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Development and Testing of a Single-Plate and a Two-Plate Ice-Maker Heat Pump

H. C. Fischer

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Department of Energy
Division of Buildings and Community Systems

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CONTENTS

LIST OF FIGURES

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H. C. Fischer

ABSTRACT

The step-by-step development of a working ice-maker heat pump is outlined for both a single-plate machine and a full-size two-plate machine built and tested at Oak Ridge National Laboratory. Test results reveal that a coefficient of performance (COP) of about 2.94 can be easily achieved with currently available components. By optimizing the design it is felt that a COP of 3.5 can be attained. Freezing cycle times of 12 to 20 min appear to be optimum from both capacity and COP standpoints. Harvesting times in excess of 40 sec need not be tolerated in a good design. A "no thermodynamic penalty" harvesting system that utilizes the enthalpy of the condensed liquid refrigerant to provide the heat to harvest the ice from the plate was developed and tested.

1. INTRODUCTION

The Annual Cycle Energy System (ACES),[1,2,3] being developed at Oak Ridge National Laboratory, provides space heating and domestic water heating by extracting energy from an insulated bin of water. As the energy is extracted, the water is converted to ice which is saved for summer air conditioning. Since both the heating and cooling outputs of the heat pump that accomplishes the extraction are used, the overall (annual) efficiency of the ACES is appreciably higher than that for conventional systems.

As originally conceived, a chilled antifreeze solution is circulated through tubing submerged in the bin of water and the ice freezes on the outer surface of the tubing. An alternate approach that may be advantageous in certain applications would have the ice formation occurring on refrigerated plates located above the water storage bin. Periodically, the ice would be harvested and dropped into the bin.

The purpose of the experiments reported herein was to explore the development of a new piece of equipment called the ice-maker heat pump, which could extract the latent as well as the sensible heat out of water and deliver this heat in a usable form to heat air or water for space heating, or domestic water heating, or both. The apparatus envisioned is one which freezes ice on a plate or other surface for a short time and then harvests the ice by thermally breaking its bond to the freezing surface so that the ice falls into the bin where it can be stored for later use in providing air conditioning or refrigeration.

Other designs of ice-maker heat pumps are possible, such as the scraped surface or auger types, which use mechanical power to remove the ice from the freezing surface.

The plate-type ice maker was chosen for this work because it is possible to design a "no thermodynamic penalty" harvesting system which does not diminish the output of the condenser (useful heat) in order to harvest the ice. The enthalpy of liquid R-22 refrigerant at the temperature that it exits the condenser, typically 38°C (100°F), is 91.14 kJ/kg (39.18 Btu/lb); the enthalpy of that same liquid at 4.4°C (40°F) is 49.72 kJ/kg (21.38 Btu/lb). The difference in enthalpy between these two temperatures is 41.42 kJ/kg (17.85 Btu/lb). The total refrigeration effect of liquid entering an expansion valve at 4.4°C (40°F) and vaporizing at −6.6°C (20°F) is 197 kJ/kg (84.96 Btu/lb). The total refrigeration effect of liquid R-22 entering an expansion valve at 38°C (100°F) and vaporizing at −6.6°C (20°F) is 155.77 kJ/kg (67.11 Btu/lb). The difference in refrigerating effect is equal to the difference in enthalpy of the liquid R-22 between the temperatures of 38°C (100°F) and 4.4°C (40°F).

A single-plate ice maker was constructed and tested to prove the effectiveness of the "no penalty" concept. After successful operation of the single-plate machine was achieved, the two-plate machine was designed and built to further test the concept on a size and configuration that could be used directly in a single-family dwelling. This two-plate machine was then tested and found to perform as expected. The actual results are covered in a later section of this report.

2. SUMMARY

2.1 Results and Conclusions

It is possible to design an ice-maker heat pump with a no thermodynamic penalty harvesting system, providing the source of heat to loosen the ice during harvesting is obtained from the reduction of the enthalpy of liquid refrigerant after it has flowed from the condenser. The available heat for harvesting is approximately 41.42 kJ/kg (18 Btu/lb) of refrigerant flowing. It was demonstrated that the output of the heat pump need not be sacrificed to provide heat for harvesting the ice.

No claim is made that the no thermodynamic penalty, hereafter referred to as "no penalty," harvesting system is the best system from an economic standpoint. There is an economic penalty in the form of slightly higher first cost which must be balanced against the better COP.

In this report, the COP is calculated as useful heat output divided by the electrical heat input of the compressor only. The COP of the ice-maker heat pump, when operating as a water chiller under conditions of the tests, varies from 3.4 at 2.8°C (37°F) to 3.8 at 8.1°C (46.5°F) with the two-plate machine. The single-plate machine COP rises to 4.0 with water temperatures of 11°C (52°F).

The COP of the ice-maker heat pump while making ice peaks at a freezing time of 19 to 20 min for the single-plate machine and at 12 to 14 min freezing time for the two-plate machine. In both machines the ice-making COP peaked at about 2.90.

This optimum COP is determined by the balance between heat transfer coefficient on the refrigerant side of the freezing plate and the steadily diminishing heat transfer coefficient on the ice-freezing side of the plate. Not all of the variables were studied in these tests.

The capacity of the machines as chillers increased with increasing temperature of the water flowing over the plates, as would be expected. Increasing the rate of water flow beyond what was required to ensure a continuous "falling film" did not noticeably affect the capacity of the plates as chillers.

The capacity of the machines as ice makers was affected by the length of freezing time. The single-plate machine peaked at 19 to 20 min freezing time, which is the same point at which the COP peaked. The two-plate machine maximum capacity was reached with an 8- to 11-min freezing time, while the COP peaked at 12 to 14 min.

The harvest time in both machines in all conditions of proper operation was accomplished in less than 40 sec. During the harvest cycle, water continued to flow over the freezing plates and aided in reducing harvest times, compared with tests where the water flow was terminated during the harvest.

The machines tested were not optimized designs. It is felt that if optimized compressor designs were combined with optimum plate designs, the COP could be pushed up to 3.5 or perhaps even higher if condensing temperatures could be lowered by the use of air-cooled condensers.

2.2 Recommendations

Further efforts should be directed toward simplifying the refrigerant circuit of the ice-maker heat pumps, especially those that are to be used to heat and cool residential buildings. The development of low-cost freezing plates may make it economically feasible to freeze ice on one side of the plate only. While ice is freezing on one side of the plate, part of the circulating 0°C (32°F) water could be heated in a heat exchanger in which the water flows countercurrent to the liquid refrigerant from the condenser. The water thus heated (while subcooling liquid refrigerant) could be stored and periodically pumped over the side of the freezing plate opposite the ice, melting the ice interface with the plate and harvesting the ice. If the costs of such a system can be brought into line, all the refrigerant valves except the expansion valve could be eliminated, thus improving the serviceability and extending the life of the equipment.

A fundamental study of the ice-freezing plate performance should be undertaken together with a variable displacement compressor to nail down optimum performance conditions. This type of work will not be undertaken by industry until the market for the ice-maker heat pump is well established and industry can see a good pay-back for such expenditures.

3. DEVELOPMENT OF EQUIPMENT

Since no ice-maker heat pumps were on the market when this project was undertaken, it was necessary to develop the equipment from common components that were available. In some cases such as the accumulator-interchanger and refrigerant receiver, some modification was required to provide optimum performance.

The single-plate ice maker was developed first. After its operating characteristics were understood, the two-plate machine was designed to serve as a full-size prototype with the characteristics required of a production machine. The results of both efforts are covered in this report.

3.1 Single-Plate Ice Maker Development

The single-plate machine, shown in Fig. 1, was breadboarded using a Coplematic water-cooled condensing unit that was in laboratory inventory. To this was installed a stainless steel freezing plate, 66 × 137 cm (26 × 54 in.), with a total active area of 1.64 m^2 (17.66 ft^2). The schematic diagram of the original test setup is shown in Fig. 2; Table 1 shows the component specifications of the parts used to make the original machine and its series of changes.

3.1.1 Original test

The first test was undertaken using components A through N, with the length of cycle being determined by the low-pressure cutout switch. When the low-pressure cutout shut the compressor down, a hold-off timer was actuated to hold the compressor off for 1 min and then let it restart. The line leading from receiver E to valve M and downstream from expansion valve N was 9.5-mm-OD (3/8-in.) copper tubing.

Ice did not form at all until the water temperature in the tank fell to 3.3°C (38°F). As ice built up, the suction pressure dropped until

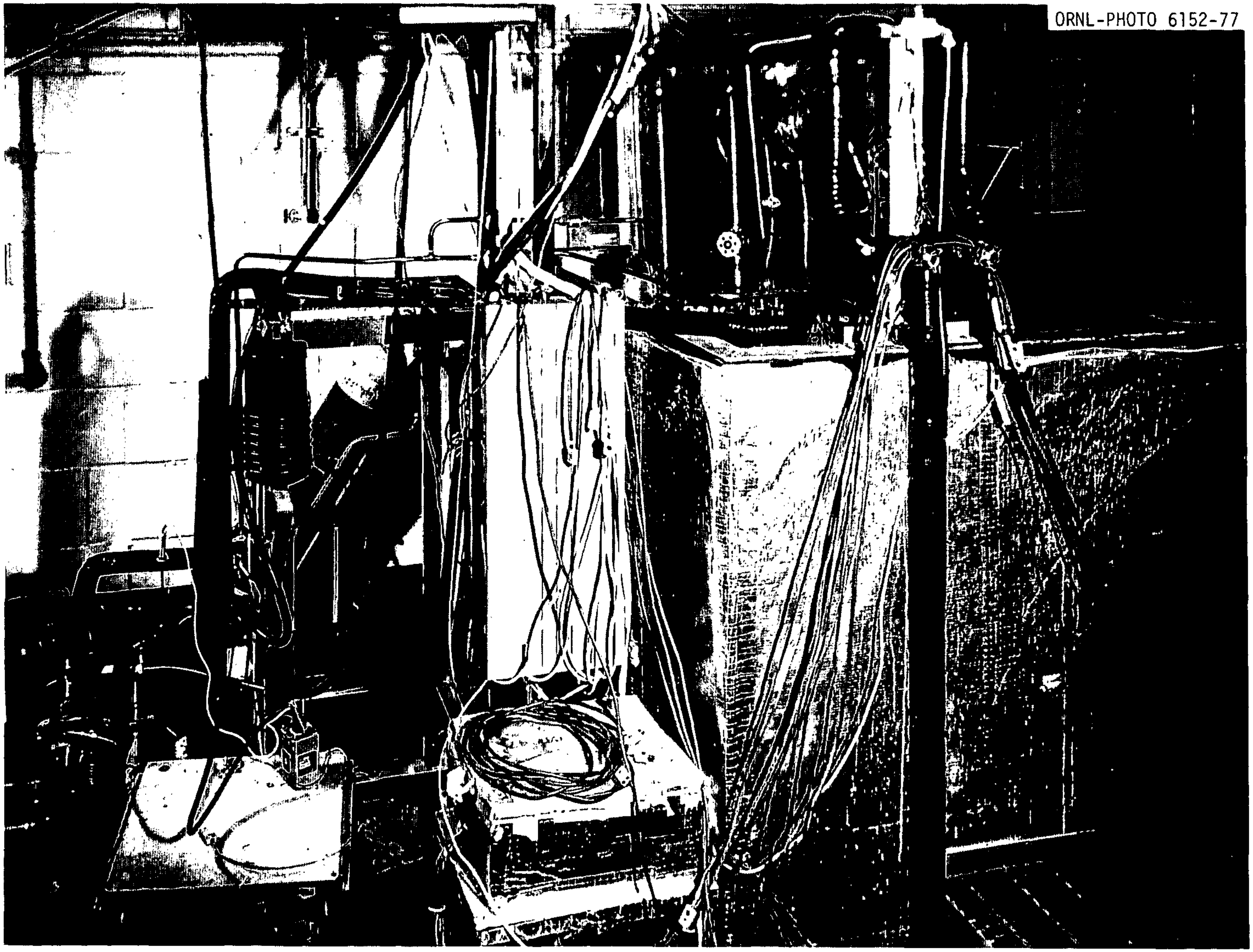

Fig. 1. One-plate ice-maker heat pump.

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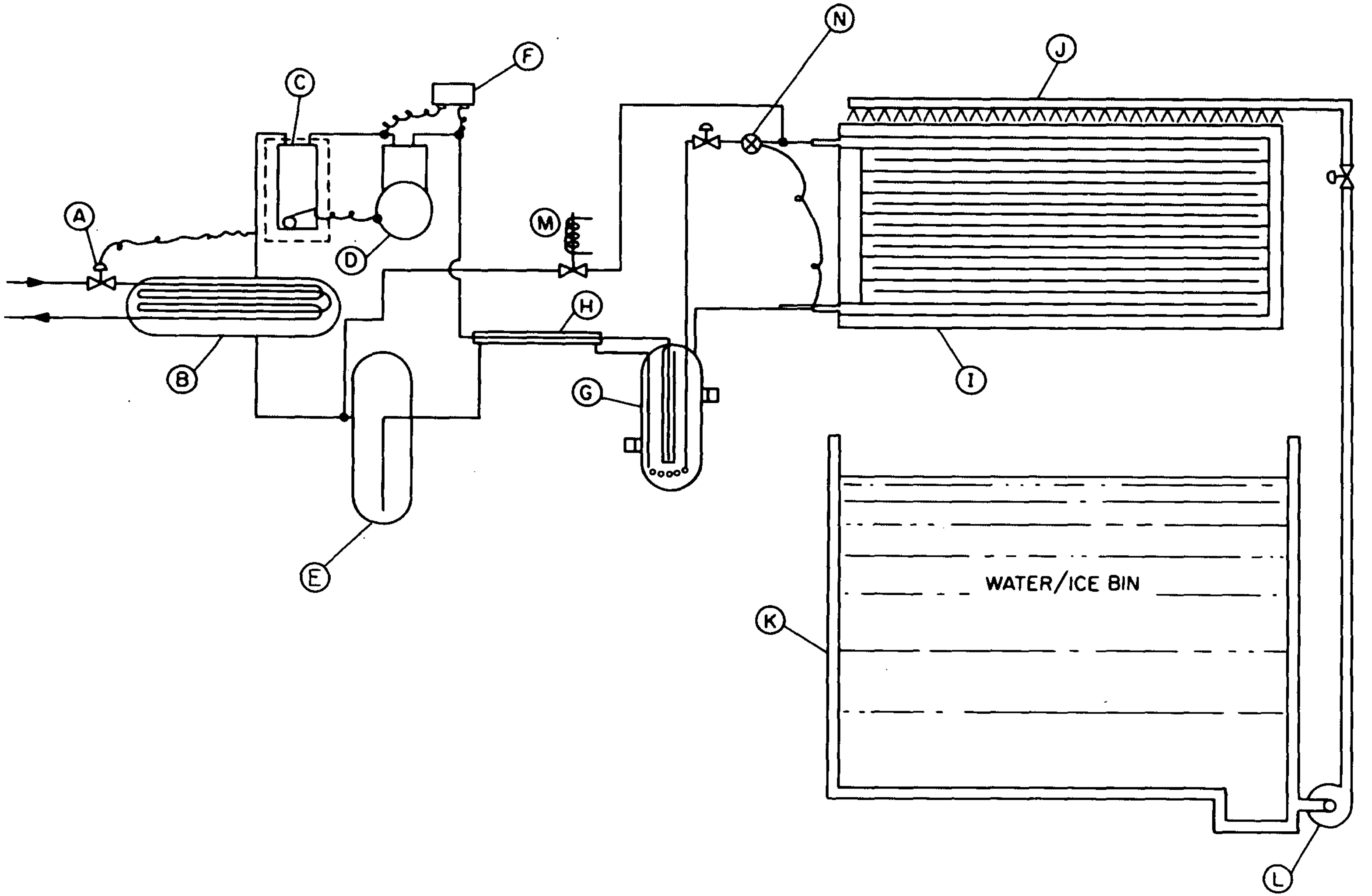

Fig. 2. Original configuration, single-plate ice-maker heat pump.

Table 1. Single-plate ice-maker heat pump component specification

A =	Penn 1/2-in. IPS water regulating valve
B =	Standard cleanable water-cooled condenser
C =	Temprite oil separator, insulated
D =	Coplematic model AAT-1-0150-TAD, 440 V, 3 phase, 1.5 hp
E =	6-in.-diam × 18-in.-high receiver, 18 lb R-22 capacity
F =	Ranco hi-lo pressure switch, hi pressure reset
G =	Refrigeration Research 6 × 10 in. accumulator-interchanger, 0.045-in. oil return orifice
H =	Suction heat exchanger, Heatron part #150-B
I =	Stainless steel "Turbo" freezing plate, 57 × 27 in. overall area, 23 × 53 in. active area, 12-pass, 2-in. weld centers
J =	1-in. pvc manifolds (2), 0.062-in. holes drilled on 1-in. centers, to flood both sides of the plate
K =	Insulated steel tank, 60 × 120 × 54 in. deep
L =	Eastern model DII 115-V centrifugal pump, 160 W
M =	Alco 1/2-in. IPS valve, 7/16-in. orifice, opens when compressor stops
N =	Singer TXV external equalizer and shutoff valve
O =	Sporlan solenoid valve, 3/16-in. orifice, closes when compressor stops
P =	Singer electric expansion valve and thermistor sensor and shutoff valve
Q =	Detroit automatic expansion valve
R =	0.090-in. capillary tube, 120 in. long, and shutoff valve
S =	Kerotest hand valve, 1/2-in.-OD flare
T =	Kerotest hand valve, 1/2-in.-OD flare
U =	1-hp Coplematic compressor model WA21-0100-TAD, 440 V, 3 phase
V =	Sporlan 3/16-in. orifice solenoid valve, closes when compressor stops

269 kPa (39 psig) was reached and the low-pressure switch stopped the compressor. The solenoid valve M opened, and vapor from the top of the receiver flowed to the plate J to cause harvesting, which took place in less than 1 min.

Upon startup, a delay in condenser water flow was noted. It was apparent that not only was the liquid in the receiver being subcooled during the harvest cycle, but the bottom of the condenser was being subcooled because of liquid still in the bottom of the condenser. The cold condenser required heat to warm it up and reduced the output of the heat pump.

During the first test series the water flow over the plate was stopped during harvesting, but this resulted in greatly extending the harvest time. It appears that the water can get behind the ice on the top of the plate, and the hydraulic pressure of the water assists in making the ice let go. The 0°C (32°F) water does condense some of the harvest gas, but the overall effect of the water flow is a plus rather than a minus in the harvest process, since it also helps supply heat to bring the ice up to 0°C (32°F) from its subcooled condition during freezing.

The decision to feed refrigerant to the top of the plate was the right one. Refrigeration Research Company, in their subsequent development work on an ice-maker heat pump, started with an upflow evaporator and had to change to downflow to get rapid harvest. The reason for this is that in downflow, the evaporator is essentially empty, and only a film of liquid refrigerant is flowing down the passes when harvesting starts. The rush of higher pressure gas into the plate at harvest pushes liquid ahead of it into the accumulator. The accumulator-interchanger should have about a 1.14-mm (0.045-in.) oil return orifice in the oil pickup tube, because the accumulator always has some liquid in it much the same as an air-to-air heat pump during the heating season. Large amounts of liquid R-22 must be kept out of the compressor, and so the smaller oil return orifice is preferred.

3.1.2 First change

Before the second test a solenoid valve was installed as shown in Fig. 3. This valve, O, is open when the compressor is running, allowing

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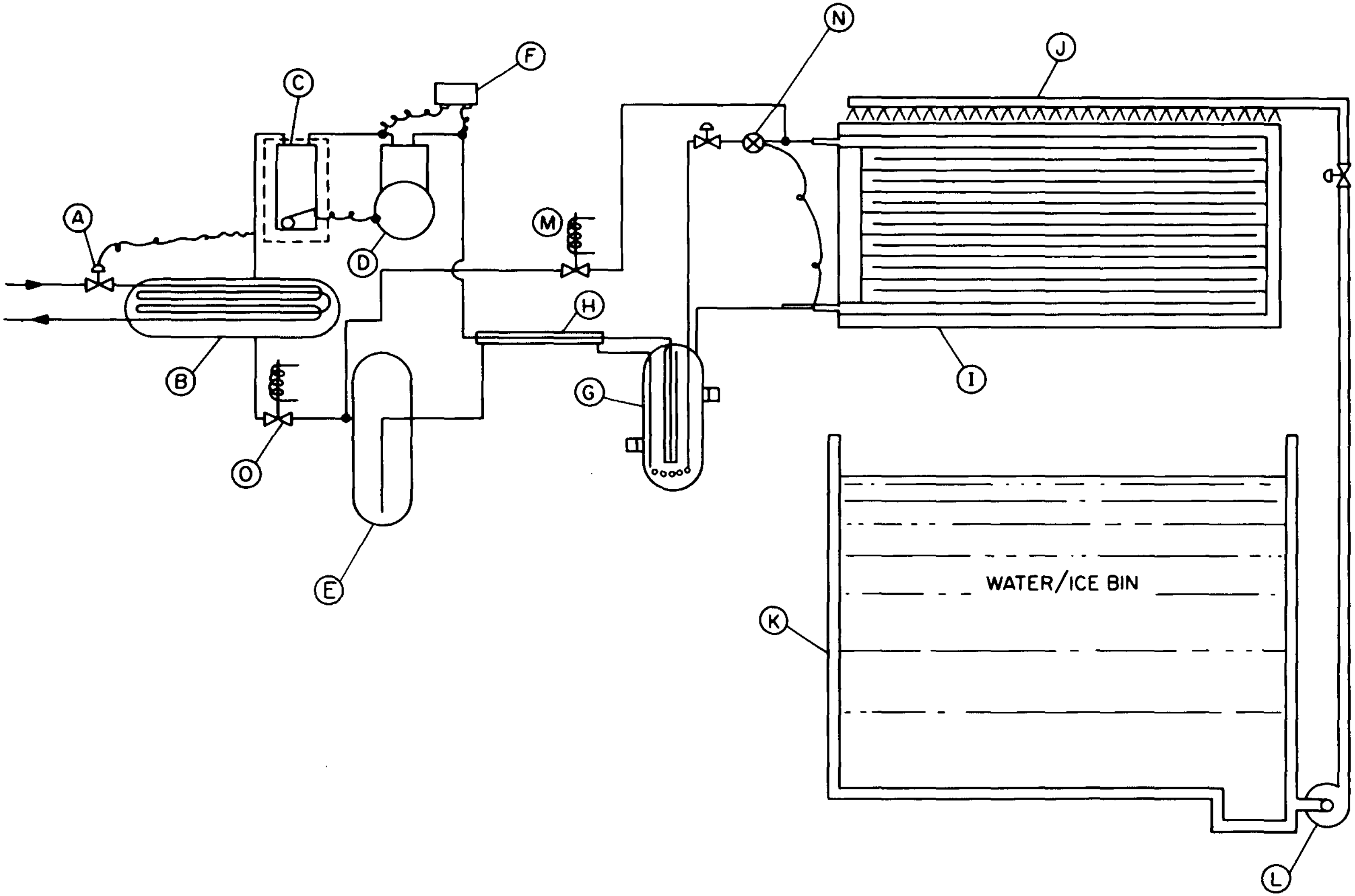

Fig. 3. First modification, single-plate ice-maker heat pump.

liquid refrigerant to flow from the condenser to the receiver. When the compressor stops, the solenoid valve closes, preventing flow of hot gas to the receiver. The receiver is connected to the inside of the cold evaporator plate when solenoid valve M opens. The pressure rise in the evaporator is rapid. If the pressure rises to 483 kPa (70 psig), the harvest will occur in less than 1 min. It was found during this test that the charge of R-22 in the receiver at the time of harvest is critical. At least 5 kg (11 lb) of R-22 must be in the receiver at the time of harvest to complete the harvest in less than 1 min. The enthalpy of the liquid at 38°C (100°F) is 91.14 kJ/kg (39.18 Btu/lb) of liquid before harvest and approximately 49.72 kJ/kg (21.38 Btu/lb) of liquid after harvest. The 41.42-kJ/kg (17.85-Btu/lb) difference in enthalpy is used to vaporize the R-22 that travels to the plate for harvesting. When not enough refrigerant is stored in the receiver, the harvesting is slow or incomplete. This is easily observed because the pressure during harvest does not reach 483 kPa (70 psig), which appears to be critical to obtain a quick harvest. This pressure corresponds to about a 5°C (41°F) R-22 saturation temperature.

Although harvesting took place most times in a normal manner with a regular thermostatic expansion valve, the valve appeared to hunt excessively, and the result was varying cycle times. The moment that 269 kPa (39 psig) was reached, the compressor stopped and harvest started, because the harvest was initiated by the suction pressure switch.

It was decided to try an electric expansion valve to see whether we could get lower superheat and more uniform freezing cycles.

3.1.3 Second change

The test 3 schematic is shown in Fig. 4. The electric expansion valve, P, reduced the superheat at the evaporator outlet, but the hunting persisted in spite of the change in thermistor sensor position on the suction tube and the change from 90- and 45-Ω ratings. The valve was too slow in responding to flooding and starving of the coil. During harvesting, the valve sensor would be flooded by the condensed liquid flowing from the harvesting plate to the accumulator, and the valve would close. Subsequently upon start-up, the plate would be starved, and the suction

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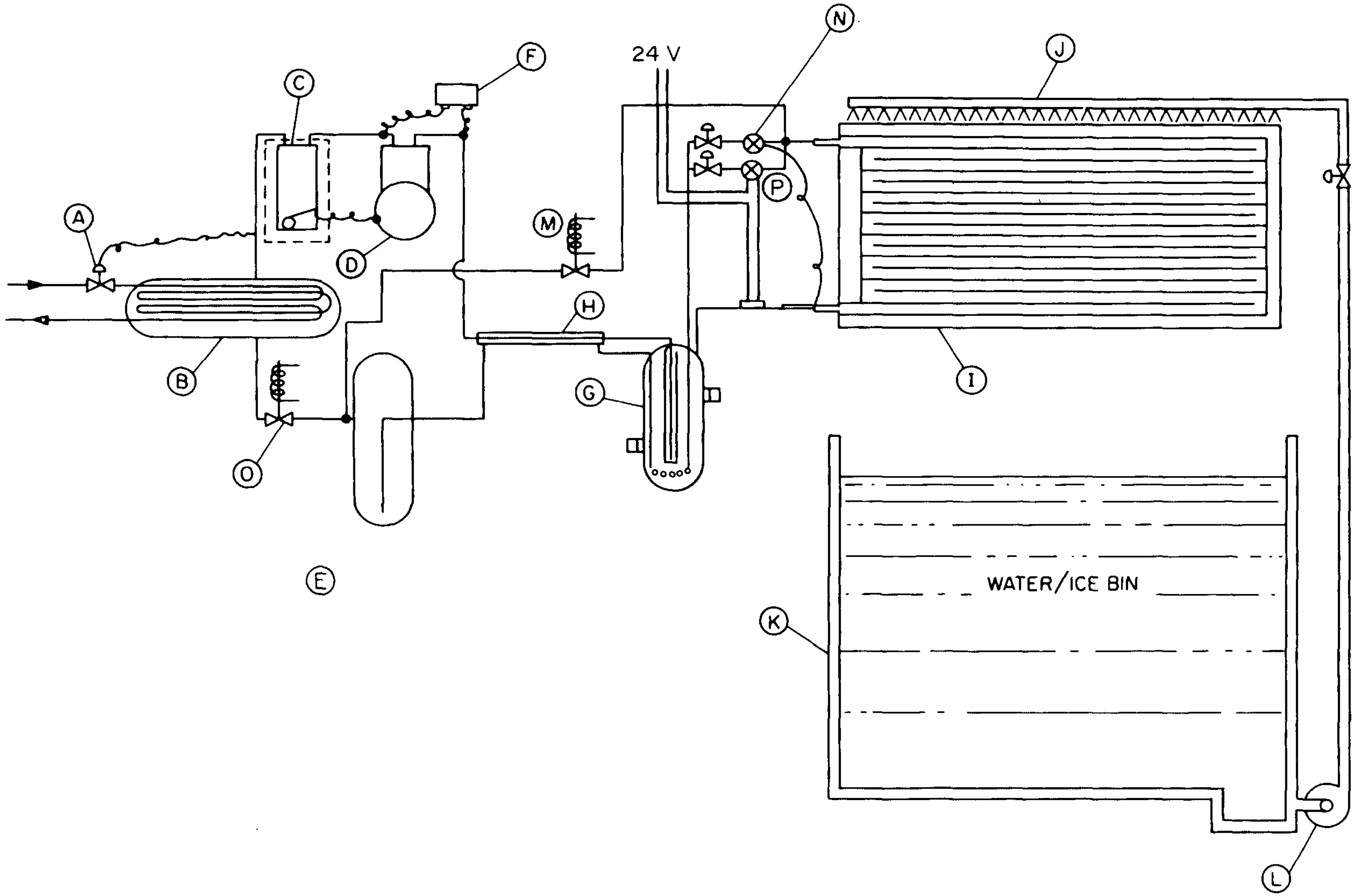

Fig. 4. Second modification, single-plate ice-maker heat pump.

pressure would drop and cause a very short cycle which could not be tolerated. To correct this condition, a relay was added to drive the expansion valve open during harvest so that on startup the coil could be flooded rapidly and the system would not short-cycle.

During this test, it was noted that on overnight shutdown all of the refrigerant would migrate to the lower-temperature evaporation plate and accumulator, and on startup large amounts of liquid refrigerant would enter the compressor. The problem was not attacked at the time because of greater interest in developing a good operating cycle.

The addition of variable resistors in the drive open and sensor circuits was not found to control the problem in all conditions and was dropped. It is felt that if a plate with faster response characteristics such as a roll-bond-aluminum or roll-bond-copper plate were used, one could have made the electric valve work properly and have a very low superheat at the plate outlet.

3.1.4 Third change

The erratic harvesting performance was still the prime problem with the suction pressure initiating the harvest cycle. A timer was installed to control freezing and harvest times. Several changes were made in the harvest gas line to speed up the process and to assure that only vapor and no liquid was fed to the plate during harvest. Figure 5 shows the modified system. The harvest gas was tapped off the top of the receiver to assure that no liquid would be entrained. The harvest gas line was increased in size to 12.7 mm (0.5 in.) in outside diameter, and two large hand valves directed the harvest gas either to the top of the plate as had been done, or to the bottom of the plate, which had been suggested but never tried. The automatic expansion valve and capillary tube were added to the expansion valve options that could be tried.

The first test with the timed cycle used the conventional thermostatic expansion valve and left the harvest gas connection to the top of the plate. The freezing performance was as before, but the harvesting took place in 30 sec, which was better than had been previously possible when the harvest gas line was 9.5 mm (3/8 in.) in outside diameter and the

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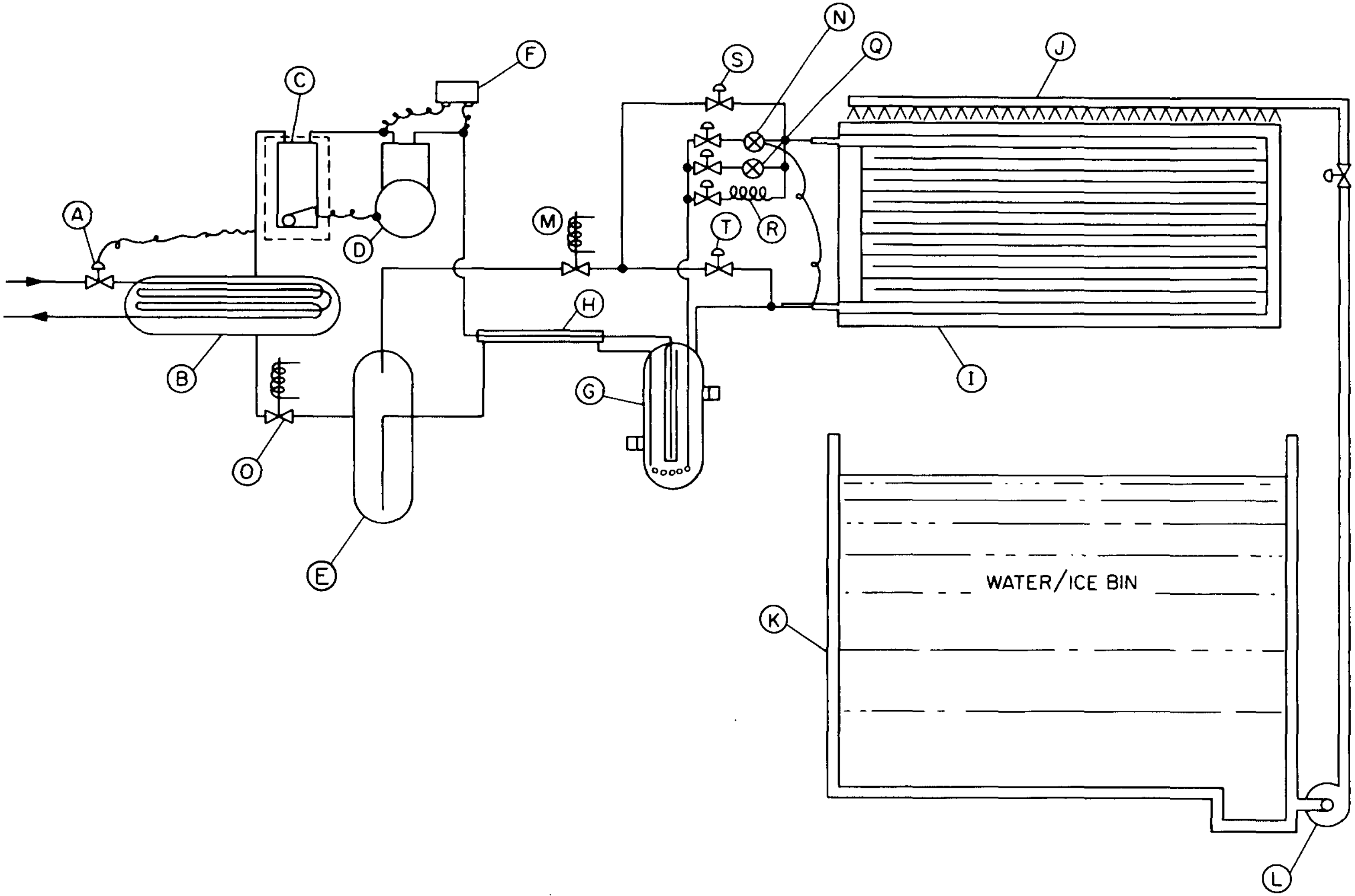

Fig. 5. Third modification, single-plate ice-maker heat pump.

harvest gas connection was tied to the inlet of the receiver. The harvesting was repeatable and rapid as long as sufficient liquid was in the receiver at the time of harvest.

The next test was run with valve S closed and valve T open, to see whether there was any improvement. The results were negative; the harvesting was slowed. Whereas the bottom pass of the plate had always been the last to let go when the harvest gas entered the top of the plate, the top passes were now slow to let go, and the harvest cycle was lengthened to over 1 min.

Next, both valves T and S were opened, but this only made the middle of the plate the last to let go on a long harvest cycle. The net result of these tests was to return to top gas harvesting. The liquid condensed in the plate when the gas entering the top of the plate was pushed by the higher gas pressure into the accumulator, G, where it was observed by sight glass, flowing into the accumulator in substantial quantities during the harvest period and before the compressor restarted.

The automatic expansion valve was put into operation next. Its pressure was set so that the plate became flooded at the end of the 12-min freezing cycle. It worked well but penalized the capacity at the beginning of the freezing cycle and offered no apparent advantage.

The next tests were run with the capillary tube and, after several additions to the length to keep it from flooding the plate too early in the cycle, it performed in a reliable manner, giving hope that in the optimization tests it would look good. The extremely subcooled liquid that flows through it at the beginning of each cycle makes it act more like a thermostatic expansion valve, but without the hunting problem of the thermostatic valve.

3.1.5 Fourth change

Attention was now turned to the problem of slugging at startup. The next day the 1119-W (1-1/2-hp) compressor failed and was replaced by a 746-W (1-hp) compressor. When this compressor, U, was installed, a solenoid valve, V, was installed in the liquid line downstream of the receiver. This prevents the migration of liquid from the receiver to the accumulator after shutdown; the change is shown in Fig. 6. The addition of solenoid V solved the problem of liquid slugging at startup, but by this time there were three solenoid valves in the system instead of one.

3.1.6 Fifth change

The last change, shown in Fig. 7, bypassed the suction line heat exchanger and allowed the liquid from the receiver to pass directly to the accumulator-interchanger. This move produced no ill effects that could be observed.

It is felt that an accumulator-interchanger with much more liquid tubing spaced uniformly up the walls of the cylinder could result in even better capillary tube performance, since the subcooling would be uniformly reduced as the freezing cycle progressed, thus reducing the flow of refrigerant to the evaporator as the ice thickness increases.

Tests were then run to determine capacity and COP of the machine at variable lengths of freezing cycle. Tests were also run at water chilling temperatures above 2.8°C (37°F) to see how the unit performed as a water chiller. Test results will be covered in a later section of this report.

After the completion of the tests where the compressor stopped during each harvest, a series of tests were run where the compressor ran continuously during both freezing and harvest cycles.

In order to do this, solenoid valve O was held open at all times. Since the orifice of this valve is only 4.76 mm (0.1875 in.) in diameter, it can allow the condensed liquid to flow to the receiver with only about 10.34 kPa (1.5 psig) pressure drop between condenser and receiver.

During harvest the flow changes from 100% liquid to a mixture of liquid and hot gas, because the pressure in the receiver is reduced by the flow of "flash gas" to the freezing plate, which is being harvested. Because of the small size of the orifice, most of the gas flowing to the plate comes from the receiver. This is evident because the temperature of the receiver drops markedly during the harvest cycle.

The harvest time remained constant at about 30 sec even though the compressor was drawing off gas at the rate of approximately 1.36 kg (3.0 lb) per minute. The ability of the receiver to produce flash gas

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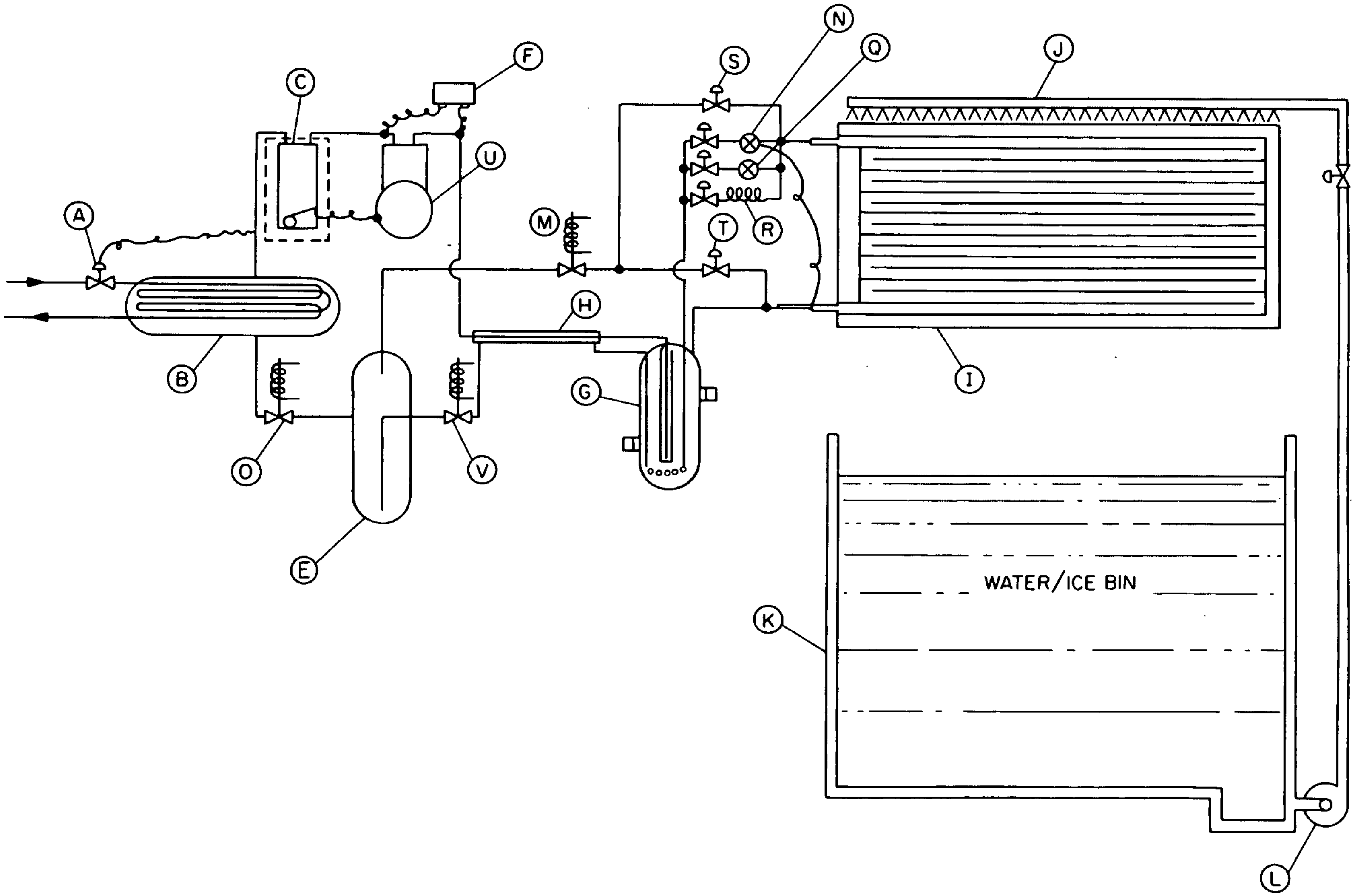

Fig. 6. Fourth modification, single-plate ice-maker heat pump.

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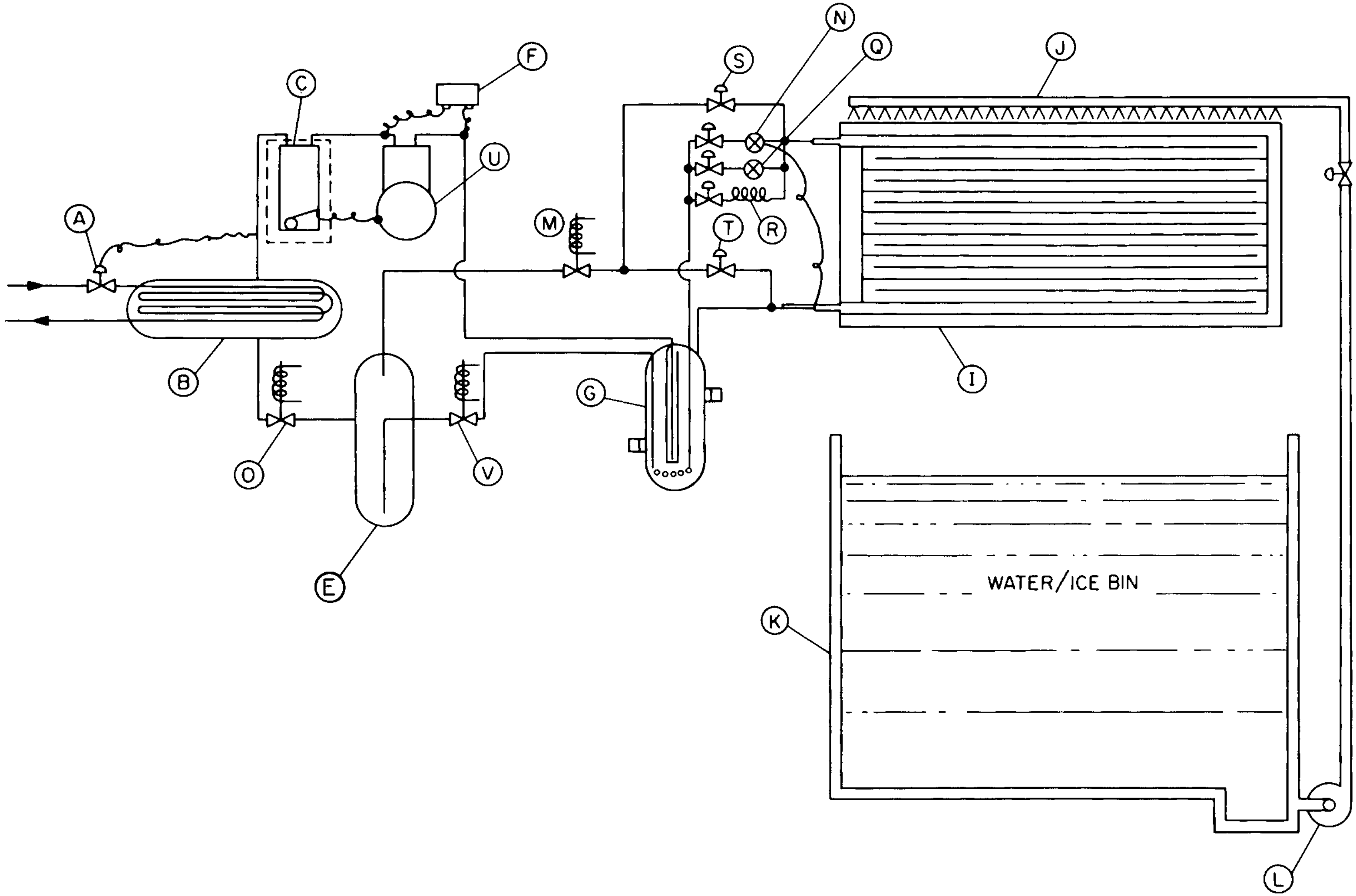

Fig. 7. Fifth modification, single-plate ice-maker heat pump.

is limited only by the size of the vapor line and the solenoid valve orifice [12 mm (15/32 in.) in diameter in this case].

Test results with constant compressor operation are reported in a later section.

3.2 Two-Plate Ice Maker Development

After the single-plate ice maker had been successfully operating for several weeks, plans were made to build a two-plate commercially sized ice maker capable of an output between 7300 and 8800 W (25,000 and 30,000 Btu/hr), using an efficient commercially available hermetic compressor and a new set of freezing plates. A photograph of the two-plate machine is shown in Fig. 8.

3.2.1 Original machine

As originally built, the two-plate unit conformed to the schematic diagram shown in Fig. 9. The component specifications for the two-plate machine are shown in Table 2.

As originally assembled, the machine froze ice satisfactorily but failed to harvest the plates completely in 45 sec. The ice on the lower third of the plate would not fall.

The trouble was felt to be due to the inability to drain the liquid refrigerant fast enough from the plate being harvested. It was also evident by watching the refrigerant flow through the sight glasses that check valves had to be added to prevent flow of hot gas from the harvesting plate to the freezing plate during the harvest cycle.

3.2.2 First change

The capillary tubes (N) were removed and replaced with 3/16-in.-orifice needle valves (Q), sight glasses (M), and check valves (K) as shown in Fig. 10.

Operation of the machine after changes were made was much improved. The needle valves were adjusted to the wide-open position in order to drain the plate fast enough to allow complete harvesting in 30 sec.

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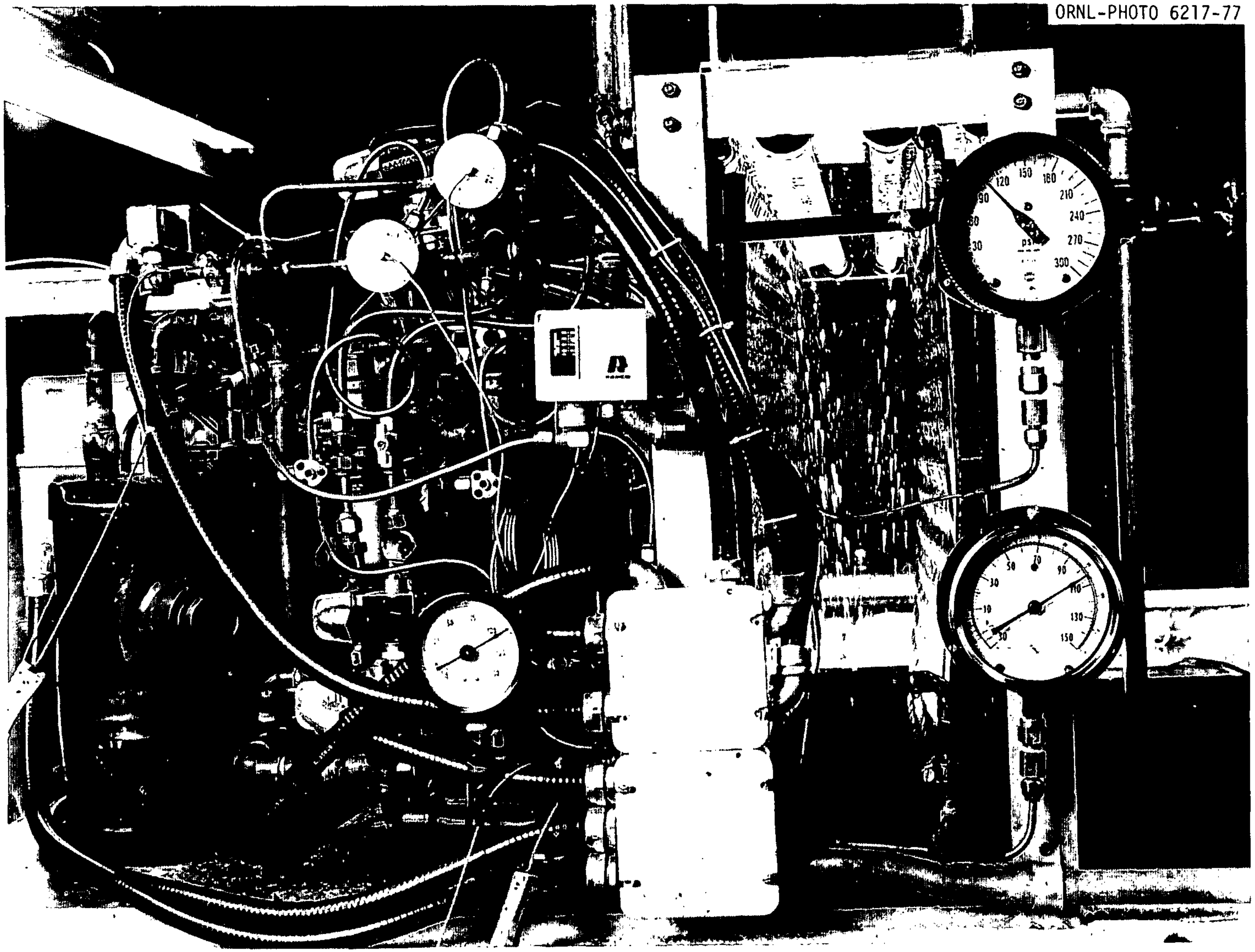

Fig. 8. Two-plate ice-maker heat pump.

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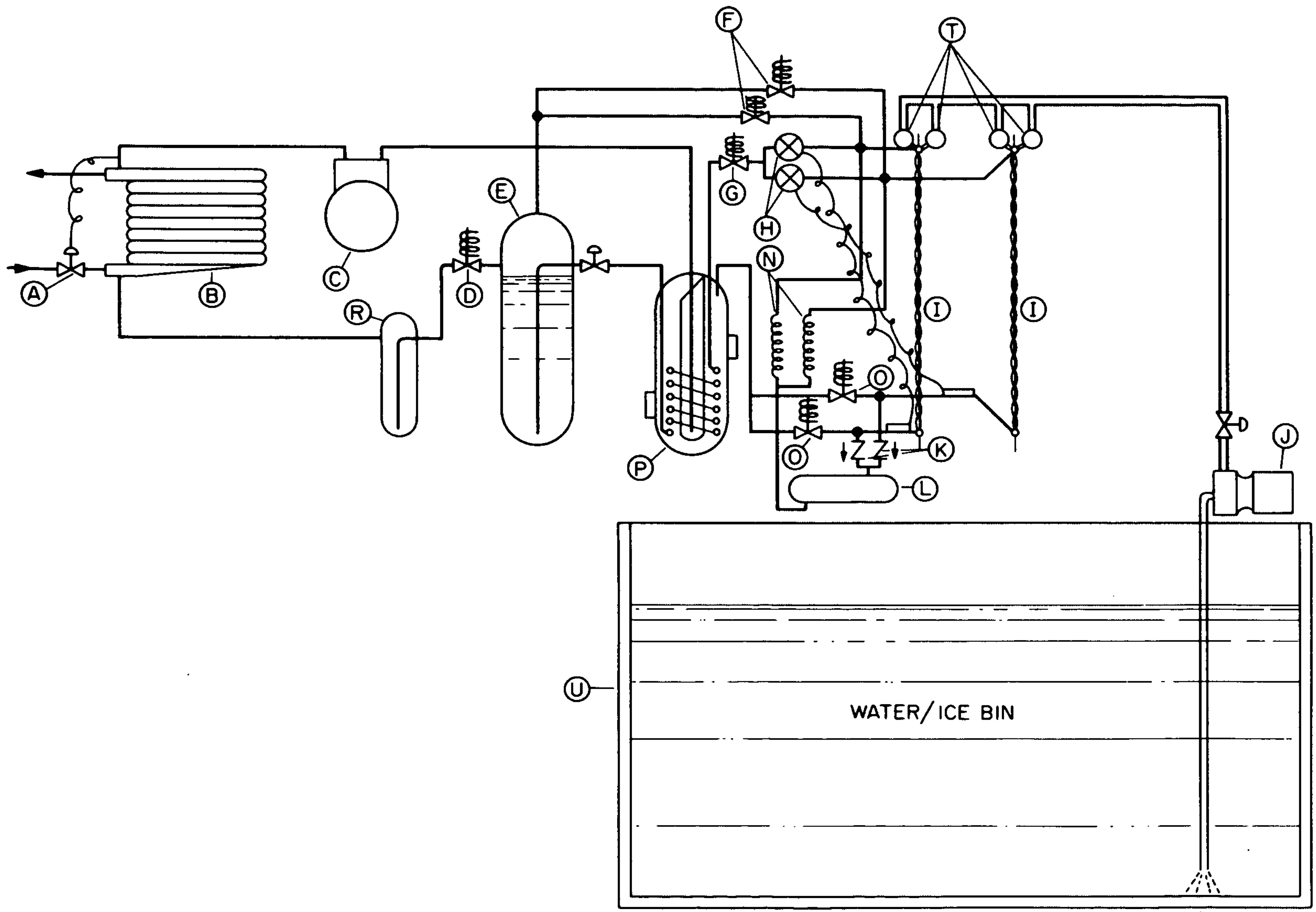

Fig. 9. Original configuration, two-plate ice-maker heat pump.

Table 2. Two-plate ice-maker heat pump component specification

(A) =	Penn 1/2-in. IPS water regulating valve
(B) =	Edwards ENG CoB7.5 water-cooled condenser
(C) =	Tecumseh AH8548E hermetic compressor
(D) =	Sporlan B6S1 solenoid valve, 3/16-in. orifice
(E) =	Refrigeration Research top-tapped receiver, 6 in. in diam × 22 in. high
(F) =	Sporlan B10S1 solenoid valves, 5/16-in. orifices, 2 "hot gas"
(G) =	Sporlan B6S1 solenoid valve, 3/16-in. orifice
(H) =	Thermostatic expansion valves, Sporlan SYE1, 1-ton size
(I) =	Steel "Turbo" freezing plates, 26 × 48 in., 18-gauge welds on 1-1/2-in. centers, 30.00 ft^2 total active area
(J) =	Marlow self-priming pump, 1/4 hp, 15 gpm at 15-ft head, 1-in. IPS
(K) =	3/8-in.-OD flare check valves
(L) =	3-in.-diam × 10-in.-long steel tank
(M) =	2 sight glasses
(N) =	Capillary tubes, 96 in. long × 0.090 in. in outside diameter, copper
(O) =	Sporlan solenoid valves OB1452, normally open, 7/16-in.-diam orifice
(P) =	Refrigeration Research accumulator-interchanger, 6 in. in diameter × 20 in. high with 14 ft of 3/8-in.-OD copper exchanger tubing
(Q) =	3/16-in.-orifice needle valves
(R) =	4-in.-diam × 10-in.-high receiver
(S) =	Tecumseh compressor AH8538E (replacement)
(T) =	1-in. IPS pvc water distribution manifolds, 0.093-in.-diam holes on 1-in. centers
(U) =	1000-gal concrete septic tank, 48 × 60 × 102 in., insulated
(V) =	Same as (P) except additional accumulator volume of 400 $in.^3$ was added

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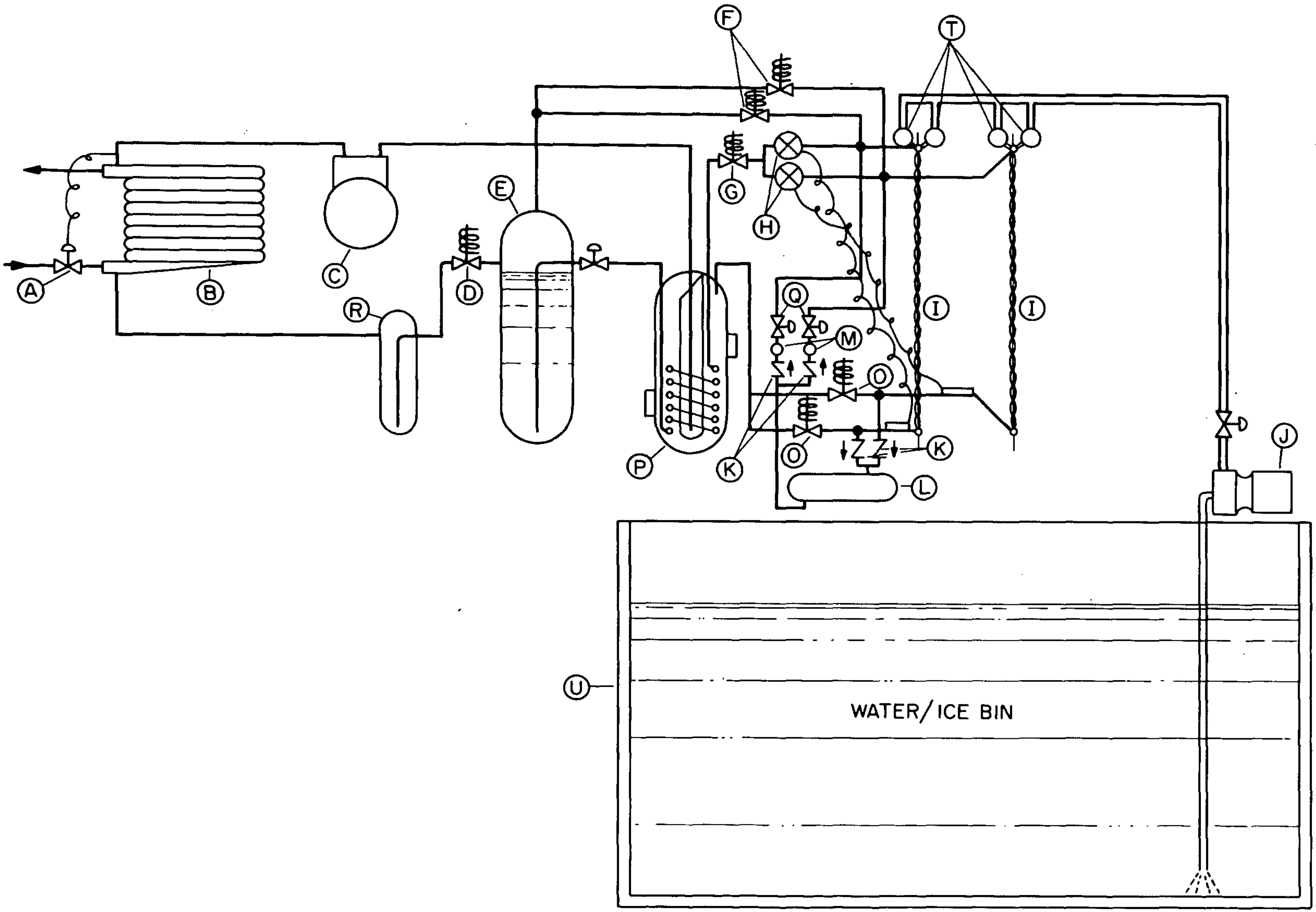

Fig. 10. First modification, two-plate ice-maker heat pump.

The liquid R-22, which is condensed in the harvesting plate, is led through the check valves and needle valves to the plate which is still freezing. The volume of liquid condensed in the harvesting plate is greater than the amount needed to flood the freezing plate. The excess liquid refrigerant passes through the freezing plate and is trapped in the accumulator-interchanger, where it boils away during the next freezing cycle. The amount of refrigerant trapped in the accumulator during the harvest cycle is between 1 and 1.5 kg (2.2 and 3.3 lb).

During the harvest operation both solenoid valves (D) and (G) are closed, thus isolating the refrigerant in the receiver as a source of harvest gas or flash gas. During harvest the freezing plate depends on the harvesting plate for its source of refrigerant liquid.

The system operated satisfactorily and continuously for several days. It was difficult to determine the presence of an overcharge of refrigerant in the system because no sight glass or test cock existed on the receiver.

After a period of several days of operation, the compressor failed and had to be replaced. An examination of the cut-open compressor revealed that a lubrication failure had occurred. This kind of failure is caused either by loss of oil from the crankcase or flood-back of liquid refrigerant to the crankcase which is in turn pumped by the oil pump to the bearings. The liquid R-22 is such a poor lubricant that bearing failures occur. One or both of these may have caused the compressor failure.

3.2.3 Second change

The changes made as a result of the compressor failure were:

1. replacement of the AH8548 compressor with an AH8538E, which has a reduced capacity better suited to the freezing-plate area of the machine;

2. addition of accumulator capacity to prevent liquid refrigerant return to the compressor;

3. addition of oil to the system to compensate for the extralarge refrigerant charge — the oil added was about 2% of the refrigerant charge of 7.2 kg (16 lb).

These changes are reflected in Fig. 11.

During subsequent operation, it became obvious that the receiver volume that can be used for storage of refrigerant cannot be over 75% of the receiver capacity. If this volume is exceeded, a mixture of gas and liquid will travel to the plate being harvested, and the excess liquid will travel to the accumulator, where it will have to be boiled off during the next freezing cycle.

It was found that the system is vulnerable to undercharge of refrigerant and to an overcharge that exceeds the limits of the receiver. Fortunately, the charge has a rather wide, satisfactory operating range and is not critical if the receiver is large enough. There must be sufficient refrigerant in the receiver at the time of harvest to supply flash gas with a heat value of 263 kJ (250 Btu) when harvesting a 1.39-m^2 (15-ft^2) steel plate.

If sufficient refrigerant is not present in the receiver to supply this gas at a pressure corresponding to 4.4°C (40°F), only part of the ice will harvest from the plate during the allotted harvest time. Lengthening the time of harvest will not correct the problem because there is no additional heat made available by this action.

The changes shown in Fig. 11 were the last made before the testing program was undertaken.

4. TEST RESULTS

The original intent of the development of the single- and the double-plate ice maker was to demonstrate the feasibility of freezing and harvesting ice simply and reliably with the smallest possible thermodynamic penalty. The experimental setup does not have an insulated enclosure around the evaporator to reduce heat pickup from ambient air. This makes it impossible to construct an accurate heat balance based on evaporator output.

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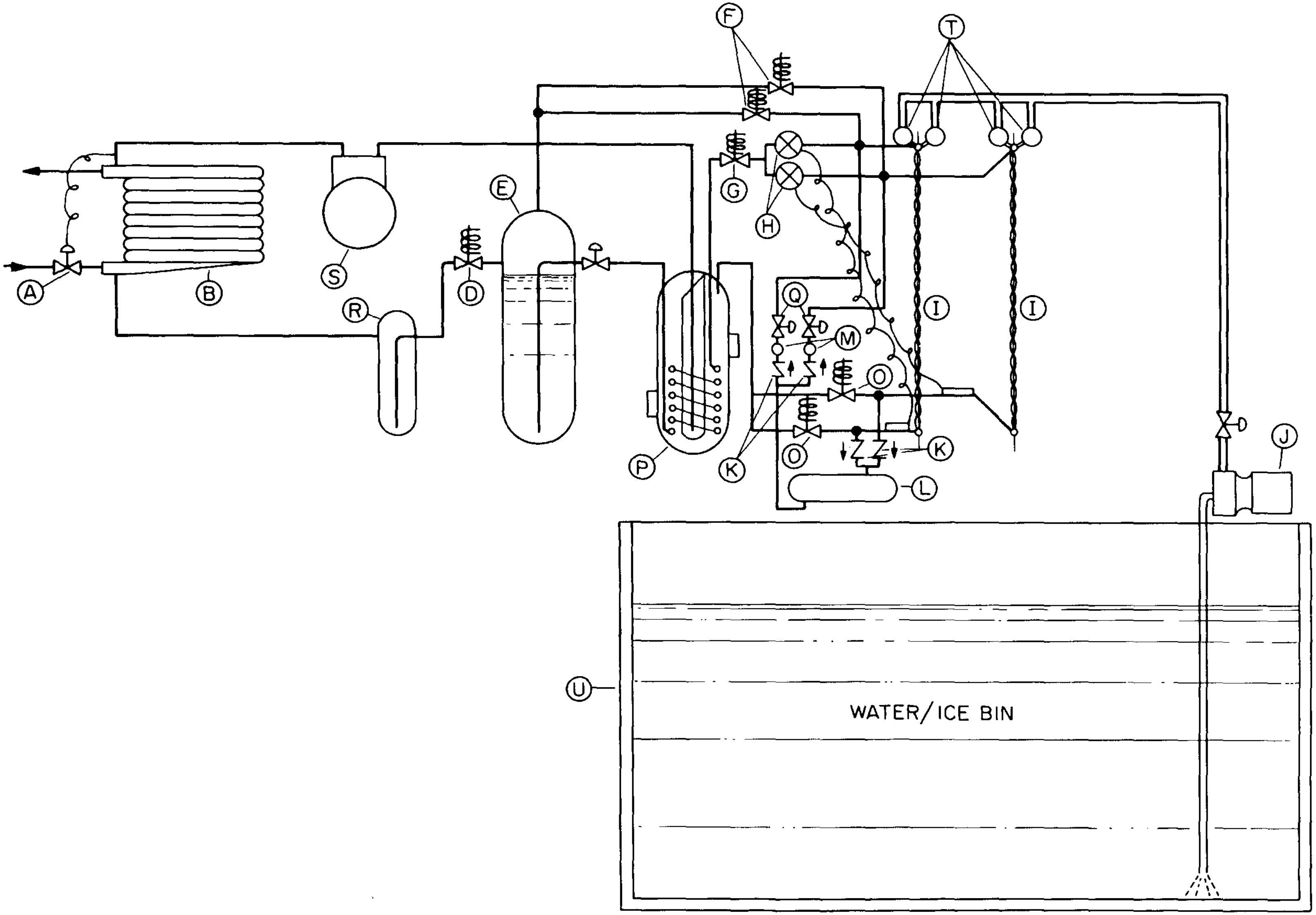

Fig. 11. Second modification, two-plate ice-maker heat pump.

The compressor shell is also exposed to ambient air, and therefore the heat radiated and convected away from the compressor is not measurable as it would have been if an air-cooled condenser had been used and the compressor shell had been located in the same condenser air stream.

David Hart, former design engineer with Janitrol Company of Columbus, Ohio, in private communications with the author, stated that in the development of the Janitrol heat-only heat pump, they were able to measure about 293-W (1000-Btu/hr) increase in output on a 1865-W (2.5-hp) unit by putting the compressor in the indoor air stream as opposed to locating it in the outdoor unit. He stated that they were able to realize an increase in output equal to 10% of the compressor power as a general rule. In the test results being reported, this extra heat output due to having the compressor in the indoor air stream has not been included in calculating the COP.

Pump power was not included in calculating COP because there is no direct relationship between pumping power used in the test setup and in an actual installation. The compressor-only COP can be used in calculation of overall performance of an actual installation once the pump and blower specifications are known.

4.1 One-Plate Ice Maker Test Results

Performance of the one-plate ice maker was determined in both water-chilling and ice-making modes. When operating as a water chiller the compressor operates continuously. When operating as an ice maker the length of the freezing time and the length of the harvest time are determined by timers. The compressor can either be operated continuously or only when the freezing is occurring.

4.1.1 Water-chiller performance

One-plate ice maker performance as a chiller is shown in Fig. 12. As would be expected, the capacity, the evaporator refrigerant temperature, and the COP increase with water temperature.

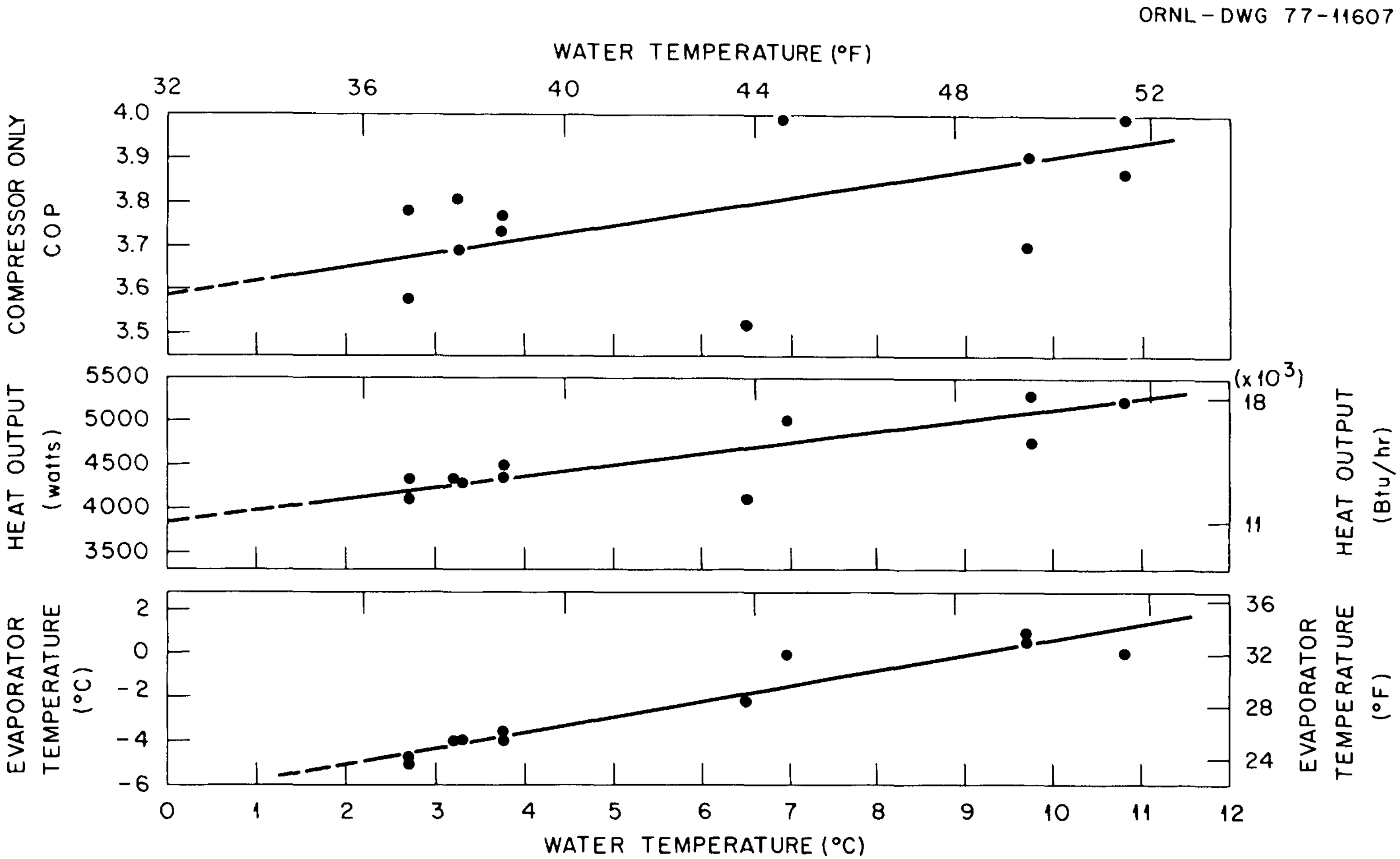

Fig. 12. Single-plate ice-maker heat pump operating to chill water.

4.1.2 Ice-maker performance

One-plate ice maker performance while freezing ice is shown in Fig. 13. The capacity increases as the freezing time increases until it reaches a maximum and then drops off rapidly as time and the thickness of ice increase.

The capacity of the unit is greater when the compressor runs continuously, by about 117 W (400 Btu/hr), than when the compressor stops during each harvest.

The refrigerant temperature in the evaporator decreases with increased freezing time, indicating the increase of ice thickness and increase of resistance to heat flow through the ice.

Figure 14 shows the variation of COP with increasing freezing time. The maximum COP roughly corresponds to the maximum capacity as shown in Fig. 13. The COP for the start-stop operation of the compressor is about 0.07 higher than with the continuous compressor operation, since the compressor operates with a COP of 1.0 during the harvest part of the cycle.

4.2 Two-Plate Ice Maker Performance

Performance of the two-plate ice maker was measured in both water-chilling and ice-making modes. When operating as a water chiller the compressor operates continuously. When operating as an ice maker the compressor operates continuously but the plates harvest alternately in a sequence as follows:

1. both plates freeze (typical time, 7.5 min);

2. one plate harvests while the second plate continues to freeze (typical time, 35 sec);

3. both plates freeze (typical time, 7.5 min);

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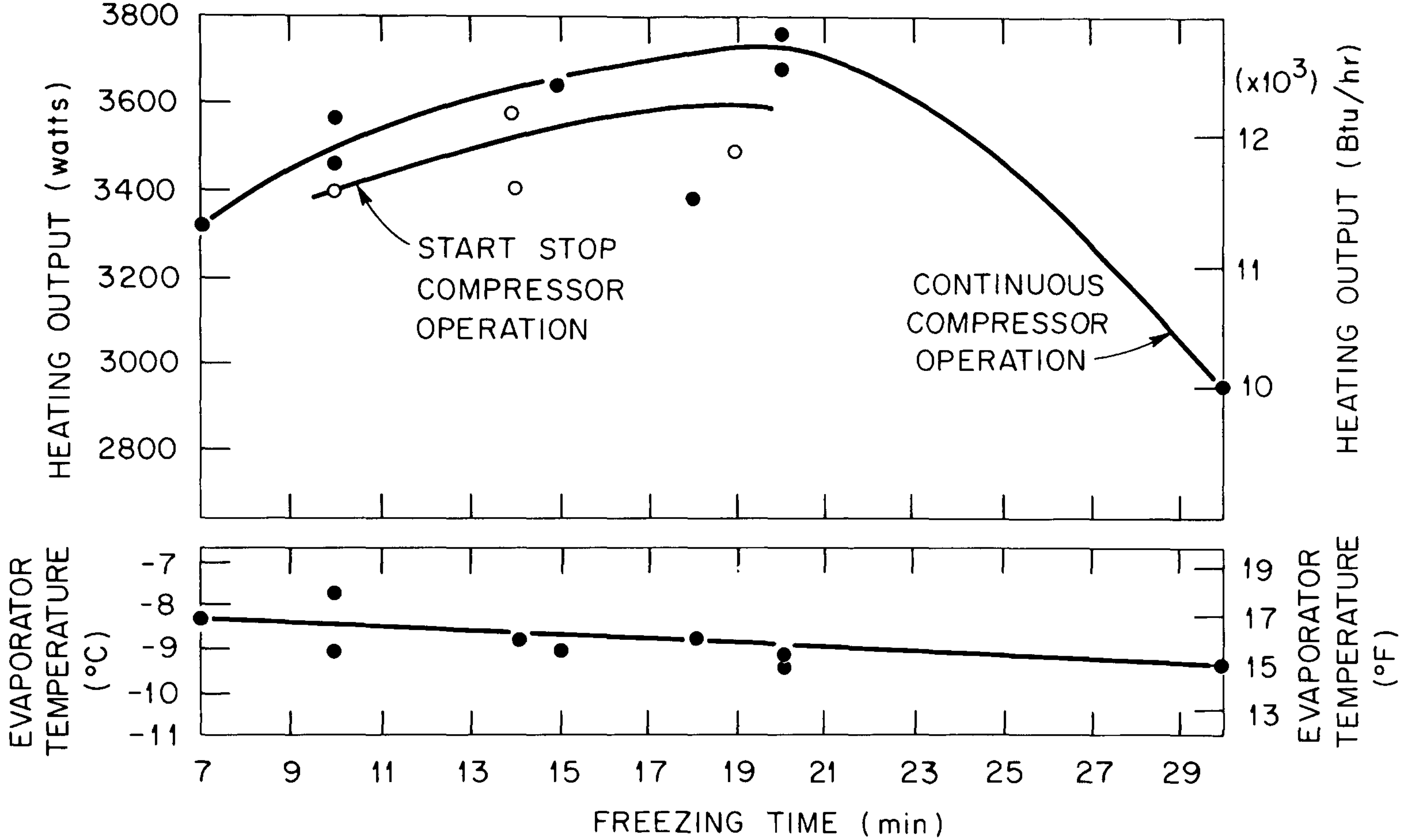

Fig. 13. Single-plate ice-maker heat pump heating output and evaporator temperature vs freezing time.

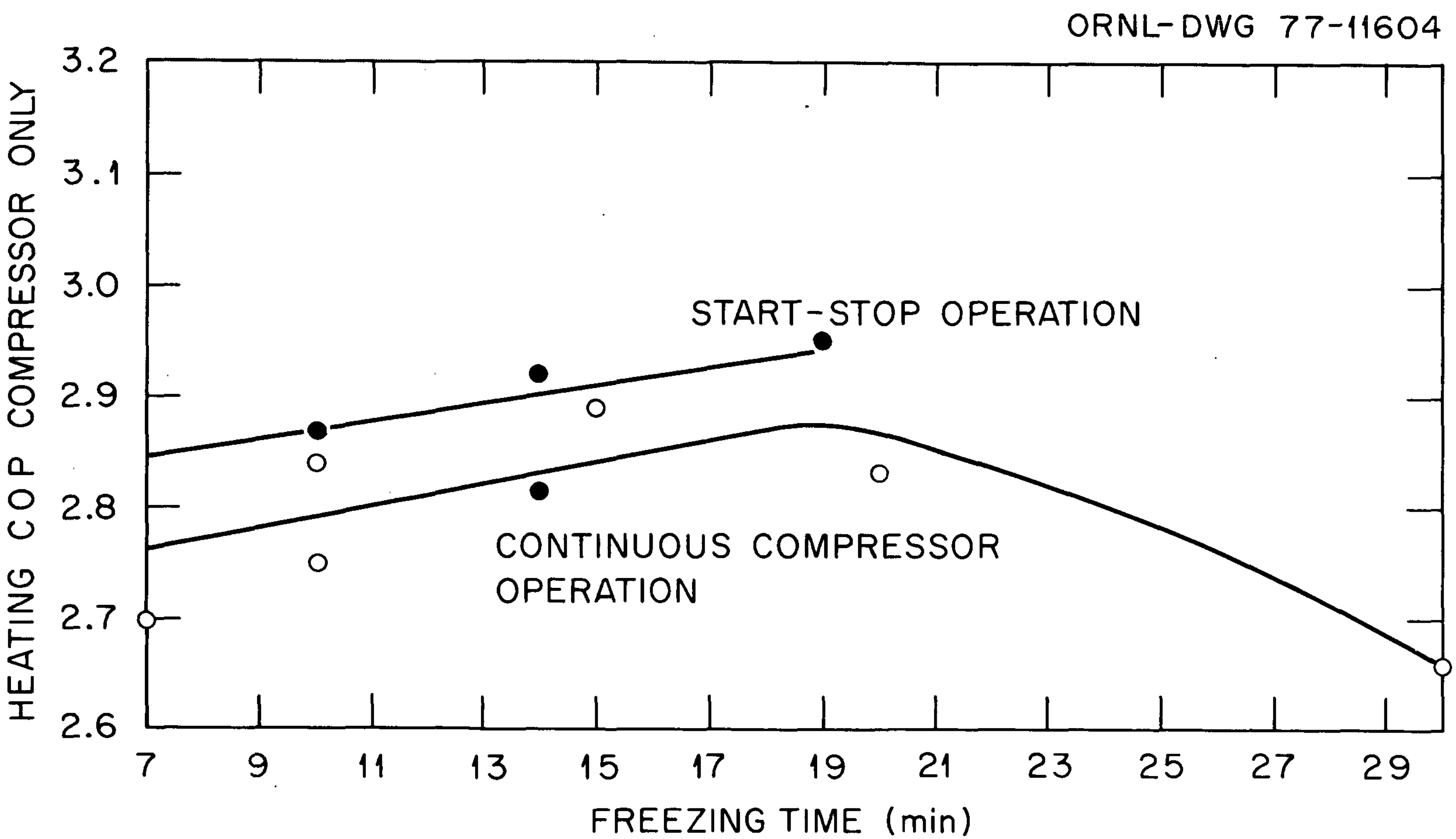

Fig. 14. COP vs time for single-plate ice-maker heat pump.

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 freezing time is determined by a timer, as is the length of harvest time.

4.2.1 Water-chiller performance

Two-plate ice maker performance as a water chiller is shown on Fig. 15. The capacity, evaporator refrigerant temperature, and COP increase with increasing water temperature as would be expected.

4.2.2 Ice-making performance

The two-plate ice maker performance while freezing ice is shown on Fig. 16. The same characteristic as for the single-plate ice maker is observed. The maximum capacity is at about 10 min freezing time and then drops off as the time increases and the ice thickness builds up and increases the resistance to heat transfer. The refrigerant temperature decreases with freezing time and the gradual buildup of the ice on the plates.

The COP results shown on Fig. 16 indicate that a rather flat curve exists within the time limits tested. It is felt that a sharp drop would occur at extended freezing times which would lower the evaporator temperature and force the compressor to supply a greater "lift" to the heat being pumped.

5. ANALYSIS OF RESULTS OF TESTS

In order to guide designers of ice-maker heat pumps, an analysis of the results outlined above will be made. The performance of an ice-maker heat pump is not that of a steady-state machine except when it is chilling water.

Optimum performance depends upon a number of factors which will vary with the design of the machine in question. The factors affecting

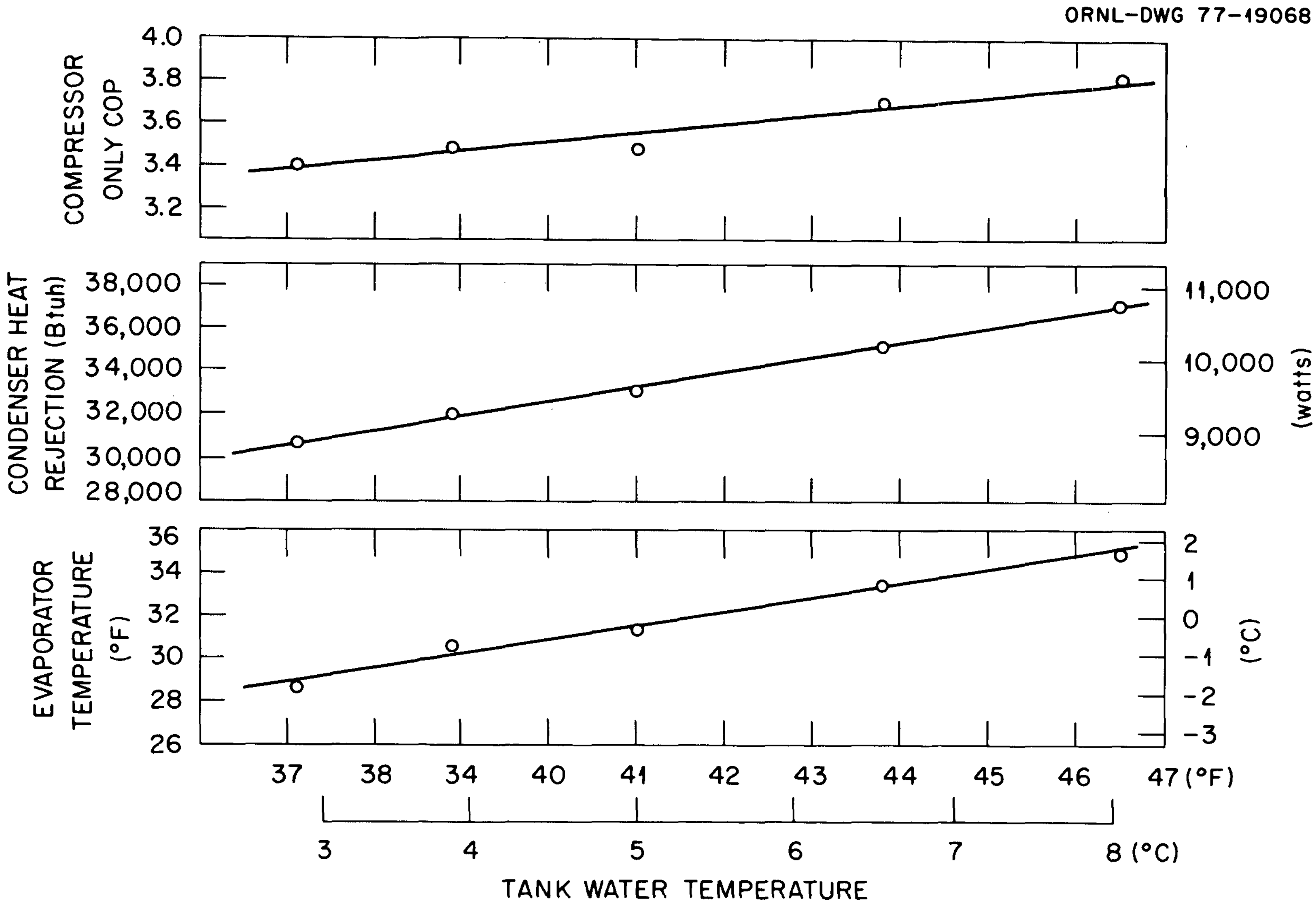

Fig. 15. Two-plate ice maker performance, no harvesting.

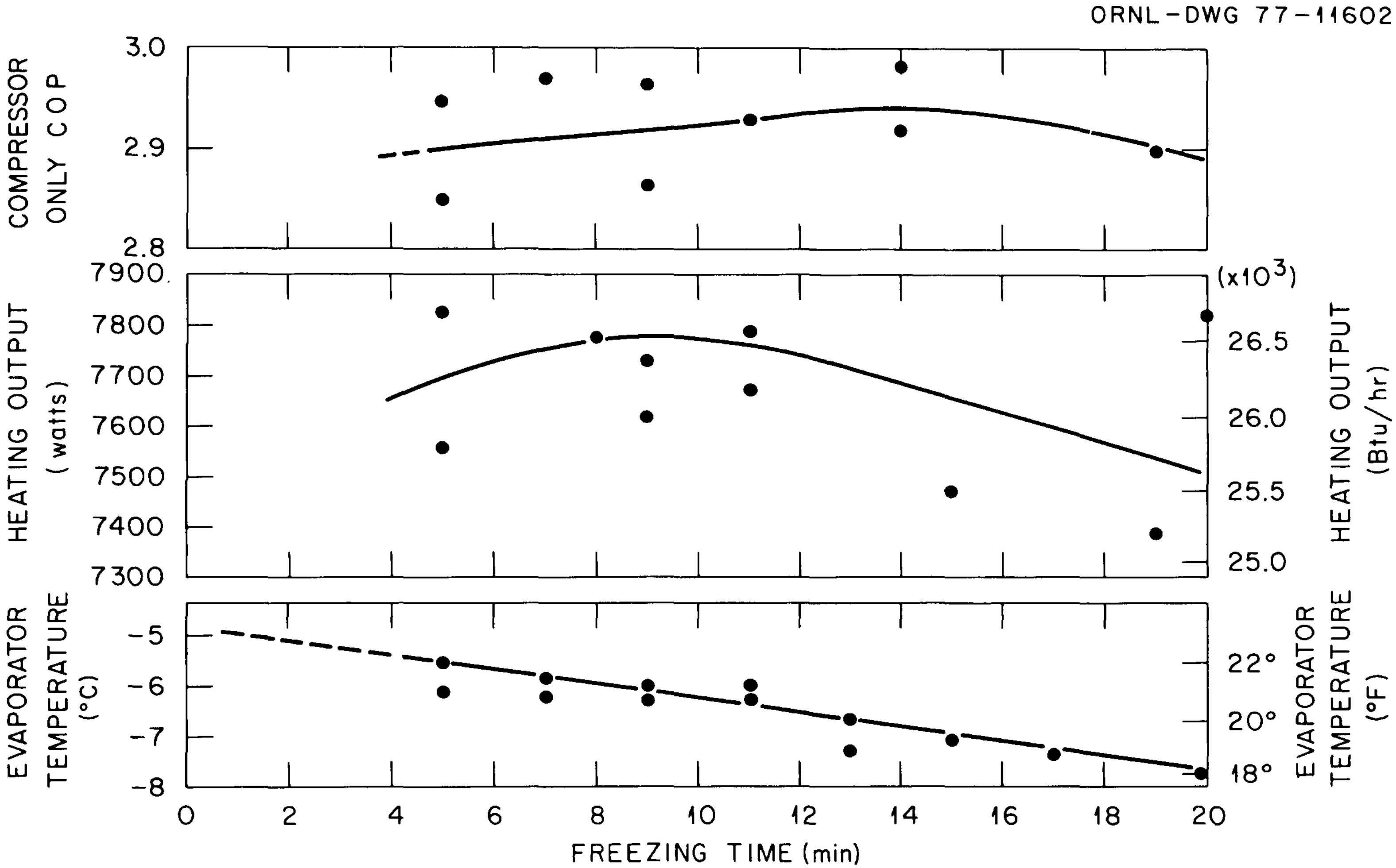

Fig. 16. Two-plate ice maker performance vs freezing time.

the machines tested will be examined, and the general rules affecting future designs will be dealt with.

The single-plate ice maker is equipped with one stainless steel evaporator plate which has a net freezing area of 1.64 m^2 (17.66 ft^2). The welds on this plate are on 51-mm (2-in.) centers. After inflation, the cross-sectional area is larger than the area of the tube connections. The resulting vapor velocity in the plate is not sufficient to keep all of the interior surface wetted with refrigerant. This is evident by examining the ice pattern on the plate. The ice is mostly built up on the lower half of each inflated section with little, if any, ice on the upper half except at the return bends, where the turbulence caused by the 180° turn enhances the heat transfer.

The calculated overall heat transfer coefficient (U) values for the single-plate unit are shown in Fig. 17, together with the thickness of ice built up as freezing progresses and the U value of the ice only as its thickness increases with time of freezing.

It can be seen that the overall U is quite constant until approximately 20 min of freezing time, when the decreasing U or coefficient of heat transfer of the accumulated ice begins to depress the overall U value. When the U value of the ice begins to dominate the heat transfer equation, it is time to harvest the ice and start the freezing process over again.

The two-plate ice maker is equipped with two mild steel plates with 2.61 m^2 (28.1 ft^2) of active surface. The welds on these plates are on 38-mm (1-1/2-in.) centers as contrasted to 51-mm (2-in.) centers on the single-plate machine. The refrigerant velocity in the two-plate machine is much greater than that for the single-plate machine so that uniform ice freezing over the entire plate results.

The pressure drop across the plate varies between 20 and 35 kPa (3 and 5 psi), which is too high for best performance. Future plates should have weld centers around 44 mm (1.75 in.) apart. A pressure drop across an evaporator of about 13.8 kPa (2 psi) has been found by personal experience to be the best compromise between pressure drop and velocity and will yield an overall U value for an evaporator that is optimum.

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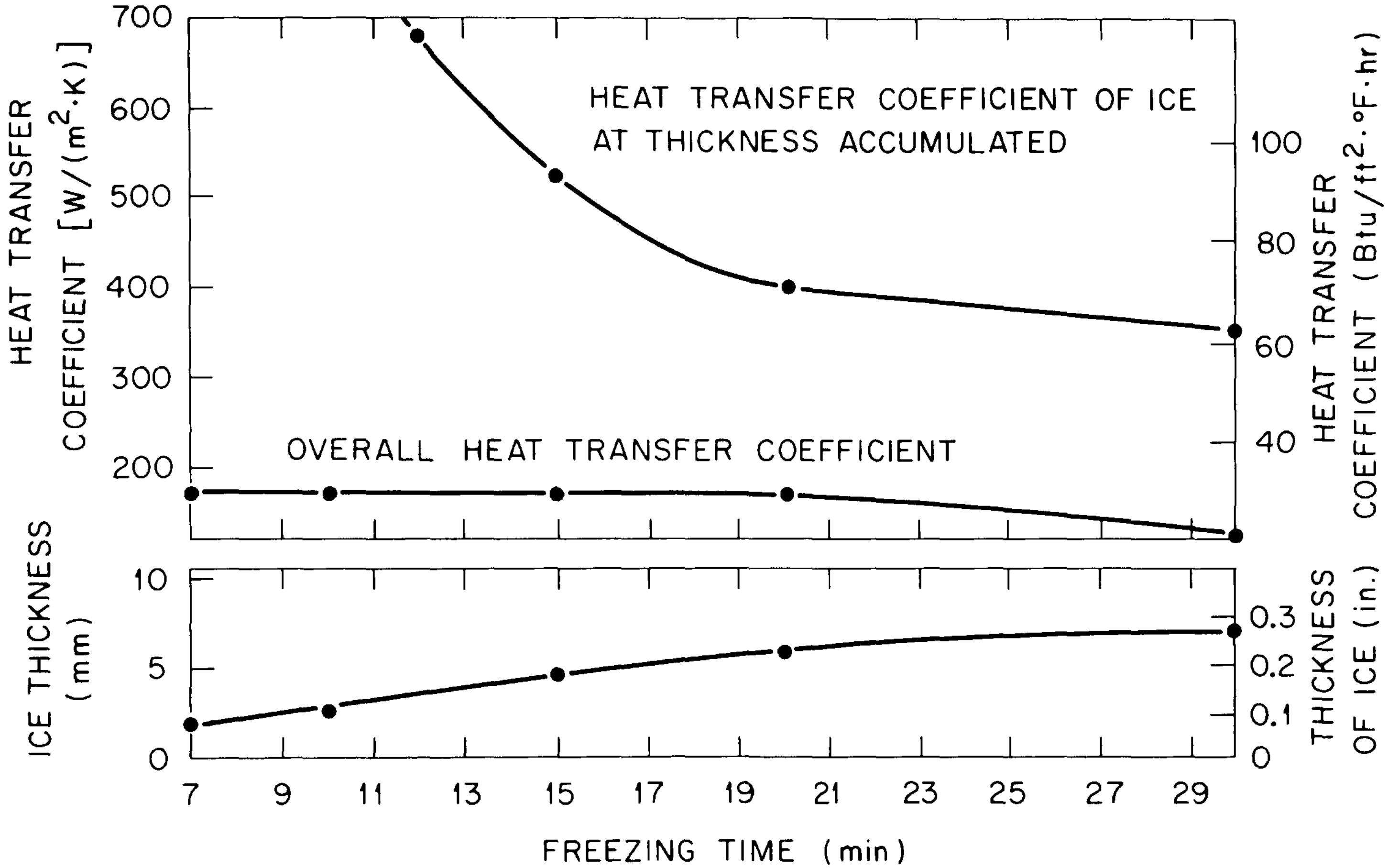

Fig. 17. Single-plate ice-maker heat pump thickness of ice and overall heat transfer coefficient vs freezing time.

The overall U values calculated from the test data are shown in Fig. 18, together with the accumulated thickness of ice with time and the U value for the ice thickness accumulated. The overall U values for the two-plate machine are double those of the single-plate machine. The same factors are at work in the two-plate machine. As the ice thickness builds up, the U of the ice begins to dominate the overall heat transfer mechanism. When that dominant factor becomes evident, it is time to harvest ice and start over. In the case of the two-plate machine, this time appears to be 12 min.

Two factors are of primary importance in obtaining a fast harvest. One is having enough heat stored to warm up the plate; the second is having a means of draining the condensed liquid from the evaporator rapidly so that all surfaces are exposed equally to the higher-temperature harvesting gas.

In the case of the two-plate machine, the weight of the freezing plate is 15.7 kg (34.6 lb). It must be warmed from −6.6°C (20°F) to 1.6°C (35°F) to loosen the ice from the plate.

The heat required to harvest the ice is obtained from the flashing of gas off the top of the stored refrigerant in the receiver. If we know the weight of refrigerant in the receiver and its temperature before and after harvest, the heat supplied to effect the harvest can be calculated.

Several harvest cycles were measured with close to 7.3 kg (16 lb) of refrigerant in the receiver at the beginning of each harvest cycle. This level was determined by the use of a small torch played against the side of the receiver. Condensate forms below the refrigerant level, but not above.

A surface temperature reading was taken before and after harvest to determine the change in temperature of the stored refrigerant. The temperature drop in the receiver averaged about 29°C (52°F). This change in enthalpy is about 37 kJ/kg (15.9 Btu/lb), which, when multiplied by the weight of refrigerant in the receiver before harvest, gives 270 kJ (256 Btu). The accuracy of this method should be approximately ±5%.

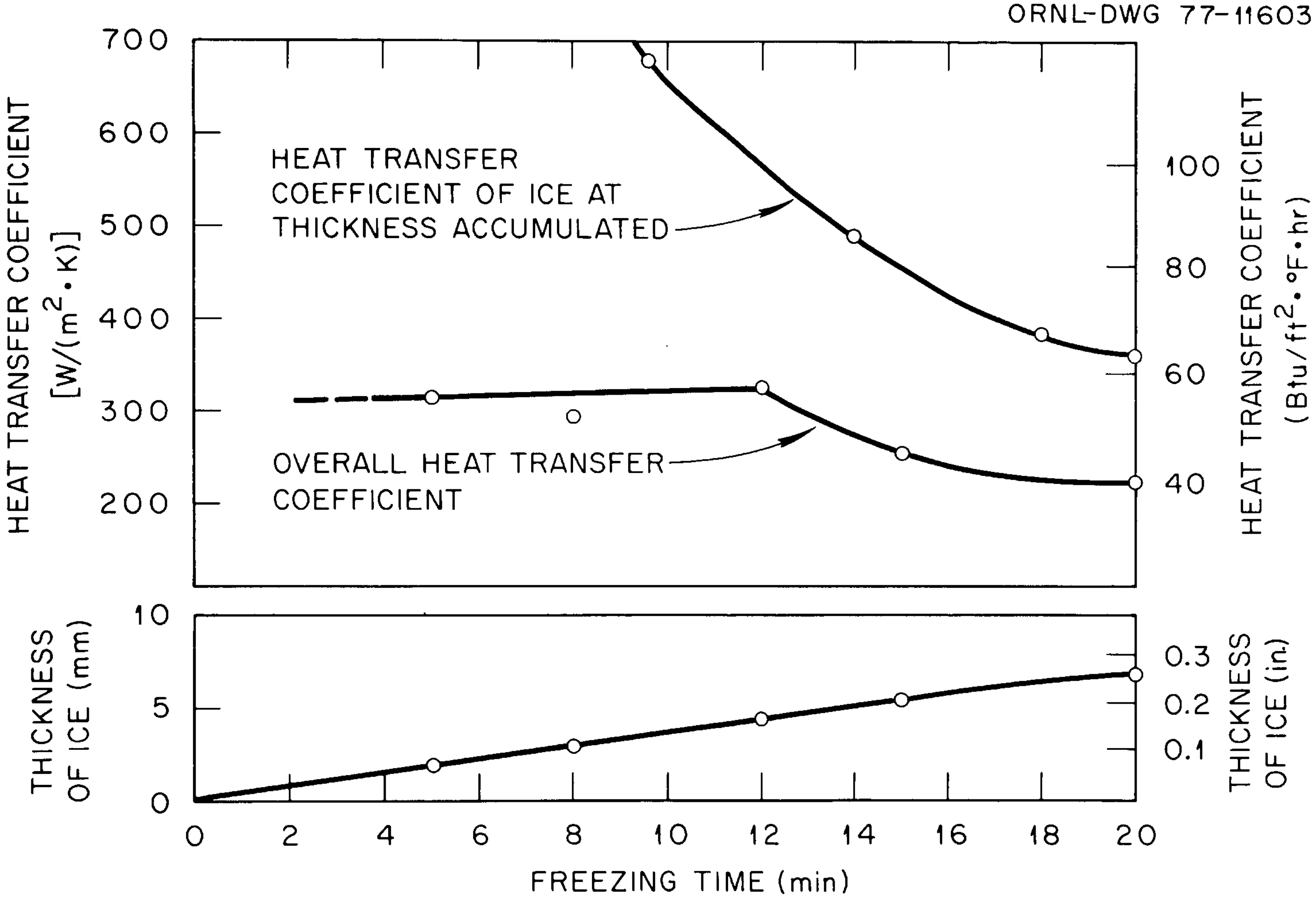

Fig. 18. Two-plate ice-maker heat pump U values vs freezing time.

We have measured the heat required to harvest the plate in 30 sec as 270 kJ (256 Btu). This is four times the heat required to warm the plate. The rest of the heat goes to melting ice and warming ice water which is still flowing over the plate during harvest. The inside surface of the plate acting as a condenser has a heat transfer coefficient of approximately 710 $W/m^2 \cdot °K$ (125 $Btu/hr \cdot ft^2 \cdot °F$). From the *ASHRAE Handbook*,[4] one would expect a 1.3-m^2 (14-ft^2) plate condensing R-22 at about 5°C (41°F) to release 8792 W (30,000 Btu/hr). This rate, applicable for 30 sec, is 264 kJ (250 Btu), which agrees closely with the measured value of 270 kJ (256 Btu) to harvest the plate.

The variation of temperatures as the cycle proceeds is shown graphically on Fig. 19. This graph was constructed from a digital computer printout during a typical freezing test. The lag in the receiver temperature drop is the result of the thermocouple being located on the liquid line leaving the receiver rather than on the receiver itself.

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2. Refrigeration Research, Brighton, Michigan — receivers and accumulators-interchangers

3. Sporlan Valve Company, St. Louis, Missouri — solenoid and expansion va˜ves

4. Singer Control Division, Milwaukee, Wisconsin — expansion valves

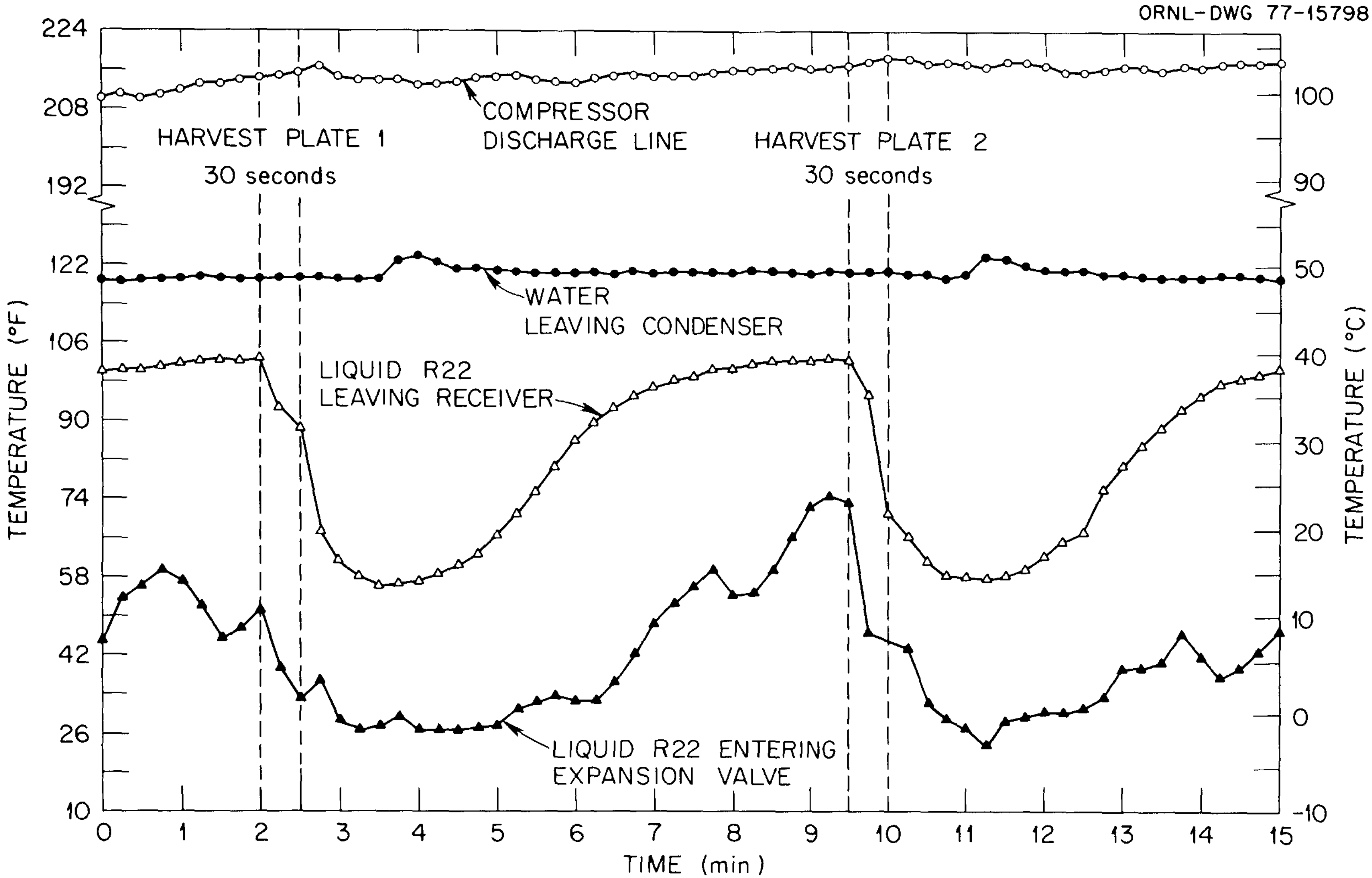

Fig. 19. Two-plate ice-maker heat pump operating with "no penalty" harvesting cycle.

5. Danfoss, Inc., Mahwah, New Jersey — expansion valves

6. Tecumseh Products Company, Tecumseh, Michigan — compressors

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