27
10/12/78
25 & NTIS

ORNL/CON-23

MASTER MASTER

Conservation
—
Division of Buildings
and
Community Systems

Intermediate Report on the Performance of Plate-Type Ice-Maker Heat Pumps

V. D. Baxter

OAK RIDGE NATIONAL LABORATORY
OPERATED BY UNION CARBIDE CORPORATION · FOR THE DEPARTMENT OF ENERGY

DISCLAIMER

Printed in the United States of America. Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22161
Price: Printed Copy $4.50 ; Microfiche $3.00

ORNL/CON-23

Contract No. W-7405-eng-26

Energy Division

INTERMEDIATE REPORT ON THE PERFORMANCE OF PLATE-TYPE ICE-MAKER HEAT PUMPS

V. D. Baxter

Date Published: October 1978

Department of Energy
Division of Buildings and Community Systems

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830
operated by
UNION CARBIDE CORPORATION
for the
DEPARTMENT OF ENERGY

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

CONTENTS

ABSTRACT 1

1. INTRODUCTION 2

2. SUMMARY 3

2.1 Results and Conclusions 3

2.1.1 Water-chilling tests 3

2.1.2 Ice-making tests 3

2.1.3 Cyclic tests 4

2.2 Future Efforts and Recommendations 4

3. DESCRIPTION OF TEST UNITS 5

3.1 Remcor IMHP 5

3.2 Two-Plate IMHP 7

4. TEST SETUP AND INSTRUMENTATION 9

5. WATER-CHILLING TESTS 10

5.1 Procedure 10

5.2 Test Results 11

6. ICE-MAKING TESTS 16

6.1 Procedure 18

6.2 Test Results 18

7. CYCLIC TESTING 21

7.1 Test Procedure 21

7.2 Test Results 24

ACKNOWLEDGMENT 28

REFERENCES 28

APPENDIX 29

INTERMEDIATE REPORT ON THE PERFORMANCE OF PLATE-TYPE ICE-MAKER HEAT PUMPS

V. D. Baxter

ABSTRACT

A prototype ice-maker heat pump obtained from Remcor Products Company and a two-plate unit developed at Oak Ridge National Laboratory were tested under the Annual Cycle Energy System program. Results were compared for the effect of harvesting scheme and evaporator plate loading on performance in both water-chilling and ice-making modes.

The Remcor scheme of using compressor discharge gas for harvesting exacts a heavy penalty on performance during short freeze cycles, whereas the two-plate unit utilizing a no-penalty harvest scheme experiences no such penalty.

However, the lower plate loading of the Remcor gives it a performance advantage over the two-plate unit in water chilling and allows it to operate on a longer freezing cycle before suffering performance penalties due to ice buildup. This latter effect allows the Remcor to overcome its harvest penalty and to achieve a slightly higher maximum coefficient of performance in the ice-making mode than does the two-plate unit. Accordingly, it is concluded that a combination of no-penalty harvest and low plate loading should provide a more optimum performance level for ice-maker heat pumps.

Cyclic operation, typical of domestic space conditioning operation, of the two-plate unit revealed significant performance loss due to interrupted compressor operation. The possibility of improving operation and reliability by combining thermostatic cycles with harvesting cycles is discussed and a possible scheme presented.

1. INTRODUCTION

The sharp increase in the price of energy since 1973 has revived interest in a number of old, but valid, ideas of how to utilize energy in more efficient ways. One of these is the Annual Cycle Energy System (ACES),[1,2] an integrated system which uses a heat pump and energy storage to provide space heating, space cooling, and domestic hot water. During the heating season, the heat pump extracts the required heat from a tank of water, converting a portion of the water to ice, which is stored to provide cooling during the summer. For this system, since both the heating and cooling outputs of the heat pump are used, the annual efficiency is considerably higher than for conventional systems.

In the original conception, a chilled antifreeze solution is circulated through tubing submerged in the water tank, and ice freezes on the tube surface. An alternative, less labor-intensive, and less expensive approach would utilize an ice-maker heat pump (IMHP). An IMHP would have the ice formation occurring on refrigerated plates located above the ice bin. Periodically, the ice would be harvested and dropped into the bin.

The purpose of this report is to compare the performance of two IMHPs, the two-plate unit developed by Fischer[3] and a sample unit built by Remcor Products Company, as observed in laboratory experiments. There are two major differences between the two machines. The first is the type of harvesting scheme employed. The two-plate machine uses warm refrigerant collected at the condenser exit in a receiver for harvesting, resulting in a "no-penalty" process in which no condenser output is lost due to the harvest. In the Remcor system, hot gas from the compressor is used, resulting in no heat output from the condenser during harvest. The second difference is due to the evaporator and compressor size. There are four plates in the Remcor with a total active area of 37.78 ft^2 (3.51 m^2), while the two-plate unit has an active area of 30 ft^2 (2.79 m^2). In addition, the compressor of the Remcor is smaller than that of the two-plate unit, resulting in an evaporator plate loading on the Remcor of 60 to 70% of that on the two-plate unit. The test results discussed in this report will attempt to ascertain the effect of these two differences on the relative performance of the machines in both water-chilling and ice-making modes.

In addition to the tests mentioned above, the effect of thermostatic-type cyclic operation on performance is investigated.

2. SUMMARY

2.1 Results and Conclusions

2.1.1 Water-chilling tests

Both units exhibited the same tendencies, in that heating capacity, coefficient of performance (COP), and evaporator temperature decreased as tank water temperature decreased. However, for a given tank water temperature, the Remcor's COP was higher. For example, at a tank temperature of 46.7°F (8.2°C), the Remcor had a COP of 4.06, while the two-plate machine had a COP of 3.82. This effect is due to lower specific plate loading on the Remcor, causing it to operate at a higher evaporator temperature.

The COP referred to here and elsewhere in this report is the compressor-only COP, that is, condenser heat rejection divided by compressor electrical input.

2.1.2 Ice-making tests

The Remcor's system of using compressor discharge gas for harvesting ice caused its condenser output to drop to zero during the harvest. This resulted in the machine suffering a significant performance penalty (up to 30% drop in COP) when operating on short freeze cycle times (20 min or less), giving COPs of about 2.2 to 2.7. The two-plate unit, due to its no-penalty harvesting scheme, experienced a negligible penalty and had a COP of around 2.94 for those freeze cycle times.

For freeze times longer than 20 min, the ice buildup on the evaporator of the two-plate machine insulated the plates and caused the COP to drop off. The lower plate loading on the Remcor reduced the rate of ice buildup on its evaporator plates, enabling it to attain freeze times of over an hour and COPs of 3 to 3.05. It is evident that both machines achieved the same level of performance despite their different harvesting methods, because the longer freeze times of the Remcor enabled it to overcome the penalty suffered during its harvest cycle. This leads to the conclusion that combining low plate loading with a no-penalty harvest scheme could provide high levels of performance across a much wider range of freezing cycle times.

2.1.3 Cyclic tests

Cyclic part-load tests run on the two-plate machine indicated that for about a 10% load, the COP for cyclic operation was 70% of that for steady-state, full-load operation. This level of performance was achieved for compressor run times of 3 min or longer. In general, as the on-off cycling frequency increased, performance of the unit decreased. It was also found that as the on-time of the compressor increased per cycle, performance increased. This leads to the obvious conclusion that the unit should be sized to match its expected load as nearly as possible in order to avoid excessive cyclic operation of the compressor.

2.2 Future Efforts and Recommendations

In an effort to reduce the amount of time spent in harvesting with the Remcor, the harvest circuit should be modified to admit the harvesting gas to the top of the plate rather than the bottom. Experience has shown that harvesting from the top down is faster. In addition, this would reduce the harvest penalty.

A study of freezing plate performance should be undertaken, using a variable-speed, open-type compressor. This would demonstrate the effect of varying plate loading on a given unit and provide information toward determining an optimum combination of plate loading (evaporator area) and capacity (compressor size).

Efforts should be undertaken to simplify the refrigerant circuitry, particularly that part devoted to harvesting. The possibility of coordinating the freeze-harvest cycle of an IMHP with the cyclic operation imposed by a thermostat should be investigated. If harvesting and cyclic operation could be lumped together in this way, there would be no need for the elaborate harvesting circuits and valving arrangements employed by both units described in this report.

Economic analyses should be combined with component studies to determine the most cost-effective and energy-efficient units for use in space heating and cooling and hot-water heating applications.

The effect of ambient air surrounding the evaporator plates on the ice buildup rate should be investigated thoroughly. As noted later in the discussion of ice-making test results, insulation of the freezing plates degraded performance. It will, therefore, be necessary to study IMHP performance, especially harvesting efficiency, with the plates exposed to the cold air of an enclosed storage bin.

3. DESCRIPTION OF TEST UNITS

3.1 Remcor IMHP

The Remcor unit, shown schematically in Fig. 1, was mounted atop a large water storage tank in the laboratory. A water pump delivered the tank water to a header arrangement which distributed the water over the evaporator plates during operation. Originally, the unit was equipped with a compressor having a heating capacity rated at approximately 30,000 Btu/hr (8800 W) for a condenser temperature of 105°F (41°C) and an evaporator temperature of 25°F (−4°C). During preliminary testing, this compressor was found to be defective and was replaced. The heating capacity of the unit with the new compressor at 105°F (41°C) condensing and 20 to 25°F (−7 to −4°C) evaporating is approximately 20,000 Btu/hr (5900 W). The compressors were of the same manufacturer and product series; consequently, it is expected that their efficiencies are nearly the same.

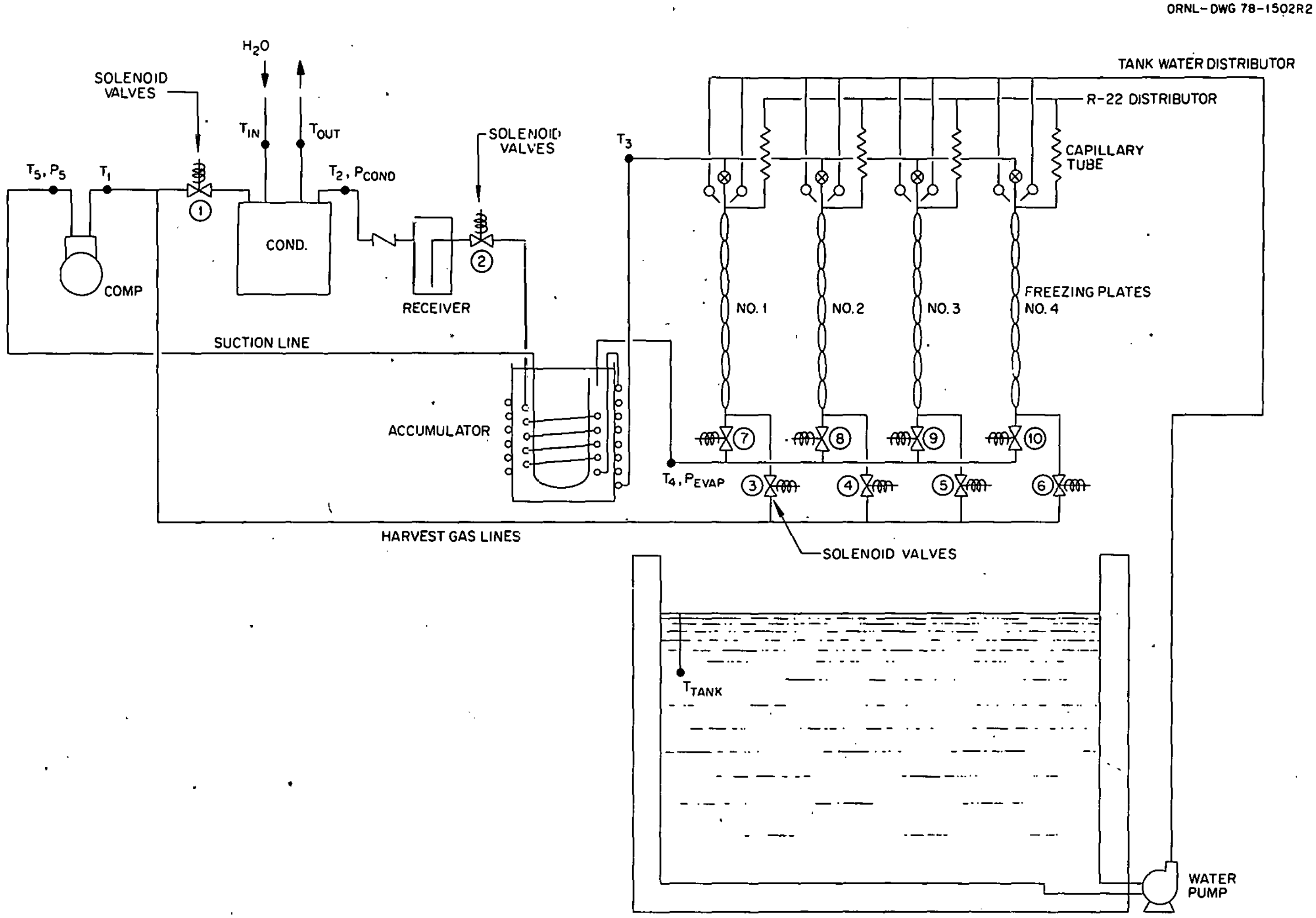

Fig. 1. Remcor schematic diagram.

The Remcor evaporator consisted of four 41.5 × 21.25 in. (1.05 × 0.54 m) copper plates having an overall area of 50 ft^2 (4.65 m^2). In operation, water covered an area 36 × 18.5 in. (0.91 × 0.47 m) per plate side for a total active plate area of 37.78 ft^2 (3.51 m^2). Refrigerant circuitry through the plates consisted of a 1/2-in. (12.7-mm) tube making 14 passes on 2-in. (50.8-mm) tube centers.

Under normal conditions of chilling water or making ice, the Remcor operates with solenoid valves 1, 2, and 7–10 open and 3–6 closed (Fig. 1). When the harvest cycle begins, valves 1 and 2 close, bypassing the condenser and thereby reducing the heating output to zero. The plates are harvested in order 1, 2, 3, and 4. For plate 1, valve 7 closes and valve 3 opens, admitting hot gas from the compressor to the bottom of the plate. As the hot gas moves up through the plate, it condenses, thereby warming the plate and causing the ice to drop into the tank. The condensed R-22 exits the plate through the capillary tube to the R-22 distributor. It then flows through the capillary tubes into the other plates, where it evaporates and returns to the compressor through the suction line. Plates 2–4 are harvested in a similar manner. After the harvest cycle is complete, valves 1 and 2 reopen and normal operation resumes. Total time for harvest was approximately 3.5 min. The total length of the freeze-harvest cycle was controlled by a timing system that allowed a maximum cycle time of 73.5 min.

3.2 Two-Plate IMHP

The two-plate unit, shown schematically in Fig. 2, was mounted atop a second water tank in the laboratory. It was equipped with a compressor having a heat output of about 25,000 Btu/hr (7325 W) under the same conditions mentioned above.

The evaporator of this unit consisted of two mild-steel plates, 26 × 48 in. (0.66 × 1.22 m), having a total active surface area of 30 ft^2 (2.79 m^2). Refrigerant circuitry was formed by welds on 1-1/2-in. (38.1-mm) centers with 12 passes per plate.

In normal operation, solenoid valves 1, 2, and 3 are open, and the evaporator plates are freezing ice or chilling water, depending on the

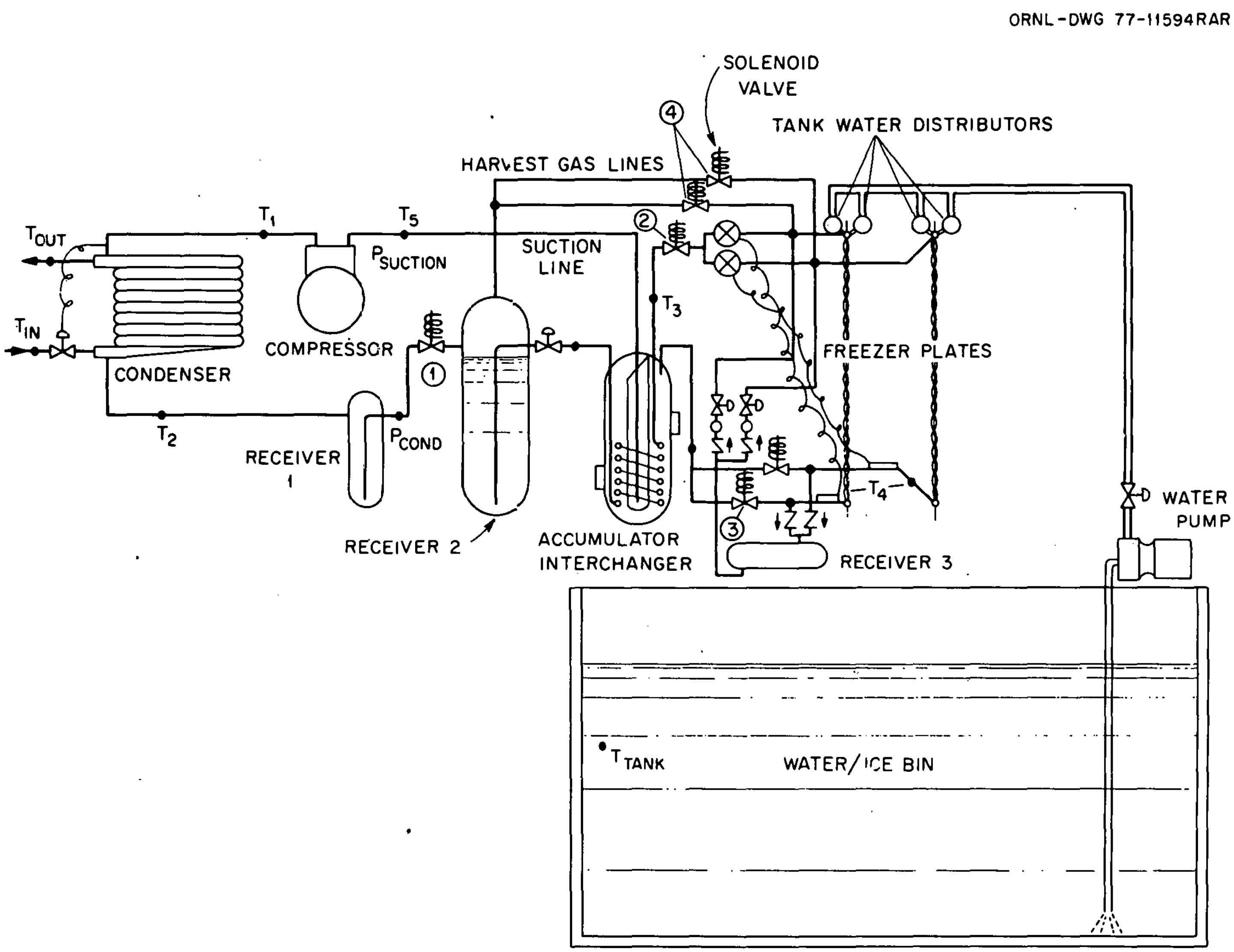

Fig. 2. Two-plate IMHP schematic diagram.

tank temperature. Refrigerant is circulating through the condenser, receivers 1 and 2, accumulator, evaporator plates, and back to the compressor with the excess charge filling up receiver 3. To effect harvest, valves 1 and 2 close and one of the valves 3 closes. The corresponding harvest valve 4 opens, connecting the plate to be harvested (at the suction-side pressure) to the gas space at the top of receiver 2 (at the high-side pressure). The liquid in the receiver immediately begins to boil, forcing the warm vapor into the top of the harvesting plate. This vapor condenses in the plate, warming it and causing the ice to slide into the bin. The condensed liquid is then routed through receiver 3 to the top of the other plate, where it evaporates to freeze ice and returns to the compressor via the accumulator. Meanwhile, the compressor and condenser continue to operate and produce heat normally, thus the harvest cycle exacts no penalty from the heating output of the machine.

The freeze-harvest cycle of the two-plate machine has been described by Fischer[3] as follows:

1. both plates freeze (typical time, 7.5 min);
2. first plate harvests while the second plate continues to freeze (typical time, 35 sec);
3. both plates freeze (typical time, 7.5 min);
4. second plate harvests while the first plate continues to freeze (typical time, 35 sec);
5. both plates freeze (typical time, 7.5 min).

The length of the freeze and harvest times was controlled by a timer with a maximum freeze time of 20.5 min.

4. TEST SETUP AND INSTRUMENTATION

The heat pump heating capacity was measured by observing the condenser cooling water temperature difference, using Cu-Const thermocouples, and monitoring the total coolant flow by weighing (for the Remcor) or with a water meter (for the two-plate unit). Cooling capacity was not directly measured in these tests. It could, however, be estimated by performing a heat balance on the refrigerant circuit.

Electrical input to the compressor was measured by a thermal-watt converter. In addition, a watt-hour meter was used as a backup and as a check on the primary instrument.

Refrigerant cycle temperatures and pressures were measured by Cu-Const thermocouples and pressure gages located as indicated in Figs. 1 and 2. These measurements were taken as a check on cycle operation and for use in refrigerant-side heat balances.

All temperatures and the compressor electrical input were recorded and averaged by the conservation laboratory data acquisition system described by Domingorena.[4]

5. WATER-CHILLING TESTS

Both units were tested in the water-chilling mode to determine the effect of evaporator temperature and plate loading on performance.

5.1 Procedure

The procedure for water-chilling tests for the Remcor unit was as follows:

1. The unit was turned on and allowed to run for approximately one-half hour to establish steady-state conditions in the refrigerant circuit. Since harvesting is unnecessary while chilling water, the harvest cycle was bypassed during these tests.
2. Every 15 min, condenser water flow was weighed for a 3-min period, during which time refrigerant cycle pressures were recorded. Refrigerant temperature, condenser cooling water temperatures, tank water temperature, and compressor electrical input were recorded every minute and averaged over the test period.
3. Step two was repeated until the unit began to make ice.

For the two-plate machine, the test procedure was somewhat different. The unit was operated without harvest; however, the tests were of 1-hr duration, with temperatures and compressor input recorded every minute and averaged at the end of a test. The tank water temperature seldom fell more than 2°F (1.1°C) during a test. Water flow was recorded by means of a flowmeter. Pressures were read from gages every 10 min.

5.2 Test Results

Water-chilling test results, as illustrated in Figs. 3 and 4, indicate that heating capacity, COP, and evaporator temperature decreased as the tank temperature decreased for both machines. Comparing the results of the two units indicated that the COP as a function of evaporator temperature was essentially the same for both units, as illustrated in Fig. 5 (upper). However, for a given water temperature, the Remcor had a higher evaporator temperature and therefore a higher COP (see Fig. 5). By looking at the heat transfer equation for the plates,

$$Q_E = UA_p(T_{H_2O} - T_{R-22}) ,$$

and noting from Table 1 that the evaporator load on the Remcor is approximately 85% that on the two-plate machine, it is possible to determine why this is so. We can define a specific plate loading, $\overline{Q}_E$, as

$$\overline{Q}_E = Q_E/A_p = U(T_{H_2O} - T_{R-22}) ,$$

where

Q_E = evaporator load,

A_p = evaporator active area,

U = overall heat transfer coefficient between water and refrigerant, based on evaporating temperature and entering water temperature,

T_{H_2O} = entering water temperature,

T_{R-22} = evaporating refrigerant temperature.

For these calculations, the evaporator load was calculated as follows:

$$Q_E = Q_o - F \cdot W_{in} ,$$

where

Q_o = condenser heat output,

W_{in} = compressor energy input,

F = fraction of electrical input to compressor that is added to refrigerant stream through compressor.

ORNL–DWG 78–1501

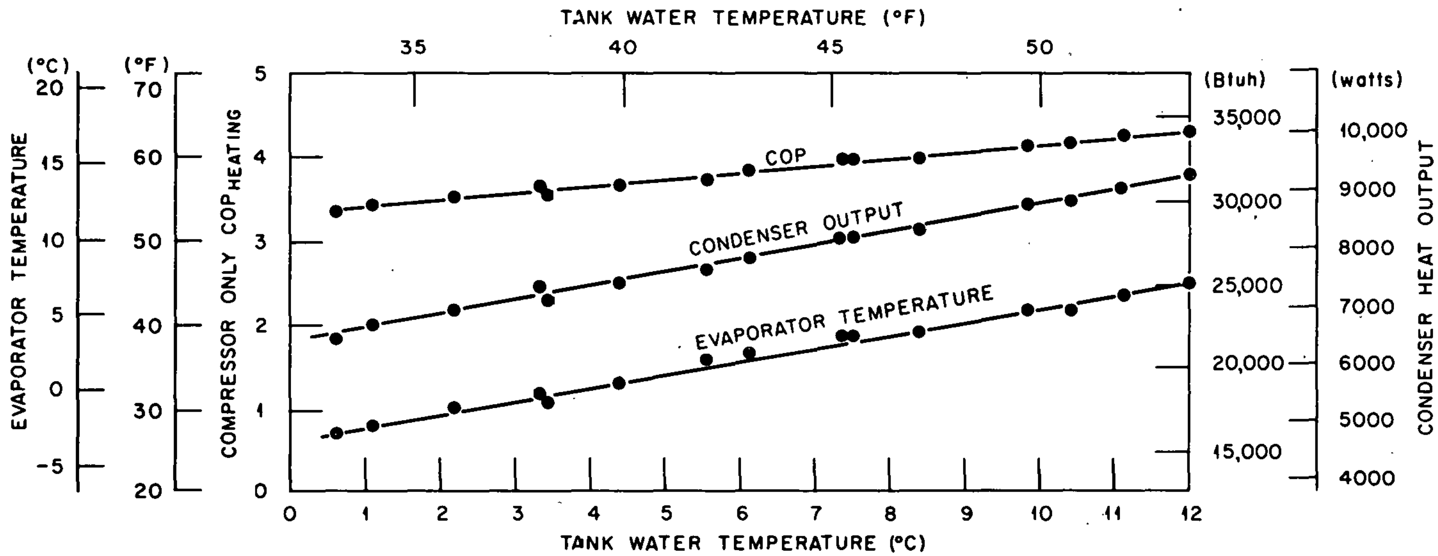

Fig. 3. Remcor COP and evaporating temperature vs tank water temperature.

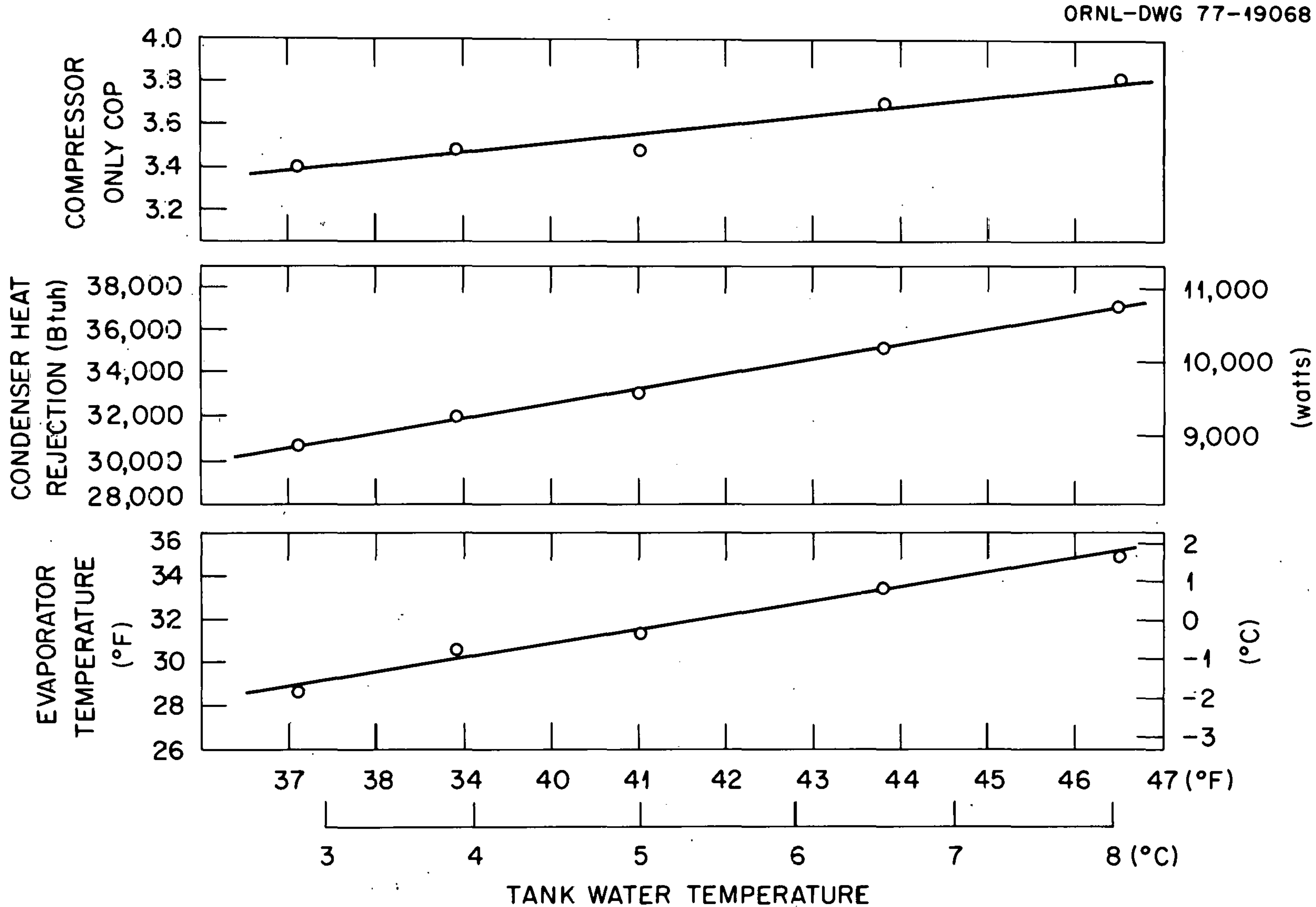

Fig. 4. Evaporator temperature, compressor-only COP, and capacity vs tank water temperature for two-plate IMHP, no harvesting.

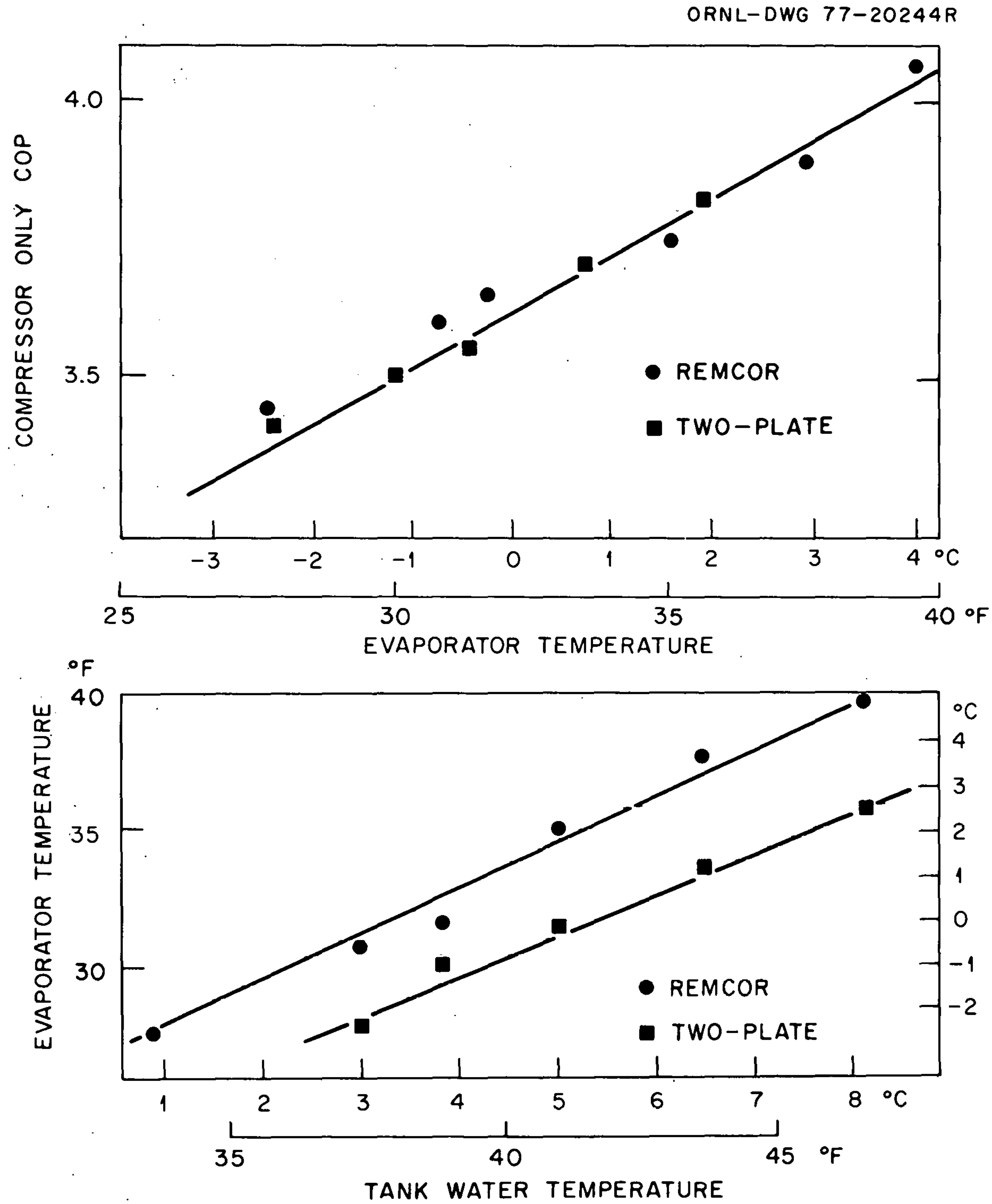

Fig. 5. Ice-maker heat pump performance as a water chiller.

Table 1. Water-chilling data from tests on two-plate and Remcor IMHPs

Tank temperature, °F (°C)	46.7 (8.2)	43.7 (6.5)	41.0 (5.0)	38.9 (3.8)	37.4 (3.0)
Heating output, Btu/hr (W)					
Two-plate	37,108 (10,876)	35,124 (10,294)	33,139 (9712)	32,079 (9402)	30,797 (9026)
Remcor	28,720 (8417)	27,134 (7953)	25,749 (7547)	24,747 (7253)	24,165 (7082)
Compressor input, Btu/hr (W)					
Two-plate	9721 (2849)	9504 (2786)	9334 (2736)	9180 (2691)	9042 (2650)
Remcor	7080 (2075)	6984 (2047)	6866 (2012)	6771 (1984)	6704 (1965)
Evaporator load, Btu/hr (W)					
Two-plate	28,539 (8364)	26,570 (7787)	24,738 (7250)	23,817 (6980)	22,654 (6441)
Remcor	23,410 (6861)	21,896 (6417)	20,600 (6038)	19,669 (5765)	19,137 (5609)
Evaporator temperature, °F (°C)					
Two-plate	35.7 (2.1)	33.5 (0.8)	31.4 (−0.3)	30.0 (−1.1)	27.8 (−2.3)
Remcor	39.6 (4.2)	37.6 (3.1)	35.1 (1.7)	31.7 (−0.2)	30.8 (−0.7)
Specific plate loading, Btu/hr·ft^2 (W/m^2)					
Two-plate	951.3 (3000.0)	885.7 (2793.5)	824.6 (2600.8)	793.9 (2504.0)	755.3 (2382.2)
Remcor	619.6 (1954.2)	579.6 (1828.1)	545.3 (1719.9)	520.6 (1642.0)	506.5 (1598.0)
U factor, Btu/hr·ft^2·°F (W/m^2·°C)					
Two-plate	86.5 (491.2)	86.8 (492.8)	85.9 (487.7)	89.2 (506.5)	78.7 (446.9)
Remcor	87.3 (495.7)	95.0 (539.4)	92.4 (524.6)	72.3 (410.5)	76.8 (436.1)
F factor, dimensionless					
Two-plate	0.91	0.90	0.91	0.89	0.87
Remcor	0.74	0.76	0.75	0.73	0.75

The factor F was found by performing heat balances on the refrigerant side of the systems. Since the condensers were well insulated from the ambient air and the surface temperature of the insulation was only 2–4°F higher than the ambient air, it can be assumed that the heat loss to the surroundings is insignificant compared to the total heat transferred to the cooling water. For this case, the refrigerant flow rate can be determined by:

$$\dot{m}_{R-22} = \frac{Q_o}{h_1 - h_2} ,$$

where the enthalpies h_1, h_2 are determined from the corresponding refrigerant temperatures and pressures taken at points shown in Figs. 1 and 2 and also in Fig. 6, a representative pressure-enthalpy diagram of the refrigerant cycle. The factor F was then calculated by:

$$F = \frac{\dot{m}_{R-22}(h_1 - h_5)}{W_{in}} .$$

From analysis of several tests, average values of F were found to be 0.75 for the Remcor and 0.90 for the two-plate heat pump. These values were then used to calculate all the evaporator loads listed in Table 1. Sample calculations may be found in the Appendix.

As seen in Table 1, the specific plate loading factor was much smaller for the Remcor than for the two-plate machine. This is due to having both lower capacity and more evaporator area. The heat transfer coefficients for both machines, obtained by:

$$U = \frac{\overline{Q}_E}{T_{H_2O} - T_{R-22}} ,$$

are approximately the same. Therefore, the Remcor must have a lower temperature difference between water and refrigerant and thus a higher evaporator temperature.

6. ICE-MAKING TESTS

The performance of the two heat pumps while making ice was checked for the effect of two different parameters: freezing cycle time and

ORNL-DWG 78-1510

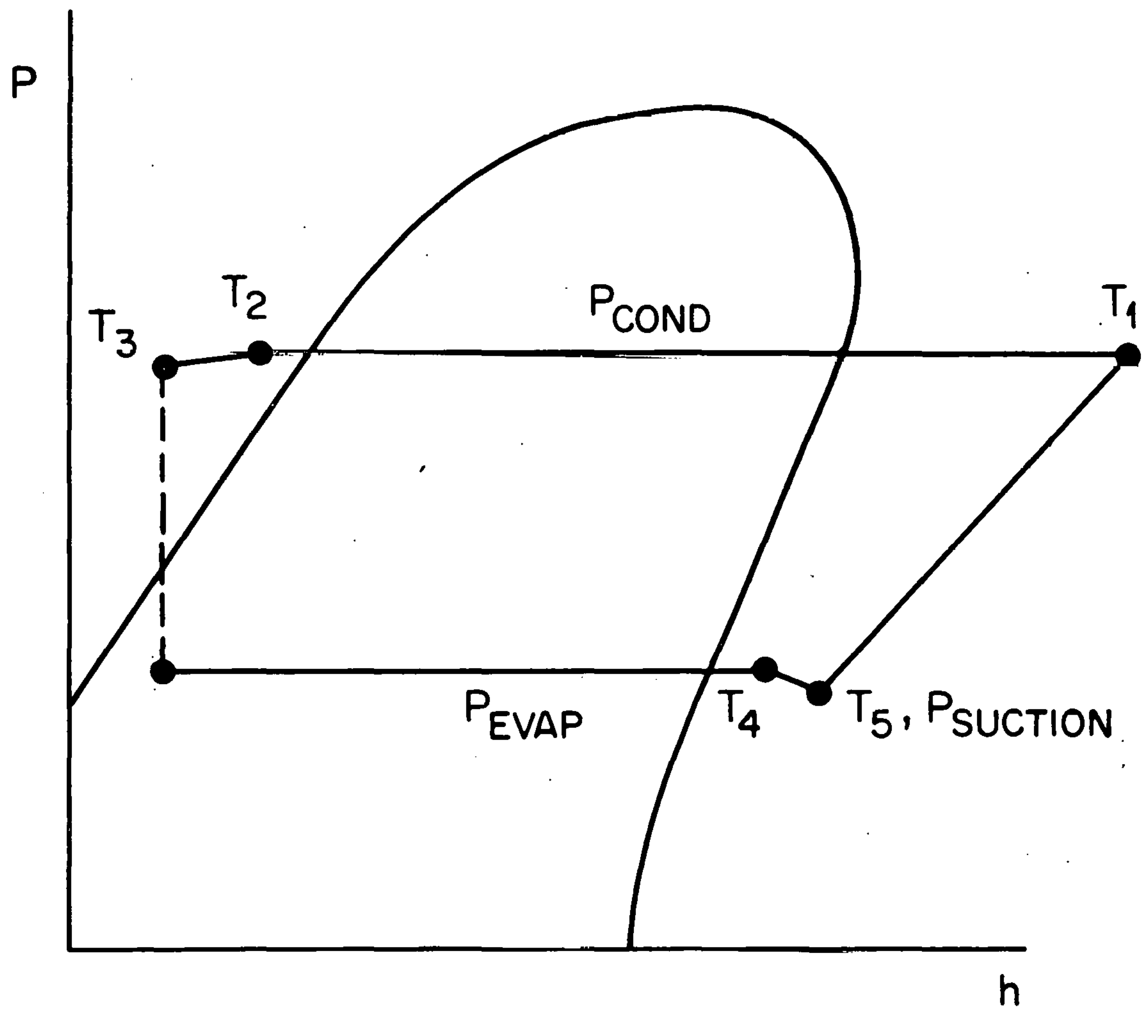

Fig. 6. Pressure-enthalpy diagram of IMHP refrigerant cycle.

condenser temperature. For both of these cases, the effects of plate loading and the two harvesting schemes were investigated.

6.1 Procedure

The procedure followed in the ice-making tests was to run the machines for at least 1 hr per test. For the longer freezing times, test duration was two full freeze-harvest cycles, starting and ending at the end of a harvest. Temperatures and compressor power usage were obtained through digital readout from the data acquisition system each minute. Pressures were read from gages as before. Cooling water flow rate was measured via water meter for the two-plate machine and by weighing for the Remcor.

6.2 Test Results

Figure 7 illustrates the performance of both units as a function of freezing time with and without harvesting. Tabular results are presented in the Appendix, Tables A.1 and A.2. The no-harvest curve for the two-plate unit was obtained experimentally by running the unit with the harvesting circuit bypassed. For the Remcor, this curve was obtained from the data taken during the freeze-harvest cycle tests by ignoring the data points taken during harvesting. This is possible since all of the heating output of the Remcor occurs during the freeze portion of the cycle. Some data points for the no-harvest curves are omitted for clarity.

As can be seen, for freezing times less than 20 min, the two-plate machine had an advantage when operating with the harvest cycle engaged. The reason is as follows: when the Remcor was harvesting, no heat was produced, that is, the unit operated with a heating COP of zero for the 3-1/2-min harvest. COPs without harvest for short freezing times were about the same for both machines.

The steady-state COP of the two-plate machine decreased gradually with freezing time while the Remcor's COP for both steady-state and freeze-harvest cycle operation increased with time. No freeze-harvest cycle tests for freeze times longer than 20 min were run on the two-plate unit, therefore it is not possible to say exactly what effect longer freeze

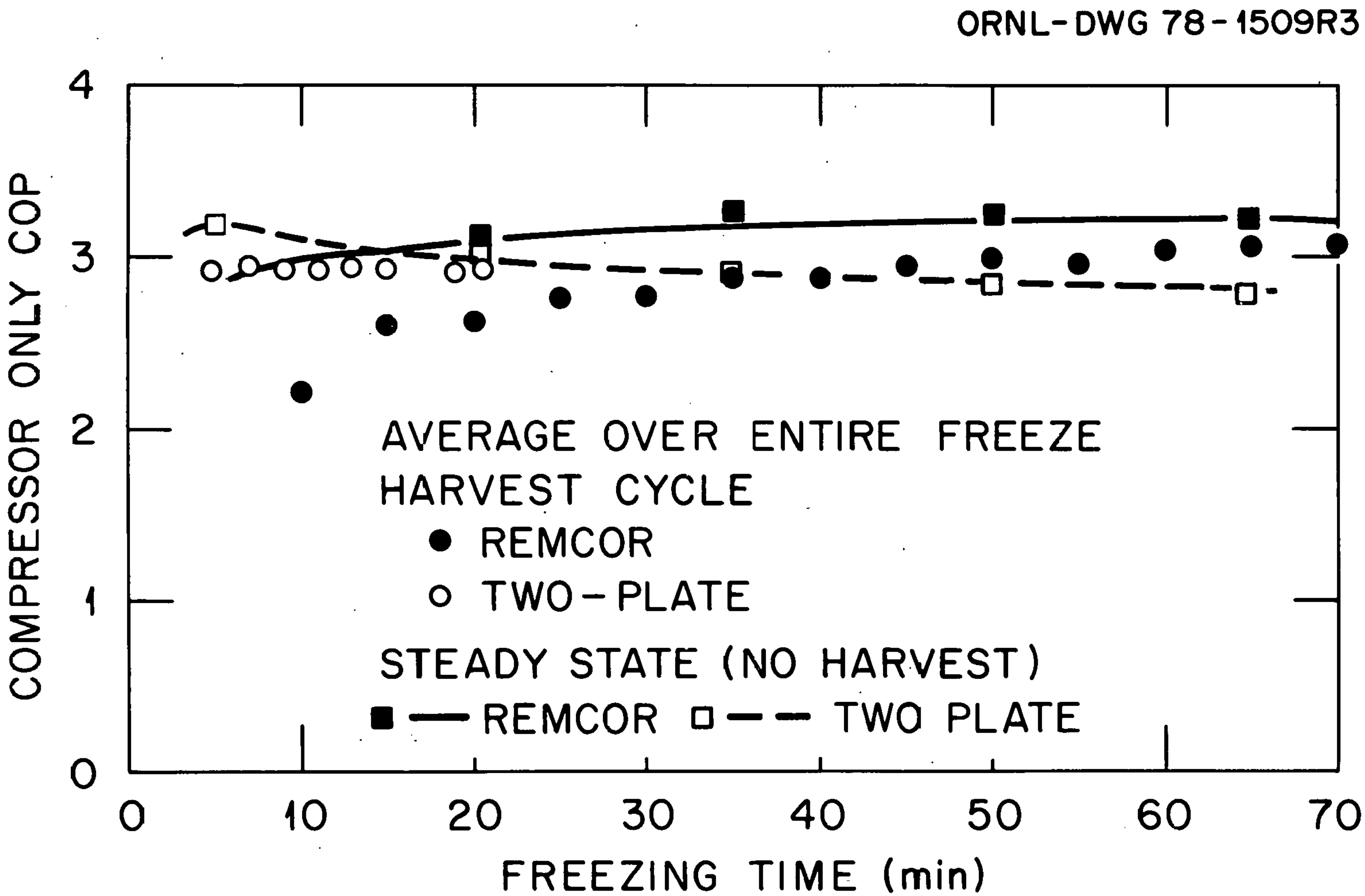

Fig. 7. Compressor-only COP vs length of freeze between harvests.

times would have had on its performance with harvesting. It is reasonable to assume, however, that freeze-harvest performance would follow the steady-state performance with slightly lower COPs.

There are two possible reasons for the improvement in the Remcor's performance with increasing freeze times:

1. The Remcor may have picked up more heat from the surrounding air than did the two-plate unit, thereby holding its evaporator at a higher temperature.
2. The lower plate loading of the Remcor retarded the buildup of ice on the plates, thus allowing it to run longer between harvests and negating the effect of the relatively short harvest period.

The data show that both machines achieved a peak COP in the ice-making mode of around 3.00. Apparently, a combination of the above two factors allowed the Remcor to overcome the deleterious effect of using compressor discharge gas for harvesting by enabling it to run such long freeze-harvest cycles.

Ambient air effects can be divided into two parts: (1) natural convection to the unwetted edges of the plates and (2) direct heat transfer from the air to the water flowing across the plates. The Remcor has about 2.5 times as much unwetted area as the two-plate unit. However, the natural-convection coefficients are on the order of 0.5 $Btu/hr \cdot ft^2 \cdot °F$ (2.84 $W/m^2 \cdot K$), and therefore the natural-convection effect is considered negligible. Direct heat transfer to the water film probably has a much greater effect, which, unfortunately, has not been quantified. Experiments with an insulated enclosure around the freezing plates of the Remcor indicated performance drops of about 1–2%, and harvesting difficulty at freeze times of over 35 min. Minimum harvesting time required increased by 35–50% for freeze times of 35 min or less, and by 15% for a 70-min freeze time.

It must be said that warm air near the evaporator plates probably had some retarding effect on ice growth on both machines with a somewhat greater effect on the Remcor due to its smaller capacity and larger evaporator. The magnitude of this effect is not known; however, it is assumed to represent a relatively small fraction of the total heat gain by the evaporators of both units. On this basis it is clear that the primary reason for the Remcor's ability to operate with such long freeze times is its much lower specific plate loading.

Experiments performed on the Remcor with reduced plate area bear out this assertion. Figure 8 shows heating capacity, COP, and evaporator temperature as functions of specific plate loading. Numerical results and sample calculations are presented in the Appendix, Table A.3. These data were obtained by testing the Remcor with all four plates freezing, three plates freezing, and so on down to one plate freezing. These tests were of the steady-state, no-harvest type, performed with the insulated enclosure around the plates. Test duration was 30 min each. As the evaporator size decreased, the performance suffered.

Ice-maker performance as a function of condenser temperature is shown in Fig. 9 (see Appendix for numerical results, Table A.4). Both machines behaved in a similar manner, with COP and capacity rising as condenser temperature fell. As can be seen, the Remcor very closely matched the COP of the two-plate machine despite its use of hot gas from the compressor for harvest. This was possible because it could freeze ice for long enough periods of time to negate the penalty involved in harvesting.

7. CYCLIC TESTING

The two-plate machine was tested under various cyclic conditions typical of the partial loading a heating system may see during periods of moderate heating demand, such as the early and late parts of a heating season. Performance effects of part loading, compressor run time, and cycling rate were examined.

7.1 Test Procedure

The machine was allowed to warm up for about 30 min; then, with a freeze cycle duration of 12 min, a full-load steady-state test was run to provide a base point for comparison with cyclic test results. Each cyclic test was run for a 1-hr period, beginning and ending at the end of an on-cycle. Cycling rates of 2, 4, and 6 cph and compressor on-times of 10, 20, 50, and 80% were used in the tests.

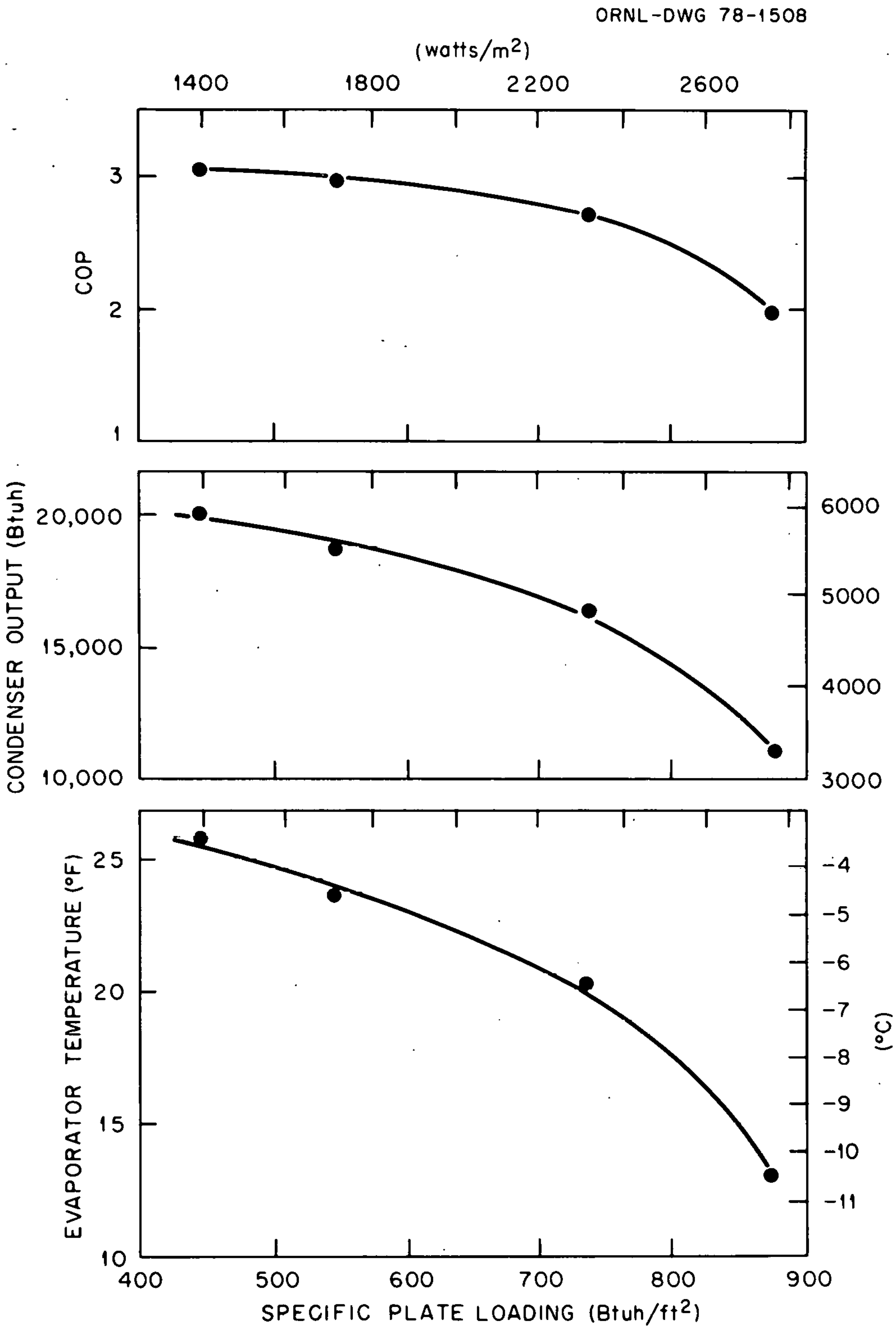

Fig. 8. Remcor heating capacity, COP, and evaporator temperature as functions of specific plate loading.

ORNL-DWG 78-1507

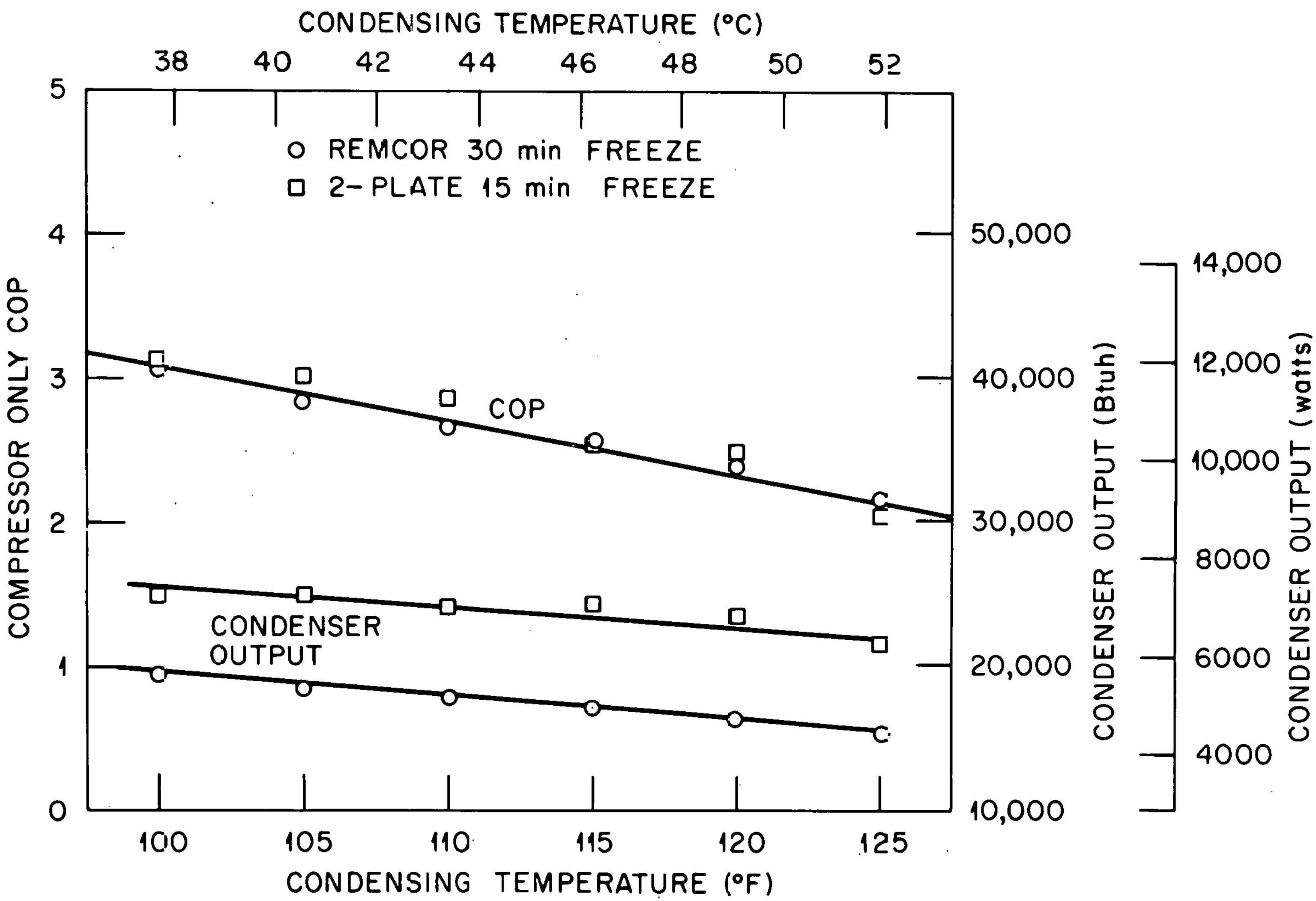

Fig. 9. Two-plate IMHP performance as a function of condenser temperature.

7.2 Test Results

In order to present the data more effectively, a part-load factor was defined as follows:

$$PF = \frac{Qo_c}{\dot{Qo}_{ss} \times \Delta t_t} ,$$

where

Qo_c = total cyclic heat output, Btu (Whr),
$\dot{Qo}_{ss}$ = steady-state, full-load heat output rate, Btu/hr (W),
Δt_t = total test duration, hr.

Test results are presented in Table 2 and Fig. 10. The figure shows the relationship of the ratio of cyclic COP to steady-state COP (COP_c/COP_{ss}) to the part-loading factor, compressor run time, and percent compressor on-time.

Table 2. Cyclic test results on two-plate machine

Comp. on-time (%)	Comp. on-time (min)	Qo_c (Btu)	Qo_c (Whr)	Cycles per hr	PF	$\frac{COP_c}{COP_{ss}}$
10	3	1,726.6	506	2	0.07	0.69
10	1.5	1,100.0	322.4	4	0.05	0.44
20	6	4,381.7	1284.2	2	0.18	0.86
20	3	3,162.2	926.8	4	0.13	0.63
50	15	11,754.5	3445.0	2	0.49	0.93
50	7.5	11,632.0	3409.1	4	0.48	0.91
50	5	11,051.5	3239.0	6	0.46	0.87
80	24	19,076.7	5591.1	2	0.79	0.94
80	12	18,867.9	5529.9	4	0.78	0.93
80	8	18,697.3	5479.9	6	0.77	0.93
100[a]		24,231.3	7101.8		1	1

[a]Full-load steady-state test.

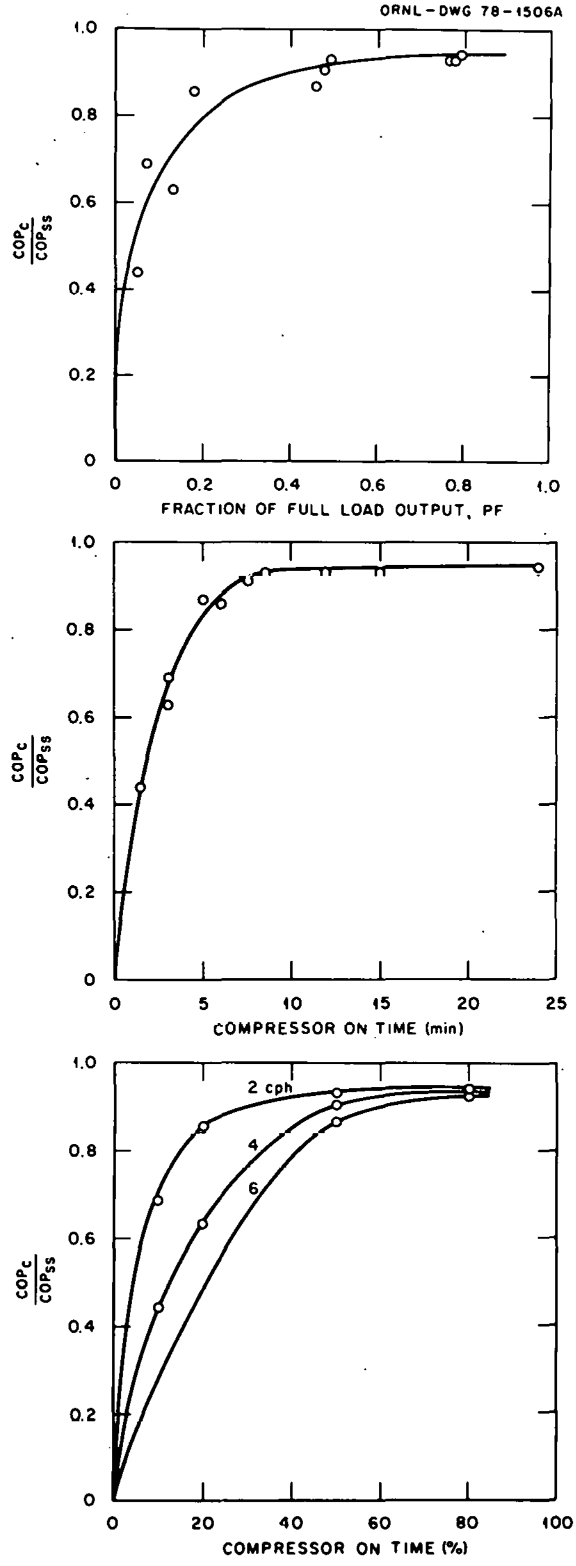

Fig. 10. Relationship of ratio of cyclic COP to steady-state COP to part-loading factor, compressor on-time, and percent compressor on-time.

For a part load of 10%, Fig. 10, the machine achieves 70% of its steady-state COP for compressor run times of 3 min or longer. The effect of cycling rate can also be seen in Fig. 10. For a given percent compressor on-time, performance improves as the cycling rate decreases.

It might be noted that as the load factor, PF, approaches 1.00, the ratio COP_c/COP_{ss} appears to approach some value less than 1.00. This can be accounted for because the test duration was limited to 1 hr. For longer test periods, allowing for longer on-cycle times, it is felt that the COP ratio would indeed approach 1.00.

The losses incurred due to cyclic operation of the two-plate machine with its no-penalty harvest lead to an interesting observation. Since ice-maker heat pump operation is inherently cyclic, and typical field operation under thermostat control is also cyclic, it should be possible to combine the cycles such that the ice is harvested simply due to the machine being turned off, thus obviating the need for an elaborate harvest scheme. If such a machine had a low enough plate loading level to allow it to run for 75 to 90 min before losing on performance, then operation need never be interrupted for harvesting except on those few very cold days when the thermostat requires the unit to run continuously. For such occasions, a timer can be incorporated to turn the unit off for approximately 5 min for harvest, after which it could resume operation.

Several such schemes are under review, and one or more should be tested in the near future. Figure 11 shows one possible application. During freezing operation, refrigerant from the receiver enters the subcooling coil, heating the liquid contained in the tank. Solenoid valve 1 is open and valve 2 is closed. When the compressor is shut off, whether by thermostat or timer, valve 2 opens and valve 1 closes. Liquid refrigerant at low pressure, just downstream of the expansion valve, flows by gravity to the bottom of the evaporating coil in the tank. As the refrigerant is evaporated by the warm liquid, its pressure will increase, and it will flow into the freezing plate from the suction side, where it condenses. This action warms the plate, causing the ice to drop into

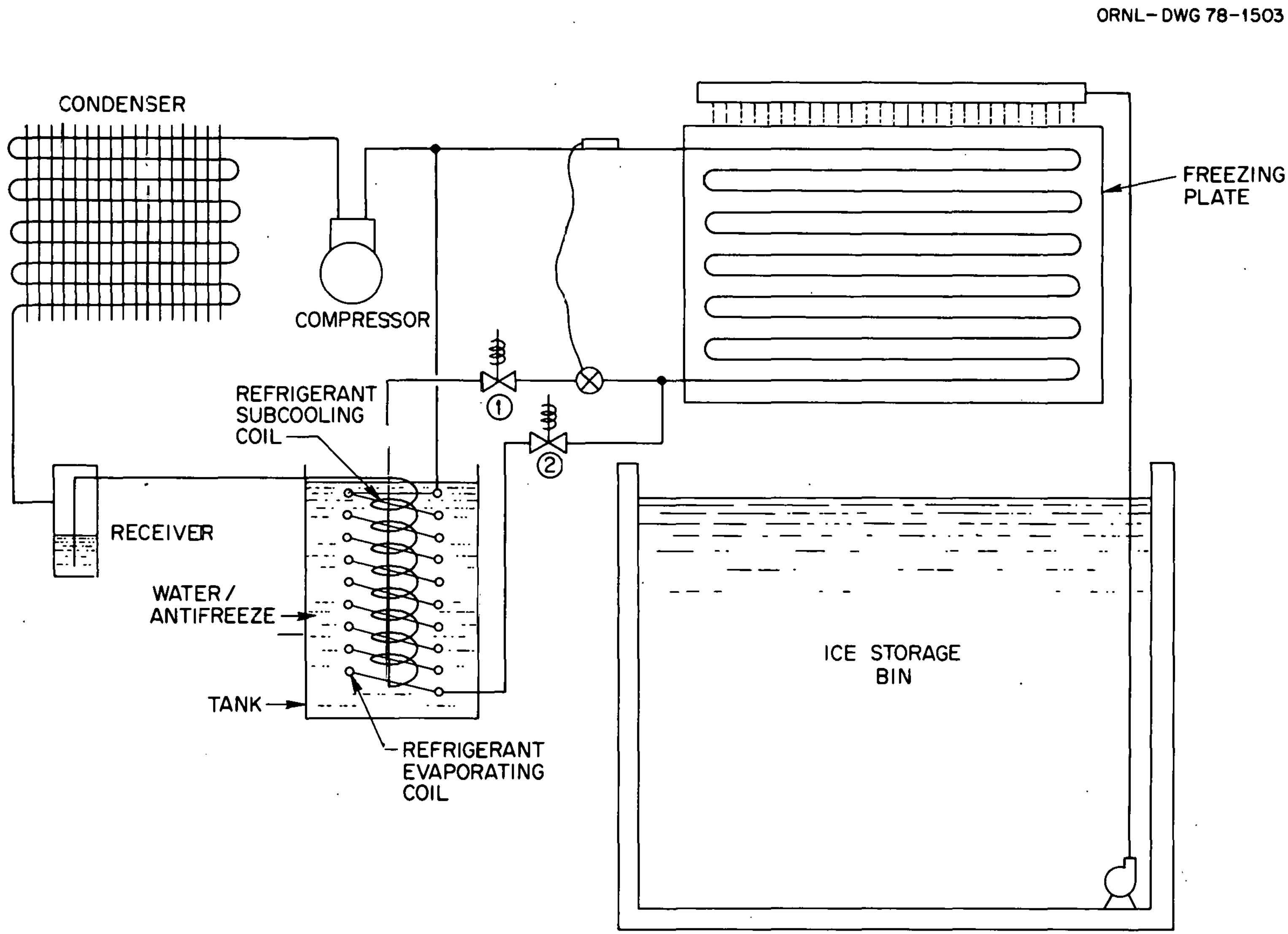

Fig. 11. Ice-maker heat pump with new harvesting technique.

the ice storage bin. The advantage of such a system is the absence of an excessive number of valves, fittings, and moving parts, greatly improving the system reliability.

ACKNOWLEDGMENT

The author gratefully acknowledges the important contributions of Phillip W. Childs in collecting and analyzing data and arranging for installation and maintenance of equipment.

REFERENCES

1. H. C. Fischer et al., *The Annual Cycle Energy System: Initial Investigations*, ORNL/TM-5525 (October 1976).

2. H. C. Fischer and E. A. Nephew, *Application of the Ice-Maker Heat Pump to an Annual Cycle Energy System*, paper presented at the ASME Winter Meeting, New York (December 1976).

3. H. C. Fischer, *Development and Testing of a Single-Plate and a Two-Plate Ice-Maker Heat Pump*, ORNL/CON-21 (April 1978).

4. A. A. Domingorena, *Performance Evaluation of a Low-First-Cost, Three-Ton, Air-to-Air Heat Pump in the Heating Mode*, ORNL/CON-18 (to be published).

APPENDIX

I. Sample calculations for Table 1

A. Data for tank water temperature of 38.9°F (3.8°C)

(1) Temperatures

(a) Cooling water entering condenser, Remcor: 61.6°F (16.4°C)

(b) Cooling water entering condenser, two-plate: 60.9°F (16.1°C)

(c) Cooling water leaving condenser, Remcor: 99.3°F (37.4°C)

(d) Cooling water leaving condenser, two-plate: 115.1°F (46.2°C)

(2) Refrigerant pressures

(a) Condenser exit, Remcor: 210 psig (1448 kPa)

(b) Condenser exit, two-plate: 210 psig (1448 kPa)

(c) Evaporator discharge, Remcor: 57 psig (393 kPa)

(d) Compressor suction, Remcor: 54 psig (372 kPa)

(e) Compressor suction, two-plate: 52 psig (359 kPa)

(3) Other

(a) Cooling water used, Remcor: 3.95 gal/0.05 hr

(b) Cooling water used, two-plate: 71 gal/hr

(c) Compressor power draw, Remcor: 1984.5 W

(d) Compressor power draw, two-plate: 2690.6 W

B. Calculations

(1) Condenser heating output

$$Q_o = (\dot{m}c\Delta T)_{\text{cooling water}}$$

(a) Remcor

$$Q_o = \left(\frac{3.94 \text{ gal}}{0.05 \text{ hr}}\right)(8.33 \text{ lb/gal})(1 \text{ Btu/lb}\cdot{}^\circ\text{F})(99.3 - 61.6)^\circ\text{F}$$

$$Q_o = 24{,}747 \text{ Btu/hr (7082 W)}$$

(b) Two-plate

Q_o = (71 gal/hr)(8.33 lb/gal)(1 Btu/lb•°F)(115.1 − 60.9)°F

Q_o = 32,079 Btu/hr (9402 W)

(2) Compressor-only COP

$$COP = Q_o/W_c \ ,$$

where W_c = compressor power draw

(a) Remcor COP = 24,747/6771 = 3.65

(b) Two-plate COP = 32,079/9180 = 3.49

(3) Evaporator load

$$Q_E = Q_o - F \cdot W_c$$

(a) Remcor: Q_E = 24,747 − 0.75(6771)
Q_E = 19,669 Btu/hr (5765 W)

(b) Two-plate: Q_E = 32,079 − 0.90(9180)
Q_E = 23,817 Btu/hr (6980 W)

(4) Evaporator temperature

(a) Remcor: at evaporator discharge, P_E = 57 psig (393 kPa); therefore, evaporator temperature T_E = 31.7°F (−0.2°C)

(b) Two-plate: suction pressure = 52 psig (359 kPa), assuming 3-psig (20.7-kPa) drop between evaporator discharge and compressor; evaporator discharge pressure = 55 psig (379 kPa); therefore, evaporator temperature T_E = 30°F (−1.1°C)

(5) Specific plate loading

$$\overline{Q}_E = Q_E/A_p \ ,$$

where A_p = active (wetted) evaporator area

(a) Two-plate: $\overline{Q}_E = 23{,}817/30$

$\overline{Q}_E = 793.9\ \text{Btu/hr·ft}^2\ (2504\ \text{W/m}^2)$

(b) Remcor: $\overline{Q}_E = 19{,}669/37.78$

$\overline{Q}_E = 520.6\ \text{Btu/hr·ft}^2\ (1642\ \text{W/m}^2)$

(6) U factor

$$U = \frac{\overline{Q}_E}{T_{H_2O} - T_{R\text{-}22}}$$

(a) Two-plate: $U = 793.9/(38.9 - 30.0)$

$U = 89.2\ \text{Btu/hr·ft}^2\text{·°F}\ (506.5\ \text{W/m}^2\text{·°C})$

(b) Remcor: $U = 520.6/(38.9 - 31.7)$

$U = 72.3\ \text{Btu/hr·ft}^2\text{·°F}\ (410.5\ \text{W/m}^2\text{·°C})$

Some of the foregoing calculational procedures were also used for the results given in Tables A.1, A.2, and A.4.

II. Sample calculations for Table A.1 (freezing time, 5 min)

A. Condenser heat output

(1) Freeze-harvest cycle

$$Q_{o_1} = \left(\frac{60.25\ \text{gal}}{1\ \ \text{hr}}\right)(8.33)(1)(110.6 - 58.3)\text{°F}$$

$$Q_{o_1} = 26{,}279\ \text{Btu/hr}\ (7702\ \text{W})$$

(2) No harvest

$$Q_{o_2} = \left(\frac{4.1\ \text{gal}}{0.0833\ \text{hr}}\right)(8.33)(1)(117.0 - 53.6)\text{°F}$$

$$Q_{o_2} = 25{,}984\ \text{Btu/hr}\ (7615\ \text{W})$$

B. Evaporator load

(1) Freeze-harvest cycle

$$Q_{E_1} = 26{,}279 - (0.9)(9030.4)$$

Table A.1. Variable freeze cycle length test results for two-plate IMHP

Freeze time (min)	Combined freeze-harvest cycle test data[a]							Steady-state (no-harvest) data						
	Condenser heat output		Evaporator load		Specific plate loading		COP	Condenser heat output		Evaporator load		Specific plate loading		COP
	Btu/hr	W	Btu/hr	W	Btu/hr·ft^2	W/m^2		Btu/hr	W	Btu/hr	W	Btu/hr·ft^2	W/m^2	
5	26,279	7702	18,152	5320	605	1908	2.91	25,984	7615	18,648	5465.5	622	1960	3.19
7	26,618	7801	18,556	5438	618.5	1951	2.97							
9	26,189	7676	18,103	5306	603	1903	2.91							
11	26,381	7732	18,244	5347	608	1918	2.92							
13	25,866	7581	17,985	5721	599.5	1891	2.95							
15	25,435	7455	17,452	5115	582	1835	2.87							
17	23,555	6904	15,613	4576	520.5	1641	2.67							
19	25,226	7393	17,419	5105	581	1831	2.91							
20.5	24,987	7323	17,342	5083	578	1823	2.94	23,390	6855	16,399	4806	547	1724	3.01
35								22,307	6538	15,445	4527	515	1624	2.93
50								21,403	6273	14,650	4294	488	1540	2.85
65								20,736	6077	14,059	4120	469	1478	2.80
80								20,312	5953	13,666	4005	456	1437	2.75
95								19,936	5843	13,295	3897	443	1398	2.70
110								19,656	5761	12,997	3809	433	1366	2.66
125								19,520	5721	12,795	3750	427	1345	2.61

[a]From ref. (3).

$$Q_{E_1} = 18{,}151.7 \text{ Btu/hr } (5320 \text{ W})$$

(2) No harvest

$$Q_{E_2} = 25{,}984 - (0.9)(8150.9)$$

$$Q_{E_2} = 18{,}648.2 \text{ Btu/hr } (5465.5 \text{ W})$$

(3) Specific plate loading

(a) Freeze-harvest cycle

$$\overline{Q}_{E_1} = \frac{18{,}151.7}{30} = 605 \text{ Btu/hr}\cdot\text{ft}^2 \ (1907.9 \text{ W/m}^2)$$

(b) No harvest

$$\overline{Q}_{E_2} = \frac{18{,}648.2}{30} = 621.6 \text{ Btu/hr}\cdot\text{ft}^2 \ (1960.3 \text{ W/m}^2)$$

(4) COP

(a) Freeze-harvest cycle

$$COP_1 = \frac{26{,}279}{9030.4} = 2.91$$

(b) No harvest

$$COP_2 = \frac{25{,}984}{8150.9} = 3.19$$

III. Sample calculations for Table A.2 (freezing time, 20 min)

A. Condenser heat output

(1) Freeze-harvest cycle

$$Q_{o_1} = (53.88 \text{ gal/hr})(8.33)(1)(100.55 - 61.46)^\circ\text{F}$$

$$Q_{o_1} = 17{,}543.7 \text{ Btu/hr } (5143 \text{ W})$$

Table A.2. Variable freeze cycle length test results for Remcor IMHP

Freeze time (min)	Combined freeze-harvest cycle test results							Steady-state (no-harvest) results						
	Condenser heat output		Evaporator load		Specific plate loading		COP	Condenser heat output		Evaporator load		Specific plate loading		COP
	Btu/hr	W	Btu/hr	W	Btu/hr·ft²	W/m²		Btu/hr	W	Btu/hr	W	Btu/hr·ft²	W/m²	
10	14,777	4331	9,773	2864	259	816	2.22	19,949	5847	14,945	4380	396	1247	2.99
15	16,675	4487	11,884	3483	315	992	2.61	20,566	6027	15,775	4623	418	1317	3.22
20	17,544	5142	12,547	3677	332	1047	2.63	20,614	6042	15,617	4577	413	1304	3.09
25	17,873	5238	12,992	3808	344	1084	2.75	20,375	5972	15,494	4541	410.1	1293	3.13
30	18,156	5321	13,238	3880	350	1105	2.77	20,274	5942	15,356	4501	406	1282	3.09
35	18,750	5495	13,866	4064	367	1157	2.88	20,625	6045	15,741	4614	417	1314	3.17
40	18,788	5507	13,901	4074	368	1160	2.88	20,432	5988	15,545	4556	411	1298	3.14
45	19,055	5585	14,220	4168	376	1187	2.96	20,537	6019	15,702	4602	416	1311	3.19
50	19,143	5611	14,347	4205	380	1198	2.99	20,483	6003	15,687	4598	415	1309	3.20
55	19,093	5596	14,269	4182	378	1191	2.97	20,307	5952	15,484	4538	410	1293	3.16
60	19,282	5651	14,476	4243	383	1208	3.01	20,407	5981	15,601	4572	413	1302	3.18
65	19,642	5757	14,793	4336	392	1235	3.04	20,700	6067	15,851	4646	420	1323	3.20
70	19,619	5750	14,810	4341	392	1236	3.06	20,600	6038	15,791	4628	418	1318	3.21

(2) No harvest

$$Q_{o_2} = Q_{o_1}\left(\frac{\text{freeze time + harvest time}}{\text{freeze time}}\right)$$

$$Q_{o_2} = (17{,}543.7)(23.5/20)$$

$$Q_{o_2} = 20{,}613.9 \text{ Btu/hr } (6042 \text{ W})$$

B. Evaporator load

(1) Freeze-harvest cycle

$$Q_{E_1} = 17{,}543.7 - (0.75)(6662.24)$$

$$Q_{E_1} = 11{,}547.7 \text{ Btu/hr } (3384 \text{ W})$$

(2) No harvest

$$Q_{E_2} = 20{,}613.9 - (0.75)(6662.24)$$

$$Q_{E_2} = 14{,}617.9 \text{ Btu/hr } (4284 \text{ W})$$

C. Specific plate loading

(1) Freeze-harvest cycle

$$\overline{Q}_{E_1} = \frac{11{,}547.7}{37.78} = 306 \text{ Btu/hr}\cdot\text{ft}^2 \ (964 \text{ W/m}^2)$$

(2) No harvest

$$\overline{Q}_{E_2} = \frac{14{,}617.9}{37.78} = 387 \text{ Btu/hr}\cdot\text{ft}^2 \ (1220 \text{ W/m}^2)$$

D. COP

(1) Freeze-harvest cycle

$$COP_1 = \frac{17{,}543.7}{6662.24} = 2.63$$

(2) No harvest

$$COP_2 = \frac{20,613.9}{6662.24} = 3.09$$

IV. Sample calculations for Table A.3 (three active plates)

A. Data taken

(1) Temperatures

(a) Refrigerant

1. compressor discharge gas (T_1) = 208.46°F (98°C)
2. liquid leaving condenser (T_2) = 99.67°F (36°C)
3. liquid entering expansion valve (T_3) = 71.63°F (22°C)
4. gas leaving evaporator plates (T_4) = 25.56°F (−4°C)
5. compressor suction gas (T_5) = 40.52°F (4.7°C)

(b) Other

1. cooling water entering condenser = 50.89°F (10.5°C)
2. cooling water leaving condenser = 97.54°F (36.4°C)
3. tank water-ice = 32.73°F (0.41°C)
4. air surrounding evaporator plates = 56.88°F (13.8°C)

(2) Refrigerant pressures

(a) Condenser exit (P_c) = 210 psig (1448.5 kPa)

(b) Evaporator entering (P_{E_1}) = 47.8 psig (329.7 kPa)

(c) Evaporator discharge (P_{E_2}) = 46.5 psig (320.7 kPa)

(d) Compressor suction (P_s) = 43.5 psig (300 kPa)

(3) Other

(a) Cooling water used = 24.1 gal

(b) Compressor power draw = 1845.3 W

B. Calculations

(1) Condenser heat output

$$Q_o = \left(\frac{24.1\ \text{gal}}{0.5\ \ \text{hr}}\right)(1\ \text{Btu/lb}\cdot°\text{F})(97.54 - 50.89)°\text{F}(8.33\ \text{lb/gal})$$

$$Q_o = 18,730.3\ \text{Btu/hr}\ (5490\ \text{W})$$

Table A.3. Variable plate loading test results for Remcor IMHP

COP	Condenser heat output		Evaporator load		Specific plate loading		Evaporator temperature		No. of plates active
	Btu/hr	W	Btu/hr	W	$Btu/hr \cdot ft^2$	W/m^2	°F	°C	
3.08	20,118.7	5896	16,736.6	4905	443	1397	25.9	− 3.4	4
2.99	18,730.3	5490	15,420.2	4519	544	1716	23.7	− 4.6	3
2.70	16,458.7	4824	13,878.9	4068	735	2318	20.4	− 6.4	2
2.05	11,092.9	3251	8,262.3	953	875	2759	13.1	−10.5	1

(2) Compressor-only COP

$$COP = \frac{5490}{1845.03}$$

$$COP = 2.98$$

(3) Refrigerant flow rate

$$\dot{m}_R = \frac{\dot{Q}_o}{h_1 - h_2}$$, from heat balance around condenser,

where h_1 and h_2 are refrigerant enthalpies corresponding to refrigerant conditions at compressor discharge and condenser exit respectively

$$\dot{m}_R = \frac{18,730.3}{132 - 39.2}$$

$$\dot{m}_R = 201.8 \text{ lb/hr } (91.7 \text{ kg/hr})$$

(4) Evaporator load

$$Q_E = \dot{m}_R(h_4 - h_3)$$

$$Q_E = 201.8(107 - 30.6)$$

$$Q_E = 15,420.2 \text{ Btu/hr } (4519 \text{ W})$$

(5) Specific plate loading

$$\overline{Q}_E = Q_E/A_p ,$$

where A_p = active (wetted) evaporator area

$$\overline{Q}_E = \frac{15,420.2}{(37.78)(0.75)}$$

$$\overline{Q}_E = 544 \text{ Btu/hr}\cdot\text{ft}^2 \ (1716 \text{ W/m}^2)$$

(6) Evaporator temperature

(a) At entrance, $P_{E_1} = 47.8$ psig (329.7 kPa);

therefore, $T_{E_1} = 24.2°F$ (−4.3°C)

(b) At exit, $P_{E_2} = 46.5$ psig (320.7 kPa);

therefore, $T_{E_2} = 23.1°F$ (−4.9°C)

$$T_E = \frac{T_{E_1} + T_{E_2}}{2}$$

$$T_E = 23.7°F\ (-4.6°C)$$

V. Sample calculations for Table A.4 (condensing temperature, 110°F)

A. Condenser heat output

(1) Remcor

$$Q_o = (54.99\ \text{gal/hr})(8.33)(1)(107.02 - 62.26)°F$$

$$Q_o = 17{,}828.8\ \text{Btu/hr}\ (5225\ \text{W})$$

(2) Two-plate

$$Q_o = (46.3\ \text{gal/hr})(8.33)(1)(122.12 - 59.68)°F$$

$$Q_o = 24{,}081.8\ \text{Btu/hr}\ (7058\ \text{W})$$

B. COP

(1) Remcor

$$\text{COP} = \frac{17{,}828.8}{6882.9} = 2.67$$

(2) Two-plate

$$\text{COP} = \frac{24{,}081.8}{8390.16} = 2.87$$

Table A.4. Variable condensing temperature test results for two-plate and Remcor IMHPs

Condensing temperature		Condensing pressure		Remcor, 30-min freeze			Two-plate, 15-min freeze		
				Condenser heat output		COP	Condenser heat output		COP
°F	°C	psig	kPa	Btu/hr	W		Btu/hr	W	
100	37.8	197	1359	19,489	5711.9	3.08	24,989	7323.9	3.18
105	40.6	210	1449	18,304	5365	2.80	25,036	7338	3.01
110	43.3	226	1559	17,828	5225	2.67	24,082	7058	2.87
115	46.1	243	1676	17,094	5010	2.52	24,228	7100	2.51
120	48.9	260	1793	16,247	4762	2.39	23,377	6851	2.49
125	51.7	278	1917	14,966	4386	2.14	21,113	6188	2.04

INTERNAL DISTRIBUTION

1. L. A. Abbatiello
2. S. I. Auerbach
3. M. Baker
4. V. D. Baxter
5. S. E. Beall
6. D. J. Bjornstad
7. J. R. Buchanan
8. R. S. Carlsmith
9. W. S. Chern
10. C. V. Chester
11. F. A. Creswick
12. R. M. Davis
13. J. G. Delene
14. C. R. Eichelberger
15. R. D. Ellison
16. W. Fulkerson
17. V. O. Haynes
18. N. E. Hinkle
19. E. A. Hirst
20. E. C. Hise
21. A. S. Holman
22. C. C. Hopkins
23. J. R. Jackson
24. P. L. Johnson
25. S. I. Kaplan
26. D. Kaserman
27. R. S. Livingston
28. H. M. Long

29–48. P. M. Love
49. J. W. Michel
50–69. R. E. Minturn
70. W. R. Mixon
71. J. C. Moyers
72. E. A. Nephew
73. D. L. O'Neal
74. H. Postma
75. M. W. Rosenthal
76. T. H. Row
77. R. L. Spore
78. E. G. Struxness
79. T. Takayama
80. L. H. Thomas
81. D. B. Trauger
82. G. U. Ulrikson
83. D. J. Walukas
84. T. J. Wilbanks
85. H. E. Zittel
86. A. Zucker
87. Biology Division Library
88–89. Central Research Library
90. Document Reference Section
91. Laboratory Records — RC
92–94. Laboratory Records Department
95. ORNL Patent Office
96–97. Energy Information Library

EXTERNAL DISTRIBUTION

98. Mr. H. W. C. Aamot, Cold Regions Research and Engineering Laboratory, U.S. Army Corps of Engineers, P. O. Box 282, Hanover, NH 03755
99. Mr. Roy W. Abbott, Manager, Advanced Technology, General Electric Company, Appliance Park, Building 6-208, Louisville, KY 40225
100. Mr. Ben Adams, President, Crouch and Adams, Inc., 106 Administration Road, Oak Ridge, TN 37830
101. Mr. Arthur G. Baitz, Assistant Vice President and Director of Planning, Robertshaw Controls Company, 1701 Byrd Avenue, P. O. Box 26544, Richmond, VA 23261
102. Mr. Hamp Barnett, Power Research Staff, Tennessee Valley Authority, 1310 Commerce Union Bank Building, Chattanooga, TN 37401
103. Mr. Ed Bottum, President, Refrigeration Research, Inc., 525 North Fifth Street, Brighton, MI 48116

104. Mr. John H. Cable, Architectural and Engineering Systems Branch, Office of Conservation, Department of Energy, 20 Massachusetts Avenue NW, Washington, D.C. 20545
105. Dr. Louis-Marie Chounet, Batiment 200, Lab. de L'Accelerateur Lineaire, 91405 Orsay, France
106. Mr. Gordon D. Duffy, Executive Editor, Air Conditioning News, 700 East Maple, Birmingham, MI 48011
107. Mr. Harry C. Fischer, Box 1687, Cocoa, FL 33932
108. Mr. Leo Foster, Elliott-Lewis Corporation, 2301 Cherry Street, Philadelphia, PA 19103
109. Mr. Joe Gracia, Department of Energy, Oak Ridge Operations, Oak Ridge, TN 37830
110. Mr. Dieter Grether, Vice President, Engineering, Friedrich Air Conditioning and Refrigeration, P. O. Box 1540, San Antonio, TX 78295
111. Mr. W. F. Hagen, President, Turbo Refrigerating Company, 1515 Shady Oaks Drive, Expressway Industrial Park, P. O. Box 396, Denton, TX 76201
112. Dr. Richard M. Kelso, Associate Professor, School of Architecture, University of Tennessee, 100 Estabrook Hall, Knoxville, TN 37916
113. Mr. Jim Leggoe, Philadelphia Electric Company, 2301 Market Street, Philadelphia, PA 19101
114. Mr. Herbert Lindahl, Manager, Advanced Engineering, Bohn Aluminum and Brass Corporation, Heat Transfer Division, 1625 East Voorhees Street, Danville, IL 61832
115. Mr. John F. Lowry, Manager, Applications Department, Philadelphia Electric Company, 2301 Market Street, Philadelphia, PA 19101
116. Mr. Calvin D. McCracken, President, Calmac Manufacturing Corporation, 150 South Van Brunt Street, Englewood, NJ 07631
117. Mr. Jim McCallum, Executive Editor, Air Conditioning and Refrigeration Business, 614 Superior Avenue West, Cleveland, OH 44113
118. Mr. Ben Miller, Manager of Engineering, REMCOR Products Company, 321 East Grand Avenue, Chicago, IL 60611
119. Mr. Dan Myers, Refrigeration Systems, 611 State Street, Newburg, IN 47630
120. Mr. Kenneth J. Nemeth, Executive Director, Southern States Energy Board, 7 Dunwoody Park, Suite 104, Atlanta, GA 30341
121. Mr. A. H. Rosenfeld, Lawrence Berkeley Laboratory, Building 50B, Room 5246, Berkeley, CA 94720
122. Dr. David P. Ross, Senior Scientist, Energy Resources and Conservation Division, Science Applications, Inc., 2028 Powers Ferry Road, Suite 260, Atlanta, GA 30339
123. Dr. Mason H. Somerville, Manager, Engineering Experiment Station, University of North Dakota, Grand Forks, ND 58202
124. Mr. Robert G. Werden, Werden & Associates, Inc., P. O. Box 414, Jenkintown, PA 19046
125. Institute of Energy Analysis, ORAU Library
126. Research and Technical Support Division, DOE-ORO

127–153. Technical Information Center, Department of Energy, P. O. Box 62, Oak Ridge, TN 37830

154–600. External Energy Conservation Distribution Mailing List and Energy Conservation Office (9102-1, Room 2)