AN INVESTIGATION OF THE OPERATING CHARACTERISTICS OF CERTAIN HEAT PUMPS USING WATER AS A HEAT SOURCE

by

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INTRODUCTION

Interest in the heat pump continues to increase throughout the country. The recent installation of several heat pumps in Kansas homes has stimulated the already active interest in this state.

Architects, builders, and home owners are interested in the heat pump because it offers the possibility of using the same equipment to supply automatic space heating and cooling without the use of ordinary fuels. The interest of rural home owners is stimulated by the relatively high cost of fuels normally available for use in automatic heating systems in rural areas.

The refrigeration industry is affected by the development of the heat pump because it has the responsibility of designing, manufacturing, installing, and servicing the equipment.

Popular acceptance of the heat pump would greatly increase the use of electricity. The electric generating and distributing industry has the problem of evaluating the probable acceptance of the heat pump and its effect upon the use of electricity. It must prepare its own facilities to take care of whatever requirements may develop. The industry is active in heat pump research.

The heat pump could prove to be a factor which will provide a sound economic foundation for central station, rural, electric service in sparcely settled areas. There are several significant reasons that the power suppliers in Kansas are concerned with heat pump development for rural areas. Among these reasons, the more important are:

1. The climate in Kansas is such as to create an interest in summer air cooling.

2. The summer and winter conditioning requirements in Kansas are such

that essentially the same size heat pump unit can fulfill the requirements for both seasons.

3. The density of customers per mile of rural line is low which emphasizes the need for greater electrical consumption per customer.

Many private organizations and public agencies in the United States are engaged in heat pump research. Research now in progress deals with investigations of heat sources, of special types and arrangements of equipment, and of the various applications of the heat pump.

THE INVESTIGATION

The object of this investigation was to determine the complete operating characteristics of several heat pumps which were being used as the sole space heating equipment in Kansas homes.

The phases of the investigation were defined under two main classifications:

1. To gather information on the cost of operation of these heat pumps and to report it in such a form as to be useful to the layman in determining the merits of this, compared to other systems of home heating.

2. To make a technical analysis of the operation of the system for the purpose of predicting the performance of this or similar units under a wide range of operating conditions.

REVIEW OF LITERATURE

One of the first heat pumps for domestic heating was employed by Haldane (3, 4) in his home in Scotland in the 1930's. The system could be used with either outside air or city water or both as a source of heat. Because Haldane used an ammonia refrigeration system, it was necessary to employ a secondary heat exchange to prevent the possibility of toxic ammonia entering the heated space through leaks in the refrigeration system. This intermediate heat exchange reduced the efficiency of the system. Values of 2 to 3 were reported as the coefficients of performance of the system.

Mangel (6) reported coefficients of performance varying from 4.95 to 5.3 for a 2 horsepower heat pump located in his Riverside, California home. The unit used city water with an entering temperature of 65 degrees Fahrenheit as a source of heat.

Walls (9) reported on a heat pump used for heating a house in Boise, Idaho through radiant panels. In this system, the condenser consisted of 6,000 feet of 3/16 inch copper tubing imbedded in the interior of the walls and the ceiling. The source of heat was well water at a temperature of 56 degrees Fahrenheit. A coefficient of performance of 3.33 was reported for the installation.

Crandall (1) presented data on a heat pump located in Indianapolis, Indiana. The system utilized the soil as a heat source. He reported a coefficient of performance of 3.6 on the system.

According to Penrod (7), the Electric Power Board of Chattanooga in tests reported by S. R. Finley found that a Marvair heat pump employing water at 58 degrees Fahrenheit as a heat source had a coefficient of performance of 4.10.

Douglas (2) reported on a heat pump located in Mojave, California, which used outside air as the heat source. The unit utilized a 3 horsepower compressor and heated a 7-room house. Laboratory tests of the unit showed that the coefficient of performance varied from 1.84 to 3.29 for air temperatures of 25° F. to 70° F., respectively.

Sporn, Ambrose, and Baumeister (8) have presented an explanation of the fundamental thermodynamic principles involved in any discussion of the heat pump. They have also discussed certain basic heat pump designs, design factors, equipment, and applications. A study of this or any standard text on elementary thermodynamics is desirable as an aid to understanding the fundamental theory of heat pump operation.

EXPLANATION OF PLATE I

View of a 5 horsepower heat pump installed in a home at Hutchinson, Kansas. At the extreme right is the water pump that supplies the heat source water. The lower part of the large cabinet in the foreground encloses the compressor, compressor motor, evaporator and expansion valves. The upper part of the cabinet is the warm air plenum chamber and encloses the condenser. The parts called the evaporator and the condenser here exchange functions when the unit is set for cooling. Wires may be seen running to the control panel located on the back side of the warm air plenum. The cabinet in the background encloses the fan, fan motor, and air filters. No return air plenum is seen because an under-floor return is used. A water meter, 2 mercury thermometers, and 1 of the thermometer wells may be seen. The solenoid valve in the water line is closed when the heat pump is not in operation.

PLATE I

DESCRIPTION OF THE HEAT PUMP INSTALLATIONS

In the fall of 1949 it was learned that heat pumps using well water as a heat source had been installed in several Kansas homes. The heat pumps were all standard production models, manufactured by the Muncie Gear Works of Muncie, Indiana, and sold under the trade name "Marvair". Specifications furnished by the manufacturer are given on page 61 and 62. The physical dimensions and general outside appearance of the units closely resemble many modern heating systems of the mechanically circulated warm air type. Plate I shows a unit installed in a home at Hutchinson, Kansas.

The path of the F-12 (Dichlorodifluoromethane) refrigerant is shown by the schematic diagram in Fig. 8, page 63. With the odd numbered valves open and the even numbered valves closed, the system is set for heating. Cooling is accomplished by the opposite setting of all numbered valves.

An investigation in October, 1949 disclosed that the following installations, listed in Table 1, were completed or were in a stage of advanced planning.

Table 1. Marvair heat pumps operated in Kansas during the 1949-50 heating season.

Unit designation	(Horse Power)	Approx. geographical location	Application
Unit No. 1	7.5	Langdon, Kansas	Air conditioning a rural residence
Unit No. 2	5	Sterling, Kansas	Air conditioning a rural residence
Unit No. 3	5	Clay Center, Kansas	Air conditioning a rural residence
Unit No. 4	5	Hutchinson, Kansas	Air conditioning a suburban home
Unit No. 5	5	Hutchinson, Kansas	Air conditioning a suburban home
Unit No. 6	7.5	Hutchinson, Kansas	Air conditioning a suburban home
Unit No. 7	10	Dodge City, Kansas	Air conditioning commercial office space
Unit No. 8	7.5	Ulysses, Kansas	Air conditioning commercial office space

In this investigation, the operation of Units No. 1, 2, and 3 were studied in detail. Less complete observations were made on the operation of Units No. 4, 5, and 6.

On the following pages, the different heat pump installations are described as well as the conditions which might have some influence upon the data taken. Since the observations made on Units No. 1, 2, and 3 were much more complete than on the others, a more detailed explanation is given of factors influencing their performance.

Unit No. 1 was installed in a farm home near Langdon, Kansas. It had been installed during the summer of 1949 and had been operated on the cooling cycle for a part of the summer. The residence was a large, 2-story, frame house with a half basement. Only the attic was insulated, but the house was well constructed and in an excellent state of repair. Aluminum storm sash had been installed on all windows. The entire house including second floor rooms was heated with the heat pump.

The heat pump compressor was driven by a 7 1/2 horsepower, 3 phase, electric motor. The electric power supply was simulated 3 phase alternating current tapped from a 2 phase rural distribution line.

Water for use as a heat source was lifted by a 1 horsepower centrifugal type pump from a depth of approximately 15 feet at a distance of approximately 100 yards from the heat pump. The water temperature was 58.8 degrees Fahrenheit. After flowing through the heat pump, the water was returned a distance of approximately 75 yards to an old, abandoned, dug well. The old well had not been a prolific source of water, but it had ample capacity for disposing of all water dumped into it by the heat pump. The water supply well and the disposal well were separated by a distance of approximately 80 yards.

Air was distributed by a centrifugal fan through a system of aluminum ducts to wall registers in the various rooms. A set of return ducts brought cool air back to a plenum chamber over the fan inlet. The fan was driven by a 1 horsepower single phase electric motor.

The thermostat was always turned down to 65° F. each night. The temperature in the house was maintained at approximately 75° F. during the day.

Unit No. 2 was installed in a farm home near Sterling, Kansas that was in the process of being remodeled. Remodeling of the house was not complete at the beginning of the heating season. It was occupied in early December, 1949, and the heat pump was put into operation at that time. The residence was a single story, ranch-type house with a small basement in which the heat pump was located.

The attic of the house was insulated, but heat losses through the walls and the floor were very large. The north, south, and west outside walls consisted of lap-siding on the exterior of 2" x 4" studs, and wood lath and plaster on the inside. The foundation was of rock construction with fairly open mortar joints.

A large double-glass window facing on the south admitted an appreciable amount of solar heat to the house on clear days.

Power was furnished by a 5 horsepower, 3 phase electric motor. The electric power supply was simulated 3 phase alternating current tapped from a 2 phase rural distribution line.

The thermostat was set to maintain a temperature of approximately 74° F. both day and night.

Water for the heat pump was lifted approximately 20 feet by a 1 horsepower single phase electric motor and a centrifugal type pump. The water temperature was 58.5° F. The well was located approximately 30 feet from the heat pump. The water leaving the heat pump, at first, was piped to an old well with 10-inch galvanized iron casing. Because the well could not accomodate the water that flowed from the heat pump, the water was piped to a point approximately 70 feet from the unit and allowed to flow out on the surface of the ground. The soil was quite sandy and the ground slope was favorable so no apparent nuisance was created by the disposal of water on the surface.

Air was forced over the condenser of the heat pump by a centrifugal fan driven by a 1/2 horsepower single phase motor. Distribution of the warm air supply was made through a plenum chamber leading from the basement to the attic and thence through round aluminum ducts to high-wall registers.

Unit No. 3 was located in a house north of Clay Center, Kansas. The house was under construction during the greater part of the heating season and was not occupied. However, the heat pump was put into operation in the early part of February, 1950, and was used to heat the house during the day while work progressed on the interior finish.

The structure was a single story, ranch-type house with a full basement. The attic and walls were insulated with 4 inches of rock wool, but storm sashes had not been installed. A large part of the 8-inch solid concrete basement wall was above grade causing a large heat loss from the basement.

The heat pump compressor was driven by a 5 horsepower single phase electric motor. Electric energy was supplied to the premises through a 40 ampere service connection from a single phase rural distribution line. This service was not adequate for the electric load and low voltage caused unsatisfactory operation of the motors during periods of peak load demand.

Water for the heat pump was supplied from a shallow well by a 1/2 horsepower, single phase combination centrifugal-jet. The temperature of the water was 56° F. The well was located in the basement, a distance of approximately 20 feet from the heat pump. After flowing through the evaporator of the heat pump, the water was piped to the outside where it was discharged into a trench with coarse rock fill. The trench followed the natural slope of the land to a highway ditch. Since the soil was relatively impervious, the water flowed from the ditch through a culvert under the highway and collected in a pasture across the road.

The air distribution system in the house did not employ return air ducts. Baseboard registers conducted the return air directly to the basement. The centrifugal fan powered by a 1/2 horsepower, single phase electric motor drew air directly from the basement. Warm air from the heat pump was discharged into a plenum chamber from which distribution was made through attic ducts to high-wall registers.

Units No. 4 and No. 5 were located in new houses built in a new housing development on the outskirts of Hutchinson, Kansas. Both houses were large, single story, ranch-type houses with insulated attics, tight wall construction, and storm sashes. Both units used 5 horsepower, 3 phase electric motors to drive the compressors. Water in each case was drawn from a shallow well located in the same room with the heat pump and was run back into the ground through another well located approximately 40 feet from the supply well. Difficulty was encountered when the disposal wells failed to handle all the water pumped to them. When the wells were sunk deeper into the coarse water-bearing gravel, they had sufficient capacity to handle the water satisfactorily.

The original water pumps used with Units No. 4 and No. 5 were capable of supplying less than 700 gallons per hour under the operating conditions that existed. In February, new pumps were installed with capacities of 1,200 gallons per hour.

Standard mechanically circulated air distribution systems were used with both units. One-half horsepower motors furnished power for the centrifugal type fans.

The house in which Unit No. 4 was located was not occupied until February, 1950. Up to the time the house was occupied, the heat pump was used to maintain an inside temperature of 60° F. to 65° F. The house in which Unit No. 5 was located was not occupied during the heating season but was used as a "Model House" in connection with the housing development. The heat pump was used to maintain an inside temperature of 60° F. to 65° F. throughout the heating season.

Both houses were located within reach of the city natural gas mains.

Unit No. 6 was located in a large ranch-type house within the city limits of Hutchinson, Kansas. The compressor was driven by a 7 1/2 horsepower, 3 phase motor. Like units No. 4 and No. 5, a larger capacity water pump was installed during the heating season. Air distribution was accomplished by conventional mechanical circulation.

Unit No. 7 was installed in an office building in Ulysses, Kansas. The heat pump was put into operation on a demonstration basis only. No study was made of this unit.

Unit No. 8 differed from all the others in that it was a larger unit and that it was driven by an internal combustion engine using natural gas for fuel. Operation of the unit was not considered satisfactory by its owners because of the difficulties arising from the many automatic controls employed. An attempt was made to secure limited operating data on the unit, but it was impossible to complete the arrangement to do so. The gasoline engine was replaced by an electric motor during the heating season.

INSTRUMENTATION

Instruments for determining day to day performance were installed on Units No. 1, 2, 3, 4, 5, and 6. A standard watt-hour meter of proper current rating, voltage, and phase was connected into the electrical circuit of each motor for the purpose of determining the electrical energy used each day. A water meter was installed in the outlet water line of each heat pump.

On Unit No. 5, two water meters were installed since the same water system was used to supply water for domestic purposes as well as for the heat pump. The second water meter was so connected as to measure all water pumped. It was then assumed that the electrical energy used by the water pump could be charged to the heat pump operation in the same ratio as the amount of water used by the heat pump to the total water pumped. When workmen replaced the small water pump on February 10, they reconnected the water meters so that only the water used by the heat pump could be measured.

A self-starting electric clock was connected into the circuit of each of the 6 heat pumps in such a way as to cause the clock to run only when the heat pump was operating. Thus, by having daily readings made on the clock, it was possible to determine the number of hours of heat pump operation each day. There was no reason to question the number of times the clock's hour hand had rotated between readings because the rate of water flow in every case was constant enough to provide a basis for estimating the water flow closer than to the nearest 12-hour interval.

Two clocks were used in connection with Unit No. 4. This unit, like Unit No. 5, was supplied by a water pump that also furnished water for domestic purposes. In the case of Unit No. 4, the second clock was used to time the operation of the water pump so that energy charges could be proportioned on the basis of water pump running time relative to heat pump running time.

Two mercury thermometers were used with each unit to measure the temperature of water flowing into and out of the heat pump. The thermometers were installed in special thermometer wells extending down into the inlet and outlet water pipes. The thermometers used were graduated to 2/10 of 1 degree Fahrenheit and could be read to 1/10 of a degree.

Recording wattmeters with appropriate current transformers were connected into the compressor motor circuits of Units No. 1, and No. 2. Similarly, a recording ammeter was connected in the circuit of Unit No. 3. These instruments were used for the following 3-fold purpose:

1. To determine the exact times of operation of the units.

2. To determine motor operating characteristics from the beginning to the end of each operating cycle.

3. To aid in the detection of any unusual conditions affecting the operation of the heat pump system.

Additional instruments and equipment used for carrying out the detailed analysis of 3 of the heat pump systems were installed on each unit only while it was undergoing the test. The instruments used and their functions were as follows:

1. Stop watches for timing.

    a. Rate of water flow.

    b. Revolutions of each watt-hour meter disc.

    c. Motor and compressor R.P.M.

2. One 16-point electronic temperature recorder with copper-constantan thermocouples used for recording temperatures.

    a. In the refrigerant system.

    b. In water system.

    c. In the air system.

3. Two pressure gauges for measuring high-side and low-side refrigerant

pressures.

4. Two special side panels for the compressor enclosure to provide access to the compressor, motor, and related equipment during the tests.

Plate II shows Unit No. 3 and some of the instruments used in the tests. Plate III shows the side panels of Unit No. 3 removed. Plate IV is a view of the instrument panel used with Unit No. 2.

PROCEDURE

The owners of the several heat pumps were interviewed and agreements were made providing for cooperation in obtaining the desired data. For various practical reasons, it was impossible to get the necessary data on 3 of the installations. Enough data was obtained through the cooperative efforts of the owners of 5 of the units to fulfill the first objective of this investigation.

Daily Operating Data

After each heat pump was put into operation and instruments were installed, the owner was supplied with data sheets and instructions for making daily readings on the instruments.

Readings were made at as nearly the same time each day as was possible. The time was usually early morning when it was most certain that the unit would be operating. The cooperating owners were instructed to make readings on the water temperatures only after it was known that the unit had been operating for a period of 5 minutes or more. This precaution was necessary because at least 5 minutes was usually required for the outlet water temperature to reach a steady value.

EXPLANATION OF PLATE II

View of Unit No. 3 with special side panel installed. Thermocouple wires may be seen between the electronic temperature recorder and the heat pump cabinet. A pressure gauge and a compound gauge are visible at the upper right hand corner of the compressor cabinet. Two mercury thermometers may be seen installed in the water line thermometer wells. Immediately adjacent to the thermometer on the left is a control switch which stops the unit if the water pressure falls below a set value. Above the thermometer and the pressure gauges is the control panel. Between the two thermometer wells is the hand valve used for throttling the water flow. A multi-meter, a recording ammeter and an indicating wattmeter are also shown. The watthour meters, water meter, and electric clock used in the tests are not visible.

PLATE II

EXPLANATION OF PLATE III

View of Unit No. 3 with the side panel removed and the end panel open. The coil below the motor and compressor is the high side liquid to low side vapor heat exchanger. Enclosed behind this heat exchanger is the unit which acts as the evaporator on the heating cycle. The hand valves for switching from heating to cooling and the expansion valves are barely visible through the open end panel.

PLATE III

EXPLANATION OF PLATE IV

View of the instrument panel for Unit No. 2. At the extreme left are two current transformers that were used with the recording wattmeter which is barely visible at the right margin. To the left of the current transformers is a recording thermometer. One bulb of the thermometer is located outside the house for recording the outside temperature. On the instrument board are 2 switch boxes, 3 watt hour meters and an electric clock. Below the board is the electronic temperature recorder. The water meter and mercury thermometers are not visible.

PLATE IV

Instrumentation was not completed on Unit No. 1 until February, 1949, and the first data were recorded on December 20. Unit No. 3 was placed in operation on January 19, 1950. The taking of data was started immediately. Instrumentation of Units No. 4 and 5 was completed in the latter part of November, 1949. Observations were started at that time. Data included in this thesis were taken between the starting dates indicated for the several units and May 1, 1950.

Under the provisions of the agreements, it was difficult at times to get complete and accurate data. However, with the knowledge gained from the detailed studies of Units No. 1, No. 2, and No. 3 along with frequent checks by the investigators, reasonably satisfactory results were obtained.

Heat Pump System Analysis

The data taken in cooperation with the owners were of value in determining the actual operating costs of the units, the quantities of water used and performance factors of the individual units. They did not, however, provide a means for answering the following important questions:

1. Were the units designed for best efficiency in the light of present knowledge of refrigeration equipment?

2. Were the installations including water supplies and air duct designs engineered to provide the best obtainable performance from the units?

3. What factors relating to design and installation would significantly influence the over-all performance of the units?

4. If differences were noted in the performance of the several units, to what factors could such differences be attributed?

The technical analysis of Units No. 1, 2, and 3 was intended to provide answers to the above questions.

Theoretical considerations indicated that, for any given vapor compression refrigeration system, the performance would be significantly influenced by the temperature limits between which it was operated. With the temperature of the water supplied to the evaporator being essentially a fixed value, it was apparent that the low temperature limit of the refrigeration cycle would be largely determined by the following considerations:

1. The rate of water flow through the evaporator which could be varied over a wide range.

2. The overall coefficient of heat transfer between the refrigerant and the water.

3. The effective area of heat transfer between the refrigerant and the water.

Effective values of the coefficient of heat transfer and the heat transfer area could be changed indirectly by adjustments of the thermostatic expansion valves to produce a change in the portion of evaporator surface that was wetted by refrigerant liquid. The relative extent to which the evaporator surface was wetted could be estimated by the degree of superheating of the refrigerant leaving the evaporator. Theoretical considerations further indicated that, with the temperature of air supplied to the heated space fixed at the lowest value compatable with satisfactory heating, the upper temperature limits of the refrigeration cycle would be largely determined by the following factors:

1. The efficiency of the condenser.

2. The temperature of air entering the condenser.

3. The rate of air flow over the condenser.

Aside from maintaining a clean condenser surface, little could be done by the investigators to influence its efficiency. The temperature of the air entering the condenser was variable from one time to another, but the investigators could exercise only limited control over this factor. Fortunately the rate of air flow could be controlled within certain limits. Since Unit No. 3 had no return air ducts, the rate of air flow could be controlled over relatively wide limits with this unit. Moreover, the effect of air temperature and air flow could be combined for the purposes of this test and the total effect could be controlled by changing the air flow alone. The relative value of this combined effect was evidenced by the condensing pressure of the refrigerant.

A test procedure based upon the foregoing considerations was established. The procedure was divided into two primary series of tests:

1. With the condensing pressure (hereinafter to be designated as the "head pressure") held at a constant value, the rate of water flow was varied in steps from a practical minimum to the highest value obtainable with the equipment. This series of tests was conducted on Units No. 1, 2, and 3 as a basis for comparing the performance of the 3 units.

2. With the rate of water flow remaining fixed, tests were conducted with head pressure adjusted to the lowest obtainable value and to the highest practical value. These tests were conducted on Unit No. 3 only.

In each test, data was not taken until the system had reached a steady state of operation for each new condition. In each series of tests, data was accurately read and recorded under the following headings:

1. Electrical energy consumption.
    a. Compressor motor.
    b. Fan motor.
    c. Water pump motor.
2. Rate of water flow over the evaporator.
3. Temperature of water.
    a. Entering evaporator.
    b. Leaving evaporator.
4. R.P.M. of compressor.
5. Refrigerant pressures.
6. Temperature of refrigerant.
    a. Liquid entering expansion valve.
    b. Liquid entering evaporator and vapor leaving evaporator.
    c. Vapor entering heat exchanger and leaving heat exchanger.
    d. Vapor entering compressor.
    e. Vapor leaving compressor.
    f. Vapor entering condenser and liquid leaving condenser.
    g. Liquid entering heat exchanger and leaving heat exchanger.
7. Temperature of air.
    a. Entering condenser.
    b. Leaving condenser.

With these data it was possible to undertake a complete analysis of the refrigeration system under a given set of test conditions.

A secondary series of tests was undertaken in which all changes in the system were observed from the beginning of an operational cycle until all functions of the cycle had reached a steady state. Tests of this nature were conducted with the rate of water flow remaining constant for any one test but with the rate changed for different tests. Further tests were conducted in which the rate of water flow remained unchanged for different tests, but the ultimate value of head pressure reached at the end of each test was changed.

The investigators were not at liberty to change the adjustments on the expansion valves of the units for the purposes of these tests. However, it was necessary to make adjustments on the expansion valves of Unit No. 3 when it was first put into operation. This gave the investigators the opportunity to adjust the valves to what they considered at the time to be near an optimum setting. The degree of superheating of the vapor leaving the evaporator turned out to be different from that in Units No. 1 and No. 2. This afforded an opportunity to make limited comparisons of the units on the basis of the relative extent of wetting by the refrigerant of the evaporator surface.

RESULTS OF THE HEAT PUMP TESTS

Costs of Operation and Water Requirements as Taken from Daily Records

The results of the daily operating data are summarized on a monthly basis in Table 2. It may be observed that in every case the fan and water pump motors were a significant part of the electrical load. Tabulated on a monthly basis, the quantities of water used appear to be large. Until the new water pumps were installed on Units No. 4, 5, and 6 in the middle of March, none of the 3 units used as much water as was recommended by the manufacturer. The water consumption of Unit No. 3 was from 20 to 30 percent under what it would have been if the manufacturer's recommendations had been followed.

The cost of heating with the heat pump in terms of dollars per million Btu may be compared with the calculated cost of heating with conventional fuels as given on page 69.

Ample allowance must be made for the accuracy of the data taken in the daily records. The data on the kilowatt hour consumption of the motors are of satisfactory accuracy. The recorded electrical energy consuption of the water pumps for Units No. 3, 4, and 5 included, in some instances, energy for pumping water that was not used by the heat pumps. It was possible to make satisfactory corrections for this so that the percentage errors in total kwhr's. were small.

The figures given for water consumption are substantially correct. It was necessary to apply corrections to the figures where water was used for purposes other than the heat pumps.

The records of the temperature difference between the water entering and that leaving the heat pump were the greatest source of error. It was impossible to obtain consistantly accurate readings on several of the units. An error of a fraction of a degree in these readings was large percentage-wise. The relative inaccuracy of these readings was reflected in the calculations for the heat taken from the water, the total heat output, the coefficient of performance, and the cost per million Btu.

From the results of the detailed tests, it is extremely doubtful that the actual coefficient of performance of any of the units ever exceeded 3.5 even though some higher values are shown in Table 2. The values of the coefficient of performance shown on Figs. 1, 2, and 3 may be taken as representative of actual performance.

Table 2. Monthly summary of heat pump operating data.

Unit 1	Dec.	Jan.	Feb. (15-28)	Mar.	Apr.
Kwhr. consumption					
Compressor motor			631.00	1,940.00	1,071.00
Fan motor			82.00	180.00	106.00
Water pump motor			107.00	274.00	209.00
Total			820.00	2,404.00	1,376.00
Water consumption (thousands of gallons)			104.70	289.00	190.20
Hours of operation			96.93	267.26	176.33
Average difference in temperature of water between inlet and outlet of evaporator (°F.)			7.20	7.40	7.50
Heat removed from water (millions of Btu)			6.28	17.82	11.88
Heat from motors (Compressor and fan, millions of Btu)			2.43	7.26	4.01
Heating output (millions of Btu)			8.71	25.08	15.89
Coefficient of performance			3.12	3.06	3.39
Cost of operation at 1.25¢ per kwhr. (dollars)			10.25	30.05	17.20
Operating cost per million Btu (dollars)			1.17	1.20	1.08
Degree days			683.00	692.00	
Average degree days at Wichita, Kansas			836.00	604.00	290.00
Unit 2					
Kwhr. consumption					
Compressor motor		2,490.00	1,641.00	1,842.00	889.00
Fan motor		134.00	93.00	82.00	41.00
Water pump motor		221.00	163.00	153.00	79.00
Total		2,845.00	1,947.00	2,077.00	1,009.00
Water consumption (thousands of gallons)		249.65	248.63	236.74	135.60
Hours of operation		415.80	277.75	261.50	127.00
Average difference in temperature of water between inlet and outlet of evaporator (°F.)		9.42	7.84	8.00	6.60
Heat removed from water (millions of Btu)		19.60	16.20	15.80	7.45

Table 2. (cont.).

Unit 2	Dec.	Jan.	Feb.	Mar.	Apr.
Heat from motors (Compressor and fan, millions of Btu)		8.95	6.80	6.57	3.17
Heating output (millions of Btu)		28.55	23.00	22.37	10.62
Coefficient of performance		2.94	3.46	3.16	3.09
Cost of operation at 1.25¢ per kwhr. (dollars)		35.60	24.35	25.95	12.60
Operating cost per million Btu (dollars)		1.25	1.06	1.16	1.18
Degree days		1,046.00	683.00	692.00	
Average degree days at Wichita, Kansas		1,016.00	836.00	604.00	290.00
Unit 3					
Kwhr. consumption					
Compressor motor			672.50	584.00	362.00
Fan motor			42.50	36.50	20.50
Water pump motor			119.00	97.50	51.00
Total			834.00	718.00	433.50
Water consumption (thousands of gallons)			72.80	60.32	31.94
Hours of operation			116.19	95.00	59.00
Average difference in temperature of water between inlet and outlet of evaporator (°F.)			10.50	12.70	13.00
Heat removed from water (millions of Btu)			6.36	6.38	3.45
Heat from motors (Compressor and fan, millions of Btu)			2.44	2.12	1.48
Heating output (millions of Btu)			8.80	8.50	4.93
Coefficient of performance			3.09	3.46	3.33
Cost of operation at 1.25¢ per kwhr. (dollars)			10.42	8.98	5.45
Operating cost per million Btu (dollars)			1.18	1.02	1.12
Degree days			683.00	692.00	
Average degree days at Wichita, Kansas			836.00	604.00	290.00

Table 2. (cont.).

Unit 4	Dec.	Jan.	Feb.	Mar.	Apr.
Kwhr. consumption					
Compressor motor	1,338.00	1,342.00	1,311.00	1,335.00	675.00
Fan motor	143.00	164.00	134.00	127.00	64.00
Water pump motor	157.00	177.00	176.00	187.00	94.00
Total	1,638.00	1,683.00	1,621.00	1,649.00	833.00
Water consumption (thousands of gallons)	180.70	216.00	193.04	269.00	
Hours of operation	239.00	310.00	275.00	213.00	108.00
Average difference in temperature of water between inlet and outlet of evaporator (°F.)	8.70	8.60	6.50	6.50	6.50
Heat removed from water (millions of Btu)	13.10	15.47	10.45	14.56	
Heat from motors (compressor and fan, millions of Btu)	5.05	5.13	4.93	4.99	
Heating output (millions of Btu)	18.15	20.60	15.38	19.55	
Coefficient of performance	3.25	3.59	2.79	3.48	
Cost of operation at 1.25¢ per kwhr. (dollars)	20.60	21.05	20.22	20.62	10.40
Operating cost per million Btu (dollars)	1.13	1.02	1.31	1.05	
Degree days	854.00	1,046.00	683.00	692.00	
Average degree days at Wichita, Kansas	947.00	1,016.00	836.00	604.00	290.00
Unit 5					
Kwhr. consumption					
Compressor motor		1,643.00	1,079.00	1,005.00	336.00
Fan motor		159.00	100.00	95.00	32.00
Water pump motor		249.00	146.00	132.00	50.00
Total		2,051.00	1,325.00	1,232.00	418.00
Water consumption (thousands of gallons)		290.00	237.55	196.50	
Average difference in temperature of water between inlet and outlet of evaporator (°F.)		8.20	5.10	5.10	
Heat removed from water (millions of Btu)		19.80	10.08	10.00	

Table 2. (concl.).

Unit 5	Dec.	Jan.	Feb.	Mar.	Apr.
Heat from motors (compressor and fan, millions of Btu)		6.14	4.02	4.20	
Heating output (millions of Btu)		25.94	14.10	14.20	
Coefficient of performance		3.70	3.12	3.34	
Operating cost at 1.25¢ per kwhr. (dollars)		25.70	16.58	15.40	5.22
Operating cost per million Btu (dollars)		0.99	1.17	1.08	
Degree days		1,046.00	683.00	692.00	
Average degree days at Wichita, Kansas		1,016.00	836.00	604.00	290.00

Technical Analysis of the Refrigeration System

The Effect of Varying the Water Flow Rate. The rate of water flow through the evaporator proved to be the most significant single variable considered in these tests. This fact had been anticipated by the investitators. It was apparent that, with the water pumps used to supply water to the heat pumps in these tests, any increase in the rate of flow of water would almost certainly improve the overall performance (including economy of operation) of any particular heat pump-water pump combination.

The problem would then appear to be that of determining the limit to which it was practical to increase the flow of water and of ascertaining the rate and extent of improvement in performance. These statements, circumscribed as they are by limiting conditions, were proved to be true. It was further proved that for the more general case, increasing the rate of water flow would increase the heating capacity but would not necessarily improve the economy of operation.

The curves plotted in Figs. 1, 2, and 3 show the changes that took place in the refrigeration systems of Units Nos. 1, 2, and 3 respectively as a result of increasing the water flow rate. The changes took place with the head pressure of each unit maintained at a constant value.

The following curves were plotted from experimental data for each of the 3 units:

1. "Water Temperature Change" which represents the number of Fahrenheit degrees that the water temperature was lowered in passing through the heat pump evaporator.

2. "Fan Motor" power input in kilowatts.

3. "Water Pump Motor" power input in kilowatts.

4. "Heat Pump Motor" which represents the power input in kilowatts to the compressor motor of the heat pump as well as the almost negligible power input to the system controls.

5. "Total Power Input" which is the sum of items 1, 2, and 3 above.

6. "Suction Pressure" which is the gauge pressure in pounds per square inch that existed in the low pressure side of the refrigeration system.

The remaining curves which are listed below are plots of values which were calculated from the experimental data:

1. "Refrigeration Output" is the product of the "Water Rate" converted to pounds per hour, the "Water Temperature Change", and the specific heat of water which was taken as unity. The refrigeration output is expressed in thousands of Btu's per hour. Use of the term "Refrigeration Output" is not intended to infer that it is any quantity supplied by the heat pump and directly useful for heating purposes. It represents the refrigeration

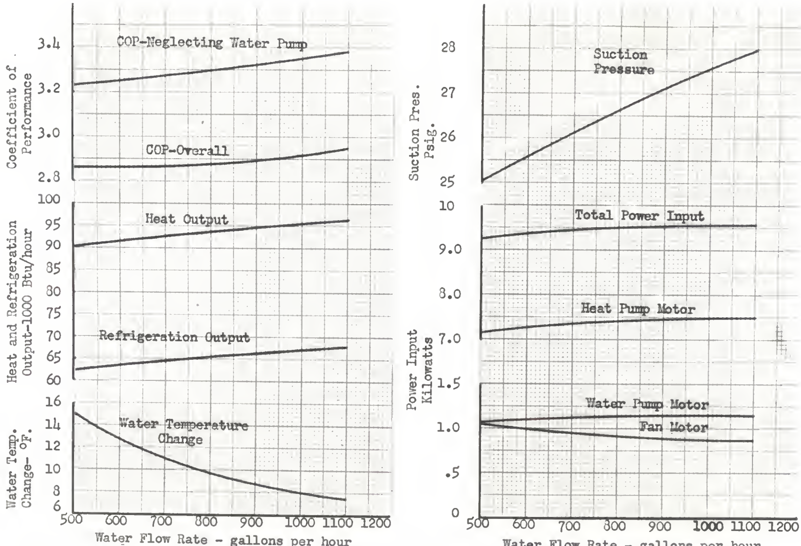


Fig. 1. Operating characteristics of Unit No. 1 at different rates of water flow.

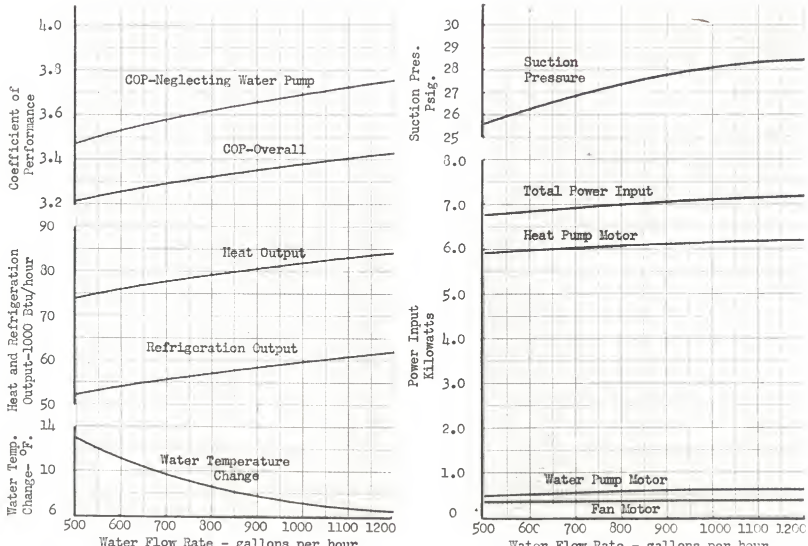


Fig. 2. Operating characteristics of Unit No. 2 at different rates of water flow.

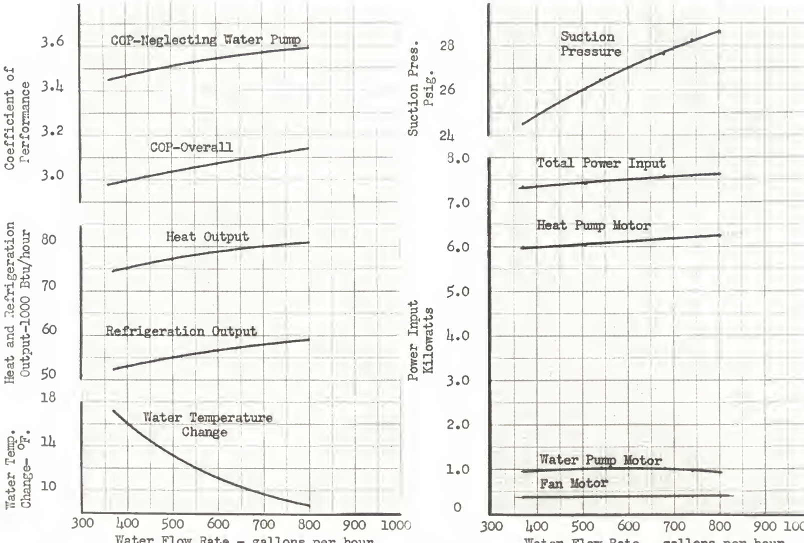


Fig. 3. Operating characteristics of Unit No. 3 at different rates of water flow.

capacity of the vapor compression system and is therefore the rate of heat removal from the water.

2. "Heat Output" is expressed in the same terms as the refrigeration output. It is the sum of the refrigeration output plus the heat equivalent of all the electrical energy supplied to equipment which was located in the heated space or in the path of air circulated to the heated space.

3. "COP-Overall" represents the overall coefficient of performance for the heat pump system. It is expressed as the ratio of the heat output to the Btu equivalent of the total electrical energy consumed by the heat pump compressor, water pump, and fan motors and controls. Since the water pump motors were not located in heated space or in the path of air circulated to heated space, the electrical energy input to the water pump motors could not be considered as contributing to the heat output.

4. "COP-Neglecting Water Pump" represents the value of the coefficient of performance that would have been obtained if the electrical energy consumption of the water pump had not been included in the calculation. These values are not significant in themselves except to the extent that they indicate the effect of water pumping costs on the overall coefficient of performance and, therefore, emphasize the importance of wise pump selection. Values taken from these curves were used in the computations for Figs. 4, 5, and 6 to determine the most economical water flow rates at different water pumping costs.

The individual components of heat pump performance that are represented by the curves were functions not only of the water rate but, in most cases, were interrelated one to another.

Significance of the Water Temperature Change. The temperature of water pumped from the well and supplied to the evaporator was always constant for any series of tests on a given heat pump. As the rate of water flow was increased, the temperature of the water at the outlet of the heat pump evaporator raised. The results are shown on the curves as a decrease in the water temperature change. The decrease in temperature change, however, was relatively less than the increase in water flow. This may be accounted for by the fact that the water velocity through the evaporator was greater for the higher rates of water flow. Thus, according to fundamental theories of fluid flow and of heat transfer, the water (stagnant) film immediately adjacent to the evaporator surface was thinner resulting in a larger water film coefficient of heat transfer.

The temperature of the water at the outlet of the evaporator was relatively high at high rates of water flow. At the same time, the inlet water temperature remained constant. It is obvious from these facts that the mean temperature of the water in the evaporator was higher for high water flow rates than it was for low water flow rates. The net effect was the same as if a heat source of higher temperature had been provided. According to theory, this should result in higher refrigeration output, higher heat output, and improved coefficient of performance.

Explanation of the Changes in Refrigeration Output. It has been stated that the refrigeration output was influenced by the mean temperature of the water in the evaporator. It was further influenced by at least two other factors which were closely related to the ascending suction pressure curve.

1. When the head pressure of the compressor remained constant and the suction pressure increased, the compression ratio was decreased. According to the Refrigeration Data Book (5), a decrease in compression ratio is

accompanied by an increase in the volumetric efficiency of a given refrigeration compressor. Thus, the increase in volumetric efficiency caused by a higher suction pressure resulted in a larger effective compressor displacement and enabled the compressor to handle a greater volume of refrigerant per unit time. Even if the specific volume of the refrigerant had remained constant, the result would have been an increase in the refrigerant output.

2. The specific volume decreased at the higher suction pressure. Thus, more pounds of refrigerant were handled for a given volume displacement.

It is unquestionable that there is a close relationship among the factors listed as influencing the refrigeration output. With a higher mean temperature of water in an evaporator, it is possible with a thermostatic expansion valve for the refrigerant to evaporate at a higher temperature. This causes an increase in the suction pressure.

Explanation of the Changes in Heat Pump Motor Input. The curves show that the measured input of electrical energy to the heat pump or compressor motor increased with the increase in water flow rate. It has been shown how the simultaneous increase in suction pressure logically resulted in the pumping of a greater weight of refrigerant vapor per unit of time. The load on the compressor was thus increased and required a greater input of energy to the motor.

The increased water flow rate was also accompanied by an increase in the heat output. Consequently it was necessary for the condenser to transfer heat to the circulated air at a greater rate. Under ordinary conditions of operation, the necessity for more rapid transfer of heat from the condenser would have resulted in a higher condensing temperature and therefore a higher head pressure. At the higher head pressure, the work of compression would have been increased accordingly. Opposing this tendency would have been the effect of increased compression ratio upon the quantity of refrigerant handled. The net effect, however, as indicated by results of other tests to be described would have been to further increase the power requirement.

In these tests the head pressure was held constant by regulating the quantity of air flowing over the condenser surface. The recorded change in heat pump motor input is therefore less than normal.

Explanation of the Changes in Heat Output. The factors that caused the increase in heat output as shown on the curves have already been mentioned indirectly in the discussions of other functions of the water flow rate and the refrigeration system. The energy components that constituted the heat output have also been discussed. However, a more specific definition is required. In these tests, the heat output was the sum of the refrigeration output, the work of compression on the refrigerant vapor, and that portion of the heat loss from electrical equipment which was directed to the heated space. Since the work of compression was supplied by the heat pump motor, which was located in the air stream, the heat loss from the motor and the work of compression constituted 100 percent of the motor input. The heat of compressions entered the calculations in this form.

Technically, the heat of friction caused by the pressure drop of the water through the evaporator contributed to the heat output. But at 1,100 gallons per hour and a friction pressure drop of 5 pounds per square inch, the heat of friction would amount to only 140 Btu per hour which is a negligible quantity.

Fig. 1 shows that, in the test on Unit No. 1, the increase in refrigeration output between the limits of 500 and 1,100 gallons per hour was 5,500 Btu per hour. The increase in heat output between the same limits was 6,000 Btu per hour. According to this the actual increase in heat output was derived almost altogether from the increase in refrigeration output. The explanation for this is found on the curves for the heat pump motor and the fan motor. Whereas, the electrical input to the heat pump motor increased 0.3 kilowatts, the input to the fan motor decreased 0.325 kilowatts, resulting in a net decrease in heat from the electrical equipment on the order of 85 Btu per hour. This leaves 585 Btu per hour unaccounted for which is within the limits of accuracy for reading the curves.

Similar reference to Fig. 2 indicates that the heat output increased 1,000 Btu per hour more between the limits of 500 and 1,200 gallons per hour than did the refrigeration output between the same limits. The energy input to the fan was the same at those limits while the energy input to the heat pump motor increased slightly less than 0.3 kilowatts. This energy increase nearly balances the difference between the heat output and the refrigeration output.

Factors Influencing the Coefficient of Performance. The coefficient of performance of each of the 3 units, that were subjected to the detailed tests, are shown on Figs. 1, 2, and 3. In every case, the coefficient of performance improved with an increase in the water flow rate. But the coefficient of performance of all 3 units was different at the same rate of water flow.

The most important factor contributing to the improvement of the coefficient of performance of any particular unit was the higher mean temperature of water in the evaporator at the increased rates of flow.

There were 2 principal factors contributing to the improvement in the coefficient of performance of the units at the higher rates of water flow. One factor was the higher mean temperature of the water in the evaporator. The other was the increased value of the water film coefficient of heat transfer. Both of these factors have been discussed previously in connection with the significance of the water temperature change.

Calculations based upon the experimental data showed that the logarithmic mean temperature difference between the refrigerant and the water in the evaporator remained essentially constant for a given heat pump unit. Stated another way, the suction temperature of the refrigerant followed directly any increase in the mean temperature of the water. This obviously contributed toward the improvement in the coefficient of performance.

The increase in the film coefficient of heat transfer was a further influence towards bringing the suction temperature closer to the water temperature.

Examination of the curves reveals that there was a significant difference in the suction pressures of the 3 units. Unit No. 1 with the lowest coefficient of performance had the lowest suction pressure. It is logical that this condition existed.

The evaporator of Unit No. 1 was identical with the evaporators of the other two units. Examination of the refrigeration output curves shows that Unit No. 1 had a higher refrigeration output than the other units. This was because the compressor was operated at a higher speed and capacity. Since the evaporators on all units had the same physical demensions, the evaporator loading was greater on Unit No. 1. Therefore, it was necessary for the suction pressure and the corresponding suction temperature to be lower in order to effect the greater heat transfer rate.

The coefficient of performance of Unit No. 2 was higher than that for Unit No. 3 even though both units had approximately equal evaporator loadings. This was true in spite of the fact that Unit No. 3 apparently had the more efficient refrigeration cycle. The relative efficiencies of the two refrigeration cycles was indicated by the difference in the suction pressures. In spite of having a lower temperature heat source, the evaporator pressure and temperature of Unit No. 3 was higher. Thus, a more efficient evaporator in Unit No. 3 allowed the refrigerant temperature to approach more closely the temperature of the heat source water.

The principal causes for the lower coefficient of performance of Unit No. 3 were that it had a lower temperature heat source than Unit No. 2, and it was powered by a single phase electric motor; whereas, Unit No. 2 was powered by a more efficient 3 phase motor. Moreover, the water pumping cost for Unit No. 3 was higher. Another factor that may have influenced the coefficient of performance of Unit No. 3 was a low voltage condition in the electric energy supply.

<u>Fan Motor, Fan, and Air Ducts</u>. Figure 1 shows a reduction in the energy input to the fan at the higher water flow rates. This reduction does not seem to be compatable with the obvious necessity for increasing the quantity of air flow over the condenser to maintain a constant head pressure at the increased rates of heat output. The explanation is not difficult. Because of certain characteristics of the air distribution system, a discussion of which follows, it was impractical to regulate the rate of air flow from the inlet side of the fan.

Air was permitted to escape between the discharge side of the fan and the condenser. More air was allowed to escape when the heat output of the condenser was low at low rates of water flow. The escape of this air reduced the static discharge pressure. The result was a greater discharge volume and a higher power input to the motor. The normal power input to the fan was the lowest point on the curve.

The fan motor curve in Fig. 2 is a straight horizontal line. This could be considered as indicating that the fan load curve was flat enough that no measurable increase in load was caused by the small increase in the air handled to maintain a constant head pressure.

The fan motor curve in Fig. 3 rises very slightly with the rising water rate. This is what would normally have been expected on all the fans. Since Unit No. 3 had no return air ducts, it is possible that the fan characteristics were influenced less by the characteristics of the duct system than was the case with the other two units.

On both Unit No. 1 and Unit No. 2, observations concerning the duct systems were made incidental to regulating the air flow. The supply duct system of Unit No. 1 appeared to have inadequate capacity. This was indicated by the fact that the head pressure normally ran higher than was necessary for supplying air at a satisfactory temperature. When small quantities of heated air were allowed to escape from the supply air plenum, it was observed that the head pressure decreased and the fan motor input increased slightly as a result of circulating a larger quantity of air. On the other hand, when cool basement air was admitted to the return air plenum, the head pressure dropped because of the cooler air; but the change in fan motor input was too small to be observable. This was assumed to indicate that the resistance to air flow in the return ducts was not excessive.

Both the supply duct system and the return duct system of Unit No. 2 appeared to have inadequate capacity. The head pressure for the system operating under normal conditions was excessively high. Long runs of return ducts passed through unheated space under the house. This caused relatively large heat losses at low outside temperature conditions. Under such conditions, the temperature of the return air was low enough to have an appreciable effect in lowering the head pressure. But when the outside temperature was high enough that the heat loss from the return air was not excessive, the head pressure would build up to 185 pounds per square inch. Since the head pressure cut out switch was set at 185 pounds per square inch, this caused the system to enter a series of rapid on-off cycles.

Water Pump Motor Input and Pump Performance. In the process of the investigation, enough data were taken on the operation of each water pump to provide fairly complete information on the individual pump characteristics at the existing suction lifts.

The water pump motor curves shown in Figs. 1, 2, and 3 indicate that the energy input to the pump motors was not appreciably lower for the lower rates of water discharge. This characteristic may be attributed mainly to the fact that the lower rates of flow were attained by throttling in the pump discharge lines. The result was that the pumps worked against higher discharge heads. Throttling is the only practical means available to the operator of a given heat pump-water pump system for achieving economy of water use, where such economy might be desired. It may be observed, however, that such a procedure cannot result in economical water pumping costs. The curves show that when the quantity of water discharged per hour was decreased by 1/2 or more, the cost per hour of pumping was only slightly lowered. The resulting cost per gallon of water was therefore more than doubled.

The combination centrifugal-jet pump that supplied water for Unit No. 3 actually consumed slightly less energy at its highest rate of discharge, which was approximately 800 gallons per hour, than it did at a discharge rate of 400 gallons per hour.

A comparison of the water pump motor curves shows that the water pump of Unit No. 1 required an energy input of 1.15 kilowatts at a water discharge rate of 1,100 gallons per hour compared to a power input of 0.6 kilowatts to the water pump motor of Unit No. 2 at the same discharge rate. The difference may be attributed, at least in part, to the fact that the water pump of Unit No. 1 pumped against a greater head because of the distance and difference in elevation between the water pump and the heat pump.

The centrifugal-jet pump of Unit No. 3 required an energy input of 0.9 kilowatts at a discharge rate of 800 gallons per hour. The unit was operated a large proportion of the time, with a water flow rate of approximately 600 gallons per hour. The pump motor input at that point was 1.0 kilowatt. Thus, the unit cost of water at the same electric rate was 366 percent greater for Unit No. 3, operated at a water flow rate of 600 gallons per hour, than it was for Unit No. 2, operated at a water flow rate of 1,200 gallons per hour.

The four dashed curves on Fig. 4 and Fig. 5 show the total energy cost for operation of Unit No. 2 at different electrical energy costs. The two dashed curves entitled "Actual Cost" on Fig. 4 show the actual heating cost in dollars per million Btu when electricity costs 1.0¢ per kwhr. and when electricity is available at 1.5¢ per kwhr. and also when it is available at

2¢ per kwhr. Figures 4 and 5 were calculated from the data for Unit No. 2.

The dashed curves on Figs. 6 and 7 show the heating cost per million Btu for Unit No. 3 when the electrical energy cost is 1.0, 1.25, 1.50 or 2.0 cents per kwhr.

It may be observed on Fig. 6 that for an electric energy cost of 1.25¢ per kwhr. the heating cost per million Btu dropped from \$1.22 to \$1.165 when the water flow rate was increased from 400 to 800 gallons per hour. Likewise, reference to Fig. 4 shows that, based upon the same cost for electricity, the cost of heating with Unit No. 2 dropped from \$1.14 to \$1.06 when the water flow rate was changed from 500 to 1,200 gallons per hour. Comparison of the corresponding curves for Unit No. 2 and Unit No. 3 reveals that at 800 gallons per hour, the most economical point reached for Unit No. 3, the operating cost was \$1.165 per million Btu compared with an operating cost at the same water rate for Unit No. 2 of \$1.11 per million Btu.

Based on the curves of Figs. 2 and 3, showing coefficients of performance neglecting the water pump and including the water pump, the coefficient of performance of Unit No. 2 was reduced 10.5 percent at 800 gallons per hour and 11 percent at the maximum water flow rate when the energy input to the pump was considered. The coefficient of performance of Unit No. 3 was reduced 12.5 percent at 800 gallons per hour, which was also the maximum water flow rate when the energy input to the pump was considered.

Considerations Influencing the Choice of the Most Economical Rate of Water Flow. It cannot be assumed from the previous discussion that the economy of operation of the heat pumps discussed would continue to improve as long as the rate of water flow was increased. Except for Unit No. 1, the coefficient of performance increased at a decreasing rate as the water rate

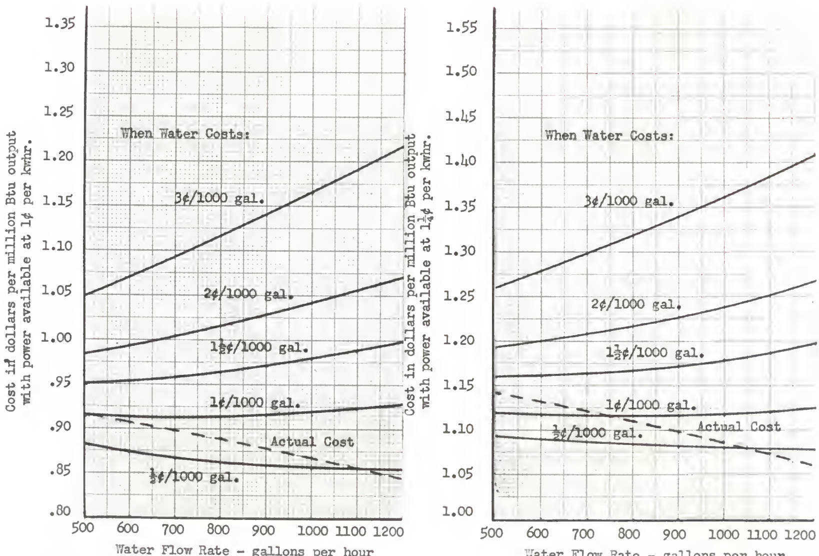
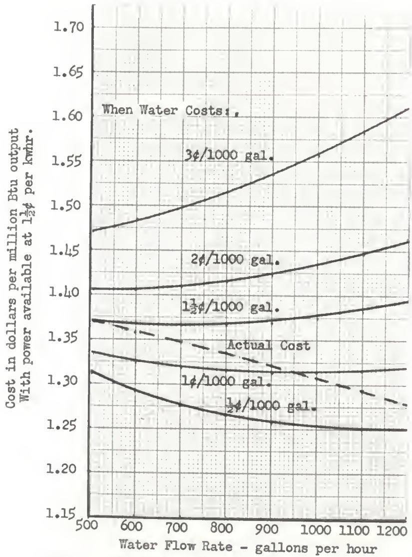


Fig. 4. Total energy cost for Unit No. 2.



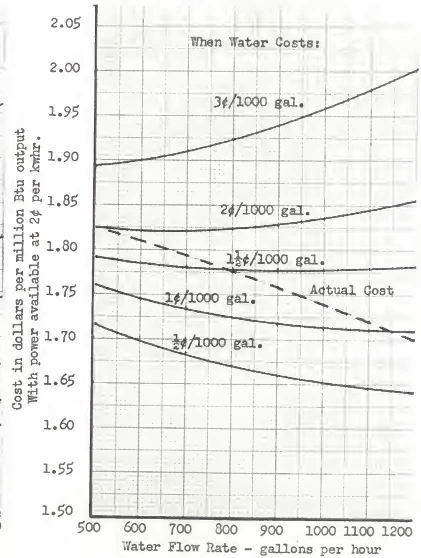


Fig. 5. Total energy cost for Unit No. 2.

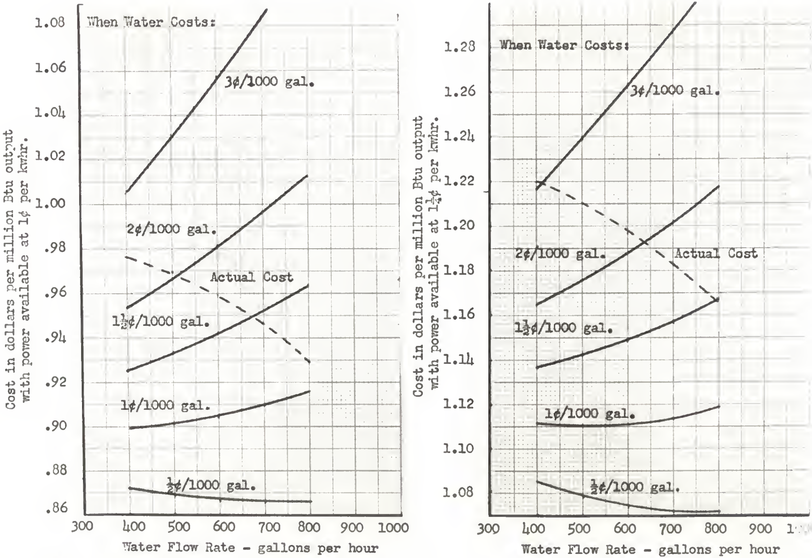


Fig. 6. Total energy cost for Unit No. 3.

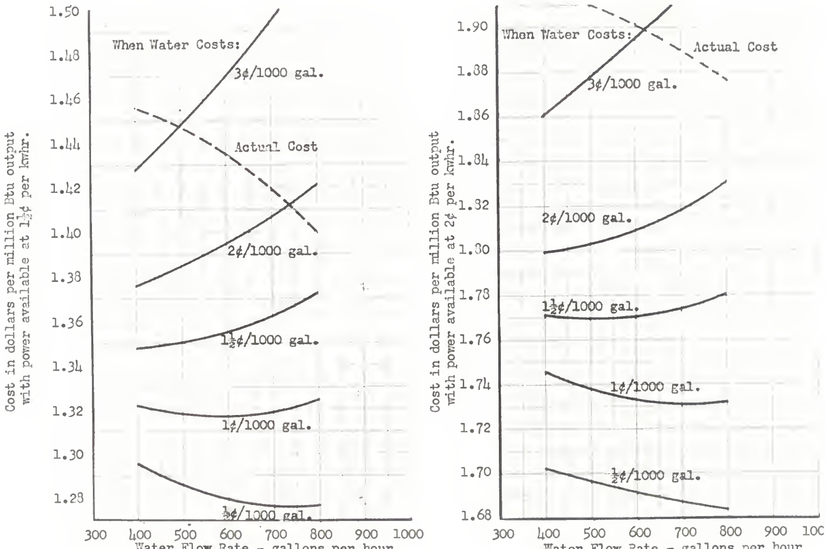


Fig. 7. Total energy cost for Unit No. 3.

increased. Obviously, a point would be reached where further increases in the water flow rate would have an exceedingly small effect on the water film coefficient of heat transfer and improvement in the coefficient of performance would be negligible. The curves for water temperature change have become relatively flat between the water rates of 1,000 and 1,200 gallons per hour. It is doubtful that there would be much to gain by using water rates in excess of 1,200 gallons per hour even if the cost of pumping water were very low.

Further study shows that under some conditions the best economy of operation would be at low rates of water flow even though the coefficient of performance of the heat pump unit itself; i.e. neglecting the water pump, would be low.

An analysis was made of the cost of heating at different electric energy costs from 1¢ per 1,000 gallons to 3¢ per 1,000 gallons, based upon the performance of Units No. 2 and No. 3. The results of this analysis are plotted on the solid line curves of Figs. 4, 5, 6, and 7.

To illustrate the use of the curves, let it be assumed that electric energy is available at a cost of 1¢ per kwhr. for operating a heat pump with performance characteristics similar to those of Unit No. 2. Let it be assumed further that by proper choice of a water pump for any chosen rate of discharge, water may be made available at 1/2¢ per 1,000 gallons at any pumping rate from 500 gallons per hour up to 1,200 gallons per hour. It will be seen from the curve representing these conditions in Fig. 4 that the most economical point of operation within the stated limits is at the highest discharge rate. On the other hand, if the conditions under which the water must be supplied are such that the water cost will be 1¢ per 1,000 gallons, the rate at which water is supplied will be of little importance. Between 500 gallons per hour and 1,000 gallons per hour, any difference in the cost of operation will be insignificant. At rates above 1,000 gallons per hour, the cost will increase slightly. According to computations made with values taken from Fig. 2, the same unit has a 13 percent higher heat output at 1,000 gallons per hour than it has at 500 gallons per hour.

In choosing the size of water pump for these conditions, it must be remembered that a large pump could be installed in consideration of high heat output. Should it then be required to economize on the use of water by throttling the pump, the total operating cost would be increased.

Now let it be assumed that the water must be drawn from a deep well or that other conditions are such as to raise the cost of pumping to 3¢ per 1,000 gallons. Obviously the most economical rate of water flow will be something less than 500 gallons per hour. This will probably be the lowest rate obtainable without freezing water in the evaporator. Again, however, it must be remembered that the water rate chosen must be sufficiently great to insure adequate heating capacity.

Comparison of Fig. 4 with Fig. 5 shows that if the cost of electrical energy for operating the heat pump is higher than 1.25¢ per kwhr. the economical advantage of using large rates of water flow applies at higher water supply costs.

Based upon the rate at which water was supplied to Unit No. 3 and upon the cost of electrical energy that was used in pumping at that rate, the cost of water supplied was 2¢ per 1,000 gallons. As examination of Fig. 6 with special reference to the curves for an electrical energy cost of 1.25¢ per kwhr. shows that the cost of operation of Unit No. 3 could be easily reduced.

In the first place, if it has been true that 2¢ per 1,000 gallons was the lowest price at which water could be pumped, then it would have been more economical to use a smaller pump which, at its full rated discharge, would have pumped about 350 to 400 gallons per hour. In this case, the saving of only 2¢ per million Btu would not have been as important as the saving on the original price of the pumps. It would have been impossible to use a much slower rate of water flow without the danger of freezing water in the evaporator.

The above is not a satisfactory solution since it is obvious that water could have been pumped at a cost lower than 2¢ per 1,000 gallons. The pump that was installed delivered water at full discharge for a cost of 1.4¢ per 1,000 gallons. Even if this had been the lowest pumping cost attainable, the curves show that a low rate of water flow from a small pump would have given the greatest economy.

The greatest falacy with either of the above solutions is that the heating capacity would have been sacrificed for small gains in economy of operation.

It is almost certain that a water pump could have been chosen that would have supplied water at a cost of not more than 1.0¢ per 1,000 gallons. At that water cost, a pump could have been selected with reference to the heating capacity required and to the capacity of the well and the water disposal system. For as far as economy of operation is concerned, the amount of water used would have had only a small influence on the total cost per unit of heat supplied.

Influence of the Head Pressure upon the Performance. One series of tests was conducted on Unit No. 3 in which the head pressure was held constant at 150 pounds per square inch, gauge, and the rate of water flow was changed in steps from 400 to 800 gallons per hour. The results of such a series of tests are plotted on Fig. 3. Another series of tests was conducted in which the same rates of water flow were used but the head pressure was held constant at 180 pounds guage pressure. The change in the head pressure caused corresponding changes in the characteristics represented by the curves of Fig. 4.

The coefficient of performance was reduced by a practically constant amount throughout the range of water flow rates. At 400 gallons per hour, the coefficient of performance was 3 at the lower head pressure and only 2.8 at the higher head pressure. Similarily at a water flow rate of 800 gallons per hour, the coefficient of performance was 3.14 at the lower head pressure and only 2.93 at the higher head pressure.

The difference in the coefficients of performance was logical because the refrigeration output was decreased to a much larger extent than the heat output. The heat removed from the water was reduced approximately 2,000 Btu per hour at all rates of water flow when the head pressure was increased from 125 to 175 pounds per square inch. On the other hand, the reduction in the heat output was less than 400 Btu per hour at a water rate of 800 gallons per hour. The heat output was maintained at the expense of greater power input to the compressor. The results was the less favorable coefficient of performance. This is proved by the fact that the recorded power input to the heat pump motor was increased by 0.57 kilowatts at the low rate of water flow and by .75 kilowatts at the high rate of water flow. These increases in the power input amounted to 9.5 percent and 12 percent respectively.

<u>The Influence of Expansion Valve Adjustment</u>. It has been stated that

the evaporator of Unit No. 3 was apparently more efficient than that of Unit No. 2. This was evidenced by its higher suction pressure and temperature in spite of its lower water temperature.

The thermocouple temperature records on Unit No. 3 showed that the temperature of the refrigerant vapor leaving the evaporator was 48° Fahrenheit at the same time that the suction pressure was 29 pounds per square inch, gauge. The saturation temperature of the refrigerant was therefore 31.5 degrees. The difference between 31.5 and 48 degrees (16.5 degrees) shows the amount of superheating done on the refrigerant vapor in the evaporator. Corresponding data on Unit No. 2 show that the refrigerant vapor leaving the evaporator was superheated 29.1 degrees when the suction pressure was 26 pounds per square inch. The smaller amount of superheating in Unit No. 3 is significant in that a greater proportion of the evaporator surface was wetted by the liquid refrigerant. It therefore had a lower average inside film coefficient of resistance. Moreover, a lower average evaporator surface temperature was maintained for a given saturation temperature. The total result was a larger effective overall coefficient of heat transfer for the evaporator.

The amount of evaporator superheating could be controlled by the expansion valves. Opening of the expansion valve orifices to permit fewer degrees of superheating raised the suction pressure for a given condition. It has been shown that raising the suction pressure has the effect of improving the coefficient of performance of the system.

DISCUSSION OF THE RESULTS

It was evident that most of the owners and some refrigeration service men had certain misconceptions about the operation of the heat pumps. This is to be expected since the principal of operation of the heat pump is a relatively new conception to the layman and since even the established conception of refrigeration is not well known to him in its fundamentals.

The effect of the rate of water flow and of the head pressure upon the operation of the heat pumps was not generally appreciated by the owners. Five of the 6 units observed were originally installed with water pumps of inadequate capacity. In several cases, the units were operated with the water flow throttled. Where the heat pump water supply was combined in one unit with the domestic water supply, the arrangement was generally unsatisfactory.

The air duct systems in several of the installations were not well designed for use with the heat pumps. All of the duct systems were new work and not simply adaptations from old mechanically circulated warm air systems. However, the ducts apparently had been designed on the basis of conventional warm air heating where bonnet temperatures are much higher than with the heat pump. The relatively low temperature at which air is delivered by the heat pump requires that a large volume of air be circulated. Ducts designed by the same rule of thumb methods that are used for mechanically circulated warm air systems are not adequate. This was the cause of the excessive head pressures encountered on Unit No. 1 and Unit No. 2.

Another misconception is that more heat is always delivered to the heated space when the delivery temperature of the air is high. The only means of securing a high delivery temperature with the heat pump is to operate at a high head pressure. It was shown in the results of the tests that the rate of removing heat from the water was reduced at high head pressures and that the only factor tending to maintain the heat output at a high level was the added power input to the system.

The manufacturer of the units sets the expansion valves at the factory. His setting is governed by the water temperature prevailing in the area to which shipment is scheduled. The expansion valves are thus used to adjust the loading of the motors for different temperatures of the heat source water.

Inproper adjustment of the expansion valves in the field could seriously impair the performance of the machines. On the other hand, the electric motors are located in the air stream and probably can be overloaded up to 25 percent without overheating. If the expansion valves on a unit are adjusted to permit the compressor motor to be somewhat overloaded, the superheating of the refrigerant can be limited and the coefficient of performance thereby improved.

The design of the evaporators of the units observed is such that the temperature of the water leaving the evaporators is rather high with respect to the refrigerant temperature. If this approach temperature could be decreased, especially in the 7 1/2 horsepower unit, the performance could be improved. The records showed that the approach temperature for Unit No. 1 using 1,000 gallons of water per hour was 22 degrees Fahrenheit. An accepted design value is approximately 10 degrees. If this value could be approached with a practical evaporator, the suction temperature and pressure would be raised. The result would be the same as if the temperature of the heat source had been raised by a similar amount. All things considered, the heat pump appeared to be well engineered and to give satisfactory service.

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APPENDIX A

Specifications for Marvair 5 H.P. Electric Unit

Heating output of unit at 50° input water.................80,000 B.T.U.
Cooling capacity...54,000 B.T.U.

Motor size - 5 H.P. single phase, repulsion-induction.
Fan motor - 1/2 H.P. G.E. single phase.
Compressor - G.E. CM802E6 - displacement 68.6 cu. in. per revolution.
Expansion valve - "Detroit" - four.
Air coil - six rows, face area 21 x 29.
Water coil - shell and tube dry type.
Controls - Allen Bradley or G.E. and Minneapolis-Honeywell.
Hi-Lo control - Ranco

Dimensions - width 27" - length 44" - height 34".

Plenum size 23 13/16" wide x 36 13/16" long.

Blower Specifications

Lau.......A12 squirrel cage blower.
1.44 sq. ft. outlet area.
2,000 cfm at 3/4" static pressure, adjustable.
Cabinet size - 30 x 30 x 35 1/2.
Return duct size 24 3/4 x 24 3/4.

Necessary water flow at 50° input to obtain heating capacity shown is 1,250 gallons per hour.

Reduced flow may be used as input temperature rises.

Specifications for Marvair 7 1/2 H. P. Electric Unit

Heating output of unit at 50° input water.................100,000 B.T.U.
Cooling capacity... 81,000 B.T.U.

Motor size - 7 1/2 H.P. G.E. three phase.
Fan motor - 1 H.P. single phase.
Compressor - G.E. CM802E6 - displacement, 68.6 cu. in. per revolution.
Expansion valves - "Detroit" - four.
Air coil - six rows, face area 21" x 29".
Water coil - shell and tube dry type.
Controls - Allen Bradley and Minneapolis-Honeywell.
Hi-Lo control - Ranco.

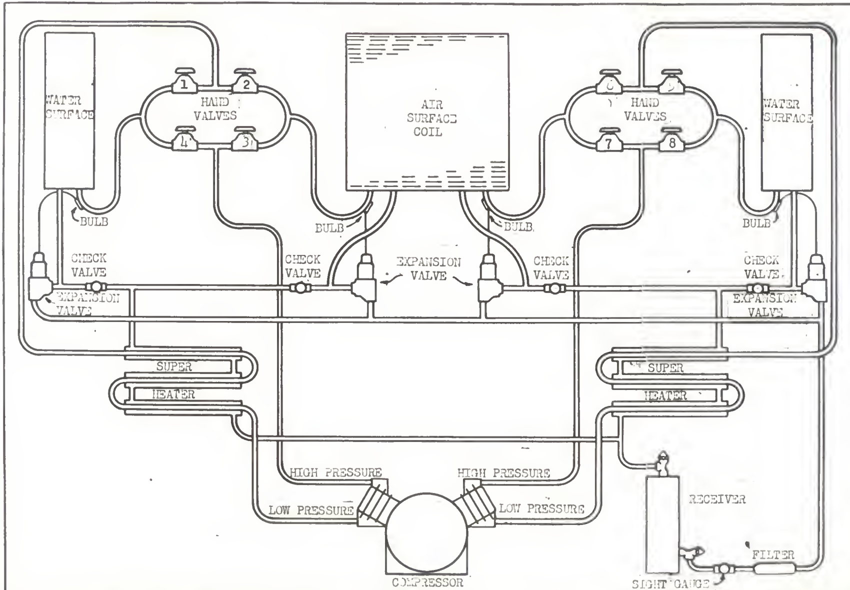
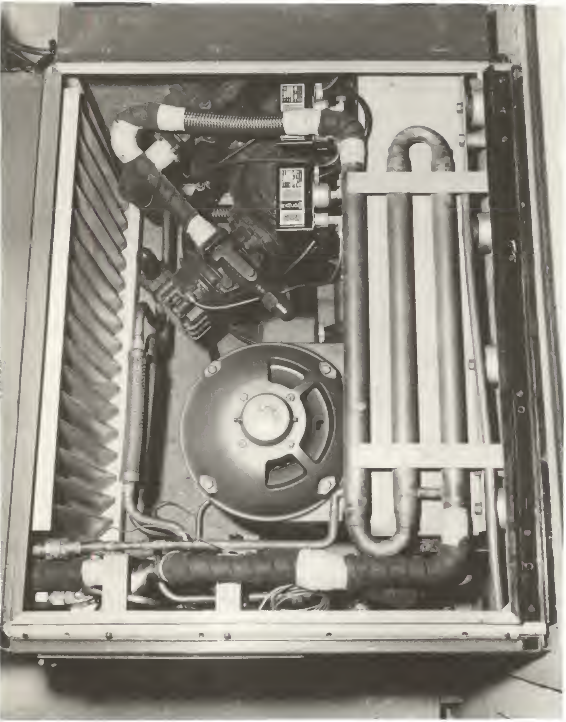


Fig. 8. Schematic diagram of the refrigeration system. (Furnished by Muncie Gear Works, Muncie, Indiana)

EXPLANATION OF PLATE V

Interior view of the compressor compartment showing the compressor, compressor motor, high side liquid to low side vapor heat exchanger, and the 2 high-low pressure cut out controls. The louvered panel above the compressor is a series of drain troughs to collect the water that drips from the evaporator on the cooling cycle. That same heat exchanger which is located directly above the troughs is the condenser on the heating cycle. The water to refrigerant heat exchanger which is the evaporator on the heating cycle and the condenser on the cooling cycle is hidden behind the metal plate below the motor and compressor.

PLATE V



Blower Specifications

Lau.......A15 squirrel cage blower.
2.01 sq. ft. outlet area.
2,750 cfm at 1" static pressure, adjustable.
Cabinet size 33 1/2" x 37" x 42" high.
Return duct size - 29 1/2" x 33".

Necessary water flow at 50° input to obtain heating capacity as is shown in 1,500 gallons per hour.

Reduced flow may be used as input temperature rises.

APPENDIX B

Calculations

<u>Heat Pump Output and Coefficient of Performance (Unit No. 1)</u>.

1. From Fig. 1, the water temperature change through the evaporator at 1,100 gallons per hour was 7.4 degrees F.

   Thus, the heat removed from the water (refrigeration output) was:

   $$7.4(1,100)\ 8.33 = 67,900 \text{ Btu per hour}$$

2. The heat output of the system will be equal to the heat removed from the water plus the electrical energy supplied to the heat pump motor and the fan motor. From Fig. 1, the power supplied to the heat pump motor at 1,100 gallons per hour was 7.50 kilowatts. At the same water rate, the fan motor drew .855 kilowatts. Then the total heat output was:

   $$(.855 + 7.500)\ 28,450 + 67,900 = 96,350 \text{ Btu per hour}$$

3. The coefficient of performance of the system neglecting the water pump was:

   $$\frac{96,350}{28,450} = 3.38$$

4. From Fig. 1, the power supplied to the water pump was 1.185 kilowatts at a water rate of 1,100 gallons per hour. Thus, the overall coefficient of performance was:

$$\frac{96,350}{(.856 \neq 7,500 \quad 1.185)\ 3,413} = 2.95$$

Seasonal Cost of Heating with Unit No. 1. The heat supplied by Unit No. 1 during March was 26,080,000 Btu. The number of degree days for March in the area was 692.4.

1. Thus, the heat loss from the building per degree day was:

$$\frac{26,080,000}{692.4} = 37,700 \text{ Btu degree day}$$

2. On the basis of a coefficient of performance of 2.95, computed above, and an electric rate of 1.2¢ per kwhr., the estimated cost for heating during March would be:

$$\frac{26,080,000}{2.95 \times 3.413} (\$.012) = \$30.95$$

3. The actual cost of heating for March with electricity figured at 1.2¢ per kilowatt hour was $28.85.

4. The average number of degree days per heating season for the area is 4,644. Therefore, the cost of heating during a normal year would be:

$$\frac{37,700 \times 4,644}{3,413 \times 2.95} = \$174.50 \text{ per season}$$

Estimating the Cost of Heating any House. The cost of heating any house with the heat pump may be estimated if the cost of electricity and the estimated heat load for the house are known. As an example, the following conditions are assumed:

1. A 6-room, well insulated house, located near Wichita, Kansas.

2. The cost of electricity is 1.2¢ per kilowatt hour.

3. The coefficient of performance may be taken to be 3.0.

4. The outside design temperature is -10 degrees Fahrenheit.

5. The inside temperature is maintained at 75 degrees Fahrenheit.

Then the design heat loss of the house would be about 60,000 Btu per hour. The normal number of degree days is 4,644. The heat loss per degree day will be:

$$\frac{60,000 \times 24}{75 \neq 10} = 16,950 \text{ Btu}$$

The cost of heating for a normal season will be:

$$\frac{16,950 \times 4,644}{3 \times 3,413} = (\$.012) = \$92.20$$

APPENDIX C

Comparative Costs of Fuels and Electricity for Home Heating

Heating System	Unit Cost	Cost Per Million Btu
Direct electrical heater	1¢ per kwhr.	$2.94
Direct electrical heater	1 1/2¢ per kwhr.	4.41
Direct electrical heater	2¢ per kwhr.	5.87
Heat pump Coefficient of performance = 3	1¢ per kwhr.	.98
Heat pump Coefficient of performance = 3	1 1/2¢ per kwhr.	1.47
Heat pump Coefficient of performance = 3	2¢ per kwhr.	1.96
Heat pump Coefficient of performance = 4	1¢ per kwhr.	.75
Heat pump Coefficient of performance = 4	1 1/2¢ per kwhr.	1.10
Heat pump Coefficient of performance = 4	2¢ per kwhr.	1.47
Stoker coal (12,000 Btu/lb.) furnace (efficiency 65%)	$10 per ton	.64
" "	$15 per ton	.96

Heating System	Unit Cost	Cost Per Million Btu
Stoker coal (12,000 Btu/lb.) furnace (efficiency 65%)	$20 per ton	$1.28
Fuel oil (140,000 Btu/gal.) furnace (efficiency 70%)	10¢ per gal.	1.02
" "	15¢ per gal.	1.53
" "	20¢ per gal.	2.04
Natural gas (1000 Btu/ft.^{3}) furnace (efficiency 80%)	40¢ per cu. ft.	.50
" "	80¢ per cu. ft.	1.00
Liquefied (100,000 Btu/gal.) petroleum (efficiency 80%) gas furnace	10¢ per gal.	1.25
" "	15¢ per gal.	1.875
" "	20¢ per gal.	2.50