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REPORT OF
AD HOC COMMITTEE

on the

APPLICATION OF MATHES HEAT PUMPS TO CAPEHART HOUSING

at

LITTLE ROCK AIR FORCE BASE

to

COMMANDING OFFICER
(ATTN: BASE PROCUREMENT OFFICER)
LITTLE ROCK AIR FORCE BASE
JACKSONVILLE, ARKANSAS

JAN 19 1959

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(Attn: Base Procurement Officer)
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Jacksonville, Arkansas

This report contains field and laboratory data obtained on the heating and cooling performance of two Mathes heat pumps installed in Capehart houses at Little Rock Air Force Base and recommendations and conclusions concurred in by the members of the Ad Hoc Committee set up to make this investigation, whose several signatures appear below.

C. Jack McDonald
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E. B. Penrod
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Frederick J. Reed
Fred J. Reed

Wm T. Smith
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P. R. Achenbach
P. R. Achenbach, Chairman

Report of
Ad Hoc Committee
on the
Application of Mathes Heat Pumps to Capehart Housing
at Little Rock Air Force Base

ABSTRACT

An Ad Hoc Committee of five members was established to carry out an agreement between the Little Rock Air Force Base and the Miles Construction Company that such a committee determine the adequacy of the Mathes Heat Pumps being installed in the Capehart housing project at Little Rock Air Force Base with respect to heating and cooling capacity. The Committee planned and conducted field heating and cooling tests in two selected houses at the project and laboratory heating and cooling tests of the two heat pumps from these houses at the laboratory of the Mathes Company in Marble Falls, Texas. One of the houses selected was a Type C house which appeared to impose the greatest heating load on the refrigeration system of the heat pump, and the other was a Type E house, so oriented as to have the greatest anticipated cooling load of any of the houses employing a single heat pump unit. The Committee concluded from the test results that the Model 27 heat pumps operating under cooling conditions would maintain an indoor temperature of 80°F in any of the houses, Type A through E, during a design summer day in which the dry bulb temperature did not exceed 105°F provided conservative use was made of the household electrical devices, such as the cooking range, washer, dryer, and electric lights, during the middle of the day, and in particular if such internal electrical load did not average more than about 5000 Btu/hr. The Committee further concluded that a Model 27 Mathes heat pump, together with strip heaters and bathroom heaters, operating under heating conditions would maintain an indoor temperature of 70°F or higher in houses of types A, A_1, B, D, and E during a design winter day in which the outdoor dry bulb temperature did not go below 5°F as a diurnal minimum, and will prevent the indoor air temperature in houses of types B_1 and C from falling below 70°F during the coldest part of such a design day if the heating system is operated in such a way as to make use of the heat stored in the structure and furnishings. The field heating tests of a type C house indicated that it had about 3200 Btu/hr less installed heating capacity than was required to maintain a constant indoor temperature of 70°F during a typical design winter day with 5°F minimum outdoor temperature. The Committee concluded that the heat capacity of the structure and furnishings in the Type C house was sufficient to prevent decrease of the indoor temperature at a rate greater than 1 degree per hour for 5 hours if the outdoor temperature did not fall below a minimum of 5°F. The Committee recommends that a fixed thermostat setting of

about 75°F be maintained day and night during the winter to make available the heat capacity of the structure and furnishings for sustaining the indoor temperature during temperature dips of a few hours' duration and to avoid the heavy electrical power demand that would occur every morning if all thermostats in the project were advanced at about the same time following night setback.

1.0 Introduction

By agreement between the Little Rock Air Force Base and the Miles Construction Company an Ad Hoc Committee was constituted to determine the adequacy of the Mathes Heat Pumps being installed in the Capehart housing project at Little Rock Air Force Base. The Committee was charged with determining by means of suitable analysis and field tests whether or not the heat pumps installed at Little Rock Air Force Base were able to heat the houses to 70°F inside for an outdoor design temperature of 5°F and cool the same houses to a temperature of 80°F and a relative humidity of 50 percent at an outside design temperature of 105°F. The Ad Hoc Committee consisted of:

E. B. Penrod, University of Kentucky
Fred J. Reed, Air Conditioning and Refrigeration Institute
W. T. Smith, U. S. Air Force
C. J. McDonald, Mathes Company
P. R. Achenbach, National Bureau of Standards

During some of the committee activities, C. W. Phillips, National Bureau of Standards, George S. Jones, Air Conditioning and Refrigeration Institute, and Leon F. Williams, U. S. Air Force, served as alternates for Achenbach, Reed and Smith, respectively.

2.0 Plan of Attack

At the initial meeting of the Committee, the following general plan of attack was agreed upon. Since the design outdoor conditions specified for summer and winter operation would be unlikely to occur during the test period, alternate methods for determining the ability of the heat pump units to perform at design conditions were devised. The procedure agreed upon consisted of the following steps:

a. Operate selected houses during hot sunny days at an interior temperature of 80°F, while maintaining 105°F air temperature at the inlet to the outdoor unit by artificial means, if necessary. Sufficient electrical resistance heat was to be provided indoors to cause the heat pump to operate steadily while maintaining the indoor temperature at 80°F. This test was to cover a period of 48 hours as a minimum to determine the excess cooling capacity of the heat pump units at the prevailing ambient temperature for comparison with the additional heat transmission loss that would be expected if the ambient temperature of the houses were 105°F.

b. Conduct one or more tests to determine, if possible, the heat storage effect of the floors, walls and ceiling of the house during the warmup period that might occur at peak cooling loads. Test houses were to be unfurnished for the tests.

c. Operate the selected houses at ambient temperatures preferably below 50°F with a thermostatted heat input sufficent to elevate the interior temperature to 85°F to determine the near steady state heat transmission coefficient of the buildings. The duration of the test was to be at least 48 hours. It was anticipated that the results obtained during the early morning hours would represent most nearly steady state conditions.

d. Conduct a heating capacity test of the heat pump unit from each house at the specified conditions of 70°F interior temperature and 5°F outdoor temperature. These tests were to be conducted at the Mathes Laboratory in Marble Falls, Texas.

e. Conduct a cooling capacity test of the heat pump unit from each house at nominal conditions of 80°F interior dry bulb temperature and 50% relative humidity and at 105°F outdoor temperature. These tests were also be to conducted at the Mathes Laboratory in Marble Falls, Texas.

The heat transmission factor determined in Item c of the above procedure was to be used after adjustment to determine whether the excess cooling capacity observed under Item a was sufficient to offset the additional cooling load that would occur at an ambient temperature of 105°F.

The heat transmission coefficient determined under Item c was also to be extrapolated to interior-exterior differences of 65°F for comparison with the heating capacity of the units as determined by the laboratory tests.

3.0 Meetings of the Ad Hoc Committee

The Ad Hoc Committee met for planning and review of the tests and test procedures at the following places and times:

1. Little Rock Air Force Base, Sept. 1 to Sept. 3, 1958

2. Little Rock Air Force Base, Sept. 8 to Sept. 9, 1958

3. Marble Falls, Texas, Mathes Co. Lab., Sept. 21 to Sept. 24, 1958

4. Little Rock Air Force Base, Oct. 30 to Nov. 1, 1958

5. Washington, D. C., Dec. 10 to Dec. 11, 1958

At the initial meeting of the Committee in Little Rock from Sept. 1 to Sept. 3, the general plan of attack outlined above was agreed upon and the houses to be used for the field tests were selected. At this same meeting, the NBS member, Achenbach, was selected as Chairman of the Committee and Phillips, his alternate, was selected as Acting Chairman in the absence of Achenbach. The National Bureau of Standards was requested by the Committee to furnish instruments for the field tests at the Little Rock Air Force Base and the Mathes Company was requested to furnish instruments and facilities for the laboratory tests at Marble Falls, Texas. Phillips and McDonald were designated to conduct and supervise the cooling tests at Little Rock and later to conduct and supervise the laboratory tests at Marble Falls and the heating tests at Little Rock. The method of reimsursement of the Committee for expenses incurred in connection with this study was discussed and agreed upon with the Miles Construction Company and the Little Rock Air Force Base.

The second, third and fourth meetings of the Committee were held to inspect and review tests and test procedures during the course of the various phases of the investigation. During the second meeting of the Committee it was agreed that normal operation of the cooling system in these houses would probably be with the thermostat set at about 75°F and that a temperature rise to the design temperature of 80°F during the peak load imposed on a day with the design outdoor temperature of 105°F would be acceptable. The anticipated heat storage effect of typical furnishings for such a 5°F rise in air temperature was to be computed, and a test to determine the heat storage of the building was to be made. The

heat storage effects thus determined were to be included, if necessary, in the determination of adequacy of the cooling capacity. At a later meeting, it was agreed that the heat storage effect of furnishings and building should be included, if necessary in the determination of adequacy of heating capacity, assuming that the thermostat setting would be 75°F and that the temperature in the house might drop to 70°F during a design day with a minimum temperature of 5°F. In each case, it was agreed to determine heat storage effects on the basis that a 5°F air temperature change would occur in 5 hours at a uniform rate of 1 degree per hour.

The final meeting in Washington, D. C. was held to review the test data, to reach conclusions on the recommendations to be made, and to agree on plans for issuing a report. At this meeting it was agreed that Achenbach and Phillips would draft a report which would be reviewed and approved by each member of the Committee before the final report was issued.

The field and laboratory tests were conducted during the following periods:

A. Cooling load tests at Little Rock Air Force Base,
Sept. 3 to Sept. 14, 1958

B. Laboratory tests at the Mathes Company laboratory,
Marble Falls, Texas,
Sept. 22 to Oct. 1, 1958

C. Heating tests at Little Rock Air Force Base,
Oct. 27 to Nov. 10, 1958

4.0 Test Procedure

4.1 Selection of Houses for Field Test

Unoccupied houses were selected for the field tests primarily to avoid interference with the normal living of the occupants, inasmuch as data were to be observed day and night. Such a selection also placed control of the miscellaneous internal electrical loads in the hands of the test personnel.

A Type E and a Type C house were selected by the Committee because the E type house appeared to have the highest cooling load of all types using a single heat pump unit and the Type C house

appeared to have the highest heating demand on the refrigeration system of the heat pump (over and above the installed strip heater capacity). These selections were based on a review by the Committee of the computations of heating loads made by the U. S. Air Force, the Mathes Co., and by Landauer and Shafer, and on the size and design of the several house types. The specific houses selected were C612 (left unit) located at 103 Illinois Drive and E642 located at 103 Indiana Drive.

Figures 1 and 2, respectively, show the front and rear of house C612. This house is a duplex and the unit tested is on the left in Fig 1 and on the right in Fig 2. Although the majority of type C units are constructed with little or no offset between the two halves of the building, it can be seen from Figs 1 and 2 that C612 has both a horizontal and vertical offset.

Figs 3 and 4, respectively, show the front and rear of house E642. This house is a single unit and is typical of the largest houses in the project which are equipped with a single heat pump unit.

The orientations of the two test houses, C612 facing northeast, and E642 facing east northeast, caused large window areas on the living room exposure of the house to face southwest and west southwest, respectively, thus contributing to a high afternoon cooling load. There was no grass or other landscaping around either house. The specimen houses were about 240 feet apart on ground sloping gently downward toward the northwest with the E house approximately 10 feet higher than the C house. The particular Type C house selected, having both a vertical and horizontal offset with respect to the other half of the duplex unit, was thought to have a heat transfer somewhat greater than a Type C house without one or both of these offsets.

All of the houses in the project were equipped with Mathes Model 27 heat pumps manufactured by the Mathes Company of Fort Worth and Marble Falls, Texas. House types A, A_1, B, B_1, C, D, and E were equipped with one such heat pump, house types F and G were equipped with two heat pump units.

Each heat pump system consisted of two separate pieces of equipment, an outdoor unit installed at the rear of the house and containing the compressor, outdoor coil and fan, and an indoor unit installed in the utility closet and containing the indoor coil, blower and resistance heating elements. The heat pump units installed in the two test houses were similar in all respects except for the number of nominal 1.8 KW resistance heating elements

in them. The unit in house C612 had two heating elements and the unit in house E642 had four heating elements, two of which were controlled by an outdoor thermostat set for about 30°F in series with the primary relay so that only two elements would be energized on house thermostat demand when the outdoor temperature was above the set point of the outdoor thermostat.

The nameplate data from the various sections of the two heat pumps were as follows:

House C612 (left unit)   Indoor unit

"The Mathes Company   FW
Division of Glen Alden Corporation
Model 27 TEB-HP
Ser. B590312
Unit Amps 18.2
Volts 230
Cycle 60
Phase 1
Blower Motor 1/4 HP 2.2 Amps
175 psi low side test pressure
400 " high " " "
$CHClF_2$ Refrigerant
All motors are equipped with overload protection"

House C612 (left unit)   Outdoor unit

"The Mathes Company   MF
Division of Glen Alden Corporation
Model 27 HAR-1E-HP
Ser. B801754
Unit Amps 14.3
Volts 230
Cycles 60
Phase 1
Compressor Horsepower 2.5
" Full Load Amps 11.6
" Locked Rotor Amps 52.4
Fan motor 1/3 HP 2.7 Amps
$CHClF_2$ Refrigerant
175 psi low side test pressure
440 psi high " " "
All motors are equipped with overload protection"

House E642 Indoor unit

"The Mathes Company FW
Division of Glen Alden Corporation
Model 27 TEB-HP
Ser. B590621
Unit Amps 34.2
Volts 230
Cycle 60
Phase 1
Fan motor 1/4 HP 2.2 Amps
$CHClF_2$ Refrigerant
175 psi low side test pressure
400 " high " " "
All motors are equipped with overload protection"

House E642 Outdoor Unit

"The Mathes Company MF
Division of Glen Alden Corporation
Model 27 HAR-1E-HP
Ser. B801740
Unit Amps 14.3
Volts 230
Cycle 60
Phase 1
Compressor Horsepower 2.5
" Full Load Amps 11.6
" Locked Rotor Amps 52.4
Fan motor 1/3 HP 2.7 Amps
$CHClF_2$ Refrigerant
175 psi low side test pressure
440 " high " " "
All motors are equipped with overload protection"

Figs 5, 6, and 7 show the indoor and outdoor units of the heat pump removed from house E642. In each view the indoor unit is on the left, the outdoor unit on the right. Fig 5 shows the air inlet and filter on the left and the air discharge opening on the top of the indoor unit and the air inlet and coil face of the outdoor unit. In Fig 6 the reverse side of the units can be seen. Fig 7 is a view of the internal construction of the two parts of the heat pump.

4.2 Cooling Load Tests

The cooling load tests were made between Sept. 3 and Sept. 14 to take advantage of the remaining hot weather of the summer season.

The Committee agreed to the following procedure for the cooling tests.

a. The outdoor unit would be operated in an ambient temperature of 105°F by surrounding it with a small enclosure that would permit controlled recirculation of the condenser air.

b. The house thermostat would be shunted to cause continuous operation of the compressor.

c. The return air to the indoor unit would be thermostatically controlled at 80°F by intermittently energizing the strip heaters in the unit or additional electric resisance heaters in the discharge plenum.

d. Actual and simulated internal loads were to be controlled as follows:

   1. Two men would live in each house and take 1 shower each early in the morning. The shower water was retained in the bath tub until it cooled to room temperature.

   2. An observer would be present in each house during the entire 24 hours of each day.

   3. Operate the electric range burners on the following schedule to simulate cooking loads:

      a. One 1250-watt burner for 15 min at about 8 AM

      b. One 1250-watt top burner and the oven thermostatted at 300°F for 30 min about noon.

      c. One 1250-watt top burner and the oven thermostatted at 300°F for 1 hour starting about 6 PM

      d. Kitchen ventilating fan for 30 min starting about 6:30 PM

      e. The observer could make coffee when desired using a top range burner.

   4. Operate washer and dryer for 1 hour about 10 AM to simulate washing loads (this load was simulated by operating the oven about 30 min because the

washing and drying equipment could not be installed in time for the tests).

5. Operate the dishwasher after noon and evening meals. (This load was simulated by extending the range oven operations for 30 min at noon and evening mealtimes, described in 3b and 3c above, because there were no dishes with which to load the dishwasher and the discharge of the hot water down the drain would have confused the evaluation of the heat released in the house).

6. Lights were to be used as required during the day. All inside lights were to be turned on from dark until 10 PM.

7. All windows were to be kept closed. The Venetian blinds on the west and south windows were to be positioned with the slats at 45° with the blinds fully down.

8. Doors were to be used as traffic demanded with prompt closing after use. The number of persons in the building was not to be restricted.

9. The refrigerator was to be in continuous operation.

10. The water heater was to be on automatic operation at a setting of about 140°F.

e. The following data were to be recorded at 30 min intervals continuously during the test:

1. Dry bulb temperature at the center of each room 60 in. above the floor.

2. Dry bulb temperature at inlet of indoor unit.

3. Wet bulb temperature or relative humidity at indoor unit inlet. (This observation was actually taken in the hallway where there was room to use a sling psychrometer).

4. Compressor discharge temperature and pressure.

5. Compressor suction temperature and pressure.

6. Outdoor temperature and humidity.

7. The electric energy used by heat pump, miscellaneous internal loads, and strip heaters for maintaining 80°F dry bulb in the house, metered so each of these three components could be determined separately.

8. Number of door openings.

9. Number of occupants in the building.

10. Other observations as required.

f. The evaporator fan and condenser fan speeds and the static pressure across the evaporator fan were to be measured at selected intervals. A continuous recording of outdoor dry bulb temperature was to be made in an open area generally between the two test houses. Continuous recordings were also to be made of house voltage and air temperature at the inlet to the indoor unit.

Data were observed and recorded continuously in accordance with the above described procedure during cooling operation for the period from Sept. 6 to Sept. 9, inclusive, on the Type E house and from Sept. 7 to Sept. 8, inclusive, on the Type C house. The outdoor temperatures did not approach 90°F during the day on the days following Sept 9 so the tests were terminated.

One test was made to obtain data on the heat capacity of the house structure under changing indoor temperature. For this test the indoor temperature was kept approximately equal to the outdoor temperature for a 24-hour period starting about 8 AM by controlling the electric energy input indoors. The outdoor temperature was essentially constant for about 12 hours starting about 6 PM on the previous evening and then rose rather rapidly after daybreak.

By maintaining equal temperatures indoors and outdoors there was no net heat transfer across the exterior walls or windows of the house and it could be assumed that the internal heat input represented heat stored in the internal partitions, floor, and in the inner half of the exterior walls. By measuring the electric energy supplied to the house under these conditions the heat storage capacity of the houses could be estimated.

A theoretical analysis was made of the heat storage effect of the interior furnishings of typical houses based on the gross weight of furnishings that the Federal Government will transport prepaid for Air Force families in the different type houses. This analysis was based on the stored heat in the furnishings (9000 lb in the Type C house and 10,500 lb in the Type E house) that would become available if the air temperature cooled at the rate of 1°F per hour for 5 hours.

Copper-constantan thermocouples made on site from calibrated wire were used to measure most of the temperatures, and readings were made with a balancing type electronic indicating potentiometer. The performance of the thermocouple system was monitored by observation of hot and cold reference bath temperatures. Electric energy measurements were made with integrating watthour meters calibrated at the conclusion of the cooling tests. Pressure measurements on the heat pump unit were made with calibrated test gages and static pressures in the air circuit were measured with a self-compensating micromanometer. Indoor and outdoor relative humidity determinations were made using sling psychrometers with 10-inch wet and dry bulb thermometers. Indoor blower and outdoor fan speeds were measured with a stroboscopic light instrument with built-in calibrator with observations being made through a glass-fitted panel on the indoor unit to permit normal static pressures during measurement.

4.3 Laboratory Tests of Heat Pump Units

The two heat pump units were removed from the test houses and shipped to Marble Falls, Texas, where capacity tests were made in the laboratory of the Mathes Co. under heating and cooling conditions using an apparatus patterned after the psychrometric calorimeter described in ASRE Standard 16-56 entitled, Methods of Rating and Testing Air Conditioners.

Cooling tests were made in the laboratory at conditions matching, as nearly as possible, those observed in the field tests at Little Rock Air Force Base. The conditions that were considered important in this regard were inlet air temperature to the outdoor unit, inlet air temperature and relative humidity to the indoor unit, compressor suction and discharge pressure, dry bulb temperature difference across the indoor coil, static pressure difference across the blower for the indoor unit, the refrigerant superheat

at the compressor suction, and line voltage. These conditions were reasonably well matched between field and laboratory tests with the exception of refrigerant superheat at the compressor suction and the rate of air flow through the indoor unit. Laboratory tests were made of each unit for a range of air circulation rates through the indoor coil.

Heating tests were made in the laboratory at the design indoor temperature of 70°F and the design outdoor temperature of 5°F for two different air flow rates through the indoor coil.

The laboratory values of heating and cooling capacity were based on measurements of mass air flow rate and enthalpy change of the air through the indoor unit. The manometer used to measure the air flow rate through the discharge orifice had been calibrated by the University of Texas within a week prior to the tests. Initially, the wet and dry bulb temperatures were measured with calibrated thermometers only and later with both the calibrated thermometers and thermocouples. The indicated readings of the thermocouples were checked against hot and cold reference baths using the calibrated thermometers as the reference instrument. Pressure gages were calibrated with a dead weight tester.

4.4 Heating Tests

The heating tests at Little Rock Air Force Base were conducted throughout the period from Oct. 29 to Nov. 9, 1958 after the weather had cooled enough to permit operation of the test houses at an indoor-outdoor temperature difference of 30 degrees F or more.

The purpose of the heating tests was to determine a heat transmission factor per unit indoor-outdoor temperature difference for each house at near steady-state conditions as a basis for extrapolation to the design winter indoor-outdoor temperature difference of 65°F and as a basis for correcting the observed heat transmission during cooling to the design outdoor temperature. For simplicity, electric resistance heating only was used to produce the desired indoor-outdoor temperature difference. Enough resistance heating was provided to produce an indoor temperature of 85°F for any outdoor temperature anticipated during the test period. The heaters were thermostatted to provide a con-

stant indoor temperature of 85°F. This constant indoor temperature was maintained day and night for about 11 days in order to permit the heat exchange between the room air and the concrete floor slab to approach more nearly a steady-state condition. The blower of the indoor unit was operated continuously during the heating tests.

Temperatures were observed at the following stations in each house during the heating tests:

a. Air entering indoor unit

b. Air leaving indoor unit

c. Outside air (average on four sides of house)

d. Air at center of each room 60-in. above the floor

e. Air at center of living room, kitchen, front bedroom, 1/2-in. above the floor

f. Floor surface at the center of living room, kitchen, front bedroom

g. Interior and exterior surface of outside wall of living room 60-in. above the floor

h. Concrete slab surface in utility closet

i. Concrete 2-in. below surface in utility closet

j. Concrete 4-in. below surface in utility closet

About midway through the test period a hole was bored through the concrete floor and into the fill beneath to a depth of 18-in. from the floor surface in each house. At this station in the utility closet, temperatures were measured by thermocouples at distances of 2, 4, 6, 12, and 18 inches below the surface, at the surface, and in the air 1/2-inch above the surface. For comparison purposes similar thermocouple installations were made in three other houses in the project. Two of these houses, E605 at 129 Illinois Drive and A1-14 (left unit) at 122 Mississippi Loop, had been occupied for several months and the occupants reported that the normal thermo-

stat setting had been 75°F. The third house, C663, (left unit) at 101 Iowa Circle, was vacant and had not been occupied, heated or cooled. It should be noted that the surface of the concrete slab was exposed in the utility closet and that the indoor unit of the heat pump and the hot water heater were located in this closet. Return air to the indoor unit entered the utility closet through louvers near the bottom of the double-section closet doors and through the clearance between the bottom of the doors and the floor surface. As opposed to the bare concrete in the utility closet, the slab was covered with adhesive bonded 3/8" wood parquet tile in the halls, bedrooms and living-dining room, with asphalt tile in the kitchen and with ceramic tile in the bathroom.

During the last two days of the test, heat flow meters were attached in turn to the floor surface in each house at three stations, 1 foot, 2 feet, and 12.5 feet from an outside wall to measure the heat flow into the floor.

Separate watthour meters were used to measure the electric resistance heating and blower power as one part of the heat supply and the miscellaneous internal electrical loads as another part of the heat supply to the houses.

The same calibrated thermocouple wire, watthour meters, pressure gages, and other instrumentation were used for the heating tests as had previously been used for the cooling tests.

5.0 Test Results and Conclusions

5.1 Heating Capacity

The results of the heating tests in houses C612 and E642 at the Little Rock Air Force Base are summarized in Tables 1 and 2, respectively. The gross heat transmission factor is based on the observed data from 3 to 6 AM each morning since the effects of solar heating during the preceding day would be largely dissipated by this time and because outdoor temperatures are frequently more stable during the early morning hours. Tables 1 and 2 also show the temperatures observed on the floor surface and 4 inches below the floor surface near the center of each house as a basis for determining whether the floor heat loss was excessive during these short term tests. The heat loss rates determined during the early morning hours of the last day of the tests by the heat flow meters are also reported.

Table 1

Summary of Heating Test
House C612

Date		Oct. 29	Oct. 30	Oct. 31	Nov. 1	Nov. 2	Nov. 3	Nov. 4	Nov. 5	Nov. 6	Nov. 7	Nov. 8	Nov. 9
Electric Power Input,	KW												
Strip Heaters & Blower		6.02	5.37	4.97	4.94	5.93	5.10	4.07	3.51	4.90	5.93	3.98	5.18
Miscellaneous Load		.73	.76	.83	.83	.97	.82	.94	.95	.95	.95	.78	.74
Total		6.75	6.13	5.80	5.77	6.90	5.92	5.01	4.46	5.85	6.88	4.76	5.92
Equivalent Heat Input,	Btu/hr	23,060	20,920	19,800	19,710	23,560	20,210	17,100	15,240	19,970	23,520	16,260	20,070
Avg. Indoor Air Temp.	°F	86.0	87.9	86.2	86.2	87.5	86.7	86.0	85.7	86.6	87.4	85.9	86.5
Avg. Outdoor Air Temp.	°F	45.2	48.5	50.2	47.5	44.0	43.9	51.3	57.1	44.3	34.6	53.9	43.4
Avg. Temp. Difference	°F	40.8	39.4	36.0	38.7	43.5	42.8	34.7	28.6	42.3	52.8	32.0	43.1
Gross Heat Transmission Factor,	Btu/hr(F)	565	531	547	509	542	472	493	533	472	445	508	465
Surface Temp. of Floor Slab	°F	79.2	79.3	80.1	81.0	81.1	81.3	81.9	81.5	81.9	81.9	82.0	81.8
Concrete Temp. 4 in. below Surface	°F	76.6	77.0	78.0	79.0	79.2	79.9	80.0	80.0	80.1	80.5	80.1	80.1
Temp. Difference across Floor	°F	2.6	2.3	2.1	2.0	1.9	1.4	1.9	1.5	1.8	1.4	1.9	1.7
Floor Heat Loss by Heat Flow Meters*	Btu/hr(ft)^2												
1 ft from cold wall													3.12
2 ft from cold wall													3.16
Average													3.14
12.5 ft from cold wall													2.94
Floor Heat Loss,	Btu/hr												
3 ft Border													1,120
Center Section													1,925
Total													3,045

*Observed between 8:30 and 9:00 a.m.

Table 2

Summary of Heating Test
House E642

Date		Oct. 30	Oct. 31	Nov. 1	Nov. 2	Nov. 3	Nov. 4	Nov. 5	Nov. 6	Nov. 7	Nov. 8	Nov. 9
Electric Power Input,	KW											
Strip Heaters & Blower		6.48	5.49	5.70	6.37	5.89	4.90	4.43	5.58	7.12	4.77	6.13
Miscellaneous Load		1.05	1.40	.80	1.08	.69	.80	.69	.72	.71	.73	.69
Total		7.53	6.89	6.50	7.45	6.58	5.70	5.12	6.30	7.83	5.50	6.82
Equivalent Heat Input,	Btu/hr	25,720	23,530	22,180	25,440	22,480	19,450	17,500	21,510	26,720	18,180	23,290
Avg. Indoor Air Temp.	°F	84.3	84.7	85.2	84.7	85.5	84.8	84.5	84.5	85.3	84.6	85.2
Avg. Outdoor Air Temp.	°F	48.4	49.7	49.6	44.9	44.6	53.0	57.4	45.6	36.4	54.4	43.6
Avg. Temp. Difference	°F	35.9	35.0	35.6	39.8	40.9	31.8	27.1	38.9	48.9	30.2	41.6
Gross Heat Transmission Factor,	Btu/hr(F)	716	672	622	639	549	611	645	552	546	621	559
Surface Temp. of Floor Slab	°F	81.5	83.2	82.8	81.8	83.2	83.6	84.1	84.2	84.0	83.8	84.3
Concrete Temp. 4 in. below Surface	°F	77.5	80.1	80.5	80.3	81.1	81.5	81.8	82.1	82.1	81.8	82.3
Temp. Difference across Floor	°F	4.0	3.1	2.3	1.5	2.1	2.1	2.3	2.1	1.9	2.0	2.0
Floor Heat Loss by Heat Flow Meters,*	Btu/hr(ft)^2											
1 ft from cold wall												2.43
2 ft from cold wall												2.16
Average												2.30
12.5 ft from cold wall												1.24
Floor Heat Loss,	Btu/hr											
3 ft Border												910
Center Section												936
Total												1,846

*Observed between 6:30 and 6:50 a.m.

The Committee agreed that the observed gross heat transmission factors for the last seven days of the tests, that is from Nov. 3 to Nov. 9 inclusive, would be averaged as a basis for further computations on heating and cooling loads. The results observed prior to Nov. 3 were deleted from further consideration because there was definite evidence that the losses were decreasing during the warmup period. Based on the last seven days of the heating tests for the average gross heat transmission factor in Btu/hr (F) was 484 for house C612 and 583 for house E642 as shown in Table 3.

Research conducted at the National Bureau of Standards* on concrete floor slab constructions similar to that in the test houses indicates that the temperature differences at near steady-state conditions across a 4-inch concrete floor do not exceed 1°F. Tables 1 and 2 show average temperature differences across the 4-in. concrete slab of 1.7°F and 2.1°F for house C612 and house E642, respectively, for the last seven days of the heating tests, indicating a heat flow through the floor approximately twice that which would be expected at near steady state for such construction in the coldest period of the heating season. To correct for this apparent excessive heat flow through the floor during the tests, the Committee agreed to reduce by 50 percent the heat flow through the center section of the floor as indicated by the heat flow meter readings. No correction was made for the 3-ft border section of the floor.

As shown in Table 3, the reduction in heat loss thus allowed was 960 and 470 Btu per hour for houses C612 and E642, respectively. This resulted in a net heat transmission factor of 460 Btu per hour (F) for house C612 and 570 Btu per hour (F) for house E642. At the design temperature difference of 65 degrees F specified for the Little Rock Air Force Base the required total heating capacity for house C612 would be 29,900 Btu per hour and for house E642, 37,100 Btu per hour.

The quantity of air being moved through the indoor unit of the heat pump during the heating tests was computed from measurements of the total heating effect of both the blower motor and strip heaters and the resulting temperature rise of the air across the indoor unit. With air entering the blower at about 85°F and a blower speed of 715 rpm, the static pressure drop across the blower was between 0.48 and 0.52 in. W.G., the external duct resistance was between 0.23 and 0.26 in. W.G., and the air flow

* Reported in Building Materials and Structures Report BMS 138

rate was 1135 cfm or 4900 lb per hour for the unit in house C612. For the unit in house E642, with air entering the blower at about 85°F and a blower speed of 690 rpm, the static pressure across the blower was between 0.51 and 0.55 in. W.G., the external duct resistance was between 0.21 and 0.24 and the air flow rate was 1335 cfm or 5760 lbs per hour.

Table 3

Corrections to Observed Heat Transmission Factor for Floor Heat Losses

House Number		C612	E642
Avg. Gross Heat Transmission Factor from Tables 1 and 2	Btu/hr(F)	484	583
Avg. Indoor-Outdoor Temp. Difference from Tables 1 and 2	°F	39.5	37.1
Reduction in Center Floor Loss Allowed,	Btu/hr	960	470
Reduction in Heat Transmission Factor Corresponding to Reduction in Floor Loss,	Btu/hr(F)	24	13
Net Heat Transmission Factor for Heating Condition,	Btu/hr(F)	460	570
Total Heat Required at Design Outdoor Temp. of 5°F,	Btu/hr	29,900	37,100

The results of the laboratory tests of the heating capacity of the heat pumps removed from houses C612 and E642 are shown in Table 4. For the unit from house C612, the total heating capacity at design temperatures of 70°F indoors and 5°F outdoors ranged from about 9800 to 10,900 Btu per hour for a range of mass air flow rates from about 3640 to 5120 lbs per hour. Interpolating for the actual air flow rate in the house of 4900 lbs per hour, indicates a total heating capacity of 10,750 Btu per hour for the unit from C612 for the conditions existing during the field test.

For the unit from house E642 the heating capacity measured in the laboratory at design temperatures of 70°F and 5°F ranged from about 9100 to 9660 Btu per hour for a range of mass air flow rates from about 3240 to 4340 lbs per hour. Extrapolating for the actual flow rate in the house of 5760 lbs per hour indicates a total heating

Table 4

Laboratory Tests of Heating Capacities
of Heat Pumps from Houses C612 and E642

		Unit from C612			Unit from E642	
Date		Sept. 24	Sept. 25	Sept. 25	Sept. 29	Sept. 29
Duration of Test,	hrs	1.5	1.0	2.0	1.5	1.9
Air Temp. entering Indoor Coil, D.B.	°F	70.1	69.8	69.2	70.2	69.7
Air Temp. entering Indoor Coil, W.B.	°F	62.0	62.8	62.9	61.7	63.2
Air Temp. leaving Indoor Coil, D.B.	°F	81.2	78.6	80.5	79.4	81.3
Air Temp. leaving Indoor Coil, W.B.	°F	65.7	63.6	66.0	65.0	67.0
Air Temp. entering Outdoor Coil, D.B.	°F	5.1	4.9	4.7	5.0	4.9
Indoor Coil Temperature,	°F	(85.0 to 80.1)	(81.6 to 75.5)	(89.8 to 78.7)	(82.5 to 82.7)	(84.1 to 85.4)
Air Temp. Change across Coil,	°F	11.1	8.8	11.3	9.2	11.6
Static Pressure across Indoor Blower,	In.W.G.	.418	.36	.487	.425	.464
Indoor Blower Speed,	rpm	720	--	723	693	694
Mass Air Flow Rate, Indoor Unit	lb/hr	3,638	5,116	3,669	4,339	3,237
Heating Capacity	Btu/hr*	9,819	10,904	9,954	9,602	9,096
Compressor Discharge Pressure,	psig	158	155	158	154	156
Compressor Suction Pressure,	psig	24.3	25.0	25.0	23.8	23.5
Refrigerant Temp. entering Compressor,	°F	< 0	< 0	< 0	< 0	< 0
Refrigerant Superheat at Compressor,	°F	0	0	0	0	0
Refrigerant Temp. leaving Compressor,	°F	--	83.8	84.5	83.9	85.3

* Includes heat from blower and blower motor

capacity of 10,380 Btu/hr for the unit from house E642 for the conditions existing during the field tests.

The available heating capacity on a continuous basis for an indoor temperature of 70°F and an outdoor temperature of 5°F is summarized in Table 5 for the two test houses. By comparing the values of available heat from Table 5 with the values of total heat required in Table 3 it was concluded that there was ample heating capacity in house E642 to maintain a 65 degree temperature difference between indoors and outdoors whereas the heating capacity in house C612 was deficient by about 3200 Btu/hr for the same conditions of operation. The heating capacity of the electric heaters in the bathrooms was considered to be a part of the available heating capacity because this was the intent of the project designers, as confirmed in writing by the Project Supervisor.

The comparisons made above do not take into account intermittent sources of heat inside the houses such as electric lights, cooking range, hot water heater, other electric appliances and the occupants themselves. These intermittent heat sources are conventionally not taken into account in selecting the heating equipment for buildings. Neither do these comparisons reflect the effect of heat storage in the furnishings of the house and in the floors, walls, partitions, and ceilings under living conditions where the normal thermostat setting was 75°F and the house was permitted to cool to 70°F under design outdoor conditions.

The Committee agreed that a design day at 5°F outdoor temperature is a day in which the minimum outdoor temperature reached, but does not go below, 5°F sometime during the 24-hour period.

The temperature change of the furnishings that would probably occur while the air temperature of the house was changing at the rate of 1°F per hour for 5 hours was computed as a transient heat flow problem. This computation indicated that a wooden member 3 inches thick would change air temperature about 3.9°F under these conditions. Because some of the furnishings would change temperature more slowly than a 3-inch wood board an average temperature change of 3.5°F was used in computing the heat storage effect of the furnishings. The average specific heat of the many kinds of material included in house furnishings was determined to be about 0.35 Btu/lb(F). Information furnished by the U. S. Air Force indicated that a family occupying a Type C house could have 9000 lb

Table 5

Available Capacity for Heating, Btu/hr
Indoor Temperature of 70°F
Outdoor Temperature of 5°F

	House C612	House E642
Measured Heat Pump Capacity* (Marble Falls Lab. Test)	10,750	10,380
Strip Heater Capacity	12,300	24,600
Bathroom Electric Heaters	3,410	6,830
Electric Refrigerator	250	250
Total Available Capacity, Btu/hr	26,710	42,060

* Corrected for air flow as measured at house; includes blower motor.

of furnishings and a family occupying a Type E house could have 10,500 lb of furnishings shipped to the base at government expense. The heat that would be transferred to the room air from the furnishings when the air temperature changed 1°F per hour for 5 hours was computed on this basis for the two test houses. The results of these computations, summarized in Table 6, show that the furnishings would probably contribute heat to the air of the house at a rate of about 2200 Btu/hr in house C612 and about 2570 Btu/hr in house E642 for 5 hours.

The warmup test that was made in each house during which the indoor and outdoor temperatures were kept equal, while the outdoor temperature rose about 13 degrees during one forenoon, showed that slightly over 3000 Btu/°F rise in air temperature was absorbed by the walls, floor and ceiling of each of the houses during this period. During a cooling period in the house the heat stored in the interior partitions and floor slab would be available to warm the air in the house and the heat stored in the entire exterior walls and ceiling construction would retard the cooling of the room air in response to a dip in outdoor temperature that lasted only a few hours. The Committee concluded from these warmup tests that the stored heat in the building constructions of the Type C and E houses would exceed 3000 Btu per degree change in interior air temperature.

It was concluded by the Committee that, whereas the heat pump with its strip heaters and the bathroom electric heater would not maintain a 65°F temperature difference continuously between indoors and outdoors in the Type C house, if the indoor temperature was maintained thermostatically at 75°F whenever possible, the heat stored in the furnishings and in the structure itself would prevent the indoor temperature from falling below 70°F during a day when the outdoor temperature decreased to a 5°F minimum in a typical daily temperature cycle. It was further concluded that the heat storage effect of the furnishings and the house structure itself did not need to be relied upon in the Type E house for maintaining an indoor temperature of 70°F at a design outdoor temperature of 5°F.

The preceding analysis does not take into account the fact that the jacket loss of the electric water heater and the sensible heat output of one or two observers, and perhaps other small heat sources or heat losses, affected the warming of the house during

Table 6

Heat Capacity of House Furnishings

		House C612	House E642
Estimated gross weight of furnishings,	lb	9,000	10,500
Avg. specific heat of furnishings,	Btu/lb(F)	0.35	0.35
Avg. temp. drop of furnishings in 5 hours,	°F	3.5	3.5
Heat transfer from furnishings in 5 hours,	BTU	11,020	12,860
Average Heat Transfer Rate from Furnishings,	Btu/hr	2,204	2,572

the tests made to determine the heat transmission factor of the houses. The magnitude of these heat sources and heat losses was not measured, but they are estimated to comprise a source of heat of the order of 500 Btu per hour.

5.2 Cooling Capacity

The results of the cooling tests in houses C612 and E642 at the Little Rock Air Force Base are summarized in Table 7. Since the inlet air temperatures to the indoor and outdoor units were maintained at 80°F and 105°F, respectively, during the cooling tests, the heat pump compressors were operating at the same compression ratio as if the design indoor and outdoor conditions actually existed. Since bright sunshine existed during the cooling tests it was concluded that the solar effects on the houses were representative of those that would exist under design summer conditions even though the outdoor dry bulb temperatures were considerably below the design value of 105°F.

The data summarized in Table 7 represent the average performance and test conditions that were observed from about noon to 6 PM on each day of the test. Operation of the electric range, dishwasher, washer and dryer, kitchen ventilating fan, electric lights, and Venetian blinds and the use of exterior doors was carried out on an actual or simulated basis as described in the section on Test Procedure. The heat contribution of the miscellaneous loads that could be measured electrically averaged from 3680 to 4530 Btu/hr in house C612 and from 3140 to 4160 Btu/hr in house E642 during the peak load period under consideration.

In addition to these miscellaneous internal loads simulating the normal daily activities of a family, resistance heating was supplied under thermostatic control in the discharge plenum of the heat pump units to maintain a return air temperature of 80°F while the units operated continuously. The amount of supplementary resistance heating required to maintain a design indoor temperature of 80°F ranged from 6900 to 8780 Btu/hr for house C612 and from 7690 to 10,270 Btu/hr for house E642 as shown in Table 7. In the case of house C612 the results of the cooling test on September 8 are considered more representative of the performance of the heat pump when properly adjusted than those observed on September 7 because the blower speed of the indoor unit was lower on September 7 than was recommended by the manufacturer.

Table 7

Summary of Cooling Tests
on Houses C612 & E642

		House C612		House E642		
Date of Test		Sept. 7	Sept. 8	Sept. 7	Sept. 8	Sept. 9
Hours of Test		1200-1800	1230-1830	1130-1730	1130-1730	1130-1730
Inlet Air Temp., Indoor Coil, D.B.	°F	78.4	79.0	79.8	79.0	80.1
Inlet Air Temp., Indoor Coil, W.B.	°F	66.0	66.8	67.9	66.9	66.7
Inlet Air Temp., Outdoor Coil, D.B.	°F	105.0	103.5	104.3	105.2	104.4
Temp. Difference across Indoor Coil, D.B.	°F	16.5	16.0	15.8	15.5	16.0
Indoor Coil Temp.	°F	54.5	54.3	52.7	53.9	53.1
Compressor Suction Pressure,	psig	85	85	83	85	84
Compressor Discharge Pressure,	psig	281	280	283	301	293
Refrigerant Temp., Comp. Suction,	°F	50.6	50.4	75.9	53.8	--
Superheat at Comp. Suction,	°F	0.5	0.2	26.8	3.5	--
Blower Speed, Indoor Unit,	rpm	--	712	690	690	690
Static Press. Difference, Indoor Blower	in.W.G.	0.44	0.55	0.52	0.52	0.52
Line Voltage	volts	243	244	245	245	246
Blower Speed, Outdoor Unit	rpm	799	799	806	806	806
Maximum Outdoor Temp., D.B.	°F	88.0	90.5	88.0	90.5	91.0
Avg. Internal Miscellaneous Elec. Load,	Btu/hr	3,680	4,530	4,160	3,470	3,140
Avg. Supplementary Strip Heating,	Btu/hr	6,900	8,780	9,530	7,690	10,270
Temp. Difference (Design Temp. - Max. Observed)	°F	17	14.5	17	14.5	14.0
Additional Heat Transmission at Design Temp.	Btu/hr	6,920	5,900	9,060	7,730	7,460

The additional heat transmission that would have occurred in the test houses if the outdoor maximum temperature had been 105°F during the cooling tests was computed from the heat transmission factors measured during the heating tests after making certain corrections for floor heat transfer. The Committee estimated the probable subfloor earth temperatures during midsummer in the latitude of Little Rock based on what data were available in the literature and the probable daily and weekly average air temperature during the hottest parts of the summer season as a basis for determining the probable direction and magnitude of the floor heat transfer during the cooling tests. It was concluded that the total heat transfer through the floor was probably so low that it could be neglected during cooling operation at design conditions. The corrections required to adjust the gross heat transmission factor observed during the heating tests for the cooling conditions with no floor heat transfer are summarized in Table 8. This computation results in net heat transmission factors of 407 and 533 Btu/hr(F) for houses C612 and E642, respectively, for the cooling condition.

The additional heat transmission that would be anticipated in the test houses if the outdoor temperature had been the design value of 105°F is summarized in Table 7. The Committee agreed that this increment of heat transmission should be based on the difference between the design temperature of 105°F and the maximum temperature observed at the test site on any given day using the temperature data obtained on a recorder located approximately midway between the two houses. The computed additional heat transmission at design summer conditions ranged from 5900 to 6920 Btu/hr for house C612 and from 7460 to 9060 Btu/hr for house E642 for the individual days of the tests. A comparison of the average supplementary strip heating and the average additional heat transmission at design temperature for all days of test in each house shows that the available cooling capacity exceeded the total cooling load by about 1400 Btu/hr in house C612 and by about 1100 Btu/hr in house E642 for the conditions established for these tests.

While the Committee recognized that the adequacy of the cooling capacity of the heat pumps was closely related to the magnitude of the miscellaneous internal loads in these houses, the schedule of internal loads set up initially was considered typical although somewhat conservative.

Table 8

Corrections to Observed Gross Heat Transmission Factor for the Cooling Condition

House Number		C612	E642
Avg. Gross Heat Transmission Factor during Heating from Tables 1 and 2,	Btu/hr(F)	484	583
Observed Total Floor Heat Loss (Tables 1 and 2),	Btu/hr	3,045	1,846
Avg. Indoor-Outdoor Temp. Difference during Heating Tests (Table 3),	°F	39.5	37.1
Reduction in Heat Transmission Factor to Eliminate Floor Loss,	Btu/hr(F)	77	50
Net Heat Transmission Factor for Cooling Condition,	Btu/hr(F)	407	533

The computed additional cooling load that would have occurred at design outdoor conditions was based on sensible heat considerations only and makes no allowance for the possibility of higher moisture content in the outdoor air entering by infiltration or door usage. However, the moisture content of the outdoor air during a part of the test period was about 90% as great as that corresponding to the design outdoor conditions of 95°F D.B. and 78°F W.B. cited in the ASHAE Guide for Little Rock. It should also be noted that the net heat transmission factor includes the effect of air infiltration at an indoor-outdoor temperature difference of 35 to 40 degrees which exceeds the temperature difference designated for design summer conditions for this housing project.

5.3 Laboratory Tests of Cooling Capacity

Laboratory tests to determine the cooling capacity of the Mathes heat pumps removed from the two test houses at Little Rock Air Force Base were made at the laboratory of the Mathes Company in Marble Falls, Texas. The temperature conditions at the inlets to the indoor and outdoor units for these laboratory tests duplicated, as nearly as possible, those observed during the field tests in the houses and observations were made over a range of air flow rates through the indoor unit.

The laboratory tests under cooling conditions were made to establish that the units installed in the test houses were function normally and also to provide a basis for determining the approximate sensible cooling load of the two test houses. The results of the cooling capacity tests are summarized in Table 9 for house C612 and in Table 10 for house E642.

Table 9 shows that the total cooling capacity of the unit from house C612 ranged from 20,210 Btu/hr to 24,070 Btu/hr for a range of mass air flow rates from 2865 to 5105 lb/hr as measured in the laboratory. For the cooling tests in house C612 the mass air flow rate through the indoor coil was about 4900 lb/hr. Interpolation of the laboratory data indicates that the heat pump unit had a total cooling capacity of about 24,000 Btu/hr at the conditions of the field test in house C612.

Table 10 shows that the total cooling capacity of the unit from house E642 ranged from 18,060 Btu/hr to 21,720 Btu/hr for a range of mass air flow rates from 2820 to 4340 lb/hr as measured in the laboratory. For the cooling tests in house E642 the mass

Table 9

Summary of Laboratory Tests of Cooling Capacity
of Mathes Model 27 Heat Pump from House C612

Date of Test		Sept.23	Sept.23	Sept.23 to Sept.24	Sept.26	Sept.26	Sept.23
Hours of Test		1515 to 1545	2200 to 2230	2245 to 0045	0115 to 0315	0445 to 0605	1600 to 1645
Inlet Air Temp., Indoor Coil, D.B.	°F	75.9	76.6	76.9	77.4	80.1	77.0
Inlet Air Temp., Indoor Coil, W.B.	°F	66.2	67.9	67.7	66.8	68.1	66.9
Outlet Air Temp., Indoor Coil, D.B.	°F	59.4	61.1	62.1	60.9	64.0	64.8
Outlet Air Temp., Indoor Coil, W.B.	°F	56.0	59.4	59.9	58.3	61.0	60.5
Temp. Diff. across Indoor Coil, D.B.	°F	16.5	15.5	14.8	16.5	16.1	12.2
Indoor Coil Temp.	°F	49.0	52.5	53.6	51.7	53.8	53.0
Inlet Air Temp., Outdoor Coil, D.B.	°F	104.2	103.4	103.4	103.9	104.6	105.6
Compressor Suction Pressure	psig	77.5	83.0	84.3	84.6	87.5	81.5
Compressor Discharge Pressure	psig	281	284	281	286	293	285
Refrigerant Temp. Compressor Inlet	°F	84.8	60.2	60.6	74.2	84.0	91.9
Superheat at Compressor Suction	°F	39.2	11.2	10.9	24.3	32.3	43.9
Blower Speed, Indoor Unit	rpm	--	723	720	724	719	716
Static Pressure Diff. Indoor Blower	In.W.G.	0.57	0.55	0.53	0.52	0.48	0.35
Mass Air Flow Rate, Indoor Unit	lb/hr	2,865	3,333	3,623	3,573	4,540	5,105
Line Voltage	volts	245	243	245	247	246	245
Sensible Cooling Capacity	Btu/hr	11,450	12,570	13,000	14,390	17,590	16,350
Total Cooling Capacity	Btu/hr	20,210	20,390	20,470	21,620	23,410	24,070

Table 10

Summary of Laboratory tests of Cooling Capacity
of Mathes Model 27 Heat Pump from House E642

Date of Test		Sept.29	Sept.27	Sept.29	Sept.27	Sept.29	Sept.29
Hours of Test		0015 to 0125	1515 to 1915	0200 to 0305	1930 to 2145	0345 to 0505	0600 to 0725
Inlet Air Temp., Indoor Coil, D.B.	°F	78.8	79.4	79.2	79.5	79.1	79.0
Inlet Air Temp., Indoor Coil, W.B.	°F	65.4	66.6	66.4	66.7	66.5	66.3
Outlet Air Temp., Indoor Coil, D.B.	°F	58.7	59.6	60.4	60.3	61.8	62.2
Outlet Air Temp., Indoor Coil, W.B.	°F	55.9	57.2	58.2	58.0	59.1	59.3
Temp. Diff. across Indoor Coil, D.B.	°F	20.1	19.8	18.8	19.2	17.3	16.8
Indoor Coil Temp.	°F	49.3	50.0	51.3	51.3	52.2	52.6
Inlet Air Temp., Outdoor Coil, D.B.	°F	105.4	105.1	105.5	104.5	104.8	104.9
Compressor Suction Pressure,	psig	79.4	82.2	82.4	83.1	84.3	85.0
Compressor Discharge Pressure,	psig	298	299	301	304	301	304
Refrigerant Temp. Compressor Inlet,	°F	47.7	49.1	50.1	50.2	58.1	56.4
Superheat at Compressor Suction,	°F	1.0	0.7	1.6	1.2	8.4	6.2
Blower Speed, Indoor Unit,	rpm	698	696	694	694	692	692
Static Pressure Diff., Indoor Blower,	In.W.G.	0.52	0.50	0.48	0.48	0.46	0.46
Mass Air Flow Rate, Indoor Unit,	lb/hr	2,819	2,992	3,495	3,500	4,198	4,344
Line Voltage	volts	246	245	245	245	246	244
Sensible Cooling Capacity	Btu/hr	13,730	14,390	15,960	16,320	17,620	17,720
Total Cooling Capacity	Btu/hr	18,060	19,330	19,960	21,230	21,720	21,130

air flow rate through the indoor coil was about 5760 lb/hr. Extrapolation of the laboratory data to this higher air flow rate indicates that the heat pump unit had a total cooling capacity of about 22,500 Btu/hr at the conditions of the field test in house E642. However, the coil of the indoor unit was probably flooded at the highest mass air flow rate used for the laboratory tests, namely 4344 lb/hr. Thus this extrapolated value of total cooling capacity is probably a little lower than actually existed during the field operating conditions.

Additional laboratory cooling tests were made of each of the two heat pump units at inlet air temperatures of 70°F on the indoor unit and 95°F on the outdoor unit with almost identical air flow rates through the indoor units. The measured total cooling capacity of the two units differed by only 1.6 percent in these tests and averaged 18,920 Btu/hr. These results show that the two units were operating normally and had equal cooling capacities within the tolerances of testing procedures and production techniques.

The operating conditions observed in the laboratory cooling tests are probably not sufficiently representative of those in the field cooling tests to permit an estimate to be made of the total cooling load of the structures under design conditions because there was a difference in the superheat condition at the outlet of the indoor coil for each of the two units between field and laboratory tests. However, the total sensible cooling obtained from each unit can be computed from the mass air flow rate and the dry bulb temperature reduction observed during the field cooling tests (See Table 7). As shown in Table 11 the computed total sensible cooling effect for dry bulb temperatures of 80°F and 105°F at the inlet to the indoor and outdoor units, respectively, was 18,820 Btu/hr for house C612 and 21,800 Btu/hr for house E642. When the sensible heating effect of the miscellaneous internal electrical loads, the occupants, and the excess cooling capacity offset by strip heaters at design conditions, is subtracted from the total sensible cooling effect, the net sensible cooling load at design conditions is shown in Table 11 to be 12,790 Btu/hr for house C612 and 16,610 Btu/hr for house E642. No precise method is available for estimating the latent load of the infiltration air during the field tests. It probably did not exceed 2000 Btu/hr for house E642 during the cooling tests, based on the difference between the total capacity of the heat pump and the measured total sensible cooling effect of 21,800 Btu/hr. This component would vary with the humidity conditions outdoors and the amount of air leakage in individual houses.

Table 11

Estimate of Sensible Cooling Load of Houses C612 and E642

House Number		C612	E642
Total Sensible Cooling Effect Observed during Field Cooling Tests,	Btu/hr	18,820	21,800
Miscellaneous Internal Electrical Loads,	Btu/hr	4,100	3,590
Estimated Sensible Heat Release of Occupants,	Btu/hr	500	500
Excess Cooling Capacity of Heat Pump at Design Conditions, Offset by Strip Heat*	Btu/hr	1,430	1,100
Net Sensible Cooling Load of House at Design Conditions,	Btu/hr	12,790	16,610

* Average Supplementary Strip Heating during Cooling Tests	Btu/hr	7,840	9,160
Average Additional Heat Transmission at Design Temperatures	Btu/hr	6,410	8,060
Difference between Strip Heating and Additional Heat Transmission	Btu/hr	1,430	1,100

These values for the net sensible cooling load of the test houses should be regarded as approximate because of the extrapolations involved and the limitations in the accuracy of field test procedures.

6.0 Discussion and Conclusions

The results of the field and laboratory tests described in this report on two installations of the Mathes heat pump units and consideration of the dimensions and other available data on the several types of houses led the Committee to draw the following conclusions:

(a) A single Model 27 Mathes heat pump will maintain 80°F or lower in any of the houses, Types A through E, during design summer conditions with a maximum dry bulb temperature of 105°F provided the electrical devices indoors such as cooking range, washer, dryer, electric lights, etc. are used conservatively during the middle of the day. The tests indicated that the heat pump would maintain 80°F indoors in the largest house equipped with a single heat pump unit, a Type E house, with a southwest exposure during such a design day if the internal electrical loading during the middle of the day did not average more than 4700 Btu/hr. The simulated internal loading during the test of this type house averaged 3600 Btu/hr. But it should be emphasized that any extended use of the cooking range, washer, dryer, or large numbers of electric lights on a design day could raise the indoor temperature above 80°F.

(b) A single Model 27 Mathes heat pump, together with strip heaters and electric resistance bathroom heaters, as scheduled in Table 12, will heat houses of types A, A_1, B, D, and E to an indoor temperature of 70°F or higher on a typical design winter day with a minimum outdoor temperature of 5°F without relying on stored heat in the house furnishings and house structure; and will prevent the air temperature from falling below 70°F in type C houses during such a design day if use is made of the heat capacity of the furnishings and structure in an appropriate way. The heating re-requirements on the refrigerating system of the heat pump in the type B_1 houses appear to be similar to those in the type C houses. Specifically, if a thermostat setting of 75°F is employed day and night in type B_1 and C houses, the heat stored in the furnishings, partitions, floors, walls, and ceilings in addition to the maximum heating capacity of the installed heat pump system and bathroom heaters will sustain the house temperature at 70°F or higher for a period of 5 hours, more or less, as the outdoor temperature approaches a minimum of 5°F on a design winter day.

Table 12

Schedule of Strip Heaters and Bathroom Heaters in House Types A through E

House Type	Heating Capacity of Strip Heaters Btu/hr at 240 Volts	Heating Capacity of Bathroom Heater Btu/hr at 240 Volts	Total Heating Capacity of Strip and Bathroom Heaters Btu/hr
A	12,290	3,410	15,700
A-1	18,440	3,410	21,850
B	18,440	3,410	21,850
B-1	18,440	3,410	21,850
C	12,290	3,410	15,700
D	18,440	3,410	21,850
E	24,590	6,830	31,420

The field teating tests of two houses indicated that the Type E house tested had about 5000 Btu/hr more installed heating capacity than required and the Type C house tested had about 3200 Btu/hr less installed heating capacity than required to maintain a constant indoor temperature of 70°F during a design winter day with 5°F minimum temperature.

Tests and computations of the heat storage capacity of the structures and the estimated amount of furnishings indicated that the furnishings would release heat to the room air at a rate of 2200 to 2570 Btu/hr and additionally the structure would release heat to the outdoors at a rate in excess of 3000 Btu/hr for a 5 hour period in which the air temperature was falling at the rate of 1 degree per hour. The Committee concluded that the heat capacity of the house and furnishings should be accepted as a means of preventing undesirably low indoor temperatures from occurring during winter extremes, since the occupants will probably maintain room temperatures of about 75°F during normal operation.

The heat available from cooking, lighting, washing, and water heating was not considered in evaluating the adequacy of the heating system in accordance with usual design practices for heating of houses.

(c) The Committee recommends that a fixed thermostat setting of about 75°F be maintained day and night during the winter in all occupied houses to avoid the heavy electrical power demand that would occur every morning if all thermostats were advanced about the same time after night setback. Furthermore, the heat capacity of the house and furnishings would not be available to help maintain a minimum indoor temperature of 70°F during extreme weather in the types of houses with insufficient instantaneous heating capacity, if the house were permitted to cool to 70°F in advance of the outdoor minimum.

By superimposing a graph of the probable heating capacity of the heat pump and resistance heaters in the range of outdoor temperature from 5°F to 15°F based on the laboratory tests on a graph of the heat loss of the Types C and E houses based on the field tests, an intersecting point can be obtained which shows the lowest design outdoor temperature at which the house could be maintained at 70°F. Such a graph indicates that this minimum design outdoor temperature is about 9.5°F for the Type C house and about -1.5°F for the Type E house. When the outdoor temperature begins to fall below these values on a cold winter day, the indoor temperature

would also begin to fall below 70°F, but at a lower rate than the outdoor temperature because of the heat capacity of the house construction and the furnishings. This illustrates the need, in the case of the Type C house, for maintaining a house temperature above 70°F during more moderate weather so the heat capacity of the structure and furnishings can be effective in sustaining the indoor temperature during temperature dips of short duration below 9.5°F.