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TESTS OF A REDUCED-SCALE EXPERIMENTAL MODEL OF A BUILDING SOLAR HEATING-COOLING SYSTEM

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16. Abstract

An experimental solar heating and cooling system model has been built and operated, combining elements that are programmable (e.g., heating and cooling load of a building and collected solar energy) with experimental equipment. The experimental system model was based on the loads and components used in the Solar Building Test Facility (SBTF), which includes a 1394 m^2 solar collector field at NASA Langley. These tests covered 5 continuous days under summer conditions. For the system model up to 55 percent of the simulated collected solar energy was used for the building load. This amount of solar energy supplied 35 percent of the building cooling load. Heat loss was significant. If tank heat loss were eliminated, which would make it similar to the actual SBTF, 75 percent of the collected solar energy would be used. This amount would supply approximately 50 percent of the building cooling load. A higher fraction of solar energy is possible with a more performance-optimized system.

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TESTS OF A REDUCED-SCALE EXPERIMENTAL MODEL OF A BUILDING SOLAR HEATING-COOLING SYSTEM

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SUMMARY

As part of the national solar heating and cooling research, an experimental solar system model has been built and operated. The model is unique in that it is capable of combining programmable elements (e.g., heating and cooling load of a building, and collected solar energy) with experimental equipment. Together, the programmable elements and the physical model provide a dynamic simulation of the heating and cooling of a given building under measured weather conditions.

In the application herein reported, the system model operation was based on the building cooling loads, the weather conditions, and the solar energy collected from the 1394-square meter collector field all used in the design of the Solar Building Test Facility (SBTF) at NASA Langley. The model was scaled to 1/50th of the loads and component capacities of the SBTF. The tests covered five continuous summer days and simulated a stage during the startup period of the system.

During this period a large part of the solar energy was diverted to the storage water. The storage tank approached a fully heated condition at the end of the fifth day. On the fifth day - which approximated a fully heated, steady-state condition of the system - 55 percent of the simulated collector solar energy was used for the building load. As a ratio of the building cooling load, solar energy provided 35 percent.

If storage tank heat loss were eliminated (the SBTF storage tank is vacuum jacketed), 75 percent of the collected solar energy could be used for the building load. This amount would supply approximately 50 percent of the building cooling load. This system, based on an initial nonoptimized design, is expected to operate with a higher fraction of solar energy when optimized for performance.

System heat loss in this test was significant and approximately constant. The loss ranged from 29 percent of the simulated solar energy on a high collection day to 47 percent on the lowest collection day. The loss was of course related to the particular storage tank design used and to the amount of insulation used on the tank and throughout the test of the system.

INTRODUCTION

An application of solar energy that has been widely accepted as having practical, significant, and near-term potential is the heating and cooling of buildings. As part of a national technological effort, NASA Lewis is conducting research on solar heating and cooling components and systems. Previous reports on facilities, equipment, and results of tests (refs. 1 to 4) have been concerned with flat-plate collectors. Reference 5 describes an experimental solar heating and cooling system model in its preliminary phase. The uniqueness of this model is its capability of combining elements that are programmable - the heat load of a building, collected solar energy - with experimental equipment. Together, the programmable elements and the physical model provide an operating simulation of the heating and cooling of a given building under measured weather conditions (within the limits of available weather data). In addition to investigating dynamic characteristics of solar heating and cooling systems, the system model can

(1) Evaluate modes of operation and controls on system performance

(2) Compare the effect that different designs of a component have on system performance

(3) Provide an experimental verification for a computer heating and cooling simulation program

The present system model configuration is based on the design of the solar heating and cooling system of the Solar Building Test Facility (SBTF) at NASA Langley. External heat, heat storage capacities, and flow rates were based on a 1/50 scale of the SBTF system. This scaling factor allowed the use of components that were of commercial size (3-ton chiller) or that were readily available (1100 to 2200 ℓ storage tanks).

This report discusses the operation and test results of the system model when applied to building loads and weather conditions that were used in the design of the SBTF. The solar energy input was based on a 1400-square-meter collector field. The model test was based on weather conditions measured in 1959 during the month of August - a period imposing a high-energy load on the system. The model system was operated over a 5-day period during which heat distribution was accounted for on an hourly basis, day and night. The results indicate the effectiveness of the solar-energy system in filling part of the building energy load.

SYMBOLS

c_p specific heat of storage water, $J/g \cdot {}^{o}C$

n number of hours for which standard deviation was calculated

ΔQ difference between Q_h and heat gain calculated by heat balance method for given hour, J

Q_h sum of heat gain in total storage tank for given hour, J

ΔQ_x heat gain in region x of storage water for given hour, J

s standard deviation between values of Q_h and heat gain calculated by heat balance, J

T_x temperature of storage water in region x, ^oC

T'_x temperature 1 hr before T_x was measured, ^oC

V volume of region x of storage water, m^3

ρ density of storage water, kg/m^3

$\Delta\tau$ time interval that solar heat supplies heat load, min

DESCRIPTION OF SYSTEM

The solar system model, shown schematically in figure 1, uses water as the working fluid and consists of a steam heat exchanger simulating the heat input from a solar collector field, a 2270 liter hot water storage tank, an expansion tank to allow for volume change of the system, a cold water heat exchanger to simulate the building heat load, and a steam heat exchanger to function as the auxiliary heater. These components are linked together with two pumps so that the heat load requirement is met either by the stored hot-water - solar input combination or by the auxiliary heater. Figures 2 and 3 are photographs of the component panel and the storage tank.

Components

The heat exchangers are of conventional tube and shell design. The working fluid for the solar loop flows through the tubes while the steam heat source or water heat sink is in the shell portion.

The cylindrical stainless-steel storage tank is 2.4 meters high and 1.2 meters in diameter. Initially, the connections for flow into and out of the tank were placed at or near the centerline at the ends. But the jet effect of the flow caused temperature oscillations in the tank as reported in reference 5. For the current test the pattern of flow into and out of the tank was modified to disperse flow horizontally at the top and bottom of the tank. A 2.5-centimeter copper tube was connected to each port of the tank and shaped to reach the upper and lower regions of the tank (as shown in fig. 4). Twentyfour 0.478-centimeter holes drilled horizontally into each tube (12 on each side) provided the dispersement of the flow.

The tank initially was covered with a 2.5-centimeter (1-in.) thick layer of elastomeric thermal insulation. While this insulation is inadequate for the heat storage use intended, it did permit us to run the system to determine its operating characteristics. Subsequently, the system was operated with a 15.2-centimeter thick fiber material insulation on the tank. The results shown herein are for the system with the thicker insulation.

Programming Units

The use of programming units (shown in fig. 5) in conjunction with the solar energy input and the heat load heat exchangers is a unique aspect of the system model. Their use enables the operator to preprogram the heat input to and the heat load demand from the system, which in turn can be based on recorded weather data. Another important asset is the repeatability of the input and load demand energies. In such a system comparisons of different components or different designs of a component can be made on a common basis.

The programmed heat inputs and load demands function as references. The actual values of input and load were obtained from the measurements of thermocouples connected differentially across the heat exchangers and of turbine flowmeters. The temperature difference and flow rate were multiplied electronically and compared with the programmed values. Any difference between the two values caused steam flow (simulating solar input) or cold water flow (simulating the heat load) to change. The change was designed to match the measured values to the programmed values.

The programming unit is in the form of a chart affixed to a rotating drum. A proper choice of drum motor speed and gears will determine the time span for one cycle. For this test one revolution took $4\frac{1}{2}$ days. The programmed heat schedule was scribed onto the chart. A probe tracing the scribed line served as an electrical "cam follower." The resulting output signal served as the reference that the measured heat inputs and outgos attempted to match.

Flow Control

Two kinds of flow control have been incorporated into the system. One is a throttle valve that insures that the fluid entering the storage tank or supplying the heat load is not cooler than a given set point (approx. 93° C). The second kind of control relies on two on-off valves. The two valves act synchronously so that the heat load receives either auxiliary heat or solar heat, depending on the availability of the solar heat.

Throttle valve. - The throttle valve is controlled by the fluid temperature discharging from the solar heat exchanger. As the sensed temperature increases, the valve allows higher flow rate to the tank or the heat load. The desired discharge temperature is set manually on a controller at the control panel. Additional features of the controller affect the response time of the valve.

Flow control, however, involves more than a simple response to temperature change. An unstable flow condition can occur where flow rate, fluid temperature, and heat input are interdependent. During the check out plan of the system, instability did occur when the flow rate through the solar heat exchanger was low. As the discharge temperature reached and passed the set temperature of the throttle valve, the valve opened, allowing the discharge to flow to the storage tank or heat load. The fluid flowing out of the solar heat exchanger circuit must be matched by fluid flow into the circuit. As the entering fluid is usually at a different temperature from the recirculating fluid, there would occur a significant change in the (mixed) fluid temperature at the solar heat exchanger inlet. This change in fluid temperature would be reflected as a change in the heat rate as measured by the differential thermocouples (as described under Programming Units). Adjusting to the change, the steam valve would respond accordingly. The change in heat rate, in turn, affected the temperature rise of the fluid through the solar heat exchanger and, closing the chain of interdependence, affected the throttle valve. This cyclic phenomenon is similar to that described in reference 6, which is concerned with the control of heat exchange and flow systems.

The solution to the system instability lay in two steps. (1) The flow in the solar heat exchanger circuit was increased. The low-flow-rate - high-temperature-rise condition was changed to a high-flow-rate - low-temperature-rise one. This had the effect of desensitizing the steam heat source and the working fluid by (a) keeping the mass of the heat exchanger at close-to-constant temperature and (b) minimizing the effect of fluid flowing into the bypass loop when the throttle valve opens to the system loop. (2) More control was exerted on the throttle valve itself. A single throttle valve, as originally designed, was expected to provide a sensitive control over the total range of flow up to 18.9 liters per minute. This requirement made it virtually impossible to provide sensitive control at the very low flow rates for the conventional valve. This meant that, in the initial stage of opening the throttle valve, the valve area change was much greater than the valve stem travel. As a result, the throttle valve tended to permit too much flow when first opened. To solve this problem, a second throttle valve was installed in parallel with and in the vicinity of the original valve. This valve was designed for a range from 0 to 5.7 liters per minute. The original valve was modified to operate from 5.7 to 18.9 liters per minute. These two steps have largely eliminated the instability. There is still a small oscillation of the valve, but it has a negligible effect on the measured heat input to the fluid.

The effect on the system of operating in this manner (high flow rate in the solar heat exchange circuit) is inconsequential as long as the flow discharged from the solar heat exchanger circuit is at a rate and temperature entirely consonant with the programmed input of solar energy.

On-off valves. - The on-off valves operate synchronously so that the heat load receives flow either from the auxiliary heat circuit or from the solar loop (solar heat input and hot water storage). The on-off valves are controlled by a thermocouple measuring the fluid temperature entering the heat load T_1 and by a thermocouple immersed in the hot water storage tank T_2 (fig. 1). The temperature controls are shown in figure 5.

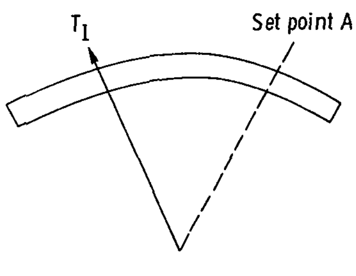
An operating logic was devised to maximize the use of the solar heat loop and minimize the use of auxiliary heat. Auxiliary heat would be used when the solar loop is inadequate to meet heat load demands. Auxiliary heat would be used also when the tank temperature is lower than that required to supply the load. This temperature limit is supplied to the controller as a set point (see sketch). To avoid possible cycling when the tank temperature is close to the set point, a dead band was used such that the solar loop was used until the tank temperature fell below a second, lower, set point. This logic can also be described by application of the following rules:

(1) Any time T_1 is below A, use auxiliary heat.

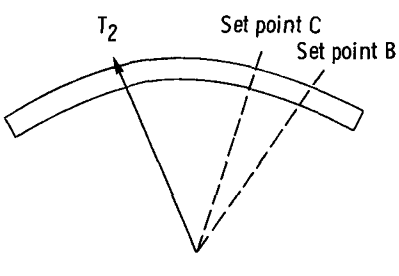
(2) When T_2 is above B, use solar loop.

(3) Once the solar loop is flowing, it must stay on until T_2 falls below C, unless rule (1) applies.

The set point values were $A = 93^{\circ}$ C, $B = 93^{\circ}$ C, and $C = 91^{\circ}$ C.



Heat load flow temperature control



Tank temperature control

INSTRUMENTATION AND DATA ACQUISITION

Temperature was measured by type E (Chromel-constantan), type K (Chromel-Alumel), and type T (copper-constantan) thermocouples. These thermocouples were taken from spools from which sample pieces were calibrated according to the National Bureau of Standards procedure. All calibrations were within Instrument Society of America specifications ± 0.21 C^o.

The use of the different types of thermocouples depended on their functions. Type E was used mainly for data measurement because of its relatively high sensitivity to temperature change. Type K was used when linearity was critical (e.g., differential measurements). Type T was used when compatibility with the controls was required.

Heat distribution around the system is the measure of system performance. Since this parameter depends on measuring the temperature change across heat exchangers, type K thermocouples were connected differentially. These temperature differential readings were used with absolute temperature measurements to determine the local slope of the thermocouple output curve. Flow mixers were used to obtain bulk temperature measurements.

Determination of heat into and out of the storage tank was accomplished by use of thermocouples attached to tubes immersed in the tank liquid. One rake (fig. 6) was positioned vertically on the centerline of the tank; it had seven thermocouple spaced 30.5 centimeters apart. These seven thermocouples were calibrated with tank volume (as shown in fig. 7) by timing a known rate of water flow into the tank until each thermocouple reading changed due to its immersion in water.

Horizontal rakes were installed through both ports in the side of the tank. Four thermocouples on each rake were used to indicate the extent of turbulence during flow into and out of the tank. Thermocouples were also attached to the outside surface of the tank and on the outside of the insulation.

Turbine flowmeters were used to measure flow rate. The accuracy of these meters is within 1 percent of the indicated readings.

Pressure gages and tranducers were used throughout the system.

The temperature differential and flow rate readings, besides providing data, provided the input with which to compare the programmed value of heat increase or decrease.

TEST PROCEDURE

The programmed heat input and load demand from the system were determined from the results of NASA Langley's NECAP (NASA's Energy Cost Analysis Program (ref. 7)). NECAP is patterned after the popularly known Post Office Program (ref. 8). It is a sophisticated building design and energy analysis tool which has embodied all of the latest ASHRAE state-of-the-art techniques for performing thermal load calculation and energy usage prediction. NECAP has been used to define the building heating and cooling requirements of the SBTF. NECAP was used to analyze the first 7 days of each month during the year 1959. Since the programming units were designed for a $4\frac{1}{2}$ day period, the building demand was scanned for a 5 consecutive-working-day period. Most of the energy expended was for cooling. Though July would have been a logical month to choose as a high energy expenditure period, it was excluded from consideration because the Fourth of July holiday interrupted the work week. August was chosen instead.

Programmed Heat Load

The computer output of the NECAP program was in terms of the energy required per hour to keep a particular zone of the building within set temperature limits. Since ventilation was not included in the output, the heat load due to this factor was calculated and added to the NECAP values. Where cooling was called for, the cooling load value was divided by 0.67 - the Coefficient of performance (= cooling output/energy input) of the lithium bromide - water absorption chiller system. Using a constant coefficient of performance is a good assumption for this type chiller. The total heat load was then divided by 50 - the scaling factor for the system model. These values for each hour, were scribed onto the heat load rotating drum chart.

Programmed Heat Input

The heat into the system depends on the solar and climatic conditions under which a collector field operates and the efficiency with which the energy is transferred into the system.

Calculating the direct solar radiation is a trigonometric exercise which accounts for the position of the Sun in relation to the point on Earth where the collectors are tilted at some angle. This procedure can be found in reference 9. Cloudiness is taken into account by the cloud cover modifier (often abbreviated as CCM; in NECAP, written as CC for each hour). Thus, the direct solar radiation incident on the collector field was calculated based on weather data. Flat-plate collectors, however, also use diffuse solar radiation. To determine the diffuse value when only the direct radiation value is known, it is necessary to use a correlation among direct, diffuse, and total radiation such as that reported in reference 10. The correlation is based on the observation that, over a large range of total radiation values, increased cloudiness causes a drop in the total solar radiation, a lower direct radiation, but a higher value of diffuse solar radiation. The appendix describes the use of the direct, diffuse, and total radiation relations used in the present test.

Once the total energy incident on the collector field was known, the efficiency of collection was calculated. The calculation was based on the results of test data such as published by F. Simon (ref. 11). His data on some 15 designs of collectors are plotted as collector efficiency as a function of (inlet temperature - ambient temperature)/incident flux. The collector chosen for use in this test was a relatively high performance, two glass pane, selective surface design. Ambient temperature was given hourly in the NECAP program. The resulting calculated values of heat input were scribed on the rotating drum chart.

Operational Procedure

The system model was operated initially with 2.5-centimeter thick insulation on the thermal storage tank. The excessive heat loss from the tank prevented proper day-today operation of the system; however, the control functions were confirmed under system operation, and further information was obtained on thermal stratification in the storage tank.

The system model was subsequently operated with 15.2 centimeters of fibrous glass insulation on the tank. The tank water was heated prior to the week of operation, the water temperature ranging from 37° C at the bottom to 94° C at the top on the eve of the first test day. This range of temperature allowed observation of the temperature history and thermal gradients of the storage water during a startup period.

Heat loss in the lines was obtained experimentally in the various branches of the system. Heat loss in the tank was calculated from the overnight temperature changes in the tank water.

The system operation, based on real time, was scheduled for 11 hours per day during the test week. Five days of data were accumulated.

The main pump in the system is the one feeding the heat load. In the auxiliary mode the working fluid circulates through the auxiliary heater and the heat load. The flow was set at a constant value, approximately 22.2 liters per minute (scaled from the Langley SBTF condition). In the solar mode the flow is directed to the solar loop: one path through the solar input heat exchanger, the other path through the storage tank. The flow varied, depending on the division of flow between the solar input and the tank, up to 24 liters per minute.

The secondary pump feeds the solar input heat exchanger. The flow was set at a constant value (approximately 16.8 ℓ/min) for fluid circulating entirely through the bypass line. This flow rate minimizes the instability as described in the section Flow

Control and yet allowed an accurate measurement of the temperature rise through the heat exchanger.

ANALYSIS

Data were analyzed by accounting for the transfer of heat from one component or subsystem to another. The heat storage was analyzed independently and compared with the results of the heat distribution analysis.

Heat Distribution

The analysis is simplified by the either-or operation of the system. Collected solar energy either enters the storage tank or goes to meet the heat load requirements. The heat load receives the required heat from either the solar heat loop or the auxiliary heater. For a particular hour the following procedure details this approach: Determine

(a) The solar energy collected (= average of the solar heat rate during the hour), kW
(b) The time that solar energy supplied heat load, $\Delta\tau$, min
(c) The solar energy supplying heat load = $(\Delta\tau/60) \times$ (a), J
(d) The solar energy delivered to tank = (a) - (c), J
(e) The heat load expended = average heat load rate during hour, kW
(f) The heat load received from solar loop = $(\Delta\tau/60) \times$ (e), J
(g) The heat load received from tank = (f) - (c), J
(h) The heat load received from auxiliary = (e) - (f), J
(i) The gross heat gain in tank = (d) - (g), J
(j) The net heat gain in tank = (i) - (line and tank losses), J

Heat Change in Hot Water Storage Tank

The extent of heat gain or loss in hot water storage was determined by the use of the thermocouples affixed to the vertical rake. For calculation purposes the tank volume was divided into seven sections chosen so that each thermocouple reading would approximate the average temperature throughout its assigned section. Figure 8 shows the thermocouple locations within the tank.

Values of heat gain or loss were calculated hourly. The change for one section was

$$\Delta Q_x = \rho V c_p (T_x - T_x')$$

The change in the heat content of the tank is the sum of the individual heat gains or losses:

$$Q_h = \sum_{x=1}^{7} \Delta Q_x$$

The standard deviation between the values of Q_h and the net heat gain (item (j)) was calculated for each day. If the hourly difference is ΔQ, the standard deviation may be calculated as follows:

$$s = \sqrt{\frac{1}{n} \sum (\Delta Q)^2}$$

The standard deviation values serve as a verification of the accuracy of the experimental results. The greatest deviation is expected when transient effects occur. The resulting time lag in the thermocouple readings within the tank, then, could be a factor.

RESULTS AND DISCUSSION

The heat distribution for each of the 5 days is tabulated on an hourly basis in table I. The Collected solar energy is distributed to either the storage tank or the heat load. The heat load expended comes from either the solar loop or the auxiliary heater. The arrows on the headings indicate the direction of heat flow. The line and tank losses are the summation of the solar energy losses in the system. The tank heat gain was arrived at by two methods. The heat balance values were those calculated from a heat balance around the system. The ΔT values were those calculated by the measured temperature change of the thermocouples immersed in the storage tank. The time corresponds to the particular hours for which the solar supply and heat loads were measured.

The standard deviation of the ΔT values of tank heat gain from the heat balance values was calculated for each day. The deviation ranged from a maximum of 2534 kilojoules to a minimum of 1439 kilojoules. For comparison a change of 1° C in the tank water is equivalent to 9263 kilojoules. The close agreement between these two methods is a measure of the accuracy of the system operation.

On any one day, the daily sum of the solar energy collected is less than the daily sum of the heat load expended. The hourly pattern, however, reveals that the solar energy is collected over a shorter period of time and has higher values than the heat load (see fig. 8). The difference between the two curves emphasizes the importance of the thermal storage component in the system.

The role of thermal storage is illustrated in figure 9. The top and bottom thermocouple readings are shown for the 5-day period. The other five thermocouples have values that range between the two, but they have been omitted from the figure for clarity. The time periods when solar energy is being collected and the periods when the heat load is being supplied by the solar loop are indicated. Since the initial condition of thermal storage reflects a startup stage of operations, the lower regions of the tank were colder than would be the case during continuous operation.

The top thermocouple responds to the solar input and to the supplying of the heat load demand as expected - increasing as hot water flows into the tank from the top and decreasing as the hot water flowing from the top is displaced by colder water from the lower regions of the tank. The bottom thermocouple increases most rapidly during the period of solar supply to the heat load. This rapid increase occurs because the heat load, in using only the energy required, discharges the solar heated water approximately 3° below the inlet temperature. The water returning from the heat load displaces and mixes with the stored water in the bottom regions of the tank, thereby increasing the temperature there. Though the temperature rise is most apparent during the first day, it is repeated on the following days. At the end of the fifth day, the bottom temperature reached 88° C. After 5 days of system operation, then, the storage tank approached a fully heated condition. Considering the energy required to attain the partially heated condition going into the first day of operation, an estimate of a minimum of 2 weeks can be made for the heating-up period of the system. If the test were extended, the temperatures throughout the tank and system could very well have progressed to over 93° C. The system model, because it is pressurized, would be able to withstand such temperatures, but a nonpressurized system would not because of temperatures approaching the boiling point of the fluid. Temperature related problems are mitigated somewhat by the lower efficiency of the solar collectors that would result from their operating at higher temperature levels.

The sum of the daytime line and tank losses (table I) was approximately constant. The tank loss overnight depended on the temperature level of the various regions within the tank. With a constant, high temperature being approached for all of the storage regions as the test progressed, the tank loss also became constant. The total line and tank losses for 24 hours was approximately 42 200 kilojoules. This loss as a percentage of the solar input ranged from 29 percent for the highest solar collection day to 47 percent for the lowest.

The consequence of the heat loss from the storage water is shown in figure 10 as a temperature drop. The extent of the loss is seen most clearly overnight. The temperature drop of the storage water means that the control temperature in the tank falls further below the minimum set point. When solar energy is collected the following morning, therefore, the energy is directed to the tank to make up for the overnight heat loss. It is only when the heat has been recovered that the solar energy input can be directed to the heat load. This explains why the solar loop supplied the heat load only toward the end of the solar input period.

The amount of collected solar energy that is absorbed by tank storage (tank heat gain) is plotted as a ratio of the collected solar energy in figure 10. The curve, plotted on a daily basis, falls off as the days progress. This trend is a measure of the extent to which the storage water is heated by solar energy. As the storage water temperature approaches its operating level, more of the solar energy is directed to supplying the heat load - the useful solar energy. The curves of useful solar energy, therefore, show a general trend opposite to that of the storage tank heat gain. The dip in the curves on August 4th is due to the low solar input, resulting in a low useful solar energy value, for that day.

The useful solar energy curves are two ways of measuring the efficiency of the system. The upper curve is a measure of the effectiveness in using solar energy that has been collected. It is a measure of the technical efficiency of the system without the dominant effect of collector efficiency. The lower curve measures the proportion of the heat load that is supplied by solar heat. This curve is related to the upper one but, since it is referenced to the heat load, this ratio is a measure of the savings of conventional energy.

The curves in this figure, for operation during startup, vary over a larger range than would be expected if it had been operating continuously. Under continuous operation, one would expect the useful solar energy curves to level out approximately at the August 7th values, that is, 55 and 35 percent for the collected solar energy and heat load ratios, respectively.

CONCLUDING REMARKS

The conduct of the test was not intended to be detailed modeling of the system installed in the SBTF; however, comparisons with the facility can be made.

Heat loss was perhaps the single most significant factor in the test, particularly from the hot water storage tank. The tank in the system model has an insulation blanket of 15.2 centimeters. The storage tank used in SBTF is vacuum jacketed. Therefore, the performance values from the system model - useful energy being 55 percent of the solar energy input and 35 percent of the building cooling load - must be thought to be minimum values of efficiency for the SBTF.

If tank losses in the model were eliminated totally, the heat loss would be approximately half its previous value. The useful energy then would be about 75 percent of the solar energy input and would amount to approximately 50 percent of the building cooling load. This system, based on an initial nonoptimized design, is expected to operate with a higher fraction of solar energy when its performance is optimized.

SUMMARY OF RESULTS

This report described a laboratory-scale experimental solar heating and cooling system where heat input (simulating collected solar energy) and heat demand (simulating the building load requirements) are controlled, independent variables. The system was operated under the simulated weather conditions and high cooling load conditions (scaled to the system model size) that are expected during the summer for the Solar Building Test Facility (SBTF), located at NASA's Langley Research Center.

Based on real time testing, the system operated for 5 days and nights (a working week) and represented a period during initial startup of the solar system. During this startup phase, a large part of the simulated collected solar energy input went toward increasing the storage water temperature. Only when the full storage capacity reached the desired temperature, was most of the simulated collected solar energy used for the heat load. This fully heated storage water temperature was approached on the fifth day of operations. For this day (which approximated a steady state condition) 55 percent of the simulated collected solar energy was used for the building load. As a ratio of the building cooling load, however, solar energy provided 35 percent, an amount reflecting both the size of the collector field and its efficiency.

Heat loss in the experimental test system was significant and approximately constant. The percentage loss to the system was inversely proportional to the solar energy collected. For the highest collection day in the test, the loss amounted to 29 percent of the collected energy, and, for the lowest collection day, the loss was 47 percent. The losses experienced in this test are, of course, a function of the particular storage tank design and the amount of insulation used throughout the system.

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APPENDIX - PROCEDURE FOR CALCULATING DIRECT AND DIFFUSE SOLAR RADIATION

As mentioned in the text, direct solar radiation can be calculated by using the ASHRAE procedure outlined in reference 9.

For diffuse radiation the findings of Buyco and Namkoong (ref. 10) were used. The data in the reference are those from Blue Hill, Massachusetts, from 1952 to 1956. The data were plotted for each month on coordinates of $diffuse_h/total_h$ as a function of $total_h/total_{max}$. The subscript h is for the particular hourly reading, and max is for the maximum reading for that hour. The Blue Hill data are plotted on these coordinates in figure 12 for the month of August. The value of this correlation lies in its ability to determine diffuse radiation when only total radiation data are available. This is precisely the situation with the National Weather Service, which has only total radiation data for all but a handful of stations in the country.

In the present case direct rather than total radiation values are available. Figure 12 can still be used, however, by an iterative approach. The method is described as follows for a particular hour in August:

(1) Calculate the maximum direct radiation. Use the ASHRAE equation, (ref. 9) assigning a value of unity to the cloud cover modifier (clear day).

(2) Multiplying this value by 1.07 gives the maximum total radiation value ($total_{max}$) for the hour. The factor 1.07 accounts for the energy that is diffuse on clear days.

(3) A first estimate of the diffuse component must be made and added to the calculated direct radiation. This sum is $total_h$.

(4) Enter the curve of figure 12 with the abscissa value of

$$\frac{total_h}{total_{max}}$$

(5) Multiply the resulting $diffuse_h/total_h$ by $total_h$.

(6) Compare the product $diffuse_h$ with the original estimate.

(7) Reiterate if closer agreement is required.

(8) Modify $diffuse_h$ by the exposure to the sky seen by the collector. Diffuse energy reflected from the ground surroundings was not considered.

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TABLE I. - HEAT DISTRIBUTION BASED ON WEATHER DATA

Time	Collected solar energy, kJ	To tank, kJ	To heat load, kJ	From solar loop, kJ	From auxiliary heater, kJ	Heat load expended, kJ	Line and tank losses, kJ	Tank heat gain, kJ	
								Heat balance method	Temperature change method
August 3, 1969									
7-8 a. m.	0	0	0	0	[a]10 970	[a]10 970	883	-883	[a]-883
8-9	0	0	0	0	[a]11 390	[a]11 390	883	-883	[a]-883
9-10	11 389	11 389	0	0	13 080	13 080	3 017	8 372	8 000
10-11	22 768	22 768	0	0	14 038	14 038	3 017	19 750	20 926
11-12 n.	29 459	29 459	0	0	14 715	14 715	3 017	26 441	26 667
12-1 p. m.	30 072	30 072	0	0	15 376	15 376	3 017	27 054	27 150
1-2	20 787	1 039	19 748	15 503	817	16 319	4 000	1 285	-5 869
2-3	20 065	0	20 065	16 310	0	16 310	4 051	-296	-1 999
3-4	11 062	6 204	4 858	7 880	10 063	17 943	3 471	-289	-333
4-5	0	0	0	0	16 917	16 917	1 055	-1 055	-2 076
5-6	0	0	0	0	15 615	15 615	1 055	-1 055	-2 092
	145 602			39 693		162 680	27 468	78 441	
August 4, 1969									
7-8 a. m.	0	0	0	0	15 012	15 012	1 055	-1 055	0
8-9	0	0	0	0	14 711	14 711	1 055	-1 055	-3 279
9-10	6 129	6 129	0	0	15 749	15 749	3 017	3 111	4 756
10-11	13 043	13 043	0	0	15 369	15 369	3 017	10 026	9 817
11-12 n.	18 180	18 180	0	0	15 473	15 473	3 017	15 163	12 492
12-1 p. m.	18 655	18 655	0	0	15 544	15 544	3 017	15 638	14 149
1-2	18 831	2 824	16 007	13 459	2 375	15 834	3 896	1 476	-1 305
2-3	14 883	9 675	5 209	5 249	9 746	14 995	3 379	6 255	8 230
3-4	0	0	0	0	15 848	15 848	3 017	-3 017	-669
4-5	0	0	0	0	[a]14 665	[a]14 665	1 055	-1 055	[a]-1 055
5-6	0	0	0	0	[a]14 771	[a]14 771	1 055	-1 055	[a]-1 055
	89 723			18 708		167 971	26 583	44 432	
August 5, 1969									
7-8 a. m.	0	0	0	0	[a]14 350	[a]14 350	1 055	-1 055	[a]-1 055
8-9	0	0	0	0	[a]13 610	[a]13 610	1 055	-1 055	[a]-1 055
9-10	8 377	8 377	0	0	16 323	16 323	3 017	5 360	5 639
10-11	20 179	20 179	0	0	16 551	16 551	3 017	17 161	15 803
11-12 n.	26 430	15 987	10 443	7 214	9 434	16 649	3 466	15 750	12 732
12-1 p. m.	21 807	0	21 807	16 294	0	16 294	4 051	1 461	1 724
1-2	22 861	0	22 861	17 367	0	17 367	0	1 442	3 136
2-3	25 378	15 822	9 555	7 143	10 714	17 857	0	14 804	12 324
3-4	2 983	2 776	207	900	17 305	17 998	0	-986	-1 172
4-5	0	0	0	0	[a]16 248	[a]16 248	1 055	-1 055	[a]-1 055
5-6	0	0	0	0	[a]16 459	[a]16 459	1 055	-1 055	[a]-1 055
	128 013			48 918		179 703	28 324	50 772	
August 6, 1969									
7-8 a. m.	0	0	0	0	16 094	16 094	1 055	-1 055	0
8-9	0	0	0	0	15 671	15 671	1 055	-1 055	0
9-10	5 758	5 758	0	0	15 742	15 742	3 017	2 741	727
10-11	13 046	13 046	0	0	16 185	16 185	3 017	10 028	8 835
11-12 n.	17 539	15 115	2 424	2 090	13 589	15 679	3 156	12 293	10 219
12-1 p. m.	22 641	0	22 641	15 966	0	15 966	4 051	2 624	3 670
1-2	22 736	0	22 736	15 643	0	15 643	4 051	3 042	2 204
2-3	21 733	737	20 995	15 916	839	16 755	4 000	1 817	2 790
3-4	6 424	6 424	0	0	16 924	16 924	3 017	3 407	5 379
4-5	0	0	0	0	[a]14 982	[a]14 982	1 055	-1 055	[a]-1 055
5-6	0	0	0	0	[a]15 087	[a]15 087	1 055	-1 055	[a]-1 055
	109 877			49 615		174 727	28 530	-31 731	
August 7, 1969									
7-8 a. m.	0	0	0	0	[a]13 716	[a]13 716	1 055	-1 055	[a]-1 055
8-9	0	0	0	0	[a]12 977	[a]12 977	1 055	-1 055	[a]-1 055
9-10	6 043	6 043	0	0	15 505	15 505	3 017	3 026	3 360
10-11	13 284	13 284	0	0	15 071	15 071	3 017	10 267	7 919
11-12 n.	18 069	7 829	10 239	8 903	6 809	15 713	3 603	5 562	3 085
12-1 p. m.	19 827	0	19 827	15 569	0	15 569	4 051	207	1 647
1-2	23 369	0	23 369	16 174	0	16 174	4 051	3 144	3 461
2-3	26 875	0	26 875	16 325	0	16 325	4 051	6 499	5 466
3-4	4 317	4 245	72	279	16 421	16 699	3 034	1 004	601
4-5	0	0	0	0	[a]15 193	[a]15 193	1 055	[a]-1 055	[a]-1 055
5-6	0	0	0	0	[a]15 298	[a]15 298	1 055	[a]-1 055	[a]-1 055
	111 785			54 263		168 239	29 046	25 489	

[a]Nominal values.

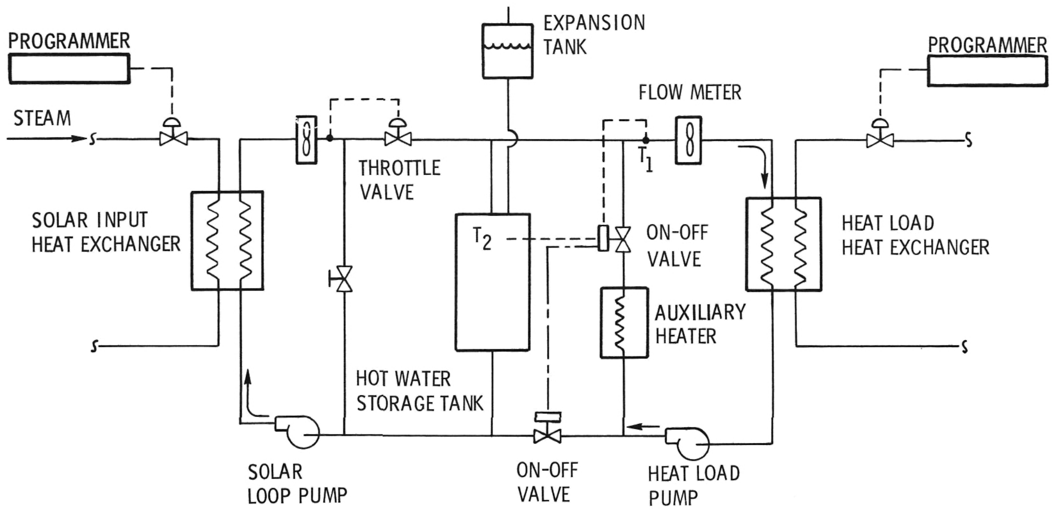


Figure 1. - Solar system model.

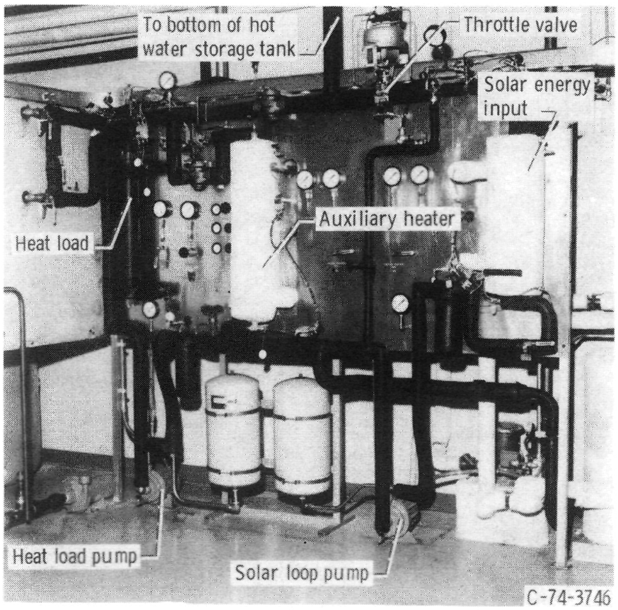


Figure 2. - Experimental solar heating and cooling system.

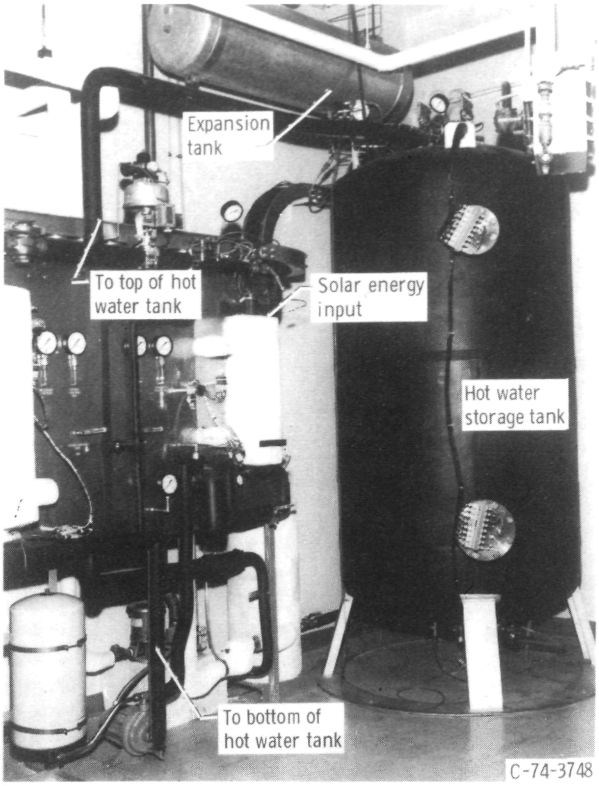


Figure 3. - Experimental solar system showing storage tank.

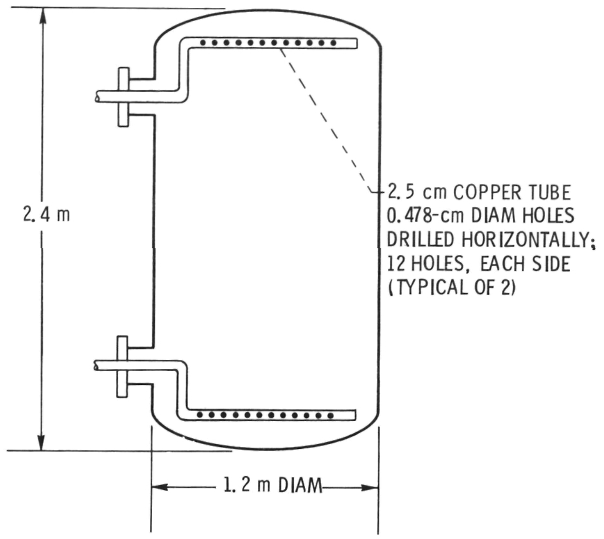


Figure 4. - Flow distribution manifold design.

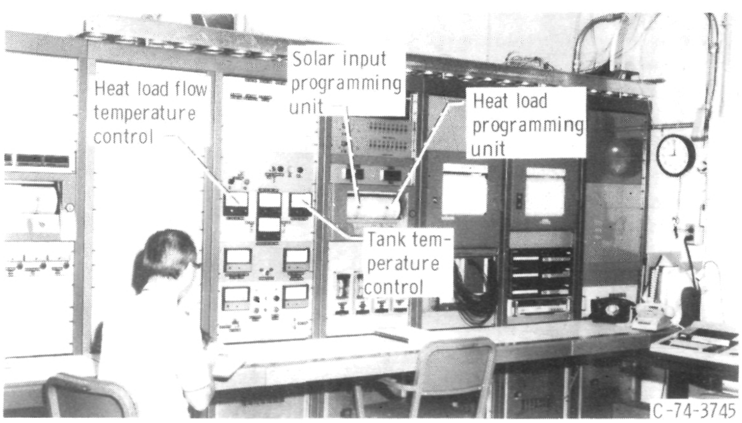
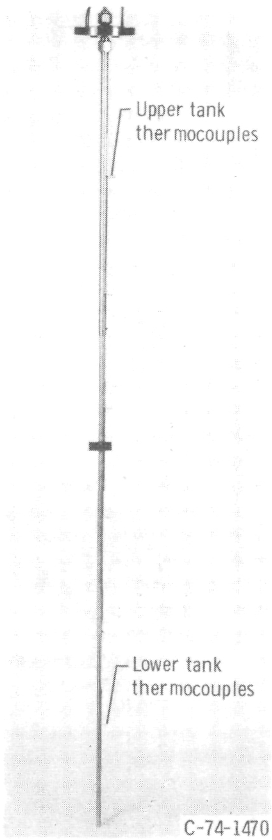
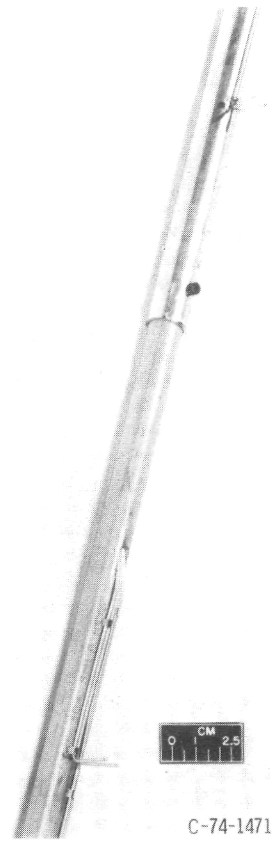


Figure 5. - Control panel of solar heating and cooling system.



(a) Overall.



(b) Closeup of two thermocouples.

Figure 6. - Temperature rake for hot water storage tank.

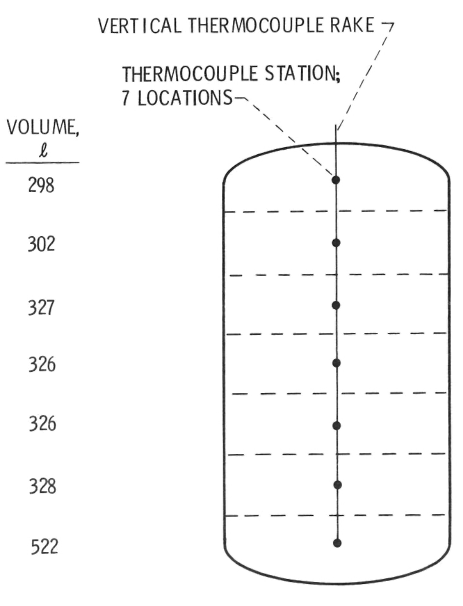


Figure 7. - Hot water storage tank. Volume increments assigned to each thermocouple location.

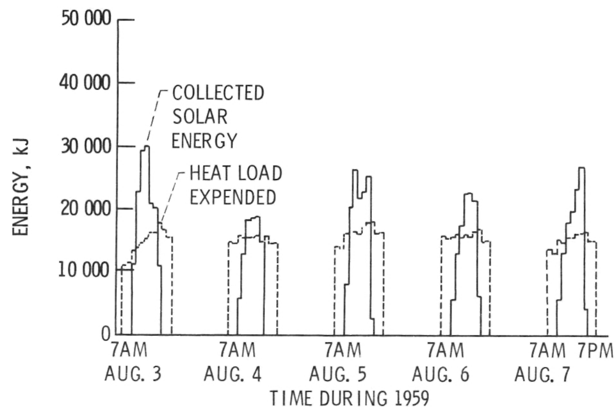


Figure 8. - Collector solar energy-heat load expenditure comparison during one week of operation.

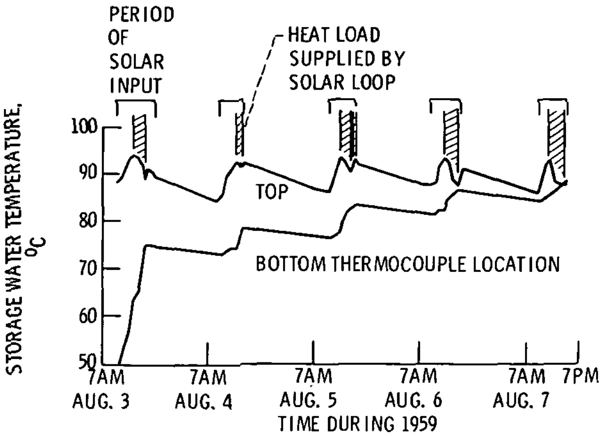


Figure 9. - Temperature variation of storage water during one week of operation.

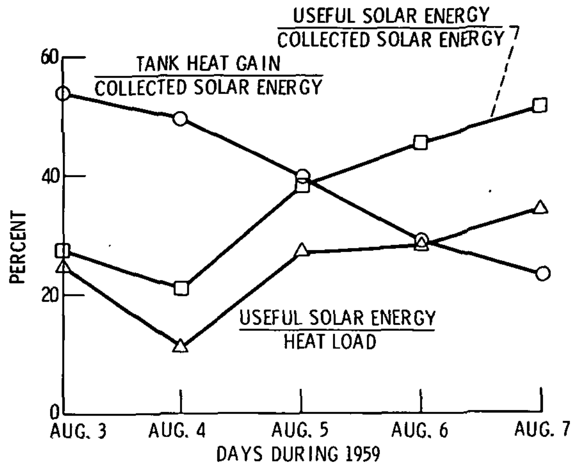


Figure 10. - Solar energy distribution during one week of operation.

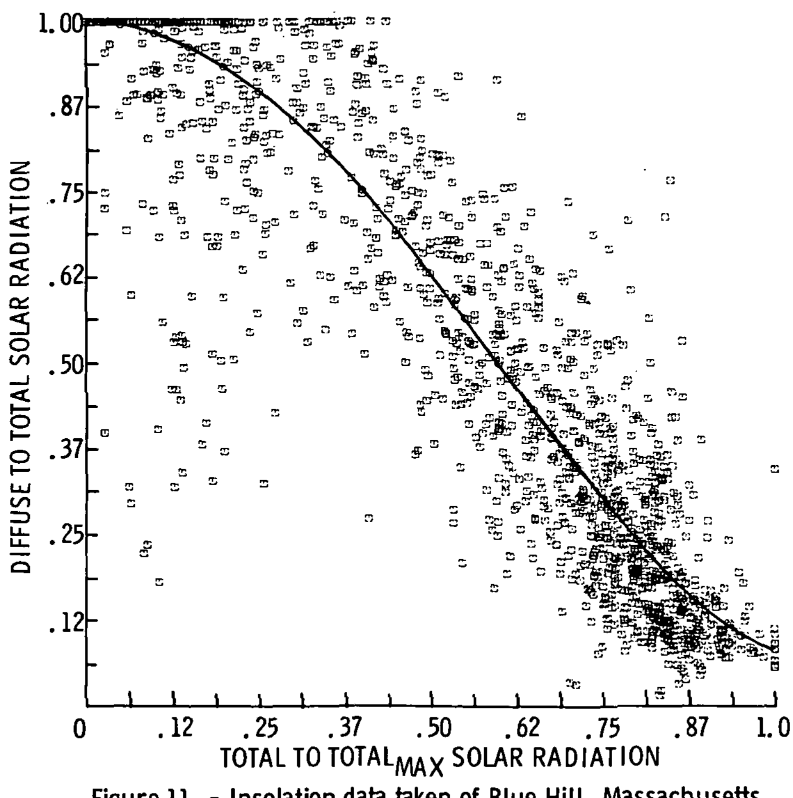


Figure 11. - Insolation data taken of Blue Hill, Massachusetts, from August 1952 to August 1956.

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