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THE DESIGN AND PERFORMANCE OF A DISTRIBUTED FLOW WATER-COOLED SOLAR COLLECTOR

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Abstract

Design of a flat plate collector which reduces the temperature differential between the absorber plate and the fluid is described. The reduced temperature differences are shown to yield increase collector performance. Flow characteristics of the collector are examined. Collector thermal performance is illustrated for typical operating and environmental conditions. A cost analysis is presented to demonstrate that material and assembly costs are substantially lower than for any collector presently on the market.

NOMENCLATURE

A_{flow}	cross-sectional flow area, m^2
A_i	tube area, m^2
c	specific heat, W-hr/kgm-°K
D	tube diameter, m
f	friction factor
g	gravitational acceleration, m/sec^2
h	tube wall to liquid heat transfer coefficient, W/m^2-°K
h_w	wind heat transfer coefficient, W/m^2-°K
HR	solar irradiation normal to collector, W/m^2
k	thermal conductivity of absorber plate, W/m-°K
L	collector length, m
m	$\sqrt{U_L/k\delta}$
$\dot{m}$	liquid mass flow rate, kgm/sec
n	number of tubes
N	number of glass covers
Nu	Nusselt number
q_ℓ	energy loss, W/m^2
q_u	useful energy, W/m^2
Q_{fin}	energy to fin, W
Q_{tube}	energy to tube, W
Q_u	useful energy to liquid, W
Re	Reynolds number
S	solar irradiation normal to plate, W/m^2
T_a	ambient temperature, °K
T_b	tube wall temperature, °K
T_i	entering liquid temperature, °K
T_o	leaving liquid temperature, °K
T_p	mean absorber plate temperature, °K
T_w	mean liquid temperature, °K
$\bar{T}$	mean temperature of region of plate between tubes, °K
U_L	collector loss coefficient, W/m^2-°K
V	mean velocity of liquid in tubes, m/sec
V_a	ambient wind speed, m/sec
W	center-to-center tube spacing, m
x	distance along plate (fin) measured from tube center, m
α	plate and tube solar absorptance
δ	thickness of absorber plate, m
ϵ_g	infrared emittance of glass (0.88)

ε_p infrared emittance of plate and tube

η fin efficiency

θ inclination of collector with respect to horizontal, deg

ν kinematic viscosity, m^2/sec

ρ liquid density, kgm/m^3

σ Stefan-Boltmann constant, $5.6697 * 10^{-8} W/m^2 - °K^4$

τ solar transmission of glass

1. INTRODUCTION

The losses from a flat plate collector increase with increasing temperature and, therefore, it is desirable that the collector absorber plate operate at as low a temperature as possible, consistent with the desired temperature of the collected thermal energy at the points of use. Temperature drops occur primarily in the various heat exchangers in the system, one of which is the absorber plate itself. This paper considers the temperature effect on the plate and presents a design which reduces the temperature differential between fluid and plate to the point where it is insignificant with respect to effect on collector performance. At the same time, the materials required and assembly costs would seem to be substantially lower than for any collector presently on the market.

The design is similar to flow between parallel plates. However, to reduce the tendency for excessive deflection for forced flow between parallel plates, flow is at negative gage pressure, and the flow channel is maintained by either of two possibilities: a) incorporating corrugations or other forms of surface indentations on one or both sheets or b) placing a porous spacer, such as screen wire, between the two sheets. The various layers are registered with each other, the ends inserted in slotted header tubes and sealed to the tubes and along the edges by appropriate means. Figures 1(a) and 1(b) show these two approaches to distributed flow.

The analytical approach to the problem for the sandwich construction shown in Figure 1(b) would logically be along the lines of flow through porous media, while that required for the flow path shown in Figure 1(a) would be the theory of flow through non-circular tubes. As for approximate approaches, both situations could probably be analyzed fairly accurately by assuming flow between parallel plates, while the flow path shown in Figure 1(a) could be represented by flow through circular tubes of individual cross-sectional area equal to that for the individual flow passage made by the corrugations.

The latter method was used in this paper. This is convenient, since one of the objectives of this work is to compare the performance of flat plate collectors which use the fin-tube absorber plate with that for collectors using distributed flow. The performance of the distributed flow design thus becomes that calculated for the fin-tube plate when the tubes are sufficiently close together. Figure 2 illustrates the general model.

In the analysis that follows, it will be assumed that the tubes are connected in parallel, and that the total cross-sectional tube flow area is 5.5 cm^2 per meter width of collector. The actual flow area for several commercially available collectors is very close to this value. As will be seen later, it is also the approximate value that is needed for the distributed flow design. In the temperature and heat transfer analysis, the calculations show that the maximum rate of collection of useful energy for high performance collectors at high values of irradiation seldom exceeds 600 watts/m^2. Accordingly, this value is chosen for purposes of calculation of a maximum design flow rate. A value for such a maximum flow rate is needed in order to determine the regime of flow, i.e., laminar or turbulent. For this latter determination, it is further assumed that at maximum flow rate, the liquid temperature increases 2.5°C per meter length. The analysis is carried out both for water and for anti-freeze solution consisting of half ethylene glycol and half water by mass.

2. THE FLOW PROBLEM

The maximum mass flow rate, per meter width of collector assuming liquid water is

$$\dot{m} = \frac{q_u L}{c(T_o - T_i)} = 0.057 \frac{Kgm}{sec\text{-}m}$$

This is approximately one gallon per minute-meter. Since the flow area is specified, the mean velocity of the fluid can be calculated:

$$V = \frac{\dot{m}}{\rho\, A_{flow}} = 0.9026\ \frac{m}{sec}$$

The number of tubes per meter of width is $n = 1/W$ where W is the center-to-center spacing of the tubes. The area of each tube, multiplied by the number of tubes per meter, is $5.5 \times 10^{-4}\ m^2$.

$$\frac{\pi D^2}{4}\ \frac{1}{W} = 5.5 \times 10^{-4} \qquad \text{or}$$

$$D = (2.65) \times 10^{-2} \sqrt{W} \tag{1}$$

The Reynolds number is

$$Re_{(anti\text{-}freeze)} = \frac{VD}{\nu} = 2460 \sqrt{W} \tag{2}^*$$

$$Re_{(water)} = 11{,}500 \sqrt{W} \tag{3}$$

For turbulent flows $2{,}460 \sqrt{W} > 2000$ (anti-freeze)
$11{,}500 \sqrt{W} > 2000$ (water).

Therefore, if flow is to be turbulent using anti-freeze,

$W > 0.667$ meters

and for water

$W > 0.174$ meters

A value of 15 cm is a realistic maximum value for tube spacing, both in terms of what is available commercially, and also from the point of view of performance. Therefore, flow can be assumed to be laminar in solar collectors that correspond to the assumptions as to flow area made here, and all calculations that follow are based on the assumption of laminar fully developed flow.

Both the Nusselt number and the friction factor can now be easily determined. The Nusselt number for constant heat flux, laminar flow is

$$Nu_D = 4.12 \ . \tag{4}$$

The friction factor for laminar flow is

$$f = \frac{64}{Re} \ . \tag{5}$$

In a later section, it is shown that when $W < 0.3$ cm the fin-tube plate is almost equivalent in thermal performance to that of a plate at the temperature of the liquid; i.e., losses due to plate-to-liquid temperature differences are practically zero. A logical question is, given $W \approx 0.3$ cm, and from Eq. (1), $D \approx 0.145$ cm, is it possible to have a flow rate of 0.057 Kgm/sec-m with the constraints on pressure that apply to distributed flow systems? The pressure of the flowing liquid must be below atmospheric pressure, but large negative values are unacceptable. The distributed flow system should be capable of generating liquid temperatures of 90°C, as required for air conditioning, and the saturation pressure is, of course, atmospheric at 100°C for water. In general, one would impose an upward flow direction on the plate to be assured of complete filling of all tubes. Upward suction in the case of distributed flow means that the fluid pressure at the top of the collector would be at least as much below atmospheric pressure as dictated by the principles of hydrostatic, i.e., for a collector in the vertical position 3 meters long, the pressure at the top would be approximately negative 0.3 atmospheres with zero flow, and perhaps considerably greater in the negative direction with an adequate flow rate. Therefore, downward flow is required for high temperature of collection, and an adequate system for purging of air during start-up must be provided.

Figure 3 shows the elements of such a flow loop. The float chamber d, and over-flow chamber e serve as pressure regulators so that the pressure at the inlet and exit are below atmospheric in amounts determined approximately by the values of h^u and h^L. During start-up, valve a is opened. The

*The value of ν for 50% by mass mixture of water and ethylene glycol was obtained by extrapolation to 60°C from the CRC Handbook ($\nu \approx 1.0 \times 10^{-6}\ m^2/sec$).

air bleed is taken from the headers of the collector at the ends opposite the liquid inlet. The vacuum at a need only exceed the value of h^u by a small amount, so that when valve a is opened, liquid is drawn into the collector from reservoir d, and raised up somewhat from reservoir e. The air is purged out as the collector fills, with the system always at negative pressures. When the air is removed, flow proceeds from top to bottom automatically.

The most desirable situation is probably for the pressure to be constant as the fluid flows downward through the collector. The head loss then becomes simply $L \sin \theta$, so in accordance with the Darcy equation

$$L \sin \theta = f \frac{L}{D} \frac{V^2}{2g} \quad .$$

Using Eqs. (1), (2), (3), and (5), one obtains the value for D for 0.057 Kgm/sec-m flow rate at constant pressure

$$D(\text{water flow}) = 2.55 \times 10^{-4}/\sqrt{\sin \theta}$$

$$D(\text{anti-freeze}) = 5.5 \times 10^{-4}/\sqrt{\sin \theta}$$

Or, for $\theta = 45°$

$$D(\text{water, p=const}) = 3.03 \times 10^{-4} \text{ m} = 0.0303 \text{ cm}$$

$$D(\text{anti-freeze, p=const}) = 6.54 \times 10^{-4} \text{ m} = 0.0654 \text{ cm}$$

These are minimum values, i.e., smaller values would reduce the flow rate below the choosen value of 0.057 Kgm/sec-m. The larger of the two values, that for anti-freeze, is somewhat smaller than required, according to the condition that $D \approx 0.145$ cm. Thus achieving adequate flow rate is not a problem, according to these calculations.

3. THE TEMPERATURE AND HEAT TRANSFER PROBLEM

The solar collector looses energy to the air and sky according to the equation

$$q_\ell = U_L(T_p - T_a) \tag{6}$$

T_p is the mean plate temperature, including the tube portion, defined by the following equation

$$T_p = \frac{(W - D)\overline{T} + D\,T_b}{W} \tag{7}$$

The value of $\overline{T}$ in Eq. (7), the mean value of the plate temperature in the region between tubes, is calculated using the usual extended surface theory. This problem has been adapted for solar absorber plates by Duffie and Beckman(1)*, and the fin temperature is given by the following equation

$$\frac{T - T_a - S/U_L}{T_b - T_a - S/U_L} = \frac{\cosh mx}{\cosh m(W - D)/2} \quad ,$$

or integrating T dx over the length of fin(W - D)/2, one obtains the mean temperature $\overline{T}$ of the fin portion of the plate:

$$\frac{\overline{T} - T_a - S/U_L}{T_b - T_a - S/U_L} = \frac{\tanh m(W - D)/2}{m(W - D)/2} = \eta \tag{8}$$

where $m = \sqrt{U_L/k\delta}$. Heat conduction in the direction of fluid flow is neglected in the derivation of Eq. (8).

The value of T_b in Eq. (7), the tube wall temperature, is obtained as follows:

$$Q_u = h\,A_i(T_b - T_W) = h\,\pi\,D\,L(T_b - T_W) \tag{9}$$

where Q_u is the useful energy collected over an area of width W and length L. From Eq. (4),

$$h = k\,Nu/D$$

so

$$h\,D_{(\text{water})} = 2.73 \text{ W/°K} \qquad \text{and} \tag{10}$$

$$h\,D_{(\text{anti-freeze})} = 1.52 \text{ W/°K} \quad . \tag{11}$$ **

*Bracketed numbers refer to entries in REFERENCES.

**A value of 0.415 watts/m-°C was used for 50% mixture of ethylene glycol and water (from the CRC Handbook).

Using the definition of fin efficiency, the thermal energy flowing to the tube from the fin is

$$Q_{fin} = (W - D)L\ [S - U_L(T_b - T_a)]\eta \quad . \qquad (12)$$

The heat flow to the tube due to the solar radiation is

$$Q_{tube} = DL[S - U_L(T_b - T_a)] \quad . \qquad (13)$$

The heat flow into the liquid is

$$Q_u = Q_{fin} + Q_{tube} \quad .$$

Combining Eqs. (9), (12) and (13) and solving for T_b, the result is

$$T_b = \frac{[(W - D)\eta + D](S + U_L T_a) + h\ D\ \pi\ T_w}{h\ D\ \pi + [(W - D)\eta + D]U_L} \quad . \qquad (14)$$

The tube diameter D in Eq. (14) can be expressed in terms of W using Eq. (1). The loss coefficient U_L was evaluated using the emperical equation developed by Klein[(2)]:

$$U_L = \left(\frac{N}{(344/T_p)[(T_p - T_a)/(N + f)]^{0.31}} + \frac{1}{h_w}\right)^{-1} + \frac{\sigma(T_p + T_a)(T_p^2 + T_a^2)}{[\varepsilon_p + 0.045N(1 - \varepsilon_p)]^{-1} + [(2N + f - 1)/\varepsilon_g] - N} \qquad (15)$$

where

$$f = (1.0 - 0.04h_w + 5.0 \times 10^{-4}\ h_w^2)(1 + 0.058N); \qquad (16)$$

and

$$h_w = 5.7 + 3.8\ V_a \qquad (17)$$

All temperatures appearing in Eq. (15) should be in degree Kelvin. The procedure used to evaluate collector efficiency for various values of tube spacing is as follows:

(1) Assume values for the following parameters:

Liquid temperature, T_w

Solar irradiation normal to the collector plate, HR

Solar transmission and absorptance product, $\tau\alpha$; $\tau\alpha$ = 0.87, 0.80, 0.75 for N = 1, 2, 3, respectively

Wind speed, V_a

Air temperature, T_a

Infrared plate emittance, ε_p

Fin conductance, $k\delta$

Water or anti-freeze

(2) Determine the loss coefficient U_L based on an assumed plate mean temperature, T_p.

(3) Calculate the value of T_b using Eq. (14).

(4) Calculate mean plate temperature using Eq. (7). The calculated value of T_p is compared with the assumed value, and a corrected estimate for T_p is made and the process repeated. After a satisfactory value of T_p has been found, the useful energy collected per unit area is found from the following equation

$$q_u = S - U_L(T_p - T_a) \quad . \qquad (18)$$

Calculations were made for the two values of liquid temperature of 60 and 90°C, and 60°C value being representative of building heating and water heating requirements, and the 90° being the approximate temperature needed for absorption air conditioning. In each of these cases, an air temperature must be assumed, and a nominal value of 10°C was chosen to correspond to the 60°C temperature of collection, and 35°C was chosen as a value appropriate for the 90°C collection temperature. For each of these pairs of values of air and liquid temperature, the value of infrared emittance, and choice of fluid (water or anti-freeze) were varied. The combinations are as follows:

Air Conditioning	T_w = 90°C, T_a = 35°C	ε_p = 0.1, Working fluid water
		ε_p = 0.95, Working fluid water
		ε_p = 0.95, Working fluid anti-freeze
Heating	T_w = 60°C, T_a = 10°C	ε_p = 0.95, Working fluid water
		ε_p = 0.10, Working fluid water

For all calculations, the wind speed was assumed to be 5m/sec, and the $\tau\alpha$ product 0.80 (two glass covers). In all cases, back and side losses were neglected.

Figures 4, 5, and 6 are the air conditioning cases. They show the efficiency versus tube spacing W for three values of solar irradiation normal to the plate, HR = 1000, 750 and 500 watts/m^2, and for four values of fin conductance, $k\delta$ = 0.001, 0.01, 0.1 and 10 watts/°K. For each value of solar irradiation normal to the plate, the horizontal line labeled $T_p = T_w$ corresponds to the efficiency for a plate which has a temperature equal to the water temperature. The various curves of efficiency versus W show that as the tube spacing decreases, the effect of fin conductance on efficiency decreases until at W = 0.3 cm, the efficiency is the same for all values of $k\delta$. Furthermore, at this same value of W(0.3 cm), the efficiency of the fin-tube design is within 1% of that for the case of $T_p = T_w$. Figures 5 and 6 show that use of anti-freeze results in a drop in performance as compared with that for water only for large $k\delta$ values. In any case, at W = 0.3 cm, the performance, even with anti-freeze as a working fluid, is essentially the same as that for distributed flow.

Figures 7 and 8 show similar results for the two cases of water heating, Figure 7 for an infrared plate emittance of 0.1, while Figure 8 is for $\varepsilon_p = 0.95$. The condition that

$$\eta_{\text{fin-tube}} \approx \eta_{\text{distributed flow}} \quad W \leq 0.3 \text{ cm}$$

applies for these conditions also.

In all the situations described above the effect of fin conductance vanishes at $k\delta$ = 10, or in other words, $k\delta = 10$ is equivalent to $k\delta = \infty$. Even at $k\delta$ = 0.1, most of the losses are associated with the tube wall-to-water temperature drop. The value of $k\delta$ = 0.1 is typical of several commercially available collectors.

The information shown in Figures 4 through 8 is shown in another way in Figures 9 and 10. These curves show the relative output increases when distributed flow is used instead of tube flow. This percent increase is plotted vs. W, for various cases previously described, and is 25% for $k\delta$=0.1, HR = 1000 W/m^2, W = 15 cm, ε_p = 0.95, T_w = 90°C, T_a = 35°C, as shown in Figure 9 for anti-freeze. When water is used instead of anti-freeze, the percent increase is about 18%.

The use of a selective black coating on the absorber plate allows the plate to operate at a high temperature without such a great loss, since the loss coefficient is low. The reduction in collection temperature, shown in Figure 10, also reduces the loss coefficient.

The percent increase for distributed flow vs. fin-tube flow for HR = 1000 W/m^2, W = 15 cm, ε_p = 0.1, T_w = 60°C, T_a = 10°C, anti-freeze, is only 12% while use of water instead of anti-freeze places the increase at 8%.

The curves in Figures 9 and 10 all merge at W $\approx$ 0.3cm at essentially zero percent increase, showing again that for W $\leq$ 0.3 cm, the losses associated with the plate-to-water temperature difference is negligible.

The above figure shows the effect of mean plate to water temperature difference without giving any direct indication of the actual value of this difference. Figures 11 and 12 show plots of plate temperature vs. W, for the various cases described above. Figure 11 is for the technically important case of $k\delta$ = 0.1 W/°K, and it shows, for example, that the plate operates at 24°C (mean value) above the water temperature when HR = 1000, ε_p = 0.95, anti-freeze, 2 covers, T_w = 60°C, T_a = 10°C.

Figure 12 is the same as Figure 11, except that the effect of $k\delta$ on $T_p - T_w$ is shown. This curve shows in a third manner that when W $\leq$ 0.3 cm, $T_p \approx T_w$.

A comparison of the percent increase in energy gain using distributed flow with that for specific commercially available absorber plates is possible. However, the particular total design in which an absorber plate is marketed, in terms of infrared emittance, number of covers, etc., is not particularly relevant, since we are making no claims in connection with these variables. Accordingly, in the following discussion, the assumption is made that the absorber plate is used in arbitrarily assumed collector, with respect to cover transmittance, infrared emittance, number of covers, etc.

The values of W and $k\delta$ of 15 cm and 0.1 W/°K spectively are appropriate for the Revere Collector and the Sun-water collector. The tubes in the Revere Collector are bonded to the plate by means of copper-loaded epoxy, while in the case of the Sun-water collector, silicone rubber and aluminum foil is used for this purpose. We have assumed a perfect thermal bond in our calculations. Neglecting this and other design differences between these two collectors and our model, the percent gain in collected energy, distributed flow vs. the given fin-tube designs, would be between 8.5% (Fig. 10) and 25.5% (Fig. 9). The smaller value corresponds to T_w = 60°C, water, and a selectively coated surface, whereas the larger value corresponds to T_w = 90°C, non-selective surface, and anti-freeze. The lower value (8.5%) corresponds to a situation that is not presently available in these collectors (selective black) whereas the larger (25.5%) is a realistic number in terms of current usage.

The case of single-cover non-selective surface collectors has not been included in the results shown in our figures. However, a single-cover collector with a selective black absorber plate is a very good combination. Let us assume that the Revere or Sun-water collector is made available with this single cover-selective black combination, and is used for high temperature collection, say T_w = 90°C. A reasonable question is, what would be the comparative performance of these collectors and the distributed flow design, should the selective black degrade? Because of the high loss coefficient for such a situation, the distributed flow would produce a substantial improvement, 40% in this case if anti-freeze is used.

In general, the higher the loss coefficient, the better the distributed flow system will appear on a comparative plot such as Figures 9 and 10. The values in these figures are lower than the actual values should be, because the back and edge losses (as well as tube-to-plate bond) were neglected.

The Pittsburgh Plate Glass Collector uses an absorber plate that corresponds closely with the assumptions used in our calculations, with $W \approx 6.3$ cm, and $k\delta \approx 0.3$ W/°K. For the situation of high fluid temperature, T_w = 90°, T_a = 35°, a non-selective surface, and HR = 1000 W/m^2, the relative percent increase using distributed flow would be between 4 and 7% depending on whether water or anti-freeze is used. The lower fluid temperature case (T_w = 60°C, T_a = 10°C), gives a similar relative percent increase of 3.7% and 6.5%.

4. DISCUSSION AND CONCLUSIONS

The experimental program using the distributed flow concept has not proceeded beyond the stage of flow experiments with certain sandwich and corrugated systems.

The flow rate has been found adequate for two experimental designs, one which represents the corrugated vs. plane surface system illustrated in Figure 1(a), and the other representing the sandwich construction shown in Figure 1(b). In the former case, the corrugated side of a 2' × 4' piece of "penticor," a commerically available decorative glass, was used with a plane sheet of 0.005" copper. The edges and headers were sealed with silicone rubber. The contour of the corrugations in the penticor are approximately sinusoidal. The peak-to-peak distance is 1/8", while the depth of the pattern is 0.027". This gives a flow area of 3.44 cm^2/m, a value somewhat below our assumed value of 5.5 cm^2/m. The flow rate, using water and for zero pressure differential (as per the discussion in the section on flow), was 0.114 Kgm/m-sec. This is for the vertical position. For θ = 45°, the flow rate should be 0.080 Kgm/m-sec for zero pressure differential. The values of W and D are 0.32 cm and 0.12 cm respectively, where D is the diameter of a circle of area equal to that of the individual flow passages in the penticor-copper sheet combination. The comparison is summarized for θ = 45°:

	Predicted Need	Penticor-Copper Sheet
Flow Rate	0.057 Kgm/m-sec	0.080 Kgm/m-sec
W	0.3 cm	0.32 cm
D	0.12 cm	0.14 cm

Experimental work is continuing with this system, since the flow area and shape are very close to what is needed. The values obtained as of this writing are approximate, and it is clear that the flow area, and therefore flow rate, is not independent of pressure level, since the pressure determines the closeness of fit of the copper sheet to the corrugations.

Flow experiments have also been made using wire mesh as a spacer. Ordinary screenwire presents too much resistance to flow, and a coarser mesh is required. Flow experiments were conducted using 1/8" × 1/8" mesh brass screen between plexiglass sheets. The mean flow area was 13 cm^2. The flow rate measured was 0.15 Kgm/m-sec. This is approximately twice the chosen maximum design value.

The ultimate criterion is cost of the collector per unit of energy collected. As indicated above, the distributed flow collector has the highest performance possible from the point of view of plate temperature distribution; but estimates of cost are difficult to make, since the anticipated useful life of the collector must be known. The PPG collector used an absorber panel made by the Olin Brass Co. In the literature describing their absorber panel, the statement is made that the panel is not guaranteed against corrosion damage, regardless of cause. On the other hand, the Sun-water and Revere collectors use copper tubes for flow passages. The distributed flow panel would probably be constructed of 0.005" copper sheet. This thickness gives adequate strength by a large margin of safety from the point of view of sustaining the compressive force due to the atmospheric-fluid pressure differential. The use of copper is required from the point of view of corrosion. A fair approach to a cost comparison would start with the dollar savings in material alone, taking into account probable life of the collector. Because of the stated no guarantee against corrosion "regardless of cause" in the Olin Brass literature, the materials used in the PPG collector will be weighted with a factor of 2. The following table gives the materials cost comparison. The cost is based on current prices 500 lb lots. The prices per 100 lbs used in the following calculation is:

0.005" copper	\$173.52
0.020" Al	174.00
0.010" copper	161.42

	Material	Wt/ft^2,lbs	Cost, \$/ft^2
Distributed Flow	Copper	0.5	0.87
PPG	Al	0.9 × 2 = 1.8	3.15
Revere	Copper	2.2	3.54
Sun-Water	Copper tubes	1.2	1.93
	Al plate	0.29	+ .50
			2.43

The rates for copper tubes were assumed equal per pound to that for 0.010 sheet copper. The cost of the 0.060" roll bond product was assumed equal to that of the 0.020 aluminum, per pound.

The table shows that a fair approximation to the dollar savings in materials is from \$2.67/ft^2 for the Revere collector to \$1.56 for the Sun-water, with the average material savings being about \$2/ft^2.

On the other hand, the collector efficiency for distributed flow will be about 44%, as compared to 38.4%, as shown for the HR = 750 W/m^2 in Figure 8. Thus, for example, if 800 square feet of collector is needed in a particular installation using fin-tube construction, area required to produce the same amount of energy using distributed flow would be

$$A(\text{dist. flow}) = 38.4/44 \times 800 = 700 \text{ ft}^2$$
$$(\$12/\text{ft}^2)$$

At current prices, this reduction in area required would represent a savings of \$1200. Furthermore, the 700 square feet of distributed flow collector would cost less by 700 × 2 = \$1400, with a total savings for the unit of \$2600.

$$\$1400 = (700 \text{ ft}^2 \times \$2/\text{ft}^2)$$

On a national scale, the savings on the 2000 units to be constructed as demonstration units would be \$5,200,000.

5. REFERENCES

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2. Klein, S. A., Duffie, J. A. and Beckman, W. A., "Transient Consideration of Flat Plate Solar Collectors," ASME J. Engr. Power, 96A (1974).

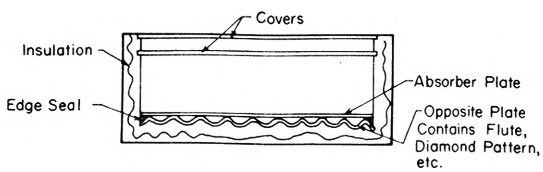

Fig. 1(a) Two plates, one or both being fluted or otherwise roughened to provide a uniform passage for flow of water.

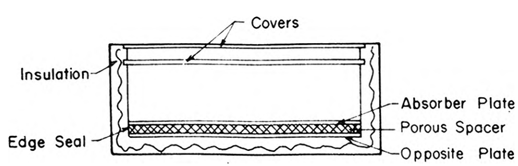

Fig. 1(b) The two parallel plates, with a porous spacer, such as screen wire, between.

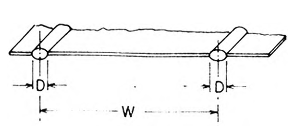

Fig. 2 Model for calculations, tube flow area $= 5.5\ cm^2/m$, $D = 2.65 \times 10^{-2} \sqrt{W}$ as W is varied.

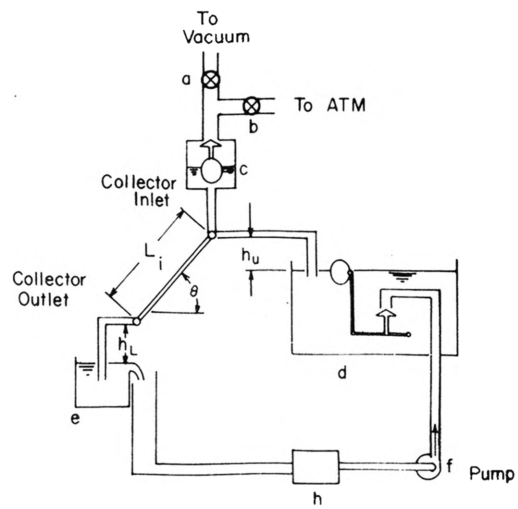

Fig. 3 Flow loop for flow downward through the collector.

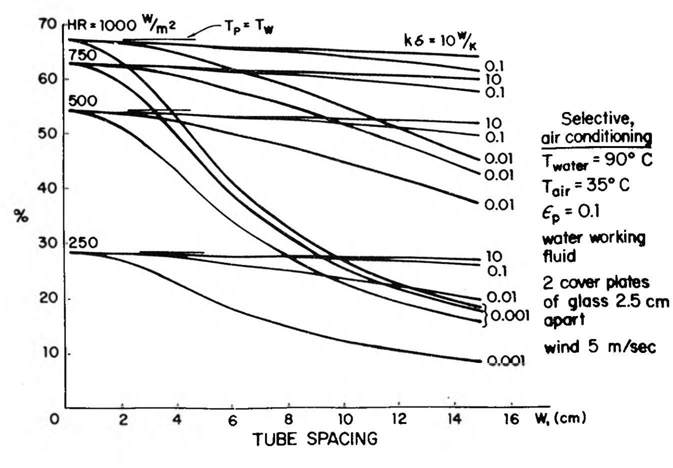

Fig. 4 Efficiency vs tube spacing for various fin conductance and irradiation.

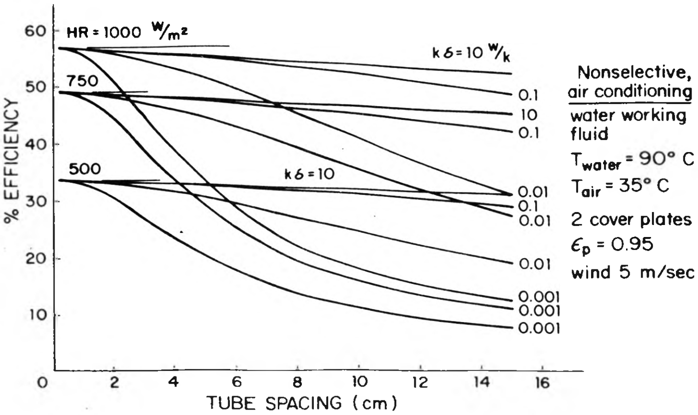

Fig. 5 Efficiency for various conductivities, solar input, and tube spacing.

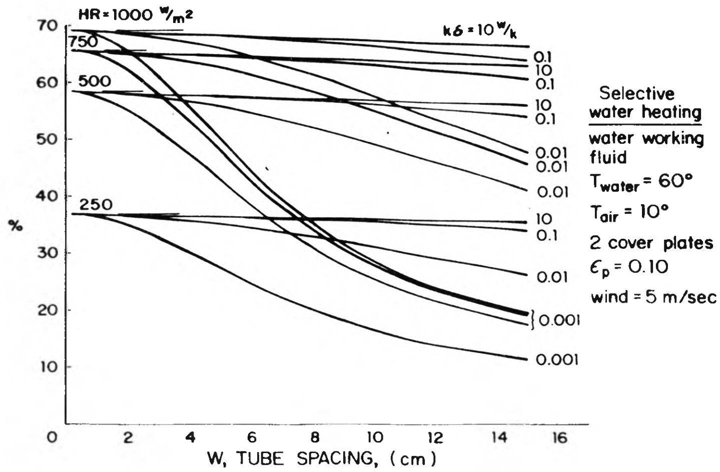

Fig. 7 Efficiency vs tube spacing for various values of conductance and irradiation.

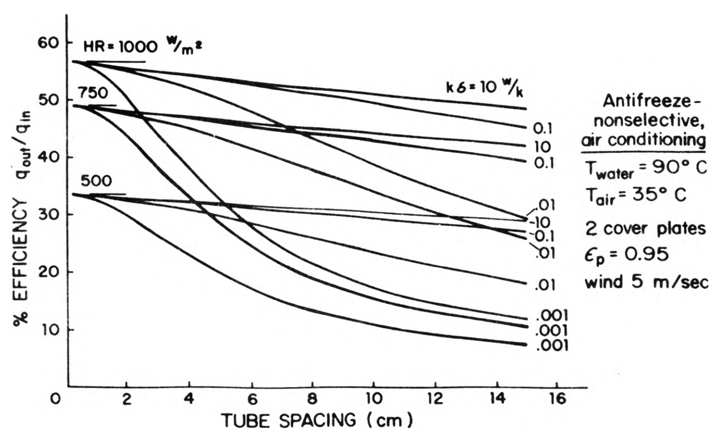

Fig.6 Efficiency for various conductivities, solar input, and tube spacing.

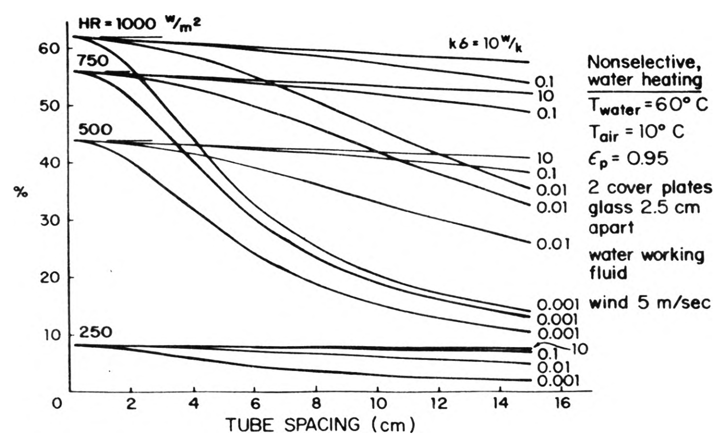

Fig. 8 Efficiency vs tube spacing for various values of kδ and HR.

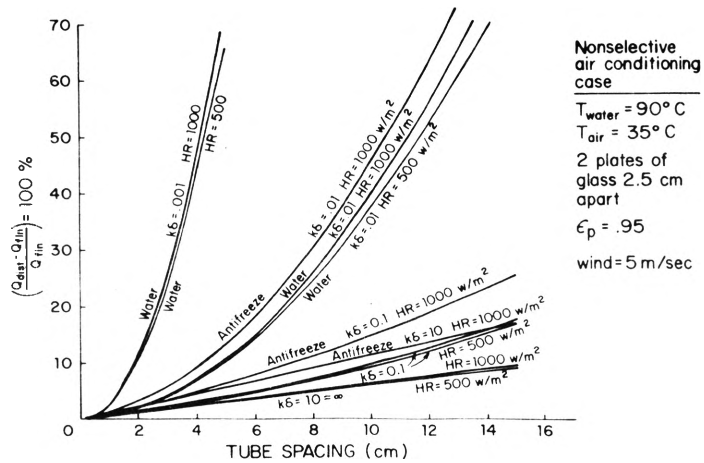

Fig. 9 Relative energy output increase using distributed flow instead of tube flow.

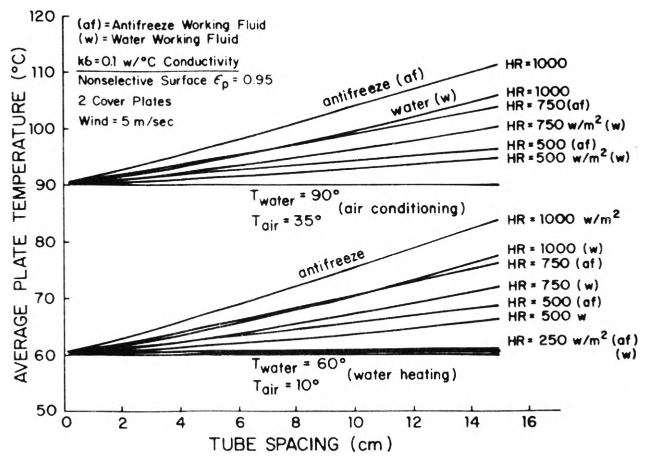

Fig. 11 Average plate temperature vs tube spacing, solar input, water vs antifreeze working fluid.

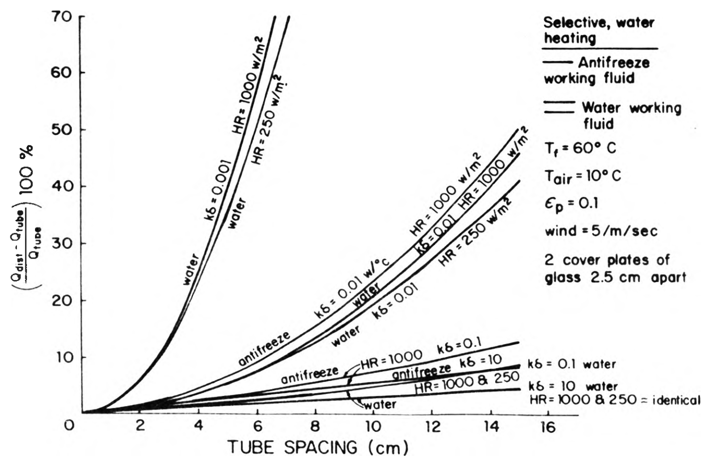

Fig. 10 Relative increase in energy output of distributed flow over tube flow.

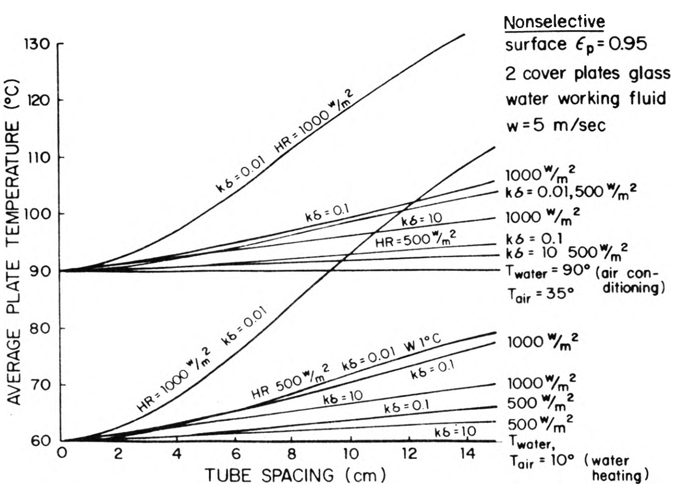

Fig. 12 Average plate temperature vs tube spacing, conductivity, and solar input.