ICE-MAKER HEAT PUMP HARVESTING SCHEME DEVELOPMENT AND ICE-PACKING DENSITY

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ABSTRACT

Four residential size test unit ice-maker heat pumps (IMHPs) were tested under the Annual Cycle Energy System (ACES) program. Alternative harvesting schemes were evaluated for effectiveness. The ice-packing density of IMHPs was also studied and compared to that of ice manufactured by commercial ice makers and brine-chiller ACES.

Three harvest schemes were tested: hot-gas, stored-refrigerant and dual-fluid off-cycle. The hot-gas scheme tended to penalize excessively the heating output of the system. Stored-refrigerant schemes eliminated that problem but caused compressor failures due to floodback and oil dilution. The dual-fluid schemes exhibited no such problems and demonstrated an ability to harvest during compressor off-cycles. Therefore, it is concluded that dual-fluid off-cycle schemes are the best for use with IMHPs using hermetic compressors.

Ice-packing density measurements indicate a maximum packing factor for plate ice from IMHPs of 0.4 compared to 0.8 for brine-chiller systems. Having the ice distributed evenly in water was found to give better packing factors than allowing it to pile up under the plates. A survey of commercial ice makers indicates that IMHPs producing cylindrical or cubical ice rather than sheets would have packing factors of 0.58 to 0.66 with obvious economic benefits. However, the cost of evaporators for making such ice is likely to offset those benefits.

Larger storage bin requirements make plate-type IMHP systems significantly more expensive than brine-chiller systems in most cases.

INTRODUCTION

The increasing cost of energy has intensified the effort to determine ways to use energy more efficiently, particularly for space heating and cooling. One of the most effective ways to conserve energy in this area is by using the Annual Cycle Energy System (ACES),[1,2] a highly efficient heating and cooling system applicable to both residential and light commercial buildings. Basically a heat pump with low-side storage, the ACES extracts heat from storage (a tank filled with water) to meet building heating and hot water loads. The ice produced as a byproduct of this operation is used to meet all or part of the building's cooling loads.

Ice Formation Methods

The ACES forms ice by one of two primary means. One means is a brine-chiller, brine-to-brine heat pump package employing a heat exchanger submerged in the storage tank such as

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This research was sponsored by the U.S. Department of Energy, Office of Buildings and Community Systems, under contract W-7405-eng-26, with the Union Carbide Corporation.

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is used at the ACES demonstration house.[3,4] Chilled brine is circulated through the heat exchanger and ice forms directly on the tubes. The other means is a plate-type ice maker heat pump (IMHP) which was developed as an alternative to the brine-chiller-type mechanical package. With the IMHP concept, the heat exchanger in the storage tank is eliminated. Ice is formed on refrigerated plates; when the plates are periodically defrosted, ice falls into the bin and floats on the surface of the water.[5,6] This ice, however, does not pack as densely as does that formed by the brine-chiller system, and thus necessitates a larger storage volume. Thus an economic trade-off exists between the in-bin heat exchanger with its more densely packed ice, and the IMHP with its larger bin. This report deals primarily with the IMHP.

IMHP System Studies

Two of the goals in the IMHP development effort have been the ice-harvesting scheme development and an improvement in ice-packing density. This report describes several candidate harvesting schemes and their effects on system performance and reliability. The results of ice-packing studies are also presented.

The ice-harvesting schemes discussed include the hot-gas, stored-refrigerant, and dual-fluid off-cycle methods. The hot-gas scheme[5] uses the output of the compressor directly for harvest, while the stored-refrigerant and dual-fluid schemes use the heat contained in the liquid refrigerant as it exits the condenser. In the stored-refrigerant system, the liquid refrigerant is used directly by flashing it into a vapor, which is sent to the evaporator; there, the vapor condenses and causes the ice to fall into the bin.[5,6] Dual-fluid systems[6] subcool the condensed refrigerant and store the heat in a secondary fluid until it is needed for harvesting, which can be done through a heat pipe arrangement or by pumping the warm fluid through parallel circuits in the evaporator plates.

Test results reported here show the effects of different harvesting schemes on the performance and reliability of IMHPs. The different harvesting schemes are evaluated to identify the best scheme for use with residential size IMHP systems.

Ice-packing density is an important parameter because it determines how large a storage volume will be needed in order to obtain a desired energy storage capacity. Obviously, the greater the packing density, the smaller (and less expensive) the storage volume required. In this report, the effect of ice geometry on packing density is discussed.

TEST UNIT DESCRIPTIONS

Test Unit One

The original configuration of test unit one is shown schematically in Fig. 1. Harvesting was accomplished by diverting the compressor discharge gas directly to the evaporator plates, one at a time. In the configuration shown, the hot gas was admitted to the bottom of the plates. The unit was also tested with the hot gas entering at the top of the plates.

The evaporator of unit one consisted of four copper plates having a total active area of 3.51 m^2. Heating capacity of the unit was about 5900 W at a 41°C condensing temperature and a -5°C evaporating temperature.

Test Unit Two

Unit two is shown schematically in Fig. 2. It employed the stored refrigerant method of harvesting. In order to harvest the ice, valves 1, 2, and one of the suction-line solenoid valves 3 closed. The harvest valve 4, connecting the plate to be harvested with the gas space at the top of receiver 2, opened. Flash boiling of the liquid in receiver 2 forced warm vapor into the top of the evaporator plate where it condensed, thus heating the plate and causing the ice to slide into the storage tank. Condensed liquid was then routed to the other plate where some of it evaporated. The remaining liquid continued on to the accumulator/interchanger where it was boiled off by subcooling liquid during the following freeze cycle.

Unit two was equipped with a slightly larger compressor from the same product line as that of unit one and an evaporator having 2.79 m^2 of active surface. The heating output of the unit was about 7325 W at a 41°C condensing temperature and a -7°C evaporating temperature.

Test Unit Three

When testing was completed on unit two, it was dismantled to make room for unit three (illustrated schematically in Fig. 3). This unit was equipped with a dual-fluid harvesting scheme.

Unit three's evaporator was composed of three copper plates each having seven parallel flow circuits, 25.4 mm on center. Three of the circuits were used for the evaporating refrigerant while the other four were used to circulate a warm 25% ethylene glycol/water solution for harvesting. During the freezing operation, refrigerant exiting the condenser passed through a coil submerged in the harvest brine reservoir, thus subcooling the refrigerant and heating the glycol solution between harvests. When harvesting, the compressor shut off and the refrigerant valve closed. The harvest pump then circulated the warm glycol from the top of the reservoir, through four circuits in each plate, and back into the bottom of the reservoir.

Unit three had a total evaporator area of about 5.57 m^2. Although the compressor and condenser from unit two were used with unit three, it had a slightly higher heating capacity of about 7900 W at 41°C condensing and -7°C evaporating due to its larger evaporator.

Test Unit Four

Unit four was the same as unit one except for its harvesting scheme. Shown schematically in Fig. 4, the unit used all of the components of unit one except the hot-gas lines and the solenoid valves that controlled them. A dual-fluid type harvesting scheme was added.

The harvest scheme used a tank containing about 75 kg of water. During the freezing operation, the water was heated by subcooling the refrigerant. When harvesting was needed, valves 1 and 2 closed and valve 3 opened. With valve 3 open, the liquid refrigerant in the evaporator drained down into the evaporating coil submerged in the harvest reservoir. There, the refrigerant evaporated and the vapor circulated back to the top of the plates and condensed, completing the heat pipe circuit.

Capacity and evaporator size of unit four were the same as for unit one. All of the test units used fully hermetic compressors.

SYSTEM TESTS

The tests discussed in this section were conducted to evaluate the performance of the various harvesting schemes and their effect on overall system performance with the goal of identifying the most attractive scheme for use with IMHPs. Most of the tests were run with the evaporator plates exposed to room-ambient air at about 23.9°C (75 F); however, some testing was done at lower air temperatures to simulate more realistic operating conditions. Condensing temperature, unless otherwise specified, was 41°C.

The heat pump heating capacity was determined by measuring the condenser coolant temperature rise with copper-constantan thermocouples and monitoring the total coolant flow with a water meter. Compressor power consumption was measured with a Lincoln thermal watt converter and/or a watt-hour meter. Refrigerant cycle temperatures and pressures were measured by copper-constantan thermocouples and pressure gauges. These data were taken to check cycle operation and for use in determining evaporator plate loading levels.

All temperatures and the compressor power draw were recorded and averaged by the data acquisition system (DAS) described by Domingorena.[7] The DAS consisted primarily of a Digital Equipment Corporation PDP8/e minicomputer, an integrating digital volt meter, a scanner, and an analog-to-digital converter.

Harvest Scheme Evaluations

Table 1 gives harvest times necessary for test units one, three and four under various operating conditions. For freeze times of up to 20 min, and with the plates exposed to room air, unit two harvest times ranged from 35 sec (water pump running) to 1 min (pump off). The unit was not tested with the plates exposed to cold air.

Hot-Gas Harvest. The hot-gas scheme tested on unit one harvested the plates quickly and thoroughly in laboratory tests averaging 45 to 50 sec per plate. As illustrated in Fig. 5, however, its performance deteriorated markedly when the plates were enclosed in an insulated box. With a 16.7°C (30 F) drop in air temperature, the harvesting time nearly doubled. The effect of an uninsulated hot gas line also caused harvest times to increase significantly. In all cases, the COP* and capacity peaked at a freeze time of 20 min.

Increased harvest time significantly degraded overall system performance as illustrated in Fig. 5.

Hot-gas harvesting (defrosting) is widely used in various applications, including air-to-air heat pumps, commercial refrigerator-freezers, frozen food display cabinets, etc. Although it does have the advantage of proven and available technology, the detrimental effect of hot-gas harvesting on thermal performance makes this scheme unattractive for use with IMHPs.

Stored-Refrigerant Harvest. The stored-refrigerant scheme of unit two harvested the plates in about 35 sec or less, which makes it the fastest scheme tested. In addition, this scheme had little effect on the heating output of the unit because it was the only scheme that required no interruption of compressor operation.

Unit two required a minimum of 7.3 kg extra refrigerant charge to store the energy for defrosting. About 25% of this extra charge was condensed during each harvest. While the unit was in operation, most of this liquid was captured by the system accumulator. During off periods, however, the refrigerant tended to migrate to the coldest spot in the system. When this was the compressor and/or accumulator, the result was massive refrigerant floodback on start-up. In addition, problems with leaking solenoid valves often allowed the stored refrigerant in receiver 2 to migrate to the low side of the system which compounded the floodback problem and caused harvesting failures. Eventually this refrigerant inventory control problem led to failure of the hermetic compressor because of lack of lubrication. This failure and continuous problems with harvest scheme control valves terminated the test program for unit two. Although the stored-refrigerant harvest scheme offers some promise of being the fastest and most efficient, it is not considered attractive for small IMHPs due to hardships imposed on system compressors by excess refrigerant inventory.

It should be mentioned that several commercial size IMHPs, including those at the Reedsburgh School, Cray Research Building, and Detroit Edison Trade School, have used the stored-refrigerant scheme successfully. These systems use semi-hermetic compressors and employ pump-down cycles to avoid compressor floodback.

Dual-Fluid Schemes. Table 1 lists the experimental harvest times for the dual-fluid, off-cycle schemes used by units three and four along with those of unit one. Harvest times for both units were comparable to those of the hot gas scheme with unit four requiring somewhat more time for harvesting than the others. The sharp increase in harvest times of unit four for freeze cycles longer than about 60 min was caused by ice forming around the edges of the plates.

Harvesting for units three and four took more than twice as long when water circulation from bin to plates was interrupted. Similar observations have been made by Fischer.[8]

To harvest the plates it was necessary to turn off the compressor of both units during the harvesting period. However, in most practical applications it is felt that the control logic could be set to initiate the harvest cycle, when necessary, upon normal compressor shutoff or after some maximum compressor on time. This should minimize the impact upon the heating output of such a system.

The dual-fluid schemes have a number of advantages over the other harvesting schemes, including much simpler refrigerant circuits with fewer valves and fittings in the system. In addition, the refrigerant charge requirement is smaller resulting in the practical

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*Refers to the average compressor-only heating COP; i.e.,

$$COP = \frac{\text{average heat output}}{\text{average compressor power input}} .$$

elimination of refrigerant floodback problems caused by the harvesting scheme without requiring a pump-down cycle.

Reliable operation, negligible impact on system performance, and absence of refrigerant floodback and control problems make dual-fluid schemes the most attractive ones for residential IMHP applications. Of the two dual-fluid schemes tested, the dual-circuited plate, pumped-glycol system is recommended because of its shorter harvest times.

Performance Tests

Performance testing was done to determine the performance of these test units as a function of freezing time. The effect of harvesting scheme on system performance is discussed.

Procedure. The procedure followed in the variable freeze cycle length tests was to run for one hour or two cycles per test, whichever was longer. Temperatures and power use rates were recorded each minute by the DAS. Condenser coolant flow was measured with water meters.

Results. Fig. 6 illustrates the performance of the units as a function of freeze cycle length. The upper curve in Fig. 6 clearly shows that the hot-gas harvesting scheme used by unit one imposed significant penalties on the performance of that unit, particularly for short freeze cycles. Unit four experienced no such performance penalty. In fact the off-cycle harvest unit exhibited higher performance levels for all freeze cycle lengths tested. Units two and three are compared in the lower curve, which indicates a distinct performance advantage for unit three. Harvesting did not significantly penalize heating output in either unit. However, unit three had about twice as much evaporator area and therefore only one-half the plate loading level of unit two. Thus, unit three was able to maintain higher performance levels over a wider range of freeze cycle lengths than was unit two, even though the same compressor-condenser combination was used in both units.

Tests run on unit three with plates and bin insulated from the surroundings show the effect of air temperature around the plates on IMHP performance. As illustrated in Fig. 7, for an air temperature of 1.7°C (35 F), average capacity and COP are about 2% lower than the values for an air temperature of 23.9°C (75 F). Plate-loading (the ratio of evaporator heat flow to evaporator surface area) increased by about 2%. Wendschlag's[9] test results indicate that this performance and capacity reduction is caused by increased plate-loading, which induces more rapid ice growth and, therefore, lower evaporating temperatures.

ICE-PACKING DENSITY

The ice-packing density or effective density of ice in gm/cm^3 (lb/ft^3) produced by an IMHP and piled in a bin is an important parameter to be considered in the design of the ice storage bins for ACES or other systems employing low-side thermal storage. Obviously, the greater the packing density of the ice produced, the smaller the storage volume necessary and thus the lower the storage bin cost.

Ice-packing density is conveniently discussed in terms of the packing factor (PF) defined as

$$PF = \frac{V_I}{V_T} ,$$

where

V_I = ice volume in storage tank, and
V_T = wetted volume of water and ice in storage bin.

The relationship of this packing factor to the weight fraction packing factor, PF_W, can be shown to be:[6]

$$PF = \frac{PF_W}{0.917 + PF_W \times 0.083} .$$

where

$$PF_W = \frac{W_I}{W_T} = \frac{\text{weight of ice in bin}}{\text{total weight of ice and water in bin}} .$$

The design packing factor of a brine-chiller coil-in-bin ACES with its cylinders of solid ice is 0.80. For IMHPs, with their free-floating sheets of ice, the packing factor is about 0.40.

This section deals with the results of ice-packing factor studies done on plate-type IMHPs and the brine-chiller system at the ACES demonstration house. The results of a survey of the packing factors of several commercial ice makers are also discussed.

Test Procedure

For the ACES house brine-chiller system, the packing factor was determined by calculating the total weight of the water in the bin and the maximum ice inventory produced. Total water weight was calculated from bin dimensions and inital water depth. The rise in the bin water level was monitored to determine ice inventory.

Two types of tests, bucket-sampling and bulk-sampling, were used to determine packing factors for IMHPs.

Bucket-sampling tests involved filling a bucket with ice produced by the test units. and weighing it. Then, the bucket was filled with 0°C water to the same level as the ice, and weighed again. The packing factor was determined by:

$$PF_w = PF_w' \times SF \ ,$$

and

$$PF = \frac{PF_w}{0.917 + PF_w \times 0.083} \ ,$$

where

PF = effective or volumetric packing factor,

$PF_w' = \frac{\text{ice sample weight}}{\text{total ice and water weight}}$ (apparent weight fraction packing factor),

SF = fraction of sample that was solid ice as determined by calorimeter tests.

The solid-ice fraction (SF) is an indication of the energy density of the ice produced and must be used to modify the apparent packing factor in order to determine the concentration of solid 3.34×10^5 J/kg (144 Btu/lb) ice. It is determined by:

$$SF = \frac{W_{si}}{W_s}$$

and

$$W_{si} = \frac{W_{H_2O}\ (T_{H_2O} - T_m) - W_s(T_m - T_s)}{h_f} \ ,$$

where

W_s = weight of calorimeter ice sample,

W_{si} = weight of solid ice in calorimeter ice sample,

W_{H_2O} = weight of hot water used in calorimeter test,

T_{H_2O} = hot water temperature,

T_s = ice sample temperaure,

T_m = ice and water mixture temperature,

h_f = heat of fusion of ice.

Bulk-sampling involved running one of the test units for a week to fill one of the small laboratory tanks with ice. At the end of the week the volume and maximum thickness of the ice pack were measured. After the ice pack melted, the water level was measured to determine the total weight of ice produced. Packing factors were calculated as follows:

$$PF_B = \frac{W_I}{\rho_I V_B} \text{ (bin-volume-based packing factor),}$$

and

$$PF_I = \frac{W_I}{\rho_I V_I} \text{ (ice-volume-based packing factor),}$$

where

W_I = ice weight,

ρ_I = ice density at 0°C,

V_B = bin volume needed based on maximum ice thickness,

V_I = ice pack volume.

Test Results

Brine-Chiller. Results of two winters of heating operation at the ACES demonstration house indicate that maximum packing factors achieved were 0.77 for 1977-78[10] and 0.87 for 1978-79.[11] Table 2 summarizes the results.

Plate-Type IMHP. Tables 3 and 4 summarize the results of bucket- and bulk-sampling tests performed on the ice generated by the test units. The bucket-sampling method yielded an average effective packing factor of about 0.44 while the ice-volume-based packing factors of the bulk samples averaged about 0.47. Modifying PF_I by the average solid-fraction of 0.848 yields an effective packing factor of 0.40 for the bulk-sampling tests, about 9% less than for the bucket samples. Averaging the results of these two methods yields an effective PF of about 42%.

Bucket samples simulate a bin scheme in which the ice is evenly distributed and floating in a water bath. Excessive piling or mounding of ice beneath the evaporator plates would be avoided through the use of this scheme.

The bulk-sampling technique models the situation in which the storage bin is only partially filled with water, of which about 80% is then frozen. Although the bulk-method packing factor, as based on the ice volume, is not much less than that of the evenly distributed water-ice mixtures, the ice-pack mounding under the plates necessitates a larger amount of freeboard volume in a rectangular bin for containment. As indicated in Table 4 this degrades the packing factor by about 32%, from 0.47 to 0.32. In addition, the ice mounding can lead to harvesting failures if the mound comes in contact with the evaporator plates. The mounding phenomenon is particularly evident in Fig. 8.

The scheme of even distribution of the ice in water, as modeled by the bucket-sampling technique, yields higher effective packing factors and avoids the formation of ice mounds, which require extra freeboard volume. Clearly, this scheme requires less bin volume for the same ice production and is therefore the recommended method for ice storage with IMHP systems.

Survey Results

A survey of various commercial ice-makers was made to determine the effect of different ice geometries (cubes, cylinders, flakes, etc.) on their packing factors. Bucket-sampling test results given in Table 5 indicate that chipped or crushed ice yields no improvement in packing factor. However, the cubical and cylindrical shapes investigated yielded packing factors of 0.58 to 0.66, an average improvement of more than 40% over the plate-type evaporator ice.

This increase in packing factor could possibly reduce storage costs for IMHP-based ACES, making them more competitive with the brine-chiller, coil-in-bin based systems. However, evaporators designed to produce small cubes or cylinders of ice are more complex and more expensive than the flat-plate evaporators, which tends to counteract the beneficial effect of the increased packing factor. As an example, the evaporator plates of test unit three cost about $300. Conversations with distributors of ice making equipment indicate that an evaporator of the same capacity, making cylindrical or cubical ice, would cost from 7 to 10 times as much.

System Economic Impact of Ice Packing Factor

Ice-packing test results indicate that an IMHP-based ACES (or any system involving ice storage) would require at least twice as much bin volume as an equivalent brine-chiller-based system. When system economics are evaluated, the increase in bin cost for plate-type IHMPs outweighs the savings realized from eliminating the in-bin coil.[12] Fig. 9 illustrates this point. For ACES applications, therefore, it is evident that brine-chiller systems will be more economical than plate-type IMHP systems.

CONCLUSIONS

As a result of the IMHP development program several conclusions have been reached about IMHP systems. These are listed here.

1. A dual-fluid harvesting scheme, one which stores energy for harvesting in a secondary fluid medium, is the best scheme tested for IMHPs.
2. Of the two dual-fluid harvesting schemes tested, the dual-circuited evaporator, pumped-glycol system has the shortest defrost times and the simplest refrigeration circuit. It is recommended over the heat pipe system.
3. Arranging the control logic to initiate harvesting during normal off periods will reduce the need to interrupt operation for defrosting purposes.
4. Distributing the ice evenly in the water/ice storage bin by floating it in water yields minimum tank sizes for IMHP systems.
5. Solid cubes or cylinders of ice require smaller tank sizes, but such evaporators are more complex and more expensive and are, therefore, not recommended.
6. Plate-type IMHP-based systems require twice as much bin volume as brine-chiller-based systems for the same energy storage capacity.
7. Brine-chillers have an economic edge over plate-type IMHPs for ACES applications because of their smaller bin volume requirements.

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Table 1. Harvest Times Necessary for Units One, Three and Four

Freeze time (min)	Unit number	Harvest time (min) 23.9°C air		1.7 - 7.2°C air	
		Pump on	Pump off	Pump on	Pump off
10	1[a]	1.35		1.40	
	3	1.50		2.50	
	4	2.50		3.00	
30	1[a]	1.56		2.35	
	3	1.50		3.00	>5.00
	4	2.50		3.50	
60	1[a]	2.70		3.90	
	3	2.00		3.00	>5.00
	4	3.00	7.50	6.00	>10.00
90	4	4.00	8.50	8.00	>10.00

[a]hot gas entry at top of plates

Table 2. Maximum Packing Factors Achieved in ACES House Ice Bin

	Total water inventory (MT)	Maximum ice inventory (MT)	PF_W	PF
1977-78	68.62	51.65[a]	0.75	0.77
1978-79	63.50	54.54	0.86	0.87

[a]Bin not quite filled to capacity

Table 3. Bucket-Sampling Test Results

Sample	Test unit	PF_W'	SF	PF_W	PF
1	1	0.45	0.803	0.36	0.38
2	1	0.47	0.814	0.38	0.40
3	1	0.49	0.876	0.43	0.45
4	1	0.47	0.933	0.44	0.46
5	3	0.51	0.873	0.45	0.47
6	1	0.55	0.795	0.45	0.47
7	1	0.51	0.845	0.43	0.45
8[a]	1	0.71	0.845	0.60	0.62
Averages excluding sample No. 8		0.49	0.848	0.42	0.44

[a]Compacted ice.

Table 4. Bulk-Sampling Test Results

Sample	Test Unit	V_W orig. water vol. (m^3)	V_I ice pack volume (m^3)	V_B bin volume needed (m^3)	Ice weight (kg)	PF_B	PF_I
1	2	1.00	2.03	2.78	850	0.33	0.46
2	2	1.13	2.40	3.28	1028	0.34	0.47
3	2	1.13	2.48	3.20	1012	0.34	0.45
4	1	1.74	4.02	4.91	1803	0.40	0.49
5	1	1.16	2.03	4.13	1016	0.27	0.55
6	2	0.51	0.93	1.52	425	0.31	0.50
7	2	0.65	1.52	2.34	567	0.26	0.41
Averages						0.32	0.47

Table 5. Geometric Effects on Ice-Packing Factors

Ice shape	Sample number	PF_w'	SF	PF_w	PF
Irregular, flat, thin (<3.0 mm) chips	1	0.50	0.81	0.41	0.43
	2	0.53	0.81	0.43	0.45
	3	0.58	0.81	0.47	0.49
Crushed flakes	1	0.50	0.75	0.38	0.40
	2	0.51	0.77	0.39	0.41
25 mm hollow cubes	1	0.35	1.00	0.35	0.37
12.5 mm solid cubes	1	0.64	1.00	0.64	0.66
12.5 mm × 16 mm × 16 mm solid rectangles	1	0.59	1.00	0.59	0.61
25 mm × 38 mm solid cylinders	1	0.59	1.00	0.59	0.61
25 mm × 25 mm solid cylinders	1	0.56	1.00	0.56	0.58
25 mm × 6 mm solid spherical section	1	0.59	1.00	0.59	0.61
Plate-type IMHP ice large corrugated sheets[a]	-	0.49	0.85	0.42	0.44
Brine-chiller ice[b] submerged solid cylinders	-	0.75-0.86		0.75-0.86	0.77-0.87

[a]Averages from Table 4.

[b]Included for comparison.

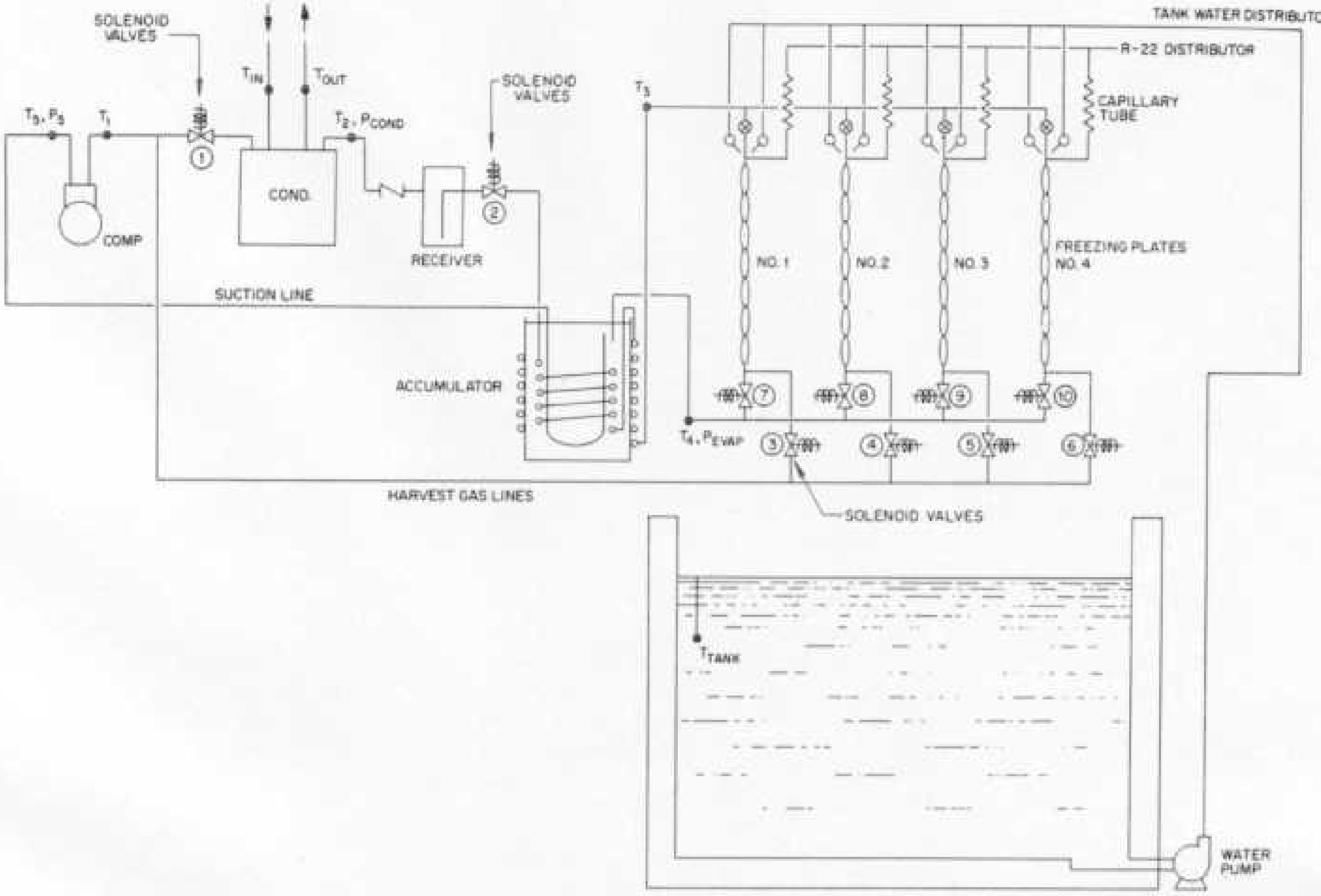

Fig. 1 Unit one schematic diagram

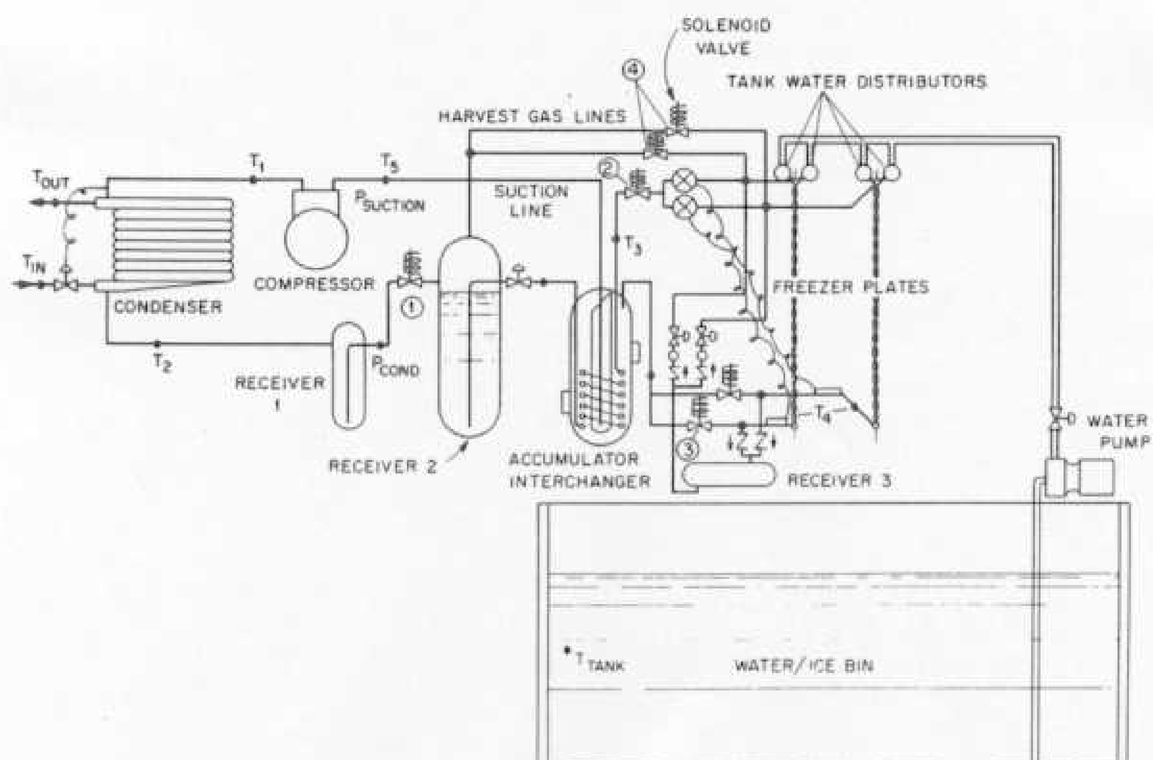

Fig. 2 Unit two schematic diagram

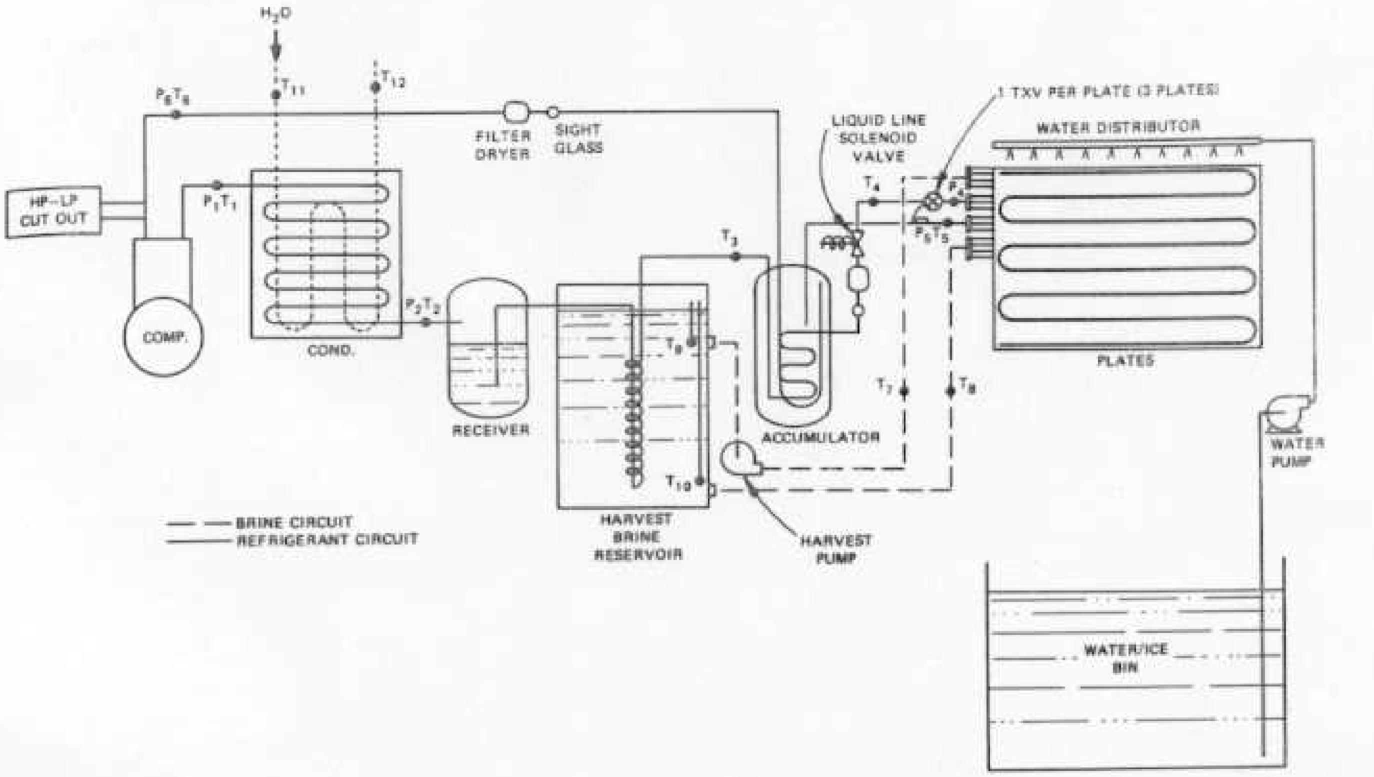

Fig. 3 Unit three schematic diagram

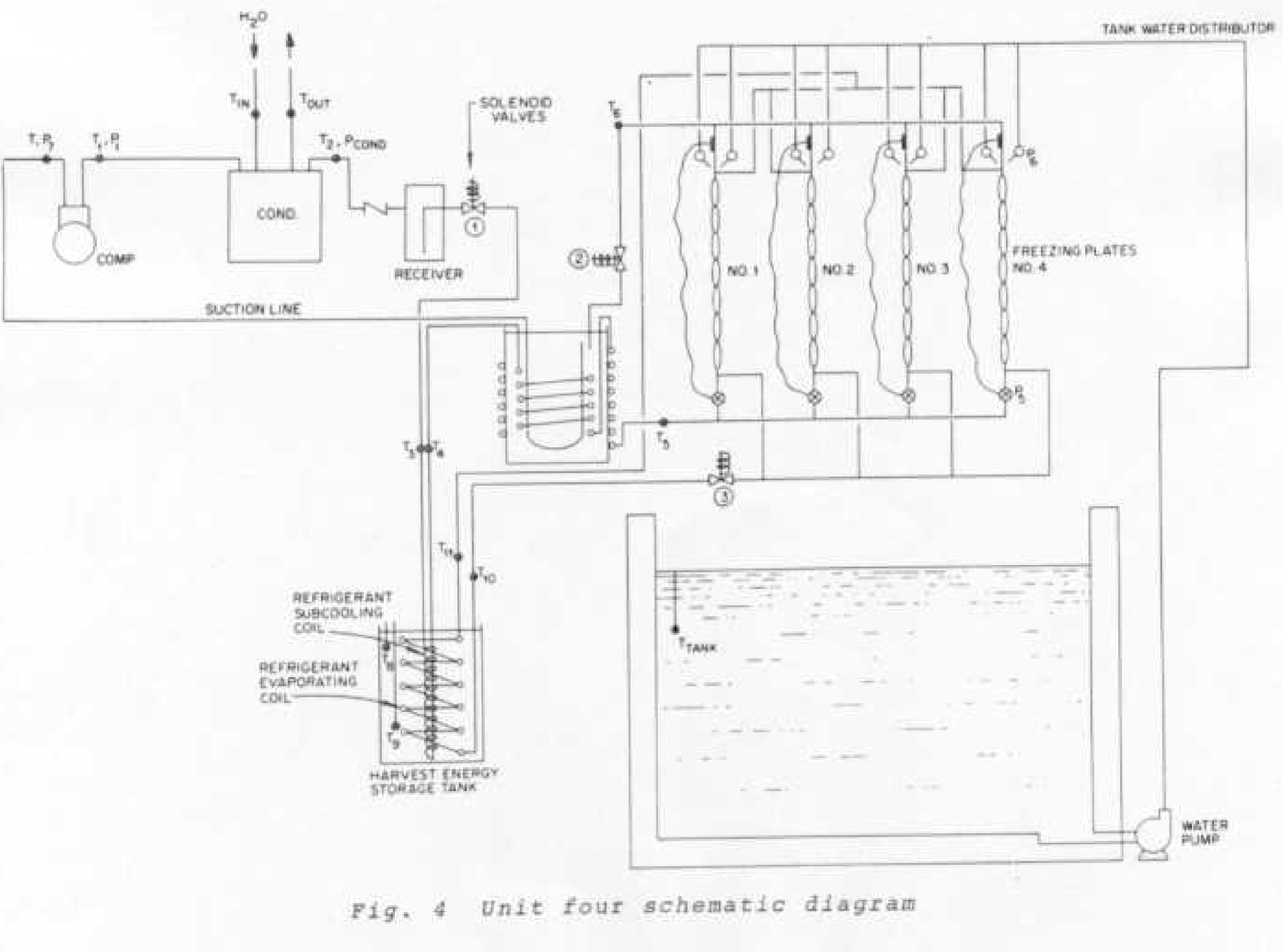

Fig. 4 Unit four schematic diagram

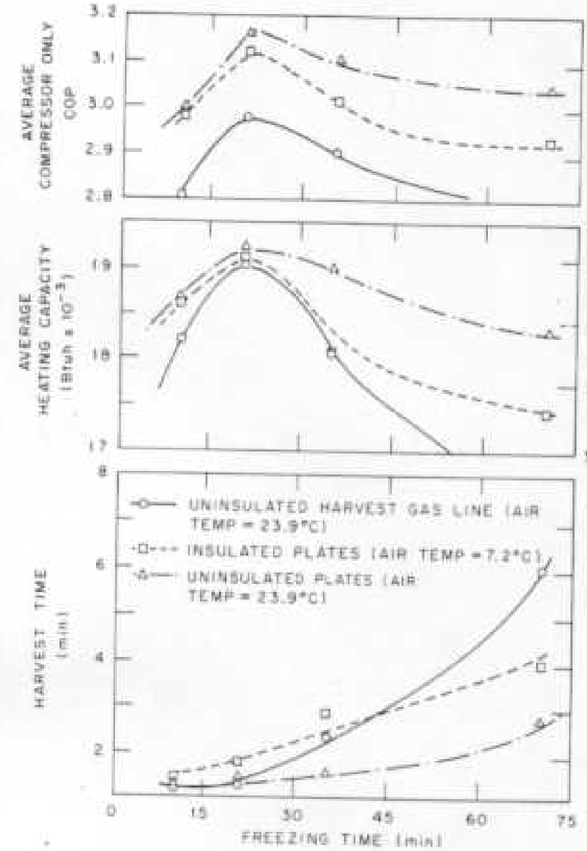

Fig. 5 Average COP, heating capacity, and harvest time of unit one vs freeze cycle time for two different air temperatures

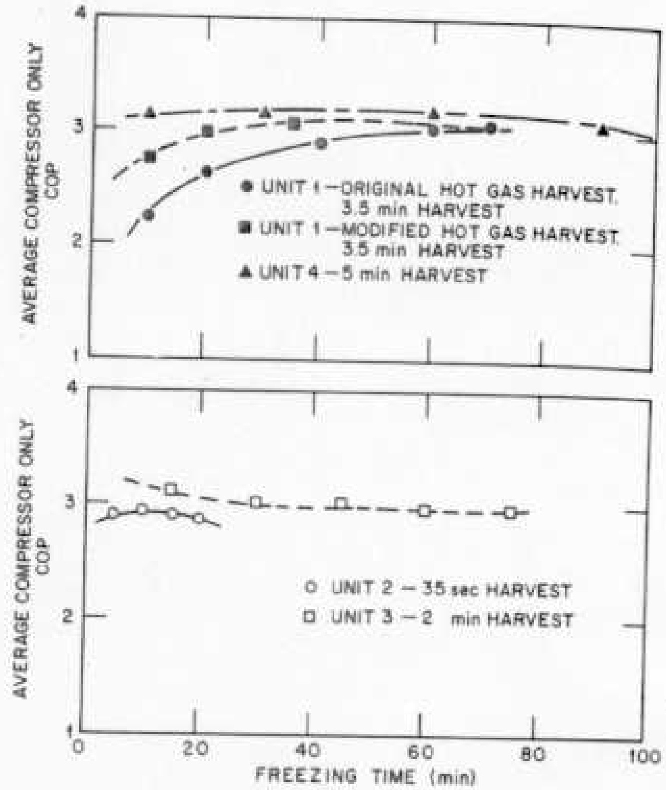

Fig. 6 Average COP vs freezing time for the IMHP test units

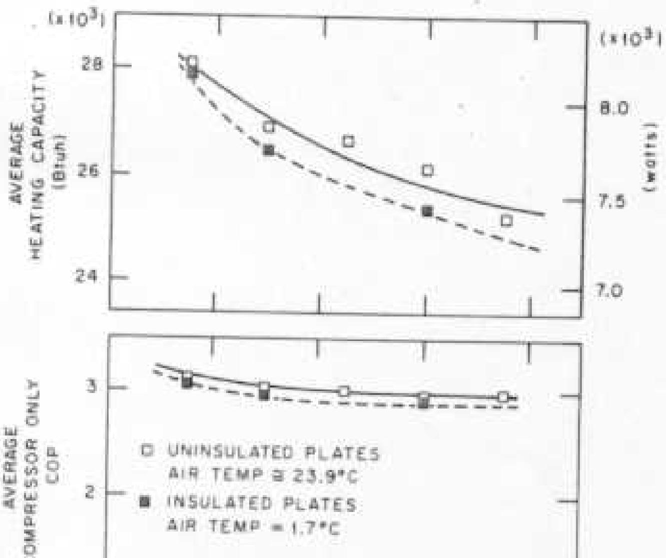

Fig. 7 Average COP and heating capacity of unit three as a function of air temperature and freeze cycle length

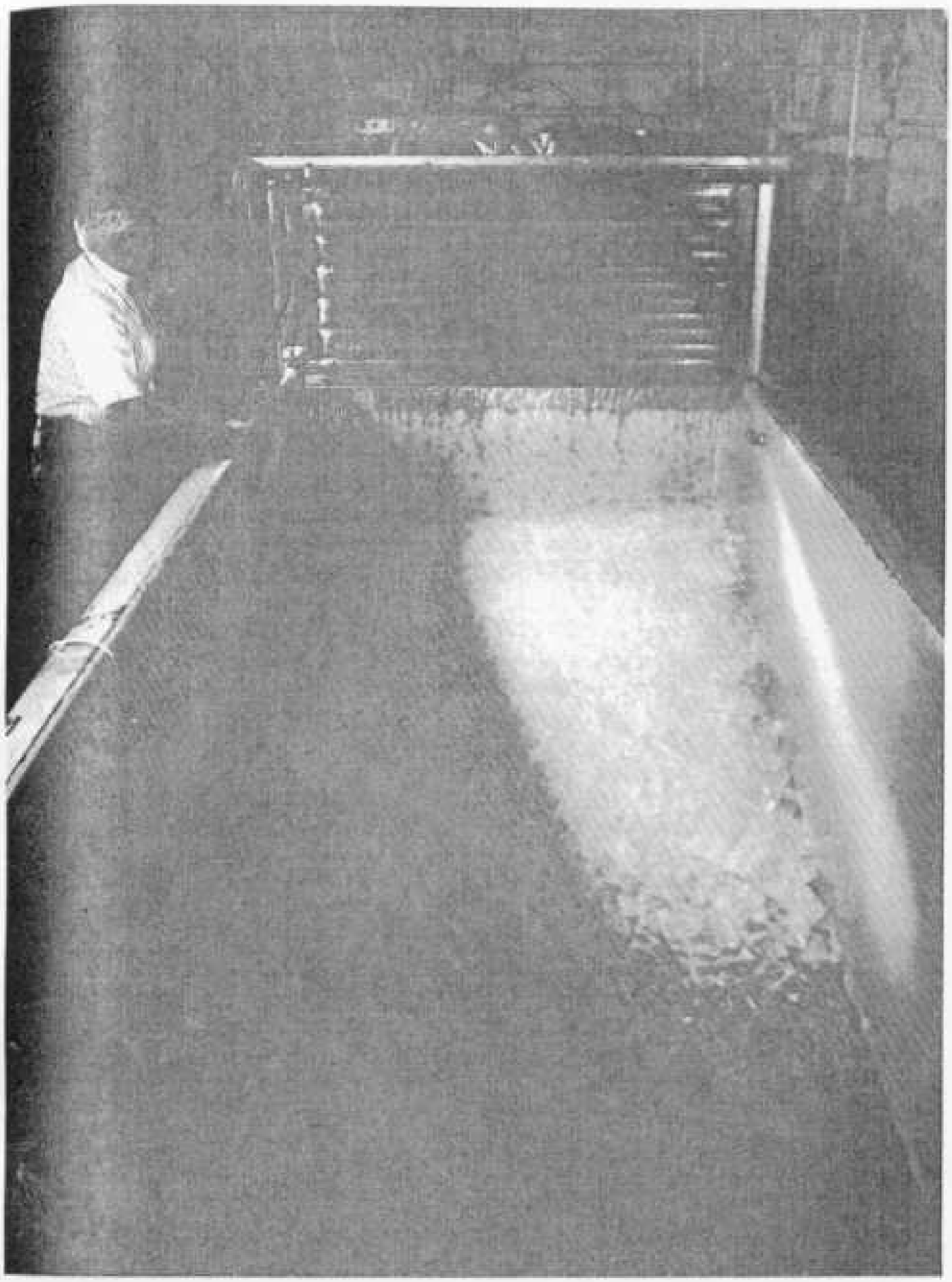

Fig. 8 Ice mound produced by IMHP during bulk sample test

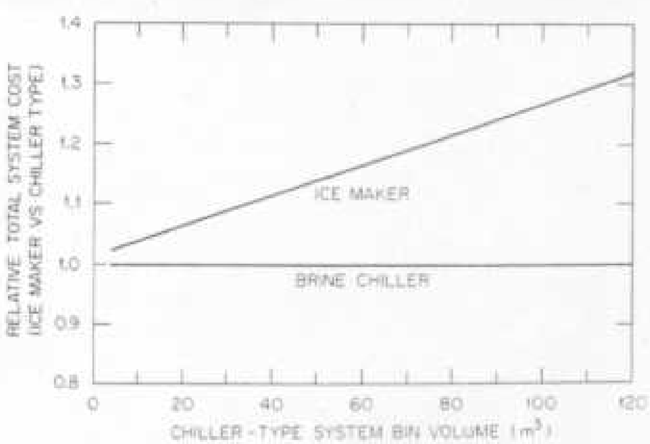

Fig. 9 Relative total system costs, ice-maker vs chiller-type for residential ACES mechanical package alternatives

DISCUSSION

M.E. RUSSELL, P.E., Dairy Equipment, Madison, WI: How do you determine initiation and completion criteria for the harvest cycle?

VAN D. BAXTER: Several methods of harvest cycle initiation and termination were tried, including suction pressure and timed cycles. Freeze and harvest cycles of fixed time duration worked best in the laboratory tests with harvest cycle duration determined by several observations. As Dr. Somerville pointed out, in a field installation where the plates are not visible, a harvest cycle control scheme based on the weight of the accumulating ice is desirable. Such a scheme appears to offer the best guarantee that all of the ice is indeed removed from the plates when they are harvested.