Thermal modelling of a night sky radiation cooling system

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Abstract
The thermal modelling of a night sky radiation cooling system suitable for a room situated in the Namib Desert at Gobabeb, Namibia, is considered in this paper. The system consists of the following components: radiator panels, a single water storage tank, room air-to-water natural convection heat exchangers or convectors, circulating pump(s), interconnecting pipe work and temperature sensors and controls. The mathematical equations describing the thermal behaviour of the various system components are given. These equations are solved using an Excel spreadsheet and the hourly panel surface, water storage tank and room temperatures are calculated for a given internally generated heat load and weather pattern. Given the maximum allowable room temperature, the sizes of the system components may be calculated. The results obtained compared favourably with values reported in the literature. It is thus concluded that the thermal model presented can be used with confidence as a design tool for the sizing of a night sky radiation cooling system.

Keywords: sky radiation cooling, night sky radiation cooling, radiation cooling system, thermal modelling

1. Introduction

When a surface on the earth faces the night sky, it loses heat by radiation to the sky, and gains heat from the surrounding air by convection. If the surface is a relatively good emitter of radiation, it will tend to radiate more heat to the sky at night than it gains from the air. The net result is that the surface temperature drops to below that of the air. This phenomenon is termed night sky radiation cooling. This is a well-known phenomenon and has been known since ancient times and has been observed and referred to by many authors (Armenta-Déu, et al., 2003). Despite its many potential applications, including storage of food, seed and medicine, air-conditioning of buildings and water desalination (Dobson, et al., 2003), its commercial exploitation is still largely untapped.

It is suggested in the literature (Etzion, et al., 1997) that rather than investing in non-renewable energy to counteract the natural environmental conditions, it is often possible harnessing natural energies by exploiting the local climate by adapting the architectural design of the buildings. In this climatically adaptive approach, a variety of technologies may be integrated in the architectural design of a building (as has been done at the J Blaustein International Centre for Desert Studies, Sede-Boqer Campus, Ben Gurion University of the Negev) to provide thermal comfort with a minimum expenditure of energy. A number of passive cooling techniques can be used, and in some cases, eliminate mechanical air conditioning requirements in areas where cooling is a dominant problem (Cooling Design Booklet, 2004). The cost and energy effectiveness of these passive options are worth considering by homeowners and builders.

The heating performance of a radiation cooling system capable of a mean nightly cooling rate of 80 W/m^2 over an 8-hour operating period was experimentally investigated by Erell and Etzion (1996). The system made use of a single 2.2 x 1.3 m commercially available flat plate solar water heater, but with the glazing removed and a water pond as part of the roof for thermal storage. A roof pond type of radiation cooler has also been built at Ha Makoe, Lesotho, by the Bethel Business and Community Centre. This system consists of plastic bags filled with water on a roof. The water filled plastic bags are covered by a roof in the daytime and, at night times, is moved away (on rollers) to expose the water to the night-sky.

Another radiation cooling system has been theoretically and experimentally investigated by Al-nimr, et al., (1998). The radiating panel was made of 1 500 x 400 mm mild steel plates, with a 40 µm polyethylene cover and rockwool back-insulation and a pump circulating water to a 120-litre water storage tank. Radiation cooling systems of some 9 other investigations were also (qualitatively) summarized. The results showed that the radiation panel was able to reduce the temperature of the water by 15 °C under spring weather conditions in

Irbid, Jordan. An acceptably good quantitative agreement between the theoretical and experimental results was claimed.

2. Objective

Arguably one of the most famous of all desert research stations is the Namibian Desert Ecological Research Unit. It is situated along-side the Kuiseb's dry riverbed some 80 km southeast from Walvis Bay in the Namib Desert at a place called Gobabeb. As part of a general review and rationalisation of the Unit's energy and cooling requirements, it was decided to consider nighttime radiation cooling. In particular, it was decided to consider a workroom utilised by the research scientists and in which a number of desktop computers are operated. This room (built in 1971) has an iron roof, it has 50 mm prefabricated walls, a 5.2 x 7.9 m floor area and a 2.27 m high ceiling.

Although radiation cooling, as stated in the Introduction, is a well-known phenomenon, no suitable information was found in the published literature on which to base the thermal design calculations. The basic objective in this paper is thus to remedy this deficiency by presenting the thermal model that was used to quantitatively validate the designed night sky cooling system. The basic equations used to model and quantitatively size the room cooling system are given (in Section 3). The system of equations is solved using an Excel spreadsheet (as discussed in Section 4). An analysis of a possible system is undertaken for a given set of weather conditions (in Section 5). A sensitivity analyses is also undertaken to assess the relative importance of the various design variables. Conclusions based on the study and analysis are then drawn and discussed in Section 6.

3. Thermal model

The important components, variables and parameters of the cooling system modelled are given in Figure 1. Water from the storage tank is circulated through the radiator panel at night where, provided the environmental conditions are right, the water-cools. The cooled water stored is in turn available to cool the air in the room, when the room heats up during the day, by circulating the cold water through the convector in the room. The cooling system thus consists of a number of subsystems: the radiator panel, radiator panel-circulating pump, cold water storage tank, room-convector, convector-circulating pump, the interconnecting pipe work and the pump temperature sensors and control system. Each of these subsystems needs to be carefully designed and sized if the cooling system is to function efficiently and effectively.

3.1 Radiator panel

In many respects, the radiator panel is much like a conventional solar water heating panel except that its surface must have a high emittance and high reflectivity, and it does not have a glass cover. Thermally the panel may be considered as a single control volume of negligible thermal capacitance as shown in Figure 2. The panel's radiating surface may be characterised by its emissivity ε_s its projected area normal to the sky A_s and an efficiency η_s. If the water is in complete contact with the surface area A_s then $\eta_s = 1$, if the water is channelled beneath the absorber in pipes much like the absorber of a tube-on-plate solar water heater absorber, then $\eta_s < 1$ and η_s can be calculated using fin theory (Mills, 2000). The underside on the panel should be insulated (with equivalent thermal resistance L_s/k_sA_s) to reduce convective heat trans-

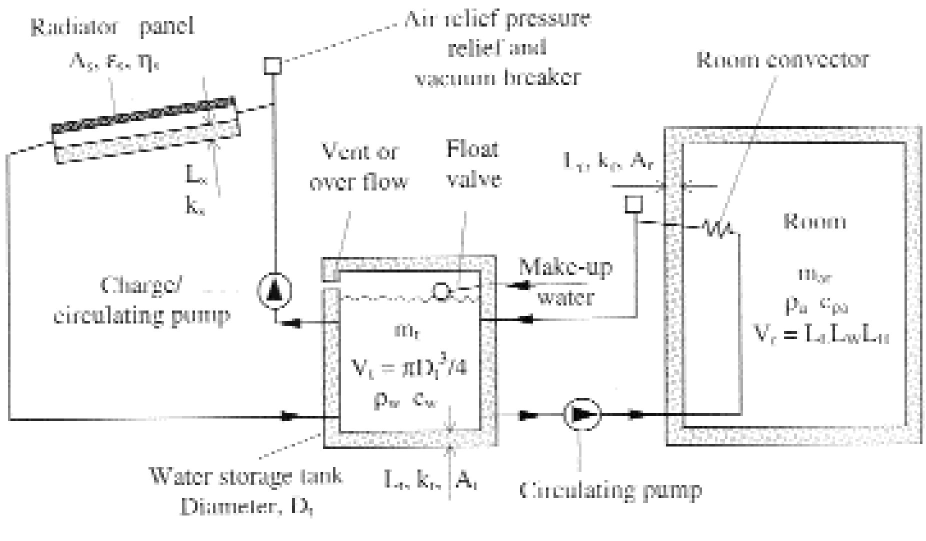

Figure 1: Thermal model of the night sky radiation cooling system showing the important components, variables and parameters

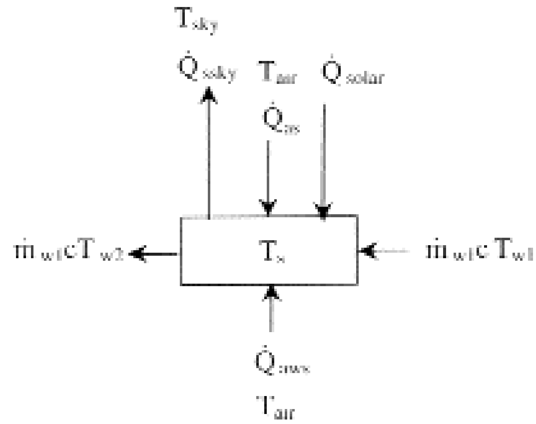

Figure 2: Control volume for the radiator panel

fer from the ambient air to the colder panel.

Referring to Figure 2, the heat transfer rates may be defined as follows:
The heat transfer rate from the radiating surface to the sky is

$$\dot{Q}_{ssky} = \left(T_s - T_{sky}\right)/R_{ssky} \quad (1)$$

Where

$$R_{ssky} = 1/h_{ssky}A_s$$

$$h_{ssky} = \varepsilon_s \sigma\left[\left(T_s + 273.15\right)^2 + \left(T_{sky} + 273.15\right)^2\right]\left[\left(T_s + 273.15\right) + \left(T_{sky} + 273.15\right)\right]$$

$$T_{sky} = \left[\varepsilon_{sky}\left(T_a + 273.15\right)^4\right]^{1/4} - 273.15$$

$\varepsilon_{sky} = 0.741 + 0.00162 T_{dp}$ at nighttime and $\varepsilon_{sky} = 0.727 + 0.00160 T_{dp}$ during the day (Mills, 2000).

Convective heat transfer between the air and the panel surface exposed to the air is

$$\dot{Q}_{as} = \left(T_a - T_s\right)/R_{as} \quad (2)$$

Where $R_{as} = 1/h_{as}A_s$ and $h_{as} = a + bv_{wind}$

The constants a and b will depend on whether the boundary layer on the panel surface is laminar or turbulent as well as its surface orientation relative to gravity. If the radiating surface is facing upwards and is colder than the air and the wind velocity, v_{wind} is less than $v_{wind} < 0.076$ m/s (to allow free convection to take place) and the flow is laminar, then a = 0.8 and b = 0. If the radiator is warmer than the ambient air and $v_{wind} < 0.45$ m/s then a = 3.5 and b =0. For a surface colder than the ambient and $1.35 < v_{wind} < 4.5$ then a = 1.8 and b = 3.8 (Erell and Etzion, 2000).

The solar radiation absorbed by the panel is

$$\dot{Q}_{solar} = \phi\alpha\tau\rho \cdot G \cdot A_s \quad (3)$$

The term $\phi\alpha\tau\rho$ represents a complicated function relating the solar irradiance G onto a horizontal surface to the radiation actually absorbed by the surface. It will depend on the position of the sun relative to the radiating surface as well as the spectral properties of the radiation and optical surface properties of the radiating surface.

The heat transfer from the ambient air to the panel through the back insulation is

$$\dot{Q}_{aws} = \left(T_a - T_s\right)/R_{aws} \quad (4)$$

Where $R_{aws} = 1/h_{aws}A_s + L_s/k_sA_s$

A steady-state energy balance of the panel is

$$0 = \dot{Q}_{aws} + \dot{Q}_{as} - \dot{Q}_{ssky} + \dot{Q}_{solar} + \dot{m}c\left(T_{w1} - T_{w2}\right) \quad (5)$$

The average temperature T_s of the radiating surface during a specified time interval (say $\Delta t = 3600$ s) is determined by solving for T_s in Equation (5) by trial and error. This can be done provided that the thermal capacity of the panel is relatively small such that the heating response time of the panel is small compared to the time interval Δt. The thermal resistance between the water and the radiating surface will be relatively small, the temperature of the water T_w in the panel will thus be more or less equal to the radiating surface temperature T_s, and the water temperature may be assumed to vary linearly along its flow path as $T_s \approx T_w \approx \left(T_{w1} + T_{w2}\right)/2$

3.2 Storage tank

The water storage tank is treated as a single control volume, with an average temperature T_t as shown in Figure 3. As the tank will be cooler than the

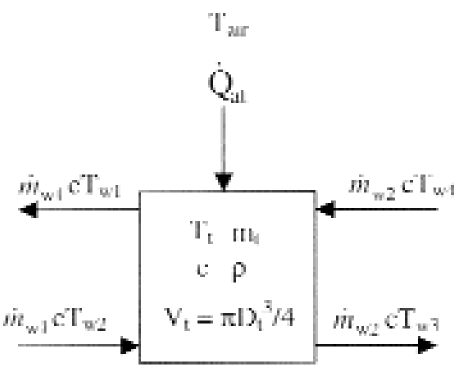

Figure 3: Water storage tank control volume

ambient air, it is insulated with a layer of insulation of thickness L_t and thermal conductivity k_t. The tank dimensions are arbitrarily chosen so that its height equals its diameter thus making it possible to express its surface area and volume in terms of only one variable, its diameter D_t.

The heat transfer rate between the water and the tank is a function of the ambient air temperature and may be given as

$$\dot{Q}_{at} = (T_a - T_t)/R_{at} \tag{6}$$

Where

$$R_{at} = 1/h_{at}A_t + L_t/k_{at}A_t \, ; \; h_{at} = a + bv_{wind} \, ;$$
$$A_t = 2\pi D_t^2/4 + \pi D_t^2 \text{ and } m_t = \rho\pi D_t^3$$

(for a tank such that its height equals its diameter)

If the tank is relatively large and the circulating flow rates are relatively low, then a significant degree of stratification of the water into layers of varying temperatures and thickness will be present. This makes it important for the cold water from the radiating panels to enter at the bottom of the tank and the warmer water to return from the top of the tank. Conversely the cold water entering the room convectors should be drawn from the bottom and the heated water returned to the top.

Considering an energy balance for the storage tank water control volume, the new temperature T^{new} of the water a finite time Δt later is given as

$$T^{new} = T^{old} + \frac{\Delta t}{m_t c_w}\left(\dot{m}_{w1}c(T_{w1} - T_{w2}) + \dot{m}_{w2}c(T_{w4} - T_{w3}) + \dot{Q}_{at}\right) \tag{7}$$

If the water circulation rates are in the order of 2 to 3 tank volumes per 8 hour-periods, then it is shown (Lunde, 1980) that there is a relatively small effect on the total amount of energy transferred to or from the tank, irrespective of whether a well mixed or stratified tank condition exits or not.

3.3 Room convector

The heat is transferred from the air in the room to the cold water circulating in the convector. The waterside heat transfer coefficient will be in the order of 4000 W/m^{2}K, however, the airside heat transfer coefficient is only in the order of 10 W/m^{2}K. This means that the cold water pipes in the convector have to be finned on their airside so as to reduce the overall thermal resistance. It is suggested that standard aluminium finned copper tubes as typically used in the air conditioning industry could be used to construct the convector heat exchanger. The fin spacing should be large enough to allow for natural convection. Rows should be kept to a minimum preferably only one row and should be horizontally orientated relatively high up in the room.

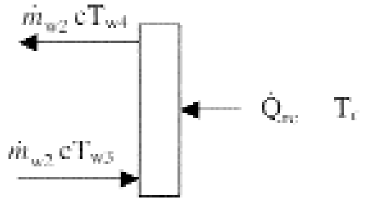

Figure 4: Room convector control volume

The control volume for the convector is given in Figure 4. The heat transferred from the air in the room to the water in the convector is

$$\dot{Q}_{rc} = (T_r - T_c)/R_{rc} \tag{8}$$

Where, as was also assumed for the radiator panels, the temperature of the water circulating through the convector may be approximated as a linear average between the inlet and outlet water temperatures as $T_w = T_c = (T_{w3} + T_{w4})/2$.

The thermal resistance of the convector may be given in terms of inside water and outside air thermal resistances as

$$R_{rc} = 1/h_{ci}A_{ci} + 1/\eta_{fin}h_{fin}A_{fin} \tag{9}$$

The heat transfer coefficient on the waterside may be determined using a standard heat transfer equation (Mills, 2000) as

$$h_{ci} = \frac{k_w}{D_{ci}} Re^{0.8} Pr^{0.35} \tag{10}$$

Where $Re = \dfrac{4\dot{m}_{w3}}{\pi D_{ci}\mu_w}$ and $Pr = \dfrac{c_w \mu_w}{k_w}$

The heat transfer coefficient on the airside may be determined also using a standard heat transfer equation (Mills, 2000) as

$$h_{fin} = 1.07(\Delta T/L_{fin})^{1/4} \text{ for } 10^4 < Gr < 10^9 \tag{11a}$$

$$h_{fin} = 1.3\Delta T^{1/3} \quad \text{for } 10^9 < Gr < 10^{12} \tag{11b}$$

Where

$$Gr = \frac{\beta \Delta T g \rho^2 L_{fin}^3}{\mu}$$

β is the coefficient of thermal expansion and can be taken (for an ideal gas) as 1/T if T is in Kelvin, ΔT is the temperature difference between the air and the surface and L_{fin} is the (vertical) length of the fin along which natural convection is taking place.

A standard chilled and hot water air conditioning coil or heat exchanger with a fin spacing of 4 fins per 25.4 mm and 0.25 mm fin thickness and tube outside diameter of 15.88 mm is however commercially available (Yucon Coil, 2003). The overall dimensions of such a heat exchanger is shown in Figure 9. It has an airside heat transfer area of 12.33 m^2, a fin efficiency of at least 90 % and an airside heat transfer coefficient of 2.8 W/m^2K. The airside heat transfer coefficient is calculated using equation (11) for an average temperature difference of 10 °C and a length L_{fin} of 0.098 m.

A steady state energy balance for the convector gives

$$0 = \dot{Q}_{rc} - \dot{m}_{w2} c \left(T_{w4} - T_{w3}\right) \tag{12}$$

3.4 Room options

A number of options may appear apparent for how best to approach the design of an enclosure situated in a hot desert environment comfortable enough for people to work in. The suggested approach is to insulate the room walls, *a priori*, and then to cool the inside air and the interior wall surfaces. It is not possible to transfer heat to/through perfectly insulated thermal mass; it is difficult to transfer heat in thermal mass made from naturally poor-conducting masonry materials. Heat transfer from a hot ceiling to the air in the room is not by convection. Hot air rises and forms an inversion layer of good-insulating stagnant air. Rather the heat is transferred from the ceiling to the floor (and occupants) by radiation and then only by convection from the floor to the air. It is still possible to feel uncomfortable in a room in which the air is cool but the walls hot. It is thus suggested that the appropriate approach to the problem is to bring the thermal mass (in this case the cold water) as close as possible to the point where it can cool the air in the room. This is especially important when the air heats up fairly rapidly due to the heat input directly into the air in the room from the computers, electrical appliances and the occupants.

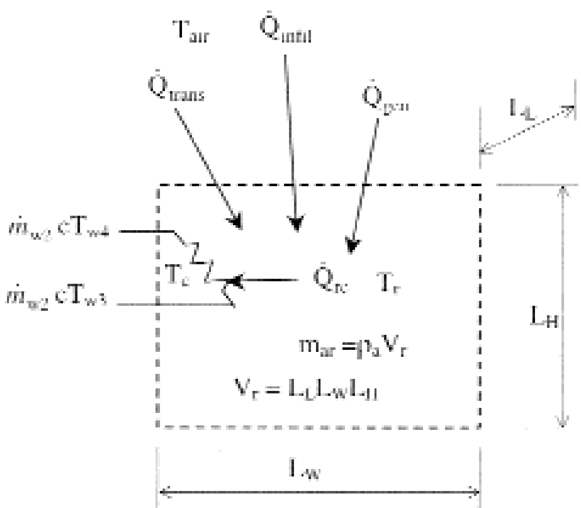

Figure 5: Room control volume including the convector

The control volume for the room containing the convector is given in Figure 5. A number of room heat source *loads* may be considered (Stoeker and Jones, 1982). These are: an internal heat source $\dot{Q}_{gen}$ due to people, electrical appliances such as computers and electric lights and which is independent of the outside ambient air temperature, a source due to infiltration of hot outside ambient air $\dot{Q}_{infil}$, and a heat source due to heat transmission from the outside ambient air into the room $\dot{Q}_{trans}$.

These heat sources are approximated as follows:

The infiltration load:

$$\dot{Q}_{infil} = C_{infil}\left(T_a - T_r\right) \tag{13}$$

Where

$$C_{infil} = \left(ACPD \cdot L_W \cdot L_H \cdot L_L \cdot \rho_a \cdot c_{pa}\right)/(24 \cdot 3600)$$

and the number of air changes of the room per 24 hours (ACPD) for a room of the size of the computer room under consideration would typically be expected to be in the order of 6.

The transmission load:

$$\dot{Q}_{trans} = \left(T_a - T_r\right)/R_r \tag{14}$$

Where

$$R_r = {L_r}/{k_r A_r}$$

and

$$A_r = 2\left(L_W L_L + L_W L_H + L_H L_L\right)$$

and the thermal resistance of the room may be approximated in terms of a wall thickness L_r of equivalent thermal conductivity k_r and surface area A_r. Equation (14) would depend on the orientation of the room, direct solar radiation through windows or fenestration, actual wall and roof construction details and inside and outside wall heat transfer coefficients and the time of day. For a reasonably well-insulated building and to limit the number of variables (without unduly losing accuracy), Equation (14) is probably quite acceptable.

The internal heat generation load is $\dot{Q}_{gen}$ on the number of occupants, electrical appliances, lights, computers, etc. in the room. It must thus be specified as a function of the time.

A steady-state energy balance for the contents of the room would yield

$$0 = \dot{Q}_{trans} + \dot{Q}_{infil} + \dot{Q}_{gen} - \dot{Q}_{rc} \tag{15}$$

With the assumption that $T_c = (T_{w3} + T_{w4})/2$ equations (12) and (15) may be (relatively easily) solved for the essentially two unknowns T_r and T_{w4} by trial and error. To simplify the trial and error solution scheme, it is convenient to note that T_{w4} can be expressed explicitly in terms of T_r as

$$T_{w4} = \frac{\dot{m}_{w2} c T_{w3} + \frac{T_r}{R_{rc}} - \frac{T_{w3}}{2R_{rc}}}{\dot{m}_{w2} c + \frac{1}{2R_{rc}}} \tag{16}$$

3.5 Inter-connecting pipe work

Air relief and vacuum breakers (either separate or combined) must be place at any of the highest points of the system, where air could collect and interfere with the water flow. In a closed system, provision must be made for thermal expansion on the water.

Pipe work should be slightly inclined to allow for any air bubble that might find itself in the system to rise under the influence of gravity to a high point where provision must be made for it to escape. Similarly, pipe work should be slightly inclined to allow the last drop of liquid to be able to drain back to suitably positioned drain points or valves.

3.6 Sensors and controls

Temperature sensors need to be provided to ensure proper operation of the circulating pumps. Two basic control options are necessary to ensure that the room temperature does not exceed a specified maximum temperature;

(i) If $T_{w2} < T_{w1}$ and $T_{w2} < T_{room\ maximum}$ then switch the radiator panel-circulating pump on, otherwise switch it off.

(ii) If $T_r > T_{room\ maximum}$ and $T_{w3} < T_{room\ maximum}$ then switch the room-circulation pump on, otherwise switch if off.

4. Computer program

The equations as given in Section 3 and that constitute the thermal model for the cooling system are quantitatively analysed using an Excel spread. Single-valued spreadsheet input variables are given in Table 1 and include the variables defining the physical size of the radiation cooling system; the radiator panels, water storage tank, computer room and the room convector. The pump mass flow rate, for convenience sake, is assumed (in the spreadsheet) as corresponding to one tank change per 3600 s but can also be varied, if required. Also to be specified is the initial storage tank temperature.

The specified weather conditions to which the system must be designed to are given graphically in Figure 6 in time steps of 3600s, and are the ambient air temperature, wind velocity and solar irradiation incident onto a horizontal surface. From this input data, the dew point temperature of the ambient air and the emissivity and temperature of the sky are calculated. Also given, in Figure 7(d), is the solar radiation incidence angle modifier $\alpha\tau\rho\phi$ to take into account the effect of the orientation of the radiator panels relative to the solar radiation and variation of radiator surface optical properties, with the angle of the sun relative to the radiator surface. The room heat loads, internal heat generation, transmission and infiltration, are given in Figures 7(a), 7(b) and 7(c).

Having now specified the input constants and time dependent variables, the steady-state energy balance for the radiator panel control volume given by Equation (5) and as depicted in Figure 2 is undertaken next. A steady-state balance is undertaken as the thermal capacity of the absorber material and water is relatively small, and the time period of 3600 s is relatively long. The heat flows for the panel are expressed as a function of only one variable, the absorber surface temperature. Using the built-in Excel "goal seek" option and a "macro" procedure, values of the surface temperature T_s are guessed for each time step until the energy inputs and outputs are equal. Similarly, a steady state energy balance using Equation (15) is then undertaken to determine the room temperature T_r at each of the time steps and, hence, also the outlet convector cooling water temperature T_{w4} using Equation (16). The new water storage tank temperature is then calculated using Equation (7).

The process as outlined in the previous three paragraphs is then repeated until the guessed radiator panel area, water storage tank volume and room convector area cannot be further reduced, subject to the condition that the specified maximum room and maximum end-of-day (18:00) storage tank temperatures are not exceeded, and the pump circulation rates are set as one tank volume per hour.

5. Results and analyses

5.1 Sample computer program

The program is essentially manually optimised by trial and error (such that the energy balances are always maintained). For this sample computer program example, it is assumed that the room temperature must not exceed 25°C, and that the pump flow rates are both one storage tank volume per hour. It is further assumed that the tank temperature at the end of the working day (18:00) is not above 17°C. This is to ensure that if the 24-hour time cycle is repeated, then the cycle will indeed be sustainable. The following optimised (lowest) values were obtained; for the radiator panel area 48 m^2, for the water storage tank volume 4.15 m^3 and room convector area 75.5 m^2, and are reflected in Table 1 as rows 9, 15 and 37 respectively.

All the important temperatures (ambient air,

Table 1: Spreadsheet inputs

Row	*Variable*	*Symbol*	*Units*	*Value*
6	**Radiator panel:**			
7	Insulation thickness on the back of the radiator	L_s	m	0.025
8	Thermal conductivity of the radiator back-insulation	k_s	W/m°C	0.030
9	Radiating area of the radiator surface	A_{s0}	m^2	48.0
10	Fin efficiency of radiator surface	η_s		0.9
11	Radiator surface emissivity	ε_s	-	0.9
12	Water-side heat transfer area	A_{si}		48
14	**Water storage tank:**			
15	Volume of water in tank	V_t	m^3	4.15
16	Density of the water	ρ_w	kg/m^3	1000
17	Specific heat of water	c_w	J/kg°C	4186
18	Insulation thickness around the storage tank	L_t	m	0.025
19	Thermal conductivity of the storage tank insulation	k_t	W/m°C	0.030
20	Mass of the water in the storage tank	m_t	kg	4150
21	Tank diameter (if diameter = height)	D_t	m	1.742
22	Surface area of the storage tank (if diameter = height)	A_t	m^2	14.296
24	**Computer Room:**			
25	Average wall thermal conductivity	k_r	W/m°C	0.1
26	Equivalent wall thickness of room	L_r	m	0.1
27	Room height	L_H	m	2.27
28	Room length	L_L	m	7.90
29	Room width	L_W	m	5.20
30	Room Volume	V_r	m^3	93.252
31	Surface area of the room	A_r	m^2	141.634
32	Air density	ρ_a	kg/m^3	1.180
33	Specific heat at constant pressure for air	c_{pa}	J/kg°C	1005
34	Mass of the air in the room	m_{ar}	kg	110.037
36	**Room convector(s):**			
37	Air-side area of the convectors	A_{co}	m^2	75.5
38	Fin efficiency based on the air-side area of the convector	η_{co}	-	0.915
39	Water-side area of the room convectors	A_{ci}	m^2	9.060
41	**Time step**	Δ_t	s	3600

computer room storage tank water, radiator panel surface, sky temperature and dew point temperatures) as calculated in the spreadsheet are given in Figure 8. The sky temperature is seen to vary from its lowest in the early morning of – 5°C to a maximum of 15°C in the afternoon. The dew point is seen to remain essentially constant at 9°C throughout the 24 hour time period. The dew point is of interest because should the radiating surface temperature drop to the dew point temperature (as the sky temperature drops to below the dew point temperature) dew will tend to form. When dew forms on a surface, its temperature will tend to remain constant at this temperature even as the sky temperature drops further due to the latent heat of condensation. The computer room temperature closely follows the ambient air temperature with the room-circulating pump off. However, when it is switched on at about 08h 00, although the ambient air temperature outside the room increases to its maximum of 32°C, the maximum room temperature only rises to its maximum of 25°C.

When the radiator panel circulating pump is switched off at about 7:00 am, the water flow through the panel ceases, it is no longer cooled by the circulating water, and its temperature attains a maximum of about 58°C at about 13:00 hours. Should the water not circulate through the radiator panels at night, the panel temperature falls and dew or frost may form. If the dew point is above 0°C,

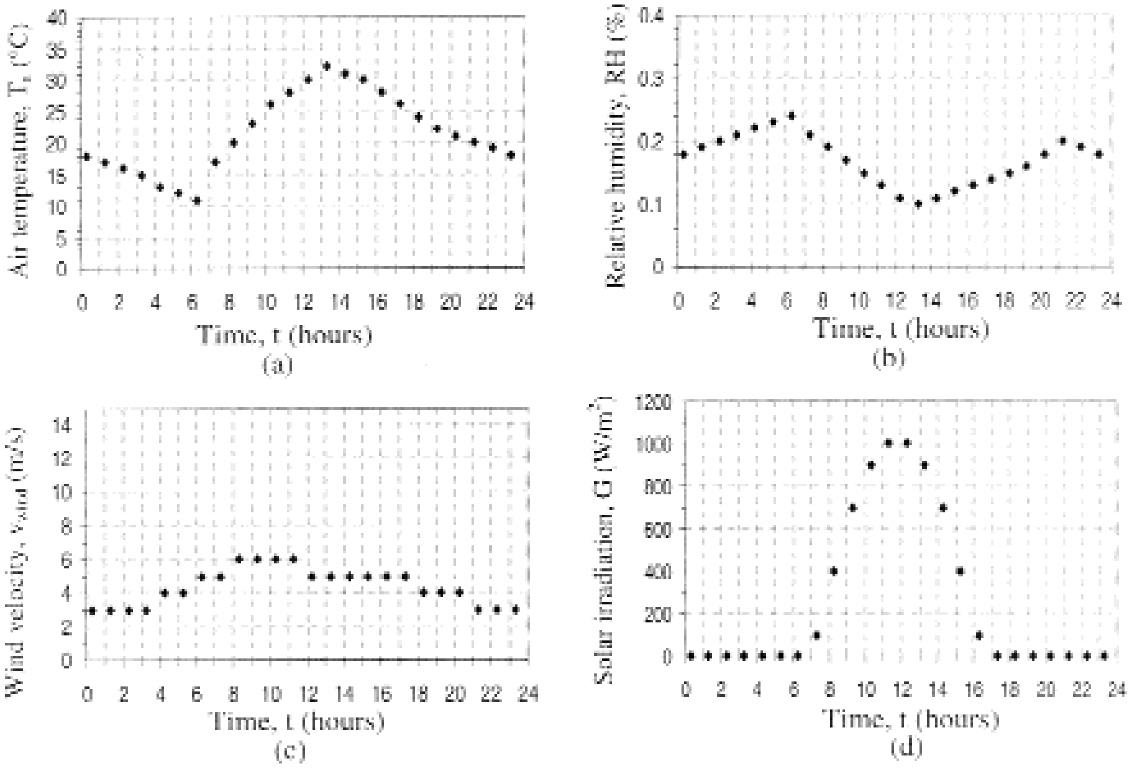

Figure 6: Hourly input values of the air temperature, relative humidity, wind velocity and solar irradiation

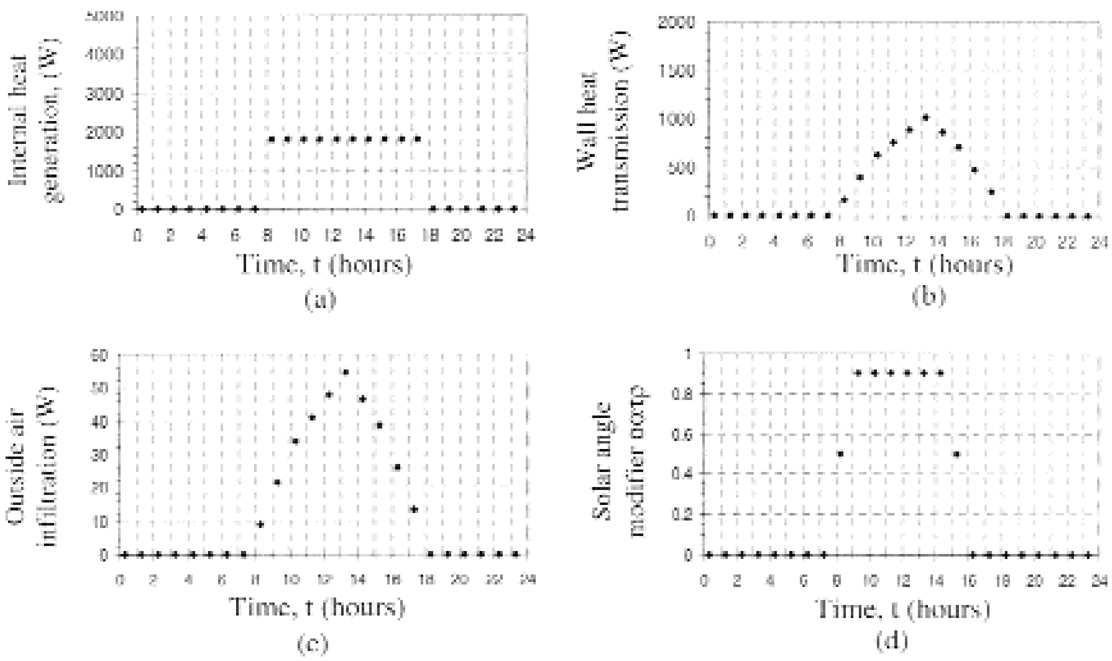

Figure 7: Hourly input values of the internal heat generation (a), heat transmission through the walls (b) outside air heat infiltration (c) and solar radiation incidence angle modifier (d)

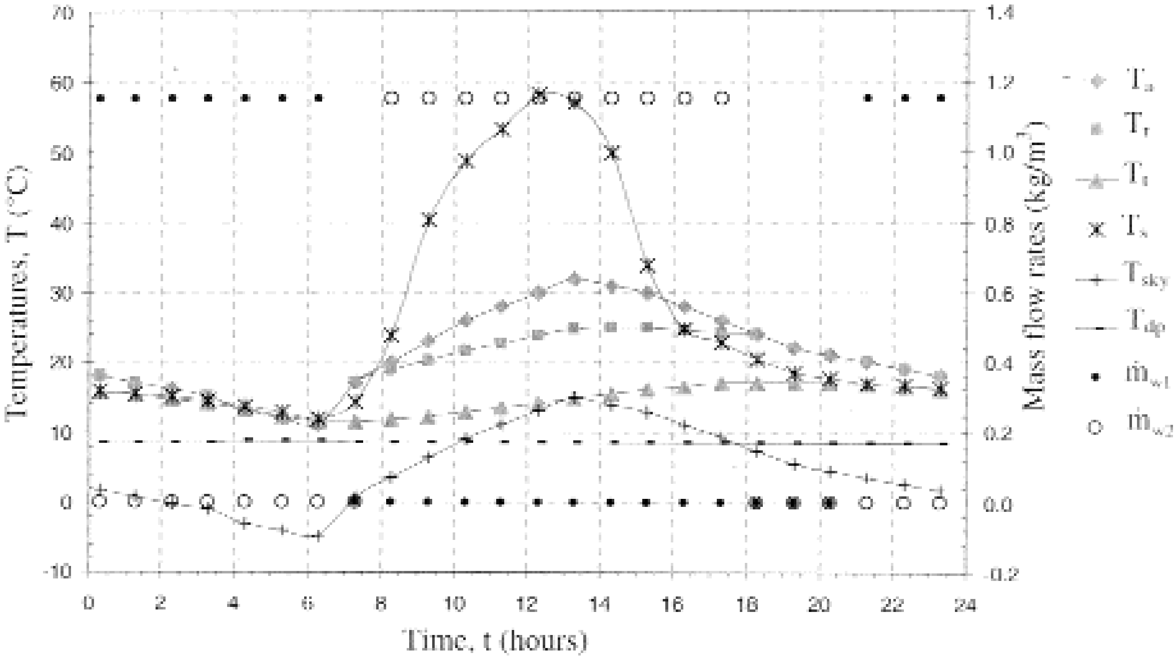

Figure 8: Hourly temperatures and mass flow rates for the base case values of the radiation cooling system

dew will form, or if the dew point is less than 0°C frost will form.

5.2 Sensitivity analysis

A sensitivity analysis was conducted for a variation of ± 25% of the more important system variables including the radiator panel surface, storage tank volume, room wall thermal conductivity, room wall thickness, convector fin surface area and the internal heat generation load. The results of this sensitivity analysis are given in Table 2 as a percentage variation of a number of the system performance parameters. These performance parameters include the sum of the difference between the outside ambient air temperature and the inside room air temperature multiplied by the time step for each of the hours of the nine hour period that the room circulation pump was switched on $\Sigma(T_a-T_r)\Delta t$. Other performance parameters include the total amount of heat gained by the storage water during the room cooling operating period $Q_{t,in}$ and the heat removed during the night sky cooling period $Q_{t,out}$, the heat removed from the room by the convectors Q_{rc}, the maximum room temperature $T_{r,max}$ and the evening (18:00) temperature of the tank at the end of the working day $T_{t,end}$.

If the water circulation rate is increased from the base case value (and keeping the water storage volume constant at 4.15 m^3) then to comply with the criteria that $T_{r,max} \leq 25$ °C and $T_{t,end} \leq 17$ °C, then both the radiator and convector surfaces have to increase as shown in Table 3.

If the water circulation rates are reduced further to less than 0.29 kg/s, the system will not be able to meet the specified operating requirement of $T_{r,max} \leq$ 25°C and $T_{t,end} \leq$ 17°C. If, however, the tank volume is increased, it is possible to reduce the mass flow rates further but at the additional expense of not only an increased tank volume, but also increased radiator and convector surface areas as shown in Table 4.

6. Discussion, conclusions and recommendations

For the 8-hour period from 11:00 to 07:00, the 48 m^2 of radiator panel surface area for the base-case example was able to remove 84 MJ from the water storage tank. This corresponds to an average heat removal rate of 60.8 W/m^2. This heat removal rate compares favourably with the value of 80 W/m^2 as reported by Erell and Etzion (1996). It is thus concluded that the thermal model presented in this paper can be used with confidence as a design tool for the sizing of a cooling system using night sky radiation.

Using the base case input values given in Table 1 and Figures 6 and 7, optimum (lowest) values were obtained for a radiator panel area of 48 m^2, a water storage tank volume of 4.15 m^3 and a room convector area of 75.5 m^2. To obtain these values, it was assumed that room temperature must not exceed 25 °C, the tank temperature at the end of the working day (18:00) is not above 17 °C and that the pump circulation rate was equal to one-storage-tank volume per hour. If the flow rates are reduced, and keeping the tank volume constant at 4.15 m^3, then increased panel and convector surface areas are required as shown in Table 3. If the flow rates

Table 2: Sensitivity analysis with the difference in value of the variables, as a result of a ± 25 % variation from the base case values, expressed as a percentage

Variable	$Q_{t,in}$ MJ	$Q_{t,out}$ MJ	$\Sigma(T_a\text{-}T_r)\Delta t$ °C·hour	Q_{rt} MJ	$T_{r,max}$ °C	$T_{t,end}$ °C
Base-case	97.70	97.76	43.24	88.04	24.98	17.00
$0.75A_s$	-3.0	-14.2	-10.7	-2.8	1.8	3.7
$1.25A_s$	2.3	10.6	8.0	2.1	-1.3	-2.8
$0.75V_t$	-2.2	-14.8	-1.9	-0.5	1.6	5.5
$1.25V_t$	1.2	11.1	-1.1	-0.3	-0.6	-2.6
$0.75k_r$	-3.6	0.0	10.8	-4.1	-2.3	-1.2
$1.25k_r$	3.0	0.0	-8.9	3.4	2.0	0.9
$0.75L_r$	3.9	0.0	-11.5	4.4	2.8	1.2
$1.25L_r$	-2.8	0.0	8.5	-3.2	-1.8	-0.9
$0.75A_{co}$	-8.5	0.0	-43.4	-9.6	7.0	-2.8
$1.25A_{co}$	6.0	0.0	25.9	6.8	-4.4	2.0
$0.75\,\dot{Q}_{gen}$	-9.6	0.0	28.7	-10.8	-5.3	-3.2
$1.25\,\dot{Q}_{gen}$	10.5	0.0	-28.7	10.8	5.3	3.2
$0.75\,\dot{m}_{w1}$	-0.4	-1.8	-1.4	-0.4	0.2	0.5
$1.25\,\dot{m}_{w1}$	-0.4	-1.8	-1.4	-0.4	0.2	0.5
$0.75\,\dot{m}_{w2}$	-0.4	-1.8	-1.4	-0.4	0.2	0.5
$1.25\,\dot{m}_{w2}$	0.2	0.0	0.7	0.2	-0.1	0.1

Table 3: Radiator panel A_{so} and convector surface A_{co} areas for reduced water circulation rates (to meet a specified operating requirement that $T_{r,max} \leq$ 25°C and $T_{t,end} \leq$ 17°C)

$\dot{m}_w$ kg/s	A_{so} m^2	A_{co} m^2
1.15	48.0	75.5
0.58	52.8	77.7
0.29	57.6	86.8

Table 4: Water circulation rates for increased tank volumes to meet a specified operating requirement that $T_{r,max} \leq$ 25°C and $T_{t,end} \leq$ 17°C

V_t m^3	$\dot{m}_w$ kg/s	A_{so} m^2	A_{co} m^2
4.15	1.15	48.0	75.5
8.30	0.23	52.8	98.2
16.60	0.12	100.8	135.9

are further decreased to below a critical value (in this case 0.29 kg/s) the specified maximum room and tank temperatures can only be achieved if the tank volume also increases as shown in Table 4. These results also tend to illustrate the relatively complex thermal relationships that exist between the various variables of a night sky cooling system.

The overall thermal management strategy adopted in order to keep the computer room to within the acceptable limits, may be succinctly described as follows. Firstly, isolate the room from heat ingress as a result of the harsh environmental conditions using insulating material. Secondly, remove the heat from the now predominantly internally generated heat sources by allowing the heated air to flow naturally over a finned heat exchanger through which cold water is circulated. The cold water, having been cooled the previous night, is stored in an insulated tank. In keeping with this strategy, it is assumed for the base case example that the computer room has been further insulated, by adding a 50 mm layer of polyurethane foam. The base case values as given in Table 1 are for this more-or-less thermally optimised system. The walls of the existing computer room at this time are of an ordinary prefabricated construction. Had the walls not been lined with additional insulation then, all things being equal, somewhat large radiator panels and convector areas and tank volumes would be required to meet the specified requirement that of the room temperature not rising above 25 °C, and the tank temperature should not fall to below 17 °C by evening time, given the typical weather data and internal heat generation as given in Figures 6 and 7.

Standard plastic swimming pool panels are potentially suitable for the radiator panels. They are heat and ultra-violet radiation stabilised polypropy-

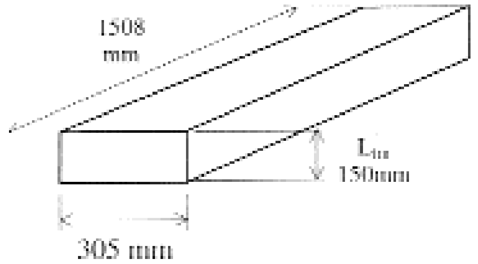

Figure 9: Basic dimensions of a convector consisting of 22 horizontal 15.88 mm outside diameter copper pipes in three rows and 0.25 mm thick plate vertical aluminium fins spaced at a pitch of 6.3 mm

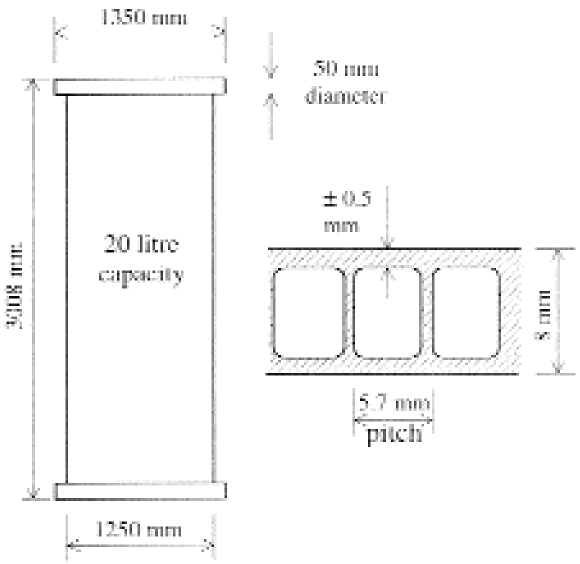

Figure 10: Radiator panel details

lene panels (with EPDM connections) with a surface emissivity of about 0.9. They are sold with a 5-year warranty (provided that they are installed in accordance with the manufacturer's instructions) and have an expected lifetime of 15 years (Sun Command). They can be placed on a relatively flat (horizontal) roof. It is claimed that damage of the plastic (as a result of water freezing in the narrow water channels) is unlikely to occur. It is, however, recommended that they be slightly inclined to ensure that the water can drain out naturally when not in use. The basic dimensions and construction of a typical swimming pool panel is shown in Figure 10.

The cooling efficiency of the radiator panels can be improved by making use of a special polyethylene cover sheet to reduce the convective heat transfer coefficient. The optical properties of the polyethylene may be further improved by specially treating the cover with, for example, a PbS film, or by pigmenting it with, for example, ZnS. In this way, it can also be made into a particularly good reflector capable of, not only significantly suppressing convection, but also capable of reflecting a significant amount of the incoming direct solar radiation away from the relatively cold infrared radiating surface beneath it (Torbjörn and Niklasson, 1995). These radiator panel-cooling improvements were not considered in the computer simulation sensitivity analyses. They could have been, but without further research and commercial development, they were deemed to be too unreliable for use at such an isolated windy desert station as the one considered in this paper.

Nomenclature

A	Area, m^2
c	specific heat, J/kgK
D	diameter, m
G	Solar radiation, W/m^2
Gr	Grashof number $Gr = \beta\Delta T g \rho^2 L_{in}^3 / \mu^2$
g	gravitational constant, 9.81 m/s^2
h	heat transfer coefficient, W/m^2K
k	thermal conductivity, W/mK
L	length, m
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
Pr	Prandt number $Pr = c_w \mu_w / k_w$
Q	energy, J
$\dot{Q}$	heat transfer rate, W
R	thermal resistance, K/W
Re	Reynolds number $Re = 4\dot{m}_w / \pi D_{ci} \mu_w$
t	time, s
T	temperature, °C
V	volume, m^3
v	velocity, m/s

Greek symbols

α	absorptivity
β	coefficient of thermal expansion
Δ	difference
ε	emissivity
ϕ	angle, °
η	efficiency
ρ	density, reflectivity
μ	viscosity, m/kgs
σ	Stefan-Boltzmann constant and equals $5.67 \times 10^{-8} W/m^2K^4$
τ	transmissivity

Subscripts

a	air
c	convector
cold	cold
dp	dew point
gen	internal heat generation
H	height
hot	hot

i	inside
infil	room air infiltration
L	length
o	outside
p	constant pressure
r	room
s	radiator panel surface
sky	sky
solar	solar
t	water storage tank
trans	wall heat transmission
W	width
w	water
wind	wind

Abbreviations

ACPD air room volume changes per day

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