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Performance of a Packaged Solar Space-Heating System Used With a Mobile Home

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ABSTRACT

As part of a continuing program to develop test methods for solar heating equipment, NBS is now developing a standard test procedure for packaged solar space-heating systems similar to test procedures now used for solar collectors and thermal storage devices, and now under development for packaged solar water-heating systems. As a first step, a mobile home, which was previously tested for thermal performance in an environmental chamber, was equipped with a packaged solar space-heating system using air-heating collectors and pebble-bed storage. The system was fully instrumented and data were collected over the 1977-78 heating season at the NBS site in Gaithersburg, Maryland. The performance of the system was determined and various methods of correlating performance were explored.

Key Words: Mobile home; packaged solar space-heating system; rating; standards; testing.

TABLE OF CONTENTS

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1. INTRODUCTION AND BACKGROUND

The National Bureau of Standards (NBS), began in 1974 a program of developing test procedures for solar heating and cooling equipment, with support from the National Science Foundation (NSF), Energy Research and Development Administration (ERDA), and later the Department of Energy (DoE). The objective of the program is to encourage the adoption of consensus standards in the solar heating and cooling industry. The first test procedures developed at NBS were for solar collectors and thermal storage devices [1-5]. They were first published in early 1975 [1,2] and made available to the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE). As a direct result of this effort, ASHRAE was able to adopt standards for both components in early 1977 [6,7].

Based on the current trend of solar energy use in this country, the next logical step is to develop a standard test procedure for solar domestic water-heating systems. ASHRAE has formed a committee to develop such a standard and NBS is actively participating in its work. There are some packaged solar space-heating systems on the market. A packaged system is defined as one in which all major components (collectors, storage, controls, etc.) are contained within a single integral unit. It was felt appropriate to begin the development of a testing standard for such systems. This report describes the first steps at NBS toward the development of such a document.

During 1975 and 1976, the staff of NBS did extensive testing of a mobile home in the Bureau's large environmental chamber to completely determine its thermal characteristics. The heating demand and part load characteristics of the gas-fired, forced-air, sealed-combustion furnace system were determined for a complete range of outdoor conditions. Air leakage measurements were performed on the home using two different experimental techniques, and thermographic equipment was used to locate defects in the installation of the wall insulation. Reports are available on the complete test series [8,9].

Following the completion of that test program, it was decided to locate the mobile home outside on the Bureau grounds and retrofit it with a packaged solar space-heating system. In this way, the Bureau could begin examining the characteristics of such systems in anticipation of the development of an applicable test standard. In January, 1977, the Bureau purchased the only packaged system known to be widely available. It was shipped to Gaithersburg in February, 1977, installed next to the mobile home, instrumented, and began operating in mid-March, 1977. Initial instrumentation and testing were designed to provide only preliminary data over the last few weeks of the 1976-1977 heating season. These data were forwarded to the Department of Housing and Urban Development (HUD) in a preliminary report [10]. During the summer of 1977, the collector was isolated from the rest of the system and tested separately. More detailed instrumentation was installed over the summer and the system was operated over the entire 1977-78 heating season. The purpose of this report is to describe these latter tests, present test results, and discuss various techniques that could be used to describe the performance of such systems.

2. TESTING AND RATING PROCEDURES

Remmers and Kier have proposed a test procedure for rating residential solar space-heating systems [11]. The proposed test is intended to be implemented in the field after installation of a complete system onto a building and with a minimum of instrumentation. The test lasts six days and begins with the thermal storage subsystem fully discharged on the morning of the first day. For the first two days, the collector is allowed to operate in its normal mode and no energy is withdrawn from storage during this period. For the following three days, the collector is not allowed to operate and no energy is withdrawn from the storage device. On the sixth day, energy is withdrawn from the storage device but the collector is still not allowed to operate. By making appropriate measurements of incident solar radiation during the first two days and of temperatures around and flow rate through the various subsystems for all six days, the data are reduced and analyzed to give:

a. collection capability in Btu/($ft^2 \cdot$day) for the first two days, and

b. storage capability in percentage of collected energy delivered after three days of operation under stagnation conditions.

This test has been conducted on at least one solar space-heating system [12]. The basic drawback to this testing procedure is that the value of the collection capability is a unique function of the value of the variables controlling the system performance that occurred during the test, such as incident radiation, ambient temperature, and to a lesser extent the wind velocity. In addition, the collection is begun under the ideal situation of a fully discharged storage device. The numerical values of such a rating would only be applicable under the conditions that existed during the test.

Wolff [13] has proposed a very similar test procedure but with some notable modifications:

a. the collector is run during the night between the first and second test days in order to gather heat loss data,

b. the storage device is fully charged through the use of an auxiliary energy source, if necessary, at both the beginning and end of the three day stagnation period, and

c. considerably more detailed data are obtained than in [11] and all data are reduced to relate the efficiency of the collector array as a function of incident solar radiation and the difference in temperature between the collector outlet and ambient air, and the performance of the storage device and space-heating and air conditioning energy delivery devices as a function of the appropriate temperatures and temperature differences.

This type of testing overcomes the basic disadvantage of that recommended in [11] in that the performance of the device can now be predicted for other operating conditions. The rating for the device is obtained by using the basic equations in a computer model in conjunction with hour-by-hour weather data for a complete year and location of interest to predict the yearly energy delivered by the system. The main disadvantage of this procedure is the significant amount of modeling required once the testing is complete. In addition, it is felt that specific requirements would have to be added concerning the environmental conditions that exist during the test and/or additional test days will be necessary in order to establish relations for the performance of the subsystems which are accurate under all types of conditions.

A third possible procedure for testing and rating solar space-heating systems is to gather data on the assembled system and establish a "correlation" of the primary performance factor of interest such as the fraction of the load met by solar energy (solar fraction) as a function of operating and environmental variables. If such a correlation were successful, yearly or at least monthly average values of the operating and environmental variables could be selected and yearly or monthly solar fractions could be obtained directly from the correlation with relatively simple calculations. Klein, Beckman, and Duffie [14, 15, 16] have shown through a large number of computer simulations using TRNSYS [17, 18] that for space-heating and water-heating systems, monthly solar fraction correlates very well with two variables. One includes the monthly incident solar radiation, optical efficiency of the collector, and the monthly load. The second variable includes the monthly heat loss from the collector (with respect to a reference temperature) and the monthly load. Liu and Hill [19] investigated how a very similar correlation could be used as the basis for a standard test procedure for solar domestic water-heating systems. Balcomb and Hedstrom [20, 21] have shown through hour-by-hour simulations that solar fraction for a space-heating system correlates very well on a month-by-month basis as a function of "solar load ratio," the ratio of the monthly total solar radiation falling on the collectors to the monthly load. Because of the potentially simpler form of this correlation compared to that of Klein, Beckman and Duffie, it has been used in Section 8 of this report to analyze the data for the space-heating system which was tested.

3. DESCRIPTION OF MOBILE HOME

The mobile home was a factory-produced two-bedroom model having nominal dimensions of 15.2 m (50 ft) by 3.7 m (12 ft) with a 2.1 m (7 ft) interior height. Nominal dimensions include a 1.2 m (4 ft) hitch, making the actual length 14 m (46 ft). Floor construction was R-4 insulation over 0.16 cm (1/16 in.) asphalt-impregnated underlayment, 5.1 x 15.2 cm (2 x 6 in.) floor joists with heating ducts running the entire length, and with carpet and vinyl floor covering on top. Exterior wall sections were R-7 insulation friction-fitted between 5.1 x 10.2 cm (2 x 4 in.) framing with corrugated aluminum outside siding. The roof section consisted of R-14 insulation between framing members and over a 6-mil polyethylene film with a 1.9 cm (3/4 in.) fiberboard ceiling material and a sheet metal roof covering the top.

The mobile home was manufactured in Ephrata, Pennsylvania, and transported by tractor to NBS. When it was originally placed in the NBS environmental chamber, all utilities--electricity, water, natural gas and drainage--were connected. At its location outside, only electricity was connected. After movement to the outdoor test site, supports were placed under the chassis to take weight off of the tires, for leveling, and to reduce vibration. Sheet metal skirting was also installed at the outer periphery between the bottom of the mobile home and the ground, and storm windows were installed, as was done in the environmental chamber. The home was factory-furnished with refrigerator, stove, two beds, dining room table and four chairs, couch and divan, and curtains and draperies for the windows. Figure 1 shows the home as it was installed in the environmental chamber. Figure 2 shows it equipped with the solar system at the outdoor test location.

The home came supplied with a warm-air, gas-fired, sealed-combustion furnace which was rated at 16,500 W (56,500 Btu/h) input and 13,200 W (45,200 Btu/h) output at the bonnet. For determining the efficiency of the gas furnace during the indoor test program, the plenum was modified to contain electric resistance heating elements. These electric heaters were used as the auxiliary energy source in the present study. The voltage supplied to the elements was controlled by a variable transformer. The thermostat was located in the hallway between the doors to the bedroom and furnace room.

4. DESCRIPTION OF SOLAR SYSTEM

The solar system used in the present study was purchased and shipped from the factory in New York and delivered to NBS by truck. The unit was a self-contained package with collector, two fans, dampers, controller, and controller sensors housed within one A-frame container. The collector was covered with a hinged reflecting surface during transport. After placement at the site, this cover was opened and laid flat on the ground in front of the collector to reflect additional solar energy onto the collector surface. The dimensions of the unit were 4.0 x 2.4 m (13 x 8 ft) at the base with a height of 2.4 m (8 ft) at the

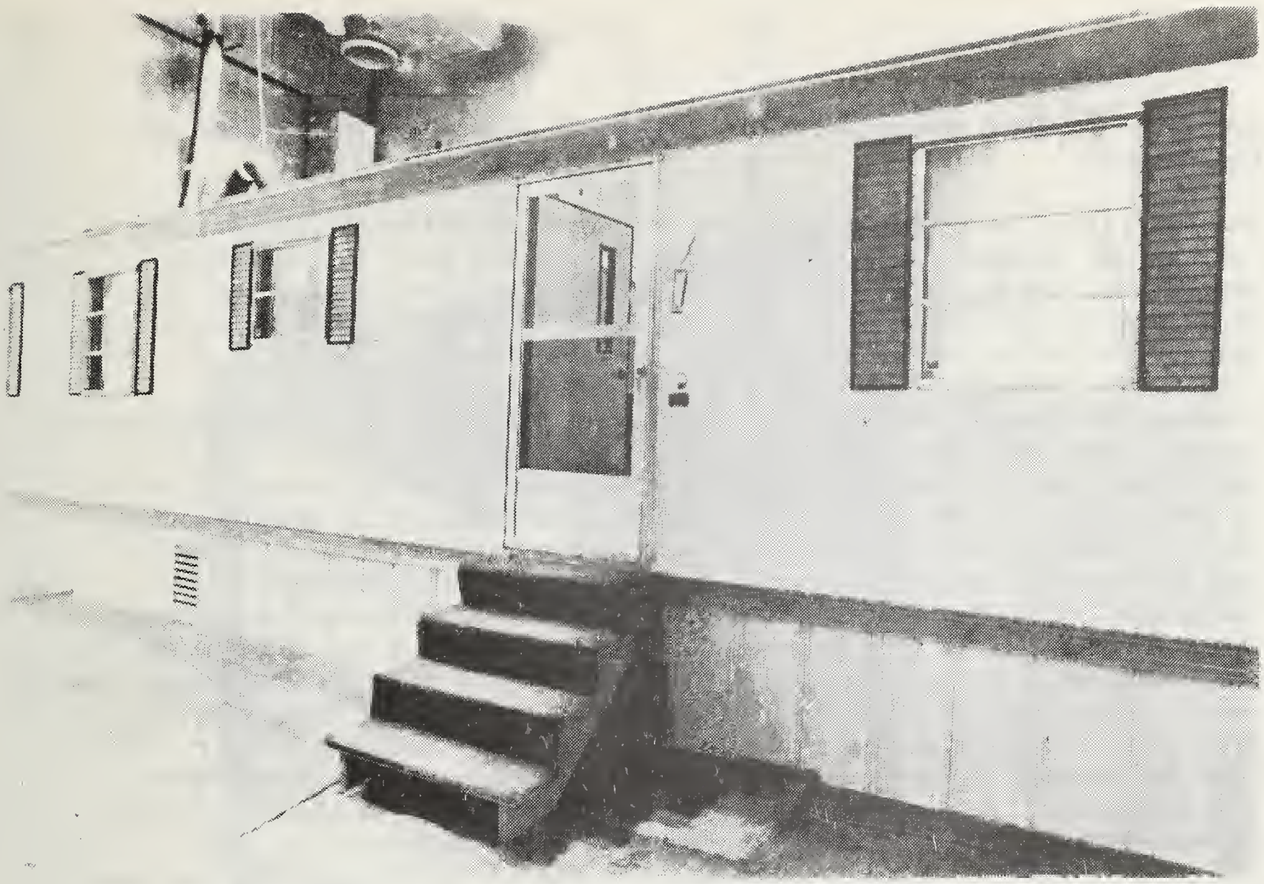

Figure 1. Mobile home in the environmental chamber.

Figure 2. Mobile home equipped with the solar system at its outdoor test site.

apex. It weighed approximately 680 kg (1500 lb) as delivered from the the factory.

The only steps necessary to put the unit into operation at the site were filling the storage bin with rock, connecting the unit to the mobile home distribution system with insulated duct, and interfacing the controls with the mobile home thermostat. All these operations were performed for NBS by the manufacturer's local contractor.

A schematic drawing of the unit is shown in Figure 3 with a cross-section looking down from above in Figure 4. It had basically four modes of operation, one for charging the pebble-bed and three for supplying heat to the house. In the charging mode, when the temperature sensor mounted on the collector absorber surface indicated a temperature 5.6°C (10°F) greater than the temperature in the "hot" side of the storage bin, the collector fan turned on and circulated air from the storage bin through the collector and back to the rocks. This continued until the temperature difference dropped below 2.8°C (5°F). In the first of the three supply modes, when the mobile home thermostat called for heat, the distribution blower in the solar unit turned on to deliver hot air from the storage bin to the home. If after 15 minutes the home thermostat was not satisfied, the auxiliary electric heater in the home turned on and space-heating was accomplished by a combination of the solar system and auxiliary energy sources. The second supply mode was designed to deliver hot air directly from the collector to the home when the pebble-bed charging mode occurred simultaneously with having to supply heat to the home. The third mode occurred when the "hot" side storage temperature dropped below 24°C (75°F). In this case, the solar system delivery blower turned off altogether and space-heating was accomplished solely by the auxiliary electric heaters and blower.

The solar collector used is shown schematically in Figure 5. It was basically a double-glazed air-heater in which the air passed over the absorber and under the inner glazing. The absorber was a finned metal plate coated with a non-selective flat-black paint. A horizontal aluminum reflector was located in front of the unit, as already mentioned, so as to reflect additional solar radiation onto the collector. The solar collector had a nominal area of 8.9 m^2 (92 ft^2) and an aperture area of 8.2 m^2 (88 ft^2). The reflector had an area of 8.9 m^2 (96 ft^2).

The pebble-bed contained approximately 11,300 kg (12.5 tons) of crushed stone with a nominal diameter of 5 cm (2 in.). The stone was crushed bluestone obtained at a local quarry and installed by the manufacturer's local distributor. The stone was not washed prior to installation in the unit. The duct work connecting the solar system to the mobile home consisted of two straight runs of approximately 3.1 m (10 ft) each of urethane foam square ducting with aluminum foil on the inside. The duct cross-sectional dimensions were 33 x 33 cm o.d. (13 x 13 in.) and 20 x 20 cm i.d. (8 x 8 in.).

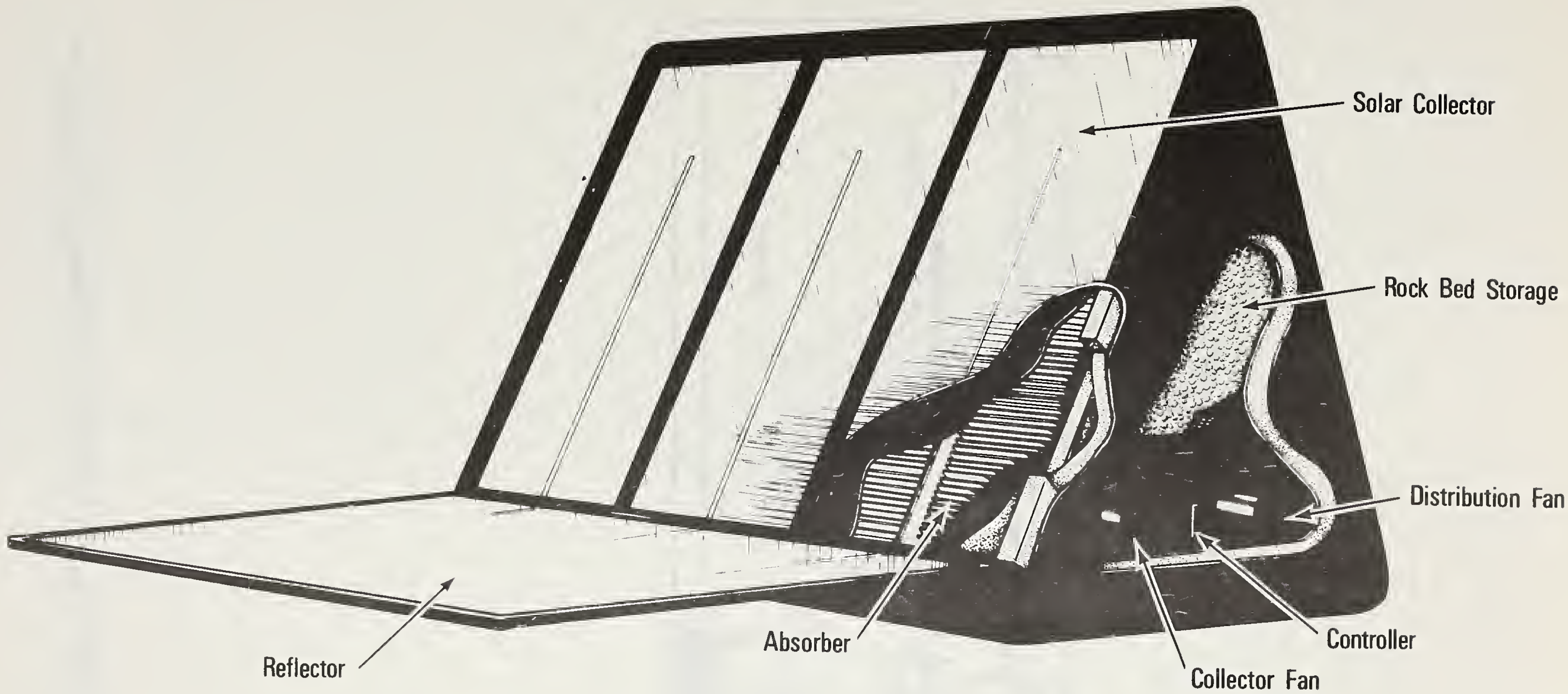

Figure 3. Schematic drawing of the packaged solar system.

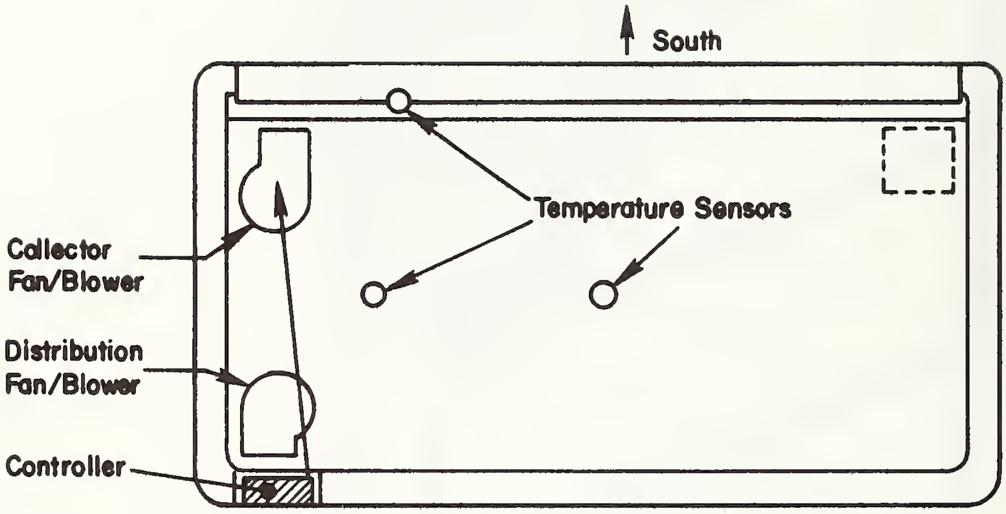

Figure 4. Schematic top view of the packaged solar system.

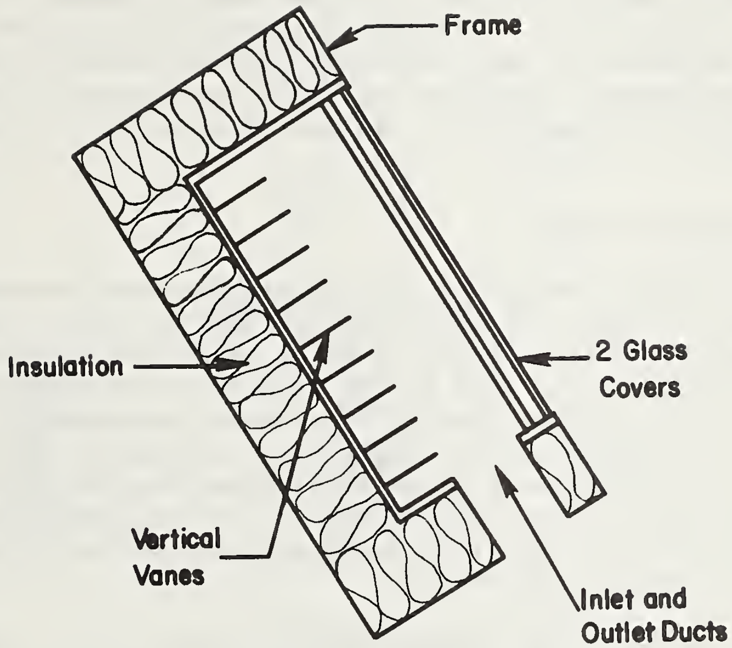

Figure 5. Schematic cross-section of the solar collector in the solar system.

5. DESCRIPTION OF INSTRUMENTATION AND TESTS

An energy flow diagram for the solar system and mobile home is shown in Figure 6. The testing of the system involved determining the energy flows across the control volume indicated, in addition to determining some of the internal energy flows. The system was instrumented using two different levels of sophistication. On one hand, a bare minimum of instrumentation necessary to determine the fraction of the load supplied by solar energy was installed and used throughout the period March, 1977 to April, 1978. In addition, detailed instrumentation was installed to determine as many important system parameters as possible so that the performance of the system could be examined in detail. The detailed level of instrumentation was used for the period December, 1977 to April, 1978.

Simplified Instrumentation

The heat loss characteristics of the mobile home were determined in previous tests in the large NBS environmental chamber over a complete range of outdoor conditions [8]. Based on these tests, equations were developed relating heating load to indoor-outdoor temperature difference. Figure 7 is a plot of total energy input under steady-state conditions as a function of inside-outside temperature difference. The heating requirements of the home were found to be approximately 9.5 MJ/(°C·day) (5000 Btu/(°F·day)) at zero wind speed and 11.4 MJ/°C·day (6000 Btu/°F·day)) at 5.8 m/s (13 mi/h) which existed on the average for the time period covered by this test.

The original furnace in the mobile home was a gas-fired, forced-air, sealed-combustion unit. As already mentioned, the plenum was modified during the tests in the environmental chamber to house electric resistance heaters in order to simplify energy measurements. These resistance heaters continued to be used as the auxiliary heaters during this solar test program, and calibrated watt-hour meters provided accurate and reliable measurements of auxiliary energy used.

The three fans in the solar system were each connected to a calibrated watt-hour meter to obtain fan operating energy requirements. Total electric energy to the home and solar system was also measured by another calibrated watt-hour meter. The internal heat generated from lights, relays, and test equipment was determined by difference.

Air temperatures were measured using laboratory-fabricated type-T thermocouples. The incident solar radiation on the 60° tilted surface (collector tilt) was measured using a precision spectral pyranometer which was located several meters from the solar system to avoid measuring reflected radiation from the collector reflector.

Through use of the watt-hour meters, monitoring of indoor and outdoor air temperature, and measurement of the incident solar radiation on the tilted surface (plus the previously-determined correlation relating

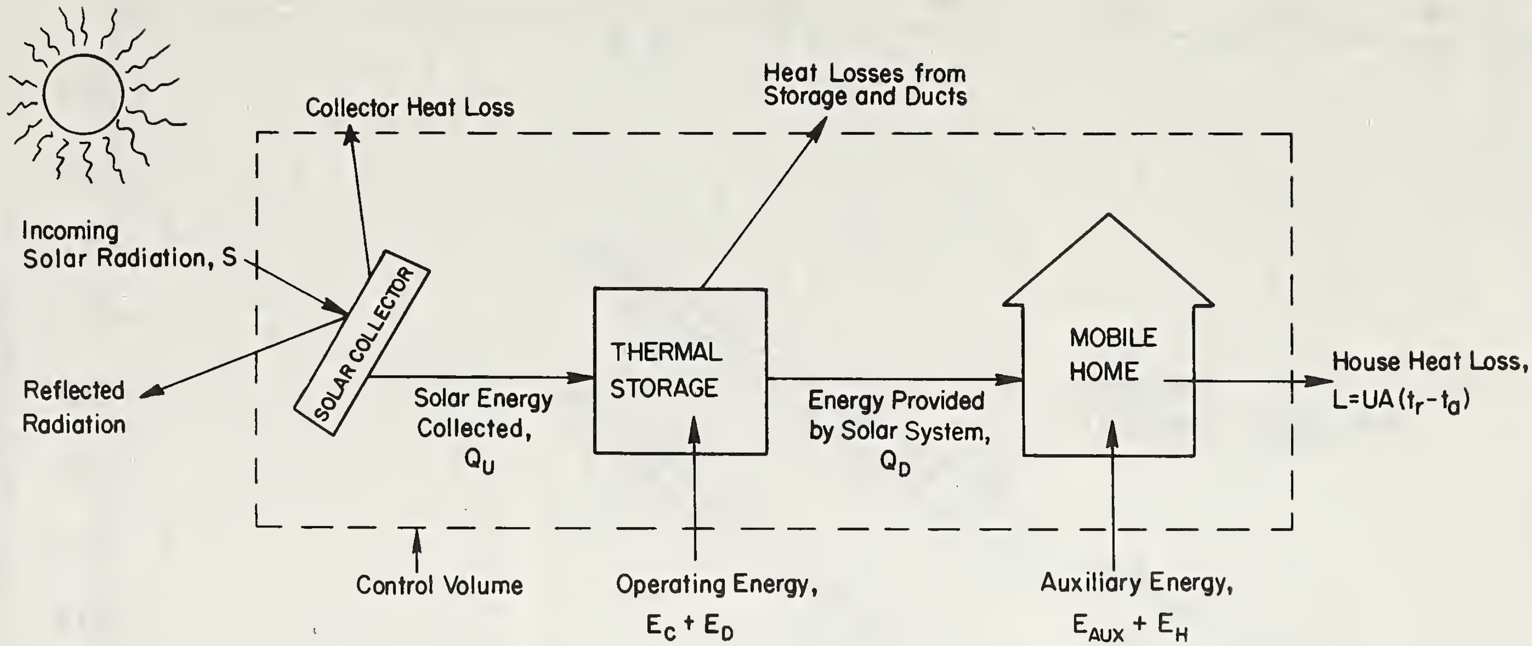

Figure 6. Schematic representation of the energy balance occurring on the mobile home equipped with the solar system.

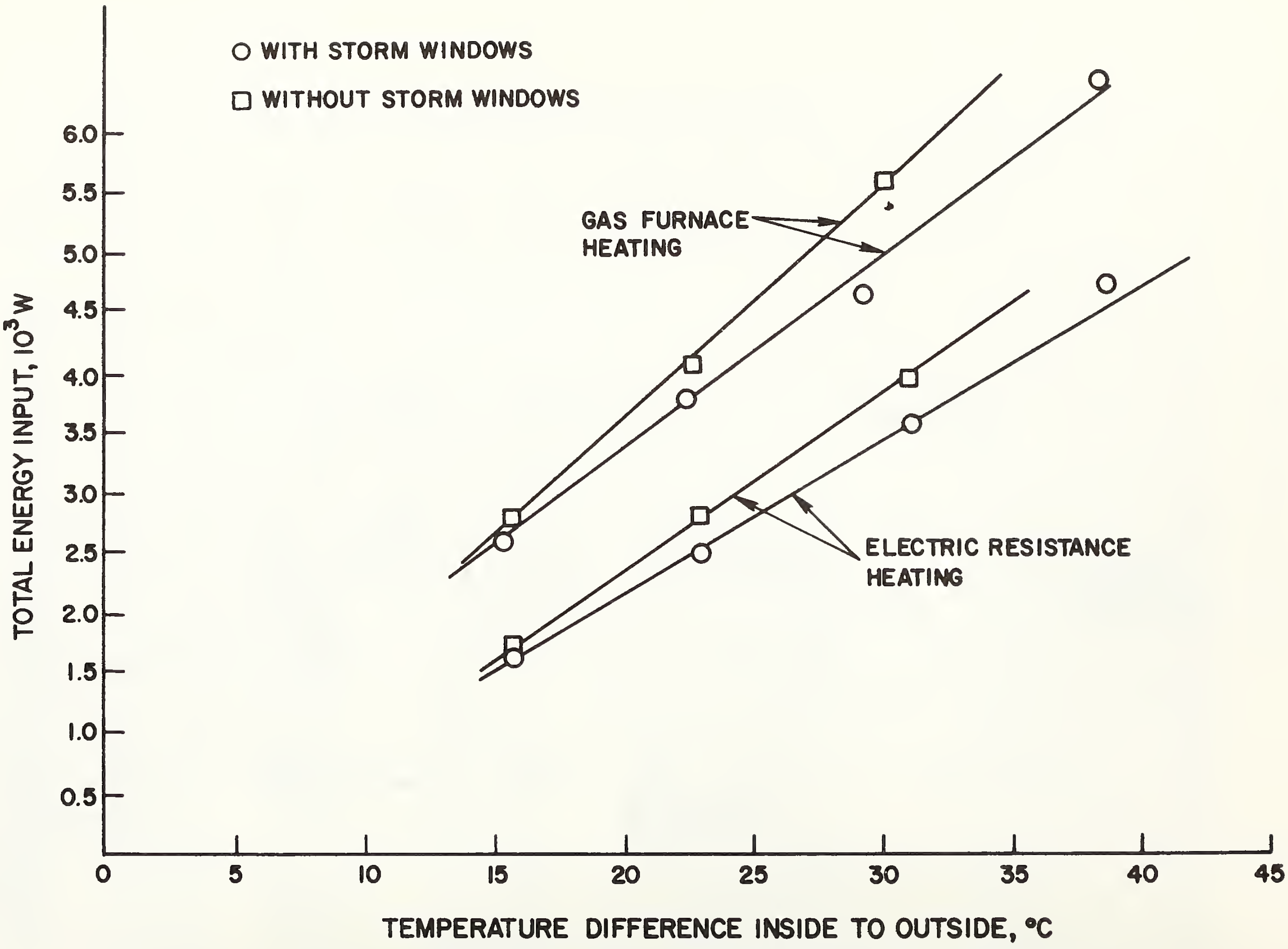

Figure 7. Space heating energy consumption characteristics for the mobile home determined in the environmental chamber.

heating load to indoor-outdoor temperature difference), it was possible to determine the solar fraction.

Detailed Instrumentation

Table 1 is a summary of the detailed instrumentation used, and the associated sensor locations can be seen in Figure 8.

Prior to beginning testing using the detailed instrumentation, the collector thermal characteristics were determined by isolating the collector from the rest of the system and performing collector efficiency tests similar to those described in ASHRAE Standard 93-77 [6]. The reflector in front of the collector complicated the determination of collector performance characteristics, since it produced a non-uniform radiation field on the collector surface which was very difficult to measure. To eliminate this complication, the reflector was covered with a flat black material during the collector tests.

The collector test loop configuration was similar to that shown in the schematic diagram of Figure 9. Type-T thermocouples were used to measure temperatures at the collector inlet and exit. A six-junction type-T thermopile was used for measuring differential temperature across the collector. The static pressure drop across the collector was measured using a slant-gage manometer. The fan was located so that it "pulled" air through the collector array, creating a slight negative gage pressure on the collector. Electric resistance heaters were used to heat ambient air to the desired collector inlet temperature. The primary flow measuring device was downstream of the collector array and consisted of a receiving chamber, discharge chamber, and a 7.6 cm (3 in.) A.S.M.E. nozzle. A commercial diaphragm type pressure transducer continuously monitored pressure difference across the nozzle and a slant-gage manometer was used to measure gage pressure at the nozzle throat. The resulting collector performance curve is shown in Figure 10.

One of the major problems in testing solar systems which use air as the working fluid is measuring air flow accurately within the system without disturbing system performance. Accurate air flow measurement with state-of-the-art methods requires either flow elements which result in high pressure drops in the system or "hot-wire" equipment which requires frequent calibration and careful installation. Neither of these methods is satisfactory for system testing over a long period of time. An alternative, used in this study, was to calibrate the collector as a flow-measuring element. The collector on the system being studied at NBS had a pressure drop of approximately .19 kPa (0.75 in. of water) under normal operation which allowed use of reliable, inexpensive electronic pressure-difference measuring devices. The high pressure drop also minimized the effects of local variations in atmospheric pressure on the measurements.

Following the collector tests, the collector was "calibrated" using the A.S.M.E. nozzle in series with the collector. A plot of air flow versus

Table 1. Detailed Instrumentation for the Solar-Heated Mobile Home

Measurement	Symbol	Instrument	Frequency of Measurement
Barometric pressure	P_b	barometer	daily
Watt-hour meter reading for collector fan	E_c	watt-hour meter	daily
Watt-hour meter reading for pebble-bed fan	E_D	watt-hour meter	daily
Watt-hour meter reading for mobile home fan	E_H	watt-hour meter	daily
Watt-hour meter reading for mobile home excluding electric auxiliary heaters	E_T	watt-hour meter	daily
Watt-hour meter reading for adjustable electric heat	E_{XA}	watt-hour meter	daily
Watt-hour meter reading for fixed electric heat	E_{XF}	watt-hour meter	daily
Collector fan on/off		relay	2 minutes
Delivery fan on/off		relay	2 minutes
Auxiliary heat on/off		relay	2 minutes
Irradiance on collector tilt w/o reflector	G_T	pyranometer	2 minutes
Collector temperature difference	Δt_c	thermopile	2 minutes
Duct temperature difference	Δt_D	thermopile	2 minutes
Collector fluid inlet temperature	$t_{f,i}$	thermocouple	2 minutes
Collector fluid exit temperature	$t_{f,e}$	thermocouple	2 minutes
Duct fluid temperature - hot side in mobile home	t	thermocouple	2 minutes
Duct fluid temperature - cold side in mobile home	t	thermocouple	2 minutes
Living room air temperature	t_r	thermocouple	2 minutes
Instrument room air temperature		thermocouple	2 minutes
Irradiance on horizontal surface	G_H	pyranometer	2 minutes
Pebble-bed centerline temperatures	t_s	thermocouple	2 minutes
Duct fluid temperature - cold side at solar system	t	thermocouple	2 minutes
Duct fluid temperature - hot side at solar system	t	thermocouple	2 minutes
Effective sky temperature	t_{sky}	pyrgeometer	2 minutes
Wind velocity	V_w	anemometer	2 minutes
Wind direction		resistance pot	2 minutes
Ambient temperature	t_a	thermocouple	2 minutes
Relative humidity in mobile home	rh	transducer	2 minutes
Beam irradiance	G_B	NIP	2 minutes
Irradiance on collector tilt w/o reflector, second measurement	G_{T2}	pyranometer	2 minutes
Pressure difference across collector	Δp_c	transducer	2 minutes

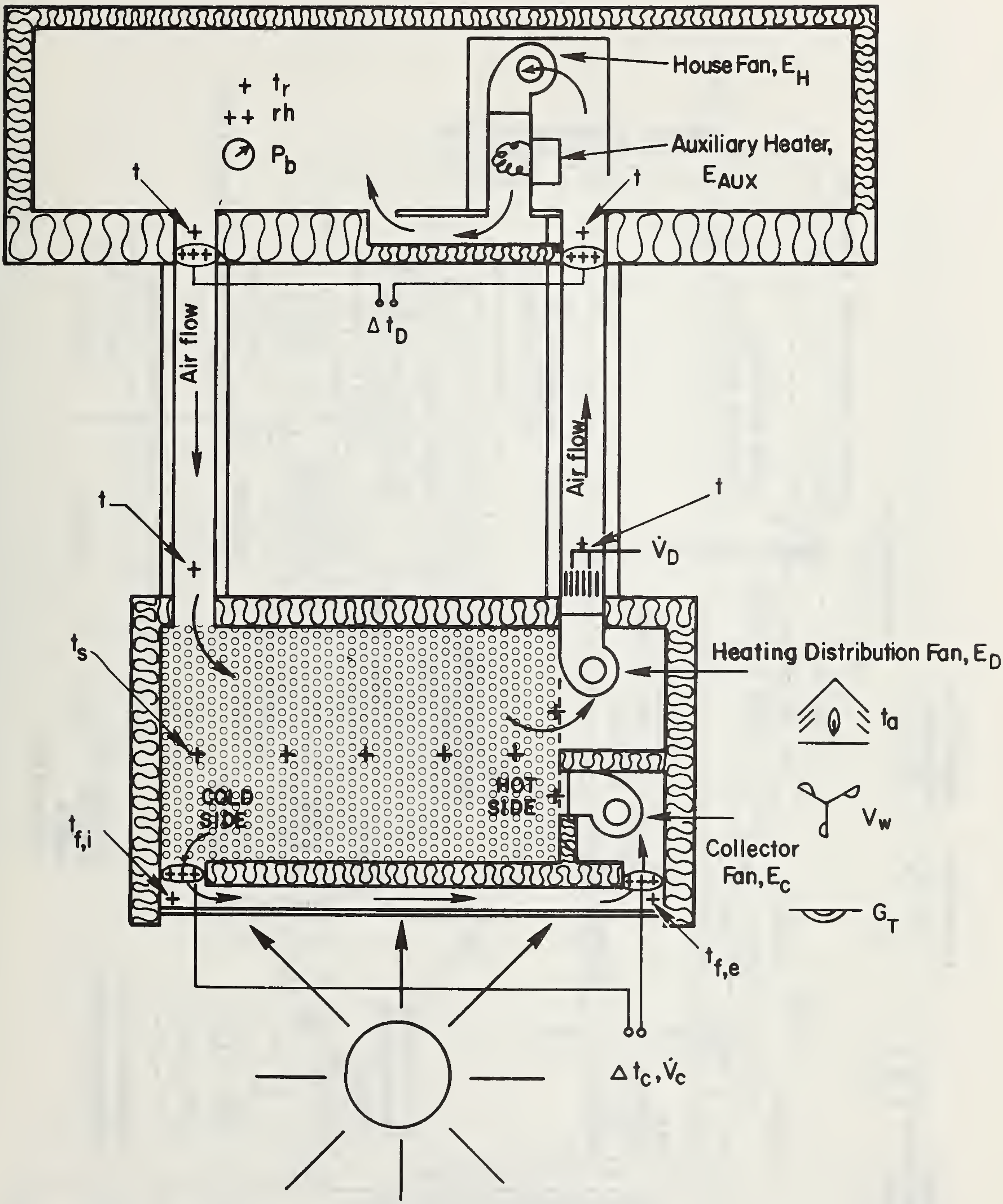

Figure 8. Instrumentation schematic of the solar system.

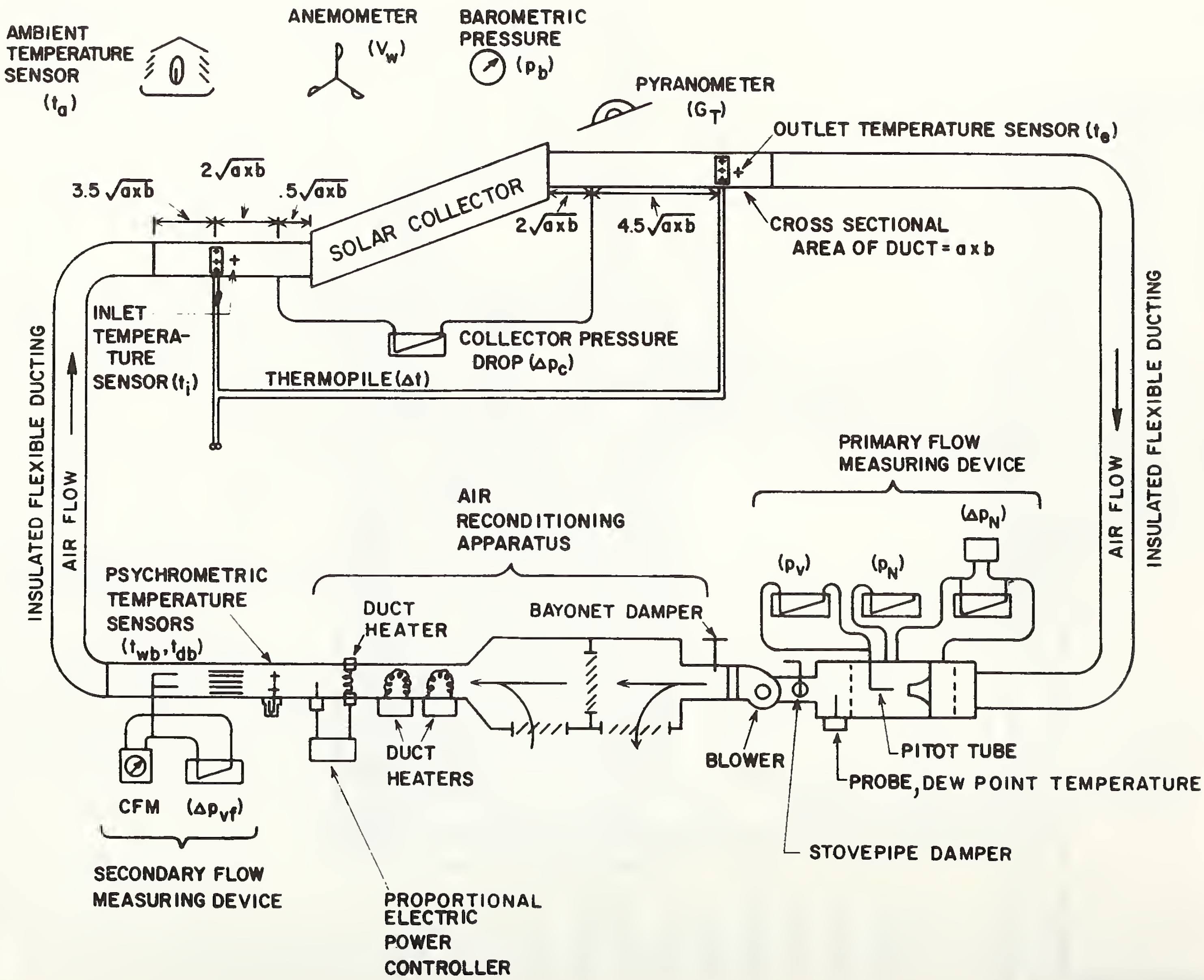

Figure 9. Collector test loop used during the separate collector test.

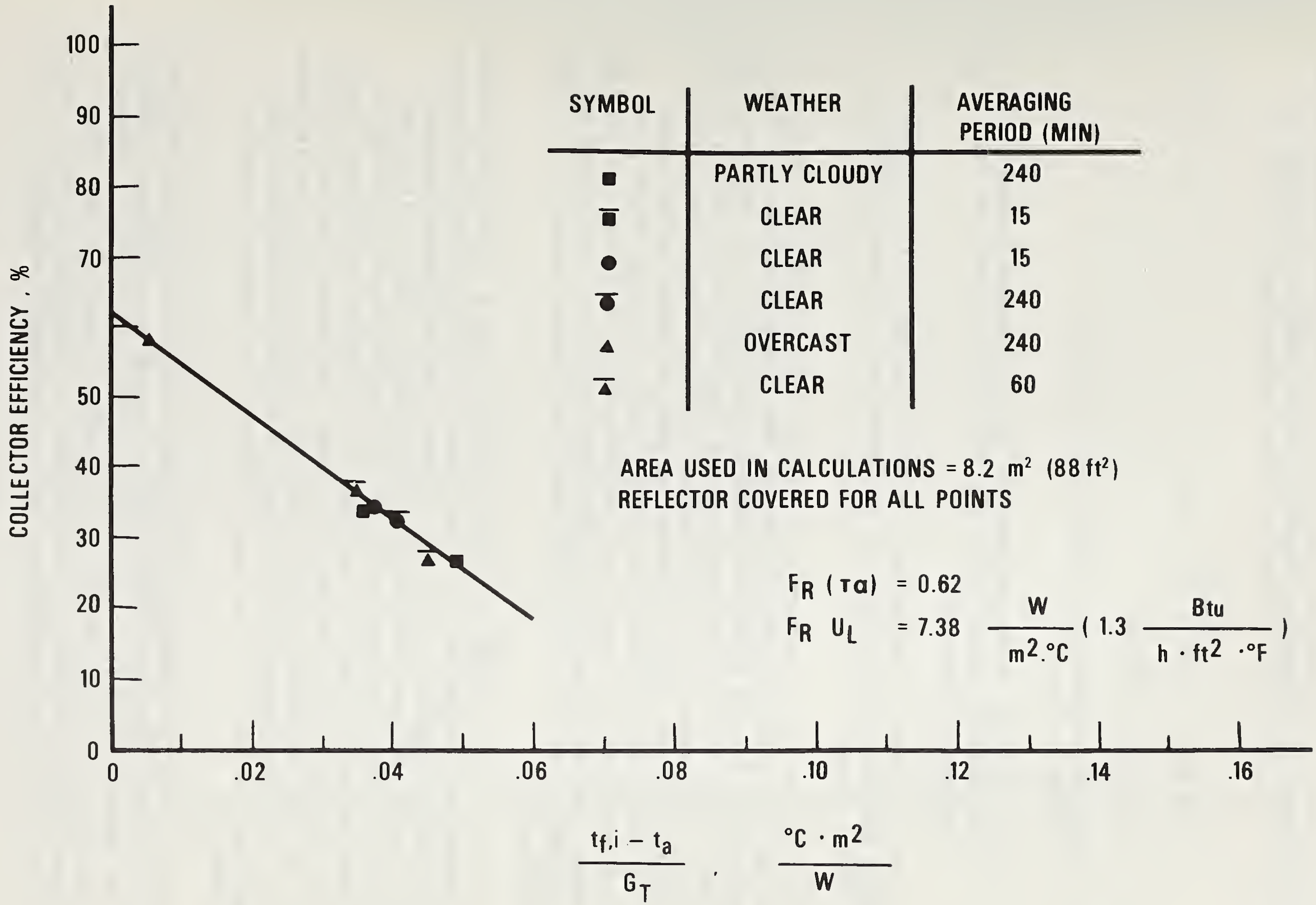

Figure 10. Solar collector efficiency curve with the reflector covered.

the square root of the pressure drop across the collector divided by air density resulted in a straight line with little scatter, as shown in Figure 11. The resulting equation for flow is therefore:

$$\dot{V} = 120.5 \sqrt{\Delta p_c/d} - 9.6 \qquad (1)$$

where $\dot{V}$ = flow rate, ft^3/min

Δp_c = static pressure drop across the collector, inches of water

d = air density in the collector, lb/ft^3 (computed using the average of the inlet and exit temperatures and the barometric pressure).

Thus it was possible to continuously measure collector flow rate by monitoring pressure drop across the collector and collector inlet and outlet temperatures.

Another major problem in determining the performance of solar heating systems using air is air leakage through the various components. The collector air flow rate which was measured and used in the performance calculations was the air flow rate exiting the collector. When the collector was "calibrated" as a flow-measuring element using the nozzle, all pressures and the flow rate were the same as those during actual operating conditions. During the 1977-78 winter heating season, the collector developed air leaks which progressively increased in magnitude. The leakage undoubtly resulted in decreased collector performance and also an underestimation of this decreased performance caused by the flow measuring method.

Air flow rate in the supply duct was measured using a commercial laminar flow element which consisted of a honeycomb air straightener upstream of a pitot tube arrangement. This flow element was calibrated using an A.S.M.E. nozzle and was found to give good results once a "discharge" correction factor was established. The correction factor, similar to the discharge coefficient in a nozzle, was a constant multiplier which corrected the indicated flow rate to actual flow rate. The correction factor for this particular element was 0.93. The low pressure differentials in the flow element (11.3 to 13.8 Pa (0.045 to 0.055 in. H_2O)) necessitated the use of a hook-gage manometer for accurate pressure measurements.

After monitoring flow measuring pressures for a period of time with the system in normal operation, equations and curves were developed relating air flow rates in both the supply duct and the collector to the respective air densities, the blowers having constant speed motors. Thus after some initial testing, it was possible to determine air flow rates knowing only duct air temperatures and the barometric pressure.

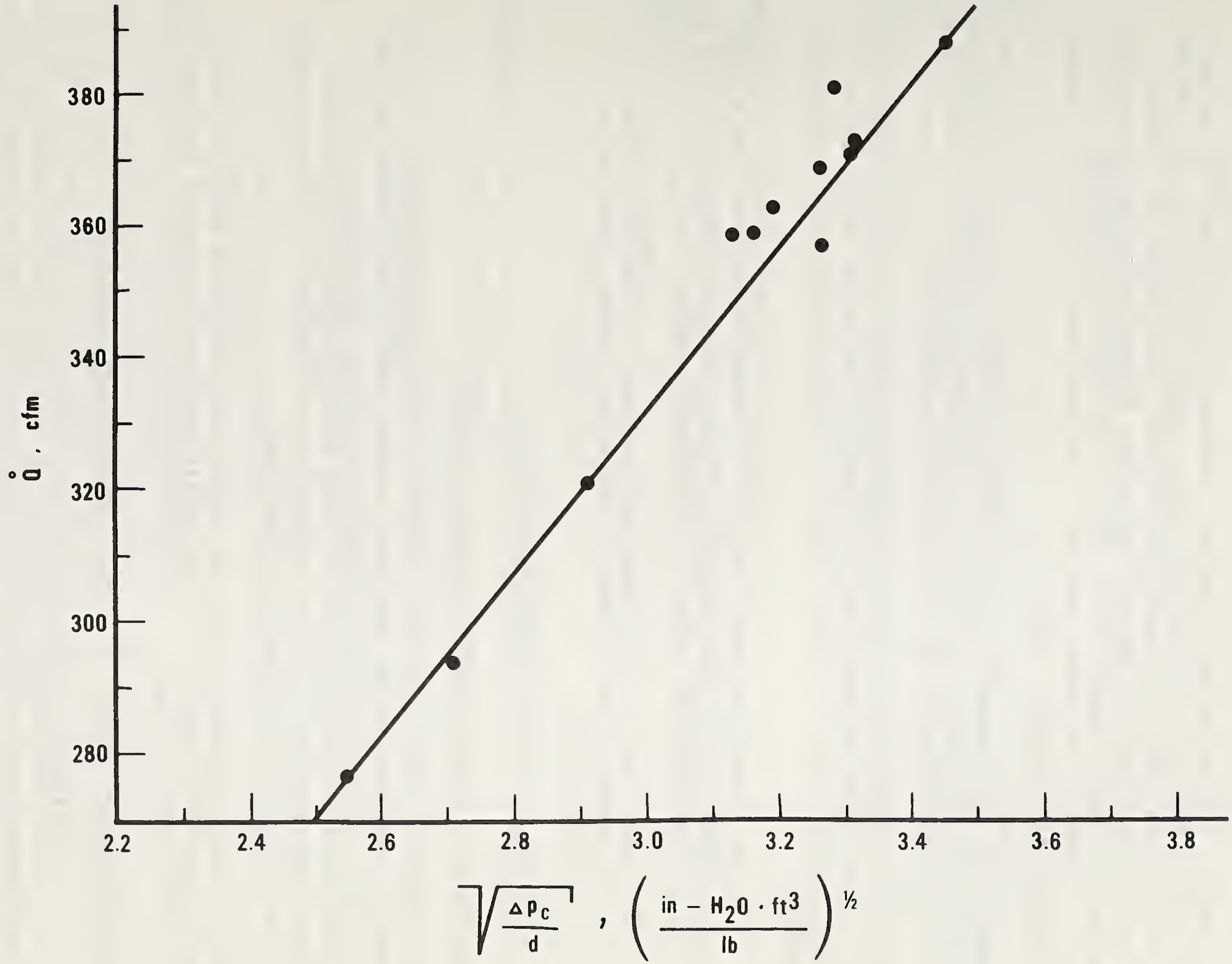

Figure 11. Solar collector air flow calibration results.

The incident solar radiation on a 60-degree tilted surface was measured as with the simplified level of instrumentation, using a pyranometer which was located several meters from the solar system.

Data were collected using a standard commercial data logger in conjunction with a reel-to-reel magnetic tape recorder. The scan rate was set at two minutes, the short auxiliary heating cycle being the critical controlling variable. The data logger had a capacity of 80 channels and handled millivolt signals (0 to 400 mv) and type-T thermocouple inputs with an internal reference junction. The data were reduced on a large-scale digital computer at NBS.

Temperatures were measured using laboratory-fabricated type-T thermocouples. In addition to the thermocouples listed in Table 1, an additional 40 thermocouples were located in the pebble-bed.

Temperature differences across the collector and across the supply-to-return ducts were measured using laboratory-fabricated six-junction type-T thermopiles. To ensure a well-mixed air stream at the measuring stations:

1) the thermopiles were located far enough downstream from a right angle turn to ensure proper mixing. Calculations showed that flow conditions were in the turbulent regime, and

2) individual thermocouples were arranged across the various cross-sections in order to detect any uneven temperature distribution.

6. DATA REDUCTION AND PERFORMANCE ANALYSIS

Data Analysis Using Simplified Instrumentation

Data resulting from using the simplified level of instrumentation consisted of watt-hour meter readings, average ambient and indoor air temperatures, incident solar radiation, and average wind speed from nearby Washington National Airport.* These data combined with the known thermal characteristics of the mobile home determined in previous tests allowed calculation of the solar fraction.

Electric power consumption of the three fans, auxiliary heaters, and the total electricity supplied to the mobile home was determined from the watt-hour meter readings.

The daily space-heating load was calculated as follows:

$$L = UA \cdot (21°C - \overline{t}_a) - E_G \qquad (2)$$

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* In a separate project at NBS conducted during the same time period [22], good agreement was found between wind speeds at the NBS site averaged over several hours and that recorded at the Airport.

where L = daily space-heating load, kJ

UA = building heat loss factor determined from the indoor tests, kJ/°C

$\bar{t}_a$ = daily average ambient temperature, °C

E_G = internal heat generated in the building from lights and instruments, kJ

After reduction of the data obtained using the detailed instrumentation, it was found that on some days the electric energy into the house far exceeded the calculated heat loss. Since the heat loss factor on which the calculated heat loss was based was determined indoors, it was assumed that outdoor wind effects caused increased air infiltration and thus additional heat loss. A correction factor was developed to take wind effects into account, as will be explained in the next section

$$L^* = L + \text{correction factor} \tag{3}$$

where L^* = corrected space-heating load, kJ

The solar fraction, f, was computed as follows, assuming that the solar contribution was the difference between the space-heating load and auxiliary energy consumed.

$$f = 1 - \frac{(E_{AUX} + E_C + E_D + E_H)}{L^*} \tag{4}$$

where E_{AUX} = daily energy consumed by the auxiliary electric heaters in the mobile home, kJ

E_C = daily energy consumed by the collector fan in the solar system, kJ

E_D = daily energy consumed by the load delivery fan in the solar system, kJ

E_H = daily energy consumed by the distribution fan in the mobile home, kJ

Data Analysis Using Detailed Instrumentation

Data from the detailed level of instrumentation were recorded on the magnetic tape. Table 1 lists data items recorded. After collection of the data, a data reduction program was used to produce both hourly and daily summaries of the input data and selected performance factors. A second program was used to calculate the performance of the system over the winter heating season and to analyze the measured performance in various ways.

The irradiation on the solar collector came from three sources: direct radiation from the sun, diffuse radiation from the sky, and reflected radiation from the reflector. As mentioned previously, the reflected radiation was difficult to characterize accurately. Thus it was not possible to measure the efficiency of the collector as an isolated unit during the heating season, although the useful solar energy collected, Q_u, was measured. A collector efficiency based on the non-reflected irradiation onto the collector aperture was calculated during this test period.

The amount of energy in thermal storage was calculated by taking the average of the pebble-bed temperatures and multiplying by the pebble-bed heat capacity. Heat losses from storage and ducts were calculated by using an energy balance around the pebble-bed. This method was not accurate, due to damper leakage that was found to occur in the system. The energy provided by the solar system to the mobile home, Q_D, was measured only when the delivery fan was operating. It was found during operation, however, that a substantial amount of hot air leaked through the dampers into the mobile home when the collector fan was operating, even though the load delivery fan was off. This leakage through the dampers also prevented any direct measurement of heating load during days of solar collection.

The heating load determination was based on the previous indoor thermal loss tests. Wind effects significantly affected the load and a wind correction factor was established using data taken during periods of no solar collection as will be explained below. Internally generated heat was considered part of the load. Auxiliary heat to the load included heat from both the furnace fan and the electric resistance heaters.

Two major correction terms, based on the data over the winter heating season, were derived and used to correct the data. The two factors were a space-heating load correction factor due to wind and a heat leakage correction factor due to air leakage through the dampers to the mobile home.

To correct the space-heating load for air infiltration due to wind, the amount of air leaking through the mobile home was assumed to be directly proportional to the average wind velocity at nearby Washington National Airport. Furthermore, it was assumed that the space-heating load correction factor was proportional to the infiltration multiplied by the inside-to-outside temperature difference. Figure 12 is a plot of the difference between measured and predicted space-heating load versus the product of average wind speed and inside-to-outside temperature difference. The following equation was developed from the curve:

$$L^* = L + 0.144 \cdot V_w \cdot (t_r - \overline{t}_a) \quad (5)$$

where V_w = daily average wind velocity, mi/h

t_r = reference temperature ≡ 70°F

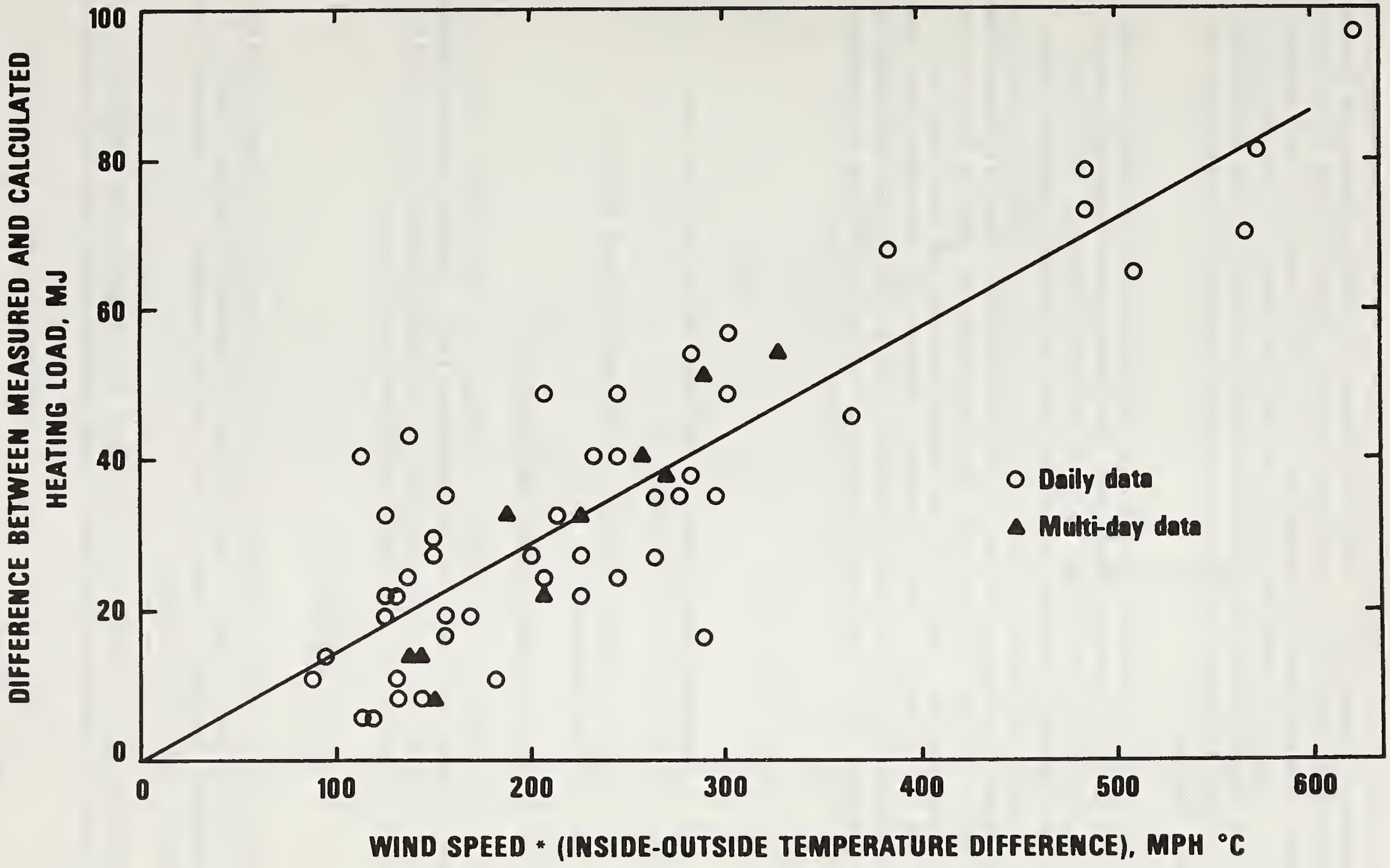

Figure 12. Effect of wind speed on heating load.

The air infiltration rate was characterized previously from tests in the environmental chamber [9]. The difference in infiltration rate between still air conditions and a simulated 3.6 m/s (8 mi/h) wind at an inside-to-outside temperature difference of 22°C (72°F) was 0.15 air changes per hour. The difference in infiltration rate due to a 3.6 m/s (8 mi/h) wind using equation (5) is 0.39 air changes per hour. The difference is due to the different conditions outside compared to inside the environmental chamber.

To correct for air leakage through the dampers from the solar system to the mobile home, it was assumed that the leakage was directly proportional to the energy imbalance around the solar system. An energy balance on the solar system indicated that the energy into the solar system from the collector array and fans exceeded energy delivered to the load plus the energy stored in the pebble-bed. The difference was assumed to be either leakage past the damper to the mobile home or heat loss to the outside. The ratio of damper leakage to heat loss was assumed to be constant. The difference between the estimated and measured space-heating load plotted against the solar system imbalance is shown in Figure 13. The resulting equation for correcting the energy delivered from the solar system to the mobile home is:

$$Q_D^* = Q_D + 0.786 \cdot \Delta E \qquad (6)$$

where

Q_D^* = corrected daily energy provided to the mobile home from the solar system, kJ

Q_D = daily energy provided to the mobile home from the solar system, kJ

ΔE = solar system energy imbalance, kJ

Several performance factors were calculated for the system. The solar fraction, f, was calculated using equation (4). A solar system fraction, f', which is defined as the fraction of the spaceheating load supplied by the solar heating system, was calculated as follows:

$$f' = \frac{L^* - \left(E_{AUX} + E_H\right)}{L^*} \qquad (7)$$

The solar system fraction represents the fraction of the load supplied by the combination of solar energy plus the energy from the fans in the solar system. The solar fraction defined by equation (4) accounts for the energy from the sun only.

A coefficient of performance, COP, which is defined as delivered solar heat divided by system solar fan energy requirements, was calculated as follows:

$$COP = \frac{Q_D^*}{E_C + E_D} \qquad (8)$$

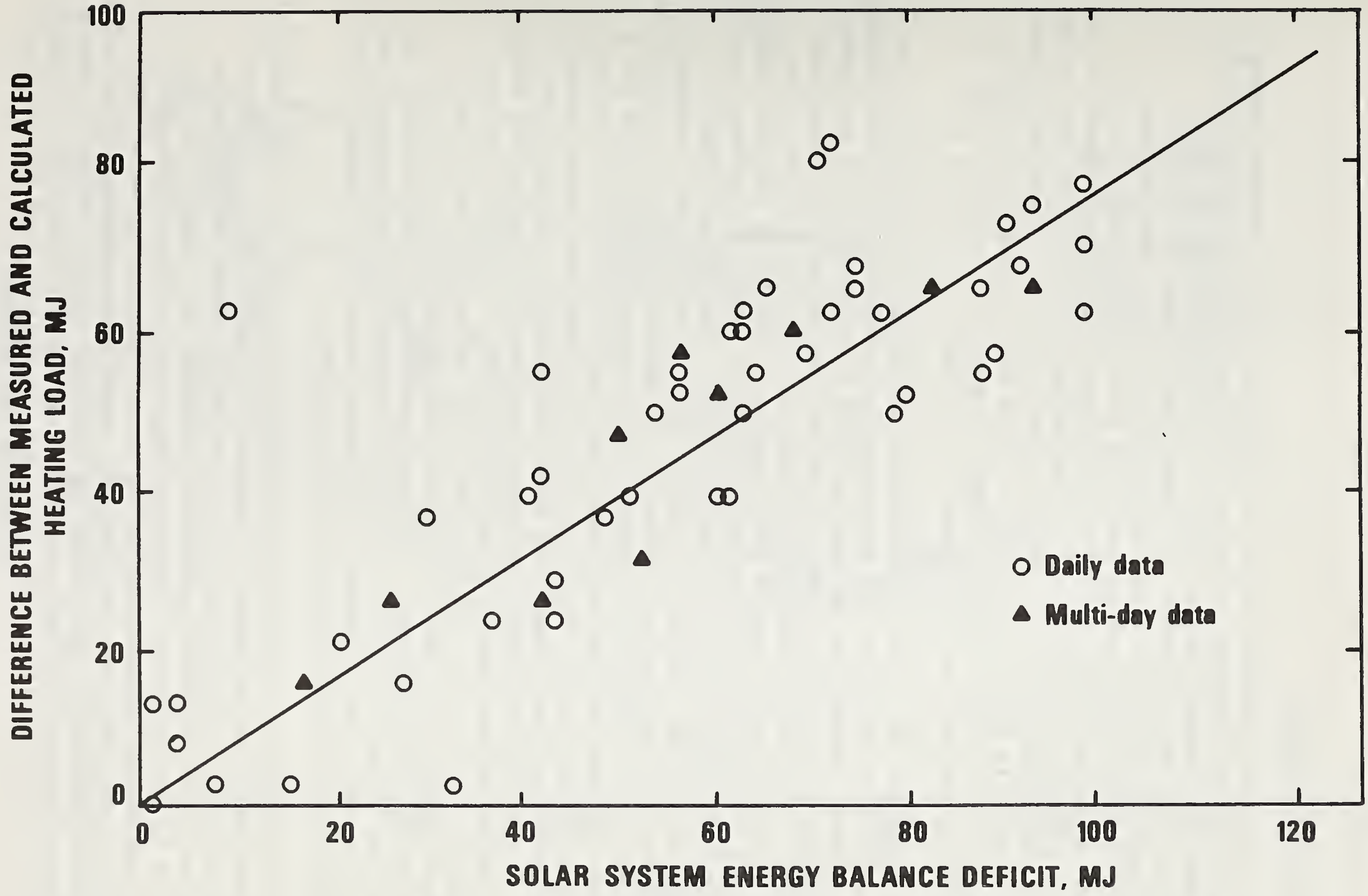

Figure 13. Correlation of solar system energy balance with measured load variation.

7. TEST RESULTS

Collector Performance

Results from the collector thermal performance tests conducted during the summer of 1977 yielded an optical efficiency (y-intercept) of 0.62 and a heat loss factor (curve slope) of 7.38 W/($m^2 \cdot$°C) (1.3 Btu/(h•$ft^2 \cdot$F)) when the collector efficiency was plotted according to ASHRAE Standard 93-77, as shown in Figure 10. These results were obtained with the reflector covered with a flat black material, as explained previously, and collector efficiency calculations based on the aperture area of 8.13 m^2 (88 ft^2). The collector was tested with a flow rate of approximately 0.19 m^3/s (400 ft^3/min) with a corresponding pressure drop of 0.19 kPa (0.75 in. H_2O). Flow rates through the collector during system operation normally ranged from 0.18 to 0.21 m^3/s (390 to 450 ft^3/min) depending on which mode the system was operating in.

It was difficult to predict the collector performance analytically due to inadequate knowledge of the heat transfer coefficient off the absorber fins. However, analytical methods outlined in reference [23] were used to calculate theoretical values of 0.58 for the optical efficiency and 4.94 W/($m^2 \cdot$°C) (0.870 Btu/(h•$ft^2 \cdot$°F)) for the heat loss factor. The values were calculated using a $(\tau\alpha)_e$ estimate of 0.72, a top loss coefficient of 6.6 W/($m^2 \cdot$°C) (1.2 Btu/(h•$ft^2 \cdot$°F)), a radiative exchange coefficient of 7.69 W/($m^2 \cdot$°C) (1.35 Btu/(h•$ft^2 \cdot$°F)), and the performance equations in Figure 7.12.1(d) of [23]. The convective heat transfer coefficient off the fins was assumed to be between 5.0 and 44.7 W/($m^2 \cdot$°C) (0.88 and 7.87 Btu/(h•$ft^2 \cdot$°F)) which are calculated coefficients for a stagnant air layer and for turbulent flow conditions based on the air flow area, respectively. The flow was in the turbulent range through the collector, although there may have been stagnant pockets between fins. Calculating collector efficiency factors, F', for the two extreme cases yielded values ranging from 0.88 to 0.90. Therefore, the heat transfer coefficient for the fins has little effect on the collector efficiency factor, the major heat loss path being from the air flow stream to the inner glazing. The large discrepancy between actual and theoretical values of collector heat loss coefficient was probably due to air leakage in the collector. Air leaks around the perimeter of the collector were observed during the test program.

The solar collector performance determined during the winter space-heating tests could not be directly compared to the performance curve of Figure 10 due to the presence of the reflector during actual system operation. The collector performance appeared to be substantially enhanced by the reflector. Figure 14 is a plot of clear-day hourly collector efficiency values taken during the winter heating season with the collector-only test curve included for comparison purposes. A reflector enhancement factor was calculated for each set of clear-day data points. The factor was based on the assumption that the difference in performance was due entirely to additional reflected irradiation from the reflector.

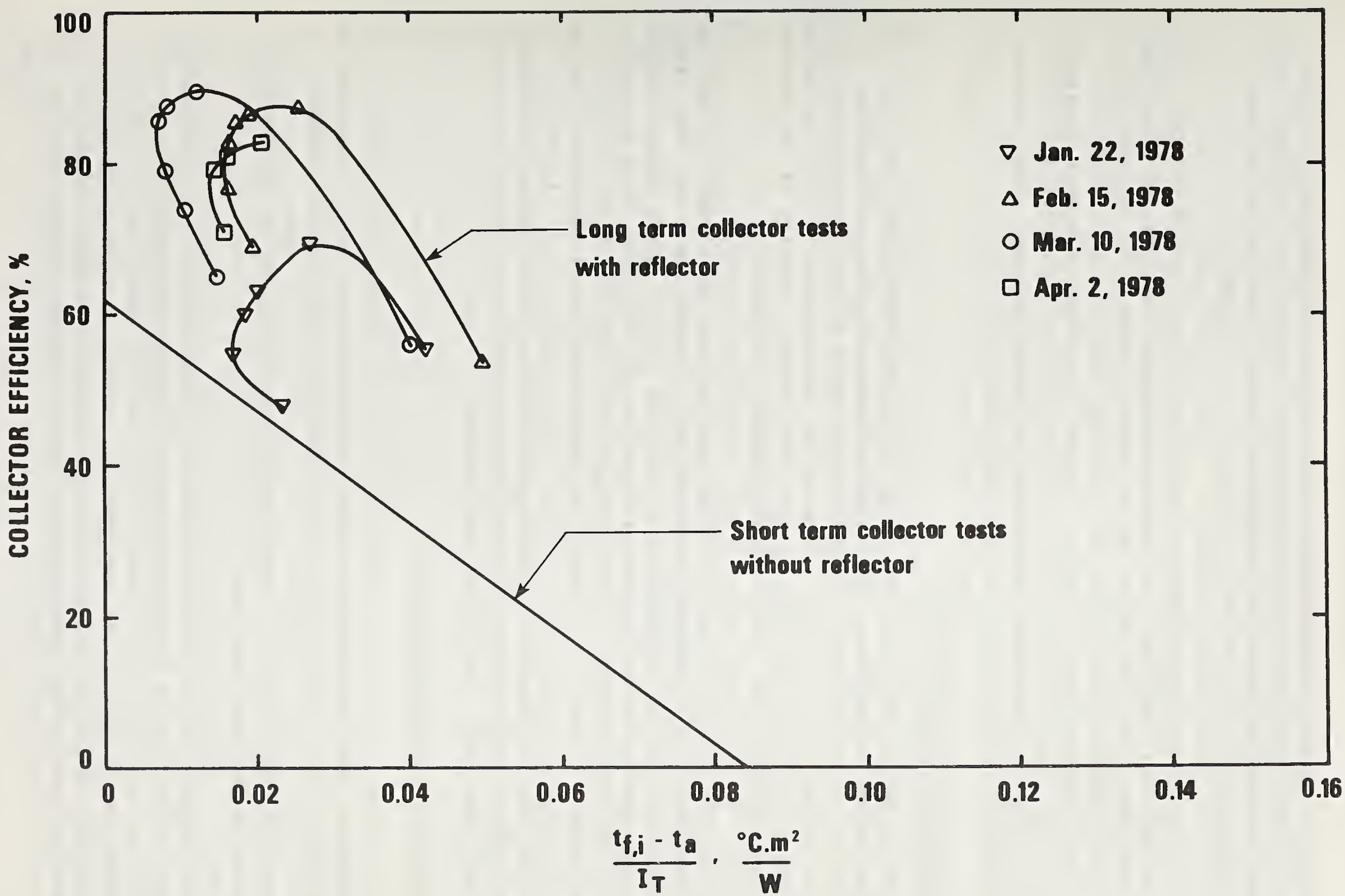

Figure 14. Solar collector performance with reflector.

Calculated clear-day enhancement factors ranged from 1.24 to 1.60 for the data shown in Figure 14. The horseshoe shape of the collector performance curves in Figure 14 is characteristic of collectors such as air-heating collectors which have large time constants.

Early in the test program during a warm sunny day in which the collector fan was not operating due to a temperature sensor failure, a dark brown liquid condensed on the inside of the inner glazing. It is suspected that this substance was driven off from the wood in the collector due to the high temperatures created. Towards the end of the test program, a dirty film slowly developed on the inner glazing, probably due to outgassing of some material within the collector housing. The film was not removed and its effect was not evaluated.

Pebble-Bed Storage Performance

The pebble-bed in this space-heating system was oversized. The ratio of storage volume to collector area commonly used in solar air-heating systems is 0.15 m^3/m^2 (0.5 ft^3/ft^2). The ratio in this particular system was 0.87 m^3/m^2 (2.8 ft^3/ft^2). Temperature measurements within the pebble-bed indicated that approximately half the storage capacity of the device was never utilized for energy storage during the tests.

The pressure losses in the air stream passing through the pebble-bed were excessive due to a relatively tortuous path through the unit. The pressure drop across the pebble-bed was 1.25 kPa (5.0 in. H_2O) under typical operating conditions as compared to a value of 0.062 kPa (0.25 in. H_2O) typical in other systems. This pressure drop required a large amount of fan power to move air through the pebble-bed, which adversely affected the performance of the entire system. After the unit was dismantled, it was noted that a great deal of dust and finely crushed stone was present in the pebble bed. The pebble-bed container and access panels appeared to remain air tight during the test period.

Auxiliary Equipment Performance

The controller functioned correctly during the entire test, except for a brief period when the collector temperature sensor failed. The cause of the failure was a broken wire on the resistance temperature element.

The two blowers within the solar system and the one in the mobile home operated without any problems.

The two dampers on the inlets of the two blowers within the solar system (magnetically latched back-draft dampers) performed poorly. When the system was installed, the damper on the load fan would open whenever the collector fan came on. Hot air from the collector would enter the mobile home during the solar collection period, causing overheating of the mobile home. This problem was solved by placing weights on the damper.

There was a great deal of leakage through the dampers and/or partitions within the solar system. Air leakage from the solar system to the mobile home during collector operation continued to be a problem even after modification of the damper as noted above; however, living space temperatures rarely exceeded 24°C (75°F). A second problem caused by leaky dampers was flow through the collector at night when the load fan was operating. The high pressure drop across the pebble-bed caused this leakage problem to be more serious than it might otherwise have been.

Daily System Performance

In order to describe the daily performance of the solar system, a typical day's in performance is shown here in detail. The day chosen was February 4, 1978, which was clear and cold.

Beginning at midnight on this day, the storage was completely discharged and the entire load was being supplied by the auxiliary electric heaters. Room temperature was approximately 20°C (68°F) and ambient temperature was approximately -10°C (14°F) at this time.

Shortly after 9:00 am, the collector fan came on, after which the delivery or load fan started operating. The average temperature of the air supplied to the mobile home by the solar system during the first hour was 17.5°C (63.5°F), which was too cool for heating. The auxiliary heaters continued to supply energy during the first hour. A time lag in supplying heat directly from the collectors to the mobile home was caused by a section of the pebble-bed being in the flow path between the collector exit and the load supply duct.

The collector fan continued to run until approximately 4:30 pm and the load fan continued to run intermittently throughout the day. Inside room temperature reached a high of 23.3°C (74°F) at 4 pm, which was above the set point temperature in the home of 21°C (70°F). This effect was caused by leakage through the load fan damper when the collector fan was operating as already mentioned.

After sundown, the load was met by energy from storage with an occasional small boost from the electric auxiliary heaters. There was a substantial amount of flow through the collector at night, induced by the load fan. In the late evening, there was a 15.5°C (28.0°F) temperature drop across the collector. Using the collector heat loss factor, and this temperature drop, the flow rate through the collectors was calculated to be 0.062 kg/s (8.2 lb/min), or approximately 30% of the normal flow rate when the collector fan was operating.

An energy audit over the 24-hour period starting at 6 am on February 4 and continuing to 6 am on the 5th yielded the results shown in Table 2. This period was chosen since there was negligible change in internal energy of the storage device between the beginning and end of this period.

Table 2. Daily Energy Balance for the Solar Space-Heating System for February 4, 1978

Measurement	Symbol	Value
Internally generated heat	E_{GEN}	28.0 MJ
Heating load	L	28.4 MJ
Additional infiltration due to wind[a]	L*-L	24.8 MJ
Corrected heating load	L*	273.2 MJ
Solar energy incident on 60° tilt away from reflector	G_T	209.6 MJ
Solar energy collected	Q_u	140.9 MJ
Energy delivered by solar system	Q_D	75.6 MJ
Heat leakage to mobile home[a]	$Q_D^*-Q_D$	59.7 MJ
Corrected energy delivered by solar system	Q_D*	135.3 MJ
Collector fan energy consumption	E_C	16.5 MJ
Load fan energy consumption	E_D	22.3 MJ
Mobile home fan energy consumption	E_H	16.9 MJ
Auxiliary heaters	E_{AUX}	127.9 MJ
Nighttime collector losses[a]		22.5 MJ
Change in stored energy	ΔU	0 MJ
Storage and supply duct losses[a]		23.2 MJ
Solar fraction	f	32.8%
Coefficient of performance	COP	3.5
Solar system fraction	f'	47.0%
Storage and delivery efficiency[b]		74.8%
Collector efficiency[c]		67.2%
Electric energy saved, fossil conventional[d]		-38.8 MJ
Fossil energy saved, fossil conventional[d,e]		223.5 MJ
Electric energy saved, electric conventional,[f,g]		145.3 MJ

[a] estimated value

[b] $$\frac{Q_D^*}{Q_D^* + \text{Nighttime Collector Losses} + \text{Storage and Supply Duct Losses}}$$

[c] $$= \frac{Q_u}{G_T}$$

[d] assuming the mobile home would be heated conventionally with a fossil energy source

[e] assumed fossil fuel conversion efficiency of 65%.

[f] assuming the mobile home would be heated conventionally with an electric energy source

[g] assumed electric conversion efficiency of 100%

Seasonal Performance

Monthly performance of the system was derived from use of the simplified instrumentation. Results are presented in Table 3 for the last half of the 1977-78 winter heating season.

The monthly solar fraction ranged from 12% to 60% with an overall solar fraction of 24% for the five months. If it is assumed that the mobile home would be heated conventionally by a fossil energy source, the total electrical energy consumption for the home and system, including an average household consumption of 300 W, increased approximately 70% by the addition of the solar system. The coefficient of performance, which is defined as the useful solar energy delivered by the system divided by the operating energy, was 1.9. Fossil fuel usage was reduced 42% by the addition of the solar system. If the mobile home were heated conventionally with a total electric system, a net energy savings of 7971 MJ (7.5 MBtu) or 28% at the building boundary would have resulted for the five month period.

8. CORRELATION OF SYSTEM PERFORMANCE

The detailed instrumentation was subject to frequent interruptions so that continuous test data were not produced for the five month period. Therefore instead of presenting test data on a monthly basis, data are presented here on either a daily basis or on a multi-day basis (3 to 7 days depending on data available).

Figure 15 is a plot of multi-day solar fractions (horizontal bars) versus the day of the year. Monthly values obtained through the use of the simplified instrumentation are also shown (cross-hatched rectangles). Note a general trend toward higher solar fractions as the winter came to an end. The multi-day solar fractions ranged from approximately 7 to 75 percent. The upper limit on the solar fraction resulted from the low COP of the solar system. In other words, if the system were operating in mild weather and no auxiliary heating was required, the solar fraction would still be limited to approximately 75 percent due to the fan operating energy requirements (included in the calculation of solar fraction).

Figure 16 is a plot of the useful energy collected by the collector divided by the heating load as a function of the solar load ratio (SLR). A similar plot can be drawn for the energy delivered to the load. However, since the thermal storage subsystem is between the collector and the mobile home, a term for the net change in stored energy must be included. Figure 17 is a plot of the energy delivered to the load plus the net change in stored energy divided by the load plotted as a function of the solar load ratio. It is evident from Figures 16 and 17 that using the solar load ratio is a good correlation technique for the present system.

Table 3. Monthly Performance of the Solar Space-Heating System

Month	Days in Month	Average Ambient Temperature (°C)	Energy to Solar System Fans (MJ)	Energy to Mobile Home Fans (MJ)	Energy to Mobile Home Excluding Electric Auxiliary Heaters (MJ)	Energy to Electric Auxiliary Heaters (MJ)	Average Wind Speed (mi/h)
12	21	-.5	425	261	322	2844	9.8
1	31	-2.9	627	428	759	5508	11.0
2	28	-3.5	829	413	741	4392	10.0
3	31	3.7	778	340	778	2574	11.2
4	30	11.1	675	193	407	295	11.7
			3334	1635	3007	15613	

Envelope Heat Loss (MJ)	Corrected Space Heating Load(MJ)	Solar Fraction	Solar System Fraction	Mobile Home Heating System Electrical Energy Consumption with Solar System Installed and Electrical Auxiliary (MJ)	Mobile Home Heating System Electrical Energy Consumption with Conventional Electric Heat (MJ)	Total Electric Energy Saved, Electric Conventional[ab] (MJ)
4289	4604	.233	.326	3530	4865	1335
7039	7453	.119	.204	6563	7881	1318
6517	6764	.167	.290	5634	7177	1543
5095	5182	.288	.438	3692	5522	1830
2822	2915	.601	.833	1163	3108	1945
25762	26918	.235	.359	20582	28553	28553

Mobile Home Heating System Fossil Consumption with Solar System Installed and Fossil Auxiliary[c] (MJ)	Mobile Home Heating System Fossil Consumption with Conventional Fossil Heat[c] (MJ)	Total Fossil Saved, Fossil Conventional[d] (MJ)	Total Electric Energy Saved, Fossil Conventional[d] (MJ)
4375	7083	2708	-425
8474	11466	2992	-627
6757	10406	3649	-829
3960	7972	4012	-778
454	4485	4031	-675
24020	41412	17392	-3334

[a] assuming the mobile home would be heated conventionally with an electric energy source

[b] assumed electric conversion efficiency of 100%

[c] assumed fossil fuel conversion efficiency of 65%

[d] assuming the mobile home would be conventionally heated with a fossil energy source

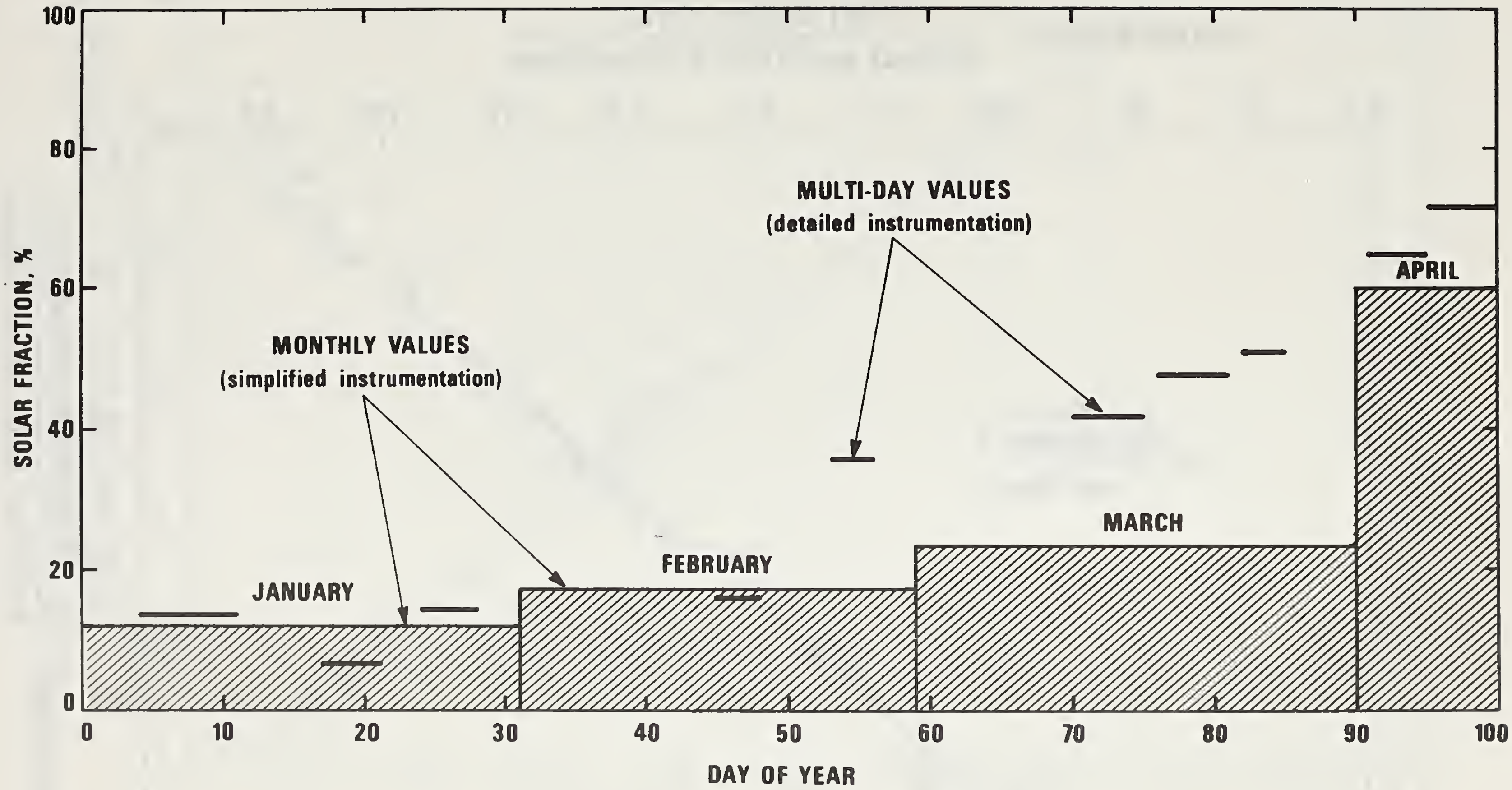

Figure 15. Variation of solar system fraction over the test period.

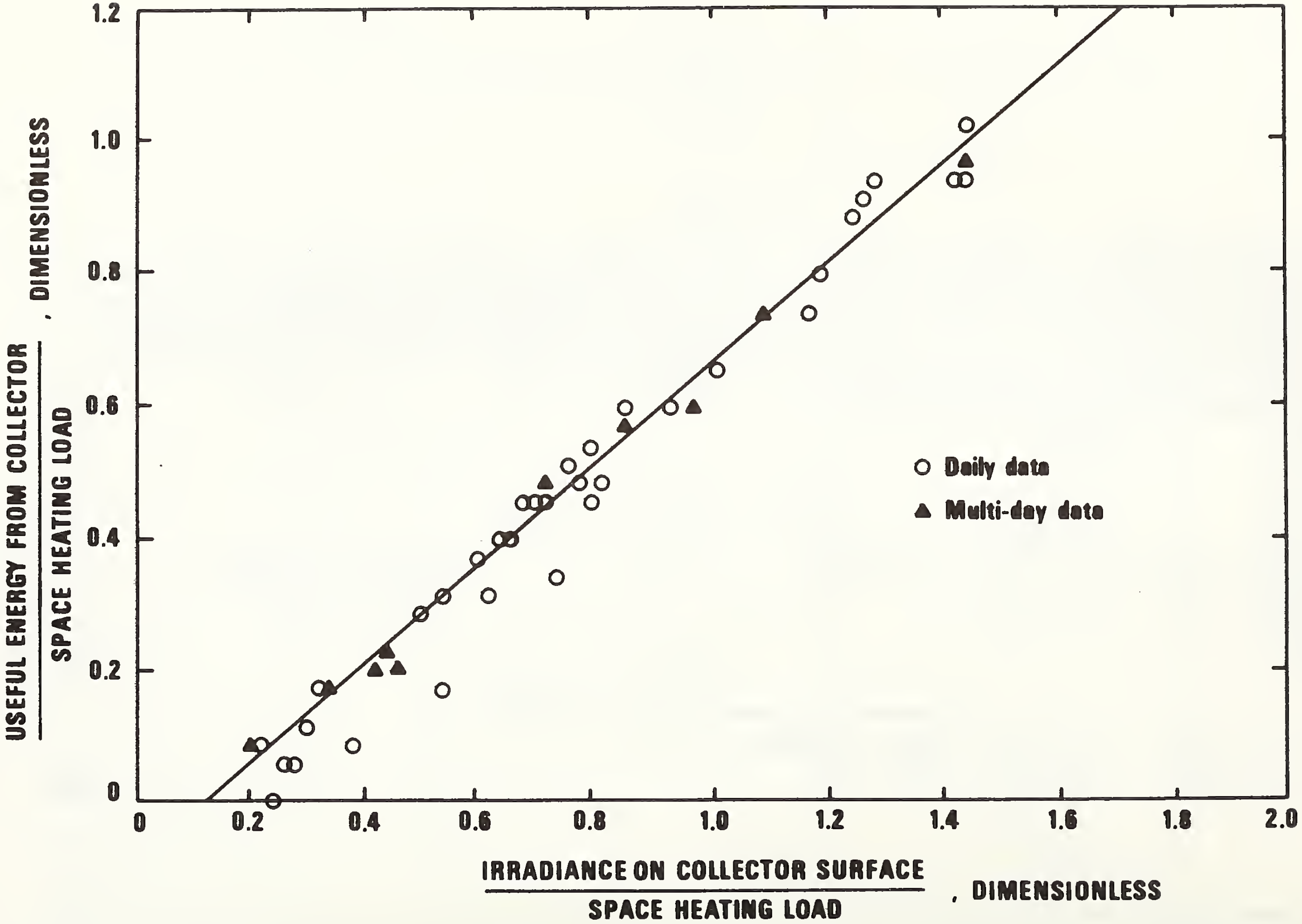

Figure 16. Useful energy delivered by the solar collector versus the solar load ratio.

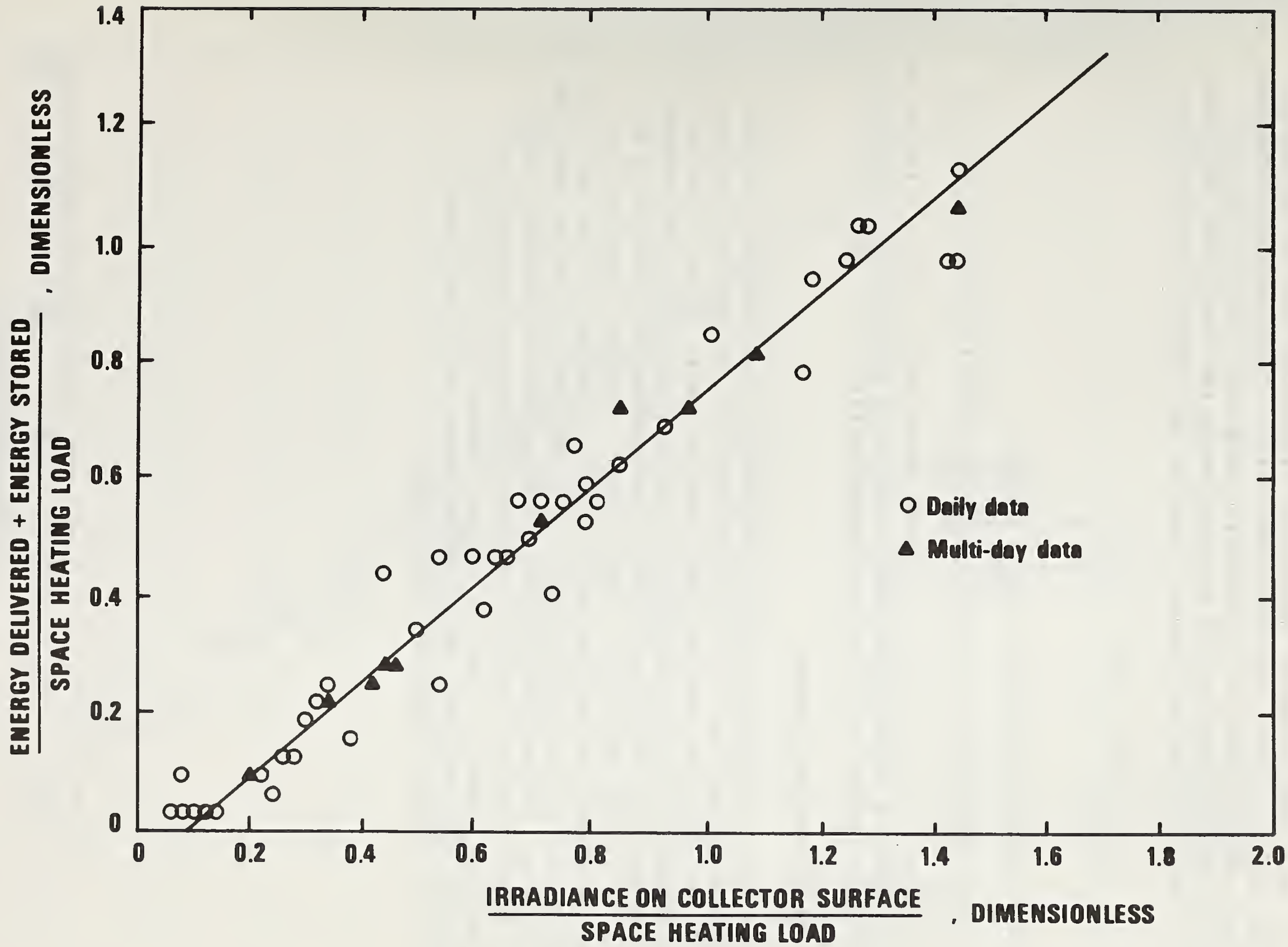

Figure 17. Energy delivered to the load and retained in storage versus the solar load ratio.

Figure 18 is a plot of solar fraction (f) (adjusted for the net change in stored energy) versus the solar load ratio. Daily and multi-day data points are shown as well as a cumulative data point for the full five month test period based on data obtained using the simplified instrumentation. The following performance equation was derived from the plot:

$$f = 0.741 \cdot SLR - 0.146 - \Delta U/L^* \qquad (9)$$

where

SLR = solar load ratio, dimensionless

ΔU = net change in stored energy over the time period, kJ.

The above result is significant since it shows that a single calculation of the five-month value of the solar load ratio enables one to evaluate the five-month solar fraction using equation (9).

9. SUMMARY AND CONCLUSIONS

Through tests on a complete solar space-heating system installed on a mobile home, it has been shown that the fraction of the load supplied by solar energy correlates very well with the ratio of the total solar radiation falling on the collector to the load imposed on the system. Such a correlation should be location independent and could serve as a basis for rating the system. For example, the solar fraction for a specific value of solar load ratio could be used as the rating value for the system. It is recommended that a detailed simulation study be undertaken now, as was done in [19], to determine the smallest number of days plus the kind of control required on the environmental and operating conditions in order to produce a meaningful and accurate correlation. In addition, the extent to which the correlation is location independent should be determined.

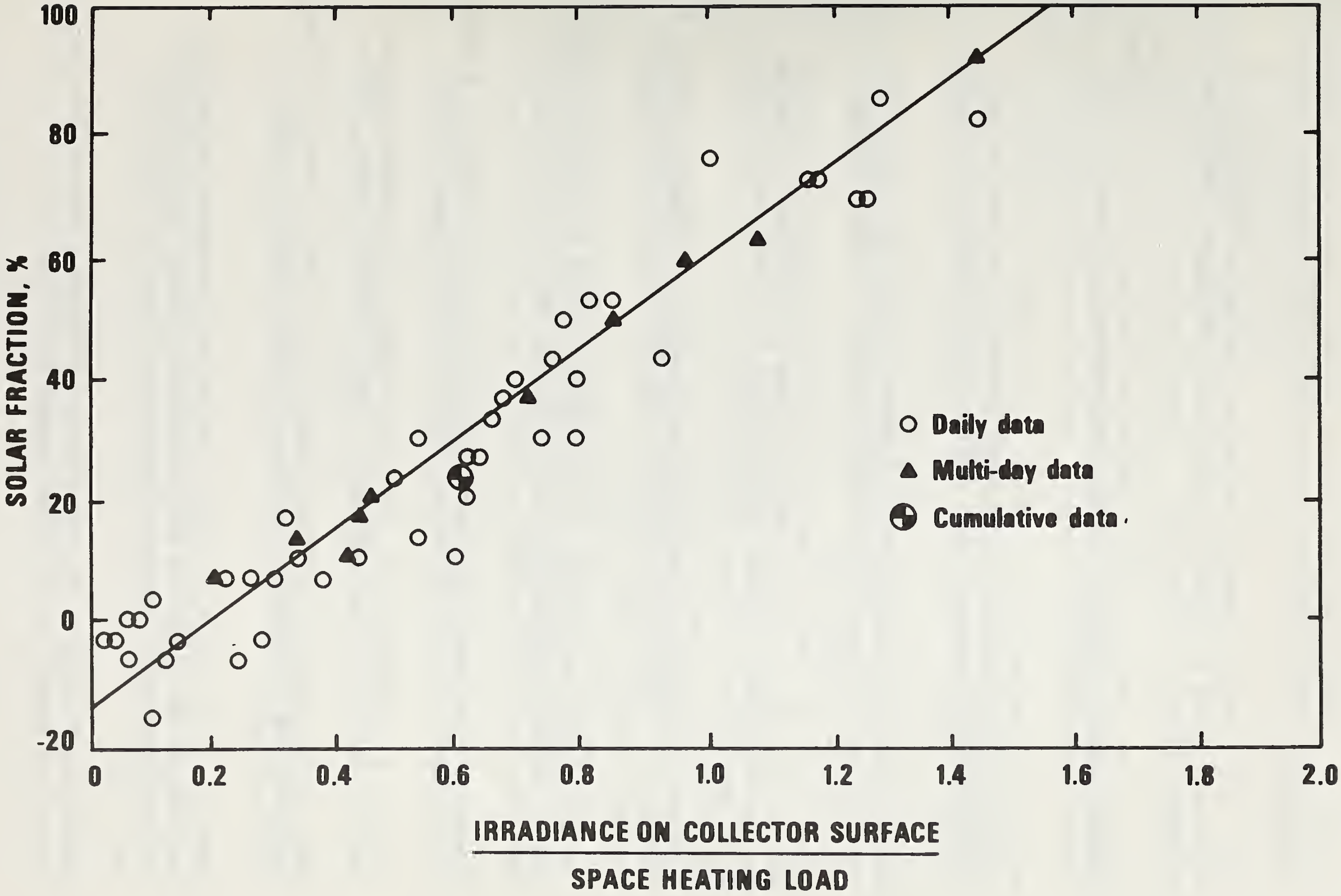

Figure 18. Solar fraction versus the solar load ratio.

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15. SUPPLEMENTARY NOTES

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

16. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.)*

As part of a continuing program to develop test methods for solar heating equipment, NBS is now developing a standard test procedure for packaged solar space-heating systems similar to test procedures now used for solar collectors and thermal storage devices, and now under development for packaged solar water-heating systems. As a first step, a mobile home, which was previously tested for thermal performance in an environmental chamber, was equipped with a packaged solar space-heating system using air-heating collectors and pebble-bed storage. The system was fully instrumented and data were collected over the 1977-78 heating season at the NBS site in Gaithersburg, Maryland. The performance of the system was determined and various methods of correlating performance were explored.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)*

Mobile home; packaged solar space-heating system; rating; standards; testing.

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