WEIGHT CONSIDERATION IN THE DESIGN OF ABSORBER PLATES

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ABSTRACT

A preliminary investigation (that has to supported later by economic analysis) on the design of an absorber plate for use in a liquid-cooled flat plate solar collector is considered. The objective of the design is to maximize collector efficiency factor, F', *while simultaneously minimizing the plate weight. By varying the plate thickness, tube spacing and tube diameter weight considerations indicate that tube spacing of 100 mm and plate thickness of 0.25 mm give a fairly large* F' *while the usage of 10 mm tubes give a reasonable low pressure loss and high inside tube heat transfer coefficient.*

INTRODUCTION

The greatest source of energy in Ethiopia, as has been mentioned time and again, is biomass and this has absolutely brought a disaster in the deforestation of the country. This trend is still continuing if not in an accelerated rate. The environmental problem that has ensued (draught, erosion) has brought about catastrophic consequences such as famine, decertification etc. To minimize this environmental destruction an alternate energy source must be sought. Solar energy is one of the candidates. Because of the low efficiency of conversion of solar energy into electrical energy, solar energy conversion devices have not become commercially competitive. However solar energy for heating water is being successfully used in conjunction with auxiliary heaters in places where the annual solar radiation is by far lower than in most regions of Ethiopia. So an attempt should be made towards the usage of such heaters wherever mild temperatures below the boiling point are required.

One of the most important factors that has a direct bearing on the actual useful energy gain by a solar collector is called the collector efficiency factor, usually designated by F'. This indicates that one of the factors on which the collection efficiency of the solar collector depends on is F'.

Duffie & Beckmann (1) have shown the variation of collector efficiency factor, F', with tube spacing, using product of thermal conductivity and the thickness of the plate as the parameter. For example, keeping F' constant while increasing the tube spacing requires an increase in plate thickness. However, what is not indicated in these figures is the effect of such changes on weight, heat transfer coefficient, pressure drop, etc. These are the kinds of consequences that will be examined in this paper. Preliminary investigation indicates that F' increases with plate thickness δ(holding the tube spacing, W, constant), but this increase diminishes with further increase in plate thickness resulting in a heavy absorber plate. In other words, for a given spacing, maximum value of F' is reached asymptotically for a given plate thickness. This suggests that a certain compromise must be made between weight and the collector efficiency factor, F'.

It is this information that initiated this investigation and by the end of this investigation, the following optimum design information will be arrived at:

i) tube spacing
ii) plate thickness
iii) diameter of tube from consideration of pressure drop and inside tube heat transfer coefficient.

METHOD OF ANALYSIS

A liquid-cooled absorber plate of the type shown in Fig. 1 is used for this investigation. The tubes and the plate are made of copper. Water is the fluid used for this analysis. The plate thickness (δ), tube spacing (W) and tube diameter (D) will be considered as variables while the plate material, length, width, heat loss coefficient (U_L) and water flow rate are constants.

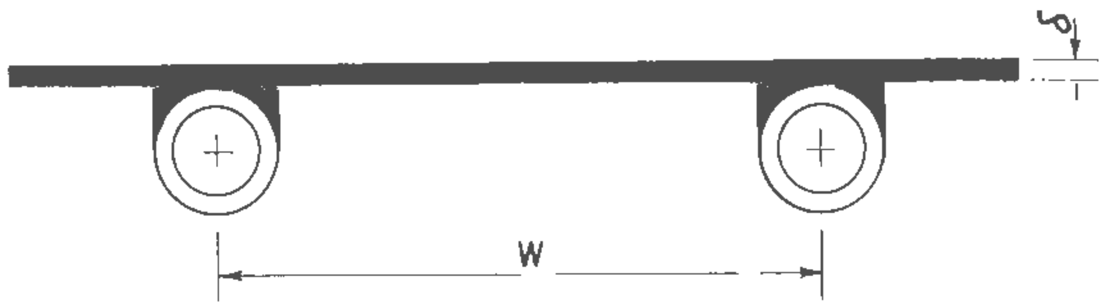

Figure 1 Liquid-cooled Collector Plate and tube Arrangement

The following are very commonly used data for solar water heaters.

Mass flow rate of water = 0.06 kg/s
Heat loss coefficient (U_L) = 4 w/m^2 °C
Width of plate = 0.9 m
Length of plate = 2.00 m

The collector efficiency factor, F', is determined from Eq. (1) according to [1].

$$F' = \frac{\frac{1}{U_L}}{W\left[\frac{1}{U_L[D_o + (W - D_o)F]} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}}\right]} \tag{1}$$

The inside heat transfer coefficient, h_{fi}, is determined from the Nusselt number, Nu, for short tubes (1) given by Eq. (2).

$$N_u = N_{u\infty} + \frac{(Re\ Pr\ D_h/L)^m}{1 + b(Re\ Pr\ D_h/L)^n} = \frac{h_{fi} D_i}{k} \tag{2}$$

The friction factor f, for determination of pressure drop uses Eq.(3) which is for laminar flow in tubes. A check on the Reynolds number (R_e) has indicated that laminar flow prevails.

$$f = \frac{64}{R_e} \tag{3}$$

The pressure drop is determined from Eq.(4) given by

$$\Delta P = f\frac{\rho L}{D}\frac{V^2}{2} \tag{4}$$

The weight of the absorber plate is determined from Eq.(5) shown below.

$$M = \rho_p L[B\delta + \frac{\pi}{4}(D_o^2 - D_i^2)N] \tag{5}$$

(There are several ways of manufacturing the absorber plate-tube assembly and the most representative one considered here is that which does not consider the weight of the bonding material.)

For this investigation, three diameters of tubes of nominal diameters 1/4", 3/8" and 1/2" where used. Particular informations about these tubes were taken from the manufacturers tables, in this case, Anaconda.

A good indicator of the relative increase of F' with change of weight is the ratio shown by Eq.(6).

$$\text{Ratio} = \frac{\text{Change in increase of } F'}{\text{Change in increase of weight}}$$

Using the above relations a computer program was run for

i) tube spacings ranging from the diameter of the tube to 20 cms.

ii) plate thickness ranging from 0.0125 mm to 1 mm.

Table 1: Variation of *F'* with tube spacing and mass for plate thickness of 0.25 mm

Tube Spacing (mm)	1/4"		3/8"		1/2"	
	mass (kg)	F'	mass (kg)	F'	mass (kg)	F'
8	7.95	0.954	10.40	0.958	13.40	0.957
10	6.86	0.940	8.65	0.942	10.90	0.945
12	6.50	0.924	8.07	0.927	10.00	0.929
14	6.13	0.906	7.48	0.910	9.13	0.913
16	5.77	0.889	6.89	0.892	8.93	0.896

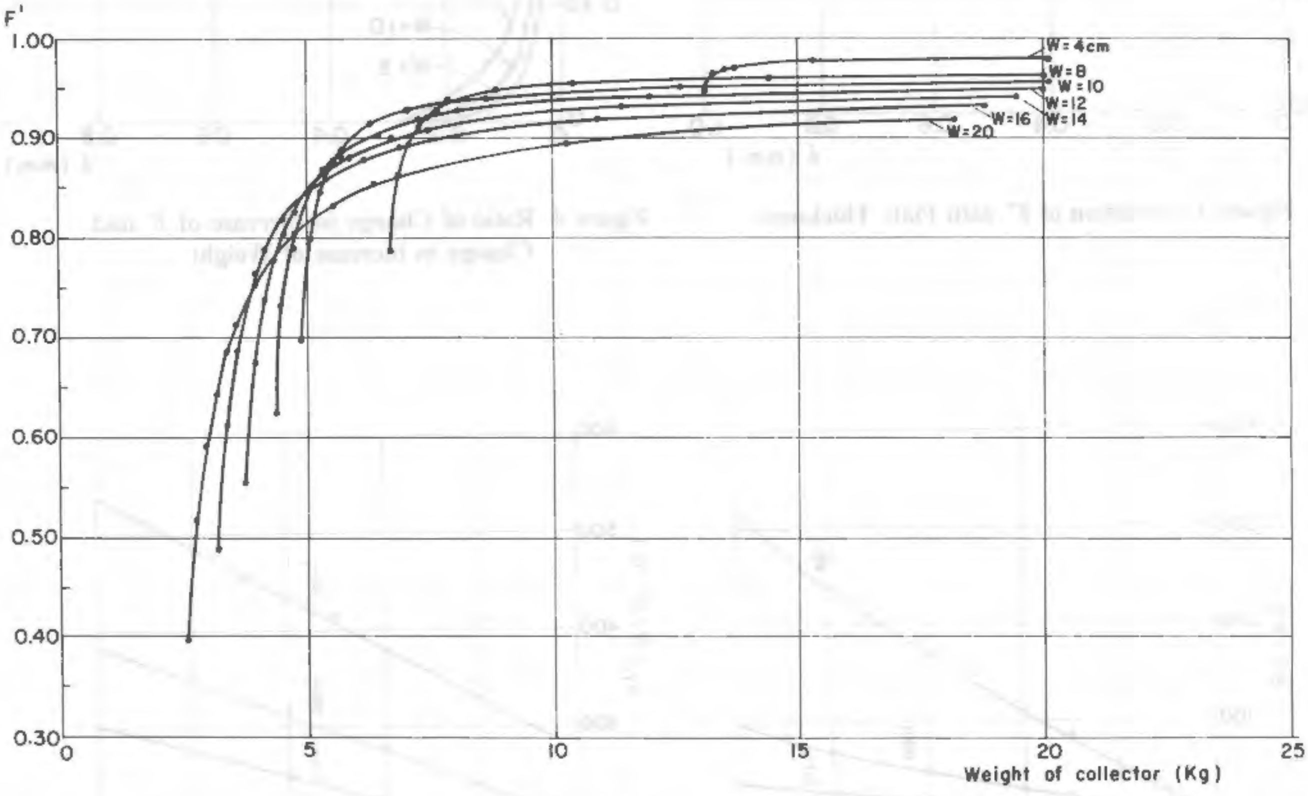

Figure 2 Variation of *F'* with Weight

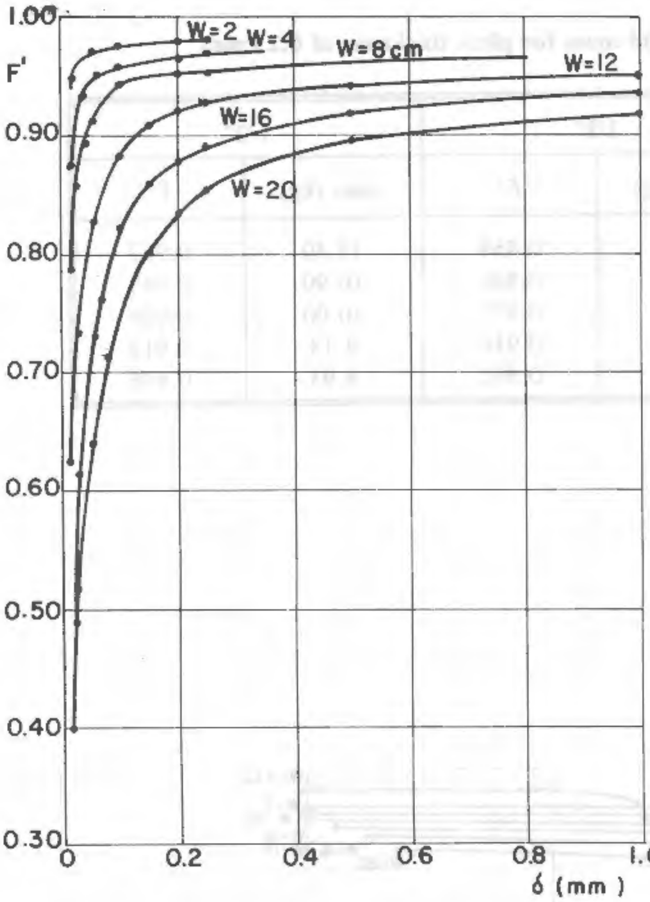

Figure 3 Variation of F' with Plate Thickness

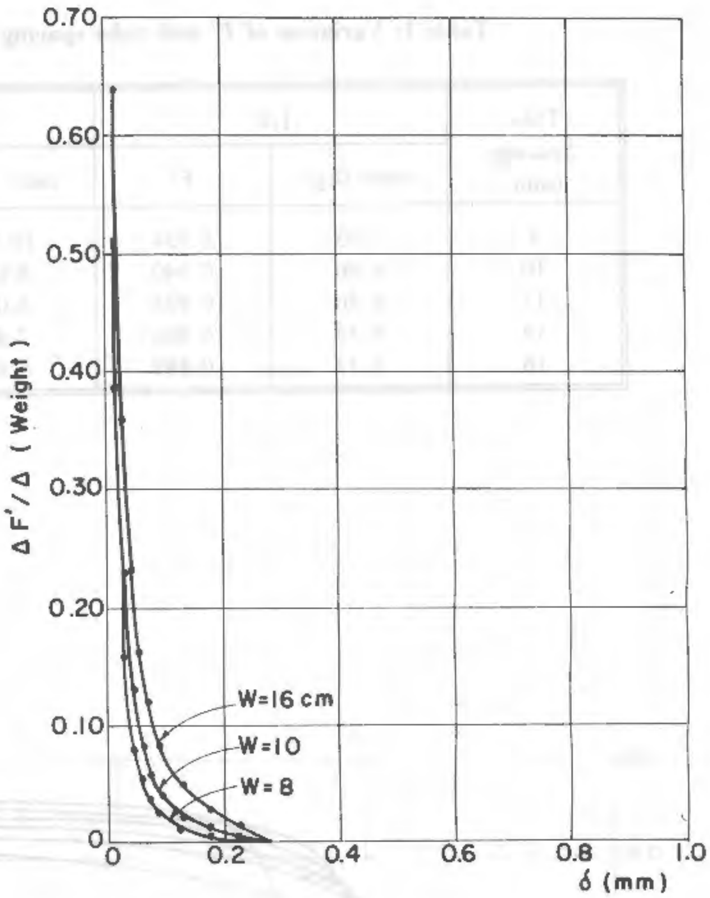

Figure 4 Ratio of Change in Increase of *F'* and Change in increase of Weight

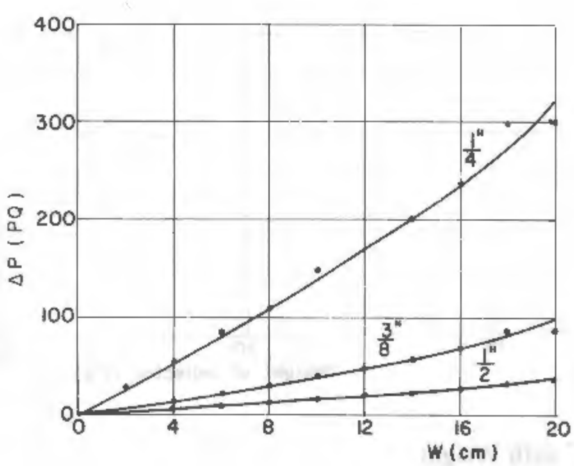

Figure 5 Pressure Loss

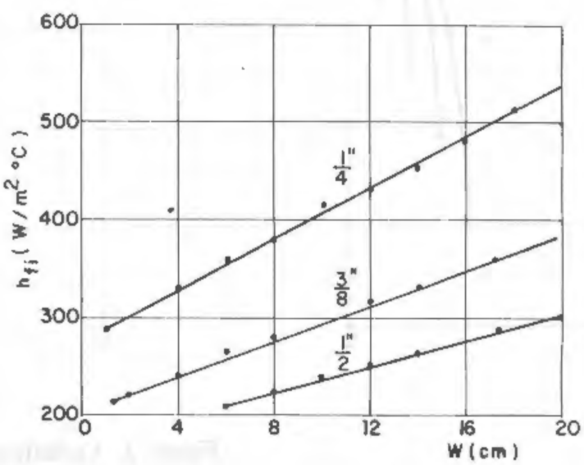

Figure 6 Inside Tube Heat Transfer Coefficient

RESULTS AND DISCUSSIONS

The data generated from execution of the computer program are shown graphically in Fig. 2 through Fig. 6. Fig. 2 is an alternate representation for variation of the collector efficiency factor, F', against weight of collector and the tube spacing as the parameter. The variation of the inside tube heat transfer coefficient, h_{fi}, is taken care of by equation (2). This curve drawn for 3/8" tube conveys the message that for a given spacing, F' increases with increase in weight of the absorber plate. It also clearly shows that the change in increase of F' decreases with increase in weight. This will then call for a compromise between F' and weight of the plate. The same trend is observed for the other tube geometries.

In Fig. 2 the variation of F' with plate thickness and the tube spacing as the parameter clearly show that for a given spacing the increase in F' again diminishes with plate thickness too. To see these diminishing returns the ratio ($\Delta F'/\Delta$ weight) was plotted as shown in Fig. 3. And these curves show that the gain in F' has already reached a zero value for a plate thickness of about 0.28 mm. for a tube spacing of 16 cm and at about 0.25 mm for tube spacings of 8 cm and 10 cm. In the succeeding discussions a tube spacing of 10 cm will be found to be the optimum spacing. With this information the optimum plate thickness is found to be 0.25 mm. With respect to spacing, Fig. 2 shows that F' increases with decrease in spacing for a fixed plate thickness. This will increase the weight of the plate as a larger number of tubes are used. Observation of Fig. 1 shows that:

i) spacings of 4 cms or less do need quite a substantial weight to arrive at their large values of F' while larger spacings with slightly lower F' have significantly low weight.
ii) the 20 cms spacing has low F' compared with the other spacings.

Therefore in the following discussions, the above spacings will not be considered.

The summary of the results of the weights and F's for different spacing and tube diameters is shown in Table 1. For all the three tube diameters, obviously the 8 cm and 16 cm tube spacings will be taken out of consideration due to relatively heavy weight that results in the 8 cm tube spacing and due to relatively low value of F' in the case of 16 cm tube spacing. This leaves us with tube spacing of 10, 12 and 14 cms. Consideration of the increase in F' as compared with increase in weight, there is negligibly small increase in F' while there is substantial increase in weight. To make this point clear, consider the 10 cm tube spacing. There is an increase of 0.21% in F' while the increase in weight is 26.1% between tube sizes of 1/4" and 3/8". This definitely suggests that the small diameter tube i.e. 1/4" tube, ought to be used.

On the other hand, the penalty for using small diameter tubes is the high pressure loss as shown in Fig. 5 and the prize being the high inside heat transfer coefficient as shown in Fig. 6. At first slight it may seem that the high inside heat transfer coefficient may compensate the high pressure loss. But this is not so as can be seen from the following argument. If we take a spacing of 12 cm, the heat transfer coefficient for 1/4" tube is 435 W/m^2 °C while it is 315 W/m^2 °C for the 3/8" tube. With this change in h_{fi}, we only observe an increase of 0.3% in F' as can be seen from Table 1. In addition to this low gain in F', we will also be very careful with large pressure drops especially where fluid circulation is due to gradient in density. So this puts the 3/8" tube in a favorable position. As far as the choice of the tube spacings is concerned, the 10 cms spacing may be selected than the other two spacings due to higher value of F' for a slightly heavier plate.

CONCLUSIONS

For optimum design of liquid-cooled absorber plates, where weight and collector efficiency factor, F', are the concerns, optimum values are determined by a compromise. In our case the compromise is a slight reduction from the maximum F' and this results in a large reduction in weight.

From the trend of increase of F' with plate thickness, δ, it was found out that the increase of F' beyond plate thickness of 0.25 mm was found to be very negligible, for all the three tube sizes used. This, therefore, suggests that 0.25 mm plate thickness is the best choice.

As for the spacing the optimum spacings are found to be 10, 12 and 14 cm for all the three diameters of tubes considered. However the choice goes to the 10 cm spacing due to a relatively large value of F' for a slightly heavier weight.

With respect to the choice on the diameter of the tube, the 1/4" tube would have been the best choice on weight basis. However the penalty on pressure loss and the insignificant increase in F' for large inside tube heat transfer coefficient, h_{fi}, puts it at a disadvantage. The next best choice is 3/8" tube.
To summarize the recommended design values for copper materials are:

Plate thickness = 0.25 mm
Nominal size of tube = 3/8"
Spacing = 100 mm

The above arguments must be supported by economical analysis and that will be the subject matter of another investigation.

REFERENCE

1. John A. Duffie & William A. Beckman: "Solar Engineering Thermal Processes", John Wiley & Sons, 1980.

NOTATIONS

a = Constant
B = Plate Width (m)
b = Constant
C = Conductance (W/m°C)
D = Diameter of tube (mm)
δ = Thickness of plate (mm)
F = Fin Efficiency
F' = Collector efficiency factor
f = Friction factor
h = Convective heat transfer coefficient (W/m^2 °C)
k = Thermal conductivity (W/m°C
L = Length of tube (m)
M = Mass of plate (kg)
m = constant
N = Number of tubes
N_u = Nusselt number
P_r = Prandtl number
ΔP = Pressure drop (Pa)
R_e = Reynolds number
ρ = Density (kg/m^3)
U = Heat loss coefficient (W/m^2 °C)
V = Velocity (m/s)
W = Tube Spacing (mm)

Subscripts

o = outside
i = inside
f_i = inside fluid
L = loss
b = bond
h = hydraulic diameter
p = plate