General Disclaimer

One or more of the Following Statements may affect this Document

- This document has been reproduced from the best copy furnished by the organizational source. It is being released in the interest of making available as much information as possible.

- This document may contain data, which exceeds the sheet parameters. It was furnished in this condition by the organizational source and is the best copy available.

- This document may contain tone-on-tone or color graphs, charts and/or pictures, which have been reproduced in black and white.

- This document is paginated as submitted by the original source.

- Portions of this document are not fully legible due to the historical nature of some of the material. However, it is the best reproduction available from the original submission.

Produced by the NASA Center for Aerospace Information (CASI)

DOE/NASA CONTRACTOR
REPORT

DOE/NASA CR-150861

December 1978

PROTOTYPE SOLAR HEATING AND COOLING SYSTEMS INCLUDING POTABLE HOT WATER (Quarterly Report)

Prepared from documents furnished by

Solaron Corporation
Solaron Energy Systems
Denver, CO 80222

Under Contract NAS8-32249 with

National Aeronautics and Space Administration
George C. Marshall Space Flight Center, Alabama 35812

For the U. S. Department of Energy

FEB 1979
RECEIVED
NASA STI FACILITY

U.S. Department of Energy

Solar Energy

TABLE OF CONTENTS

QUARTERLY REPORT

2 NOVEMBER 1976 - 30 APRIL 1977

NAS8-32249

CONTRACT TITLE:

DEVELOPMENT, DELIVERY AND SUPPORT
OF TWO (2) PROTOTYPE SOLAR HEATING AND COOLING
SYSTEMS, INCLUDING POTABLE HOT WATER

PREPARED BY: Don Bloomquist and Rodney L. Oonk

APPROVED BY: L.E. Shaw

v

Part I - Summary

1.0 This report will summarize and bring up-to-date the reporting on this contract from the effective date of the contract, 2 November 1976, through the month of April 1977. Current and cumulative costs, contract changes, schedules, and technical performance are included.

Part II - Contract

2.0 The original contract as awarded, signed and dated 11/2/76 remains unchanged except for:

    (1) Ammendment/Modification No. 1 dated and signed Nov. 8 1976.

    (2) Technical Directive No.   1/AH-00183 dated 12/17/76.

    (3) Technical Directive No.   2/AH-00184 dated 12/17/76.

    (4) Technical Directive No.   3/AH-00225 dated 1/21/77.

2.1 Cost -

Part III - Schedules

3.0 The contract schedule was reviewed and revised at the request of the Technical Manager, Mr. Mitchell Cash. This revision was completed in April, but not submitted to NASA at that time because of information from Mr. Cash that NASA was changing the installation schedule of the heating systems, originally scheduled to be completed in the third quarter of 1977, to coincide with the installation of the cooling systems in the second quarter of 1978. This 4/77 revised schedule is attached for reference purposes. The schedule will be revised again to reflect the new sequence of events and will be coordinated with the Technical Manager.

Part IV - Technical Performance

4.0 The technical performance under this contract is reported in this part in an essentially chronological order.

4.1   The period of time from the effective date of the Contract (11/2/76) until the first Technical Interchange Meeting with the Technical Manager, Mr. Cash, was spent in preliminary design activities. These activities included the preparation of schematics and psychometric charts for the baseline desiccant cooling system as presented in the proposal for review at the first Technical Interchange Meeting (TIM) which was scheduled for 12/22/76.

4.   The first TIM was held at Solaron, Denver, as scheduled, with Mr. Cash representing NASA. The following significant items were discussed and dealt with in this meeting.

4.2.1   Site selection for the two program demonstration sites - ERDA /NASA expressed interest in Cleveland, Ohio and New Orleans, Louisianna as sites. Solaron expressed dissatisfaction with these sites because for New Orleans there is a minimal heating load and for Cleveland there is extensive fog and cloud cover which causes solar insolation to be inordinately low. These matters were to be considered by NASA and ERDA in making final site selection.

4.2.2   Mr. Cash indicated that not enough time was allotted in the Contract program schedule (Ref: Contract NAS8-32249, SHC 3014 pgs. 2 & 3) for study and analysis and that the schedule should be revised by Solaron subject to NASA approval. Solaron pointed out that any delay in site selection could have a direct impact on the program schedule.

4.2.3   Solaron has negotiated an OEM arrangement with Honeywell to supply control components for solar heating systems. This arrangement involves components for both heat pump and desiccant cooling systems as well as space and water heating systems.

4.2.4   Mr. Cash said that the definition in the Contract that the two systems shipped for installation should be "identical" is not necessarily true, that the systems shipped may have to be "tailored" for the site locations.

4.2.5   Mr. Oonk reported on the computer simulation work being done under subcontract by Carrier Corp. and the University of Wisconsin as follows:

> Carrier - Carrier is working on developing the computer model of the three (3) basic solar heat pump systems called out in the proposal. After initial inter-component parametric studies, plans are to complete a one year simulation of the three systems.
>
> U. of Wisconsin - The University of Wisconsin has a computer program for solar assisted desiccant cooling systems. This program will be modified as necessary and will be used to study several solar assisted desiccant cooling systems.

4.2.6   Mr. Cash asked if Solaron would test the life cycle of desiccants. Solaron responded that this would require an extremely long time span, and that our approach to this was to get the data from manufacturers and users of the desiccants.

4.2.7   Dr. Löf (Solaron) commented that the whole solar assisted cooling concept is "wide open" and that Congress expects to have this concept in effect within five years, which he felt was a relatively short time.

4.3   Work continued in defining psychrometric air cycles that would be adaptable for solar assisted cooling using a desiccant drying process. Additional study and investigation was conducted to identify suitable equipment components for the various cycle processes. These processes are:

4.3.1   Desiccant drying of the process air to a degree that will permit effective evaporative cooling. (By its inherent nature, this process adds sensible heat to the air stream.)

4.3.2   Air-to-air heat exchange, transferring heat from the dried process air to the regeneration air stream.

4.3.3   Evaporative cooling and humidification of the air to a condition (cycle state point) suitable for delivery into the conditioned space.

4.3.4   The "room line" along which the air gains sensible and latent heat as a function of the characteristic cooling loads of the conditioned space.

4.3.5   Sensible heating of the regenerating air stream in two stages:

     (a) by air-to-air transfer of heat from the process air stream leaving the desiccant dryer, and

     (b) the addition of the required balance of heat by the solar heating/storage system when available, or by the auxiliary heating system when solar energy is not available.

4.3.6   Regeneration of the desiccant drying system with the hot air stream developed in process 4.3.5 above.

4.4   For all of the above processes, the primary criteria for selection of hardware components to be used in subscale testing and ultimately in prototype system design are:

     a. that preference will be given to equipment available on the commercial market,

     b. that preference will be given to equipment that has a proven track record over a sufficient period of time to establish its reliability,

     c. that equipment will be cost effective for the process involved,

d. that the equipment will be reasonably compact in size and configuration, and

e. that the equipment can be adapted to a solar assisted cooling system with minimum modification.

4.5 For the desiccant drying process (4.3.1 above) several concepts were considered:

4.5.1 Triethylene glycol (TEG) used as a regenerable drying agent. This is not considered a viable system for the following reasons:

a. It is a wet system subject to corrosion and leaks.

b. It requires a complex system of pumps, valves, heat exhangers and piping.

c. There could be some liquid (mist) carryover into the conditioned space, which would condense out onto walls, windows furniture and drapes. TEG is expensive, so that make-up costs would be high.

d. System controls would be complex and expensive.

4.5.2 The Munters Environmental Control (MEC) system. This is not currently under consideration for the following reasons:

a. It is not commercially available on the open market.

b. It is still in a laboratory test stage of development.

c. It employs asbestos compounded wheels through which the process air flows and this could have environmental consequences.

d. The MEC unit/system would have to be manufactured on a sub-contracted "model shop" basis which flags many potential problems and certainly points to a high cost.

4.5.3 A dual fixed bed silica gel system. This system employs two fixed silica gel beds which must be periodically switched from process air flow to regeneration air flow. The fixed bed systems are not considered to be viable candidates for the following reasons:

a. They are not available on the commercial market for use in air conditioning systems, although they are available for compressed air systems in heavy wall tank configurations.

b. The R & D of a fixed bed system would be expensive and the systems would have no predictable advantage over commercially available equipment.

4.5.4   The Cargocaire lithium chloride rotating wheel unit. This
unit uses the same asbestos wheel structure as the Munters unit,
and, in fact, is manufactured under license from Munters. This
unit is not under active consideration for the following reasons:

a. Because of the asbestos structure of the wheel. (See
4.5.2 - c above)

b. The fact that the lithium chloride coating on the wheel
structure will deliquesce and drain off the wheel if inlet air
ever approaches saturation. Replacing the wheel or recoating
of the original wheel is quite expensive.

4.5.5   The Bry-Air unit with silica gel rotating bed. This unit is
now the prime candidate, and is scheduled to be used in Solaron's
subscale testing of system components. The following reasons are
cited for the selection of this unit.

a. It has a good track record of performance and reliability
on the commercial market.

b. Silica gel is considered to be the best available choice
of desiccant for the desiccant cooling system.

c. The University of Wisconsin Computer modeling program has
silica gel data in place and would require a great deal of
time to obtain and load data on other desiccants.

d. U.S. Navy experience with the "mothball fleet" over a long
period indicates that silica gel has an extremely good life
cycle, on the order of 30 years under proper operating condi-
tions.

4.6   For the heat exchange process in which heat is transferred
from the effluent air leaving the desiccant dryer to the regener-
ation air stream, the following considerations are of prime im-
portance:

a. The exchange of heat must be essentially 100% sensible.

b. The heat transfer efficiency must be high (on the order of
85 to 95% optimization of overall system cost effectiveness -
vs. - heat transfer efficiency depends on the output of the
computer modeling program.)

Available air-to-air heat exchangers have been studied. These in-
clude counterflow plate type, rotary wheel type, and heat pipes.
The most promising at this time appears to be the heat pipe, and
our investigation of heat pipes is continuing. Pros and cons of
these types of heat exchangers are as follows:

a. The plate type is essentially the simplest but for the required efficiencies becomes excessively long (large) in the direction of air flow.

b. The rotary wheel type is efficient for its depth in direction of air flow and has a low pressure drop character- istic, but, on the negative side, it is large in the dimen- sions normal to the direction of air flow, and requires a separate, powered drive system and rubbing seals which tend to degrade its reliability and maintainability. Also there is some moisture contamination of the dry side because of seal leakage and the hygroscopic characteristics of contaminants that tend to build up on the wheel in a time related manner. The first cost also is higher for the wheel than for the plate type exchanger.

c. Evaluation of heat pipes has not been completed, but pre- liminary information indicates that they will be superior in compactness and heat transfer efficiency. Also, the heat transfer is 100% sensible. Cost is expected to be comparable to rotary wheels.

4.7   The next process in the cycle is the evaporative cooling process. Several hardware candidates for this process have been studied and evaluated. These are: Air washer with water traps at the outlet end, spinning disc atomizers, wetted aspen fiber media, wetted metal media, combinations of the above, and the wetted Munters celulose media. Of these the Munters media appears at this time to be the optimum choice for the following reasons:

a. This media has the highest saturation efficiency per unit volume.

b. It is available in "building blocks" of variable face dimensions and depth in direction of air flow which makes it adaptable for building units of virtually any size and satura- tion efficiency.

c. Manufacturers who use this media say that it is the very best available (to quote one, "The only media thats any good!"). Also they say that it is self cleaning and otherwise trouble free provided that proper water treatment and/or bleed pro- cedures are followed with respect to the local water condition, and that the media will last for years if proper procedures are followed.

d. Other manufacturers, not now using the media, express in- terest in using it in the design and marketing of their own evaporative cooling units because of its recognised high saturation efficiency, reliability, and good record of main- tainability.

e. The Munters media has been used all over Europe for many years and is recognized there as a standard in the cooling tower and evaporative cooler industries.

4.8 The next process in the cycle is the "room 1 ne", (or "house line") which is characteristic of the ambient design condition of the locality, the design and construction of the house, and the size and life style of the family living there. This process is independent of the heating/cooling system supplying the house except for the mass flow of air supplied to it and the psychrometric condition of that air.

4.9 The next process is the adiabatic cooling of the effluent air from the house. It is used, first of all, as a heat sink for the air-to-air heat exchange processes and, finally, after further heating by the solar system and/or auxiliary heating systems as the regeneration air for the desiccant drying system. The various cycles under study could differ from each other on this process. For instance the effluent air from the house could be returned directly to the desiccant drying process, and a separate air stream of 100% outside ambient air could be used for regeneration. In all cases the regeneration air, after being preheated by the heat exchange process, would be further heated by the Solaron solar air collector or rock storage to the required regeneration temperature when heat was available from either of these sources. An auxiliary heating system will be supplied to suppliment the solar system when energy is not available from that source. This air is then dumped to atmosphere after passing through the desiccant bed in the regeneration process.

4.10   The first Quarterly Review was held at Solaron, Denver on 17
        February 1977 with Mr. Mitchell Cash representing NASA.

        Progress on the design analysis of the solar assisted desiccant
        system was presented using handouts of the psychrometric cycles,
        block schematic diagrams, and a step-by-step write-up of the
        various processes and hardware components involved.

        Two distinctly different systems were presented as follows: (Note:
        See Attachments for relevant psychrometric cycles and schematic
        block diagrams)

            a. A closed (recirculating) cycle in which the house air is
            continuously recirculated through the cycle without the
            addition of fresh air, and a separate air stream, using 100%
            outside ambient air, to regenerate the desiccant sub-system,
            (See Figures R-1 & R-2) and

            b. An open (ventilating) cycle is which 100% outside ambient
            air is used to condition the house, and, in the same continu-
            ing stream, to regenerate the desiccant sub-system (See
            Figures V-1 & V-2).

        These two options were discussed in detail with reference to psy-
        chrometric cycle diagrams (Figures R-1 & V-1) and to block schematic
        diagrams (Figures R-2 & V-2) along with a descriptive narrative,
        copies of which are attached for reference.

4.10.1 Mr. Cash pointed out that our desiccant systems were quite
        similar to the MEC system and he was concerned that there might be
        a problem with patent infringement. Mr. Shaw replied that Solaron's
        intent was to use commercially available equipment components to
        the maximum extent possible, and that this should obviate any
        possibility of patent infringement. It was agreed, however, that
        this potential problem would be investigated in depth. Mr. Cash
        agreed to identify and send to Solaron reports that might help to
        analyze this potential problem.

        Solaron did immediately thereafter have a meeting with its patent
        attorney and the substance of the outcome of that meeting was that
        Solaron probably did not have a patent infringement problem.
        However, Solaron will institute a patent search when a specific
        desiccant system configuration has been chosen.

4.10.3 The "house" load of 24,000 Btu/hr. (2 tons) as specified in
        the Contract was discussed from the standpoint that it might not
        match the actual load of the houses selected for this demonstration
        program. Mr. Cash pointed out that the 2 ton load specified in
        the Contract is not "poured in concrete" and that the load and
        other parameters could, and would, be adjusted after the site
        selections were completed (two separate houses in different loca-
        tions).

Relative to a probable site location in Cleveland, Ohio (or the Cleveland Area), Solaron expressed concern that it was not a good demonstration site from the standpoint of solar insolation because of extensive fog and cloud cover.

Mr. Cash asked that Solaron supply him with increased cost estimates for 3 & 4 ton systems in anticipation of a possible increase in tonnage to match an actual site selection. Solaron agreed to supply him with this information.

Mr. Cash asked if our proposed desiccant system would function properly under partial load conditions. Solaron assured that it would. (The capacity of the system can be controlled by modulating the amount of heat going into the desiccant regeneration process.)

4.10.4   Mr. Cash mentioned that in his Naval duties he had experience in the use of silica gel as a desiccant for drying air in "mothballed" ships. He also mentioned the 30 years experience of Mr. Tuggle of San Francisco in the "mothball" fleet. Mr. Tuggle has stated that they never changed the silica gel charge, but that they occasionally added new gel to make up for settling of the bed if and when it occurred.

4.10.5   Solaron reported on the information that it had been able to get relative to the life expectancy of silica gel in a dynamic recycling application. Our contact was Mr. Dick Schoofs, a Western States employee of W.R. Grace & Co. of which Davison Chemical is a division. Davison is a long term manufacturer of silica gel products in various types and grain sizes. Mr. Schoofs seems very knowledgeable in the subject, and his statements to us were as follows:

> a. Oil and acids are serious contaminants, and should be trapped out of the flow stream prior to entry into the silica gel bed.

> b. Liquid water droplets, however fine, (as mist) cause high localized heating because of the adsorbtive thermal process. This can cause localized fracturing due to thermal stresses in the gel. All such "mist" should be eliminated prior to air stream entry into the gel bed.

> c. The gel decrepitates after long exposure to high temperature (on the order of 500^0 to 600^0F). "Decrepitate" is standard terminology in the industry meaning cracking and fracturing.

> d. Vibration causes mechanical breakage on a time of exposure and degree of vibration basis. This should not exceed 20% per year in an application using a rotating bed, which could be corrected by the replacement of lost fines with new gel on a periodic basis, estimated to be annually.

e. Extraneous adsorbates such as oil vapor or paint mist as might be found in an industrial atmosphere should be avoided.

4.10.6   Of the above causes for degradation of the gel, the only ones applicable to our specific application are water droplets (mist) and vibration.

As to "mist", the prototype system design must be such that it precludes the possibility of impingement of water droplets or mist on the gel bed. This is not a difficult problem as any efficient filter could overcome this problem.

As to the "vibration" problem, this will be worked with the supplier of the dehumidifier unit.

4.10.7   Solaron asked Mr. Cash for some criteria on which to base the final selection of the type of system to be developed for the prototype operating system, whether "desiccant" or "heat pump", and what type of each. Mr. Cash said that Solaron should make the first cut at this and then negotiate the criteria with NASA.

4.10.8   Solaron stated that Mr. John Nelson, who has been doing our subcontract work at the University of Wisconsin, is leaving the University as of 1 March 1977, and that his work will be taken over by Mr. Pat Hughes. Mr. Nelson will remain in Madison and will be available as a consultant to the extent that he might be needed. Mr. Hughes is competent to do the work and has been briefed by Mr. Nelson on everything that has been done to this point. Solaron anticipates no problems from this change in personnel.

4.10.9   Design Study Building Heating and Cooling Loads and Location.

4.10.9.1 On both the studies being carried on at Wisconsin and
          Carrier, involving a Solaron-Desiccant system and a Solaron Heat
          Pump system, respectively, identical buildings and locations are
          used. This was done to insure that both types of systems face the
          same type of load and weather conditions allowing the results of
          performance calculations to be directly compared.

4.10.9.2 The location of the analyses was chosen to be New York
          City. This choice was agreed upon by Solaron and NASA for several
          reasons. First, the study and analysis of these proposed systems
          is being performed by using a complex hour by hour computer simu-
          lation program. This program necessitates the use of hourly weather
          data. Data of this type is only available for a limited number of
          U.S. locations, of which New York is one. Secondly, New York has a
          sizeable heating and cooling requirement. It was felt that since
          solar cooling systems would only be cost effective when they are
          incorporated with solar heating systems, the location of the analysis
          should be one which has both a heating and cooling requirement.
          Thirdly, present costs of conventional fuels in the New York area
          make the use of alternative heating and cooling systems very attrac-
          tive. Finally, at the time of the need to pick a design location,
          the eventual installation locations were not yet selected.

4.10.9.3 It is recognized that performance data obtained from the
          New York study will not be directly applicable to the eventual
          installation locations. However, this data will give insight into
          which system has the potential to save the most energy, and which
          will be the most cost effective. This data will also give insight
          into characteristic performance of the various systems proposed.

4.10.9.4 The building chosen for the modeling study was one on which
          Solaron had considerable data. As was the case with the location,
          the building to be modeled was chosen prior to actual site selec-
          tion. As a result, a representative building using modern con-
          struction techniques was chosen from Solaron's files. Drawings of
          this building are enclosed (see Figures B-1 - B-5).

4.10.9.5 The modeling method employed for this building is based on the
          use of transfer functions. This model is based on the Laplace
          transformation of the governing equations, as described in Chapter
          22 of the ASHRAE Handbook of Fundamentals. The transfer function
          technique is particularly compatable with computer simulation
          studies and load calculations performed with it have been shown to
          be quite accurate. Transfer function models run much more quickly
          and inexpensively than other techniques of load modelling such as
          the finite difference method.

4.10.9.6 The building model calculates the hourly heating or cooling load on the building as a function of the hourly changes in the weather. Included in the model are the thermal response characteristics of the walls and roof, the solar heat gains, internal sensible and latent heat gains and infiltration loads, both sensible and latent.

4.11   Status of the Solaron-University of Wisconsin study on the Solar-Desiccant systems.

4.11.1 ·   The initial cycles to be investigated by the University of Wisconsin are shown in Figures W-1 through W-9. A brief description of each cycle is given below.

W-1 - This figure is a schematic of a Solaron heating system integrated with any of the proposed desiccant air conditioning systems. The basic modes of operation of the system allow the desiccant system to be regenerated directly from the collector or from the stored heat in the pebble bed. The system can also store heat from the collector in the pebble bed, for periods when there is solar collector heat available and there is no simultaneous need for cooling.

W-2, 3 - Vent Cycle - This proposed cycle is called a vent cycle, due to its use of 100% fresh air. Referring to the system flow diagram and psychometric chart, ambient air at state 1 is drawn through a desiccant wheel, where it is dryed and heated to state 2. The air then is cooled by a sensible heat exchanger to state 3 followed by evaporative cooling to state 4, where it enters the room. Return air from the room (state 5) is first evaporatively cooled to 6 and then is heated by the sensible heat exchange process to state 7. At this point it is further heated by the solar system (either by the collector or storage, see W-1) and an auxiliary heater if needed, to state 8. This hot air then regenerates the desiccant wheel and is discharged to the atmosphere.

W-4, 5 - Recirculation Cycle, option 1 - This is one of three proposed recirculation cycles. Recirculation in this case refers to the room air stream flowing in a closed loop, while a second open loop air stream is used for regeneration. Referring to the room air loop first, return air at 6 is drawn through a desiccant wheel where it is dryed and heated to 7; the hot dry air at state 7 is then sensibly cooled to state 8 and evaporatively cooled to state 9 where it re-enters the rooms. The regeneration cycle consists of ambient air at state 1 which is evaporatively cooled (state 2) and then sensibly heated (state 3). Air at state point 3 is then heated by the solar system and/or auxiliary (if required) to state 4; this air then is used to regenerate the desiccant wheel and is then discharged.

W-6, 7 - Recirculation Cycle, option 2 - In this cycle a slight change is made to the system in option 1 in an attempt to increase the overall COP. Return air at state 5 is dryed to state 6 and sensibly cooled as before, however at state 7 the air is sensibly cooled further by an indirect evaporative cooler, which allows state 8 to closely approach the outdoor wet bulb temperature. Air at state 8 is then evaporatively cooled to 9 where it enters the room. Ambient air used in the regenerative loop actually takes two paths. A relatively large flow is used in the indirect evaporative cooler to provide cooling to the room air stream, while an additional amount of air equal in flow to the room air rate is directed into the sensible heat wheel. This air is heated to the same temperature at state 2 as was the air at state 3 of option 1, but since it has not been evaporatively cooled in this cycle it is at a lower humidity. This allows the air at state 3 of this cycle to be at a lower temperature and still achieve the required regeneration of the wheel.

W-8, 9 - Recirculation Cycle, option 3 - This cycle is exactly the same as option 1 on the room side, but is significantly different on the regeneration side. Specifically, ambient air enters at two different points in the regeneration process. One ambient air stream is evaporatively cooled from state 1 to 2 and then heated from 2 to 3 by the first sensible heat exchanger. Ambient air at state 1 also enters a second sensible heat exchanger and is heated to state 4; this air is then further heated by the solar system and/or auxiliary to state 5. This air then regenerates the wheel and leaves at state 6 where it enters the second sensible heat exchanger, is cooled and is discharged at state 7.

4.11.2   The cycles described here are all being integrated with the Solaron space and domestic water heating systems in a computer study at Wisconsin. In these studies, system parameters such as collector size, rock storage volume and building load characteristics will be held constant. Parameters of the desiccant systems, notably the performance of the sensible heat exchangers, will be varied to determine the best trade-off of heat exchanger performance versus solar collector area. All the cycles will be intercompared at constant collector areas to determine the most thermally efficient desiccant cycle.

4.11.3   Initial simulations have been made on the vent cycle system integrated with the Solaron system for the New York City location. Results although very preliminary, indicate that the system being sized for an approximate 70% annual heating savings, results in a 80% annual cooling savings. These results also indicate a very low desiccant system COP (on the order of 0.2). It is felt that these low COP's are the result of an inappropriate control sequence, causing the system to "over dry" hence reducing COP. Work is underway to improve the control sequence of the system. Simulation of the first option of the recirculation cycle is also under way. Performance Data on this system will be available soon.

4.12   Status of the Solaron-Carrier Study and Analysis of the
          Solaron-Heat Pump system.

4.12.1   The combination of solar and heat pump systems can be accomp-
          lished in many ways. Three possible systems which appear to be
          most promising as shown schematically in Figures C-1, C-2 and C-3.

          In Figure C-1, the parallel system, the heat pump replaces the
          normal furnace in the Solaron system. The heat pump operates as
          the back up heating source, and when called upon to deliver heat,
          operates in a conventional fashion, drawing heat from the outside
          air. In this system solar heat is not ever used as a source for the
          evaporator section of the heat pump. Also, under conditions where
          the heat pump is heating the rooms, house return air is fed directly
          to the inlet of the heat pump via the bypass duct and damper MD-3.
          This is done to prevent the compressor from running when the inlet
          air temperature to the indoor section is above room air temperature.
          If warm air from the solar system is fed to the heat pump indoor
          section while it's compressor is running, pressures in the freon
          loop tend to exceed design limits, and consequently the compressor
          loop would shut down. A brief sequence of operation for this
          system is given in Figure C-4.

4.12.2   Figure C-2, the dual source system allows the combined solar
          heat pump system to be operated in either of three basic modes.
          Solar heat from the collector or storage can be used to directly
          heat the space, or, solar heat can be fed to the evaporative sec-
          tion of the heat pump, presumably allowing the heat pump to operate
          at a high COP as it heats the rooms, or, finally the heat pump can
          supply heat in a conventional fashion using outside air as the
          source. When the heat pump operates in a solar boosted mode,
          dampers MD-2 and MD-6 will be modulated to prevent air warmer than
          70°F from entering the evaporator section. This is necessary to
          prevent the freon loop from experiencing excessive pressures. A
          mode of operation table for this system is shown on Figure C-5.

4.12.3   A system design to utilize off-peak electricity is shown in
          Figure C-3. This system is exactly like the parallel system
          during the heating season. In the summer, however, the system has
          the ability to store off-peak powered cooling. In this mode,
          presumably at night, the heat pump would cool air passing through
          the outdoor section. This cool air is then passed into the top of
          the pebble bed through damper MD-5. Air from the bottom of the
          rock bed then flows up through collector bypass MD-6, through the
          solar air handler and back to the heat pump. The condensing section
          dumps the heat into the outdoor air. This cycle of cooling the
          storage will continue until the storage is entirely cooled or until
          the daytime peak on electric use approaches. Then, when the rooms
          require cooling, air is drawn through the pebble bed (now cold) and
          supplied to the space. This system has two potential advantages.

First, a cost savings can be realized if the customer's electric utility includes an off-peak rate structure. Secondly there is a slight energy savings realized by running the heat pump only at night, when outdoor air temperatures are the coolest and hence heat pump operation the most efficient. Modes of operation of this system are covered in Figure C-6.

4.12.4   Analysis of the first of the three Solaron-heat pump systems, the parallel system, is underway. Preliminary results on this system are expected in mid-May.

5.0   Other pertinent Items/Events

5.1   Solaron Corporation moved its corporate and operating head-
          quarters from 4850 Olive Street, Commerce City, Colorado to the
          Galleria Tower on South Colorado Boulevard. The move was made in
          March 4-5, 1977. The new mailing address/Phone are:

                         300 Galleria Tower
                    720 South Colorado Boulevard
                       Denver, Colorado 80222
                       Phone: (303) 759-0101

5.2   Seminars - Two seminars were hosted by Solaron to gain expert
          information and advice from people who had special knowledge and
          experience in fields that relate to solar powered desiccant cooling
          systems. NASA personnel were invited to attend and participate in
          these seminars.

5.2.1   The first seminar was held in Denver, Colorado on 4 March 1977.
          The speaker was Mr. Bill Rush of the Institute of Gas Technology
          (IGT). Mr. Rush gave a slide illustrated lecture on the work the
          he and his colleagues have been doing over a period of years to
          develop a gas fired total air conditioning system (heating, cooling,
          and humidity control). Their system/unit, called the Munters En-
          vironmental Control (MEC) is licensed under Munters patents. The
          MEC system/unit is adaptable to solar assisted heating and cooling,
          and IGT is now working on this aspect. There was a thorough dis-
          cussion of the MEC system following Mr. Rush's lecture. The seminar
          was attended by Dr. Löf and all R & D engineers of Solaron, and
          by Mr. Cash representing NASA.

5.2.2   The second seminar was held in Denver at Solaron's new corporate
          headquarters on 22 March 1977. The guest specialists were Mr. Al
          Newton, a private consultant who also sits on one or more ASHRAE
          Technical Committies, and Mr. John Mitchell, who is in direct charge
          of Solaron's subcontract for computer modeling and simulation
          work at the University of Wisconsin. The seminar was primarily
          a discussion meeting on the subject of desiccant cooling systems
          and the adaptability of these to solar air heating systems. Mr.
          Mitchell reported on the progress of the computer modeling and
          simulation work being done at the University of Wisconsin. The
          other attendees at this seminar were Dr. Löf and all R & D engineers
          of Solaron, and Mr. Cash and Mr. Clark representing NASA.

6.0   Attachments -

The following listed attachments are a part of and supplimentary to Part IV of the foregoing Quarterly Report.

QUARTERLY REPORT

1 MAY 1977 - 30 JUNE 1977

NASA CONTRACT NO. NAS8-32249

CONTRACT TITLE:

DEVELOPMENT, DELIVERY AND SUPPORT
OF TWO (2) PROTOTYPE SOLAR HEATING
AND COOLING SYSTEMS, INCLUDING
POTABLE HOT WATER

PREPARED BY: Don Bloomquist and Rodney L. Oonk

APPROVED BY: L.E. Shaw

Part I - Summary

1.0 This report will summarize and bring up-to-date the reporting on this contract for the period from 1 May 1977 through 30 June 1977. The following information, as applicable, is included: Contract changes, Current and Cumulative Costs, schedules, and technical performance.

Part II - Contract

2.0 <u>Contract Changes</u> - There have been no contract changes during the reporting period.

2.1 <u>Cost</u>

Part III - Schedules

3.0 The schedule of Contract milestones has been reviewed and negotiated with the Technical Manager, Mr. Mitchell Cash, and is as follows: (See following page)

Development Plan

SHC-3014

Paragraph 2.0 - Schedule

2.0 The schedule below is established to meet the Contract require-
ments. A variation of minus one week or plus two weeks from the
below schedule is authorized upon agreement with the Technial
Manager.

Event	1st System	2nd System
Design Approach Review	2/17/77	2/17/77
Quarterly Review	2/17/77	2/17/77
Quarterly Review	6/27/77	6/27/77
Preliminary Design Review	10/5/77	10/5/77
Quarterly Review	10/5/77	10/5/77
Submit Prelim. Instrumentation Plan	10/5/77	10/5/77
Prototype Design Review	11/9/77	11/9/77
Instrumentation Required	11/30/77	1/15/78
First Article Review	2/15/78	?
Quarterly Review	2/15/78	2/15/78
Deliver/Prototype System	2/28/78	2/28/78
Installation Review	4/27/78	4/28/78
Quarterly Review	6/15/78	6/15/78
Operation Test Review	4/25/79	4/25/79

3.1 The Solaron Gantt chart schedule has been revised to be consistent
with the above schedule of Contract milestones, and is included as
Attachment I to this report.

Part IV - Technical Performance

4.0 The technical performance under this Contract is reported under separate headings of (4.1) Desiccant System - Solaron Engineering Design, (4.2) Heat Pump Systems - Carrier Subcontract, and (4.3) Desiccant Systems - University of Wisconsin Sub Contract. The content is essentially a composite and summary of the technical performance as reported in the May and June, 1977 monthly reports.

4.1 Desiccant System - Solaron Engineering Design

4.1.1 Refinement of the desiccant system cooling cycles proceeded during the reporting period. The direction of this effort was influenced by the first firm site selection of Akron, Ohio and by the data coming out of the University of Wisconsin computer simulation program. This simulation program is based on a house in New York, N.Y. because of the need for hourly weather data in the structure of the programs. New York is one of the few locations for which this detailed data is available, and its use in the computer simulation program was previously approved by NASA.

The climatic conditions for Akron and New York are reasonably comparable, with both having a predominant heating load.

4.1.2 The Wisconsin data was presented in report form at the Technical Review/Quarterly Review Meeting (TR/QRM) held at Solaron/Denver on 27 June 1977, and is included as Attachment II to this report under the title of "Simulation of Four Open Cycle Desiccant Cooling Systems". The original report as received from Wisconsin has been slightly amplified by the addition of psychrometric cycle diagrams (Fig's. 3, 5, 7 & 9) and a block schematic diagram showing the integration of the Recirculation (Option 1) Cooling cycle with the standard Solaron heating system (Fig. 18).

4.1.3 Based on the data presented in Para. 4.1.2 it has been determined that the optimum system cycle for the Akron, Ohio site is the Recirculation cycle, option 1. (Fig's. 4, 5 & 18 of Attachment II). Comparison of Figures 13, 15 & 17 show better performance and efficiency for options 2 & 3, but it is felt the increased capital cost for these options is not justified. This was discussed in the TR/QRM of 27 June 1977, and it was agreed to proceed on that basis.

4.1.4   In the Monthly report for May 1977 and subsequently in the TR/QRM of 27 June 1977 it was pointed out that the baseline atmospheric design conditions of 95° FDB & 60% RH were not realistic for a site in Akron, Ohio, and that the size and cost of the necessary hardware would be inordinately high if these specifications were rigidly enforced. Solaron recommended that design be based on 2½ percent summer design atmospheric conditions and the use of 24 hour design using the "flywheel" effect of the mass of house and interior furnishings as a heat sink. (These design parameters were subsequently and officially approved by NASA letter dated July 15, 1977)

Thus, system design of the first prototype system is being tailored for the first site in Akron, Ohio, for which the design conditions are as follows:

Latitude: 41° 0'

Elevation: 1210 ft. above sea level

Winter Design

   Median of annual extremes, °F: -5

      97½% design, °F   : +6

   Coincident wind velocity   : M

      (M = Moderate, 50 to 74 percent Cold extreme
       hours > 7 mph)

   Degree days: 6037

Summer Design: 2½%

   Design dry bulb, °F   87

   Design wet bulb, °F   73

   Outdoor daily range, °F   21

The design load on the Akron house is less than the anticipated 2-Ton load (approx. 21,600 Btu/Hr. or 1.8 Tons). For this reason, and also the interim results of the University of Wisconsin computer model, which indicate an optimum air flow rate of approximately 450 CFM/Ton, the flow rate for the Akron house has been reduced to 60 lb./min. (800 SCFM). The original design for a 2-Ton load was 80 lb./min. (1067 SCFM) at 533 CFM/Ton.

Solaron has prepared a data sheet for the Akron site/house which is attached as Attachment III.

4.1.5   The fresh air intake and exhaust system for the Solaron
laboratory, as required for sub-scale and prototype system testing,
is scheduled for "installation complete" by mid July. This system
consists of two wall penetrations with external weather hoods,
modulating damper control and intake and exhaust points, and three
(3) take-off positions each in the supply and exhaust headers.

4.1.6   The two (2) evaporative coolers ordered in May were received
in June and will be scheduled for sub-scale testing upon completion
of the fresh air supply and exhaust systems.

4.1.7   A contract Technical Review Meeting was held at Solaron,
Denver
on 27 June 1977 with attendees as follows:

Name	Representing
Mitchell Cash	NASA
Ralph Murphy	NASA
Dr. George O.G. Lof	Solaron
Les Shaw	Solaron
Rod Oonk	Solaron
Don Bloomquist	Solaron

Solaron reviewed the status and progress on the contract and NASA
presented the status of the Akron site. NASA also scheduled a one
day on-site meeting at Akron for 20 July 1977. A "work book" for
the meeting, was given to each participant of the meeting.

4.2 Heat Pump Systems - Carrier Subcontract

It was reported in the May report and subsequently discussed in the
TR/QRM of 27 June 1977 that preliminary analysis of the off-peak
solar heat pump cooling system has led to a concern about cooling
the rocks in the pebble bed. Experience with moisture from the
room air condensing on heat exchangers at Carrier has presented a
real danger of odor problems. With cold rocks in the pebble bed
and warm moist return air entering the bed, the same odor problem
may well occur. Consequently, consideration is now being given to
an off-peak cooling system using water as the cold storage media
and using an air-to-water heat exchanger to cold charge and to
extract cooling from the water storage. The pebble bed would then
only be used as a heat storage device, a function that has been
free of odor problems.

As was reported in the June report, Carrier is continuing effort in modelling of three basic solar - heat pump concepts: (1) Solar and heat pump used independently of each other in a parallel fashion (solar system with a heat pump back up); (2) solar and heat pump combined in a dual source fashion, allowing the heat pump to extract heat from the solar system or outdoors; (3) combining the solar system and heat pump as in (1) but further adding the capability to the system to store off-peak cooling in a water tank, and thereby avoiding running the heat pump during the peak cooling periods of the day.

Preliminary performance results were obtained for the parallel system for two different heat pump sizes and a constant size solar system. The results indicated that the extra energy savings of the larger capacity heat pump only produced an $11 per year reduction in electricity cost at 4¢ per kilowatt hour. With this information it was decided to not use the larger pump in future simulations.

Carrier expects to be finished with their simulation by July 31, 1977.

4.3 Desiccant Systems - University of Wisconsin Subcontract

4.3.1   As was reported in the May report, simulations of the ventilation cycle and option 1 and option 3 of the recirculation cycles were performed by the University of Wisconsin. Simulations were performed at collector areas ranging from 40 m^2 to 135 m^2. These collector areas would yield a heating energy savings ranging from 40 to 70% in the New York Climate. The corresponding cooling savings for the New York climate ranged from 85-95%. The simulations also indicate that excessively large collector areas carried less of the cooling load than some smaller areas. This unexpected result was explained by observing that since the system flow rate was proportional to the collector area large collector areas meant excessively large flow rates for the New York house. This meant that the flow rate per ton of cooling delivered increased substantially, which in turn reduced the ability to exchange heat wthin the cycle. This reduced heat exchange forced the systems to use more auxiliary heat at extremely high collector areas.

With this information it was apparent that system flow rate should be held more or less constant and allow instead the collector flow rate per unit area to vary. Simulations of the change are underway and should be available shortly.

4.3.2   As was reported in the June report, simulations of the four different solar-desiccant cycles at four different collector areas and a constant system flow rate were completed while additional simulations were performed with system flow rate being varied while collector area was held constant in order to study the effect on overall performance. The results showed that the recirculation cycles performed better than the ventilation cycle for the New York climate, with the more expensive options 2 and 3 of the recirculation cycles doing a bit better than option 1. The study also showed that for New York, solar systems design for 50% or better heating energy savings, carried nearly all (96% - 98%) of the cooling load, regardless of the desiccant cycle type. From these results it was decided that for Akron, the simplest recirculation cycle would be the best choice, due to the Akron climate also being predominately a heating requirement. The results of these simulations are presented in report form in attachment II, as previously mention in para. 4.1.2.

Appendix A

This page deleted

Figure R-1
CLOSED (RECIRC.)
CYCLE
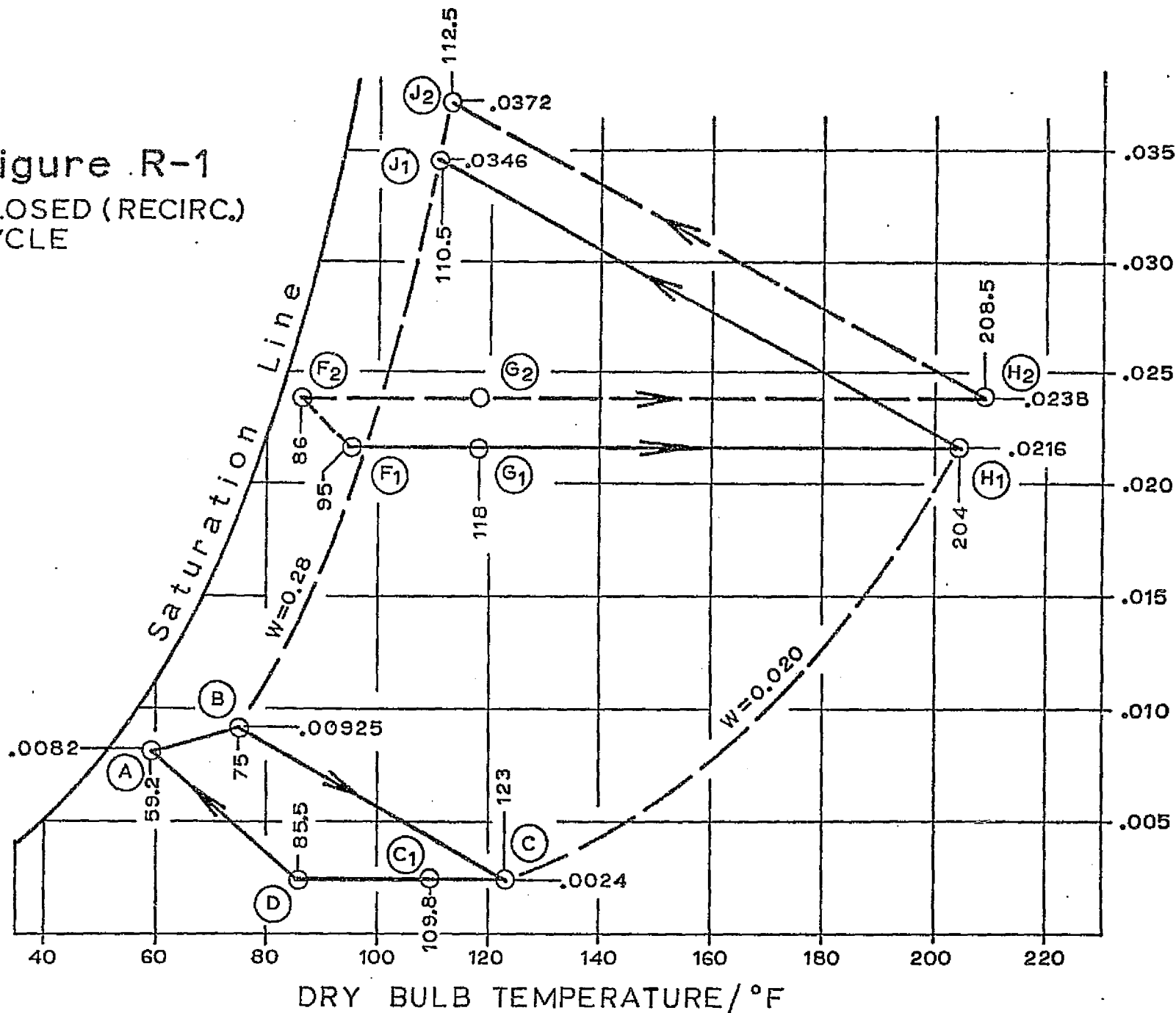
CONDITIONED SPACE
COOLING LOAD – 24,000 BTU/HR (SPECIFIED)
INSIDE DESIGN – 75°F DB – 50% RH
AIR FLOW DESIGN 80 LBS./MIN
(1067 SCFM)

INDIRECT EVAP COOLER

AMBIENT AIR

EXHAUST

EVAP COOLER

SUPPLY STREAM
REGEN. STREAM

ENERGY SUPPLY SOLAR OR AUX

SENSIBLE HEAT EXCHANGER

DEHUMIDIFIER

AMBIENT AIR
46 LBS/MIN
(613 SCFM)
DESIGN 95 F DB
60% RH
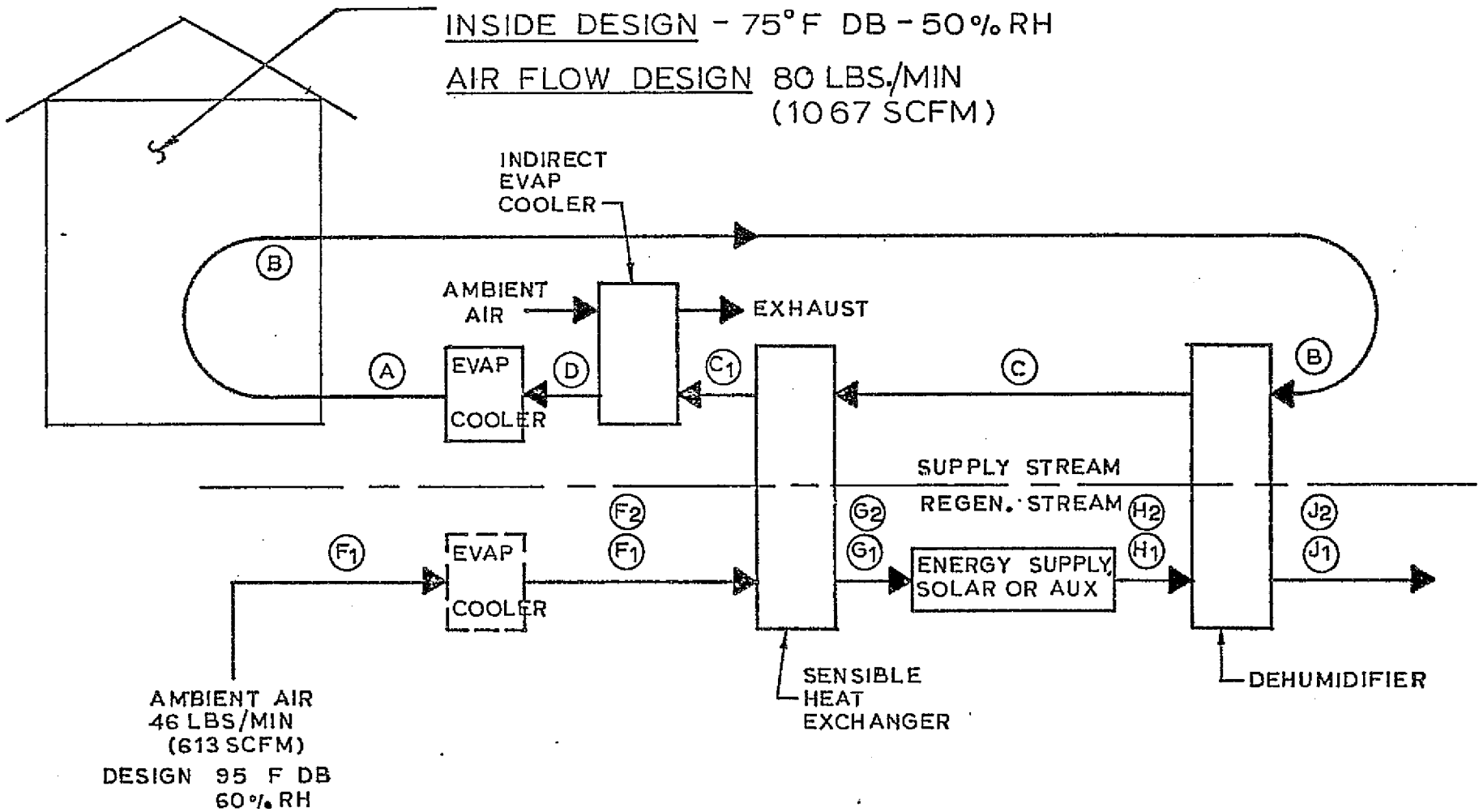
Figure R-2
CLOSED (RECIRC.) CYCLE
(See Fig R-1 For Psychrometric Analysis)

Figure V-1
OPEN (VENT) CYCLE

EVAP COOLER

⑥

ENERGY SUPPLY, SOLAR OR AUX

⑦   ⑧

SENSIBLE HEAT EXCHANGER

⑤

DEHUMIDIFIER

⑨

EVAP COOLER

④   ③   ②   ①

AMBIENT AIR
DESIGN — 95°F DB
60% RH

CONDITIONED SPACE
COOLING LOAD – 24,000 BTU/HR
(SPECIFIED)

INSIDE DESIGN – 75°F DB
50% RH

AIR FLOW DESIGN – 80 LBS./MIN
(1067 SCFM)
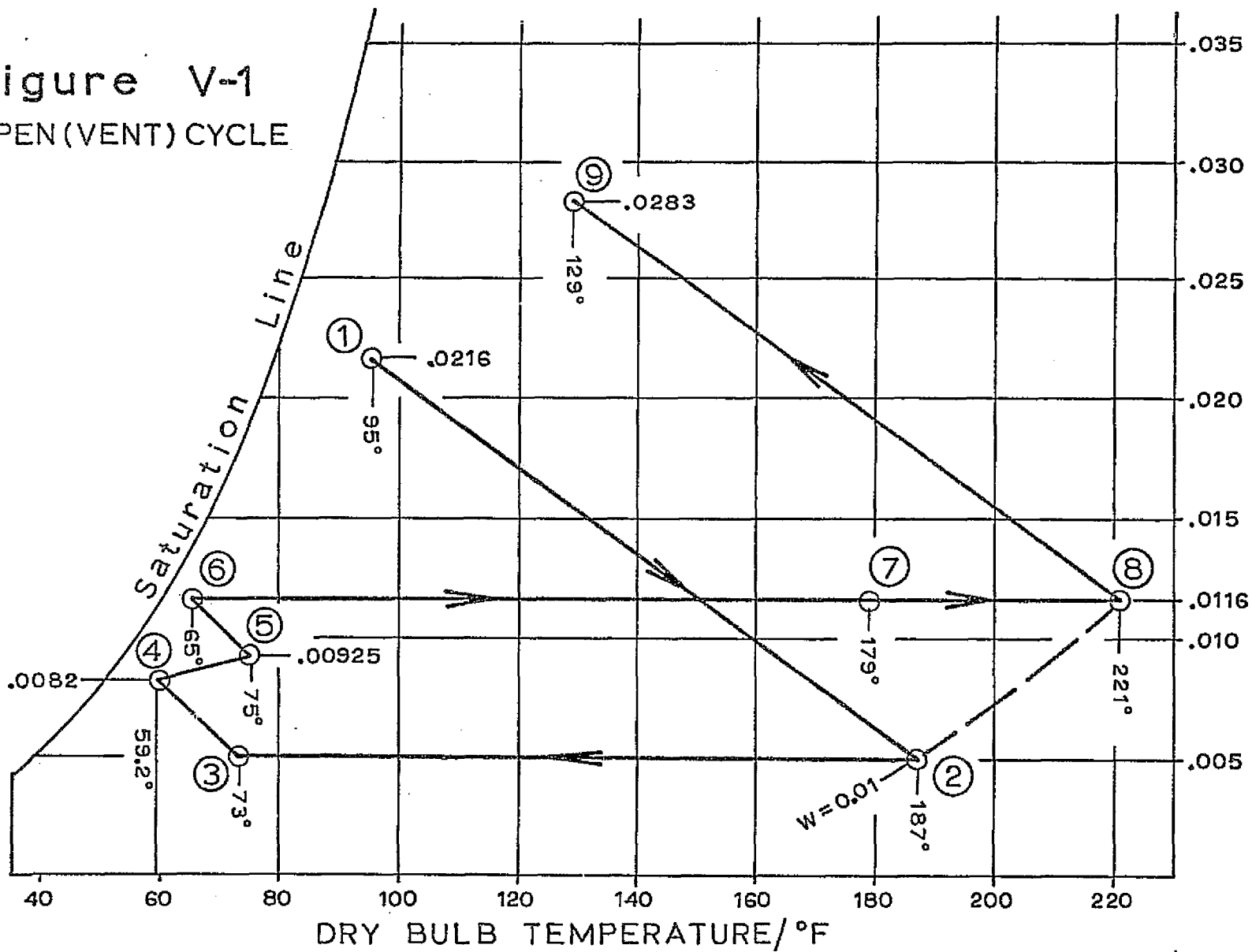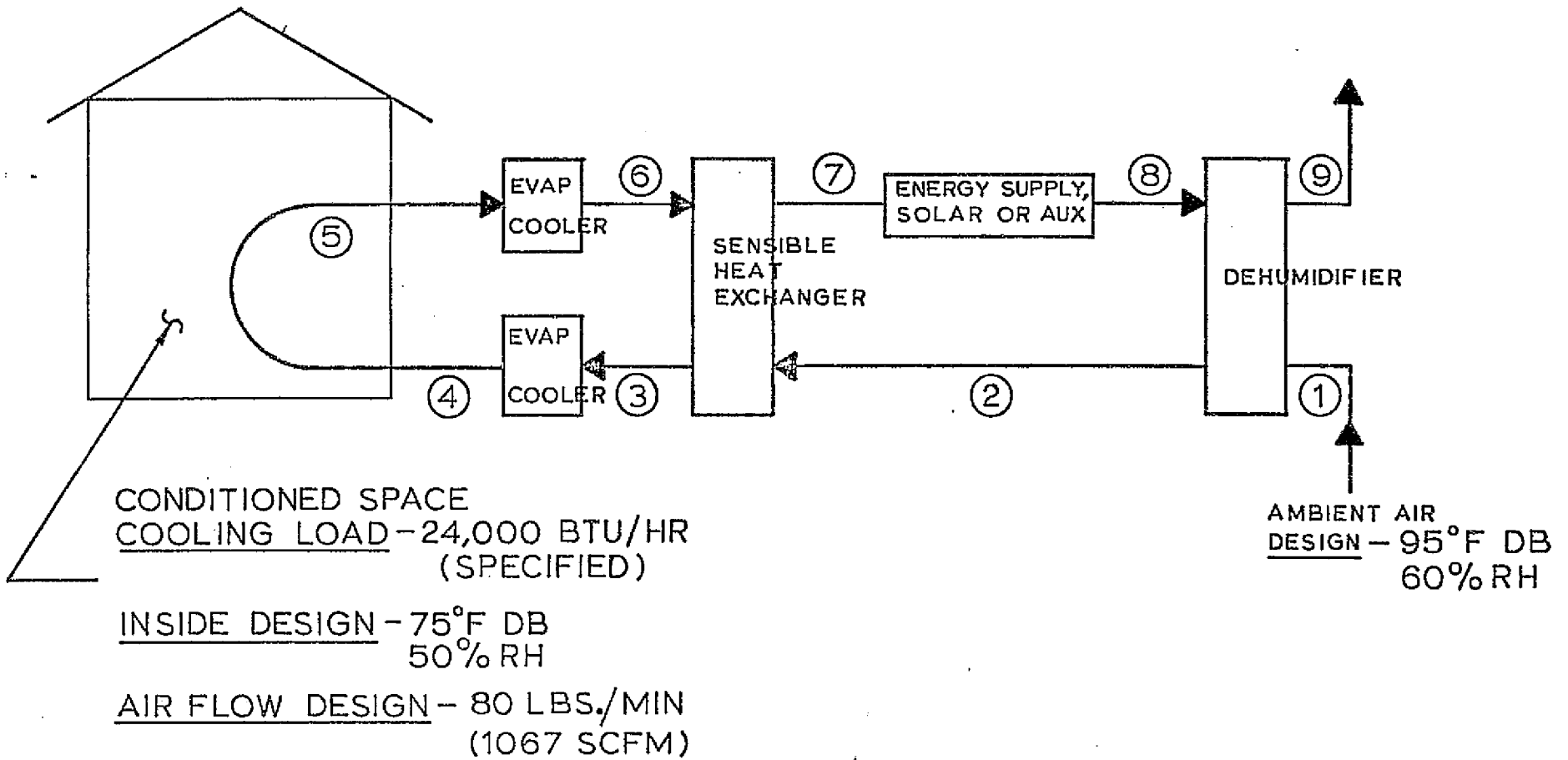
Figure V-2 OPEN (VENT) CYCLE

(See Fig V-1 For Psychrometric Analysis)

DESCRIPTIVE NARRATIVE FOR FIGURES R-1, R-2, V-1 & V-2

The following discussion is relative to the desiccant system only. Two options are under investigation to determine which will be the most effective in energy conservation. One is a closed (recirculating) system and the other an open (ventilating) system.

1.   A closed (recirculating) cycle in which the house air is continuously recirculated through the cycle without the addition of fresh air, and a separate air stream using 100% outside ambient air to regenerate the desiccant sub-system, and

2.   An open (ventilating) cycle in which 100% outside ambient air is used to condition the house, and in the same continuing stream to regenerate the desiccant subsystem.

These two options are discussed separately, and in detail as follows:

Option 1. - The closed (recirculating) cycle

This cycle is detailed psychrometrically in Fig. R-1 and a block schematic is shown in Fig. R-2. The state condition points are identical in each figure for ready cross reference.

The house air is maintained at a design condition of $75^{o}F$ dry bulb and 50% RH (equivalent to $62.5^{o}F$ wet bulb, $55^{o}F$ dew point, 0.00925 lb. moisture/ lb. dry air, or 64.75 grains/lb.).

House air at state condition (B) is delivered to a desiccant dehumidifier where it is dehumidified to a moisture level of .0024 $^{\#H_2O}/\#DA$ (16.8 gr./#, or $22.3^{o}F$ dew point) and simultaneously heated to approximately $123^{o}F$ dry bulb at state condition (C).

From the dehumidifier it flows through an air-to-air heat exchanger which reduces the temperature to approximately $109.8^{o}F$ at constant moisture level. Air leaves the exchanger at state point (C_1). The heat removed on the "hot" side is transferred to the inlet ambient air for the regeneration stream, heating this air from (F_1) to (G_1) on the "cold" side.

The supply air stream is further cooled from (C1) to (D), without change in moisture content, using an evaporative cooler with separate "wet" and "dry" sides. The heat is transferred to the wet side, which has a separate supply of outside ambient air entering at 95^OFDB and 60% RH (82.7^OFWB, 79^OFDP, .0216 #/#, or 151.2 gr./#). The "wet" side air is exhausted to atmosphere.

The "house" (supply) air is then cooled and simultaneously humidified from point (D) at 85.5^OFDB and .0024 #/# (55.0^OFWB, 22.3^OFDP, or 16.8 gr./#) to point (A) at 59.2^OFDB and .0082#/# (55.0 FWB, 51.8^OFDP, 57.4 gr/#, and 76%RH).

The air delivered to the house at point (A) is heated from (A) to (B) along the "room line" due to the house design load, thus completing the "House" (supply) air cycle.

The air for regeneration of the desiccant dehumidifier is outside ambient air entering the regenerating stream at design condition (F1) which is 95^OFDB & 60%RH (82.7^OFWB, 79^OFDP, .0216#/#, or 151.2 gr./#). This air is heated without change in moisture content from (F_1) to (G_1) by the heat exchanger, taking heat from the house (supply) air stream.(G_1) is at 118^OFDB & 87.7^OFWB.

The energy to heat the regeneration air from (G_1) to (H_1) will be supplied by the solar collectors and storage system to the maximum extent possible, and augmented by an auxiliary energy supply to make up any required difference. The required input to the desiccant regeneration system at (H_1) is estimated to be 204^OFDB at .0216#/#, (151.2 gr./#).

The regeneration process from (H_1) to (J_1) will be a characteristic of the selected dehumidifier. The estimated condition for the exhaust air (J_1) is 110.5^OFDB, 96^OFWB (92.5^OFDP, .0346#/#, 242.2 gr./#, or 60%RH).

A possibly viable alternate to the regeneration stream would be to evaporatively cool the air from (F_1) to F_2) to improve the heat transfer from the house (supply) air stream. This however would result in a higher moisture level and consequently higher required entering air temperature (H_2) for the regeneration process.

Option 2. The Open (ventilating) cycle

This cycle is detailed psychrometrically in Fig. V-1 and a block schematic is shown in Fig. V-2.

The inside house design condition is the same as for the closed cycle (75°FDB & 50%RH) and the air flow rate is the same (80 lb/min. or 1067SCFM). The essential difference is that in this case 100% outside air is used for the conditioning process and is exhausted to atmosphere after the cycle is completed.

Outside air at state condition (1) is delivered to a desiccant dehumidifier where it is dehumidified to a moisture level of .005$^{\#}$H$_2$O/#DA (35.0 gr./lb.) or 39°F dew point) and simultaneously heated to approximately 187°F dry bulb at state (2).

From the dehumidifier it flows through an air-to-air heat exchanger which reduces the temperature to approximately 73°F at constant moisture level to state (3). The heat removed on the "hot" side is transferred to the air for regenerating the dehumidifier, heating this air from 65°F to approximately 179°F, states (6) to (7).

The "house" (supply) air is then cooled and simultaneously humidified from point (3) at 73°FDB and .005$^{\#}$/# (54.8°FWB, 35 gr./lb. and 39° FDP) to state (4) at 59.2°FDB and .0082$^{\#}$/# (55.0°FWB, 51.8°FDP, 57.4 gr./#, and 76% RH).

The air delivered to the house at state (4) is heated from (4) to (5) along the "room line" due to the house load.

The "house" air is then delivered to an evaporative cooler which cools (and simultaneously humidifies) the air to state (6) at 65°FDB and .0116$^{\#}$/# (62.6°FWB, 81.2 gr./lb., 61°FDP).

Air from the evaporative cooler goes to the "cool" side of the air-to-air heat exchanger which heats it from 65°FDB to approximately 179°F without change in moisture content. This process is in the regeneration portion of the cycle.

The additional heat (energy) required for regeneration of the desiccant dehumidifier is supplied by the solar collectors and storage system to the maximum extent possible, and augmented by an auxiliary energy supply to make up any required difference. The required input to the regeneration system is at state (8) is estimated to be 221^OFDB at $.0116^{\#H_2O}/\#DA$.

The regeneration process from (8) to (9) will be a characteristic of the selected dehumidifier. The estimated condition for the exhaust air, (9), is 129^OFDB at $.0283^{\#H_2O}/\#DA$.

A - 11

FINISHED BASEMENT FLOOR PLAN   1/4" = 1'-0"

MAIN FLOOR PLAN   1/4" = 1'-0"

Figure B-2

REAR/SOUTH ELEVATION 1/8" = 1'-0"

RIGHT/WEST ELEVATION 1/8" = 1'-0"

LEFT/EAST ELEVATION 1/8" = 1'-0"

GARAGE ELEVATIONS - LOTS 6, 7 & 8 1/8" = 1'-0"

EAST - LOT 7 (MIRRORED FOR LOT 6)

WEST - LOT 8

NORTH - LOTS 7 & 8

EAST - LOT 6

SOUTH - LOT 6

SOUTH - LOTS 7 & 8

FRONT/NORTH ELEVATION 1/8" = 1'-0"

FOUNDATION PLAN 1/8" = 1'-0"

FOUNDATION WALL KEY

MODEL A-B
LOT 6

MODEL AR
LOT 7

MODEL B
LOT 8

A - 12

Figure B-3

Figure B-4

Figure B-4

FIREPLACE FAMILY/REC. RM.

FIREPLACE BEDROOM 1

OPTIONAL FIREPLACE ELEVATIONS

Figure B-5
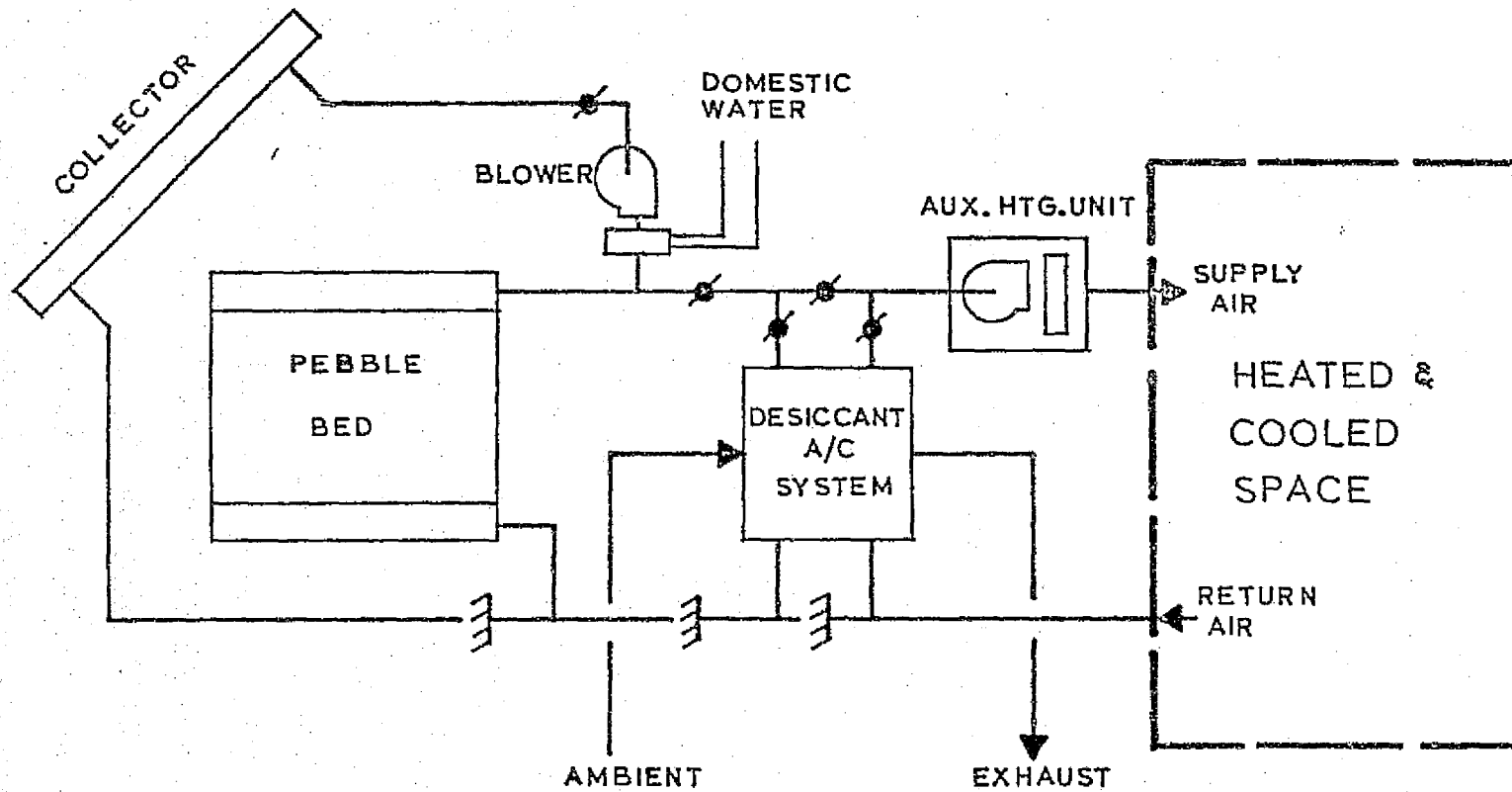
Figure W-1   SOLARON System Integrated With Desiccant Air Conditioning System

MODES: HEATING
          OPERATION — HEAT FROM COLLECTOR
                    — STORE HEAT
                    — HEAT FROM STORAGE

          COOLING
          OPERATION — REGENERATE FROM COLLECTOR
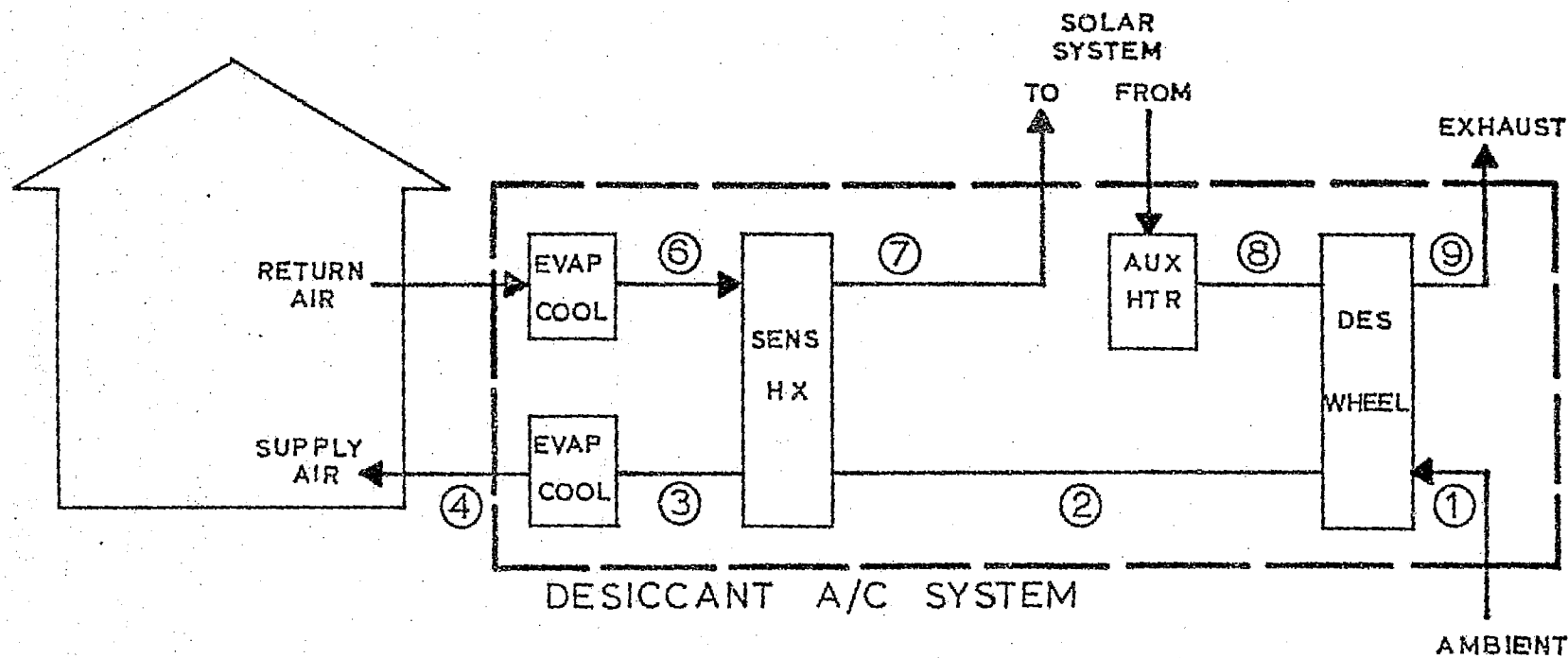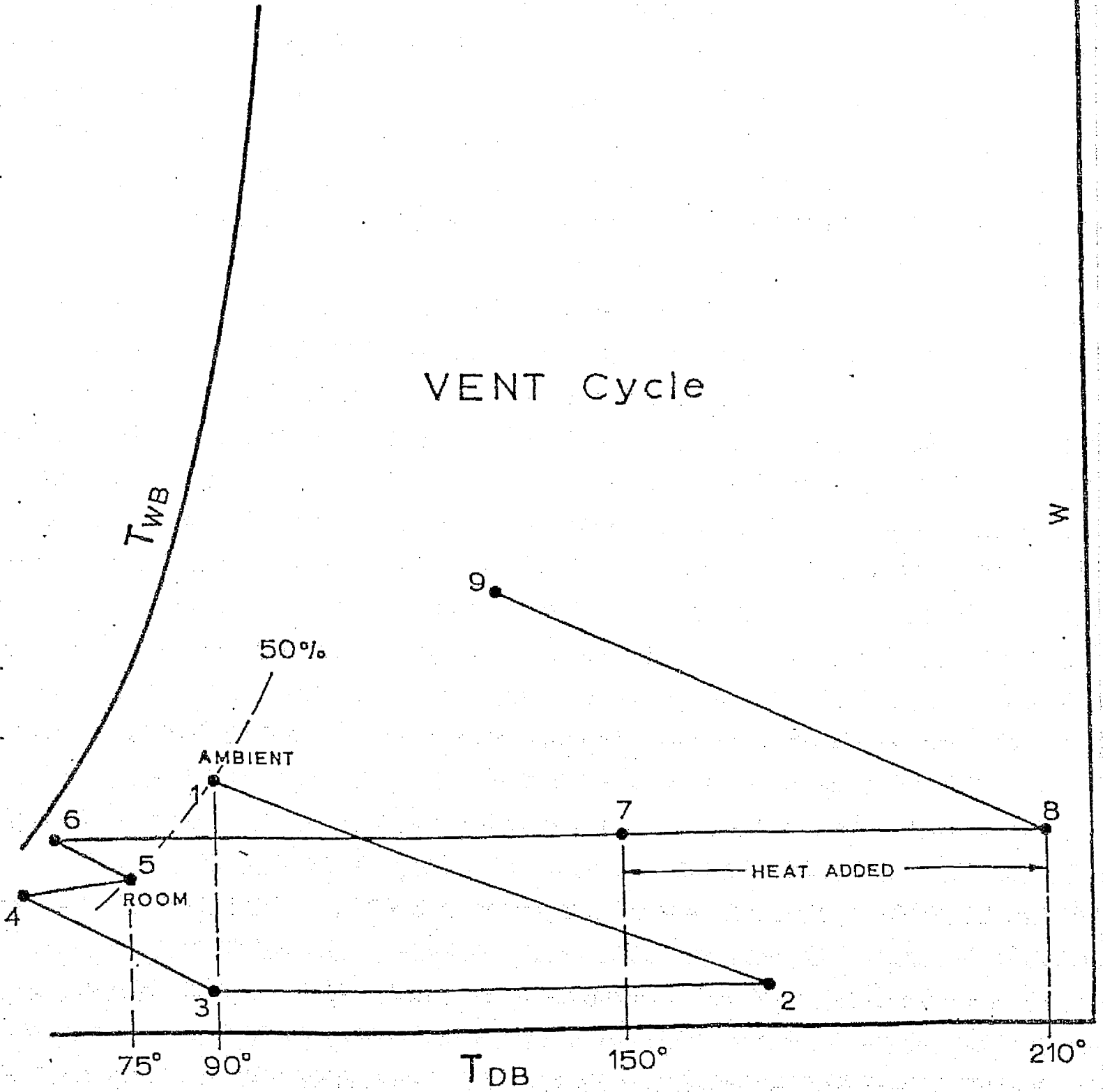   — STORE HEAT
                    — REGENERATE FROM STORAGE

Figure W-2 VENT Cycle

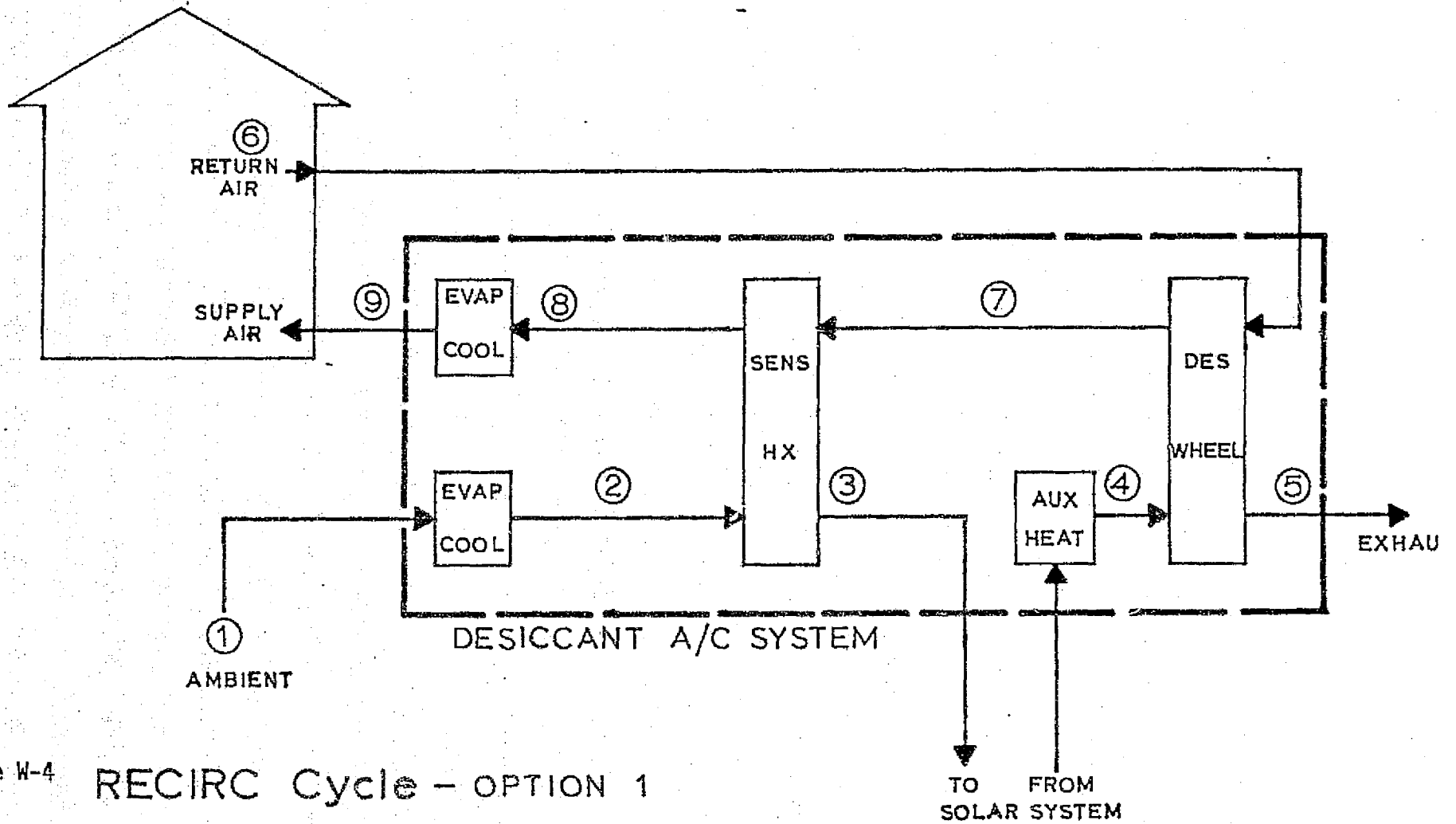SENSIBLE HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

VENT Cycle

Figure W-4   RECIRC Cycle – OPTION 1

SENSIBLE   HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

RECIRC Cycle
OPTION 1

T_{WB}

W

50%

2

3

4

5

AMBIENT

1

HEAT ADDED

ROOM

9

6

8

7

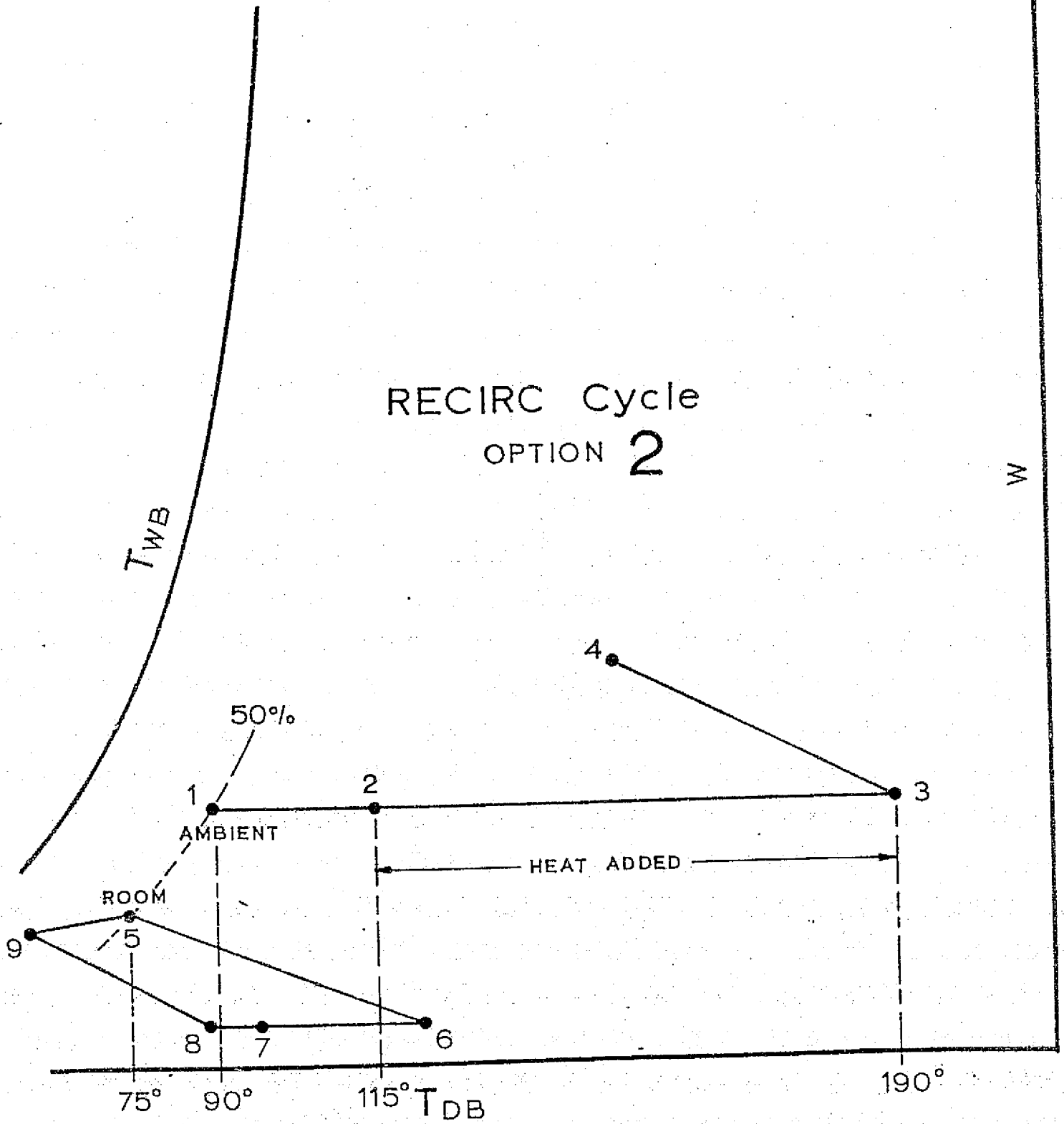75°   90°   115° T_{DB} 195°
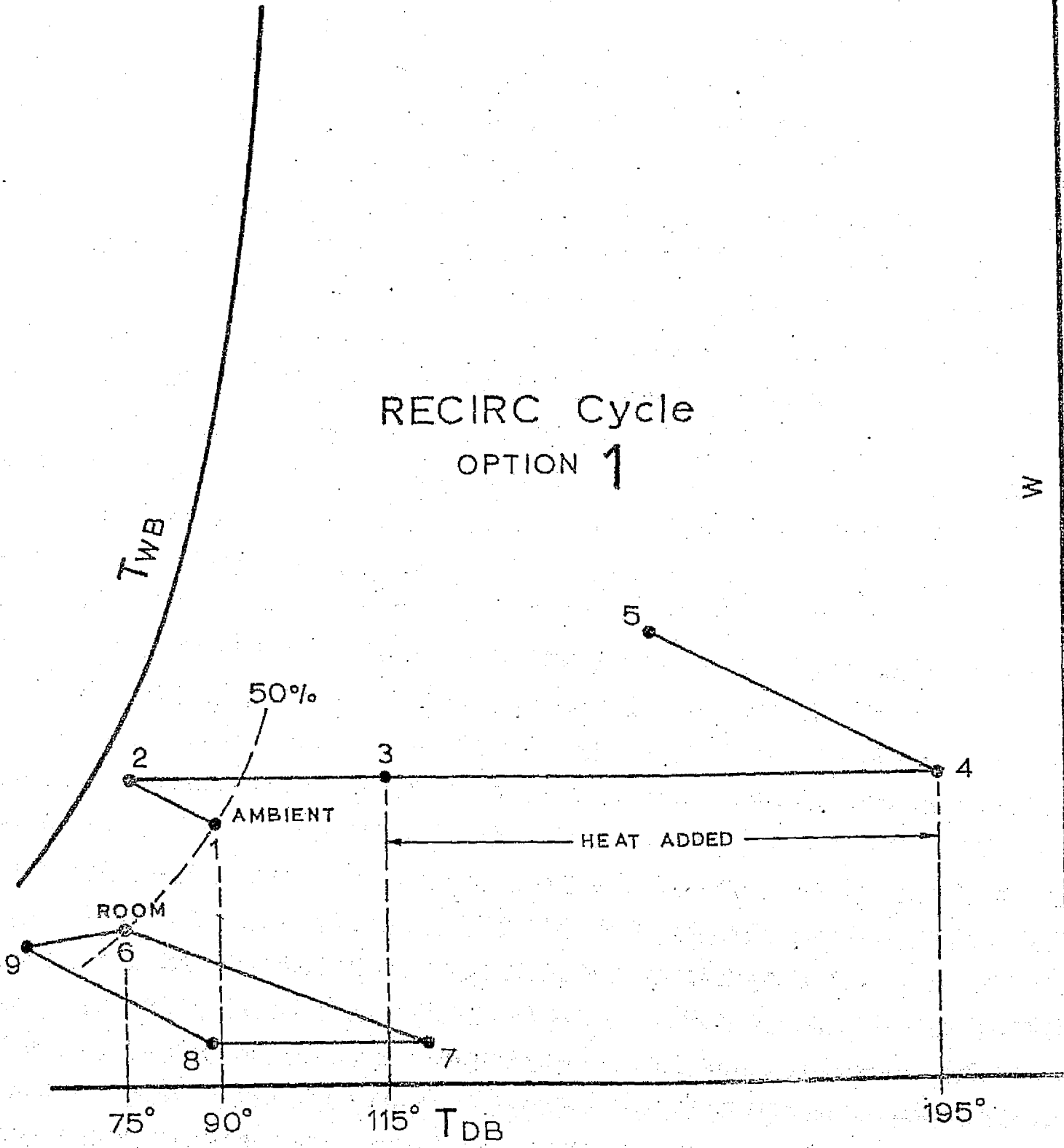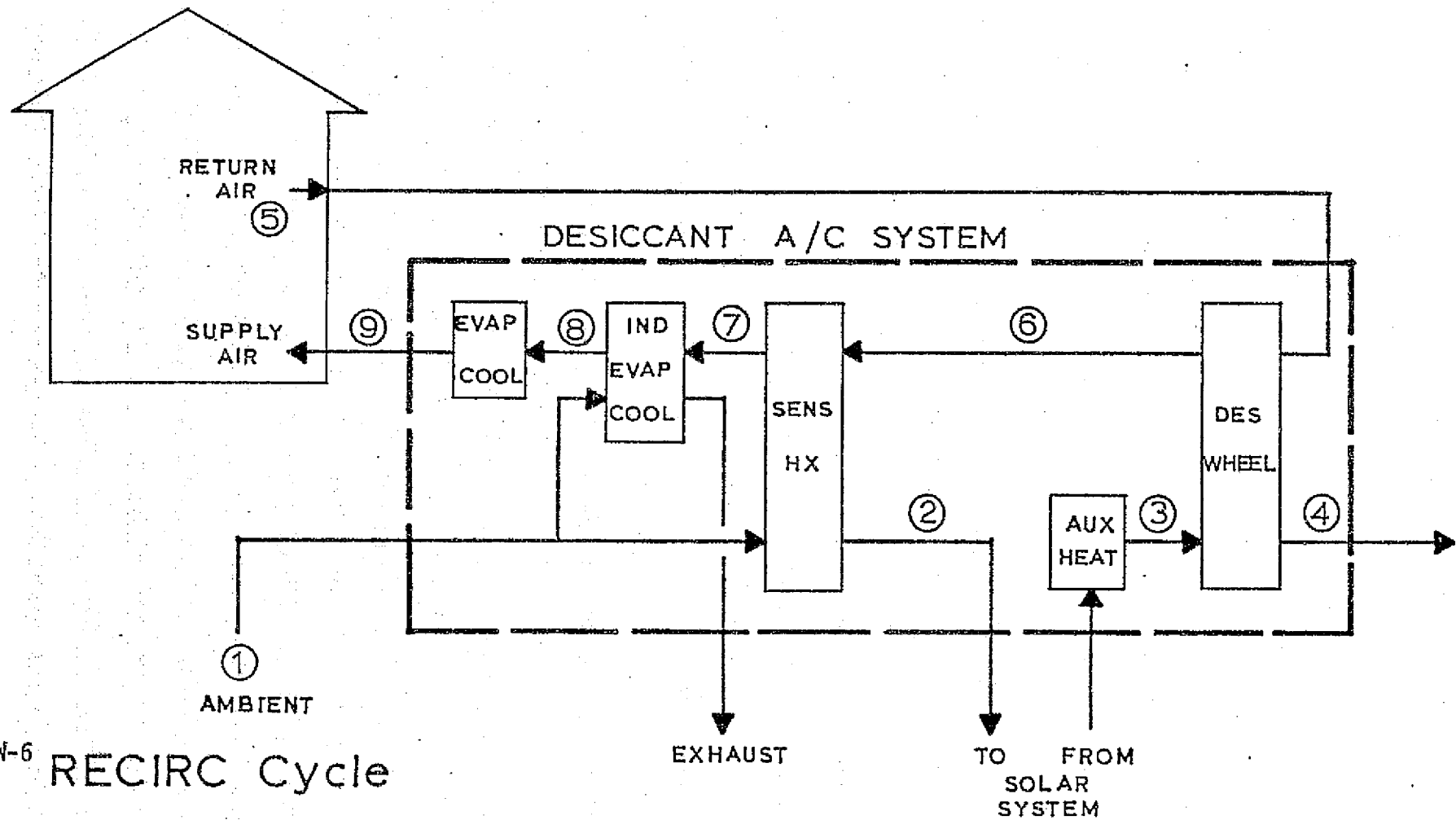
Figure W-6 RECIRC Cycle

OPTION 2

SENSIBLE HX $\varepsilon = 0.70$
$\varepsilon = 0.85$

INDIRECT
EVAP COOLER $\eta = \dfrac{T_{DB7} - T_{DB8}}{T_{DB7} - T_{WB1}}$ $= 0.63$
$= 0.86$
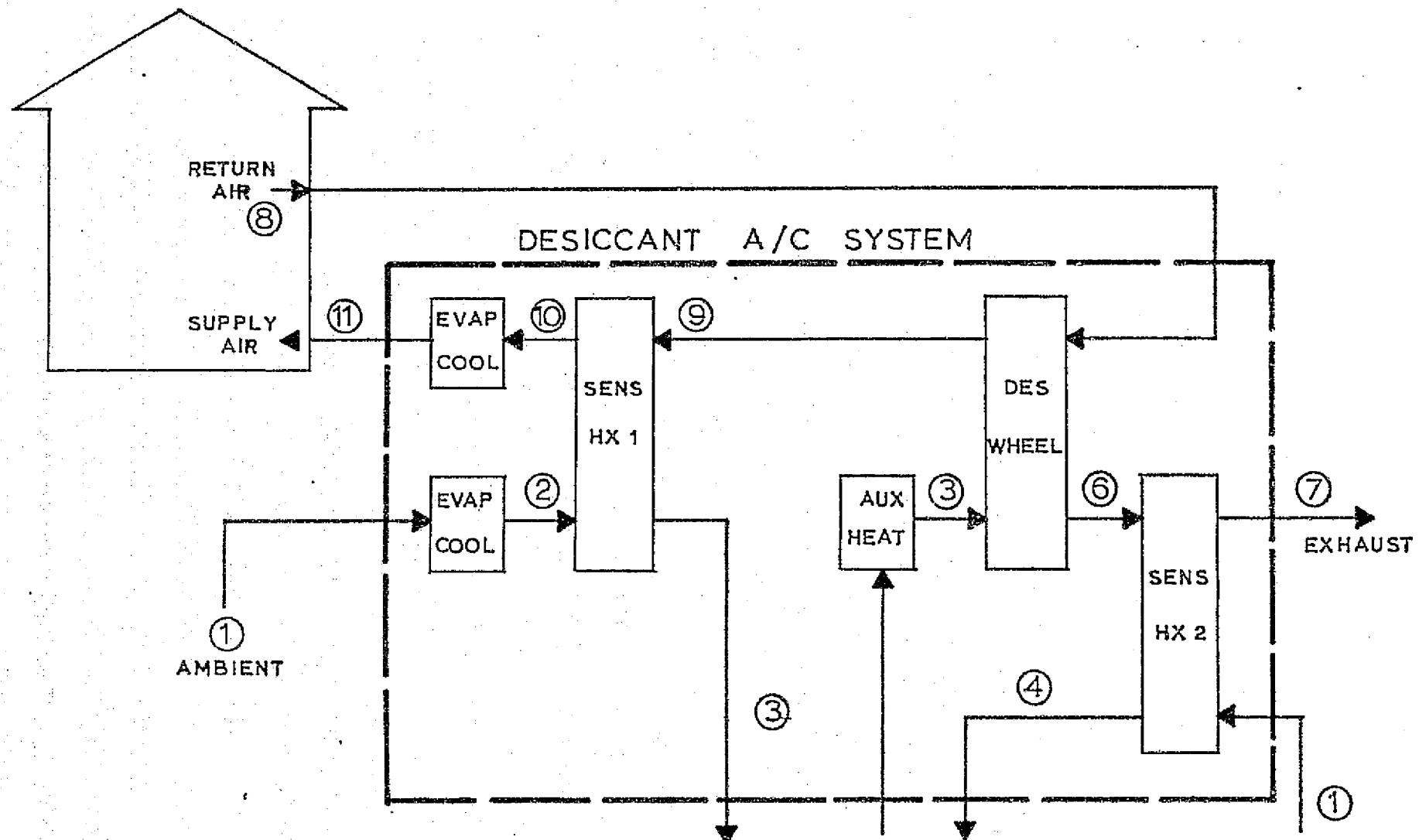$= 0.95$

RECIRC Cycle
OPTION 2
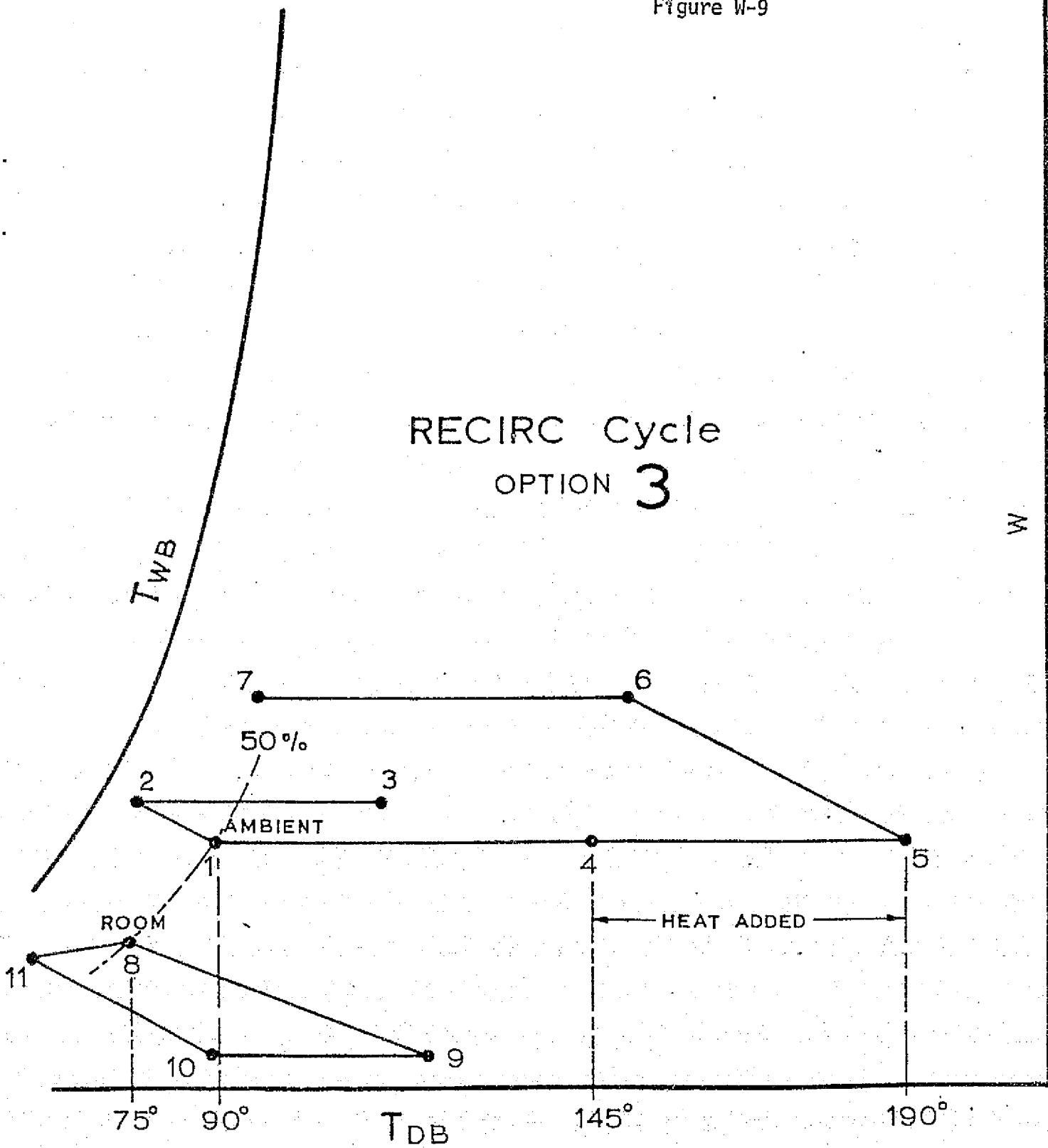
Figure W-8

RECIRC Cycle

OPTION 3

SENS HX 1 $\varepsilon = 0.70$
  $\varepsilon = 0.85$
HX 2 $\varepsilon = 0.70$
  $\varepsilon = 0.85$

A - 22

Figure W-9

RECIRC Cycle
OPTION 3
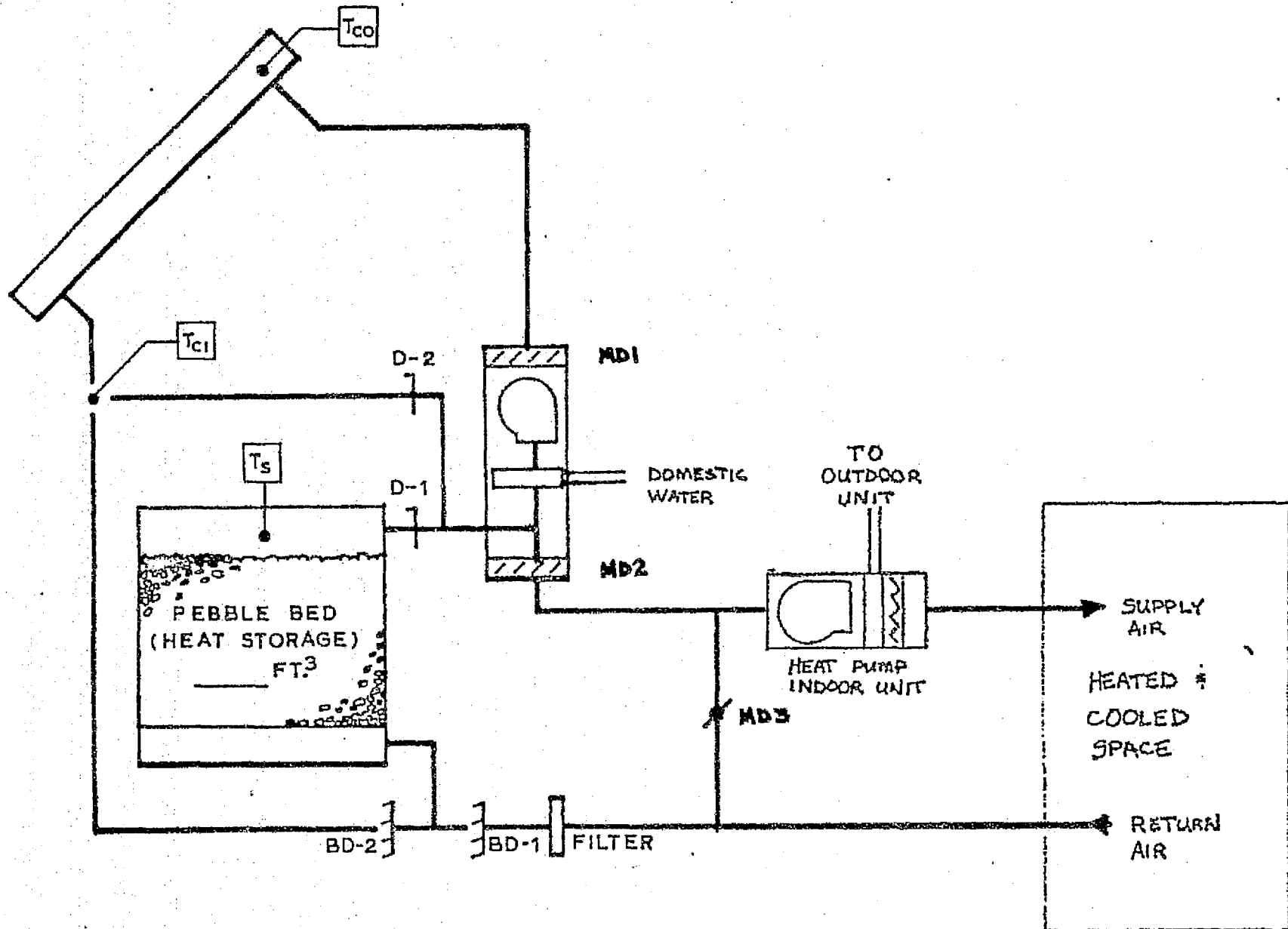
Figure C-1
PARALLEL SYSTEM
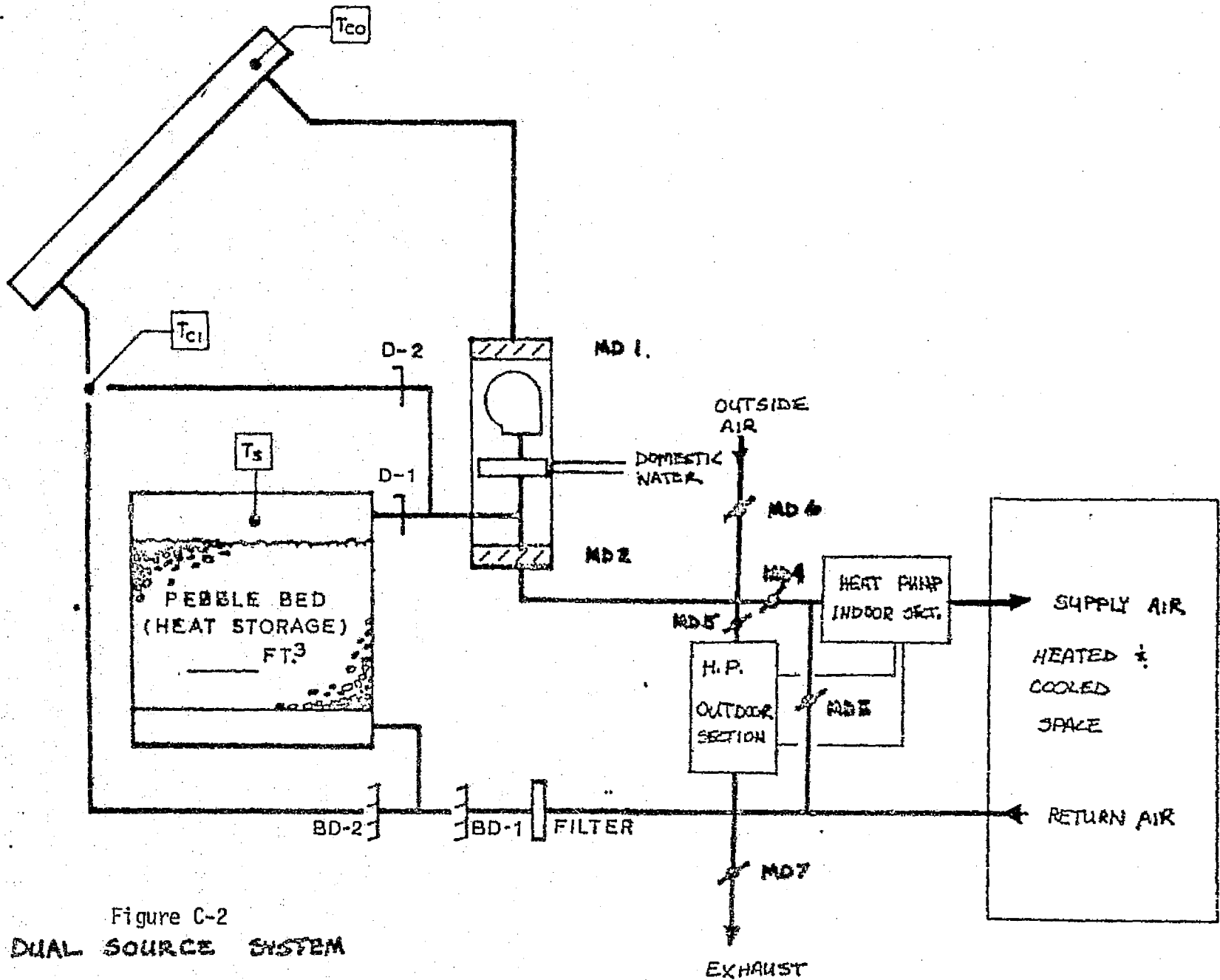
Figure C-2
DUAL SOURCE SYSTEM
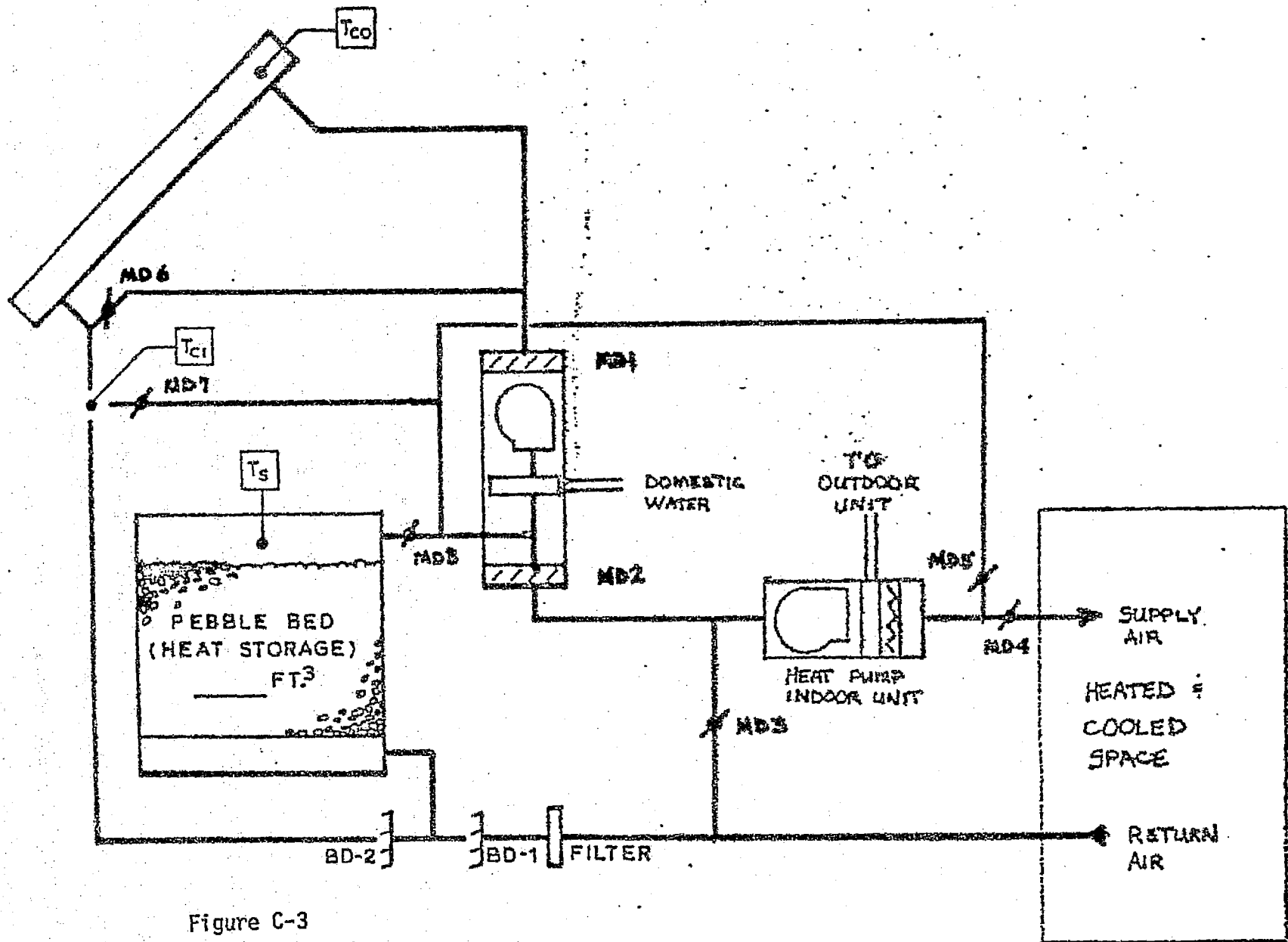
Figure C-3
OFF-PEAK SYSTEM

Figure C-4   PARALLEL SYSTEM

MODE	MD 1	MD 2	MD 3	SOLAR FAN	H.P. FAN	H.P. COMP	
Heat From Collector	0	0	C	1	1	0	
Heat From Storage	C	0	C	0	1	0	
Store Heat	0	C	0	1	0	0	
Heat From H.P. & St.Ht	0	C	0	1	1	1	
Cool & Store Heat	0	C	0	1	1	1	
Dead State	C	0	0	0	0	0	

Figure C-5   DUAL SOURCE SYSTE

MODE	MD1	MD2	MD3	MD4	MD5	MD6	MD7	SOLAR FAN	H.P. FAN	H.P. COMP
Heat F Coll	0	0	C	0	C	C	C	1	1	1
Heat F Stor	C	0	C	0	C	C	C	0	1	0
Store Heat	0	C	0	C	C	C	C	1	0	0
Solar Assist From Coll	0	0*	0	C	0	C*	C*	0	1	1
Solar Assist From Store	C	0*	0	C	0	C*	C*	0	1	1
Heat From H.P. F Outside	C	C	0	C	0	0	0	0	1	1

* These dampers will be modulated in these modes to not allow
  too warm of air to enter the evaporator (outdoor) section of
  the heat pump.

Figure C-6   OFF PEAK SYSTEM

MODE	MD1	MD2	MD3	MD4	MD5	MD6	MD7	MD8	SOLAR FAN	H.P. FAN	H.P. COMP
Heat F Coll	0	0	C	0	C	C	C	0	1	1	0
Heat F Stor	C	0	C	0	C	C	C	0	0	1	0
Store Heat	0	C	0	0	C	C	C	0	1	0	0
Heat From H.P. & St. Ht.	0	C	0	0	C	C	C	0	1	1	1
Cool F Stor	C	0	C	0	C	C	C	0	0	1	0
Stor Cool	0	0	C	C	0	0	C	0	1	1	1

Appendix B

SIMULATION OF FOUR OPEN CYCLE DESICCANT
COOLING SYSTEMS

Report Submitted To

SOLARON CORPORATION

by

John W. Mitchell

Patrick J. Hughes

William A. Beckman

Solar Energy Laboratory
University of Wisconsin
Madison, WI

June 20, 1977

1. Computer Simulation for this study is based on hourly weather data for New York City, New York from 1 July through 22 Sept. 1957.

    Total year data is available from 1 July 1957 through 30 June 1958.

2. Psychrometric cycles have been added to correspond with the various system schematics (Figures 3, 5, 7 & 9).

3. Figure 21 added to show integration of solar heating and cooling systems.

4. Figures and numbers on the figures themselves have been added/revised (Figures 1 through 21), but reference numbers have not been changed in the text written by the University of Wisconsin.

5. A descriptive narrative has been added for figures 1 through 21, and Tables 1 & 2.

6. The COP line of Figure 16 is now plotted correctly. As presented at the TR/QRM meeting of 27 June 1957 the COP line was erroneously plotted against the wrong scale. The tabular data is correct.

7. Coefficient of performance (COP) as shown and used in these figures is based on the total energy input to the desiccant cycle, including both conventional and collected solar heat and is equal to total cycle cooling output divided by total heat input to the cycle. This parameter is useful in a comparative evaluation of the various cycles.

    For evaluating systems relative to conventional energy usage, the COP should be evaluated as total system output divided by conventional energy input only. These values of COP will be generated in future analyses.

DESCRIPTION OF SYSTEMS

The general arrangement of the open cycle, desiccant cooling system using solar energy simulated in these studies is shown schematically in Fig. 1. The solar energy collection system is a Solaron system and the space heating load is that of the HUD AAD cluster for New York City, New York.

The solar desiccant system is built up of available components in four configurations. These are shown schematically in Figures 2-5. The control strategy is the same as that used by J. R. Nelson in his thesis for the Miami simulations.

Preliminary calculations were done with f-chart to determine reasonable sizes for the collector. It was found that 50 m^2 of collector allowed 40% of the space heating load to be met, 100 m^2 met 60%, and 135 m^2 met 70%.

SYSTEM PARAMETERS

The major parameter values used for all system simulations are given below.

Collector

$$F' = 0.781$$

$$U_L = 0.85 \text{ Btu/hr ft}^2 \text{F}$$

$$\alpha = 0.95$$

$$\tau = 0.815$$

$$\overline{\tau\alpha} = 0.753$$

Rock Bed

$$L = 6 \text{ ft}$$

$$A = 0.125 \, A_c$$

Rock Bed, cont'd.

$U = 0.0832$ Btu/hr ft^{2}F

$k = 0.022$ Btu/hr ft F

$C_r = 0.22$ Btu/lb F

$\rho_r = 95$ lb/ft^3

Domestic Hot Water System

Heat Exchangers: $UA = 440$ Btu/hr F

Tank: $V = 80$ gallons

$U = 0.1$ Btu/hr ft^{2}F

Water Flow Rate: Scheduled over period 6 AM to 7 PM

Total flow is 667 lb/day

Desiccant Systems

Dehumidifier: Area/Vol $= 1028$ m^2/m^3

Flow Length $= 0.0381$ m

Total Frontal Area $= 1.56$ m^2

Void Fraction $= 0.4$

Rotational Speed $= 1.023 \times 10^{-3}$ rev/sec

Conductance $= 121$ w/m^{2}K

Friction Factor $= 0.9$

Desiccant Mass/Air Mass $= 1607$

Sensible Heat Regenerator:

Effectiveness $= 0.90$

Evaporative Coolers

Maximum Effectiveness $= 0.90$

Indirect Evaporative Cooler Effectiveness $= 0.95$

House Load

New York City weather for July 1 to September 22, 1955

House UA = 1700 kJ/hrC

Thermostat Setting = 23.8C

Humidistat Setting = 0.009 Kg/Kg

Sensible Load = 2.0×10^6 W-hr

Latent Load = 1.2×10^6 W-hr

(The loads were found to be constant for all systems simulated.)

RESULTS

Three sets of simulations were performed as follows:

Set A: Ventilation mode, recirculation mode 1 and recirculation

mode 3 for a constant air flow rate per unit collector area

of 2 cfm/ft^2 and for variable collector area.

Set B: All four modes for a constant mass flow rate of 1540 cfm

and variable collector area.

Set C: All four modes for a constant collector area and variable

flow rate.

The results for Set A are given in Table 1 and presented graphically

in Figures 6, 7 and 8. Note that there are no entries for the water con-

sumption. A programming error occurred, and the previous values reported

are wrong. The coefficient of performance is the total load met divided

by the sum of the solar and auxiliary energy supplied.

The results for Sets B and C are given in Table 2 and shown in Figures

9, 10, 11, and 12. The results are grouped by mode to show the influence

of area and flow rate on each mode. For the recirculation mode, option 3,

the water flow rate does not include that used in the indirect evaporative

cooler. The amount of water used was not calculated as the evaporative

cooler was assumed to have a constant 95% effectiveness.

CONCLUSIONS

The following are conclusions that can be drawn from this study.

1. The results from Set A show that the flow rate through the machine and load is important. Either too large or too small a flow rate requires additional auxiliary energy. For large collector areas, it is better to reduce collector performance by lowering flow rate than to maintain a design value of flow per unit collector area.

2. The results from Set B and C show that the solar system can provide essentially all of the energy required. The collector area required for 90% solar depends on mode as follows:

	$A_c(m^2)$
Ventilation	60
Recirculation, Option 1	55
Recirculation, Option 2	40
Recirculation, Option 3	40

The implication is that for these collector areas and larger, the systems will probably operate satisfactorily with no auxiliary and acceptable diurnal changes in room temperature.

3. The coefficients of performance are somewhat lower than expected, and show a decrease with increased area. This is due, in part, to the available solar temperature being much greater than needed much of the time. Thus, solar energy is not effectively used. Improved control could alleviate this situation.

4. The water useage for all modes is comparable. The average useage is about 50 kg/day, or 10-15 gal/day.

5. The hours of operation decrease with increased area. The machine
is operating six hours per day on the average. However, some days it is on
more than 12 hours, and others on as little as one hour, depending on the
weather for that day.

6. The results of Set C show that, generally, the lowest flow rate
yields the best performance. The ventilation mode is the most sensitive
to flow rate. Option 3 shows a slight decrease in auxiliary with increased
flow.

7. The pressure drops through the dehumidifier and regenerators are
about 0.1 and 0.07 inches water, respectively, per side. The overall pressure
drop through a desiccant system with one dehumidifier and one regenerator at
1540 cfm is then about 0.34 inches water. For 400 hours of operation the
fan power would be about 24,000 w-hr, or about 0.8% of the load.

8. The control strategy needs considerable further research before
an optimum design is obtained. The strategy employed is that used in Nelson's
thesis for Miami, and is obviously not satisfactory for a climate like New
York. An actual system could undoubtably be made to operate better using
an improved control strategy.

NARRATIVE DESCRIPTION OF FIGURES

Fig. 1   This figure is a schematic of a Solaron heating system integrated with any of the proposed desiccant air conditioning systems. The basic modes of operation of the system allow the desiccant system to be regenerated directly from the collector or from the stored heat in the pebble bed. The system can also store heat from the collector in the pebble bed, for periods when there is solar collector heat available and there is no simultaneous need for cooling.

Fig's. 2 through 9 This group of figures shows the variety of solar assisted desiccant cooling system modes presently under study and development by Solaron Corporation. Each mode is described by a system flow schematic (Fig's. 2, 4, 6 & 8) followed by a corresponding psychrometric cycle diagram (Fig's. 3, 5, 7 & 9).

Fig's. 2 & 3   Vent Cycle - This proposed cycle is called a vent cycle, due to its use of 100% fresh air. Referring to the system flow diagram and psychrometric chart, ambient air at state 1 is drawn through a desiccant wheel, where it is dried and heated to state 2. The air then is cooled by a sensible air-to-air heat exchanger to state 3 followed by evaporative cooling to state 4, where it enters the room. Return air from the room (state 5) is first evaporatively cooled to 6 and then is heated by the sensible heat exchange process to state 7. At this point it is further heated by the solar system (either by the collector or storage, Fig. 1) and an auxiliary heater if needed, to state 8. This hot air then regenerates the desiccant wheel and is discharged to the atmosphere at state 9.

Fig's. 4 & 5   Recirculation Cycle, Option 1 - This is one of three proposed recirculation cycles. Recirculation in this case refers to the room air stream flowing in a closed loop, while a second open loop air stream is used for regeneration. Referring to the room air loop first, return air at 6 is drawn through a desiccant wheel where it is dried and heated to 7; the hot dry air at state 7 is then sensibly cooled to state 8 and evaporatively cooled to state 9 where it re-enters the rooms. The regeneration cycle consists of ambient air at state 1 which is evaporatively cooled (state 2) and then sensibly heated (state 3). Air at state point 3 is then heated by the solar system and/or auxiliary (if required) to state 4; this air then is used to regenerate the desiccant wheel and is then discharged to atmosphere at state 5.

<u>Fig's. 6 & 7</u>   Recirculation Cycle, option 2 - In this cycle a slight
change is made to the system in option 1 in an attempt to increase
the overall COP. Return air at state 5 is dried to state 6 and
sensibly cooled as before, however at state 7 the air is sensibly
cooled further by an indirect evaporative cooler, which allows
state 8 to closely approach the outdoor wet bulb temperature. Air
at state 8 is then evaporatively cooled to 9 where it enters the
room. Ambient air used in the regenerative loop actually takes two
paths. A relatively large flow is used in the indirect evaporative
cooler to provide cooling to the room air stream, while an addi-
tional amount of air equal in flow to the room air rate is directed
into the sensible heat exchanger. This air is heated to the same
temperature at state 2 as was the air at state 3 of option 1, but
since it has not been evaporatively cooled in this cycle it is at a
lower humidity. This allows the air at state 3 of this cycle to be
at a lower temperature and still achieve the required regeneration
of the wheel.

<u>Fig's. 8 & 9</u>   Recirculation Cycle, option 3 - This cycle is exactly the
same as option 1 on the room side, but is significantly different
on the regeneration side. Specifically, ambient air enters at two
different points in the regeneration process. One ambient air
stream is evaporatively cooled from state 1 to 2 and then heated
from 2 to 3 by the first sensible heat exchanger. Ambient air at
state 1 also enters a second sensible heat exchanger and is heated
to state 4; this air is then further heated by the solar system
and/or auxiliary to state 5. This air then regenerates the wheel
and leaves at state 6 where it enters the second sensible heat ex-
changer, is cooled and is discharged at state 7.

<u>Tables 1 & 2</u>   The results of the three sets of Simulations (Sets A,
B & C) are presented in tabular form in these Tables. Table 1
gives the results of Set A, and Table 2 gives the results of Sets
B & C.

<u>Fig's. 10 through 20</u>   present graphically the preliminary results of
a computerized cycle simulation program at the University of Wiscon-
sin. This particular study was run on a specific house in New York
City, N.Y. using hourly weather data recorded in the period of 1
July through 22 September 1957. In this way comparison of various
cycle parameters for the modes and system cycles is directly related
to the same set of operating conditions and loads. The parameters
analyzed in this study are COP, hours of system operation, water
consumption, and auxiliary energy usage, all totalized over the
entire period of the study, 1 July through 22 September.

<u>Figs. 10, 11 & 12</u>   present the above data (Set A) for a standard
(normal) air flow rate of 2 SCFM/ft.^{2} (21.52 SCFM/meter^2) of
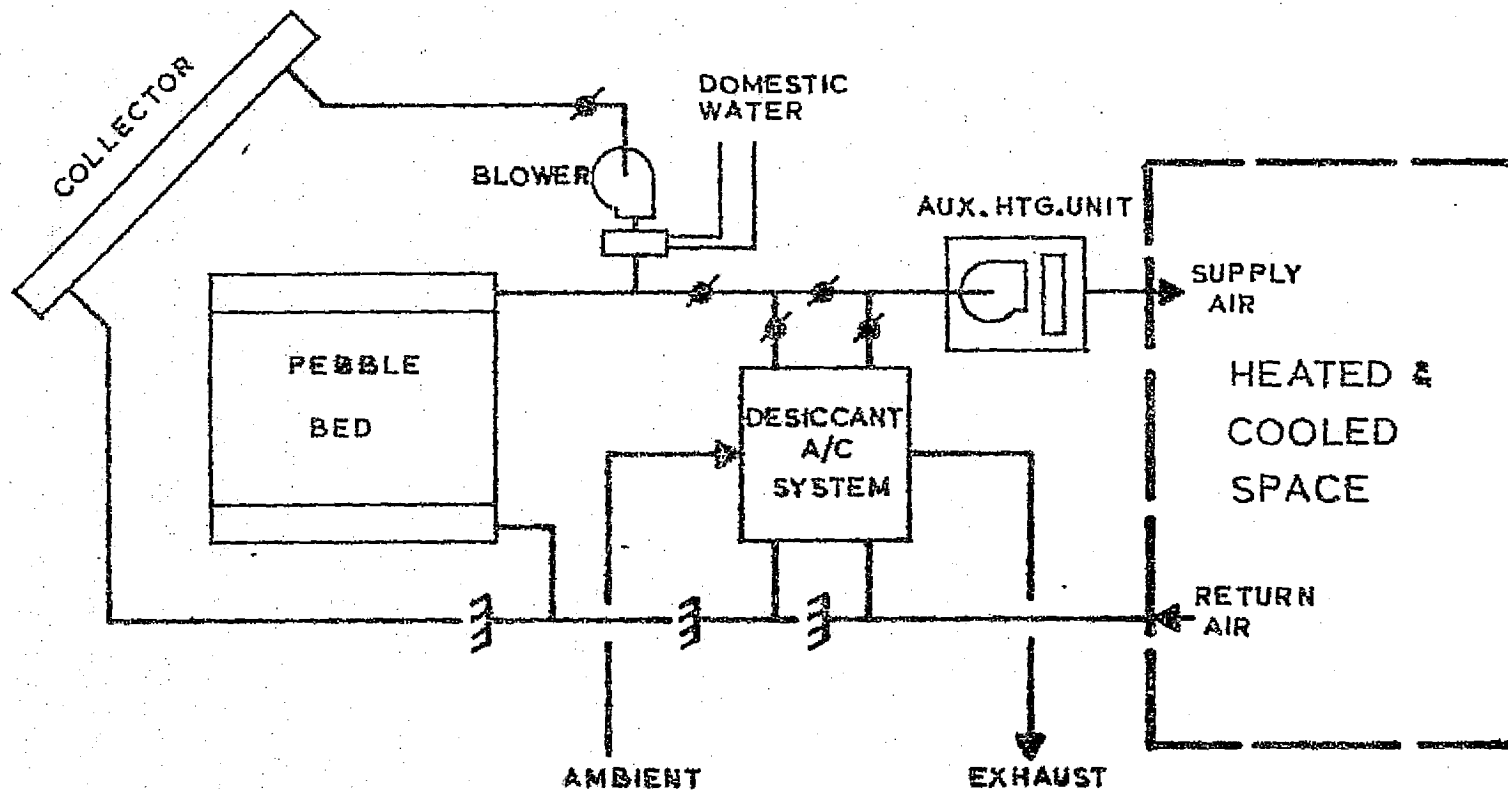Collector area, and for variable collector area.

Fig's. 13, 15, 17 & 19   present the above data (Set B) for a constant air flow rate of 1540 SCFM and a variable collector area from 25 to 75 square meters.

Fig's. 14, 16, 18 & 20   present the same data (Set C) for a constant collector area of 55 square meters and a variable air flow rate from 1260 to 1680 SCFM.

Fig. 21   This is a schematic flow diagram for an integrated solar heating and cooling system with additional provision for domestic hot water.   The cooling portion of this system is Solaron desiccant recirculation cycle, option 1, as shown in Fig's. 2 & 3.

COP   - Coefficient of Performance (COP) as shown and used in these figures is based on the total energy input to the desiccant cycle, including both conventional and collected solar heat and is equal to total cycle cooling output divided by total heat input to the cycle.   This parameter is useful in a comparative evaluation of the various cycles.

For evaluating systems relative to conventional energy usage, the COP should be evaluated as total system output divided by conventional energy input only.   These values of COP will be generated in future analyses.

SOLARON System Integrated With Desiccant Air Conditioning System

MODES: HEATING
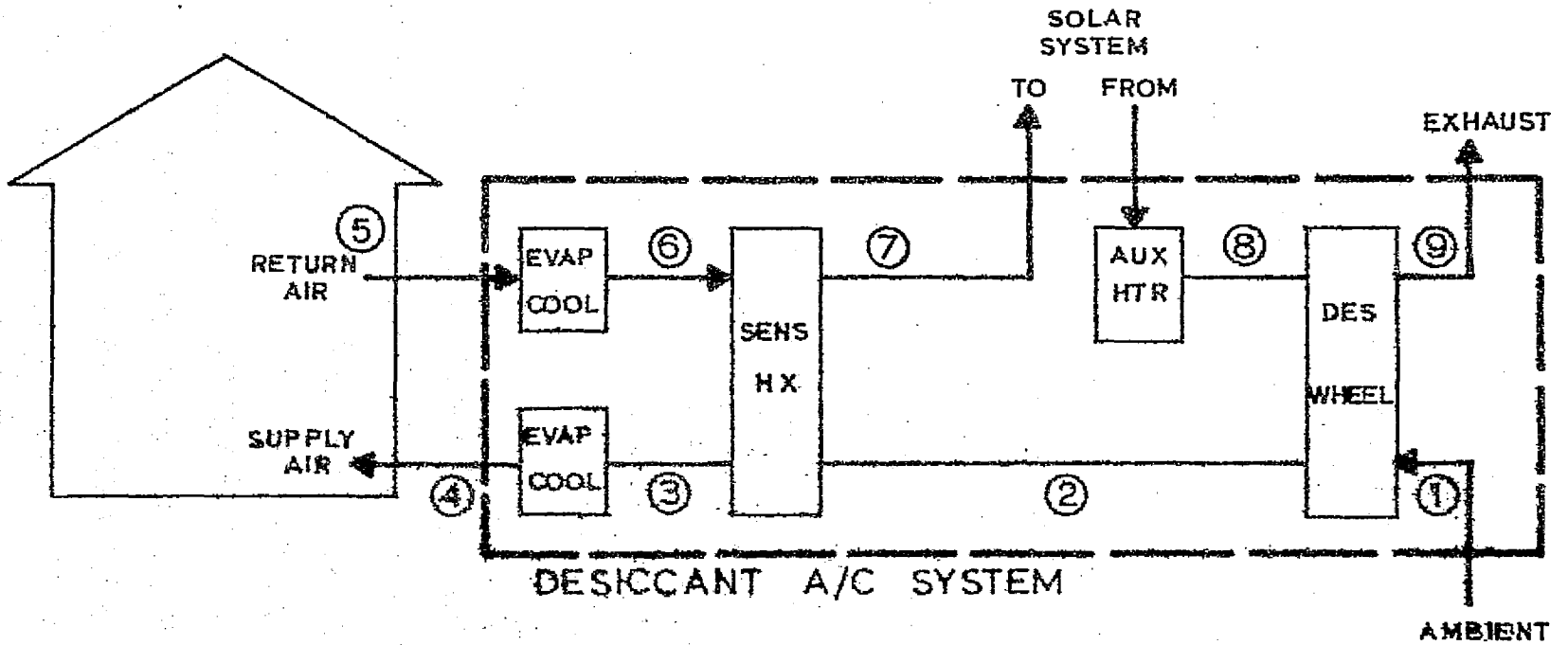   OPERATION — HEAT FROM COLLECTOR
                     — STORE HEAT
                     — HEAT FROM STORAGE

          COOLING
          OPERATION — REGENERATE FROM COLLECTOR
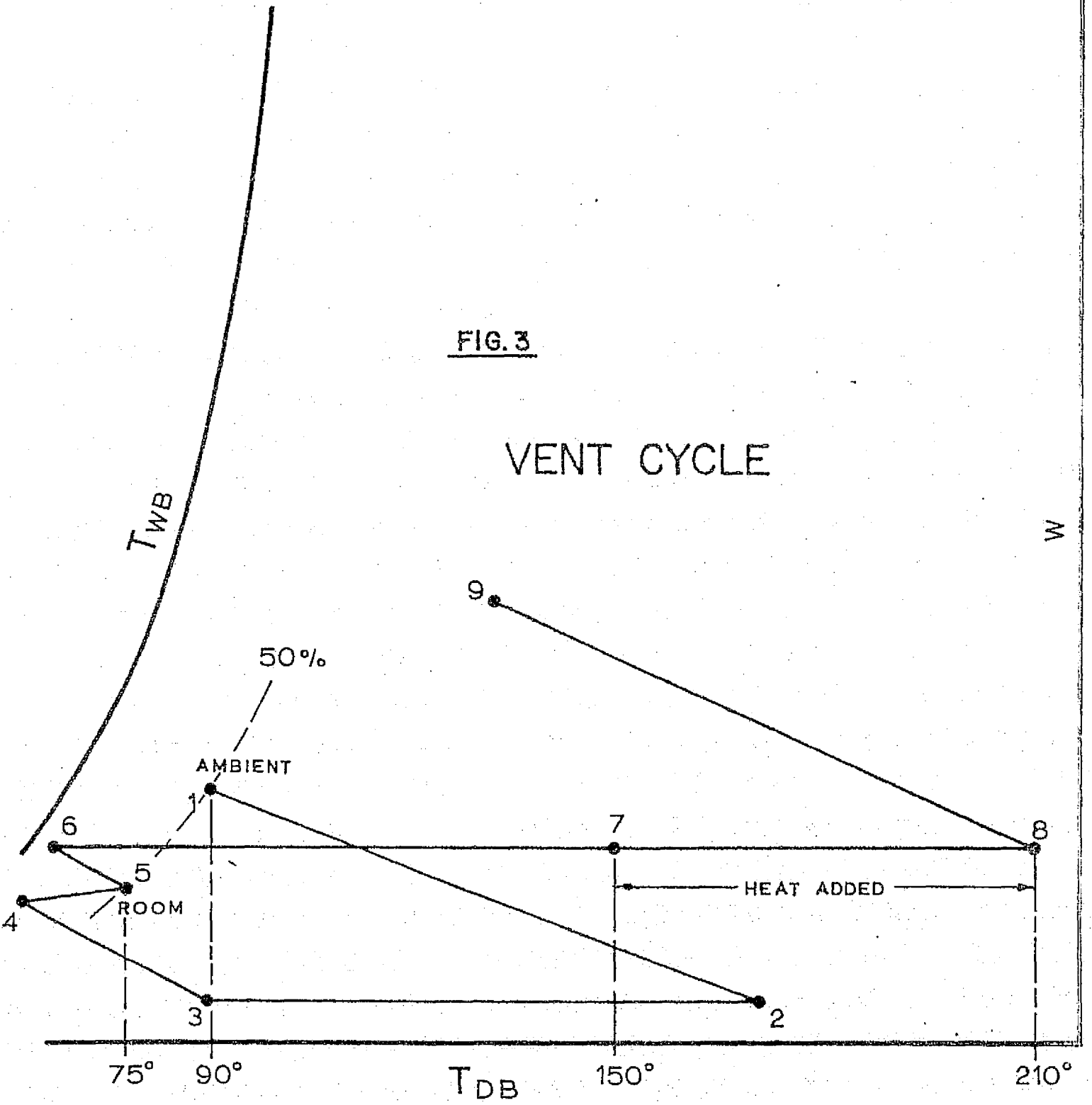   — STORE HEAT
                     — REGENERATE FROM STORAGE

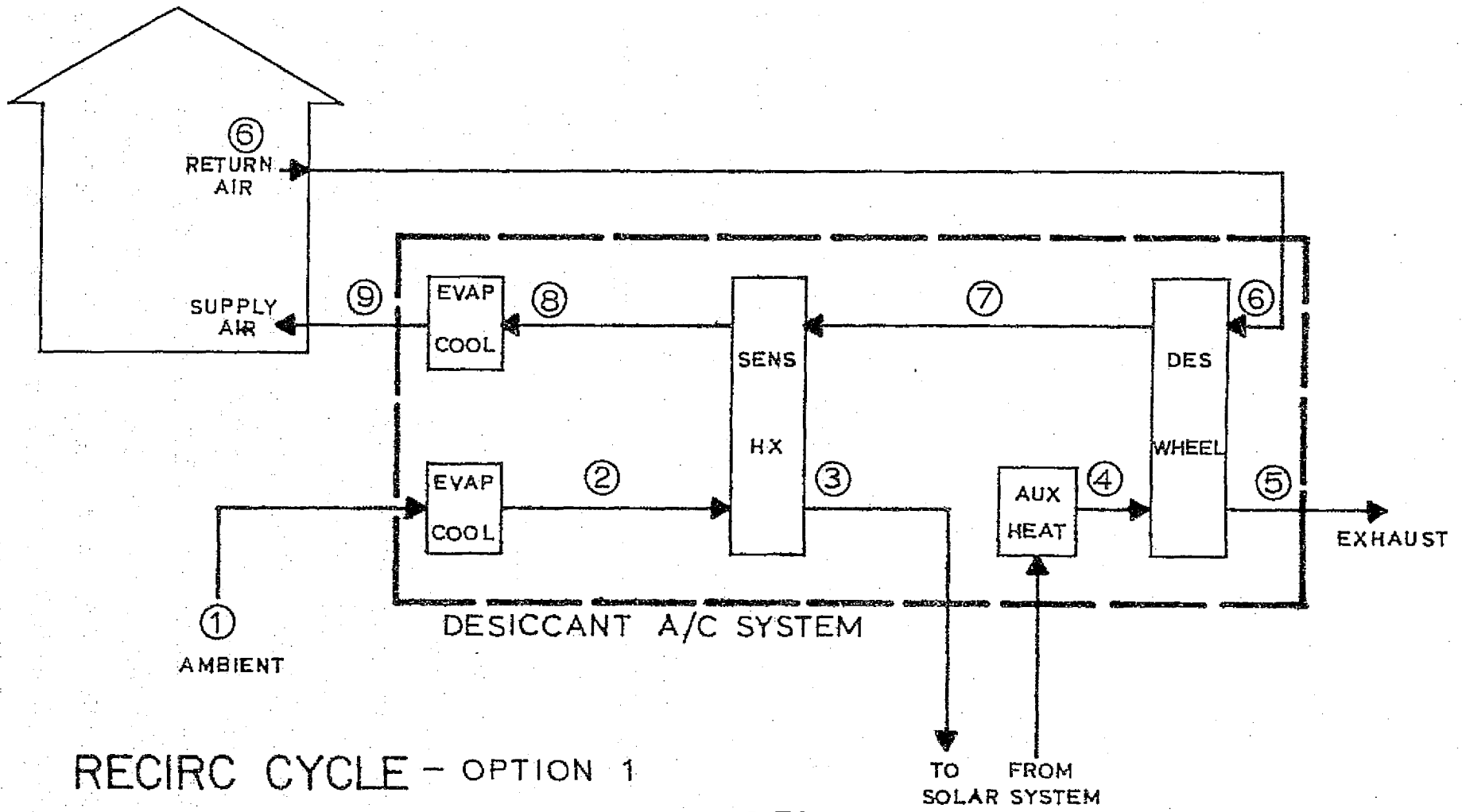FIG. 1

VENT CYCLE

SENSIBLE HX $\varepsilon = 0.70$

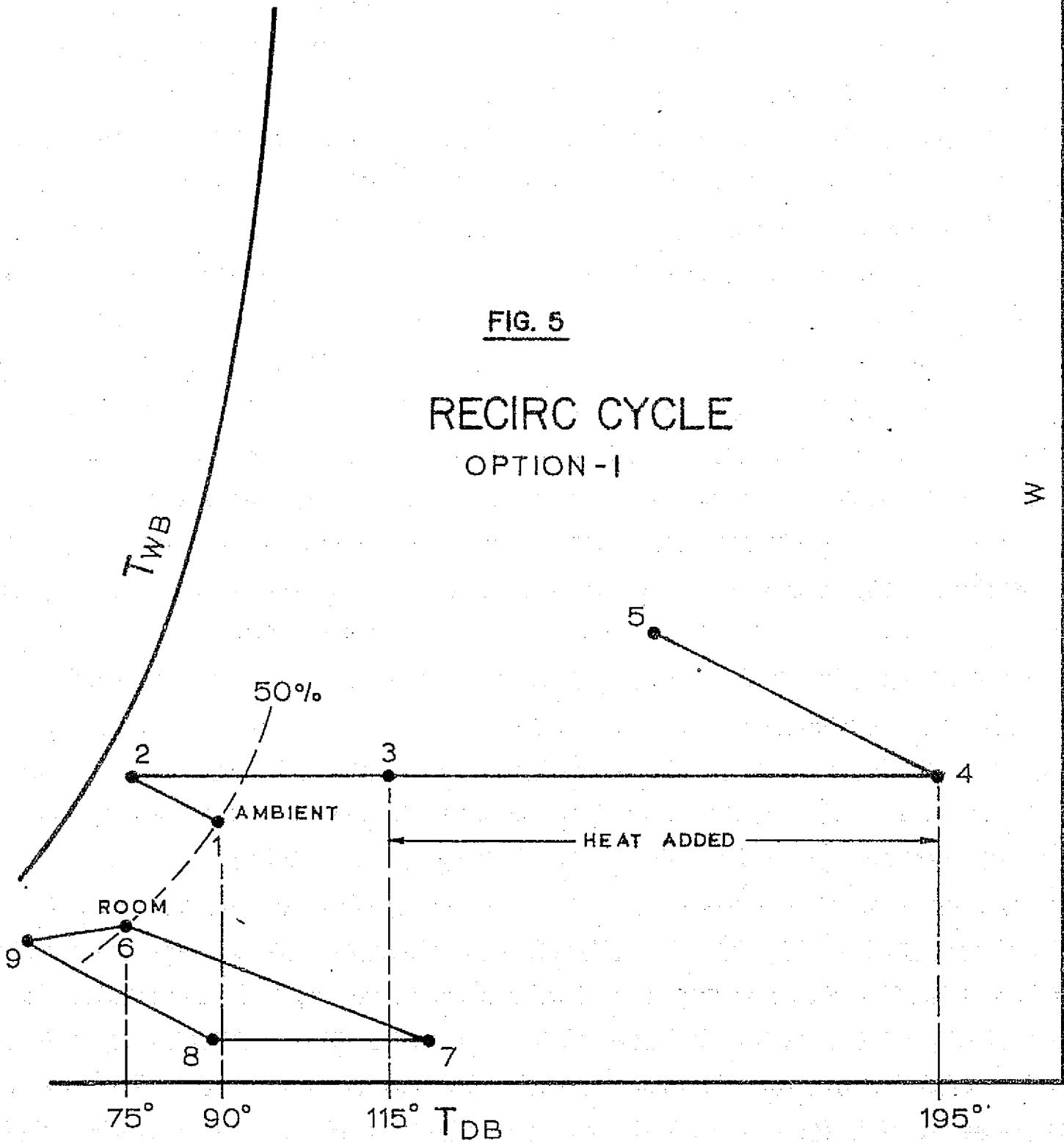$\varepsilon = 0.85$

FIG. 2

FIG. 3

VENT CYCLE

B-14

RETURN AIR ⑥

SUPPLY AIR ⑨ EVAP COOL ⑧ SENS HX ⑦ DES WHEEL ⑥

EVAP COOL ① AMBIENT ② ③ AUX HEAT ④ ⑤ EXHAUST

DESICCANT A/C SYSTEM

TO FROM
SOLAR SYSTEM

RECIRC CYCLE – OPTION 1

FIG. 4

SENSIBLE HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

FIG. 5

RECIRC CYCLE
OPTION-I
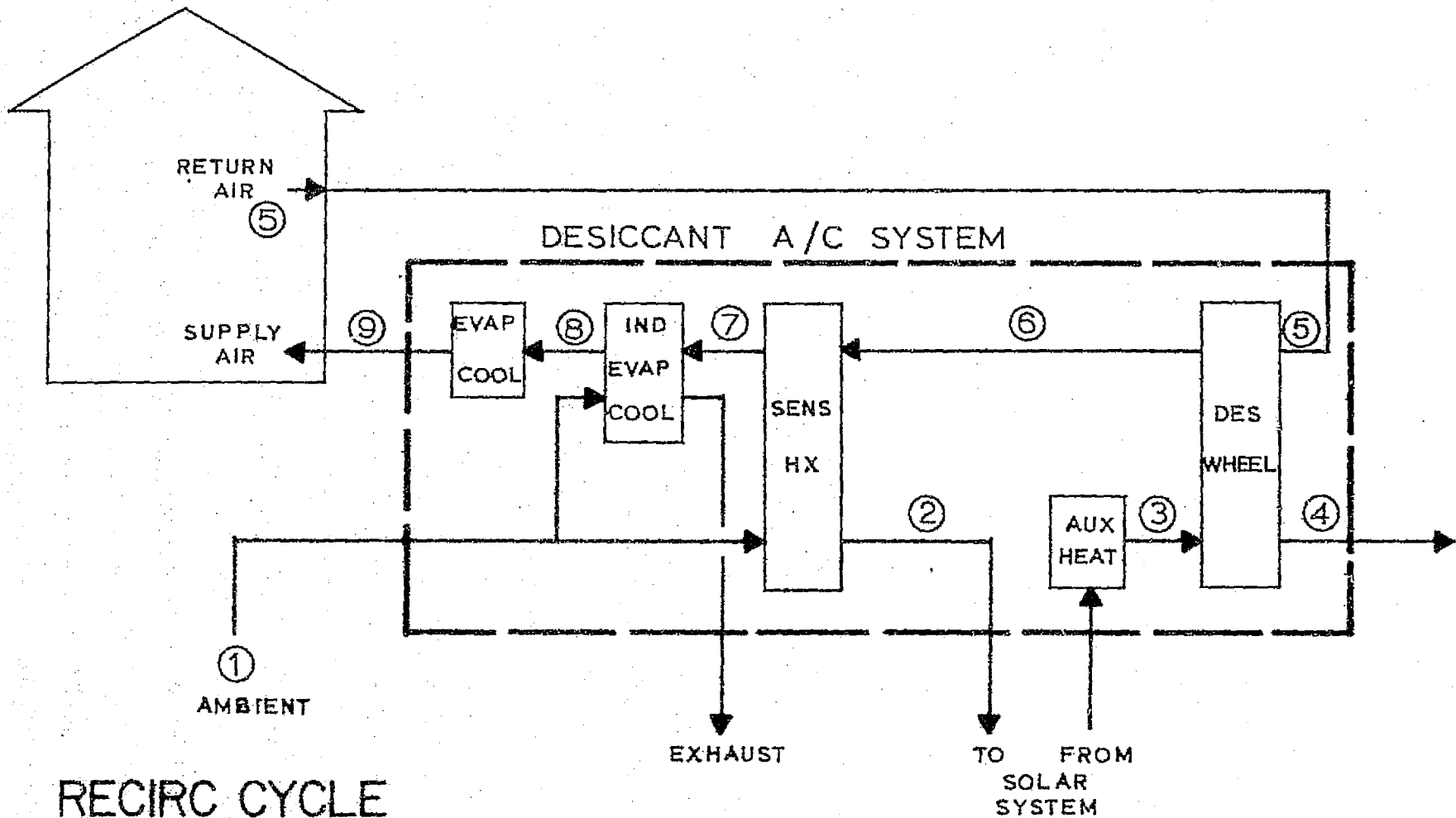
RETURN AIR ⑤

SUPPLY AIR

DESICCANT A/C SYSTEM

EVAP COOL ⑧ IND EVAP COOL ⑦ SENS HX ⑥ DES WHEEL ⑤

⑨

② AUX HEAT ③ ④

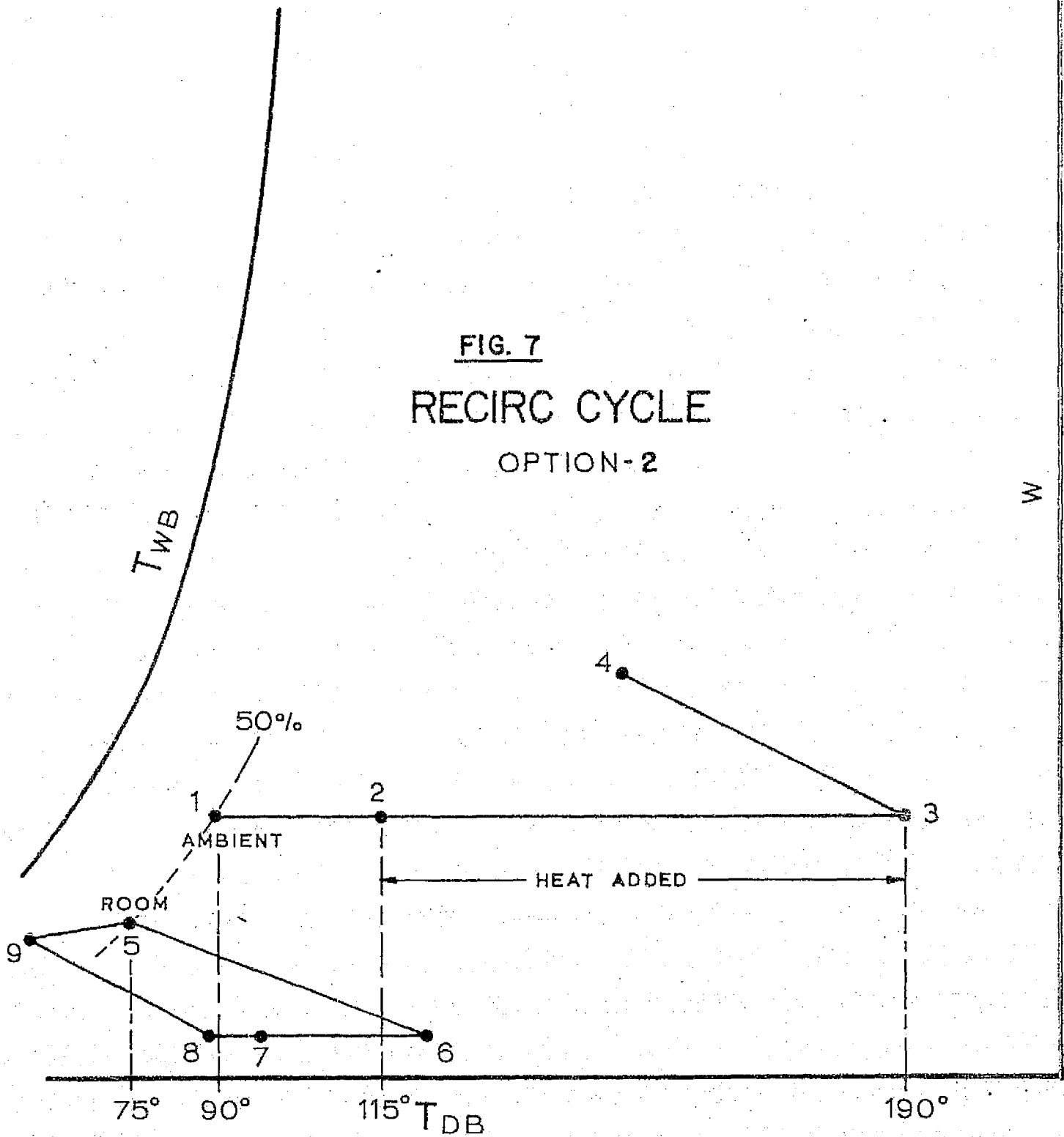① AMBIENT

EXHAUST

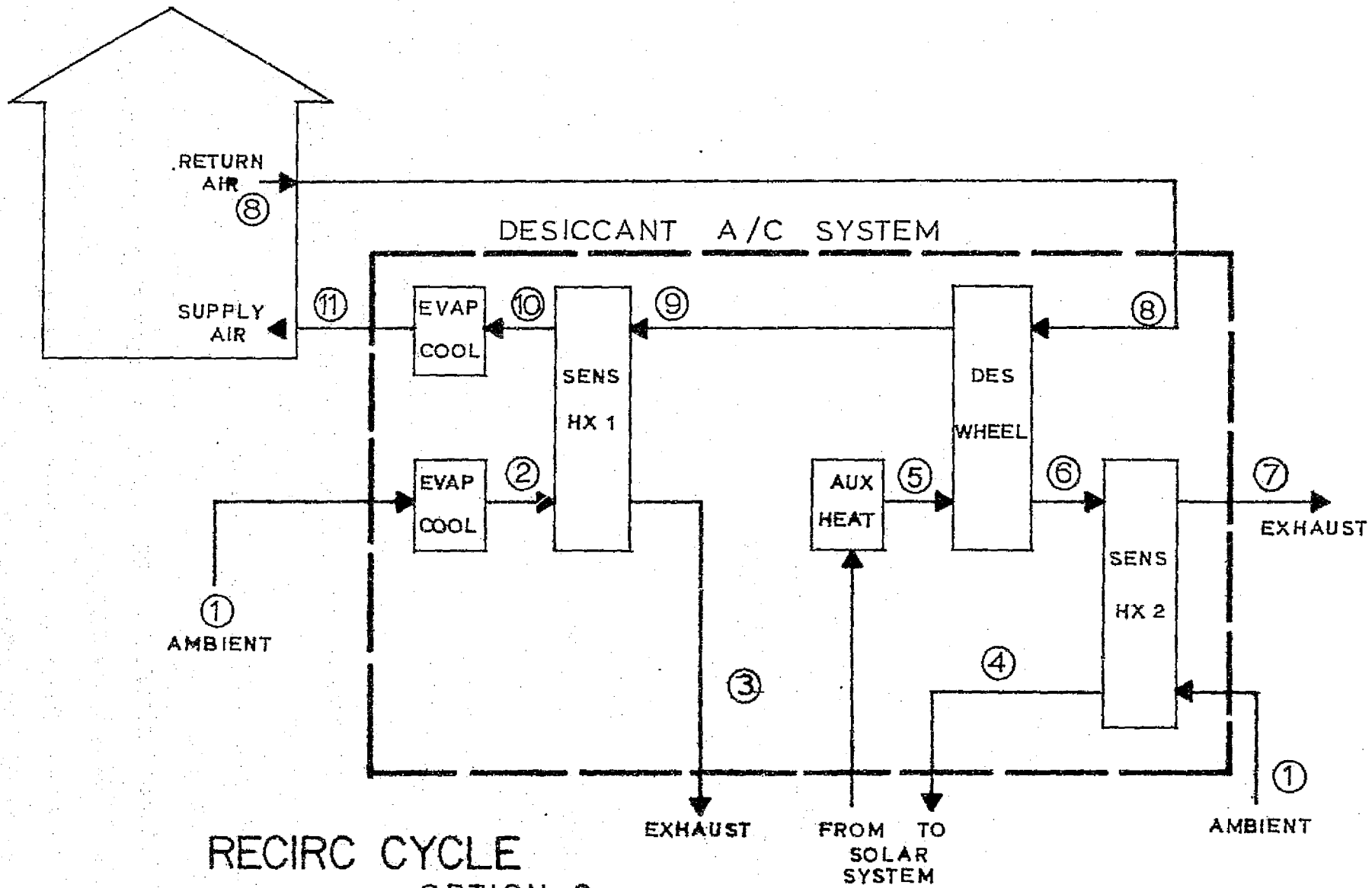TO FROM SOLAR SYSTEM

RECIRC CYCLE
OPTION 2

SENSIBLE HX $\varepsilon = 0.70$
$\varepsilon = 0.85$

INDIRECT EVAP COOLER $\eta = \dfrac{T_{DB7} - T_{DB8}}{T_{DB7} - T_{WB1}}$ $= 0.63$
$= 0.86$
$= 0.95$

FIG. 6

FIG. 7

RECIRC CYCLE
OPTION-2

T_{WB}

W

50%

1 AMBIENT

2

4

3

HEAT ADDED

ROOM

9

5

8 7

6

75° 90° 115° T_{DB} 190°

DESICCANT A/C SYSTEM

RECIRC CYCLE
OPTION 3

SENS HX 1 $\varepsilon = 0.70$
$\varepsilon = 0.85$

HX 2 $\varepsilon = 0.70$
$\varepsilon = 0.85$
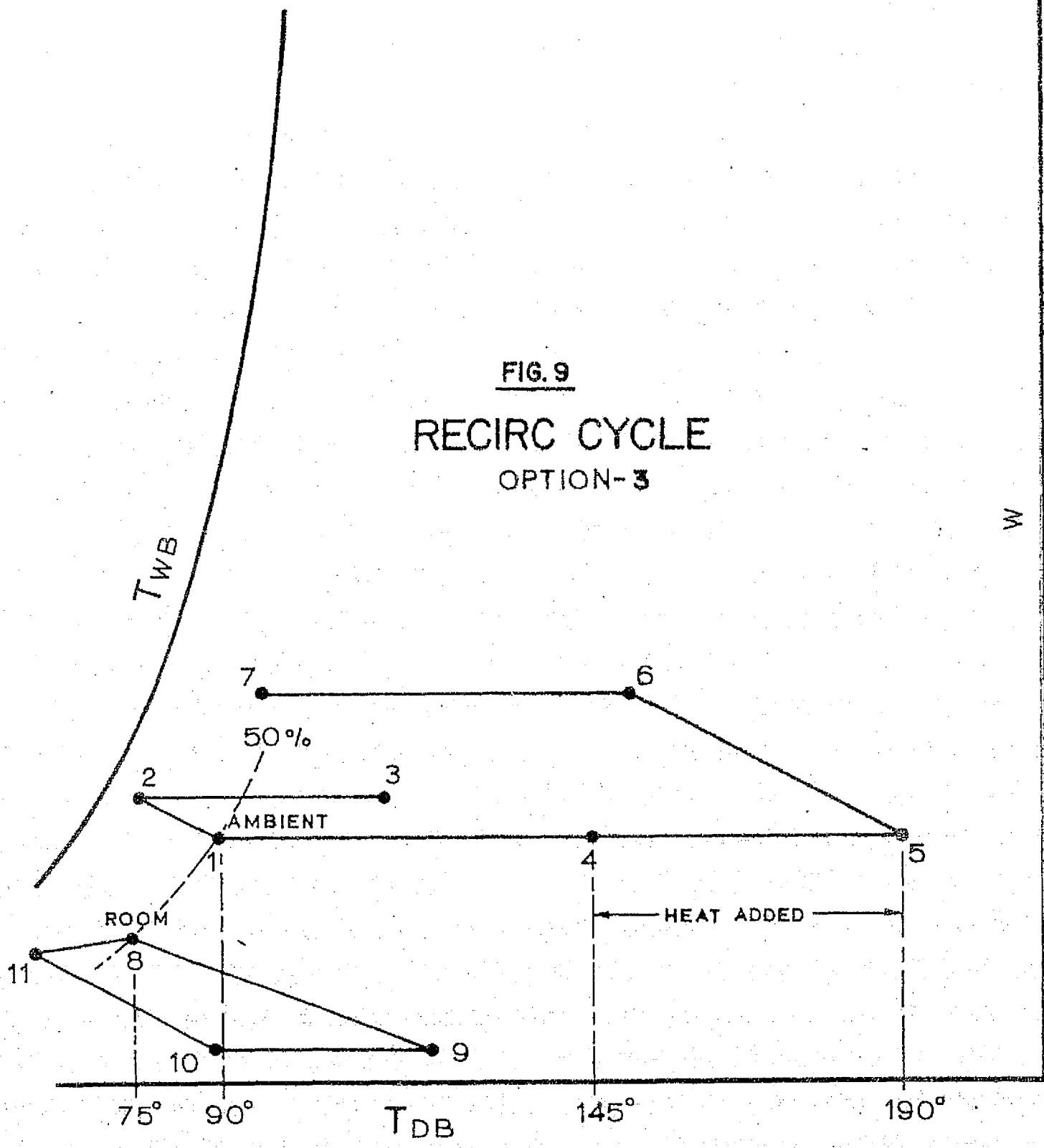
FIG. 8

FIG. 9

RECIRC CYCLE

OPTION-3

TABLE 1

RESULTS FOR SET A

Ventilation Mode

A_c (m^c)	Qsolar (10^6W-hr)	Auxiliary (10^6W-hr)	COP	Percent Solar	Hours of operation
135	19.78	5.11	0.13	79	672
100	13.80	1.68	0.21	89	611
75	9.73	0.96	0.30	91	615
40	4.74	1.51	0.30	76	847

Recirculation Mode, Option 1

A_c	Qsolar	Auxiliary	COP	% Solar	Hours
135	21.12	0.98	0.15	96	440
100	15.06	0.72	0.21	95	444
75	11.21	0.73	0.27	94	495
40	5.88	1.90	0.40	76	763

Recirculation Mode, Option 3

A_c	Qsolar	Auxiliary	COP	% Solar	Hours
135	9.99	0.16	0.33	98	365
100	8.35	0.14	0.33	98	386
75	7.25	0.18	0.44	98	440
40	5.15	1.37	0.48	79	728

TABLE 2

RESULTS FOR SETS B AND C

Ventilation Mode

A_c (M^2)	Flow Rate (cfm)	Qsolar (10^6W-hr)	Auxiliary (10^6W-hr)	COP	Percent solar	Hours of operation	Water Used (kg)
75	1540	10.12	0.45	0.31	96	561	3620
55	1540	8.32	1.36	0.33	86	646	4238
40	1540	6.54	2.65	0.34	71	743	4759
25	1540	3.90	4.96	0.35	44	836	5045
55	1680	8.70	1.81	0.30	83	666	5173
55	1400	7.94	0.98	0.36	89	637	3353
55	1260	7.52	0.70	0.39	91	632	2594

Recirculation Mode, Option 1

A_c	Flow	Qsolar	Auxiliary	COP	% Solar	Hours	Water
75	1540	11.84	0.54	0.26	96	484	4614
55	1540	9.68	1.00	0.30	91	526	4668
40	1540	7.70	1.94	0.33	80	584	4710
25	1540	4.93	4.22	0.35	54	640	4736
55	1680	9.70	1.18	0.29	87	520	4748
55	1400	9.37	0.86	0.32	92	540	4577
55	1260	9.01	0.80	0.33	92	559	4520

TABLE 2 (CONT.)

Recirculation Mode, Option 2

A_c	Flow	Qsolar	Auxiliary	COP	% Solar	Hours	Water*
75	1540	10.39	0.28	0.31	97	371	1776
55	1540	8.55	0.41	0.37	95	394	1743
40	1540	7.02	0.64	0.42	92	432	1714
25	1540	4.77	1.94	0.48	71	483	1629
55	1680	8.95	0.40	0.35	96	386	1756
55	1400	8.30	0.32	0.38	96	405	1737
55	1260	8.30	0.24	0.40	97	423	1750

*Does not include water use of indirect evaporative cooler.

Recirculation Mode, Option 3

A_c	Flow	Qsolar	Auxiliary	COP	% Solar	Hours	Water
75	1540	7.25	0.17	0.43	98	436	4484
55	1540	6.87	0.26	0.46	96	468	4569
40	1540	6.08	0.66	0.48	90	520	4581
25	1540	4.22	2.20	0.49	66	610	4639
55	1680	7.03	0.20	0.45	97	451	4573
55	1400	6.76	0.26	0.47	96	486	4490
55	1260	6.66	0.27	0.47	96	512	4444

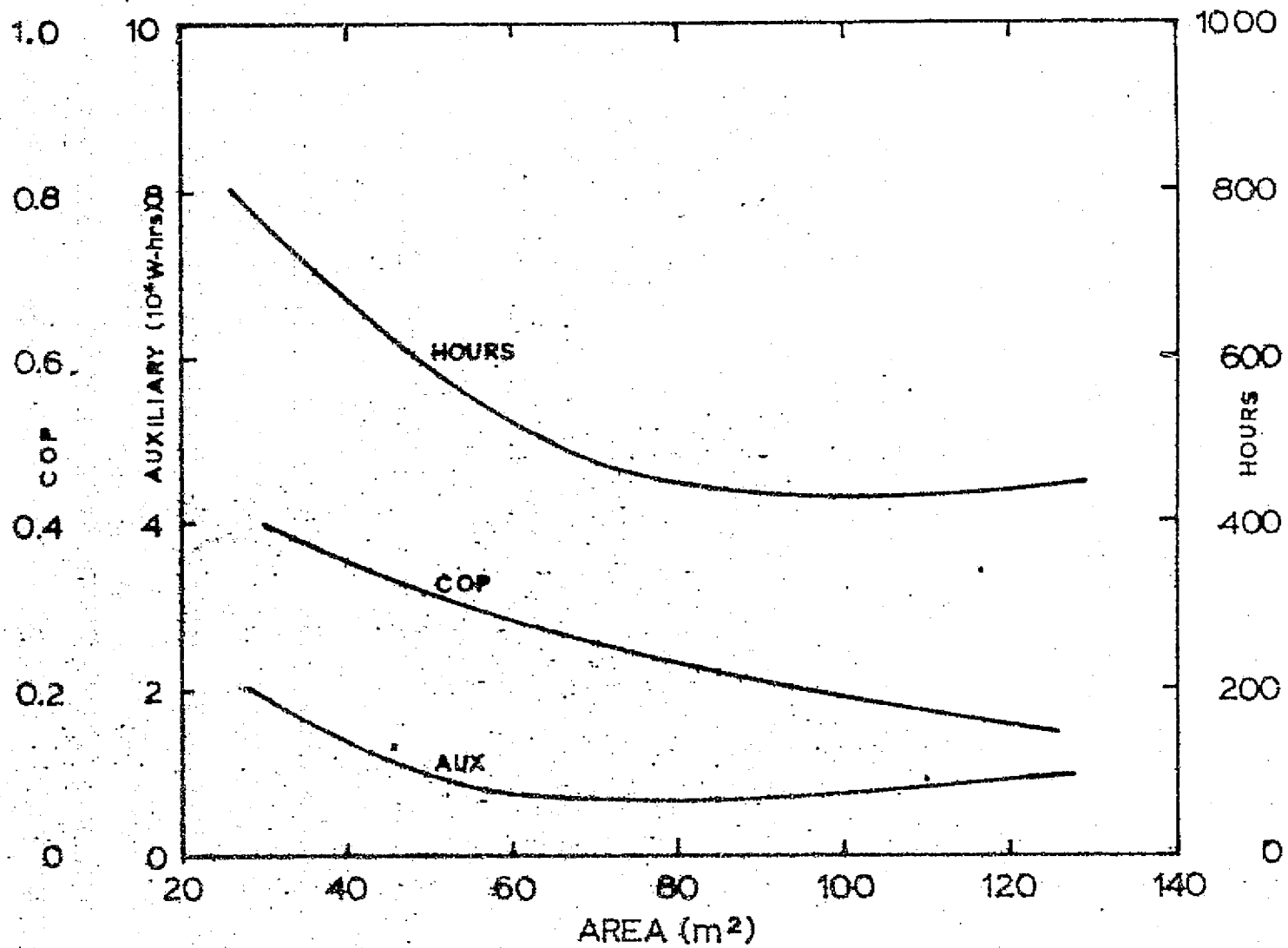# VENTILATION MODE
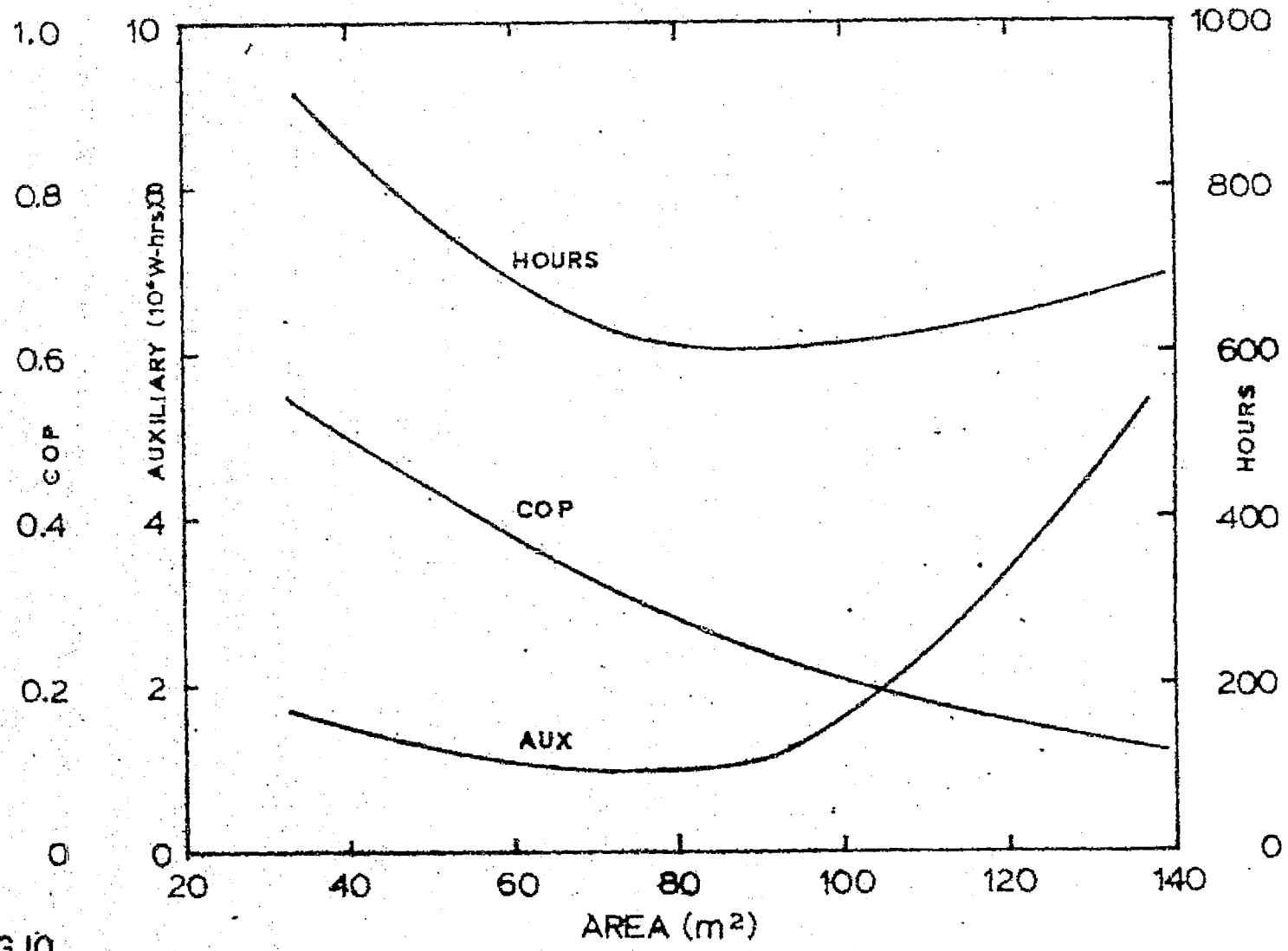
FIG. 10
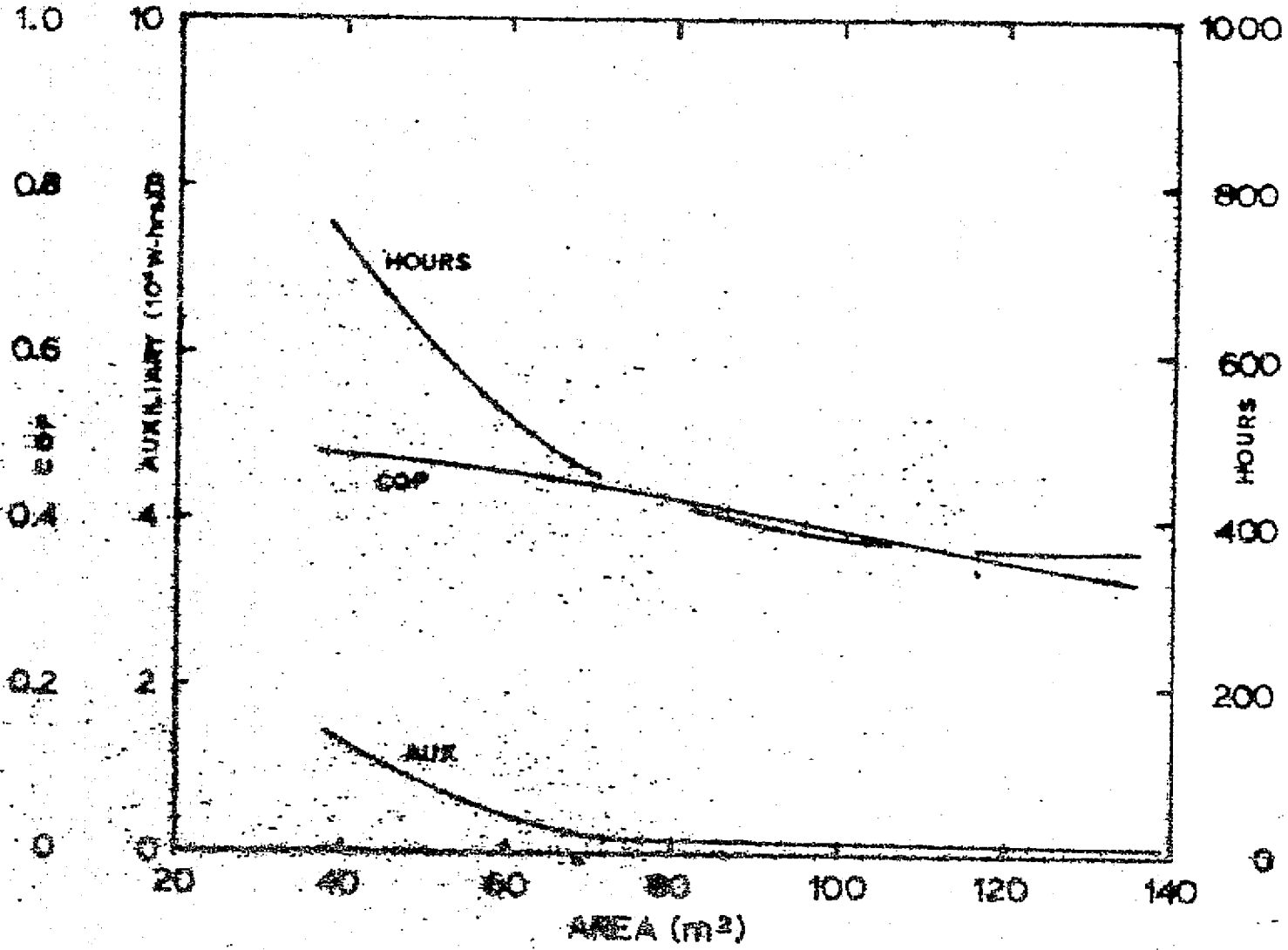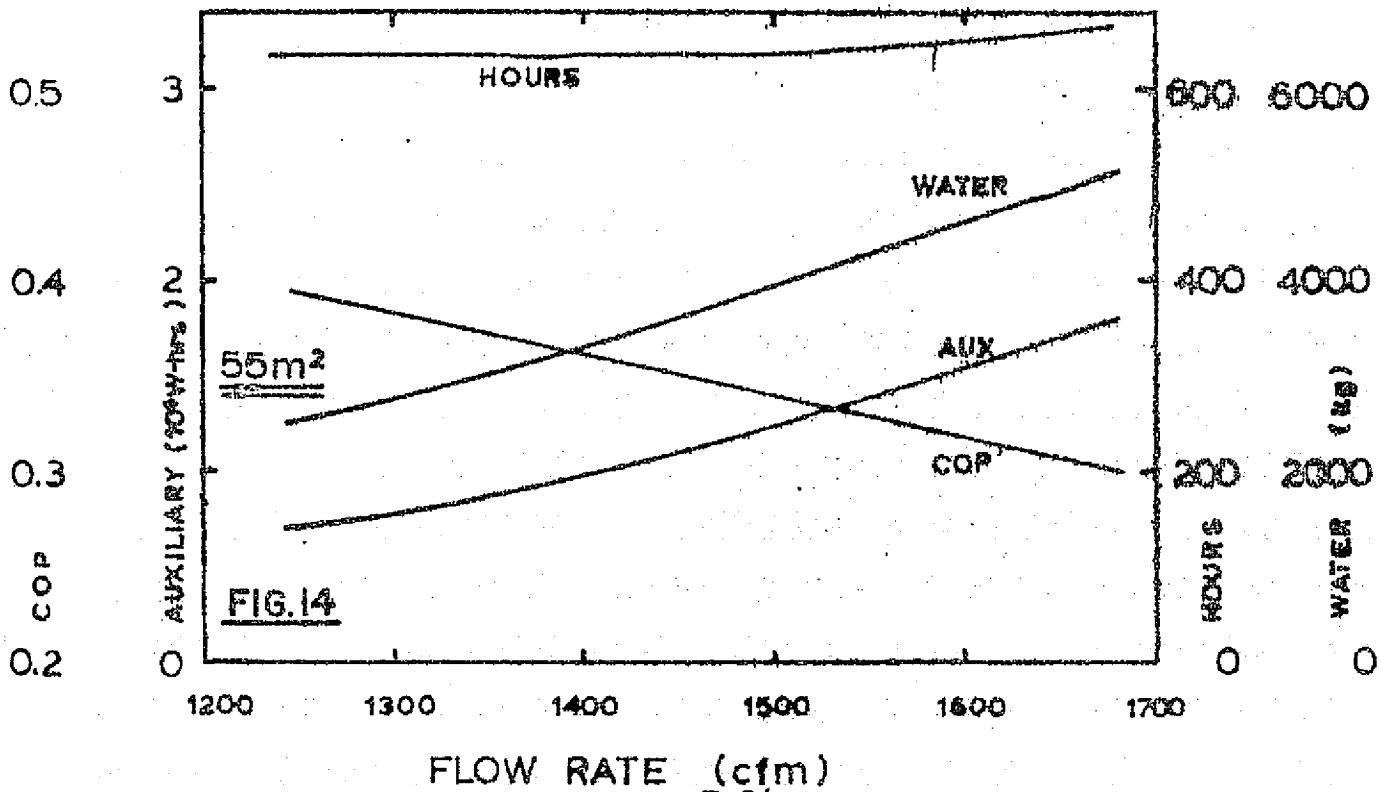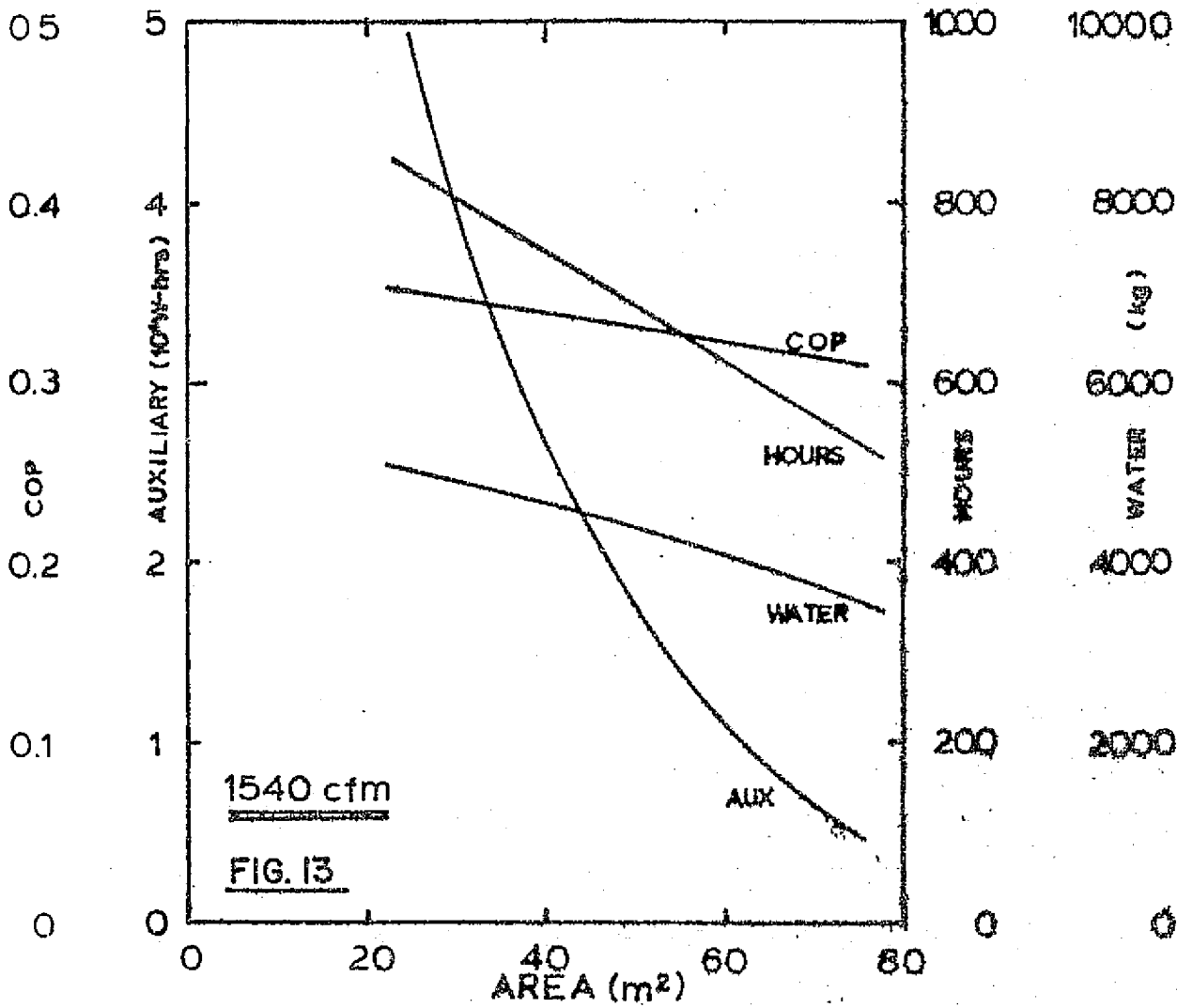
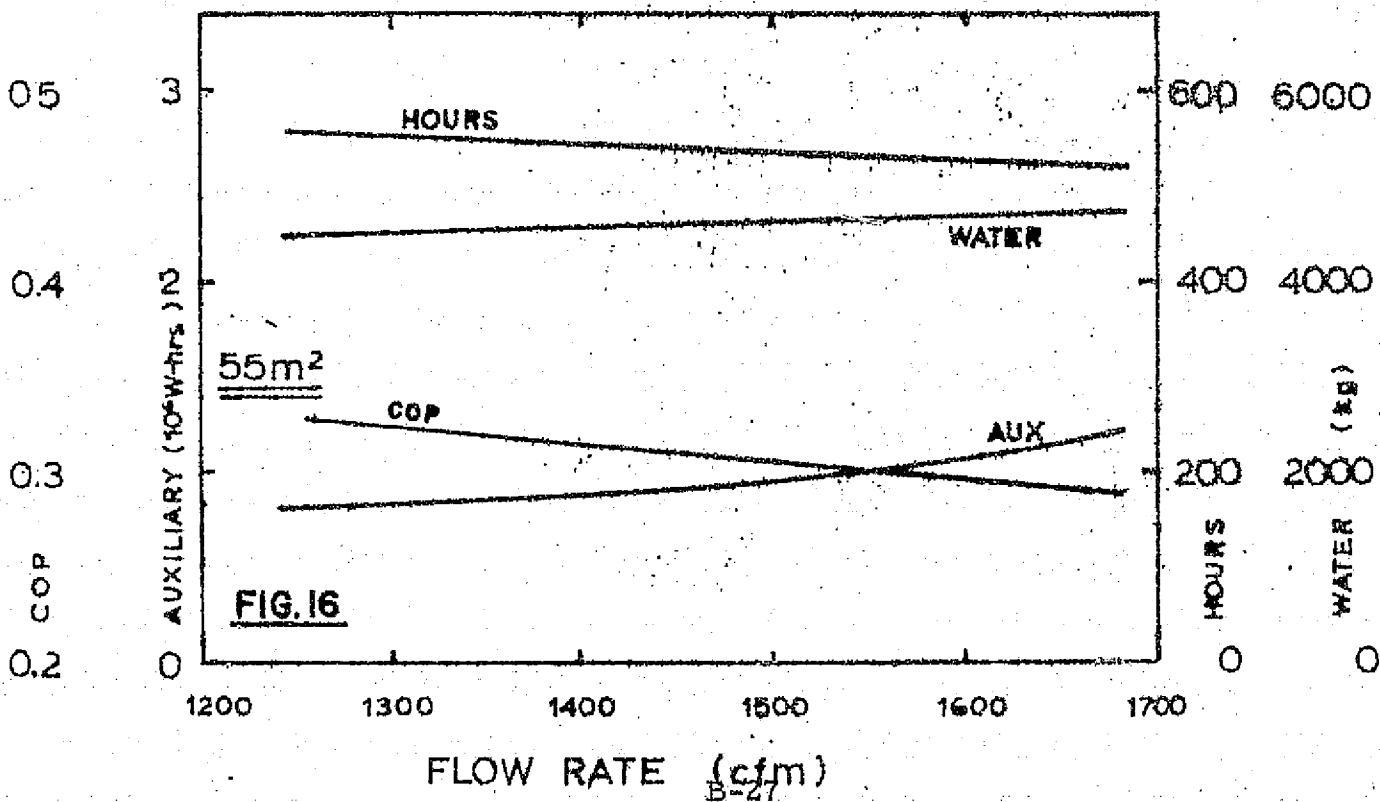RECIRCULATION MODE OPTION 1
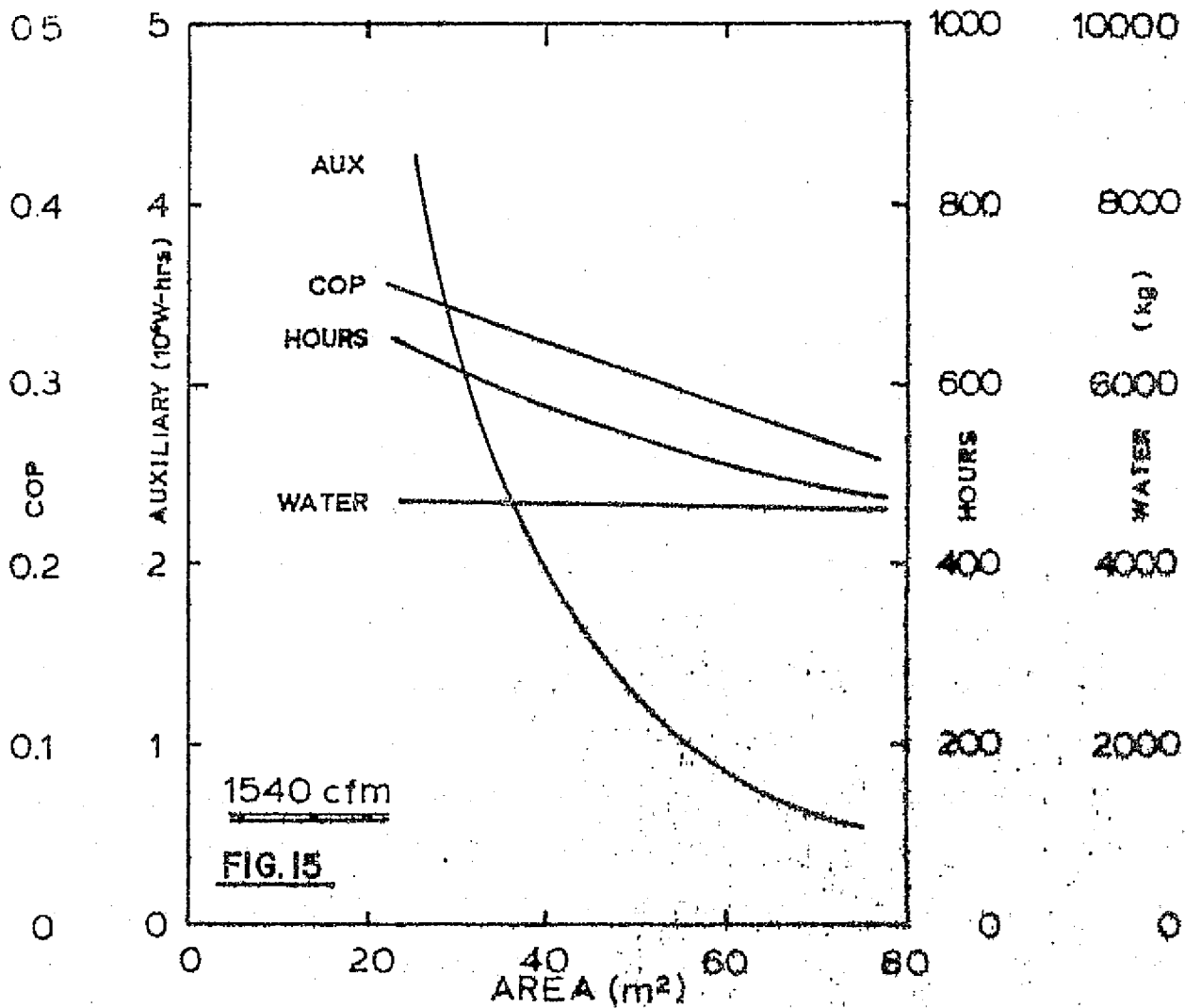
FIG. II

B-24

RECIRCULATION MODE OPTION 3

FIG.12

B-25

VENTILATION MODE

1540 cfm

FIG. 13

55m²

FIG. 14

FLOW RATE (cfm)

RECIRCULATION MODE OPTION 1

FIG. 15

1540 cfm

FIG. 16

55m²

FLOW RATE (cfm)

B-27

RECIRCULATION MODE OPTION 2

1540 cfm

FIG. 17

55m²

FIG. 18

FLOW RATE (cfm)

B-28

RECIRCULATION MODE OPTION 3

FIG. 19 — 1540 cfm

FIG. 20 — 55 m²

FLOW RATE (cfm)
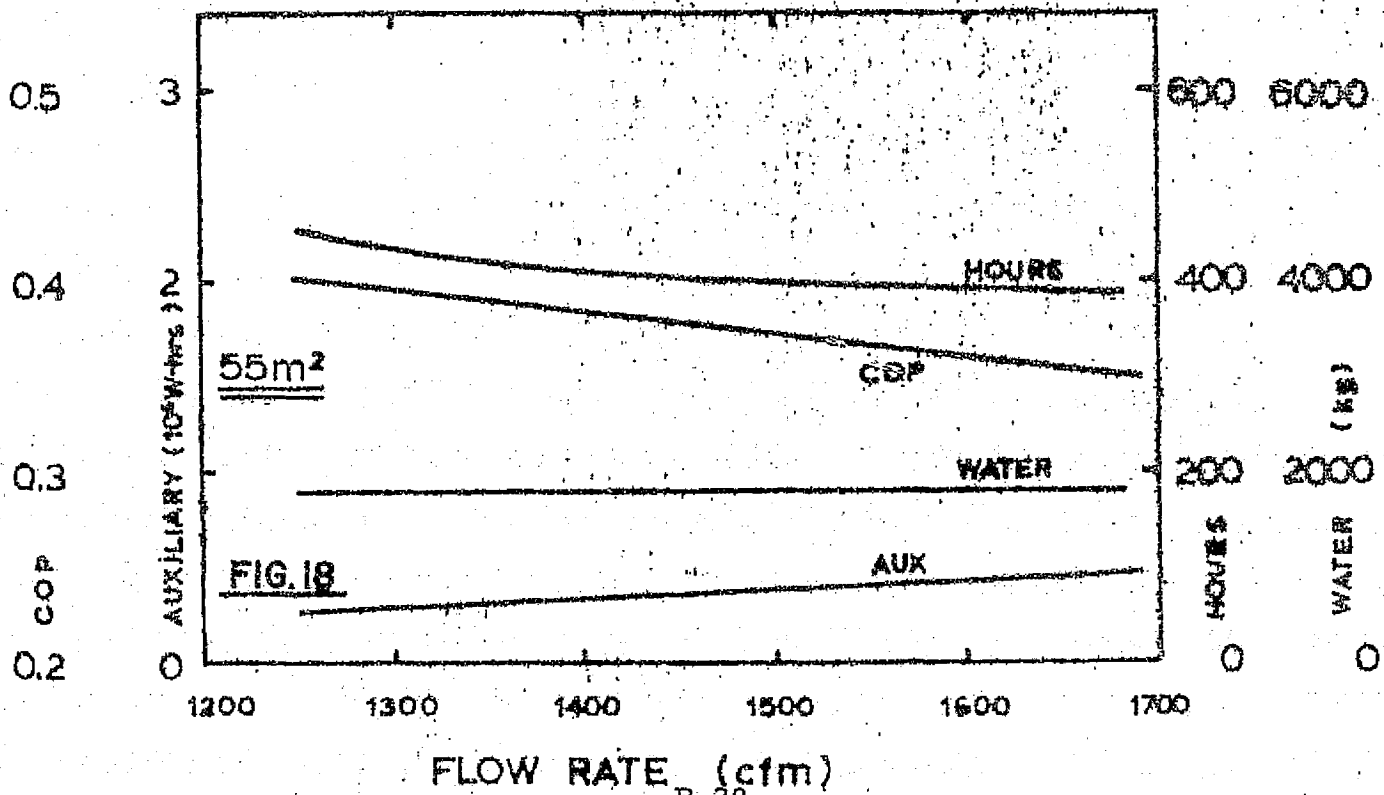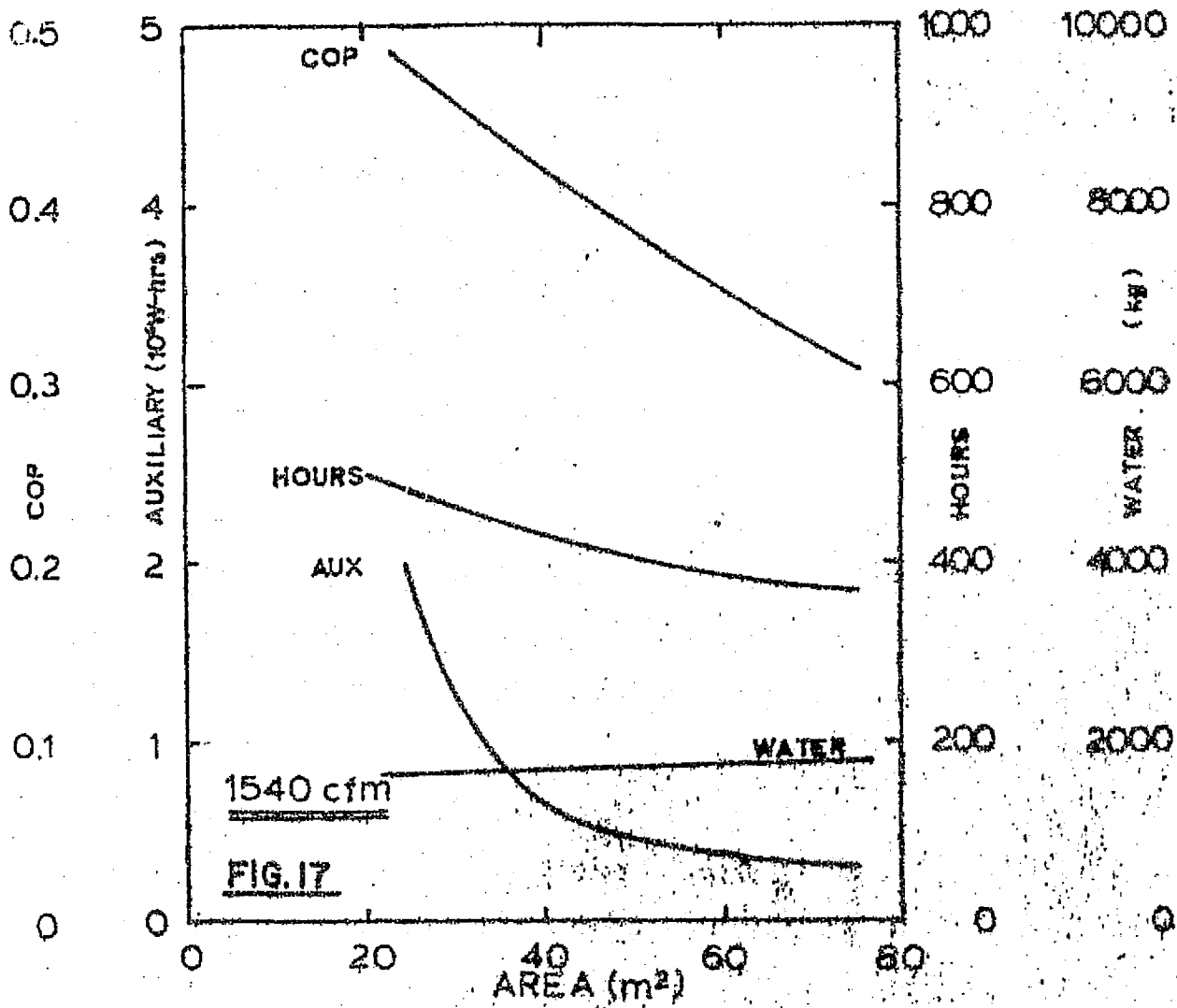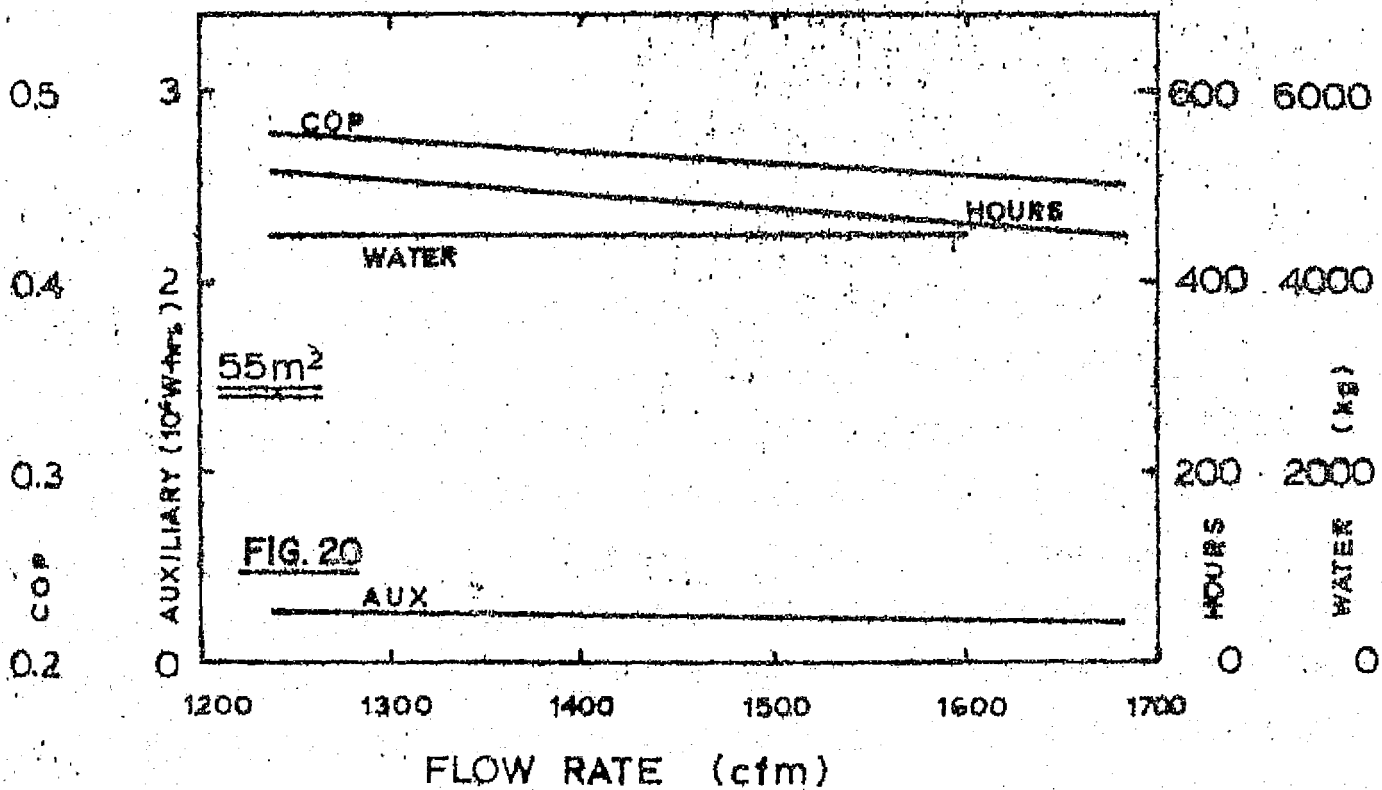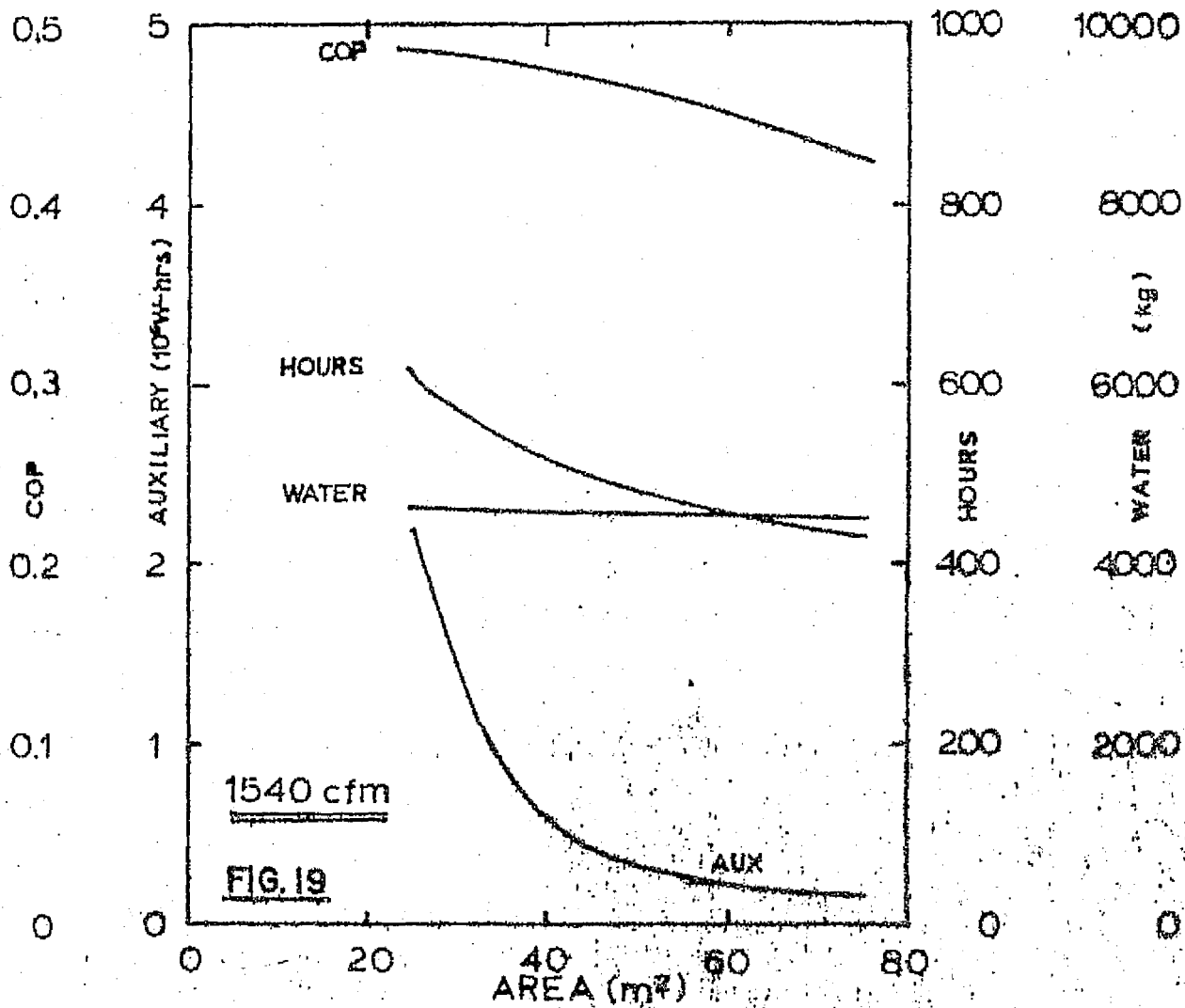
SOLARON DESICCANT RECIRCULATION CYCLE
HEATING AND COOLING SYSTEM

COLLECTOR

MDIa

DOMESTIC
WATER
COIL

MDIb

OUTSIDE
SUPPLY
AIR

MD2a MD2b MD7 MD6

AUX. HEATER

SUPPLY
AIR

ROCK
BED

MD3

MD5

EVAP.
COOLERS

SENSIBLE HEAT
EXCHANGER

CONDITIONED
SPACE
(HEATING & COOLING)

AUX.
HEATER

DESICCANT
WHEEL

BD2 BDI

RETURN
AIR

MD4

EXHAUST

FIG. 21

B-30
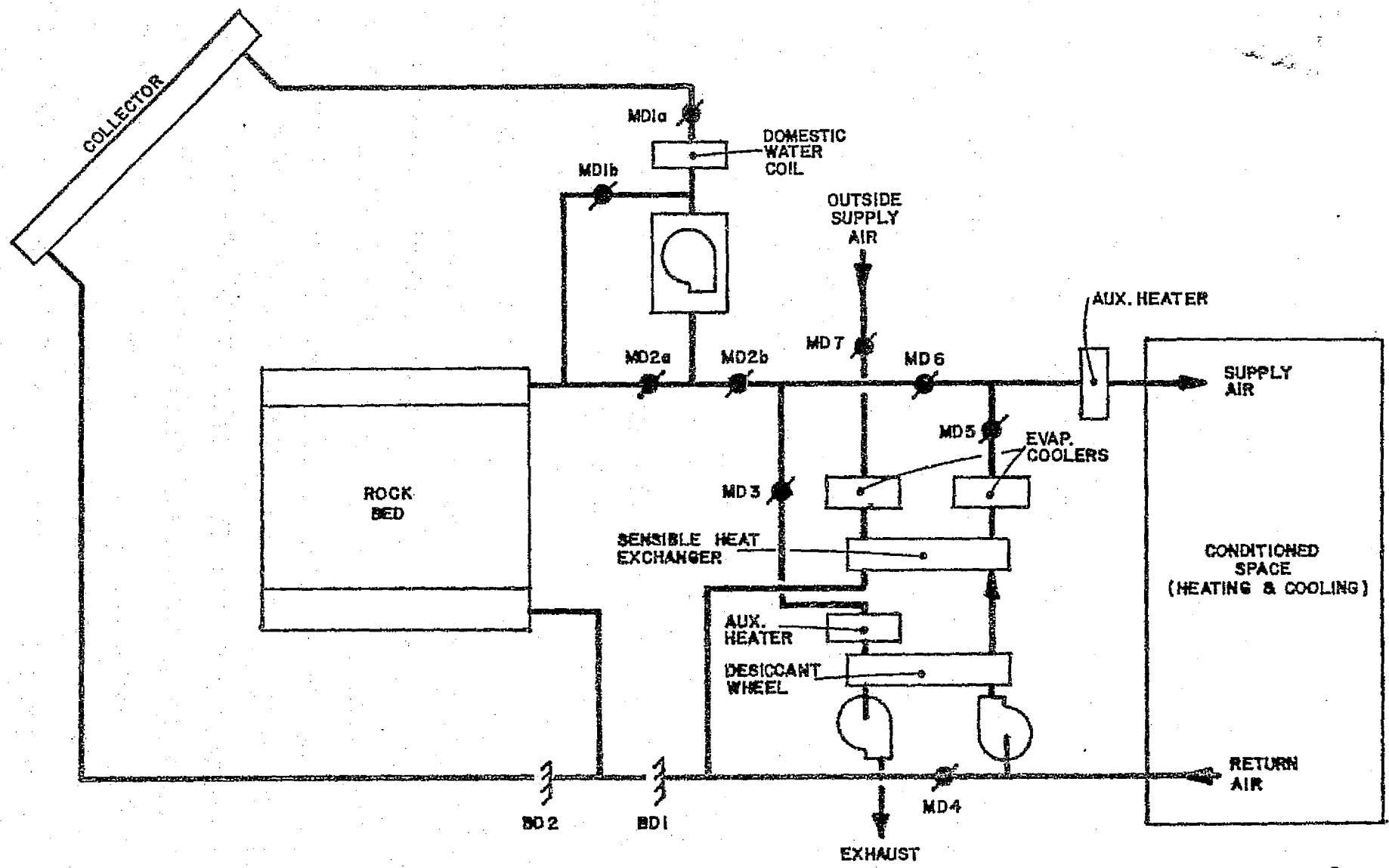
ATTACHMENT III

AKRON HOUSE DATA SHEET

1. OWNER.......AKRON METROPOLITAN HOUSING AUTHORITY

2. LOCATION....ORCHARD PARK, NE AKRON, OHIO

3. LIVING AREA....APPR. 1200 FT.^{2}

4. ROOF ORIENTATION....FACING DUE SOUTH
        SLOPE.........45^0

5. CLIMATIC DESIGN CONDITIONS:

    LATITUDE.....................41^0 0' N
    ELEVATION....................1210 FT. ABOVE SEA LEVEL

    WINTER DESIGN:

        MEDIAN OF ANNUAL EXTREMES.....-5^{0}F
            97½% DESIGN..............+6^{0}F

        COINCIDENT WIND VELOCITY......M (M = MODERATE,
                    50 TO 74^0 EXTREME HOURS > 7 MPH)
        DEGREE DAYS..................6037

    SUMMER DESIGN (2½% DESIGN):

        DESIGN DRY BULB..............87^{0}F
        DESIGN WET BULB..............73^{0}F
        OUTDOOR DAILY RANGE..........21^{0}F

6. HOUSE DESIGN LOADS

    WINTER (HEATING)   SPACE............28,759 BTU/HR.
                        HOT WATER........ 2,231 BTU/HR.

                        TOTAL   30,990 BTU/HR.

    SUMMER (COOLING)......................21,600 BTU/HR.

7. SOLARON COLLECTORS: 28 @ 19.5 FT.^{2} EA. = 546 FT.^{2} TOTAL
    SOLAR ANNUAL PERCENT OF HEATING LOAD: 54.4%
    SOLAR ANNUAL PERCENT OF COOLING LOAD: APPR. 90%
    (BASED ON COLUMBUS, OHIO SOLAR DATA)