COMPARISON OF FIELD PERFORMANCE TO STEADY-STATE PERFORMANCE FOR TWO DEALER-INSTALLED AIR-TO-AIR HEAT PUMPS

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ABSTRACT

Two dealer-supplied and installed commercially available air-to-air heat pumps were tested under both steady-state and long-term field conditions at a site near Knoxville, Tennessee. Results from these tests indicate that in the Knoxville area during the heating season losses caused by duty cycling, frosting, and defrosting may exceed 40% in some spring months and cause seasonal efficiencies to fall about 20% below the efficiency of steady-state performance. Approximately another 5% decrease in efficiency results from resort to backup resistance heaters at low ambient temperatures. In the cooling mode, cycling losses reduce efficiency by about 10 to 20%.

INTRODUCTION

Heat pumps, which were installed in about 24% of new single family homes in 1980, are being marketed at about four times the 1973 levels according to a recent industry report [1]. In spite of this dramatic increase in use, there appears to be a dearth of information on the actual heating and cooling efficiencies of these systems under field conditions. Steady-state performance data for individual heat pump systems on the market are reported by the Air Conditioning and Refrigeration Institute (ARI), but, while these ratings are indicators of relative efficiencies between different models, they are not accurate indicators of seasonal performance for either heating or cooling. They are not because a heat pump's seasonal performance is degraded from its rated steady-state performance [2, 3, 4]. In the heating season, the degradation is caused by decreased efficiency at low outdoor temperatures, by frosting of the outdoor coil, by cycle reversal to defrost, by duty cycling, and by resort to electric resistance elements to supplement heating at low outdoor temperatures. In the cooling season, the degradation results from duty cycling and occasional high ambient air temperatures at the condenser. The National Bureau of Standards (NBS) has developed for the Department of Energy methods for estimating the seasonal performances of heat pumps which attempt to take into account degradation due to cycling and supplemental heating [5, 6, 7]. Heating seasonal performance factors for heat pumps, however, are not available in the literature. Cooling seasonal performance factors are currently being estimated by many heat pump manufacturers.

Except for some early tests on a heat pump in a house in the Washington, DC area [2], there appears to be little well-documented field data on the seasonal performance of air-to-air residential heat pump systems. The purpose of this paper is to report on the field performances of two very well instrumented, commercially available, dealer-installed heat pumps at a residential test site near Knoxville, Tennessee. Since 1976, the Tennessee Energy Conservation in Housing (TECH) complex, pictured in Fig. 1, has been used to obtain long-term

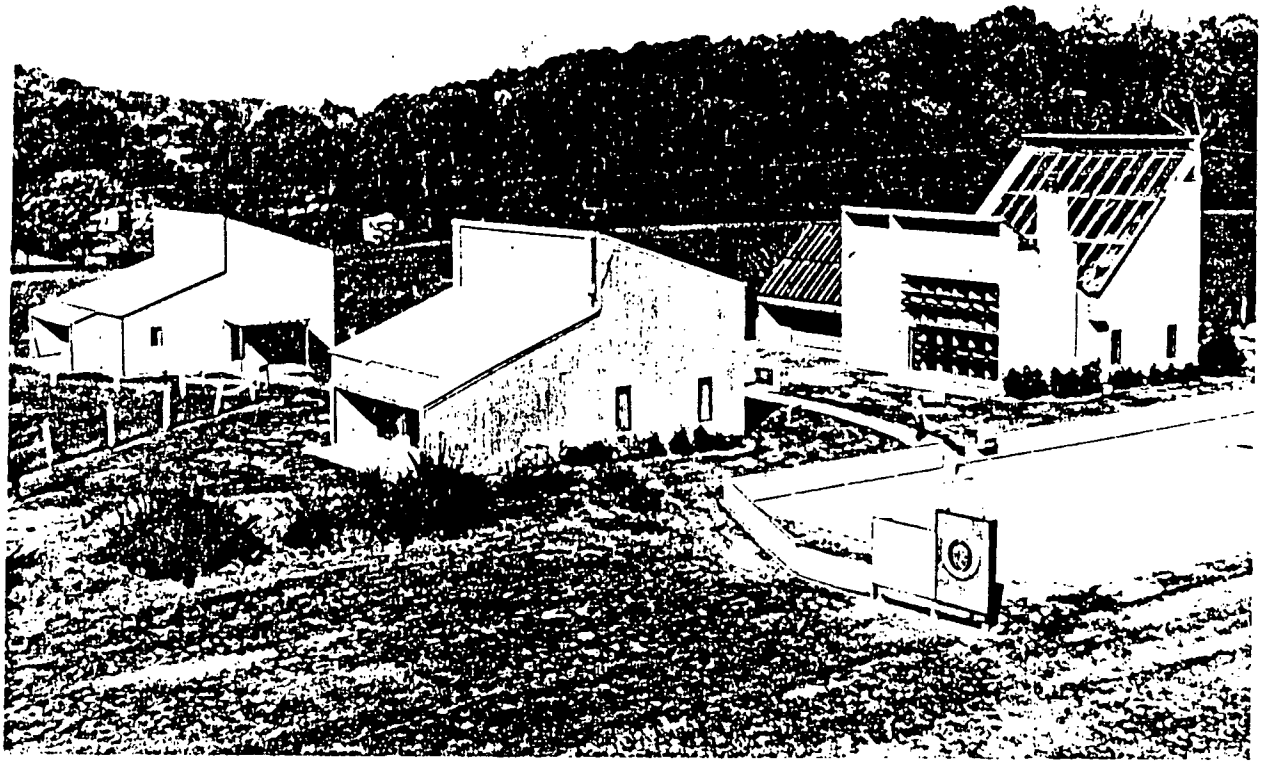

Figure 1. The TECH complex on the University of Tennessee Campus in Knoxville, Tennessee. From left to right, the Control House, the ACES House and the Solar House. Behind the Solar House is a garage fitted for additional solar experiments.

field performance data for a number of residential heating, cooling, and water heating systems. Systems tested have included, in addition to the conventional heat pumps reported here, the Annual Cycle Energy System (ACES) [8-11], an active solar energy heating system [12], and two solar assisted heat pumps [13].

TEST FACILITIES AND CONDITIONS

The TECH facility consists of three 167 m^2 well-insulated houses, two of which, the ACES House and a reference house called the Control House, are essentially identical while the third, the Solar House, has the same floor plan, volume, and construction details, but employs passive solar features and differs in roof line in order to accommodate the solar panels. For the purposes of this paper, only the ACES and Control Houses will be considered further. Neither the ACES nor the Control House was occupied, although a "package" simulating the loads imposed by a typical family of four (e.g., water usage, electrical appliances, etc.) was placed in each. Each house was monitored by a computer-controlled data acquisition system (DAS) [14] located in the ACES House, and data were collected and stored hourly. A weather station at the site provided concurrent information on wet and dry bulb outdoor temperatures, insolation, wind speed and direction, and rainfall. The thermostat in each house was set at 21°C (70°F) during the heating season and 26°C (78°F) during the cooling season.

The heat pumps tested in these experiments were representative of the commonly used air-to-air residential types which have as principal components a hermetically sealed compressor and motor, an outdoor and an indoor fan coil, a refrigerant expansion device, and a reversing valve. They were designed to operate in both the heating and cooling modes. The outdoor heat exchangers were of spine fin design, the indoor heat exchangers were of plate fin design, and R-22 was the refrigerant. As is conventionally done, the heat pumps selected for these tests were sized

to meet anticipated design day cooling loads for the test house. This is done to assure good humidity control during the cooling season, but it usually results in a compressor/heat exchanger combination that will not meet all of the cold season heating needs. To meet these needs, auxilliary electric resistance heaters are usually installed, and were in these tests.

Two thermostats were used to control the heating process. The outdoor thermostat, in the outdoor heat exchanger housing, determined the availability of the electric resistance elements (strip heaters) to supply supplemental or backup heating. This thermostat was set at 0°C; at outdoor temperatures above this the strip heaters could not be activated by a call for heating from the indoor thermostat except under an emergency setting. The other thermostat, wall-mounted indoors, had two stages. In the heating season, the heat pump operated at all indoor temperatures below the thermostat set point, 21°C. The second stage of the thermostat was activated at indoor temperatures about 2°C below the set point. When the second stage of the thermostat was activated, the supplemental strip heaters came on if the outdoor temperature was below 0°C. During defrosting of the outdoor heat exchanger, the strip heaters provided heat to the house. Nighttime setback of the thermostat was not employed. During the cooling season, compressor operation began whenever the indoor temperature exceeded the set point, about 26°C. In both the heating and cooling modes, the indoor fan was operated only when the compressor operated.

During the first full year of operation [10], 1977/78, the Control House was fitted with an air-to-air heat pump (details below), but only the backup resistance heating elements were used for space heating so that the house space heating loads could be measured directly. The ACES House heating loads in each year of these tests were also measured directly. This was done by the DAS, which continuously measured brine flow rates from the energy storage bin to the heat pump evaporator and

brine temperature differences across the storage bin heat exchanger. From these data, the DAS calculated the energy input to the evaporator and added to it all electrical energy inputs to the mechanical equipment. The resulting sum was the energy delivered to the house. Data were accumulated for each hour and then recorded on magnetic tape. Reference 14 gives further details on data acquisition. The time averaged seasonal heating loads for the two houses were found to be within about 2% of each other. In the succeeding years of the tests reported here, the Control House was heated by the heat pump, backed up as necessary during colder weather by the strip heaters, and the loads met were assumed to be equal to the loads measured at the ACES House. The cooling loads at the ACES House were calculated from measured brine flows to and temperature differences across the indoor cooling fan coil, with appropriate corrections for thermal leakage from pump and fan motors. During the first year of these tests, the cooling load of the Control House was assumed equivalent to that measured at the ACES House. This assumption was validated during the following summer when the Control House cooling loads were actually measured by means of an additional (refrigerant-to-liquid) heat exchanger inserted between the heat pump and the house air supply fan coil. The cooling loads of the Control House were found to be within about 3% of those of the ACES House [11]. The addition of the intermediate heat exchanger degraded the performance of the heat pump, and none of the data on heat pump performance collected while the exchanger was in place are used in this paper. To reiterate, in the heat pump results reported in this paper, the heating and cooling loads handled by the heat pumps are assumed equal to those measured for the same time periods for the ACES House.

Two commercially available heat pumps were tested, and their specifications are listed in Table I. Heat Pump I, utilized during the period 1977-79, had an ARI rated Coefficient of Performance (COP)* of 2.46 at 8.3°C and 1.7 at -8.3°C. This heat pump is typical of the most

*The Coefficient of Performance (COP) is defined as the useful energy delivered divided by the electrical energy required to deliver it in identical units. The COPs used in this paper include the electricity used by the indoor and outdoor fan motors as well as by the compressors and backup strip heaters.

efficient heat pump systems available in the period 1972-77 and is of the grade now normally installed by speculative builders. Heat Pump II, a high-efficiency model, had an ARI rated COP of 3.11 at 8.3°C and 2.07 at -8.3°C. This type of higher efficiency heat pump became generally available during the years 1979-80, but did then, and does now, command a premium price. No heat pumps with substantial improvements in steady-state efficiency have reached the market since the introduction of heat pumps of this type.

Table I. Description of heat pump systems tested at TECH complex during period 1977-80.

	HEAT PUMP I 1977-79	HEAT PUMP II 1979-80
Manufacturer	General Electric	General Electric
Model name	Weathertron	Weathertron II
Indoor unit	WE636G	BWE936C
Blower capacity	1200 CFM	1200 CFM
Outdoor unit	BGWA830R1B	BWR930A100A
Outdoor thermostat setting	0°C (32°F)	0°C (32°F)
Strip heaters	12 kW	9 kW
Indoor thermostat settings		
Heating	21.1°C (70°F)	21.1°C (70°F)
Cooling	25.6°C (78°F)	25.6°C (78°F)

Both of the units tested were delivered and installed by an authorized local dealer. The relationships of the steady-state heating capacities as a function of temperature to measured house loads are shown in Fig. 2, which shows that the house balance point, the temperature at which a continuously operating heat pump can just meet the house load, is about -2°C for Heat Pump I and -3°C for Heat Pump II. As detailed below, both heat pumps, upon installation, were tested for steady-state COPs, and these were found to agree closely with the ARI values.

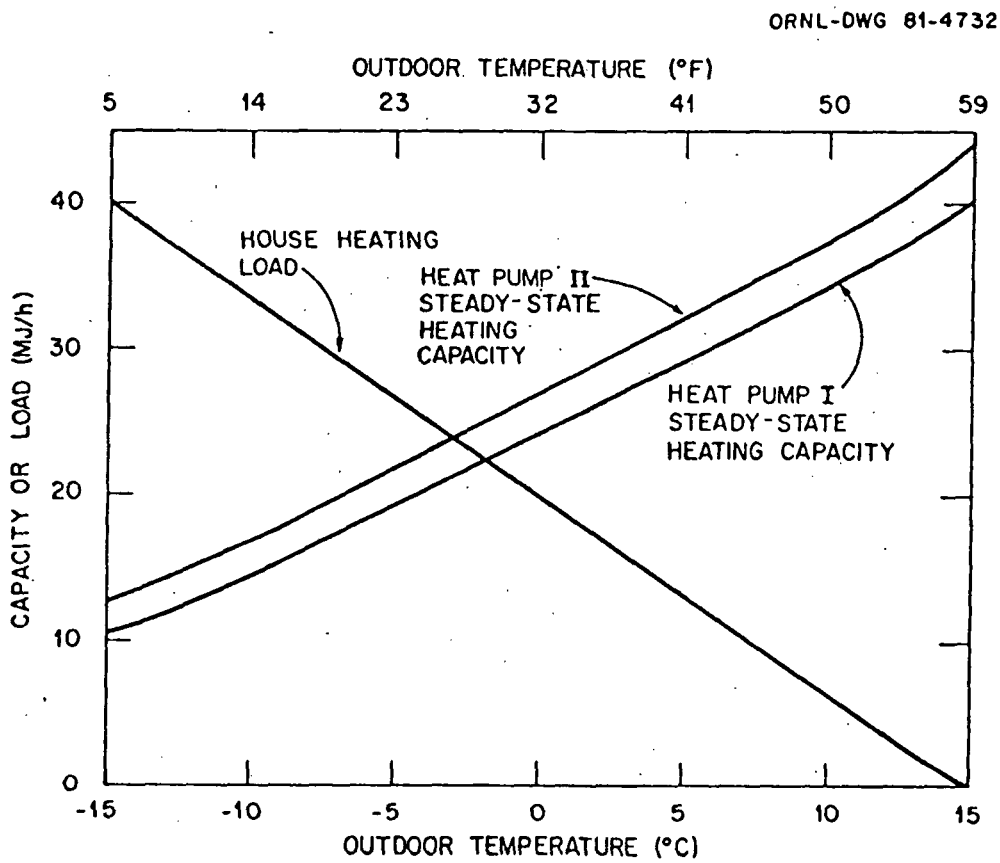

Figure 2. Control House heating load and calculated steady-state heating capacities of heat pumps I and II as a function of outdoor dry bulb temperature.

RESULTS

The ARI ratings and the steady-state performance test results for the two heat pumps are given in Table II. The deviations of field-measured COPs from the ARI values are small and reasonable in light of the deviations between the outdoor temperatures during measurements and the standard temperatures used for the ARI tests. These results indicate that the equipment was properly installed and operating at expected efficiencies.

Table II. ARI ratings and steady-state performance test results for heat pumps installed in the Control House.

	HEAT PUMP I		HEAT PUMP II		
	ON-SITE TEST	ARI	ON-SITE TEST 1	ON-SITE TEST 2	ARI
HEATING MODE					
OUTDOOR TEMPERATURE (°C)	11.1	8.3	9.7	8.0	8.3
HEAT OUTPUT (W)	10,856	9,085	10,892	10,328	0,965
OUTDOOR UNIT ENERGY USE (W)	3,680	—	3,147	3,086	—
INDOOR UNIT ENERGY USE (W)	530	—	395	389	—
TOTAL ENERGY USE (W)	4,210	3,700	3,542	3,475	3,200
SPACE HEATING COP	2.58	2.46	3.08	2.97	3.11
COOLING MODE					
OUTDOOR TEMPERATURE (°C)	31.3	35	35.2	26.5	35
COOLING CAPACITY (W)	7,500	8,795	9,379	10,424	9,818
OUTDOOR UNIT ENERGY USE (W)	—	—	3,411	3,037	—
INDOOR UNIT ENERGY USE (W)	—	—	405	404	—
TOTAL ENERGY USE (W)	4,075	4,300	3,816	3,441	3,900
SPACE COOLING COP	1.86	2.04	2.45	3.03	2.52

From May 1978 through April 1979, the Control House was heated and cooled by Heat Pump I. Performance data for the period are listed in Table III. Two outdoor dry-bulb temperatures are listed, one the average temperature during the month and the other the average temperature during only those hours in which the compressor operated. The house loads are

those met by the heating and cooling system, and the two electric energy usages listed are one, the total use for each month for heating and cooling, and two, the amount used monthly during the heating season by the electric resistance supplemental heaters. The COPs are based on total electricity used. No data are reported for the mild transition months of October and November because the ACES was being modified during this period and no house loads data were available. In general, demands on residential heating and cooling systems in the Knoxville area are very light in spring and fall transition months, and the omission of data for these months has very little effect on the evaluation of system performance.

Table III. Loads and performance data for Heat Pump I during period May 1978 through April 1979.

Period	Avg. Outdoor Temp. (°C) During Month	Avg. Outdoor Temp. (°C) During Hours of Operation	House Loads (GJ)[a]	Electric Energy Used (kWh) Supplemental	Electric Energy Used (kWh) Total	COP
COOLING SEASON						
May 1978	16.6	23.8	1.58	0	293	1.50
June	21.1	25.7	5.25	0	879	1.66
July	24.3	26.2	6.90	0	1,167	1.64
August	25.1	26.8	6.65	0	1,131	1.63
September[b]	24.4	26.2	3.52	0	581	1.68
SEASONAL	22.6	26.1	23.90	0	4,051	1.64
HEATING SEASON						
December 1978	3.3	1.8	7.29[c]	120[c]	1,211	1.67[c]
January 1979	-0.2	-0.9	12.12	336[c]	2,042	1.65
February	2.6	-0.4	8.41	221	1,592	1.47
March	10.5	4.9	3.92	14	727	1.50
April	14.4	8.6	1.11	4	214	1.44
SEASONAL	6.2	0.6	32.85	695	5,787	1.58

[a]Gigajoules; GJ = 0.948 x 10^6 Btu.
[b]Through September 18, 1978, only.
[c]Estimated.

From December 1, 1979 through September 15, 1980, the Control House was heated and cooled by Heat Pump II. Loads and performance data for this period are listed in Table IV. The test houses needed neither heating nor cooling during May 1980, and data for this month are not included.

Table IV. Loads and performance data for Heat Pump II during period December 1, 1979 through September 15, 1980.

Period	Avg. Outdoor Temp. (°C) During Month	Avg. Outdoor Temp. (°C) During Hours of Operation	House Loads (GJ)	Electric Energy Used (kWh) Supplemental	Electric Energy Used (kWh) Total	COP
HEATING SEASON						
December 1979	3.9	0.9	7.70	118	1,122	1.91
January 1980	4.7	1.9	8.35	160	1,090	2.13
February	0.8	-2.0	9.95	266	1,401	1.97
March	8.3	2.1	6.49	144	896	2.01
April	14.3	8.7	1.68	4	256	1.82
SEASONAL	6.4	0.9	34.18	692	4,765	1.99
COOLING SEASON						
June 1980	23.5	26.5	5.26	0	646	2.26
July	26.7	29.3	8.35	0	1,086	2.13
August	25.8	28.2	6.70	0	772	2.41
September[a]	24.5	27.3	2.39	0	270	2.46
SEASONAL	25.2	28.3	22.70	0	2,775	2.27

[a]Through September 15, 1980, only.

DISCUSSION

Monthly heating COPs, as a function of outdoor temperature during the hours of operation, are plotted for Heat Pumps I and II in Figs. 3 and 4, respectively. COPs with and without electric resistance supplemental

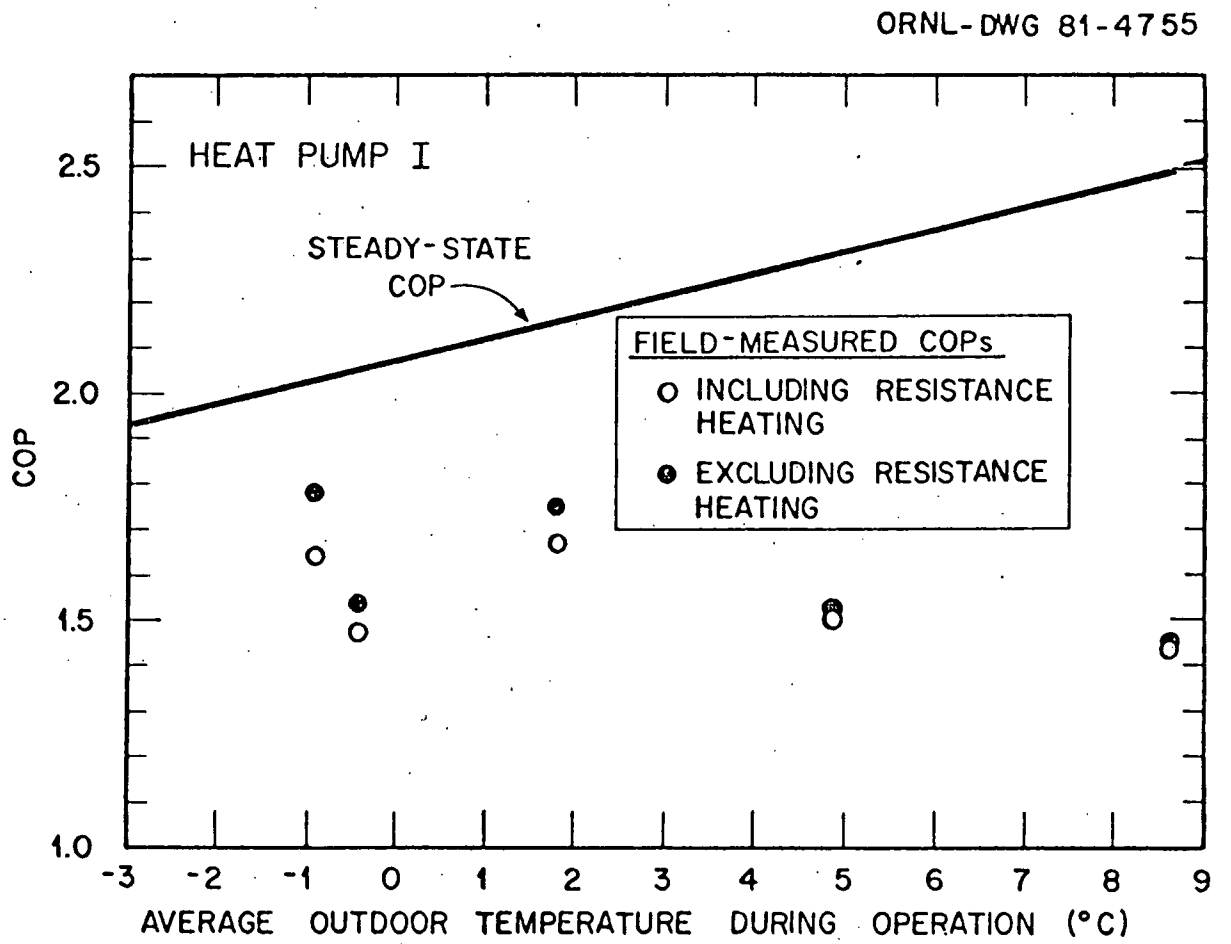

Figure 3. Heat Pump I heating season steady-state COP as a function of temperature (line) and field test COPs as a function of average temperature during compressor operation.

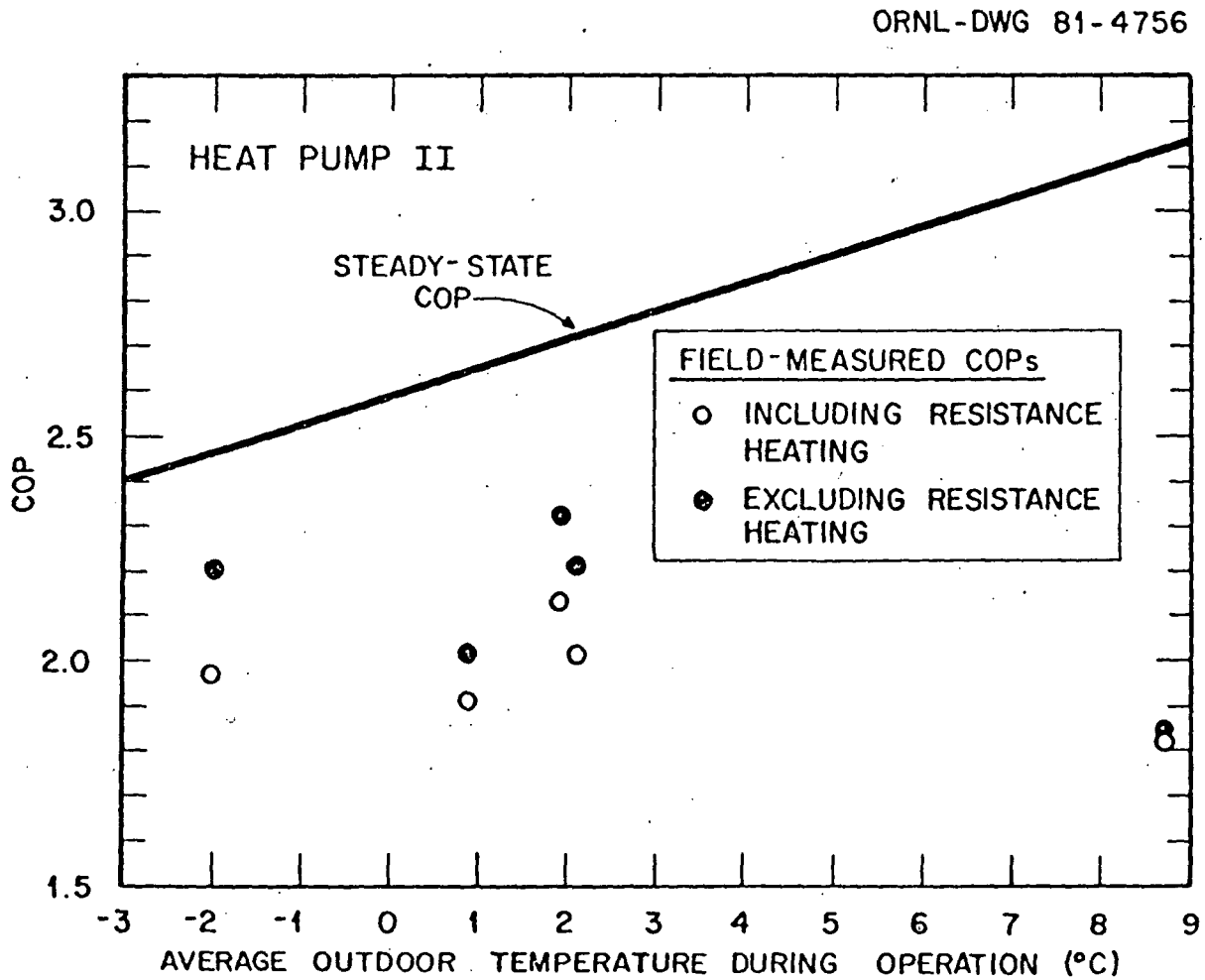

Figure 4. Heat Pump II heating season steady-state COP as a function of temperature (line) and field test COPs as a function of average temperature during compressor operation.

heating included are given, along with the estimated steady state COP values as a function of temperature. The latter do not include energy used by resistance heating elements. For both heat pumps, the field test values fall well below the steady state values for all temperatures encountered in these tests, and the discrepancy tends to become greater at higher temperatures. In fact, the monthly COPs appear to decrease in absolute value during warmer months, although steady-state COPs increase with temperature. As expected, at the higher temperature there is little separation between the COPs with and without backup resistance heating included, but heating system efficiency degradation from this cause does become appreciable at lower temperatures.

The degradation in heat pump efficiency from steady state values for both heat pumps is illustrated more clearly in Fig. 5, in which the percentage degradation from steady state values is plotted against the average outdoor temperature during the hours of operation in a given month. Both efficiencies and temperatures during operation are monthly averages. Between 8 and 9°C average operating temperature, the percent degradation from steady state values for both heat pumps exceeded 40%. The degradation decreased with temperature to 10 to 12% for the lowest average temperatures reached in this study (-2°C). There appears to be an anomolous increase in degradation in the temperature range -1°C to +1°C.

Since the values plotted in Fig. 5 do not include decreases in system efficiency caused by resort to electric resistance heating in cold weather, the degradation must be caused by frosting, defrosting, and duty cycling. Since losses caused by cycling should approach zero at the house balance point, which is about -2°C for Heat Pump I and -3°C for Heat Pump II (see Fig. 2), the losses seen at lower temperatures must be caused primarily by frosting and defrosting. W. A. Miller has shown recently [15] in laboratory tests that frosting and defrosting losses, with frosting losses predominating, may together decrease steady-state efficiencies by 25% or more in the ambient temperature range -2°C to +2°C, and have much less impact outside this range. It is likely that

such losses are reflected in the high degradation values around 0°C seen in Fig. 5. Miller's work shows that frosting/defrosting losses become negligible above 3 or 4°C, so the very large losses in efficiency found at higher average temperatures must be caused primarily by duty cycling.

The heating seasonal COPs for the system (heat pump plus resistance backup), commonly called the seasonal performance factors (SPF), also fell well below the ARI ratings. For Heat Pump I, it can be seen from Table III that the heating SPF is 1.58 at an average temperature during operation of 0.6°C. The ARI steady-state rating interpolates to about 2.1 at this temperature, indicating a 25% degradation in performance due to all causes. For Heat Pump II (Table IV), the SPF is 1.99 at an average temperature of 0.9°C, again down about 25% from the 2.64 value expected under steady-state operation. If the supplemental electric heating is subtracted from both the loads met and the electricity required, the SPF for heating for Heat Pump I becomes 1.66 at the 0.6°C average temperature. This is down 21% from the steady-state value. The heat pump only COP for Heat Pump II was 2.16 at 0.9°C average temperature, down 19% from the expected steady-state value. These values are essentially identical to those found by Kelly and Bean of NBS in their 1976 study [2].

The monthly cooling COPs listed in Tables III and IV are plotted as a function of average temperature during operation in Fig. 6. For each heat pump, the average COPs found in these field tests fall below the ARI 35°C steady-state values even though the average operating temperatures were appreciably lower. The cooling SPF for Heat Pump I at 1.64 (26.1°C) is 20% below the 35°C steady-state value, 2.04. For Heat Pump II, the field value was 2.27 (28.3°C), 10% below the 35°C steady-state value, 2.52. No DOE/NBS seasonal energy efficiency ratios (SEER) are available for the heat pumps used in these tests, but Parten et al. in some laboratory tests reported in 1977 [3] found degradations in efficiency of a 3-ton heat pump of from 10% at 80% compressor on time to 30% at 15% on time.

The ARI steady-state ratings indicate that Heat Pump II is about 24% more efficient than Heat Pump I in the cooling mode at 35°C. The field tests indicate that Heat Pump II was more than 38% more efficient

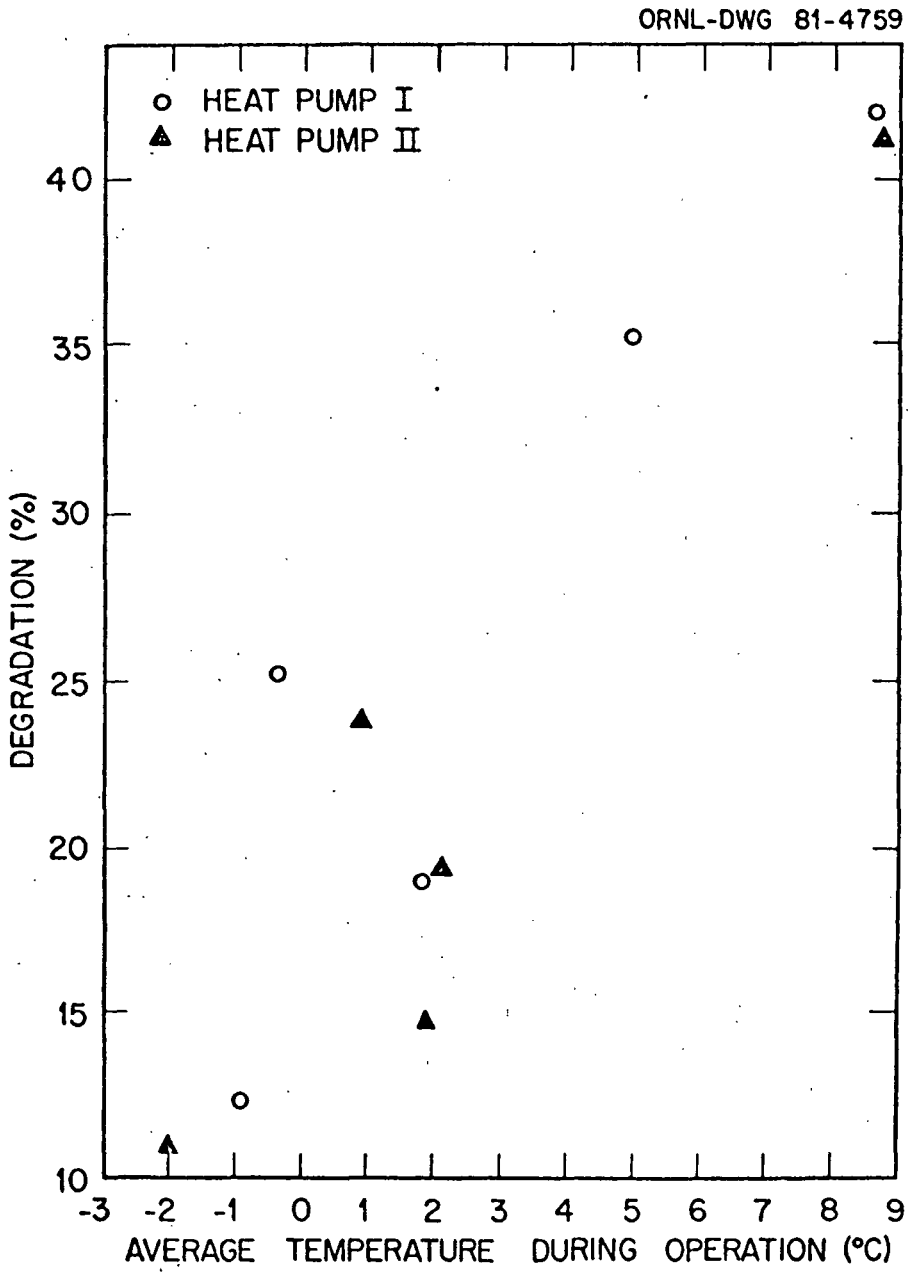

Figure 5. Degradation in heating efficiency from steady-state values as a function of average temperature during operation. Efficiencies and temperatures are monthly averages.

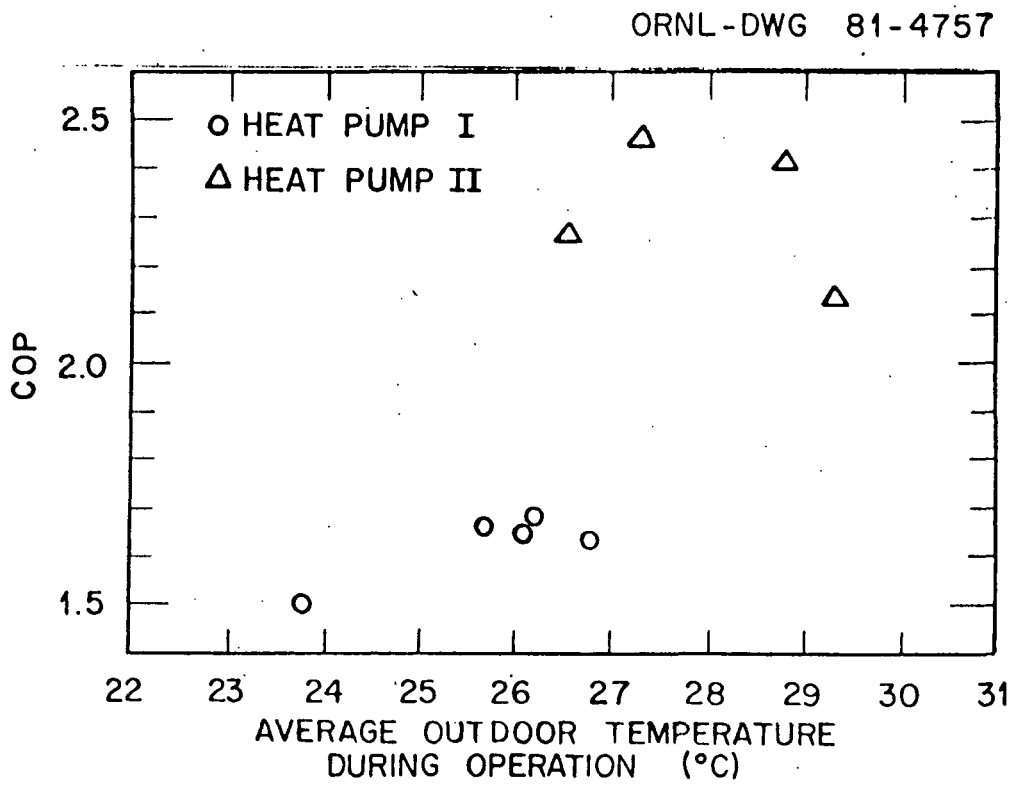

Figure 6. Cooling season COPs for Heat Pump I and Heat Pump II as a function of average temperature during compressor operation.

in seasonal cooling than Heat Pump I, even though the average temperature was 2.2°C higher. In the heating mode, Heat Pump II enjoys an efficiency advantage from about 22% (at -8.3°C) to about 26% (at 8.3°C) in steady-state operation. Again, that efficiency advantage is maintained under field conditions, with the advantage about 26% at approximately 1°C average temperature. These results are encouraging in that a consumer who pays a premium for higher heat pump efficiencies does indeed reap a benefit under field conditions.

SUMMARY

From these detailed long-term tests on a limited number of units, the following conclusions regarding air-to-air heat pumps can be reached:

(1) In the Knoxville, Tennessee area (1835 Centigrade degree-days heating), frosting, defrosting, and cycling losses degrade heating season performance by about 20% from steady-state values.

(2) In the same area, resort to electric backup heating during cold weather increases seasonal degradation by about 5%.

(3) Cycling losses play such an important part in heating season performance that heat pump system efficiency may actually decrease as ambient temperatures rise. In Knoxville, degradation from steady-state values exceeded 40% in April of both years of these tests.

(4) In the Knoxville area (845 Centigrade degree-days cooling), seasonal performance in the field in the cooling mode fell about 20% below steady-state values for Heat Pump I and about 10% below steady-state values for Heat Pump II. Most of these losses are due to cycling.

(5) The high efficiency heat pump tested here retained its efficiency advantage over the middle-of-the-line model under field conditions.

(6) Dealers can and do install heat pumps in a manner that allows the equipment to reach the manufacturer's performance specifications when operated under steady-state conditions.

In climates harsher than Knoxville, one would expect cycling losses during the heating season to decrease somewhat from the values reported here, but performance degradation due to resistance backup heating would be greater. Frosting and defrosting losses depend strongly upon relative humidity as well as upon climate, and areas with appreciably dryer winter-time climates would experience less degradation from these causes. In areas with milder winter climates than Knoxville, cycling losses would probably be greater while the others would be decreased.

ACKNOWLEDGEMENTS

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