

# PERFORMANCE OF AN UNGLAZED SOLAR COLLECTOR FOR RADIANT COOLING

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## ABSTRACT

Under night conditions, when there is no solar radiation and the sky temperature is low, unglazed solar collectors can radiate heat to the sky thus cooling a storage tank to provide cooling the following day. This study theoretically and experimentally examines the performance of an unglazed solar collector for cooling. It shows that such systems can provide a cooling capacity in the order of  $50\text{W/m}^2$  and are able to cool to well below the ambient temperatures experienced during the cooling season. Finally it explores the contribution such a system could make to cooling loads in typical New Zealand and Australian buildings.

## 1. INTRODUCTION

Unglazed solar collectors have long been relegated to providing low cost heating in applications such as swimming pool heating systems. This is due to their high heat loss: firstly by convection to the surrounding air and secondly by radiant heat transfer to the atmosphere. The magnitude of these losses leads to them having a very limited heating temperature range. However, it is possible to take advantage of the radiant cooling of unglazed solar collectors by operating them at night. Moreover, by integrating such devices into roofing materials it is possible to achieve a multifunctional heating, cooling and roofing solution.

The integration of solar devices into roofs has a number of advantages over conventional ‘bolt on’ systems; in particular the installation of both the roof and solar panel occur at the same time meaning reduced labour and reduced material cost. Despite the potential benefits the use of water cooled solar collectors as building elements has, until recently, been largely ignored.

Many systems essentially integrate panels onto a building rather than into the building. A case in point being Kang et al. [1] who discussed the performance of a roof integrated solar collector which utilised an array of “standalone” solar water heaters as a roof. This system although integrated onto the building was not integrated into the building. Probst and Roecker [2] found this method of integrating solar collectors to be “acceptable” to architects. However they emphasise that in the future, building integrated solar collectors “should be conceived as part of a construction system” and although somewhat self-evident their findings appear to have been overlooked by the research community. Medved et al. [3] however examined a large area unglazed solar thermal system that could be truly integrated into a building. In their system they utilised a standard metal roofing system as a solar collector for a swimming pool heating system. They found that they were able to achieve payback periods of less than 2 years, a reduction of 75% in the time taken to pay for a glazed solar collector system.

In New Zealand and Australia long run metal roofing is widely used for domestic, commercial and industrial applications. Moreover, its low cost, high durability, aesthetics and relatively good thermal conductivity make the material well suited to use as the basis for a large area solar heating system. However for many of the warmer areas of Australia, demand for cooling can be more significant than for heating [4], meaning that the use of large area solar collectors solely for heating water is of little value in standalone houses. As such, there has been significant attention given to the development of solar cooling systems particularly those using absorption and adsorption cycles

[5,6]. The down side of such systems however, is that they require large amounts of high grade heat, which results in the need for large (and expensive) collector areas.

Previously, the authors have shown that long run metal roofing can be used for heating over a wide temperature range, from low temperature pool heating [7] through to domestic water heating [8]. However, it has been noted that these collectors have relatively high overall heat loss coefficients, part of which can be attributed to the high emissivity of the polyester paint surface (Colorcote®).

This high emissivity is undesirable for heating systems due to the increased radiant heat loss; however it could be beneficial if heat is to be “dumped” from the collectors to provide cooling. Systems such as this are often referred to as radiant cooling systems or night/sky radiators [9-14] and rely on heat being lost from a panel by radiant heat transfer to the “sky”. Hence, under night conditions, when there is no solar radiation and the sky temperature is low, the collector can radiate heat to the sky and cool a storage tank to provide cooling in the building the following day.

## 2. THEORETICAL ASSESSMENT OF AN UNGLAZED SOLAR COLLECTOR

For an unglazed solar collector the useful heat gain ( $Q$ ) can be represented as a function of the collector area ( $A$ ), the heat removal efficiency factor ( $F_R$ ), the absorptance of the collector ( $\alpha$ ), the solar radiation ( $G''$ ), the collector heat loss coefficient ( $U_L$ ) and the temperature difference between the collector inlet temperature ( $T_{in}$ ) and the ambient temperature ( $T_a$ ) as shown in Equation 1 [15].

$$Q = AF_R [G''(\alpha) - U_L(T_{in} - T_a)] \quad (1)$$

The heat removal factor ( $F_R$ ) can be derived from Equation 2, which accounts for the mass flow rate in the collector ( $\dot{m}$ ) and the specific heat of the collector fluid ( $C_p$ ).

$$F_R = \frac{\dot{m}C_p}{AU_L} \left[ 1 - e^{-\frac{AU_L F'}{\dot{m}C_p}} \right] \quad (2)$$

To determine the heat removal factor it is necessary to calculate a value for the corrected fin efficiency ( $F$ ) (Equation 3) taking into account the influence of the tube pitch ( $W$ ) and the tube width ( $d$ ). Furthermore, the coefficient ( $M$ ) accounts for the thermal conductivity of the absorber (Equation 4).

$$F = \frac{\tanh\left(M \frac{W-d}{2}\right)}{\left(M \frac{W-d}{2}\right)} \quad (3)$$

$$M = \sqrt{\frac{U_L}{K_{abs} L_{abs}}} \quad (4)$$

The collector efficiency factor ( $F'$ ) can be calculated using Equation 5 where the overall heat loss coefficient ( $U_L$ ) of the collector is the summation of the collector's edge, bottom and top losses. The bottom loss coefficient is given by the inverse of the insulations R-value (i.e.  $K_b/L_b$ ) and the edge losses are represented by Equation 6, where  $p$  is the collector perimeter and  $t$  is the absorber thickness.

$$F' = \frac{\frac{1}{U_L}}{W \left[ \frac{1}{U_L(d + (W - d)F)} \right] + \frac{1}{\pi d h_{fluid}}} \quad (5)$$

$$U_{edge} = \frac{K_{edge} p t}{L_{edge} A_{collector}} \quad (6)$$

For unglazed collectors the top loss coefficient is a function of radiation, natural convection and forced convection (wind). Hence it is necessary to calculate the top loss coefficient ( $U_{top}$ ) by taking the summation of the individual contributions.

The forced convection heat transfer coefficient ( $h_w$ ) can be calculated using a correlation in terms of wind velocity ( $v$ ), as shown in Equation 7 [16], while the natural convection loss ( $h_{nat}$ ) can be represented by a function of the temperature difference between the mean collector plate temperature ( $T_{pm}$ ) and the ambient temperature ( $T_a$ ) as shown in Equation 8 [17].

$$h_w = 2.8 + 3.0v \quad (7)$$

$$h_{nat} = 1.78(T_{pm} - T_a)^{1/3} \quad (8)$$

By combining both forced and natural convection heat transfer it is possible to determine an overall convection heat transfer coefficient ( $h_c$ ) as shown in Equation 9 [17]. Taking the sum of both the convection and radiation losses, it is possible to determine the overall top loss heat transfer coefficient ( $U_{top}$ ) for the unglazed collector and subsequently the overall heat loss coefficient ( $U_L$ ).

$$h_c = \sqrt[3]{h_w^3 + h_{nat}^3} \quad (9)$$

The heat loss due to radiation can be expressed as a radiation heat transfer coefficient in terms of the sky temperature ( $T_s$ ), the mean collector plate temperature ( $T_{pm}$ ) and the plate emissivity ( $\varepsilon_p$ ) as shown in Equation 10.

$$h_r = \sigma \varepsilon_p (T_{pm}^2 + T_s^2)(T_{pm} + T_s) \quad (10)$$

The sky temperature can be calculated from Equation 11, that has been shown to work well across a wide range of climate conditions [18, 19], where the emissivity of the sky ( $\varepsilon_s$ ) is dependent on the dew point temperature ( $T_{dp}$ ) and the fraction of cloud cover ( $n$ ) (between 0 and 1) (Equation 12).

$$T_s = \varepsilon_s^{0.25} T_a \quad (11)$$

$$\varepsilon_s = \left( 0.711 + 0.56(T_{dp}/100) + 0.73(T_{dp}/100)^2 \right) \times \left( 1 + 0.0224n - 0.0035n^2 + 0.00028n^3 \right) \quad (12)$$

Finally the mean plate temperature is determined from Equation 13.

$$T_{pm} = T_{in} + \frac{Q/A_{collector}}{F_R U_L} (1 - F_R) \quad (13)$$

Now in the absence of solar radiation the collector acts as a radiator and the heat transferred from the absorber ( $S$ ) is given in terms of a linearised form of the Stefan-Boltzman law, as shown in Equation 14 [20].

$$S = 4\varepsilon_p \sigma T_a^3 (T_{pm} - T_s) \quad (14)$$

where the heat loss is a function of the sky temperature ( $T_s$ ), the mean collector plate temperature ( $T_{pm}$ ), the plate emissivity ( $\epsilon_p$ ) and the ambient temperature ( $T_a$ ). By utilising this knowledge it is then possible to calculate the useful heat gain or loss from the absorber.

### 3. EXPERIMENTAL TESTING AND MODEL VALIDATION

The system developed in this study is designed to be directly integrated into a troughed sheet metal roof. During the manufacturing process in addition to the normal troughed shape, channels are added to the trough for the thermal cooling medium to travel through. An absorber sheet, analogous to the fin of a finned tube absorber, is bonded into the trough. The channels formed in the trough are thus enclosed by this sheet; forming a riser tube having an inlet and outlet at opposite ends of the trough to which heat can be transferred (Figure 1). Water is pumped through a manifold (header tube), through these “riser” tubes and out through a manifold (header tube) before being fed to a heat exchanger that removes the heat from the fluid.

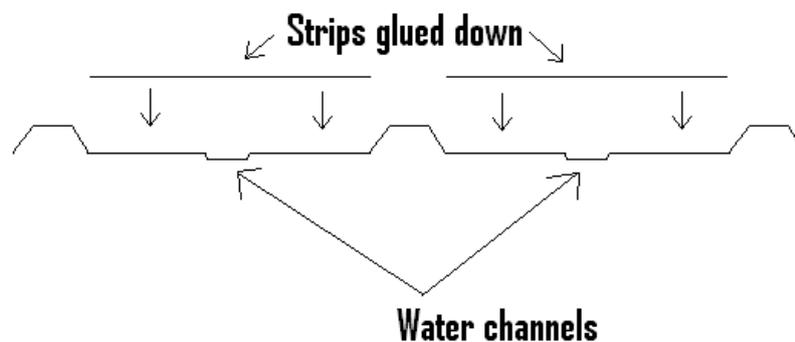


Figure 1: Schematic cross section of panel

Now, in the previous section the theoretical performance of the radiant cooling system was discussed. However, for the design methodology to be confirmed it was necessary to validate the model against experimental data. To achieve this validation, an outdoor thermal test system similar to that recommended in AS/NZS 2535.1 [21] was used. T-type thermocouples were used to measure the temperature of the fluid entering and exiting the panels and the temperature of the panels themselves (details of the panel are shown in Table 1). A pyranometer mounted in plane with the collector surface monitored the global solar radiation and a flow meter at the inlet was used to determine the flow rate in the collector. Wind speed over the collector was taken from a nearby weather monitoring station, as were the ambient temperature, dew point temperature and cloud cover. Additionally the rear surface of the collector panel was insulated with mineral fibre insulation held in place with plywood.

Table 1: Design parameters for building integrated collector

Parameter	Symbol	Value	Unit
Emittance of plate	$\epsilon_p$	0.95	
System flow rate	$m$	340	l/h
Collector Area	$A$	6.3	m <sup>2</sup>
Absorber thickness	$t$	0.5	mm
Tube Hydraulic Diameter	$d_h$	8.5	mm
Tube Spacing	$W$	0.22	m
Insulation Conductivity	$k$	0.045	W/mK
Back Insulation Thickness	$L_b$	0.1	m
Absorber Conductivity	$k_{abs}$	50	W/mK

The collectors were coupled to a 60-litre storage cylinder that was heated during the day using energy collected by the absorber, then cooled at night by radiation. In Figure 3 it can be seen that during the morning, the tank temperature increases due to the increasing solar radiation. However, in the afternoon, the decreasing levels of solar radiation lead to more heat being rejected by the collector to the atmosphere than is gathered, and so a decrease in the temperature below the ambient temperature.

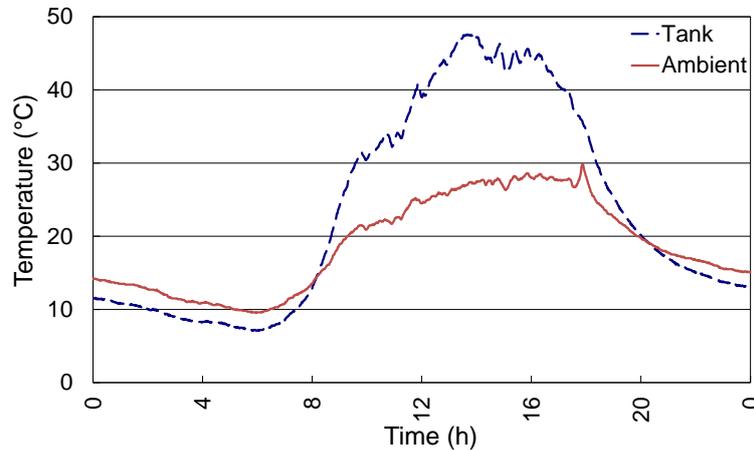


Figure 2: Tank water temperature

This can be better understood through examination of Figure 4, where it can be seen that early and late in the day there is a net heat loss from the collector of approximately  $50 \text{ W/m}^2$  due to radiant heat loss.

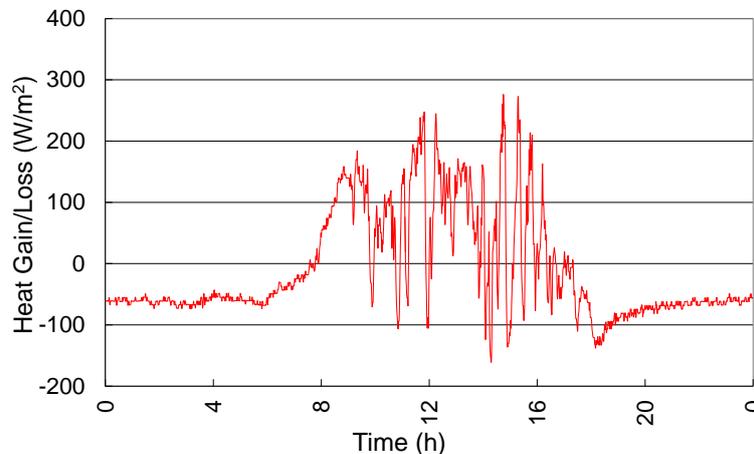


Figure 3: Heat transfer to and from the collector

In the absence of solar radiation, heat loss is principally due to the radiation heat transfer from the absorber to the cold sky as discussed in the theoretical analysis. This is best illustrated by determining the heat loss due to radiation. In Figure 5, it can be seen that there is very good agreement between the predicted and the calculated values for the collector radiant heat loss.

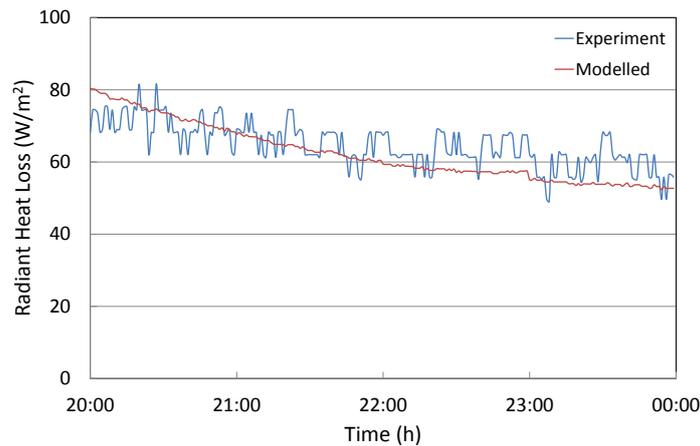


Figure 4: Cooling provided by radiant cooling panel

#### 4. DISCUSSION

From the proceeding description it can be seen that there is significant potential for unglazed solar collectors to act as a radiant cooling system at night. In doing so, it was found that when operated in the absence of solar radiation, the collector was able to achieve cooling in the order of  $50\text{W/m}^2$  with temperatures of approximately  $10^\circ\text{C}$  being observed in the storage tank. This would appear to be well suited for moderate cooling loads. Further, given the flexibility of the system to be used as a roofing structure, there is significant potential to use large areas of the collector to operate in heating mode during daytime hours, with storage in a “hot” cylinder and subsequently at night with storage in a “cold” cylinder. This cold water could subsequently be used either directly, for example, in radiant cooling panels in the building, in cooling coils for ducted air-conditioning, to provide chilled water to an industrial process, or indirectly, as a sink for the condenser of a vapour compression air conditioning system, thus allowing it to achieve higher COP's.

In considering the use of such systems however, it is important to understand the climatic conditions where a system might be implemented. For this study four cities were chosen for examination, Auckland, Sydney, Brisbane and Perth. To gauge the suitability of each location, histograms showing the frequency of sky temperature depression were prepared from typical and representative meteorological year data. In Figure 6 it can be seen that for times when there is zero solar radiation, the sky temperature is typically 15-20 degrees below ambient for all locations. It is also interesting to note that Perth has a relatively high number of hours with large sky temperature depressions, suggesting it may be particularly well suited to the use of radiant cooling.

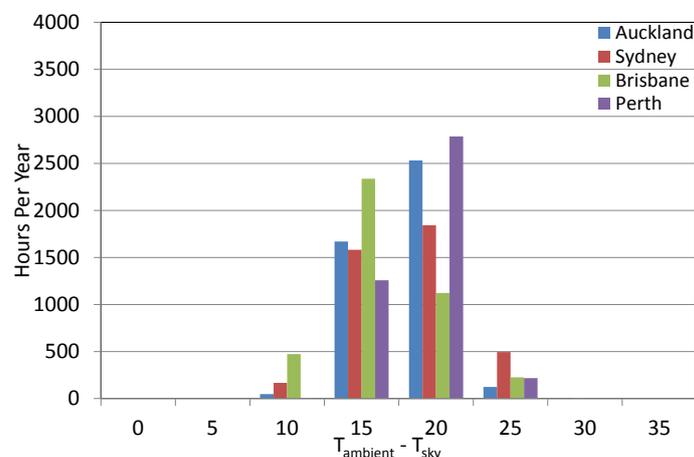


Figure 5: Frequency of sky temperature depressions

Using sky temperature depression as the sole measure of the suitability of a location for radiant cooling though has some shortcomings, in that it ignores the influence of convective heat transfer that can also influence the performance of radiant cooling systems. As such it was decided to examine the frequency at which the stagnation temperature ( $T_{stag}$ ) (Equation 15) [10] of the radiant panels was below the ambient temperature, in the absence of solar radiation.

$$T_{stag} = T_a - \frac{\varepsilon_p \sigma (T_a^4 - \varepsilon_s T_a^4)}{h_c + 4\varepsilon_p \sigma T_a^3} \quad (15)$$

In Figure 7 it can be seen that Perth does appear to be well suited to radiant cooling systems, as does Sydney. In contrast both Auckland and Brisbane would appear to be less well suited, possibly due to higher levels of cloud cover, higher humidity and dew point temperatures.

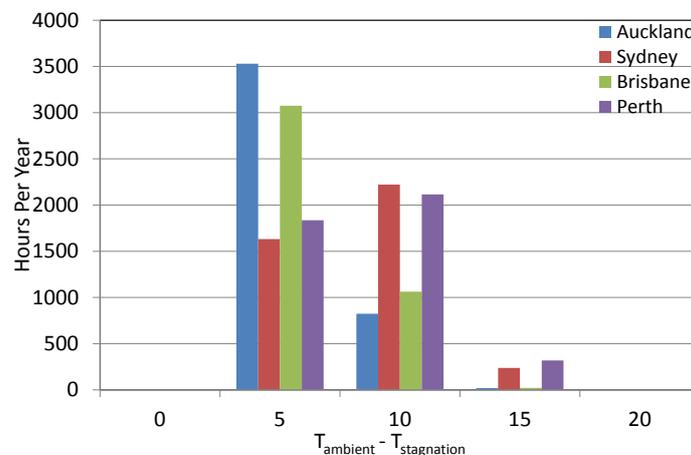


Figure 6: Frequency of sky temperature depressions

On this basis consider a simple case where the radiant cooling panel has an area of  $100\text{m}^2$  and is used for cooling over an entire year, averaging a radiant heat loss of  $50\text{W}/\text{m}^2$ . In Sydney and Perth such a system may be able to be used for approximately 2000h annually; this would lead to cooling of approximately 10000kWh/annum. Though simplistic, it serves to illustrate the point that radiant cooling systems represent a relatively untapped means of energy savings in the built environment.

## 5. CONCLUSION

Unglazed solar collectors have tended to be relegated to applications such as swimming pool heating systems due to their low temperature range. This limited temperature range is due to heat loss: firstly by convection to the surrounding air and secondly by radiant heat transfer to the cold sky. This study showed that under night conditions when there is no solar radiation and the sky temperature is low, the collector can be used to radiate heat to the sky and cool a cold storage tank to provide cooling in the building the following day.

In this regard there is scope to increase the use of such systems, by integrating them into low cost building materials thereby delivering (large area) low cost cooling (and heating) systems for buildings. There would appear to be significant scope for this, particularly in Australian locations, such as Sydney and Perth.

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