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A Study of an Integrated Appliance, the Air Conditioner/Heat Pump-Heat Recovery Unit-Water Heater

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A STUDY OF AN INTEGRATED APPLIANCE, THE AIR CONDITIONER/HEAT PUMP-HEAT RECOVERY UNIT-WATER HEATER

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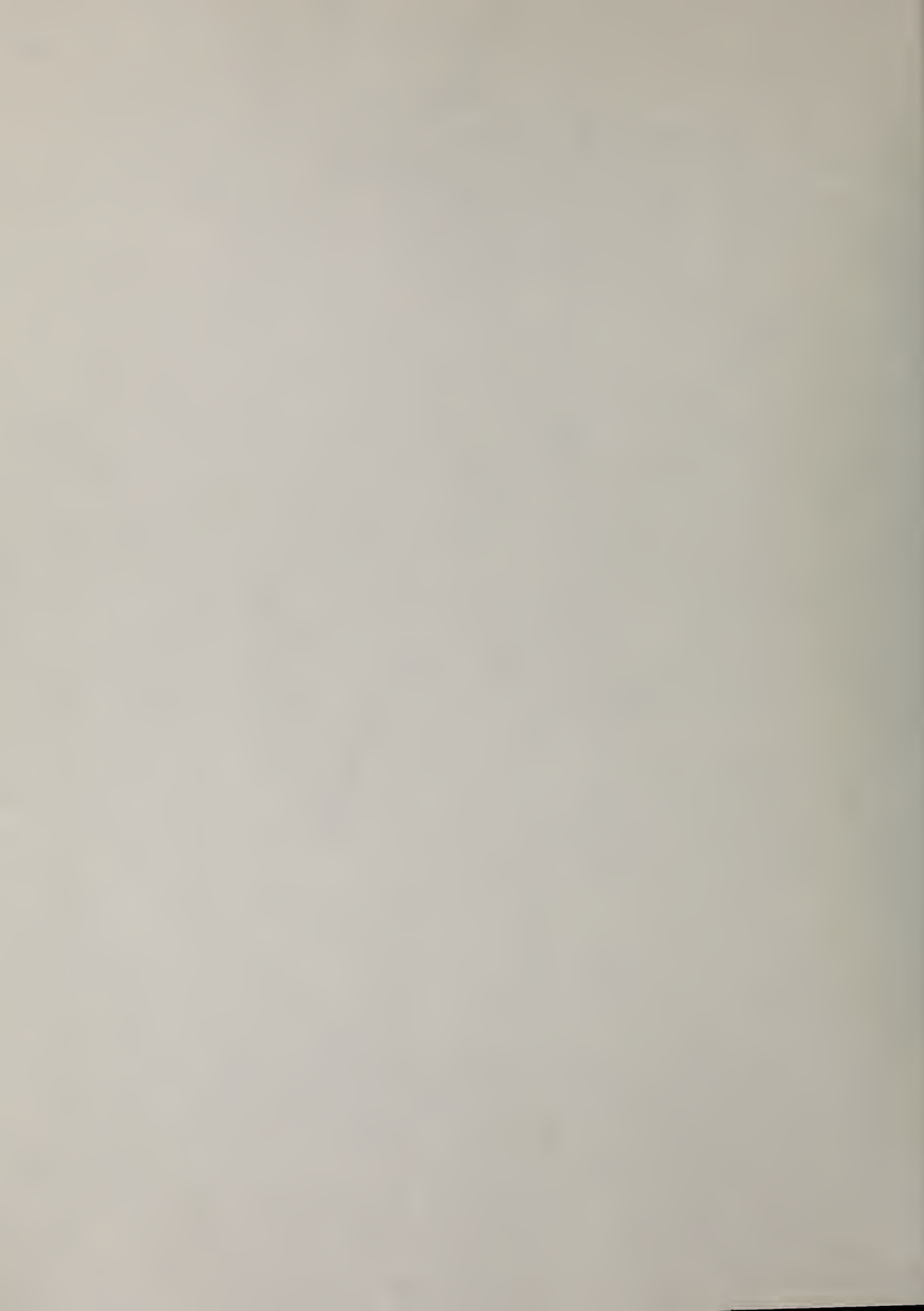
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

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Abstract

Three integrated heat pump-heat recovery unit-water heater appliances were
tested under various environmental conditions to measure the functional
parameters and study the operating characteristics of these systems. It was
found that the heat recovery, heat recovery rate and heat recovery efficiency
were dependent on the heat recovery unit's characteristics. The use of the heat
recovery unit in the system resulted in a reduced work load for the heat pump's
compressor and slightly improved the heat pump's performance.

A computer simulation model of the integrated system was developed to study
the interactions between several of the pertinent system variables on an hourly
basis for selected situations and to estimate energy savings. Two alternative
estimation methods that utilize five-degree temperature bin data were also
developed. The estimated savings determined by using the alternative methods
were about the same as those estimated using the hourly data. Conclusions were
also reached concerning the use of water heaters with different tank capacities
and on methods of increasing potential energy savings.

Summary

Three different integrated heat pump—heat recovery unit—water heater appliances were tested under various environmental conditions in order to measure and study the operating characteristics and parameters of these systems. Tests were conducted on the same heat pump and water heater with two different heat recovery units to determine the heat recovery, heat recovery rate and heat recovery efficiency. For these same two integrated systems, the heat recovery, heat recovery rates and heat recovery efficiencies were substantially greater during the cooling mode operation than during the heating mode operation. The heat recovery units were installed and used in the systems as received from the manufacturers. No adjustments were made to optimize or otherwise alter the as-received performance, Since the third system was discovered to be operating improperly, only a limited discussion of this system is included in this report.

It was also determined that the heat recovery units had no adverse effects on the heat pumps' performance. In fact, they slightly improved the performance by reducing the work load on the compressor.

A modified integrated appliance system that utilizes a water heater with an upper electrical resistance heating element and a heat pump with supplemental resistance heat was analyzed to estimate energy savings potentials. A computer simulation model was developed to study the interactions among several of the pertinent system variables on an hourly basis for selected situations. Information generated by the computer simulations for four locations in the U.S. indicated yearly point-of-use savings in the range 800-1200 kWh, representing 40-60 dollars for an electricity cost rate of 5 cents per kWh, for a particular residential application. Since hourly data are available for few locations, specific house characteristics and operating strategies, two alternative estimation methods were developed that use five-degree temperature bin data, which are available for numerous locations throughout the U.S.

It was also concluded from the analysis that: (1) the use of a 52 gallon water heater tank would permit greater flexibility in scheduling residential hot water usage than would a 40 gallon tank but it would add about 3 dollars a year in operating costs due to jacket losses: (2) the use of tanks with more than a 52 gallon capacity increase energy costs without providing sufficient additional benefits; (3) energy savings could be increased by controls which account for the thermal efficiency of the heat recovery unit and utilizing ventilation when it can achieve and/or maintain satisfactory indoor conditions; and (4) the use of averaged weather data for energy estimation and decision purposes, by consumers, is justified.

I. Introduction

The work described and analyzed in this report was performed by the Product Performance Engineering Division (PPED) of the Center for Consumer Product Technology (CCPT), National Bureau of Standards (NBS), in partial fulfillment of "Task B, Air Conditioner/Heat Pump - Water Heater Project." Funding for and management of the project, in accordance with Interagency Agreement DOE NO. EW-78-I-05-5750, was provided by the Department of Energy through the Oak Ridge National Laboratory (ORNL).

The purposes of the project were to gain a better understanding of AC/HP-HRU-WH systems and how they may contribute to energy conservation, and to identify areas in which potential improvements may result in increased energy savings.

The "Air Conditioner/Heat Pump - Water Heater" specified in the task title is a new and currently available integrated major home appliance whose principal components are a central air conditioner (AC) or heat pump (HP), a heat recovery unit (HRU), and a storage type water heater (WH). In view of these principal components, the integrated appliance will hereinafter be abbreviated as AC/HP-HRU-WH or, when a heat pump is used, as HP-HRU-WH.

The purpose of integrating the three principal components is to construct an appliance that will use less energy than would ordinarily be used by the AC/HP and the WH. In the AC/HP-HRU-WH, the AC performs its normal cooling function and the HP its normal heating and cooling functions, and the WH stores and provides potable hot water for normal use. The HRU is the added unifying component that makes energy savings possible; its function is to provide the means for and to control the transfer of thermal energy from the superheated refrigerant gas discharged by the compressor of the AC/HP to the water in the WH tank. Figures I-1 and I-2 show, schematically and respectively, the major components of an AC/HP and a typical pressure - enthalpy diagram for the refrigerant cycle in an AC/HP.

The HRU has two inlets, one for superheated refrigerant gas that is discharged from the compressor of the AC/HP and the other for water from the drain outlet of the WH, and two outlets, one to allow the refrigerant gas to flow to the condenser of the AC/HP, and the other to allow the water to flow to the hot water outlet of the WH. Inside the HRU are a counterflow heat exchanger (HX) and the gas and water flow control system.

Utilization of thermal energy that would otherwise be wasted by discharging it into the outdoor air during cooling mode operation results in energy savings as less conventional energy is required to attain and maintain the desired water temperature in the WH tank. As a result of the advertized claims of energy savings and user interest in saving energy, thousands of AC/HP-HRU-WH systems have been installed in residences, supermarkets, restaurants, hotels, hospitals and other structures. Some of the installed AC/HP-HRU-WH systems are of the retrofit type, and others are supplied as complete or almost complete systems. In the retrofit installation, an appropriately sized HRU is added to interface with a previously installed AC/HP and WH. In the complete or almost complete

system installation, an entire AC/HP-HRU-WH system is installed, or an integrated AC/HP-HRU system is interfaced with an existing WH. Few companies manufacture the entire AC/HP-HRU-WH system or an integrated AC/HP-HRU system. Many companies manufacture an assortment of HRUs for retrofit purposes.

The obvious goal in using the AC/HP-HRU-WH integrated appliance in commercial or residential applications is to optimize energy usage, i.e. maximize energy and dollar savings, while performing the required cooling or heating functions and satisfying normal hot water needs. However, there are many technical factors involved in the operation and assessment of potential energy savings of such systems that have not been fully addressed and, therefore, require investigation. Some of these factors are ambient conditions; incoming, storage and operating water temperatures; hot water draw schedules; water storage tank size; and the heat recovery unit design.

Ambient conditions have a significant effect on the thermal energy available for heating water. During the cooling season, for example, higher outdoor temperatures result in greater quantities of thermal energy available for recovery. During the heating season, thermal energy available for heating water decreases.

Incoming water temperatures are higher in southern than in northern states; storage temperatures, in the water heater, are dependent upon the settings of the water heater thermostatic control; and operating water temperatures will depend upon the user.

Hot water draw schedules vary from application to application dependent upon the number of users and appliances. Furthermore, even draw schedules for a particular application may vary daily, and from month to month, depending upon any number of water usage variables, such as for bathing, cooking, dishwashing and clothes washing, and different hot water temperature preferences.

Water storage tank size will also affect the energy savings potential. Since the storage tank is to provide adequate amounts of water at sufficiently elevated temperatures, it is important to select a WH tank size that will have sufficient capacity with an appropriate backup heater to handle major draws. If the heat recovery unit in the AC/HP-HRU-WH is designed to heat large quantities of water using a high flow rate, the increase in water temperature will be slight. However, if the design is to handle smaller flow rates, the increase in water temperature will be large. In either case, a prolonged draw rate may result in depletion of the hot water if the storage tank capacity is too small for the application.

The HRU must be properly designed for effective heat recovery. Some manufacturers of HRU have claimed that their use in the integrated appliance not only provides "free heat" for heating water, but it also improves the capacity and efficiency of the AC or HP. However, others have indicated that such improvement is marginal at most. Manufacturers of compressors used in AC and HP have expressed concerns about the possibility that use of the HRU may increase compressor head pressure which may lead to premature compressor failure.

Two different types of HP and three different HRU were selected for this investigation because they were judged to be typical of the integrated AC/HP-HRU-WH systems presently in the field. When operated in the cooling mode, the HP is considered to be an AC. These systems are described in the section dealing with the apparatus and test setup.

The investigation consisted of two major phases. One was an experimental phase during which the HP was operated with and without the HRU and WH. This was to verify the manufacturers data for the HP, to determine how the integrated appliance performed, and to provide the technical basis on which a test procedure for evaluating the performance of the HRU may be developed. In the other phase a computer model was developed to study the interactions between several pertinent system variables for selected situations. Analytic methods were developed to utilize the performance data in estimating and optimizing energy savings.

II. Performance of Existing Systems

A. Typical AC/HP—HRU—WH System Operation

A pressure-enthalpy diagram for a refrigeration cycle is presented in Figure II-1. The enthalpy differences h_C-h_B, h_D-h_C, and h_D-h_A represent, respectively, the cooling capacity, the compressor work input and the heating capacity. If losses are ignored, the thermodynamic coefficient of performance for cooling, COP_C, is calculated as

$$COP_C = \frac{h_C-h_B}{h_D-h_C}$$

and for heating, COP_H, is calculated as

$$COP_H = \frac{h_D-h_A}{h_D-h_C} \ .$$

The superheat portion for the refrigerant in this cycle, following discharge from the compressor, is between points D and F in the figure. Typical refrigerant temperatures are $220^\circ F$ at point D, the compressor discharge, and $115^\circ F$ at point F, the condensing temperature. The amount of thermal energy that might be utilized in heating water is determined by the enthalpy difference, (h_D-h_F), times the refrigerant mass flow rate. If the controls in the HRU confine the water flow only to desuperheating the refrigerant gas, then the quantity of thermal energy available for heat recovery would be reduced to (h_D-h_F) times the refrigerant mass flow. An important parameter that is used in Section III of this report concerns the thermal energy available for water heating and is called the "water heating factor"; it is defined as the ratio of the heat available for water heating per pound of refrigerant to the

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corresponding cooling or heating effect. Its variability with respect to outdoor dry-bulb temperature is shown in Figure III-3a.

That part of the cycle between points G and A represents refrigerant subcooling. In passing through a metering device, generally a capillary tube or thermostatic expansion valve, the refrigerant pressure drops from the high side (or discharge) pressure, at point A, to the low side (or suction) pressure, at point B. From point B to point B', the refrigerant passes through the evaporator absorbing thermal energy and vaporizing. Between points B' and C the refrigerant is superheated (about 15°F in the example shown) before entering the compressor. The compressor raises the refrigerant pressure and temperature from point C to point D.

B.   Apparatus and Test Setup

The performance of each of three HP-HRU-WH integrated appliances was investigated in this phase of the study. The common major components of the first two integrated appliances, or systems, are a HP (cooling capacity of 35,000 and 38,000 Btu/h at outdoor temperatures of 95 and 75°F, respectively; and heating capacity of 34,000 and 39,300 Btu/h at an outdoor temperature of 40 and 50°F, respectively) and a nominal 52 gallon storage type WH whose electric heating elements were disconnected for this work. The third major component of the first system was a retrofit HRU intended for use with an AC or HP rated for 18,000 to 60,000 Btu/h. The third major component of the second system was an HRU produced by the manufacturer of the HP described above and intended for use with any AC or HP rated for 18,000 to 42,000 Btu/h.

A schematic representation of the these two integrated systems, referred to as System 1 and System 2, are presented in Figure II-2. The figure shows an indoor unit, called an air handler, in the "indoor room" of a three room environmental chamber. This chamber was set and maintained at a nominal 80°F and 50% relative humidity (RH) when the HP was operated in the cooling mode, and a nominal 70°F and 50% RH when the HP was operated in the heating mode. The coil in the indoor unit served as the evaporator coil during cooling mode operation and the condenser coil during heating mode operation. The outdoor unit was located in the "outdoor room" of the environmental chamber. This chamber was set and maintained at nominal 95, 85 and 75°F, and 50% RH, during cooling mode operation, and a nominal 20°F during heating mode operation. The reversing valve, accumulator, compressor and capillary tube of the outdoor unit are identified in the figure. The coil in the outdoor unit served as the condensing coil during cooling mode operation, and the evaporator coil during heating mode operation. The "basement room" of the environmental chamber, which was set and maintained at a nominal 65°F and 50% RH, contained the HRU and the WH. Water flow direction is shown in the "basement room", and refrigerant flow direction is shown in all rooms for cooling mode (black arrows) and heating mode (white arrows) operations.

Schematics of the HRUs used in System 1 and System 2 are presented in Figures II-3 and II-4, respectively. The water and refrigerant inlets and outlets of the HRU are appropriately identified. For both systems, the refrigerant flow through the heat exchanger in the HRU is governed by the condenser and evaporator temperatures seen by the compressor.

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In System 1, the water circulating pump of the HRU operates when the HP compressor operates and the inlet water temperature sensed by the thermally controlled switch is less than about 150°F. When the inlet water temperature rises to about 150°F, the switch deactivates the circulating pump. When the inlet water temperature decreases to about 140°F, the switch reactivates the circulating pump. Although the circulating pump is the driving force for circulating water, the water temperature at the modulating flow control valve must exceed some set value which in this case was $140 + 5^\circ$F before the valve opens allowing the water to flow. For this HRU the control valve temperature is settable in the range of $120-160^\circ$F. The circulating pump has a sealess type impeller with magnetic coupling to the drive motor which has thermal overload protection; it is rated at 1/200 HP, 1725 RPM, 230 V, 60 Hertz, 1 phase and 0.26 amps. Although the maximum rated flow rate of the circulating pump is 3 gallons per minute, the actual flow rate is intermittent and modulated from less than 0.1 gallon per minute to the maximum by the temperature sensitive flow control valve. The heat exchanger in HRU 1 is well insulated, being completely encapsulated in polyurethane foam insulation.

In System 2, the water circulating pump of the HRU operates whenever the HP compressor operates. The flow of water is controlled by a temperature sensitive modulating flow control valve. This valve, commonly called a reverse acting valve, opens when the temperature at its sensor reaches the control temperature, which can be set between 130 and 160°F. When the water temperature at the valve equals or exceeds the set temperature, the valve allows the water to flow. For more rapid thermal response, the valve contains a small internal bleed port, through which a small steady stream of water flows over a temperature sensitive bulb. Another important component in this HRU is the 3-way flow control valve, which directs the incoming water to either the heat exchanger or the heat exchanger by-pass loop, or both, depending upon the temperature of the inlet water. The recommended temperature set point for this valve is 160°F. When the inlet water temperature is below 160°F, it flows through the heat exchanger; between about $160-170^\circ$F both paths are open; and when the inlet water exceeds the set-point by about 10°F, the water by-passes the heat exchanger and will not extract any heat from the hot refrigerant gas. The heat exchanger in HRU 2 is enclosed in a fiberglass insulated compartment that is open on one side and allows venting of the HRU through louvers in the HRU casing.

In both Systems 1 and 2, the heat exchanger is of the counterflow tube-in-tube type. Hot refrigerant gas passes through the outer tube and water flows through the concentric inner tube which has a convoluted outer surface for increased heat transfer on the refrigerant side.

The schematic representation of the third integrated system investigated is shown in Figure II-5. The HP in this system contains a thermal expansion valve and a capillary tube for metering the refrigerant flow during heating and cooling mode operation, respectively. In addition, the heat pump is a single package unit that mounts completely within the outdoor chamber. The rated cooling capacity of this HP is a nominal 36,000 Btu/h. A schematic of the HRU is shown in Figure II-6. The water circulating pump of the HRU operates only when the HP compressor operates. When water is circulating at a temperature below the set temperature, the two solenoids valves are normally open (N.O.) and

normally closed (N.C.), as shown in Figure II-6. When the temperature of the leaving water exceeds the set temperature it causes the temperature sensors on the water outlet line to reverse the two solenoid valves positions and the hot refrigerant gas to by-pass the heat exchanger. When the temperature sensor on the refrigerant gas outlet senses the set temperature of the gas in the gas outlet line it deactivates the circulating pump. It should be noted that all flow control set temperatures may be manually adjusted. The heat exchanger in this HRU is of the counterflow 2-tube type.

For Systems 1 and 2 a twelve foot 0.5 inch O.D. copper tube was used to connect the refrigerant outlet from the HP compressor discharge to the refrigerant inlet of the HRU. A similar length of tubing was used to connect the refrigerant outlet of the HRU back to the HP reversing valve. By-pass valves were installed near the compressor discharge to permit the HRU to be switched either into or out of the system. For System 3, two 25 foot 0.5 inch O.D. copper tubes were used to connect the inlet and outlet of the HRU to the refrigerant lines of the HP; no HRU by-pass provisions were made for this system, since such provision is an integral part of the HRU used in this system.

For all systems a six foot 0.5 inch O.D. copper tube was installed to connect the water inlet of the HRU with the drain outlet of the WH, and a ten foot 0.5 inch O.D. copper tube was installed to connect the water outlet of the HRU with the hot water outlet of the WH. All of the interconnecting tubing between the HP, HRU and WH was insulated with RF-16 Rubatex FS 4003, 5/8 ID x 3/8" wall thickness.

For Systems 1 and 2, sheet metal ducting was installed on the indoor air handling unit to obtain the manufacturer's specified airflow rate and pressure drop across the indoor coil. For System 3, the airflow rate through the indoor side was adjusted to the manufacturer's specifications by use of a baffle installed in the indoor section. The proper airflow pattern was obtained by using an air deflector.

Standard copper-constantan thermocouples were used to measure the following: (1) dry-bulb and wet-bulb temperatures of the air on the downside of the coil within the indoor unit and at the air inlet and outlet of the HP indoor unit, (2) dry-bulb temperatures in each of the three environmental compartments ("outdoor", "indoor" and "basement"), (3) water temperatures entering and leaving the HRU, (4) water temperatures at 6 different levels in the WH, and (5) temperatures at various locations along the refrigerant flow line.

Calibrated pressure gauges were used to measure high side and low side refrigerant pressures in the refrigerant line. Calibrated watt-hour meters were used to record the electrical energy used by (1) the compressor and outdoor fan and (2) the indoor side blower. The locations of the thermocouples, pressure gauges and air flow meters are indicated in Figure II-2 for Systems 1 and 2. The six thermocouples in the WH are attached to a "thermocouple tree"; their junctions were located at the level of the central horizontal plane of each of 6 non-overlapping but approximately equal volumes from the top to the bottom of the WH tank.

Finally, after making all refrigerant line connections to the HRU, refrigerant R22 was added to the system in accordance with the HRU manufacturer's instructions. During this refill, all valves between the HP compressor and reversing valve were open and the "indoor", "outdoor" and "basement" room temperatures were equal, at approximately 75°F.

C.   Test Procedure

The test procedure generally followed during the experimental phase of this study was to begin operation of the HP with the HRU in the line (not by-passed) and record the relevant data every half during the first 5.5 hours of HP operation, after which time the HRU was by-passed. With the HRU by-passed, the HP operation was continued for an additional 1.5 hours with measurements made each half hour before the run was terminated.

Heating elements in the water heater were disconnected and no water was drawn during the test periods. The indoor conditions were controlled to maintain nominal values of 50 percent relative humidity with an indoor room temperature of 80°F during cooling mode operation and 70°F during heating mode operation. The series of tests for each of the systems was run in order with a nominal "outdoor" temperature of 20, 95, 85, and finally 75°F.

D.   Test Results and Analysis

The data recorded during the experimental investigation of integrated HP-HRU-WH Systems 1 and 2 were used to develop Figures II-7A-C through II-16-A-C, which depict water temperatures, heat recovery, refrigerant temperatures, refrigerant pressures, and energy consumption as a function of time during continuous HP operation. In addition, these data were also utilized in the development of Table II-1, which presents thermodynamic performance data for the HP with and without the HRU in the refrigerant line; Table II-2, which presents HP capacity, power, and efficiency performance data with and without the HRU in the refrigerant line; and Table II-3, which presents mean heat recovery rate, thermal energy rate, energy consumption rate and heat recovery efficiency.

Furthermore, it should be noted that due to time constraints of stabilizing the environmental chamber in which the tests were conducted and limitations on personnel availability it was not possible to carry out any runs to the point where the HRU control systems would shut down the heat recovery operation. Thus, the design cutout temperatures were never determined. This problem could have been alleviated by using a smaller water storage tank, but this would not have been consistent with the objective of determining the time required for a complete recovery with a standard 52 gallon storage type water heater. The recovery time for the two systems tested is obviously greater than 5.5 hours., the maximum of any of the runs.

It was found that there were unexplanable inconsistencies in the data for the cooling mode run with nominal outdoor temperature of 85°F for System 1 and the heating mode run for System 2; thus, no data for these runs are presented.

Before discussing the figures and tables, an explanation of why they do not include System 3 is appropriate. While running System 3, it was apparent that

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the heat recovery was much less than expected. However, the runs were continued in the belief that the system was performing as it should and that its heat recovery performance was just poor. This unsatisfactory performance, which was highlighted during the analysis of data, led to a trouble shooting investigation of why the performance was so poor. From this investigation it was determined that the automatic by-pass control, which by-passes the refrigerant around the heat exchanger when the water in the tank is sufficiently hot, operated incorrectly during these runs, resulting in invalid test data. In other words, the poor heat recovery had resulted from the refrigerant by-passing the heat exchanger when it should not have, due to a faulty automatic by-pass control. Since the data taken for System 3 are invalid they are not presented or further discussed in this report, nor is it planned to repeat these tests at this time.

1. Figures for System 1

    Figures II-7A-C show the water temperatures within the storage type WH as a function of time during continuous HP operation over a period of 5 or 5.5 hours. As may be expected, the system provides more hot water at usable temperature when the outdoor temperature (T_o) is higher than when it is lower. When $T_o = 20°F$, little usable hot water is provided. In addition, the figures also show the thermal stratification in the water tank and the decrease in the time required for all of the water in the tank to reach an "equilibrium" temperature, of about 140-145°F, as the T_o increases. The cut-in and cut-out temperatures are the maximum and minimum temperatures respectively, at the water outlet of the HRU. These temperatures are influenced by the location and temperature setting of the flow control valve, its operational temperature tolerances, the heat exchanger insulation and heat transfer characteristics, the lengths of piping between the heat exchanger and the flow control valve, and the insulation on the copper tubing within the HRU. As the water in the heat exchanger increased in temperature, the water near and at the flow control valve also increased in temperature, by thermal conduction. When the flow control valve sensed a high enough temperature it opened. Since the temperature of the water between the flow control valve and the heat exchanger and that in the heat exchanger was higher than that at which the flow control valve opened, flow continued until water passing through the heat exchanger gained insufficient thermal energy to maintain a temperature high enough to keep the flow control valve open.

    Figures II-8A-C show the increase in heat recovery as a function of time of system operation. The values used to plot the curves were calculated using the water heater thermocouple tree temperature data taken each half hour. It is apparent that heat recovery during cooling mode operation is substantially larger than during heating mode operation. These curves also serve as a basis for determining some of the heat recovery rates shown in Table II-3 determined for the period between 1 and 3 hours following the start of system operation, as will be described below.

    Figures II-9A-C show the temperatures of the refrigerant gas discharged from the compressor (T_{CDG}), entering the HRU (T_{RGE}), leaving the HRU (T_{RGL}), and entering the compressor (T_{SGE}) as a function of HP

continuous operating time. In the heating mode of operation with T_o = 20°F, the temperature of the compressor discharge gas at the inlet to the HRU is relatively low, approximately 155°F; thus, the relatively small magnitude of the refrigerant gas enthalpy in the superheat region results in a low heat recovery rate. In the cooling mode of operation the heat recovery rates are much larger than for the heating mode, since the higher refrigerant gas temperature at the compressor discharge provides a much higher enthalpy and, consequently, greater quantities of thermal energy are available for the HRU. After some time of operation, the rise in the temperature of the refrigerant gas leaving the HRU (T_{RGL}) reflects a decrease in thermal energy transfer due to the rise in the temperature of the water returning to the HRU from the WH tank. This occurred after approximately 2.0-3.5 hours for this system when operating in the cooling mode, Figures II-9B-C. When the HRU was removed from the refrigerant line, there appeared to be a slight initial drop, followed by a slight rise, in the refrigerant gas temperature at the compressor discharge port, T_{CDG}.

Figures II 10A-C show the refrigerant pressures at the compressor discharge (P_{CD}), downstream of the 12-foot gas line from the HRU (P_{RG}), just downstream of the condenser (P_{RLC}), and in the suction line to the compressor (P_{CS}) as a function of time of continuous HP operation. The discharge pressure is more than 50 psi higher for T_o = 95°F than for T_o = 75°F, while the suction pressures differ by only 10 psi for this 20°F variation of outdoor temperatures. The pressure drop due to the HRU and the two 12 foot tubing lines connecting the HRU to the HP is about 5-8 psi at T_o = 75°F and 8-10 psi at T_o = 95°F; it is only 2-4 psi at T_o = 20°F. The pressure drop between the compressor discharge and the condenser outlet decreased from an average of about 17.4 psi with the HRU in the refrigerant line to an average of about 15.3 psi with the HRU by-passed, during cooling mode operation. During heating mode operation the pressure difference averaged about 40.8 psi. Suction pressure generally rose slightly when the HRU was by-passed.

Figures II-11A-C show the total electrical energy consumed by the indoor blower, condenser fan and compressor as a function of HP continuous operating time. As may be noted, the rate of energy consumption shown in Table II-3 remained essentially constant whether or not the HRU is in the refrigerant line.

Mean heat recovery rates for the runs at nominal T_o = 20, 75 and 95°F for System I were determined from the difference between the total heat recovery at 1 and 3 hours following the start of system operation, see Figures II-8A-C, and are presented in Table II-3. The thermal energy rate of the refrigerant gas was determined from the difference between the inlet and outlet enthalpy and the refrigerant mass flow rate at 2 hours following the start of system operation. The refrigerant pressure used in the determination of the inlet and outlet enthalpies was taken as the mean of the compressor discharge pressure, P_{CD}, and the outlet pressure of the HRU, P_{RG}, at the 2 hour mark, see Figures II-10A-C. The inlet and outlet temperatures used in determining the enthalpies were T_{RGE} and T_{RGL}, were taken at the 2-hour mark, see Figures II-9A-C. Refrigerant mass flow rate

was determined from the manufacturer's data for this compressor at the appropriate evaporator and condenser temperatures.

2. Figures for System 2

Figures II-12A-C are similar to Figures II-7B,C, showing that the higher the T_o, the more hot water was provided at usable temperatures. The hot water in the WH tank generally reached a first pass mean temperature of about $135-138^{\circ}F$ with a cut-in temperature of $140^{\circ}F$ as compared to the $140-145^{\circ}F$ mean temperature and $149^{\circ}F$ cut-in temperature for System 1. Obviously, the temperature settings of the flow control valves in System 1 and 2 as set by the two HRU manufacturers, were not identical.

Figures II-13A-C are similar to Figures II-8B,C and were used in the same manner to calculate the experimental mean heat recovery rates shown for System 2 in Table II-3.

Figures II-14A-C are similar to Figures II-9B,C. The compressor discharge gas temperature, T_{CDG}, for the $T_o = 75^{\circ}F$ run was higher than expected because of the increased load that resulted from an excessively high indoor temperature (Table II-1). The result of this condition was an increased heat recovery for the $T_o = 75^{\circ}F$ run to about the same magnitude as for the $T_o = 85^{\circ}F$ run, as may be seen by comparing Figures II-13A-B. For the $T_o = 95^{\circ}F$ run, the magnitude of T_{RGL} at 5 hours of operation is much greater than for the other runs, due to the increased temperature of incoming water (Figure II-12C) that resulted from a high heat recovery rate. Thereafter, the small difference between T_{RGE} and T_{RGL} resulted in a decrease in heat recovery, as seen in Figure II-13C. About 2.5-3.5 hours of cooling mode operation were required before T_{RGE} began to increase.

Figures II-15A-C are similar to Figures II-10B,C. As occurred in System 1, the discharge pressure of System 2 increased substantially with T_o while the suction pressure increased only slightly. The pressure drop due to the HRU and the two 12 foot lines connecting the HRU to the HP was relatively constant at 8-10 psi for the runs at $T_o = 75$ and $85^{\circ}F$ and 6-8 psi for the run at $95^{\circ}F$. Removal of the HRU from the refrigerant line resulted in an apparent increase in the discharge pressure for the $T_o = 75$ and 85° runs and little change for the $T_o = 95^{\circ}F$ run. The pressure drop between the compressor discharge and the condenser outlet, during cooling mode operation, decreased from an average of about 16.8 psi to an average of about 8.4 psi, when the HRU was by-passed. Suction pressure generally increased slightly when the HRU was by-passed.

Figure II-16A-C are similar to Figures II-11B,C. The total energy consumed by the indoor blower, condenser fan and compressor increased linearly with time and directly with T_o. The rates of energy consumption for cooling mode operation did not noticebly change when the refrigerant gas was rerouted to by-pass the HRU.

The same methods used for calculating the mean heat recovery rates for the WH and thermal energy rates for the refrigerant for System 1 were

used for System 2. The results of these calculations are also summarized in Table II-3.

3.   Tables for Systems 1 and 2

Table II-1 shows the time averaged ambient conditions and thermodynamic performance data for the HP operating by itself and as part of an integrated HP-HRU-WH system. As may be noted, the condenser and evaporator pressures and temperatures, the compressor discharge pressures, the compression ratios, and the refrigerant mass flow rates were consistently lower when the HRU were in the refrigerant line than when they were by-passed. The compressor discharge temperatures, work of compression, and refrigeration effect were almost always larger when the HP operated as part of the HP-HRU-WH system rather than as an independent unit. Finally, the values calculated for the thermodynamic coefficient of performance during cooling mode operation and the cooling capacity when the HP operated with the HRU in the refrigerant line were about equal to those calculated for the HP when it operated as an independent unit.

Table II-2 presents overall performance data, which includes all fans and pumps, and shows the capacity, power and efficiency of Systems 1 and 2, and the HP data provided by the HP manufacturer. Agreement of the experimental data with the manufacturer's data for the HP was almost spectacular when it is considered that this was a reproducibility experiment which compared a specific unit with a representative average for the manufacturer's product. As for the average power, it may be noted that using the HRU resulted in a decrease in power usage. The EER, calculated by dividing the capacity by the power, almost always increased (mainly due to the decreased power requirement) when the HRU were used.

These data indicate that the HRU effectively increased the capacity of the condenser, which led to an improvement in the performance of the HP, since the increase in pressure drop through the condenser was not excessive. In addition, the reductions in the compressor discharge pressure and compression ratio indicate less loading on the compressor, which would tend to improve the operation and prolong the useful lifetime of the compressor.

Table II-3 summarizes the mean heat recovery rate, thermal energy rate, energy consumption rate, and heat recovery efficiency for Systems 1 and 2. The table shows consistently better performance, i.e., greater heat recovery rate and heat recovery efficiency, for System 1 when the HP is operated in the cooling rather than the heating mode. The mean heat recovery rates for the WH and the thermal energy rates for the refrigerant also appear to increase with increasing outdoor temperature. Furthermore, it is apparent that the thermal energy rates for the refrigerant are consistently greater than the mean heat recovery rates for the WH; the differences, which average 7.2 and 8.3 percent during cooling mode operation for Systems 1 and 2, respectively, are to be expected and may be attributed to generalized heat loss characteristics of the HRU. The sizable percentage difference for heating mode operation, for System 1, may

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be attributed to the uncertainty in the determination of the mean heat recovery rate and the heat loss characteristics of the HRU. Regarding the heat recovery efficiency, which is the ratio of the mean heat recovery rate to the energy consumption rate, there appears to be no consistent trend with varying nominal outdoor temperature; although the heat recovery efficiency increased from 1.86 to 1.96 Btu/Wh for System 1 operation at $T_o \cong$ 75 and $95^{\circ}F$, respectively, it was an average of 1.68 Btu/Wh for System 2 operation at $T_o \cong$ 75,85 and $95^{\circ}F$.

E.   Conclusions

The major conclusions drawn from the experimental phase of this work are as follows:

1.   The HRUs used in HP-HRU-WH Systems 1 and 2 were adequately controlled so that they produced no discernable adverse effects on either the HP or the WH.

2.   The use of the HRU in Systems 1 and 2 resulted in improved performance of the HP, mostly due to a decrease in the power requirement for its compressor.

3.   The use of the HRU in Systems 1 and 2 resulted in a reduced work load for the HP compressor, as shown by the decreased discharge pressure and compression ratio.

4.   The heat recovery rate and heat recovery efficiency for System 1 is substantially lower during HP heating mode operation than during HP cooling mode operation.

5.   To increase the water temperature from 50 to $140^{\circ}F$ for a 52 gallon WH tank using a continuously operating HP-HRU-WH system the equivalent of Systems 1 and 2 would require in excess of 6 hours during the cooling season and much longer during the heating season.

III.   Estimation of Seasonal Energy Costs and Savings

A.   Introduction and Principal Findings

Seasonal energy costs and savings refer to a full annual cycle of heating and cooling a house with a HP and meeting hot water requirements. The savings are the differences between total household electricity usage with and without heat recovery, expressed in kWh per year. Energy savings may be converted to dollar savings using local rate schedules for the time frame of the estimate. Note that marginal cost rates vary with locality, amount of energy used, power company success in obtaining higher rates during the estimation period, and in some cases the type or class of service provided to the particular residence. Hourly simulations were conducted in which house construction and operation were held fixed.

Weather was varied by using data from Washington, D. C., Atlanta, GA, Chicago, IL, and Seattle, WA. Estimated savings are in the approximate range of 800 to 1200 kWh per year, or $40 or $60 per year at $0.05 per kWh. (See Table III-1)

Simulation details about the house, the heat recovery system and house operation are given below and in Appendix A. Significant operating assumptions were to set the thermostat for cooling at 10°F above the setting for heating and to use ventilation where the results would be acceptable. These assumptions led to a zero house load for more than a third of the hours per year at each of the tested locations. This consideration is important because heat recovery is possible only if the house places a demand on the HP.

Tank size was varied at Washington and Atlanta. Different analytical methods were compared to test the need for detailed simulations in making decisions on heat recovery systems.

Principal Findings

1.   Tank size selection involves compromise. Increasing the tank size increases the ability to accept recovered thermal energy whenever it is offered and increases the amount of "reserve" hot water that is available when no recovered heat is being provided, but decreases efficiency by increasing heat losses through the storage tank's insulating jacket. The ability to accept offered thermal energy was not a problem under the simulated conditions using tanks of 40 gallons or larger. Based on a quarter tank reserve, a 40 gallon system with a good heating rate from its backup resistance heater might meet dishwasher or clotheswasher demands. A 52 gallon system would permit more flexibility in scheduling hot water use, at an additional cost of about $3 per year (at 5 cents per kWh) in jacket losses. Larger tanks are not particularly attractive, increasing energy costs without providing important benefits.

2.   The fraction of the offered thermal energy that was accepted was at or near 1.0 for all the cases covered in the simulations. This is the "decoupling factor" shown in Table III-1. This means that the tanks were quite successful in accepting thermal energy whenever it was offered and making it available for later use. In this sense the time sequence of offerings is decoupled or rendered independent of the time sequence of water uses. Where time sequences are unimportant, the hourly data may be arranged in ways that are analytically more convenient and the benefits of hourly simulations for system performance estimation purposes are minimal. Holding the house load profiles fixed, the decoupling factor will remain high for higher hot water usage. Fixing hot water usage, the decoupling will also be high for lower house load profiles and for lower HP capacities.

3.   It should be expected that thermal energy recovered during the heating season will not meet hot water requirements even under

favorable conditions. This means that heat from the backup heater in the WH will be needed on a regular basis and that the tank will commonly be at or near its "reserve" status during this season.

4.   Some thermal inefficiencies must be expected in the loop connecting the HRU and the WH. If this loop is less efficient than the heating element in the tank in providing thermal energy to the water, the heat recovery system should be shut off when supplemental resistance heat is being provided to the house.

5.   The operating cost of the HP is unchanged by heat recovery operations during the cooling season, except for the possibility of some second order effects involving performance of the refrigerant coils. Total operating costs during the heating season for the HP, supplementary heater and backup heater vary with tank size because jacket losses depend upon tank size. However, since all the offered heat was accepted during the heating season, there was no difference in the combined operating costs of the HP and the supplemental heater with respect to tank size in the simulated cases. (Coil defrosting costs were not estimated. The modeling assumption is that heat recovery operations would be suspended during defrosting and that defrosting cycles do not interfere with HP heating cycles.)

6.   It is possible during some parts of the cooling season to achieve satisfactory indoor conditions by ventilation instead of keeping the house closed and depending on the HP. Ventilation is economically sound since more energy is saved by not running the HP than would be provided "free" to the WH.

7.   The weather at any given location can vary considerably from year to year and within the service area of a weather data source, with corresponding effects on HP loads and heat recovery savings. Since system paybacks will take years, the data and analysis should cover a significant number of years and the use of averaged weather data in the estimating procedures is justified.

8.   Simplified estimation methods that would be adequate for consumer decision purposes appear practical. Development of such methods is suggested.

B. System Description and Operation

        A schematic diagram of the integrated HP-HRU-WH system to be discussed is shown in Figure III-1. The main elements are a HP, a HRU and a conventional electric WH with its lower heating coil disconnected (and omitted from the diagram). The upper electric heater is called the backup heater in the discussion. A supplemental resistance heater is provided to augment the HP when the overall demand for heat by the house and the heat recovery system exceeds the HP capacity. House thermostats control the HP and the supplemental heater. The circulating pump in the HRU operates steadily when the HP is operating. Water is allowed to flow from the HRU only when it is hot enough (nominally 145°F in these simulations).

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The symbol, Q, on the scale next to the tank represents the heat status of the tank. There will be some mixing between the hot water at the top and the cold water at the bottom of the tank, but a sharp temperature transition is used for purposes of exposition. Q increases (i.e., moves down in the diagram) when the HRU is providing hot water and moves up when hot water is drawn or thermal energy is lost during standby. The backup heater will come on when Q is less than the "reserve" hot water level, Q1, and will turn off when Q reaches Q1. If the HRU supplies enough thermal energy to increase Q to Q9, then the pump will draw hot water, signaling that the tank has reached its maximum heat capacity, and no further thermal energy offerings by the HRU will be accepted.

Figures III-2a-c show the hour-by-hour heat status of the tank in percent of heat capacity, Q9, for selected days from the Washington, D.C. simulation. Figures III-2a and b are for a 40 gallon tank on different days, and Figures III-2b and c are for 40 and 52 gallon tanks on the same day. Status is shown after all gain and loss transactions for an hour have been completed. Therefore, Q is never shown below the reserve status, Q1, and Q is never allowed to rise above the heat input limit, Q9. Except for extraordinary cases, backup heat is needed whenever the ending status is less than Q1, and some of the offered heat is not used when the ending status is Q9. It is seen that the tank heat status can be quite dynamic when the HP is active and that larger tanks are able to accept more of the offered heat than smaller capacity tanks.

Further understanding of the general interactions involving the house system and local weather may be gained from Figure III-3[1], in which several heat recovery factors are presented in schematic form as functions of the primary independent variable, the outdoor dry-bulb temperature. The HP operating regimes of heating and cooling are separated by a zone of zero house load, in which the HP does not operate.

The water heater factor, WHF, is shown in the top graph, III-3a. WHF is the fraction of the HP output that is offered to the water tank. WHF rises with outdoor temperature because the discharge temperature of the refrigerant coming from the HP compressor increases with the temperature of the outdoor coil whereas the minimum temperature to which the refrigerant may be cooled in the heat exchanger is determined by the required output water temperature.

House load and HP capacity vary with outdoor temperature, as indicated in Figure III-3b. The two capacity curves are for a nominal three-ton HP. House loads are shown here and in Figure III-4 solely as functions of outdoor temperature.[2] The general effect of a ventilation option during the cooling season is shown. The load curve slopes and intercepts on the temperature axis are determined mainly by house size, construction and operation. Geographic location may affect the load through wind, air quality, cloud cover and outdoor humidity characteristics of the locality. Latitude affects the amount of daylight through the seasons and the incident angle of solar radiation. Houses have thermal masses that cause loads to lag the forcing functions. Even though

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[1] Information from several sources was used to develop Figure III-3, including Bibliography items 6 and 14, test data, engineering calculations and externally supplied simulation results.

[2] Hourly house load data and corresponding outdoor dry-bulb temperature were obtained from another NBS project. These data were sorted according to five-degree temperature bins, averaged within bin, and plotted.

the load can vary considerably at any given temperature, the outdoor temperature can serve alone as an effective indicator of house load for present purposes. (Estimating methods using outdoor temperatures are discussed in Section E and numerical results are compared in Table III-2 with those from simulations using hourly heating, cooling loads and temperature.)

Supplemental resistance heat is needed when the combined house heating and HRU loads exceed HP capacity. The effect of offering thermal energy to the WH during heating mode operation is shown as a cross-hatched area, reducing the HP heating capacity available to the house. This capacity reduction does not occur during cooling mode operation. If the house load exceeds the HP capacity in the cooling mode, the indoor temperature will rise appropriately.

There is a ten-to-one scale expansion from III-3b to III-3c. The dome-shaped curve in III-3c labeled "From Heat Pump" corresponds exactly to the heat flow segment in III-3b labeled "To Water Heater". Figure III-3c shows the average hourly heat requirement for the WH without regard for time of day, and the thermal energy sources in the integrated water heating system. Thermal energy offerings from the HP (the curves labeled "From Heat Pump") are found by multiplying the WHF by the lesser of the HP load or the HP capacity. The HP load is the same as the house load in the cooling mode, but includes the thermal energy given to the water by the HRU and the house load in the heating mode. Some of the offered thermal energy at the higher outdoor temperatures is more than can be accepted on a steady-state basis. However, some of this excess thermal energy can be used if the tank status before the offering is below the maximum level that is allowed. (High outdoor temperatures accounted for all cases in the simulations in which some of the offered heat could not be used, thus reducing the decoupling factor.) The backup heater must be capable by itself of meeting the full water-heating load when the HP will not be operating.

A sample climate profile in terms of hours per year as a function of outdoor temperatures is shown in Figure III-3d. This profile is for Washington, D. C. and the single year of 1957. Profiles of this type for other locations in the United States are given in Figures A-3 through A-13 of Appendix A. These latter profiles, representing data over at least ten years, provide a useful set of weighting factors for judging system attractiveness in various regions. Data for these curves came from Item 6 in the Bibliography, in which plots for many more sites are available. The attractiveness of HP-WH systems is greatest at those locations having many hours per year in the temperature ranges where the HP offerings (III-3c) are high.

House load data for the simulations were generated for the same hypothetical house at four locations, using a particular weather year at each location. These load data were sorted into five-degree temperature bins, averaged within bin and plotted against bin temperature in Figure III-4 for Washington, D.C. and Atlanta, GA. Sensible and latent loads were added to obtain the total load for each hour. Differences in climate and sun elevation may account for the differences between sites.

Some general inferences that are borne out by the simulations may be deduced directly from Figure III-3.

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1. The WHF declines as the outdoor temperature goes down, for both heating and cooling applications, reaching zero where the temperature of the refrigerant coming out of the compressor is too low to transfer thermal energy to water in the HRU at the required water output temperature. Hours at temperatures below these zero points could be dropped from consideration for purposes of heat recovery analysis.

2. There is some temperature below which supplemental resistance heat will be needed, given reasonable sizing of the HP with respect to the house load profile. The supplemental heater must make up any shortfall in the heat that the HP supplies to the house. Therefore, no advantage is seen in giving HP thermal energy to the WH if it must be replaced by the supplemental heater. The simulation does not include any allowances for thermal inefficiencies between the HRU and the WH, and thus treats the energy used by the tank's backup heater or the supplemental heater as equivalent in the energy totals. Any such inefficiencies mean that the heat recovery system should be turned off when the supplemental heater is needed.

3. The backup heater will have to carry much of the water heating load during the heating season, with the further implication that the tank heat status will be at or near the reserve level, Q1, during this season. Referring to the Water Heater Operation graphs in Figure III-3, the dashed line labeled Average Hourly Requirement is one twenty-fourth of (64.3 gallons per day x 8.25 pounds per gallon x 90°F temperature rise plus 1 percent per hour for heat loss during standby), or a total of about 2100 Btu per hour. The highest offerings during the heating season occur in the 25 to 35°F temperature range and are about half of these average demands. Some combination of major reductions in water use and increased thermal energy offerings (by increasing both house heating loads and heat pump capacity) would be needed to close the gap in this limited temperature range. Also, where all the thermal energy offered during the heating season is accepted, the total annual electricity consumption by the HP and the supplemental heater is independent of tank size. However, WH jacket losses do vary with tank size, affecting annual energy use by the backup heater and hence the total annual consumption of electricity.

4. The practice of ventilation during the cooling season is economically sound as long as satisfactory indoor temperature and humidity conditions can be maintained. Some "free" thermal energy to the WH would be missed, but this loss would be outweighed by the savings in HP operation.

C. Simulation Methodology

There are many variables affecting the amount of thermal energy that is available from a HP-HRU-WH system and the operating cost of obtaining this heat. A computer simulation model was developed to study the interactions of several of these variables on an hourly basis for selected situations. The hourly steps

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in this simulation are outlined in Figure III-5 and the simulation is described further in Appendix A. The purpose of the simulation is to generate numerical information bearing on the economics of HP-HRU-WH systems in terms of major system variables. Several of these variables and their interactions have already been introduced.

The thermal performance of houses and other buildings is a significant study topic at NBS for which a highly detailed computer program, called the NBS Load Determination, or NBSLD (Bibl. 8), program, has been written. NBSLD was used on another NBS project (a retrofitting study using 1800 square foot houses and four-person families) to generate hour-by-hour house load values for selected locations and weather years. Thermostats inside the house were set at $68^{\circ}F$ during the heating season and $78^{\circ}F$ during the cooling season. Ventilation was used when a 68 to $78^{\circ}F$ indoor temperature and an indoor relative humidity below 60 percent could be maintained by this method. Neither ventilation nor the HP were used when satisfactory conditions could be maintained without them. The present simulation makes use of these house load data and the corresponding outdoor temperatures. An average hot water usage rate of 64.3 gallons per day (450 gallons per week) with a $90^{\circ}F$ temperature rise was used. The key parameters of HP coefficient of performance (COP) and output capacity (CAP) and the water heater factor (WHF) were expressed as functions of outdoor temperature. (See Appendix A, Table A2.) The COP and CAP values in the simulation are consistent with NBS laboratory data. An hourly hot water draw schedule developed by NBS was used for each day of the simulation year. (See Appendix A, Table A1.) The heat status of the WH at the end of the previous hour is the beginning status in the current hour.

The computational steps for each hour are:

1. Subtract from the initial heat status, Q, the standby heat loss and the thermal energy in any hot water that was drawn during the hour to produce the pre-offer heat status. A standby thermal energy loss rate of one percent per hour of the initial heat status of the tank was used.

2. If there is any load on the HP, determine which alternative will set the limit on the amount of thermal energy that the tank will accept.

    a) The tank can only accept thermal energy up to its heat capacity, Q9.

    b) The maximum thermal energy offering is the product of the HP capacity and the WHF at the temperature for the hour.

    c) If the HP load is at or below the HP capacity, then the thermal energy offering is determined by the product of HP load and the WHF. The HP load is solely the house load in the cooling mode, but includes thermal energy for the WH in the heating mode.

3. Compute the new tank status and the following items for accumulation and eventual printout:

a. Heat recovery unit offering

b. Heat recovery unit output (= water tank acceptance)

c. Backup heater load

d. Supplemental heater load

e. Heat pump electricity used.

The backup heater is used only when necessary to achieve the minimum (or reserve) heat status, Q1, at the end of the hour. Calculations of electricity use by the HP include adjustments for efficiency degradation during cyclic operation.

4. Go on to the next hour.

D. Results

Results from ten simulation runs are summarized in Table III-1. Two runs were conducted for each city using systems with 52 gallon tanks, one without heat recovery (the base case) and the other with heat recovery. These pairs indicate regional variations in energy use and savings potential. Tank size effects were also tested with runs for Washington and Atlanta using heat recovery systems with 40 gallon and 80 gallon tanks. Both the 52 and 80 gallon tank systems accepted all offered thermal energy. Jacket losses for the 80 gallon tank system were higher than for the 52 gallon tank system, so the 80 gallon results are not presented. The operating rules with and without heat recovery are identical except that no thermal energy is accepted from the HRU in the base case. All water thermal energy in the base case comes from the backup heater and the tank is restored to the reserve status, Q1, at the end of each hour.

Two system performance measures are provided. The one of greatest interest to a homeowner is annual electric energy saved, measured relative to a HP and WH system without heat recovery and in which the lower coil in the WH is similarly disconnected. The second will be called the decoupling factor and is the ratio of thermal energy accepted by the WH to the thermal energy offered by the HRU. This factor is an index of the system's ability to decouple the two time series of thermal energy offerings and thermal energy uses by storing offered thermal energy for later use.

Electric energy savings are shown on line M and decoupling factors are shown on line N. Savings are affected by tank size to a minor degree through differences in the energy lost during standby and the ability to accept offered thermal energy. Larger tanks have larger hot water reserves and therefore have more hot water exposed to standby losses. Hot water reserves of 10 and 13 gallons were used in the simulation respectively for the 40 and 52 gallon systems. Line O, labeled "Q1 Hours," shows hours for the year ending in the reserve status. At least for these hours in common, the 52 gallon system has

three more gallons of hot water losing heat during standby than the 40 gallon system. Based upon the savings indicated on line M and energy costs of 3.5 to 5 cents per kilowatt-hour, use of the smaller tank would save $2 - $3 per year over the larger tank in Washington and Atlanta.

The obvious next question is why not go to still smaller tanks? It is seen that any tank will spend much of its time in the reserve status, Q1, and that major hot water draws such as for washing clothes and dishes must be anticipated when in this status. The largest single draws used in the simulation are one 15 gallon draw in the morning and another in the evening. Both exceed the 13 or 10 gallon reserves, and could be a basis for insisting on a tank of 60 gallons or larger. However. the tanks under consideration have recovery rates of 17 to 25 gallons per hour, and recovery heat will be available during the actual period of the draw to reduce or eliminate the possible shortfall. As a practical matter, the 52 gallon system offers some flexibility in scheduling hot water use, whereas the 40 gallon system requires more scheduling care. The general conclusion is that the most economical system uses the smallest tank that has enough reserve hot water as long as the decoupling factor is high.

The decoupling factors on line N are high for both tank sizes and at all locations, indicating that, at least for the particulars of these simulations, there is little problem in making full use of the thermal energy offered by the HRU. The simulation for Washington indicates a minor inability of the 40 gallon tank system to take all of the offered thermal energy. Thirteen kBtu that were accumulated during eight hours (line P) during the year could not be used (line E-line D). These hours occurred during the hottest part of the summer. The 52 gallon systems were able to accept all of the offered heat in Atlanta and Seattle, and all but one kBtu in Washington and Chicago.

Unused thermal energy and the Q9 hours increase as tank size decreases and also as the hours at very high outdoor temperatures increase. The decoupling factor decreases under these circumstances, making the time phasing between thermal energy offerings and the water draws more important.

Combined electric energy used by the HP (line I) and the supplemental heater (line K) is independent of tank size as long as all the unusable thermal energy is offered during cooling operations. Unusable thermal energy, if any, offered during heating operations leads to a decrease in the year's combined HP and supplemental heater energy use and to increased energy use by the backup heater.

E. Alternative Estimation Methods

One objective in detailed simulation is to preserve the effects that events in one time period (hour-to-hour or day-to-day, for example) have on events in another period. Another is to provide a baseline from which the acceptability of alternative analytical methods may be judged. As will be shown, certain aspects of the simulation procedure may be reshaped to good effect and without changing the numerical results enough to affect most consumer decisions.

Hourly data on house loads and corresponding weather data are available for only a few locations and quite specific house characteristics and operating

strategies. This places severe limits on the ability to extend results to other locations and situations. Therefore, the numerical significance of making two adjustments to the more rigorous hour-by-hour simulation was examined. The first of these was to sort the load data by five-degree temperature bins, compute bin averages and then compute the various heat recovery output results. Average house loads by bin are shown in Figure III-4 for Washington and Atlanta. The issues being examined are the importance of specific sequences of hourly loads and the effects of nonlinear relationships on the results. The second change was to replace the bin averages of the house loads with loads derived from linear regressions of the bin averages. If the first adjustment is acceptable, then annual summaries of temperature data could be used. If the second adjustment is acceptable, the characterization of houses and their operations can be much more flexible and parametric studies become more practical.

Comparative results are summarized in Table III-2 for four locations and three computation methods. Systems with 52 gallon tanks were used throughout. The reference case for computing savings is a 52 gallon tank maintained at a 13 gallon reserve level, Q1, by a backup heater. Both the Bin Average and the Bin Linear methods are based on steady state conditions of heat recovery within each temperature bin and do not allow for thermal energy acceptances within the dynamic range of the tank heat capacity when the average offering rate exceeds the acceptance rate, accounting for most of the difference in savings among the methods. Nonlinearities and approximations of constants account for the remaining differences in savings estimates.

The difference in numerical results among the methods do not appear to be important with respect to the most important consumer decision on whether a heat recovery system of this kind should be installed or not. The differences are well within common year-to-year weather variations. Although these few examples hardly constitute proof, the strong inference is that the simpler computing methods are adequate. Furthermore, they offer opportunities for parameteric examination of several house and recovery system variables without expensive and sometimes unavailable weather data and preparatory load calculations. The various curves in Figure III-3 and COP values could be hypothesized, for example, in order to explore the circumstances of economic feasibility for such systems. Bibliographic item 6 or its equivalent provides a rich compilation of weather data, house load curves could be hypothesized, and so on. All of the reported approaches use discrete weighting and summation methods to produce the main results. There is an untested possibility that the various curves could be expressed in continuous algebraic form and dealt with at that level for sensitivity analysis purposes.

Table II-1: Time-Averaged Conditions and Thermodynamic Performance Data for Heat Pump 1 By Itself and as Part of Integrated HP-HRU-WH Systems 1 and 2

Heat Pump	Heat Recovery Unit	Outdoor Temperature °F	Indoor Temperature °F	Indoor Relative Humidity %	Condenser Pressure psia	Condenser Temperature °F	Evaporator Pressure psia	Evaporator Temperature °F	Compressor Discharge Pressure psia
1	HRU 1	96.5	84.5	52.2	257.4	115	92.3	46	270.4
1	no HRU	97.3	84.0	52.3	271.1	119	95.5	48	278.0
1	HRU 1	77.8	78.2	75.0	199.3	96	78.9	37	210.4
1	no HRU	84.3	80.6	76.0	225.5	105	84.7	41	236.0
1	HRU 1	21.1	72.0	80.0	170.4	85	55.6	18	217.6
1	HRU 2	97.7	78.8	59.0	267.6	118	93.9	47	279.9
1	no HRU	97.3	78.7	58.5	274.6	120	95.5	48	280.0
1	HRU 2	82.4	77.9	55.0	225.5	105	87.7	43	236.2
1	no HRU	89.2	77.3	56.0	254.1	114	92.3	46	256.7
1	HRU 2	76.6	83.6	54.0	210.6	100	84.7	41	222.2
1	no HRU	83.8	83.4	55.0	241.0	110	90.7	45	240.5

Heat Pump	Heat Recovery Unit	(Discharge-Condenser Exit) ΔP psi	Compression Ratio	Discharge Temperature °F	R22 Mass Flow Rate lb/h	Refrigeration Effect Btu/lb	Work of Compression Btu/lb	Thermodynamic COP_H or COP_C	Cooling or Heating Capacity Btu/h
1	HRU 1	18.3	2.79	220.6	480	75.0	21.0	3.57	36000
1	no HRU	13.3	2.84	213.0	490	73.5	20.0	3.68	36015
1	HRU 1	16.4	2.53	207.5	435	81.5	20.5	3.98	35453
1	no HRU	17.3	2.66	209.9	450	79.0	20.0	3.95	35550
1	HRU 1	40.8	3.07	165.0	310	92.4	21.0	4.40	28644
1	HRU 2	17.5	2.85	193.9	485	69.5	20.0	3.48	33708
1	no HRU	10.6	2.88	181.3	490	69.8	17.7	3.94	34202
1	HRU 2	16.6	2.57	200.9	470	78.2	20.0	3.91	36754
1	no HRU	7.7	2.75	190.8	480	72.8	19.9	3.66	34944
1	HRU 2	16.4	2.49	204.3	460	82.0	20.0	4.10	37720
1	no HRU	6.8	2.66	202.0	480	78.0	19.0	4.10	37440

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Table II-2: Comparison of Heat Pump Capacity, Power and Efficiency
Performance Data for Systems 1 and 2 with Available
Data Supplied by Manufacturers

	Capacity, Btu/h at $T_O^{(1)} \cong$				Average Power, W, at $T_O^{(1)} =$				$COP_H(1)$ at $T_O=$	EER, Btu/wh		at $T_O^{(1)} =$
	20F	75F	85F	95F	20F	75F	85F	95F	20F	75F	85F	95F
Mfr. HP1 data	25000	38100	37200	35000	3520	4100	4275	4375	2.08	9.3	8.7	8.0
System 1 with HRU1	28644	35453	- - -	36000	3620	3870	- - -	4350	2.32	9.16	- - -	8.28
System 1 HP	- - -	35550	- - -	36015	- -	4110	- - -	4450	-	8.65	- - -	8.09
System 2 with HRU2	- - -	37720	36754	33708	- -	4010	4090	4400	-	9.41	8.99	7.66
System 2 HP	- - -	37440	34944	34202	- -	4210	4290	4420	- -	8.89	8.15	7.74

(1) T_O is a nominal temperature, except for Mfr. HP1 data.

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Table II-3: Mean Heat Recovery Rate, Thermal Energy Rate,
Energy Consumption Rate, and Heat Recovery
Efficiency for Integrated Systems 1 and 2

| | System | Nominal Outdoor Dry-Bulb Temperature | | | |
		20°F	75°F	85°F	95°F
Mean Heat Recovery Rate, kBtu/h (1)	1	2.26	7.20	----	8.50
	2	----	6.80	6.80	7.40
Thermal Energy Rate, kBtu/h (2)	1	3.26	7.49	----	9.59
	2	----	7.34	7.77	7.79
Energy Consumption Rate, kWh/h	1	3.64	3.88	----	4.28
	2	----	4.00	4.10	4.40
Heat Recovery Efficiency, Btu/Wh (3)	1	0.62	1.86	----	1.96
	2	----	1.70	1.66	1.68

←— Heating —→ ←———— Cooling ————→

Notes:

(1) The Mean Heat Recovery Rate was determined from the difference in total heat recovery by the water between 1 and 3 hours following the start of system operation.

(2) The Thermal Energy Rate was determined from the refrigerant mass flow rate and the difference in the inlet and outlet enthalpy of the HRU 2 hours following the start of system operation.

(3) The Heat Recovery Efficiency is the ratio of the Mean Heat Recovery Rate to the Energy Consumption Rate.

Table III-1: Tank Size Simulation Results[a]

Item	Washington, D.C.			Atlanta, GA			Chicago, IL		Seattle, WA	
A. City[b]										
B. Tank size, gallons[c]	52 (Base)	40	52	52 (Base)	40	52	52 (Base)	52	52 (Base)	52
Total Heats for the Year, kBtu										
C. House load	46770	46770	46770	35842	35842	35842	59138	59138	39734	39734
D. Heat accepted	0	5760	5772	0	4399	4399	0	5418	0	5375
E. Heat offered	5773	5773	5773	4399	4399	4399	5419	5419	5375	5375
F. Heat from backup heater	18273	12477	12659	18273	13794	13986	18273	12992	18323	13078
G. Supplemental heater	302	354	354	36	49	49	2348	2493	0	0
H. Heat to water heater	18273	18237	18431	18273	18193	18385	18273	18410	18323	18453
Electric Energy for the Year, kWh										
I. Heat pump	6554	6979	6979	4973	5255	5255	8232	8759	5612	6263
J. Backup heater	5355	3657	3710	5355	4043	4099	5355	3808	5370	3833
K. Supplemental heater	89	104	104	10	14	14	688	731	0	0
L. Totals	11998	10740	10793	10338	9312	9368	14275	13298	10982	10096
M. Savings	NA	1258	1205	NA	1026	970	NA	977	NA	886
N. Decoupling factor (D/E)[d]	0	1.0	1.0	0	1.0	1.0	0	1.0	0	1.0
O. Q1 hours	8760	4249	4291	8760	5138	5195	8760	4413	8784	4220
P. Q9 hours	0	8	1	0	0	0	0	1	0	0
Q. Hours with no house load	3112	3112	3112	4730	4730	4730	3336	3336	2948	2948

(a) Results are for a whole simulation year. Standby loss rate = one percent per hour of the heat in the water. The house was ventilated when the resulting indoor dry-bulb temperature could be maintained between 68°F and 78°F and humidity was acceptable. Cooling and heating loads are added.

(b) Related average temperature profiles in Appendix A: Washington-Andrews AFB, Atlanta-Dobbins AFB, Chicago-O'Hare IAP, Seattle-Everett, WA.

(c) Base = No heat recovery. Backup heater in a 52 gallon tank heats all the water.

(d) The test reference years for the four locations are, respectively, 1957, 1975, 1974, and 1960 (a leap year).

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Table III-2: Alternative Calculation Comparisons

Item Method[a] Heat Recovered	Hourly		Bin Average		Bin Linear	
	No	Yes	No	Yes	No	Yes
Location: Washington, D.C.						
kWh for Year:						
Heat Pump	6554	6979	6647	7063	6779	7245
Backup Heater	5355	3710	5356	3710	5356	3673
Supplemental Heater	89	104	58	80	36	40
Total	11998	10793	12061	10873	12171	10958
Saving	NA	1205	NA	1188	NA	1213
kBtu Accepted	0	5772	0	5616	0	5740
kBtu Offered	5773	5775	5777	5777	5906	5906
Decoupling Factor	0	1.0	0	0.97	0	0.97
Location: Atlanta, GA						
kWh for Year:						
Heat Pump	4973	5255	5053	5345	4728	5033
Backup Heater	5355	4099	5356	4093	5356	4149
Supplemental Heater	10	14	0	4	0	0
Total	10338	9368	10409	9442	10084	9182
Saving	NA	970	NA	967	NA	902
kBtu Accepted	10	4399	0	4309	0	4116
kBtu Offered	4399	4399	4401	4401	4208	4208
Decoupling Factor	0	1.0	0	0.98	0	0.98
Location: Chicago, IL						
kWh for Year:						
Heat Pump	8232	8759	8363	8917	8344	8910
Backup Heater	5355	3808	5356	3787	5356	3784
Supplemental Heater	688	731	594	609	451	466
Total	14275	13298	14313	13313	14151	13160
Saving	NA	977	NA	1000	NA	991
kBtu Accepted	0	5418	0	5352	0	5361
kBtu Offered	5419	5419	5432	5432	5439	5439
Decoupling Factor	0	1.0	0	0.99	0	0.99
Location: Seattle, WA[b]						
kWh for Year:						
Heat Pump	5612	6263	5680	6349	5636	6304
Backup Heater	5370	3833	5370	3804	5370	3814
Supplemental Heater	0	0	0	0	0	0
Total	10982	10096	11050	10153	11006	10118
Saving	NA	886	NA	897	NA	888
kBtu Accepted	0	5375	0	5344	0	5309
kBtu Offered	5375	5375	5374	5374	5338	5338
Decoupling Factor	0	1.0	0	0.99	0	0.99

(a) Results are for 52 gallon systems. Hourly data are from the hour-by-hour simulation. "Bin Average" data are based on hour-by-hour house loads averaged for each five degree outdoor temperature bin. "Bin Linear" refers to house load inputs generated by linear regression of "Bin Average" data.

(b) Seattle data are for a leap year of 8784 hours.

ENTHALPY (Btu/lb)

Figure I-2: Pressure vs. Enthalpy Schematic for Refrigerant Cycle in a Heat Pump

In the schematic above, the enthalpy differences between points B and C, C and D, and D and A represent, respectively, the cooling capacity, the compressor work input and the heating capacity. When an HRU is used with the HP, it is connected between points D and E. Although not shown in the schematic, the refrigerant gas temperature is a high at point D, from which it passes to the HRU refrigerant input, and lower, at about the condensing temperature (Point E), after returning from the HRU refrigerant outlet. The difference in temperatures at points D and E indicates heat recovery by the HRU.

Figure I-1: Major Components of a Heat Pump

The only major component not shown in the schematic above is the reversing valve that reverses the refrigerant flow path through the evaporator and condenser coils when changing from the heating to the cooling mode, or vice versa. The arrow indicates the refrigerant flow path, and the metering device is either a capillary tube or a thermostatic expansion valve. The letters indicate points in the Pressure vs. Enthalpy refrigerant cycle shown in Figure I-2. When the heat recovery unit is used with the HP, its input and output lines leave and return to the refrigerant flow path at a break between the compressor discharge port (Point D) and the reversing valve that precedes the condenser (Point E).

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ENTHALPY (BTU/lb above saturated liquid at −40°F)
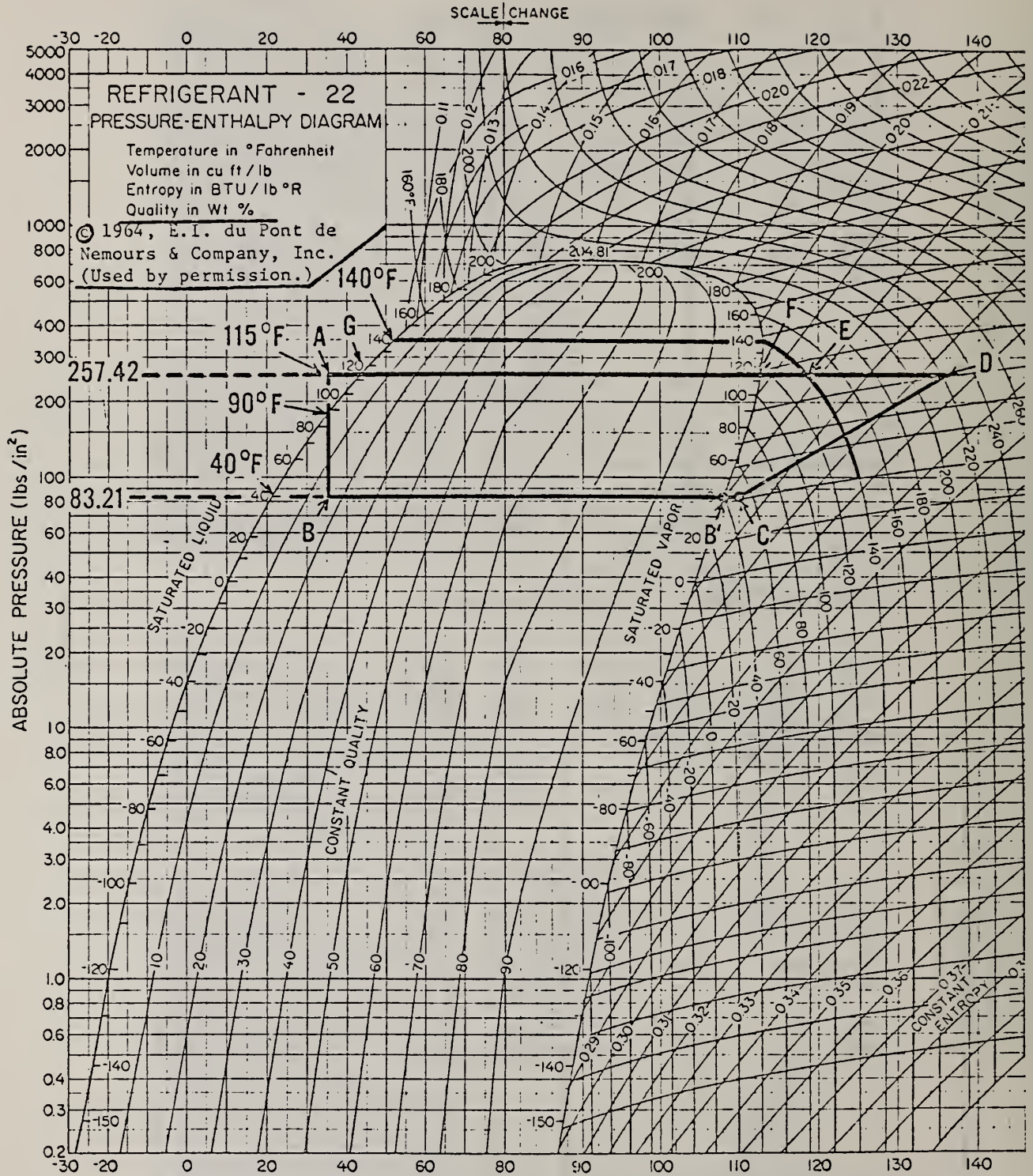
Figure II-1 :   Detailed Sample Pressure vs. Enthalpy Diagram

Figure II-2: Schematic of Integrated Systems 1 and 2

☒ BYPASS VALVE
✕ TEMPERATURE MEASURED
● TEMPERATURE & PRESSURE MEASURED

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SCHEMATIC REPRESENTATIONS
OF HEAT RECOVERY UNITS 1-3
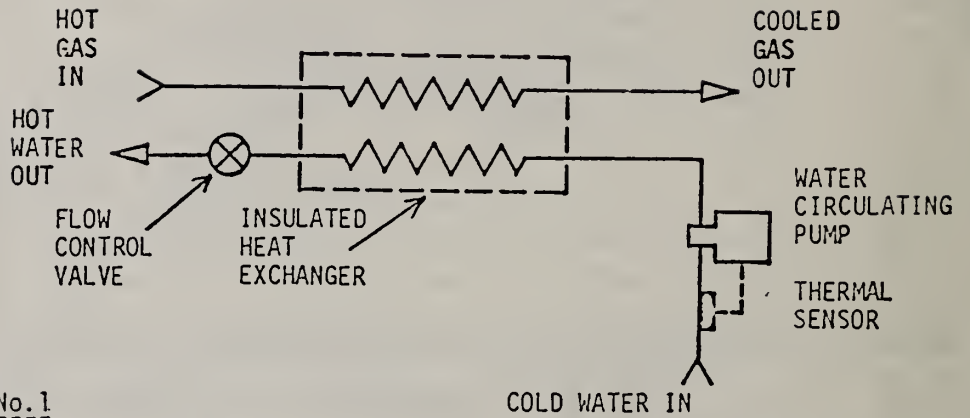
Figure II-3: HRU No.1
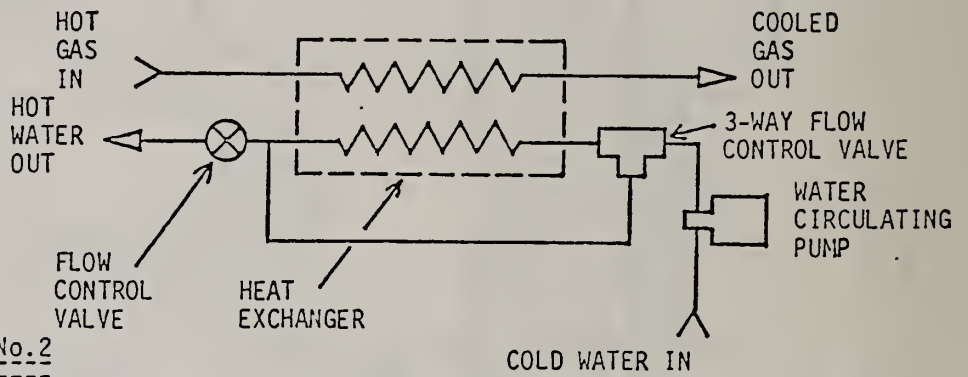
Figure II-4: HRU No.2
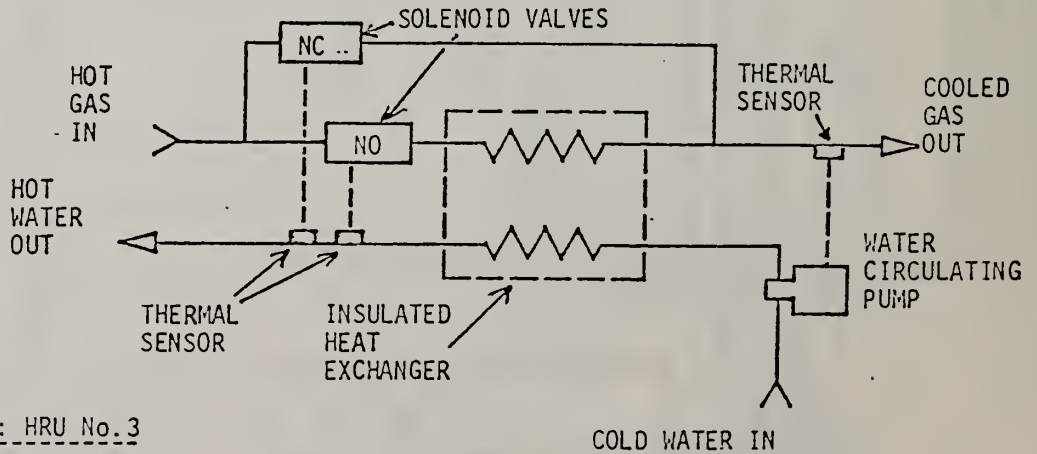
Figure II-6: HRU No.3
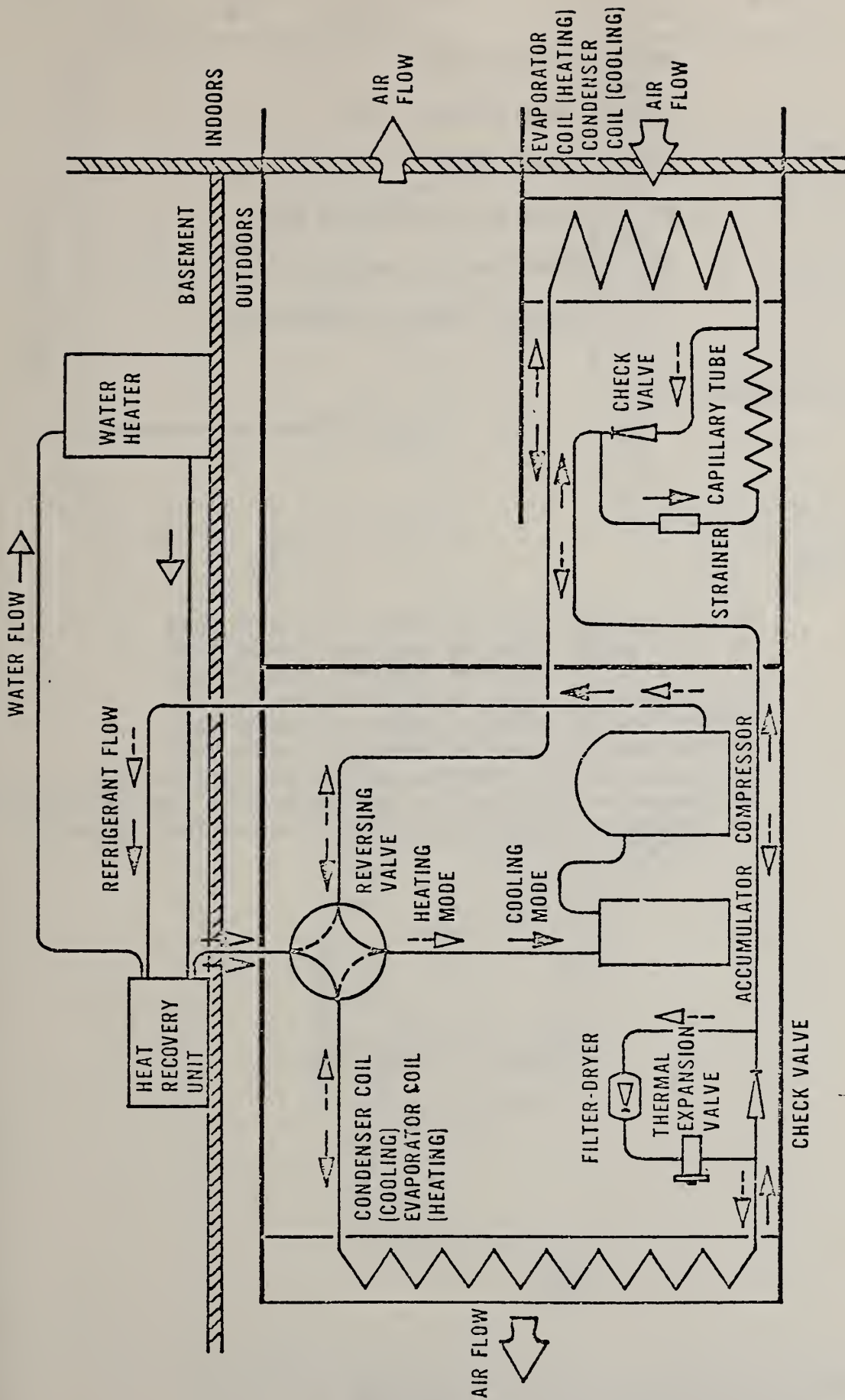
Figure II-5: Schematic for Integrated System 3

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TC = Thermocouple

HRU = Heat Recovery Unit

WH = Water Heater

HP_c = Heat Pump in Cooling Mode

HP_h = Heat Pump in Heating Mode

T_o = Nominal Outdoor Temperature

TC 1 is 3-7/8" from the Bottom of the WH Tank
TC 2 is 11-5/8" from the Bottom of the WH Tank
TC 3 is 19-3/8" from the Bottom of the WH Tank
TC 4 is 27-1/8" from the Bottom of the WH Tank
TC 5 is 34-7/8" from the Bottom of the WH Tank
TC 6 is 42-5/8" from the Bottom of the WH Tank
TC 7 measures the water temperature entering the HRU
TC 8 measures the water Cut-Out Temperature leaving the HRU
TC 9 measures the water Cut-In Temperature leaving the HRU

Addendum to Figure II-7A-C
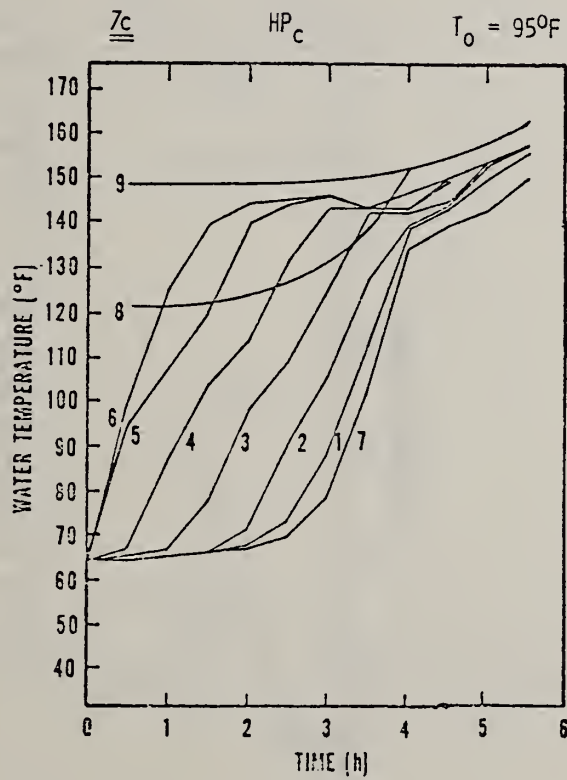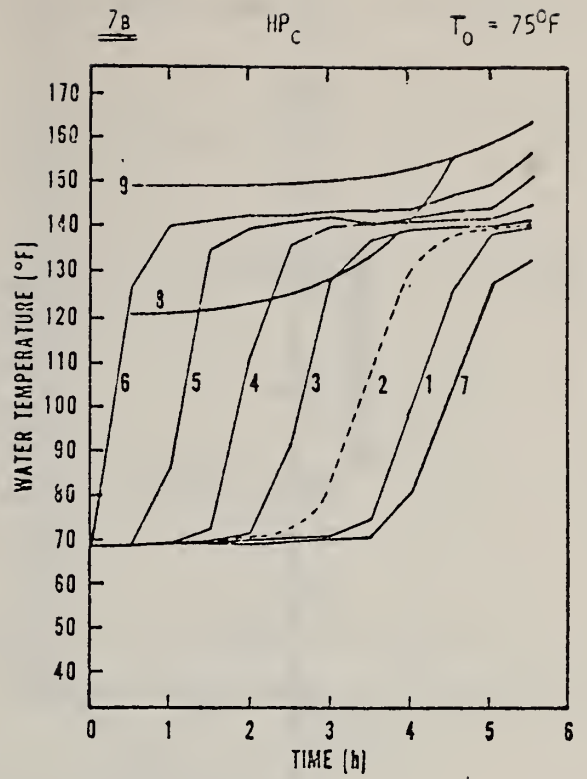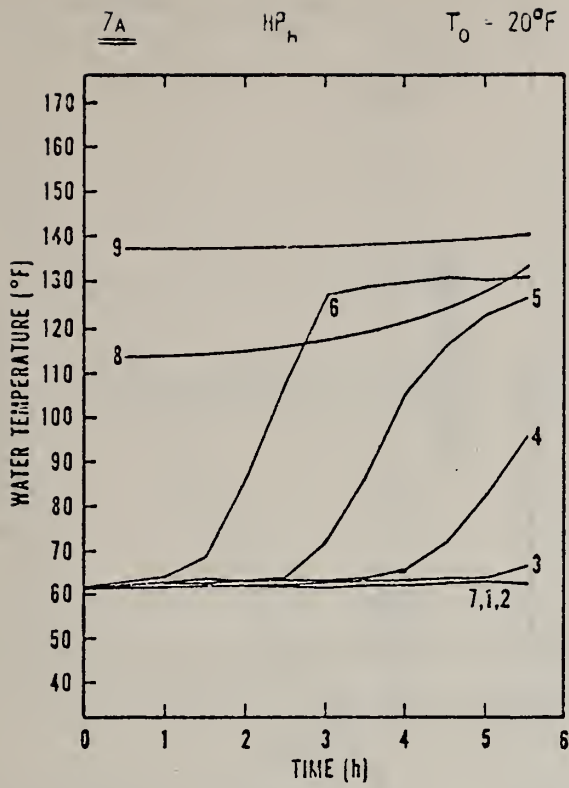
FIGURES II-7A-C: WATER TEMPERATURES AT VARIOUS LOCATIONS IN THE WH VERSUS
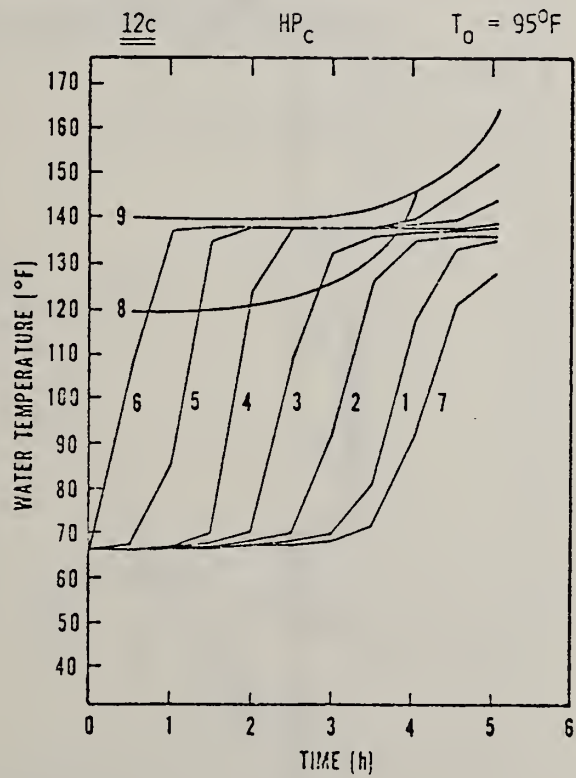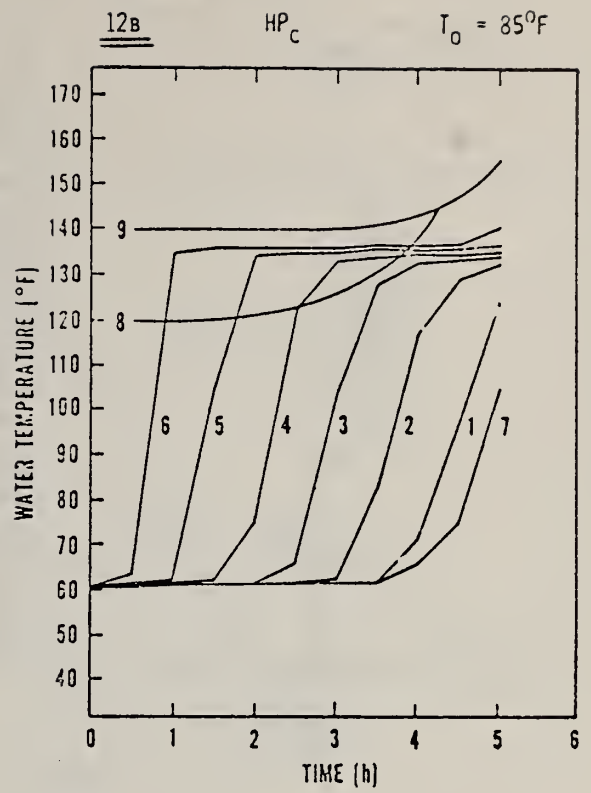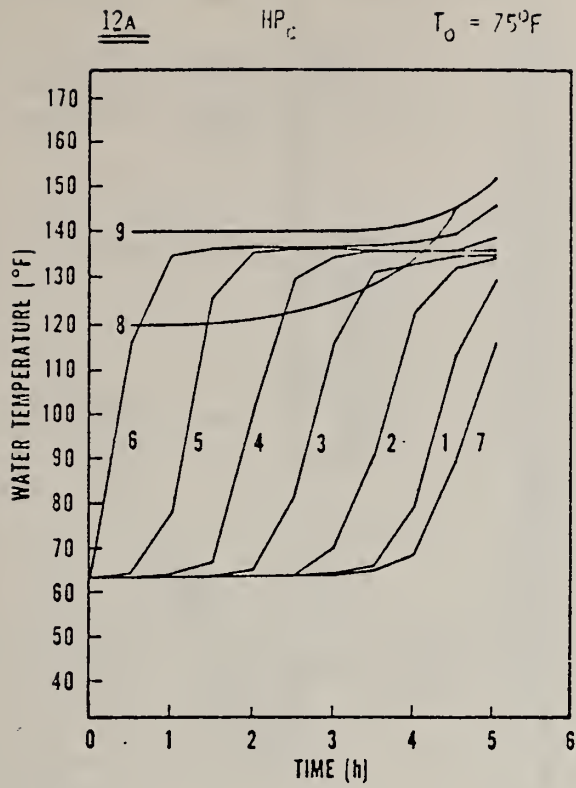TIME OF OPERATION OF THE HP - HRU - WH INTEGRATED
APPLIANCE, USING HRU NO. 1

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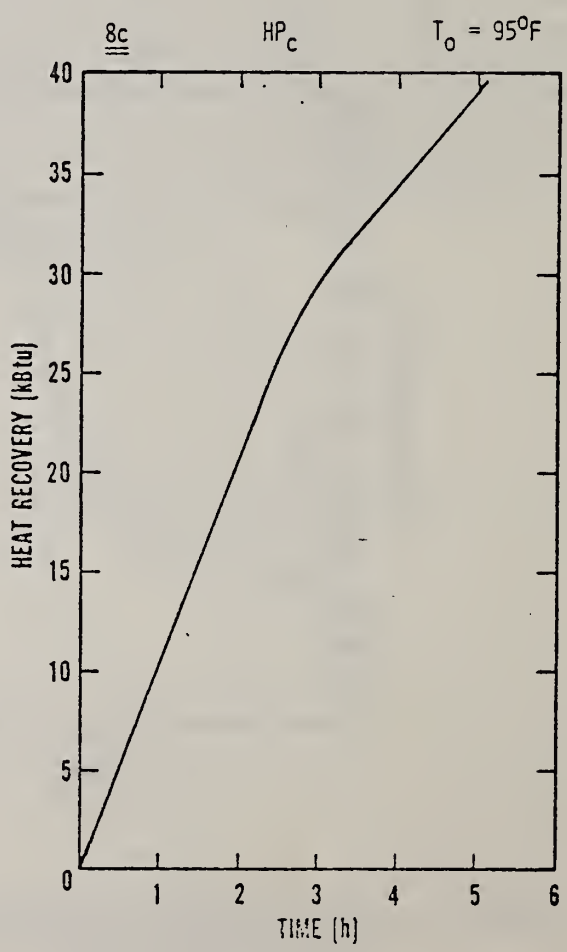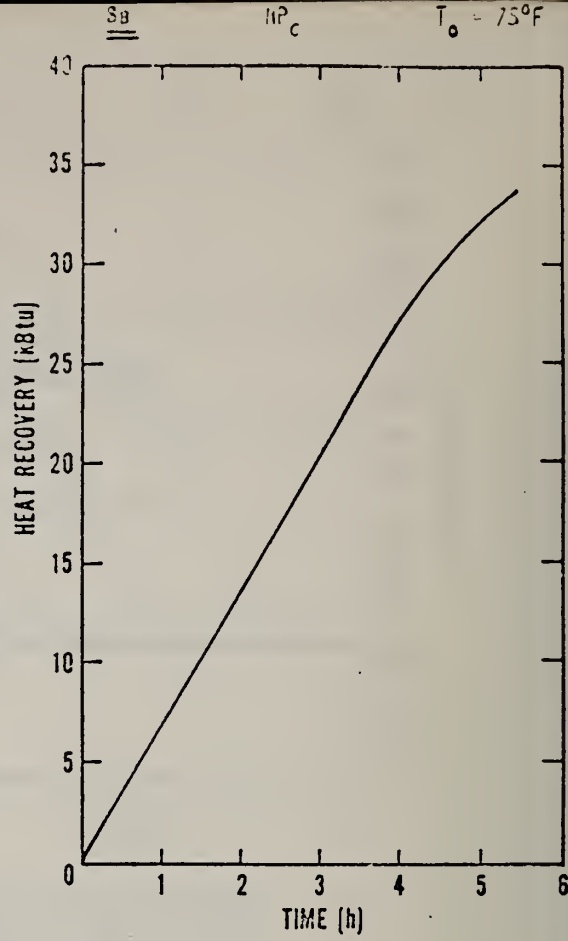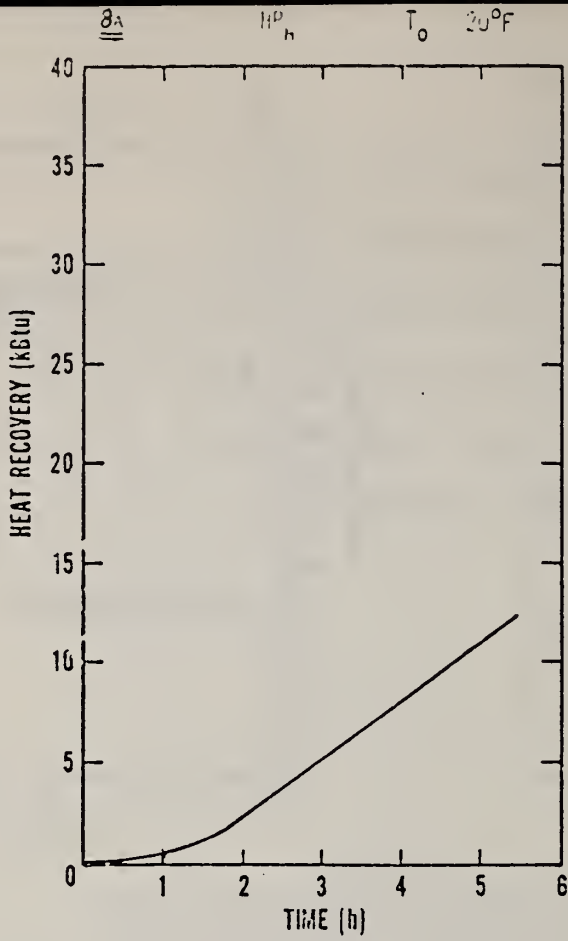
FIGURES II-8a-c: HEAT RECOVERY VERSUS TIME OF OPERATION OF THE
HP - HRU - WH INTEGRATED APPLIANCE, USING HRU No. 1
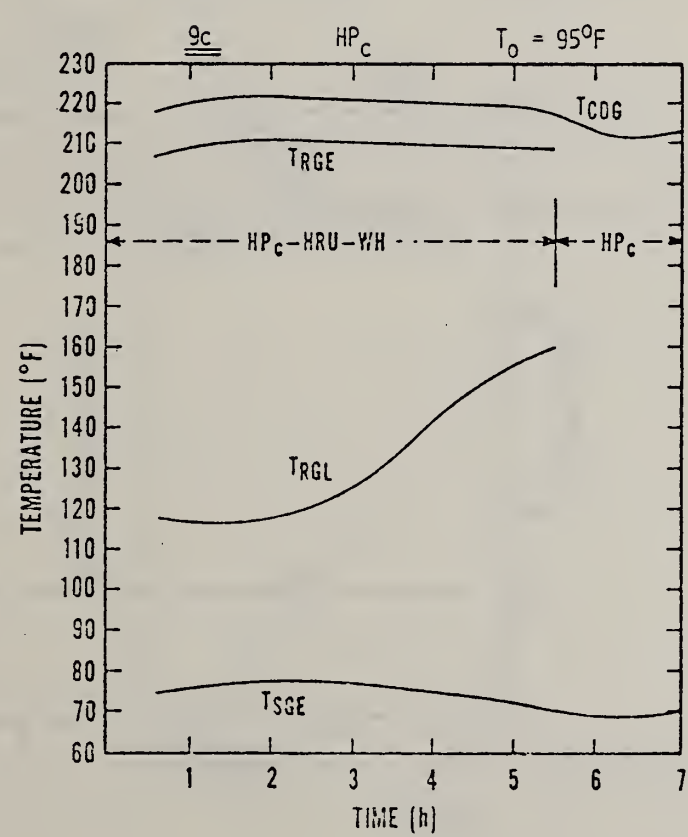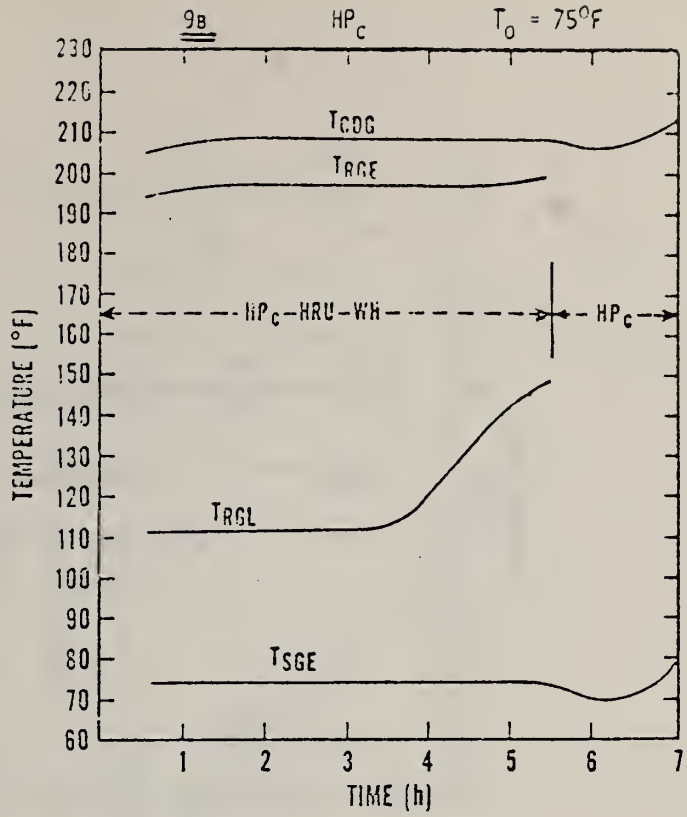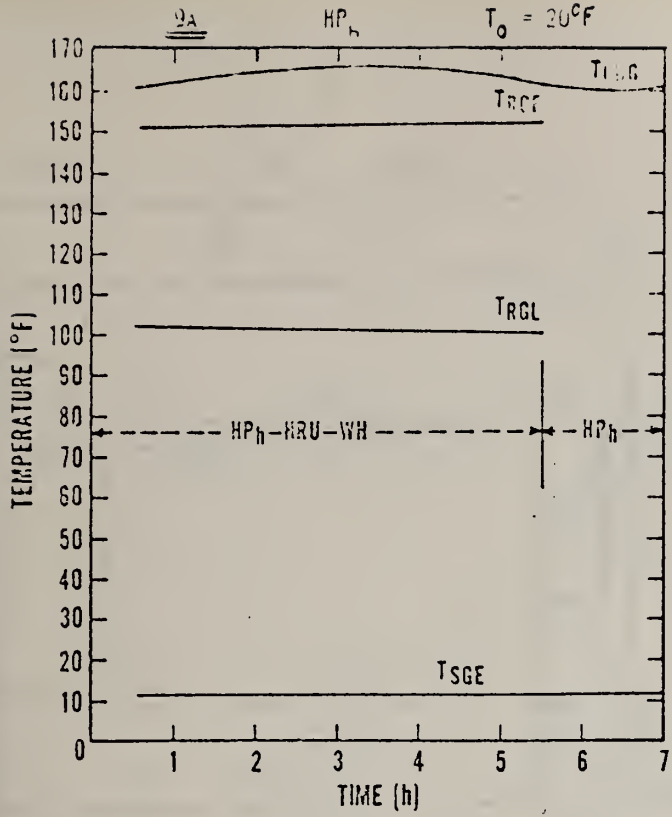
FIGURES II-9A-C: REFRIGERANT TEMPERATURES VERSUS TIME OF OPERATION FOR THE HP - HRU - WH INTEGRATED APPLIANCE, USING HRU No. 1

NOTES: T_{CDG} = TEMPERATURE OF COMPRESSOR DISCHARGE GAS

T_{RGE} = TEMPERATURE OF REFRIGERANT GAS ENTERING HRU

T_{RGL} = TEMPERATURE OF REFRIGERANT GAS LEAVING HRU

T_{SGE} = TEMPERATURE OF SUCTION GAS ENTERING COMPRESSOR
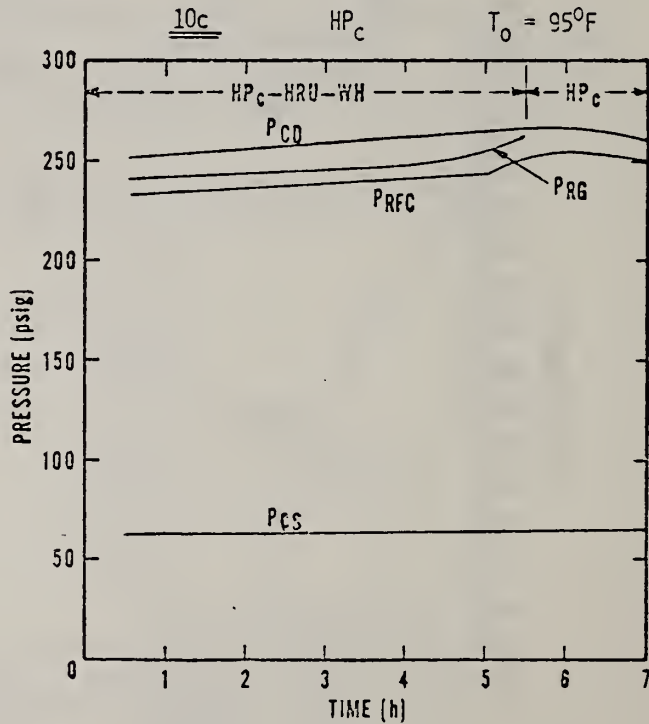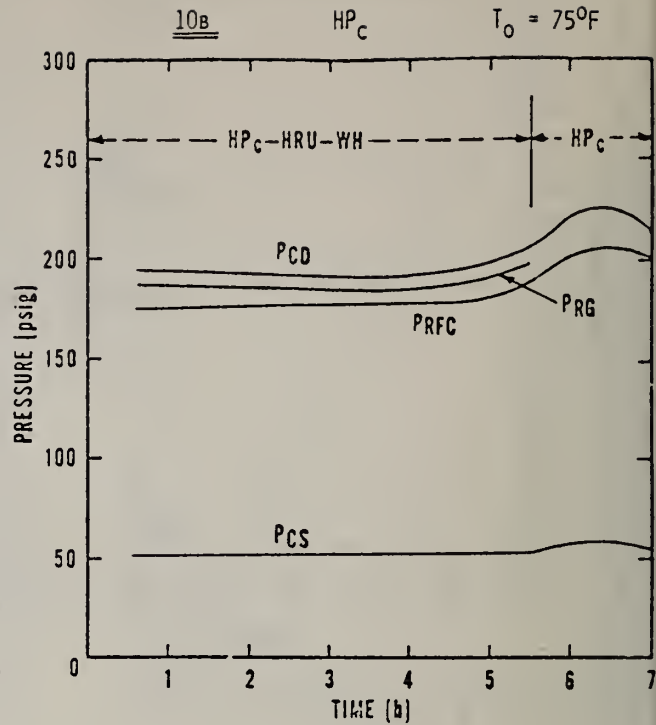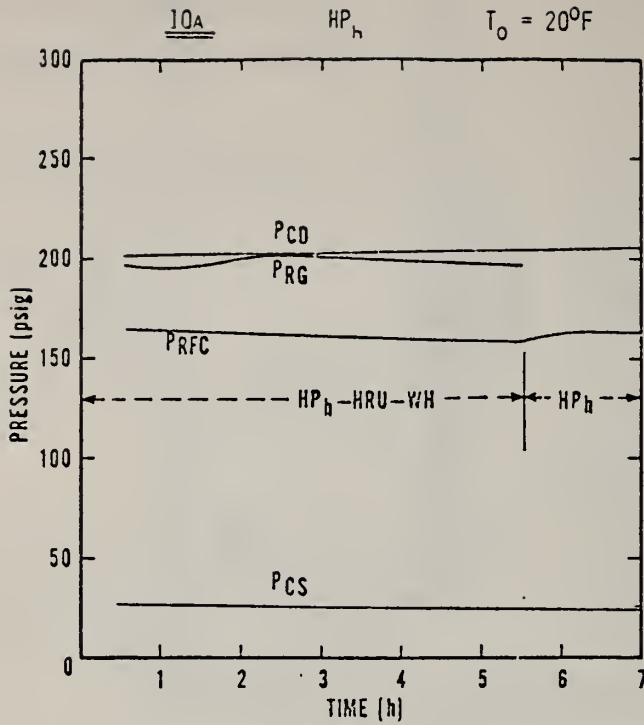
FIGURES II-10A-C: REFRIGERANT PRESSURES VERSUS TIME OF OPERATION OF THE
HP - HRU - WH INTEGRATED APPLIANCE, USING HRU No. 1

NOTES: P_{CD} = COMPRESSOR DISCHARGE PRESSURE PRECEDING GAS LINE TO HRU

P_{RG} = REFRIGERANT GAS PRESSURE FOLLOWING GAS LINE FROM HRU

P_{CS} = COMPRESSOR SUCTION PRESSURE

P_{RFC} = REFRIGERANT PRESSURE FOLLOWING CONDENSER

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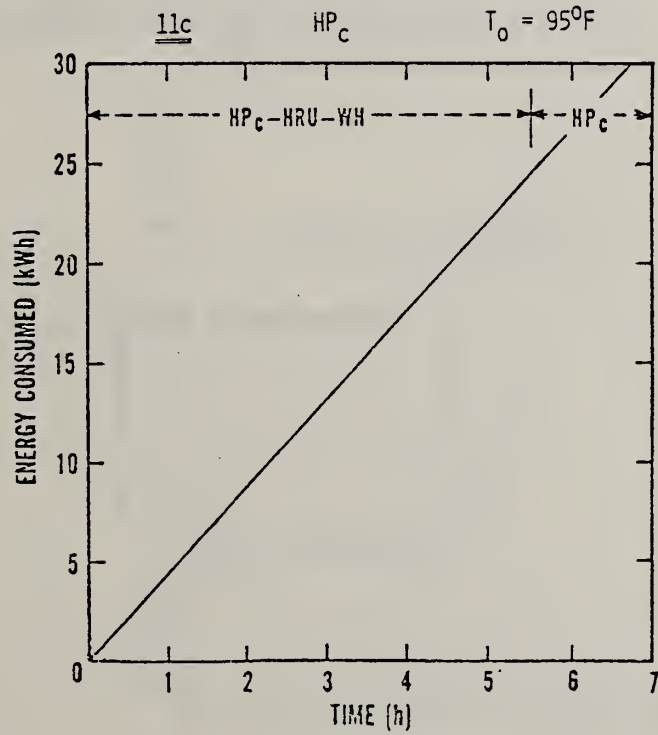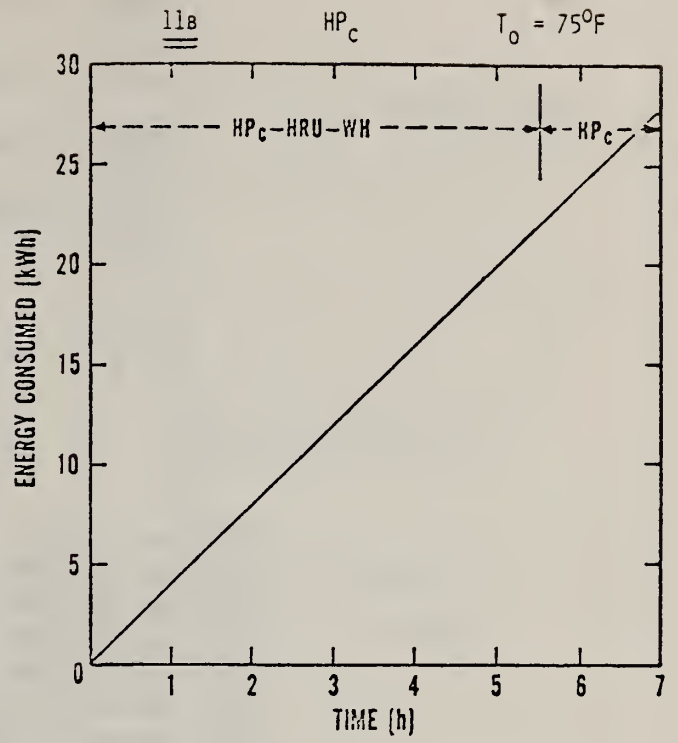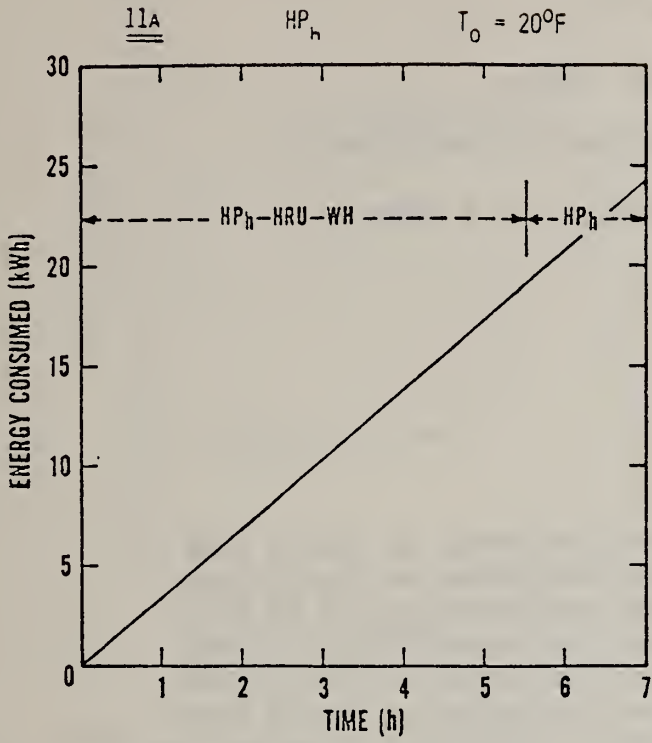
Figures II-11a-c: Energy Consumed versus Time of Operation of the HP - HRU - WH Integrated Appliance, Using HRU No. 1

TC   =   Thermocouple

HRU   =   Heat Recovery Unit

WH   =   Water Heater

HP_c =   Heat Pump in Cooling Mode

T_o =   Nominal Outdoor Temperature

TC 1 is 3-7/8" from the Bottom of the WH Tank
TC 2 is 11-5/8" from the Bottom of the WH Tank
TC 3 is 19-3/8" from the Bottom of the WH Tank
TC 4 is 27-1/8" from the Bottom of the WH Tank
TC 5 is 34-7/8" from the Bottom of the WH Tank
TC 6 is 42-5/8" from the Bottom of the WH Tank
TC 7 measures the water temperature entering the HRU
TC 8 measures the water Cut-Out Temperature leaving the HRU
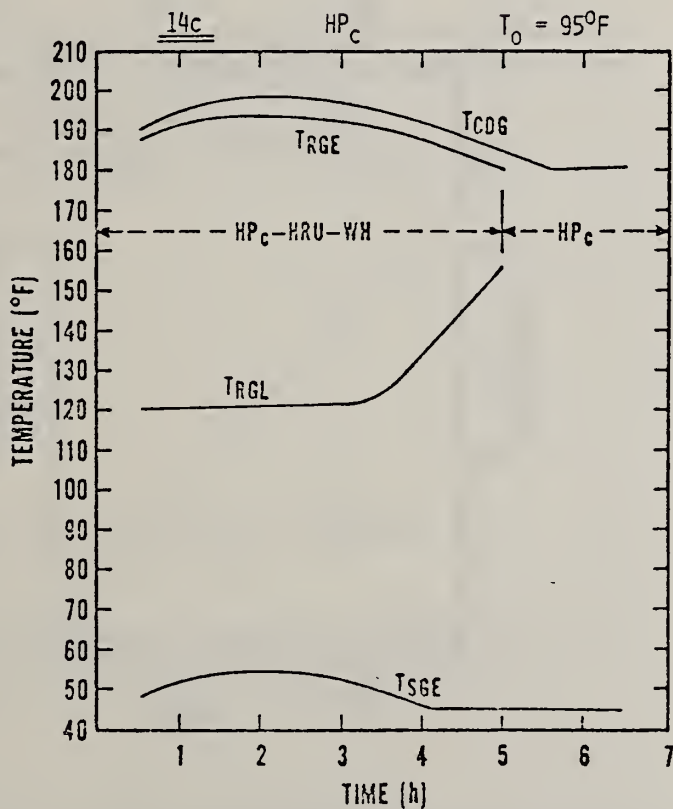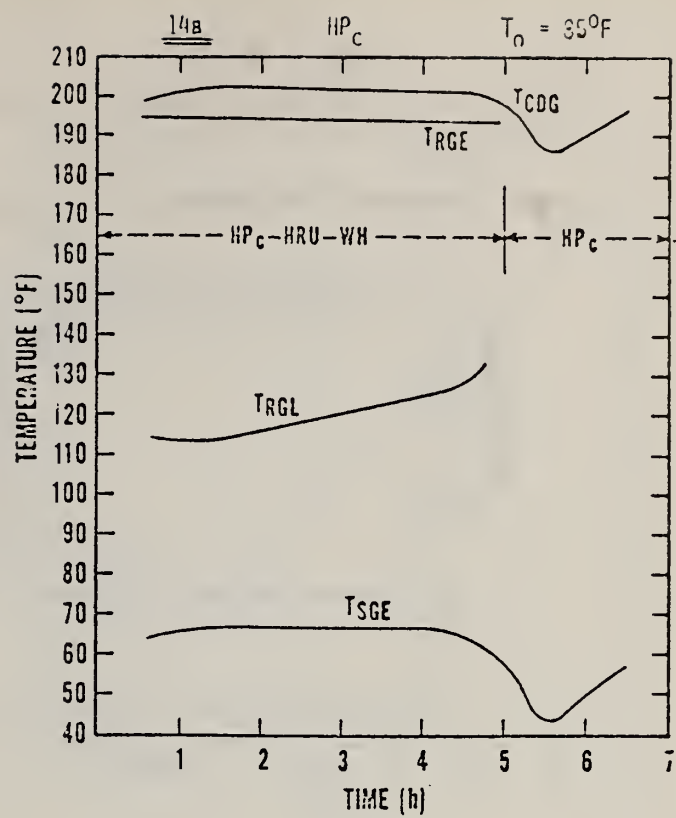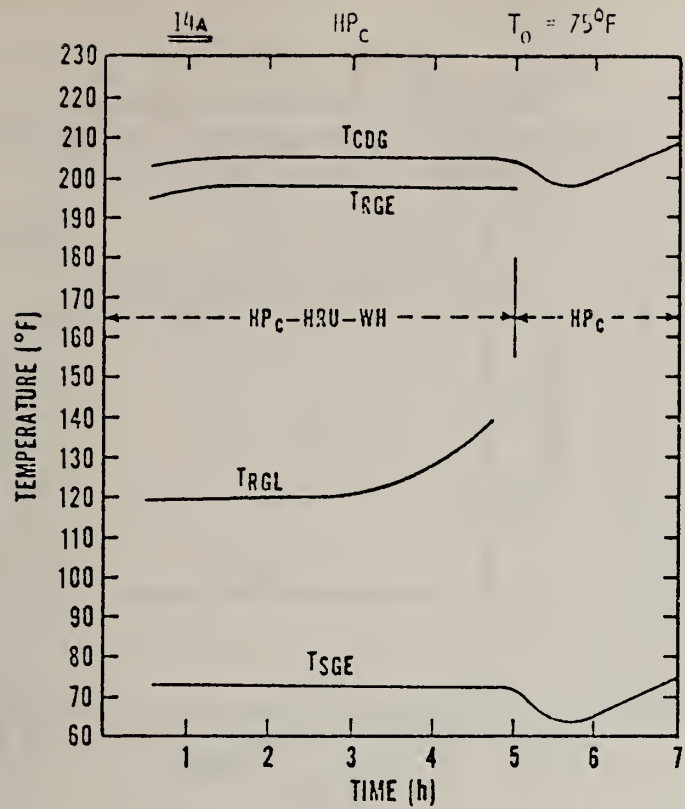TC 9 measures the water Cut-In Temperature leaving the HRU

Addendum to Figure II-12A-C

Figures II-12a-c: Water Temperatures at Various Locations in the WH versus Time of Operation of the HP - HRU - WH Integrated Appliance, Using HRU No. 2
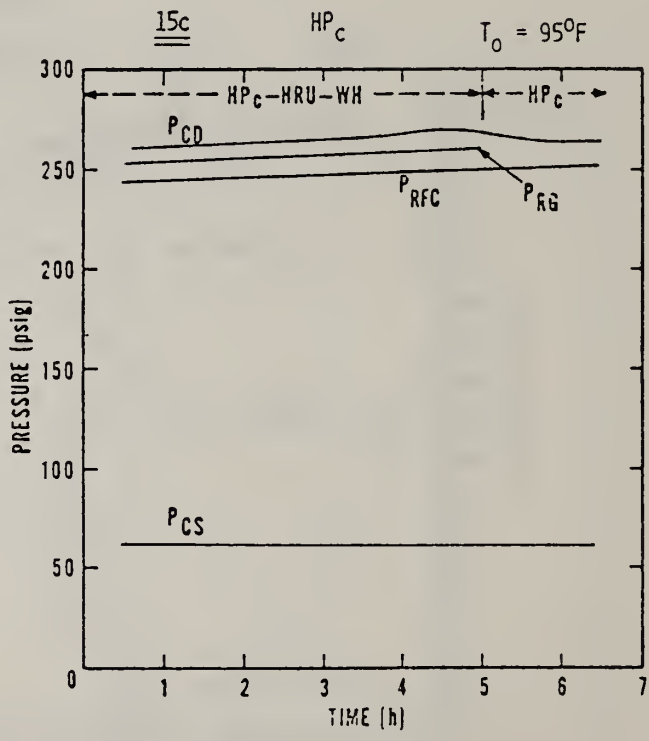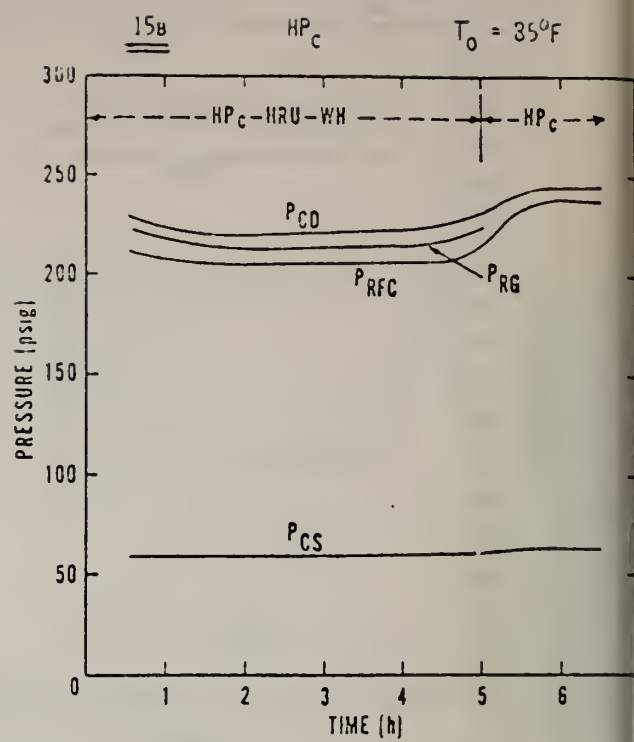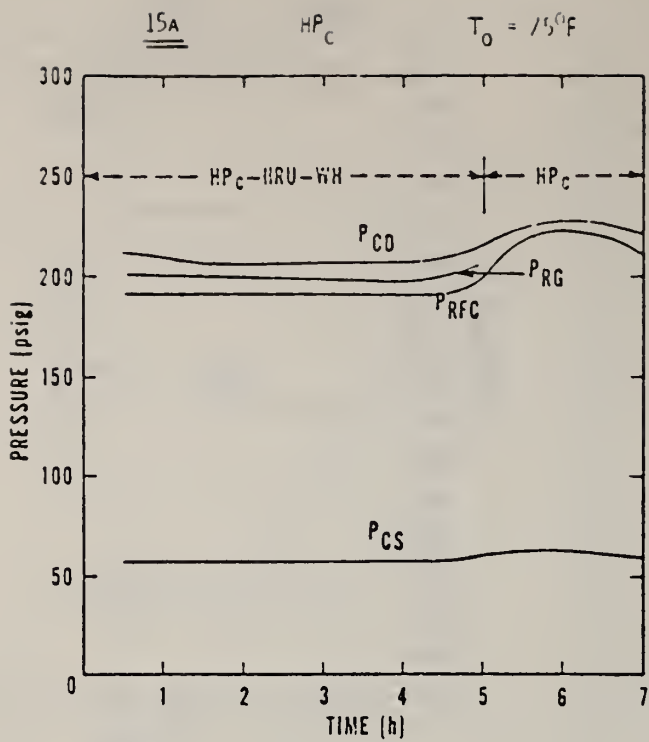
39
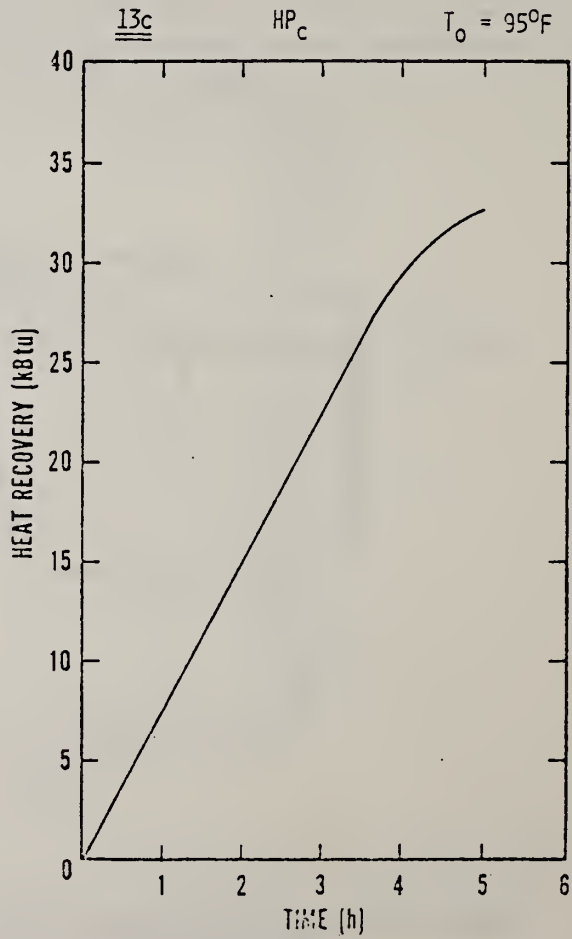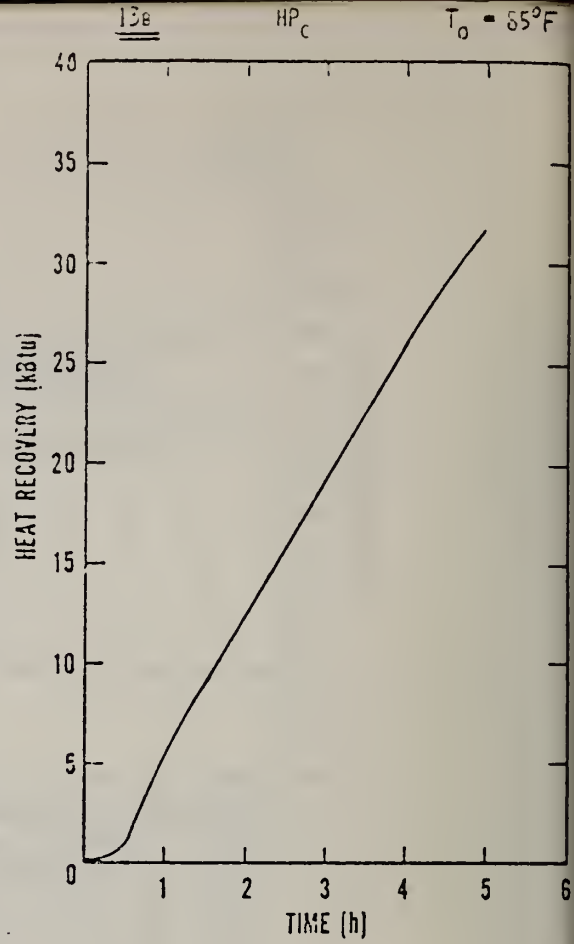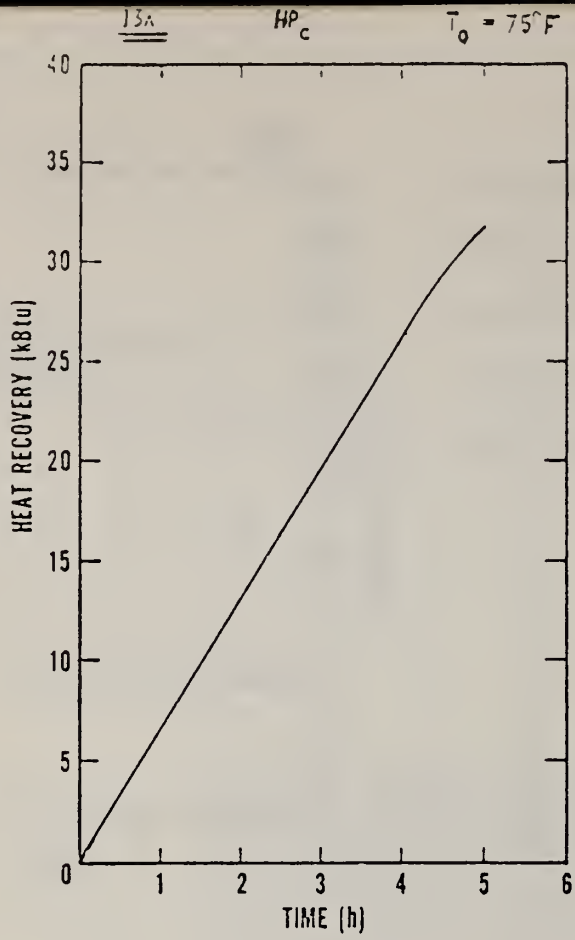
FIGURES II-13A-C: HEAT RECOVERY VERSUS TIME OF OPERATION OF THE
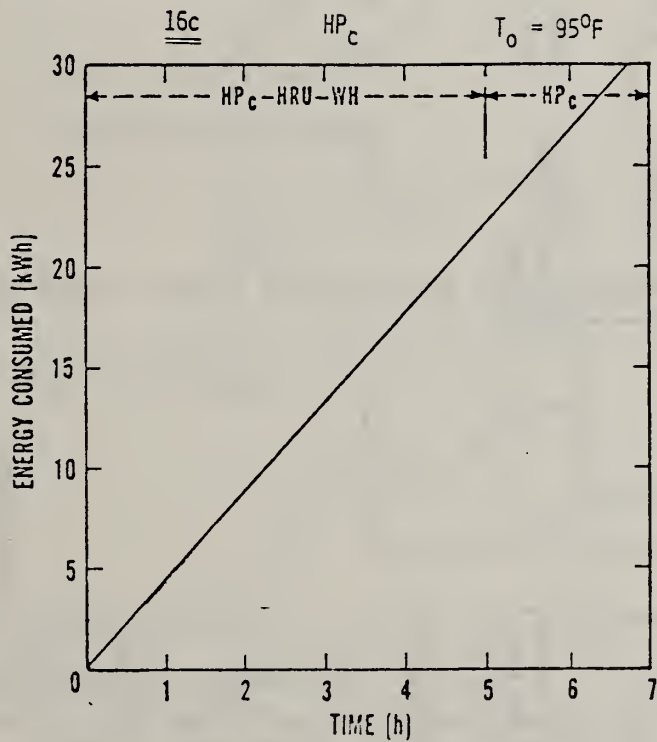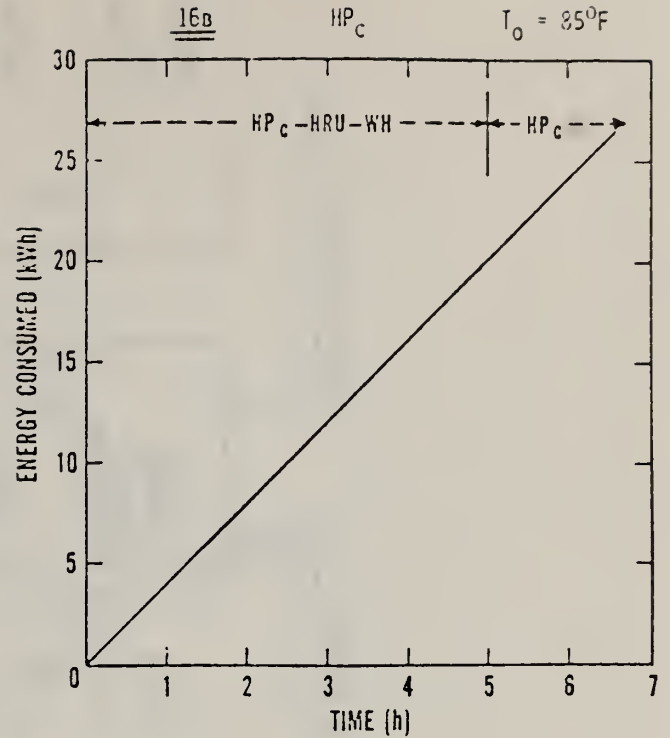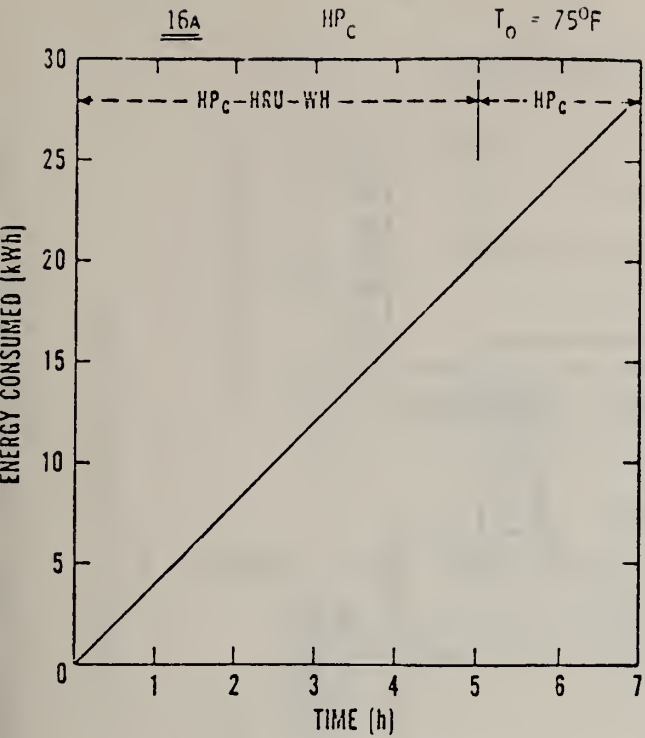HP - HRU - WH INTEGRATED APPLIANCE, USING HRU No. 2

FIGURES II-14A-C: REFRIGERANT TEMPERATURES VERSUS TIME OF OPERATION FOR THE
HP - HRU - WH INTEGRATED APPLIANCE, USING HRU NO. 2

NOTES: T_{CDG} = TEMPERATURE OF COMPRESSOR DISCHARGE GAS

T_{RGE} = TEMPERATURE OF REFRIGERANT GAS ENTERING HRU

T_{RGL} = TEMPERATURE OF REFRIGERANT GAS LEAVING HRU

T_{SGE} = TEMPERATURE OF SUCTION GAS ENTERING COMPRESSOR

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FIGURES II-15A-C: REFRIGERANT PRESSURES VERSUS TIME OF OPERATION OF THE
HP - HRU - WH INTEGRATED APPLIANCE, USING HRU NO. 2

NOTES: P_{CD} = COMPRESSOR DISCHARGE PRESSURE PRECEDING GAS LINE TO HRU

P_{RG} = REFRIGERANT GAS PRESSURE FOLLOWING GAS LINE FROM HRU

P_{CS} = COMPRESSOR SUCTION PRESSURE

P_{RFC} = REFRIGERANT PRESSURE FOLLOWING CONDENSER

Figures II-16a-c: Energy Consumed versus Time of Operation of the
HP - HRU - WH Integrated Appliance, Using HRU No. 2

43
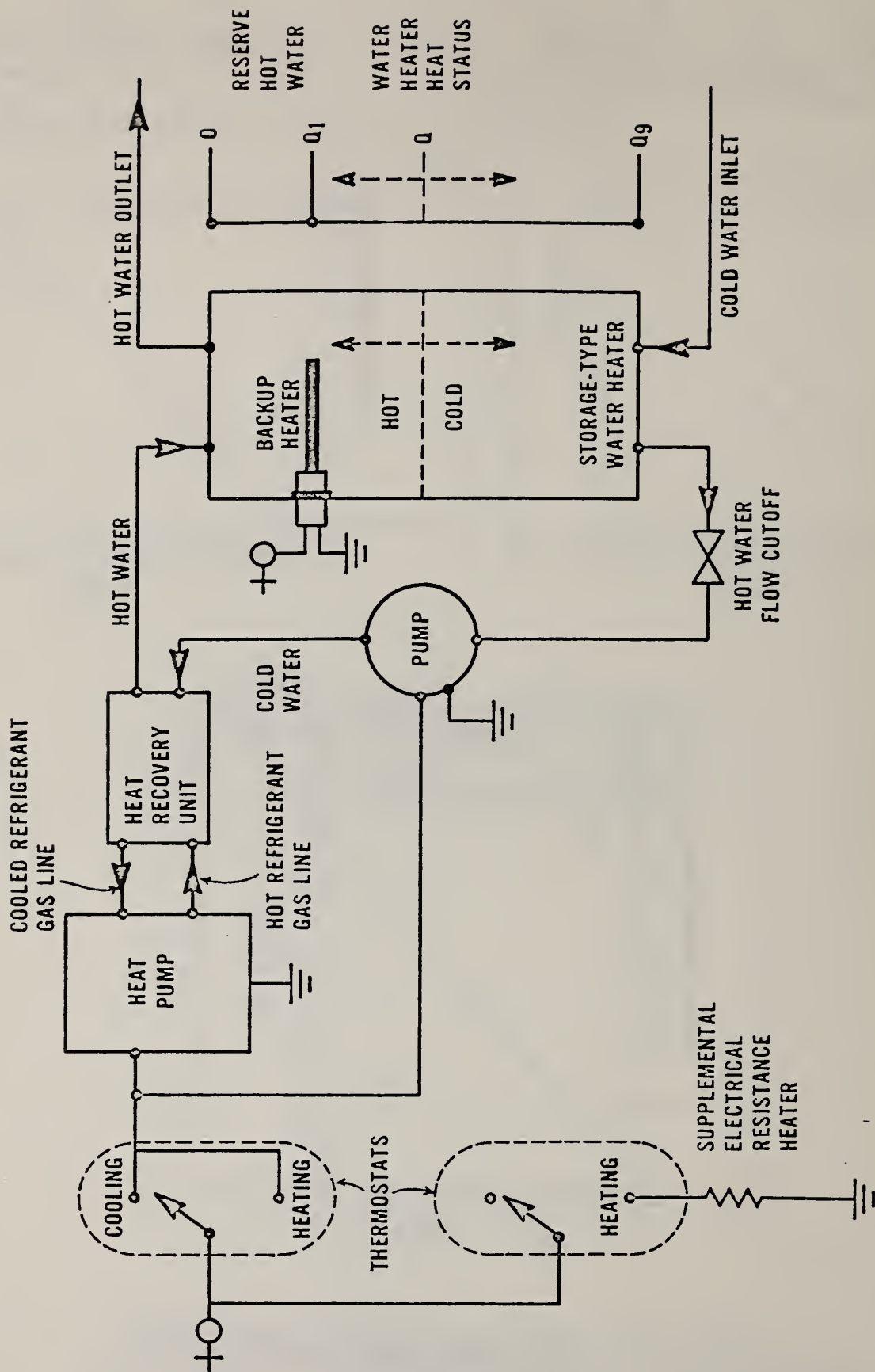
Figure III-1: MODIFIED AC/HP – HRU –. WH INTEGRATED APPLIANCE SYSTEM

44
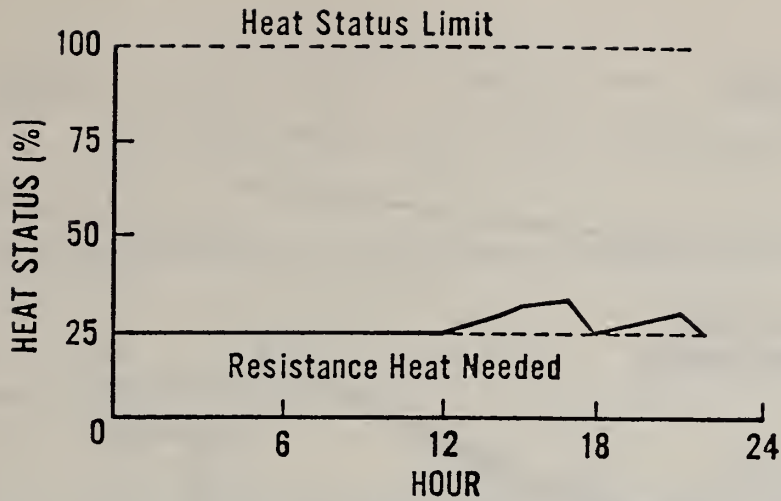
a. Sample hourly water heater status using a 40 gallon tank
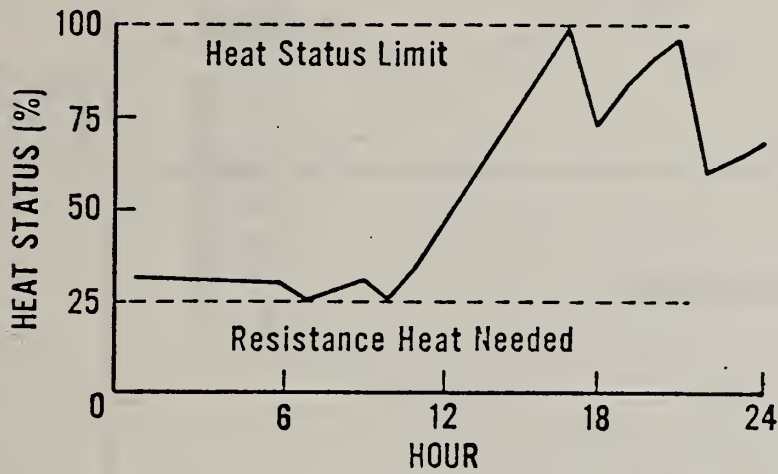
b. Sample hourly water heater status using a 40 gallon tank
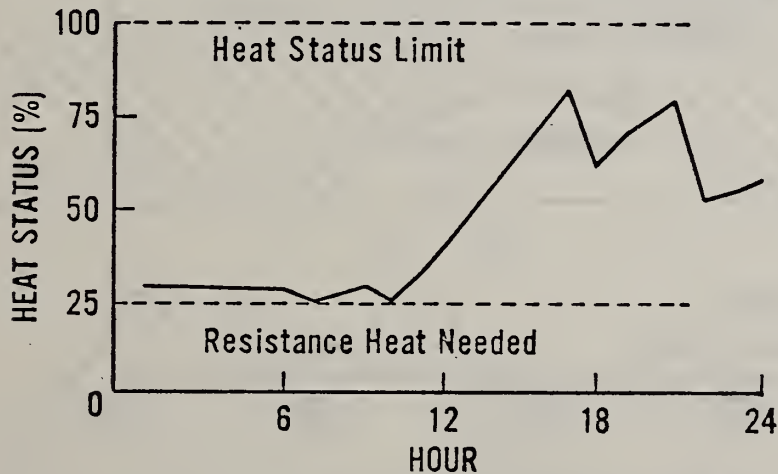
c. Sample hourly water heater status using a 52 gallon tank

Figure III-2 a, b, and c: Heat Status vs. Time of Day, for Washington, D.C.

a and b: different days,   b and c: same day

45
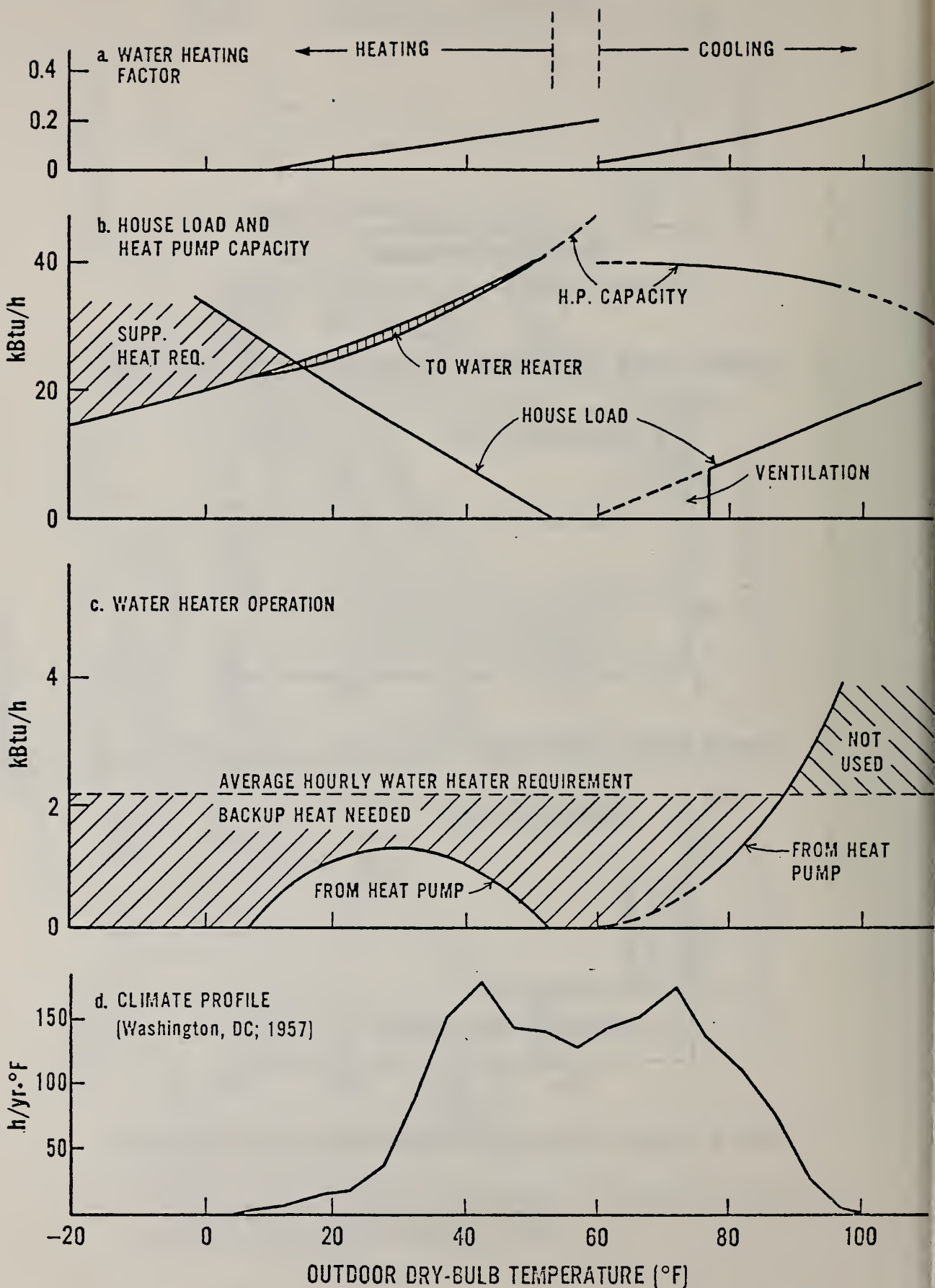
Figure III-3:   Heat Recovery Factors vs. Outdoor Temperature

46
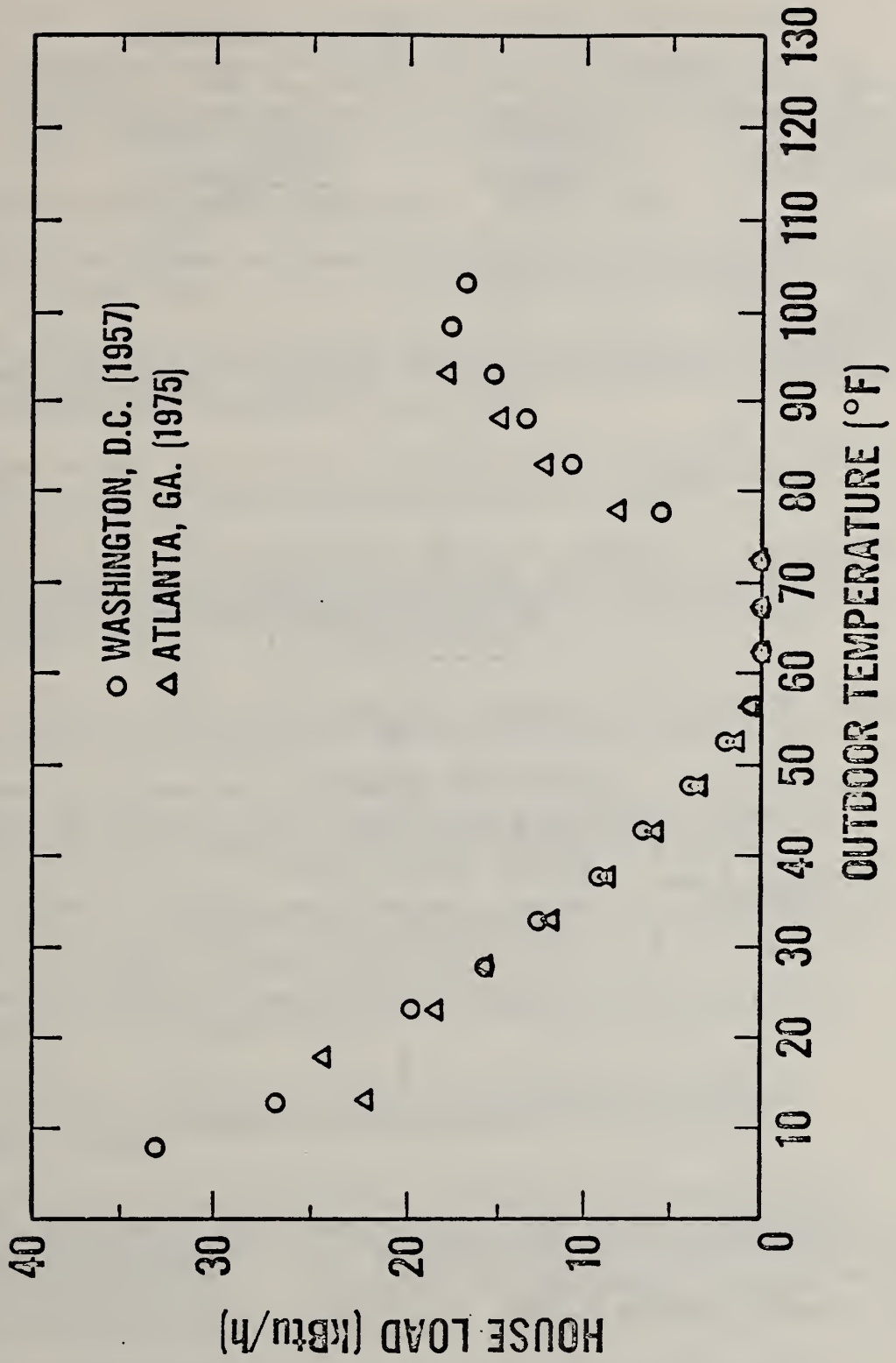
Figure III-4: NBSLD House Heating and Cooling Loads vs. Outdoor Dry-Bulb Temperature

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```
HOURLY
INPUTS
```

```
OUTDOOR            HOUSE          WATER DRAW        W H TANK
TEMPERATURE        LOAD           SCHEDULE          HEAT STATUS
```

```
H. P. COP                                          TANK STANDBY
& CAP,                                             HEAT LOSS
& THE WHF*
```

```
WATER DRAW
SCHEDULE
```

```
PRE-OFFER HEAT
STATUS OF TANK
```

```
CONTROLLING**
LIMITATION
```

```
HRU HEAT OFFERING
HRU HEAT OUTPUT
HOURLY          BACKUP HEATER LOAD
OUTPUTS         SUPPLEMENTAL HEATER LOAD
H.P. ELECTRICAL ENERGY USED
NEW TANK HEAT STATUS
```

```
NEXT HOUR'S INPUT
```
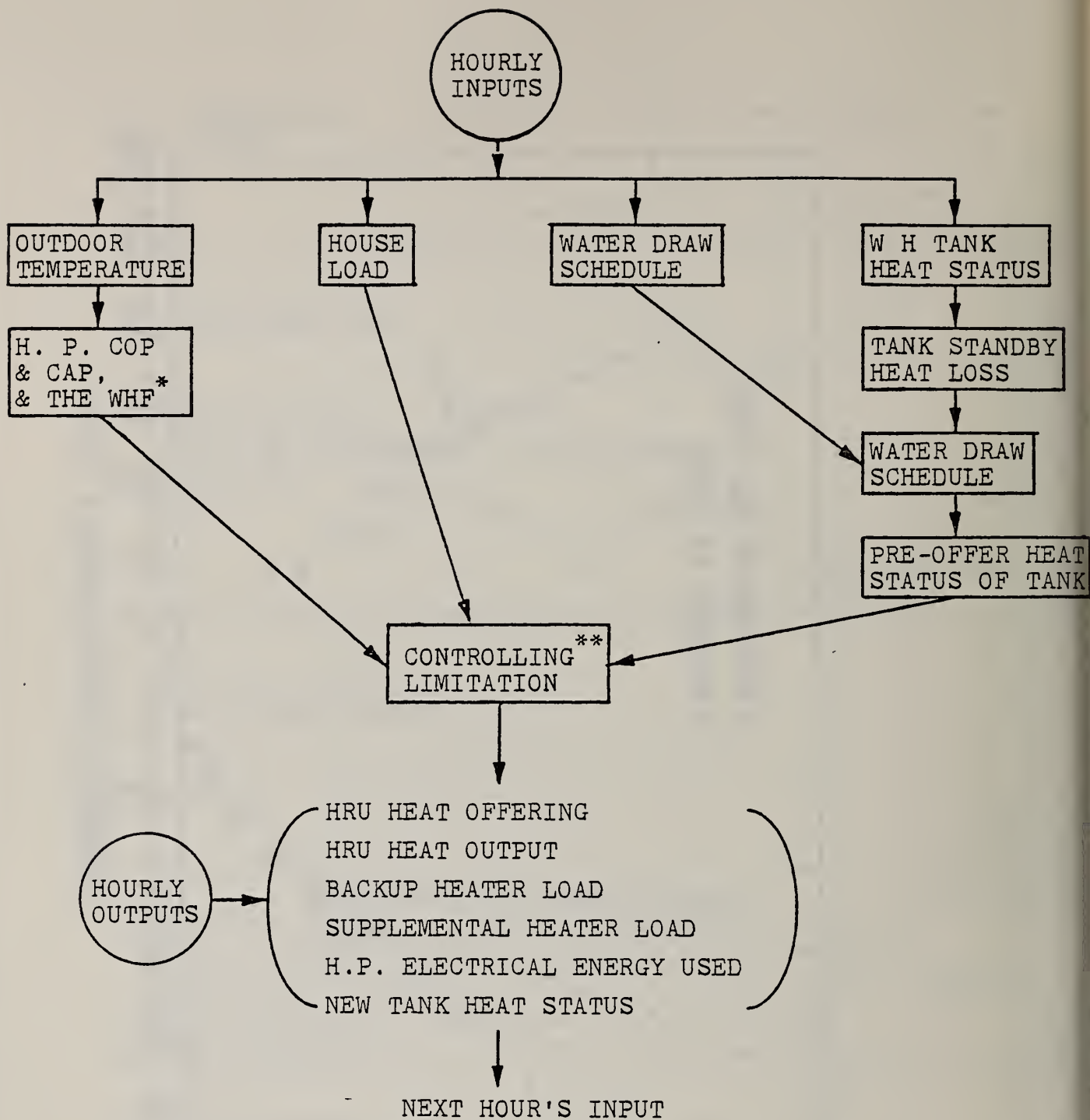
*COP = COEFFICIENT OF PERFORMANCE   **HEAT PUMP CAPACITY,
 CAP = CAPACITY (HEATING OR COOLING)   HOUSE LOAD, OR
 WHF = WATER HEATER FACTOR   W.H. TANK STATUS

Figure III-5:   Simulation Outline Diagram

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Appendix A. Simulation Background and Related Data

A-I. House Heating, Cooling and Hot Water Loads

Hourly data developed on another project at NBS were used as the source of outdoor temperatures and loads on the heating and cooling system. Detailed weather data and a suitable description of a house and its operation were analyzed using portions of the NBSLD (National Bureau of Standards Load Determination) program. Weather data for several years have been examined for several locations in the United States, and for each location a reasonably typical year, called a Test Reference Year (TRY) has been chosen for use in weather dependent studies.   The TRY's are 1957 for Washington, DC, 1975 for Atlanta, GA, 1974 for Chicago, IL, and 1960 for Seattle, WA. Dry- and wet-bulb temperatures, wind speeds, cloud cover and solar radiation are among the important environmental input data.

The same house was used in all locations for which the load data are available and has the following features:

o   Single family, one story detached dwelling

o   1800 square feet of floor over a crawl space (49' x 36' 9")

o   Insulation includes R-38 for the ceiling, R-13 for the walls, R-19 for the floor and double glazing for windows and glass doors.

o   Infiltration was calculated separately for each hour, depending on wind speed and the indoor/outdoor temperature differential.

o   Thermostat settings were 68°F for heating and 78°F for cooling.

o   Indoor relative humidity was maintained between 20 percent and 60 percent.

o   Natural ventilation was used only when it could keep the house temperature at 68 to 78°F, while maintaining the indoor relative humidity in the specified range. The amount of ventilation was variable and vents were closed if the humidity requirement could not be met with ventilation or the ventilated indoor dry-bulb temperature would be either above or below 78°F.

o   Occupants were two adults and two children.

o   The use of lights and equipment was given on an hourly schedule.

Data used in the present simulation include the hourly streams of outdoor temperature and the heating and cooling loads on the environmental control equipment. A daily cycle of hourly draws of hot water that was developed at NBS was added to complete the description of the environment in which the heat recovery system operates. This schedule is given in Table A-1, in which the fractional draws were first translated into gallons on the basis of 64.3 gallons of hot-water use per day, aggregated by hour of the day and translated into kBtu

on the basis of a 90°F temperature rise and 8.25 pounds per gallon. Heat losses from the WH were computed on the basis of one percent per hour of the thermal energy in the water (i.e., the tank heat status, Q) at the beginning of each hour. NBS tests indicate that this one percent factor is reasonable for well insulated WH.

The input data from the NBS house load simulation studies consist of the temperature for each hour and the corresponding sensible and latent heat movement loads on the heat pump. Sensible loads are the amounts of thermal energy that must be added or removed to control the indoor dry-bulb temperature. Latent loads refer to humidity control. Both loads were computed using the NBSLD program and are the resultants remaining after all other thermal energy gains and losses have been considered. These loads were added for each hour to yield the total load on the heat pump. Total loads for the year are the sums of the absolute values of these hourly loads.

A-II.   Heat Pump and Heat Recovery System

Three parameters were used to describe the essential characteristics of the HP and HRU. These are the HP capacity, CAP, the HP coefficient of performance, COP, and the water heating factor, WHF, for the HRU. WHF is the fraction of the HP output that is available for heating water. CAP and COP depend on both the indoor and outdoor temperatures and WHF depends on the temperature of the refrigerant leaving the compressor. CAP, COP and WHF were evaluated at the mid-points of five degree temperature bins from 110°F down to -40°F to cover the main climate range in the U.S., taking into consideration the indoor temperatures associated with heating and cooling situations. The reference HP for these estimates is one of the two used in the experimental phase of this study. It has a cooling capacity of 35,000 Btu per hour under standard ARI rating conditions of 95°F outdoor dry-bulb temperature and indoor temperatures of 80°F dry-bulb and 67°F wet-bulb. Rated heating capacities for 70°F indoors and outdoor dry- and wet-bulb combinations of 47°F/43°F and 17°F/15°F are 37,000 and 24,000 Btu per hour, respectively. The values for CAP, COP and WHF that were used in the simulation and in the alternate calculation procedures are given in Table A-2 and are compatible with the results of laboratory tests of a sample HP-HRU-WH system. The values for a given bin were used for all hours with outdoor temperatures falling within the range of the bin. Since it would be more economical to rely on supplemental resistance heat when the COP is less than 1.0, CAP and COP are set to zero for these cold-weather conditions.

The COPs apply to steady state operation at the corresponding CAPs. However, when the load, L, is less than the CAP, the HP will cycle on and off and this part load COP will be less than the full load COP. (If L exceeds CAP, the HP output is limited to CAP.) This means that actual seasonal operating costs will be somewhat higher than would be estimated using full load COP's throughout the year. A part load COP adjustment factor developed at NBS was used in the calculation of energy consumption for each hour or each bin, depending on the computing model in use.

$$\frac{\text{COP (part load)}}{\text{COP (steady state)}} = 0.75 + 0.25 \times \frac{L}{\text{CAP}} \qquad (1)$$

A-III. Simulation Processing

    The main loop in the hourly simulation was described in Section III, along
with two bin procedures. Figures A-1 and A-2 are more detailed flow diagrams of
the computational logic of the hourly simulation and the alternate methods,
respectively. Details of the initializing print array generation and printout
are not shown. A page of symbol definitions accompanies each flow diagram. The
bin procedures do not consider the hourly variations of the heat status, Q, of
the water heater tank. Annual results are derived by computing results for one
hour under stable conditions at the selected location for each bin and then
weighting the results by the number of hours per year at the location.
Otherwise, the engineering concepts are identical for the hourly and bin
methods.

    Briefly, the steps in the hourly simulation are:

    1.   Specify the COP, CAP and WHF (described above) together with the
          initial value of the tank heat status, Q, and the maximum and reserve
          heat statuses, Q9 and Q1, respectively, for the tank being modeled.

    2.   Read the outdoor temperature and the house load for each successive
          hour, carrying out the steps and making the accumulations described
          in Section III.

A-IV. Additional Temperature Profiles

    The outdoor dry-bulb temperature correlates quite well with the average
total sensible and latent house loads. Data from Bibliography Item 1 were used
to generate annual temperature profiles for several locations in the country.
The data were compiled in each case from hourly readings covering a period of
ten years or more. These profiles (Figures A-3—A-13) correspond in their
application to the climate profile at the bottom of Figure III-3.

Table A1: Water Draw Schedule

Time	Gallons			Hour Starting	kBtu
7: 1: 0	0.6			1	0
7: 1:24				2	0
7: 5: 0	15			3	0
7:10: 0				4	0
7:25: 0	0.6	· 16.2		4	0
7:25:24				5	0
8: 0: 0	0.6	0.6		6	0
8: 0:24				7	12.029
10: 0: 0	8			8	0.446
10: 2: 0				9	0
10:22: 0	4			10	9.356
10:23: 0				11	0
10:30: 0	0.6	12.6		12	0.891
10:30:24				13	0
12: 0: 0	0.6			14	0
12: 0:24				15	0
12:45: 0	0.6	1.2		16	0.446
12:45:24				17	0.520
16: 0: 0	0.6	0.6		18	11.583
16: 0:24				19	0
17:20: 0	· 0.7	0.7 ·		20	0
17:20:24				21	0
18: 0: 0	0.6			22	12.029
18: 0:24				23	0.446
18:20: 0	3			24	0
18:22: 0				Total	47.746
18:28: 0	3				
18:30: 0					
18:36: 0	3				
18:38: 0					
18:44: 0	3				
18:46: 0					
18:52: 0	3	15.6			
18:54: 0					
22: 0: 0	0.6				
22: 0:24					
22: 5: 0	15				
22:10: 0					
22:25: 0	0.6	16.2			
22:25:24					
23:30: 0	0.6	0.6			
23:30:24					
Total 64.3	64.3				

Table A2: Heat Pump Capacities (CAP), Heat Pump
Coefficients of Performance (COP) and
Water Heating Factors (WHF)

BIN	TDO[a]	CAP[b]	COP[c]	WHF[c]
1	107.5	30.15	2.05	0.33
2	102.5	32.25	2.17	0.28
3	97.5	34.15	2.29	0.24
4	92.5	35.65	2.40	0.20
5	87.5	36.75	2.51	0.16
6	82.5	37.45	2.59	0.13
7	77.5	37.9	2.68	0.10
8	72.5	38.3	2.77	0.07
9	67.5	38.6	2.86	0.05
10	62.5	38.8	2.94	0.03
11	57.5	42.1	2.80	0.18
12	52.5	40.2	2.73	0.16
13	47.5	38.0	2.66	0.14
14	42.5	35.3	2.57	0.12
15	37.5	32.7	2.47	0.11
16	32.5	30.0	2.36	0.09
17	27.5	27.8	2.25	0.07
18	22.5	25.9	2.14	0.05
19	17.5	24.1	2.02	0.04
20	12.5	22.3	1.91	0.02
21	7.5	20.7	1.79	0 (d)
22	2.5	19.4	1.66	0
23	-2.5	18.1	1.54	0
24	-7.5	16.8	1.41	0
25	-12.5	15.5	1.28	0
26	-17.5	14.3	1.15	0
27	-22.5	13.1	1.01	0
28	-27.5	0 (d)	0 (d)	0
29	-32.5	0	0	0
30	-37.5	0	0	0

(a) Outdoor dry-bulb temperatures at the midpoints of the bins.

(b) Full-load capacity, kBtu/h.

(c) Dimensionless.

(d) CAP and COP were set to zero for COP less than 1.0 because
the use of resistance heat alone would be more economical
than using the heat pump at all under these conditions.
WHF is zero when water at the required temperature cannot
be delivered to the tank.

Notes for Figure A-1

INT(xx) = Find the integer component of (xx).

TDO = Outdoor dry-bulb temperature for bin, deg. F.

HLS = Sensible house load, kBtu/h.

HLL = Latent house load, kBtu/h.

HLN = HLS + HLL = Net house load, kBtu/h.

WDR = Hot water draw for hour, kBtu.

BIN = Five-degree temperature bin containing TDO.

Q = Heat status of WH at start of hour, kBtu.

Q4 = Heat status of WH after jacket losses and WDR for the hour, kBtu.

Q9 = WH maximum heat status, kBtu.

Q1 = WH reserve heat status = Q9/4, kBtu.

BHLT = Cumulative backup heater load, kBtu.

   (Accumulation formulas take the form; New Total = Old Total + Increment.)

Q6 = Q9-Q4 = Heat that the WH can accept, kBtu.

HPL = Heat pump (HP) load, kBtu/h.

HPK = HP electricity use, kWh/h.

HPKT = Cumulative HPK, kWh.

HOF = Heat offered to WH, kBtu/h.

HOFT = Cumulative HOF, kBtu.

WHF = Water heater factor, dimensionless.

WHR = WH heat refusal, kBtu/h.

WHRT = Cumulative WHR, kBtu.

SHL = Supplemental heater load, kBtu/h.

SHLT = Cumulative SHL, kBtu.

SHKT = Cumulative SHK, kWh.

BHKT = Cumulative BHK, kWh.

Start

Read fixed data.
Initialize constants and arrays.

Next Day

(E) → Day Loop
1-365 or 366

Read 24 sets of
TDO, HLS, HLL

Next Hour

(A) → Hour Loop
1-24

$BIN = 22 - INT \times (TDO/5)$
$HLN = HLS + HLL$
$Q4 = 0.99 \times Q - WDR$

$HLN \neq 0$? — Yes → $Q6 = Q9 - Q4$ → $HLN > 0$? — Yes → (B) (Heating)

No ↓

$Q = Q4$

(C) → $Q \geq Q1$? — Yes

No ↓

$BHL = Q1 - Q$
$BHLT = BHLT + BHL$
$Q = Q1$

(A)

(E)

$SHKT = SHLT/3.412$
$BHKT = BHLT/3.412$ → To Printout

(B)

$HOF = WHF \times$ lesser of $(CAP$ and $\frac{HLN}{(1 - WHF)})$
$Q5 =$ lesser of $(Q6$ and $HOF)$
$HPL =$ lesser of $(CAP$ and $(HLN + Q5))$
$SHL =$ greater of $((HLN + Q5 - HPL)$ and $0)$
$SHLT = SHLT + SHL$
$HPK = \dfrac{4 \times HPL \times CAP}{3.412 \times COP \times (3 \times CAP + HPL)}$
$HPKT = HPKT + HPK$

(D)

(Cooling) ↓

$HPL =$ lesser of $(-HLN$ and $CAP)$
$HPK = \dfrac{4 \times HPL \times CAP}{3.412 \times COP \times (3 \times CAP + HPL)}$
$HPKT = HPKT + HPK$
$HOF = WHF \times HPL$
$HOFT = HOFT + HOF$

$Q6 > HOF$? — Yes → $Q = Q4 + HOF$ → (C)

No ↓

$Q = Q9$
$WHR =$ greater of $((HOF - Q6)$ or $0)$
$WHRT = WHRT + WHR$

(A)

(E)

Figure A-1. Computational Flow Diagram for Hourly Simulation Method

Notes for Figure A-2

There are 30 five-degree temperature bins covering the range from 110°F to -40°F. Data from curves are evaluated at the mid-point of each bin. There is one value per bin for the following data items:

COP = HP coefficient of performance, dimensionless.

CAP = HP capacity, kBtu/h.

WHF = Water heater factor, dimensionless.

HRS = Site bin-hours per year.

HL1 = House load, bin average from detailed simulation input tape, kBtu/h.

HL2 = House load, from linear regressions of HL1, kBtu/h.

HLX = HL1 for Bin Average method = HL2 for Bin Linear method, kBtu/h.

Bin variables derived in the course of the analysis include:

HPL = HP load, kBtu/h. (HPL is not allowed to exceed CAP).

HPK = HP kWh for the bin, corrected for part load effects on the HP, kWh/yr.

$$= \text{HRS} \times \frac{\text{HPL}}{3.412 \times \text{COP}} \times \frac{4 \times \text{CAP}}{3 \times \text{CAP} + \text{HPL}}$$

SHK = Supplemental heater load for bin, kWh/yr.

HOF = Heat offered to WH, kBtu/h.

WHD = Average hourly heat demand by WH, covering water draws and jacket losses. For a 52 gallon tank,

$$= (\frac{52 \text{ gal}}{4} \times .01 \text{ loss per hour} + 365 \times 64.3 \text{ gal per day}/8760 \text{ hrs per yr.})$$

$$\times 8.25 \text{ lb per gal} \times 90°F \text{ rise}/1000 = 2.086 \text{ kBtu/h}.$$

WHA = Water heater acceptance, kBtu per hour.

WHK = Electricity demand on backup heater = HRS x WHA /3.412, kWh/yr.

Comparable bin results without heat recovery are also computed;

SHKO = Supplemental heater electrical load, kWh/yr.

HPLO = Heat pump load, kBtu/h.

HPKO = Heat pump electrical load, kWh/yr.

WHKO = Backup heater electrical load, kWh/yr.

Start → Read site and system data.
Initialize arrays and constants.

Next Bin
(C) → Main loop
Bins 1-30

Next Method
(D) → Methods
1, 2

Method 1 = Bin Average; HLX = HL1
Method 2 = Bin Linear ; HLX = HL2

Set HLX

Heating Load? — Yes →

No ↓

HPL = lesser of (HLX and CAP)
HOF = WHF x HPL

$HOF = WHF \times$ lesser of $(\frac{HLX}{(1-WHF)}$ and CAP$)$

WHA = lesser of (HOF and WHD)

SHK = greater of
$(0$ and $\frac{HRS}{3.412} \times (HLX + WHA - CAP))$

HPL = lesser of ((HLX + WHA) and CAP)

$HPK = \frac{HRS \times 4 \times HPL \times CAP}{3.412 \times COP \times (3 \times CAP + HPL)}$

$WHK = \frac{HRS}{3.412} \times$ greater of
$(0$ and $(WHD - WHF \times HPL))$

Compute kWh loads without heat recovery
(Suffix = 0):

$SHKO = \frac{HRS}{3.412} \times$ greater of $(0$ and HLX - CAP$)$

HPLO = lesser of (HLX and CAP)

$HPKO = \frac{HRS \times 4 \times HPLO \times CAP}{3.412 \times COP \times (3 \times CAP + HPLO)}$

$WHKO = \frac{HRS \times WHD}{3.412}$

(C) ← (D)

Accumulate item totals

to printout

Figure A-2. Flow Diagram for Bin Average and Bin Linear Methods

58
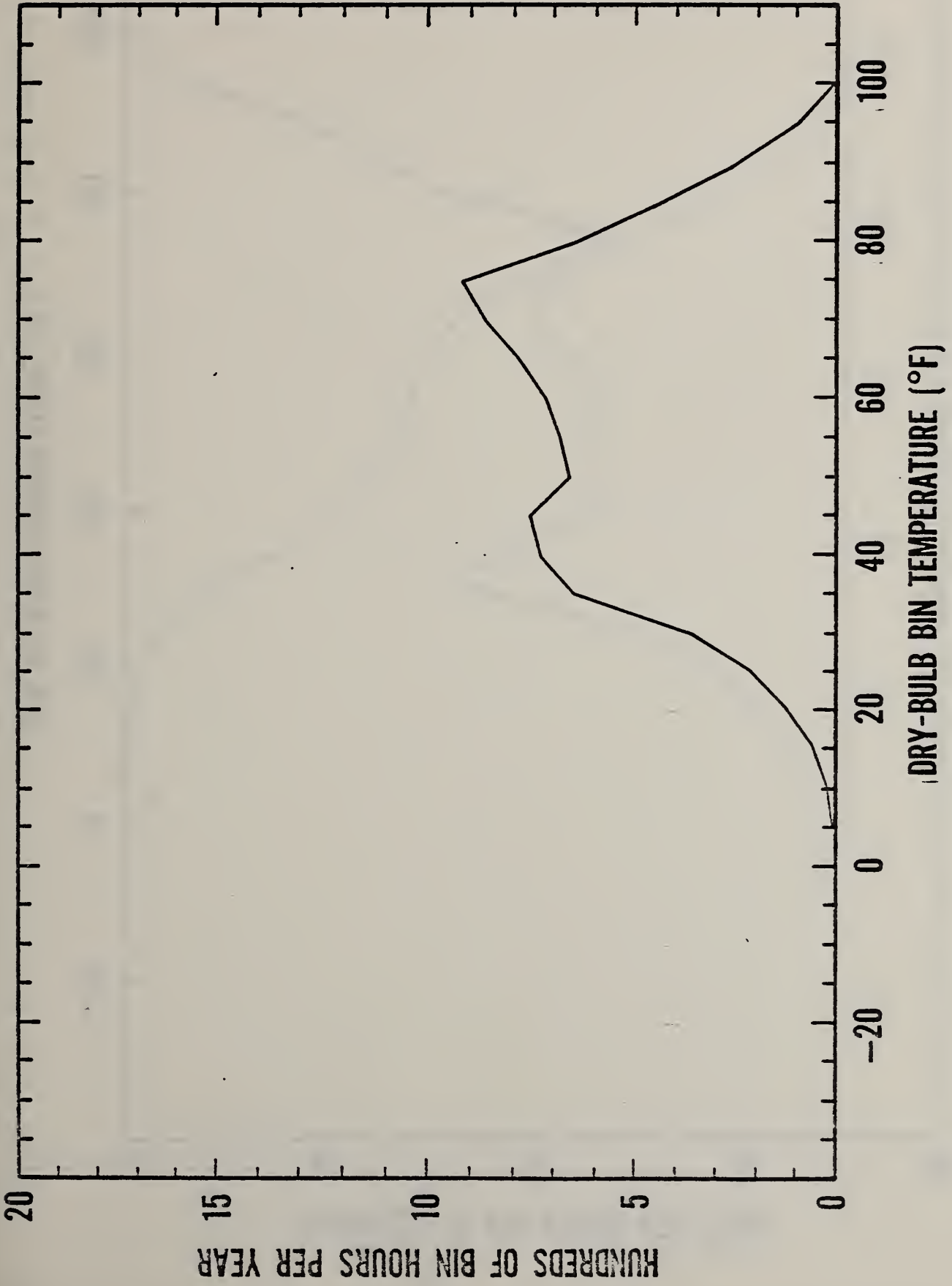
Figure A3: Bin Hours per Year versus Dry-Bulb Bin Temperature for Andrews AFB, MD.
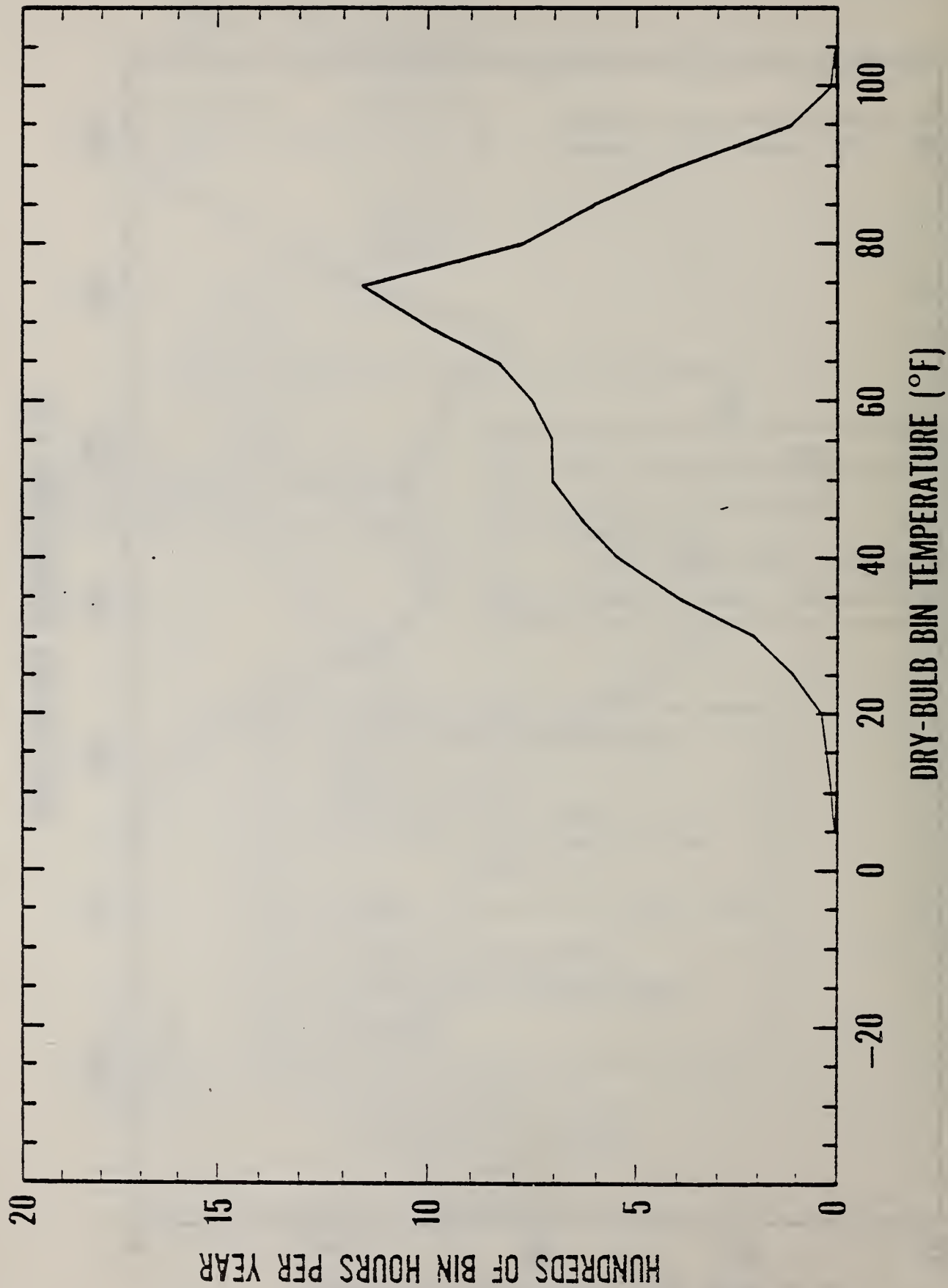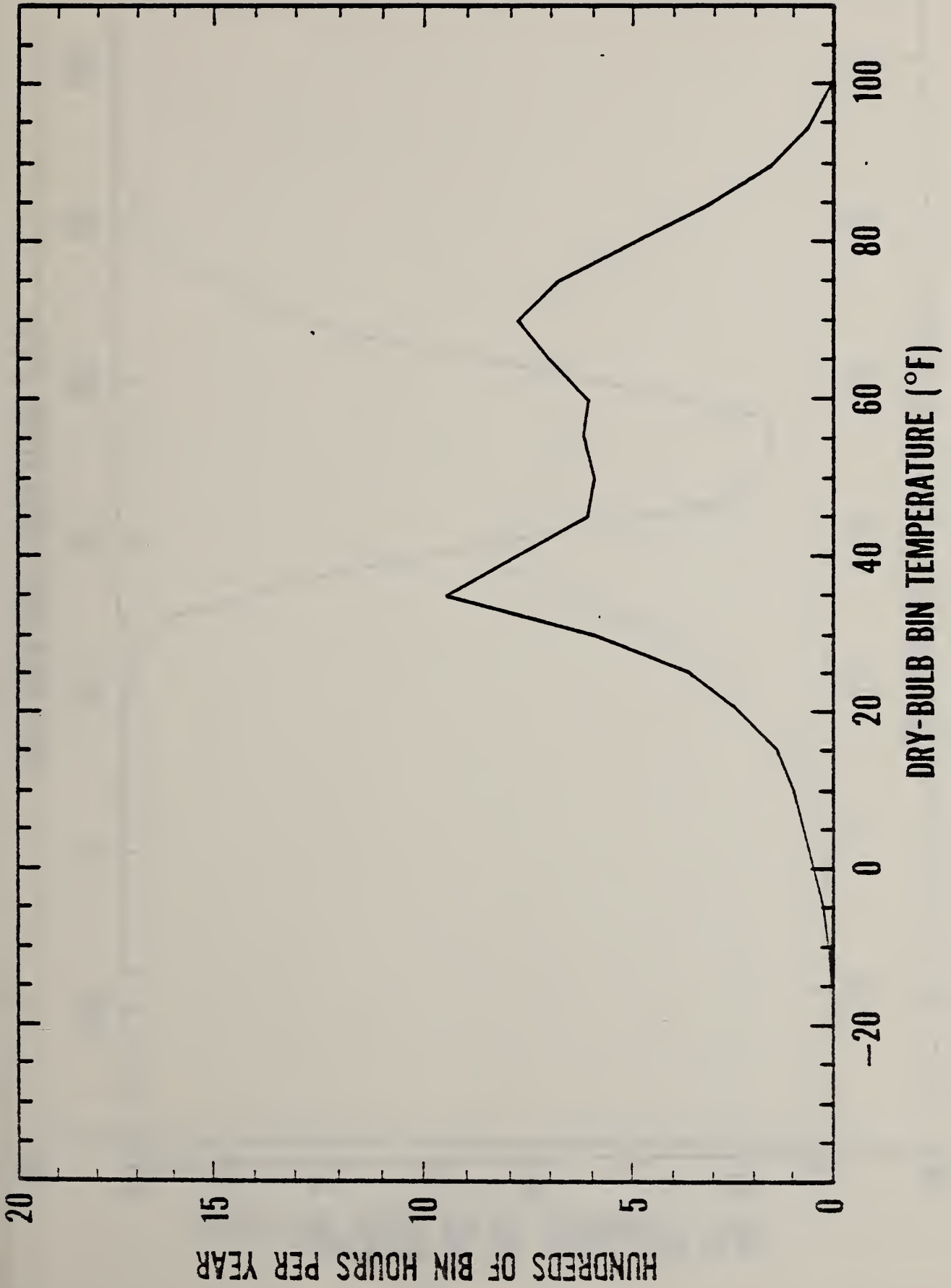
Figure A5: Bin Hours per Year versus Dry-Bulb Bin Temperature for O'Hare IAP, IL.
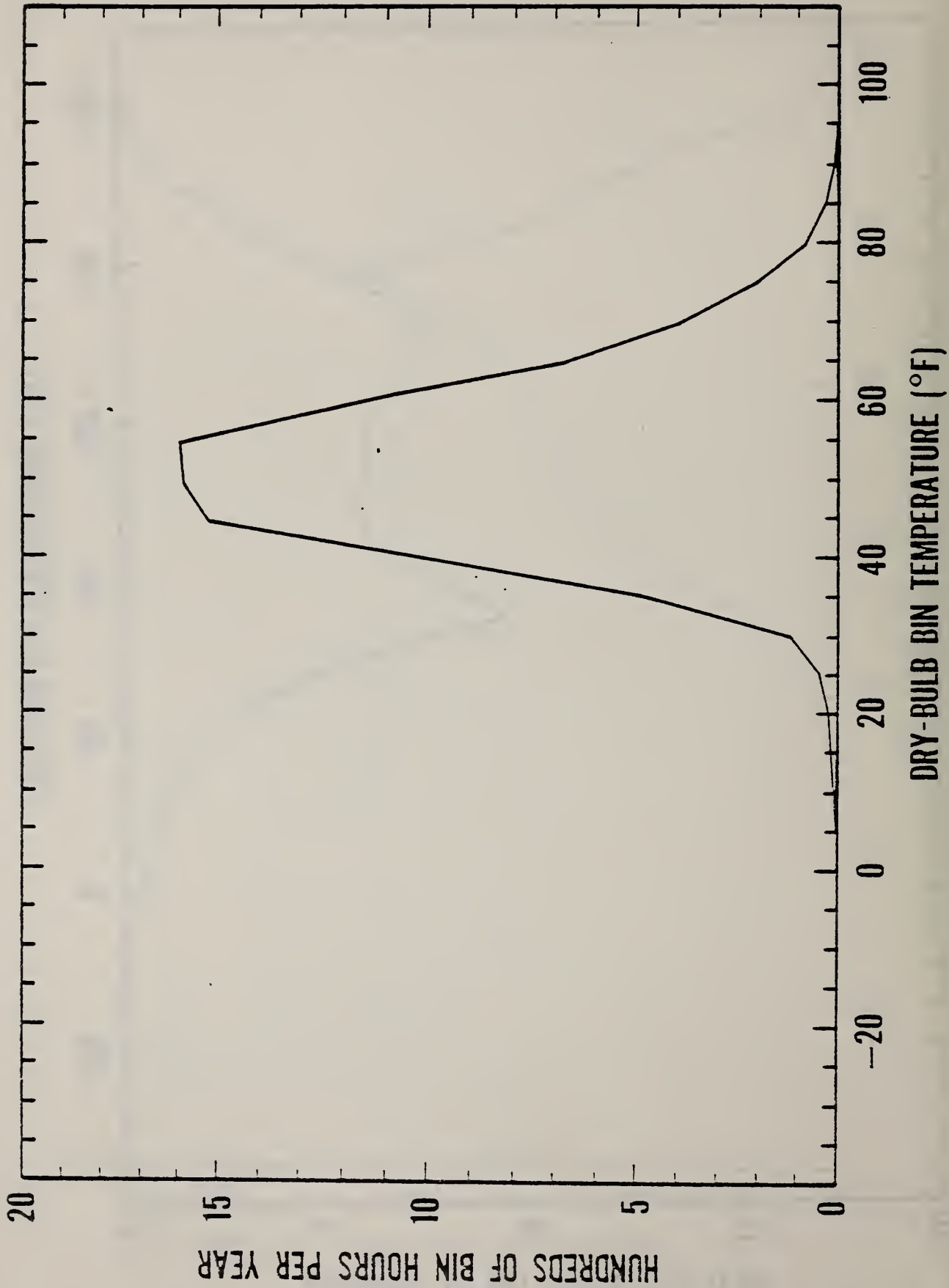
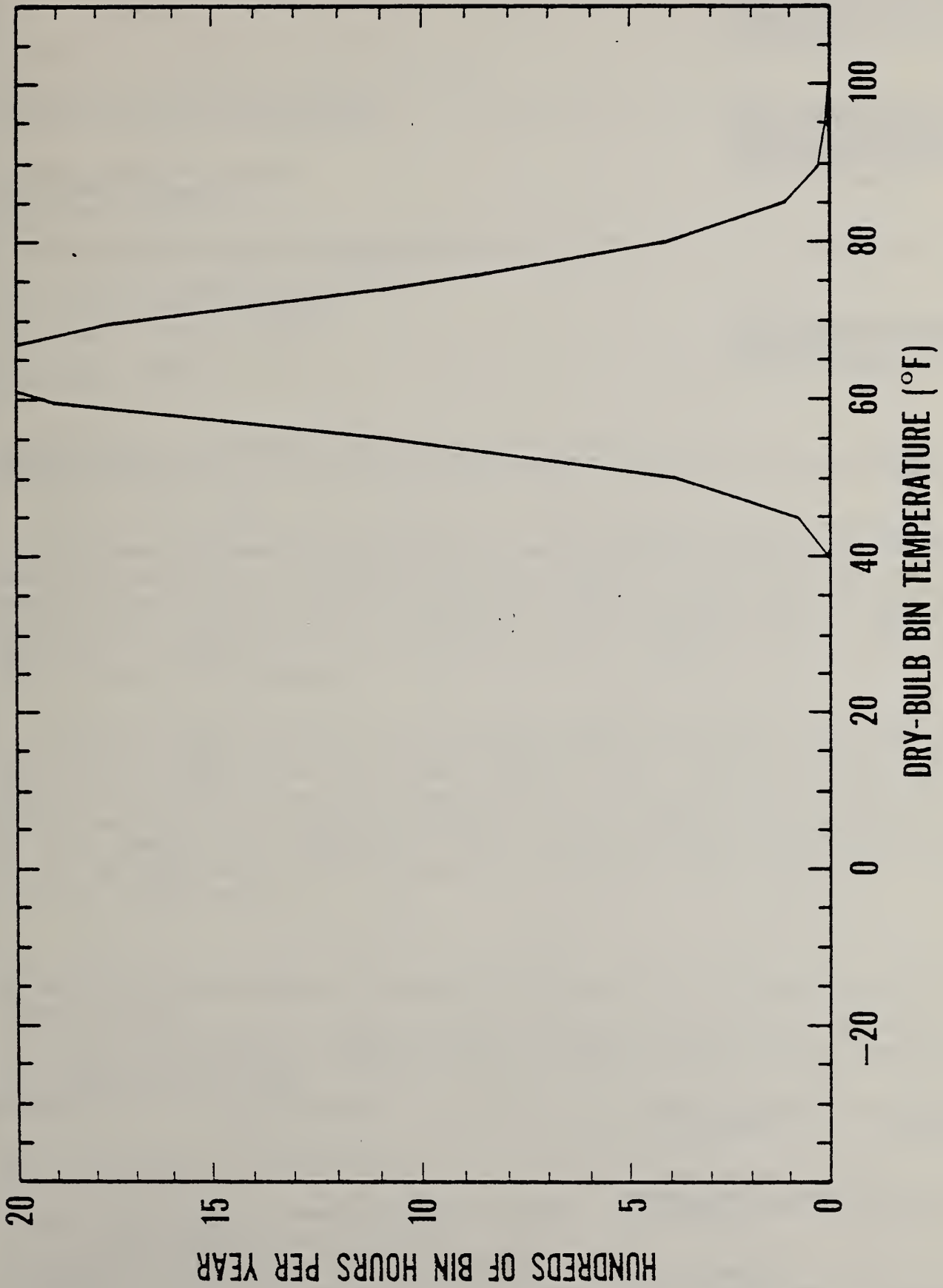Figure A6: Bin Hours per Year versus Dry-Bulb Bin Temperature for Everett, WA
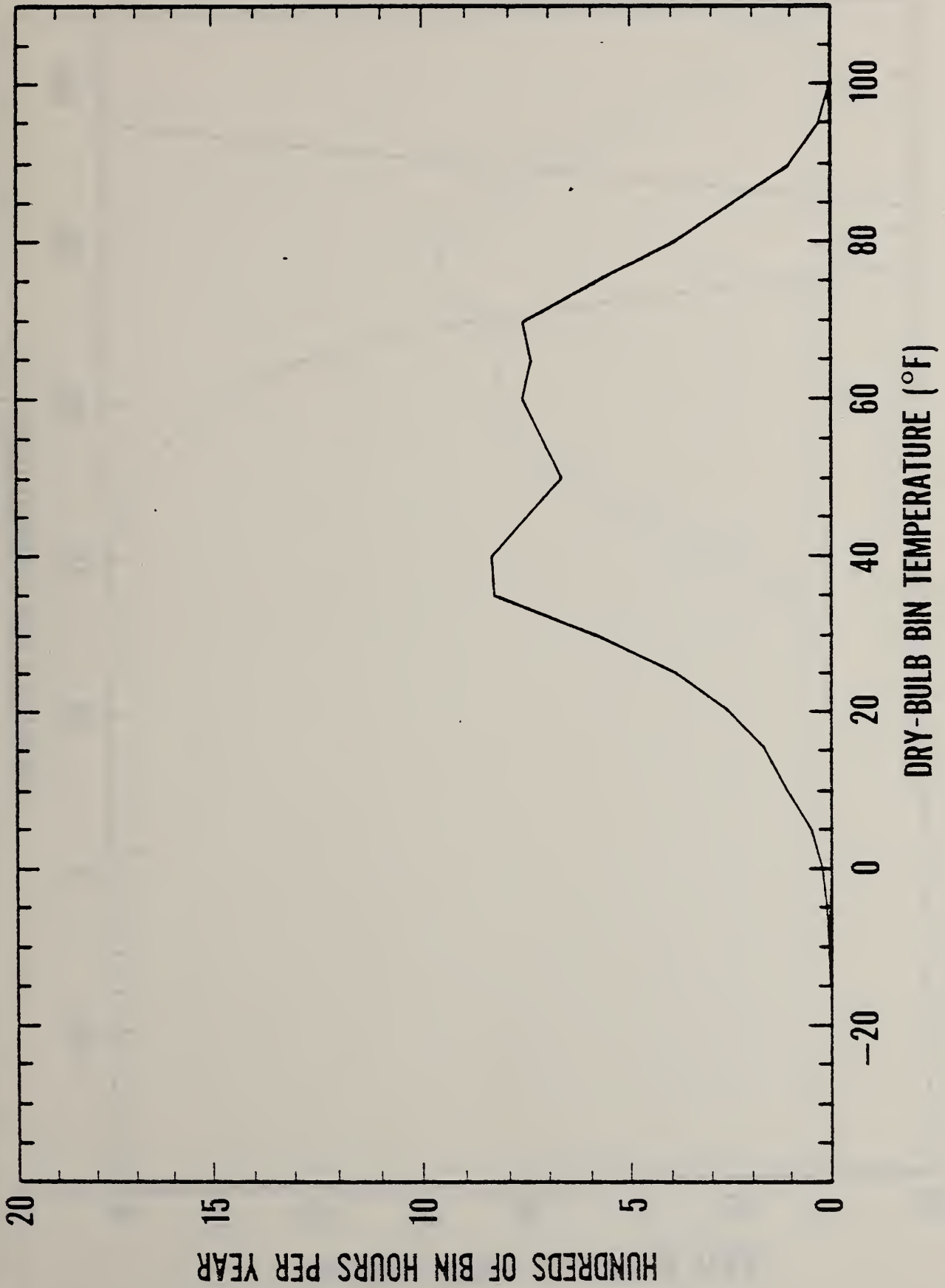
Figure A7: Bin Hours per Year versus Dry-Bulb Bin Temperature for Hanscom Field, MA
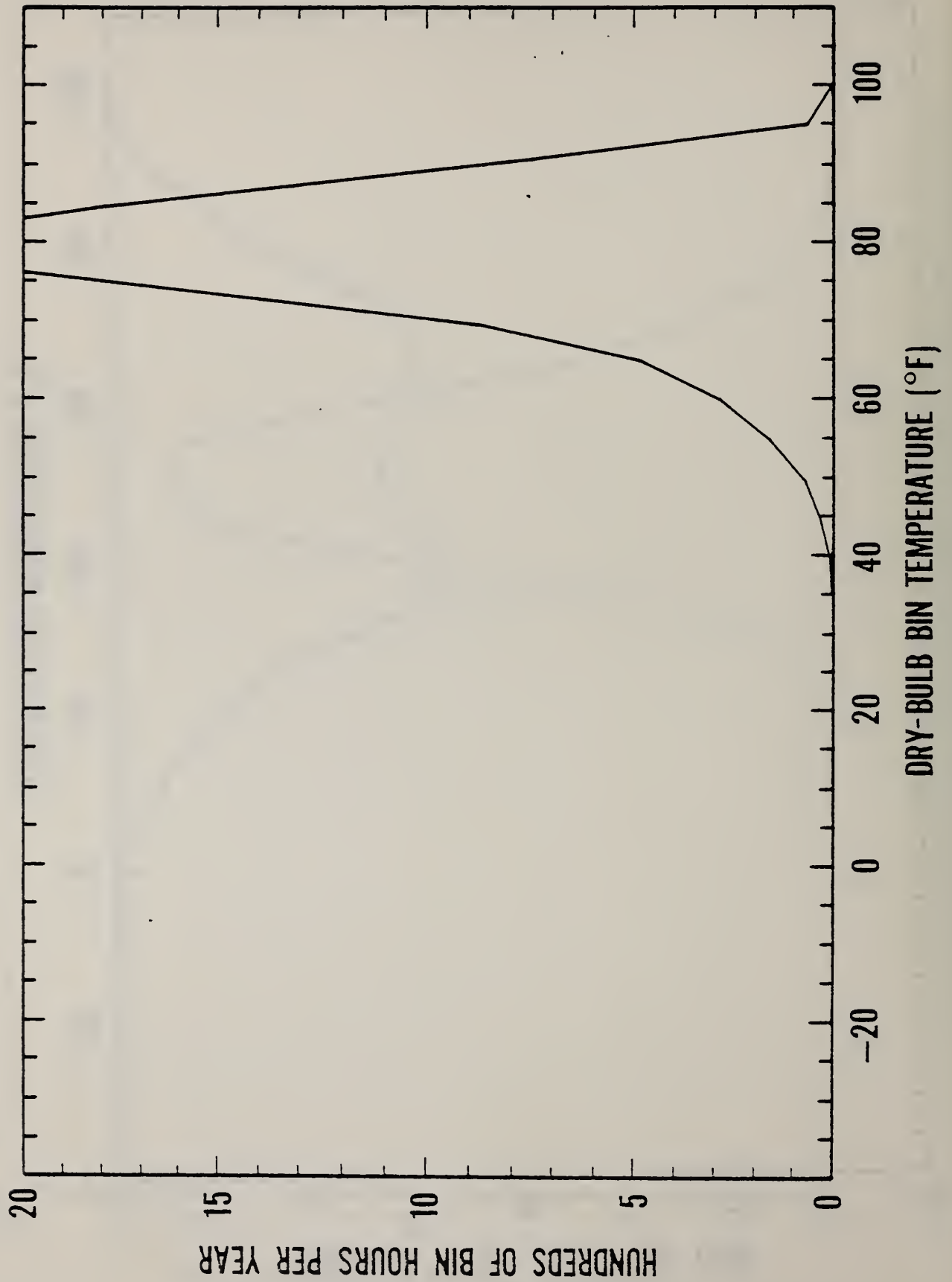
Figure A8: Bin Hours per Year versus Dry-Bulb Bin Temperature for Homestead AFB, FL.
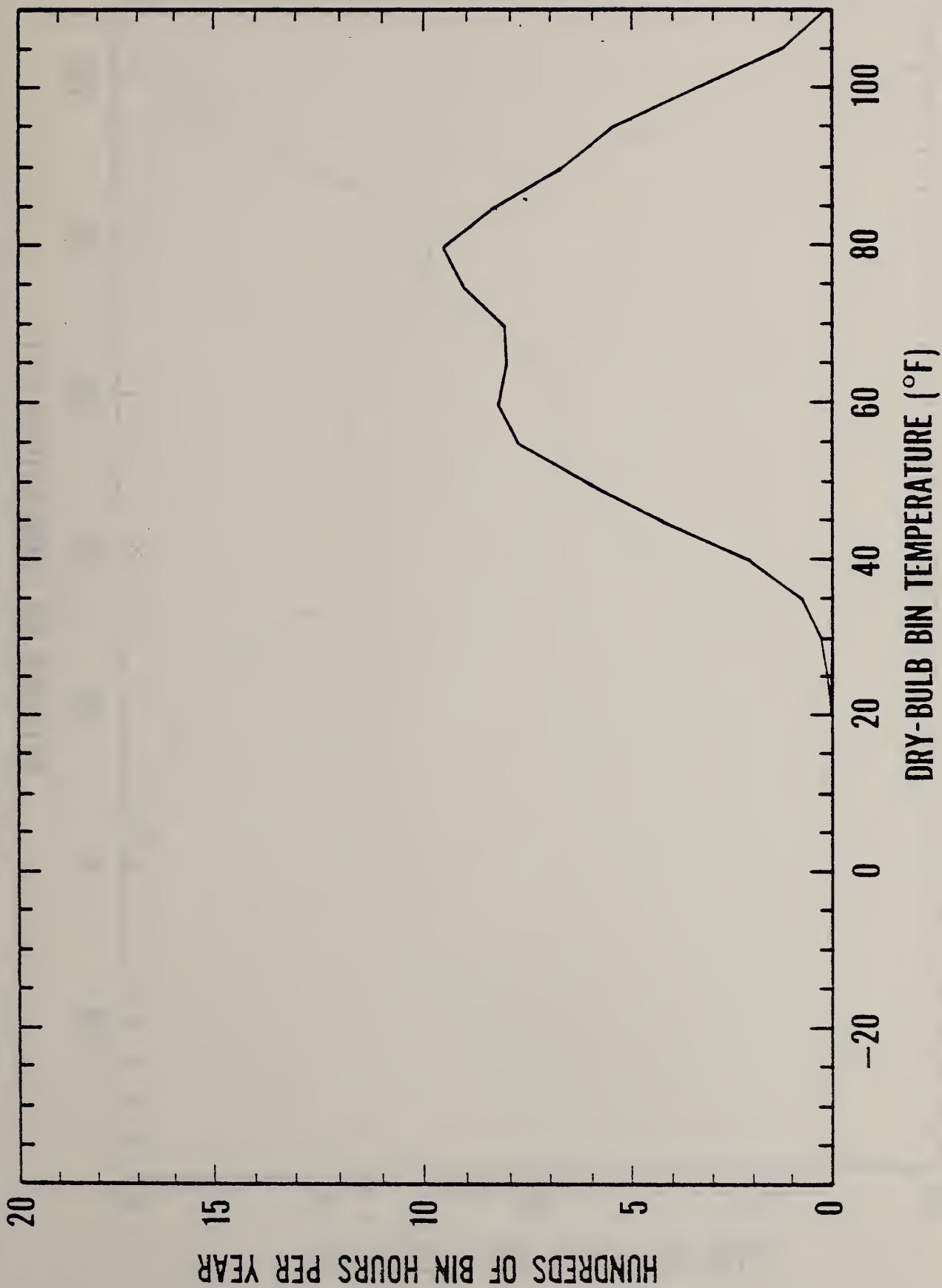
Figure A9: Bin Hours per Year versus Dry-Bulb Bin Temperature for Davis Monthan AFB, AZ
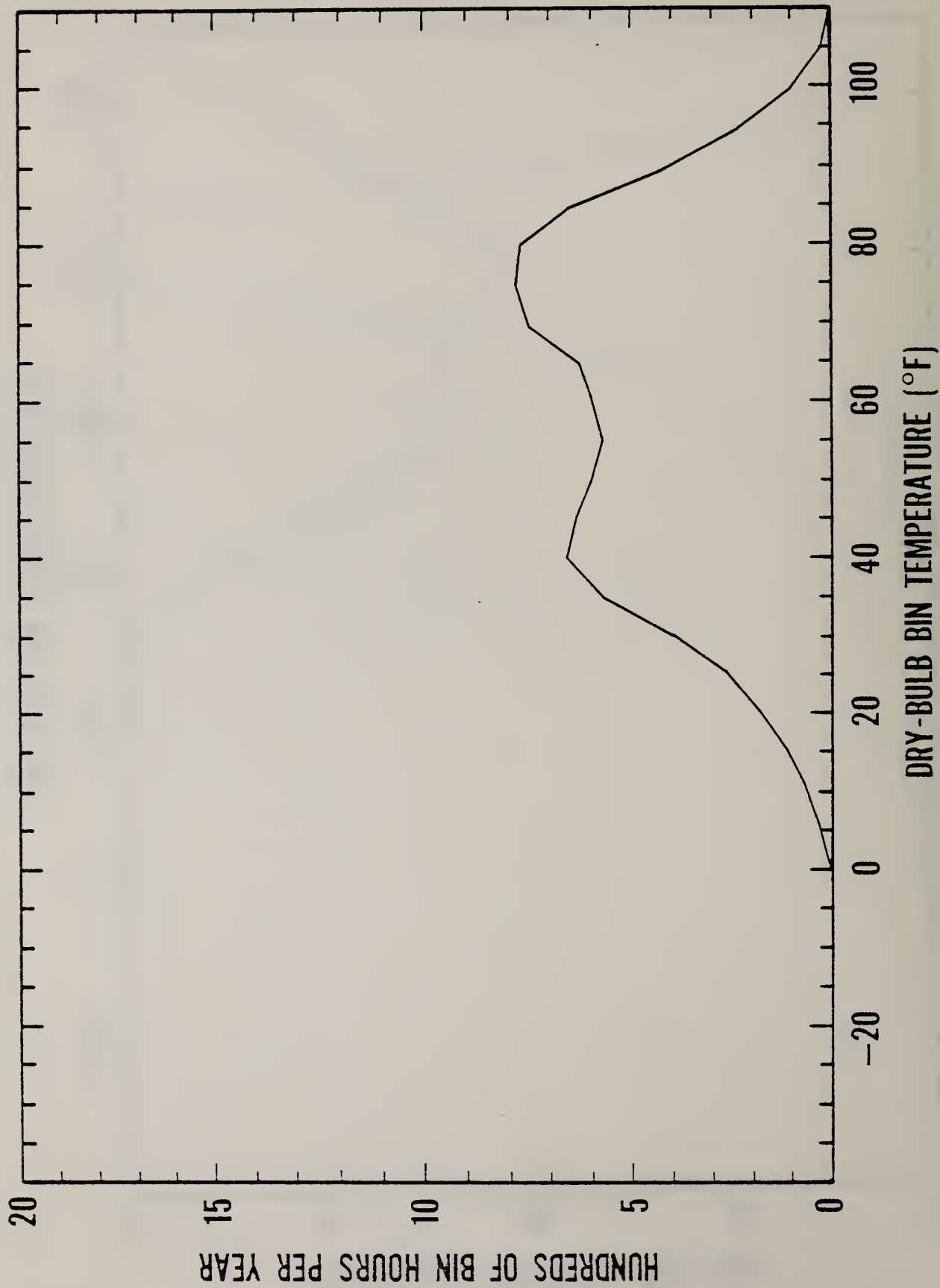
Figure A10: Bin Hours per Year versus Dry-Bulb Bin Temperatures for Kansas City, MO
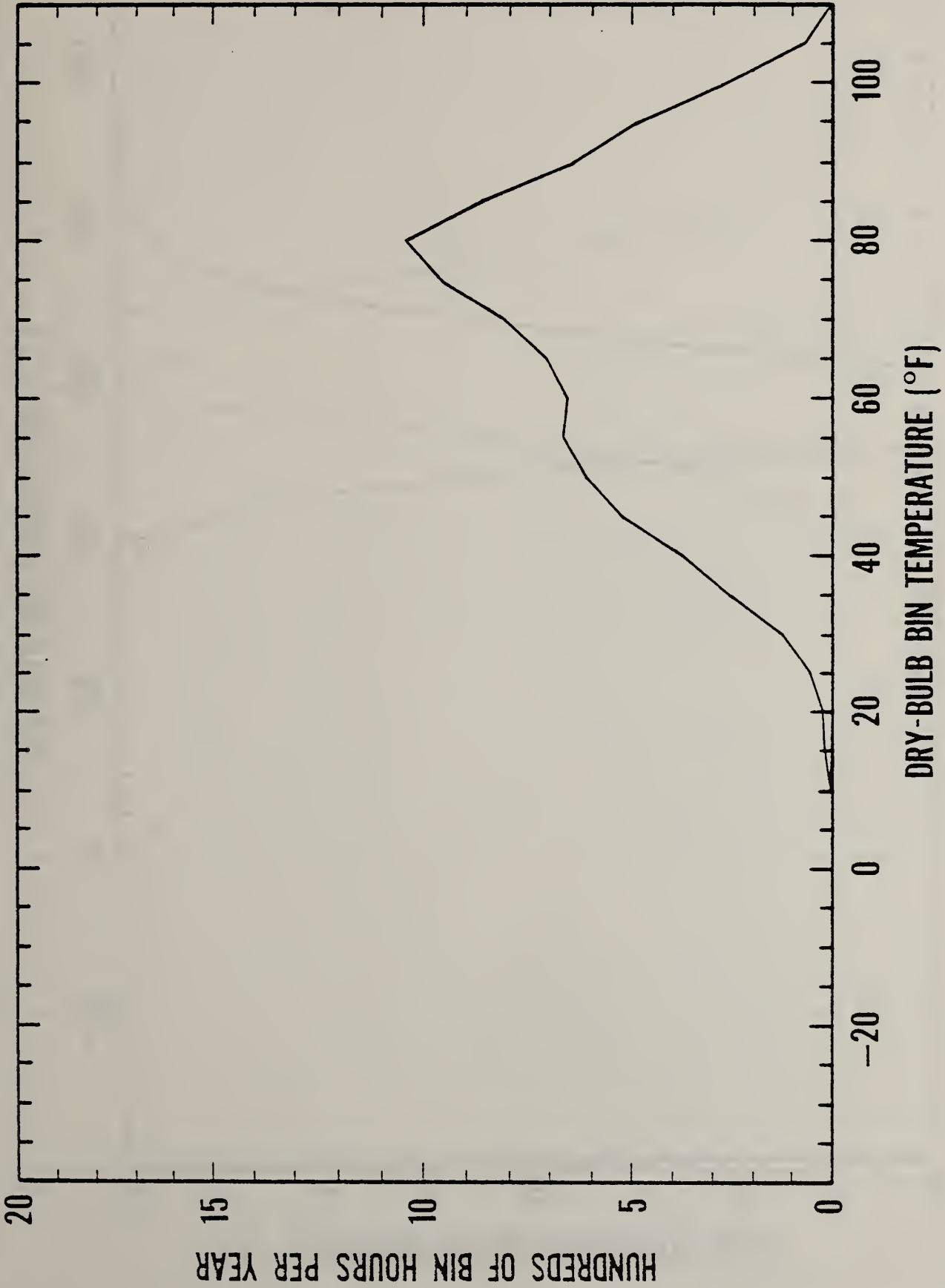
Figure A11: Bin Hours per Year versus Dry-Bulb Bin Temperature for Carswell AFB, TX
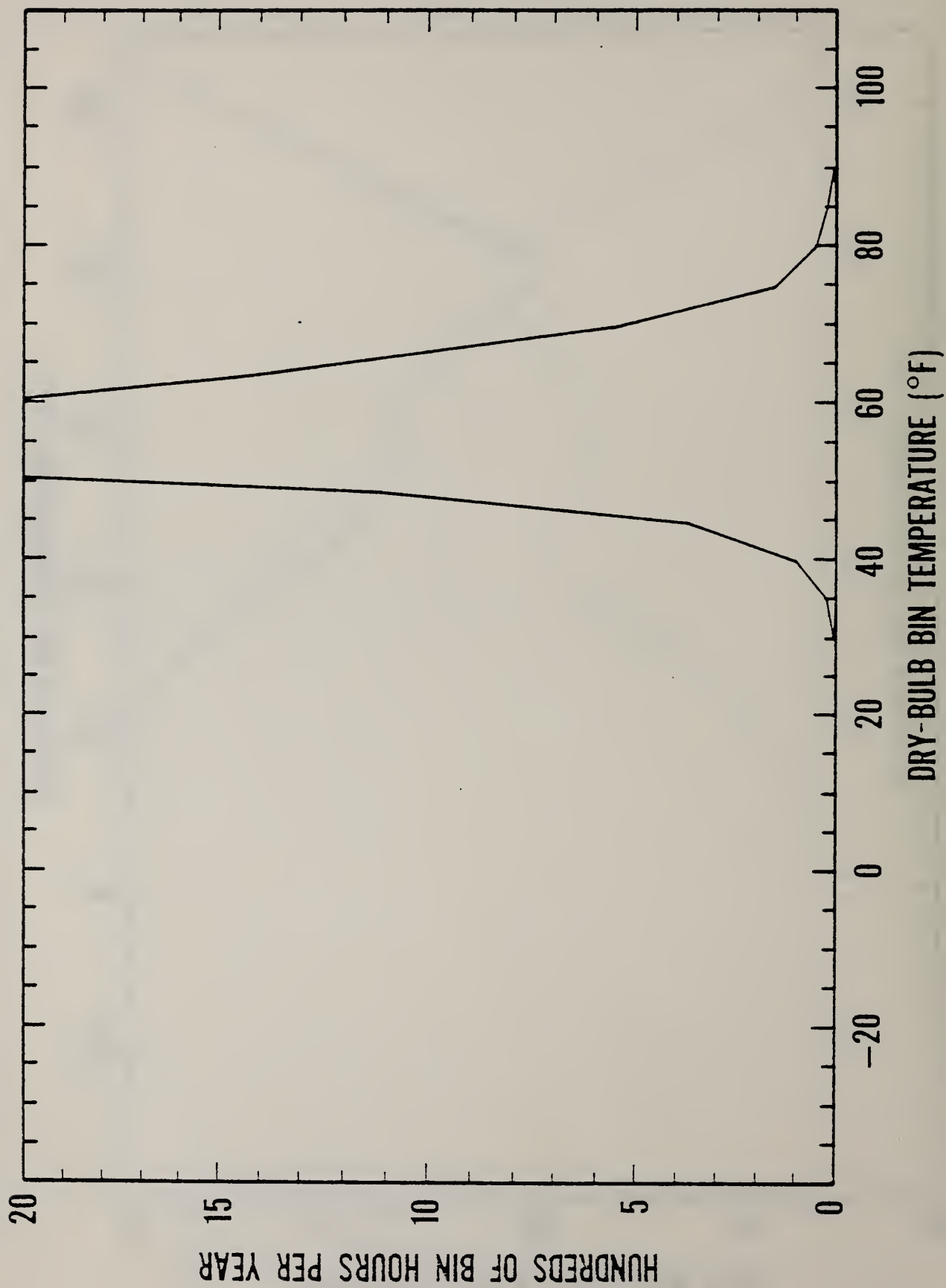
Figure A12: Bin Hours per Year versus Dry-Bulb Bin Temperature for Monterey NAF, CA

Figure A13: Bin Hours per Year versus Dry-Bulb Bin Temperature for Los Angeles, CA

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

Three integrated heat pump-heat recovery unit-water heater appliances were tested under various environmental conditions to measure the functional parameters and study the operating characteristics of these systems. It was found that the heat recovery, heat recovery rate and heat recovery efficiency were dependent on the heat recovery unit's characteristics. The use of the heat recovery unit in the system resulted in a reduced work load for the heat pump's compressor and slightly improved the heat pump's performance.

A computer simulation model of the integrated system was developed to study the interactions between several of the pertinent system variables on an hourly basis for selected situations and to estimate energy savings. Two alternative estimation methods that utilize five-degree temperature bin data were also developed. The estimated savings determined by using the alternative methods were about the same as those estimated using the hourly data. Conclusions were also reached concerning the use of water heaters with different tank capacities and on methods of increasing potential energy savings.