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Test Procedures for Rating Residential Heating and Cooling Absorption Equipment

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
Center for Building Technology
Building Equipment Division
Washington, DC 20234

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NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

ABSTRACT

Test and rating procedures are presented for gas-fired absorption devices operating in either the heating or cooling modes. These procedures are designed to include the effects of part-load and cyclic operation, variations in outdoor temperature, and frost formation during the heating mode. Both air-source and ground water source absorption heat pumps are considered, as well as air cooled and ground water cooled air conditioners and water chillers. A calculation procedure is presented for estimating the heating and cooling seasonal performance and cost of operation of residential water chillers, air conditioners, and heat pump units.

Key words: Central air conditioners; central heating equipment; heat pumps; heating seasonal performance; cooling seasonal performance; rating procedure; seasonal cost of operation; test method.

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NOMENCLATURE

$BL(T_j)$	Building load at an outdoor dry-bulb temperature T_j, kW (kBtu/hr).
C	Cooling.
C_D	Degradation factor for cyclic operation, defined by equation (5.13).
C_e	Cost of electricity in \$/Whr.
C_f	Cost of fuel in \$/Btu.
CLF	Cooling load factor, defined as the ratio of the total cyclic cooling done in a complete cycle or specified period consisting of an 'on'-time and an 'off'-time to the steady-state cooling done over the same time period at constant ambient conditions. See equation (5.11).
CLH	Cooling load hours, defined as the number of hours in a cooling season that a building requires cooling. See Table 4.
COP	Coefficient of performance, defined as the net heating done over a specified period of time divided by the total electrical energy and fuel energy input over the same time interval.
C_p	Specific heat (heat capacity) kJ/kg °C (Btu/lbm °F).
C_{pa}	Specific heat at constant pressure of air-water mixture per pound of dry air.
CSPF	Cooling seasonal performance factor, defined as the ratio of the total cooling done to the total energy usage over a cooling season.
DHR	Design Heating Requirement (steady-state heating capacity at outdoor design temperature, T_h), kW (kBtu/hr).
E	Total energy consumption during entire season kWhr, (kBtu).
$\dot{E}_{ss}(T)$	Steady-state total energy input at a given water temperature T, kW (kBtu/hr).
$E(T_j)$	Total energy consumption at an outdoor dry-bulb temperature, T_j. kW (kBtu/hr).
h	Heating.
HHV	Higher heating value of fuel on a mass basis.

HLF	Heating load factor, defined as the ratio of the total cyclic heating done in a complete cycle of specified period consisting of an 'on'-time and 'off'-time to the steady-state heating done over the time period at constant ambient conditions. See equation (8.10).
HLH	Heating load hours. See Table 7.
HSPF	Heating seasonal performance factor, defined as the ratio of the total heating done to the total energy usage over a heating season.
j	Outdoor dry-bulb temperature bin number. See Tables 4 and 7.
$\dot{m}_f$	Fuel mass flow rate. See equation (5.1).
N	Total number of temperature bin hours.
n_j	Number of temperature bin hours in a particular bin.
n	Number of non-zero temperature bins.
$\frac{n_j}{N}$	Fractional number of temperature bin hours. See Tables 4 and 7.
PLF	Part-load factor, defined as the ratio of the cyclic COP to the steady-state COP, see equation (5.14).
P_t	Total amount of electrical energy being supplied averaged over the test duration, kW (kBtu/hr).
$\dot{Q}_{cyc}$	Cyclic total capacity defined as the ratio of the total cooling done over a given time period to the duration of time the burner is on in that period, kW (kBtu/hr).
$\dot{Q}_f$	Residual energy in the products of combustion (flue gas) leaving the system, kW (kBtu/hr).
$\dot{Q}_j$	Jacket heat loss representing the convective and radiative losses from heated metal surfaces, kW (kBtu/hr).
$\dot{Q}_{sc}$	Air flow rate across the condenser and absorber coils, kg/hr (lbm/hr or CFM).

$\dot{Q}_{ss}(T)$ Total steady-state cooling or heating capacity at a water temperature T, kW (kBtu/hr).

$\dot{Q}_g$ Rate of thermal energy supplied to the generator, kW (kBtu/hr).

$\dot{Q}_{ss}$ Steady-state capacity, kW (kBtu/hr).

RH Auxiliary electric resistance heating, kW.

$RH(T_j)$ Resistance heat energy usage in temperature bin, T_j, kWhr, see equation (8.15).

SOC Seasonal Operating Costs of direct-fired absorption systems.

SPF Seasonal performance factor, defined as the ratio of the total cooling or heating done to the total energy over a cooling or heating season.

T_c specified outdoor change-over temperature.

T_h Outdoor design temperature (also T_{OD}).

T_j Representative outdoor dry-bulb temperature for temperature bin j, °C (°F). See Tables 4 and 7.

t Time, hours.

$\dot{V}$ Indoor air flow rate, m^3/s (CFM), at the dry-bulb temperature, humidity ratio, and pressure existing in the region of measurement.

v_n Specific volume of air-water mixture, at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flowrate, m^3/kg (ft^3/lbm).

W_n Humidity ratio, (the 'n' means at the nozzle [i.e., point of measurement for air-flowrate]). kg/kg (lbm/lbm).

SI CONVERSION FACTORS

MULTIPLY	BY	TO OBTAIN
Btu/h, Btuh	0.293	W
Btu/lbm°F, [C_p, specific heat]	4.19	kJ/kg°C
°F	°C = (°F - 32)/1.8	
ft	0.3048	m
ft/min, fpm	0.00508	m/s
ft^3/lbm	0.0623	m^3/kg
ft^3/min, CFM	0.472	m^3/s
gpm (US)	0.0631	L/S
inch	25.4	mm
inch of water	3.38	kPa
kBtu/h	1055	kJ
lbm/h	0.126	g/s
ton of refrigeration capacity	3516	W

1. INTRODUCTION

1.1 Background

Absorption cycle air-conditioning and refrigerating equipment have been in standard production for decades, and have been designed for residential, commercial, and industrial applications. Absorption cycle heat pumps are currently being developed for introduction into the marketplace. Absorption equiment has typically been driven by such energy sources as fossil fuels and waste and process steam. Capacities have ranged from three kW to approximately seven MW (3 to 2000 tons). Coefficients of performance based on full load steady-state operation testing, have varied from 0.4 to values near 1.0, depending upon whether the machine is a single or double-effect design. The efficient performance of absorption equipment is dependent on many operating and design variables including the temperatures at which energy is supplied and rejected, the load imposed by the conditioned space, the required chilled fluid temperature, various solution and fluid flow rates, and the refrigerant and absorbent fluids selected.

Because of worldwide energy concerns and the increasing scarcity of some forms of energy, the efficient use of absorption equipment has recently come into sharper focus. As a result, new and more energy efficient designs are being developed by various groups and agencies in this country and abroad. Accordingly, there is an increasing need by industry, government, and the consumer to be able to evaluate and compare these improved systems on the same technical basis. A standardized test and rating procedure which incorporates provisions specifically tailored to the nature of each type of absorption system is required in order to effectively compare overall system performance. Such a test and rating procedure is the subject of this report.

1.2 Comparison of Different Residential Size Systems

In order to compare different residential size systems such as absorption or vapor compression air-conditioners or heat pumps, the following caution should be taken into account. The performance of a cooling or heating system is not only dependent upon the unit itself but also of the characteristics of the thermostat which controls the cycling rate. Differences in thermostats can result in cycling time lengths of unequal duration leading to fluctuations in indoor temperatures, and consequently different energy performance of the cooling or heating device. In this test procedure the unit is operated according to the charcteristics of the thermostat provided by the manufacturer; however, the results of this procedure do not show differences in the comfort for inhabitants of a room due to humidity and temperature variations during a cycle, nor does it measure the effects on equipment life as a result of the total number of cycles required to meet the seasonal load.

1.3 Classification of Absorption Systems

Absorption systems are identified according to the following classification:

- o Type of Service
  - cooling
  - heating
  - both heating and cooling (reversible)
- o Energy Source
  - direct-fired (gas or oil combustion)
  - hot water (steam)
- o Application
  - residential
  - commercial

- industrial

o Sink/Source Medium

- air
- water

A further characterization of a system includes whether the generator unit is single-effect or double-effect, and a specification of the refrigerant/absorbent fluid pair.

All commercially available absorption systems are currently designed for cooling service only. There are, however, several heat-only and reversible absorption systems currently under development. These systems are designed for direct firing with oil or gas, or the use of waste or process steam to provide high temperature hot water.

In this report the terminology and classification scheme used for cooling and heating systems is as shown below:

Air cooled ≡ refers to units rejecting heat to air

Water cooled ≡ refers to units rejecting heat to water

Air source ≡ refers to units absorbing heat from air

Water source ≡ refers to units absorbing heat from water

<u>Heating Systems</u>

Outdoor Side

Air Source - outdoor air temperature entering evaporator

Water Source - ground water temperature entering evaporator

Indoor Side

Hot Air Heat - indoor return air temperature

Hot Water Heat - indoor return water temperature

Cooling Systems

Outdoor Side

Air Sink - (i.e., air cooled) outdoor air temperature entering condenser

Water Sink - (i.e., water cooled) ground water temperature or cooling tower water temperature entering condenser

Indoor Side

Air Conditioners - indoor return air temperature

Water Chillers - indoor return water temperature

1.4 Objective and Scope

The objective of this study is to develop generic test and rating procedures for absorption cooling and heating systems which are likely to be employed in residential and light commercial buildings both now and in the future. Where feasible, the proposed procedures include the formulation of calculation procedures to estimate the seasonal performance and seasonal cost of operation of these systems. The intent of the proposed procedures is to provide a means whereby the performance of prototype and production type absorption systems having different design characteristics may be meaningfully compared on the same technical basis, and sound decisions may be made regarding which systems are worthy of further development or application.

The test, rating, and calculation procedures recommended herein apply only to residential and light commercial cooling and heating applications of the 10.6-52.8kW (3-15 tons) capacity range. The procedures are restricted to direct-fired systems, (since these are the most likely to emerge in the residential and small commercial markets). They are sufficiently general, however, to include both air-cooled and water-cooled systems. The procedures are intended to be applicable to both single-effect and double-effect machines using any refrigerant/absorbent fluids, and essentially treat the absorption system as a 'black box' with energy inputs and outputs. Therefore, these procedures are as independent of thermodynamic and thermal design specifics as possible.

2. RECOMMENDED RATING REQUIREMENTS

2.1 General

The recommended rating requirements for the absorption systems considered in this study are classified according to whether the systems are designed for cooling or heating applications. For cooling applications (i.e., water-chilling and air-conditioning), those systems which reject heat to water are further divided according to whether heat is rejected to ground water or to an air-cooled device (cooling tower). For heating applications, the systems are divided according to whether they deliver heat to air or water.

2.1.1 Cooling Systems

The rating requirements for these systems include a single steady-state maximum load rating test at specified condenser unit cooling fluid inlet temperature and chilled fluid conditions, one or two steady-state part load performance tests (to account for frost accumulation and high temperature operation) and a cyclic test. The rating test is a maximum load test, run at the 'design' conditions that are established close to the most severe operating conditions likely to be found in the field. The steady-state part load tests cause the unit to operate at the same maximum energy input but at a different refrigerant flow rate. The cyclic test will result in the unit operating to meet load either by cycling or modulation depending on the unit's control logic.

2.1.2 Heating Systems

The rating requirements for heating systems include a steady-state rating test (low-temperature test) at conditions specified in Table 2. This test is at maximum load at conditions that are established close to the most severe ambient conditions to be found in the field. A frost accumulation test is required only for air-source systems and is to be run at an outdoor DB/WB temperature of 1.7°C (35°F)/-1.1°C (30°F). A second steady-state test (high-temperature test) is to be run with both air and water source systems. In order to account for the performance degradation due to the unit cycling on and off, a cycling test is also specified. This test is to be run at the same conditions as the high temperature test for both air-source systems and water-source systems.

These tests on direct-fired systems provide data necessary to construct performance curves which provide a basis for calculating the system's seasonal performance factor and seasonal cost of operation.

A summary of the recommended testing requirements for direct-fired systems is presented in Tables 1 and 2. Recommended rating sheets for the various systems are shown in Tables 3 and 6.

It should be noted that this document differs from the ARI Standard (Ref. 9) for absorption water chillers in its specification of the evaporator water temperature. Whereas the ARI Standard specifies the water temperature leaving the evaporator, here the temperature of the water entering the evaporator is specified. This alteration was necessary to specify a definite chilled water temperature condition for the cyclic test, which is not covered in the ARI document.

2.2 Cooling Systems

2.2.1 Standard Rating Test

A Standard Rating Test shall be conducted according to the test procedures specified in Section 3.2, and the performance calculated according to the procedures discussed in Section 5. In order that the performance of different absorption systems may be compared, the results of the rating tests and the calculated performance shall be reported. Results to be reported are discussed in Section 2.4 and illustrated in Table 3.

2.2.2 Steady-State Tests

A total of three steady-state tests (A, B, C, see Table 1) shall be conducted for direct-fired absorption air-conditioners. Test A is run at the standard rating conditions listed in Table 1. Test B is conducted at different condenser fluid inlet conditions, selected to conform with other (non-absorption type) air-conditioner standards (Ref. 3, 5, 7, 8). The condenser fluid inlet temperature values in these other standards were selected as typical field application values. Test C is intended to provide data for the calculation of the seasonal performance factor and to determine the performance of air-conditioners under dry evaporator coil conditions.

For direct-fired absorption water chillers, Tests A and C shall be conducted. Test B is not applicable.

For systems rejecting heat to air the dry-bulb temperature of the ambient air shall be 35°C (95°F) for Test A, and shall be 27.8°C (82°F) for Tests B and C. The first ambient condition coincides with the ARI Standard rating point for cooling equipment that rejects heat to the outside air. The 27.8°C (82°F) dry-bulb temperature for Test C is chosen because it approximates the average operating temperature of many climates within the U.S. during the cooling season. The data of both steady-state tests are necessary to evaluate the seasonal performance of the system.

For systems which reject heat to a separate air-cooled condenser (cooling tower), the condenser unit inlet water temperatures shall be 35°C (95°F) for Test A and 23.9°C (75°F) for Tests B and C. For systems which reject heat to ground water, the condenser inlet temperatures shall be 21.1°C (70°F) for Test A and 15.6°C (60°F) for Tests B and C respectively. The 21.1°C (70°F) condition does not coincide with the ANSI Standard rating point of 23.9°C (75°F) (Ref. 2) for water cooled equipment; however, it does agree with reference (5) which in turn concurs with the ARI Standard (Ref. 7). The 15.6°C (60°F) condition selected is representative of ground water temperatures that are likely to occur in the field much of the time.

For air-conditioning equipment, the dry-bulb and wet-bulb temperatures of the air entering the cooling coil shall be 26.7°C (80°F) and 19.4°C (67°F), for Tests A and B. For Test C the dry-bulb and wet-bulb temperatures of the entering air shall be 26.7°C (80°F) and a low enough wet-bulb temperature to insure that the indoor cooling coil is not condensing moisture (dry coil). For water-chillers, the temperature of chilled water entering the evaporator coil shall be 12.8°C (55°F) for all three tests (A, C, D). The return air dry-bulb/wet-bulb condition of 26.7°C/19.4°C (80°F/67°F) for air-conditioners was chosen to coincide with the ANSI standard rating points for air-conditioners.

2.2.3 Cyclic Test

To complete the series of tests one cyclic test, D, is recommended for an evaluation of the performance degradation due to off-cycle refrigerant migration and heat loss. The cyclic performance test shall be performed immediately following Test C. The steady-state Test C results and the cyclic test results are used together to find the performance degradation.

There are two possible designs for adjusting the capacity of direct-fired absorption air-conditioners and water-chillers to the building load. The first is modulating the fuel flow rate to the burner and the second by operating the system only for limited periods of time at full capacity so that the time-averaged capacity meets the building load. The last method is the type found in current production models of the capacity range of concern, and is therefore the only one addressed here.

In order to calculate cycling losses and incorporate its effect in an air-conditioner's seasonal performance and seasonal operating costs, it is necessary to couple the results of the cyclic test with the steady-state tests which is done by means of the part load factor:

$$PLF = \frac{COP_{cyc}}{COP_{ss}}$$

In the case of air-conditioners the following assumption is made in order to consider the fact that a dry or wet cooling coil might occur:

$$PLF = \left.\frac{COP_{cyc}}{COP_{ss}}\right|_{dry} = \left.\frac{COP_{cyc}}{COP_{ss}}\right|_{wet} \quad ;$$

For air-conditioners and chillers the dry-bulb and wet-bulb temperatures for Test D are the same as those of Test C. Similarly, the condenser ambient condition is identical to Test C: for units rejecting heat to air the outdoor dry-bulb temperature shall be 27.8°C (82°F); for units rejecting heat to ground water the entering water temperature shall be 15.6°C (60°F) and for units rejecting heat to cooling tower water the entering water temperature shall be 23.9°C (75°F).

During the cyclic test the systems shall be operated by the control-devices supplied by the manufacturer. Tests shall be conducted with the burner on 20% of the cycle time and off 80% of the time. Current absorption air conditioners incorporate a thermostat set at a maximum cycling rate of 1 1/2 cycles per hour at a 50% on-time. Assuming a parabolic thermostatic control curve, the resulting burner on and off-times are 12 minutes and 48 minutes respectively. However, if comparison with vapor compression systems is intended, the on-off time should be identical to the latter system. References (3) and (4) recommend a 20% part load test to be achieved with six minutes on-time and 24 minutes off-time for these system comparisons.

2.2.4 Summary

The test requirements for direct-fired air-conditioning and water-chilling systems are summarized in Table 1. The results of these tests shall be used to calculate system cooling capacity and coefficient of performance of each system tested, and shall also be used to calculate the seasonal performance factor and seasonal operating cost. Pertinent test data and calculated results shall be reported according to the recommended rating sheets discussed in Section 2.4 and illustrated in Table 3.

2.3 Heating Systems

2.3.1 Standard Rating Test

The Standard Rating Test shall be conducted according to the test procedures specified in Section 6.2 and the performance calculated according to the procedure discussed in Section 8. In order that the performance of different absorption systems may be effectively compared, the results of the rating tests and the calculated performance shall be reported. Results to be reported are discussed in Section 2.4 and illustrated in Table 4.

2.3.2 Steady-State Tests

Two steady-state tests (A and C, see Table 2), shall be conducted for direct-fired absorption heat pumps.

For air source systems the dry-bulb and wet-bulb temperatures of the ambient air shall be 8.3°C (47°F) and 6.1°C (43°F), respectively, for Test A, and shall be -8.3°C (17°F) and -9.4°C (15°F) for Test C. For water source systems the evaporator inlet water temperature shall be 21.1°C (70°F) during Test A and 15.6°C (60°F) for Test C. The 15.6°C (60°F) condition is representative of ground water temperatures that frequently occur in the field.

2.3.3 Frost-Accumulation Test

The frost accumulation Test B applies only to air-source systems. Air temperatures entering the evaporator and surrounding the outdoor portion of the unit shall have ambient dry-bulb and dew-point temperatures of 1.7°C (35°F) and -1.1°C (30°F), respectively. Air entering the indoor heating coil of an air heating system shall have a dry-bulb temperature of 21.1°C (70°F) and maximum indoor wet-bulb temperature of 15.6°C (60°F). For a hot water heating system the entering water temperature shall be 40.6°C (105°F).

2.3.4 Cyclic Test

The final test required is a cyclic Test D, which evaluates the performance degradation due to off-cycle refrigerant migration and heat losses.

The cyclic performance test shall be performed immediately following the high temperature Test A. The dry-bulb temperature and the wet-bulb temperature of the air entering the outdoor portion of an air-source unit shall be the same as in the high temperature test. Similarly, the dry and wet-bulb temperatures of the air entering and surrounding the indoor portion of air-source units shall be the same as Test A (21.1°C (70°F)) dry-bulb, (15.6°C (60°F)) maximum wet-bulb. The entering water temperature for hot water systems shall again be 40.6°C (105°F). For water-source systems the cyclic performance test shall also be run immediately following the high temperature (steady-state) test. The entering water temperature and flow rate during the on-period as well as the dry-bulb and wet-bulb temperature of the air entering and surrounding the unit shall be the same as the high temperature test conditions. The temperature of the entering water of a hot water system shall remain the same as in Test A within the tolerance specified in Section 6.3.4. During the cyclic test, the water flow and the indoor fan cycle 'on' and 'off' as the generator cycles 'on' and 'off', except that the indoor fan cycling times may be delayed due to controls that are normally installed with the unit. The generator cycling times shall be 12 minutes 'on' and 48 minutes 'off', unless the thermostat supplied by the manufacturer specifies on-off times at 80% burner on-time that are different.

2.3.5 Summary

The test requirements for air-source and water-source direct-fired heat pump systems are summarized in Table 2. The results of these tests shall be used to calculate system heating capacity and coefficient of performance as well as the seasonal performance factor and seasonal operating cost. Pertinent test data and calculated results shall be reported according to the recommended rating sheets discussed in Section 2.4 and illustrated in Table 6.

2.4 Suggested Rating Sheets

Rating sheets for the systems examined in this study are illustrated in Tables 3 and 6.

Table 3 is applicable to Direct-Fired Absorption Cooling Systems, and requires that the system under test be rated relative to its:

- Steady-state cooling capacity and coefficient of performance at standard (Test A) rating conditions.
- Steady-state cooling capacity and coefficient of performance at Test B and Test C conditions.
- Chilled fluid, condenser unit, and fuel flow rates at rated conditions.
- Coefficient of performance at cyclic Test D.
- On-off times at 20% burner on time.
- Seasonal performance factor and seasonal operating cost.

Table 6 is applicable to Direct-Fired Absorption Heating Systems, and requires that the system under test be rated relative to its:

- Steady-state heating capacity and coefficient of performance at standard (Test A) rating conditions.
- Steady-state heating capacity and coefficient of performance at Test B and Test C conditions.

- Source fluid, evaporator, and fuel flow rates at rated conditions.
- Coefficient of performance at cyclic Test D.
- On-off times at 20% burner on time.
- Seasonal performance factor and seasonal operating cost.

3. RECOMMENDED TEST PROCEDURE FOR ABSORPTION COOLING SYSTEMS

3.1 Introduction

The purpose of this section is to describe standard test procedures and methods for determining accurate and reliable test data on the performance of prototype and production type direct-fired absorption cooling systems. The cooling capacity of each system shall be directly determined from the results of a primary test, and indirectly determined from a simultaneously conducted heat balance confirming test. The primary test shall be considered valid when the cooling capacity from the confirming test (heat balance) agrees within six percent of the primary test results. The primary test shall be used as a basis for rating the equipment as recommended in Section 2. Where feasible, the test procedures have been taken or adapted from ASHRAE Standards (Ref. 10, 11, and 12), ARI Standards (Ref. 9) and ANSI Standards (Ref. 2).

3.2 Steady-State Test Procedure

3.2.1 Applicable Test Method

The test method most appropriate for determining the steady-state cooling capacity of absorption air-conditioners is the air-enthalpy method. In this method, cooling capacities are determined from measurements of the air flow rate and the wet- and dry-bulb temperatures of the air stream entering and leaving the cooling coil. This method shall be used as the primary test method for absorption air-conditioning equipment covered by this report. When required as part of the confirming test, the air enthalpy method shall also be used to determine the heat rejected by the condenser unit of systems rejecting heat to air. A description of this method and its associated test room and measurement requirements is presented in Section 4 of the ASHRAE Standard (Ref. 10).

The primary test for determining the cooling capacity of absorption water-chillers shall be the simultaneous measurement of the water flow rate and the temperature difference between entering and leaving chilled water. This method shall also be used in the confirming test to determine the energy rejected by the condenser unit circuit of systems which reject heat to water. For direct-fired systems, the energy input to the refrigeration cycle shall be determined from the fuel's steady-state flow rate, its higher heating value and the electric power consumption.

3.2.2 Instrumentation and Required Data

The steady-state performance tests shall have the same instrumentation and data requirements as those specified in Section 9 and Table 1 of ASHRAE Standard (Ref. 10). Provision shall be made to determine the cooling capacity of absorption water-chilling systems.

3.2.3 Test Operating Procedure and Results

The equipment under test and the test room reconditioning apparatus shall be operated until 'equilibrium conditions' are attained but not for less than one hour, before any test data is recorded. Data shall then be recorded at 10 minute intervals until seven consecutive sets of readings within the tolerances specified in Section 3.2.4 are attained.

The steady-state results of a performance test at specified conditions shall include each of the following quantities as are applicable to the equipment under test:

1) Total cooling capacity, kW (Btu/hr).

2) Condenser unit heat rejection, kW (Btu/hr).

3) Energy input to the generator, kW (Btu/hr).

4) Total electric power input to all components and accessories, kWh.

5) Coefficient of performance.

6) Flow rate of medium to be cooled (water for chillers--air for air-conditioners), kg/hr (lbm/hr. or CFM).

7) Ground water or outside air flow rate over the condenser unit, kg/hr (lbm/hr or CFM).

8) Fuel flow rate, kg/hr (lbm/hr or CFM).

9) Flue gas CO_2, %

10) Flue gas temperature, °C (°F).

11) Surface temperatures of the jacket, °C (°F).

Sections 11.1.3, 11.1.4, and 11.2.1 of the ASHRAE Standard (Ref. 10) shall apply for all performance tests.

3.2.4 Test Tolerances

All steady-state tests shall be conducted within the tolerances specified in Table II of the ASHRAE Standard (Ref. 10). Test operating tolerance is defined as the greatest permissible difference between maximum and minimum instrument observations during the test. Test condition tolerance is defined as the maximum permissible variation of the average of the test observations from the desired test conditions. Variations greater than those described shall invalidate the test.

3.3 <u>Cyclic Test Procedure</u>

3.3.1 Applicable Test Method

As outlined in Section 3.2.1 the air-enthalpy method shall be used as a primary test to determine the capacity of absorption air-conditioners. The primary test for determining the capacity of absorption water-chillers shall be the simultaneous measurement of the water flow rate and the temperature difference between inlet and outlet chilled water. For direct-fired systems the energy input to the unit shall be determined from the fuel's flow rate and its higher heating value and the electric power consumption.

3.3.2 Instrumentation and Required Data

The cyclic test set up shall have the same instrumentation that is provided for the steady-state tests. In addition, care must be taken to ensure that during the on-time, sufficient data are taken to evaluate capacity and COP with the required accuracy. Usually this is done by recording the temperature difference between inlet and outlet of the chilled fluid continuously while the fluid flows at a constant rate.

3.3.3 Test Operating Procedure and Results

The cyclic performance test, Test D, shall be performed immediately following Test C. The equipment under test and the test room reconditioning apparatus shall be operated until 'equilibrium conditions' are obtained before any test data are recorded. 'Equilibrium conditions' means in the case of cycling tests that during subsequent cycles the same set of data within tolerances specified in Section 3.3.4 during the on- and off-period are obtained. Once at 'equilibrium conditions' the data of the subsequent fourth cycle shall be recorded. The results of any part load test shall include all quantities listed in Section 3.2.3 except items 2, 3, 9, 10 and 11.

3.3.4 Test Tolerances

One minute after start up of the burner the same test tolerances shall be applied as specified in Section 3.2.4.

4. MODEL LOADS AND CLIMATE SPECIFICATIONS FOR ABSORPTION COOLING SYSTEMS

4.1 Introduction

The seasonal performance and seasonal cost of operation of any direct-fired absorption cooling system depends not only upon the instantaneous performance of the system under specific indoor and outdoor conditions, but also upon the type of building in which it is installed, its thermal load, and the climate in which the building is situated. Because of the wide range of climates in the United States, and the even wider range of building types and thermal requirements, it becomes extremely difficult to adequately characterize the performance of a cooling system in all regions of the country with one or two seasonal indicators. In order to provide the manufacturer of direct-fired equipment some latitude and flexibility in establishing the seasonal performance of his product, two separate evaluation approaches are recommended for seasonal calculations of such systems:

1) The generalized climate of the United States shall be adopted using the average cooling load hours, CLH, determined for the climate. (Table 4).

2) Assuming a more localized climate, the Fractional Temperature Bin Hours and Cooling Load Hours for that climate shall be used. (Generally available Air Force Manual 88-29 and Table 5.)

In both cases, the assumed Outdoor Design Cooling Temperature and Ground Water Temperature shall be 35°C (95°F) and 15.6°C (60°F), respectively. The temperature bin method illustrated in Section 5.6 shall be used to determine the seasonal performance factor and the seasonal operating cost based upon the appropriate climate.

4.2 Building Loads

Cooling requirements are determined by assuming a linear relationship between building load and outdoor dry-bulb temperature. The cooling load line extends from zero load at a specified outdoor temperature T_c, to a value that is 10% below the steady-state cooling capacity at an outdoor design temperature of 35°C (95°F). The building load-temperature relationship is given by:

$$BL(T_j) = \frac{\dot{Q}_{ss}(T_h)(5j-3)}{1.1(T_h-T_c)} \quad ; \quad T_j > T_c \qquad \text{(Temperatures in °F check equivalent units for °C)} \tag{4.1}$$

where j = 1, 2, . . . n. $\dot{Q}_{ss}(T_h)$ is the measured steady-state cooling capacity of the direct-fired absorption system at the assumed design temperature, n represents the total number of non-zero temperature bins, and 1.1 represents an arbitrary oversizing factor. T_j is the representative temperature of the jth bin and is given by:

$$T_j = T_c - 3 + 5j \quad ; \quad T_j > T_c \tag{4.2}$$

The change-over temperature T_c is assumed to be 18.3°C (65°F).

The fractional building cooling load at a representative outdoor temperature, T_j, is expressed as:

$$BL(T_j) \cdot \frac{n_j}{N} = \frac{\dot{Q}_{ss}(T_h)(5_j - 3)}{1.1\ (T_h - T_c)} \cdot \frac{n_j}{N} \tag{4.1a}$$

where n_j/N is the ratio of bin hours of the jth temperature bins to the total seasonal cooling hours.

5. CALCULATION PROCEDURE FOR ABSORPTION COOLING SYSTEMS

5.1 Introduction

The calculation procedure in this section describes methods for calculating cooling capacity, condenser unit heat flow, input energy to the generator, electric power consumption, and coefficient of performance of all the absorption cooling systems considered in this study. In addition, a procedure is defined for calculating the seasonal performance factor and seasonal operating cost of direct-fired units rejecting heat to air and direct-fired units which reject heat to ground water.

5.2 Calculation Procedure for Steady-State Tests

5.2.1 Fuel Energy Input

The rate of thermal energy $\dot{Q}_g$ supplied to the generator of direct-fired systems under steady-state conditions is:

$$\dot{Q}_g = \left[\dot{m}_f \cdot \mathrm{HHV}\right] \tag{5.1}$$

where $\dot{m}_f$ is the fuel mass flow rate and HHV is the higher heating value of the fuel on a mass basis.

5.2.2 Electrical Energy Input

The total electrical power input to the unit is defined by:

$$P_t = P_e + P_c + P_{aux} \tag{5.2}$$

where P_t is the total amount of electrical power being supplied averaged over the test duration. For water chillers P_e is the power to the chilled water pump. If this pump is not furnished with the unit, a value of 11.4 watts/kW

(40 watts/ton) shall be assumed. For air-conditioners P_e is the power of the indoor fan. If a fan is not furnished as part of the model, a value of 0.777 W per ℓ/sec (1250 Btu/hr per 1000 cfm) shall be assumed. In the case of units which reject heat to water P_c is the power to the condenser water pump. If this pump is not furnished with the unit, a value of 20 watts/kW (70 watts/ton) shall be assumed. For units which reject heat to outdoor air P_c is the power to the outdoor fan. P_{aux} is the electrical power required by the various controls and auxilliaries.

5.2.3 Cooling Capacity

The steady-state cooling capacity $\dot{Q}_{ss}$ of absorption air-conditioners shall be determined according to the air-enthalpy method outlined in Section 4 of ASHRAE Standard (Ref. 10) using the appropriate equations specified in Section 4.6 and 7.4. For air-conditioning systems which may not have indoor-air circulating fans furnished as part of the system, their measured cooling capacity shall be adjusted by subtracting 0.777 W per ℓ/sec (1250 Btu/hr per 1000 cfm) of indoor air-flow from the measured value.

For water chillers, the steady-state cooling capacity shall be determined from

$$\dot{Q}_{ss} = \dot{m}_e c_p (T_{e1} - T_{e2}) \tag{5.3}$$

where $\dot{m}_e$ is the water mass flow rate through the cooling coils and T_{e1} and T_{e2} are the cooling coil inlet and outlet water temperatures, respectively.

It is assumed that the effect of the chilled water circulation pump on the capacity is negligible.

5.2.4 Heat Balance - Confirming Test

This method is used to confirm the direct measurement of steady-state cooling capacity by algebraically combining the measured values of generator input energy $\dot{Q}_g$, condensing unit heat $\dot{Q}_c$ (includes both condenser and absorber heats), and the total electrical energy P_t input to the unit. Accordingly the steady-state cooling capacity by the heat balance method is:

$$\dot{Q}_{ss} = \dot{Q}_c - \dot{Q}_g - 3.413 \cdot P_t + \dot{Q}_j + \dot{Q}_f \tag{5.4}$$

where the $\dot{Q}$'s are measured in Btu/hr and P_t in watts.

$\dot{Q}_g$ is discussed in Section 5.2.1. $\dot{Q}_f$ is the residual energy in the products of combustion (flue gas) leaving the system, and is determined from measurement of flue gas temperature and CO_2 content. $\dot{Q}_j$ is jacket heat loss and represents the convective and radiative losses from heated metal surfaces. Its determination is based upon appropriate surface temperature readings and calculation methods presented in Appendix B of reference (2). The jacket surface temperature measurement is performed in the manner described in part 2.12 of reference (7).

$\dot{Q}_c$ represents the amount of energy rejected from the refrigeration machine including energy added to the cooling fluid from fans or pumps.

For units rejecting heat to air the heat rejection shall be calculated from:

$$\dot{Q}_c = 1.08\ \dot{Q}_{sc}(T_{c2} - T_{c1}) \tag{5.5}$$

(Temperatures in °F, $\dot{Q}_{sc}$ in CFM; check equivalent units for °C).

where $\dot{Q}_c$ is in Btu/hr.

$\dot{Q}_{sc}$ is the air flow rate across the condenser and absorber coils (in some units it includes the combustion flue gas as well) corrected to standard conditions, and which is calculated from equations specified in Section 7.4 of the ASHRAE Standard (Ref. 10). T_{c1} and T_{c2} are the inlet and outlet temperatures of the cooling air respectively.

For those units, where the flue gases are mixed with the condenser unit cooling air, $\dot{Q}_f$ is already included in the measured value of $\dot{Q}_c$. Therefore, term $\dot{Q}_f$ in equation (5.4) shall be neglected.

For units rejecting heat to water, the heat rejection shall be determined from:

$$\dot{Q}_c = \dot{m}_c C_p (T_{c2} - T_{c1}) \qquad (5.6)$$

where $\dot{m}_c$ is the mass flow rate of the cooling water and T_{c1} and T_{c2} are the inlet and outlet temperatures of the cooling water, respectively, and C_p is the specific heat (liquid) of the water.

5.3 Calculation Procedure for Cyclic Tests

Since the cooling capacity varies with the unit on-time the cooling done over a complete cycle for water chillers is evaluated by:

$$\dot{Q}_{cyc} = \dot{m}_e C_p \int_{t_{pump\ on}}^{t_{pump\ off}} \Delta T(t)dt \qquad (5.7)$$

$\dot{Q}_{cyc}$ is the cooling done over a complete cycle, $\dot{m}_e$ is the flow rate of the chilled water, assumed to be constant with time, ΔT is the temperature difference which is a function of time (t) and $t_{pump\ on}$ is the on-time of the chilled water pump over a complete cycle.

For air-conditioners the following equation should be used to determine cyclic cooling done.

$$\dot{Q}_{cyc} = \frac{60\ \dot{V}\ C_{pa}}{v_n(1 + W_n)} \int_{t_{fan\ on}}^{t_{fan\ off}} \Delta T(t)dt \qquad (5.8)$$

$\dot{V}$ is the air flow rate (which is assumed to be constant), C_{pa} is the specific heat at constant pressure of the air-water mixture per pound of dry air (C_{pa} = $1.01 + 1.86\ W_n$, kJ/kg. °C, [$0.24 + 0.444\ W_n$, Btu/lbm °F]), and v_n and W_n are the specific volume and humidity ratio of the air-water mixture at the position where the flow rate measurements are taken. ΔT is the temperature difference which is a function of time and $t_{fan\ on}$ is the on-time of the indoor fan over a complete cycle.

For air-conditioning systems which may not have indoor air circulating fans furnished as part of the system, their measured cooling capacity shall be adjusted by subtracting 0.777 W per ℓ/sec (1250 Btu/hr per 1000 SCFM) of indoor air flow from the measured value.

5.4 Coefficient of Performance

The coefficient of performance for direct-fired systems shall be based on the total fuel energy input to the system plus electrical energy supplied to fans, pumps, controls, etc. Accordingly,

$$COP_{ss} = \frac{\dot{Q}_{ss}}{\dot{m}_f \cdot HHV + 3.413\ P_t} \qquad (5.9)$$

where ($\dot{m}_f \cdot HHV$) is in Btu/hr and P_t is in watts; 3.413 is conversion from watts to Btu/hr. P_t is the total amount of electrical energy consumed during the test duration as given by equation (5.2).

The COP_{cyc} for Test D shall be evaluated according to equation (5.9) with the following modifications.

$\dot{Q}_{ss}$ is replaced by $\dot{Q}_{cyc}$ (eq. 5.7), $\dot{m}_f$ is replaced by the total amount of fuel consumed during one total cycle and P_t is replaced by the total amount of electricity consumed during the entire cycle.

5.5 Seasonal Performance Factor and Seasonal Operating Costs

This section describes a calculation procedure for determining the seasonal performance factor (SPF) and seasonal operating costs (SOC) of direct-fired absorption cooling systems.

The fractional energy consumption in the jth temperature bin shall be evaluated by:

$$E(T_j) \cdot \frac{n_j}{N} = \frac{CLF(T_j)}{1 - C_D(1 - CLF(T_j))} \cdot \dot{E}_{ss} \cdot \frac{n_j}{N} \tag{5.10}$$

where $\dot{E}_{ss} = (\dot{m}_f \cdot HHF) + 3.413\ P_t$ and is the steady-state energy input to the unit; it is assumed to be independent of temperature, because fuel flow rate and electrical power consumption do not vary significantly with outdoor temperature. The cooling load factor CLF is obtained by the following equation:

$$CLF(T_j) = \frac{BL(T_j)}{\dot{Q}_{ss}(T_j)} \quad ; \quad \dot{Q}_{ss}(T_j) \geq BL(T_j) \tag{5.11}$$

$$CLF(T_j) = 1 \quad ; \quad \dot{Q}_{ss}(T_j) < BL(T_j)$$

$\dot{Q}_{ss}(T_j)$ is evaluated by interpolation or extrapolation of the capacities of Tests A and B according to the following equation:

$$\dot{Q}_{ss}(T_j) = \dot{Q}_{ss}(82°F) + a(T_j - 82°F) \tag{5.12}$$

with $a = \dfrac{\dot{Q}_{ss}(95°F) - \dot{Q}_{ss}(82°F)}{95°F - 82°F}$

The degradation factor of equation (5.10) is given by:

$$C_D = \frac{1 - PLF(82°F)}{1 - CLF(82°F)} \tag{5.13}$$

$$\text{with PLF}(82°F) = \frac{COP_{cyc}(82°F)}{COP_{ss}(82°F)} \tag{5.14}$$

The total amount of energy consumed during an entire season divided by the total number of temperature bin hours, N, is evaluated by:

$$\frac{E}{N} = \sum_{j=1}^{n} E(T_j) \cdot \frac{n_j}{N} \tag{5.15}$$

where n is the number of non-zero temperature bins. The SPF is then given by:

$$SPF = \frac{BL/N}{E/N} \tag{5.16}$$

$$\text{where } \frac{BL}{N} = \sum_{j=1}^{n} BL(T_j) \cdot \frac{n_j}{N}$$

An estimation of the seasonal operating cost is given by:

$$SOC = \frac{E}{N}\left[xC_f + (1 - x)\frac{C_e}{3.413}\right] \cdot CLH \tag{5.17}$$

E is the total energy consumption per season, divided by the total number of temperature bin hours, N, C_f and C_e are the costs for fuel in \$/Btu and elecricity in \$/Whr and CLH is the number of total cooling season hours.

Factor x in eq. (5.17) is the ratio of the seasonal primary fuel energy consumption to the total energy consumption. Since this ratio can only be approximated by a few laboratory tests, it is recommended that the cyclic test data ratio of these energies (e.g., the fuel to total) should be used.

The SPF and SOC shall be determined from eqs. (5.16) and (5.17) for the climate discussed in Section 4.1. Table 5 is a calculation sheet that may be used for calculating the SPF and SOC for direct-fired absorption systems.

5.6 Sample Calculations

The following sample calculations for a chilled water-air heat rejection unit are based on hypothetical data obtained from Tests A, C and D. Test B results are not given as no test is required. The total number of seasonal cooling hours is assumed to be 3825 hours.

Test No.

A:	$\dot{Q}_{ss}(95°F) = 32800$ Btu/hr	$\dot{E}_{ss}(95°F) = 81300$ Btu/hr	$COP_{ss}(95°F) = 0.403$
C:	$\dot{Q}_{ss}(82°F) = 36900$ Btu/hr	$\dot{E}_{ss}(82°F) = 81300$ Btu/hr	$COP_{ss}(82°F) = 0.454$
D:	$\dot{Q}_{cyc}(82°F) = 33415$ Btu/hr	$\dot{E}_{cyc}(82°F) = 81300$ Btu/hr	$COP_{cyc}(82°F) = 0.411$

The fractional building load for a given temperature bin is evaluated according to equation (4.1a). For the first bin temperature at 19.4°C (67°F):

$$BL \cdot \frac{n_1}{N} = \frac{32800 \text{ Btu/hr} \cdot [(5 \cdot 1) - 3]°F}{1.1\ (95°F - 65°F)} \cdot 0.214 = 425 \text{ Btu/hr.}$$

The results for all bin temperatures are listed in Table 10 column D. The $CLF(T_j)$ is obtained according to equations (5.11) and (5.12).

For example:

$$\dot{Q}_{ss}(67°F) = 36900 \text{ Btu/hr} + \left[\frac{32800 \text{ Btu/hr} - 36900 \text{ Btu/hr}}{95°F - 82°F}\right] \cdot$$

$$(65°F - 82°F) = 41631 \text{ Btu/hr}$$

and

$$CLF(67°F) = \frac{32800 \text{ Btu/hr} \cdot (67°F - 65°F)}{1.1(95°F - 65°F)} \cdot \frac{1}{41631 \text{ Btu/hr}} = 0.048$$

Further values of $CLF(T_j)$ are listed in column E of Table 10. The energy consumption $E(T_j) \cdot n_j/N$ is obtained by applying equations (5.10), (5.13), and (5.14):

$$PLF = \frac{0.411}{0.454} = 0.905$$

$$C_D = \frac{1 - 0.905}{1 - 0.458} = 0.175$$

$$E(67°F) \cdot \frac{n_j}{N} = \frac{0.048}{1 - 0.175(1 - 0.048)} \cdot 81300 \text{ Btu/hr} \cdot 0.214 = 1002 \text{ Btu/hr}$$

The results for other temperature bins are again listed in Table 10 column F. The SPF is then calculated by equation (5.16):

$$SPF = \frac{11717}{28502} = 0.41$$

According to equation (5.17) the seasonal operating cost, assuming

$C_f = 0.005 \frac{\$}{\text{kBtu}}$ and $C_e = 0.06 \frac{\$}{\text{kWh}}$ and $x = 0.95$ and $CLH = 3825$

$$SOC = 28502 \left[.95(.005) + .05 \left(\frac{.06}{3.413}\right)\right] 3825 = \$613.67$$

where it is assumed that the cyclic test data showed that 95% (i.e., $x = 0.95$) of the total energy input is fuel.

Example calculations for an air-conditioning unit would be similar to those for the vapor compression air-conditioners as illustrated in ASHRAE Standard (Ref. 12).

6. RECOMMENDED TEST PROCEDURE FOR ABSORPTION HEATING SYSTEMS

6.1 Introduction

The purpose of this section is to describe standard test procedures and methods for determining accurate and reliable test data on the performance of prototype and production type direct-fired absorption heating systems. The heating capacity of each system shall be directly determined from the results of a primary test, and indirectly determined from a simultaneously conducted heat balance confirming test. The primary test shall be considered valid when the heating capacity from the confirming test (heat balance) agrees within six percent of the primary test results. The primary test shall be used as a basis for rating the equipment as recommended in Section 2. Where feasible, the test procedures have been taken or adapted from ASHRAE Standards (Ref. 10 and 11).

6.2 Steady-State Test Procedure

6.2.1 Applicable Test Method

The test method most appropriate for determining the steady-state heating capacity of absorption heat pumps is the air-enthalpy method--indoor side (for forced air heating systems). In this method, heating capacities are determined from measurements of the air flow rate and the dry bulb temperatures of the airstream entering and leaving the heating coil. This method shall be used as the primary test method for absorption heat pump equipment covered by this report. When required as part of the confirming test, the air enthalpy method shall be used to determine the heat absorbed by the evaporator unit of air-source systems. A description of this method and its associated test room and measurements requirements is presented in Section 3 of ASHRAE Standard (Ref. 11).

The method used as a confirmation test to determine the energy absorbed by the evaporator unit of water-source systems shall be the simultaneous measurement of the water flow rate and the temperature difference between entering and leaving water of the indoor coil.

For direct-fired systems the energy input to the refrigeration cycle shall be determined from the fuel's steady-state flow rate, its higher heating value and the electric power consumption.

6.2.2 Instrumentation and Required Data

The steady-state performance tests shall have the same instrumentation and data requirements as those specified in Section 10 and Table II of ASHRAE Standard (Ref. 11). Provision shall be made to determine the heating capacity of absorption heat pump systems.

6.2.3 Test Operating Procedure and Results

The equipment under test and the test room reconditioning apparatus shall be operated until equilibrium conditions are attained, but not for less than one hour, before any test data is recorded. Data shall then be recorded at 10 minute intervals until seven consecutive sets of readings within the tolerances specified in Section 6.2.4 are attained.

The steady-state results of a performance test at specified conditions shall include each of the following quantities as are applicable to the equipment under test:

1) Total heating capacity, kW (Btu/hr)

2) Evaporator unit heat absorption, kW (Btu/hr)

3) Energy input to the generator, kW (Btu/hr)

4) Total electric power input to all components and accessories, kWh.

5) Coefficient of performance

6) Flow rate of medium to be heated (hot water or indoor air), kg/hr (lbm/hr or CFM).

7) Ground water or outside air flow rate over the evaporator unit, kg/hr (lbm/hr or CFM).

8) Fuel flow rate, kg/hr (lbm/hr or CFM).

9) Flue gas CO_2, %

10) Flue gas temperature, °C (°F)

11) Surface temperatures of the jacket, °C (°F)

Sections 12.15, 12.16, and 15 of the ASHRAE Standard (Ref. 11) shall apply for all performance tests.

6.2.4 Test Tolerances

All steady-state tests shall be conducted within the tolerances specified in Table III of the ASHRAE Standard (Ref. 11). Test operating tolerance is defined as the greatest permissible difference between maximum and minimum instrument observations during the test. Test condition tolerance is defined as the maximum permissible variation of the average of the test observations from the desired test conditions. Variations greater than those described shall invalidate the test.

6.3 Cyclic Test Procedure

6.3.1 Applicable Test Method

As outlined in Section 6.2.1 the air enthalpy method shall be used as a primary test to determine the capacity of absorption heat pumps. Air-source units coil must be clean of frost both before and throughout the test duration.

6.3.2 Instrumentation and Required Data

The cyclic test set up shall have the same instrumentation that is provided for the steady-state tests. In addition, care must be taken to ensure that during the on-time, sufficient data are taken to evaluate capacity and COP with the required accuracy. Usually this is done by recording the temperature difference between inlet and outlet water, circulating through the condensing unit, while the fluid flows at a constant flow rate.

6.3.3 Test Operating Procedure and Results

The cyclic performance test, Test D, shall be performed immediately following the high temperature test, Test A. The equipment under test and the test room reconditioning apparatus shall be operated until 'equilibrium conditions' are obtained before any test data are recorded. 'Equilibrium conditions' means in the case of cycling tests that during subsequent cycles the same set of data within tolerances specified in Section 6.3.4 during the on- and off-period are obtained. Once at 'equilibrium conditions' the data of the subsequent fourth cycle shall be recorded. The results of any part-load test shall include all quantities listed in Section 6.2.3 except items 2 and 3.

6.3.4 Test Tolerances

One minute after start up of the burner the same test tolerances shall be applied as specified in Section 6.2.4.

6.4 Frost Accumulation Test Procedure

6.4.1 Applicable Test Method

This test is to be conducted solely for air-source systems using the method described in Section 6.2.1--Indoor side--as a primary test. A capacity determination based on the indoor air enthalpy measurements is the only permissible test. During this test any apparatus disturbing normal outdoor air flow on the equipment must not be connected. The indoor airflow is to be allowed to continue with no changes in the air flow settings for test equiment or associated test apparatus, except that if the defrost controls provide for stopping the indoor fan, provision shall be made to shut off flow of air through the indoor coil from the test apparatus while the indoor fan is stopped. The same is true for the water flow in a hot water heating system.

6.4.2 Instrumentation and Required Data

The frost accumulation performance tests shall have the same instrumentation and data requirements as specified in Section 10 and Table II of ASHRAE Standard (Ref. 11).

6.4.3 Test Operating Procedure and Results

The equipment under test and the test room reconditioning apparatus shall be operated for a test period of three hours. If the unit is in defrost at the end of this test period, the cycle shall be completed. Data shall be recorded at normal ten minute intervals (see Section 11.3.2 of ASHRAE Standard (Ref.

11), except that during the defrost cycle data shall be recorded continuously to establish accurately the start and completion of the defrost cycle, the time-temperature pattern of the indoor air stream (if the indoor fan is running) and the electrical and fuel input to the equipment.

The results of this performance test at specified conditions shall include all quantities listed in Section 6.2.3 as are applicable to the air source units.

6.4.4 Test Tolerances

All test observations shall be within the tolerances specified in Table III of the ASHRAE Standard (Ref. 11), as appropriate to the test methods and type of equipment. The maximum permissible variation of any observation during the capacity test is listed under 'Test Operating Tolerance' in the table. This represents the greatest permissible difference between maximum and minimum instrument observations during the test. The maximum permissible variations of the average of the test observations from the standard or desired test conditions are shown in the table under 'Test Condition Tolerance'. Variations greater than those described shall invalidate the test.

7. MODEL LOADS AND CLIMATE SPECIFICATIONS FOR ABSORPTION HEATING SYSTEMS

7.1 Introduction

The seasonal performance and seasonal cost of operation of any direct-fired absorption heating system depends not only upon the instantaneous performance of the system under specific indoor and outdoor conditions, but also upon the type of building in which it is installed, its thermal load, and the climate in which the building is situated. Because of the wide range of climates in the United States, and even wider range of building types and thermal requirements, it becomes extremely difficult to adequately characterize the performance of a heating system in all regions of the country with one or two seasonal indicators. In order to provide the manufacturer of direct-fired equipment some latitude and flexibility in establishing the seasonal performance of his product, the following evaluation approach is recommended for seasonal performance calculations of such systems:

For Heating Only Systems -

- Identify climatic region of the United States based on Tables 7 and 8, in which the direct-fired absorption heating system is to be located.
- Use the representative outdoor heating design temperature for this region (listed in table) to determine the building heating requirement.
- Size the system heating capacity at 80% of the building heating requirement and use add-on heat (gas furnace or electric resistance heat as appropriate to the unit) to make up the difference.
- Use the temperature bin method to determine the seasonal performance factor and the seasonal operating cost based upon the appropriate climate.

For Heating and Cooling Systems -

- The system shall be sized based on the cooling load (refer to Section 4.1).

7.2 Building Loads

Heating requirements are determined by assuming a linear relationship between building load and outdoor dry-bulb temperature. The heating load line extends from zero load at a specified change-over temperature, T_c, to a value that equals the steady-state heating capacity or design heating requirement (whichever is smaller) at the outdoor design temperature, T_h. The building load-temperature relationship is given by:

$$BL(T_j) = \left(\frac{T_c - T_j}{T_c - T_h}\right) \cdot (C) \cdot (DHR) \tag{7.1}$$

where T_j is the representative temperature of the jth bin and is given by:

$$T_j = T_c + 2 - 5_j \ ; \ T_j < T_c \tag{7.2}$$

The change-over temperature T_c is assumed to be 18.3°C (65°F). DHR is the Design Heating Requirement or steady-state heating capacity at T_h. C is an 'experience factor' which tends to improve the agreement between calculated and measured building loads (Ref. 8). The value for C is 0.77.

The fractional building heating load at a representative outdoor dry-bulb temperature, T_j, is expressed as:

$$BL(T_j) \cdot \frac{n_j}{N} = \left[\left(\frac{T_c - T_j}{T_c - T_h}\right) \cdot (C) \cdot (DHR)\right] \cdot \frac{n_j}{N} \tag{7.1a}$$

n_j/N is the ratio of bin hours of the jth temperature bins to the total seasonal heating hours.

8. CALCULATION PROCEDURE FOR ABSORPTION HEATING SYSTEMS

8.1 Introduction

The calculation procedure in this section describes methods for calculating heating capacity, evaporator unit heat flow, input energy to the generator, electric power consumption, and coefficient of performance of all the absorption heating systems considered in this study. In addition, a procedure is defined for calculating the seasonal performance factor and seasonal operating cost of direct-fired air-source systems, and direct-fired water-source systems that absorb heat from ground water.

8.2 Calculation Procedures for Steady-State Tests

8.2.1 Fuel Energy Input

The rate of thermal energy $\dot{Q}_g$ supplied to the generator of direct-fired systems under steady-state conditions is:

$$\dot{Q}_g = \left[\dot{m}_f \cdot \mathrm{HHV}\right] \tag{8.1}$$

where $\dot{m}_f$ is the fuel mass flow rate and HHV is the higher heating value of the fuel on a mass basis.

8.2.2 Electrical Energy Input

The total electrical power input to the unit is defined by:

$$P_t = P_e + P_c + P_{aux} \tag{8.2}$$

where P_t is the total amount of electric power being supplied averaged over the test duration. P_e is the power to the supply water pump. If this pump is not furnished with the unit, a value of 20 watts/kW (70 watts/ton) shall be assumed. P_c is the power to the evaporator water pump, for water source units. If this pump is not furnished with the unit, a value of 11.4 watts/kW (40 watts/ton) shall be assumed. P_{aux} is the electrical power required by the various controls and auxilliaries. For heat pump systems which may not have indoor air circulating fans furnished as part of the system, the total energy used shall be adjusted by adding 0.777 W per ℓ/sec (1250 Btu/hr per 1000 cfm) of indoor air flow.

8.2.3 Heating Capacity

The steady-state heating capacity $\dot{Q}_{ss}$ of absorption heat pumps shall be determined according to the air-enthalpy method outlined in Section 3 of ASHRAE Standard (Ref. 11) using the appropriate equations specified in Section 3.8 and 7.4. For heat pump systems which may not have indoor air-circulating fans furnished as part of the system, their measured heating capacity shall be adjusted by adding 0.777 W per ℓ/sec (1250 Btu/hr per SCFM) of indoor airflow to the measured value.

8.2.4 Heat Balance - Confirming Test

This method is used to confirm the direct measurement of steady-state heating capacity by algebraically combining the measured values of generator input energy $\dot{Q}_g$, evaporating unit heat absorption $\dot{Q}_e$ and the total electrical power power P_t input to the unit. Accordingly, the steady-state heating capacity by the heat balance method is:

$$\dot{Q}_{ss} = \dot{Q}_e + \dot{Q}_g + 3.413\ P_t - \dot{Q}_j - \dot{Q}_f \tag{8.3}$$

where the $\dot{Q}$'s are measured in Btu/hr and P_t in watts.

$\dot{Q}_g$ is discussed in Section 5.2.1. $\dot{Q}_f$ is the residual energy in the products of combustion (flue gas) leaving the system, and is determined from measurement of flue gas temperature and CO_2 content. $\dot{Q}_j$ is the jacket loss and represents the convective and radiative losses from heated metal surfaces. Its determination is based upon appropriate surface temperature readings and calculation methods presented in Appendix B of reference (2). The jacket surface temperature measurement is performed in the manner described in part 2.12 of reference (7).

$\dot{Q}_e$ represents the amount of energy absorbed by the refrigeration machine from either the outside air or ground water source. For air-source systems, the heat absorption shall be calculated from:

$$\dot{Q}_e = 1.08\ \dot{Q}_{sc}(T_{e_2} - T_{e_1}) \qquad (8.4)$$

(Temperatures in °F, $\dot{Q}_{sc}$ in CFM; check equivalent units for °C)

where $\dot{Q}_{sc}$ is in Btu/hr.

$\dot{Q}_{sc}$ is the air flow rate across the evaporator coils corrected to standard conditions, and is calculated from equations specified in Section 7.4 of the ASHRAE Standard (Ref. 11). T_{e_1} and T_{e_2} are the inlet and outlet temperatures of the air, respectively.

For water-source systems, the heat absorption shall be determined from:

$$\dot{Q}_e = \dot{m}_e C_p (T_{e_2} - T_{e_1}) \qquad (8.5)$$

where $\dot{m}_e$ is the mass flow rate of the source water and T_{e_1} and T_{e_2} are the inlet and outlet temperatures of the source water, respectively. C_p is the specific heat (liquid) of the source (ground) water.

8.3 Calculation Procedure for Cyclic Test

Since the heating capacity varies with the unit on-time, the heating done over a complete cycle for a hot water heat system is evaluated by:

$$\dot{Q}_{cyc} = \dot{m}_c C_p \int_{t_{pump\ on}}^{t_{pump\ off}} \Delta T(t)dt \tag{8.6}$$

$\dot{Q}_{cyc}$ is the heating done over a complete cycle, $\dot{m}_c$ is the flow rate of the supply water, assumed to be constant with time, ΔT is the temperature difference which is a function of time (t) and $t_{pump\ on}$ is the on-time (t) of the supply water pump, over a complete cycle.

For heat pumps in a hot air system the following equation should be used to determine cyclic heating done.

$$\dot{Q}_{cyc} = \frac{60\ \dot{V}\ C_{pa}}{v_n(1 + W_n)} \int_{t_{fan\ off}}^{t_{fan\ on}} \Delta T(t)dt \tag{8.7}$$

$\dot{V}$ is the air flow rate (which is assumed to be constant), C_{pa} is the specific heat at constant pressure of the air-water mixture per pound of dry air, C_{pa} = $1.01 + 1.86\ W_n$, kJ/kg, °C, ($0.24 + 0.444\ W_N$, Btu/lbm °F), and v_n and W_n are the specific volume and humidity ratio of the air-water mixture at the same position where the flow rate measurements are taken. ΔT is the temperature difference which is a function of time and $t_{fan\ on}$ is the on-time of the indoor fan over a complete cycle.

For heating systems which may not have indoor air circulating fans furnished as part of the system, their measured heating capacity shall be adjusted by adding 0.777 W per ℓ/sec (1250 Btu/hr per SCFM) of indoor air flow from the measured value.

8.4 Coefficient of Performance

The coefficient of performance for direct-fired systems shall be based on the total fuel energy input to the system plus electrical energy supplied to fans, pumps, controls, etc. Accordingly,

$$COP_{ss} = \frac{\dot{Q}_{ss}}{\dot{m}_f \cdot HHV + 3.413\ P_t} \qquad (8.8)$$

where ($\dot{m}_f \cdot HHV$) is in Btu/hr and P_t in watts; 3.413 is conversion from watts to Btu/hr.

P_t is the total cmount of electrical energy consumed during the test duration as given by equation (8.2).

The COP_{cyc} for Test D shall be evaluated according to equation (8.8) with the following modifications:

$\dot{Q}_{ss}$ is replaced by Q_{cyc} (eq. 8.6), $\dot{m}_f$ is replaced by the total amount of fuel consumed during one total cycle and P_t is replaced by the total amount of electricity consumed during the entire cycle.

8.5 Seasonal Performance Factor and Seasonal Operating Costs

This section describes a calculation procedure for determining the seasonal performance factor (SPF) and seasonal operating costs (SOC) of direct-fired absorption heating systems.

The fractional energy consumption in the jth temperature bin shall be evaluated by:

$$E(T_j) \cdot \frac{n_j}{N} = \frac{HLF(T_j)}{1 - C_D(1 - HLF(T_j))} \cdot \dot{E}_{ss} \cdot \frac{n_j}{N} \qquad (8.9)$$

where $\dot{E}_{ss} = (\dot{m}_f \cdot HHV) + 3.413\ P_t$ and is the steady-state energy input to the unit; it is assumed to be independent of temperature, because fuel flow rate and electrical power consumption do not vary significantly with outdoor temperature. The heating load factor, HLF, is obtained by the following equation:

$$HLF(T_j) = \frac{BL(T_j)}{\dot{Q}_{ss}(T_j)} \quad ; \quad BL(T_j) \leq \dot{Q}_{ss}(T_j)$$

$$HLF(T_j) = 1 \quad ; \quad BL(T_j) > \dot{Q}_{ss}(T_j) \qquad (8.10)$$

$\dot{Q}_{ss}(T_j)$ is evaluated by interpolation or extrapolation of the capacities of Tests A and C according to the following equation:

$$\dot{Q}_{ss}(T_j) = \dot{Q}_{ss}(47°F) + a(T_j - 47°F) \tag{8.11}$$

with $a = \dfrac{\dot{Q}_{ss}(17°F) - \dot{Q}_{ss}(47°F)}{17°F - 47°F}$

The degradation factor of equation (8.9) is given by:

$$C_D = \frac{1 - PLF(47°F)}{1 - HLF(47°F)} \tag{8.12}$$

$$\text{with } PLF(47°F) = \frac{COP_{cyc}(47°F)}{COP_{ss}(47°F)} \tag{8.13}$$

The total amount of energy consumed during an entire season divided by the total number of temperature bin hours, N, is evaluated by:

$$\frac{E}{N} = \sum_{j=1}^{n} E(T_j) \cdot \frac{n_j}{N} \tag{8.14}$$

where n is the number of non-zero temperature bins. The SPF is then given by:

$$SPC = \frac{BL/N}{E/N + (RH \cdot 3.413)^{*}} \tag{8.15}$$

where $\dfrac{BL}{N} = \sum_{j=1}^{n} BL(T_j) \cdot \dfrac{n_j}{N}$

*If unit is gas-fired use $\left[\dot{m}_f \cdot HHV\right]$.

and

$$RH = \sum_{j=1}^{n} \frac{RH(T_j)}{N} = \sum_{j=1}^{n} \frac{(BL(T_j) - \dot{Q}_{ss}(T_j))}{3.413} \cdot \frac{n_j}{N}$$

where $BL(T_j) \geq \dot{Q}_{ss}(T_j)$

An estimation of the seasonal operating cost is given by:

$$SOC = \frac{E}{N}\left[xC_f + (1 - x)\frac{Ce}{3.413}\right] \cdot HLH + \left[RH \cdot 3.413\right]^{*} \cdot HLH \qquad (8.16)$$

E/N is the total energy consumption divided by the total number of temperature bin hours, N, per season, C_f and C_e are the costs for fuel in \$/Btu and electricity in \$/Wh and HLH is the number of total heating season hours. Factor x in equation (8.16) is the ratio of the seasonal primary fuel energy consumption to the total energy consumption. Since this ratio can only be approximated by a few laboratory tests, it is recommended that the cyclic test data ratio of these energies (e.g., the fuel to total) should be used.

Table 5 is a calculation sheet that may be used for calculating the SPF and SOC for direct-fired absorption systems.

8.6 Sample Calculations

The following sample calculations for an air-source hot-water absorption heat pump unit are based on hypothetical data obtained from Tests A, C and D. The frost accumulation test (Test B) is not being considered in this example.

---

*If unit is gas-fired use $\left[\dot{m}_f \cdot HHV\right]$.

We are assuming a location in Maine, which is Region V. The heating load hours and outdoor design temperature corresponding to this region are obtained from Table 7.

Test No.

A: $\dot{Q}_{ss}(47°F) = 52901$ Btu/hr $\quad \dot{E}_{ss}(47°F) = 61365$ Btu/hr $\quad COP_{ss}(47°F) = 1.16$

C: $\dot{Q}_{ss}(17°F) = 46075$ Btu/hr $\quad \dot{E}_{ss}(17°F) = 43771$ Btu/hr $\quad COP_{ss}(17°F) = 0.95$

D: $\dot{Q}_{cyc}(47°F) = 35000$ Btu/hr $\quad \dot{E}_{cyc}(47°F) = 25550$ Btu/hr $\quad COP_{cyc}(47°F) = 0.73$

HLH = 2750 hours $\qquad T_{OD} = T_h = -23.3°C(-10°F)$

The fractional building load for a given temperature bin is evaluated according to equation (7.1a). For the first bin temperature at 16.7°C(62°F).

$$BL \cdot \frac{n_1}{N} \left[\left(\frac{65°F - 62°F}{65°F - (-10°F)}\right) \cdot \left(0.77\right) \cdot \left(39932 \text{ Btu/hr}\right)\right] \cdot 0.106 = 130.4 \text{ Btu/hr}$$

where $$DHR = \dot{Q}_{ss}(-10°F) = 52901 \text{ Btu/hr} + \left[\frac{46075 \text{ Btu/hr} - 52901 \text{ Btu/hr}}{17°F - 47°F}\right] \cdot (-10°F - 47°F) = 39932 \text{ Btu/hr}$$

is determined from equation (8.11).

The results for all bin temperatures are listed in Table 11 column D. The $HLF(T_j)$ is obtained according to equations (8.10) and (8.11). For example:

$$\dot{Q}_{ss}(62°F) = 52901 \text{ Btu/hr} + \left[\frac{46075 \text{ Btu/hr} - 52901 \text{ Btu/hr}}{17°F - 47°F}\right] \cdot (62°F - 47°F)$$

$$= 56313.5 \text{ Btu/hr}$$

and

$$HLF(62°F) = \left[\left(\frac{65°F - 62°F}{65°F - (-10°F)}\right) \cdot (0.77) \cdot (39932 \text{ Btu/hr})\right] \cdot \frac{1}{56313.5 \text{ Btu/hr}} = 0.022$$

Further values of $HLF(T_j)$ are listed in column E of Table 11. The energy consumption $E(T_j) \cdot n_j/N$ is obtained by applying equations (8.9), (8.12), and (8.13):

$$PLF = \frac{0.73}{1.16} = 0.63$$

$$C_D = \frac{1 - 0.63}{1 - 0.14} = 0.43$$

$$E(62°F) \cdot \frac{n_j}{N} = \frac{0.022}{1 - 0.43(1 - 0.022)} \cdot 61365 \text{ Btu/hr} \cdot 0.106 = 247.0 \text{ Btu/hr}$$

The results for other temperature bins are again listed in Table 11 column F. The SPF is then calculated by equation (8.15):

$$SPF = \frac{12155.7}{21727.1} = 0.56$$

According to equation (8.16) the seasonal operating cost, assuming $C_f = 0.005$ \$/kBtu and $C_e = 0.06$ \$/Kwh and $x = 0.96$ and $HLF = 2750$ is:

$$SOC = 21.727 \left[.96(.005) + .04 \frac{.06}{3.413}\right] \cdot 2750 = \$328.81$$

where it is assumed that the cyclic test data showed that 96% (i.e., $x = .96$) of the total energy input is fuel.

9. LIMITATIONS OF THE RECOMMENDED TEST AND RATING PROCEDURES

The test requirements described in Section 2 and illustrated in Tables 1-2 assume that the behavior of the particular system can be sufficiently determined with only four test points. As more laboratory and field experience is acquired on the latest absorption systems, the number of test points may have to be increased to adequately describe the system. The rating procedure may then need to be reviewed to verify that it adequately describes field performance.

Additionally, the climate specifications outlined in Sections 4 and 7 are sufficiently general to warrant further investigation. While these seasonal rating approaches suffer the inevitable disadvantages associated with climate generalization, they do provide a suitable technique for assessing both relative Carnot effect (i.e., the use of various temperature reservoir differences) and cycling effect of different absorption systems that are designed for the same market and the same general climate.

These limitations indicate a need for further investigation.

REFERENCES

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11. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment, ASHRAE 37-78.

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TABLE 1

SUMMARY OF TESTING REQUIREMENTS FOR DIRECT-FIRED

AIR-COOLED AND WATER-COOLED ABSORPTION COOLING SYSTEMS

a) Air-Cooled Systems

TEST	(CONDENSER) OUTDOOR DB TEMPERATURE °C (°F)	WATER CHILLERS (EVAPORATOR COIL)	AIR CONDITIONERS (COOLING COIL)
		ENTERING WATER TEMPERATURE °C (°F)	DB/WB TEMPERATURE OF RETURN AIR °C (°F)
A*) Steady-State	35.0 (95)	12.8 (55)	26.7/19.4 (80/67)
B Steady-State	27.8 (82)	**	26.7/19.4 (80/67)
C Steady-State	27.8 (82)	12.8 (55)	26.7/19.4 (80/<67)(dry coil)+
D***) Cyclic	27.8 (82)	12.8 (55)	26.7/19.4 (80/<67)(dry coil)+

b) Water-Cooled Systems

TEST	CONDENSER UNIT INLET WATER TEMPERATURE °C (°F)		WATER CHILLERS (EVAPORATOR COIL)	AIR CONDITIONERS (COOLING COIL)
	GROUND WATER	COOLING TOWER (Air-Cooled)	ENTERING WATER TEMPERATURE °C (°F)	DB/WB TEMPERATURE OF RETURN AIR °C (°F)
A*) Steady-State	21.1 (70)	35 (95)	12.8 (55)	26.7/19.4 (80/67)
B Steady-State	15.6 (60)	23.9 (75)	**	26.7/19.4 (80/67)(wet coil)
C Steady-State	15.6 (60)	23.9 (75)	12.8 (55)	26.7/19.4 (80/ 67)(dry coil)+
D***) Cyclic	15.6 (60)	23.9 (75)	12.8 (55)	26.7/19.4 (80/ 67)(dry coil)+

*) Standard Rating point.

**) Test B is not conducted for water chillers.

***) Test D is conducted with 20% burner on-time and 80% burner off-time; the pumps, fans, etc. should run in accordance with their normal control system as it responds to the burner condition.

+) A wet-bulb temperature sufficiently low so as to result in a completely dry (non-condensing) evaporator coil surface.

TABLE 2

SUMMARY OF TESTING REQUIREMENTS FOR DIRECT-FIRED AIR-SOURCE AND WATER-SOURCE ABSORPTION HEATING SYSTEMS

a) Air-Source Systems

TEST		AIR TEMPERATURE ENTERING EVAPORATOR °C (°F)	HEAT PUMPS	
			ENTERING WATER TEMPERATURE °C (°F)	DB/WB TEMPERATURE OF INDOOR RETURN AIR °C (°F)
A)	High-Temperature (Steady-State)	T_{DB} 8.3 (47) T_{WB} 6.1 (43)	40.6 (105)	21.1 (70)/<15.6 (60)
B)	Frost Accumulation	T_{DB} 1.7 (35) T_{DP} -1.1 (30)	40.6 (105)	21.1 (70)/<15.6 (60)
C*)	Low Temperature (Steady-State)	T_{DB} -8.3 (17) T_{WB} -9.4 (15)	40.6 (105)	21.1 (70)/<15.6 (60)
D**)	Cyclic	T_{DB} 8.3 (47) T_{WB} 6.1 (43)	40.6 (105)	21.1 (70)/<15.6 (60)

b) Water-Source Systems

TEST		EVAPORATOR INLET GROUND WATER °C (°F)	HEAT PUMPS	
			ENTERING WATER TEMPERATURE °C (°F)	DB/WB TEMPERATURE OF INDOOR RETURN AIR °C (°F)
A)	High Temperature (Steady-State)	21.1 (70)	40.6 (105)	21.1 (70)/<15.6 (60)
B)	Frost Accumulation	--	--	--
C*)	Low Temperature (Steady-State)	15.6 (60)	40.6 (105)	21.1 (70)/<15.6 (60)
D**)	Cyclic	21.1 (70)	40.6 (105)	21.1 (70)/<15.6 (60)

* Standard Rating Point

** Test D is conducted with 20% burner on-time and 80% burner off-time. The entire length of the cycle is determined by the thermostat characteristics supplied by the manufacturer; all pumps, fans, etc. should run in accordance with their normal control system as it responds to the burner conditions.

TABLE 3

RATING SHEET FOR DIRECT-FIRED ABSORPTION COOLING SYSTEMS

FUEL: ☐ GAS ☐ OIL ☐ OTHER ________

FUEL HIGHER HEATING VALUE: ______________________ Btu/lbm or Btu/Ft^3

FUEL COST: ______ $/kBtu   COST OF ELECTRICAL POWER: ____________ $/kWh

DESIGN: ☐ Single-Effect ☐ Double Effect

COOLING LOAD HOURS ☐

THERMOSTAT ON-OFF TIME: ____________

TYPE: ☐ Water Chiller
- ☐ Ground Water Sink
- ☐ Cooling Tower Sink
- ☐ Outdoor Air Sink

☐ Air Conditioner
- ☐ Ground Water
- ☐ Cooling Tower (or water sprayed on outdoor coil)
- ☐ Outdoor Air

REFRIGERANT/ABSORBENT: ______________________

PERFORMANCE PARAMETER	TEST			
	A	B**	C	D
COOLING CAPACITY kW (kBtu/hr)				
COEFFICIENT OF PERFORMANCE				
FUEL FLOWRATE kg/hr (lbm/hr or CFM)*				
POWER CONSUMPTION (kWh)				
EVAPORATOR (Chilled Fluid) FLOWRATE kg/hr (lbm/hr or CFM)*				
CONDENSER UNIT FLOWRATE kg/hr (lbm/hr or CFM)				

SEASONAL PERFORMANCE FACTOR   SEASONAL OPERATING COST $

*Flowrates determined at rating conditions.
**Test is not provided for water chillers.

TABLE 4

FRACTIONAL TEMPERATURE BIN HOURS FOR COOLING SEASON CALCULATION

Bin No., j	Bin Temperature, T_j, °F	Fraction of Total Temperature Bin Hours n_j/N
1	67	0.214
2	72	0.231
3	77	0.216
4	82	0.161
5	87	0.104
6	92	0.052
7	97	0.018
8	102	0.004

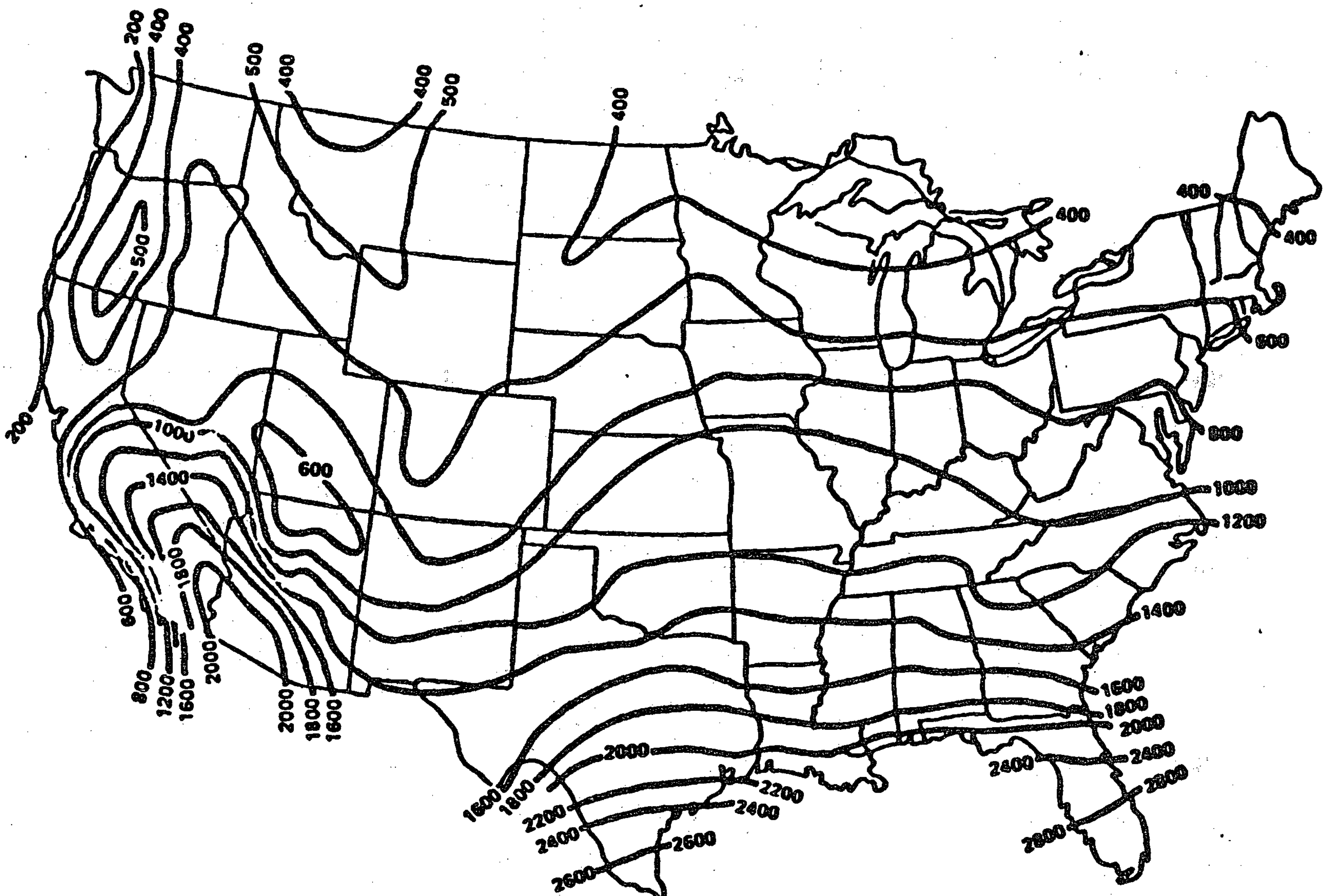
Cooling Load Hours, CLH - 3825

Outdoor Design Cooling Temperature - 35°C (95°F)

Ground Water Temperature - 15.6°C (60°F)

TABLE 5

DISTRIBUTION OF ACTUAL COOLING LOAD HOURS (CLH) THROUGHOUT THE UNITED STATES



Alaska	– 0	Cooling	Load	Hours	
Canal Zone	– 6,000	"	"	"	
Guam	– 6,600	"	"	"	
Hawaii	– 2,300	"	"	"	
Puerto Rico	– 6,000	"	"	"	
Samoa	– 6,600	"	"	"	
Virgin Island	– 6,000	"	"	"	

TABLE 6

RATING SHEET FOR DIRECT-FIRED ABSORPTION HEATING SYSTEMS

FUEL: ☐ Gas ☐ Oil ☐ Other ________

FUEL HIGHER HEATING VALUE: ____________________ Btu/lbm or Btu/ft^3

FUEL COST: ________ \$/kBtu COST OF ELECTRICAL POWER: ________ \$/kWh

DESIGN: ☐ Single Effect ☐ Double Effect

MAJOR CLIMATIC REGION (USA): ____________________ (See Table 7, 8)

THERMOSTAT ON-OFF TIME: ____________________

REFRIGERANT/ABSORBENT: ____________________

TYPE: ☐ Hot Air Heat ☐ Hot Water Heat

☐ Ground Water Source ☐ Ground Water Source

☐ Outdoor Air Source ☐ Outdoor Air Source

PERFORMANCE PARAMETER	TEST			
	A	B**	C	D
HEATING CAPACITY kW (kBtu/hr)				
COEFFICIENT OF PERFORMANCE				
FUEL FLOWRATE kg/hr (lbm/hr or CFM)*				
POWER CONSUMPTION (kWh)				
EVAPORATOR (Source Fluid) FLOWRATE kg/hr (lbm/hr or CFM)*				
CONDENSER UNIT FLOWRATE kg/hr (lbm/hr or CFM)				

SEASONAL PERFORMANCE FACTOR SEASONAL OPERATING COST \$

*Flowrates determined at rating conditions.
**Test B is not for water source.

TABLE 7

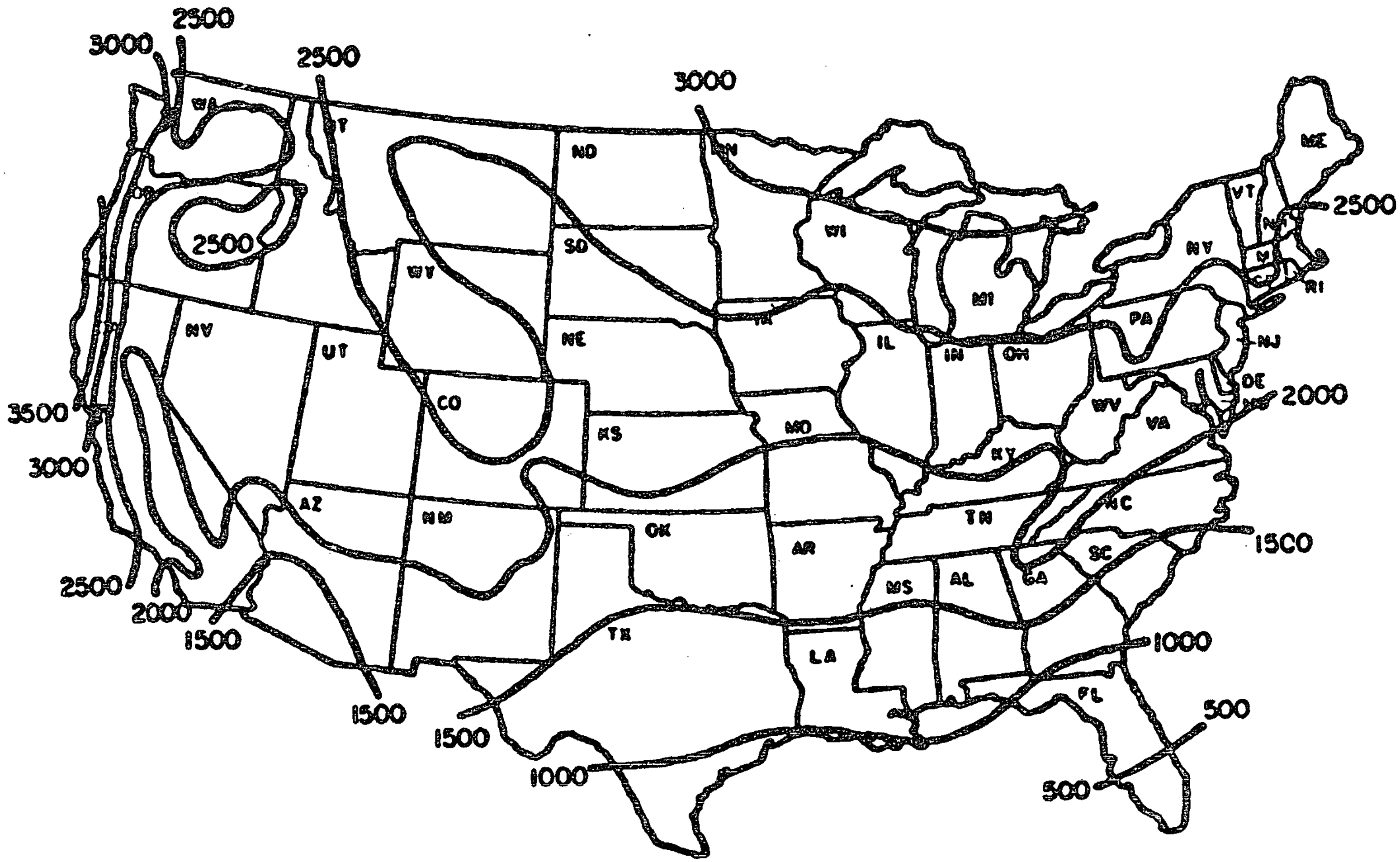
MAJOR CLIMATIC REGIONS IN THE CONTINENTAL USA

REGION		I	II	III	IV	V	VI
HEATING LOAD HOURS, HLH		750	1250	1750	2250	2750	2750*
OUTDOOR DESIGN TEMPERATURE, T_H FOR THE REGION, °F		37	27	17	5	-10	30
FRACTIONAL HOURS: n_j/N							
Bin #	Bin Temp.						
j = 1	T_j(°F) = 62	.291	.215	.153	.132	.106	.113
2	57	.239	.189	.142	.111	.092	.206
3	52	.194	.163	.138	.103	.086	.215
4	47	.129	.143	.137	.093	.076	.204
5	42	.081	.112	.135	.100	.078	.141
6	37	.041	.088	.118	.109	.087	.076
7	32	.019	.056	.092	.126	.102	.034
8	27	.005	.024	.047	.087	.094	.008
9	22	.001	.008	.021	.055	.074	.003
10	17	0	.002	.009	.036	.055	0
11	12	0	0	.005	.026	.047	0
12	7	0	0	.002	.013	.038	0
13	2	0	0	.001	.006	.029	0
14	-3	0	0	0	.002	.018	0
15	-8	0	0	0	.001	.010	0
16	-13	0	0	0	0	.005	0
17	-18	0	0	0	0	.002	0
18	-23	0	0	0	0	.001	0

*In Pacific Coast Region

TABLE 8

HEATING LOAD HOURS (HLH) FOR THE UNITED STATES



This map is reasonably accurate for most parts of the United States but is necessarily highly generalized and consequently not too accurate in mountainous regions, particularly in the Rockies.

TABLE 9

RECOMMENDED CALCULATION SHEET FOR DETERMINING SPF AND SOC OF DIRECT-FIRED ABSORPTION SYSTEMS

APPLICATION: ☐ Cooling ☐ Heating

CLIMATIC REGION: ______________________________

TOTAL BIN HOURS: ______________________________

FUEL: ☐ Gas ☐ Oil ☐ Other ____________

FUEL HEATING VALUE (HHV): ______ Btu/lbm or Btu/ft^3 Cost of Fuel ______ $/Btu Cost of Electricity ______ $kWh

A	B	C	D	E	F	G	H
( ) BIN NUMBER	( ) BIN TEMPERATURE (°F)	( ) $\frac{n_j}{N}$*	( ) BUILDING LOAD $\cdot\, n_j/N$ (Btu/hr)	( ) HLF OR CLF (T_j)	( ) TOTAL ENERGY CONSUMPTION $\cdot\, n_j/N$ (Btu/hr)	( ) FUEL CONSUMPTION $\cdot\, n_j/N$ (Btu/hr)	( ) ELECTRICITY CONSUMPTION $\cdot\, n_j/N$ (Btu/hr)
			ΣD		ΣF	ΣG	ΣH

$$SPF = \frac{\Sigma D}{\Sigma F}$$

*Fractional Bin Hours

Degradation Factor ( ) C_D = ____________

$$SOC = \Sigma F\left[xC_f + (1 - x)\frac{C_e}{3.413}\right] \cdot N + \underbrace{\begin{matrix}[RH \cdot 3.413] \cdot N \\ \text{electric or gas} \\ [\dot{m}_f \cdot HHV] \cdot N\end{matrix}}_{\text{For Heating Only}}$$

TABLE 10

SAMPLE CALCULATIONS FOR ABSORPTION COOLING SYSTEMS

A	B	C	D	E	F	G	H
BIN NUMBER	BIN TEMPERATURE (°F)	$\frac{n_j}{N}$*	BUILDING LOAD · n_j/N (Btu/hr)	CLF (T_j)	FUEL ENERGY CONSUMPTION · n_j/N (Btu/hr)	FUEL CONSUMPTION · n_j/N (Btu/hr)	ELECTRICITY CONSUMPTION · n_j/N (watts)
1	67	.214	425	0.048	1002	952	15
2	72	.231	1607	0.174	3820	3629	56
3	77	.216	2576	0.310	6192	5882	91
4	82	.161	2720	0.458	6623	6292	97
5	87	.104	2274	0.619	5608	5328	82
6	92	.052	1395	0.795	3486	3312	51
7	97	.018	573	0.986	1446	1374	21
8	102	.004	147	1	325	309	5
Σ			11717		28502	27078	418

*Fractional Bin Hours

$PLF = \frac{0.411}{0.454} = 0.905$ $C_D = \frac{1 - 0.905}{1 - 0.458} = 0.175$ $SPF = \frac{11717}{28502} = 0.41$ SOC = \$613.67

CLH = 3825 Hours $C_f = 0.005$ \$/kBtu $C_e = 0.006$ \$/kWh

TABLE 11

SAMPLE CALCULATIONS FOR ABSORPTION HEATING SYSTEMS

A	B	C	D	E	F	G	H
BIN NUMBER	BIN TEMPERATURE (°F)	$\frac{n_j}{N}$*	BUILDING LOAD $\cdot n_j/N$ (Btu/hr)	HLF (T_j)	FUEL ENERGY CONSUMPTION $\cdot n_j/N$ (Btu/hr)	FUEL CONSUMPTION $\cdot n_j/N$ (Btu/hr)	ELECTRICITY CONSUMPTION $\cdot n_j/N$ (watts)
1	62	.106	130.4	.022	274.9	237.1	2.9
2	57	.092	301.7	.059	559.5	537.1	616
3	52	.086	458.3	.099	852.9	818.8	10.0
4	47	.076	560.8	.140	1306.1	994.7	12.1
5	42	.078	735.5	.182	1343.8	1290.1	15.8
6	37	.087	998.7	.227	1815.3	1742.7	21.3
7	32	.102	1380.0	.273	2485.9	2386.5	29.1
8	27	.094	1464.4	.322	2621.7	2516.8	30.7
9	22	.074	1304.5	.373	2319.0	2226.2	27.2
10	17	.055	1082.3	.427	1912.3	1835.8	22.4
11	12	.047	1021.3	.484	1794.0	1722.2	21.0
12	7	.038	903.6	.543	1575.9	1512.9	18.5
13	2	.029	749.0	.605	1296.9	1245.0	15.2
14	-2	.018	501.9	.672	864.2	829.6	10.1
15	-7	.010	299.3	.741	511.7	491.2	6.0
16	-13	.005	159.9	1	306.8	294.5	3.6
17	-18	.002	68.1	1	122.7	117.8	1.4
18	-23	.001	36.1	1	61.4	58.9	0.7
Σ			12155.9		21727.1	20857.0	254.6

*Fractional Bin Hours

$PLF = \frac{0.73}{1.16} = 0.63$ $C_D = \frac{1 - 0.63}{1 - 0.14} = 0.43$ $SPF = \frac{12155.7}{21727.1} = 0.56$ SOC = $328.81

HLH = 2750 C_f = 0.005 \$/kBtu C_e = 0.06 \$/kWh

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10. SUPPLEMENTARY NOTES

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)*

Test and rating procedures are presented for gas-fired absorption devices operating in either the heating or cooling modes. These procedures are designed to include the effects of part-load and cyclic operation, variations in outdoor temperature, and frost formation during the heating mode. Both air-source and ground water source absorption heat pumps are considered, as well as air cooled and ground water cooled air-conditioners and water chillers. A calculation procedure is presented for estimating the heating and cooling seasonal performance and cost of operation of residential water chillers, air-conditioners, and heat pump units.

12. KEY WORDS *(Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)*

central air-conditioners; central heating equipment; cooling seasonal performance; heat pumps; heating seasonal performance; rating procedure; seasonal cost of operation; test method

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