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ABSTRACT

Test and rating procedures are presented for electrically-driven residential air-to-air heat pumps operating in the heating mode. The procedures are designed to include the effects of part-load (cyclic) operation, variations in outdoor temperature, and frost formation on the heating performance. Using the test procedure results, a calculation procedure is presented for estimating the heating seasonal performance (HSPF) and cost of operation of residential heat pump units.

Key Words: Central air conditioners; central heating equipment; heat pumps; heating seasonal performance; residential heating; seasonal performance; test method.

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NOMENCLATURE

$BL(T_j)$	building heating load at an outdoor temperature T_j, (kBtu/h)
C	factor equal to 0.77 used to improve agreement between calculated and measured building loads.
C_D	degradation factor used to characterize the degradation in cyclic performance; defined by equation 3.3.
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 of air-water mixture per pound of dry air, (Btu/lb °F).
DHR	design heating requirement, defined by equations (4.2), (4.3) and Table 6, (kBtu/h).
$\dot{E}(T_j)$	total electrical power input to a heat pump at an outdoor temperature T_j, (kW).
$\dot{E}^k(T_j)$	total electrical power input to a heat pump at low speed (single-compressor operation or one cylinder loaded) k=1, or at high speed (two-compressor operation or all cylinders loaded) k=2, at a representative outdoor temperature T_j, (kW).
$\dot{E}_{SS}(T_j)$	total electrical power input to a heat pump operating in a steady-state mode at an outdoor temperature, T_j, (kW).
$\dot{E}_{DEF}$	total electrical energy usage of a heat pump divided by the defrost test time. The defrost test time starts at a defrost termination and ends at the next defrost termination, (kW).
E_{CYC}	total electrical energy usage of a heat pump used in one complete on-off cycle during a cyclic test including any adjustments for fan energy usage, (kWh).
$\frac{E(T_j)}{N}$	the ratio of the total electrical energy usage of a heat pump in temperature bin j to the total number of temperature bin hours in the heating season, (kW).
HLF	heating load factor; defined as the ratio of the cyclic heating done over a complete cyclic test period consisting of an "off" and an "on" period to the steady-state heating that would have been done over the same period of time. See equation 3.4 for basic definition.
HLH	heating load hours, presented in Table 5 and Figure 1 and equation 4.1, (h).

HSPF	heating seasonal performance factor for heat pumps evaluated by equations (4.4) or (4.13).
j	outdoor dry bulb temperature bin number, (e.g.: 1, 2, 3,... n.)
N	total number of hours in the heating season, $N = \sum_j n_j$, (h).
$\frac{n_j}{N}$	fractional number of hours in the j^{th} temperature bin.
n_j	total number of hours in the j^{th} temperature bin, (h).
PLF, PLF(X)	part load factor, defined as the ratio of the cyclic COP to the steady-state COP. See equation 4.9 for basic definition.
$\dot{Q}(T_j)$	heating capacity at an outdoor temperature T_j, (kBtu/h).
$\dot{Q}^k(T_j)$	heating capacity at either low speed (or single compressor operation or all cylinders loaded) k=1, or high speed (two compressor operation or all cylinders loaded) k=2, (kBtu/h).
$\dot{Q}_{SS}(T_j)$	steady-state heating capacity at an outdoor temperature T_j, (kBtu/h).
$\dot{Q}_{DEF}(T_j)$	the ratio of the net heating done during frosting to the total defrost cycle time starting at defrost termination and ending at the next defrost termination, (kBtu/h).
Q_{DEF}	the net heating done over a complete defrost cycle; defined by equation 3.5, (kBtu).
Q_{CYC}	the net cyclic heating during a cyclic test, including any adjustment for fan energy, (kBtu).
$\frac{RH(T_j)}{N}$	the ratio of supplementary resistance heat required at an outdoor temperature T_j to the total number of hours in the heating season, (kW).
T_j	representative outdoor dry-bulb temperature for temperature bin j, (°F). See Table 5.
T_{OD}	97 1/2%-outdoor design temperature for a particular geographical region, (°F).
T_{OFF}	outdoor temperature at which the heat pump compressor is automatically turned off, (°F).
T_{ON}	outdoor temperature at which the heat pump compressor is automatically turned on after having been off, (°F).

$T_{a1}(t)$ — dry-bulb temperature of the air entering the indoor portion of a heat pump, (°F).

$T_{a2}(t)$ — dry-bulb temperature of the air leaving the indoor portion of a heat pump, (°F).

t — time, (h).

t_{ON} — duration of time the compressor is on in one on-off cycle, (h).

v_n' — specific volume of air-water vapor mixture, at the dry-bulb temperature, humidity ratio, and pressure measured at the nozzle, (ft^3/lb).

$\hat{V}$ — indoor air flow rate at the dry-bulb temperature, humidity ratio and pressure existing at the nozzle during the test, (CFM).

W_n — humidity ratio, (lb. of moisture/lb. of dry air)

X, $X(T_j)$, — load factor at an outdoor dry bulb temperature, T_j. See equation 4.8 for basic definition.

Γ — defined by equations 3.2 and 3.6, (°F-hr).

$\delta(T_j)$, $\delta'(T_j)$, $\delta''(T_j)$ — heat pump low temperature compressor cut-off factor. See equations 4.7, 4.19, 4.33, respectively.

$\sum_j$ — indicates summation over all applicable temperature bins.

SUPERSCRIPTS AND SUBSCRIPTS

cyc — cyclic

DEF — frost-defrost

j — temperature bin number, see Table 5.

k — mode of operation for two-speed compressor, two compressor heat pumps, or cylinder unloading heat pumps. Low speed or single compressor operation or one cyclinder loaded is designated as k=1 while high speed or dual compressor operation or all cylinders loaded is designated as k=2.

ss — steady-state.

1. SCOPE AND PURPOSE

The recommended test and rating procedures are intended to be applied to electrically driven single phase air-to-air heat pumps, which have cooling capacities (at test A-type conditions (NBSIR 77-1271)) of 65,000 Btu/h (19.045 kW) or less. For electrically driven heat pumps which supply a heating only function, the procedures apply for those units having capacities of 70,000 Btu/h (20.510 kW) or less (at the 47°F (8.3°C) High Temperature Test conditions (see section 2.2.2)). They do not apply to water source heat pumps or hybrid systems which supplement the heat pump output by burning fossil fuel. For two speed compressor units, units with two compressors, or units incorporating compressor cylinder unloading, the cooling and heating capacities referred to above are the capacities obtained with the heat pump operating on the high compressor speed or with both compressors in operation, or with all cylinders in operation.

The purpose of the recommended test and rating requirements are generally, to provide performance information from which a heat pump unit may be properly sized to a heating load, to provide information from which a consumer may estimate his or her heating 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.

1.1 DEVELOPMENT OF TEST PROCEDURES

The test procedures presented herein have been developed within the framework that the tests should not be unduly burdensome for the manufacturer to perform and should yield accurate and repeatable results. The results are to be applied to a heating seasonal performance calculation and to provide sizing information for residential heating loads. The test procedures should result in a fair and accurate comparison between units having different design features. The recommended test procedures have been adapted from the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Standard 37-69 entitled "Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment" and from a NBS report entitled "Method of Testing, Rating and Estimating the Seasonal Performance of Central Air Conditioners and Heat Pumps Operating in the Cooling Mode", NBSIR 77-1271. This latter report formed the basis for the test procedure for residential central air conditioning equipment published in the Federal Register Vol. 42, No. 227, November 25, 1977.

A steady-state test at 47°F (8.3°C) is required and is the same as the ARI (Air Conditioning and Refrigeration Institute) Standard 240-76 high temperature test. In addition, in order to account for the performance degradation due to the unit cycling on and off, a cycling test is also specified at the same ambient condition. The percent on-time for the test was chosen to be 20% which resulted in a cycling rate of approximately 2 cph, an on-period of 6 minutes and an off-period of 24 minutes. In lieu of performing this cycling test, a manufacturer may use an assigned degradation factor of $C_D = 0.25$.

For two-speed compressor units, units with two compressors, or units incorporating cylinder unloading, a steady-state test at 62°F (16.7°C) is also required in order to evaluate the units' performance at this temperature when cycling on and off at low compressor speed, with a single compressor in operation, or with a cylinder unloaded.

A frost accumulation test is included and conducted at an outdoor temperature of 35°F (1.7°C) and a relative humidity of approximately 80%. These conditions are commonly found throughout the country and are also conditions which can result in large accumulations of frost. At this condition, the unit is to be tested for a complete defrost cycle, starting and ending at a defrost termination. By testing for a complete defrost cycle, instead of a minimum fixed time interval, units containing a particular frost sensing control will not be favored over other systems except as they affect the energy efficiency. For the defrost test, the recommended procedures provide for a tighter control on the outdoor relative humidity during the heating period than required by the ASHRAE standard, since it has been determined from laboratory tests that (depending on the method of defrost initiation) the performance of some units is related to the outdoor relative humidity. Nevertheless, the test should not be significantly more difficult to perform than the defrost test currently required in the ASHRAE Standard, since test room tolerances have been relaxed during defrost and for a short time after the defrosting process is completed.

Because little is known about the effects of part-load operation on a heat pump's performance when the unit is operating at conditions which are likely to result in frost build-up, the recommended calculation procedures, described in section 4, account for the effect of frost formation and defrosting at outdoor temperatures ranging from 17°F (-8.3°C) to below 45°F (7.2°C) and for the effect of part-load operation for building loads less than the heat pump capacity. While the separate handling of these two effects is not rigorously correct, it appears to be a fair and reasonably accurate approach to employ until additional research is carried out to clarify the interactive nature of these effects.

A low temperature test, similar to the ARI Standard 240-76 low temperature test condition is also required. Although frost may form on the outdoor coil the effect on most units' performance at this low temperature condition is not large. Seasonal performance calculations have been made including and excluding the typical reduction in performance at these low temperatures due to frost. Comparison of the results indicates that the accompanying drop in performance which appears in the frost accumulation test at 35°F (1.7°C) and extrapolated up to 45°F (7.2°C) tends to compensate for the exclusion of the performance drop due to frost at the low temperature. As a result, the effect of omitting a full frost accumulation test at this low temperature has only a small effect on the seasonal performance. The recommended procedures therefore require a steady-state test conducted for a duration of 1/2 hour. By not requiring a longer frost accumulation test (e.g., 3 hours), the test procedures also do not unfairly favor some units with demand defrost systems which may not defrost until there is a severe degradation in their performance. Such units would tend to be adversely affected by a variety of weather conditions not accounted for in the proposed test and calculation procedures (such as freezing rain, snow, drifting snow, high winds) which occur in many climatic regions. While the type of defrost control design, and its reliability and performance under the above described weather conditions can significantly affect the seasonal performance of a heat pump, there does not, at the present time, appear to be any easy way of fairly distinguishing between the performance of different defrost systems at low outdoor temperatures. Additional research is needed to clarify the important parameters and to develop defrost test procedures which will encourage manufacturers to install more effective defrost initiation systems.

2. RECOMMENDED TESTING AND RATING REQUIREMENTS

2.1 INTRODUCTION

The heat pump test operating conditions described in Section 2.2.1 through 2.2.5 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 heating seasonal performance with a minimum amount of testing.

Since properly sized heat pumps will cycle on and off to meet small heating loads, a cycling test is included at an outdoor temperature of 47°F (8.3°C). The percent on-time for the test was chosen to be 20% in order to maximize the part-load degradation that would be measured and thereby minimize the associated experimental error. 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 results in a cycling rate of approximately 2 cph at a percent-on-time of 20 percent, which corresponds to cycling period of 30 minutes. The on and off-times for the cycling test were therefore chosen to be 6 minutes and 24 minutes, respectively.

The results of the cycling tests are used in conjunction with High Temperature Test results to determine a degradation factor,

$$C_D = \frac{1 - \dfrac{COP_{cyc}(47)}{COP_{ss}(47)}}{1 - HLF}.$$

This factor is then used in the performance calculations described in Section 4. Laboratory tests have shown that this degradation factor tends to be independent of outdoor temperature and, for the purposes of estimating a unit's seasonal performance, may be considered constant over the range of heating load factors typically encountered in the field.

Manufacturers are given the option of either performing the cyclic test or accepting an assigned degradation factor of $C_D = 0.25$. It was felt that this option was necessary in order to accommodate the small manufacturer which might not have test facilities capable of performing the cycling test within the tolerances required for accuracy.

The steady-state high temperature test at 47°F (8.3°C) and the steady-state low temperature test at 17°F (-8.3°C) provide the data necessary to define the instantaneous steady-state capacity and power curves. For outdoor temperatures equal to or below 17°F (-8.3°C) and outdoor temperatures equal to or greater than 45°F (7.2°C) the steady-state performance of the unit is assumed to be identical to that predicted by these performance curves.

Accumulation of frost on the outdoor coil can result in a significant change in performance over non-frosting conditions. Moreover, frost buildup occurs in a range of outdoor temperatures and relative humidities for which many regions of the country can expect many hours of operation. Hence, frost build-up can have a significant influence on the heating seasonal performance. To account for this, a frost accumulation test is included and conducted at an outdoor temperature of 35°F (1.7°C) and relative humidity of approximately 80%. The degradation in performance is expected to be significant at this temperature and relative humidity and the results of this test and the low temperature test are used to predict the heat pump's performance under frosting conditions for outdoor temperatures greater than 17°F (-8.3°C) and less than 45°F (7.2°C). The outdoor relative humidity of 80% was determined to be an average value commonly occurring in most regions of the country which have a significant number of hours at outdoor temperatures of 35°F (1.7°C). The average was determined by weighting the relative humidity found in the 5°F temperature bins occurring between 27°F (-2.7°C) and 37°F (2.8°C)* by the number of hours in each bin and the number of heat pumps sold. The results were calculated for 50 regions throughout the continental United States and averaged.

2.2 REQUIREMENTS FOR HEAT PUMPS WITH SINGLE-SPEED COMPRESSORS, TWO-SPEED COMPRESSOR, TWO COMPRESSORS AND COMPRESSORS INCORPORATING CYLINDER UNLOADING

Heat pumps with single-speed compressors shall be subjected to the following tests, described in Sections 2.2.2 through 2.2.5, respectively:

High Temperature Test at 47°F (8.3°C),
Cyclic Test,
Frost Accumulation Test, and
Low Temperature Test.

As an alternative to performing the cycling test for single speed units, a degradation factor, C_D, shall be applied whose value shall be 0.25.

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* Facility Design and Construction Engineering Weather Data, AFM 88-8 Chapter 6, TM5-785, NAVFAC P-89., Depts of the Air Force, the Army, and the Navy, Washington, D.C., 1967.

Heat pumps with two-speed compressors or two compressors or cylinder unloading shall be subjected to the following tests, described in Section 2.2.2, 2.2.4 and 2.2.5, respectively:

High Temperature Test at 47°F (8.3°C),
Frost Accumulation Test, and
Low Temperature Test,

with the unit operating on the high compressor speed or with both compressors in operation, or with both cylinders loaded (activated). In addition, the following tests, described in Section 2.2.2 and 2.2.3, shall be conducted:

High Temperature Test at 62°F (16.7°C),
High Temperature Test at 47°F (8.3°C),
Cyclic Test at 47°F (8.3°C),

with the unit operating on the low compressor speed or with the compressor which normally operates at low loads (high outdoor temperatures), or with the cylinder that normally operates at low loads (cylinder unloading).

Two-speed compressor, two-compressor, or cylinder-unloading heat pumps, for which the normal mode of operation requires cycling "on" and "off" of the compressor at high speed, or simultaneous cycling on and off of both compressors or simultaneous cycling on and off of both cylinders, shall also be subjected to the Cyclic Test at 47°F (8.3°C) described in Section 2.2.3, with the unit operating on the high compressor speed or with both compressors in operation, or with both cylinders loaded.

An additional requirement for a heat pump with a two-speed compressor or with two compressors or cylinder unloading applies if the unit is designed to operate on low compressor speed or with a single compressor or a cylinder unloaded at outdoor temperatures below 40°F (4.4°C). If the unit's low capacity performance at and below 40°F (4.4°C) is needed to calculate its seasonal performance as specified in Section 4.3, then it shall be subjected to the following additional tests described in Section 2.2.4 and 2.2.5, respectively:

Frost Accumulation Test, and
Low Temperature Test,

with the unit operating on the low compressor speed or with the compressor which normally operates at low loads or the cylinder which normally operates at low loads (high outdoor temperatures).

In lieu of performing cyclic tests as described in the above requirements, an assigned degradation factor of $C_D^{K=1}$=0.25 shall be applied for low compressor speed (single compressor operation, or one cylinder loaded). Similarly, if a cyclic test is required at high compressor speed (two-compressor operation, or all cylinders loaded) an assigned degradation factor of $C_D^{K=2}$= 0.25 shall be used in lieu of performing the cyclic test.

2.2.1 General Test Condition Requirements for All Tests

All tests shall be conducted at the indoor-side air quantities specified in Sections 5.1.4.3 and 5.1.4.6 and Table 2 of ARI Standard 240-76. Additional sections of this ARI Standard which apply to all tests are:

Voltage and Frequency - Section 5.1.4.2
Outdoor-Side Air Quantity - Section 5.1.4.4
Requirements for Separated Assemblies - Section 5.1.4.5.

For split-system heat pumps, the condenser-evaporator-coil combinations selected for testing and rating shall be the combination likely to have the largest volume of retail sales.

For split-system heat pumps, if the indoor portion of the unit is designed for either horizontal flow or vertical upflow operations, it shall be tested in the vertical-upflow position.

The equipment under test shall be installed according to the requirements of Section 11.2 of ASHRAE Standard 37-69 and Section 5.1.4.5 of ARI Standard 240-76. Briefly, this requires that the unit be installed according to manufacturer's instructions, and with at least 25 feet (7.6 m) of interconnecting tubing (for split systems), no change in external resistance to air flow to account for barometric changes should be made; pressure gages should be installed with short lengths of small-diameter tubing; the unit is to be charged in accordance with the manufacturer's recommended procedures.

In all tests, the specified dry-bulb temperature entering the outdoor portion of the unit also applies to the air temperature surrounding the outdoor portion of the unit. Similarly, models where portions are intended to be installed indoors shall have the air temperature surrounding that portion of the unit the same as the indoor air temperature. Table 1 summarizes the test condition requirements for the different tests.

2.2.2 High Temperature Test Conditions at 47°F (8.3°C) or 62°F (16.7°C)

The steady-state test at 47°F (8.3°C) shall be conducted at an outdoor dry-bulb temperature of 47°F (8.3°C) and an outdoor wet-bulb temperature of 43°F (6.1°C). The steady-state test at 62°F (16.7°C) shall be conducted at an outdoor dry-bulb temperature of 62°F (16.7°C) and an outdoor wet-bulb temperature of 56.5°F (13.6°C). For both tests, the dry-bulb air temperature entering and surrounding the indoor portion of the unit shall be 70°F (21.1°C) and a maximum wet-bulb temperature of 60°F (15.6°C).

Both tests shall be conducted according to the test procedures outlined in Section 3.2 for the high temperature steady-state tests. The duration shall be for a minimum of 1/2 hour. If the unit accumulates frost then the test shall be preceded by a defrost and tested according to the procedures outlined in Section 3.2.

2.2.3 Cyclic Performance Test Conditions

The cyclic performance test at 47°F (8.3°C) shall be performed immediately following the High Temperature Test at 47°F (8.3°C). The high temperature performance test results and the cyclic test results are used together to find the performance degradation resulting from cyclic operation. The dry-bulb temperature and wet-bulb temperature of the air entering the outdoor portion of the unit shall be the same as the high temperature test and the temperature of the air entering and surrounding the indoor portion shall also be the same as the high temperature test (70°F (21.1°C) dry-bulb, 60°F (15.6°C) maximum wet-bulb). The cyclic performance test is conducted according to the test procedures outlined in Section 3.3.

During the cyclic test, 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".

As an alternative to performing the cycling tests, a degradation factor, C_D, shall be applied whose value shall be 0.25. However, if this alternative is chosen the high-temperature steady-state test is still required.

2.2.4 Frost Accumulation Test Conditions

The dry-bulb temperature and dew-point temperature of the air entering the outdoor portion of the unit shall be 35°F (1.7°C) and 30°F (-1.1°C) respectively. The indoor dry-bulb temperature shall be 70°F (21.1°C) and the maximum indoor wet-bulb shall be 60°F (15.6°C).

Briefly, the frost accumulation test procedures require that the unit undergo a defrost prior to the actual test. The test then begins at defrost termination and ends at the next defrost termination. Defrost termination occurs when the controls normally installed within the unit are actuated to cause it to change from defrost operation to normal heating operation. During the test, auxiliary resistance heaters shall not be employed during either the heating or defrost portion of the test.

2.2.5 Low Temperature Test Conditions

The low temperature test shall be conducted at a dry-bulb temperature entering the outdoor portion of the unit of 17°F (-8.3°C) and a wet-bulb temperature of 15.0°F (-9.4°C). The air entering the indoor portion of the unit shall have a dry-bulb temperature of 70°F (21.1°C) and maximum wet-bulb temperature of 60°F (15.6°C). The test shall be conducted according to the procedures outlined in Section 3.5.

3. HEAT PUMP HEATING PERFORMANCE TEST PROCEDURES

3.1 INTRODUCTION

The following steady-state test procedures have been adapted largely from ASHRAE Standard 37-69. The cyclic performance test procedures have been adapted from a report entitled "Method of Testing, Rating and Estimating the Seasonal Performance of Central Air Conditioners and Heat Pumps Operating in the Cooling Mode," NBSIR 77-1271.

The test procedures are categorized as follows:

Steady-State Test Procedure at Outdoor Temperatures of 47°F (8.3°C) or 62°F (16.7°C) - Section 3.2

Cyclic Performance Test - Section 3.3

Frost Accumulation Test - Section 3.4

Low Temperature Test - Section 3.5

The tests are designed to determine the performance of the unit under conditions of typical use, as well as to establish a performance curve from which the heating seasonal performance and cost of operation may be determined. The high temperature test and cyclic performance test are conducted to determine the steady-state and cyclic performance that would typically occur at high outdoor temperatures.

At temperatures below 45°F (7.2°C), units generally experience a degradation in performance due to accumulation of frost on the outdoor coil. The frost accumulation test at 35°F (1.7°C) is designed to account for this effect. The duration of the test is for a complete cycle from defrost termination to defrost termination or for a period of time equal to twelve hours, whichever is less. The maximum test time is made long enough so that virtually all units will be tested for a complete defrost cycle, thereby preventing the test from favoring one type of defrost control over another.

The low temperature test at 17°F (-8.3°C) is a compromise between establishing a "reasonably accurate" value for the performance and the possible great expense needed to establish a "true" value. The test itself is a steady-state test which lasts for a minimum of 1/2 hour. While frost may accumulate and affect the performance over a much longer period, the period of time between defrosts for some units could extend well beyond 12 hours. Other units, however, may have control systems which require them to defrost every 90 minutes. Both of these defrost control strategies will generally have a small effect on the heating seasonal performance when the more accurate determination of performance is obtained by testing from the end of one defrost cycle to the end of the next defrost cycle. Since the effect on the heating seasonal performance of neglecting frost formation at low outdoor tempertures has been calculated to be, in most cases, small and because the effect of requiring a test from the end of one defrost to the end of the next defrost cycle would place an unfair test burden on some units, a short test period has been selected in order to obtain the unit's performance under essentially frost-free conditions.

3.2 STEADY-STATE HIGH TEMPERATURE TEST PROCEDURE AT OUTDOOR TEMPERATURES OF 47°F (8.3°C) OR 62°F (16.7°C)

3.2.1 Available Test Methods

Listed below and in the ASHRAE Standard 37-69 are the recommended test methods for conducting steady-state tests.

i) Air Enthalpy Method - Indoor Side

ii) Air Enthalpy Method - Outdoor Side

iii) Compressor Calibration Method

iv) Volatile Refrigerant Flow Method.

The Air Enthalpy Method - Indoor Side shall be used for all tests. In addition, one other method chosen from the above list shall simultaneously be employed 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 37-69, 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. Three-piece units and units containing the compressor located with the indoor section and separately ventilated shall not use a test method (such as the calorimeter air enthalpy 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 indoor side air enthalpy method are given in the ASHRAE Standard 37-69 under the following Sections, subject to the above condition on compressors located indoors and separately ventilated.

Volatile Refrigerant Flow Method - Sections 5.1 - 5.3 and 5.5

Compressor Calibration Method - Sections 4.1, 4.2.1 (except that only 4 sets of readings are required) 4.2.2-4.2.6, 4.3, 4.4 and 4.6

Outdoor Air Enthalpy Method - Section 3.1 (except 3.1.3e), 3.2 - 3.5, 3.6.1 (except that the preliminary and actual tests shall be a minimum of 30 minutes), 3.6.3, 3.7.3, 3.8.2, 3.8.3.

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 of ASHRAE Standard 37-69. 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 pressures across the nozzle(s), the velocity pressure and static pressure measurements measured at the nozzle throat shall be measured with manometers which will result in errors which are no greater than ± 1% of reading and having a minimum scale division not exceeding 2.0% of the reading. Static pressure and temperature measurements must be taken at the nozzle throat in order to obtain air density. The area(s) of the nozzle(s) shall be determined by measuring their diameter within an error no greater than ± 0.2% in four places approximately 45 degrees apart, around the nozzle in each of two places through the nozzle throat, one at the outlets and the others in the straight section near the radius.

Electrical Measurements. The 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 ± .5 percent of the quantity measured. Likewise, the energy usage of the compressor, outdoor fan and all other equipment components (including controls, crankcase heater, transformers, etc.) shall be measured with a watt-hour meter of the same accuracy.

Indoor-Air Entering and Discharge Temperatures. Measurements of the air temperature entering and leaving the indoor coil or the difference between these two shall be made in accordance with the requirements of ASHRAE Standard 41 Part 1. These temperatures shall be continuously recorded with instrumentation having a total system accuracy within ± .3°F (± .17°C) and a response time of 2.5 seconds or less. Response time is defined as the time required for the instrumentation to obtain 63% 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 and Outdoor-Side Ambient Conditions. The indoor-side dry-bulb temperature and outdoor-side dry-bulb temperature shall be continuously recorded with instrumentation which will result in an error no greater than ± .3°F (± .17°C). In addition, the outdoor wet-bulb temperature or some other suitable measurement from which the test condition wet-bulb temperature may be derived shall be continuously recorded.

Static Pressure Measurements in Ducts. Static pressure measurements in the ducts and across the unit shall be made in accordance with Section 8 of ASHRAE Standard 37-69 using a manometer which will result in an error no greater than ± 0.01 inches of water (± 2.5Pa).

Additional Instrumentation and Data Acquisition. All other data not continuously recorded shall be recorded in 10 minute intervals. Additional requirements for instrumentation and data acquisition are listed in Sections 10.1 and 10.2 and Table II of ASHRAE Standard 37-69. Requirements for the volatile refrigerant flow method are listed in Section 5.2.1, 5.2.2, 5.2.3 of the above standard. For the compressor calibration method the requirements are found in Sections 4.2.2 through 4.2.6 of the ASHRAE Standard. Requirements for instrumentation for the Air Enthalpy Method - Outdoor Side are the same as for the Air Enthalpy Method - Indoor Side.

3.2.3 Test Preparation and Performance

Test chamber requirements are the same as given in Section 11.1 of the ASHRAE Standard 37-69. When the outdoor air enthalpy method is used the outdoor chamber must not interfere with the normal air circulating pattern during the preliminary test. It is necessary to determine and adjust for system resistance when the outdoor air measuring apparatus is attached to the outdoor portion of the unit.

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 intermittant data being recorded at 10-minute intervals. For all units, especially those having controls which periodically cause the unit to operate in a defrost mode, attention should be given to prevent defrost during the steady-state test. Units which have undergone a defrost should operate in the heating mode for at least 10 minutes after defrost termination prior to the start of the test.

The Air Enthalpy Method - Indoor Side shall be used along with one other test method listed in Section 3.2 conducted simultaneously. When the outdoor air enthalpy method is used as a second test, then a preliminary test must be conducted for a minimum of 30 minutes with 4 or more sets of data recorded at 10-minute intervals. All remaining requirements of Section 3.6.1 in the ASHRAE Standard 37-69 shall then apply in conducting the preliminary test for the outdoor air enthalpy method.

For some units, at the ambient condition of the test, frost may accumulate on the outdoor coil. If the supply air temperature or the difference between the supply air temperature and the indoor air entering temperature has decreased by more than 1.5°F (.83°C) at the end of the test, then the unit shall be defrosted and the steady-state test restarted. Only the results of this second steady-state test shall be reported and used in the heating seasonal performance calculation described in Section 4. Alternatively, the unit may first be defrosted and the steady-state test then conducted. Prior to beginning the steady-state test, a unit shall operate in the heating mode for at least 10 minutes after defrost termination to establish equilibrium conditions for the unit and the room reconditioning apparatus. The steady-state test may begin only when the test unit and room conditions are within the test condition tolerances specified in Table 2.

3.2.4 Test Tolerances

All steady-state tests shall be conducted within the tolerances specified in Table 2. Variation greater than those given in the Table shall invalidate the test. The heating capacity results by the indoor air enthalpy method shall agree within 6 percent of the value determined by any other simultaneously conducted capacity test in order for this test to be valid.

3.2.5 Data Analysis and Results

The results of the steady-state tests shall include:

1) Heating capacity, $\dot{Q}_{ss}(47)$ or $\dot{Q}_{ss}(62)$, (kBtu/h)

2) Electrical power input to all components, $\dot{E}_{ss}(47)$ or $\dot{E}_{ss}(62$, (kW)

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

4) Outdoor dry-bulb temperature, outdoor wet-bulb temperature and indoor dry-bulb temperature, (°F)

5) Data to be recorded as specified in Table 11 of ASHRAE Standard 37-69.

Only the capacity results obtained by the indoor air enthalpy method shall be reported and used in the calculation procedure described in Section 4. The formulae to be used in calculating the heating capacity, and the indoor air flowrate are presented in Sections 3.8.1 and 7.4 of ASHRAE Standard 37-69. The coefficient of performance obtained for the High Temperature Test, $COP_{ss}(47)$ or $COP_{ss}(62)$, is the ratio of the heating capacity in kBtu/h to the product of 3.413 and the sum of the power inputs to the indoor fan in kW and the power inputs to the remaining equipment components (including auxiliary power for crankcase heaters, controls, transformers, etc.) in kW.

Units not having an indoor fan as part of the model tested shall add 1.250 kBtu/h per 1000 SCFM (0.78 watts per dm^3/s) of indoor air handled to the measured capacity to obtain the total heating capacity, $\dot{Q}_{ss}(47)$ or $\dot{Q}_{ss}(62)$, and add 0.365 kW per 1000 SCFM (0.78 watts per dm^3/s) of indoor air handled to the measured power to obtain the total power input, $\dot{E}_{ss}(47)$, or $\dot{E}_{ss}(62)$, to the unit.

3.3 CYCLIC PERFORMANCE TEST PROCEDURE

3.3.1 Available Test Methods

The cyclic performance test must be carried out immediately following the steady-state High Temperature Performance Test. This is necessary since the results of both tests must be used to determine the part-load degradation factor, C_D, described in Section 3.3.5. The test method for measuring the capacity during the cyclic operation shall be the indoor air enthalpy method. No additional simultaneously conducted test method need be employed since the cycling of the unit tends to invalidate the results. 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 High Temperature Test, 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 (agreement within $\pm$ 1%) as the air flow rate measured during the previously conducted High Temperature Test. Data to be recorded and instrumentation requirements are identical to the requirements of Section 3.2 for steady-state High Temperature Test, where applicable. The instrumentation required to continuously 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. (Response time is defined as the time required for the instrumentation to attain 63% of the final steady-state temperature difference when subjected to a step change in temperature difference of 15°F or more.) The electrical energy usage of the indoor fan and the energy usage of the remaining equipment components shall be recorded with watt-hour meters.

3.3.3 Test Preparation and Performance

The test rooms and equipment installation requirements are identical to the Steady-State High Temperature Test procedures described in Section 3.2.3. It is suggested, however, that electric resistance heaters be installed in the outdoor chamber and indoor chamber where applicable, in order to reduce the fluctuating heating and cooling loads imposed on the room reconditioning equipment by the cycling of the test equipment. The indoor-side heater should be approximately the same capacity as the heating capacity of the equipment under test. The capacity of the outdoor-side heater should be approximately equal to the product of the total power input and the quantity obtained by subtracting unity from the heat pump's COP at 47°F (8.3°C).

The Cylic Performance Test shall follow the Steady-State High Temperature Performance test, conducted according to the procedures described in Section 3.2. At the conclusion of the High Temperature Test, the unit shall be cycled "off" and "on", using the "off" and "on" time periods specified in Section 2.2.3, until steadily repeating ambient conditions are achieved for both the indoor and outdoor test chambers, but for not less than two 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 and outdoor test room ambient conditions shall remain within the tolerances specified in Table 3.

During the cyclic tests 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 test installation shall be designed such that there will be no airflow through the indoor unit due to natural or forced convection while the indoor fan is "off" by installing dampers located upstream and downstream of the indoor portion of the unit.

If, during the preliminary phase of the steady state test the unit underwent a defrost cycle (either automatic or manually induced) to rid the outdoor coil of any accumulated frost, then prior to starting the process described above of cycling the unit "off" and "on" it should be made to undergo a defrost. After defrost is completed and before starting the cycling process the unit shall be operated continuously in the heating mode for at least 10 minutes to assure that equilibrium conditions have again been established for the unit and the room conditioning apparatus. Cycling of the unit may begin when the test unit and room conditions are within the steady-state test condition tolerances specified in Table 2. For all units, especially those having controls which periodically cause the unit to operate in a defrost mode, attention should be given to prevent defrost after the cycling process has begun.

During the off-period portion of the cyclic heating test at 47°F (8.3°C) the refrigerant switch-over valve shall remain in the heating mode unless the controls normally supplied with the units are designed to reverse it, in which case the controls shall operate the valve.

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 3. Note that the test condition requirements in Sections 2.2.2 and 2.2.3 and the test condition tolerance requirements in Table 2 and 3 require that the average outdoor dry-bulb temperature measured for the steady-state test and the average outdoor dry-bulb temperature for the on-period portion of the cyclic test shall be between 46.5°F (8.06°C) and 47.5°F (8.61°C). Similarly, the average indoor dry-bulb temperatures for the High Temperature Test and the on-period portion of the Cyclic Test shall be between 69.5°F (20.83°C) and 70.5°F (21.39°C). Variations exceeding any specified test tolerance shall invalidate the test results.

3.3.5 Data Analysis and Results

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

$$Q_{cyc}(47) = 60 \frac{\dot{V} C_{pa} \Gamma}{v_n'(1 + W_n)} \frac{1}{1000} \tag{3.1}$$

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

and $\dot{V}$ is the flow rate during the on-period calculated in accordance with Section 7.4 of ASHRAE Standard 37-69.

For heat pump units tested without an indoor fan, the value determined above for $Q_{cyc}(47)$ shall be increased by a quantity of heat equal to the product of 1.250 kBtu/h per 1000 SCFM (0.78 watts per dm^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.

The total energy usage, 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 (including controls, crankcase heaters, transformers, etc.) during the test cycle. Units not having an indoor fan as part of the model tested, shall set the energy required for indoor air circulation equal to the quantity given by the product of 0.365 kW per 1000 SCFM (0.78 watts per dm^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 cyclic coefficient of performance, $COP_{cyc}(47)$ shall be the ratio of the total heating done in kBtu to the product of 3.413 and the total energy usage, in kW, when the total heating done and total energy usage have been adjusted as described above. The cyclic degradation factor shall be calculated as follows:

$$C_D = \frac{1 - \dfrac{COP_{cyc}(47)}{COP_{ss}(47)}}{1 - HLF} \tag{3.3}$$

where

$$HLF = \frac{Q_{cyc}(47)}{\dot{Q}_{ss}(47) \times (t_{on} + t_{off})} \tag{3.4}$$

and $\dot{Q}_{ss}$ (47) and COP_{ss} (47) are the steady-state capacity (kBtu/h) and coefficient of performance respectively, determined from the Steady-State High Temperature Performance Test conducted at an outdoor temperature of 47°F (8.3°C) and t_{on} and t_{off} are the "on" and "off" times, respectively, in hours. The degradation coefficient, C_D, shall be reported to the nearest .02.

3.4 FROST ACCUMULATION TEST PROCEDURE

3.4.1 Available Test Methods

The performance of the unit shall be evaluated using the Air Enthalpy Method-Indoor Side. No other test method need be simultaneously employed, since the non-steady conditions encountered with equipment undergoing defrost tend to invalidate the results. In order to assure the accuracy of the instrumentation, the Frost Accumulation Test shall take place after the High Temperature Test has been performed. The requirement that the capacity, as measured by two simultaneously conducted test methods during the High Temperature Test, shall agree within 6% is considered proof that test setup and instrumentation are functioning properly. During the period of time between conducting the High Temperature Test and the Frost Accumulation Test, modifications shall be limited to those necessary to conduct the tests and which will have no effect on the accuracy and precision of the instrumentation.

3.4.2 Air Flow Rate, Instrumentation and Data to be Recorded

The air flow rate shall be the same (agreement within ± 1%) as the air flow rate measured during the previously conducted High Temperature Test. The instrumentation and data to be recorded shall be identical to the applicable requirements for the High Temperature Test specified in Section 3.3.2.

The temperature of the air leaving the indoor portion of the unit or the temperature difference between the entering and leaving air shall be measured in accordance with the requirements outlined in Section 3.2.2. As mentioned previously, the response time of the above temperature measuring instruments shall be 2.5 seconds or less when subjected to a step-change in temperature difference of 15°F or more.

The indoor-side dry-bulb temperature and outdoor-side dry-bulb temperature shall be continuously recorded with instrumentation having a total system uncertainty within ± 0.3°F (± 0.17°C). In addition, the outdoor dew point temperature shall be continuously measured or be determined from other continuously recording instrumentation with an error which is no greater than ± 0.5°F (0.28°C). All other data shall be recorded at 10-minute intervals during the heating cycle, except as noted below during defrost.

Defrost initiation, termination and complete test cycle time (from defrost termination to defrost termination) shall be recorded. Defrost initiation is defined as the actuation (either automatic or manual) of the controls normally installed with the unit which cause it to alter its normal heating operation in order to eliminate possible accumulations of frost on the outdoor coil. Similarly, defrost termination occurs when the controls normally installed within the unit are actuated to change from defrost operation to normal heating operation. Provisions should be made so that instrumentation is capable of recording the cooling done during defrost as well as the total electrical energy usage during defrost. These data and the continuously recorded data mentioned above need be the only data obtained during defrost.

3.4.3 Test Preparation and Performance

The test rooms and equipment installation requirements are identical to the steady-state High Temperature Test procedures described in Section 3.2.3. The equipment shall be operated according to the manufacturer's recommendation for normal residential use. The defrost controls shall be set at the normal settings which most typify those encountered in region IV of the U.S.A. (see Table 5 and Figure 1).

The test room reconditioning equipment and the unit under test shall be operated for at least 1/2 hour prior to the start of a "preliminary" test period. The preliminary test period and the test period itself are to be conducted within the test tolerances given in Section 3.4.4. In some cases, described below, the preliminary defrost cycle may be manually induced; however, it is important that the normally operating controls govern the defrost termination in all cases.

For units containing defrost controls which are likely to cause defrost at intervals less than one hour when the unit is operating at the required test conditions, the preliminary test period shall start at the termination of a defrost cycle which occurs automatically and shall end at the termination of the next automatically occurring defrost cycle. The test period then begins at the latter defrost termination and ends at the termination of the next automatically occurring defrost cycle.

For units containing defrost controls which are likely to cause defrost at intervals exceeding one hour when operating at the required test condition, the preliminary test period consists of "heating-only" preliminary operation for at least one hour, after which a defrost may be manually or automatically induced. The test period then begins at the termination of this defrost cycle and ends at the termination of the next automatically occurring defrost cycle. If the unit has not undergone a defrost after 12 hours, then the test shall be concluded and the results calculated for this 12-hour period.

For those units which turn the indoor fan off during defrost, the indoor supply duct shall be blocked during all defrost cycles to prevent natural or forced convection through the indoor unit. During defrost, auxiliary resistance heaters which may be normally installed with the unit shall be prevented from operating.

3.4.4 Test Tolerances

Test condition and test operating tolerances for frost accumulation tests are specified in Table 4. Test Operating Tolerances During Heating applies when the heat pump is in the heating mode, except for the first 5 minutes after the termination of a defrost cycle. Test Operating Tolerance During Defrost applies during a defrost cycle and during the first 5 minutes after defrost termination when the heat pump is operating in the heating mode. In determining whether the Test Condition Tolerances are met, only the heating portion of the test period shall be used in calculating the average values.

Variations exceeding the tolerances presented in Table 4 shall invalidate the test.

3.4.5 Data Analysis and Results

The results to be reported for the Frost Accumulation Test are:

1) Net heating capacity, $\dot{Q}_{DEF}(35)$, (kBtu/h)

2) Net electrical power to all components, $\dot{E}_{DEF}(35)$, (kW)

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

4) Outdoor dry-bulb temperature, outdoor dew-point temperature and indoor dry-bulb temperature, (°F)

5) Data to be recorded as specified in Table II of ASHRAE Standard 37-69.

The net heating, $Q_{DEF}(35)$ in kBtu done during the test period shall be obtained using the equation:

$$Q_{DEF}(35) = \frac{60\ \dot{\bar{V}}\ C_{pa}\ \Gamma}{v_n'\ (1 + W_n)}\ \frac{1}{1000} \tag{3.5}$$

$$\text{where} \quad \Gamma = \int_{(\text{time, defrost termination})}^{(\text{time, next defrost termination})} [T_{a2}\ (t) - T_{a1}\ (t)]\ dt \tag{3.6}$$

The flowrate, $\hat{\bar{V}}$, in CFM shall be the average of the air flowrates calculated at four or more time intervals throughout the heating phase of the test using the equations in Section 7.4 of ASHRAE Standard 37-69.

For units tested without indoor fans, the value determined above for $Q_{DEF}(35)$ shall be increased by a quantity of heat equal to the product of 1.250 kBtu/h per 1000 SCFM (0.78 watts per dm^3/s), the length of time in hours during the Frost Accumulation Test that there was indoor air circulating, and the average flow rate of indoor air circulated in units of 1000 SCFM.

The net heating capacity, $\dot{Q}_{DEF}$ (35), is the total net heating done over the test period (including any credit for the indoor fan heat) divided by the total length of the test period, in hours.

The total energy usage, E_{DEF} (35) shall be the sum of the energy usage required for indoor-air circulation during the test period and the energy used by the remaining equipment components (including controls, crankcase heaters, transformers, etc.) during the test period. Units not having an indoor fan as part of the model tested, shall set the energy required for indoor air circulation equal to the quantity given by the product of 0.365 kW per 1000 SCFM (0.78 watts per dm^3/s), the length of time in hours during the Frost Accumulation Test that there was indoor air circulating, and the average flow rate of indoor air circulated in units of 1000 SCFM.

The net electrical power to all components, $\dot{E}(35)_{DEF}$ is the total energy usage for the test period (including the energy to the indoor fan) divided by the total length of the test period, in hours.

3.5 LOW TEMPERATURE TEST PROCEDURE

3.5.1 Available Test Methods

The Air Enthalpy Method-Indoor Side shall be used as the primary test method and the results from only this method used in reporting results. One other test method, chosen from the list in Section 3.2.1, shall be used simultaneously. The capacity results calculated from the two simultaneously conducted tests must agree within 6% in order to constitute a valid test. The requirements and applicability of the test methods are described in Section 3.2.1.

When the air-enthalpy method is employed for the outdoor side tests, it is important to determine whether attachment of an air flow measuring device affects the performance. A preliminary test shall be conducted according to the requirements specified in Section 3.6 of ASHRAE Standard 37-69, except that the test duration shall be 1/2 hour and only four sets of readings are required. In addition, the preliminary test shall start shortly after a defrost termination (either induced manually or occurring automatically) in a manner similar to that described in Section 3.2.3 for the High Temperature Test.

3.5.2 Instrumentation and Data to be Recorded

Where applicable, instrumentation and data to be recorded are identical to the High Temperature Test described in Section 3.2. Intermittently acquired data shall be recorded at 10-minute intervals.

3.5.3 Test Preparation and Performance

Where applicable, the Steady-State High Temperature Test test preparation and performance requirements shall also be used in the Low Temperature Test, except where noted below. The test room and reconditioning equipment shall first be operated in a steady-state manner for at least one half hour at equilibrium and at the specified test conditions. The unit shall then undergo a defrost, either automatic or manually induced. It is important, however, that the unit terminate the defrost sequence by the action of its own defrost controls. At a time no earlier than 10 minutes after defrost termination, the test shall start. The test duration is one half hour. For all units, especially those having controls which periodically cause the unit to operate in a defrost mode, attention should be given to prevent defrost during the one-half hour test period.

3.5.4 Test Tolerances

During the test period for the Low Temperature Test, the operating conditions shall be within the tolerances specified in Table 2.

3.5.5 Data Analysis and Results

Data analysis and results are the same as those described for the Steady-State High Temperature Test in Section 3.2.5. The heating capacity and indoor air flow rate are calculated using the appropriate equations presented in Sections 3.8.1 and 7.4 of ASHRAE Standard 37-69. Units tested without indoor fans shall have 1.250 kBtu/h per 1000 SCFM (0.78 watts per dm^3/s) of indoor air handled added to their measured capacity to obtain their total heating capacity, $\dot{Q}_{ss}(17)$, and shall have 0.365 kW per 1000 SCFM (0.78 watts per dm^3/s) of indoor air handled added to their measured power to obtain their total power input, $\dot{E}_{ss}(17)$. The total power input includes power to the indoor fan and all remaining equipment components (including controls, crankcase heaters, transformers, etc.)

4. CALCULATION OF THE HEATING SEASONAL PERFORMANCE FACTOR (HSPF) AND SEASONAL COST OF OPERATION

4.1 INTRODUCTION

The heating seasonal performance factor, HSPF, and seasonal operating cost of a heat pump are strongly dependent upon the climatic region in which the heat pump operates, the type of heat pump system employed (e.g., single-speed or variable-speed compressor, whether the compressor operates at all temperatures or is turned off and on at certain temperatures, etc.), and the heating requirement of the building relative to the unit's capacity at different outdoor temperatures. Because of these factors, it is recommended that the HSPF and seasonal cost of operation of a heat pump be calculated between the ranges of the maximum and minimum "design heating requirement" in each major climatic region of the country using the temperature bin method. A "design heating requirement" is the heating requirement that must be met by the heat pump system at the 97-1/2 percent outdoor design temperature. <u>In addition, a consumer purchasing a heat pump which provides both heating and cooling should be reminded to select a cooling capacity which will satisfy his home's cooling requirements and maintain a comfortable indoor level of relative humidity.</u>

The heating load hours (HLH) for a specific geographical region are determined by using the following expression:

$$HLH = \frac{24 \times D\text{-}D}{65 - T_{OD}} \qquad (4.1)$$

where D-D and T_{OD} are the number of degree-days and outdoor temperature (°F) respectively for the region. The number of heating load hours was computed for many regions of the country, resulting in the plot presented in Figure 1. Standard values of the number of heating load hours and the outdoor design temperatures were developed and are presented in Table 5.

The general procedure for estimating the seasonal operating cost of heat pumps is to first calculate the heating seasonal performance factor (HSPF) for the heat pump for the desired geographical region. The seasonal heating required is next determined and is equal to the product of the number of heating load hours (HLH) for the region, the design heating requirement (DHR) and a correction factor equal to 0.77. The correction factor, based on experience, improves the agreement between calculated and measured building loads. The seasonal operating cost is then determined by the product of the seasonal heating required and the fuel (electricity) cost divided by the HSPF. This calculation has the advantage of allowing direct comparison of the seasonal operating cost of heat pumps and furnaces, since the latter is calculated in a similar manner.*

4.2 CALCULATING THE HSPF AND SEASONAL OPERATING COST FOR HEAT PUMPS WITH SINGLE-SPEED COMPRESSORS

Table 5 lists six major U.S. climatic regions and their associated heating load hours, outdoor design temperatures, and fractional hours in each temperature bin. Figure 1 is a map of heating load hours (HLH) for the continental United States that may be used to locate these six regions.

The minimum and maximum design heating requirements of a residence in which a heat pump is likely to be installed will depend upon the climatic region and the capacity of the unit. They may be obtained for each of these six climatic regions by using the following equations:

$$\text{minimum design heating requirement} = \begin{cases} \dot{Q}_{SS}(47)\,\dfrac{(65-T_{OD})}{60}, & \text{for regions I, II, III, IV, and VI} \\ \dot{Q}_{SS}(47), & \text{for region V} \end{cases} \tag{4.2}$$

and

$$\text{maximum design heating requirement} = \begin{cases} 2\,\dot{Q}_{SS}(47), \dfrac{(65-T_{OD})}{60}, & \text{for regions I, II, III, and IV} \\ 2.2\dot{Q}_{SS}(47), & \text{for region V} \end{cases} \tag{4.3}$$

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* Recommended Testing and Calculation Procedures for Determining the Seasonal Performance of Residential Central Furnaces and Boilers; Kelly, Chi, Kuklewicz; NBSIR 78-1543, National Bureau of Standards, Oct. 1978.

and rounding the results off to the nearest standardized design heating requirement given in Table 6. In the above equations, T_{OD} is the outdoor design temperature given in Table 5 for each major climatic region and $\dot{Q}_{SS}(47)$ is the heat pump capacity measured during the high temperature performance test at 47°F.

For each climatic region and for all standardized design heating requirements (see Table 6) ranging from the minimum to the maximum design heating requirements for the region, the heating seasonal performance factor, HSPF, and seasonal operating cost of a heat pump shall be determined using:

$$\mathrm{HSPF} = \frac{\sum_j \frac{n_j}{N} \, BL(T_j)}{(3.413)\left[\sum_j \frac{n_j}{N} \frac{X(T_j)}{PLF(X)} \, \delta(T_j) \, \dot{E}(T_j) + \sum_j \frac{RH(T_j)}{N}\right]}, \qquad (4.4)$$

and

$$\text{Seasonal Operating Cost} = \frac{(\mathrm{HLH})(C)(\text{Design Heating Requirement})(\text{Cost per KWH in \$'s})}{(3.413)(\mathrm{HSPF})}, \qquad (4.5)$$

where

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 = 67 - 5j is the representative temperature of the j^{th} bin, (°F)

$\sum_j$ = indicates the quantity following the symbol is to be summed over all temperature bins

$\frac{RH(T_j)}{N}$ = supplementary resistance heat term at temperature T_j required in those cases where the heat pump automatically turns off ($T_j < T_{ON}$) or 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 $N \equiv \sum_j n_j$ 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, (h)

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

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

T_{OD} = is the outdoor design 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)

$\delta(T_j)$ = heat pump low temperature cut-out factor

$X(T_j)$ = heat pump heating load factor

PLF(X) = heat pump part-load factor

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

$$BL(T_j) = \left(\frac{65 - T_j}{65 - T_{OD}}\right)(C)(DHR) \tag{4.6}$$

$$\delta(T_j) = \begin{cases} 0 \;; & T_j \le T_{OFF} \text{ or } \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} < 1 \\[2ex] \frac{1}{2} \;; & T_{OFF} < T_j \le T_{ON} \text{ and } \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} \ge 1 \\[2ex] 1 \;; & T_j > T_{ON} \text{ and } \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} \ge 1 \end{cases} \tag{4.7}$$

$$X(T_j) = \begin{cases} \dfrac{BL(T_j)}{\dot{Q}(T_j)} \;; & \dot{Q}(T_j) > BL(T_j) \\[2ex] 1 \;; & \dot{Q}(T_j) \le BL(T_j) \end{cases} \tag{4.8}$$

$$PLF(X) = 1 - C_D(1 - X(T_j)) \tag{4.9}$$

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* ibid.

$$\frac{RH(T_j)}{N} = \frac{[BL(T_j) - \dot{Q}(T_j)\, X(T_j)\, \delta(T_j)]\frac{n_j}{N}}{3.413} \qquad (4.10)$$

where

T_{OFF} = the outdoor temperature that the compressor is automatically shut off (If no such temperature exists, T_j is always greater than T_{OFF} and T_{ON}) , (°F)

T_{ON} = the outdoor temperature that the compressor is automatically turned on (if applicable) if designed for low temperature automatic shut-off, (°F)

C_D = degradation factor determined as described in section 3.3.5.

In using the above equation to calculate HSPF, the heat pump capacity in kBtu/hr, $\dot{Q}$, and the power in kW, $\dot{E}$, shall be obtained as follows:

$$\dot{Q}(T_j) = \begin{cases} \dot{Q}_{SS}(17) + \dfrac{(\dot{Q}_{SS}(47) - \dot{Q}_{SS}(17))(T_j - 17)}{30}, & T_j \geq 45°F \text{ or } T_j \leq 17°F \\ \dot{Q}_{SS}(17) + \dfrac{(\dot{Q}_{DEF}(35) - \dot{Q}_{SS}(17))(T_j - 17)}{18}, & 17°F < T_j < 45°F \end{cases} \qquad (4.11)$$

$$\dot{E}(T_j) = \begin{cases} \dot{E}_{SS}(17) + \dfrac{(\dot{E}_{SS}(47) - \dot{E}_{SS}(17))(T_j - 17)}{30}, & T_j \geq 45°F \text{ or } T_j \leq 17°F \\ \dot{E}_{SS}(17) + \dfrac{(\dot{E}_{DEF}(35) - \dot{E}_{SS}(17))(T_j - 17)}{18}, & 17°F < T_j < 45°F \end{cases} \qquad (4.12)$$

where $\dot{Q}_{SS}(47)$ and $\dot{E}_{SS}(47)$, $\dot{Q}_{DEF}(35)$, and $\dot{E}_{DEF}(35)$, and $\dot{Q}_{SS}(17)$ and $\dot{E}_{SS}(17)$ are the capacities (in kBtu/hr) and powers (in kW) measured during the high temperature test, the frost accumulation test, and the low temperature test, respectively. For units not incorporating an indoor fan as part of the model, the above measured capacities and power inputs are to be adjusted as described in Sections 3.2.5, 3.3.5, 3.4.5, 3.5.5. The above measured power inputs include indoor fan power, compressor and outdoor fan power to all other remaining equipment power such as controls, crankcase heaters, transformers, etc. It should be noted that if the above definition of $\dot{Q}(T_j)$ and $\dot{E}(T_j)$ result in the quantity $\dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j)}$ less than unity for a temperature T_j, then the value of δ (T_j) used in the above HSPF calculation is equal to zero 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 to low outdoor temperature, could cause the heat pump to have a COP which is less than unity.

Appendix A presents example calculations for the heating seasonal performance factors and seasonal operating costs corresponding to the range between maximum and minimum design heating requirements for one of the six regions. specified in Table 5. 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 7 to assist the consumer in selecting the most cost-effective heating appliance for his residence.

If a single HSPF and a single seasonal operating cost figure is required for labeling purposes, it is recommeded that the HSPF corresponding to Region IV and the standardized design heating requirement nearest the capacity measured during the high temperature performance test, $\dot{Q}_{SS}(47)$, be used and that the operating cost be based upon this HSPF and on a value of HLH equal to 2080. It must be pointed out, however, that the use of a single HSPF and a single operating cost figure, without a breakdown by region or design heating requirement, will mislead the consumer. A heat pump which is the best performing unit in one climatic region will not necessarily be the best performer in another climatic region. In addition, since the performance of a heat pump is so dependent upon the climatic region and the design load, a fair and accurate comparison with electric and fuel-fired furnaces or boilers is impossible without a breakdown similar to the one given in Appendix A.

4.3 CALCULATING THE HSPF AND SEASONAL OPERATING COST FOR HEAT PUMPS WITH TWO-SPEED COMPRESSOR OR TWO COMPRESSORS, OR COMPRESSORS INCORPORATING CYLINDER UNLOADING

The minimum and maximum design heating requirements of a residence in which a heat pump is likely to be installed shall be determined for the six climatic regions listed in Table 5 using the same procedure outlined in Section 4.2 for units with single-speed compressors. The only difference is that $\dot{Q}_{SS}^{(k=2)}(47)$ (which is the capacity measured in the high temperature performance test at 47°F (+8.3°C) with the unit operating at the high compressor speed or with both compressors in operation or with all cylinders in operation) shall be used in place of $\dot{Q}_{SS}(47)$ in equations 4.2 and 4.3.

For each climatic region and for design heating requirements ranging from both the minimum and maximum design heating requirements for each region, the heating seasonal performance factor, HSPF, and seasonal operating cost of a heat pump shall be determined using:

$$\mathrm{HSPF} = \frac{\sum_j \frac{n_j}{N} BL\ (T_j)}{(3.413)\left[\sum_j \frac{E(T_j)}{N} + \sum_j \frac{RH(T_j)}{N}\right]} \qquad (4.13)$$

and

$$\text{Seasonal Operating Cost} = \frac{(\text{HLH})(\text{C})(\text{Design Heating Requirement})(\text{Cost per kWh in \$'s})}{(3.413)(\text{HSPF})} \tag{4.14}$$

where T_j, $\frac{n_j}{N}$, 3.413, $BL(T_j)$, C, $\sum_j$, $\delta(T_j)$ have been previously defined.

The terms $\frac{RH(T_j)}{N}$, $PLF^{k=1}$, $PLF^{k=2}$, $X^{k=1}$, $X^{k=2}$, are consistent with the definition for single-speed heat pumps and are expanded to include high-speed or two-compressor operation or both cylinders in operation (k=2) or low-speed or single-compressor operation or a cylinder unloaded (k=1). These terms along with the term $\frac{E(T_j)}{N}$ (denoting the heat pump electrical energy usage in the j^{th} temperature bin divided by the total number of bin hours) are evaluated according to the four possible cases of heat pump operation denoted below.

CASE I

$BL(T_j) \leq \dot{Q}^{k=1}(T_j)$; and the unit is operating at low compressor speed or with a single compressor or with a cylinder unloaded, i.e., k=1, for which the building heating load, BL(Tj) is less than or equal to the heating capacity, $\dot{Q}^{k=1}(T_j)$.

$$\frac{E(T_j)}{N} = \frac{\dot{E}^{k=1}(T_j)\, X^{k=1}(T_j)\, \delta'(T_j)}{PLF^{k=1}} \frac{n_j}{N} \tag{4.15}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N} BL(T_j)[1 - \delta'(T_j)]}{3.413} \tag{4.16}$$

$$X^{k=1}(T_j) = \frac{BL(T_j)}{\dot{Q}^{k=1}(T_j)} \tag{4.17}$$

$$PLF^{k=1} = 1 - C_D^{k=1}(1 - X_j^{k=1}) \tag{4.18}$$

$$\delta'(T_j) = \begin{cases} 0\ ; & T_j \leq T_{OFF} \\ 1/2; & T_{OFF} < T_j \leq T_{ON} \\ 1\ ; & T_j > T_{ON} \end{cases} \tag{4.19}$$

CASE II

$\dot{Q}^{k=1}(T_j) < BL(T_j) < \dot{Q}^{k=2}(T_j)$; and the unit is alternating between high-speed or two-compressor operation or with both cylinders loaded (k=2) and low-speed or single-compressor operation or a cylinder unloaded (k=1) to satisfy the building heating load at temperature T_j

$$\frac{E(T_j)}{N} = [\dot{E}^{k=1}(T_j)X^{k=1}(T_j) + \dot{E}^{k=2}(T_j)\ X^{k=2}(T_j)]\ \delta'(T_j)\ \frac{n_j}{N} \tag{4.20}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N}\ BL(T_j)[1 - \delta'(T_j)]}{3.413} \tag{4.21}$$

$$X^{k=1}(T_j) = \frac{\dot{Q}^{k=2}(T_j) - BL(T_j)}{\dot{Q}^{k=2}(T_j) - \dot{Q}^{k=1}(T_j)} \tag{4.22}$$

$$X^{k=2}(T_j) = 1 - X^{k=1}(T_j) \tag{4.23}$$

$$\delta'(T_j) = \begin{cases} 0\ ; & T_j \leq T_{OFF} \\ 1/2\ ; & T_{OFF} < T_j \leq T_{ON} \\ 1\ ; & T_j > T_{ON} \end{cases} \tag{4.24}$$

CASE III

$\dot{Q}^{k=1}(T_j) < BL(T_j) < \dot{Q}^{k=2}(T_j)$; and the unit is cycling on and off at high compressor speed or cycling both compressors on and off simultaneously, or both cylinders cycling on and off simultaneously, (k=2) in order to satisfy the building heating load at temperature T_j.

$$\frac{E(T_j)}{N} = \frac{\dot{E}^{k=2}(T_j)\ X^{k=2}(T_j)\ \delta'(T_j)\ \frac{n_j}{N}}{PLF^{k=2}} \tag{4.25}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N}\ BL(T_j)[1 - \delta'(T_j)]}{3.413} \tag{4.26}$$

$$X^{k=2}(T_j) = \frac{BL(T_j)}{\dot{Q}^{k=2}(T_j)} \tag{4.27}$$

$$PLF^{k=2} = 1 - C_D^{k=2} \left(1 - X^{k=2}(T_j)\right) \tag{4.28}$$

$$\delta'(T_j) = \begin{cases} 0 \; ; & T_j \le T_{OFF} \\ 1/2 \; ; & T_{OFF} < T_j \le T_{ON} \\ 1 \; ; & T_j > T_{ON} \end{cases} \tag{4.29}$$

CASE IV

$BL(T_j) \geq \dot{Q}^{k=2}(T_j)$; and the unit operates continuously at high compressor speed or with both compressors in continuous operation or with both cylinders loaded (k=2) in order to satisfy the building heating load at temperature T_j.

$$\frac{E(T_j)}{N} = \dot{E}^{k=2}(T_j)\; X^{k=2}(T_j)\; \delta''(T_j)\; \frac{n_j}{N} \tag{4.30}$$

$$\frac{RH(T_j)}{N} = \frac{\left[BL(T_j) - \dot{Q}^{k=2}(T_j)\; X^{k=2}(T_j)\; \delta''(T_j)\right] \frac{n_j}{N}}{3.413} \tag{4.31}$$

$$X^{k=2}(T_j) = 1.0 \tag{4.32}$$

$$\delta''(T_j) = \begin{cases} 0 \; ; & T_j \le T_{OFF} \text{ or } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}T_j))} < 1 \\ \frac{1}{2} \; ; & T_{OFF} < T_j \le T_{ON} \text{ and } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))} \ge 1 \\ 1 \; ; & T_j > T_{ON} \text{ and } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))} \ge 1 \end{cases} \tag{4.33}$$

In each of the above cases, the heating capacity in kBtu/h, $\dot{Q}^k(T_j)$ and the power input in kW, $\dot{E}^k(T_j)$, corresponding to low speed, single compressor operation, or unloaded cylinder (k=1) and high speed, two compressors in operation, or all cylinders loaded (k=2), shall be calculated (when required) as follows:

$$\dot{Q}^{k=1}(T_j) = \begin{cases} \dot{Q}_{SS}^{k=1}(47) + \dfrac{(\dot{Q}_{SS}^{k=1}(62) - \dot{Q}_{SS}^{k=1}(47))\,(T_j - 47)}{15}; & T_j \geq 40°F \quad (4.34a) \\ \dot{Q}_{SS}^{k=1}(17) + \dfrac{(\dot{Q}_{DEF}^{k=1}(35) - \dot{Q}_{SS}^{k=1}(17))\,(T_j - 17)}{18}; & 17°F \leq T_j < 40°F \quad (4.34b) \\ \dot{Q}_{SS}^{k=1}(17) + \dfrac{(\dot{Q}_{SS}^{k=1}(47) - \dot{Q}_{SS}^{k=1}(17))\,(T_j - 17)}{30}; & T_j < 17°F \quad (4.34c) \end{cases}$$

$$\dot{Q}^{k=2}(T_j) = \begin{cases} \dot{Q}_{SS}^{k=2}(17) + \dfrac{(\dot{Q}_{SS}^{k=2}(47) - \dot{Q}_{SS}^{k=2}(17))\,(T_j - 17)}{30}; & T_j \geq 45°F \text{ or } T_j \leq 17°F \quad (4.34d) \\ \dot{Q}_{SS}^{k=2}(17) + \dfrac{(\dot{Q}_{DEF}^{k=2}(35) - \dot{Q}_{SS}^{k=2}(17))\,(T_j - 17)}{18}; & 17°F < T_j < 45°F \quad (4.34e) \end{cases}$$

$$\dot{E}^{k=1}(T_j) = \begin{cases} \dot{E}_{SS}^{k=1}(47) + \dfrac{(\dot{E}_{SS}^{k=1}(62) - \dot{E}_{SS}^{k=1}(47))\,(T_j - 47)}{15}; & T_j \geq 40°F \quad (4.35a) \\ \dot{E}_{SS}^{k=1}(17) + \dfrac{(\dot{E}_{DEF}^{k=1}(35) - \dot{E}_{SS}^{k=1}(17))\,(T_j - 17)}{18}; & 17°F \leq T_j < 40 \quad (4.35b) \\ \dot{E}_{SS}^{k=1}(17) + \dfrac{(\dot{E}_{SS}^{k=1}(47) - \dot{E}_{SS}^{k=1}(17))\,(T_j - 17)}{30}; & T_j < 17°F \quad (4.35c) \end{cases}$$

$$\dot{E}^{k=2}(T_j) = \begin{cases} \dot{E}_{SS}^{k=2}(17) + \dfrac{(\dot{E}_{SS}^{k=2}(47) - \dot{E}_{SS}^{k=2}(17))\,(T_j - 17)}{30}; & T_j \geq 45°F \text{ or } T_j \leq 17°F \quad (4.35d) \\ \dot{E}_{SS}^{k=2}(17) + \dfrac{(\dot{E}_{DEF}^{k=2}(35) - \dot{E}_{SS}^{k=2}(17))\,(T_j - 17)}{18}; & 17°F < T_j < 45°F \quad (4.35e) \end{cases}$$

where $\dot{Q}_{SS}^{k}(62)$ and $\dot{E}_{SS}^{k}(62)$, $\dot{Q}_{SS}^{k}(47)$ and $\dot{E}_{SS}^{k}(47)$, $\dot{Q}_{DEF}^{k}(35)$ and $\dot{E}_{DEF}^{k}(35)$, and $\dot{Q}_{SS}^{k}(17)$ and $\dot{E}_{SS}^{k}(17)$ are the capacities (in kBtu/hr) and power (in kW) measured during the high temperature tests at 62°F, the high temperature tests at 47°F, the frost accumulation tests, and the low temperature tests, respectively. For units not incorporating an indoor fan as part of the model the above measured capacities and power inputs are to be adjusted as described in Sections 3.2.5, 3.3.5, 3.4.5, 3.5.5. The above measured power inputs include indoor fan power, compressor and outdoor fan power and power to all other remaining equipment components such as controls, crankcase heaters, transformers, etc. It should be noted that if these definitions of $\dot{Q}^k(T_j)$ and $\dot{E}^k(T_j)$ result in the quantity $\dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))}$ being less than unity for a temperature T_j, then the value of $\delta''(T_j)$ used in the above HSPF calculation is set equal to zero at this particular temperature T_j. This avoids the possibility of having a COP for the heat pump at an outdoor temperature T_j which is less than unity because of any errors introduced by the straight line extrapolation of the measured capacities and power inputs to low outdoor temperatures.

In the four cases described above, T_{OFF} and T_{ON} are, respectively, the outdoor temperatures at which compressor operation automatically stops and automatically starts. If no such temperatures exists, then T_j is always greater than T_{ON} and T_{OFF}. The quantity $C_D^{k=2}$ is the part-load degradation factor for the unit cycling at high compressor speed or with both compressors simultaneously cycling or both cylinders simultaneously cycling (if applicable) and $C_D^{k=1}$ is the part-load degradation factor for unit cycling at low compressor speed or with the single compressor or single cylinder that normally operates at low heating loads (high outdoor temperatures).

Appendix B presents an example to aid in calculating the heating seasonal performance factors and seasonal operating costs for a range of design heating requirements from the minimum to the maximum (standardized according to the values given in table 6) for one of the regions specified in Table 5. It is recommended that all seasonal operating cost figures be rounded off to the nearest five dollars and the information be presented in a form similar to Table 7 to assist the consumer in selecting the most cost-effective heating appliance for his residence.

If a single HSPF and a single seasonal operating cost figure is required for labeling purposes, it is recommended that the HSPF corresponding to Region IV and the standardized design heating requirement nearest the capacity, $\dot{Q}_{SS}^{k=2}(47)$, (measured during the high temperature performance test at 47°F with the unit operating at the high compressor speed or with both compressors in operation or both cylinders in operation) be used. The operating cost shall then be based upon this HSPF and a value of HLH equal to 2080. It should be pointed out, however, that the use of a single HSPF and a single operating cost for heat pumps with two speed compressors or two compressor speeds or cylinder unloading compressors has the same draw backs as those discussed in section 4.2 for units with single-speed compressors.

Table 1. Summary of Test Requirements

Test	Outdoor Conditions, (°F)	Description	Section Pertaining to Test Requirements	Section Pertaining to Test Procedures
1. High Temperature	$T_{DB}=47, T_{WB}=43$ or $T_{DB}=62, T_{WB}=56.5$	Steady-state	2.2.1, 2.2.2	3.2
2. Cyclic Option 1	$T_{DB}=47, T_{WB}=43$	6 min-on/24 min-off	2.2.1, 2.2.3	3.3
Option 2	—	Assume $C_D = 0.25$	—	—
3. Frost Accumulation	$T_{DB}=35, T_{DP}=30$	Frost accumulation	2.2.1, 2.2.4	3.4
4. Low Temperature	$T_{DB}=17, T_{WB}=15$	Steady-state	2.2.1, 2.2.5	3.5

Table 2. Steady-State High Temperature and Low Temperature Test Tolerances

	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	---
Leaving	1.0	---
Outdoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	2.0	---
Outdoor wet-bulb, °F		
Entering	1.0	0.3
Leaving	1.0	---
External resistance to air flow, inches of water	.05	.02
Electrical voltage, %	2.0	---

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* 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 3. Test Tolerances for the On-Period Portion of Cyclic Performance Tests

	Test Operating Tolerances* (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 dry-bulb, °F		
Entering	2.0	0.5
Leaving	---	---
Outdoor wet-bulb, °F		
Entering	2.0	1.0
Leaving	---	---
External resistance to air-flow, inches of water	.05	.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 Frost Accumulation Tests

	Testing Operating Tolerance* During Heating	Testing Operating Tolerance* During Defrost	Test Condition Tolerance** (Heating Portion Only)
Indoor dry-bulb, °F			
Entering	2.0	4.0***	0.5
Leaving	---	---	---
Indoor wet-bulb, °F			
Entering	1.0	---	---
Leaving	---	---	---
Outdoor dry-bulb, °F			
Entering	2.0	10.0	1.0
Leaving	---	---	---
Outdoor dew-point, °F			
Entering	1.5	---	0.7
Leaving	---	---	---
External resistance to air-flow, inches of water	.05	---	.02
Electrical voltage, %	2.0	---	---

* Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value. Test Operating Tolerance During Heating applies when the heat pump is in the heating mode, except for the first 5 minutes after termination of a defrost cycle. Test Operating Tolerance During Defrost applies during a defrost cycle and during the first 5 minutes after the termination of a defrost cycle when the heat pump is operating in the heating mode.

** Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition. Test Condition Tolerance applies only when the heat pump is operating in the heating mode.

*** Not applicable during defrost if the indoor fan is off.

Table 5. Major Climatic Regions In The Continental U.S.A.

Region		I	II	III	IV	V	VI
Heating Load Hours, HLH		750	1250	1750	2250	2750	2750*
Outdoor Design Temperature, T_{OD} for the region		37	27	17	5	-10	30
Fractional Hours:							
Bin #	T_j(°F)						
j = 1	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 6. Standardized Design Heating Requirements (kBtu/h)

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

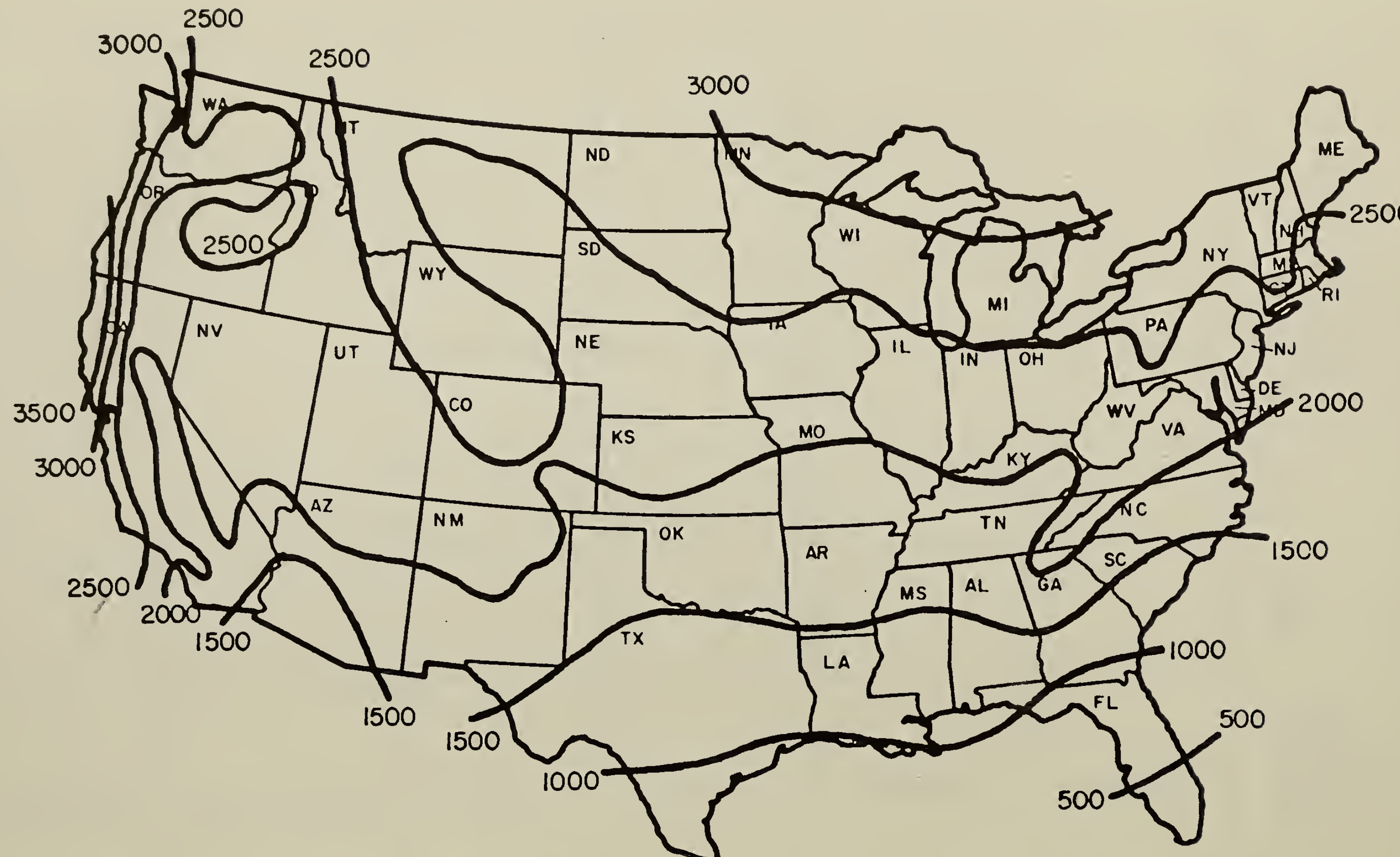
Table 7. Example of Information Which Would Assist A Consumer in Purchasing A Heating Appliance

Cooling Capacity at 95°F = * * * * (if applicable)

Heating Capacity at 47°F = 36,000 Btu/h

Region	Design Heating Requirement (kBtu/h)	HSPF	Cost of Fuel ($/kWh) .02	.04	.06	.08	.1
I (750 HLH)	15 20 25 30 35						
II (1250 HLH)	25 30 35 40 50						
III (1750 HLH)	30 35 40 50 60						
IV* (2250 HLH)	35 40 50 60 70						
V* (2750 HLH)	35 40 50 60 70 80						
VI* (2750 HLH in Pacific Coast Region)	20 25 30 35 40						

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



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.

Figure 1. Heating Load Hours (HLH) for the United States

APPENDIX A

Sample Calculations for a Single-Speed Compressor Heat Pump

The following sample calculation for a single-speed compressor heat pump illustrates the type of experimental data required and the use of the tables in calculating the HSPF and seasonal cost of operation for different regions of the country. The performance data and mode of operation has been contrived to illustrate the calculation procedure and is not meant to be representative of typical or desired operational characteristics.

The experimental data required in the recommended test procedures is presented in Table A1 for the fictitious single-speed compressor heat pump. It is assumed that the manufacturer has chosen to exercise the option of using a value of 0.25 for the degradation factor, C_D. (It should be noted that in lieu of employing an assigned value of C_D the manufacturer has the option of performing the cyclic performance test described in section 3.3.) The heat pump is designed to stop operation at an outdoor temperature, T_{OFF} equal to 5°F. It is assumed that the heat pump will start operating when the temperature increases to a value of T_{ON} equal to 12°F. During the time the heat pump is not operating or not providing sufficient heating capacity to meet the building heating load, supplementary electric resistance heaters will supply the balance of the heating requirements.

Table A2 is a sample worksheet used to evaluate the HSPF and seasonal cost of operation. The HSPF and operating cost is to be evaluated for a geographic location encompassed by Region V (see Table 5 and Figure 1). The heating load hours, HLH, and outdoor design temperature, T_{OD}, obtained from Table 5 are 2750 hours and -10°F, respectively.

The HSPF and seasonal cost of operation are to be calculated for the geographic region for a minimum design heating requirement, DHR_{min}. This minimum value corresponds to the smallest design building load that a particular heat pump is likely to encounter in most practical applications. Equation 4.2 along with the capacity data from Table A1 is used to calculate a DHR_{min} of 50 (kBtu/h) when standardized to the values presented in Table 6.

Columns (a) and (b) of the sample worksheet, Table A2, contain fixed numbers representing the temperature bin number, j, and the corresponding representative outdoor temperature, T_j in °F. The fractional bin hours, $\frac{n_j}{N}$, are obtained for Region V from the data presented in Table 5.

In the following outline, examples are given to illustrate the use of the equations employed to fill in the remaining columns of the worksheet in Table A2 and arriving at a value of the HSPF and seasonal cost of operation.

Column d: Building Load

The building load is calculated for each representative outdoor temperature, T_j, by equation 4.6.

Example: $T_j = 42°F$

$$BL(42) = \frac{65 - T_j}{65 - T_{OD}} (C)(DHR)$$

$$= \frac{65 - 42}{65 + 10} (.77)(50)$$

$$= 11.807 \text{ (kBtu/h)}$$

Columns (e) and (f): Capacity and Power Input

The heat pump capacity $\dot{Q}(T_j)$ and power input, $\dot{E}(T_j)$ are determined from equations (4.11) and (4.12), respectively, and the experimental data presented in Table A1.

Example: $T_j = 52°F$

$$\dot{Q}(52) = \dot{Q}(17) + \frac{[\dot{Q}_{SS}(47) - \dot{Q}_{SS}(17)](52 - 17)}{30}$$

$$= 30.00 + \frac{[48.00 - 30.00](52-17)}{30} = 51.00 \text{ (kBtu/h)}$$

Example: $T_j = 37°F$

$$\dot{Q}(37) = \dot{Q}(17) + \frac{[\dot{Q}_{DEF}(35) - \dot{Q}_{SS}(17)](37 - 17)}{18}$$

$$= 30.00 + \frac{(36.00 - 30.00)(37 - 17)}{18} = 36.67 \text{ (kBtu/h)}$$

Column g: Heating Load Factor

The heating load factor, $X(T_j)$, is evaluated by equation (4.8) using the values from columns (d) and (e):

Example: $T_j = 27°F$

$$X(27) = \frac{BL(27)}{\dot{Q}(27)} = \frac{19.505}{33.33} = .585$$

Column (h): Part Load Factor

The part-load factor PLF(X) is evaluated by applying equation (4.9), the data from Table A1, and values from column (g).

Example: $T_j = 47°F$

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

$$= 1 - 0.25\ [1 - .193] = .798$$

Column (i): Low Temperature Compressor Cut-Out Factor

The low-temperature compressor cut-out factor, $\delta(T_j)$, is evaluated for each representative outdoor temperature T_j by equation 4.7 and the data: $T_{OFF} = 5°F$, $T_{ON} = 12°F$.

Example: $T_j = 17°F$

$$\delta(17) = 1$$

Example: $T_j = 12°F$

$$\delta(12) = \frac{1}{2}$$

Example: $T_j = 2°F$

$$\delta(2) = 0$$

Column (j): Heat Pump 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 (4.4), rewritten here for convenience:

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

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

$$(\text{Col. j}) = \frac{(\text{Col. c})(\text{Col. g})(\text{Col. i})(\text{Col. f})}{(\text{Col. h})}$$

Example: $T_j = 12°F$

$$(\text{Col. j}) = \frac{(.047)(1)(0.5)(4.290)}{1} = .1008\ (\text{kW})$$

Column k: Supplementary Resistance Heat

The supplementary resistance heat, $\frac{RH(T_j)}{N}$, is evaluated by using equation 4.10.

Example: $T_j = 12°F$

$$\frac{RH(12)}{N} = \frac{[BL(12) - \dot{Q}(12)\, X(12)\, \delta(12)]}{3.413} \frac{n_j}{N}$$

$$= \frac{[27.205 - (27.00)(1)(0.5)]\ (.047)}{3.413} = .1887\ (kW)$$

Column ℓ: Heating Load

The heating load at each temperature bin, j, is represented by the term within the summation sign in the numerator of equation 4.4, written as $\frac{n_j}{N} BL(T_j)$.

Example: $T_j = 12°F$

(Col. ℓ) = (Col. c)(Col. d) = (.047)(27.205) = 1.2786 [kBtu/h]

Equation (4.4) is evaluated for the HSPF by summing the values in columns (j), (k), and (ℓ) of Table A2.

Thus:

$$HSPF = \frac{\sum_j \text{Col. } \ell}{3.413\ [\sum_j \text{Col. } j + \sum_j \text{Col. } k]} = \frac{15.219}{3.413\ [1.7519 + 1.0522]} = 1.59$$

The seasonal operating cost is evaluated by using equation 4.5. As indicated also in Table A2 and by assuming a cost of electricity to be $.04/kW, the seasonal operating cost is approximately $780.

The HSPF and operating cost for the geographic Region V has been calculated in the above example for a minimum design heating requirement, DHR_{min}. In order to determine the range of seasonal performance and operating cost values for the unit within the same Region V, the above HSPF and cost calculation sequence should be repeated using the maximum design heating requirement, DHR_{max}. The quantity DHR_{max} is calculated by equation 4.3 and rounded to the nearest value appearing in Table 6. The same calculation sequence is then used for determining the HSPF's and operating costs for all the standardized design heating requirements appearing in Table 6 which fall between these maximum and minimum design heating requirements.

Continuing with the present example, the maximum standard design heating requirement for this unit was found from equation (4.3) and Table 6 to equal 110 (kBtu/h). The HSPF and seasonal operating cost for this value of the

DHR was calculated, using the procedure first described in generating Table A2 and is equal to 1.49 and \$1830., respectively. The values for the HSPF for this example heat pump in Region V are presented in Table A3. The calculation procedure is repeated to generate the HSPF's and seasonal operating costs between the maximum and minimum values. The calculation procedure should also be repeated for each of the climatic regions specified in Table 5.

Table A1. Sample Data Required to Calculate the HSPF for a Single-Speed Heat Pump

$\dot{Q}_{SS}$(47°F), kBtu/h	48.00
$\dot{Q}_{DEF}$(35°F), kBtu/h	36.00
$\dot{Q}_{SS}$(17°F), kBtu/h	30.00
$\dot{E}_{SS}$(47°F), kW	5.023
$\dot{E}_{DEF}$(35°F), kW	4.732
$\dot{E}_{SS}$(17°F), kW	4.395
Degradation Factor, C_D	0.25
T_{ON}, °F	12.
T_{OFF}, °F	5.

Table A2. Sample Worksheet Used to Evaluate the HSPF and the Seasonal Cost of Operation for a Single-Speed Compressor Heat Pump

column	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)
	Bin Number, j	Representative Temperature, T_j; (°F)	Fractional Bin Hours, $\frac{n_j}{N}$	$BL(T_j)$, (kBtu/h)	$\dot{Q}(T_j)$, (kBtu/h)	$\dot{E}(T_j)$, (kW)	$X(T_j)$	PLF(X)	$\delta(T_j)$	Heat Pump Elec. Energy Input, (kW);(c)(g)(i)(f)/(h)	Supplementary Resist. Heat, (kW);(c)((d)-(e)(g)(i))/3.413	Heating Load, (kBtu/h);(c)(d)
	1	62	.106	1.540	57.00	5.337	.027	.757	1	.0202	0	.1632
	2	57	.092	4.106	54.00	5.232	.076	.769	1	.0476	0	.3778
	3	52	.086	6.673	51.00	5.128	.131	.783	1	.0738	0	.5739
	4	47	.076	9.239	48.00	5.023	.193	.798	1	.0923	0	.7022
	5	42	.078	11.807	38.33	4.863	.308	.827	1	.1413	0	.9209
	6	37	.087	14.372	36.67	4.769	.392	.848	1	.1918	0	1.2504
	7	32	.102	16.939	35.00	4.676	.484	.871	1	.2650	0	1.7278
	8	27	.094	19.505	33.33	4.582	.585	.896	1	.2812	0	1.8335
	9	22	.074	22.072	31.67	4.489	.697	.924	1	.2506	0	1.6333
	10	17	.055	24.638	30.00	4.395	.821	.955	1	.2078	0	1.3551
	11	12	.047	27.205	27.00	4.290	1.	1.	1/2	.1008	.1887	1.2786
	12	7	.038	29.771	24.00	4.186	1.	1.	1/2	.0795	.1979	1.1313
	13	2	.029	32.338	21.00	4.081	1.	1.	0	0	.2748	.9378
	14	-3	.018	34.904	18.00	3.976	1.	1.	0	0	.1841	.6283
	15	-8	.010	37.471	15.00	3.872	1.	1.	0	0	.1098	.3747
	16	-13	.005	40.037	12.00	3.767	1.	1.	0	0	.0587	.2002
	17	-18	.002	42.604	9.00	3.662	1.	1.	0	0	.0250	.0852
	18	-23	.001	45.170	6.00	3.558	1.	1.	0	0	.0132	.0452
Totals										1.7519	1.0522	15.219

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

Heating Load Hours = 2750

Regional Outdoor Design Temperature = -10 (°F)

√ Minimum or Maximum

Design Heating Requirement = 50 (kBtu/h)

$$\text{HSPF} = \frac{15.219}{3.413 \times [1.7519 + 1.0522]} = 1.59$$

$$\text{Seasonal Operating Cost} = \frac{(2750)(.77)(50)(.04)}{(1.59)(3.413)}$$

$$= \$780.$$

Table A3. HSPF Values for the Example Single-Speed Compressor Heat Pump

Capacity at 47°F = 48(kBtu/h)			
REGION	DHR (kBtu/h)	HSPF	SEASONAL OPERATING COST ($)* (based on $.04/kWh)
V	110	1.49	1830.
HLH = 2750	100	1.52	1635.
	90	1.55	1440.
	80	1.58	1225.
	70	1.60	1085.
	60	1.60	930.
	50	1.59	780.

* Operating costs in this example have been rounded off to the nearest $5.

Appendix B

Sample Calculations for a Two-Speed Compressor Heat Pump

The following sample calculation illustrates the type of experimental data required and the use of the tables provided in order to calculate the HSPF and seasonal cost of operation for a two-speed heat pump installed in Region IV.

The experimental data required to calculate the HSPF for a fictitious two-speed heat pump is presented in Table B1. The data and mode of operation are for illustrative purposes only and are not meant to represent typical or desired performance characteristics. It is assumed that capacity, $\dot{Q}^k(T_j)$ and power, $\dot{E}^k(T_j)$ at the high speed (k=2) and low speed (k=1) were determined from the previously described test procedures at outdoor temperatures of 17°F (-8.3°C), 35°F (1.7°C), and 47°F (8.3°C). In addition, a steady-state test was conducted at the low (k=1) compressor speed at an outdoor temperature of 62°F (16.7°C). It is further assumed that the unit in this example is designed to cycle on and off only at the low compressor speed (k=1), and that a cyclic test was performed at an outdoor temperature of 47°F with the unit operating at low speed. (In lieu of conducting the cycling test, a degradation factor of $C_D^{k=1}$ equal to 0.25 could have been used.) In this example, the unit shall be designed to operate at outdoor temperature below -23°F so that the compressor cut-out temperature T_{OFF}, and compressor cut-in temperature T_{ON} are not applicable.

Figure B1 is a graph of the high speed (k=2) and low speed (k=1) capacity of the two-speed unit versus outdoor temperature. The curves were generated from the example data in Table B1 and from equations 4.34a through 4.34e. Also shown in this figure is a hypothetical building load curve, which will be discussed later. The purpose of the Figure is to illustrate, for this particular unit and building load configuration, the various heat pump operating regions that occur for each temperature bin. Case I operation occurs at relatively high outdoor temperatures where the unit cycles on and off at low compressor speed (k=1) to meet the small building heating loads. At lower temperatures, the building load is greater than the low-speed capacity but less than the high-speed (k=2) capacity. The unit is designed to run continuously by cycling between the low and high compressor speeds to satisfy the building load. This mode of operation is designated by the region labeled Case II.

Case IV operation occurs at still lower outdoor temperatures where the building load exceeds the high speed (k=2) capacity and supplementary electric resistance heat is required to supply the balance of the heating requirements.

Table B2 and the following outline are used to evaluate the HSPF and seasonal cost of operation for this example two-speed unit.

From Table 5 the number of heating load hours, HLH, is 2250 and the outdoor design temperature, T_{OD}, is 5°F for Region IV. The HSPF shall be calculated for a minimum design heating requirement, DHR_{min}. From equation 4.2 and the high-speed capacity test results for $\dot{Q}_{SS}^{k=2}(47)$, DHR_{min} is calculated to be 70 kBtu/h , when rounded to the nearest standardized value appearing in Table 6.

Column a and b: Temperature Bin Number, j, and Representative Outdoor Temperature, T_j

The values are fixed for all calculations.

Column c: Fractional Bin Hours

The fractional number of hours in each temperature bin is determined for Region IV from Table 5.

Column (d): Building Load

$BL(T_j)$ is determined for each representative outdoor temperature from equation 4.6.

Example: $T_j = 57°F$

$$BL(57) = \frac{65 - 57}{65 - 5} (.77)(70)$$

$$= 7.187 \text{ (kBtu/h)}$$

Columns (e), (f), (g), (h): High Speed (k=2) and Low Speed (k=1) Capacity, $\dot{Q}^k(T_j)$, and Power Input, $\dot{E}^k(T_j)$

The low speed capacity and power input, $\dot{Q}^{k=1}(T_j)$, $\dot{E}^{k=1}(T_j)$, (columns (e) and (g)) for each representative outdoor temperature T_j are determined from equations 4.34a, b, c, and 4.35a, b, c, and the experimental data presented in Table B1.

Example: $T_j = 57°F$

$$\dot{Q}^{k=1}(57) = \dot{Q}_{SS}^{k=1}(47) + \frac{[\dot{Q}_{SS}^{k=1}(62) - \dot{Q}_{SS}^{k=1}(47)]}{15} (57 - 47)$$

$$= 30.00 + \frac{[42.00 - 30.00]}{15} (10)$$

$$= 38.00 \text{ (kBtu/h)}$$

Example: $T_j = 32°F$

$$\dot{Q}^{k=1}(32) = \dot{Q}_{SS}^{k=1}(17) + \frac{[\dot{Q}_{DEF}^{k=1}(35) - \dot{Q}_{SS}^{k=1}(17)]}{18}(32 - 17)$$

$$= 17.00 + \frac{[22.00 - 17.00]}{18}(15)$$

$$= 21.17 \text{ (kBtu/h)}$$

Example: $T_j = 12°F$

$$\dot{Q}^{k=1}(12) = \dot{Q}_{SS}^{k=1}(17) + \frac{[\dot{Q}_{SS}^{k=1}(47) - \dot{Q}^{k=1}(17)]}{30}(12 - 17)$$

$$= 17.00 + \frac{[30.00 - 17.00]}{30}(-5)$$

$$= 14.83 \text{ (kBtu/h)}$$

The high speed capacity, $\dot{Q}^{k=2}(T_j)$ and power input $\dot{E}^{k=2}(T_j)$ (columns (f) and (h), respectively) are determined for each respective outdoor temperature, T_j by equations 4.34 d, e and 4.35 d, e.

Example: $T_j = 57°F$

$$\dot{Q}^{k=2}(57) = \dot{Q}_{SS}^{k=2}(17) + \frac{[\dot{Q}_{SS}^{k=2}(47) - \dot{Q}_{SS}^{k=2}(17)]}{30}(57 - 17)$$

$$= 42.00 + \frac{[65.00 - 42.00]}{30}(40)$$

$$= 72.67 \text{ (kBtu/h)}$$

Example: $T_j = 32°F$

$$\dot{Q}^{k=2}(32) = \dot{Q}_{SS}^{k=2}(17) + \frac{[\dot{Q}_{DEF}^{k=2}(35) - \dot{Q}_{SS}^{k=2}(17)]}{18}(32 - 17)$$

$$= 42.00 + \frac{[50.00 - 42.00]}{18}(15)$$

$$= 48.67 \text{ (kBtu/h)}$$

Column (i): Applicable Case

A two-speed heat pump will operate in four possible cases, depending on the unit design and on a comparison of the building load, $BL(T_j)$, relative to the low and high speed capacities, $\dot{Q}^k(T_j)$, at each representative outdoor temperature, T_j. The four possible modes are described in Section 4.3. The equation to be used for evaluating the quantities in each of the subsequent columns (j) thorugh (p) will depend on which of the four cases is applicable for that temperature bin. Figure B1 illustrates the applicable cases for this particular sample problem.

Example: $T_j = 57°F$

$BL(57) = 7.187$ (kBtu/h)

$\dot{Q}^{k=1}(57) = 38.00$ (kBtu/h)

The low speed capacity is greater than the building load, and the unit is cycling on and off at low speed. The applicable equations for subsequent calculations are found under Case I.

Example: $T_j = 32°F$

$BL(32) = 29.645$ (kBtu/h)

$\dot{Q}^{k=1}(32) = 21.17$ (kBtu/h)

$\dot{Q}^{k=2}(32) = 48.67$ (kBtu/h)

Since BL(32) is less than $\dot{Q}^{k=2}(32)$ but larger than $\dot{Q}^{k=1}(32)$ and cycles between high speed and low speed, the equation to be used for subsequent calculations are found under Case II.

Example: $T_j = 12°F$

$BL(12) = 47.612$ (kBtu/h)

$\dot{Q}^{k=2}(12) = 38.17$ (kBtu/h)

The equation found under Case IV should be used in subsequent calculation since BL(12) is larger than $\dot{Q}^{k=2}(12)$ and supplementary resistance heat is required to meet the balance of the building heating requirements.

Columns (j) through (p): Applicable Case I

Example: $T_j = 57°F$

$\delta'(57) = 1.0$

$$X^{k=1}(57) = \frac{BL(57)}{\dot{Q}^{k=1}(57)}$$

$$= \frac{7.187}{38.00} = 0.189$$

$$PLF^{k=1} = 1 - C_D^{k=1}(1 - X^{k=1}(T_j))$$

$$= 1 - 0.2\ (1 - 0.189) = 0.838$$

$$\frac{E(57)}{N} = \frac{\dot{E}^{k=1}(57)\ X^{k=1}(57)\ \delta'(57)}{PLF^{k=1}}\ \frac{n_j}{N}$$

$$= \frac{(3.028)(.189)(1)}{.838}\ (.111) = 0.0758\ (kW)$$

$$\frac{RH(57)}{N} = \frac{n_j}{N}\ \frac{BL(57)\ [1 - \delta'(57)]}{3.413}$$

$$= \frac{(.111)(7.187)[1 - 1]}{3.413} = 0\ (kW)$$

Example: $T_j = 42°F$

$\delta'(42) = 1.0$

$$X^{k=1}(42) = \frac{BL(42)}{\dot{Q}^{k=1}(42)}$$

$$= \frac{20.662}{26.00} = 0.795$$

$$PLF^{k=1} = 1.0 - C_D^{k=1}\ (1 - X^{k=1}(42))$$

$$= 1 - .20\ (1 - .795) = 0.959$$

$$\frac{E(42)}{N} = \frac{\dot{E}^{k=1}(42)\ X^{k=1}(42)\ \delta'(42)}{PLF^{k=1}}\ \frac{n_j}{N}$$

$$= \frac{(2.881)(0.795)(1.0)}{.959}(.100) = 0.2388 \text{ (kW)}$$

$$\frac{RH(42)}{N} = \frac{n_j}{N} \; \frac{BL(42)[1 - \delta'(42)]}{3.413}$$

$$= \frac{(1.00)(20.662)[1 - 1]}{3.413} = 0 \text{ (kW)}$$

Columns (j) through (p): Applicable Case II

Example: $T_j = 32°F$

$$\delta'(32) = 1.0$$

$$X^{k=1}(32) = \frac{\dot{Q}^{k=2}(32) - BL(32)}{\dot{Q}^{k=2}(32) - \dot{Q}^{k=1}(32)}$$

$$= \frac{48.67 - 29.645}{48.67 - 21.17} = 0.692$$

$$X^{k=2}(32) = 1 - X^{k=1}(32)$$

$$= 1 - 0.692 = 0.308$$

$$\frac{E(32)}{N} = [\dot{E}^{k=1}(32)\; X^{k=1}(32) + \dot{E}^{k=2}(32) X^{k=2}(32)]\; \delta'(32)\; \frac{n_j}{N}$$

$$= [(2.803)(0.692) + (6.163)(0.308)](1)(.126)$$

$$= 0.4836 \text{ (kW)}$$

$$\frac{RH(32)}{N} = \frac{n_j}{N} \; \frac{BL(32)[1 - \delta'(32)]}{3.413} = 0 \text{ (kW)}$$

Columns (k) through (p): Applicable Case IV

Example: $T_j = 7°F$

$$\delta''(7) = 1.0$$

$$X^{k=2}(7) = 1.0$$

$$\frac{E(7)}{N} = \dot{E}^{k=2}(7)\; X^{k=2}(7)\; \delta''(7)\; \frac{n_j}{N}$$

$$= (4.486)(1)(1)(.013) = .0583 \text{ (kW)}$$

$$\frac{RH(T_j)}{N} = \frac{[BL(7) - \dot{Q}^{k=2}(7)\; X^{k=2}(7)\; \delta''(7)]}{3.413} \; \frac{n_j}{N}$$

$$= \frac{[52.103 - 34.33\ (1)(1)]}{3.413}\ (.013)$$

$$= 0.0677\ (\mathrm{kW})$$

Column (q): Building Heating Requirements for the j^{th} Temperature Bin

Col. (q) = (Col. d) x (Col. c)

Example: $T_j = 57°F$

$$BL(57)\ \frac{n_j}{N} = (7.187)(.111)$$
$$= .7978\ (\mathrm{kBtu/h})$$

The sum of the terms in column (o), in (kW), represents the ratio of the total heat pump electrical energy usage, $E(T_j)$, over the heating season to the total number of hours in all the temperature bins, N. The sum of the values in column (p), in (kW), represent the ratio of the total amount of supplemental resistance heating, $RH(T_j)$, to the total number of temperature bin hours, N. The ratio of the total building heating requirements to the total number of temperature bin hours, N, is determined from the sum of the terms in column (q), in (kBtu/h).

The HSPF is then evaluated as indicated in Table B2 or equivalently, by equation 4.13. The HSPF for this example two-speed heat pump in Region IV with a minimum standard DHR is shown in the Table to be 2.29. The seasonal cost of operation for a two-speed heat pump is calculated using equation 4.14. Performing the operations indicated in the Table and assuming a cost of electricity of \$.04/kWh results in a seasonal operating cost of approximately \$620.

The HSPF and seasonal operating cost of this example two-speed heat pump should be re-calculated for the same region (Region IV) with the heat pump installed in a house having the maximum standard design heating requirement, DHR_{max}. In calculating DHR_{max}, equation 4.3 should be used, rounding off to the values closest to those appearing in Table 6. The HSPF values and annual operating costs between DHR_{max} and DHR_{min} should then be calculated to determine the HSPF values and annual operating costs obtained between DHR_{max} and DHR_{min}.

Table B1. Sample Data Required to Evaluate the HSPF for a Two-Speed Compressor Heat Pump

	k=1	k=2
$\dot{Q}^K(62°F)$, kBtu/h	42.00	-(2)-
$\dot{Q}^K(47°F)$, kBtu/h	30.00	65.00
$\dot{Q}^K(35°F)$, kBtu/h	22.00	50.00
$\dot{Q}^K(17°F)$, kBtu/h	17.00	42.00
$\dot{E}^K(62°F)$, kW	3.077	-(2)-
$\dot{E}^K(47°F)$, kW	2.930	7.054
$\dot{E}^K(35°F)$, kW	2.865	6.370
$\dot{E}^K(17°F)$, kW	2.491	5.128
Degradation Factor, C_D^K	.20	-(3)-
T_{ON}, °F	-(1)-	
T_{OFF}, °F	-(1)-	

(1) Unit does not contain a low ambient temperature compressor cut-out

(2) Test not required

(3) Depending on the unit's operation, a value of $C_D^{K=2}$ may be required. This may be obtained by either a high speed cyclic test or, optionally, using a pre-assigned value of 0.25. In the example illustrated in Appendix B, no $C_D^{k=2}$ value is required.

Table B2. Example Worksheet for Evaluating the HSPF and Seasonal Cost of Operation for a Two-Speed Heat Pump

(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	(q)
Temp. Bin, j	Outdoor Temp. T_j, (°F)	Frac. Temp. Bin Hours $\frac{n_j}{N}$	$BL(T_j)$ (kBtu/h)	$Q^{k=1}(T_j)$, (kBtu/h)	$Q^{k=2}(T_j)$, (kBtu/h)	$E^{k=1}(T_j)$, (kW)	$E^{k=2}(T_j)$, (kW)	Applicable Case	$\delta(T_j)$	$X^{k=1}$	$X^{k=2}$	$PLF^{k=1}$	$PLF^{k=2}$	$E(T_j)\frac{n_j}{N}$, (kW)	$RH(T_j)\frac{n_j}{N}$, (kW)	$BL(T_j)\frac{n_j}{N}$, (kBtu/h)
1	62	.132	2.695	42.00	76.50	3.077	8.017	I	1	.064	NA	.812	NA[1]	.0320	0	.3557
2	57	.111	7.187	38.00	72.67	3.028	7.696	I	1	.189	NA	.838	NA[1]	.0758	0	.7978
3	52	.103	11.678	34.00	68.83	2.979	7.375	I	1	.344	NA	.869	NA[1]	.1215	0	1.2028
4	47	.093	16.170	30.00	65.00	2.930	7.054	I	1	.539	NA	.908	NA[1]	.1618	0	1.5038
5	42	.100	20.662	26.00	53.11	2.881	6.853	I	1	.795	NA	.959	NA[1]	.2388	0	2.0662
6	37	.109	25.153	22.56	50.89	2.907	6.508	II	1	.909	.091	NA[1]	NA[1]	.3526	0	2.7417
7	32	.126	29.645	21.17	48.67	2.803	6.163	II	1	.692	.308	NA[1]	NA[1]	.4836	0	3.7353
8	27	.087	34.137	19.78	46.44	2.699	5.818	II	1	.462	.538	NA[1]	NA[1]	.3808	0	2.9699
9	22	.055	38.628	18.39	44.22	2.595	5.473	II	1	.217	.783	NA[1]	NA[1]	.2667	0	2.1246
10	17	.036	43.120	17.00	42.00	2.491	5.128	IV	1	NA[1]	1.0	NA[1]	NA[1]	.1846	.0118	1.5523
11	12	.026	47.612	14.83	38.17	2.418	4.807	IV	1	NA[1]	1.0	NA[1]	NA[1]	.1250	.0719	1.2379
12	7	.013	52.103	12.67	34.33	2.345	4.486	IV	1	NA[1]	1.0	NA[1]	NA[1]	.0583	.0677	.6773
13	2	.006	56.595	10.50	30.50	2.272	4.165	IV	1	NA[1]	1.0	NA[1]	NA[1]	.0250	.0459	.3396
14	-3	.002	61.087	8.33	26.67	2.198	3.844	IV	1	NA[1]	1.0	NA[1]	NA[1]	.0077	.0202	.1222
15	-8	.001	65.578	-	22.83	-	3.523	IV	1	NA[1]	1.0	NA[1]	NA[1]	.0035	.0125	.0656
16	-13	0	70.070	-	19.00	-	3.202	IV	1	NA[1]	1.0	NA[1]	NA[1]	0	0	0
17	-18	0	74.562	-	15.17	-	2.881	IV	1	NA[1]	1.0	NA[1]	NA[1]	0	0	0
18	-23	0	79.053	-	11.33	-	2.560	IV	1	NA[1]	1.0	NA[1]	NA[1]	0	0	0
													TOTALS	2.5177	.2300	21.4928

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

Heating Load Hours, (HLH) 2250 (hrs)

Regional Outdoor Design Temperature, (T_{OD}) 5 (°F)

✓Minimum or Maximum

Design Heating Requirement, (DHR) 70 (kBtu/h)

$$HSPF = \frac{\Sigma \text{Col. (q)}}{3.413\ [\Sigma \text{Col. (o)} + \Sigma \text{Col. (p)}]} = \frac{21.4928}{3.413\ [2.5177 + .2300]} = 2.29$$

$$\text{Seasonal Operating Cost} = \frac{(HLH)(DHR)(C)(\$/kWh)}{3.413\ HSPF} = \frac{(2250)(70)(.77)(.04)}{(3.413)(2.29)} = \$620.$$

[1] NA means "not appropriate"

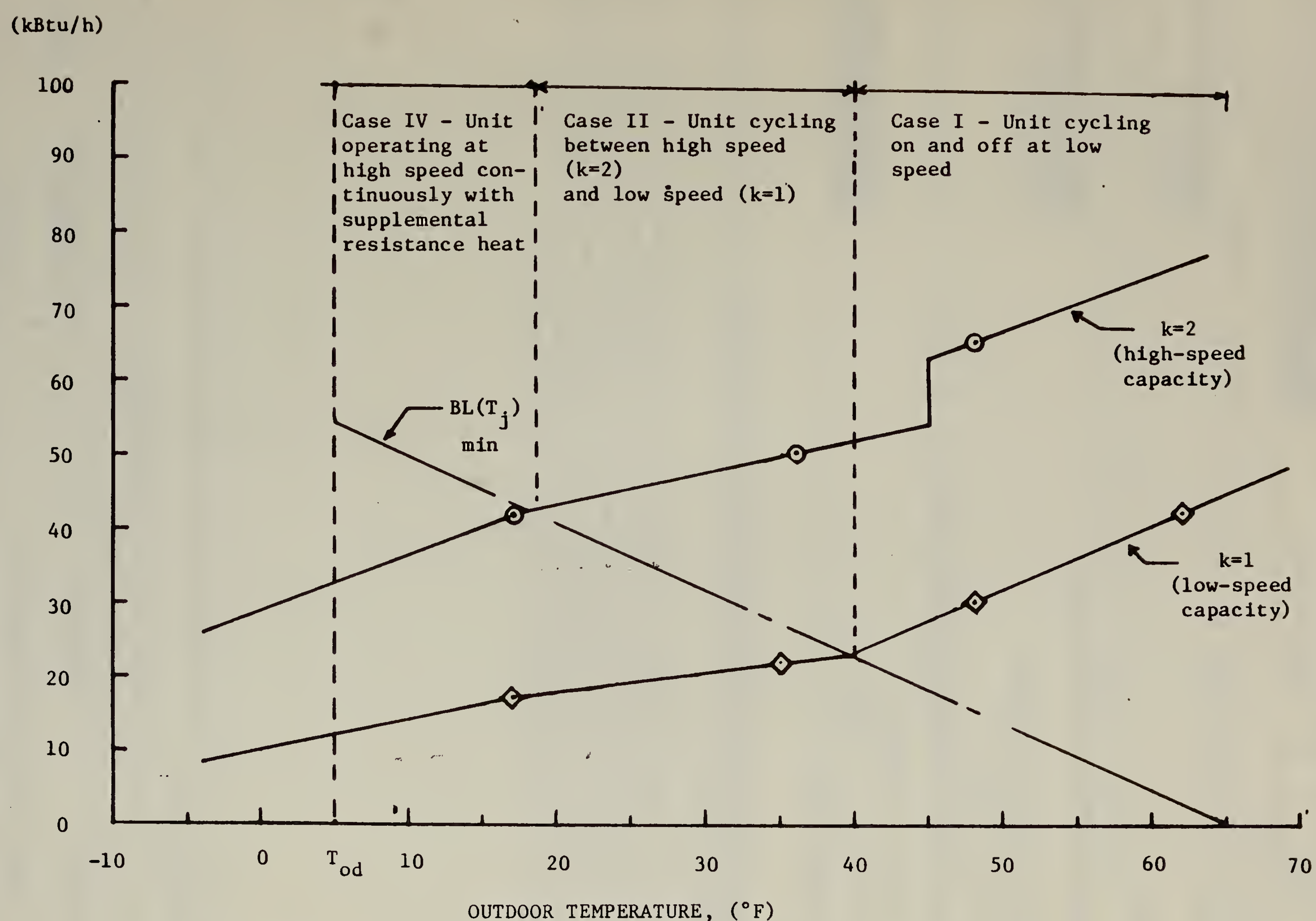


Figure B1. Sample Building Load and Two-Speed Heat Pump Performance Curves

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15. SUPPLEMENTARY NOTES

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

16. 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 electrically-driven residential air-to-air heat pumps operating in the heating mode. The procedures are designed to include the effects of part-load (cyclic) operation, variations in outdoor temperature, and frost formation on the heating performance. Using the test procedure results, a calculation procedure is presented for estimating the heating seasonal performance (HSPF) and cost of operation of residential heat pump units.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)*

Central air conditioners; central heating equipment; heat pumps; heating seasonal performance; rating procedure; residential heating; searonal performance; test method

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