Air-Heating Solar Collectors for Humidification-Dehumidification Desalination Systems

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ABSTRACT

Relative to solar water heaters, solar air heaters have received relatively little investigation and have resulted in few commercial products. However, in the context of a Humidification-Dehumidification (HDH) Desalination cycle, air heating accounts for advantages in cycle performance. Solar collectors can be over 40% of an air-heated HDH system's cost, thus design optimization is crucial. Best design practices and sensitivity to material properties for solar air heaters are investigated, and absorber solar absorptivity and glazing transmissivity are found to have the strongest effect on performance. Wind speed is also found to have an impact on performance. Additionally a well designed, and likely low cost, collector includes a double glazing and roughened absorber plates for superior heat transfer to the airstream. A collector in this configuration performs better than current collectors with an efficiency of 58% at a normalized gain of 0.06 K m^2/W.

NOMENCLATURE

Symbols

A_p	Collector area [m^2]
c_p	Specific heat capacity of air at constant pressure [J/kg K]
$\mathcal{F}_{1-2}$	Radiation transfer factor
F'	Heat gain factor
F_O	Equivalent effectiveness
F_R	Heat removal factor
h	Average convective heat transfer coefficient [W/m^2 K]
h_1	Absorber to fluid convective heat transfer coefficient [W/m^{2}K]
h_2	Inner glazing to fluid convective heat transfer coefficient [W/m^{2}K]
h_{amb}	Convective heat transfer coefficient to ambient air [W/m^{2}K]
h_r	Absorber to glazing radiative heat transfer coefficient [W/m^{2}K]
$h_{r,c1-c2}$	Radiation heat transfer coefficient, between glazing layers [W/m^{2}K]
$h_{r,c2-sky}$	Radiation heat transfer coefficient, outer glazing to sky [W/m^{2}K]
I_T	Solar irradiation [W/m^2]
$\dot{m}$	Mass flow rate of air through the collector [kg/sec]
NG	Normalized gain [K m^2/W]
q_u	Useful heat gain by the fluid per unit collector area [W/m^2]
S	Solar flux absorbed by the absorber [W/m^2]
T	Temperature [K]
$\bar{T}$	Mean Temperature [K]
U_b	Overall bottom loss heat transfer coefficient [W/m^{2}K]
U_L	Overall heat loss coefficient [W/m^{2}K]

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U_t	Overall top loss heat transfer coefficient [W/m^{2}K]
α	Solar absorptivity
ε	IR emissivity
η	Collector efficiency
τ	Solar transmissivity
$(\tau\alpha)$	Transmittance Absorptance Product

Subscripts

a	Absorber
air	Ambient air
amb	Sol-air
*c*1	Inner glazing
*c*2	Outer glazing
f	Fluid or air stream
in	Air inlet
out	Air outlet
sky	Sky

INTRODUCTION

Solar water heaters have been thoroughly investigated and developed commercially [1, 2, 3] whereas there has been relatively little investigation and almost no commercial development of solar air heaters. In the context of Humidification-Dehumidification (HD) Desalination, heating the air as opposed to the water streams shows significant performance gains [4]. These heaters can amount to over 40% of the total cost [5] of an HDH system and so the development of a cost effective and efficient solar collector is essential to the system's overall feasibility.

COMPARING EXISTING COLLECTORS

Nayaran et al. [4] reviewed potential solar air heaters and compared their efficiency and top temperature output to other collectors in the literature as well as those manufactured commercially. The standard metric of a solar air heater's performance is the collector thermal efficiency. It is defined by Equation 1.

$$\eta = \frac{\dot{m}c_p(T_{out} - T_{in})}{I_T A_p} \tag{1}$$

This definition of performance is that used by the ASHRAE 93-2003 Standard for solar collector testing [6] and it defines both the instantaneous and time averaged efficiencies when evaluating dynamically changing solar radiation inputs and temperature profiles.

In solar collectors, efficiency decreases with fluid temperature gain, as heat losses are directly proportional to temperature. The most common way of showing solar air heater efficiency is to plot the efficiency versus the normalized heat gain as defined by Equation 2. It has has the units K m^2/W.

$$NG = \frac{(T_{out} - T_{air})}{I_T} \tag{2}$$

The normalized gain will decrease with increasing air mass flow rate. Figure 1 shows the reported efficiencies of solar air heaters in the research literature [5, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18] as a function of normalized heat gain, where better performing heaters are more to the top right portion of the graph, as they deliver the highest air temperature at the highest efficiency. The highest efficiency commercial solar collector, the SunMate Sm-14 [15], is included for comparison. The highest performing heaters are indicated as grey points on the graph, which will set the standard by which new designs will be compared.

Two outliers [9, 11] that do not follow the trend of the other data were excluded from the grey shaded group. Mohamad [9] is a theoretical study that claims an extremely large efficiency improvement with an addition of porous media as an absorber, with 75% efficiency at 0.12 K m^2/W of normalized gain. However, experiments conducted on a collector in a similar configuration [19] show only 60% efficiency at a normalized gain of 0.017, which is significantly lower. Romdhane [11] provided an experimental study with various types of surface roughening. He reports a near constant efficiency through increasing normalized gain to his highest normalized gain and efficiency. However, when experiments are done by varying mass flow rate, the same collector shows a linearly increasing trend with increasing mass flow rate, which is expected. The increase in mass flow rate is accompanied by a decrease in temperature rise (and normalized gain) as the air has a shorter residence time in the collector. This appears to be inconsistent with the reported results for varying normalized gain.

By comparing the designs of the five best heaters a list of apparent best design practices can be obtained.

Air flow over the absorber plate: Having air flow above the absorber decreases losses from the top of the absorber plate and eliminates conduction resistance through the plate. Many modern air heaters use this method. [7, 8, 16]

Packing materials: Packing materials in the air stream improve heat transfer by mixing the air and providing more surface area to absorb radiation. Packing also provides sensible heat storage but comes at the cost of high pressure drop [8, 12]. In the context of HDH, the materials have to be moisture and corrosion resistant. Since they add effective

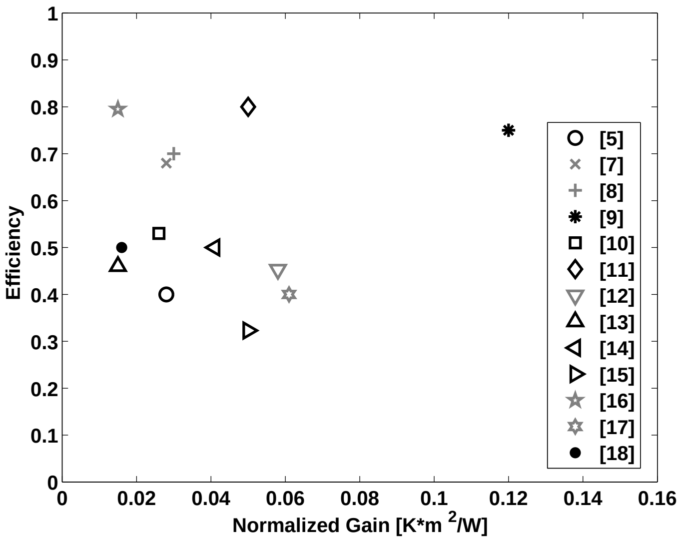

Figure 1. Normalized comparison of solar air heaters in literature

energy storage to the collector they will be considered in a separate paper on transient solar collector designs.

Roughened absorber plate: improves convection heat transfer into the air. Rough configuration also increases pressure drop, but only marginally when compared to a smooth plate for duct cross sections used in solar air heating. Roughening for increased convection has been extensively investigated, and has shown performance improvements in collectors [9, 16].

Multiple passes of air through the collector: improves heat gain by increasing contact with the absorber, and makes the absorber run cooler, decreasing losses [12]. However the same can be accomplished with a rough absorber plate without having a very thermally conductive absorber. This allows many more materials to be used as absorber surfaces, such as those with low thermal conductivity.

Multiple glazing layers: reduces heat loss by infrared radiation and traps an insulating air layer between the glazings. However this comes at a greater material cost and lower solar transmissivity. All of the top performing heaters except Sahu and Bhagoria [16] use a double glazing.

Glass and metal construction: provides better heat transfer characteristics and better durability. All the best performing collectors used glass and metal construction, as polymer alternatives, especially for glazings, suffer from low durability although providing optical properties comprable to glass [20].

SENSITIVITY OF HEATER PERFORMANCE TO MATERIAL PROPERTIES AND ENVIRONMENTAL CONDITIONS

Baseline Design

Using information gleaned from the literature review, a simple baseline design was devised. To obtain the required temperature rise, a long a narrow collector was necessary and it has the cross section as illustrated in Figure 2. In reality this long effective collector can be achieved by placing shorter modules in series. The total length of the collector is 10 m, and it consists of an aluminum absorber coated with carbon black paint, and low-iron glass glazing panels. The outside is insulated with fiberglass insulation. The outdoor wind speed is assumed to be a moderate 5 m/s, which is consistent with averages for a desert climate such as Saudi Arabia [21]. The characteristic length over which wind blows is the average of the width and length of the collector, as wind direction is highly variable. The absorber is roughened with transverse ribs to increase turbulence and heat transfer. The ribs have a constant height and pitch throughout the channel. Tables 1 and 2 outline the fixed parameters of the baseline design. The analysis only varies one of the material properties in Table 2 at a time, keeping all the others constant.

Constants	Values
Solar Irradiation	900 W/m^2
Ambient Wind Speed	5 m/s
Latitude	27 °
Solar Declination	23 °
Collector Tilt Angle	45 °
Collector Inlet Temperature	30 °C
Ambient Air Temperature	30 °C
Dew Point Temperature	4 °C
Insulation Conductivity	0.02 W/mK

Table 1. Constant parameters for simulating baseline design

Governing Equations

In steady state, the heat transfer processes in the collector can be modeled as a series of thermal resistances. They are shown in Figure 3. If a control volumes are taken around the two glazings, the absorber and the air stream, and the heat flows

Material Properties	Value
Glazing Refraction Index	1.526
Glazing Extinction Coefficient	4
Absorber Solar Absorptivity	0.94
Glazing IR Emissivity	0.92
Absorber IR Emissivity	0.86
Transmittance Absorptance Product - $(\tau\alpha)$	0.77

Table 2. Baseline values of varied material properties

Roughening Parameters	Value
Rib Height, h	0.0032 m
Rib Pitch, p	0.02 m
p/h	6.3
Roughening Regime	Fully Rough

Table 3. Roughening parameters

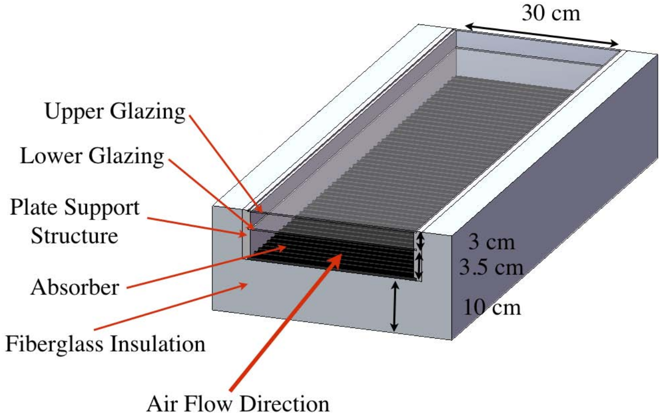

Figure 2. Diagram of Heater Cross Section

between each of the control volumes balanced; Equation 3 is obtained.

$$U_t(T_{amb} - \bar{T}_{c1}) + h_r(\bar{T}_a - \bar{T}_{c1}) + h_2(\bar{T}_f - \bar{T}_{c1}) = 0 \qquad (3a)$$

$$S + U_b(T_{amb} - \bar{T}_a) + h_r(\bar{T}_{c1} - \bar{T}_a) + h_1(\bar{T}_f - \bar{T}_a) = 0 \qquad (3b)$$

$$h_2(\bar{T}_{c1} - \bar{T}_f) + h_1(\bar{T}_a - \bar{T}_f) = q_u \qquad (3c)$$

where terms are defined in nomenclature. S is defined by multiplying the irradiation I by $(\tau\alpha)$, which is the combined solar transmissivity and absorptivity of the absorber and cover system. The above equations can be solved for the fluid temperature and integrated along the length of the collector. However, in steady state, where heat capacity can be neglected, the integrated differential equation can be expressed explicitly using several lumped parameters as recommended by Duffie and Beckman [20].

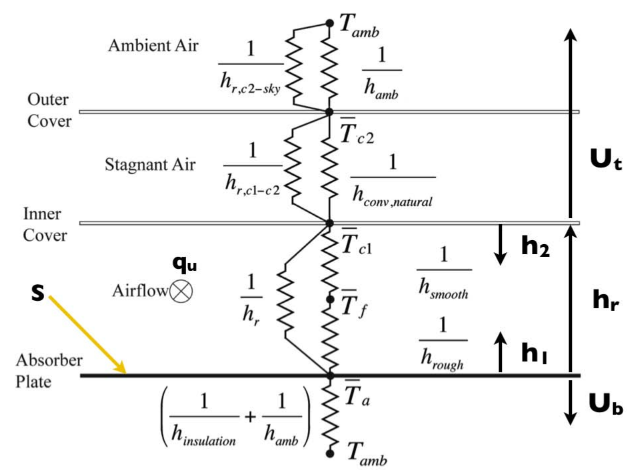

Figure 3. Heat transfer resistances with lumped parameters

Figure 3 shows radiation to the same ambient air temperature as convection which is required for the use of simple lumped parameters. Duffie and Beckman [20] state that sky temperature is relatively unimportant for calculating collector performance. However this may become important as the collector is required to run hotter and radiative loss is more important. Therefore sky temperature is included in these calculations and T_{amb} is defined as a sol-air temperature of the environment by Equation 4. A correlation [20] for sky temperature based on dew point temperature is used. The sol-air temperature is used for the total loss, despite the fact that there is no radiation from the back surface to the sky. This does not have a large effect on the loss, as the bottom loss only 3% of total loss in the baseline configuration.

$$T_{amb} = T_{air} + \frac{-h_{rad,c2-sky}(T_{air} - T_{sky})}{h_{amb} + h_{rad,c2-sky}} \quad (4)$$

Heat transfer coefficients are calculated using commonly used correlations for natural and forced convection heat transfer [22]. Radiation heat transfer coefficients are linearized radiation heat transfer expressions in the form of $h_{rad} = 4\sigma\bar{T}^3\mathcal{F}_{1-2}$, as given by Lienhard and Lienhard [23].

The surface was modeled as having transverse rib roughening using the equations developed by Dalle Donne and Meyer [24]. The roughening parameters are given in Table 3

Equation 5 is a combined loss coefficient resulting from algebraic manipulations of Equations 3.

$$U_L = \frac{(U_t + U_b)(h_1 h_2 + h_1 h_r + h_2 h_r) + U_b U_t (h_1 + h_2)}{h_1 h_r + h_2 U_t + h_2 h_r + h_1 h_2} \quad (5)$$

With a combined loss coefficient a simple energy balance leads to Equation 6

$$q_u = [S - U_L(\bar{T}_a - T_{amb})] \quad (6)$$

where q_u is Since the mean plate temperature is unknown it needs to be found iteratively. Equation 7 gives the integrated solution in terms of a heat removal factor and loss factor given by Equation 8.

$$\bar{T}_a = T_{amb} + \frac{q_u}{F_R U_L}(1 - F_R) \quad (7)$$

$$F_R = \frac{\dot{m}c_p}{A_p U_L}\left[1 - \exp\left(\frac{-A_p U_L F'}{\dot{m}c_p}\right)\right] \quad (8a)$$

$$F' = \frac{h_r h_1 + h_2 U_t + h_2 h_r + h_1 h_2}{(U_t + h_r + h_1)(U_b + h_r + h_2) - h_r^2} \quad (8b)$$

After convergence the final plate temperature is used in Equation 6 and the useful heat is divided by the total solar irradiation to obtain an efficiency. The air temperature rise is given by dividing the useful heat by the capacity rate of the air, $\dot{m}c_p$. To characterize how the efficiency behaves vs the temperature rise in the collector, an equivalent effectiveness [20] is defined by Equation 9 .

$$F_O = \frac{\frac{\dot{m}c_p}{A_p U_L}\left[1 - \exp\left(\frac{-A_p U_L F'}{\dot{m}c_p}\right)\right]}{\exp\left(\frac{-A_p U_L F'}{\dot{m}c_p}\right)} \quad (9)$$

The efficiency vs. normalized gain curve takes the form of Equation 10.

$$\eta = F_O(\tau\alpha) - F_O U_L\left[\frac{T_{out} - T_{amb}}{I_T}\right] \quad (10)$$

The normalized gain in Equation 10 is different from Equation 2 in that it is based on the outlet and ambient temperature difference. However since the calculations on the baseline collector were done by fixing the air inlet T_{in} to T_{amb} this equation can describe the efficiency vs. normalized gain with normalized gain as it is defined in Equation 2. F_O is analogous to a heat exchanger effectiveness. Therefore the collector performs better at higher temperature gains if it has high glazing transmissivity and absorber absorptivity of solar radiation, losses are minimized, and the heat transfer coefficent from the absorber to the air is high.

Sensitivity Analysis Results

The sensitivity study investigates the effect of various material properties on performance of the collector, as well as how environmental conditions effect the performance when certain materials are used.

Material Properties. The material properties that have the greatest impact on performance are the infrared emissivities of the glazing and absorber plates, the glazing stack solar transmissivity, and absorber solar absorptivity. Figure 4 shows the relative effect of each parameter as it is varied from 0 to 1. The operating point was based on the operation of an HDH cycle in a desert environment. For the 10 m long collector a normalized gain of 0.06 K m^2/W is obtained at a mass flow rate of 0.029 kg/sec. The calculations assume that the conduction resistances of the glazings and absorber are negligible, owing to their small thickness. Heater dimensions can be optimized for the desired temperature rise, and optima are easily found for the dimensions of roughness features or spacing between the plates, and will not be discussed here.

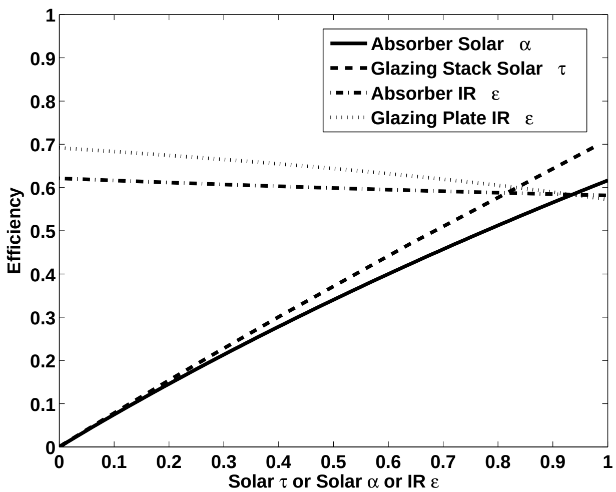

Figure 4. Effect of emissivity, absorber absorptivity, and glazing transmissivity on efficiency

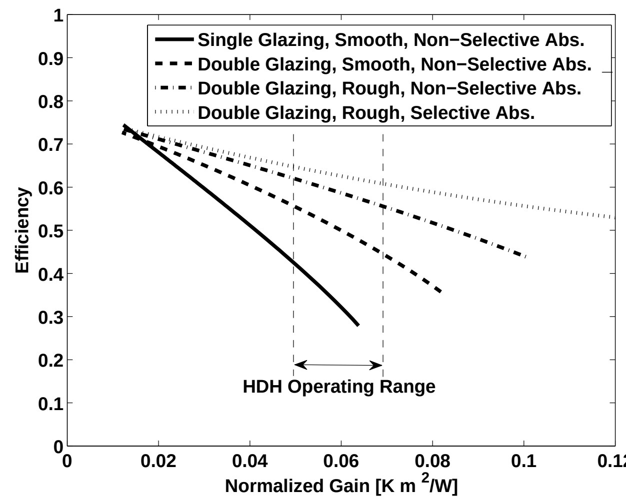

Figure 5. Effect of design enhancements on collector performance

The graph shows that the two most important parameters are absorber solar absorptivity and the glazing solar transmissivity. Using an absorber with a selective coating (α = 0.9-1, ε = 0.02-0.3) [25] does not offer significant performance gains with only a 4% efficiency improvement. This is typically a very expensive design addition, as selective surfaces are often made of exotic materials, such as quartz, can involve expensive manufacturing processes, and are limited to only a few substrates. Using a low ε coating for the glazing plates offers a larger improvement of 10%, but also can be an expensive addition due to materials and manufacturing processes used.

To ascertain the efficiency "value" of various design attributes, heaters were simulated in different configurations, adding various design improvements onto a collector with a smooth, non-selective absorber with a single glazing. As can be seen from Figure 5, the addition of surface roughness increases performance by the greatest amount in the HDH operating range. A typical value for a fully roughened surface using the parameters in Table 3 can increase the heat transfer coefficient 8 times over that for a smooth plate. The use of a selective absorber coating (ε=0.05) also improves performance by a small amount. For low normalized gain a selective surface does not improve performance for over a roughened absorber, although it is more important at higher temperatures where radiative losses dominate.

Figure 6 shows how the baseline design compares with existing air heaters that include Chafik's HDH collector [5] and the SunMate commercial collector [15] for which the performance curve is available. It is clearly shown that the baseline collector, which incorporates all of the design enhancements beside a selective absorber, outperforms existing collectors in the HDH operating range.

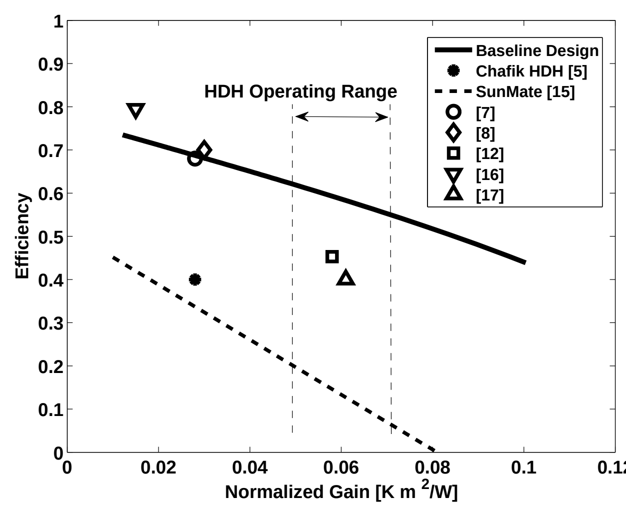

Figure 6. Comparison of Baseline Design (Double Glazed, Rough, Non-Selective Absorber) with exiting air heaters

Environmental Conditions. Environmental conditions also influence how a collector can be designed, and the importance of design enhancements. The environmental parameters of most importance are the ambient air temperature, the dew point temperature, which affects sky temperature, and ambient wind speed. As suggested in the literature [20] ambient air temperature has a small effect on performance when compared to ambient wind speed. A variation in dew point temperature from -4 to 36 °C translates into a 1-2% efficiency change in the HDH operating range, and a change in ambient air temp from 0 to 40 °C translates into a 3-4% efficiency change in the HDH operating range.

Wind speed has a larger effect on performance. Figure 7 shows that wind speed can have a substantial effect on performance in the HDH operating range, with an efficiency change of 10-12%. Figures 8 and 9 show the effect of wind speed on performance at varying infrared emissivities for glazing and absorber plates. The lines each represent a different wind speed from 2 m/s (h_{amb} = 3.07 W/m^{2}K) to 20 m/s (h_{amb} = 39.74 W/m^{2}K) in increments of 2 m/s. The graphs show that using low ε surfaces is of low importance in calm environments with only marginal improvement in windy ones.

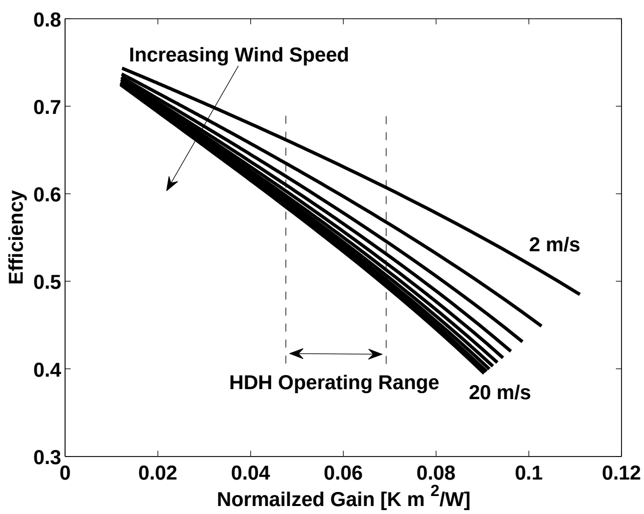

Figure 7. Efficiency vs normalized gain for different wind speeds.

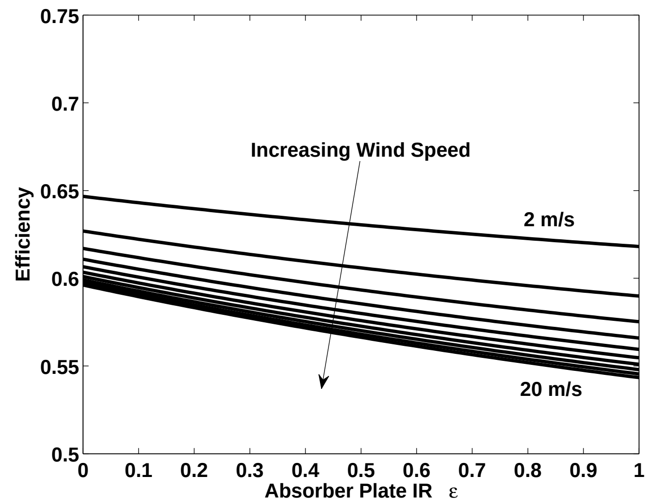

Figure 8. Effect of wind speed on collector performance at different absorber IR emissivity

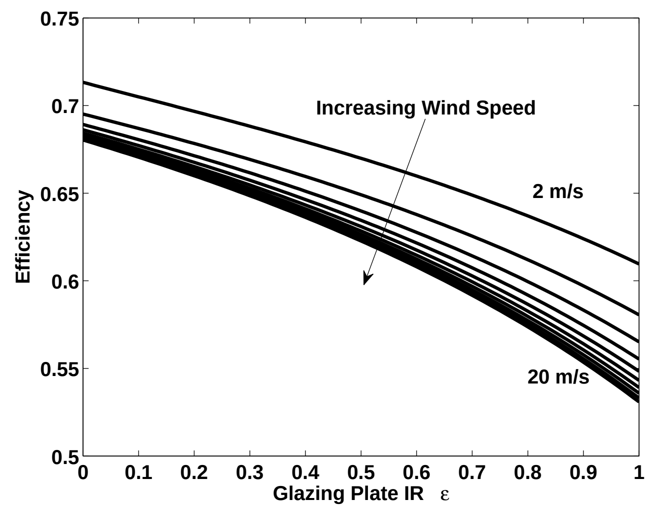

Figure 9. Effect of wind speed on collector performance at different glazing IR emissivity

CONCLUSION

Air heating solar collectors have been studied for conditions typical of an HDH desalination system. Overall, improving the transmissivity of the glazing by using highly transmissive polymer films or low iron glass, and using a very absorptive absorber, which is inexpensively accomplished by including a carbon black coating, would have the largest impact on performance. The greatest improvement to a collector's performance can be accomplished by using a double glazing, resulting in a 20% efficiency increase in the HDH operating range compared to a single glazed collector. This reduces radiative losses as glass is opaque to infrared radiation. An insulating layer of trapped air between the plates also lowers the outer collector temperature and further decreases convective and radiative losses. The second most important enhancement is the addition of a rough surface on the absorber plate, which improves convection from the heat source to the air. This offers up to a 12% efficiency increase without the need for a selective surface on the absorber. Adding a selective surface to the absorber, or a low ε coating to the glazing can add a great deal of cost due to the use of exotic materials. In the HDH operating range these improvements have limited impact on performance, even as environmental conditions change adversely. Of the environmental conditions that effect performance, wind speed has the greatest impact. The wind speed increases the already dominant convective losses from the collector. A collector with a double glazing, a highly roughened absorber, and a carbon black coated absorber, results in a collector efficiency of 58% at a normalized gain of 0.06 K m^2/W. This offers significant performance gains over existing solar air heaters, which is accomplished in a simple and possibly inexpensive design.

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