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KEYWORDS

Solar, Standards, Modeling, Refrigerant, Thermal Response

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BACKGROUND

Soin et al. (1979) noted experimentally that the efficiency of a two-phase thermosiphon appeared to follow a linear relationship, and suggested that an analytical study was needed. Abramzon et al. (1983) numerically solved a set of equations that represent a collector with both boiling and sensible heat transfer, and found that different values of subcooling of the inlet liquid resulted in nearly parallel, linear plots of efficiency versus $(T_i-T_a)/I$. However, no closed form expression was given for efficiency, fluid flow rate, or other features of collector operation. Al-Tamimi (1982) and Al-Tamimi and Clark (1983) developed the following modified Hottel-Willier equation for the efficiency of a boiling collector:

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$$\eta = \mathcal{F}_R[(\tau\alpha) - U_L(T_i-T_a)/I] \ , \tag{1}$$

in which $\mathcal{F}_R$ is a generalized heat removal factor that depends on collector properties, fluid properties, weather parameters, and subcooling. Note that the explicit linear term in Equation 1 contains the difference between inlet and ambient temperatures. Based on the work of Al-Tamimi and Clark, ASHRAE Standard 109P was developed for testing the thermal performance of flat-plate solar collectors containing a boiling liquid (Al-Tamimi and Clark 1984; Spears and Waldin 1984a; Spears and Waldin 1984b; Youngblood 1984). This Standard requires the experimental determination of five separate plots of efficiency versus $(T_i-T_a)/I$, with each plot obtained at specified values of insolation and subcooling. Price, et al.(1985, 1986)) extended the analysis to include the effects of a condenser. From system studies, they concluded that the effect of subcooling on long-term performance would be small. Price (1984) concluded that ASHRAE 109P does not provide a sufficient improvement over Standard 93-77 to be useful.

For a collector with sensible cooling, the efficiency equation is

$$\eta = F_R[(\tau\alpha) - U_L(T_i-T_a)/I] \tag{2}$$

(Duffie and Beckman 1980). For a fixed circulation rate, F_R is constant. Therefore, the single plot of η versus $(T_i-T_a)/I$ as specified by ASHRAE 93-77R in effect measures two constants, $F_R(\tau\alpha)$ and F_RU_L. Knowledge of these constants provides some understanding of the properties of the collector and may guide the manufacturer in improving the collector if he wishes to do so. In contrast, η for a boiling collector is a complicated function of many parameters, including subcooling. At zero subcooling, $\mathcal{F}_R = F_b'$. Therefore the collector efficiency plot at zero subcooling specified by ASHRAE 109P would in effect measure $F_b'(\tau\alpha)$ and $F_b'U_L$ and might thereby reveal some of the general properties of the collector (although F_b' is not necessarily constant). The other plots required by ASHRAE 109P at specific values of insolation and subcooling permit comparison of different collectors at the specified test conditions, but do not provide means for predicting collector performance under other conditions. The purpose of this paper is to show that Equation 1 may be rearranged so as to enable a more intuitive understanding of collector behavior and so as to indicate the situations in which testing with various values of subcooling and insolation may not be necessary.

DEVELOPMENT OF THE EFFICIENCY EQUATION

In this paper, the collector is assumed to provide saturated vapor to an external condenser (or engine) whose properties establish the temperature (or pressure) at which the vapor will move out of the collector. The variation of boiling temperature due to hydrostatic head within the collector is ignored (Al-Tamimi 1982). The desired results could be obtained by manipulation of the equations of Al-Tamimi and Clark (1983). However, derivation following first principles (Duffie and Beckman 1980) is outlined here in order to provide clarity and consistent notation.

We regard a fractional length of the collector, z^*, as having sensible heat transfer in which the liquid is warmed from the inlet temperature, T_i, to the saturation (boiling) temperature, T_b. In the subsequent fractional length, $(1-z^*)$, boiling heat transfer to the two-phase fluid occurs at constant temperature T_b. The rate of sensible heating of the liquid is

$$\dot{m}C_p(T_b-T_i) = A_c z^* F_{R,nb}[S - U_L(T_i-T_a)] , \tag{3}$$

in which the non-boiling heat removal factor is

$$F_{R,nb} = \frac{\dot{m}C_p}{A_c z^* U_L}\left[1 - \exp\left(\frac{-z^* A_c U_L F'}{\dot{m}C_p}\right)\right] . \tag{4}$$

The rate of latent heat transfer to the fluid is given by

$$\dot{m}L = A_c(1-z^*)F_b'[S - U_L(T_b-T_a)] , \tag{5}$$

in which F_b' appears explicitly because the fluid temperature is assumed to be constant in the boiling portion of the collector. At this point, Equations 3 and 5 could be utilized to solve for m and z^*. However, we will first make two important substitutions. The stagnation temperature, T_s, is defined by

$$T_s - T_a = S/U_L . \tag{6}$$

The dimsionless subcooling (or temperature rise), x, is defined as the ratio of the subcooling to the difference between stagnation and inlet temperatures:

$$x = (T_b - T_i)/(T_s - T_i) \ . \tag{7}$$

Note that $x=0$ when there is no subcooling, and $x=1$ when the boiling temperature equals the stagnation temperature. The quantity x is a measure of the temperature rise of the liquid, as a fraction of the temperature rise that would occur if the boiling temperature were increased until the flow stopped.

From Equations 3, 4, 6, and 7, we find

$$F_{R,nb} = -xF'/\ln(1-x) \ , \tag{8}$$

and Equation 3 can be solved for z^*:

$$z^* = -\frac{\dot{m}C_p}{A_c U_L} \frac{\ln(1-x)}{F'} \ . \tag{9}$$

Equations 5, 6, 7, and 9 can be combined to give a dimensionless flow rate:

$$\frac{\dot{m}C_p}{A_c U_L} = \frac{F_b'}{\frac{L}{C_p(T_s - T_i)} \frac{1}{1-x} - \frac{F_b'}{F'} \ln(1-x)} \ , \tag{10}$$

which can in turn be substituted into Equation 9 with the result

$$z^* = \frac{1}{\frac{L}{C_p(T_s - T_i)} \frac{1}{(x-1)\ln(1-x)} \frac{F'}{F_b'} + 1} \ . \tag{11}$$

Finally, with substitution of Equations 6 and 7, Equations 3 and 5 can be added to form the rate of total useful energy yield per unit area of the collector:

$$q_u/A_c = [z^* F_{R,nb} + (1-z^*)(1-x)F_b'][S - U_L(T_i - T_a)] \tag{12}$$

$$= \mathcal{F}_R [S - U_L(T_i - T_a)] \ . \tag{12a}$$

It can be shown that the first term in brackets on the right-hand side of Equation 12 is equal to $\mathcal{F}_R$. At this point, we have simply expressed $\mathcal{F}_R$ as a function of the independent parameter, x. When Equations 8, 9, and 11 are substituted into Equation 12, we find

$$\frac{q_u}{A_c} = F_b' \frac{\left(\frac{L}{C_p(T_b-T_i)} + 1\right)\left(1 - x\right)}{\frac{L}{C_p(T_b-T_i)} + \frac{F_b'}{F'}\frac{x-1}{x}\ln(1-x)} \left[S - U_L(T_i-T_a)\right] . \tag{13}$$

Note that

$$(1-x) = \frac{S - U_L(T_b-T_a)}{S - U_L(T_i-T_a)} . \tag{14}$$

Upon substitution of Equation 14 into Equation 13, a final expression for efficiency results:

$$\eta = q_u/A_c I = F_b' E_f [(\tau\alpha) - U_L(T_b-T_a)/I] , \tag{15}$$

in which

$$E_f = \frac{\frac{L}{C_p(T_b-T_i)} + 1}{\frac{L}{C_p(T_b-T_i)} + \frac{F_b'}{F'}\frac{x-1}{x}\ln(1-x)} . \tag{16}$$

Note that the independent variable in Equation 15 is $(T_b-T_a)/I$, whereas ASHRAE 109P is based on Equation 1 in which the independent variable is $(T_i-T_a)/I$. The remainder of this paper shows that the product $F_b'E_f$ of Equation 15 is usually nearly constant, and that therefore considerable simplification in a test procedure can be achieved if the independent variable is based on the saturation temperature rather than on the inlet temperature. Physically this is because the latent heat gain is usually much larger than the sensible heat gain, and the efficiency is much more sensitive to saturation temperature than to inlet temperature. An equation very similar to Equation 15 was derived and subjected to limited experimental verification by Kishore et al. (1984a; 1984b). In their derivation, they assumed that $F_b'=F'$, and they used an approximate form of Equation 15 in which the behavior of E_f is less evident than it is in Equation 16. However, the investigations by Kishore and colleagues included the effects of superheating, which we ignore here because most solar systems with boiling collectors are not intended to produce superheated vapor.

BEHAVIOR OF THE DIMENSIONLESS TERMS

Because F_R itself depends on many parameters, expression of efficiency in the form of Equation 1 does not permit the impacts of weather, collector characteristics, and operating parameters to be examined independently. Equations 9, 10, and 16 were developed in terms of the dimensionless temperature rise, x, so as to make several aspects of collector behavior more easily understood. Figure 1 shows the behavior of the nonboiling fractional length, z^*, and the dimensionless flow rate as functions of the subcooling ratio, x. For fixed (T_s-T_i), z^* increases as the subcooling is increased from zero, as we might expect. Without the benefit of these calculations, we might also expect that the nonboiling fractional length, z^*, would approach unity whenever the subcooling approached the stagnation temperature difference (whenever x approached unity). However, as the subcooling is made larger (as boiling temperature is made to approach stagnation temperature), z^* decreases because the flow rate decreases. Thus, we see that a subcooling ratio near unity does not necessarily force most of the collector length to operate in the nonboiling mode. Rather, Figure 1 and Equation 11 indicate that z^* will approach unity only if (T_s-T_i) becomes large compared to L/C_p, and that the maximum value of z^* always occurs where x is equal to 0.632.

Physically, these results can be understood as follows. If the boiling temperature is close to the inlet temperature (if x is small), most of the collector length is involved in boiling heat transfer (z^* is small) as shown in Figure 1. If the boiling temperature is somewhat increased, the flow rate decreases and the fraction of collector required to bring the fluid to boiling initially increases. If the boiling temperature is further increased to nearly the stagnation temperature, the flow rate decreases almost to zero. Due to the low flow rate, only a small fraction of the collector is again required to bring the fluid to the boiling temperature. Most of the collector is again involved in boiling heat transfer, but the latent energy gain is small because the boiling portion of the collector is at a temperature close to stagnation, and the absorbed solar energy in this major portion of the collector is largely lost. Because the nonboiling fractional length approaches zero as the boiling temperture approaches either the inlet temperature or the stagnation temperature, the nonboiling fraction must reach a maximum at some intermediate temperature. Furthermore, this maximum approaches unity only if L/C_p is sufficiently small.

Algebraically, the collector loss coefficient and the weather conditions have been lumped into the stagnation temperature. Equations 10 and 11 show that for a given collector, fluid, and stagnation temperature, the flow rate and nonboiling fraction are functions of x only. The nonboiling fraction is a maximum when the quantity $(x-1)\ln(1-x)$ is a maximum, which occurs at $x = 0.632$. Various functions of x are plotted in Figure 2 for use in visualizing the behavior of various terms.

Under most conditions of significant energy output, the collector operates in the boiling mode over most of its length (z^* is small), and warming of the subcooled liquid consumes a minor fraction of the collected energy. This physical fact corresponds to the fact that E_f of Equation 15 is usually close to unity. E_f depends on the ratio of boiling efficiency factor to non-boiling efficiency factor, F_b'/F'; on the subcooling ratio, x; and on the ratio $L/C_p(T_b-T_i)$. Figures 3-5 show lines of constant E_f in the space of two variables, with F_b'/F' as a parameter. The ratio F_b'/F' is nearly constant for a given collector. These contour plots show that E_f is nearly constant over a wide range of collector operation. If E_f is nearly constant, a plot of efficiency vs $(T_b-T_a)/I$ should closely approximate a single straight line under all conditions. Calculations are presented below to test that approximation.

It should be noted that when x is small, (T_b-T_i) is also relatively small, so that collector operation does not occur in the lower left-hand corner of Figures 3-5. Figures 3-5 also show that as F_b'/F' is increased, the spacing between the coutours of E_f becomes smaller, permitting E_f to depart farther from unity.

The entries in Table 1 for R-11 fluid show that $L/C_p(T_b-T_i)$ will be greater than 10 for operating temperatures up to 188 F (87°C) and subcooling up to 29 F (16°C). For many collectors, F_b'/F' will be approximately 1.2, as represented by Figure 4. Thus, for space- or water-heating applications of many collectors using R-11, Figure 4 shows that E_f will not deviate from unity by more than 5% unless x is greater than 0.8, which would then imply that the saturation temperature is close to the stagnation temperature. R-12 is usually unsuitable for solar systems due to its low critical temperature. Of the other refrigerants listed in Table 1, R-114 has the lowest values of L/C_p and therefore offers the greatest potential for variation of E_f. In most space- and water-heating applications, $L/C_p(T_b-T_i)$ for R-114 would be greater than 5, and E_f would deviate from unity by at most 10%. Therefore, tbe assumption that E_f is a constant equal to 1.0 in Equation 15 will usually be accurate to $\pm$ 10%.

COLLECTOR EFFICIENCY

Because the product $F_b'E_f$ remains nearly constant under the circumstances of collector testing (whether or not it is close to unity), it is attractive to consider a test procedure based on Equation 15. If $F_b'E_f$ is nearly constant, data points representing various degrees of subcooling and insolation should form a single line on a plot of η versus $(T_b-T_a)/I$. In this case, testing at multiple values of insolation and subcooling would not be necessary. Table 2 presents the assumed properties of one actual and three hypothetical collectors for which the efficiency was calculated allowing variation of all parameters. Because F_b' and F' depend on F, U_L, and on the ratio of fluid heat transfer area to plate area, F_b' and F' were calculated for each point of numerical data as explained in the appendix. The thermal properties of the fluid were also varied according to temperature and type of refrigerant. Collector efficiency was calculated with the values of T_b, I, and subcooling shown in Table 3.

Collector A represents a commercial flat-plate unit used at the author's laboratory as part of a downward-acting passive transport system (Neeper and Hedstrom 1985). For this collector, F_b'/F' was approximately 1.2 over the range of calculated conditions, which leads us to expect from Figure 4 that E_f should be nearly constant. The minimum and maxiumum values of E_f that occurred during the calculations for Collector A were 0.98 and 1.12. As $(T_b-T_a)/I$ increased, E_f increased slightly while F_b' decreased, causing the product to decrease. Figure 6 shows the calculated efficiency plotted as a function of $(T_i-T_a)/I$, as prescribed by ASHRAE 109P. The values of subcooling and insolation are more extreme than required by ASHRAE 109P. The three lines of Figure 6 are horizontally displaced from each other by $(T_b-T_i)/I$, as expected if $F_b'E_f$ were constant in Equation 15. Figure 7 is a similar plot at lower insolation and higher saturation temperature, T_b. The line for zero subcooling is nearly identical to the corresponding line of Figure 6, indicating that the change in fluid properties with temperature had little effect. In contrast to Figures 6 and 7, which illustrate data as prescribed by ASHRAE 109P, Figure 8 presents the efficiency calculated at various values of insolation and subcooling, plotted against

$(T_b-T_a)/I$ as suggested by the form of Equation 15. It can be seen that the magnitude of the systematic deviation of the points from a single straight line is less than or similar to the magnitude of the random scatter to be expected in an actual experimental test. Calculations using other values of T_b are very close to the points shown in Figure 8. It can be seen that the efficiency at various values of saturation temperature, insolation, subcooling, and ambient temperature behaves in practice as a single linear function of $(T_b-T_a)/I$. Therefore, little would be learned by testing Collector A at multiple values of insolation and subcooling as required by ASHRAE 109P.

Each point of Figure 9 indicates the average of the calculated data generated by three values of T_b, four values of I, and five values of subcooling. The data for all of the efficiency plots were generated according to Equation 15 at intervals of $(T_b-T_a)/I$ that represented evenly spaced fractions of the stagnation value. When the subcooling was greater than (T_b-T_a) at a particular point, no data could be generated. Consequently, a varying number (between 23 and 56 inclusively) of data points entered the average to form each point of Figure 9. Although the data being averaged did not constitute a random statistical distribution, the standard deviation of each average was computed in order to indicate the spread of the data around the average. The standard deviation of the data is indicated in Figure 9 by the vertical extent of each symbol along the line of the graph. Because R-11 and R-114 represent the extremes of L/C_p in Table 1, we conclude from Figure 9 that the efficiency of Collector A is insensitive to the choice of refrigerant at temperatures between 63 and 189 F (17 and 87°C).

EXTRAORDINARY COLLECTORS

Figures 3-5 show that E_f becomes more sensitive to subcooling as the ratio F_b'/F' is increased. This ratio is maximized by a large tube-to-plate bond conductance, by a large coefficient of boiling heat transfer, and by F=1. Under these maximizing conditions,

$$\frac{F_b'}{F'} = 1 + \frac{U_L W}{D_i h} = \frac{\text{heat loss rate per tube}}{\text{heat transfer rate to fluid per tube}}, \qquad (17)$$

in which h is the coefficient for sensible heat transfer to the liquid. It can be seen, therefore, that F_b'/F' will be largest and consequently the sensitivity of E_f will be greatest for the maximum values of W/D_i and U_L, and for the minimum value of h. To maximize this sensitivity for Collectors A-C, the Nusselt number for sensible heat transfer was chosen to be 4.0, near its minimum possible value (Duffie and Beckman 1980, p. 134). For hypothetical Collector B, W/D_i was assigned double the value for Collector A, and U_L was also doubled to 1.76 Btu/ft^2 hr F (10 W/m^2 °C). This large value of U_L might occur for a single-glazed collector with flat black absorber (Duffie and Beckman 1980, p. 208). Indeed, Figure 10 shows that, for Collector B, different degrees of subcooling result in slightly separated efficiency plots, indicating that the product $F_b'E_f$ is not effectively constant as it was for Collector A. Figure 11 presents average data and standard deviations for Collector B, which may be compared to the similar data shown in Figure 9 for Collector A. Each point of Figure 11 represents the average of at least 17 and at most 50 individual points. The straight line is drawn through the end points so as to reveal the systematic departure of the data from linearity. The standard deviations of Figure 11 and the spacing between the lines of Figure 10 are sufficiently small that they might be within the errors of an actual experimental test. Thus, for a collector such as B with unusually large U_L, the tests prescribed by ASHRAE 109P might or might not reveal the actual small dependence of $F_b'E_f$ on subcooling and insolation.

Although ASHRAE 109P was probably not intended to apply to unglazed collectors, it is interesting to explore the conditions under which $F_b'E_f$ of Equation 15 might vary sufficiently with subcooling or insolation so as to require multiple tests. Collector C represents an extreme case, with U_L = 2.64 Btu/ft^2 hr F (15 W/m^2 °C). This might represent an unglazed collector. For this collector, F_b'/F' varied between 2.6 and 5.5 as operating conditions changed, and E_f varied between 0.62 and 1.0. Experimental tests should be able to measure the relatively large effect of subcooling on the efficiency of Collector C, as shown in Figure 12. The results of calculations (not shown) with insolation of 149 Btu/ft^2 hr (470 W/m^2) are almost identical to the lines of Figure 12, indicating that E_f is insensitive to insolation. Therefore, tests at multiple values of insolation would reveal little information.

The assumed dependence of h_b on heat flux at the tube wall (Du Pont Inc. undated) in principle causes lower values of F_b', and thus lower efficiency, under conditions of low heat flux. This may be why extrapolations of some of the linear plots for Collectors B and C intercept the abscissa prior to the theoretical stagnation point. However, in the range of fluxes useful for the significant collection of energy, the dependence of F_b' on the heat flux is not sufficient to cause noticeable curvature of the lines.

According to Al-Tamimi (1982), the Nusselt number for sensible heat transfer should usually be close to 6. In the calculations for Collectors A-C, the Nusselt number was assumed to be 4.0 in order to accentuate the dependence of E_f on subcooling. The sensitivity of efficiency to Nu would be greatest for the collector with the largest U_L. Consequently, Collector D was chosen to have the same extreme values of W/D_i and U_L as Collector C, but Nu was increased to 6. The increase of Nu from the mimimum possible value of 4.0 to the expected value of 6.0 reduced the sensitivity to subcooling by approximately half (not shown in the graphs). This again indicates that testing at multiple values of subcooling should seldom be required.

CONCLUSIONS

It has been shown that for a conventional flat-plate collector with U_L near 0.88 Btu/ft^2 hr F (5 W/m^2 °C), the efficiency may be approximated as a single linear function of $(T_b-T_a)/I$, with the leading coefficient insensitive to insolation or subcooling. This suggests that testing at multiple values of insolation and subcooling is unnecessary. If U_L is approximately 1.76 Btu/ft^2 hr F (10 W/m^2 °C) (which would be unusual for a glazed collector), then the linear approximation may become sufficiently sensitive to subcooling that testing at one non-zero value of subcooling might provide useful information. Even an extreme loss coefficient of 2.64 Btu/ft^2 hr F (15 W/m^2 °C) does not cause the linear approximation to become sensitive to insolation in the range 149-251 Btu/ft^2 hr (470-790 W/m^2). Therefore, testing at multiple values of insolation may not be necessary in any case.

Elements of an efficiency test less elaborate than that specified by ASHRAE 109P are therefore suggested as follows. With insolation > 251 Btu/ft^2 hr (750 W/m^2) and zero subcooling, the efficiency is measured and plotted as a function of $(T_b-T_a)/I$. A single data point with subcooling of 27 F (15°C) is subsequently measured under conditions with $(T_b-T_a)/I$ less than 60% of the stagnation value inferred by linear extrapolation of the plot obtained with zero subcooling. If this point deviates from the plotted data by more than 10% (that is, if $\{\eta(0) - \eta(15)\}/\eta(0) > 0.1$), then a complete set of efficiency data at 27 F (15°C) subcooling should be obtained. If any data set does not form a suitably straight line when plotted as a function of $(T_b-T_a)/I$, the heat transfer within the collector may be sensitive to heat flux, and a complete test per ASHRAE 109P should be conducted.

The above paragraph is intended as a broad suggestion, not as a precise specification of procedure. A testing procedure based on this suggestion could reveal those collectors for which the product $F_b'E_f$ is sensitive to subcooling and/or insolation, while not requiring unnecessary tests for the majority of collectors. Whether the suggested procedure can be modified to include a collector with integral condenser has not yet been investigated.

NOMENCLATURE

A_c	Collector area.
C_p	Specific heat of the liquid working fluid.
D_i	Internal diameter of a tube of a fin-tube flat-plate collector.
E_f	Factor defined by Equation 16 that relates efficiency to subcooling.
F	Fin efficiency factor.
F'	Collector efficiency factor for the nonboiling portion of the collector, in which only sensible heat transfer is assumed to occur.
F_b'	Collector efficiency factor for the boiling portion of the collector, in which only boiling heat transfer is assumed to occur.
F_R	Heat removal factor for a collector with sensible cooling.
$F_{R,nb}$	Heat removal factor for the nonboiling portion of the collector.
$\mathcal{F}_R$	Generalized heat removal factor for the efficiency expression based on inlet temperature.
h	Coefficient of heat transfer from tube wall to nonboiling liquid.
h_b	Coefficient of heat transfer from tube wall to the boiling fluid.
I	Insolation (power per unit area) incident on the collector.
L	Latent heat of vaporization of the working fluid.
$\dot{m}$	Time rate of mass flow of working fluid through the collector.

Nu	Nusselt number for heat transfer to the liquid.
q_u	Total useful energy yield of the collector per unit time.
S	Solar radiation absorbed per unit time per unit area. S = ( )I.
T_a	Ambient temperature.
T_b	Saturation (boiling) temperature of the working fluid.
T_i	Temperature of the liquid at the inlet of the collector.
T_s	Stagnation temperature of the collector.
U_L	Collector loss coefficient.
W	Spacing between centerlines of tubes of the absorber plate.
x	Dimensionless subcooling ratio, $(T_b-T_i)/(T_s-T_i)$.
z^*	Fraction of the collector length in the nonboiling state.
η	Thermal efficiency of the collector.
$\eta(\Delta T)$	Thermal efficiency of the collector at a particular subcooling, T.
$(\tau\alpha)$	Transmittance-absorptance product.

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APPENDIX A: DETAILS OF HEAT TRANSFER MODELING

Linear approximations were used to represent L and C_p as functions of temperature. A piecewise linear approximation for h_b as a function of heat flux was used for all refrigerants, based on Du Pont data for R-114 (Du Pont

Inc. undated). These data may not represent the several modes of boiling in a tube and the dependence of h_b on tube diameter, as given by more elaborate correlations (Al-Tamimi 1982). However, over the limited range of heat fluxes measured by Al-Tamimi (1982), the Du Pont data for R-114 approximately agree with measurements using R-11 in a solar collector. Therefore, the Du Pont data for h_b were used for all refrigerants in this study. T_b, I, and $(T_b-T_a)/I$ were established before calculating each value of efficiency, with the consequence that T_a occasionally had an unrealistic value. C_p was calculated at the average of T_b and T_i. Collector efficiency was calculated in an iterative loop in which h_b was adjusted according to the heat flux until the change in F_b'/F' was $< 10^{-3}$.

TABLE 1. RATIO OF LATENT HEAT TO SPECIFIC HEAT+

Refrigerant		11		12		113		114	
T(F)	T(°C)	L/C_p (units of temperature difference)							
		F	°C	F	°C	F	°C	F	°C
62.2	16.8	378	210	270	150	292	162	230	128
98.2	36.8	355	197	236	131	274	152	205	114
188.2	86.8	292	162	123	68.5	227	126	146	81.2
224.2	106.8	265	147	0	0	207	115	120	66.9

+Based on data from the 1981 ASHRAE Handbook of Fundamentals.

TABLE 2. ASSUMED PROPERTIES OF COLLECTORS

Collector	$(\tau\alpha)$	F	U_L †		D_i		W		Nu
			Btu	W	in	m	in	m	
A	0.81	0.98	0.88	5.0	0.374	0.0095	2.00	0.0508	4
B	0.81	1.0	1.76	10.0	0.394	0.01	3.94	0.10	4
C	0.81	1.0	2.64	15.0	0.394	0.01	7.87	0.20	4
D	0.81	1.0	2.64	15.0	0.394	0.01	7.87	0.20	6

†Btu/ft^2 hr F or W/m^2 °C.

TABLE 3. VALUES OF SATURATION TEMPERATURE, INSOLATION, AND SUBCOOLING USED IN CALCULATING EFFICIENCY

T_b		I †		$(T_b - T_i)$	
F	°C	Btu	W	F	°C
62.2	16.8	317	1000	0	0
98.2	36.8	251	790	10.8	6
188.2	86.8	149	470	27.0	15
		63.5	200	43.2	24
				59.4	33

†Btu/ft^2 hr or W/m^2.

FIGURE CAPTIONS

Fig. 1. Non-boiling fractional length of collector and dimensionless flow rate as functions of x for two values of $L/C_p(T_s-T_i)$.

Fig. 2. Three functions of x.

Fig. 3. Contours of E_f for $F_b'/F' = 1.0$.

Fig. 4. Contours of E_f for $F_b'/F' = 1.2$, as occurs for many flat-plate collectors.

Fig. 5. Contours of E_f for $F_b'/F' = 2.0$.

Fig. 6. Efficiency of Collector A versus $(T_i-T_a)/I$ with I = 317 Btu/ft^2 hr (1000 W/m^2) and T_b = 62.2 F (16.8°C).

Fig. 7. Efficiency of Collector A versus $(T_i-T_a)/I$ with I = 149 Btu/ft^2 hr (470 W/m^2) and T_b = 98.2 F (36.8°C).

Fig. 8. Efficiency of Collector A versus $(T_b-T_a)/I$ with T_b = 62.2 F (16.8°C).

Fig. 9. Efficiency of Collector A averaged over saturation temperatures, insolation, and subcooling. Data are shown for R-11 and R-114 fluids.

Fig. 10. Efficiency of Collector B versus $(T_b-T_a)/I$ with I = 251 Btu/ft^2 hr (790 W/m^2) and T_b = 98.2 F (36.8°C).

Fig. 11. Efficiency of Collector B averaged over saturation temperatures, insolation, and subcooling.

Fig. 12. Efficiency of Collector C versus $(T_b-T_a)/I$ with I = 251 Btu/ft^2 hr (790 W/m^2) and T_b = 98.2 F (36.8°C).

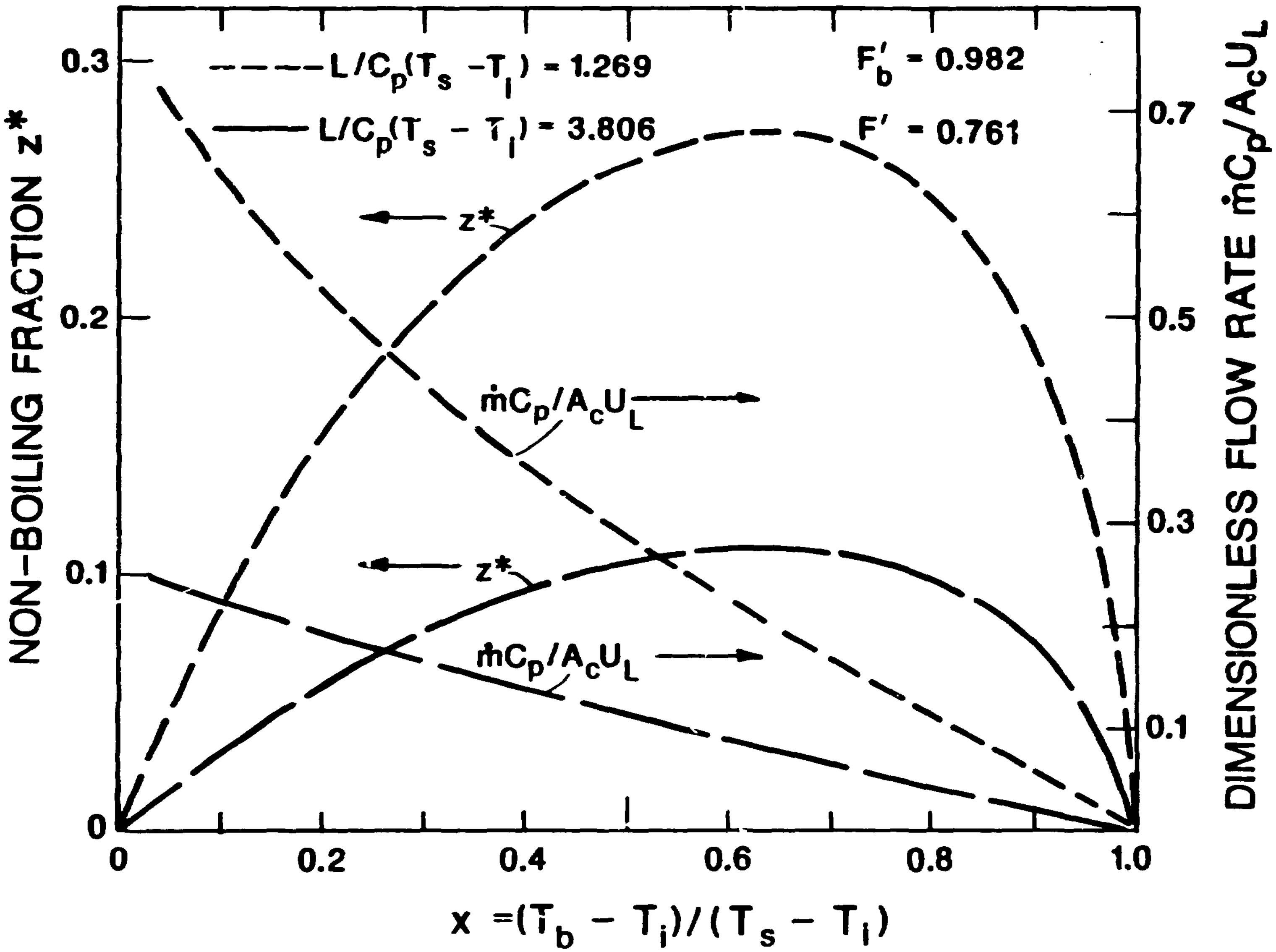


Fig. 1. Non-boiling fractional length of collector and dimensionless flow rate as functions of x for two values of $L/C_p(T_s-T_i)$.

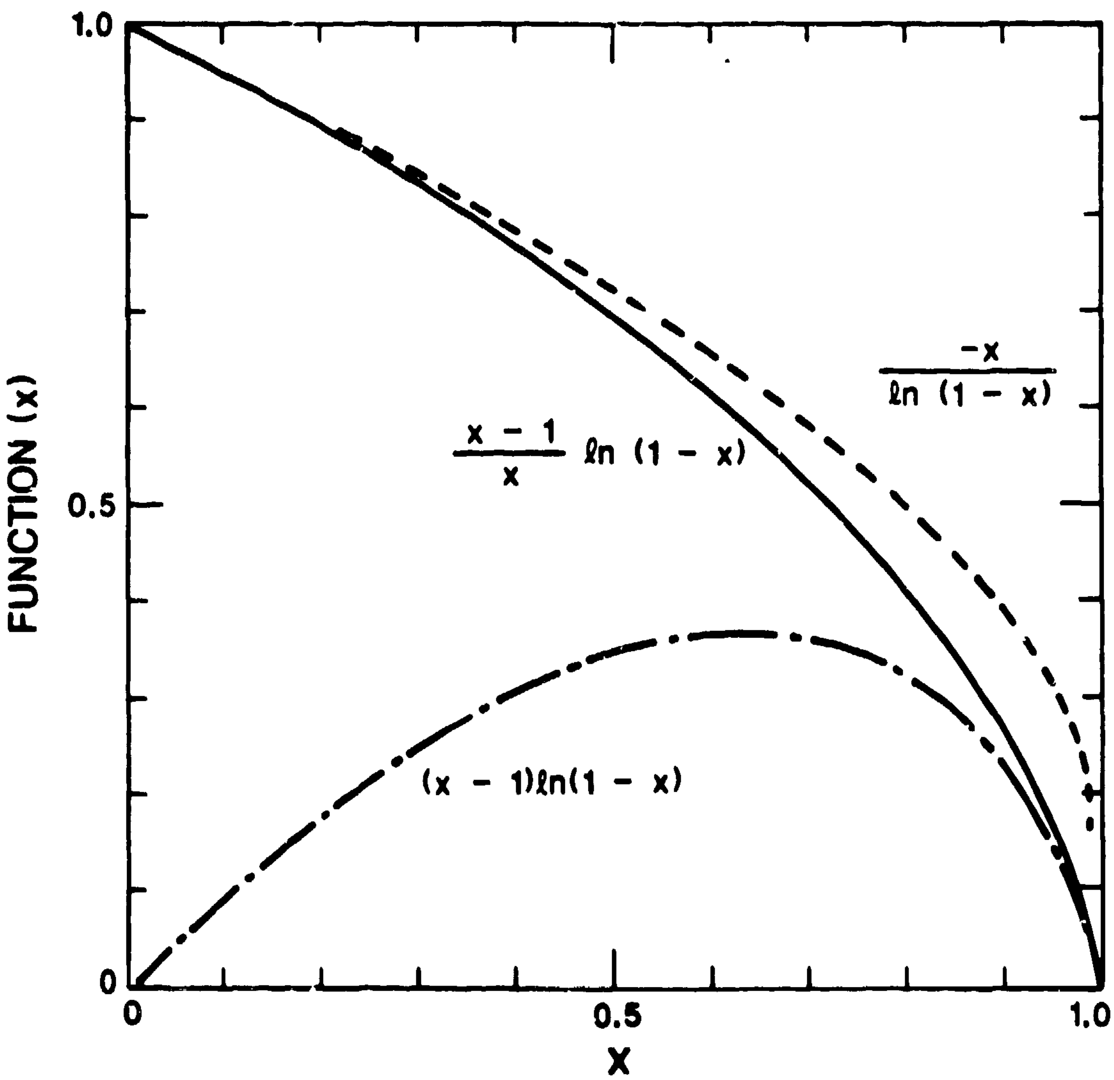


Fig. 2. Three functions of x.

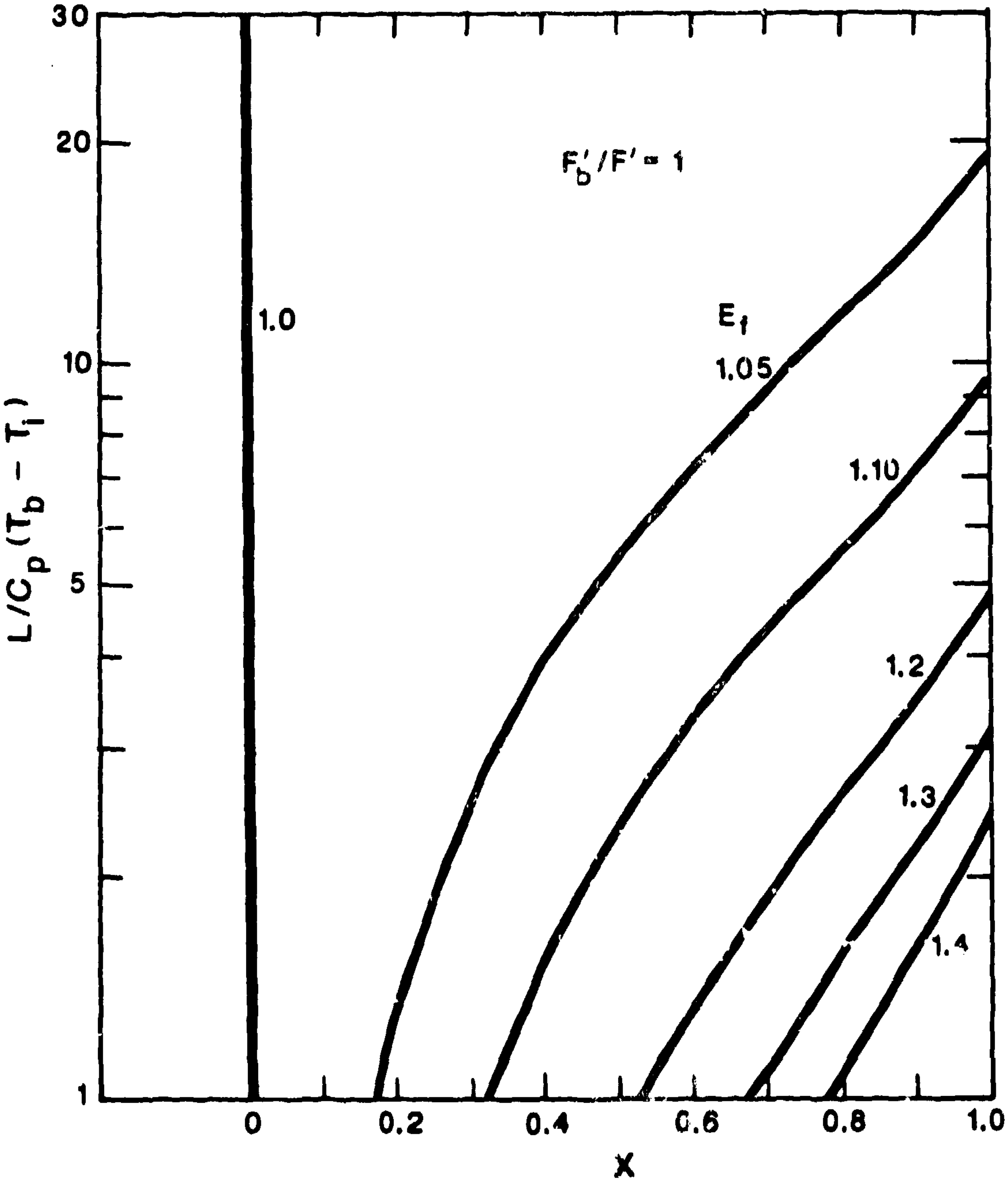


Fig. 3. Contours of E_f for $F_b'/F' = 1.0$.

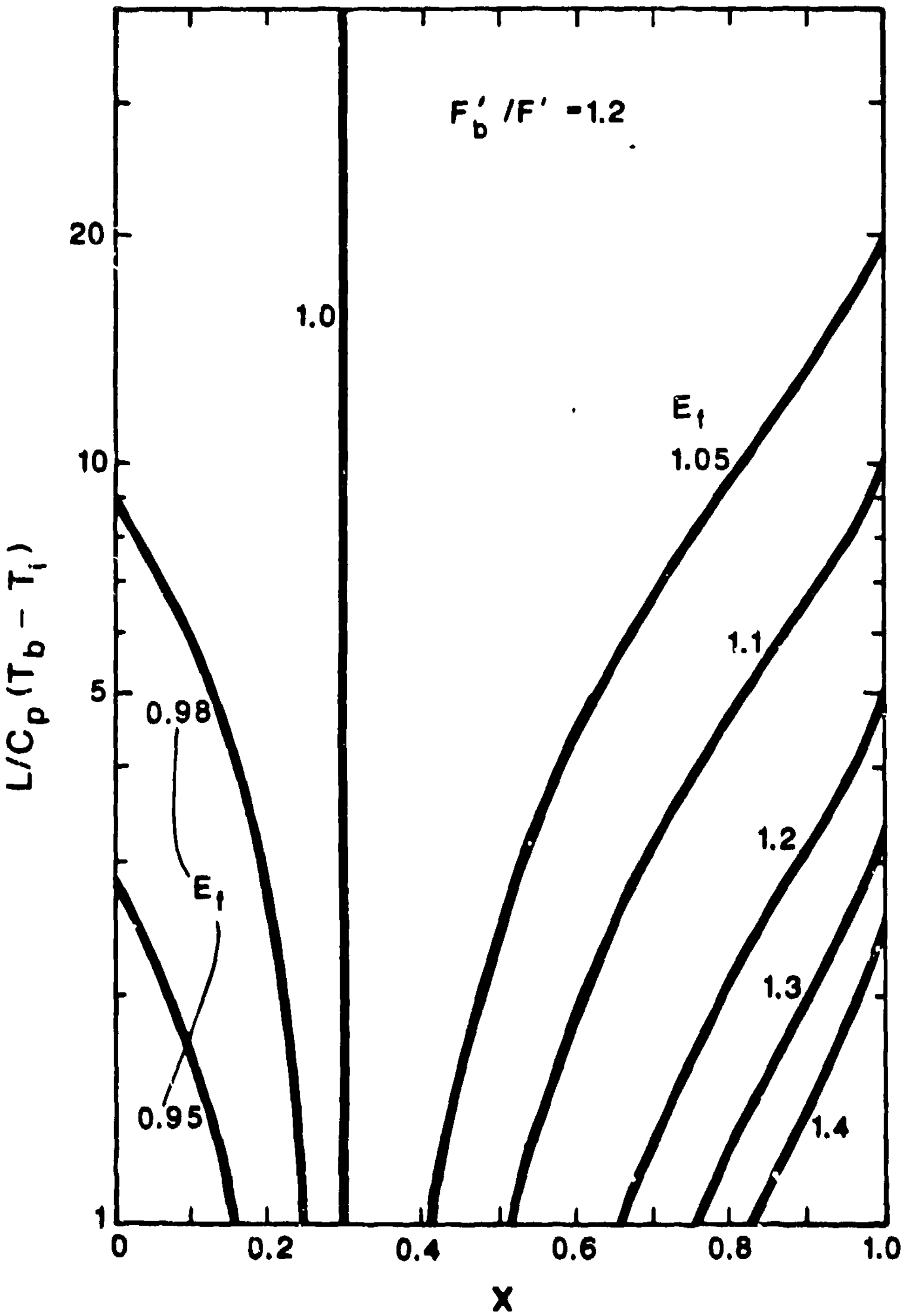


Fig. 4. Contours of E_f for $F_b'/F' = 1.2$, as occurs for many flat-plate collectors.

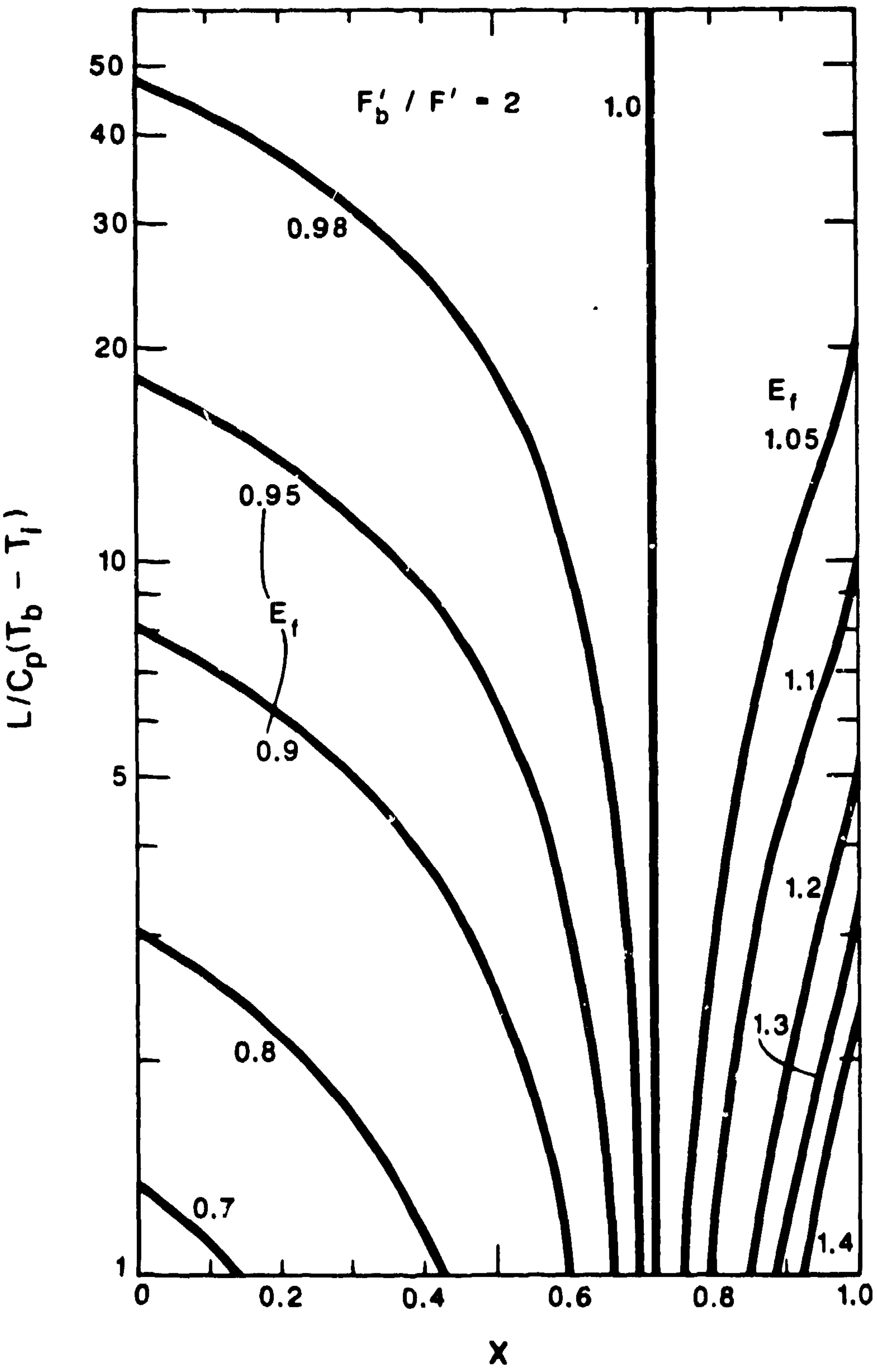


Fig. 5. Contours of E_f for $F_b'/F' = 2.0$.

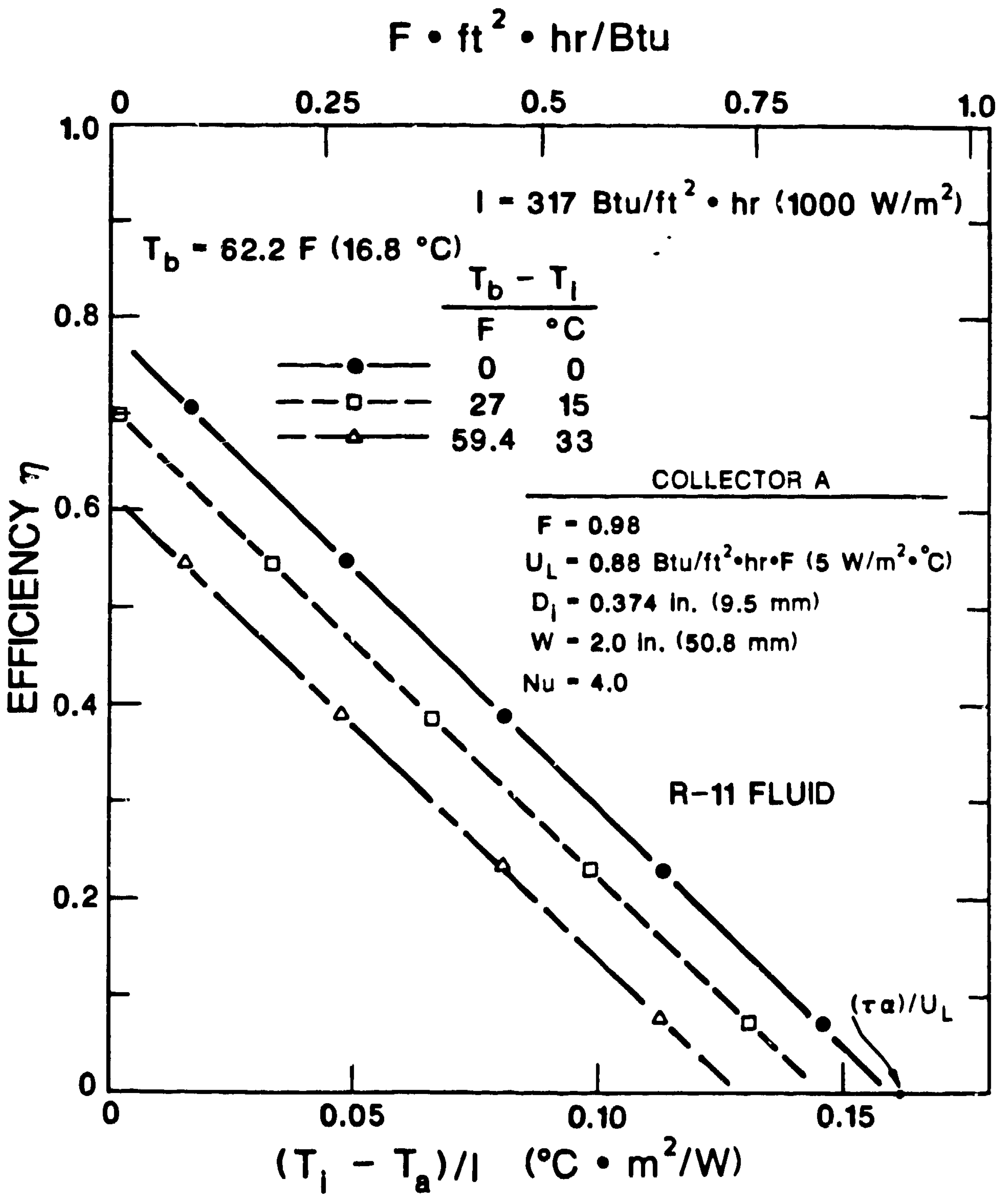


Fig. 6. Efficiency of Collector A versus $(T_i - T_a)/I$ with I = 317 Btu/ft$^2\cdot$ hr (1000 W/m^2) and T_b = 62.2 F (16.8°C).

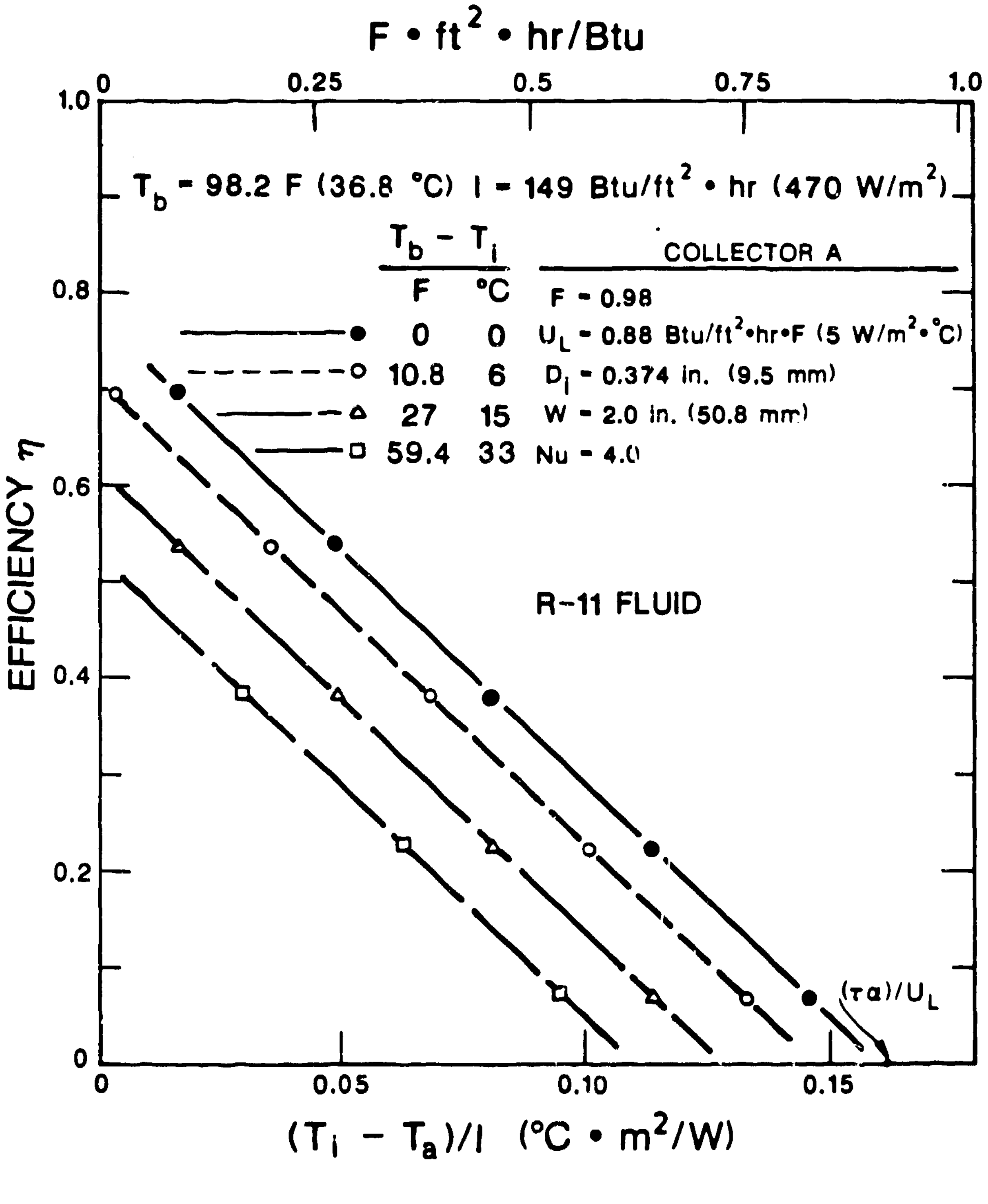


Fig. 7. Efficiency of Collector A versus $(T_i-T_a)/I$ with I = 149 Btu/ft^2· hr (470 W/m^2) and T_b = 98.2 F (36.8°C).

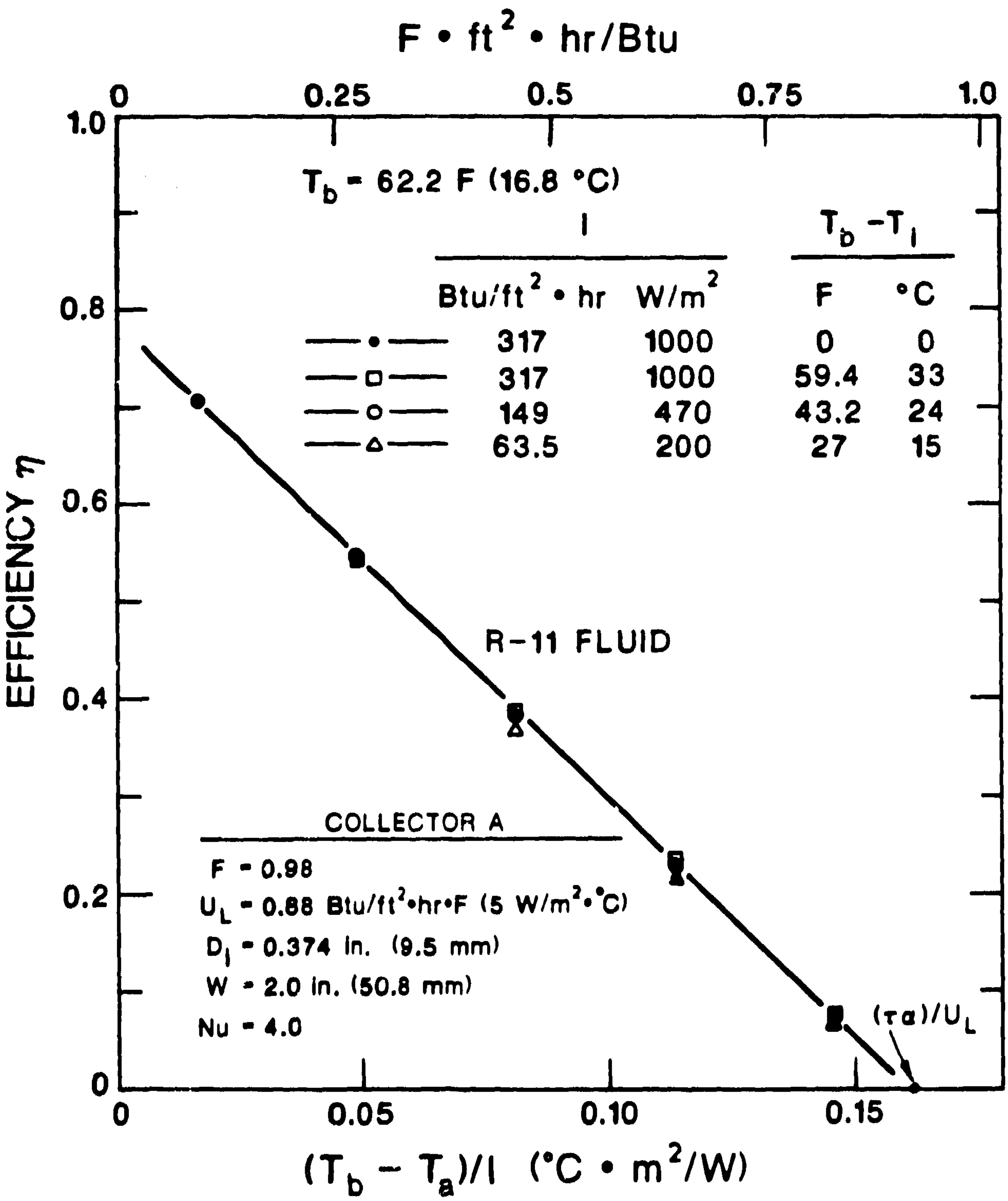


Fig. 8. Efficiency of Collector A versus $(T_b-T_a)/I$ with T_b = 62.2 F (16.8°C).

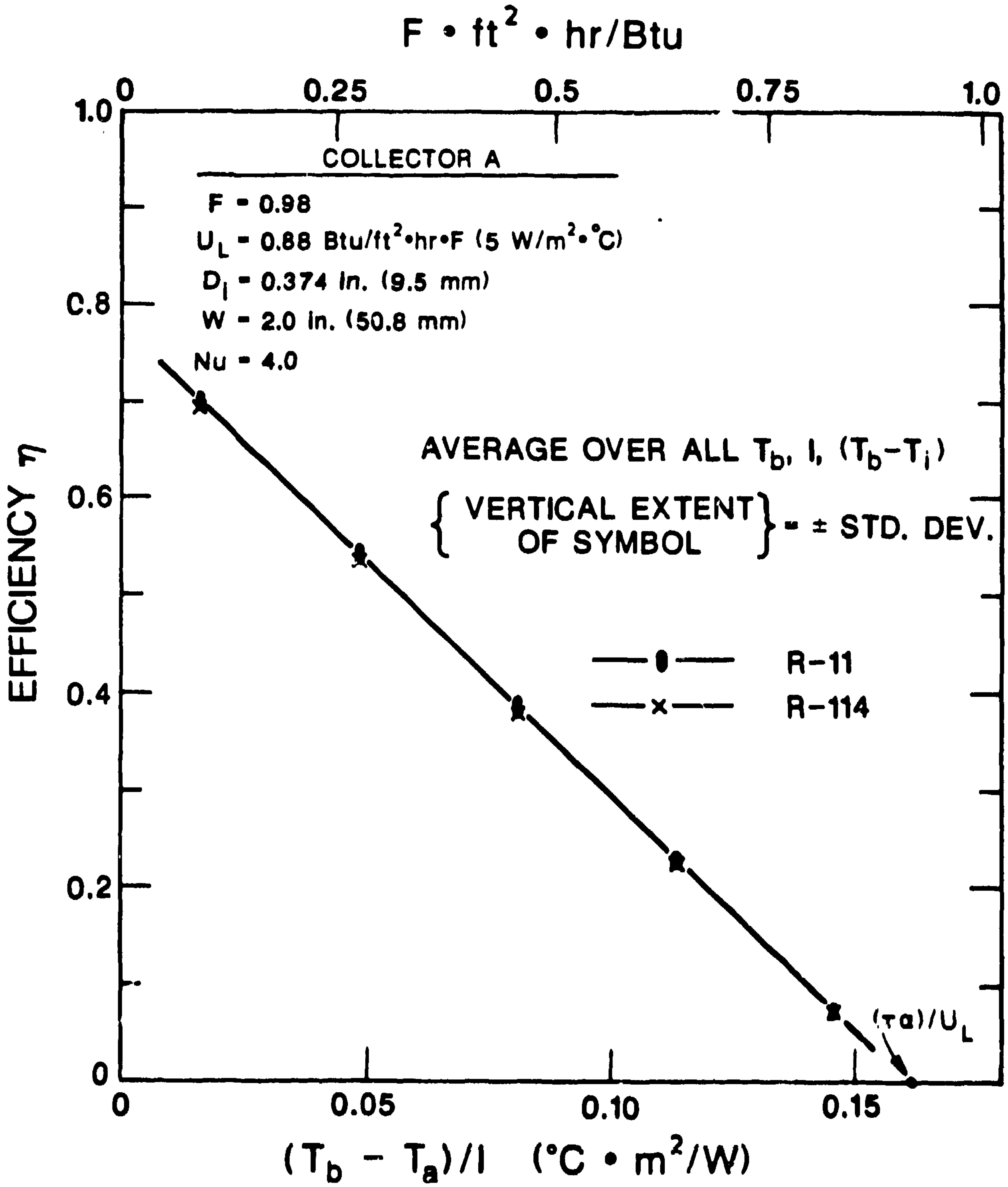


Fig. 9. Efficiency of Collector A averaged over saturation temperatures, insolation, and subcooling. Data are shown for R-11 and R-114 fluids.

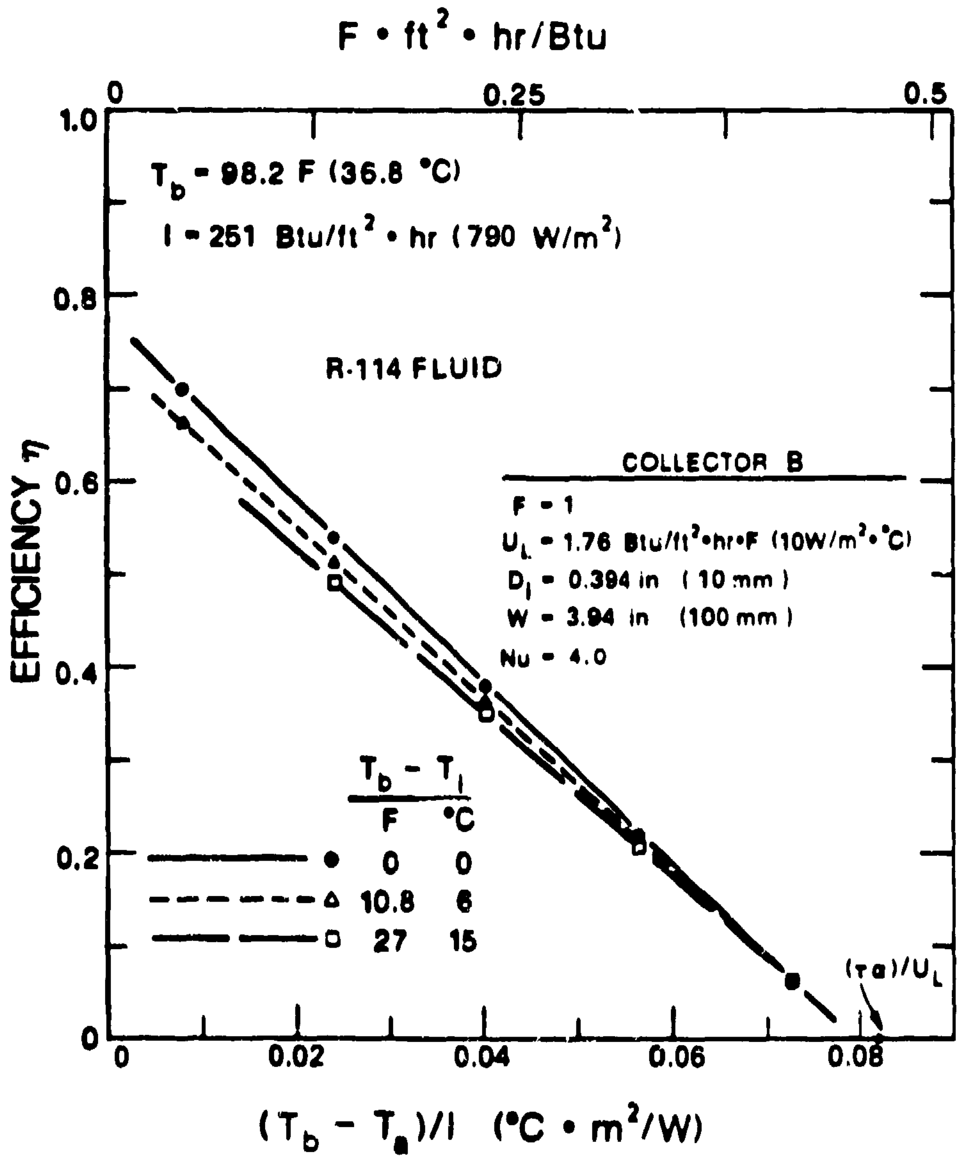


Fig. 10. Efficiency of Collector B versus $(T_b-T_a)/I$ with I = 251 Btu/ft$^2\cdot$hr (790 W/m^2) and T_b = 98.2 F (36.8°C).

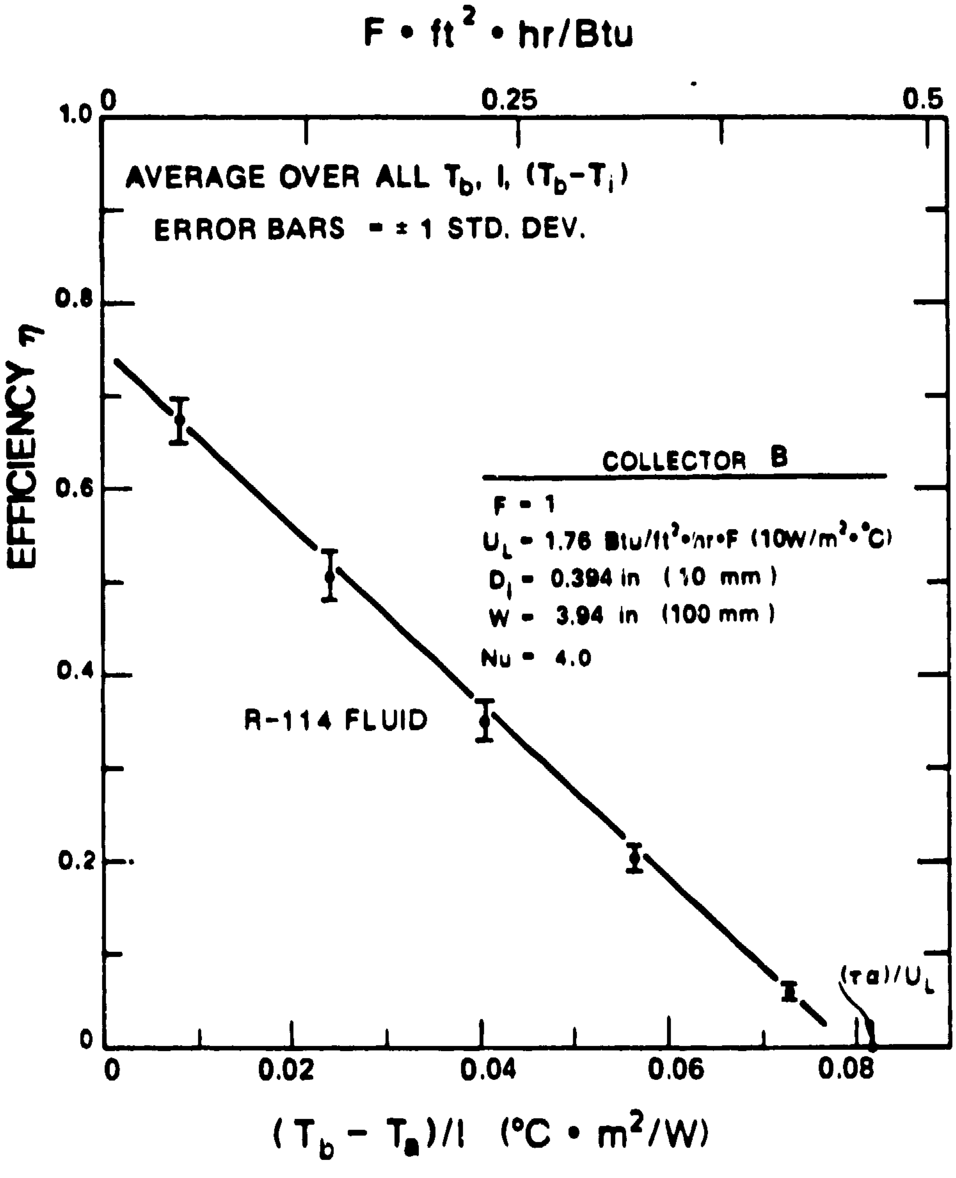


Fig. 11. Efficiency of Collector B averaged over saturation temperatures, insolation, and subcooling.

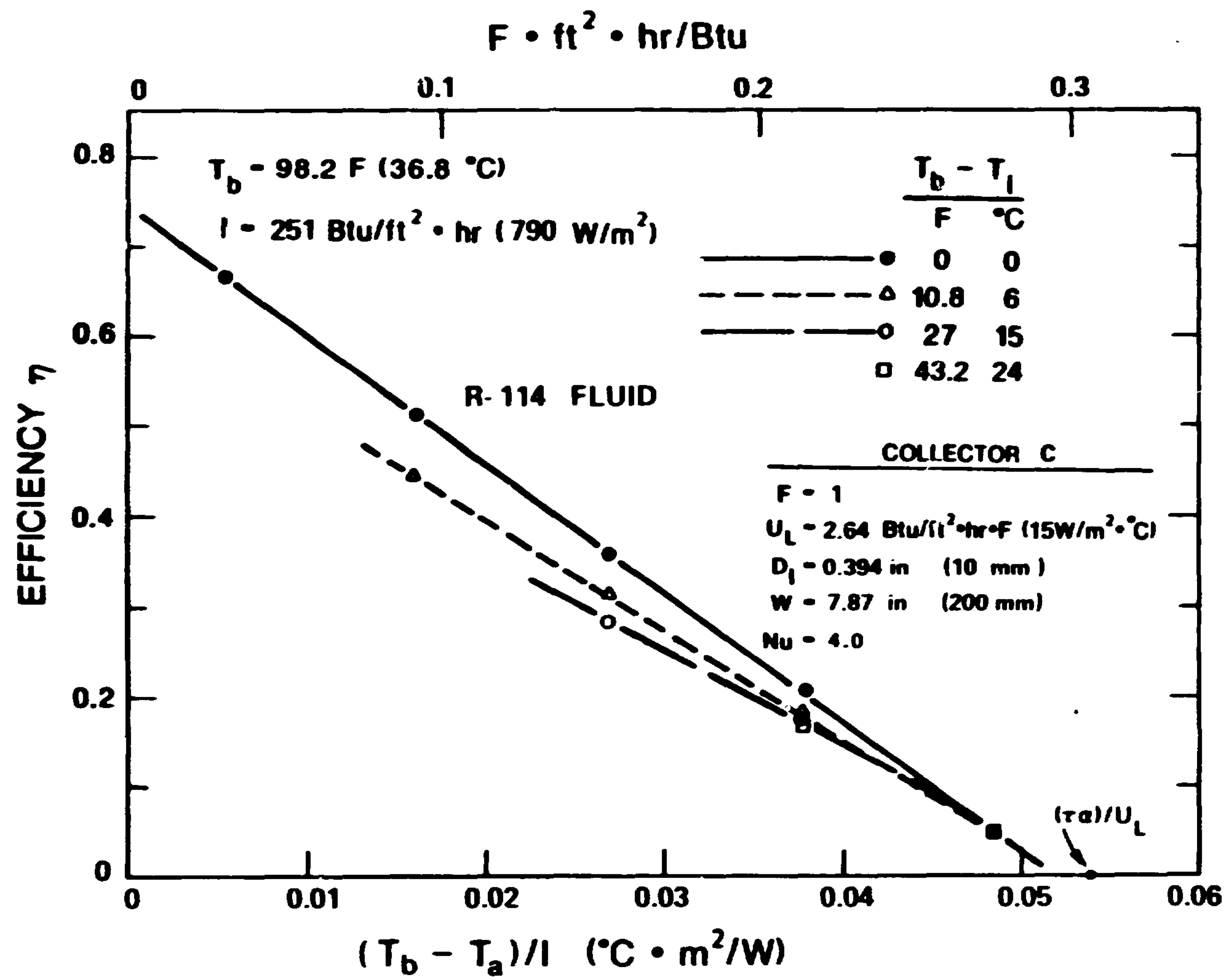
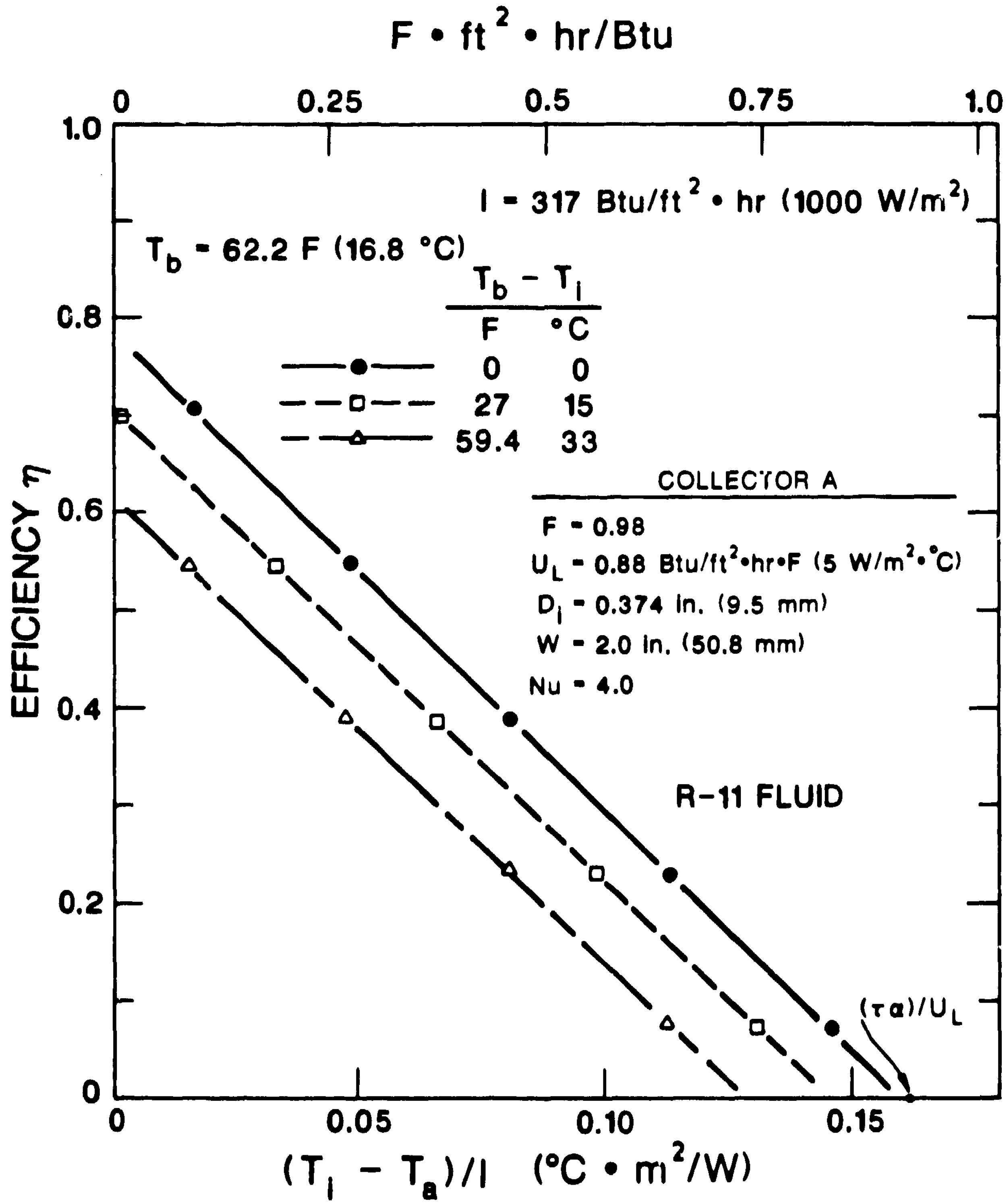


Fig. 12. Efficiency of Collector C versus $(T_b-T_a)/I$ with I = 251 Btu/ft^2· hr (790 W/m^2) and T_b = 98.2 F (36.8°C).



F • ft² • hr/Btu
0
0.25
0.5
0.75
1.0
1.0
0.8
0.6
0.4
0.2
0
I = 317 Btu/ft² • hr (1000 W/m²)
T_b = 62.2 F (16.8 °C)
T_b − T_i
F °C
0 0
27 15
59.4 33
COLLECTOR A
F = 0.98
U_L = 0.88 Btu/ft²•hr•F (5 W/m²•°C)
D_i = 0.374 in. (9.5 mm)
W = 2.0 in. (50.8 mm)
Nu = 4.0
R-11 FLUID
(τα)/U_L
EFFICIENCY η
0
0.05
0.10
0.15
(T_i − T_a)/I (°C • m²/W)