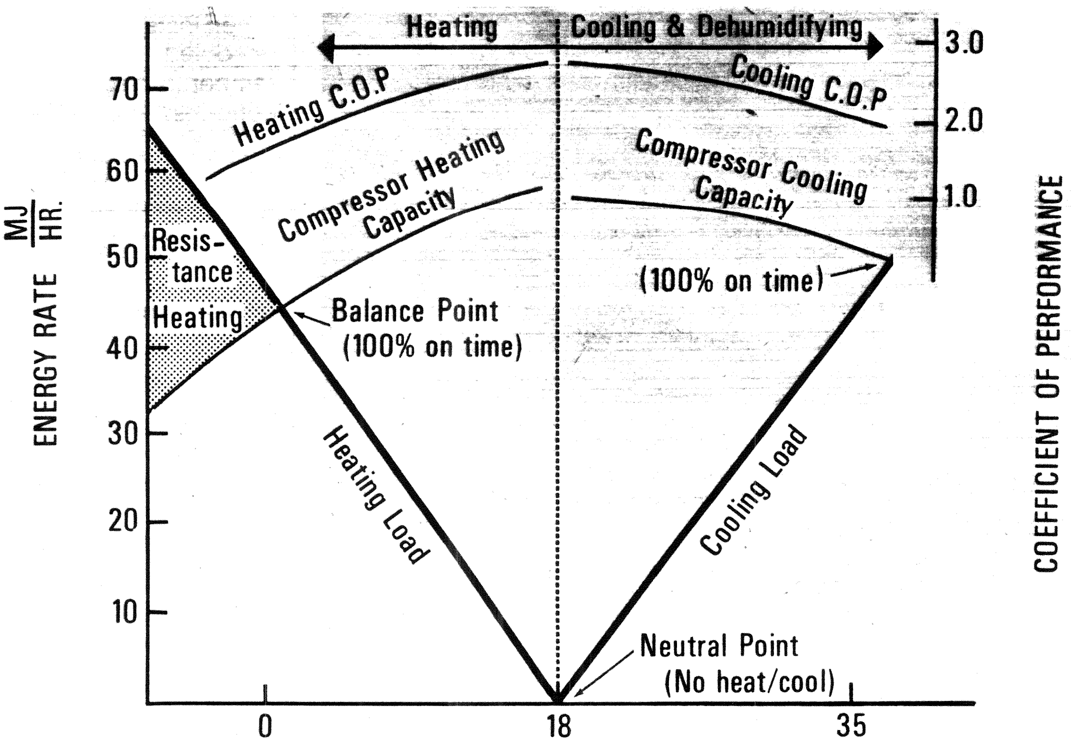

Fig. 1

New Testing and Rating Procedures for Seasonal Performance of Heat Pumps

If more efficient products are to be developed, it is essential that fair credit be given to as many energy conservation innovations as possible. In order to give such credit, an evaluation procedure is necessary which considers the causes of inefficiencies, or stated in a positive sense, considers the modifications to the unit which will tend to minimize all the inefficiencies that may exist as operated in the field.

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PRESENT testing and rating procedures for heat pumps[1,2] have served the industry and consumer well for many years prior to the energy conservation era. These steady-state (SS), full-load evaluation procedures permit capacity determination for proper sizing of the unit to match building loads. They also allow efficiency evaluations between various units. Increasing energy costs now require a life-cycle-cost approach rather than the traditional first-cost method. To provide the consumer with a better estimate of operational costs, the heat pump industry needs a more comprehensive evaluation technique to obtain data during part-load, cyclic operation. The wide variety of heat pump installations and climates makes true field performance data impractical as a basis for this cost analysis. However, it does appear that approximating the unit's field performance by laboratory part-load simulation can increase the amount of information for a reasonable testing investment.

Part-load laboratory evaluation offers two additional benefits. First, the resulting data allow for a more accurate estimate of energy usage on an absolute basis. Thus one obtains a better comparison between heat pumps and other equipment—something needed at least in the heating mode. Second, development of a more efficient product will be encouraged, since individual manufacturers will have greater opportunity to get credit for their innovations. In effect, it is a step towards making **efficiency** part of a "sales-pitch". Additional information can, of course, always be put to good use. The question as to how much investment is worthwhile to obtain that information is difficult to answer. The test procedure[3,4] presented here in abbreviated form was developed with that question in mind.

HEAT PUMP PERFORMANCE

Fig. 1 is a graphical representation of most of the information necessary for determining the seasonal performance of a heat pump. The equipment load lines are assumed to be a linear function of outdoor dry bulb temperature and to have their maximum values determined by the building load at the summer and winter design day temperatures. Due to highly variant and complex patterns of internal loads, the neutral point for the load lines is not necessarily coincident

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nor is the balance point of a single fixed value; however, unless known to be otherwise, they are usually assumed to be as shown. It is also assumed that the maximum cooling load is the basis for sizing the heat pump and that resistance heat coils will compensate for the deficiency in the heating capacity of the compressor. The system, as shown, is sized exactly to meet the cooling load for the summer design day.

As can be seen in Fig. 1, the steady-state test evaluation procedure will characterize a typical residential heat pump's two major inherent disadvantages. That is, a decrease in capacity and in coefficient of performance with an increase in operating temperature difference. These degradations result from the refrigerant vapor mass flow rate decrease and the Carnot effect, respectively. With these effects accounted for, the traditional rating procedure has been to weight the different steady-state coefficients of performance (COP) with operating time and load, which will vary depending on climate. The technique used is to describe the hourly outdoor dry-bulb occurrence of a given site by lumping the number of hours (n_j) of a season into 5°F temperature bins and summing up the respective COP'S at each temperature bin to obtain a seasonal performance factor (SPF)*. This calculation procedure may be expressed in an algebraic form of the mean value theorem as:

$$\frac{1}{SPF} = \frac{\sum_{j=1}^{K} n_j \, \text{Load}\,(T_j)\,[1/COP(T_j)]}{\sum_{j=1}^{K} n_j \, \text{Load}\,(T_j)} \qquad (1)$$

where: $SPF \equiv \frac{\text{seasonal output}}{\text{seasonal input}}$

*Various terms are currently being used, such as seasonal coefficient of performance, seasonal cooling performance factor, etc., which are identical in numerical value and are dimensionless as well. The seasonal energy effectiveness ratio (SEER) is conceptually the same thing but differs numerically because it is the cooling SPF multiplied by 3.413 Btu/watt-hr.

j = is the bin number of sequential integer value; each bin is 5°F wide.

T_j = 62 + 5j is the representative temperature within the jth bin. For the first temperature bin, T_1 = 67°F corresponding to j = 1.

In 1973-74, NBS conducted an extensive evaluation[5] of a residential heat pump, indicating that there are two additional phenomena which significantly degrade the unit's performance: (1) the "cycling effect" resulting from the need to establish a dynamic equilibrium condition after the system had returned to static equilibrium during the off- cycle; (2) the "frosting effect" on the outdoor coils, in the heating mode, which increases both the heat conduction resistance and the air flow passage resistance. Fig. 2 illustrates part of the results of this study and the two phenomena. The portion of the results to the left of the break, in the lower heat load or warmer outdoor temperature (O.T. > 4.4°C) region, has a slope which, if extrapolated, would indicate that the unit's COP at full load (Heating Load Factor = 1) would be about 95% of the manufacturer's SS rating, but at zero load (Heating Load Factor = 0) it would be about 86% of manufacturer's SS rating. The results in the frosting region (O.T. < 4.4°C) indicate a more or less constant degradation to about 74% of manufacturer's SS rating.

These results deserved a more detailed investigation of these phenomena under controlled laboratory conditions. The NBS results[6] of a laboratory evaluation of a similar unit (and subsequently several units of different manufacturers) are shown in Fig. 3 and 4. Cycling degradation occurs whether in the cooling or heating mode and is a stronger

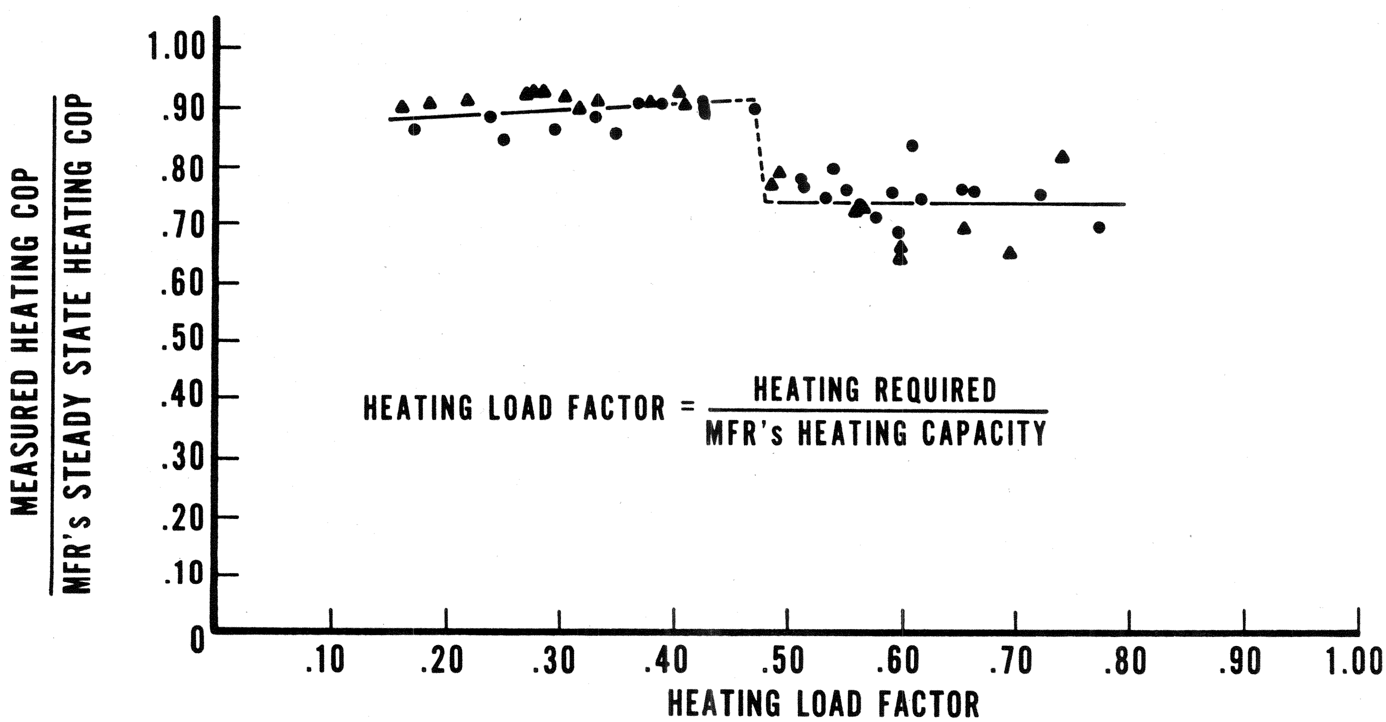


Fig. 2

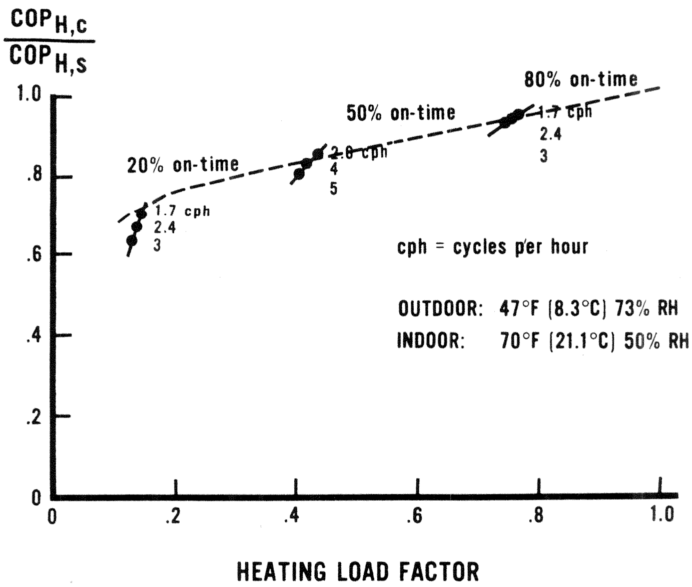


Fig. 3

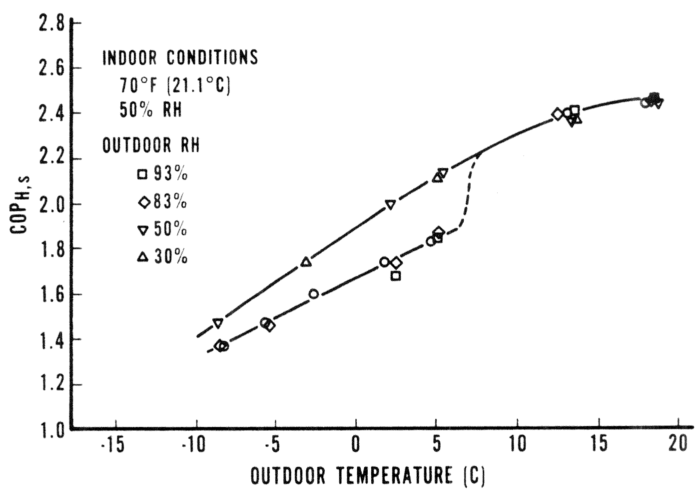


Fig. 4

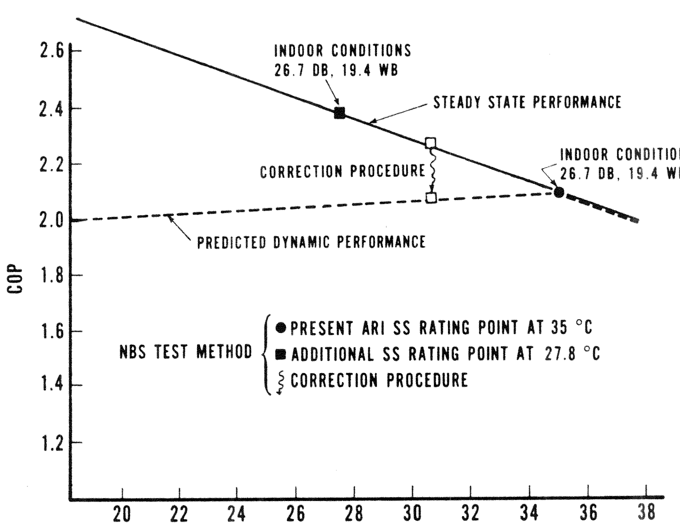


Fig. 5

function of percent on-time than of cycling rate. The significance of the frosting effect was isolated and highlighted by repeating a series of tests varying only the relative humidity and *not* including energy consumption for defrost in the steady-state heating COP determination. Fig. 4 illustrates the degradation caused by frost build-up on a heat pump using a demand-defrost system. A time-defrost unit would also have a degradation but it would be expected to be a stronger function of relative humidity.

As a result of these field and laboratory investigations it was concluded that actual field operation added two more inherent degradations to present-day vapor compression heat pumps and that a separate evaluation method which would measure these phenomena was required—if fair credit was to be given to all the different designs.

PART-LOAD COOLING TESTING AND RATING

Current industry testing procedure for heat pumps in the cooling mode requires only one certification point taken under full-load steady-state conditions at 35°C DB (95°F) outdoor temperature and 26.7°C DB (80°F)/19.4°C WB (67°F) indoor conditions. In order to establish a measure of the part-load cyclic effect it is necessary to evaluate the unit at one more outdoor temperature condition. The outdoor temperature at which this test is run is somewhat arbitrary; a value of 27.8°C (82°F) was selected, because it is the weighted mean of the U.S.A. national summer dry-bulb temperatures.** As in the existing full-load test, the indoor coil should be wet (same indoor conditions as the 35°C test) during the test, since dehumidifying is inseparable in the cooling mode. Unfortunately, cycling a unit under wet-coil conditions presents both accuracy and redundancy problems. Wet-bulb instrumentation systems have long time constants so that both room control and indoor air stream enthalpy measurements are unreliable under periodic conditions. It was however, noted through a series of tests, run with the meticulousness that only a research lab could afford, that:

$$\left.\frac{COP_{cyc}}{COP_{SS}}\right|_{\substack{dry\\coil}} = \left.\frac{COP_{cyc}}{COP_{SS}}\right|_{\substack{wet\\coil}}$$

With the assumption that the above relationship is true in general, one can then deduce the part-load wet-coil performance

$$\left(COP_{cyc}\Big|_{\substack{wet\\coil}}\right)$$

from a set of two steady-state tests (one wet, one dry) and a cyclic dry-coil test, all at the same outdoor temperature condition. Based on typical thermostat designs which control units to cycle at approximately 3 CPH at 50% on-time, the recommended cyclic test operation is 2 CPH at 20% on-time, which corresponds to 6 minutes on/24 minutes off.

With this cyclic wet-coil value calculated by means of the previous equation, a performance line may be defined by it and the steady-state data point at 35°C (95°F) (See Fig. 5). A load weighting process similar to the traditional bin method may be conducted for the rating procedure. This rating procedure may be expressed as:

$$CSPF = \frac{\sum_{j=1}^{K} n_j CBL(T_j)}{\sum_{j=1}^{K} n_j \dfrac{X(T_j)\,\dot{E}_{SS}(T_j)}{PLF(X)}} \tag{2}$$

**Weighted in proportion to unitary air conditioner sales around the country.

where:

CSPF is the cooling seasonal performance factor which is the new figure of merit accounting for cyclic as well as steady-state effects.

$CBL(T_J) = X(T_j)\,\dot{Q}_{SS}(T_j)$

ODT is the outdoor design temperature (dry bulb) which is applicable to the rating climate. It is expressed in °F When using the constants as shown in the equation for $CBL(T_j)$.

$\dot{Q}_{SS}(ODT)$ is the unit's steady-state capacity at the outdoor design temperature.

α is the oversizing factor, a value of 0.1 is suggested whn using the ASHRAE method for determining building loads.

$X(T_j)$ is the load factor which is equal to the ratio of building load, $CBL(T_j)$, to the steady-state capacity, $\dot{Q}_{SS}(T_j)$, when the steady-state capacity is greater than the building load; that is

$$X(T_j) = \left(\frac{5_j - 3}{ODT - 65}\right) \frac{\dot{Q}_{SS}(ODT)}{1 + \alpha} \frac{1}{\dot{Q}_{SS}(T_j}$$

if the part-load effect were ignored, $X(T_j)$ would be the percent run-time. When the capacity is less than the building load, a value of 1 is assigned.

PLF(X) is the part-load factor, which is a function of the load factor. More explicitly, $PL = 1 - C_D\ (1 - X(Tj))$ where C_D is the cyclic degradation coefficient defined by the expression:

$$C_D = \frac{1 - [COP_{cyc,\,Dry}(82)]/[COP_{SS,\,Dry}(82)]}{1 - CLF\,(82)}$$

The CLF (82) is the cooling load factor, which is the ratio of the cyclic capacity to the steady-state capacity at 82°F. Typically, values for PLF run from 1 at full-load operation to 0.75 at zero load.

$\dot{E}_{SS}(T_j)$ is the steady-state power input for the particular outdoor temperature T_j.

Although the number of terms in Eq. (2) make the expression somewhat complicated, it is still of the same basic form as the inverse of Eq. (1). The numerator is the seasonal output or building load that must be met, and the denominator is the seasonal input which includes the penalty factor for cycling effects.

The value of C_D is assumed to be constant when in fact it is a variable. Complete characterization of the C_D variation over the entire load range would require an unreasonable amount of testing. The cyclic test specifications given above should result in the best known single valued representation of the $C_D$variation. The Federal Document[3] offers an option to the cyclic testing by accepting an assigned value of .25 for C_D. Based on measurements made on a variety of models presently in production, it would appear that different designs can have an average C_D value anywhere from .02 to at least .35. The optional assigned value (.25) is not intended to be a median or a goal of any sort. It was selected so as to encourage the manufacturer to compete in the marketplace by designing (and testing to verify) a more seasonally efficient unit,[8] (*i.e.* one whose $C_D < .25$). On the other hand, those manufacturers who find the additional testing too burdensome still have a way to avoid increasing their testing costs without too drastic a performance penalty. Thus the 0.25 value was selected as a compromise between these opposing political-economic forces rather than for purely technical reasons.

For a heat pump having a single-speed compressor, it is possible to employ a simplified method to evaluate its CSPF for an average U.S. climate. It can be shown that multiplying the COP obtained in the wet- coil test at 27.8°C (82°F) by the PLF evaluated at X = 0.5 (*i.e.*, PLF = 1 − 0.5 C_D) yields a result which is virtually identical to the value of CSPF obtained by applying Eq. (2) to national average bin data. This is equivalent to saying that the CSPF can be found by evaluating the heat pump's dynamic COP at the national average cooling season temperature of 27.8°C (82°F) and the load factor (X = 0.5) corresponding to this average temperature.

PART-LOAD-HEATING TESTING AND RATING

Testing procedures for heat pumps in the heating mode require two certified test points taken under steady-state full-load conditions at 8.3°C (47°F) and −8.3°C (17°F) outdoor temperatures and 21.1°C (70°F) DB/15.6°C (60°F) maximum WB indoor conditions. In order to establish a part-load performance curve, it is necessary to require two additional test points to account for the cyclic effect and the frosting effect. A plot of the capacity and power curves defined by these points is illustrated in Fig. 6. The cyclic test point is reached directly from one test since the indoor coil is dry in the heating mode. The outdoor temperature value of 8.3°C (47°F) is prescribed as a matter of convenience since the traditional steady-state point (still used for capacity rating) is measured at this condition. This point does tend to be a good upper bound point, since typically the capacity curve will flatten at warmer temperatures. The frosting-point test is recommended to be at 1.7°C (35°F). This is the point at which the maximum rate of frost might be expected to occur. A lower temperature condition would have less water vapor in the air, while a higher temperature condition might result in natural melting during the off-cycle. The hypothetical capacity and power lines may be defined for conceptual purposes, as shown in Fig. 6. The solid lines represent performance obtained by using steady-state values, the dashed lines represent the performance that considers the cyclic and frosting effects. The frost buildup/defrost effect begins with the steady-state −8.3°C (17°F) point where no frost is assumed to occur, and causes the performance curves to deviate from the existing steady-state values in a linear direction through the 1.7°C (35°F) test point until 7.2°C (45°F), where a step change out of the frost region is assumed to occur. For convenience, this step change should be defined at an edge of a temperature bin, and assuming that most outdoor coils have a 5.6 to 8.3 degrees C (10 to 15 degrees F) temperature difference between the air and the refrigerant, the 7.2°C (45°F) value seems to be reasonably representative of field behavior. The cyclic effect is superimposed on the frosting effect between the balance point and 7.2°C (45°F). At 7.2°C (45°F) and above, the heat pump's performance is degraded only by the part-load cycling effect.

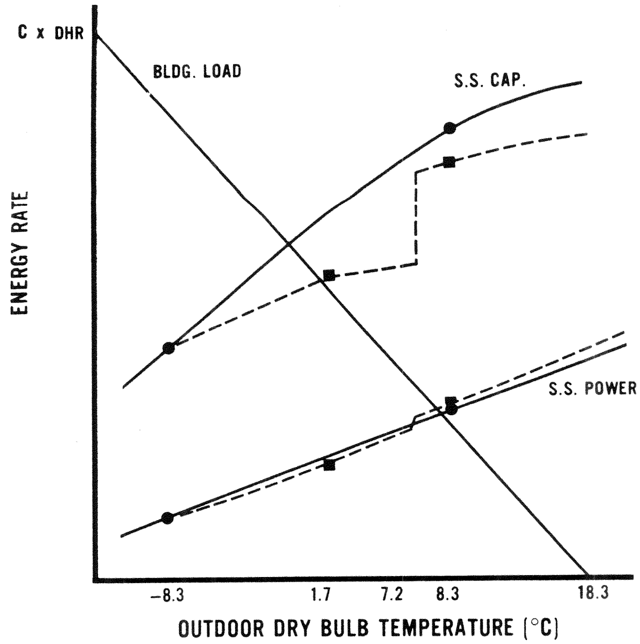
The rating procedure is, as before, a matter of considering the cyclic capacity, cyclic power, and number of operating hours for each temperature bin and determining the weighted average for the heating seasonal performance factor. Algebraically, this may be expressed as:

$$HSPF = \frac{\sum_{j=1}^{K} n_j HBL(T_j)}{\sum_{j=1}^{K} n_j \frac{X(T_j)}{PLF(X)}\ \delta(T_j)\,\dot{E}_{SS}(T_j) + \sum_{j=1}^{K} RH(T_j)} \tag{3}$$

where:

HSPF, n_j, $X(T_j)$, PLF(X), $\dot{E}_{SS}(T_j)$ have the same definitions as those in Eq. (2) except are now applied to the heating mode.

$BL(T_j)$ is the building heating load requirement which is shown in Fig. 1 and is defined by a zero value at 18°C (65°F) and the design heating requirement (DHR) value at the outdoor winter design temperature (T_{OD}). This may be expressed



● PRESENT ARI RATING POINT

■ ADDITIONAL RATING POINT NEEDED

Fig. 6

as $$HBL(T_j) = \left(\frac{5j - 2}{65 - ODT}\right) C\,(DHR)$$

where C is an experience factor to improve agreement between calculated and measured building loads. A value of .77 is recommended, but open to future adjustment.

$\varsigma(T_j)$ is the heat pump low temperature cut-out factor to account for those systems which have a compressor that shuts off at a given outdoor temperature. It has a vlue of 0 if $T_j <$ the cut-off temperature, a value of 1 if $T_j >$ the cut-on temperature, and a value of 1/2 if T_j is in between these two temperatures.

T_j = 67 − 5j is the representative temperature within the jth bin.

$RH(T_j)$ is the quantity of energy for resistance (supplemental) heat required for each bin. As illustrated in Fig. 1, it has a zero value at temperatures above the balance point and a finite value in the shaded triangular region below the balance point. It is, of course, quite sensitive to the sizing criteria and affects the HSPF significantly.

Both Eq. (2) and (3) are applicable to single-speed compressor/fan units only. The concepts discussed for these single-speed procedures are equally applicable for the two-speed units, but require additional testing and more complex expression for the rating to account for the differences in operation at both speeds. Details are described in references.[3,4]

CONCLUSIONS

Table 1 lists several innovations that heat pump manufacturers are presently working on or have recently developed. As shown, only the first two innovations would have received credit under the existing steady-state procedures. The next three innovations are effective under part-load conditions and would not be observed under full-load test conditions. On the other hand, all five innovations are given credit under the proposed test and rating procedures. As to the applicability of this procedure to new systems or cycles, it is difficult to predict; but certainly the concepts should be considered. We at NBS have already adapted these test procedures to evaluate engine-driven heat pump systems.[7]

Table 1

Heat Pump Innovations

	Test Method Existing	Test Method New (SPF)
• Increase Heat Exchange Area	X	X
• More Efficient Compressors	X	X
• 2-Speed Motors (Fan and Compressor)		X
• Defrost Controls		X
• Variable Flow Control		X
• New Systems		?
• New Cycles		?

Testing and rating procedures are dynamic documents; periodic reviews and revisions are constantly required if they are to continue to serve an everchanging product line. In the U.S.A., the procedure discussed herein has been proposed as a mandatory test and rating procedure by the Department of Energy for all of industry to follow. Simultaneously, these procedures are being used as a starting position for the ASHRAE Standards Committees 103.1P and 103.2P for their development of a new testing procedure to either complement or replace reference.[1] A similar process is under development at ARI for reference.[2] It is hoped that these second-generation documents coming out of the voluntary consensus process will then be substituted back into the mandatory sector as an improved updated procedure based on the experience the manufacturers are presently gaining with the part-load testing outlined here. It is hoped that improvements to ASHRAE/ARI documents (proposed completion in 1980) will be made when necessary, to accommodate new innovations in equipment and simplications in the testing procedure, as theoretical knowledge of heat pump performance is increased.

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