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INVESTIGATION OF METHODS TO TRANSFER HEAT FROM SOLAR LIQUID-HEATING COLLECTORS TO HEAT STORAGE TANKS

Final Report

By
John D. Horel
Francis de Winter

April 20, 1978

Work Performed Under Contract No. EY-76-C-03-1238

Altas Corporation
Santa Cruz, California

MASTER

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Solar Energy

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JOHN D. HOREL AND FRANCIS DE WINTER

APRIL 20, 1978

ALTAS CORPORATION
500 CHESTNUT STREET
SANTA CRUZ, CA 95060

PREPARED FOR THE
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SUMMARY

A study was made of the methods available to transfer heat from the collector to the water storage tank in water heating systems. In counterflow heat exchangers used in double loop water heating systems, it was found to be more important to use a high water flowrate than a high heat transfer fluid flowrate. It was earlier thought to be best to have matched WC_p (mass flowrate-specific heat) products in the loops. It was shown in this study that the water WC_p product should be about twice as large as that of the heat transfer fluid. It was found that neither the heat exchanger type nor the size was very critical, so that very simple criteria were adequate in determining optimum heat exchanger size. It was found that there is a definite system size below which one should use a traced tank or a coil in a tank. Equations and optimization criteria were developed for traced tanks or tanks with coils. At present, there is no quantitative understanding of liquid to liquid (direct contact) heat exchangers, though they are clearly quite effective. Draindown systems are discussed, and several appendices are included on heat transfer and other characteristics of fluid and of equipment.

TABLE OF CONTENTS

Nomenclature

A_c - Collector Area.

A_{cutoff} - Collector area at which traced tank and heat exchanger systems are equally cost effective.

A_{min} - Minimum cross-sectional area on shell side of heat exchanger through which fluid passes.

A_t - Traced tank area.

A_x - Heat exchanger area.

B - Tube spacing of helical coil in traced tank system.

C_c - Collector cost per unit area.

C_L - Tube length cost of helical coil per unit length.

C_p - Cost of extra pump for double loop heat exchanger system, specific heat of fluid.

C_t - Total cost of system.

C_x - Exchanger cost per unit area.

C' - Capacity rate ratio.

d - Inner tube diameter (inches).

D_i - Inner tube diameter (Ft or meters).

D_o - Outside tube diameter (Ft or meters).

D_a - Outer shell diameter of heat exchanger.

D_t - Diameter of storage tank.

F - Friction factor.

F_2 - Section efficiency of the collector.

F_R - Collector heat removal factor.

F_t - Helical coil efficiency factor.

F_t' - Traced tank penalty factor.

F_{ex}' - Heat exchanger penalty factor.

F - Deviation from matched capacity rates.

g - Acceleration due to gravity.

G - Mass flowrate per unit collector area.

G_s - Total mass flowrate through shell side of the heat exchanger.

G' - Total mass flowrate.

G'' - Total mass flowrate per unit tube cross-sectional area.

G_{max} - Total mass flowrate per unit minimum cross-sectional area on the shell side of the heat exchanger.

h_{is} - Inside tube scaling coefficient.

h_{so} - Outside tube scaling coefficient.

h_{ti} - Inside tube heat transfer coefficient.

h_{to} - Outside tube heat transfer coefficient.

H_T - Height of the storage tank.

k - Thermal conductivity, storage tank height to radius ratio.

L - Collector, exchanger or helical coil tube length.

LD_{50} - Quantity of chemical substance which kills 50% of dosed animals within 14 days.

N - Number of collector or heat exchanger tubes.

N_{BAF} - Number of exchanger baffles.

NTU - Number of heat transfer units.

P - Design pressure of the storage tank.

Δp - Pressure drop.

Q - Rate of heat transfer, total flowrate (gallons/minute).

Q_n - Flowrate within each collector or exchanger tube (gallons/minute tube).

r_t - Radius of the storage tank.

R_{wall} - Resistance to heat transfer by tube wall within heat exchanger.

R_e - Reynolds number.

T_a - Ambient temperature.

T_{ci} - Temperature of the fluid entering the collector.

T_{co} - Temperature of the fluid leaving the collector.

T_{si} - Temperature of the water entering the collector.

T_{so} - Temperature of the water leaving the collector.

T_w - Wall temperature of the fluid.

T_{tube} - Thickness of the tube of the helical coil.

T_{wall} - Thickness of the storage tank wall.

U_c - Collector heat loss coefficient.

U_t - Natural convection coefficient within the storage tank.

U_x - Overall heat transfer coefficient of the heat exchanger.

W - Mass flowrate (same as G'), tube spacing of the collector.

$(WC_p)_c$ - Capacity rate of the collector loop.

$(WC_p)_{min}$-Greater of the two capacity rates.

$(WC_p)_s$ - Capacity rate of the storage loop.

$\overline{WC}_p$ - Mean capacity rate of heat exchanger system.

$\alpha\tau$ - Absorptivity - transmissivity product of the collector.

ϵ - Exchanger or traced tank effectiveness.

ϵ' - Joint efficiency of the storage tank.

β - Volumetric coefficient of expansion.

μ - Viscosity.

ρ - Density.

σ - Maximum allowable hoop stress of the storage tank.

1.0 Introduction

In low temperature useage of solar energy one can use an air heating collector with a rock pile storage system, or a liquid heating collector with a water storage system. The liquid heating collector - water storage system combination has two main drawbacks:

a. Water is corrosive and requires expensive plumbing materials to achieve long lifetimes.

b. When water freezes it expands and can easily break the plumbing in the collector.

In virtually all parts of the U.S.A. it freezes regularly in Winter. The easiest way to confirm this is to consult horticultural guides for the areas in which frost susceptible tropical plants can be kept outdoors unprotected year-round. Outside of the southern part of Florida, the low lying areas in Hawaii, and _very_ narrow coastal regions in California and along the Gulf Coast, one invites disaster with outdoor frost susceptible tropical plants _or_ with water filled collection systems.

There are several ways to produce reliable operation despite freezing weather. One can use a separate heat transfer fluid loop, using a heat exchanger to transfer the heat. One can use a draining system, in which the water is drained out of the collection system whenever there is insufficient solar energy to heat the collector above freezing.

A draindown system must be designed, specified and constructed well enough so that it is totally foolproof. When it is supposed to drain, air or some other gas _must_ be able to get into the system, and the water in all of the parts in which there is danger of frost _must_ be able to flow out by gravity.

A system using a separate frost-proof heat transfer fluid is different. A cost effective fluid must be chosen. It must be matched to the plumbing materials, and such maintenance must be supplied as necessary, to prevent corrosion. If the fluid is toxic it must be kept from contaminating drinking water. Finally, the heat exchanger must be optimized. It must be large enough so as not to impose an excessive collection penalty through a large temperature rise in the collection, and yet must not be so large as to constitute an unreasonably large investment compared to other parts of the system.

Several heat exchanger arrangements can be used. One can use a double-loop system (one loop through the collector with antifreeze, one loop through the storage tank with water) with a heat exchanger between the two pumped loops. One can use a single pumped loop which features a coil in the storage tank or coils fastened to the outside of the tank, or one can use a liquid-to-liquid heat exchanger, using a fluid immiscible in water and of a different density, sprayed through the water tank in droplet form to exchange the heat. In all cases the loops could be of the thermosyphon type rather than pumped, but this requires the storage to be above the collectors.

The use of a heat exchanger leads to a collection penalty, as shown in Figure 1.1. The efficiency of collection decreases with increasing collection temperature, as shown in the curve in the lower part of the figure. The presence of the heat exchanger increases the collection temperature, and hence produces the collection penalty.

In a draindown system there is no such a heat exchanger penalty. There is the additional advantage that when the collector cools down at night the fluid does not cool down with it in the collector. In the morning the empty collector hence warms up faster and can begin to collect earlier. A draindown system must however be totally foolproof. One simply can not afford failure, except on a statistically low level (say one collection system every few hundred years of operation).

De Winter (1975)* first analyzed the case of a double loop heat exchanger system, and found that if capacity rates were used in the two loops so that:

$$(WC_p)_{coll} \leqslant (WC_p)_{sto} \tag{1.1}$$

*Included in full in Appendix A.

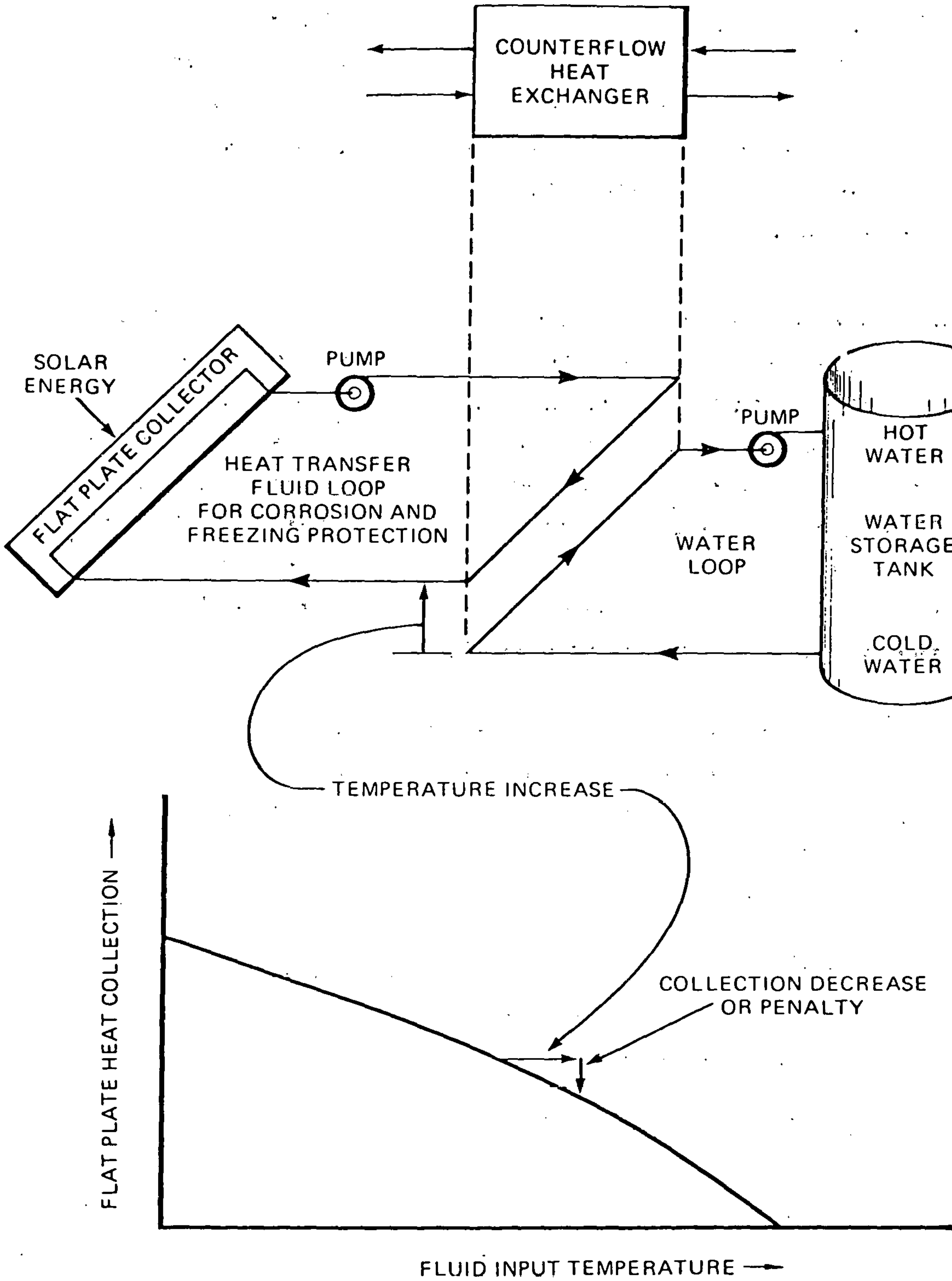


Figure 1.1 Heat Collection Decrease Due to a Double–Loop Heat Exchanger

(where W is the mass flowrate, C_p the specific heat, coll refers to the loop through the collector and sto to the loop through the storage tank, as shown in Figure 1.1) then the heat collected by the collector-heat exchanger combination was simply reduced by the factor:

$$\frac{F_R'}{F_R} = \frac{1}{1 + \frac{F_R U_c A_c}{(WC_p)_{coll}} \left[\frac{1}{\epsilon} - 1 \right]} \tag{1.2}$$

In this equation F_R is the standard collector efficiency factor of the Hottel Whillier flat plate collector model, F_R' is the same factor modified by the heat exchanger effect, A_c is the area of the collector, U_c the collector heat loss coefficient, and ϵ the heat exchanger effectiveness. Klein, Beckman and Duffie (1976) extended this to systems in which Equation (1.1) above does not hold, and determined that for this more general case:

$$\frac{F_R'}{F_R} = \frac{1}{1 + \frac{F_R U_c A_c}{(WC_p)_{coll}} \left[\frac{(WC_p)_{coll}}{\epsilon (WC_p)_{min}} - 1 \right]} \tag{1.3}$$

Equation (1.3), being completely general, is shown in Figure 1.2. In the general case, the heat exchanger effectiveness is an exponential function of the parameters $NTU=(U_x A_x)/(WC_p)_{min}$ and of $(WC_p)_{min}/(WC_p)_{max}$ as shown in Equation (1.4 a) and (1.4 b) (it should be noted that A_x is the heat exchanger heat transfer area and U_x the associated overall heat transfer coefficient):

$$\epsilon = \frac{1 - e^{-N}}{[1 - (WC_p)_{min}/(WC_p)_{max} e^{-N}]} \tag{1.4 a}$$

with

$$N = NTU\,[1 - (WC_p)_{min} / (WC_p)_{max}]. \tag{1.4 b}$$

The effectiveness increases with the heat exchanger A_x, and this reduces the collection penalty (it increases F_R'/F_R, bringing it closer to 1), so that it increases the heat collection. On the other hand, increasing the heat exchanger size increases the system cost. An optimum heat exchanger size can be found as illustrated in Figure 1.3.

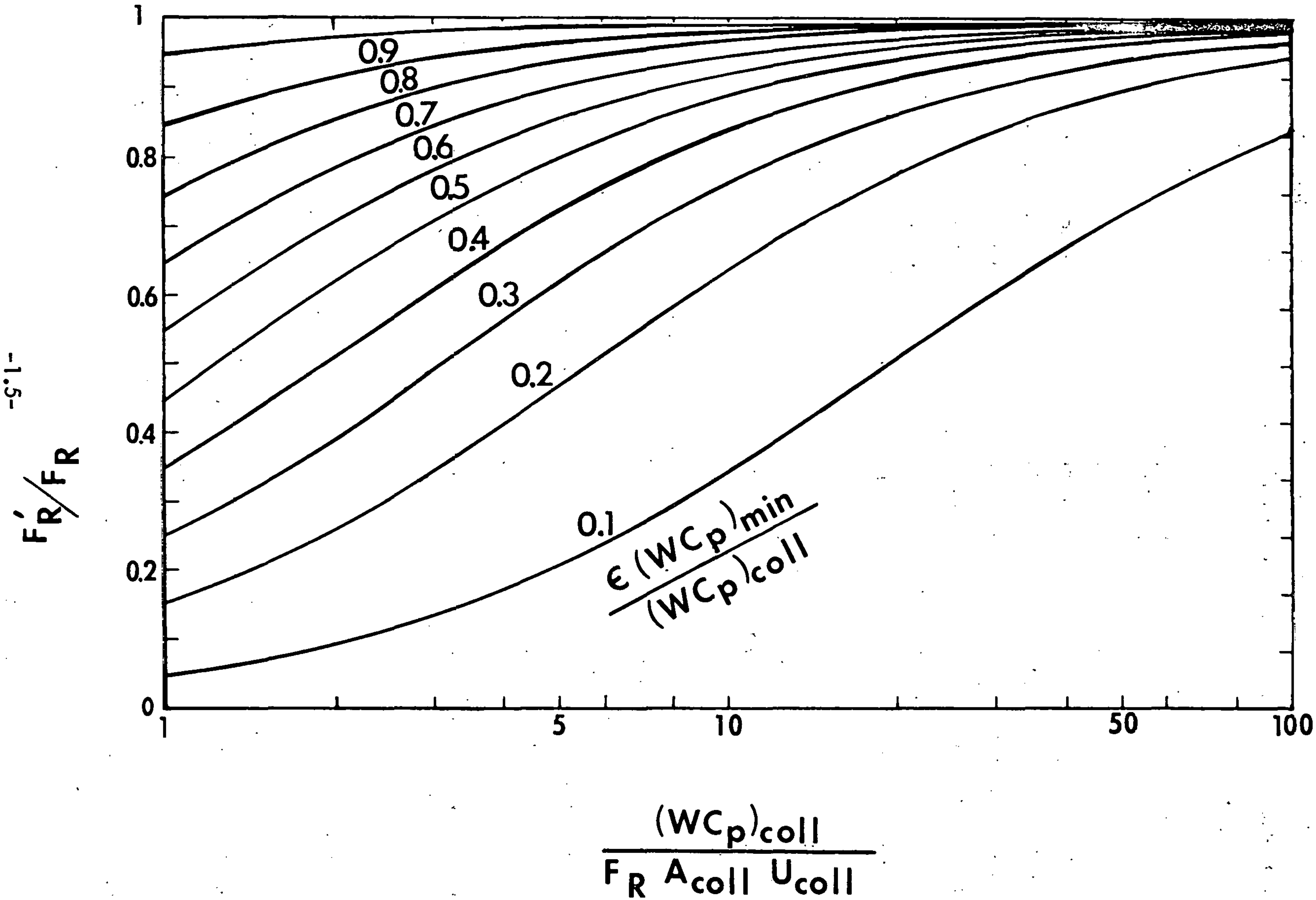


FIGURE 1.2 COLLECTOR HEAT EXCHANGER FACTOR

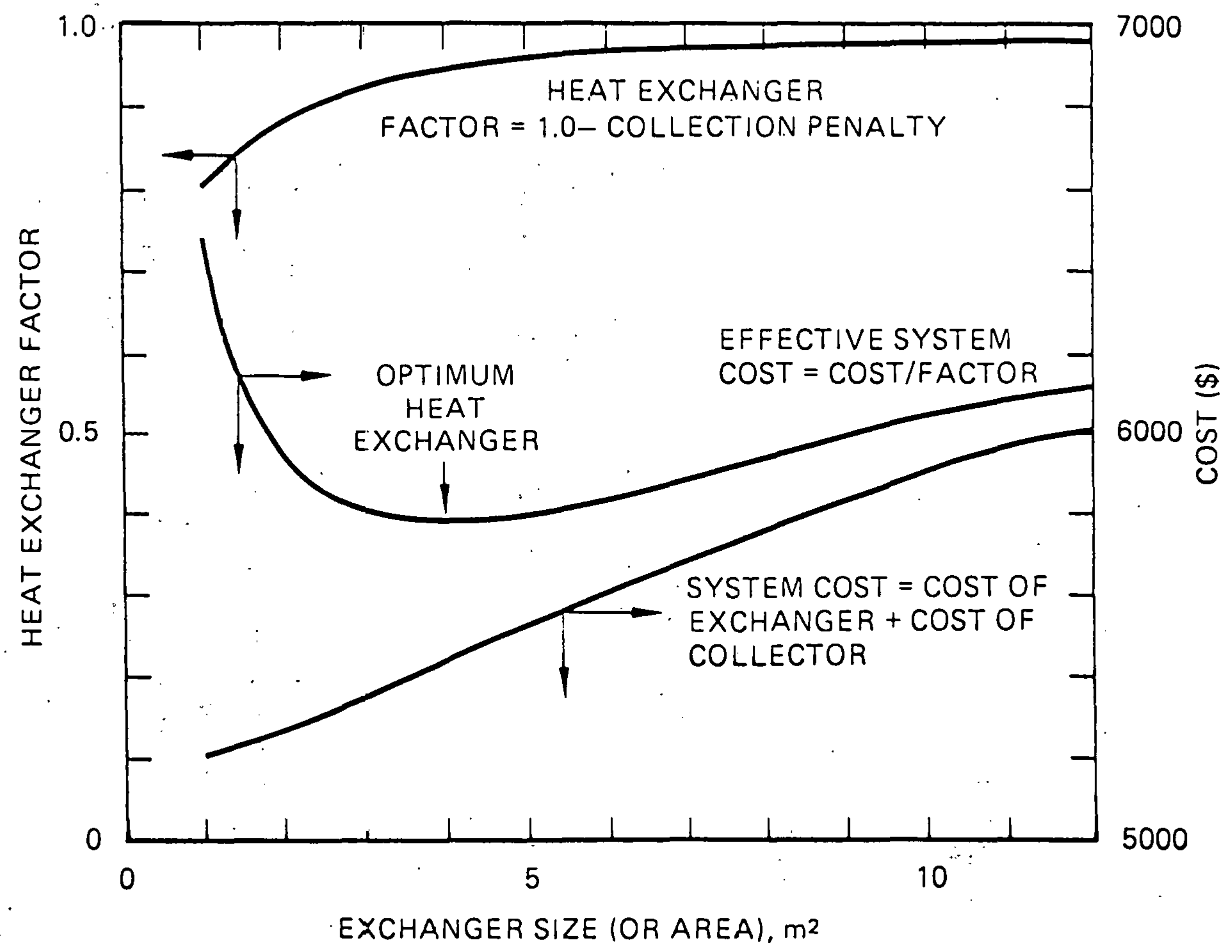


Figure 1.3 Typical Heat Exchanger Optimization plot, showing the Heat Exchanger Factor, Total System Cost, and Effective System Cost as a function of Heat Exchanger Size or Area.

For the specific case in which:

$$(WC_p)_{coll} = (WC_p)_{sto}, \tag{1.5}$$

de Winter (1975) found that:

$$\frac{F_R'}{F_R} = \frac{1}{1 + \frac{F_R U_c A_c}{U_x A_x}} \tag{1.6}$$

Since

$$\epsilon = \frac{1}{1 + \frac{(WC_p)_{coll}}{U_x A_x}} \tag{1.7}$$

When the cost per unit area for the collector (C_c) and the cost per unit area of heat exchanger (C_x) are constant, de Winter (1975) further found that if the heat transfer coefficient U_x did not vary with the area A_x the optimum heat exchanger area A_x could be calculated from the equation:

$$A_x = A_c \sqrt{\frac{F_R U_c C_c}{U_x C_x}} \tag{1.8}$$

In the present study it was found that with a given <u>average</u> WC_p product, the optimum heat exchanger invariably had a higher storage capacity rate $(WC_p)_{sto}$ than a collector capacity rate $(WC_p)_{coll}$, so that Equation (1.1) was invariably satisfied and Equation (1.2) applied. For typical values of the collector capacity rate $(WC_p)_{coll}$, it was found that the value of $C' = (WC_p)_{coll}/(WC_p)_{sto}$ ranged from 0.5 to 0.6. For all practical purposes it was found that it was still possible to use Equation (1.8) to find the optimum heat exchanger area, since this was only about 1% different from that found for the optimum (unmatched capacity rate) case. This area is discussed in Section 2.1.1.

An analysis for a single-loop system, using a traced tank or a coil in a tank, was performed in the present study (see Section 2.2). It was found that the same heat exchanger factor determined for a double loop system in Equation (1.2) could be used for the single loop system. Again one can determine an optimum heat exchanger area using the methodology shown in Figure 1.3. The main difficulty now lies in the fact that the heat transfer coefficients used to determine U_x are no longer straightforward forced convection coefficients, but that on the water (storage) side one has a natural convection coefficient which is harder to determine. This area is addressed in Section 2.2.

There is still another heat exchanger case one can use. One can use an unpressurized storage tank filled with water. This can feature inert gas (instead of air) above the water so as to limit oxidation and corrosion. The water can be deionized to further limit corrosion. The collector can be of the draindown type. A heat exchanger can be used between this tank and the domestic water system so as to heat the water on a once through basis. There are too many design permutations to permit generalizations on such a design without a thorough design study.

The conduction problem between the inside tank wall and the fluid in tracing tubes is analogous to that obtained in a flat plate collector with the tubes bonded below the plate. The heat transfer is given by: (inside water coefficient U_t)(inside tank area A_t)(F_t)(Fluid to water temperature diff.). According to Duffie and Beckman (1974) F_t is given by:

$$F_t = \frac{1}{\frac{BU_t}{\pi D_o h} + \frac{BU_t}{C_{bond}} + \frac{B}{D_o + (B - D_o) F}} \tag{1.9}$$

D_o - Outside diameter of the coil tube (m)

h - Heat transfer coefficient of fluid circulating through the coil $\left[\frac{\text{Watts}}{m^2\ {}^oC} \right]$

C_{bond} - Conductance of tank to coil bond $\approx \frac{4\ T_{wall}\ K_s}{D_o}$ $\left[\frac{\text{Watts}}{m\ {}^oC} \right]$

(This value of the bond conductance was determined by de Winter (1978).)

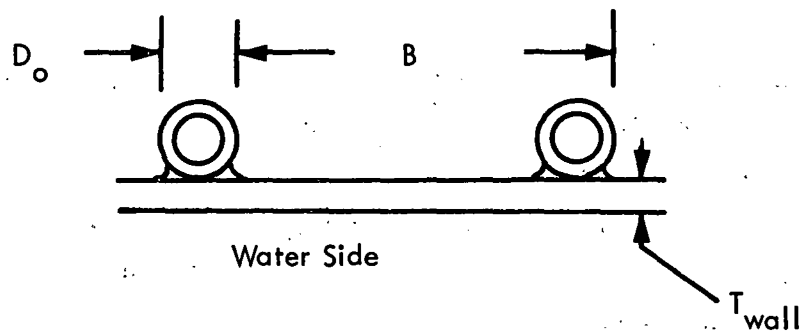
T_{wall} - Thickness of tank wall [m]

K_s - Conductivity of steel tank - $\left[\frac{\text{Watts}}{m\ {}^oC} \right]$

F - Fin efficiency of tank wall between the tubes, for heat losses to the water.

Figure 1.4 shows the relationship between the parameters of the traced tank.

Figure 1.4 Specification of Several Coil Parameters



Work on the liquid-to-liquid heat exchanger concept at the Colorado State University (Buchan et al, 1976; Ward et al, 1977) has revealed that these heat exchangers can be very efficient and can lead to very low collection penalties. There is however as yet no quantitative understanding of their performance.

This is the final report on a DOE-sponsored project to perform an analytical study on the different alternatives existing in this area. Section 2 deals with the behavior of systems using a heat exchanger with segregated fluids. Section 3 deals with direct contact liquid-to-liquid heat exchangers, Section 4 with draindown systems, and Section 5 with recommendations for further work. Section 6 and 7 and the appendices provide background information, and information on heat transfer and fluid characteristics and computer programs used in the study.

2.0 HEAT EXCHANGER SYSTEMS WITH SEGREGATED FLUIDS

In double loop solar water heating systems the physics was well understood at the beginning of the study. In single loop systems there were essentially no previous guidelines, although single loop systems had been built in the 1930's and probably even earlier.

For double loop systems the requirements of optimum designs were studied in some detail. For single loop systems the theory of operation was developed, and optimum design guidelines were developed. Guidelines were developed for determining the point at which one should change from single to double loop systems.

It should be noted that many practical problems are glossed over in this section. For example: in any pumped antifreeze loop one way valves are essential to avoid reverse thermosyphon at night, which can (and often does) freeze heat exchangers. In any liquid heating system, lines would have to be cost-optimized in order to make sure that one is not paying too much in pumping power or too much in plumbing.

2.1 Heat Transfer Analysis and Optimization of Double-Loop Solar Water Heating Systems

In double loop water heating systems, much of the basic analysis had already been performed by de Winter (1975) and by Klein, Beckman and Duffie (1976). It remained to do systematic sensitivity studies on the model to find out what parameters, if any, were important, to determine what was the simplest way to determine a reliable optimum, and to determine what was the minimum size at which a full fledged double loop (also double pumped) system with a heat exchanger should be used. Below this minimum size it would become better to switch to a single loop system, using a traced tank or a coil to transfer the heat to the water. These questions are treated in the following subsections.

2.1.1 Criticality of the Matched Capacity Rate Concept

The capacity rates (mass flowrate-specific heat products) on either exchanger side are important in determining the optimum performance characteristics of the exchanger-collector systems. For a given average capacity rate, if the capacity rate on the storage side is much larger than that on the collector side (or viceversa) the net effect will be an increase in collector temperature and a corresponding decrease in collection. This becomes clearer examining Figure 2.1.1-1. De Winter (1975) assumed the optimum counterflow exchanger would be one operating at a capacity rate ratio ($C' = (WC_p)_{min}/(WC_p)_{max}$ equal to one, i.e. matched capacity rates. In this study, the effect of the capacity rate ratio (C') on the optimum performance was investigated further.

In order to determine the performance of the collector-exchanger system with capacity rate ratios other than one, it was assumed that there existed a fixed average capacity rate ($\overline{WC}_p$) at which the exchanger operated. The storage and collector loop capacity rates were allowed to vary from this mean state, but restricted so that when one capacity rate is increased, the other must decrease. Thus the collector loop capacity rate is:

$$(WC_p)_c = \overline{WC}_p(1.0 + F) \quad (\text{Watts}/\ ^\circ C) \qquad (2.1.1\text{-}1)$$

F - deviation from the mean capacity rate $-1 < F < 1$

and the storage side capacity rate becomes:

$$(WC_p)_s = \overline{WC}_p\ (1.0 - F) \quad (\text{Watts}/\ ^\circ C) \qquad (2.1.1\text{-}2)$$

There is nothing magic about this assumption of an average capacity rate. If one wants to limit the total pumping power one would be concerned about some weighted average capacity rate (or flow rate). In any case one is interested in determining whether it is useful to increase the flowrate of one of the fluids, at the expense of the flowrate of the other one. If pumping power is no concern collection can always be increased by increasing either or both of the flowrates.

For the matched case, F equals 0, and the capacity rates both equal the mean capacity rate. Note when:

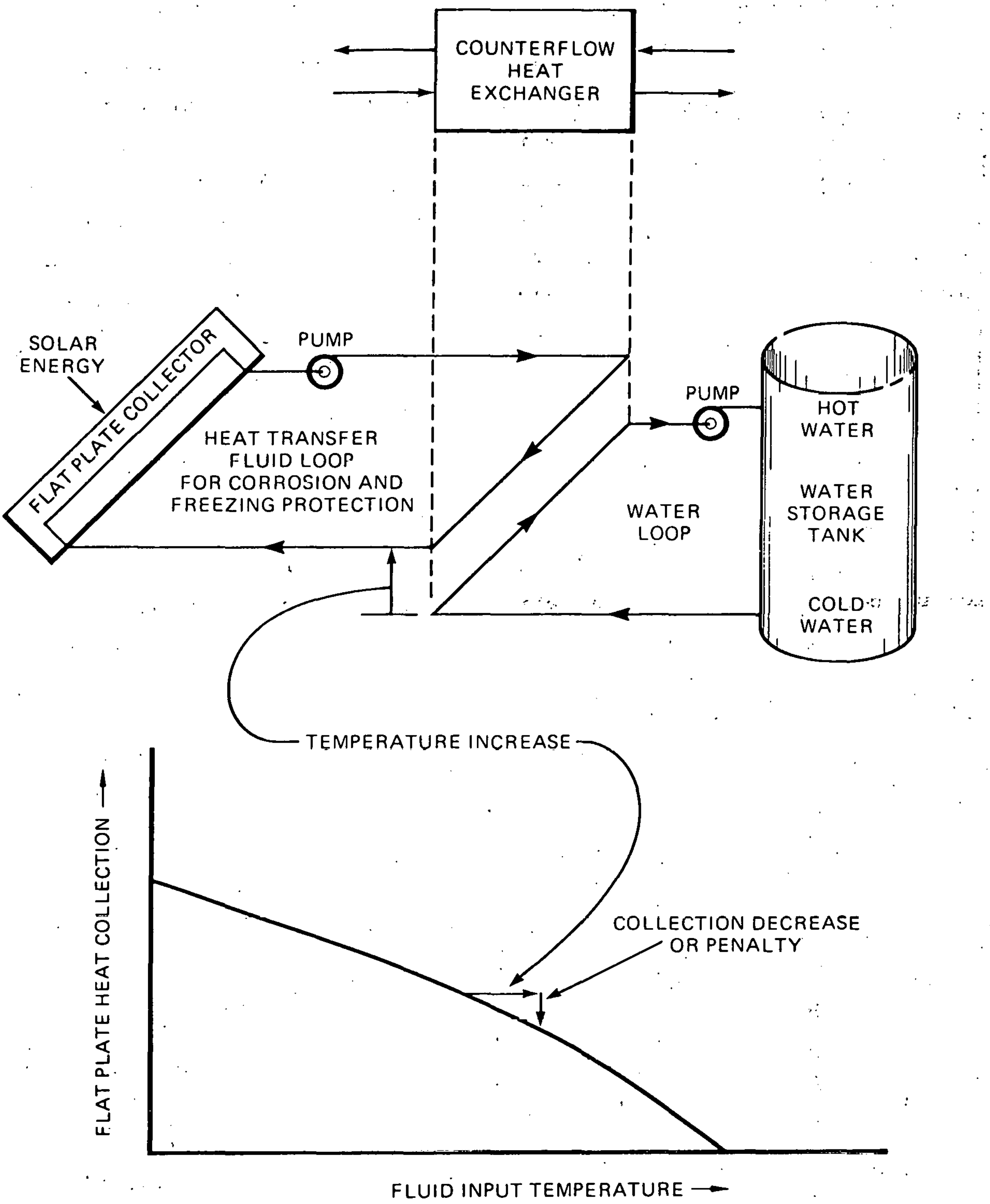


Figure 2.1.1.1 Heat Collection Decrease due to a Double-Loop Heat Exchanger

$(WC_p)_c > (WC_p)_s$, then $C' = (1 - F)/(1 + F)$ while for

$(WC_p)_c < (WC_p)_s$, $\quad C' = (1 + F)/(1 - F)$

A brief discussion of the factors influenced by the capacity rates precedes the determination of the optimum capacity rate ratios.

The major effect of the capacity rates on the rate of heat transfer is through the heat exchanger penalty factor-collector efficiency product F_R'. From Section 2.1.4 this becomes:

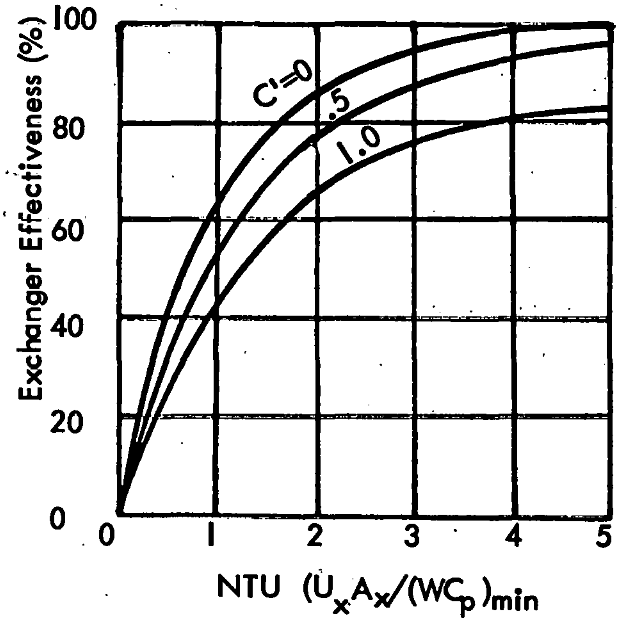
$$F_R' = \frac{F_R}{1 + \frac{F_R U_c A_c}{(WC_p)_c} \left[\frac{(WC_p)_c}{(WC_p)_{min} \epsilon} - 1 \right]} \qquad (2.1.1\text{-}3)$$

This term depends on the capacity rates through the variables $(WC_p)_c$, $(WC_p)_{min}$, F_R and ϵ. From Equation (2.1.1-3) it can be seen that increasing $(WC_p)_c$ and $(WC_p)_{min}$ will reduce the penalty to heat transfer producted by the heat exchanger, i.e. it will produce higher heat collection. Similarly, F_R increases with increasing capacity rates which also increases the heat collection.

The effect of the capacity rate ratio on the exchanger effectiveness follows. In Section 2.1.3 the exchanger effectiveness is presented for a counterflow exchanger. Figure 2.1.1-2 using Equation (2.1.3-1) is presented here to show the effect of the capacity rate ratio (C') on the exchanger effectiveness for various NTU's $[(U_x A_x)/(WC_p)_{min}]$. For a given NTU, the efficiency of the counterflow exchanger increases with increasing $(WC_p)_{max}$. It is only when some restraint is imposed on total flowrate, such as might for example exist when it is necessary to minimize the total fluid pumping power on side one plus side two, that matched (or closely matched) capacity rates are of any potential usefulness.

Another effect of the capacity rate is its influence upon the collection temperatures. Assuming minimal heat losses through the pipes connecting the heat exchanger to the collector and storage tank, the effect of the capacity rates on the temperature characteristics of the system can be determined.

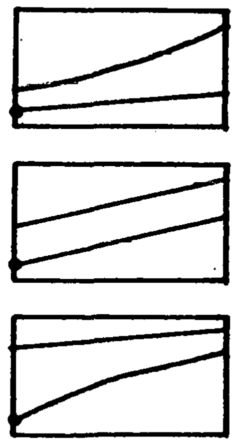
Figure 2.1.1.2 Effectiveness of a Counter-flow Heat Exchanger versus NTU and the Capacity rate Ratio C'... Also Temperature Profiles versus C'.



a) $(WC_p)_c < (WC_p)_s$

b) $(WC_p)_c = (WC_p)_s$

c) $(WC_p)_c > (WC_p)_s$



To consider the effect of the capacity rate ratio on collection, consider Equations (2.1.1-4a), (2.1.1-4b), (2.1.1-5a) and (2.1.1-5b), which describe the exchanger effectiveness. The effectiveness is simply the ratio of the heat transfered to the maximum heat transfer the second law of thermodynamics will allow. Equations (2.1.1-4a) and (2.1.1-4b) show this definition of the effectiveness. In the first equation, the storage side has the minimum capacity rate, in the second equation the collector side

$$\epsilon = \frac{(WC_p)_s (T_{si} - T_{so})}{WC_{p\ min} (T_{co} - T_{so})} \qquad (2.1.1\text{-}4a)$$

if $(WC_p)_s < (WC_p)_c$

$$\epsilon = \frac{(WC_p)_c \ (T_{co} - T_{ci})}{(WC_p)_{min} (T_{co} - T_{so})} \qquad (2.1.1\text{-}4b)$$

if $(WC_p)_c < (WC_p)_s$

Where:

T_{so} - temperature of the water leaving storage - °C

T_{si} - temperature of the water entering storage - °C

T_{co} - temperature of the leat transfer fluid leaving collector-°C

T_{ci} - temperature of the heat transfer fluid entering collector - °C

In the above equations the WC_p products can of course be cancelled out. They are left in because in this way the heat quantities are more easily identifiable.

The effectiveness as a function of heat exchanger parameters is given by:

$$\epsilon = \frac{1}{1 + \frac{WC_p}{U_x A_x}} \tag{2.1.1-5 a}$$

if $(WC_p)_s = (WC_p)_c$

and

$$\epsilon = \frac{[1 - e^{-NTU(1 - C')}]}{[1 - C' e^{-NTU(1 - C')}]} \tag{2.1.1-5b}$$

if $(WC_p)_s \lessgtr (WC_p)_c$

Equations (2.1.1-5 a) and (2.1.1-5b) are represented by Figure 2.1.1.2, with the temperature profiles in the exchanger represented in the lower part of the figure for three different capacity rate ratio cases. The effect of the capacity rate ratio can be understood most readily by considering these three cases, a, b, and c and by assuming that as a first approximation the heat transfered in the collector does not vary much with changes in flow rate ratio, and that the U_x value* also stays approximately constant. Consider that in all cases T_{so} is the same, and that $(WC_p)_s$ $(T_{si} - T_{so})$ is the same. The upper line represents the temperature of collection as well as the temperature of the collector fluid in the heat exchanger. It is quite easy to see, in considering cases a, b, and c, that if one goes from b to c the average temperature of collection is raised somewhat, so that less energy will be collected. If on the other hand one moves from b to a, with a moderately mismatched flowrate, it is reasonable to expect that the collection temperature will be pulled down somewhat, so that a greater heat collection will result. Simple calculations can serve to confirm this explanation.

In the following subsection, the optimum capacity rate ratio is determined for typical collector and exchanger characteristics.

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* Note that the effectiveness increases somewhat at unmatched capacity rates.

Determination of the Optimum Capacity Rate Ratios

In this section, the capacity rate ratio (C') will be defined as:

$$C' = \frac{(WC_p)_c}{(WC_p)_s}$$ which from Equation (2.1.1-1) (2.1.1-2) reduces to:

$$C' = \frac{1 + F}{1 - F}$$

The optimum capacity rate ratio is one for which the following is true.

$$\frac{d}{dF}\left[\frac{C_t}{Q}\right] = 0 \qquad (2.1.1\text{-}6)$$

Where:

$$C_T = C_c A_c + C_x A_x$$

$$Q = F_{ex}' \, F_R Q_o$$

$$Q_o \neq f(F)$$

Equation (2.1.1-6) reduces to:

$$\frac{d}{dF}\left\{[C_c A_c + C_x A_x]\left[\frac{1}{F_R} + \frac{U_c A_c}{(WC_p)_c}\left[\frac{(WC_p)_c}{(WC_p)_{min}\epsilon} - 1\right]\right]\right\} = 0 \qquad (2.1.1\text{-}7)$$

Solving this for F reduces to:

$$C_x \frac{dA_x}{dF}\left[\frac{1}{F_R} + \frac{U_c}{GC_p(1 + F)}\left[\frac{1}{\epsilon} - 1\right]\right] + [C_c A_c + C_x A_x]\left[-\frac{1}{F_R^{\,2}}\frac{dF_R}{dF} - \frac{U_c X'}{GC_p(1 + F)^2}\right] = 0$$

(2.1.1-8)

Where:

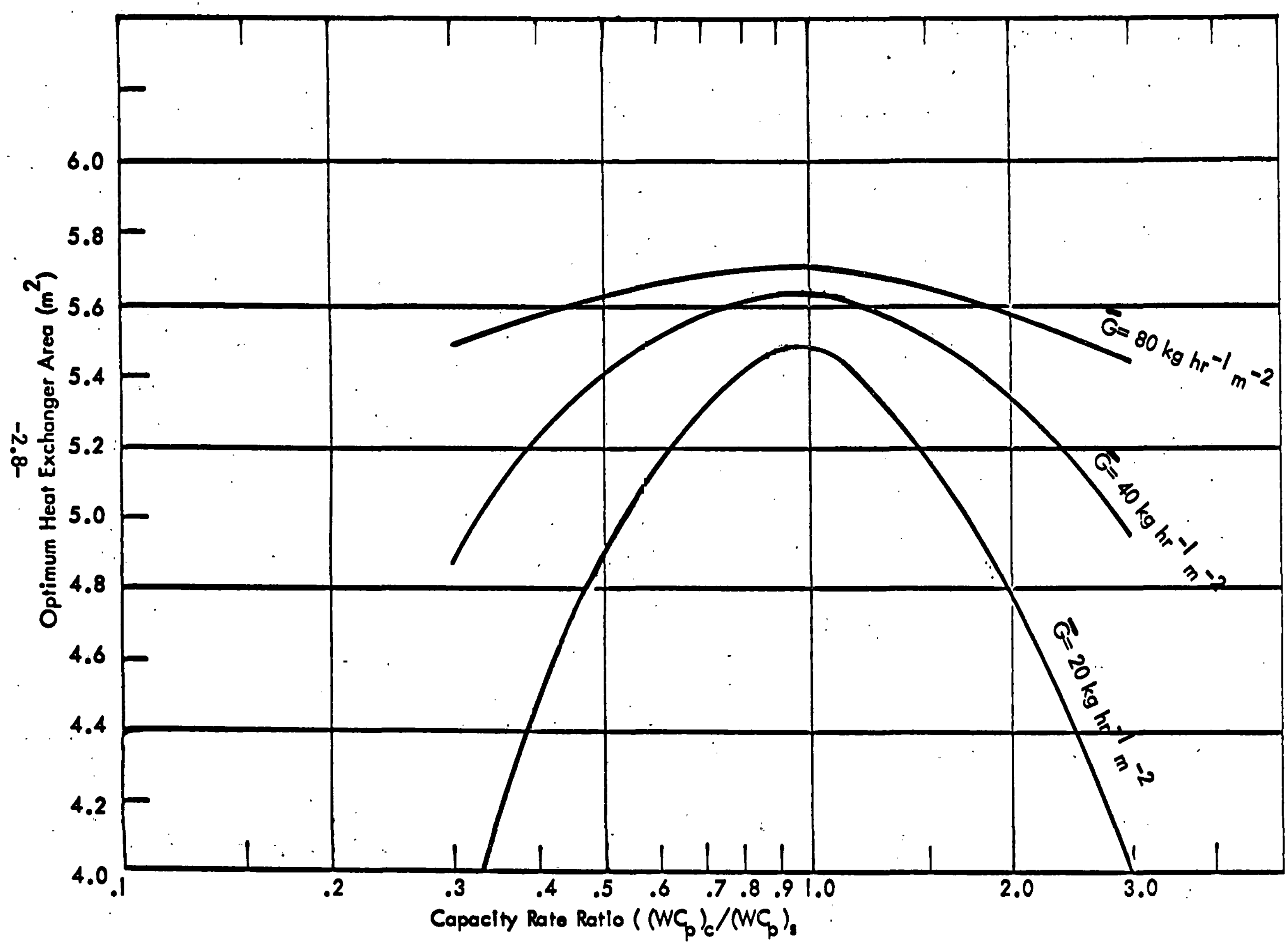
$$\frac{dF_R}{dF} = \frac{F_R}{1 + F} - \exp\left[-\frac{F_2 U_c}{GC_p(1 + F)}\right]\left[\frac{F_2}{1 + F}\right]$$

F_2 = section efficiency of the collector

$$X' = \frac{1}{\epsilon} - 1 + \frac{(1 + F)\frac{d\epsilon}{dF}}{\epsilon^2}$$

Figure 2.1.1.3

Optimum Heat Exchanger Area Versus the Capacity Rate Ratio and Mass Flowrate



$$\frac{\frac{d\epsilon}{dF}}{\epsilon^2} = \frac{\exp y'}{(1 - \exp y')^2} \left[- \frac{dy'}{dF} + (1 - \exp y')\left(\frac{2}{1 - F^2}\right) + \frac{1 + F}{1 - F}\frac{dy'}{dF} \right]$$

$$y' = \frac{A_x U_x}{A_c \overline{GC}_p (1 + F)} \left[1 - \frac{1 + F}{1 - F} \right]$$

$$\frac{dy'}{dF} = \frac{ZU_x}{A_c \overline{GC}_p} \left[\frac{dA_x}{dF} \left(\frac{F}{1-F^2}\right) + A_x \left[\frac{1 + F^2}{(1 - F^2)^2} \right] \right]$$

This series of equations could not be reduced much further and was solved by iteration. A_x was determined from equation (2.1.4-3) by iteration, but for simplicity dA_x/dF was found from the matched optimum heat exchanger area:

$$A_x = A_c \sqrt{\frac{F_R U_c C_c}{U_x C_x}}$$

It was found that the optimum capacity rate ratio C' was a function of C_c, C_x, $\overline{GC}_p$, F_2, U_c, and U_x, where $\overline{GC}_p$ is the mean capacity rate per unit collector area. In order to keep the overall heat transfer coefficient constant for varying exchanger size, the number of tubes within the exchanger was found from Equation (2.1.4-8). Thus the heat exchanger area was varied by varying the tube length, not the number of tubes. Although the overall heat transfer coefficient of the exchanger will vary due to the varying capacity rates, it is assumed to be a small variation since the increase in heat transfer on one side will be balanced by a decrease on the other side.

As shown in Section 2.1.4 the optimum heat exchanger area is a function of the capacity rate ratio. Figure 2.1.4.1 is reproduced here as Figure 2.1.1.3 to show the effect of mismatched capacity rates on the optimum heat exchanger area. As the capacity rate ratio $(WC_p)_c/(WC_p)_s$ deviates from matched, the optimum heat exchanger area goes down. Note that for normal mean capacity rates (operating mass flowrates of 80 kg m^{-2} h^{-1}, the effect of the capacity rate ratio on the optimum heat exchanger area is quite small. For C'=1, A_x=5.69 m^2, while for C'=0.5, A_x=5.62 m^2. This is a 1.25% change in the optimum heat exchanger area while the capacity rate ratio decreased by 50%. Note also that Figure 2.1.1.3 has no direct cost effectiveness implications. If with a constant average capacity rate one begins working with exchangers

Figure 2.1.1.4

Optimum Capacity Rate Ratio Versus the Mean Flowrate -Specific Heat Product

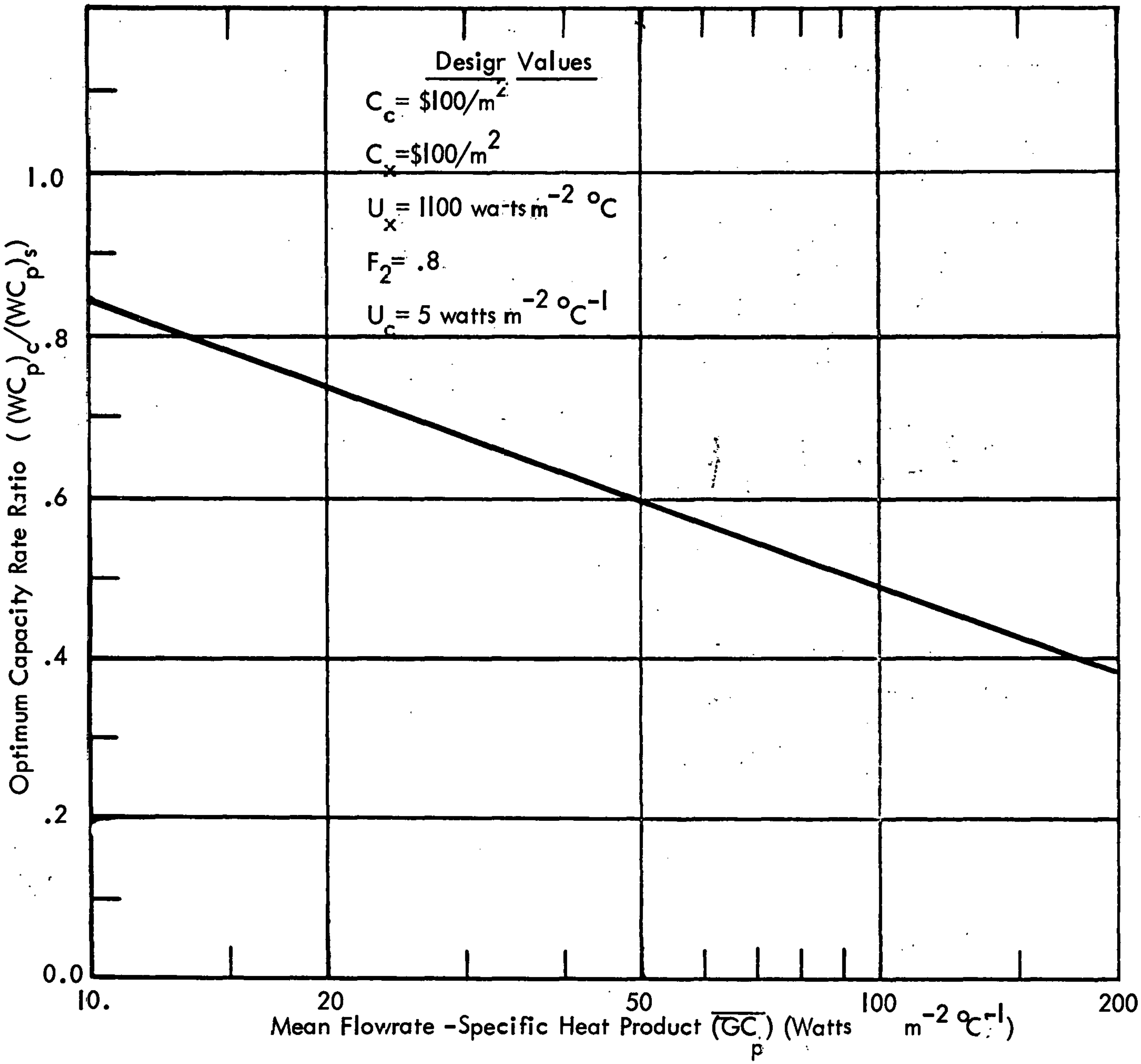


Figure 2.1.1.5

Correction Factors for the Optimum Capacity Rate Ratio (C')

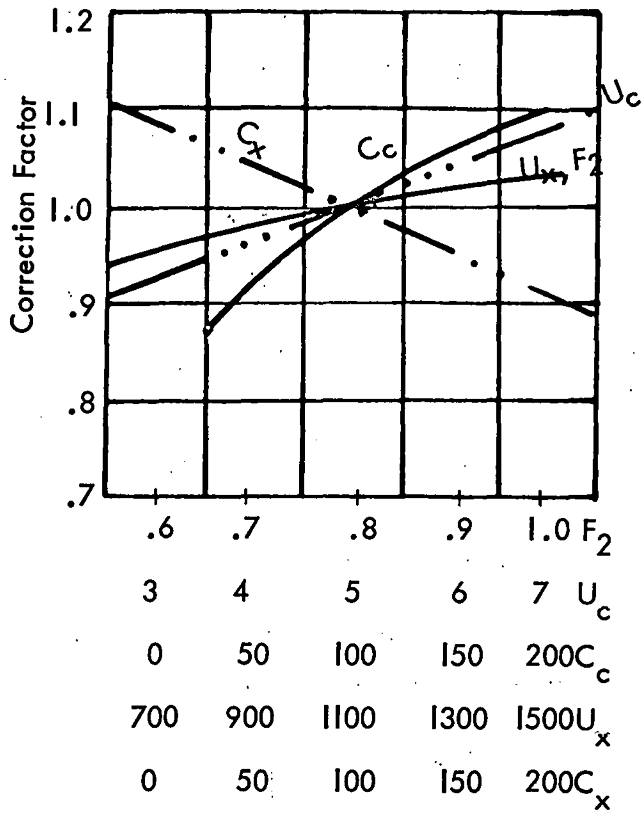


Figure 2.1.1.6

Cost to Heat Exchanger Factor Ratio Versus Capacity Rate Ratio and Mean Mass Flowrate

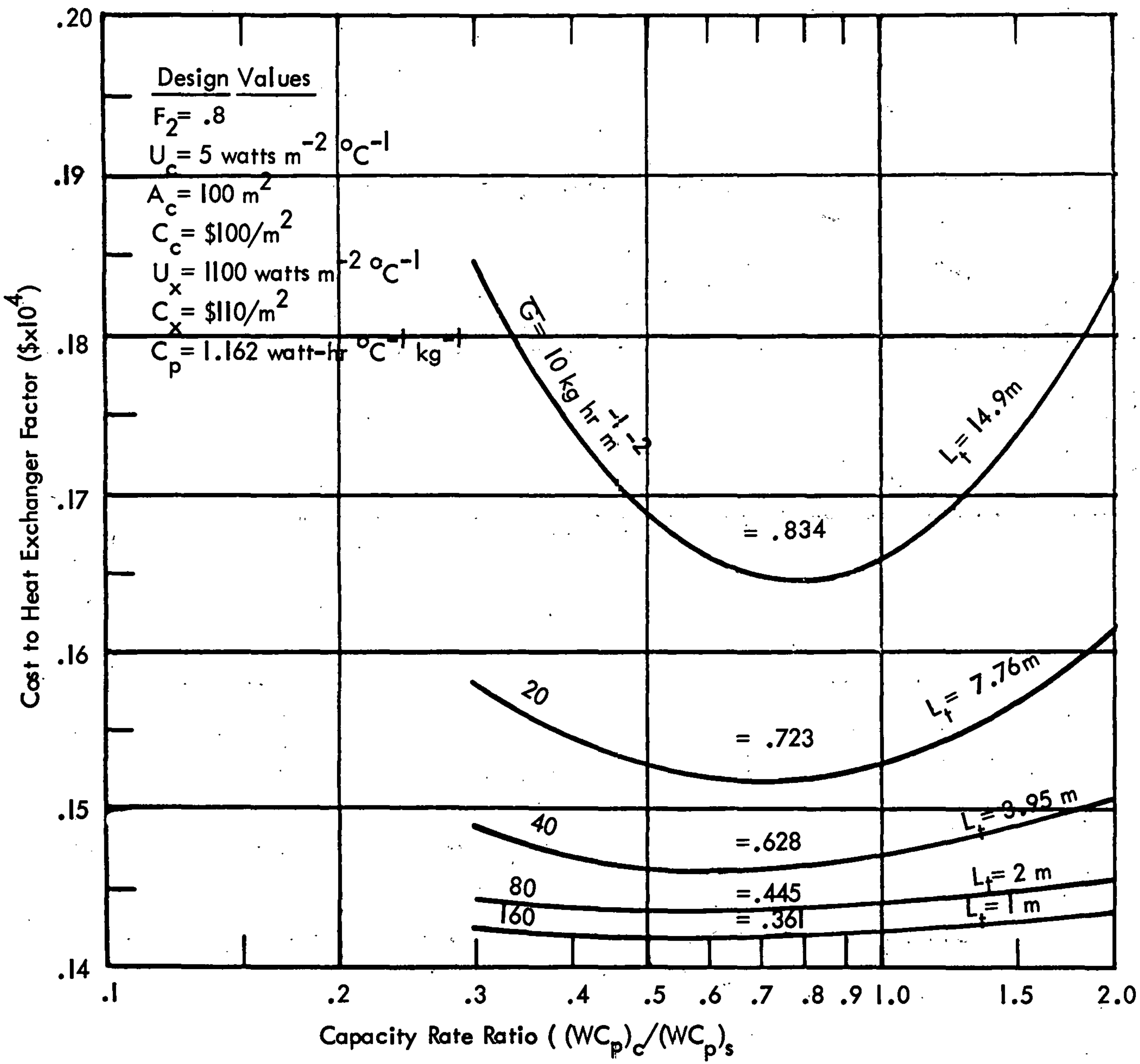
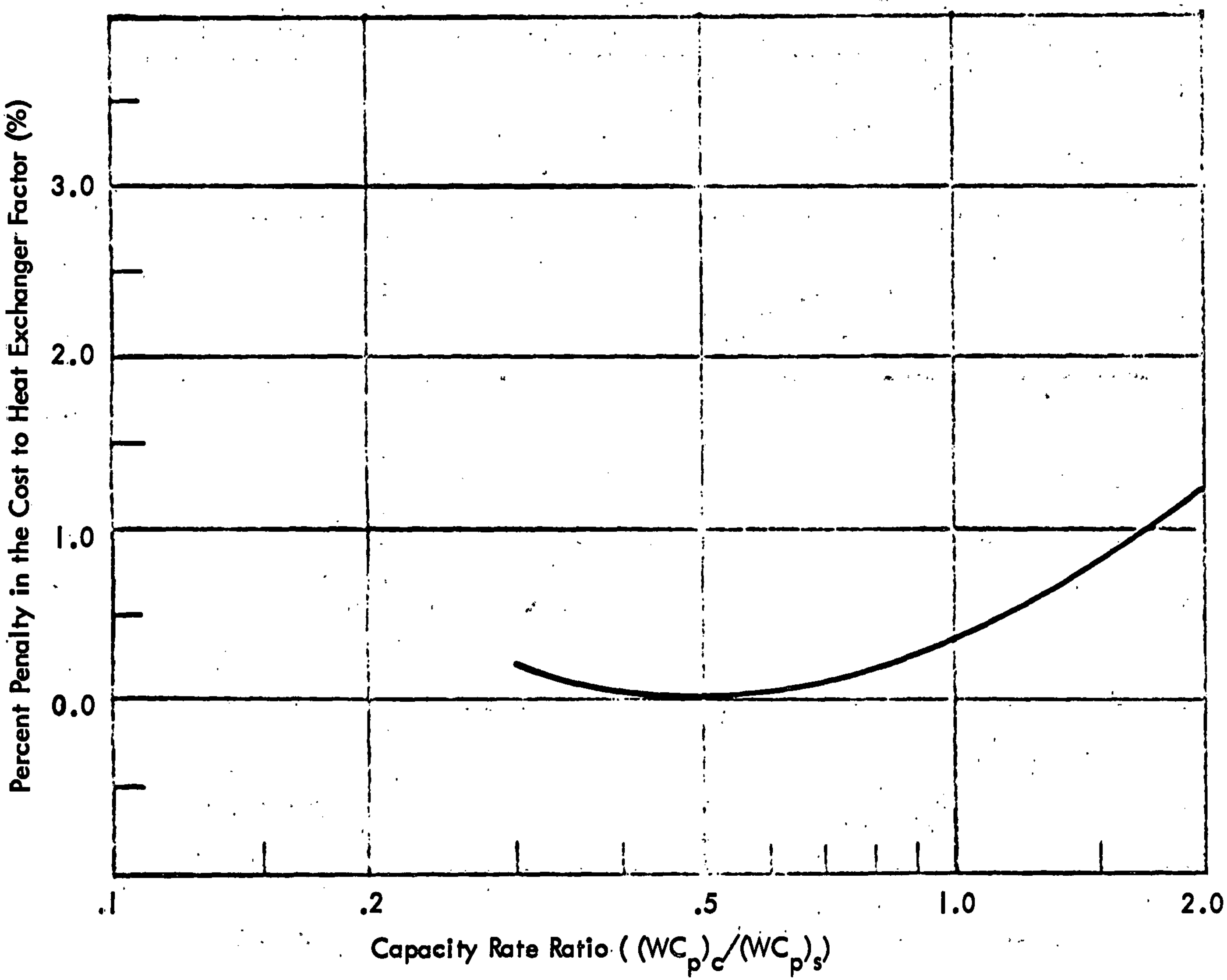


Figure 2.1.1.7

Percent Penalty in the Cost to Heat Exchanger Factor Ratio Due to Off-Optimum Capacity Rate Ratio

($\overline{GC_p}$ = 80 watts $m^{-2}\ {}^{o}C^{-1}$)



in which the capacity rates are more and more apart, temperature profiles in the exchanger become sharper and the law of diminishing return sets in earlier.

Figure (2.1.1.4) shows the optimum capacity rate ratio versus mean capacity rate per unit collector area for typical design parameters. From this figure it can be seen that the optimum capacity rate ratio lies between 0.5 and 0.6 for typical systems. Note that for matched capacity rates to be optimum, the mean capacity rate must be small. For an optimized system with small mean capacity rates, the heat exchanger tends to be of high efficiency (see Figure 2.1.1.6 for examples). Such systems are not typical. Although the terms C_c, C_x, U_x, F_2 and U_c are not completely independent of $\overline{GC}_p$ and the other terms, mean correction factors over representative values are shown in Figure 2.1.1.5 From this it can be seen that the controlling parameter is $\overline{GC}_p$, with the other terms causing less than 10% changes in the optimum capacity rate ratio when varied up to 100%.

Figure 2.1.1.6 shows the cost/heat exchanger factor versus capacity rate ratio for typical design conditions. It shows that for typical systems (i.e. G= 8-16 lb ft^{-2} hr^{-1}) the effect of off optimum capacity rate ratios is very small. For smaller mean operating mass flowrates, the effect of off-optimum capacity rates is increased. Also displayed on this figure are the effectiveness of the matched capacity rate case for each mass flowrate and tube length required to keep the overall heat transfer coefficient fixed. To use even a 8 lb ft^{-2} hr^{-1} mass flowrate under these conditions requires the tube length to be doubled from that required at 16 lb ft^{-2} hr^{-1}.

Figure 2.1.1.7 presents the percent penalty due to off-optimum capacity rate ratios for a mass flowrate of approximately 16 lb ft^{-2} hr^{-1}. This figure reiterates that the capacity rate ratio can be varied from approximately 0.25 to 1.1 with the per cent penalty to the rate of heat transferred being less than 0.5%. Thus considerable tolerance is allowable near the optimum capacity rate ratio and that even matched capacity rates would not lead to a considerable penalty to the rate of heat transferred. Note capacity rates greater than one would require larger penalties than are shown here because the collector heat losses would be increased due to the higher operating collector temperatures.

For the remainder of this report, it has been assumed that the optimum capacity rate ratio is 0.5. In many instances comparisons will be based on matched capacity rates for ease of comparison.

2.1.2 Effect of Other-Than-Optimum Heat Exchanger Design

As discussed above, the optimum heat exchanger design is a counterflow heat exchanger with a heat transfer area picked so as to minimize the total investment cost per unit heat collected. There are many possible ways in which an other-than-optimum heat exchanger may end up in a solar system. Some of these include:

1. The method of determining the optimum heat exchanger area.
2. No exchanger models available with precisely the same size as required for optimum performance.
3. Uncertainties in the determination of the input parameters.
4. Increased scale deposits with time within the heat exchanger, reducing the heat exchanger performance.

Each of these effects on the system performance is investigated in the following sections. Figure 2.1.4.2 is reproduced here to show the difference in the optimum heat exchanger area calculated when the capacity rate ratio is 0.5 or 1. Assuming the capacity rates are matched leads to an optimum heat exchanger area slightly higher than required if the actual capacity rate ratio is less than one. Thus, using Equation (2.1.4.5) to determine the optimum heat exchanger leads to a slight off-optimum result with more heat exchanger area than required for mismatched cases. For the case of Figure 2.1.4.2, this effect is very small. The extra heat exchanger area predicted by Equation (2.1.4.5) is less than 1% higher, representing an increase in cost of $5.5 and an increase in the cost to heat transferred ratio of approximately 0.35%. Thus the effect of the extra cost is reduced by slightly increased heat transfer. For typical systems, such as this, the slight off-optimum design predicted by Equation (2.1.4.5) when the capacity rate ratio is other than one can be ignored.

Off-optimum heat exchanger sizes can occur if there are no heat exchanger models available with precisely the same size or characteristics as is required by Equation (2.1.4.5). As noted in Section 2.1.4, not only is the exchanger area important, but the number of tubes required is also critical. If the number of tubes used greatly exceeds the number determined by Equation (2.1.4.9), the heat

Figure 2.1.4.2

Percent Increase in the Cost to Heat Exchanger Factor Ratio Versus the Exchanger Area and the Capacity Rate Ratios of .5 and 1 for a Typical System

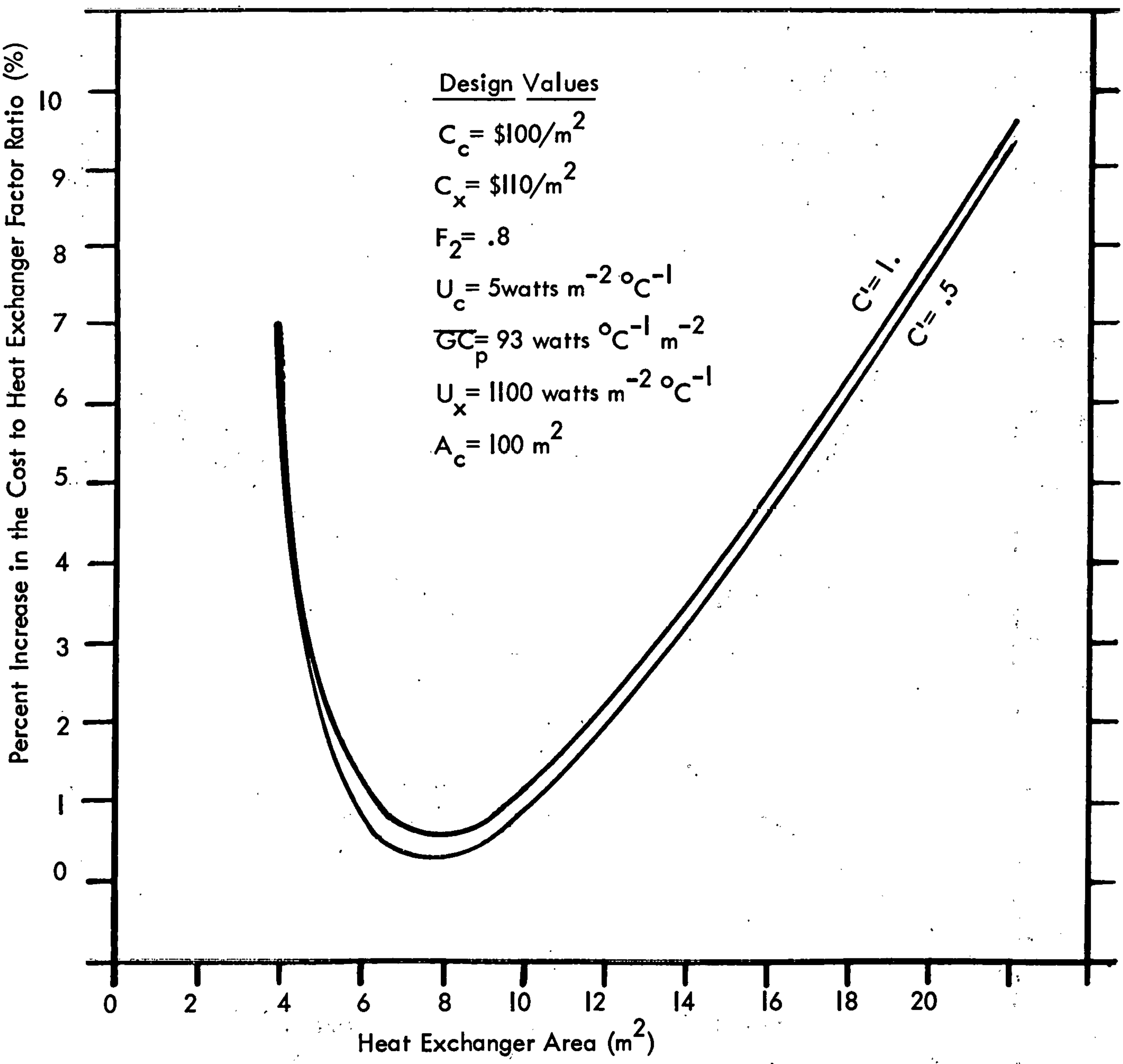
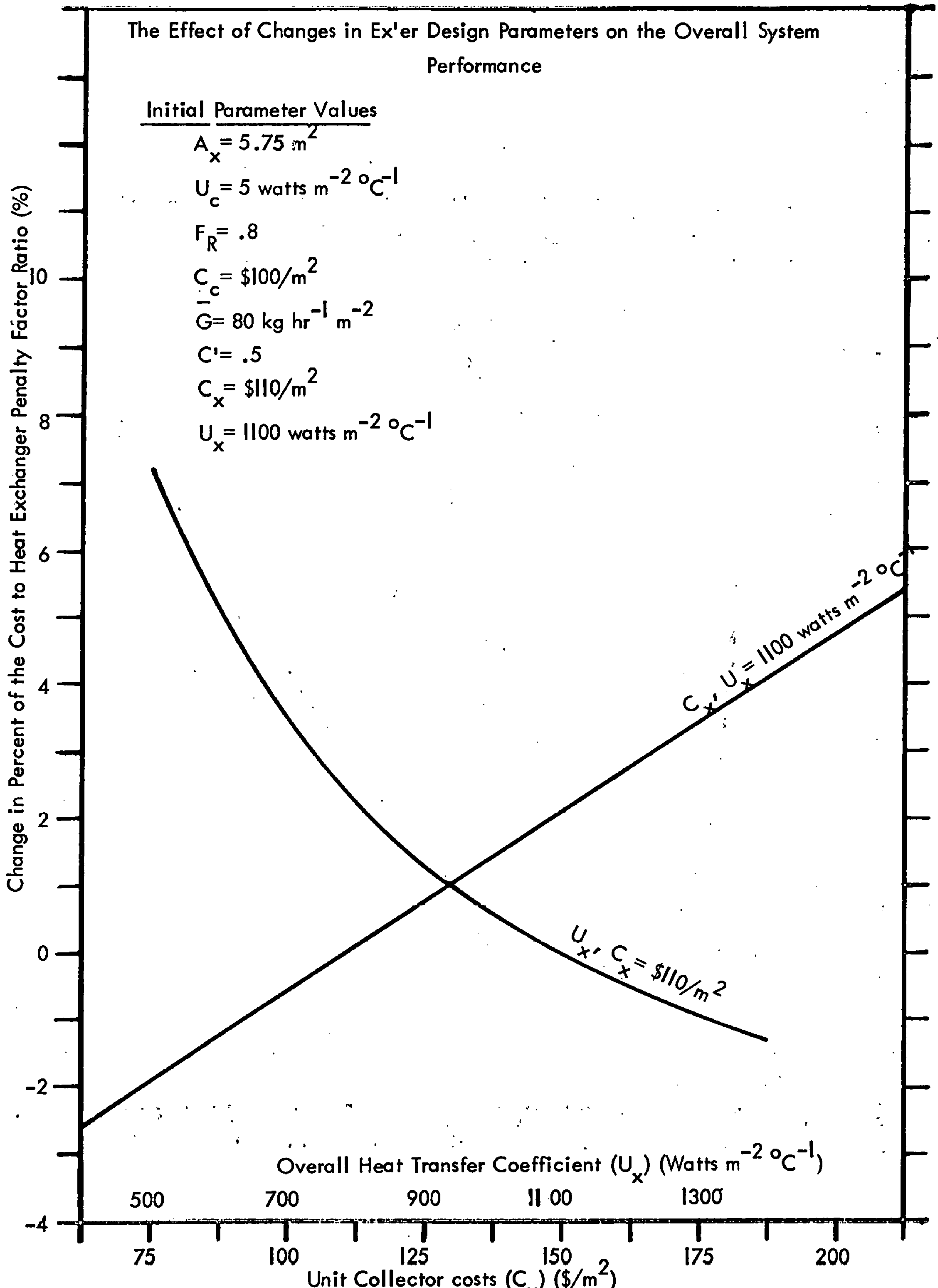


Figure 2.1.2.1

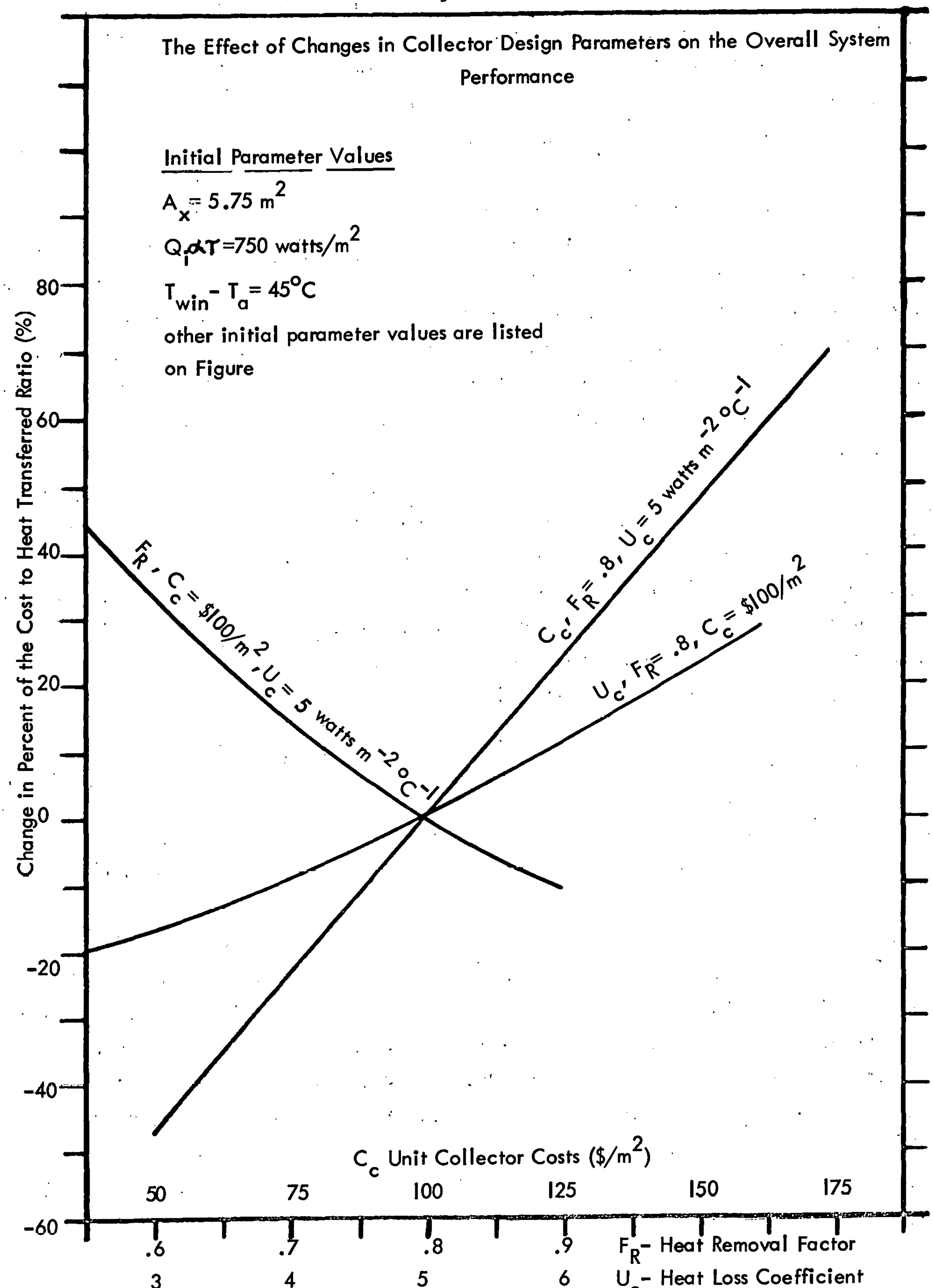


The Effect of Changes in Ex'er Design Parameters on the Overall System Performance
Initial Parameter Values
A_x = 5.75 m^2
U_c = 5 watts m^-2 °C^-1
F_R = .8
C_c = $100/m^2
G = 80 kg hr^-1 m^-2
C' = .5
C_x = $110/m^2
U_x = 1100 watts m^-2 °C^-1
Change in Percent of the Cost to Heat Exchanger Penalty Factor Ratio (%)
10
8
6
4
2
0
-2
-4
C_x, U_x = 1100 watts m^-2 °C^-1
U_x, C_x = $110/m^2
Overall Heat Transfer Coefficient (U_x) (Watts m^-2 °C^-1)
500
700
900
1100
1300
75
100
125
150
175
200
Unit Collector costs (C_x) ($/m^2)

transfer within the heat exchanger will suffer. If too few tubes are used, the length of the exchanger tubing can be excessive to meet the exchanger area required. Figure 2.1.4.2 also shows that off-optimum heat exchanger areas increase the cost to heat transfered ratio gradually. For example if the exchanger area actually used is $7 m^2$ instead of $5.63\ m^2$ (an increase of approximately 25%), the cost to heat transferred ratio will increase by less than 1%. Thus there is room to vary the exchanger area slightly to meet the standard exchanger sizes available and still not significantly affect the cost to heat transferred ratio. If possible, when the number of tubes needed lies between two standard exchanger tube numbers, the fewer number of tubes should be used which will increase the heat exchanger performance. This will also increase the tube length required. Since tube lengths are generally available in foot intervals, a close match between the needed exchanger area and that available from the manufacturer can be obtained.

The input parameters used in determining the optimum heat exchanger area can also affect the optimum performance if they change with time or are incorrectly specified initially. Figure 2.1.2.1 shows the effect on the cost to heat transferred ratio if the unit exchanger cost (C_x) or overall heat transfer coefficient (U_x) are mis-specified initially. This case was similar to that used as an example above, with the optimum exchanger area calculated from Equation 2.1.4.5 with the input parameters C_c=\$100/m^2, $F_R U_c$=4 watts $m^{-2}\ {}^oC^{-1}$, C_x=\$110/m^2, U_x=1100 watts $m^{-2}\ {}^oC^{-1}$, and C'=0.5. This yielded an exchanger area of $5.75\ m^2$. With A_x fixed, C_x and U_x were then allowed to vary to show the effect if these parameters were mis-specified initially. For example, if the actual exchanger cost was \$150/m^2 rather than \$110/m^2, then the exchanger would be too large for the optimum case and the cost to heat transferred ratio would be increased by approximately 2% due to the increased costs. If U_x was actually lower than the design value of 1100 watts $m^{-2}\ {}^oC^{-1}$ and equal to 800 (this is true if a non-aqueous fluid was used instead of an aqueous solution) then this reduction in the heat transfer of the exchanger would increase the cost to heat transferred ratio again by approximately 2%. In this case the amount of heat exchanger determined by Equation 2.1.4.5 initially would be too low. This would also be the case if the overall heat transfer coefficient was reduced due to increased scaling on either the tube or shell side

Figure 2.1.2.2


The Effect of Changes in Collector Design Parameters on the Overall System Performance
Initial Parameter Values
$A_x = 5.75\ m^2$
$Q_i \alpha\tau = 750\ watts/m^2$
$T_{win} - T_a = 45°C$
other initial parameter values are listed on Figure
Change in Percent of the Cost to Heat Transferred Ratio (%)
80
60
40
20
0
-20
-40
-60
$F_R,\ C_c = \$100/m^2,\ U_c = 5\ watts\ m^{-2}\ °C^{-1}$
$C_c,\ F_R = .8,\ U_c = 5\ watts\ m^{-2}\ °C^{-1}$
$U_c,\ F_R = .8,\ C_c = \$100/m^2$
C_c Unit Collector Costs ($\$/m^2$)
50
75
100
125
150
175
.6
.7
.8
.9
F_R - Heat Removal Factor
3
4
5
6
U_c - Heat Loss Coefficient

of the heat exchanger. Also if U_x is underpredicted or C_X is overpredicted then the system will have increased performance due to increased heat transfer for the former case and extra heat exchanger area for the latter.

The effect of mis-specification initially of the collector design parameters on the overall system performance is shown in Figure 2.1.2.2. Similar to the exchanger parameter case above, the exchanger area was assumed fixed and found from the initial values of the parameters shown on Figure 2.1.2.2. The unit collector cost (C_c), heat removal factor (F_R), and heat loss coefficient (U_C) were then allowed to vary. The heat loss coefficient (U_c) not only affects the heat exchanger penalty factor but also the rate of heat transferred even if water were to circulate directly through the collector (see Appendix D, for this term) for this case design values of 750 watts m^{-2} for $Q_i\alpha\tau$ and 45^oC for $T_{win} - T_a$ were used. For example, if the unit collector cost was increased from \$100/m^2 to 150 the cost to heat transferred ratio would increase by approximately 47%. For this case, the heat exchanger area used was too small and would have to be increased in order to reduce the effects of the increased collector costs. Similarly, if the heat loss coefficient increased (for example, if no cover plates were used) from 5 to 7 watts m^{-2} $^oC^{-1}$, this would increase the cost to heat transferred ratio by approximately 25%. Lastly, if the heat removal factor was reduced from 0.8 to 0.6, the cost to heat transferred ratio would increase by over 30%. Comparing the results of Figures 2.1.2.1 and 2.1.2.2, it can be seen that changes in the collector parameters affect the cost to heat transfer ratio more than changes in the exchanger terms. Since for a given collector, the F_RU_C product and C_c are quite readily obtained, these large initial errors in the design parameters listed above should not occur. In contrast, U_x and C_x can be quite difficult to determine or obtain, so that mis-specification of these parameters could be quite common. Since the error caused by these variables is less, this also should allow adequate tolerance to the user in specifying heat exchanger equipment.

Conclusions

With care in the selection of the initial parameters used in the heat exchanger sizing process (especially the collector terms), the heat exchanger so chosen should be close to optimum. Determining the heat exchanger area assuming the capacity rates are matched leads to slightly higher heat exchanger areas than required for the mismatched cases. For typical systems, this effect is very small.

2.1.3 Criticality of Heat Exchanger Type

A comparison of heat exchangers of different flow geometries was conducted. Counterflow, parallel flow, crossflow (one fluid mixed), and parallel-counter-flow heat exchangers were compared. See Figure 2.1.3-1 for the flow patterns through these exchangers. The penalty imposed on heat transfer by the different exchanger types was determined.

The counterflow heat exchanger is the most effective heat exchanger type for transferring heat between two streams of matched capacity rates. The effectiveness of a single tube and shell pass counterflow exchanger (ϵ) from Kays and London (1964) is:

$$\epsilon = 1 - \exp\left[-NTU\left[1 - \frac{(WC_p)_{min}}{(WC_p)_{max}}\right]\right] \Big/ \left[1 - \frac{(WC_p)_{min}}{(WC_p)_{max}} \exp\left[-NTU\left[1 - \frac{(WC_p)_{win}}{(WC_p)_{max}}\right]\right]\right]$$

NTU - Number of heat transfer units

$$NTU = \frac{A_x U_x}{(WC_p)_{min}} \qquad (2.1.3\text{-}1)$$

A_x = Surface area of heat exchanger (m^2)

U_x = Overall heat transfer coefficient of the Heat exchanger $\frac{\text{watts}}{m^2\ {}^\circ C}$

$(WC_p)_{min}$ = smaller capacity rate watts/°C

$(WC_p)_{max}$ = larger of the two capacity rates watts/°C

Equation 2.1.3-1 reduces to the following for matched capacity rates:

$$\epsilon = \frac{1}{1 + \frac{1}{NTU}}, \qquad (WC_p)_{min} = (WC_p)_{max} \qquad (2.1.3\text{-}2)$$

For the parallel flow single tube and shell pass heat exchanger the effectiveness of the exchanger according to Kays and London (1964) is:

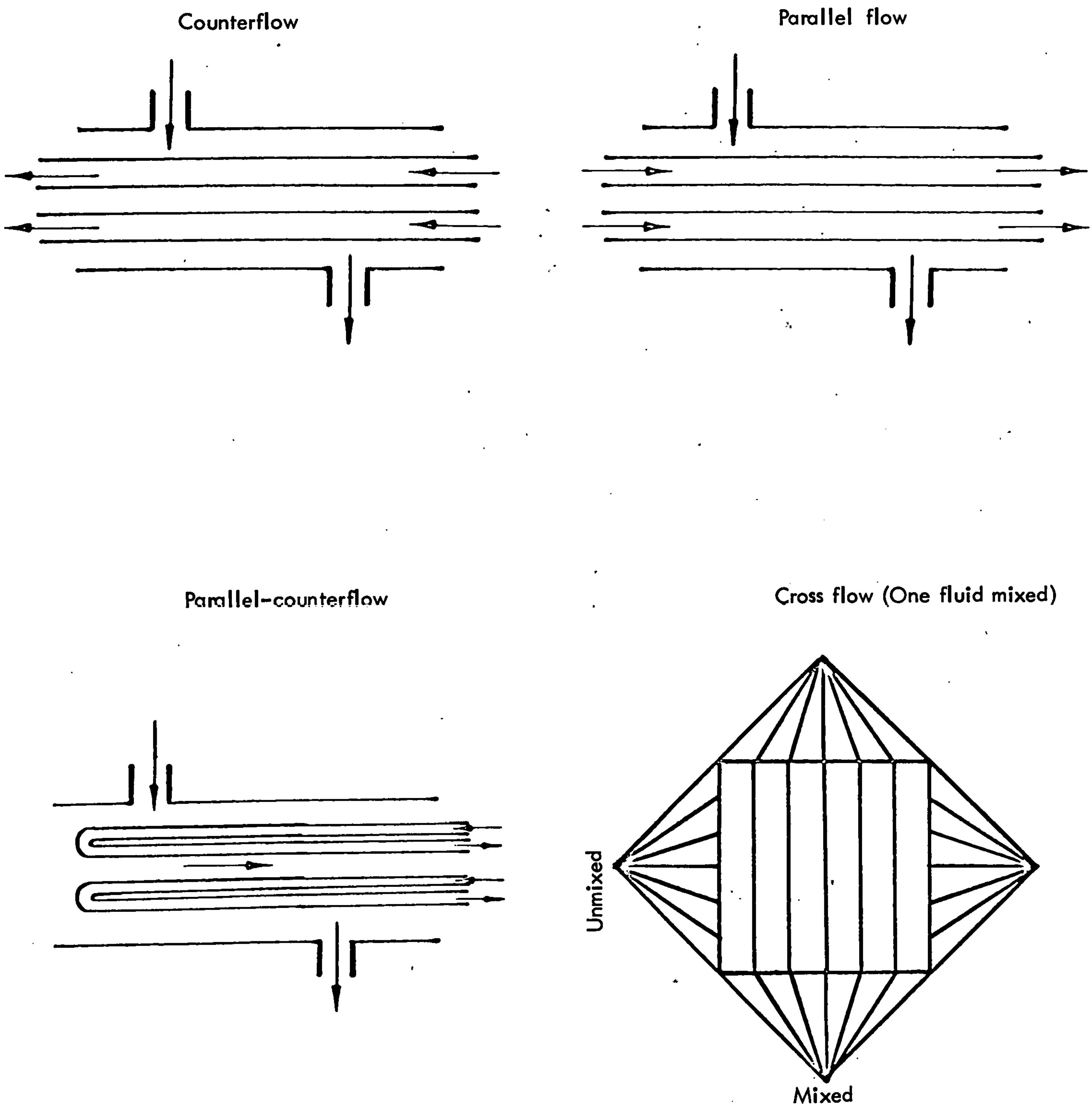
$$\epsilon = 1 - \exp\left[-NTU\left[1 + \frac{(WC_p)_{min}}{(WC_p)_{max}}\right]\right] \Big/ \left[1 + \frac{(WC_p)_{min}}{(WC_p)_{max}}\right] \qquad (2.1.3\text{-}3)$$

which reduces to the following for matched capacity rates:

$$\epsilon = 1 - \exp\left[-2NTU\right]/2, \qquad (WC_p)_{min} = (WC_p)_{max} \qquad (2.1.3\text{-}4)$$

Figure 2.1.3.1

Flow Geometries of Selected Heat Exchanger Types



One of the most common exchanger designs is one utilizing a "U" tube design, a parallel-counterflow exchanger. The effectiveness of the parallel-counterflow exchanger according to Kays and London (1964) is:

$$\epsilon = \frac{2}{1 + \frac{(WC_p)_{min}}{(WC_p)_{max}} + \sqrt{1 + \left[\frac{(WC_p)_{min}}{(WC_p)_{max}}\right]^2} \left[\frac{1 + \exp[-\Gamma]}{1 - \exp[-\Gamma]}\right]} \qquad (2.1.3\text{-}5)$$

where $\Gamma = NTU \sqrt{1 + \left[\frac{(WC_p)_{min}}{(WC_p)_{max}}\right]^2}$

For the matched capacity rate case equation 2.1.3-5 reduces to:

$$\epsilon = \frac{2}{2 + \sqrt{2} \left[\frac{1 + \exp[-\Gamma']}{1 - \exp[-\Gamma']}\right]} \qquad (2.1.3\text{-}6)$$

where $\Gamma' = NTU\sqrt{2}$

For the cross flow exchanger (1 fluid "mixed," the other "unmixed") with $(WC_p)_{max} = (WC_p)_{unmixed}$, $(WC_p)_{min} = (WC_p)_{mixed}$. The effectiveness becomes:

$$\epsilon = 1 - \exp\left[-\Gamma \frac{(WC_p)_{max}}{(WC_p)_{min}}\right] \qquad (2.1.3\text{-}7)$$

where: $= 1 - \exp\left[-NTU \frac{(WC_p)_{min}}{(WC_p)_{max}}\right]$.

For matched capacity rates the effectiveness of the cross flow heat exchanger becomes:

$$\epsilon = 1 - \exp\left[-\Gamma'\right] \qquad (2.1.3\text{-}8)$$

$$\Gamma' = 1 - \exp\left[-NTU\right]$$

The penalty imposed by these different heat exchangers on heat transfer was determined by using equation 2.1.1-3 i.e.:

$$F_{ex}' = \frac{1}{1 + \frac{F_R U_c A_c}{(WC_p)_c} \left[\frac{(WC_p)_c}{\epsilon (WC_p)_{min}} - 1\right]}$$

where F_{ex}' = penalty imposed on heat transfer due to the heat exchanger.

$(WC_p)_c$ - capacity rate of collector fluid [watts/°C]

For comparison of the heat exchanger types, the capacity rates of the heat transfer fluid and water were assumed matched. The effect of the exchangers on heat transfer was determined, assuming that NTU was constant between heat exchanger types.

Figures 2.1.3-2 through 2.1.3-4 show the variation of the heat exchanger penalty factor [F_{ex}'] versus heat exchanger type, NTU, and collector term $(WC_p)_c/(F_R U_c A_c)$ From these figures it is apparent that the single tube and shell pass counterflow heat exchanger is more effective in transferring heat between two streams of matched flow rates than the cross flow, parallel flow or parallel-counterflow heat exchanger . Note the larger difference in performance between heat exchangers for lower collector term values, while for large collector term values the relative difference in performance is less. Thus for less efficient collectors [where $\frac{(WC_p)_c}{F_R U_c A_c}$ is small], the heat exchanger type becomes most critical in optimizing heat transfer.

An example to show the effect of heat exchanger type for a typical collector system follows.

Assume: $A_c = 100\ m^2$ $\quad F_R U_c = 5$ watts $m^{-2}\ {}^{o}C^{-1}$

$(WC_p)_c$ = 5000 watts/°C

An optimum heat exchanger size for this collector, assuming U_x = 1200 watts $m^{-2}\ {}^{o}C$ from equation 2.1.4-5 is: $A_x = 8.4\ m^2$

Therefore $NTU = \frac{A_x U_x}{(WC_p)_{min}} = 2$ for this matched capacity rates case.

For these conditions the heat exchanger penalty for the differing types of heat exchangers becomes:

F_{ex}' = .9524 Counterflow

F_{ex}' = .9060 Parallel flow

F_{ex}' = .9263 Parallel-counterflow

F_{ex}' = .9322 Cross flow

Figure 2.1.3.2

Heat Exchanger Penalty Factor Versus Collector Term and Heat Exchanger Type with NTU = 5

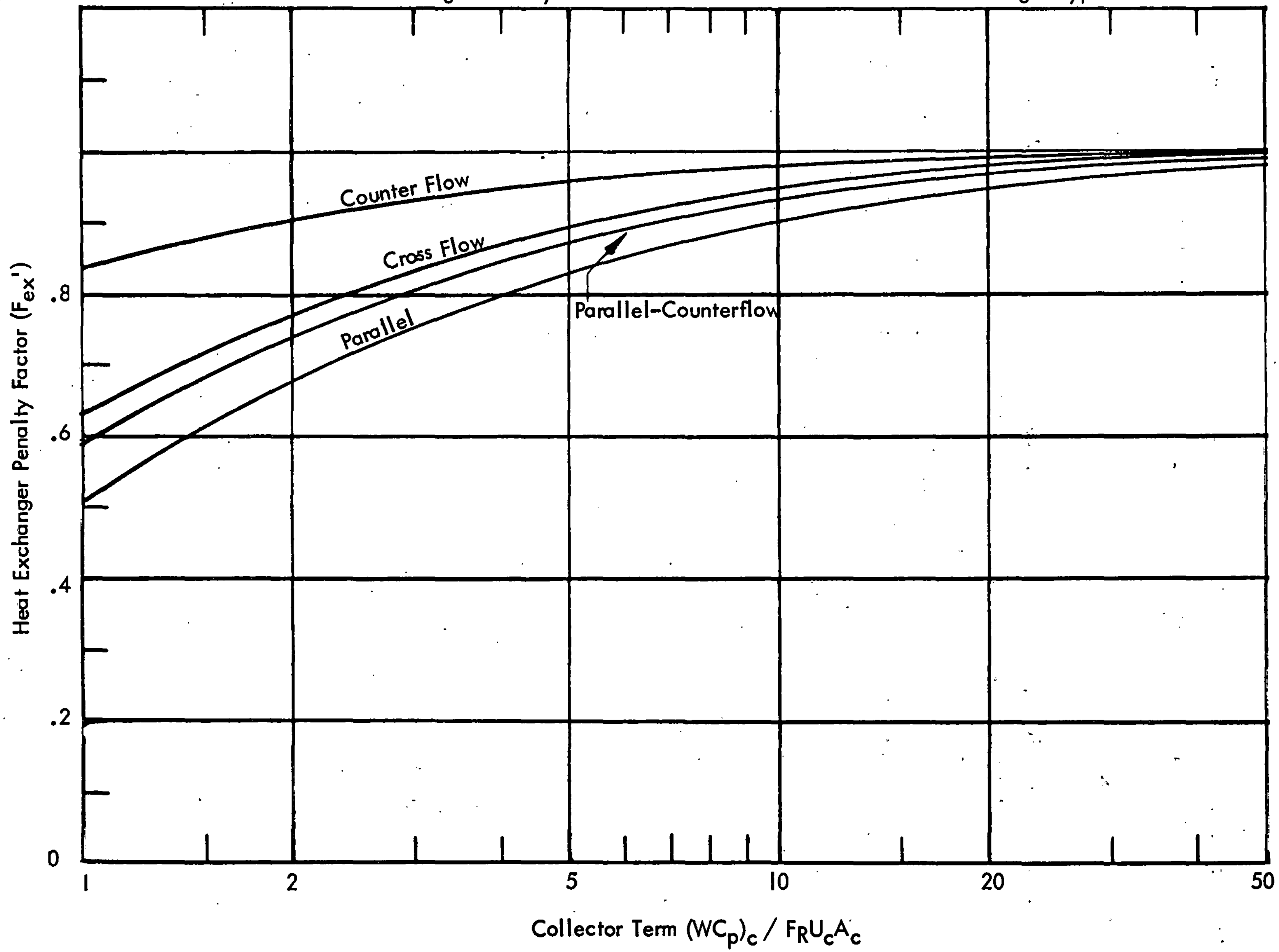


Figure 2.1.3.3

Heat Exchanger Penalty Factor Versus Collector Term and Heat Exchanger Type with NTU = 2

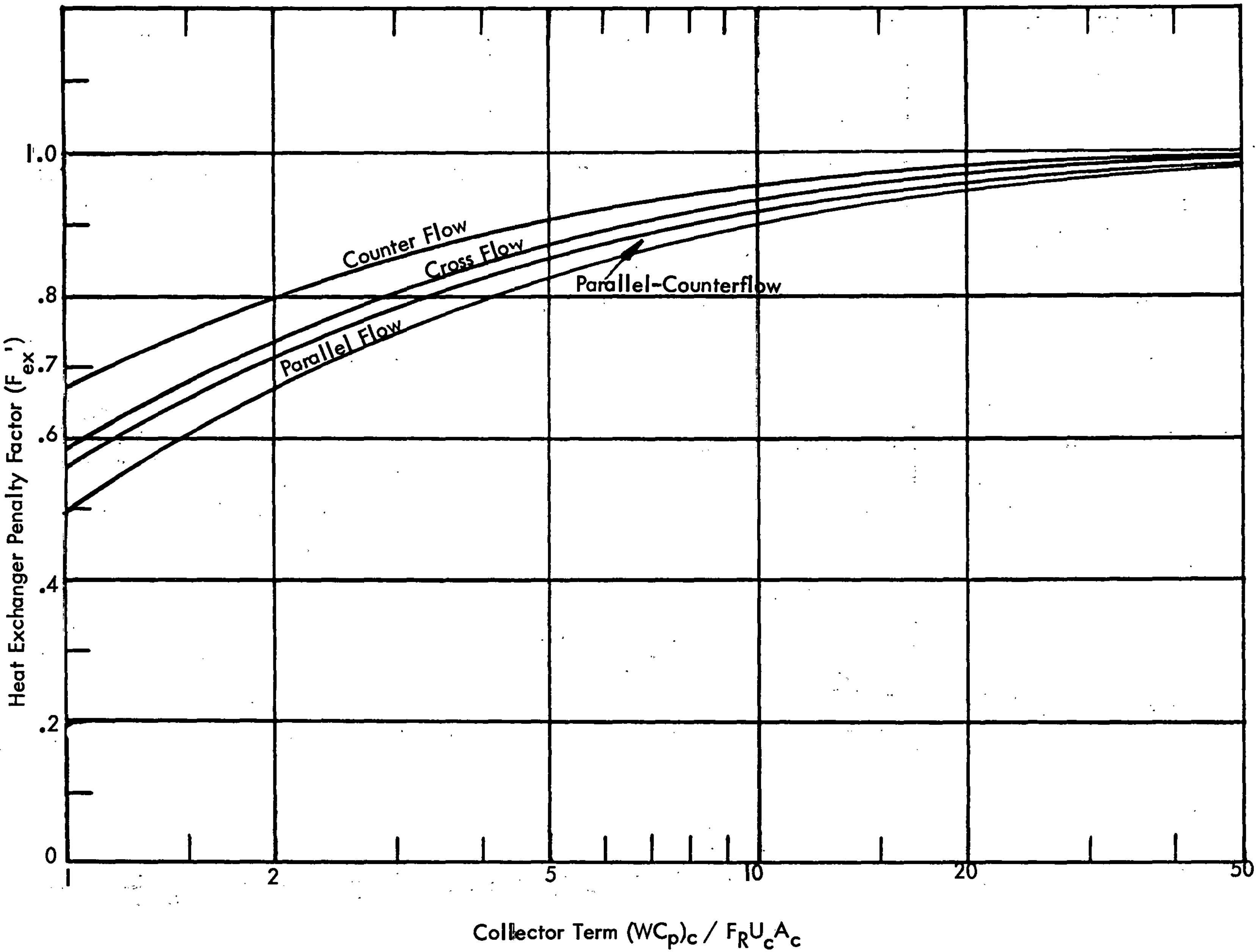
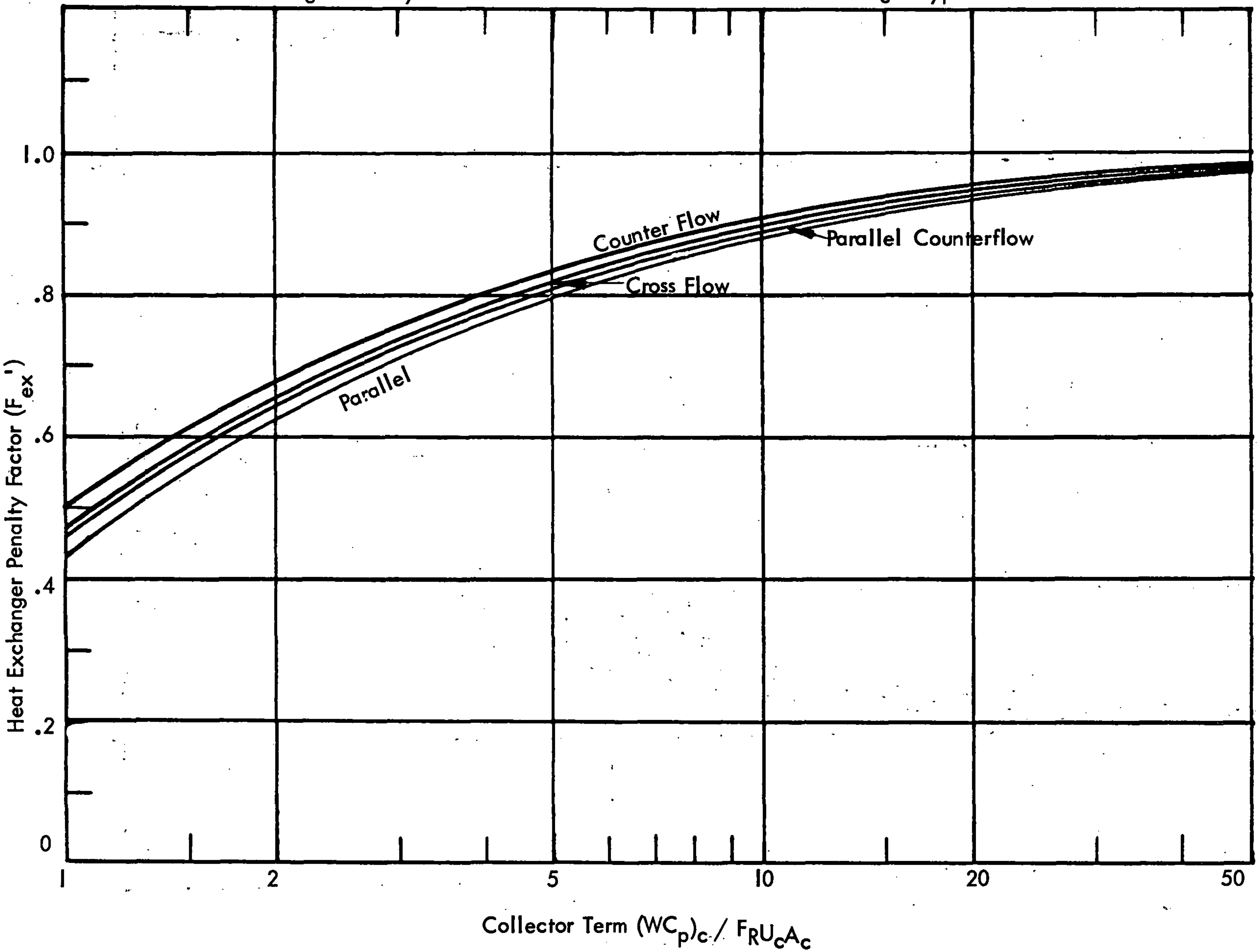


Figure 2.1.3.4

Heat Exchanger Penalty Factor Versus Collector Term and Heat Exchanger Type with NTU = 1



Thus the use of a parallel flow heat exchanger rather than a counterflow exchanger will result in a further reduction in heat transfer of 5%. The cross flow will reduce performance by 2% compared to the counterflow heat exchanger while the parallel-counterflow heat exchanger will reduce the heat transfer by 3%.

2.1.4 Determination of Simple and Complete Procedures for Establishing the Specifications of Optimum Heat Exchangers

An important part of this study was to establish a method to determine the optimum heat exchanger area for applications to solar energy design. De Winter (1975) (included as Appendix A) determined the optimum heat exchanger area assuming the optimum heat exchanger would be a counterflow exchanger with the capacity rates of the two loops (storage and collector) matched. As was shown in Section 2.1.1, a mismatch in the capacity rates of the two loops leads to more optimum performance, although the effect is small for typical systems.

In the following section, the optimum heat exchanger area is determined for the case when the capacity rates are mismatched. Following this, a discussion of the terms affecting the optimum exchanger area is presented. Lastly, simple procedures for establishing specifications of the optimum heat exchanger are further discussed.

Determination of Optimum Heat Exchanger Area for Mismatched Capacity Rates Within Double Loop Heat Exchanger Systems

The optimum heat exchanger area is one for which the total cost to heat transferred ratio is a minimum. If the heat exchanger area exceeds this, then the cost increases more rapidly than the amount of heat transferred and the overall performance diminishes. If the heat exchanger area is less than optimum, although the total cost is reduced, then the heat transfer suffers. As shown in Section 2.1.1, the optimum counterflow exchanger should operate with the capacity rate ratio ($C' = (WC_p)min/(WC_p)max$) less than one (matched) for all typical applications. De Winter (Ref. C-a-1) assumed the matched case would be optimum due to higher exchanger performance. Although the effect of the mismatched capacity rates is small on the optimum heat exchanger area, it is investigated thoroughly in the following section.

The optimum heat exchanger area (Ax) occurs when:

$$\frac{d}{dA_x}\left[\frac{C_t}{Q}\right] = 0 \qquad (2.1.4.1)$$

Where:

C_t = total cost of the system including all components of the double loop heat exchanger system and collector = $C_c A_c + C_x A_x$

C_c = collector costs/m^2 of collector (includes additional cost of the second pump required for the double loop system)

A_c = collector area (m^2)

C_x = heat exchanger cost/m^2 of exchanger

Q = rate of heat transferred (watts)

$= F_{ex}' F_R A_c [Q_i F_R \alpha\tau - U_c(T_{win} - T_a)]$

F_{ex}' = heat exchanger penalty factor (See Section 2.1 for derivation of this term.)

$$= \frac{1}{1 + \frac{F_R U_c A_c}{(WC_p)_c} [\frac{(WC_p)_c}{(WC_p)_{min}} - 1]}$$

ϵ = exchanger exchanger effectiveness

$$\epsilon = \frac{1}{1 + \frac{(WC_p)_c}{U_x A_x}} \quad \text{for} \quad C' = \frac{(WC_p)_{min}}{(WC_p)_{max}} = 1 \quad \text{(matched case)}$$

$$\epsilon = \frac{1 - \exp[-\frac{A_x U_x}{(WC_p)_{min}}(1 - C')]}{1 - C' \exp[-\frac{A_x U_x}{(WC_p)_{min}}(1 - C')]} \quad \text{for } C' \neq 1 \text{ (mismatched case)}$$

See Section 2.1.3 for a discussion of the heat exchanger effectiveness.

F_R = collector efficiency factor

U_c = collector heat loss coefficient (watts/m^2 ^oC)

U_x = overall heat transfer coefficient of counterflow heat exchanger (watts/m^2 ^oC)

A discussion of the capacity rates is included in Section 2.1.1

For this section let:

$(WC_p)_c = A_c \overline{GC_p} (1 + F)$

$(WC_p)_s = A_c \overline{GC_p} (1 - F)$

F = Deviation from the mean capacity rate $= \dfrac{C'-1}{C'+1}$

$\overline{GC}_p$ = mean capacity rate per unit collector area (watts/m^2 °C)

Equation (2.1.4.1) from the above definitions becomes:

$$\frac{d}{dA_x} \left(C_cA_c + C_xA_x \right) \left\{ 1 + \frac{F_RU_cA_c}{(WC_p)_c} \left[\frac{(WC_p)_c}{(WC_p)_{min}\,\epsilon} - 1 \right] \right\} = 0 \qquad (2.1.4.2)$$

Solving Equation (2.1.4.1) leads to:

$$\frac{(WC_p)_{min}}{(WC_p)_c} - \frac{(WC_p)_{min}}{F_RU_cA_c} - \frac{1}{\epsilon} + \frac{U_x}{(WC_p)_{min}} \left[\frac{A_cC_c}{C_x} + A_x \right] \text{expy} \left[\frac{1 - C'}{1 - \text{expy}} \right]^2 = 0$$

(2.1.4.3)

where $Y = - \dfrac{A_xU_x}{(WC_p)_{min}} (1 - C')$

Equation (2.1.4.3) could not be reduced further but was solved by iteration for the optimum heat exchanger area (A_x). By inspection it was found that:

$$A_x = F(A_c, U_c, F_R, U_c, U_x, C_x, GC_p, C')$$

Thus in contrast to the matched optimum heat exchanger area which from de Winter (1975) is:

$$A_x = A_c \sqrt{\frac{F_RU_cC_c}{U_xC_x}} \qquad (2.1.4.4)$$

the optimum heat exchanger area is now dependent on both the mean capacity rate per unit collector area (GC_p) and the capacity rate ratio (C'). Actually, even the matched optimum heat exchanger area is slightly dependent on the mean capacity rate per unit collector area and capacity rate ratio since:

$$F_R = \frac{(WC_p)_c}{U_cA_c} \left[1 - \exp\left(- \frac{F_2U_cA_c}{(WC_p)_c} \right) \right]$$

Where F_2 = Section efficiency of the collector.

Figure 2.1.4.1 shows the optimum exchanger area versus the capacity rate ratio C' and $\overline{GC}_p$ for a typical collector (A_c = 100 m^2, F_2 = .8, U_c = 5 watts/m^2 °C, C_c = \$110/m^2) and exchanger (U_x = 1100 watts/m^2 °C, C_x = \$110/m^2).

The overall heat transfer coefficient corresponds to an aqueous heat transfer fluid such as an ethylene or propylene glycol solution. Figure 2.1.4.1 shows that for mismatched capacity rates, the optimum heat exchanger area is less. It should be noted that Figure 2.1.4.1 does not give an indication of cost effectiveness. The most cost effective capacity rate ratio is always below 1, as shown in Figure 2.1.1.4. At capacity rate ratios greater than one the optimum area may decrease, but the capacity rate ratio is far from optimum anyway so the cost effectiveness is lower. Also note that at lower mean capacity rates per unit collector area, the effect of the mismatched capacity rates is increased and that the optimum exchanger area is reduced. Again this does not imply that lower flowrates are better or more cost effective, but simply that if one chooses lower flowrates, increasing heat exchanger area is not as beneficial. As discussed in Section 2.1.1 and later in this Section these lower flow rates are clearly impractical since the length of the tubes required within the heat exchanger becomes much too long to keep the flow turbulent on the storage side (water) of the heat exchanger. Thus for the higher operating flowrates, the effect of the capacity rate ratio on the optimum heat exchanger area is markedly reduced.

Figure 2.1.4.2 shows the increase in the cost to heat transferred ratio (in percent) versus exchanger area for capacity rate ratios of 1.0 and 0.5. The operating conditions are similar to those of Figure 2.1.4.1 with the mean mass flowrate $G = 80$ kg/hr m^2 (C_p was assumed to correspond to water and equal to 1.162 watt hr/kg ^oC. Note that the optimum heat exchanger area for a capacity rate ratio equal to 1 is slightly less than 1% larger than that required if the capacity rate ratio used was 0.5. The increase in the cost to heat transferred ratio for the optimum case with C'= 1.0 is 0.35% compared to that obtained if the capacity rate ratio was 0.5. Also note that the effect of off-optimum heat exchanger areas for both C' equal to 1 and 0.5 on the increase in the cost to heat transferred ratio are similar.

Figure 2.1.4.1

Optimum Heat Exchanger Area Versus the Capacity Rate Ratio and Mass Flowrate

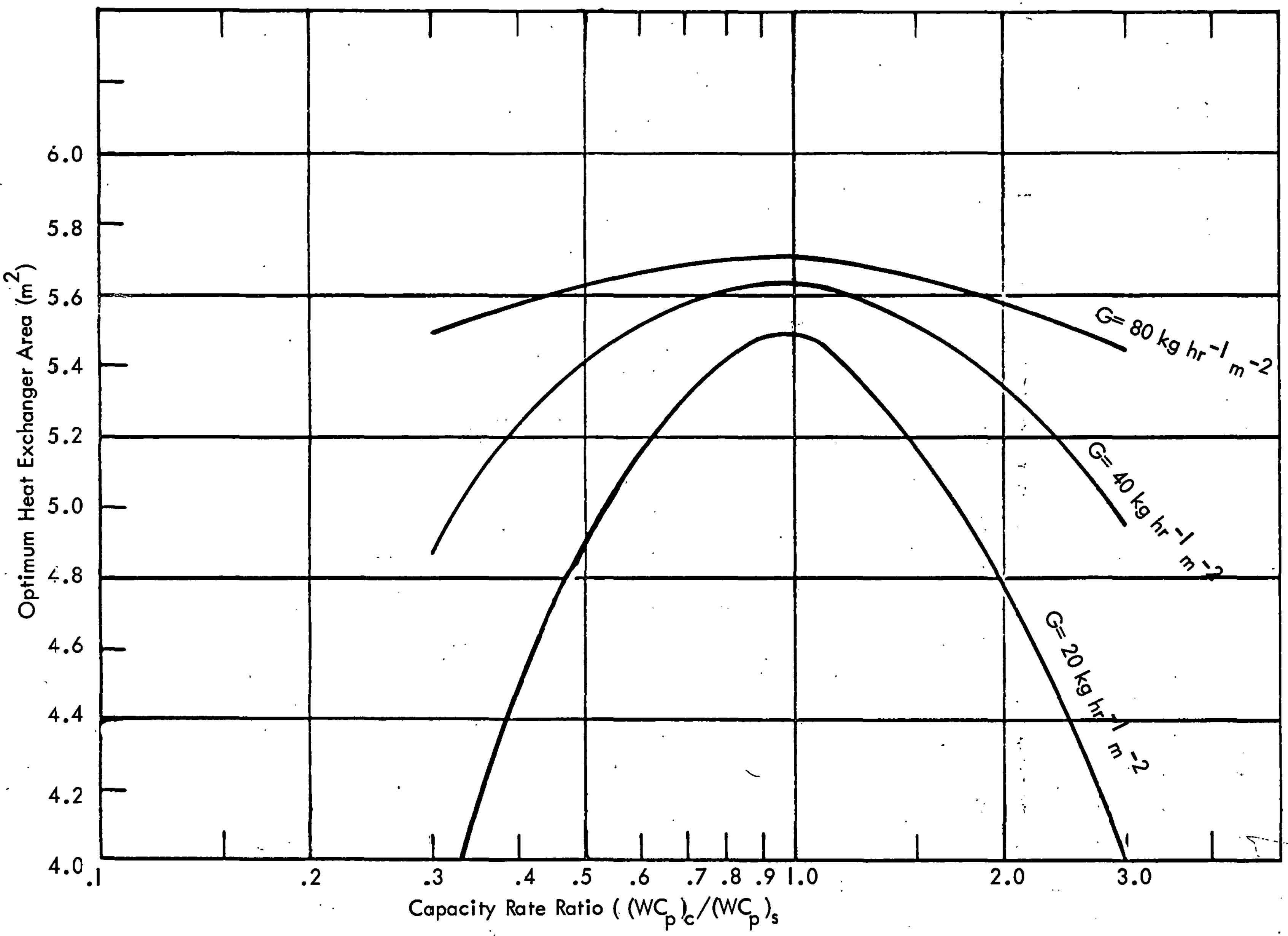
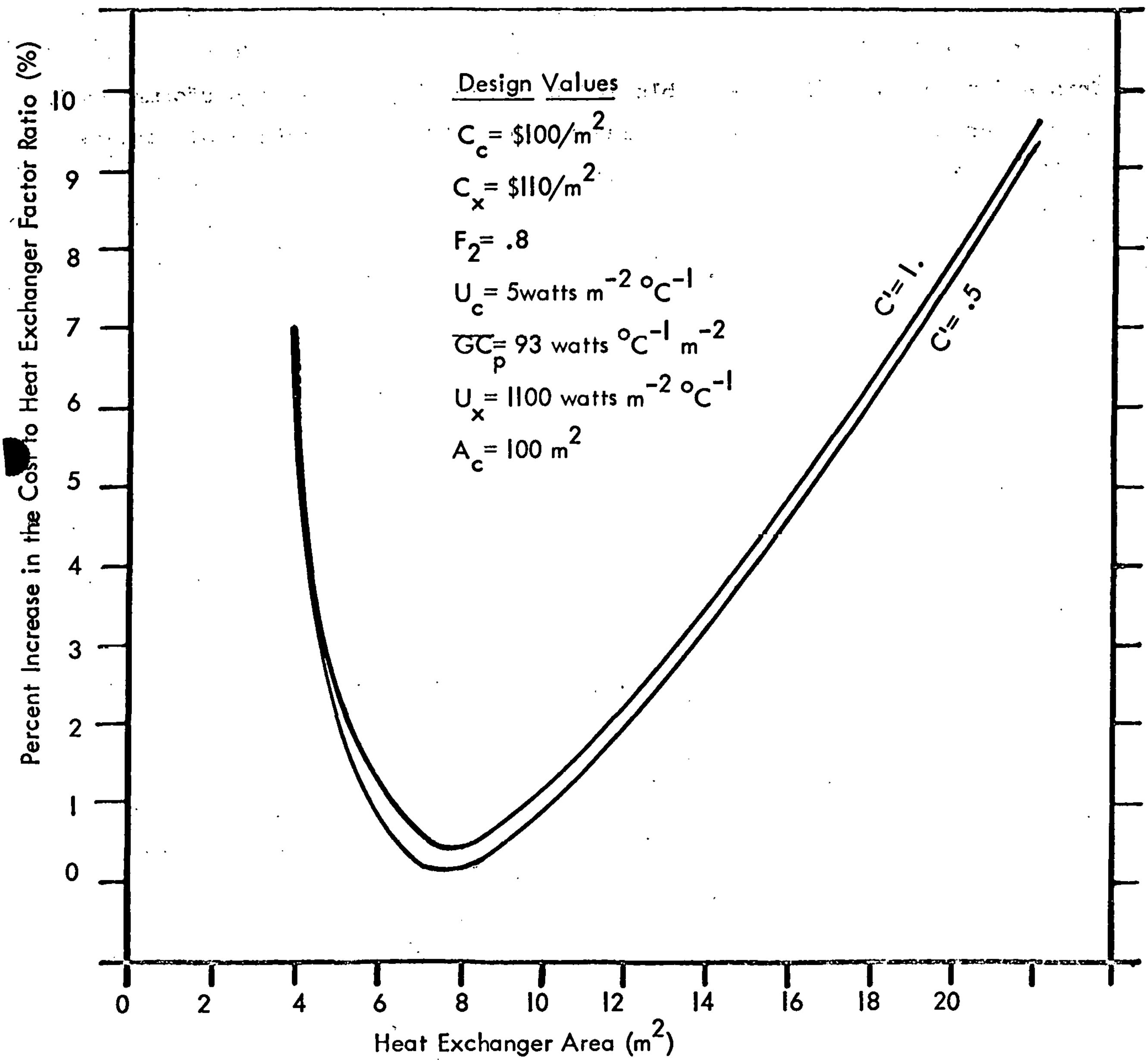


Figure 2.1.4.2

Percent Increase in the Cost to Heat Exchanger Factor Ratio Versus the Exchanger Area and the Capacity Rate Ratios of .5 and 1 for a Typical System



Conclusions

For the higher operating flowrates, which are required when the heat exchanger double loop system is used, it is apparent that there is little difference between the optimum heat exchanger area predicted by the matched case developed by de Winter (1975) and that predicted by Equation (2.1.4.3). For the above typical case using equation 2.1.4.4. leads to an additional cost of $5.5 and increases the cost to heat transfer ratio by 0.35%. This additional cost is negligible and the slight increase in performance due to the larger exchanger area will reduce this effect for most applications, while the ease in using equation (2.1.4.4) versus (2.1.4.3) is marked.

Discussion of the Terms Affecting the Optimum Heat Exchanger Area

As shown by de Winter (1975), the optimum heat exchanger can be found if the collector area (A_C), cost (C_C), and performance characteristics ($F_R U_c$) and the exchanger cost (C_x) and performance characteristics (U_x) are known. As noted before, A_x becomes:

$$A_x = A_c \sqrt{\frac{F_R U_c C_c}{U_x C_x}} \qquad (2.1.4.5)$$

A discussion of each of these terms and their effects on the optimum heat exchanger area follows. In later sections, the penalty imposed by variations in these quantities is investigated.

Increasing unit collector costs (C_c) make larger exchanger areas more cost effective. Since the collector costs vary widely due to the materials of construction and size (see Appendix B-3), it is felt that the unit collector costs should be determined for each collector for which a heat exchanger is to be sized. In many parts of this study, a constant unit collector cost of $\$100/m^2$ was used to allow simple comparisons of the cost effectiveness of the other components. For many collector types and sizes, this value would be clearly unacceptable.

The $F_R U_c$ product is a critical component of the heat exchanger sizing. For a manufactured collector, this quantity is not difficult to determine from the manufacturers' specification sheets for a given temperature of operation. Appendix B-3 further discusses the determination of the $F_R U_c$ product.

The unit exchanger costs (C_x) also affect the optimum heat exchanger area. Unfortunately, C_x is not completely independent of the exchanger size. Generally, the unit exchanger costs are lower for longer, larger shell diameter exchangers than their smaller counterparts (See Appendix B-3). Also there is a wide variation in exchanger costs depending on both O.E.M. discounts and the quantity ordered. Because of these difficulties, it is not recommended that a single unit area list price be used. Current costs and availability should be considered each time a heat exchanger is to be sized. In many parts of this study, a design cost of $\$110/m^2$ was used. This value should not be considered as a viable unit exchanger cost for all types and sizes of exchangers.

The last term, U_x (overall heat transfer coefficient of the exchanger), is also dependent on the exchanger area because of its dependency on the number of exchanger tubes. This effect and a method to eliminate the dependency of U_x on the exchanger area follows.

The exchanger area is completely specified when the outer tube diameter (D_o), tube length (L), and tube number (N) are known. Based on the outer surface area of the exchanger, A_x becomes:

$$A_x = \pi D_o L N \qquad (2.1.4.6)$$

Not only is the heat exchanger area important but the components D_o, L, N affect the heat transfer and final performance of the double loop heat exchanger system. The overall heat transfer coefficient is discussed in Appendix B-1 and depends on the tube side heat transfer coefficient, the shell side heat coefficient, and the tube diameter, thickness and thermal conductivity. Most importantly, the water within the tubes must be circulated at a high enough flowrate through each tube so that the water is kept within the turbulent regime and U_x high. The shell side coefficient can be maintained at a high enough level by manipulation of the baffle spacing and is independent of the exchanger area. Figure B.1-1 shows the inside tube heat transfer coefficient for water. From figure B.1-1, an empirical relationship was determined to find acceptable flowrates as a function of the inner tube diameter to maintain the water within the turbulent regime. It was found that the following relationship allowed a quick determination of plausible flowrates.

$$Q_N = D_i + 0.125$$

where

Q_N - flowrate through each exchanger tube (gallons/minute)

D_i - inner tube diameter (in.)

Q_N can be expressed in terms of the mean flowrates operating in the double loop system by the expression:

$$Q_N = \frac{0.2494\ \bar{G}\ A_c}{N(C' + 1)} = D_i + 0.125 \qquad (2.1.4.7)$$

where

$\bar{G}$ - mean mass flowrate (lb/ft^2 hr)

A_c - collector area (ft^2)

ρ - density of water = 61.3 lb/ft^3 at 150^oF

N - number of exchanger tubes

C'- capacity rate ratio

The number of tubes needed to maintain turbulent flow within the exchanger tubes becomes:

$$N = \frac{0.2494\ \bar{G}\ A_c}{(C' + 1)(D_i + 0.125)} \qquad (2.1.4.8)$$

If the number of tubes used is much larger than the above formula would determine, the heat transfer within the tubes would suffer, resulting in poorer heat exchanger performance.

Using equations (2.1.4.6) and (2.1.4.8), the tube length required to meet the optimum heat exchanger area required and to keep the flow within turbulent flow in the tubes becomes:

$$L = \sqrt{\frac{F_R U_c C_c}{U_x C_x}}\ \rho\ \frac{(C' + 1)(D_i + 0.125)}{D_o\ 0.2494\ \bar{G}}$$

If tube lengths larger than this are used, fewer tubes are required and the heat transfer increases, but if the tube length is too long, the heat exchanger will be unwieldy and difficult to use in most solar energy applications since the space available for the heat exchangers is usually quite small.

Equations (2.1.4.8) and (2.1.4.9) can be reduced using typical water properties and characteristics of the tubing. For typical tube thicknesses with D_o expressed in feet:

$$\frac{D_i + 0.125}{D_o} \approx 12.$$

From Figure 2.1.1.4, the optimum capacity rate ratio versus mean mass flowrate for a typical system is given. For that example, with $G=16\ lb/ft^2hr$, the optimum capacity rate ratio is approximately 0.5 with $\sqrt{F_R U_c C_c/(U_x C_x)}$ equal to 0.0568. For these conditions with 3/8" nominal O.D. tubing, the number of tubes and tube lengths become:

$$N = 0.0868\ A_c\ (ft^2) \qquad L = 5\ ft.$$
$$= 0.9343\ A_c\ (m^2) \qquad =1.5\ m$$

Thus, for this example for any size of collector, the optimum tube length would be 1.5 meters with the number of tubes 0.93 of the collector area (in m^2). If the flowrate was reduced, C' would increase slightly while the number of tubes would decrease and the tube length increase. For $G=8\ lb/ft^2\ hr$ (C'=0.6), the following would be true:

$$N = 0.4380\ A_c\ (in\ m^2) \qquad L = 3.2m = 10.5\ ft$$

Although exchangers are available with tube lengths of this length (see Appendix B-3) the reduction in tube number would reduce this type of heat exchanger's use for smaller collector sizes since the minimum number of tubes available for off the shelf heat exchangers is approximately 30. Thus the minimum collector area below which the heat transfer would be reduced would be 32 m^2 for the first case and 70 m^2 for the second. This would either eliminate many possible applications for home heating or reduce the performance of the double loop system in comparison to one using a higher flowrate.

To eliminate the dependency of U_x on the heat exchanger area and to insure adequate heat transfer from the heat exchanger, not only the optimum heat exchanger area but the number of tubes and tube lengths required should also be calculated. When U_x does not depend on the heat exchanger area, U_x can be calculated as shown in Appendix B-1. With the inside tube heat transfer coefficient fixed for a given tube diameter, U_x is most affected by the heat transfer fluid circulating.

Discussion of the Simple and Complete Procedures for Establishing the Optimum Heat Exchanger

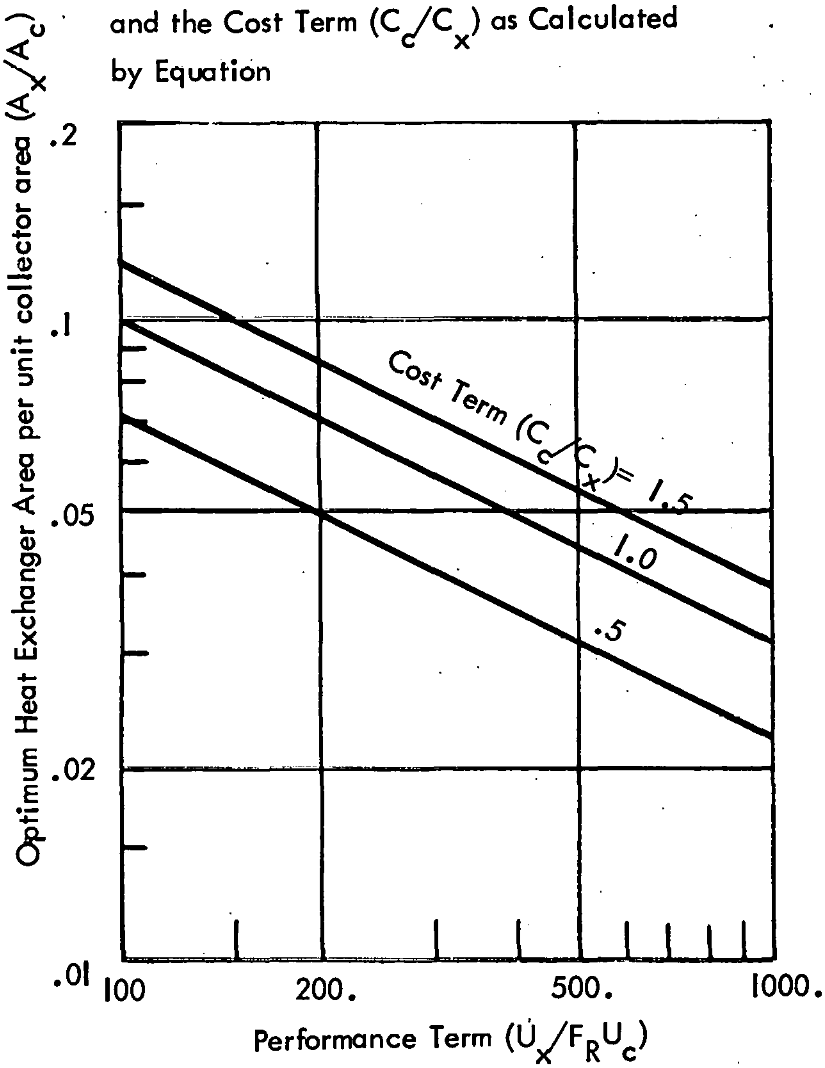
On the shell side. As shown in Table B-2-7, U_x can vary for typical systems from 1000-1200 watts/m^2 ^oC for aqueous fluids and from 600 to 900 watts/m^2 ^oC for nonaqueous fluids.

From the above discussion it has been shown that the optimum heat exchanger area should be determined from Equation (2.1.4.5), taking into consideration the number of tubes and tube lengths required to meet the optimum area. Simpler methods of determining the optimum heat exchanger area were explored, but in all cases, they were felt inferior to Equation (2.1.4.5). One such method would be to say that the amount of heat exchanger area required would be 5 or 8% of the total collector area. Figure 2.1.4.3 explores this possibility. In Figure 2.1.4.3, the ratio A_x/A_c was calculated from Equation (2.1.4.5) for typical variations of $U_x/(F_R U_C)$ and C_c/C_x. It is clear from Figure 2.1.4.3 that to assume that A_x/A_c is a constant would lead to large off optimum results.

In this age of pocket calculators, Equation (2.1.4.5) should present little problem, in determining the optimum heat exchanger area. Collector manufacturers can readily supply the unit collector costs and $F_R U_c$ products for each of their products. Since the overall heat transfer coefficient can be assumed constant for design purposes with values depending on the heat transfer fluid circulating through the collector loop, and unit exchanger cost are available from exchanger manufacturers, the optimum exchanger area can be determined.

Figure 2.1.4.3

Ratio of the Optimum Exchanger to Collector Area Versus the Performance Term (U_x/F_RU_c) and the Cost Term (C_c/C_x) as Calculated by Equation



2.1.5 Determination of Minimum System for which a Heat Exchanger rather than a Traced Tank System should be used

For large collector systems such as used for heating or cooling buildings, the use of a heat exchanger is a more cost effective approach than using a traced tank system. For single dwelling potable hot water systems, a traced tank becomes more viable. The collector sizes, at which a switch in design approach should be made, were determined for the various cost parameters.

The optimum heat exchanger or traced tank performance has been shown to occur when the total cost to heat transferred ratio is a minimum (see deWinter 1955 and Section 2.2). From this condition the optimum heat exchanger area and optimum total coil length can be determined. The collector area (A_{cutoff}) (Collector area at which switch over from a traced tank design to a heat exchanger system should be made) can be found when the total cost to heat transferred ratios on the two competing systems are equal.

Thus: (2.1.5-1)

$$\frac{C_cA_c + C_xA_x + C_p}{F_{ex}'[Q_iF_R \quad A_c - F_RU_cA_c(T_{win}-T_a)]} = \frac{C_cA_c + C_LL}{F_t'[Q_iF_R \quad A_c - F_RU_cA_c(T_{win}-T_a)]}$$

Assuming the heat exchanger and traced tank systems have similar collector performances this reduces to:

$$\frac{C_cA_c + C_xA_x + C_p}{F_{ex}'} = \frac{C_cA_c + C_LL}{F_t'} \qquad (2.1.5\text{-}2)$$

Where C_p = Cost of extra pump and tubing needed for double loop system

A computer program was developed to solve equation (2.1.5-2) for $A_c = A_{cutoff}$ for typical operating conditions. The optimum tube length needed for the traced tank and optimum exchanger area were found for these operating conditions. It was assumed that the traced tank and heat exchanger would have similar long term performance.

Using the relationships for F_{ex}' and F_t' the cutoff collector area (A_{cutoff}) can be found from equation (2.1.5-2). Thus:

$A_{cutoff} = f\ (F_R, U_c, B, GC_{pt}, GC_{pex}, F_tU_t, U_x, C_c, C_p, C_x, C_L)$

For the purposes of this section the following variables were held constant.

$$F_R = 0.8U_c = 5 \text{ watts m}^{-2}\ {}^oC^{-1},\ B = 0.0699 \text{ m} = 2.75 \text{ inches}$$
$$GC_{pt} = 80 \text{ watts m}^{-2}\ {}^oC^{-1},\ GC_{pex} = 80 \text{ watts m}^{-2}\ {}^oC^{-1}$$

Thus for this analysis:

$$A_{cutoff} = f\ (F_tU_t, U_x, C_c, C_p, C_x, C_L)$$

As an example of the relative performance of the heat exchanger versus a traced tank system, Figure 2.1.5-1 shows the total cost to penalty factor ratio versus collector area for several unit collector costs with the other variables fixed at representative values. This Figure was developed from the computer analysis and shows that the traced tank performs better at lower collection areas (the cost to penalty factor ratio is lower for the traced tank than the heat exchanger) while the heat exchanger is better at higher collection areas. The effect of increased collector costs is to reduce the cutoff collector area. For the $\$50/m^2$ case, A_{cutoff} equals 16.8 m^2 while for $C_c = \$150/m^2$, A_{cutoff} is 11.26 m^2. Thus at higher collector costs, the heat exchanger can operate optimally to lower collector areas.

Remember for collector areas above a given A_{cutoff}, a heat exchanger should be used while for lower collector areas a traced tank should be used.

The effects of U_x and F_tU_t on A_{cutoff} is shown in Figure 2.1.5-2 for representative cost parameters. For fixed F_tU_t, Figure 2.1.5-2 shows the effect of increasing U_x is a lower A_{cutoff}. Thus increased heat exchanger performance allows the heat exchanger to be used to lower collection areas more effectively. For fixed U_x and increased F_tU_t (better performance of the traced tank system) the A_{cutoff} is higher. Thus increased traced tank performance allows the traced tank to be used to higher collector areas more effectively.

Figure 2.1.5.1

Cost to Penalty Factor Ratio Versus Collector Area and System Type (i.e. Traced Tank or Double loop Heat Exchanger System) for Typical Design Conditions With Varying Unit Collector Costs

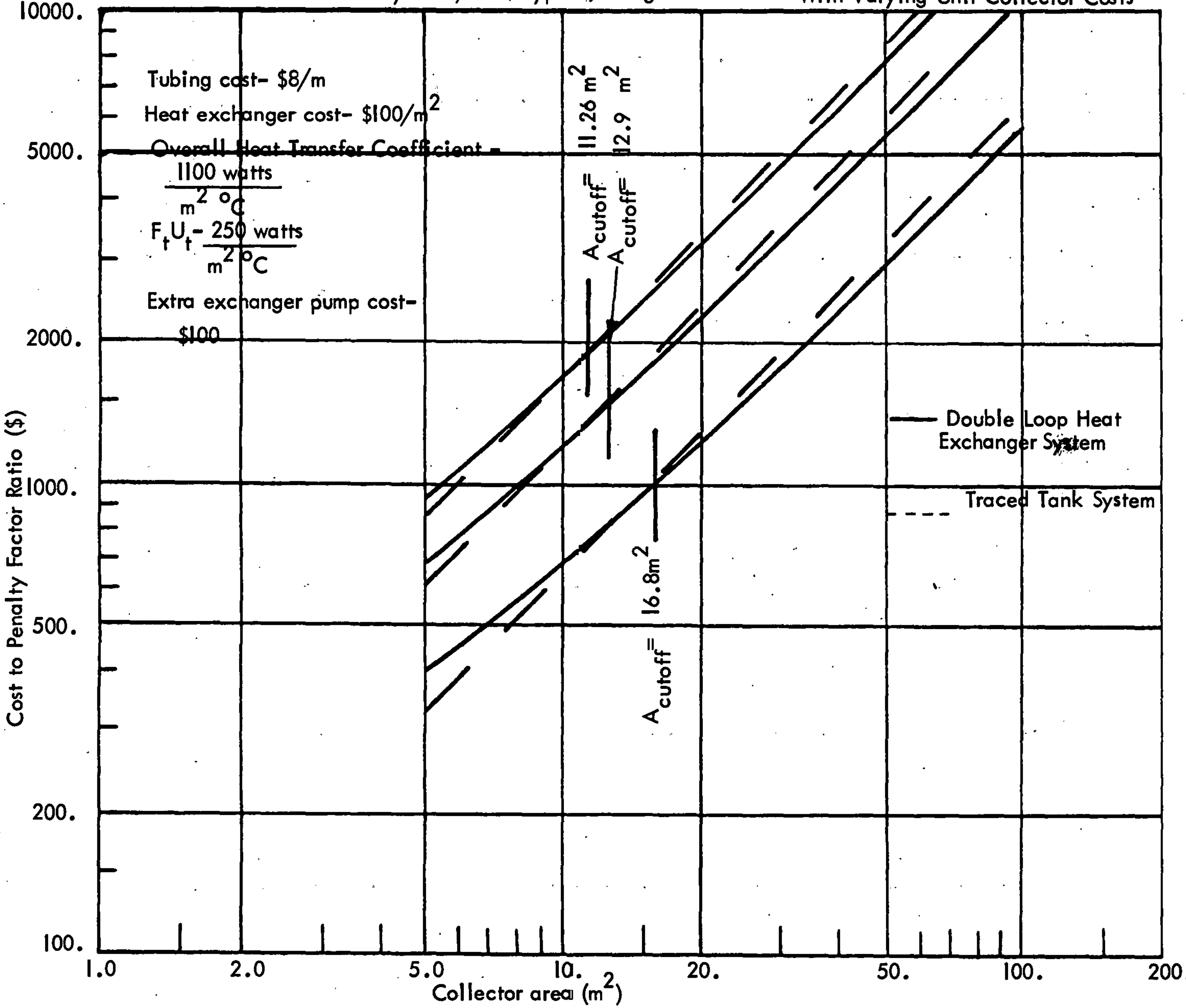
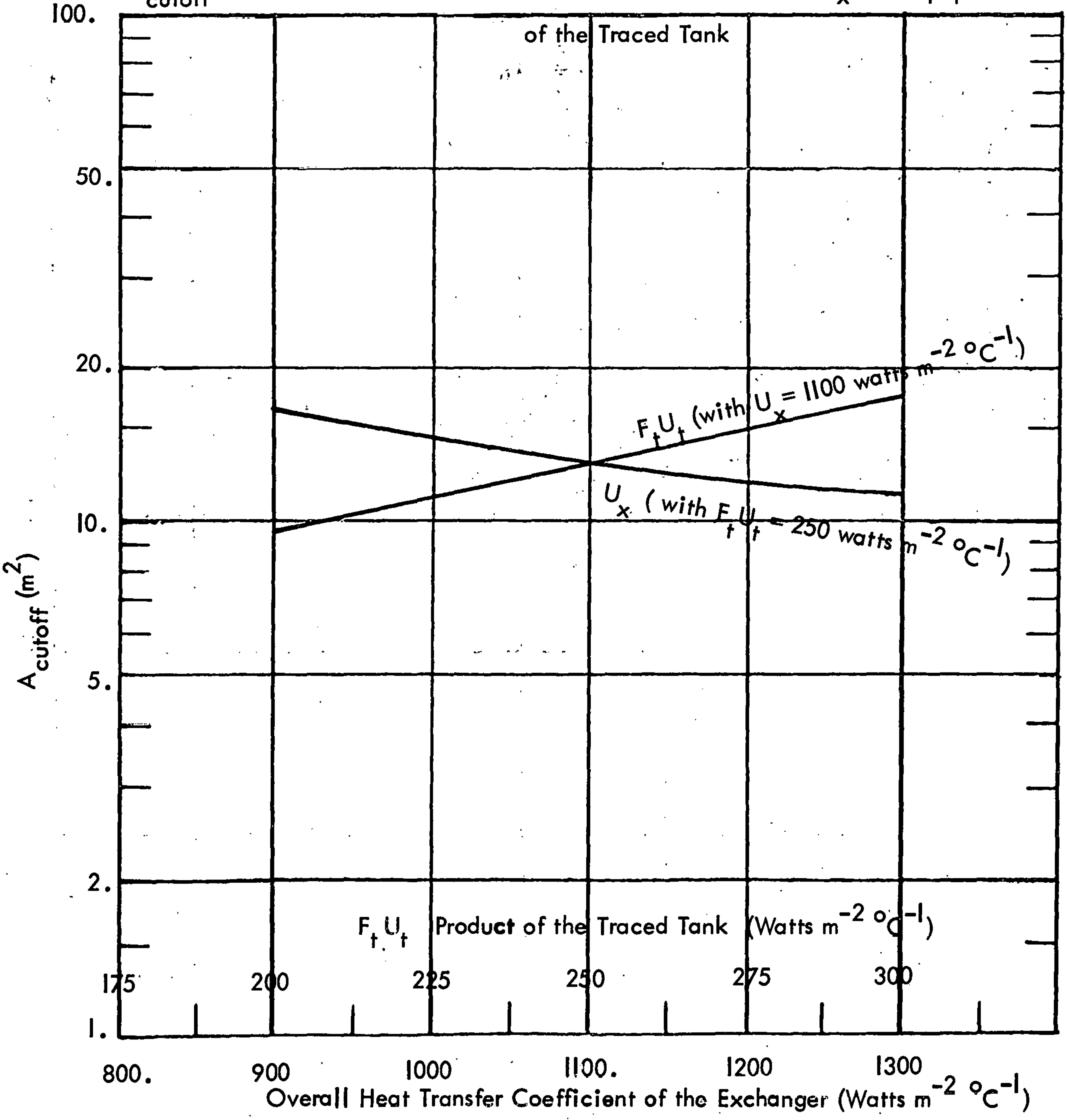


Figure 2.1.5.2

A_{cutoff} Versus Overall Heat Transfer Coefficient of the Ex'er(U_x) and F_tU_t Product of the Traced Tank



The effects of U_x and F_tU_t on A_{cutoff} are not constant for varying cost parameters. Thus 2.1.5-2 is an illustrative example only of the effect of U_x and F_tU_t for this case and should not be used as a correction factor for varying F_tU_t or U_x.

The critical effects of the cost parameters on the collector area at which design switch over occurs is investigated in Figure 2.1.5-3 and 2.1.5-4. Figure 2.1.5-3 shows $A_{cutoff}(m^2)$ versus unit collector costs ($\$/m^2$), unit heat exchanger costs ($\$/m^2$) and unit tubing and solder costs ($/m). The extra pump cost of the double loop heat exchanger system is kept fixed at $100, while the heat transfer parameters (F_tU_t and U_x) are fixed at 250 watts/(m^2 ^oC) and 1100 watts/(m^2 ^oC respectively. The effect of increased unit collector cost on A_{cutoff} is to reduce A_{cutoff} as shown in both Figures 2.1.5-1 and 2.1.5-3 Thus the heat exchanger is more cost effective at lower collector areas when the unit collector cost is high.'

The effect of increased unit heat exchanger costs is to increase A_{cutoff}, as the cost effectiveness of the heat exchanger system decreases. The effect of increased unit tubing and solder costs on A_{cutoff} is to reduce the collection area at which a heat exchanger can be used, since the cost effectiveness of the traced tank system is poorer.

The effect of the cost of the extra pump and tubing required in a double loop heat exchanger system is investigated in Figure 2.1.5-4. This effect is fixed for all other parameters. Thus a correction factor on A_{cutoff} can be determined and is shown in Figure 2.1.5-4.

If no extra pump was required, the reduced heat exchanger system would always be more cost effective than the traced tank system (i.e. A_{cutoff} =0). Since the extra area is required there exist some collector areas for which the traced tank system is most cost effective.

Figure 2.1.5.3

Cutoff Collector Area Versus Unit Collector Cost (C_c), Unit Exchanger Cost (C_x), and Tubing Cost (C_l)

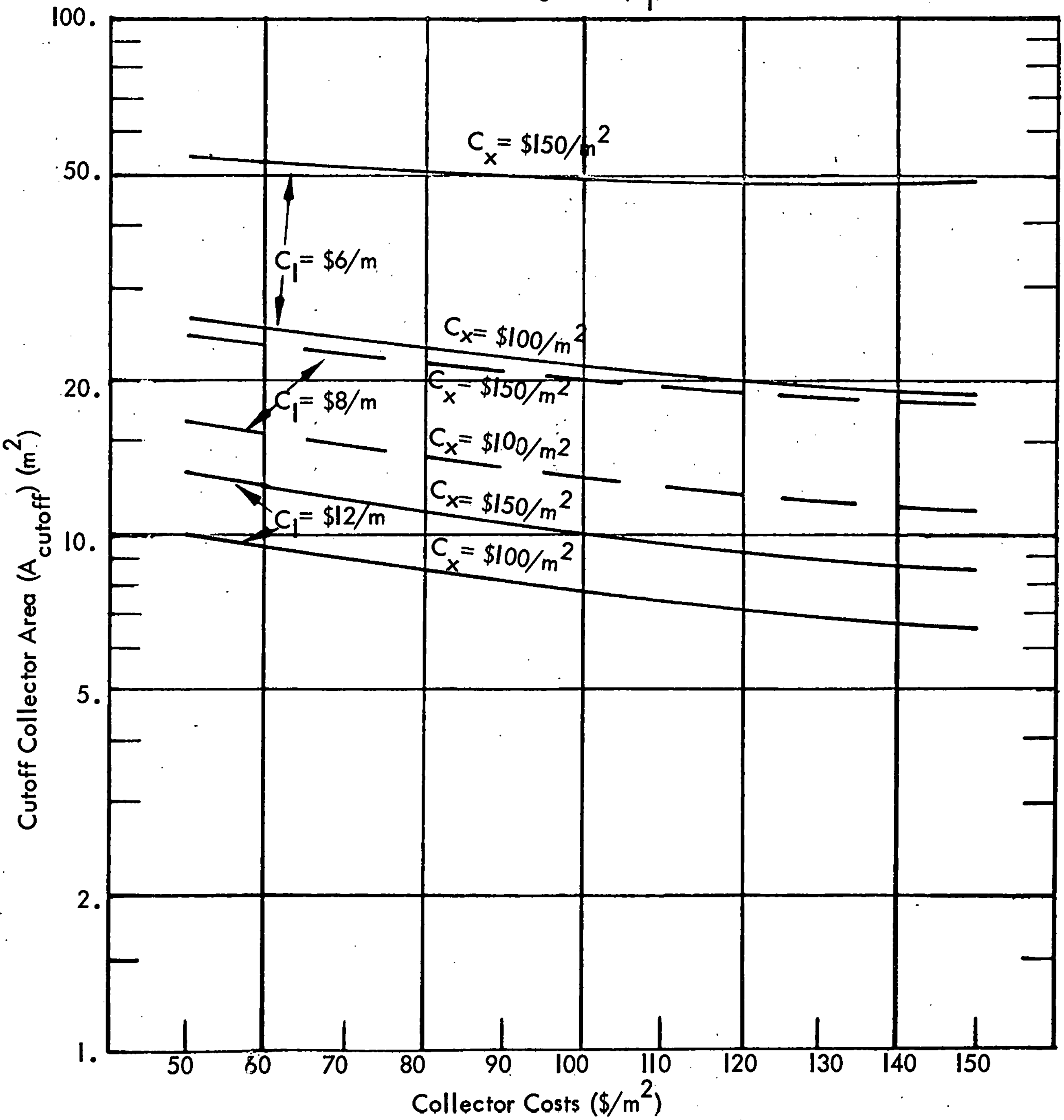
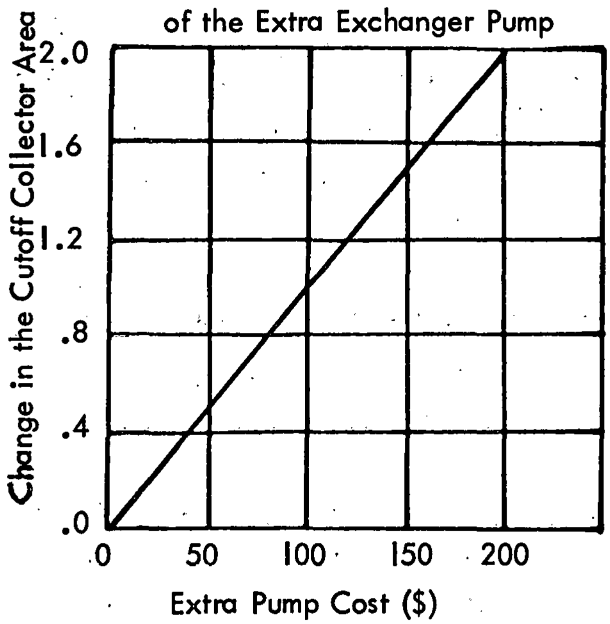


Figure 2.1.5.4

Change of Cutoff Collector Area Versus The Cost of the Extra Exchanger Pump



2-2 Heat Transfer Analysis and Optimization of Traced Tanks or Tanks with Internal Coils

Introduction

For large collector systems, a heat exchanger may be cost effective to transfer heat from the collector fluid to the water circulating in storage. For smaller system, particularly domestic hot water heating systems, a traced tank can become more cost effective than a heat exchanger to transfer heat from the collector fluid to the storage water.

The traced tank unit is composed of a helical coil soldered to the storage tank through which the heat transfer fluid circulates from the collector. The coil is of fixed spacing B and total length L. See Figure 2.2.1 for typical traced tank configurations.

Some of the advantages of the traced tank system in comparison to a heat exchanger system are:

1.) The extra pump and associated piping needed in the heat exchanger unit are eliminated.

2.) For smaller collector systems optimum heat exchanger areas are difficult to design due to the lower operating flow rates.

3.) Because two walls (tube and tank) separate the heat transfer fluid from the storage water, toxic fluids can be used and meet code requirements.

Analysis

The components of the traced tank unit affecting heat transfer are:

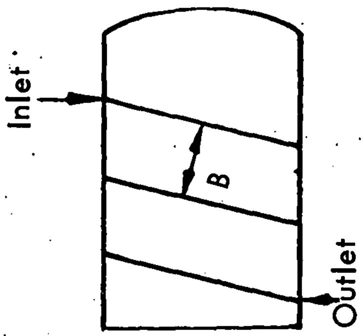
1.) Storage tank

2.) Water within tank

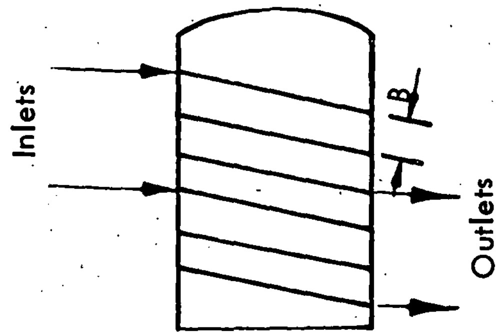
3.) Heat transfer fluid within helical coil

4.) Helical coil bonded to the tank

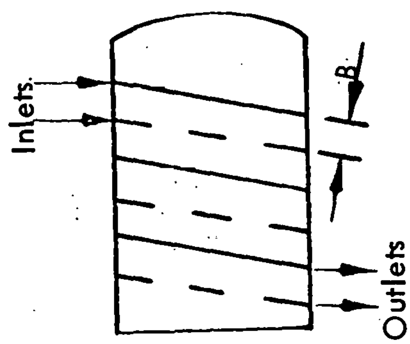
Figure 2.2.1
Possible Traced Tank Designs



Single Helical Coil



Double Helical Coil
Non-Overlapping Design



Double Helical Coil
Overlapping Design

Storage Tank

Typical storage tank configurations and optimum parameter values were determined in Appendix c-1. These results were applied in all subsequent calculations in this section.

Water

The water inside the storage tank was assumed to be nonstratified in this analysis. The water was heated by the tank walls and subjected to a variable hot water load. The rate at which heat is added to the tank water is:

$$Q_t - U_t A_t \; [\, T_w - T_B \,] \tag{2.2.1}$$

Q_t - Rate of heat transfer (watts)

U_t - Natural convection coefficient of water within tank $[\frac{\text{watts}}{m^2 \; {}^oC}]$

A_t - Area of heat transfer (m^2) = BL

T_w - Wall temperature (^oC)

T_B - Bulk temperature of water (^oC)

The natural convection heat transfer coefficient (U_t) for a fluid inside a vertical cylinder according to McAdams (1954) is :

$$U_t = 0.13 \frac{K_F}{L} [\, X \,]^{1/3} \qquad 10^9 < x < 10^{12} \tag{2.2.2}$$

$$\text{where } X = \frac{L^3 \rho_F{}^2 g \beta_F \Delta t}{\mu_F{}^2} \left[\frac{C_p \mu}{K} \right]_F \tag{2.2.3}$$

U_t - in $\frac{BTU}{hr \; ft^2 \; {}^oF}$

K_F - Thermal conductivity of fluid at film temperature $\frac{BTU}{hr \; ft \; {}^oF}$

L - Height of Cylindrical tank (ft)

ρ_F - Density of fluid at film temperature $\frac{lb}{ft^3}$

g - Acceleration due to gravity = 4.18×10^8 ft/hr^2

β_F - Volumetric coefficient of expansion of fluid at film temperature $\frac{1}{{}^oF}$

Δt - Difference in temperature between tank wall and fluid [^{o}F]

$[\frac{C_p\mu}{K}]_F$ - Prandtl number at film temperature of fluid

μ - Viscosity of fluid at film temperature $\frac{lb}{ft\ hr}$

t_F - Film temperature

For water then: $U_t = F\ [\Delta t$, heat transfer properties of water at $t_F]$
Figure 2.2.2 shows U_t as a function of Δt and t_F as determined by McAdams.

Heat Transfer Fluid

The heat transfer fluids studied in this section were the same as those investigated in the heat exchanger system. The fluid has little effect on performance within the helical coil if the flow is turbulent. Since the coil can be one single tube turbulent flow can be obtained for some fluids within allowable pressure drops. Higher viscosity fluids (such as silicone fluids) will have reduced performance because of their lower Reynold's number. See Appendix B-1 for the performance of fluids within the helical coil.

Because for smaller collector systems less heat transfer fluid is required, more expensive fluids can become more cost effective if their other properties are desirable.

The inside tube heat transfer coefficient of the fluid can be determined from Appendix B-1 for a particular application. Because of the properties of the helical coil McAdams (1954) recommends a correction factor be used to heat transfer with a helical coil. This is:

$$h_t' = h_t\ [\ 1 + 3.5\ \frac{D_i}{D_t}\ ]$$

h_t' = Inside tube heat transfer coefficient in helical coil $[\frac{watts}{m^2\ {}^{o}C}]$

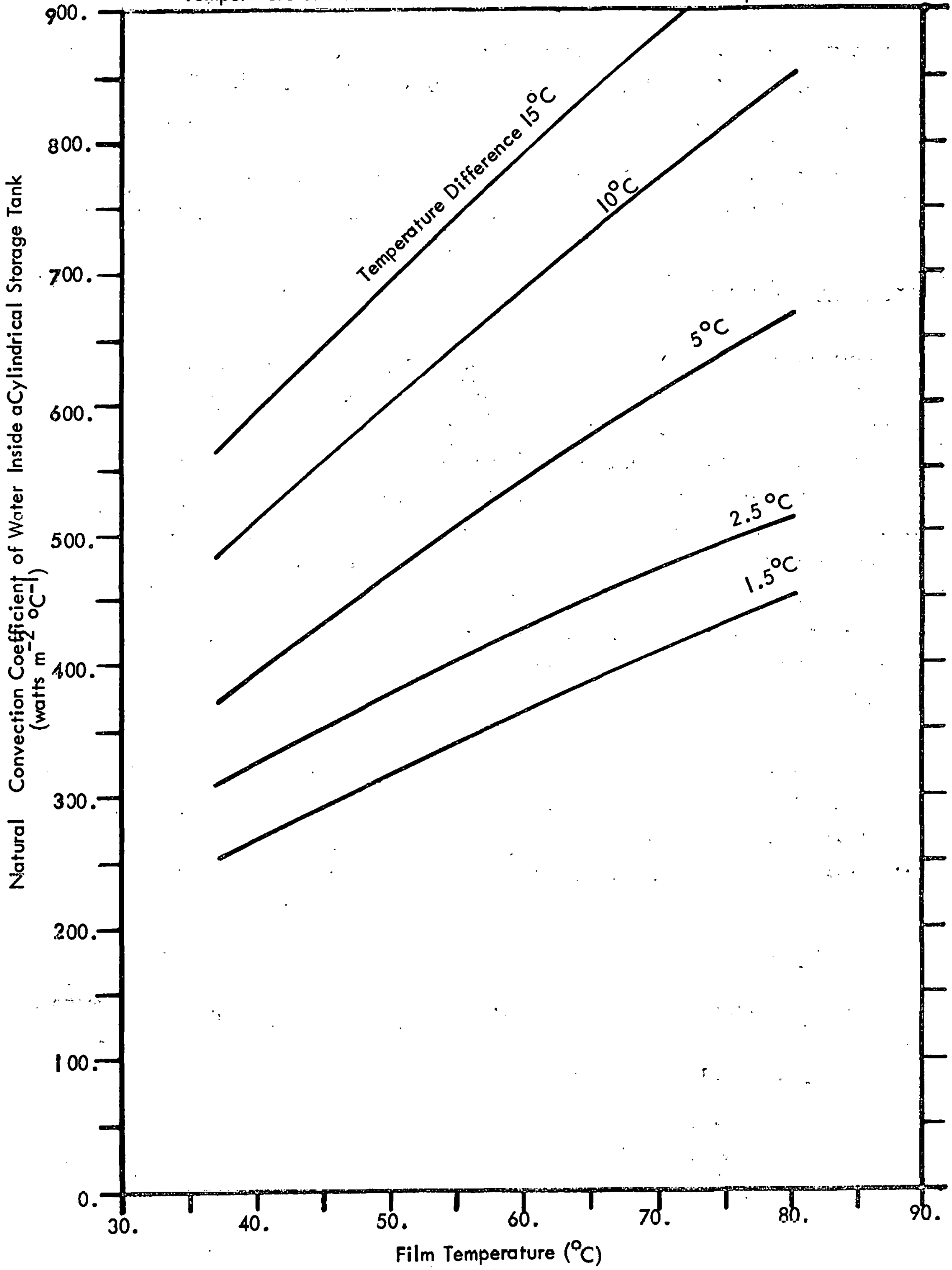
h_t = Inside tube heat transfer coefficient from Figures Appendix B-1

D_i = Tube inner diameter (m)

D_t = Diameter of storage tank (m)

Figure 2.2.2

Natural Convection Coefficient Inside the Storage Tank Versus the Film Temperature and the Difference in Fluid and Tank Wall Temperatures



Helical Coil - Heat Transfer

To optimize the heat transfer and properties of the helical coil, an analysis similar to that developed for the heat exchanger by de Winter (1975) was conducted.

Whillier's (1953) linear expression for a flat plate collector was used in the form:

$$Q_c = Q_i F_R \alpha \tau A_c - F_R U_c A_c [T_{in} - T_a] \qquad (2.2.4)$$

The rate at which heat is transferred through the tank wall is:

$$Q_c = WC_p \epsilon \; [T_{out} - T_B] \qquad (2.2.5)$$

Where WC_p - Capacity rate of fluid within coil

ϵ - Effectiveness of the helical coil to transfer heat.

T_{out} - Temperature of the fluid leaving the collector. Assuming there is no heat loss between the collector and the helical coil, T_{out} is the temperature entering the helical coil.

The effectiveness (ϵ) of the traced tank can also be expressed as:

$$\epsilon = \frac{T_{out} - T_{in}}{T_{out} - T_B} \qquad (2.2.6)$$

also

$$\epsilon = 1 - \exp \left[- \frac{F_t U_t A_t}{WC_p} \right] \qquad (2.2.7)$$

according to Kays and London (1964). This assumes that the water in the tank is essentially isothermal at any time. In Equation (2.2.7), it should be noted that F_t is the efficiency of the traced tank for transfer heat.

Substitution of [2.2.6] and [2.2.5] into [2.2.4] yields:

$$Q_c = \left[\frac{1}{1 + \frac{F_R U_c A_c}{(WC_p)_c} (\frac{1}{\epsilon} - 1)} \right] [Q_i F_R \alpha \tau A_c - F_R U_c A_c (T_B - T_a)] \qquad (2.2.8)$$

This is exactly the same equation as developed (de Winter, 1975) for the heat exchanger problem. The first term in parentheses is the penalty due to the traced tank system:

$$\frac{F_R'}{F_R} = \frac{1}{1 + \frac{F_R U_c A_c}{(WC_p)_c} \left[\frac{1}{\epsilon} - 1 \right]} \qquad (2.2.9)$$

The second term in parentheses is the rate of heat transferred if no traced tank system was used, i.e. the water in the tank circulated to the collector.

With equation (2.2.8) the rate of heat transfer can be determined for various traced tank designs. Appendix E shows the computer program to solve this task.

The conduction problem between the inside tank wall and the fluid in the tubes is analogous to that obtained in a flat plate collector with the tubes bonded below the plate. The heat transfer is given by: (inside water coefficient U_t)(inside tank area A_t)(F_t)(Fluid to water temperature diff.). According to Duffie and Beckman F_t is:

$$F_t = \frac{1}{\frac{BU_t}{\pi D_o h} + \frac{BU_t}{C_{bond}} + \frac{B}{D_o + (B - D_o) F}} \qquad (2.2.10)$$

D_o - Outside diameter of the coil tube (m)

h - Heat transfer coefficient of fluid circulating through the coil $\left[\frac{\text{Watts}}{m^2\ {}^oC} \right]$

C_{bond} - Conductance of tank to coil bond $\approx \frac{4\ T_{wall}\ K_s}{D_o} \left[\frac{\text{Watts}}{m\ {}^oC} \right]$

This value of the bond conductance was determined by de Winter (1978).

T_{wall} - Thickness of tank wall [m]

K_s - Conductivity of steel tank - $\left[\frac{\text{Watts}}{m\ {}^oC} \right]$

F - Fin efficiency of tank wall between the tubes, for heat losses to the water.

Figure 2.2.3 shows the relationship between the parameters of the traced tank.

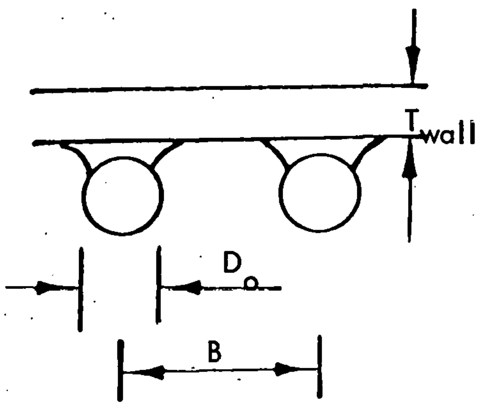


Figure 2.2.3 Specification of Several Coil Parameters

Specification and Optimization of Coil Variables

The coil is completely specified if the coil length (L), coil spacing (B), tube outside diameter (D_o), and tube thickness (T_{Tube}) are known.

Tube Thickness

For a coil of fixed outside tube diameter (D_o), the effect of increased tube wall thickness is to increase the heat transfer (because of larger operating velocities) but also increases the pressure drop. Table 2.2.1 shows the effect of tube wall thickness on the traced tank penalty factor (F_R'/ F_R) and pressure drop for a tube diameter (D_o) of 0.625 in. This case was for a 50% ethylene glycol aqueous solution. Note that the increased wall thickness increased the performance by 0.1% whereas the pressure drop increased 75%. In order to keep all fluids operating within acceptable pressure drops, it was decided to use Type "M" copper tubing.

Table 2.2-1
Effect of Tube Thickness on Traced Tank Performance for an Outside Tube Diameter of 0.625 inches (i.e. 1/2 inch nominal diameter)

	Thickness of copper tubing		
	"M" [0.0275 in]	"L" [0.0425 in]	"K" [0.0575]
F_R'/F_R - Traced tank penalty factor	0.940	0.940	0.941
Δp - Pressure drop [psi/100 ft]	0.328	0.425	0.564

Outside Tube Diameter

For a coil of fixed tube thickness, the effect of decreased tube outer diameter (D_o) is to increase the heat transfer of the coil but also increase the pressure drop. Table 2.2.2 shows the effect of outer tube diameter on the traced tank penalty factor F_R'/F_R and on the pressure drop for Type "M" tubing, also for a 50% ethylene glycol, 50% water mixture. Note that decreasing the outer tube diameter from 0.625 to 0.375 inches increased the traced tank factor by 0.7% whereas the pressure drop increased 16 times. For this study, the helical coil was assumed to be with 0.625 in. outer tube diameter to allow for plausible heat transfer and pressure drop for the various fluids.

Table 2.2.2

Effect of Outer tube Diameter on traced tank performance.

Tube thickness Type "M".

	Actual outer tube diameter (inches)				
	0.375	0.5	0.625	0.75	0.875
F_R'/F_R	0.947	0.944	0.940	0.935	0.930
p [psi/100 ft]	5.0	1.06	0.324	0.124	0.056

Tube Spacing

The optimum tube spacing (B) was determined by extensive computer analysis using a computer program developed for optimization and sensitivity studies by de Winter et al (1967). Physically, the effects of tube spacing can be seen in equations [2.2.1] and [2.2.10]. When the tube spacing is too small (for fixed tube length), not enough heat transfer area will be covered which will lower the rate of heat transfer. If the tube spacing is too large then the efficiency of heat transfer will be reduced.

Table 2.2.3 shows the results of several test runs to determine the optimum tube spacing. All cases show the optimum to be approximately 3 inches. Closer analysis revealed 2.75 inches to be the optimum tube spacing for heat transfer. Figure 2.2.4 depicts a typical case to show the error tolerance. From Table 2.2.2 and Figure 2.2.4 it is apparent that going to larger tube spacing than optimum results in smaller cost effectiveness penalties than in going to too small spacings.

Figure 2.2.4 Total Cost to Heat Transfered Ratio versus Tube Spacing.

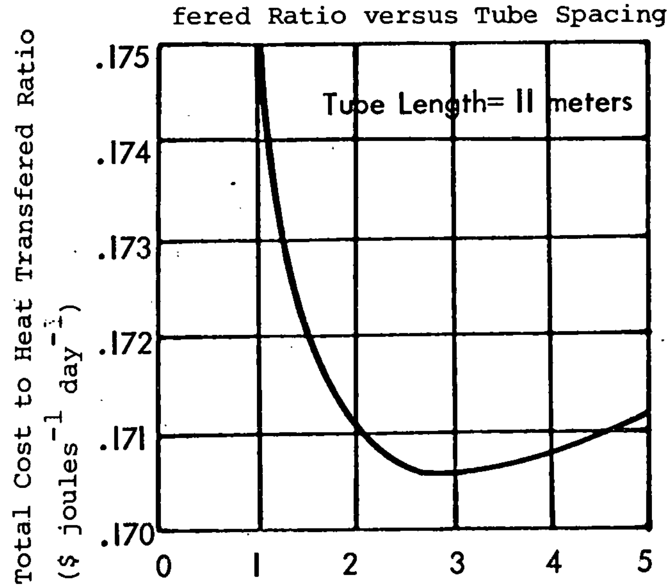


Table 2.2.3

Determination of Optimum Tube Spacing

Coll. Area (m^2)	Fluid	tube length (m)	tube spacing (in)	cost/heat transfered $/joule day
5	water	11	1	0.1748
			2	0.1710
			2.5	0.1706
			2.75	0.1705 ←
			3	0.1706
			3.5	0.1707
			4	0.1708
			4.5	0.1710
			5	0.1711
5	water	9	1	0.1777
			3	0.1710 ←
			5	0.1717
5	Dowtherm J	11	1	0.1910
			3	0.1873
			5	0.1877
5	propylene glycol 40% 60% water	11	1	0.1806
			3	0.1766
			5	0.1771
10	water	9	1	0.1854
			3	0.1720 ←
			5	0.1726

Optimum Total Coil Length

The most important parameter of the helical coil is the total tube length required. If the tube length is too short the heat transfer is poor, whereas for long tube lengths, the cost becomes too high. To optimize the total tube length required, the analysis was similar to that to optimize the heat exchanger area. The optimum total coil length is one for which the total cost/heat transferred ratio is a minimum. For the traced tank this becomes:

$$\frac{d}{dL} \left[\frac{C_T}{Q_c} \right] = 0 \tag{2.2.11}$$

C_t = Total cost of system = $C_c A_c + C_L L = C + C_L L$ (\$)
C - Total cost of collector system (\$)
C_L - Cost per unit length of tubing and solder (\$/m)
Substitution of equation (2.2.8) into (2.2.11) yields:

$$\frac{d}{dL} \left[[C + C_L L] \; [1 + \frac{F_R U_c A_c}{WC_p} (\frac{1}{\epsilon} - 1)] \right] = 0 \tag{2.2.12}$$

Solving this equation leads to:

$$\frac{2}{Y \cdot Z} [\cosh (ZL) - 1] + \frac{1}{Z} [1 - \exp(-ZL)] - L = \frac{C}{C_L} \tag{2.2.13}$$

$Z = F_t U_t B / WC_p$ = traced tank term (m^{-1})

$Y = \frac{A_c F_R U_c}{WC_p}$ = collector term

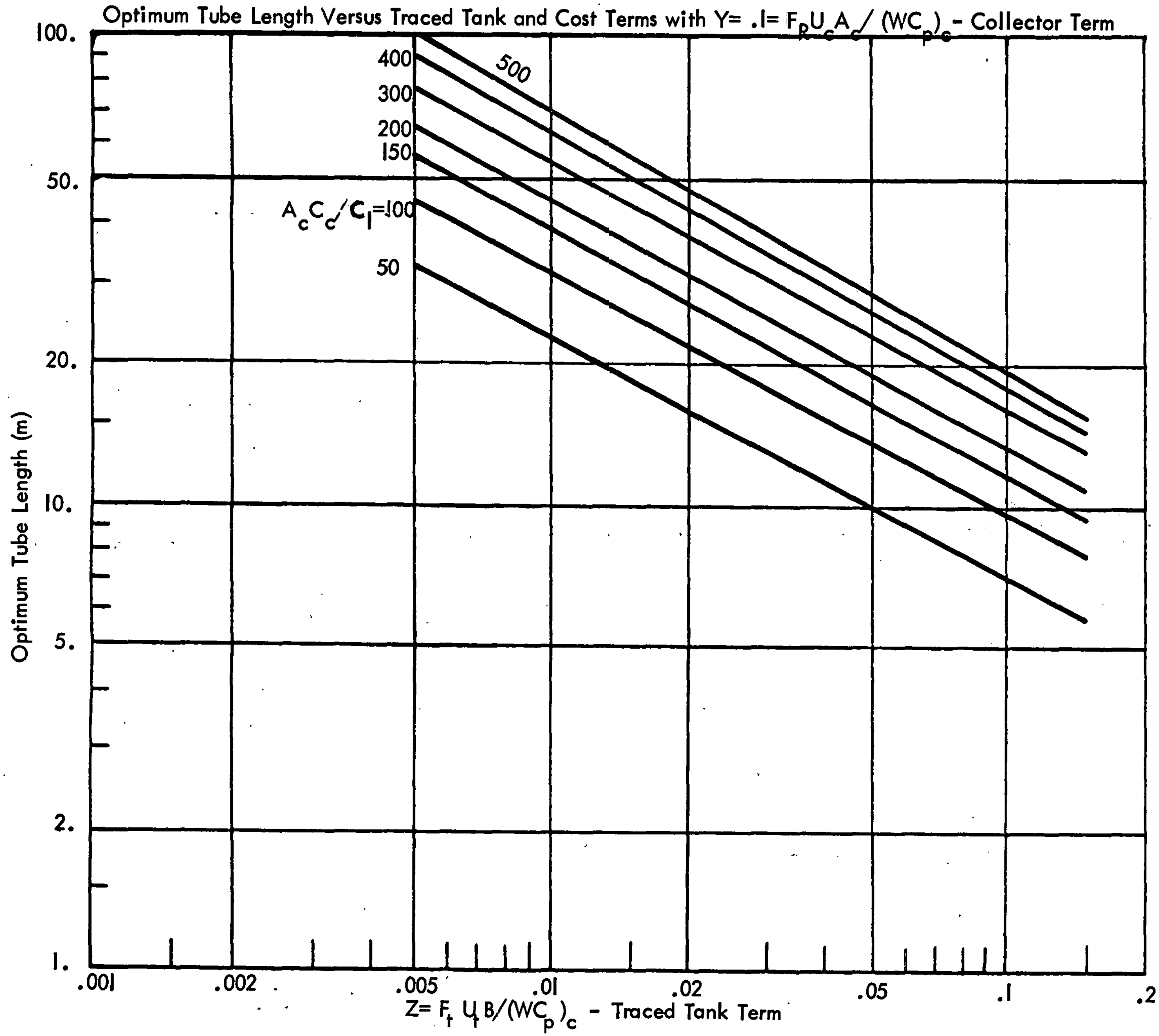
Equation (2.2.13) is not easily solved for the optimum tube length (L), but L can be determined by iteration if Z, Y, and C/C_L are known. By computer analysis the optimum tube length was found for varying values of Z, Y and C/C_L. These results are shown in Figures 2.2.5 through 2.2.7.

In order to use these figures a discussion of each term is necessary. The collector term (Y) is completely determined for a given collector and flowrate.

It reduces to $Y = \frac{F_R U_c}{GC_p}$ since $WC_p = A_c GC_p$

G - Flowrate through collector $(kg/hr\ m^2)$
C_p - Specific heat of heat transfer fluid [watt - hr/(kg °C)]
As long as the flow is turbulent within the traced tank GC_p has little effect on the optimum tube length.

Figure 2.2.5



-2.60-

Figure 2.2.6

Optimum Tube Length Versus Traced Tank and Cost Terms with $Y = .08 = F_R U_c A_c / (WC_p)_c$ - Collector Term

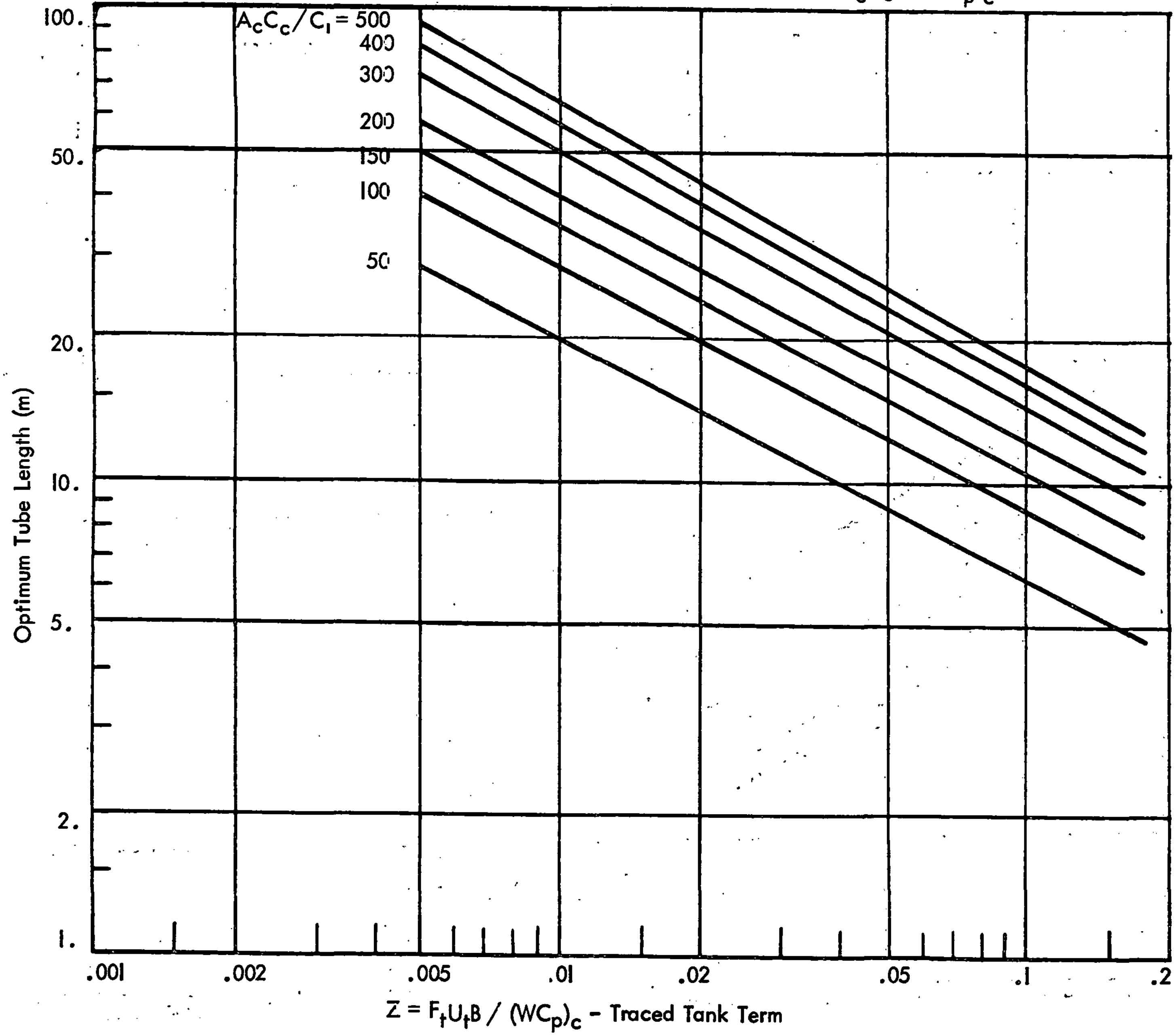
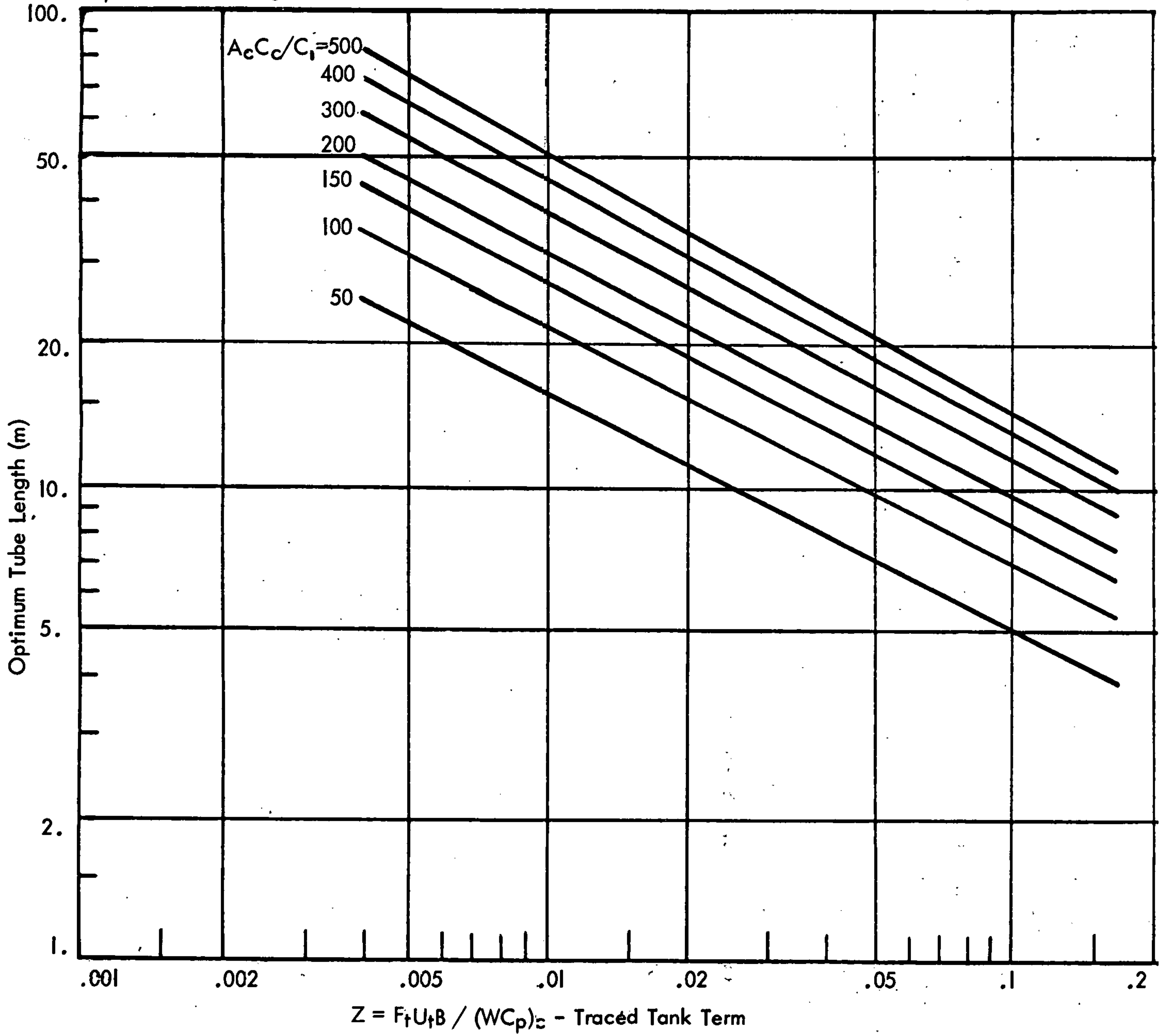


Figure 2.2.7

Optimum Tube Length Versus Traced Tank and Cost Terms with Y = .05 = $F_R U_c A_c / (WC_p)_c$ - Collector Term



Typical design parameters which can be used for the collector term are:

$F_R U_c = 4 \frac{\text{watts}}{m^2 \, {}^oC}$, $GC_p = 80 \frac{\text{watts}}{m^2 \, {}^oC}$, $Y = 0.05$

The cost ratio C/C_L depends on collector size and cost and tube and solder costs. This must be determined using up-to-date cost parameters to insure optimum cost effectioness of the traced tank system.

The traced tank term (Z) is more complicated than the other terms. The natural convection coefficient (U_t) must be determined knowing the temperature of the water within the tank and the difference in temperature between the tank wall. and the water. Also the efficiency of the traced tank system (F_t) depends on many parameters (See equation 2.2.10). Figure 2.2.8 shows the relationship between U_t and F_t for typical design parameters. Figure 2.2.2 and Figure 2.2.8 can be used to determine the $F_t U_t$ product. For conservative design purposes the $F_t U_t$ product can be assumed to be 250 watts/m^2 ^oC. Using the optimum tube spacing B = 2.75 inches and GC_p = 80 watts/m^2 ^oC, the traced tank term (Z) becomes $Z = 0.2183/A_c \; [\frac{1}{m}]$

Thus for design purposes the optimum tube length required for the traced tank can be determined knowing the cost and size of the collector and the cost of tubing and solder per meter.

Figure 2.2.9 shows the cost to heat transferred ratio, traced tank penalty factor and total cost of a traced tank unit versus the tube length as determined by computer analysis. For this case Y = 0.08 and Z = 0.07/m when the cost ratio C/C_L equals 100 m. From Figure 2.2.9 the optimum tube length was 10.5 m. For the same parameters using Figures 2.2.5 and 2.2.6 the optimum tube length was 10.25 m. Figure 2.2.9 shows, for off optimum conditions, longer tube lengths can be tolerated more readily than shorter ones. Thus the extra cost is absorbed by slightly better heat transfer.

Summary 2.2

In this section the heat transfer characteristics of a traced tank were discussed. The optimum characteristics of the helical coil were determined. By computer analysis the following values were chosen as design values.

1.)	Helical coil tube thickness	Type "M" copper tubing
2.)	Helical Coil outer tube diameter	0.625 in. actual O.D. 1/2" nominal O.D.
3.)	Helical coil tube spacing	2.75 in.

Figure 2.2.8

Efficiency of the Traced Tank Versus the Tank Natural Convection Coefficient For a Typical System

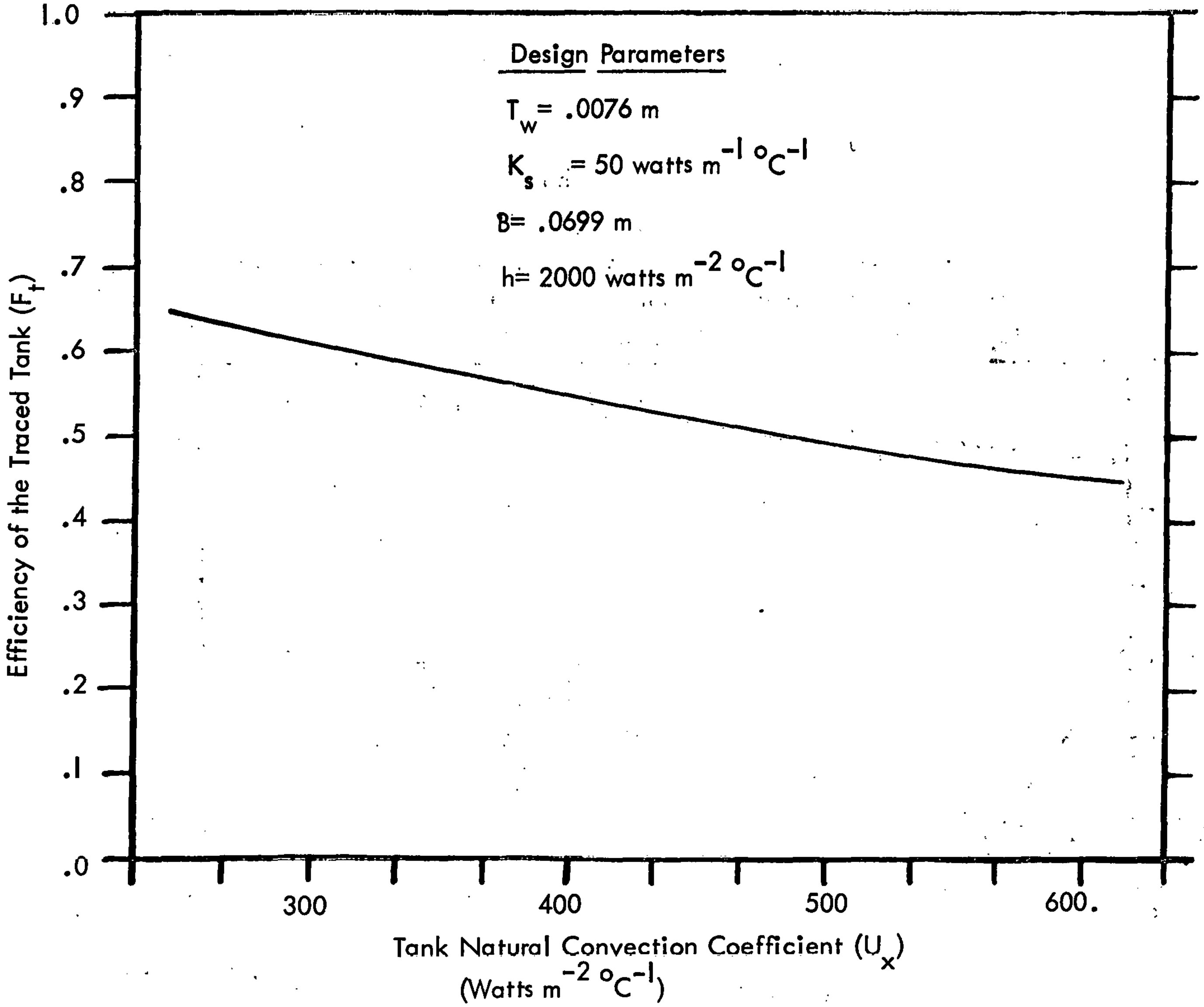
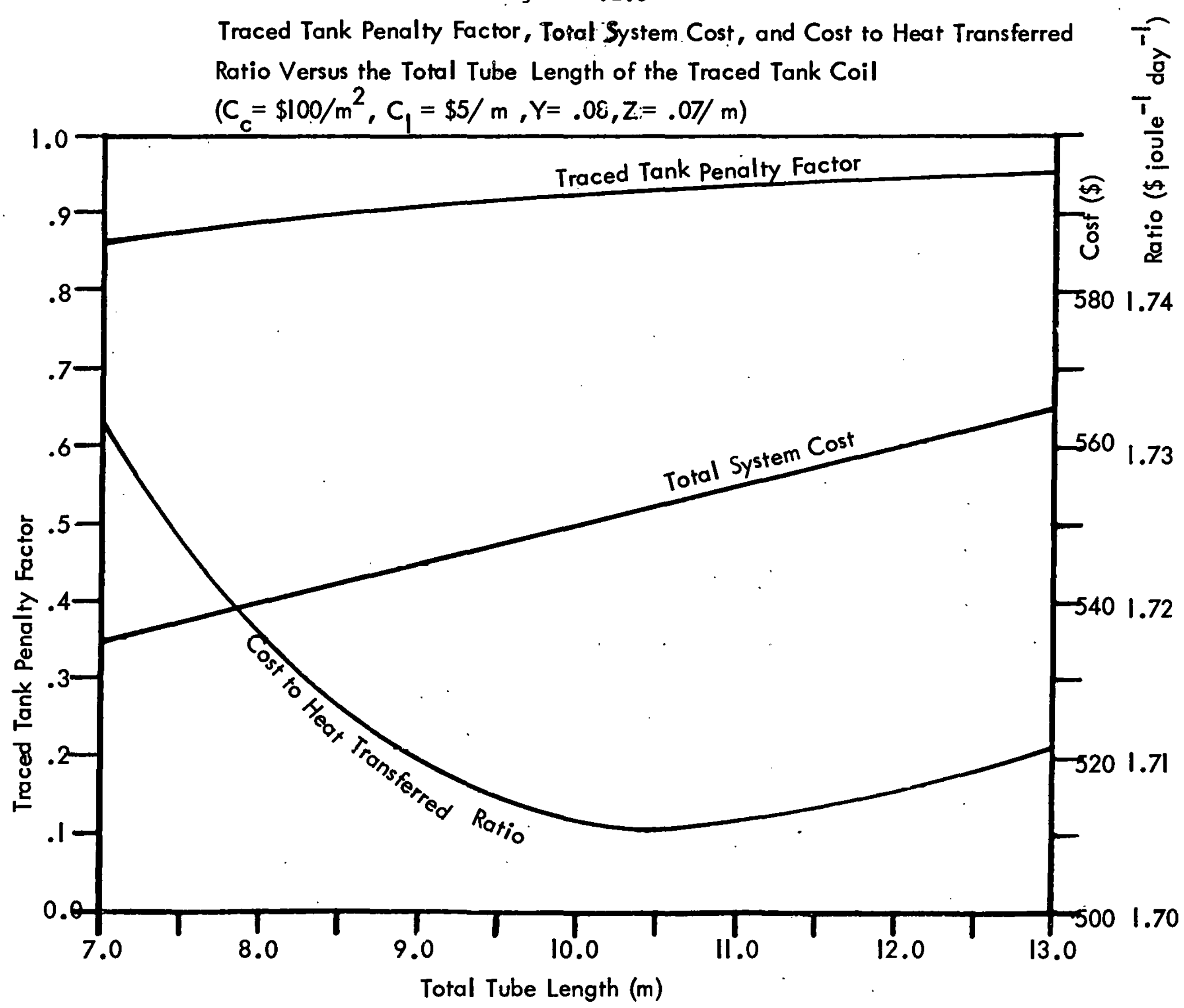


Figure 2.2.9

Traced Tank Penalty Factor, Total System Cost, and Cost to Heat Transferred Ratio Versus the Total Tube Length of the Traced Tank Coil

(C_c = \$100/m^2, C_l = \$5/ m ,Y= .08,Z= .07/ m)



The optimum total coil length (L_{opt}) was determined. It is a function of the total cost of the collector system ($C = C_c A_L$), cost of copper tubing and solder per meter (C_L), collector term $Y = \frac{A_c F_R U_L}{(WC_p)}$, and the traced tank term $Z = \frac{F_t U_t B}{WC_p}$

A closed form solution was not found. Computer runs developed Figures 2.2.5 through 2.2.8 to determine the optimum tube length as a function of C/C_L, Y, and Z.

3 Direct Contact Liquid to Liquid Heat Exchanger

In 1974 G.O.G. Löf proposed the use of a "direct contact" liquid to liquid heat exchanger. This requires an antifreeze fluid which is immiscible in water, and of a different density. This is then circulated in the collector loop, and sprayed into the water storage tank in such a way that a stream of drops flows through the tank to the opposite side, at which it is collected again. The droplet spray can produce an enormous heat transfer area at little cost, and a very efficient heat exchanger can be produced between the two fluids.

Much of the previous work on direct contact heat exchangers was performed in the desalination program of the 1960's, in which the desalination method involving freezing processes generally involved direct contact heat exchange between water (brine) and an immiscible liquid less dense than water. In solar energy applications, since the liquid from the collector would be hot and since a (partially) stratified water tank would be hottest at the top, it seemed best to use a heat transfer fluid more dense than water.

Potential advantages of this direct contact exchanger were:

(a) There is no direct cost associated with the heat transfer surface, as there is with units involving metal walls. Hence a cheaper and more efficient exchanger might be possible.

(b) One circulation pump is eliminated.

(c) If the fluids were indeed immiscible, if it was easy to separate out all the water, and if the collector fluid were non-corrosive, it might be possible to avoid any corrosion problems in the collector.

For several years, research work has been performed on this scheme at CSU under ERDA contract E(11-1)-2867, which has been reported by Buchan et al (1967) and by Ward et al (1977). A number of heat transfer fluids have been examined. Three phthalates were identified which seemed to have good heat transfer properties, were denser than water and immiscible in it, which had low toxicity and low cost. They were diethyl phthalate, butyl phthalate, and dimethyl phthalate. Pilot plant runs have been made which seem to indicate that the method is viable and that the heat transfer can be exceedingly efficient. Followup tests are planned on full size hardware tied into one of the CSU houses.

The program seems to be giving promising results, but it is as yet impossible to perform accurate scaleup projections or cost and performance estimates. In the pilot plant tests, there was an exit temperature difference of essentially zero degrees. This is an encouraging result; it is however also of limited usefulness, for it is impossible to use it to get any quantitative results on the heat transfer rates in the system. It may well be that heat transfer area and performance are inherently so cheap in the direct contact exchanger that it is quite practical to design exchangers of essentially 100% effectiveness; until the rates are understood it is however not possible to say with any certainty what it takes to get high effectiveness. In full size hardware, with a cross sectional area much larger than the 22.28 cm diameter pilot plant, it may be necessary to prevent flow instabilities to get high effectiveness.

An accurate description of the heat transfer is probably quite difficult. In the normal heat exchanger analysis a constant heat transfer coefficient is assumed. This leads to linear differential equations, with the well known exponential solutions. Application of these equations is made somewhat questionable by the fact that the heat transfer coefficients close to the fluid entrance vary considerably, and by the fact that in the exchanger the fluid properties change, and that one has neither the constant heat flux nor the constant wall temperature case for which results are normally available. Despite these factors, the exponential results are quite useful, since they are not that far off. In the direct contact case things are however much worse. There are four distinct stages of drop-to-surroundings heat transfer, with radically different fluid dynamics and heat transfer behavior (see Ward, et al, op. cit. pages 28-29). Any effort to force this behavior into an exponential heat exchanger model may be quite useless. It may be necessary to consider all four regimes independently, and to determine the heat transfer in all of them and the transition points between them, to get any understanding of equipment performance.

It is not only the CSU material which is of limited usefulness in the characterization of heat transfer results. Based on the CSU literature search, it was concluded that: "The heat exchange data that have been reported are not sufficiently complete to permit generalized correlations to be made." As a result, although it can be said that the direct contact heat exchanger is a promising device, it will take more work before it can be compared on a quantitative basis with other devices. Beyond heat transfer tests, it seems desirable to conduct tests specifically designed to verify that the water carryover in the collector loop is indeed small enough to ensure that corrosion problems are indeed eliminated. These corrosion tests can be done on very simple hardware.

It should be noted that the liquid to liquid system, like the draining systems, has one disadvantage: that there is a direct pressure transmission between the storage tank and the collector loop. In both of these systems, the tank may have to be pressurized to get the fluid to the collector without flashing. In a system with a standard heat exchanger, pressurization can be limited to the collector loop.

4 The Use of Drain-Down Systems

One way of avoiding freezing problems in the collector is to drain down the collector whenever there is a danger of frost. The easiest way to do this is probably to drain it whenever no useful energy can be collected and not just when there is danger of freezing. One can save the investment corresponding to one heat exchanger and one associated pump, the heat exchanger penalty is avoided, and one does not need an antifreeze charge - which can be quite expensive. One must however invest sufficient money in equipment and design so that the drain down provision is essentially 100% reliable. It should be noted that it is rarely likely to be cost effective to provide antifreeze protection to all of the water in a storage tank, generally representing 1-2 gallons per square foot of collector.

Normally the water storage tank is at a level below the collector. The customary way to make a draining system involves a design in which air (or some other gas) is let into the collector circuit when the circulation pump stops, so that the collector (and all other parts of the plumbing which might be subject to freezing) can drain back into the system. Some precautions to be used in these designs, and some characteristics of such systems, are discussed below.

It is <u>essential</u> for draining systems to be built and operated with the utmost of care. In the CDA Decade 80 House in Tucson the swimming pool heater featured serpentine loops which could not possibly drain completely. The panels froze, and the rectangular passages bulged as shown in Figure 4.1, requiring replacement of the collectors. In the Walnut Creek demonstration home of PG & E, somebody pressure - tested the collectors with water just before the weekend and forgot to drain the panels, which were frozen and destroyed by the time people came back on Monday. There are dozens of other examples of systems which have failed due to faulty design or operation, causing damages far greater than the amount of money which can be saved by avoiding an antifreeze loop with a heat exchanger.

It is essential in draining systems to connect and slope <u>all</u> lines so that they will drain fully, and to arrange things also so that on filling it is

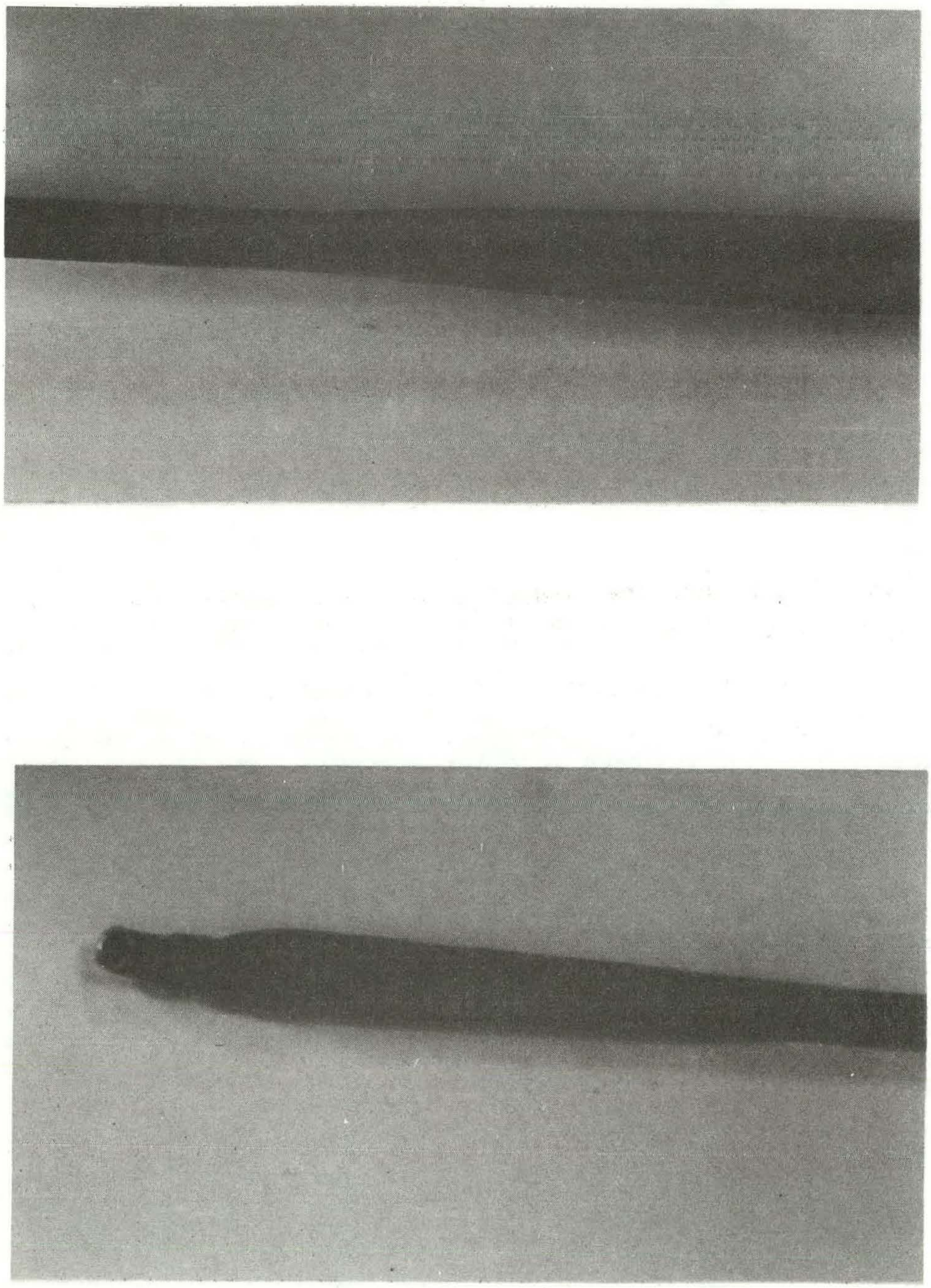


Figure 4.1 Bulged and Distorted Rectangular Copper Tubes After a Freezing Incident with Water Inside.

almost impossible for any part of the collection system to operate dry. Possible arrangements for the plumbing of the individual panels are shown in figures 4.2 and 4.3 below, taken from de Winter (1974).

Figures 4.2 and 4.3 constitute an exercise in heater plumbing logic. The objective is twofold:

(1) The heater should empty completely when the pump stops and the one-way valve lets air into the plumbing. Otherwise freezing damage may result.

(2) The heater should fill completely with water when the pump turns on. All air bubbles should be driven towards the exit, and all tubes should end up carrying water. Otherwise part of the heater could have no water flowing through it, and do no useful work.

Some examples follow. Consider the undesirable arrangements (Figure 4.2). Note that all the heaters are drawn so that the top of the heater is at the top in the figure. Undersirable arrangement B will function, but it will never empty completely. Undersirable arrangements A, C, D, and E will empty, but might end up with some dry tubes during operation.

By contrast, the 3 desirable arrangement (Figure 4.3) all fill completely when the water is turned on, and all are able to empty completely if built properly. This requires that the horizontal tubes in the design, if tilted at all, be tilted in the right direction. For example, in desirable arrangements A and B the manifolds or headers are nominally horizontal. Both inlet and outlet headers should be able to drain completely by being tilted slightly so that the water will flow in the proper direction. In desirable arrangement C the heater tubes are nominally horizontal. They should be tilted slightly so that the water can flow towards the inlet header. The lower part of the outlet header is shown as a dead end which will stay filled with water when the pump is turned off. This dead end section should be as short as possible.

The reader should confirm that between the desirable and undesirable arrangements all possible arrangements have been covered. The recommended arrangements are desirable arrangements A or B.

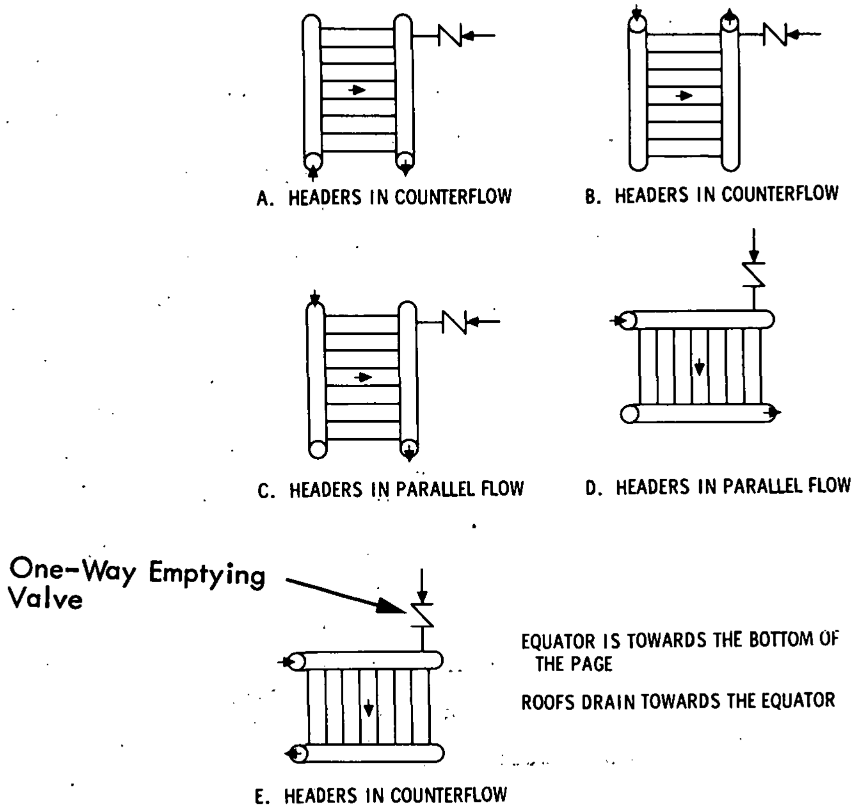


Figure 4.2 Undesirable Heater Plumbing Arrangements (see text)

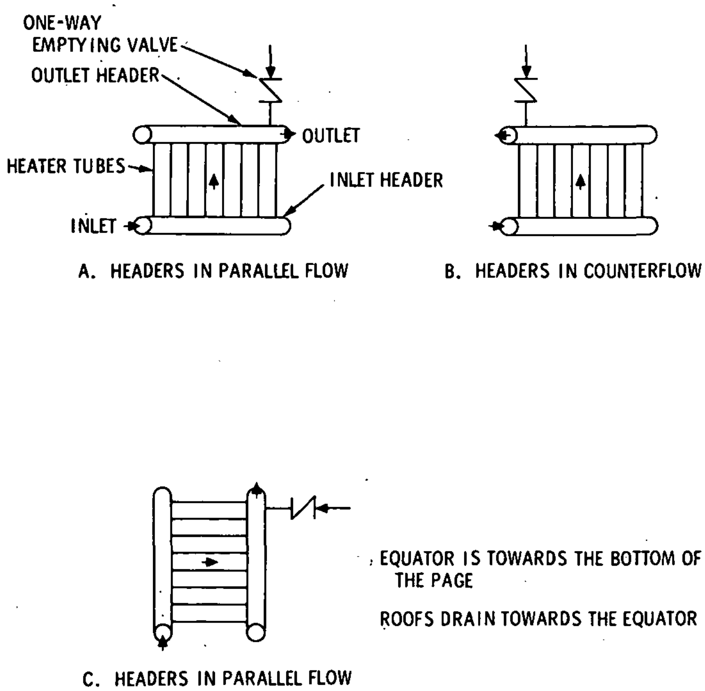


Figure 4.3 Desirable Heater Plumbing Arrangements (see text)

In the figures the one-way (air-fill) valve is shown on the heater manifold. This is not necessary, as it can be located anywhere in the heater circuit, just so long as it is above the water level of the pool. It should not be very much above the water level of the pool. Otherwise it may let in air continuously when the pump is operating.

If your pump is above the pool water level this automatic air filling approach is not recommended, since your pump might lose its prime. In this case it is best to drain your heater manually and by-pass it in cold weather, when you probably are not using the pool anyway. With a self-priming pump there may be no problems.

One advantage of any draining system is the reduction of the thermal capacitance of the collector, and hence of the transient losses. With the collector (and the piping to and from the collector) empty, the solar heat needed to get the system up to operating energy in the morning is less, and this increases the amount of energy which can be collected correspondingly (typically by perhaps around 1%). This effect was already noted by Whillier in his 1953 ScD thesis at M. I. T.

A draining system will generally involve a storage tank with some gas at the top, so that the water in the collector circuit can empty into the tank. This gas need not be air: one can use a charge of inert gas and the system can be hermetic. This makes it possible to use an all steel system with essentially no corrosion problems. It is also possible to have a pressurized system, so as to avoid or minimize vapor flashing in the collector. It should be noted that a tank which is pressurized is likely to be much more expensive than one which is not.

To avoid possible malfunction problems with a vacuum breaking, one way, air filling valve (used to empty the system), one can use a failsafe way to fill the system with air (or some other gas) by simply keeping the return line <u>above</u> the water level in the storage tank. This way, the moment the flow stops air flows in and the system empties. This has been used by Prof. Frank Hooper in his seasonal storage system house in Canada. A diagram of such a system is shown in Figure 4.4.

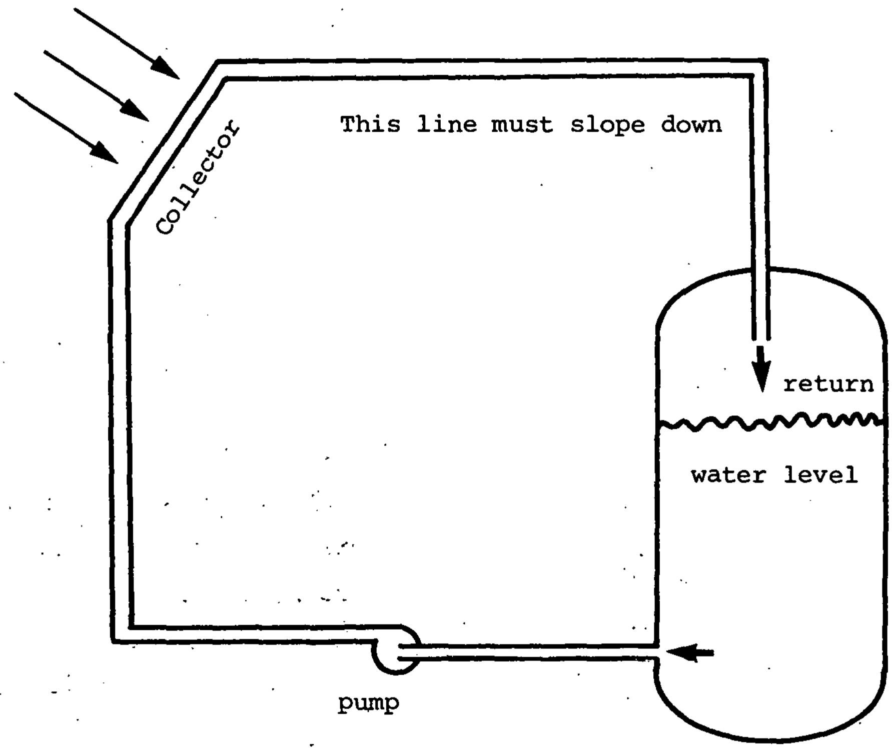


Figure 4.4 Frank Hooper Draining System

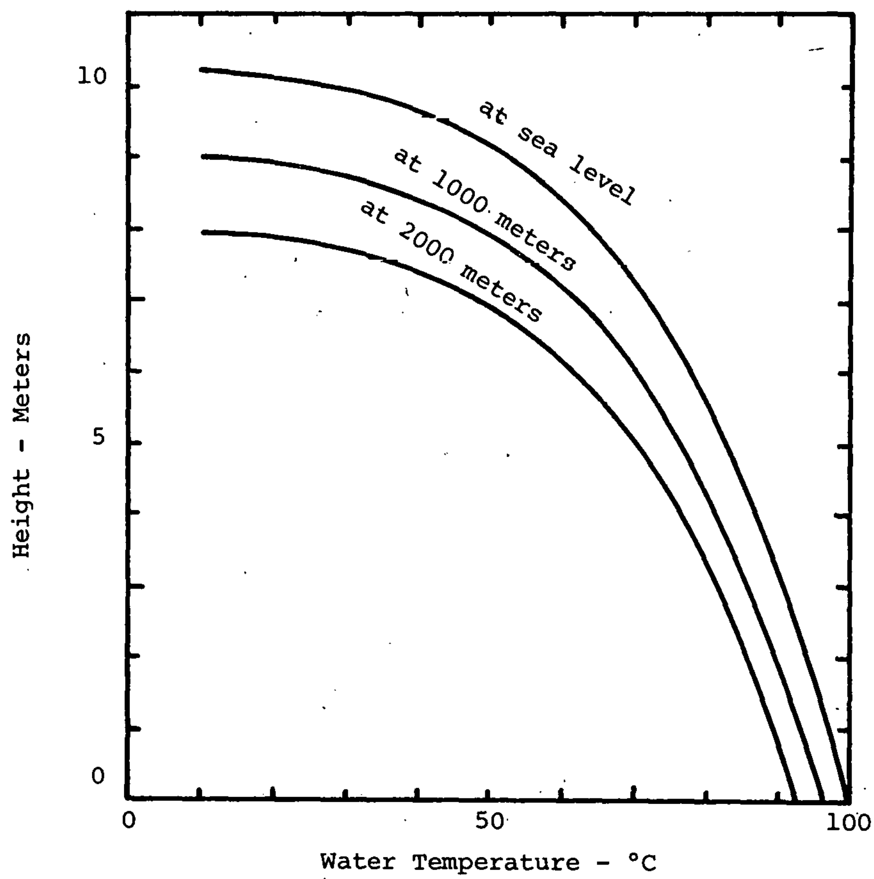


Figure 4.5 Maximum Suction Height Which can be Achieved in Gas-Free Water at Different Temperatures and Altitudes

Pumping requirements can be quite different before and after the descending leg of the syphon fills up with water. It may be desirable to use two pumps: one for normal operation and one to assist in filling the syphon. If the syphon can be made hermetic all that is needed is a small vacuum pump. If the return line is above the tank water surface, or if the vacuum breaker valve can not be kept tightly closed, then a booster pump is needed. It should be noted that the syphon action is limited by the flashing behavior of water.

If vapor flashing is to be avoided in the collector, only a limited use should be expected of the syphon "pull-down" action of the descending leg of the collector circuit. Unless the pressure drop in this leg is made to be high enough, or unless the leg is short enough, or unless the storage tank is pressurized, or unless the water in the collector is cold enough, the water can flash in the collector. Figure 4.5 shows the height of the column of water which can be supported without vapor flashing as a function of water temperature. If the water flashes, the pumping power requirements may be more than expected, one may have a noisy system, one may collect much less energy than expected if the collector runs dry in part, and one may end up with deposits in the collector.

In swimming pool heating systems, hygienic considerations require circulation even on days (or in seasons) when no useful heat can be collected or is desired. In this case a drain down and a bypass system may be desirable. If this is done, then it must be ensured that during operation of the bypass line it is impossible for the collector lines to fill with stagnant water due to slow leaks in the bypass valving. This can only be ensured if there are failsafe draining valves in the collector plumbing which will drain away any water that may leak in. A diagram is shown in Figure 4.6.

In domestic hot water systems a drain down system, with an exchanger between the storage tank and the domestic water, has a number of potential benefits: by controlling water chemistry, it is easy to avoid corrosion or deposits in the collector. The storage tank can be unpressurized, an inert gas fill can be used, also resulting in fewer corrosion or deposition problems. The domestic hot water is then a completely separate circuit, connected only with some heat exchanger to the storage tank. One idiosyncracy concerns the behavior of this heat

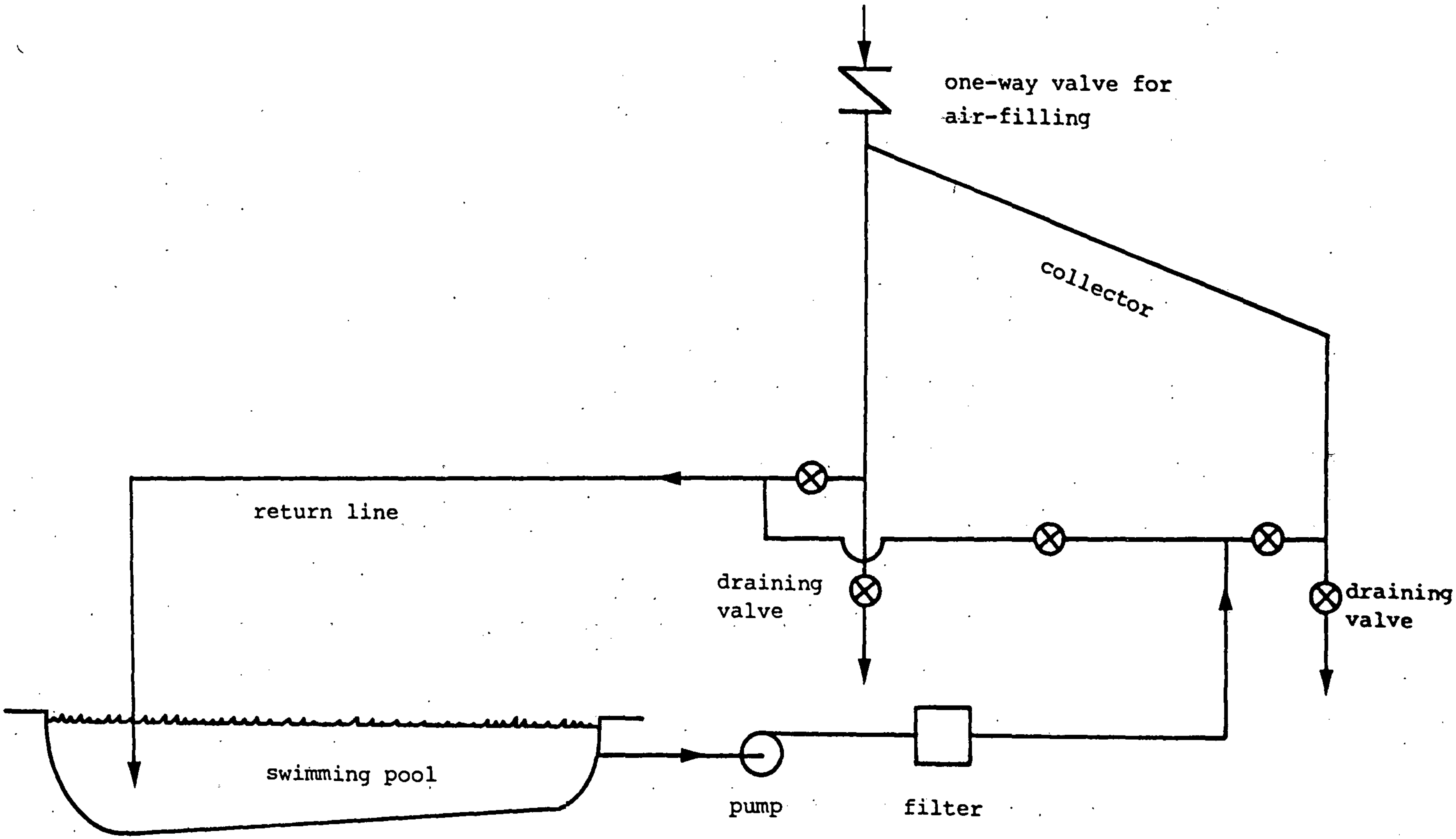


Figure 4.6 Swimming Pool Circuit with Failsafe Collector Circuit Draining Valves for By-Pass Operation

exchanger. When hot water is just turned on (after no hot water has been used for a while) the hot water is at maximum tank temperature. After a while of operation, a temperature difference becomes established in the heat exchanger. The hot first slug of water may be useful for preheating the cold plumbing.

In many of the plastic swimming pool heaters on the market, there is little or no concern with freezing. There are a number of reasons which may be responsible for the fact that this seems to work. On freezing, water does not bond well to most plastics (witness plastic ice trays) so that high stresses produced by water freezing can be relieved by sliding. Most plastics can stretch significant amounts. Finally the low thermal conductivity of plastics makes it likely that freezing processes are pretty slow, so that volume expansion can be relieved over a long period of time, before the passages are closed off with solid ice. Whatever the reason(s), freezing does not seem to be a frequent cause of plastic panel failure. Nature has provided others.

In an unpressurized system, one can produce drain down provisions by using only passive valves, such as one-way valves. In a pressurized system, such as one might have in a domestic hot water system in which the storage tank is at line pressure, multiple solenoid valves are needed: several to isolate the collector circuit, and one or more to drain it. (Note that this is not the case in a hermetic system with a pressurized gas charge in which the water being drained does not leave the system). It is likely to be much more expensive to achieve a given level of reliability in a system in which a number of solenoids must open and close reliably, than in a system in which things happen more or less automatically. It should be noted that the one way valves need not open fully nor close fully for the drain down provision to work properly. In a pressurized system, slight leakage in solenoid valves could be disastrous.

It is premature at this point to establish quality requirements for the valves and other components needed to make draindown systems reliable. Flow passage diameters below about 3/8 inch should probably not be used anywhere in the plumbing or collector, since capillary forces may hold the water in and prevent proper drainage. It is unlikely that in the forseable future there will be sufficient understanding of the requirements of the valves to be able to specify them properly. It is probable that the only reliable specifications which can imposed at present are those imposed on vacuum breakers and backflow preventers by the American Society of Sanitary Engineers (see ASSE 1970-ASSE 1974).

5 Recommendations for Further Study

Six areas of further concern are discussed below.

5.1 Further Analysis and Investigation of Heat Transfer Fluid Properties

Most of the fluid properties used in the discussion of the heat transfer fluids were those obtained from the fluid manufacturers. In most cases, this was sufficient for a proper comparison of the different fluid types. For some fluids, inadequate information was sometimes supplied in the product literature. For example, important thermophysical properties were sometimes listed at only one or two temperatures. Complete data should be obtained as it becomes available for all fluids to be considered for solar energy applications for the following parameters:

(1) Viscosity
(2) Specific heat
(3) thermal conductivity
(4) density
(5) coefficient of volumetric expansion
(6) vapor pressure

Also, for some parameters (particularly flammability and localized corrosion), independent analysis and investigation of the fluids' properties when operating within typical collector systems should be conducted. Further information on code requirements should be obtained for potentially toxic or flammable fluids as they become available. Fluids which do not meet these safety requirements should be discarded as potential fluids for solar energy applications.

It seems desirable to arrange for a continuing effort to collect, measure and determine, and update the data on the properties of heat transfer fluids. Perhaps this could be done at one of the National Laboratories. Perhaps it could be a contracted effort at a thermophysical property measurement laboratory, or at an independent contractor.

5.2 Verification of the Optimum Capacity Rate Ratio Being Less Than One

As shown in Section 2.1.1, a double loop system operates more efficiently when the capacity rate of the collector loop is less than that of the storage loop. Although this effect is small for typical systems, it needs to be investigated in an actual double loop system to develop a further understanding of the effect of the capacity rate ratio on the optimum performance. The penalty imposed by off-optimum capacity rate ratios could then be verified.

5.3 Further Investigation of the Viability of The Traced Tank System

The traced tank system can be more cost effective than the double loop heat exchanger system for smaller collector areas. This study has investigated the cost effectiveness of the traced tank system and developed design parameters and a method to determine the optimum tube length for the helical coil. The traced tank system allows the use of toxic fluids since two walls separate the two fluids. Also, the traced tank system allows easy conversion of existing hot water tanks, allowing reduced costs when retrofitting a hot water system to solar use. For a new system, non-toxic fluids could be used, circulating within a coil immersed inside the potable water storage tank. Further investigations should be conducted to compare these two systems for use in hot water heating systems. It seems desirable to run detailed tests on some geometries to test out the accuracy of the natural convection correlations and predictions.

5.4 Investigation of Plate and Fin Heat Exchangers for Use With Toxic Fluids

Altas Corporation is already continuing its study on the viability of toxic fluids in double loop heat exchanger systems. In particular, plate and fin heat exchangers are being considered for the double loop system which would allow the use of toxic fluids and still meet code requirements. Using plate and fin heat exchanger manufacturing techniques, it is easy to make a "double wall" exchanger, in which there is a vented layer between water and antifreeze passages. Such exchangers have already been built, and are called "buffered" exchangers. In the heat exchange between liquid metals and water, as well as in the heat exchange between lubricating oil and jet fuel, it is essential to prevent mixing of the streams in case of a leak. For these purposes these buffered exchangers were developed. Possible manufacturers of plate and fin exchangers have been contacted to develop prototype models and preliminary performance and cost estimates. They can be used to eliminate the risk of using a toxic fluid near a potable water supply, at a cost and performance penalty probably much lower than that for other double wall heat exchangers. A proposal is being prepared on a development effort for such an exchanger.

5.5 Investigation of Direct Contact Liquid to Liquid Heat Exchangers

An effort should be made to develop a basic understanding of the heat transfer mechanisms in direct contact heat exchangers. At present it is only known there is a high rate of heat transfer, there is no knowledge as to how high it is. Up to now at CSU all the work has involved fluids denser than water. In view of the high heat transfer rates and the well stirred tank, a fluid less dense than water can be used just as well. This should be considered in the further work.

5.6 Heat Exchanger Workshop

When the first paper on heat exchanger penalties and optimization (de Winter, 1975) was first distributed in 1974, there were probably only a few dozen heat exchangers installed, many of which were grossly undersized or oversized. Since then there has been an enormous expansion in the use of heat exchangers, and many people have had the occasion to choose optimum models, to choose heat transfer fluids, or to adapt the equations or optimization criteria. The present study is the first formal attempt to explore questions on heat exchanger in solar energy systems. There is however an enormous amount of other work going on, and it seems most desirable to arrange a workshop in this area so workers can get together and swap notes. There does not seem to be any specific group in the country which is doing a large share of the work. It may well be desirable to have the American Section of the International Solar Energy Society organize the workshop.

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Appendix A Heat Exchanger Penalties in Double-Loop Solar Water Heating Systems

On the following three pages is given in full the paper in Solar Energy which originally led to the study described in the present report. In reading the paper, several things should be kept in mind:

(1) While the matched flowrate case is interestingly simple, it is not the optimum. This is discussed further in section 2.1.1.

(2) In Equation (7), Ac should be changed to Ax.

HEAT EXCHANGER PENALTIES IN DOUBLE-LOOP SOLAR WATER HEATING SYSTEMS

FRANCIS DE WINTER
Altas Corporation, 2060 Walsh Avenue, Santa Clara, CA 95050, U.S.A.

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Abstract—In many solar water heating systems, it may prove desirable to use a double-loop system with a heat exchanger between the flat-plate collector and the water storage tank. This approach, using a second fluid which does not freeze in service and which does not lead to corrosion of metals, may be the most convenient way to avoid freezing or corrosion problems in the collector. Because of the heat exchanger, the collector is, however, forced to operate at a higher temperature with a corresponding performance penalty.

A heat exchanger factor has been developed, which makes it possible to determine the collection performance penalty in a straightforward manner. When the heat exchanger is of the counterflow type and is operated so that the mass flowrate-specific heat products of the two streams are equal, the expression becomes very simple, and lends itself to direct optimization of heat exchanger size. Several sample optimization calculations are shown.

Solar Energy, Vol. 17, pp. 335–337. Pergamon Press 1975. Printed in Great Britain

INTRODUCTION

In solar water heating systems (or, more generally, in systems collecting solar energy to heat water in a heat-storage tank) there are several factors which make it desirable to consider a double-loop system, in which a fluid other than water is circulated through the flat-plate solar energy collector, and then heats the water in a heat exchanger.

(a) If water is used in the collector, it might freeze on cold nights and damage the collector.†

(b) If water is used in the collector, then the collector must be made of corrosion-proof materials.

There are several penalties to be paid in a two-loop system. Antifreeze fluids or other heat transfer liquids are all relatively expensive. Most are combustible to some degree, and some are toxic. Most have a higher viscosity than water, and a lower thermal conductivity and specific heat. Finally, the use of a heat exchanger between the storage tank and the collector raises the collection temperature, hence lowering the collection efficiency. This effect is analysed below.

ANALYSIS

A simple linear expression was proposed for a flat-plate collector by Whillier[1] in the form

$$Q_c = Q_i F_R \alpha \tau A_c - F_R U_c A_c (T_{in} - T_a). \quad (1)$$

In this equation

- Q_c is the rate of heat collection in W
- Q_i is the incident solar heat flux in W/m^2
- F_R is a dimensionless heat removal efficiency factor—a function of collector design and fluid flowrate and properties
- α is the solar absorptivity of the collection plate
- τ is the transmissivity of the glazing system
- A_c is the area of the collector in m^2
- U_c is the heat loss coefficient of the collector in W/m^{2}C
- T_{in} is the fluid inlet temperature in C
- T_a is the ambient temperature in C

The linear equation is an approximation, in part because U_c is a function of T_{in} and T_a (F_R is also a function of U_c). U_c increases with increasing temperatures, so that at high temperatures collector efficiencies are poorer than eqn (1) might suggest. Since one can always linearize an equation over small regions (see below), an equation such as (1) is adequate for present purposes.

The exit temperature from the collector can be obtained from a simple heat balance:

$$T_{out} = T_{in} + \frac{Q_c}{WC_p}. \quad (2)$$

In this equation

- T_{out} is the fluid exit temperature in C
- W is the fluid flowrate in kg/s
- C_p is the fluid specific heat in J/kgC

Combining eqns (1) and (2)

$$Q_c = \frac{1}{\left[1 - \frac{F_R U_c A_c}{WC_p}\right]}[Q_i F_R \alpha \tau A_c - F_R U_c A_c (T_{out} - T_a)]. \quad (3)$$

The outlet temperature of the collector is the inlet temperature of the heat exchanger.‡ If in the heat exchanger, the WC_p product of water is equal to or greater than that of the collector-loop fluid, then the performance of the heat exchanger can be described by the equation [2]:

$$Q_c = WC_p \epsilon [T_{out} - T_{win}]. \quad (4)$$

†Freezing can also be avoided by emptying the collector at night or by supplying moderate amounts of heat from storage.

‡Assuming zero heat loss in the piping.

Note that the heat transfer in the heat exchanger equals the heat collection Q_c.† Other terms in this equation are:

ϵ is the heat exchanger effectiveness
T_{win} is the temperature at which water flows into the heat exchanger, in C

Now we can solve for T_{out} in eqn (4) and substitute in eqn (3), with the result

$$Q_c = \frac{1}{\left[1+\frac{F_R U_c A_c}{WC_p}\left(\frac{1}{\epsilon}-1\right)\right]}[Q_i F_R \alpha \tau A_c - F_R U_c A_c (T_{win} - T_a)]. \quad (5)$$

The last term in brackets is the collection rate if water were used directly in the collector, with no heat exchanger.‡ The first term is a penalty factor imposed by the exchanger in the double loop system:

$$\text{Heat exchanger factor} = \frac{1}{\left[1+\frac{F_R U_c A_c}{WC_p}\left(\frac{1}{\epsilon}-1\right)\right]}. \quad (6)$$

The best heat exchanger to use is a counterflow heat exchanger, since it yields the highest effectiveness values. When in a counterflow heat exchanger, the WC_p products of the streams are *not* equal, the effectiveness expression has exponential terms, and eqn (6) cannot be simplified any further. There are however, advantages in having matched WC_p products for the two fluids. Using matched WC_p products, one gets more heat transfer (or lower temperature differences) in the same heat exchanger, and the heat exchanger effectiveness becomes simply[2]:

$$\epsilon = \frac{1}{1+\frac{WC_p}{(U_x A_c)}}. \quad (7)$$

In this equation $U_x A_x$ is the UA product of the heat exchanger in W/C.

Combining eqns (6) and (7), the exchanger factor for a two-loop system of matched WC_p products becomes simply:

$$\text{Heat exchanger factor} = \frac{1}{1+\frac{F_R(U_c A_c)}{(U_x A_x)}}. \quad (8)$$

Equation (8) (or eqn (6) combined with more complex exchanger effectiveness equations) can be used for the purposes of optimizing heat exchanger investment.

As one further step in optimization, one can consider the case of a collector with a constant cost per unit area C_c, and a family of heat exchangers having a constant cost per unit area C_x. The optimum heat exchanger is one for which the parameter: (total cost)/(heat exchanger factor) is minimized. Writing

$$\frac{d}{dA_x}\left[(A_c C_c + A_x C_x)\left(1+F_R\frac{U_c A_c}{U_x A_x}\right)\right] = 0, \quad (9)$$

we get

$$\frac{A_x}{A_c} = \sqrt{\left(\frac{F_R U_c C_c}{U_x C_x}\right)}. \quad (10)$$

Two sample calculations are shown below.

Sample calculation with constant cost factors. Consider a flat-plate collector system with the following parameters:

$A_c = 100\ \text{m}^2$ $\quad F_R = 0{\cdot}8$
$U_c = 4\ \text{W/m}^2\text{C}$ $\quad U_x = 1320\ \text{W/m}^2\text{C}$
$C_c = \$50/\text{m}^2$ $\quad C_x = \$109/\text{m}^2$.

Assume that the collector and exchanger have the same useful life. What is the optimum exchanger area, and what is the heat exchanger factor and cost?

Using eqn (10), $A_x = 3{\cdot}32\ \text{m}^2$, and using eqn (8), the heat exchanger factor is 0·93. The cost is \$362, compared to a collector cost of \$5000.

As a final assumption in the analysis, it should be noted that the linearization of eqn (1) is adequate so long as the exchanger factor is not far from unity. Being between 0·9 and 1·0 is certainly close enough to unity. One must merely take care to use as U_c a value obtained from the slope of the curve "locally", i.e. at the temperature level at which the collector is intended to operate.

Sample calculation with more realistic cost factors. Reliable heat exchanger costs were obtained for Whitlock Type HT exchangers from the Walter W. Perkins Corporation of Los Angeles, and are shown in Fig. 1.§

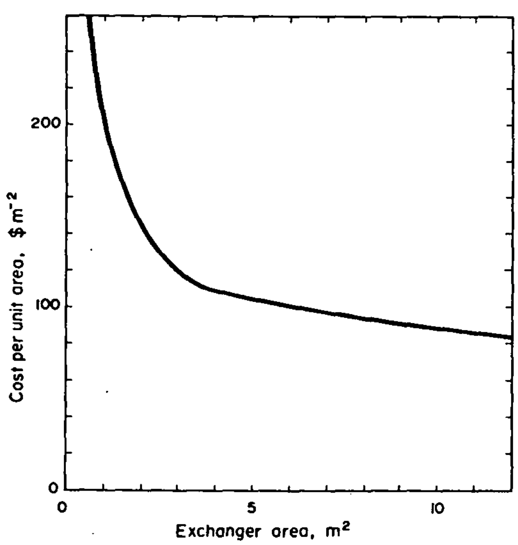

Fig. 1.

†Assuming zero heat loss in the piping.
‡It would have to be circulated at a (WC_p) product equal to that used in the secondary loop. Fluid *heat transfer* coefficients are not generally controlling in the flat plate collector, so a mismatch in convection coefficients is relatively unimportant.
§Values include standard "Original Equipment Manufacturer" discounts.

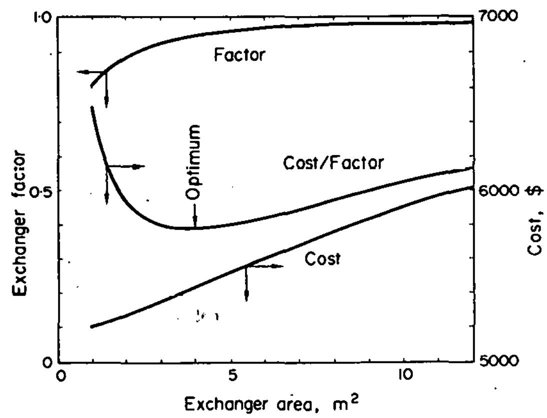

Fig. 2.

Water, circulated at 40C and 1·2 m/s through the ($\frac{3}{8}$ in.) tube side (according to McAdams[3], eqn 9-19) leads to a heat transfer coefficient of 6700 W/m^2C. The "fouling factor" to be expected with city water at 1·2 m/s is given in the TEMA Standards ([4], p. 60), to be 5678 W/m^2C. Dowtherm *J* circulated at 65C at a maximum velocity of 0·6 m/s through the shell side yields (according to McAdams[3], Fig. 10-21) a heat transfer coefficient of 2320 W/m^2C. The overall U_x value then becomes approximately equal to

$$U_x = \frac{1}{\frac{1}{6700} + \frac{1}{5678} + \frac{1}{2320}} = 1320\ W/m^2C.$$

It is quite reasonable to keep the overall heat transfer coefficient fixed while changing the heat exchanger area. In these shell and tube heat exchangers, area is varied by making the exchanger longer; the cross section stays the same.

Let us assume again that the collector has the same useful life as the heat exchanger, and that the collector specifications are as before.

Final optimization results are shown in Fig. 2. The optimum heat exchanger has a heat transfer area of about 4 m^2. The "heat exchanger factor" is 0·94, so that the collector collects 6 per cent less heat than it would have if water had been used directly. The heat exchanger investment is \$436, compared to \$5000. in collector cost, an increase of 8·7 per cent over a simple system. Comparing the two sample calculations, it can be seen that the use of constant heat exchanger cost-per-unit-area leads to exchangers which are somewhat too small. This was to be expected.

CONCLUSIONS

A simple factor has been developed to describe the solar energy collection penalty imposed by a heat exchanger in a double-loop system used with a flat-plate solar heat collector. It is shown that this factor makes it possible to determine the optimum size heat exchanger in a straightforward manner.

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Appendix B Fluid and Equipment Characteristics

There were a number of fluid and equipment characteristics that merited description in some detail in this report. They were included in this appendix, following the outline shown below.

B.1 Determination of Heat Transfer Coefficients

Heat transfer coefficients are required to optimize the heat transfer of a collector to storage system.. They also allow easy comparison of heat transfer fluids. In this study, heat transfer coefficients within the tubes, outside the tubes (i.e., shell-side heat transfer coefficients) and overall heat transfer coefficients were used. In the following sections, each of the above coefficients are discussed.

Inside Tube Heat Transfer Coefficients

For a double loop heat exchanger system, inside tube heat transfer coefficients must be specified for the collector tubes and the exchanger tubes. For a traced tank, the inside tube heat transfer coefficients must be determined for the helical coil also.

The inside tube heat transfer coefficient is dependent upon:

1. The flowrate through the tube
2. Cross-sectional area of the tube
3. Temperature of operation
4. Properties of the fluid at the operating temperature

Depending on the state of the fluid (i.e., laminar, transition, or turbulent) different correlations have been used to determine the inside tube heat transfer coefficients (h_{ti}). For the laminar region (Reynolds number less than 2500) the following correlation from McAdams (1954) was used:

$$h_{ti} = \frac{KC_2}{D_i} \tag{B.1-1}$$

Where:

$$C_2 = 1.75 \left(\frac{G'C_p}{KL} \right)^{1/3} \qquad \text{for } 1.75 \left(\frac{G'C_p}{KL} \right)^{1/3} > 3.66$$

$$C_2 = 3.66 \qquad \text{for } 1.75 \left(\frac{G'C_p}{KL} \right)^{1/3} < 3.66$$

h_{ti} - inside tube heat transfer coefficient (Btu hr^{-1} ft^{-2} $°F^{-1}$)
K - thermal conductivity of fluid (Btu hr^{-1} ft^{-1} $°F^{-1}$)
D_i - inside tube diameter (ft)
G' - flowrate (lb/hr)
C_p - specific heat of fluid (Btu/lb °F)
L - tube length (ft)

Thus, in the upper laminar region, the inside tube heat transfer coefficient depends upon the tube length also.

For the transitional region (2500 < Reynolds number < 7100), h_{ti} becomes:

$$h_{ti} = (C_p\mu/K)^{-2/3} \; C_p G''J' \qquad \text{(B.1-2)}$$

Where:

$$J' = 0.116 \; (Re^{2/3} - 125)/Re$$

and

$Re = G''D_i/\mu$ = Reynolds number
μ = viscosity of fluid lb/ft hr
G'' = flowrate lb/ft^2 hr tube

For the turbulent region (Re > 7100) the inside tube heat transfer coefficient from McAdams is:

$$h_{ti} = \frac{0.023 \; K \; Re^{0.8}}{D_i} (C_p\mu/K)^{0.4} \qquad \text{(B.1-3)}$$

Since in general, the transition region should be avoided, it was included only to provide continuity from the laminar to the turbulent regimes. Also note that at the interface between transitional and turbulent (Re=7100) and the interface between laminar and transitional (Re=2500), the equations do not predict similar inside tube heat transfer coeffients. For Re=7100 there is a 10% difference between the two equations, whereas around Re=2500 the error is larger. The selection of the transition region between Reynolds numbers 2500 and 7100 was completely arbitrary. It was chosen to minimize the errors at the two boundaries and to allow reasonable heat transfer in the lower turbulent region.

Because of the difficulty in using these equations by the average user, computer analyses were conducted for the fluids under consideration for various operating conditions. The temperature of operation, tube inner diameter, and operating flowrate were varied for each heat transfer fluid to determine the inside transfer coefficients. The results of these runs are shown in Figures B.1-1 through B.1-9. Note that the transitional region results were omitted since it was decided not to design systems to operate within this region. Correction factors for varying temperatures are included along with a tube length correction factor for laminar flow with $C_2 > 3.66$. Also included is an acceptable upper limit to flow within tubes i.e. 5 ft/second. Flow above this should be avoided due to increased chance of erosion-corrosion and limitations of the tubing. Of the non-aqueous solutions, only Dowtherm J can operate within the turbulent regime for the smaller tube sizes. Since the heat transfer fluids in a double loop heat exchanger system operate within tubes only in the collector, the effect of the fluids operating in the laminar regime is reduced. This is further discussed in Section B.2.

For the traced tank system, the effect of the fluids operating in the laminar regime is greater since the heat transfer coefficients can affect the helical coil efficiency. Fluids operating in laminar flow have much lower efficiencies than those operating in the turbulent regime. See Section 2.2 for a more thorough discussion of this.

Some simple relationships between the flowrate in gallons/(minute tube) and the other flowrates follow:

$$Q = Q_n N$$

$$Q_n = \frac{0.1247\ G'}{\rho N}$$

$$G'' = \frac{4G'}{\pi D_i^{\,2} N}$$

Figure B.1-1

Inside Tube Heat Transfer Coefficient Versus Flowrate for Water

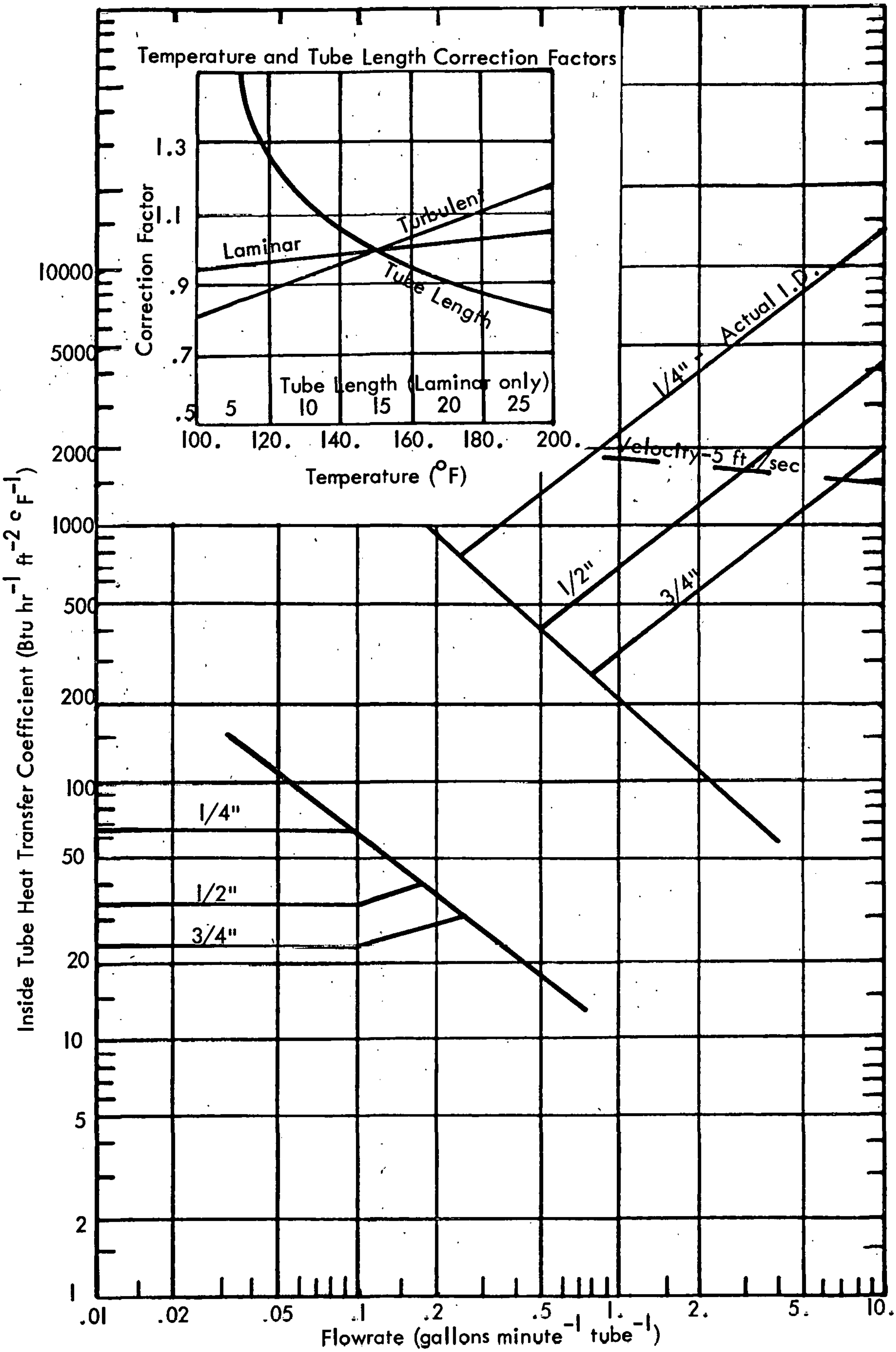


Figure B.1-2

Inside Tube Heat Transfer Coefficient Versus Flowrate For 50% ethylene glycol

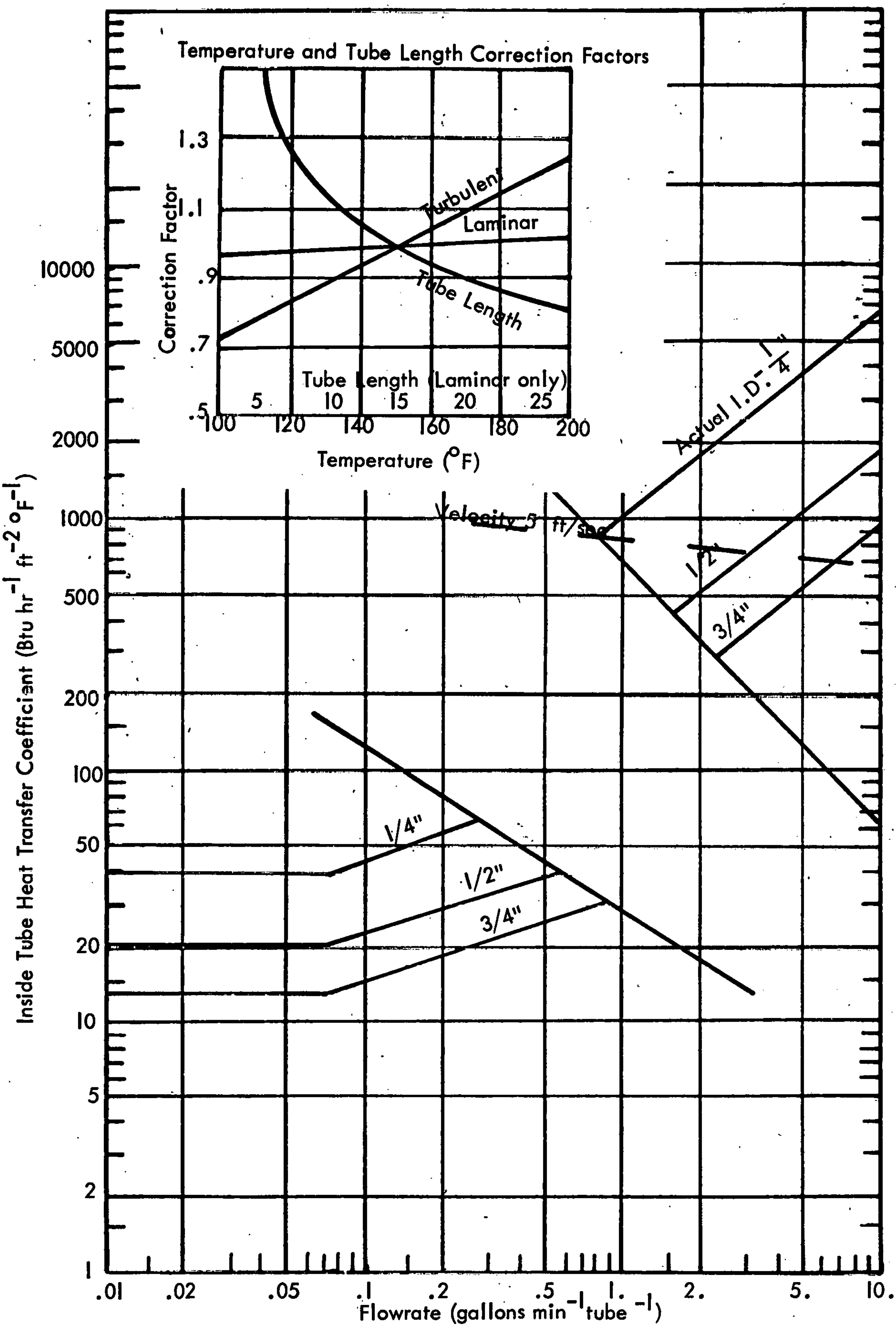


Figure B.1-3

Inside Tube Heat Transfer Coefficient Versus Flowrate for 50% propylene glycol

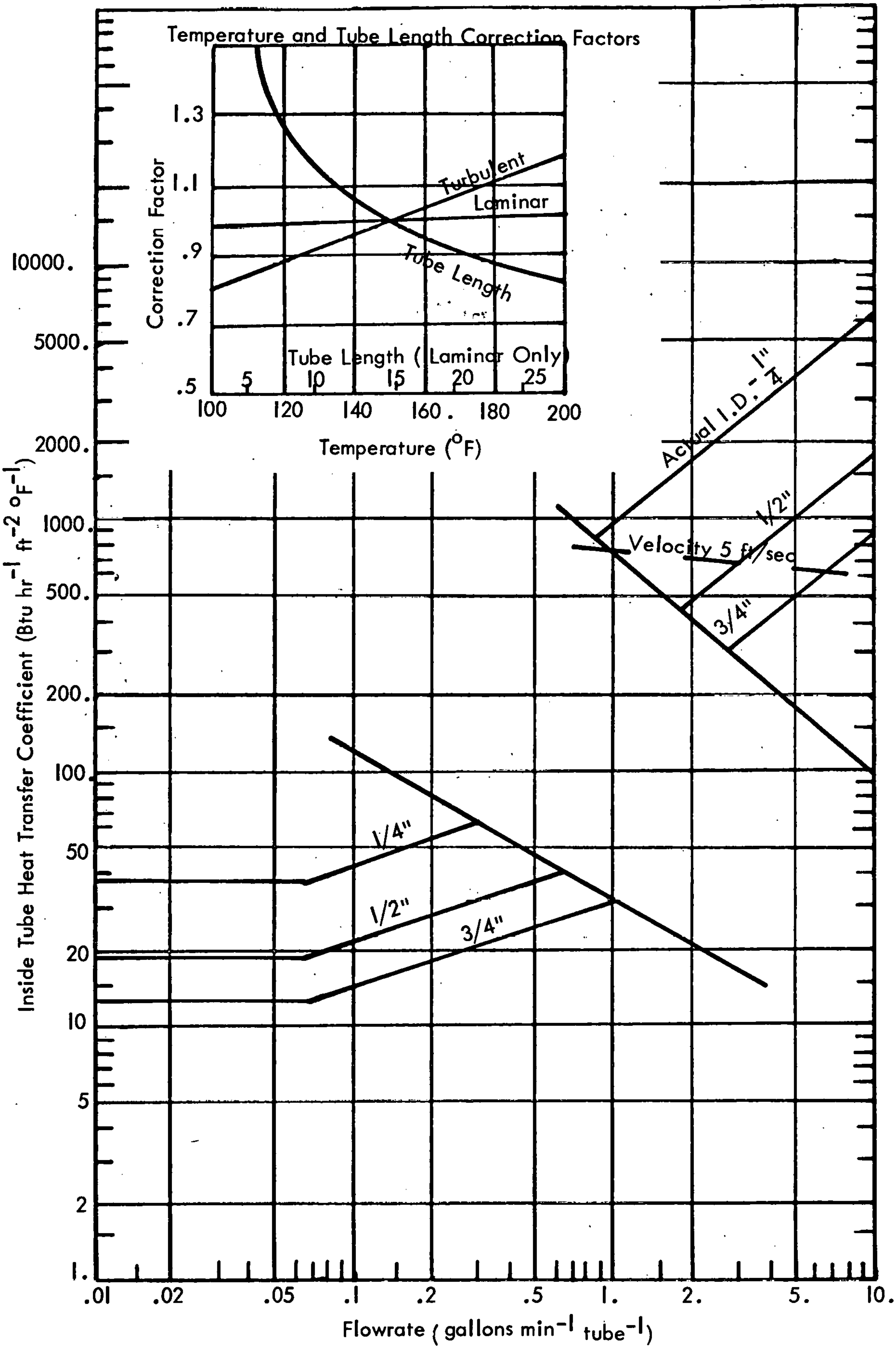


Figure B.1-4

Inside Tube Heat Transfer Coefficient Versus Flowrate For Mobiltherm Light

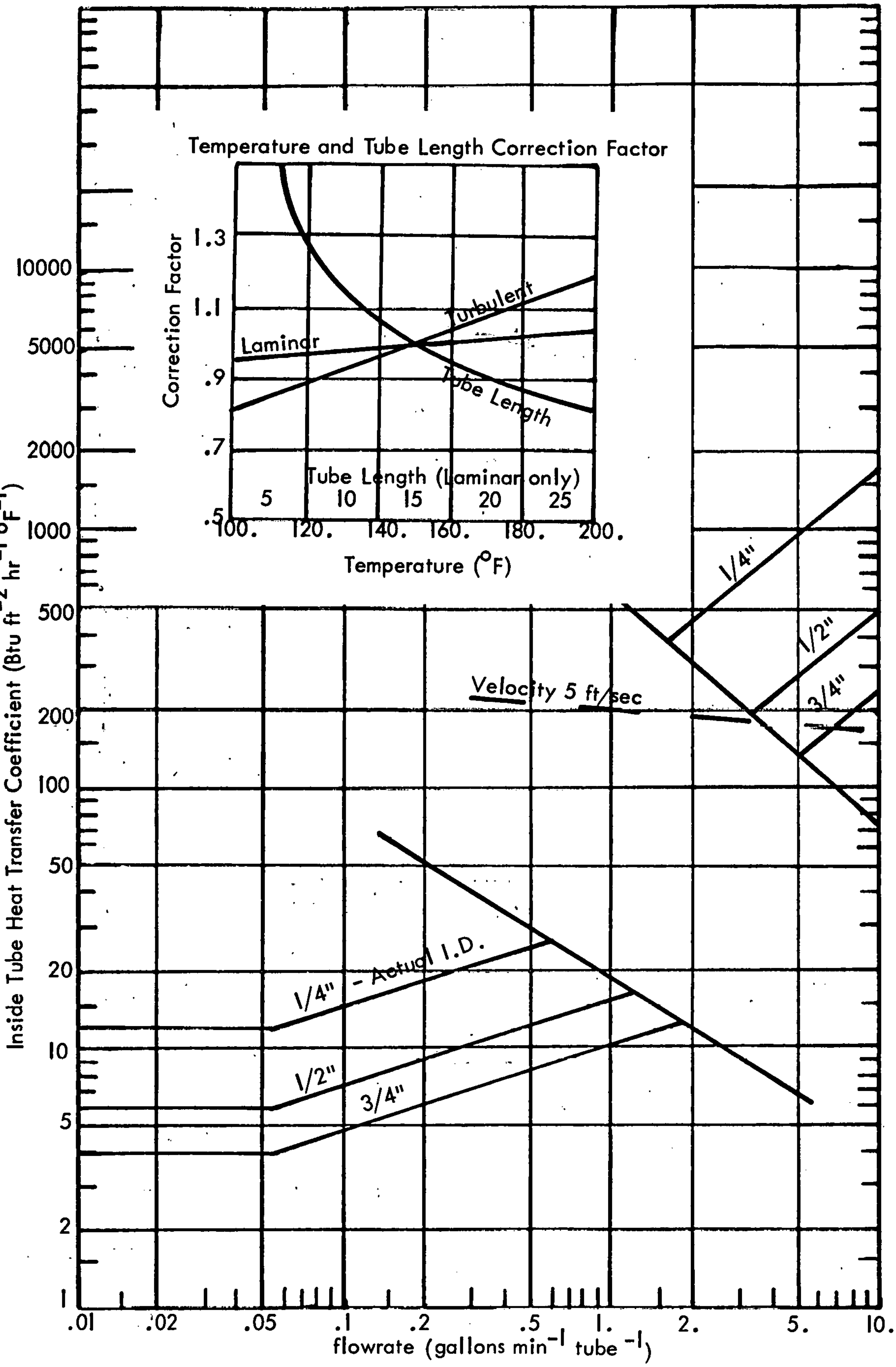


Figure B.1-5

Inside Tube Heat Transfer Coefficient Versus Flowrate for Q2-1132 (Silicone)

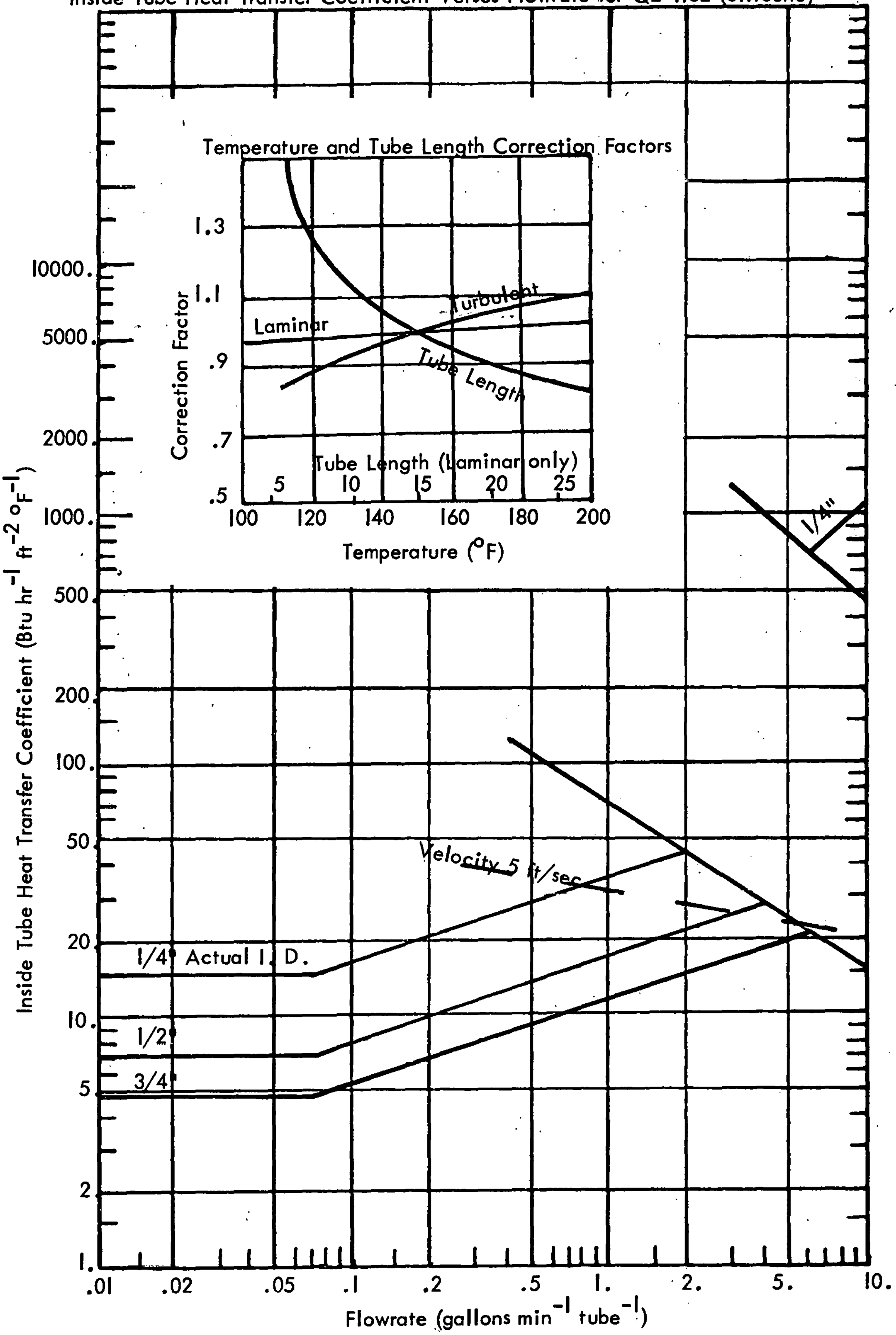


Figure B.1-6

Inside Tube Heat Transfer Coefficient Versus Flowrate and Inner Tube Diameter for Dowtherm

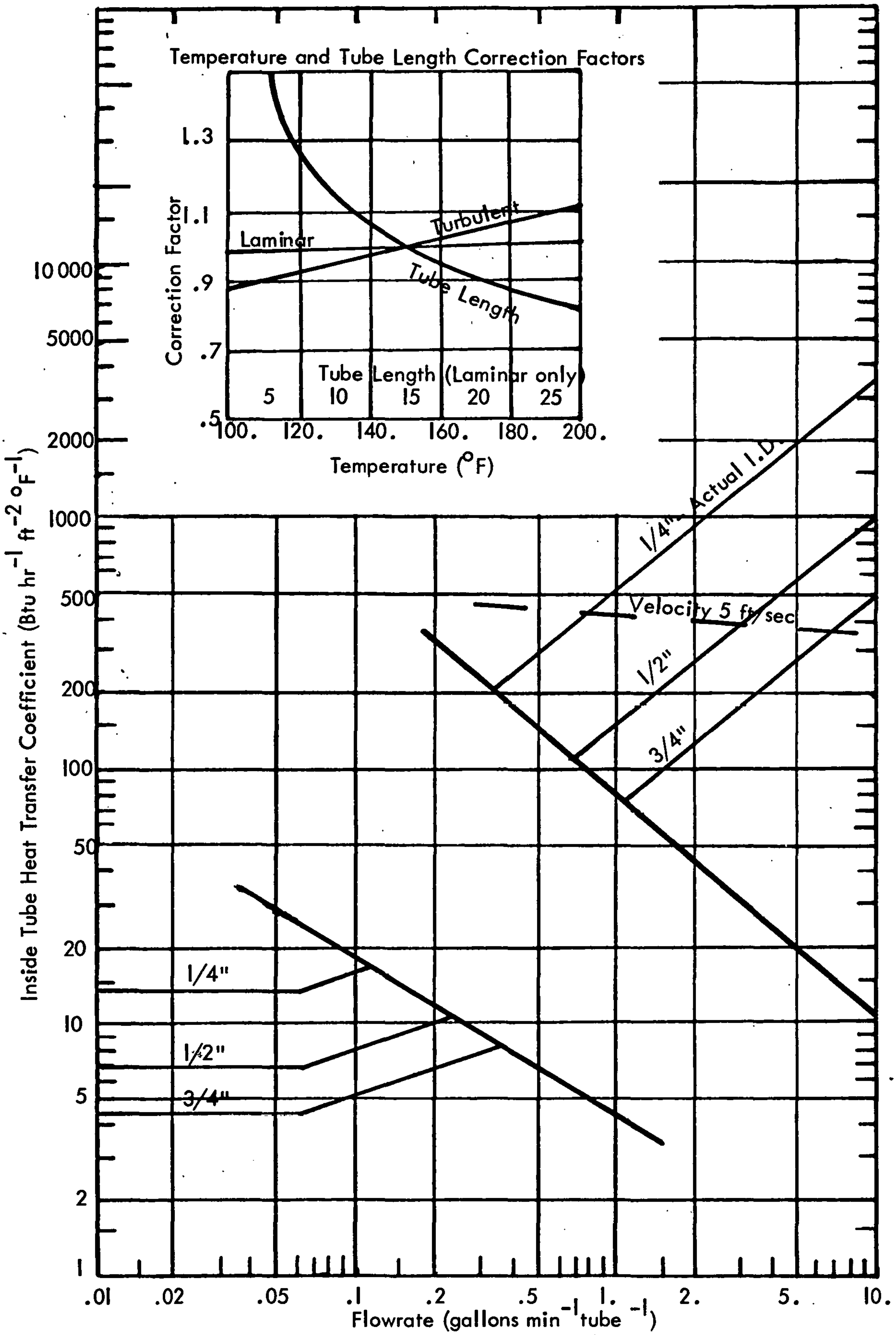


Figure B.1-7

Inside Tube Heat Transfer Coefficient Versus Flowrate for Sun-Temp

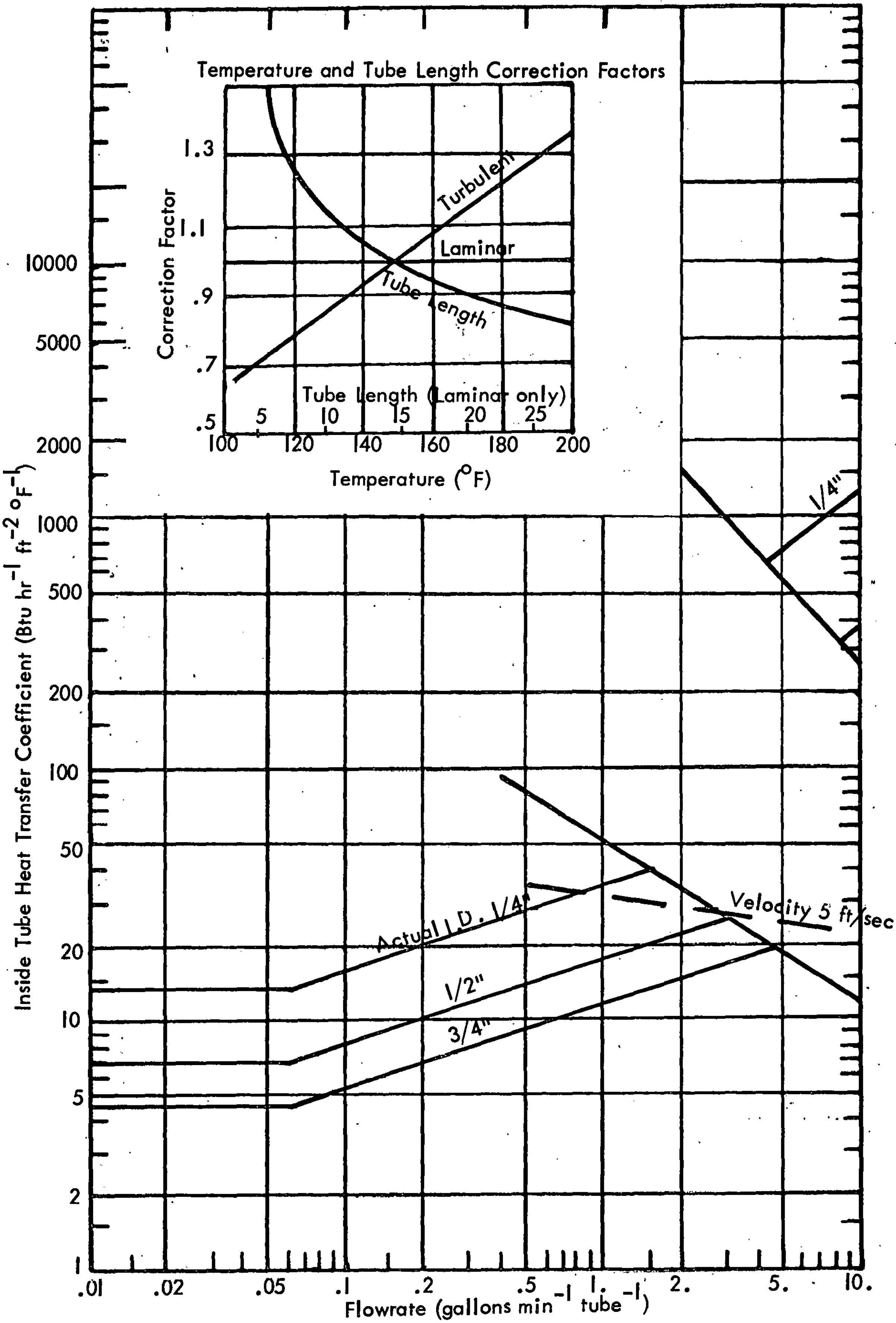


Figure B.1-8

Inside Tube Heat Transfer Coefficient Versus Flowrate for Therminol 55

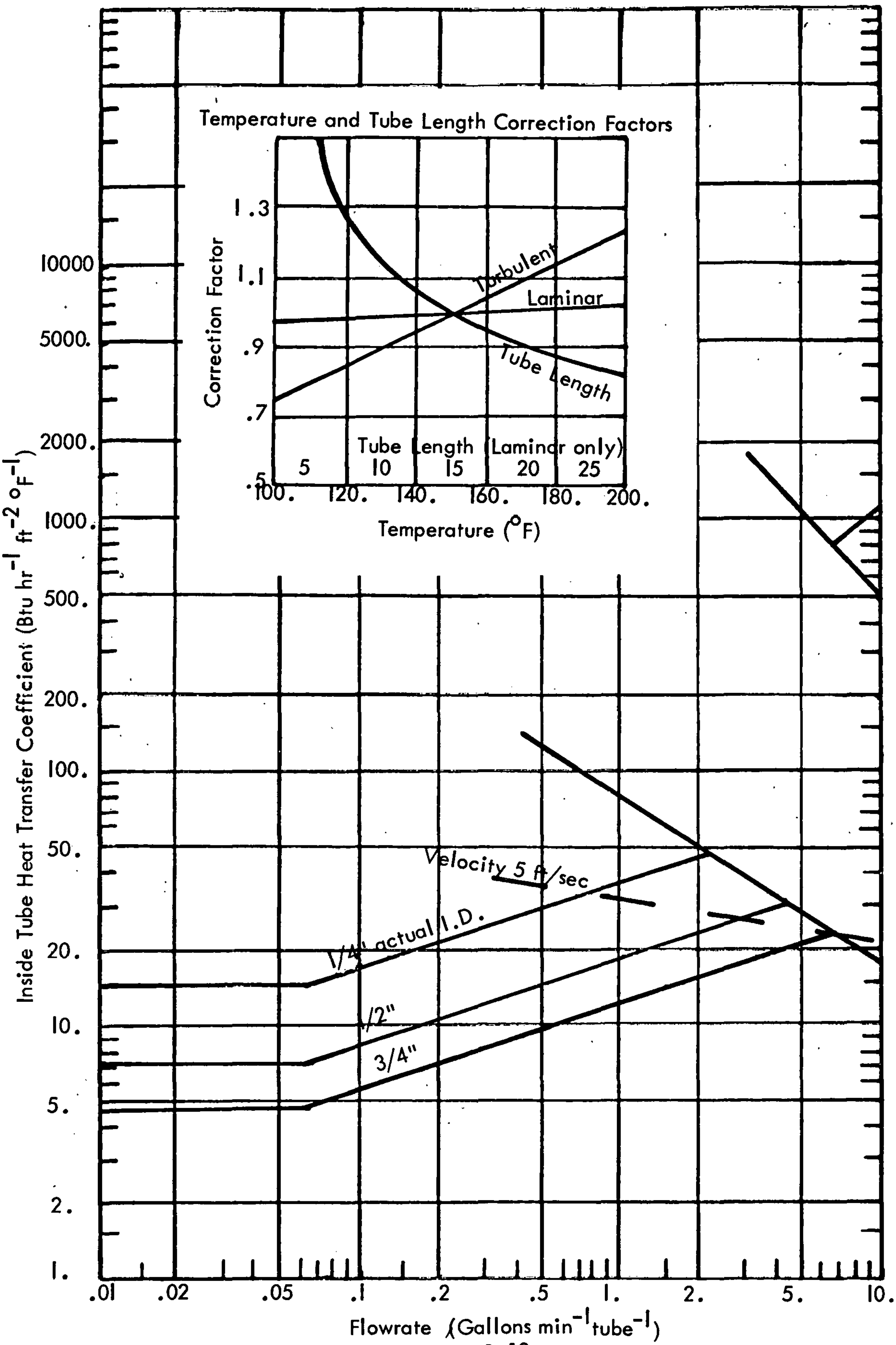
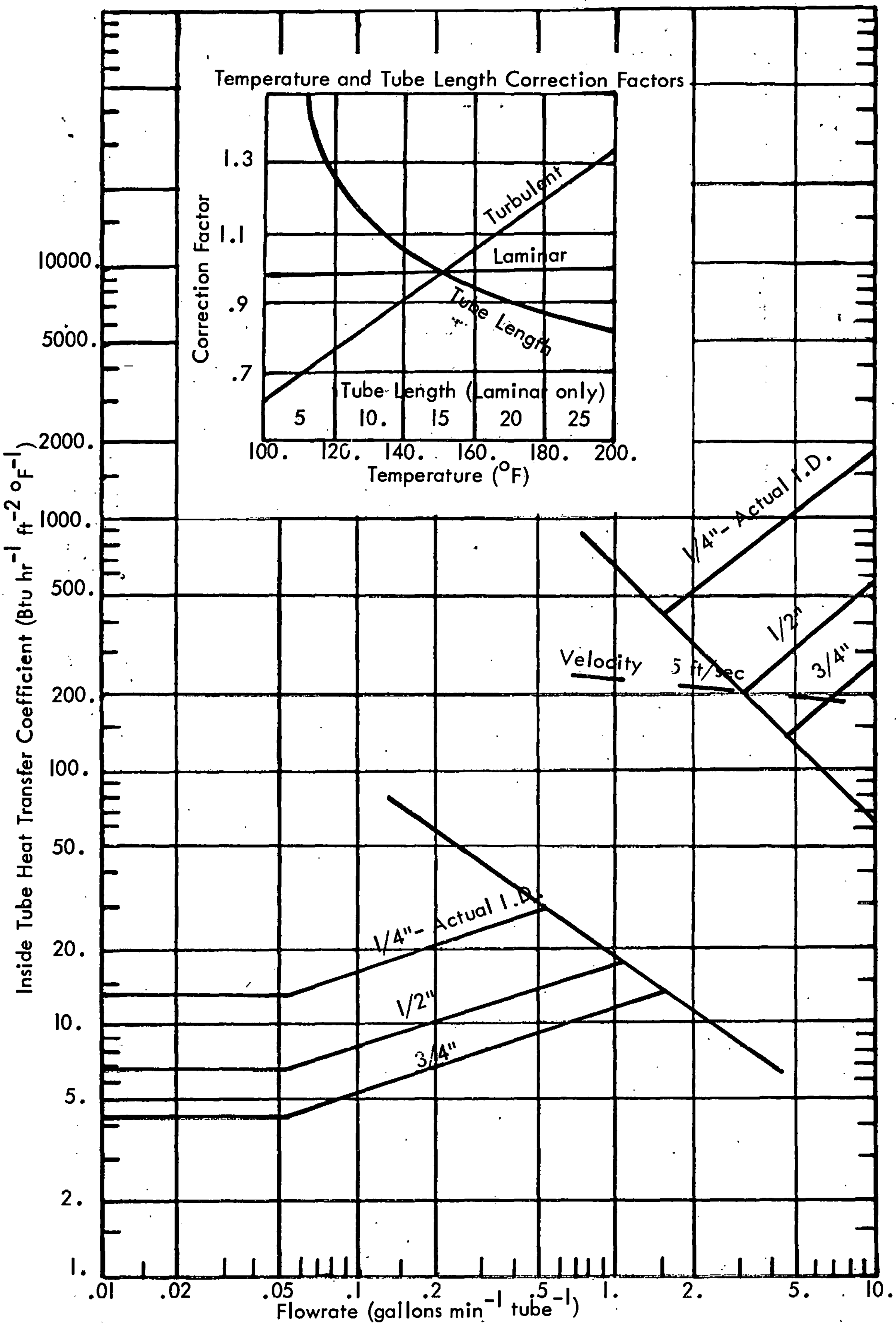


Figure B.1-9

Inside Tube Heat Transfer Coefficient Versus Flowrate and Inner Tube Diameter For Therminol 60



Where:

Q - total system flowrate (gallons/minute)

N - total number of tubes

Q_n - flowrate gallons/(minute tube)

ρ - density of the fluid lb/ft^3

Simple determinations of the inside tube heat transfer coefficients are shown in Section 2.1.4.

Shell Side Heat Transfer Coefficient

The shell side heat transfer coefficient within the heat exchanger (h_{to}) was determined for those fluids studied. h_{to} is a function of:

1. shell entrance flowrate
2. temperature of operation
3. fluid properties at the operating temperature
4. characteristics of the heat exchanger
   a. Tube pitch
   b. Baffle spacing
   c. Outer tube diameter
   d. Number of tube rows

A correlation was found from Kreith & Kreider:

$$h_{to} = 0.33\ Re'^{\,0.6}\ (C_p\mu/K)^{0.33}\ K/D_o \qquad \text{(B.1-4)}$$

$Re' = \frac{G_{max} D_o}{\mu}$ = Reynolds number through minimum cross-sectional area of heat exchanger

G_{max} - flowrate through the minimum cross-sectional area of the heat exchanger lb/ft^2 hr

$$G_{max} = \frac{G_s}{A_{min}\ (N_{row} + 1)}$$

G_s - total shell flowrate lb/hr

Q_s = total shell flowrate gal/min

$= 0.1247\ G_s/\rho$

A_{min} - minimum cross-sectional area = $S_{baf}S_{min}$ (ft^2)

S_{baf} - baffle spacing (ft)

S_{min} - tube spacing (ft)
= (pitch -1) D_o

Pitch= equilateral triangular pitch = 1.25

N_{row} = number of tube rows across diameter of shell. This is a conservative estimate of the number of tube openings available for the fluid to flow through.

Figure B.1-10 shows the exchanger characteristics more readily. Computer analyses were conducted to determine the shell side heat transfer coefficients for varying flowrates, temperature and exchanger characteristics. Figure B.1-11 shows these results for several fluids. Correction factors follow in Figure B.1-12. The temperature correction factor was the average change in shell side heat transfer coefficient for all fluids for varying temperatures.

Overall Heat Transfer Coefficient

The overall heat transfer coefficient of a heat exchanger (U_x) can be determined from the following equation by Kays and London (1964):

$$U_x = \frac{1}{\frac{1}{h_{to}} + \frac{1}{h_{so}} + \frac{D_o}{D_i h_{ti}} + \frac{D_o}{D_i h_{is}} + R_{wall}} \qquad \text{(B.1-5)}$$

Where:

h_{so} - shell side scaling coefficient (Btu/hr ft^2 °F)

h_{is} - inside tube scaling coefficient (Btu/hr ft^2 °F)

R_{wall} - tube wall heat transfer resistance (hr ft^2 °F/Btu)

$$R_{wall} = \frac{D_o}{2\,K_{tex}} \ln \frac{D_o}{D_i}$$

K_{tex} - thermal conductivity of the tube wall (Btu/hr ft^2 °F)

Figure B.1-10

Shell Side Exchanger Characteristics

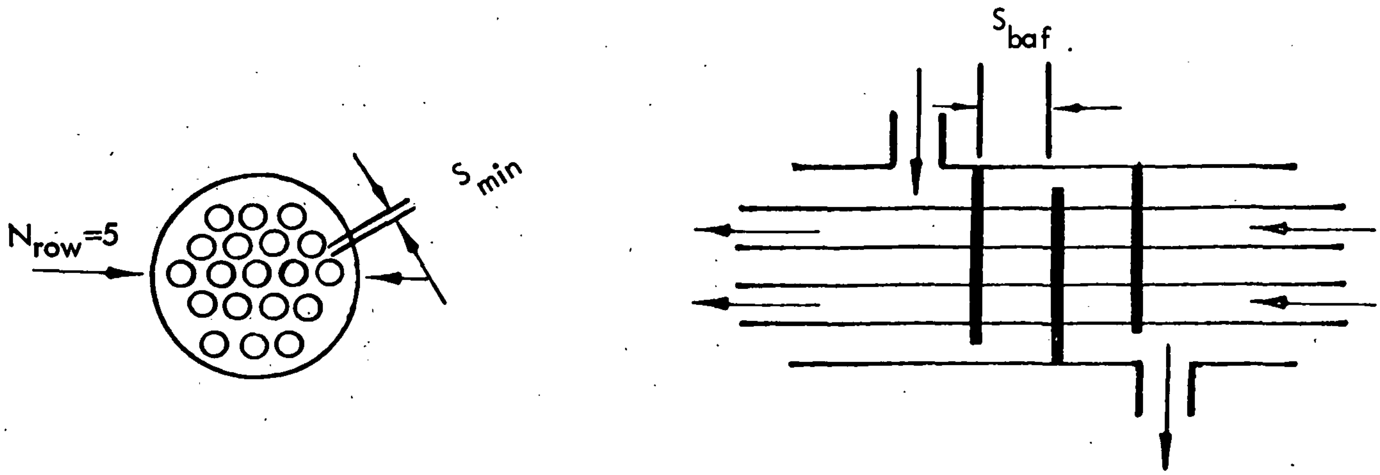


Figure B.1-11

Shell Side Heat Transfer Coefficient for the Heat Transfer Fluids Versus the Total Shell Flowrate

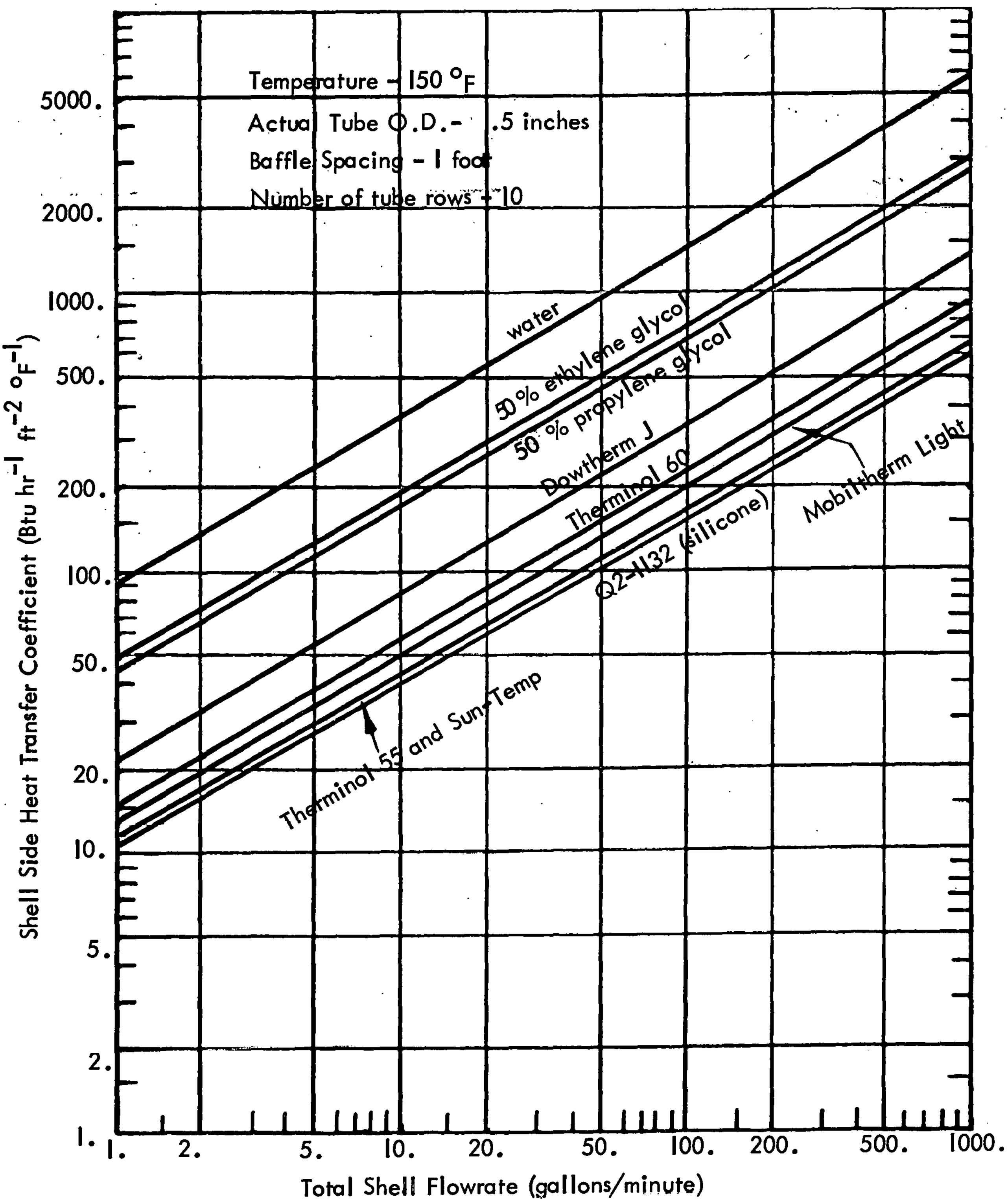
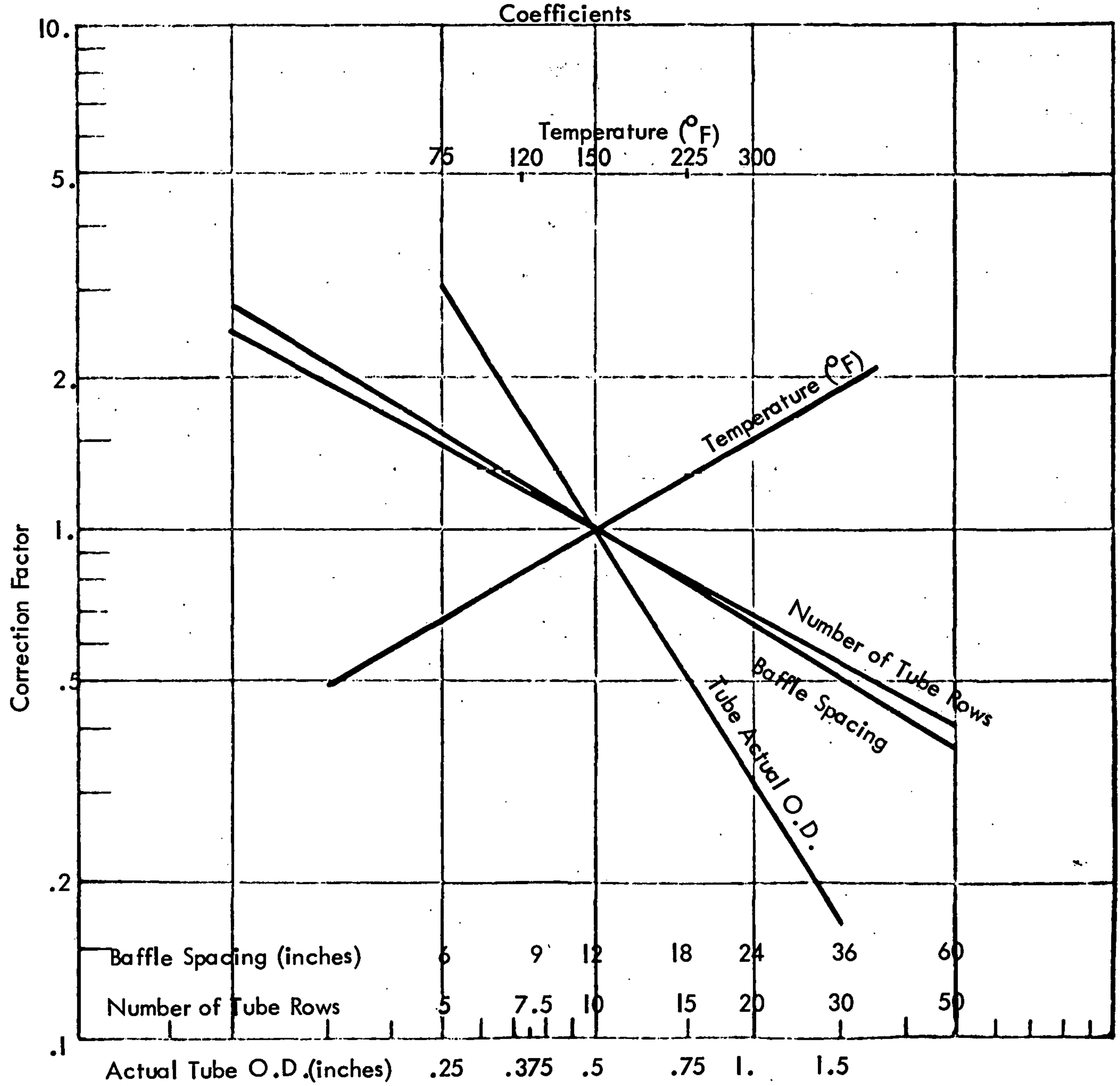


Figure B.1-12

Temperature and Exchanger Characteristics Correction Factors For Shell Side Heat Transfer Coefficients



In this study the scaling coefficients were assumed constant for all fluids and tube sizes and equal to 1000 BTU/hr ft^2 °F. Normally scaling coefficients decrease with time due to increased scaling deposits on the inner and outer tube walls, if maintenance is not periodically performed. This can reduce the performance of the heat exchanger and increase the possibility of corrosion.

For copper tubing within the heat exchanger, the wall resistance is generally negligible, but the equation for the overall heat transfer coefficient can not be reduced further. For type "L" (medium thickness) copper tubing of 3/8" norminal outer diameter, Figure B.1-13 shows the variation of U_{ex} versus h_{ti} and h_{to}. Since curves of this sort would have to be generated for all differing tube wall thicknesses and outer tube diameters, it is better to use equation (B.1-5) to determine the overall heat transfer coefficient. An example is shown below to determine the heat transfer coefficients and the overall heat transfer coefficients.

Example of how to find U_x

Find Inside Tube Heat Transfer Coefficient of Heat Exchanger

A rule of thumb was developed in Section 2.1.4 to find an acceptable flowrate within the exchanger tubes. It is:

$$Q_n = D_i + 0.125 \text{ (gallons min}^{-1}\text{ tube}^{-1})$$

D_i = actual inside diameter (inches)

From Figure B.1-1 the inside tube heat transfer coefficient can be obtained if the flowrate and tube inner diameter are known.

Example:

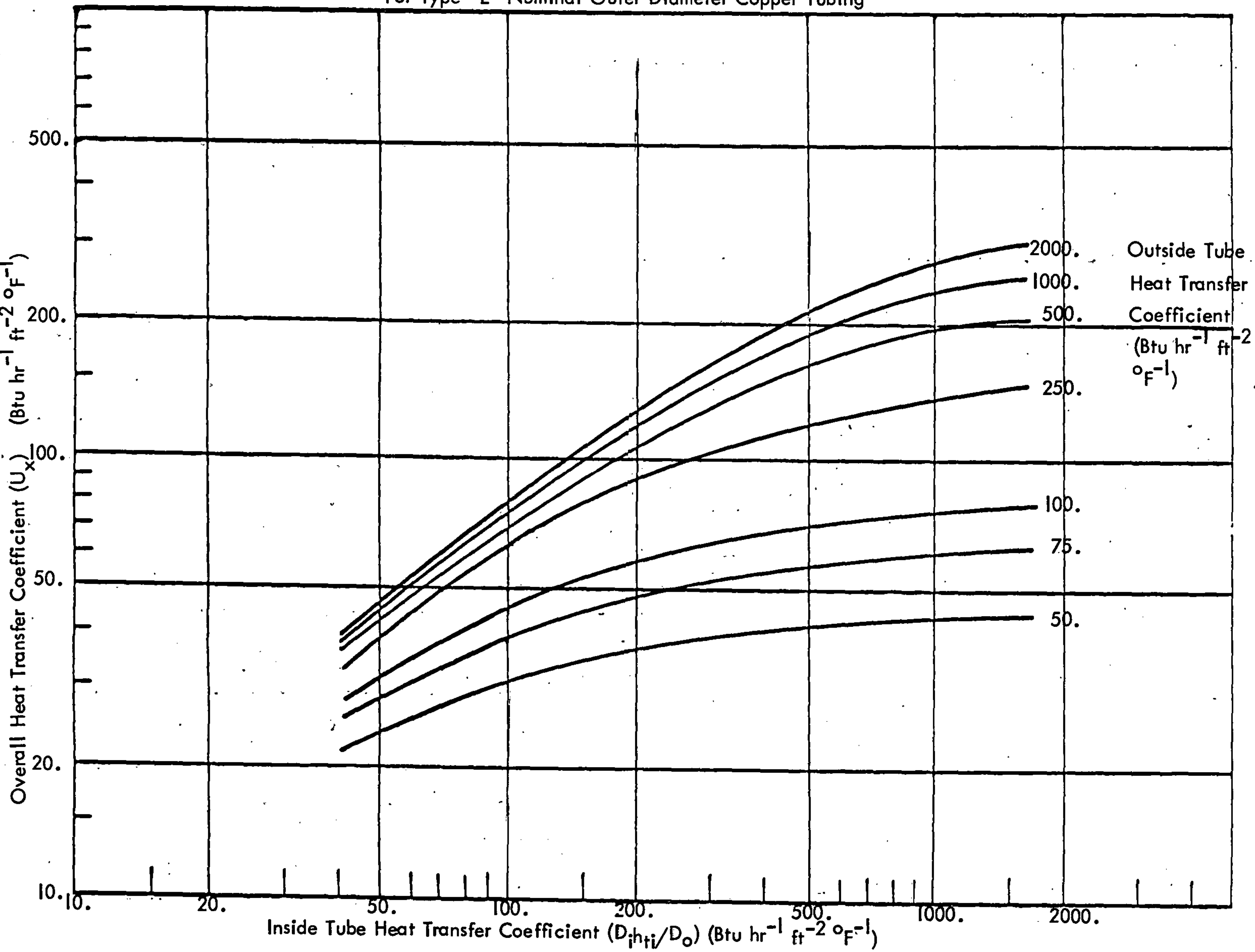
Assume 3/8" actual I.D. tube is to be used

then:

Q_n = 0.5 gallons/minute. By interpolation on Figurc B.1-1.

Figure B.1-13

Overall Heat Transfer Coefficient (U_x) Versus Inside and Outside Tube Heat Transfer Coefficients For Type "L" Nominal Outer Diameter Copper Tubing



then:

$$h_{ti} = 750 \text{ BTU hr}^{-1} \text{ ft}^{-2} \text{ °F}^{-1}$$

If this design method is unacceptable,

then:

Q_n can be found from $Q_n = \dfrac{0.2493\ \bar{G}A_c}{(C' + 1)\rho N}$

G = mass flowrate lb ft^{-2} hr^{-1}

A_c = collector area ft^2

ρ = density of water lb/ft^3

C' = capacity rate ratio

N = No. of exchanger tubes

<u>Outside Tube Heat Transfer Coefficient of Heat Exchanger</u>

The capacity rates of the two loops are interrelated. To find the total shell side flowrate (Q_s), the capacity rate ratio, collector area, area flowrate, density of the collector loop fluid (ρ_f), and the specific heat ratio (C_{pw}/C_{pf}) must be known. Q_s can be found from:

$$Q_s = \frac{0.2493\ C'}{1 + C'\rho_f}\ \bar{G}\ \frac{C_{pw}}{C_{pf}}\ A_c$$

For example, if C' = 0.5, A_c = 1000 ft^2, $\bar{G}$ = 16 lb/ft^2, C_{pw} = 1 BTU/lb °F with a 50% ethylene glycol acqueous solution with ρ_f = 65 lb/ft, and C_{pf} = 0.85 BTU/lb °F @ 150 °F then Q_s = 24 gal/min.

Assuming that all the heat exchanger characteristics are the same as those listed on Figure B.1-10, then h_{to} becomes:

$$h_{to} = 350 \frac{\text{BTU}}{\text{hr ft}^2 \text{ °F}}$$

From Equation (B.1-5) with D_i = 0.375 in and D_o = 0.5 in. U_x becomes:

$$U_x = \frac{1}{\frac{1}{350} + \frac{1}{1000} + \frac{1.33}{750} + \frac{1.33}{1000}} = 144 \frac{\text{BTU}}{\text{hr Ft}^2 \text{ °F}} = 818 \frac{\text{watts}}{\text{m}^2 \text{ °C}}$$

With manipulation of the baffle spacing this could be higher.

Overall Heat Transfer and Cost Characteristics of the Heat Transfer Fluids

Water and other aqueous solutions have good heat transfer because of water's low viscosity, high thermal conductivity and specific heat, and minimal cost. Other properties of aqueous solutions can reduce the cost effectiveness of these heat transfer fluids, i.e. inadequate freeze protection or increased corrosion and high vapor pressure at high temperatures. In the following sections, water and other aqueous solutions are compared versus other heat transfer fluids for heat transfer, cost, and overall cost effectiveness. In this study, the heat transfer fluids are compared under similar operating conditions. The capacity rates are assumed matched for the double loop systems for comparison purposes. In order to keep the capacity rates matched with the different specific heats of the fluids, the flowrate is determined from the following form of equation:

$$G_{fluid} = G_{water} \frac{(C_p)_{water}}{(C_p)_{fluid}} \qquad (B.1\text{-}6)$$

For low specific heat fluids, the fluids flowrates can be over twice that of water.

Computer iterations were conducted to determine the heat transfer characteristics of each of the fluids under similar operating conditions (see Appendix D and E for the computer listings). Table B.1-1 shows one such calculation for each of the fluids versus collector area. The heat transfer properties are noontime steady state calculations for the typical systems. For the $10m^2$ collector, a traced tank system was used with 20 meters of tubing in 2 helical coils around the storage tank and connected to the collector (see Section 2.2 for other traced tank optimum design parameters used). For the larger collector sizes, a double loop system utilizing a heat exchanger was used with an exchanger area equal to 8% of the collector area. The number of tubes was fixed at 7/10 of the collector (in m^2) for collector areas greater than 40 m^2. Below 40 m^2, the number of tubes within the exchanger was constant and equalled 28. This is the minimum number of tubes currently available for off-the-shelf heat exchangers (see Section B.3). Both the tube length for the helical coil and the heat exchanger area in the double loop system were slightly above optimum in size.

Table B.1-1

Effect of Heat Transfer Fluids on Heat Transfer and Cost for Typical Systems

Heat Transfer Fluid	Collector Area (m^2)	Collector efficiency F_R	U_x (Ex'er) F_tU_t (Traced) (Tank) ($\frac{Watts}{m^2\ {}^oC}$)	F_{ex}' (Ex'er) F_t' (Traced) (Tank)	Rate of Heat Transfer Q (Watts) x10^4	(Q_{water}) (potable) Q_{fluid}	Initial Fluid Fillup Cost ($/year)	Pumping Cost ($/year)	Total Cost C_t ($/year)
Water	10*	0.7734	-	-	0.4089	1	-	0.1924	125.19
Circulating	20	0.7739	-	-	0.7171	1	-	0.005	250.0
From	40	0.7739	-	-	1.436	1	-	0.022	500.0
Storage	80	0.7739	-	-	2.891	1	-	0.044	1000.0
Water	10*	0.7734	370	0.9469	0.3872	1.0561	-	0.1924	137.69
Used as	20	0.7739	928	0.9504	0.6818	1.0522	-	0.005	282.5
HTF Within	40	0.7739	1218	0.9618	1.380	1.0397	-	0.022	552.5
Double loop or Traced Tank System	80	0.7739	1228	0.9621	2.781	1.0394	-	0.044	1092.5
50%	10*	0.7529	331	0.9409	0.3769	1.085	2.25	0.367	140.1
Ethylene	20	0.7535	827	0.9461	0.6615	1.085	3.33	0.015	285.7
Glycol	40	0.7535	1056	0.9573	1.360	1.063	5.63	0.065	557.9
	80	0.7535	1068	0.9578	2.720	1.062	10.13	0.225	1103.0
50%	10*	0.7502	331	0.9369	0.3745	1.092	2.13	0.368	140.0
Propylene	20	0.7502	810	0.9453	0.6582	1.090	3.17	0.017	285.7
Glycol	40	0.7502	1031	0.9565	1.355	1.067	5.32	0.074	557.9
	80	0.7502	1035	0.9570	2.710	1.066	9.58	0.945	1103.0
mobiltherm	10*	0.6662	267	0.9332	0.3419	1.195	1.67	3.98	143.1
light	20	0.6662	590	0.9341	0.6071	1.182	2.42	0.156	295.1
	40	0.6662	725	0.9457	1.227	1.178	4.08	0.656	557.2
	80	0.6662	738	0.9466	2.456	1.178	7.33	2.31	1102.1
Q2-1132	10*	0.6837	165	0.884	0.3205	1.276	28.75	7.329	173.6
Silicone	20	0.6837	576	0.9309	0.6187	1.160	42.58	0.698	325.8
fluid	40	0.6837	581	0.9428	1.251	1.155	71.92	2.92	627.3
	80	0.6837	590	0.9435	2.503	1.155	129.42	10.18	1221.9
Dowtherm	10*	0.6766	321	0.9445	0.3514	1.164	5.63	2.292	145.4
J	20	0.6766	686	0.9420	0.6194	1.158	8.33	0.0462	290.9
	40	0.6766	864	0.9534	1.252	1.154	14.08	0.2067	566.8
	80	0.6766	870	0.9540	2.504	1.155	25.32	0.617	1118.5
Suntemp	10*	0.670	145	0.8708	0.3151	1.298	4.38	2.873	144.7
	20	0.670	542	0.9283	0.6068	1.183	6.50	0.26	289.3
	40	0.670	656	0.940	1.227	1.178	10.92	1.11	564.5
	80	0.670	670	0.941	2.456	1.177	19.68	3.90	1116.1
	10*	0.6789	70.8	0.7547	0.2747	1.489	3.5	2.22	143.2
Therminol 55	20	0.6789	549	0.9283	0.6137	1.168	5.18	0.48	288.2
	40	0.6789	668	0.9402	1.241	1.164	8.75	2.03	563.3
	80	0.6789	679	0.9412	2.484	1.164	15.75	7.10	1115.3
Therminol 60	10*	0.676	274	0.9333	0.3481	1.174	8.5	3.63	149.6
	20	0.676	613	0.9355	0.6154	1.166	12.59	0.144	295.2
	40	0.676	757	0.9471	1.244	1.161	21.25	0.608	574.4
	80	0.676	770	0.9480	2.489	1.161	38.25	2.12	1132.9

*Traced tank single loop system. All other collector sizes utilize double loop heat exchanger systems.

Table B.1-1 shows that two cases for water were included. One case allows the potable water to be circulated directly from the storage to the collector assuming freeze protection is facilitated by a reserve draindown system. For comparison purposes, water was also used as a heat transfer fluid circulating through the collector loop of the double loop heat exchanger system or circulating through the traced tank loop for the smaller collector sizes. There would be no advantage in operating a system with water this way, since the penalty imposed by the use of the heat exchanger or traced tank would reduce the rate of heat transfer while still running the risk of inadequate freeze protection. Water used in this fashion does show the best possible heat transfer and the lowest initial costs of any heat transfer fluid to be circulated through the double loop or helical coil systems.

The potable water case, for simplicity, was assumed to have similar collector performance compared to the water in the double loop or single loop traced tank systems. Thus the only difference for the two cases is that the potable water case has no heat exchanger penalty factor or traced tank penalty factor imposed. In reality, the heat exchanger or helical coil would also increase the operating temperature of the collector and thus reduce the collector performance. This effect was ignored in this section.

Heat Transfer

In Table B.1-1, the effects of the heat transfer fluids upon the heat transfer of the systems are shown. Note that the collector efficiency (F_R) varies little with collector area since the same type of collector was used for all cases and the number of collector tubes was a function of the collector size. The number of collector tubes equalled twice the collector area (m^2) to reduce the pressure drop within the collector and to keep all the fluids in the laminar regime within the collector. Some fluids, water and Dowtherm J, could easily operate in the turbulent regime within allowable pressure drops if the number of tubes was reduced. This would increase the collector efficiency by up to 10%. All of the heat transfer fluids had collector efficiencies within 15% of that of water.

Also included for design purposes on Table B.1-1 are the overall heat transfer coefficients (U_{ex}) for the heat exchanger cases, and the helical coil efficiency-tank natural convection coefficient product ($F_t U_t$) for the traced tank loop. For the aqueous solutions at the larger collector areas, the overall heat transfer coefficients are from 1000 to 1200 watts m^{-2} $°C^{-1}$, while for the non aqueous fluids, the overall heat transfer coefficients lie between 650 and 850. The $F_t U_t$ product for the traced tank case varies from 370 for water to 71 watts m^{-2} $°C^{-1}$ for therminol 55. This large variation is due to the reduced efficiency of the traced tank when the fluids are not operating in the turbulent regime within the helical coil.

Also listed in Table B.1-1 are the exchanger penalty factor (F_{ex}') and the traced tank penalty (F_t'). As discussed above, the use of water within the single loop traced tank or double loop exchanger system reduced the amount of heat transferred by up to 5 1/2% over that possible if water was circulated directly from the storage tank to the collector. The other heat transfer fluids reduced the rate of heat transfer by up to 25% in comparison to the water circulating directly.

To show the overall effect of the heat transfer fluids on the rate of heat transfer, the noon time steady state rate of heat transfer is included along with the ratio of the rate of heat transferred by the potable water to that rate of heat transferred by the other heat transfer fluids. Note that the use of Therminol 55 in the traced tank case reduced the rate of heat transfer by approximately 50% due to its operation within the laminar region in the helical coil. Other fluids do not reflect this large of a drop in performance with aqueous solutions reducing the rate of heat transfer by less than 10%. Most of the non-aqueous fluids for the larger collector sizes reduce the rate of heat transfer by less than 20%.

Cost

The effect of the initial and expected costs of each of the fluids on the total system cost was also considered. The collector cost for these cases were assumed to be $100/m^2$, with the exchanger costs similarly $100/m^2$, and the traced tank tubing costing $5/m. An extra pump and tubing was required for the double loop exchanger system and was included at an additional cost of $100. All of the components, including the initial fluid fillup, were assumed to be paid on an annual basis of 12.5%. This corresponds to a current money cost of 10% with the useful life of each of the components being 15 years. Although some of the fluids could be expected to last this long (such as silicone fluids), others, such as glycols, will be subject to further fluid and inhibitor addition to reduce the corrosion, and will have useful lives much less than 15 years. For purposes of comparison, it was assumed that the user would continue to pay for the fluid on an annual basis for the entire system lifetime with additional costs for more fluid (if needed) paid in the later years. Note that the potable water case requires only the collector cost since neither a heat exchanger or a helical coil is used.

The initial fillup costs were determined from Table B.2-2, assuming the amount of fluid required (in gallons) equalled one half the collector area (in m^2) plus 5 gallons. This was slightly more fluid required than needed for the smaller collector sizes.

The pumping cost per year of each of the fluids were estimated under the operating conditions. It was assumed that the pumping costs of the storage loop (water side) in the double loop system were minimal. The pumping costs were calculated from:

$$C_{pump} = \frac{C_{elec} h_{op} d_{op} \Delta P \, Q_t}{2298.7 \, P_{eff}} \qquad \text{(B.1-7)}$$

Where:

C_{pump} - cost of pumping the fluid (\$/year)

C_{elec} - cost of the electricity required = \$0.03/kilowatt-hr

h_{op} - hours of operation per day = 8 hours/day

d_{op} - days of operation per year = 365 days/year

Δp - pressure drop through the collector loop (psi)

Q_t - flowrate through the collector loop (gallons/minute)

P_{eff} - pump efficiency = 0.7

Using the above assumptions, the pumping costs become:

$$C_{pump} = 0.0544 \quad P \, Q_t \tag{B.1-8}$$

The pumping costs were calculated for each of the heat transfer fluids under the operating conditions and is shown in Table B.1-1. Note that the pumping costs of the aqueous solutions are quite negligible (less than \$1/year). Even for the higher viscosity fluids, such as silicone fluids, which have higher pumping requirements, the cost of the additional needed pumping is generally less than \$10/year.

The total cost of the system including the initial fluid fillup costs and the estimated pumping costs for each of the heat transfer fluids is also shown in Table B.1-1. Q2-1132 (a silicone fluid) with its higher initial and pumping costs has the highest estimated cost per year, with the glycols and water the least. Other silicone fluids have lower initial fillup costs such as SF-96(50), but their heat transfer is lower due to increased viscosity. The net effect is similar to the Q2-1132 fluid. Other costs due to the use of particular heat transfer fluids are possible. The effects of these additional costs are considered in the next section.

Overall Cost Effectiveness of the Heat Transfer Fluids

As outlined in the above sections, certain fluids have properties other than heat transfer and initial cost worth considering before a heat transfer fluid is chosen to replace water. Water systems require draindown for both freeze and boiling protection to reduce the risk to the collector. If these draindown systems were to fail, the additional cost of replacing part of the collector would be large. Corrosion also can be enhanced in aqueous solutions which can increase costs in the long run due to reduced performance and the need to replace the damaged equipment. Other fluids also require additional equipment which upon failure or lack of maintenance, would reduce the performance and increase the risk of additional costs to the users of such systems. Table B.1-2 lists the fluids and additional possible equipment needed which could increase the total costs attributable to the heat transfer fluids. The silicone fluids are shown to have less expected additional costs in Table B.1-2 than the aqueous solutions. If the expected yearly additional costs of the heat transfer fluids can be estimated, then the overall cost-effectiveness of the fluids can be determined.

In comparison to potable water circulated directly from the storage tank to the collector, any other heat transfer fluid is as cost effective when the cost to heat transfer ratio of each of the fluids for a particular system match, i.e.:

$$\frac{(C_t)_{water} + X_w}{Q_{water}} = \frac{(C_t)_{fluid} + X_f}{Q_{fluid}} \qquad \text{(B.1-9)}$$

Where:

$(C_t)_{water}$ = total cost of system including initial water fillup ($0) cost and water pumping costs

$(C_t)_{fluid}$ = total cost of system including initial fluid fillup cost and fluid pumping costs

Q_{water} = rate of heat transfer of potable water if water is circulated directly from the storage to the collector (i.e. no heat exchanger or traced tank penalty)

Q_{fluid} = rate of heat transfer of fluid

X_w = additional cost of the potable water system to make the other heat transfer fluid as cost effective as water ($/year) could be due to inadequate freeze or corrosion protection

X_f = additional cost of the fluid system to be expected ($/year) i.e. cost of inadequate freeze or corrosion protection, maintenance, additional fluid or inhibitor required, expansion tank, etc.

Table B.1-2

Possible Requirements for HTF's Affecting Yearly HT and Cost

Fluid	Inhibitors (to reduce corrosion) or Additional Fluid (due to decomposition) Required	Reserve Drain Down For: Freeze Protection	Boiling Protection	Expension Tank	Inert Blanketing to reduce decomposition upon exposure to air	Double Wall Required (due to toxicity)	Type of Maintainance Required	Comments
Potable water	no	yes	yes	no	no	no	cleaning, corrosion, freeze protection maintenance	
water used as HTF	yes if steel, Aluminum to be used	yes	yes	no	no	yes if inhibitors used	cleaning, inhibitor addition freeze protection	Reduced HT if double wall required
50% Ethylene Clycol	yes	no	yes	no	yes	yes	inhibitor addition, cleaning of tars, etc.	reduced HT if double wall used, or fluid decomposed
50% Propylene Glycol	yes	no	yes	no	yes	no*	inhibitor addition, cleaning of tars, etc.	reduced HT if fluid decomposed
Mobiltherm	no	no	no	yes	yes	probably*	cleaning,	reduced HT if fluid decomposed
Silicone fluids	no	no	no	no	yes	no*	minimal	
Dowtherm J	no	no	yes	yes	yes	yes	cleaning of tars etc.	reduced heat transfer if fluid decomposed
Sun temp	no	no	no	no	yes	no*	minimal	
Therminol 55	no	no	no	no	yes	probably*	cleaning	
Therminol 60	no	no	no	no	yes	probably*	cleaning	

*Further study on code requirements should be made before final determination of whether double walls are required for a particular heat transfer fluid

The additional water system cost needed to make the heat transfer fluid as cost effective becomes:

$$X_w = \frac{Q_{water}\,[(C_t) + X_f]}{Q_{fluid}} - (C_t)_{water} \qquad \text{(B.1-10)}$$

For each of the heat transfer fluids studied, all of the above quantities can be found easily except for X_f. The ratio Q_{water}/Q_{fluid} can be the ratio of the total integrated amount of heat transferred for the entire year, but for this Section, it was simply the ratio shown in Table B-1-1. This is assuming that the ratio of the performance of the two competing systems would be constant throughout the year. Figure B.1-14 and B.1-15 were developed for the heat transfer fluids using equation (B.1-10) and the results of Table B.1-1 for varying possible additional heat transfer fluids costs (X_f). Figure B.1-14 is for the traced tank case, while Figure B.1-15 corresponds to the double loop heat exchanger system of 40 m^2. For a particular additional heat transfer fluid cost, the aqueous solutions are still the most cost effective fluids requiring the least additional yearly expense of the potable water system to be as cost effective.

Since the silicone fluids can be expected to have much less yearly additional costs than the glycols, it is possible that these fluids can be as cost effective as the glycols. For example, if the silicone fluids can be expected to have additional costs of $10/year for the traced tank case, then the silicone fluids are as cost effective as the 50% ethylene glycol solutions if their additional costs are $76/year. Although this would require a high annual maintenance or inhibitor and fluid addition costs, if the ethylene glycol solutions were to increase corrosion significantly due to inadequate inhibitor level or operating under high temperature stagnation conditions, it is very possible that the silicone fluids could be as cost effective as the glycols.

Other low maintenance fluids such as Sun Temp could also be as cost effective as the glycols or water for small collector applications. For the traced tank case if the glycols annual expected costs were greater than approximately $45/year, then the Sun Temp fluid could be as cost effective.

From Figure B.1-14 and B.1-15 it is apparent that the propylene glycol solutions are nearly as cost effective as the ethylene glycols. Since the propylene glycol fluids are much less toxic, there appears to be little incentive to use the toxic ethylene glycol solutions for most applications.

Other fluids which require special precautions such as Dowtherm J, due to its high toxicity and low flash point, would have higher expected annual costs. Thus it would require higher glycol or water maintenance costs or inadequate freeze or corrosion protention costs to replace the aqueous solutions.

Also included in Figures B.1-14 and B.1-15 are the annual replacement costs of the collectors. Thus if the silicone fluids require additional yearly costs in excess of $25/year, then the whole collector using potable water could be replaced and still be more cost effective than the silicone system. For the large collector sizes, such as 40 m^2, the total collector replacement costs is much higher than the expected additional heat transfer fluid costs.

Thus Figure B.1-14 and B.1-15 allow comparison of the cost effectiveness, of each of the heat transfer fluids and their possibility as replacements for the use of water in solar energy installations, if the expected annual costs of these fluids can be determined. Although some fluids have major drawbacks, such as Dowtherm J with its low flash point and high toxicity, no fluids should be ruled out if proper system design can reduce these effects. Further investigation of the code requirements for the use of toxic and flammable fluids should be conducted.

At present, it is not felt that any one fluid deserves special attention as a possible fluid candidate for solar energy applications. Many manufacturers are beginning to market products which are directly applicable to solar energy uses and these fluids should be considered. It is felt that in the next few years many fluids will be weeded out which are not cost effective or which present hazardous conditions under normal operating conditions, leaving those fluids which meet the solar energy industries' needs.

Figure B.1-14

The Additional Yearly Costs of A Single Loop System With Water Circulating Directly from Storage Required to Make a Traced Tank Single Loop System as Cost Effective Versus the Expected Additional Yearly Costs of the Heat Transfer Fluid Used

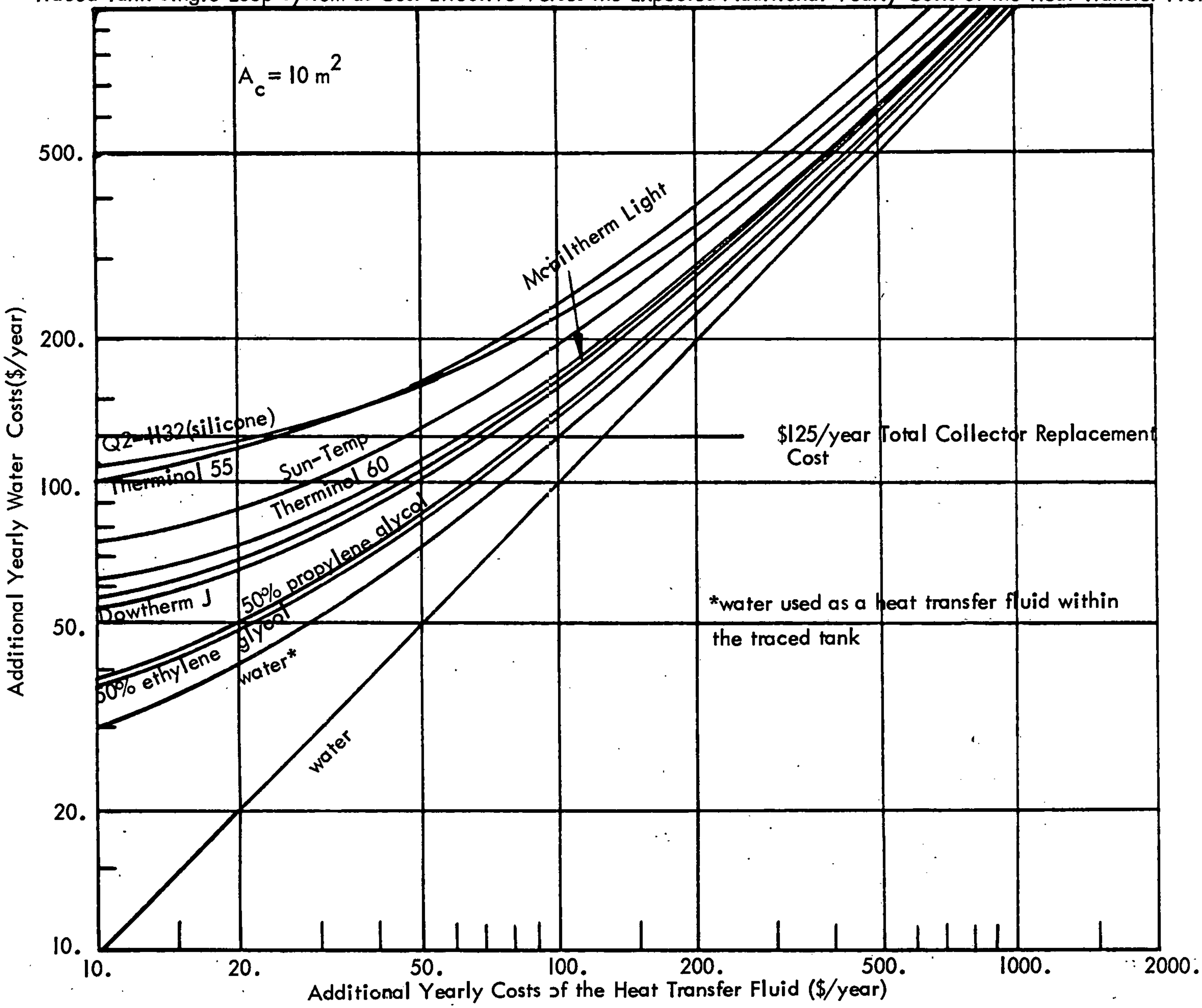
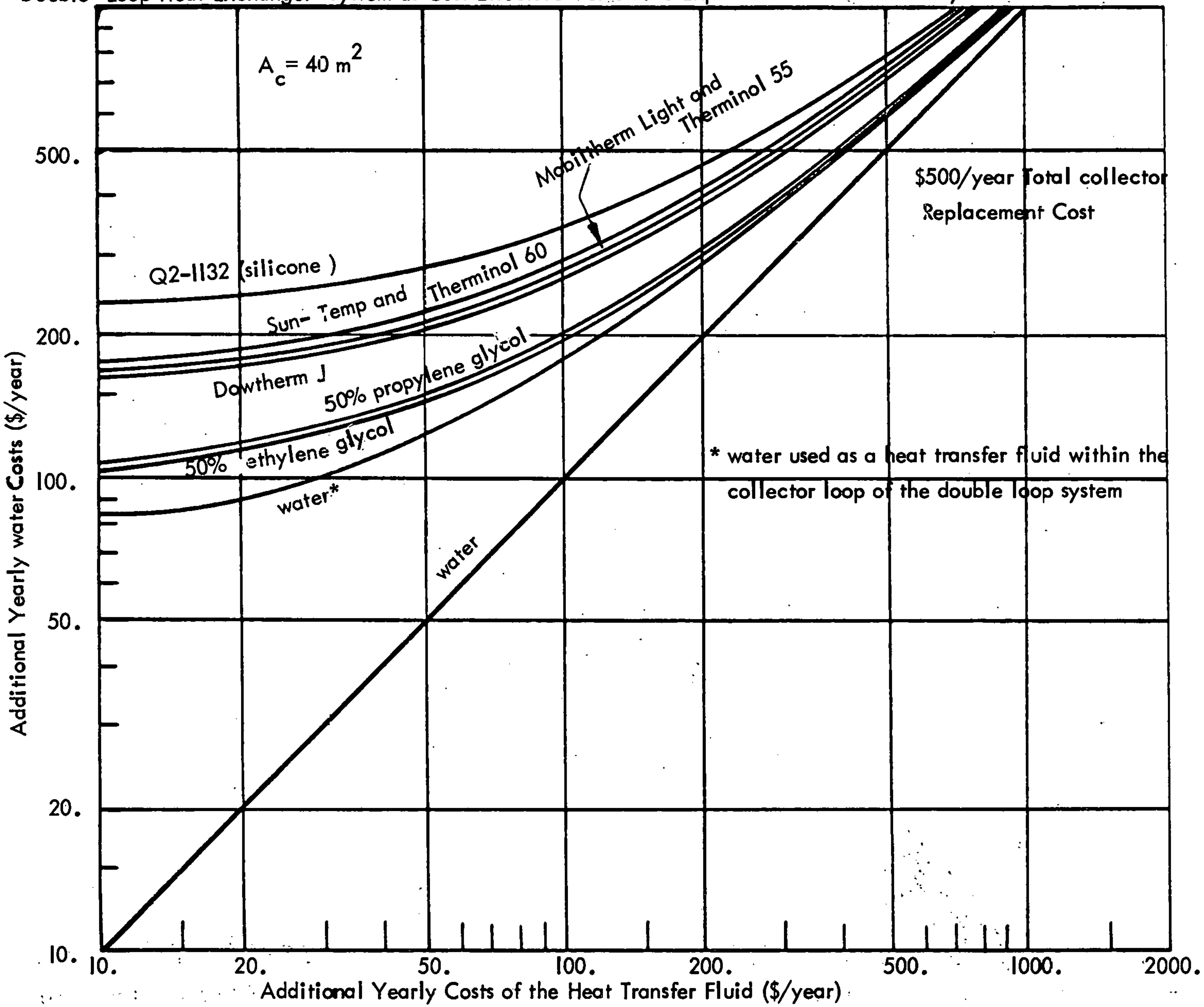


Figure B.1-15

The Additional Yearly Costs of A single Loop System With Water Circulating Directly from Storage Required to Make a Double Loop Heat Exchanger System as Cost Effective Versus The Expected Additional Yearly Costs of the Fluids Used



Heat transfer fluids were studied in depth to determine possible fluids (other than water) for use in flat plate collectors. Because water corrodes many metals and expands upon freezing, other fluids have been investigated for use in collector systems. A brief discussion follows of several candidates along with a description of the characteristics of water as a heat transfer medium. After this introduction, a comparison of the fluids for the following fluid properties is listed:

1. Thermophysical properties
2. Flowrate
3. Cost
4. Toxicity
5. Flammability
6. Corrosion
7. Vapor pressure
8. Freeze protection
9. Overall heat transfer and cost characteristics

Water

Water is a readily available fluid with good heat transfer properties (i.e., high specific heat and thermal conductivity and low viscosity). Its major drawbacks are a high freezing temperature, expansion upon freezing and its corrosive nature to common engineering materials (except copper). Also a low boiling point can cause large pressures within the collector system under zero flow conditions. Water has no adverse biological or environmental effects.

Ethylene Glycol

The heat transfer fluid most commonly in use, other than water, in flat plate collectors are water ethylene glycol solutions. There are common colorless, odorless anti-freeze solutions used in many other applications. Ethylene glycol is relatively inexpensive and available from many manufacturers. A sample of the manufacturers marketing ethylene glycol is shown in Table B-2.1. With inhibitors, aqueous ethylene glycol solutions can reduce the corrosive nature and freezing

Table B.2.1

A Sample of Manufacturers Marketing Glycol Fluids

Manufacturer	Specific Product: Ethylene Glycols	Specific Product: Propylene Glycols
Dow Chemical Corp.	Dowtherm SR-1	Dowfrost
Union Carbide Corp.	Thermofluid 17	UCAR Thermofluid 35
Jefferson Chemical Co.		
B.A.S.F. Wyandotte Corp.		

temperature of potable water. They are usually available in a wide range of concentrations and inhibitor levels. The thermal properties of these solutions (specific heat, thermal conductivity, and viscosity) are poorer than water. The boiling and flash point of aqueous ethylene glycol mixtures are low, and can be easily reached under zero flow conditions. Glycols can oxidize to organic acids (such as glycolic acids) when exposed to air near boiling temperatures. The inhibitors used are designed to neutralize these extremely corrosive acids. Periodic maintenance and addition of inhibitors must be done to use these fluids. Another major drawback to the use of ethylene glycol is its high toxicity.* Near potable water most plumbing codes require double walls to separate the two fluids.

Propylene Glycol

Propylene glycol has similar properties as compared with ethylene glycol except for higher viscosity and being less toxic. With inhibitors, propylene glycol can be used with most common engineering materials. Periodic maintenance and inhibitor addition must be performed to limit corrosion. Propylene glycol will also form acids at higher temperatures in oxygen-rich atmospheres. Because of its lower toxicity, propylene glycol has been widely used in the food industry. Most manufacturers who produce ethylene glycol also market propylene glycol as listed in Table B.2.1. The higher viscosity of propylene glycol reduces the heat transfer properties of aqueous propylene glycol mixtures compared to ethylene glycol.

Other Glycols

Other glycol solutions have been used as heat transfer fluids in industry applications. These include diethylene and triethylene glycol. With inhibitors, both of these fluids can be used with higher boiling points than ethylene glycol. The thermal properties of these aqueous solutions are similar to that of ethylene glycol at similar concentrations. The vapor pressure of each are slightly higher

*The U.S. Federal Food, Drug and Cosmetic Acts of 1938, a big step in the formation of the U.S. Food and Drug Administration (FDA), was prompted mainly by a poisoning episode in 1937 involving at least 73 deaths and perhaps as many as 107 deaths due to diethylene glycol contained in a drug known as "Elixir Sulfanilamide." (Campbell, 1938) Diethylene glycol is somewhat less toxic than ethylene glycol.

than that of ethylene glycol. The toxicity of these fluids are in between that of ethylene and propylene glycol. Cost of these glycols is slightly higher than that of ethylene and propylene glycols.

Other glycol heat transfer compounds include polyalkylene glycols such as Ucon fluids (by Union Carbide) and Jeffox (by Jefferson Chemical Co.). With inhibitors, the corrosion of common engineering materials is reduced. They are low in toxicity and are available in a wide range of viscosities. The price of these fluids applicable to heat transfer purposes is higher than for the other glycol compounds.

Petroleum (mineral) Oils

A class of heat transfer fluids used in industry applications is petroleum oils. They generally are fluids designed to operate at high temperatures with some able to offer lower temperature operation. As a group, they have poorer heat transfer than water with lower specific heat and thermal conductivity and higher viscosity. The flash point and boiling points lie below possible zero flow temperatures of a collector. Upon exposure to air at high temperatures, these fluids are subject to oxidation and cracking, forming tars and other by-products which would reduce collector performance and increase corrosion. The toxicity of these fluids is generally low and their prices are relatively low. Mobiltherm Light (by Mobil Oil Corporation) was chosen in this study as a good representative of this class of fluid for low temperature applications.

Silicone Fluids

Some flat plate collector installations have used silicone fluids as the heat transfer fluid. Among others they are produced by Dow Corning and General Electric. These fluids have low freezing and pour points, low vapor pressure, low general corrosion, long term stability, and low toxicity. Their major drawbacks are high viscosity causing poor heat transfer and requiring higher flowrates, and high cost. Also, leakage through fittings can create problems because silicone fluids have lower surface tension than aqueous solutions. Joints and fittings must be adequate to insure minimal leakage.

Other Fluids

Another possible fluid to be used in flat plate collectors is Dowtherm J manufactured by Dow Chemical Corporation. It is an alkylated aromatic compound with low viscosity, specific heat, and thermal conductivity. It is relatively inexpensive but has low flash and fire point. Oxidation at high temperatures upon exposure to air can lead to formation of insoluble materials and increased fluid viscosity. Also upon overheating, the flash point can be lowered and vapor pressure increased. Upon contamination by other fluids (such as water) corrosion can be enhanced (in the case of water, steel). The toxicity of Dowtherm J is high. Like aqueous ethylene glycol solutions double walls would most likely have to separate the potable water from the Dowtherm J.

Other possible fluids are manufactured by Monsanto Corporation. They include Therminol 44 (ester based), Therminol 55 (alkylated benzene), and Therminol 60 (hydrogenated aromatic). They have low specific heat and thermal conductivity and high viscosity with low freezing temperatures. The flash points of these fluids is at the upper range of possible zero flow temperatures. The costs of Therminol 44 and 60 are relatively high while Therminol 55 is much less costly.

Sun Temp fluid (a saturated hydrocarbon) marketed by Research Technology Corporation is another possible heat transfer fluid available to flat plate collector users. It has low specific heat and thermal conductivity and high viscosity. It has a low freezing temperature and a high boiling temperature. It is of low toxicity and low corrosivity with aluminum. It is relatively inexpensive with low vapor pressure. Because of its high viscosity, larger flow rates are required to produce turbulent flow and to increase the heat transfer.

Recently, inorganic aqueous salt solutions have been proposed as possible heat transfer fluids. According to Kauffman (1977) 23% sodium acetate and 38% sodium nitrate aqueous solutions with suitable additives are possible heat transfer fluids. The cost of these solutions is comparable to ethylene glycol, with low toxicity, and heat transfer properties similar to the glycols. Pumping costs would be low but like other aqueous solutions they are subject to boiling at lower temperatures with large vapor pressures. These fluids are still being investigated for solar energy applications.

In the above qualitative discussion of the heat transfer fluids, general characteristics of each fluid studied have been discussed. In the following sections, a more quantitative description of each fluid is presented. In order to choose a heat transfer fluid, the following characteristics of each fluid must be considered:

(1) Thermophysical properties
(2) Fluid flow properties
(3) Corrosion
(4) Toxicity
(5) Flammability
(6) Cost
(7) Vapor pressure
(8) Freeze protection
(9) Overall heat transfer and cost characteristics
(10) Maintenance requirements

In the following subsections, the fluids are compared to offer a quantitative description of probable performance in double loop heat exchanger collector systems. In some subsections, representative fluids were chosen for the comparison. For ethylene glycol also a 50% aqueous solution with inhibitors was used. Because most of the glycol properties are not drastically different from manufacturer to manufacturer, it was not felt necessary to compare each available ethylene or propylene glycol product in some subsections. A 50% solution for both ethylene and propylene glycols was chosen since this allows adequate freeze protection for most cases. For some applications, lower concentrations might be plausible, so these results will be slightly conservative for heat transfer and flowrate properties. Also, since the properties of diethylene and triethylene glycol are close to those of ethylene glycol, it was not felt necessary in some of the sections to compare these fluids.

Thermophysical properties

The thermophysical properties of the fluids were found from the manufacturers' specifications over the operating temperature range of flat plate collectors. For heat transfer, water is the best fluid. It has a high specific heat and thermal conductivity, and low viscosity. Water and the other heat transfer fluids are compared in Figures B.2.1 through B.2.4 for the following thermo-physical properties:

(1) Viscosity

(2) Specific heat

(3) Thermal conductivity

(4) Density

Generally, aqueous solutions (such as ethylene and propylene glycol) have thermophysical properties better than the rest of the heat transfer fluids with the exception of Dowtherm J. Dowtherm J has a lower viscosity than glycol solutions but also lower specific heat and thermal conductivity. Other simple comparisons of the heat transfer fluids can be made from figures B.2.1 through B.2.4.

In section B.1, heat transfer coefficients for the fluids are presented which will also show the applications of the thermophysical properties of each fluid under operating conditions. In the overall heat transfer and cost section, the penalty imposed on heat transfer by each fluid will be discussed.

Flowrate

One of the important parameters to be considered in selecting a heat transfer fluid is the operating pressure drop due to friction within the fluid channel. The pressure drop of the fluids was investigated for various flowrates and fluid channel sizes. From McAdams (1954) the pressure drop per tube length within tubes is:

$$\frac{\Delta P}{L} = \frac{f\, G''^2}{2\, D_i\, g\, \rho\, 144} \qquad \text{(psi/ft)} \qquad \text{(B.2.1)}$$

This is neglecting entrance and exit effects. This equation is applicable for collector, heat exchanger and traced tank tubes where:

f - friction factor

$f = 16/Re$ Laminar flow $Re < 2500$

$f = .0014 + .125/Re^{.32}$ $Re > 2500$ (for smooth-walled tubes)

Re - Reynolds No.

$Re = G'' D_i/\mu$

μ - fluid viscosity [lb/(ft hr)]

G" - mass flowrate through tube [lb/(ft^2 hr)

D_i - Inner tube diameter [ft]

g - acceleration due to gravity

$g = 4.18 * 10^8$ ft/hr^2

Figure B.2.1

Viscosity of the Heat Transfer Fluids Versus Temperature (°F)

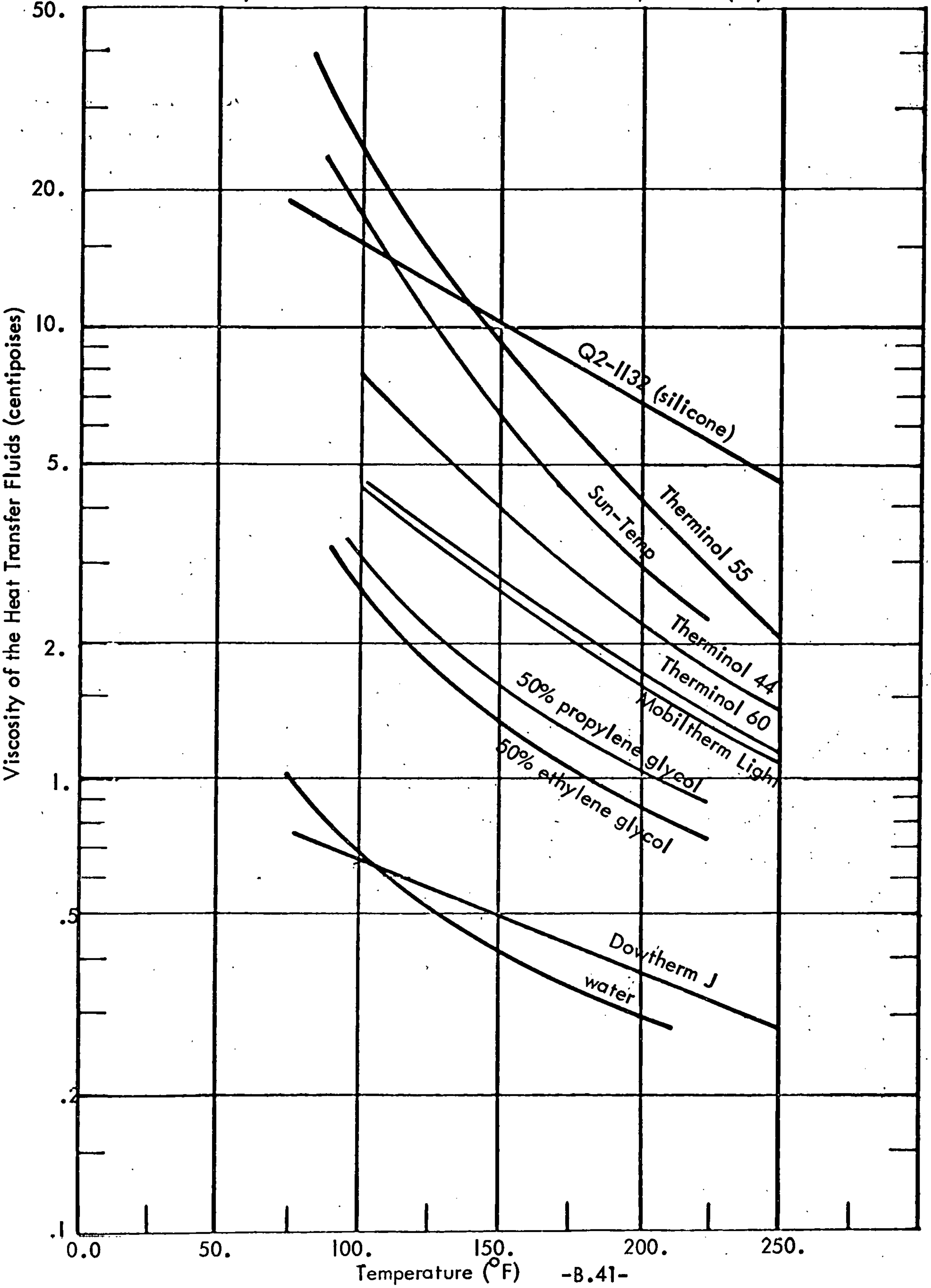


Figure B.2.2

Specific Heat of the Heat Transfer Fluids Versus Temperature (°F)

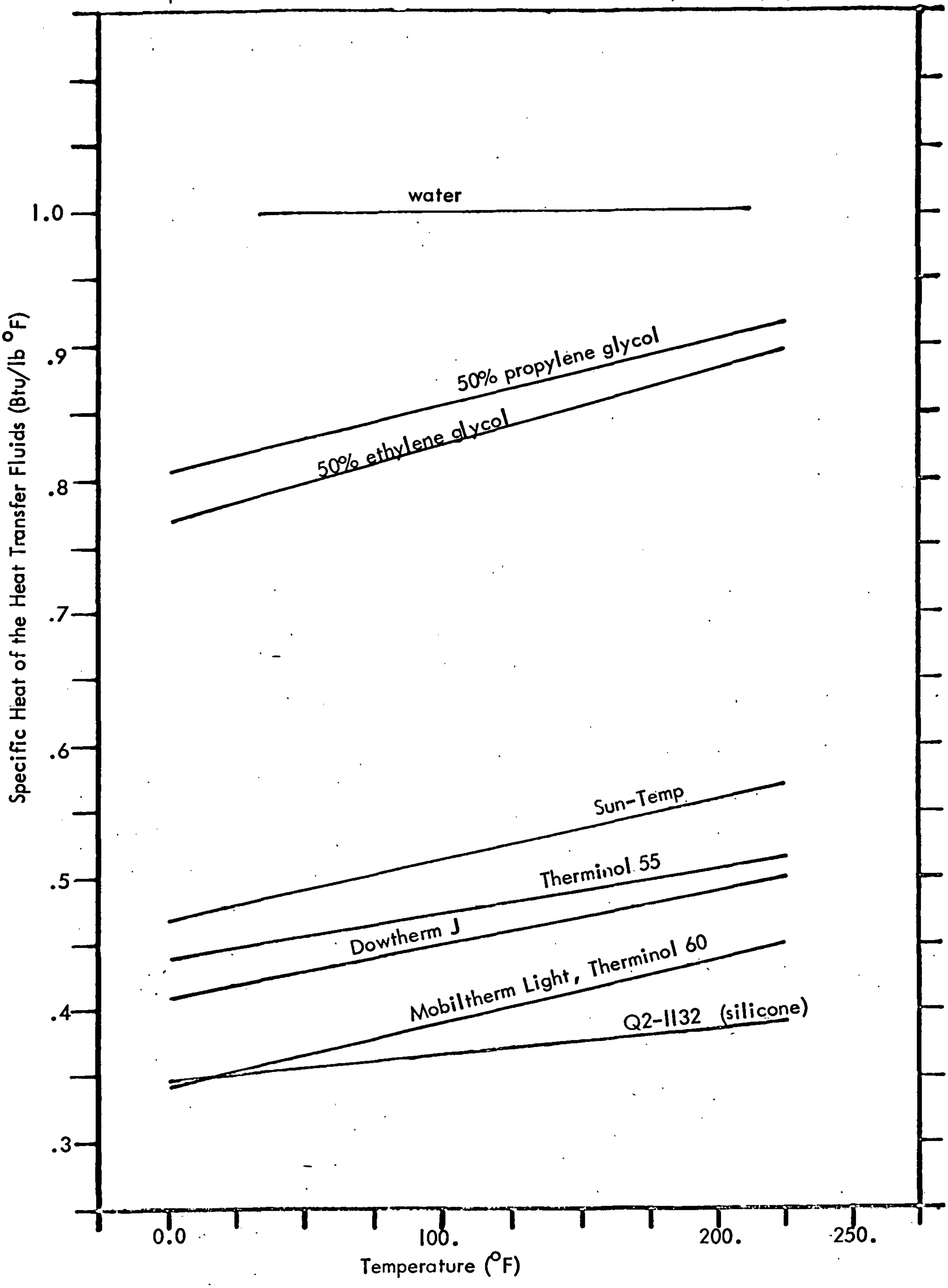


Figure B.2.3

Thermal Conductivity of the Heat Transfer Fluids Versus Temperature (°F)

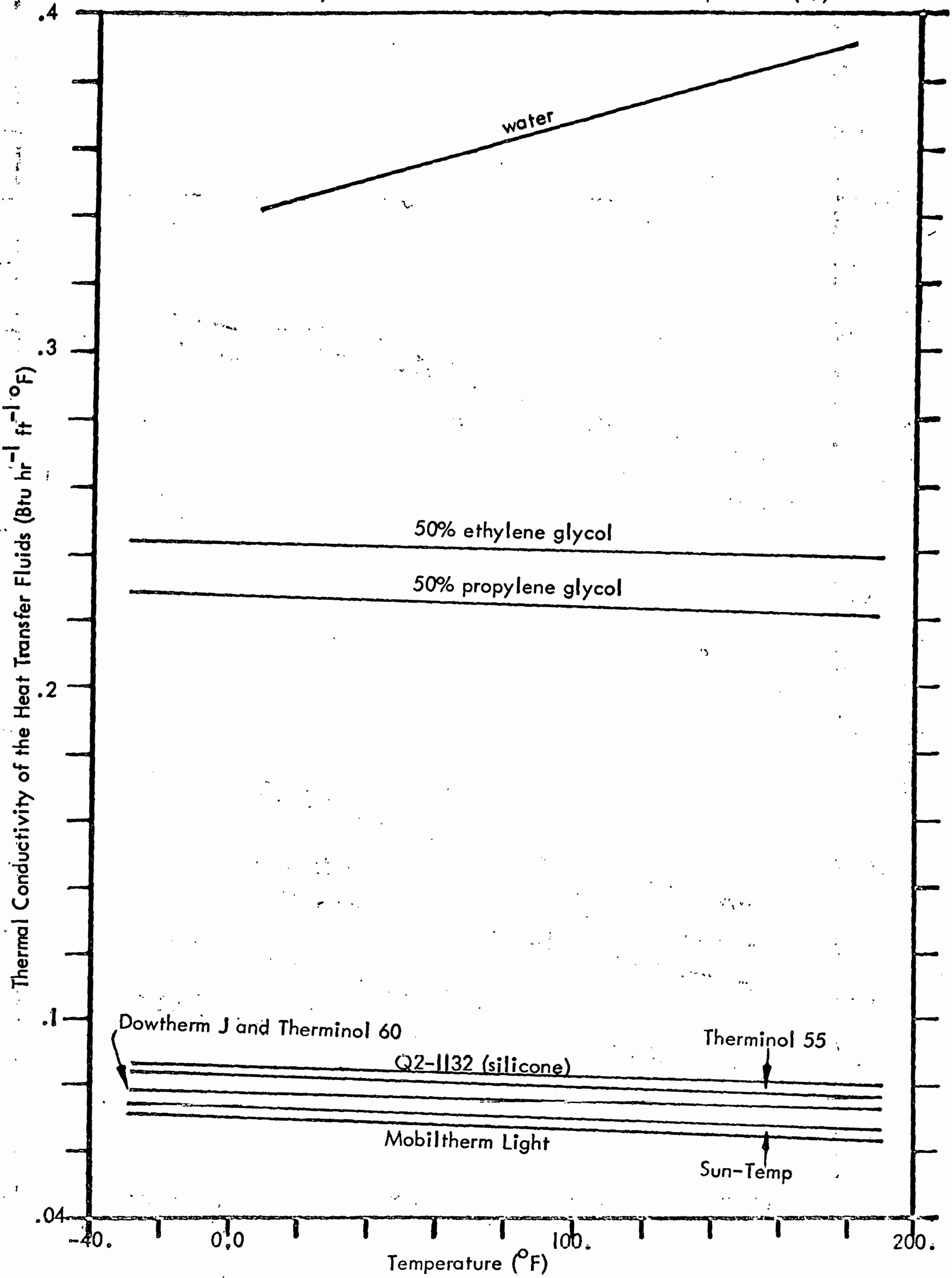
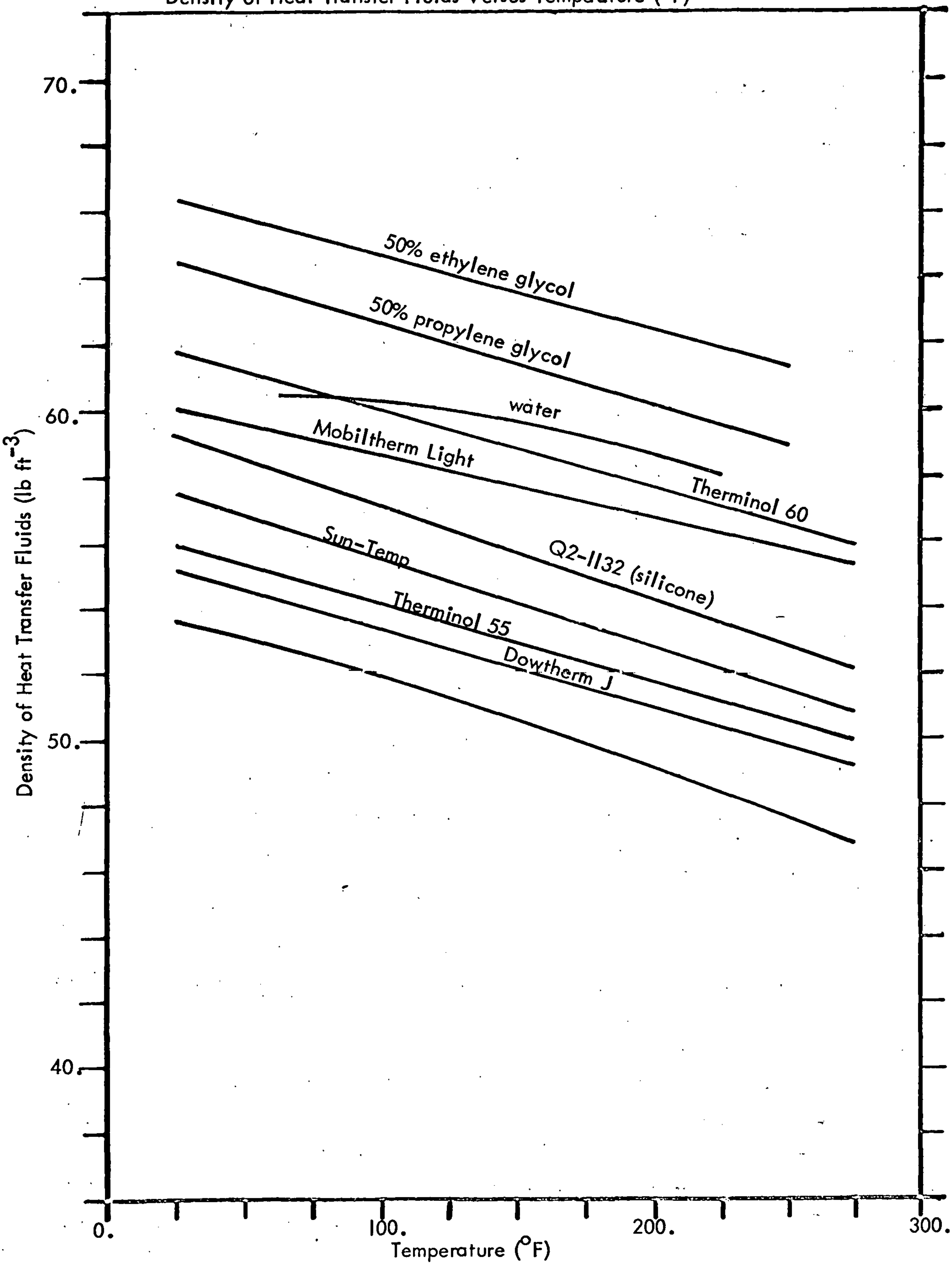


Figure B.2.4

Density of Heat Transfer Fluids Versus Temperature (°F)



ρ - density of fluid [lb/ft^3]

Equation (B.2.1) can be reduced to the Darcy equation in the form:

$$\frac{\Delta P}{L} = \frac{.0216\ f\ Q^2}{d^5} \qquad \text{(B.2.2)}$$

Where:

Q - flowrate [gal/min]

d - Tube inner diameter [in]

From Equation (B.2.2) it is easily seen that the tube size greatly affects the pressure drop within the tube. For some fluids, because of their higher pressure drops, larger tube sizes than needed for water must be used.

The pressure drop was determined for the representative fluids versus inner tube diameter and flowrate from Equation (B.2.1) and are shown in figures B.2.5 through B.2.13. In these figures the transition region ($2500 < Re < 7100$) was not included. Temperature correction factors for the laminar and turbulent regimes are included. From these figures, it is apparent that viscous fluids (such as silicone fluids or Sun Temp) have much higher pressure drops for the same operating conditions as compared with water. Also because of their increased viscosity, these fluids operate in the laminar regime over much larger ranges of flowrate than aqueous fluids.

The shell side pressure drop can be found from the following equation from <u>Process Heat Transfer</u>, D.Q. Kern (1950).

$$\Delta p = \frac{f\ G_{max}^2\ D_s\ (Nbaf + 1)}{2\ g\ \rho\ D_o\ 144.} \quad \text{(psi)} \qquad \text{(B.2.3)}$$

G_{max} - Maximum flowrate through shell side [lb/(ft^{2}hr)] See appendix B.1

D_s - Shell diameter [ft]

N_{baf} - Number of baffles within heat exchanger

D_o - Outer tube diameter [ft]

f - friction factor

$f = .0014 + 0.125/Re^{0.32}$

Re' - Shell side Reynolds Number

$Re' = G_{max} D_o / \mu$

Figure B.2.5.
Pressure Drop Versus Flowrate Within Tubes for Water

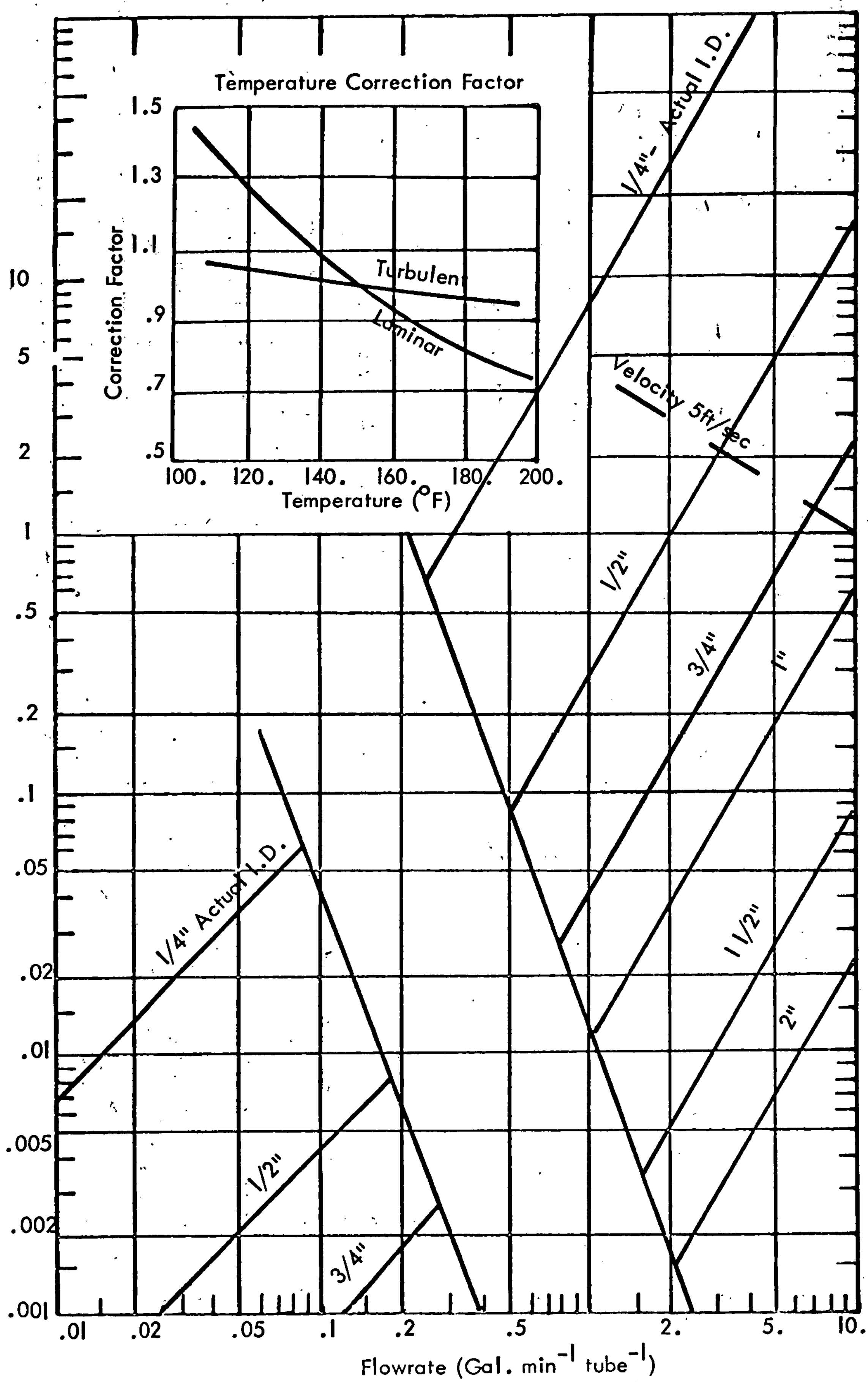


Figure B.2.6
Pressure Drop Versus Flowrate Within Tubes for 50% Ethylene Glycol

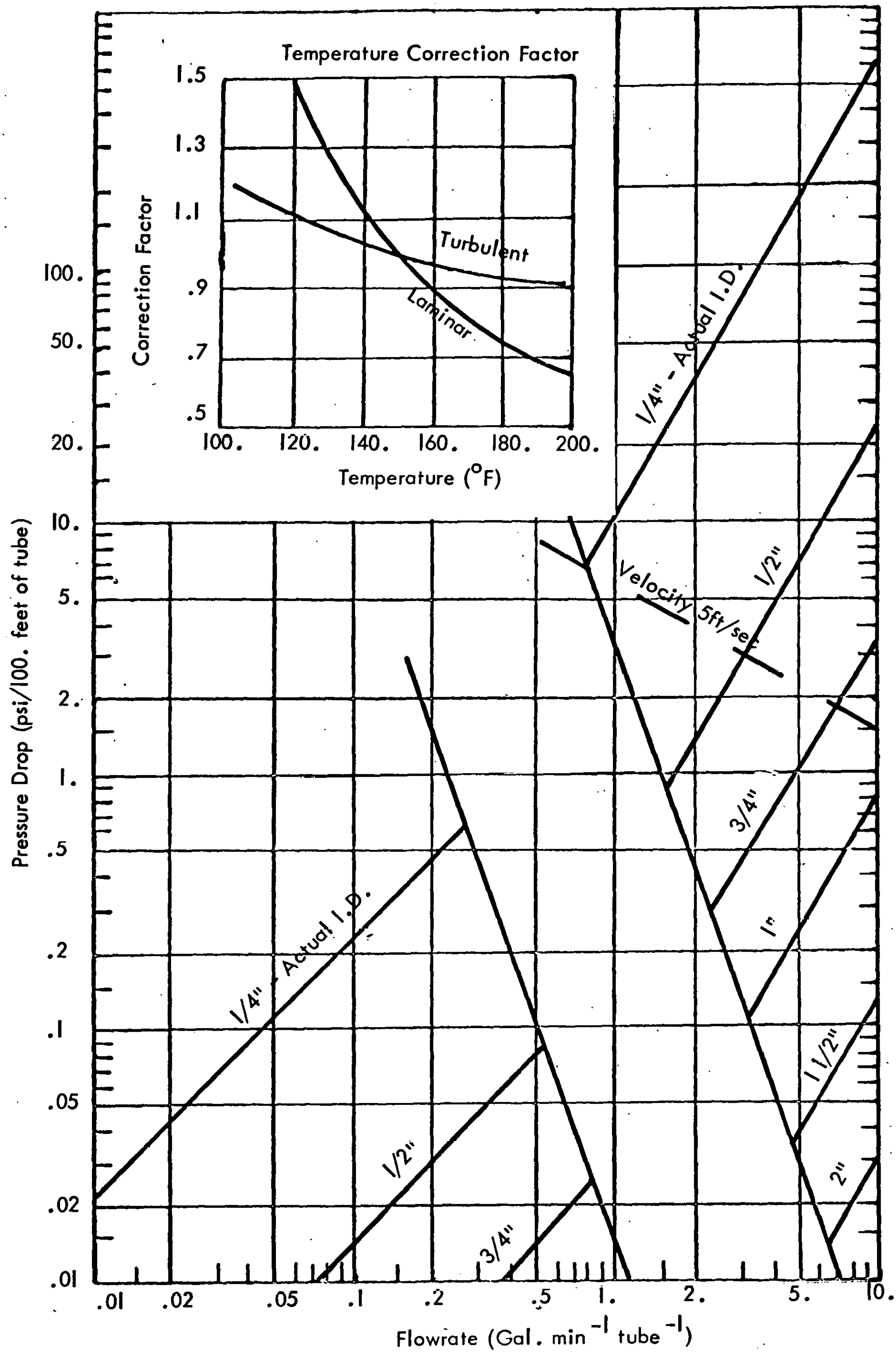


Figure B.2.7
Pressure Drop Versus Flowrate Within Tubes For 50% Propylene Glycol

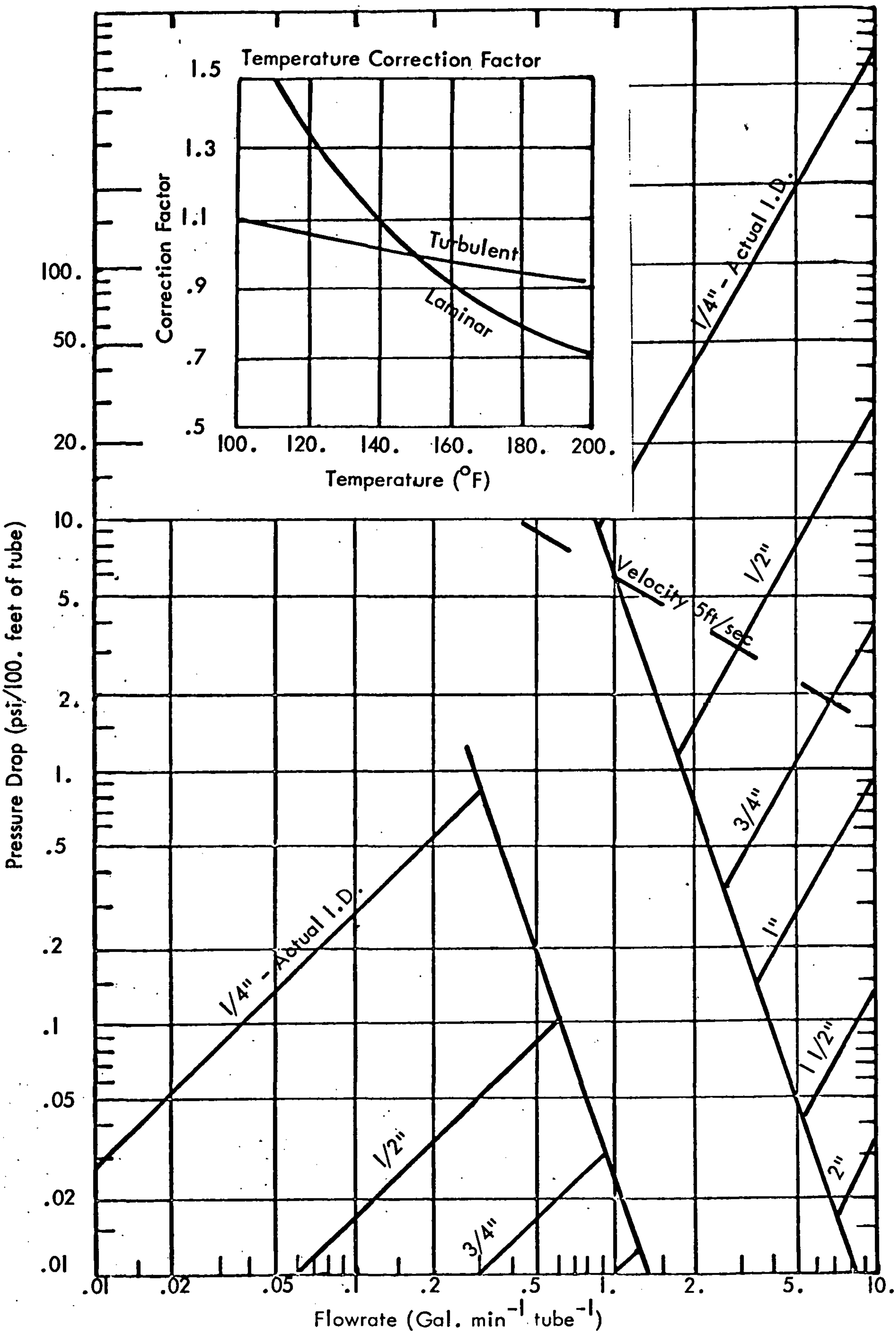


Figure B.2.8
Pressure Drop Versus Flowrate Within Tubes for Mobiltherm Light

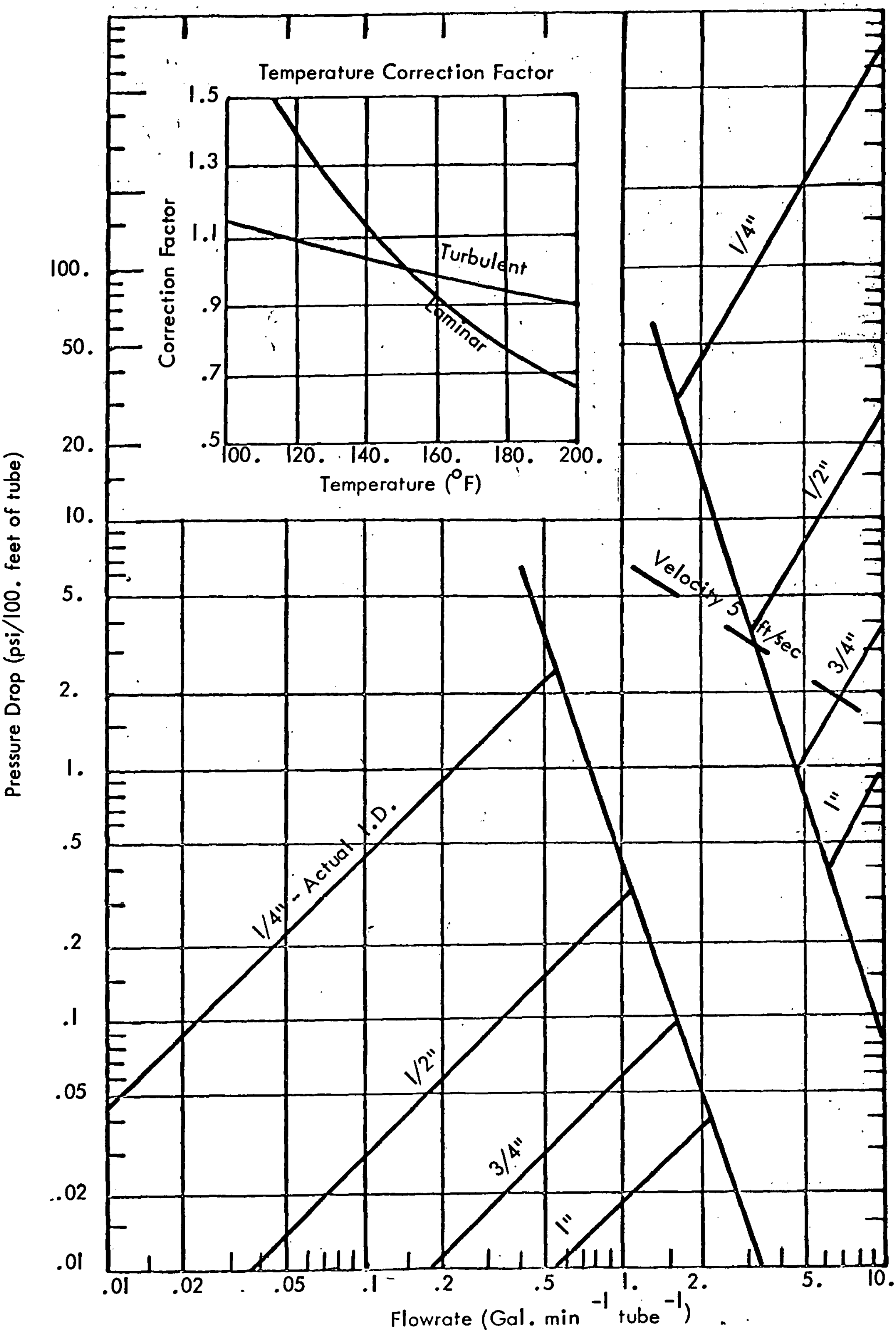


Figure B.2.9

Pressure Drop Versus Flowrate Within Tubes for Q2-1132 (Silicone)

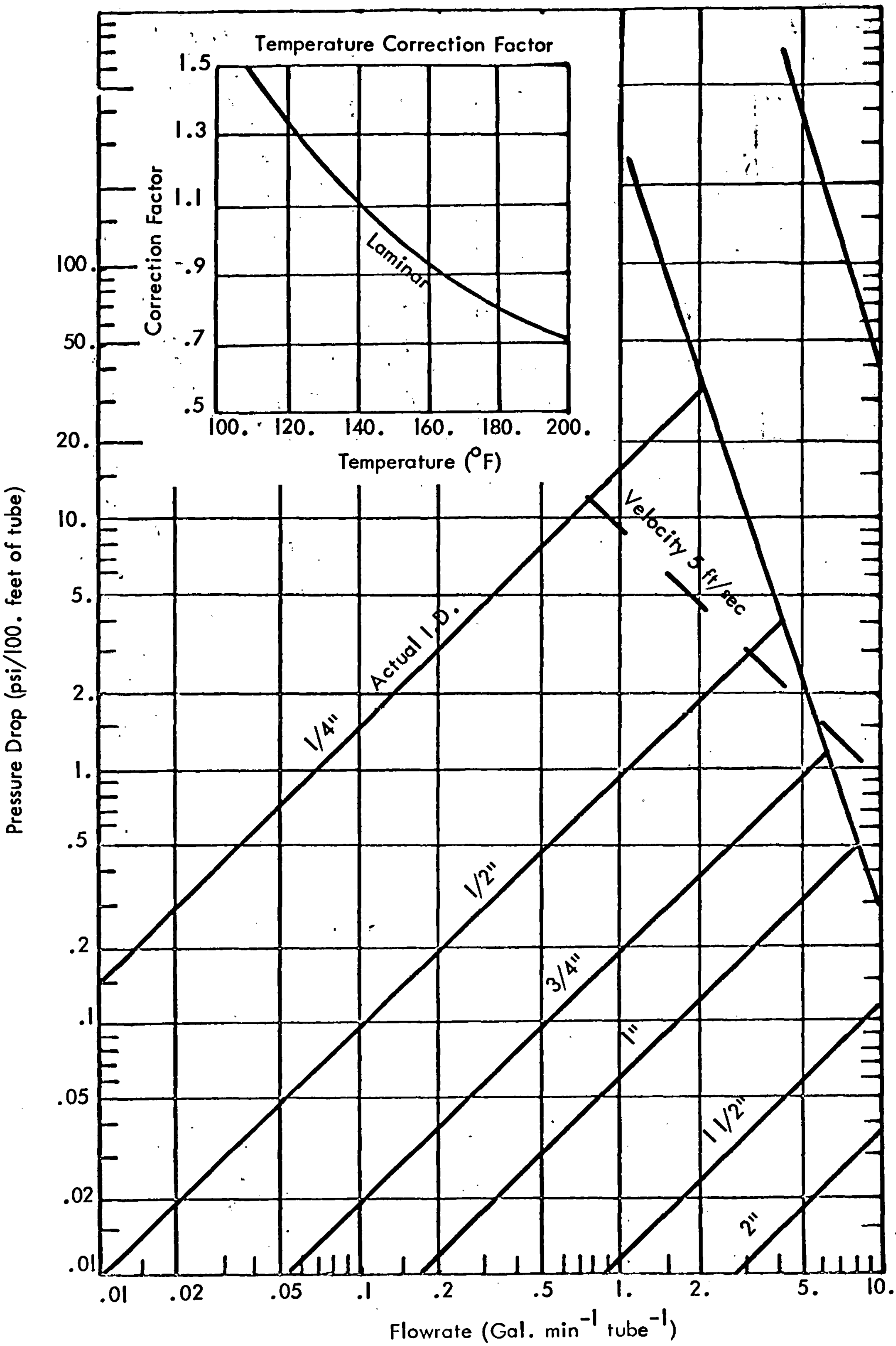


Figure B.2.10
Pressure Drop Versus Flowrate Within Tubes for Dowtherm J

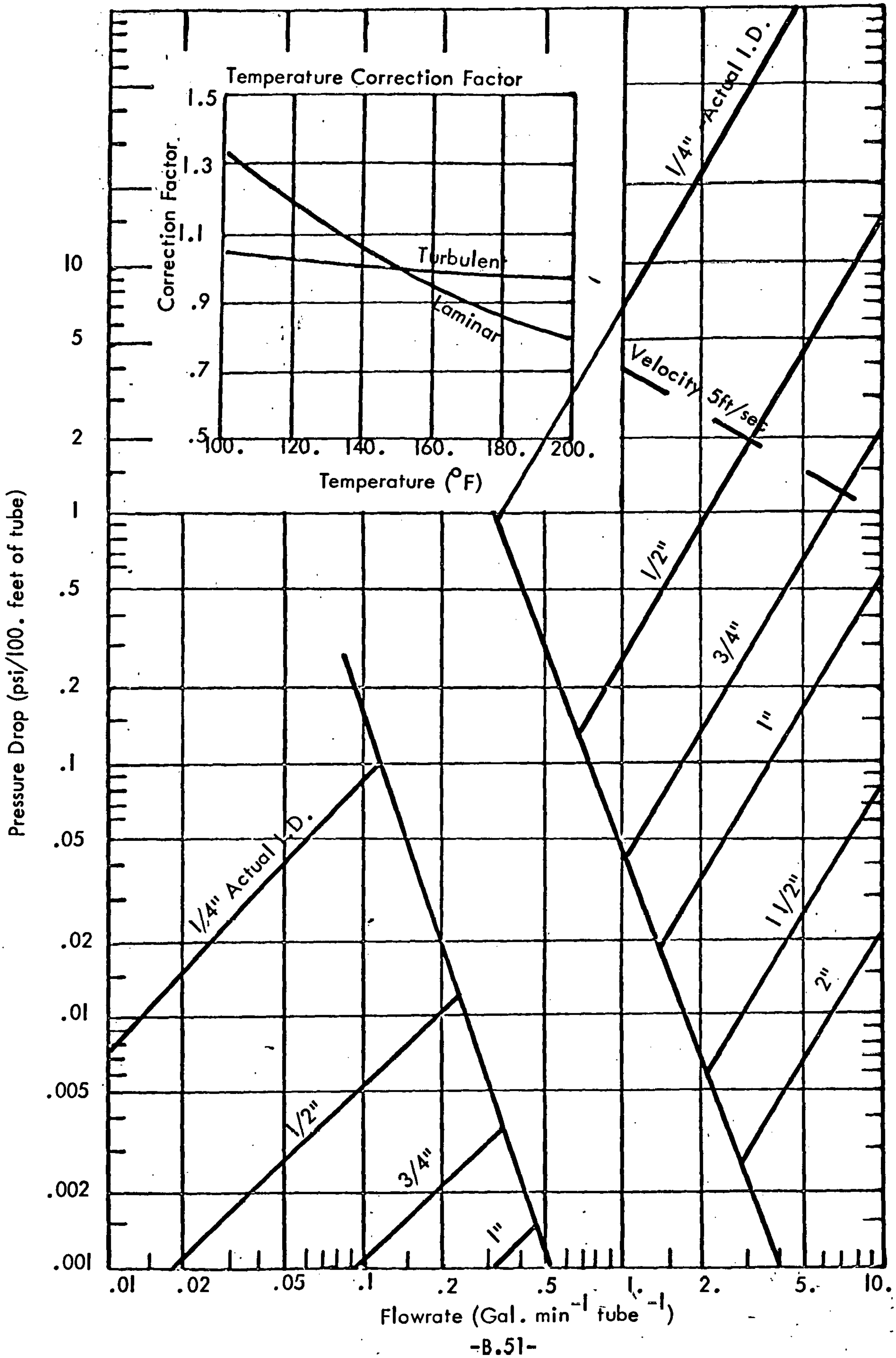


Figure B.2.11

Pressure Drop Versus Flowrate Within Tubes for Sun Temp

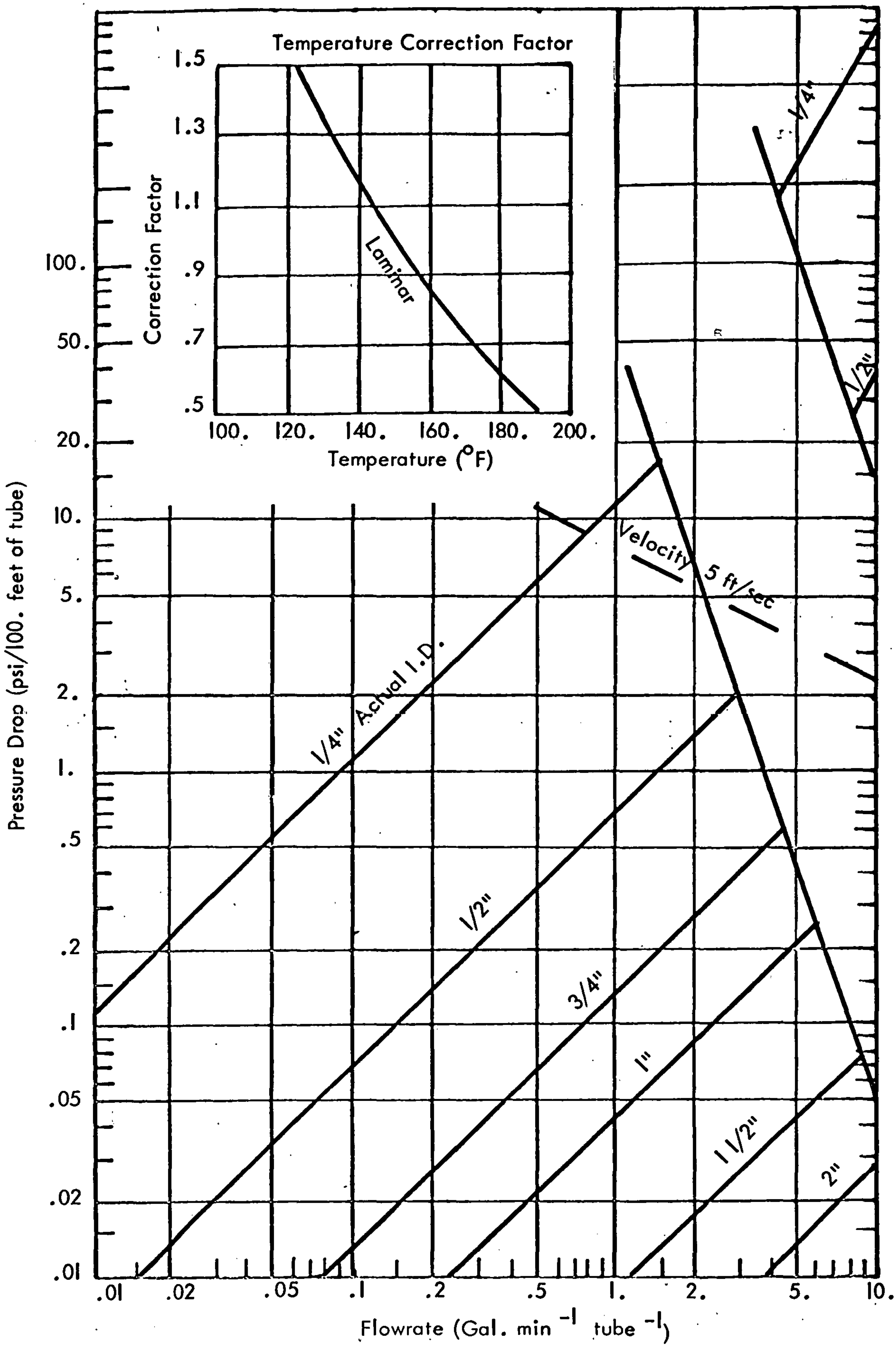


Figure B.2.12

Pressure Drop Versus Flowrate Within Tubes for Therminol 55

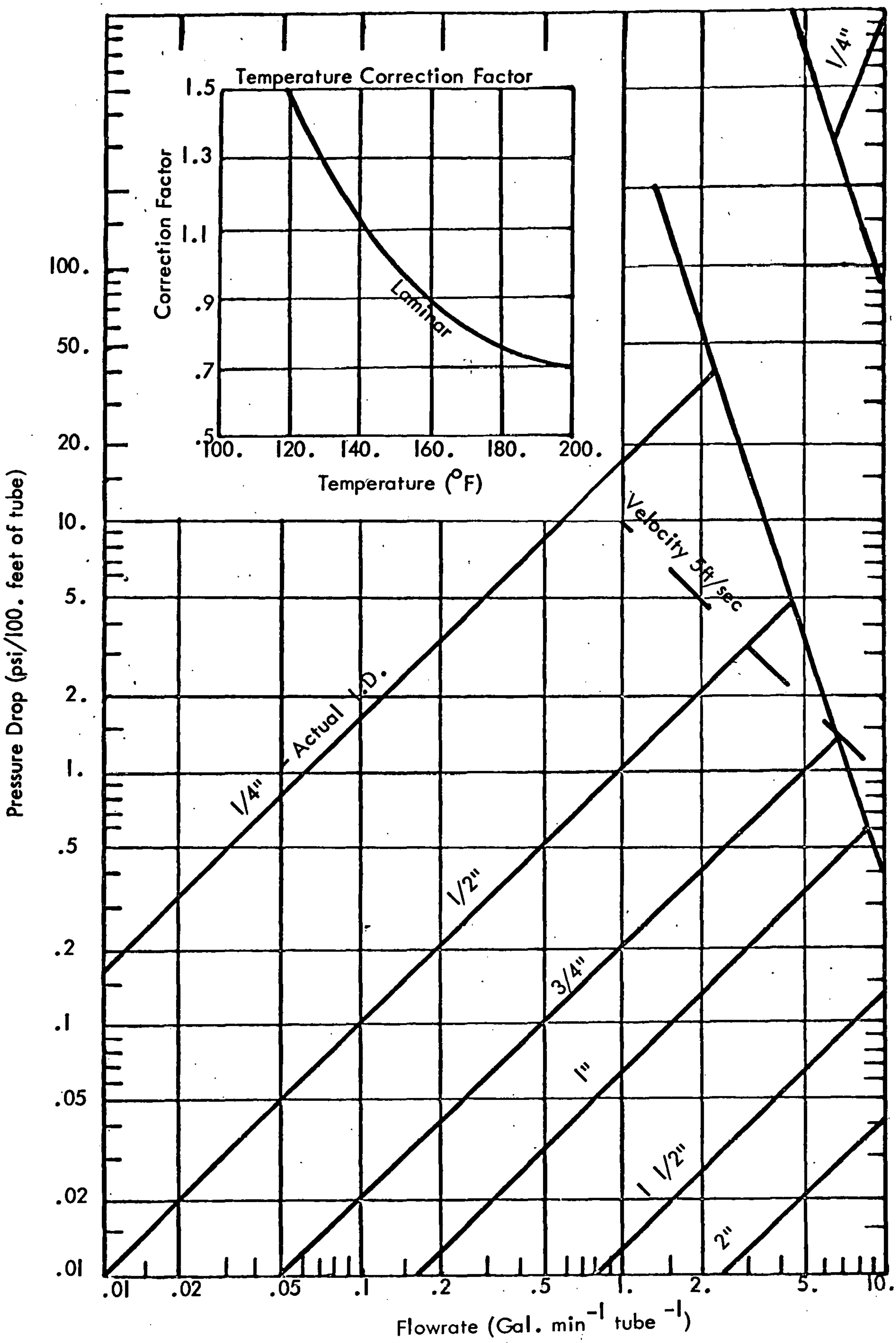
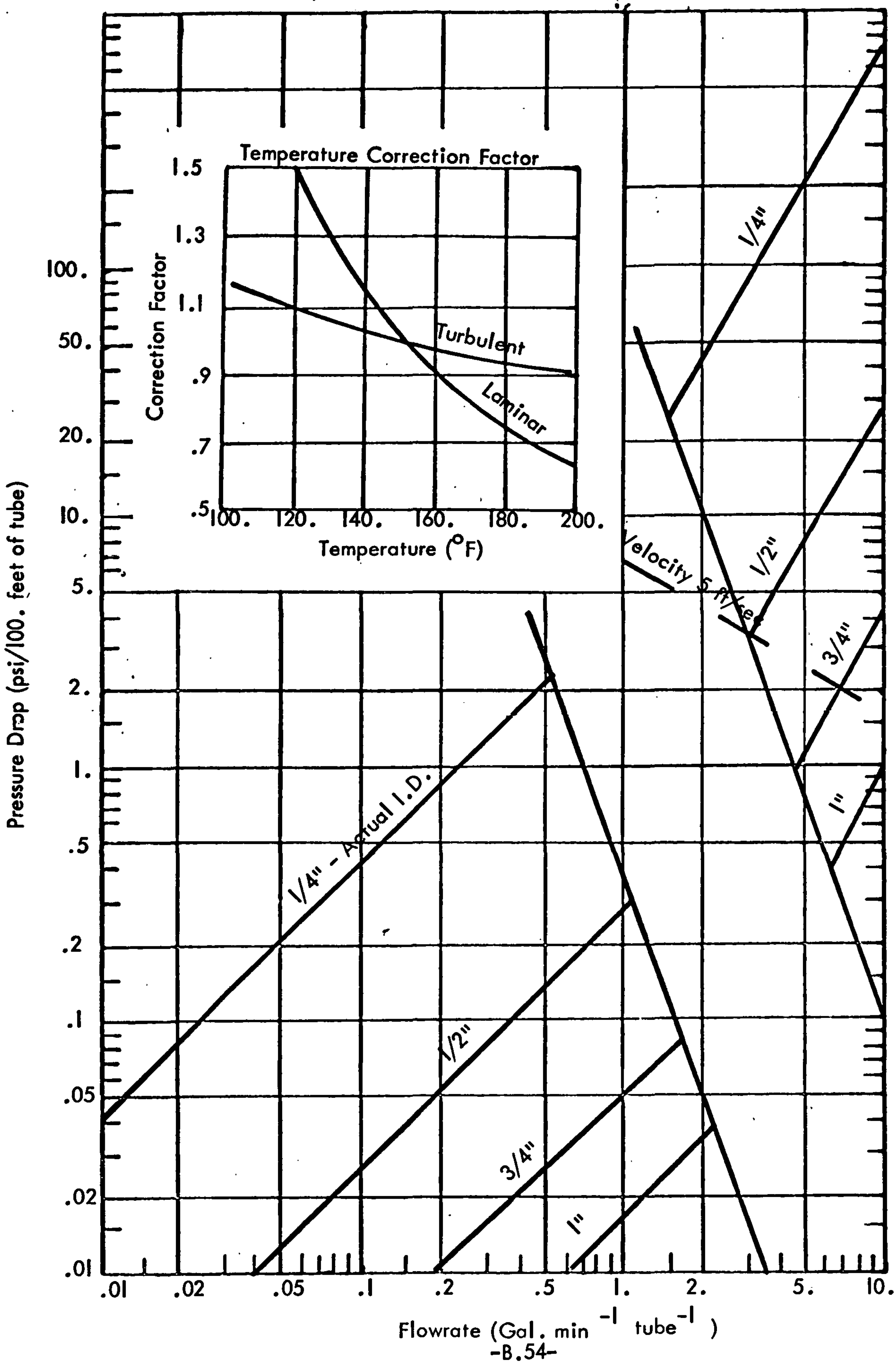


Figure B.2.13

Pressure Drop Versus Flowrate Within Tubes for Therminal 60



No figures were developed for the shell side pressure drop within a heat exchanger but the following example will suffice to show the use of this equation.

If Dowtherm J was circulated with a total shell flowrate of 50 gal/min through the shell side of the heat exchanger, with the mean operating temperature equal to 150^{o}F then:

$$\mu = \frac{1.16 \text{ lb}}{\text{Fthr}} \qquad \rho = 52 \frac{\text{lb}}{\text{Ft}^3} \qquad \text{and} \quad Q_s = 50 \frac{\text{gal}}{\text{min}}$$

The heat exchanger had a baffle spacing of 1 foot, 3/8" nominal O.D. tubing, tube length of 5ft and nominal shell diameter of 5". The tubes were spaced equilaterally with 10 tube rows. Then:

$$A_{min} = .0104 \text{ Ft}^2 \qquad G_{max} = 1.8226 \times 10^5 \frac{\text{lb}}{\text{Ft}^2\text{hr}} \qquad f = .0089$$

Then the total pressure drop becomes $\Delta p = .0024$ psi

This is a quite minimal pressure drop and is less than that exhibited within the collector.

Cost

In some applications, more expensive fluids can be more competitive with their less costly competitors. In order to determine the relative cost of a heat transfer fluid, the volume of fluid required for a particular application must be known. For a flat plate collector, the volume of fluid required for tubes bonded to the collector surface with fixed spacing (W) can easily be shown to be:

$$\text{Volume of fluid} = \frac{5.8748 \, D_i^2 \, Ac}{W} \quad \text{[Gallons]} \qquad (B.2.4)$$

A_c - Collector area [ft^2]

D_i - Actual inner tube diameter [ft]

W - Tube spacing [ft]

For any other tubing (including that for the traced tank and the heat exchanger) the volume of fluid required is:

$$\text{Volume of fluid} = 5.8748\, D_i^2 L \qquad (B.2.5)$$

L - length of tubing [ft]

It has been assumed in this study that for small systems the volume of fluid within the heat exchanger on the shell side is 1 gallon. As an example, Figure B.2.14 shows the volume of fluid required versus collector size with an inner tube diameter of .569 inches, and a tube spacing of .33 ft. This system uses a heat exchanger with 50 ft. of 2 inch pipe connecting the collector to the storage system. For a collector of 500 ft^2, Figure B.2.14 shows the amount of fluid required is approximately 30 gallons.

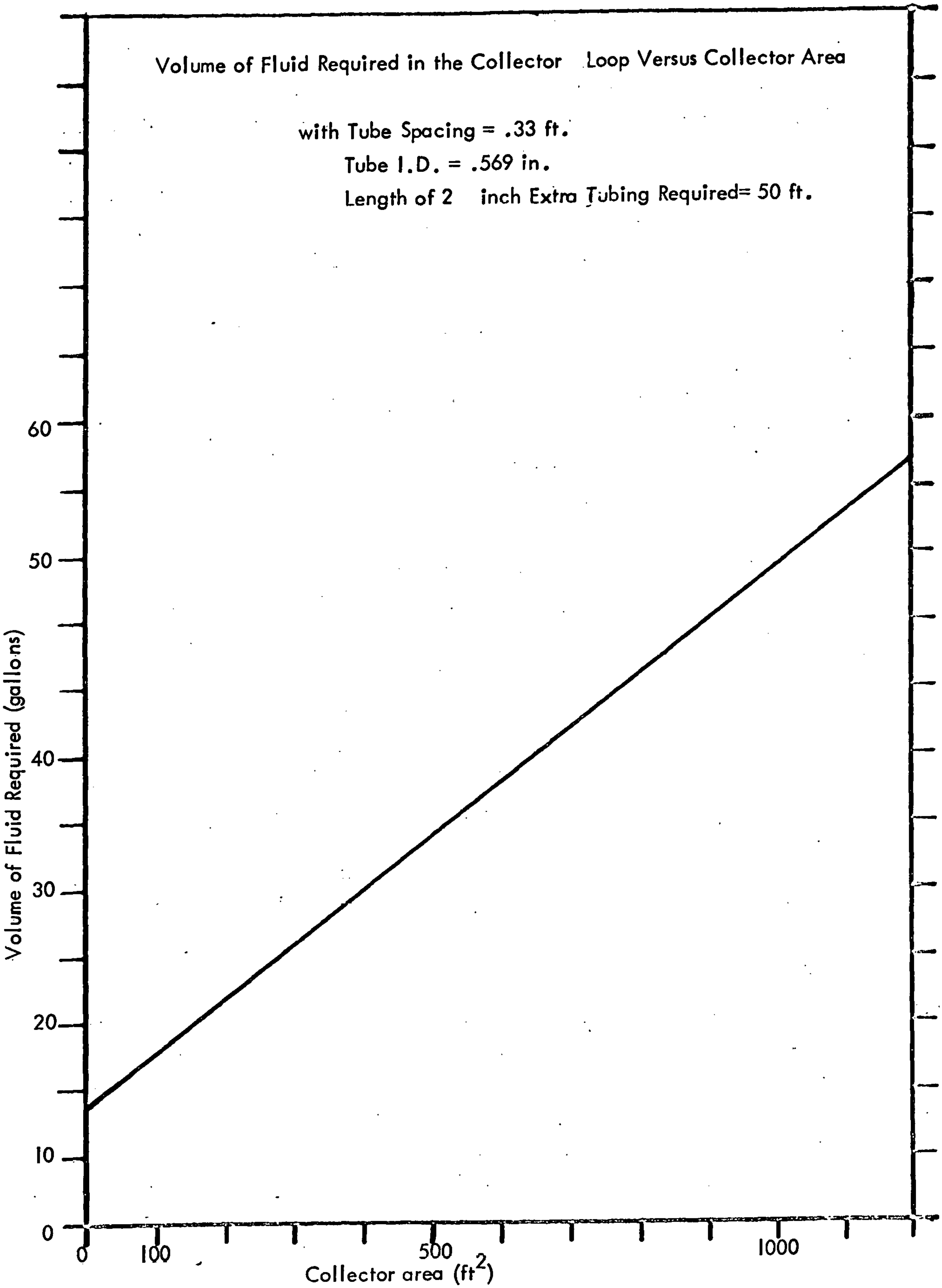
For some applications (such as domestic hot water heating) the amount of heat transfer fluid required will be small since the collector area needed is small. Using a traced tank system (see Section 2.2) more costly fluids can be used if their other properties are desirable.

The following Table (Table B.2.2) shows the current costs of many of the fluids in single 55 gallon drum quantities. Note for the glycol solutions the final costs will generally be lower since a 100% solution of the glycols is not necessary. Thus Mobiltherm light and the glycols are the least expensive heat transfer fluid for initial installation with the silicone fluids the most expensive.

There are other fluid costs besides those of the initial fillup. If periodic maintenance and inhibitor addition is needed, this can add to the total cost of the fluid over a specific time period. Also, if corrosion and freeze protection is inadequate leading to collector failure, this additional cost must be considered. Also, more viscous fluids will require higher flowrates and increased pumping costs. Thus the total fluid investment over a given time period is equal to the sum of the initial fluid cost plus any additional costs of added fluid or inhibitor, increased pumping costs, maintenance, cost of replaced parts needed because of inadequate freezing or corrosion protection, or cost of reserve draindown or expansion tanks needed by some fluids.

These added costs will be further developed in the Overall Heat Transfer and Cost Section where the effects of those factors will be considered on the optimum performance of the double loop system.

Figure B.2.14



Volume of Fluid Required in the Collector Loop Versus Collector Area
with Tube Spacing = .33 ft.
Tube I.D. = .569 in.
Length of 2 inch Extra Tubing Required= 50 ft.
Volume of Fluid Required (gallons)
60
50
40
30
20
10
0
0
100
500
1000
Collector area (ft2)

TABLE B.2.2

INITIAL FILLUP COST OF HEAT TRANSFER FLUIDS

FLUID	COST / GALLON (Single 55 gallon drum quantities)	MANUFACTURER
Water	---	--
100% Ethylene Glycol	2.56	Union Carbide
100% Propylene Glycol	2.45	Union Carbide
100% Diethylene Glycol	2.82	Union Carbide
100% Triethylene Glycol	3.70	Union Carbide
100% UCAR Thermofluid (Ethylene glycol & inhibitors)	3.81	Union Carbide
100% UCAR Foodfreeze (Propylene Glycol & inhibitors)	3.63	Union Carbide
100% Dowtherm SR-1 (Ethylene Glycol & inhibitors)	3.65	Dow Chemical
100% Dow frost (Propylene Glycol & inhibitors)	3.45	Dow Chemical
Mobiltherm Light	1.29	Mobil Oil
SF-96(50) (Silicone)	14.00	General Electric
Q2-1132 (Silicone)	23.00	Dow Corning
Dowtherm J	4.5	Dow Chemical
Therminol 44	7.65	Monsanto
55	2.80	Monsanto
60	6.80	Monsanto
Suntemp	3.50	Resource Technology Corporation

Toxicity

The toxicity of a heat transfer fluid can greatly affect the design and operation of a double loop flat plate collector system. Most plumbing codes require double walls or vented surfaces to separate a toxic fluid from potable water supplies. Also the possibility of poisonous fumes escaping from the heat transfer fluid must be considered. These problems require different heat exchangers which will transfer heat less optimally than ones which operate without a toxic fluid. The following discussion describes the toxicity of the heat transfer fluids studied. The information was obtained from the manufacturers.

In a discussion of toxicity the following definitions are useful (from United States Codes Annotated, 1974):

Hazardous substance - Any substance or mixture of substances which:

(1) is toxic
(2) is corrosive (will cause destruction of living tissue by chemical action)
(3) is an irritant
(4) is strong sensitizer
(5) is flammable or combustible
(6) generates pressure through decomposition, heat, or other means

Toxic - Any substance which has the capacity to produce personal injury or illness to man through, ingestion, inhalation, or absorption through any body surface.

Highly Toxic - any substance which produces death within 14 days in half or more than half of a group of ten or more laboratory white rats each weighing between 200 and 300 grams at a single dose of 50 milligrams or less per kilogram of body weight when orally administered, or when inhaled continuously for a period of 1 hour or less at an atmospheric concentration of 200 parts per million by volume or less of gas or vapor, or 2 milligram per liter by volume or less of dust or mist.

LD_{50} - Quantity of chemical substance which kills 50% of dosed animals within 14 days. Dosage is expressed in grams or milliliters per Kilogram of body weight.

Single dose (acute) oral LD_{50} - Quantity of substance which kills 50% of dosed animals within 14 days when administered orally in a single dose.

Because the primary hazard of the heat transfer fluids is the possibility of accidental ingestion of the heat transfer fluid due to leakage into a potable water supply, acute oral toxicity is the primary concern in this section. Table B.2.3 lists the LD_{50} values for selected fluids for acute oral toxicity. From this table it is apparent that no substance is highly toxic according to the above definition, but several are still quite toxic. From Table B.2.3 it can be seen that Dowtherm J is the most toxic fluid listed with the ethylene glycol mixtures second. The least toxic fluids are silicone fluids, Sun-Temp and propylene glycol. Propylene glycol is routinely used in the food industry.

In deciding whether a toxic fluid should be used the other fluid properties and cost should be considered.

Flammability

The possibility of the heat transfer fluid being a fire hazard was considered. In a discussion of the flammability of a heat transfer fluid the following definitions are useful:

Boiling point - the temperature at which the vapor pressure of a liquid equals the absolute external pressure at the liquid vapor interface.

Flash point - the lowest temperature at which a combustible vapor above a liquid ignites and burns when ignited momentarily in air.

Fire point - lowest temperature at which combustible vapors flash and burn continuously.

Self-ignition point - temperature at which self-sustained ignition and combustion in ordinary air takes place independent of a heating source.

Extremely flammable - any substance which has a flash point at or below $20^{o}F$ as determined by the TOCT (Togliabue Open Cup Tester)

Flammable - any substance which has a flash point between $20^{o}F$ and $80^{o}F$ as determined by the TOCT

Combustible - any substance which has a flash point between $80^{o}F$ and $150^{o}F$ as determined by the TOCT.

TABLE B.2.3

Acute Oral Toxicities of Heat Transfer Fluids

FLUID	LD_{50} (g/kg)
Water	--
100% Ethylene Glycol (No inhibitors)	8
100% Propylene Glycol (No inhibitors)	34.6
100% Diethylene Glycol (No inhibitors)	30.
100% Triethylene Glycol (No inhibitors)	30.
100% Dowtherm SR-1	4
Mobiltherm Light	20.
SF-96(50)(Silicone)	50 g/kg
Q2-1132 (Silicone)	50 g/kg
Dowtherm J	1.1
Therminol 44	13.5
Therminol 55	15.8
Therminol 60	13.0
Suntemp	No test information available

Table B.2.4 lists the fluids studied and their boiling or flash points which ever were supplied by the manufacturer. From Table B.2.4 it is apparent that none of the fluids are extremely flammable or flammable. Only Dowtherm J is combustible with a flash point of 145°F. With the exception of the silicone fluids, Sun Temp and Therminol 44, most fluids have flash points below possible stagnation temperatures.

The HUD minimum property standards for FHA eligibility according to Kauffman (1977) precludes the use of fluids whose flash points are not at least 100°F higher than the highest temperature to which they might be exposed. Thus the use of fluids with low flash points is limited unless adequate safeguards limit the exposure of these heat transfer fluids to high temperatures and exposure to the atmosphere.

Corrosion

Butt and Popplewell (1970) state that general corrosion is usually slow in most systems, but that localized corrosion is the prime cause for corrosion problems in flat plate collector systems. According to Popplewell (1975) there are four basic types of localized internal corrosion that can be affected by the heat transfer fluid. These are:

(1) Galvanic
(2) Pitting
(3) Crevice
(4) Erosion - corrosion

Galvanic corrosion occurs when two dissimilar metals are joined together in an electrolyte (a fluid which conducts electricity such as aqueous solutions). Depending on the type of metals in contact, corrosion can occur quite rapidly at the interface. To avoid this problem, insulating couplings should separate any dissimilar metals in an electrolytic solution according to Popplewell (1975).

Pitting corrosion is characterized by rapid localized metal loss which leads to perforation of metals in uninhibited aqueous solutions. For aluminum, the presence of chloride ions in the heat transfer fluid will aggravate this type of corrosion. Also, metal ions (copper and iron) will cause pitting to begin on aluminum surfaces

TABLE B.2.4

FLAMMABILITY OF HEAT TRANSFER FLUIDS

FLUID	BOILING POINT	FLASH POINT (^{o}F) (Cleveland Open Cup)
Water	212	--
100% Ethylene Glycol	388	240
50%	225	
100% Propylene Glycol	370	225
100% Diethylene Glycol	475	290
100% Triethylene Glycol	550	330
100% Downtherm SR-1	325	240
50% Dowtherm SR-1	230	
100% Dowfrost	--	214
Mobiltherm Light	250	
SF-96(50)		600
Q2-1132		450
Dowtherm J		145
Therminol 44	425	405
55	600	355
60	650	310
Suntemp	500	

according to Popplewell. Steel is also susceptible to pitting corrosion in aqueous heat transfer fluids with chloride ions.

Crevice corrosion, according to Popplewell, is similar to pitting corrosion in that rapid metal loss occurs in localized areas (inside crevices). Crevices can occur in blockages within internal channels or gaskets through which the heat transfer fluid passes. Aluminum and carbon steel are more susceptible to this form of corrosion in aqueous environments. This problem can be reduced by eliminating possible crevices by proper design.

Erosion corrosion is caused by the joint action of corrosion coupled with mechanical removal of the protective product film. It occurs under high velocity or turbulent liquid flow conditions. Partial obstructions within the fluid channel can cause localized high velocities and enhanced corrosion. Aluminum, copper, and steel are all subject to this form of corrosion. According to Popplewell a maximum velocity of 2 ft/sec. is considered relatively safe if the system is relatively free of abrasions.

General Wastage

Most of the fluid manufacturers show that the general wastage of common engineering materials by their fluids is small. Table B.2.5 shows a couple of examples of general wastage of metallic surfaces by different fluids. Thus, little is known at present of the possibilities of localized corrosion by the non-aqueous solutions

Vapor Pressure

Under zero flow conditions within the collectors, temperatures in excess of 300°F are possible. For aqueous solutions the vapor pressure under stagnation conditions can reach several atmospheres. Some collectors would not be able to withstand these pressures. Figure B.2.15 shows the absolute vapor pressure versus temperatur for several of the fluids. Other than the aqueous solutions and Dowtherm J, the vapor pressures of the fluids are quite low even under zero flow conditions.

Freeze Protection

One of the major drawbacks of water is its high freezing temperature (32°F). In

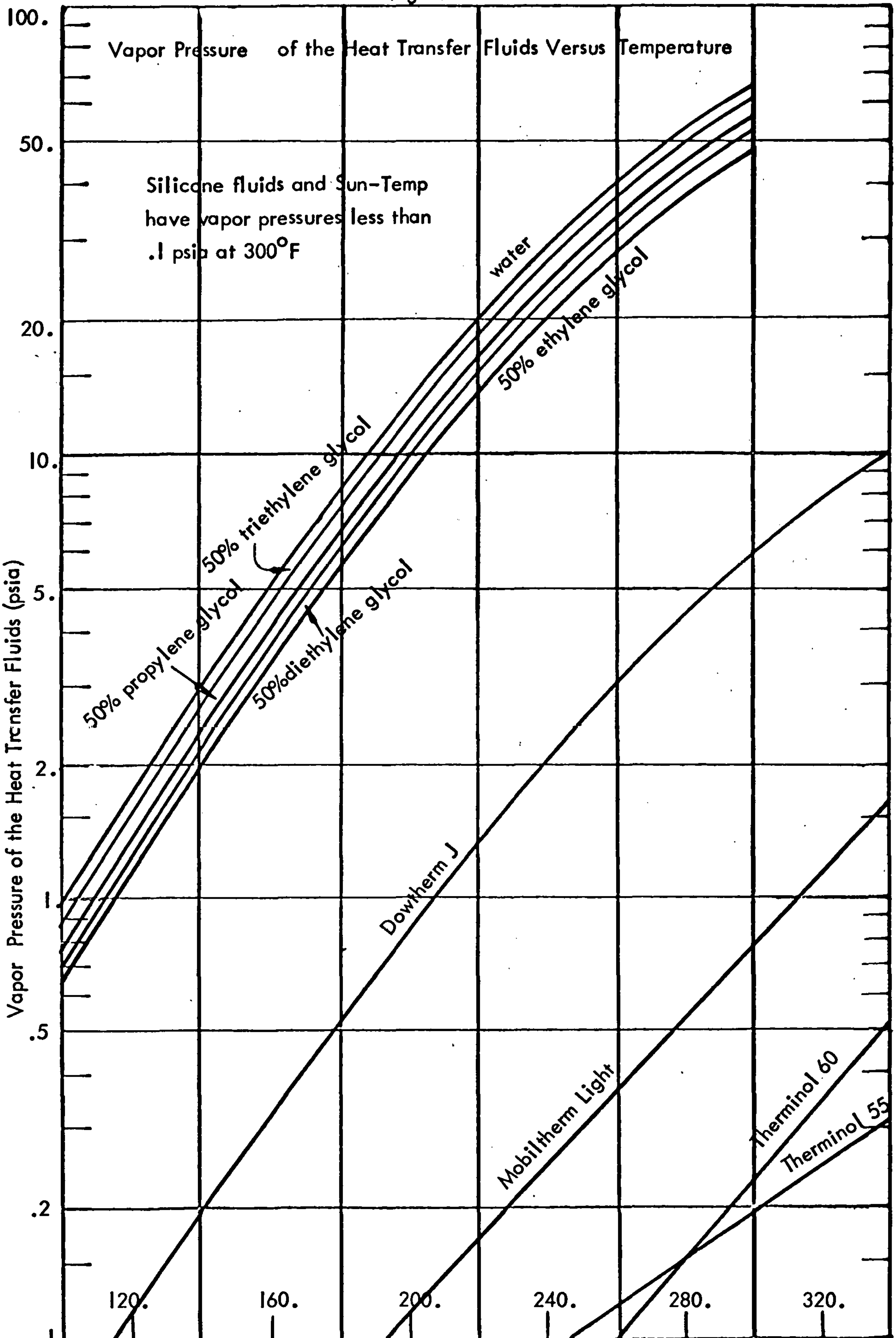
TABLE B.2.5

GENERAL CORROSION OF VARIOUS METALS BY HEAT TRANSFER FLUIDS

Metal	Silicone (Q2-1132) mg/cm^2	50% Propylene Glycol mg/cm^2 perday
Aluminum	.01 Bright	.25
Cast Iron	.01 Bright	--
Steel	.01 Bright	.002
Copper	.02 Medium Stain	.124

Silicone humidified fluid corrosion test results obtained as per SAE xj 1705 (from Dow Corning Form No. 22-380A-76).

Figure B.2.15



Vapor Pressure of the Heat Transfer Fluids Versus Temperature
Silicone fluids and Sun-Temp have vapor pressures less than .1 psia at 300°F
water
50% ethylene glycol
50% triethylene glycol
50% propylene glycol
50% diethylene glycol
Dowtherm J
Mobiltherm Light
Therminol 60
Therminol 55
100.
50.
20.
10.
5.
2.
1.
.5
.2
.1
120.
160.
200.
240.
280.
320.
Vapor Pressure of the Heat Transfer Fluids (psia)
Temperature (°F)

the continental United States there are few locations which have had no recorded below freezing temperatures. Figure B.2.16 Ruffner and Bair (1977) shows the record minimum temperatures for selected stations in the continental United States. This figure should be used as a guide only, since there are large local variations in minimum temperatures due to terrain. From Figure B.2.16, it can be seen that there are no stations located at these major cities which did not have a record of freezing temperatures.

Anti-freeze solutions have been commonly added to water to lower the freezing temperature of water. In some cases these solutions can retard the expansivity of the water and create a slush which will not rupture the fluid vessel. Most non-aqueous fluids do not expand upon freezing and thus reduce the risk of damaged piping.

Because some fluids become so viscous that the freezing temperature is not easily measured, the pour point temperature of the fluid is used as the lower operating limit of the fluid. The pour point temperature is the temperature of the fluid at which the fluid fails to flow when the container is tilted to horizontal and held for 5 seconds.

Figure B.2.17 shows the freezing and pour point temperatures (whichever was reported by the manufacturer) for the heat transfer fluids. For the glycol solutions the freezing temperature is shown as a function of concentration.

In determining the possibility of damage to a collector system by the heat transfer fluid at low temperatures, it was assumed for consistency that the fluids caused no damage at temperatures above the pour point or freezing temperatures.

Figures B.2.16 and B.2.17 also allow a cursory determination of whether a particular heat transfer fluid can be used and whether it will allow adequate freeze protection. For example, a 30% ethylene glycol aqueous solution should allow adequate freeze protection for most low elevation locations in California and southern Florida. A 30% ethylene glycol solution would clearly not suffice in the upper plains states in the winter.

The additional cost of backup freeze protection is considered in the next section.

Figure B.2.16

Record Minimum Temperatures (°F) for Selected Stations In The Continental United States

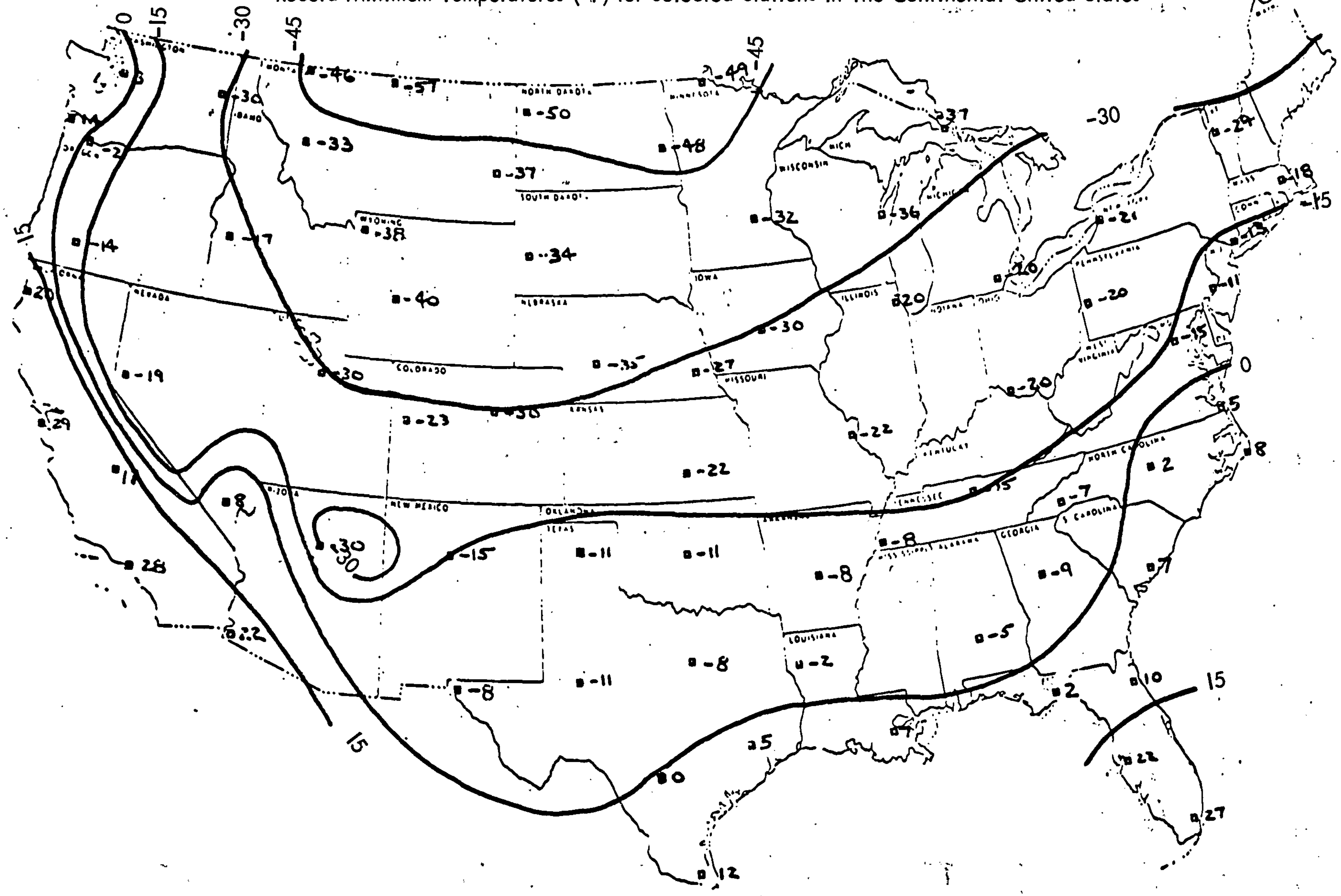
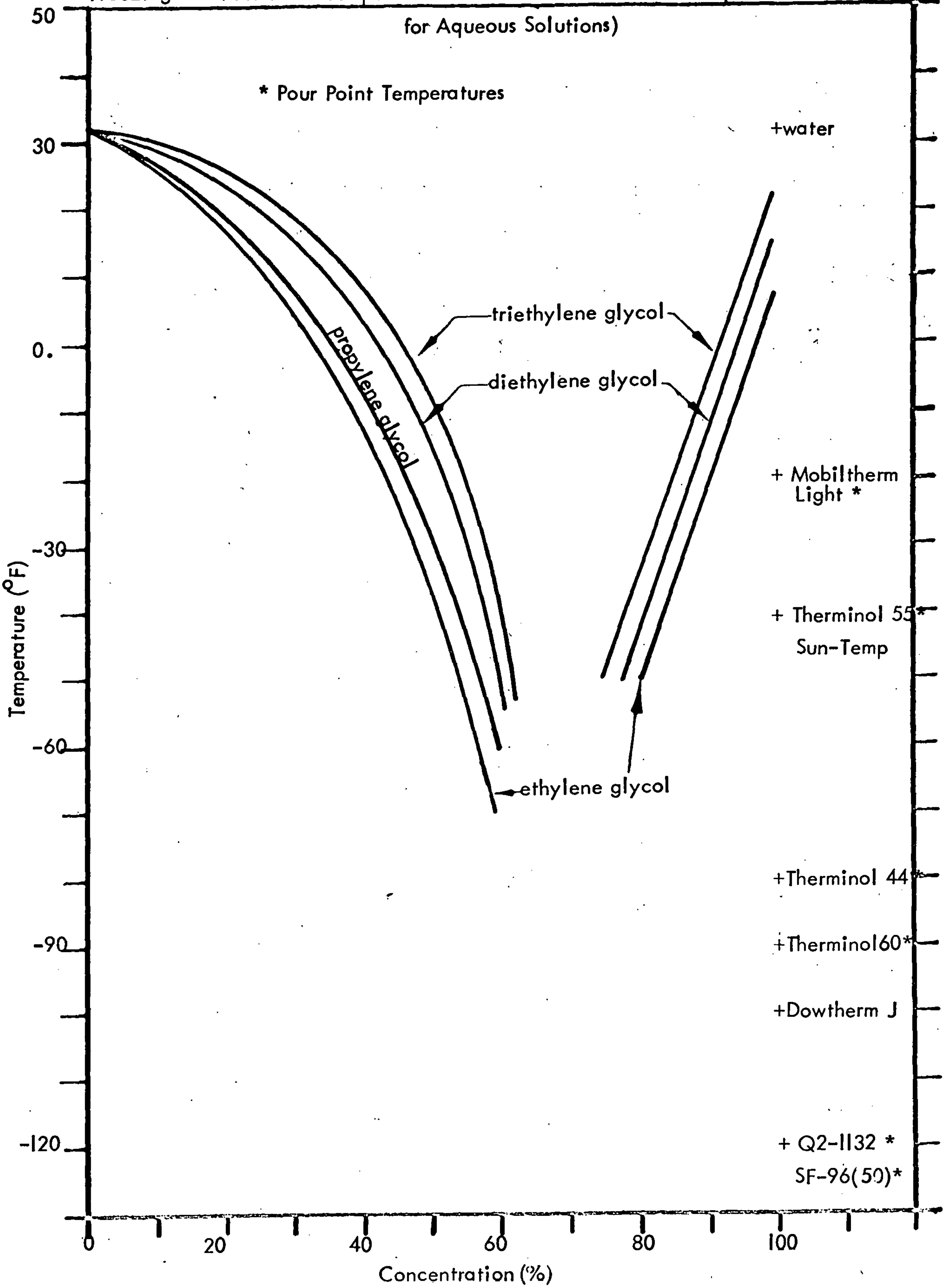


Figure B.2.17

Freezing and Pour Point Temperatures of the Heat Transfer Fluids (Versus Concentration for Aqueous Solutions)



B-3. Determination of Typical Collector and Heat Exchanger Characteristics

Both the collector and heat exchanger characteristics can affect the performance and cost effectiveness of a particular system. In the following two subsections, design and cost information is presented for typical collectors and heat exchangers.

Typical Collector Characteristics

The efficiency and cost characteristics of collectors have a major effect upon system performance and total cost. As shown in Section 2.1.4 and Section 2.2, the cost of the collector and the $F_R U_c$ product (rate of change of collection with increasing temperature) can affect the optimum heat exchanger size for a double loop system or also affect the optimum tube length of a single loop traced tank system. Typical collectors were compared for efficiency and cost in this study.

Table B-3-1 compares the type of collectors, their basic design materials and their cost for various manufacturers' products. Most of the collectors were flat plate collectors with the exception of the polyethylene pipe coil and a representative concentrating collector. Differing materials of construction were used in these typical collectors, with some all copper units while others used aluminum or steel. Most of these collectors were designed for use with a liquid heat transfer fluid with the exception of the Solaron Corporation unit which uses air as the heat transfer medium. Most of the units were compared for single glazed collectors, if available. Many manufacturers market collectors with other types or number of glazings. Costs vary widely depending on the materials of construction. In most sections of this report, the costs of the collector are allowed to vary to reflect the wide difference in prices available at present.

Figure B-3-1 shows the instantaneous collector efficiency (%) versus the collector-ambient temperature difference per unit incident radiation for the typical collectors as specified by the manufacturers. Figure B-3-1 shows that most of the collectors have similar performance within roughly 40%, with major exceptions for the low efficiency polyethylene coil (Solar Energy, Inc.), the concentrating collector (Northrup, Inc.) and the Daystar flat plate collector.

The F_RU_c products for each of the collectors can be determined from Figure B-3-1. The F_RU_c product is the slope of the performance curve. For example, the F_RU_c product for the Garden Way Lab collector is approximately 5 watts $m^{-2}\ {}^oC^{-1}$ (0.95 Btu $hr^{-1}\ F_t^{-2}\ {}^oF^{-1}$). For those collectors with non-linear dependencies on the temperature difference, the F_RU_c product should be determined locally i.e. near the temperature of operation.

For the purposes of comparison of other system parameters, in this study a copper absorber plate with copper tubes bonded to the collector with a tube spacing of approximately 0.2 meters was used. De Winter (1974, 1978) determined other needed design parameters for this type of collector. These other design parameters are shown in the computer programs in Appendices D and E.

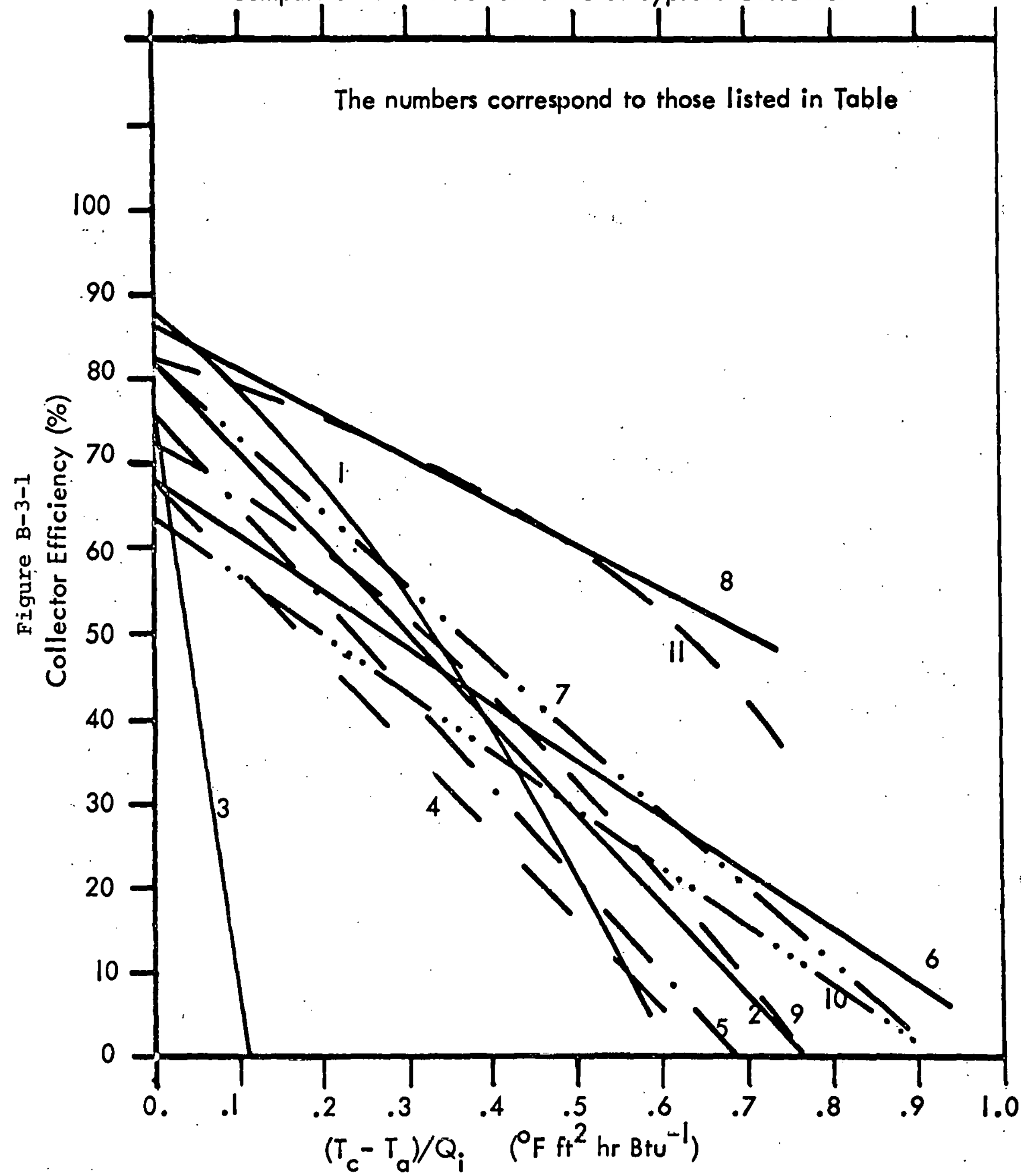
Table B-3-1

TYPICAL COLLECTORS AVAILABLE AT PRESENT

Number	Manufacturer	Product	Type	Price
1.	Kennecott	Terra-Light	Copper Flat Plate Collector with Single glass cover	2.75 (Absorber Plate only)
2.	Solar Energy Products, Inc.	Cu30	Flat Plate Collector, Aluminum plate, Copper Tubing single glass cover	
3.	Solar Energy, Inc.	Sunburst Solar Coil	Polyethylene Pipe	
4.	Solar Energy, Inc.	Sunburst Solar Collector	Aluminum Flat Plate Collector single Tedlar coated Fiberglass Cover	
5.	Sunworks	Solector	Copper Flat Plate Collector (single glazed)	12.00
6.	Chamberlain	Solar Collector Panels	Steel Flat Plate Collector (one cover, black paint)	9.00
7.	Garden Way Lab.	SunEarth Collector Model 3290	Steel Plate, Copper Tube Flat Plate Collector with single glazing	
8.	Daystar Corp.	Daystar 20	Flat Plate Collector Single Glazing	
9.	PPG	Baseline Solar Collector	Aluminum Flat Plate Collector Single Glazed Cover	7.20
10.	Solaron Corp.	Series 2000	Air Circulating Steel Flat Plate Collector Double Glazed Cover	
11.	Northrup, Inc.	Solar Collector	Concentrating Solar Collector with copper absorber	

Figure B.3.1

Comparison of the Performance of Typical Collectors



Typical Heat Exchanger Characteristics

Over fifty possible heat exchanger manufacturers were contacted to develop viable design procedures for heat exchanger selection applicable to those heat exchangers presently available. Table B-3-2 lists several heat exchanger manufacturers interested in the use of their products for solar energy applications. Table B-3-3 lists typical characteristics of shell and tube counterflow single pass heat exchangers available from several manufacturers. The choice of a heat exchanger can affect the following:

(1) Corrosion

(2) Heat Transfer

(3) Cost

In selecting a heat exchanger, corrosion enhanced by particular heat transfer fluids must be considered. In Table B-3-3, heat exchangers were chosen which would be applicable for the use of potable water to circulate through the tubes (i.e. copper or stainless steel tubing). Such exchangers should reduce the possibility of corrosion within the tubes. On the shell side, because several different heat transfer fluids are viable choices, various materials of construction were allowed.

The effect of heat exchanger size on heat transfer was discussed in Section 2.1.4 As mentioned in that section, the number of tubes and lengths available for a particular tube size are important in determining the overall performance of the system. If too many tubes are used, the heat transfer within the tubes is reduced due to lower operating flowrates. If too few tubes are used, the tube lengths required to meet the optimum heat exchanger area can become too long. From Table B-3-3 it can be seen that the minimum number of tubes is approximately 30. Also for the larger shell diameters, the number of tubes is similar for the different manufacturers. Tube lengths are generally available from 12 feet for the small shell diameters, to 20 feet for the larger shell diameters.

The cost of many of the heat exchangers is also given in Table B-3-3. Note that the cost of the heat exchangers per unit area depends on both tube length and shell diameter. Thus the longer, larger shell diameter exchangers are less costly on a per unit area basis than the smaller units. Note also that the price listed is for single list price off the shelf heat exchangers. It is possible that custom heat exchangers for solar energy applications could become available in sizes and costs other than listed in the above table.

Because of the availability of heat exchangers in a wide range of sizes and cost, the design procedures for selecting heat exchangers were kept general so that they would equally apply to most heat exchangers being marketed at present. See Section 2.1.4 for further discussion of the design procedures for heat exchanger selection.

Table B.3.2

HEAT EXCHANGER SUPPLIERS INTERESTED IN DEVELOPING DESIGNS FOR SOLAR ENERGY APPLICATIONS

	Domestic Hot Water X	Home Heating X	Industrial Heat X
(1) Agric Machinery Corp. Madison, New Jersey 07940	No	Yes	Yes
(2) Atlas Industrial Mfg. Co. Clifton, New Jersey 07012	No	Yes	Yes
(3) American Heat Reclaiming Corp. New York, N.Y. 10020	No	No	Yes
(4) Chromalox South El Monte, Calif. 91733	No	No	Yes
(5) Ecodyne, MRM Division Massillon, Ohio 44646	Yes	No	No
(6) Harris Thermal Transfer Products Tualatin, Oregon 97062	No	No	Yes
(7) R.W. Holland Co. Houston, Texas 77041	Yes	Yes	Yes
(8) Richard S. Dawson Co. Los Angeles, CA 90026 (Bell and Gossett)	Yes	Yes	Yes
(9) Packless Industries, Inc. Mount Wolf, Penn. 17347	Yes	Yes	Yes
(10) Patterson Kelley Company East Stroudsburg, Penn. 18301	No	No	Yes
(11) PEMCO, Inc. Elizabeth, New Jersey 07201	No	No	Yes
(12) Tranter Inc. Lansing, Michigan	Yes	Yes	Yes
(13) WSF Industries Inc. Tonawanda, N.Y. 14150	No	No	Yes
(14) House of Hydraulics (Ametek Whitlock) Cerritos, CA 90701	Yes	Yes	Yes
(15) C.H. Bull Company So. San Francisco, CA 94080	No	Yes	Yes
(16) Young Radiator Company Racine, Wisconsin 53404	Yes	Yes	Yes
(17) Airesearch Mfg. Company Torrance, California 90509	No	Yes	Yes

Table B.3.3

TYPICAL HEAT EXCHANGER DESIGNS AVAILABLE AT PRESENT

Company	Ident. No.	Materials Used Shell	Tubes	Heads	Baffles	O.D. Shell in.	O.D. Tubes	Shell Thickness in.	Tube Thickness in.	Length of Tubes in.	No. of Tubes	Total Area Ft^2	Cost (List Price) $\$/Ft^2$
Bell & Gossett	STH-310-1	Copper	Copper	Cast Iron	Brass	3	1/4	1/16	.02	12	60	4.2	48.6
Fluid	STH-320-1	"	"	"	"	"	1/4	"	"	24	50	7.6	26.8
Handling	STH-515-1	"	"	"	"	5	3/8	"	"	18	80	13	23.3
Division	STH-530-1	"	"	"	"	"	"	"	"	36	80	24	16.6
	STH-620-1	"	"	"	"	6	"	"	"	24	140	28.2	16.0
	STH-650-1	"	"	"	"	"	"	"	"	60	140	66	9.8
Harris Thermal Transfer	8F9	Carbon Steel	Copper	Carbon Steel	Carbon Steel	8	3/4	1/16	"	108	51	89	
	24F18					24	"	"	"	216	479	1672	
	03008	Brass	Copper	Cast Iron	Brass	3	1/4	1/16	.02	8	56	2.4	32.8
American	03024	"	"	"	"	3	"	"	"	24	56	7.4	18.2
Standard	05014	"	"	"	"	5	3/8	"	.03	14	80	9.1	23.6
	05036	"	"	"	"	"	"	"	"	36	80	24	13.1
	06024	"	"	"	"	6	"	"	"	24	115	23	13.1
	06060	"	"	"	"	"	"	"	"	60	115	56	10.0
	08024	"	"	"	"	8	3/8	"	"	24	210	41	16.0
	08072	"	"	"	"	8	"	"	"	72	210	124	8.8
Pemco	LL6-96	Steel	Stainless Steel	Carbon Steel	Carbon Steel	6 3/8	5/8	1/4	.03	96	37	47	
	LL6-240	"	"	"	"	"	"	"	"	240	37	107	
	LL8-96	"	"	"	"	8	"	"	"	96	64	82	
	LL8-240	"	"	"	"	"	"	"	"	240	64	208	
Young Radiator	F301	Brass	90-10 CuNi	Cast Iron	Brass	3"	3/8	1/16	.02	9	35	3.6	34.7
	F303	"	"	"	"	"	3/8	"	"	27	35	7.8	30.8
	F502	"	"	"	"	5	"	"	"	18	75	11.2	31.9
	F504	"	"	"	"	"	"	"	"	36	75	22.4	24.4
	F602	"	"	"	"	6	"	"	"	18	120	17.5	21.2
	F608	"	"	"	"	"	"	"	"	72	180	70.2	18.2
	F802	"	"	"	"	8	"	"	"	18	230	34.1	21.1
	F810	"	"	"	"	"	"	"	"	90	230	166.1	12.1
Ametek Whitlock	2-W-8	Brass	Inhib. Admiralty	Cast Iron	Brass	2	1/4	1/16	.02	8	31	1.35	74.1
	2-W-48	"	"	"	"	"	"	"	"	48	31	8.1	22.3
	3-Y-8	"	"	"	"	3	3/8	"	"	8	28	1.8	92.8
	3-Y-48	"	"	"	"	"	"	"	"	48	28	11	31.0
	5-Y-14	"	"	"	"	5	"	"	"	14	84	9.6	35.7
	5-Y-48	"	"	"	"	"	"	"	"	48	84	33	18.3
	8-Y-24	"	"	"	"	8	"	"	"	24	224	44	23.0
	8-Y-72	"	"	"	"	"	"	"	"	72	224	132	13.2

B-4 Determination of Optimum Insulation Thicknesses

The insulation thickness required for a particular component of the collector-storage system depends on the cost of the insulation and the amount of heat lost through the insulation. If the insulation is too thin, although the cost is low, the amount of heat lost can be large. For thick insulation, which is more costly, the amount of heat lost is reduced. Thus there exists some optimum insulation thickness for a particular application depending on the cost of the insulation and the cost of the lost heat. The optimum thickness of insulation occurs when:

$$\frac{d}{dT_{ins}} [C_{total}] = 0 \qquad \text{(B-4-1)}$$

When T_{ins} - Thickness of insulation (ft.)

C_{total} - Total cost affected by thickness ($)

$$C_{total} = \frac{\text{cost of heat "lost"}}{\text{year}} + \frac{\text{cost of insulation}}{\text{year}}$$

$$\frac{D_y h_{op} A_{suf} K_{ins} [T_L - T_a] C_{Btu}}{T_{ins}} + T_{ins} A_{suf} C_{ins} C_{eqy} \qquad \text{(B-4-2)}$$

D_y - Number of operating days per year.

h_{op} - Number of operating hours per day.

A_{suf} - Surface area covered by insulation ($Ft.^2$).

K_{ins} - Thermal conductivity of insulation Btu/hr Ft $^\circ$F.

$T_h - T_a$ - Temperature difference between fluid inside insulated system component and the ambient temperature [$^\circ$F].

C_{Btu} - Cost of B_{tu} lost [\$/B_{tu}]. This is the cost of additional heating needed to compensate for the lost heat.

C_{ins} - Cost of insulation installed [\$/Ft^3]. It is assumed to not be a funtion of insulation thickness.

C_{eqy} - Yearly percentage cost of equipment.

Substituting Equation (B-4-2) into (B-4-1) and solving for the thickness of insulation [T_{ins}] yields:

$$T_{ins} = \left[\frac{D_y h_{op} K_{ins} (T_h - T_a) C_{Btu}}{C_{ins} C_{eqy}} \right]^{1/2} \qquad \text{(B-4-3)}$$

De Winter (1974) presents a simple scheme for determining the yearly percentage cost of equipment (C_{eqy}) knowing the yearly interest rate (%) and the expected useful life of the equipment, based conservatively on a zero scrap value at the end of the equipment's useful life. This is reproduced as Figure B-4-1. For a typical system, with a useful life of 10 years and a yearly interest rate of 8%, the yearly percentage cost of equipment is 15%, thus C_{eqy} is .15.

The cost (C_{ins}) and thermal conductivity (K_{ins}) of the insulation varies between the products of different manufacturers. Table B-4-1 shows a small sample of present insulation available. Obviously the best insulation is one in which both the thermal conductivity and cost are low. Note that the cost of insulation ($\$/Ft^3$) is a function of insulation thickness especially for pipe insulation. Assuming a constant cost of insulation will result in slight off optimum design especially for small insulation thicknesses.

The cost of Btu's lost (C_{Btu}) can be determined from current costs of fossil fuels. A typical design value for C_{Btu} is $\$3/10^6$Btu.

The temperature difference (T_h-T_a) can be assume to be constant for varying insulation thicknesses for design purposes. In this study a temperature difference of 75°F was assumed.

The number of days of operation per year (D_y) should be 365 for design purposes while the hours of operation (h_{op}) vary depending on the component for which the insulation is to be sized. For a storage tank H_{op} should be 24 hrs/day while for piping from collector to storage tank the hours of operation can be assumed to be 8 hrs/day.

Using the assumptions listed above Equation (B-4-2) becomes:

$$T_{ins} = \left[\frac{13.14\ K_{ins}}{C_{ins}} \right]^{1/2} \quad \text{for storage tank} \quad \text{(B-4-4)}$$

$$T_{ins} = \left[\frac{4.38\ K_{ins}}{C_{ins}} \right]^{1/2} \quad \text{for collector, piping, heating exchangers} \quad \text{(B-4-5)}$$

Figure B-4-2 shows the optimum insulation thickness (in.) versus the thermal conductivity of (Btu) and the cost of the insulation ($\$/ft^3$) for the storage tank case. Figure B-4-3 presents the same for the collector, piping or heat exchanger case.

For example, assume that a storage tank is to be covered by Johns-Manville Thermo 12 block insulation. The cost and thermal conductivity (at 100 $^{\circ}$F) of this insulation is $\$10./ft^3$ and 0.0317 Btu/hr ft $^{\circ}$F respectively. Assume all the other parameters are the same as above. From Figure B-4-2 the thickness of insulation needed is 2.5 in. Note it was assumed that the cost of insulation was approximately constant above a thickness of 1.5 in. and equal to $\$10/ft^3$.

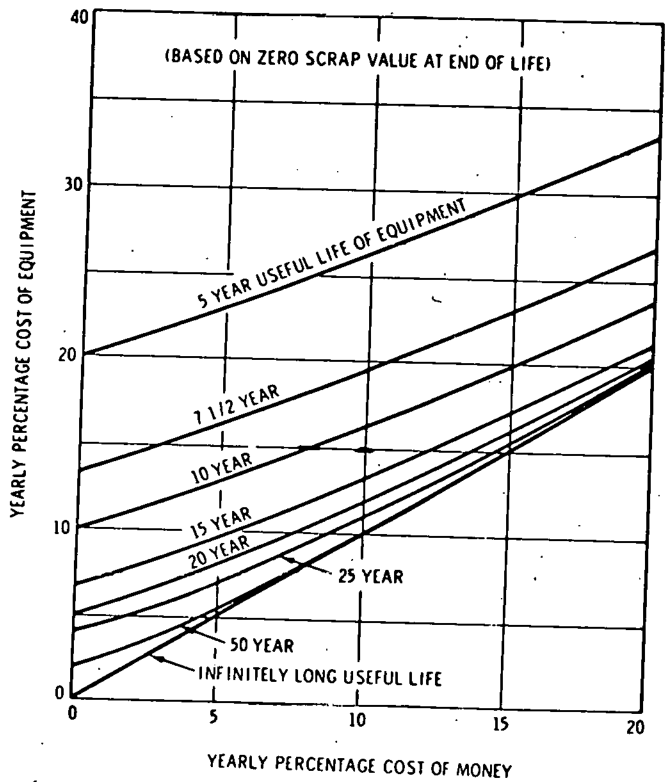


Figure B-4-1 How the Yearly Percentage Cost of Equipment varies Depending on the Annual Cost of Money (Interest) and on the Lifetime of the Equipment.

Figure B-4-2

Thickness of Insulation Required for a Storage Tank Versus the Cost (C_{ins}) and Thermal Conductivity of the Insulation

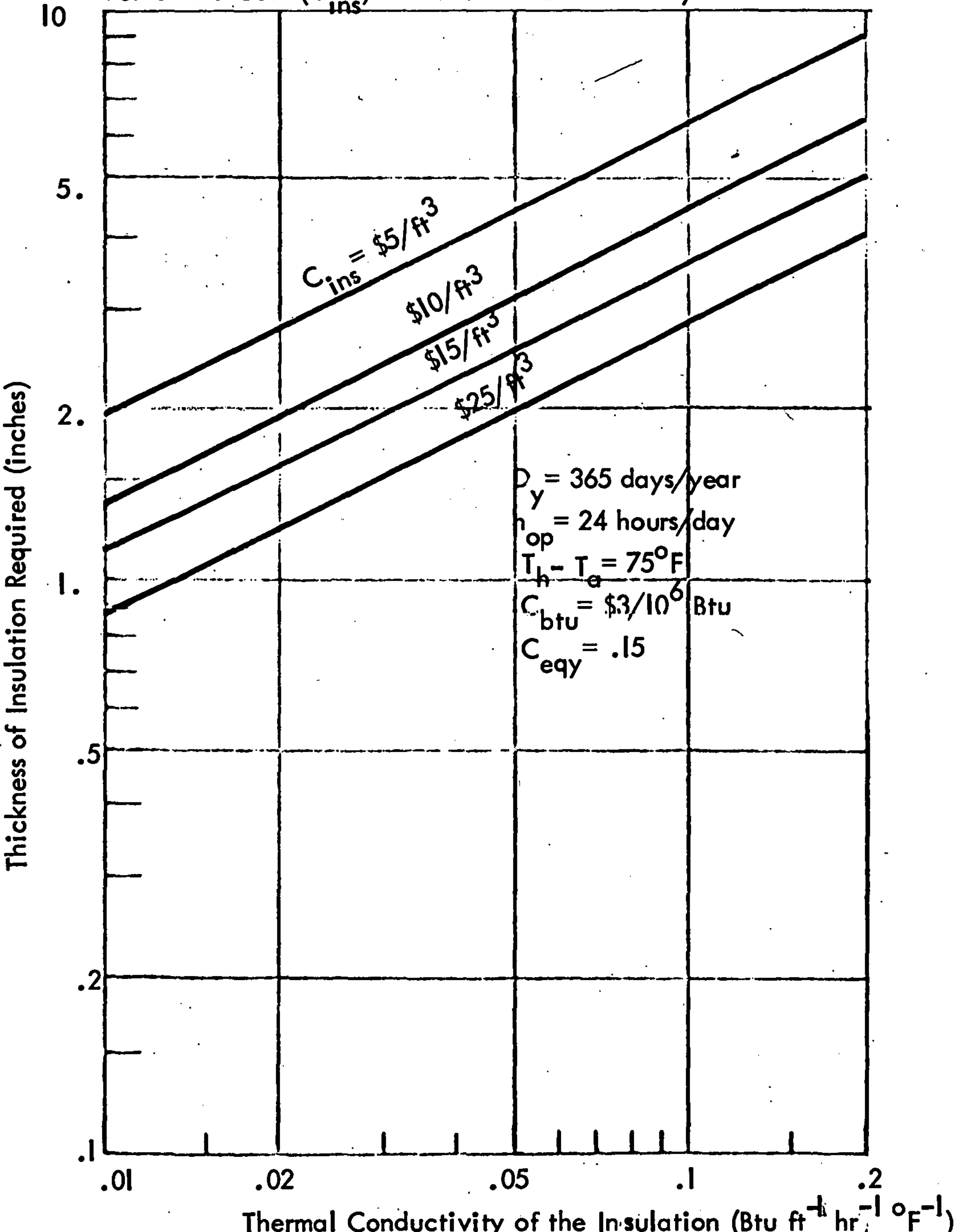


Figure B-4-3

Thickness of Insulation Required for a Heat Exchanger or Pipes Versus the Cost (C_{ins}) and Thermal Conductivity of the Insulation

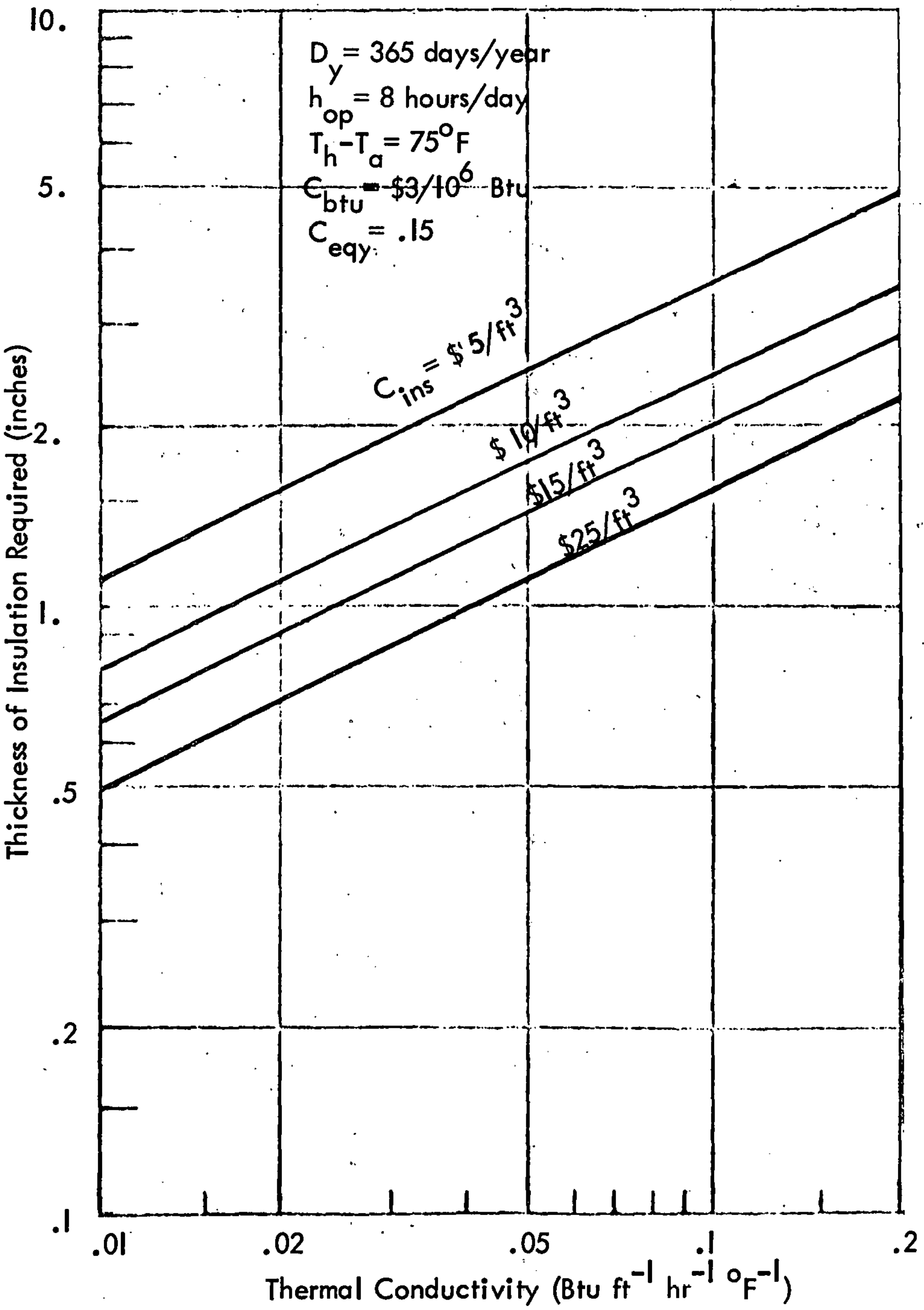


Table B-4-1

Representative Thermal Conductivities and Cost of Insulation for avrious Insulation Products

Manufacturer	Product	Type of Insulation	Standand Sizes (inches)	Thermal Conductivity (Btu/hr Ft °F) 100°F	Thermal Conductivity (Btu/hr Ft °F) 200°F	Thickness (inches)	Cost ($/Ft^3)
Johns-Manville	Thermo-12	Block	12 x 36 inches Thickness 1-4 inches	0.0317	0.0342	1	13.1
						1.5	10.2
						2	10.1
Johns-Manville	Thermo-12	pipe	Fit 36 inches, tube nominal O.D. 3/8 to 6 inches	0.0317	0.0342	$\frac{3}{4}$" O.D. tubes 1"	21
						3	5.66
Johns-Manville	Flame-Saf	pipe	$5\frac{1}{8}$" O.D. to $12\frac{1}{8}$" O.D. $\frac{1}{2}$" to 3" Thickness	0.02	0.0242	$\frac{3}{4}$" O.D. Tubes 1"	13.17
						3"	4.26
Pittsburgh Corning	Foam Glass	Block	12"x18"x$1\frac{1}{2}$ to 5 in	0.0333	0.0359	$1\frac{1}{2}$	3.84
						5	3.88
		pipe	$\frac{1}{4}$" to 36 in O.D. in 24" lengths	0.0333	0.0359	$\frac{3}{4}$" O.D. 1"	10
						3"	8.0

Appendix C Storage Tank Wall Thicknesses

C-1 Design of Pressurized Tanks

The storage tanks analyzed in this section were assumed to be cylindrical, vertical tanks. The height to diameter ratio was assumed a constant. The volume of storage needed for most collectors is usually known and here was assumed to be constant. Thus the tank diameter (D_t) is given by:

$$D_t = \left(\frac{.0519\ K_2 A_c}{K_1}\right)^{1/3} \quad [m] \qquad [C\text{-}1\text{-}1]$$

where K_1 = ratio of tank height to diameter = H_t/D_t

H_t = Height of cylindrical tank = $K_1 D_t$

K_2 = Volume storage required/ft^2 collector $\left(\frac{\text{Gal}}{\text{Ft}^2 \text{ collector}}\right)$

A_c = collector area [m^2]

in this report it was assumed that K_1 = 3 and

K_2 = 2 gal/(ft.^{2} collector)

Figure [C-1-1] shows the tank diameter required versus storage required per unit collector area and the collector area for K_1 equal to 3.

The wall thickness needed for the cylindrical tank system can be computed knowing the radius of the tank and properties of the material of construction. From Roark [1954] Formulas for Stress and Strain the wall thickness [T_{wall}], neglecting joints and end effects is:

$$T_{wall} = \frac{pr}{\sigma} \quad [in] \qquad [C\text{-}1\text{-}2]$$

where p = Design pressure of tank [psi]

r = Radius [in.]

σ = Maximum allowable hoop stress of wall material [psi]

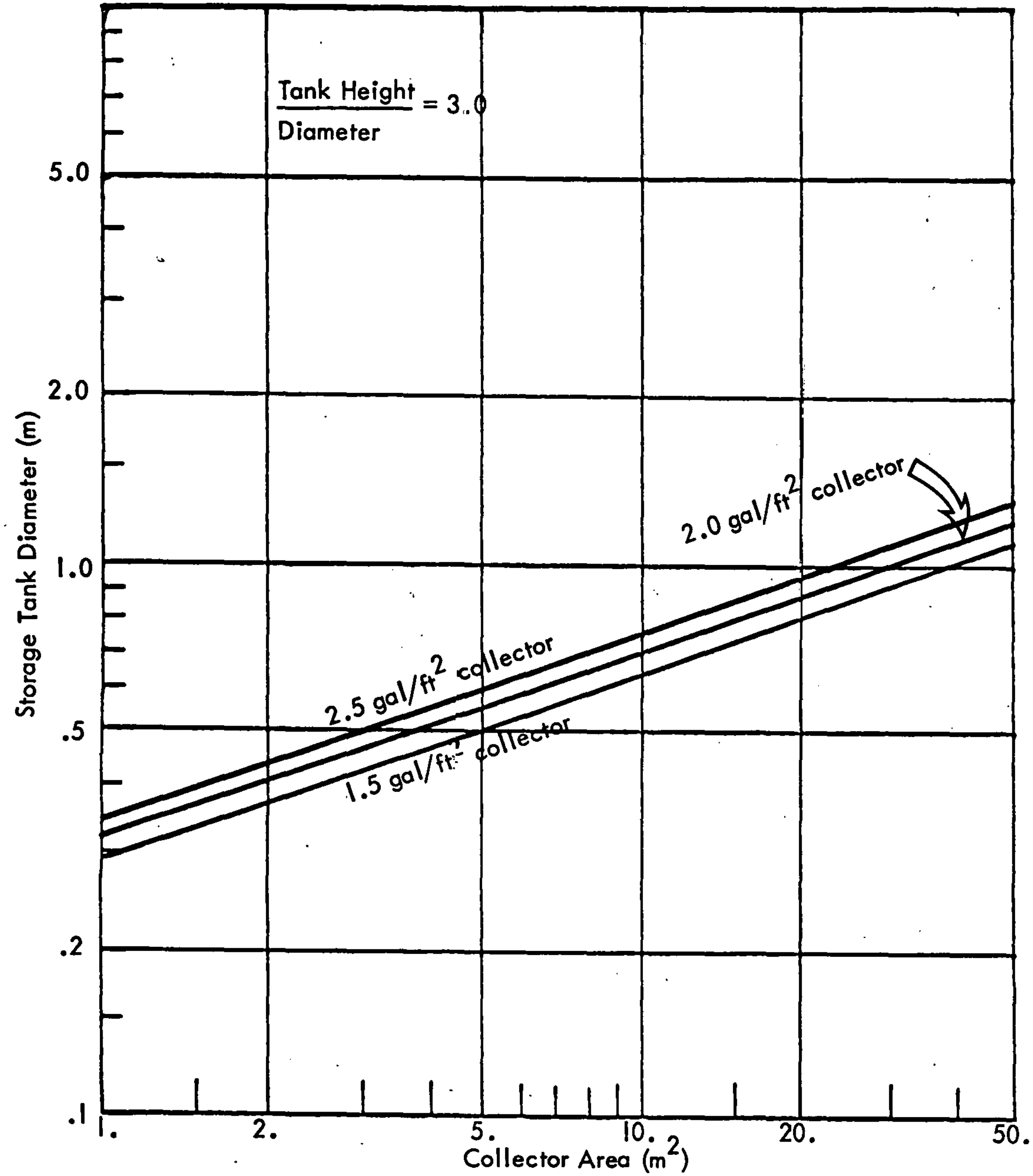
A more conservative estimate of the thickness taking into account joint efficienc is:

$$T_{wall} = \frac{pr}{\epsilon'\sigma} \qquad [C\text{-}1\text{-}3]$$

ϵ' - Joint efficiency. In this study it is assumed to be .7 for longitudinal welded joints.

Figure C-1-1

Storage Tank Diameter Versus Collector Area and Storage Required per unit collector area

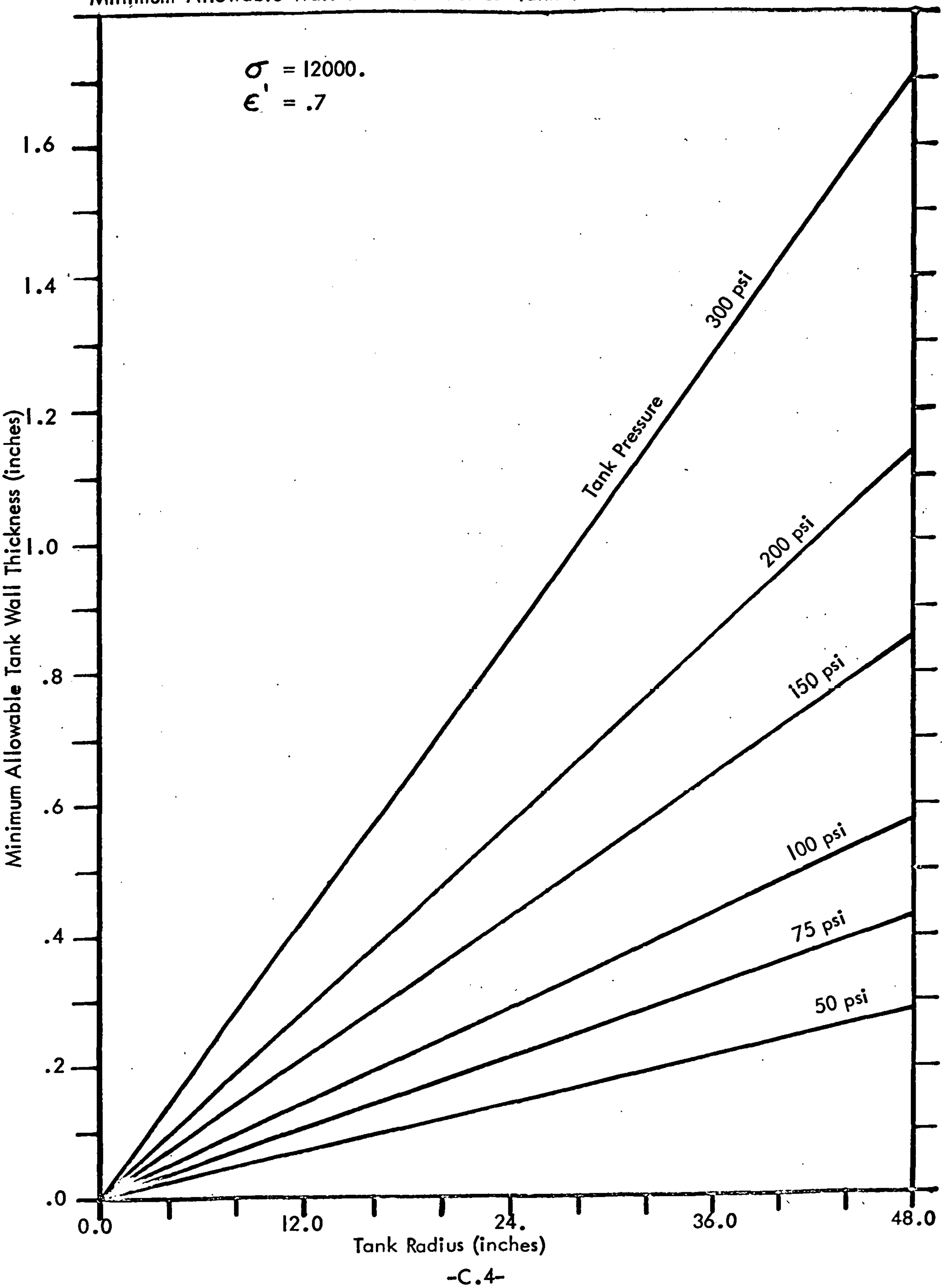


For this study the tank was assumed to be carbon steel. From The Standards of Tubular Exchanger Manufacturer's Association, Anon(1954) the maximum allowable stress (σ) can be determined from the grade of the steel. For this study a design maximum allowable stress (σ) was assumed equal to 12,000 psi with a joint efficiency of .7 . Figure [C-1-2] shows the tank wall thickness versus the radius of the tank and the design pressure of the tank.

Thus the storage tank can be completely specified from equations C-1-1 and C-1-3 or Figures C-1-1 and C-1-2.

Figure C-1-2

Minimum Allowable Wall Thickness Versus Tank Pressure and Radius



C-2 Determination of the Effect of Storage Tank Pressure on Storage Tank Cost

Except in those cases in which a heat exchanger produces a pressure isolation between the collector and storage loop, the storage tank may have to operate at a pressure dictated by the hydrostatic head imposed by the collector level. Depending on the vertical distance between the collector and storage tank, the hydrostatic pressure can easily be two or three atmospheres. The cost effect of this added design pressure is reflected in the increased wall thickness needed.

As shown in Appendix C-1, the wall thickness (T_{wall}) is a function of the design pressure (p), the maximum allowable hoop stress (σ), the joint efficiency (ϵ') and the radius of the tank (r_t). The cost of the tank increases with increasing thickness and radius of the tank and is a function of the cost per unit pound and weight (lbs) of the tank.

To determine the effect of varying height difference between the collector and the storage tank on the cost of the storage tank, the following assumptions were made:

1. The tanks studied were cylindrical with a flat top and bottom.
2. The length to diameter ratio of the tank was a constant.

The increased cost of the tank (C_{inc}) was determined from:

$$C_{inc} = cost_1 - cost \ (\$) \tag{C-2-1}$$

$Cost_1$ - cost of tank with increased tank wall thickness due to hydrostatic head

Cost - cost of tank without increased wall thickness

Another parameter developed was the percent increase in tank cost (P_{inc}). This can be found from:

$$P_{inc} = \frac{Cost_1 - cost}{cost} \times 100\% \tag{C-2-2}$$

Using the relationships shown in Appendix C-1 and the above assumptions equations (C-2-1) and (C-2-2) can be reduced to:

$$C_{inc} = \rho' \frac{C_{lb}\ Gal\ \rho\ Z}{1077\ \sigma \epsilon'} \left(2 + \frac{P}{\sigma\epsilon'} + \frac{\rho\ Z}{\sigma\epsilon'\ 144} + \frac{2}{K}\right) \qquad (C\text{-}2\text{-}3)$$

ρ' - Density of tank material (lb)/ft^3

C_{lb} - Cost of tank ($/lb)

Gal - Storage capacity of tank (gallons)

ρ - Density of water (lb) (61.5 $\frac{lb}{ft^3}$ a 150 $^\circ$F

Z - Difference in height between storage tank and collector (ft)

P - Design pressure of storage tank (Psi)

σ - Maximum allowable hoop stres (Psi)

ϵ' - Joint efficiency

K - Tank height to radius ratio

and

$$P_{inc} = \frac{\rho\ Z}{144\ P} \left(\frac{2 + \frac{p}{\sigma\epsilon'} + \frac{Z}{\sigma\epsilon'\ 144} + \frac{2}{K}}{2 + \frac{p}{\sigma\epsilon'} + \frac{2}{K}} \right) \qquad (C\text{-}2\text{-}4)$$

For a typical steel tank with:

$\rho' = 8\rho$, $\sigma = 12000$ Psi, $\epsilon' = 0.7$, $K = 6$

The increased tank cost becomes:

$$C_{inc} = 7.6394 \times 10^{-3}\ C_{lb}\ Gal\ Z \qquad (C\text{-}2\text{-}5)$$

And the percent increase in tank cost becomes:

$$P_{inc} = 43.09\ Z/P \qquad (C\text{-}2\text{-}6)$$

From (C-2-5) the effect of increased storage capacity, storage to collector height difference, and cost of the tank can be seen. For larger tanks, there is a lower cost difference between a tank with the collector and storage at the same level and one with a large collector-storage height difference.

Figure (C-2-1) shows the increased cost ($) due to collector-storage height difference versus storage capacity and vertical height difference (ft) developed from Equation (C-2-3).

Figure C-2-1

Increased Storage Tank Cost Versus Volume of Storage and the Vertical Difference Between the Collector and Storage Tank

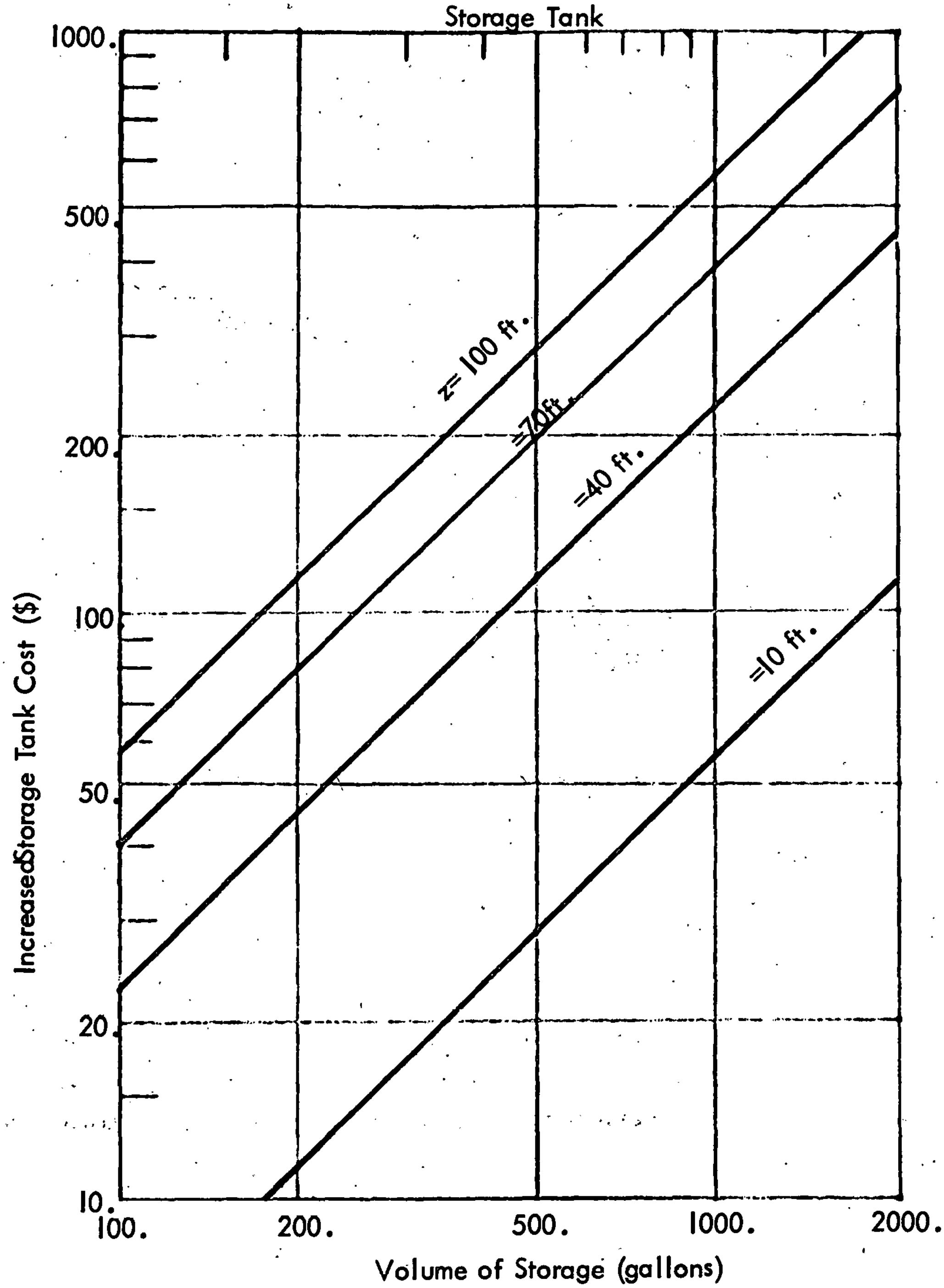
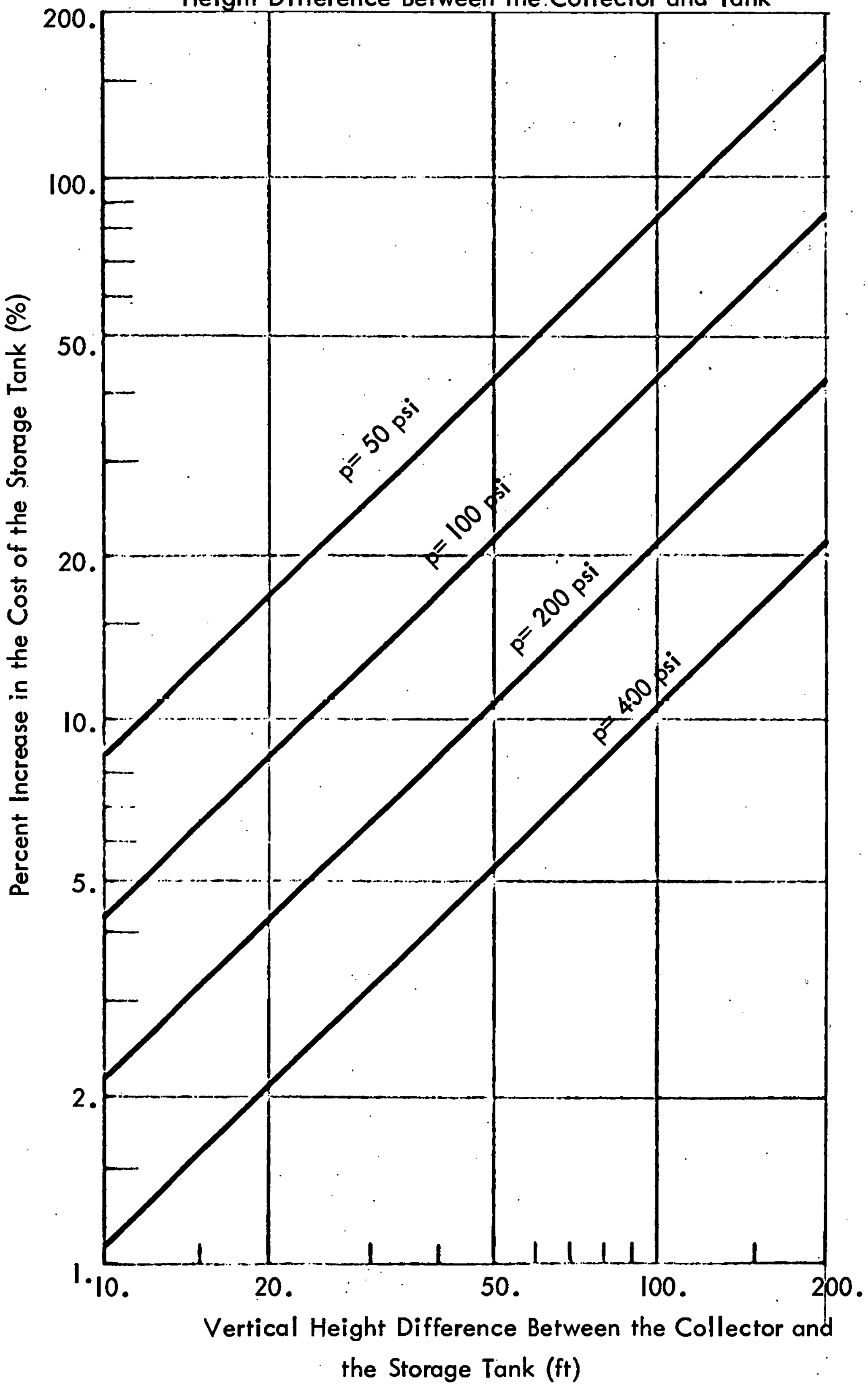


Figure C-2-2

Percent Increase in the Cost of the Storage Tank Versus the Initial Tank Pressure and the Vertical Height Difference Between the Collector and Tank



Equation (C-2-6) allows a simple determination of the percent increase in tank cost due to collector-storage height difference. For large design pressures the percent increase will be smaller while for large height differences the % increase is larger. Figure C-2-2 shows the % increase in cost versus height difference and design pressure from equation (C-2-4).

An example of these two relationships follows. Assume the cylindrical steel tank has a capacity of 1000 gallons with a design pressure of 300 Psi. Assume also that the cost of the tank is \$0.75/lb with a collector to storage tank vertical drop of 40 ft. From the above relationships and figures the increase in cost is \$239 which is a percent increase in tank cost of 5.75%.

APPENDIX D

DOUBLE LOOP MODEL

The following computer program modeled the performance of the double loop heat exchanger system. Most of this program and the input paramaters used have been discussed in other sections of this report. In this particular case, it was used to show the difference in performance between several heat transfer fluids. Sample printouts of the results follow the computer program listing for the cases when water, 50% ethylene glycol and mobiltherm light were used within the collector loop.

```
DIMENSION QINCID(24)
DIMENSION QFAC(24)
DIMENSION A1(15), A2(15), A3(15), A4(15), A5(15)
DIMENSION E1(15),E2(15),E3(15),E4(15),E5(15)
DIMENSION F1(15),F2(15),F3(15),F4(15),F5(15)
DIMENSION DEN1(15),DEN2(15),DEN3(15),DEN4(15),DEN5(15)
DIMENSION FUNC(10),AREA(10),G(10)
DIMENSION TEMP( 9, 24)
DIMENSION TAMB(24)
DIMENSION TW(24),TL(24),P1(24),P112(24),P2(24),P22(24),GM1(24)
DIMENSION GM2(24),REYN(24),PRAN(24),XH(24),VIS1(24)      ,EC(24)
DIMENSION QCP(24),QCP2(24),QC3(24),VIS2(24),REY(24),XH1(24)
DIMENSION EXC(24),U(24),XLS(24),XLD(24),WLS(24),WAS(24),RLS(24)
DIMENSION TH(24) ,EXA(24),TRET(24),QAR(24),ALO(24),QHP(24)
DIMENSION TL1(24)
REAL NROW
```

```
      REAL NRAF
      ACCG= 4.18E8
      PI= 3.141593
C        DELTIM- INTERVAL OF TIME IN WHICH PROGRAM IS EVALUATED IN HR
      DELTIM= 1
C        CHARACTERISTICS OF COLLECTOR
C        DIAMO- OUTSIDE DIAM. OF COLLECTOR TUBE IN M
      DIAMO= .625 * .0254
C        DIAMI- INSIDE DIAM. OF TUBE IN COLLECTOR IN M
      DIAMI= .569 * .0254
C         DIAMX- INSIDE DIAM. OF COLLECTOR TUBE IN FT.
      DIAMX= .569 /12.
C        SIGMAO  FIN THICKNESS OF COLLECTOR IN M
      SIGMAO=  .0108 * .0254
C        XLENGT- FIN LENGTH OF COLLECTOR IN M
      XLENGT= .1
C        COND- COND. OF COPPER TUBING AND SOLDER IN WATTS/M C
      COND= 396.
C        CSUBS- BOND CONDUCTANCE
      CSUBS= 4. * SIGMAO * COND/DIAMO
C        W- SPACING OF TUBING ON COLLECTOR IN M
      W= 2. * XLENGT + DIAMO
C        USLOPE- HEAT LOSS COEFFICIENT FOR COLLECTOR....
      USLOPE= 5.
C
C        CHARACTERISTICS OF FLUID WATER HEAT EXCHANGER
C        KEXC- CONDUCTIVITY OF EXCHANGER WALL IN WATTS/M C
      KEXC= 396.
C        XEXC IS THE THICKNESS OF THE EXCHANGER WALL IN FT
      XEXC= .03/ 12.
C        SIZE- O.D. OF EX'ER TUBE IN FT.
      SIZE= .375/12.
C        XD IS THE INNER DIAMETER OF THE TUBE INTHE EXCHANGER IN FT
      XD= SIZE - 2. * XEXC
      XD= XD * .3048
      XE= XD
C        XDP- I.D. OF EX'ER TUBE IN FT
      XDP= XE/.3048
C        XNUT- CROSS-SECTIONAL AREA  OF TUBE IN EX'ER
      XNUT= 4./( PI * ( XD ** 2.))
C        SSIZE- O.D. OF EX'ER IN M
      SSIZE= SIZE * .3048
C        PITCH- PITCH OF TUBING WITHIN EXHER
      PITCH= 1.25
C        SPACING BETWEEN TUBES (FT)
      SMIN= (PITCH - 1.) * SIZE
C        DSHEL- I.D. OF EX'ER SHELL
      DSHEL= 6./12.
C        CHARACTERISTICS OF WATER
C        B1 THRU B5 ARE CONSTANTS FOR THE SPECIFIC HEAT EQUATION
C        THEY HAVE BEEN CALCULATED FROM HANDBOOK OF HEAT TRANSFER
      READ 1080, B1,B2, B3, B4, B5
C        D1 - D5 - COEFF. FOR CONDUCTIVTY EQUATIONS OF WATER
      READ 1080, D1, D2, D3, D4, D5
C        R1 - R5 - COEFF. FOR DENSITY EQUATION OF WATER
      READ 1080, R1, R2, R3, R4, R5
C        Z1-Z5 -COEFF. FOR VISCOSITY OF WATER EQUATION FOR TEMP. >60F
```

```
      READ 1080, Z1, Z2, Z3, Z4, Z5
C        CHARACTERISTICS OF HEAT TRANSFER FLUID
C        N IS THE NUMBER OF FLUIDS TO BE COMPARED THRU THE COLLECTOR
      N= 3
C        A1 THRU A5 ARE THE FLUID COEFF. FOR THE SPECIFIC HEAT EQUATIONS
      READ 1080,(A1(I),A2(I),A3(I),A4(I),A5(I),I= 1,N)
C         E1 -E5 -COEFF. FOR CONDUCTIVITY EQUATION OF FLUID
      READ 1080,(E1(I),E2(I),E3(I),E4(I),E5(I),I= 1,N)
C        F1-F5 -COEF. FOR VISCOSITY EQUATION OF FLUID
      READ 1080,(F1(I),F2(I),F3(I),F4(I),F5(I),I= 1,N)
C        DEN1-DEN5- COFF. FOR DENSITY EQUATION OF THE FLUID
      READ 1080,(DEN1(I),DEN2(I),DEN3(I),DEN4(I),DEN5(I),  I= 1,N)
 1080 FORMAT( 5E10.4)
C        QFAC- VARIABLE HOT WATER LOAD ARRAY
      READ 197,(QFAC(J),J= 1,24)
 197  FORMAT(12F6.2)
C        CHARACTERISTICS OF ENVIRONMENTAL PARAMETERS
C        XPHASE AND XOMEGA -USED IN DETERMINING AMBIENT AIR TEMP.
      XPHASE= - 3. * PI/2.
      XOMEGA= PI/ 12.
C        LOOP TO READ IN INCIDENT RAD. AND AMBIENT TEMP.
      DO 1090 J= 1,24
C        QINCID-AMOUNT OF INCIDENT RADIATION RECEIVED AT COLLECTOR SITE.....
C         FOR THIS CASE IT IS SYMMETRIC AROUND 12:00
      QINCID(J)= 750. *COS((PI *J/12.)-PI)
C        TAMB- AMBIENT TEMP. AT COLLECTOR IN C
C        TAMB IS ASSUMED TO BE SINUSOIDAL, WITH AMAX AT 18:00 AND MIN. AT 6:00
      TAMB(J)=  5.+ 5.* COS(XPHASE + J * XOMEGA)
C        IF THE INCIDENT RADIATION IS NEGATIVE IT IS ASSUMED TO BE 0
      IF( QINCID(J) .GT. 0) GO TO 1090
      QINCID(J)= 0.
 1090 CONTINUE
C        TSTART-TEMP. DIFF. BETWEEN WATER TEMP. AND FLUID TEMP. IN ORDER TO TURN
C        PUMP ON
      TSTART= 18.
C        TSTOP -TEMP. DIFF. BETWEEN WATER TEMP. AND FLUID TEMP. IN ORDER TO TURN
C        PUMP OFF
      TSTOP= 3.
C        THIGH- MAX. ALLOWABLE FLUID TEMP. OR ELSE BOILING AND PRESSURE
C        EFFECTS WILL BECOME NOTICEABLE
      THIGH= 90.
C         NFLOW- NO. OF FLOWRATES TO BE USED
      NFLOW= 1
C        G-ARRAY WHICH HOLDS VARYING FLOWRATES OF FLUID
      READ 1078,(G(JL),JL=1,NFLOW)
 1078 FORMAT(F10.4)
C        NSIZE- NO. OF DIFFERENT SIZES OF COLLECTORS
      NSIZE= 1
C        AREA- ARRAY WHICH HOLDS SIZES OF COLLECTORS
      READ 1077,(AREA(JA),JA= 1, NSIZE)
 1077 FORMAT(4F10.4)
C        NTOP- NO. OF DIFFERENCES BETWEEN CAPACITY RATES.
      NTOP= 1
C        FUNC- AN ARRAY WHICH HOLDS DIFF. BETWEEN CAPACITY RATES
      READ 1078,(FUNC(IL),IL=1,NTOP)
      PRINT 1913
 1913 FORMAT(50X,'COLLECTOR TUBE CHARACTERISTICS'/10X,'O.D.TUBE(M)',10X,
```

```
     1'I.D.TUBE(M)',10X,'SPACING OF TUBING(M)')
      PRINT 1914,DIAMO,DIAMI,W
 1914 FORMAT( 7X,F10.4,11X,F10.4,16X,F10.4/)
C         LOOP FOR THE DIFFERENT FLUIDS
      DO 850 JK= 1,N
C         LOOP TO VARY SIZE OF COLLECTOR
      DO 1890 JA= 1, NSIZE
C         AREACO- AREA OF COLLECTOR IN M**2
      AREACO= AREA(JA)
C         K- RATIO OF THE TANK HEIGHT TO RADIUS
      K= 6
C         DIAMT - DIAM. OF TANK IN M
      DIAMT= ((2.8781 *AREACO/(K * PI))**(1./3.)) * 2.
      DIAMT= DIAMT* .3048
C         TANKH- HEIGHT OF TANK IN M
      TANKH= K * DIAMT/2.
C         OAREAT- TOTAL SURFACE AREA OF TANK IN M**2
      OAREAT= PI * DIAMT * TANKH + PI * (DIAMT**2.)/2.
      PRINT 1911
 1911 FORMAT(50X,'TANK DIMENSIONS'/10X,'HEIGHT(M)',10X,'DIAMETER(M)',
     110X,'TOTAL SURFACE AREA(M**2)')
      PRINT 1912,TANKH,DIAMT,OAREAT
 1912 FORMAT(6X,F10.4,7X,F10.4,13X,F10.4/)
C         XNTUBE- NO. OF TUBES IN COLLECTOR
      XNTUBE= .5 * AREACO
C         XLCOL-LENGTH OG COLLECTOR TUBES(FT)
      XLCOL=  AREACO/(W * XNTUBE)
C         CCOL- COLLECTOR UNIT COSTS
      CCOL= 100.
      CCOST=  CCOL * AREACO
      IZCONT= 1
C         XNEX- NO. OF TUBES IN EX'ER
      XNEX= 30.
      IF( AREACO .GE. 40) XNEX= .7 * AREACO
C         AREAEX IS THE AREA OF THE EXCHANGER IN M**2
      AREAEX= .08 * AREACO
C         TUBLEN- LENGTH OF TUBES IN EX'ER
      TUBLEN = AREAEX/(PI * SSIZE * XNEX)
      CEX= 110.
      CTEX= CEX * AREAEX
      TCOL= CCOST + CTEX
      CRAT= CCOL/ CEX
C         LOOP FOR VARYING THE MATCHED FLOWRATES
      DO 795 IL= 1,NTOP
C         JL- COUNTER FOR NUMBER OF FLOWRATES
      JL= 1
C         ICOUNT-COUNTER FOR KEEPING TRACK OF FLOWRATES
      ICOUNT= 0
C         LOOP TO VARY FLOWRATE OF FLUID THRU COLLECTOR
 1062 CONTINUE
C         TWAT- STARTING TEMP. OF WATER IN TANK.
      TWAT= 60.
C         TLIQ IS THE TEMP OF FLUID ENTERING EXCHER IN C
      TLIQ= TWAT
C         TLIQIN IS THE TEMP OF FLUID LEAVING EXCHANGER
      TLIQIN= TLIQ
C
```

```
C        IX- COUNTER FOR CONVERGENCE LOOP FOR MULTIPLE DAY RUNS
      IX= 0
C      AN ITERATION BEGINS WHICH WILL CONTINUE UNTIL THE DAY'S TEMP, AND THE
C        PREVIOUS DAY'S TEMP ARE WITHIN AN ACCEPTABLE LIMIT.....
 1063 CONTINUE
C        IF IT TAKES MORE THAN 7 DAYS TO CONVERGE IT IS TOO LONG
      IF( IX .GE. 7) GO TO 1822
 1064 CONTINUE
      IX= IX + 1
C        ICOUNT-COUNTER FOR KEEPING TRACK OF FLOWRATES
      IF( ICOUNT .NE. 0) GO TO 1065
      JL= 1
 1065 CONTINUE
C        QTTRIM- TOTAL RATE OF HEAT TRANSFERRED /DAY FROM FLUID TO TANK WATER
      QTTRIM= 0
C         QTINC- TOTAL INCIDENT RAD. FOR 1 DAY
      QTINC= 0.
C         ATLOAD-TOTAL LOAD FOR HOME HEATING FOR ONE DAY
      ATLOAD= 0.
C         QCOL-TOTAL RATE AT WHICH HEAT IS COLLECTED(WATTS/D
      QCOL= 0.
C        PUMP- CONTROLS WHETHER PUMP IS ON OR OFF.
      PUMP= 0
C        IC- COUNTER FOR NO. OF HOURS PUMP IS ON PER DAY
      IC= 0
C        LOOP TO VARY TIME OF DAY
C        STARTING WITH 1:00 TO 24:00
      DO 1830 J= 1,24
      QC2= 1.
C        TEMP.-AN ARRAY WHICH HOLDS ALL DAILY TEMPERATURES UNTIL THE TEMP.
C        CONVERGE......
      TEMP( IX, J) = TWAT
C        QINCID IS INCIDENT RADIATION RECIEVED BY COLLECTOR IN WATTS/M**2
      QI= QINCID(J)
C        TAMBCN- AMBIENT AIR TEMP. FOR GIVEN IN C
      TAMBCN= TAMB(J)
C        PUMP-SIGNIFIES WHETHER PUMP IS ON OR OFF.
      IF( PUMP .NE. 0) GO TO 1854
C        TLIQ-TEMP. OF FLUID WHEN PUMP IS OFF,ASSUMING ISOTHERMAL COLLECTOR
      TLIQ=(QI  /USLOPE)+ TAMBCN
C        THIGH- MAX. ALLOWABLE FLUID TEMP. OR ELSE BOILING AND PRESSURE
C        EFFECTS WILL BECOME NOTICEABLE
      IF( TLIQ .GE. THIGH) TLIQ= THIGH
C        TLIQIN- TEMP. OF FLUID AT ENTRANCE TO COLLECTOR
      TLIQIN= TWAT
      TENT1= TLIQ - TWAT
C        IF THE TEMP. OF THE FLUID IN THE COLLECTOR IS MUCH GREATER THAN THAT OF
C        THE WATER IN THE STORAGE TANK THEN THE PUMP IS TURNED ON.
      IF( TENT1 .LE. TSTART) GO TO 1099
C        LOOP TO FIND FLOWRATE FOR TURBULENT FLOW
 1854 CONTINUE
C        IF PUMP IS 1  PUMP IS ON,PUMP=0,PUMP OFF.
      PUMP= 1
      TLIQI1=  1.8 * TLIQIN + 32.
      SH66= B1 + B2 * (B3 + TLIQI1)+ B4 *(B5+ TLIQI1) **2.
      SH77= A1(JK)+ A2(JK) *(A3(JK)+ TLIQI1)+ A4(JK)*(A5(JK)+TLIQI1)**2.
      SH6= SH66 * 1.162
```

```
      SH7= SH77 * 1.162
C        CAPOPT- MEAN OPERATING CAPACITY RATE
      CAPOPT= SH6 * G(JL)* AREACO
C        CAP1- CAPACITY RATE OF COLLECTOR LOOP
      CAP1= CAPOPT *( 1. + FUNC(IL))
C        G1 - FLUID FLOWRATE IN KG/HR
      G1= CAP1/ SH7
C        G11- FLOWRATE OF FLUID THRU COLLECTOR IN KG/HR-M**2.
      G11= CAP1/(SH7 * AREACO)
      GACT= CAP1/SH6
C        G111-FLOWRATE OF FLUID IN EACH TUBE OF COLLECTOR,ASSUMING FLOW IS
C        EVENLY DISTRIBUTED THRU COLLECTOR
      G111= 4. *G1/(PI * XNTUBE *(DIAMI**2.))
C        G112-FLOWRATE OF FLUID THRU TUBES IN COLLECTOR IN LB/FT**2.-HR
      G112= G111/4.883
C        DETERMINATION OF COLLECTOR EFFICIENCY
      COEF= SQRT(USLOPE/(COND * SIGMAO))
      ETA1= TANH(COEF * XLENGT)/(COEF * XLENGT)
      YPART1= 1./((1. - DIAMO/W)* ETA1)
      YPART2= 1./(((W * USLOPE)/CSUBS)+ YPART1)
      YPART3= 1./((DIAMO/W)+ YPART2)
C        ITERATION TO DETERMINE TEMP. DISTRIBUTION OF  FLUID IN EX'ER
C        TLI-WALL TEMP. OF FLUID IN EX'ER IN C
      TLI= TWAT
C        TWA-WALL TEMP. OF WATER  IN EX'ER IN C
      TWA= TLIO
C        TWAFIN- TEMP. OF WATER LEAVING EX'ER
      TWAFIN= TWAT
C         IC5-COUNTER FOR NO. OF TIMES LOOP PERFORMED
      IC5= 0
  720 CONTINUE
      IC5= IC5+ 1
C        TWMEAN- MEAN BULK TEMP OF WATER INEX'ER
      TWMEAN= (TWAT + TWAFIN)/2.
C        TWMEA1 -TWMEAN IN F
      TWMEA1= 1.8 * TWMEAN+ 32.
C         TLMEAN - MEAN TEMP OF FLUID WITHIN EX'ER
      TLMEAN= (TLIO + TLIOIN)/2.
C        TLMEA1- TLMEAN IN F
      TLMEA1= 1.8* TLMEAN+ 32.
C        TLI1- TLI IN F
      TLI1= 1.8*TLI+ 32.
C        TWA1- TWA IN F
      TWA1= TWA * 1.8+ 32.
C        VISFL1- VISCOSITY OF FLUID IN LB/FT-HR
      VISFL1=F1(JK)+F2(JK)*(F3(JK)+TLMEA1)+ F4(JK)*(F5(JK)+TLMEA1)**2.
      VISFL1= 1./ VISFL1
      SH55= A1(JK)+A2(JK)*(A3(JK)+TLMEA1)+A4(JK)*(A5(JK)+TLMEA1)**2.
      SH5= SH55 * 1.162
C        CONFL1-CONDUCTIVITY OF FLUID IN BTU/HR-FT-F
      CONFL1= E1(JK)+E2(JK)*(E3(JK)+TLMEA1)+E4(JK)*(E5(JK)+TLMEA1)**2.
      DENFL1= DEN1(JK)+DEN2(JK)*(DEN3(JK)+TLMEA1)+DEN4(JK)*(DEN5(JK)+
     1TLMEA1)**2.
C        REYNOL - COLLECTOR REYNOLDS NO.
      REYNOL = G112 * DIAMX/VISFL1
      IF( REYNOL .LT. 2500) GO TO 157
C        FRIC1- FRICTOON FACTOR IN COLLECTOR TUBES
```

```
      FRIC1= .0014 + .125/(REYNOL **.32)
      PRANDT= SH55 * VISFL1/CONFL1
      IF( REYNOL .GT. 7100) GO TO 166
C        TRANSITION REGION
      QJPRIM= .116*(((REYNOL)**(2./3.))-125.)/  REYNOL
      XHCW= ( PRANDT**(-2./3.)) * SH55 * G112 * QJPRIM
      XHCW= XHCW * 5.678
      GO TO 158
 166  CONTINUE
C         TURBULENT REGION
      XHCW= CONFL1 * .023 *(REYNOL **.8) * ( PRANDT**.4)/DIAMX
      GO TO 158
 157  CONTINUE
C        LAMINAR REGION
      FRIC1= 16./REYNOL
      G121= G1/(XNTUBE * .4536)
      TUBECO= AREACO/(((2 * XLENGT + DIAMU)* XNTUBE)* .3048)
      CPART1= G121 * SH55/(CONFL1 * TUBECO)
      CPART2=  1.75 * CPART1 **(1./3.)
      IF( CPART2 .LE. 3.66) CPART2= 3.66
      XHCW= CONFL1 * CPART2 / DIAMX
C        XHCW- INSIDE TUBE HEAT TRANSFER COEFF. IN COLLECTOR
      XHCW= XHCW * 5.678
 158  CONTINUE
C        XPD- PRESSURE DROP IN EACH COLLECTOR TUBE
      XPD= FRIC1 * XLCOL *(G112**2.)/(2.*DIAMX * ACCG* DENFL1 *144.)
C        CXPD- TOTAL PRESSURE DROP IN COLLECTOR
      CXPD= XNTUBE * XPD
C        VFLOW1- FLOWRATE OF FLUID THRU COLLECTOR IN FT**3/SEC
      VFLOW1= G1 * 2.2046/(DENFL1 * 3600.)
C        XHP1- HORSEPOWER REQUIRED TO CIRCULATE FLUID THEU COLLECTOR
      XHP1= CXPD * VFLOW1 * 144./550.
      ETA2= 1./(((W * USLOPE)/(PI * DIAMI * XHCW))+ YPART3)
C        DETERMINATION OF ETA3
      AX1= -(ETA2 * USLOPE)/(G11 * SH5)
      AX2= 1 - EXP(AX1)
      ETA3= (G11 * SH5/USLOPE) * AX2
      XPAR= ETA3 * AREACO * QI
      QC= XPAR - USLOPE * AREACO * ETA3 * ( TLIQIN - TAMBCN)
      TLIQ= TLIQIN + QC/CAP1
      IF( TLIQ .GE. THIGH) TLIQ = THIGH
C        IF TEMP. OF LIQUID ENTERING EX'ER IS LESS THAN THAT OF WATER ENTERING
C        EX'ER THEN THE PUMP WILL NOT BE TURNED ON.
      TENT2= TLIQ - TWAT
      IF( TENT2 .LE. TSTOP) GO TO 1100
C        SH22 IS THE SPECIFIC HEAT IN BTU/(LB F)
      SH22=  B1+ B2 * (TWMEA1 + B3) + B4 * (TWMEA1+ B5) **2
C        SH2 IS THE SPECIFIC HEAT OF WATER IN WATT HR/( KG C)
      SH2= 1.162 * SH22
      DENW1= R1 + R2 *(R3+ TWMEA1)+R4*(R5+ TWMEA1)**2.
      CONDW= D1+ D2 *(TWMEA1+ D3) + D4*(TWMEA1+ D5) **2
C        VISB2 IS THE VISCOSITY OF WATER AT THE WATER TEMP. IN LB/(FT HR)
      VISB2= Z1+ Z2*( Z3+TWMEA1) + Z4*(Z5+TWMEA1)**2.
C        VISB IS THE VISCOSITY OF WATER IN CENTIPOISES
      VISB2= 1./ VISB2
C        CAP2-CAPACITY RATE OF WATER AS A FUNCTION OF THE CAP. RATE OF FLUID
C        CAP2  IN WATTS/ C
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```
      CAP2= CAPOPT * ( 1.- FUNC(IL))
C        G2 IS THE WATER FLOWRATE IN KG/HR
      G2= CAP2/SH2
C        G222 IS THE WATER FLOWRATE IN KG/ HR* M**2
      G222= G2 * XNUT/XNEX
C        G22 IS CONVERTED TO LB/(HR * FT**2)
      G22= G222/4.883
C        DETERMINATION OF WHICH FLUID FLOWRATE IS SMALLEST
      IF( CAP1 .GE. CAP2) GO TO 725
      CAPMIN= CAP1
      CAPMAX= CAP2
      GO TO 750
 725  CAPMIN= CAP2
      CAPMAX= CAP1
 750  CONTINUE
      DELQ= ( QC - QC2)/ QC2
      DELQ= ABS( DELQ)
      IF ( DELQ  .LE. .001)  GO TO 735
C        CONVERSION OF LENGTH MEASURES TO METERS
      RXD= XD/2.
C        XEXC- THICKNESS OF EXHER WALL IN M
      XEXC= .3048 * .03/12.
C        DX IS THE OUTSIDE DIAMETER IN METERS
      DX= SSIZE
      RDX= DX/2.
C        HOS IS THE SHELLSIDE SCALING COEFF. IN WATTS/ C M**2
      HOS= 5678.
C        DETERMINATION OF SHELL SIDE HEAT TRANSFER COEFFICIENT
C        DEN1-DEN5-COEFF. FOR DENSITY OF FLUID THRU COLLECTOR
      DENFL5= DEN1(JK)+DEN2(JK)*(DEN3(JK)+TLMEA1)+DEN4(JK)*(DEN5(JK)+
     1TLMEA1)**2.
      CONFL5= E1(JK)+ E2(JK)*(E3(JK)+TLMEA1)+E4(JK)*(E5(JK)+TLMEA1)**2.
      VISFL5= F1(JK)+F2(JK)*(F3(JK)+TLMEA1)+F4(JK)*(F5(JK)+TLMEA1)**2.
      VISFL5= 1./VISFL5
      SH98= A1(JK)+A2(JK)*(A3(JK)+TLMEA1)+A4(JK)*(A5(JK)+TLMEA1)**2.
C        BAF- BAFFLE SPACING IN FT
      BAF= 1.
C        NBAF- NO. OF BAFFLES IN HEAT EX'ER
      NBAF= TUBLEN    /( BAF * .3048)
      NROW= ( 7./40.) * AREACO
C        AMIN- MIN. AREA THRU WHICH FLUID FLOWS
      AMIN= BAF * SMIN
C        GMAX- MAX. ALLOWABLE FLOWRATE OF FLUID ON SHELL SIDE.
      GMAX= ( G1 * 2.2046)/(AMIN * ( NROW + 1))
C        OREY- REYNOLDS NO. ON SHELL SIDE FOR FLUID IN EX'ER
      OREY= GMAX * SIZE/VISFL5
C        OPRAN- PRANTDL NO. ON SHELL SIDE FOR FLUID IN EX'ER
      OPRAN= SH98 * VISFL5/CONFL5
      IF( OREY .LT. 2500) GO TO 163
      FRIC2= .0014 + .125/(OREY**.32)
      GO TO 164
 163  CONTINUE
      FRIC2= 16./OREY
 164  CONTINUE
C        SPD- SHELL SIDE PRESSURE DROP
      SPD= FRIC2 *(GMAX**2.)*DSHEL*(NBAF + 1.)/(2.*ACCG*DENFL5*SIZE*144)
C        HO IS THE SHELL SIDE COEFF. IN WATTS/ C M**2
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      HO= .33*(OREY**.6)*(OPRAN**(1./3.)*CONFL5/SIZE
C        VISFLB-VISCOSITY OF FLUID AT WALL TEMP.
      VISFLB= F1(JK)+F2(JK)*(F3(JK)+TLI1 )+F4(JK)*(F5(JK)+TLI1)**2.
      VISFLB= 1./VISFLB
C        VPRIM-FACTOR ACCOUNTING FOR VARIATION OF VISCOSITY WITH TEMP.
      VPRIM= ABS(VISFLB/VISFL5)
      VPRIM= VPRIM ** -.14
      HO= HO * 5.678 * VPRIM
C        HIS IS THE INSIDE TUBE SCALING COEFF IN WATTS/C M**2
      HIS= 5678.
      HIS= HIS * XD/DX
C        RWALL- RESISTANCE OF WALL TO HEAT TRANSFER
      RWALL= ( RDX * ALOG( DX/ XD))/ KEXC
      RWALL= RWALL * DX/XD
C        DETERMINATION OF INSIDE  TUBE HEAT TRANSFER COEFFICIENT
      TUBLEX= TUBLEN/.3048
C        REYNO - REYNOLDS NO.  WITHIN EXHER TUBES
      REYNO= G22 * XDP/VISB2
      IF( REYNO .LT. 2500) GO TO 161
      FRIC3= .0014+ .125/(REYNO**.32)
      PRAND= SH22 * VISB2/CONDW
      IF( REYNO .GT. 7100) GO TO 167
C        TRANSITION REGION
      PJPRIM= .116 * ((REYNO **(2./3.)) - 125.)/REYNO
      HI= (PRAND **(-2./3.)) * SH22 * G22 * PJPRIM
      GO TO 162
  167 CONTINUE
C        TURBULENT REGION
      HI= CONDW * .023 * ( REYNO**.8) *(PRAND **.4)/XDP
      GO TO 162
  161 CONTINUE
C        LAMINAR REGION
      FRIC3= 16./REYNO
      G212= G2/(XNEX * .4536)
      DPART1= (G212 * SH22)/(CONDW * TUBLEX)
      DPART2= 1.75 * (DPART1) **(1./3.)
      IF( DPART2 .LE. 3.66) DPART2= 3.66
      HI= CONDW * DPART2 / XDP
  162 CONTINUE
C        HI-INSIDE TUBE HEAT TRANSFER COEFF.
      HI= HI * XE/DX
      TBEPD= FRIC3*(G22**2.)*TUBLEX/(2.*ACCG*DENW1*XDP *144.)
C        TBEPD- PRESSURE DROP WITHIN EXER TUBES
      TBEPD= TBEPD * XNEX
      VFLOW2= G2 * 2.2046/(DENW1 * 3600.)
      XHP2= TBEPD * VFLOW2 * 144./ 550.
C        VISBA- VISCOSITY OF WATER AT WALL TEMP.(LB/FT HR)
      VISBA=  Z1+ Z2*(Z3+ TWA1)+Z4*(Z5+TWA1)**2.
      VISBA= 1./ VISBA
C        VPRIME-FACTOR ENCOMPASSING INFLUENCE OF VISCOSITY
      VPRIME= (VISBA/VISB2)**-.14
      HI= HI * 5.678 * VPRIME
C        UX- OVERALL HEAT TRANSFER COEFF. IN EX'ER IN WATTS/M**2 C
      UX= 1./((1./HI)+(1./HIS)+ (1./HOS)+(1./HO)+  RWALL)
C        DETERMINATION OF EXCHANGER EFFECTIVENESS
      IF( CAPMIN .NE. CAPMAX) GO TO 680
C        IF THE CAP. RATES MATCH THEN THE EX'ER  EFFECTIVENESS IS THE FOLLOWING
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      EXCHEF= 1./(1.+(CAPMIN/(UX * AREAEX)))
      GO TO 682
  680 CONTINUE
C         IF THE CAP. RATES DO NOT MATCH THE EX'ER EFFECTIVENESS IS THE FOLLOWING
      Z= 1. - (CAPMIN/CAPMAX)
      Y= -(AREAEX* UX/CAPMIN) *Z
      X= 1.-((CAPMIN/CAPMAX) * EXP(Y))
      EXCHEF= (1.- EXP(Y))/X
  682 CONTINUE
C         TWAFIN- TEMP. OF WATER LEAVING EXCHANGER
      TWAFIN= TWAT + EXCHEF*(CAPMIN/CAP2)* (TLIQ - TWAT)
      TLIQIN= TLIQ - EXCHEF * (CAPMIN/CAP1) *( TLIQ - TWAT)
      QC2= XPAR- USLOPE * AREACO * ETA3 * ( TLIQIN- TAMBCN)
      TLMEAN= (TLIQ + TLIQIN)/2.
      TLMEA1= 1.8* TLMEAN+ 32.
      SH55= A1(JK)+A2(JK)*(A3(JK)+TLMEA1)+A4(JK)*(A5(JK)+TLMEA1)**2.
      SH5= SH55 * 1.162
      TLIQ= TLIQIN + QC/CAP1
      IF( TLIQ .GE. THIGH) TLIQ = THIGH
      TENT3= TLIQ - TWAT
      IF (TENT3 .LE. TSTOP) GO TO 1100
      DELQ= ( QC- QC2)/ QC2
      DELQ= ABS( DELQ)
      IF ( DELQ  .LE. .001)  GO TO 735
C         DETERMINATION OF EXCHANGER EFFECTIVENESS
      IF( CAPMIN .NE. CAPMAX) GO TO 751
C         IF THE CAP. RATES MATCH THEN THE EX'ER  EFFECTIVENESS IS THE FOLLOWING
      EXCHEF= 1./(1.+(CAPMIN/(UX * AREAEX)))
      GO TO 752
  751 CONTINUE
C         IF THE CAP. RATES DO NOT MATCH THE EX'ER EFFECTIVENESS IS THE FOLLOWING
      Z= 1. - (CAPMIN/CAPMAX)
      Y= -(AREAEX* UX/CAPMIN) *Z
      X= 1.-((CAPMIN/CAPMAX) * EXP(Y))
      EXCHEF= (1.- EXP(Y))/X
  752 CONTINUE
      TWAFIN= TWAT + EXCHEF*(CAPMIN/CAP2)* (TLIQ - TWAT)
      TLIQIN= TLIQ - EXCHEF * (CAPMIN/CAP1) *( TLIQ - TWAT)
C         DETERMINATION OF ETA4
      V= (CAP1/(CAPMIN * EXCHEF))- 1.
      V1= 1+ (( ETA3 * USLOPE * AREACO)/ CAP1) * V
C         ETA4-FINAL  PENALTY IMPOSED BY HEAT EX'ER ON HEAT TRANSFER,
      ETA4= ETA3/ V1
C         QCFIN- RATE OF HEAT TRANSFERRED FROM COLLECTOR TO TANK ASSUMING
C         NO HEAT LOSSES IN PIPES ONLY IN THE TANK ITSELF.
      QCFIN= ETA4 * AREACO * ( QI - USLOPE *(TWAT    - TAMBCN))
C         TWA-WALL TEMP. OF WATER  IN EX'ER IN C
      TWA= TWMEAN+ QCFIN/(HI * AREAEX)
C         TLI-WALL TEMP. OF FLUID IN EX'ER IN C
      TLI= TLMEAN - QCFIN/(HO * AREAEX)
      GO TO 720
  735 CONTINUE
  736 CONTINUE
C         IF CONTROL HAS PASSED TO THIS LOCATION THEN THE PUMP HAS BEEN ACTIVATED
C         AND HEAT IS TRANSFERRED TO THE TANK
      PUMP= 1
C         IC- COUNTER FOR NO. OF HOURS PUMP IS ON PER DAY
```

```
      IC= IC + 1
C         DETERMINATION OF ETA4
C          TCPD= TOTAL COLLECTOR LOOP PRESSURE DROP
      TCPD= CXPD + SPD
      XHP3= TCPD * VFLOW1 * 144./550.
C         COP1= TOTAL COST TO PUMP FLUID FOR ONE YEAR
      COP1= .03 * 8. * 365. * .7457 * XHP3/.7
      V= (CAP1/(CAPMIN * EXCHEF))- 1.
      V1= 1+ (( ETA3 * USLOPE * AREACO)/ CAP1) * V
C         ETA4=FINAL  PENALTY IMPOSED BY HEAT EX'ER ON HEAT TRANSFER,
      EFIN= 1./V1
      ETA4= ETA3/ V1
      QCFIN= ETA4 * AREACO * ( QI - USLOPE *(TWAT   - TAMBCN))
      GO TO 1110

 1099 CONTINUE
C        IF CONTROL HAS PASSED TO THIS LOCATION THEN THE PUMP WASN'T ACTIVATED...
C          BLANKING OUT VARIABLES NOT USED IF PUMP IS OFF
      REYNOL= 0.
      [illegible] 0.
        [illegible]1= 0.
      ETA3= 0.
      PRANDT= 0.
      XHCW= 0.
      QC= 0.
      QC2= 0.
 [illegible] CONTINUE
C         IC= COUNTER FOR NO. OF HOURS PUMP IS ON PER DAY
      IC= IC
      PUMP= 0
      QCFIN = 0.
      VISH2= 0.
      G222= 0.
      HI= 0.
      RWALL= 0.
      UX= 0.
      EXCHEF= 0.
      ETA4= 0.
      REYNO= 0.
      G22= 0.
      OREY= 0.
      HO= 0.
      CAP1= 0
      CAP2= 0
      TLMEAN= 0.
      TWMEAN= 0.
      TWA= 0.
      TLI= 0.
 1110 CONTINUE
C         QTTRIM= TOTAL HEAT TRANSFERRED FOR 1 DAY
      QTTRIM= QTTRIM + QCFIN  * DELTIM * 3600.
C          QTINC= TOTAL INCIDENT RAD. FOR 1 DAY
      QTINC= QTINC + QI * DELTIM * 3600. * AREACO
      DENSW= R1 + R2 *(R3+ TWAT) + R4* ( R5 + TWAT) **2.
C         DENSW= DENSITY OF WATER IN KG/M**3
      DENSW= 16.05212 * DENSW
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```
      SH55= B1 + B2 *( B3+ TWAT) + B4 *(B5 + TWAT) **2.
C        SH5= SPECIFIC HEAT OF WATER
      SH5= SH55 * 1.162
      PFAC= QFAC(J)
      RBASE= 50. * AREACO
C        PLOAD= VARIABLE HOT WATER LOAD
      PLOAD= PFAC * RBASE
C         ATLOAD=TOTAL LOAD FOR HOME HEATING FOR ONE DAY
      ATLOAD= ATLOAD + 3600. * PLOAD * DELTIM
      FEX= TCOL/ QTTRIM
C        VOLW= VOLUME OF WATER INSIDE TANK ASSUMING NO EXPANSION OF WATER WITH
C        TEMPERATURE AND FILLED TO THE TOP OF THE TANK
      VOLW= PI * ((DIAMT **2.)/4.) * TANKH
      ALPH= ( QCFIN - PLOAD) /(SH5 * DENSW * VOLW)
      TW(J)= TWAT
C        TWATNE= TEMP OF WATER AT END OF 1 HOUR
      TWATNE= TWAT + DELTIM * ALPH
      TWMAX= 70.
      TWAT= TWATNE
      IF( TWAT .GE. TWMAX) TWAT= TWMAX
      TL(J)= TLIO
      P1(J)= CXPD
      P112(J)= TCPD
      P2(J)= TLMEAN
      P22(J)= TWMEAN
      GM1(J)= XHP1
      GM2(J)= XHP2
      REYN(J)= QCFIN
      PRAN(J)= EFIN
      XH(J)= DENFLS
      VIS1(J)= ETA3
      EC(J)= CAP2
      QCP(J)= TBEPD
      QCP2(J)= G1
      QC3(J)= G112
      VIS2(J)= CAP1
      REY(J)=  G2
      XH1(J)=    G22
      EXC(J)= EXCHEF
      U(J)= REYNO
      TL1(J)= XHCW
      XLS(J)=  HI
      XLO(J)=  HO
      WLS(J)= UX
      WAS(J)= OREY
      RLS(J)= REYNOL
      TH(J)= COP1
      EXA(J)= VISR2
      TRET(J)= CONDW
      QAR(J)= ETA2
      ALO(J)= VISFLS
      QHP(J)= CONFLS
 1830 CONTINUE
      IF( IX .LE. 1) GO TO 1063
      J= 1
 1821 CONTINUE
      IF( J .GE. 25) GO TO 1822
```

```
      DIFF= TEMP(IX,J) - TEMP(IX-1,J)
      ABS1= ABS( DIFF)
C        IF ALL TEMP. MATCH  WITHIN A CERTAIN TOLERANCE BETWEEN THOSE OF ONE
C        DAY AND THOSE OF THE PREVIOUS THEN THE PROGRAM CONTINUES  TO THE
C        NEXT ITERATION
      IF( ABS1 .GT. 2) GO TO 1063
      J= J + 1
      GO TO 1821
 1822 CONTINUE
      PRINT 2
 2    FORMAT(1H0,49X,'EX ER CHARACTERISTICS'/3X,'O.D.TUBE(M)',4X,'I.D.TU
     1BE(M)',6X,'NO.TUBES',5X,'TUBE LENGTH(M)',2X,'EXER AREA(M2)')
      PRINT 3,SSIZE,XD,XNEX,TUBLEN,AREAEX
 3    FORMAT(E12.5,4(4X,E12.5)//)
      PRINT 1043
 1043 FORMAT(  1X,'COLLECTOR AREA(M**2)',2X, 'FLUID FLOWRATE(KG/HR)',2X,
     1'WATER FLOWRATE(KG/HR)'2X,'CAP.RATE RATIO')
      PRINT 1044,AREACO,G1,G2,CXYRAT
 1044 FORMAT(5X,2(E10.4,8X),8X,E10.4,10X,E10.4)
      PRINT 1042
 1042 FORMAT(50X,'HEAT TRANSFER CHARACTERISTICS(JOULES/DAY)')
      PRINT 1040
 1040 FORMAT(3X, 'INC.RAD',2X, 'HEAT COLL.',2X, 'HEAT TRAN.', 2X,
     1'HEATING LOAD',2X,'COST TO HEAT TRANSFERRED RATIO')
      PRINT 1041,QTINC,QCOL,QTTRIM,ATLOAD,FEX
 1041 FORMAT(E10.4,3(2X,E10.4),6X,E10.4/////)
      PRINT 4,(J,TW(J),TL(J),TL1(J),XLS(J),XLD(J),WLS(J),VIS1(J),EXC(J),
     1PRAN(J),REYN(J),J= 1, 24)
 4    FORMAT(2X,I2,10E10.4)
      PRINT 5
 5    FORMAT(1H1)
      IF( JL .GE. NFLOW) GO TO 795
      JL= JL + 1
      ICOUNT= ICOUNT + 1
      GO TO 1062
 795  CONTINUE
 1238 CONTINUE
 1234 CONTINUE
 1890 CONTINUE
 850  CONTINUE
      END
```

PILATION: NO DIAGNOSTICS.

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 07/15/77  12:43:19 FRDA02 0293AAAHSEMI 000293            S29   60

@MAP,SX    ,NAME
MAP28R1C RI 72-8 07/15/77 12:44:16
   1.           IN TPF$,


ADDRESS LIMITS     001000 015661       6578 IBANK WORDS DECIMAL
                   040000 050005       4102 DBANK WORDS DECIMAL
STARTING ADDRESS   012456



                SEGMENT $MAIN$           001000 015661       040000 050005

NSWTC$/FOR69              $(1)     001000 001032
NRBLK$/FOR-E2             $(1)     001033 001150      $(0)     040000 040001
                                                      $(4)     040002 040054
NRWND$/FOR-E2             $(1)     001151 001232      $(2)     040055 040066
NWEF$/FOR-E2              $(1)     001233 001466      $(2)     040067 040106
NCLUS$/FOR-E3             $(1)     001467 001732      $(2)     040107 040136
NWBLK$/FOR68              $(1)     001733 002121      $(0)     040137 040140
                                                      $(4)     040141 040212
NBSBL$/FOR-E3             $(1)     002122 002174      $(0)     040213 040213
                                                      $(4)     040214 040251
NUPDA$/FOR68              $(1)     002175 002230
NBDCV$/FOR-E3             $(1)     002231 002361      $(2)     040252 040327
NFCHK$/FOR-E3             $(1)     002362 003352      $(2)     040330 040501
                          $(3)     003353 003353      $(4)     040502 040553
NBF00$/ISD                                            $(2)     040554 042761
NFTV$/FOR-E2              $(1)     003354 003376
NCNVT$/FOR68              $(1)     003377 003620      $(2)     042762 043057
NOUT$/FOR-E3              $(1)     003621 005377      $(2)     043060 043123
NIOER$/FOR-E3             $(1)     005400 005627      $(2)     043124 043273
NFMT$/FOR-E3              $(1)     005630 006512      $(2)     043274 043347
NINPT$/FOR-E3             $(1)     006513 010121      $(2)     043350 043403
NTAB$/ISD                                             $(2)     043404 043443
FORCOM$/FORFTN                                        $(2)     043444 043451
NERR$/FOR-E3              $(1)     010122 010465      $(2)     043452 043642
ERU$/SYS72-8
NERCOM$/FOR-TE3           $(1)     010466 010545      $(2)     043643 043656
FORVCOM$/FOR-TE3                                      $(2)     043657 043666
NSTOP$/FOR-TE3            $(1)     010546 010611      $(2)     043667 043675
ALOG$/FOR-E3              $(1)     010612 010730      $(2)     043676 043736
EXP$/FOR59                $(1)     010731 011020      $(2)     043737 043757
TANH$/FOR59               $(1)     011021 011124      $(2)     043760 044005
NEXP6$/FOR-E3             $(1)     011125 011322      $(2)     044006 044057
NOSYM$/FOR-E3             $(1)     011323 011565      $(2)     044060 044061
SINCOS$/FOR-E3            $(1)     011566 011722      $(2)     044062 044104
SQRT$/FOR59               $(1)     011723 011763      $(2)     044105 044116
NIER$/FOR-E3              $(1)     011764 012141      $(2)     044117 044237
NISYM$/FOR-E3             $(1)     012142 012315      $(2)     044240 044241
NINTR$/FOR-E3             $(1)     012316 012455      $(2)     044242 044272
BLANK$COMMON(COMMONBLOCK)
MAIN                      $(1)     012456 015661      $(0)     044273 050005
                                                      $(2)     BLANK$COMMON

SYS$*RLIB$, LEVEL 72-8
```

*XQT NAME

COLLECTOR TUBE CHARACTERISTICS

O.D.TUBE(M)	I.D.TUBE(M)	SPACING OF TUBING(M)
.0159	.0145	.2159

TANK DIMENSIONS

HEIGHT(M)	DIAMETER(M)	TOTAL SURFACE AREA(M**2)
3.3429	1.1143	13.6527

EX ER CHARACTERISTICS

O.D.TUBE(M)	I.D.TUBE(M)	NO.TUBES	TUBE LENGTH(M)	EXER AREA(M2)
.95250-02	.80010-02	.28000+02	.38192+01	.32000+01

COLLECTOR AREA(M**2)	FLUID FLOWRATE(KG/HR)	WATER FLOWRATE(KG/HR)	CAP.RATE RATIO
.4000+02	.3200+04	.3201+04	1.0000

HEAT TRANSFER CHARACTERISTICS(JOULES/DAY)

INC.RAD	HEAT COLL.	HEAT TRAN.	HEATING LOAD	COST TO HEAT TRANSFERRED RATIO
.8203+09	.0000	.2428+09	.1728+09	.1792-04

HEAT TRANSFER FLUID - WATER

	TWAT	TLIQ	XHCW	HI	HO	UX	ETA3	EXCHEF	EFIN	QCFIN
1	.6523+02	.3706+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
2	.6523+02	.2500+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
3	.6523+02	.1464+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
4	.6523+02	.6699+00	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
5	.6470+02	.1704+00	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
6	.6417+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
7	.6365+02	.3899+02	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
8	.6312+02	.7567+02	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
9	.6259+02	.6452+02	.4343+03	.0000	.0000	.0000	.7992+00	.0000	.9613+00	.0000
10	.6206+02	.6767+02	.4085+03	.4245+04	.5060+04	.1217+04	.7969+00	.5119+00	.9607+00	.1077+05
11	.6332+02	.7014+02	.4142+03	.4292+04	.5096+04	.1223+04	.7975+00	.5130+00	.9609+00	.1307+05
12	.6519+02	.7236+02	.4196+03	.4346+04	.5131+04	.1230+04	.7979+00	.5143+00	.9610+00	.1377+05
13	.6884+02	.7537+02	.4274+03	.4438+04	.5180+04	.1240+04	.7986+00	.5163+00	.9613+00	.1264+05
14	.7000+02	.7536+02	.4285+03	.4452+04	.5186+04	.1241+04	.7987+00	.5165+00	.9613+00	.1035+05
15	.7000+02	.7360+02	.4254+03	.4445+04	.5179+04	.1240+04	.7984+00	.5163+00	.9613+00	.6847+04
16	.7000+02	.7230+02	.4253+03	.0000	.0000	.0000	.7984+00	.0000	.9613+00	.0000
17	.6947+02	.4865+02	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
18	.6788+02	.1000+02	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
19	.6629+02	.9830+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
20	.6523+02	.9330+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
21	.6523+02	.8536+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
22	.6523+02	.7500+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
23	.6523+02	.6294+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000
24	.6523+02	.5000+01	.0000	.0000	.0000	.0000	.0000	.0000	.9613+00	.0000

TANK DIMENSIONS

HEIGHT(M)	DIAMETER(M)	TOTAL SURFACE AREA(M**2)
3.3429	1.1143	13.6527

EX ER CHARACTERISTICS

O.D.TUBE(M)	I.D.TUBE(M)	NO.TUBES	TUBE LENGTH(M)	EXER AREA(M2)
.95250-02	.80010-02	.28000+02	.38192+01	.32000+01

COLLECTOR AREA(M**2)	FLUID FLOWRATE(KG/HR)	WATER FLOWRATE(KG/HR)	CAP.RATE RATIO
.4000+02	.3717+04	.3201+04	1.0000

HEAT TRANSFER CHARACTERISTICS(JOULES/DAY)

INC.RAD	HEAT COLL.	HEAT TRAN.	HEATING LOAD	COST TO HEAT TRANSFERRED RATIO
.8203+09	.0000	.2465+09	.1728+09	.1766-04

HEAT TRANSFER FLUID - 50% ETHYLENE GLYCOL

	TWAT	TLIQ	XHCW	HI	HO	UX	ETA3	EXCHEF	EFIN	QCFIN
1	.6523+02	.3706+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
2	.6523+02	.2500+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
3	.6523+02	.1464+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
4	.6523+02	.6699+00	.0000	.0900	.0000	.0000	.0000	.0000	.9549+00	.0000
5	.6470+02	.1704+00	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
6	.6417+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
7	.6365+02	.3899+02	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
8	.6312+02	.7567+02	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
9	.6259+02	.6457+02	.9096+03	.0000	.0000	.0000	.8181+00	.0000	.9549+00	.0000
10	.6206+02	.6819+02	.7447+03	.4245+04	.3113+04	.1058+04	.8141+00	.4769+00	.9541+00	.1093+05
11	.6336+02	.7078+02	.7809+03	.4294+04	.3140+04	.1064+04	.8150+00	.4782+00	.9543+00	.1326+05
12	.6528+02	.7307+02	.8173+03	.4350+04	.3165+04	.1071+04	.8159+00	.4797+00	.9545+00	.1398+05
13	.6898+02	.7610+02	.8704+03	.4442+04	.3202+04	.1080+04	.8171+00	.4819+00	.9549+00	.1283+05
14	.7000+02	.7583+02	.8662+03	.4454+04	.3203+04	.1081+04	.8170+00	.4819+00	.9549+00	.1052+05
15	.7000+02	.7391+02	.8443+03	.4453+04	.3190+04	.1080+04	.8165+00	.4816+00	.9549+00	.6955+04
16	.7000+02	.7258+02	.8444+03	.0000	.0000	.0000	.8165+00	.0000	.9549+00	.0000
17	.6947+02	.4865+02	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
18	.6788+02	.1000+02	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
19	.6629+02	.9830+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
20	.6523+02	.9330+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
21	.6523+02	.8536+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
22	.6523+02	.7500+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
23	.6523+02	.6294+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000
24	.6523+02	.5000+01	.0000	.0000	.0000	.0000	.0000	.0000	.9549+00	.0000

TANK DIMENSIONS

HEIGHT(M)	DIAMETER(M)	TOTAL SURFACE AREA(M**2)
3.3429	1.1143	13.6527

EX ER CHARACTERISTICS

O.D.TUBE(M)	I.D.TUBE(M)	NO.TUBES	TUBE LENGTH(M)	EXER AREA(M2)
.95250-02	.80010-02	.28000+02	.38192+01	.32000+01

COLLECTOR AREA(M**2)	FLUID FLOWRATE(KG/HR)	WATER FLOWRATE(KG/HR)	CAP.RATE RATIO
.4000+02	.7592+04	.3201+04	1.0000

HEAT TRANSFER CHARACTERISTICS(JOULES/DAY)

INC.RAD	HEAT COLL.	HEAT TRAN.	HEATING LOAD	COST TO HEAT TRANSFERRED RATIO
.8203+09	.0000	.2364+09	.1728+09	.1841-04

HEAT TRANSFER FLUID - MOBILTHERM LIGHT

	TWAT	TLIQ	XHCW	HI	HO	UX	ETA3	EXCHEF	EFIN	QCFIN
1	.6523+02	.3706+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
2	.6523+02	.2500+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
3	.6523+02	.1464+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
4	.6523+02	.6699+00	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
5	.6470+02	.1704+00	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
6	.6417+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
7	.6365+02	.3899+02	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
8	.6312+02	.7567+02	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
9	.6259+02	.6452+02	.4325+03	.0000	.0000	.0000	.7995+00	.0000	.9376+00	.0000
10	.6206+02	.6923+02	.3615+03	.4245+04	.1358+04	.7352+03	.7924+00	.3878+00	.9369+00	.1044+05
11	.6323+02	.7196+02	.3793+03	.4288+04	.1363+04	.7378+03	.7942+00	.3885+00	.9370+00	.1270+05
12	.6501+02	.7420+02	.3950+03	.4340+04	.1372+04	.7419+03	.7958+00	.3898+00	.9372+00	.1342+05
13	.6856+02	.7694+02	.4157+03	.4430+04	.1384+04	.7483+03	.7978+00	.3918+00	.9375+00	.1236+05
14	.7000+02	.7686+02	.4143+03	.4453+04	.1385+04	.7491+03	.7975+00	.3919+00	.9376+00	.1008+05
15	.7000+02	.7462+02	.4002+03	.4453+04	.1381+04	.7480+03	.7962+00	.3916+00	.9376+00	.6659+04
16	.7000+02	.7252+02	.4014+03	.0000	.0000	.0000	.7963+00	.0000	.9376+00	.0000
17	.6947+02	.4865+02	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
18	.6788+02	.1000+02	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
19	.6629+02	.9830+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
20	.6523+02	.9330+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
21	.6523+02	.8536+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
22	.6523+02	.7500+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
23	.6523+02	.6294+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000
24	.6523+02	.5000+01	.0000	.0000	.0000	.0000	.0000	.0000	.9376+00	.0000

APPENDIX E

TRACED TANK MODEL

The following computer program modeled the performance of the single loop traced tank system. Most of this program and the input parameters used have been discussed in other sections of this report. In this particular case, it was used to show the difference in performance between several heat transfer fluids for a collector of $10m^2$. Sample printouts of the results follow the computer program listing for the cases when water, 50% ethylene glycol and mobiltherm light were used within the heat transfer fluid loop.

```
DIMENSION TX(2)
DIMENSION TM(24),C4(24),CX(24),CZ(24),QI1(24),PL(24),FA(24),FP(24)
DIMENSION SH(24) ,CON(24),VIS(24),BE(24),DE(24),CO(24),S(24)
DIMENSION VI(24)
DIMENSION QINCID(24),TMEAN(2)
DIMENSION A1(15), A2(15), A3(15), A4(15), A5(15)
DIMENSION E1(15),E2(15),E3(15),E4(15),E5(15)
DIMENSION F1(15),F2(15),F3(15),F4(15),F5(15)
DIMENSION TEMP( 9, 24)
DIMENSION TAMB(24)
DIMENSION G(10),AREA(10)
DIMENSION TW(24),TL(24),REYN(24),XH1(24),REY(24),XH(24),U(24)
DIMENSION EX(24),EC(24),QC5(24),QC6(24)
DIMENSION QFAC(24)
REAL IC
DIMENSION DEN1(15),DEN2(15),DEN3(15),DEN4(15),DEN5(15)
PI= 3.141593
   ACCG- ACCELERATION OF GRAVITY IN FT/ HR**2
```

```
      ACCG= 4.17E8
C        DELTIM- INTERVAL OF TIME IN WHICH PROGRAM IS EVALUATED IN HR
      DELTIM= 1
C        PROPERTIES OF COLLECTOR
C        DIAMO- OUTSIDE DIAM. OF COLLECTOR TUBE IN M
      DIAMO= .625 * .0254
C        DIAMI- INSIDE DIAM. OF TUBE IN COLLECTOR IN M
      DIAMI= .569 * .0254
C        XNUT-CROSS-SECTIONAL AREA OF TUBE IN COLLECTOR IN M**2.
      XNUT= 4./(PI * DIAMI**2.)
C        DIAMX-INSIDE DIAM. OF TUBE IN FT.
      DIAMX= .569 /12.
C        SIGMAO  FIN THICKNESS OF COLLECTOR IN M
      SIGMAO=  .0108 * .0254
C        XLENGT- FIN LENGTH OF COLLECTOR IN M
      XLENGT= .1
C        COND- COND. OF COPPER TUBING AND SOLDER IN WATTS/M C
      COND= 396.
C        CSUBS- BOND CONDUCTANCE
      CSUBS= 4. * SIGMAO * COND/DIAMO
C        W- SPACING OF TUBING ON COLLECTOR IN M
      W= 2. * XLENGT + DIAMO
C        USLOPE- OVERALL HEAT TRANSFER COEFF. FOR COLLECTOR WATTS/M**2 C
      USLOPE= 5.
C        NSIZE- NO. OF DIFFERENT SIZES OF COLLECTORS
      NSIZE= 1
C        AREA- ARRAY WHICH HOLDS SIZES OF COLLECTORS
      READ 1078,(AREA(JA),JA=1,NSIZE)
 1078 FORMAT(F10.4)
C        PROPERTIES OF WATER
C        B1 THRU B5 ARE CONSTANTS FOR THE SPECIFIC HEAT EQUATION
C        THEY HAVE BEEN CALCULATED FROM HANDBOOK OF HEAT TRANSFER
      READ 1080, B1,B2, B3, B4, B5
C        D1 - D5 - COEFF. FOR CONDUCTIVTY EQUATIONS OF WATER
      READ 1080, D1, D2, D3, D4, D5
C        R1 - R5 - COEFF. FOR DENSITY EQUATION OF WATER
      READ 1080, R1, R2, R3, R4, R5
C        S1 - S5 - COEFF. FOR BETAF EQUATION FOR WATER
      READ 1080, S1, S2, S3, S4, S5
C        Z1-Z5 -COEFF. FOR VISCOSITY OF WATER EQUATION FOR TEMP. >60F
      READ 1080, Z1, Z2, Z3, Z4, Z5
 1080 FORMAT( 5E10.4)
C         PROPERTIES OF HEAT TRANSFER FLUID
C        N- NO. OF FLUIDS CIRCULATED THRU COLLECTOR
      N= 9
C        A1 THRU A5 ARE THE FLUID COEFF. FOR THE SPECIFIC HEAT EQUATIONS
      READ 1080,(A1(I),A2(I),A3(I),A4(I),A5(I),I= 1,N)
C         E1 -E5 -COEFF. FOR CONDUCTIVITY EQUATION OF FLUID
      READ 1080,(E1(I),E2(I),E3(I),E4(I),E5(I),I= 1,N)
C        F1-F5 -COEF. FOR VISCOSITY EQUATION OF FLUID
      READ 1080,(F1(I),F2(I),F3(I),F4(I),F5(I),I= 1,N)
C        DEN1-DEN5- COEF. FOR DENSITY EQUATION OF THE FLUID
      READ 1080,(DEN1(I),DEN2(I),DEN3(I),DEN4(I),DEN5(I), I= 1,N)
C        DETERMINATION OF ENVIRONMENTAL PARAMETERS
C        XPHASE AND XOMEGA -USED IN DETERMINING AMBIENT AIR TEMP.
      XPHASE= - 3. * PI/2.
      XOMEGA= PI/ 12.
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```
C         LOOP TO READ IN INCIDENT RAD. AND AMBIENT TEMP.
      DO 1090 J= 1,24
C         QINCID-AMOUNT OF INCIDENT RADIATION RECEIVED AT COLLECTOR SITE.....
C          FOR THIS CASE IT IS SYMMETRIC AROUND 12:00
      QINCID(J)= 750.* COS((PI *J/12.) - PI)
C         TAMB- AMBIENT TEMP. AT COLLECTOR IN C
C         TAMB IS ASSUMED TO BE SINUSOIDAL, WITH AMAX AT 18:00 AND MIN. AT 6:00
      TAMB(J)= 15. + 10. * COS( XPHASE + J * XOMEGA)
C         IF THE INCIDENT RADIATION IS NEGATIVE IT IS ASSUMED TO BE 0
      IF( QINCID(J) .GT. 0) GO TO 1090
      QINCID(J)= 0.
 1090 CONTINUE
C
C         TSTART-TEMP. DIFF. BETWEEN WATER TEMP. AND FLUID TEMP. IN ORDER TO TURN
C         PUMP ON
      TSTART= 18.
C         TSTOP -TEMP. DIFF. BETWEEN WATER TEMP. AND FLUID TEMP. IN ORDER TO TURN
C         PUMP OFF
      TSTOP= 3.
C         THIGH- MAX. ALLOWABLE TEMP. OF FLUID
      THIGH= 90.
C          NFLOW- NO. OF FLOWRATES TO BE USED
      NFLOW= 1
C         G-ARRAY WHICH HOLDS VARYING FLOWRATES OF FLUID
      READ 1078,(G(JL),JL=1,NFLOW)
C         QFAC- ARRAY WHICH HOLDS VARYING HOT WATER LOAD
      READ 197,(QFAC(J),J= 1,24)
 197  FORMAT(12F6.2)
      PRINT 1913
 1913 FORMAT(50X,'COLLECTOR TUBE CHARACTERISTICS'/10X,'O.D.TUBE(M)',10X,
     1'I.D.TUBE(M)',10X,'SPACING OF TUBING(M)')
      PRINT 1914,DIAMO,DIAMI,W
 1914 FORMAT( 7X,F10.4,11X,F10.4,16X,F10.4/)
C         LOOP TO VARY FLUIDS IN COLLECTOR
      DO 1900 I= 1,N
C         LOOP TO VARY SIZE OF COLLECTOR
      DO 1890 JA= 1, NSIZE
C         AREACO- AREA OF COLLECTOR IN M**2
      AREACO= AREA(JA)
C         XNTUBE- NO. OF COLLECTOR TUBES
      XNTUBE= 4. * AREACO
C         LENGTH IF COLLECTOR TUBES IN FT
C         XLCOL- LENGTH OF COLLECTOR TUBES
      XLCOL= AREACO/(XNTUBE * W * .3048)
C         XPANEL- NO. OF SEPERATE HELICAL COILS
      IF( JA .EQ. 1) XPANEL= 2
C         PROPERTIES OF TANK AND COIL AROUND TANK
C         K- RATIO OF TANK HEIGHT TO RADIUS
      K= 6
C         DIAMT - DIAM. OF TANK IN M
      DIAMT=((2.8781 * AREACO /(K * PI))**(1./3.))*2.
      DIAMT= DIAMT * .3048
C         TANKH- HEIGHT OF TANK IN M
      TANKH= K * DIAMT /2.
C         TCOND CONDUCTIVITY OF COPPER TUBING AND COPPER WALLS IN WATTS/M C
      TCOND= 396.0
C         SCOND- CONDUCTIVITY OF STEEL TANK IN WATTS/M C
```

```
      SCOND = 50.
C        TWALL- THICKNESS OF TANK IN M
      TWALL= .017143 * DIAMT
      BWID= .625
C        BWIDTH - OUTSIDE DIAM. OF TUBING AROUND TANK IN M
      BWIDTH= .625 * .0254
      YK= .00125
C        T- HELICAL COIL TUBE THICKNESS
      T= YK + .02 * BWID/12.
C        BWIDX-DIAM. OF TUBE IN FT.
      BWIDX= BWID/12. - 2. * T
      T= T * .3048
C        BWIDIN- INSIDE DIAM. OF TUBING AROUND TANK IN M
      BWIDIN= BWIDX * .3048
C        XNUT1-CROSS-SECTIONAL AREA OF TUBE AROUND TANK IN M**2.
      XNUT1= 4./(PI* BWIDIN**2.)
C        BONDCO- BOND CONDUCTANCE OF TANK BOND TO THE TUBING
      BONDCO= 4. * TWALL * SCOND/BWIDTH
C        OAREAT- TOTAL SURFACE AREA OF TANK
      OAREAT= PI * DIAMT * TANKH+ PI *(DIAMT **2.)/2.
      PAREAT= PI * DIAMT * TANKH
      PRINT 1
 1    FORMAT(1H0)
      PRINT 1911
 1911 FORMAT(50X,'TANK DIMENSIONS'/10X,'HEIGHT(M)',10X,'DIAMETER(M)',
     110X,'TOTAL SURFACE AREA(M**2)')
      PRINT 1912,TANKH,DIAMT,PAREAT
 1912 FORMAT(6X,F10.4,7X,F10.4,13X,F10.4/)
C        CCOL- COST OF COLLECTOR SYSTEM
      CCOL= 100. * AREACO
C        TUBEL -LENGTH OF TUBING IN M
        TUBEL= 20.
      FTUBEL= TUBEL / .3048
C        CTUBE- COST OF TUBING AND SOLDER/METER OF TUBING
      CTUBE= 5.
C        BMAX - SPACING BETWEEN TUBES IN M
      BMAX= 2.75 * .0254
C        TLENGT- FIN LENGTH OF TANK IN M
      TLENGT= ( BMAX - BWIDTH  )/2.
C        AREAT- AREA OF HEAT TRANSFER IN M**2
      AREAT= BMAX * TUBEL
C        T1- FRACTION  OF TANK  COVERED BY TUBING
      T1= AREAT/ PAREAT
      PRINT 1915, T1
 1915 FORMAT(50X,'COIL CHARACTERISTICS'/50X,'AMOUNT OF TANK COVERED BY
     1TUBING',F7.3/ 5X,'O.D.TUBE(M)', 5X,'I.D.TUBE(M)', 5X,'TOTAL LENGTH
     2OF TUBING(M)',10X,'SPACING BETWEEN COILS',10X,'FIN LENGTH')
      PRINT 1916,BWIDTH,BWIDIN,TUBEL,BMAX,TLENGT
 1916 FORMAT(3X,F10.4,5X,F10.4,11X,F10.4,17X,F10.4,18X,F10.4/)
C        JL- COUNTER FOR NUMBER OF FLOWRATES
      JL= 1
C        ICOUNT- COUNTER TO DETERMINE HOW MANY FLOWRATES TO USE
      ICOUNT= 0
 1062 CONTINUE
C        TWATF- STARTING TEMP OF WATER
      TWATF= 60.
C        TWAT - TEMP OF WATER IN TANK
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      TWAT = TWATF
C        TLIQIN- TEMP. OF FLUID ENTERING COLLECTOR C
      TLIQIN= TWAT
C        IX- COUNTER FOR CONVERGENCE LOOP FOR MULTIPLE DAY RUNS
      IX= 0
C       AN ITERATION BEGINS WHICH WILL CONTINUE UNTIL THE DAY'S TEMP. AND THE
C        PREVIOUS DAY'S TEMP ARE WITHIN AN ACCEPTABLE LIMIT.....
 1063 CONTINUE
C        IF IT TAKES MORE THAN 7 DAYS TO CONVERGE IT IS TOO LONG
      IF( IX .GE. 7) GO TO 1822
      IX= IX + 1
C        FOR MULTIPLE DAY RUNS, FLOWRATE COUNTER IS SET BACK TO 1
C        SO THAT THE FLOWRATE IS JUST TURBULENT
C        ICOUNT- COUNTER TO DETERMINE HOW MANY FLOWRATES TO USE
      IF( ICOUNT .NE. 0) GO TO 1064
      JL= 1
 1064 CONTINUE
C        TBEG- BEGINNING TEMP. OF WATER IN TANK FOR EACH DAY
      TBEG= TWAT
C        IC-COUNTER OF NUMBER OF HOURS PUMP IS TURNED ON
      IC= 0
C        QTPRIM- TOTAL RATE OF HEAT COLLECTED AND TRANSFERED TO FLUID/DAY
      QTPRIM= 0
C        QTTRIM- TOTAL RATE OF HEAT TRANSFERRED /DAY FROM FLUID TO TANK WATER
      QTTRIM= 0
C        QTLOSS-TOTAL LOSS FROM TANK FOR 1 DAY
      QTLOSS= 0.
C        QTLOAD-TOTAL LOAD FROM TANK FOR 1 DAY
      QTLOAD= 0.
C         QTINC- TOTAL INCIDENT RAD. FOR 1 DAY
      QTINC= 0.
C        EAVE- AVE. COLLECTOR EFFICIENCY
      EAVE= 0.
C        FPAVE- AVE. FIN EFFICIENCY OF TUBE-TANK SYSTEM
      FPAVE= 0.
C        EXAVE- AVE. EFFICIENCY OF TUBE TANK SYSTEM
      EXAVE= 0.
C        TMAX- MAX. TEMP. OF WATER FOR A GIVEN DAY.
      TMAX= TWAT
C        TMIN- MIN. TEMP. OF WATER FOR A GIVEN DAY
      TMIN= TWAT
C        PUMP-SIGNIFIES WHETHER PUMP IS ON OR OFF.
C        IF PUMP IS 1  PUMP IS ON,PUMP=0,PUMP OFF.
      PUMP= 0
C        LOOP TO VARY TIME OF DAY
C        STARTING WITH 1:00 TO 24:00
      DO 1830 J= 1,24
C        TEMP.-AN ARRAY WHICH HOLDS ALL DAILY TEMPERATURES UNTIL THE TEMP.
C        CONVERGE......
      TEMP( IX, J) = TWAT
C        QINCID IS INCIDENT RADIATION RECIEVED BY COLLECTOR IN WATTS/M**2
      QI= QINCID(J)
C        TAMBCN- AMBIENT AIR TEMP. FOR GIVEN IN C
      TAMBCN= TAMB(J)
C        PUMP-SIGNIFIES WHETHER PUMP IS ON OR OFF.
      IF( PUMP .NE. 0) GO TO 1854
C        TLIQ-TEMP. OF FLUID WHEN PUMP IS OFF,ASSUMING ISOTHERMAL COLLECTOR
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      TLIQ=(QI  /USLOPE)+ TAMBCN
C        IF TEMP. OF FLUID IS GREATER THAN THE MAX. ALLOWABLE TEMP. ABOVE
C        WHICH PRESSURE AND BOILING EFFECTS ARE SIGNIFICANT,IT IS ASSUMED
C        THAT THERE IS A RELEASE VALVE COOLING FLUID TO THE MAX. TEMP.
      IF( TLIQ .GE. THIGH) TLIQ= THIGH
C        TLIQIN- TEMP. OF FLUID AT ENTRANCE TO COLLECTOR
      TLIQIN= TWAT
      TENT1= TLIQ - TWAT
C        IF THE TEMP. OF THE FLUID IN THE COLLECTOR IS MUCH GREATER THAN THAT OF
C        THE WATER IN THE STORAGE TANK THEN THE PUMP IS TURNED ON.
      IF( TENT1 .LE. TSTART) GO TO 1399
C        LOOP TO FIND FLOWRATE FOR TURBULENT FLOW
 1854 CONTINUE
C        IF PUMP IS 1  PUMP IS ON,PUMP=0,PUMP OFF.
      PUMP= 1
C        TLIQI1 - TLIQIN IN F
      TLIQI1= 1.8 * TLIQIN + 32.
C        SH111- SPECIFIC HEAT OF WATER AT LIQUID TEMP.
      SH111= B1+ B2 *(B3+ TLIQI1) + B4* (B5+ TLIQI1)**2.
C        SH11- SPECIFIC HEAT OF FLUID IN BTU/LB F
      SH11= A1(I)+ A2(I) *( A3(I) + TLIQI1) + A4(I) *( TLIQI1+ A5(I))**2
C        SH1- SPECIFIC HEAT OF FLUID IN WATT HR/KG C
      SH1= SH11 * 1.162
C        G1 - FLUID FLOWRATE IN KG/HR
      G1= G(JL) * AREACO *(SH111/SH11)
C        G11- FLOWRATE OF FLUID THRU COLLECTOR IN KG/HR-M**2.
      G11= G(JL) * (SH111/SH11)
C        G111-FLOWRATE OF FLUID THRU COLLECTOR IN LB/FT**2OFTUBE HR
      G111= G1 * XNUT/XNTUBE
      G111= G111/4.883
C        VISFL1- VISCOSITY OF FLUID IN LB/FT-HR
      VISFL1= F1(I)+ F2(I)*(F3(I)+ TLIQI1)+F4(I)*(F5(I)+TLIQI1)**2.
      VISFL1= 1./VISFL1
C        CONFL1-CONDUCTIVITY OF FLUID IN BTU/HR-FT-F
      CONFL1= E1(I)+ E2(I)*(E3(I)+TLIQI1)+E4(I)*(E5(I)+TLIQI1)**2.
      DENFL1= DEN1(I)+DEN2(I)*(DEN3(I)+TLIQI1)+DEN4(I)*(DEN5(I)+TLIQI1)
     1**2.
C        CAP1 - CAPACITY RATE OF FLUID IN WATTS/C
      CAP1= G1 * SH1
C        DETERMINATION OF ETA2
      COEF= SQRT(USLOPE/(COND * SIGMAO))
      ETA1= TANH(COEF * XLENGT)/(COEF * XLENGT)
      YPART1= 1./((1. - DIAMO/W)* ETA1)
      YPART2= 1./(((W * USLOPE)/CSUBS)+ YPART1)
      YPART3= 1./((DIAMO/W)+ YPART2)
      REYNOL= G111 * DIAMX/VISFL1
      IF( REYNOL .LT. 2500) GO TO 157
      PRANDT= SH11 * VISFL1/CONFL1
      FRIC= .0014+ .125/REYNOL**.32
      IF( REYNOL .GT. 7100) GO TO 166
C        TRANSITION REGION
      QJPRIM= .116*(((REYNOL)**(2./3.))-125.)/  REYNOL
C        XHCW- INSIDE TUBE HEAT TRANSFER COEFF. FOR COLLECTOR
      XHCW1= (PRANDT**(-2./3.)) * SH11 * G111 * QJPRIM
      GO TO 158
  166 CONTINUE
C         TURBULENT REGION
```

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      XHCW1=CONFL1 * .023 *( REYNOL **.8) *(PRANDT **.4)/DIAMX
      GO TO 158
 157  CONTINUE
C         LAMINAR REGION
      FRIC= 16./REYNOL
      G121= G1/(XNTUBE * .4536)
      TUBECO= AREACO/(((2 * XLENGT + DIAMU)* XNTUBE)* .3048)
      CPART1= G121 * SH11/(CONFL1 * TUBECO)
      CPART2=  1.75 * CPART1 **(1./3.)
      IF( CPART2 .LE. 3.66) CPART2= 3.66
      XHCW1= CONFL1 * CPART2/ DIAMX
 158  CONTINUE
      XHCW1= XHCW1 * 5.678
C         PRESSURE DROP IN EACH TUBE OF COLLECTOR
      XPD= FRIC*XLCOL*(G111**2.)/(2.*DIAMX*ACCG*DENFL1*144.)
      CXPD= XNTUBE * XPD
      ETA2= 1./(((W *USLOPE)/(PI *DIAMI * XHCW1))+YPART3)
C         DETERMINATION OF ETA3
C         ETA3 - COLLECTOR EFFICIENCY
      AX1= -( ETA2 * USLOPE)/( G11 * SH1)
      AX2= 1 - EXP(AX1)
      ETA3= ( G11 * SH1/ USLOPE) * AX2
C         QCOLL AMOUNT OF HEAT COLLECTED BY COLLECTOR
      QCOLL= ETA3 * AREACO *( QI- USLOPE *(TLIQIN - TAMBCN))
      IF( QCOLL .LE. 0) GO TO 1399
      QTPRIM= QCOLL * DELTIM * 3600. + QTPRIM
 1112 CONTINUE
C         TLIQF- TEMP OF LIQUID LEAVING COLLECTOR ASSUMIMG CAP. RATE AT TEMP. OF
C         FLUID ENTERING COLLECTOR
      TLIQF= TLIQIN + QCOLL/ CAP1
C         CONVERGENCE LOOP TO DEVELOP MEAN CAPACITY RATE IN ORDER TO CALCULATE
C         THE TEMP. OF FLUID LEAVING COLLECTOR.....
C       IC1- COUNTER TO DETERMINE NUMBER OF ITERATIONS....
      IC1= 0
 1140 CONTINUE
      IC1 = IC1 + 1
      TINCR= TLIQF - TLIQIN
      TLIQF1= 1.8 * TLIQF + 32.
      SH77= A1(I) + A2(I)*(A3(I)+ TLIQF1)+ A4(I)*(TLIQF1 + A5(I))**2.
C         SH7- SPECIFIC HEAT OF FLUID IN WATT-HR/KG-C AT TLIQF
      SH7= SH77 * 1.162
C         CAP7- CAPACITY RATE AT TLIQF IN WATTS/C
      CAP7= SH7 * G1
C         CAP8- MEAN CAPACITY RATEUSED IN DETERMINING THE FINAL TEMP. OF FLUID
C         LEAVING COLLECTOR...
      CAP8= ( CAP7 + CAP1) /2.
C         TLIQ- TEMP. OF LIQUID LEAVING COLLECTOR IN C
      TLIQ= TLIQIN + QCOLL / CAP8
      TVAL= (TLIQF - TLIQ)/TINCR
      TVAL = ABS(TVAL)
C         IF TVAL).1( OF TEMPS. THEN LOOP HAS CONVERGED
      IF( TVAL .LE. .0001) GO TO 1141
      TLIQF= TLIQ
      GO TO 1140
 1141 CONTINUE
C         TLIQ- TEMP OF FLUID LEAVING COLLECTOR
      TLIQ1= TLIQ * 1.8 + 32.
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C          IF THE TEMP. OF FLUID IS LESS THAN THAT OF THE TANK WATER , THE
C          PUMP WILL NOT BE TURNED ON AND NO HEAT WILL BE COLLECTED....
C          IF TEMP. OF FLUID IS GREATER THAN THE MAX. ALLOWABLE TEMP. ABOVE
C          WHICH PRESSURE AND BOILING EFFECTS ARE SIGNIFICANT,IT IS ASSUMED
C          THAT THERE IS A RELEASE VALVE COOLING FLUID TO THE MAX TEMP.
      IF( TLIQ .GE. THIGH) TLIQ= THIGH
      TENT2= TLIQ- TWAT
      IF( TENT2 .LE. TSTOP) GO TO 1400
C          G1111-FLOWRATE IN LB/FT**2-HR
      G1111= G1 * XNUT1/XPANEL
      G1111= G1111/4.883
      VISFL2= F1(I)+F2(I)*(F3(I)+TLIQ1)+F4(I)*(F5(I)+TLIQ1)**2.
      VISFL2= 1./ VISFL2
      DENFL2= DEN1(I)+DEN2(I)*(DEN3(I)+TLIQ1 )+DEN4(I)*(DEN5(I)+TLIQ1 )
     1**2.
      SH44= A1(I) + A2(I) *(A3(I) + TLIQ1) + A4(I) *(A5(I)+ TLIQ1) **2.
C          SH4- SPECIFIC HEAT OF FLUID AT TLIQ
      SH4= 1.162 * SH44
C          CAP3- CAPACITY RATE OF FLUID AT TLIQ
      CAP3= SH4 * G1
      CONFL2= E1(I)+E2(I)*(E3(I)+TLIQ1)+E4(I)*(E5(I)+TLIQ1)**2.
      REYNO= G1111 * BWIDX/VISFL2
      IF( REYNO .LT. 2500) GO TO 161
      PRAND= SH44 * VISFL2/CONFL2
      FRIC= .0014 + .125/REYNO**.32
      IF(REYNO .GT. 7100)  GO TO 167
C          TRANSITION REGION
      TJPRIM=.116*(((REYNO)**(2./3.))-125.)/REYNO
C          XHCW- INSIDE HEAT TRANSFER COEFF OF FLUID IN TUBES
      XHCW= (PRAND **(-2./3.))* SH44 * G1111 * TJPRIM
      GO TO 162
 167  CONTINUE
C           TURBULENT REGION
      XHCW=  CONFL2 * .023 *(REYNO **.8) *(PRAND **.4)/BWIDX
      GO TO 162
 161  CONTINUE
C          LAMINAR REGION
      FRIC= 16./REYNO
      G1121= G1/(XPANEL * .4536)
      TUBEL1 = TUBEL/(.3048 * XPANEL)
      DPART1= G1121 * SH44/(CONFL2 * TUBEL1)
      DPART2= 1.75 * DPART1 **(1./3.)
      IF( DPART2 .LE. 3.66) DPART2 = 3.66
      XHCW= CONFL2 * DPART2/BWIDX
 162  CONTINUE
C          FOR HELICAL COIL,THE INSIDE TUBE HEAT TRANSFER COEFF. IS INCREASED BY
C          EFFECTS OF CURVATURE ACCORDING TO MCADAMS BY THE FOLLOWING
      XHCW= XHCW * 5.678
      XHCW= XHCW * (1. + 3.5 * ( DIAMI/DIAMT))
      WXPD= FRIC*FTUBEL*(G1111**2.)/(2.*BWIDIN*ACCG*DENFL2*144.)
C         IC2- COUNTER TO DETERMINE NUMBER OF ITERATIONS....
      IC2= 0
C          BEGINNING OF CONVERGENCE LOOP TO DETERMINE THE HEAT TRANSFER COEFF.(UL)
C          FROM FLUID TO TANK WATER
C          TX- TEMP. OF WALL OF TANK
 1111 TX(1)= TLIQ
      M= 1
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 1120 CONTINUE
:        TMEAN(M)- MEAN TEMP. OF WATER
      TMEAN(M)= ( TX(M) + TWAT) / 2.
      IC2= IC2 + 1
:        BETAC- VOL. COEFF. OF EXPANSION IN 1/C
      BETAC= S1 + S2 *( S3 + TMEAN(M)) +S4 *(TMEAN(M) + S5) **2.
:        BETAF- VOLUMETRIC COEFF. OF EXPANSION AT WALL TEMP
      BETAF= 5. * BETAC/ 9.
      TMEAN(M)= TMEAN(M) * 1.8 + 32.
:        DENSW3 - DENSITY OF WATER AT WALL TEMP
      DENSW3= R1 + R2 *(R3+ TMEAN(M)) + R4 *(TMEAN(M) + R5) **2.
:        CONDW3- CONDUCTIVITY OF WATER  AT WALL TEMP
      CONDW3= D1 + D2 *(D3+ TMEAN(M)) + D4 *(TMEAN(M) + D5) **2.
:        SH33- SPECIFIC HEAT OF WATER AT WALL TEMP.
      SH33  = B1 + B2 *(B3+ TMEAN(M)) + B4 *(TMEAN(M) + B5) **2.
      SH3= SH33 * 1.162
:        VISW3- VISCOSITY OF WATER AT WALL TEMP
      VISW3= Z1 + Z2 *( Z3 + TMEAN(M)) + Z4 *( Z5 + TMEAN(M)) **2.
      VISW3= 1./VISW3
:        TANKH- HEIGHT OF TANK IN FT
      TANKH= 4.5
:        APART1,APART2,APART3, ARE PARTS OF NATURAL CONVECTION HEAT TRANSFER
:        EQUATION FOUND IN MCADAMS. FOR TURBULENT CASE(7-4A),LAMINAR(7-4B)
:
      APART1= (SH33 * VISW3/CONDW3) * TANKH **3.
      TWAT1= TWAT * 1.8 + 32.
      TXP= 1.8 * TX(M) + 32.
      APART2= APART1 * ACCG * BETAF * (TXP   - TWAT1) *(DENSW3**2.)
      APART2= ABS(APART2)
      APART3= APART2 /(VISW3 **2.)
:        SEPERATION FOR TURBULENT AND LAMINAR RANGES
      IF( APART3 .LT. 1E9) GO TO 1170
:        TURBULENT RANGE
      CCON= .13
      DCON= 1./3.
      GO TO 1200
:        LAMINAR RANGE
 1170 CCON= .59
      DCON= .25
 1200 CONTINUE
:        UL- NATURAL CONVECTION COEFF.  OF WATER FOR A VERTICAL CYLINDER
      UL= CONDW3 * CCON *(APART3 ** DCON)/TANKH
      UL= UL * 5.678
      TANKH= K * DIAMT/2.
:        AHALF,ARGEF,ETA50,ELE1,ELE2,ELE3, ARE PARTS OF EQUATION TO DETERMINE
:                    FPRIME AS FOUND IN
      AHALF= SQRT(UL/(SCOND * TWALL))
      ARGEF= AHALF * TLENGT
      ETA50= TANH(ARGEF)/ARGEF
      ELE1= BMAX/( BWIDTH + 2. * TLENGT * ETA50)
      ELE2= BMAX * UL/ BONDCO
      ELE3= BMAX * UL /(PI * BWIDTH * XHCW)
      FPRIME= 1/( ELE1 + ELE2 + ELE3)
      TMEAN(M)= (5./9.) *(TMEAN(M) -32.)
      EXETA= 1. - EXP(-UL * AREAT * FPRIME/ CAP3)
         TLIQW- TEMP. OF FLUID LEAVING TANK AT BOTTOM  IN C
      TLIQW= TLIQ - EXETA *( TLIQ - TWAT)
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C           LOOP TO DETERMINE MEAN CAPACITY RATE OF FLUID IN ORDER TO DETERMINE
C           AMOUNT OF HEAT TRANSFERRED TO TANK....
C         IC3- COUNTER TO DETERMINE NUMBER OF ITERATIONS....
      IC3= 0
 1220 CONTINUE
      IC3= IC3+ 1
      TLIQW1= 1.8 * TLIQW + 32.
      SH66= A1(I) + A2(I) *( A3(I)+TLIQW1)+A4(I)*(A5(I)+TLIQW1)**2.
C           SH6- SPECIF HEAT OF FLUID AT TLIQW
      SH6= SH66 * 1.162
C           CAP6- CAPACITY FLOW RATE AT TLIQW
      CAP6= SH6 * G1
C           CAP4-MEAN CAPACITY RATE OF FLUID
      CAP4= ( CAP3 + CAP6)/2.
C           EXETA- EFFECTIVENESS OF HEAT TRANSFER FROM TUBES TO TANK
      EXETA= 1. - EXP(-UL * AREAT * FPRIME/CAP4)
C           TLIQIN- TEMP. OF FLUID LEAVING TANK.
      TLIQIN= TLIQ - EXETA * (TLIQ - TWAT)
      TABS1= TLIQ - TLIQIN
      TABS=(TLIQIN - TLIQW)/TABS1
      TABS= ABS( TABS)
C           IF TABS ) .1( THEN THE LOOP HAS CONVERGED
      IF( TABS .LE. .0001) GO TO 1230
      TLIQW= TLIQIN
      GO TO 1220
 1230 CONTINUE
      Z= UL * BMAX * FPRIME / CAP4
      EITA= ((1./EXETA) - 1.)
C           FACTOR- PENALTY IMPOSED BY JACKETED TANK SYSTEM ON HEAT TRANSFER
      FACTOR= 1./(1. + (AREACO * ETA3 * USLOPE/CAP4) * EITA)
      TPD= CXPD + WXPD
      VFLOW1= G1 * 2.2046 /( DENFL1 * 3600.)
      XHP3= TPD  * VFLOW1 * 144. / 550.
      COP1= .03 * 8. * 365. * .7457 * XHP3/.7
C           QTOTAL - AMOUNT OF HEAT TRANSFERED FROM FLUID TO WATER IN TANK
      QTOTAL= FACTOR * AREACO * ETA3 * (QI - USLOPE *(TWAT - TAMBCN))
      M= M+1
      TX(M)=  QTOTAL/(UL * AREAT) + TWAT
      AARD= TX(M) - TX(M- 1)
      AARD= ABS( AARD)
C           IF THE DIFF. BETWEEN THE NEW AND OLD WALL TEMP.) .1C THEN THE LOOP
C           HAS CONVERGED...
      IF( AARD .LE. .1) GO TO 1500
      M= 1
      TX(M)= TX(M+ 1)
      GO TO 1120
 1399 CONTINUE
C         IF CONTROL HAS PASSED TO THIS LOCATION THEN THE PUMP WASN'T ACTIVATED...
      REYNOL= 0
      XHCW1= 0
      QCOLL= 0
      ETA3= 0
 1400 CONTINUE
      PUMP= 0
      QTOTAL= 0.
      REYNO= 0
      XHCW= 0
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      UL= 0
      EXETA= 0
      FACTOR= 0
      OC= 0
      IC= OC + IC
      GO TO 1510
 1500 CONTINUE
C        IF CONTROL HAS PASSED TO THIS LOCATION THEN THE PUMP HAS BEEN ACTIVATED
C        AND HEAT IS TRANSFERRED TO THE TANK
C        IC-COUNTER OF NUMBER OF HOURS PUMP IS TURNED ON
      IC= IC+ 1
C        EXAVE- AVE. EFFICIENCY OF TUBE TANK SYSTEM
      EXAVE= EXAVE + EXETA
C        FPAVE- AVE. FIN EFFICIENCY OF TUBE-TANK SYSTEM
      FPAVE= FPAVE + FPRIME
C        EAVE- AVE. COLLECTOR EFFICIENCY
      EAVE= ETA3 + EAVE
C        FINAL DETERMINATION OF FACTORS FOR GIVEN HOUR
 1510 CONTINUE
C         QTINC- TOTAL INCIDENT RAD. FOR 1 DAY
      QTINC= QTINC + QI * DELTIM * 3600. * AREACO
      QTTRIM= QTTRIM + QTOTAL * DELTIM * 3600.
      DENSW= R1 + R2 *(R3+ TWAT) + R4* ( R5 + TWAT) **2.
C        DENSW- DENSITY OF WATER IN KG/M**3
      DENSW= 16.05212 * DENSW
      SH55= B1 + B2 *( B3+ TWAT) + B4 *(B5 + TWAT) **2.
C        SH5- SPECIFIC HEAT OF WATER
      SH5= SH55 * 1.162
C        VOLW- VOLUME OF WATER INSIDE TANK ASSUMING NO EXPANSION OF WATER WITH
C        TEMPERATURE AND FILLED TO THE TOP OF THE TANK
      VOLW= PI * ((DIAMT **2.)/4.) * TANKH
C        TCOL- TOTAL COST OF SYSTEM
      TCOL= CCOL + TUBEL * CTUBE
      FEX= TCOL/QTOTAL
      GEX= TCOL/QTTRIM
      ZCOL= CCOL/ CTUBE
      PFAC= QFAC(J)
      RBASE= 60. * AREACO
C        PLOAD- VARIABLE HOT WATER LOAD
      PLOAD= PFAC * RBASE
      QTLOAD= QTLOAD + 3600. * PLOAD * DELTIM
      TW(J)= TWAT
C        TWATNE- TEMP OF WATER AT END OF 1 HOUR
      ALPH= (QTOTAL - PLOAD)/(SH5 * DENSW * VOLW)
      TWATNE= TWAT + DELTIM * ALPH
      TWAT = TWATNE
C        TWMAX- MAX. ALLOWABLE TANK WATER TEMP.
      TWMAX= 70.
      IF( TWAT .GT. TWMAX) TWAT= TWMAX
C        TMAX- MAX. TEMP. OF WATER FOR A GIVEN DAY.
      IF( TWAT .GE. TMAX) TMAX= TWAT
C        TMIN- MIN. TEMP. OF WATER FOR A GIVEN DAY
      IF( TWAT .LE. TMIN) TMIN= TWAT
      TL(J)= TLIQ
      REYN(J)= REYNOL
      XH1(J)= XHCW1
      REY(J)= REYNO
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      XH(J)= XHCW
      U(J)= UL
      EX(J)= EXETA
      EC(J)= ETA3
      QC5(J)= FACTOR
      QC6(J)= QTOTAL
      TM(J)= TMEAN(1)
      C4(J)= CAP4
      CX(J)= WXPD
      CZ(J)= G1111
      QI1(J)= CUP1
      PL(J)= TPD
      FA(J)= FACTOR
      FP(J)= FPRIME
      SH(J)= SH4
      CON(J)= CONFL2
      VIS(J)= VISFL2
      BE(J)= BETAF
      DE(J)= DENSW3
      CO(J)= CONDW3
      S(J)= SH33
      VI(J)= VISW3
 1830 CONTINUE
      IF( IX .LE. 1) GO TO 1063
      J= 1
 1821 CONTINUE
      IF( J .GE. 25) GO TO 1822
      DIFF= TEMP(IX,J) - TEMP(IX-1,J)
      ABS1= ABS( DIFF)
:        IF ALL TEMP. MATCH  WITHIN A CERTAIN TOLERANCE BETWEEN THOSE OF ONE
:        DAY AND THOSE OF THE PREVIOUS THEN THE PROGRAM CONTINUES  TO THE
:        NEXT WATER INLET TEMP.
      IF( ABS1 .GT. 2.) GO TO 1063
      J= J + 1
      GO TO 1821
 1822 CONTINUE
      IF( IC .EQ. 0) GO TO 1823
:        EXAVE- AVE. EFFICIENCY OF TUBE TANK SYSTEM
      EXAVE= EXAVE /  IC
:        FPAVE- AVE. FIN EFFICIENCY OF TUBE-TANK SYSTEM
      FPAVE= FPAVE/IC
         EAVE- AVE. EFFICIENCY OF COLLECTOR
      EAVE= EAVE/IC
 1823 CONTINUE
 1910 CONTINUE
      PRINT 1917
 1917 FORMAT(50X,'PERFORMANCE CHARACTERISTICS'/10X,'COLLECTOR AREA',10X,
     1'FLOWRATE OF FLUID(KG/HR)',5X,'NO. OF HOURS PUMP IS ON')
      PRINT 1918,AREACO,G1,IC
 1918 FORMAT(10X,F10.4,16X,E10.4,17X,F10.4/)
      PRINT 1832
 1832 FORMAT(1X,' J',3X,'TWAT',6X,'TLIQ',5X,'REYNOL',5X,'XHCW1',5X,
     1'REYNO',5X,'XHCW',7X,'UL',7X,'EXETA',5X,'ETA3',5X,'FACTOR',4X,
     2'QTOTAL'/)
      PRINT 1833,(J,TW(J),TL(J),REYN(J),XH1(J),REY(J),XH(J),U(J), EX(J)
     1,EC(J),QC5(J),QC6(J),J=1,24)
 1833 FORMAT(1X,I2,11E10.4)
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145 ERDA02 0293AAAHSEMI 000293 825 40

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      PRINT 1919
 1919 FORMAT(45X,'HEAT TRANSFER CHARACTERISTICS(JOULES)'/10X,'INC,RAD
     1/DAY',5X,'HEAT COLL',5X,'HEAT TRANS./DAY',5X,'HEAT LOAD/DAY',
     25X,'COST TO HEAT GAIN RATIO')
      PRINT 1920,QTINC,QTPRIM,QTTRIM,QTLOAD,GEX
 1920 FORMAT( 5X,E10.4, 9X,E10.4,15X,E10.4,10X,E10.4,10X,E10.4/)
      PRINT 1921
 1921 FORMAT (1H1)
      IF( JL .GE. NFLOW) GO TO 839
      JL= JL + 1
      ICOUNT= ICOUNT+ 1
      GO TO 1062
  839 CONTINUE
  820 CONTINUE
 1890 CONTINUE
 1900 CONTINUE
      END
```

PILATION: NO DIAGNOSTICS.

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07/15/77  11:10:45 ERDA02 0293AAARSEMI 000293          S25   40

@MAP,SX   ,NAME
MAP28R1C RL72-8 07/15/77 11:11:34
   1.         IN TPFS.

ADDRESS LIMITS     001000 015332      6363 IBANK WORDS DECIMAL
                   040000 047576      3967 DBANK WORDS DECIMAL
STARTING ADDRESS   012337

            SEGMENT $MAINS          001000 015332     040000 047576

NSWTCS/FOR69            $(1)    001000 001032
NRBLKS/FOR-E2           $(1)    001033 001150     $(0)    040000 040001
                                                  $(4)    040002 040054
NRWNDS/FOR-E2           $(1)    001151 001232     $(2)    040055 040066
NWEFS/FOR-E2            $(1)    001233 001466     $(2)    040067 040106
NCLOSS/FOR-E3           $(1)    001467 001732     $(2)    040107 040136
NWBLKS/FOR68            $(1)    001733 002121     $(0)    040137 040140
                                                  $(4)    040141 040212
NBSBLS/FOR-E3           $(1)    002122 002174     $(0)    040213 040213
                                                  $(4)    040214 040251
NUPDAS/FOR68            $(1)    002175 002230
NBDCVS/FOR-E3           $(1)    002231 002361     $(2)    040252 040327
NFCHKS/FOR-E3           $(1)    002362 003352     $(2)    040330 040501
                        $(3)    003353 003353     $(4)    040502 040553
NBFOOS/ISD                                        $(2)    040554 042761
NFTVS/FOR-E2            $(1)    003354 003376
NCNVTS/FOR68            $(1)    003377 003620     $(2)    042762 043057
NOUTS/FOR-E3            $(1)    003621 005377     $(2)    043060 043123
NIOERS/FOR-E3           $(1)    005400 005627     $(2)    043124 043273
NFMTS/FOR-E3            $(1)    005630 006512     $(2)    043274 043347
NINPTS/FOR-E3           $(1)    006513 010121     $(2)    043350 043403
NTABS/ISD                                         $(2)    043404 043443
FORCOMS/FORFTN                                    $(2)    043444 043451
NERRS/FOR-E3            $(1)    010122 010465     $(2)    043452 043642
ERUS/SYS72-8
NERCOMS/FOR-TE3         $(1)    010466 010545     $(2)    043643 043656
FORVCOMS/FOR-TE3                                  $(2)    043657 043666
NSTOPS/FOR-TE3          $(1)    010546 010611     $(2)    043667 043675
EXPS/FOR59              $(1)    010612 010701     $(2)    043676 043716
TANHS/FOR59             $(1)    010702 011005     $(2)    043717 043744
NEXP6S/FOR-E3           $(1)    011006 011203     $(2)    043745 044016
NOSYMS/FOR-E3           $(1)    011204 011446     $(2)    044017 044020
SINCOSS/FOR-E3          $(1)    011447 011603     $(2)    044021 044043
NIERS/FOR-E3            $(1)    011604 011761     $(2)    044044 044164
NISYMS/FOR-E3           $(1)    011762 012135     $(2)    044165 044166
SQRTS/FOR59             $(1)    012136 012176     $(2)    044167 044200
NINTRS/FOR-E3           $(1)    012177 012336     $(2)    044201 044231
BLANK$COMMON(COMMONBLOCK)
MAIN                    $(1)    012337 015332     $(0)    044232 047576
                                                  $(2)    BLANK$COMMON

SYS$*RLIB$. LEVEL 72-8
END MAP
```

HEAT TRANSFER FLUID - MOBILTHERM LIGHT

TANK DIMENSIONS

HEIGHT(M)	DIAMETER(M)	TOTAL SURFACE AREA(M**2)
2.1059	.7020	4.6441

COIL CHARACTERISTICS

AMOUNT OF TANK COVERED BY TUBING .301

O.D.TUBE(M)	I.D.TUBE(M)	TOTAL LENGTHOF TUBING(M)	SPACING BETWEEN COILS
.0159	.0145	20.0000	.0698

PERFORMANCE CHARACTERISTICS

COLLECTOR AREA	FLOWRATE OF FLUID(KG/HR)	NO. OF HOURS PUMP IS ON
10.0000	.1937+04	5.0000

J	TWAT	TLIQ	REYNOL	XHCW1	REYNO	XHCW	UL	EXETA	ETA3	FACTOR
1	.5625+02	.1241+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
2	.5625+02	.1000+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
3	.5625+02	.7929+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
4	.5625+02	.6340+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
5	.5562+02	.5341+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
6	.5498+02	.5000+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
7	.5435+02	.4416+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
8	.5371+02	.5471+02	.3217+03	.7816+02	.0000	.0000	.0000	.0000	.6663+00	.0000
9	.5308+02	.5526+02	.3151+03	.7818+02	.0000	.0000	.0000	.0000	.6664+00	.0000
10	.5245+02	.5558+02	.3085+03	.7820+02	.6653+04	.8739+03	.5100+03	.2933+00	.6664+00	.9208+00
11	.5338+02	.5833+02	.3315+03	.7814+02	.7427+04	.1156+04	.5408+03	.3249+00	.6663+00	.9310+00
12	.5488+02	.6059+02	.3527+03	.7808+02	.7690+04	.1184+04	.5503+03	.3291+00	.6662+00	.9322+00
13	.5849+02	.6242+02	.3730+03	.7803+02	.8257+04	.1205+04	.5574+03	.3322+00	.6661+00	.9332+00
14	.6128+02	.6429+02	.3973+03	.7797+02	.8622+04	.1225+04	.5424+03	.3301+00	.6660+00	.9325+00
15	.6355+02	.6561+02	.4191+03	.7791+02	.0000	.0000	.0000	.0000	.6659+00	.0000
16	.6292+02	.6420+02	.4153+03	.7792+02	.0000	.0000	.0000	.0000	.6659+00	.0000
17	.6228+02	.6348+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
18	.6036+02	.2500+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
19	.5848+02	.2466+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
20	.5721+02	.2366+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
21	.5721+02	.2207+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
22	.5721+02	.2000+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
23	.5721+02	.1759+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
24	.5721+02	.1500+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

HEAT TRANSFER CHARACTERISTICS(JOULES)

INC.RAD /DAY	HEAT COLL	HEAT TRANS./DAY	HEAT LOAD/DAY	COST TO HEAT GAIN RATIO
.2051+09	.8155+08	.5512+08	.5184+08	.1996-04

HEAT TRANSFER FLUID - 50% ETHYLENE GLYCOL

TANK DIMENSIONS

HEIGHT(M)	DIAMETER(M)	TOTAL SURFACE AREA(M**2)
2.1059	.7020	4.6441

COIL CHARACTERISTICS

AMOUNT OF TANK COVERED BY TUBING .301

O.D.TUBE(M)	I.D.TUBE(M)	TOTAL LENGTHOF TUBING(M)	SPACING BETWEEN COILS
.0159	.0145	20.0000	.0698

PERFORMANCE CHARACTERISTICS

COLLECTOR AREA	FLOWRATE OF FLUID(KG/HR)	NO. OF HOURS PUMP IS ON
10.0000	.9360+03	4.0000

J	TWAT	TLIQ	REYNOL	XHCW1	REYNO	XHCW	UL	EXETA	ETA3	FACTOR
1	.5931+02	.1241+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
2	.5931+02	.1000+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
3	.5931+02	.7929+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
4	.5931+02	.6340+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
5	.5868+02	.5341+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
6	.5804+02	.5000+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
7	.5741+02	.4416+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
8	.5677+02	.5777+02	.3366+03	.1802+03	.0000	.0000	.0000	.0000	.7527+00	.0000
9	.5614+02	.5848+02	.3314+03	.1801+03	.0000	.0000	.0000	.0000	.7527+00	.0000
10	.5551+02	.5892+02	.3262+03	.1801+03	.7096+04	.2065+04	.5314+03	.3648+00	.7526+00	.9342+00
11	.5674+02	.6171+02	.3440+03	.1803+03	.7555+04	.2476+04	.5679+03	.3864+00	.7528+00	.9397+00
12	.5859+02	.6404+02	.3613+03	.1806+03	.7937+04	.2529+04	.5806+03	.3913+00	.7529+00	.9409+00
13	.6258+02	.6598+02	.3785+03	.1808+03	.8249+04	.2571+04	.5903+03	.3950+00	.7530+00	.9417+00
14	.6569+02	.6809+02	.4007+03	.1812+03	.0000	.0000	.0000	.0000	.7531+00	.0000
15	.6506+02	.6761+02	.4041+03	.1812+03	.0000	.0000	.0000	.0000	.7531+00	.0000
16	.6442+02	.6581+02	.3990+03	.1811+03	.0000	.0000	.0000	.0000	.7531+00	.0000
17	.6379+02	.6348+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
18	.6188+02	.2500+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
19	.5998+02	.2466+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
20	.5871+02	.2366+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
21	.5871+02	.2207+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
22	.5871+02	.2000+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
23	.5871+02	.1759+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
24	.5871+02	.1500+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

HEAT TRANSFER CHARACTERISTICS(JOULES)

INC.RAD /DAY	HEAT COLL	HEAT TRANS./DAY	HEAT LOAD/DAY	COST TO HEAT GAIN RATIO
.2051+09	.8874+08	.4980+08	.5184+08	.2209-04

NAME

COLLECTOR TUBE CHARACTERISTICS

O.D.TUBE(M)	I.D.TUBE(M)	SPACING OF TUBING(M)
.0159	.0145	.2159

HEAT TRANSFER FLUID - WATER

TANK DIMENSIONS

HEIGHT(M)	DIAMETER(M)	TOTAL SURFACE AREA(M**2)
2.1059	.7020	4.6441

COIL CHARACTERISTICS

AMOUNT OF TANK COVERED BY TUBING .301

O.D.TUBE(M)	I.D.TUBE(M)	TOTAL LENGTHOF TUBING(M)	SPACING BETWEEN COILS
.0159	.0145	20.0000	.0698

PERFORMANCE CHARACTERISTICS

COLLECTOR AREA	FLOWRATE OF FLUID(KG/HR)	NO. OF HOURS PUMP IS ON
10.0000	.8000+03	4.0000

J	TWAT	TLIQ	REYNOL	XHCW1	REYNO	XHCW	UL	EXETA	ETA3	FACTOR
1	.5977+02	.1241+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
2	.5977+02	.1000+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
3	.5977+02	.7929+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
4	.5977+02	.6340+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
5	.5914+02	.5341+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
6	.5850+02	.5000+01	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
7	.5787+02	.4416+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
8	.5723+02	.5824+02	.1027+04	.2458+03	.0000	.0000	.0000	.0000	.7732+00	.0000
9	.5660+02	.5899+02	.1016+04	.2456+03	.0000	.0000	.0000	.0000	.7731+00	.0000
10	.5597+02	.5946+02	.1006+04	.2454+03	.2120+05	.4935+04	.5375+03	.4058+00	.7731+00	.9425+00
11	.5729+02	.6218+02	.1039+04	.2460+03	.2206+05	.5028+04	.5735+03	.4194+00	.7732+00	.9455+00
12	.5926+02	.6450+02	.1072+04	.2466+03	.2280+05	.5106+04	.5914+03	.4261+00	.7734+00	.9469+00
13	.6335+02	.6644+02	.1106+04	.2472+03	.2341+05	.5170+04	.5982+03	.4287+00	.7735+00	.9474+00
14	.6656+02	.6864+02	.1152+04	.2479+03	.0000	.0000	.0000	.0000	.7737+00	.0000
15	.6592+02	.6851+02	.1165+04	.2481+03	.0000	.0000	.0000	.0000	.7738+00	.0000
16	.6529+02	.6668+02	.1154+04	.2480+03	.0000	.0000	.0000	.0000	.7737+00	.0000
17	.6465+02	.6348+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
18	.6275+02	.2500+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
19	.6084+02	.2466+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
20	.5957+02	.2366+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
21	.5957+02	.2207+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
22	.5957+02	.2000+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
23	.5957+02	.1759+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
24	.5957+02	.1500+02	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000

HEAT TRANSFER CHARACTERISTICS(JOULES)

INC.RAD /DAY	HEAT COLL	HEAT TRANS./DAY	HEAT LOAD/DAY	COST TO HEAT GAIN RATIO
.2051+09	.9051+08	.5119+08	.5184+08	.2149-04

This program is designed to model the following system. A flat plate collector of variable size (on the order of $5m^2$) is connected to a hot water tank by a single copper tube. The tube follows a serpentine path over the collector and as a continuous helical coil around the tank, in the tank is water assumed to be nonstratified subjected to a variable hot water load corresponding to an average home's use of hot water. The tank and coil are of known dimensions and the heat transfer characteristics are determined.

In the pipe is a heat transfer fluid at varying flow rates. A pump is used to pump the fluid when the fluid temperature in the collector is sufficiently greater than that of the water in the tank. Due to proper positioning of tank and collector density difference will not induce natural circulation.

As incident radiation is received on the collector, covered by a single glass plate, the collector and fluid are heated. Since the pump is off until a certain temperature difference is achieved, the rate at which heat is transferred is 0. The collector fluid system is assumed isothermal, until the pump is turned on, where upon heat is transferred to the tank with the temperature of the fluid in the collector falls below that needed to keep the pump going. This process is continued for a one day period, evaluating the temperature properties of water and fluid every hour.

Initially many variables pertaining to the collector must be given. Most of these variables have been determined in previous Altas studies. For this case a variable size collector (in the program the size is held in spray "area" and variables "areaco") is used and the effect collector size has on the system is studied. The metal tube is in a serpentine path to effectively utilize the area of the collector which maintains turbulent flow and a realistic pressure drop. The spacing between bends in the tubing ("w"), fin length ("xlenct"), fin thickness ("sigmao"), tube outer diameter ("diamo") and collector tube inner diameter ("diamt") were optimized in previous Altas studies and the values determined are used in this program. See the program for the actual values.

The cylindrical tank characteristics, diameter ("diamt") and height ("tankh"), were determined for holding 80 gallons of hot water. The thickness of the tank wall ("twall") needed to hold the water was determined for a water pressure of 300 psi within a stress of 20,000 psi. It is assumed at present that there is no heat loss from the tubing from the collector to the tank. Later various insulation types will be used to optimize heat transfer to cost.

The tube coil characteristics were determined from flow rate and heat transfer considerations. The optimum spacing ("BMAX") between successive coils for maximum heat transfer/cost tubing was determined from "BMAX" the length of tubing ("TUBEL"), and the total surface area covered by the coil ("DREAT") were determined knowing how much of the tank was covered by the coil ("XM").

The heat transfer fluid properties were determined from the manufacturer's specifications. Equations were determined from tables and graphs to approximate these properties within an error of 1%, for the thermal conductivity (CONFLI") and specific heat ("SHS") and within 5% for the liquid viscosity ("VISFLI," "VISFL2") for the temperature range 50-80°C. The large viscosity error is due to inadequate methods to approximate the rapid variation with temperature. At this time three fluids are used. One is a 40% propylene clycol, 60% water solution. This has adequate freeze protection (0°C), with high specific heat and thermal conductivity but a high viscosity which results in larger flow rates needed for turbulent flow. It is nontoxic with a low boiling point (105°C). The second fluid is a 40% ethylene glycol, 60% water mixture, with similar properties to the propylene mixture but it is very toxic with a somewhat higher boiling and' freezing point. The third product is Dowtherm J, a fluid with low viscosity, low specific heat, and low thermal conductivity. Although the flow rate needed for turbulent flow for Dowtherm J are much lower the heat transfer characteristics are very poor.

Water properties were determined from the HANDBOOK OF HEAT TRANSFER by Ceiringer pg. 105-110. These are correlated for thermal conductivity (CONDW3), specific heat (SH3), viscosity ("VISW3"), coefficient of volumetric expansion (BETAF) and density (DENSW3). Water is a good heat transfer fluid with low viscosity, high thermal conductivity and specific heat but is inadequate for use in the collection when freezing temperatures are possible.

At present environmental parameters are covered by sinusdidal variations. Later the ambient all temperature (array "TAMB" and variable TAMBLN") and incident radiation received on the collector (array "QINCID", variable "QI") will be for actual locations throughout the U.S. for the time being the sinusdidal variations are adequate. For "QI," the radiation is a maximum at noon tapering off to 0 at 6:00 and 1800. "TAMBLN" is a minimum at 6:00 and maximum at 1800.

A variable load of hot water was determined for the average home. According to consumer reports an average of 100 kg of hot water/day is used in a typical home. For this study 110 kg/day was used with the following pattern:

After the following types of parameters are determined the actual iterations of the program begin:

1) Collector Variables
2) Tank Variables
3) Coil Variables
4) Fluid and water characteristics
5) Environmental parameters

The first iteration of the main program is to vary the fluids used in the collector. It begins with: "DO 1900 I=i,N" and ends with statement "IEOn continue." All processes within the loop are done until all "N" fluids have been used. The next iteration is to vary the collector size as follows: "DO 1890 JA=1, Nsize" where Nsize is the number of different collector sizes to be used.

The following iteration is much more complex beginning with statement "1062 continue". The flowrate of the fluid ("G11" in kg/rm^2) is varied by assuming the flowrate is the first segment of the array "G" which holds all of the flow rate to be used in the study. When the pump is turned on this assumption governs the pump until turbulent flow results. This will be explained more later. Within these loops the water temperature within the tank ("TWAT") is initilized for the first hour of the first day. The value chosen is of little effect since the program continues until a steady state condition exists.

At statement "1063 CONTINUE" another loop begins. This is a convergence loop for the steady state case, where the program will run up to seven days until the temperature of one day match those of the previous day within a given tolerance.

Many variables are initialized for each day and before the actual hourly iteration begins at statement "DO 1830 J = 1, 24."

Within this loop many variables previously determined for each hour are evaluated for the particular hour such as the incident radiation and ambient temperature. An array "TEMP" holds the temperature history of the tank so that the convergence of successive days can be determined later.

The first major statement for the day is a control statement to determine whether the pump is on or off. PUMP is a variable which shows whether the pump is on (PUMP = 1), or off (PUMP = 0). If the pump is off then no heat is being transferred and the temperature of the collector is determined from the following relationship for an isothermal collector:

TLIQ = (QI / USLOPE) + TAMBCN,

where QI is the rate at which radiation is received on the collector ($W.m^{-2}$), USLOPE is the heat loss coefficient ($W.m^{-2}.C^{-1}$), TAMBCN is the ambient temperature in degrees C, and TLIQ is the temperature of the fluid in the collector in degrees C.

Thus "TLIQ" temperature of fluid can be obtained if the other variables are determined. If the difference between the fluid temperature and the water temperature exceeds the difference needed to turn the pump on (TSART) then a flowrate is determined for the pump on. If not then control will pass to statement "1400 CONTINUE" which will evaluate the performance for the pump off. If the pump was already on there was no need to recalculate the fluid exit temperature (TLIQ) and control would revert to "1854 CONTINUE". Variou' relationships for the flowrate are needed for different equations. Thus "GII" is the flowrate of the fluid in (kg/hrm^2), "G1" is the flowrate in (kg/hr m^2 of tube in collector), and "GLLLL" is the fluid flowrate in the tube around the storage tank in (kg/hrm^2).

The viscosity of the fluid is determined next in (lb/ft hr) in order to determine the minimum flowrate needed for turbulent flow ("GMIN11) in the collector tube. If the flowrate "G111" is greater than "GMIN11" then the flow is turbulent and control goes to statement "1855 CONTINUE." Otherwise, a new flowrate is used if turbulent flow does not occur by 1200. This is used so that early morning flow does not affect optimum collecting hour operation.

When the flow is turbulent then the other fluid properties are determined at the fluid inlet temperature (TL1Q1N). This results in some error since the fluid properties should be evaluated at some mean fluid temperature but this effect on the efficiency estimate of the collector is negligible so further refinement is not necessary. The heat transfer coefficient for the collector fluid (KHCW) is determined from these fluid properties and it can be shown that a 100% change in the heat transfer coefficient would result in a 1 or 2% change in the collector efficiency factor (ETA3). "KHCW" is determined by a correlation Pg. 219 of McAdam's Heat Transmission. This correlation is for smooth tubes with fully developed turbulent flow for the collector loop this correlation is adequate.

After the collector efficiency (ETA3) is determined, the rate heat is transferred from the collector to the fluid (QCOLL) can be determined. An iteration is then performed to find the actual temperature distribution through the collector starting with "1112 CONTINUE". The temperature of the fluid entering the collector (TLIQ) is known, which along with "QCOLL" can be used to determine the exit temperature of the fluid (TLIQ). Since it is assumed at present that no heat loss occurs in the pipe the temperature leaving the collector of the fluid is the entering temperature of the fluid of the coil around the tank.

Also after "TL1Q" is known, the water to fluid temperature difference is checked so that heat will be transferred from the collector to the tank. If "TLIQ - TWAT (Water temperature)" is greater than TSTOP then the pump is left on otherwise the pump is shut off and control reverts to "1400 CONTINUE" The flowrate through the coil is determined as are the fluid properties at "TLIQ". Thus will give erroneously high vales since "TLIQ" is greater than the mean temperature of the fluid but this effect is not great as long as the temperature drop is not excessive. The heat transfer coefficient for the fluid in the helical coil is increased over that of a horizontal tube by the correlation Page 228 McAdam's <u>Heat Transmission.</u> An iteration begins at statement "1120 CONTINUE" to determine the natural convection coefficient ("UL") for water inside of the tank. The wall temperature of the tank must be known in order to calculate ("UL") according to Page 124 of McAdams. The efficiency of the coil-tank system ("FPRIME") is then determined assuming the flowrate of the water in the tank due to natural convection is small compared to that of the fluid. An iteration to determine the temperature at which the fluid leaves the coil is then started similar to that of the collector exit temperature of the fluid. The rate at which heat is transferred to the water (QTOTIAL) is determined and a new mean wall temperature of the tank is found. This loop continues until the mean wall temperature converges. After convergence control passes to statement "1500 CONTINUE" which determines the final characteristics when the pump is on.

The temperature of the water is then determined after evaluating the rate heat is lost to the environment (XLOSS), and the rate heat is removed for the home's hot water use (XLOAD) by UMLIZINL as simple euler method. This final temperature of the water for the particular hour is then the beginning temperature of the water for the next hour for the 24 hour period. The program continues until all iterations are complete while printing of results needed for further evaluation.

APPENDIX F. Conversion Factors

Multiply	By	To Obtain
Flowrates		
$\frac{lb}{hr}$	0.4536	$\frac{kg}{hr}$
$\frac{lb}{hr}$	$\frac{0.124667}{\rho}$	$\frac{gal}{min}$
	ρ -Density of fluid $\frac{(lb)}{ft^3}$	
$\frac{lb}{ft^2 hr}$	4.8824	$\frac{kg}{hr\ m^2}$
Energy		
BTU	1055.06	Joules
Joule	1	Watt-Sec
Power		
$\frac{BTU}{hr}$	0.293071	Watts
Heat Flux		
$\frac{BTU}{hr\ ft^2}$	3.1546	$\frac{watts}{m^2}$
Coefficient of Heat Transfer		
$\frac{BTU}{hr\ ft^2\ {}^oF}$	5.67826	$\frac{Watts}{m^2\ {}^oC}$

CONVERSION FACTORS

Multiply	by	To Obtain
Specific Heat		
$\frac{BTU}{lb\ ^{o}F}$	1.162	$\frac{Watt\ hr}{Kg\ ^{o}C}$
$\frac{BTU}{lb\ ^{o}F}$	1	$\frac{cal}{gm\ ^{o}C}$
$\frac{BTU}{lb\ ^{o}F}$	4.1868	$\frac{kj}{kg\ ^{o}C}$
Thermal Conductivity		
$\frac{BTU}{hr\ ft\ ^{o}F}$	1.73073	$\frac{Watts}{m\ ^{o}C}$
Viscosity		
$\frac{lb}{ft\ hr}$	0.4132	Centipoises
$\frac{lb}{ft\ hr}$	$\frac{25.8065}{\rho}$	Centistokes
Density		
$\frac{lb}{Ft^3}$	16.0185	$\frac{kg}{m^3}$

ρ -Density of fluid $\frac{(lb)}{ft^3}$

APPENDIX G. SIMPLIFIED METHODS OF DETERMINING HEAT EXCHANGER SIZE

One of the objectives of the program was to produce an abstracted version of the conclusions of the study, for the benefit of those preparing handbooks or other compendia.

During the course of the study such a section was prepared for inclusion in the "Design and Installation Manual for Thermal Energy Storage," Report ANL-79-15 of the Solar Energy Group of Argonne National Laboratory, February 1979, which was included as Appendix D in that report. In Appendix G of the present report this section is reproduced in its entirety.

SIMPLIFIED METHODS OF DETERMINING HEAT EXCHANGER SIZE

Using a heat exchanger either as a means of separating antifreeze solution from the storage water or as a means of separating potable water from nonpotable water requires choosing a heat exchanger of the proper size. For purposes of calculating heat exchanger size there are two main types of heat exchanger systems, double-loop and single-loop. A double-loop system, illustrated in Figure G-1, requires two pumps (forced convection) to maintain positive control of the flow on both sides of the heat exchanger. A single-loop system has only one pump and typically features either a coil inside the tank or a coil fastened to the outside of the tank. Single-loop systems rely on bouyancy of the heated water to maintain flow on the tank side of the heat exchanger (natural convection). Forced convection is maintained on the other side of a single-loop system by a pump.

The use of a heat exchanger leads to a collection penalty, as shown in Figure G-1. The efficiency of collection decreases with increasing collection temperature, as shown in the curve in the lower part of the figure. The presence of the heat exchanger increases the collection temperature and hence produces the collection penalty.

DOUBLE-LOOP HEAT EXCHANGER SYSTEMS

De Winter first analyzed the case of a double-loop heat exchanger system and found that if capacity rates were used in the two loop so that:

$$(WC_p)_{coll} \leq (WC_p)_{sto} \qquad \text{G-1}$$

where:

W is the mass flow rate,

C_p is the heat capacity,

coll is the loop through the collector, and

sto is the loop through the storage tank, as shown in Figure G-1, then the heat collected by the collector-heat exchanger combination was simply reduced by the factor:

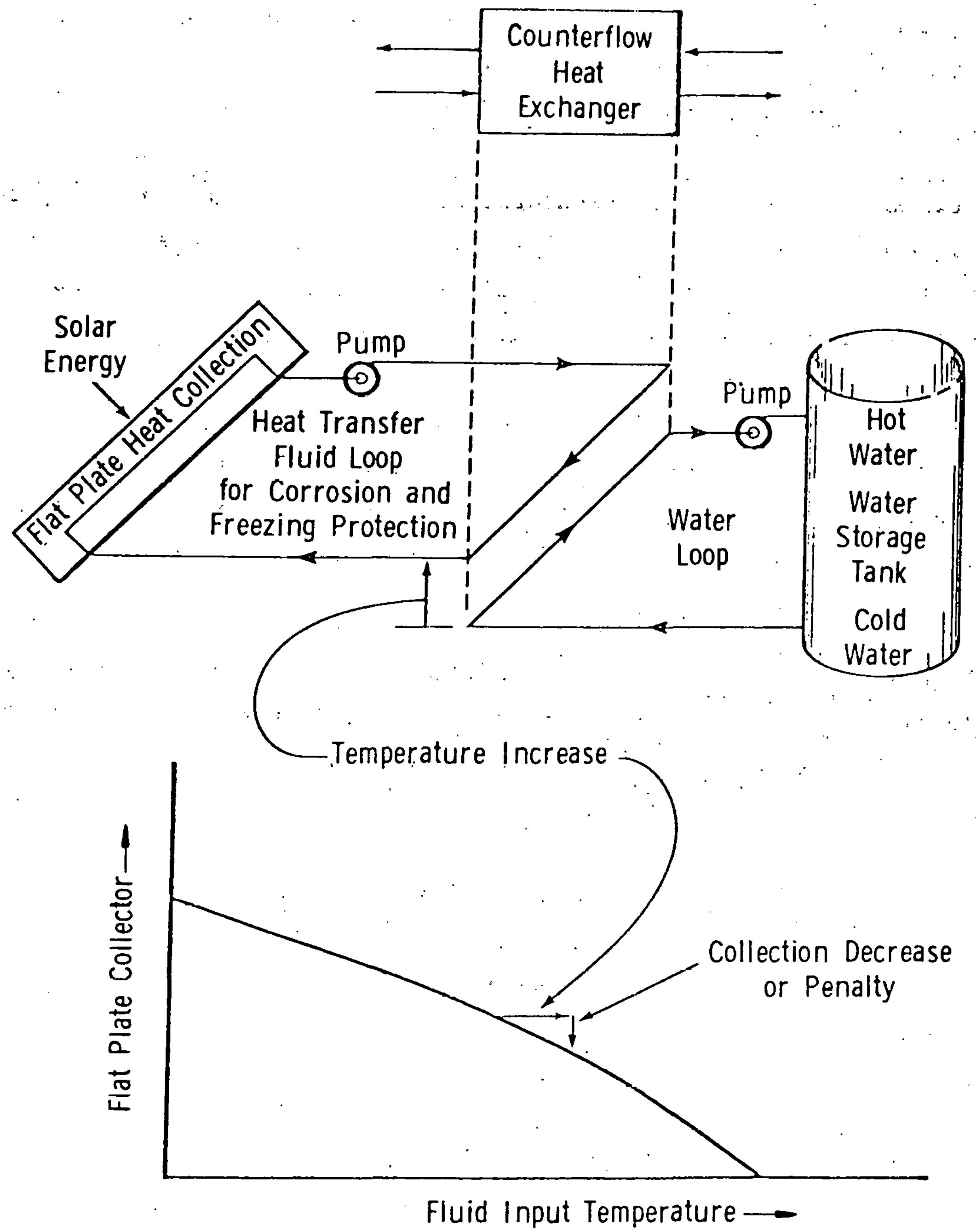

Figure G-1. Heat Collection Decrease Caused by a Double-Loop Heat Exchanger

$$\frac{F_R'}{F_R} = \frac{1}{1 + \frac{F_R U_c A_c}{(WC_p)_{coll}} \left[\frac{1}{\varepsilon} - 1\right]} \qquad \text{G-2}$$

where:

- F_R is the standard collector efficiency factor of the Hottel-Whillier flat plate collector model.
- F_R' is the same factor modified by the heat exchanger effect.
- A_c is the area of the collector.
- U_c is the collector heat loss coefficient.
- ε is the heat exchanger effectiveness.

Klein, Beckman, and Duffie extended this to systems in which Equation G-1 does not hold and determined that for this more general case:

$$\frac{F_R'}{F_R} = \frac{1}{1 + \frac{F_R U_c A_c}{(WC_p)_{coll}} \left[\frac{(WC_p)_{coll}}{\varepsilon\ (WC_p)_{min}} - 1\right]} \qquad \text{G-3}$$

Equation G-3, which is completely general, is shown in Figure G-2. In the general case, the heat exchanger effectiveness is an exponential function of the parameters NTU = $(U_x A_x)/(WC_p)_{min}$ and of $(WC_p)_{max}$ as shown in Equations G-4a and G-4b. (A_x is the heat exchanger heat transfer area and U_x the associated overall heat transfer coefficient.)

$$\varepsilon = \frac{1 - e^{-N}}{\left[1 - (WC_p)_{min}/(WC_p)_{max} e^{-N}\right]} \qquad \text{G-4a}$$

with

$$N = NTU \left[1 - (WC_p)_{min}/(WC_p)_{max}\right] \qquad \text{G-4b}$$

The effectiveness increases as the heat exchanger heat transfer area increases. This reduces the collection penalty--it increases F_R'/F_R, bringing it closer to 1--so that heat collection is increased. However, increasing the heat exchanger size increases the system cost. By doing

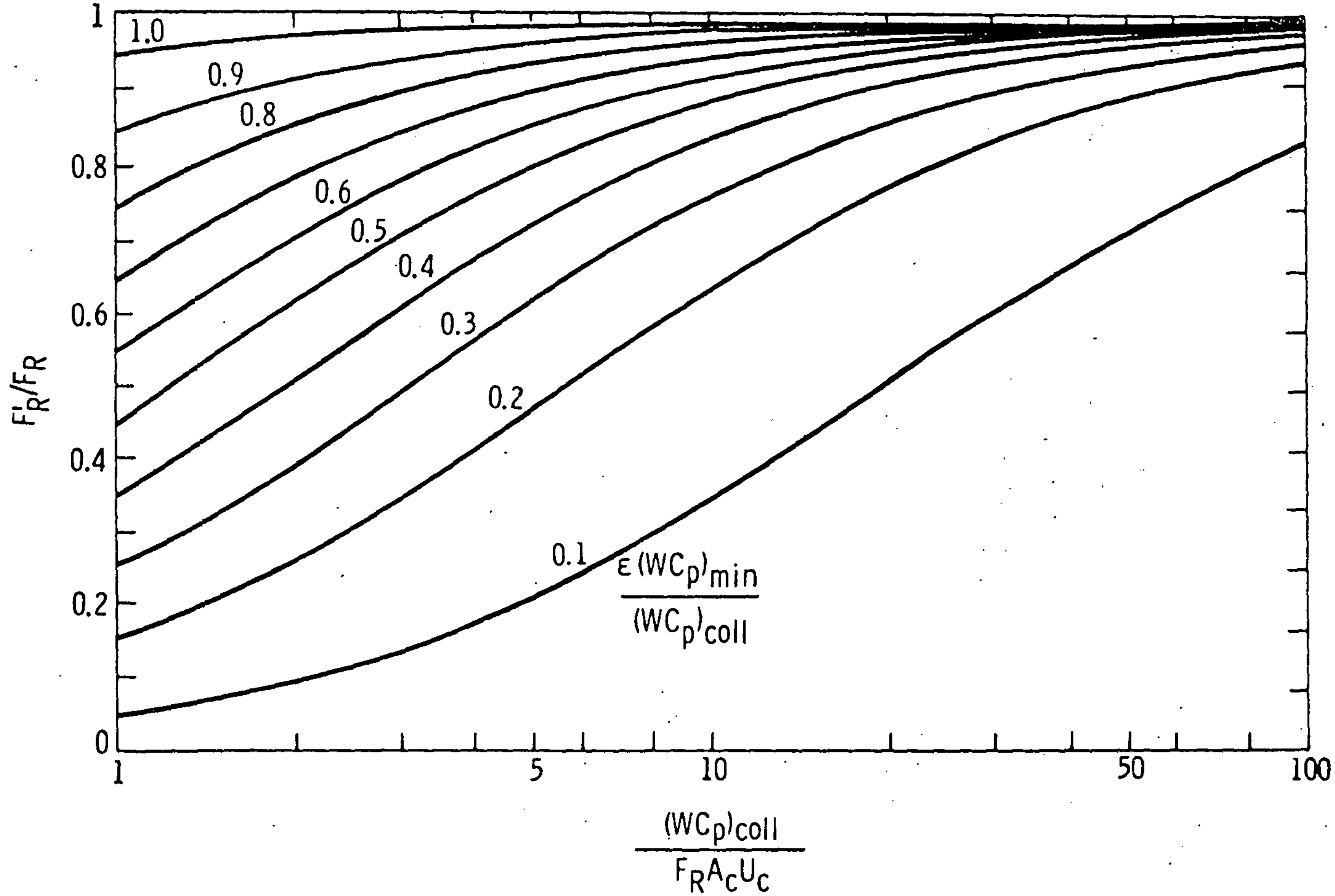

Figure G-2. Collector Heat Exchanger Factor F'_R/F_R

a computer simulation the designer can find an optimum heat exchanger size as illustrated in Figure G-3.

For the specific case in which:

$$(WC_p)_{coll} = (WC_p)_{sto} \qquad \text{G-5}$$

de Winter found that:

$$\frac{F_R'}{F_R} = \frac{1}{1 + \dfrac{F_R U_c A_c}{U_x A_x}} \qquad \text{G-6}$$

since:

$$\varepsilon = \frac{1}{1 + \dfrac{(WC_p)_{coll}}{U_x A_x}} \qquad \text{G-7}$$

When the cost per unit area of the collector (C_c) and the cost per unit area of the heat exchanger (C_x) are constant, de Winter further found that if the heat transfer coefficient U_x did not vary with the area A_x the optimum heat exchanger area A_x could be calculated from the equation:

$$A_x = A_c \left[\frac{F_R U_c C_c}{U_x C_x}\right]^{1/2} \qquad \text{G-8}$$

According to Horel and de Winter, with a given <u>average</u> WC_p product, the optimum heat exchanger invariably had a storage capacity rate $(WC_p)_{sto}$ higher than its collector capacity rate $(WC_p)_{coll}$, so that Equation G-1 was invariably satisfied and Equation G-2 applied. For typical values of the collector capacity rate $(WC_p)_{coll}$, they found that the value of $C' = (WC_p)_{coll}/(WC_p)_{sto}$ ranged from 0.5 to 0.6 and that, for all practical purposes, Equation G-8 could still be used to find the optimum heat exchanger area, since this was only about 1 percent different from that found for the optimum (unmatched capacity rate) case.

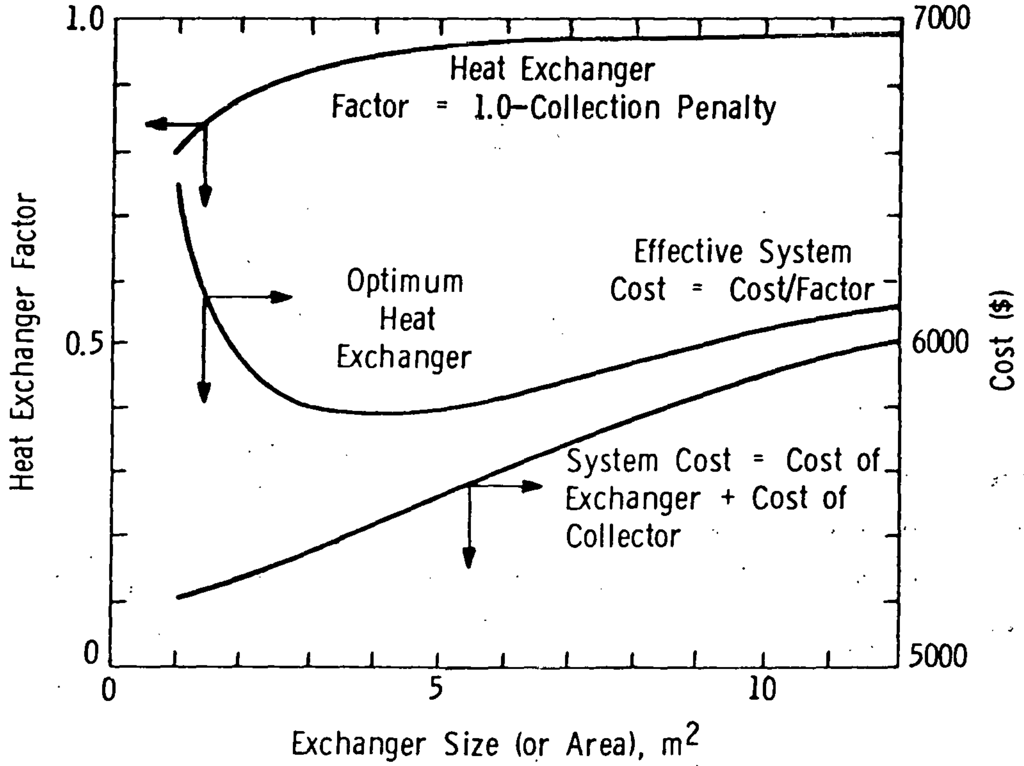

Figure G-3. Typical Heat Exchanger Optimization Plot, Showing the Heat Exchanger Factor, Total System Cost, and Effective System Cost as a Function of Heat Exchanger Size or Area

SINGLE-LOOP HEAT EXCHANGER SYSTEMS

An analysis for a single-loop system, using a traced tank or a coil in a tank, was performed by Horel and de Winter. They found that the same heat exchanger factor determined for a double-loop system in Equation G-2 could be used for the single-loop system. Again the designer can determine an optimum heat exchanger area using the methodology shown in Figure G-3. The main difficulty in this case lies in the fact that the heat transfer coefficients used to determine U_x are no longer straightforward forced convection coefficients, since on the water (storage) side there is a natural convection coefficient that is harder to determine. This area is addressed in the next section.

FORCED CONVECTION IN HEAT EXCHANGERS

Using heat transfer coefficients will enable the designer to optimize the heat transfer of a collector-to-storage system. They also allow easy comparison of heat transfer fluids. Heat transfer coefficients within the tubes and outside the tubes (i.e., shell-side heat transfer coefficients) as well as overall heat transfer coefficients must be used. In the following sections, each of these coefficients is discussed.

Inside Tube Heat Transfer Coefficients

For a double-loop heat exchanger system, inside tube heat transfer coefficients must be specified for the collector tubes and the exchanger tubes. For a traced tank, the inside tube heat transfer coefficients must also be determined for the helical coil.

The inside tube heat transfer coefficient is dependent upon:

- Flow rate through the tube
- Cross-sectional area of the tube
- Temperature of operation
- Properties of the fluid at the operating temperature.

Depending on the state of the fluid (i.e., laminar, transitional, or turbulent) different correlations must be used to determine the inside tube heat transfer coefficients (h_i). For the laminar region (Reynolds number < 2500) the following correlation from McAdams can be used:

$$h_i = \frac{KC_2}{D_i} \qquad \text{G-9}$$

where:

$$C_2 = 1.75 \left(\frac{G'C_p}{KL} \right)^{1/3} \quad \text{for } 1.75 \left(\frac{G'C_p}{KL} \right)^{1/3} > 3.66$$

$$C_2 = 3.66 \quad \text{for } 1.75 \left(\frac{G'C_p}{KL} \right)^{1/3} < 3.66$$

h_i = inside tube heat transfer coefficients (Btu/hr·°F·ft^2)
K = thermal conductivity of fluid (Btu/hr·°F·ft)
D_i = inside tube diameter (ft)
G' = flow rate (lb/hr)
C_p = heat capacity of fluid (Btu/lb·°F)
L = tube length (ft)

Thus, in the upper laminar region, the inside tube heat transfer coefficient also depends upon the tube length.

For the transitional region (2500 < Reynolds number < 7100), h_i becomes:

$$h_i = (C_p \mu/K \qquad \text{G-10}$$

where:

$$J' = 0.116 (Re^{2/3} - 125)/Re$$

and

Re = Reynolds number
= $G''D_i/\mu$
μ = viscosity of fluid (lb/ft·hr)
G'' = flow rate per tube (lb/ft^2.hr)

For the turbulent region (Re > 7100), the inside tube heat transfer coefficient from McAdams is for values of L/D greater than 60.

$$h_i = \frac{0.023\ K\ Re^{0.8}}{D_i} (C_p\ \mu/K)^{0.4} \qquad \text{G-11}$$

Since, in general, the transitional region should be avoided, it was included only to provide continuity from the laminar to the turbulent regimes. Also note that, at the interface between transitional and turbulent (Re = 7100) and the interface between laminar and transitional (Re = 2500), the equations do not predict similar inside tube heat transfer coefficients. For Re = 7100 there is a 10-percent difference between the two equations, whereas around Re = 2500 the error is larger. The selection of the transitional region between Reynolds numbers 2500 and 7100 was completely arbitrary. It was chosen to minimize the errors at the two boundaries and to allow reasonable heat transfer in the lower turbulent region.

For the traced tank system, the effect of the fluids operating in the laminar regime is greater, since the heat transfer coefficients can affect the helical coil efficiency. Heat exchangers operating in laminar flow have much lower effectiveness than those operating in the turbulent regime.

Some simple relationships between the flow rate in gallons/(minute·per tube) and the other flow rates follow.

$$Q = Q_n N \qquad \text{G-12}$$

$$Q_n = \frac{0.1247\ G'}{\rho N}$$

$$G'' = \frac{4G'}{D_i^2 N \pi}$$

where:

Q = total system flow rate (gallons/minute)

N = total number of tubes

Q_n = flow rate per tube (gallons/minute)

ρ = density of the fluid (lb/ft^3)

<u>Shell-Side Heat Transfer Coefficient</u>

The shell-side heat transfer coefficient within the heat exchanger (h_o) was determined for those fluids studied. h_o is a function of:

- Shell entrance flow rate
- Temperature of operation
- Fluid properties at the operating temperature
- Characteristics of the heat exchanger
  - (1) Tube pitch
  - (2) Baffle spacing
  - (3) Outer tube diameter
  - (4) Number of tube rows.

A correlation was found from Kreider and Kreith.

$$h_o = 0.33\ Re'^{\,0.6}\ (C_p\ \mu/K)^{0.33}\ K/D_o$$

Re' = Reynolds number through minimum cross-sectional area of heat exchanger

G_{max} = flow rate through the minimum cross-sectional area of the heat exchanger ($lb/ft^2 \cdot hr$)

$$= \frac{G_s}{A_{min}\ (N_{row} + 1)}$$

G_s = total shell flow rate (lb/hr)

Q_s = total shell flow rate (gal/min)

$= 0.1247\ G_s/\rho$

A_{min} = minimum cross-sectional area (ft^2)

$= S_{baf} S_{min}$

S_{baf} = baffle spacing (ft)

S_{min} = tube spacing (ft)

= (pitch -1) D_o

Pitch = equilateral triangular pitch

= 1.25

N_{row} = number of tube rows across diameter of shell. This is a conservative estimate of the number of tube openings available for the fluid to flow through.

Figure G-4 shows the exchanger characteristics more clearly.

Overall Heat Transfer Coefficient

The overall heat transfer coefficient of a heat exchanger (U_x) can be determined from the following equation by Kays and London.

$$U_x = \frac{1}{\frac{1}{h_o} + \frac{1}{h_{so}} + \frac{D_o}{D_i h_i} + \frac{D_o}{D_i h_{is}} + R_{wall}} \qquad \text{G-13}$$

where:

h_{so} = shell side scaling coefficient (Btu/hr·ft^2·°F)

h_{is} = inside tube scaling coefficient (Btu/hr·ft^2·°F)

R_{wall} = tube wall heat transfer resistance (hr·ft^2·°F/Btu)

$$= \frac{D_o}{2\,K_{tex}} \ln \frac{D_o}{D_i}$$

K_{tex} = thermal conductivity of the tube wall (Btu/hr·ft·°F)

The scaling coefficients can be assumed constant for nearly all fluids and tube sizes and equal to 100 Btu/hr·ft^2·°F. If the water is very hard (over 15 grains/gallon), a scaling coefficient of 330 Btu/hr·ft^2·°F can be specified. The reciprocal of the scaling coefficient, known as the fouling factor, is frequently specified instead of the scaling coefficient. Normally, scaling coefficients decrease with time if maintenance is not periodically performed because of increased scaling deposits on the inner and outer tube walls. This can reduce the performance of the heat exchanger and increase the possibility of corrosion.

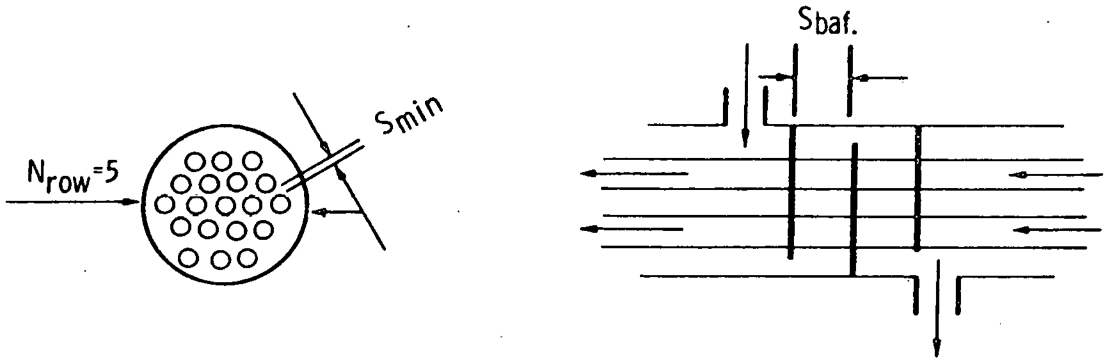

Figure G-4. Shell-Side Heat Exchanger Dimensions

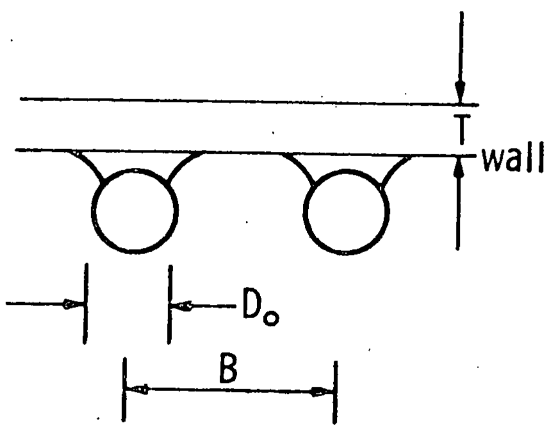

Figure G-5. Traced-Tank Heat Exchanger Dimensions

Since the wall resistance for copper tubing within the heat exchanger is generally negligible, the equation for the overall heat transfer coefficient cannot be reduced further.

NATURAL CONVECTION IN TANKS WITH INTERNAL COILS OR IN TRACED TANKS

One difficulty with coils in tanks or with traced tanks involves the natural convection heat transfer coefficient on the tank side. Forced convection heat transfer coefficients are normally determined entirely by the flow conditions. Natural convection coefficients, however, are determined by the geometry of the heating (or cooling) surface, by the temperature difference between the surface and the fluid, and by the fluid properties.

Natural Convection Equations

The conduction problem between the inside tank wall and the fluid in the tubes is analogous to that obtained in a flat plate collector with the tubes bonded below the plate. The heat transfer rate is given by the product of inside water film coefficient h_t, inside tank heat transfer area A_t, F_t, and fluid to water temperature difference. According to Duffie and Beckman:

$$F_t = \frac{1}{\frac{Bh_t}{\pi D_o h_i} + \frac{Bh_t}{C_{bond}} + \frac{B}{D_o + (B - D_o) F}} \qquad \text{G-14}$$

where:

B = spacing between tubes (ft)

D_o = outside diameter of the coil tube (ft)

h_i = heat transfer coefficient of fluid circulating through the coil (Btu/hr·ft^2·°F)

C_{bond} = conductance of tank to coil bond $\approx \frac{4T_{wall}k_{wall}}{D_o}$ (Btu/hr·ft·°F)

This value of the bond conductance was determined by de Winter.

T_{wall} = thickness of tank wall (ft)

k_{wall} = conductivity of tank wall (Btu/hr·ft·°F)

F = fin efficiency of tank wall between the tubes, for heat losses to the water.

Figure G-5 shows the relationship among the parameters of the traced tank.

The natural convection coefficients are given by McAdams. Equations 7-4a and 7-4b in his book pertaining to vertical plates are reproduced here as Equations G-17 and G-20. Equation 7-6a in his book pertaining to horizontal cylinders can be replaced with Equation G-20 if the tube diameter is replaced by a "flow length" L equal to half the tube perimeter:

$$L = \pi \frac{D_o}{2} \qquad \text{G-15}$$

For the turbulent regime, defined by

$$10^9 < (KL^3 \Delta T) < 10^{12}, \qquad \text{G-16}$$

the heat transfer coefficient is given by McAdams's Equation 7-4a for vertical plates:

$$\frac{h_t L}{k} = 0.13(KL^3 \Delta T)^{1/3} \qquad \text{G-17}$$

or, in a simplified form,

$$h_t = 0.13kK^{1/3} \Delta T^{1/3} = A_i \Delta T^{1/3}. \qquad \text{G-18}$$

For the laminar regime, defined by

$$10^4 < (KL^3 \Delta T) < 10^9, \qquad \text{G-19}$$

the heat transfer coefficient for both vertical plates and horizontal tubes is given by

$$\frac{h_t L}{k} = 0.59(KL^3\Delta T)^{1/4} \qquad \text{G-20}$$

or, in simplified form,

$$h_t = 0.59kK^{1/4}(\frac{\Delta T}{L})^{1/4} = A_1(\frac{\Delta T}{L})^{1/4}. \qquad \text{G-21}$$

It should be noted that tubes are almost certain to stay in the laminar natural convection regime in solar applications unless the tank is stirred up.

In the above equations:

L is the natural convection flow length along the surface (ft) for vertical plates and vertical tubes, and L must be calculated from Equation G-15 for horizontal tubes.

D_o is the outside diameter of the tube (ft)

$K = \frac{\rho^2 g}{\mu^2} \beta (\frac{C_p \mu}{k})$ See Table G-1.

β is the fluid thermal expansion coefficient ($ft^3/ft^3 \cdot °F$).

ΔT is the temperature difference between the wall and the fluid (°F).

h_t is the natural convection heat transfer coefficient ($Btu/hr \cdot ft^2 \cdot °F$).

ρ is the fluid density (lb/ft^3).

g is the acceleration of gravity = 4.17×10^8 ft/hr^2.

C_p is the heat capacity ($Btu/lb \cdot °F$).

μ is the viscosity of the fluid ($lb/hr \cdot ft$).

k is the thermal conductivity of the fluid ($Btu/hr \cdot ft \cdot °F$).

The values of K, of k, and of A_i and A_1 used in Equations G-18 and G-21 are given in Table G-1 for water as a function of temperature T in degrees Fahrenheit.

Table G-1. Convection Factors for Water

T	K	k	A_i	A_1
60	$.337 \times 10^9$	0.338	30.58	27.02
80	$.557 \times 10^9$	0.351	37.54	31.81
100	$.959 \times 10^9$	0.363	46.54	37.69
120	1.453×10^9	0.372	54.77	42.85
140	2.189×10^9	0.379	63.97	48.37
160	2.785×10^9	0.385	70.42	52.18
180	3.660×10^9	0.390	78.13	56.60

Recommended Iteration Procedure

Since the natural convection heat transfer coefficient is a function of the temperature difference, it is necessary to iterate to determine the final heat transfer situation. The recommended scheme below will lead to convergence to within about 1 percent within 4 or 5 iterations.

(1) Calculate the heat transfer coefficient h_i on the forced convection side (usually Equation G-11, but possibly Equation G-9 or G-10).

(2) Assume a natural convection heat transfer coefficient h_t of 100 Btu/hr·ft²·°F to start the calculation process.

(3) Calculate $U_x = h_t F_t$, based on h_i, h_t, and the conduction geometry.

(4) Calculate $NTU = \frac{U_x A_t}{W C_p}$.

(5) Calculate the effectiveness for the coil or traced tank from $\varepsilon = 1 - e^{-NTU}$.

(6) Calculate F_R' / F_R. (Use Equation G-2, G-3, or G-6.)

(7) Calculate the collected heat Q from the collector performance map.

(8) With Q and h_t and the natural convection area, calculate $\Delta T_{avg} = \frac{Q}{h_t A}$.

(9) Calculate the natural convection heat transfer coefficient h_t obtained with this temperature difference. (Use Equation G-20 or G-21.)

(10) Go back to Step 3 and go through the calculations until the numbers in successive iterations no longer change appreciably.

It should be noted that this calculation applies to two types of systems:

- The case in which an antifreeze loop heats a traced storage tank or a storage tank with a coil. This involves water being heated by natural convection.
- The case in which a domestic water line is being heated by a storage tank with a coil or by a traced storage tank. This involves water being cooled by natural convection.

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