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Recommended Procedure for Rating and Testing of Variable Speed Air Source Unitary Air Conditioners and Heat Pumps

Piotr A. Domanski

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
National Engineering Laboratory
Center for Building Technology
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U.S. DEPARTMENT OF COMMERCE, C. William Verity, *Secretary*
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

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VARIABLE SPEED AIR SOURCE UNITARY AIR CONDITIONERS AND HEAT PUMPS

Abstract

A procedure is presented for testing and rating variable speed, residential air conditioners and heat pumps. The procedure is derived in part from existing procedures for single speed and two speed systems. The main addition to the existing procedures is a new algorithm for representation of variable speed unit performance in the intermediate speed operation range. Analysis and background which led to the formulation of the procedure are included as well as calculation examples for the cooling and heating mode. This procedure has been prepared for the Department of Energy for consideration in the rule making process.

Key Words: Air conditioner, heat pump, rating procedure, seasonal performance, variable speed system.

DISCLAIMER

In view of the presently accepted practice of the building industry in the United States, common U.S. units of measurement are used in this report. In recognition of the United States as a signatory to the General Conference of Weights and Measures, which gave official status to the SI system of units in 1960, appropriate conversion factors are provided in the table below. The reader interested in making further use of the coherent system of SI units is referred to: NBS SP330, 1972 Edition, "The International System of Units," or E380-72, ASTM Metric Practice Guide (American National Standard 2210.1).

METRIC CONVERSION FACTORS

Length	1 inch (in) = 25.4 millimeters (mm) 1 foot (ft) = 0.3048 meter (m)
Area	1 ft^2 = 0.092903 m^2
Volume	1 ft^3 = 0.028317 m^3
Temperature	°F = 9/5°C + 32
Temperature Interval	1°F = 5/9°C or K
Mass	1 pound (lb) = 0.453592 kilogram (kg)
Mass Per Unit Volume	1 lb/ft^3 = 16.0185 kg/m^3
Energy	1 Btu = 1.05506 kilojoules (kJ)
Specific Heat	1 Btu/[lb · °F] = 4.1868 kJ/[kg · K]
Gallon	1 gallon = 0.0037854 m^3

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LIST OF SYMBOLS

$A_{95}^{k=2}$	=	steady state test in the cooling mode at conditions specified in Table 1
a	=	coefficient used in eq. (3.17) and eq. (4.19), (-)
BL(t)	=	building load at outdoor temperature t, (Btu/h)
$B_{82}^{k=1}$	=	steady state test in the cooling mode at conditions specified in Table 1
$B_{82}^{k=2}$	=	steady state test in the cooling mode at conditions specified in Table 1
$B_{67}^{k=1}$	=	steady state test in the cooling mode at conditions specified in Table 1
b	=	coefficient used in eq. (3.17) and eq. (4.19), (-)
C	=	0.77, an experience factor which improves the agreement between calculated and measured building loads, (-)
C_D	=	cyclic degradation coefficient of performance, (-)
CFM	=	volumetric flow rate of air, (ft^3/min)
COP	=	coefficient of performance, (-)
$C_{67}^{k=1}$	=	cyclic test in the cooling mode at conditions specified in Table 1
c	=	coefficient used in eq. (3.17) and eq. (4.19), (-)
DHR	=	design heating requirement, (Btu/h)
$D_{67}^{k=1}$	=	cyclic test in the cooling mode at conditions specified in Table 1
E(t)	=	power input to a system at outdoor temperature t during steady state operation, different subscripts may be used, (W)
EER	=	energy efficiency ratio, (Btu/W•h)
$e(t_j)$	=	power usage in bin temperature t_j, (W)
HSPF	=	heating seasonal performance factor, (Btu/W•h)
$I_{87}^{k=i}$	=	intermediate speed test in the cooling mode at conditions specified in Table 1
j	=	sequential number of a temperature bin, (-)

n_j	=	fractional bin hours in bin j, per Table 2 or Table 3, (-)
n_s	=	cycling rate of a single speed system, (1/h)
n_v	=	cycling rate of a variable speed system, (1/h)
PLF	=	part load factor, (-)
Q(t)	=	steady state capacity at outdoor temperature t, different subscripts and superscripts may be used, (Btu/h)
$q(t_j)$	=	cooling done in bin temperature t_j, (Btu/h)
$RH(t_j)$	=	power input to the supplementary resistant heater required in those cases when the heat pump automatically turns off ($t_j \leq t_{off}$), or when it is needed to meet the balance of the building heating requirement, (W)
RPM	=	speed, (revolutions/min)
SEER	=	seasonal energy efficiency ratio, (Btu/(W•h))
T_D	=	outdoor design temperature as specified in Table 4, (°F)
T_{off}	=	compressor time-off, (h)
T_{on}	=	compressor time-on, (h)
t	=	outdoor temperature, (°F)
t_j	=	representative outdoor temperature for temperature bin j (°F)
t_1	=	outdoor temperature in cooling at which the building load is equal to capacity of the system operating at the minimum compressor speed, (°F)
t_2	=	outdoor temperature in cooling at which the building load is equal to capacity of the system operating at the maximum compressor speed, (°F)
t_3	=	outdoor temperature in heating at which the building load is equal to capacity of the system operating at the minimum compressor speed, (°F)
t_4	=	outdoor temperature in heating at which the building load is equal to capacity of the system operating at the maximum compressor speed, (°F)
t_{on}	=	outdoor temperature at which the compressor is automatically turned on after low-temperature shut-off, (°F)

t_{off} = outdoor temperature at which the compressor is automatically turned off to avoid operation at low temperatures, (°F)

t_{vc} = outdoor temperature at which the building cooling load is equal to system capacity while operating at the compressor speed prescribed for the intermediate speed test, (°F)

t_{vh} = outdoor temperature at which the building heating load is equal to system capacity while operating at the compressor speed prescribed for the intermediate speed test, (°F)

$X(t_j)$ = cooling or heating load factor at a bin temperature t_j, (-)

$\delta(t_j)$ = heat pump low temperature cut-off factor, (-)

Superscripts:

k=n = nominal compressor speed

k=1 = the minimum compressor speed

k=2 = the maximum compressor speed

k=i = the intermediate speed specified for the intermediate speed test

k=v = an intermediate speed at which the unit capacity matches the building load at specified outdoor temperature

Subscripts:

def = defrost

f = indoor fan

i = intermediate speed test

max = maximum

min = minimum

s = single speed

ss = steady state

v = variable speed

1. INTRODUCTION

Most residential air conditioners and heat pumps employ single speed compressors and single speed fans. Such systems can provide only one level of capacity at given operating conditions. More advanced systems employing two speed compressors can provide two levels of capacity at given operating conditions. The distinguishing feature of variable speed systems is that their compressors can operate at different speeds and allow capacity to modulate within a certain range.

The interrelation between the building load and capacity of a variable speed system in the cooling mode is shown graphically in Figure 1. The diagonal line originating at the intersection of the ordinate and abscissa is the building load line. The line defined by system capacities at the maximum compressor speed, $Q_{ss}^{k=2}(82)$ and $Q_{ss}^{k=2}(95)$, provides a simplified representation of system capacity at the maximum compressor speed at different outdoor temperatures. Similarly, capacities $Q_{ss}^{k=1}(82)$ and $Q_{ss}^{k=1}(67)$ prescribe the system capacity line at the minimum compressor speed.

Because of the capacity range a variable speed system can provide, it has two balance points in a given installation, and three modes of operation as shown for the cooling mode in Figure 1. The low speed balance point, depicted as t_1, is the outdoor temperature at which the capacity line at the minimum compressor speed intersects the building load line. The high speed balance point, depicted as t_2, is the outdoor temperature at which the capacity line at the maximum compressor speed intersects the building

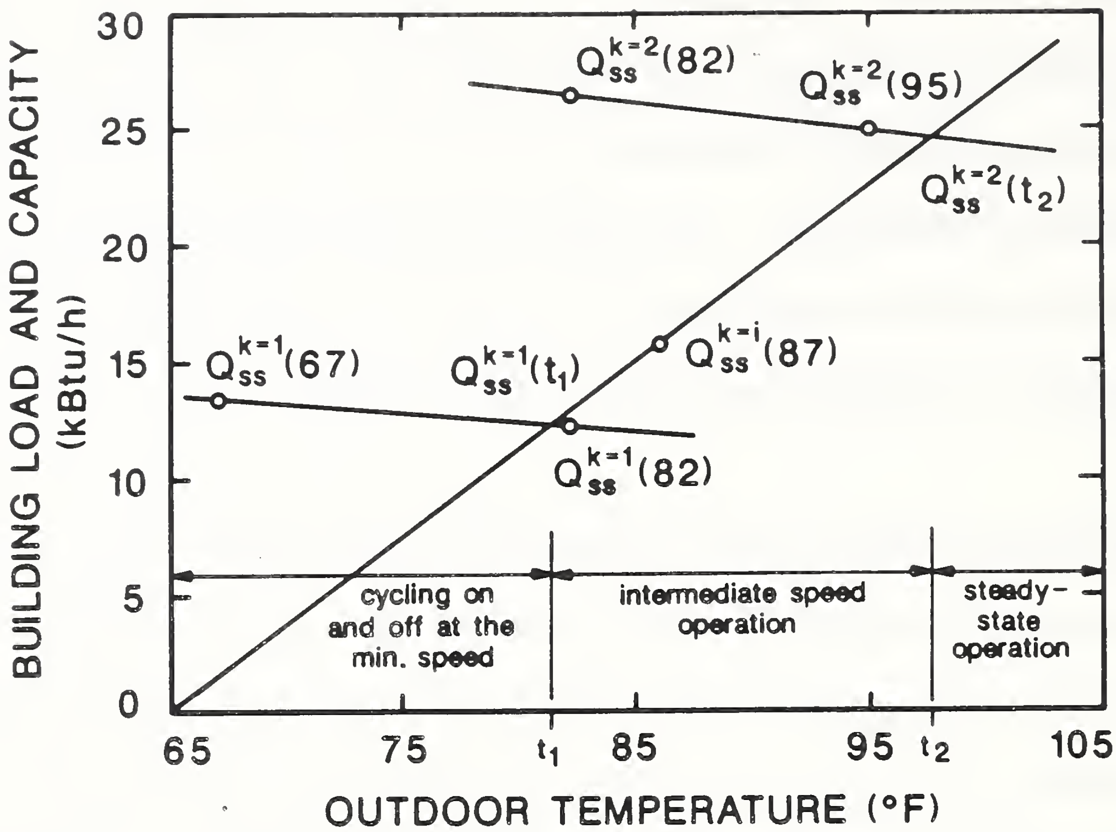


Figure 1. Building cooling load and capacity of a variable speed air conditioner.

load line. These two balance points separate three outdoor temperature ranges corresponding to three cases of operation of a variable speed unit:

65°F to t_1 - system cycles on and off at the minimum compressor speed to match the building load, case I

t_1 to t_2 - system operates continuously at an intermediate compressor speed to deliver the capacity that matches the building load, case II,

t_2 to 105°F - system operates continuously at the maximum compressor speed, case III.

Operation of a variable speed system at the outdoor temperature ranges from 65°F to t_1 (case I) and from t_2 to 105°F (case III) is similar to the operation of a two speed unit. As a result, appropriate portions of a two speed unit rating procedure can be, with no or little modifications, applied to represent performance of variable speed equipment. However, for the outdoor temperature range between the two balance points (case II) the existing U.S. Department of Energy (DoE) procedures [1] can not represent adequately the performance of a variable speed system. Similarly, three cases of operation of variable speed equipment can be identified in the heating mode with no procedure available to account for performance in the intermediate speed region. Consequently, new procedures are developed in this report to allow rating of variable speed systems.

This report presents analysis and development of a rating procedure for a variable speed system. At the time the analysis was performed no variable speed system was available on the market, making it impossible to base the procedure on NBS tests of actual units. The rating procedure was prepared based on NBS experience with other residential heat pump products for which rating procedures exist, and also based on industry waiver petitions to DoE test procedures, and on comments on the DoE proposed procedure for variable speed systems [2].

This procedure evolved from the existing DoE procedures for single speed and two speed systems [1]. Section 2 of this document describes, as a background, only modifications and enhancements made to the existing procedures to rate variable speed equipment. Reader's familiarity with the existing procedures is assumed. This procedure has been prepared for DoE as a NBS recommendation in the rule making process.

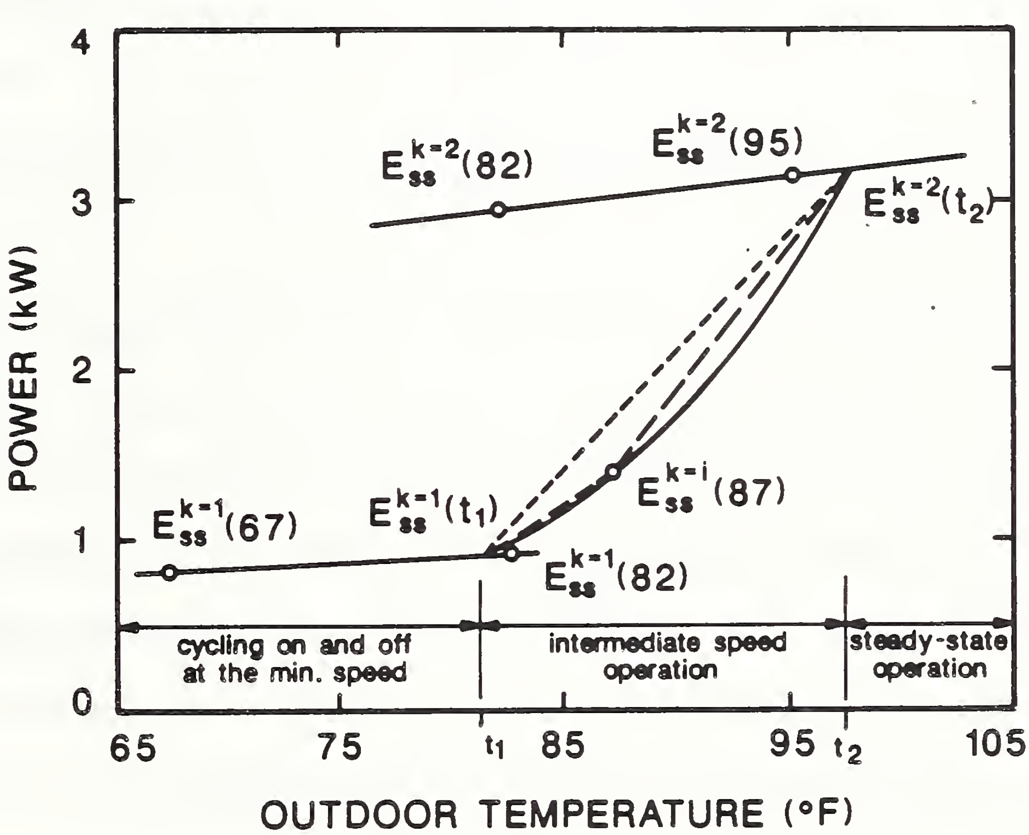
2. BACKGROUND OF THE PROCEDURE DEVELOPMENT

2.1. The Intermediate Speed Test

The main shortcoming of the existing rating procedure [1] with respect to variable speed systems is that it cannot adequately represent performance of variable speed equipment in the intermediate speed range. Consequently, each manufacturer who planned to develop and market a variable speed system formulated his own rating procedure and petitioned DoE for approval as a waiver.

The first variable speed rating procedure was proposed by Borg-Wagner [3] and was basically adopted in ARI Standard 210/240 [4]. As the major concern was an accurate representation of a non-linear power input profile in the intermediate speed operation range, the procedure introduced an intermediate speed test in the cooling and heating mode at 87°F and 35°F outdoor temperature, respectively. Taking into account bin hour distributions and practical considerations, the selection of these outdoor temperatures as test conditions was appropriate and was followed by other manufactures in their waiver petitions.

The available points for power interpolation are shown for the cooling mode in Figure 2. The two points at the edges of the intermediate speed region, $E_{ss}^{k=1}(t_1)$ and $E_{ss}^{k=2}(t_2)$, can be calculated from the minimum and maximum speed capacity and power input equations, and the building load algorithm. The point, $E_{ss}^{k=i}(87)$, is the additional datum obtained from the intermediate speed test. This third point allows for a better representation of the non-linear power input profile, which otherwise would have to be prescribed by a straight line connecting $E_{ss}^{k=1}(t_1)$ and $E_{ss}^{k=2}(t_2)$, as shown by the short dashed line in Figure 2, overestimating the power input.



---- linear interpolation of power using two points
— — linear interpolation of power using three points
—— transposed from linear interpolation of EER using three points; expected shape of typical characteristic curve

Figure 2. System power at intermediate speed operation using different interpolations.

The ARI procedure [4] requires the system capacities during the intermediate speed tests to be equal to the respective building loads calculated for the outdoor temperatures at which the cooling and heating tests are performed. The matching of loads and capacities should be obtained by the proper selection of compressor speeds. The matching requirement is much easier to satisfy in cooling than in the heating mode. To avoid an iterative testing at different speeds while searching for the speed at which matching occurs, Urbs et al, [5], proposed an algorithm which allows use of test data obtained when the capacity does not match the load. The algorithm uses capacity slopes at the maximum and minimum compressor speeds. By weighing these slopes, the capacity line for the intermediate compressor speed at which the test was performed is determined. The power line at the intermediate speed is similarly obtained. Having prescribed capacity and the building load lines, the temperature can be found at which both lines intersect, i.e. in which the system operating at the tested intermediate speed matches the building load. The power input to the system is then evaluated at this temperature, and this value is used in interpolation of power for the intermediate speed operation range. For details on this algorithm the reader may refer to the original publication. NBS found this procedure adequate and highly practical, and adopted it in this proposed rating procedure for variable speed systems.

Once it is accepted that a match between the building load and the system capacity is not necessarily needed, the selection of the compressor speed for the intermediate speed test had to be otherwise defined. A strict prescription of the speed is preferable for rating verification.

Again, the objective of speed selection is to obtain capacity that would match or be close to the building load. The task of prescribing such speed for different systems in a standardized way appeared to be quite difficult because different systems possess different capacity vs RPM characteristics, dependent on electric drive and compressor characteristics, control strategy relating fan speeds to the compressor speed, expansion device characteristics, etc.

After a review of the waiver procedures for variable speed systems of two manufacturers [6,7], the following equation is proposed for prescribing the speed for the intermediate speed test in the cooling mode:

$$\text{intermediate speed} = \text{minimum speed} + (\text{maximum speed} - \text{minimum speed})/3 \tag{2.1}$$

For the heating mode, the same speed as for cooling is prescribed. If the unit controls allow frequency to be changed only in discrete steps such that the prescribed speed cannot be attained in the tested equipment, the test should be performed at the next higher input frequency level available in the system.

The above paragraphs identify the compressor speed at which the intermediate speed test should be attempted to run. However, constant compressor speed may be difficult to maintain and, therefore, the following variations from the prescribed speed should be allowed:

- those associated with variations of the input line frequency and instability of controls. These variations should not result in decrease of the compressor speed below the speed calculated by equation (2.1); if this happened to be the case, the test should be performed at the next higher available speed step.
- those associated with control strategies which control speed in the field during frosting/defrosting operation: during the initial period after switching from defrosting into heating, in the final period of the heating portion of the frost accumulation cycle, and during defrosting. Variation of the compressor speed from the test value shall be allowed in these instances to the same extent and for the same period of time as such variations would occur in the field due to control strategy.

2.2 Interpolation of the Intermediate Speed Power Input

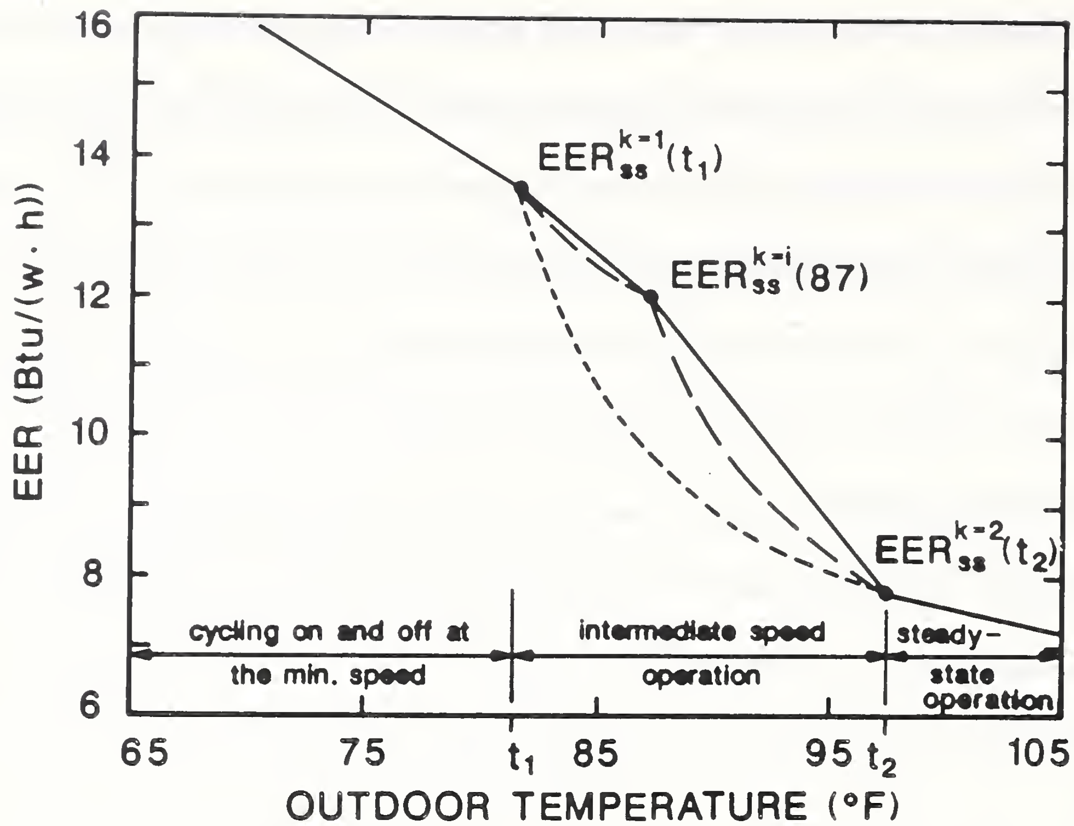
The ARI procedure for variable speed equipment [4] as well as waiver procedures of individual manufacturers [3,6,7] evaluated the power input in the intermediate speed range by applying linear interpolation between the three known data points; the power at the minimum speed balance point, $E_{ss}^{k=1}(t_1)$, the power at the intermediate speed test, $E_{ss}^{k=i}(87)$, and the power at the maximum speed balance point, $E_{ss}^{k=2}(t_2)$. This method is shown for the cooling mode in Figure 2 by the long dashed line.

It was recognized from the beginning that additional intermediate speed tests at other than 87°F and 35°F temperatures would further improve the rating of a variable speed product. However, because of the increased testing burden, the manufacturers who submitted waver petitions for their variable speed products opted for one intermediate test point and a lower rating. An improved calculation procedure for seasonal performance of a variable speed system (that would provide a rating closer to real system performance) can be formulated by introducing two modifications to the existing methods:

1. instead of interpolating the power input, performing EER or COP interpolation and then calculating the power input values by dividing capacity by EER or COP, respectively,
2. using parabolic instead of a linear interpolation.

The advantage of interpolating EER rather than power can be deduced by examining Figure 2 and Figure 3. The figures show system power and efficiency lines in the intermediate speed operation range in the cooling mode. The shape of the lines depends on the interpolation method used. The short dashed line in Figure 2 connecting system power inputs, $E_{ss}^{k=1}(t_1)$ and $E_{ss}^{k=2}(t_2)$, at the minimum and maximum compressor speeds, respectively, represents most conservative power input assessment. The long dashed line connecting system powers at t_1, 87°F and t_2 represents the interpolation method used in ARI Standard 210/240. Experimental evidence has shown that none of these lines prescribe adequately the system power but rather the solid line provide a realistic representation of the power input.

As system power and EER are related through system capacity, (EER = capacity/power) it is of interest to examine the shape of EER lines (Figure 3) in the intermediate speed range in relation to selected



---- transposed from linear interpolation of power using two points

— — transposed from linear interpolation of power using three points

—— linear interpolation of EER using three points

Figure 3. System EER at intermediate speed operation using different interpolations.

interpolating methods of the system power shown in Figure 2. There is a direct relationship between the line pattern shown in Figure 2 and Figure 3; i.e. the short dashed line representing EER in Figure 3 corresponds to the assumption that the system power changes linearly between t_1 and t_2 as shown by the short dashed line in Figure 2. Linear interpolation of power between the three points results in an unrealistic shape of the EER line in Figure 3. The continuous line in the intermediate region in Figure 3 represents a linear interpolation of EER between three points. This linear interpolation of EER results in a realistic representation of the system power as shown by the solid concave line in Figure 2.

In brief, the power input to a variable speed system in the intermediate speed operating range, can be estimated by interpolating either system power or system EER. The parameter selected for interpolation should be the one which is more linear in the interpolation region. The experimental evidence is that EER is a more linear parameter than power.

Having chosen EER for interpolation in the intermediate speed operation range, it is not inconsequential which interpolation method is applied. The parabolic interpolation method, as compared to the linear, has two important advantages:

- unlike the power line, the EER line in the intermediate speed range may be either convex or concave. The reason that this line may be concave or convex is that, since it is nearly linear, a small deviation in system characteristics in either direction will cause it to be either above (convex) or below (concave) linearity. A straight line interpolation of EER would unduly benefit systems with a concave EER line.
- from the comments received from the industry on the proposed procedure for variable speed systems it appears that the EER line is indeed of a second or higher order. Although different systems may have different characteristics, a parabolic fit should give the best estimate of the EER line with three performance points available.

Although the above analysis refers to the cooling mode, the same conclusions hold for the heating mode. Consequently, the recommended rating procedure uses parabolic interpolation of an efficiency descriptor (EER or COP) for representation of system performance over the intermediate speed range in both cooling and heating. The recommended form of the interpolation equation is $y = a + b \cdot x + b \cdot x^2$.

2.3 Cyclic Test

The cooling cyclic test for single speed systems and two speed systems is prescribed at an outdoor 82°F temperature [1]. The cyclic test for variable speed systems is specified in ARI Standard 210/240 [4] at an outdoor temperature of 67°F. Although the 67°F temperature constitutes a departure from the standard cyclic test 82°F condition, it has been adopted in the procedure contained in this report. This selection is based on the practical consideration of assigning a cyclic test at the same outdoor temperature as one of the wet coil tests. Because variable speed systems are expected to operate at the minimum speed in the outdoor temperature range from 65°F to approximately 80°F (depending on the capacity modulation ratio), minimum speed wet coil test outdoor temperatures were selected to be 82°F and 67°F. Since a variable speed system is not likely to cycle in the field at an outdoor temperature above 80°F, the temperature of 67°F seems to be the better selection for the cyclic test. For similar reasons, the ARI Standard 210/240 62°F outdoor temperature for heating cyclic test was also adopted.

It should be pointed out that NBS laboratory cyclic tests performed on a single speed system showed insignificant dependence of the C_D value on the outdoor temperature as long as the compressor time-on and time-off are preserved. The C_D values for tests at outdoor temperatures of 95°F, 82°F, and 70°F were the same within the experimental error.

Another departure from the single speed cyclic test is the length of the time-on and time-off. Since a variable speed system will start to cycle "on" and "off" at a much lower outdoor temperature than a single speed system, the variable speed unit should be allowed more than 6 minutes time-on during the cyclic test. An analysis using simple thermostat equations will allow derivation of appropriate time-on, time-off, and cycling rate values for variable speed equipment.

Figure 4 presents a typical parabolic curve representing the compressor cycling rate as a function of the cooling load. The upper graph describes the compressor cycling rate for a single speed system whose capacity line coincides with the capacity of a variable speed system operating at the maximum speed. The lower graph describes the cycling rate of the variable speed system. Cycling starts at the building load equal to $Q_{ss}(t_2)$ for the single speed system (which is equal to $Q_{ss}^{k=2}(t_2)$ of the equivalent variable speed system), and at the building load equal to $Q_{ss}^{k=1}(t_1)$ for the variable speed system.

In order to find the cycling rate of the variable speed system, the relationship between the time-on and time-off periods for the variable and single speed systems must be first determined. Since both units would be installed in the same structure, the same building load would have to be satisfied. Also, assuming that the thermostat band is the same for both systems, the same amount of heat would have to be transferred to the house (independently of the outdoor temperature) to trigger a compressor in each system, i.e. $BL(t_j) \cdot T_{off}(t_j) = \text{const.}$

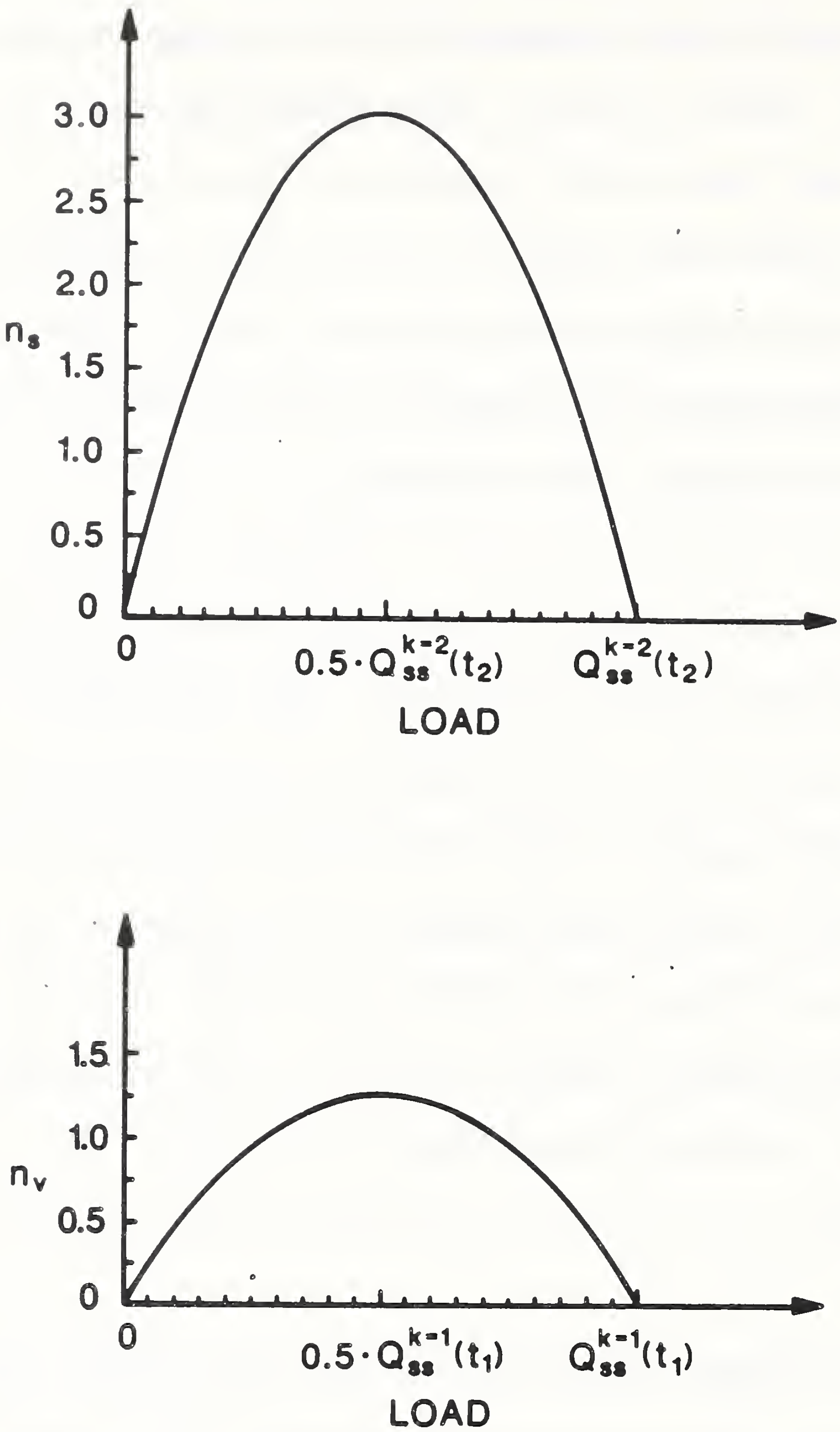


Figure 4. Cycling rate for a single speed system, n_s, and a variable speed system, n_v.

In order to find time-off for the variable speed system, we may write this equation for both systems for the same load fraction and equate both sides; for example, for a 20% fraction,

$$T_{off,s} \cdot 0.2 \cdot Q_{ss}^{k=2}(t_2) = T_{off,v} \cdot 0.2 \cdot Q_{ss}^{k=1}(t_1) \qquad (2.2)$$

where: $T_{off,s}$ and $T_{off,v}$ are time-off of single speed and variable speed systems, respectively.

Consequently,

$$T_{off,v} = T_{off,s} \cdot Q_{ss}^{k=2}(t_2) \;/\; Q_{ss}^{k=1}(t_1) \qquad (2.3)$$

Assuming the start-up characteristics of both systems are similar and the percentages of time-on and time-off are preserved, time-on for the variable speed system, $T_{on,v}$, may be expressed as a function of the time-on of the equivalent single speed system, $T_{on,s}$.

$$T_{on,v} = T_{on,s} \cdot Q_{ss}^{k=2}(t_2) \;/\; Q_{ss}^{k=1}(t_1) \qquad (2.4)$$

Employing equations (2.3) and (2.4), the cycling rate of the variable speed system, n_v, in relation to the cycling rate of the single speed system, n_s, is determined.

$$n_v = n_s \cdot Q_{ss}^{k=1}(t_1) \;/\; Q_{ss}^{k=2}(t_2) \qquad (2.5)$$

The above equations, although derived with some simplifications, could serve as an algorithm for determining time-on and time-off during the cycling test of a variable speed system. The equations would allow a variable speed system with a greater capacity modulation ratio, $Q_{ss}^{k=2}(t_2) / Q_{ss}^{k=1}(t_1)$, to have longer compressor time-on during the test which would result in a lower value of the cyclic degradation coefficient, C_D. However, applying these equations could result in a long and costly cyclic test. For a capacity modulation ratio of 2, the cyclic test would last one hour, but two hours would be required for a system having a capacity modulation ratio of 3. Conversely, the benefit of a lower C_D value is smaller for systems with a higher capacity modulation ratio since the systems cycles less often. The decreasing effect of C_D on the Seasonal Energy Efficiency Ratio, SEER, is shown in Figure 5. The figure shows the maximum theoretical penalty in SEER which we define as a difference in the SEER value calculated with $C_D = 0.25$ and $C_D = 0$. The figure shows that, from a practical standpoint, there is no reason to run long cyclic tests for high capacity modulation systems since the C_D improvement will not justify the increased burden of a prolonged test. As a result, a fixed time-on equal to 12 minutes is prescribed in the proposed procedure for systems having a capacity modulation ratio of 2 or

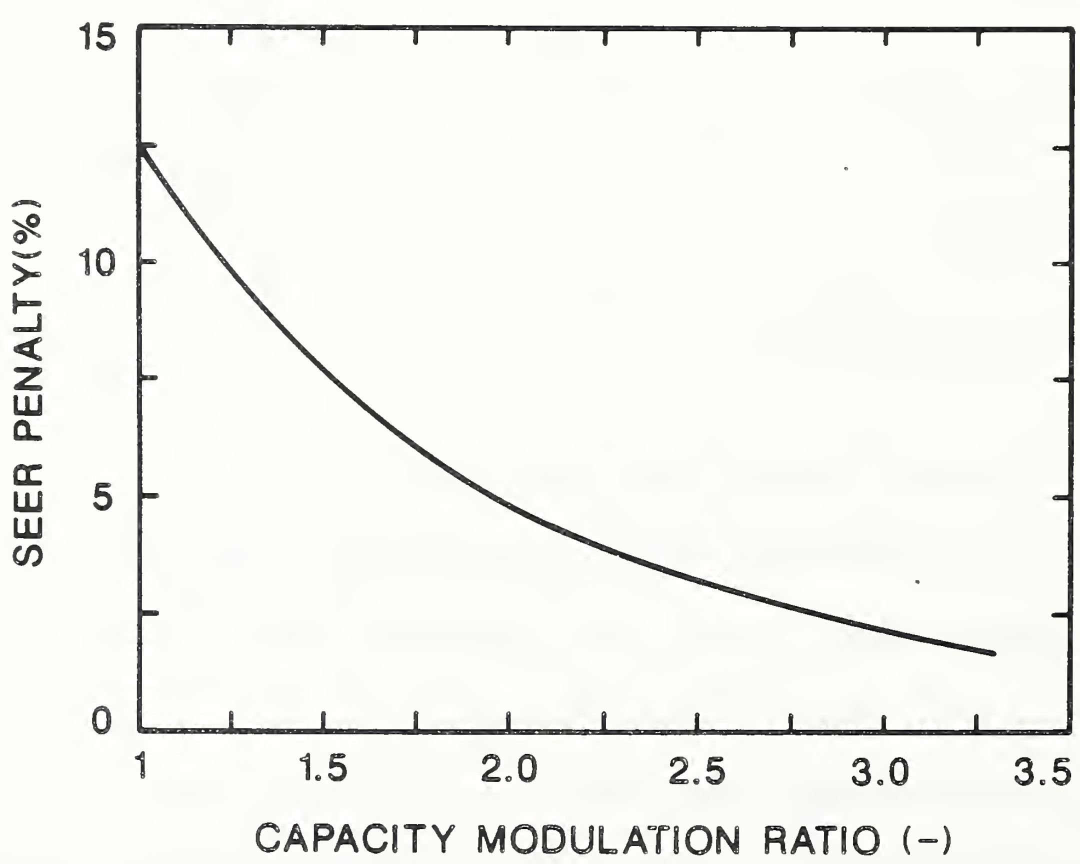


Figure 5. Maximum theoretical SEER penalty at default C_D.

more. The equations presented above are recommended for systems having capacity modulation ratio smaller than 2.

As was done for the cooling mode, a similar analysis can be repeated for the heating mode. The situation in the heating mode is more complicated because of a range of building loads (between the maximum and minimum) prescribed by the DoE rating procedure [1]. If the steps of the cooling mode analysis are followed rigorously, different values for time-on and time off could be prescribed for different building heating loads. A heat pump with a capacity modulation ratio of 2 in the cooling mode could have a compressor time-on between 9 minutes and 12 minutes in the heating mode, depending on the design heating requirement. For simplicity, the proposed procedure prescribes the heating mode time-on and time-off to be the same as for cooling cycling test.

2.4 Capacity and Power Input Lines at the Maximum Compressor Speed in the Heating Mode

The rating procedure for variable speed equipment contained in ARI Standard 210/240 [4] prescribes the capacity and power lines based on results of two tests at temperatures of 17°F and 47°F. The same method was proposed in two waiver procedures [6,7] submitted to DoE. This prescription of the capacity and power lines does not include degradation due to frost accumulation and, therefore, is inconsistent with reality and the rating procedures already in place for other equipment. The simplification of the linear representation of capacity and power characteristics based on 17°F and 47°F points, however, does not affect the Heating Seasonal

Performance Factor (HSPF) value calculated for the minimum heating design requirement. This is due to the intersection between the minimum building load line and the maximum speed capacity line occurring in the neighborhood of 14°F for region V, and in the neighborhood of 19°F for the other regions. Thus, the method of representation of capacity and power used in previous procedures [4,6,7] does not affect the HSPF value for the minimum design heating requirement in region IV which is prominently used for marketing purposes. For higher design heating requirements, the linear representation underestimates the power input required to the electric heater and, as a result, overestimates the efficiency descriptor.

NBS believes that comparability of variable speed equipment and single speed equipment ratings should be maintained by including the frost accumulation test results at the maximum speed, $Q_{def}^{k=2}(35)$ and $E_{def}^{k=2}(35)$, in prescription of the capacity and power input lines. Since results of this test affect HSPF values derived for lesser used design requirements, NBS recommends that, as an alternative to test, the use of equations (2.6) and (2.7) be allowed to obtain the values for $Q_{def}^{k=2}(35)$ and $E_{def}^{k=2}(35)$.

$$Q_{def}^{k=2}(35) = 0.90 \circ [Q_{def}^{k=2}(17) + [Q_{def}^{k=2}(47) - Q_{def}^{k=2}(17)][35-17]/[47-17]] \quad (2.6)$$

$$E_{def}^{k=2}(35) = 0.985 \circ [E_{def}^{k=2}(17) + [E_{def}^{k=2}(47) - E_{def}^{k=2}(17)][35-17]/[47-17]] \quad (2.7)$$

Correction factors of 10 percent for capacity and 1.5 percent for power in these equations were selected based on review of test data of ten single speed heat pumps [8] equipped with demand defrost (Figure 6 and 7). These correction factors result in COP degradation of 8.6% which is in line with COP degradation observed in the tested systems, as is shown in Figure 8.

2.5 Building Load in the Heating Mode.

It is likely that some variable speed systems which will be available commercially will be able to run at a higher compressor speed in the heating mode than in the cooling mode. As sizing of heat pumps is usually performed based on the cooling load, the ability to run at a higher maximum speed in the heating mode lowers the balance point - the temperature at which the electric heater is turned on by system controls. The efficiency benefit associated with the higher compressor speed in the heating mode depends on the relative position of the building load line

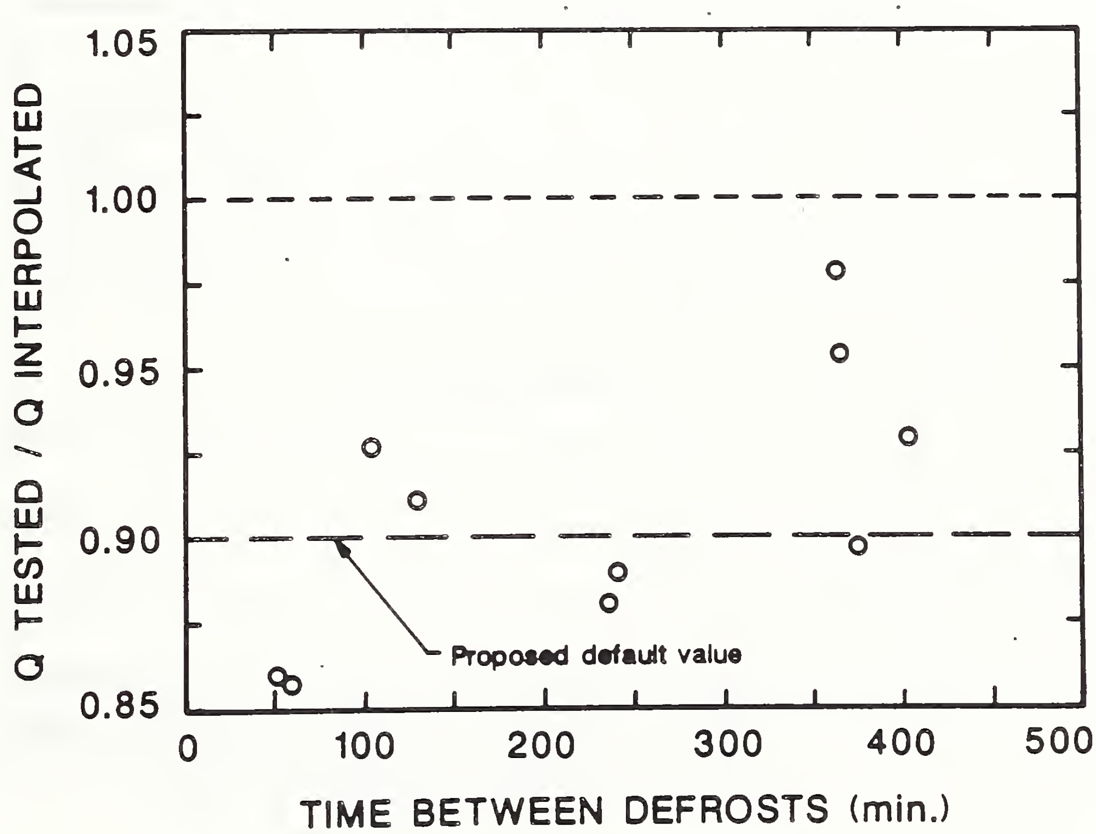


Figure 6. Ratio of measured capacity to linearly interpolated capacity (using 17°F and 47°F data) for the frost accumulation test.

and the capacity line. The building load line, as prescribed in the existing procedures [1], is a function of capacity at 47°F outdoor temperature. If the system capacity at the maximum compressor speed, $Q_{ss}^{k=2}(47)$, were used for the building load calculations, the calculated HSPF value would not be much different from the HSPF value calculated for a system operating at the same maximum speed in both heating and cooling.

In order to allow for an appropriate credit, the proposed procedure prescribes an optional, nominal capacity test at 47°F outdoor temperature. This test is applicable only if controls of the unit allow the compressor to run in the heating mode at a higher speed than in the cooling mode. The nominal capacity test should be performed at the compressor speed

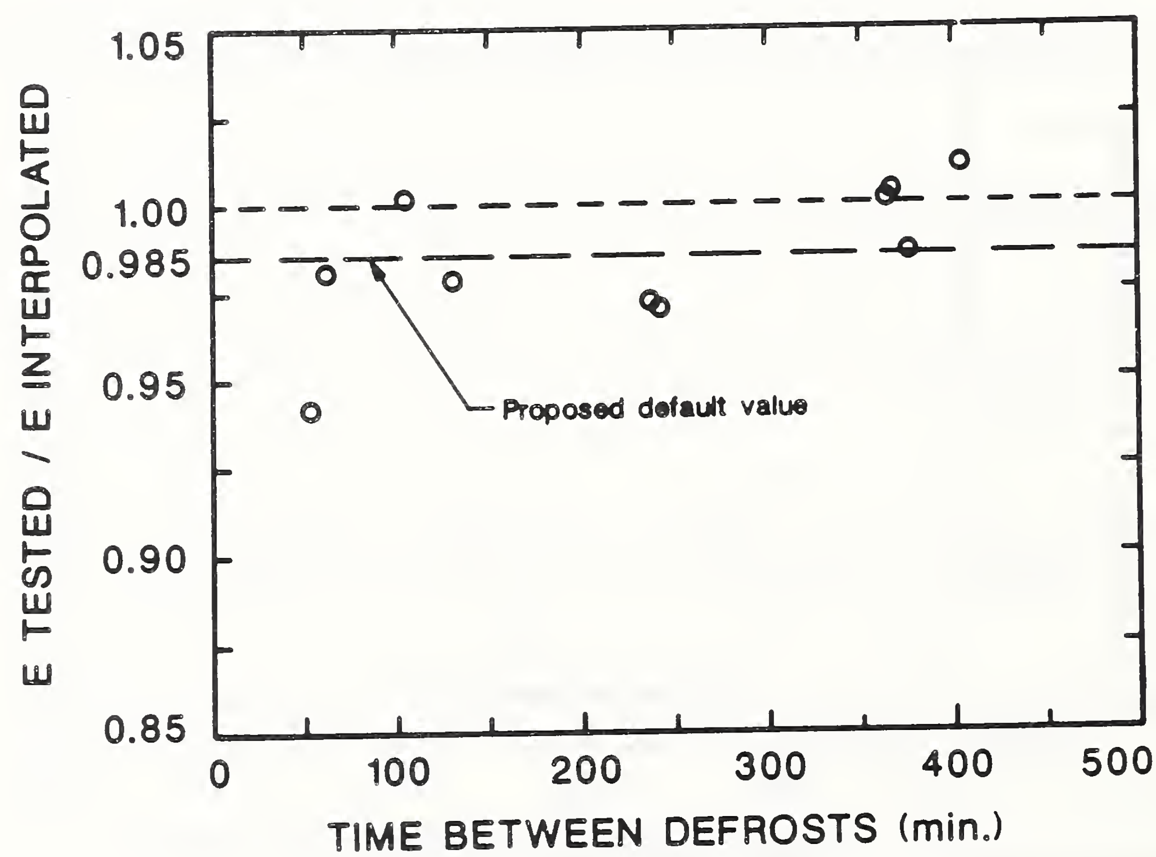


Figure 7. Ratio of measured power to linearly interpolated power (using 17°F and 47°F data) for the frost accumulation test.

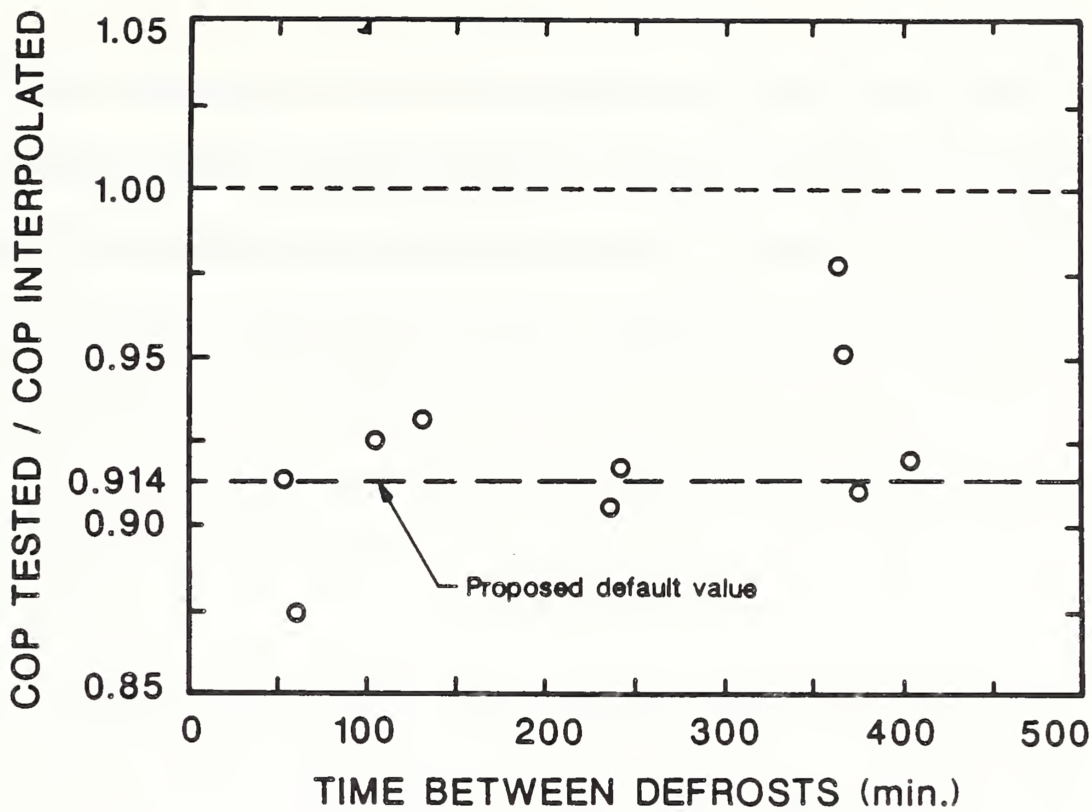


Figure 8. Ratio of measured COP to linearly interpolated COP (using 17°F and 47°F data) for the frost accumulation test.

equal to the compressor speed during the cooling test at 95°F temperature. The system capacity obtained in the nominal capacity test is to be used exclusively for evaluation of the design heating requirement.

2.6 Measurement of the Electrical Energy Input

The existing DoE rating procedure [1] requires measurement of the energy input to a system within 0.5 percent uncertainty. This requirement is not difficult to meet for single and two speed systems and, most likely, can be satisfied with an induction (rotating disc) type watthour meter.

Among the various types of variable speed drives possible, drives employing some kind of a solid state inverter are expected to be used in heat pumps.

An undesirable feature of electric inverters is that they may inject large amounts of harmonic currents into the utility system. The amount of current wave distortion from its sinusoidal shape depends on the type of the inverter and its detailed design. For illustration, Figure 9 presents deformations of a current waveform recorded at the input to a three phase inverter. An informative presentation of different types of solid state inverters and their input and output current waveforms may be found in [9].

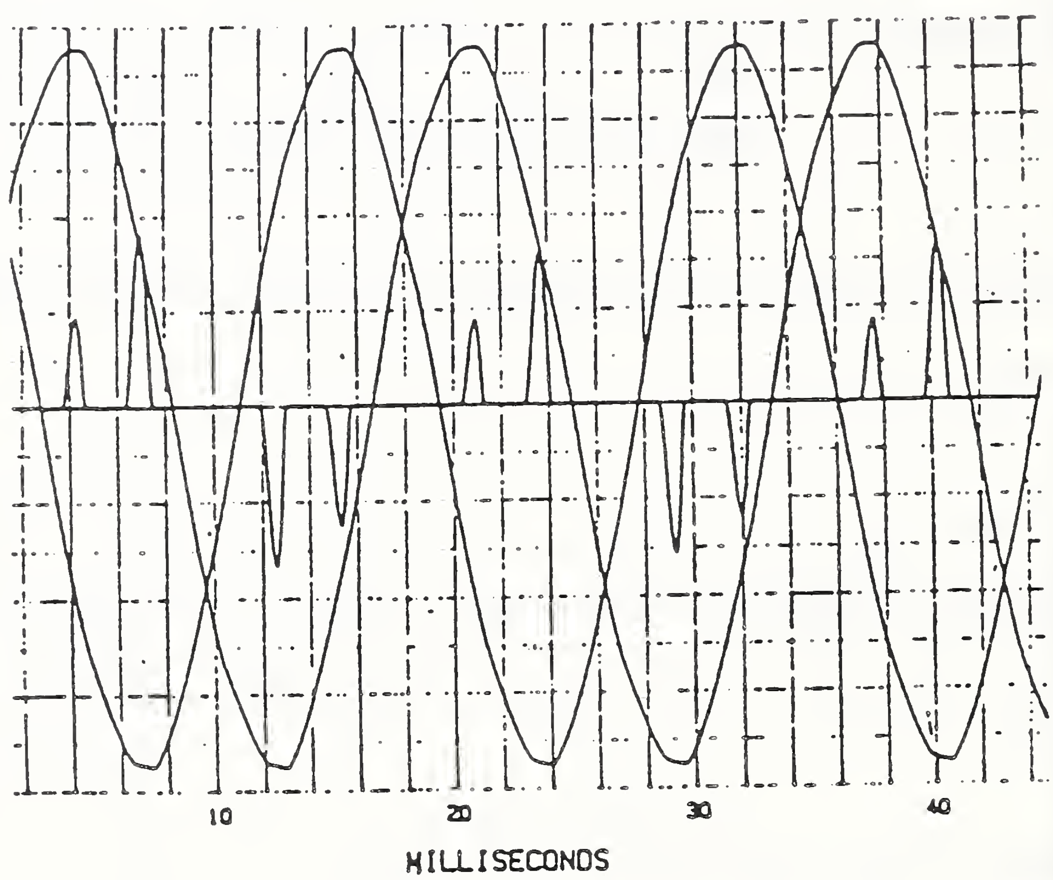


Figure 9. Deformation of the current waveform at input to a three phase inverter at 30 Hz output frequency (two voltage and one current waveforms are recorded).

Distortion of the current waveform on the line side makes the measurement of the energy input more difficult. Significant errors may occur if an inductive watthour meter if used. Baldwin et al. [10] conducted a series of tests on commercially available, self-contained, induction watthour meters. When exposed to current waveform distortion due to a variable-speed controller (three-phase), the meters over registered even if the voltage was undistorted at 60 Hz. The average error was 2 percent with an individual meter over registering by as much as 6.7 percent.

Better accuracy of energy input measurement for distorted wave forms can be obtained with an electronic-type watthour metering device. However, our review of specifications of a number of electronic meters of different manufacturers showed that the measurement uncertainty of 0.5 percent cannot be obtained even with an electronic meter unless an excessively priced meter is used. Therefore, the proposed rating procedure specifies a relaxed uncertainty of 1 percent of the energy input measurement of systems employing a solid state inverter.

2.7 Needed Research

Numerous assumptions were made in the development of the proposed testing and rating procedure described in this report. Some assumption were related to performance characteristics in the intermediate speed region. One of the unknown aspects of system operation is its ability to closely follow the building load in the intermediate speed range. The proposed procedure assumes that the heat pump capacity exactly matches the building load at all times in the intermediate speed operation range. The penalty due to the compressor speed hunting in search of the matching capacity has not been evaluated.

The frost accumulation test at the intermediate speed, as prescribed by the procedure, represents a significant departure from the intermediate speed operation in the field. To simplify the procedure, a constant compressor speed during the test is prescribed, while in field operation the compressor speed would increase with accumulation of frost to provide capacity matching the building load. Accumulation of frost provided significant complexity to the rating procedure for single speed systems; accurate representation of variable speed system performance when frosting of the outdoor coil occurs is even more difficult.

Since there was no variable speed equipment available on the market at the time of formulation of this rating procedure, NBS did not have its own laboratory data for support of the procedure development and relied on data and information made available by others [11,12,13,14]. It is believed that the proposed procedure captures the major performance characteristics common to most of the forthcoming variable speed products. It has to be realized, however, that continuous developments in electronics, sensors and control strategies will provide future systems with performance advantages that might not be accounted for within the presently proposed rating methodology. Thus the impact of system controls, frosting and load matching capability are all issues for further investigations.

3. RATING PROCEDURE FOR VARIABLE SPEED SYSTEMS IN THE COOLING MODE

3.1 Requirements for Testing Conditions and Testing Procedures

Requirements regarding instruments and data acquisition systems, test apparatus, methods of test, test procedures, and data analysis are as prescribed in ASHRAE Standard 116-1983 [15] with the exception to the provisions specified in the following sections.

3.1.1 Tests and Testing Conditions

The tests and test conditions are presented in Table 1. The required tests include five wet coil tests ($A_{95}^{k=2}$, $B_{82}^{k=2}$, $B_{82}^{k=1}$, $B_{67}^{k=1}$, $I_{87}^{k=i}$) and two dry coil tests ($C_{67}^{k=1}$, $D_{67}^{k=1}$).

3.1.2 Cyclic Test

The cyclic test, $D_{67}^{k=1}$ (see Table 1), shall be conducted by cycling the unit "on" and "off" by manual or automatic operation of the normal control circuit of the unit. The unit shall cycle "on" and "off" with the compressor time-on, T_{on}, and time-off, T_{off}, determined as follows:

$$T_{on} = \begin{cases} 6 \cdot \dfrac{\dot{Q}_{ss}^{k=2}\ (95)}{\dot{Q}_{ss}^{k=1}\ (82)} & \text{or} \\ 12, & \text{whichever is smaller} \end{cases} \tag{3.1}$$

$$T_{off} = 4 \cdot T_{on} \tag{3.2}$$

The indoor fan and outdoor fan shall also cycle "on" and "off"; the duration of the fans "on" and "off" periods shall be governed by the automatic controls which the manufacturer normally supplies with the unit. The installation shall be designed to prevent air flow through the indoor unit due to natural or forced convection while the indoor fan is "off". This may be accomplished by installing dampers upstream and downstream of the test unit to block the "off" period air flow. Capacity integration shall be performed during the indoor fan "on" period. Power integration shall be performed over the total period of the cycle (time-on and time-off).

During "on" time the compressor shall operate at the minimum speed unless the system controls have a preprogrammed start-up routine which causes the compressor to run at other than the minimum speed during the initial portion of the start-up period. In such a case, the compressor may follow the start-up routine during the test for the time and to the degree this would happen in the field.

The results of the cyclic test, $D_{67}^{k=1}$, shall be used in conjunction with the results of test $C_{67}^{k=1}$ to evaluate the cyclic degradation coefficient, C_D. Evaluation of C_D shall be performed by using the algorithms prescribed in section 9.2.2 of ASHRAE Standard 116-1983 [15]. An assigned value of 0.25 may be used for the degradation coefficient, C_D, in lieu of conducting tests $C_{67}^{k=1}$ and $D_{67}^{k=1}$.

3.1.3 Intermediate Speed Test

The intermediate speed test, $I_{87}^{k=i}$ (see Table 1), shall be conducted at the compressor speed evaluated by the equation:

$$RPM_i = RPM_{min} + [RPM_{max} - RPM_{min}]/3 \tag{3.3}$$

where: RPM_i = compressor speed during the intermediate speed test, (revolutions/min)

RPM_{min} = minimum compressor speed in the cooling mode, (revolutions/min)

RPM_{max} = maximum compressor speed in the cooling mode, (revolutions/min)

The compressor input frequency may be used in lieu of the compressor speed to evaluate the compressor speed for the intermediate speed test. If the system controls allow varying the compressor speed only in discrete steps such that the calculated RPM_i cannot be attained by the system, the intermediate speed test shall be performed at the next higher compressor speed available. Variations from this speed are not allowed with the exception of those associated with line frequency variations and instability of the system controls. If these variations result in a decrease in the speed below the calculated RPM_i value, the compressor speed selected for the test shall be increased.

3.1.4 Air Quantity Measurement

The air flow through the indoor and outdoor sections at any test shall be governed by system controls. The air flow shall meet the requirements of sections 5.1.3.3, 5.1.3.4, and 5.1.3.6 of ARI Standard 210/240-84 [4]. The minimum external pressure requirement (section 5.1.3.6 of [4]) applies only to tests with the maximum indoor air flow. The indoor air flow at lower than the maximum fan speed shall be determined by the equation:

$$CFM = CFM_{max} \cdot RPM_f / RPM_{f,max} \tag{3.4}$$

where: CFM = volumetric flow rate at a lower than the maximum fan speed, (ft^3/min)

CFM_{max} = volumetric flow rate at the maximum fan speed, $RPM_{f,max}$, (ft^3/min)

RPM_f = indoor fan speed during a test, (revolutions/min)

$RPM_{f,max}$ = maximum fan speed, (revolutions/min)

3.1.5 Power Measurement

The power input to a system which employs an electrical inverter shall be measured with the aid of an instrument which is accurate to within ± 1.0 percent of the quantity measured.

Due to the possibility of distorted voltage and current waveforms when an inverter is employed, an induction (rotating disc) type watthour meter may not provide the required accuracy for energy measurement for a variable speed system. The instrument which most likely can satisfy this accuracy requirement would cover a bandwidth of at least to 1 kHz. Because of the possibility of small dc-currents resulting from nonsymetrical switching, the use of current transformers may contribute additional errors.

3.2 Calculation of Seasonal Energy Efficiency Ratio, SEER

The building cooling load in bin temperature t_j, $BL(t_j)$, shall be calculated by the following equation:

$$BL(t_j) = Q_{ss}^{k=2}(95)\ \frac{t_j - 65}{1.1[95-65]} \quad (Btu/h) \tag{3.5}$$

where: 1.1 = size factor for 10% oversizing

The seasonal Energy Efficiency Ratio, SEER, shall be found by the following equation:

$$SEER = \frac{\sum_{j=1}^{8} n_j \cdot q(t_j)}{\sum_{j=1}^{8} n_j \cdot e(t_j)} \tag{3.6}$$

where: t_j = representative outdoor dry-bulb temperature for temperature bin j, (°F)

n_j = fractional cooling bin hours in bin j, per Table 2.

$q(t_j)$ = cooling done in bin temperature t_j, (Btu/h)

$e(t_j)$ = power input in bin temperature t_j, (W)

The terms $q(t_j)$ and $e(t_j)$ are evaluated at each bin temperature, shown in Table 2, by the following equations:

$$q(t_j) = X(t_j) \cdot Q(t_j) \tag{3.7}$$

$$e(t_j) = X(t_j) \cdot E(t_j)/PLF(X) \tag{3.8}$$

where: $Q(t_j)$ = system cooling capacity at bin temperature t_j at the speed at which capacity matches the building load, (Btu/h)

$E(t_j)$ = system power input at bin temperature t_j during steady state operation, (W)

$X(t_j)$ = cooling load factor, (-)

$$X(t_j) = \begin{cases} BL(t_j)/Q(t_j) & \text{if } BL(t_j)/Q(t_j) < 1 \\ 1 \text{ otherwise} & \end{cases} \tag{3.9}$$

$PLF(X)$ = part load factor, (-)

$$PLF(X) = 1 - C_D[1 - X(t_j)] \tag{3.10}$$

C_D = cyclic degradation coefficient, (-)

Quantities $Q(t_j)$ and $E(t_j)$ shall be evaluated according to three possible cases depending on compressor operation, as shown in Figure 1. These three cases can be identified in terms of the three outdoor temperature ranges or the two temperatures, t_1 and t_2, which separate them.

The outdoor temperature, t_1, at which the building load equals system capacity with the compressor operating at the minimum (k=1) speed shall be calculated by the equation:

$$t_1 = \frac{975 \cdot Q_{ss}^{k=2}(95) + 495 \cdot Q_{ss}^{k=1}(67) + 2211 \cdot [Q_{ss}^{k=1}(67) - Q_{ss}^{k=1}(82)]}{15 \cdot Q_{ss}^{k=2}(95) + 33 \cdot [Q_{ss}^{k=1}(67) - Q_{ss}^{k=1}(82)]} \tag{3.11}$$

The outdoor temperature, t_2, at which the building load equals system capacity with the compressor operating at the maximum speed (k=2) shall be calculated by the equation:

$$t_2 = \frac{845 \cdot Q_{ss}^{k=2}(95) + 429 \cdot Q_{ss}^{k=2}(82) + 2706 \cdot [Q_{ss}^{k=2}(82) - Q_{ss}^{k=2}(95)]}{33 \cdot Q_{ss}^{k=2}(82) - 20 \cdot Q_{ss}^{k=2}(95)} \qquad (3.12)$$

The equations for t_1, and t_2 were derived by equating the building load (eq. (3.5)) with system capacities at the minimum and maximum speeds, respectively (eqs. (3.13) and (3.32)).

Case I, $BL(t_j) \leq Q_{ss}^{k=1}(t_j)$, $(t_j \leq t_1)$

Capacity of the unit at the minimum compressor speed is greater than or equal to the building load. The unit cycles on and off.

$$Q(t_j) = Q_{ss}^{k=1}(67) + [Q_{ss}^{k=1}(82) - Q_{ss}^{k=1}(67)] \cdot [t_j - 67]/15 \qquad (3.13)$$

$$E(t_j) = E_{ss}^{k=1}(67) + [E_{ss}^{k=1}(82) - E_{ss}^{k=1}(67)] \cdot [t_j - 67]/15. \qquad (3.14)$$

CASE II, $Q_{ss}^{k=1}(t_j) < BL(t_j) < Q_{ss}^{k=2}(t_j)$, $(t_1 < t_j < t_2)$

The unit is able to match the building load by modulating compressor speed between the minimum (k=1) and maximum (k=2) speed.

$$Q(t_j) = BL(t_j) \qquad (3.15)$$

$$E(t_j) = \frac{Q(t_j)}{EER_{ss}^{k=v}(t_j)} \qquad (3.16)$$

where: $EER_{ss}^{k=v}(t_j) = a + b \cdot t_j + c \cdot t_j^2,$ (3.17)

steady state energy efficiency ratio at temperature t_j and at an intermediate speed at which the unit capacity matches the building load, (Btu/W•h)

a, b, c = coefficients to be calculated

The following is a procedure for evaluation of coefficients a, b, and c. The unit performance has to be evaluated first at the compressor speed (k=i) at which the intermediate speed test was conducted. The capacity of the unit at any temperature t with the compressor operating at the intermediate speed (k=i) shall be determined by:

$$Q_{ss}^{k=i}(t) = Q_{ss}^{k=i}(87) + M_Q[t - 87] \tag{3.18}$$

where: $Q_{ss}^{k=i}(87)$ = capacity of the unit at 87°F determined by the intermediate speed steady state test, (Btu/h)

$$M_Q = \frac{Q_{ss}^{k=1}(82) - Q_{ss}^{k=1}(67)}{82 - 67} \cdot (1 - N_Q) + \frac{Q_{ss}^{k=2}(95) - Q_{ss}^{k=2}(82)}{95 - 82} \cdot N_Q \tag{3.19}$$

$$N_Q = \frac{Q_{ss}^{k=i}(87) - Q_{ss}^{k=1}(87)}{Q_{ss}^{k=2}(87) - Q_{ss}^{k=1}(87)} \tag{3.20}$$

$Q_{ss}^{k=1}(87)$ and $Q_{ss}^{k=2}(87)$ shall be calculated by equations (3.9) and (3.28), respectively.

Once the equation (3.18) for $Q_{ss}^{k=i}(t)$ has been determined, the temperature at which $Q_{ss}^{k=i}(t) = BL(t)$ can be found. This temperature, designated as t_{vc}, shall be calculated by the equation:

$$t_{vc} = \frac{33 \cdot Q_{ss}^{k=i}(87) - 2871 \cdot M_Q + 65 \cdot Q_{ss}^{k=2}(95)}{Q_{ss}^{k=2}(95) - 33 \cdot M_Q} \qquad (3.21)$$

The electrical power input for the unit operating at the intermediate compressor speed (k=i) and the temperature (t_{vc}) is determined by:

$$E_{ss}^{k=i}(t_{vc}) = E_{ss}^{k=i}(87) + M_E[t_{vc} - 87] \qquad (3.22)$$

where: $E_{ss}^{k=i}(87)$ = electrical power input of the unit at 87°F determined by the intermediate speed, steady state test, (W)

$$M_E = \frac{E_{ss}^{k=1}(82) - E_{ss}^{k=1}(67)}{82 - 67} \cdot [1 - N_E] \qquad (3.23)$$

$$+ \frac{E_{ss}^{k=2}(95) - E_{ss}^{k=2}(82)}{95 - 82} \cdot N_E$$

$$N_E = \frac{E_{ss}^{k=i}(87) - E_{ss}^{k=1}(87)}{E_{ss}^{k=2}(87) - E_{ss}^{k=1}(87)} \qquad (3.24)$$

$E_{ss}^{k=1}(87)$ and $E_{ss}^{k=2}(87)$ shall be calculated by equations (3.14) and (3.33), respectively.

The energy efficiency ratio of the unit, $EER_{ss}^{k=i}(t_{vc})$, at the speed (k=i) and temperature t_{vc} shall be calculated by the equation:

$$EER_{ss}^{k=i}(t_{vc}) = \frac{Q_{ss}^{k=i}(t_{vc})}{E_{ss}^{k=i}(t_{vc})} \tag{3.25}$$

Similarly, energy efficiency ratios at temperatures, t_1 and t_2 shall be calculated by the equations:

$$EER_{ss}^{k=1}(t_1) = \frac{Q_{ss}^{k=1}(t_1)}{E_{ss}^{k=1}(t_1)} \tag{3.26}$$

$$EER_{ss}^{k=2}(t_2) = \frac{Q_{ss}^{k=2}(t_2)}{E_{ss}^{k=2}(t_2)} \tag{3.27}$$

where: $EER_{ss}^{k=1}(t_1)$ = steady state energy efficiency ratio at the minimum compressor speed at temperature t_1, (Btu/W•h)

$EER_{ss}^{k=2}(t_2)$ = steady state energy efficiency radio at the maximum compressor speed at temperature t_2, (Btu/W•h)

$Q_{ss}^{k=1}(t_1)$ = $Q(t_1)$, steady state capacity at the minimum compressor speed at temperature t_1, calculated by eq. (3.13), (Btu/h)

$Q_{ss}^{k=2}(t_2)$ = $Q(t_2)$, steady state capacity at the maximum compressor speed at temperature t_2, calculated by eq. (3.32), (Btu/h)

$E_{ss}^{k=1}(t_1)$ = $E(t_1)$, electrical power input at the minimum compressor speed at temperature t_1, calculated by eq. (3.14), (W)

$E_{ss}^{k=2}(t_2)$ = $E(t_2)$, electrical power input at the maximum compressor speed at temperature t_2, calculated by eq. (3.33), (W)

Finally, coefficients a, b, and c shall be calculated using equations (3.28) through (3.31):

$$d = \frac{t_2^2 - t_1^2}{t_{vc}^2 - t_1^2} \tag{3.28}$$

$$b = \frac{EER_{ss}^{k=1}(t_1) - EER_{ss}^{k=2}(t_2) - d \cdot [EER_{ss}^{k=1}(t_1) - EER_{ss}^{k=i}(t_{vc})]}{t_1 - t_2 - d \cdot [t_1 - t_{vc}]} \tag{3.29}$$

$$c = \frac{EER_{ss}^{k=1}(t_1) - EER_{ss}^{k=2}(t_2) - b \cdot [t_1 - t_2]}{t_1^2 - t_2^2} \tag{3.30}$$

$$a = EER_{ss}^{k=2}(t_2) - b \cdot t_2 - c \cdot t_2^2 \tag{3.31}$$

CASE III, $BL(t_j) \geq Q_{ss}^{k=2}(t_j)$, $(t_j \geq t_2)$

Capacity of the unit at the maximum (k=2) compressor speed is equal to or smaller than the building load.

$$Q(t_j) = Q_{ss}^{k=2}(82) + [Q_{ss}^{k=2}(95) - Q_{ss}^{k=2}(82)][t_j - 82]/13 \quad (3.32)$$

$$E(t_j) = E_{ss}^{k=2}(82) + [E_{ss}^{k=2}(95) - E_{ss}^{k=2}(82)][t_j - 82]/13 \quad (3.33)$$

4. RATING PROCEDURE FOR VARIABLE SPEED SYSTEMS IN THE HEATING MODE

4.1 Requirements for Testing Conditions and Testing Procedures

Requirements regarding instruments and data acquisition systems, test apparatus, methods of test, test procedures, and data analysis are as prescribed in ASHRAE Standard 116-1983 [15] with the exception to the provisions specified in the following sections.

4.1.1 Tests and Testing Conditions

The tests and test conditions are presented in Table 3.

4.1.2 Cyclic Test

The cyclic test shall be conducted according to provisions specified in section 3.1.2 of this document and with the compressor time-on and time-off specified for the cyclic test in the cooling mode. Results of the heating mode cyclic test shall be used with results of the maximum temperature heating test to evaluate the cyclic degradation coefficient, C_D, as specified in section 9.2.4 of ASHRAE Standard 116-1983 [15]. An assigned value of 0.25 may be used for the degradation coefficient, C_D, in lieu of conducting the heating cyclic test.

4.1.3 Intermediate Speed Frost Accumulation Test

The compressor speed shall be the same as during the intermediate speed test in the cooling mode. Compressor speed variations are allowed as specified in section 3.1.3. Also, variations associated with control strategies (if employed) during the initial period after switching from defrosting to heating, in the final period of the heating cycle, and during defrosting are permitted. Variations of the compressor speed from the prescribed speed are allowed to the same extent and for the same period of time as such variation would occur in the field due to control strategy.

4.1.4 Maximum Speed Frost Accumulation Test

In lieu of conducting the test the following equations may be used to obtain heat pump capacity and power at the test conditions:

$$Q_{def}^{k=2}(35) = 0.90 \cdot [Q_{def}^{k=2}(17) + [Q_{def}^{k=2}(47)-Q_{def}^{k=2}(17)][35-17]/[47-17]] \quad (4.1)$$

$$E_{def}^{k=2}(35) = 0.985 \cdot [E_{def}^{k=2}(17) + [E_{def}^{k=2}(47)-E_{def}^{k=2}(17)][35-17]/[47-17]] \quad (4.2)$$

4.1.5 Nominal Speed High Temperature Test

This test is applicable if the maximum compressor speed in the heating mode is greater than the maximum speed in the cooling mode. In such a case the nominal speed is the maximum speed the compressor may run in the cooling mode. The heating capacity obtained during the nominal speed high temperature test shall be used only to calculate design heating requirements. This test is optional. If this test is not opted to be performed or is not applicable, capacity obtained during the maximum speed high temperature heating test shall be used for design heating requirement calculations.

4.1.6 Air Quantity Measurement

Air flow quantity and measurement method shall comply with provisions specified in section 3.1.4.

4.1.7 Power Measurement

Power measurement shall be performed as prescribed for the cooling mode in section 3.1.5.

4.2 Calculation of Heating Seasonal Performance Factor, HSPF

The Heating Seasonal Performance Factor, HSPF, shall be calculated by the following equation:

$$HSPF = \frac{\sum_{j} n_j \cdot BL(t_j)}{\sum_{j} n_j \cdot [e(t_j) + RH(t_j)]} \qquad (4.3)$$

where: t_j = representative outdoor dry-bulb temperature for temperature bin j, (°F)

n_j = fractional heating bin hours in bin j, per Table 4

$BL(t_j)$ = building load at bin temperature t_j, (Btu/h)

$e(t_j)$ = power input to the system (excluding the supplemental resistance heater) operating in temperature t_j, (W)

$RH(t_j)$ = power input to the supplementary resistance heater required in those cases when the heat pump automatically turns off ($t_j < t_{on}$), or when it is needed to meet the balance of the building heating requirement, (W)

The building load, $BL(t_j)$, shall be calculated by the following equation:

$$BL(t_j) = \frac{65 - t_j}{65 - T_D} \cdot C \cdot DHR \tag{4.4}$$

where: C = 0.77, an experience factor which improves the agreement between calculated and measured building loads, (-)

T_D = outdoor design temperature (shown for different regions in Table 4), (°F)

DHR = design heating requirement, (Btu/h)

The minimum and maximum design heating requirements, DHR_{min} and DHR_{max}, of a residence in which a heat pump is likely to be installed shall be obtained by evaluating the following two equations and rounding off the results to the nearest standardized values given in Table 5.

$$DHR_{min} = \begin{cases} Q_{ss}^{k=n}(47) \cdot [65 - T_D]/60, & \text{for regions I, II, III, IV, and VI} \\ Q_{ss}^{k=n}(47), & \text{for region V} \end{cases} \tag{4.5}$$

$$DHR_{max} = \begin{cases} 2 \cdot Q_{ss}^{k=n}(47)[65 - T_D]/60, & \text{for regions I, II, III, IV and VI} \\ 2.2 \cdot Q_{ss}^{k=n}(47), & \text{for region V} \end{cases} \tag{4.6}$$

where:

$$Q_{ss}^{k=n}(47) = \begin{cases} \text{system capacity during the optional nominal speed high temperature test} \\ Q_{ss}^{k=2}(47) \text{ if the nominal speed high temperature test was not performed, (Btu/h)} \end{cases}$$

The quantities $e(t_j)$ and $RH(t_j)$ shall be calculated by the following equations:

$$e(t_j) = \frac{\delta(t_j) \cdot X(t_j) \cdot E(t_j)}{PLF(X)} \tag{4.7}$$

$$RH(t_j) = \frac{BL(t_j) - \delta(t_j) \cdot X(t_j) \cdot Q(t_j)}{3.413} \tag{4.8}$$

where: $\delta(t_j)$ = heat pump low temperature cut-off factor, (-)

$$\delta(t_j) = \begin{cases} 0 \text{ if } t_j \le t_{off} \text{ or } Q(t_j)/(3.413 \cdot E(t_j)) < 1 \\ 0.5 \text{ if } t_{off} < t_j \le t_{on} \text{ and } Q(t_j)/(3.413 \cdot E(t_j)) \ge 1 \\ 1 \text{ if } t_j > t_{on} \text{ and } Q(t_j)/(3.413 \cdot E(t_j)) \ge 1 \end{cases} \tag{4.9}$$

t_{off} = outdoor temperature at which the compressor is automatically stopped to avoid operation at low temperatures

t_{on} = outdoor temperature at which the compressor is automatically turned on after low-temperature shut-off

$X(t_j)$ = heating load factor, (-)

$$X(t_j) = \begin{cases} BL(t_j)/Q(t_j) \text{ if } BL(t_j)/Q(t_j) < 1 \\ 1 \quad \text{otherwise} \end{cases} \tag{4.10}$$

$PLF(X)$ = part load factor

$$PLF(X) = 1 - C_D[1 - X(t_j)] \tag{4.11}$$

C_D = cyclic degradation coefficient, (-)

$Q(t_j)$ = system capacity at temperature t_j during continuous compressor operation at the speed needed to match the building load, (Btu/h)

$E(t_j)$ = power input to the system at temperature t_j (excluding the supplemental resistance heater) during continuous compressor operation at the speed needed to match the building load, (W)

Quantities $Q(t_j)$ and $E(t_j)$ shall be calculated according to three possible cases depending on compressor operation, as shown in Figure 10. These three cases can be identified in terms of the three outdoor temperature ranges or the two temperatures, t_3 and t_4, which separate them.

The outdoor temperature, t_3, at which the building load equals system capacity with the compressor operating at the minimum (k=1) speed shall be calculated by the equation:

$$t_3 = \frac{975 \cdot C \cdot DHR + 47 \cdot [65-T_D][Q_{ss}^{k=1}(62)-Q_{ss}^{k=1}(47)] - 15 \cdot Q_{ss}^{k=1}(47)[65-T_D]}{15 \cdot C \cdot DHR + [65-T_D][Q_{ss}^{k=1}(62)-Q_{ss}^{k=1}(47)]} \tag{4.12}$$

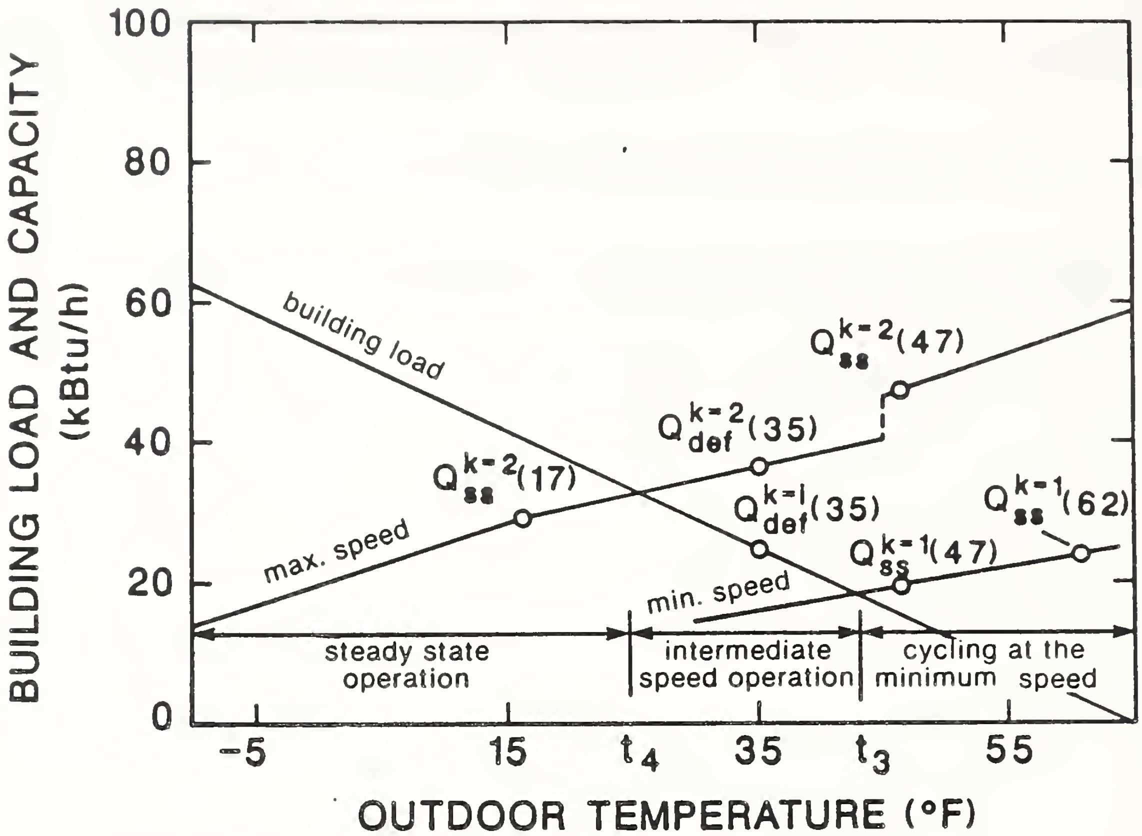


Figure 10. Building heating load and capacity of a variable speed heat pump.

The outdoor temperature, t_4, at which the building load equals system capacity with the compressor operating at the maximum speed (k=2) shall be calculated by either equation (4.13) or (4.14), as appropriate:

If the calculated value for t_4 is smaller than or equal to 17:

$$t_4 = \frac{1950 \cdot C \cdot DHR + 17[65-T_D][Q_{ss}^{k=2}(47) - Q_{ss}^{k=2}(17)] - 30 \cdot Q_{ss}^{k=2}(17)[65-T_D]}{30 \cdot C \cdot DHR + [65-T_D][Q_{ss}^{k=2}(47) - Q_{ss}^{k=2}(17)]} \tag{4.13}$$

If the calculated value for t_4 is greater than 17, t_4 shall be calculated by the following equation:

$$t_4 = \frac{1170 \cdot C \cdot DHR + 17[65-T_D][Q_{def}^{k=2}(35)-Q_{ss}^{k=2}(17)] - 18\ Q_{ss}^{k=2}(17)[65-T_D]}{18 \cdot C \cdot DHR + [65-T_D][Q_{def}^{k=2}(35) - Q_{ss}^{k=2}(17)]} \tag{4.14}$$

The equations for t_3 and t_4 were derived by equating the building load (eq. (4.4)) with respective system capacities and the minimum and maximum speeds (eq. (4.15) and eq. (4.34) or (4.36)).

CASE I, $Q_{ss}^{k=1}(t_j) \geq BL(t_j)$, $(t_j \geq t_3)$

Capacity of the unit at the minimum compressor speed (k=1) is greater than or equal to the building load. The unit cycles on and off.

$$Q(t_j) = Q_{ss}^{k=1}(47) + [Q_{ss}^{k=1}(62) - Q_{ss}^{k=1}(47)][t_j - 47]/15 \tag{4.15}$$

$$E(t_j) = E_{ss}^{k=1}(47) + [E_{ss}^{k=1}(62) - E_{ss}^{k=1}(47)][t_j - 47]/15. \tag{4.16}$$

CASE II, $Q_{ss}^{k=1}(t_j) < BL(t_j) < Q^{k=2}(t_j)$, $(t_4 < t_j < t_3)$

The unit is able to match the building load by modulating compressor speed between the minimum (k=1) and maximum (k=2) speed.

$$Q(t_j) = BL(t_j) \tag{4.17}$$

$$E(t_j) = Q(t_j)/(3.413 \cdot COP^{k=v}(t_j)) \tag{4.18}$$

where: $COP^{k=v}(t_j)$ = coefficient of performance at an intermediate speed at which the unit delivers capacity matching the building load at temperature t_j

$$COP^{k=v}(t_j) = a + b \cdot t_j + c \cdot t_j^2 \tag{4.19}$$

a, b, c = coefficients which have to be calculated separately for each design building requirement

The following is a procedure for evaluation of coefficients a, b, and c.

Before the coefficient of performance, $COP^{k=v}(t_j)$, can be calculated, the unit performance has to be evaluated at the compressor speed (k=i) at which the intermediate speed test was conducted. The capacity of the unit at any temperature t when the compressor operates at the intermediate speed (k=i) shall be determined by:

$$Q_{def}^{k=i}(t) = Q_{def}^{k=i}(35) + M_Q[t - 35] \tag{4.20}$$

where: $Q_{def}^{k=i}(35)$ = capacity of the unit at 35°F determined at the intermediate compressor speed (k=i) in the frost accumulation test, (Btu/h)

$$M_Q = \frac{Q_{ss}^{k=1}(62) - Q_{ss}^{k=1}(47)}{62 - 47} \cdot (1 - N_Q) \tag{4.21}$$

$$+ \frac{Q_{def}^{k=2}(35) - Q_{ss}^{k=2}(17)}{35 - 17} \cdot N_Q$$

$$N_Q = \frac{Q_{def}^{k=i}(35) - Q^{k=1}(35)}{Q_{def}^{k=2}(35) - Q^{k=1}(35)} \tag{4.22}$$

$Q^{k=1}(35) = Q(35)$, capacity of the unit at 35°F at the minimum compressor speed, calculated by equation (4.15), (Btu/h)

Once the equation for $Q_{def}^{k=i}(t)$ has been determined, the temperature at which $Q_{def}^{k=i}(t) = BL(t)$ can be found. This temperature, designated as t_{vh}, shall be calculated by the equation:

$$t_{vh} = \frac{65 \circ C \circ DHR + [65 - T_D][35 \circ M_Q - Q_{def}^{k=i}(35)]}{C \circ DHR + M_Q[65 - T_D]} \qquad (4.23)$$

A separate t_{vh} shall be determined for each design heating requirement.

The electrical power input for the unit operating at the intermediate compressor speed (k=i) and at the temperature t_{vh} shall be determined by:

$$E_{def}^{k=i}(t_{vh}) = E_{def}^{k=i}(35) + M_E[t_{vh} - 35] \qquad (4.24)$$

where: $E_{def}^{k=i}(35)$ = electrical power input of the unit at 35°F determined at the intermediate compressor speed (k=i) in the frost accumulation test, (W)

$$M_E = \frac{E_{ss}^{k=1}(62) - E_{ss}^{k=1}(47)}{62 - 47} \circ (1 - N_E) \qquad (4.25)$$

$$+ \frac{E_{def}^{k=2}(35) - E_{ss}^{k=2}(17)}{35 - 17} \circ N_E$$

$$N_E = \frac{E_{def}^{k=i}(35) - E^{k=1}(35)}{E_{def}^{k=i}(35) - E^{k=1}(35)} \tag{4.26}$$

The coefficient of performance, $COP_{def}^{k=i}(t_{vh})$, at the intermediate speed (k=i) and temperature t_{vh} shall be calculated by the equation:

$$COP_{def}^{k=i}(t_{vh}) = \frac{Q_{def}^{k=i}(t_{vh})}{3.413 \cdot E_{def}^{k=i}(t_{vh})} \tag{4.27}$$

Similarly, coefficients of performance at temperature t_3 and t_4 shall be calculated by the equations:

$$COP^{k=1}(t_3) = \frac{Q^{k=1}(t_3)}{3.413 \cdot E^{k=1}(t_3)} \tag{4.28}$$

$$COP^{k=2}(t_4) = \frac{Q^{k=2}(t_4)}{3.413 \cdot E^{k=2}(t_4)} \tag{4.29}$$

where: $COP^{k=1}(t_3)$ = coefficient of performance at the minimum compressor speed at temperature t_3, (-)

$COP^{k=2}(t_4)$ = coefficient of performance at the maximum compressor speed at temperature t_4, (-)

$Q^{k=1}(t_3)$ = $Q(t_3)$, system capacity at the minimum compressor speed at temperature t_3, calculated by eq. (4.15), (Btu/h)

$Q^{k=2}(t_4) = Q(t_4)$, system capacity at the maximum compressor speed at temperature t_4, calculated by eq. (4.34) for $t_4 > 17$, or by eq. (4.36) for $t_4 \le 17$, (Btu/h)

$E^{k=1}(t_3) = E(t_3)$, power input at the minimum compressor speed at temperature t_3, calculated by eq. (4.16), (W)

$E^{k=2}(t_4) = E(t_4)$, power input at the maximum compressor speed at temperature t_4, calculated by eq. (4.35) for $t_4 > 17$, or by eq. (4.37) for $t_4 \le 17$, (W)

Finally, coefficients a, b, and c shall be calculated using equations (4.30) through (4.33):

$$d = \frac{t_3^2 - t_4^2}{t_{vh}^2 - t_4^2} \tag{4.30}$$

$$b = \frac{COP^{k=2}(t_4) - COP^{k=1}(t_3) - d \cdot [COP^{k=2}(t_4) - COP_{def}^{k=i}(t_{vh})]}{t_4 - t_3 - d \cdot [t_4 - t_{vh}]} \tag{4.31}$$

$$c = \frac{COP^{k=2}(t_4) - COP^{k=1}(t_3) - b \cdot [t_4 - t_3]}{t_4^2 - t_3^2} \tag{4.32}$$

$$a = COP^{k=2}(t_4) - b \cdot t_4 - c \cdot t_4^2 \tag{4.33}$$

CASE III, $BL(t_j) \geq Q^{k=2}(t_j)$, $(t_j \leq t_4)$

Capacity of the system at the maximum (k=2) compressor speed is equal to or less than the building load. Evaluation of $Q(t_j)$ and $E(t_j)$ depends on the value of t_j as prescribed in equations (4.34) through (4.37).

If $t_j \geq 17°F$

$$Q(t_j) = Q_{ss}^{k=2}(17) + [Q_{def}^{k=2}(35) - Q_{ss}^{k=2}(17)][t_j - 17]/18 \quad (4.34)$$

$$E(t_j) = E_{ss}^{k=2}(17) = [E_{ss}^{k=2}(35) - E_{ss}^{k=2}(17)][t_j - 17]/18 \quad (4.35)$$

If $t_j < 17°F$

$$Q(t_j) = Q_{ss}^{k=2}(17) + [Q_{ss}^{k=2}(47) - Q_{ss}^{k=2}(17)][t_j - 17]/30 \quad (4.36)$$

$$E(t_j) = E_{ss}^{k=2}(17) + [E_{ss}^{k=2}(47) - E_{ss}^{k=2}(17)][t_j - 17]/30 \quad (4.37)$$

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APPENDIX A. EXAMPLE OF SEER CALCULATION

This appendix contains an example of calculations required for determination of the Seasonal Energy Efficiency Ratio, SEER.

Table A1 presents a summary of calculations performed for each temperature bin. Calculation steps required for the 92°F temperature bin are shown in detail below.

The unit performance data needed by the calculation procedure are listed below:

$Q_{ss}^{k=2}(95) = 27000$ Btu/h	$E_{ss}^{k=2}(95) = 3288$ W
$Q_{ss}^{k=2}(82) = 28397$ Btu/h	$E_{ss}^{k=2}(82) = 3082$ W
$Q_{ss}^{k=1}(82) = 13130$ Btu/h	$E_{ss}^{k=1}(82) = 958$ W
$Q_{ss}^{k=1}(67) = 14492$ Btu/h	$E_{ss}^{k=1}(67) = 859$ W
$Q_{ss}^{k=i}(87) = 17888$ Btu/h	$E_{ss}^{k=i}(87) = 1450$ W

$C_D = 0.2$

Step 1. Determine the minimum and maximum speed balance points, t_1 and t_2.

$$t_1 = \frac{975 \cdot 27000 + 495 \cdot 14492 + 2211 \cdot (14492-13130)}{15 \cdot 27000 + 33 \cdot (14492-13130)} = 81.14°F \qquad \text{(eq. (3.11))}$$

$$t_2 = \frac{845 \cdot 27000 + 429 \cdot 28397 + 2706 \cdot (28397-27000)}{33 \cdot 28397 - 20 \cdot 27000} = 97.65°F \qquad \text{(eq. (3.12))}$$

Since $t_j = 92$, $t_1 < t_j < t_2$, Case II applies

Step 2. Calculate t_{vc}.

$$Q_{ss}^{k=1}(87) = 14492 + (13130-14492)(87-67)/15 = 12676.0 \quad \text{Btu/h} \qquad \text{(eq. (3.13))}$$

$$Q_{ss}^{k=2}(87) = 28397 + (27000-28397)(87-82)/13 = 27859.7 \quad \text{Btu/h} \qquad \text{(eq. (3.32))}$$

$$N_Q = \frac{17888 - 12676}{27859.7 - 12676} = 0.34326 \qquad \text{(eq. (3.20))}$$

$$M_Q = \frac{13130 - 14492}{82 - 67} \circ (1-034326) +$$

$$\frac{27000 - 28397}{95 - 82} \circ 0.34326 = -96.519 \qquad \text{(eq. (3.19))}$$

$$t_{vc} = \frac{33 \circ 17888 - 2871 \circ (-96.519) + 65 \circ 27000}{27000 - 33 \circ (-96.519)} = 86.88°\text{F} \qquad \text{(eq. (3.21))}$$

Step 3. Calculate power at temperature t_{vc}.

$$E_{ss}^{k=1}(87) = 859 + (958-859)(87-67)/15 = 991.0 \quad \text{W} \qquad \text{(eq. (3.14))}$$

$$E_{ss}^{k=2}(87) = 3082 + (3288-3082)(87-82)/13 = 3161.2 \text{ W} \qquad \text{(eq. (3.33))}$$

$$N_E = \frac{1450 - 991}{3161.2 - 991} = 0.21150 \qquad \text{(eq. (3.24))}$$

$$M_E = \frac{958 - 859}{82 - 67} (1.-0.21150) + \frac{3288 - 3082}{95 - 82} \circ 0.21150 = 8.556 \qquad \text{(eq. (3.23))}$$

$$E_{ss}^{k=i}(86.88) = 1450 - 8.556 \circ (86.88-87.0) = 1451.0 \quad \text{W} \qquad \text{(eq. (3.22))}$$

Step 4. Calculate EER at temperatures t_1=81.14, t_{vc}=86.88, and t_2=97.65 .

$Q_{ss}^{k=1}(81.14) = 14492 + (13130-14492)(81.14-67)/15 = 13208.1$ Btu/h (eq. (3.13))

$E_{ss}^{k=1}(81.14) = 859 + (958-859)(81.1-67.0)/15 = 952.1$ W (eq. (3.14))

$EER_{ss}^{k=1}(81.14) = 13208.1/952.1 = 13.87$ Btu/(W•h) (eq. (3.26))

$Q_{ss}^{k=i}(86.88) = BL(86.88) = 27000 \circ (86.88-65)/(1.1 \circ 30) = 17901.8$ Btu/h (eq. (3.5))

$E_{ss}^{k=i}(86.88) = 1451.0$ W (step 3)

$$EER_{ss}^{k=i}(86.88) = \frac{17901.8}{1451.0} = 12.34 \text{ Btu/(W•h)} \qquad \text{(eq. (3.25))}$$

$Q_{ss}^{k=2}(97.65) = 28397 + (27000-28397)(97.65-82)/13 = 26715.2$ Btu/h (eq. (3.32))

$E_{ss}^{k=2}(97.65) = 3082 + (3288-3082)(97.65-82)/13 = 3330.0$ W (eq. (3.33))

$EER_{ss}^{k=2}(97.65) = 26715.2/3330.0 = 8.02$ Btu/(W•h) (eq. (3.27))

Step 5. Evaluate parameters for a parabolic fit.

$$d = (97.65^2 - 81.14^2)/(86.88^2 - 81.14^2) = 3.0607 \qquad \text{(eq. (3.28))}$$

$$b = \frac{13.87 - 8.02 - 3.0607 \bullet (13.87-12.34)}{81.14 - 97.65 - 3.0607 \bullet (81.14-86.88)} = 1.1027 \qquad \text{(eq. (3.29))}$$

$$c = \frac{13.87 - 8.02 - 1.1027 \bullet (81.14-97.65)}{81.14^2 - 97.65^2} = -8.1494 \bullet 10^{-3} \qquad \text{(eq. (3.30))}$$

$$a = 8.02 - 1.1027 \circ 97.65 - (-8.1494 \bullet 10^{-3}) \circ 97.65^2 = -21.950 \qquad \text{(eq. (3.31))}$$

Step 6. Calculate capacity and energy input at 92°F temperature.

$$Q(92) = BL(92) = 27000\ (92-65)/33 = 22090.9 \quad \text{Btu/h} \qquad \text{(eq. (3.5))}$$

$$EER_{ss}^{k=v}(92) = -21.950 + 1.1027 \cdot 92 - 8.1494 \cdot 10^{-3} \cdot 92^2 = 10.52 \quad \text{Btu/(W·h)} \qquad \text{(eq. (3.17))}$$

$$E(92) = \frac{22090.9}{10.52} = 2099.9 \quad \text{W} \qquad \text{(eq. (3.16))}$$

Step 7. Calculate seasonally weighted capacity and energy input at 92°F temperature bin.

$$X(92) = 1 \qquad \text{(eq. (3.9))}$$

$$PLF(1) = 1 \qquad \text{(eq. (3.10))}$$

$$q(92) = 1 \cdot 22090.9 = 22090.9 \quad \text{Btu/h} \qquad \text{(eq. (3.7))}$$

$$n_6 = 0.052 \qquad \text{(Table 2)}$$

$$n_6 \cdot q(92) = 0.052 \cdot 22090.9 = 1148.7 \quad \text{Btu/h} \qquad \text{(eq. (3.6))}$$

$$e(92) = 1 \cdot 2099.9/1 = 2099.9 \quad \text{W} \qquad \text{(eq. (3.8))}$$

$$n_6 \cdot e(92) = 0.052 \cdot 2099.9 = 109.2 \quad \text{W} \qquad \text{(eq. (3.6))}$$

It should be noted that some values in the above example differ from the respective values shown in Table A1 due to smaller number of carried digits and the round-off error of hand-done calculations.

While performing calculations for all temperature bins, Steps 1 through 5 have to be followed one time. Once coefficients a, b and c are evaluated, calculations for Case II start at Step 6.

APPENDIX B. EXAMPLE OF HSPF CALCULATIONS

Example of HSPF Calculations.

This section contains an example of calculations required for determination of the Heating Seasonal Performance Factor, HSPF. The example calculations were performed for the minimum design heating requirement in region IV. The following are the unit performance data needed for calculating HSPF:

$Q_{ss}^{k=2}(47) = 30838$ Btu/h	$E_{ss}^{k=2}(47) = 3236$ W
$Q_{def}^{k=2}(35) = 22930$ Btu/h	$E_{def}^{k=2}(35) = 2900$ W
$Q_{ss}^{k=2}(17) = 17112$ Btu/h	$E_{ss}^{k=2}(17) = 2544$ W
$Q_{ss}^{k=1}(62) = 13274$ Btu/h	$E_{ss}^{k=1}(62) = 1101$ W
$Q_{ss}^{k=1}(47) = 10355$ Btu/h	$E_{ss}^{k=1}(47) = 960$ W
$Q_{def}^{k=i}(35) = 10233$ Btu/h	$E_{def}^{k=n}(35) = 1106$ W
$Q_{ss}^{k=n}(47) = 25670$ Btu/h	

$C_D = 0.25$

$t_{off} = 0°F$ $\qquad t_{on} = 10°F$

First, the minimum and maximum heating requirements have to be determined. Since the nominal speed test was performed, capacity from this test, $Q_{ss}^{k=n}(47)$, will be used in calculations. For region IV after rounding off to the standardized values we obtain:

$$DHR_{min} = 25670 \cdot (65 - 5)/60 = 25670 \approx 25000 \text{ Btu/h} \qquad \text{(eq. (4.5) and Table 5)}$$

$$DHR_{max} = 50000 \text{ Btu/h} \qquad \text{(eq. (4.6) and Table 5)}$$

Table B1 presents a summary of calculations performed for each temperature bin for minimum heating design requirement. Calculation steps required for 27°F temperature bin are shown in detail below.

Step 1. Determine the minimum and maximum speed balance points, t_3, and t_4.

$$t_3 = \frac{975 \cdot 0.77 \cdot 25000 + 47 \cdot (65-5)(13274-10355) - 15 \cdot 10355 \cdot (65-5)}{15 \cdot 0.77 \cdot 25000 + (65-5)(13274-10355)}$$

$$t_3 = 38.11°F \qquad \text{(eq. (4.12))}$$

$$t_4 = \frac{1950 \cdot 0.77 \cdot 25000 + 17(65-5)(30838-17112) - 30 \cdot 17112 \cdot (65-5)}{30 \cdot 0.77 \cdot 25000 + (65-5)(30838-17112)}$$

$$t_4 = 14.80°F \qquad \text{(eq. (4.13))}$$

Since calculated t_4 is smaller than 17, the applied equation is the right one. If obtained value were greater than 17, equation (4.14) would be used.

Since $t_4 < t_j = 27 < t_3$ we have Case II.

Step 2. Evaluate t_{vh}.

$$Q^{k=1}(35) = 10355 + (13274-10355)(35-47)/15 = 8019.8 \ \text{Btu/h} \qquad \text{(eq. (4.15))}$$

$$N_Q = (10233-8019.8)/(22930-8019.8) = 0.14844 \qquad \text{(eq. (4.22))}$$

$$M_Q = (13274-10355)(1-0.14844)/15 + (22930-17112) \cdot 0.14844/18 = 213.69 \qquad \text{(eq. (4.21))}$$

$$t_{vh} = \frac{65 \cdot 0.77 \cdot 25000 + (65-5)(35 \cdot 213.69-10233)}{0.77 \cdot 25000 + 213.69 \cdot (65-5)} = 33.86°F \qquad \text{(eq. (4.23))}$$

Step 3. Calculate power at temperature t_{vh}

$$E^{k=1}(35) = 960 + (1101-960)(35-47)/15 = 847.2 \ \ W \qquad \text{(eq. (4.16))}$$

$$N_E = (1106-847.2)/(2900-847.2) = 0.12607 \qquad \text{(eq. (4.26))}$$

$$M_E = (1101-960)(1-0.12607)/15 + (2900-2544) \cdot 0.12607/18 = 10.708 \qquad \text{(eq. (4.25))}$$

$$E_{def}^{k=i}(33.86) = 1106 + 10.708 \cdot (33.86-35) = 1093.8 \ \ W \qquad \text{(eq. (4.24))}$$

Step 4. Calculate COP at temperature t_3=38.11, t_{vh}=33.86, and t_4=14.8 .

$$Q^{k=1}(38.11) = BL(38.11) = (65-38.11) \cdot 0.77 \cdot 25000/(65-5) = 8627.2 \ \ Btu/h \qquad \text{(eq. (4.4))}$$

$$E^{k=1}(38.11) = 960 + (1101-960)(38.11-47)/15 = 876.4 \ \ W \qquad \text{(eq. (4.16))}$$

$$COP^{k=1}(38.11) = \frac{8625.0}{3.413 \cdot 876.4} = 2.883 \qquad \text{(eq. (4.28))}$$

$$Q_{def}^{k=i}(33.86) = BL(33.86) = (65-33.86) \cdot 0.77 \cdot 25000/(65-5) = 9990.8 \ \ Btu/h \qquad \text{(eq. (4.4))}$$

$$COP_{def}^{k=i}(33.86) = \frac{9990.8}{3.413 \cdot 1093.8} = 2.676 \qquad \text{(eq. (4.27))}$$

$$Q^{k=2}(14.80) = BL(14.80) = (65-14.80) \cdot 0.77 \cdot 25000/(65-5) = 16105.8 \ Btu/h \qquad \text{(eq. (4.4))}$$

$$E^{k=2}(14.80) = 2544 + (3236-2544)(14.80-17)/30 = 2493.3 \ \ W \qquad \text{(eq. (4.37))}$$

(If t_4 were greater than 17, equation (4.35) would be used)

$$COP^{k=2}(14.80) = \frac{16105.8}{3.413 \cdot 2493.3} = 1.893 \qquad \text{(eq. (4.29))}$$

Step 5. Evaluate parameters for a parabolic fit.

$$d = (38.11^2 - 14.80^2)/(33.86^2 - 14.80^2) = 1.3298 \qquad \text{(eq. (4.30))}$$

$$b = \frac{1.893 - 2.883 - 1.3298 \cdot (1.893 - 2.676)}{14.80 - 38.11 - 1.3298 \cdot (14.80 - 33.86)} = 2.5164 \cdot 10^{-2} \qquad \text{(eq. (4.31))}$$

$$c = \frac{1.893 - 2.883 - 2.5164 \cdot 10^{-2} \cdot (14.80 - 38.11)}{14.80^2 - 38.11^2} = 3.2711 \cdot 10^{-4} \qquad \text{(eq. (4.32))}$$

$$a = 1.893 - 2.5164 \cdot 10^{-2} \cdot 14.8 - 3.2711 \cdot 10^{-4} \cdot 14.8^2 = 1.4489 \qquad \text{(eq. (4.33))}$$

$$Q(27) = BL(27) = (65-27) \cdot 0.77 \cdot 25000/(65-5) = 12191.7 \text{ Btu/h} \qquad \text{(eq. (4.4))}$$

$$COP^{k=v}(27) = 1.4489 + 2.5164 \cdot 10^{-2} \cdot 27 + 3.2711 \cdot 10^{-4} \cdot 27^2 = 2.367 \qquad \text{(eq. (4.19))}$$

$$E(27) = 12191.7/(3.413 \cdot 2.367) = 1509.1 \text{ W} \qquad \text{(eq. (4.18))}$$

Step 7. Calculate seasonally weighted capacity, the energy input to the heat pump, and energy input to the electric heater.

$$BL(27) = (65-27) \cdot 0.77 \cdot 25000/(65-5) = 12191.7 \text{ Btu/h} \qquad \text{(eq. (4.24))}$$

$$n_8 = 0.087 \qquad \text{(Table 4)}$$

$$n_8 \cdot BL(27) = 0.087 \cdot 12191.7 = 1060.7 \text{ Btu/h} \qquad \text{(eq. (4.3))}$$

$$\delta(27) = 1 \qquad \text{(eq. (4.9))}$$

$X(27) = 1$ (eq. (4.10))

$PLF(27) = 1 - 0.25 \cdot (1-1) = 1$ (eq. (4.11))

$e(27) = 1 \cdot 1 \cdot 1509.1/1 = 1509.1$ W (eq. (4.7))

$n_8 \cdot e(27) = 0.087 \cdot 1509.1 = 131.3$ W (eq. (4.3))

$RH(27) = (12191.7 - 1 \cdot 1 \cdot 12191.7)/3.413 = 0$ W (eq. (4.8))

$n_8 \cdot RH(27) = 0$

It should be noted that calculations steps related to evaluation of the minimum and maximum speed balance points, and evaluation of coefficients for the parabolic fit of COP in the intermediate speed range (Steps 1 through 5) have to be performed one time for each design heating requirement and climate region for which seasonal performance of a heat pump is evaluated. Once the parabolic fit is determined for a given region and DHR, calculations for succeeding temperature bins in the intermediate speed range include only steps 6 and 7.

Table 1
Summary of Test Conditions for Rating Variable Speed Systems
in the Cooling Mode.

Test	Compressor Operation (1)		Outdoor Temperature				Indoor Temperature				Quantity obtained and used in calculation procedure	
			dry - bulb		wet - bulb (2)		dry - bulb		wet - bulb			
	Speed	Mode	°F	°C	°F	°C	°F	°C	°F	°C	capacity (Btu/h)	power input (W)
$A^{k=2}_{95}$	max	steady state	95.0	35.0	75.0	23.9	80.0	26.7	67.0	19.4	$Q^{k=2}_{ss}(95)$	$E^{k=2}_{ss}(95)$
$I^{k=i}_{87}$	interme-diate (3)	steady state	87.0	30.6	69.0	20.6	80.0	26.7	67.0	19.4	$Q^{k=i}_{ss}(87)$	$E^{k=i}_{ss}(87)$
$B^{k=2}_{82}$	max	steady state	82.0	27.8	65.0	18.3	80.0	26.7	67.0	19.4	$Q^{k=2}_{ss}(82)$	$E^{k=2}_{ss}(82)$
$B^{k=1}_{82}$	min	steady state	82.0	27.8	65.0	18.3	80.0	26.7	67.0	19.4	$Q^{k=1}_{ss}(82)$	$E^{k=1}_{ss}(82)$
$B^{k=1}_{67}$	min	steady state	67.0	19.4	53.5	11.9	80.0	26.7	67.0	19.4	$Q^{k=1}_{ss}(67)$	$E^{k=1}_{ss}(67)$
$C^{k=1}_{67}$	min	steady state	67.0	19.4	53.5	11.9	80.0	26.7	(4)	(4)	$Q^{k=1}_{ss,dry}(67)$	$E^{k=1}_{ss,dry}(67)$
$D^{k=1}_{67}$	min	cyclic	67.0	19.4	53.5	11.9	80.0	26.7	(4)	(4)	$Q^{k=1}_{cyc,dry}(67)$	$E^{k=1}_{cyc,dry}(67)$

(1) - all maximum speed tests and minimum speed tests shall be performed at the same speeds, respectively
(2) - applies only to those units which reject condensate to the outdoor coil
(3) - speed selection is prescribed in section 3.1.3
(4) - wet bulb temperature sufficiently low that no condensate forms on the evaporator

Table 2.

Distribution of Fractional Hours in Temperature Bins to be Used for Calculation of SEER.

Bin number j	Bin temperature t_j (°F)	Fraction of total temperature bin hours n_j
1	67	0.214
2	72	0.231
3	77	0.216
4	82	0.161
5	87	0.104
6	92	0.052
7	97	0.018
8	102	0.004

Table 3
Summary of Test Conditions for Rating Variable Speed Systems
in the Heating Mode.

Test	Compressor Speed(1)	Outdoor Temperature dry-bulb °F	Outdoor Temperature dry-bulb °C	Outdoor Temperature wet-bulb °F	Outdoor Temperature wet-bulb °C	Indoor Temperature dry-bulb °F	Indoor Temperature dry-bulb °C	Indoor Temperature wet-bulb °F	Indoor Temperature wet-bulb °C	Quantity obtained and used in calculation procedure: capacity (Btu/h)	Quantity obtained and used in calculation procedure: power input (W)
Max Temperature Heating	minimum	62.0	16.7	56.5	13.6					$Q_{ss}^{k=1}(62)$	$E_{ss}^{k=1}(62)$
Cyclic Heating(2)	minimum									$Q_{cyc}^{k=1}(62)$	$E_{cyc}^{k=1}(62)$
High Temperature Heating	minimum									$Q_{ss}^{k=1}(47)$	$E_{ss}^{k=1}(47)$
High Temperature Heating(3)	nominal(4)	47.0	8.3	43.0	6.1	70.0	21.1	60(5) (max)	15.6(5) (max)	$Q_{ss}^{k=n}(47)$	
High Temperature Heating	maximum									$Q_{ss}^{k=2}(47)$	$E_{ss}^{k=2}(47)$
Frost Accumulation	maximum	35.0	1.7	30.0(6)	-1.1					$Q_{def}^{k=2}(35)$	$E_{def}^{k=2}(35)$
Frost Accumulation	intermediate(7)									$Q_{def}^{k=i}(35)$	$E_{def}^{k=i}(35)$
Low Temperature Heating	maximum	17.0	-8.3	15.0	-9.4					$Q_{ss}^{k=2}(17)$	$E_{ss}^{k=2}(17)$

(1) - all maximum speed tests and minimum speed tests shall be performed at the same speeds, respectively
(2) - details on the cycling rate are discussed in section 4.1.2
(3) - the test is optional; used for calculation of DHR only
(4) - the nominal speed is the lesser of the maximum speed in the cooling mode and maximum speed in the heating mode
(5) - should not exceed specified values
(6) - the specified temperature is a dew-point
(7) - selection of the speed is prescribed in section 4.1.3

Table 4. Distribution of Fractional Heating Hours in Temperature Bins, Heating Load Hours and Outdoor Design Temperature for Different Climate Regions.

Region*		I	II	III	IV	V	VI
Heating Load Hours, HLH		750	1250	1750	2250	2750	2750
Outdoor Design Temperature, T_D 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

*Heating domestic regions are shown in Figure 11.

Table 5

Standard Design Heating Requirements
(Btu/h)

5,000	25,000	50,000	90,000
10,000	30,000	60,000	100,000
15,000	35,000	70,000	110,000
20,000	40,000	80,000	130,000

Table A1. Bin Calculation Worksheet for Evaluation of SEER.

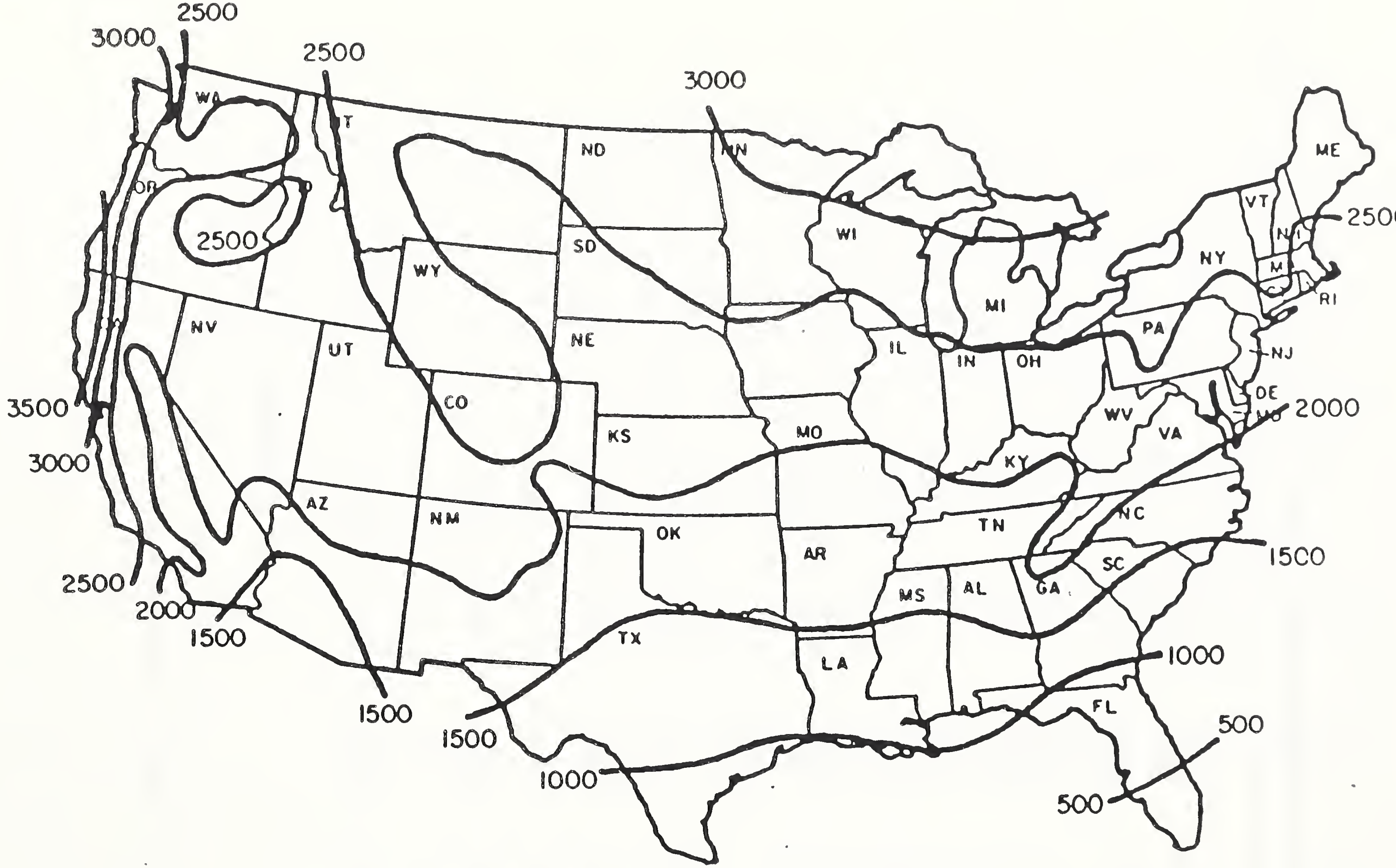
j	t_j	n_j	$BL(t_j)$	$Q(t_j)$	$E(t_j)$	$X(t_j)$	PLF	$n_j \cdot q$	$n_j \cdot e$
1	67.	.214	1636.	14492.	859.0	.113	.8	350.2	25.2
2	72.	.231	5727.	14038.	892.0	.408	.9	1323.0	95.4
3	77.	.216	9818.	13584.	925.0	.723	.9	2120.7	152.9
4	82.	.161	13909.	13909.	1017.0	1.000	1.0	2239.4	163.7
5	87	.104	18000.	18000.	1461.6	1.000	1.0	1872.0	152.0
6	92	.052	22091.	22091.	2096.7	1.000	1.0	1148.7	109.0
7	97	.018	26182.	26182.	3139.5	1.000	1.0	471.3	56.5
8	102	.004	30273.	26248.	3398.9	1.000	1.0	105.0	13.6
						TOTALS:		9630.3	768.3

SEER (Btu/(W•h)) = 9630.3/768.3 = 12.53

Table B1. Bin Calculation Worksheet for Evaluation of HSPF.

j	t_j	n_j	$BL(t_j)$	$Q(t_j)$	$X(t_j)$	PLF	$\delta(t_j)$	$BL \cdot n_j$	$e \cdot n_j$	$RH \cdot n_j$
1	62.	.132	963.	13274.	.073	.768	1.0	127.0	13.7	.0
2	57.	.111	2567.	12301.	.209	.802	1.0	284.9	30.4	.0
3	52.	.103	4171.	11328.	.368	.842	1.0	429.6	45.4	.0
4	47.	.093	5775.	10355.	.558	.889	1.0	537.1	56.0	.0
5	42.	.100	7379.	9382.	.787	.947	1.0	737.9	75.9	.0
6	37.	.109	8983.	8983.	1.000	1.000	1.0	979.2	101.4	.0
7	32.	.126	10588.	10588.	1.000	1.000	1.0	1334.0	151.0	.0
8	27.	.087	12192.	12192.	1.000	1.000	1.0	1060.7	131.3	.0
9	22.	.055	13796.	13796.	1.000	1.000	1.0	758.8	102.9	.0
10	17.	.036	15400.	15400.	1.000	1.000	1.0	554.4	82.4	.0
11	12.	.026	17004.	14824.	1.000	1.000	1.0	442.1	63.1	16.6
12	7.	.013	18608.	12537.	1.000	1.000	.5	241.9	15.0	47.0
13	2.	.006	20213.	10249.	1.000	1.000	.5	121.3	6.6	26.5
14	-3.	.002	21817.	7961.	1.000	1.000	.0	43.6	.0	12.8
15.	-8.	.001	23421.	5674.	1.000	1.000	.0	23.4	.0	6.9
							TOTALS:	7675.9	875.2	109.8

HSPF (Btu/(W•h)) = 7675.9/(875.2 + 109.8) = 7.793



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 11. Heating climatic regions and distribution of the Heating Load Hours (HLH) for the United States.

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A procedure is presented for testing and rating variable speed, split residential air conditioners and heat pumps. The procedure is derived in part from existing procedures for single speed and two speed systems where these procedures could be applied, and introduces a new algorithm for representation of variable speed unit performance in the intermediate speed operation range. Analysis and background which led to the formulation of the procedure are included as well as calculation examples for the cooling and heating mode. This procedure has been prepared for the Department of Energy for consideration in the rule making process.

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air conditioner; central air conditioner; heat pump; rating procedure; variable speed system

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