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Experimental Evaluation of Natural Heat Transfer in Façade Integrated Triangular Enclosures

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Abstract

The use of building integrated solar concentrators 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 flat static reflector used in conjunction with a photovoltaic/thermal absorber and glazing, thus forming a triangular enclosure similar to a single sided V-trough enclosure. Such systems may be well suited to adoption in façade applications.
However, in order to precisely predict the performance of such building integrated façade collectors, it is crucial to understand the heat losses from the absorber. Unlike a flat plate collector, the glazing is not parallel to the absorber plate hence we need to develop a relationship that describes the thermal losses through the vertical glazed cover. As the air between the collector and the glass is isolated from the atmosphere, the thermal loss from the absorber to the glass cover will be due to natural convection and radiation. The radiation heat transfer can be readily determined using view factors, however there is no relationship in the literature to describe the natural convection heat transfer in triangular enclosures similar to the one illustrated.
In this study, a relationship to describe the natural convection heat transfer in such triangular enclosures was experimentally determined. The relationship shows that the heat transfer, expressed in terms on the Nusselt number, is strongly dependent on the Rayleigh number and can be expressed by a relationship of the form of $Nu = aRa^b$.

1. Introduction

In an urban environment, with limited supply of roof space, low concentrating façade integration of solar collectors opens an innovative way of harnessing solar energy and is an area of developing interest within the sustainable building practices. One way of achieving such a device could be through the use of static reflectors used in conjunction with glazed photovoltaic/thermal absorbers.

Systems of a similar premise to this with photovoltaic absorbers have been discussed by researchers in the UK and Ireland (Zacharopoulos, et al. 2000, Mallick, et al. 2004), Sweden (Brogren 2004) (Brogren and Karlsson 2002), (Gajbert 2008) and Brazil (Zanesco and Lorenzo 2002), with respect to increasing their electrical output. Interestingly, although some of the studies (Brogren and Karlsson 2002), (Gajbert 2008) have used water to cool the absorber in order to improve their electrical performance, none of these studies has taken steps to predict their precise thermal performance by analysing the heat balance of the absorber plate.

In an ideal scenario the majority of the energy that reaches the absorber plate should be absorbed and utilised. One promising way of achieving this is by using photovoltaic/thermal absorber plates that can deliver both thermal and electrical energy simultaneously. Here the thermal energy can be carried away by the transport medium as useful energy.

However, thermal losses from the collector to the environment are inevitable. Minimizing the thermal losses and knowing the parameters governing the heat losses will help understand the operation of the collector from the design phase. Previously, (Piratheepan and Anderson, 2014) proposed a design for a low concentration ratio façade-integrated photovoltaic/thermal concentrator which when glazed would form a triangular enclosure filled with air, as shown in Figure 1.

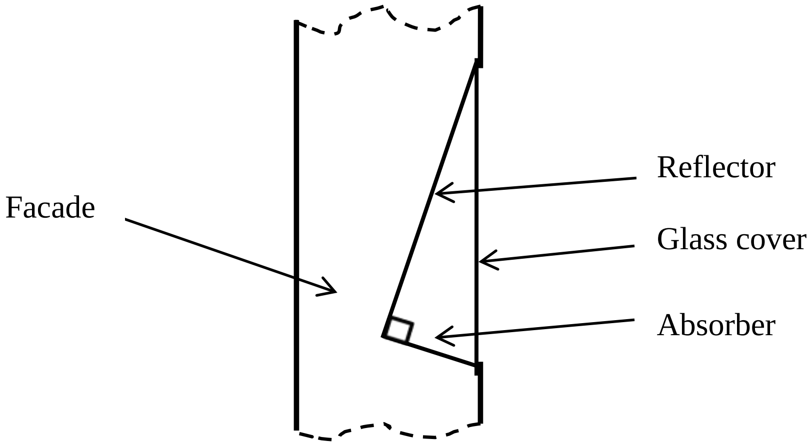

Figure 1 Façade Integrated Concentrator

Of the available literature relating to convection heat transfer coefficients in triangular enclosures, the majority have been aimed at attic shaped enclosures. In a recent study (Anderson et al, 2010) examined an attic shaped enclosure with a heated inclined surface and suggested that a correlation given by (Ridouane and Campo, 2005) could be used in the determination of the convective heat transfer coefficient. In another study, (Anderson, 2013) found a relationship between Nusselt number and Rayleigh number in the form of $\mathrm{Nu}=a\mathrm{Ra}^{b}$ that described the heat loss in enclosed V-trough concentrators operating at a range of orientations. However, there are no existing relationships to describe the convection heat transfer between the absorber and glazing in an enclosure such as that shown in Figure 1.
In light of this lack of experimental data relating to tilted right triangular enclosures with a heated base, this study experimentally evaluates the convection heat transfer coefficient in the proposed collector, in order to facilitate accurate prediction of the collector's performance.

2. Experimental Method

In order to determine the convective heat loss from the absorber, an analogue of the collector was fabricated, where a heating element was used to represent the absorber plate in the proposed collector. It was decided to build a 1.2m long enclosure with cross sectional dimensions of 200mm x 600mm as shown in Figure 2. To achieve a temperature gradient between the cold plate and the hotplate, the hotplate was made with flexible resistance heating wires attached to a 2mm polished aluminium plate. Aluminium plate was used to ensure a uniform temperature along the heater plate (owing to its high thermal conductivity) and also for it's low emissivity, to reduce radiation heat transfer.

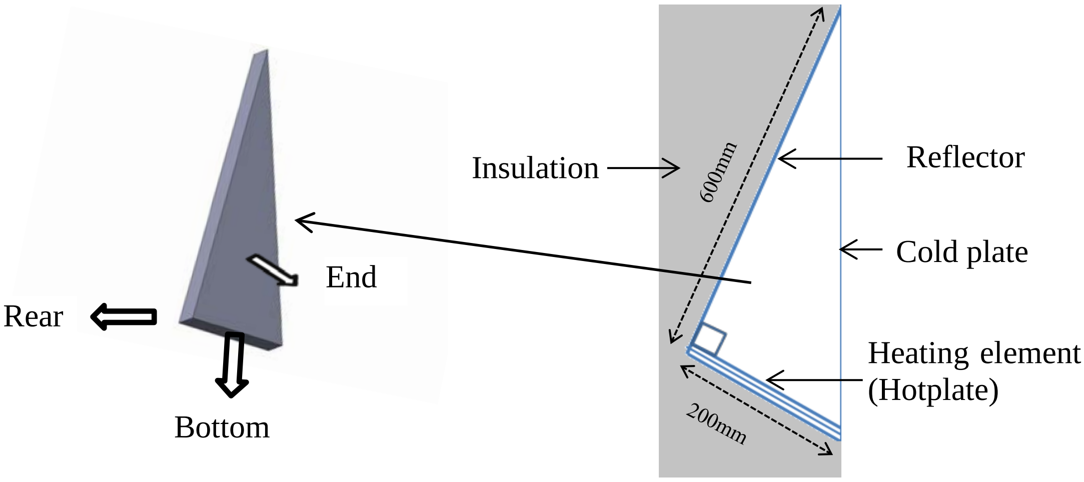

Figure 2: Module to be used for the heat transfer experiment

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 filled with 100mm thick mineral wool fibre with nominal R-value of 2.8 and enclosed by 18mm plywood. The ends were fabricated with 18mm plywood. The inclined reflector module was constructed with 2mm alumium plate to minimize thermal radiation from it (low emissivity). The vertical façade (cold plate) was built with 2mm aluminium sheet in order to keep the temperature uniform over the entire area. Furthermore, this surface was cooled by two fans to ensure the majority of the heat from the enclousure was taken away through this surface. Finally the edges of the whole enclosure were sealed with high temperature aluminium duct tape to reduce any undesirable heat losses by air leakage.

Now, 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. The schematic diagram of the experimental configuration is illustrated in Figure 3. The difference between hot and cold plate temperature (ΔT) was calculated using the mean hot and cold plate temperatures.

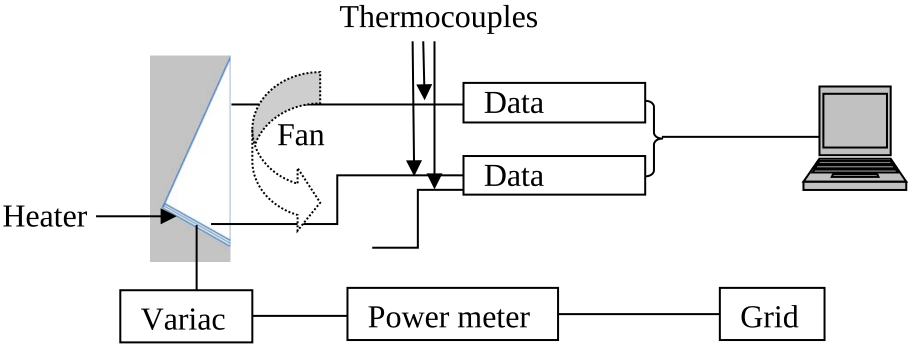

Figure 3 Schematic of the experimental setup

The series of 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 was not fluctuating more than 0.6K over a 60 minute time period.

3. Analysis

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

$$U = Q_e/(A_h\ \Delta T)\ (1)$$

This provides 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 ($Q_{convection}$) however, a complete heat balance should be undertaken. Thus, the $Q_{convection}$ can be expressed in terms of total electrical energy input and the other means of heat losses as given by Equation 2.

$$Q_{convection} = Q_e - Q_{conduction} - Q_{radiation}\ (2)$$

Here $Q_{conduction}$ 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 T_a is the ambient air temperature and in the case of rear and bottom heat losses T_i was the temperature of the hot plate. However, the end loss was calculated using the bulk temperature of the air inside the enclosure as T_i.

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

Now, the radiation heat transfer between the hot and cold plate ($Q_{radiation}$) is another portion of the thermal losses from the hotplate. However it was assumed to be negligible in this case given that both hot and cold plate were fabricated using low emissivity aluminium plate ($\varepsilon_p \approx 0.06$). Furthermore, 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.

To confirm the decision to discount the radiation from the heat balance calculation, the enclosure was assumed to be a two-surface enclosure consisting of the aluminium heater and the cold plate. Hence the expression of the $Q_{radiation}$ can be written in terms of area of the hotplate (A_h), area of the cold plate (A_c), both made up of aluminium with the emissivity of ε_p and the view factor (F) (Cengel 2007), as given by Equation 4.

$$Q_{radiation} = \frac{\sigma(T_h^4 - T_c^4)}{\left(\frac{1-\varepsilon_p}{\varepsilon_p}\right)\left(\frac{1}{A_h} + \frac{1}{A_c}\right) + \frac{1}{A_h F}}\ (4)$$

Here T_h and T_c are hot and cold plate temperatures.

The analysis found that $Q_{radiation}$ accounted for less than 4% of the heat transferred by convection and hence it is reasonable to assume it to be negligible. Based on this assumption, Equation 2 can be reduced to Equation 5 without the radiation component.

$$Q_{convection} = Q_e - Q_{conduction}\ (5)$$

By combining Equations 1 and 5, 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 ($\varDelta T$) between the hot and cold plate as given in Equation 6.

$$h_c = (Q_e - Q_{conduction})/(A_h \varDelta T) \quad (6)$$

4. Results

In order to determine the natural convection heat transfer coefficient of the proposed enclosure, the heat supplied to the hotplate was varied. 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.

Figure 5 indicates that there is an increasing trend in heat transfer coefficient as the temperature difference increases. This matches with the characteristics of the natural convection flows. As the temperature difference increases, the degree of turbulence in the fluid increases, and hence the heat transfer increases (Cengel 2007).

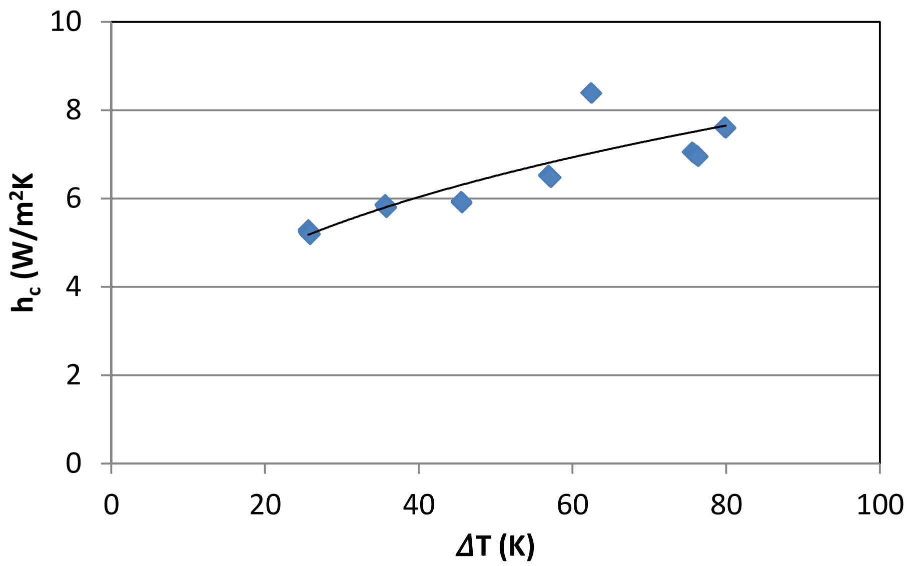

Figure 4 Temperature difference v heat transfer coefficient

Although the results given in Figure 4 relate the temperature gradient to the heat transfer coefficient, this relationship cannot be used directly to find the heat transfer coefficient in different sized enclosures. Therefore it was decided to find a relationship between the dimensionless parameters (Nusselt number and Rayleigh number) so that it can be used to obtain the heat transfer coefficient of geometrically similar enclosures of various sizes.

In achieving this, the air in the enclosure was assumed to be a real fluid and its thermophysical properties were calculated at the bulk enclosure temperature (the average temperature of hot and cold plate). These properties were then used to find the Rayleigh and Nusselt number, where the temperature change from 48°C to 112°C, corresponded a Rayleigh number change from 5 x 10^7 to 1 x 10^9 taking the characteristic length to be the height of the cold plate.

From Figure 5, it can be seen that there exists a relationship between the Nusselt number and Rayleigh number that can be expressed in the general form given in Equation 7.

$$Nu = 0.11Ra^{0.35} (7)$$

This is a typical relationship of the correlations that exist for turbulent natural convection in enclosures, with an exponential value close to 1/3 often being used in enclosures with isothermal surfaces (Cengel 2007).

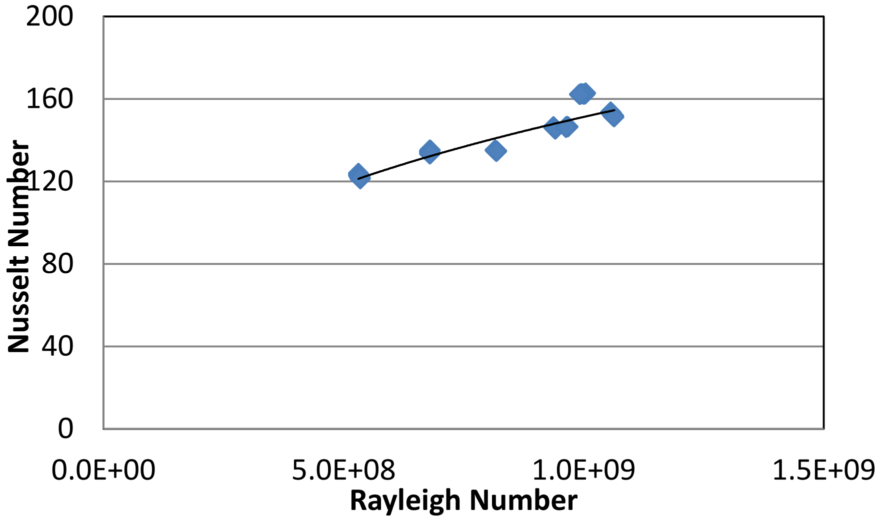

Figure 5 Rayleigh number v Nusselt number

5. Conclusion

The triangular shaped enclosure 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, in order to precisely predict the performance of the system, it is essential to understand the heat losses from it.

In light of the unavailability of the correlations relating the convection heat transfer coefficient in proposed façade enclosed system, this study shows that the natural convection heat loss can be predicted by the relationship $Nu = 0.11Ra^{0.35}$ for the Rayleigh number range of 5 x 10^7 to 1 x 10^9. Furthermore, this correlation can be applied for geometrically similar enclosures with different dimensions, given that the Rayleigh number is within the range of experimental values. The empirical equation suggests this simple relationship is a good representation of the heat transfer coefficient in the triangular enclosures and is similar to existing relationships for enclosures in general.

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