

MEASUREMENT OF NATURAL CONVECTION HEAT TRANSFER IN AN ATTIC SHAPED ENCLOSURE

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ABSTRACT

Heat transfer by natural convection in triangular enclosures is an area of significant importance in applications such as the design of greenhouses, attics and solar water heaters. However, given its significance to these areas it has not been widely examined. In this study, the natural convection heat transfer coefficients for air in an attic shaped enclosure were determined for Grashof Numbers over the range of 10^7 to 10^9 . It was found that the measured heat transfer coefficients could be predicted to within 5% by Ridouane and Campo' (2005) equation (Eqn. 1) for natural convection in a triangular enclosure previously developed for Grashof Numbers in the range 10^5 and 10^6 .

$$Nu = 0.286A^{-0.286}Gr^{1/4} \quad (1)$$

As such, it is suggested that this equation may be suitable for predicting the natural convection heat transfer coefficients in full scale attic enclosures.

INTRODUCTION

The problem of heat transfer due to natural convection in triangular enclosures is quite widespread and has application to buildings, solar collectors and greenhouses. However, unlike rectangular, square and cylindrical enclosures, the problem of natural convection heat transfer in triangular or attic-style enclosures has received relatively little attention.

It should be noted that the problem has not been completely ignored, in fact many of the studies undertaken in the area have relied upon the use of computational fluid dynamics to further the understanding the flow and heat transfer in triangular enclosures.

In their computational study Akinsete and Coleman (1982) found that for a right triangular enclosure with a cooled base, heated inclined wall and adiabatic vertical side that there was an increase in heat transfer near the apex of the heated and cooled side. In this study they observed a single convection cell as did Poulidakos and Bejan (1983a).

However, in a separate computational study Poulidakos and Bejan (1983b) found that at higher Rayleigh numbers that a Benard-type instability resulted in multiple cells forming in their enclosure. They found that the formation of these cells was related to the aspect

ratio of the right triangular enclosure, as did Asan and Namli (2000, 2001) and Aramayo et.al (2002a and b).

The flow behaviour and presence of convection cells has also been discussed in recent times for isosceles triangular enclosures, similar to those formed by pitched roofs. Holtzman et. al. (2000) showed computationally that the flow in such enclosures was asymmetric and undertook flow visualisation to validate this conclusion. A similar phenomenon was computationally observed by Ridouane et. al. (2006) and Ridouane and Campo (2007).

Given the large number of computational studies that have examined natural convection in triangular enclosures, there is a distinct lack of generalised correlations to predict the heat transfer in these spaces (Cengel 1998, Holman 1997, Incopera and DeWitt 2002). Al-Shariah and Ecevit (1992) were perhaps the first to present a truly generalised equation for heat transfer in a triangular enclosure. In their study they examined a right triangular enclosure with one heated and one cooled side and found that the heat transfer could be expressed as a function of the Rayleigh number, the enclosure aspect ratio and the inclination of the cavity to the horizontal.

For isosceles triangular enclosures, perhaps the most notable experimental work is that of Flack et. al. (1979) and Flack (1980); in these studies one side of an isosceles triangular enclosure was heated while the other was cooled and the base was treated as being adiabatic. They performed a number of experiments for varying aspect ratios and also Grashof Numbers in the range 2.9×10^6 to 9×10^6 . Using their data they subsequently developed a series of correlations for predicting heat transfer in such enclosures.

Recently, Ridouane and Campo (2005) re-examined the data from Flack's work and presented their results as a single generalised correlation expressed in terms of the enclosure Grashof number and aspect ratio. One key shortcoming of this correlation is that it is based on data from a relatively narrow range of Grashof numbers. In a recent study, Anderson et. al. (2008) proposed using the low natural convection heat transfer due to air inside an attic space as a means of passively insulating the rear surface of an uninsulated Building Integrated Photovoltaic/Thermal (BIPVT) Solar Collector. However, for large attic spaces in a tropical or warm climate such as those of Australia or New Zealand, the Grashof numbers could exceed those previously studied by several orders of magnitude, with values of up to and possibly exceeding 10^{10} .

In light of the lack of experimental data relating to higher Rayleigh and Grashof number flows, it was decided to experimentally examine the issue of natural convection in an isosceles triangular enclosure and test the validity of Ridouane and Campo's correlation over a wider range of Grashof number conditions.

EXPERIMENTAL METHOD

To determine the heat transfer by natural convection it was necessary to build an appropriate experimental apparatus. For this study, the aim was to examine the applicability of Ridouane and Campo's correlation of Flack's work at higher Grashof number conditions. As such the scale of the enclosure was increased from 152.4mm x 76.2mm x 254mm in the previous studies to an enclosure 707.1mm x 353.6mm x

1500mm as a means of changing the characteristic length and therefore the Grashof number.

In order to create a temperature gradient within the enclosure, a flexible aluminium foil resistance heater (750W nominal) was attached to a 2mm aluminium plate and used as one of the inclined sides of the enclosure. The aluminium plate, having a high thermal conductivity, was used to maximise temperature uniformity on the heated surface inside the enclosure (an examination of the temperature data found that the temperature variation was less than 5% across the surface of aluminium plate). To minimise heat loss from the rear surface of the heater, it was insulated with approximately 100mm of mineral wool fibre insulation with a nominal R-value of 2.2 and a sheet of 20mm plywood.

Similarly, the base of the enclosure was made from 100mm of the mineral wool fibre insulation between two layers of 20mm plywood. The ends were fabricated from a single layer of 20mm plywood. The second inclined surface was constructed from 2mm aluminium plate to ensure that there would be minimal thermal resistance across this surface. Additionally, the inclined aluminium plate was cooled by forced convection by a fan providing a free stream velocity of approximately 4 m/s. This step was taken to ensure that the surface would be the main source of heat loss in the enclosure. Finally, to reduce unwanted heat loss, all edges of the enclosure were sealed with high temperature duct tape to ensure that no air leakage from the enclosure could occur. The layout of the physical enclosure is illustrated in Figure 1.

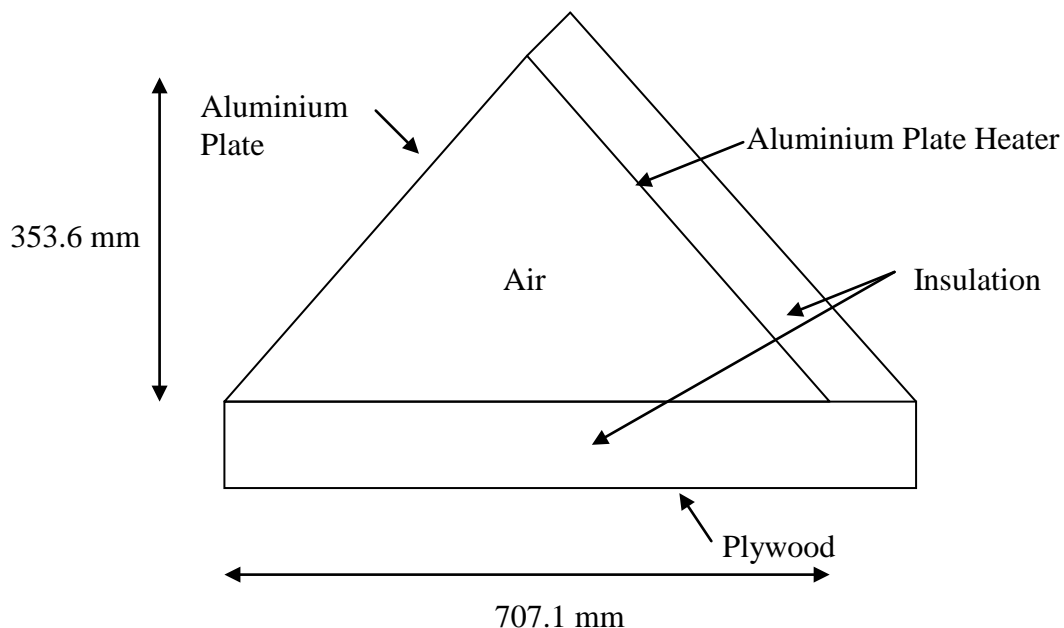


Fig. 1: Experimental enclosure

In order to vary the Grashof number conditions, it was necessary to vary the temperature difference across the enclosure. In order to achieve this, the power being supplied to the electrical resistance heater was varied using a variable power transformer (Variac) and was measured using a single phase EMU Elektronik power meter.

To determine the temperature gradient occurring with the enclosure the mean temperature of the heater was measured using a series of 6 copper-constantan (T-type)

thermocouples that were uniformly distributed over and attached to the heater surface, while a seventh thermocouple was used to measure the ambient temperature. Before undertaking the measurements these thermocouples were calibrated against a platinum resistance thermometer and were found to be accurate to within $\pm 0.3\text{K}$ across the temperature measurement range. Subsequently, the thermocouples were connected to Picolog TC-08 eight channel thermocouple DAQ system and recorded by a computer via the USB interface. The configuration of the power and temperature measurement system and instrumentation is illustrated schematically in Figure 2.

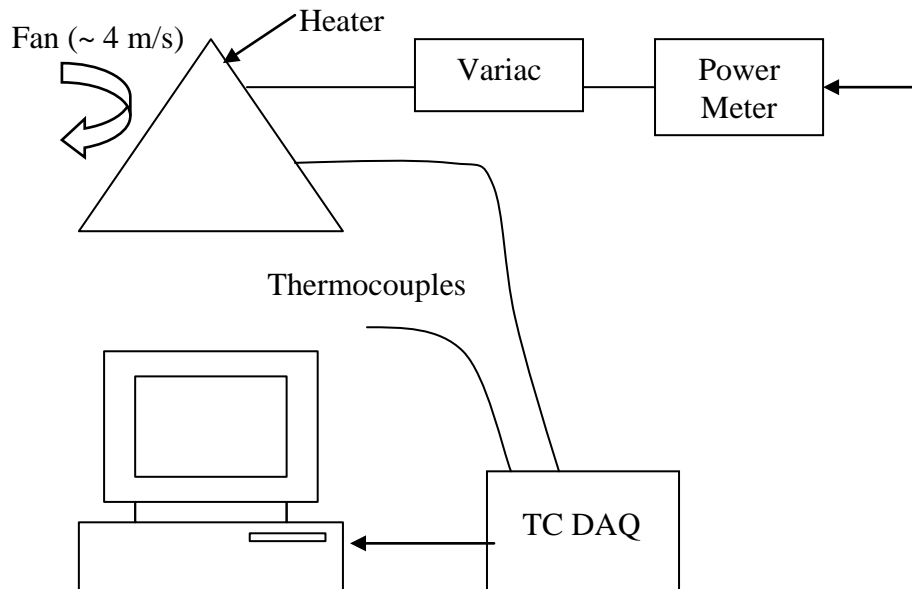


Fig. 2: Schematic of measurement system

Finally, to accurately determine the heat transfer coefficient within the enclosure it was necessary to allow the system to reach steady state conditions. To do this, the heater power was set and the ambient and heater temperatures were monitored. When the variation of the temperature difference between the heater and the ambient was not more than 0.6K over a 30 minute period, it was assumed that the system had reached a steady state, as shown in Figure 3. Subsequently, the readings taken during this period were used to determine the natural convection heat transfer coefficient.

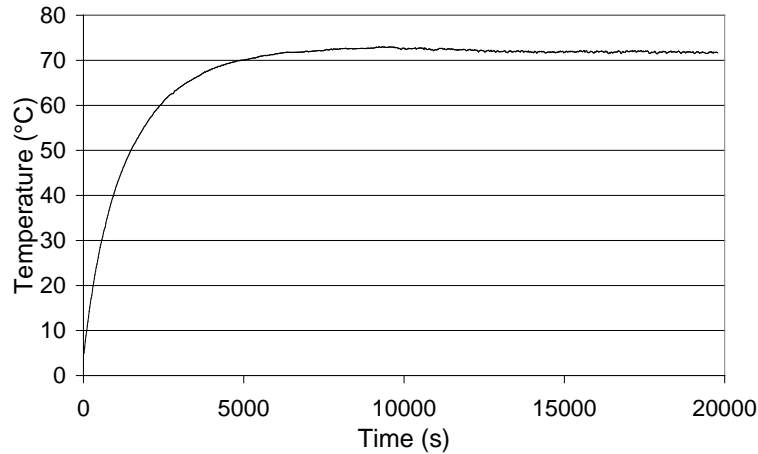


Fig. 3: Temperature v time showing steady state

ANALYSIS

From the experimental measurements it is relatively straight-forward to determine the overall heat loss coefficient for the experimental rig. Under steady state conditions the rig heat loss coefficient (U) is represented by a function of the input electrical power (Q_e) the difference in temperature between the heaters mean (T_p) and the ambient temperature (T_a) and the heater area (A_p), as shown in Eqn. 1.

$$U = \frac{Q_e}{A_p(T_p - T_a)} \quad (1)$$

Although this provides a crude estimate of the heat transfer coefficient inside the enclosure, it is necessary to undertake a full heat balance to identify the heat that is being transferred by convection. For this study the heat balance could be given by Eqn. 2: where Q_e is the electrical input power, $Q_{end,cond}$ is the heat loss through the ends of the enclosure by conduction based on the mean enclosure temp (T_e , taken as the average of T_p and T_a), $Q_{back,cond}$ is the heat loss from the heater by conduction through the insulation on its rear surface, $Q_{base,cond}$ is the heat loss through the base by conduction based on the mean enclosure temp and Q_{rad} is the radiation heat loss.

$$Q_e - Q_{end,cond} - Q_{back,cond} - Q_{base,cond} - Q_{rad} = Q_{convection} \quad (2)$$

The remaining term, $Q_{convection}$ is the heat transferred from the heated to cooled inclined wall by convection. In actuality this term is the sum of the thermal resistance due to natural convection, the inclined aluminium plate wall and the external forced convection. However, the thermal resistance through the aluminium wall and due to external forced convection are relatively small and as such these terms were neglected (testing showed that neglecting these terms resulted in less than a %3 increase in the internal heat transfer coefficient).

The calculation of heat loss by conduction (Q_{cond}) through a wall is a relatively simple application of Fourier's law (Eqn. 3) over the area (A) of the wall of interest. Where k is the thermal conductivity of the wall and L is the thickness of the wall or alternatively R (in m^2K/W) is the thermal resistance of the wall.

$$Q_{cond} = \frac{kA(T_e - T_a)}{L} = \frac{A(T_e - T_a)}{R} \quad (3)$$

The radiation heat transfer in an enclosure of this nature is somewhat more complicated. However it was recognised that for this case, heat transfer by radiation between the heated and cooled sides is effectively negligible, given that both inclined sides were made from aluminium plate ($\varepsilon_p \approx 0.06$). Furthermore, the base of the enclosure is essentially adiabatic and therefore could be discounted from any radiation calculations. Finally the view factor from the heated surface to the end walls for a cavity of this size are relatively small (Siegel and Howell, 2002) as is their area, thereby reducing the potential radiation heat transfer.

Therefore, to verify that radiation could be neglected, it was assumed that radiation heat transfer would only occur between the wooden ends and base and the aluminium heater and could be treated as being analogous to having two parallel plates, essentially a “worst case” scenario. Under these conditions it was assumed that the equation for the radiation heat transfer could be simplified to Eqn. 3 for a two surface enclosure where the two surfaces are: the heater of area (A_p) made of aluminium with emissivity ε_p , and the other made of wood with emissivity ε_w and an area (A_w) equal to the sum of the areas of the ends and base of the enclosure and where the view factor (F) between the plates is equal to unity (Cengel 1998).

$$Q_{rad} = \frac{\sigma(T_p^4 - T_e^4)}{\frac{1 - \varepsilon_p}{A_p \varepsilon_p} + \frac{1 - \varepsilon_w}{A_w \varepsilon_w} + \frac{1}{A_p F}} \quad (3)$$

From the analysis of Q_{rad} it was found that, based on the assumptions listed above, that radiation heat transfer accounted for approximately 5% of the heat being transferred and so could be assumed to be negligible. If however the term were to be included, it would increase the heat transfer coefficient by approximately 5%, although this would be reduced by 3% if the forced convection term discussed earlier were also included.

Based on the energy balance it is then possible to determine the heat transfer by convection between the heated and cooled surfaces. Subsequently the heat transfer coefficient by natural convection (h_c) can be determined from Eqn. 4.

$$h_c = \frac{Q_{convection}}{A_h(T_p - T_a)} \quad (4)$$

RESULTS

Having established an appropriate experimental methodology and means of analysing the natural convection heat transfer in the enclosure a number of experiments were undertaken to determine the heat transfer coefficient.

As mentioned previously, the power to the heater was varied in order to change the temperature gradient. In this study the power was varied such that the mean heater temperatures were between 30°C and 120°C under steady state conditions. By recording

this period of steady state data it was then possible to determine the heat transfer coefficient. For this study, the range of plate temperatures equated to Grashof Numbers in the range of 10^7 to 10^9 , where the characteristic length was taken to be the vertical height of the enclosure and the physical properties were taken at the mean enclosure temperature.

In Figure 4, it can be seen that with an increasing temperature gradient in the enclosure, that there is an increase in the natural convection heat transfer coefficient. This is characteristic of natural convection flows, as at higher temperature gradients there is a higher degree of turbulence and therefore heat transfer.

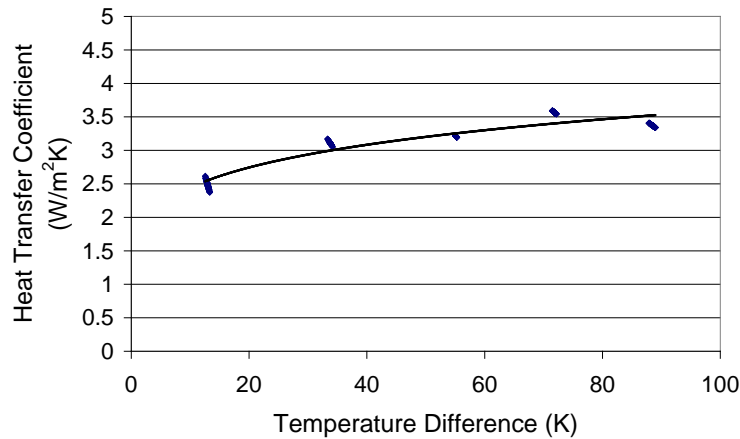


Fig. 4: Heat transfer coefficient v enclosure temperature difference

Although this demonstrates that the heat transfer coefficient increases for increasing temperature gradient it is not a dimensionless correlation and so cannot be readily translated to enclosures of a different size. This is readily overcome by translating the data into a non-dimensional form. In Figure 5 it can be seen that there exists a relationship between the Nusselt number and Rayleigh number that can be represented in the general form of Eqn. 5 and more specifically as shown in Eqn. 6.

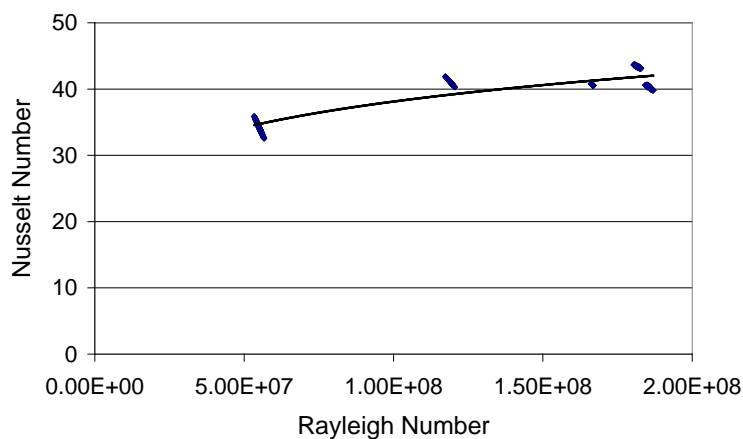


Fig. 5: Nusselt number v Rayleigh number

$$Nu = cRa^n \quad (5)$$

$$Nu = 1.33Ra^{0.2} \quad (6)$$

This is typical of the relationship that exists for turbulent natural convection where an exponent value of $n = 0.2$ would commonly be used for surfaces of constant heat flux, as was the case here (Bejan, 2004).

A key shortcoming of this correlation however, is that it is based on a relatively narrow range of Grashof (or Rayleigh) numbers and does not account for aspect ratio. As such, it was decided to examine how well an existing correlation (Eqn. 7) from Ridouane and Campo (2005), and that accounts for aspect ratio, could predict the heat transfer in the experimental enclosure. A similar shortcoming of Ridouane and Campo's work was that it too was based on a narrow range of Grashof numbers.

$$Nu = 0.286A^{-0.286}Gr^{1/4} \quad (7)$$

In Figure 6 it can be seen that although Eqn. 7 was developed for Grashof Numbers in the range 2.9×10^6 to 9×10^6 it is actually able to provide a prediction to within 5% of the values achieved in the experiments conducted in this study, where the Grashof Numbers were in the range of 10^7 to 10^9 . This suggests that the correlation is well suited to use over a much wider range of conditions and is a suitable generalised representation of heat transfer in a triangular enclosure.

Furthermore, in Figure 6 it can be seen that there is a good relationship between the predicted values from this correlation, as well as the CFD results from Anderson et. al. (2008), and the experimental data presented in this work.

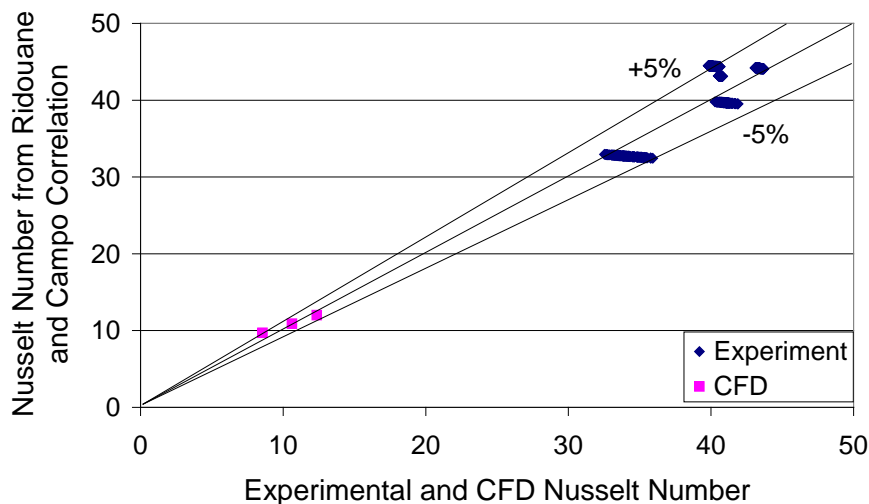


Fig. 6: Relationship between Nusselt number from Ridouane and Campo's correlation versus CFD prediction and measured Nusselt number

CONCLUSION

Recently Anderson et. al. (2008) suggested that natural convection in the attic of a cold roof building could serve as an insulating medium for BIPVT collectors installed in a cold roof building. Ridouane and Campo (2005) had previously developed a correlation to describe natural convection in attic style enclosures based on Grashof Numbers in the range 2.9×10^6 to 9×10^6 . However for a tropical or warm climate such as those of Australia or New Zealand, the Grashof numbers in an attic space could conceivably exceed those previously studied by several orders of magnitude.

In light of the shortcomings of the data relating to natural convection in attic shaped enclosures, the experimental work in this study showed that Ridouane and Campo' correlation was also valid for Grashof Numbers in the range of 10^7 to 10^9 . Based on this finding it is therefore possible to utilise this correlation in determining the heat loss from the uninsulated rear surface of a BIPVT collector in a cold roof building.

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Tim Anderson studied Mechanical Engineering as an undergraduate and postgraduate research student at the University of New South Wales. He is currently completing his PhD at The University of Waikato for a thesis entitled "Thermal Aspects of Building Integrated Photovoltaic/Thermal Solar Collectors".