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

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METHOD OF TESTING, RATING AND ESTIMATING THE SEASONAL PERFORMANCE OF GROUND-WATER-SOURCE HEAT PUMPS

William J. Mulroy

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U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, *Secretary*
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

ABSTRACT

The National Bureau of Standards has made a study of the part-load and seasonal performance of residential water source heat pumps operating in both heating and cooling modes. This document outlines methods for testing and rating these units which account for the variation in performance due to part-load operation and change in source water temperature. A calculation procedure is presented which can be used to estimate the seasonal performance and seasonal cost of operation of residential water source heat pumps.

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

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NOMENCLATURE

$BL(T_j)$ Building load at an outdoor dry-bulb temperature T_j, kBtu/h (kW)

C_D Degradation factor for cyclic operation, defined by equation 3.3

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 3.4.

CLH Cooling load hours, defined as the number of hours in a cooling season that a building requires cooling. See table 5.

COP Coefficient of performance, defined as the net heating done over a specified period of time divided by the total electrical energy input over the same time interval.

C_{pa} Specific heat at constant pressure of air-water mixture per pound of dry air, $C_{pa} = 0.24 + 0.444\ W_n$, Btu/lb°F, ($1.01 + 1.86\ W_n$, kJ/kg•°C).

CSPF Cooling seasonal performance factor; defined as the ratio of the total cooling done to the total energy usage over a cooling season.

E Total seasonal electrical energy usage, kW.

$\dot{E}_{ss}(T)$ Steady-state total power input at a water temperature T, kW.

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 cooling done over the same time period at constant ambient conditions. See equation 3.4.

HLH Heating load hours, defined by table 5.

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 6 and 7.

LF Load Factor, defined as the ratio of the total cyclic cooling or heating done in a complete cycle of specified period consisting of an "on"-time and "off"-time to the steady-state cooling or heating done over the same time period at constant ambient conditions. See equation 1.2.

N Total number of temperature bin hours.

$\frac{n_j}{N}$ Fractional number of temperature bin hours, see tables 6 and 7.

PLF Part-load factor, defined as the ratio of the cyclic COP to the steady-state COP, see equation 4.30.

Q Total seasonal cooling or heating done, kBtu (kJ).

Q_{cyc} Total cooling or heating done over a cycle consisting of one compressor "off" period and one compressor "on" period, kBtu (kJ).

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

$\dot{Q}_{cyc}$ Cyclic total capacity defined as the ratio of the total cooling done over a given time period to the duration of time the compressor is on in that period, kBtu/h (kW).

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

$RH(T_j)$ Resistance heat energy usage in temperature bin; kW - h, see equation 4.31.

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

T_j Representative outdoor dry-bulb temperature for temperature bin j, °F (°C). See tables 6, 7.

$T_{a1}(t)$ Dry-bulb temperature of air entering the indoor coil, °F (°C).

$T_{a2}(t)$ Dry-bulb temperature of air leaving the indoor coil, °F (°C).

t Time, hours

t_{on} Duration of time the compressor is on in one cycle consisting of a compressor on-time and a compressor off-time, hours

$\dot{V}$ Indoor air flow rate, cfm (m^3/s), 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, ft^3/lb (m^3/kg).

W_n Humidity ratio, lb/lb (kg/kg).

$X(T_j)$ Load factor at an outdoor dry-bulb temperature T_j, defined by equation 4.29.

j - Temperature bin number, see Table 3

c - cooling

H - heating

X - Same as $X(T_j)$

τ - Time period consisting of a compressor on-time and compressor off-time [h]

Γ - Defined by equation 3.2, °F.

SUBSCRIPTS AND SUPERSCRIPTS

ss - steady-state

cyc - cyclic

dry - dry-coil

SI CONVERSION FACTORS

Multiply	By	To Obtain
Btu, IT	1.055	kJ
Btu/h, Btuh	0.293	W
Btu/lb°F, c, 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/lb	0.0623	m^3/kg
ft^3/min, cfm	0.472	--
gpm (US)	0.0631	--
inch	25.4	mm
inch of water	3.38	kPa
kBtu/h	1055	kJ
lb/h	0.126	g/s

1. INTRODUCTION

1.1 SCOPE AND PURPOSE

The recommended test and rating procedures proposed herein are intended to be applied to packaged, unitary, ground-water-to-air heat pumps, driven by single phase electric power which are intended for residential applications. They do not apply to water source heat pumps used for intrastructure heat transfer, air source heat pumps, hybrid systems which supplement the heat pump output by burning fossil fuel, to two speed compressor units, or to units with two compressors.

The steady-state rating points proposed herein were chosen to be in conformance with ARI (Air Conditioning and Refrigeration Institute) Standard 325-82, "Ground Water Source Heat Pumps" [1]. This rating procedure differs from that of ARI 325-82 in that it includes procedures for calculating seasonal efficiency and cost of operation and does not include quality, reliability or safety criteria. Also the procedure herein includes test series (Tests I, J, K, L, M, N) which may be employed if one wants to predict either the potential degradation because of cycling phenomena in situ or the merits of a particular off-cycle innovation (e.g., auto-switch for crankcase heater). Typically, these cycling effects could cause a difference between seasonal and steady-state of the order of 10%. The steady-state tests (E, F, G, H) would be expected to yield the same ratings as the ARI procedure.

The recommended testing and calculation procedures are intended to provide ratings comparable to those resulting from the current air source heat pump test procedures [3, 4].

Rating points and calculation procedures for solar-water-source heat pumps are not covered because of the lack of standardization in system design for this application. It is believed, however, that the data resulting from tests according to this Test Procedure would allow calculation of the seasonal efficiency and energy cost for many solar systems once their design has been established. Water-source heat pumps for heat transfer within structures are not covered because of the predominantly commercial nature of their application.

The purpose of these recommended test and rating requirements is, generally, to provide performance information from which a heat pump unit may be properly sized to a heating or cooling load, to provide information from which a consumer may estimate heating and cooling costs, and to provide information from which seasonal energy efficiency and operating cost comparisons can be made between heat pump models and between different types of heating systems.

2. RECOMMENDED TESTING AND RATING REQUIREMENTS

2.1 INTRODUCTION

The heat pump test operating conditions described in section 2.2.1 through 2.2.4 were selected to reflect the various modes of operation a heat pump may commonly encounter. These operating points were also chosen so as to be able to estimate the seasonal performance with a minimum amount of testing.

A steady-state cooling test at a 70°F (21.1°C) entering water temperature is required (test E). In addition a low temperature steady-state cooling test (test F) is required at a 50°F (10°C) entering water temperature. The addition of the second test makes possible interpolation (or extrapolation) of the test data to conditions other than those used for the tests.

Similarly two steady-state heating tests are required so that data interpolation or extrapolation is possible for heating applications. An entering water temperature of 70°F (21.1°C) is specified for the steady-state high temperature heating test (test G) and an entering water temperature of 50°F (10°C) is specified for the steady-state low temperature heating test (test H).

These required steady-state test conditions are the same as those required by ARI (Air Conditioning and Refrigeration Institute) Standard 325-82.

Since properly sized heat pumps will cycle on and off to meet small heating and cooling loads, cycling tests are included to determine the part load performance degradation at each steady-state rating point. The percent on-time for these tests was chosen to be 20 percent. In order to limit temperature swings inside the house, it is common practice to design thermostats so that they cause the unit to cycle at approximately 3 cph at a 50 percent on-time. This, assuming the control thermostat produces a parbolic relationship between cph and on-time, results in cycling rate of approximately 2 cph at a percent on-time of 20 percent, which corresponds to a cycling period of 30 minutes. The on and off-time for the cycling tests were therefore chosen to be 6 minutes and 24 minutes, respectively.

The cyclic cooling tests are to be performed with the same entering air dry-bulb temperature and entering water temperatures as the steady-state cooling rating point tests, but, because of inaccuracies in available methods of cyclic latent cooling capacity measurement, with the entering air wet-bulb temperature reduced sufficiently to insure a dry coil (no condensate collection). For comparison, two additional steady-state cooling tests are required that are identical to the other steady-state rating point tests but with the entering air wet-bulb temperature reduced sufficiently to insure a dry coil condition.

The cyclic heating tests are to be performed at the same entering air and water conditions as the steady-state heating tests.

The ratio of the coefficients of performance measured during comparable cyclic and steady-state tests is used with the load factor, LF, to calculate the degradation coefficient, C_D, of the heat pump using the following formulas:

$$CD = \frac{1 - \frac{COP_{cyc}}{COP_{ss}}}{1 - LF} \tag{1.1}$$

$$LF = \frac{\%RT \times \dot{Q}_{cyc}}{100 \times \dot{Q}_{ss}} \tag{1.2}$$

where COP_{cyc}, COP_{ss}, $\dot{Q}_{cyc}$, and $\dot{Q}_{ss}$ are repectively the unit's coefficient of performance, and capacity, measured at comparable cyclic and steady-state conditions, and percent RT is the percent run time of the unit during cyclic operation. The degradation coefficient is used in the seasonal performance calculations described in section 4. Laboratory tests have shown that the degradation factor varies with water temperature. In order to reduce the test burden this variation is assumed to be linear in this test method.

It is recommended that when C_D has been determined at both high and low temperature test points in an operational mode (heating or cooling), values of C_D for operation in the same mode at other water temperatures be obtained by linear interpolation or extrapolation from the two measured values.

There are a total of six tests to determine the cyclic performance of the water-source heat pump unit (four cyclic and two steady-state dry coil tests). Manufacturers may perform all six tests or may select one of three options (see section 2.2) which reduce or eliminate cyclic testing, but require simplifying assumptions to be made about the degradation coefficient of the unit. It is felt that these options were necessary in order to accommodate the small manufacturer who might not have test facilities capable of performing the cycling tests within the tolerances required for accuracy, or for units produced in production runs too small to justify this additional testing burden.

2.2 TEST REQUIREMENTS

Heat pumps shall be subjected to the following tests described in sections 2.2.1 through 2.2.4 (test identification letters have been selected starting with E to avoid confusion with air-source unit tests A through D of the DOE Test Procedure for Air Conditioners and Heat Pumps). All cooling mode tests are to be omitted for heating-only units:

Test E. High Temperature Cooling Test (omit for heating only units)
Test F. Low Temperature Cooling Test (omit for heating only units)
Test G. High Temperature Heating Test
Test H. Low Temperature Heating Test

Depending on any of the four alternatives discussed below some or all of the following optional tests may also be performed.

Test I. High Temperature Dry Coil Cooling Test
Test J. Low Temperature Dry Coil Cooling Test
Test K. High Temperature Cyclic Cooling Test
Test L. Low Temperature Cyclic Cooling Test
Test M. High Temperature Cyclic Heating Test
Test N. Low Temperature Cyclic Heating Test

In lieu of performing all six of the additional tests (test I through test N) which are required to determine the cyclic performance at each rating point the manufacturer may choose one of the following three options:

1) Perform the high temperature cooling cyclic test (test K) and the comparable high temperature cooling dry coil steady-state (test I) and assume that the resulting coefficient of degradation, C_D, is typical of all cooling operation. Perform the low temperature cyclic heating test (test N) and assume that the resulting C_D is typical of all heating operation.

2) Assume a constant cooling value of $C_D = 0.25$. Perform the low temperature cyclic heating test (test N) and assume that the resulting C_D is typical of all heating operation.

3) Assume a constant $C_D = 0.25$ for both heating and cooling.

2.2.1 General Test Condition Requirements for All Tests

All tests shall be conducted at the standard rating conditions specified in section 5.1.4 of ARI Standard 325-82. Table 1 summarizes the test condition requirements for the different tests.

2.2.2 Steady-State Test Conditions

2.2.2.1 <u>Test E, High Temperature Cooling Test Conditions</u>

The high temperature steady-state cooling test shall be conducted with an entering water temperature of 70°F (21.1°C). The dry-bulb air temperature entering and surrounding the unit shall be 80°F (26.7°C) and the wet-bulb air temperature entering the unit shall be 67°F (19.4°).

The test shall be conducted according to the test procedures outlined in section 3.2 for steady-state tests. The duration of the test shall be a minimum of 1/2 hour.

2.2.2.2 <u>Test F, Low Temperature Cooling Test Conditions</u>

The low-temperature steady-state cooling test shall be conducted with an entering water temperature of 50°F (10°C). The dry-bulb air temperature entering and surrounding the unit shall be 80°F (26.7°C) and the wet-bulb air temperature entering the unit shall be 67°F (19.4°C).

The test shall be conducted according to the test procedures outlined in section 3.2 for steady-state tests. The duration of the test shall be a minimum of 1/2 hour.

2.2.2.3 Test G, High Temperature Heating Test Conditions

The high temperature steady-state heating test shall be conducted with an entering water temperature of 70°F (21.1°C). The dry-bulb air temperature entering and surrounding the unit shall be 70°F (21.1°C) and the wet-bulb air temperature entering the unit shall not exceed 60°F (15.6°C).

The test shall be conducted according to the test procedures outlined in section 3.2 for steady-state tests. The duration of the test shall be a minimum of 1/2 hour. If elected (option 1 in table 2), the High Temperature Cyclic Heating test (test M, section 2.2.4.1) shall be performed immediately following this test.

2.2.2.4 Test H, Low Temperature Heating Test Conditions

The low temperature steady-state heating test shall be conducted with an entering water temperatures of 50°F (10°C). The dry-bulb air temperature entering and surrounding the unit shall be 70°F (21.1°C) and the wet-bulb air temperature entering the unit shall not exceed 60°F (15.6°C).

The test shall be conducted according to the test procedures outlined in section 3.2 for steady-state tests. The duration of the test shall be a minimum of 1/2 hour. If elected (option 1, 2, or 3 in table 2), the High Temperature Cyclic Heating Test (test N, section 2.2.4.2) shall be performed immediately following this test.

2.2.3 Conditions for Optional Dry Coil Steady-State and Cyclic Cooling Tests

2.2.3.1 Optional Test I, High Temperature Dry Coil Cooling Test Conditions

The steady-state dry coil cooling test at a 70°F (21.1°C) entering water temperature shall be conducted with the same water flow rate as was used for test E, High Temperature Cooling Test. The dry-bulb air temperature entering and surrounding the unit shall be 80°F (26.7°C) and the wet-bulb temperature entering the unit shall at no time exceed that value which results in the production of condensate by the cooling coil at this test condition.

The test shall be conducted according to the test procedures outlined in section 3.2 for steady-state tests. The duration of the test shall be a minimum of 1/2 hour. The High Temperature Cyclic Cooling Test (test K, section 2.2.3.3) shall be performed immediately following this test.

2.2.3.2 Optional Test J, Low Temperature Dry Coil Cooling Test Conditions

The steady-state low temperature dry coil cooling test shall be conducted with the same water flow rate and 50°F (10°C) entering water temperature as was used for test F, Low Temperature Cooling Test. The dry-bulb air temperature entering and surrounding the unit shall be 80°F (26.7°C) and the wet-bulb temperature entering the unit shall at no time exceed that value which results in the production of condensate by the cooling coil at this test condition. The test shall be conducted according to the test procedures outlined in section 3.2 for steady-state tests. The duration of the test shall be a minimum of 1/2 hour. The Low Temperature Cyclic Cooling Test (test L, section 2.2.3.4) shall be performed immediately following this test.

2.2.3.3 Optional Test K, High Temperature Cyclic Cooling Test Conditions

The cyclic cooling performance test at a 70°F (21.1°C) entering water temperature shall be performed immediately following the steady-state High Temperature Dry Coil Cooling Test (test I, section 2.2.3.1). The steady-state test results and the cyclic test results are used together to find the performance degradation resulting from cyclic operation at high water temperatures. The water flow rate during the on-period shall be the same as the rate during test I.

The dry-bulb and wet-bulb temperature of the air entering and surrounding the unit shall be the same as test I. The cyclic performance test is conducted according to the test procedures outlined in section 3.3.

During the cyclic test, the water flow and the indoor fan shall cycle "on" and "off" as the compressor 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 compressor cycling times shall be 6 minutes "on" and 24 minutes "off".

2.2.3.4 Optional Test L, Low Temperature Cyclic Cooling Test Conditions

The low temperature cyclic cooling performance test shall be performed immediately following the steady-state Low Temperature Dry-Coil Cooling Test (test J, section 2.2.3.2). The steady-state test results and the cyclic test results are used together to find the performance degradation resulting from cyclic operation at low water temperatures. The entering water temperature and flow rate during the on-period shall be the same as that during test J.

The dry-bulb and wet-bulb temperature of the air entering and surrounding the unit shall be the same as test J. The cyclic performance test is conducted according to the test procedures outlined in section 3.3.

During the cyclic test, the water flow and the indoor fan shall cycle "on" and "off" as the compressor cycles "on" and "off", except that the indoor fan cyclic times may be delayed due to controls that are normally installed with the unit. The compressor cycling times shall be 6 minutes "on" and 24 minutes "off".

2.2.4 Conditions for Optional Cyclic Heating Tests

2.2.4.1 Optional Test M, High Temperature Cyclic Heating Test Conditions

The cyclic heating performance test at a 70°F (21.1°C) entering water temperature shall be performed immediately following the steady-state High Temperature Heating Test (test G, section 2.2.2.3). The steady-state test results and the cyclic test results are used together to find the performance degradation resulting from cyclic operation at high water temperatures. The water flow rate during the on-period shall be the same as for test G. The dry-bulb and wet-bulb temperature of the air entering and surrounding the unit shall also be the same as test G. The cyclic performance test is conducted according to the test procedures outlined in section 3.3.

During the cyclic test, the water flow and indoor fan shall cycle "on" and "off" as the compressor 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 compressor cycling times shall be 6 minutes "on" and 24 minutes "off".

2.2.4.2 Optional Test N, Low Temperature Cyclic Heating Test Conditions

The low temperature cyclic heating performance test shall be performed immediately following the steady-state Low Temperature Heating Test (test H, section 2.2.2.4). The steady-state test results and the cyclic test results are used together to find the performance degradation resulting from cyclic operation at low water temperatures. The entering water temperature and flow rate during the on-period shall be the same as for test H. The dry-bulb and wet-bulb temperature of the air entering and surrounding the unit shall also be the same as test H. The cyclic performance test is conducted according to the test procedures outlined in section 3.3.

During the cyclic test, the water flow and the indoor fan cycle "on" and "off" as the compressor 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 compressor cycling times shall be 6 minutes "on" and 24 minutes "off".

3. HEAT PUMP PERFORMANCE TEST PROCEDURES

3.1 INTRODUCTION

The test procedures are categorized as follows:

Steady-State Test Procedure - section 3.2

Cyclic Test Procedure - section 3.3

The tests are designed to determine the performance of the heat pump unit under conditions of typical use, as well as to establish a performance curve from which the seasonal performance and cost of operation may be determined.

3.2 STEADY-STATE TEST PROCEDURE

3.2.1 Available Test Methods

Listed below and in the ASHRAE Standard 37-78 are the recommended test methods for conducting steady-state tests of heat pumps.

i) Air Enthalpy Method - Indoor-Side
ii) Water Coil Method - Outdoor-Side
iii) Compressor Calibration Method
iv) Volatile Refrigerant Flow Method

The Air Enthalpy Method - Indoor Side shall be used as the primary method for all tests. In addition, one other method chosen from the above list shall simultaneously be employed in tests E, F, G, and H as a check on the Air Enthalpy Method - Indoor Side. The steady-state capacities, as measured by the two methods and calculated according to the procedures outlined in ASHRAE Standard 37-78, shall agree within 6 percent in order to constitute a valid test. The results of the Air Enthalpy Method - Indoor Side shall, however, be the only results reported and utilized in the heating seasonal performance calculations of section 4. Units containing the compressor located with the indoor section and separately ventilated shall not use a test method which includes compressor heat rejection in the determination of heating capacity.

Requirements to be followed in employing the various test methods listed above as suitable for checking the results of the Air Enthalpy Method - Indoor Side are given in ASHRAE Standard 37-78 under the following sections.

Volatile Refrigerant Flow Method - section 5.1 - 5.5.1

Compressor Calibration Method - sections 4.1 - 4.6.1 except that only 4 sets of reading are required in section 4.2.1

Outdoor Water Coil Method - sections 6.1 - 6.6.1

results and the cyclic test results are used together to find the performance degradation resulting from cyclic operation at high water temperatures. The water flow rate during the on-period shall be the same as for test G. The dry-bulb and wet-bulb temperature of the air entering and surrounding the unit shall also be the same as test G. The cyclic performance test is conducted according to the test procedures outlined in section 3.3.

During the cyclic test, the water flow and indoor fan shall cycle "on" and "off" as the compressor 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 compressor cycling times shall be 6 minutes "on" and 24 minutes "off".

2.2.4.2 Optional Test N, Low Temperature Cyclic Heating Test Conditions

The low temperature cyclic heating performance test shall be performed immediately following the steady-state Low Temperature Heating Test (test H, section 2.2.2.4). The steady-state test results and the cyclic test results are used together to find the performance degradation resulting from cyclic operation at low water temperatures. The entering water temperature and flow rate during the on-period shall be the same as for test H. The dry-bulb and wet-bulb temperature of the air entering and surrounding the unit shall also be the same as test H. The cyclic performance test is conducted according to the test procedures outlined in section 3.3.

During the cyclic test, the water flow and the indoor fan cycle "on" and "off" as the compressor 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 compressor cycling times shall be 6 minutes "on" and 24 minutes "off".

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Outdoor Water Coil Method - sections 6.1 - 6.6.1

3.2.2 Air Flow Rate, Instrumentation and Data to be Recorded

Air Flow Rate

The indoor air flow rate shall be determined as described in sections 7.1 through 7.4.3 of ASHRAE Standard 37-78. In general, this requires the construction of an air receiving chamber and discharge chamber separated by a partition in which one or more nozzles are located. The receiving chamber is connected to the indoor air discharge side of the test specimen through a short plenum. The exhaust side of the air flow rate measuring device contains an exhaust fan with some means to vary its capacity to obtain the desired external resistance to air flow rate. The exhaust side is then left open to the test room or is ducted through a conditioning apparatus and then back to the test specimen inlet.

The static pressure across the nozzle(s), the velocity pressure, and static pressure at the nozzle throat shall be measured with manometers which will result in errors which are no greater than ± 1 percent of the reading and having minimum scale divisions not exceeding 2.0 percent of the reading or another measurement system of equivalent accuracy. Static pressure and temperature measurements must be taken at the nozzle throat in order to obtain density of the air. The area(s) of the nozzle(s) shall be determined by measuring their diameter with an error no greater than ± 0.2 percent. Four measurements will be made approximately 45 degrees apart around the nozzle in each of two planes through the nozzle throat, one plane at the outlet and the other plane in the straight section near the radius.

Electrical Measurements

The electrical energy usage of the indoor fan shall be measured with a watt-hour meter which will result in an error which is no greater than ± 0.5 percent of the quantity of energy measured. Likewise, the energy usage of the compressor and all other equipment components utilizing electrical energy shall be measured with a watt-hour meter of the stated accuracy.

Indoor-Air Entering and Discharge Temperature

Measurements of the air temperature entering and leaving the indoor coil or of the difference between the entering and leaving temperatures shall be made in accordance with the requirements of ASHRAE Standard 41.1-74 [5]. These temperatures shall be recorded at intervals of 5 minutes or less with instrumentation having a total system error no greater than ± .3°F (± .17°C) and a response time of 2.5 seconds or less in air at an average airspeed encountered in the region of measurement. Response time is defined as the time required for the instrumentation to obtain 63 percent of the final steady-state temperature difference when subjected to a step change in temperature of 15°F (8.3°C) or more. Temperature measurements are to be made upstream of the static pressure tap on the inlet and downstream of the static pressure taps on the outlet.

Indoor-Side Ambient Conditions

The indoor-side dry-bulb temperature shall be recorded at intervals of 5 minutes or less with instrumentation which will result in an error no greater than $\pm$ .3°F ($\pm$ .17°C).

Outdoor-Side Entering and Discharge Water Temperatures

The water temperatures entering and leaving the outdoor side heat exchanger and, if the Outdoor Coil Method (section 3.2.1) is used, the difference between these two temperatures shall be measured in accordance with the requirements of ASHRAE Standard 41.1-74 using sheathed thermocouples inserted through pipe plugs that are located within 12 inches (305 mm) of the inlet and outlet of the unit. A minimum of 6 inches (152.4 mm) of sheathed thermocouple lead, immediately upstream of the thermocouple junction, shall be immersed in the water between the pipe plug and the unit inlet/outlet. The remainder of the thermocouple lead shall be run along the surface of the pipe (and under any insulation) for a distance of not less than 3 feet (183 mm). The pipe three feet immediately before the inlet and three feet after the outlet shall be covered with 0.5 inch (13 mm) of closed cell foam rubber insulation or equivalent. These temperatures shall be recorded with instrumentation having a total system error no greater than $\pm$ 0.3°F ($\pm$ 0.17°C).

Liquid Flow Measurement

Liquid flow measurements shall be made in accordance with section 10.6 of ASHRAE Standard 37-78.

Water and brine flow rates shall be measured with a liquid flow meter or quantity meter having an error no greater than $\pm$ 1.0 percent of the quantity measured.

Static Pressure Measurements in Ducts

Static pressure measurement in the ducts and across the unit shall be made in accordance with section 8 of ASHRAE Standard 37-78 using a manometer which will result in an error no greater than $\pm$ 0.01 inches of water ($\pm$ 2.5 Pa) or another measurement system of equivalent accuracy.

Additional Instrumentation and Data Acquisition

Hand recorded data shall be taken at 5 minute intervals. The data recording interval for computerized data recording systems shall be 1 minute. Additional requirements for instrumentation and data acquisition are listed in sections 10 and table II of ASHRAE Standard 37-78. Requirements for the Volatile Refrigerant Flow method are listed in section 5.2 through 5.5 of ASHRAE Standard 37-78. Requirements for instrumentation when using the Outdoor Water Coil Method are found in sections 6.2 and 6.6 of ASHRAE Standard 37-78.

3.2.3 Test Preparation and Performance

The equipment under test shall be installed according to the requirements of section 11.2 of ASHRAE Standard 37-78. Briefly, this requires that the unit be installed according to the manufacturer's instructions, that no change in external resistance to air flow to account for barometric changes should be made, that pressure gages should be installed with short lengths of small diameter tubing, and that the unit is to be charged in accordance with the manufacturer's recommended procedures.

Test chamber requirements are the same as given in section 11.1 of the ASHRAE Standard 37-78. The test room apparatus and test specimens must be operated for at least one hour with at least 1/2 hour at equilibrium and at the specified test conditions prior to starting the test. The steady-state test shall then be conducted for a minimum of 1/2 hour with intermittent data recorded at 10-minute intervals. The Air Enthalpy Method - Indoor Side shall be used along with one other test method listed in section 3.2 conducted simultaneously. The steady-state test may only begin when the test unit and room conditions are within the test conditions tolerances specified in section 3.2.4.

3.2.4 Test Tolerances

All steady-state tests shall be conducted within the tolerances specified in table 3. Variation greater than those given in table 3 shall invalidate the test. The heating capacity results by the Air Enthalpy Method-Indoor side shall agree within 6 percent of the value determined by any other simultaneously conducted capacity test in order for the test to be valid.

3.2.5 Data Analysis and Results

The results of the steady-state tests shall include:

i) Cooling capacity, $\dot{Q}_{SS,C}(70)$ and $\dot{Q}_{SS,C}(50)$, kBtu/h

ii) Dry coil cooling capacity, $\dot{Q}_{SS,dry,C}(70)$ and $\dot{Q}_{SS,dry,C}(50)$, kBtu/h

iii) Heating capacity, $\dot{Q}_{SS,H}(70)$ and $\dot{Q}_{SS,H}(50)$, kBtu/h

iv) Electrical power input to all components, $\dot{E}_{SS,C}(70)$, $\dot{E}_{SS,C}(50)$, $\dot{E}_{SS,dry,C}(70)$, $\dot{E}_{SS,dry,C}(50)$, $\dot{E}_{SS,H}(70)$, $\dot{E}_{SS,H}(50)$, kW

v) Indoor air flow rate (SCFM) and external resistance to indoor air flow, inches of water

vi) Water flow rate, gpm

vii) Water pressure drop through the unit, psi

viii) Indoor wet bulb temperature and indoor dry-bulb temperature, °F

ix) Data to be recorded as specified in table 11 of ASHRAE Standard 37-78

Only the capacity results obtained by the Air Enthalpy Method Indoor side shall be reported and used in the calculation procedure described in section 4. The steady-state dry coil cooling tests (test I and test J) may be omitted and a constant degradation factor assumed (table 2).

The formulae to be used in calculating the heating capacity and the indoor air flow-rate are presented in sections 3.8.1 and 7.4 of ASHRAE Standard 37-78. The coefficient of performance, COP, is defined as the ratio of the heating capacity in kBtu/h to the product of 3.413 kBtu/h-kW (1000 W/kW) and the power inputs to the indoor fan and other equipment components in kW. For heat pumps units not having an indoor fan as part of the model tested, 1.250 kBtu/h per 1000 SCFM (7.77 watts per m^3/s) of indoor air handled shall be subtracted from the measured cooling capacities to obtain the total cooling capacities, $Q_{SS,C}(70)$, $Q_{SS,dry,C}(70)$, $Q_{SS,dry,C}(50)$, and $Q_{SS,C}(50)$, and 1.250 kBtu/h per 1000 SCFM(7.77 watts per m^3/s) of indoor air shall be added to the measured capacities to obtain the total heating capacities, $Q_{SS,H}(70)$ and $Q_{SS,H}(50)$. Also for the units not having an indoor fan as part of the model tested 0.365 kW per 1000 SCFM (7.77 watts per m^3/s) of indoor air handled shall be added to the measured power to obtain the total power inputs to the unit, $\dot{E}_{SS,C}(70)$, $\dot{E}_{SS,C}(50)$, $\dot{E}_{SS,dry,C}(70)$, $\dot{E}_{SS,dry,C}(50)$, $\dot{E}_{SS,H}(70)$ and $\dot{E}_{SS,H}(50)$.

For all units a penalty shall be added to the measured power input in the amount of 60 watts/gpm of water flow (951 w/ℓ/s) to obtain the total power inputs to the unit, $\dot{E}_{SS,C}(70)$, $\dot{E}_{SS,C}(50)$, $\dot{E}_{SS,dry,C}(70)$, $\dot{E}_{SS,dry,C}(50)$, $\dot{E}_{SS,H}(70)$, and $\dot{E}_{SS,H}(50)$.

3.3 CYCLIC TEST PROCEDURE

3.3.1 Available Test Methods

The cyclic performance tests must be carried out immediately following the steady-state tests at comparable conditions as described in section 2.2.2 through section 2.2.4. This is necessary since the results of both tests must be used to determine the part load degradation coefficient, C_D, described in section 3.3.5. The cyclic tests, comparable steady-state tests, and resulting degradation coefficients may be summarized as follows:

Cyclic Test	Follows Comparable Steady-State Test	Resulting in Degradation Coefficient
K, High Temp. Cyclic Cooling	I, High Temp. Dry-Coil Cooling	$C_{D,C}(70)$
L, Low Temp. Cyclic Cooling	J, Low Temp. Dry-Coil Cooling	$C_{D,C}(50)$
M, High Temp. Cyclic Heating	G, High Temp. Heating	$C_{D,H}(70)$
N, Low Temp. Cyclic Heating	H, Low Temp. Heating	$C_{D,H}(50)$

The test method for measuring the capacity during the cyclic operation shall be the Air Enthalpy Method Indoor side. No additional simultaneously conducted test method need be employed since the cycling of the unit tends to invalidate results of other test methods. Instead, it is assumed that the test setup and instrumentation has been proven accurate by the requirement that the capacities measured by the two test methods simultaneously employed in the steady-state tests must agree within 6 percent.

3.3.2 Air Flow Rate, Instrumentation and Data to be Recorded

The air flow rate during the on-period of the cyclic test shall be the same (agree within $\pm$ 1 percent) as the air flow rate measured during the previously conducted steady-state test. Data to be recorded and instrumentation requirements are identical to the requirements of section 3.2 for steady-state tests where applicable. The instrumentation required to record the air temperature entering and leaving the indoor portion of the unit or the difference between these two temperatures shall have a response time of 2.5 seconds or less in air at an average airspeed encountered in the region of measurement. (Response time is defined as the time required for the instrumentation to attain 63 percent of the final steady-state temperature difference when subjected to a step change in temperature difference of 15°F or more). The electrical energy useage of the indoor fan and the energy usage of the remaining equipment components shall be recorded with watt-hour meters.

If a data acquisition system is not used, data used in monitoring room control shall be recorded at 5 minute intervals, and data used directly in calculations shall be recorded continuously by a multi-pen recorder.

If a data acquisition system is used, data used in monitoring room control shall be recorded at 20 second intervals, and data used in calculations shall be recorded for the first minute after start-up, shut down, or fan delay, at an interval of 5 seconds, for the second and third minutes thereafter at an interval of 10 seconds, and thereafter at an interval of 20 seconds. The integration of temperature-time profiles should be the arithmetic average of the current temperature value and the previous temperature value multiplied by their respective time difference, and these products summed over the entire fan on-period. Note that this discrete data recording scheme is not applicable to electrical power measurements which still should be done with watt-hour meters.

3.3.3 Test Preparation and Performance

The test rooms and equipment installation requirements are identical to the steady-state test procedures described in section 3.2.3. It is suggested, however, that electric resistance heaters be installed in the test room where applicable and be cycled "on" when the unit is cycled "off", in order to reduce the fluctuating heating and cooling loads imposed on the room reconditioning equipment by the cycling of the test equipment. The heater should be approximately the same capacity as the heating capacity of the equipment under test.

During the cyclic tests, the water flow through the unit shall cycle "on" and "off" as the compressor cycles "on" and "off". If the unit does not have an integral solenoid that accomplishes this purpose, such a solenoid must be added to the water supply circuit. It is suggested that a solenoid controlled bypass circuit be installed which will open when the unit cycles "off" to maintain a constant flow through the inlet water temperature control system. It is also suggested that the water system be "total loss" (non-recirculating) to reduce deviation of the inlet water temperature from the value measured during the comparable steady-state test.

These suggestions are made in recognition of the fact that most water temperature control systems will not be able to control within the test tolerance required by this standard if either the water flow rate past the control system temperature sensing element or the supply water temperature vary excessively.

At the conclusion of the appropriate steady-state test, the unit shall be manually cycled "off" and "on", using 24 minute "off" and 6 minute "on" time periods until steadily repeating ambient conditions are achieved for both the indoor and outdoor test chamber, but for not less than 2 complete "off"/"on" cycles. Without a break in the cycling pattern, the unit shall be operated through an additional "off"/"on" cycle, during which the required test data shall be recorded. During the last cycle, which is referred to as the test cycle, the indoor test room ambient conditions shall remain within the tolerance specified in section 3.3.4.

During the cyclic tests, the water flow and the indoor fan shall cycle "on" and "off" as the compressor cycles "on" and "off" except that the indoor fan cycling times (but not the water flow) may be delayed due to controls that are normally installed with the unit. The test installation shall be designed such that there will be no air flow through the unit due to natural or forced convection while the fan is "off".

3.3.4 Test Tolerances

The test condition tolerances and test operating tolerances for the on-period portion of the test cycle are specified in table 4. Variations exceeding any specified test tolerance shall invalidate the test results.

3.3.5 Data Analyses and Results

The reuslts of the cyclic tests shall consist of the Degradation Coefficients, $C_{D,C}(70)$, $C_{D,C}(50)$, $C_{D,H}(80)$ and $C_{D,H}(50)$.

The actual heating or cooling done during the test cycle, Q_{cyc} in kBtu, shall be determined using the following equation:

$$Q_{cyc}(T) = 60 \frac{\dot{V} C_{pa} \ \Gamma}{v_n'(1 + W_n)} \ \frac{1}{1000} \tag{3.1}$$

$$\text{where } \Gamma = \int_{\text{(time, indoor fan on)}}^{\text{(time, indoor fan off)}} [T_{a2}(t) - T_{a1}(t)] \, dt \tag{3.2}$$

and $\dot{V}$ is the flow rate during the on-period as measured during the comparable steady-state test and calculated in accordance with section 7.4 of ASHRAE Standard 37-78.

For heat pump units without an indoor fan which are tested in the cyclic cooling test (test K and test L), the cyclic capacity determined above, $Q_{cyc}(T)$, shall be decreased by an amount equal to the product of 1.250 kBtu/h per 1000 SCFM (7.77 watts per m^3/s), the length of the on-period of the test cycle in hours, and the flow rate of indoor air circulated in units of 1000 SCFM. For cyclic heating tests (test M and test N) of units without an indoor fan the cyclic capacity shall be increased by the same amount.

The total energy usage during the test cycle, E_{cyc}, shall be the sum of the energy usage required for air circulation during the test cycle and the energy used by the remaining equipment components during the test cycle. Units not having a fan as part of the model tested shall set the energy required for air circulation equal to the quantity given by the product of 0.365 kW per 1000 SCFM (7.77 Watts per m^3/s), the length of the on-period of the test cycle in hours and the rate of indoor air circulated in units of 1000 SCFM. The energy required for water circulation shall be set equal to the quantity given by the product of 60 watts per gpm of water flow (951 w/1/s) measured at each test condition and the length of the on-period cycle in hours.

The cyclic coefficient of performance, COP_{cyc}, shall be the ratio of the total heating or cooling done in kBtu (w) during the test cycle to the product of 3.413 kBtuh/Kw (1000 w/kW) and the total energy usage, in kW, during the test cycle when the total heating or cooling done and total energy usage have been adjusted as described above. The cyclic degradation coefficient for heating, $C_{D,H}$, or cooling $C_{D,C}$, shall be calculated as follows at the entering water temperature of the test:

$$C_D = \frac{1 - \frac{COP_{cyc}(T)}{COP_{ss}(T)}}{1 - (HLF \text{ or } CLF)} \tag{3.3}$$

where HLF, CLF =

$$\frac{Q_{cyc}(T)}{\dot{Q}_{ss}(T) \times (t_{on} + t_{off})} \tag{3.4}$$

$\dot{Q}_{ss}(T)$ and $COP_{ss}(T)$ are the steady-state cooling or heating capacity (kBtu/h) and coefficient or performance, respectively, determined from the steady-state performance test (E, F, I or J) conducted at entering temperature, T, and t_{on} and t_{off} are the "on" and "off" times during the test cycle, respectively, in hours. The degradation coefficient, C_D, shall be reported to the nearest .02.

4. CALCULATION OF CSPF, HSPF AND SEASONAL COST OF OPERATION

4.1 DISCUSSION OF THE CALCULATION PROCEDURE FOR SHALLOW GROUND WATER SOURCE APPLICATIONS

The temperature of shallow ground water in a particular location is determined mainly by the mean annual air temperature of the region, and in most locations ground-water temperature at a depth of 30 to 60 feet (9 to 18 meters) usually remains constant within approximately one degree fahrenheit (0.6°C) during the year. Due to these reasons, the entering water temperature in the calculation procedure is assumed to be constant year round at an assigned value for each major climatic region of the country.

The calculation procedures assume that a water source heat pump will be sized to meet either the design heating or cooling load. It is also assumed that resistance heat will be provided to maintain conditions when the building heating load is greater than the heat pump heating capacity.

Since the ratio of the house thermal gain factor (UA, Btu/h°F, W/°C) during the cooling season to the loss factor during the heating season will have a range of from nearly unity to approximately two, it is recommended that for each design load (heating or cooling), a range of seasonal performance factors be calculated corresponding to the range of possible ratios of thermal loss and gain factors between seasons.

Six climatic regions of the U.S. are identified for use in performance factor calculations. Figure 1 is a map of heating load hours (HLH) for the continental United States that defines the six regions for which seasonal bin calculations are to be performed. Figures 2 and 3 are maps of cooling load hours (CLH) and ground water temperatures, respectively, which may be useful in estimating the performance of a heat pump at a specific location. The typical performance calculations for entry in table 9 are to be performed using the cooling load hours and ground water temperatures listed in table 5; figures 2 and 3 are not to be used for this purpose.

The seasonal performance factors are to be calculated by the bin method. A single set of bins is to be used in all regions for the cooling season (see figure 6). Two sets of bins are to be used for heating seasonal calculations, one for climatic regions I through V and a second for region VI (see table 7). The additional set of bins for region VI is provided because of the atypical temperature bin - hour distribution in this region.

4.2 CALCULATING CSPF, HSPF AND SEASONAL OPERATING COST FOR GROUND WATER SOURCE HEAT PUMP APPLICATIONS

Table 5 lists the cooling load hours, heating load hours, heating outdoor design temperatures and design ground water temperatures required in the performance factor calculations for each of the six major U.S. climatic regions. Table 6 lists the fractional hours in each temperature bin for the single set of cooling bins and table 7 lists the fractional hours in each temperature bin for the two sets of heating bins.

In order to evaluate the seasonal unit performance in each region, its full load, steady-state heating and cooling capacities, and coefficients of performance at the water temperature for the region must be known. Laboratory tests have shown that these quantities are approximately a linear function of the mean water temperature, the average of the water temperature in and out of the unit. Since the capacity is linear with mean water temperature, for a unit with a constant water flow the difference between the entering water temperature and mean water temperature will be linear with either of these temperatures. Consequently the capacity and COP will be linear with entering water temperature for a constant water flow application. In such cases, the heating and cooling capacities and coefficients of performance for a specific entering water temperature shall be calculated for the seasonal performance calculations using the following equations:

$$\dot{Q}_{SS,C}(T) = \dot{Q}_{SS,C}(50) - \frac{\dot{Q}_{SS,C}(50) - \dot{Q}_{SS,C}(70)}{20}(T - 50) \quad (4.1)$$

$$COP_{SS,C}(T) = COP_{SS,C}(50) - \frac{COP_{SS,C}(50) - COP_{SS,C}(70)}{20}(T - 50) \quad (4.2)$$

$$\dot{Q}_{SS,H}(T) = \dot{Q}_{SS,H}(50) + \frac{\dot{Q}_{SS,H}(70) - \dot{Q}_{SS,H}(50)}{20}(T - 50) \quad (4.3)$$

$$COP_{SS,H}(T) = COP_{SS,H}(50) + \frac{COP_{SS,H}(70) - COP_{SS,H}(50)}{20}(T - 50) \quad (4.4)$$

$$\text{and } E(T) = \frac{\dot{Q}(T)}{3.413\ COP(T)} \quad (4.5)$$

If the water flow rate is not constant but is a known non-linear function of temperature, the data shall be interpolated or extrapolated as required using the following equations:

$$\dot{Q}_{SS,C}(T_m) = \dot{Q}_{SS,C}(50) - \frac{\dot{Q}_{SS,C}(50) - \dot{Q}_{SS,C}(70)}{T_{m,c}(70) - T_{m,c}(50)}(T_m - T_{m,c}(50)) \quad (4.6)$$

$$COP_{SS,C}(T_m) = COP_{SS,C}(50) - \frac{COP_{SS,C}(50) - COP_{SS,C}(70)}{T_{m,c}(70) - T_{m,c}(50)}(T_m - T_{m,c}(50)) \quad (4.7)$$

$$\dot{Q}_{SS,H}(T_m) = \dot{Q}_{SS,H}(50) - \frac{\dot{Q}_{SS,H}(50) - \dot{Q}_{SS,H}(70)}{T_{m,h}(70) - T_{m,h}(50)}(T_m - T_{m,h}(50)) \quad (4.8)$$

$$COP_{SS,H}(T_m) = COP_{SS,H}(50) - \frac{COP_{SS,H}(50) - COP_{SS,H}(70)}{T_{m,h}(70) - T_{m,h}(50)}(T_m - T_{m,h}(50)) \quad (4.9)$$

where:

$$T_{m,c} = \frac{2\dot{m}\ T_i + a_c}{2\dot{m} - b_c} \quad (4.10)$$

$$T_{m,h} = \frac{2\dot{m}T_i - a_h}{2\dot{m} + b_h} \tag{4.11}$$

and:

$$a_c = \dot{Q}_{SS,C}(70) + b\ T_{m,c}(70) \tag{4.12}$$

$$b_c = \frac{\dot{Q}_{SS,C}(50) - \dot{Q}_{SS,C}(70)}{T_{m,c}(70) - T_{m,c}(50)} \tag{4.13}$$

$$a_h = \dot{Q}_{SS,H}(70) - b\ T_{m,h}(70) \tag{4.14}$$

$$b_h = \frac{\dot{Q}_{SS,H}(70) - \dot{Q}_{SS,H}(50)}{T_{m,h}(70) - T_{m,h}(50)} \tag{4.15}$$

where: T_m = Mean (Average) water temperature through the unit, °F (°C)

T_i = Inlet water temperature, °F (°C)

$\dot{m}$ = Water flow rate, lbs/h

$T_{m,c}(70)$ = T_m at test condition I

$T_{m,c}(50)$ = T_m at test condition J

$T_{m,h}(70)$ = T_m at test condition G

$T_{m,h}(50)$ = T_m at test condition H

If these non-linear flow equations are used, the value of $\dot{m}$ used in making the pump power correction (Section 3.2.5) on Q and COP values shall be that which is used in these non-linear flow equations instead of the actual test values.

When a heat pump is sized to meet either the design cooling or design heating load, the minimum and maximum loads to be met in the other non-design season will depend upon the range of the ratio of the building summer heat gain factor to the building winter heat loss factor. To account for this, it is recommended that the seasonal performance and cost of operation be calculated for a number of design loads between minimum and maximum load values determined as follows:

Maximum design cooling load = $\dot{Q}_{ss,c}$ (4.16)

Minimum design cooling load = The lesser of $\dfrac{60\ \dot{Q}_{ss,H}}{2(65°F - T_{OD})}$ or $\dot{Q}_{ss,c}$ (4.17)
(Regions I, II, III, IV, VI).

Minimum design cooling load = The lesser of $\dfrac{\dot{Q}_{ss,H}}{2.2}$ or $\dot{Q}_{ss,c}$ (4.18)
(Region V)

Maximum design heating load = The greater of $\dfrac{2\ \dot{Q}_{ss,c}}{60}(65°F - T_{OD})$ or $\dot{Q}_{ss,H}$ (4.19)
(Regions I, II, III, IV, VI)

Maximum design heating load = The greater of $2.2\dot{Q}_{ss,c}$ or $\dot{Q}_{ss,H}$ (4.20)
(Region V)

Minimum design heating load = The lesser of $\frac{\dot{Q}_{ss,c}}{60}(65°F - T_{OD})$ or $\dot{Q}_{ss,H}$ (4.21)
(Regions I, II, III, IV, VI)

Minimum design heating load = The lesser of $Q_{ss,c}$ or $Q_{ss,H}$ (4.22)
(Region V)

Where $Q_{ss,c}$ and $Q_{ss,H}$ are the steady-state cooling and heating capacity, respectively, calculated at the regional water temperature given in table 5. T_{OD} is the outdoor design temperature given in table 5.

The above calculated maximum and minimum design loads are to be rounded to the nearest standardized value from table 8.

The seasonal performance factors, CSPF and HSPF, are to be calculated by the bin method at the maximum and minimum design loads and at all standard design loads from table 8 between the maximum and minimum loads.

For each climatic region and for design heating and cooling loads for which bin method calculations are required, the heating seasonal performance factor, HSPF, cooling seasonal performance factor, CSPF, and seasonal operating cost of a heat pump shall be determined using:

$$HSPF = \frac{\Sigma_j \frac{n_j}{N} BL_H(T_j)}{(3.413) \sum_j \frac{n_j}{N} \frac{X_H(T_j)}{PLF_H(X)} \dot{E}_H + \Sigma_j \frac{RH(T_j)}{N}} \quad (4.23)$$

$$\text{Heating Seasonal Operating Cost} = \frac{(HLH)\ (C)\ (\text{Design Heating Requirements})\ (\text{Cost per kWh in \$'s})}{(3.413)(HSPF)}, \quad (4.24)$$

$$CSPF = \frac{\Sigma_j \frac{n_j}{N} BL_c^*(T_j)}{(3.413) \sum_j \frac{n_j}{N} \frac{X_c(T_j)}{PLF_c(X)} \dot{E}_c} \quad (4.25)$$

$$\text{Cooling Seasonal Operating Cost} = \frac{(CLH)\ (C)\ (\text{Design Cooling Requirements})\ (\text{Cost per kWh in \$'s})}{(3.413)(CSPF)}, \quad (4.26)$$

$$\text{Where } BL_c^*(T_j) = \begin{cases} BL_c(T_j), & BL(T_j) \le \dot{Q}_{ss,c} \\ \dot{Q}_{ss,c}, & BL(T_j) > \dot{Q}_{ss,c} \end{cases}$$

$j = 1, 2, 3, ---, n$ corresponds to the j^{th} temperature bin

n = total number of non-zero temperature bins in the climatic region

T_j for heating = $(67 - 5_j)$

T_j for cooling = $(62 + 5_j)$

Σ_j indicates the quantity following the symbol is to be summed over all temperature bins, starting with $j = 1$

$\frac{RH(T_j)}{N}$ = supplementary resistance heat term at temperature T_j required when it is needed to meet the balance of the building heating requirements, (kW)

$\frac{n_j}{N}$ = is the number of hours in the j^{th} temperature bin divided by $N = \Sigma_j n_j$ and is referred to as the "fractional hours in j^{th} temperature bin"

HLH = is the number of heating load hours for the region as given in table 5

CLH = is the number of heating load hours for the region as given in table 5

DHR = is the design heating requirement (kBtu/h)

DCR = is the design cooling requirement (kBtu/h)

3.413 = is a conversion factor which converts kilowatt hours to kBtu

T_{OD} = is the outdoor design heating temperature given in table 5 for each major climatic region, (°F)

C = 0.77 is an experience factor which tends to improve the agreement between calculated and measured building loads

$BL(T_j)$ = building load at temperature T_j, (kBtu/h)

$X(T_j)$ = heat pump load factor

$PLF(X)$ = heat pump part load factor

The quantities $BL(T_j)$, $X(T_j)$, $PLF(X)$ and $\frac{RH(T_j)}{N}$ are defined by the following equations:

$$BL_H(T_j) = \frac{65 - T_j}{65 - T_{OD}} (C) (DHR) \tag{4.27}$$

$$BL_C(T_j) = \frac{T_j - 65}{(1.1)(30)} (DCR) \tag{4.28}$$

$$X(T_j) = \begin{cases} \dfrac{BL(T_j)}{\dot{Q}_{ss}} ; & \dot{Q}_{ss} > BL(T_j) \\ 1 ; & Q_{ss} \leq BL(T_j) \end{cases} \quad \text{for either heating or cooling} \tag{4.29}$$

$$PLF(X) = 1 - C_D(1 - X(T_j)) \quad \text{for either heating or cooling} \tag{4.30}$$

$$\frac{RH(T_j)}{N} = \frac{BL_H(T_j) - \dot{Q}_{ss}}{3.413} \frac{n_j}{N}, \; BL_H(T_j) > \dot{Q}_{ss} \tag{4.31}$$

It should be noted that if the quantity $\dfrac{\dot{Q}(T)}{(3.413)(\dot{E}(T))}$ is less than unity the value of $\dot{E}(T)$ used in the above HSPF calculation should be set equal to $\dot{Q}(T)/3.413$ at this particular temperature T_j. This is done to avoid the possibility that any errors, introduced by the straight line extrapolation of the measured capacities and power inputs could cause the heat pump to have a COP which is less than unity.

A sample calculation showing the use of these equations is given in appendix A as an aid to their application in the calculation of heating and cooling seasonal performance factors and seasonal operating costs for each of the six regions specified in table 5. Once the maximum, intermediate and minimum design heating and cooling requirements, HSPF, CSPF, and corresponding operating cost values have been obtained for each standardized design requirement (see tables 5, 8) in each region, it is recommended that all seasonal operating cost figures be rounded off to the nearest five dollars and the information be arranged in a form similar to table 9 to assist the consumer in selecting the most cost-effective heating appliance for his residence.

REFERENCES

1. Air Conditioning and Refrigeration Institute Standard 325-82, "Ground Water Source Heat Pumps."

2. American Society of Heating, Refrigeration and Air Conditioning Engineers Standard 37-1978, "Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment."

3. G. E. Kelly, W. H. Parken Jr., "Method of Testing, Rating and Estimating the Seasonal Performance of Central Air Conditioners and Heat Pumps Operating in the Cooling Mode," NBSIR 77-1271.

4. Walter H. Parken, George E. Kelly, and David A. Didion, "Method of Testing, Rating and Estimating the Heating Seasonal Performance of Heat Pumps," NBSIR 80-2002.

5. American Society of Heating, Refrigeration, and Air Conditioning Engineers Standard 41.1-1974, "ASHRAE Standard Measurement Guide: Section on Temperature Measurement."

Table 1. Summary of Test Conditions

	Entering Water Temp., °F	Water Flow Rate	Description	Entering Air DB Temp., °F	Entering Air WB Temp., °F	Section Pertaining to Test Requirements	Section Pertaining to Test Procedure
Test E. High Temperature Cooling	70	(a)	Steady-State	80	67	2.2.1, 2.2.2.1	3.2
Test F. Low Temperature Cooling	50	(a)	Steady-State	80	67	2.2.1, 2.2.2.2	3.2
Test G. High Temperature Heating	70	(a)	Steady-State	70	60 maximum	2.2.1, 2.2.2.3	3.2
Test H. Low Temperature Heating	50	(a)	Steady-State	70	60 maximum	2.2.1, 2.2.2.4	3.2
Test I. Dry Coil High Temp. Cooling	70	Same as used in Test E	Steady-State	80	(b)	2.2.1, 2.2.3.1	3.2
Test J. Dry Coil Low Temp. Cooling	50	Same as used in Test F	Steady-State	80	(b)	2.2.1, 2.2.3.2	3.2
Test K. High Temperature Cyclic Cooling	70	Same as used in Test E, I	6-min "on"/24-min "off"	80	(b)	2.2.1, 2.2.3.3	3.3
Test L. Low Temperature Cyclic Cooling	50	Same as used in Test F, J	6-min "on"/24-min "off"	80	(b)	2.2.1, 2.2.3.4	3.3
Test M. High Temperature Cyclic Heating	70	Same as used in Test G	6-min "on"/24-min "off"	70	60 maximum	2.2.1, 2.2.4.1	3.3
Test N. Low Temperature Cyclic Heating	50	Same as used in Test H	6-min "on"/24-min "off"	70	60 maximum	2.2.1, 2.2.4.2	3.3

(a) As specified by the manufacturer.

(b) The entering air wet-bulb temperature for tests I, J, K, and L shall at no time exceed that value of the wet-bulb temperature which results in the production of condensate by the cooling coil at the dry-bulb temperatures existing for the air entering the unit.

Table 2. Cyclic Test Options

1) Perform Tests I, J, K, L, M, N.

2) Perform Tests I and K, assume constant cooling $C_{D,C}$ at value given by Tests I and K perform Test N, assume constant heating $C_{D,H}$ at value given by Tests H and N.

3) Assume constant cooling $C_{D,C} = 0.25$. Perform Test N, assume constant heating $C_{D,H}$ at value given by Test H and N.

4) Omit Tests I, J, K, L, M, N, assume constant $C_D = 0.25$.

Table 3. Steady-State Test Tolerance for Tests E, F, G, H, I, J

	Test Operating Tolerance*	Test Condition Tolerance**
Indoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	2.0	-
Indoor wet-bulb, °F		
Entering	1.0	0.3
Leaving	1.0	-
Outdoor water, °F		
Entering	0.5	0.2
Leaving	0.5	-
Water flow, %	2	-
External resistance to air flow, inches of water	0.05	0.02
Electrical voltage, %	2.0	-

* Test Operating Tolerance is the maximum permissible variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value.

** Test Condition Tolerance is the maximum permissible variation of the average value of the measurement from the standard or desired test condition.

Table 4. Test Tolerances for the On-Period Portion of Cyclic Performance Tests, K, L, M, and N

	Test Operating Tolerance* (Applies after the 1st 30 sec. after compressor start-up	Test Condition Tolerance**
Indoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	-	-
Indoor wet-bulb, °F		
Entering	1.0	-
Leaving	-	-
Outdoor water, °F		
Entering	0.5	0.2
Leaving	-	-
Water flow, %	2	-
External resistance to air flow, inches of water	0.05	0.02
Electrical voltage, %	2.0	-

* Test Operating Tolerance is the maximum permissible variation of any measurement. When expressed as a percentage, the maximum allowable variation is specified percentage of the average value.

** Test Condition Tolerance is the maximum permissible variation of the average value of the measurement from the standard or desired test condition.

Table 5. Design Conditions for Six Standardized Climatic Regions

Region	CLH	HLH	$T_{OD,H}$	T_W
I	2400	750	37	72
II	1800	1250	27	68
III	1200	1750	17	62
IV	800	2250	5	53
V	400	2750	-10	45
VI	200	2750	30	55

Table 6. Fractional Temperature Bin Hours for Cooling Season Calculation

Bin No., j	Bin Temp., 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

Table 7. Fractional Temperature Bin Hours for Heating Season Calculations

		Region I - V	Region VI
Bin No., j	Bin Temp. T_j, °F	Fraction of Total Temperature Bin Hours n_j/N	Fraction of Total Temperature Bin Hours n_j/N
1	62	0.132	0.113
2	57	0.111	0.206
3	52	0.103	0.215
4	47	0.093	0.204
5	42	0.100	0.141
6	37	0.109	0.076
7	32	0.126	0.034
8	27	0.087	0.008
9	22	0.055	0.003
10	17	0.036	0
11	12	0.026	0
12	7	0.013	0
13	2	0.006	0
14	-3	0.002	0
15	-8	0.001	0

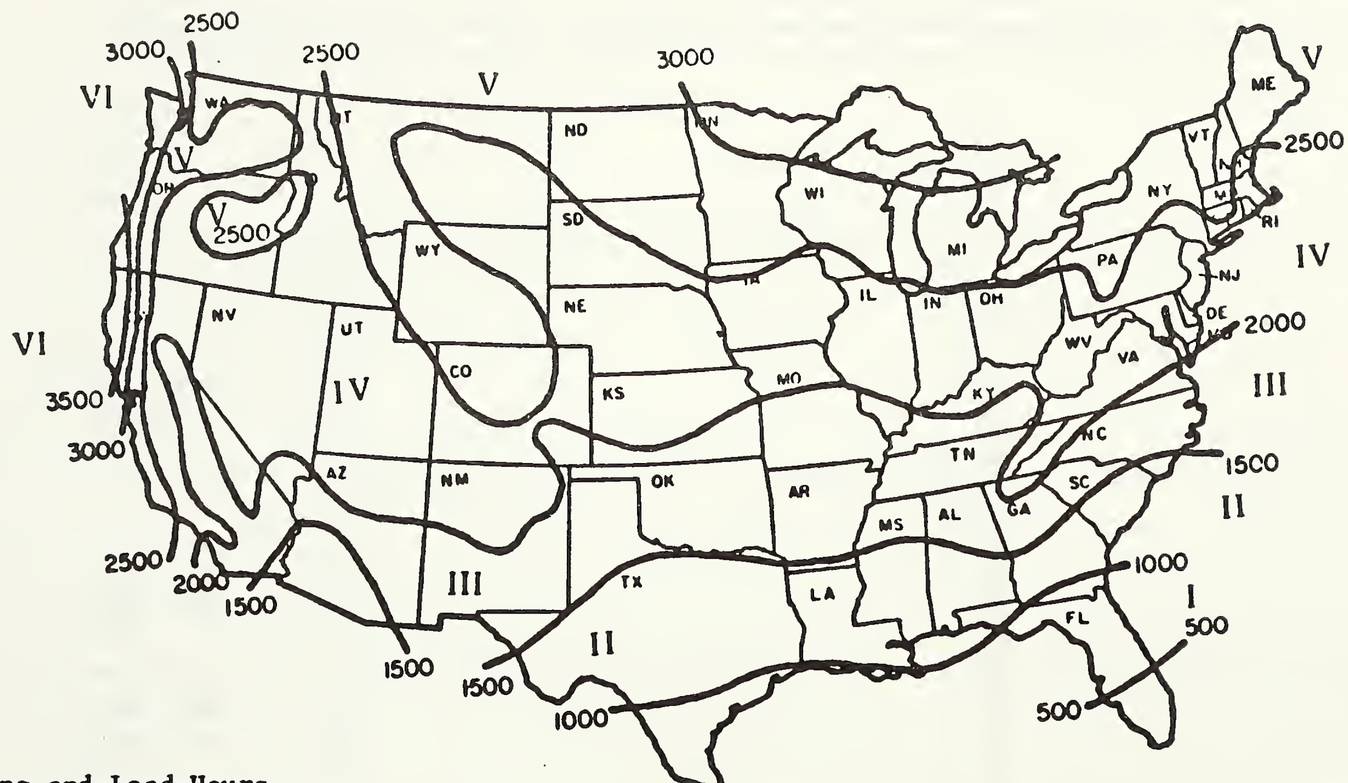
Table 8. Standardized Design Heating and Cooling Requirement Values (kBtu/h)

5	25	50	90
10	30	60	100
15	35	70	110
20	40	80	130

Table 9. Example of Information Which Would Assist A Consumer In Purchasing A Heat Pump

Region	Design Cooling Requirement (kBtu/h)	Operating Cost in Cooling Season Cost of Fuel ($/kWh)						Design Heating Requirement (kBtu/h)	Operating Cost in Heating Season Cost of Fuel ($/kWh)					
		CSPF	.02	.04	.06	.08	.1		CSPF	.02	.04	.06	.08	.1
								25						
I								30						
750 HLH	50							35						
240000 CLH								40						
								50						
								60						
								70						
								35						
II								40						
1250 HLH	50							50						
1800 CLH								60						
								70						
								40						
III	35							50						
1750 HLH	40							60						
1200 CLH	50							70						
								80						
								50						
IV	25							60						
2250 HLH	30							70						
800 CLH	35							80						
	40							90						
	50							100						
	60							110						
								70						
V *	15							80						
2750 HLH	20							90						
400 CLH	25							100						
	30							110						
	35							130						
	40							150						
	50							170						
	60													
								30						
VI								35						
2740 HLH	50							40						
200 CLH	60							50						
								60						

* Manufacturers must state that "This heat pump is not recommended for use in this region" or some equivalent phase for all regions in which the ground water temperature as defined in Table 5 in below the minimum entering water temperature at which the manufacturer recommends application.



Regional Heating and Load Hours

Region	HLH_R
I	750
II	1250
III	1750
IV	2250
V	2750
VI	2750

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

Alaska - 3500 HLH
Hawaii and Territories - 0 HLH

Figure 1. Actual heating load hours (HLH_A) and regional heating load hours (HLH_R) for the United States

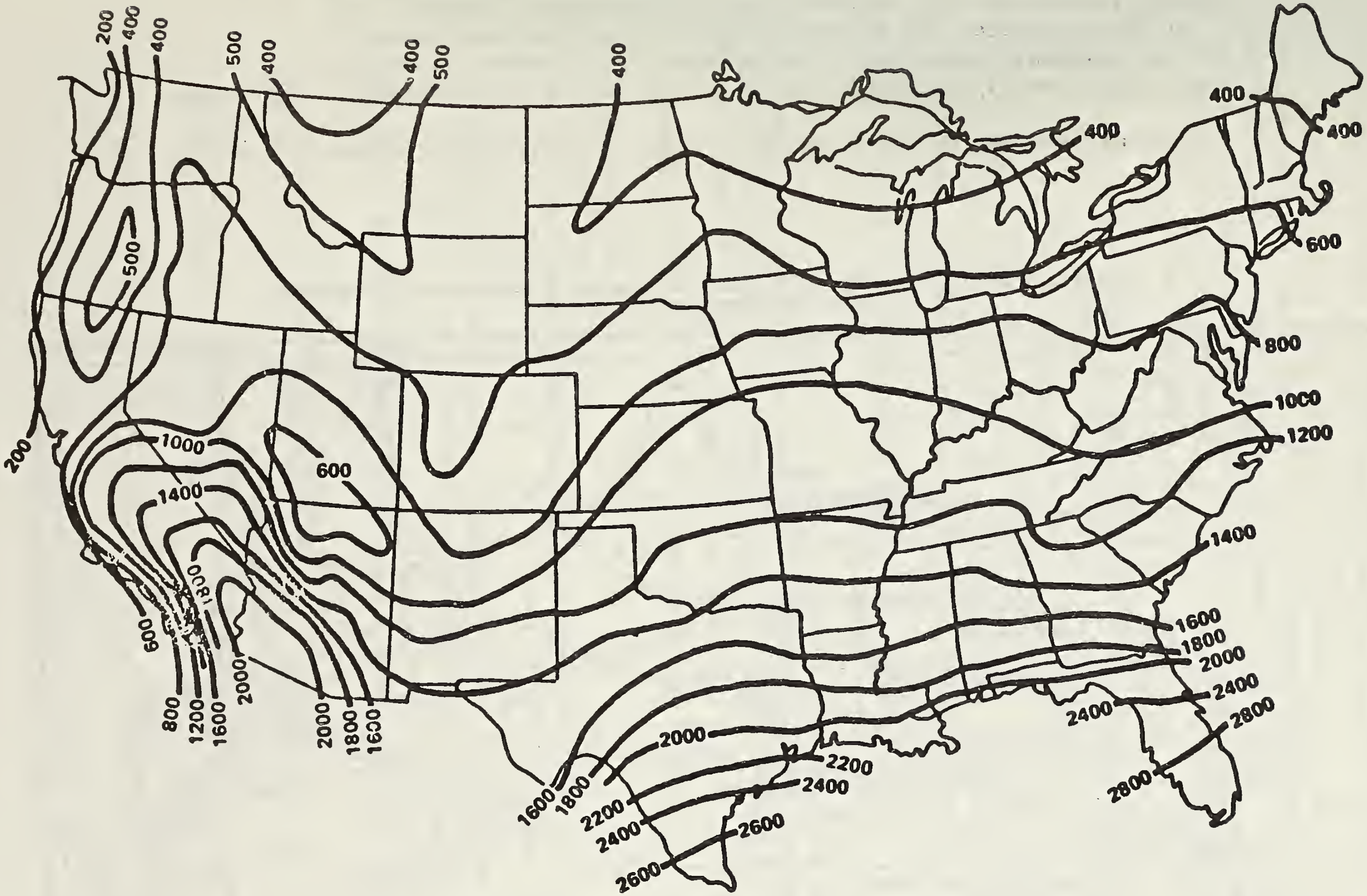
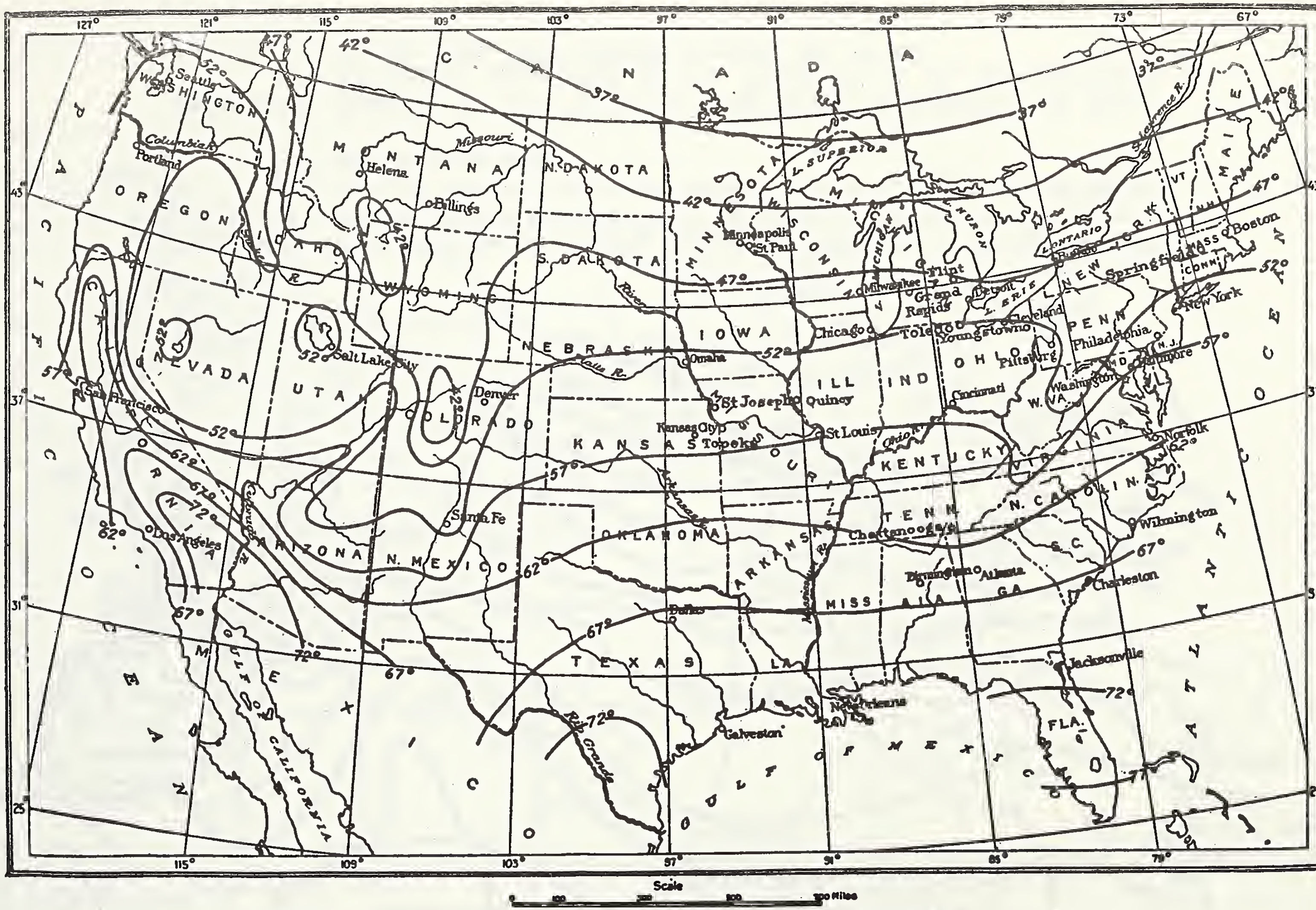


Figure 2. Distribution of cooling load hours throughout the continential United States to be used in estimating the seasonal operating cost of central air conditioners and heat pumps operating in the cooling mode.

Reference: Prepared by York Division, Borg-Warner Corporation and based on results calculated by R.R. McConnell et al, and published in the June, 1976 ASHRAE Journal, pp. 61-65.



APPROXIMATE TEMPERATURE OF WATER FROM NONTHERMAL WELLS AT DEPTHS OF 30 TO 60 FEET

Figure 3. Average temperature of shallow ground water in degrees Fahrenheit

Reference: "Contributions to the Hydrology of the United States, 1923-1924," Nathan C. Grover, Ed., Water Supply Paper 520-F, Temperature of water available for industrial use in the United States, by W.D. Collins; Department of the Interior, U.S. Geological Survey, Washington, D.C., 1925

APPENDIX A

SAMPLE CALCULATIONS

The following sample calculations illustrate the type of experimental data required and the use of the formulas and tables presented to calculate the CSPF, HSPF and seasonal cost of operation for different regions of the United States. The heat pump performance data used to illustrate the calculation procedure are fictional and are not meant to be representative of typical or desired operational characteristics.

Experimental data which would result from the recommended test procedures is presented in table A1 for the fictitious heat pump. It is assumed that the manufacturer has chosen the option of using a value of 0.25 for the cooling degradation factor, $C_{D,C}$ and a measured value of the heating $C_{D,H}$ of 0.16. (It should be noted that the manfacturer has the option of choosing any of several combinations of assigned and measured C_D as given in section 2.2.) If this heat pump does not have sufficient heating capacity to meet the imposed building heating load, at any time, auxiliary electric resistance heaters will supply the balance of the heating requirements.

Tables A2 and A3 are sample worksheets used to evaluate the CSPF, HSPF and seasonal cost of operation. In this example, the CSPF, HSPF and operating cost will be evaluated for a geographic location encompassed by Region IV (see table 5 and figure 1).

The design conditions for region IV given in table 5 are:

Cooling Load Hours, CLH:	800
Heating Load Hours, HLH:	2250
Temp., Outdoor Design, Heating, $T_{OD,H}$:	5°F
Ground-Water-Temperature, T_W	53°F

The heating and cooling capacity and COP at this ground-water-temperature are by interpolation from the data given in table A1.

$$\dot{Q}_{ss,c}(53) = 57.72 - \frac{(57.72 - 47.30)}{(70 - 50)}(53 - 50) = 56.14 \text{ kBtu/h} \tag{4.1}$$

$$COP_{ss,c}(53) = 3.16 - \frac{(3.16 - 2.30)}{(70 - 50)}(53 - 50) = 3.031 \tag{4.2}$$

$$\dot{Q}_{ss,H}(53) = 48.40 + \frac{(75.70 - 48.40)}{(70 - 50)}(53 - 50) = 52.50 \text{ kBtu/h} \tag{4.3}$$

$$COP_{ss,H}(53) = 2.88 - \frac{(3.21 - 2.88)}{(70 - 50)}(53 - 50) = 2.930 \tag{4.4}$$

And:

$$\dot{E}_{ss,c}(53) = \frac{52.50}{(3.413)(3.031)} = 5.427 \text{ kW} \quad (4.5)$$

$$\dot{E}_{ss,H}(53) = \frac{52.50}{(3.413)(3.031)} = 5.250 \text{ kW} \quad (4.6)$$

The maximum design cooling load for which a bin calculation is to be performed is evaluated by equation (4.16) and is equal to the unit cooling capacity = 56.14 kBtu/h.

The minimum design cooling load for which a bin calculation is to be performed is evaluated by equation (4.17) as equal to the lesser of the unit cooling capacity = 56.14 kBtu/h or 60 $Q_{ss,H}/[2(65-T_{OD})]$ = [(60)(52.50)]/[(2)(60)] = 26.25 kBtu/h.

The maximum design heating load for which a bin calculation is to be performed is evaluated by equation (4.18) as equal to the greater of the unit capacity = 52.50 kBtu/h or $[(2)(Q_{ss,c})(65 - T_{OD})]/60$ = [(2)(56.14)(60)]/60 = 112.28 kBtu/h.

The minimum design heating load for which a bin calculation is to be performed is evaluated by equation (4.20) as equal to the lesser of the unit capacity = 52.50 kBtu/h or $[Q_{ss,c})(65 - T_{OD})]/60$ = [(56.4)(60)]/60 = 56.14 kBtu/h.

The above values are rounded to the nearest Standardized Design Requirements from table 8 below. Bin calculations of the CSPF, HSPF and seasonal cost are required at the maximum and minimum and Intermediate Standardized Design Requirements from table 8 as follows:

Maximum standardized design cooling load:	60 kBtu/h
	50 kBtu/h
	40 kBtu/h
	35 kBtu/h
	30 kBtu/h
Minimum standardized design cooling load:	25 kBtu/h
Maximum standardized design heating load:	110 kBtu/h
	110 kBtu/h
	90 kBtu/h
	80 kBtu/h
	70 kBtu/h
	60 kBtu/h
Minimum standardized design heating load:	50 kBtu/h

In this example, the bin calculations will be carried through only for the maximum standardized design cooling and heating loads of 60 kBtu/h and 110 kBtu/h respectively.

In the following sample, heating and cooling season calculations for comparable quantities are shown together to emphasize the differences and similarities between the procedures for the two seasons. To the extent possible, the same column numbers are used for similar columns between the sample cooling seasonal performance factor worksheet, table A2, and the sample heating seasonal performance factor worksheet, table A3. In the sample calculations the subscript C is used to denote quantities being calculated for the cooling season and the subscript H to denote those being calculated for the heating season.

The identification numbers of the equations from the test procedure are indicated in parentheses for each calculation.

Column d: Building Load

The building load is calculated for ecah representative outdoor temperature, T_j, by equation (4.27) for heating and (4.28) for cooling.

Example: $T_j = 42°F$

$$BL_H(42) = \frac{65 - T_j}{60}(C)(DHR)$$

$$= \frac{65 - 42}{60}(.77)(110)$$

$$= 32.468(kBtu/h)$$

$T_j = 82°F$

$$BL_C(82) = \frac{T_j - 65}{(1.1)(30)}(DCR)$$

$$= \frac{82 - 65}{(1.1)(30)}(60)$$

$$= 30.909(kBtu/h)$$

Column e: Load Factor

The load factor, $X(T_j)$, is evaluated by equation (4.29) using the values from columns (d) and the steady-state capacity, $\dot{Q}_{ss,c}$ or $\dot{Q}_{ss,H}$.

Example: $T_j = 42°F$ for heating season

$$X_H(42) = \frac{BL(42)}{\dot{Q}_{ss,H}} = \frac{32.468}{42.50} = .618$$

$T_j = 82°F$ for cooling season

$$X_C(82) = \frac{BL_C(82)}{\dot{Q}_{ss,c}} = \frac{30.909}{56.14} = .551$$

The load factor for both heating and cooling has a maximum value of 1.

Column f: Part Load Factor

The part load factor PLF(X) is evaluated by applying equation (4.30), the data from table A1, and values from column (e).

Example: $T_j = 47°F$

$$PLF_H(X) = 1 - C_D(1 - X(T_j))$$

$$= 1 - 0.16\ [1 - .484] = .917$$

Example: $T_j = 87°F$

$$PLF_c(X) = 1 - 0.25\ [1 - 0.7/13] = .928$$

Column g: Electrical Energy Input

The heat pump electrical energy input for each temperature bin, j, is represented by the terms within the first summation sign in the denominator of equation (4.23), re-written here for convenience:

$$\frac{n_j}{N} \frac{X(T_j)(\dot{E}_{ss})}{PLF(X)}$$

Using table A2, the above expression may be evaluated as follows:

$$(\text{Col. g}) = \frac{(\text{Col. c})(\text{Col. e})(\dot{E}_{ss})}{(\text{Col. f})}$$

Example: $T_j = 12°F$

$$(\text{Col. g})_H = (.026)(1)(5.250) = .1365$$

Example: $T_j = 82°F$

$$(\text{Col. g})_c = \frac{(.161)(.551)(5.427)}{(.888)} = .5422$$

Column h (Table A3): Supplementary Resistance Heat

The supplementary resistance heat, $\frac{RH(T_j)}{N}$, is evaluated by using equation (4.32).

Example: $T_j = 12°F$

$$\frac{RH(12)}{N} = \frac{[BL(12) - \dot{Q}_{ss}]}{3.413} \frac{n_j}{N}$$

$$\frac{74.818 - 52.50}{3.413} (0.026) = .1700$$

Column h (Table A2): Cooling Load

The cooling load at each temperature bin, j, is represented by the term within the summation sign in the numerator of equation (4.25, written as $\frac{n_j}{N} BL_c^*(T_j)$ with a maximum value of $\frac{n_j}{N}(\dot{Q}_{ss,c})$.

Example: $T_j = 87°F$

$BL_c(T_j) = 40.000 < \dot{Q}_{ss,c} = 56.14$

$\frac{n_j}{N} BL_c^* (T_j) = (0.104)(40.000) = 4.1600$

Example: $T_j = 97°F$

$BL_c(T_j) = 58.182 > \dot{Q}_{ss,c} = 56.14$

$\frac{n_j}{N} = BL_c \frac{n_j}{N} \dot{Q}_{ss,c} = (0.018)\ (56.14) = 1.0105$

Column i (Table A3): Heating Load

The heating load at each temperature bin, j, is represented by the term within the summation sign in the numerator of equation 4.23 written as $\frac{n_j}{N} BL_H(T_j)$.

Example: $T_j = 12°F$

(Col. i) = (Col. c)(Col. d) = (.026)(74.818) = 1.94533 [kBtu/h]

CSPF and HSPF

Equation 4.25 is evaluated for the CSPF and equation 4.23 for the HSPF using the summation of the values in columns (g), (h), and (i) of tables A2 and A3. Thus:

$$\text{CSPF} = \frac{\sum_j \text{Col. } h_c}{3.413[\sum_j \text{Col. } g_c]} = \frac{21.3551}{3.413\ [2.3378]} = 2.68$$

And:

$$\text{HSPF} = \frac{\sum_j \text{Col. } i_H}{3.413[\sum_j \text{Col. } g_H + \sum_j \text{Col. } h_H]} = \frac{33.7774}{3.413\ [3.2564 + .7086]} = 2.50$$

The seasonal operating costs are calculated by using equations 4.26 and 4.24. Continuing the example, if the cost of electricity is assumed to be \$.04/kWh then the cooling seasonal operating cost is approximately $210 and the heating seasonal operating cost is approximately $893.

The CSPF, HSPF and operating cost for geographic region IV has been calculated in the above example for maximum design heating and cooling requirements. In order to determine the range of seasonal performance and operating cost values for the unit within the same region IV, the above CSPF, HSPF and cost calculation sequence should be repeated using the minimum and intermediate design requirements as previously discussed. Also, for each value of design requirement, a range of fuel costs should be used to produce a series of seasonal heating and cooling costs.

The complete set of calculated data for the fictitious heat pump operating in region IV is given in table A4. Similar sets of data should be produced for each region where it is appropriate to operate the unit under test.

Table A1. Sample Data Required to Calculate the CSPF and HSPF for a Heat Pump

	Sample Value*
$\dot{Q}_{ss,c}(70)$, kBtu/h	47.30
$\dot{Q}_{ss,c}(50)$, kBtu/h	57.70
$\dot{Q}_{ss,H}(70)$, kBtu/h	70.00
$\dot{Q}_{ss,H}(50)$, kBtu/h	48.40
$COP_{ss,c}(70)$	2.30
$COP_{ss,c}(50)$	3.16
$COP_{ss,h}(70)$	3.21
$COP_{ss,H}(50)$	2.88
Degradation Factor, $C_{D,c}$	0.25
Degradation Factor, $C_{D,H}$	0.16

* Sample values do not represent any particular heat pump unit.

Table A2. Sample Worksheet Used to Evaluate the CSPF and Seasonal Cost of Operation

column (a) Bin Number J	(b) Representative Temperature, T_j (°F)	(c) Fractional Bin Hours $\frac{n_j}{N}$	(d) BLT (T_j), (kBtu/h)	(e) X(T_j)	(f) PLF(X)	(g) Heat Pump Electrical Energy Input. (kW) (c) (e) (Ess)/(f) =(g)	(h) Cooling Load, (kBtu/h) (c) (d) = (h)
1	67	0.214	3.636	.065	.766	.0986	.7782
2	72	0.231	12.727	.227	.807	.3523	2.9400
3	77	0.216	21.818	.389	.847	.5378	3.7127
4	82	0.161	30.909	.551	.888	.5422	4.9764
5	87	0.104	40.000	.713	.928	.4333	4.1600
6	92	0.052	49.091	.874	.969	.2548	2.5527
7	97	0.018	58.182	1	1	.0977	1.0105
8	102	0.004	67.273	1	1	.0217	.2246
						2.3384	21.3551 Totals

Region (I, II, III, IV, V, VI). IV

Cooling Load Hours. 800 (hrs)

Regional Outdoor Temperature. 95 (°F)

/_/ Minimum /_/ Maximum

Design Cooling Requirement. 60 (kBtu/h)

$$\text{CSPF} = \frac{21.3551}{(2.3384)(3.413)} = 2.68$$

$$\text{Seasonal Operating Cost} = \frac{(800)(60)(.04)}{(2.68)(3.413)} = \$210$$

Table A1. Sample Data Required to Calculate the CSPF and HSPF for a Heat Pump

	Sample Value*
$\dot{Q}_{ss,c}(70)$, kBtu/h	47.30
$\dot{Q}_{ss,c}(50)$, kBtu/h	57.70
$\dot{Q}_{ss,H}(70)$, kBtu/h	70.00
$\dot{Q}_{ss,H}(50)$, kBtu/h	48.40
$COP_{ss,c}(70)$	2.30
$COP_{ss,c}(50)$	3.16
$COP_{ss,h}(70)$	3.21
$COP_{ss,H}(50)$	2.88
Degradation Factor, $C_{D,c}$	0.25
Degradation Factor, $C_{D,H}$	0.16

* Sample values do not represent any particular heat pump unit.

Table A2. Sample Worksheet Used to Evaluate the CSPF and Seasonal Cost of Operation

column (a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	
Bin Number J	Representative Temperature, T_j (°F)	Fractional Bin Hours $\frac{n_j}{N}$	BLT (T_j), (kBtu/h)	$X(T_j)$	PLF(X)	Heat Pump Electrical Energy Input. (kW) (c) (e) (Ess)/(f) =(g)	Cooling Load, (kBtu/h) (c) (d) = (h)	
1	67	0.214	3.636	.065	.766	.0986	.7782	
2	72	0.231	12.727	.227	.807	.3523	2.9400	
3	77	0.216	21.818	.389	.847	.5378	3.7127	
4	82	0.161	30.909	.551	.888	.5422	4.9764	
5	87	0.104	40.000	.713	.928	.4333	4.1600	
6	92	0.052	49.091	.874	.969	.2548	2.5527	
7	97	0.018	58.182	1	1	.0977	1.0105	
8	102	0.004	67.273	1	1	.0217	.2246	
						2.3384	21.3551	Totals

Region (I, II, III, IV, V, VI). IV

Cooling Load Hours. 800 (hrs)

Regional Outdoor Temperature. 95 (°F)

/ / Minimum / / Maximum

Design Cooling Requirement. 60 (kBtu/h)

$$CSPF = \frac{21.3551}{(2.3384)(3.413)} = 2.68$$

$$\text{Seasonal Operating Cost} = \frac{(800)(60)(.04)}{(2.68)(3.413)} = \$210$$

Table A3. Sample Worksheet Used to Evaluate the HSPF and Seasonal Cost of Operation

column (a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)
Bin Number j	Representative Temperature, T_j, (°F)	Fractional Bin Hours $\frac{n_j}{N}$	$BL(T_j)$, (kBtu/h)	$X(T_j)$	PLF(X)	Heat Pump Electrical Energy Input, (kW) $(c)(e)(E_{ss})/(f)$	Supplementary Resistance Heat, (kW) $[(c)((d)-Q_{ss})]/3.413=(h)$	Heating Load, (kBtu/h) (c) (d) = (i)
1	62	0.132	4.235	.081	.853	.0655		.5590
2	57	0.111	11.293	.215	.874	.1434		1.2536
3	52	0.103	18.352	.350	.896	.2110		1.8902
4	47	0.093	25.410	.484	.917	.2576		2.3631
5	42	0.100	32.468	.618	.939	.3458		3.2468
6	37	0.109	39.527	.753	.960	.4486		4.3084
7	32	0.126	46.585	.887	.982	.5977		5.8697
8	27	0.087	53.643	1	1	.4568	.0291	4.6670
9	22	0.055	60.702	1	1	.2888	.1322	3.3386
10	17	0.036	57.760	1	1	.1890	.1610	2.4394
11	12	0.026	74.818	1	1	.1365	.1700	1.9453
12	7	0.013	81.877	1	1	.0682	.1119	1.0644
13	2	0.006	88.935	1	1	.0315	.0641	.5336
14	-3	0.002	95.993	1	1	.0105	.0255	.1920
15	-8	0.001	103.052	1	1	.0052	.0148	.1031
						3.2564	.7086	33.7774

Region (I, II, III, IV, V, VI) = VI

$$\text{HSPF} = \frac{33.7774}{(3.413)(3.2561 + 7086)} = 2.50$$

Cooling Load Hours = 2250 (hrs)

Regional Outdoor Design Temperature = 5 (°F)

$$\text{Seasonal Operating Cost} = \frac{(2250)(.77)(110)(.04)}{(2.50)(3.413)} = \$210$$

/_/ Minimum /X/ Maximum /_/ Intermediate

Design Heating Requirement = 110 (kBtu/h)

Table A4. Example of Information Which Would Assist A Consumer In Purchasing A Heat Pump.

Region	Design Cooling Requirement (KBTU/hr)	CSPF	Operating Cost in Cooling Season					Design Heating Requirement (KBTU/hr)	HSPF	Operating Cost in Heating Season Cost of Fuel ($/Kwh)				
			.02	.04	.06	.08	.1			.02	.04	.06	.8	.1
								25						
I								30						
750	50							35						
2400 CLH								40						
								50						
								60						
								70						
								35						
II								40						
1250 HLH	50							50						
1800 CLH								60						
								70						
								40						
III	35							50						
1750 HLH	40							60						
1200 CLH	50							70						
								80						
	25	2.45	50	95	145	190	240	50	2.65	190	385	575	765	960
IV	30	2.48	55	115	170	225	285	60	2.69	225	455	675	905	1130
2250 HLH	35	2.51	65	130	195	260	325	70	2.71	260	525	785	1050	1310
800 CLH	40	2.55	75	145	220	295	370	80	2.70	300	600	900	1200	1500
	50	2.61	90	180	270	360	450	90	2.66	345	685	1030	1375	1715
	60	2.67	105	210	315	420	525	100	2.58	390	790	1180	1575	1970
								110	2.50	445	895	1340	1785	2235
	15							70						
V*	20							80						
2750 HLH	25							090						
400 CLH	30							100						
	35							110						
	40							130						
	50							150						
	60							170						
VI								30						
2750 HLH	50							35						
200 CLH	60							40						
								50						
								60						

* Manufacturers should be given the option of stating that "This heat pump is not recommended for use in this region" or some equivalent phrase.

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5. AUTHOR(S)

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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)

The National Bureau of Standards has made a study of the part-load and seasonal performance of residential ground water source heat pumps operating in both heating and cooling modes. This document outlines methods for testing and rating these units which account for the variation in performance due to part-load operation and change in source water temperature. A calculation procedure is presented which can be used to estimate the seasonal performance and seasonal cost of operation of residential ground water source heat pumps.

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

Central air conditioners; heat pumps; rating procedure; seasonal cost of operation; test method

13. AVAILABILITY

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