SOLAR ENERGY DEHUMIDIFICATION EXPERIMENT

on the

CITICORP CENTER BUILDING

FINAL REPORT

Prepared for

NATIONAL SCIENCE FOUNDATION

Energy Laboratory
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ENERGY LABORATORY

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

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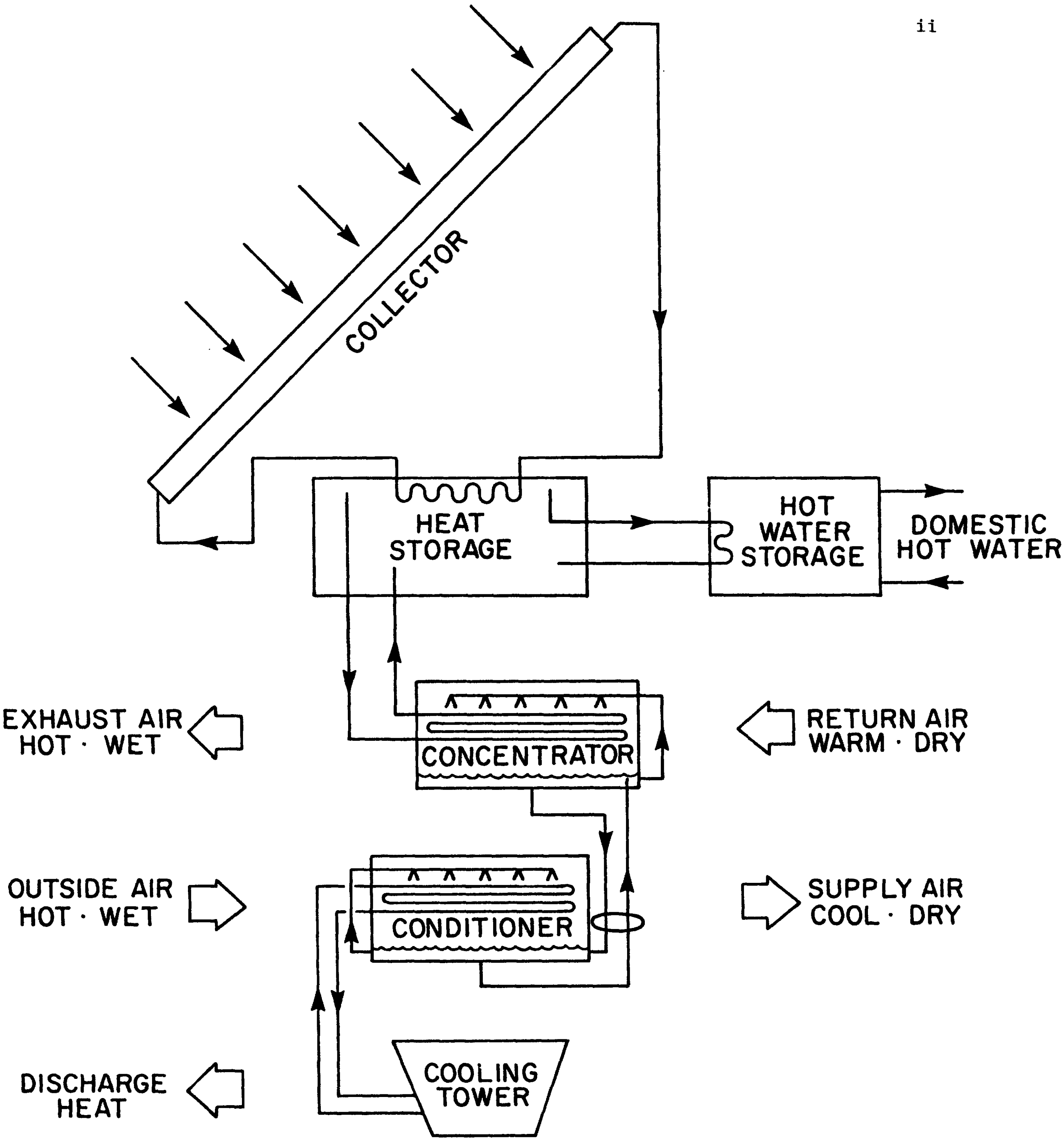
Abstract

The technical and economic feasibility of using solar energy to reduce conventional energy consumption of a large urban commercial building were studied in depth. Specifically, solar assisted dehumidification of ventillation air to reduce conventional air conditioning requirements for the Citicorp Center in New York City was investigated. A detailed computer simulation of yearly operation was made on an hourly basis using New York City temperature, humidity and solar data. Several system configurations were examined and were defined each operating in its most efficient fashion.

It was found that maximum energy savings was achieved by optimizing the operation of the conventional system.

Maximum energy savings could be achieved by the following, in order of decreasing impact: optimization of the operation of the conventional system, use of additional conventional equipment for energy savings, and use of the solar assisted system.



Conceptual Diagram for Solar Dehumidification

The solar dehumidification system consists of a conditioner which dries outside air taken into the building to replace air exhausted for ventilation purposes. Since the air is dry, the energy consumption by the conventional cooling system is reduced.

In the conditioner, the air is dried by contacting a cool liquid desiccant. The desiccant is heated in the concentrator by solar energy and the moisture driven off.

SOLAR ENERGY DEHUMIDIFICATION EXPERIMENT

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Executive Summary

Reported here are the results of a detailed study of the technical and economic feasibility of the application of solar energy to supplement conventional energy supplies of a large urban commercial building. A preliminary study was done which had as an objective the comparative evaluation of solar energy and steam for the production of chilled water and for the dehumidification by means of a liquid desiccant dehumidifier, of the building's ventilation air. The results of the preliminary study indicated the need to assess the solar application for dehumidification in more detail to determine if it was indeed a viable solar energy demonstration experiment.

Citicorp Center, under construction at a site at Lexington Avenue and 53rd Street in midtown Manhattan, was the proposed location for the solar dehumidifier experiment. The upper twenty-five floors of the central office tower were studied for this application. The study, led by the MIT Energy Laboratory was carried out in cooperation with Citicorp (First National City Corporation, New York) and its architects Hugh Stubbins and Associates, Consolidated Edison and Loring-Meckler Associates.

Initial architectural design studies projected terraced dwelling units on top of the office tower of Citicorp Center. The terraces were to be glass enclosed. The dwelling unit concept was abandoned, leaving the building with a sloping top looking West, which suggested the possibility of a sizeable solar collector installation (20,000 square feet) if the crown were rotated to face South. The preliminary study cost an estimated $110,000. Citicorp spent about $90,000 on design studies by the architects to rotate the crown of the building and to support work by their energy consultants, Loring-Meckler Associates, Inc., who initially proposed the application of solar dehumidification at Citicorp Center. The remaining $20,000 was furnished by Con Edison and the MIT Energy Laboratory for supporting work, program development and planning.

The preliminary study (Phase 0) estimated a net capital cost of a solar regenerated dehumidifier system at $900,000 and annual savings at $47,000. While this is not as rapid a payout as is usually desired, these initial results were encouraging enough to the design team and to Citicorp to warrant a closer look. Nearly six billion Btu might be saved by the solar dehumidification system annually, which is the equivalent in fuel of 56,000 gallons of residual oil at the steam plant.

Dehumidification in a conventional system is accomplished by chilling air to condense the moisture. To remove the right amount of moisture, incoming air often must be chilled to a temperature well below that desired in the occupied space. This over-chilled air then has to be reheated to satisfy comfort conditions in the occupied space. Thus energy is used, not only for reheat, but also to chill the air for removal of moisture.

Air can also be dehumidified by passing it through a liquid desiccant sprayed on coils containing cooling tower water. When saturated, the desiccant can be dried by heating to temperatures readily available from flat plate solar collectors and spraying the desiccant into air being exhausted from the building. A system of this kind, using triethylene glycol as the desiccant, is commercially available and widely used in industry where humidity control is particularly important, e.g., the food processing industry. These systems have not been similarly adopted for air conditioning in buildings because of high first cost and because a separate chilled water system is also usually necessary for air cooling. Where purchased energy is used for reheat and solar energy is available for drying the desiccant, this system provides additional savings and significant economic leverage for the solar system.

Because peak loads on many urban electric utilities occur during the summer as a result of high demand from air conditioning units, the widespread adoption of solar dehumidifiers in new buildings and their adaptation to existing buildings using electrically driven chillers can substantially moderate summer peak electrical loads.

The detailed study required careful investigation of the loads on the space conditioning system and the design of the conventional equipment to meet these loads most efficiently. During the mechanical design evolution it was discovered that the necessity of reheat could be eliminated by reducing air flow in the air handling units and by allowing room conditions to deviate moderately from design values under unusual load and outside air conditions. This approach, in effect, makes use of internally generated heat from lighting, equipment, etc. as a source of reheat energy. Some control is sacrificed by this approach, and if further conservation measures reduce internally generated heat, reheat may again be required.

Simulation programs were developed for all the building systems considered. Weather and climatological effects were included with data obtained for nearby Central Park. A basic objective of the simulations was to combine systems and components and to develop operational strategies to minimize purchased energy use in the building. In addition to solar dehumidification, absorption refrigeration and evaporative cooling systems were considered in conjunction with optimization of the outside air-return air mix. In an earnest desire to apply solar energy, opportunities to reduce energy use by equivalent amounts at possibly less cost should not be overlooked.

Outside air conditions are often such as to result in large savings from spray units. Evaporative cooling was investigated at two places in the system: spray on the conventional chiller coils, and spray on the dehumidifier coil. The results indicate that evaporative cooling can be an important method of saving purchased energy whether or not a dehumidifier is used. Also investigated was the value of achieving lower cooling tower water temperatures by reducing the approach temperature to one-half its conventional value. In the summary of projected savings for the systems below steam costs have been projected to average $11 per million Btu over the next decade.

Solar Assisted Dehumidifier	Gross Savings	O&M Costs	Net Savings
No Spray (7° Approach)	$18,400	$15,000	$ 3,400
Spray on Chiller Coil	$51,600	$20,300	$31,300
Spray on Dehumidifier Coil	$29,900	$17,650	$12,250
No Spray (3.5° Approach)	$18,900	$15,000	$ 3,900
Absorption Chiller @ 180°F			
No Spray	$20,400	$15,000	$ 5,400
Spray on Chiller Coil	$50,200	$20,300	$29,900

System capital costs were estimated to be $1,727,500 including evaporative cooling coil units. (Cost of coil units: $150,000). For half-system, net savings would be halved, but the net capital cost would be $1,032,000.

A number of things contributed to the serious decline of the benefit/cost ratio. Important amoung them is the loss of leverage for the solar system by the elimination of reheat in the system design. Credit for the reduction in required chiller capacity affected by the dehumidifier equipment was not taken because the number of chillers will remain the same to allow the plant to be used to condition also the Citicorp building across the street at 399 Park Avenue.

Because of the above savings projections, the rather large capital cost, and uncertainties regarding the procurement and installation of the solar collector on a schedule compatible with ongoing building erection, the solar dehumidification experiment for Citicorp Center is not recommended at this time.

This study has shown the importance of a detailed design study and a careful appraisal of other conservation options in the context of a specific building to the practical and economic demonstration of solar systems.

There is no question about the technical feasibility of the systems studied here. In fact, these systems should be carefully considered for other applications and in different contexts, where they might be far more favorable economically.

The potential of evaporative cooling as a conservation measure is great, particularly when implemented with an optimum outside-return air mixing strategy. Some professionals and building operators, however, are concerned about O&M problems associated with spray units. These concerns may deter widespread adoption.

Many questions were raised about the operation of conventional systems that cannot be answered in the detail desired without making critical measurements on the building while in actual operation. How effective certain operational strategies or energy conservation features employed are, will not be known until such measurements are made. An experimental evaluation of this building while in regular operation should be made.

ACKNOWLEDGMENT

The M.I.T. team wishes to acknowledge the substantial contribution of the following people in the carrying out of this study: Gershon Meckler, Loring-Meckler Associates; Adolph Nuchtern, Joseph Felner, Joseph R. Loring & Associates; Robert Bell, Paul Torpey and William Messner, Consolidated Edison Company of New York; W. Easley Hamner, Hugh Stubbins & Associates. The work of Gershon Meckler and others of Loring-Meckler Associates was funded by a subcontract from M.I.T. The time of the Consolidated Edison personnel was provided by Con Edison. The contributions of the architect and mechanical engineering firms were supported by Citicorp.

We are especially indebted to Thomas Creamer of Citicorp whose untiring interest and support made our investigations not only possible, but pleasant in the face of the exigencies of a tight schedule for a building already in design and construction.

Other individuals at Citicorp who were most helpful were Henry DeFord III, Robert Dexter, George Herbst and Edward Zimmerman.

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Introduction

1.1 The Urban Energy Problem

The recent shortages in petroleum supplies serve to emphasize both the need for new sources of energy and more efficient utilization of available energy supplies. To reduce both the peak and base demands for energy, the use of solar energy has been proposed. Solar energy now appears to be a technically practical supplement to conventional systems for space heating in buildings and can reduce conventional energy demand when heating is required. Solar energy for space cooling, however, has not been as well developed. A practical technique for solar space cooling of buildings in metropolitan areas would reduce utility energy demands during periods of the year when the demand is greatest. In particular, a solar cooling system which could be retrofited to existing buildings would have the potential for a significant near-term impact on urban peaks during the summer.

To date the most solar space cooling systems have used absorption refrigeration processes. At the present time these systems have relatively low efficiencies. The absorption process needs input water temperatures in the 200°F range. At these high operating temperatures a flat plate solar collector has a low efficiency. A reduction of the collector temperature to increase efficiency does not help, because as the input temperature to the absorption refrigeration equipment is lowered, its capacity decreases.

The system presented in this report employs solar energy to dehumidify air before cooling by the conventional cooling coils. Moist outside air is dehumdified by spraying triethylene glycol, a liquid desiccant, into the air. The desiccant becomes diluted as it absorbs moisture. The solution is dried and reconcentrated by heating it to 140°F with solar energy and using the exhaust air to carry off the moisture. In this system, solar collectors may be operated in the range of 150°F which results in substantially higher flat plate collector efficiencies. At high collector efficiencies, a solar dehumdification system may be a feasible means to reduce the conventional energy requirements of high-rise buildings in an urban environment. The dehumidifier system utilizes components which are commercially available, eliminating the need for a lengthy development process.

In urban areas, many electric utilities experience the peak load in the summer. Net energy savings resulting from the use of the proposed dehumidification system will reduce electrical requirements when the peaking is most severe. The system can also be easily retrofited to existing air conditioning systems. For high-rise structures having limited space for collector installation, the system would make use of all or most of the available solar energy.

Because of the potential indicated by a preliminary study done by Loring Meckler Associates in conjunction with M.I.T. and Consolidated Edison, an experimental solar energy laboratory was proposed to further evaluate solar dehumidification in a high-rise metropolitan building. The office tower of Citicorp Center being built in New York City was selected as the site of the proposed experiment. Although the performance of the system components is well known, the proposed system represents a unique combination of these components. Therefore, a detailed design study and performance simulation was necessary to substantiate the initial costs and energy saving projected for the system before its construction.

The Massachusetts Institute of Technology in cooperation with Citicorp and its architects Hugh Stubbins and Associates, Consolidated Edison of New York, and Loring-Meckler Associates, an energy consultant, has conducted the study of the solar dehumidification system. The study was sponsored by the National Science Foundation (RANN).

1.2 General Description of Building

Citicorp Center, Fig. 1-1, has three buildings, St. Peter's Church, a seven-story skylighted galleria, and an office tower over nine hundred feet tall. From the Center's inception, energy conservation has been a primary consideration. Life cycle costing has been used as a basis for decisions on designs offering the greatest potential for energy conservation and management.

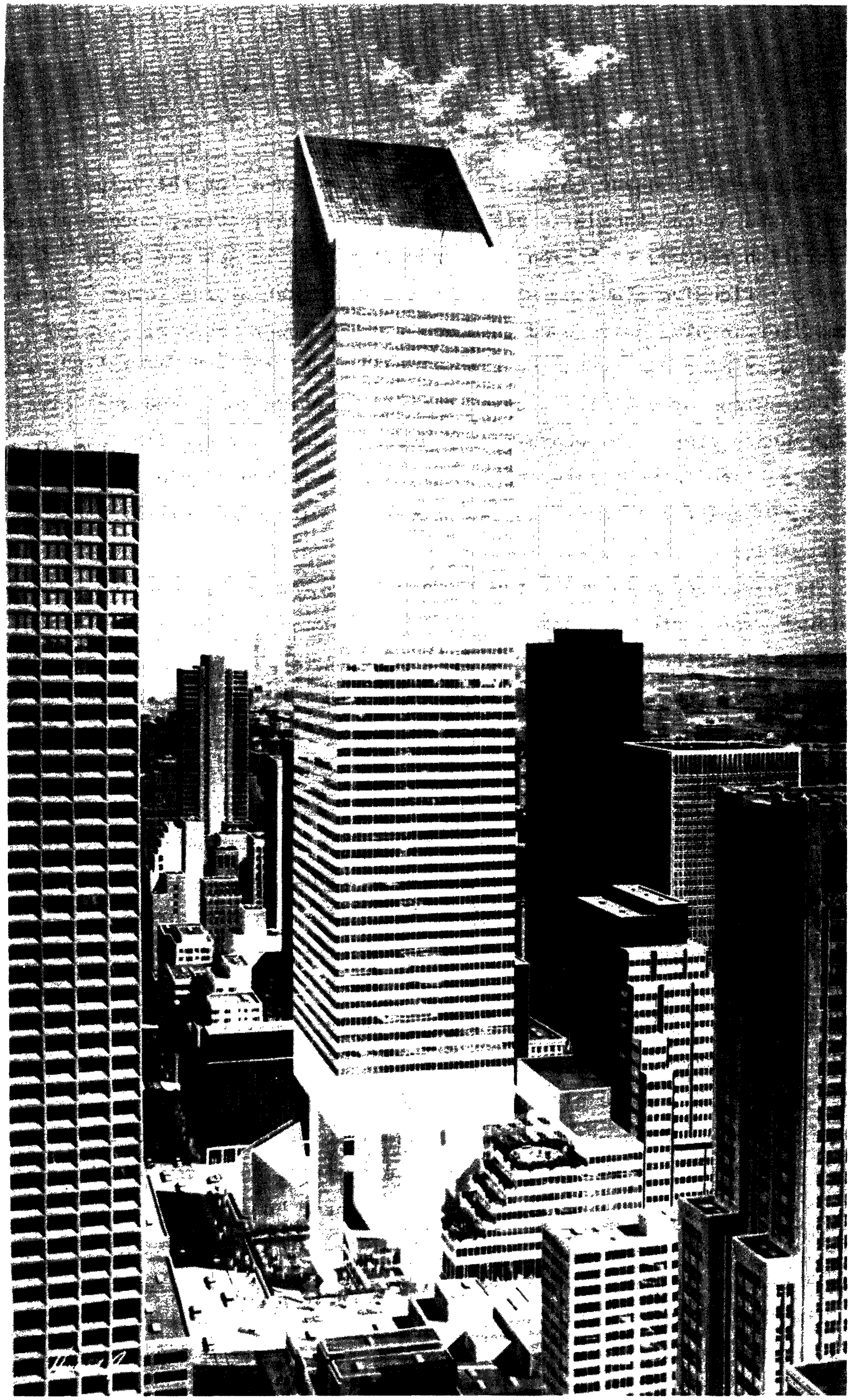


FIG. 1.1: CITICORP CENTER'S THREE BUILDINGS, ST. PETER'S CHURCH, A SEVENTEEN STORY GALLERIA, AND THE OFFICE TOWER

The tower is square to minimize the surface area for the volume enclosed and only 46% of the facade is glass. The highly reflective, double-glazed fenestration reduces radiation and conduction heat losses and gains. Light colored, reflective aluminum spandrels clad the rest of the tower. Two-inch thermafiber insulation is used throughout. The tower has a sloping roof facing 29° west of south extending from elevations of 791'-6" to 913'-0", with plan dimensions of 156'-10" square, giving a sloping roof area well over 20,000 square feet. The weatherproof decks of the roof are flat and the sloping surface is ventilated to meet fire codes. The sloping surface can support the solar collectors or can be cladded with spandrels as used for the remainder of the curtain wall.

The space enclosed under the sloping roof, Fig. 1-2, contains the cooling tower, water storage tanks, a tuned mass damper to control building motion in high winds, and other mechanical equipment. Floors 56, 57 and 58 immediately under the sloping part, house the remainder of the mechanical equipment which serves the top half of the tower building. A tenth floor mechanical equipment room contains three 2000 ton steam driven centrifugal chillers which provide chilled water for the entire center. A tunnel under Lexington Avenue to an existing building at 399 Park Avenue permits both buildings to be serviced from a single plant under conditions other than peak demand periods to further improve the efficiency of operation.

An air-and-water system was chosen to meet the buildings' space conditioning requirements. The primary air system for the upper twenty-five floors consists of two interior zone air handlers each with a capacity of 164,450 CFM. Dampers control the mix of outside and return air. A minimum of 22,000 CFM of outside air per unit (44,000 CFM total) is used. The system has the advantages of positive ventilation, central dehumidification, winter humidification and good temperature control. Ordinarily dehumidification is performed at the chiller coils and winter humidification is performed with steam. The solar driven dehumidifier consists of two separate units installed in the outside air ductwork of the interior zone air handlers.

Selected floors in the tower can be operated while minimizing air conditioning supplied to other floors. Controls are provided to selectively

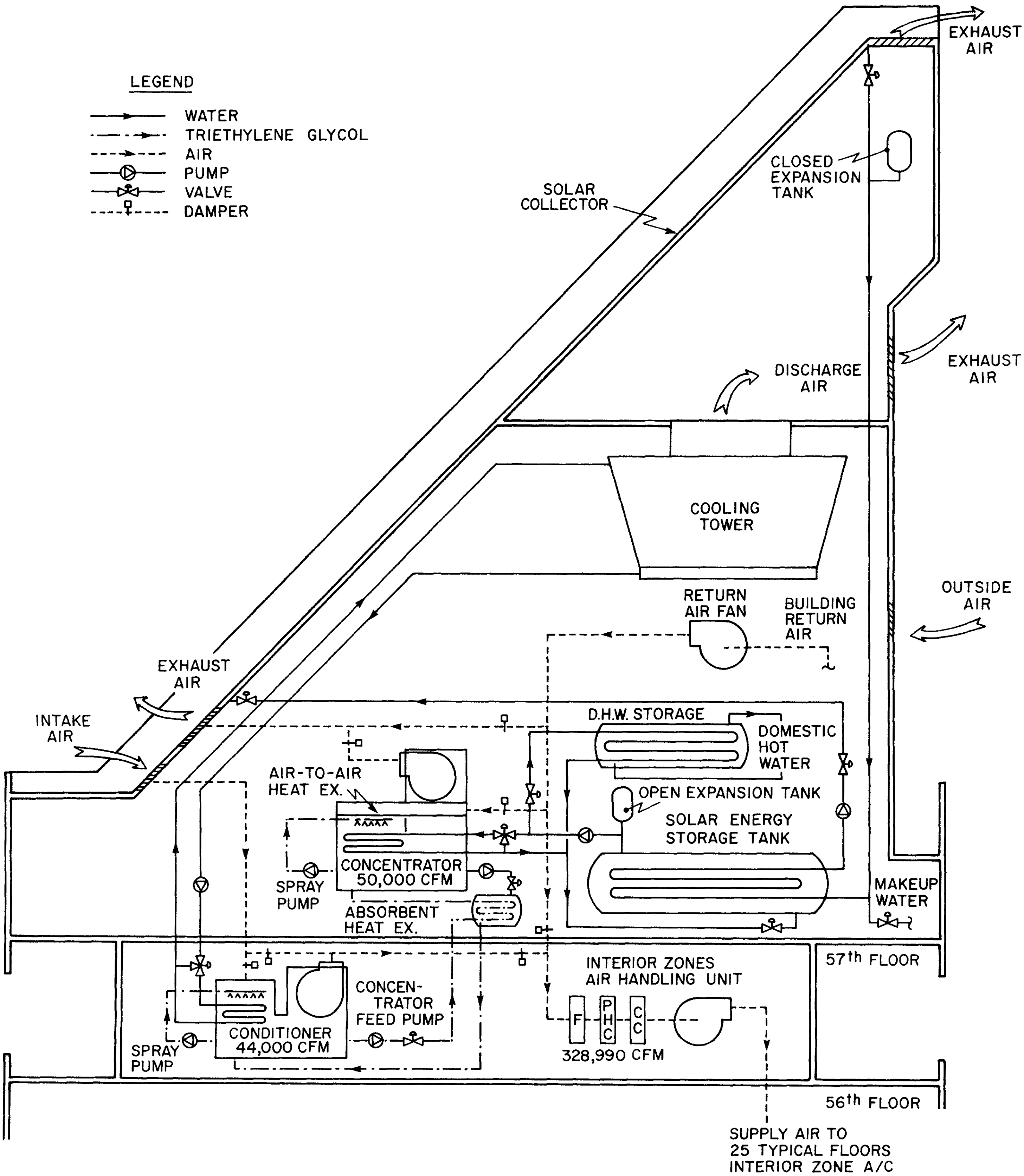


FIGURE I-2

PHASE I SYSTEM SCHEMATIC DIAGRAM

shut off secondary water supplied to perimeter units on unused floors and to shut off air supplied to interior systems serving unused floors. Variable volume inlet vanes are included in interior fan systems. This will permit the use of the so-called "economiser cycle" which makes use of appropriate volumes of outside air for space conditioning when outdoor temperatures and humidities are suitable. The two-pipe, low-temperature change-over induction system was chosen for the perimeter to reduce air quantitites, duct work, and fan horsepower.

Low brightness, baffled, efficient lighting fixtures produce a satisfactory light level at 2.5 watts per square foot rather than the 4 watts per square foot now commonly used.

A computer controlled building managment system has been designed for the building as an additional energy conservation feature which can ensure efficient operation of building systems, moderate peak demand, and schedule maintenance operations for low-load occasions. Lighting can also be controlled by this system to reduce lighting levels during off-hours cleaning and to shut lights off when cleaning operations are completed. Studies indicate that master switching by floor or quadrant will permit an 8% saving of total building energy by control of cleaning operations alone.

1.3 Purpose of Study

The application of the solar-dehumidification concept for the Citicorp Center was first suggested by Citicorp's consultant, Loring-Meckler Associates. The concept was initially evaluated in a first order feasibility study, Phase 0*, completed last year which indicated that the proposed solar dehumidification system would save more conventional energy than a solar-absorption chiller system. It was estimated that the yearly energy savings of the solar system would represent 5% of the initial solar system cost.

This report describes the results of the Phase 1 study, which was a one-year program definition and experimental design phase. The objectives of the Phase 1 study were the accurate simulation of the solar system performance and the final design of the system so that capital costs for equipment and

*See Appendix I for a condensation of this report.

installation could be obtained from potential contractors.

1.4 Basic Concept

The basic concept is compared with a conventional air conditioner in Fig. 1-3. In a conventional unit the cooling coil is required to both remove moisture and lower the temperature of air to be circulated to the building space. In some instances, to achieve proper moisture control, the air must be over-cooled and a reheat coil is needed to achieve a comfortable dry bulb temperature*. In the new system the dehumidifying and cooling functions are separated. Outside makeup air is dehumidified and then mixed with return air. Ideally, when the dry air leaving the dehumidifier is mixed with the return air the mixture has the desired moisture content. The conventional cooling coil only has to do sensible cooling to lower the dry bulb temperature of the mixture.

If excess dehumidifying capacity exists, one alternative is to overdry the mixture and then spray water to provide evaporative cooling.

A schematic of the dehumidifying equipment and solar collector is shown in Fig. 1-4. The dehumidification units are commercially available. In the dehumidifier, concentrated triethylene glycol (TEG) is sprayed over coils cooled by a cooling tower. Air is blown over the coils and contacts the TEG. Due to differences in the water vapor pressure between the humid air and the cool TEG, water is absorbed by the TEG and the air is dried.

The dilute TEG goes to the regenerator. Solar energy collected on the sloping roof to the Citicorp Building is transferred to a heat storage tank. The hot water in the tank heats the coils in the regenerator to about 140 to 150°F. At elevated temperatures the vapor pressure of the TEG rises and water is driven off into the exhaust air. Since the system uses relatively low temperature solar energy, it has an advantage over an absorption system which needs a temperature in excess of 200°F. The TEG and air-to-air heat exchangers shown in the figure serve as heat recuperators or preheaters to lower the amountof solar heat required.

*The design concept at the time of the Phase 0 Study included reheat, but in the final design, reheat coils were eliminated.

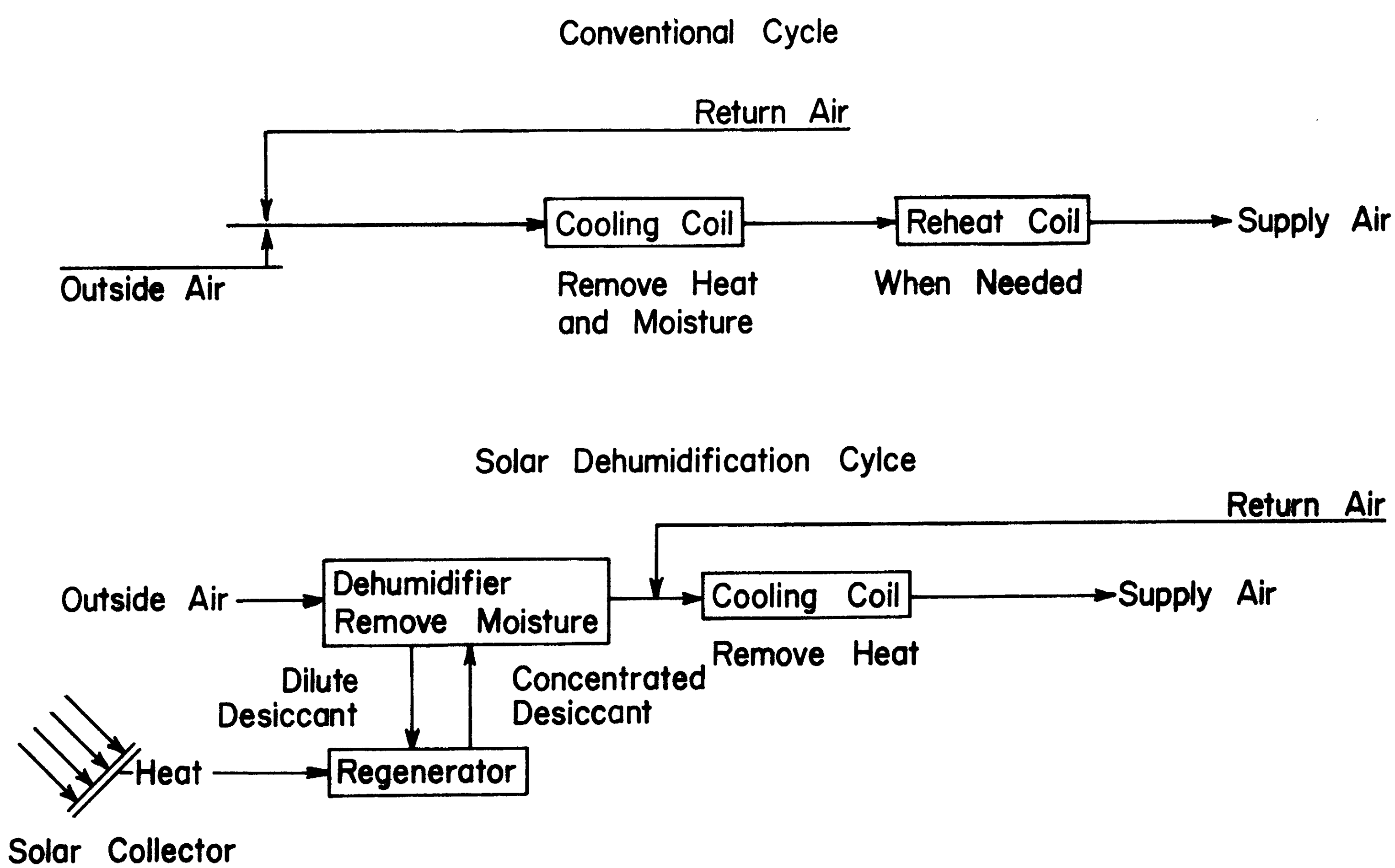


FIGURE 1-3: CONVENTIONAL AND SOLAR DEHUMIDIFICATION CYCLE BLOCK DIAGRAMS

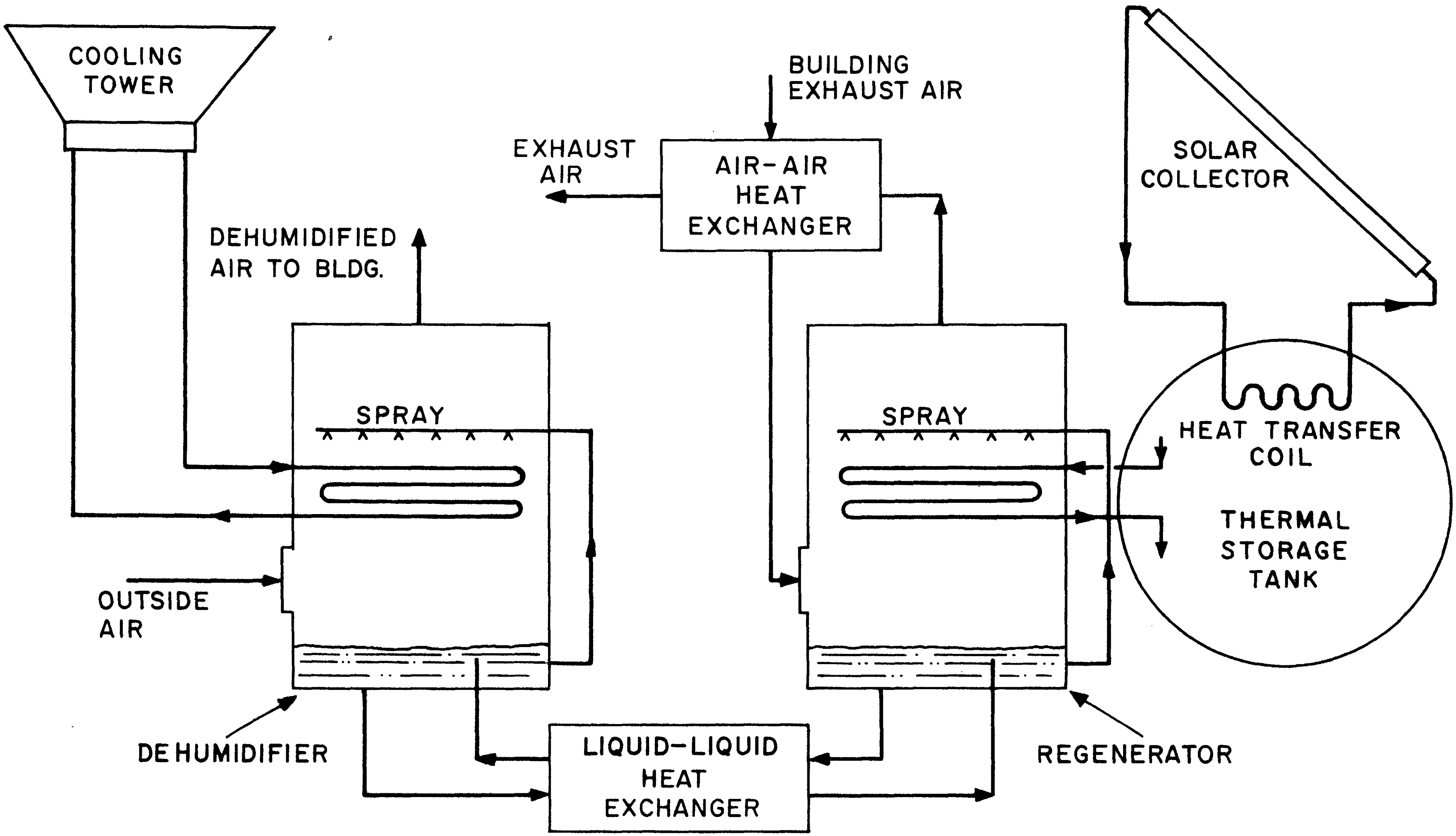


FIGURE 1-4: LIQUID DESICCANT DEHUMIDIFICATION EQUIPMENT

TEG is a non-toxic substance with no known health hazards. In fact, it is frequently used as a germicide in hospitals.

Fig. 1-5 illustrates components and different operational modes of the system. Spray coolers can be added either to the cooling coil or after the dehumidifier. When the absolute humidity is below the desired levels for the building, the spray performs some sensible cooling. The outside air flow can be increased on moderate and low temperature and humidity days to reduce the cooling load of the conventional coil.

1.5 Key Design Questions

In order to develop an accurate simulation of the system and arrive at a final design, a number of key design questions were addressed.

The solar collector must represent a good trade-off between efficiency and reasonable cost. Various designs of flat plate collectors, e.g., with one or two glass cover plates, with and without selective coatings, were evaluated along with other, non-flat plate configurations. Also the panels msut be available to meet the tight construction schedule of the building.

The dehumidifier and regenerator will be operated at conditions which differ somewhat from conventional installations. Operating curves of the units at the new conditions must be verified. Both physical characteristics and health effects of TEG are needed.

All of the feasible design options, e.g., with spray and with variable make-up air flow, must be identified. For each option, the optimum system operation as a function of ambient conditions must be defined. These conditions combined with an hourly simulation of the New York City weather yield a simulation of the system over an entire year. Cost savings can then be found by comparison with a conventional system also operated in its optimum fashion.

The following sections of the report examine each of the questions given above.

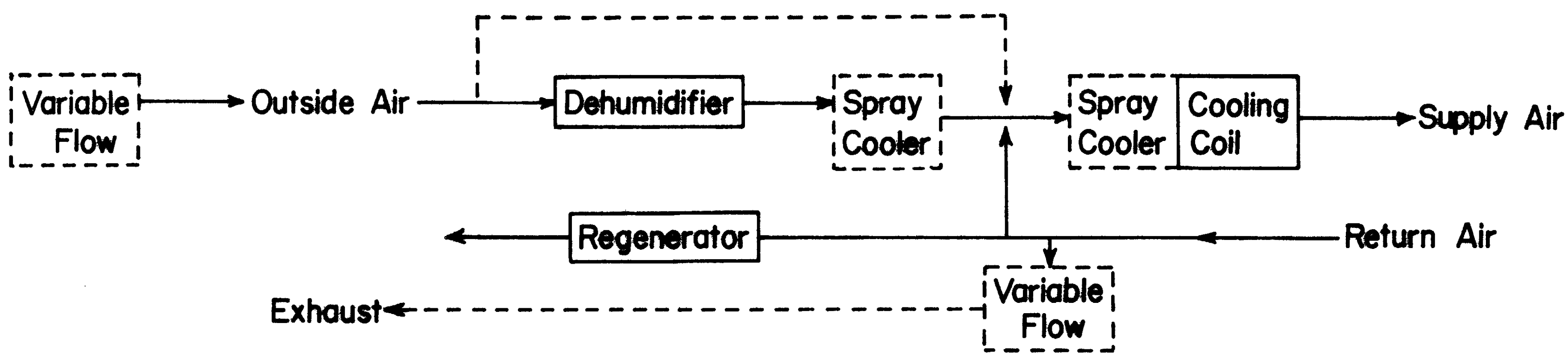


FIGURE 1-5: ADDITIONAL COMPONENTS AND OPERATIONAL MODES OF THE SYSTEM

II. Component and System Models

It was necessary to model each of these major components and their combination into systems in order to predict the energy saving potential of various proposed systems compared with a conventional system with reasonable accuracy and to simulate overall system operation under conditions determined by weather data for New York City.

The major components of the proposed system include: the solar collector storage system, the dehumidifier which reduces the moisture content of the inlet makeup air, the regenerator or conditioner which removes the moisture from the desiccant used in the dehumidifier and the HVAC system.

The HVAC system model consists of a number of sub-system models which can be combined to represent both conventional and solar assisted systems. These basic sub-system models include: a building load model, a cooling coil model, an operating strategy for the conventional system, a dehumidification system model, and an absorption system model.

Models were developed for the many variations of flat plate solar collectors and for one version of the evacuated tube collector (the flat plate inside an evacuated glass tube) and for insulated water tank storage. The development of models for the dehumidifier/regenerator required more detailed information on the vapor pressure characteristics of triethylene glycol than were available. Special measurements of these characteristics were made.

In this chapter the model used for each of the components is described and the strategy of system operation formulated to optimize the energy savings. The technical details of the analysis are contained in the appendices to this report.

Readers interested in the overall results can omit this chapter and turn directly to the next chapter.

2.1 Collector-Storage Models

2.1.1 Solar Collector Analysis

A computer simulation for flat plate solar collectors was developed by closely following the design equations set forth in the solar energy literature. A summary of these equations is given in Appendix II along with a flow chart indicating their usage. In our present simulation a detailed method of determining the heat losses from the collector plate to the ambient was used rather than a semi-empirical overall heat loss coefficient. Recent data on heat transfer between inclined parallel plates heated from below was used and the collector heat losses were then determined through the simultaneous solution of a set of coupled non-linear algebraic equations. The simulation allows for different absorber plate flow-passage designs, specified radiative properties of the collector surface (i.e., selective surfaces), arbitrary radiative properties of the glass covers, directional dependencies of the above properties, any amount and type of insulation, arbitrary collector tilt, number of glass covers and fluid flow rate.

In addition to the simulation for flat plate collectors, an analysis was made of a collector plate contained in an evacuated glass cylinder. The detailed development of this calculation procedure is given in Appendix II which indicates the method for modifying the flat plate program to accommodate the cylindrical collector. The calculation procedure becomes complex since one must determine at each point in time the transmission of the sun's rays through the curved glass surface to the absorbing plate. However, calculation of the heat loss to the surrounding atmosphere is greatly simplified since there is only one "cover" and the absorbing plate is in a vacuum.

The generality of the programs allowed various parametric studies to determine the optimum collector construction and mode of installation for Citicorp. Taken as inputs in the studies were typical hourly weather data for Central Park in New York City and the fact that the "South" facing slope of the building is 29 degrees west of south. Also, the angle of slope of the basic crown structure is 45 degrees. To determine how the collectors performed without regard to the operation of the rest of the system, the simulations were run throughout an entire year assuming that the temperature of the fluid supplied to the collectors could be maintained at any given value.

To compare the performance of the flat plate and the cylindrical collectors, effective collecting areas for the two were established. Based on manufacturers' specifications, it was estimated that net areas of 18,000 ft^2 of flat plate and 10,000 ft^2 of cylindrical collectors could be placed on the crown of the office tower. The cylindrical configuration does not permit as much effective area because of shadowing of adjacent cylinders if placed too close together. The flat plate collectors were initially assumed to be installed at 45 degrees to conform to the crown slope. The cylindrical collectors, on the other hand, had the flexibility that the cylinders could be oriented either horizontally or up and down the crown slope. In addition, the orientation of the collector plate inside the cylinder could also be specified. A parametric study showed that either cylinder orientation gave nearly the same yearly collection rate as long as the collector plate was oriented optimally inside the tube. The horizontal orientation was selected for the cylinder since it performed better during the summer months, and a plate angle of 30° was chosen since it yielded the maximum yearly collection rate.

A basic tenet of the project is to make effective use of the lowest temperature solar heat possible. Table 2-1 shows the effects of collection temperature on total energy collector for the two basic systems.

Table 2-1

TOTAL ENERGY COLLECTED FOR THE YEAR

(Millions Btu's)

Collection temperature	Flat Plate 18,000 ft^2	Evacuated Cylindrical 10,000 ft^2
140°F	2571	1968
180°F	1687	1444
220°F	960	954

(Figures showing the monthly variations corresponding to the tabulated yearly totals are given in Appendix IID). For the flat plate studies, a collector construction similar to the PPG Baseline collector using double glazing and a selective surface was assumed. The results indicate that the superior efficiency of the evacuated tube collector cannot make up for the loss in effective collector area at the lower collection temperature which can be used in this system. At the higher temperature, such as that required for efficient operation of an absorption type chiller, the total energy collected for the year for the two collectors is the same.

The importance of the angle of tilt of the flat plate collector was also investigated over the year. The results of this parametric analysis are shown in Table 2-2 (and Figure 3 of Appendix IID). A tilt angle of 15°

Table 2-2

EFFECT OF ANGLE OF TILT OF COLLECTOR
ON ENERGY COLLECTED
(Millions Btu's)

Angle of tilt	April-Sept	Oct-March	Total for year
45°	1661	910	2571
30°	1949	800	2749
15°	2071	587	2658

For: Flat Plate Collector -- 18,000 ft^2

T_c = 140°F

ε = 0.2

2 glass covers

from horizontal is best for the summer months when the load on the dehumidification system is the greatest. However, on an annual basis, the tilt angle does not affect very much the energy collected. There may be cost advantages in keeping the angle of tilt of the collector the same as the basic structure of the crown, 45 degrees.

Next, parametric studies were performed to find the effect on the energy collected of the total number of cover plates used, the application of selective surfaces and various combinations of these two parameters. The results are shown in Table 2-3 (and in Figures 4-6 of Appendix IID).

Table 2-3

EFFECT OF NUMBER OF GLASS COVERS AND ABSORBER PLATE COATING ON YEARLY ENERGY COLLECTION (Millions Btu's)

	1 Glass Cover	2 Glass Covers
Selective Surface	2351	2571
Non-Selective Surface	1399	2082

For: Flat Plate Collector -- 18,000 ft^2

T_c = 140°F

45° Tilt

A summary of the annual collection of energy is shown in the table.

These studies show that at our low collection temperatures the evacuated cylindrical collector cannot make up in efficiency for its smaller effective collecting area. If all the solar heat collected can be utilized either in the dehumidifer or to supply heat for the domestic hot water of the building, a 45° tilt angle for the collector is acceptable. In terms of annual collection the two cover plate, selective surface flat plate collector is the best, followed closely by the one cover combined with a selective surface. The one cover plate reduces the weight of the collector which may be necessary to ease installation problems on the crown of this tall building. A non-selective surface with two glass covers is acceptable, but about a 20% loss in the total annual solar heat collected occurs.

2.1.2 Storage Analysis

The above studies assumed that the inlet fluid temperature to the collectors could be maintained at any given value. This assumption can be interpreted as drawing and returning the fluid to an infinitely large storage tank at that temperature. In the following total system simulations, a finite size tank was used. It was assumed to be 20 feet in diameter, 16 feet long containing 37,000 gallons and to be covered with 3 inches of Fiberglas insulation. The fluid drawn from the tank and sent either to the TEG concentrator or back through the collectors was taken to be at the mean temperature of the tank. The governing equations for the nonstratified storage tank are included in Appendix IIA.

2.2 Liquid Desiccant Dehumidifier Model

2.2.1 General Description of the Equipment

The other main component in the proposed system other than the solar collector is the dehumidifier-regenerator. The dehumidifierand regenerator are commercial products manufactured by Niagara Blower Company of New York. The very low vapor pressure liquid desiccant used is triethylene glycol, TEG. At low temperatures, the water vapor partial pressure of a concentrated TEG-water mixture is low allowing the fluid to absorb water. The water vapor partial pressure rises as the mixture is heated to permit the water to be driven off. The vapor pressur of TEG remains about an order of magnitude lower than that of water minimizing the loss of TEG during the concentration process. See Figure 2-1 for a schematic of the dehumidifier equipment.

2.2.1.1 The Conditioner (Dehumdifier)

In the dehumdifier, air is dehumidifed by drawing it through a chamber in which triethylene glycol is sprayed over finned-tube cooling coils.

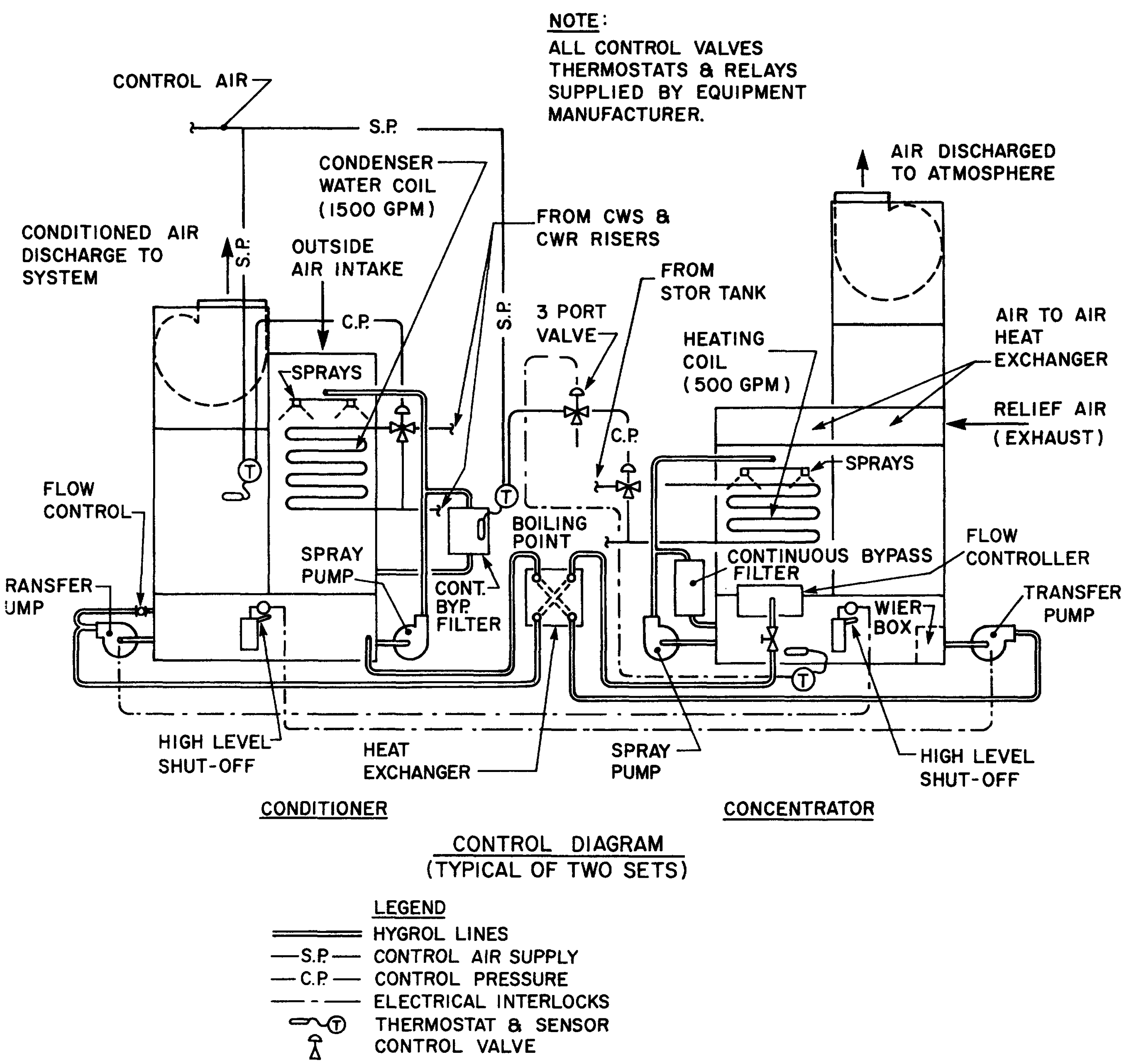


Fig. 2.1: Schematic & Control Diagram of the Dehumidifier Equipment

The TEG is kept cool by the cooling water passing through the finned tubes. Hence the partial pressure of the water vapor above the cool TEG is kept low compared with the air stream thereby providing the driving potential for the mass transfer. The cooling water in the tubes carries away the latent heat of the moisture which is condensed as well as the sensible heat associated with the lowering of the bulk air temperature. The dew point and dry bulb temperatures of the air leaving the unit are controlled by two factors: (1) the spray contact temperature and (2) the TEG concentration. The spray contact temperature (which is essentially the temperature of the TEG as it drops off the last fin) is established by the cooling load and the volume and temperature of cooling water available. The cooling water is obtained from the cooling tower which is primarily used for the conventional refrigeration units. The concentration is fixed by the operation of the TEG concentrator or regenerator.

2.2.1.2 The Concentrator (Regenerator)

Dilute TEG is constantly pumped from the conditioner to the concentrator where the water vapor it has absorbed is then driven off. The internal components and operation of the concentrator are qualitatively similar to the conditioner. Again, air is drawn through a chamber in which TEG is sprayed over plate-finned-coils. Now the air is relatively dry and hot water flows through the coils so that the equilibrium partial pressure of the water vapor above the TEG is greater than that of the air stream, and hence water is driven out of the TEG. The concentrated TEG is then pumped back to the conditioner. The operation of the concentrator is controlled by the spray contact temperature and the TEG concentration - which are in turn determined by the cooling load and the volume and temperature of hot water available.

The concentrator has an additional section not found in the conditioner. Due to the elevated temperatures in the unit, some TEG is inevitably evaporated. To reclaim this, the concentrator has a reflux coil to cool the air as it leaves the unit and thereby retrieve the entrained TEG by condensation. A source of relatively cool water (90°F) is needed for this coil.

Finally, the flow of warm TEG to the conditioner passes the flow of cool TEG to the concentrator in a heat exchanger between the two units. This in effect reduces the cooling load in the conditioner and the heating load in the concentrator.

The hot humid air leaving the concentrator before being exhausted from the building passes through a heat exchanger to preheat the incoming air to the concentrator which is exhaust air from the occupied space.

2.2.2 Detailed Model of Dehumidifier/Regenerator

A detailed analysis of the heat and mass transfer mechanisms in the Niagara units was performed to verify their operating characteristics. The analysis was an extension of conventional cooling tower analysis to the case of non-adiabatic operation (energy which is either added or removed by the water in the coils). The results of the steady-state numerical calculations indicate that the sizes of the units are such that the moisture content of the leaving air corresponds closely to the equilibrium vapor pressure of the TEG which drops off the bottom coil. This gave confidence that the equilibrium vapor pressure curves for triethylene glycol could be used to determine the dew point of the air leaving the unit rather than having to use a detailed numerical simulation.

2.2.2.1 Vapor Pressure of Triethylene Glycol

The accuracy of the equilibrium vapor pressure curves is crucial to the prediction of unit performance. Hence the Niagara curves were compared with the vapor pressure curves published by Union Carbide and Dow Chemical in their booklets on properties of glycols. There was significant disagreement between the three sources, and in fact, there was internal disagreement in the latter two. As a result, an independent evaluation was necessary. PVT Incorporated of Houston, Texas was capable of performing accurate vapor pressure measurements, and hence TEG samples supplied by Niagara Blower Co. were sent to them. Their results indicated that using the Niagara data would predict conservatively the operation of each unit. (They measured lower vapor pressures at the temperatures characteristic of the concentrator and higher vapor pressures at conditioner temperatures. Therefore, it was acceptable to use the Niagara curves (and their extrapolation to higher temperatures) for predicting unit performance.

2.2.2.2 Combined Dynamic Simulation

Finally, dynamic simulation of the combined conditioner, heat exchanger, concentrator system was developed, and results indicated that the system could respond rapidly to variations in weather conditions. Thus to study annual system performance from hourly weather data, sufficient accuracy could be obtained by solving statically for the operating point corresponding to the instantaneous weather conditions.

2.3 HVAC System Model

The performance of the individual components was integrated into the HVAC system model. This model simulated the operation of the system for given weather conditions to meet the load requirements of the building. Each of the subsystem models are described below.

2.3.1 Building Load Model

The building load to the HVAC system consists of sensible heat gains from lighting and electric power consumption, heat added by supply and exhaust fans, and both sensible and latent heat due to occupants. There are no radiant or skin losses from the building since the system considered controls an interior space surrounded by a perimeter which is separately controlled to the same temperature.

The portion of the building under study is an interior zone with two air handling units; each supplies 165,000 square feet. The loads on each unit are:

Occupants: The occupancy density is 0.01 person/s.f. Each person dissipates 245 Btu/hr of sensible heat and 205 Btu/hr of latent heat (1400 grains/hour of moisture).

Lighting and Electricity: The electric power load is 2 W/s.f. for lights and 0.5 W/s.f. for electric utilities. Of the lighting load, 70% goes to the occupied space and 30% to the return plenum. A lag occurs in the buildup of temperature when lights are turned on in the morning due to the specific heat of the building and furnishings. It is assumed that the lag is exponential starting at 70% of full power with a time constant of 1 hour.

Supply and Return Fans: Each air handler has two supply fans and one return fan. For the initial design flow of 1 cfm/s.f., the supply fans produce a 9" W.G. head and consume 136 H.P. each. The return fan produces a 3 1/2 W.G. head and requires 40 H.P. At a lower flow rate of 0.75 cfm/s.f., the heads can be reduced to 5" and 2" in which case, from the fan characteristic curves, the power is reduced to 75 H.P. each for supply air and 20 H.P. for return air.

2.3.2 Cooling Coil Model

The cooling coil is represented as a device which takes entering air from its initial state to an exit state in which the dry bulb temperature is controlled. Preliminary psychrometric studies suggest a leaving temperature of 60.1°F. If the entering air has less than approximately 77 gr/lb of moisture, then only sensible cooling is required and the air exits with the same moisture as it entered. In this case the energy required by the coil for sensible cooling to 60.1°F is computed. If the entering air has greater than approximately 77 gr/lb, then exit conditions at 60.1°F correspond to a saturation condition and some water is removed from the air. In this case the energy required by the coil for both cooling and condensation is computed following the process represented by standard cooling coil curves as shown on the Trane psychrometric chart described in detail in Appendix III.

An optional case has been considered for conditions in which entering air has less than approximately 77 gr/lb and in which spray may be used with the cooling coil. In this case the air is taken from its initial state to a final state of 60.1°F and approximately 90% relative humidity. This process uses less energy from the cooling coil because of evaporative cooling by the spray. The cooling coil spray process is described in Appendix III.

The operation of the spray system is somewhat unconventional since, if water is sprayed on the cooling coil resulting in a combination of sensible and evaporative cooling, it is possible to use spray even when the incoming wet bulb is above the desired outlet wet bulb temperature. The situations with and without spray are shown in Figure 2-2. In the summer operation, energy is saved since the air exits the cooling coil near 90% relative humidity, whereas on a warm dry day without spray, the exit humidity would be less than 90%. The savings in energy come from the difference in enthalpy at the specified exit temperature. Spray can also be used during winter operation with two beneficial results. Extremely low humidity may be avoided which results from mixing cold dry outside air with the return air. When mixing fresh and return air in preparation for spray, it is possible to mix to a higher temperature, thus using less outside air so that when the outside temperature is below 40°F, less preheat energy is required.

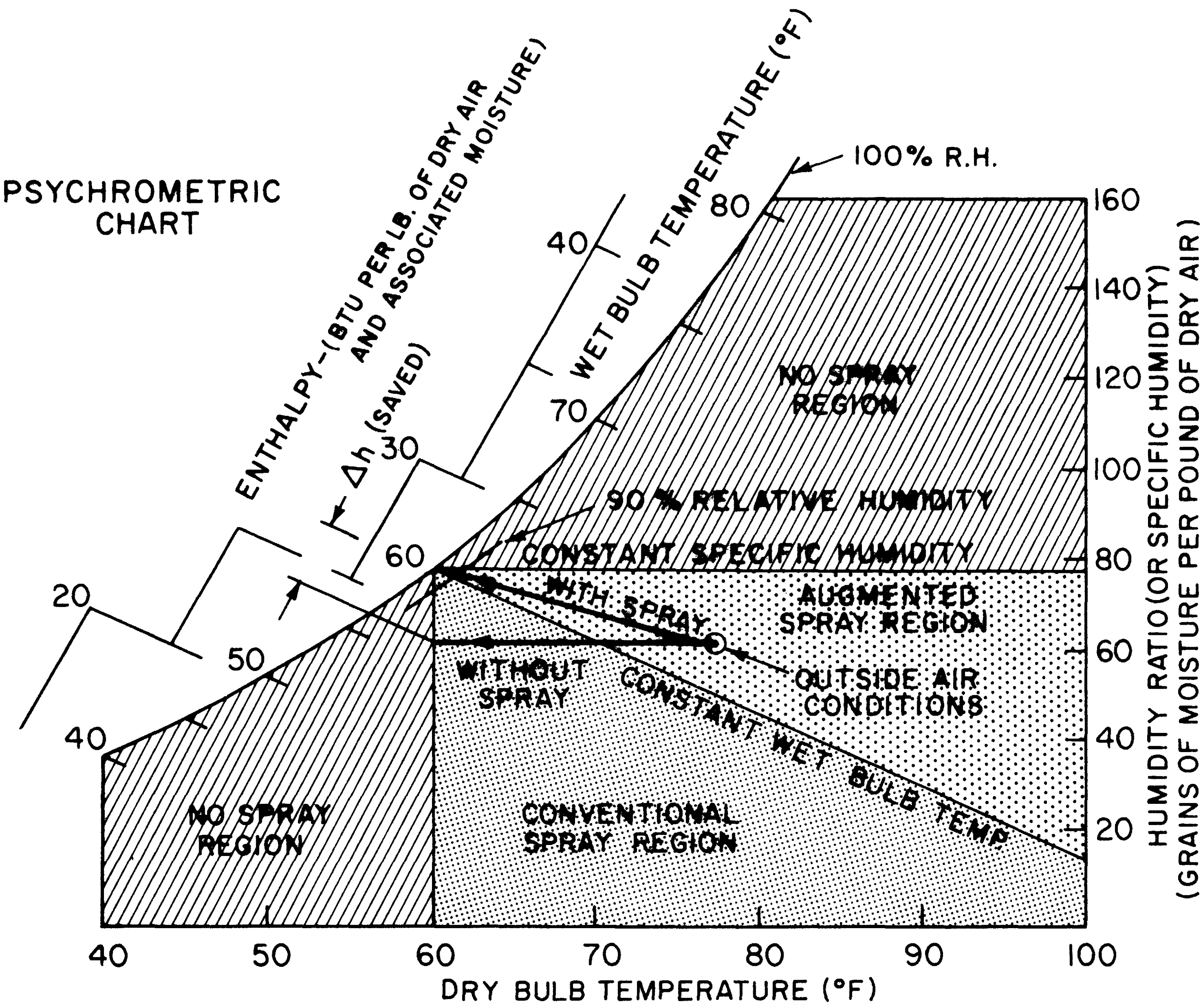
Thus for cases with or without spray, the cooling coil model computes the amount of energy required to take entering air to a state to 60°F dry bulb and the appropriate relative humidity depending upon the use of spray. Energy required to drive the cooling coil is obtained from steam. In computing the amount of steam energy required a COP of 0.7 was used for the cooling unit.

2.3.3 Conventional Operation System

In the conventional system, outside air and building return air are mixed and passed through the cooling coil to achieve desired exit conditions. The total air flow is set at 0.75 cfm/sq. ft. by design, of which at least 0.134 cfm/sq. ft. must be fresh air by code. The conventional system is controlled by adjusting inlet dampers which control the relative amounts of fresh and makeup air combined and sent to the cooling coil. It has been assumed that the dampers in the system are modulated on an hourly basis to provide operation which minimizes the energy required by the cooling coil while remaining within the set total and minimum fresh air requirements. The modulation is based upon measurement of the building return air and fresh air states and an operating strategy which essentially:

(1) for the case without spray, selects a mixture ratio which yields a dry bulb temperature as close as possible to 60.1°F.

FIGURE 2-2: ILLUSTRATION OF THE USE OF SPRAY TO CONSERVE ENERGY



(2) for the case with spray, selects a mixture ratio which yields an enthalpy as close as possible to the desired cooling coil-spray exit set point enthalpy.

The operating strategy for this system can be implemented using a computer control system which utilizes building and outside dry and wet bulb temperature data to determine the optimum setting of air dampers on an hourly basis.

In the winter time, conditions may occur in the conventional system without spray in which the relative humidity may fall below 30%. In these cases, steam injection is used to raise the relative humidity to 30%.

2.3.4 Solar Assisted Dehumidification System

In the solar assisted dehumidification system three air streams are mixed to form the air entering the cooling coil: outside air, building return air and air from the dehumidifier. The mixing strategy adopted is based upon minimizing total energy consumption while meeting conditioning temperature, humidity and flow rate specifications. The states of the three air streams were used to determine the optimum mixing ratios. In the paragraphs below, first the overall operating strategy is defined, and then the detailed operating model for the solar assisted dehumidification system is outlined.

In establishing the general operating strategy three types of operating situations have been identified:

A. Outside dry bulb less than cooling coil outlet temperature; 60.1°F.

B. Enthalpy of air leaving the dehumidifier is greater than the enthalpy of air entering the dehumidifier.

C. Outside dry bulb is greater than the cooling coil outlet temperature and the enthalpy of dehumidified air is less than the enthalpy of the fresh air.

For either condition A or B, the dehumidification unit is not used and the system is operated in the conventional conditioning mode.

In condition C, the dehumidification unit can be used to save energy. Three operating strategies were employed:

Case C1: The dehumidified air is mixed with either return air or fresh air in order to obtain a minimum enthalpy for the mixture. If the mixed temperature is less than the design coil exit temperature, then the mixing ratio is adjusted so that the mixed temperature is equal to the design coil exit temperature.

Case C2: With local spray at the dehumidifer, the procedure is similar to case C1, but is repeated assuming that the dehumidified air is sprayed to 90% relative humdity at a constant wet bulb temperature. The spray is used if it results in less energy consumption by the main cooling coil. Typically this case occurs when the mixed air with spray off is drier than 90% RH at the cooling coil exit temperatures.

Case C3: Again the dehumidified air is mixed with either fresh or return air to obtain a minimum enthalpy, but not less than the enthalpy corresponding to 90% RH at the coil exit temperature. The main spray is used if the mixed humidity is below the humidity corresponding to 90% RH at the cooling coil exit temperature. It is assumed that the main spray is sufficient to produce 90% RH at the coil exit temperature.

The detailed model is summarized in Appendix III. Results illustrating the operation of the unit for various weather conditions are summarized in Table 2.4. As the weather changes, the performance of the unit may be limited by either available solar water temperature or the cooling tower water temperature. The energy required from both these sources is tabulated.

2.4 Solar Absorption Air Conditioning

A possible alternative to the solar dehumidification system is an absorption refrigeration unit derated to use low temperature heat supplied by solar collectors. The research divisions of York and Trane were contacted to obtain data or projections of the performance of commercial absoprtion units in the temperature range of interest. As the inlet hot water temperature is decreased, the capacity of the units deteriorate rapidly. However, the COP of the units remain constant due to the improved heat exchanger effectiveness at reduced capacity.

The selection of the optimum operating temperature would involve a detailed cost and performance trade-off. For the purposes of this, the hot water supplied to the unit was assumed to be 180°F, which is at the low end of

TYPICAL DAILY RESULTS FOR SELECTED SYSTEMS

Day	System	Energy: 10^6 BTU				Solar Water Temperature °F
		Cooling Tower	Chiller	Solar	Hot Water	
Pleasant 5/21/68 67°FDB 54°FWB 12:00	Conventional	---	25	---	---	---------
	Solar No Spray	7.5	25	9.1	---	136
	Solar Spray:Niagara	7.6	18	10.2	---	134
	Solar Spray:Chiller	4.0	2.5	10.1	---	129
Unpleasant 8/7/68 90°FDB 76°FWB 12:00	Conventional	---	98	---	---	----------
	Solar No Spray	5.7	89	10	---	134
	Solar Spray:Niagara	5.5	89	10	---	134
	Solar Spray:Chiller	4.6	87	9	---	137
Cool 5/5/68 62°FDB 53°FWB 12:00	Conventional	---	20	---	---	----------
	Solar No Spray	2.9	20	16	10	163
	Solar Spray Niagara	2.9	14	14	10	163
	Solar Spray:Chiller	3.2	2	16	6	161

TABLE 2.4

the inlet temperature range for the absorption unit. Further, it was assumed that the COP was 0.7 which lies between the values given by the two manufacturers. The COP was assumed to remain constant over the year and not vary with the temperature of the cooling tower water supplied to the abosrption unit.

Conceptually the system can be pictured as shown in Figure 2-3.

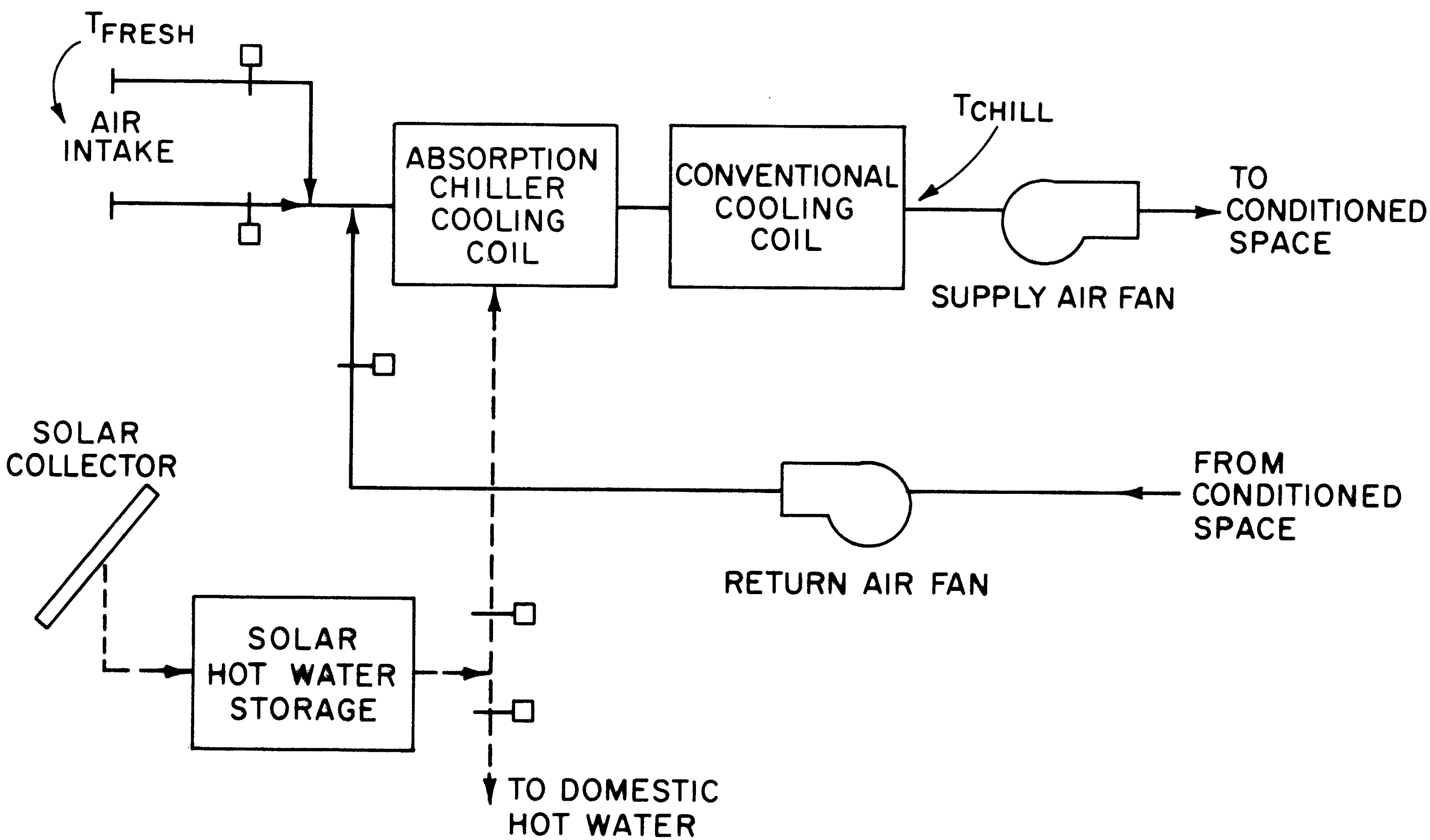


FIGURE 2-3: CONCEPTUAL BLOCK DIAGRAM OF SOLAR ABSORPTION SYSTEM

The absorption cycle was run only with solar water at 180°F or greater. Since domestic hot water is heated to only 160°F (or to 140°F between October and April) one has to make the decision whether to use storage water for domestic hot water heating, or to run the absorption chiller. Since our task is to aid in cooling the building, not just to collect a maximum of solar energy, domestic hot water is heated only when outside air dry bulb is less than the cooling coil exit temperature, for in that case there should be no demand for the chiller.

When the chiller is needed, no solar water is used until the solar water reaches 180°, and then it is used only to run the absorption chiller. The capacity of the absorption unit was assumed to be large enough to use all of the solar energy available.

The operating strategy was to use the chiller coils of the absorption unit to their maximum capacity (limited by the solar energy supply) and then use the conventional chiller coils to provide any further chilling to achieve the desired end-state for the air. Rarely does the absorption chiller handle the entire cooling load. However, the demand on the conventional cooling coil is reduced by the amount the absorption chiller could contribute. This amount was computed as this system can be run with either spray or steam humidification at the main coil. The overall system operation with the absorption chiller was optimized in the same fashion as the conventional system.

Steam was never used to supplement the solar energy to the chiller coils. The additional fan power needed to overcome the air flow of the absorption chiller coils was not considered in the operating cost.

2.5 Environmental Data

An environmental data base was constructed for use in evaluating the conventional and solar assisted systems. Weather conditions at the Central Park weather station including dry bulb temperatures and wind velocity measured hourly were obtained for the period 1960-1974 on a weather tape from Consolidated Edison Company. Solar insolation data measured hourly at Central Park was obtained from the U.S. Department of Commerce Climatic

Center for the years 1962, 1965 and 1968.

The weather and solar insolation data were combined into a single self consistent data base. This base was checked for internal consistency, i.e., the solar radiation was found to decrease in cloudy, rainy weather, etc. The hourly data for the year 1968 was used for most of the comparative analyses of system performance. The weather data for 1968 was checked against monthly ten year averages and found to have monthly average temperatures which did not deviate from ten year averages by more than 15%.

The average daily solar insolation data for 1968 was also compared for each month with multiyear averages. As shown in Figure 2.4, the data for 1968 is close to data based upon multiyear averages, and over the entire year the average daily insolation for 1968 was 325 Langleys per day based upon a multiyear average.

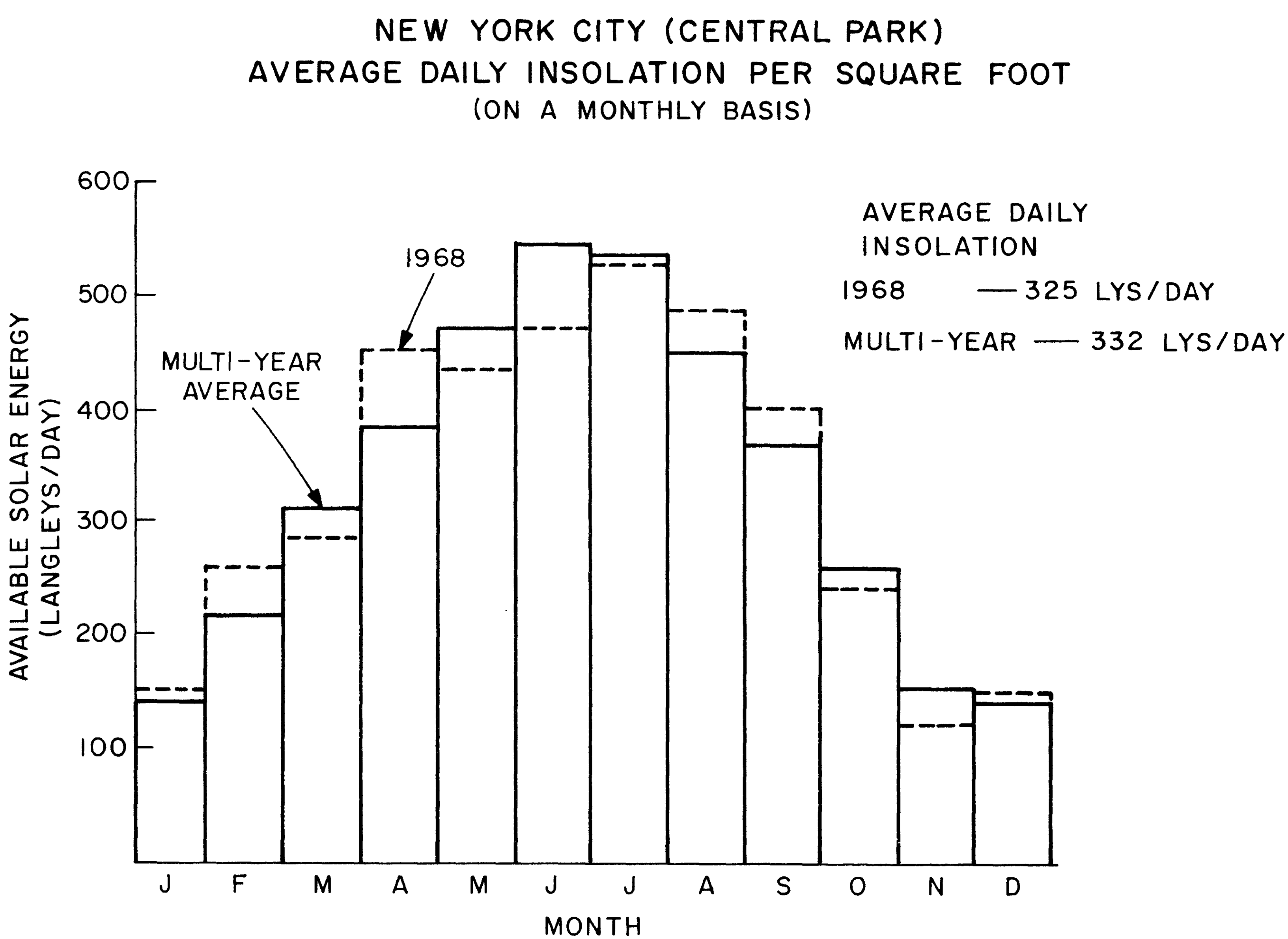


FIGURE 2-4: AVERAGE DAILY INSULATION IN EACH MONTH FOR THE YEAR 1968 AND FOR MULTIPLE YEAR

III. Simulation of Conventional and Solar Assisted Systems Operation Over Periods of One Year

3.1 Operational Simulation

A computer simulation was developed to evaluate the comparative performances of the conventional and solar assisted systems described in Chapter II. The simulation was developed to determine on an hourly basis the operation of a specified system over the period of a year*. The basic components of the simulation are summarized in Fig. 3-1 and have been described in detail in Chapter II. The simulation program was developed by combining:

(1) Environmental Weather and Solar Data

(2) Solar Collector Models

(3) Conventional and Solar Assisted HVAC System Models

(4) Building Load Models

The specific cases for which simulations have been run are summarized in Table 3.1. Each system was simulated using the environmental data from 1968 and using a building load model based upon a flow rate of 0.75 CFM per square foot of space. The detailed operating strategy for each system and the levels of comfort attained in the building are described in Chapter II. For the systems without spray, steam humidification is used in the winter to keep the relative humidity about 30% in the building. In the summer the various systems achieve slightly different levels of humidity, however, with all the systems the relative humidity is maintained below 60%, thus year round the humidity is maintained between 30-60% for all systems. Year round the dry bulb design temperature was 74°F. At no time is external heat required to raise the temperature because the building load requires cooling to reach 74°F at all times of the year.

Each of the HVAC systems were operated in the following manner:

(1) The HVAC systems were run Monday-Friday from 8 a.m. to 6 p.m. and Saturday from 8 a.m. to Noon.

(2) The HVAC systems were not run from Saturday noon until Monday morning and were not run on legal holidays. When a solar

*The simulation may also be run for shorter time periods such as a month or a day.

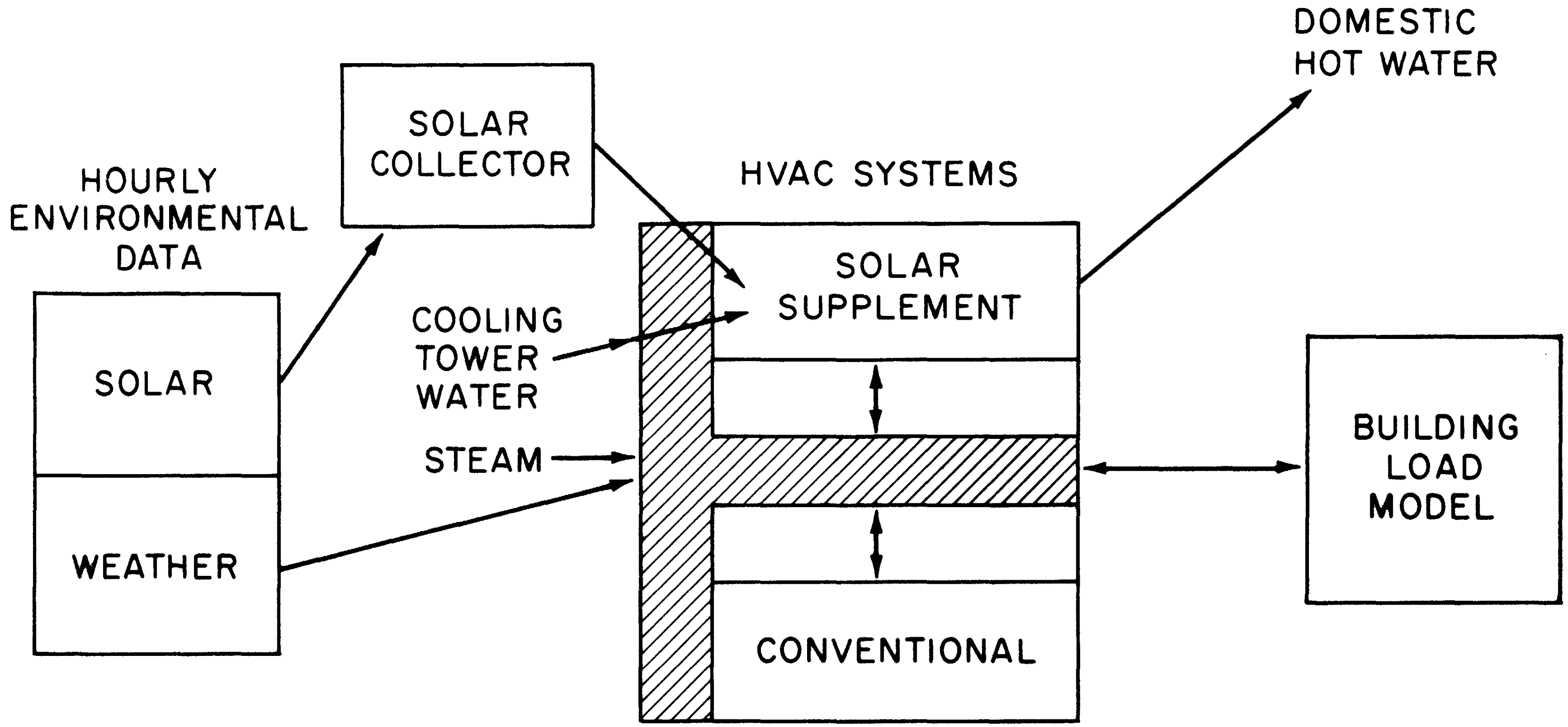


FIGURE 3-1: BASIC ELEMENTS IN SYSTEM COMPUTER SIMULATION

Table 3.1

HVAC Systems Selected for Simulation

<u>Case #1</u>	Conventional HVAC system with optimal mixing of outside and return air to minimize cooling coil steam requirements while maintaining building conditions. Case 1(a) No spray 1 (b) Use of spray at cooling coil
<u>Case #2</u>	Solar assisted using two cover plate flat collector at 45° inclination and using the Niagara dehumidification unit with a cooling tower approach temperature of 7°F. Case 2(a) No spray Case 2(b) Spray at cooling coil Case 2(c) Spray after Niagara Unit
<u>Case #3</u>	Solar assisted using two cover plate flat collector at 45° inclination and using the Niagara dehumidification unit with cooling tower approach temperature of 3.5°F Case 3(a) No spray
<u>Case #4</u>	Solar assisted using two cover plate flat collector at 45° inclination and using an absorption unit with COP of 0.7 Case 4(a) No spray Case 4(b) Spray at cooling unit

assisted HVAC was turned off, solar energy was collected for use during the next operating period.

Hour by hour simulations based on 1968 weather data were run for each system in Table 3.1 and the following data computed:

(1) Space conditions achieved including temperature and relative humidity (This data was used to check the operation of the system).

(2) Amount of steam required for operating the cooling coil

(3) Amount of steam required for humidification

(4) Amount of solar energy collected

(5) Amount of solar energy available for heating domestic hot water

(6) Amount of solar energy used directly for space conditioning.

Results of the simulations are presented in the following sections.

3.2 Monthly Simulation Results

Results of the simulation runs are summarized in terms of monthly energy requirements for the four basic systems described in Table 3.1 in Figures 3-2 through 3-8. The utilization of solar energy for Case 2(a), the basic solar assisted dehumidification system and for Case 4(a), the solar assisted absorption system are summarized in Figures 3-2 and 3-3*. For both cases solar energy is used primarily to heat domestic hot water from January-April and November-December. In these time periods both types of systems use the same amounts of solar energy since the collectors are operated to yield 160°F temperature water. In the period May-October, energy in system 2(a) is used primarily in the dehumidification unit and in system 4(a) in the absorption unit. Since energy is collected on system 2(a) at 140°F and in system 4(a) at 180°F, in the May-October time period, the dehumidification system is able to collect and use more solar energy than the absorption system which must operate collectors at a higher temperature of 180°F. Over the complete year system 2(a) collects 2.2×10^9 Btu while the absorption system

*The solar energy utilization for Systems 2(b) and 2(c) and 3(a) are nearly identical to System 2(a) and for System 4(b) is nearly identical to System 4(a).

FIGURE 3-2: SOLAR ENERGY COLLECTED AND USED PER MONTH FOR SYSTEM 2(a) - A SOLAR-ASSISTED DEHUMIDIFICATION SYSTEM

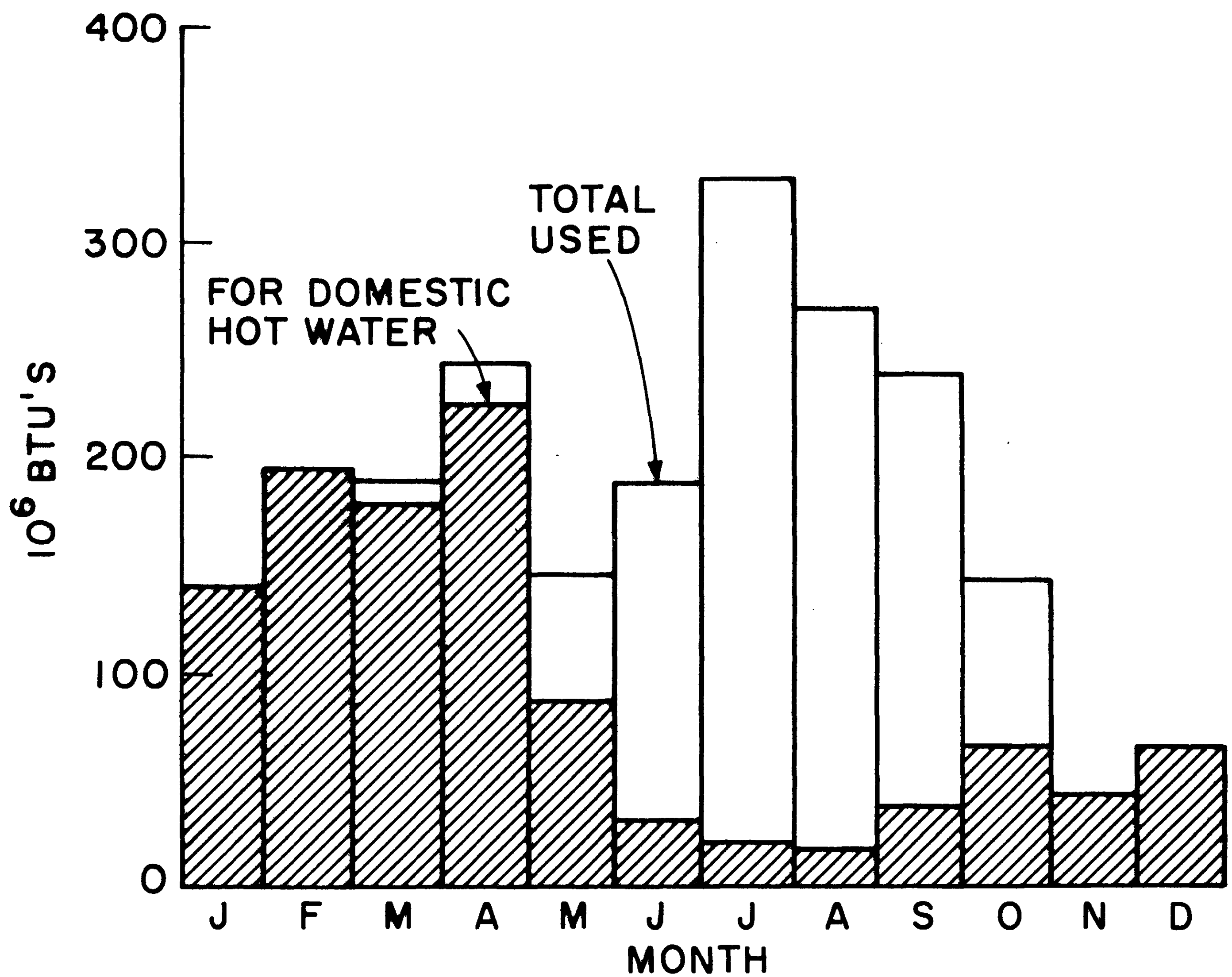
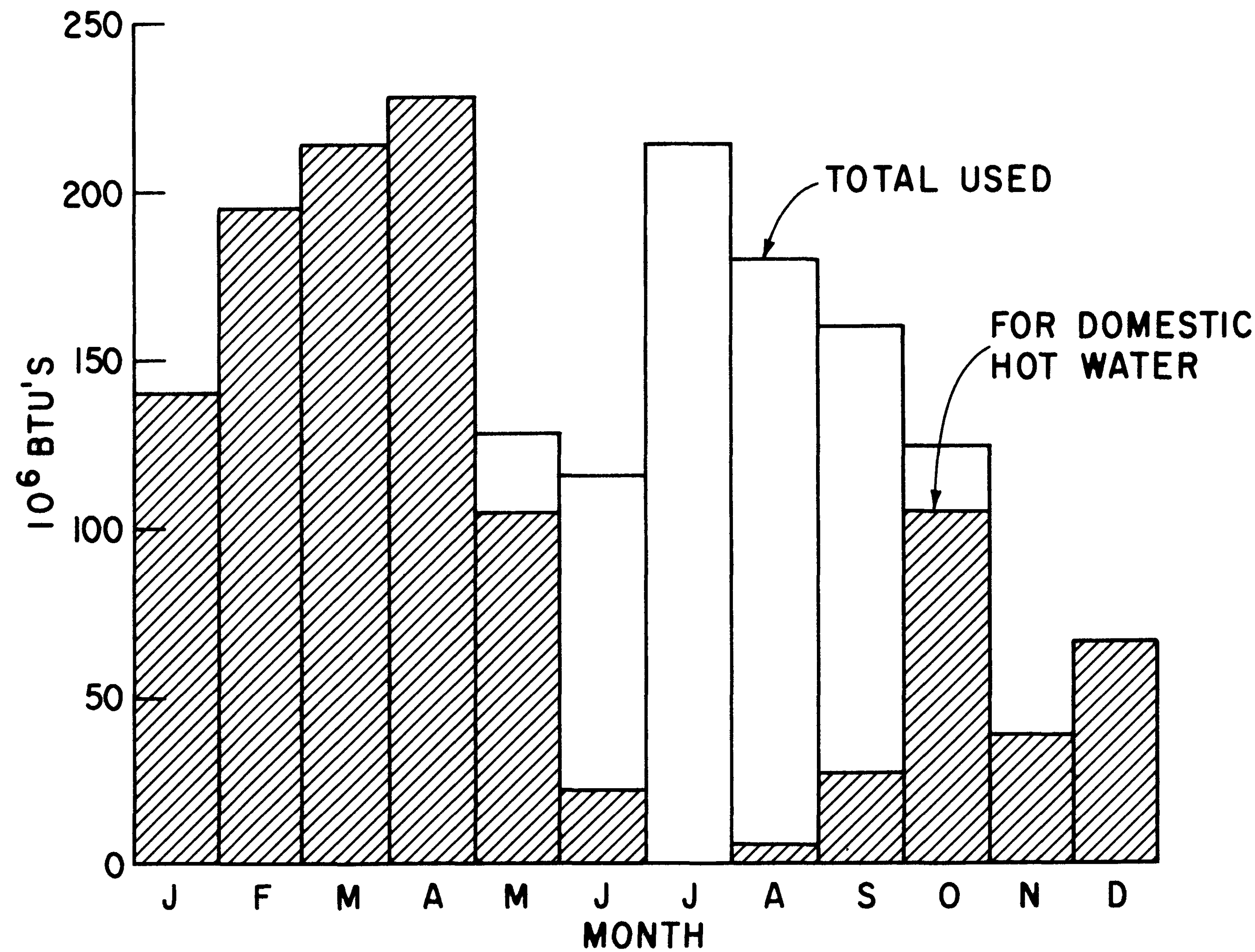


FIGURE 3-3: SOLAR ENERGY COLLECTED AND USED PER MONTH FOR SYSTEM 4(a) - a 180°F ABSORPTION CYCLE ASSIST TO CHILLER



collects 1.8 x 10^9 Btu because of the lower efficiency of collectors when operated at higher temperatures during the months May-October.

The amounts of steam energy required per month for cooling and/or humidification in the winter are summarized in Fig. 3-4 for the conventional system 1(a) and the solar assisted humidification system 2(a) with no spray use. The figure shows that primary energy consumption occurs in the period April-October and outside this period steam is used principally for humidification. In the months of June-September, the principal savings in steam energy is obtained with the solar assisted dehumidification system. For example, in July the conventional system uses about 2400 x 10^6 Btu's while the solar system uses 2225 x 10^6 Btu's.

Fig. 3-5 presents data for the systems with spray at the cooling coil - the conventional system 1(b) and the solar assisted dehumidifcation system 2(b). With the use of spray, no steam is required for humidification, i.e., 30% relative humidity may be maintained in the winter using spray. It is also noted that while the energy savings between the conventional and solar assisted systems without spray are comparable for each month to those achieved between the conventional and solar systems without spray shown in Fig. 3-4, the use of spray decreases the steam energy requirement for both conventional and solar assisted systems. For example, in July the conventional system required 2400 x 10^6 Btu with no spray and 2250 x 10^6 Btu with spray and the solar assisted system without spray required 2225 x 10^6 Btu and with spray 1980 x 10^6 Btu. Thus, the use of spray for either conventional or solar assisted systems results in an energy savings.

Fig. 3-6 presents data for the solar assisted dehumidification system when a spray is used after the dehumidification unit. This case is compared with a conventional system with no spray. The use of the spray produces increased savings in comparison to the savings achieved with no spray. However, when only spray after the dehumidification unit is used, in the winter some steam is required for humidification, thus spray at the main cooling coil produces greater savings than the use of spray at the dehumidification units.

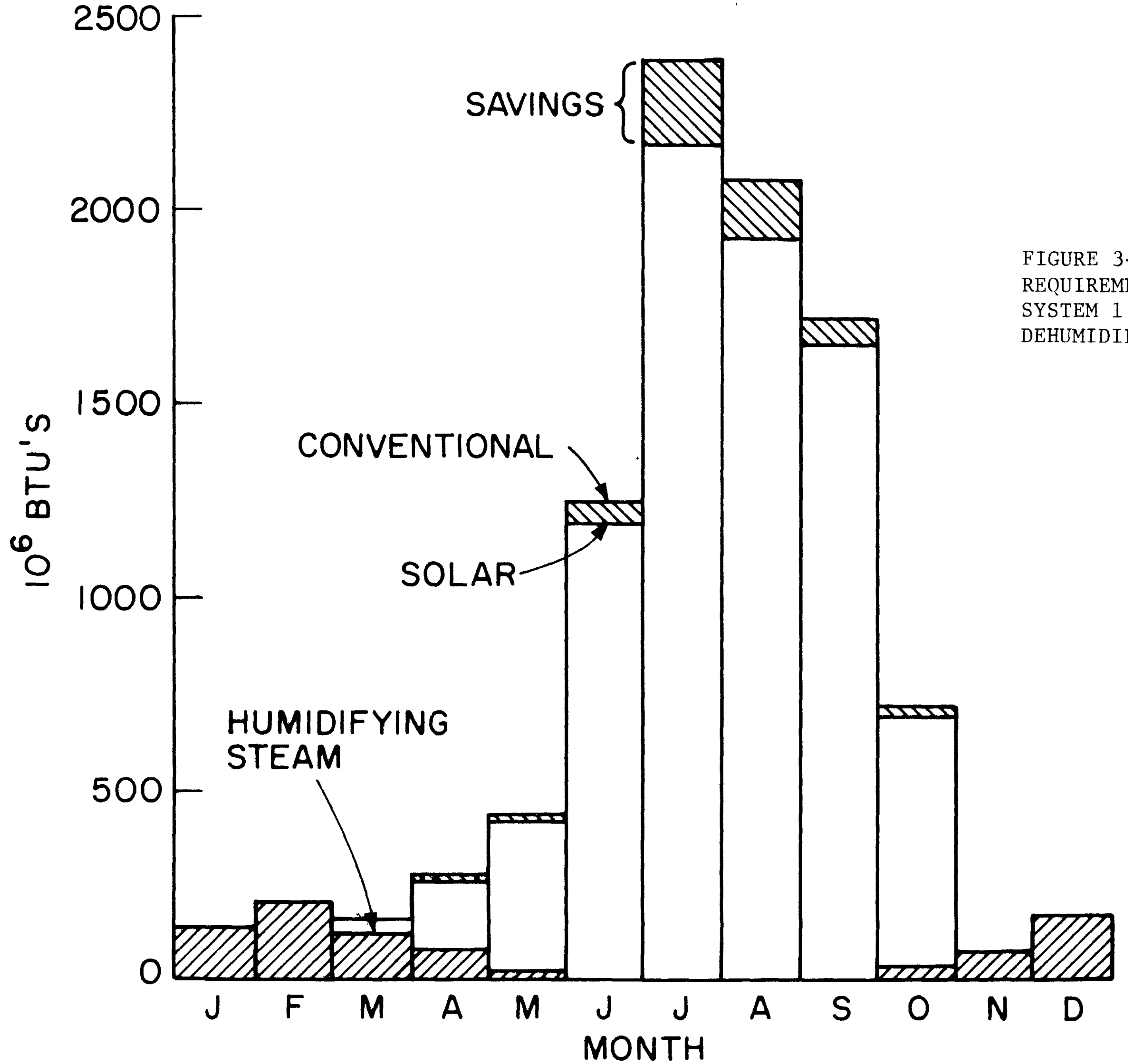


FIGURE 3-4: TOTAL STEAM REQUIREMENTS FOR CONVENTIONAL SYSTEM 1(a) and SOLAR ASSISTED DEHUMIDIFICATION SYSTEM 2(a)

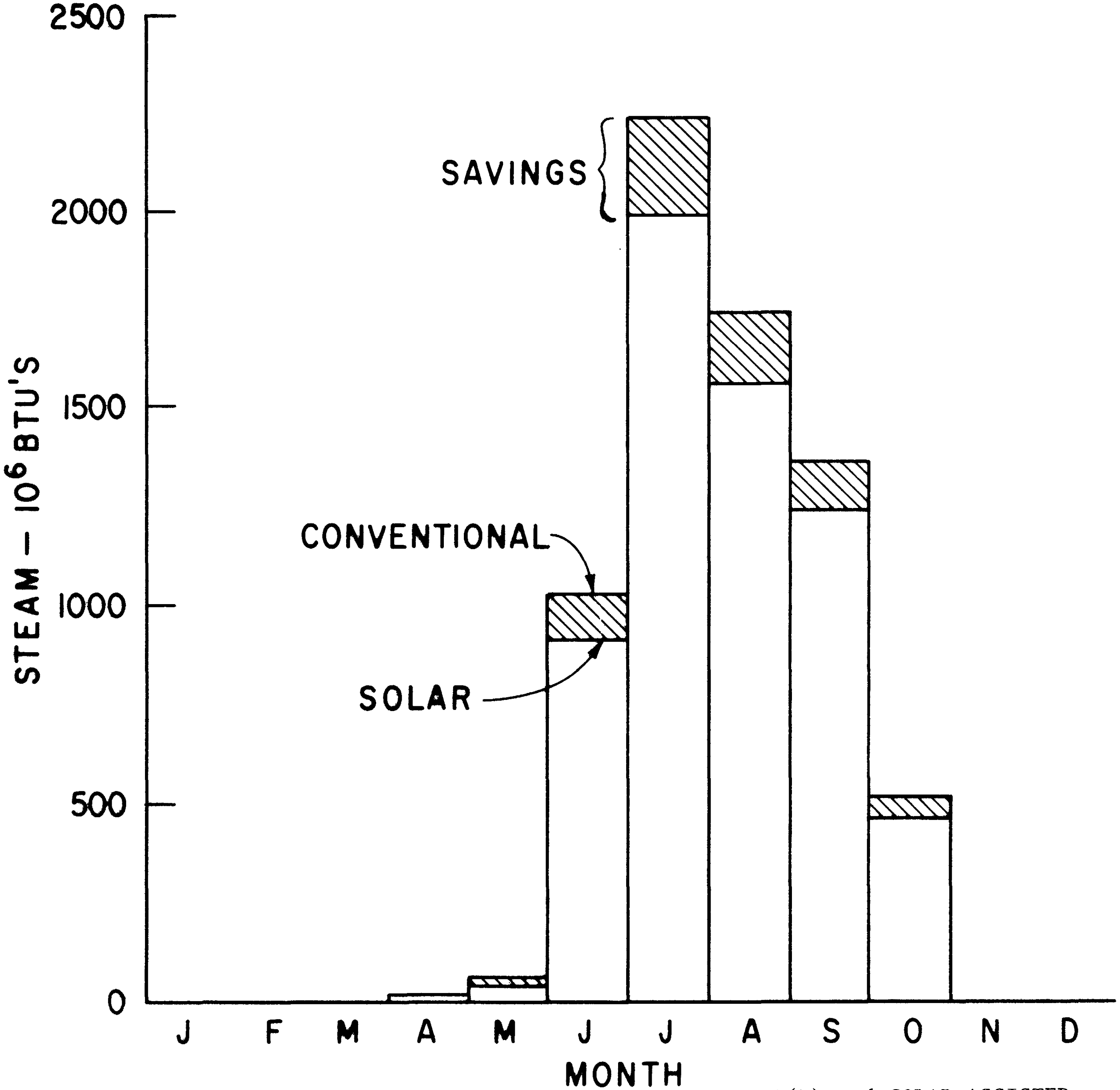


FIGURE 3-5: TOTAL STEAM REQUIREMENTS FOR CONVENTIONAL SYSTEM 1(b) and SOLAR-ASSISTED DEHUMIDIFICATION SYSTEM 2(b) WITH STRAY ON COOLING

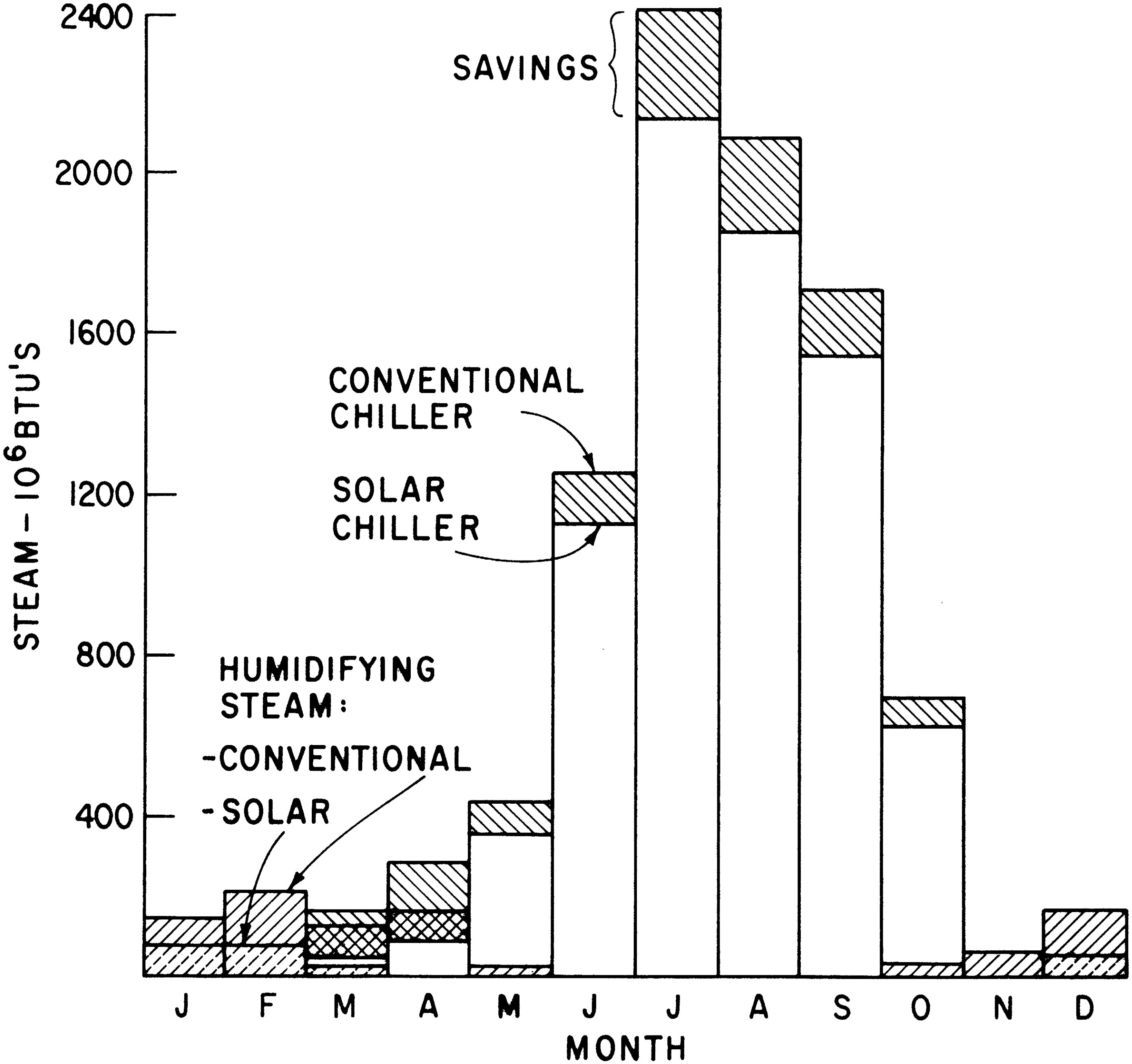


FIGURE 3-6: TOTAL STEAM REQUIREMENTS FOR CONVENCTIONAL SYSTEM 1(a) and SOLAR-ASSISTED DEHUMIDIFICATION SYSTEM 2(c) WITH SPRAY AT THE DEHUMIDIFICATION UNIT

Finally Figs. 3-7 and 3-8 present steam requirements of solar assisted absorption systems 4(a), no spray, and 4(b), spray at the cooling coil, with the equivalent conventional systems. These figures show that:

(1) Principal savings with an absorption system occur in June-October when the solar water temperature can be readily heated to 180°F.

(2) The savings in these months is comparable to those achieved with the solar assisted dehumidification systems.

(3) The use of spray at the main cooling coil is effective in reducing steam energy requirements.

3.3 Summary of Yearly Simulation Results

The solar energy collected and the steam used for the complete year are summarized in Table 3.2 for each of the basic conventional and solar assisted systems. Also included is the net consumption of energy which is equal to the steam required for the cooling coil and humidification less the credit for domestic hot water supplied by solar energy. The following results are shown by the data:

(1) In all of the solar assisted systems about one-half the solar energy collected is used to assist in heating of domestic hot water and half to assist with conditioning the building air.

(2) The savings between a solar assisted dehumidification and a conventional system is 1674×10^6 Btu with no spray on both systems and is 1670×10^6 Btu with spray on both systems at the cooling coil. Thus, the amount of savings between systems with spray and without is similar, however, both the systems with spray use about 2700×10^6 Btu less steam than both the systems without spray.

(3) The savings between a solar assisted absorption system and a conventional system is 1850×10^6 Btu with no spray on both systems and is 1851×10^6 Btu with spray on the main

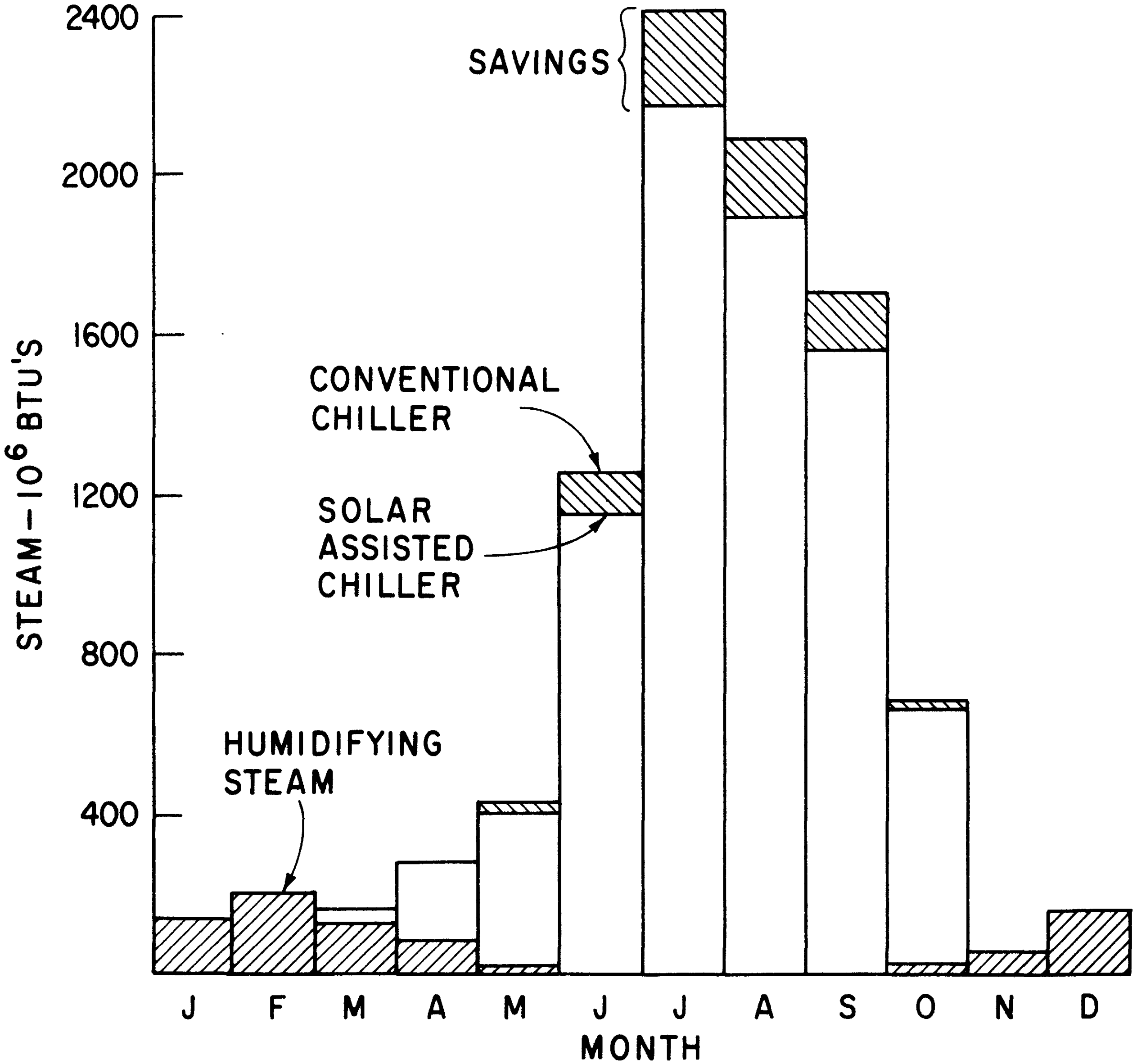


FIGURE 3-7: TOTAL STEAM REQUIREMENTS FOR CONVENTIONAL SYSTEM 1(a) and FOR SOLAR-ASSISTED ABSORPTION SYSTEM 4(a)

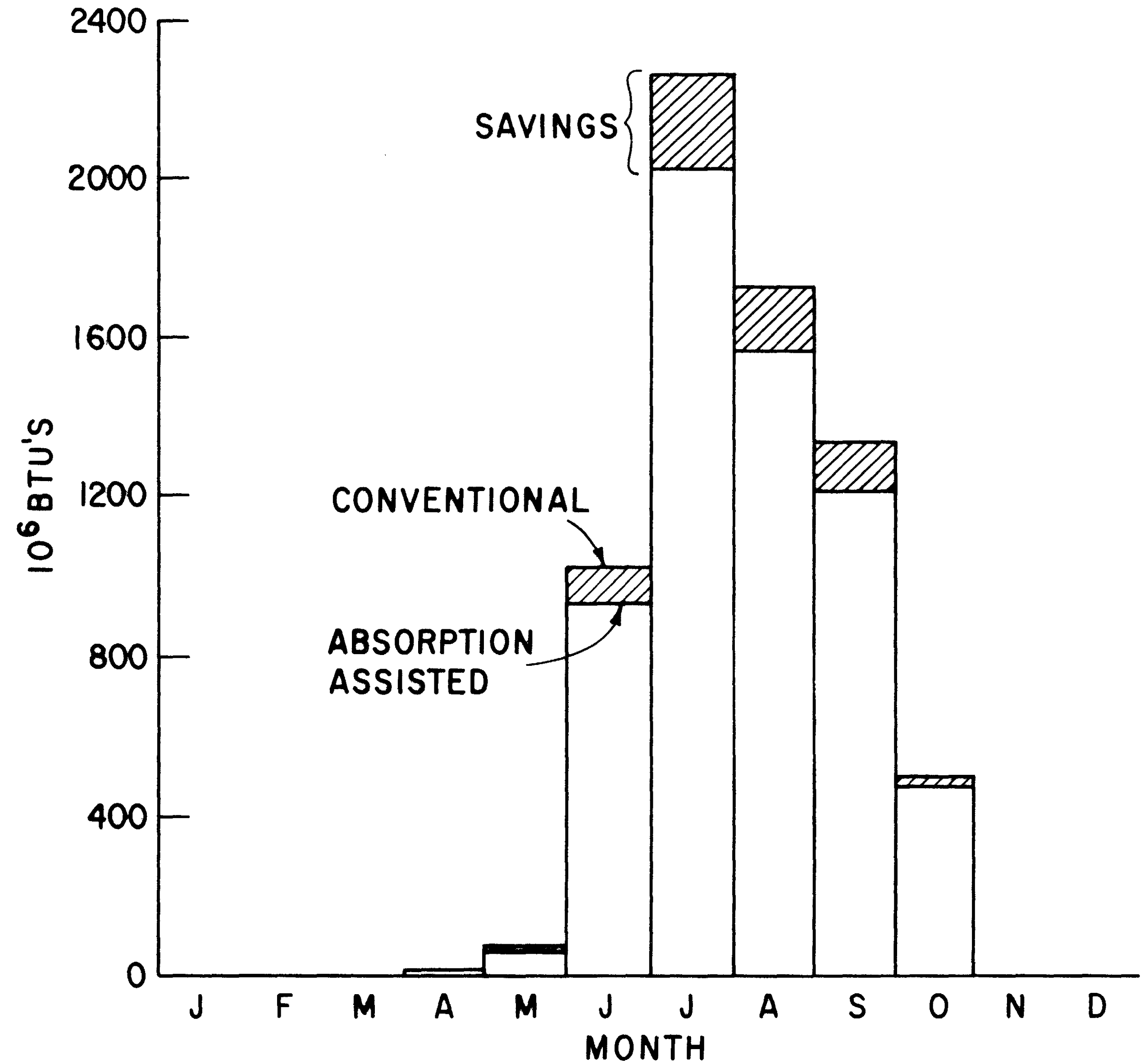


FIGURE 3-8: TOTAL STEAM REQUIREMENTS FOR CONVENTIONAL SYSTEM 1(b) and FOR SOLAR ASSISTED ABSORPTION SYSTEM 4(b) WITH SPRAY ON THE COOLING COIL

TABLE 3.2: SYMMARY - SOLAR ENERGY COLLECTED AND STEAM REQUIREMENTS FOR CONVENTIONAL AND SOLAR-ASSISTED SYSTEMS FOR 1968 WEATHER DATA (in 10^6 BTU's)

SYSTEM				SPRAY†	TOTAL SOLAR HEAT USED	PURCHASED STEAM: COOLING‡ COIL	PURCHASED STEAM: HUMIDIFI-CATION	SOLAR HEAT: DOMESTIC HOT WATER	NET PURCHASED STEAM: CONSUMPTION LESS SAVINGS
CONVENTIONAL OPTIMALLY MIXED			1(a)	NO	—	8804	827	—	9631
CONVENTIONAL OPTIMALLY MIXED			1(b)	ON COIL	—	6918	—	—	6918
SOLAR SUPPLEMENT	DEHUMIDIFIER: COOLING TOWER APPROACH TEMP. °F	7	2(a)	NO	2210	8268	820	1132	7956
SOLAR SUPPLEMENT	DEHUMIDIFIER: COOLING TOWER APPROACH TEMP. °F	7	2(b)	ON COIL	2206	6169	—	1231	4938
SOLAR SUPPLEMENT	DEHUMIDIFIER: COOLING TOWER APPROACH TEMP. °F	7	2(c)	ON DEHUMID.	2237	7802	226	1112	6916
SOLAR SUPPLEMENT	DEHUMIDIFIER: COOLING TOWER APPROACH TEMP. °F	3.5	3(a)	NO	2245	8156	827	1073	7910
SOLAR SUPPLEMENT	ABSORPTION CHILLER*	180°F	4(a)	NO	1805	8062	827	1148	7781
SOLAR SUPPLEMENT	ABSORPTION CHILLER*	180°F	4(b)	ON COIL	1774	6256	—	1189	5067

† SPRAY MAY BE ON COOLING COIL OR AT DEHUMIDIFIER
‡ REFRIGERATION COP = 0.62
* ABSORPTION UNIT COP = 0.70

cooling coil for both systems. These savings with the absorption system are slightly greater than those with the solar assisted dehumidification system. While the absorption system collects less total solar energy than the dehumidification system, the absorption system makes better effective use of the energy (COP = 0.7) than the dehumidification system and thus reduces the total steam energy required for cooling in comparison to the dehumidification system. The results for the absorption system are only approximate due to the simplification used in modeling this system.

(4) Use of spray at the dehumidification unit is not as effective as using spray at the main cooling coil and still requires use of steam for humidification in winter. However, use of spray at the dehumidification unit allows a reduction in steam requirements in comparison to system 2(a) with no spray.

(5) A reduction in the cooling tower approach temperature from 7°F to 3.5°F so that the cooler water is supplied to the dehumidification system decreases steam requirements by a relatively small amount, i.e., system 2(a) requires 7956 x 10^6 Btu while system 3(a) with the 3.5°F cooling tower approach temperature requires 7910 x 10^6 Btu*.

In conclusion the principal results of the yearly simulation show that the solar assisted dehumidification and absorption systems have similar energy savings of about 1.8 x 10^9 Btu in comparison to a comparable optimized conventional system. About half the savings is due to heating of domestic hot water. In addition it has been shown that the use of spray on the main cooling coil results in a savings in both conventional and solar assisted systems of about 2.7 x 10^9 Btu of steam.

*In the Citicorp application a separate cooling tower used solely for the solar assisted system is required if a 3.5°F cooling tower approach temperature is to be achieved.

IV. Capital Costs and Savings

4.1 Capital Costs

Cost estimates were made on the basis that the solar system is supplementary to the building's conventional system. No reduction in capacity of the conventional system was assumed because surplus capacity resulting from the contribution of the solar system or from less than design building loads would be used to service an existing building at 399 Park Avenue.

The design called for the installation of commercially available dehumidification equipment in the ductwork of the two interior air handling units in such a way that if the solar related equipment were down for any reason the system could operate conventionally with no penalty.

No pro-rata share of the cooling tower cost was assigned the solar system because it was determined that the cooling tower designed for the conventional system had sufficient capacity to handle the dehumidifiers. A separate cooling tower to handle the dehumidifiers alone was considered briefly in an effort to reduce the spray contact temperature under conditions of high wet bulb temperature, but rejected for the final design.

The concentrator units of the dehumidification system will not be supplied with supplementary steam in the absence of solar heat as was the plan in the Phase O study. The elimination of reheat in the system made it less economical to use supplementary steam in the dehumidifier than to apply it directly to the conventional system.

Surplus solar energy not needed to regenerate the liquid desiccant will be applied to the heating of domestic hot water. The conventional system design specifies steam driven semi-instantaneous hot water heaters. Two units rate at 56 gallons per minute for a temperature rise of 100°F (40°F to 140°F) will be installed on the 56th floor mechanical equipment room to serve floors 31 through 55; and two at 45 gpm on the 9th floor serving floors 9 through 30. Solar energy would preheat the supply water to these heaters.

Based on standard design criteria for commercial buildings* it was

*See for example "Sizing of Service Water Heating Equipment in Commercial and Institutional Buildings", by R.G. Werden and L.G. Spielvogel, ASHRAE Transactions, Vol. 75, p. 181 (1969).

assumed that the building hot water demand would be great enough to utilize all the surplus solar energy. Actual demand profiles for comparable commercial buildings would put this assumption on much firmer ground, but detailed data were not available to us. No surplus heat dump was designed for the system.

4.1.1 Solar Collector Costs

Because the solar collectors would replace building cladding on the crown a credit was taken for the curtain wall saved. Specifications were drawn up for the solar collectors and their installation (Appendix IV) to form the basis for budgetary cost estimates from a number of potential suppliers. Twenty firms were requested to provide a budgetary estimate if they were interested in supplying collectors for this installation. The results were disappointing. Less than half the twenty potential collector suppliers responded to our request for a budgetary estimate. The most frequent reason given for not responding was that the special nature of the collector would require departure from established production items and therefore would possibly increase costs and delay deliveries. Others felt that their current resources were too limited to propose on a project of this size. Two firms were organizing for large scale production and while they felt they would be able to deliver on our schedule they could not give an estimate at that time. Responses recieved indicated a range of \$7.50 to \$13.00 per square foot FOB jobsite in New York City. One estimate included installation and plumbing to an inlet and outlet manifold at \$500,000. From these results we would only conclude that the industry was still too immature to provide many alternatives for a building under construction. In light of this we could neither be very confident about holding to a tight schedule nor could we be sure about maintaining costs.

4.1.2 System Characteristics

The system on which cost estimates are based has the following characteristics:

Solar Collector

20,000 square feet gross area, 18,000 square feet net area performance equivalent to non-selective coating double cover, or single cover selective coating flat plate collector.

Total collector heat transfer fluid flow rate of 250 gallons/ minute. Heat transfer fluid a mixture of water and ethylene glycol. Individual panels connected in a series-parallel arrangement. Collectors to be installed from the inside. (4 panels in series horizontally, paralleled vertically).

Construction and installation to be in sequence with the completion of the building. Outside construction hoists to be available for hoisting.

Thermal storage:

Two 20,000 gallon water storage tanks. Insulated. Knocked down tanks to be hoisted by outside hoist. One 5,000 gallon domestic hot water storage tank. All storage tanks equipped with heat exchangers.

Dehumidifer:

A system consisting of two conditioners, two concentrators and two heat exchangers, each set having its individual controls, transfer pumps, flow controllers and thermostats. (Model 55009's and Model 56012's from Niagara Blower Company). Units to be hoisted knocked down by an outside hoist. An air-to-air heat exchanger for each concentrator.

Spray units:

One corrosion-resistance spray coil unit for each of the air handler units main coils.

A New York sales tax of 8% was included on purchased equipment. All material and labor costs were based on the 1975 Dodge Manual. For mechanical and electric trades, 30% has been added to cover insurance, taxes, supervision and small tools. An adjustment factor of 1.22 for labor and 1.11 for materials was used to adjust Dodge figures to New York City construction.

4.1.3 Estimated Capital Costs

The estimated capital cost of the full system is given in Table 10 as $1,727,500.

Table 4-1

SUMMARY OF FULL SOLAR SYSTEM CAPITAL COSTS

1.	Solar Collectors (FOB jobsite)	216,000
2.	Dehumidifier Equipment	203,000
3.	Air-to-Air Heat Exchangers	60,000
4.	Spray-Cooling Coil Units	150,000
5.	Installation, Tanks, Ducts, and Plumbing	930,000
6.	Control Interface with Building Management System	20,000
7.	Electrical	20,000
8.	Gratings for Access and Working Platforms	36,000
9.	Permanent Scaffolding at Interior of Crown	10,000
10.	Concrete Pads for Equipment & Cradles for Tanks	14,000
	Subtotal:	$1,659,000
11.	General Contractor, General Conditions @ 6%	99,500
12.	Consultants' Fees	169,000
	Subtotal:	$1,927,500
	Credit for replacement of curtain wall on crown with solar collector: 20,000 sq. ft. @ $10 per sq. ft.	200,000
	Total Net Cost:	$1,727,500

We also made an estimate of the cost of a half-system, i.e., 10,000 square feet of collector, 20,000 gallons of storage, dehumidifier equipment and spray units on only one of the two interior air handlers. Most, but not all the capital costs were halved for this case resulting in a total net capital cost of $1,032,500. The figure in this case to be compared with the first subtotal of the full system ($1,659,000) is $909,000.

4.2 Operating Costs

Table 4-2 shows the estimated annual operating and maintenance costs for the solar dehumidification system including spray cooling. Water consumption and treatment costs refer to the spray cooling system. Electric power consumption is due to two major components: pumping power to circulate water through the solar collectors, to the storage tanks, and then to the desiccant concentrator; and fan power to move the air through the dehumidiifer and regenerator units. The latter consumes about two-thirds of the total electric power. It is conceivable that in future projects which have a longer lead time between design and construction, that the dehumidifier and regenerator units could be redesigned to reduce fan power requirements.

Our simulations indicated that the total annual operating hours for the solar collector would be 1575 hours, and the total for the dehumidification system would be 845 hours.

Table 4-2

ESTIMATED ANNUAL OPERATION AND MAINTENANCE COSTS

1.	Maintenance	
	Spray cooling units	$1,000
	Solar equipment	5,000
2.	Water consumption	200
3.	Water treatment	1,800
4.	Electrical consumption	
	8¢/Kwhr	12,300
		$20,300

In reducing the building demand for steam, the solar system has added to the electrical consumption. From the utility's point of view, the substitution of electric power for part of the steam demand does not help moderate peak summer electric power demand. Peak steam demand occurs in winter in New York City.

If spray cooling is omitted, estimated O&M costs drop by $5,300 to about $15,000 per year.

The mechanical engineering consultants and the building operators were apprehensive about the maintainability and useful life of spray cooling units. The building at 399 Park Avenue does have spray units, and a phased replacement program is currently in progress.

4.3 Savings

4.3.1 Projection of Average Steam Costs to 1985

To arrive at dollar savings equivalent to the proejcted steam savings we projected steam costs at $11 per million Btu. In the Phase 0 study, these costs were projected at $8. Experience in the intervening period indicated that these costs should be revised upward. To make a new projection from current costs of $6.27 per thousand pounds of steam, the fuel adjustment currently at $3.10 was projected to rise to $4.10 in 1978, to $4.90 in

1980, and to $5 in 1985. If the average base price of $2.40, a surcharge of 6.1%, and the sales tax of 8% remained the same, total estimated costs of 1000 pounds of steam were: $8.45 (1978), $9.70 (1980), and $13.60 (1985). The average is about $11.

4.3.2 Summary of Savings

When total savings are compared with a conventional system, optimally mixed, no spray and the cost of steam is taken to be $11 per million Btu we have the following annual monetary savings for the various approaches:

Solar Assisted Dehumidifier	Gross Savings	O&M Costs	Net Savings
No Spray (7° Approach)	$18,400	$15,000	$ 3,400
Spray on Chiller Coil	$51,600	$20,300	$31,300
Spray on Dehumdifier Coil*	$29,900	$17,650	$12,250
No Spray (3.5° Approach)	$18,900	$15,000	$ 3,900
Absorption Chiller @ 180°F**			
No Spray	$20,400	$15,000	$ 5,400
Spray on Chiller Coil	$50,200	$20,300	$29,900

*O&M costs for spray on dehumdifier coil estimated at one-half O&M for spray on chiller coil: $2,650/yr.

**O&M costs for absorption chiller assumed to be the same as for the dehumidification units.

V. Conclusions

5.1 Application to Citicorp Center

The solar dehumidification system studied here is technically feasible with currently available equipment operating within its known performance range.

For this application, the projected annual savings of the solar part of the total system are less than two percent of the capital cost of the system. A solar absorption air conditioning system would yield similar savings at comparable capital costs. Solar system savings projected by the initial feasibility study have been greatly reduced due to improvements in efficiency of the conventional system resulting from redesign.

5.2 Other Applications

Although solar dehumidification does not appear economically feasible for this building it deserves consideration for other applications. Evaluation of this technique must take into account climatic conditions, conventional system alternatives and erection costs which may be far different than those for Citicorp Center. Solar dehumidification and solar absorption air conditioning may well be economically viable for other specific applications.

5.3 Design for Economic Viability

Because of the capital intensiveness of solar systems, it is essential to design for year-round utilization of the installation and the energy it collects. The major space conditioning load in commercial high-rise buildings is air cooling and dehumidification. The solar dehumidification system energy demand matches nicely the solar energy collected during May through October. For the remainder of the year, simulations show subtantial amounts of solar energy available for domestic hot water in winter. We did not consider using the solar heat for space heating because there are other sources available to satisfy this very nominal load. We have assumed, however, that all the solar energy not utilized in the dehumidifier could be used for domestic hot water. We have been unable to confirm this with actual hot water demand figures for a comparable building. If a restauraunt or cafeteria is part of the domestic hot water load, there is little question that all the solar heat available from Citicorp Center's collector would be utilized.

5.4 Reheat Elimination

Cost savings achieved by the solar dehumidification system were significantly decreased by the elimination of reheat in the conventional system. The simulations confirm that Citicorp Center's conventional system can operate without reheat. However, the opportunity for further energy conservation is constrained because heat loads such as lighting, fans, etc., provide essential reheat for the conventional cycle.

5.5 Spray Cooling

The simulations indicate that the use of spray cooling for removal of sensible heat during moderate outside conditions can increase the utilization of the solar dehumidifier. If conventional energy costs increase as currently projected, spray cooling can provide substantial energy and cost savings for both conventional and solar systems. Maintenance problems experienced with past spray cooling installations should be resolved before widespread application of the technique is proposed.

5.6 The Conservation Context for the Application of Solar Energy

It has become clear that the application of solar energy to space conditioning of buildings must be done in conjunction with more conventional energy conservation measures. For example, the analysis and simulation developed during this project indicated that the modification of a conventional system to use optimum mixing of outside and return air to meet air conditioning requirements and the use of a spray cooler would yield more energy savings than the addition of a solar system.

Because appreciable energy savings result from the application of conventional energy conservation measures, it is important to assure that the systems are always operated in a manner to achieve the savings they were designed to produce.

APPENDIX I

CONDENSED PHASE 0 REPORT

ANALYSIS OF CONVENTIONAL STEAM DRIVEN TURBINE REFRIGERATION, SOLAR-CHEMICAL DEHUMIDIFICATION AND SOLAR LiBr ABSORPTION REFRIGERATION SYSTEM OPTIONS FOR THE CITICORP CENTER BUILDING, NEW YORK CITY

Loring-Meckler Associates, Inc. in cooperation with M.I.T. and Consolidated Edison prepared a Phase 0 Study for Citicorp Center.

The work of Loring-Meckler Associates, Inc. was supported by the First National City Bank (Citibank), that of Consolidated Edison by the company, and that of M.I.T. by Energy Laboratory funds.

The original Phase 0 Study report was dated July 1974.

The condensed version presented here was prepared in January 1976.

ANALYSIS OF CONVENTIONAL STEAM DRIVEN TURBINE REFRIGERATION, SOLAR-CHEMICAL DEHUMIDIFICATION AND SOLAR LiBr ABSORPTION REFRIGERATION SYSTEM OPTIONS FOR THE CITICORP CENTER BUILDING, NEW YORK CITY

This study documents a comparative analysis of solar augmented space conditioning systems with the conventional system proposed for Citicorp Center currently under design by Hugh Stubbins and Associates and Emery Roth and Sons.

The energy requirements of each system option have been estimated and the feasibility of a solar energy supplement for high-rise air conditioning systems investigated. Each cooling system option was considered for application to the two interior zone air handling units serving the upper floors of Citicorp Center. Analyses were based on the following design criteria:

Space Condition	76°F D.B./50% RH - 63.4 °F W.B. -67.4 GR/LB
People - 1 Person/ 100 Sq. Ft.	245 BTU/Hr SENS 205 BTU/Hr LAT
Lighting*	2.5 Watts/Sq. Ft.
Power	0.5 Watts/Sq. Ft.
Supply Air	1 CFM/Sq. Ft.
Total Supply Air/Unit	164,450 CFM
O.A./Unit (minimum)	22,000 CFM (13.377%)
Motor Heat to Supply Air	4.89°F
Motor Heat to Return Air	2°F
Outside Air Condition	92°F D.B. 76°F W.B.

Description of System Options

The conventional system mixes outside and recirculated air and cools the mixture to the dewpoint temperature required to maintain the proper humidity level in the occupied space. This cooled and dehumidified mixture is reheated to the dry bulb temperature required to meet the interior space conditions.

The air distribution network consists of two interior zone air handling units located in the mechanical equipment room on the fifty-sixth floor. Each air handling unit has an air filter, return and outside air damper sections capable of providing a minimum or 22,000 CFM and up to a maximum of 164,450 CFM of outside air, and a cooling coil section capable of producing conditioned air at a constant volume of 1 CFM per square foot of 58° dry bulb and 54.7° dewpoint. Each air handling unit supplies cooled and dehumidified air through a medium-pressure duct distribution system extending downward in vertical shafts from floor fifty-six. A reheat coil at each floor heats the supply air to 67.42°F.

The process is illustrated graphically in Figure 1.

*60% of Load to Room - 40% of Load to Plenum

Citicorp Center employs three steam turbine driven 2000-ton centrifugal chillers located in the mechanical equipment room on the tenth floor. A central chilled water distribution network provides chilled water to cooling coils in all central air handling units. A diagram of the conventional system is shown in Figure 2.

An alternative system employs an absorption chiller operated on solar heated water at 200°F to produce chilled water to supplement that from the steam turbine refrigeration plant, Figure 3.

A third alternative is to dehumidify outside ventilation air to a dewpoint temperature sufficiently low that, after being mixed with the recirculated air, the design humidity level of the occupied space is maintained. The liquid desiccant is cooled by the cooling tower to remove the heat of condensation. In this way, some equivalent refrigeration is obtained from the cooling tower. (Chilled water is not used in the dehumidifier although it could be.) Outside air and recirculated air are then mixed and cooled by a chilled water coil to the dry bulb temperature required by the interior space cooling load. The chilled water is produced by the steam turbine chillers. Because the air is not supercooled when dehumidified by a desiccant, reheat is not required. Only the required minimum outside air is dehumidified. The liquid desiccant is regenerated (dried) continuously by circulating the diluted desiccant through the regenerator (concentrator) where it is heated and sprayed into the exhaust air stream to remove the water. This heat can be provided by steam, Figure 4, or by solar hot water at a temperature of 140°, Figure 5. Solar heat can be supplemented by steam when required.

COMPUTER SIMULATION

Solar energy from the collector was computed utilizing a New York City weather data tape prepared by the National Oceanic and Atmospheric Administration (NOAA). (U.S. Department of Commerce) for Central Park, New York City.

PSYCHROMETRIC CHARTS FOR THE SYSTEMS STUDIED

The psychrometric chart for the conventional system, Figure 6, has outside air entering the system at point 1 and, when mixed with return air from the occupied spaces, the mixture reaches point 6. The sensible heat from the plenum and return air fan motor is added to the system air to create the final return air temperature entering the air handling unit, point 5.

The mixed air is then chilled and dehumidified by the chilled water coil to point 2, the cooling coil leaving air temperature. Sensible heat produced by the supply fan motors heats the supply

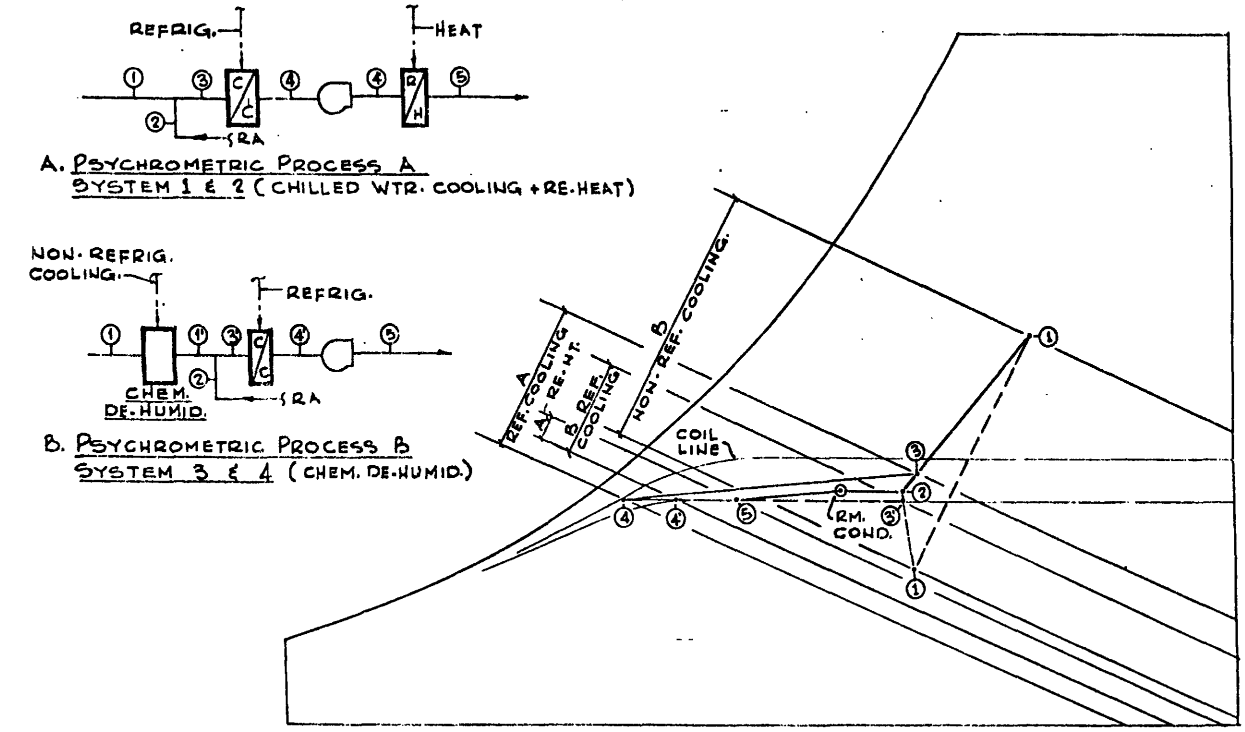


FIG. 1 - PSYCHROMETRIC PROCESS DIAGRAM

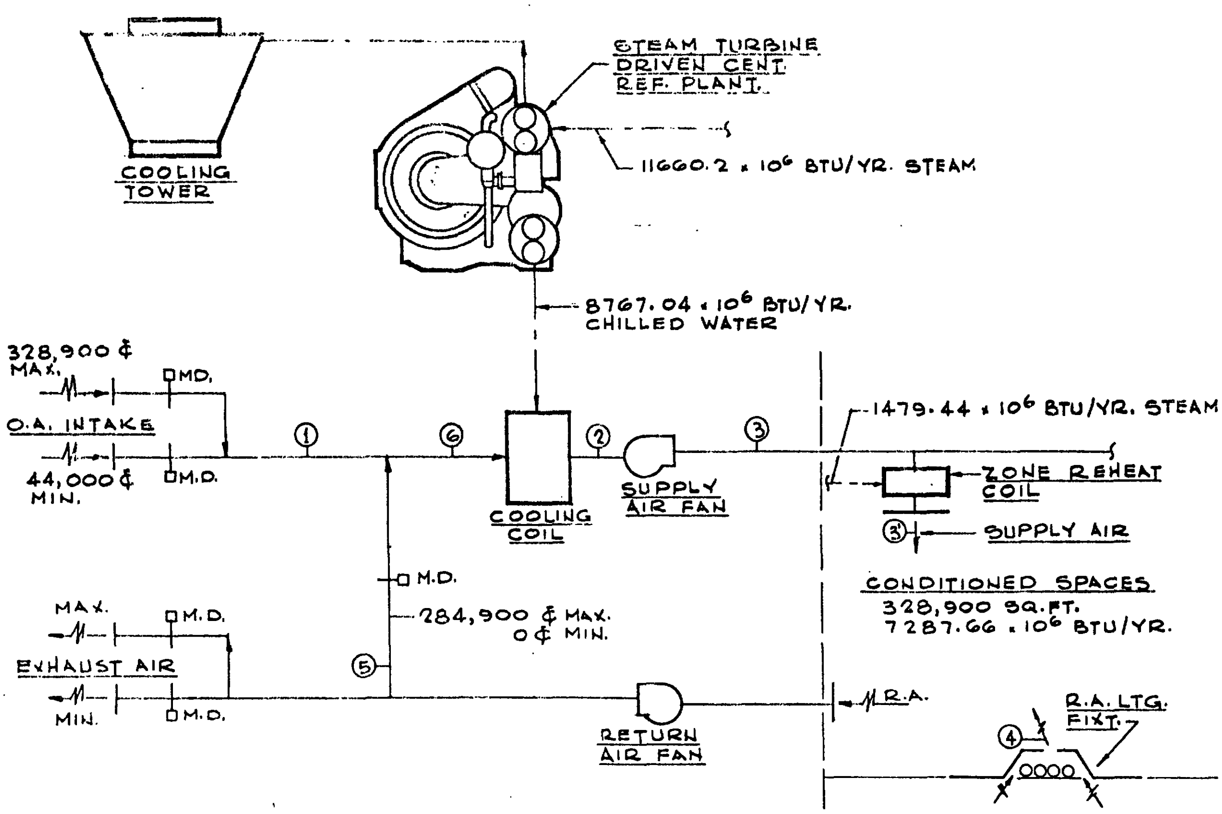


FIG. 2 - CONVENTIONAL SYSTEM ENERGY FLOW DIAGRAM - (CHILLED WATER COOLING)

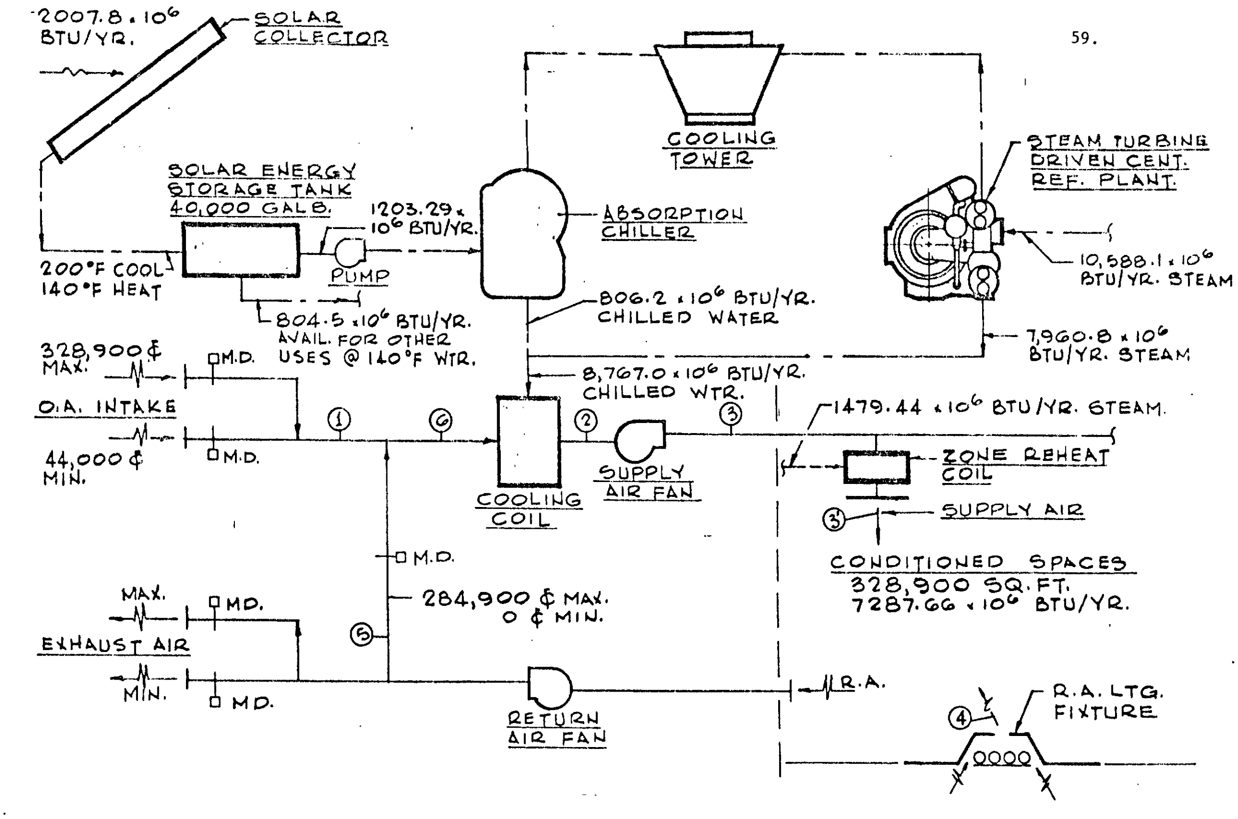


FIG. 3 - ENERGY FLOW DIAGRAM USING SOLAR ENERGY WITH ABSORPTION CHILLER (CHILLED WATER COOLING)

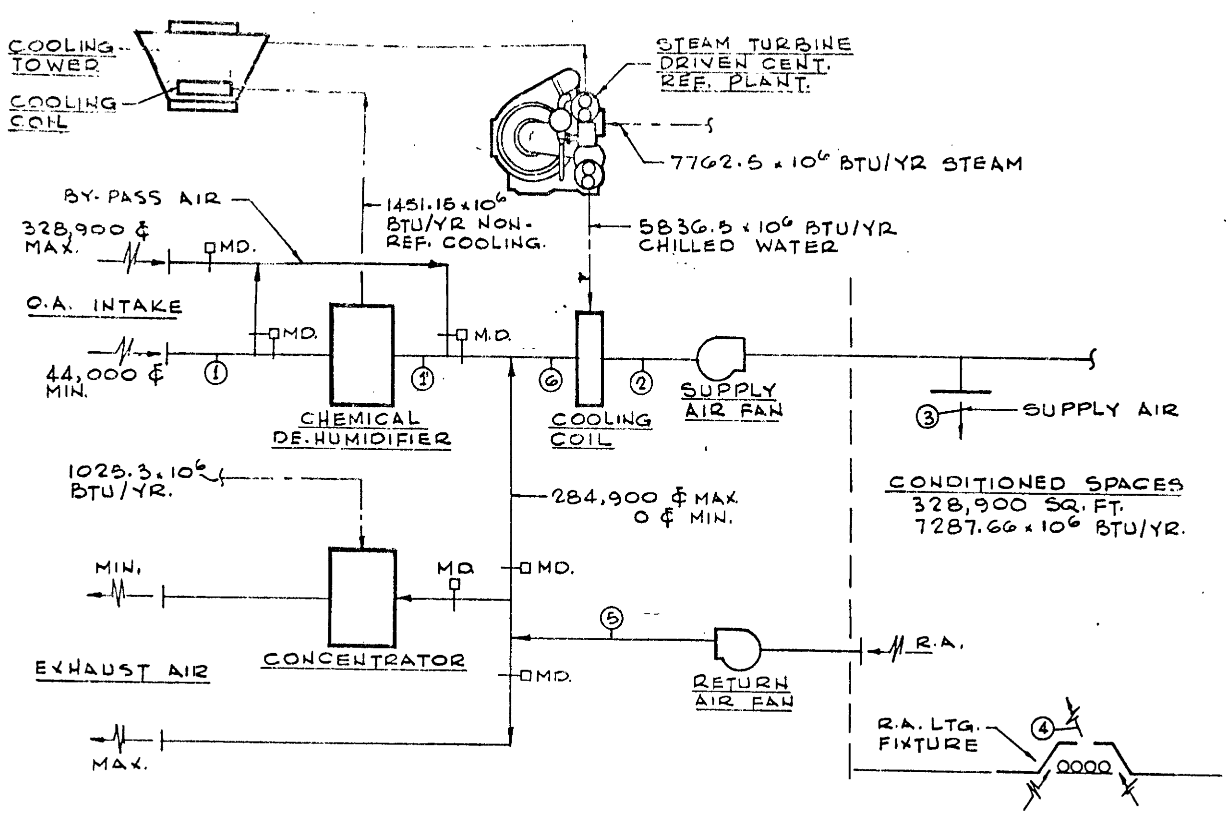


FIG. 4 - STEAM CONCENTRATED CHEMICAL DEHUMIDIFICATION

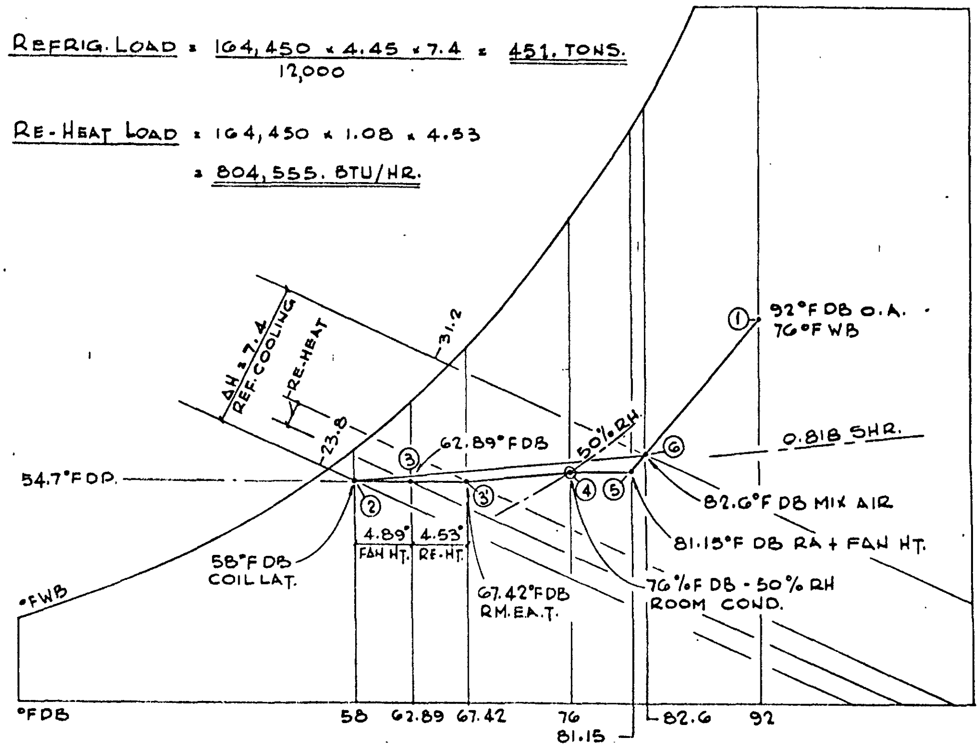


FIG. 6 PSYCHROMETRIC CHART - CONVENTIONAL SYSTEM (CHILLED WATER COOLING + RE-HEAT)

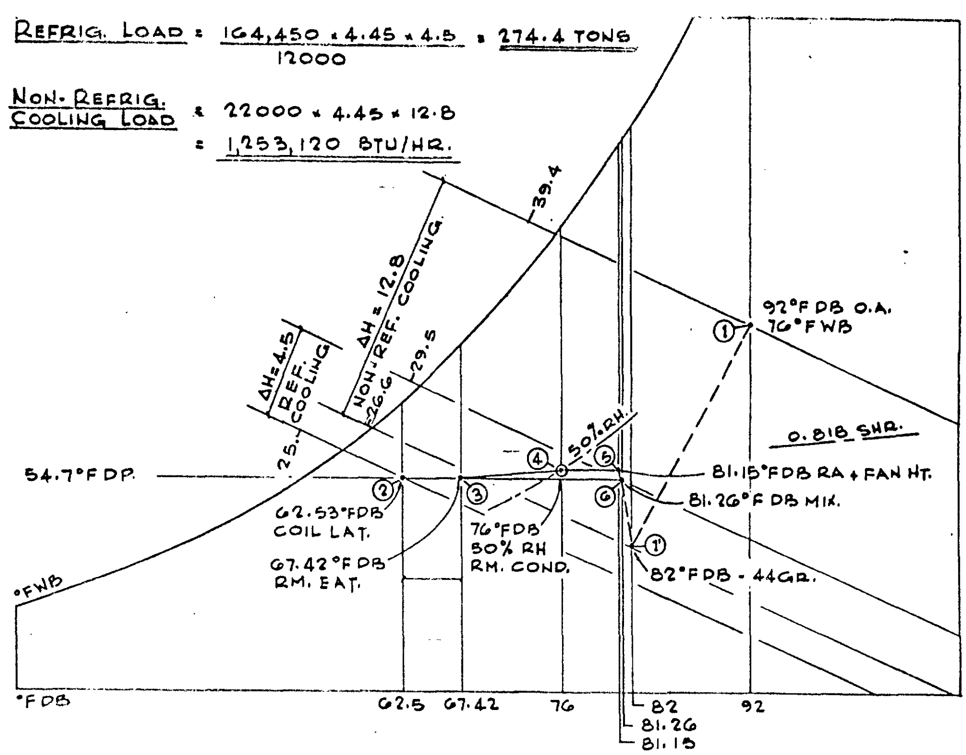


FIG. 7 PSYCHROMETRIC CHART (CHEMICAL DE-HUMIDIFICATION)

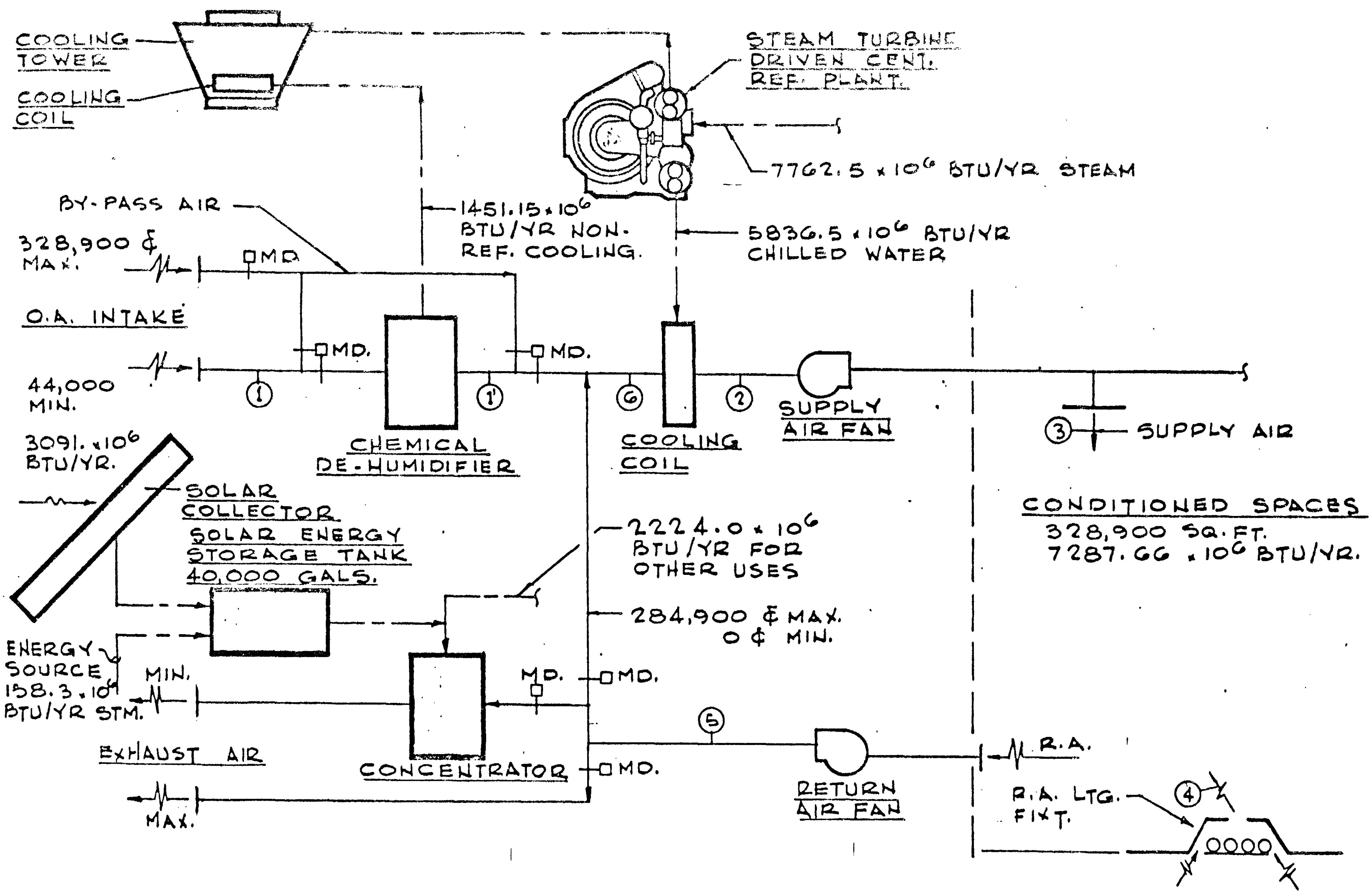


FIG. 5 - SOLAR CONCENTRATED CHEMICAL DEHUMIDIFIER

air to point 3. The reheat* necessary to maintain the space design conditions is indicated by the line from 3 to 3'.

The building load is represented by the line between 3' and 4.

In the dehumidifier system the minimum required outside air for building ventilation enters the dehumidifier at point 1, Figure 7, and is cooled and dehumidified by a liquid desiccant sprayed into the air stream. The desiccant is cooled to 83° by the cooling tower, point 1'. (The non-refrigeration cooling indicated is that provided by the cooling tower). This dehumidified air is then mixed with return air, which has been heated by the fan motor at point 5 to produce a system mixed air condition at point 6. The relative humidity of the mixed air at this point is under 50%; dehumdification is unnecessary. The chilled water coil therefore has only to cool the system air to satisfy the sensible load requirements as represented on the chart between points 6 and 2.

Because the cycle path passes through the room entering air condition at point 3, it is only necessary to cool to point 2 to compensate for heat from the supply fan motor. The sensible and latent heat load of the conditioned space then takes the path from point 3' to point 4' to complete the cycle.

The constant volume air circulation system can now meet the cooling demands without reheat. Variation in the space conditions will not affect the system. Moreover, any reduction of interior heat gain through further conservation measures will result in refrigeration load savings. Systems using reheat must compensate, at least in part, for any reduction in occupied space heat gain.

When compared with the conventional system requiring reheat chemical dehumidification produces significant energy savings. It appears feasible to regenerate the chemical desiccant with low temperature solar heat, thus increasing solar collector efficiencies, or with steam to effect energy savings.

CHEMICAL DEHUMDIFICATION

Moist air may be dehumidified by spray contact with a liquid desiccant. A diagram of one system is shown in Figure 8. The system has two sections, one for dehumidifying and one for desiccant regeneration (concentration). In the dehumidifying section, moist outside air is dehumidified by spray contact with the absorbent solution over cooling coils maintained at the cooling tower water temperature. The desiccant solution is circulated between the sump of the dehumidifying section and that of the regenerator (concentrator). The desiccant is heated to 140°F by heating coils in the regenerator and is sprayed so the evaporated water is removed by building exhaust air. The hot water for regeneration may be obtained either from steam or from solar collectors.

The extent of dehumidification depends on the concentration and temperature of the desiccant. Moisture is absorbed from the air by the solution as long as a vapor pressure difference exists between the air and the desiccant.

The moisture content of the air leaving the dehumidifer can be controlled by varying the concentration (density) of the water-desiccant solution.

The heat of absorption of moisture from the outside air is mainly the latent heat of condensation of water vapor. This heat is dissipated by circulating the solution through a closed circuit coil cooled with cooling tower water.

The regenerator has an air-to-air recovery system. A heat exchanger in the regenerator exhaust air stream preheats the inlet air to the regenerator which improves the energy efficiency of the process. A liquid-to-liquid heat exchanger is placed in the pumping circuit between the sumps of the dehumidifier and the regenerator.

In this application outside air is dehumidified more than in a conventional system before being mixed with return air. This permits control of the humidity of the mixture. The mixed air is then cooled by chilled water cooling coils to the desired temperature.

In figure 9, an example of chemical dehumidification process is shown diagrammatically. In this example Hygrol (triethylene glycol) is the liquid desiccant. For the Citicorp building the outside air design condition is 92°F D.B. and 76°F W.B. With a cooling tower approach of 7°, the spray contact temperature will be 85°F. At 92.5% Hygrol concentration the moisture drops from 110 grains to 44.1 grains, which is lower than that obtained with a conventional system, thus affording space latent heat control when mixed with building return air.

Figure 9a shows the system schematically. The computer simulation consists of:

A. A dynamic solar collector model developed jointly, by the Westinghouse Geophysics Laboratory at Boulder, Colorado and the Westinghouse Center for Advanced Studies and Analysis, Falls Church, Virginia;

B. A dynamic HVAC load model for each of the system options studied, developed by Loring-Meckler Associates, Inc. Each system option was simulated for all operating hours of the year to establish annual energy use.

*During the early design stages consideration was given to bypassing the cooling coil with some of the mixed air, possibly to reduce or eliminate the necessity for reheat. An air bypass arrangement causes humidity or latent heat removal problems, and the control required offers no flexibility for maintaining room conditions or compensating for potential possible variations. This is particularly true for the constant volume system which was selected for this building to maintain a good air circulation within the occupied space. For these reasons, reheat was retained.

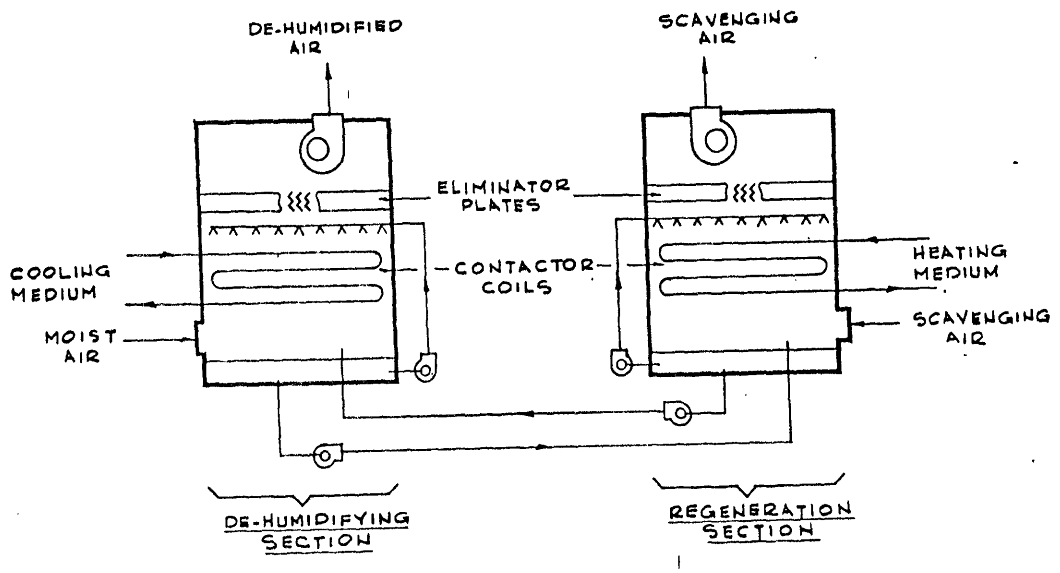


FIG. 8. SCHEMATIC DIAGRAM OF A CHEMICAL DE-HUMIDIFICATION SYSTEM.

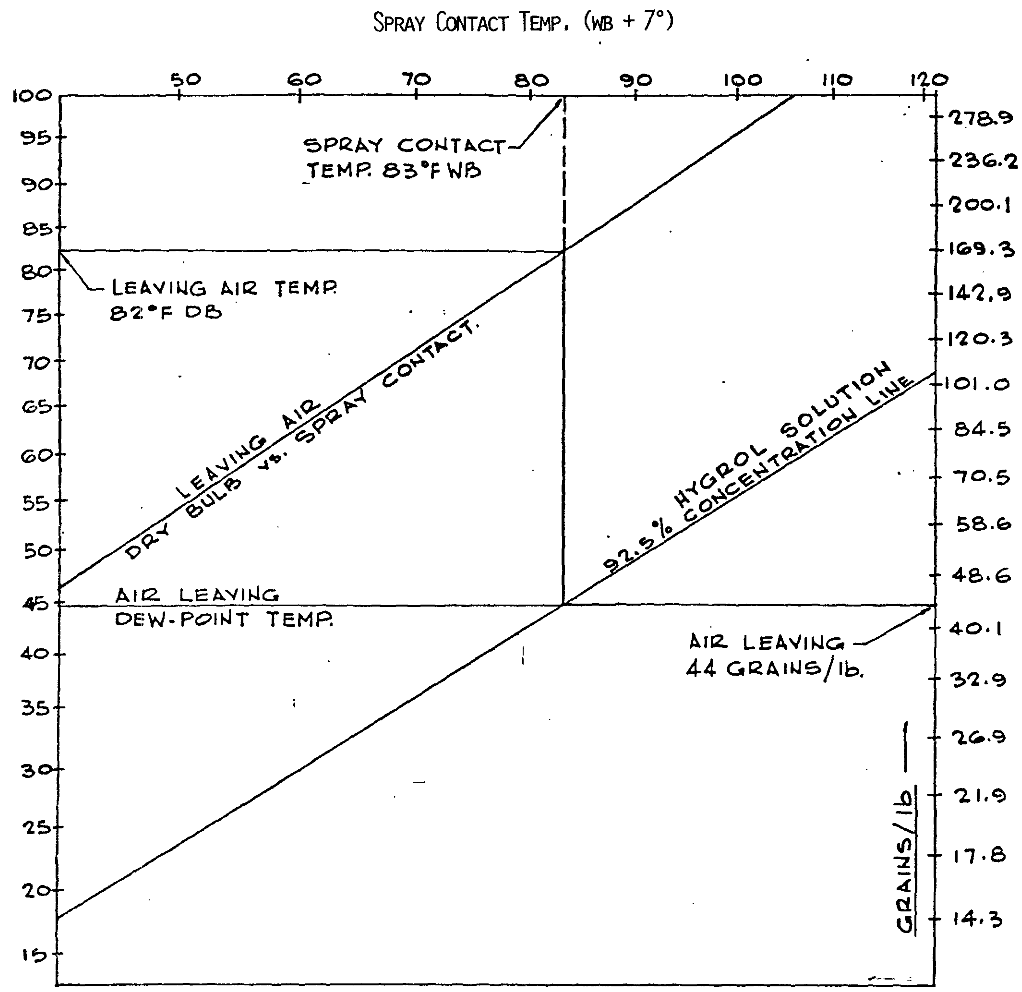


FIG. 9 - CHEMICAL DE-HUMIDIFICATION PROCESS

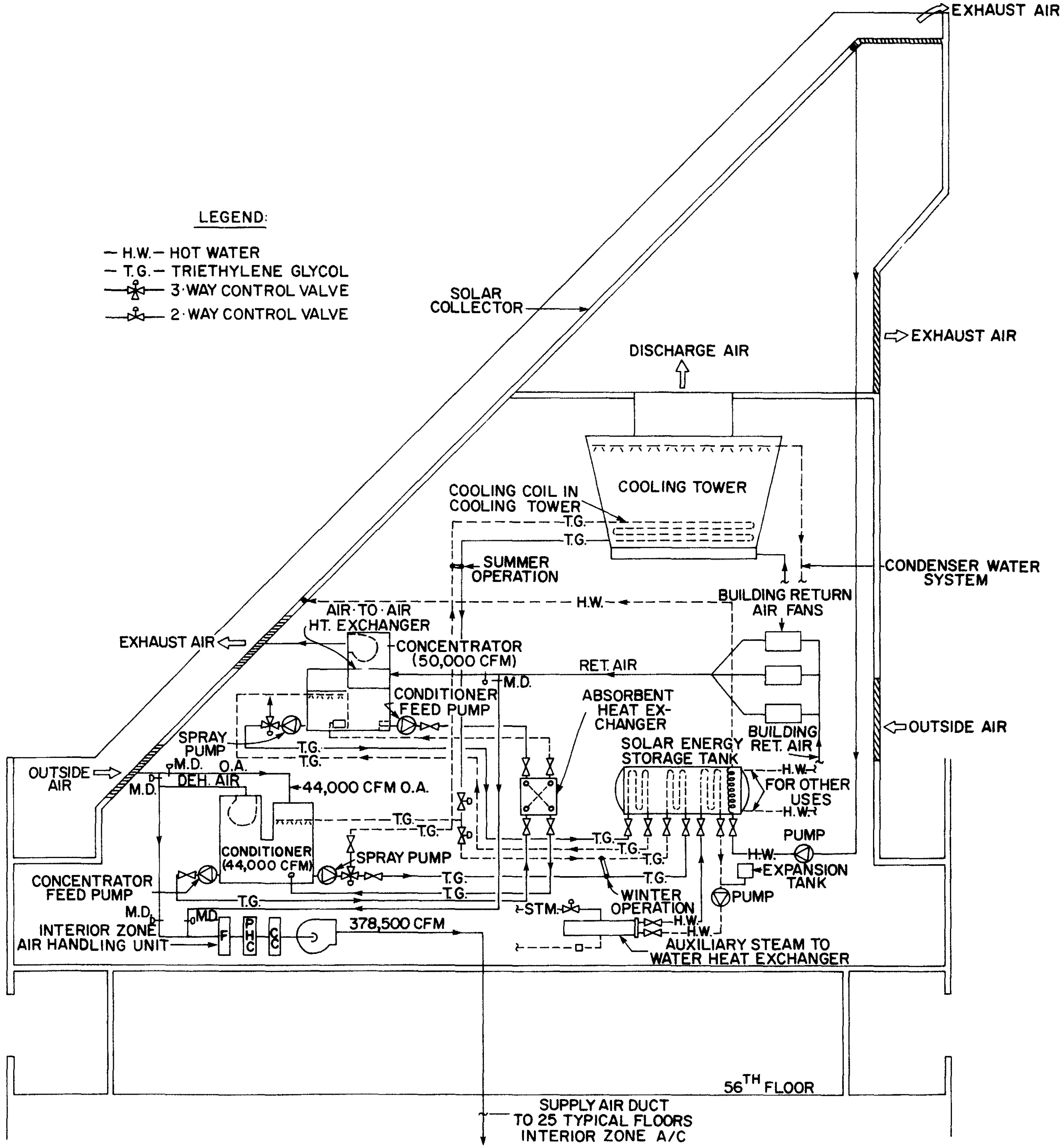


FIGURE 9A; PHASE O SYSTEM SCHEMATIC DIAGRAM

APPROXIMATION OF SOLAR ENERGY AVAILABILITY

Solar energy available in New York City was calculated using "monthly modifying factors" on the hourly clear day direct insolation.

1. The total monthly clear day direct insolation on a horizontal surface at 40° North Latitude was calculated using data given in the ASHRAE Handbook of Fundamentals. The average daily clear day direct insolation was then calculated.

2. The average daily actual insolation (both direct and diffuse) was obtained from the weather data tape.

3. ASHRAE data is only direct insolation. Five percent of the average actual insolation was assumed to be diffuse radiation. Monthly modifying factors were then determined as the ratio of the average actual direct (95% of the average daily actual) insolation for the month to the average clear day direct insolation for the month. The factors used are listed below.

MONTHLY MODIFYING FACTORS FOR NEW YORK CITY CENTRAL PARK

January	- 0.69	July	- 0.75
February	- 0.71	August	- 0.73
March	- 0.71	September	- 0.76
April	- 0.71	October	- 0.75
May	- 0.73	November	- 0.66
June	- 0.75	December	- 0.68

This procedure to calculate cloud cover modifier has been developed and published by ASHRAE*.

SOLAR COLLECTOR SIMULATION

The solar collector modeled is a conventional double glazed flat plate collector with the following parameters:

PARAMETERS USED IN SOLAR COLLECTOR ANALYSIS

Collector emissivity	.2
Collector absorptivity	.92
Emissivity of inside glass	.9
Emissivity of outside glass	.9
Collector azimuth angle	209°
Collector tilt angle	40°
Back and edge heat loss rate-10% of radiation losses	
Efficiency of fin design	.90
Flow rate efficiency	.96
Shading and dirt loss factor	.96

Shading and dirt loss factor, flow rate efficiency, etc. were obtained empirically by Westinghouse.

In the calculation of useful energy collected by a solar collector, the following procedures were followed:

1. Compute the internal temperatures and heat loss rates of the collector as established by ambient weather conditions.

2. Determine the incident angle of the sunlight on the collector from the sun position and collector orientation.

3. Determine the insolation rates, reflection and transmission losses at the incident angle and loss conditions.

4. Using the flow and fin efficiency factors, determine the useful energy absorbed by the heat transfer fluid.

A typical computer output showing the values related to solar energy, for one day only is given in the Table below.

M	D	HR	SOL	ANGLE	QU	CEFF	SOLAR
7	21	1	0.00	0.00	0.00	0.0	0.
7	21	2	0.00	0.00	0.00	0.0	0.
7	21	3	0.00	0.00	0.00	0.0	0.
7	21	4	0.00	0.00	0.00	0.0	0.
7	21	5	0.00	0.00	0.00	0.0	0.
7	21	6	11.93	117.96	0.00	0.0	0.
7	21	7	133.67	104.33	0.00	0.0	0.
7	21	8	189.27	90.42	0.00	0.0	0.
7	21	9	216.12	76.40	0.00	0.0	0.
7	21	10	234.29	62.39	1.84	.8	.369E+05
7	21	11	242.08	48.54	44.10	18.2	.882E+06
7	21	12	246.05	35.17	80.93	32.9	.162E+07
7	21	13	251.26	23.22	106.85	42.5	.214E+07
7	21	14	249.88	16.29	115.23	46.1	.230E+07
7	21	15	245.77	20.43	107.95	43.9	.216E+07
7	21	16	235.11	31.52	84.99	36.2	.170E+07
7	21	17	220.49	44.64	50.63	23.0	.101E+07
7	21	18	193.18	58.40	8.86	4.6	.177E+06
7	21	19	106.60	72.38	0.00	0.0	0.
7	21	20	.55	86.41	0.00	0.0	0.
7	21	21	0.00	0.00	0.00	0.0	0.
7	21	22	0.00	0.00	0.00	0.0	0.
7	21	23	0.00	0.00	0.00	0.0	0.
7	21	24	0.00	0.00	0.00	0.0	0.
DAILY TOTAL			2776.22		601.38		.120E+08

*Procedure for Determining Heating & Cooling Loads for Commercial Energy Calculations, Algorithms for Building Heat Transfer Subroutines, ed. M. Lokmanhekim, 1971.

M - Month, D - Day, H - Hour, SOL - Energy Rate BTU's per hour—Sq. Ft. for 100% efficient collector at normal incidence, ANGLE - Solar Collector Angle of Incidence, QU - Energy Rate of Actual Collector, CEFF - Solar Collector Efficiency (Collected Solar Energy ÷ Incident Solar Energy), SOLAR - Total Solar Energy Collected with 20,000 Sq. Ft. Collector.

The data thus generated can be plotted as shown in Figures 10-14.

ESTIMATION OF ENERGY REQUIREMENTS OF HVAC SYSTEMS

Each system has been simulated for each hour of the data base year. In addition to weather parameters, the variables listed below were used.

Floor Area	164,450	Sq. Ft.
Total air flow	164,450	CFM
Minimum outside air flow	22,000	CFM
Lighting Load	2.5	Watts/Sq. Ft.
Equipment Load	0.5	Watts/Sq. Ft.
No. of people in space	1645	
Fraction of lighting load into space	60	%
Room temperature	76	°F
Room relative humidity	50	%
Minimum room relative humidity		%
Supply fan temperature rise	4.89	°F
Return fan temperature rise	2	°F
Cold duct temperature gain		°F
Coil bypass factor		
Heat input/cooling output		Btu
Supply air temperature	58	°F

The total energy required is summed monthly and yearly. Monthly maximum demand of each energy source is developed for each system studied.

An example of computation results for 1300 hours on August 20th is shown below.

TYPICAL HOURLY COMPUTER PRINTOUT

MONTH	8	
DAY	20	
HOUR	13	
OUTSIDE AIR		
Humidity Ratio	0.01322	lbs/lb
Temperature	75.0	°F
DEHUMDIFIED AIR		
Humidity Ratio	0.00523	lbs/lb
Temperature	76.0	°F
% OF OUTSIDE AIR		
Pass through Dehumidifier	13.4	%
By-Passes Dehumidifier	27.5	%
MIXED AIR		
Humidity Ratio	0.00998	lbs/lb
Temperature	78.8	°F
CONVENTIONAL SYSTEM		
Cooling Energy Required	4773690	BTU
Reheat Energy Required	2378483	BTU
Total Energy Required	7152173	BTU
DEHUMIDIFIER		
Cooling Energy Required	2884686	BTU
Dehumidification Energy Required	1875982	BTU
Total Energy Required	3836632	BTU
DIFFERENCE IN TOTAL ENERGY REQUIRED BY CONVENTIONAL AND SOLAR DEHUMIDIFIER	3,315,541	BTU

In the computer runs, calculations were made for each hour of a selected year to be compatible with the weather data to be used for the solar energy study. Monthly and yearly totals are shown in the next table. The columns are identified as follows:

A-COOL	-Actual refrigeration produced by conventional system as designed, BTU x 10^6/month
A-REHEAT	-Actual reheat required by conventional system as designed, BTU x 10^6/month
A-TOTAL	-Total heat energy input required by conventional system as designed, BTU x 10^6/month
B-COOL	-Actual refrigerated sensible cooling produced by chemical dehumidification system as designed, BTU x 10^6/month
B-DEHUMID	-Actual heat energy required to regenerate chemical dehumidifcation system, BTU x 10^6/month
B-TOTAL	-Total heat energy input required by chemical dehumidification system, BTU x 10^6/ month.

Monthly summaries are as follows in the next table.

*Note: Figures 11 and 13 are intentionally omitted.

Phase 0

Building Chilled Water Requirements by Month (10^6 Btu)

Month	Conventional	With Dehumid	(Non. Refr. Cooling) Cooling Tower Contribution
1	0	0	0
2	0	0	0
3	0	0	0
4	80	80	0
5	640	510	90
6	1,860	1,340	230
7	1,890	1,410	220
8	2,190	1,470	400
9	1,040	810	140
10	1,040	700	180
11	30	30	0
12	0	0	0
TOTAL:	8,770	6,350	1,120

Dehumidifier Energy Requirements by Month

Solar Energy Available by Month

($x10^6$ Btu)

Mo.	Dehumidifier	Solar for Dehumid	Steam for Dehumid	Solar Avail* @140°F 20,000	Solar Hot Water
1	0	0	0	140	140
2	0	0	0	200	200
3	0	0	0	280	280
4	0	0	0	300	300
5	140	140	0	320	180
6	340	320	20	320	0
7	320	320	0	350	30
8	600	300	300	300	0
9	210	210	0	330	120
10	270	270	0	270	0
11	0	0	0	160	160
12	0	0	0	120	120

*Note: Solar heat energy loss by the storage tanks was taken to be 3% of heat available for other uses.

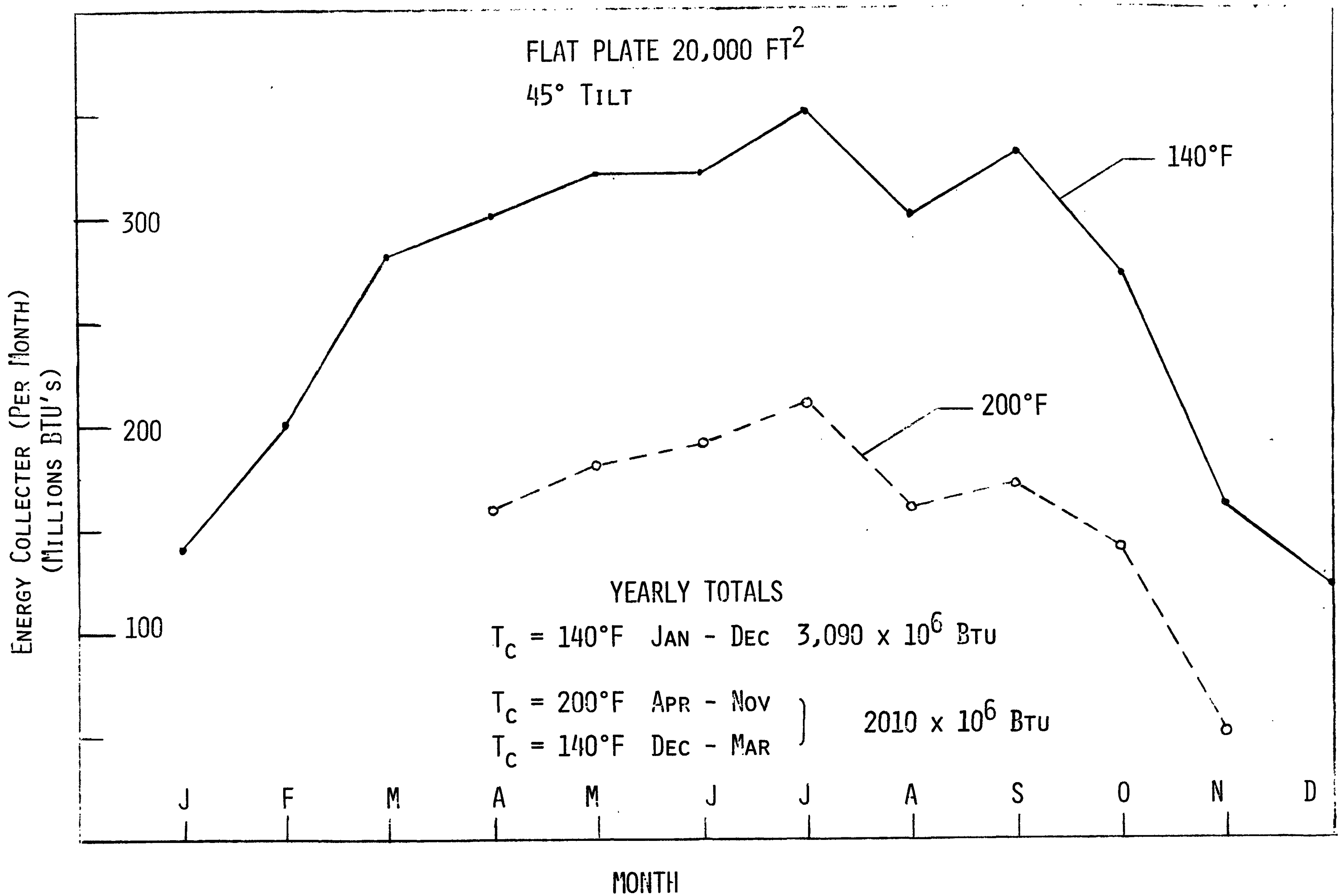


FIGURE 14: COLLECTED SOLAR ENERGY - BTU/MONTH AT 140°F FOR DEHUMIDIFIER AND DOMESTIC HOT WATER AND AT 200°F FOR ABSORPTION CHILLER OPERATED DURING SUMMER MONTHS.

ESTIMATED CAPITAL COSTS

Dehumidification Equipment

Equipment

Conditioners, concentrators, heat exchangers, controls, pumps, etc.	220,000.

Installation

Ductwork, piping & hookup, installation electrical work	205,000.
	$ 425,000.

Credits

Reduced steam driven chiller capacity 345 tons @ $400/ton	-141,600.
	$ 283,400.

Solar Collector System

Solar Collector Equipment (20,000 sq. ft.)

Collector panels	$10.00/sq. ft.
Piping & hookup	5.00/sq. ft.
Installation	10.00/sq. ft.
	$25.00/sq. ft.

20,000 sq. ft. collector @ $25/sq. ft.	500,000.
Structural steel	200,000.
Storage tank & related work	75,000.
	$ 775,000.

Credits

Curtain wall elimination - 20,000 sq. ft. 20,000 sq. ft. @ $8/sq. ft.	-160,000.
	$ 615,000.

Note: In the phase 0 study, the estimated operating and maintenance cost of the dehumidifcation equipment was taken to be exactly offset by the reduced central plant electric power requirements and the maintenance of the system was considered included in the general maintenance of the building system.

No estimates of O & M costs for the solar system were made.

DISCUSSION OF THE RESULTS

The results of the simulations are tabulated as follows:

The solar driven absorption chiller saves 1,070 x 10^6 Btu/yr compared with the conventional system. In this case 810 x 10^6 Btu/yr is available for other purposes such as domestic water heating.

The steam driven dehumidifier saves 2830 x 10^6 Btu/yr. The solar driven dehumidifier saves 4,390 x 10^6 Btu/yr and provides an additional 1,530 x 10^6 Btu/yr of 140°F hot water.

The solar collector temperature has been taken to be 140°F when used for dehumidification or when used during non-cooling months. When used for absorption refrigeration chilling, the collection temperature is taken to be 200°F.

If we take $8/$10^6$ Btu as the projected cost of steam in 1977, annual savings over the conventional system are as follows:

	Annual Savings	Est. Capital Costs	Cost to Annual Savings Ratio
Solar Driven Absorption Chiller (reheat required)	$8,560		
Hot water	6,480		
	$15,040	715,000	48 yrs.
Steam Driven Dehumidifier (reheat not required)	$22,640	253,000*	11 yrs.
Solar Driven Dehumidifier (steam supplement) (reheat not required)	$35,120		
Hot water	$12,240		
	$47,360	$898,400	19 yrs.

*Capital cost of $395,000 reduced by $141,600 the cost of 354 tons of steam driven chiller capacity replaced by dehumidifier at $400/ton.

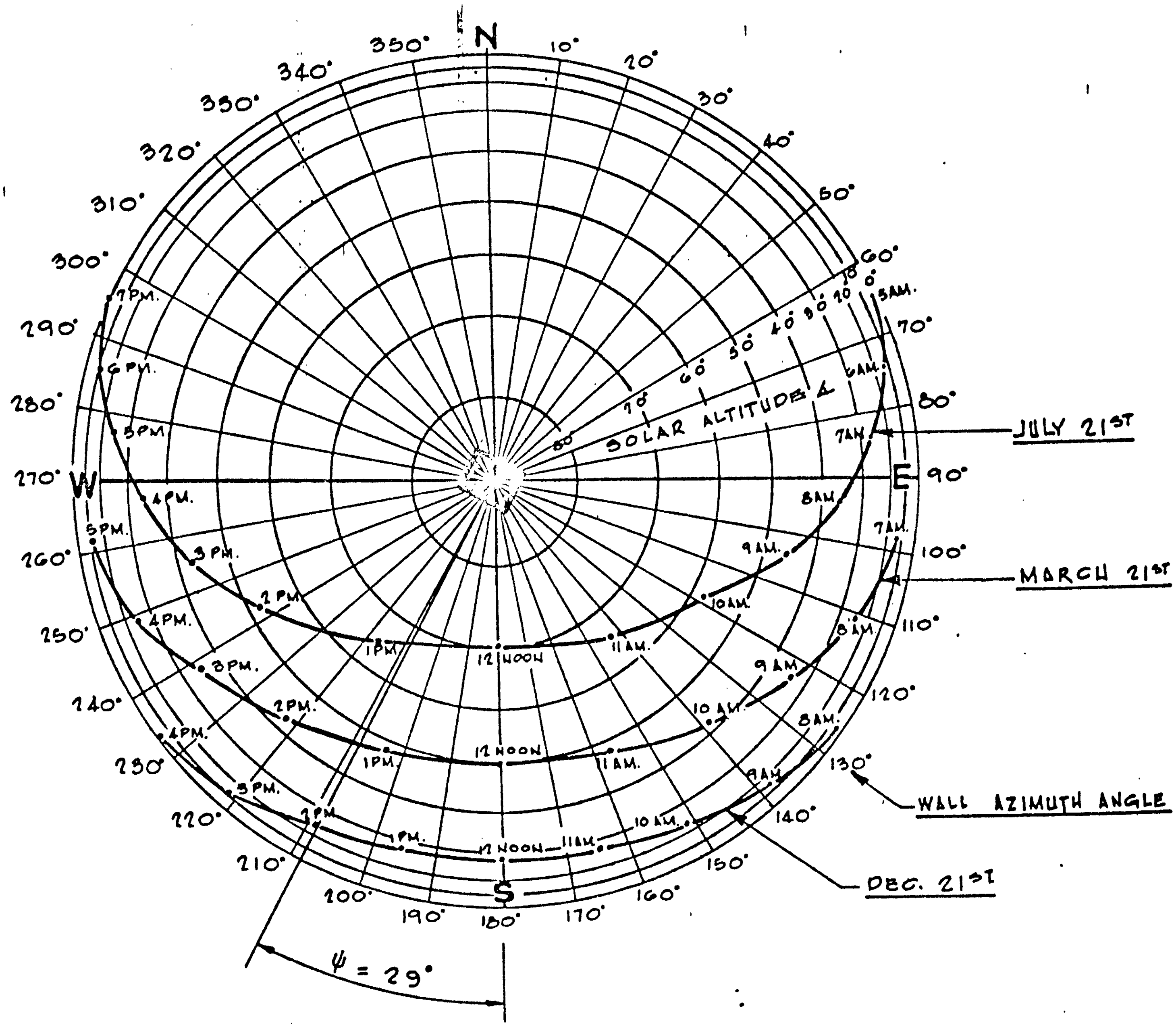


FIG. 10 - SUN'S PATH ON DIFFERENT DAYS OF A YEAR

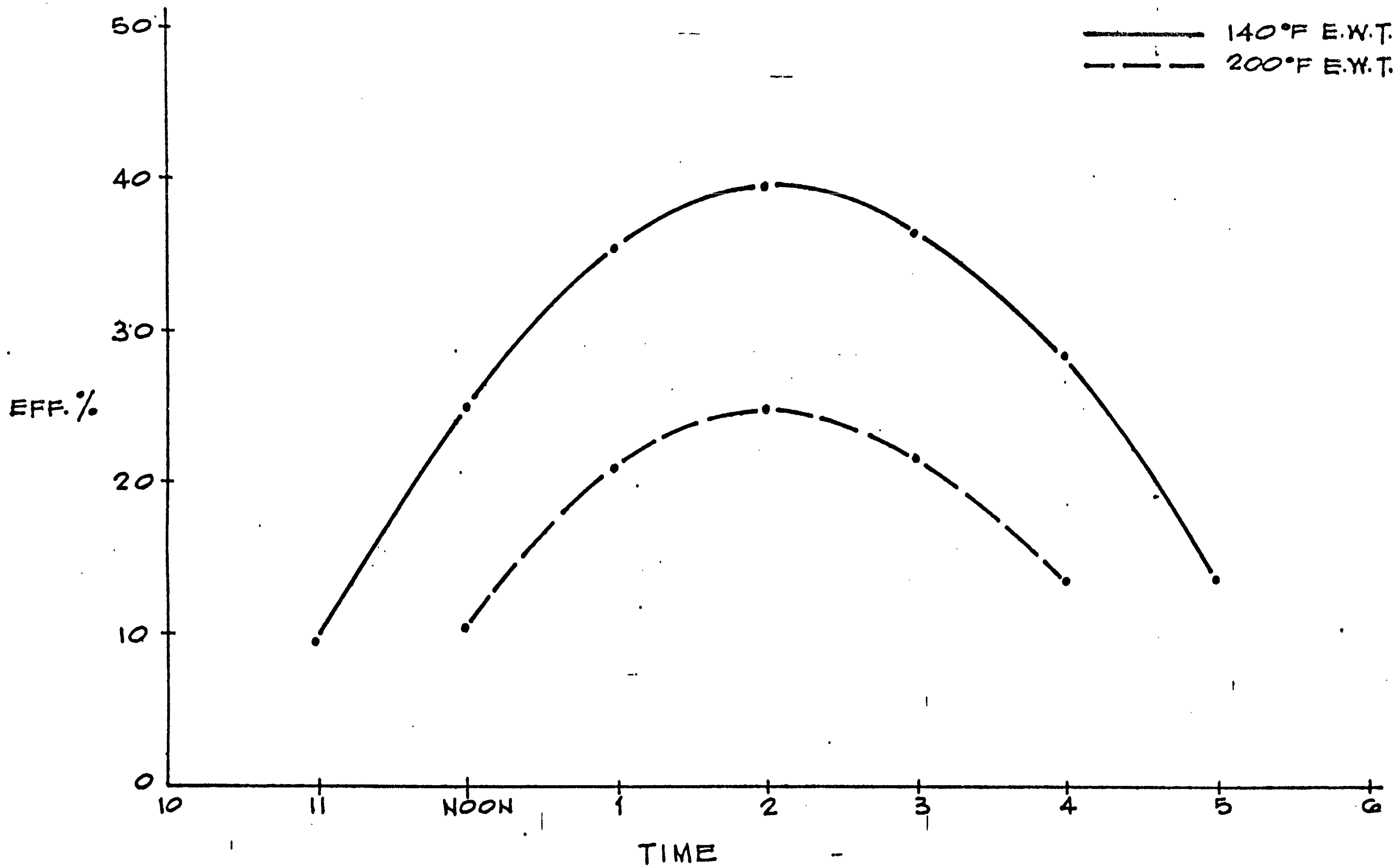


FIG. 13 - COLLECTOR EFFICIENCIES - MARCH 21ST

FOR COLLECTOR AT 40° TILT ANGLE 209° (29° W OF S) ORIENTATION
40° N. LATITUDE.

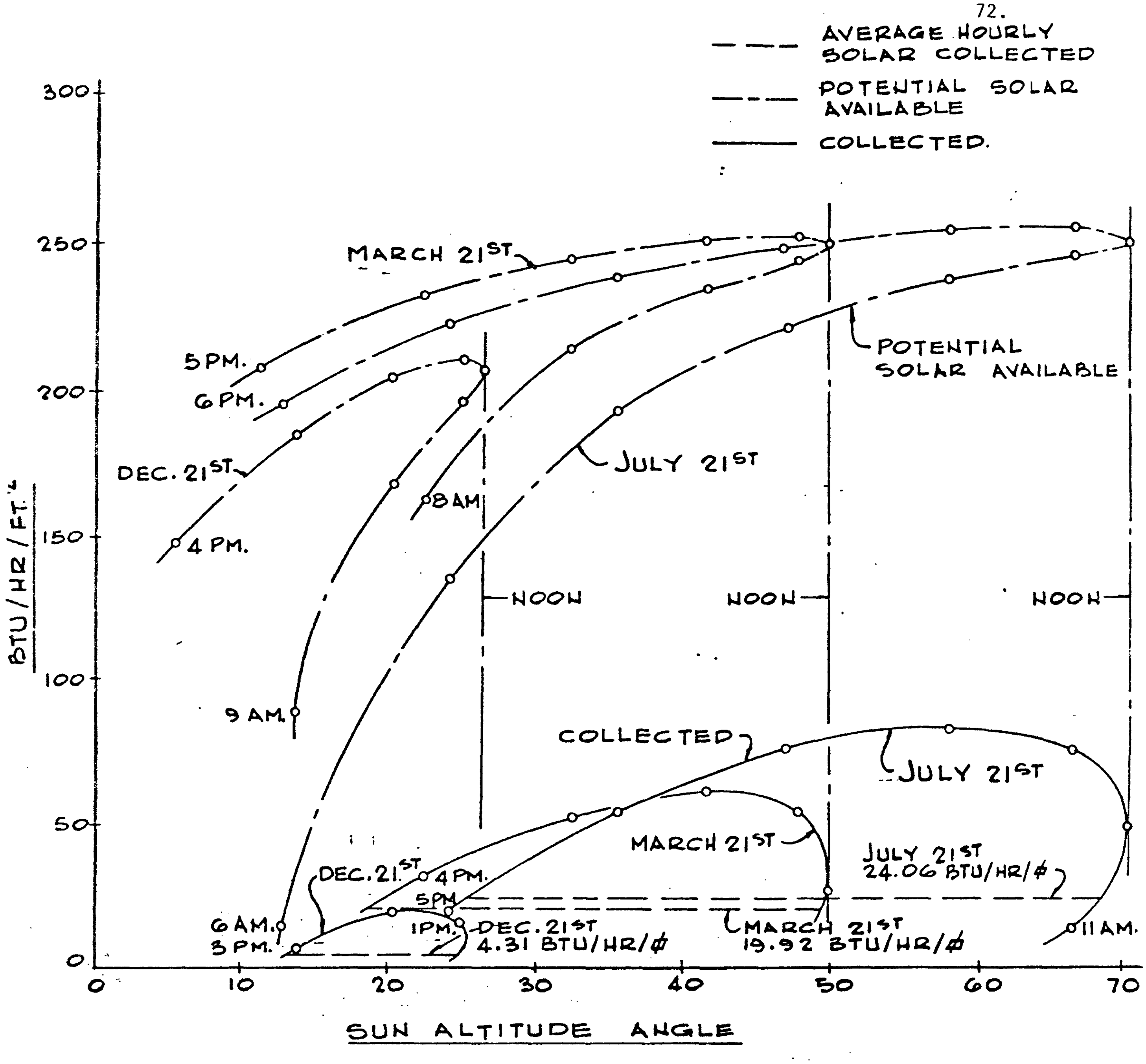


FIG. 12. - INTENSITY OF TOTAL SOLAR INSOLATION INDICATING COMPARISON OF POTENTIAL SOLAR AVAILABLE AND COLLECTED ENERGY IN BTU/HR/FT² FOR COLLECTOR AT 40° TILT ANGLE 209° (29° W OF S) ORIENTATION 40° N. LATITUDE 200°F E.W.T.

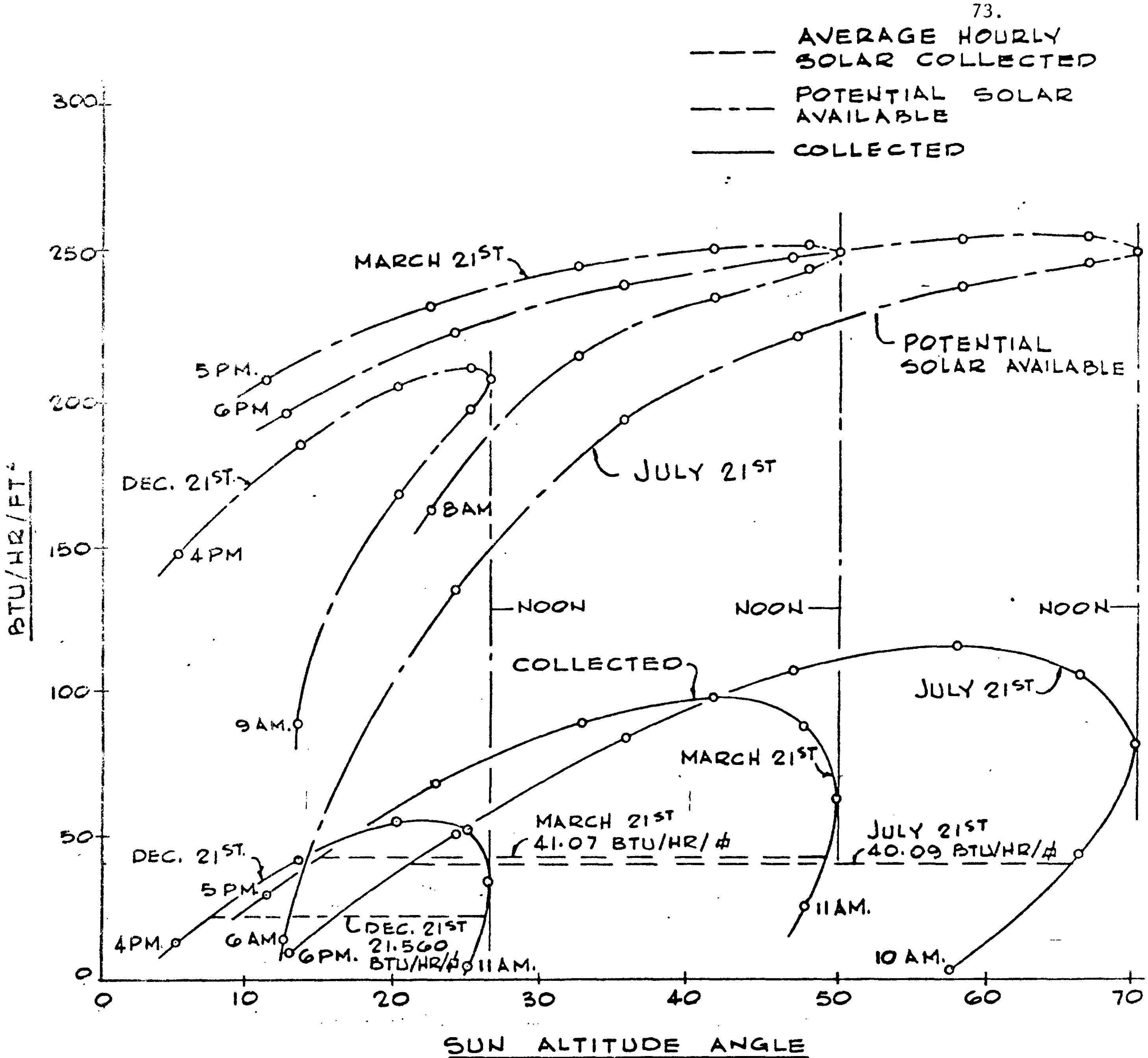


FIG. 11 .- INTENSITY OF TOTAL SOLAR INSOLATION INDICATING COMPARISON OF POTENTIAL SOLAR AVAILABLE AND COLLECTED ENERGY IN BTU/HR/FT² FOR COLLECTOR AT 40° TILT ANGLE 209° (29° W OF S) ORIENTATION 40° N. LATITUDE 140°F E.W.T.

APPENDIX II

II-A PERFORMANCE OF A FLAT PLATE SOLAR COLLECTOR MODULE

Reference: Solar Energy Thermal Processes, Duffie and Beckman [1]

I. Required Data

A. Meteorological

1. Beam and diffuse radiation per unit area of horizontal surface: H_b and H_d, respectively. These were estimated from the total radiation per unit area of horizontal surface H and the extraterrestrial radiation H_{ext} per unit area of horizontal surface.

2. Ambient air temperature T_a

3. Wind velocity, V

4. Latitude ϕ; day of year n; time of day.

B. Properties of Glass Cover Plates

1. Average optical properties for solar radiation - average refractive index in visible spectrum, n_g; average absorption coefficient in the visible, K.

2. Low temperature hemispherical emissivity, ε_g

3. Mass per sheet, m; thickness, L; specific heat, c.

C. Properties of Collector Plate

1. Directional absorptivity for solar radiation, $\alpha(\theta)$

2. Low temperature hemispherical emissivity, ε_p

3. Plate construction - material, size, tube spacing, bonding, insulation, etc. These will affect the mass m, specific heat c, the efficiency factors F, F', F_R and the back loss coefficient U_b.

D. Flow Conditions

1. Inlet temperature $T_{f,i}$

2. Mass flow rate per unit area of collector surface, G

3. Specific heat of the fluid, c.

E. Properties of Storage Tank

1. Mass of fluid in tank, M

2. Specific heat of fluid, c

3. Surface area of tank, A_t

4. Insulation thermal conductivity, k_{ins}, and thickness, ℓ_{ins}.

F. Miscellaneous

1. Angle of tilt of collector from the horizontal, s.

2. Surface azimuth angle, γ; zero being due south, east positive and west negative. (-29°)

3. Diffuse reflectivity of surroundings (i.e., buildings, ground), ρ_o

4. Number of glass cover plates, N.

5. Spacing of glass cover plates, ℓ

II. Estimation of Beam and Diffuse Radiation from Weather Data.

The weather data tape provided values of the hourly total radiation per unit area of horizontal surface, H, and the hourly extraterrestrial radiation per unit area of horizontal surface, H_{ext}. To determine the beam and diffuse components of such radiation an approximation to the method of Liu and Jordan [2] was used. Taking their curve based on daily values and assuming that it would be approximately valid for hourly values, their curve was divided into three regions and approximated in each region by a straight line. The following equations resulted:

$$r = \frac{H}{H_{ext}} \qquad \text{(IIa-1)}$$

$$r \le 0.1 \qquad H_d = H \qquad \text{(IIa-2)}$$

$$0.1 \le r \le 0.75 \qquad H_d = [1.0-1.29(r-0.1)]H \qquad \text{(IIa-3)}$$

$$0.75 \le r \qquad H_d = 0.16\,H \qquad \text{(IIa-4)}$$

$$H_b = H-H_d \qquad \text{(IIa-5)}$$

III. Calculation of Energy Losses from the Collector System

A. Losses Through the TOP by Convection and Radiation

1. The formulation of the top loss coefficient can be approached in a very straightforward manner by noting that in a quasi-steady state in which absorption by the glass covers is taken as negligible that the heat loss from the plate to the first cover equals that of the first cover to the second...equals that from the top cover to the ambient. Equations for the radiative and free convective heat transfer for enclosed air spaces tilted with respect to a gravitational field can readily be written plus the equation for the combined radiative and forced convective transfer from the uppermost cover. Doing this

results in a set of N + 1 coupled non-linear algebraic equations in N + 2 unknowns: the N unknown cover temperatures, the unknown mean plate temperature $T_{p,m}$ and the as yet to be determined heat flux loss

$$\frac{Q_t}{A_c}.$$

The required N + 2nd equation, however, cannot be directly written. This is because an equation for the mean plate temperature $T_{p,m}$ must be written and it just isn't coupled directly to the other N + 1 equations, but rather to variables such as the mean fluid temperature $T_{f,m}$ and the useful energy collected Q_u through the heat transfer resistance between the plate and the fluid. As a result, an iteration of the complete solution to the problem must be performed in order to determine the proper $T_{p,m}$. And, within each iterative step, the solution to the N + 1 coupled non-linear equations is required.

The system of equations relating the top losses Q_t/A_c, the mean plate temperature $T_{p,m}$, the N glass cover temperatures T_i, and the ambient air temperature T_a are:

$$Q_t/A_c = \{Nu_1 \frac{k_{a1}}{\ell} + \frac{\sigma(T_{p,m} + T_1)(T_{p,m}^2 + T_1^2)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_g} - 1}\} \times (T_{p,m} - T_1) \qquad \text{(IIa-6)}$$

where from [3]:

$$Nu_1 = 0.142\,(Gr_1)^{0.285} \qquad \text{(IIa-7)}$$

$$Gr_1 = \frac{g \cos s\ \beta_1\ (T_{p,m} - T_1)\ell^3}{\nu_1^2} \qquad \text{(IIa-8)}$$

For $2 \leq i \leq N$:

$$Q_t/A_c = \{Nu_i \frac{k_{ai}}{\ell} + \frac{\sigma(T_{i-1}+T_i)(T_{i-1}^2+T_i^2)}{\frac{2}{\varepsilon_g} - 1}\}(T_{i-1}-T_i) \qquad \text{(IIa-9)}$$

where

$$Nu_i = 0.142(Gr_i)^{0.285} \qquad \text{(IIa-10)}$$

$$Gr_i = \frac{g \cos s\ \beta_i(T_{i-1}-T_i)\ell^3}{\nu_i^2} \qquad \text{(IIa-11)}$$

For the N + 1st equation

$$Q_t/A_c = \{h_w + \frac{\sigma(T_N+T_a)(T_N^2+T_a^2)}{\frac{1}{\varepsilon_g}}\}(T_N-T_a) \qquad \text{(IIa-12)}$$

where

$$h_w = 5.7 + 3.8\ V \qquad V(\text{m/sec}) \quad h_w(\text{w/m}^{-2}\text{-°C}) \qquad \text{(IIa-13)}$$

The air properties - coefficient of thermal expansion β, kinematic viscosity ν, thermal conductivity k_a - should be evaluated at the mean of the temperature between which the corresponding heat transfer occurs (i.e.,

$$\beta_1(T) = \beta_1((T_{p,m} + T_1)/2)$$

Equations (IIa-6) through (IIa-13) represent a system of N + 1 equations in N + 2 unknowns - the N glass cover temperatures T_i, the mean plate temperature $T_{p,m}$, and the energy losses through the top Q_t/A_c. The mean plate temperature $T_{p,m}$ will be found by iteration of the solution to the entire problem. Hence at this point in the calculational procedure, a value has been assigned to $T_{p,m}$ thus making it a known quantity. The solution of the remaining N + 1 coupled non-linear algebraic equations can then perhaps be best accomplished by assigning a value to Q_t/A_c, calculating T_1 from equations (IIa-6) -(IIa-8), then T_2 from equations (IIa-9) -(IIa-11), and so on until a T_a is found. Iteration proceeds until the so-calculated T_a matches with the actual T_a. In beginning the procedure, the estimated value of Q_t/A_c may be obtained from the following system of equations [1]:

$$U_t(45°) = \{\frac{N}{(344/T_{p,m})[(T_{p,m} - T_a)/(N+f)]^{0.31}} + \frac{1}{h_w}\}^{-1} + \frac{\sigma(T_{p,m}+T_a)(T_{p,m}^2 + T_a^2)}{[\varepsilon_p+0.0425N(1-\varepsilon_p]^{-1} + [(2N+f-1)/\varepsilon_g] - N} \qquad \text{(IIa-14)}$$

and

$$\frac{U_t(s)}{U_t(45°)} = 1 - (s-45°)(0.00259 - 0.00144\ \varepsilon_p) \qquad \text{(IIa-15)}$$

where

$$f = (1.0-0.04h_w + 5.0\times10^{-5}h_w^2)(1+0.058N) \qquad \text{(IIa-16)}$$

Then,

$$Q_t/A_c = U_t(s)\,(T_{p,m} - T_a) \qquad \text{(IIa-17)}$$

2. With the T_i's known, the required loss coefficients for the ith cover to (i + 1)th cover may then be determined using the following equations:

$$U_i = \{Nu_{i+1}\frac{k_{ai+1}}{\ell_{i+1}} + \frac{\sigma(T_i+T_{i+1})(T_i^2+T_{i+1}^2)}{\frac{2}{\varepsilon_g} - 1}\} \qquad \text{(IIa-18)}$$

$$U_N = h_w + \frac{\sigma(T_N+T_a)(T_N^2+T_a^2)}{\frac{1}{\varepsilon_g}} \qquad \text{(IIa-19)}$$

These are needed for other calculations which are to follow.

B. Loss Through the BOTTOM by Conduction and Convection

1. The overwhelming resistance to heat flow through the back of the collector is the conduction resistance offered by the insulation. Convective resistance is generally only a small percentage of the conductive.

2. Therefore the back loss coefficient is:

$$U_b = \frac{k_{ins}}{\ell_{ins}} \qquad \text{(IIa-20)}$$

and the heat transfer is considered to occur between the plate and the ambient temperatures.

C. Loss through the EDGES of the collector are generally less than 1% of the top and bottom losses for large collectors. Hence the edge loss coefficient will be taken equal to zero.

D. The overall loss coefficient is then:

$$U_L = U_t + U_b \qquad \text{(IIa-21)}$$

IV. Determination of Solar Energy Absorbed by the Collector Plate

A. Calculation of radiation incident on a tilted surface per unit area of the tilted surface (used only for reference):

$$_t = H_bR_b + H_d\,(\frac{1 + \cos s}{2})$$

$$+ (H_b + H_d)\rho_o\,(\frac{1-\cos s}{2}) \qquad \text{(IIa-22)}$$

where

$$R_b = \frac{\cos \Theta_t}{\cos\phi\cos\delta\cos\omega + \sin\phi\sin\delta} \qquad \text{(IIa-23)}$$

$$\cos\Theta_t = (\cos\gamma\sin\phi\sin s + \cos\phi\cos s)\times$$

$$\cos\delta\cos\omega + \sin\gamma\sin s\;\times \qquad \text{(IIa-24)}$$

$$\cos\delta\sin\omega +$$

$$(\sin\phi\cos s - \cos\gamma\cos\phi\sin s)\sin\delta$$

ω = hour angle; solar noon being zero, and each hour equaling 15° of longitude; mornings positive, afternoons negative (IIa-25)

$$\delta = 23.45\sin[360\,\frac{284 + n}{365}] \qquad \text{(IIa-26)}$$

Solar time, and hence ω, can be determined from the equation of time:

$$\text{Solar time} = \text{standard time} + E+4(L_{st}-L_{loc})/60 \qquad \text{(IIa-27)}$$

where E = equation of time, from Fig. 2.7.1 of reference [1]

L_{st} = standard meridian for the local time zone (=75°W for eastern time).

L_{loc} = longitude of the location question (= 74°W for New York City).

B. Effective Transmittance - Absorptivity Product - $(\tau\alpha)_{eff}$

1. An effective value of $(\tau\alpha)$ is required for two reasons:

   a) Of the radiation passing through the cover system and striking the plate, some is reflected back to the cover system. But, not all of this radiation is lost since some is reflected back to the plate.

   b) Not all of the energy absorbed by the glass covers is a loss to the system as a whole. Part of it serves to increase the glass temperature, reducing the temperature difference between the plate and the covers and thereby reducing the losses from the plate. For bookkeeping purposes this can be shown to add directly to the straightforward $(\tau\alpha)$ calculation.

2. Calculation of $(\tau\alpha)_{eff}$ for BEAM radiation at angle Θ_t.

a)

$$(\tau\alpha)_{eff,b} = (\tau_N\alpha)_{\Theta_t} + [1-\tau_\alpha(\Theta_t)] \sum_{i=1}^{N} a_i \tau_{N-i}(\Theta_t) \qquad \text{(IIa-28)}$$

where

$$a_i = U_t \sum_{j=1}^{N} \frac{1}{U_j} \qquad \text{(IIa-29)}$$

(see section II-A-2 for U_j); and the remaining terms are evaluated as follows:

b)

$$(\tau_N\alpha)_{\Theta_t} = \frac{\tau_N(\Theta_t)\ \alpha(\Theta_t)}{1-[1-\alpha(\Theta_t)]\rho_D} \qquad \text{(IIa-30)}$$

where ρ_D is the reflectivity of the glass for diffuse solar radiation (approximately equal to the spectral reflectivity at 60°) and $\alpha(\Theta_t)$ is the absorptivity of the plate at angle Θ_t (see Fig. 5.7.1 of reference [1]).

c)

$$\tau_N(\Theta_t) \doteq \tau_{rN}(\Theta_t)\ \tau_{\alpha N}(\Theta_t) \qquad \text{(IIa-31)}$$

d)

$$\tau_{rN}(\Theta_t) = \frac{1-\rho(\Theta_t)}{1+(2N-1)\rho(\Theta_t)} \qquad \text{(IIa-32)}$$

where

$$\rho(\Theta_t) = \frac{1}{2}\left[\frac{\sin^2(\Theta_2-\Theta_t)}{\sin^2(\Theta_2+\Theta_t)} + \frac{\tan^2(\Theta_2-\Theta_t)}{\tan^2(\Theta_2+\Theta_t)}\right] \qquad \text{(IIa-33)}$$

with

$$\sin\Theta_2 = \frac{n_{air}}{n_g}\sin\Theta_t \qquad (n_{air} = 1.0) \qquad \text{(IIa-34)}$$

e)

$$\tau_{\alpha N}(\Theta_t) = \exp(-KLN/\cos\Theta_t) \qquad \text{(IIa-35)}$$

3. Calculation of $(\tau\alpha)_{eff}$ for DIFFUSE radiation.

a) According to a graphical integration by Hottel and Woertz, this quantity can be reasonably approximated by using the "beam" equations above evaluated for Θ_t= 58. This average angle is a function of the properties of the collector system but can be expected to be on the order of 60°. This latter value will be used (it is more conservative than the 58° approximation).

b) Therefore:

$$(\tau\alpha)_{eff,d} = (\tau_n\alpha)_{60^\circ} + [1-\tau_{\alpha 1}(60^\circ)] \sum_{i=1}^{N} a_i\tau_{N-i}(60^\circ) \qquad \text{(IIa-36)}$$

where equations (Ia-30) through(IIa-35) are again used.

C. Effects of Dust and Shading

1. Due to the complex nature of these effects, they are generally taken into account by empirical factors which reduce directly the amount of energy absorbed by the plate.

2. Dust factor = (1-d) where $d \simeq 0.02$ (IIa-37)

3. Shading factor = (1-s) where $s \simeq 0.03$. (IIa-38)

D. Solar Energy Absorbed by the Collector Plate, S.

$$S = (1-s)(1-d)\left\{H_b R_b(\tau\alpha)_{eff,b} + H_d\left(\frac{1+\cos s}{2}\right)(\tau\alpha)_{eff,d} + (H_b+H_d)\rho_o\left(\frac{1-\cos s}{2}\right)(\tau\alpha)_{eff,d}\right\} \qquad \text{(IIa-39)}$$

V. Determination of the Collector Heat Removal Factor, F_R.

A. The heat removal factor relates the useful energy gain of a collector to the useful gain if the whole collector plate were at the fluid inlet temperature:

$$F_R = \frac{Gc\,(T_{f,o} - T_{f,i})}{[S - U_L(T_{f,i} - T_a)]} \qquad \text{(IIa-40)}$$

B. A useful expression for this quantity can be derived:

$$F_R = F'[1 - e^{-\frac{F'U_L}{Gc}}]/(\frac{F'U_L}{Gc}) \qquad \text{(IIa-41)}$$

where F' is a collector efficiency factor which accounts for the heat transfer to the fluid at a given section of the collector due to the non-uniform plate temperature and the temperature difference between the plate above the fluid and the fluid itself. It is therefore a function of plate and tube construction and the heat transfer coefficient between the fluid and the surface it is in contact with (hence a function of flow rate G). F' factors are tabulated in given references for various collector geometries. Calculation of F' for the assumed flat plate geometry is given in section IX.

VI. Determination of the Steady-State Useful Energy Collection by Iteration.

A. The useful energy collection is:

$$Q_u/A_c = F_R\,[S - U_L(T_{f,i} - T_a)] \qquad \text{(IIa-42)}$$

B. The iteration procedure using specified values of $T_{f,i}$, T_a and G is:

1. Guess a $T_{p,m}$ and then determine in order:

   a) T_i, U_i and U_L through the previously described iteration procedure (see section III)

   b) S

   c) F_R

   d) Q_u/A_c

   e) $T_{f,m} = T_{f,i} + \frac{Q_u/A_c}{U_L F_R}\,[1 - \frac{F_R}{F'}]$

$$\text{(IIa-43)}$$

2. Calculate a $T_{p,m}$ from the approximate relation:

$$T_{p,m} \doteq T_{f,m} + Q_u R_{p-f} \qquad \text{(IIa-44)}$$

where R_{p-f} is the heat transfer resistance between the plate and the fluid. The major part of this resistance is the convective heat transfer coefficient to the fluid. Thus for high fin efficiency and high bond conductance (if tubes are bonded to the plate) the R_{p-f} can be given as:

$$R_{p-f} \doteq \frac{1}{h_{f,i}\pi D\eta\lambda} \qquad \text{(IIa-45)}$$

where η and λ are the number of tubes and their length, respectively.

3. Compare the calculated $T_{p,m}$ with the guessed one and iterate until they are approximately equal. The value of Q_u/A_c at that time is then the steady state useful energy collection.

VII. Transient Response of the Collector - Heat Capacity Effects

A. A simple transient analysis can be performed in which it is assumed that the absorber plate, the water in the tubes, and the back-insulation are all at the same temperature T_p. Also, each cover has its own unique temperature. By introducing auxiliary steady state relationships between the upward heat losses, the N + 1 differential equations resulting from energy balances on the plate and each cover can be reduced to a simple differential equation whose solution is:

$$\frac{S - U_L(T_p - T_a)}{S - U_L(T_{p_{initial}} - T_a)} = \exp(-\frac{A_c U_L t}{(mc)_{eff}}) \qquad \text{(IIa-46)}$$

where $(mc)_{eff}$ is an effective heat capacity of the collector:

$$(mc)_{eff} = (mc)_p + \sum_{i=1}^{N} a_i (mc)_{c,i} \qquad \text{(IIa-47)}$$

where $(mc)_p$ is the heat capacity of the plate-water-insulation combination, $(mc)_{c,i}$ is the heat capacity of the i^{th} cover, and a_i is the same quantity as given in section IV-B-2(a).

B. The above result assumes that S and T_a remain constant for some period, say one hour. Thus knowing the average hourly values of S, T_a and U_L plus the value of T_p at the beginning of the hour, the "collector plate temperature" T_p may be estimated at the end of the hour.

C. It should be noted that heat capacity effects are substantial during early morning warm-up and evening cool-down. However the effects of mid-day weather fluctuations are usually small. As a result, the above formulation was used only during the daylight hours when the collectors were not operating. During operation, the heat capacity effects of the collectors were approximately accounted for by simply subtracting the quantity

$$(Q/A_c)_{\substack{heat\\capacity}} = (mc)_{eff}(T_{p,m_{new}} - T_{p,m_{old}}) \quad \text{(IIa-48)}$$

from the previously calculated useful heat gain to yield an actual useful energy collection rate given by:

$$(Q_u/A_c)_{act} = (Q_u/A_c) - (Q/A_c)_{\substack{heat\\capacity}} \quad \text{(IIa-49)}$$

During non-daylight hours the collector temperature equaled the ambient temperature.

VIII. Thermal Storage Tank Dynamics

Modeling the nonstratified storage tank as an insulated cylinder which gains an amount of solar energy Q_u and loses an amount equal to the sum of the energy taken to the concentrator plus thermal losses results in the following equation for the variation in tank temperature

$$T_t - T_{a,o} = (T_t - T_a)_o\, e^{-\frac{UA_t}{Mc}t} + \left(\frac{Qu-\Delta Q}{UA_t}\right)\left(1-e^{-\frac{UA_t}{Mc}t}\right) \quad \text{(IIa-50)}$$

where $(T_t - T_a)_o$ = Difference between tank and ambient temperature at the beginning of the time interval

M = Mass of fluid in tank

c = Specific heat of fluid

A_t = Surface area of tank

$U = \dfrac{k_{ins}}{\ell_{ins}}$

ΔQ = Energy taken from storage tank to concentrator

Q_u = Energy gain from solar collectors

t = Time increment

IX. Assumed Parameters for the Flat Plate Solar Collector

A. Calculation of collector efficiency factor, F'

1. The collector plate geometry was assumed to be of the tube and sheet type-identical to the PPG Roll-Bond plate except with circular fluid passageways of diameter D.

2. The fin efficiency, F, for such a construction can be determined from:

$$m_1 = \sqrt{\frac{U_L}{k_p\ \ell_p}}\ (W-D)/2 \quad \text{(IIa-51)}$$

where W is the spacing between the tubes, k_p is the plate thermal conductivity and ℓ_p is the plate thickness. Then

$$F = \frac{1}{m_1}\ \tanh\ m_1$$

3. The collector efficiency factor, F', may then be shown to be:

$$F' = \frac{1}{U_L W\left[\left(\frac{1}{U_L[D+(W-D)F]}\right)+\frac{1}{h_{f,i}\pi D}\right]} \quad \text{(IIa-52)}$$

where from [4]:

$$h_{f,i} = \frac{k_f}{D}\left\{4.36 + \frac{0.067[(D/\lambda)RePr]}{1+0.04\ [(D/\lambda)RePr]^{2/3}}\right\} \quad \text{(IIa-53)}$$

in which Re is the Reynolds Number and Pr is the Prandtl number.

B. Parameter Values

1. Glass

$n_g = 1.526$

$K = 0.206\ in^{-1}$

$\varepsilon_g = 0.90$

$L = 0.125$ in

$m_g = 1.64\ lbm/ft^2$

$c_g = 0.16\ \frac{BTU}{lbm\text{-}°F}$

2. Absorbing Plate

$k_p = 100\ \frac{BTU}{hr\text{-}ft\text{-}°F}$

$\ell_p = 0.060$ in

D = .44 in

W = 2.50 in

λ = 6 ft.

η = 13

$m_p = .85\ lbm/ft^2$

$c_p = .208\ \frac{BTU}{lbm\text{-}°F}$

3. Fluid (30% ethylene glycol-water)

$k_f = 0.28\ \frac{BTU}{hr\text{-}ft\text{-}°F}$

$Pr = .639$

$G = 15\ \frac{lbm}{hr\text{-}ft^2}$

$m_f = 0.33\ \frac{lbm}{ft^2}$

$c_f = 0.94\ \frac{BTU}{lbm\text{-}°F}$

$Re = 2160$

4. Insulation

$k_{ins} = .025\ \frac{BTU}{hr\cdot ft\text{-}°F}$

$\ell_{ins} = 3$ in

$m_{ins} = 1.5\ \frac{lbm}{ft^2}$

$c_{ins} = 0.2\ \frac{BTU}{lbm\text{-}°F}$

5. Storage Tank

Diameter = 20 ft.

Length = 16 ft.

$U = 0.10\ \frac{BTU}{hr\text{-}ft^2\text{-}°F}$ (3" Fiberglas insulation)

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REFERENCES

1. Duffie, J.A. and Beckman, W.A., Solar Energy Thermal Processes, John Wiley and Sons, Inc., New York (1974).

2. Liu, B.Y.H. and Jordan, R.C., Solar Energy 4, No. 3 (1960).

3. Buchberg, H., Catton, I., and Edwards, D.K., "Natural Convection in Enclosed Spaces: A Review of Application to Solar Energy Collection", ASME Paper 74-WA/HT-12.

4. Whillier, A., "Design Factors Influencing Collector Performance", in Low Temperature Engineering Applications of Solar Energy, ASHRAE, New York (1967).

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NOMENCLATURE

A_c	collector area
A_t	surface area of storage tank
a_i	ratio of the top loss coefficient to the loss coefficient from the i^{th} cover to the ambient
c	heat capacity
D	tube diameter
d	quantity in dust factor
E	equation of time
F	fin efficiency
F'	collector efficiency factor
F_R	collector heat removal factor
f	function defined in section III-A-1
G	flow rate per unit area of collector surface
Gr	Grashof number
g	gravitational acceleration
H	total radiation per unit area of horizontal surface
H_b	beam radiation per unit area of horizontal surface
H_d	diffuse radiation per unit area of horizontal surface
H_{ext}	extraterrestrial radiation per unit area of horizontal surface
H_T	radiation incident on a tilted surface per unit area of tilted surface
$h_{f,i}$	heat transfer coefficient between tube wall and fluid
h_w	heat transfer coefficient for wind on top cover
K	absorption coefficient for glass
k_a	thermal conductivity of air
k_{ins}	thermal conductivity of the insulation
k_p	thermal conductivity of the absorbing plate
L	thickness of a single glass cover
L_{loc}	longitude of location in question
L_{st}	standard meridian for the local time zone
ℓ	thickness of insulation
ℓ_p	thickness of absorbing plate
M	mass of fluid in storage tank

m mass per unit area

N number of glass cover plates

Nu Nusselt number

n number of the day of the year

n_{air} index of refraction for air

n_g index of refraction for glass

Q_u useful energy collected

Q_t energy loss through the top

R_b ratio of beam radiation intercepted by a tilted surface to that intercepted by a horizontal surface

R_{p-f} heat transfer resistance between the plate and the fluid

S solar energy absorbed by the collector plate

s angle of tilt of collector from horizontal; also a quantity in the shading factor

T_a ambient air temperature

$T_{f,i}$ inlet fluid temperature

$T_{f,m}$ mean fluid temperature

T_i temperature of the i^{th} glass cover

$T_{p,m}$ mean plate temperature

t time

U_b back loss coefficient

U_i loss coefficient from i^{th} cover to the (i+1)th cover

U_t top loss coefficient

U_L overall loss coefficient

V wind velocity

W spacing between tubes

$\alpha(\theta)$ directional absorptivity of plate surface

β coefficient of thermal expansion

γ surface azimuth angle

δ declination of the sun

ε_g low temperature hemispherical emissivity of the glass

ε_p low temeprature hemispherical emissivity of the plate

η number of tubes in the collector

θ_t angle between the incoming bean radiation and the normal to the collector

λ length of a single riser tube

ν kinematic viscosity

ρ reflectance

σ Stefan-Boltzmann constant

τ_r transmittance of single glass cover allowing only for reflection losses

τ_α transmittance of single glass cover allowing only for absorption losses

ω hour angle of the sun

II-B PERFORMANCE OF CORNING EVACUATED CYLINDRICAL COLLECTOR

Many of the calculations in the analysis of flat plate collectors are also required when analyzing cylindrical collectors. As a result, the same basic computer program can be used. The flow chart given in Appendic I-C was therefore written to encompass both the flat plate and the cylindrical collector calculation schemes.

There are two basic points of departure between the schemes. The first is in the calculation of the beam radiation absorbed by the collector plate and the second is in determining the energy losses from the plate. These are discussed below. One interesting result is that the solar energy absorbed by the collector plate may be calculated from the same equation as for flat plates by using an appropriate mean value for the transmittance of beam radiation through the glass cylinder.

I. Determination of Solar Energy Collected by the Absorbing Plate

BEAM RADIATION

To find the amount of direct solar energy which will be transmitted through the cylindrical surface, the angle of incidence which the solar rays make with each element of the cylinder at any point in time must first be determined. The position of the sun as seen from the ground may be expressed by the unit vector (see Figure 1):

$$n_{sun} = (\cos\alpha \cos z)\,\hat{i} + (\cos\alpha \sin z)\hat{j} + (\sin\alpha)\hat{k} \qquad \text{(IIb-1)}$$

where it can be shown that:

$$\sin\alpha = \cos\phi \cos\delta \cos\omega + \sin\phi \sin\delta \qquad \text{(IIb-2)}$$

$$\sin z = -\frac{\cos\delta \sin\omega}{\cos\alpha} \qquad \text{(IIb-3)}$$

in which

ϕ = latitude

δ = declination of the sun

ω = hour angle from solar noon; afternoons positive.

Using the coordinate system of Figure 1, the "reference" position of the collector will be as given in Figure 2. The axis of the cylinder is aligned with the east-west axis, and the normal to the absorbing plate points in the direction of the zenith. Figures 3 and 4 show the coordinate systems associated with an arbitrary position of the cylinder and the absorbing plate within it. The unit vectors for the various coordinates may be expressed in terms of the reference unit vectors as follows:

$$\hat{i}' = \cos\gamma_1\,\hat{i} + \sin\gamma_1\,\hat{j} \qquad \text{(IIb-4)}$$

$$\hat{j}' = -\sin\gamma_1\,\hat{i} + \cos\gamma_1\,\hat{j} \qquad \text{(IIb-5)}$$

$$\hat{k}' = \hat{k} \qquad \text{(IIb-6)}$$

$$\hat{i}'' = \hat{i}' \qquad \text{(IIb-7)}$$

$$\hat{j}' = \cos\eta\,\hat{j}' + \sin\eta\,\hat{k}' \qquad \text{(IIb-8)}$$

$$\hat{k}' = -\sin\eta\,\hat{j}' + \cos\eta\,\hat{k}' \qquad \text{(IIb-9)}$$

Angles γ_1 and η define the position of the cylinder. Angle γ_1 is the counterclockwise rotation of the cylinder about the x-axis. And, angle η is the counterclockwise rotation of the cylinder a-about the x'-axis. The general expression for the vector normal to the cylindrical surface for any point on the cylinder will then be given by:

$$\begin{aligned}\hat{n}_{cyl} &= \sin\beta\,\hat{i}'' + \cos\beta\,\hat{k}'' \\ &= (\sin\beta \cos\gamma_1 + \cos\beta \sin\eta \sin\gamma_1)\hat{i} \qquad \text{(IIb-10)} \\ &+ (\sin\beta \sin\gamma_1 - \cos\beta \sin\eta \cos\gamma_1)\hat{j} \\ &+ (\cos\beta - \cos\eta)k\end{aligned}$$

where β defines the direction of an arbitrary radius vector in the x" - z" plane.

The angle of incidence between the solar beam radiation and an element of cylindrical surface will then be:

$$\begin{aligned}\cos\Theta_i &= \hat{n}_{sun} \cdot \hat{n}_{cyl} \qquad \text{(IIb-11)} \\ &= (\cos\alpha \cos z)(\sin\beta \cos\gamma_1 + \cos\beta \sin\eta \sin\gamma_1) \\ &+ (\cos\alpha \sin z)(\sin\beta \sin\gamma_1 - \cos\beta \sin\eta \cos\gamma_1) \\ &+ (\sin\alpha)(\cos\beta \cos\eta)\end{aligned}$$

Since the glass cylinder is thin-walled, the transmission of a ray through the glass wall at any point along its surface may be reasonably approximated as the transmission of a ray through a planar glass sheet by simply using the local angle of incidence:

$$\tau(\theta_i) = \frac{1 - \rho(\theta_i)}{1 + \rho(\theta_i)} \qquad \text{(IIb-12)}$$

where $\rho(\theta_i)$ may be determined from eq.(IIa-33).

Not all rays passing through the glass will directly strike the absorbing plate. It is assumed that those which don't will never strike it. Figures 5 are views looking down the axis of the cylinder and they depict the limiting angles of intercept of the solar rays in the two cases of interest. In these figures, angle c prescribes the orientation of the absorbing plate inside the cylinder. Angle ξ denotes the angle between the z"-axis and the projection of the solar vector onto the x" - z" plane. The latter is given by:

$$\tan\xi = \frac{\hat{i}'' \cdot \hat{n}_{sun}}{\hat{k}'' \cdot \hat{n}_{sun}} = \qquad \text{(IIb-13)}$$

$$= \frac{\cos\gamma_1 \cos\alpha \cos z + \sin\gamma_1 \cos\alpha \sin z}{\sin\eta \sin\gamma_1 \cos\alpha \cos z - \sin\eta \cos\gamma_1 \cos\alpha \sin z + \cos\eta \sin\alpha}$$

Because of symmetry, the total energy transmitted through the glass which strikes the plate can be obtained by doubling the results of looking at the half-interval from β_1 to β_2 where either:

$$c \geq \xi \qquad\qquad c \leq \xi$$

$$\beta_1 = -\frac{\pi}{2} + c \quad \text{or} \quad \beta_1 = -\frac{\pi}{2} - c + 2\xi$$

$$\beta_2 = \xi \qquad\qquad \beta_2 = \xi \tag{IIb-14}$$

Integration must be performed with regard to the energy contained within an increment of angle $\Delta\beta$. As such, the integration path should lie along the curve generated by the intersection of a plane of the sun's rays with the surface of the cylinder (see Figure 6). The particular plane of rays which is chosen is the one for which: (a) the "central" ray is the vector produced by starting on the centerline of the cylinder and pointing directly at the sun, and (b) the rest of the rays of interest lie in a plane which is perpendicular to the x"-y" plane and are clearly parallel to the "central" ray. The energy transmitted and intercepted by the surface is then:

$$E_\tau = 2 \int_{\beta_1}^{\beta_2} I_{sun}\, \tau(\beta)\, dA_p(\beta) \tag{IIb-15}$$

where

I_{sun} = Beam radiation per unit area normal to the solar rays

dA_p = Element of projected area normal to solar rays

This can be used to define a mean transmittance for the cylinder:

$$\bar{\tau} = \frac{\int_{\beta_1}^{\beta_2} \tau(\beta)\, dA_p(\beta)}{\int_{\beta_1}^{\beta_2} dA_p(\beta)} \tag{IIb-16}$$

To complete the above integrations, only an expression for $dA_p(\beta)$ need be developed - $\tau(\beta)$ already being known. It should be noted that at a given point in time, all quantitites have definite values except β. Hence β becomes the only independent variable.

The angle ψ of Figure 6 may be found as shown in Figure 7:

$$\tan\psi = \frac{\hat{n}_{sun} \cdot \hat{j}''}{\hat{n}_{sun} \cdot i''} = \tag{IIb-17}$$

$$\frac{-\cos\alpha\cos z \cos\eta\sin\gamma_1 + \cos\alpha\sin z \cos\alpha\cos\gamma_1 + \sin\alpha\sin\eta}{\cos\gamma_1 \cos\alpha \cos z + \sin\gamma_1 \cos\alpha \sin z}$$

The vectors shown in Figure 6 may be found as functions of β to be:

$$\vec{V}_1 = -r(1 - \sin\beta)\tan\psi\, j'' \tag{IIb-18}$$

$$\vec{V}_2 = -r \tan\psi j'' \tag{IIb-19}$$

$$\vec{V}_3 = r\, n_{cyl} + \vec{V}_1 - \vec{V}_2$$

$$= r(\sin\beta\, i'' + \cos\beta\, k'') + r \sin\beta \tan\psi\, \hat{j}'' \tag{IIb-20}$$

$$\vec{V}_4 = \frac{d\vec{V}_3}{d\beta}\, d\beta \tag{IIb-21}$$

$$= r[\cos\beta\, \ddot{i}'' + \cos\beta \tan\psi\, \ddot{j}'' - \sin\beta\, k'']d\beta$$

$$|\vec{V}_4| = r[1+(\cos\beta \tan\psi)^2]^{1/2} d\beta \tag{IIb-22}$$

and the angle λ may be found from:

$$\cos\lambda = \frac{\hat{n}_{sun} \cdot \vec{V}_4}{|n_{sun} \cdot \vec{V}_4|} \tag{IIb-23}$$

$$= \frac{1}{[1+(\cos\beta \tan\psi^2]^{1/2}},$$

$$\{(\cos\alpha \cos z)(\cos\beta \cos\gamma_1 - \cos\beta \tan\psi$$

$$\cos\eta \sin\gamma_1 - \sin\beta \sin\eta \sin\gamma_1) + (\cos\alpha\sin z)$$

$$(\cos\beta \sin\gamma + \cos\beta \tan\psi \cos\eta \cos\gamma_1 +$$

$$\sin\beta \sin\eta \cos\gamma_1) + (\sin\alpha)(\cos\beta\tan\psi$$

$$\sin\eta - \sin\beta \cos\eta\,)\}$$

The element of projected area is then calculated as

$$dA_p = \sin\lambda |\vec{V}_4| \tag{IIb-24}$$

Hence the mean transmittance for the cylinder becomes:

$$\bar{\tau} = \frac{\int_{\beta_1}^{\beta_2} \tau(\beta)\sin\lambda(\beta)\ [1+(\cos\beta \tan\psi)^2]^{1/2}\, d\beta}{\int_{\beta_1}^{\beta_2} \sin\lambda(\beta)\ [1+(\cos\beta \tan\psi)^2]^{1/2}\, d\beta} \tag{IIb-25}$$

Not all the solar energy first striking the plate is absorbed by it. However, some of that which isn't will get back to the plate by multiple reflection and subsequently be absorbed by it. Assuming the multiple reflection behavior in the cylinder yields the same additional absorption behavior as for flat plates results in an effective transmittance absorptivity product for beam radiation given by:

$$(\bar{\tau}\alpha)_{eff,b} = \frac{\tau\ \alpha(\theta_T)}{1-[1-\alpha(\theta_T)]\rho_D} \qquad \text{(IIb-26)}$$

where the nomenclature is as in Appendix Ia. The quantity θ_T, however, takes on a slightly more complicated form due to the greater number of angles required to define the position of the absorbing plate in space. From Figure 4, the normal to the plate is seen to be:

$$n_p = \sin c\ i'' + \cos c\ k'' \qquad \text{(IIb-27)}$$

Hence,

$$\begin{aligned}\cos\theta_T &= n_p \cdot n_{sun} \\ &= (\cos\alpha\cos z)(\sin c\ \cos\gamma_1 + \cos c\ \sin\eta\sin\gamma_1) \\ &+ (\cos\alpha\sin z)(\sin c\ \sin\gamma_1 - \cos c\ \sin\eta\cos\gamma_1) \\ &+ \sin\alpha\ \cos c\ \cos\eta\end{aligned} \qquad \text{(IIb-28)}$$

Therefore, the beam energy transmitted through the glass cylinder and absorbed by the plate per unit area of absorbing plate becomes:

$$\frac{E_{\bar{\tau}\alpha}}{A_c} = (\bar{\tau}\alpha)_{eff,b}\frac{I_{sun}A_p}{A_c} = (\bar{\tau}\alpha)_{eff,b}H_bR_b \qquad \text{(IIb-29)}$$

where Appendix Ia nomenclature is again used, where R_b is defined by eq. (IIa-23) except that $\cos\theta_T$ is now given by eq. (IIb-28).

Ia. Diffuse Radiation

If the diffuse component of the solar radiation is truly diffuse, then whether the absorbing plate is covered by a glass cylinder or by a flat glass plate will not affect the amount of diffuse energy absorbed by the plate. Hence, the results of Appendix I may be used directly - the transmittance being evaluated as the transmittance of a beam of radiation passing through a single sheet of thin plate glass at an angle of incidence of 60°.

Ib. Solar Energy Absorbed by the Collector Plate

Accounting for the possible effects of dust and shading, the total solar energy absorbed by the collector plate will be given by:

$$\begin{aligned}S = (1-s)(1-d)\{&H_bR_b(\bar{\tau}\alpha)_{eff,b} \\ &+ H_d\frac{(1+\cos c_1)}{2}(\tau\alpha)_{eff,d} + (H_b+H_d)\rho_o \\ &\frac{(1-\cos c_1)}{2}(\tau\alpha)_{eff,d}\}\end{aligned} \qquad \text{(IIb-30)}$$

which is identical to the expression used for flat plate collectors except that for beam radiation $\bar{\tau}$ must be used. Angle c_1 in the above is again the angle of the collector plate from the horizontal and may be determined from:

$$\cos c_1 = n_p \cdot k = \cos\eta\ \cos c \qquad \text{(IIb-31)}$$

II. Calculation of Energy Losses from the Collector

The system of equations relating the heat losses, the mean plate temperature, $T_{p,m}$, the cylindrical glass cover temperature T_g, and the ambient air temperature T_a are:

$$Q_L = \frac{2A_c}{\frac{1}{\varepsilon_p} + (\frac{1}{\varepsilon_g}-1)2\frac{A_c}{A_g}}\sigma(T_{p,m}^4 - T_g^4) \qquad \text{(IIb-32)}$$

where

A_c = area of one side of the absorbing plate = 2.33 ft^2

A_g = area of entire glass cylinder = 8.143 ft^2

Also,

$$Q_L = A_g\varepsilon_g\sigma(T_g^4 - T_a^4) + A_gh_w(T_g-T_a) \qquad \text{(IIb-33)}$$

where h_w is as in Appendix II-A.

With an assumed value of $T_{p,m}$, T_g may be evaluated directly from either equation and the overall loss coefficient determined from:

$$U_L = \frac{(Q_L/A_c)}{T_{p,m} - T_a} \qquad \text{(IIb-34)}$$

III. Determination of Steady-State Useful Energy by Iteration

The useful energy collection is then given by eq.(IIa-42) using eq. (IIb-30) for S and (IIb-34) for U_L. The iteration procedure for finding the actual $T_{p,m}$ then proceeds in the same way as in in Appendix IIa, eqs.(IIa-43) to(IIa-45). For the cylindrical collector, F_R was assumed to equal 0.96. The collector efficiency factor F' and the heat transfer resistance R_{p-f} were taken to be the same as for the flat plate case.

IV. Transient Response of the Collector - Heat Capacity Effects

The simple transient analysis used in Appendix IIa was also used for the cylindrical collector.

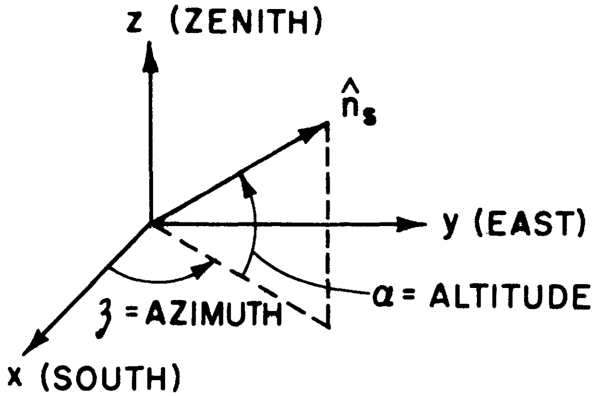


FIGURE 1: POSITION OF SUN IN SKY

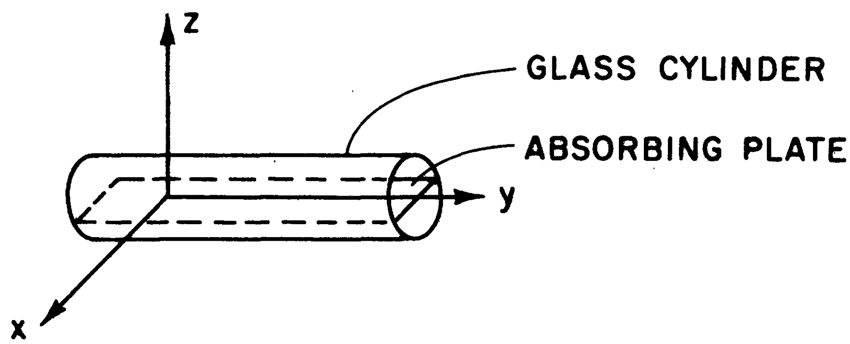


FIGURE 2: REFERENCE POSITION OF CYLINDER AND ABSORBING PLATE

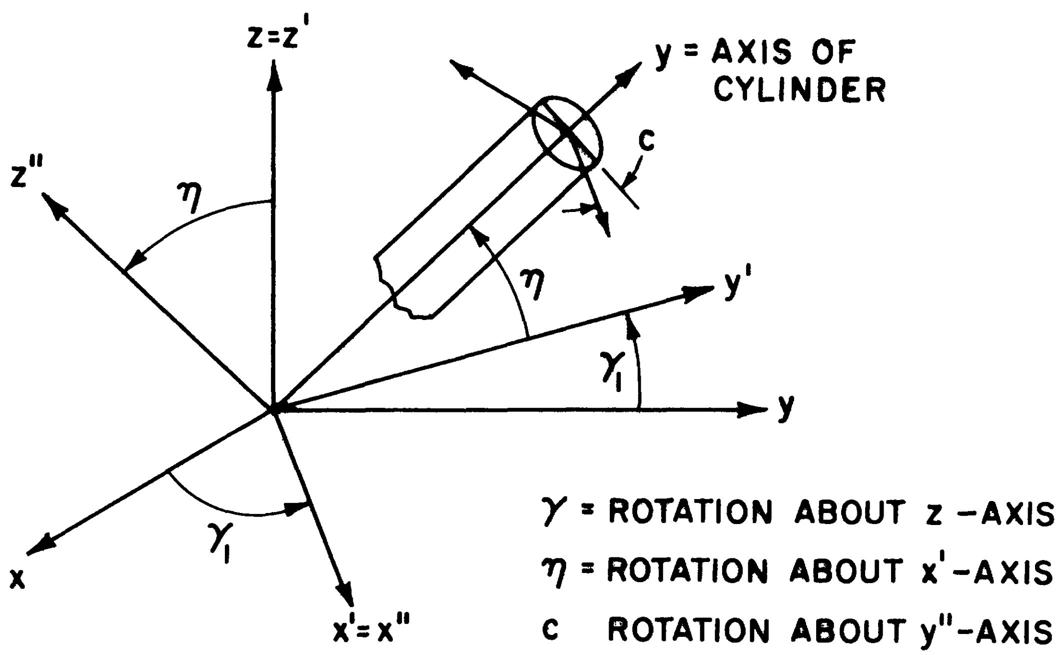


FIGURE 3: ARBITRARY ORIENTATION OF CYLINDER

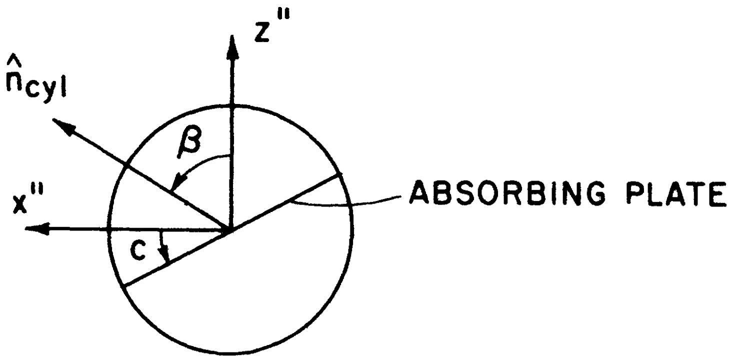
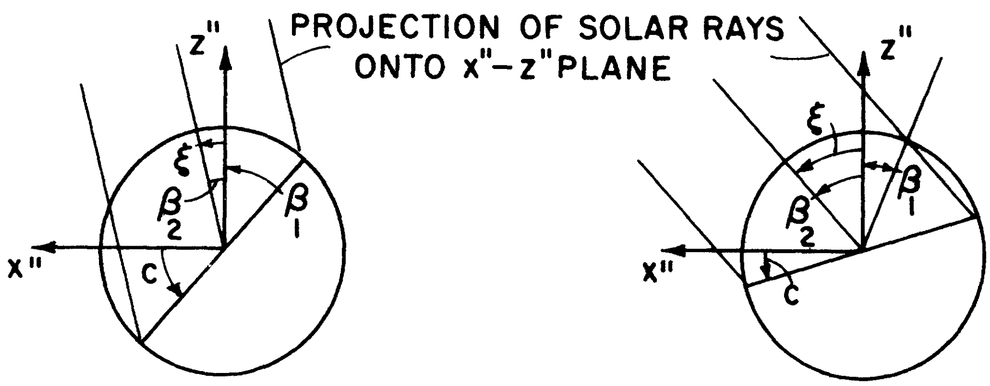


FIGURE 4: LOOKING DOWN THE AXIS OF THE CYLINDER



$c \geqslant \xi$

$\beta_1 = -\frac{\pi}{2} + c$

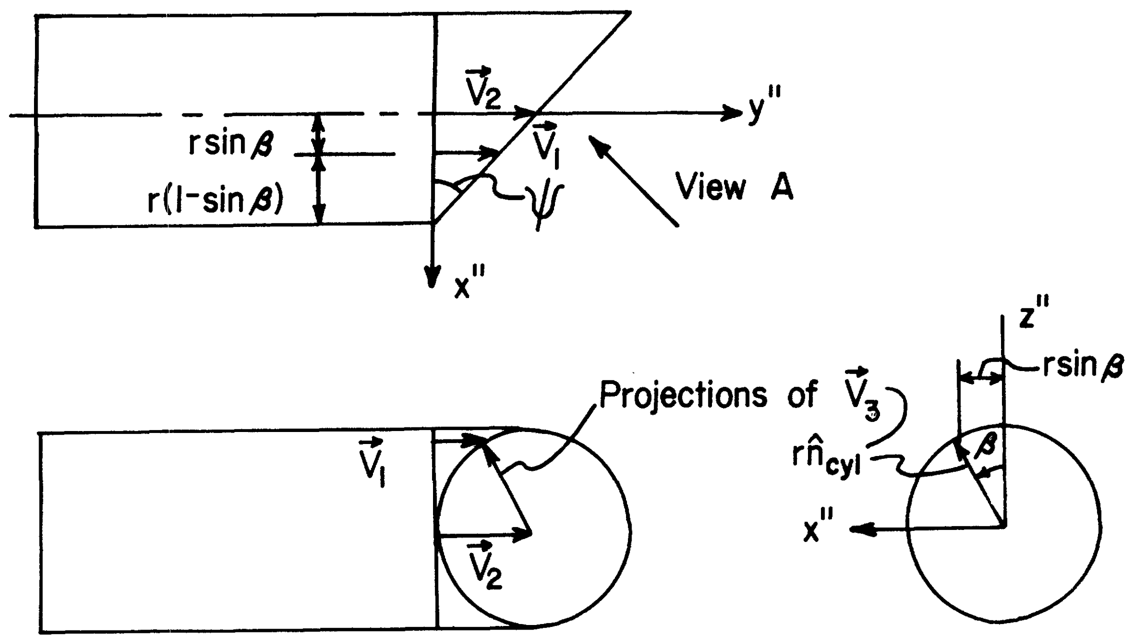
$\beta_2 = \xi$

$c \leqslant \xi$

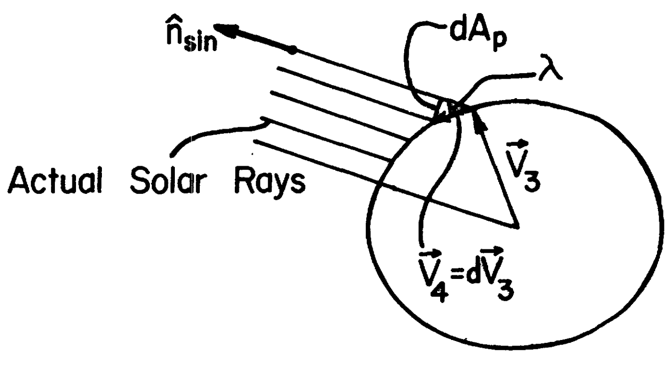
$\beta_1 = -\frac{\pi}{2} - c + 2\xi$

$\beta_2 = \xi$

FIGURE 5: SOLAR RAYS WHICH STRIKE THE ABSORBING PLATE (PROJECTION INTO THE X" - Z" PLANE)



ORTHOGRAPHIC PROJECTIONS OF CYLINDER SLICED BY A PLANE OF THE SUN'S RAYS



View A

FIGURE 6: A PLANE OF SOLAR RAYS CUTTING THE CYLINDER

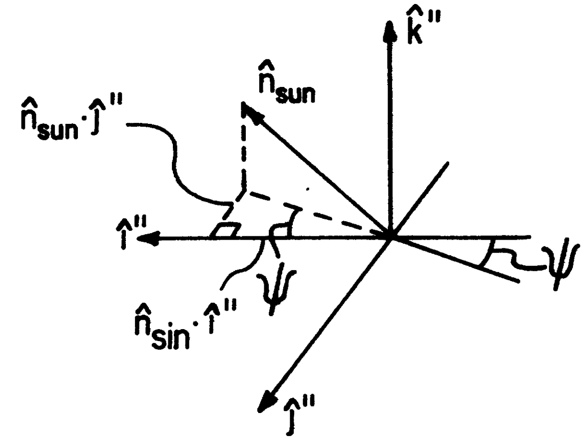
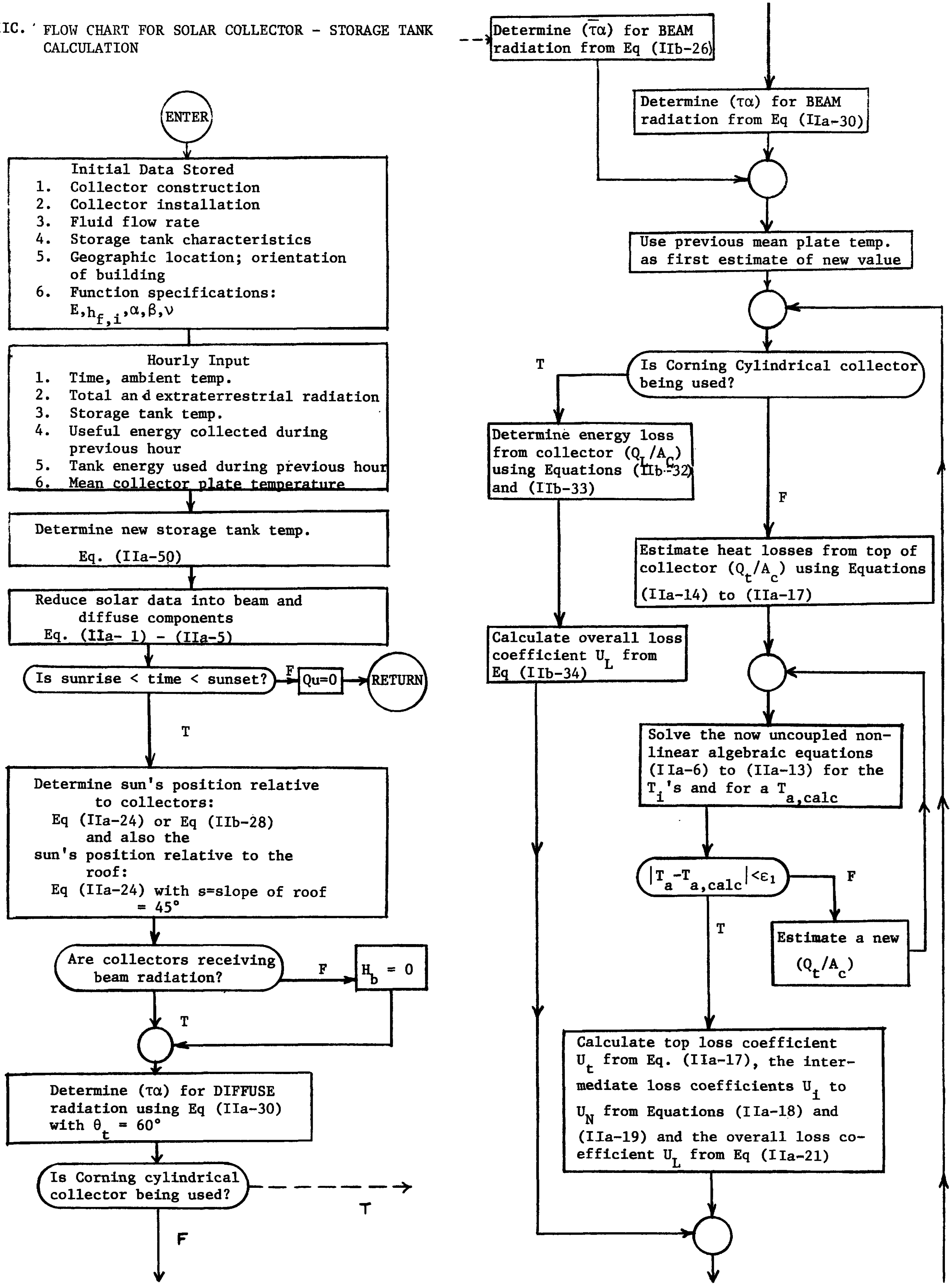
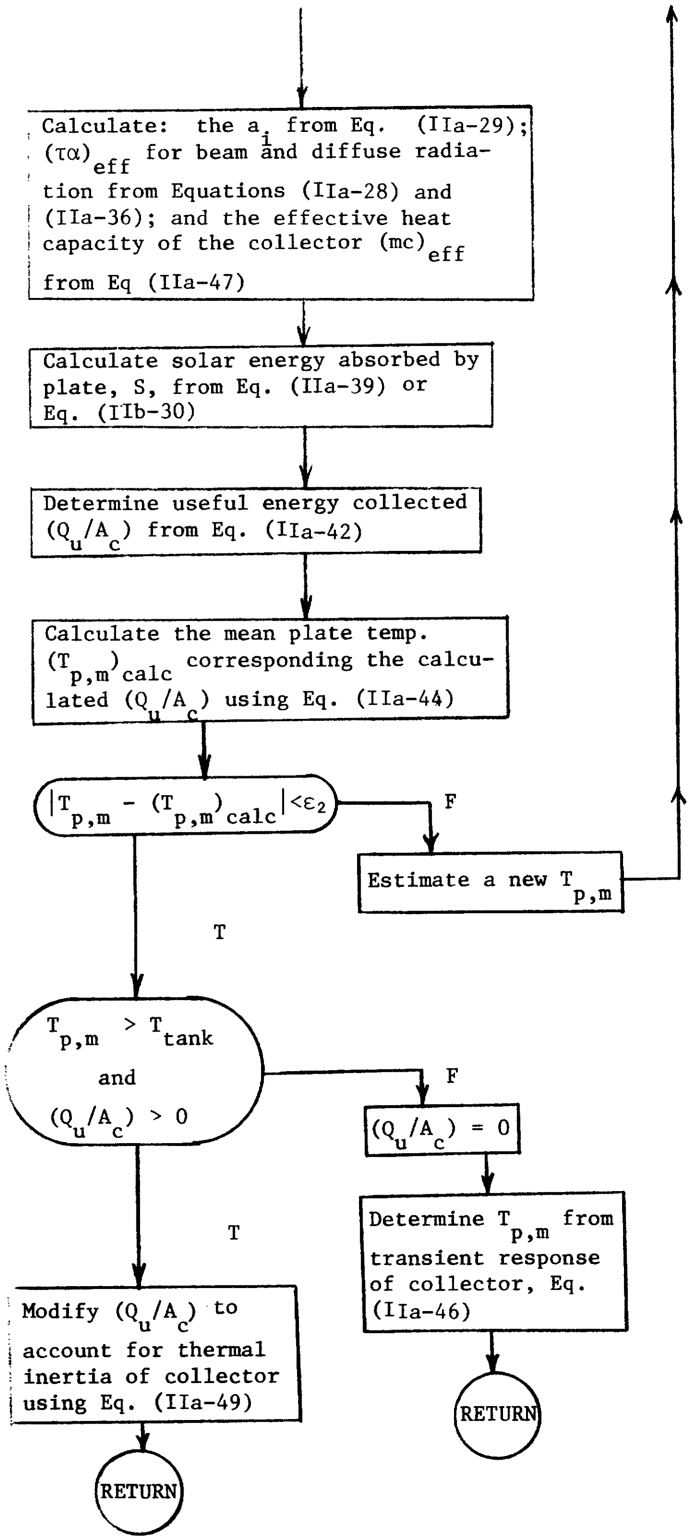


FIGURE 7: CALCULATION OF TAN ψ

IIC. FLOW CHART FOR SOLAR COLLECTOR - STORAGE TANK CALCULATION

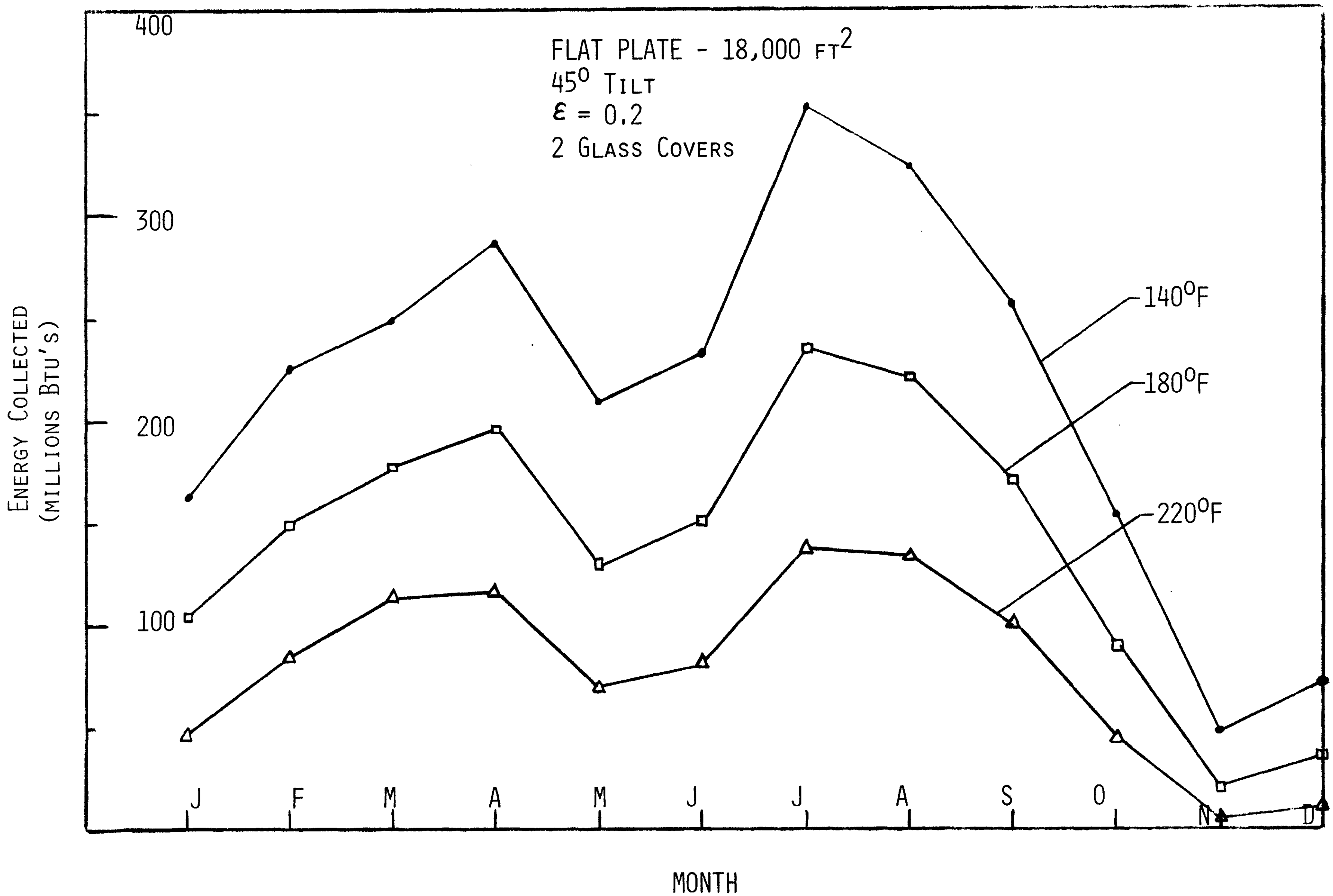




Calculate: the a_i from Eq. (IIa-29); $(\tau\alpha)_{eff}$ for beam and diffuse radiation from Equations (IIa-28) and (IIa-36); and the effective heat capacity of the collector $(mc)_{eff}$ from Eq (IIa-47)
Calculate solar energy absorbed by plate, S, from Eq. (IIa-39) or Eq. (IIb-30)
Determine useful energy collected (Q_u/A_c) from Eq. (IIa-42)
Calculate the mean plate temp. $(T_{p,m})_{calc}$ corresponding the calculated (Q_u/A_c) using Eq. (IIa-44)
$|T_{p,m} - (T_{p,m})_{calc}| < \varepsilon_2$
F
Estimate a new $T_{p,m}$
T
$T_{p,m} > T_{tank}$ and $(Q_u/A_c) > 0$
F
$(Q_u/A_c) = 0$
T
Determine $T_{p,m}$ from transient response of collector, Eq. (IIa-46)
Modify (Q_u/A_c) to account for thermal inertia of collector using Eq. (IIa-49)
RETURN
RETURN

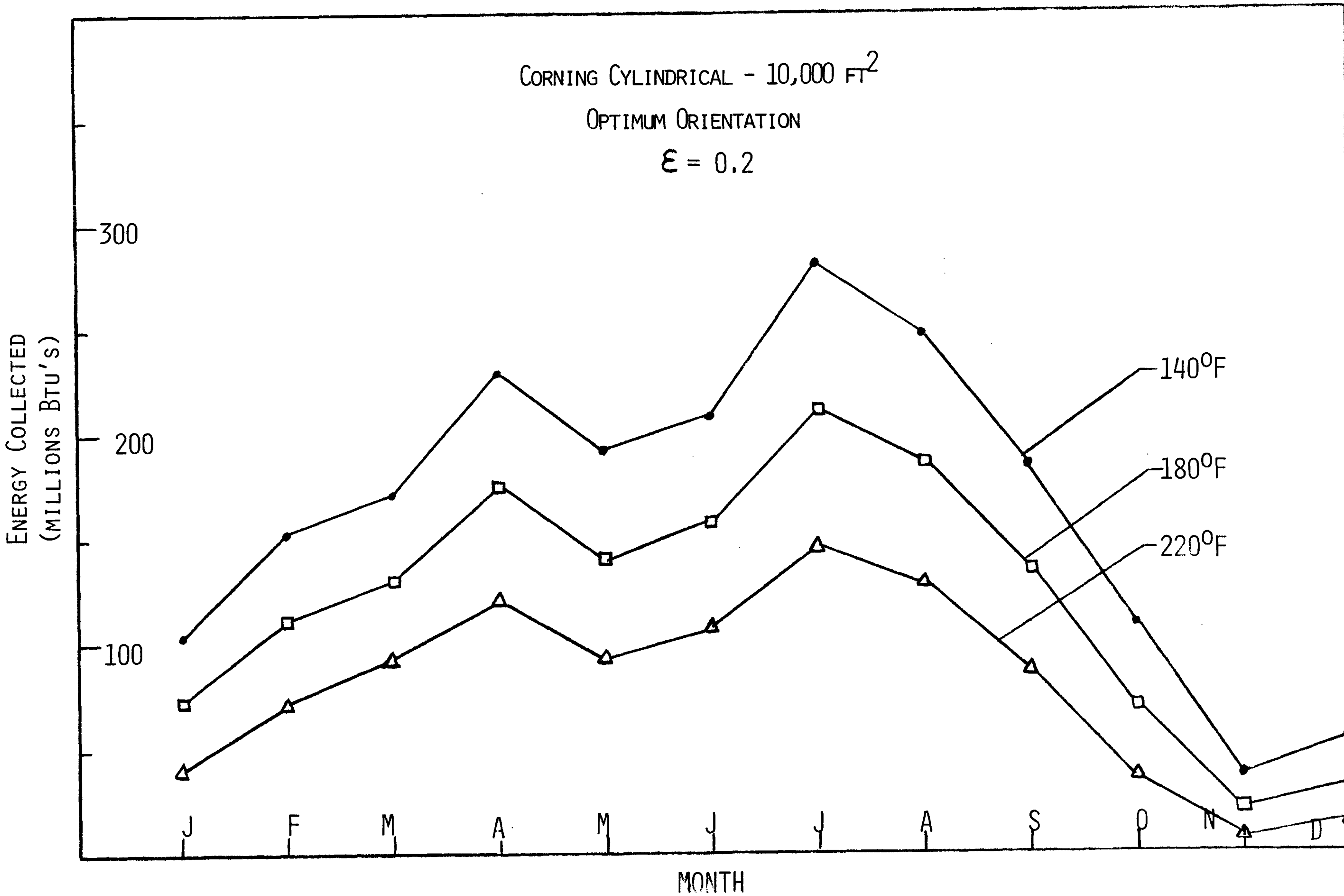
Appendix IId

Monthly Variations Corresponding to the Yearly Totals of Tables I - III



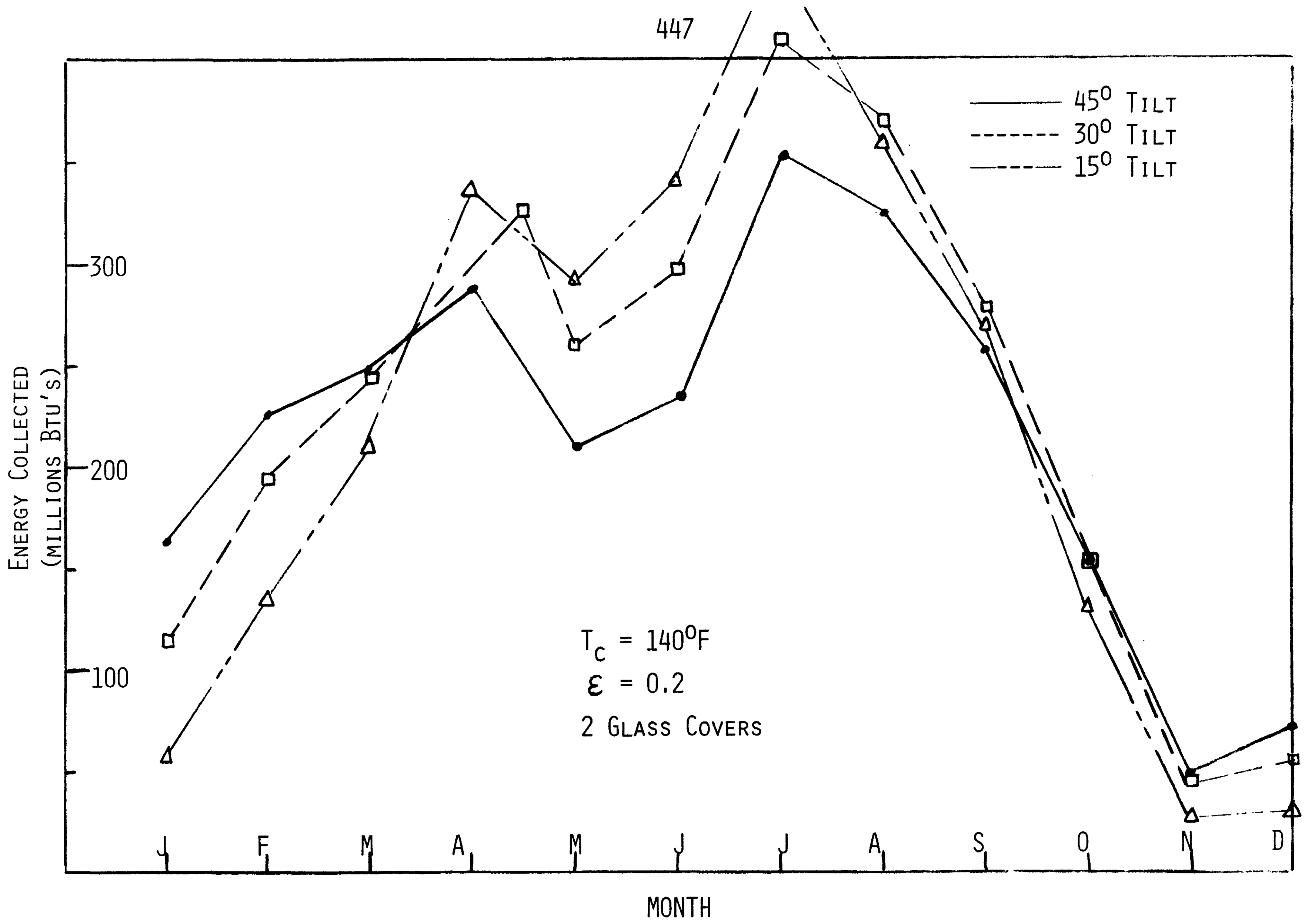
EFFECT OF COLLECTION TEMPERATURE

FIGURE IId-1



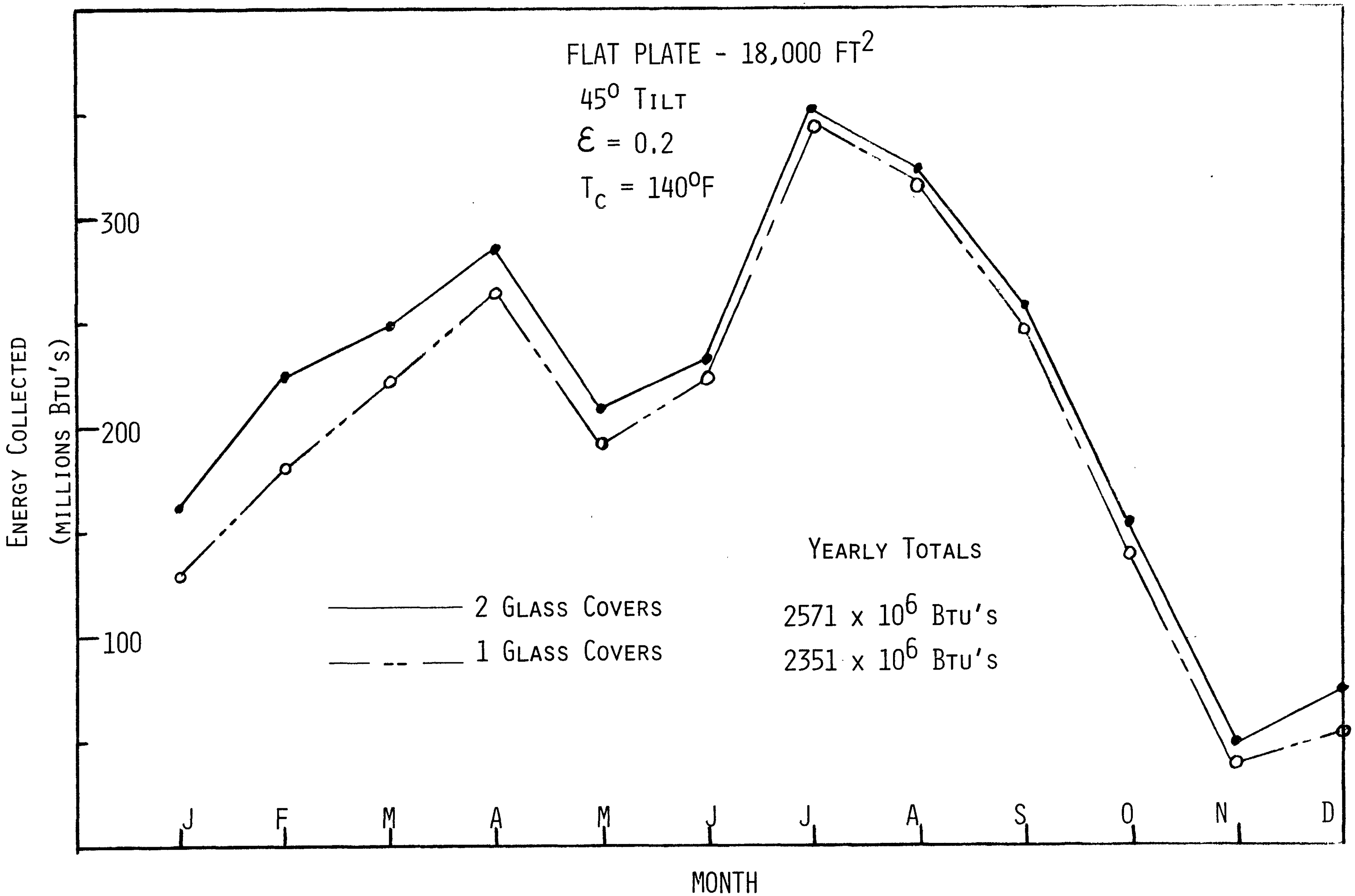
EFFECT OF COLLECTION TEMPERATURE

FIGURE IId-2



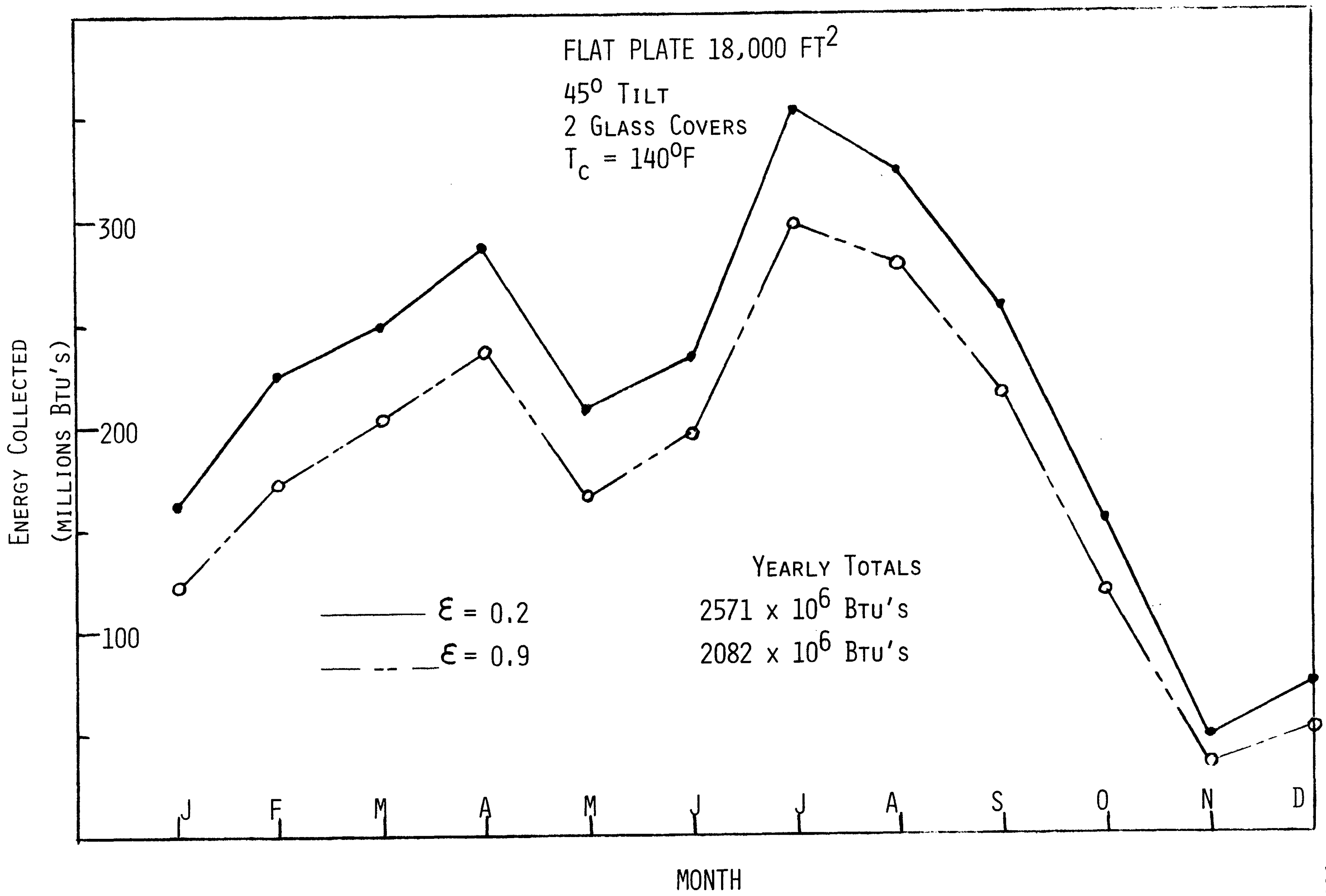
EFFECT OF ANGLE OF TILT OF COLLECTOR

FIGURE IId-3



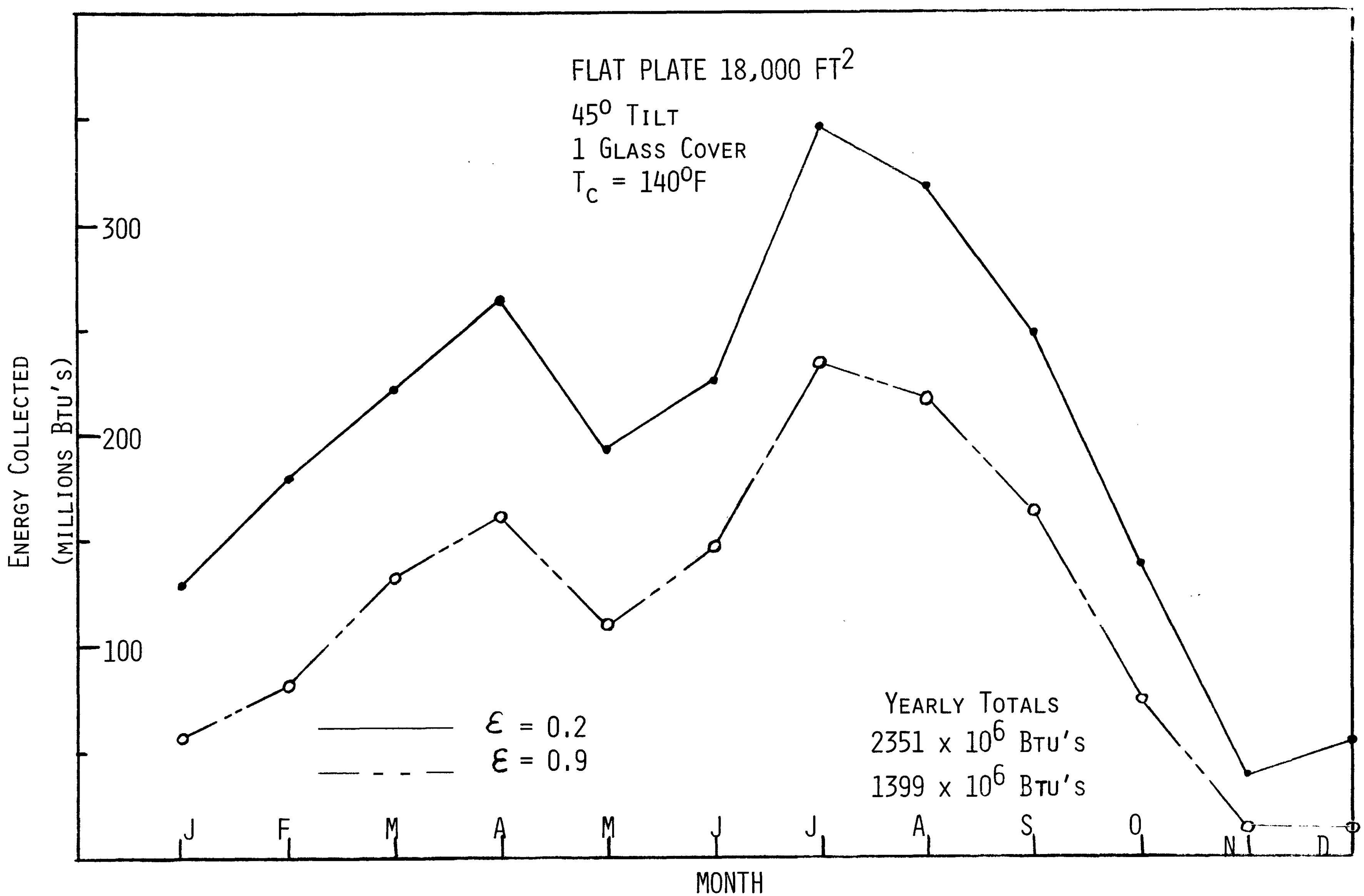
EFFECT OF NUMBER OF GLASS COVERS

FIGURE IId-4



EFFECT OF ABSORBER PLATE COATING - 2 GLASS COVERS

FIGURE IId-5



EFFECT OF ABSORBER PLATE COATING - 1 GLASS COVER

FIGURE IId-6

APPENDIX III. DEHUMIDIFIER, BUILDING LOAD, COIL AND SPRAY MODELS, AND COMPLETE HVAC SYSTEM MODEL

III-a. ANALYSIS OF THE LIQUID DESICCANT DEHUMIDIFIER, CONCENTRATOR AND ASSOCIATED HEAT EXCHANGERS

The analysis of the dehumidification systems requires combining the constitutive equations from the dehumidifier, the concentrator, the liquid-liquid heat exchanger between the concentrator and the dehumidifier and the air-air heat exchanger used with the concentrator. Each of these subsystems will first be treated and then the interconnection and control considerations will be discussed.

The dehumidifier functions as a multiport heat exchanger. For a steady state model, the net power and mass flow into the dehumidifier from the air streams, the TEG and the cooling water must be zero. In addition, it is assumed that the air leaving the dehumidifier is in chemical equilibrium with the TEG spray.

Air enters the dehumidifier at the outdoor ambient state of dry bulb temperature, T_a, and specific humidity, σ_a, and at a flow rate expressed in pounds of dry air per hour, W_a. Associated with this air state is an enthalpy, h_a, expressed in BTU per pound of dry air. The enthalpy is related to the temperature and humidity by the equation

$$h_a = 0.240\ T_a + \sigma_a(0.1517 + 63\text{E-}6\ T_a) \qquad \text{(IIIa-1)}$$

Air leaves the dehumidifier at a new state (T_d, σ_d, h_d) which is determined by the TEG equilibrium. The net power gained by the air (including the moisture which is removed at temperature T_{SC_d} through dehumidifier sump) is

$$q_{ad} = ((h_d - h_a) + (T_{SC_d} - 32)\ (\frac{\sigma_d - \sigma_a}{7000}))W_a \qquad \text{(IIIa-2)}$$

Energy is also transferred between the sumps of the dehumidifier and concentrator. TEG solution enters the sump with a volume flow rate of Q_{cd} of which a volume fraction, X_c, is TEG and the balance is water. TEG solution leaves the dehumidifier sump at a concentration, X_d, and a volume flow rate of Q_{dc}. The difference in flow rates must, from conservation of mass, be equal to the rate at which moisture is removed from the air, or

$$Q_{dc} = Q_{cd} + W_a(\sigma_a - \sigma_d)/(70003) \qquad \text{(IIIa-3)}$$

Furthermore, the net flow of pure TEG must be zero. Therefore

$$X_d Q_{dc} = X_c Q_{cd} \qquad \text{(IIIa-4)}$$

In normal operation, the concentration, X_c, is maintained at a set level by boiling point analyzer in the concentrator and the flow rate, Q_{cd}, is determined by a pump in response to measurement of the air leaving condition. Then equations (IIIa-3) and (IIIa-4) can be combined to determine the concentration, X_d, in terms of controlled or manipulated variables, X_c and Q_{cd} and ambient conditions

$$X_d = X_c\ Q_{cd}/(Q_{cd} + W_a(\sigma_a - \sigma_d)/(70003)) \qquad \text{(IIIa-5)}$$

Note that this is not an explicit equation since σ_d is a function of X_d. The power added to the dehumidifier by the TEG depends on the pump temperatures and the flow rates between the units:

$$q_{dc} = \rho C_p\ Q_{cd}(T_{dc} - T_{SC_d}) \qquad \text{(IIIa-6)}$$

This assumes that Q_{cd} is nearly equal to Q_{dc} and that T_{SC_d} is nearly the same as the sump temperature. Also note that the energy associated with the removal of moisture has been included in equation (IIIa-2).

When TEG absorbs water an additional heat load, called a heat of mixing, of about 100 BTU per pound is released. This is normally a negligible load, but if it is included the total heat load in the dehumidifier becomes:

$$q_d = ((h_d - h_a) + (T_{SC_d} - 32 + 100)\left(\frac{\sigma_d - \sigma_a}{7000}\right))W_a + \rho C_p Q_{cd}(T_{dc} - T_{SC_d}) \quad \text{(IIIa-7)}$$

Substituting for the air enthalpies gives

$$q_d = (0.240(T_d - T_a) + (0.1614 + T_{SC_d}/7000)(\sigma_d - \sigma_a) + 63E\text{-}6(T_d\sigma_d - T_a\sigma_a))W_a$$

$$+\rho C_p Q_{cd}(T_{dc} - T_{SC_d}) \quad \text{(IIIa-8)}$$

For a dehumidifier with sufficient contact area between the TEG and the air stream, the air leaving dry bulb temperature, the leaving humidity and the spray contact temperature are related by equilibrium vapor pressure curves of the form shown in Figure III-1. As is indicated, the humidity, but not the dry bulb leaving temperature, is a function of the TEG concentration. As is discussed in a later section, these curves can be approximated by simple functions over the range of contact temperatures normally encountered. By using these functions, it is possible to use X_d and any one of T_d, T_{SC_d} or σ_d to solve for the remaining two variables.

Thus, to complete the analysis, it is only necessary to add one additional equation, namely a relation between spray contact temperature and the energy removed by the cooling water. This equation can be derived from a detailed analysis of the heat exchanger properties of the contact area. For typical units, an empirical correlation has been derived in the form:

$$q_d = B_d \cdot C_{k_d} \cdot C_{a_d} \cdot T_{LM_d} \qquad \text{(IIIa-9)}$$

In this equation, B_d is a basic rating factor for the unit; B_d is normally proportional to the number of heat exchanger coils used and is a function of the geometry of the coils. C_{k_d} is a correction factor which is a function of the cooling water flow rate for any given unit. A typical curve for C_{k_d} is shown in Figure III-2. C_{a_d} is a correction factor to correct for off-design air flow rates. C_{a_d} (W_{a_d}) is typically of the form:

$$C_a(W_a) = (W_a/W_{a_{max}})^{0.419} \qquad \text{(IIIa-10)}$$

Finally, T_{LM_d} is the log mean temperature across the heat exchanger

$$T_{LM_d} = \frac{\Delta T_d}{\ln\left(1 - \dfrac{\Delta T_d}{T_{SC_d} - T_{CT}}\right)} \qquad \text{(IIIa-11)}$$

ΔT_d = drop in cooling water temperature

T_{CT} = cooling water entering temperature

ΔT_d is determined by equating the heat removed by the cooling water to the total heat load from equation (IIIa-8)

$$\Delta T_d = \frac{q_d}{\rho_w \, C_{p_w} \, Q_{c_w}} \qquad \text{(IIIa-12)}$$

Combining equations (IIIa-9) through (IIIa-12) gives an implicit non-linear equation

$$\frac{B_d \, C_{k_d} \, C_{a_d}}{\ln\left[1 - \dfrac{q_d}{\rho_w \, C_{p_w} \, f(C_{k_d})(T_{SC_d} - T_{CT})}\right]} + \rho_w \, C_{p_w} \, f(C_{k_d}) = 0 \qquad \text{(IIIa-13)}$$

where q_d is given by equation (IIIa-8) and $f(C_{k_d})$ is the inverse relation between cooling water flow rate and C_{k_d}.

$$Q_{cw} = f(C_{k_d}) \qquad \text{(IIIa-14)}$$

Equations (IIIa-5), (IIIa-8), (IIIa-13) and Figure III-1 provide four nonlinear relationships which must be solved simultaneously. The assumed inputs are T_a, W_a, σ_a, T_{CT}, Q_{cd} and T_{dc}. There are four unknowns: T_d, σ_d, T_{SC_d}, and C_{k_d}.

The normal procedure is to specify an outlet humidity, σ_d, and a dehumifier concentration. Then C_{k_d} may be found by an iterative search. If C_{k_d} is excessive (>1.25) then either X_d can be increased by increasing Q_{cd} or σ_d can be increased until a reasonable value of C_{k_d} will suffice.

The basic function of the concentrator is identical to the dehumidifier. How water is used in place of cold water so that moisture will be driven out of the TEG. A boiling point analyzer is used to determine the concentration X_c and the hot water flow rate modulated accordingly. Thus the concentrator equations are, by analogy to Equation (IIIa-8).

$$q_c = [0.240(T_{c1}-T_{ca})+(0.147+T_{SC_c}/7000)(\sigma_c-\sigma_r)+63E-6(T_{c1}\sigma_c-T_{ca}\sigma_r)]W_c$$

$$+ \rho C_p Q_{dc}(T_{cd}-T_{SC_c}) \qquad \text{(IIIa-15)}$$

where

q_c = heat flux in concentrator

T_{c1} = dry bulb temperature leaving the concentrator

T_{ca} = dry bulb temperature entering the concentrator

T_{SC_c} = spray contact temperature in the concentrator

σ_c = specific humidity leaving the concentrator

σ_r = specific humidity entering the concentrator

W_c = mass flow of dry air through the concentrator

ρC_p = heat capacity of TEG solution

Q_{dc} = volume flow rate dehumidifier to concentrator

T_{cd} = temperature of TEG entering the concentrator

The 0.1471 is used in equation (IIIa-15) in place of 0.1614 in equation (IIIa-8) because the term corresponding to the heat of mixing included in equation (IIIa-8) is less than 5%.

The heat flux must be carried off by heat exchanger action and transferred to the hot water supply. This gives rise to an equation similar to equation (IIIa-13).

$$\frac{B_c C_{k_c} C_{a_c}}{\ln\left[1 - \dfrac{q_c}{\rho_w C_{p_w} f_c(C_{k_c})(T_{SC_c} - T_{HW})}\right]} + \rho_w C_{p_w} f_c(C_{k_c}) = 0 \qquad \text{(IIIa-16)}$$

Where the subscript c is used to refer to parameters and variables in the concentrator. T_{HW} is the temperature of the available hot water supply. The drop in hot water temperature, also determined by an energy balance, is

$$\Delta T_c = \frac{q_c}{\rho_w C_{p_w} Q_{HW}} \qquad \text{(IIIa-17)}$$

While the dehumidifier and concentrator could be directly connected, this is not done since it would result in an unacceptably low thermal efficiency for the system. To raise the efficiency, two heat exchangers are employed. One is a liquid-liquid heat exchanger used on the TEG exchanger flow and the other is an air-air heat exchanger installed at the concentrator which uses concentrator exhaust air to preheat the air entering the concentrator.

The heat exchangers have been analyzed by the NTU (Rohsenow & Choi) method. For the liquid-liquid exchanger,

$$T_{dc} - T_{SC_d} = \frac{(1 - \varepsilon_\ell)\, Q_{cd}/Q_o}{\varepsilon_\ell + (1 - \varepsilon_\ell)Q_{cd}/Q_o} (T_{SC_c} - T_{SC_d}) \qquad \text{(IIIa-18)}$$

and

$$T_{cd} - T_{SC_c} = \frac{(1 - \varepsilon_\ell)\, Q_{cd}/Q_o}{\varepsilon_\ell + (1 - \varepsilon_\ell)Q_{dc}/Q_o} (T_{SC_d} - T_{SC_c}) \qquad \text{(IIIa-19)}$$

where

ε_ℓ = heat exchanger effectiveness

Q_o = design flow rate for the heat exchanger

This analysis assumes a negligible difference in density and specific heat as TEG concentration varies in the humidifier and the concentrator.

The air to air heat exchanger is analyzed in a similar fashion. In this case, however, the flow rates on both sides are equal since the entire concentrator air flow passes in both directions. The difference in specific heat due to change in humidity is ignored. Thus the equation for the concentrator entering air temperature in terms of the building return air temperature and the concentrator dry bulb leaving temperature is:

$$T_{ca} = T_{c\ell} - \frac{(1 - \varepsilon_a)W_c/W_{co}}{\varepsilon_a + (1 - \varepsilon_a)W_c/W_{co}} (T_{c\ell} - T_{exh}) \qquad \text{(IIIa-20)}$$

where

T_{exh} = building exhaust temperature

ε_a = heat exchanger effectiveness

W_{co} = design mass flow rate

X_c	TEG concentration in concentrator (% vol.)
X_d	TEG concentration in dehumidifier (% vol.)
ε_a	Heat exchanger effectiveness (air-air)
ε_ℓ	Heat exchanger effectiveness (liquid-liquid)
ρ	Density of TEG solution lb/ft^3
ρ_w	Density of water (lb/ft^3)
σ_a	Ambient specific humidity (gr/lb)
σ_c	Specific humidity leaving the concentrator (gr/lb)
σ_d	Specific humidity leaving dehumidifier (gr/lb)
σ_o	Specific humidity in occupied space
σ_s	Entering air specific humidity (grains)

The preceding equations serve to define the performance of the dehumidification system in terms of the environmental conditions and the temperature of the heating and cooling water supplies for any choice of operating strategy. To obtain numberial results, a strategy was adopted in which the humidity of the air leaving the dehumidifier was minimized using fixed concentrations of 92.5% and 96.0% for the TEG. (See Flow chart in Figure III-3). It was assumed that the cooling water would be a fixed increment of 7°F above the outside wet bulb temperature. In order to maintain the dehumidifier concentration, the flow rate between the units was adjusted. Table III-1 and III-2 shows the system parameters and dehumidification system. Table III-2 shows the results for the four relative humidities and three different ambient temperatures. For the first three pages, the cooling tower water is assumed to be 7°F above the outside wet bulb. For the fourth case, chilled water at 55°F is used. For all cases, the results are shown as functions of the available solar water temperature. Two coefficients of performance have been defined. In the first case, the COP assumes that cooling water energy is free so the COP measures the ratio of energy removed from the inlet air to energy supplied by the heating water. The second COP includes the cooling energy in the denominator. This report assumes the use of cooling water, not chilled water, and so the first COP was used for the economic analysis.

In each situation, heating water temperature above the temperature for which the dehmidifier performance was limited by the cooling water were not considered. As can be seen from the tables, the attainable humidity decreases and the COP improves as the heating water temperature is increased.

III-b DYNAMIC MODEL OF THE BUILDING HVAC SYSTEM

In order to accurately determine energy savings for the proposed system, models of the various components of the HVAC system were developed and combined in several configurations. Each configuration was used in a simulation in which recorded hourly weather and solar data were used to determine the energy requirements of the HVAC system. The simulation covered a full year of operation with a building schedule supplied by the architects. The configurations were individually optimized in such a way as to correspond to realistic operational policies for each configuration. As a result, the temperature and humidity in the occupied space were slightly different in each case.

The component and subsystem models are described below. For each configuration, the operating strategy is also discussed. These basic subsystems include:

(1) A building load model

and (2) A cooling coil model with and without spray

which are used in each system and also:

(3) A conventional system operating strategy model

(4) A dehumidification system model with and without spray and with operating strategy

(5) An absorption system model with operating strategy

Building Load Model

The building load to the HVAC system consists of sensible heat gains from lighting and electric power consumption, heat added by supply and exhaust fans, and both sensible and latent heat due to occupants. There are no radiant or skin losses since the system considered controls for an interior space surrounded by a perimeter which is separately controlled to the same temperature.

To obtain thermal comfort in the occupied space of the building, it is necessary to control both the temperature and the humidity. The conventional method is to cool the air until enough water is removed to establish the desired humidity level and then, if necessary, to reheat the air to obtain temperature control. The necessity for reheat depends on the combination of sensible and latent cooling and on the air flow rate which is desired.

If the temperature and humidity in the occupied space are specified, then the temperature and humidity leaving the cooling coil can be determined from the heat load, the moisture load and the air flow rate as:

$$T_c = T_o - (q_s + q_d)/c_p W \qquad \text{(IIIb-1)}$$

$$\sigma_c = \sigma_o - g_s/W \qquad \text{(IIIb-2)}$$

where:

T_o = Drybulb in occupied space

σ_o = Specific humidity in occupied space

T_c = Drybulb leaving cooling coil

σ_c = Specific humidity leaving cooling coil

q_s = Heat generated in occupied space per s.f.

q_d = Heat added to supply duct fan per s.f.

g_s = Moisture generated in occupied space per s.f.

c_p = Specific heat of air

W = Flow rate of air per s.f.

The heat and moisture loads in the occupied space, q_s and g_s, are governed primarily by the occupancy and the electric requirements. For Citicorp, the occupancy density is expected to be 0.01 person/s.f. Each person will dissipate 240 BTU/hr of sensible heat and 205 BTU/hr of latent heat (1400 gr/hr of moisture which must be removed by the HVAC system). The electrical load will be 2 W/s.f. for lighting and 0.5 W/s.f. for electrical utilities. The lighting load will be divided with 70% going to the occupied space and 30% going directly into the return plenum. In the dynamic model, it was assumed that in the morning the lighting load would build exponentially from 70% of full power with a time constant of 1 hour. This is due to the absorption of heat by the building and furnishings since the building begins each day cooler than the normal operating temperature.

The heat load due to the supply duct fan is a substantial fraction of the total load in typical situations. This load is the product of air flow rate and the pressue drop in the ducting. Since the pressure drop is approximately proportional to the square of the flow rate, the fan power will vary with the cube of the flow rate for a given duct cross section. During the design phase, economic considerations may favor designing for a particular fan power and then reducing the duct size. For Citicorp, each air handler has two supply fans and one return fan, at the proposed flow rate of 0.75 CFM/s.f., the pressure heads will be 5"W.G. for the supply fans and 2"W.G. for the return fan. The fan power will be 75 HP for each supply fan and 25 HP for the return fan. Since the air handlers serve 164000 s.f., this converts to a total supply fan load of 2.33 BTU/hr/s.f. and a return fan load of 0.39 BTU/hr/s.f.

Equations (IIIb-1) and (IIIb-2) give the required state of the air leaving the cooling coil as a function of flow rate. If these equations are plotted on a psychrometric chart, then the intersection of the resulting curve with operating characteristic of the cooling coil indicates the flow rate which is required to balance the building load without reheat. Figure III-4 shows the result for the Citicorp Center. Curve A is for constant fan power and would only be appropriate if the duct size were adjusted to a specific flow rate. Curve B assumes the duct size will not be changed from the current design and that the fan efficiency is not a function of flow rate. This would mean selecting an appropriate size fan from a family with similar characteristics. For a variable flow fan, the efficiency would decrease at low flow rates giving a curve intermediate between A and B.

The optimal flow rate for Citicorp Center is about 0.3 CFM/s.f. The minimum flow rate of fresh air required by the building code for the planned occupancy is 0.14 CFM/s.f. The design flow rate is 0.75 CFM/s.f. which is above the optimal flow rate. The result of increasing the flow rate in the absense of reheat, is to increase the humidity since the temperature is being regulated.

It is not practical to lower the air flow rate below 0.75 CFM/s.f. since the lower flows would not provide sufficient air motion in the occupied space to avoid stagnant areas in regions where funiture or partitions might reduce circulation. Further, the low temperature of the air entering the space might result in a comfort problem near the air inlets. However some air could bypass the main cooling coil, preferably within the local terminal boxes on each floor. Then by just specifying the mixing ratio room temperature, air flow, entering temperature and humidity could all be controlled. Such a modification would require considerable additional hardware as well as design changes and so is not proposed for Citicorp, but should be considered for future buildings.

Conditioning Coil Model

The conditioning coil accepts a mixture of fresh air and building return air. The mixture is cooled and dehumidified according to the model below. The flow rate of chilled water in the conditioning coil is regulated to maintain an exit air dry bulb temperature which is determined from equation (IIIb-1)

Figure III-5 shows a psychrometric plot of a conditioning coil. In a non-spray conditioning coil, air at the entering state is cooled sensibly until the state approaches 100% relative humidity. Then a combination of latent and sensible cooling takes place. The details of the process depend on the characteristics of a particular conditioning coil. In a typical case, the cooling is sensible until the relative humidity reaches 75%. Then the humidity increases to 90% while the temperature drops 10°F. If it is assumed that the humidity approaches 100% exponentially, then a model for the process is:

$$r = 100 - 25e^{0.09163(T_{CHILL} - T_{75\%})} \qquad \text{(IIIb-3)}$$

where

r = relative humidity in % at exit

T_{CHILL} = exit temperature

$T_{75\%}$ = temperature at which entering air is 75% saturated

If the entering air is already more than 75% saturated, the equation can be modified to:

$$r = 100 - (100 - r_s)e^{0.09163(T_{CHILL} - T_s)} \tag{IIIb-4}$$

where

r_s = relative humidity of entering air (%)

T_s = temperature of entering air

If the entering air is so dry that the relative humidity at the exit temperature is still less than 75%, then the cooling is sensible all the way.

$T_{75\%}$ is a function of the entering air state. If the specific humidity of the entering air is given as σ_s, in grains, then the vapor pressure entering is:

$$P_{vs} = B \cdot \sigma_s/(4354 + \sigma_s) \tag{IIIb-5}$$

where

B = barometric pressure (psi)

P_{vs} = entering vapor pressure (psi)

σ_s = entering specific humidity (grains)

Then

$$T_{75\%} = T_{vap}\ (P_{vs}/0.75) \tag{IIIb-6}$$

where

$T_{vap}(p)$ = inverse saturation curve for water,(saturation temperature as a function of saturation pressure).

III-c EVAPORATIVE (SPRAY) COOLING

The conditioning coil can be run with an integral spray unit. The moisture content of the entering air is less than the saturation level at the desired exit temperature. The process uses less energy in the conditioning coil because of evaporative cooling by the spray.

The operation of the spray system is somewhat unconventional since, if water is sprayed on the cooling coil, resulting in a combination of sensible and evaporative cooling, it is possible to use spray even when the incoming wet bulb is above the desired outlet wet bulb temperature. The situations with and without spray are shown in Figure III-6. In the summer operation, energy is saved since the air exits the cooling coil near 90% relative humidity whereas on a warm dry day without spray, the exit humidity would be less than 90%. The saving in energy comes from the difference in enthalpy at the specified exit temperature. Spray can also be used during winter operation with beneficial results. The extremely low humidity which results from mixing cold dry outside air with the return air, can be avoided by using spray.

For cases with or without spray the configuration coil model computes the amount of energy required to take entering air to the exit dry bulb temperature and the appropriate relative humidity depending upon the use of spray. Energy required to drive the cooling coil is obtained from a steam driven chiller.

III-D FUNCTIONAL MODEL FOR TEG EQUILBRIUM DATA

To model the dehumidifier and the concentrator it is necessary to know the relationship between the temperatures and the vapor pressure for TEG under equilibrium conditions. Some typical data was provided by a manufacturer in graphical form over a limited temperature range. In order to extrapolate the data, and to obtain a conventional form for computation, a regression analysis was performed.

From the thermodynamic principles, one would expect an equilibrium relation of the form

$$\ln P = A + B/T \qquad \text{(IIId-1)}$$

where

P = vapor pressure of water over TEG

T = absolute temperature

A,B = empirical constants to be determined.

By using a standard regression analysis, values of A and B were determined so that equation(IIId-1)agreed with the data as well as possible, in a least squares sense, over the available range. Then the equation was used to determine the relationship at temperatures outside the calculated range. The coefficients A and B are functions of the concentration of

TEG. Figure III-7 and III-8 show the values of A and B. These values were used to produce the equilibrium vapor pressure curves of figure III-1.

Nomenclature for Appendix III

A	Emperical constant in equation IIId-1
B	Barometric Pressure (psi)
B_c	Basic rating factor for concentrator heat exchanger (Btu/°F)
B_d	Basic rating factor for dehumidifier (Btu/°F)
Ca_c	Air flow coefficient for concentrator
C_{ad}	Air flow coefficient for dehumidifier
C_{kc}	Water flow coefficient for concentrator
C_{kd}	Water flow coefficient for dehumidifier
Cp	Specific heat of TEG solution Btu/ft^3
Cp_w	Specific heat of water (Btu/lb/°F)
f	Heat exchanger energy balance function
f_c	Concentrator energy balance function
g_s	Moisture generated in occupied space (gr/hr ft^2)
h_a	Ambient enthalpy (Btu/lb)
h_d	Enthalpy leaving dehumidifier
P	Saturation pressure (psi)
P_{vs}	Entering air vapor pressure (psi)
q_{ad}	Heat load due to air in dehumidifier (Btu/hr)
q_c	Heat load in concentrator (Btu/hr)
q_d	Total dehumidifier heat load (Btu/hr)
q_s	Heat generated in occupied space (Btu/hr ft^2)
Q_{cd}	Flow rate of TEG entering dehumidifier (ft^3/hr)
Q_{cw}	Cooling water flow rate (ft^3/hr)
Q_{dc}	Air flow coefficient for concentrator
Q_o	Design flow rate for heat exchanger (ft^3/hr)

r	Relative humidity at the chiller exit
r_s	Relative humidity of air entering the chiller
T	Absolute temperature
T_a	Ambient (outside) temperature (°F)
T_c	Dry bulb leaving cooling coil
T_{ca}	Temperature of air entering the concentrator (°F)
T_{cd}	Temperature of TEG entering concentrator pump (°F)
T_{CHILL}	Temperature at the chiller exit (°F)
T_{cl}	Temperature of air leaving the concentrator (°F)
T_{cM_d}	Log mean temperature drop across cooling coil (°F)
T_{cT}	Temperature of cooling water (°F)
Tex_h	Temperature of air as heat exchanger exhaust (°F)
T_{HW}	Temperature of concentrator hot water supply (°F)
T_o	Drybulb temperature in occupied space (°F)
T_s	Temperature of air entering the chiller (°F)
$T_{s_{c_c}}$	Spray contact temperature in the concentrator (°F)
$T_{s_{c_d}}$	Spray contact temperature in dehumidifier (°F)
T_{vap}	Saturation temperature (°F)
$T_{75\%}$	Temperature at which the entering air is 75% saturated (°F)
ΔT_d	Temperature drop across dehumidifier cooling coil (°F)
W	Flow rate of air per square foot of occupied space (lb/hr ft^2)
W_a	Makeup air flow (lb/hr)
$W_{a_{max}}$	Maximum rated air flow for dehumidifier (lb/hr)
W_c	Air flow rate through concentrator (lb/hr)
W_{c_o}	Design mass flow rate for concentrator (lb/hr) (°F)

TABLE III-1

SYSTEM PARAMETERS FOR STEADY STATE ANALYSIS OF SOLAR POWERED DEHUMIDIFICATION

SYSTEM PARAMETERS	
Air Flow Rate	92,500 #/hr
Maximum Air Flow - Dehumidifier	106,000 #/hr
Concentrator	112,000 #/hr
Basic Rating Factor-Dehumidifier	300×10^5 BTU/hr-°F
Concentrator	1.01×10^5 BTU/hr-°F
Return Air Temperature	79°F
Return Air Humidity	67.4 gr/#
Desired Humidity Leaving Dehumidifier	51.4 gr/#
Heat Exchanger Effectiveness	
Liquid/Liquid ε_ℓ	0.75
Air/Air ε_a	0.70
TEG Conditions	
Dehumidifier	92.5% by volume
Concentrator	96.0% by volume
Maximum Flow Rate	50 gpm

MAJOR HEADINGS FOR TABLE III-2

R-HUM	Outside relative humidity
WET BULB	Outside wet bulb (°F)
GRAINS	Outside humidity (gr/#)
ENTHALPY	Outside enthalpy (BTU/#)
COOLING WATER	Cooling Water Temperature (°F)

Column Headings for Table III-2

SOLAR	Solar water temperature (°F)
DRYBULB	Temperature leaving dehumidifier (°F)
GRAINS	Humidity leaving dehumidifier (gr/#)
ENTHALPY	Enthalpy leaving dehumidifier (BTU/#)
CK-D	Coil Coefficient in dehumidifier C_{k_d}
CK-C	Coil Coefficient in concentrator C_{k_c}
COP-1	Coefficient of performance, neglecting cooling water
COP-2	Coefficient of performance, including cooling water

OUTSIDE TEMPERATURE = 70.

R-HUM= 25.% WET BULB= 52. GRAINS= 27.6 ENTHALPY= 21.1 COOLING WATER= 59.5

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	65.1	25.	19.5	0.35	0.05	1.18	0.53
135.0	65.1	25.	19.5	0.35	0.05	1.18	0.53
140.0	65.1	25.	19.5	0.35	0.04	1.18	0.53

R-HUM= 50.% WET BULB= 59. GRAINS= 55.1 ENTHALPY= 25.4 COOLING WATER= 65.5

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	70.4	31.	21.7	0.86	0.55	0.62	0.36
135.0	70.4	31.	21.7	0.86	0.44	0.62	0.35
140.0	70.4	31.	21.7	0.86	0.35	0.62	0.35

R-HUM= 75.% WET BULB= 65. GRAINS= 82.7 ENTHALPY= 29.7 COOLING WATER= 71.7

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	83.4	45.	27.1	0.25	1.21	0.30	0.21
135.0	77.8	36.	24.6	0.74	1.23	0.49	0.30
140.0	75.1	35.	23.5	1.24	1.04	0.55	0.32
145.0	75.1	35.	23.5	1.24	0.81	0.55	0.32
150.0	75.1	35.	23.5	1.24	0.67	0.55	0.32

R-HUM=100.% WET BULB= 70. GRAINS=110.3 ENTHALPY= 34.0 COOLING WATER= 77.0

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	98.6	71.	34.9	0.00	1.24	-0.10	-0.11
135.0	95.5	65.	33.2	0.05	1.22	0.03	0.07
140.0	91.6	56.	31.1	0.23	1.22	0.25	0.19
145.0	87.2	50.	28.8	0.49	1.21	0.38	0.24
150.0	81.3	42.	26.1	1.13	1.23	0.49	0.29
155.0	80.4	41.	25.7	1.24	1.04	0.50	0.29
160.0	80.4	41.	25.7	1.24	0.88	0.50	0.29
165.0	80.4	41.	25.7	1.24	0.75	0.50	0.29

TABLE III-2 1/4

DEHUMIDIFIER OPERATION AS A FUNCTION OF AMBIENT TEMPERATURE, HUMIDITY, SOLAR WATER TEMPERATURE AND COOLING WATER TEMPERATURE

OUTSIDE TEMPERATURE = 92.

R-HUM= 25.% WET BULB= 67. GRAINS= 58.0 ENTHALPY= 31.2 COOLING WATER= 73.5

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	77.3	37.	24.4	1.15	0.42	1.33	0.55
135.0	77.3	37.	24.4	1.15	0.34	1.33	0.55
140.0	77.3	37.	24.4	1.15	0.28	1.33	0.55

R-HUM= 50.% WET BULB= 77. GRAINS=116.1 ENTHALPY= 40.4 COOLING WATER= 84.1

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	101.4	76.	35.6	0.16	1.21	0.44	0.29
135.0	99.0	70.	34.6	0.32	1.24	0.57	0.34
140.0	94.6	63.	32.7	0.51	1.22	0.65	0.36
145.0	90.4	55.	30.4	0.84	1.23	0.72	0.38
150.0	85.8	50.	28.7	1.24	1.15	0.75	0.39
155.0	86.8	50.	28.7	1.24	0.95	0.76	0.39
160.0	86.8	50.	28.7	1.24	0.80	0.76	0.39

R-HUM= 75.% WET BULB= 85. GRAINS=174.1 ENTHALPY= 49.5 COOLING WATER= 92.3

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	119.4	136.	50.3	0.00	1.16	-0.10	-0.11
135.0	117.6	129.	46.7	0.02	1.18	0.08	0.07
140.0	115.5	121.	45.9	0.07	1.20	0.23	0.18
145.0	113.4	113.	45.1	0.15	1.19	0.34	0.24
150.0	111.1	105.	43.3	0.25	1.18	0.42	0.28
155.0	108.2	96.	41.1	0.38	1.20	0.50	0.30
160.0	105.1	87.	39.0	0.54	1.20	0.55	0.32
165.0	101.7	78.	35.8	0.74	1.20	0.59	0.33
170.0	98.2	70.	34.7	1.03	1.19	0.62	0.33
175.0	96.7	67.	33.8	1.17	1.09	0.62	0.34
180.0	96.7	67.	33.8	1.17	0.97	0.62	0.34
185.0	96.7	67.	33.8	1.17	0.87	0.52	0.34

R-HUM=100.% WET BULB= 92. GRAINS=232.2 ENTHALPY= 58.6 COOLING WATER= 99.0

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	130.7	194.	62.3	0.00	1.17	-0.46	-0.83
135.0	129.4	186.	60.8	0.00	1.19	-0.23	-0.30
140.0	128.0	178.	59.2	0.01	1.18	-0.05	-0.06
145.0	126.5	170.	57.5	0.02	1.19	0.09	0.03
150.0	124.6	161.	55.7	0.06	1.18	0.21	0.17
155.0	122.9	152.	53.7	0.13	1.19	0.30	0.22
160.0	121.0	143.	51.7	0.22	1.20	0.38	0.25
165.0	118.8	133.	49.7	0.32	1.20	0.43	0.28
170.0	115.4	124.	47.6	0.43	1.20	0.48	0.30
175.0	113.9	114.	45.5	0.55	1.19	0.51	0.30
180.0	111.2	105.	43.4	0.70	1.19	0.54	0.31
185.0	109.2	96.	41.2	0.83	1.20	0.55	0.31
190.0	105.2	87.	39.0	1.10	1.19	0.57	0.31
195.0	104.2	85.	38.4	1.19	1.11	0.57	0.31
200.0	104.2	85.	38.4	1.19	1.02	0.57	0.31
205.0	104.2	85.	38.4	1.19	0.94	0.57	0.31

TABLE III-2 2/4

OUTSIDE TEMPERATURE = 80.

R-HUM= 25.% WET BULB= 59. GRAINS= 38.9 ENTHALPY= 25.3 COOLING WATER= 65.5

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	69.3	30.	21.3	1.10	0.16	1.46	0.58
135.0	69.3	30.	21.3	1.10	0.13	1.46	0.59
140.0	69.3	30.	21.3	1.10	0.11	1.45	0.58

R-HUM= 50.% WET BULB= 67. GRAINS= 77.8 ENTHALPY= 31.4 COOLING WATER= 74.0

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	79.7	40.	25.4	0.73	1.21	0.68	0.37
135.0	77.4	37.	24.4	1.21	0.79	0.73	0.38
140.0	77.4	37.	24.4	1.21	0.75	0.73	0.39
145.0	77.4	37.	24.4	1.21	0.51	0.73	0.39

R-HUM= 75.% WET BULB= 74. GRAINS=116.7 ENTHALPY= 37.5 COOLING WATER= 81.0

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	101.6	78.	35.7	0.03	1.23	0.09	0.08
135.0	98.6	71.	34.9	0.12	1.21	0.25	0.19
140.0	94.8	63.	32.7	0.31	1.23	0.40	0.26
145.0	91.2	57.	30.9	0.52	1.20	0.49	0.30
150.0	85.2	49.	28.3	0.98	1.21	0.58	0.33
155.0	84.6	46.	27.6	1.21	1.05	0.60	0.33
160.0	84.6	46.	27.6	1.21	0.88	0.60	0.33
165.0	84.6	46.	27.6	1.21	0.76	0.60	0.33

R-HUM=100.% WET BULB= 80. GRAINS=155.6 ENTHALPY= 43.6 COOLING WATER= 87.0

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	114.7	117.	45.2	0.00	1.18	-0.31	-0.44
135.0	112.7	110.	44.6	0.00	1.18	-0.10	-0.11
140.0	110.5	103.	42.8	0.02	1.18	0.07	0.06
145.0	107.9	95.	40.9	0.10	1.19	0.20	0.16
150.0	104.8	86.	38.8	0.24	1.20	0.32	0.22
155.0	101.6	78.	36.7	0.39	1.19	0.40	0.26
160.0	97.8	70.	34.5	0.60	1.19	0.47	0.28
165.0	93.7	61.	32.2	0.92	1.19	0.52	0.30
170.0	91.2	57.	30.8	1.19	1.12	0.54	0.30
175.0	91.2	57.	30.8	1.19	0.99	0.54	0.30
180.0	91.2	57.	30.8	1.19	0.98	0.54	0.30

TABLE III-2 3/4

OUTSIDE TEMPERATURE = 92.

R-HUM= 50.% WET BULB= 77. GRAINS=116.1 ENTHALPY= 40.4 COOLING WATER= 55.0

SOLAR	DRY BULB	GRAINS	ENTHALPY	CK-D	CK-C	COP-1	COP-2
130.0	101.4	78.	35.6	0.05	1.21	0.44	0.29
135.0	98.0	70.	34.5	0.11	1.24	0.57	0.34
140.0	94.6	63.	32.7	0.19	1.22	0.65	0.35
145.0	90.4	55.	30.4	0.29	1.23	0.72	0.36
150.0	85.3	47.	27.9	0.41	1.23	0.78	0.39
155.0	79.4	40.	25.3	0.58	1.23	0.82	0.40
160.0	72.9	33.	22.6	0.84	1.21	0.85	0.40
165.0	66.7	26.	20.1	1.24	1.19	0.87	0.40
170.0	65.7	26.	20.1	1.24	1.04	0.87	0.40
175.0	65.7	26.	20.1	1.24	0.91	0.87	0.40

TABLE III-2 4/4

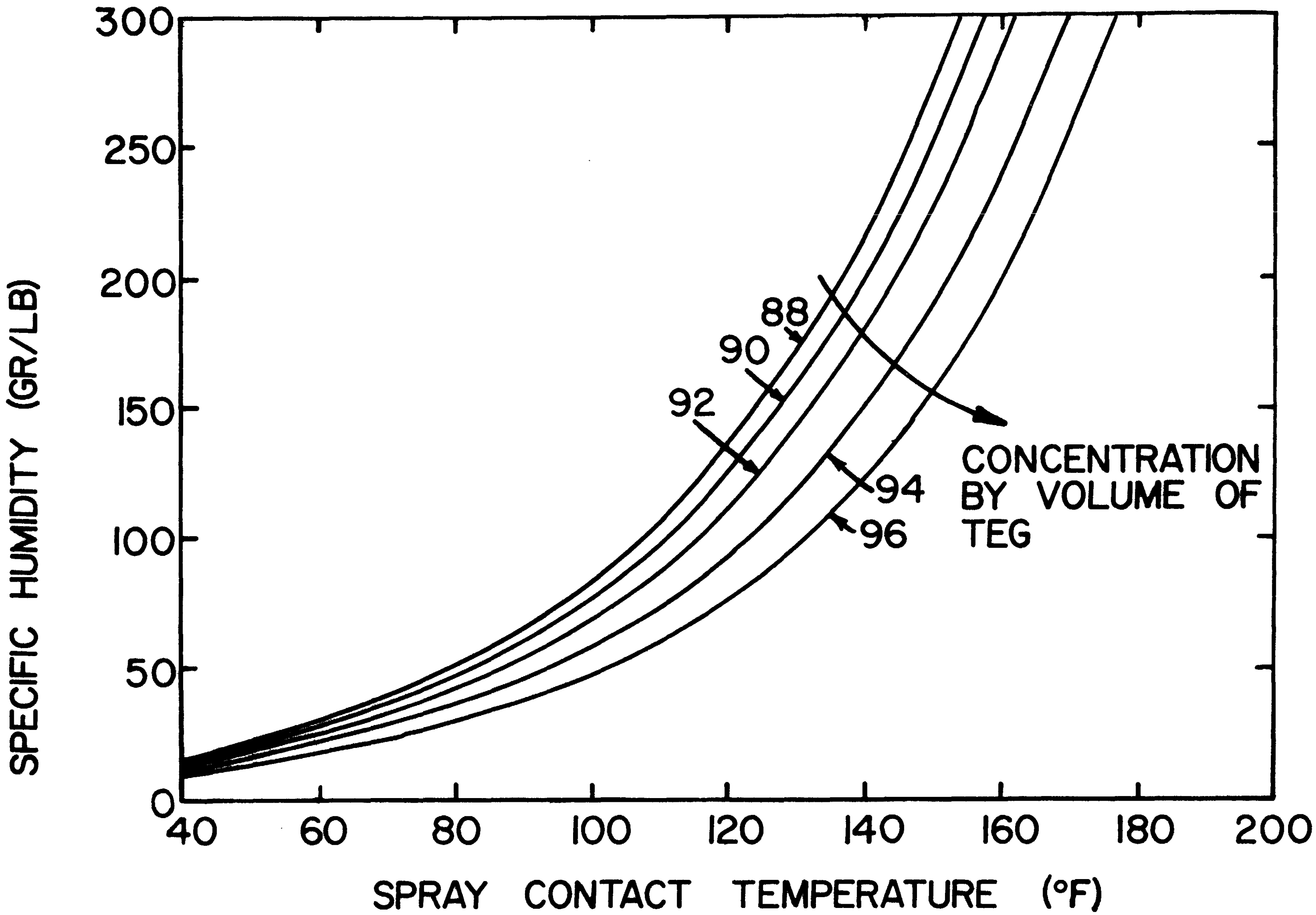


FIGURE III-1 SPECIFIC HUMIDITY OF AIR LEAVING DEHUMIDIFIER AS A FUNCTION OF SPRAY CONTACT TEMPERATURE AND CONCENTRATION BY VOLUME OF TEG

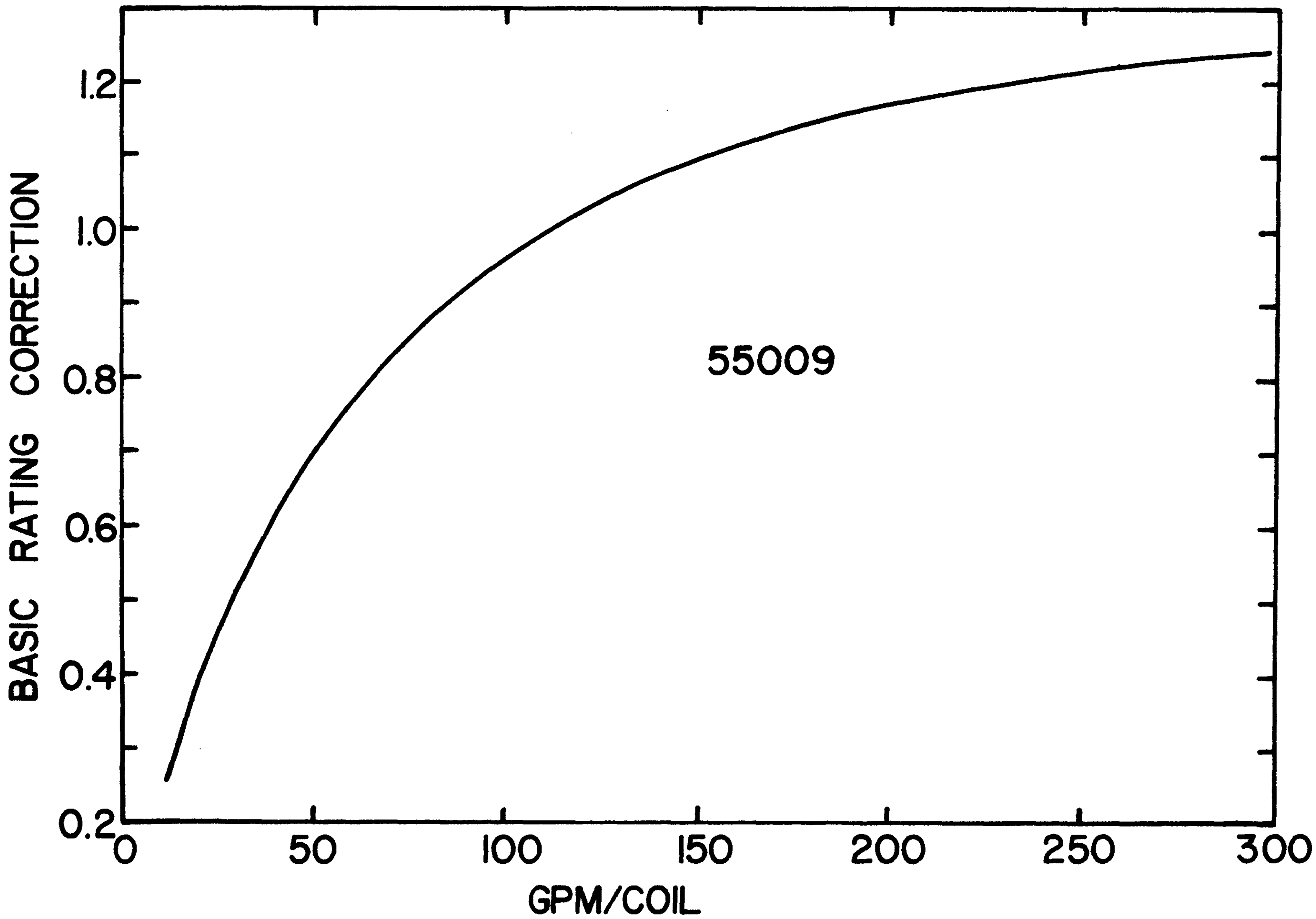


FIGURE III-2 COIL CAPACITY CURVE FOR TYPICAL DEHUMIDIFIER HEAT EXCHANGER

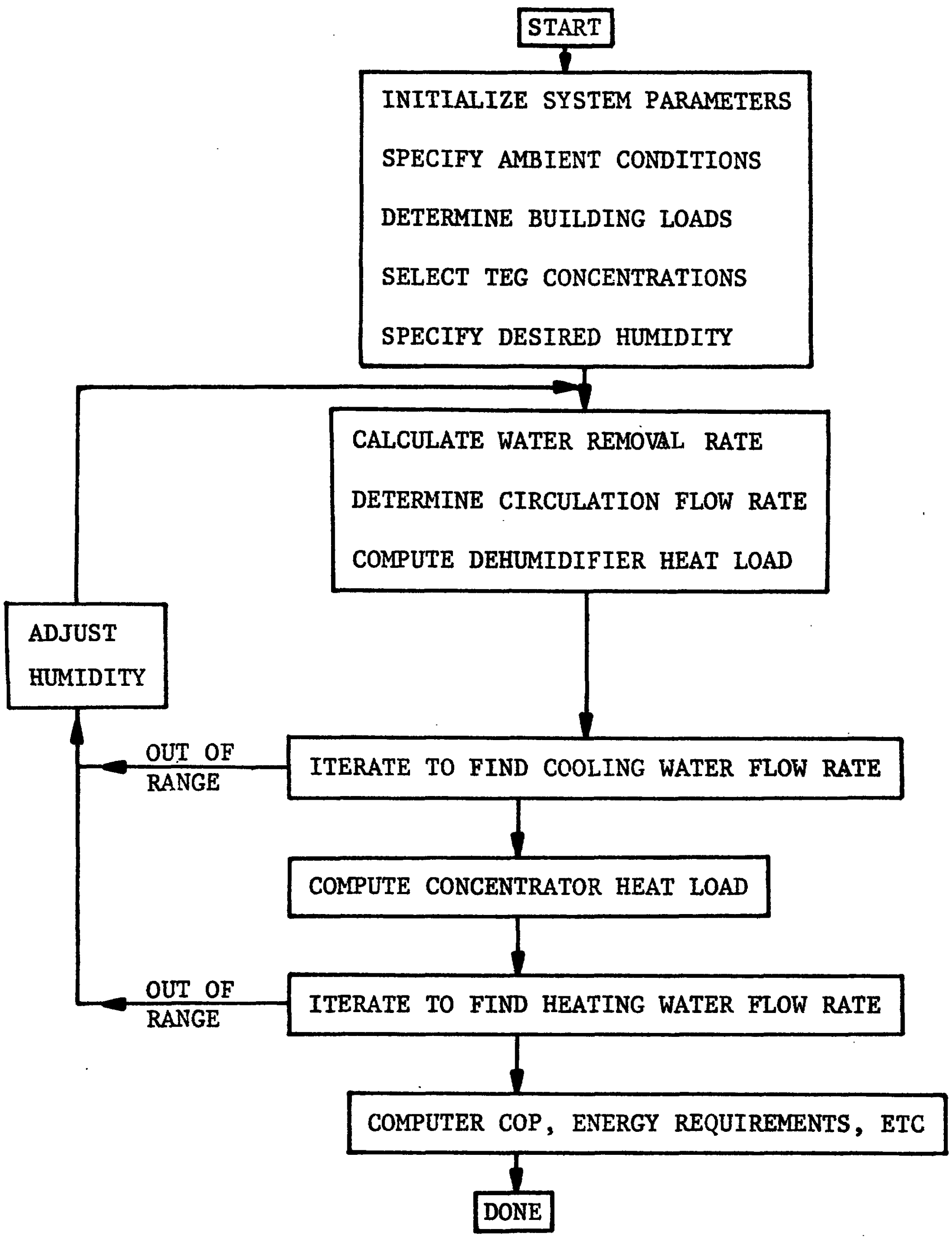


Figure III-3: FLOW CHART FOR DETERMINING DEHUMIDIFIER/CONCENTRATOR PERFORMANCE

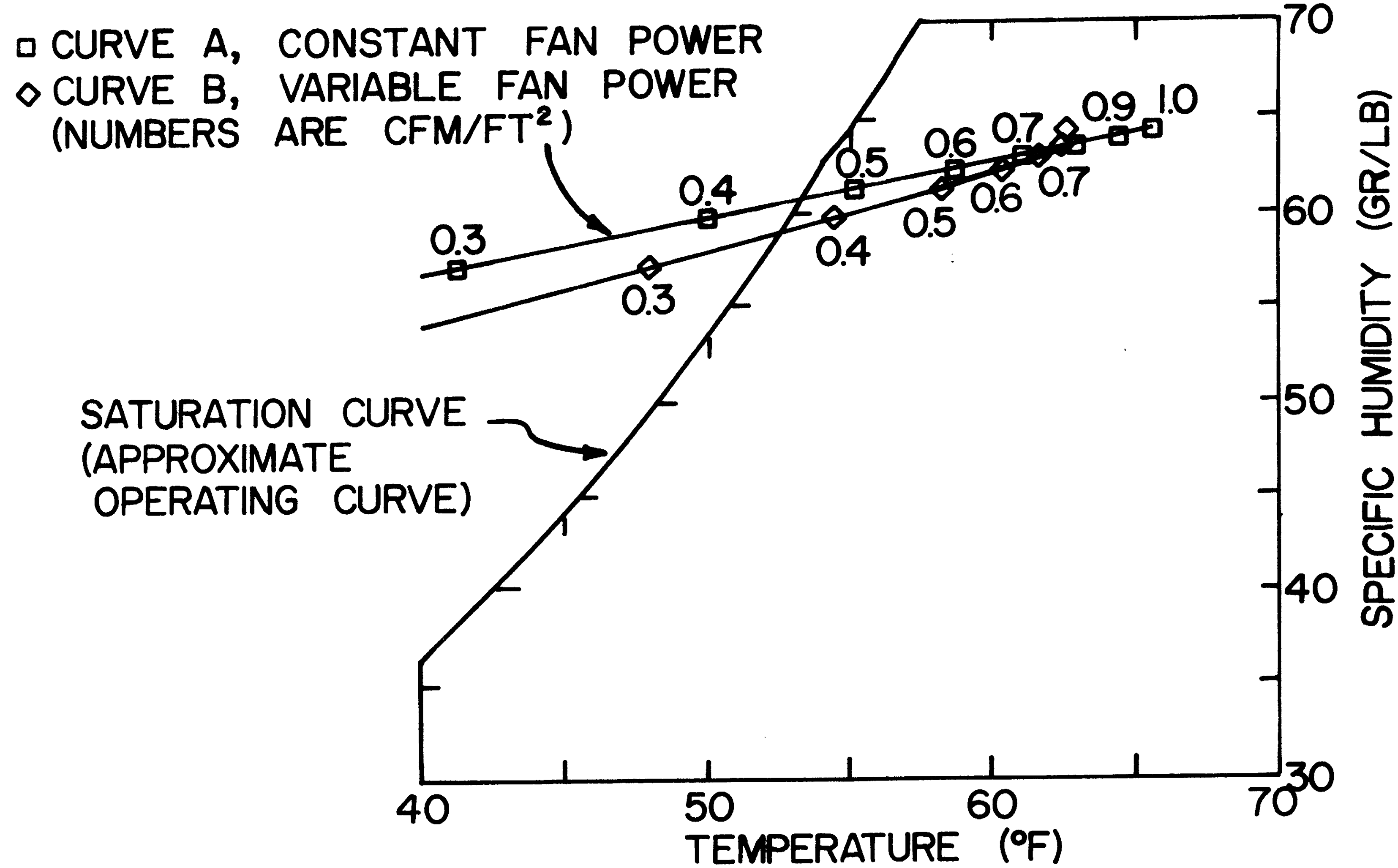


FIGURE III-4 SKELETON PSYCHROMETRIC CHART SHOWING AIR FLOW RATE REQUIRED TO BALANCE HEAT AND MOISTURE LOSS

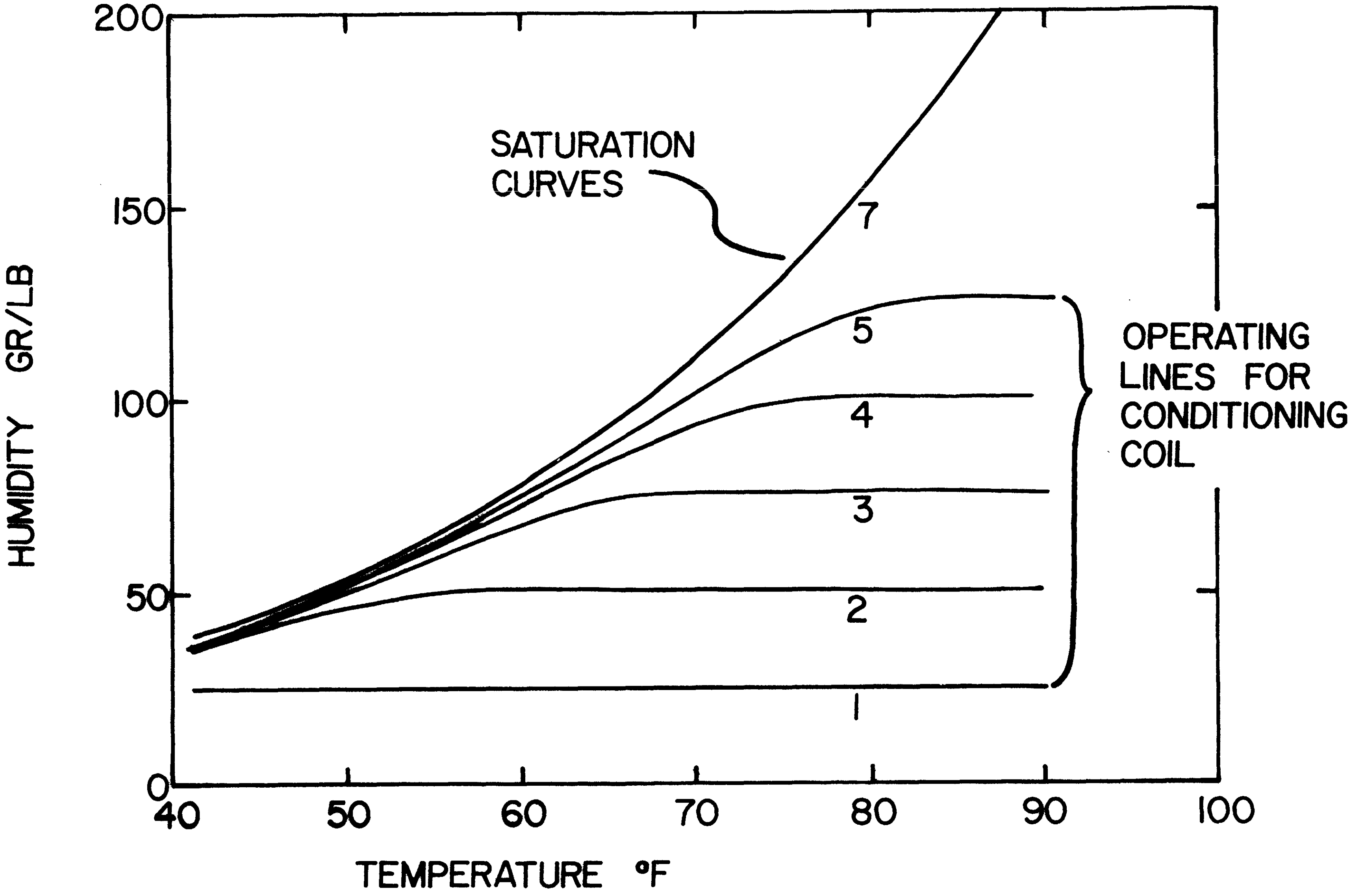


FIGURE III-5 COEFFICIENT A FOR VAPOR PRESSURE EQUILIBRIUM CURVES AS A FUNCTION OF CONCENTRATION BY VOLUME OF TEG

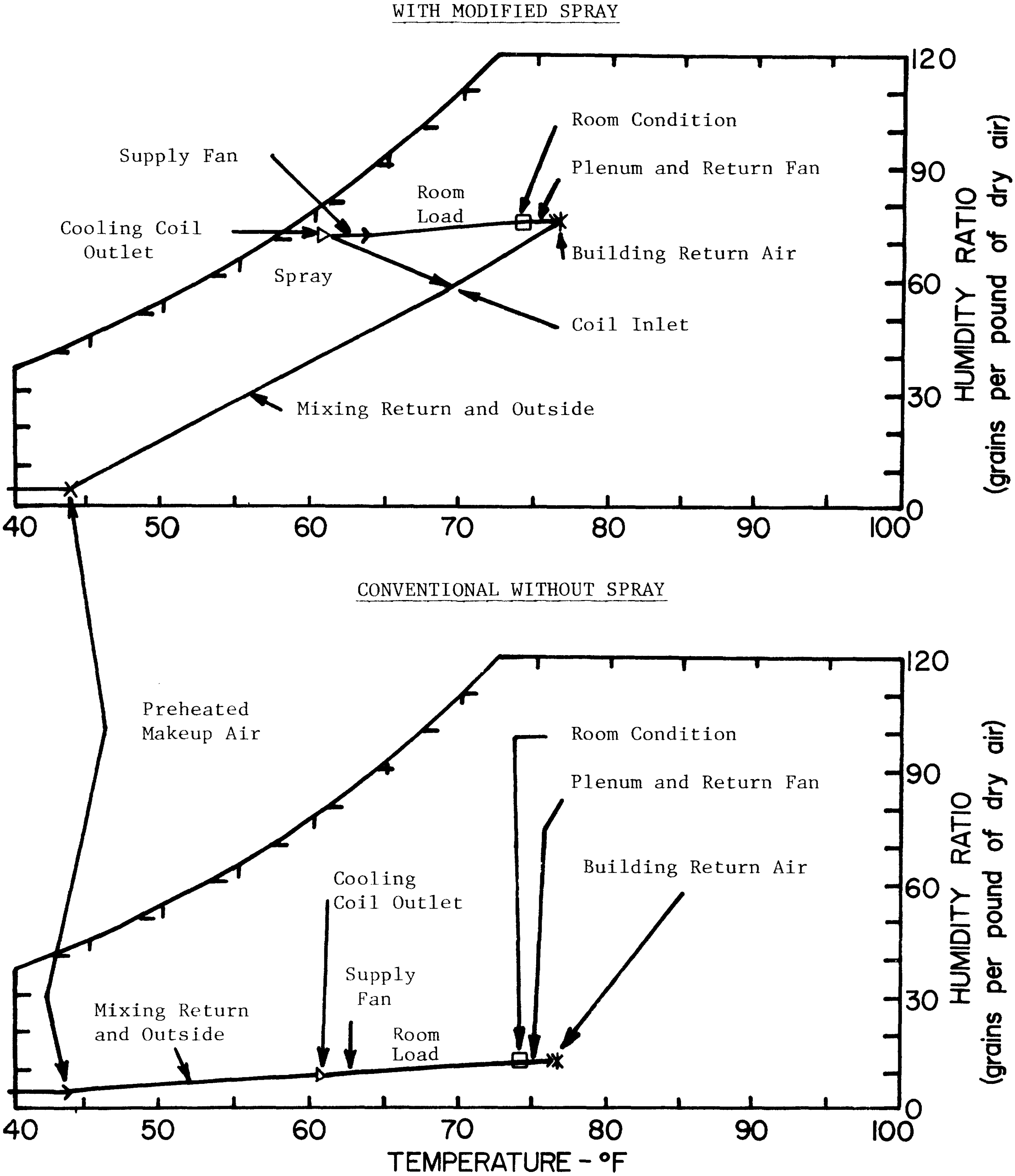


FIGURE III-6 USE OF MODIFIED SPRAY TO CONTROL HUMIDITY DURING HOT WEATHER

1968 TYPICAL YEAR
January 2, 10:00 am.m.
Outside Air: T = 12 °F, 9 grains

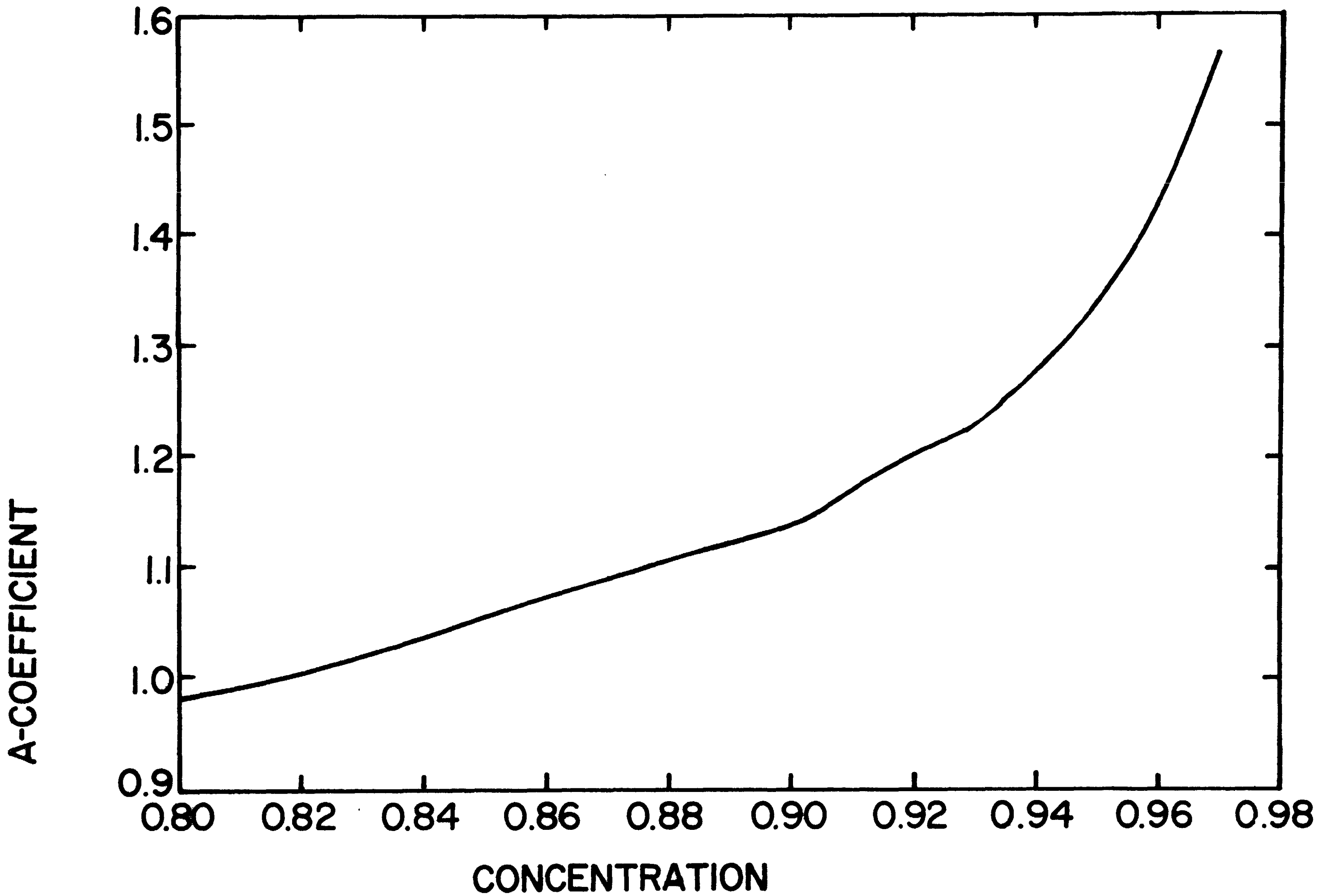


FIGURE III-7 Coefficient "A" for TEG Vapor Equilibrium Model

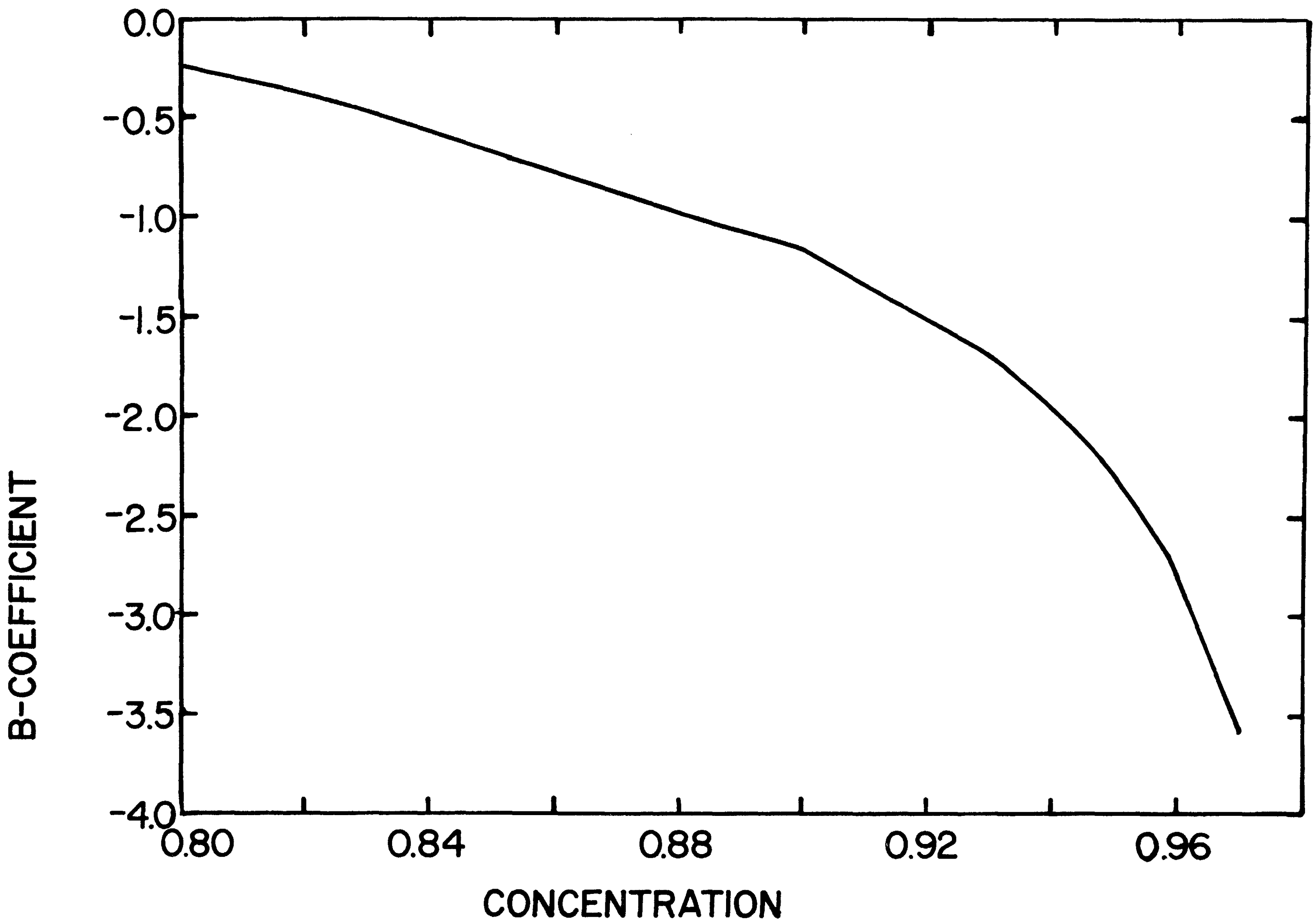


Figure III-8 Coefficient "B" for TEG Vapor Equilibrium Model

APPENDIX IV

THE SOLAR COLLECTOR PRACTICAL DESIGN CONSIDERATIONS AND DRAFT REQUEST FOR PROPOSAL

I. Factors Affecting the Configuration and Installation of the Solar Collector

Early design studies of the office tower of Citicorp Center explored the possibility of terraced dwelling units on top which were to be glass enclosed for protection from the elements. This concept was abandoned leaving the building crown with a 45 degree slope facing West. The sloping top suggested the possibility of a solar collector installation. Citicorp had its architects and engineers rotate the crown by 90 degrees so that the slope would face South. Because the tower is aligned with the city streets, the slope is actually oriented 29 degrees west of South.

The basic structural steel design was to support aluminum cladding similar to that used to clad the remainder of the building. The sloping top does not have to be weather tight; in fact, codes call for it to be ventilated if the structural steel is not to be fireproofed. Specifically, at least 3 inch gaps are required every 6 feet 6 inches along the vertical slope. There is substantial cost advantage in not requiring the solar collector to serve as the weather roof in addition to the mechanical convenience during the installation of panels and should they have to be replaced. The main support of the solar collectors consists of I-beam purlins 9'-3" on centers running down the sloping crown of the building for a distance of 174'-8". The I-beams used are 12 inches deep and conform to the 45 degree slope of the crown. There is space for 16 columns of collector panels. The top flange of the I-beams is left exposed to form a track for a crown washing rig that might also be used to access the front side of the installed solar panels. The webs of the purlins require bracing at points not less than 5 inches, nor higher than 7 inches above the bottom of the purlin. Bracing points must not be further apart than 16'-3" as measured along the slope. Framing of the collector modules or structures provided for their installation could serve as braces as long as the structural design meets the requirements of the New York City Building Code. The code requires the following load sustaining capability:

Live Load: 5 psf horizontal projection, acting downward

Wind Load: 24 psf pressure, acting normal to the surface or 16 psf suction.

A number of design studies were carried out to select a collector configuration that would meet the criteria described above while minimizing installation costs. Configurations of standard collector panels are shown in Figs. IV-1 and 2. There are installation cost advantages in allowing for installation from the inside and reducing handling weight of individual collector panels to the vicinity of 150 pounds. This should allow two persons working on interior scaffolding to install the panels. One attractive suggested de-

CLUSTER CONFIGURATIONS OF
STANDARD PANELS

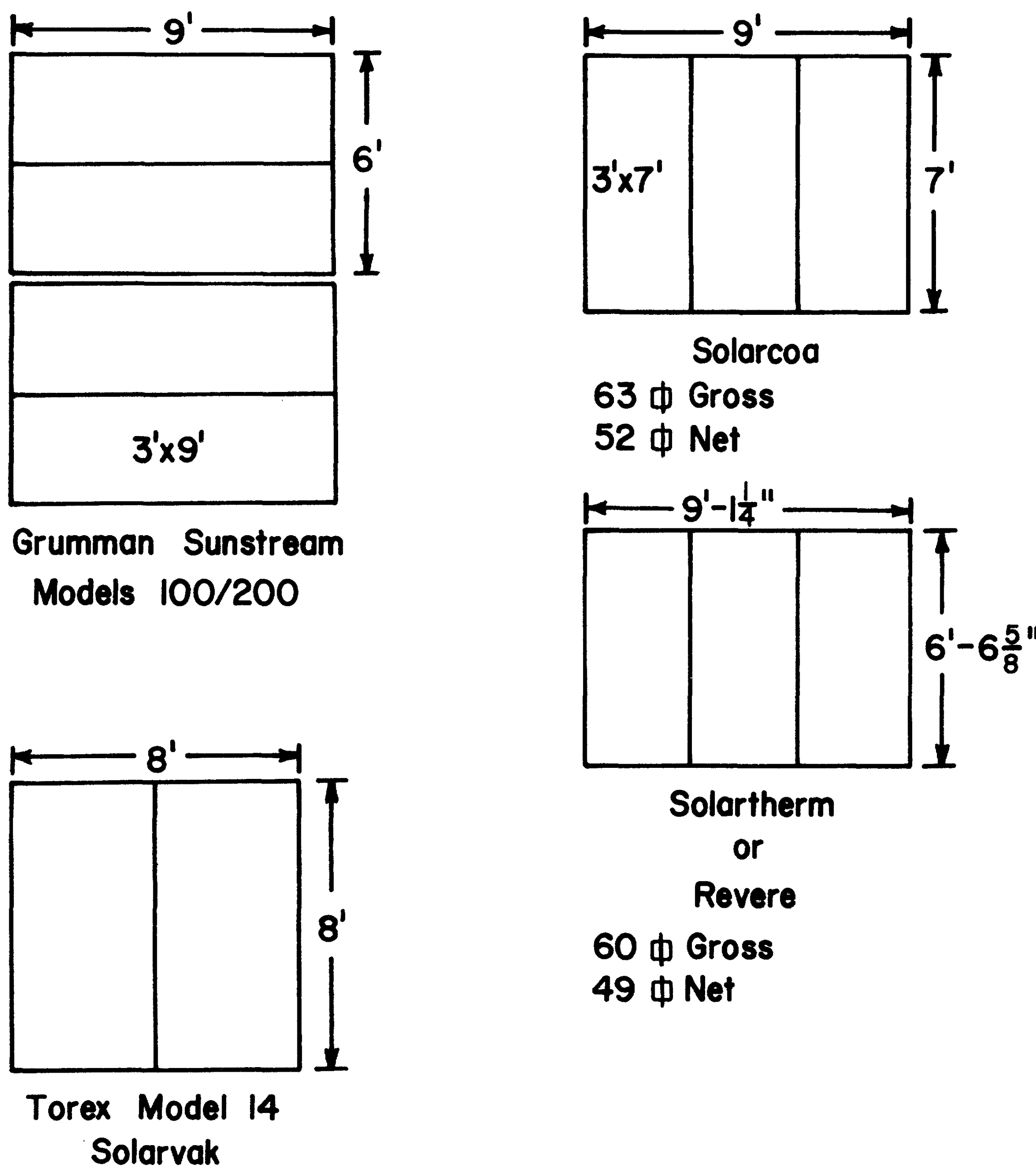


FIGURE IV-1

CLUSTER CONFIGURATION
OF STANDARD PANELS

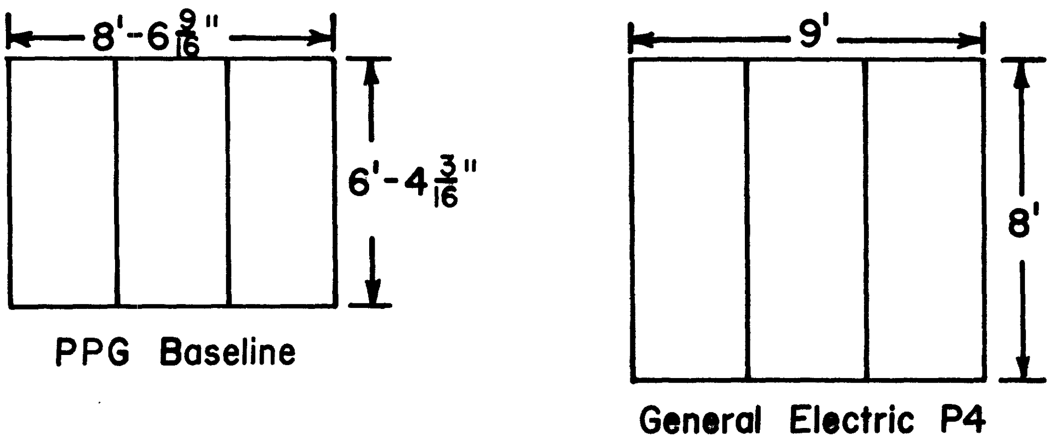


FIGURE IV-2

sign involved the use of "S" beam laterals which serve the dual function of bracing the purlins and supporting structurally the midline of a pair of 3' x 9' collector panels, Fig.IV-3. This approach reduces the angle of tilt of the collectors from 45 degrees to about 36 degrees which improves summertime performance but not sufficiently in aggregate to justify changing the angle at increased cost. The 3' x 9' panels can also be installed with cleats fastened to the web of the purlin and with the back edge of the panel resting on the lower flange. The purlins will have a white finish which subtantially eliminates the shadowing effect of the recessed mounting in both approaches.

Spacing of the collector panels must also be adjusted to drain the water collected on them during heavy rain (2 inches/hr.) In particular, water buildup and cascading past the gaps down the length of the collector must be avoided to ensure more even distribution of water over the main roof below. To avoid overloading storm sewers, city codes require that building roofs retain 3 inches water depth before discharging.

In all these considerations mounting configurations were explored which would not require additional structural steel or substantial changes in the existing design for the crown. The requirement that the collector assemblies be structurally self supporting and in addition supply the necessary purlin bracing would in our estimation accomplish this. In retrospect it is clear that this caused some problems for potential collector suppliers in that it represented a departure from the standard collector module of some of them.

II. Factors Affecting the Performance of the Solar Collector

The 55th floor of the tower is the uppermost office space. The 56th floor houses two Hygrol conditioner units in the mechanical equipment rooms and has around its perimeter a track area for a window washing rig. On the 57th floor are located the two concentrator units, two 10,000 gallon thermal storage tanks, a 500 gallon hot water tank and an emergency generator room and the return air chamber. A small part of the weatherproof roof is at this level. The 58th floor houses transformers, network protectors and elevator machinery. The slope of the crown begins at about the 59th floor level on which the building's tuned mass damper is located. The main roof tops the 59th floor. A 4-bay cooling tower is installed above the main roof. The cooling tower exhausts at a safing floor level at about the midpoint of the slope of the crown through louvers in the top and in the north face of the crown.

The potential effects of the cooling tower system on solar collector performance and reliability were explored. We specify for example that the enclosure of the solar panels resist corrosion when operated in the vicinity of a forced draft evaporative cooling tower. The panels in the upper half of the collector will be most severely affected by the cooling tower exhaust. The potential obscuration of the collector by cooling tower fog was also considered. Fogging usually occurs when ambient temperatures are low. Fog density is greatest during periods of high humidity. Fog dissipation time is longest when the wind is light and air is subsiding or

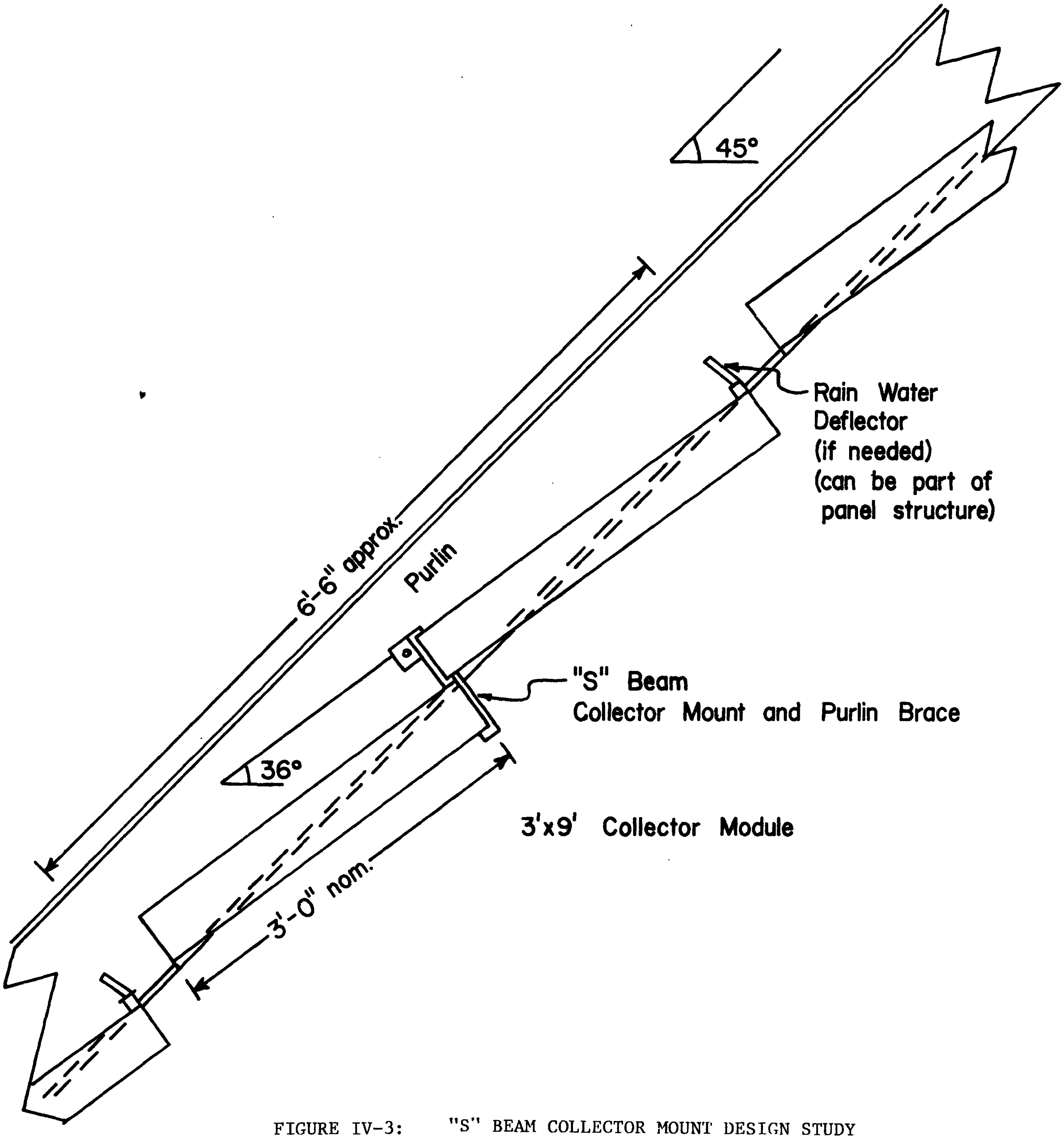


FIGURE IV-3: "S" BEAM COLLECTOR MOUNT DESIGN STUDY during periods of light wind and cloud cover. Fortunately prevailing summer winds are from the southwest which would tend to blow the fog away from the collector, and often solar heating of the ambient air dissipates the fog soon after sunrise. Fog under overcast conditions can have only a small effect on the solar energy collected even if the fog covered a large fraction of the collector. The probability remains that the cooling tower exhaust will pass through the upper half of the collector in spaces left for ventilation. A wall may be installed along the south side of the cooling tower fan opening to deflect the exhaust We would have preferred ducted exhaust but this proved potentially expensive.

Cooling tower water is treated for corrosion and scale control with Liquid Vaporene 72 in the ratio of one gallon of Vaporene per 1000 gallons of water. We were concerned about the possibility of a Vaporene precipitate forming on the coverplates of the collector which might reduce its efficiency. A worst case test was performed by brushing a coating of concentrated Vaporene on one-half of a piece of cover glass. When the coating had dried, a direct comparison of the solar transmission of the coated half with the uncoated half indicated a reduction of only about 5%. It is expected that in practice, the precipitate will never get as dense as in our test and normal rainfall will assist in keeping the collector covers clean so that attenuation from this source will be negligible.

III. Solar Collector Procurement

The procurement and installation of the quarter-acre solar collector contained the major construction uncertainties and the greatest questions of final in-place costs. Factors affecting these uncertainties were:

1. The basic design of the building was complete, structural steel ordered, the general contractor and principal subcontractors identified and working, and construction was underway.

2. The collector delivery and installation schedule had to be closely coordinated with the building construction to realize the cost savings possible through the utilization of installed hoisting and rigging equipment. The solar system should not incur costs caused by a delay in the building construction schedule.

3. The collector was to be installed at the top of a 900 foot high building.

4. There was some uncertainty regarding which trades group or combination would claim the complete collector installation.

5. Certain constraints had to be placed on the size and weight of collector modules to facilitate manhandling during installation and reduce labor costs.

7. We required delivery of nearly 1000 modules on a tight schedule having expressed, or implied by company reputation, some reasonable warranty.

6. Premium construction costs prevail in New York City.

8. The collector modules were to be designed and constructed in a way to minimize the need for additional structural steel over that for a curtain wall the collector would replace.

In addition to the problems raised by the special nature of this project, we were aware of a number of problems being experienced in solar installations currently operational. Among these problems were:

1. Corrosion of heat exhcange fluid passages. This problem appears most severe for aluminum passages. Inhibitors have been used in water-ethylene glycol mixtures with some success, but we were concerned about the problem of the longer term maintenance of a properly inhibited heat transfer fluid by regular building operating and maintenance personnel.

2. System leaks. A solar collector has twice the number of interconnections as there are panels. The connections must have some flexibility to allow for thermal expansion and should allow for replacement of individual panels in case of damage and/or deterioration. Some collectors use neoprene rubber or plastic for interconnections. Often the interconnectors severely limit the pressure that the collector system can withstand.

3. Collector damage on interruption of flow or loss of heat transfer fluid. If the heat transfer fluid is missing or is not circulating, the collector, depending on the details of its design, can reach rather high temperatures. Steam has been generated in non-circulating fluids and the resulting pressure has ruptured the collector. Equilibrium temperatures reached by collectors under normal insolation and without heat transfer fluid have been high enough to damage some types of insulation used and expansion has deformed some collector cases.

4. Condensation between cover plates. If not sealed or properly vented, water can condense between the cover plates and result in poor performance.

IV. Draft Request for Proposal

A draft request for proposal was prepared in which we attempted to be specific enough to avoid the problems foreseen above yet allow the collector supplier to make use of any innovative concepts his particular design might offer.

In recognition of the problems the above factors might raise, we initiated a survey of potential collector suppliers at the beginning of the study and held informal discussions with many of them as the final design of the system was evolving. From the survey we were able to identify twenty potential suppliers from whom we would request a budgetary estimate based on a draft specification. We also asked for comments on the draft specification in an effort to ensure that we did not foreclose any option unintentionally.

The draft request for proposal and the list of potential suppliers to whom it was sent is contained in Attachments I and II.

A number of factors affected our choice of collector suppliers to be solicited. We knew that in such a fledgling industry customary warranties would be difficult to obtain. We believed, however, that the larger, well established firms having experience in this, or a closely related field, would support their product because of its large public exposure when installed and operating in Citicorp Center. Yet, we did not want to rule out smaller and newer firms who because they had a single product line, had something special to offer that would be worth some risk.

We obtained numerous useful comments on our draft RFP. A frequent recommendation was that we specify performance rather than approach. It is better, for example, to require that the collector module withstand a full day of maximum solar radiation expected for New York City without circulating heat exchange fluid, than to specify that the module components be able to withstand, say, 400°F. Different collector designs will reach different maximum temperatures under these conditions and those with lower equilibrium temperatures could employ safely, lower temperature materials. The specification was rewritten as follows: "The panel components shall withstand temperatures reached by the assembly during loss of heat transfer fluid or an interruption of circulation of the heat transfer fluid in the absorber plate under maximum insolation conditions for one day for New York City without damage or deterioration." "The insulation materials or any other materials used in the fabrication of the collector shall not outgas or give off vapors at temperatures reached if the heat transfer fluid flow is interrupted during a day of maximum insolation in New York City."

Other comments suggested that the specified 150 pound pressure was too high. The specification was rewritten to read, "Heat transfer tubing shall withstand a normal operating pressure of 75 psig." Our specification of the characteristics of a selective surface ruled out some promising surface treatments so the specification of absorptivity and emissivity will be deleted. The actual performance of the collector and its cost will make it possible to evaluate the supplier's tradeoff between surface treatment and the number of covers.

V. Budgetary Estimates

We recieved two budgetary estimates for supplying the collector modules and installing them.

Owens/Corning Fiberglas, Construction Services Division, the marketing outlet for the Corning evalucated tube collector, estimated a cost between \$1-million and \$1.175-million for the furnishing and installing of 845, 6-tube collector units, supply and return headers, thermal insulation, drain piping, supply and return valves, vent valves, balancing valves, and expansion tank. The installation would provide 10,647 effective square feet of collector. The evalcauted tube construction makes this an efficient unit for higher collection temperatures and our analyses showed that it was equivalent to 20,000 square feet of a good standard flat plate collector at collection temperatures of 220°F. At the lower collection temperatures needed for our project, the improved efficiency could not compensate for the smaller area. A decision was made later not to produce and market this collector.

International Environment Corporation estimated a cost of \$497,000 for furnishing and installing 752 Solar Therm II collecotrs (20,304 sq. ft.), supply and return headers and hangers for same, fittings, cooper tube connections to collectors, header and connector tube insulation, angle clips and fasteners to connect collector frames to purlins, and miscellaneous accessories. The estimate also included shop drawings, hoist expense, structural engineer supervision and New York City sales tax on materials. This collector is based on the company's extensive experience in the manufacture and installation of heat transfer panels for radiant heating and cooling systems over the past decade. The company has experience in New York City construction and in working with the general contractor for Citicorp Center.

H. Sand & Co., Inc., the mechanical subcontractor working on Citicorp Center, was asked to make a budgetary estimate of collector installation costs. The cost of receiving, unloading, hoisting and installing 864 solar panels was estimated at \$198,500.00. The cost of furnishing and installing piping, fittings, valves and strainers, panel shut off and balancing valves, pipe hangers and supports, anchors and guides, expansion tank, gauges, thermometers, circulating pumps and vibration bases, glycol including mixing tank and accessories, insulation and temperatures controls was estimated at \$572,000.

Other respondents gave estimates for supplying the collector panels alone:

PPG Industries, Inc. quoted a price of \$170 per collector panel, FOB factor, for the PPG Standard Copper Solar Collector. Each panel has a gross area of 18 sq. ft. Because of their size (34 3/16" x 76 3/16" x 1 5/16" not including back insulation) these panels would have to be mounted in clusters of three abreast to span the distance between the supporting beams.

Chamberlain Manufacturing Company estimated the cost of 953 panels (3' x 7') FOB job site, New York City, at \$277,630.00 for single glazed panels with a black chrome selective surface and \$247,240.00 for double glazed panels with a black paint collector surface. The estimate is for panels only, no hardware for assembling into a 7' x 9' array or installation. Job site supervision is not included.

Sunsource estimated the cost of 740 collectors (3' x 9') at $182,336.00. Soon after this estimate was received we were notified by the company that a new general manager had taken over at Sunsource.

Reynolds Metals Company quoted a basic price for their standard Torex 14 panel at $5.25/sq. ft. for quantities from 500 to 749. The panel could be made in a 4' x 9' size to fit the purlin spacing. The unit has a double Tedlar cover, and would not withstand 400°F, but only reachs about 180°F over ambient under no-flow conditions.

Honeywell reported that it had reached a licensing agreement with Lennox Industries where Lennox would manufacture flat plate solar collectors based on Honeywell's technology. Production was expected to begin in early 1976.

For various reasons, none of the other companies solicited responded with a budgetary estimate. Several of them were very helpful with comments on our draft RFP. A frequent reason given for not responding with a budgetary estimate was that production had not yet reached the point where they could respond to a job of this magnitude on the time schedule needed. Many of the firms' activities involved R & D and demonstration programs as total system efforts and hence the companies weren't particularly attracted to supplying the collector alone. Others were apprehensive about problems that might arise in the installation of this novel equipment on an office tower in New York City. In addition, rumors were circulated that the solar project had been cancelled well before that decision had been reached which caused some consternation and confusion for at least one potential supplier.

It is our judgment, based on this experience and informal conversations with people in the industry, that the manufacturing supply and marketing of solar collectors was too immature at that time to expect much competitive response to a task of this magnitude, and on a tight and relatively immediate time schedule. Demonstration programs and feasibility experiments appear to be far more attractive to those companies having research, development and engineering personnel needed to transform preproduction prototypes into mass produced, marketable units.

Appendix IV Attachment I

Budgetary Estimate Solicitation Package

Sources Solicited for Budgetary Estimate

General Electric Company
Space Division
Valley Forge Space Center
P.O. Box 8661
Philadelphia, PA 19101
Attn: W.F. Moore

Kalwall Corporation
1111 Candia Road
Manchester, N.H. 02103
Attn: Keith W. Harrison

PPG Industries, Inc.
One Gateway Center
Pittsburgh, PA 15222
Attn: R.G. Gallagher

Daystar Corporation
41 Second Ave.
Burlington, MA 01803
Attn: Gary Nelson

Owens/Corning Fiberglas
Construction Group
Fiberglas Tower
Toledo, Ohio 43659
Attn: Dan E. Morgenroth

International Environment Corp.
129 Halstead Ave.
Mamoroneck N.Y.
Attn: Richard D. Rothschild

Revere Copper and Brass, Inc.
Building Products Department
Rome, N.Y. 13440
Attn: William J. Heidrich

Libbey-Owens-Ford
1701 East Broadway
Toledo, Ohio 43605
Attn: Howard Swift

Grumman Aerospace Corp.
Bethpage, N.T. 11714
Attn: Gregory W. Knowles

A.S.G. Industries, Inc.
P.O. Box 929
Kingsport, Tenn. 37662

Reynolds Aluminum
Mill Products Division
Reynolds Metals Company
Richmond, VA 23261
Attn: C.H. Holtyn

Solarsystems, Inc.
1802 Dennis Drive
Tyler, Texas 75701

SOL-THERM CORP.
7 West 14th St.
New York, N.Y. 10011
Attn: Itamar Sittenfeld

Aluminum Company of America
ALCOA Building
Pittsburgh, PA 15219
Attn: W.M. Foster

SUNSOURCE
9606 Santa Monica Blvd.
Beverly Hills, CA 90210

Solar Corporation of America
100 Main Street
Warrenton, VA 22186
Attn: Anthony Easton

Honeywell Corporation
2700 Ridgeway Parkway
Minneapolis, Minn. 55413
Attn: James Ramsey

Owens-Illinois, Inc.
P.O. Box 1035
Toledo, OH 43666
Attn: Richard E. Ford

Solar Conversion Corp. of America
SOLARCOA
16260 Raymer St.
Van Nuys, CA 91406
Attn: Kenneth E. Parker

Chamberlain Manufacturing Corp.
845 Larch Avenue
Elmhurst, IL 60216
Attn: Norman A. Buckley

3-137

Energy Laboratory
Massachusetts Institute of Technology
Cambridge, Massachusetts 02139
(617) 253-3400

August 29, 1975

Gentlemen:

You are hereby invited to submit a budgetary estimate of the cost of supplying approximately 20,000 square feet of your solar collector modules for the Citicorp Center building, and at your option a budgetary estimate of the cost to install them on the building at Lexington Avenue and 53rd Street in New York City. The installation estimate should include the cost of piping the collector up to and including the main collector supply and return headers. Estimates of the cost of only part of the task (furnish collectors, installation, or piping) are acceptable if you do not wish to be considered for the whole job.

Enclosed for your information is a draft copy of major portions of a formal request for proposal which can serve as a guide to the technical and special requirements of the project, to permit you to make a more accurate budgetary estimate. The inclusion of this material does not imply a request for a formal proposal at this time. Your comments on the draft are solicited.

Enclosed also are sketches of several collector installation concepts. Your appraoch need not necessarily follow any of these suggestions as long as the technical conditions of the draft RFP are met.

Please let us know by September 10, 1975, if you intend to submit a budgetary estimate. A telephone call to us at (617) 253-3403 followed by a confirming letter is adequate. We would like to recieve the budgetary estimates no later than the close of the working day on September 22, 1975.

Very truly yours,

/s/ James W. Meyer

James W. Meyer
Project Manager

Enclosures: Attachment I; Attachment II

Attachment I

Your Budgetary Estimate shall include all labor, materials, equipment and services required by the following listed Documents:

A. Specifications

B. General Conditions - A.I.A. Document A-201, dated April, 1970.

Edition and supplementary General Conditions, dated March 19, 1974.

C. Rider No. 1-A.

D. Special Requirements- - Solar Collector Panels dated July 18, 1975.

AI-2

Draft Collector RFQ

The technical specifications have been drafted and circulated for comment.

DRAFT

Request for Quotation/Proposal

Solar Collector for Citicorp Center

Technical Conditions

Materials:

The collector panels shall have an enclosure ensuring the mechanical integrity of the unit under normal building construction methods. The enclosure shall resist corrosion when operated in the vicinity of a forced draft evaporative cooling tower. The panels shall have a frame of sufficient strength to support the panel over a 9'-0" clear span when supported by clips at the ends of the panel.

The weight of the panel shall not exceed 15#/ft^2 including the fill with the heat transfer fluid. The handling weight of the entire panel as rigged for installation shall not exceed 150#.

Structural Requirements:

a. Cover frames, casings, glass covers, and all other parts shall be designed to sustain dead, live, and wind loads in accordance with New York City Building Code. (For the bidder's convenience, NYC Code requires the following:

   Live Load: 5 psf horizontal projection, acting downward

   Wind Load: 24 psf pressure, acting normal to surface or 16 psf suction.)

b. Provide connections to structural steel to sustain loads in accordance with NYC Building Code. In addition, provide connections to structural steel purlins to brace these purlins as follows:
   1. The webs of the purlins shall be braced at points no less than 5 inches above the bottom of purlin, nor higher than 7 inches above the bottom of purlin.

   2. Bracing points shall be no further apart than 16'-3", measured on the slope.

3. Bracing may be accomplished by the collector modules as mounted or by combined bracing/collector mounting members meeting the requirements above.

c. All structural design shall be under the direct supervision of a professional engineer registered in the State of New York and employed by the Contractor. Submit calculations for review by the Architect.

d. All fabrication and installation shall conform with the New York City Building Code.

The absorber plate may have either one or two transparent covers. The outside cover shall be of tempered glass or approved equivalent of a thickness required for the panel size to support the snow and wind loads specified above when the panels are mounted at an angle of 45° from horizontal. The interior cover shall withstand temperatures up to 400°F on the collector plate (which may be reached during loss of heat transfer fluid or an interruption of circulation of the heat transfer fluid in the absorber plate) without damage or deterioration. The collector plate backing is required to withstand temperatures up to 400°F. If the collector plate proposed is treated with a selective surface, only one cover plate will be required; (if the selective surface has an absorptivity to solar energy of 0.9 or larger and an emissivity in the infrared of 0.2 or less). The anticipated lifetime of the absorber surface treatment shall be specified and the bases for the projection explained.

The circulating heat transfer medium will be 30% ethylene glycol-water mixture with inhibitors added. All parts of the collector in contact with the heat transfer medium shall be copper or an approved equivalently corrosion resistant material. Heat transfer tubing shall withstand a normal operating pressure of 150 psig. The nominal flow rate of heat transfer fluid through the panels will be 0.017 gpm per square foot with a pressure drop through the unit greater than 0.5 ft. of water but not exceeding 2 ft. of water. The supply and return connections shall extend through the rear of the panel.

The panel insulation shall be completely enclosed in the casing. A "U" factor of 0.1 or less is required of the insulation. Insulation thickness will be determined by the type of insulation proposed. The insulation materials or any other materials used in the fabrication of the collector shall not outgas or give off vapors at temperatures up to 400°F if these materials reach this temperature during a 10 hour day of maximum insolation and with no flow of heat transfer fluid.

Dimensions:

The basic support of the solar collector consists of I beams 9'-3' on centers running down the sloping crown of the building for a distance of 174'-8". The I beams are 12 inches. There is space for 16 columns of collector panels. The slope of the structural members is 45° from horizontal. For detailed information refer to architectural, structural, and mechanical drawings.

Collector panel width shall be sized such that total individual collector panel weight does not exceed 150# configured for installation between the I beams. Codes require a gap of 3" between panels every 6'-6" along the slope of the collector. Within the above constraints, sizing of the panels should be done to maximize the ratio of net effective collector area to gross installation area. The number of panels required to cover the installation area should be as small as possible, keeping within the 150 pound limit, to minimize installation costs.

The space between the cover plate/plates and the collector plate shall be hermetically sealed containing dry air or condensation shall be avoided by another approved technique.

The collector assembly does not serve as the weatherproof roof of the building. Individual collector units shall be weathertight. The collector must be designed to provide the required crown ventilation (3" space every 6'-6") when installed. Spacing must also be adjusted to drain the water collected on the solar collectors during heavy rain (2 inches/hr.) and in particular to avoid water buildup and cascading past the gaps and down the length of the collector. The collectors must perform satisfactorily, without major repair, for the duration of the proposed experiment--three years as an absolute minimum. The ultimate objective is collector life equivalent to that of the cladding it will replace.

Experimental or field substantiation of collector performance, maintainability and life should be supplied as available.

General Objectives

The collector is to provide solar thermal energy for units operating in the temperature range 140°F to 180°F. Maximum energy demands on the system occur during the period from 15 June to 15 September of the year. The combination of relatively low collection temperatures needed and the high ambient temperatures during the peak load period allow for reasonable or high efficiencies with relatively unsophisticated assemblies. Durable, lightweight, easy-to-install and service panels are desired requiring a minimum of maintenance over a reasonable lifetime. We are looking for an experienced supplier with an established reputation for durable, reliable products and for delivery on schedule.

Data Sheet: The following data sheet shall be filled out to assist the owner in evaluating the proposal. The contractor may include any other data such as performance tests, photographs, engineering drawings or similar items that lend credence to claims made in the proposal.

DATA SHEET

1. Dimensions:

   Type of Collector____________________

   Unit Length____________Width________, Thickness______

   Pipe Size of Supply and Return Connections____________

   Weight of Assembled Panel____________________

   Net Aperture Area per Panel____________________

   Net Absorber Area per Panel____________________

   Gross Area: Panel____________; Assembly (if any)________

2. Cover Plates:

   Number of covers________Optical Properties of Cover____________

   ____

3. Absorber:

   Type of Material for Absorber____________________

   Absorber Surface Treatment____________________

   Absorptivity____________Emissivity____________

   Maximum Allowable Temperature____________________

4. Insulation Data:

   Material________,Thickness________,Thermal Properties________

5. Collector Thermal Performance:

   Expected performance for collector tilted 29° west of south at an angle of 45° with horizontal and at an angle of 30° with horizontal with indicated variable solar flux (Btu/hr. sq. ft.) total insolation as indicated for a flow rate of 0.017 gpm/sq. ft. with steady state conditions. The flux listed on the next page is radiation incident at a plane normal to the incident beam.

The efficiency is to be defined as the amount of energy removed by the heat transfer fluid per unit aperture over a 15-minute period divided by the total incident solar radiation onto the collector for the 15-minute period.

Temp. Diff. Entering Fluid & Ambient Air Temp. °F		Temp. Ent. Coll. Panel	Temp. Lv. Coll. W/ Variable Solar Flux (Btu/Sq.Ft./Hr.)			Efficiencies W/ Variable Solar Flux		
			275	200	100	275	200	100
		45° Tilt						
80	T June 21	170°F	____	____	____	____	____	____
70	T Mar 21	150°F	____	____	____	____	____	____
70	T Sept. 21	120°F	____	____	____	____	____	____
80	T Dec. 21	100°F	____	____	____	____	____	____

6. Hydraulic Data:

Material in Contact with Circulating Fluid ____________________

Pressure Drop Thru Panel: ________ ft. of water w/ ______________ gpm

Maximum Flow Thru Panel: ________ gpm w/ ____________ ft. of water PD.

7. The complete collector will be ready for shipment _______ days after receipt of order.

8. Total price, of collector panels including supervision of installation at the job site in New York City, including subassembly on site if proposed and installation on site if proposed. (Specify)

____

____

____

____

____

9. Name of Proposer

Address

Signature

Title

SPECIAL REQUIREMENTS
SOLAR COLLECTOR PANELS

CITICORP CENTER

1. The Solar Collector manufacturer is to furnish to the job site all panels and appurtenances in truckload lots for unloading and distribution by others. The delivery of the panels is to be coordinated with the erection sequence. All trucking to be in trucks with hydraulic tailgate lifts.

2. The Subcontractor will coordinate his work with the Subcontractor that is to install the work and is to submit detailed shop drawings, after coordination, for the Architects' and Engineers' approval.

3. For maintenance purposes, and to replace parts damaged during erection, the Subcontractor is to guarantee availability of replacement parts of the system for a period of not less than 10 years.

4. The individual panels are to be palletized to facilitate unloading and transporting to the roof. The pallets are not to exceed 1500 lbs. and are not to exceed 10' long by 48" wide by 50" high.

5. Payment Clause:
Upon receipt of payment from the Owner, monthly payments, on account, in the amount of 90% of the value of the work delivered to the building site during the preceding month, and the balance of 10% thirty (30) days after completion of the project and acceptance by the Owner.

6. The Subcontractor's attention is particularly called to the following items which are a part of the Construction Contract between the Owner and the Contractor and the provisions of said Contract relating to such items are hereby extended to and shall apply equally to this subcontract and the parties hereto:

A. GUARANTEE BY SUBCONTRACTOR

In addition to, and not in limitation of, any other obligations to the Owner under any of the provisions of this Agreement and the General Conditions (A.I.A. and Supplementary) the Subcontractor agrees as follows:

1. The Subcontractor agrees to indemnify and hold harmless the Owner and/or Contractor against any and all claims which may at any time be made against the Owner and/or Contractor whether during the time of this Agreement, or thereafter, for labor performed or materials furnished in connection with the Work, provided the Owner and/or Contractor has not refused or failed to pay the Subcontractor for the cost thereof, as provided in this Agreement.

2. The Subcontractor accepts, as between the parties to this Agreement,

exclusive liability for the payment or withholding of any tax, federal, state, municipal or otherwise, with respect to compensation, wages or other remuneration for any work to be performed by the Subcontractor or its employees under the terms of this Agreement, and the Subcontractor further agrees to indemnify and hold harmless the Owner and/or Contractor against all such taxes and to comply with all governmental regulations with respect thereto, including the filing of the necessary returns,, provided that the foregoing shall not be deemed to affect the Owner's and/or Contractor's obligation to pay the Subcontractor for all such taxes to the extent provided in Article VI.

3. The Subcontractor and every sub-subcontractor shall execute and deliver to the Owner a written guarantee covering all work under this Agreement and under each sub-subcontract. Such guarantee shall be for such periods as specified in the several trade sections of the specifications. Should any defect develop during the period or periods during which any such guarantee agreement shall be in effect, due to improper materials, workmanship, arrangement (excluding design) or failure to comply with the plans and specifications, the same shall be made good by the Subcontractor without cost or expense to the Owner and/or Contractor. Any other work affected in making good such imperfections shall also be made good by the Subcontractor in like manner. The Guarantee agreements required by the provisions of this Article shall be in addition to any other guarantees, bonds or other undertakings required of the Subcontractor or sub-subcontractors under any other provisions of this Agreement.

4. The Subcontractor shall be responsible for his work and the work of each sub-subcontractor and every part thereof, and for all materials, tools, appliances and property of every description used in connection therewith.

5. The Subcontractor and his subcontractors shall obtain written consent from the Contractor before subletting any part of their work. Where such consent is granted, the Subcontractor and his subcontractors shall be bound by the same terms and conditions covering all insurance described and set forth in the Owner's Job Insurance Program identified as Rider No. 3 of this Agreement.

7. The Bidders are to submit with their Proposal, Unit Prices for adding or deleting Solar Panels on a per square foot or per panel basis.

RIDER NO. 1-A

1. Seller agrees that the articles and merchandise to be furnished hereunder will not infringe any valid patent or trademark, that Seller will, at its own expense, defend any and all actions or suits charging such infringement, and that Seller will indemnify us, our customers and those for whom we may act as agent in the purchase of said articles and merchandise against, and save us (and each of us) harmless from, any loss, liability or damage in case of any such infringement, or claim of infringement.

2. It is agreed that time is of the essence of this order, and if delivery is not effected within the time specified, we reserve the right to cancel this order and to hold Seller liable for damages sustained by us as a result of Seller's failure in this respect. If the occurrence of any event or cause which would constitute an excuse in law for a delayed delivery or nondelivery is to be relied on by Seller, it is agreed, as a condition of such reliance, that Seller notify us in writing thereof within five (5) days after such occurrence, and in such case we may cancel this order at our option, either retain partial shipments theretofore made and accepted by us as satisfactory at the contract price thereof, or return the same at Seller's expense, and may also at our option, request the delivery of articles or merchandise partially processed pursuant to this order, at the cost thereof to Seller as determined by Seller's usual accounting methods, in their existing state, and/or the delivery of such articles which have been completely manufactured by Seller, at the contract price. We reserve to ourselves the right, at any time before shipment, to postpone delivery dates for a reasonable time.

3. Seller warrants that all articles and merchandise sold hereunder will conform to the specifications, drawings, samples or other descriptions furnished and in all respects will be of good material and workmanship and free from defects. All articles and merchandise hereunder shall be subject to inspection and tests by us and by the United States Government or other customers. Rejected articles may be returned at Seller's risk and expense. We reserve the right to inspect any such articles and merchandise (including materials or equipment made to special specifications) at any stage in the process of manufacture, but any such inspection or any approval, shall be provisional only and shall not constitute final acceptance nor shall it be construed as a waiver of any rights we may have after receipt thereof at specified designation. Rejection and return of any portion of shipment which may be defective or which fails to comply with specifications may be made without invalidating the remainder of the order, at our election.

4. This order may not be filled at prices higher than those last charged for the same material, without notification and acceptance of the advanced prices.

5. We reserve the right, prior to the delivery of the articles of merchandise herein ordered to make changes in quantities and/or specifications relating to such articles and merchandise, and in such event, any variation in the cost of furnishing the articles and merchandise specified shall be equitably reflected in adjustment of price.

6. Transportation must be prepaid by Seller on all shipments to which a delivered price applies. Charges for prepaid transportation must be substantiated by Seller by attaching to the invoice original transportation bills duly receipted by the carrier.

   Except as shall be expressly provided for in writing duly signed by us, we will not accept charges for packing, packaging, or transportation, including C.O.D. shipments.

   Seller shall insure at full value all shipments by Parcel Post, express or commercial steamship lines. The articles and merchandise to be furnished hereunder shall be at Seller's risk until actual delivery to us or our designee at such place or destination as may be specified in this Purchase Order or may be specified by us in writing at a later date.

7. Seller agrees that no part of this order shall be sublet without our written approval and that the articles and merchandise to be furnished hereunder must be furnished only by the person or firm to whom this Purchase Order is addressed.

8. If this order is placed under, or in connection with the performance by us of, a Government prime contract or sub-contract, or in regard to a project whose mortgage is insured by a Governmental Agency (hereinafter referred to as Government Contract), and whether or not disclosure thereof is made to Seller, we reserve the right, in case of a cancellation or other termination of such Government Contract to cancel this order in whole or in part upon notice to Seller at any time prior to the completion of the work to be performed by Seller hereunder and upon receipt of such notice Seller shall stop all work hereunder except as otherwise directed by us in writing. In the event of such cancellation or termination, the terms and conditions of the underlying contract or contracts, and of any applicable statutes and regulations then in force, as to the terms of settlement between us, and as to the manner, method and amount of payment shall be controlling of our obligations hereunder and any right of action against us that Seller might otherwise have on account of our cancellation hereof for the reasons aforesaid shall be suspended until our own settlements under said contracts have been consummated.

9. Seller agrees to comply with, and warrants that all articles and merchandise covered hereby and labor employed in the manufacturing, processing, handling and delivery thereof shall be subject to all applicable laws, regulations and ordinances of the United States of America, or any state, or municipality, or governmental authority or agency in respect of the manufacture, procurement or sale of any goods to be supplied pursuant to this Purchase Order, more particularly, and without limiting the generality hereof, the Fair Labor Standards Act, Public Contracts Act, Buy America Act, and Davis-Bacon Act.

   Seller agrees, upon our request, to furnish us with appropriate affidavits of such compliance.

10. In the event of any strike, fire, accident or other cause beyond our control which shall affect our ability to receive and/or use the articles and merchandise herein ordered, we shall have the right, upon written notification to Seller by telegram or letter and without penalty to us, to order a delay in scheduling shipment of articles and merchandise to us.

11. In case of entry by the Seller, or any of the Seller's agents or employees, upon our property or property upon which we are engaged in construction, alteration or other work of any kind, nature or description, for the purpose of making delivery of articles and merchandise covered by this Purchase Order, or for the purpose of making inspection thereof, or inspection of the site itself, the Seller agrees to provide all necessary and sufficient safeguards and to take all proper precautions, against the occurrence of accidents, injuries or damages to any persons or property and to be responsible for and to indemnify and save us harmless from all loss or damage and any or all claims arising by reason of accidents, injuries, or damage to any person or property in connection with such work at such location(s) or site(s) except such as may be the sole and direct result of negligence on our part, and from all fines, penalties or losses incurred by reason of the violation of any law, regulation, or ordinance, and Seller further agrees to defend at the Seller's expense, any and all suits or actions civil or criminal arising out of such claims or matters and further agrees to procure and carry such insurance of employees as may be required by any Workmen's Compensation Act or other law, regulation, or ordinance, which may apply in the premi s.

   Should the Owner of the project for which the articles and merchandise to be delivered hereunder are intended, or any other person or persons at any time, assert a claim or institute any action, suit or proceeding against us involving the manner or sufficiency of the performance of the work comprehended by the contract between us and such Owner or any of the contract documents, insofar as the same may relate to or affect any part or portion of the articles and merchandise hereby undertaken to be furnished by the Seller, the Seller will, upon our request, promptly take over the defense of any such claim, action, suit or proceeding at the sole cost and expense of the Selle d will also indemnify us and save us harmless from and against any and all liability, damages, judgments, costs or expenses a ng out of or in connection with any such claim, action, suit or proceeding.

RIDER NO. 1-A (Cont'd.)

12. The waiver or acceptance by us of any breach on Seller's part of any of the terms of this order shall not relieve Seller of responsibility hereunder for any prior or subsequent breach.

13. It is agreed by the Seller that any right, cause of action, or remedy under the warranties or undertakings assumed or imposed upon the Seller under this order shall extend without exception to any company affiliated with us or upon whose behalf this order is issued by us, as the interest of such company shall appear.

14. Should the Seller be adjudged bankrupt or make a general assignment for the benefit of creditors, or should a petition under the Bankruptcy Act, or under any other act relating to insolvency, be filed by or against the Seller, or should a Receiver be appointed on account of the Seller's insolvency, or should the Seller fail in any respect to furnish articles and merchandise hereunder with promptness and diligence, we may, after three (3) days written notice to the Seller, terminate this agreement and make other arrangements for the procurement of such articles and materials, or any part thereof, and deduct the cost thereof from any money due or thereafter to become due to the Seller. In the event of such termination, Seller shall not be entitled to receive any future payment under this agreement until all of the articles and merchandise to be furnished under this agreement have been obtained by us, at which time, if the unpaid balance of the amount to be paid under this agreement shall exceed the expense incurred by us in obtaining such articles and merchandise, such excess shall be paid by us to the Seller, but if such expense shall exceed such unpaid balance, the Seller shall pay the difference to us.

15. This order is confidential between us and the Seller, and it is agreed by the Seller that none of the details of this transaction shall be published or disclosed without our written permission. Any violation of this condition shall be deemed a material breach of contract.

16. It is agreed that the foregoing are in addition to, and not in limitation of, all and whatever rights, not herein specifically dealt with, as are conferred upon us by law.

Attachment II

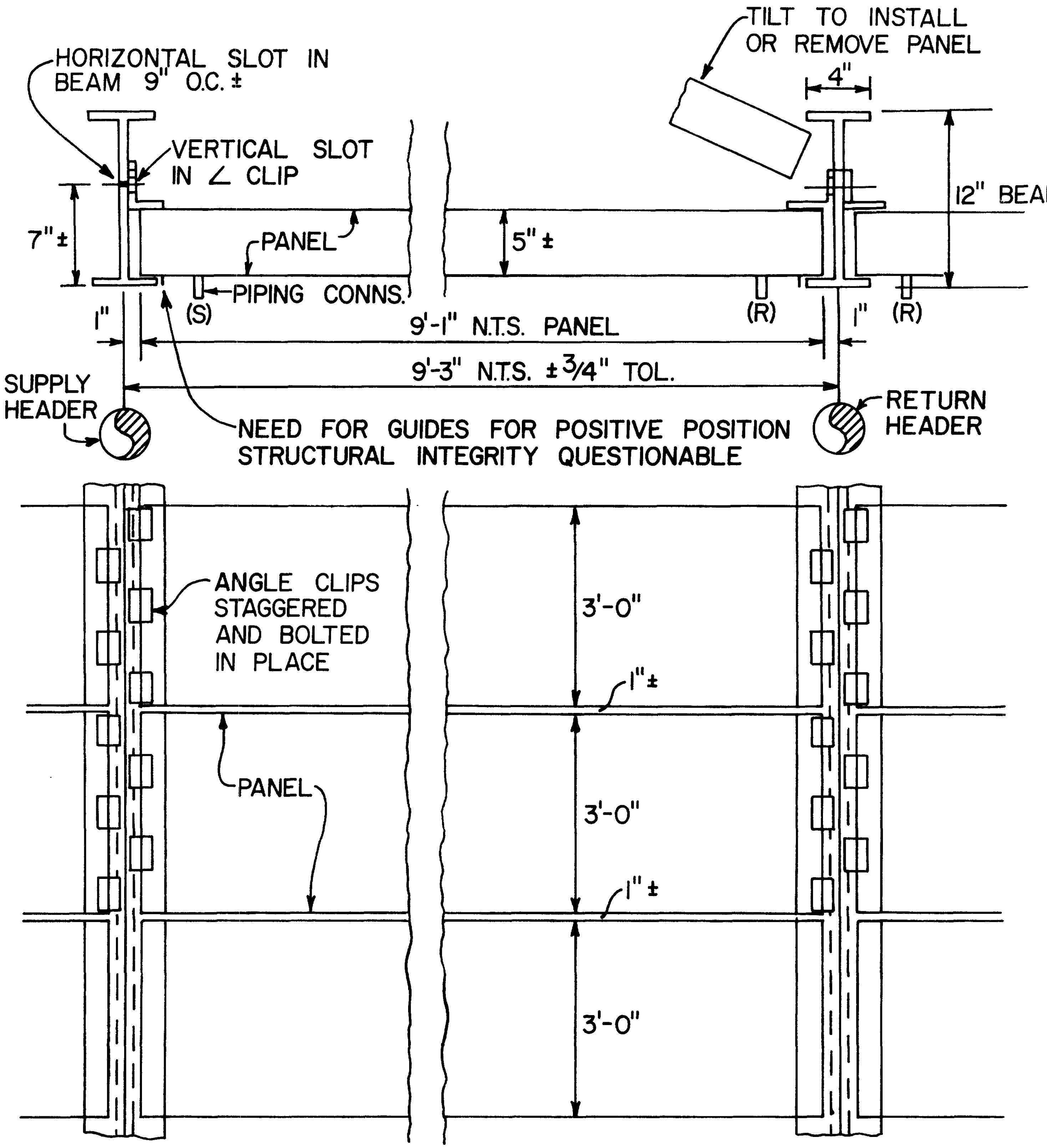
Suggested alternatives for solar panel installation.

Solar Panel Mounting:

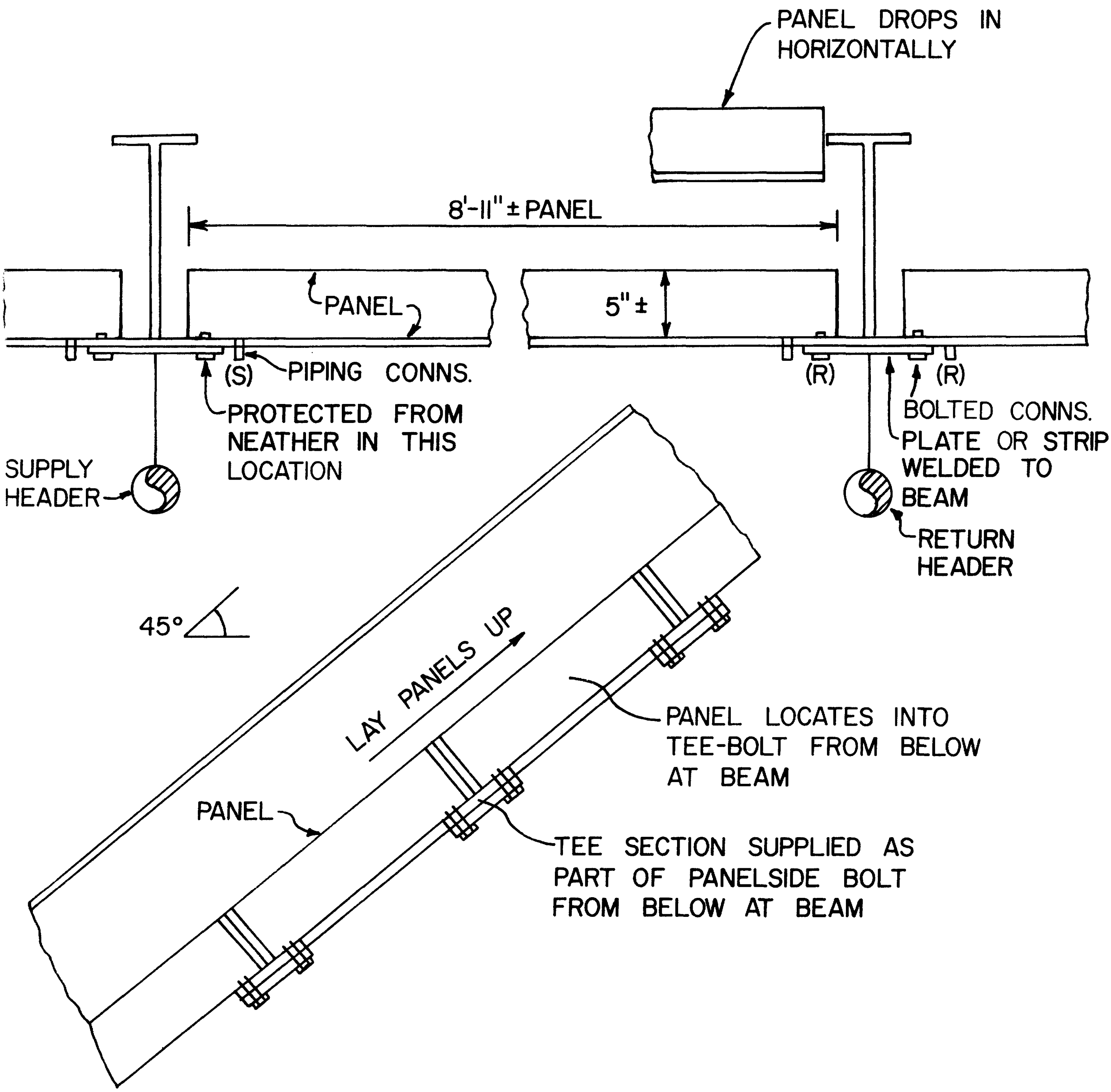
The requirement that the solar collector panels support themselves and the standard loads over a clear span of nine feet, the need to brace the structural steel purlins near the center of their webs, the requirement to ventilate the crown with at least 3 inch gaps every 6 feet 6 inches, and the desire to overlap some of the collector framing to increase the effective collector area to physical area ratio led to the preliminary exploration of and "S" beam concept. See sketch.* The "S" beam not only would provide the necessary purlin bracing, but also stiffen the collector modules by giving them greater effective depth in addition to the stiffness provided by the beam itself. This configuration lends itself to mounting collector panels from the rear and allows individual panels to be small enough for manhandling. This mounting provides a slightly smaller collector angle than the 45 degrees from horizontal provided by the steel structure itself.

Outstanding questions are those of cost and the potential for developing cascading of water during heavy rain.

* Fig. IV-3 of this appendix


HORIZONTAL SLOT IN BEAM 9" O.C. ±
VERTICAL SLOT IN ∠ CLIP
TILT TO INSTALL OR REMOVE PANEL
4"
12" BEA
7" ±
PANEL
5" ±
PIPING CONNS.
(S)
(R)
(R)
1"
1"
9'-1" N.T.S. PANEL
9'-3" N.T.S. ± 3/4" TOL.
SUPPLY HEADER
RETURN HEADER
NEED FOR GUIDES FOR POSITIVE POSITION
STRUCTURAL INTEGRITY QUESTIONABLE
ANGLE CLIPS STAGGERED AND BOLTED IN PLACE
3'-0"
1" ±
PANEL
3'-0"
1" ±
3'-0"

SCHEME 1


PANEL DROPS IN HORIZONTALLY
8'-11" ± PANEL
PANEL
5" ±
(S)
PIPING CONNS.
(R)
(R)
PROTECTED FROM NEATHER IN THIS LOCATION
BOLTED CONNS.
PLATE OR STRIP WELDED TO BEAM
SUPPLY HEADER
RETURN HEADER
45°
LAY PANELS UP
PANEL LOCATES INTO TEE-BOLT FROM BELOW AT BEAM
PANEL
TEE SECTION SUPPLIED AS PART OF PANELSIDE BOLT FROM BELOW AT BEAM

SCHEME 2

Appendix V Health Effects of Triethylene Glycol

I. Health Effects

The dehumidifier recommended for installation in Citicorp Center manufactured by Niagara Blower Co. employs a liquid desiccant "Hygrol" (TM), a stable, hygroscopic liquid solution based on triethylene glycol. Triethylene glycol, also known as triglycol is a member of a large chemical family which includes the familiar ethylene glycol, a base for automobile antifreezes.

Because Hygrol is sprayed into the air passing through the dehumidifier unit and a fraction of the Hygrol, depending mainly on the spray contact temperature, droplet size, and air flow, is carried over with the conditioned air. For this reason a literature search for potential health effects of triethylene vapors and aerosols was undertaken. No adverse indications were found in this search. Only salutory short term effects have been reported. We have no information on long term effects.

In The Toxic Substances List, 1974 Edition[1] no danger from vapor is specified and only massive ingestion of the liquid (10gm/kg) would cause problems.

The discussion of triethylene glycol in Industrial Hygiene and Toxicology (2nd Rev. Ed.) it is pointed out that triglycol is less volatile and less toxic than diethylene glycol. Research in the 1940's by O.H. Robinson and coworkers indicated that triethylene glycol was an effective air sterilizer. Other researchers exposed numerous persons to triglycol vapors in air with no ill effects. "It is generally concluded that indefinitely long exposure to air at room temperatures substantially saturated with vapors of triethylene glycol are harmless." See page 1510.

Hamilton and Hardy in Industrial Toxicology, Third Edition report that ethylene glycol has a low vapor pressure, and significant air concentrations are not achieved unless the compound is heated or sprayed as a mist. Respiratory exposures or application of the material to the skin are not considered to be of toxicological significance. Triethylene glycol has an even lower vapor pressure.

In 1961 experiments were conducted on apparatus set up in Niagara Blower Company's research laboratory to measure the Hygrol dehumidifier's effectiveness with micro-organisms of the types ordinarily present in the atmosphere of an industrial area under normal operating conditions. The tests were conducted by Biology Professor Valentine J. Nadolinski of the State University of New York in Buffalo. Nadolinski found that over 98% of such organisms, including molds, yeasts, and bacteria were removed.

While a variety of glycols will function as air disinfectants, triethylene glycol is generally preferred because of the extremely small amounts required. Only 0.18 pounds per million cubic feet of air at 70°F and 40%

humidity will saturate it or only 1/2 part per million on a volume basis. Best antibacterial action is had for humidities between 20 and 50 percent and dry bulb temperatures betweeen 60 and 85°F.

From the experience above it appears the triglycol based air conditioing systems can provide an added service, that of disinfecting the air. In the Citicorp Center system only the minimum outside ventilation air passes through the dehumidifier which represents only about 13% of the air circulated by the interior air handler systems.

II. Selected Bibliography on Health Effects of Triethylene Glycol

George O. Curme, Jr., Ed., "Glycols", Reinhold, New York, 1953.

Herbert E. Christensen, Ed., "The Toxic Substances List, 1974 Edition", National Institute for Occupational Safety and Health, Rockville, Maryland, (June 1974).

Alice Hamilton, and Harriet L. Hardy, "Industrial Toxicology, Third Edition", Publishing Sciences Group, Acton, Mass. (1974).

O.H. Robertson, E. Bigg, B.F. Miller, Z. Baker and T.T. Puck, "Sterilization of Air by Certain Glycols Employed as Aerosols and Vapors", Trans. Assoc. Am. Physicians 56 353-357 (1941).

O.H. Robertson, Theodore T. Puck, Henry F. Lemon and Clayton G. Loosli, "The Lethal Effect of Triethylene Glycol Vapor on Air-Borne Bacteria and Influenza Virus", Science 97, 142-144 (1943).

M. Hamburger, T.T. Puck, V. Hurst, O.H. Robertson, "The Effect of Triethylene Glycol Vapor on Air-Borne Beta Hemolytic Streptococci in Hospital Wards I, II, III, J. Infections Diseases, 76, 208-225; 77, 177-180 (1945).

E. Bigg et al., "Triethylene Glycol Vapor Distribution for Air Sterilization", Heating, Piping, Air Conditioning No. 5, 103-107, (1947).

Saul Krugman, and Bertha Swerdlow, "Lethal Effect of Triethylene Glycol Vapor on Air-Borne Mumps Virus and Newcastle Disease Virus", Proc. Soc. Exp. Biol. and Med., 71, 680-684 (1949).

W.J. McConnell, "Air Experiment with Triethylene Glycol Vapor for the Control of Colds Among Office Employees", Ind. Med. 18, 192=196 (1949).