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Commercial Cool Storage Laboratory Test Procedure

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COMMERCIAL COOL STORAGE LABORATORY TEST PROCEDURE

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NOMENCLATURE

ASHRAE = American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ARI = Air Conditioning, and Refrigerating Institute

CAP = average capacity (tons)

DX = direct expansion

h_{eA} = exiting air enthalpy (Btu/lb_m)

h_{eR} = refrigerant enthalpy leaving the storage tank (Btu/lb_m)

h_{ew} = exiting water enthalpy (Btu/lb_m)

h_{fg} = enthalpy of vaporization

h_{iA} = entering air enthalpy (Btu/lb_m)

h_{iR} = refrigerant enthalpy entering the storage tank (Btu/lb_m)

h_{iw} = entering water enthalpy (Btu/lb_m)

ISTF = Ice Storage Test Facility

L_s = static heat gains to tank (Btu/h)

$\dot{m}_{eA}$ = air mass flow rate from storage (lb_m/h)

$\dot{m}_{iA}$ = air mass flow rate into storage from agitator (lb_m/h)

$\dot{m}_{eR}$ = refrigerant mass flow rate leaving the storage tank or entering the compressor (lb_m/h)

$\dot{m}_{iR}$ = refrigerant mass flow rate entering the storage tank or leaving the condensing unit (lb_m/h)

$\dot{m}_{eW}$ = water mass flow rate exiting the storage tank (lb_m/h)

$\dot{m}_{iW}$ = water mass flow rate returning to storage tank from the load (lb_m/h)

OR = overfeed ratio

$\hat{Q}$ = heat flow from ambient into storage (Btu/h)

$\dot{Q}_c$ = heat rejected from condensing unit (Btu/h)

$\dot{Q}_L$ = building total cooling load (Btu/h)

ton = 12,000 Btu/h

ton-h = 12,000 Btu or 83.3 lb of ice

U(t) = storage internal energy (Btu) at time, t

$\dot{W}$ = mechanical power expended in storage (Btu/h)

$\dot{W}_c$ = power input to compressor (Btu/h)

$\dot{W}_p$ = power input to pump (Btu/h)

EXECUTIVE SUMMARY

BACKGROUND

Many utilities have identified cool storage in commercial buildings as one of the most attractive load management options. However, these same utilities have stated that they must be reasonably assured that the cool storage equipment will function as claimed before they expend the effort necessary to structure incentives and develop a marketing strategy.

Many of the guidelines for the design and installation of cool storage systems have not been adequately verified by either laboratory or field testing. Although cool storage has been used in a large number of commercial buildings, in many installations the performance of the cool storage system is not adequately monitored and/or the results are not widely reported. In a number of cases where performance data were taken, significant design, installation, and operating problems were noted, and the quality of the performance data was generally poor. Even if additional monitoring equipment were to be added to existing field installations, many performance issues would remain unanswered because tests cannot generally be made once a system is installed in the field and being relied upon to provide space conditioning.

One solution to the problems of identifying design deficiencies, documenting performance, and providing improved cool storage installation and design guidelines is laboratory testing cool storage systems following a uniform test precedure.

COOL STORAGE PERFORMANCE ISSUES

The majority of manufacturers of cool storage equipment were contacted for their ideas on both the need for system testing and the required test procedures. The manufacturers contacted included Baltimore Aircoil Company, Inc.; Thermal Energy Storage Inc.; Turbo Manufacturing; Chester-Jensen Company; Sunwell Engineering Company; National Integrated Systems, Inc.; Calmac Manufacturing Corporation; Girton Manufacturing;

TDS, Inc.; Continental Equipment Corporation; Chicago Bridge and Iron; and Midwesco Inc. The performance issues most frequently cited by the manufacturers were charging efficiency, discharge rates and temperatures, state-of-charge sensing, and evaporator circuit balancing.

Twenty architects, designers, and engineers experienced in the installation and operation of commercial cool storage equipment were contacted to determine the unresolved performance issues for cool storage systems. Although most of their systems were reportedly now performing satisfactorily, many problems were found in past designs and installations. These designers and builders described problems obtaining adequate information for many issues including energy efficiency, optimal operation for flooded refrigeration systems, optimal suction temperatures, tank insulation requirements, refrigeration system slugging problems, oil return problems, nonuniform ice formation and melting, tank and circuit balancing methods, parasitic energy requirements, corrosion, and expansion valve problems when operated over a wide range of suction temperatures.

ICE STORAGE TEST FACILITY DESCRIPTION

A preliminary test procedure was drafted to address most of the cool storage performance issues. Using this procedure, a functional test facility specification was made. The final ice storage test facility was designed to meet this functional specification.

The basic function of the testing facility is to monitor the ice storage system's thermal performance during charging and discharging of the storage unit. The facility must also provide variations in charging and discharging conditions. The fundamental components for the facility include the ice storage unit, refrigeration system, versatile piping circuits, discharge heater, condensers, instrumentation, control devices, and a data acquisition system to monitor and control the facility. The instrumentation for the facility includes measurements of electrical power, temperature, flow rate, pressure, and stored cooling.

TEST PROCEDURES

Available cool storage equipment is based on a number of different concepts and designs. Several manufacturers produce ice tanks with integral evaporators to be mated with field-supplied refrigeration systems. Other systems include only the condensing unit and ice manufacturing components, while the tank is field erected. Still others operate with an intermediate brine loop between the condensing package and ice tank so that the refrigerant circuit is external to the storage tank.

Because of the multiplicity of available system types, meaningful comparisons are difficult. For example, poor thermal performance of an ice-on-coil evaporator configuration may be compensated somewhat by the use of an efficient evaporative condenser or an efficient chiller design. On the other hand, the performance of an otherwise efficient ice-forming process may be compromised by poor heat exchange characteristics in the balance of the system design. Consequently, it is a requirement that the cool storage test procedures be structured so the detailed performance of all components of the system, including the chiller, evaporator, flow control devices, and heat exchangers, be determined by the procedures as well as overall combined system performance.

Without exception, ice storage system designs are composed of three generic components: (1) the ice manufacturing/storage component; (2) a chiller package or condensing unit; and (3) a building loads component, which, although not a piece of system hardware, must nevertheless be simulated in any procedure to characterize the discharging process. Test procedures were designed to test each of these three components as well as overall system performance. The test procedures developed recognize the unique attributes of direct expansion, flooded, secondary loop, and dynamic systems.

COMMERCIAL COOL STORAGE LABORATORY TEST PROCEDURE

T. K. Stovall J. J. Tomlinson

ABSTRACT

Many utilities have identified cool storage in commercial buildings as one of their most attractive load-management options. Widespread adoption of this technology can be enhanced by greater certainty about the performance of cool storage systems and the resolution of relevant technical issues. To address these issues, cool storage equipment manufacturers, architects, designers, and engineers experienced in cool storage implementation recommended a program of laboratory testing to identify design deficiencies, document performance, and provide improved cool storage installation and design guidelines. The Ice Storage Test Facility was designed to address these issues through a thorough testing program independent of the manufacturers. Test procedures presented in this document are aimed at answering all outstanding performance issues.

1. INTRODUCTION

Cool storage systems based on ice have been used for a number of years in processing industries such as dairies and breweries and have more recently been used in building heating and cooling. Contacts with manufacturers of ice storage systems have indicated that the dairy application identified as much as 30 years ago was the principal reason that ice storage machines were developed. In this particular application, fresh milk must be cooled quickly to prevent bacterial growth during storage so that the process cooling load (tons of refrigeration) is large and the duration of the load (hours) is relatively short.* Without cool storage a substantial capital outlay for large liquid chillers would be needed to meet this cooling requirement. Although the total

*Units used throughout this report are common to and exclusive in the industry.

cooling energy requirements with or without storage remain unchanged, the storage option accommodates this energy requirement by using a chiller of reduced capacity to freeze ice over a long time period. Upon discharge, the ice melts quickly so that a much higher cooling capacity is available, albeit for a smaller period of time.

The typical cooling load found in commercial buildings is similar to that in the dairy process discussed above. This load is greatest during the daytime because of the air conditioning needed to offset the building solar gains and internal heat generated from people, electrical equipment, and lighting. Because electric utility demand charges are often exacted on the commercial customer, it is more economically feasible to shift the electrical demand for building air conditioning to the night when the building baseload is small.

Although arguments can be made for both latent and sensible storage systems in buildings, retrofit applications of cool storage are generally restricted to latent storage. A principal reason for this is the large volume required for sensible storage compared with latent storage. Because the permissible temperature range for sensible cool storage in water is ~18°F (from 32 to 50°F), the volumetric storage capacity is 1120 Btu/ft^3. With ice storage, this number becomes 10,100 Btu/ft^3, a factor of 10 increase in storage density. In practice, this factor is reduced to between 6 and 8 because of considerations such as the presence of heat exchangers in the ice storage tank and ice-packing factors less than unity. The relatively small-size requirement for latent cool storage systems has led to the manufacture of ice storage systems that can be shipped and installed in the mechanical room of a building, on the roof, or outside. The size range of latent cool (principally ice) storage systems being manufactured is broad. The number of manufacturers is <20.

A survey of cool storage installations in the United States and Canada completed in 1980 identified a relatively large number of ice storage installations. However, a major problem has been lack of verifiable performance data and design guidelines. For utilities to structure incentives and develop a marketing strategy for cool storage equipment, they must be reasonably assured that the cool storage equipment will function as claimed. Under the sponsorship of the Electric Power Research Institute, the Ice Storage Test Facility (ISTF) was developed to provide uniform performance data in response to this need.[1] This report includes a brief description of the ISTF development history and a description of the completed ISTF, and the main focus is on the updated test procedures and analysis methods described in Chap. 4. The test results will be made available to utilities, building cooling system designers, and cool storage system manufacturers.

2. COOL STORAGE TESTING

Manufacturers of cool storage systems were contacted, and their ideas on both the need for system testing and the procedure to be followed were evaluated. A description of these manufacturers and their products is found in the Appendix. Twenty architects, designers, and engineers experienced in the installation and operation of commercial cool storage systems were also contacted to determine unresolved performance issues for the systems. The problems and questions they encountered related to the system's thermal performance, economics, water treatment, compressor operation, ice formation, controls, and component sizes.

The need for a cool storage test program was indicated without exception by all of the cool storage system manufacturers contacted as well as a majority of system designers. While some testing has been done on ice storage equipment by the manufacturer, the test results were largely unavailable, or, if available, they were often found to be limited in scope.

The performance and field application issues varied somewhat among the various types of ice storage systems, including direct expansion (DX), liquid overfeed, dynamic, and secondary fluid designs.

2.1 SUMMARY OF PERFORMANCE ISSUES

2.1.1 System Efficiency

For most systems, global system efficiency in terms of electrical energy input during charging and discharging vs thermal energy removed by the fan coil unit during discharge was unknown, or unavailable, but was identified as an important storage parameter for which tests should be directed. Clearly, many separate factors specific to a design influence this parameter.

More specific performance questions concerned optimizing the operation of flooded refrigeration systems, optimal suction temperatures, and the amount of tank insulation required. Most of the questions on thermal performance involved the need for detailed knowledge of the systems efficiency to correctly size the system and to improve efficiency.

2.1.2 Refrigerant Conditions

In static ice makers, particularly, knowledge of refrigerant saturated suction temperatures and pressures is needed during the ice building process as a measure of charging performance. It is somewhat misleading to represent these quantities as an average of refrigerant conditions at the beginning and at the end of ice buildup. An accurate estimate of these quantities is needed as a function of time during charging so that compressor or chiller derating can be determined on a dynamic basis and system performance can be predicted at all times during the charging process.

2.1.3 System Thermal Losses

Thermal gains (defined as thermal losses in this context) to the ice storage tank are generally unknown. This is a difficult quantity to measure directly, and available numbers are estimates based on tank thermal resistance and convective heat transfer coefficients. Tank thermal losses and the effect of possible moisture condensation on the outside of the tank walls on these thermal losses should be determined.

2.1.4 Parasitic Energy Requirements

In several ice storage system designs, agitation is used to combat thermal stratification and maintain uniform water temperature during freezing and melting. Techniques used include water recirculation across baffles and air agitation. Although both techniques require that extra energy be expended, some manufacturers indicated that using compressed air for this task was particularly inefficient because the injected air carried the heat of compression to the ice tank. Other system inefficiencies are associated with water or brine circulation during charging and discharging.

Although ice-on-coil storage systems are by far the most common, at least one manufacturer produces an ice-harvesting heat pump that builds a relatively thin layer of ice on an evaporator surface and periodically releases the ice into a storage bin. The defrosting operation removes surface fouling and increases heat transfer. The disadvantage is the extra energy required for ice harvesting, which is accomplished by using hot-gas bypass from the vapor compression cycle. This technique requires that a portion of the refrigerant vapor leaving the compressor be directed to the evaporator to release the ice.

2.1.5 Uniform Ice Formation and Melting

The uniform ice buildup on coils and its effect on the efficiency of ice formation are major concerns of both manufacturers and engineers. If ice builds in a nonuniform pattern on the coils, sections of the evaporator with little ice are being underutilized, while the efficiency of ice manufacture at the more thickly covered coil regions degrades the storage performance significantly. This is a complex issue because to simply determine how uniformly ice builds would require many detailed point measurements inside the storage tank. A parallel task is to determine the reasons for possible nonuniform ice buildup. In multiple evaporators or multiple evaporator tube passes in the same refrigerant circuit, unbalancing can occur because once uneven ice formation is initiated, portions of the coil can be starved of refrigerant. This phenomenon is more of a problem in direct expansion than in flooded systems. To evaluate the tendency for evaporator unbalancing, particularly in multiple evaporator systems, refrigerant mass flow rates through each expansion valve or evaporator circuit must be determined continuously during charging.

Nonuniform ice burnoff was cited as a significant performance issue by manufacturers. Without uniformity during melting, chilled water-flow passages tend to form through complex hydrodynamic processes within the storage tank and lead to tank "short circuiting," wherein elevated tank discharge temperatures are found even when a substantial quantity of ice remains. Tests to monitor chilled water-flow paths within a discharging tank may lead to recommendations for simple flow restricters located throughout the tank.

2.1.6 Ice Inventory Sensors

One of the most useful ice storage controls is a device to monitor ice inventory during charging, standby, and discharging. If reliable, such a device can be used to control the quantity of ice formed so that only that amount of ice required for building cooling during a peak period is manufactured in the prior offpeak period. A variety of ice inventory monitoring devices, including liquid level sensors, pressure transducers, point temperature sensors, and manually operated mechanical linkages that can be brought into contact with the ice, are being used by manufacturers. It was recognized by a majority of manufacturers that measurement of the amount of ice in a storage tank with these devices is imprecise; however, quantitative estimates of the accuracies of such devices are virtually unknown. Detailed laboratory tests that determine ice inventory by using direct techniques such as water displacement can be used to judge the accuracy of point sensors supplied with the storage units.

2.1.7 Latent Heat Capacity

Efforts to develop latent cool storage materials that freeze at temperatures higher than ice have resulted in the Transphase system based on a glauber salt eutectic and the Thermal Energy Storage Inc. (TESI) system based on a clathrate of refrigerant R-11 and water. The latent heat of the clathrate material is not accurately known, so the quoted storage capacity of a system based on this compound is questionable. The storage capacity of a Transphase module could be determined by calorimetry, but it is not clear that in a cool storage application all of the latent heat in the module would be available because of the small temperature gradients present. It may be that heat transfer limitations on the exterior of the module as well as through the glauber salt will reduce the effective storage capacity of the module. To evaluate this hypothesis, *in situ* thermal performance tests of modules located in a chilled water tank should be performed in addition to the basic calorimetry tests on a single module. Another issue concerning glauber salt eutectic is that the storage capacity tends to decrease with cycling because of incongruent melting.

The ratio of usable storage capacity (ton-h) to the overall system volume for latent (ice and other) cool storage systems is a consideration. The less effectively a cool storage system utilizes the storage volume available, the greater the overall system size must be and the smaller the advantage of latent storage over chilled-water (sensible) storage. Tests that measure storage capacity within defined delivery temperatures followed by measurements of available storage volume can be used to evaluate the volumetric efficiency of a system design.

2.2 FIELD APPLICATION ISSUES

2.2.1 Heat Exchanger Penalties

Heat exchangers are commonly found in ice storage system applications. Their presence between the condensing unit and storage tank or storage tank and load can exact energy penalties because the cooling system is required to operate at lower temperatures. These penalties are manifested through condensing unit derating that reduces the overall system efficiency. Although efficient plate heat exchangers with low approach temperatures are used to separate the water in the storage tank from that in the chilled-water distribution system, system thermal performance falls short of what is possible without heat exchangers. In certain ice storage system designs, ice is formed on tubes carrying a brine, and a refrigerant-to-brine heat exchanger is used. With these systems, heat exchangers incorporated into the design usually require additional circulation pumps. The negative impact on the system efficiency is therefore the result of heat exchanger penalties as well as additional pumping power requirements.

2.2.2 Refrigerant Oil Circulation

Condensing units comprised of reciprocating or screw compressors must be lubricated. In addition to reducing friction and wear, the oil helps cool the compressor. Although most of the oil remains in the compressor crankcase, a portion is recirculated with the refrigerant throughout the system. The oil has only limited solubility in the refrigerant; therefore, to ensure that oil is returned to the compressor crankcase, the refrigerant velocity (particularly in the vapor phase) must be high enough to sweep the oil through the system. This problem may be especially acute in the evaporator of direct expansion machines where, because of low refrigerant temperatures, the solubility of oil in the refrigerant is reduced, and at low cooling capacities refrigerant velocities are insufficient to keep the oil moving through the evaporator. This can cause the evaporator to become oil bound, thereby reducing the heat transfer effectiveness. This is a greater problem with R22 than with R12.

2.2.3 Onset of Corrosion

In some cases, inadequate water treatment in the ice tank was found to be a problem, leading to tank corrosion and heat exchanger fouling. Inadequate water treatment caused problems in a few of the systems. Some concrete tanks required protective coatings. One set of heat exchanger coils rusted through in only 2 1/2 years; improperly treated steel in the coils was the suspected cause. Corrosion of inactive air agitation pumps and piping was an additional concern. The short duration of any unit's installation on the test facility will provide an insufficient basis for meaningful analysis of this problem. Field installations should be examined to learn more about such corrosion problems.

2.2.4 Postinstallation Modifications

Inappropriately sized refrigeration components were identified in some of the installations. One system had an undersized liquid receiver that caused the refrigerant to back up into the condenser, decreasing the condenser's effectiveness. An evaporative condenser in another system was undersized and had to be replaced with a larger condenser. The performance of air-cooled condensers was especially difficult to predict because of changing ambient conditions. Difficulties were also expressed in trying to operate thermal expansion valves over a wide range of suction temperatures. Problems can also occur in the piping of chilled water. These problems include freezing of exposed pipes and drainage problems occurring when the system shuts off. Without proper controls, the storage tanks can overflow during shutdown, or the chilled water distribution pipes can fill with air.

Compressor-related problems occasionally occurred, primarily with DX refrigeration systems. One of the problems encountered was "slugging" or liquid refrigerant entering the compressor. By using accumulators and flooded system designs, slugging problems were reduced. One engineering firm converted all of their DX systems to liquid overfeed systems to avoid slugging problems. It was also suggested that only open-type compressors be used with ice storage systems. Screw compressors were commonly applied in larger systems.

Most of the designers and engineers contacted produce ice storage systems that operate effectively, but many also admitted to some problems encountered in earlier projects. Detailed information about the systems would allow engineers to reduce these initial problems and to optimize their systems for greater economy. This information, if developed, was expected to be particularly helpful to heating, ventilating, and air conditioning (HVAC) specialized engineering firms and would encourage strong consideration of ice storage systems by these engineering firms and commercial developers.

2.3 GENERAL CONSIDERATIONS

Ice storage equipment currently being manufactured (described in the Appendix) becomes part of a total system when placed in a building. In its simplest form, the total system consists of a condensing package that removes heat from the ice tank during the charging process, an evaporator section usually contained in a tank where ice is formed and stored, and a chilled water coil or liquid distribution system to the building itself.

From one manufacturer to another, there are variations in this basic design and configuration. The Calmac system, for example, employs an intermediate brine loop between the condensing package and the ice bin so the refrigerant circuit is external to the storage tank. Other manufacturers, including Chester-Jensen, recommend that a liquid-liquid heat exchanger be installed on the discharge of the ice storage tank so that ice water from the storage tank is not circulated directly to the building fan coil units (the cooling load) during discharge. In other systems (e.g., Sunwell), the condensing unit and ice-manufacturing components are packaged as an integral unit, and ice is stored as slush in an external tank. In such systems, a test procedure to evaluate only global system performance parameters such as system electrical energy input during charging and heat removed from the load during discharging would be overly simplistic and have limited application. For example, poor thermal performance of an ice-on-coil evaporator configuration may be somewhat compensated for through the use of an efficient evaporative condenser or an efficient chiller design. On the other hand, the performance of an otherwise efficient ice forming process may be compromised by poor heat exchange characteristics in the balance-of-system design. Therefore, the cool storage test procedure must be structured such that the detailed performance of all components of the system including chiller, evaporator, flow control devices, and heat exchangers be determined as well as overall system performance. Only in this way can specific design weaknesses be identified for redress by the manufacturer and the effect of component performance on total system efficiency be determined.

Without exception, ice storage system designs are composed of three generic components: (1) the ice manufacturing/storage component; (2) a chiller package or condensing unit; and (3) a loads component, which although not a piece of system hardware, must nevertheless be simulated in any procedure written to characterize the discharging process. The following sections describe each of these components separately and outline the test parameters and objectives.

2.3.1 Storage Performance

Ice storage systems can be thought of as storage heat exchangers in which a quantity of heat removed from a building is used to melt ice in a bin, and after a period of time this heat is removed from the bin (thereby freezing the ice once again) and rejected to the outdoor environment by a chiller. A simple thermodynamic description of this process considers a control volume drawn around the storage tank of a direct expansion ice maker with agitation as shown in Fig. 2.1. An

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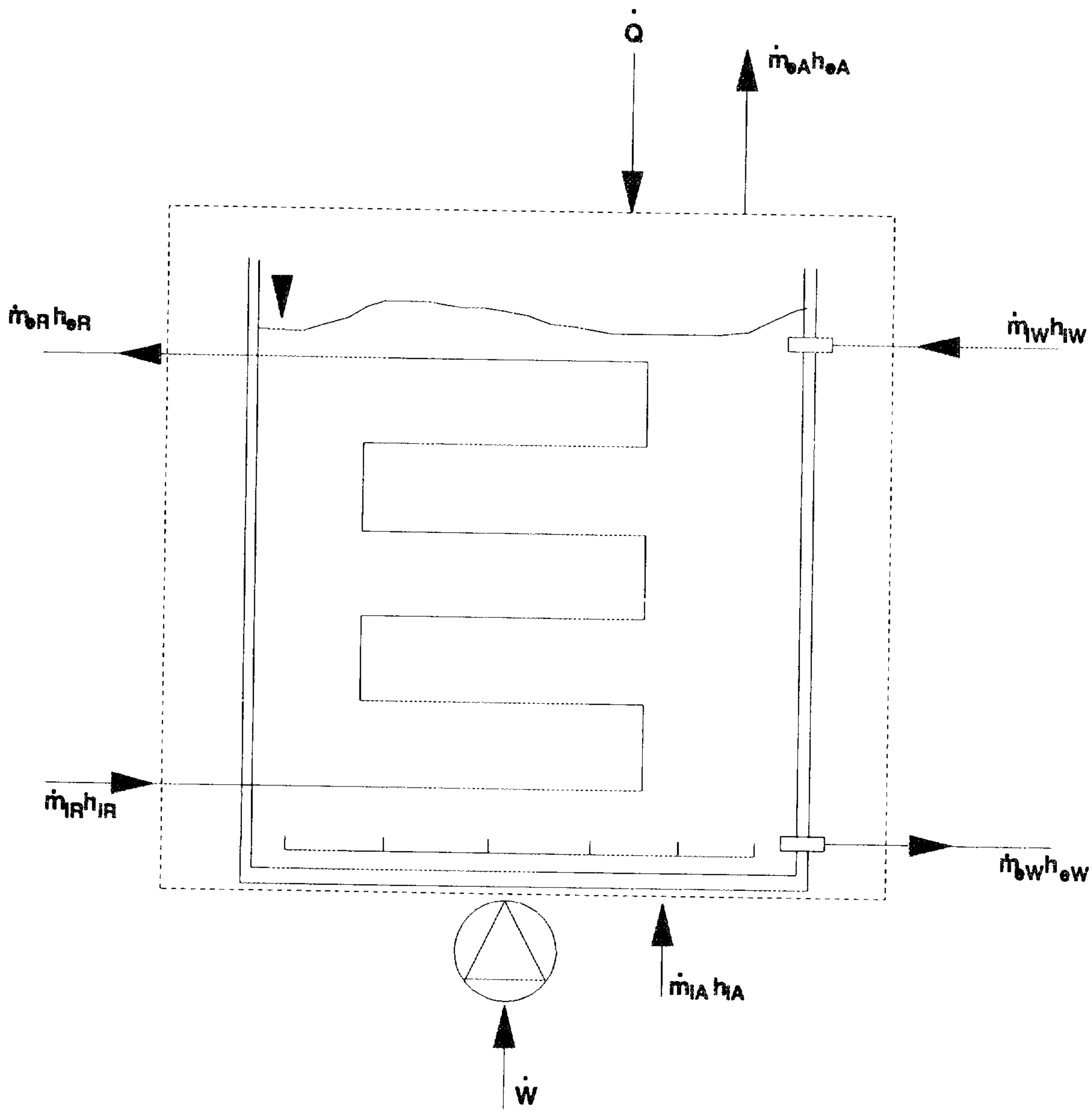

Fig. 2.1. Control volume surrounding static ice builder.

energy balance is maintained at all times for this system, including ice building, standby with a full charge of ice, and melting, and is given by

$$(\dot{m}_{iA}h_{iA} - \dot{m}_{eA}h_{eA}) + (\dot{m}_{iR}h_{iR} - \dot{m}_{eR}h_{eR}) + (\dot{m}_{iW}h_{iW} - \dot{m}_{eW}h_{eW})$$

$$+ \dot{Q} + \dot{W} = \frac{dU}{dt} , \qquad (2.1)$$

where

dU/dt = rate of change of internal energy in the ice tank,

$\dot{m}$ = mass flow rate,

$\dot{Q}$ = energy flow as heat gains through tank wall,

$\dot{w}$ = energy flow as mechanical work of agitation,

h = enthalpy,

i = inlet,

e = exit,

A = air,

R = refrigerant,

W = water.

This relation assumes that water, air, and refrigerant are the only materials that can enter or exit the storage vessel.

The reason for using Eq. (2.1) to describe test procedures for an ice storage system is obvious. One of the most important storage parameters is the system internal energy U(t) and the rate at which this parameter changes during charging with a chiller and discharging by a heat load impressed on the system. For an ice machine, this is equivalent to determining at any time the quantity of ice present in the tank, the quantity of water present, and the temperature distribution of each. With a sensible heat storage system, internal energy changes dU/dt can be determined simply from the temperature changes in the system. However, in the case of ice storage involving latent heat, storage temperatures alone are insufficient to determine the thermodynamic state and internal energy of the system at any time. However, the instantaneous rate of change of internal energy can be determined indirectly from Eq. (2.1) by measuring all of the enthalpy flows to and from the system, including work input. This equation can be integrated over time to yield the system internal energy at any instant — a parameter that indicates how much ice is present in the storage tank at any time during charging, standby, and discharging.

2.3.2 Chiller Performance

There are many configurations of vapor compression systems used in ice storage systems, and each is characterized by the type of compressor and the technique used for rejecting heat of compression and condenser heat to the ambient. Reciprocating compressors are commonplace in cooling applications up to 200 tons and are well-suited for ice storage systems. Reciprocating compressors are classified as open if the compressor requires an external drive (usually an electric motor through a coupling or belt) or closed (hermetic or semihermetic) if the compressor and electric motor are built into an integral housing. Although either type can be packaged as a condensing unit that includes the compressor plus an interconnected condenser, open compressors are better suited for ice making because the heat of the motor is not added to the refrigerant, as it is in a closed compressor.

A major performance issue identified by manufacturers and cool storage system designers is the degree to which condensing unit performance suffers with low evaporator temperatures commonplace in ice system designs and high condensing temperatures corresponding to peak summer-day conditions. The cool storage test procedure is structured to accommodate measurement of the performance of both air- and water-cooled condensing units if they are part of the manufactured cool storage system. A schematic of a refrigeration system with work and heat flows through a control volume surrounding the package indicated is shown in Fig. 2.2. Excluding transient startup and shutdown periods, the system is considered to be a steady-state, steady-flow device in which useful power input (or shaft horsepower), $\dot{W}_c$, to the compressor is expended as net energy flowing out of the control volume because of refrigerant mass transfer ($\dot{m}_{iR}h_{iR} - \dot{m}_{eR}h_{eR}$) and heat rejected, $\dot{Q}_c$, or from the first law,

$$\dot{m}_{iR}(h_{iR} - h_{eR}) + \dot{W}_c = \dot{Q}_c \ , \tag{2.2}$$

because the refrigerant mass flow is the same at all points in the loop for steady-state conditions. If unsteady flow exists, or if receivers are present, the refrigerant mass flows into and out of the control volume should be adjusted accordingly. The first two terms in Eq. (2.2) represent the refrigerating capacity of the condensing unit and are also the rate at which heat is withdrawn from the storage tank during charging. Refrigerating capacity is an important testing parameter that can be determined directly by measuring the mass flow $\dot{m}_{iR}$ of the subcooled

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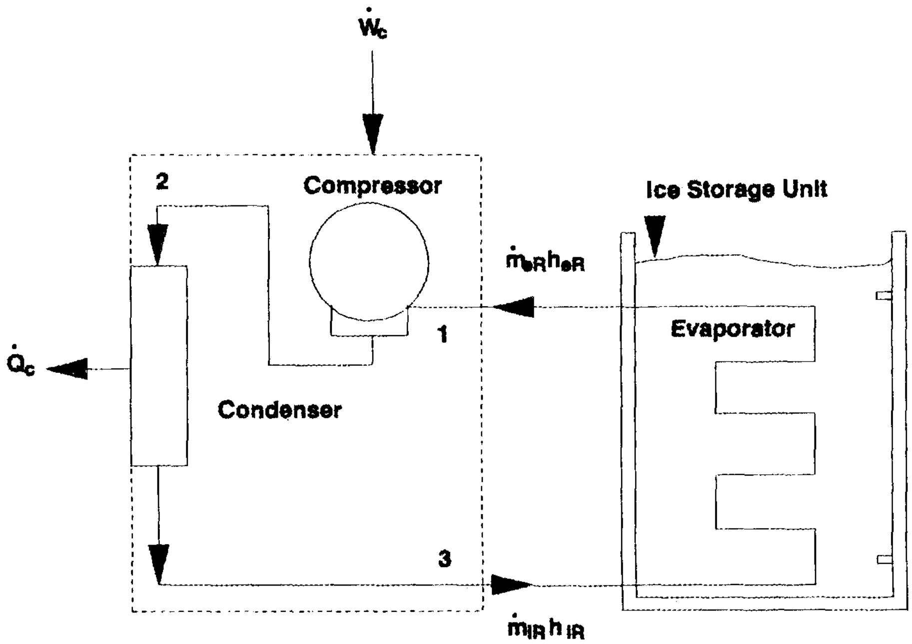

Fig. 2.2. Refrigeration system.

refrigerant at point 3 of Fig. 2.2 and the state of the refrigerant at points 1 and 3. Because the refrigerant is single phase at these points, pressure and temperature uniquely determine the refrigerant state. Through Eq. (2.2), the refrigerating capacity of the package can also be determined indirectly by measuring $\dot{W}_c$ and $\dot{Q}_c$. This technique could be used to corroborate refrigerant measurements or to determine refrigerating capacity when penetration of refrigerant lines is not practical.

The technique used to measure rejected heat, $\dot{Q}_c$, depends on the chiller package used. For air-cooled condensing units, $\dot{Q}_c$ can be determined by measuring the flow and rise in dry bulb temperature of air flowing through the condensing unit and the air flow rate. For liquid-cooled condensers, $\dot{Q}_c$ can be measured on the liquid side of the condenser with appropriate corrections made for heat losses to ambient.

2.3.3 Load Characterization

In building cooling applications, the total cooling load comprised of sensible and latent heat components is satisfied by a water coil (water-to-air heat exchanger) housed in an air handler or fan coil unit. The sensible heat ratio of the air is the fraction of the total cooling load that is sensible heat. With ice storage systems, low coil surface temperatures are available, and the air sensible heat ratio can be smaller than would otherwise be possible with nonice systems. However, discussions with some cool storage system designers indicate that low air sensible heat ratios may lead to overdehumidification of the building supply air. Fan coil temperature control can be accomplished quite simply through use of a recirculation loop and three-way valve. A schematic of a chilled-water distribution system is shown in Fig. 2.3. The three-way valve modulates to divert a portion of the warmed water from the load exit to point A so constant load exit and inlet temperatures can be maintained for any cooling load, $\dot{Q}_L$, by varying the pump power and recirculation valve. From the energetics of the discharge process within the dotted control volume of Fig. 2.3, it can be seen

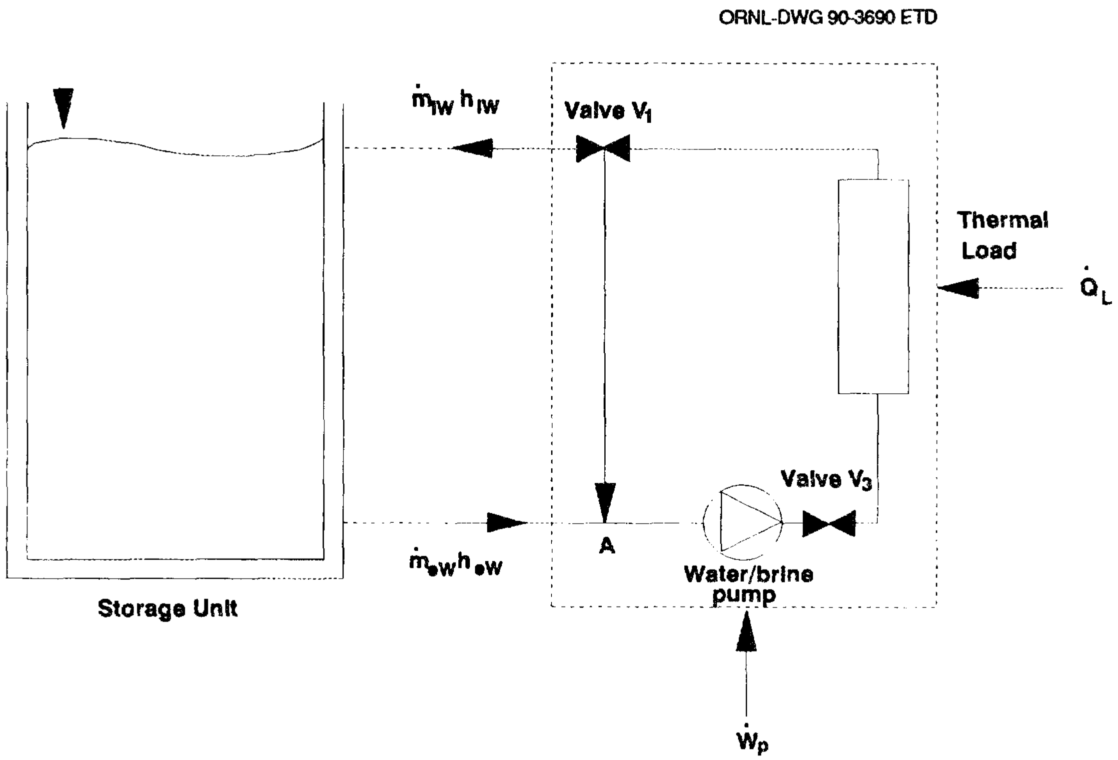

Fig. 2.3. Building load control.

that

$$\dot{m}_{iW}h_{iW} - \dot{m}_{eW}h_{eW} = \dot{W}_p + \dot{Q}_L \ , \tag{2.3}$$

where $\dot{W}_p$ is the power input to the pump assuming that thermal losses from the pump are part of $\dot{Q}_L$. In most ice storage systems, all quantities in Eq. (2.3) can be measured using standard techniques.

3. TESTING FACILITY DESCRIPTION

The ISTF was designed and built for the sole purpose of testing latent cool storage systems.[1] Wherever possible, the measurement techniques and accuracies for the testing have been based on ASHRAE Standards.

The refrigeration compressors, piping, and other components are versatile enough to satisfy most ice storage manufacturer's specifications. The test facility provides adequate support and instrumentation to accommodate direct expansion, dynamic, secondary loop, and liquid overfeed ice storage systems. Instrumentation on the refrigeration system (compressor and condenser) and ice storage unit (evaporator) includes inlet/outlet refrigerant pressures, temperatures, and flow rates. In addition, chilled water or brine temperatures and flow rates are measured. Water level, temperature, and ice thicknesses in the storage tank are measured whenever possible. Also measured are the power input to the system for the compressors and pumps. The installation of the ice storage system is subject to approval by the storage manufacturer to ensure the proper system performance. To discharge the storage unit a discharge piping circuit and an electric resistance heater are used.

3.1 GENERAL FACILITY DESCRIPTION

The basic function of the ISTF is to monitor the ice storage system's thermal performance during charging and discharging. The facility also provides variations in charging conditions and discharging loads. The facility's fundamental components are an ice storage unit, a refrigeration system, versatile piping circuits, a thermal loading device, condensers, instrumentation, control devices, and a data acquisition system to monitor and control the system and facility. A schematic of the facility and its main components are shown in Figs. 3.1–3.3. (The instrumentation shown in these figures is discussed later and listed in Table 3.1.) An examination of these figures show many components that are used to test one type of system but not another. For example, the low-pressure receiver and refrigerant pump are used only when testing liquid overfeed systems. The evaporator/chiller is

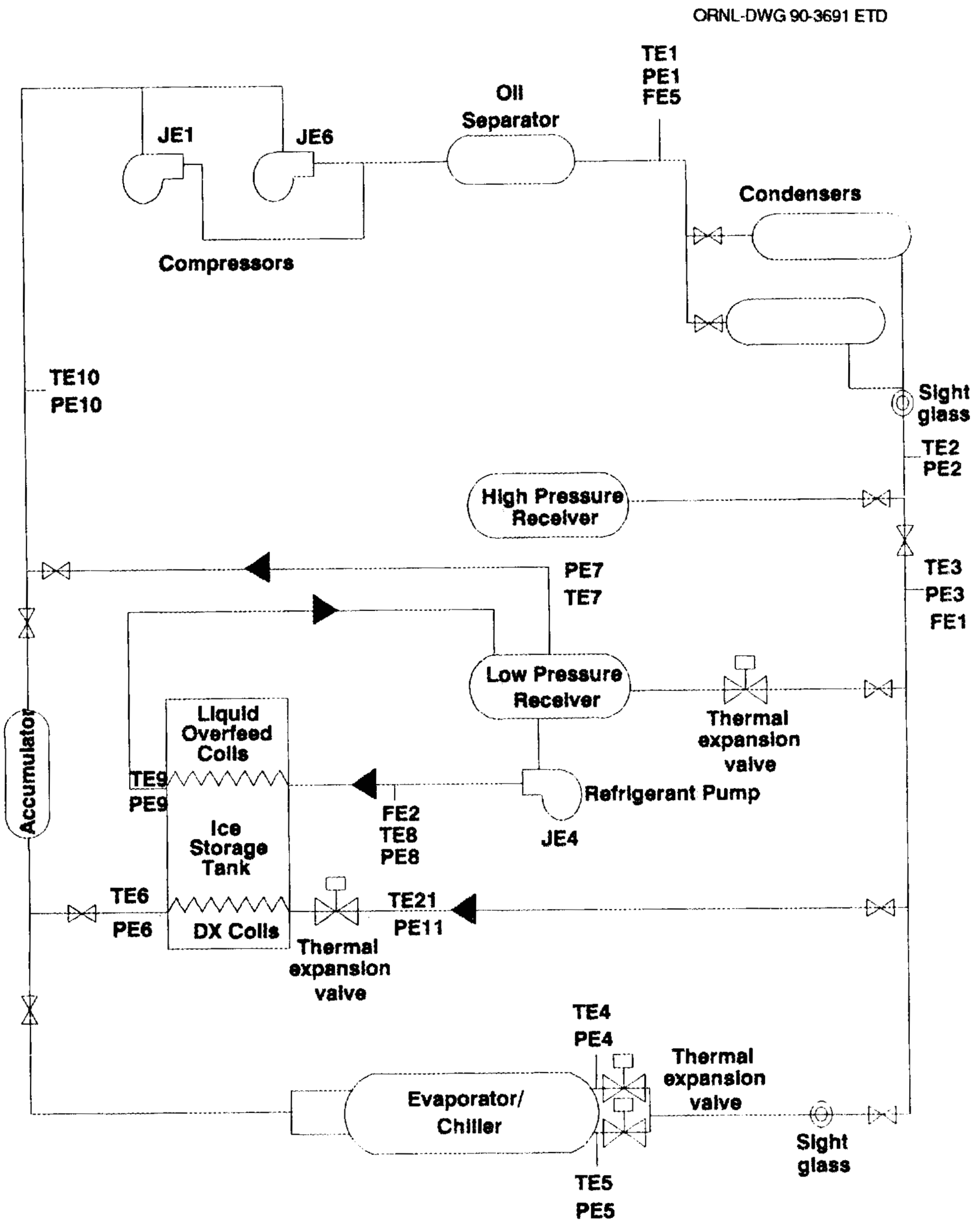

Fig. 3.1. Ice storage testing facility, refrigerant components, and instrumentation.

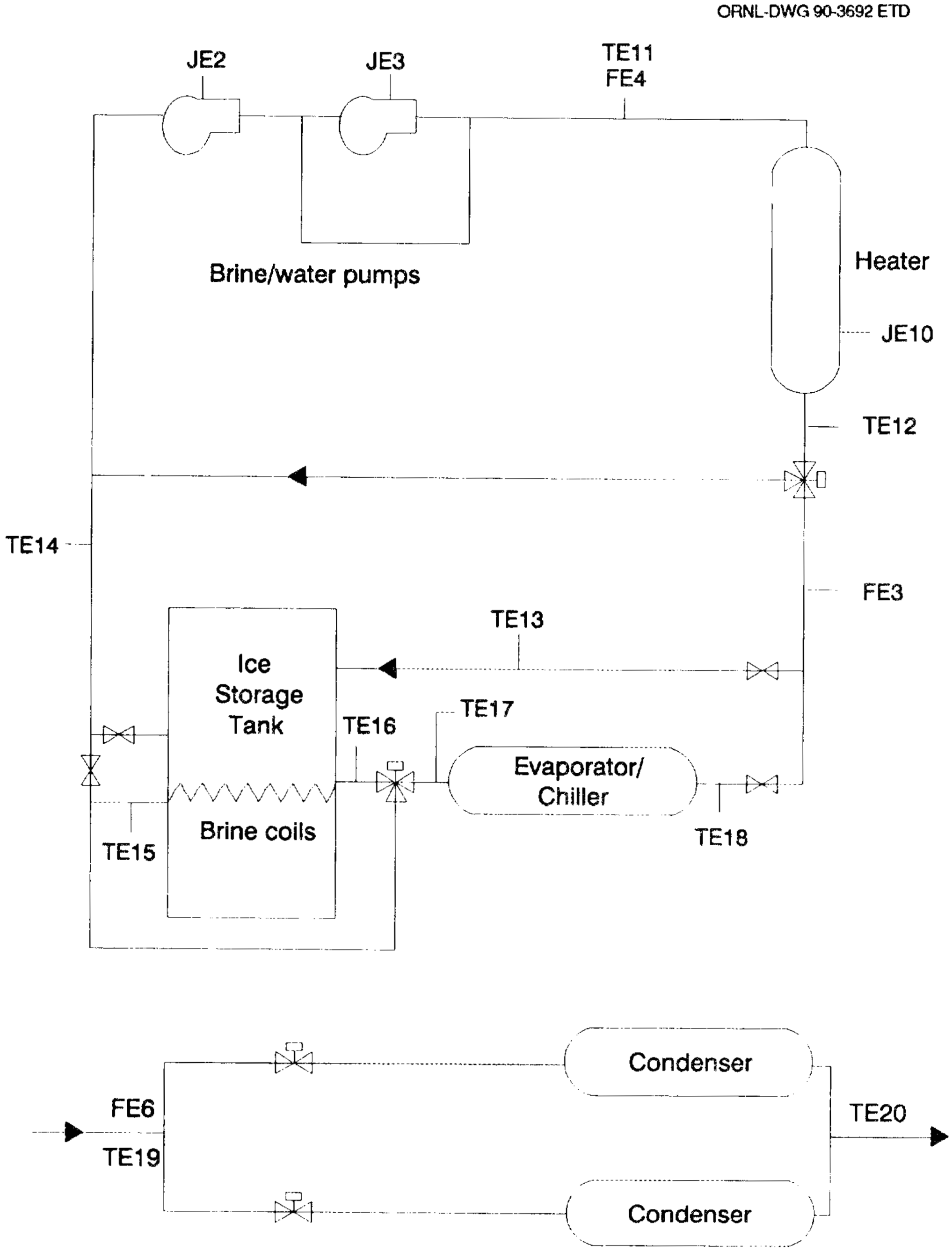

Fig. 3.2. Ice storage testing facility, water/brine components, and instrumentation.

Table 3.1. ISTF monitoring points

Point label	Measured quantity
FE1	Refrigerant flow to expansion valve(s), mass
FE2	Refrigerant flow, to ice tank for liquid overfeed system, or hot-gas defrost flow for dynamic ice maker, mass
FE3	Water flow to ice tank or brine flow to evaporator/chiller
FE4	Brine or water flow at pump discharge
FE5	Compressor discharge gas flow, volume
FE6	Condenser(s) inlet water flow
FE7	Water recirculation flow for dynamic ice maker
JE1	Compressor energy and power
JE2	Brine/water pump energy and power
JE3	Brine/water pump energy and power
JE4	Refrigerant pump energy and power for liquid overfeed system
JE5	Agitation air compressor power, or water pump energy and power for dynamic ice maker
JE6	Compressor energy and power
JE10	Heater energy and power
PDE1	Differential pressure (used to measure tank water height)
PE1	Compressor discharge pressure
PE2	Condenser outlet refrigerant pressure
PE3	Refrigerant pressure to expansion valve(s)
PE4	Refrigerant pressure to evaporator/chiller, after expansion valve
PE5	Refrigerant pressure to evaporator/chiller, after expansion valve
PE6	Refrigerant pressure out of DX evaporator coils
PE7	Low-pressure receiver pressure
PE8	Ice tank inlet refrigerant pressure for liquid overfeed coils
PE9	Ice tank exit refrigerant pressure for liquid overfeed coils

Table 3.1 (continued)

Point label	Measured quantity
PE10	Compressor suction pressure
PE11	Refrigerant pressure to DX expansion valve
TE1	Compressor discharge temperature
TE2	Condenser outlet refrigerant temperature
TE3	Refrigerant temperature to expansion valve or hot gas refrigerant temperature for dynamic ice maker
TE4	Liquid refrigerant temperature to evaporator/chiller
TE5	Liquid refrigerant temperature to evaporator/chiller or refrigerant temperature at evaporator plate exit for dynamic ice maker
TE6	Refrigerant temperature at DX coil outlet or at evaporator plate exit for dynamic ice maker
TE7	Low-pressure receiver temperature or refrigerant temperature at evaporator plate exit for dynamic ice maker
TE8	Ice tank inlet refrigerant temperature for liquid overfeed system or refrigerant temperature at evaporator plate exit for dynamic ice maker
TE9	Ice tank outlet refrigerant temperature for liqid overfeed coils
TE10	Compressor suction temperature
TE11	Heater inlet water or brine temperature
TE12	Heater outlet water or brine temperature
TE13	Ice tank inlet water temperature
TE14	Ice tank outlet water or brine temperature
TE15	Brine temperature at ice storage tank outlet
TE16	Brine temperature at ice storage tank inlet
TE17	Water or brine temperature at evaporator/chiller outlet
TE18	Water or brine temperature at evaporator/chiller inlet
TE19	Condenser inlet water temperature
TE20	Condenser outlet water temperature
TE21	Refrigerant temperature to DX expansion valve

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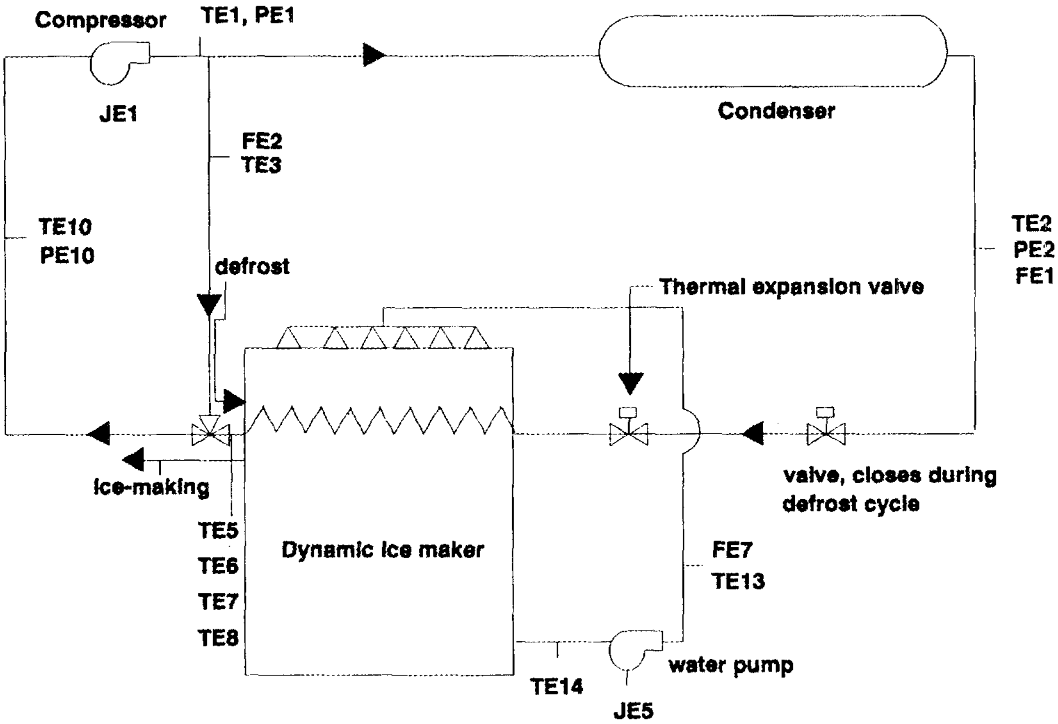

Fig. 3.3. Instrumentation for dynamic ice-maker refrigeration system.

used only when testing secondary loop systems. Each component of the facility will be further described in the following paragraphs.

An ice storage unit, with a latent capacity up to 240 ton-h, is provided by its manufacturer for testing. A testing floor is provided that is capable of carrying the expected weight loads. All piping connections between the storage unit and the test facility are made at this testing floor location.

Two reciprocating, open-drive compressors, nominally rated at 40 and 60 tons, can be used individually or in parallel. Each compressor can be loaded at 50, 75, or 100% capacity. An accumulator is positioned in the test facility to prevent liquid refrigerant exiting either a DX evaporator or a brine chiller barrel from entering the compressor suction. An oil separator is used to improve oil return to the compressor.

Two water-cooled condensers, rated at 60 and 40 tons, were installed in parallel. Either, or both, may be used depending on the system load. The flow of cooling water to the condenser(s) is varied to control the compressor discharge pressure.

The high-pressure, surge-type receiver serves as a buffer during changes in refrigerant flow rate and as a storage location for the facility's refrigerant inventory.

The low-pressure receiver serves to separate the liquid and gas refrigerant for liquid overfeed systems. The low-pressure receiver therefore both prevents liquid slugging at the compressor suction and provides an adequate liquid head to the refrigerant pump. A metering expansion valve feeds refrigerant to the low-pressure receiver and is controlled by the level of the fluid in the receiver. A rotary, positive displacement pump with sliding vanes is used to pump liquid refrigerant from the low-pressure receiver through the evaporator of a liquid overfeed system. The pump is equipped with a variable speed motor to permit testing at different capacities and at different liquid overfeed ratios.

The evaporator/chiller is nominally rated at 60 tons and is used only when testing secondary loop systems. This evaporator/chiller is equipped with two separate and independent evaporator coil circuits. One or both circuits of the evaporator/chiller are used, depending on the system load. Identical thermostatic expansion valves feed each chiller circuit. These thermostatic expansion valves are controlled by the measured superheat at the chiller exit. The thermostatic expansion valve on a DX evaporator coil will also be controlled by the superheat at the exit of the evaporator coil.

The refrigerant lines are sized for minimum pressure drop and adequate oil return under low-capacity conditions. Most of the refrigerant suction lines are 3 1/8-in. copper tubing, most of the discharge gas lines are 2 1/8-in. copper tubing, and the refrigerant condensate lines are 3 1/8-in. copper tubing. Some short stretches of smaller-diameter tubing are required near flowmeters. The system is well equipped with isolation valves to permit the versatility necessary to test vastly different system arrangements.

The water/brine piping loop is shown in Fig. 3.2 and is equipped with a variable-speed 5-hp centrifugal pump and a 2-hp centrifugal booster pump, in series with an optional bypass. These pumps serve to circulate either water or brine when discharging any system and to circulate brine when charging a secondary loop system. Most brine/water piping is made of 4-in. polyvinyl chloride (PVC). Short piping sections of smaller diameters have been placed near flowmeters as necessary. Isolation valves permit the brine/water flow to be circulated through the storage tank (exterior melt) or through coils within the storage tank (interior melt and secondary loop charging). Two controlled three-way valves are included in the brine/water piping. The first is located at the heater exit and can divert a portion of the heater outlet flow to the pump inlet. This permits exact control of the heater inlet temperature. The second three-way valve is located between the chiller and the ice storage tank. This valve can be controlled to permit emulation of a partial storage system. A partial storage mode of operation would allow the chiller to meet a reduced heating load while simultaneously charging the ice tank at a reduced rate. The thermal loading device consists of an electric resistance heater with a variable power supply and a maximum rating of 135 kW.

Instrumentation on the test facility includes a host of temperature, pressure, flow, and power measurements. The instrument locations are indicated in Figs. 3.1–3.3 and are described in Table 3.1. Some of the instrumentation listed in Table 3.1 is not shown in these figures, including agitation air compressor power use, the differential pressure transducer used to measure tank water height, and a thermocouple used to measure ambient air temperatures and to take point measurements of tank fluid temperature. All of the instrumentation described in Table 3.1 is connected to the automatic data acquisition system. Other measurements are made via observation and recorded during a test. These observations include tank water height (when the differential pressure transducer cannot be used), the agitation air flow rate and temperature, and the examination of the refrigerant flow through the sight glasses. Together, the facility's instrumentation includes all the measurements required in the test procedures with some redundancy built in to aid in checking instrument accuracy. This instrumentation and the data processing and control equipment are discussed in Sect. 3.2.

Many of the ISTF components can be seen in Fig. 3.4. The large tank shown in the center of this photograph is the low-pressure receiver used with liquid overfeed systems. The smaller tank directly below the low-pressure receiver is the evaporator chiller used with secondary fluid systems. The vertical cylinder behind the low-pressure receiver (near the man) is the heater, used to test all system types. The condensers are located directly above the low-pressure receiver. In the foreground of the photograph is the small suction accumulator that prevents slugging to the compressors. The test floor with an ice storage tank installed and piped into the system is shown behind these components. This photograph was taken before the pipes were insulated.

The compressors are shown in Fig. 3.5. The two parallel compressors share an oil separator (vertical cylinder near the center of the photograph) and an oil still (horizontal cylinder partially visible behind the compressors).

The ISTF equipment is operated from the control panel and personal computer shown in Fig. 3.6. The left control panel includes controls for compressor loading and pump and heater power, as well as a selector switch to identify the system type (either secondary fluid, liquid overfeed, or DX). The middle panel includes annunciator alarms, instantaneous Coriolis flowmeter readings, the Autodata-10 data logger, and the Metrabyte instrumentation. The Metrabyte is used to transmit signals that control valve positions and pump speeds from the personal computer.

3.2 TEST FACILITY INSTRUMENTATION

3.2.1 Data Acquisition and Control

A data acquisition system and computer are used to control the thermal loading rate, the brine and refrigerant circulation pump speeds, recirculation valve positions, and the condensing temperature, and to collect the data from system instrumentation. The computer allows short sampling times (<30 s) of the instrumentation to provide data for

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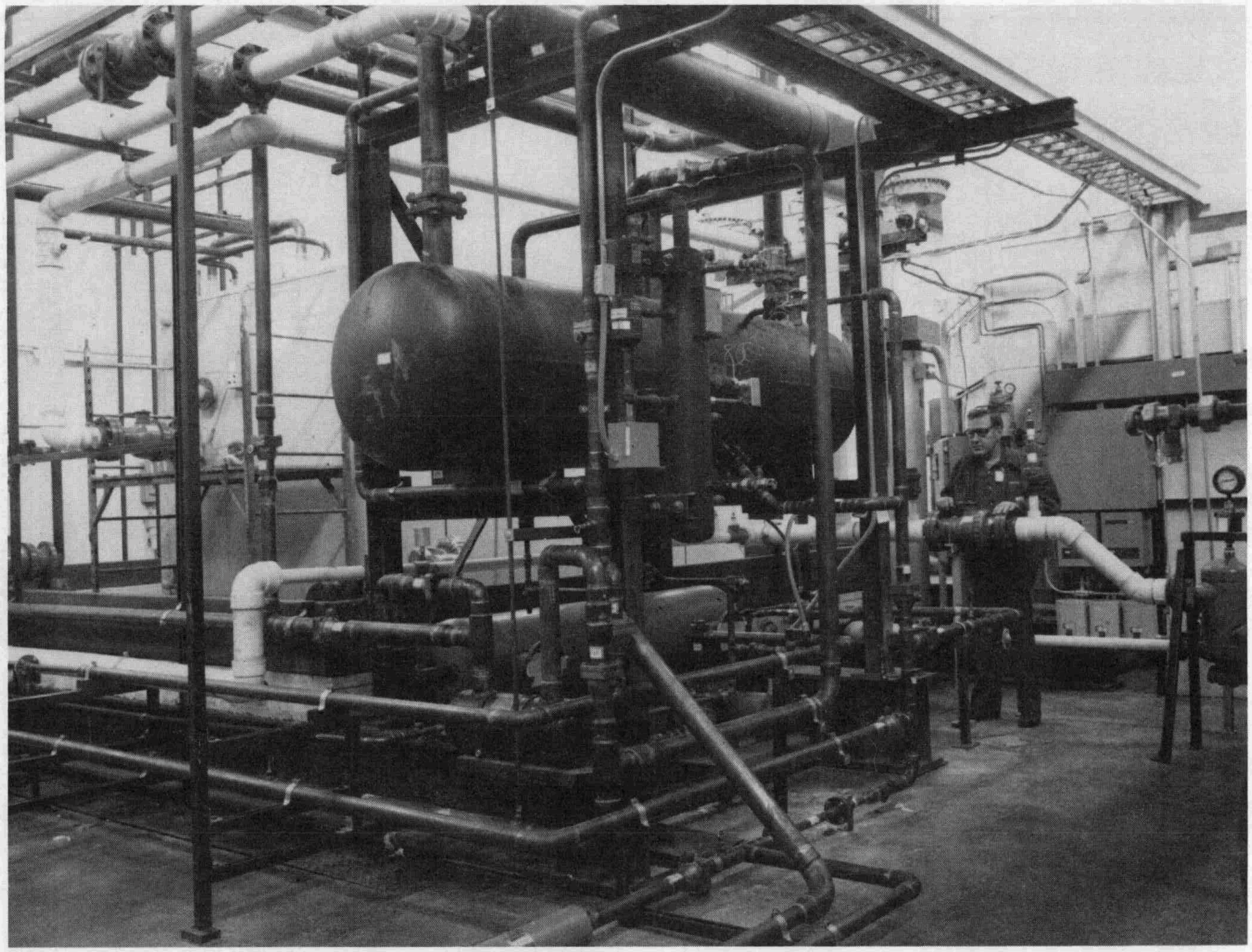

Fig. 3.4. Photograph of ISTF components.

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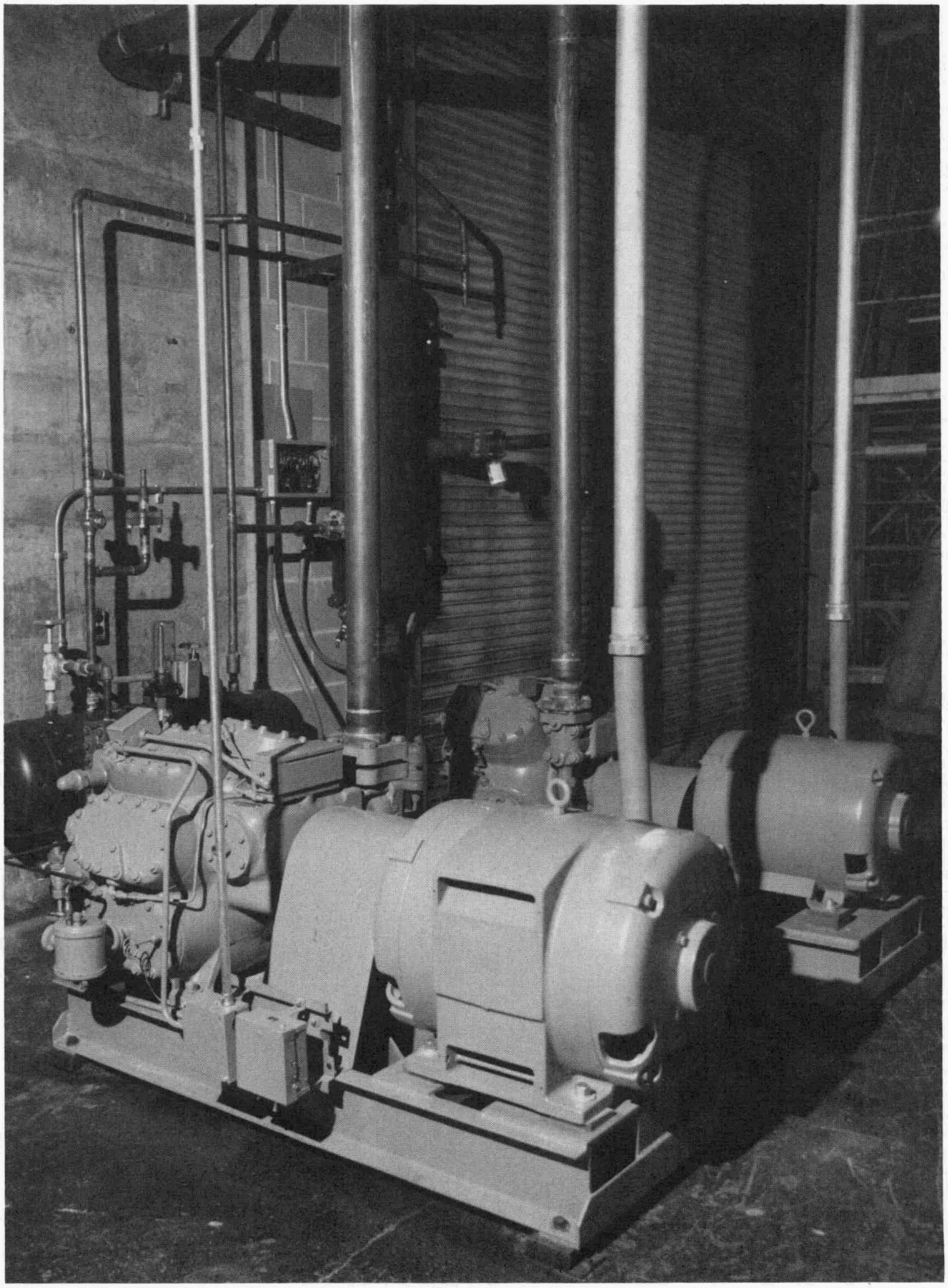

Fig. 3.5. Photograph of ISTF compressors.

ORNL PHOTO 9996-87

Fig. 3.6. Photograph of ISTF control room.

detailed analysis and feedback during transient system operation. Direct controls, outside of the data acquisition/computer system, are available for compressor loading, booster pump operation, and auxiliary portions of the test facility.

3.2.2 Temperature Measurements

Temperature measurements are made by resistance temperature devices (RTDs) with calibration values good to within 0.5°F. These RTDs are checked periodically in an ice bath and against each other. Water and brine temperature measurements are made by insertion of the sensing element within the fluid stream. Refrigerant temperature measurements are made by RTDs bonded to the exterior of the copper piping. Those RTDs bonded to suction piping are covered by insulation.

3.2.3 Flow Measurements

Vortex-shedding flowmeters are used to measure the condenser cooling water flow, the water/brine flow to the heater, the water/brine flow to the ice tank, and the gaseous refrigerant flow to the condenser. These flowmeters are accurate to ±0.8% of the reading for liquid flows and ±1.5% of the reading for gaseous flows.

Liquid refrigerant flows to the evaporator, low-pressure receiver, or ice tank are measured with Coriolis flowmeters that directly measure mass flow. The same type of flowmeter is used to measure the hot gas defrost flow for a dynamic ice maker. A sight glass is positioned to provide a visual confirmation of single-phase flow downstream of the meter. Measurement accuracy is ±1.5% of the measured flow.

Air flow measurements for the agitation air supply are made by a rotameter with a calibrated accuracy of about ±1% of the measured value.

3.2.4 Pressure Measurements

Refrigerant pressure measurements are made with pressure transducers to allow the electric recording of the values. The accuracy of these absolute pressure readings is nominally rated at ±0.11% of full scale. However, the calibration certificates supplied with each transducer show accuracies of ±0.004% or better. Depending on the service location, some of the transducers are rated for 0 to 500 psia and others for 0 to 250 psia. During testing, the pressure measurements are periodically compared with other measurements within the loop and to the expected refrigerant properties.

A differential pressure meter can be used to measure the change in tank water depth during charging. The meter measures from 0 to 10 in. of water, with an accuracy of ±0.5% of full range output (i.e., ±0.05 in. of water).

3.2.5 Oil Concentration Measurement

When testing a DX ice storage system, oil concentration will be measured by sampling the refrigerant flow and weighing the sample using the instrumentation and technique specified in ASHRAE Standard 41.4-75.

3.2.6 Electrical Measurements

Electrical measurements for the compressor power (rated at 40 and 75 hp), circulating pump(s) power (from 2 to 5 hp), agitation air compressor power (1/2 hp), and heater power (0 to 135 kW) are measured by watt/watt-hour transducers. The watt-hour measurements are accurate to (±0.2% of the reading + 0.01% of the rated output)/(power factor). However, the use of this measurement to determine the heater power was unreliable because of the phase angle power controller used to vary the heater power. Heater energy use measurements are therefore based on the fluid flow rate and temperature change.

3.2.7 Cool Stored Measurement

The change in storage medium volume is used to measure the amount of expansion caused by ice formation for ice on coil systems. The quantity of ice formed, along with the sensible heat removed from the storage medium, indicates the quantity of cool stored in, or energy removed from, the tank.

Where possible, a stand pipe is attached to the storage tank, and the differential pressure transducer described in Sect. 3.2.4 is mounted at the storage system manufacturer's recommended initial water level. This permits the data recording system to continuously record the tank water height. When such an arrangement is not possible, other methods of measuring the change in water depth are used, including manual sight tube recordings.

For systems where the ice is free-floating in the storage tank (i.e., dynamic or ice-harvesting machines), no direct measurement of the ice storage inventory is available. However, the rate of ice production can be directly measured if the storage tank access arrangements permit the operator to catch and weigh the ice as it is harvested.

4. COOL STORAGE TEST PROCEDURES

These ice storage system test procedures were developed to test the thermal performance of off-the-shelf ice storage systems or components that can be assembled into a system. In addition, the test procedures are sufficiently flexible so that existing nonice (but latent) systems can be tested as well as prototypical systems under development. Because many of the performance issues summarized in Chap. 2 were both systemic and component specific, the test procedure is designed to evaluate the three components - refrigeration system, storage tank, and discharge system - individually as well as their combined overall performance. The test procedure is structured to determine all of the nominal performance parameters available from ice storage manufacturers including latent and sensible storage capacity (kWh or ton-h), cooling capacity (kW or Btu/h), and compressor suction pressures and temperatures. Additional detailed component-level tests have been structured to resolve the performance issues identified earlier. During these tests, it is important to distinguish between the performance of the manufacturer's system and the performance of test facility components.

The test procedure is structured to test systems operating in the full storage mode of operation. From the individual performance characteristics of the refrigeration system, storage system, and discharge system (or load), the performance of the entire system operating in the demand limiting or partial storage mode of operation can be predicted.

4.1 TEST PROCEDURE FOR DX SYSTEM

A DX ice storage system utilizes the basic components of a mechanical refrigeration system including a compressor, condenser, expansion valve, ice storage tank, and discharge system (Figs. 3.1 and 3.2). The following test sequence is written for DX ice builders. This basic procedure is also used as the basis for flooded systems, dynamic ice builders, and systems charged with a secondary working fluid, with slight modifications described later in this report. Major test categories are Refrigeration System Tests and Storage System Tests. The steps to be followed in each test are detailed below.

Step 1

The system shall be physically examined to confirm that the model submitted for testing corresponds to the model represented in the manufacturer's literature. Tube diameters and spacing shall be measured where possible. Exterior dimensions shall be measured and recorded. When filling the system, the volume of water shall be measured and compared with that listed by the manufacturer. The shipping weight shall be recorded and compared with the manufacturer's data.

Install the system and fill the tank with water to the level specified by the manufacturer. Cover the tank. Measure the ambient temperature and relative humidity of the room weekly.

Mount a water level sensor on the tank so that as ice forms, the change in volume of water/ice can be measured. Calibrate the change in tank height to the change in tank volume by overfilling the tank by the volume change that corresponds to the manufacturer's rated latent storage capacity shown in Eq. (4.1).

$$\Delta V = 0.908 \times CAP \ , \qquad (4.1)$$

where ΔV is the change in volume (in gallons) and CAP is manufacturer's rated latent storage capacity (in ton-h). This equation is based on an ice density of 57.2 lb/ft^3 and a water density of 62.4 lb/ft^3. The change in tank volume shall be recorded for each change of 1/2 in. in water depth because of irregularities within the tank caused by piping and piping supports. Drain the excess water (rechecking the level calibration in the process) to return the water depth to the manufacturer's recommended level.

4.1.1 Refrigeration System Tests

Refrigeration system tests are directed at defining compressor capacity and power use as a function of condensing and evaporating temperatures. These tests shall be made once for the test facility equipment and again whenever the refrigeration system is provided by the storage system manufacturer. To study the operation of the compressor with high evaporator temperatures, a charging sequence will be initiated with the storage tank at room temperature. Sensible heat is withdrawn by the compressor from the tank until the tank reaches 32°F. Refrigerant evaporating and condensing pressures, temperatures and mass flow rates, compressor power, and the cooling capacity will be determined so that compressor curves can be constructed.

Step 2

Begin charging cycle with water well-mixed at room temperature and continue charging under a constant condensing temperature of 105°F. Measure and record refrigerant temperatures TE1, TE2, TE3, TE21, TE6, and TE10; pressures PE1, PE2, PE3, PE11, PE6, and PE10; flow rates FE5 and FE1; and compressor power JE1 and JE6, as indicated in Fig. 3.1, every 5 min until the mixed water temperature reaches 32°F. In addition, measure the flow and temperature change in the condenser water by monitoring FE6, TE19, and TE20 as shown in Fig. 3.2. This measurement is used to corroborate refrigerant-side measurements and to complete the system energy balance.

Step 3

Discharge the system completely by operating the water pump at maximum speed, applying maximum heater power and adjusting the three-way valve at the heater exit so that the water mass flow through the thermal load, FE4, equals FE3 (i.e., no bypass). Allow the discharge to continue until the storage water reaches room temperature.

Step 4

Repeat Steps 2 and 3 with constant refrigerant condensing temperature of 95°F and Step 2 with a condensing temperature of 85°F.

4.1.2 Storage System Tests

Storage system tests are designed to evaluate standby heat gains to the tank from ambient and to characterize the performance of the ice tank/evaporator coil. These tests are summarized in Tables 4.1–4.3 and will apply to every DX unit tested. Note that a few tests are repeated in these tables because it is necessary to check test results for consistency.

Table 4.1. Charge test sequence: DX, liquid overfeed, and secondary loop systems

Test No.	Test duration (h)	Saturated condensing temperature[a] (°F)	Refrigerant flow to storage tank[b] (gpm)	Brine flow to storage tank[c] (gpm
1	6	85	MR[d]	MR
2	6	100	1.25 × MR	1.5 × MR
3	10	95	MR	MR
4	14	85	MR	MR
5	14	100	0.8 × MR	0.5 × MR
6	18	95	MR	MR
7	10	95	MR	2 × MR
8	6	100	MR	MR

[a]Specified for DX tests only.

[b]Specified for liquid overfeed tests only.

[c]Specified for secondary loop tests only.

[d]Manufacturer's recommended flow rates.

Table 4.2. Discharge test sequence: DX, liquid overfeed, secondary loop, and dynamic systems

Test No.	Test duration (h)	TE12 (°F)	TE11 (°F)
1	6	60	38
2	9	60	45
3	12	60	45
4	6	50	38
5	9	50	38
6	12	50	45

Table 4.3. Standby test sequence: DX, liquid overfeed, secondary loop, and dynamic systems

Test No.	Test duration (h)	Initial tank condition
1	>60	Fully frozen
2	>60	Fully frozen

The tank heat gains are evaluated by starting with a fully charged tank of ice and recording the change in ice inventory over a prolonged period of time.

Characterization of the system's thermal performance during charge cycles can be accomplished by determining the saturated suction temperature at which ice is formed under constant compressor loading (i.e., percent of full compressor capacity) and constant discharge pressure. The independent variables in the test will be the compressor loading and saturated discharge temperature, which will be set, and the dependent variables are time, ice formation rate, and saturated suction temperature. To accomplish this objective, the storage tank is completely charged by maintaining constant saturated discharge temperatures at the condenser exit. (In the context of these procedures, completely charged means charged to 100% of the manufacturer's rating as determined by changes in tank water depth. The charge controller or ice inventory sensor must be overridden as a refrigeration system control.) This test will define the saturated suction temperature profile for different average capacities and different saturated discharge temperatures.

On the thermal load side of the system (see Fig. 3.2), the exit water temperature, TE12, and the heater power will be fixed during discharge. Discharge tests will be performed at TE12 equal to 60 and 50°F and thermal loads set to fully discharge the tank in 6, 9, and 12 h.

Step 5. Standby Heat Gain Tests

Standby heat gains shall be determined over a period of at least 2 d with the storage tank charged with an adequate ice inventory so that a significant amount of ice remains at the end of the test (tests 1 and 2). The rate of melting can be measured with water-level sensors described in Chap. 3. The ambient temperature and tank water depth shall be recorded every 1 to 2 h, and the ambient relative humidity shall be measured at the beginning and end of the test. This standby test shall be repeated at least once. This method allows a constant tank temperature (32°F) during the testing period. It also allows losses to be calculated under conditions that would normally occur in the field.

Step 6. Charging Tests 1–8

Initiate a charging cycle with a compressor capacity slightly higher (recognizing that the capacity will decrease during the course of the test) than the manufacturer's rated latent storage capacity divided by the desired test duration (see Table 4.1). In selecting the compressor operating conditions, assume a saturated suction temperature of about 28°F. Compressor capacity control can be accomplished in condensing unit/storage tank packaged systems through control of condensing conditions and, if available, compressor unloading controls. In the ISTF, both discharge pressure and compressor loading can be controlled over a wide range of capacities, as described in Chap. 3. Maintain a constant compressor loading and saturated condensing temperature throughout the test. If the freezing time varies significantly from the desired test duration, adjust the compressor loading and repeat the test. Record refrigerant mass flow FE1 and FE5; refrigerant temperatures TE1, TE2, TE3, TE21, TE6, and TE10 and pressures PE1, PE2, PE3, PE11, PE6, and PE10; compressor energy consumption JE1 and JE6; agitation air compressor energy consumption JE5; condenser cooling water flow FE6 and temperatures TE19 and TE20; and change in tank water height every 5 min during charging. Note the agitation air flow rate and inlet temperature. For ice-on-coil systems, observe and record ice thickness throughout the coil (where possible) every 2 h to characterize ice uniformity. Because oil problems in DX evaporators have been encountered in the field, withdraw refrigerant samples at the evaporator exit at regular intervals for analysis to characterize oil concentration changes. Stop the test when 100% of the storage capacity stated by the manufacturer is reached. Note the reading of the ice inventory sensor supplied with the unit.

Step 7. Discharge Tests 1–6

Select the discharge power, JE10, required by dividing the manufacturer's stated storage capacity (in ton-h) by the desired test duration from Table 4.2. Turn on the water/brine pump and adjust the flow rate such that TE12 is maintained at the desired heater outlet temperature from Table 4.2, and adjust the mixing valve at the heater outlet so that the heater inlet temperature, TE11, is maintained at the desired temperature from Table 4.2. Record the water loop flow rates FE3 and FE4; the temperatures TE11 and TE12 (which are fixed), TE13, and TE14; the thermal discharge energy JE10, the electrical pumping energy JE2 and JE3; and (if agitation is specified during discharge) agitation air compressor energy consumption JE5 at 5-min intervals until TE14 reaches TE11. Note the agitation air flow rate and inlet temperature. At this point, the flow rates measured by FE3 and FE4 should be equal and the temperature TE14 should be equal to TE11. Continue the test, adjusting the pump speed as necessary until the desired heater outlet temperature cannot be maintained. During the discharge cycle, note ice thickness throughout the tank to characterize melting uniformity.

4.2 TEST PROCEDURE FOR FLOODED SYSTEMS

The refrigerant feed in flooded systems is designed to keep the evaporator volume in the storage tank ~60% full of liquid refrigerant, which has the benefit of greater heat transfer from the water in the tank during charging because the coil is wetted on the inside with liquid refrigerant along its entire length. Flooded evaporator systems are classified as either gravity feed or liquid overfeed systems. In gravity feed systems, a refrigerant liquid head on the evaporators is maintained by a surge drum located above the evaporator coils that keeps the evaporator flooded. Refrigerant circulates through the evaporator by the naturally occurring gravity head pressure difference between the evaporator inlet and exit because of the change in refrigerant density as it picks up heat from the tank. Liquid overfeed systems employ a pump to move the liquid refrigerant through the evaporator.

The following test procedure for flooded systems was written for the refrigerant overfeed configurations employing a mechanical refrigerant pump but, in principle, could be modified to test any of the flooded systems. As indicated in Figs. 3.1 and 3.2, a liquid overfeed ice storage system contains many of the same components as a DX system. In the overfeed system, the liquid/vapor mixture exiting the evaporator is separated by a low-pressure receiver with the dry vapor passing to the compressor. The liquid refrigerant pools in the bottom of the receiver for recirculation to the evaporator coils by the refrigerant pump shown in the figures.

The major exception in using the DX test procedure for testing flooded systems is the limited use that can be made of measurements TE9 and PE9. Because the refrigerant is two-phase at the evaporator exit and its quality is unknown, the enthalpy of the refrigerant is also unknown. However, the refrigerating capacity can be determined from the water level (or ice inventory) in the storage tank or by monitoring the low-pressure receiver conditions (including fluid inventory and refrigerant flows, temperatures, and pressures) and the refrigerant temperature, pressures, and flow rate through the compressor(s).

Step 1

Install the system and calibrate the level changes as described in Step 1 for DX system tests.

4.2.1 Refrigeration System Tests

Refrigeration system tests shall be made once for the test facility equipment and again when the refrigeration system is provided by the storage system manufacturer. To study the operation of the compressor with high evaporator temperatures, a charging sequence will be initiated with the storage tank at room temperature. Sensible heat is withdrawn by the compressor from the tank until the tank reaches 32°F. Refrigerant evaporating and condensing pressures, temperatures and mass flow rates, compressor power, and the cooling capacity will be determined so compressor curves can be constructed.

Steps 2, 3, and 4

Collect the data needed for these compressor curves using the procedures described in Steps 2, 3, and 4 for DX systems. Referring again to Fig. 3.1, however, note that TE6, TE21, PE6, and PE11 will not be monitored. Instead, PE7, TE7, JE4, FE2, TE8, and PE8 will be recorded.

4.2.2 Storage System Tests

Storage system tests are designed to evaluate standby heat gains to the tank from ambient and to characterize the performance of the ice tank/evaporator coil. These tests are summarized in Tables 4.1–4.3 and will apply to every liquid overfeed unit tested. Note that several tests are repeated in this table because it is necessary to check test results for consistency.

Evaluation of tank heat gains is done by starting with a fully charged tank of ice and recording the change in ice inventory over a prolonged period of time.

Characterization of the system's thermal performance during charge cycles can be accomplished by determining the saturated suction temperature at which ice is formed under constant compressor loading and constant discharge pressure. The independent variables in the test, including the compressor loading, saturated discharge temperature, and refrigerant flow through the ice tank, will be set; the dependent variables are time, ice formation rate, and saturated suction temperature. To accomplish this objective, the storage tank is completely charged while maintaining a constant refrigerant flow rate to the storage tank. (In the context of these procedures, completely charged means charged to 100% of the manufacturer's rating as determined by changes in tank water depth. The charge controller or ice inventory sensor must be overridden as a refrigeration system control.) This test will define the saturated suction temperature profile for different average capacities.

On the thermal load side of the system, the exit water temperature, TE12, and the thermal load, JE10, will be fixed during discharge. Discharge tests will be performed at TE12 equal to 60 and 50°F and thermal loads set to fully discharge the tank in 6, 9, and 12 h.

Step 5

Conduct standby heat gain tests (1 and 2) as described in Step 5 for DX system tests.

Step 6

Conduct charging tests 1–8 as described in Step 6 for DX tests. The refrigerant pump speed will be selected for each test using the following equation and Table 4.1:

$$FE2 = OR \times CAP \times 200/h_{fg} \,, \tag{4.2}$$

where FE2 is the refrigerant mass flow rate (lb/min) to the storage tank, OR is the manufacturer's recommended overfeed ratio, CAP is the average capacity (tons) during the test, and h_{fg} is the refrigerant heat of evaporation at 20°F (Btu/lb). This equation is based on the manufacturer's recommended liquid overfeed ratio (usually about 3 to 1) and on refrigerant conditions at a temperature of 20°F. The overfeed ratio will vary during the test as the suction temperature varies. Refrigerant samples will not be required for oil analysis for liquid overfeed systems. However, oil return mechanisms for the system should be described in test results.

Record refrigerant mass flows FE1, FE2, and FE5; refrigerant temperatures TE1, TE2, TE3, TE7, TE8, TE9, and TE10; and pressures PE1, PE2, PE3, PE7, PE8, PE9, and PE10; compressor energy consumption JE1 and JE6; air agitator energy consumption JE5; and refrigerant feed pump energy JE4 every 5 min during charging. Note the agitation air flow rate, ice thickness, agitation air temperature, and tank water depth changes during the test. Stop the test when 100% of the storage capacity stated by the manufacturer is reached. Note the reading of the ice inventory sensor supplied with the unit.

Step 7

Conduct discharge tests 1–6 as described in Step 7 for DX systems.

4.3 TEST PROCEDURE FOR SECONDARY LOOP SYSTEMS

In secondary loop ice storage systems, a refrigerant-to-brine heat exchanger (chiller barrel) is located between the refrigeration system and the ice tank so that in practice, the configuration is an ice tank connected to a chiller. A brine (generally a glycol) serves as the working fluid between this heat exchanger, the storage system, and the discharge system shown in Figs. 3.1 and 3.2.

In the charging mode, the brine pump(s) circulates the brine through the deenergized heater, the chiller, the storage tank, and then back to the pump suction. If partial storage tests are desired, the mixing valve following the chiller can be used. During discharge, the heater is energized. As before, the pump speed and mixing valve following the heater modulate to provide the desired heater inlet and outlet temperatures. With this system, several operating options are available including compressor-assisted discharging or partial storage in which the storage tank is a thermal "flywheel" and the refrigeration system operates continuously 24 h/d.

The test procedure for a secondary loop system can be based largely on that for a DX system. The procedure must be modified only slightly to account for the additional heat exchanger and pump electrical energy measurements.

Step 1

Install the system and calibrate the level changes as described in Step 1 for DX system tests. The brine loop must be charged according to the manufacturer's specifications. The volume of brine needed to charge the system shall be compared against the manufacturer's specifications. The brine pressure drop through the storage system will be measured over a range of flow rates that includes the manufacturer's recommended flow rate.

4.3.1 Refrigeration System Tests

Refrigeration system tests shall be made once for the test facility equipment and again when the refrigeration system is provided by the storage system manufacturer. To study the operation of the compressor with high evaporator temperatures, a charging sequence will be initiated with the storage tank at room temperature. Sensible heat is withdrawn by the compressor from the tank until the tank reaches 32°F. Refrigerant evaporating and condensing pressures, temperatures and mass flow rates, compressor power, and the cooling capacity will be determined so compressor curves can be constructed.

Steps 2, 3, and 4

Collect the data needed for these compressor curves using the procedures described in Steps 2, 3, and 4 for DX systems. The data collection will include temperatures TE1, TE2, TE3, TE4, TE5, TE10, TE15, TE16, TE17, TE18, TE19, and TE20; pressures PE1, PE2, PE4, PE5, and PE10; flow rates FE1, FE3, RE4, FE5, and FE6; and energy consumptions JE1, JE6, JE2, and JE3. If air agitation is used, the same agitation data described for DX systems is collected.

4.3.2 Storage System Tests

Storage system tests are designed to evaluate standby heat gains to the tank from ambient and to characterize the performance of the ice tank/evaporator coil. These tests are summarized in Tables 4.1–4.3 and will apply to every secondary loop unit tested; note that many tests are repeated because it is necessary to check test results for consistency.

Tank heat gains are evaluated by starting with a fully charged tank of ice and recording the change in ice inventory over a prolonged period.

Characterization of the system's thermal performance during charge cycles is accomplished by determining the brine inlet temperature to the storage tank at which ice is formed under conditions of constant compressor loading, constant condensing temperature, and constant brine flow rate. The independent variables in the test are the compressor loading, saturated discharge temperature, and brine flow rate (which will be set); the dependent variables are time, ice formation rate, and brine inlet temperature. To accomplish this objective, the storage tank is completely charged. (In the context of these procedures, completely charged means charged to 100% of the manufacturer's rating as determined by changes in tank water depth. The charge controller, or ice inventory sensor, must be overridden as a refrigeration system control.) This test will define the brine inlet temperature profile for different average capacities and brine flow rates. This information can then be used with chiller performance data and compressor curves to predict storage system performance with any specified compressor and chiller package.

On the thermal load side of the system, the exit water temperature, TE12, and the thermal load, JE10, will be fixed during discharge. Discharge tests will be performed at TE12 equal to 60 and 50°F, and thermal loads will be set to fully discharge the tank in 6, 9, and 12 h.

Step 5

Conduct standby heat gain tests 1 and 2 as described in Step 5 for DX system tests.

Step 6

Conduct charging tests as described in Step 6 for DX tests. The brine pump speed will be selected for each test based on the manufacturer's recommended brine flow rate and Table 4.1 (tests 1–8).

Refrigerant samples will not be required for oil analysis for secondary loop systems. However, oil return mechanisms for the system should be described in test results.

Record refrigerant mass flows FE1 and FE5; refrigerant temperatures TE1, TE2, TE3, TE4, TE5, and TE10; pressures PE1, PE2, PE3, PE4, PE5, and PE10; compressor energy consumption JE1 and JE6; air agitator energy consumption JE5; brine pump energy JE2 and JE3; brine flow rate and temperatures FE3, FE4, TE11, TE18, TE17, TE16, TE15, and TE14; and tank water depth changes every 5 min during charging. Note the agitation air flow rate and air temperature to the tank during the test. Stop the test when 100% of the storage capacity stated by the manufacturer is reached. Note the reading of the ice inventory sensor supplied with the unit. If possible, note the ice thickness several times during the test.

Step 7

Conduct discharge tests as described in Step 7 for DX systems. Some secondary loop systems discharge using water from the ice storage tank, and others discharge using the same brine loop used for charging. The ISTF can accommodate both types of systems. In either case, the discharge pumping power and storage system pressure drops should be measured.

4.4 TEST PROCEDURE FOR DYNAMIC ICE BUILDERS

Dynamic ice makers or ice-harvesting machines manufacture ice by direct expansion plate evaporators that are close together. Water is pumped over these vertical plates, and sheets of ice ~0.25 in. thick form and adhere to these surfaces. Periodically (typically at 20-min intervals), ice is removed from these plates by a 20- to 40-s defrosting process in which hot refrigerant gas from the compressor is routed to the evaporator inlet, melting a thin layer of the ice so that the remainder slips away from the metal surface and falls into a holding bin or storage tank placed beneath the ice maker. Because defrosting is accomplished regularly, the maximum ice thickness for each cycle remains the same under constant condensing conditions.

The principal difference between a test procedure for testing ice-harvesting machines and the procedures described for testing static ice builders is due to the charging technique. In the static ice-on-coil machine, the heat-exchange effectiveness of the coil in the storage tank drops throughout the charging process, and the charging system becomes less efficient. In the case of ice-harvesting systems, the heat exchanger effectiveness of the plate evaporators reaches a minimum at the end of each ice-build cycle when the ice has reached its maximum thickness. Thus, in harvesting machines, the real time dynamics of ice formation are more rapid than in static ice builders, and the charging test procedure and data-sampling rate must be modified accordingly.

Step 1

Install the system and perform a physical examination as described in Step 1 for DX tests. Because ice floats freely in the storage tank located beneath a dynamic ice maker, the liquid level will remain unchanged, and no level calibration will be required.

4.4.1 Refrigeration System Tests

Because ice-harvester systems are usually unitary packages, the compressor performance must be evaluated through a series of tests at elevated evaporator temperatures and at selected discharge pressures.

To evaluate the system charging performance, the rate at which ice forms on the evaporator plates must be determined. The total quantity of ice manufactured during an 8- to 12-h charging interval can be determined by allowing the ice and water falling from the ice builder to collect in the storage tank. When charging is completed, the amount of water remaining in the storage tank is determined by pumping or draining the water out of the tank and measuring the volume of water removed. The amount of ice formed during the 8- to 12-h charging interval is simply the difference between the original inventory of water before system charging and that pumped out of the tank following the charging interval. This method can be used to compare the accuracy of capacity measurements based on refrigerant flows, pressures, and temperatures. This freeze, drain, and refill procedure should also be done to measure the amount of ice in the tank at the time the ice-level sensor indicates a full charge.

Step 2

Initiate a charging cycle under a constant condensing temperature of 105°F and with the water well-mixed at room temperature. The defrost cycle should be disabled for the early portion of these tests to better define compressor performance (no ice should form at this water temperature). Measure and record refrigerant temperatures TE1, TE2, TE3, TE5, TE6, TE7, TE8, and TE10; pressures PE1, PE2, and PE10; and flow rates FE1 and FE2, as well as compressor and pumping power JE1 and JE5, as indicated in Fig. 3.3, in as short a time as possible (a maximum time interval of 1 min). Also, measure water flow rates FE6 and FE7 and temperatures TE14, TE19, and TE20. (Note that refrigerant measurements may be impractical because the ice harvester is a unitary package. In this case, condensing performance can be determined through similar measurements on the water side of the condenser.) After ~3 h of charging, measure the cumulative ice formation by draining the tank and measure the amount of water removed. Refill the tank, measuring the amount of water added, and melt the ice in the tank using the heater at maximum power.

Steps 3 and 4

Note the new water volume and repeat Step 2 for condensing temperatures of 95 and 85°F.

4.4.2 Storage System Tests

Storage system tests are designed to evaluate standby heat gains to the tank from ambient and to characterize the performance of the ice maker. These tests are summarized in Tables 4.2–4.4. Note that several tests are repeated in these tables because it is necessary to recheck test results for consistency.

Tank heat gains are evaluated by starting with a fully charged tank of ice and recording the change in ice inventory over a prolonged period of time.

Table 4.4. Charge test sequence: dynamic ice builders

(Defrost cycle time is as recommended by manufacturer)

Test No.	Saturated discharge temperature (°F)	Total cycle time (h)
1	105	MR[a]
2	95	MR
3	85	MR
4	105	0.5 × MR
5	95	0.5 × MR
6	85	0.5 × MR
7	105	2 × MR
8	95	2 × MR
9	85	2 × MR
10	105	1.5 × MR
11	95	1.5 × MR
12	85	1.5 × MR
13	105	MR
14	95	MR
15	85	MR

[a]Manufacturer's recommended cycle times.

Characterization of the system's thermal performance during charge cycles is accomplished by determining the saturated suction temperature at which ice is formed under constant compressor loading and constant discharge pressure. The independent variables in the test will be the compressor loading, saturated discharge temperature, defrost cycle time, and total cycle time that will be set, and the dependent variables are ice-formation rate and saturated suction temperature. To accomplish this, the storage tank is charged over several defrost periods (enough periods so that each plate section in the machine is defrosted at least once) while maintaining constant saturated discharge temperatures at the condenser exit. Several different charging conditions can be tested while charging the tank to the fully charged condition, as determined by the charge controller, or ice inventory sensor. This test will define the saturated suction temperature profile for different compressor discharge conditions under various defrost schedules.

On the thermal load side of the system, the exit-water temperature, TE12, and the thermal load, JE10, will be fixed during discharge. Discharge tests will be performed at TE12 equal to 60 and 50°F and thermal loads set to fully discharge the tank in 6, 9, and 12 h.

Step 5

Repeat the standby heat gain tests described for the DX system (tests 1 and 2).

Step 6. Charging Tests 1–15

Note the charging test schedule shown in Table 4.4. Each test shall last long enough to permit every plate section to go through a complete defrost cycle at least three times to permit visual determination of the effectiveness of the defrost cycle (i.e., to ensure the surfaces are completely free from ice). During each test, record refrigerant mass flow rates FE1 and FE2; refrigerant temperatures TE1, TE2, TE3, TE5, TE6, TE7, TE8, and TE10; pressures PE1, PE2, and PE10; compressor electrical energy consumption JE1; water flow rate and temperatures FE7 and TE13; and water recirculation pump energy consumption JE5 every minute (or more frequently) during charging. Perform a discharge test when 100% of the manufacturer's stated tank storage capacity is reached as indicated by the ice inventory sensor supplied by the manufacturer. The water recirculation rate shall be that specified by the manufacturer. If time permits, a few tests should be repeated at different water recirculation rates to determine the impact of this variable on the system performance.

Step 7. Discharge Tests 1–6

Conduct discharge tests as described in Step 7 for DX systems, referring to Table 4.2 for the test durations and heater inlet and outlet temperatures. Make a visual inspection of the amount of ice remaining in the tank at the end of the discharge cycle. If a significantly large amount remains, drain the tank to determine the volume of the remaining ice.

4.5 CALCULATIONS AND ANALYSIS

Data collected in the test procedure can be used to resolve many of the ice storage performance issues summarized in Chap. 2. The analysis technique is described for DX, flooded, secondary loop, and dynamic systems.

The charging efficiency of any ice storage system is a function of the refrigeration system characteristics and efficiency, and the rate at which ice can be made and stored is a function of refrigerant evaporating conditions.

4.5.1 Refrigeration System Performance

The refrigeration system performance should be analyzed once for the test facility and again when the refrigeration equipment is supplied by the manufacturer. It is important to distinguish between the test loop refrigeration performance and the storage system performance in all analysis and reporting.

From Steps 2 through 4, the relationship between the refrigeration system rate of charging (or capacity in tons), evaporating temperatures (saturated suction temperatures), condensing temperatures, and electrical power can be determined and plotted as shown in Fig. 4.1. The

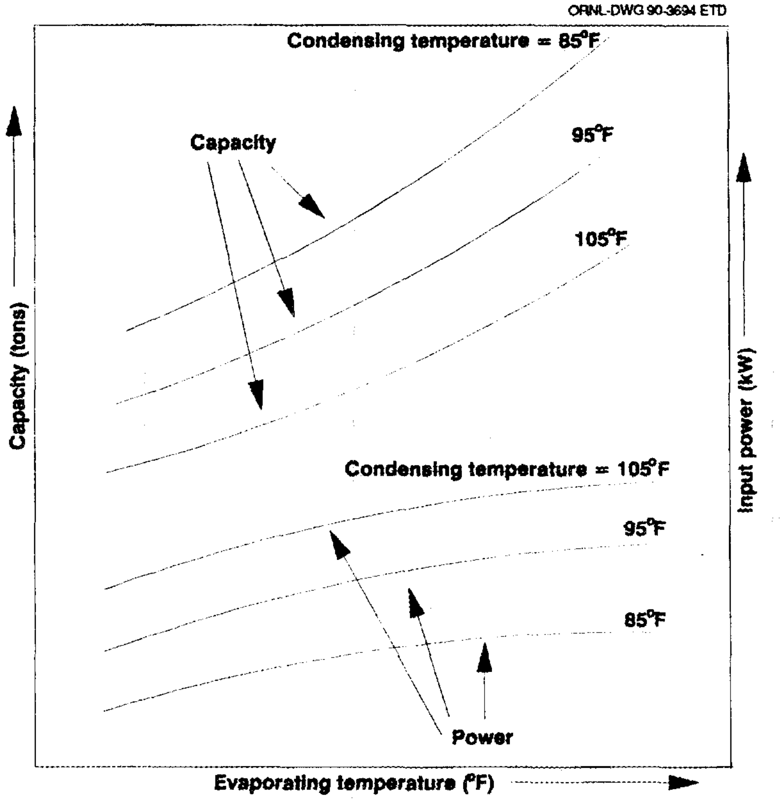

Fig. 4.1. Compressor performance.

capacity can be determined in several ways, depending on the type of system tested. The most straightforward technique would use temperature and pressure to calculate refrigerant enthalpy before the thermal expansion valve and to use PE6 and TE6 to calculate refrigerant enthalpy at the exit from the evaporator. Refrigeration capacity is then the product of refrigerant mass flow (from FE1) and this enthalpy difference. A second technique would use the power to the compressor and the heat rejected by the condenser (both of which were measured). Refrigeration capacity is the difference between these quantities. A third technique would exploit the water displacement sensor on the storage tank to determine the quantity of ice present at any time. The discharge energy required can be used to confirm the other methods. Although these capacity measurements seem redundant, they serve as a check and provide at least one means of measuring any system type where all three techniques might be difficult to apply.

4.5.2 Storage System Performance

Step 5 is used to determine static heat gains to the tank from ambient. The amount of ice that melts over a long period is a direct measure of the standby heat gains. Because the test facility is within a sheltered location, no measurements of heat gains caused by solar radiation in exposed locations will be possible. However, such gains should be considered by any system designer.

Step 6 is designed to determine the performance of the storage tank under dynamic (ice building) conditions. From the data collected in the charging tests, the storage system inlet temperature (saturated suction temperature for DX, liquid overfeed, or dynamic systems and brine inlet temperature for secondary loop systems) can be plotted as a function of storage system inventory and average charging rate, as shown in Fig. 4.2. The rate at which ice is made during charging (refrigerating capacity) can be plotted as a function of time and saturated discharge temperature as shown in Fig. 4.3, which shows the rate normalized relative to the capacity averaged over the course of the test. The compressor performance is inherent in this figure, but it is still useful

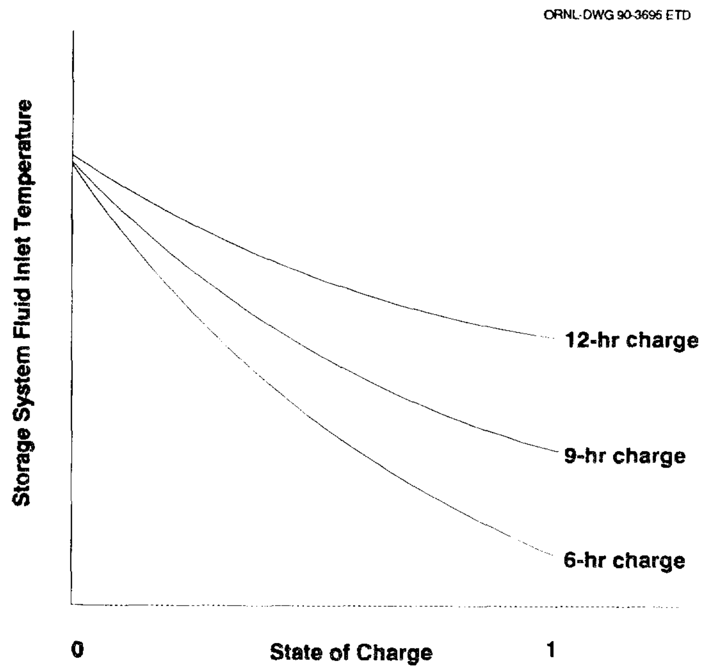

Fig. 4.2. Storage system inlet temperature vs stored energy for different charging rates

ORNL-DWG 90-3696 ETD

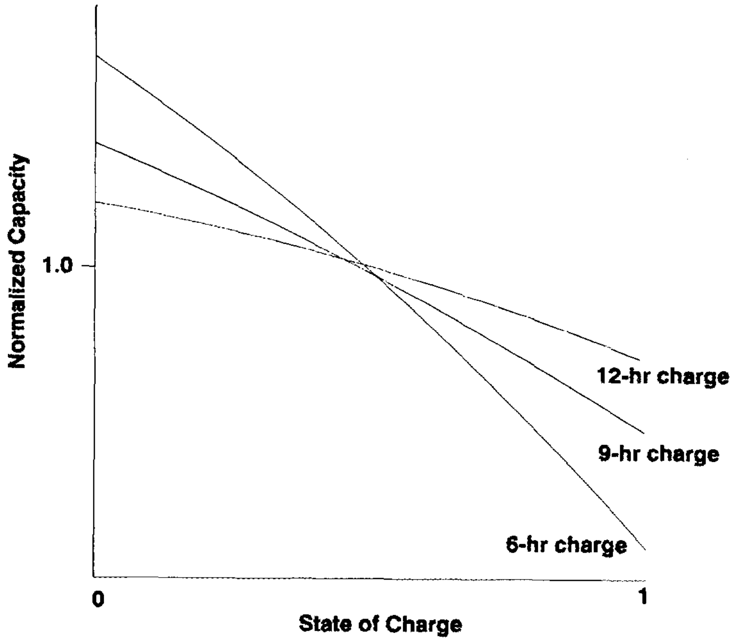

Fig. 4.3. Normalized charging capacity vs stored energy for different charging rates

in giving an indication of the amount of compressor derating that will occur over the course of a charge cycle. A careful application of the temperature data from Fig. 4.2 and the compressor curves of Fig. 4.1, taken together with the expected condensing conditions (and the chiller specifications for a secondary loop system), provide adequate and accurate information for selecting a compressor for a given storage system application. The cumulative capacity can also be plotted as a function of time and average capacity, as shown in Fig. 4.4, to verify that the manufacturer's rated capacity is attainable.

The saturated suction temperature, instantaneous capacity, and cumulative capacity are available at all times during each charging test. If the data points corresponding to the tank being 10, 25, 50, 75, and 100% charged are used, linear regressions can be developed that show the capacity as a function of saturated suction temperature and storage tank state of charge. This relationship can be plotted as shown in Fig. 4.5. This plot is useful for liquid overfeed and DX systems (it will also be useful for brine systems if the evaporator/chiller is part of the storage package). If a manufacturer's compressor curve (as in

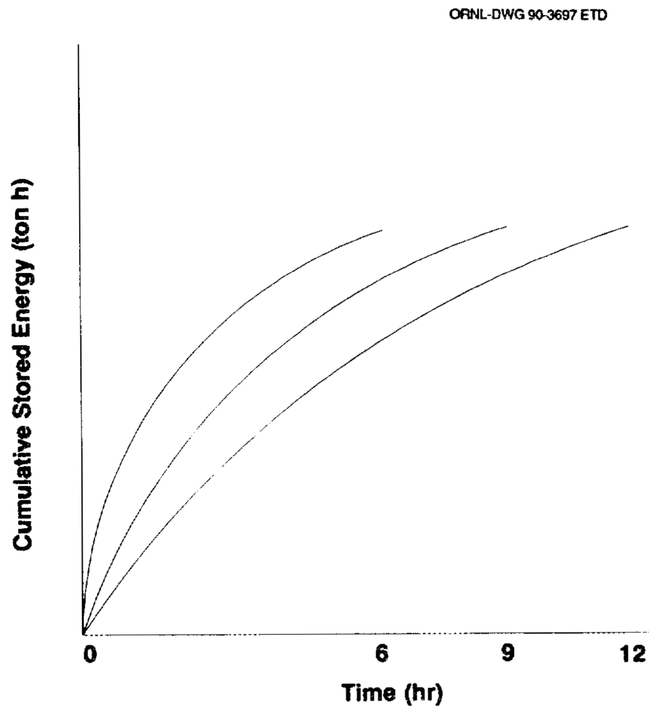

Fig. 4.4. Cumulative stored energy vs time for different charging rates.

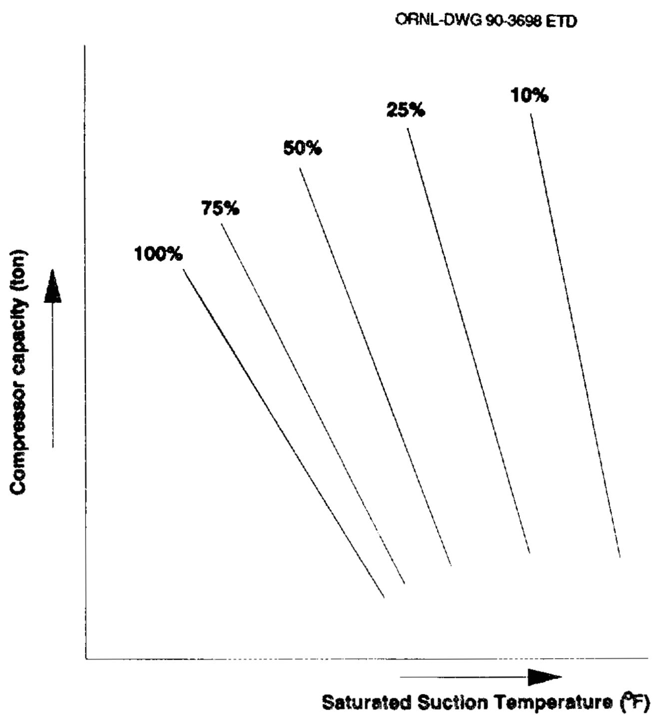

Fig. 4.5. Compressor capacity as a function of saturated suction temperature and state of charge.

Fig. 4.1) for a given discharged pressure is placed on top of this graph, the varying capacity and suction conditions during a charge cycle can be approximated by the intersections of the manufacturer's curve and linear regressions developed from the test data. The average capacity for that given compressor under ice-making conditions can then be estimated. Figure 4.5 also helps to show that the tank charges slightly faster in the beginning of a charge cycle because that is when the ice layer on the coil is thinnest and the suction temperature higher.

A somewhat similar procedure is used to analyze data taken during discharge testing. All discharge data were taken under conditions of (1) fixed load, (2) fixed exit-water temperature from the load and, (3) a minimum acceptable entering chilled-water temperature to the load. Although, under these conditions, ice may remain in the tank at the end of discharging (where the test end is defined by the fixed exit-water temperature from the load), it is of no use. That is, the "cool" remaining in the tank at the end of discharging is thermodynamically unavailable. The method used to help analyze the discharge data is illustrated in Fig. 4.6. The discharge rate (tons) was set to three levels so the tank would be discharged in 6, 9, and 12 h, yielding the three dotted lines in Fig. 4.6 for TE12 set at 60°F. The time at which the allowable TE11 was exceeded during discharge is used to plot the solid curves in Fig. 4.6. Presented in this way, the data show that as

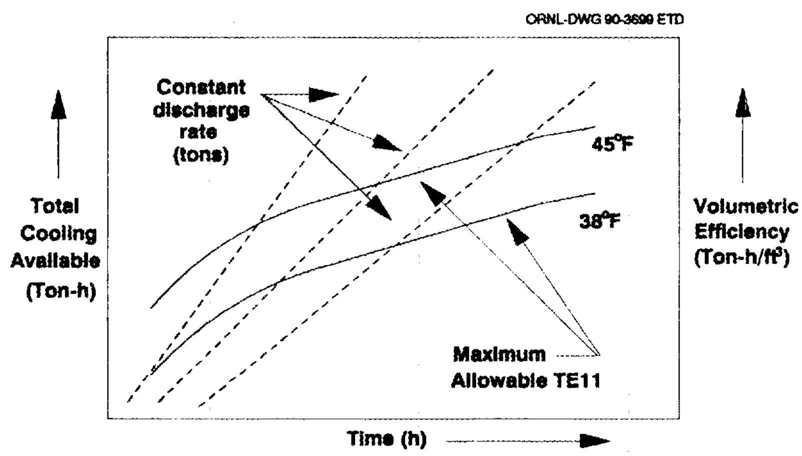

Fig. 4.6. Discharge characteristics of cool storage system, TE12 = 60°F.

the permissible temperature to the load, TE11, is lowered, the usable capacity of the storage system is also lowered. With some systems, this will only eliminate a portion of the sensible storage; for others, it could include a portion of the latent storage. If the vertical axis of Fig. 4.6 is divided by the storage system volume, this same plot will reveal the volumetric efficiency (ton-h/ft^3) of the storage tank for the varying discharge conditions.

It is important to distinguish between the energy stored as latent energy, sensible energy of fluids within the storage tank, and sensible energy in the piping loop's liquid inventory (for secondary fluid systems). The sensible energy in the loop's inventory is available if the discharge cycle starts immediately after the charge cycle is complete. If the system rests for a period of time, however, the piping contents will come to ambient conditions. Therefore the available energy of the piping contents will depend on the operating schedule, the most recent charging fluid temperature, and the volume of the piping system. Because each building will have a unique piping system design and operating schedule, the tests should only account for energy storage within the boundary of the storage tank.

The storage tank outlet temperature should also be plotted vs time and cumulative discharge energy (or percent frozen) to show typical temperature profiles for the various discharge rates. The shape of these profiles will vary according to way the ice is stored and the way the discharge fluid is circulated. The cumulative discharge energy should be plotted as a function of time and constant discharge capacity to verify that the manufacturer's rated capacity is available (note the dashed lines in Fig. 4.6).

The data can also be used to determine the global parameter of kilowatt-hour per ton-hour. The total electrical energy input including that from the compressor and pumps(s) during charging can be compared with the total cooling provided during discharge (in ton-h) to determine this parameter. This presentation is relatively straightforward for unitary systems. However, for storage systems tested with the test facility refrigeration system, it will be necessary to distinguish between the various system components and to identify each component's efficiency.

As stated in Chap. 2, knowledge of average refrigerant suction temperatures during charging is a major performance issue. Although the preceding techniques for describing system performance accommodate the fact that suction temperatures drop, it will also be useful to plot the saturated suction temperatures as a function of time during charging. From this plot the time-averaged charging suction temperature can be determined and compared with that stated by the manufacturer. For secondary fluid systems, the brine inlet temperature profile should be used.

All system parasitic energy requirements and heat exchanger penalties have been measured and should be evaluated relative to the capacity of the storage system. Data were taken on all electrical inputs, temperatures, and flows through system heat exchangers so that estimates of their effects on total system efficiency can be made.

The test procedure was structured to test individual cool storage systems and subsystems in detail. To reduce the number of independent variables in any charge test, some parameters had to be known beforehand or held constant. The ice inventory was one of these. In each complete charge test, the tank was charged to 100% of its capacity as stated by the manufacturer. This state of charge for testing purposes was always the same for a particular system. The ice storage inventory sensor was not used to control the amount of ice stored because this would have introduced another variable into the test. The ice inventory sensor performance can be determined by noting the time that charging *would have* terminated if it had been connected. From this time measurement and with knowledge of the rate of ice formation from earlier analyses, the over- or undercharge precision can be determined for the ice inventory sensor.

Maintaining adequate compressor lubrication throughout charging for DX systems was another performance issue cited in Chap. 2. Chemical analysis to determine oil concentration in the refrigerant from samples collected periodically on each side of the evaporator should be made. The rate at which oil pools in the evaporator during charging can then be determined and reported.

REFERENCE

1. J. J. Tomlinson et al., *Commercial Cool Storage Laboratory Testing, Draft Final Report*, EPRI Research Project 2036-11, May 1985.

APPENDIX

EPRI sponsored a survey of cool-storage system manufacturers that was reported by Tomlinson et al. in 1985.[1] The results of that survey demonstrated the need for an ice storage test facility and helped to define the scope of the testing program.

BALTIMORE AIRCOIL COMPANY, INC. (BAC)

BAC, founded in 1938, specialized in design and production of heat transfer equipment for industrial and commercial refrigeration, air conditioning, and industrial processes. In 1981 they began to manufacture ICE CHILLER™ thermal storage units and currently offer cool storage units in 22 standard sizes from 145 to 1200 ton-h (12,000 to 100,000 lb of ice, respectively). The BAC ice chiller business is relatively minor but is expected to grow.

The basic ice builder system consists of an insulated rectangular tank containing evaporator tubes arranged in a serpentine pattern. The tubes are unfinned, and ice is grown as a log surrounding the tube. The BAC ice builders are static systems, as opposed to dynamic, where the ice is stored remote from the evaporator. The inner tanks walls are made of reinforced heavy-gage, hot-dipped galvanized steel. The tank walls are covered with expanded polystyrene insulation 3 in. thick on tank sides and ends (R-13) and 2 in. thick on the bottom and top (R-8). The exterior is covered with thinner-gage, hot-dipped galvanized steel. The skin and insulation are supported by a steel framework of boxed sections to resist the forces resulting from the static water pressure in the tank. The serpentine evaporator tubing is encased in a steel frame for support, pressure tested to 350 psig, and galvanized as a fabricated unit to ensure protection from corrosion. The state-of-charge (amount of ice present) is determined by an ice-thickness sensor mounted on the upper branch of a single tube to give a point measurement of ice thickness and is noted by BAC as one of the weaker aspects of the system. Also, such point measurements provide reasonable performance during ice manufacture but do not work well during cool storage discharge. The problems from this type of sensor are manifested especially during midseasons (spring and fall), where the ice bin is infrequently totally charged or discharged. A water-level sensor is also being used by BAC. This device is a liquid-level sensor that, over a 5-in. liquid level change, yields a 4- to 20-ma output signal. Although this sensor itself appears to work well according to BAC, other problems have been experienced with liquid-level sensing as a technique to determine ice inventory. Condensation and evaporation of water, plus charging inventory of water in the chilled water piping distribution system in a building, all affect the tank water level.

BAC indicated that a typical refrigerant temperature at the exit of the evaporator coil is 18.5°F during ice production, although Table 1 in the BAC bulletin S140/1-0 indicates that the evaporator temperature can be designed from 12 to 24°F (the latter figure is for an undersized ice chiller unit with an ice-build time of 14 h or more.

The tank exterior height on all units is fixed at 7 ft 9 in., and the tank width is either 5 ft 3 in., 7 ft 5 in., or 9 ft 11 in. Within these parameters, the required water capacity (up to 18,830 gal) is determined by the length of the tank (up to 41 ft 5 in.). If larger single-tank installations are needed, BAC indicates that a field-erected concrete storage tank with their evaporator tubes inside may be the economic choice.

THERMAL ENERGY STORAGE INCORPORATED (TESI)

TESI has been in the storage business for several years, manufacturing heat-storage systems based on the latent heat of fusion of certain hydrated salts. These systems have been successfully marketed by TESI in the southern California area, where TESI is located. Approximately 2 years ago, TESI entered the cool storage field with a unique system based on the R-11 clathrate as the storage medium. The system consists of a closed vertical tank containing a copper heat exchanger at the top for charging and another at the bottom of the tank for discharging. The tank is partially filled with a mixture consisting principally of water and the low-pressure refrigerant R-11. The upper heat exchanger is a direct expansion coil from the chiller, which, during the charging process, extracts heat from the storage vessel. As heat is extracted and "cool" is stored, the water/R-11 mixture cools to 47°F, at which point the clathrate forms. Put simply, the clathrate can be thought of as a "warm ice" because it is composed principally of water. The crystalline structure formed with the R-11 is very close to that of pure ice. The latent heat of fusion of this material is nominally 120 Btu/lb. A brine circulating through the lower coil and fan coil units in the building is used to discharge the system and melt the clathrate formed.

To date, TESI has manufactured three of these units: one for Southern California Edison, one for San Diego Gas and Electric (SDG&E), and one for TVA (scheduled for delivery in February 1985). The sizes range from 30 ton-h for the SDG&E system to 60 ton-h for the TVA system. A 1000-ton-h system is under study because TESI has determined from utilities that 100- to 200-ton cooling loads are frequently found in buildings. TESI feels that the 1000-ton-h system will be their mainstay system in the future.

All testing done on the clathrate cool storage system has been global in the sense that only cooling capacities for kilowatt-hour in and ton-hour out have been measured. To do this, a 22-ton-h device was charged with a 5-ton condensing unit, and the above parameters were measured. Standby losses were calculated (not measured), and the literature value of specific latent heat capacity of the clathrate was used. Discharge tests consisted of placing a constant heat flux on the discharge circuit and measuring the time taken to discharge the unit. Victor Ott, a company representative, indicated that some additional data have been taken by Southern California Edison that consist of heat exchanger temperatures, entering and leaving fan coil temperatures, and system parasitic power needs.

TESI sees several outstanding performance issues facing current designs. In many cases, the ratio of kilowatt-hour to ton-hour is unknown and needs to be measured. Suction temperatures and pressures in ice-on-coil systems is often unknown, as well as the effect of reduced pressures and temperatures on system performance.

TURBO ICE-MAKING HEAT PUMPS

The Turbo ice-making heat pump is used for the building heating and cooling market. Development was initiated in 1975, and the first units were sold in 1977. The unit is an ice-harvesting type that uses a hot-gas bypass to periodically release sheets of ice from the evaporator plates. The condenser heat is removed in the form of hot water up to 130°F and can be used for space heating and/or for domestic hot water. The ice in this operation can be used for air conditioning during the summer months. The heat pump can also function as a water chiller. The evaporator coils are multiple stainless steel plates that are mounted above an insulated storage tank. During the charging process, water is pumped from the tank over the plates and frozen into sheets ~1/4 in. thick on each side. Turbo noted that keeping the ice this thin helps maintain evaporating temperatures at ~20°F and charging efficiencies high. Ice-build time is set for 20 min, and it takes 20 to 40 s for the ice to be harvested. During discharge in the cooling mode, the cooling load melts the ice. In the heating mode, the ice continues to build as the condenser heat is exploited. An ideal application is one in which heating and cooling are required simultaneously.

Turbo manufactures the ice-making heat pumps in eight sizes from 7 to 80 tons of refrigerating capacity so that in a 12-h charge period, a maximum of 1000-ton-h storage capacity is available. There are 144 rectangular evaporator plates in the 80-ton model, each with an area 10 ft^2.

CHESTER-JENSEN COMPANY

Chester-Jensen has been manufacturing cool storage systems for 25 years and selling them for 35 years. Known originally as the Chester Dairy Supply Company, they sold and installed ice systems that were used primarily for chilling milk quickly to inhibit bacteria growth. Chester-Jensen states that their involvement in the HVAC field with cool storage systems is relatively new and is expected to grow. Although cool storage for the dairy industry, as well as cool storage applications in buildings, is one of the major product lines for Chester-Jensen, they also manufacture chillers and plate heat exchangers, both of which are used in cool storage heat pump systems.

Icetech, a spinoff division of Chester-Jensen, was formed to promote cool storage development and sales. All Chester-Jensen ice systems are made and tested at Cattaraugus, New York.

The basic Chester-Jensen system is a static ice builder that forms ice in logs around tubing coiled inside a tank. Air agitation is provided by a single sinuous, stainless steel, perforated tube that permits air to be distributed throughout the water in the unit. Air agitation is recommended during charging and discharging operations. Agitation during charging builds a thicker, denser ice. The storage tanks are constructed with liners of heavy steel plate at the sides, ends, and bottom. Thermal insulation is provided by layers of fiberglass and styrofoam, 3 in. on the sides and end and 2 in. on the bottom. The ice-building coils are 1- or 1.25-in. schedule-40 steel pipe welded at the seams. Capacities range from 1000 to 100,000 lb of ice stored (12 to 1200 ton-h) in 43 sizes. An ice-thickness control, based on the thermal resistivity of ice, limits ice formation to a nominal thickness on the coils of 2.5 in. Performance data on these ice storage systems were unavailable.

SUNWELL ENGINEERING COMPANY

Sunwell was established in 1978 in Concord, Ontario, for the purpose of manufacturing cool storage systems for processing and HVAC applications. The major market product is an ice slush system for the fishing and poultry industries; the secondary market, which Sunwell hopes will grow, is commercial cool storage in buildings. The Sunwell system represents a new approach to ice storage by using a binary mixture consisting of a 3 to 5% ethylene glycol solution in water as the cool storage medium and the cooling process occurring in an agitated heat exchanger (turbulator) external to the storage tank. The ethylene glycol is a freezing-point depressant; so, as the solution is cooled to

~27°F in the turbulator, ice crystals are formed and remain dispersed throughout the remaining fluid. The slush is pumped to a storage tank where the ice crystals accumulate, thereby storing coolness. System discharge is accomplished by circulating the slush from the tank to a fan coil unit in the building. The Sunwell system is marketed as a package containing the condensing unit, the turbulator, the pumps, and the controls. The storage tank and associated insulation are separate items not manufactured by the Company but rather specified by the design engineer. System sizes range from 5 to 40 tons in 5-ton increments at capacities up to 1500 ton-h.

A moderate level of system testing is being performed by Sunwell. The performance of the condensing unit used for charging tests is taken from manufacturer-supplied data rather than from a Sunwell test procedure. Heat removed from storage, however, is measured on the binary solution side of the turbulator; therefore, a rough indication of charging efficiency can be determined for current condensing conditions. In the Sunwell brochure, saturated suction temperatures are quoted as 15°F throughout the charging process. However, as the ice crystals form, the remaining liquid in the slush will become more highly concentrated, and the freezing temperature will continue to drop, resulting in lower and lower evaporator temperatures and lowered performance. The technical specifications given in the Sunwell data sheet are brief, giving nominal compressor horsepower, charging evaporator temperature, ice-build time, and available ton-hours for each of the 16 Sunwell models.

Sunwell confirmed that charging suction pressures and temperatures differ from one manufacturer to another. Thus it is difficult to determine the performance of an ice storage design. An otherwise marginally performing storage unit may be compensated for by a high-efficiency condensing unit or very effective plate heat exchangers. Other cool storage issues noted by Sunwell were the effect of storage temperature (in their case 27°F) on system performance and, more important, the effect of available discharge temperature on overall storage performance.

MIDWESCO CORPORATION

Midwesco manufactures pressurized closed-circuit ice storage vessels. The ice-building evaporator coil is enclosed in an insulated, large-diameter pipe that contains the chilled-water pressure. The thermal storage vessels, called Perma-Ice, are manufactured by the Perma-Pipe Division of Midwesco, Inc., which is located in Niles, Illinois. The closed-system ice storage vessels can provide lower operating temperatures and enhanced ice storage capacity because they are sealed from the atmosphere. The fully insulated vessels also permit the use of smaller piping and ductwork than required in open-system ice banks.

The Perma-Ice pressurized-ice unit builds ice ~1.7 in. thick at a suction temperature of 25°F. Each vessel is 45 ft long, and the enclosed ice coils are 1 1/4 in. in diameter. The vessels are insulated with 2-in.-thick, closed-cell urethane foam insulation with an R-value of 15. Each vessel is covered with a corrosion- and abrasion-proof filament-wound, fiberglass-reinforced, polyester resin jacket.

The vessels are suited for direct burial or aboveground installations. Each vessel has a capacity of 220 ton-h. Units are typically piped together for larger storage capacities. The units are equipped with integral supports to facilitate vertical stacking. Operation of the ice storage Perma-Ice system does not require an air agitator, a secondary pumping loop, or water treatment and makeup as do some open-system ice banks.

NATIONAL INTEGRATED SYSTEMS (NIS)

Since 1951, NIS has been manufacturing custom ice storage systems used principally for church cooling. They estimate that 300 cool storage systems designed by NIS are in place and operating in Texas, where the company originated, and in California, their current location. The system design has evolved and improved over this period to include fiberglass storage tanks, refrigerant DX into copper coils supported by PVC fixtures, and a computerized ice-inventory sensor. A 100 ton-h system was the smallest installed and a 7200 ton-h the largest. Note that each system is custom-designed for a particular installation and that there are no "off-the-shelf" systems available. Bud Kennedy, principal owner of NIS, indicated that there are several outstanding performance issues facing current designs. Electrically operated expansion valves actuated by thermisters downstream of the condenser would provide positive superheat control and, he estimates, would result in a 2% system efficiency gain alone. He also indicated that use of compressed air for agitation during ice manufacture is inefficient because extra power is required not only for operation of an air compressor but also for removing the heat of compression from the air that is bubbled through the water being frozen in the storage tank. Although Kennedy has indicated that the controls are less of a problem in NIS installations, development of effective ice inventory (thickness) controls are needed.

CALMAC MANUFACTURING CORPORATION

Calmac has been in the field of heating and cooling since 1947, and began marketing ice-storage systems for cooling buildings 5 years ago. The Calmac ice banks use standard packaged chillers with a conventional shell-and-tube evaporator and a glycol-based brine. The brine freezes ice solid in a high-density polyethylene tank with a patented temperature-averaging counterflow coil of closely spaced inexpensive flexible plastic tubes. The advantage of this type of coil is that ice is built uniformly throughout the tank, and water does not become surrounded by ice during the freezing process and can move freely, preventing stress on the tank. In this system no agitation is required. The tanks are 3/8 in. thick and insulated with 1 in. of polystyrene and 1 in. of polyurethane on the sides, and 2 in. of polystyrene on the bottom. The counterflow heat exchanger tubes are 3/8-in. ID and, in the most popular tank, 1.6 miles long if uncoiled. The heat exchanger is used to both charge and discharge the system. One advantage of this type of system is that ice is formed and melted from the same surface; after a discharge cycle, ice begins to form on a clean surface. Calmac manufactures three models of ice storage tanks with latent capacities of 51, 76, and 85 ton-h. Installations requiring more cool storage capacity require multiple tanks.

Calmac finds the largest market for cool storage to be in the 25- to 30-ton range, and multiple-tank systems are being installed. Calmac indicated that the generic problems with ice storage systems in general are low evaporator temperatures (leading to condensing unit derating), water channelling during discharge so that all of the ice formed cannot be used, and ice inventory controls that are not accurate or reliable.

GIRTON MANUFACTURING

Girton has been manufacturing ice storage systems since 1938 for the processing industry and in 1974 entered the ice storage market for building HVAC applications. The Girton "King Zeero" ice builder, originally designed for the dairy industry, is now being marketed under the trade name "Caloskills-Thermaster" for building cooling or heating. The Girton unit is a static ice builder constructed of 3/16-in. steel plate and insulated with 2 in. of cork if used for cool storage only, or with 6 in. of foamed-in-place polyurethane if the tank is used to store ice as well as hot water during the heating season. The evaporators are coil-plate fixtures with baffles to create internal water passages and induce turbulence as water flows through the passages. This type of system is advertised as eliminating the need for air agitation and apparently reduces thermal stratification when the tank is used in the heating mode. Each evaporator circuit has a separate expansion valve branching out from a common inlet header. There are 20 models available ranging from 48 to 540 ton-h. Nominal cool storage ratings are done at 2 1/2 in. of ice on each side of the evaporator plate.

Most of the systems installed are the DX type, although liquid overfeed systems have been provided. Girton indicated that ~9000 of their units have been installed in industry over the life of the company and that most of the problems with their cool storage systems have been solved. As a result, their current testing program is small, and they indicated a willingness to participate in a program of independent laboratory testing of their units.

TDS, INCORPORATED

TDS is a company in Florida that manufacturers an ice-on-coil system for building cooling. Two sizes are currently being manufactured: a 66- and a 51-ton-h system. The storage tank is cylindrical and made of fiberglass-reinforced polyester resin with 2 in. of foamed-in-place urethane insulation sandwiched between the tank interior and a 0.020-in. vinyl liner. The evaporator is constructed of 3/8-in. copper tube formed into 60-ft flat spirals spaced 3 in. apart. The spiral evaporator circuits are fed with a multiple outlet distributor from a wide-range thermostatic expansion valve. The evaporator is designed for 20°F and can be used to heat water in the tank during the winter when operated in a reversible cycle. The basic system requires a 7 1/2-hp condensing unit operating in the DX mode that drives a single tank. Larger-capacity needs are accommodated by using multiples of this basic system. Properly controlled, a multiple-system installation will operate efficiently because charging can be directed to the unit having the least amount of ice. TDS has indicated the need for cool storage system testing and is willing to participate in such a program.

OTHER AVAILABLE SYSTEMS

There are several manufacturers of large ice storage systems currently being used. Continental Equipment Corporation produces an air-agitated, DX ice builder that has been used for cool storage in buildings. The tank is of conventional rectangular construction and contains steel pipe evaporator coils. Chicago Bridge and Iron now manufactures a unique ice-building system that forms ice on a smooth-surface evaporator so that as ice is formed, it slides off into a storage tank.

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