DOE/CS/31720-72

DEVELOPMENT OF MARKETABLE SOLAR

ASSISTED HEAT PUMPS

FEBRUARY 20, 1981

PHASE II SUMMARY REPORT - TECHNICAL RESULTS

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CARROLLTON, TEXAS

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PREPARED FOR

THE DEPARTMENT OF ENERGY

UNDER CONTRACT NO. DE-AC-03-78-CS3-1720

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INTRODUCTION

Lennox Industries Inc. has substantially completed the technical requirements of Phase II of the subject contract and submits this report to summarize the progress and accomplishments made during Phase II. This report does not contain marketability information because the market plan required performance data only recently made available to the marketing personnel working on the project. We will advise when the market plan will be available completing Phase II.

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ABSTRACT

This report summarizes our progress to date under Phase II of our contract for the development of a marketable solar-assisted heat pump. Our goal in this phase of the project was to design a water source heat pump capable of operating over the range from 40°F to 110°F entering water temperature and to have a heating coefficient of performance greater than six in the upper portion of this range. This has been accomplished.

A summary of the key tasks completed during Phase II are listed and discussed below.

- 1) A computerized heat pump balance program was written to allow the performance of either a water-to-water or a water-to-air heat pump to be predicted in either the heating or cooling mode. The basic design for the program is similar to the ORNL heat pump model. Modifications and additions to this program were added to allow us to directly model the systems of interest. A detailed program description, flow charts, and sample outputs are included in the appendices of this report.
- 2) The balance program was used to specify components for a high efficiency water-to-water and a high efficiency water-to-air heat pump. Performance predictions for both units in heating and in cooling are included in this report.

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The water-to-water and water-to-air performance predictions were compared. The water-to-air approach was clearly superior for several reasons:

1. Higher COP in heating.
 2. Significantly higher EER in cooling coupled with better latent heat removal characteristics.
 3. Lower initial cost.
 4. Easier applicability to other water source applications.
- 3) A detailed design and layout was done for the three ton water-to-air heat pump. A horizontal configuration was chosen for marketability reasons. The design was made consistent with high quantity production equipment available in our factories and should result in the lowest possible manufacturing cost. This design will be refined and modified as necessary during the prototype and test programs in Phase III.
- 4) A key question to be answered in this project is whether a reciprocating compressor can operate without damage at the high suction pressures characteristic to the inline SAHP. Reciprocating compressors are normally applied in applications with maximum suction pressures equal to a 55°F saturated suction temperature. An accelerated life test was performed on low speed at 85°F saturated suction temperature and 100°F saturated

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condensing temperature. After 465 hours at these severe conditions no performance degradation or physical damage was noted.

Tests on high speed at 70°F saturated suction temperature are in progress and should define the limits where the compressor may be safely operated.

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HEAT PUMP SIMULATION PROGRAM

In order to properly design a high efficiency water source heat pump and evaluate its performance, a refined computer program was needed to be able to predict the effect of component changes on system performance. Sufficient detail in the modeling of the major refrigeration cycle components was necessary to make the program sensitive to small design changes. Such a program was not available so we set out to develop our own program.

The Oak Ridge National Lab air-to-air heat pump model was found to be an excellent guide in our efforts. We used a similar control scheme to the ORNL model and utilized the refrigerant service routines from this model. The compressor subroutine was developed internally and made use of published compressor pumping rate and power consumption curves to model its performance. The refrigerant-to-air condenser and evaporator routines made use of commonly used capacity curves which were digitized and stored for commonly used coil types. The water-to-refrigerant evaporator and condenser routines were developed using a constant effectiveness method.

The operation of the model starts with the input of the physical parameters of the system to be studied. Initial assumptions incorporated in the model allow an initial evaporator performance to be modeled. This information is passed to the compressor model as an input. Compressor mass flow rate and power consumption are calculated and passed on to the

condenser routine as inputs. Condensing capacity is calculated and the enthalpy of the liquid refrigerant produced by the condenser is compared to the initial enthalpy of the refrigerant which was introduced to the evaporator. An iterative process is continued until convergence of these values meets a specified error. The amount of error is currently set at .25% but may be made larger to reduce the number of iterations at the expense of decreased accuracy or the allowed error may be reduced for greater convergence at the expense of increased execution time.

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SYSTEM DESIGN AND PERFORMANCE PREDICTION

Our primary objective was to design a two speed water source heat pump capable of both heating and cooling which could be used with water entering temperatures between 40°F and 110°F and to have a heating coefficient of performance (COP) of greater than six at elevated water temperatures. The two speed, three ton nominal compressor was used to obtain the capacity modulation necessitated by the wide range of inlet temperatures.

Since the size and type of the compressor was already determined, the computer simulation was utilized in selecting the proper size and design of the heat exchanger surfaces. In the case of the water-to-refrigerant (coaxial) heat exchanger, the choice was based on desired water flow rate, pressure drop characteristics, cost, and heat transfer performance as an evaporator and a condenser. The indoor heat exchanger was selected in the same manner, but the additional parameters involved were latent heat removal in the cooling mode and compactness of size.

Figures 1 through 4 describe the performance characteristics of both the water-to-air and water-to-water heat pump in the heating and cooling mode. As can be seen, the figures show the water-to-air heat pump model as the base design. Applying the same size and design heat exchangers for the water-to-water heat pump model resulted in a significant reduction

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in performance, especially in the cooling mode. This is primarily due to the introduction of an additional heat exchanger into the system for the water-to-water heat pump model. Also, the water-to-refrigerant heat exchanger does not perform as well in the evaporator mode as it does in the condenser mode. This occurs due to higher pressure drops across the exchanger as an evaporator. The other set of performance curves incorporated larger heat exchangers for the water-to-water heat pump model in order to obtain the performance characteristics of the water-to-air heat pump model in the heating mode. The figures show that even though the performance characteristics were matched in the heating mode, there is still a performance degradation in the cooling mode.

Lennox has chosen to design the solar-assisted heat pump for the water-to-air model. This decision was made due to more favorable cost and performance characteristics. A listing of the water-to-air heat pump simulation data is included in the appendix of this report.

HEATING CAPACITY

FIGURE 1

HEATING PERFORMANCE

FIGURE 2

FIGURE 3

COOLING PERFORMANCE

FIGURE 4

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DETAILED DESIGN

Input from marketing indicated that a horizontal configuration of the indoor water source heat pump is the better approach. This type of unit could be located in an attic, a crawl space, or between floors of a multi-story building.

Our design approach was to hold the vertical height of the unit as small as possible, to hold the width less than 36 inches, and to allow the length to assume that necessary to fit all the components into a compact but accessible unit. A blow-through configuration was used to allow the unit to be mated with a straight section of ductwork.

Dimensioned drawings with labeled components are illustrated in Figure 5 through 7.

LENNOX SOLAR-ASSISTED HEAT PUMP

TOP VIEW

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FIGURE 5

LENNOX SOLAR-ASSISTED HEAT PUMP
SIDE VIEW

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FIGURE 6

LENNOX SOLAR-ASSISTED HEAT PUMP COMPONENTS & PIPING

FIGURE 7

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COMPRESSOR TESTING

Testing was performed with an L7-03322A three-ton, two-speed compressor with two goals in mind. One was to determine compressor performance at elevated evaporating temperatures so that an accurate compressor performance map would be available to the computer balancing routine. This would allow heat pump performance to be accurately predicted for evaporating temperatures well in excess of that normally encountered in an air-to-air heat pump. The performance maps are shown in Figure 8.

Another testing goal was to investigate the reliability of the compressor at the high evaporating temperatures sure to be encountered in an inline SAHP. A particular concern was the durability of the compressor suction valves since the mass flow of refrigerant introduced into the compressor at elevated evaporating temperatures, and hence at high suction gas densities, would be much greater than normally encountered. The greater mass flow would exert a greater force and cause a greater deflection in the reed valve reducing its fatigue life.

To check this possibility, the compressor was operated first on low speed at 85°F evaporating temperature and 100°F condensing temperature. These particular temperatures were taken to represent practical maximum operating conditions that the compressor might see in application. Performance data were recorded initially and were compared at several intervals

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over the 465 hours that the compressor was run. No performance degradation occurred over the duration of the test. At the end of the test the compressor was disassembled and checked for damage or wear. No problems were noted.

The compressor was then reassembled and a high speed operating test was initiated at 70°F evaporating temperature and 100°F condensing temperature. The lower evaporating temperature was justified for two reasons. At the high entering water temperatures the inline SAHP might see (up to 110°F), the capacity of the heat pump increases dramatically even on low speed. At temperatures above 90°F entering water temperature, the low speed should have ample capacity to meet the building load.

A second reason was that we were concerned about the extremely high mass flow of refrigerant on high speed at 85°F evaporating temperatures. The 70°F condition should give a reasonable test and when this test is completed, a higher temperature test will be performed.

At the present time 65 hours have been logged on the high speed life tests with no performance degradation. This test will be continued until 400 operating hours are reached or an indication is seen of performance degradation or damage to the compressor.

COOLING PERFORMANCE GRAPH
L7A-03322A-B1

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20°F Superheat
15°F Subcooling
R-22

FIGURE 8

CONCLUSIONS AND RECOMMENDATIONS

The major conclusions from the work done in Phase II are:

- 1) It is possible to design a high efficiency heat pump to meet the performance goals of the project using existing components.
- 2) The compressor life tests show that the compressor has the necessary durability to function under the extreme operating conditions specified in the low speed mode.
- 3) The overall package design is one which can be mass produced.

Recommendations for future work are as follows:

- 1) A prototype of the final design should be built and tested to verify that our performance goals can be met and to validate and adjust our computer modeling program.
- 2) This sample should be sent to Brookhaven National Lab for independent testing to confirm our work.

APPENDIX A

HEAT PUMP SIMULATION PROGRAM DESCRIPTION

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COMPUTER SIMULATION

Lennox has developed a program to predict water source heat pump performance for given system characteristics and input conditions.

Performance of the entire system depends upon the performance characteristics of the system components. The main program loops between the component subprograms until there is an energy and pressure balance over the entire cycle. The main program, HPSIM, and the evaporator, compressor, and condenser subprograms are detailed in this section of the report.

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PROGRAM INPUTS

For each run of the program, there are several inputs to be made in order to properly simulate the desired condition. A description of each input is listed below:

- Heat Pump Type

- (1) Water-to-Air -- This type of heat pump has a water-to-refrigerant heat exchanger on the water source side and a refrigerant-to-air heat exchanger on the indoor side.
- (2) Water-to-Water -- This type of heat pump has a water-to-refrigerant heat exchanger on the water source side and a refrigerant-to-water heat exchanger coupled with a water-to-air heat exchanger on the indoor side.

- Heating or Cooling Mode

This inputs to the program the mode of heat pump operation that will be simulated.

- Compressor Speed

This inputs to the program the compressor speed to simulate (low or high) and to retrieve the associated compressor data; i.e., mass flow and power consumed.

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- Indoor Air Temperature

This inputs to the program the temperature (dry bulb and wet bulb) of the air entering the heat pump.

- Air Flow

This inputs to the program the flow of air (cubic feet per minute) through the heat pump.

- Entering Water Temperature

This inputs to the program the temperature of the water entering the water-to-refrigerant heat exchanger of the water source side of the heat pump.

- Entering Water Flow Rate

This inputs to the program the flow of water (gallons per minute) through the water-to-refrigerant heat exchanger on the water source side of the heat pump.

- Water-to-Refrigerant Heat Exchanger Effectiveness

This inputs to the program the effectiveness of the water-to-refrigerant heat exchanger on the water source side of the heat pump.

- Refrigerant Data

These inputs are used in fine tuning the program and are dependent on heat exchanger characteristics and system charge. The actual values to be input are determined from the judgment and past experience of the operator.

(1) Degrees Subcooling:

This inputs to the program the amount of subcooling that the refrigerant has obtained below the saturated liquid temperature as it leaves the condenser of the heat pump.

(2) Degrees Superheat:

This inputs to the program the temperature rise that the refrigerant obtains above the saturated vapor temperature as it leaves the evaporator of the heat pump.

(3) Evaporator side total pressure drop:

This inputs to the program the amount of pressure drop (pounds per square inch) that is assumed across the evaporator of the heat pump.

(4) Condenser side total pressure drop:

This inputs to the program the amount of pressure drop (pounds per square inch) that is assumed across the condenser of the heat pump.

• For Water-to-Air Heat Pump

This inputs to the program the physical configuration of the refrigerant-to-air (DX) coil for a water-to-air type heat pump.

The coil physical configuration is incorporated within the program and to change any of these parameters would require editing of the program.

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- For Water-to-Water Heat Pump

- (1) Water-to-air coil physical configuration:

This inputs to the program the physical configuration of the water-to-air (hydronic) coil for a water-to-water type heat pump.

- (2) Hydronic flow rate:

This inputs to the program the flow of water (gallons per minute) through the hydronic indoor side of the water-to-water type heat pump which inputs the flow rate through the refrigerant-to-water and the water-to-air heat exchangers.

- (3) Refrigerant-to-water heat exchanger effectiveness:

This inputs to the program the effectiveness of the refrigerant-to-water heat exchanger on the hydronic indoor side of the water-to-water type heat pump.

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EVAPORATOR SUBPROGRAM

The evaporator subprogram has three possible data flow paths for a given set of inputs. Each path represents a mode or type of heat pump operation which is to be simulated. The evaporator subprogram flow chart shown on the following page, Figure 9., best describes the flow of data from inputs to outputs.

The decision blocks that determine the path of data use the inputs given. If the heat pump simulation is in the heating mode, then the evaporator (water-to-refrigerant heat exchanger) will be operating on the water source side for either water-to-water or water-to-air type of heat pump. If in the cooling mode, then another input is used to determine whether the heat pump is a water-to-water type or a water-to-air type. For a water-to-water type, the evaporator is a refrigerant-to-water heat exchanger coupled to a water-to-air heat exchanger on the indoor hydronic side. For a water-to-air type, the evaporator is a refrigerant-to-air heat exchanger (DX coil) on the indoor side.

As the flowchart indicates, there are four subroutines called, depending on the path of data. Each subroutine will be detailed further in this report.

COMPUTER SIMULATION

EVAPORATOR SUBPROGRAM FLOW CHART

INPUTS:

WATER-TO-REFRIGERANT
 EVAPORATING TEMP.
 INLET WATER TEMP.
 WATER FLOW RATE
 W-R EXCH. EFFECTIVENESS

WATER-TO-WATER
 EVAPORATING TEMP.
 W-A COIL CONFIGURATION
 INLET AIR TEMP. (W.B.)
 AIR FLOW RATE
 WATER FLOW RATE
 R-W EXCH. EFFECTIVENESS

WATER-TO-AIR
 EVAPORATING TEMP.
 R-A COIL CONFIGURATION
 INLET AIR TEMP. (W.B.)
 AIR FLOW RATE

OUTPUTS:

WATER-TO-REFRIGERANT
 TOTAL CAPACITY (BTUH)
 EXIT WATER TEMP.

WATER-TO-WATER
 TOTAL CAPACITY (BTUH)
 INLET WATER TEMP.
 EXIT WATER TEMP.

WATER-TO-AIR
 TOTAL CAPACITY (BTUH)

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FIGURE 9

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SUBROUTINE WATE

Water-to-Refrigerant Heat Pump Evaporator

The coaxial (tube in tube) water-to-refrigerant heat exchanger subroutine has been developed using the effectiveness-NTU method where heat exchanger effectiveness is defined as:

$$\text{Effectiveness} = \frac{Q_{\text{Act}}}{Q_{\text{Max}}} = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}}$$

The maximum possible heat transfer for the heat exchanger is the maximum value that could be obtained if one of the fluids were to undergo a temperature change equal to the maximum temperature difference present in the exchanger, which is the difference in the entering temperatures for the hot and cold fluids. The fluid which undergoes the maximum temperature change is the one that has the minimum heat capacity rate. The heat capacity rate is the product of the fluid mass flow rate ($\dot{m}$) and the fluid's specific heat (C_p) so the maximum possible heat transfer is expressed as:

$$Q_{\text{max}} = (\dot{M} C_p)_{\text{min}} (T_{\text{Hinlet}} - T_{\text{Cinlet}}).$$

During evaporation, the refrigerant is accepting heat without increasing its temperature. Therefore, the refrigerant specific heat value goes to infinity during evaporation. With this in mind, it is assumed that the water loop will always be the minimum fluid, and the minimum temperature ($T_{\text{c inlet}}$) of the heat exchanger is the evaporating temperature or

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$$Q \text{ max} = (\dot{m} \text{ Cp}) \text{ water} (T \text{ water in} - T \text{ evap})$$

Heat exchanger effectiveness can be determined for selected heat exchangers provided that the vendor can supply performance data for a requested set of running conditions. Such data has been provided to Lennox from a coaxial heat exchanger manufacturer and a sample was later acquired and tested confirming performance data and analysis. On the following page, Figure 10 illustrates the vendor data used to determine heat exchanger effectiveness for a selected coaxial coil.

As shown in Figure 9, inputs required for each run of subroutine WATE are evaporating temperature, inlet water temperature, water flow rate, and water-to-refrigerant heat exchanger effectiveness. Other program inputs such as degrees superheat leaving the evaporator and evaporator side total pressure drop affect the outputs of subroutine WATE by increasing or decreasing the refrigerant evaporating temperature in order for the program to reach an energy balance over the entire cycle. With these inputs, subroutine WATE will output the total heat capacity and the exit water temperature.

WATER-TO-REFRIGERANT EVAPORATOR PERFORMANCE DATA

WATER FLOW RATE, GPM	ENTERING WATER TEMP., °F	LEAVING WATER TEMP., °F	EVAPORATOR CAPACITY/BTUH	SATURATED SUCTION, °F	SUPERHEAT °F
6.0	60.0	47.8	36,600	34.3	5.0
8.0	60.0	50.8	36,800	36.5	4.7
10.0	60.1	52.6	37,500	38.3	3.8
6.0	70.1	56.2	41,700	42.1	4.5
8.0	70.1	59.2	43,600	44.7	5.6
10.0	70.0	61.0	45,000	46.3	4.9
6.0	80.1	63.4	50,100	48.8	6.1
8.0	80.0	67.1	51,600	51.8	5.3
10.0	80.0	69.4	53,000	54.1	4.7

NOTE: THE TEST DATA ARE FOR NOMINAL 3-TON COMPRESSOR USING R-22. THE REFRIGERANT SIDE PRESSURE DROP ACROSS THE HEAT EXCHANGER RANGED FROM 10 TO 12 PSID.

FIGURE 10

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SUBROUTINE REFWA

Water-to-Water Heat Pump Evaporator

This path of data (See Figure 9) calls two subroutines which are coupled in a looping process or energy balance. Subroutine WATE, refrigerant-to-water heat exchanger, simulates heat exchange between the refrigerant and a closed loop water circuit using the effectiveness - NTU method described previously in this section. The other subroutine WACOI is a water-to-air heat exchanger routine that simulates heat exchange between the closed loop chilled water circuit and the indoor air. The looping process continues until the heat exchange capacities of the two subroutines match to within a specified tolerance. When the tolerance is satisfied, the looping stops and the capacity is transferred to the output.

The water-to-air heat exchanger subroutine WACOI is developed from hydronic coil capacity curves derived from "Bohn's Air Conditioning Coil Engineering Manual"¹. The capacity curves and associated correction factors are specified for seamless copper tubes with plate type aluminum fins with self-spacing die formed collars completely covering the tubes. All other parameters such as size, configuration, and inlet conditions can

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be manually changed. Though Lennox manufactures its own finned tube coils, the performance curves of the Bohn manufactured coils have correlated closely to actual test data and are used industry wide in design selection of coils in air conditioning systems.

Both the capacity and correction factor curves shown in Figures 11 and 12 are reproduced in the form of first, second, and third degree equations depending on the best curve fit. The subroutine inputs are as follows:

- (1) Distance between the coil and plates:
90 inches maximum
- (2) Coil tube diameter:
3/8 inch or 1/2 inch nominal
- (3) Coil fin pattern:
For 1/2 inch tubing - 1.25 by 1.08 inch
For 3/8 inch tubing - 1.00 by .866 inch
or 1.25 by 1.08 inch
- (4) Coil fin configuration:
Corrugated or flat fin
- (5) Number of tube passes per row:
30 tubes maximum per coil slab
- (6) Number of tube rows:
2 to 4 row
- (7) Number of water flow circuits:

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(8) Fins per inch:

10 to 15 fins per inch

(9) Water flow rate - gallons per minute:

Limited so that water velocity is from 1 to 8 feet per second.

(10) Air flow rate - cubic feet per minute:

Limited so that the air velocity is from 200 to 600 feet per minute.

(11) Inlet water temperature:

32 to 65 degrees Fahrenheit

(12) Inlet air temperature (wet bulb):

50 to 100 degrees Fahrenheit

(13) Refrigerant evaporating temperature:

Again, Figure 9 lists the inputs required for each run of the water-to-water heat pump evaporator subroutine REFWA. The looping process begins between the refrigerant-to-water heat exchanger subroutine WATE and the water-to-air heat exchanger subroutine WACOI. Once the heat exchange capacities match to a specified tolerance, the looping stops and subroutine REFWA outputs total cooling capacity and the temperatures of the water entering and leaving the water-to-air heat exchanger.

WATER COOLING CAPACITY—WET SURFACE

1/2" O.D. TUBE COILS

1.25 x 1.08 STAGG.

2/20/80

FIGURE 11

WATER COOLING CORRECTION FACTORS

F_v

$\frac{CFM}{GPM}$

F_r

2 ROW	951	982	1000	1011	1016	1028
3 ROW	983	993	1000	1003	1006	1011
4 ROW	1000	1000	1000	1000	1000	1000
5 ROW	1026	1007	1000	998	994	988
6 ROW	1044	1012	1000	994	990	980
	1	2	3	4	5	8

F_v

WATER VELOCITY—FPS

F_t

2 ROW	.97	1.00	1.00	1.01	1.03	1.05	1.06	1.09	1.13
3 ROW	.99	1.00	1.00	1.01	1.01	1.02	1.03	1.05	1.08
4 ROW	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
5 ROW	1.01	1.01	1.00	1.00	.99	.98	.97	.96	.95
6 ROW	1.01	1.01	1.00	1.00	.99	.97	.95	.94	.93
	50	100	133	150	200	250	300	350	400

F_r

$\frac{CFM}{GPM}$ RATIO

FIGURE 12

SUBROUTINE ECOIL

Water-to-Air Heat Pump Evaporator

The water-to-air heat pump evaporator calls on a refrigerant-to-air heat exchanger subroutine ECOIL shown in Figure 9, which simulates a finned tube evaporator (DX coil) on the indoor side. For the given inputs, this subroutine will output the total cooling capacity as seen by the inside air.

Subroutine ECOIL is developed from evaporator finned tube coil capacity curves derived from "Bohn's Air Conditioning Coil Engineering Manual"¹. Again, the capacity curves and temperature correction factor curves are specified for aluminum finned copper tubes and are shown in Figures 5 and 6. All other inputs such as size, configuration, and inlet conditions are as follows:

- (1) Distance between the coil end plates:

90 inches maximum

- (2) Coil tube diameter:

3/8 inch or 1/2 inch nominal

- (3) Coil fin pattern:

For 1/2 inch tubing - 1.25 by 1.08 inch

For 3/8 inch tubing - 1.00 by .866 inch

or 1.25 by 1.08 inch

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(4) Coil fin configuration:

Corrugated or flat fin

(5) Number of tube passes per row:

30 tubes maximum per coil slab

(6) Number of tube rows:

2 to 4 row

(7) Fins per inch:

10 to 15 fins per inch

(8) Air flow rate - cubic feet per minute:

Limited so that the air velocity is from 200 to
600 feet per minute.

(9) Inlet air temperature (wet bulb):

50 to 100 degrees Fahrenheit

(10) Refrigerant evaporating temperature:

Of the inputs shown above and in Figure 9 for the water-to-air heat pump evaporator, only the last three, 8, 9, and 10, can change from one condition to another. The other inputs, 1 to 7, which describe the coil configuration are incorporated into the program and any changes would require editing the program. This is done so fewer inputs are needed per run and to avoid repetition.

EVAPORATOR CAPACITY—WET SURFACE

1/2" O.D. TUBE COILS

1-1/4 x 1.08 STAGG.

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FIGURE 13

EVAPORATOR CORRECTION FACTORS ALL FIN TYPES

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FIGURE 14

CONDENSER SUBPROGRAM

The condenser subprogram, like the evaporator subprogram, has three separate paths through which data flows for a given set of inputs. The flow chart shown on the following page, Figure 15, indicates the flow of data from inputs to outputs. Each path represents a mode or type of heat pump operation which is to be simulated.

The control statements that determine the path of data use the inputs given. If the heat pump simulation is in the cooling mode, then the condenser (a water-to-refrigerant heat exchanger) will be operating on the water source side for either water-to-water or water-to-air type of heat pump. If in the heating mode, then another input is used to determine whether the heat pump is a water-to-water or water-to-air type. For a water-to-water type, the condenser is a refrigerant-to-water heat exchanger coupled to a water-to-air heat exchanger on the indoor hydronic side. For a water-to-air type, the condenser is a refrigerant-to-air exchanger on the indoor side.

As the flowchart indicates, there are four subroutines used depending on the path of data. Each subroutine will be detailed further in this report.

COMPUTER SIMULATION

CONDENSER SUBPROGRAM FLOW CHART

INPUTS:

WATER-TO-REFRIGERANT
CONDENSING TEMP.
INLET WATER TEMP.
WATER FLOW RATE
W-R EXCH.EFFECTIVENESS

WATER-TO-WATER
CONDENSING TEMP.
W-A COIL CONFIGURATION
INLET AIR TEMP. (D.B.)
AIR FLOW RATE
WATER FLOW RATE
R-W EXCH. EFFECTIVENESS

WATER-TO-AIR
CONDENSING TEMP.
R-A COIL CONFIGURATION
INLET AIR TEMP. (D.B.)
AIR FLOW RATE

OUTPUTS:

WATER-TO-REFRIGERANT
TOTAL CAPACITY (BTUH)
EXIT WATER TEMP.

WATER-TO-WATER
TOTAL CAPACITY (BTUH)
INLET WATER TEMP.
EXIT WATER TEMP.

WATER-TO-AIR
TOTAL CAPACITY (BTUH)

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FIGURE 15

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SUBROUTINE WATC

Water-to-Refrigerant Heat Pump Condenser

The coaxial (tube-in-tube) water-to-refrigerant heat exchanger subroutine has been developed again using the effectiveness - NTU method of analysis. The approach is the same as that used in the water-to-refrigerant heat pump evaporator subroutine WATE described previously in this report. During condensation, the refrigerant is rejecting heat without decreasing its temperature. Therefore, the water loop will always be the minimum fluid and the maximum temperature (TH inlet) of the heat exchanger is the condensing temperature of the refrigerant or

$$Q_{\max} = (M C_p)_{\text{water}} (T_{\text{cond}} - T_{\text{water In}}).$$

Again Lennox has been provided with data from a coaxial heat exchanger manufacturer. Figure 16 illustrates the vendor data used to determine heat exchanger effectiveness for a selected coaxial coil.

The inputs required for each run of the water-to-refrigerant heat exchanger subroutine, shown in Figure 15, are condensing temperature, inlet water temperatures, water flow rate, and water-to-refrigerant heat exchanger effectiveness. Other inputs such as degrees subcooling leaving

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the condenser and condenser side total pressure drop affect the outputs of subroutine WATC by increasing or decreasing the refrigerant condensing temperature in order for the program to reach an energy balance over the entire cycle. With these inputs, the subroutine will output the total heat capacity and the exit water temperature.

WATER-TO-REFRIGERANT CONDENSER PERFORMANCE DATA

WATER FLOW RATE, GPM	ENTERING WATER TEMP., °F	LEAVING WATER TEMP., °F	CONDENSER CAPACITY/BTUH	SUBCOOLING °F	REF. SIDE ΔP, PSID
6.0	85.0	97.6	37,800	5.1	3.0
8.0	84.9	95.7	43,200	5.1	3.0
9.0	85.1	95.3	45,900	4.9	3.0
9.5	85.1	94.9	46,550	5.0	3.5
6.0	85.0	97.2	36,600	10.1	2.5
8.0	85.0	95.4	41,600	10.0	3.0
9.0	85.0	94.8	44,100	10.1	3.5
10.0	85.0	94.2	46,000	10.0	3.5

NOTE: THE TEST DATA ARE FOR NOMINAL 3-TON COMPRESSOR USING R-22. THE SATURATED DISCHARGE TEMPERATURE FOR ALL RUNS WAS 105.1°F.

FIGURE 16

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SUBROUTINE REFWA

Water-to-Water Heat Pump Condenser

This part of the program shown in Figure 15, calls two subroutines which are coupled in a looping process or energy balance. The refrigerant-to-water heat exchanger subroutine WATC simulates heat exchange between the condensing refrigerant and a closed loop water circuit using the effectiveness - NTU method described previously. The other subroutine WACOI is a water-to-air heat exchanger subroutine that simulates heat exchange between the closed loop hot water circuit and the indoor air. Again, the looping process continues until the heat exchange capacities of the two subroutines match to within a specified tolerance. When the tolerance is satisfied, the looping stops and the capacity is transferred to the output.

The water-to-air heat exchanger subroutine WACOI is used for both chilled and hot water hydronic coils. There are differences associated with the calculations of hydronic heating and cooling capacities. The methodology is similar in that capacity and correction factor curves are used, but the values are different as shown in Figures 17 and 18. The subroutine inputs are as follows:

- (1) Distance between the coil end plates:

90 inches maximum

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(2) Coil tube diameter:

3/8 inch or 1/2 inch nominal

(3) Coil fin pattern:

For 1/2 inch tubing - 1.25 by 1.08 inch

For 3/8 inch tubing - 1.00 by .866 inch

or 1.25 by 1.08 inch

(4) Coil fin configuration:

Corrugated or flat fin

(5) Number of tube passes per row:

30 tubes maximum per coil slab

(6) Number of tube rows:

2 to 4 row

(7) Number of water flow circuits:

(8) Fins per inch:

10 to 15 fins per inch

(9) Water flow rate - gallons per minute:

Limited so that water velocity is from 1 to 8 feet
per second.

(10) Air flow rate-cubic feet per minute:

Limited so that the air velocity is from 200 to
600 feet per minute.

(11) Inlet water temperature:

90 to 250 degrees Fahrenheit.

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(12) Inlet air temperature (dry bulb):

40 to 100 degrees Fahrenheit.

(13) Refrigerant condensing temperature:

The outputs as listed in Figure 15 of the water-to-water heat pump condenser are total heating capacity and the temperatures of the water entering and leaving the water-to-air heat exchanger.

WATER HEATING CAPACITY

1/2" O.D. TUBE COILS

1.25 x 1.08 STAGG.

FIGURE 17

WATER HEATING CORRECTION FACTORS

1 ROW	0.99	0.99	1.00	1.00	1.00	1.00
2 ROW	1.00	1.00	1.00	1.00	1.00	1.00
3 ROW	1.01	1.00	1.00	1.00	1.00	1.00
4 ROW	1.02	1.01	1.00	0.99	0.99	0.99
	1	2	3	4	5	8

WATER VELOCITY-FPS

F_v

FIGURE 18

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SUBROUTINE CON

Water-to-Air Heat Pump Condenser

The water-to-air heat pump condenser calls on a refrigerant-to-air heat exchanger subroutine CON, shown in Figure 15, which simulates a finned tube condenser on the indoor side. For the given inputs, this subroutine will output the heating capacity as seen by the inside air.

Subroutine CON is developed from condenser finned tube coil capacity curves, shown in Figure 19, that are derived from "Bohn's Air Conditioning Coil Engineering Manual"¹. All other inputs and limitations are as follows:

- (1) Distance between the coil end plates:
90 inches maximum.
- (2) Coil tube diameter:
3/8 inch or 1/2 inch nominal.
- (3) Coil fin pattern:
For 1/2 inch tubing - 1.25 by 1.08 inch.
For 3/8 inch tubing - 1.00 by .866 inch
or 1.25 by 1.08 inch
- (4) Coil fin configuration:
Corrugated or flat fin
- (5) Number of tube passes per row:
30 tubes maximum per coil slab.

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(6) Number of tube rows:

2 to 4 row

(7) Fins per inch:

10 to 15 fins per inch

(8) Air flow rate - cubic feet per minute:

Limited so that the air velocity is from 200 to
600 feet per minute.

(9) Inlet air temperature (dry bulb):

(10) Refrigerant condensing temperature:

Like the water-to-air heat pump evaporator subroutine ECOIL, only the three inputs, 8, 9, and 10, can change from one condition to another. The other inputs, 1 to 7, which describe the coil configuration are incorporated into the program and any changes require editing the program.

CONDENSER CAPACITY—DRY SURFACE

1/2" O.D. Tube Coils

1-1/4 x 1.08 STAGG.

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FIGURE 19

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COMPRESSOR SUBPROGRAM

The compressor subprogram is a performance model of a Lennox nominal three-ton, two-speed compressor. The family of curves shown previously in Figure 8 reflect the mass flow rate and power consumption rate, respectively, of the compressor for a given set of conditions.

The subprogram COMPR consists of an array of numbers which represent refrigerant mass flow rates and compressor power consumptions. Through interpolation routines, these outputs are calculated given compressor speed, condensing temperature, evaporating temperature, and degrees superheat in the suction line. The compressor subprogram flow chart with inputs and outputs is shown in Figure 20.

COMPUTER SIMULATION
COMPRESSOR SUBPROGRAM FLOW CHART

INPUTS: COMPRESSOR SPEED
CONDENSING TEMPERATURE
EVAPORATING TEMPERATURE
DEGREES SUPERHEAT (SUCTION)

OUTPUTS: REFRIGERANT MASS FLOW RATE
COMPRESSOR POWER INPUT

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FIGURE 20

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PROGRAM HPSIM

Water Source Heat Pump Simulation

The water source heat pump performance and capacity properties are determined through a balancing process between the evaporator, compressor, and condenser subprograms described previously. Figure 21 illustrates the flow chart of the main program HPSIM which incorporates each subprogram in the looping process. Refrigerant thermodynamic state properties are also calculated during this process using saturated and superheated service routines for Refrigerant 22.

After the inputs are read, the program begins the simulation with some initial assumptions. These assumptions do not affect the outputs of the program, but allow a reasonable starting point from which the convergence can begin in order to reduce execution time. The assumptions are listed below:

For heating mode:

- (1) Condensing Temp. = Inlet air temp. (DB) + 10°
- (2) Evaporating Temp. = Inlet water temp. - 20°

For cooling mode:

- (1) Condensing Temp. = Inlet water temp. + 10°
- (2) Evaporating Temp. = Inlet air temp. (WB) - 20° .

COMPUTER SIMULATION

HEAT PUMP FLOW CHART

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FIGURE 21

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At this point, the refrigerant thermodynamic state properties are calculated over the specific cycle using the inputs given and the initial assumptions. An illustration of the refrigerant cycle can be seen in Figure 22. The figure shows the effects of the condenser side pressure drop, degrees subcooling at state point 3, evaporator side pressure drop, and degrees superheat at state point 1. The refrigerant thermodynamic state properties are determined using service routines adapted from "A Computer Simulation of Steady-State Performance of Air-to-Air Heat Pumps" developed for the Department of Energy at Oak Ridge National Laboratory².

Now that the initial evaporating and condensing temperatures are assumed, the compressor subprogram COMPR is called and refrigerant mass flow ($\dot{M}_1$) and compressor power input ($\dot{W}_1$) are determined. The power input ($\dot{W}_1$) to the compressor is assumed to be transmitted to the refrigerant as an enthalpy increase. This is shown in Figure 16 as ΔH_{comp} from state points 1 to 2.

From this information, state point 2 is determined and the energy balance process begins. The evaporator subprogram ECOIL determines a capacity for the given evaporating temperature. If this does not match the amount of heat absorbed by the refrigerant ($\dot{M} \cdot \Delta H_{\text{evap}}$), a new evaporating temperature is assumed and the program loops back to the refrigerant properties service routine to determine revised state points 1 and 4. This looping process continues until the evaporator capacity is within a specified tolerance of the heat absorbed by the refrigerant.

COMPUTER SIMULATION
REFRIGERATION CYCLE
P-H DIAGRAM

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FIGURE 22

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With the new evaporating temperature, the compressor subroutine is called again and a revised mass flow ($\dot{M}2$) and power input ($\dot{W}2$) is determined. The new mass flow ($\dot{M}2$) is compared to the old ($\dot{M}1$) and if the tolerance is not satisfied, then the program loops back with the new compressor data to the evaporator subprogram and begins the process over again. Once the capacity and mass flow tolerance is satisfied, then the program execution continues.

The energy balance process continues when the condenser subprogram determines a capacity for the given condensing temperature. If this capacity does not match the amount of heat rejected by the refrigerant ($\dot{M} \cdot \Delta H_{\text{cond}}$), then a new condensing temperature is assumed and the looping process continues until the condenser capacity is within a specified tolerance of the heat absorbed by the refrigerant. When the tolerance is satisfied, new state points at 2 and 3 are determined for the revised condensing temperature and the initial condensing pressure is compared to the final condensing pressure. If the tolerance is not satisfied, then all the revised state points are looped back to the beginning of the program execution and the entire process is repeated until all tolerances are satisfied.

When the energy and pressure balance is complete, the program calculates and outputs the following:

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Water-to-Air Heat Pump:

- Heating mode
 - (1) Exit air temp. - dry bulb
 - (2) Exit water temp.
 - (3) Net capacity (Btuh) including blower work
 - (4) Net efficiency (COP) including blower work

- Cooling mode
 - (1) Exit air temp. - dry bulb
 - (2) Exit air temp. - wet bulb
 - (3) Exit water temp.
 - (4) Total cooling capacity (Btuh) including blower work
 - (5) Sensible cooling capacity (Btuh)
 - (6) S/T ratio
 - (7) Net efficiency (EER) including blower work

Water-to-Water Heat Pump:

- Heating mode
 - (1) Exit air temp. - dry bulb
 - (2) Exit water temp. (source)
 - (3) Water temp. entering the hydronic coil
 - (4) Water temp. leaving the hydronic coil
 - (5) Net capacity (Btuh) including blower work
 - (6) Net efficiency (COP) including blower work

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- Cooling mode

- (1) Exit air temp. - dry bulb
- (2) Exit air temp. - wet bulb
- (3) Exit water temp. (source)
- (4) Water temp. entering the hydronic coil
- (5) Water temp. leaving the hydronic coil
- (6) Total cooling capacity (Btuh) including blower work
- (7) Net efficiency (EER) including blower work

The program also outputs the final thermodynamic properties of each state point in the refrigerant cycle, the mass flow rate, and the inlet conditions of both water and air.

APPENDIX B

HPSIM OUTPUTS

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1 HIGH SPEED HEATING

STATE	T DEG F	P PSIA	X	H	S	Q BTU/HR
-------	------------	-----------	---	---	---	-------------

	57.19	94.2		110.54	.2221	
						10985.4
	192.0	252.6		129.74	.2353	
						50391.2
	107.0	247.6		41.49	0.000	
	53.22	104.2	0.00	41.49	0.000	
						39565.9

REFRIGERANT MASS FLOW = 571. LBM/HR
HIGH SPEED COMPRESSOR

INLET AIR-DRY BULB TEMP. 70.00 DEG F

INLET AIR-WET BULB TEMP. 60.00 DEG F

OUTLET AIR-DRY BULB TEMP. 64.43 DEG F

OUTLET AIR-WET BULB TEMP. 0.00 DEG F

INLET WATER TEMPERATURE = 70.00 DEG.F.

OUTLET WATER TEMPERATURE = 62.11 DEG.F.

WATER FLOW RATE = 10.00 GPM.

HEAT EXCHANGER EFFECTIVENESS = .400

INDOOR AIR FLOW = 1350. CFM.

COMPRESSOR + BLOWER WORK = 12673. BTU/HR.

HEATING + BLOWER WORK = 52079. BTU/HR.

UNIT COP = 4.11

DEGREES SUPERHEAT -10.0 DEG F

DEGREES SUBCOOLING 5.00 DEG F

LOW SPEED HEATING

STATE	T DEG F	P PSIA	X	H	S	Q BTU/HR
-------	------------	-----------	---	---	---	-------------

	64.46	106.3		111.15	.2208	
						4165.3
	151.3	198.1		123.46	.2302	
						29627.0
	88.75	193.1		35.77	0.000	
	60.00	116.3	0.00	35.77	0.000	
						25466.4

REFRIGERANT MASS FLOW = 338. LBM/HR
LOW SPEED COMPRESSOR

INLET AIR-DRY BULB TEMP. 70.00 DEG F
INLET AIR-WET BULB TEMP. 60.00 DEG F
OUTLET AIR-DRY BULB TEMP. 64.43 DEG F
OUTLET AIR-WET BULB TEMP. 0.00 DEG F
INLET WATER TEMPERATURE = 70.00 DEG.F.
OUTLET WATER TEMPERATURE = 64.91 DEG.F.
WATER FLOW RATE = 10.00 GPM.
HEAT EXCHANGER EFFECTIVENESS = .400
INDOOR AIR FLOW = 1350. CFM.
COMPRESSOR + BLOWER WORK = 5853. BTU/HR.
HEATING + BLOWER WORK = 31314. BTU/HR.
UNIT COP = 5.35

DEGREES SUPERHEAT -10.0 DEG F
DEGREES SUBCOOLING 5.00 DEG F

HIGH SPEED COOLING

STATE	T DEG F	P PSIA	X	H	S	Q BTU/HR
	57.47	94.6		110.56	.2220	
	177.2	220.1		127.84	.2351	10486.5
	96.55	215.1		38.19	0.000	54564.8
	50.53	99.6	0.00	38.19	0.000	43887.2

REFRIGERANT MASS FLOW = 609. LBM/HR
HIGH SPEED COMPRESSOR

FACE**AREND08R CB169-SDAET6*****
FACE AIR VELOCITY IS 241.49 FT/MIN.
TOTAL CAPACITY IS 42447. BTU/HR.
SENSIBLE COOLING IS 31483. BTU/HR.
S/T RATIO IS .74170

INLET AIR-DRY BULB TEMP. 80.00 DEG F
INLET AIR-WET BULB TEMP. 67.00 DEG F
OUTLET AIR-DRY BULB TEMP. 58.84 DEG F
OUTLET AIR-WET BULB TEMP. 57.22 DEG F
INLET WATER TEMPERATURE = 85.00 DEG.F.
OUTLET WATER TEMPERATURE = 95.94 DEG.F.
WATER FLOW RATE = 10.00 GPM.
HEAT EXCHANGER EFFECTIVENESS = .600.
INDOOR AIR FLOW = 1350. CFM.
COMPRESSOR + BLOWER WORK = 12174. BTU/HR.
COOLING - BLOWER WORK = 42200. BTU/HR. UNIT EER = 11.83
DEGREES SUPERHEAT -10.0 DEG F
DEGREES SUBCOOLING 5.00 DEG F

LOW SPEED COOLING

STATE	T DEG F	P PSIA	X	H	S	Q BTU/HR
	65.29	107.8		111.22	.2207	
	149.5	196.6		123.16	.2298	4127.7
	88.22	191.6		35.61	0.000	30151.2
	58.08	112.8	0.00	35.61	0.000	
						26036.8

REFRIGERANT MASS FLOW = 344. LBM/HR
LOW SPEED COMPRESSOR

*****ARENDOOR C8159-SBAET6*****
FACE AIR VELOCITY IS 241.49 FT/MIN.
TOTAL CAPACITY IS 26241. BTU/HR.
SENSIBLE COOLING IS 25409. BTU/HR.
S/T RATIO IS .96828

INLET AIR-DRY BULB TEMP. 80.00 DEG F
INLET AIR-WET BULB TEMP. 67.00 DEG F
OUTLET AIR-DRY BULB TEMP. 62.93 DEG F
OUTLET AIR-WET BULB TEMP. 61.19 DEG F
INLET WATER TEMPERATURE = 85.00 DEG.F.
OUTLET WATER TEMPERATURE = 91.03 DEG.F.
WATER FLOW RATE = 10.00 GPM.
HEAT EXCHANGER EFFECTIVENESS = .600
INDOOR AIR FLOW = 1350. CFM.
COMPRESSOR + BLOWER WORK = 5815. BTU/HR.
COOLING - BLOWER WORK = 24349. BTU.HR. UNIT EER = 14.29

DEGREES SUPERHEAT -10.0 DEG F
DEGREES SUBCOOLING 5.00 DEG F

REFERENCES

1. Bohn Aluminum and Brass Co., Heat Transfer Division, Bohn's Air Conditioning Coil Engineering Manual, Danville, Illinois, 1970.
2. R. D. Ellison and F. A. Creswick, A Computer Simulation of Steady-State Performance of Air-to-Air Heat Pumps, Oak Ridge National Laboratory, Oak Ridge, Tennessee, 1978.