NASA TECHNICAL MEMORANDUM

NASA TM X-71427

NASA TM X-71427

FLAT-PLATE COLLECTOR PERFORMANCE EVALUATION
THE CASE FOR A SOLAR SIMULATION APPROACH

by F. F. Simon and Paul Harlamert
Lewis Research Center
Cleveland, Ohio

TECHNICAL PAPER proposed for presentation at
International Solar Energy Society
Cleveland, Ohio, October 3-4, 1973

(NASA-TM-X-71427) FLAT-PLATE COLLECTOR PERFORMANCE EVALUATION. THE CASE FOR A SOLAR SIMULATION APPROACH (NASA) 21 p HC $3.25 CSCL 03B

N73-32655

Unclas

G3/29 18577

E-7670

FLAT-PLATE COLLECTOR PERFORMANCE EVALUATION

THE CASE FOR A SOLAR SIMULATION APPROACH

By

F. F. Simon and Paul Harlamert

ABSTRACT

A method is proposed for determining the performance of flat-plate solar collectors using a simulated "sun". Collector test variables that will help establish the basis for the indoor test facility at the Lewis Research Center are discussed. The use of the indoor testing should permit a standard test for the convenient and accurate determination of collector performance. Preliminary test results are reported as an example of the type of collector performance data to be expected from the simulation approach.

FLAT-PLATE COLLECTOR PERFORMANCE EVALUATION

THE CASE FOR A SOLAR SIMULATION APPROACH

By

F. F. Simon and Paul Harlamert

Lewis Research Center, Cleveland, Ohio 44135

INTRODUCTION

The NASA LeRC is investigating the possible application of solar energy as a clean, nondepleting source to help our nation's energy needs. This investigation includes the generation of electricity, the heating and cooling of buildings and the production of clean fuel (ref. 1). Approximately 25 percent of the nation's total energy is used for heating and cooling of homes (ref. 2). Solar energy captured by flat-plate collectors have been and are presently being used to heat homes. To make a significant impact on our energy needs in this area, efficient, economical, reliable solar collectors must be developed.

The experimental determination of collector efficiency is an important requirement in the evaluation of a collector design. Much of the past collector performance data are of limited value because the data are applicable only for a given location under a given weather condition. This limitation does not permit a standardized comparison between various collector designs. One approach to get around this difficulty is the simultaneous testing of different collector designs (ref. 3). This method, however, has practical limitations. Another approach is to use the "standard test" suggested by Whillier (ref. 4) in which the data collected are used to calculate performance parameters. This approach was used by Gupta and Garg (ref. 5) for the comparison of various collector designs. Because of the variation of such parameters as wind speed, air temperature, etc., the performance data of Gupta and Garg show much scatter: What is needed for an evaluation and comparison of collector performance is a standard method under controlled environmental conditions. This paper attempts to demonstrate that the requirements for standard collector performance tests and the determination of key performance parameters are best satisfied by a solar simulation approach. After a discussion of the collector testing requirements a description is given of the facility presently being built up at the Lewis Research Center for the indoor testing of collectors under simulated conditions. Preliminary tests on a prototype facility system are reported to help validate the solar simulation approach.

COLLECTOR TESTING REQUIREMENTS

A discussion of testing requirements is a useful exercise in helping to determine the variables that require controls. In deciding on the testing approach, two goals of the testing program need to be kept in mind. It is necessary to have standard conditions that will permit a comparison of different collectors, and it is required that collector performance parameters be determined that will allow a prediction of collector performance under other than standard conditions. This discussion will use a collector model of the traditional design shown in figure 1. This collector design has channels for the heat exchange medium that collects the energy absorbed on the plate, one or more windows to allow the entry of solar radiation and reduce convective and radiation losses and insulation to reduce rear and side conduction losses.

Collector Performance Variables

The basic equation relating collector performance to the absorber plate temperature is as follows:

$$\eta = \alpha\tau_e - U_L(\bar{T}_p - T_a)/q_T \tag{1}$$

Calculation of the collector efficiency (η) by equation 1 requires a knowledge of the product of absorptivity (α) and transmittance (τ_e), the overall heat loss coefficient (U_L), the plate and ambient temperatures (T_p, T_a) and the amount of solar radiation (q_T). In the practical determination of collector performance, it is more convenient to measure the temperature of the fluid as it enters and leaves the collector rather than measuring the plate temperature. Therefore, equation 1 is neither practical or convenient for the determination of collector efficiency. However, as is shown later, if an experimental determination of the product of absorptivity and transmittance ($\alpha\tau_e$) is required, then equation 1 is needed. Whillier (ref. 6) expressed the collector efficiency in terms of the average fluid temperature and a plate efficiency factor (F') as follows:

$$\eta = F'\left[\alpha\tau_e - U_L\ (\bar{T}_f - T_a)/q_T\right] \tag{2}$$

An even more convenient expression of collector efficiency is given in reference 6 in terms of the inlet fluid temperature and a factor called the heat removal efficiency factor (F_R), as follows:

$$\eta = F_R\left[\alpha\tau_e - U_L\ (T_1 - T_a)/q_T\right] \tag{3}$$

The heat removal efficiency factor is a function of flow rate (G), heat loss coefficient (U_L), plate efficiency factor (F') and fluid heat capacity (C_p).

$$F_R = f(G, U_L, F', C_p) \tag{4}$$

For a given collector design, the plate efficiency factor is expressed in terms of fluid convective coefficient (h_f) and the heat loss coefficient.

$$F' = f(h_f, U_L) \tag{5}$$

The product of absorptivity and transmittance is a function of the angle of incidence (θ_i) and the relative quantities of diffuse (q_{df}) and direct radiation (q_{dr}).

$$\alpha\tau_e = f(\theta_i, q_{df}, q_{dr}) \tag{6}$$

For a given collector, the overall heat loss coefficient is a function of the absorber plate temperature (T_p), the ambient temperature (T_a) and the exterior convective coefficient or what is commonly called the wind coefficient (h_o).

$$U_L = f(T_p, T_a, h_o) \tag{7}$$

For an accurate evaluation of collector heat loss, the ambient temperature requires a humidity corrective to determine the "effective" sky temperature (T_s). The effective sky temperature is different for a clear day than for a cloudy day. Equation 7 is more correctly stated as:

$$U_L = f(T_p, T_s, h_o, T_a) \tag{7a}$$

The amount and spectral quality of solar radiation is a function of such things as air mass, cloud cover, air pollution, water vapor, dust and the molecular components of the atmosphere. All the above functional relationships indicate the large number of variables which can affect the performance of a given collector design. In summary, the collection efficiency of a given collector design is a function of the following:

1. fluid flow rate
2. ambient temperature
3. plate temperature
4. inlet fluid temperature
5. wind speed
6. amount and spectral quality of solar radiation
7. incident angle of solar radiation
8. relative quantities of diffuse and direct radiation

Consideration of the above variables gives the basis for a standard testing approach. For a method that can be used for comparing different collector designs, we need to maintain constant values of the flow rate, ambient temperature and wind speed. With these standard conditions we can vary the inlet fluid temperature and the amount of thermal radiation so as to generate standard performance curves. Such performance curves allows us to compare collector performance for a given inlet temperature and value of thermal radiation. The standard method should employ a thermal radiation source which is equivalent to the spectral output of air mass 2. Air mass is approximately equal to sec θ_z , where θ_z is the zenith angle of the Sun. Therefore, a normal path of the Sun through the atmosphere is air mass 1. Benning (ref. 7) has suggested that the "average" sunlight for the North American continent is best represented by air mass 2 spectra. The standard radiation source should give a fixed ratio of direct to diffuse radiation (or all direct) and permit a control of the incident angle of radiation. A universal agreement on the above considerations for standard testing conditions would allow, in addition to comparative testing at one site, a basis for the comparison of collector test results made in different parts of the world. In addition, since the solar energy work at the LeRC will also involve long-range outdoor testing of collectors, it is important to have a testing standard that can be used for judging the performance life of a collector. This can be determined from indoor tests prior to and after the outdoor tests.

Collector Performance Parameters for Standard Tests

With the standard test method described above and a well instrumented collector, it is possible to experimentally determine the collector performance parameters of equations 1, 2 and 3. From the slopes and intercepts of the plots of collection efficiency versus the ratio of temperature difference to heat flux, the following parameters may be determined:

1. $\alpha\tau_e$

2. F'

3. U_L

4. F_R

These parameters are "indices of performance" of a collector and except for the heat loss coefficient (U_L) should be made as high as possible. These indices become important in the comparison of different collector designs as discussed by Whillier (ref. 6). Since the collector plate efficiency factor (F') is a measure of how collector geometry effects performance, we can use this factor for comparing designs if pumping costs are not a consideration. Whillier (ref. 6) states that the best collector design is one which has the lowest value of the quotient dollars/(square feet-F'). Because the heat-removal efficiency factor is related to flow rate, as well as to collector geometry effects, it can be used for comparing collector designs when pumping costs are a consideration. Whillier (ref. 6) states if flow rates are used which give the same pumping costs for all collectors then the best collector design is the one that has the lowest value of the quotient dollars/(square feet-F_R). In addition to the collector design considerations of Whillier, if a collector is to perform well at high temperatures, then it is necessary that it have low heat loss (low U_L) and as high a value of transmittance (τ_e) and absorptivity (α) as possible. Since these "indices of performance" will be determined for the standard conditions mentioned above, it should be possible to conveniently select the best collector designs for a given system.

Another important collector performance parameter is the collector heat capacity which is used for evaluating collector warm-up time and the effect of clouds. The heat capacity is easily determined with a solar simulator and a well instrumented collector. The effect of cloud cover may be investigated by varying the power to the simulator so as to simulate the variation of the radiant flux caused by clouds.

Collector Performance Parameters for Outdoor Tests

The two efficiency factors (F' & F_R) determined from the standard tests are, strictly speaking, valid only for the conditions of the standard tests. However, since they have a small dependence on such variables as temperature, solar radiation and wind velocity, the parameters can be used for the prediction of collector efficiency for a wide range of environmental conditions. Since the collector heat loss coefficient is dependent on the ambient temperature (approximately 1 percent change for every 10 degree change in T_a), the question arises as to what heat loss coefficient can be expected for ambient temperatures far different than that used in the standard tests? A possible way to answer this is to separate out in the standard tests the heat loss due to radiation and convection from the overall heat loss. With a knowledge of the radiation properties of the collector it becomes possible to calculate the radiation losses and correlate the convection losses. If we take a simple single window collector as an example, where the heat loss can be expressed as follows:

$$q_L = h_c(T_p - T_g) + \frac{1}{\frac{1}{\epsilon_p} + \frac{1}{\epsilon_g} - 1} \sigma(T_p'^4 - T_g'^4) \qquad (8)$$

where

$$h_c = c\,(T_p - T_g)^n \qquad (8a)$$

$$q_L = h_o(T_g - T_a) + \epsilon_g\, \sigma\,(T_g'^4 - T_a'^4) \qquad (9)$$

To solve for the heat loss for a given wind condition, ambient temperature and plate temperature, we need to solve equations 8 and 9 for three unknowns (q_L , T_g , h_c). By using a well instrumented and well insulated collector in the standard tests, we can determine the total heat loss of equations 8 and 9; and, after making a calculation for the radiation loss of these equations, a correlation can be made of the convective heat transfer. This correlation of the convective heat transfer coefficient (equation 8a) will yield the constants c and n for a given collector design. With this information the calculation approach of equations 8 and 9 can be used for determining the heat loss and heat loss coefficient ($U_L = q_L/T_p - T_a$) at other ambient temperatures.

If the standard tests utilize a thermal radiation source with all direct radiation then the value of the product of absorptivity times transmittance ($\alpha\tau_e$) obtained from the standard tests is valid for the portion of solar radiation which is direct. The value of this product ($\alpha\tau_e$) to be used for diffuse radiation may be obtained by integrating with respect to the radiation incident angle (θ_i) the value of the product $\alpha\tau_e$ determined indoors for different angles of incident.

INDOOR FACILITY FOR COLLECTOR TESTING

The indoor test facility at the Lewis Research Center is so designed that the test requirements outlined in the previous section can be met. Figure 2 gives a view of the facility with the collector in a position to receive radiation from the simulator at a zero incidence angle. The simulator is designed (see ref. 8 for details).for air mass 2 type solar radiation. It consists of

143 tungsten-Halogen 300 watt lamps (43 kW) placed in a modular array with fresnel lenses placed at the focal distance so as to collimate the radiation. A blower at the top of the facility prevents overheating of the lamps. Because the lamps have a dichroic coating to reduce the infrared output, it is possible to get a good spectral simulation of the Sun's radiation. A calculation of the absorptivity of a selective surface under simulated conditions indicates an error of only 0.1 percent in α when compared to air mass 2 radiation. There is virtually no change in the spectral output as a result of varying the power of the lamps which can deliver radiant fluxes up to 320 BTU/hr-ft.

The simulator which is normally fifteen feet from the collector provides a collimated beam that covers an area four feet by four feet with a flux uniformity of $\pm5\%$. It was determined that a great economic advantage could be achieved if some latitude were permitted on the degree of collimation of the simulator. The collimation half angle is $4\frac{1}{2}$ degrees, which is judged to be adequate for flat-plate collector testing. For normal incident tests, the error due to imperfect collimation is small and does not become noticeable for angles of incident up to sixty degrees. Beyond an incident angle of sixty degrees, flat-plate collectors are not effective in collecting solar energy and therefore testing beyond this incident angle is not required. Controls have been provided which permit an adjustment of the radiation plane of the simulator so as to accommodate different collector tilt angles.

The collector to be tested is placed on a stand (fig. 2) which allows for adjustment of the collector tilt angle and variation of the incident angle of radiation. The base of the collector stand is on a turntable that rotates to simulate the rotation of the Earth with respect to the Sun. This allows for variation of the angle of incidence. The angle of incidence can also be varied by changing the tilt angle of the collector. This procedure has the disadvantage that the collector convective heat loss varies slightly with the angle of tilt and thus adds another performance factor for comparing different collector designs. The purpose of the fan shown blowing on the collector in figure 2 is to provide controlled convective thermal losses that would be created by a wind of constant speed. At one end of the facility large doors are available which can be opened for outdoor heat loss determinations.

For the testing of collectors utilizing water systems, a thermal storage unit is provided. The water contents can be flowed through a heat exchanger (fig. 3) for quick adjustment of the inlet temperature to the collector. Without the heat exchanger, it would be required to vary the temperature of the water in storage. This would involve long waiting times. In collectors utilizing air as a heat exchange medium, the air is sent through a heat exchanger to add thermal energy for control of the inlet temperature (fig. 4). In addition to evaluating collector efficiencies, fluid pressure drop within the collector is also evaluated to determine pumping costs.

Data from a well instrumented collector including glass temperatures, plate temperatures, inlet and outlet fluid temperatures, flow rates, pressure drop, etc., are recorded on tape with an automatic voltage digitizer system. This is done because of the large number of tests and measurements to be made. The data can be quickly and accurately reduced with a high-speed computer. Some of the key data will also be recorded with pen recorders. The thermal radiation flux will be measured by using a water-cooled pyrheliometer.

PRELIMINARY COLLECTOR TESTS

A Apparatus

Preliminary tests were run with a small version (prototype) of the solar simulator to check out the test equipment and the test approach. The collector used in the test was one foot by one foot insulated by styrofoam and with one cover glass (fig. 5). The absorber plate was made by soldering 3/8" o.d. copper tubes to a 0.03-inch copper plate with the copper tubes being connected to two 3/4-inch copper headers which were used for flow distribution. The absorber surface was formed by coating the copper plate with a carbon black-silicon dioxide paint. The collector window was 1/8-inch thick glass placed one-half inch away from the collector plate. A 3-mil chromel-alumel thermocouple was placed near the outer surface of the glass and shielded from radiation by covering the cavity in which the thermocouple lay with a small dot of white paint. Thermocouples were also placed on the collector plate and at the inlet and outlet positions of the flow channel (fig. 5). A thermocouple for measuring the ambient temperature was placed a few feet away from the collector. The overall flow loop used for the collector tests is shown in figure 6. A gear pump driven by a variable speed drive was used to control the flow of water through the collector and a rotameter used for flow determination. Water at a controlled temperature was provided by having a temperature controller vary the power to an electric heater in the water tank. The radiant energy delivered by the simulator was measured with a water-cooled pyrheliometer. The emf signals from the pyrheliometer and thermocouples were recorded on an electronic self-balancing recorder.

The solar simulator composed of twelve 300-2att lamps was placed approximately ten feet from the collector. This simulator (ref. 8) was used for checking out the design of the large simulator described in the previous section. The simulator was too small to give a uniformity of ±5 percent on a one square foot surface, but did provide ±10 percent uniformity of the radiant flux.

Procedure

All test runs were made at a constant flow rate (G) of 33 lb/hr ft^2 with the collector in a vertical position. The simulator was placed in a vertical position so that the radiation incident on the collector would have a zero incident angle. Once a controlled inlet temperature was established the power to the simulator was varied to determine collector performance at different radiant flux levels. For each adjustment of the simulator power level, a time of ten minutes was found to be sufficient to insure thermal equilibrium. Once equilibrium was determined from a visual inspection of the electronic pen recorder, a record of the following quantities was made:

1. $T_g - T_1$
2. $T_o - T_1$
3. $T_1 - T_a$
4. $T_p - T_1$
5. $T_1 - T_{ref.}$ (water-ice mixture used for reference conditions)
6. q_{dr}

The radiant flux was varied from 47 to 294 BTU/hr ft^2 and the inlet temperature varied from 73°F to 109°F. The collector efficiency was calculated with the following equation:

$$\eta = G\,C_p\,(T_o - T_1)/q_{dr} \tag{10}$$

Results and Discussion

The collector performance data taken under simulated conditions was plotted according to equations 1, 2, and 3 as shown in figures 7a, 7b and 7c. The correlations are good, especially when one considers the small size of the collector and the marginal radiation uniformity produced by the simulator. The slope of the correlating line of figure 7a represents the overall heat loss coefficient (U_L) which is constant for the range of absorber plates temperatures encountered. A constant heat loss coefficient for the range of temperatures of the present experiments is consistent with the collector heat loss results of Ostad-Hosseini (ref. 9). When radiation losses become more evident at higher plate temperatures than those of the present experiments, the overall heat loss coefficient should begin to exhibit non-linearity. The heat loss results of ref. 9 show this to occur.

According to equations 1, 2, and 3 the intercepts and slopes of figures 7a, 7b and 7c represent performance parameters. These parameters are tabulated in Table I and compared to the analytical results of Bliss (ref. 10) for the simple collector tested. The comparison between the experimental and analytical results of Table I is good. It is clear that an analytical approach is very successful for a simple collector system. Difficulty in evaluating collector performance is presented with more complex collector designs where a more sophisticated analysis is required. For more complex collector designs, tests as outlined in this paper will conveniently and accurately give the performance parameters. Gupta and Garg (ref. 5) have shown that the method of obtaining performance parameters by the correlating technique of figure 7 can be used for most types of solar collectors.

Some comments on the collector design may be made based on the performance parameters listed in Table I. The high values of the plate efficiency factor (F') and plate heat-removal efficiency (F_R) indicate a good design for efficient heat collection by the heat transfer medium (water in this case). It would appear that there is room for improvement in the transmission qualities of the glass. A very clear glass can give a value of the product of absorptivity and transmittance ($\alpha\tau_e$) of about .86. The factor which makes this collector design poor is the overall heat loss coefficient (U_L) which is unacceptably high.

The overall heat loss coefficient of Table I is higher than one would expect from a calculation of convective losses, radiation losses, edge losses and rear losses. The higher loss encountered is possibly due to the copper headers on each side of the collector plate. If we assume that all the collector heat losses (radiation and convection) are via the collector window, it is possible to make an estimate of the wind coefficient (h_o). This assumption is made on the basis that the collector is well insulated. Total heat losses are calculated by the use of equation 1, where $q_L = U_L(\overline{T}_p - T_a)$. Therefore, the total heat loss equation is as follows:

$$q_L = q_{dr}(\alpha\tau_e - \eta) \tag{11}$$

The total heat loss at the outside surface of the glass window is composed of convective heat losses and radiation losses as follows:

$$q_L = h_o(T_g - T_a) + \epsilon_g \sigma (T_g'^4 - T_a'^4) \tag{12}$$

The radiation losses were calculated to be about 10-15% of the total and were subtracted from the total heat loss to give the convective heat loss ($q_{L,c}$). The convective heat loss is plotted versus the glass temperature (assuming a constant ambient temperature) in figure 8. The slope of the correlating line gives a wind coefficient of 4.0 BTU/hr ft^2 ^{o}F. This coefficient is equivalent to a wind of ten miles an hour which appears to be reasonable. The use of the wind coefficient has more significance than a statement of air speed, because of the difficulty in simulating wind conditions in indoor tests such as these.

The wind coefficient and the conditions of flow, ambient temperature and incident angle are all required for correctly stating collector performance and for comparing the performance of different collector designs.

CONCLUSIONS

This paper has endeavored to demonstrate the advantage of indoor testing of flat-plate collectors under simulated conditions. It is believed that this type of testing allows:

1. Convenient and standard conditions of radiation, ambient temperature, flow rates and wind speeds so as to permit a definitive evaluation and comparison of the performance of different collector designs.

2. A determination of the key collector parameters that can be used for the comparative evaluation of collector designs and the prediction of the outdoor performance of collectors.

REFERENCES

1. Thomas, R. L.: The Utilization of Solar Energy to Help Meet Our Nation's Energy Needs, NASA TMX-68230, 1973.

2. Solar Energy as a National Energy Resource, NSF/NASA Solar Energy Panel, Dec. 1972.

3. Selcuk, M. K.: Flat-Plate Solar Collector Performance at High Temperatures. Solar Energy 8, No. 2, 57-62, 1964.

4. Whillier, A., and Richards, S. J.: A Standard Test for Solar Water Heaters. U. N. Conference on New Sources of Energy, Vol. 5, 111-113, Rome, 1961.

5. Gupta, G. L., and Garg, H. P.: Performance Studies on Solar Air Heaters. Solar Energy 11, No. 1, 25-31, 1967.

6. Whillier, A.: Solar Energy Collection and Its Utilization for House Heating. Sc Thesis, Dept. Mech. Engr. Dept, MIT, 1953.

7. Benning, F. N.: Simulation of Average Sea-Level Sunlight (Air Mass Two), J. of Materials, JMLSA, Vol. 3, No. 3, 571-584, 1968.

8. Yass, K. and Curtis, H. B.: A Low-Cost Air Mass 2 Solar Simulator. Presented at International Solar Energy Conf., Cleveland, Ohio, Oct. 3-4, 1973.

9. Ostad-Hosseini, A.: Solar Heat Utilization in Residential Heating Systems, Ph.D Thesis, Univ. of Penn., 1972.

10. Bliss, R. W., Jr.: The derivation of Several Plate-Efficiency Factors Useful in the Design of Flat-Plate Solar Collectors. Solar Energy 3, No. 4, 55-64, 1959.

SYMBOLS

C_p	Heat capacity, BTU/lb ^oF
F'	Collector plate efficiency factor, dimensionless
F_R	Collector plate heat-removal efficiency, dimensionless
G	Flow rate of collector fluid, lb/hr-sq ft of collector
h_c	Free convection coefficient, BTU/hr-ft^2 ^oF
h_f	Fluid convection coefficient, BTU/hr-ft^2 ^oF
h_o	Wind coefficient, BTU/hr-ft^2 ^oF
q_{df}	Incident diffuse solar radiation, BTU/hr-ft^2
q_{dr}	Incident direct solar radiation, BTU/hr-ft^2
q_T	Incident total solar radiation, BTU/hr-ft^2
q_L	Collector heat loss, BTU/hr-ft^2
T_f	Collector fluid temperature, ^oF
T_1	Fluid inlet temperature, ^oF
T_o	Fluid outlet temperature, ^oF
T_p	Collector plate temperature, ^oF
T_a	Ambient temperature, ^oF
T_g	Glass temperature, ^oF
T_s	Effective sky temperature, ^oF
U_L	Overall collector heat loss coefficient, BTU/hr-ft^2 ^oF
α	Collector surface absorptivity, dimensionless
ϵ_p	Emissivity of collector surface
ϵ_g	Emissivity of glass
η	Collector efficiency, dimensionless

θ_i Angle of incidence

τ_e Transmittance

σ Stefan-Boltzmann constant, BTU/hr-ft^2 $^oR^4$

Superscripts:

______ Average quantity

' Absolute temperature, ^oR

TABLE I

COLLECTOR PERFORMANCE PARAMETERS

Parameter	Figure	Experimental		Calculated (ref. 10)
		Intercept	Slope	
U_L	7(a)	----	1.9	---
$\alpha\tau_e$	7(a)	0.82	----	---
F'	7(b)	.92	.94	.94
F_R	7(c)	.89	.89	.91

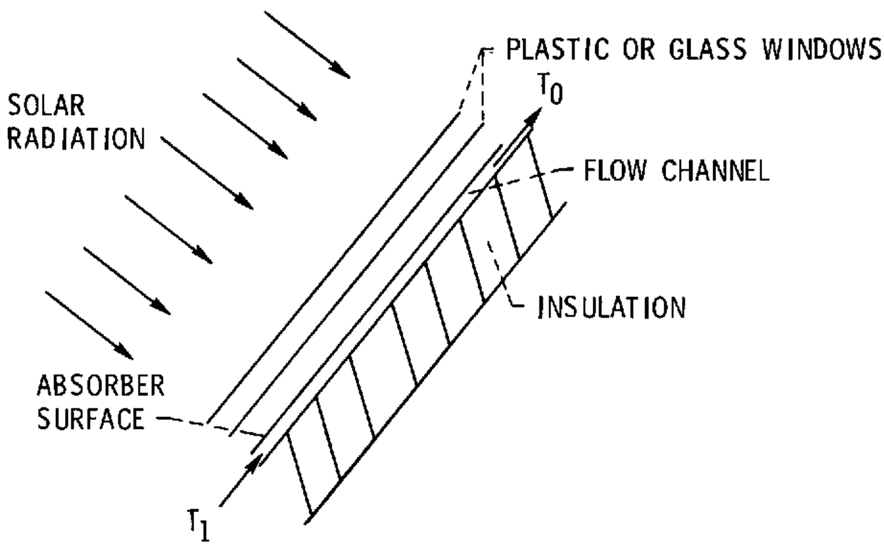


Figure 1. - Basic components of a flat plate collector.

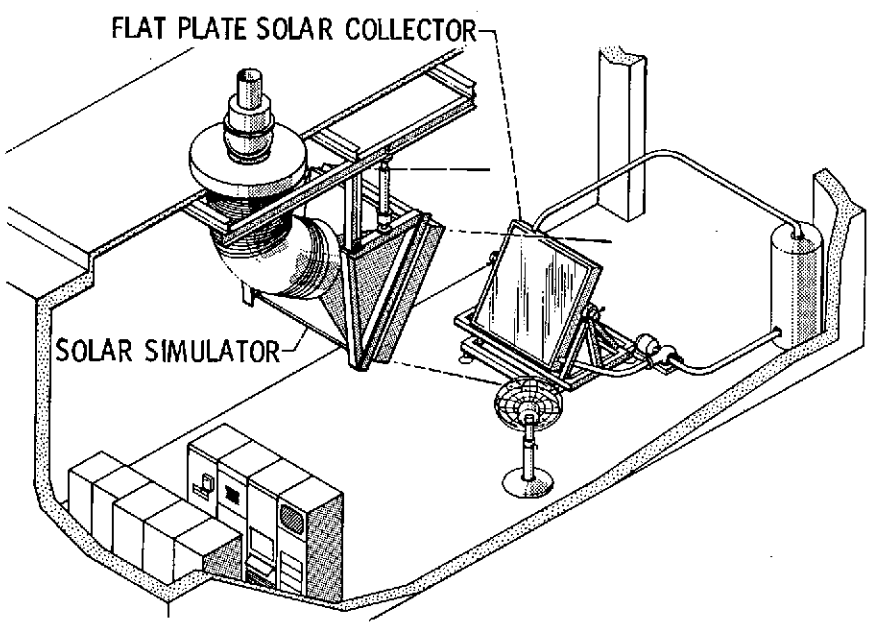


Figure 2. - Solar simulation facility for testing solar collectors.

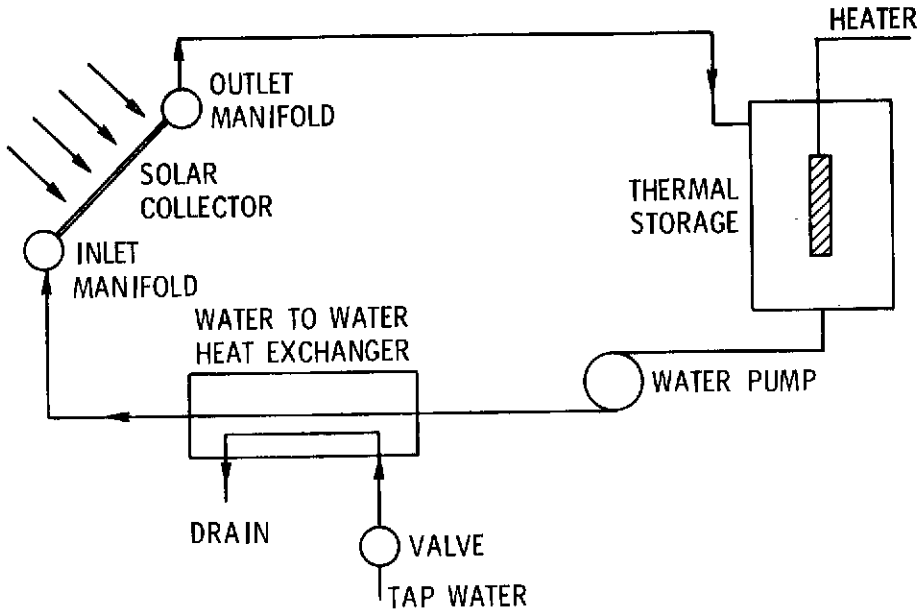


Figure 3. - Liquid flow system for flat plate collector tests.

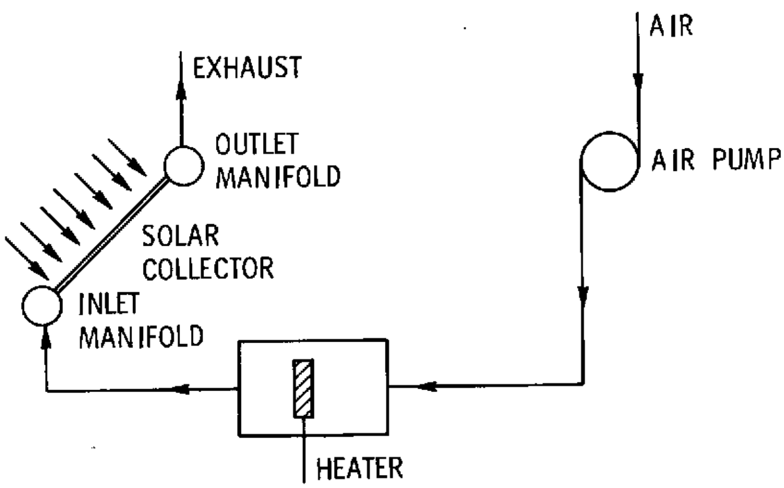


Figure 4. - Air flow system for flat plate collector tests.

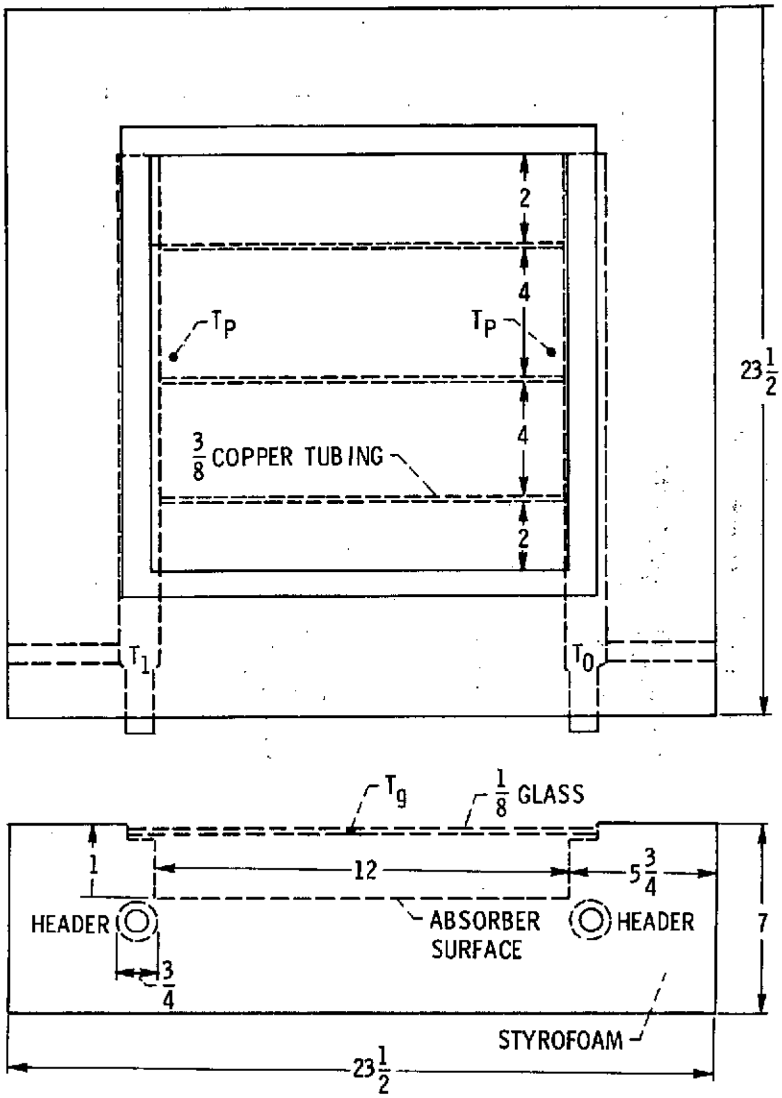


Figure 5. - Experimental flat plate collector. (Dimensions in in.)

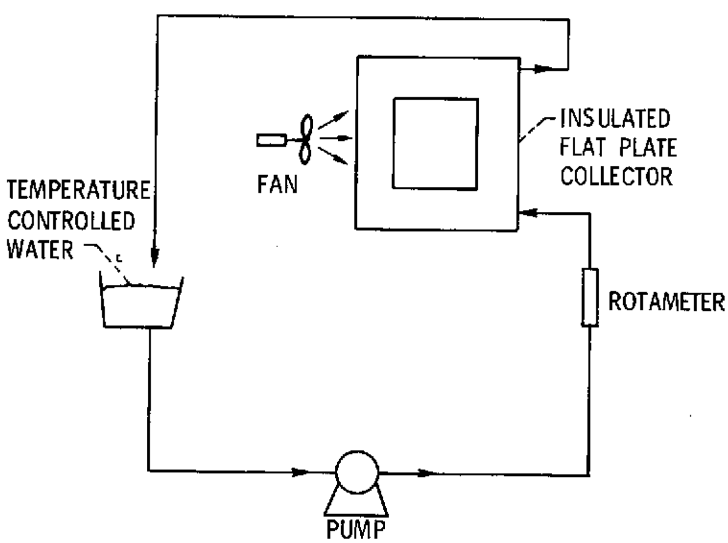
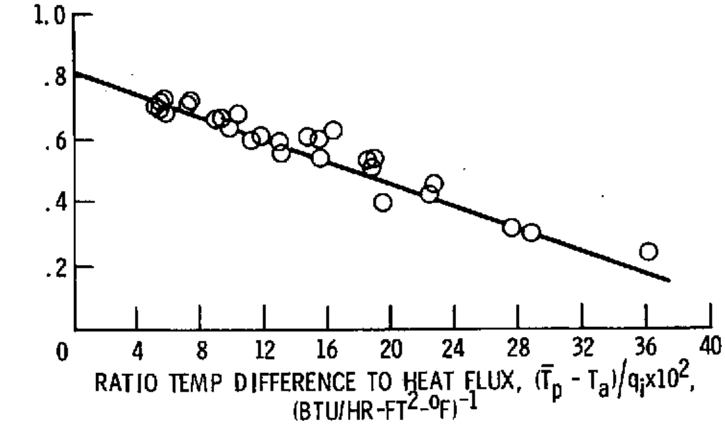
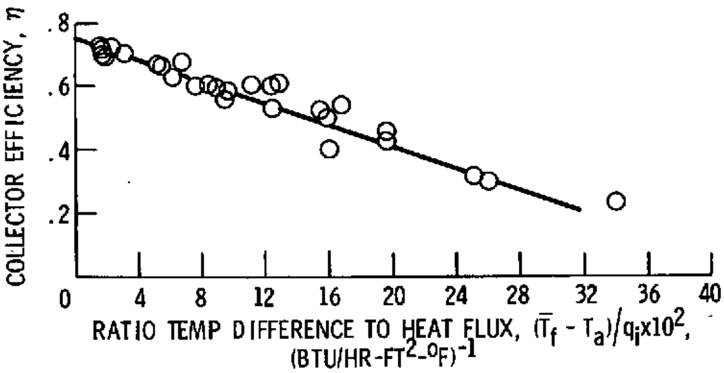


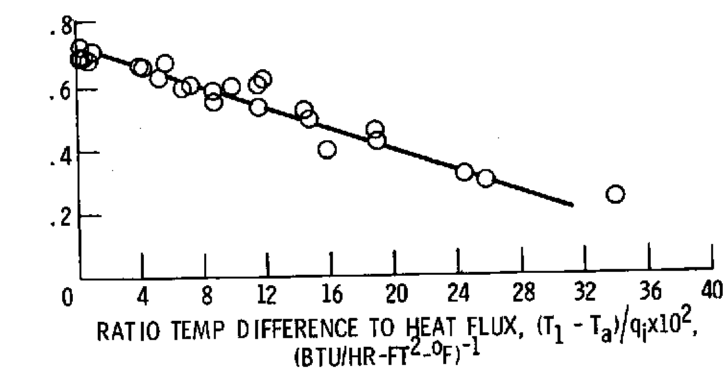
Figure 6. - Flow system - preliminary collector tests.



(A) TEMP DIFFERENCE BASED ON PLATE TEMP.



(B) TEMP DIFFERENCE BASED ON AVG FLUID TEMP.



(C) TEMP DIFFERENCE BASED ON INLET TEMP.

Figure 7. - Collector efficiency as function of the ratio of temperature difference to heat flux. T_a, 73^o F, G, 33 lb/hr-ft^2; h_0, 4.0 BTU/hr-ft^2-^{o}F.

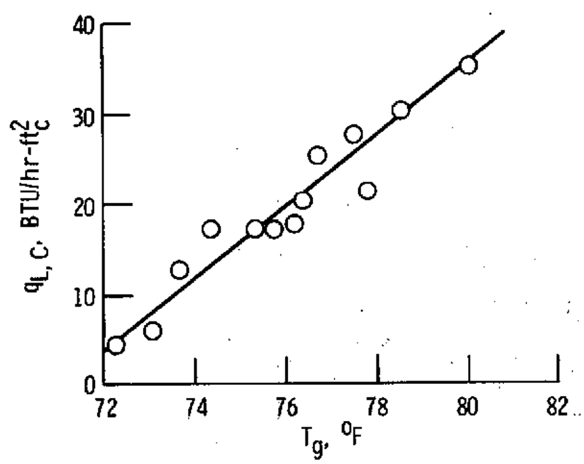


Figure 8. - Wind coefficient determination.