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PROCEDURES FOR TESTING, RATING, AND ESTIMATING THE SEASONAL PERFORMANCE OF ENGINE-DRIVEN HEAT PUMP SYSTEMS

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ABSTRACT

A generic test and rating procedure is developed for heat engine-driven air-to-air heat pump systems. The procedures are classified according to whether the systems have single-speed, two-speed, or variable-speed capability, and whether they are operating in the heating or cooling mode. The test requirements generally consist of a series of steady-state tests to establish the impact of outdoor temperature on performance, two or more part-load (cyclic) tests to determine the effect of "on-off" cycling, two steady-state intermediate speed tests to determine part-speed performance, and a single frost accumulation test to estimate the effect of frost. A generalized calculation and rating procedure is developed. The system is rated in both modes based upon its steady-state performance at the ARI standard rating points, its seasonal performance factor, and its seasonal operating cost. A frost degradation coefficient is also established. The seasonal parameters are based upon either a residential or commercial/industrial building application which is located in either a generalized northern or southern climate.

Key Words: Building heating and cooling; engine-driven heat pump; heat pump; heating and cooling equipment; heating, ventilating and air conditioning.

NOMENCLATURE

BHC	Average total bin hours for cooling, (hrs)
BHH	Average total bin hours for heating, (hrs)
$BL(T_j)$	Building load at dry-bulb temperature T_j, (Btu/hr)
C_D	Part-load degradation coefficient
C_{def}	Frost degradation coefficient
C_E	Electric energy cost, (\$/kWh)
C_F	Engine fuel cost, (\$/kBtu)
C_S	Supplemental energy cost, (\$/kBtu)
c_{pa}	Specific heat of air-water mixture, $c_{pa} = .24 + .444\, W_n$, (Btu/lbm - F)
COP_{cyc}	Cyclic coefficient of performance for one "off"/"on" cycle
$COP_{cyc}(T_j)$	Cyclic coefficient of performance at temperature T_j
$COP_{def}(32)$	Coefficient of performance under frost conditions
$COP_{ss}(T_j)$	Steady-state coefficient of performance at temperature T_j
$E_{cyc}(T_{cyc})$	Total input electric energy during one "off"/"on" cycle at T_{cyc}, (watt-hr)
$E_{def}(32)$	Total input electric energy during the frost accumulation and defrost test, (watt-hr)
$\dot{E}_{ss}(T_j)$	Steady-state electrical power input at temperature T_j, (watts)
E_T	Total electric and fuel energy required during a season, (Btu)
$E_T(T_j)$	Total input electric and fuel energy in the jth temperature bin, (Btu)
$E_T'(T_j)$	Cost of total input electric and fuel energy in the jth temperature bin (\$)
HDT	Heating design temperature, (F)
j	Bin number
LHV	Lower heating value of fuel (Btu/lbm or Btu/ft^3)
$\dot{m}_f(T_j)$	Fuel mass (or volume) flow rate (lbm/hr or ft^3/hr)
n	Total number of non-zero temperature bins
n_j	Number of hours in the jth temperature bin
N	Total heating or cooling season temperature bin hours, (hrs)
N^0	Minimum compressor speed, (rpm)

N^1	Maximum compressor speed, (rpm)
N^i	Intermediate compressor speed, (rpm)
$PLF(X(T_j))$	Heating or cooling part-load factor
Q	Total heating or cooling done during a season, (Btu)
$Q(T_j)$	Total heating or cooling done in the jth temperature bin, (Btu)
$Q_{cyc}(T_{cyc})$	Total heating or cooling during one "off"/"on" cycle at T_{cyc}, (Btu)
$Q_{def}(32)$	Net heating during the frost accumulation and defrost test, (Btu)
$Q_f(T_{cyc})$	Total input fuel energy at T_{cyc}, (Btu)
$Q_f(32)$	Total input fuel energy during the frost accumulation and defrost test, (Btu)
$\dot{Q}_f(T_j)$	Input fuel energy at temperature T_j, (Btu/hr)
Q_R	Total resistance heat required during a season, (Btu)
$Q_R(T_j)$	Total resistance heat in the jth temperature bin, (Btu)
$\dot{Q}_{ss}(T_j)$	Steady-state heating or cooling capacity at temperature T_j, (Btu/hr)
SOC	Seasonal operating cost ($)
SPF	Seasonal performance factor
t	Time
$T_{a1}(t)$	Dry-bulb temperature of air entering the indoor side, (F)
$T_{a2}(t)$	Dry-bulb temperature of air leaving the indoor side, (F)
T_c	Change-over temperature (temperature corresponding to zero heating and cooling requirements), (F)
T_{cyc}	Outdoor dry-bulb temperatures at which cyclic tests are conducted, (F)
T_j	Representative temperature for bin j, (F)
T_{off}	Outdoor temperature at which compressor is automatically turned off (heating only)
$T_H(t)$	The higher of the dry-bulb temperatures of the air entering and leaving the indoor side
$T_L(t)$	The lower of the dry-bulb temperatures of the air entering and leaving the indoor side
$\dot{V}$	Indoor air flow rate, (cfm)
v_n'	Specific volume of air-water mixture, (ft^3/lbm)
W_n	Humidity ratio, (lbm water/lbm dry air)

$X(T_j)$	Heating or cooling load factor at temperature T_j
Z	Normalized speed at intermediate speed N^i
α^i	Normalized part-speed capacity factor
η_s	Efficiency of producing supplemental heat from primary fuel (η_s = 0.3 if electric resistance heat is used)
ψ^i	Normalized part-speed input energy factor

Subscripts

H	Heating
C	Cooling
ss	Steady-state
cyc	Cyclic
j	Temperature bin number

Superscripts

0	Low speed (smallest capacity)
i	Intermediate speed (intermediate capacity)
1	High speed (greatest capacity)
e	Designates superscripts 0 and 1

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1. INTRODUCTION

1.1 BACKGROUND

The application of a heat engine to a heat pump has the inherent advantages of engine waste heat recovery to supplement the refrigeration cycle heating mode output, and increased heat pump efficiency in both the heating and cooling modes due to capacity modulation. There are currently a number of innovative engine-driven heat pump development projects in progress around the country [ref. 1-5], and there is as yet no standardized test and evaluation procedure available for these systems. Therefore, industry and government are unable to evaluate and compare these new heat pumps, and to make sound decisions regarding which are worthy of further development. A standardized test and rating procedure, incorporating provisions specifically tailored to the nature of each heat pump system, is required so that the results of different research groups working on different types of engine-driven heat pump systems may be effectively compared on the same technical basis.

1.2 OBJECTIVE AND SCOPE

The objective of this study is the development of a generic test and rating procedure for engine-driven heat pump systems which are likely to be employed in residential and small commercial buildings in the future. Inherent in the development of these procedures is the formulation of calculation procedures to estimate the seasonal performance and seasonal cost of operation of these systems. The test procedures and the rating and calculation procedures recommended herein apply only to prototype, heat engine-driven, air-to-air, mechanical compression, Rankine-cycle heat pumps operating in the heating and cooling modes. Since consideration is restricted to prototype systems, it is expected that the recommended test and rating procedures will be used by the industry as a tool to evaluate and compare the performance of such systems, and may eventually provide a foundation for a future less burdensome test and rating standard for factory-made systems which reach the market place.

1.3 CLASSIFICATION OF ENGINE-DRIVEN HEAT PUMP SYSTEMS

Engine-driven heat pump systems are typically classified according to the thermodynamic power cycle of the prime mover and the thermodynamic cycle of the refrigerator. In addition, they are often classed according to the nature of their heat source and sink (air-to-air, liquid-to-liquid, etc.). For residential and small commercial applications, an air heat source and sink is the most practical and common. Since current development efforts [1-5] involve air-to-air designs, the recommended test and rating procedures will be restricted to these heat pumps.

Of the many power cycles available for heat pump applications, the five most common are the Brayton, Diesel, Otto, Rankine, and Stirling cycles. The Brayton engine is normally rotary in design; the Rankine engine may be reciprocating or rotary, and the Otto, Diesel, and Stirling engines are all reciprocating, and they may be of a free-piston design. Since there is no reason to exclude any of these engines from consideration in heat pump applications, the recommended test and rating procedures will be sufficiently general to include all of them. For purposes of generality, however, the prime mover will be conceptualized as a "black box" which converts input fuel energy (liquid, solid, or gaseous) into useful work and recoverable and non-recoverable waste energy. References to a particular engine type will not normally be made except in those instances where uniqueness of design or operation requires it.

The refrigeration cycle most commonly employed for residential and small commercial applications is the vapor compression Rankine cycle. Since all current development efforts are using this cycle, the test and rating procedures recommended in this study will be restricted to it. The procedures will be sufficiently general, however, to be applicable to conventional positive and variable displacement compressors, as well as high-speed centrifugal or free-piston types.

1.4 PERFORMANCE CONSIDERATIONS

The current requirements for testing and rating of electrically-driven unitary heat pump equipment are described in the Air Conditioning and Refrigeration Institute (ARI) Standard 240-76. Test methods used to evaluate this equipment are described in the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Standard 37-69. For air-source heat pumps the current heating mode rating requirements are based upon two steady-state tests conducted at indoor dry-bulb and wet-bulb temperatures of 70°F (21.1°C) and 60°F (15.6°C), respectively. One test is conducted at outdoor dry-bulb and wet-bulb temperatures of 47°F (8.3°C) and 43°F (6.1°C), and the other is conducted at 17°F (-8.3°C) and 15°F (-9.4°C), respectively. In the cooling mode, the rating requirements are based on one steady-state test conducted at indoor dry-bulb and wet-bulb temperatures of 80°F (26.7°C) and 67°F (19.4°C), respectively, and an outdoor dry-bulb temperature of 95°F (35°C). In addition, for units which reject condensate to the outdoor air stream, an outdoor wet-bulb temperature of 75°F (23.9°C) is required.

Knowledge of the steady-state performance of a heat pump system at the ARI rating points is useful for single point rating and comparative purposes, regardless of the power source. Such knowledge is insufficient, however, to assess the non-steady-state performance of the system, and is insufficient to determine the seasonal performance of the system. Experimental investigations of electric heat pumps have indicated that the "on-off" cycling that these units must undergo to satisfy comfort requirements often has a significant effect on the performance of the units in both modes of operation. Because of the dynamic characteristics of the heat engine, it's expected that cycling will have an even greater effect on the performance of an engine-driven system. Experimental investigations of engine-driven heat pump systems have shown that heating and cooling capacity and input energy are generally nonlinear functions of outdoor dry-bulb temperature. This is particularly true in the heating mode where heat pump capacity is supplemented with recovered engine heat. Because of differences in engine brake thermal efficiency and heat recovery characteristics, as well as differences in heat pump efficiency, the effect of outdoor temperature must be considered in the test and rating procedures and in the seasonal performance determinations. Additional factors which affect instantaneous and seasonal performance are the climate in which the system operates, the building load classification, capacity modulation, and defrost requirements. All of these factors are addressed in the recommended test, rating, and calculation procedures which follow.

2. RECOMMENDED TESTING REQUIREMENTS

2.1 GENERAL

The recommended testing and rating requirements for engine-driven heat pumps are classified basically according to whether the system has single-speed, two-speed, or variable-speed capability, and whether it is operating in the heating or cooling mode. The test requirements generally consist of a series of steady-state wet-coil (or no-frost) tests which provide data necessary to define the steady-state performance curve, combined with two or more cyclic (dynamic) tests to determine the part-load performance. For variable speed systems the test requirements also include a provision for steady-state, intermediate speed tests to determine part-speed performance. The performance of the system under frost accumulation and subsequent defrost operation is characterized by a single test under high relative humidity conditions. The system is then rated in both operating modes based upon its steady-state performance at the ARI standard rating points, its seasonal performance factor, and its seasonal operating cost. A summary of the testing requirements is presented in table 1 and is illustrated in figures 1-3.

2.2 STEADY-STATE OPERATION

The steady-state, wet-coil (or no-frost) testing requirements are dependent on whether the heat pump system has single-speed, two-speed, or variable-speed capability. The respective requirements are discussed below.

2.2.1 Single-Speed Operation

A total of six steady-state, wet-coil (or no-frost) tests shall be conducted according to the test procedures specified in section 3.2. Of this total, two wet-coil tests shall be conducted in the cooling mode and the remaining four in the heating mode.

In the wet-coil cooling mode tests, the air entering the indoor portion of the unit shall have a dry-bulb temperature of 80°F (26.7°C) and a wet-bulb temperature of 67°F (19.4°C). The dry-bulb temperature of the air surrounding and entering the outdoor portion of the unit shall be 95°F (35°C) in the first test, and 80°F (26.7°C) in the second. For those units which reject condensate to the outdoor air stream, the outdoor wet-bulb temperatures entering the outdoor unit shall be 75°F (23.9°C) and 64°F (17.8°C), respectively. The outdoor dry-bulb and wet-bulb temperatures of the first test point coincide with the ARI standard rating point for air-source units in the cooling mode. The 80°F (26.7°C) dry-bulb temperature of the second point was chosen because it approximates the average operating temperature in northern climates during the cooling season. The corresponding wet-bulb temperature was chosen so that the relative humidities of both testing points are approximately equal.

In the heating mode tests, the air entering the indoor portion of the unit shall have a dry-bulb temperature of 70°F (21.1°C) and a maximum wet-bulb temperature of 60°F (15.6°C). The four wet-coil (or no-frost) tests shall be conducted with the dry-bulb temperature of the air surrounding and entering the outdoor unit equal to 47°F (8.3°C), 32°F (0°C), 17°F (-8.3°C), and -13°F (-25°C), respectively. The corresponding wet-bulb temperatures of the entering air shall be 43°F (6.1°C), 30°F (-1.1°C) or less, and 15°F (-9.4°C), respectively, for the first three tests, and shall be left uncontrolled for the remaining one. The outdoor conditions of the first and third testing points coincide with the ARI standard high and low temperature rating points for air-source units in the heating mode. The second test point was chosen in order to provide a reference for a subsequent frost accumulation and defrost test, and the fourth point was chosen in order to establish low temperature system performance. The four testing points shall be used to determine the system performance as a function of outdoor dry-bulb temperature.

2.2.2 Two-Speed/Dual-Capacity Operation

In order to properly characterize the performance of engine-driven systems with two-speed compressor capability, the test requirements are separated into high-speed and low-speed requirements, and they shall be conducted according to the test procedures specified in section 3.2. In the cooling mode, the steady-state tests for both compressor speeds are the same as those described in section 2.2.1 for single speed operation, and are the minimum tests necessary to estimate the steady-state performance curve for each speed. In the heating mode, however, three high-speed compressor tests are specified over a range of low ambient temperatures, and three low-speed compressor tests are specified over a range of higher ambient temperatures. This approach reflects the way in which most two-speed systems are designed to operate, and also reduces the number of tests necessary to establish the performance curve for each compressor speed.

The three wet-coil (or no-frost) low-speed tests shall be conducted with the dry-bulb temperature of the air surrounding and entering the outdoor unit equal to 47°F (8.3°C), 32°F (0°C) and 17°F (-8.3°C), respectively. The corresponding wet-bulb temperatures of the entering air shall be 43°F (6.1°C), 30°F (-1.1°C) and 15°F (-9.4°C). A maximum wet-bulb temperature of 30°F (-1.1°C) was chosen for the second test in order to reduce the accumulation of frost on the outdoor coil. Although frost may form during this test, and during the subsequent test at 17°F (-8.3°C) dry-bulb temperature, its effect during the time span of the tests is not expected to be great.

The three no-frost high-speed tests shall be conducted at dry-bulb temperatures of 17°F (-8.3°C), 2°F (-16.7°C), and -13°F (-25.0°C). The wet-bulb temperature of the first point shall be 15°F (-9.4°C) and shall be left uncontrolled for the remaining two.

Systems which use twin compressors, staged compression, dual firing rates, suction/discharge valve control, etc., to achieve dual capacity performance are subject to the same test requirements as those systems with two-speed compressors. In these instances the term "high-speed" shall mean operation with both compressors, or at the greatest capacity level, and "low-speed" shall mean one compressor, or smallest capacity operation. As indicated in table 1 and illustrated in figure 2, a total of ten steady-state tests shall be conducted for systems with two-speed capability.

2.2.3 Continuously Variable-Speed Operation

Experimental investigations of engine-driven heat pump systems have shown that heating and cooling capacity, input fuel energy, and input electric energy are generally nonlinear functions of compressor speed as well as of each other. Therefore, for those systems whose speed is continuously variable, the test requirements and rating procedures must include provisions to estimate the system's performance at speeds intermediate to the maximum and minimum speed.

The steady-state test requirements for variable-speed heat pump systems are divided into two sets. The requirements for the first set are exactly the same as those described in section 2.2.2 for two-speed operation. High and low compressor speed shall then mean maximum and minimum compressor speed. As in section 2.2.2, a total of ten steady-state tests shall be conducted according to the test procedures specified in section 3.2.

The second set of requirements consists of steady-state intermediate speed tests conducted at a normalized speed Z according to the test procedures specified in section 3.2. The normalized speed Z is defined as:

$$Z = \frac{N^i - N^0}{N^1 - N^0} \tag{2.1}$$

where Z varies from 0 to 1, and N^0, N^1, and N^i represent minimum speed, maximum speed, and any intermediate speed, respectively. One wet-coil intermediate speed test shall be conducted in the cooling mode. The indoor conditions shall be as specified in section 2.2.1, and the outdoor dry-bulb and wet-bulb temperatures shall be 80°F (26.7°C) and 64°F (17.8°C), respectively. The test shall be conducted at a normalized speed of Z = 1/2.

One no-frost intermediate speed test shall be conducted in the heating mode at the indoor conditions specified in section 2.2.1. The outdoor dry-bulb and wet-bulb temperatures shall be 17°F (-8.3°C) and 15°F (-9.4°C), and the test shall be conducted at Z = 1/2. The above intermediate speed test requirements are summarized in table 1 and illustrated in figure 3. The results will be used to calculate part-speed capacity and input energy with the calculation procedure described in section 5.3.

2.3 PART-LOAD (CYCLIC) OPERATION

Laboratory investigations (refs. 6 and 7) have shown that the heating and cooling performance of electric heat pumps is significantly reduced when operated under part-load, or cyclic, conditions. Performance degradation has been shown to be directly dependent on load and to be relatively insensitive to outdoor ambient conditions. During cyclic wet-coil cooling tests of electric heat pumps, it has been found that constant indoor conditions are difficult to maintain, and that dry-bulb and wet-bulb temperature measurement errors can create large uncertainties in the cyclic sensible and latent capacities. Further cooling tests have indicated, however, that cyclic wet-coil tests may be replaced by simpler and more accurate dry-coil tests. Specifically, the results showed that when steady-state and cyclic dry coil tests were conducted, the ratio of the cyclic COP to the steady-state COP was essentially the same as the COP ratio determined from wet-coil tests at the same cooling load factor.

The part-load test requirements during the cooling mode shall include one or more steady-state dry-coil tests and an equal number of cyclic dry-coil tests. These tests shall be performed in conjunction with each other according to the test procedures specified in sections 3.2 and 3.3, respectively. The indoor dry-bulb temperature shall be 80°F (26.7°C) and the wet-bulb temperature shall be a value which does not cause condensate to form on the indoor coil. It is recommended that the temperature be equal to or less than 57°F (13.9°C). The outdoor dry-bulb-temperature shall be 80°F (26.7°C) and the wet-bulb temperature shall be unspecified. For systems with single and continuously-variable speed capability, there shall be one steady-state dry-coil test and one cyclic dry-coil test. For variable-speed systems both tests shall occur at the minimum speed since this is the speed at which cycling will normally occur under low load conditions. For systems with two-speed (two-capacity) capability there shall be two steady-state dry-coil tests and two cyclic dry-coil tests, for a total of four. Since the dynamic performance of an engine-driven system is a function of speed (capacity), a steady-state and cyclic test shall occur at each of the two speeds of operation, and they shall be performed in conjunction with each other.

The cyclic test requirements during the heating mode shall include one cyclic test for single-speed and continuously-variable speed systems. The outdoor dry-bulb and wet-bulb conditions shall be 47°F (8.3°C) and 43°F (6.1°C), respectively. Two cyclic tests shall be conducted for two-speed (dual-capacity) systems, one at each compressor speed. The low-speed test shall occur at the dry-bulb and wet-bulb conditions specified above, and the high-speed test shall occur at dry-bulb and wet-bulb conditions of 17°F (-8.3°C) and 15°F (-9.4°C), respectively. The indoor dry-bulb and wet-bulb temperatures shall be 70°F (21.1°C) and 60°F (15.6°C), respectively, for all cyclic heating tests.

The conditions specified for these cyclic heating tests are the same as those outlined in section 2.2.1 for two of the steady-state tests. Each cyclic heating test shall, therefore, be performed in conjunction with its corresponding steady-state test according to the procedures specified in section 3.2. The cyclic test requirements for heating and cooling are summarized in table 1 and figures 1-3. The results will be used to calculate a part-load degradation coefficient C_D with the calculation procedures described in section 5.2.

2.4 FROST ACCUMULATION AND DEFROST OPERATION

Accumulation of frost on the outdoor coil has been shown (ref. 7) to cause a significant drop in the performance of electric heat pumps compared to their non-frosted performance. Since frost formation occurs most rapidly over a range of outdoor conditions in which most space heating systems accumulate many years of yearly operation, a frost accumulation and defrost test should be included in the test and rating procedures for engine-driven heat pump systems.

One proposed approach to account for the deteriorating effects of frost would be to apply a correction factor to all steady-state performance data obtained within the frost range. The correction would be a function of outdoor dry-bulb temperature and relative humidity, and would be based upon a series of frost accumulation tests. Because of the current lack of experimental data regarding the performance of engine-driven heat pump systems under frosted-coil and defrost conditions, it is very uncertain how such a correction should vary as a function of ambient temperature and humidity conditions. It is equally uncertain at this time as to what minimum tests should be required of the system developer in order to define these functions. Therefore, this approach is not recommended for the prototype systems addressed in these procedures.

Instead, a single-point frost accumulation and defrost test is specified for frost rating purposes. As actual operating data become available under frosting conditions, the development of a more general frost correction should be investigated.

A single-point frost accumulation and defrost test shall be performed with the dry-bulb and wet-bulb temperatures of the air entering the outdoor unit equal to 32°F (0°C) and 30°F (-1.1°C), respectively. These conditions correspond to a relative humidity of 80%. This is an average value for most regions of the country which have a significant number of their heating hours occurring at outdoor temperatures less than 35°F (1.7°C). The indoor conditions shall be 70°F (21.1°C) dry-bulb and 60°F (15.6°C) maximum wet-bulb temperature. For systems with two-speed (dual-capacity) or continuously-variable speed capability, the test shall occur at the lower speed (capacity) since most engine-driven systems would normally operate at this speed when the outdoor temperature is 32°F (0°C). The test shall be pseudo steady-state over a complete frost accumulation and defrost cycle, and it shall begin at defrost termination and continue until the next defrost termination. The time-until-defrost, the method of defrost, and the operating procedures during the defrost period shall not be specified. In this way, no single heat pump developer or type of engine-driven system will be favored. The frost accumulation test shall be conducted according to the test procedures specified in section 3.4, and the results are to be reported in connection with the corresponding non-frosting steady-state test at 32°F (0°C) specified in section 2.2.1. The calculation procedure is described in section 5.5.

2.5 INDOOR-SIDE AND OUTDOOR-SIDE AIR QUANTITY

All steady-state and cyclic heating and cooling tests shall be conducted at the outdoor-side air quantity requirements specified in section 5.1.4.4 of ARI Standard 240-76 and ARI Standard 210-74, respectively. All steady-state and cyclic heating and cooling tests shall be conducted at the indoor-side air quantity specified in section 5.1.4.3 of ARI Standard 240-76 and ARI Standard 210-74, respectively. Rated heating and cooling capacity shall refer to the capacity of the system as measured at the highest speed at an outdoor dry-bulb temperature of 47°F (8.3°C) and 95°F (35°C), respectively.

3. RECOMMENDED TEST PROCEDURES

3.1 INTRODUCTION

The recommended test procedures have been divided into steady-state procedures, part-load (cyclic) procedures, and frost accumulation and defrost test procedures. They have been established within the general framework of providing adequate and reliable test data on the performance of prototype heat pump systems at a reasonable investment of funds, time, and effort by the developer. Where feasible, the steady-state test procedures have been adapted from the ASHRAE Standard 37-69, and the part-load procedures have been adapted from references 6 and 7. As additional performance data and operating experience are acquired on engine-driven systems, additions or modifications to these procedures may be required.

3.2 STEADY-STATE TEST PROCEDURE

The following test procedures pertain to the steady-state heating and cooling tests specified in section 2.2 and illustrated in table 1 and figures 1-3.

3.2.1 Applicable Test Method

ASHRAE Standard 37-69 describes the following four test methods that may be used to determine the heating and cooling capacities of unitary electric heat pump equipment:

a) Air-Enthalpy Method - Indoor Side
b) Air-Enthalpy Method - Outdoor Side
c) Compressor Calibration Method
d) Volatile Refrigerant Flow Method.

Because of differences in engine-driven heat pump types and differences in heat recovery equipment configuration, the test method most applicable to all equipment types is the Air-Enthalpy Method - Indoor Side. Therefore, all steady-state heating and cooling tests on engine-driven heat pump equipment shall employ this method as the required test method. Prototype developers are encouraged, however, to simultaneously employ additional measurement and analysis techniques as a check on the overall experimental system. Performance calculations should be based, however, only on the results of the Indoor Side Air-Enthalpy Method. Test room requirements are the same as those specified in section 11.1 of the ASHRAE Standard 37-69.

3.2.2 Instrumentation and Required Data

The steady-state performance tests shall have the same instrumentation and data requirements as those specified in section 10 and table II of ASHRAE Standard 37-69, with the following additions or modifications:

(a) Liquid or gaseous fuel quantity shall be measured with a suitable integrating type meter having an accuracy within $\pm$ 2.0% of the quantity measured.
(b) The lower heating value (LHV) of a representative sample of fuel shall be determined by suitable chemical or calorimetric analysis and shall be used in all subsequent input fuel energy calculations.
(c) The dry-bulb temperature of the air entering and leaving the indoor side (or the difference between these two temperatures), and the dry-bulb temperature of the air entering

the outdoor refrigerant coil shall be continuously recorded with instrumentation having a total system accuracy and precision within $\pm$ 1.0°F (0.6°C).

(d) The electric energy usage of all components and accessories (fans, pumps, blowers, control circuits, etc.) shall be measured with watt-hour meters that are accurate to within $\pm$ 2.0% of the quantity measured.

3.2.3 Test Operating Procedure and Results

The engine-driven heat pump system and any associated test room reconditioning equipment shall be operated for at least one-half (1/2) hour under equilibrium conditions before any steady-state heating or cooling performance data are recorded. Data shall then be recorded at ten-minute intervals until four consecutive sets of readings within the tolerances specified in table 2 have been attained.

Under some conditions of heating, a small amount of frost may accumulate on the outdoor coil, and a distinction needs to be made between frosting and non-frosting operation during the test period. For purposes of these procedures, the test is considered non-frosting provided that the dry-bulb temperature of the air leaving the indoor and outdoor units does not deviate by more than 2.0°F (1.1°C) during the test. When this tolerance is exceeded because of frost, the defrost cycle shall be initiated manually and the steady-state test repeated. For those systems with controls which periodically initiate a defrost cycle based upon time, the control system shall be modified to prevent defrost during the steady-state heating tests.

The steady-state heating and cooling results shall include each of the following quantities calculated using the procedures described in section 5.1:

(a) Total heating or cooling capacity, Btu/hr (J/sec).
(b) Input fuel energy, Btu/hr (J/sec).
(c) Total electric power input to all components and accessories, watts.
(d) Steady-state Coefficient of Performance.

Sections 12.1.5-12.1.7 of the ASHRAE Standard 37-69 shall apply for all steady-state heating and cooling test results.

3.2.4 Test Tolerances

All steady-state heating and cooling tests shall be conducted within the applicable test operating and test condition tolerances specified in table 2. Test operating tolerance is defined as the greatest permissible difference between maximum and minimum instrument observations during the test. Test condition tolerance is defined as the maximum permissible variation of the average of the test observations from the desired test condition. Variations greater than those prescribed in table 2 shall invalidate the test.

3.3 PART-LOAD (CYCLIC) TEST PROCEDURE

The following test procedures pertain to the cyclic dry-coil and non-frosting tests specified in section 2.3 and illustrated in table 1 and figures 1-3.

3.3.1 Applicable Test Method

All part-load (cyclic) tests shall employ only the Air-Enthalpy Method - Indoor Side, and each test shall be performed in conjunction with its corresponding steady-state test as described in section 2.3.

3.3.2 Instrumentation and Required Data

The instrumentation and data requirements for the cyclic test shall be the same as those specified in section 3.2.2 for steady-state tests. An additional requirement is that the instrumentation used to measure the air temperature entering and leaving the indoor side, or the difference between these two temperatures, shall have a response time of 2.5 seconds or less. Response time is defined as the time required for the instrumentation to attain 63% of the final steady-state temperature difference when subjected to a step change of 15°F (-9.4°C) or more.

3.3.3 Test Operating Procedure and Results

Each cyclic test shall immediately follow the corresponding steady-state test described in section 2.3 and illustrated in table 1. At the conclusion of the steady-state test the engine-driven heat pump system shall be manually cycled "off" and "on" until steadily repeating ambient conditions are achieved in both the indoor and outdoor test chambers, but for not less than two complete "off"/"on" cycles. Without disrupting the cycling process, the system shall then be operated for an additional "off"/"on" cycle during which the required test data shall be taken. During this test cycle the indoor and outdoor conditions shall remain within the tolerances specified in table 2.

A complete "off"/"on" cycle shall be defined as an "off" period of 24 minutes and an "on" period of six minutes. This results in a cycling rate of two cycles per hour which is approximately the rate at which thermostats are set in order to maintain building temperature variations within 3°F (1.7°C) when operating under a 20% heating or cooling load.

For those engine-driven systems which permit the compressor to be quickly disengaged from the prime-mover, cycling "off" and "on" shall refer to the refrigerant compressor operation. For those systems whose refrigerant compressor is permanently coupled to the prime-mover, cycling "off" and "on" shall refer to the initiation of that action which results in the most rapid stopping and starting of the refrigerant compressor, respectively. Depending on the type of engine, this may require decreasing or stopping the fuel flow to the engine, stopping or diverting the working fluid from the engine's expander, equalizing pressures across the expander, etc. The indoor fan shall cycle "off" and "on" with the compressor, but its cycling times may be delayed or modified by controls which are normally installed with the system. Operation and control of the engine and associated components during the "off" and "on" periods shall be determined by the manufacturer. Any fuel consumed by the engine and any electric power used by pumps, fans, blowers, controls, etc., during the "off" period shall be measured, however, and included with "on" period energy in subsequent performance calculations. Also, provisions should be made to measure and include any "coast-down" heating or cooling which may occur after action has been initiated to stop the refrigerant compressor. If, during the cyclic heating tests, the engine continues to operate during the "off" period, the recovered engine waste energy shall be measured and included with "on" period energy.

The cyclic test results shall include the following quantities calculated for the complete test cycle using the procedures described in section 5.2:

(a) Total heating or cooling capacity over the test cycle, Btu (J).
(b) Total input fuel energy over the test cycle, Btu (J).
(c) Total electric power input to all components and accessories over the test cycle, kWh.
(d) Cyclic Coefficient of Performance.

The steady-state and cyclic test results shall be used to calculate a part-load degradation coefficient C_D according to the calculation procedures outlined in section 5.2.

3.3.4 Test Tolerances

The cyclic heating and cooling tests shall be conducted within the test operating and test condition tolerances specified in table 2. In order to meet the required air temperature tolerances, it is recommended that electric resistance heaters be installed in the indoor and outdoor chambers to offset the fluctuating heating and cooling loads imposed by the cycling equipment.

3.4 FROST ACCUMULATION AND DEFROST TEST PROCEDURE

The following test procedures pertain to the single-point frost accumulation and defrost test specified in section 2.4. The results of the test will be combined with the results of a steady-state non-frosting test, performed under the same ambient conditions, to form a single-point frost degradation coefficient C_{def} according to the procedures outlined in section 5.5.

3.4.1 Applicable Test Method

The frost accumulation and defrost test shall be conducted using the Air-Enthalpy Method - Indoor Side.

3.4.2 Instrumentation and Required Data

The instrumentation and data requirements shall be the same as those specified in section 3.2.2 for steady-state tests. They shall also include the 2.5-second response requirement for temperature instrumentation specified in section 3.3.2 for cyclic tests.

3.4.3 Test Operating Procedure and Results

The engine-driven heat pump system shall be operated for at least one-half (1/2) hour under equilibrium ambient conditions prior to the actual test. Then, in order to assure that the outdoor refrigerant coil is frost-free at the start of the test, the system shall be operated through a manually induced but normally terminated defrost period. The test shall then commence at defrost termination and shall continue through the frost accumulation and subsequent defrost periods. It shall terminate at defrost termination. Defrost initiation and termination shall be governed by the normal controls associated with the system. Data shall be taken at ten-minute intervals during the frost accumulation heating period, and the times at defrost initiation and termination shall be recorded. Since defrost method, control, and mode of operation will generally be dependent upon specific equipment configuration and engine type, they shall be determined by the manufacturer and shall not be part of these test procedures. However, the total fuel consumed by the engine, the total electric power used by associated components (excluding resistance heaters which may be installed), and the total heating and cooling done during the defrost period shall be measured and included in the overall performance calculations.

The test results shall include the following quantities calculated for the frost accumulation and defrost cycle (defrost termination to defrost termination) using the procedures described in section 5.5

(a) Net heating capacity, Btu (J)
(b) Total input fuel energy, Btu (J)
(c) Total electric power input, kWh
(d) Frosting-defrost Coefficient of Performance

These results shall be combined with the results of the steady-state non-frosting test to determine a frost degradation coefficient C_{def} using the procedures outlined in section 5.5.

3.4.4 Test Tolerances

The test operating and test condition tolerances which must be maintained during the frost accumulation and defrost test are specified in table 2. The indoor and outdoor conditions shall be based only on the values occurring during the heating period.

4. BUILDING LOADS AND CLIMATE SPECIFICATIONS

4.1 INTRODUCTION

The seasonal performance and seasonal cost of operation of any heat pump system depends not only upon the instantaneous performance of the system under specific indoor and outdoor conditions, but also upon the type of building in which it's installed and its thermal load, and the climate in which the building is situated. Because of the wide range of heating and cooling climates in the United States, and the even wider range of building types and thermal requirements, it becomes impossible to adequately characterize the performance of a heat pump system for all regions of the country with one or two seasonal indicators. Because of waste heat recovery, however, engine-driven heat pump systems are most applicable to northern climates where the energy requirements are predominantly heating and the engine's recovered waste heat can be effectively utilized. There will be applications, however, in which the system is located in a southern climate. In these instances, the cooling requirements may be approximately equal to, or even greater than, the heating requirements. In order to provide the prototype developer some latitude and flexibility in establishing the seasonal performance of his product, the following evaluation approach is recommended for seasonal calculations:

(a) assume a generalized northern climate and a generalized southern climate in which an engine-driven heat pump system is most likely to be located.
(b) for each climate, assume representative heating and cooling design temperatures at which the system heating and cooling capacities equal the building energy requirements.
(c) assume heating and cooling loads typical of a residential building, and of a light commercial/industrial building.
(d) for both seasons, use the temperature bin method to determine the seasonal performance factor and the seasonal operating cost based upon the appropriate building type and the choice of one of the two generalized climates.

While this seasonal rating approach suffers the inevitable disadvantages associated with climate and building generalization, it does provide a suitable technique for assessing the relative seasonal performance of different heat pump systems which are designed for the same market and the same general climate.

4.2 GENERALIZED CLIMATE

Tables 3 and 4 list temperature bin information and the fractional hours in each bin during the heating and cooling season for the generalized northern and southern climates. The tables are based upon residential and commercial/industrial applications, respectively. The fractional hours in each bin for northern climate were determined by averaging weather data from six U.S. cities, and the fractional hours for the southern climate were obtained by averaging data from eight cities. Table 5 lists average heating degree days DD, heating design temperature HDT, and total bin hours in heating BHH, and cooling BHC, for each climate. The cooling design temperature is 95°F (35°C) for both climates.

4.3 BUILDING LOADS

Heating and cooling requirements are determined for residential applications and for light commercial/industrial applications by assuming a linear relationship between building load and outdoor temperature. The heating and cooling load lines extend from zero load at a specified change-over temperature T_c, to values which equal the steady-state heating and cooling capacities at assumed outdoor design temperatures of HDT and 95°F (35°C), respectively. The building load-temperature relationship is given by:

$$BL(T_j) = \frac{\dot{Q}_{ss}(HDT)\,[5j - 2]}{T_c + 5} \qquad T_j < T_c$$

$$BL(T_j) = \frac{\dot{Q}_{ss}(95)\,[5j - 3]}{95 - T_c} \qquad T_j > T_c \tag{4.1}$$

where j = 1, 2, 3,...,n. $\dot{Q}_{ss}$ (HDT) and $\dot{Q}_{ss}$ (95) are the measured steady-state heating and cooling capacities of the system at the assumed design temperatures, and n represents the total number of non-zero temperature bins. T_j is the representative temperature of the jth bin and is given by:

$$T_j = T_c + 2 - 5j, \qquad T_j < T_c$$

$$T_j = T_c - 3 + 5j, \qquad T_j > T_c \tag{4.2}$$

The change-over temperature T_c, for residential applications is assumed to be 65°F (18.3°C). For light commercial/industrial applications, T_c was determined to be 55°F (12.8°C). This results from assuming a constant building internal load $\dot{Q}_i$ equal to one-third of the building shell cooling load at 95°F (35°C), or equivalently: $\dot{Q}_i$ = BL(95)/4. The residential heating and cooling load lines are illustrated in figures 1-3.

5. CALCULATION PROCEDURE

5.1 GENERAL

The steady-state heating and cooling capacity $\dot{Q}_{ss}(T_j)$ at each test point shall be calculated using the appropriate equations specified in section 3.7 of the ASHRAE Standard 37-69. For heat pump systems which may not have indoor-air circulating fans furnished as part of the system, their measured heating and cooling capacities shall be adjusted by adding or subtracting, respectively, 1250 Btu/hr per 1000 SCFM (0.78 watts per dm^3/s) of indoor air flow to or from the measured values. Similarly, 366 watts of electrical energy per 1000 SCFM shall be added to the total measured steady-state electrical power input $\dot{E}_{ss}(T_j)$.

The input fuel energy rate $\dot{Q}_f(T_j)$ under steady-state conditions at bin temperature T_j shall be determined from:

$$\dot{Q}_f(T_j) = \dot{m}_f(T_j) \cdot (LHV) \tag{5.1}$$

where $\dot{m}_f(T_j)$ is the fuel mass (or volume) flow rate at temperature T_j and is determined from the fuel quantity measured over the specified test period. LHV is the lower heating value of the fuel expressed on a mass (or volume) basis.

The steady-state coefficient of performance $COP_{ss}(T_j)$ of an engine-driven heat pump system based upon source energy shall be determined from:

$$COP_{ss}(T_j) = \frac{\dot{Q}_{ss}(T_j)}{\dot{Q}_f(T_j) + 3.413 \left[\frac{\dot{E}_{ss}(T_j)}{0.3}\right]} \tag{5.2}$$

Where $\dot{E}_{ss}(T_j)$ is the steady-state electrical power input at temperature T_j, 3.413 converts watts to Btu/hr, and the factor 0.3 is an approximation for the efficiency of electric power generation and transmission.

The steady-state capacity, input electric power, and input fuel energy at bin temperatures other than the test points shall be determined through linear interpolation of data from the test points immediately adjacent (above and below) to the bin temperature in question. In those instances where the bin temperature is either greater than or less than all the test points, the data shall be linearly extrapolated from the two closest points. For residential heating applications, table 3 indicates that steady-state tests are conducted at bin temperatures T_j where j = 4, 7, 10, 13, 16. For commercial/industrial applications, table 4 indicates heating tests are conducted at bin temperatures T_j where j = 2, 5, 8, 11, 14.

5.2 PART-LOAD (CYCLIC) PERFORMANCE

In order to determine the seasonal performance of engine-driven heat pump equipment, several part-load parameters must first be defined or developed. A heating or cooling load factor $X(T_j)$ is defined by:

$$X(T_j) = \begin{cases} 0 & ; \quad T_j < T_{off} \\ \dfrac{BL(T_j)}{\dot{Q}_{ss}(T_j)} & ; \quad BL(T_j) < \dot{Q}_{ss}(T_j) \\ 1 & ; \quad BL(T_j) \geq \dot{Q}_{ss}(T_j) \end{cases} \qquad (5.3)$$

where T_{off} is the outdoor temperature at which the compressor is automatically turned off during the heating season (if applicable).

The results of the cyclic and steady-state heating and cooling tests shall be used in the following equations to calculate (a) Total heating or cooling during one "off"/"on" cycle $Q_{cyc}(T_{cyc})$, (b) cyclic coefficient of performance for the test cycle $COP_{cyc}(T_{cyc})$, (c) cyclic degradation coefficient C_D, and (d) a part-load factor $PLF[X(T_j)]$ at bin temperature T_j. T_{cyc} denotes the dry-bulb temperature at which cycling tests are conducted (80°F and 47°F).

$$Q_{cyc}(T_{cyc}) = \frac{60\ \dot{V}\ c_{pa}\ \Gamma}{v_n'\ [1 + W_n]} \qquad (5.4)$$

$$\text{where } \Gamma = \int_{\text{(time indoor fan goes on)}}^{\text{(time indoor fan goes off)}} [T_H(t) \quad T_L(t)]\ dt$$

and $\dot{V}$, c_{pa}, v_n', W_n, $T_H(t)$, and $T_L(t)$ are defined in the nomenclature.

$$COP_{cyc}(T_{cyc}) = \frac{Q_{cyc}(T_{cyc})}{Q_f(T_{cyc}) + 3.413\left[\dfrac{E_{cyc}(T_{cyc})}{0.3}\right]} \qquad (5.5)$$

where $Q_f(T_{cyc})$ is the total input fuel energy during the complete 30 minute test cycle. It is based upon the mass (or volume) of fuel consumed and the fuel's LHV. $E_{cyc}(T_{cyc})$ is the total input electric energy measured during the test cycle. For those heat pump systems which may not have indoor-air circulating fans, the calculated heating or cooling, and the measured input electric energy during one cycle shall be adjusted for the effect of a fan. The correction procedure shall be the same as that described in section 5.1 for steady-state tests, except that the correction shall equal the total energy transferred during the time of fan operation instead of a fixed energy rate.

Heating and cooling tests of electric heat pumps have indicated (refs. 6 and 7) that the cyclic-to-steady-state COP ratio [$COP_{cyc}(T_j)/(COP_{ss}(T_j)$] at any temperature T_j decreases approximately linearly as the load factor $X(T_j)$ decreases. Assuming a similar variation for engine-driven heat pumps, a part-load degradation coefficient C_D shall be calculated as follows:

$$C_D = \frac{1-\left[\dfrac{COP_{cyc}(T_{cyc})}{COP_{ss}(T_{cyc})}\right]}{1-\left[\dfrac{Q_{cyc}(T_{cyc})}{0.5\dot{Q}_{ss}(T_{cyc})}\right]} \tag{5.6}$$

where $0.5\ \dot{Q}_{ss}(T_{cyc})$ is the total heating or cooling which would occur during the 0.5 hour test cycle if steady-state conditions existed. C_D is determined from the results of the steady-state and cyclic heating and cooling tests specified in section 2.3, and it is equal to one minus the zero-load factor intercept of the cyclic-to-steady-state COP ratio [$1 - COP_{cyc}(T_j)/COP_{ss}(T_j)$ at $X(T_j) = 0$]. Laboratory tests of electric heat pumps [ref. 6] have shown that C_D is constant over a wide range of load factors [$X(T_j)$], and is relatively independent of outdoor temperature. Denoting $COP_{cyc}(T_j)/COP_{ss}(T_j)$ as a heating or cooling part-load factor $PLF[X(T_j)]$, then C_D and the linear relationship between $PLF(X(T_j))$ and $X(T_j)$ may be used to express $PLF(X(T_j))$ as follows:

$$PLF(X(T_j)) = \frac{COP_{cyc}(T_j)}{COP_{ss}(T_j)} = 1 - C_D[1-X(T_j)] \tag{5.7}$$

This expression shall be used to determine the heating or cooling cyclic coefficient of performance $COP_{cyc}(T_j)$ at any temperature T_j.

5.3 PART-SPEED PERFORMANCE

The capacity, input electric energy, and input fuel energy of engine-driven heat pump systems are generally nonlinear functions of compressor speed as well as one another. The performance of variable-speed heat pumps at compressor speeds intermediate to the maximum and minimum speed shall be determined through the use of a normalized part-speed input energy factor ψ^i. When the total input fuel and electric energy at temperature T_j is denoted by $E_T^i(T_j)$, where superscript i designates intermediate speed (Z=i), then $E_T^i(T_j)$ may be determined from

$$E_T^i(T_j) = E_T^0(T_j) + \psi^i[E_T^1(T_j) - E_T^0(T_j)] \tag{5.8}$$

The superscripts 0 and 1 denote values at minimum speed (Z=0) and maximum speed (Z=1), respectively. ψ^i shall be determined from:

$$\begin{aligned} \psi^i &= \left[\frac{\psi^{1/2}}{\alpha^{1/2}}\right] \alpha^i \; ; \; 0 \leq \alpha^i \leq \alpha^{1/2} \\ \psi^i &= \psi^{1/2} + \left[\frac{1 - \psi^{1/2}}{1 - \alpha^{1/2}}\right] (\alpha^i - \alpha^{1/2}) \; ; \; \alpha^{1/2} < \alpha^i \leq 1 \end{aligned} \tag{5.9}$$

where α^i is a normalized part-speed capacity factor defined at temperature T_j by:

$$\alpha^i = \frac{BL(T_j) - \dot{Q}_{ss}^0(T_j)}{\dot{Q}_{ss}^1(T_j) - \dot{Q}_{ss}^0(T_j)} \tag{5.10}$$

The values of $\psi^{1/2}$ and $\alpha^{1/2}$ required in equation (5.9) are determined for both operating modes from equations (5.8 and 5.10), respectively. Their evaluation is based upon intermediate speed tests at Z=1/2 specified in section 2.2.3 and conducted at outdoor-dry-bulb temperatures of 80°F (26.7°C) and 17°F (-8.3°C).

Equations (5.8 - 5.10) indicate that ψ^i and α^i vary from zero to one and that the relationship between them is approximated by two straight-line segments with the discontinuity at Z=1/2. Experimental data indicate that the relationship for heating is different than for cooling, but that each is essentially independent of outdoor temperature.

The purpose of equations (5.8 - 5.10) is to provide means for estimating the total input fuel and electric energy $E_T^i(T_j)$ at any speed and temperature. Accordingly, these equations shall be used in the following calculation procedure:

(a) Determine $\psi^{1/2}$ and $\alpha^{1/2}$ from equations (5.8 and 5.10) for heating and cooling using measured values of $\dot{Q}_{ss}^Z(T_j)$ and $E_T^Z(T_j)$, (Z=0, 1).

(b) Determine α^i at intermediate speed Z=i from equation (5.10).

(c) Determine ψ^i from equation (5.9) based upon the results of (a) and (b).

(d) Determine $E_T^i(T_j)$ from equation (5.8).

5.4 SEASONAL PERFORMANCE FACTOR AND SEASONAL OPERATING COST

5.4.1 Single-Speed Operation

The total heating or cooling done in the jth temperature bin is

$$Q(T_j) = n_j \; BL(T_j) \tag{5.11}$$

where n_j is the number of heating or cooling hours in the jth bin, and $BL(T_j)$ and T_j are given by equations (4.1 and 4.2), respectively. The total heating or cooling done during a season is equal to the summation of the energy in each non-zero temperature bin and may be expressed as

$$Q = N \sum_{j=1}^{n} \frac{Q(T_j)}{N} = N \sum_{j=1}^{n} \left[\frac{n_j}{N}\right] BL(T_j) \tag{5.12}$$

n is the number of non-zero temperature bins, $N = \sum_{j=1}^{n} n_j$ is the total heating or cooling season hours in all the bins. (n_j/N) are the fractional bin hours and are listed in tables 3 and 4 for residential and commercial/industrial applications, respectively.

When the steady-state heating or cooling capacity at temperature T_j is greater than the building load $[\dot{Q}_{ss}(T_j) > BL(T_j)]$, the total input fuel and electric energy in the jth temperature bin is:

$$E_T(T_j) = \frac{n_j \; BL(T_j)}{COP_{cyc}(T_j)} \tag{5.13}$$

Equations (5.3, 5.7, and 5.2) may be used to express equation (5.13) in terms of the load factor $X(T_j)$, part-load factor $PLF[X(T_j)]$, and the input fuel and electric energy $\dot{Q}_f(T_j)$ and $\dot{E}_{ss}(T_j)$. Accordingly:

$$E_T(T_j) = \frac{n_j \; X(T_j)}{PLF(X(T_j))} \left[\dot{Q}_f(T_j) + \frac{3.413}{0.3} \; \dot{E}_{ss}(T_j)\right] \tag{5.14}$$

When $BL(T_j) > \dot{Q}_{ss}(T_j)$, equations (5.3 and 5.7) indicate that $X(T_j) = PLF(X(T_j)) = 1$. Equation (5.14) is, therefore, also applicable under these conditions (assuming that $T_j > T_{off}$). It does not, however, include the supplemental energy required when $BL(T_j) > \dot{Q}_{ss}(T_j)$. The total input fuel and electric energy (excluding resistance heat) for the entire heating or cooling season becomes (for $T_j > T_{off}$):

$$E_T = N \sum_{j=1}^{n} \frac{E_T(T_j)}{N} = N \sum_{j=1}^{n} \left[\frac{n_j}{N}\right] \left[\frac{X(T_j)}{PLF[X(T_j)]}\right] \left[\dot{Q}_f(T_j) + \frac{3.413}{0.3} \dot{E}_{ss}(T_j)\right] \quad (5.15)$$

During heating when $BL(T_j) > \dot{Q}_{ss}(T_j)$, the supplemental energy required in the jth bin is:

$$Q_R(T_j) = n_j \, [BL(T_j) - \dot{Q}_{ss}(T_j)]/\eta_s \quad (5.16)$$

η_s is the efficiency of producing the supplemental energy from primary fuel. If electrical resistance is used, it is assumed that η_s=0.3. The supplemental energy required for the entire heating season becomes:

$$Q_R = N \sum_{j=n'}^{n} \frac{Q_R(T_j)}{N} = N \sum_{j=n'}^{n} \left(\frac{n_j}{N}\right) [BL(T_j)-\dot{Q}_{ss}(T_j)]/\eta_s \quad (5.17)$$

where n' is the lowest temperature bin at which $BL(T_j) > \dot{Q}_{ss}(T_j)$. The seasonal performance factor SPF is defined as:

$$SPF = \frac{Q}{E_T + Q_R} \quad (5.18)$$

Substituting equations (5.12, 5.15, 5.17), into equation (5.18) yields:

$$SPF = \frac{\sum_{j=1}^{n} \left[\frac{n_j}{N}\right] BL\ (T_j)}{\sum_{j=1}^{n} \left(\frac{n_j}{N}\right) \frac{X(T_j)}{PLF(X(T_j))} \left[\dot{Q}_f(T_j) + \frac{3.413}{0.3} \dot{E}_{ss}(T_j)\right] + \sum_{j=n'}^{n} \left[\frac{n_j}{N}\right] \left[BL(T_j) - \dot{Q}_{ss}(T_j)\right] /\eta_s} \quad (5.19)$$

The second summation in the denominator is zero for cooling calculations. An estimate of the seasonal operating cost SOC, is given by:

$$SOC = N \sum_{j=1}^{n} \left[\frac{n_j}{N}\right] \frac{X(T_j)}{PLF(X(T_j))} \left[\dot{Q}_f(T_j)(C_F) + (C_E) \ \dot{E}_{ss}(T_j)\right]$$

$$+ \sum_{j=n'}^{n} \left[\frac{n_j}{N}\right] \left[\frac{C_s}{\eta_s}\right] \left[BL\ (T_j) - \dot{Q}_{ss}(T_j)\right] \quad (5.20)$$

C_F, C_E, and C_s denotes the cost of engine fuel ($/Btu), the cost of electric energy ($/Wh), and the cost of supplemental energy ($/Btu). If the supplemental energy is electric resistance heat, then $C_s = C_E$ and $\eta_s = 1$. As before, the second summation is zero during cooling. The average total bin hours for heating and cooling, and for either climate and for both building applications are given in table 5. The steady-state capacity $\dot{Q}_{ss}(T_j)$, input electric energy $\dot{E}_{ss}(T_j)$, and input fuel energy $\dot{Q}_{fs}(T_j)$ at bin temperatures other than the test points shall be determined from linear interpolation of the test data as described in section 5.1.

The residential or commercial/industrial SPF and SOC shall be determined from equations (5.19) and (5.20) for both the heating and cooling seasons, and for either climate. Table 6 is a calculation sheet which may be used for systematically calculating these seasonal parameters.

5.4.2 Two-Speed/Dual-Capacity Operation

The seasonal performance factor SPF and the seasonal operating cost SOC of those engine-driven systems with two-speed compressor capability, or with some other means of achieving dual-capacity operation, are determined from:

$$\mathrm{SPF} = \frac{\sum_{j=1}^{n} \left[\frac{n_j}{N}\right] BL(T_j)}{\sum_{j=1}^{n} \frac{E_T(T_j)}{N} + \sum_{j=n'}^{n} \frac{Q_R(T_j)}{N}} \tag{5.21}$$

$$\mathrm{SOC} = N\left\{\sum_{j=1}^{n} \frac{E_T'(T_j)}{N} + C_s \sum_{j=n'}^{n} \frac{Q_R(T_j)}{N}\right\} \tag{5.22}$$

where n_j, N, n', and C_s are defined in section 5.4.1, and $[n_j/N]$ are listed in tables 3 and 4. $E_T(T_j)$ represents the total fuel and electric energy input to the jth temperature bin, $E_T'(T_j)$ represents the total cost of that energy, and $Q_R(T_j)$ denotes any supplemental energy required. Evaluation of these quantities is discussed subsequently. The building load $BL(T_j)$ is given by:

$$\begin{aligned} BL(T_j) &= \dot{Q}_{ss}^{1}(HDT)(5j-2)/(T_c+5)\ , \quad T_j<T_c \\ BL(T_j) &= \dot{Q}_{ss}^{1}(95)(5j-3)/(95-T_c)\ , \quad T_j>T_c \end{aligned} \tag{5.23}$$

where the superscript 1 denotes high-speed (greatest capacity) operation, and superscript 0 will subsequently denote low-speed (smallest capacity) operation.

The steady-state heating and cooling capacities $\dot{Q}^{Z}_{ss}(T_j)$, electrical power input $\dot{E}^{Z}_{ss}$ and input fuel energy rate $\dot{Q}^{Z}_{f}(T_j)$ required in subsequent calculations at bin temperatures other than the test points shall be calculated (as described in section 5.1) through interpolation of the two test points adjacent to the bin temperature in question. As shown in figure 2, low speed (Z=0) tests are conducted over the 47°F to 17°F temperature range, and high speed (Z=1) tests are conducted over the 17°F to -13°F range. Tables 3 and 4 give the bin temperatures which correspond to the test points for residential and commercial/industrial applications, respectively. If the bin temperature in question lies outside the range of test points, the result shall be linearly extrapolated from the two closest test points.

$E_T(T_j)/N$, $E_T'(T_j)/N$, and $Q_R(T_j)/N$ required in equations (5.21 and 5.22) are determined according to the operating regimes I — IV discussed below:

Regime I:

$BL(T_j) < \dot{Q}^{0}_{ss}(T_j)$ and the system cycles between off and low-speed (smallest capacity) operation in order to meet the building load. It is assumed that $T_j > T_{off}$.

$$\frac{E_T(T_j)}{N} = \left[\frac{n_j}{N}\right] \frac{X^0(T_j)}{PLF^0[X^0(T_j)]} \left[\dot{Q}^{0}_{f}(T_j) + 3.413 \left[\frac{\dot{E}^{0}_{ss}(T_j)}{0.3}\right]\right] \qquad (5.24)$$

$$\frac{E_T'(T_j)}{N} = \left[\frac{n_j}{N}\right] \frac{X^0(T_j)}{PLF^0[(X^0(T_j)])} \left[\dot{Q}^{0}_{f}(T_j)(C_F) + (C_E)\,\dot{E}^{0}_{ss}(T_j)\right] \qquad (5.25)$$

$$\frac{Q_R(T_j)}{N} = 0 \qquad (5.26)$$

where $$X^0(T_j) = \frac{BL(T_j)}{\dot{Q}^{0}_{ss}(T_j)} \qquad (5.27)$$

$$PLF^0[X^0(T_j)] = C^0_D\,[1-X^0(T_j)] \qquad (5.28)$$

Regime II:

$BL(T_j) > \dot{Q}^1_{ss}(T_j)$ and the system operates continuously at high-speed (greatest capacity) with supplemental heat required during the heating mode to meet the building load.

$$\frac{E_T(T_j)}{N} = \left[\frac{n_j}{N}\right]\left[\dot{Q}^1_f(T_j) + 3.413 \left(\frac{\dot{E}^1_{ss}(T_j)}{0.3}\right)\right] \tag{5.29}$$

$$\frac{E_T'(T_j)}{N} = \left[\frac{n_j}{N}\right]\left[\dot{Q}^1_f(T_j)\,[C_F] + [C_E]\,\dot{E}^1_{ss}(T_j)\right] \tag{5.30}$$

$$\frac{Q_R(T_j)}{N} = \left[\frac{n_j}{N}\right]\left[\frac{BL(T_j) - \dot{Q}^1_{ss}(T_j)}{\eta_s}\right] \tag{5.31}$$

If $T_j \leq T_{off}$, $\dot{Q}^1_f(T_j) = \dot{E}^1_{ss}(T_j) = \dot{Q}^1_{ss}(T_j) = 0$, and $SPF = \eta_s$

Regime III:

$\dot{Q}^0_{ss}(T_j) \leq BL(T_j) \leq \dot{Q}^1_{ss}(T_j)$ and the system cycles between off and high-speed (greatest capacity) operation in order to meet the building load. It is assumed that $T_j > T_{off}$.

$$\frac{E_T(T_j)}{N} = \left[\frac{n_j}{N}\right]\frac{X^1(T_j)}{PLF^1(X^1(T_j))}\left[\dot{Q}^1_f(T_j) + 3.413\left[\frac{\dot{E}^1_{ss}(T_j)}{0.3}\right]\right] \tag{5.32}$$

$$\frac{E_T'(T_j)}{N} = \left[\frac{n_j}{N}\right]\frac{X^1(T_j)}{PLF^1(X^1(T_j))}\left[\dot{Q}^1_f(T_j)(C_F) + (C_E)\,\dot{E}^1_{ss}(T_j)\right] \tag{5.33}$$

$$\frac{Q_R(T_j)}{N} = 0 \tag{5.34}$$

where $$X^1(T_j) = \frac{BL(T_j)}{\dot{Q}^1_{ss}(T_j)} \tag{5.35}$$

$$PLF^1\,(X^1(T_j)) = 1 - C^1_D\,[1 - X^1(T_j)] \tag{5.36}$$

Regime IV:

$\dot{Q}^0_{ss}(T_j) \leq BL(T_j) \leq \dot{Q}^1_{ss}(T_j)$ and the system cycles between high-speed (greatest capacity) operation and low-speed (smallest capacity) operation. It is assumed that $T_j > T_{off}$.

$$\frac{E_T(T_j)}{N} = \left[\frac{n_j}{N}\right] \left\{ X^0(T_j) \left[\dot{Q}^0_f(T_j) + 3.413\left(\frac{\dot{E}^0_{ss}(T_j)}{0.3}\right)\right] + X^1(T_j) \left[\dot{Q}^1_f(T_j) + 3.413 \left(\frac{\dot{E}^1_{ss}(T_j)}{0.3}\right)\right] \right\} \tag{5.37}$$

$$\frac{E_T'(T_j)}{N} = \left[\frac{n_j}{N}\right] \left\{ X^0(T_j) \left[\dot{Q}^0_f(T_j)\ (C_F) + (C_E)\ \dot{E}^0_{ss}(T_j)\right] + X^1(T_j) \left[\dot{Q}^1_f(T_j)\ (C_F) + (C_E)\ \dot{E}^1_{ss}(T_j)\right] \right\} \tag{5.38}$$

$$\frac{Q_R(T_j)}{N} = 0 \tag{5.39}$$

$$\text{where } X^0(T_j) = \frac{\dot{Q}^1_{ss}(T_j) - BL(T_j)}{\dot{Q}^1_{ss}(T_j) - \dot{Q}^0_{ss}(T_j)} \tag{5.40}$$

$$X^1(T_j) = 1 - X^0(T_j) \ . \tag{5.41}$$

The residential and commercial/industrial SPF and SOC of two-speed/dual-capacity heat pump systems shall be determined from equations 5.21 and 5.22 for both seasons and for either climate using equations 5.23 - 5.41 as required. Table 7 is a calculation sheet which may be used for systematically calculating these seasonal parameters.

5.4.3 Variable-Speed Operation

The evaluation of the seasonal performance factor SPF and the seasonal cost of operation SOC of variable speed heat pumps is very similar to the procedure described in section 5.4.2 for two-speed heat pumps. The SPF and SOC are given by equations 5.21 and 5.22 , respectively. The building load $BL(T_j)$ is given by equation 5.23 , and the steady-state heating and cooling capacity $\dot{Q}_{ss}(T_j)$, electrical power input $\dot{E}_{ss}(T_j)$, and input fuel energy $\dot{Q}_f(T_j)$ are determined from linear interpolation of adjacent test points. Superscripts 0 and 1 shall designate minimum and maximum speed, respectively. The total input fuel and electric energy $E_T(T_j)/N$, its total cost $E_T'(T_j)/N$, and the supplemental heat required are determined according to operating regimes I-III. Regimes I $[BL(T_j) < \dot{Q}^0_{ss}(T_j)]$ and II $[BL(T_j) > \dot{Q}^1_{ss}(T_j)]$ are exactly the same as in section 5.4.2. $E_T(T_j)/N$, $E_T'(T_j)/N$, and $Q_R(T_j)/N$ are given by equations 5.24 - 5.26 for regime I and by equations 5.29 - 5.31 for regime II.

Regime III:

$\dot{Q}^0_{ss}(T_j) \leq BL(T_j) \leq \dot{Q}^1_{ss}(T_j)$, $T_j > T_{off}$, and the system is operating continuously at a speed such that $\dot{Q}^i_{ss}(Tj) = BL(T_j)$. Therefore, $Q_R(T_j)/N=0$ and $X(T_j)=PLF[X(T_j)]=1$. Superscript i denotes normalized intermediate-speed (Z=i). The total input fuel and electric energy are obtained from equation 5.8 .

$$\frac{E_T(T_j)}{N} = \frac{E^i_T(T_j)}{N} = \frac{E^0_T(T_j)}{N} + \psi^i \left[\frac{E^1_T(T_j)}{N} - \frac{E^0_T(T_j)}{N}\right] \tag{5.42}$$

$$\frac{E_T'(T_j)}{N} = \frac{E^i_T(T_j)'}{N} = \frac{E^0_T(T_j)'}{N} + \psi^i \left[\frac{E^1_T(T_j)'}{N} - \frac{E^0_T(T_j)'}{N}\right] \tag{5.43}$$

where
$$\left.\begin{aligned} \frac{E^0_T(T_j)}{N} &= \left[\frac{n_j}{N}\right]\left[\dot{Q}^0_f(T_j) + 3.413\left(\frac{\dot{E}^0_{ss}(T_j)}{0.3}\right)\right] \\ \frac{E^1_T(T_j)}{N} &= \left[\frac{n_j}{N}\right]\left[\dot{Q}^1_f(T_j) + 3.413\left(\frac{\dot{E}^1_{ss}(T_j)}{0.3}\right)\right] \end{aligned}\right\} \tag{5.44}$$

$$\left.\begin{aligned} \frac{E^0_T(T_j)'}{N} &= \left[\frac{n_j}{N}\right]\left[\dot{Q}^0_f(T_j)(C_F) + (C_E)\,\dot{E}^0_{ss}(T_j)\right] \\ \frac{(E^1_T(T_j)'}{N} &= \left[\frac{n_j}{N}\right]\left[\dot{Q}^1_f(T_j)(C_F) + (C_E)\,\dot{E}^1_{ss}(T_j)\right] \end{aligned}\right\} \tag{5.45}$$

ψ^i is the normalized part-speed input energy factor and is given by equation 5.9 as a function of the normalized part-speed capacity factor α^i. α^i, in turn, is determined at intermediate speed Z=i from equation 5.10. The values of $\psi^{1/2}$ and $\alpha^{1/2}$ required in equation 5.9 are determined for both heating and cooling from equations 5.8 and 5.10. Table 7 is a calculation sheet which may be used to systematically calculate the heating and cooling SPF and SOC of variable-speed systems for both seasons, and for either climate or building application.

5.5 FROST ACCUMULATION AND DEFROST CALCULATIONS

The frost degradation coefficient C_{def}, is defined by:

$$C_{def} = \frac{COP_{def}(32)}{COP_{ss}(32)} \qquad (5.46)$$

where $COP_{ss}(32)$ is the steady-state coefficient of performance at the 32°F (0°C) test point. It is determined from equation 5.2 using the calculation procedures described in section 5.1. $COP_{def}(32)$ is the coefficient of performance under frosting-defrost conditions, and is determined from:

$$COP_{def}(32) = \frac{Q_{def}(32)}{Q_f(32) + \frac{3.413}{0.3} E_{def}(32)} \qquad (5.47)$$

$Q_f(32)$ is the total input fuel energy during the complete frost accumulation and defrost test. It is based upon the mass (or volume) of fuel consumed and the fuel's LHV. $E_{def}(32)$ is the total input electric energy measured during the test. $Q_{def}(32)$ is the net heating done during the test period, and is determined from:

$$Q_{def}(32) = \frac{60\ \dot{V}\ c_{pa}\ \Gamma}{v_n'\ (1 + W_n)} \qquad (5.48)$$

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

and $\dot{V}$, c_{pa}, v_n', W_n, $T_{a2}(t)$, and $T_{a1}(t)$ are given in the nomenclature. The flowrate $\dot{V}$ shall be an average value calculated at several intervals throughout the heating portion of the test. For those systems without indoor-air circulating fans, the calculated heating done and the measured input electric energy shall be adjusted for the effect of a fan. The correction procedure shall be the same as that described in section 5.1 for steady-state tests, except that the correction shall equal the total energy transferred during the time of fan operation instead of a fixed energy rate.

6. RECOMMENDED RATING REQUIREMENTS

The objective of the procedures recommended in this study has been to provide fair and accurate methods for testing prototype engine-driven heat pump systems having different design characteristics, and for estimating their seasonal performance and seasonal operating cost. Inherent in this objective is a requirement that the results of the tests and calculations be reported in such a way that the performance of different prototype systems may be effectively compared. Table 8 is a rating sheet which requires that the system under test be rated relative to its:

(a) Steady-state capacity $\dot{Q}_{ss}(T_j)$, at the ARI rating points of 95°F (35°C), 47°F (8.3°C), and 17°F (-8.3°C), respectively.
(b) Steady-state coefficient of performance $COP_{ss}(T_j)$, at the ARI rating points.
(c) Seasonal performance factor SPF, for both heating and cooling.
(d) Seasonal operating cost SOC, for both heating and cooling.
(e) Frost degradation factor, C_{def}.

The seasonal parameters SPF and SOC shall be based upon the residential building application for those systems which have cooling capacities at the ARI rating point of 60,000 Btu/hr (17.6 kW) or less. The light commercial/industrial building application shall apply for those systems with cooling capacities greater than 60,000 Btu/hr (17.6 kW).

7. LIMITATIONS OF THE RECOMMENDED TEST AND RATING PROCEDURES

There are a number of limitations associated with the recommended test and rating procedures. Some are inherent in any test and rating program which attempts to characterize the performance of a system based upon a finite and small number of test points. These limitations are almost unavoidable and will not be discussed. The remaining limitations are more fundamental and indicate a need for further investigation. These are discussed below:

(a) As discussed in section 2.2 and illustrated in figures 1-3 and table 1, there are a large number of steady-state tests required. It is felt that these tests are required to adequately predict the performance of the system over a broad range of outdoor temperatures. As more operating data becomes available on engine-driven systems, however, it may be possible to eliminate some of the steady-state tests.

(b) Cyclic testing of different engine-driven heat pump systems is required under a variety of ambient conditions and cycling rates. These tests should confirm and improve the accuracy and generality of the procedure discussed in section 5.2 for determining cyclic performance degradation. It is conceivable that sufficient testing may lead to a default C_D which may be predicted based upon a system's design and operating mode. If such a prediction was possible, it might eliminate the need for costly and time-consuming cyclic testing.

(c) It was noted in section 2.4 that the ideal way to account for the deteriorating effects of frost on the outdoor coil would be to apply a correction factor to all steady-state performance data obtained within the frost range, but that insufficient data currently exists to establish that correction factor. Laboratory and field testing over a range of outdoor temperatures and relative humidities is therefore required in order to develop a general frost correction which may be effectively applied to a broad range of engine-driven heat pump systems.

(d) The recommended test and rating procedures have assumed that the engine's waste heat is diverted to the outdoor side during the cooling mode, and is therefore not utilized. While this approach is generally justified for engine-driven systems designed for northern climates where the cooling load is low relative to the heating load, it will be unfair to those systems which are located in southern climates and are designed to utilize the waste heat for such purposes as domestic or commercial hot water heating. Under these circumstances the test and rating procedures should be modified to credit the heat pump system with engine waste heat utilization. This will require changes in the instrumentation requirements, test requirements, and calculation procedures. In addition, seasonal performance factor and seasonal operating cost determinations will require the specification of a generalized domestic or commercial hot water heating load. Also required is the specification of an alternate means of supplying energy to heat the hot water when the heat pump system is on the "off" part of its operating cycle.

(e) The recommended test and rating procedures have been limited to prototype, air-to-air, mechanical compression heat pump systems. These procedures could readily be modified for application to possible future commercially produced systems. Considerable work would be required, however, to adapt these procedures to absorption systems, or to mechanical compression systems which do not use ambient air as the heat source or heat sink medium.

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TABLE 1. SUMMARY OF TEST REQUIREMENTS

	Outdoor DB/WB Temp. (F)	MODE OF OPERATION: SINGLE SPEED: STEADY STATE	SINGLE SPEED: CYCLIC	TWO SPEED: STEADY STATE	TWO SPEED: CYCLIC	VARIABLE SPEED: STEADY STATE	VARIABLE SPEED: CYCLIC	VARIABLE SPEED: Z=1/2	FROST ACCUMULATION AND DEFROST TEST
COOLING MODE									
Indoor DB = 80°F	*95/75	1		2		2			
Indoor WB = 67°F	80/64	1	2**	2	4**	2	2**	1	
	*47/43	1	1	1	1	1	1		
HEATING MODE	32/30	1		1		1			1**
INDOOR DB = 70°F	*17/15	1		2	1	2		1	
MAX. INDOOR WB = 60°F	2/-			1		1			
	-13/-	1		1		1			
TOTAL NUMBER OF TESTS		6	3	10	6	10	3	2	1
TESTING REQUIREMENTS	Section	2.2.1	2.3	2.2.2	2.3	2.2.3	2.3	2.2.3	2.4
TESTING PROCEDURES	Section	3.2	3.3	3.2	3.3	3.2	3.3	3.2	3.4
CALCULATION PROCEDURE	Section	5.1	5.2	5.1	5.2	5.1	5.2	5.3	5.5
SEASONAL PROCEDURE AND COST	Section	5.4.1		5.4.2		5.4.3			
PRODUCT RATING	Section	6		6		6			6

* ARI Standard Rating Points

** Includes a steady-state dry coil test; indoor DB=80°F; indoor WB < 57°F.

TABLE 2

TEST OPERATING AND TEST CONDITION TOLERANCES

READINGS	STEADY-STATE TESTS		PART LOAD (CYCLIC) TESTS		FROST ACCUMULATION AND DEFROST TESTS	
	Test Operating Tolerance	Test Condition Tolerance	Test Operating Tolerance	Test Condition Tolerance	Test Operating Tolerance During Heating	Test Condition Tolerance During Heating
Outdoor Dry-Bulb Temp. (F) Entering	2.0	1.0	4.0	2.0	2.0	1.0
Leaving	---	---	---	---	---	---
Outdoor Wet-Bulb Temp. (F) Entering	2.0 (-)*	1.0 (-)*	---	---	2.0	1.0
Leaving	2.0 (-)*	---	---	---	---	---
Indoor Dry-Bulb Temp. (F) Entering	2.0	1.0	4.0	2.0	2.0	1.0
Leaving	2.0	---	---	---	---	---
Indoor Wet-Bulb Temp. (F) Entering	2.0 (-)**	---**	---**	---**	2.0	---
Leaving	2.0 (-)**	---	---	---	---	---

* Dry-coil cooling tests

** For dry-coil cooling tests the indoor wet-bulb temperatures shall at no time exceed the value which results in the production of condensate on the indoor coil. For heating tests the maximum value of indoor entering wet-bulb temperature is 60°F (15.6°C).

TABLE 3

GENERALIZED CLIMATES FOR RESIDENTIAL APPLICATIONS

	Temperature Bin Number j	Representative Bin Temperature T_j (F)	Bin Temperature Range (F)	Fraction of Total Temperature Bin Hours, n_j/N	
				Northern Climate	Southern Climate
	1	62	60-64	.115	.155
	2	57	55-59	.106	.139
	3	52	50-54	.096	.136
	*#4	47	45-49	.092	.133
	5	42	40-44	.097	.129
H	6	37	35-39	.109	.115
E	* 7	32	30-34	.118	.091
A	8	27	25-29	.086	.052
T	9	22	20-24	.058	.026
I	#*10	17	15-19	.039	.013
N	11	12	10-14	.028	.005
G	12	7	5-9	.020	.002
	*13	2	0-4	.015	.001
	14	-3	(-5) - (-1)	.010	--
	15	-8	(-10) - (-6)	.006	--
	*16	-13	(-15) - (-11)	.003	--
C	1	67	65-69	.293	.213
O	2	72	70-74	.271	.238
O	3	77	75-79	.200	.201
L	4	82	80-84	.131	.153
I	5	87	85-89	.071	.110
N	6	92	90-94	.028	.059
G	7	97	95-99	.005	.023
	8	102	100-104	0	.003

* Steady-state testing point
\# Standard ARI rating point

TABLE 4

GENERALIZED CLIMATES FOR COMMERCIAL/INDUSTRIAL APPLICATIONS

	Temperature Bin Number j	Representative Bin Temperature T_j (F)	Bin Temperature Range (F)	Fraction of Total Temperature Bin Hours, n_j/N	
				Northern Climate	Southern Climate
	1	52	50-54	.124	.193
	#*2	47	45-49	.120	.189
	3	42	40-44	.126	.184
H	4	37	35-39	.141	.164
E	*5	32	30-34	.151	.129
A	6	27	25-29	.110	.074
T	7	22	20-24	.074	.036
I	#*8	17	15-19	.049	.023
N	9	12	10-14	.036	.007
G	10	7	5-9	.026	.003
	*11	2	0-4	.019	.001
	12	-3	(-5) - (-1)	.013	---
	13	-8	(-10) - (-6)	.008	---
	*14	-13	(-15) - (11)	.003	---
	1	57	55-59	.170	.125
C	2	62	60-64	.186	.139
O	3	67	65-69	.189	.156
O	4	72	70-74	.175	.175
L	5	77	75-79	.129	.148
I	6	82	80-84	.084	.113
N	7	87	85-89	.046	.081
G	8	92	90-94	.018	.044
	9	97	95-99	.004	.017
	10	102	100-104	---	.002

* Steady-state testing point
\# Standard ARI rating point

TABLE 5

SUMMARY WEATHER DATA FOR REPRESENTATIVE CLIMATES

Generalized Climate	Degree Days (DD)	Heating Design Temperature (F) (HDT)	Building Application	Avg. Total Bin Hours: Heating (BHH)	Avg. Total Bin Hours: Cooling (BHC)
Northern	7000	-5	Residential	6240	2497
			Comm./Ind.	4877	3860
Southern	3500	15	Residential	4800	3980
			Comm./Ind.	3388	5391

TABLE 6. CALCULATION SHEET FOR SINGLE-SPEED HEAT PUMPS

MODE OF OPERATION: ☐ Heating ☐ Cooling

BUILDING APPLICATION: ☐ Residential ☐ Commercial/Industrial

CLIMATE: ☐ Northern ☐ Southern

Bin Hours Heating: BHH = ________(HRS)

Bin Hours Cooling: BHC = ________(HRS)

Supplemental Heat Generating Efficiency: η_S = ________(%)

A	B	C	D	E	F	G	H	I
Bin Nbr. j	Bin Temp[1] T_j (F)	(n_j/N)[1]	Capacity[2] $\dot{Q}_{ss}(T_j)$ (kBtu/hr)	Input Power[2] $\dot{E}_{ss}(T_j)$ (kW)	Fuel Energy[2] $\dot{Q}_f(T_j)$ (kBtu/hr)	Building Load[3] $BL(T_j)$ (kBtu/hr)	Load Factor[4] $X(T_j)$	Part Load Factor[5] $PLF[X(T_j)]$
1								
2								
3								
.								
.								
.								
16								

$$(SPF)_{Heating} = \frac{\underset{(J)}{\Sigma}}{\underset{(K)}{\Sigma} + \underset{(L)}{\Sigma}} = \frac{(\quad\quad)}{(\quad) + (\quad)} = \_\_\_\_\_$$

$$(SPF)_{Cooling} = \frac{\underset{(J)}{\Sigma}}{\underset{(K)}{\Sigma}} = \frac{(\quad)}{(\quad)} = \_\_\_\_\_$$

$$(SOC)_{Heating} = BHH \left[\underset{(M)}{\Sigma} + \underset{(N)}{\Sigma}\right] = (\quad)[(\quad) + (\quad)] = \$\_\_\_\_\_$$

$$(SOC)_{Cooling} = BHC \left[\underset{(M)}{\Sigma} + \underset{(N)}{\Sigma}\right] = (\quad)[(\quad) + (\quad)] = \$\_\_\_\_\_$$

TABLE 6. (continued)

FUEL COST: C_F = ________ (\$/kBtu)

C_E = ________ (\$kWh)

C_S = ________ (\$/kBtu)

J	K	L	M	N
[(Heating or Cooling)/N][6] $Q(T_j)/N$ (kBtu/N)	[(Fuel + Electric Energy)/N][7] (kBtu/N)	[(Supplemental Energy)/N][8] (kBtu/N)	[(Fuel + Electric Energy Cost)/N][9] (\$/N)	[(Supplemental Energy Cost)/N][10] (\$/N)
Σ= (J)	Σ= (K)	Σ= (L)	Σ= (M)	Σ= (N)

Ref. 1: Table 3 or 4

Ref. 2: Interpolation of Test Data

Ref. 3: Use eqn. (4.1)

Ref. 4: Use eqn. (5.3)

Ref. 5: Use eqn. (5.7)

Ref. 6: $Q(T_j)/N = (\text{col. C})(\text{col. G})$

Ref. 7: $(\text{Fuel}+\text{Electric Energy})/N = \left[\frac{(\text{col.C})(\text{col.H})}{(\text{col.I})}\right]\left[\text{col.F} + \frac{3.413(\text{col.E})}{0.3}\right]$

Ref. 8: $(\text{Supplemental Energy})/N = (\text{col.C})(\text{col.G} - \text{col.D})/\eta_S$

Ref. 9: $(\text{Fuel + elect. energy cost})/N = \left[\frac{(\text{col.C})(\text{col.H})}{(\text{col.I})}\right]\left[(\text{col.F})(C_F) + (C_E)(\text{col.E})\right]$

Ref. 10: $(\text{Supplemental energy cost})/N = (C_S)(\text{col.L})$

TABLE 7. CALCULATION SHEET FOR TWO-SPEED/DUAL CAPACITY OR VARIABLE-SPEED HEAT PUMPS

SYSTEM TYPE: ☐ Two-Speed ☐ Variable Speed

MODE OF OPERATION: ☐ Heating ☐ Cooling

BUILDING APPLICATION: ☐ Residential ☐ Commercial/Industrial

CLIMATE: ☐ Northern ☐ Southern

Bin Hours - Heating: BHH = ________(HRS)

Bin Hours - Cooling: BHC = ________(HRS)

Supplemental Heat Generating Efficiency: η_S = ________(%)

A	B	C	D	E	F	G	H	I	J	K	L
Bin Nbr. j	Bin Temp[1] T_j (F)	(n_j/N)[1]	Capacity[2] $\dot{Q}^0_{ss}(T_j)$ (kBtu/hr)	Capacity[2] $\dot{Q}^1_{ss}(T_j)$ (kBtu/hr)	Input Power[2] $\dot{E}^0_{ss}(T_j)$ (kW)	Input Power[2] $\dot{E}^1_{ss}(T_j)$ (kW)	Fuel Energy[2] $\dot{Q}^0_f(T_j)$ (kBtu/hr)	Fuel Energy[2] $\dot{Q}^1_f(T_j)$ (kBtu/hr)	Building[3] Load BL(T_j) (kBtu/hr)	Regime[4]	Load Factor[5] $X^0(T_j)$
1											
2											
3											
.											
.											
.											
16											

$$(SPF)_{Heating} = \frac{\underset{(P)}{\Sigma}}{\underset{(Q)}{\Sigma} + \underset{(R)}{\Sigma}} = \frac{(\quad)}{(\quad) + (\quad)} = \underline{\qquad}$$

$$(SPF)_{Cooling} = \frac{\underset{(P)}{\Sigma}}{\underset{(Q)}{\Sigma}} = \frac{(\quad)}{(\quad)} = \underline{\qquad}$$

$$(SOC)_{Heating} = BHH\left[\underset{(S)}{\Sigma} + \underset{(T)}{\Sigma}\right] = (\quad)[(\quad) + (\quad)] = \$\underline{\qquad}$$

$$(SOC)_{Cooling} = BHC\left[\underset{(S)}{\Sigma} + \underset{(T)}{\Sigma}\right] = (\quad)[(\quad) + (\quad)] = \$\underline{\qquad}$$

TABLE 7. (Continued)

FUEL COST: C_F = ______(\$/kBtu)

C_E = ______(\$/kWh)

C_S = ______(\$/kBtu)

FOR VARIABLE-SPEED OPERATION:

$^{1/2}$ = ______(Heating), ______(Cooling)

$\Psi^{1/2}$ = ______(Heating), ______(Cooling)

M	N	O	P	Q	R	S	T
Load Factor[6] $X^1(T_j)$	Part Load Factor[7] PLF^0	Part Load Factor[8] PLF^1	[(Heating or Cooling)/N][9] $Q(T_j)/N$ (kBtu/N)	[(Fuel + Elect. Energy)/N][10] $E_T(T_j)/N$ (kBtu/N)	[(Supplemental Energy)/N][11] $Q_R(T_j)/N$ (kBtu/N)	[(Fuel + Elect. Energy Cost)/N][12] $E'_T(T_j)/N$ (\$/N)	[Supplemental Energy Cost/N][13] (\$/N)
			Σ = (P)	Σ = (Q)	Σ = (R)	Σ = (S)	Σ = (T)

Ref. 1: Table 3 or 4

Ref. 2: Interpolation of Test Data

Ref. 3: Use Eqn. (5.23)

Ref. 4: Regime I: $BL(T_j) < \dot{Q}^0_{SS}(T_j)$

Regime II: $BL(T_j) > \dot{Q}^0_{SS}(T_j)$

Regime III or IV: $\dot{Q}^0_{SS}(T_j) \leq BL(T_j) \leq \dot{Q}^1_{SS}(T_j)$

Ref. 9: $Q(T_j)/N$ = (Col. C)(Col. J)

Ref. 11: $Q_R(T_j)/N$ = (Col. C)[Col. J - Col. E]/η_S

Ref. 13: Supplemental Energy Cost/N = (C_S)(Col. R)

Ref No.	Term	Value of Equation to be used for: Two-Speed Regime I	II	III	IV	Variable-Speed Regime I	II	III
5	$X^0(T_j)$	5.27	0	0	5.40	5.27	0	--
6	$X^1(T_j)$	0	1	5.35	5.41	0	1	--
7	PLF^0	5.28	0	0	0	5.28	0	--
8	PLF^1	0	1	5.36	0	0	1	--
10	$E_T(T_j)/N$	5.24	5.29	5.32	5.37	5.24	5.29	5.42
12	$E'_T(T_j)/N$	5.25	5.30	5.33	5.38	5.25	5.30	5.43

TABLE 8

RATING SHEET FOR ENGINE-DRIVEN HEAT PUMP SYSTEMS

ENGINE TYPE: BRAYTON DIESEL RANKINE STIRLING OTHER____________

SYSTEM TYPE: SINGLE-SPEED TWO-SPEED VARIABLE-SPEED

BUILDING APPLICATIONS: RESIDENTIAL COMMERCIAL/INDUSTRIAL

BUILDING APPLICATIONS:

CLIMATE: NORTHERN SOUTHERN

FUEL COST: ELECTRIC____________$/kWh ENGINE________$/kBtu

SUPPLEMENTAL ________

PERFORMANCE PARAMETER	SYMBOL	UNITS	HEATING	COOLING
ARI	$\dot{Q}_{ss}(95)$	kBtu/hr		
STANDARD RATING	$\dot{Q}_{ss}(47)$	kBtu/hr		
CAPACITY	$\dot{Q}_{ss}(17)$	kBtu/hr		
ARI STANDARD	$COP_{ss}(95)$	-		
COEFFICIENT OF	$COP_{ss}(47)$	-		
PERFORMANCE	$COP_{ss}(17)$	-		
SEASONAL PERFORMANCE FACTOR	SPF	-		
SEASONAL COST OF OPERATION	SOC	$		
FROST DEGRADATION COEFFICIENT	C_{def}	-		

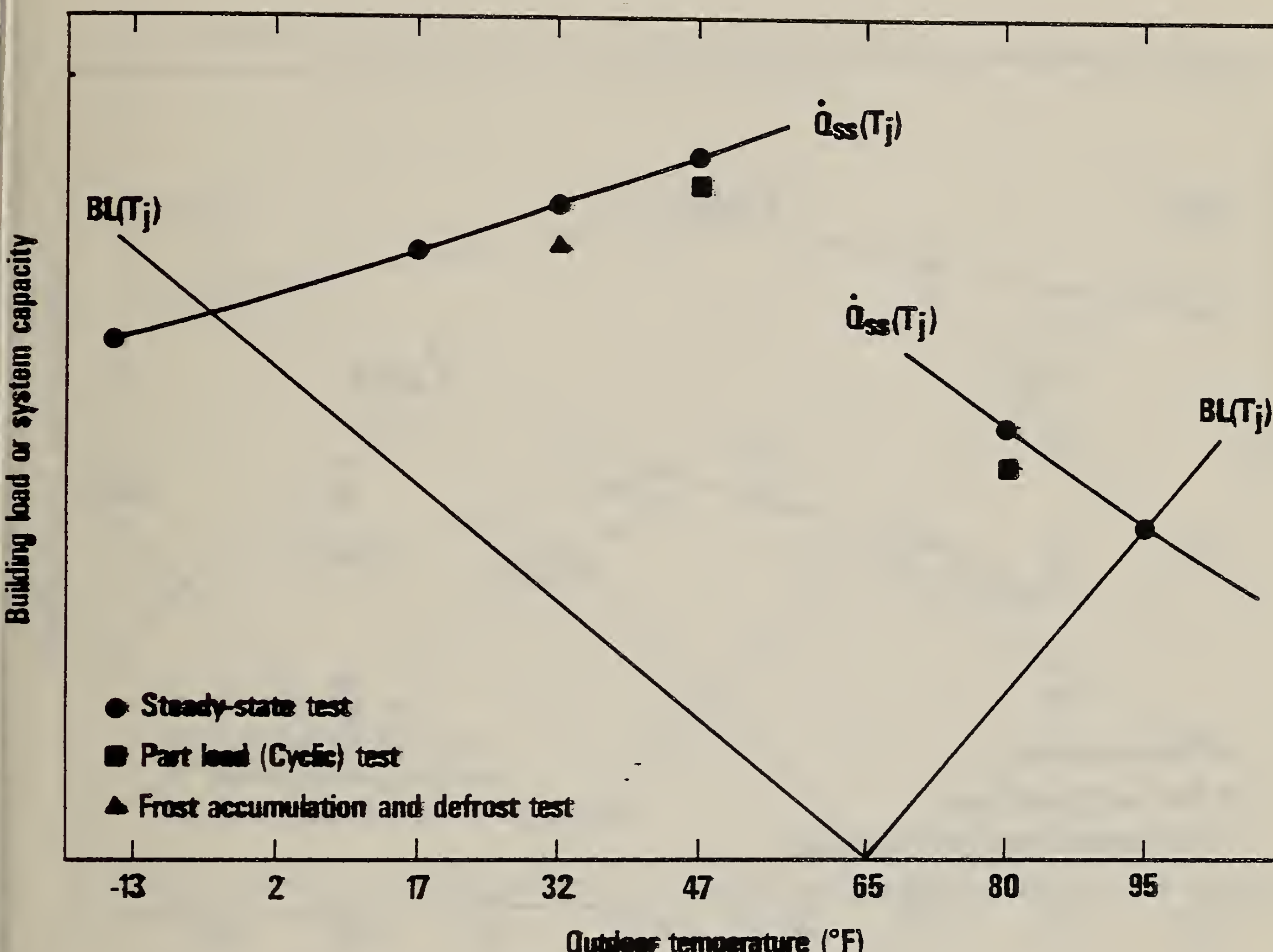

Figure 1. Test requirements for single-speed heat pump systems.

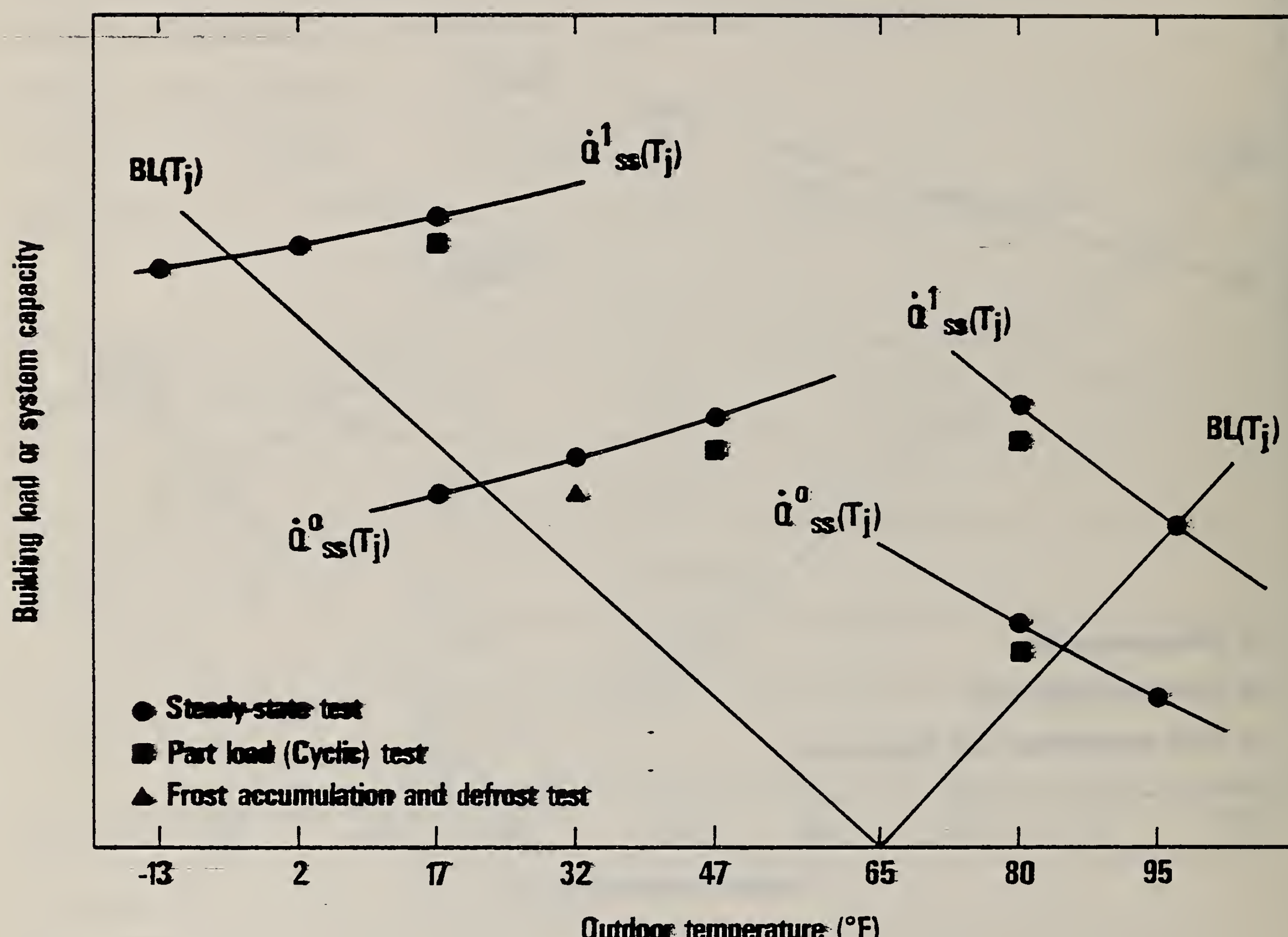

Figure 2. Test requirements for two-speed (dual capacity) heat pump systems.

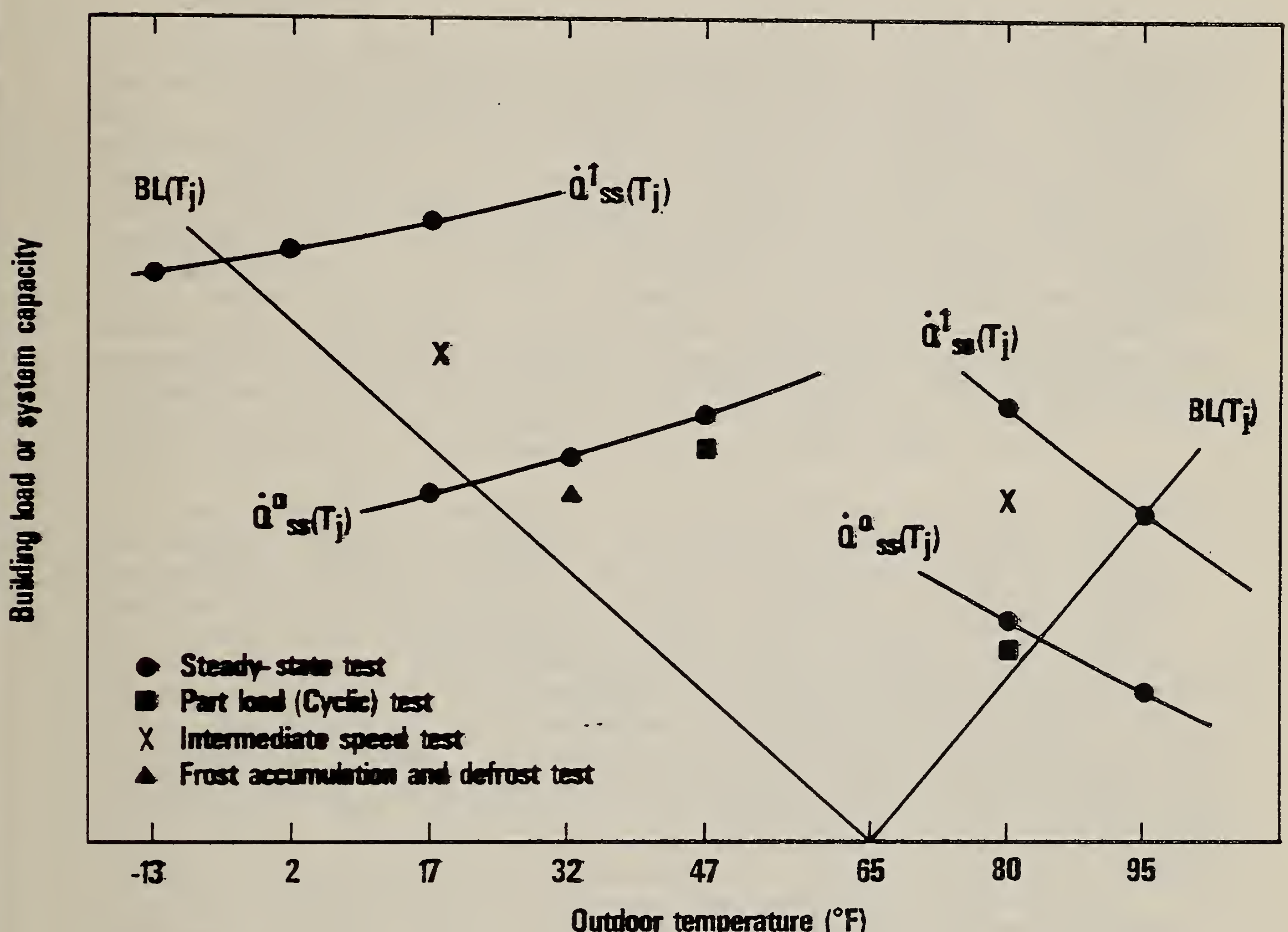

Figure 3. Test requirements for variable-speed heat pump systems.

APPENDIX A

PROPAGATION OF MEASUREMENT UNCERTAINTIES INTO THE STEADY-STATE COEFFICIENTS OF PERFORMANCE $COP_{SS}(T_j)$

A.1 Introduction

Questions often arise regarding the reliability of laboratory measurements and the calculated performance of equipment which is based on these measurements. In multi-sample measurements, sufficient readings of the variable are normally taken so that reliability can be established by statistical means. Unfortunately, the time required and the operating and personnel costs of many engineering experiments are often too great to permit multi-sample measurements. The experimenter is then restricted to single-sample measurements and the questions of reliability and measurement uncertainty often become significant. This appendix will discuss valid and acceptable methods of estimating and describing the uncertainties in single-sample measurements, as well as discuss a recommended method for calculating the propagation of these uncertainties into the results. These methods will then be applied to estimating the uncertainty associated with the steady-state coefficient of performance $COP_{SS}(T_j)$ given by eqn. (5.2).

In order to eliminate confusion regarding terminology, several fundamental concepts should be reviewed. As defined in ASHRAE Standard 41.5-75, the difference between the true value of a measured quantity and the observed value of the quantity is termed the "error" of the measurement. The "accuracy" of a measurement is indicated by the size of the error, and is often termed the systematic error. This type of error persists in all repetitions of the measurement and can be corrected through calibration. The "precision" of a measurement is indicated by how sharply the measurement is defined. It is an error which causes the readings to take random values on either side of some mean value, and is often termed the random error. The "uncertainty" of the measurement is the estimated value of the error. Although the uncertainty of a measurement may reflect both accuracy and precision errors (systematic or random) we will only consider precision errors, since it is presumed that accuracy errors could be eliminated through careful calibration. A "variable" will mean a basic quantity observed in the laboratory (pressure, temperature, volume, etc.), and its recorded values or readings will be called "data". The term "result" will refer to the value obtained by making corrections to or calculations with the data. "Propagation of uncertainty" shall mean the way in which uncertainties in the variables affect the uncertainty in the results.

The conventional means for stating the uncertainty of a variable is a statement of the best estimate of the true value together with a statement about the magnitude of the error in the estimate (uncertainty). The best estimate of the true value is normally the mean of the readings and the uncertainty is estimated based upon specified odds (confidence level). The recommended format for describing the best value of a variable is:

$$V = M \pm W, \text{ (P percent)} \tag{A1}$$

where V is the variable, M is the best value, W is the uncertainty, and P is the confidence level expressed as a percentage. The implication of eqn. (A1) is that the best value of V is M with a P percent probability that the true value lies within $\pm$ W of the value M. Determination of the uncertainty W with a given level of confidence (given odds) is often based upon the experience and judgment of the experimenter, and is a value selected such that he would be willing to wager (with P percent confidence) that the error is less than W. Eqn. (A1) provides a method by which the experimenter can describe the uncertainties associated with his basic measurements. It then becomes necessary to determine how these uncertainties propagate into the results.

If we let the result R be a function of n independent variables v_1, v_2, . . ., v_n, and each of the independently measured variables has an uncertainty Δv_1, Δv_2, . . ., Δv_n, then the uncertainty in the result ΔR is shown in Ref. 8 to be:

$$\Delta R = \left[\left(\frac{\partial R}{\partial v_1}\Delta v_1\right)^2 + \left(\frac{\partial R}{\partial v_2}\Delta v_2\right)^2 + \ldots + \left(\frac{\partial R}{\partial v_n}\Delta v_n\right)^2\right]^{1/2} \tag{A2}$$

The effect of large uncertainties in the variables is emphasized by this equation, and it shows that a specified reduction in a large uncertainty has a greater effect than a comparable reduction in a small uncertainty. It has been found from experience that a nondimensional form of eqn. (A2) is often more useful because it expresses the uncertainties on a percentage basis, and is generally easier to use. Dividing every term of eqn. (A2) by R gives:

$$\frac{\Delta R}{R} = \left[\left(\frac{\partial R}{\partial v_1}\frac{\Delta v_1}{R}\right)^2 + \left(\frac{\partial R}{\partial v_2}\frac{\Delta v_2}{R}\right)^2 + \ldots + \left(\frac{\partial R}{\partial v_n}\frac{\Delta v_n}{R}\right)^2\right]^{1/2} \tag{A3}$$

Eqns. (A2) and (A3) are important in instrumentation selection since they show the experimenter where large uncertainties occur, and he may then improve the instrumentation or experimental techniques in these areas.

A.2 Steady-State Heating Coefficient of Performance

The steady-state coefficient of performance $COP_{SS}(T_j)$ of an engine-driven system is given by eqns. (5.1) and (5.2), and is repeated here for convenience:

$$COP_{SS}(T_j) = \frac{Q_{SS}(T_j)}{\dot{m}_f(T_j)LHV + \frac{3.413}{.3}\dot{E}_{SS}(T_j)} \tag{A4}$$

The $COP_{SS}(T_j)$ is a function of several variables, each of which is subject to an uncertainty:

$$COP_{SS}(T_j) = f(\dot{Q}_{SS}(T_J), \dot{m}_f(T_j), LHV, \dot{E}_{SS}(T_j)) \tag{A5}$$

By forming the required derivatives $\frac{\partial COP_{SS}(T_j)}{\partial \dot{Q}_{SS}(T_j)}$, $\frac{\partial COP_{SS}(T_j)}{\partial \dot{m}_f(T_j)}$,

$\frac{\partial COP_{SS}(T_j)}{\partial LHV}$, and $\frac{\partial COP_{SS}(T_j)}{\partial \dot{E}_{SS}(T_j)}$, substituting into eqn. (A3) and simplifying,

it can be shown that:

$$\frac{\Delta COP_{SS}(T_j)}{COP_{SS}(T_j)} = \left[\left[\frac{\Delta\dot{Q}_{SS}(T_j)}{\dot{Q}_{SS}(T_j)}\right] + \left[\frac{LHV \cdot \Delta\dot{m}_f(T_j)}{\dot{m}_f(T_j)LHV + 3.413\,\dot{E}_{SS}(T_j)/.3}\right]\right.$$

$$\left. + \left[\frac{\dot{m}_f(T_j) \cdot \Delta LHV}{\dot{m}_f(T_j)LHV + 3.413\,\dot{E}_{SS}(T_j)/.3}\right]^2 + \left[\frac{3.413\,\Delta\dot{E}_{SS}(T_j)/.3}{\dot{m}_f(T_j)LHV + 3.413\,\dot{E}_{SS}(T_j)/.3}\right]^2\right]^{1/2} \tag{A6}$$

The term $\Delta\dot{Q}_{SS}(T_j)/\dot{Q}_{SS}(T_j)$ is determined for the heating mode with the procedure described below.

Using the appropriate equations given in ASHRAE Standard 37-69, the steady-state heating capacity $Q_{SS}(T_j)$ is:

$$\dot{Q}_{SS}(T_j) = 60 \cdot 1096\ CA_n\left[.24 + .444\ W_n\right]\left[T_{a2} - T_{a1}\right]\left[\frac{P_v P_n}{R_a T_n\left[1+W_n\right]}\right]^{1/2} \tag{A7}$$

where C is the discharge coefficient of the air flow measuring nozzle, A_n is the nozzle area in ft^2, P_n and T_n are the static pressure and temperature at the nozzle throat in in. Hg absolute and °R, respectively, P_v is the velocity pressure at the nozzle throat in in. H_2O, and R_a is the universal gas constant in units of (in. Hg $-ft^3$)/(lbm$-$°R). The remaining terms are defined in the nomenclature. Eqn. (A7) may be simplified by letting:

$$k = 60 \cdot 1096\ C$$

$$\beta = T_{a2} - T_{a1}$$

$$\alpha = \frac{.24 + .444W_n}{(1 + W_n)^{1/2}} \tag{A8}$$

$$\theta = \left[\frac{P_v P_n}{R_a T_n}\right]^{1/2}$$

$\dot{Q}_{SS}(T_j)$ then becomes:

$$\dot{Q}_{SS}(T_j) = KA_n\ \alpha\beta\theta \tag{A9}$$

Letting the uncertainties associated with the parameters A_n, α, β, and θ be denoted by ΔA_n, $\Delta\alpha$, $\Delta\beta$, and ΔQ, respectively, and assuming K is a constant, eqn. (A3) can be used to show that the nondimensionalized uncertainty in $\dot{Q}_{SS}(T_j)$ is:

$$\frac{\Delta\dot{Q}_{SS}(T_j)}{\dot{Q}_{SS}(T_j)} = \left[\left[\frac{\Delta A_n}{A_N}\right]^2 + \left[\frac{\Delta\alpha}{\alpha}\right]^2 + \left[\frac{\Delta\beta}{\beta}\right]^2 + \left[\frac{\Delta\theta}{\theta}\right]^2\right]^{1/2} \tag{A10}$$

The nondimensionalized uncertainties in α, β, and θ may be determined through the use of eqns. (A8) and (A3), and are:

$$\frac{\Delta\alpha}{\alpha} = \left[\frac{.111\ W_n + .162}{.222\ W_n^2 + .342\ W_n + .12}\right] \Delta W_n \tag{A11}$$

$$\frac{\Delta\beta}{\beta} = \frac{\Delta\left[T_{a2} - T_{a1}\right]}{\left[T_{a2} - T_{a1}\right]} \qquad \text{(A12)}$$

$$\frac{\Delta\theta}{\theta} = \left[\frac{1}{4}\left[\frac{\Delta P_v}{P_v}\right]^2 + \frac{1}{4}\left[\frac{\Delta P_n}{P_n}\right]^2 + \frac{1}{4}\left[\frac{\Delta T_n}{T_n}\right]^2\right]^{1/2} \qquad \text{(A13)}$$

Eqns. (A11 to A13) may now be substituted into eqn (A10) to yield $\Delta\dot{Q}_{SS}(T_j)/\dot{Q}_{SS}(T_j)$, and this in turn may be substituted into eqn. (A6) to give the final nondimensional expression for the uncertainty in the steady-state coefficient of performance. Accordingly:

$$\frac{\Delta COP_{SS}(T_j)}{COP_{SS}(T_j)} = \left[\left[\frac{\Delta A_n}{A_n}\right]^2 + \left[\frac{.111\ W_n + .162}{.222\ W_n^2 + .342\ W_n + .12}\right]^2 \Delta W_n^2\right.$$

$$+ \left[\frac{\Delta\left[T_{a2} - T_{a1}\right]}{\left[T_{a2} - T_{a1}\right]}\right]^2 + \frac{1}{4}\left[\frac{\Delta P_v}{P_v}\right]^2 + \frac{1}{4}\left[\frac{\Delta P_n}{P_n}\right]^2 + \frac{1}{4}\left[\frac{\Delta T_n}{T_n}\right]^2$$

$$+ \left[\frac{LHV \cdot \Delta\dot{m}_f(T_j)}{\dot{m}_f(T_j)LHV + 3.413\dot{E}_{SS}(T_j)/.3}\right]^2 \qquad \text{(A14)}$$

$$+ \left[\frac{\dot{m}_f(T_j) \cdot \Delta LHV}{\dot{m}_f(T_j)LHV + 3.413\ \dot{E}_{SS}(T_j)/.3}\right]$$

$$\left.+\left[\frac{3.413\ \Delta E_{SS}(T_j)/.3}{\dot{m}_f(T_j)LHV + 3.413\ \dot{E}_{SS}(T_j)/.3}\right]^2\right]^{1/2}$$

Eqn. (A14) shows that the uncertainty in $COP_{SS}(T_j)$ is a function of nine measured parameters, each with its own measurement uncertainty. Therefore,

$$COP_{SS}(T_j) = f\ (A_n,\ W_n,\ T_{a2} - T_{a1},\ P_v,\ P_n,\ T_n,\ \dot{m}_f(T_j),\ LHV,\ \dot{E}_{SS}(T_j)) . \qquad (A15)$$

A.3 Sample Calculation

As an illustration of the previous concepts, consider the following example taken from the experimental results of a Stirling engine-driven heat pump system operating at an outdoor temperature of 50°F. The measured values of the variables required in eqn. (A4) are:

$$\dot{Q}_{SS}(50) = 67{,}800 \text{ Btuh}$$

$$\dot{m}_f(50) = 2.44 \text{ lbm/hr}$$

$$LHV = 19{,}930 \text{ Btu/lbm}$$

$$\dot{E}_{SS}(T_j) = 820 \text{ W}$$

Substitution into eqn. (A4) yields $COP_{SS}(50) = 1.17$ as the calculated result. The uncertainty in this result is given by eqn. (A14). The uncertainties of the individual measurements were estimated with a 95% confidence level (19 to 1 odds) to be:

$$\frac{\Delta A_n}{A_n} = 0.01\ (.1\%)$$

$$\Delta W_n = 3.5 \times 10^{-5} \text{ lbm moisture per lbm dry air}$$

$$\frac{\Delta [T_{a2} - T_{a1}]}{[T_{a2} - T_{a1}]} = 0.01\ (1\%)$$

$$\frac{\Delta P_v}{P_v} = 0.005\ (0.5\%)$$

$$\frac{\Delta P_n}{P_n} = 0.01\ (1\%)$$

$$\frac{\Delta T_n}{T_n} = 0.005\ (0.5\%)$$

$$\Delta \dot{m}_f(T_j) = .0244 \text{ lbm/hr } (1\%)$$

$$\Delta LHV = 100 \text{ Btu/lbm } (0.5\%)$$

$$\Delta \dot{E}_{SS}(T_J) = 8 \text{ W } (1\%)$$

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15. SUPPLEMENTARY NOTES

Dr. Maxwell conducted this study during a year in residence at NBS under the IPA (Intergovernment Personnel Act) program.

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

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

A generic test and rating procedure is developed for heat engine-driven air-to-air heat pump systems. The procedures are classified according to whether the systems have single-speed, two-speed, or variable-speed capability, and whether they are operating in the heating or cooling mode. The test requirements generally consist of a series of steady-state tests to establish the impact of outdoor temperature on performance, two or more part-load (cyclic) tests to determine the effect of "on-off" cycling, two steady-state intermediate speed tests to determine part-speed performance, and a single frost accumulation test to estimate the effect of frost. A generalized calculation and rating procedure is developed. The system is rated in both modes based upon its steady-state performance at the ARI standard rating points, its seasonal performance factor, and its seasonal operating cost. A frost degradation coefficient is also established. The seasonal parameters are based upon either a residential or commercial/industrial building application which is located in either a generalized northern or southern climate.

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

Building heating and cooling; engine-driven heat pump; heat pump; heating and cooling equipment; heating, ventilating and air conditioning.

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