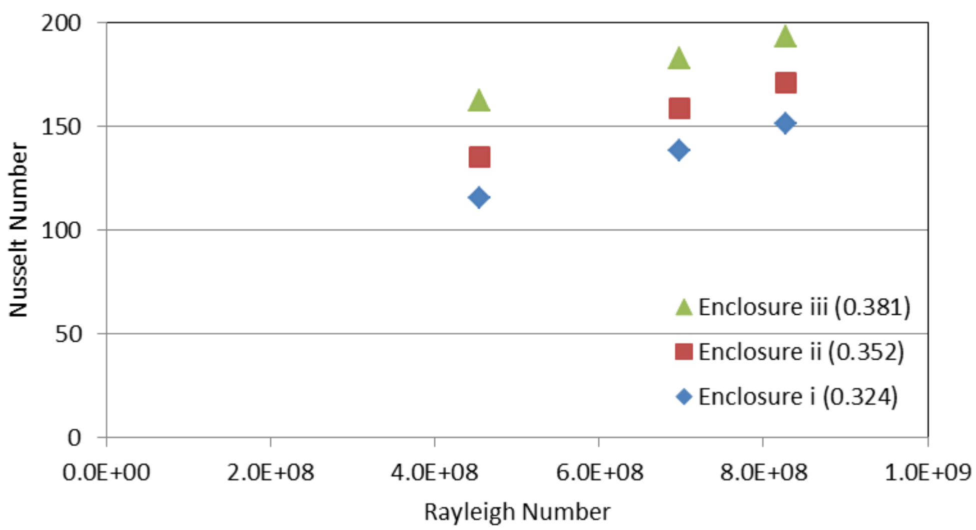

Figure 6 Nusselt v Rayleigh number for enclosures with varying aspect ratios

In achieving this, the air in the enclosure was assumed to be a real fluid and its thermo-physical properties were calculated at the bulk temperature (the average temperature of hot and cold plate) of the enclosure and the Rayleigh and Nusselt number were calculated on this basis. This corresponded to Rayleigh numbers from 5 x 10^8 to 1 x 10^9, taking the characteristic length to be the height of the cold plate.

Now, in Figures 5 and 6, the heat transfer characteristics of triangular enclosures (enclosures i-iii) were examined, however an enclosure with a parabolic reflector was also modelled (enclosure iv). Therefore, the natural convection heat transfer coefficients in this enclosure, under the same conditions as the three triangular enclosures, were also determined.

In Figure 7, it can be seen that the natural convection heat transfer coefficients in the parabolic enclosure, under the same conditions, are essentially the same as those of enclosure i, and varied only by having a flat rather than curved reflector. On this basis it is apparent then, that a single relationship could be developed that describes the natural convection heat transfer coefficient for enclosures with flat and parabolic reflectors.

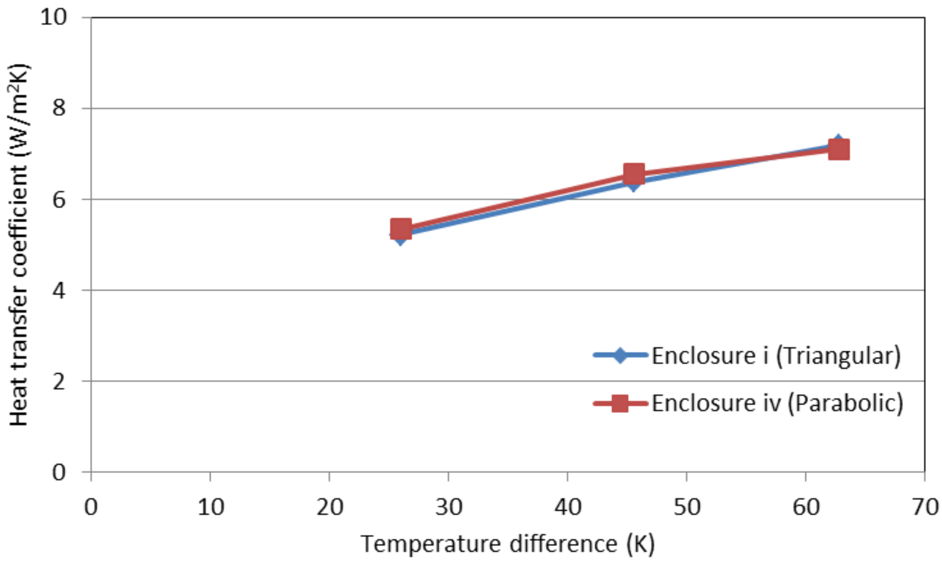

Figure 7 Natural convection heat transfer coefficient v Temperature difference in enclosures i and iv

As shown in Figures 6 and 7, all four enclosures follow the same trend of increased Nusselt number with Rayleigh number, and clearly indicate that Nusselt number also strongly depends on the aspect ratio. Therefore, a relationship was determined from the computational analysis that expressed the natural convection heat transfer in terms of both the Rayleigh number and the cavity aspect ratio as given by Equation 6.

$$Nu = 0.67Ra^{0.26}(A/H)^{1.75} \quad (6)$$

6.2 Experimental results

In order to validate the CFD results, the natural convection heat transfer coefficient of the proposed enclosure was calculated by varying the heat supplied to the hotplate. By changing the heater power, the mean temperature of the hotplate was varied from 48°C to 112°C. Using the recorded temperature difference and the amount of heat supplied, it was possible to calculate the convection heat transfer coefficient from Equation 6. At each temperature difference under steady state, the Nusselt numbers were plotted against the Rayleigh number.

As shown in Figure 8 the experimental results closely match the previous CFD results for the enclosure of the same geometry. As such, the experiments validate both the results of the computational analysis and also the correlation developed from this analysis to describe the natural convection heat transfer coefficient for an enclosed solar concentrator with a static reflector.

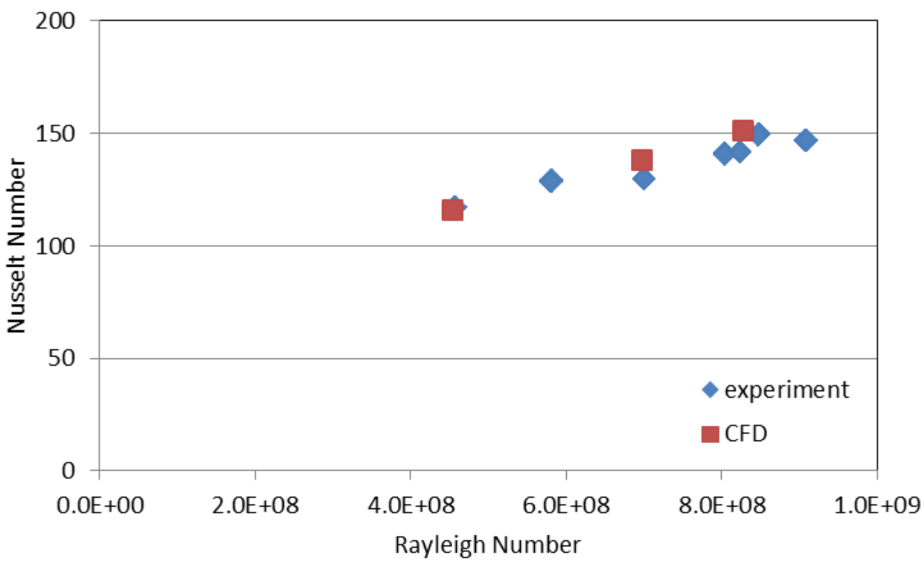

Figure 8 Nusselt v Rayleigh number for experimental and CFD analysis

7. Conclusion

The flat reflector solar concentrator concept proposed in this study offers a simple solution to increase the solar radiation on the absorber plate in a façade integrated solar system. In this way, the electrical/thermal output from such façade collector system can be improved. However, to precisely predict the performance of the system, it is essential to understand the heat losses from it.

In light of the lack of correlations relating the natural convection heat transfer coefficient for the solar collector geometries, this study found that the natural convection heat loss could be predicted by the relationship $Nu = 0.67Ra^{0.36}(A/_{H})^{1.75}$ for Rayleigh numbers in the range of 1 x 10^8 to 1 x 10^9. The experimental results suggest that this relationship is capable of determining the natural convection heat transfer coefficient in enclosed solar concentrators with flat reflectors. Furthermore, the inclusion of the aspect ratio term allows this relationship to be used in enclosures with different concentration ratios. As such, this correlation can be applied for geometrically similar enclosures with different dimensions, assuming that the Rayleigh number is within the range of the tested values.

Finally, there is close agreement between the natural convection heat transfer coefficient in enclosures with flat reflectors and the enclosure with the parabolic reflector. This shows that the relationship can be used in the determination of the natural convection heat transfer coefficient in enclosures with parabolic reflectors that may also be used in façade applications, as has been discussed in previous research work.

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