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TITLE: THE SOLAR LOAD RATIO METHOD APPLIED TO COMMERCIAL BUILDING ACTIVE SOLAR SYSTEM SIZING.

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THE SOLAR LOAD RATIO METHOD APPLIED TO COMMERCIAL BUILDING ACTIVE SOLAR SYSTEM SIZING

by

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INTRODUCTION

The design of an active solar energy system for a commercial building is an extremely complex process. The performance of the system will depend on a large number of variables, including the weather (solar radiation, dry bulb temperature, wind velocity, etc.); details of the building that affect the heating and cooling loads; the type of heating, ventilating, and air-conditioning (HVAC) system; the control strategy; and the capacity and performance of the solar components. The final, detailed design is best obtained by using a solar system simulation computer program such as those contained in DOE-2 [1],* BLAST [2], or TRNSYS [3], which perform hourly simulations for a design year. In the preliminary design stage, however, a much quicker and less expensive procedure is needed.

One method of providing results in a simplified manner is to perform hourly simulations for a variety of values of the important system parameters and to present correlations of the results. In 1977, simulations were performed at the Los Alamos Scientific Laboratory (LASL) for a Building Service Hot Water (BSHW) System and a combined space heating and hot water system using liquid collectors for a commercial building [4]. The results were presented in the form of solar load ratio curves based on the method of Balcomb and Hedstrom [5]. The simulation program used to produce those results calculated the loads in a somewhat approximate manner and did not have the capability to accurately model a variety of HVAC systems or control strategies.

The work described is a refinement and extension of the method and results reported in Ref. 4. The hourly simulation procedure that is used

*Numbers in brackets designate references listed at the end of the paper.

is the DOE-2 building energy analysis computer program. It is capable of calculating the loads and of simulating various control strategies in detail for both residential and commercial buildings and yet is computationally efficient enough to be used for extensive parametric studies. In addition to the two types of systems analyzed previously, a space heating system using an air collector is analyzed. A series of runs is made for systems using evacuated tube collectors for comparison to the flat-plate collectors previously considered, and the effects of additional system design parameters are investigated. Also, the generic collector types are characterized by standard efficiency curves, rather than by the detailed collector specifications used in the previous study.

THE DOE-2 COMPUTER PROGRAM

The primary tool used to analyze the systems considered here is the DOE-2 building energy analysis computer program. This program has been selected as the standard evaluation technique for the determination of design energy budgets under the Building Energy Performance Standards (BEPS). The simulation portion of DOE-2 consists of four programs that are normally run in sequence. The LOADS program calculates hourly heating and cooling loads for each zone of a building based on a fixed-zone air temperature. The input required by LOADS includes a detailed description of the building (orientation; geometry; and material properties of walls, floors, windows, ceilings; etc.); internal loads from lights, people, and equipment; and hourly weather data. ASHRAE algorithms [6] are used to calculate heat gains and losses through walls, roofs, floors, windows, and doors. Transient effects in massive walls are accounted for using response factors. The results of the LOADS program are written to a file that can be accessed by the SYSTEMS program.

The SYSTEMS program contains algorithms for simulating the performance of the secondary HVAC equipment (ducts, fans, coils, controls, etc.) used to control the temperature and humidity of each zone within the building. It uses air temperature weighting factors and the heat gains calculated by LOADS to determine the actual air temperature variations in each zone. The output information from LOADS and a list of user-defined system characteristics (flow rates, thermostat settings, schedules of equipment operation, and temperature setback schedules) are used to calculate the hour-by-hour energy requirements of the secondary HVAC system. Those results are printed on an output file for later use by the PLANT program.

The PLANT program contains algorithms that are used to calculate the performance of the primary energy conversion equipment. The operation of each plant component (boiler, chiller, hot water storage tank, solar collector, etc.) is modeled on the basis of operating conditions and part-load performance characteristics. The solar system is modeled by the component based simulator (CBS). CBS allows the user to select a variety of components (collector, heat exchangers, pumps, storage tanks, rock beds, etc.) and interconnect them in a variety of ways.* The PLANT program uses hourly results from the LOADS and SYSTEMS programs and instructions for equipment operating schedules to compute the electrical, thermal, and solar energy used by the building.

THE SIMULATIONS AND RESULTING CORRELATIONS

Because the number of design parameters is quite large, it is necessary to limit the number that are varied in the parametric studies. A "standard liquid system" is therefore defined by specifying values for those parameters that do not significantly affect the results. In addition, parameters such as the collector tilt, collector flow rate, and storage tank volumes are assigned values that were found to be optimum in previous parametric studies of residential systems [7, 8]. Values of these parameters are given in Table 1.

The design parameters that were varied include the hot water supply temperature, collector area, and collector type. The weather data used were Typical Meteorological Year (TMY) data for Bismarck, North Dakota; Fresno, California; Madison, Wisconsin; Medford, Oregon; Miami, Florida; Nashville, Tennessee; and New York, New York. These cities were chosen to provide a representative range of weather conditions for the US.

*CBS is modeled after TRNSYS, the solar energy system simulation program developed at the University of Wisconsin [3].

TABLE 1

THE STANDARD LIQUID SYSTEM

Parameter	Nominal Value
SOLAR COLLECTOR	
Orientation	Due south
Tilt (from horizontal)	Latitude + 10°
*Coolant flow rate	112.6 kg/h per m^2 of collector area (0.046 gpm per ft^2 of collector area)
HEAT EXCHANGER	
Effectiveness	0.70
Cold side flow rate	112.6 kg/h per m^2 of collector area (0.046 gpm per ft^2 of collector area)
STORAGE TANKS	
Capacity	73.4 kg of water per m^2 of collector area (15 lb. of water per ft^2 of collector area)
Height to diameter ratio	3.0
Loss coefficient	0.28 W/m^2-K (0.05 Btu/h-ft^2 F)
Environment temperature	21.1°C (70°F)**
Cold water supply temperature (to BSHW system)	15.6°C (60°F)

*Water, water/glycol, or nonaqueous collector coolant could be used.

**Assumes storage losses do not contribute to meeting the heating load.

The collector type is designated by its efficiency curve specified by

$$E = A + Bx + Cx^2, \tag{1}$$

where

$$x = (T_f - T_a)/I. \tag{2}$$

The collector coefficients, A, B, and C, depend on the specific collector and whether one uses an average or inlet fluid temperature for T_f. The symbols T_a and I correspond to the ambient temperature (C) and the total solar radiation (W/m^2). The collector coefficients used here were intended to correspond to typical collectors of four generic types. These coefficients were based on a study of performance data supplied by manufacturers of a large number of different collectors. The coefficients selected for the collector types used in this study are given in Table 2.

Hourly simulations were run for various combinations of the design parameters using the DOE-2 program. A range of collector areas was used, resulting in a corresponding range of solar load ratio values. Results were computed in the form of both monthly and annual solar fractions versus solar load ratios. The solar fraction is the fraction of the load (total energy required for water heating or combined water and space heating) that is supplied by solar. The Solar Load Ratio (SLR) is the ratio of total solar energy incident on the collector to the load.

The BSHW System

The first system studied was the BSHW system shown in Fig. 1. A heat exchanger separates the collector coolant from the storage water, and the solar heated storage tank acts as a preheater for a conventional hot water tank. Thus the system is classified as a double tank, indirect system. A control scheme provides auxiliary heat as needed to maintain the delivery water temperature at a specified level. The demand schedule (see Fig. 2) is a typical hot water profile for an office building based on ASHRAE data [9]. Hourly simulations were performed for various combinations of design parameters using the CBS part of the PLANT program.

TABLE 2

EFFICIENCY CURVE COEFFICIENTS USED TO CHARACTERIZE THE GENERAL TYPES OF COLLECTORS

Type	Fluid	A	B		C	
			$(W/m^2\text{-}K)$	$(Btu/h\text{-}ft^2\text{-}°F)$	$(W/m^2\text{-}K)^2$	$(Btu/h\text{-}ft^2\text{-}°F)^2$
Single-glazed, selective	LIQ	0.705	-5.04	-0.887	0.	0.
Single-glazed, nonselective	LIQ	0.780	-7.50	-1.32	0.	0.
Double-glazed, nonselective	LIQ	0.643	-5.00	-0.880	0.	0.
Evacuated tube, single-glazed	LIQ	0.642	-0.343	-0.0604	-3.475	-0.1076
Single-glazed, selective	AIR	0.55	-4.89	-0.86	0.	0.
Single-glazed, nonselective	AIR	0.59	-6.25	-1.1	0.	0.
Double-glazed, nonselective	AIR	0.475	-4.15	-0.73	0.	0.

The fluid temperature used in Eq. (2) is the inlet temperature except for the evacuated tube case for which the average fluid temperature is used.

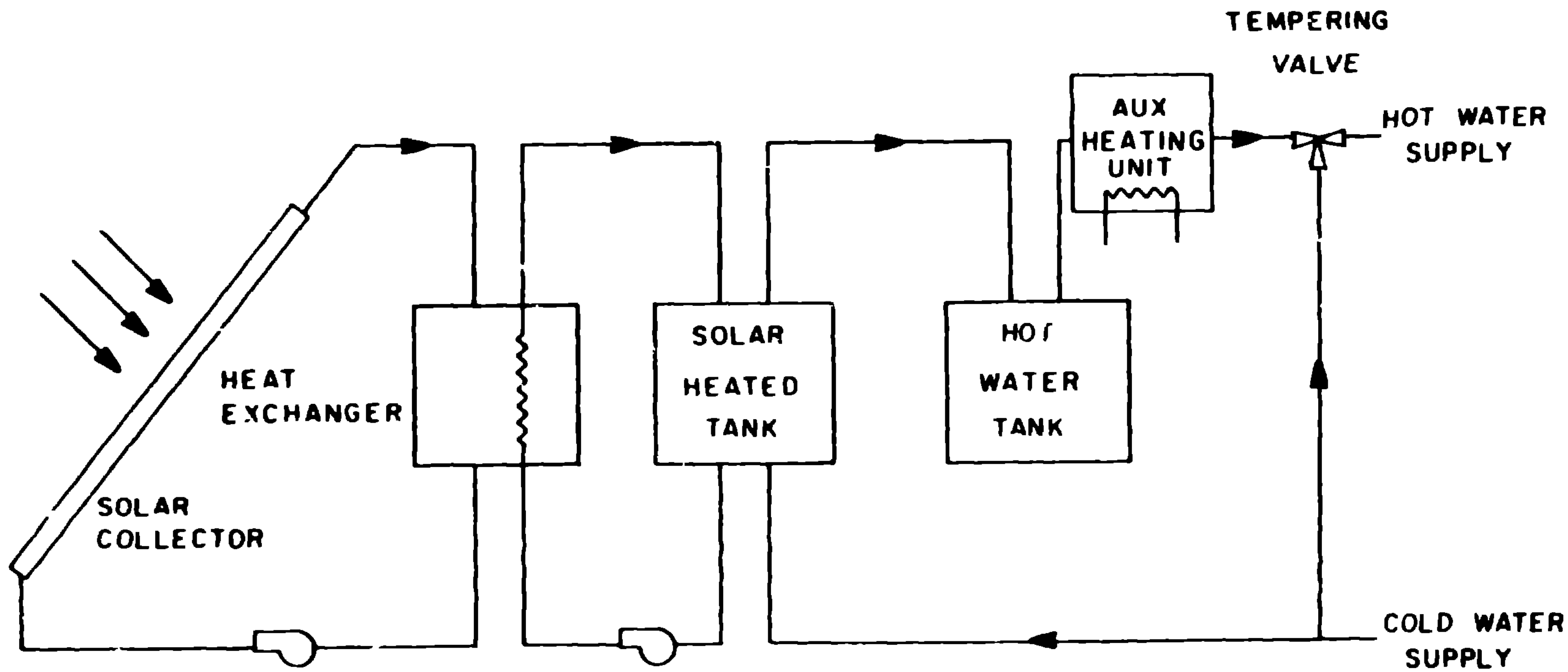

Fig. 1. Schematic diagram of the BSHW system.

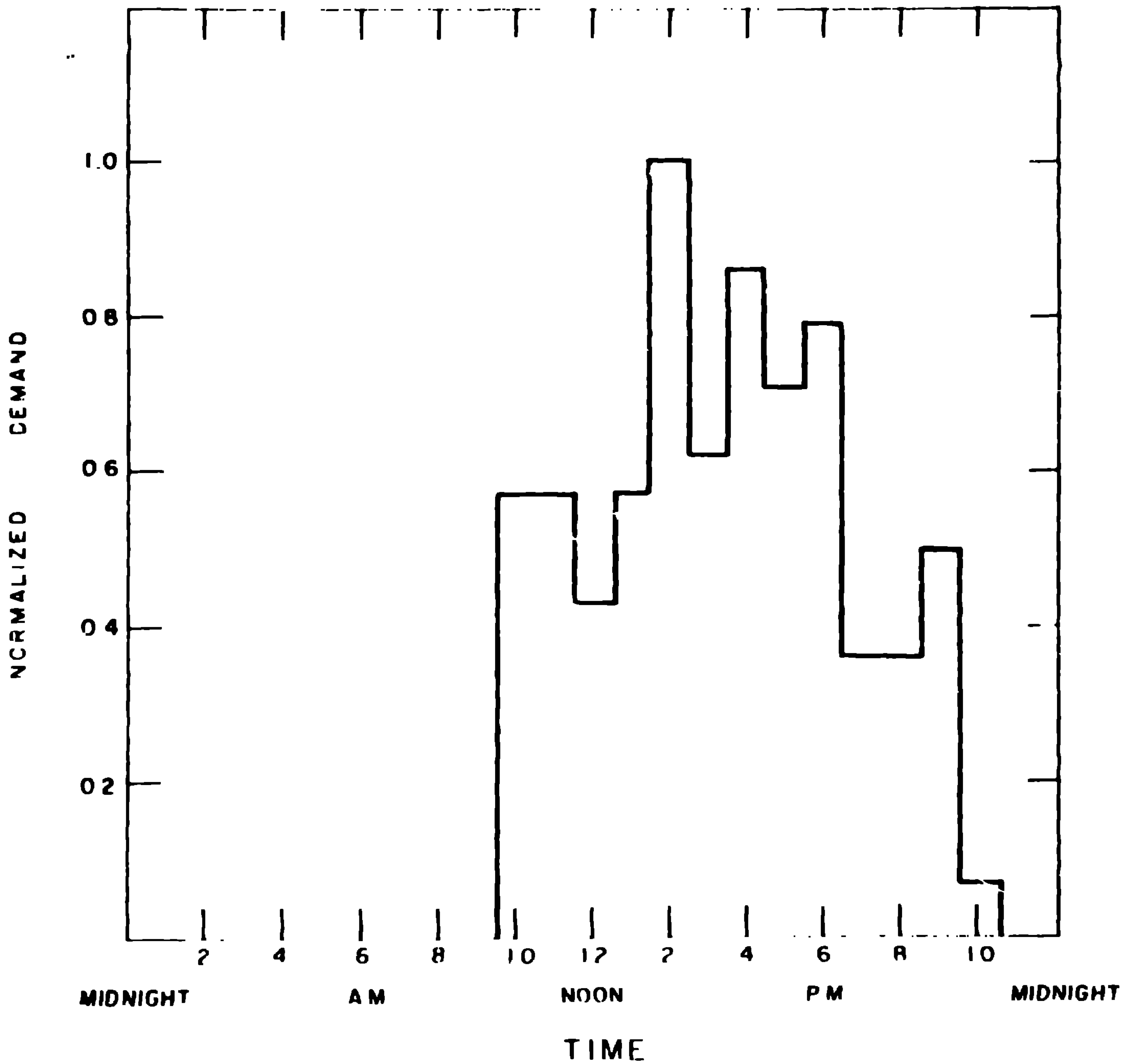

Fig. 2. RCHW use profile.

Typical results showing the effect of location on BSHW system performance are shown in Fig. 3. Because of the uniform load characteristics thoughout the year, the performance of BSHW systems is best correlated using annual load parameters. The performance is better in warmer climates because the ambient temperature is closer to the collector temperature, with resulting smaller heat losses and higher collector efficiencies. Values of heating degree days for the cities are shown in the figure. The curves shift downward with increasing degree days and therefore higher losses. It appears that results for locations other than those considered here can be obtained by characterizing the location by its number of heating degree days. A set of design curves, recommended for various ranges of heating degree days, is given in Fig. 4 for a hot water supply temperature of 54°C (130°F) and single-glazed, selective-surface collector.

The next parameter investigated was the hot water supply temperature. A series of runs was made for a single-glazed, selective surface collector using water as the coolant in New York City. Results are shown in Fig. 5. As expected, a lower supply temperature results in lower collector operating temperatures and higher efficiencies. This has a significant effect on overall system performance as represented by the solar fraction.

The effect of collector type on system performance is shown in Fig. 6. Note that the difference in performance for the three generic collectors considered is relatively minor.

These BSHW system performance curves differ from the earlier results presented in Ref. 4 in two key respects. First, it has been found that a better correlation is obtained on an annual rather than a monthly basis because of the uniform load. This makes the curves considerably easier to use. Second, the considerable scatter in the data that occurred with geographical location was obscured in the earlier study. A more accurate correlation is obtained by presenting a family of curves for varying degree-day locations, rather than the single performance curve presented in Ref. 4.

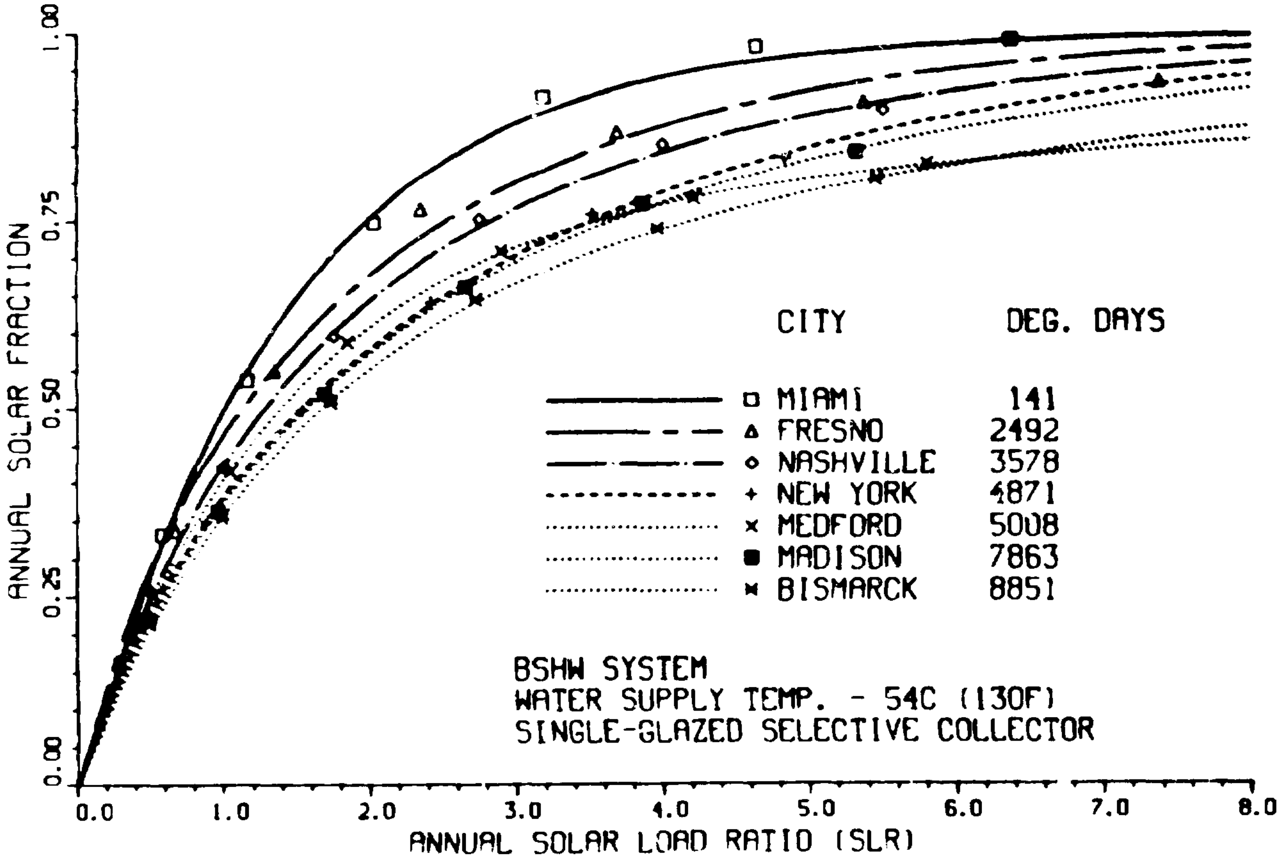

Fig. 3. The effect of location on the performance of the BSHW system.

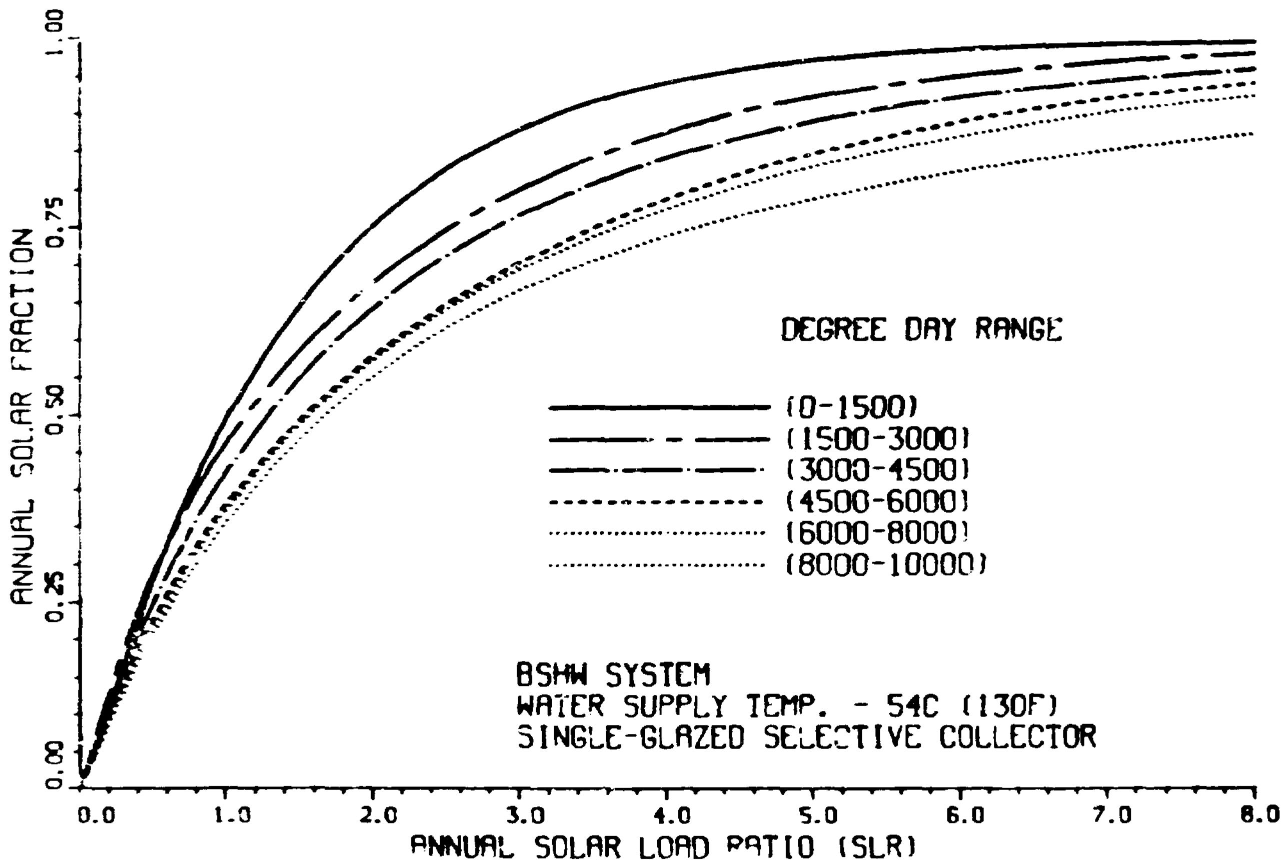

Fig. 4. Design curves for the BSHW system for various degree-day ranges.

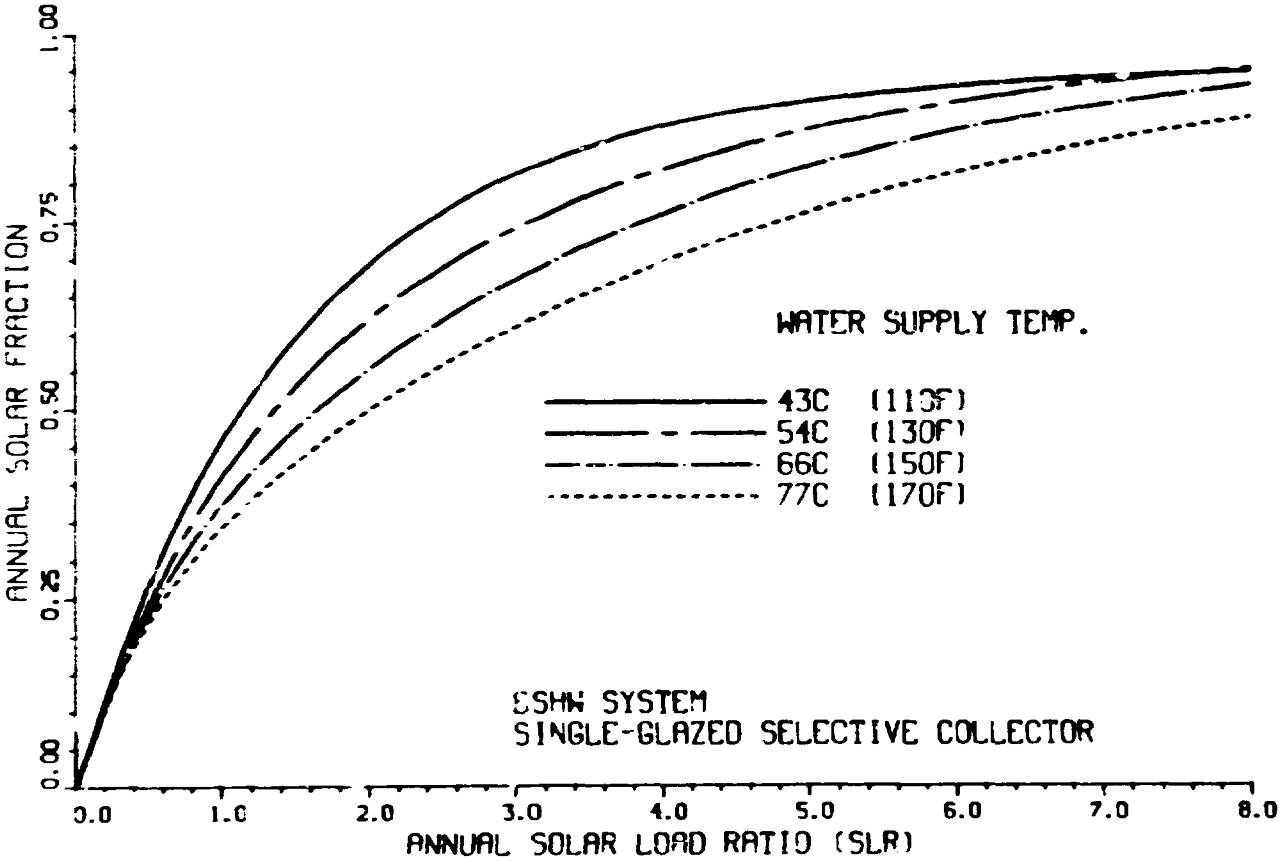

Fig. 5. The effect of water supply temperature on the performance of the BSHW system.

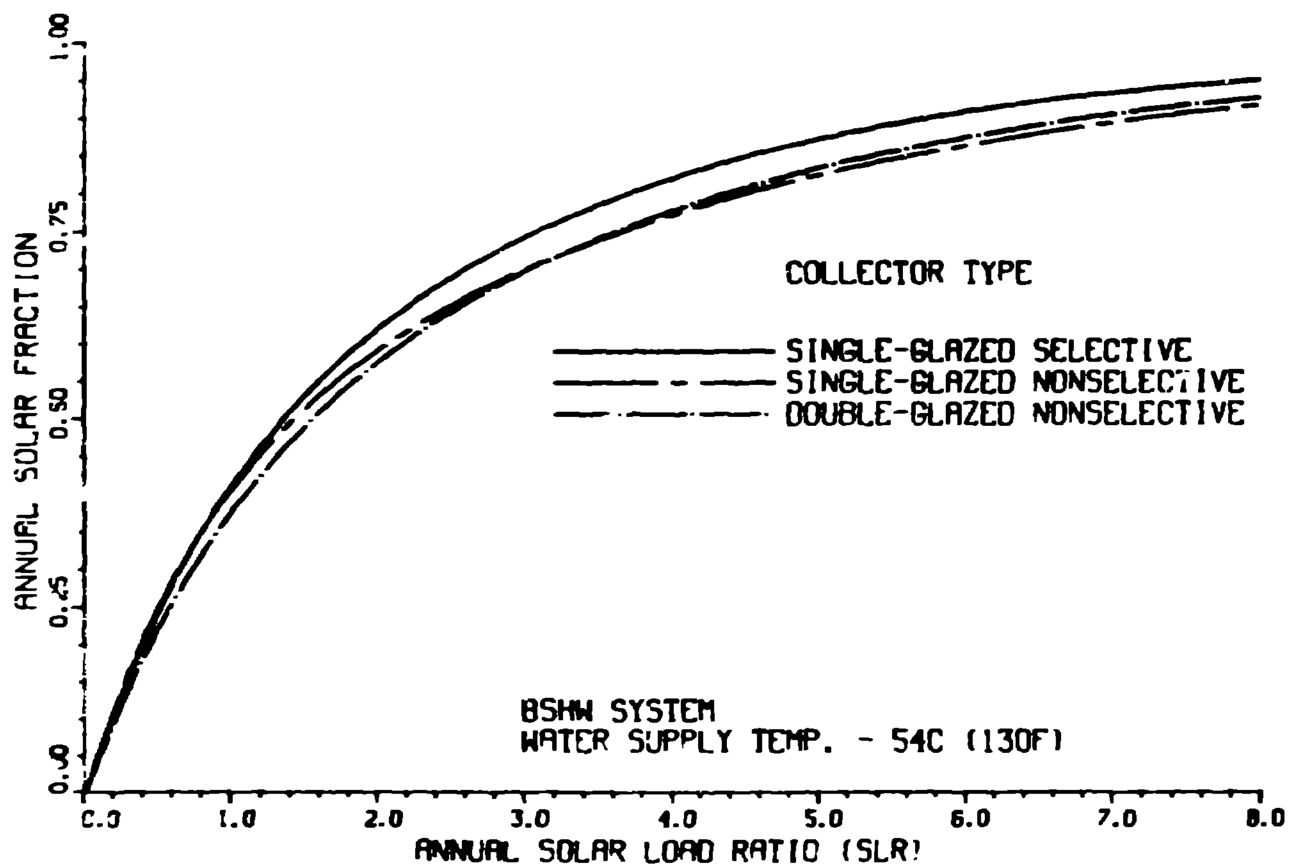

Fig. 6. The effect of collector type on the performance of the BSHW system.

Design Procedure for the BSHW System

A BSHW system using a flat-plate collector may be sized by using the following procedure. A sample problem using this recommended procedure is given in the Appendix.

1. Given the usage schedule, the cold water temperature, and the hot water supply temperature, calculate the annual BSHW load.

2. Determine the solar radiation incident on the collector at a given geographic location and having a specified tilt. These data are available in Refs. 4 and 10. Methods for converting total horizontal radiation to tilted surface radiation are given in Ref. 4.

3. Select a value of collector area. Multiply the area by the result of Step 2 to determine annual solar radiation incident on the tilted collector. Divide the annual insolation by the load to obtain the annual SLR.

4. Use Fig. 4 to determine the solar fraction for the specified location (degree-day range) and the SLR from Step 3.

5. Correct for hot water supply temperatures other than 54°C (130°F) using Fig. 5. Measure the vertical distance (solar fraction correction) between the curve being used and the curve for 54°C (130°F) at the given SLR. The correction will be positive for temperatures less than 54°C and negative for higher temperatures.

6. Correct for the collector type using Fig. 6. The uncorrected value was obtained for a single glazed selective surface collector having an efficiency curve with an intercept of 0.705 and slope of 5.04 W/m^2·K (0.887 Btu/h ft^2·F). If the efficiency curve of the collector is known, compare with the cases available in Fig. 6 and interpolate or extrapolate to make the correction.*

*It is recognized that this procedure is approximate and requires some judgment on the part of the user. The corrections are usually small, so the effect of some error in this step will not greatly affect overall accuracy.

7. Add the corrections obtained in Steps 5 and 6 to the solar fraction obtained in Step 4.

8. Repeat Steps 3 through 8 for other collector areas to determine an optimum collector size. This generally is done on a life-cycle cost basis (see Ref. 4).

The case of the evacuated tube collector must be treated separately because the collector efficiency curve has a much smaller slope and because operation at higher temperatures does not result in such a large drop in collector performance. The effect of geographic location for the evacuated tube collector is shown in Fig. 7 for a delivery temperature of 54°C (130°F). The effect of water supply temperature is shown in Fig. 8 for New York City. An approximate design for this type of collector can be obtained by determining the solar fraction from Fig. 7 for the specified location and SLR, and adding a supply temperature correction determined from Fig. 8. The procedure is the same as discussed above except for the elimination of Step 6.

This procedure has been tested for a wide variety of cases by comparing the results to those obtained by direct hour-by-hour calculations using DOE-2. Differences in the solar fractions were less than 7 per cent for all cases and were usually within the readability of the graphs.

The Space Heating Liquid System (SHLS)

The space heating liquid system provides both space heating and service hot water for a commercial building using solar collectors with a liquid coolant. A schematic diagram of the system studied is given in Fig. 9. This case is much more complex because the space heating load depends on numerous details of construction of the building, as well as the type of HVAC system and control system used.

The building selected as the basis for this analysis is a medium sized office building at the Los Alamos Scientific Laboratory in Los Alamos, New Mexico. It is a three story, double loaded corridor building having 1542 m^2 (16,600 ft^2) of floor space. It consists of offices, a computer room, a small unfinished space, rest rooms, hallways, and stairwells. Internal loads are primarily from people, lighting, and the computer

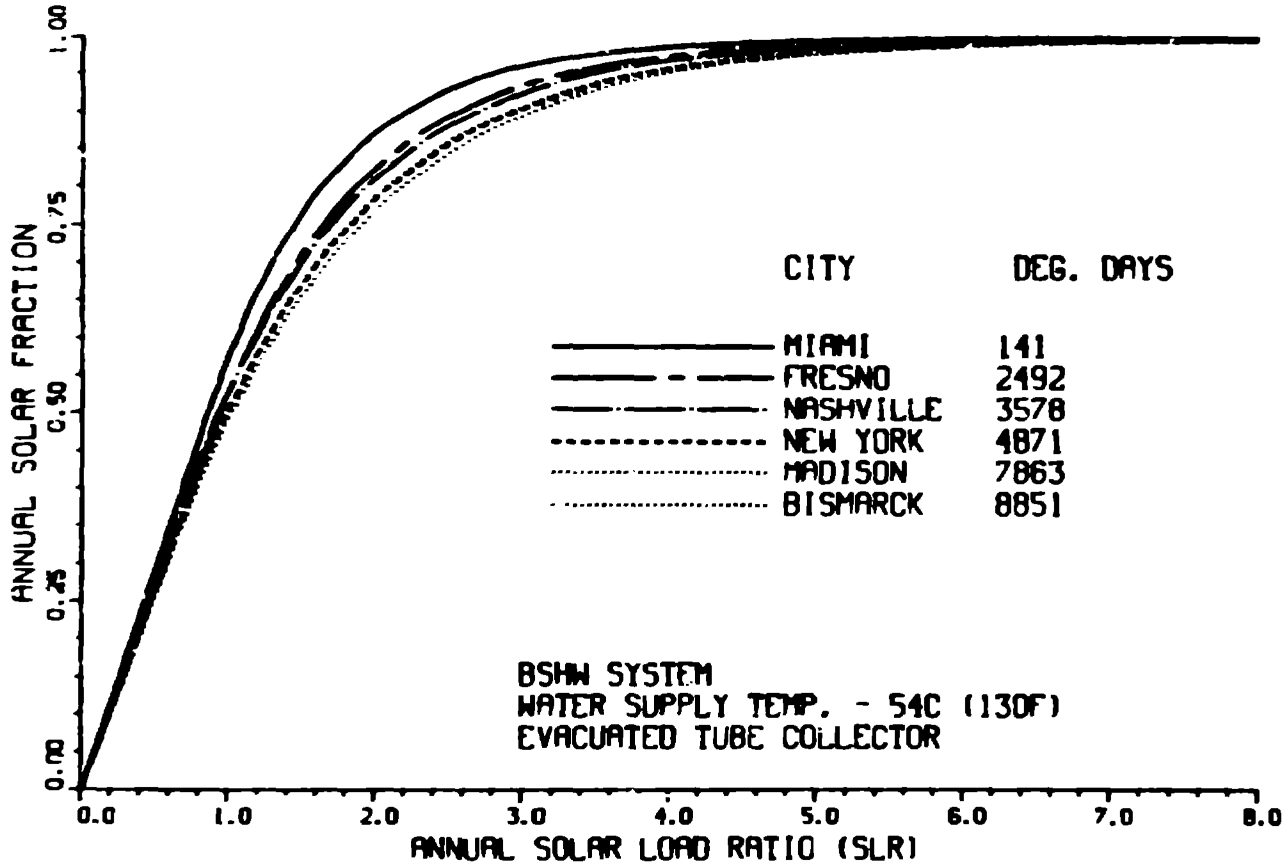

Fig. 7. The effect of location on the performance of a BSHW system using an evacuated tube collector.

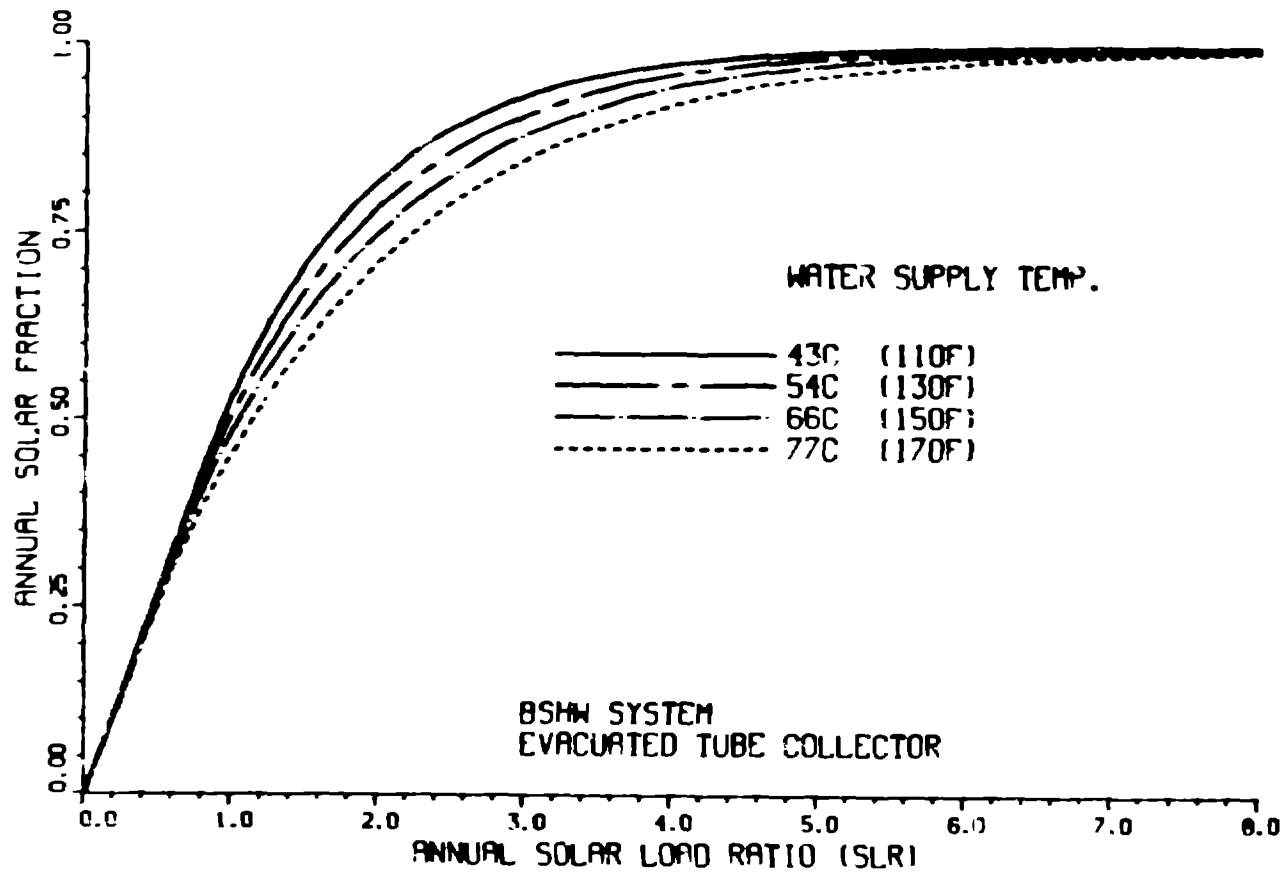

Fig. 8. The effect of water supply temperature on the performance of a BSHW system using an evacuated tube collector.

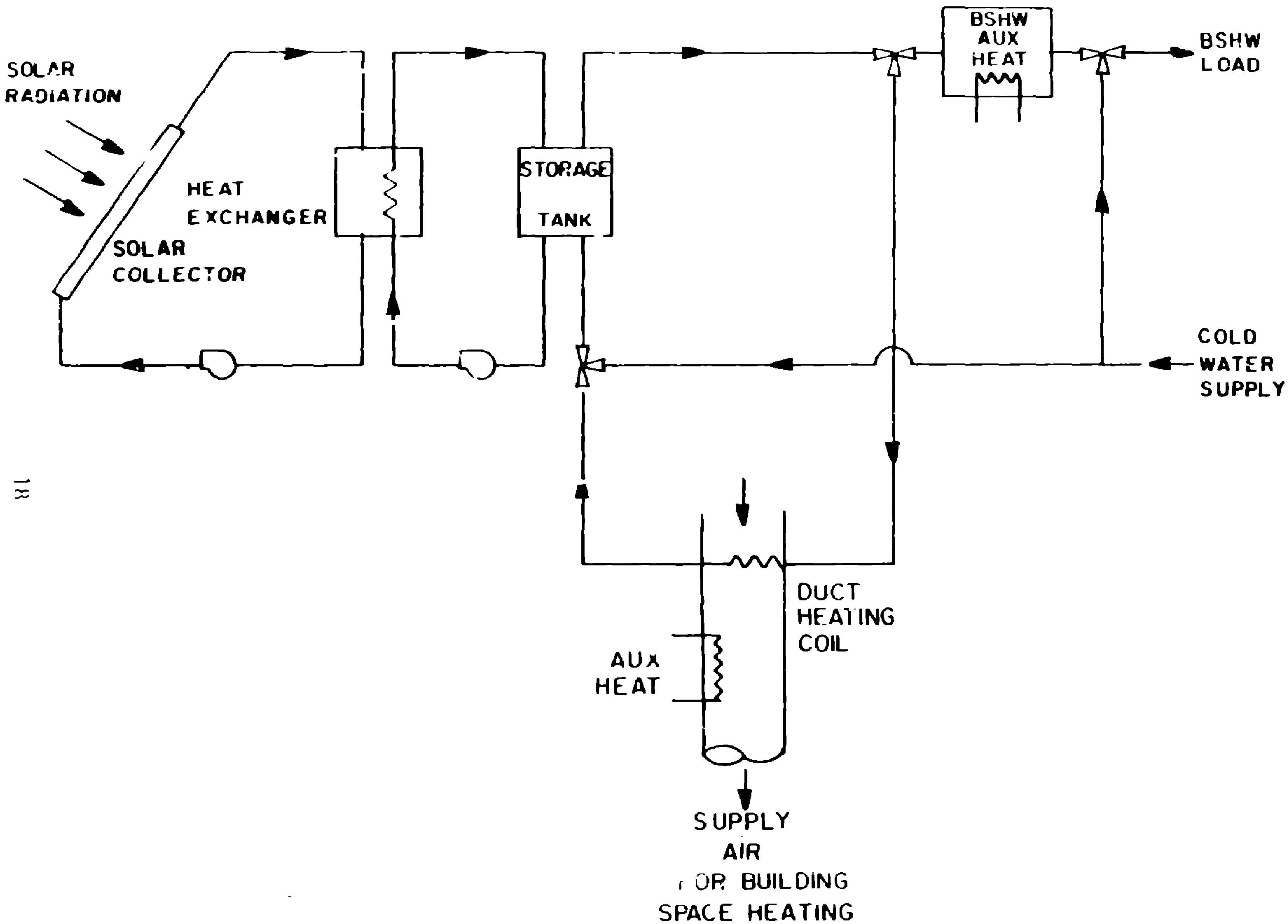

Fig. 3. Schematic diagram of the SHLS.

equipment. The HVAC system is a multizone system with a proportional thermostat having a 2.2°C (4°F) throttling range and setpoints of 20.0°C (68°F) in winter and 25.6°C (78°F) in summer. Hot water for space heating is supplied to the duct heating coil according to an outside air schedule. The BSHW demand schedule was obtained by multiplying values in the normalized schedule of Fig. 2 by a usage factor of 107 kg/h (0.47 gpm). The geometry and loss coefficient for the storage tank, the flow rates and effectiveness of the heat exchanger, and the orientation and tilt of the collector are identical to those used for the standard liquid system (Table 1).

An additional parameter that must be specified is the minimum water supply temperature for the main heating coil. A series of preliminary runs was made to determine the effect of that temperature on the solar fraction. It was found that the solar fraction was relatively insensitive to changes in the coil temperature for values less than 37.8 C (100°F), but began to drop sharply as the coil supply temperature was increased above that level. Because a 37.8°C (100°F) minimum coil temperature is typical, this value was selected for all subsequent runs.

Analysis of this system was carried out using a two step procedure. First, a LOADS and SYSTEMS run was made for a given location and a particular HVAC system. (The sensitivities of the results to building type and HVAC system are discussed below.) Resulting heating and cooling loads and energy requirements were stored in a file. The second step was a series of parametric runs of the PLANT program (including analysis of the solar equipment) to determine the solar system performance for various collector types, collector sizes, etc. The output file from the LOADS and SYSTEMS run was used as input to the PLANT program.

Monthly loads for the SHLS case vary widely because little, if any, space heating is required in the summer months. The collector array is oversized for that season, the solar fraction is unity, and excess heat must be dumped. Thus the results are better correlated in terms of monthly, rather than annual, load parameters for the SHLS case.

The results of a series of runs used to determine the effect of location are shown in Fig. 10. Curves are presented for all the cities listed previously except Miami, Florida, which is not included because virtually no space heating is required there. In all cases, the BSHW supply water temperature was 54°C (130°F) and a single glazed, selective surface collector

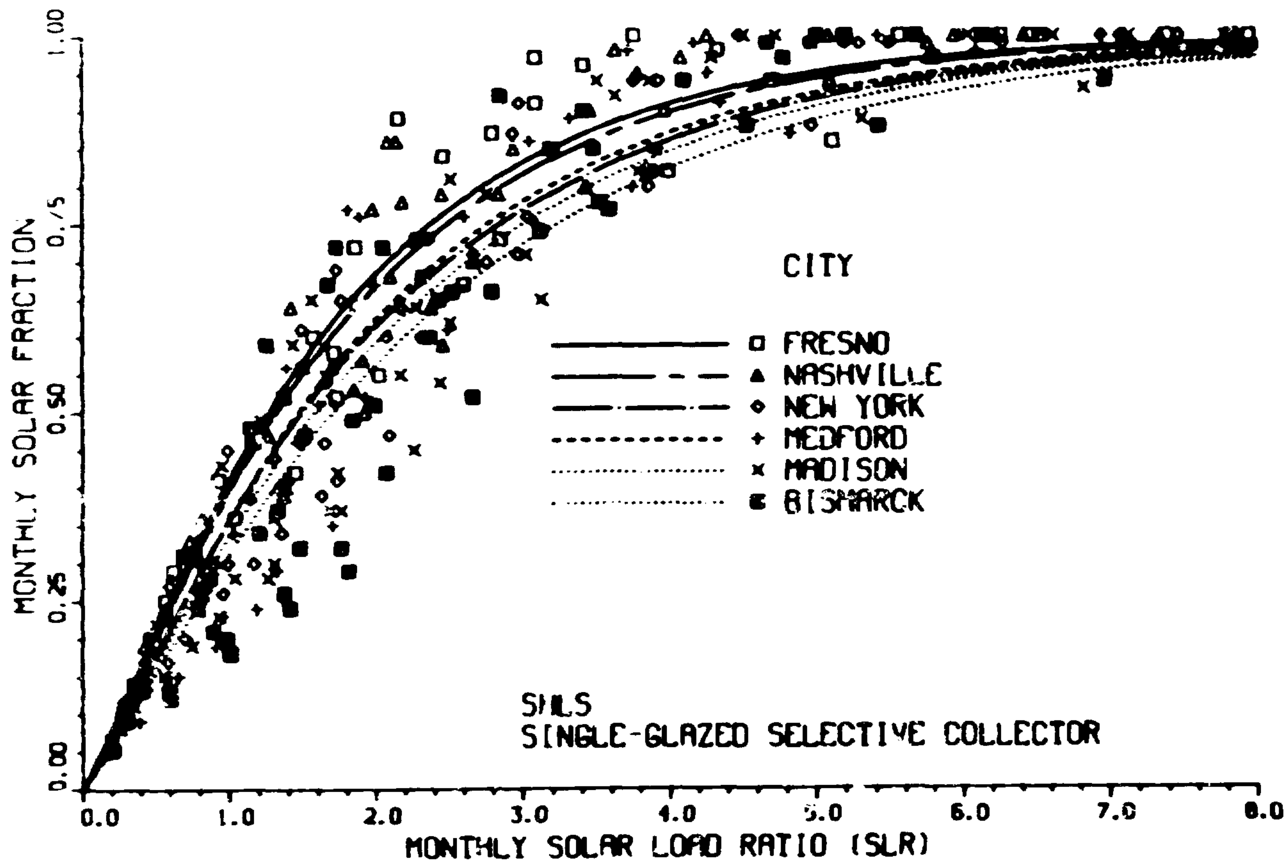

Fig. 10. The effect of location on the performance of the SHLS.

was used. A range of collector areas, from 279 m^2 (3000 ft^2) to 650 m^2 (7000 ft^2), was run to obtain an appropriate distribution of SLR values. The curves were obtained by a least-squares fit of an exponential curve with a straight line segment for low values of the SLR. Standard deviations of the data points from these curves ranged from 3 to 6 per cent. The solar fractions are again higher for warmer climates, but the variation with location is much less than for the BSHW system. This is because the collector losses vary with the load and are incorporated in the SLR. Because of the reduced data scatter compared w[illegible]e BSHW case, a single performance curve, for a given collector, adequ[illegible]esents all cities.

The effects c[illegible]e collector type are shown in Fig. 11 where the curve for each collector type represents the best fit for 216 data points representing 6 cities, 3 collector areas, and 12 months. The standard deviations between the monthly points and the curves are in the range of 4 to 7 per cent. Note that universal performance curves for each collector type apply to all locations.

The effects of several other design parameters were also investigated. The BSHW supply temperature was varied from 43 to 77 C (110 to 170 F) and the usage was varied between 0 and 227 kg/h (1.0 gpm). The effect of these changes on the performance curves was found to be negligible. This is mainly caused by the fact that the energy required for BSHW heating constituted a relatively small percentage (2 to 3 per cent for the cases considered here) of the total load.

The effect of the type of HVAC system was investigated by considering both a multizone system and a variable air volume system. For the same load the difference in energy requirements for these two systems was very large; the multizone system required approximately twice the energy of the variable air volume system. Despite this, differences in the performance curves for the two systems were less than 5 per cent.

The effect of the type of building was investigated by varying infiltration rates and internal energy generation rates caused by lights, people, and equipment. Although these changes caused large changes in the load, they did not cause a significant change in the performance curves.

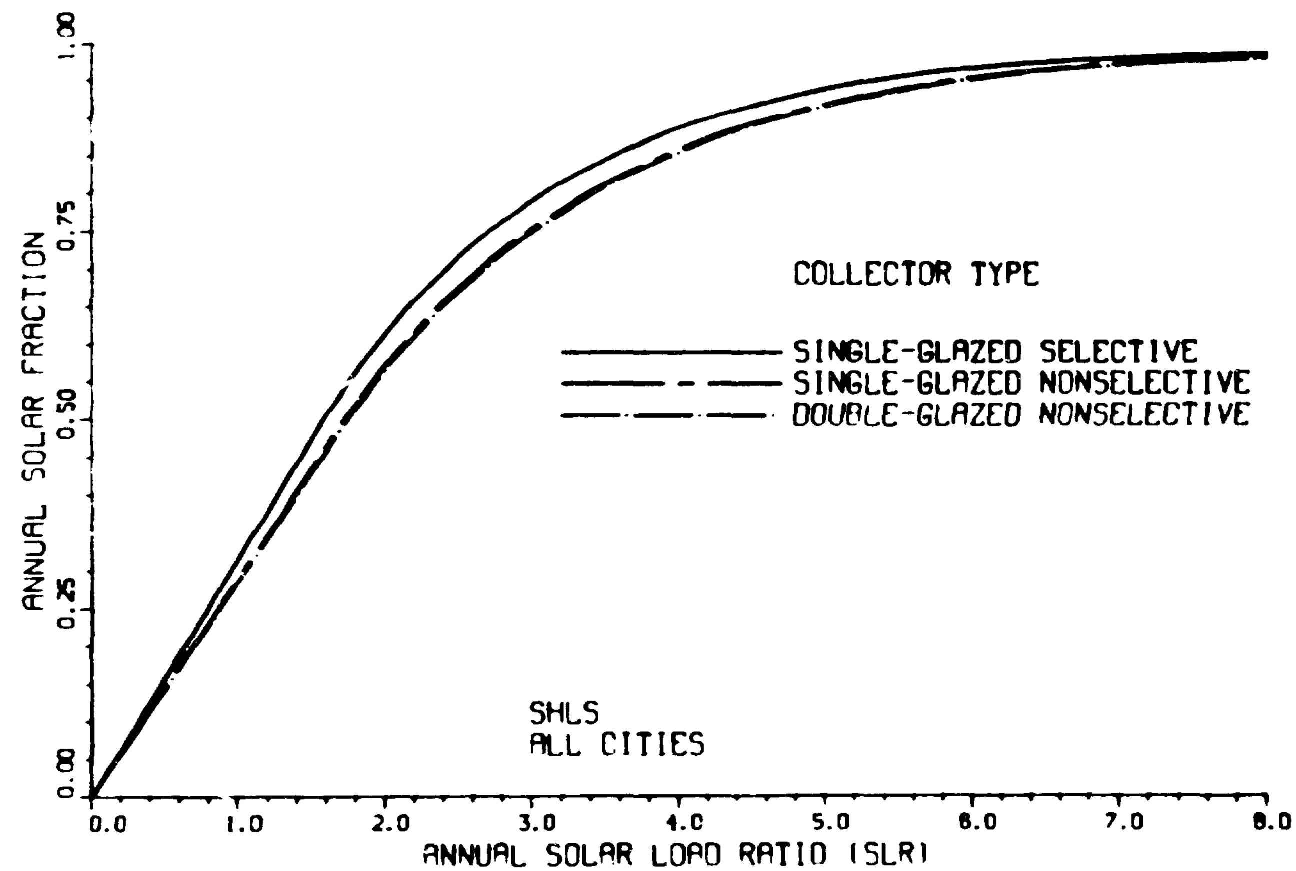

Fig. 11. Design curves for the SHLS for various collector types.

Therefore, it is recommended that the curves of Fig. 11 be used for the preliminary design and sizing of liquid-based solar systems for all commercial buildings in all locations. However, the following precautions should be observed.

- The results are not expected to be accurate if the BSHW load is greater than about 20 per cent of the total load.
- The results do not apply to buildings with passive solar energy features that provide a significant portion of the load.
- Heat losses in pipes and ducts have not been included in this analysis. If they are significant, suitable adjustments should be made to the results presented here.

The Space Heating Air System (SHAS)

Although the vast majority of solar systems in commercial buildings use a liquid as the collector coolant, a system using air collectors was also studied. A schematic diagram of a space heating air system (SHAS) used to provide both space heating and service hot water for a commercial building is shown in Fig. 12: the reference building used in this analysis is the same as for the SHLS. Supply air for space heating may come directly from the collector or the rock-bed storage unit. A heat exchanger transfers energy from the air to the hot water storage tank. A rock bed volume of 0.22 m^3 per m^2 of collector (0.72 ft^3 per ft^2 of collector) and a hot water storage mass of 3.67 kg of water per m^2 of collector (0.75 lb of water per ft^2 of collector) ~~is~~ are used. All other parameters are the same as for the SHLS.

Results obtained for the SHAS are quite similar to those for the SHLS. A comparison of the results of these two systems is shown in Fig. 13 for a single-glazed selective surface collector in New York City. The collectors that were selected with air as the coolant have slightly lower efficiencies than those using a liquid. This disadvantage is apparently more than offset by the fact that the air leaving the collector may be supplied directly to the zones without loss (for the system simulated here), while the liquid system has two heat exchangers with an effectiveness significantly lower than 100 per cent.

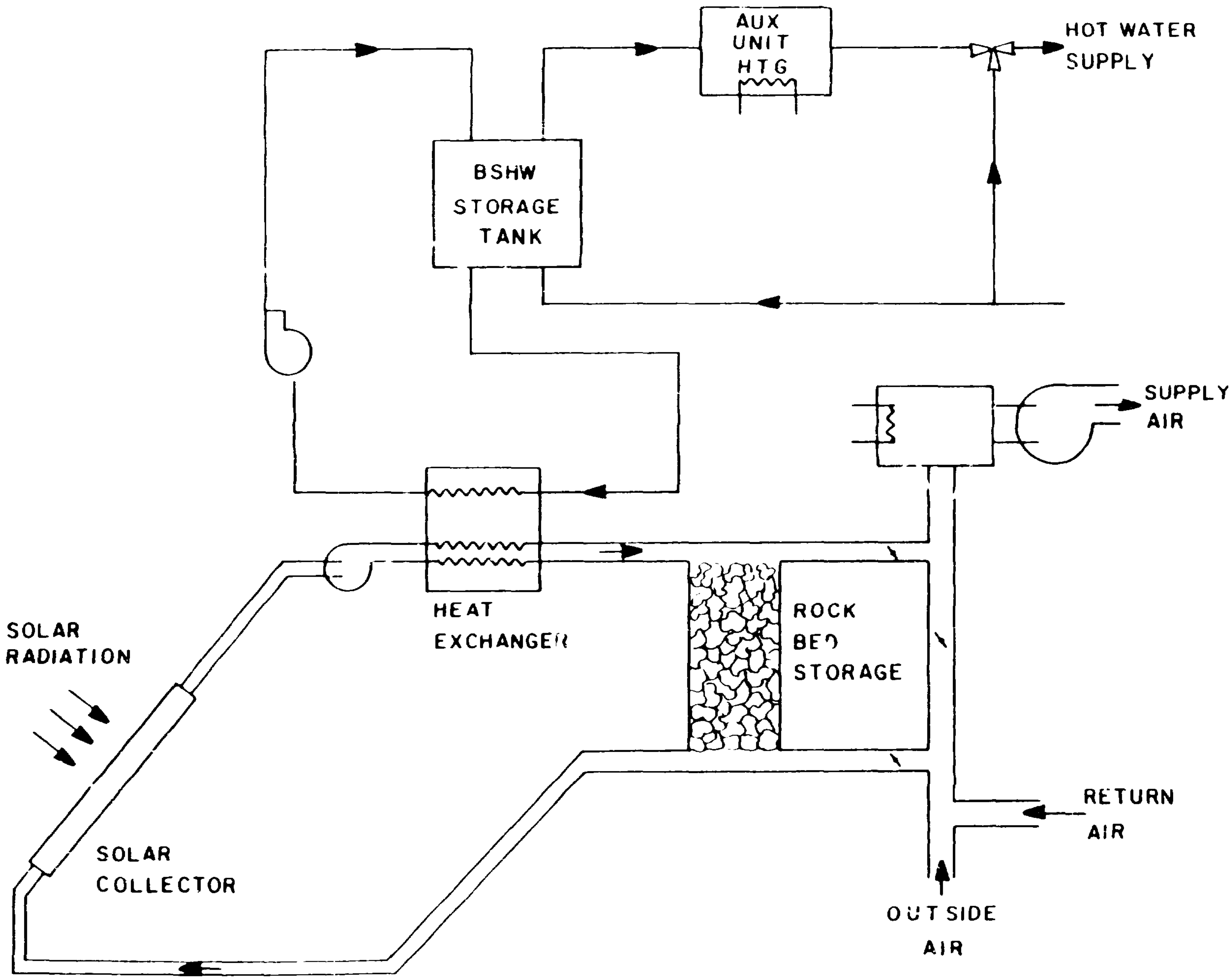

Fig. 12. Schematic diagram of the SHAS.

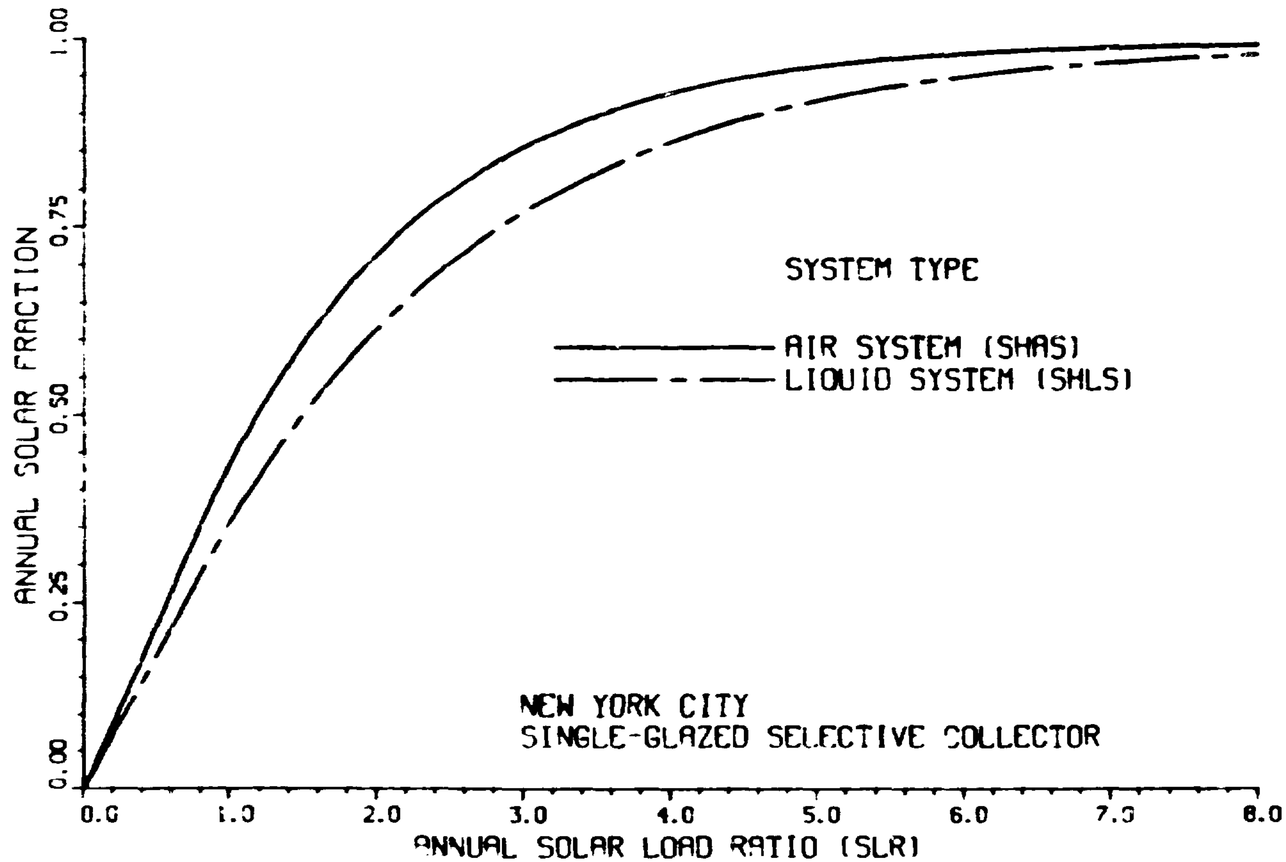

Fig. 13. A comparison of results for the SHLS and SHAS.

Results for the three types of air collectors* listed in Table 2 are shown in Fig. 14. These may be considered universal design curves and may be used for the preliminary design of air heating systems for commercial buildings.

For the SHAS, the effects of BSHW demand and supply temperature on the solar system performance were found to be minor.

CONCLUSIONS

The DOE-2 building energy analysis computer program, with its active solar system simulator (CBS), has been an effective analysis tool for predicting the performance of a wide range of BSHW and combined space heating/BSHW systems in the commercial building environment. A series of parametric runs for these two system classifications has resulted in universal performance curve correlations, on an annual basis, for four generic collector types. Performance sensitivities to the BSHW system design parameters of supply temperature, geographic location, and collector type are presented in a set of universal performance curves.

For combined space heating/BSHW systems, a common set of performance curves for all locations, on a monthly basis, ~~have~~ has been developed for four generic collector types. Sets of curves were developed for both liquid and air heating systems. Sensitivities of system performance to BSHW supply temperature, HVAC system type, and building type have been determined.

The universal performance curves developed here provide a simplified sizing procedure for active solar heating systems in commercial buildings.

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*Evacuated tube collectors using air as the coolant are rarely used and are therefore not shown.

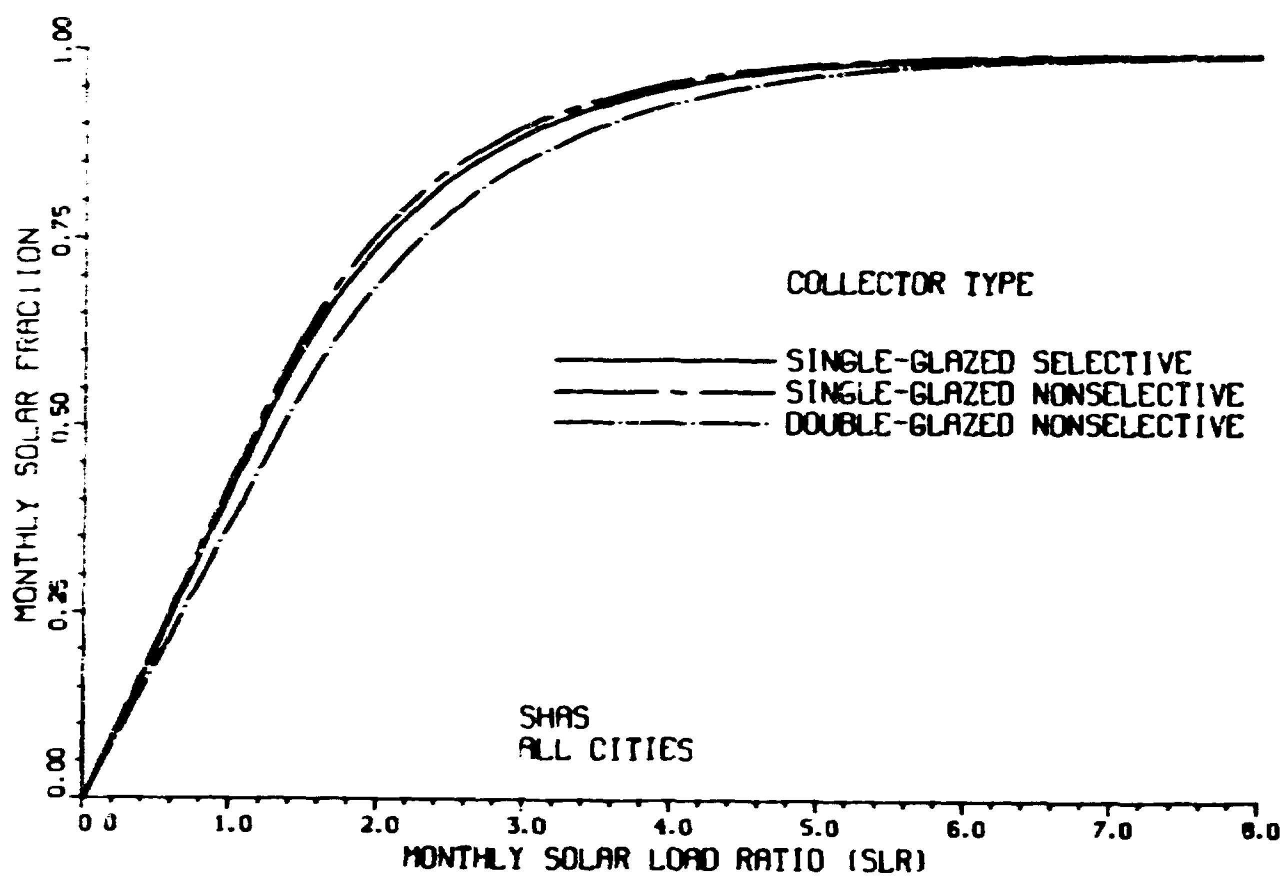

Fig. 14. Design curves for the SHAS for various collector types.

ACKNOWLEDGMENTS

The authors wish to acknowledge the contributions of L. D. Fought in performing the calculations and of J. D. Balcomb and J. C. Hedstrom for helpful discussions.

APPENDIX

Sample Problem

Determine the solar fraction supplied by an array of single-glazed nonselective collectors having an area of 200 m^2 for a BSHW system for a commercial building in Bismarck, North Dakota. The water supply temperature is 77°C (170°F), and the usage is 10,000 kg/day for weekdays.

SOLUTION

The annual load, assuming 260 weekdays, is

$$Q = M\,C_p\,\Delta T = (260)(10,000)(4.1889)(77 - 16)$$
$$= 666,000 \text{ MJ } (631.4 \text{ MBtu}).$$

The city water temperature is assumed to be 16°C (60°F).

The annual insolation on a collector facing south and having a tilt of (Lat + 10°) for Bismarck is found from Table A-1, p. 92 of Ref. [4], to be 6836 MJ/m^2 (602,000 Btu/ft^2). The number of heating degree days is 4917 C-day (8851 F-days). The SLR is

$$SLR = (260)(6836)/(666,000) = 2.05.$$

From Fig. 4 for SLR = 2.05 and a water temperature of 77°C, the uncorrected solar fraction is 0.56. The correction for water supply temperature is obtained from Fig. 5 by determining the difference between the solar fraction for 77°C (170°F) and that for 54°C (130°F) at SLR = 2.05. The correction is 0.47 - 0.59 = -0.12. The correction for collector type is obtained from Fig. 6 by determining the difference between the solar fraction for the single-glazed nonselective collector and that of the reference case, the single-glazed selective type. This correction is 0.55 - 0.59 = -0.04. Therefore, the corrected solar fraction is 0.56 - 0.12 - 0.04 = 0.40.

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