APPENDIX B

WIND SPEED MODIFICATION DEVELOPMENT METHODOLOGY

B.0 INITIAL ASSUMPTIONS

By definition, the "y" intercept of an unglazed collector's performance curve (Figure B.1) is considered to represent the following operating conditions:

1. Average collector temperature ≅ average air temperature
2. All collector heat gain is from solar radiation
3. Virtually all heat loss may be attributed to reradiation or reflection losses.

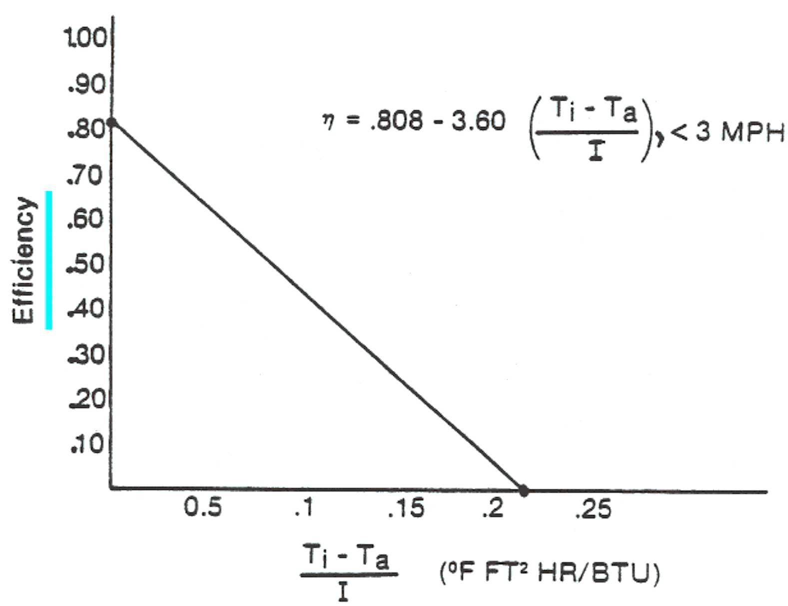

Figure B.1

Typical Unglazed Collector Performance Curve

B.1 CALCULATION OF LOSSES AT "Y" INTERCEPT

Using these assumptions and ASHRAE 96-80 test performance results for a given collector, we may evaluate the magnitude of the radiation and reflection losses for that collector under "y" intercept conditions

Table B presents averaged test conditions and results for unglazed plastic and flexible mat collectors in widespread use at the end of September 1983 The numbers are adjusted slightly to make it easier for the reader to follow the calculations

TABLE B.1

Averaged Test Conditions (English Engineering Units)

1) composite first order efficiency equation:

$$\text{collector decimal efficiency} = .808 - 3.60 \ \frac{t_i - t_a}{I}$$

2) insolation rate: 300 $Btu/ft^2 \cdot hr$
3) solar absorptance: 0.9
4) temperature rise across collector 4°F
5) total loss (at "y" intercept): 192 x insolation
6) assumed reflectance: 10 x insolation
7) radiation loss (at "y" intercept): 092 x insolation
8) ambient temp ≅ inlet temp = 90°F
9) average collector temperature = 90°F + 4°F/2 = 92°F

- The radiation loss may be calculated from Table B.1, item 7

$$300 \times .092 = 27.6 \ Btu/ft^2 \cdot hr$$

- The effective sky temperature may be estimated from an accepted radiant heat exchange formula

$$Q_{reradiation} = .1714 \times A \times \varepsilon \left[\left(\frac{t_{coll}}{100} \right)^4 - \left(\frac{t_{sky}}{100} \right)^4 \right]$$

Substituting the known values

$$1714 \times 1 \times 9 \left(\frac{92+460}{100} \right)^4 \qquad \left(\frac{t_{sky}}{100} \right)^4$$

$$.1543 \quad 928.5 \quad -\left(\frac{t_{sky}}{100} \right)^4$$

$$1543 \left(\frac{t_{sky}}{100} \right)^4 = 143-27.6$$

$$t_{sky} = 100 \times \sqrt[4]{\frac{115.4}{.1543}}$$

$$= 523°R = 64°F$$

- This sky temperature may be used in combination with information from the collector's performance curve to evaluate reflection and reradiation losses from the collector at other fluid parameters -- for example, at .05 and .1 (English Engineering Units). That having done, convection losses may be calculated for the same fluid parameter values using the formula

$$Q_{cv} = h_{cv} \times A \times \Delta T$$

- New values for h_{cv} (accurate enough for these calculations) for increased wind speeds may be developed using the formula:

h_{cv} 1 + .3 x wind speed (in mph)

- The new h_{cv} values may be used to calculate convective losses at wind speeds higher than 3 mph and at specific fluid parameters (say .05 and 1). The losses may be totaled and new efficiency curves and equations generated for 5-, 10- and 15-mph wind speeds It should be remembered that the wind speeds must be those which occur across the <u>surface of the unglazed collector array</u> if the new curves and equations are to yield useful results. (The 5, 10 and 15 mph efficiency curves and their corresponding first order efficiency equations were used in an FSEC program for a PDP 11/34 computer to generate the nomograph which appears as Figure 5.3 on page 5.11

CALCULATION OF LOSSES AT OTHER FLUID PARAMETERS

During testing to ASHRAE 96-80 specifications, it is usually easier to raise the inlet temperature of liquid entering the collector undergoing test than it is to lower the ambient air temperature Establishment of a data point corresponding to a fluid parameter of 05, an ambient temperature of 90°F, and an insolation rate of 300 Btu/ft²·hr would require a fluid inlet temperature of:

$$05 \quad \frac{t_i - 90}{300}; \; t_i = 105°F$$

Under actual pool heating conditions this could correspond to an insolation rate of 300 Btu/ft²·hr, a pool temperature (assumed to be the collector inlet temperature) of 85°F, and an ambient temperature of 70°F (This is characteristic of daytime, winter conditions in much of the Sun-belt. It also could correspond to a spa temperature of 100°F and an air temperature of 85°F at an insolation rate of 300 Btu/ft²·hr

The total osses at the 05 point on the performance curve are

$(1.00-.628) \times 300 = 111.6$ Btu/ft²·hr

(Efficiency at .05 = $.808-(.05 \times 3.60) = .628$)

- The average temperature of the collector surface is approximately:

105°F (inlet temp) + $\frac{4°F}{2}$ (temp rise) or 107°F

- Thus, the reradiation loss may be approximated as

$$Q_{rerad} = .1714 \times 1 \times .90\ (1034 - 748) = 44 \text{ Btu/ft}^2\cdot\text{hr}$$

- The convective loss at 3 mph is:

$Q_{cv} = h_{cv} \times A \times \Delta t$; wind speed = 3 mph

$Q_{cv} = (1 + .3 \times 3) \times 1 \times (107 - 90) = 32$ Btu/ft²·hr

- We now may calculate the conduction loss at a fluid parameter of .05 by using a heat balance

$Q_{fluid} + Q_{reflect} + Q_{rerad} + Q_{conv} + Q_{cond} = Q_{rad}$

$Q_{cond} = 300 - (.628 \times 300 + .1 \times 300 + 44 + 32)$

$Q_{cond} = 6$ Btu/ft²·hr

- Employing the formula $h_{cv} = 1. + .3 \times$ wind speed (mph) we may calculate the convective losses to be expected at a fluid parameter of 05 and wind speeds of

5 mph; $h_{cv} = 1 + .3 \times 5 = 2.5$

10 mph $h_{cv} = 1 + \ 3 \times 10 = 4.0$

15 mph $h_{cv} = 1 + .3 \times 15 = 5.5$

By ratio $_{cv}$ =

at 5 mph; $32 \times \frac{2.5}{1.9} = 42$ Btu/ft²·hr

at 1(mph; $32 \times \frac{4.0}{1.9} = 67$ Btu/ft²·hr

at 1! mph; $32 \times \frac{5.5}{1.9} = 93$ Btu/ft²·hr

Total losses at 5 mph are:
44 (rerad) + 30 (reflect.) + 6 (cond.) + 42 (conv.) = 122 Btu/ft²·hr

Total losses at 10 mph are:
44 (rerad) + 30 (reflect.) + 6 (cond.) + 67 (conv.) = 147 Btu/ft²·hr

Total losses at 15 mph are:
44 (rerad) + 30 (reflect.) + 6 (cond.) + 93 (conv.) = 173 Btu/ft²·hr

Table B. gives fractional losses and corresponding collector efficiencies for a fluid parameter of 05 (for the collector in Figure B.1)

The convective losses at increased wind speed may be added to the reflective, reradiative and conductive losses at a fluid parameter point of .05 and a new efficiency value derived that is substantially correct.

TABLE B.2

Calculated Collector Efficiencies with Fluid Parameter of .05

Wind Speed	Fractional Loss	Collector Efficiency
5	$\frac{122}{300}$ = 41	1 - .41 = 59
10	$\frac{147}{300}$ = .49	1 - 49 = 51
15	$\frac{173}{300}$ = 58	1 - .58 = 42

The process may be repeated for a fluid parameter of 10

10 300 = t_i - 90

120° = t_i

The efficiency at a fluid parameter of 10 is 448

Total losses ar(

$$(1 - 448) \times 300 = 166 \text{ Btu/ft}^2 \text{ hr}$$

temperature rise across the surface is slightly less (only about 2°F) because the quantity of energy absorbed at the lower collector efficiency corresponding to a fluid parameter of .1 is about half that absorbed at the "y" intercept point but the rate of fluid flow has remained constant. Thus, the average collector temperature is 121°F.

One set of operating conditions corresponding to this test point is a pool temperatur of 85°F and an ambient temperature of 55°F

- Reradiation losses may be approximated as:

$$Q_{rad} = .1714 \times 1 \times .9\left[\left(\frac{121 + 460}{100}\right)^4 - \left(\frac{64+460}{100}\right)^4\right] = 1543 \times 386$$

$$= 60 \text{ Btu/ft}^2\cdot\text{hr}$$

- The convective loss at 3 mph is:

$$= h_{cv} \times A \times \Delta t; \text{ wind speed 3 mph}$$

$$= (1 + .3 \times 3) \times 1 \times (121-90)$$

$$= 59 \text{ Btu/ft}^2\cdot\text{hr}$$

- A heat balance establishes the conductive loss as

$$Q_{cond} = 300 - (.448 \times 300 - .1 \times 300 + 60 + 59)$$

$$Q_{cond} = 17 \text{ Btu/ft}^2\cdot\text{hr}$$

- Again the convective loss may be ratioed to bring it into correspondence with:

$$5 \text{ mph}; 59 \times \frac{2.5}{1.9} = 78 \text{ Btu/ft}^2\cdot\text{hr}$$

$$10 \text{ mph}; 59 \times \frac{4.0}{1.9} = 124 \text{ Btu/ft}^2\cdot\text{hr}$$

$$10 \text{ mph}; 59 \times \frac{5.5}{1.9} = 171 \text{ Btu/ft}^2\cdot\text{hr}$$

The total losses are

5 mph; 60 + 30 + 17 + 78 = 185 $Btu/ft^2 \cdot hr$

10 mph; 60 + 30 + 17 + 124 = 231 $Btu/ft^2 \cdot hr$

15 mph; 60 + 30 + 17 + 171 = 278 $Btu/ft^2 \cdot hr$

Table B gives fractional losses and corresponding collector efficiencies for a fluid parameter of .1 (for the collector in Figure B.1

B.3 TABULATION OF DERIVED EFFICIENCIES

TABLE B.3

Calculated Collector Efficiencies with Fluid Parameter of

Wind Speed	Fractional Loss	Collector Efficiency
5	$\frac{185}{300}$ = .62	1 - .62 = 38
10	$\frac{231}{300}$ = .77	1 - .77 = 23
15	$\frac{278}{300}$ = 93	1 -.93 = 07

Figure 5.2 (page 5-10) presents 3, 5, 10 and 15 mph performance curves for the composite collector performance curve represented by Figure B.1 (page B-1) It is a plot of the values from Tables B.2 and B.3 The slope correction factors in Table 5.3 are derived from Figure 5.2.

B.4 QUALIFICATIONS

In the preceding calculations, average static conditions have been assumed. In truth, heat transfer occurs under highly variable insolation, ambient temperature and surface wind speed conditions. For a high degree of accuracy, analysis should be based on the dynamic nature of operating conditions However, because of the unpredictability of weather, approximate results derived from the static analysis often are accurate enough for solar pool heating design and installation

B.5 COMPARISON BETWEEN CALCULATED VALUES AND MEASURED RESULTS

Holbaugh ınd Huggins[1], of the Testing and Laboratories Division of FSEC have developed experimental data under Florida weather conditions for an unglazed plastic sheet collector whose ASHRAE 96-80 performance approximates that of the hypothetical collector the performance of which is represented by Figure B.1. Figure B.2 presents that emperical data

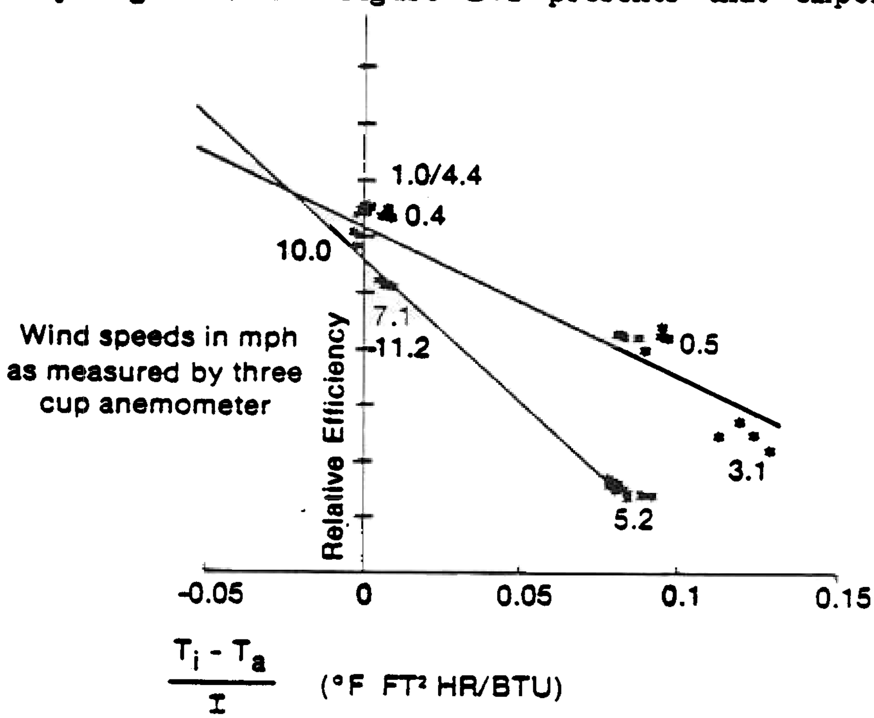

Figure B.2

Measured Variation of Unglazed Collector Performance with Wind Speed

Lumsdaine[2] has presented results of theorectical analysis and experimental testing of glazed flat-plate collectors (under conditions acceptable to the ASHRAE 95-73 rating procedure) He presents convincing arguments that testing under varying wind conditions (1-10 mph), varying tilt angles, and varying ratios of diffuse to direct radiation can lead to fairly substantial discrepancies in comparative test results The magnitude of discrepancies caused by varying wind speeds is inversely related to the number of cover plates Figure B.3 is an adaption of his data for a flat-plate collector with no and with two cover plates

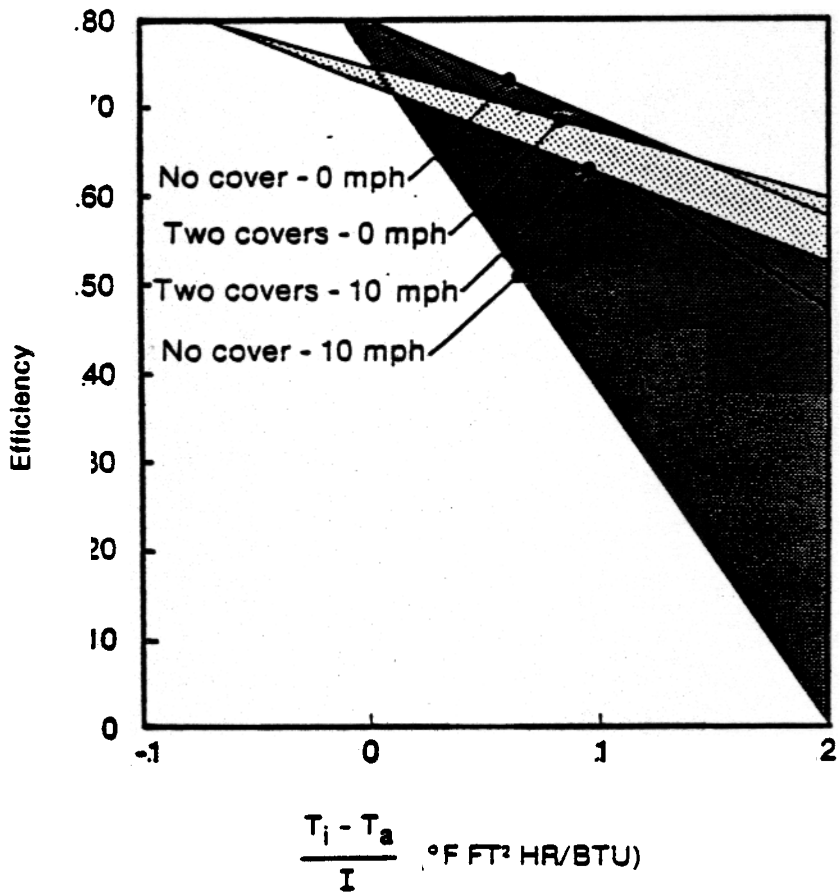

Figure B.3

Adaptation of Lumsdaine's Data

As shown by the data presented in Figure B.3, the effect of wind speed on unglazed collectors is substantial but that effect on glazed collectors is of little consequence in the pool heating range of fluid parameters (less than about .1 in EE units)

It should be noted that the information presented in Figure B.3 may not be compared directly with that presented in Figures B.1 and B.2 The values were developed for different kinds of collectors

Jenkins and Reed[3] also have reported on the effect of wind speed on unglazed collectors. Figures B.4 and B.5 are reprinted from their report. Their findings agree with those reported by Holbaugh and Huggins, and with those reported by Lumsdaine

An examination of the figures discloses that wind speed effects both the slope of the performance curve <u>and</u> its y intercept. The increase in slope is translated into poor performance beyond certain windspeed/temperature limits. The decrease intercept has a far less dramatic impact on collector performance. The reasons are these: the decrease in intercept is only a few percent and is applicable only when the term

$$\frac{t_i - t_a}{I}$$

approaches 0 -- that is when unglazed collectors are operating at about ambient temperature. Under these weather conditions the collector performance is near its maximum level, windy or not. Additionally, under windy conditions collector efficiency actually improves if the air is warmer than the pool.

A comparison of Figure 5.2 with Figures B.2, .3, 4 and 5 shows acceptable agreement between calculated and measured curves for purposes of predicting windy weather performance of unglazed collectors in Florida

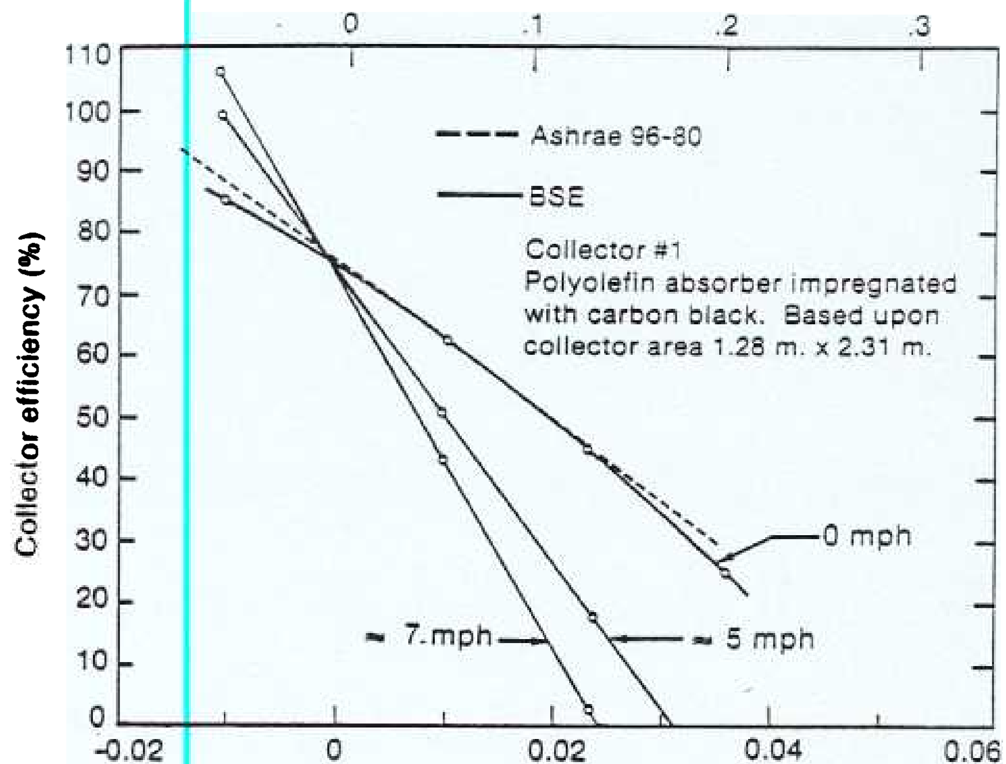

Figure B.4

Performance Curves: Unglazed Plastic

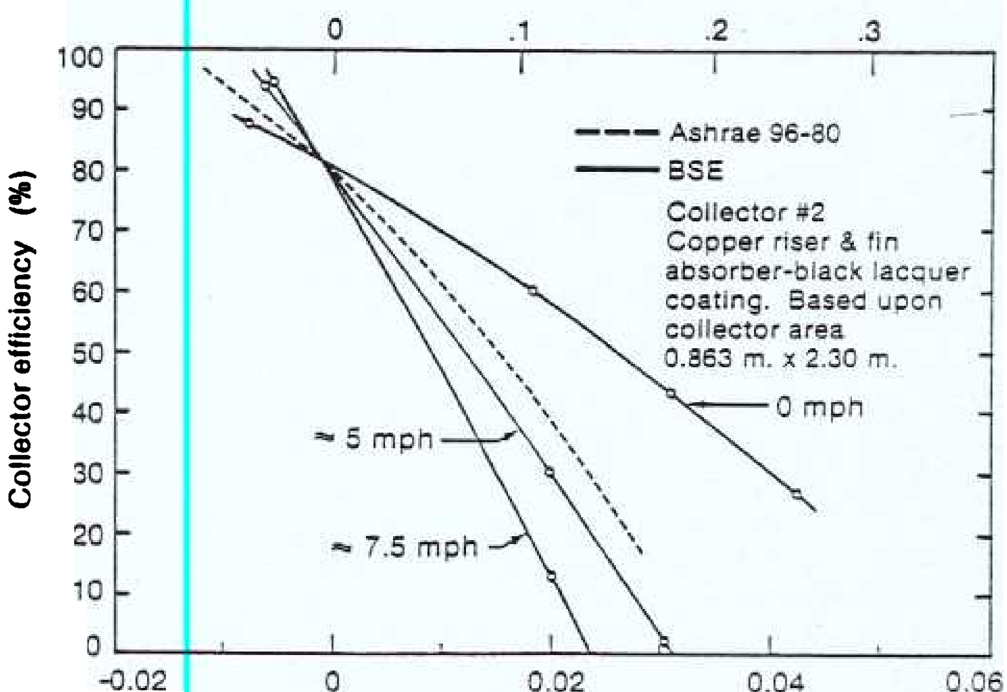

Figure B.5

Performance Curves: Unglazed Metal

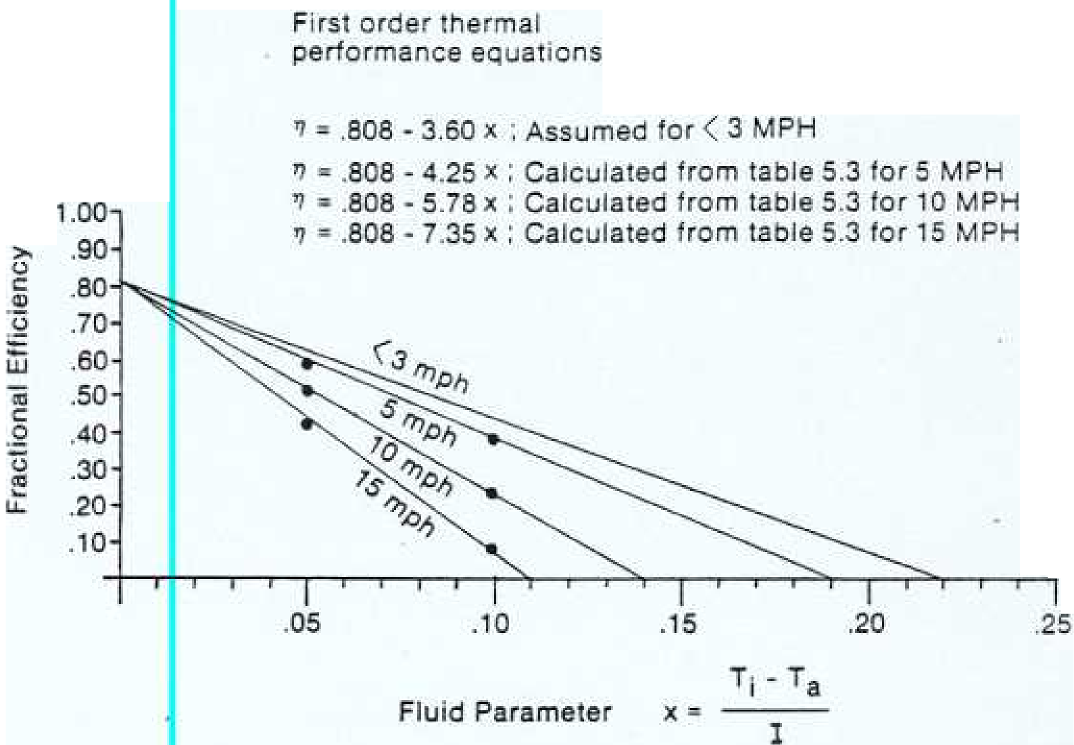

Figure 5.2

Modified Performance Graphs and Equations

(for unglazed collectors)

B.6 REFERENCES

1. Holbaugh and Huggins, Private Communication.
2. Lumsdaine, Edward, "On the Testing of Solar Collectors to Determine Thermal Performance", Proceedings of ISES Annual Conference, Denver, Co., June 1978.
3. Jenkins, John P. and Reed, Kent, A Comparison of Unglazed Flat Plate Liquid Solar Collector Thermal Performance Using ASHRAE Standard 96-1980
NBSIR 82-2522, May 1982.