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# COMPARISON OF FIELD PERFORMANCE OF A HIGH EFFICIENCY HEAT PUMP WITH AND WITHOUT A DESUPERHEATER WATER HEATER*

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## ABSTRACT

An air-to-air heat pump (COP=3.11 at 47°F (8.3°C)) run alternately with an electric-resistance water heater and a desuperheater water heater was tested under field conditions at a site near Knoxville, TN. Results from these tests indicate that the desuperheater system has an annual performance factor (measured over a six-month period consisting of three winter and three summer months) 14% better than that of the system using electric resistance water heating. An analytical model predicts a 21% improvement over an entire year with average weather for the desuperheater system. The simple payback period ranges from seven years (for a 21% improvement) to about nine years (for a 14% improvement).

## INTRODUCTION

Recent energy price increases have led researchers in both private and public sectors to look for ways to make more efficient use of equipment for building space conditioning and water heating. One way is to use a desuperheater, or heat recovery water heater, with air conditioners and heat pumps, thereby making effective use of normally rejected energy. There are numerous desuperheater units on the market for retrofitting existing systems, and some HVAC equipment manufacturers are offering desuperheater water heaters (DSWH) as options for their central air conditioners and heat pumps.

Analytical studies at the Oak Ridge National Laboratory (ORNL) have shown that a residential space conditioning and water-heating system consisting of a heat pump having a rated COP of 2.75 at 47°F (8.3°C) and a DSWH can achieve an annual coefficient of performance (ACOP) improvement of 15% over that of the same heat pump with an electric resistance water heater (Nephew and Abbatiello 1982). It is, therefore, desirable to obtain field performance data for such a system to verify its potential for energy conservation.

The purpose of this paper is to report on the seasonal and annual field performance of a high-efficiency heat pump*/desuperheater water heater (HP/DS) system and compare them to those of a high-efficiency heat pump*/electric-resistance water heater (HP/$I^2R$) system.

## TEST FACILITIES AND CONDITIONS

These field tests were performed at the Tennessee Energy Conservation in Housing (TECH) facility near Knoxville, TN, pictured in figure 1. This facility has been used for a number of years to obtain field performance data on several systems, including conventional heat pumps (Baxter et al. 1982), the Annual Cycle Energy System (ACES) (Baxter 1981), and a number of solar systems (Bedinger et al. 1978 and 1981). The equipment for the tests reported in this paper was installed in TECH house III.

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*Rated COP of 3.11 at 47°F (8.3°C).

Although the house was unoccupied during the tests, a loads package simulating the effects of a family of four was installed. The house and equipment were monitored by a data acquisition system installed in TECH house II (Bledsoe and Miller 1981).

The heat pump used in these experiments was typical of a high efficiency, split system, air-to-air residential type. Its specifications are listed in table 1 and its ratings in table 2. Two thermostats controlled the heating process: an outdoor thermostat set at 32°F (0°C) and an indoor, two-stage thermostat with a first-stage heating set point of 70°F (21°C). At house temperatures below 70°F (21°C), the heat pump was operated. If the house temperature fell more than 2°F (1°C) below the first-stage set point, the second stage of the indoor thermostat was activated, energizing the supplemental electric-resistance heaters if the outdoor temperature was below 32°F (0°C). For cooling operation, the indoor thermostat was set at 78°F (26°C). A water-heating load was imposed by drawing 70 gal (265 L) of water from the hot water tank each day.

For the HP/DS system, the desuperheater water pump drew water from the bottom of a standard 82-gal (260-L) water tank and injected water into the top of the tank whenever there was a call for space heating or space cooling—it was not controlled by the tank water temperature. A temperature-controlled valve set at 120°F (49°C) regulated the temperature of the water leaving the desuperheater. Supplemental electric resistance heat for water heating consisted of the upper electric element in the water tank and was controlled by a thermostat set at 120°F (49°C). For the HP/$I^2R$ system, both top and bottom elements were employed and each was set at 120°F (49°C).

Both the heat pump and the desuperheater unit, shown in figure 2, were installed by an authorized local dealer. The relationship of the temperature-dependent steady-state heating capacity of the heat pump, both with and without the desuperheater, to measured house loads is shown in figure 3. This figure shows that the balance point for the heat pump alone is about 27°F (−3°C), while that of the heat pump with desuperheater is about 30°F (−1°C).

The experiment ran from November 1981 through October 1982. Each system was operational in every other month; i.e., the HP/DS was used in November, the HP/$I^2R$ system was used in December, etc.

## TEST RESULTS

Steady-state performance test results are given in table 3. These results show that the overall heat pump efficiency in the heating mode is not changed by adding a desuperheater, although the space-heating capacity is reduced by about 20%. Cooling mode efficiency is enhanced by about 35%, due to the desuperheater's recovery of normally rejected heat.

From November 1981 through October 1982, TECH house III had its space-conditioning and water-heating energy needs met by an HP/DS system and an HP/$I^2R$ system on an alternating monthly cycle. Monthly performance data for the two systems are listed in tables 4 and 5. Two outdoor dry-bulb temperatures are listed: (1) the average temperature for the month and (2) the average temperature during the hours of heat pump operation only. The loads are those met by the space-conditioning and water-heating system. Electric energy usages are (1) the amount used by the electric-resistance water heater, (2) the amount used by the electric-resistance space heaters, and (3) the total energy used for space conditioning and water heating. COPs listed are based on total energy used.

## DISCUSSION

Average weekly COP for space conditioning and water heating, as a function of average temperature during heat pump operation, is plotted in figure 4. The same information, as a function of overall average outdoor temperature, is given in figure 5.

In the heating season, for average operating temperatures between 37°F and 44°F (3°C and 6.5°C) or overall average temperatures between 41°F and 52°F (5°C and 11°C), the HP/DS system had a slight advantage over the HP/$I^2R$ system. At higher and lower temperatures, however, both systems had nearly the same efficiency. At lower temperatures—below the house balance point—the HP/DS system used more supplemental electric-resistance space heat to make up for the heat diverted to the domestic hot water by the desuperheater. In other words, the resistance heat saved in water heating

was counterbalanced by the additional resistance heat required for space heating. This is clearly illustrated in table 6, which shows that, for two weeks with nearly identical heating loads, the HP/DS system used 109 kWh of supplemental heat while the HP/$I^2R$ system used only 25 kWh. For operating temperatures above 44°F (6.5°C) (or overall average temperatures above 52°F (11°C)), little space heating was required by the test house; thus most of the water heating was done by electric-resistance heating elements and both system COPs approached 1.0.

The HP/DS system had about a 3% better HSPF than that of the HP/$I^2R$ system. This slight advantage in efficiency is partially due to the fact that the HP/DS system delivered less water heating in winter than did the HP/$I^2R$ system. For an HP/DS system, installed as was done for this test, only the upper resistance-heating element in the water tank is energized. Thus the water in the lower portion of the tank may not always be heated to as high a temperature as it would be by an HP/$I^2R$ system and standby losses would thereby be lower. Additionally, the supply of available hot water would be less. Another reason for the HP/DS system's slightly better heating-season performance is that for the temperature ranges mentioned above, the desuperheater delivered its share of the water-heating load at a COP greater than 1.0.

During the cooling season, HP/DS system performance is 30 to 45% better than that of the HP/$I^2R$ system, except for weeks when the average temperature was low (little or no cooling load).

The desuperheater supplied 63% of the water-heating energy delivered over the six-month test period. Monthly values ranged from 100% in July to 34% in March.

Tables 4 and 5 show that the HP/DS system has about a 30% higher cooling-season performance factor (CSPF) than the HP/$I^2R$ system for similar total loads. Table 7 contains measured and calculated ACOPs for both systems. Calculations were done by the monthly ACES design (MAD) computer code (Ballou et al. 1981) using average weather conditions. Test-year weather was similar to the long-term average for Knoxville, having 8% more heating degree-days and 7% fewer cooling degree-days than the average year. The measured ACOPs (for one-half year) show a 14% improvement in performance through the use of a desuperheater as compared to a predicted improvement of 21% for a full year under average weather conditions. As indicated in table 7, the calculated winter performance of the HP/DS system was higher than measured, while the calculated cooling performance was lower than measured for both systems. Overall, the program slightly overpredicts HP/DS annual performance while slightly underpredicting the HP/$I^2R$ annual performance. In both cases, the differences are less than 5%, which is within the experimental accuracy of the data system (Baxter 1981).

Cost comparisons for the two systems based on performance predictions generated by the MAD program (Ballou et al. 1981) and on measured half-year performance, are given in table 8. The system's installed cost includes the equipment cost and all necessary installation costs. First-year energy costs are calculated using electricity rates current as of January 1981 for Knoxville (3.78 ¢/kWh). Life-cycle costs are the sum of installed costs and 25 years of energy and maintenance costs discounted at a 3.5% real interest rate (after tax and inflation) (Nephew and Abbatiello 1982).

These calculations show that the additional cost of installing a desuperheater water heater with an air-to-air heat pump system can be paid for with energy savings in seven to nine years. In addition, the 25-year life-cycle cost of the HP/DS system is slightly less than that of an HP/$I^2R$ system. Both these observations indicate that the relatively modest additional cost required by the HP/DS system could be a good investment for the consumer.

## CONCLUSIONS

From the foregoing test results and analysis, the following conclusions can be reached:

1. In the Knoxville, TN, area (1835 C-days heating, 845 C-days cooling), adding a desuperheater to a high-efficiency air-to-air heat pump improved the annual performance by about 14%, almost exclusively as a result of recovering normally rejected energy for domestic water heating in the cooling season.

2. Reductions in the amount of electric-resistance water heating in winter by the desuperheater are counterbalanced by increased use of electric-resistance heat for space heating for the system tested. Using a larger capacity heat pump could conceivably eliminate this effect but may result in a system that is oversized for cooling.

3. Cooling-season performance improvement with the desuperheater averaged about 35% over that of the heat pump with electric-resistance water heating.

4. Based on cost comparisons using 1981 equipment and electricity costs, a desuperheater would pay for itself in electricity savings in about seven to nine years in the Knoxville area.

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## ACKNOWLEDGMENTS


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**TABLE 1**

**Description of the Heat Pump System Tested at TECH Complex during 1981–82**

| | |
|---|---|
| Indoor unit coil | 3-row, plate-fin, "A" type |
| Blower capacity, cfm ($m^3/s$) | 1200 (0.57) |
| Outdoor unit coil | 1-row, spine-fin |
| ARI* cooling/heating capacity, Btu/h (kW) | 33,500/34,000 (9.8/10.0) |
| Refrigerant charge, lb (kg) | 10.0 (4.6) |
| Outdoor thermostat setting, °F (°C) | 32 (0) |
| Strip heaters, Btu/h (kW) | 30,710 (9.0) |
| Indoor thermostat settings, °F (°C) | |
| Heating | 70 (21.1) |
| Cooling | 78 (25.6) |

*Air Conditioning and Refrigeration Institute.

**TABLE 2**

**ARI Ratings and Steady-State Performance Test Results for the Heat Pump Installed in TECH House III**

| | On-site Test 1 | On-site Test 2 | ARI |
|---|---|---|---|
| HEATING MODE | | | |
| Outdoor temperature, °F (°C) | 49.5 (9.7) | 46.4 (8.0) | 47.0 (8.3) |
| Heat output, Btu/h (W) | 37,164 (10,892) | 35,239 (10,328) | 34,000 (9,965) |
| Outdoor unit energy use, Btu/h (W) | 10,738 (3,147) | 10,529 (3,086) | - |
| Indoor unit energy use, Btu/h (W) | 1,347 (395) | 1,327 (389) | - |
| Total energy use, Btu/h (W) | 12,085 (3,542) | 11,856 (3,475) | 10,918 (3,200) |
| Space heating COP | 3.08 | 2.97 | 3.11 |
| COOLING MODE | | | |
| Outdoor temperature, °F (°C) | 95.4 (35.2) | 79.7 (26.5) | 95.0 (35.0) |
| Cooling capacity, Btu/h (W) | 32,000 (9,379) | 35,567 (10,424) | 33,500 (9,818) |
| Outdoor unit energy use, Btu/h (W) | 11,638 (3,411) | 10,362 (3,037) | - |
| Indoor unit energy use, Btu/h (W) | 1,382 (405) | 1,378 (404) | - |
| Total energy use, Btu/h (W) | 13,020 (3,816) | 11,740 (3,441) | 13,307 (3,900) |
| Space cooling COP | 2.45 | 3.03 | 2.52 |

**TABLE 3**

**Heat Pump Steady-State Performance with and without Desuperheater**

| | Heat Pump Only | Heat Pump with Desuperheater |
|---|---|---|
| HEATING MODE | | |
| Outdoor temperature, °F (°C) | 49.5 (9.7) | 48.6 (9.2) |
| Capacity, Btu/h (Wh/h) | | |
| Space heating | 37,164 (10,892) | 28,507 (8,355) |
| Water heating | - | 7203 (2,111) |
| Energy Consumption, Btu/h (Wh/h) | 12,085 (3,542) | 11,556 (3,387) |
| COP | 3.08 | 3.09 |
| COOLING MODE | | |
| Outdoor temperature, °F (°C) | 95.4 (35.2) | 91.0 (32.8) |
| Capacity, Btu/h (Wh/h) | | |
| Space cooling | 32,000 (9,379) | 32,578 (9,548) |
| Water heating | - | 7581 (2,222) |
| Energy Consumption, Btu/h (Wh/h) | 13,020 (3,816) | 12,317 (3,610) |
| COP | 2.45 | 3.26 |

**TABLE 4**

**Loads and Performance Data for HP/DS System: November 1981 through September 1982**

| Period | Avg. Outdoor Temp. During Month °F | Avg. Outdoor Temp. During Month °C | Avg. Outdoor Temp. During Heat Pump Operation °F | Avg. Outdoor Temp. During Heat Pump Operation °C | Loads: Space Heating MBtu | Loads: Space Heating GJ | Loads: Space Cooling MBtu | Loads: Space Cooling GJ | Loads: Water Heating MBtu | Loads: Water Heating GJ | Electricity Used: Electric Water Heater* MBtu | Electricity Used: Electric Water Heater* kWh | Electricity Used: Electric Space Heater* MBtu | Electricity Used: Electric Space Heater* kWh | Electricity Used: Total MBtu | Electricity Used: Total kWh | COP |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 10/30-11/30/81 | 45.7 | 7.6 | 39.4 | 4.1 | 3.97 | 4.19 | 0.00 | 0.00 | 1.30 | 1.37 | 0.68 | 198 | 0.24 | 70 | 3.00 | 878 | 1.76 |
| 12/31/81-1/29/82 | 30.9 | −0.6 | 26.2 | −3.[illegible] | 10.52 | 11.10 | 0.00 | 0.00 | 1.51 | 1.59 | 0.29 | 84 | 2.51 | 735 | 7.31 | 2,143 | 1.64 |
| 3/1-4/2/82 | 53.1 | 11.7 | 43.2 | 6.2 | 3.12 | 3.29 | 0.00 | 0.00 | 1.39 | 1.47 | 0.92 | 270 | 0.08 | 24 | 2.70 | 791 | 1.71 |
| Heating season: | 43.5 | 6.4 | 32.5 | 0.3 | 17.61 | 18.58 | 0.00 | 0.00 | 4.20 | 4.43 | 1.88 | 552 | 2.83 | 829 | 13.01 | 3,812 | 1.68 |
| 5/3-6/1/82 | 69.4 | 20.8 | 78.1 | 25.6 | 0.00 | 0.00 | 3.12 | 3.29 | 1.14 | 1.20 | 0.48 | 142 | 0 | 0 | 2.02 | 593 | 2.10 |
| 7/1-7/28/82 | 76.8 | 24.9 | 80.4 | 26.9 | 0.00 | 0.00 | 7.42 | 7.83 | 1.43 | 1.51 | 0.00 | 1 | 0 | 0 | 3.05 | 894 | 2.90 |
| 9/3-10/4/82 | 71.8 | 22.1 | 76.6 | 24.8 | 0.00 | 0.00 | 2.59 | 2.73 | 1.06 | 1.12 | 0.53 | 154 | 0 | 0 | 1.65 | 483 | 2.00 |
| Cooling season: | 72.5 | 22.5 | 79.0 | 26.1 | 0.00 | 0.00 | 13.13 | 13.85 | 3.63 | 3.83 | 1.02 | 298 | 0 | 0 | 6.72 | 1,970 | 2.50 |
| Annual | | | | | 17.61 | 18.58 | 13.13 | 13.85 | 7.83 | 8.26 | 2.90 | 850 | 2.83 | 829 | 19.73 | 5,782 | 1.96 |

*Supplemental resistance heat.

**TABLE 5**

**Loads and Performance Data for HP/I²R System: December 1981 through October 1982**

| Period | Avg. Outdoor Temp. During Month °F | Avg. Outdoor Temp. During Month °C | Avg. Outdoor Temp. During Heat Pump Operation °F | Avg. Outdoor Temp. During Heat Pump Operation °C | Loads: Space Heating MBtu | Loads: Space Heating GJ | Loads: Space Cooling MBtu | Loads: Space Cooling GJ | Loads: Water Heating MBtu | Loads: Water Heating GJ | Electricity Used: Electric Water Heater[a] MBtu | Electricity Used: Electric Water Heater[a] kWh | Electricity Used: Electric Space Heater[a] MBtu | Electricity Used: Electric Space Heater[a] kWh | Electricity Used: Total MBtu | Electricity Used: Total kWh | COP |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 11/30-12/31/81 | 35.6 | 2.0 | 32.2 | 0.1 | 8.61 | 9.08 | 0.00 | 0.00 | 1.69 | 1.78 | 1.69 | 494 | 0.55 | 160 | 6.17 | 1,807 | 1.67 |
| 1/29-3/1/82 | 42.3 | 5.7 | 37.0 | 2.8 | 6.83 | 7.20 | 0.00 | 0.00 | 1.69 | 1.78 | 1.69 | 493 | 0.30 | 89 | 5.09 | 1,492 | 1.67 |
| 4/2-5/3/82 | 55.2 | 12.9 | 44.6 | 7.0 | 1.97 | 2.08 | 0.00 | 0.00 | 1.48 | 1.56 | 1.48 | 433 | 0.02 | 6 | 2.50 | 734 | 1.38 |
| Oct. 15-31 | 51.4 | 10.8 | 44.6 | 7.0 | 1.22 | 1.29 | 0.00 | 0.00 | 0.73 | 0.77 | 0.73 | 213 | 0.01 | 2 | 1.15 | 337 | 1.70 |
| Heating season: | 45.3 | 7.4 | 35.4 | 1.9 | 18.63 | 19.65 | 0.00 | 0.00 | 5.58 | 5.89 | 5.58 | 1,633 | 0.88 | 257 | 14.91 | 4,370 | 1.63 |
| 6/1-7/1/82 | 72.0 | 22.2 | 77.7 | 25.4 | 0.00 | 0.00 | 5.15 | 5.43 | 1.14 | 1.20 | 1.14 | 334 | 0.00 | 0.00 | 3.31 | 969 | 1.90 |
| 8/4-9/3/82 | 72.9 | 22.7 | 76.6 | 24.8 | 0.00 | 0.00 | 5.46 | 5.76 | 1.07 | 1.13 | 1.07 | 314 | 0.00 | 0.00 | 3.29 | 964 | 1.99 |
| Oct. 4-15 | 66.7 | 19.3 | 75.7 | 24.3 | 0.00 | 0.00 | 0.73 | 0.77 | 0.45 | 0.47 | 0.45 | 130 | 0.00 | 0.00 | 0.76 | 222 | 1.57 |
| Cooling season: | 71.2 | 21.8 | 77.0 | 25.0 | 0.00 | 0.00 | 11.34 | 11.96 | 2.65 | 2.80 | 2.65 | 778 | 0.00 | 0.00 | 7.35 | 2,155 | 1.90 |
| Annual: | | | | | 18.63 | 19.65 | 11.34 | 11.96 | 8.24 | 8.69 | 8.24 | 2,411 | 0.88 | 257 | 22.26 | 6,525 | 1.72 |

[a]Supplemental resistance heat.

**TABLE 6**

**Loads and Performance Data for Weeks of Dec. 14, 1981, and Jan. 4, 1982**

| Week beginning | System | Avg. temp. during heat pump operation | | Space-Heating Load | | Water-Heating Load | | | | Supplemental Space Heat | | Total Energy Used | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | D.S.[a] | | $I^2R$[b] | | | | | |
| | | °F | °C | MBtu | kWh | MBtu | kWh | MBtu | kWh | MBtu | kWh | MBtu | kWh |
| Dec. 14, 1981 | HP/$I^2R$ | 26.6 | −3.0 | 2.25 | 659 | 0 | 0 | 0.34 | 113 | 0.09 | 25 | 1.63 | 479 |
| Jan. 4, 1982 | HP/DS | 26.9 | −2.9 | 2.21 | 647 | 0.25 | 74 | 0.09 | 27 | 0.37 | 109 | 1.54 | 450 |

[a]By desuperheater.

[b]By electric resistance.

**TABLE 7**

**Measured versus Calculated Seasonal and Annual Performance Factors**

| | HP/$I^2R$ | | HP/DS | |
|---|---|---|---|---|
| | Calculated[a] | Measured[b] | Calculated[a] | Measured[b] |
| Seasonal Performance Factor: | | | | |
| Heating | 1.63 | 1.63 | 1.86 | 1.68 |
| Cooling | 1.76 | 1.90 | 2.41 | 2.50 |
| Annual Performance Factor | 1.69 | 1.72 | 2.04 | 1.96 |

[a]Calculated by MAD[7] program for average weather year in Knoxville.

[b]Measured over half-seasons during 1981-1982 test year in Knoxville.

**TABLE 8**

**System Cost Comparisons**

| System | | Installed Cost (1981 $) | First-Year Energy Cost (1981 $) | Life-Cycle Cost (1981 $) |
|---|---|---|---|---|
| HP/$I^2R$[a] | | 3710 | 460 | 14,415 |
| HP/DS | 1)[a] | 4270 | 380 | 13,810 |
| | 2)[b] | 4270 | 395 | 14,090 |

[a]Based on MAD program performance prediction.

[b]Based on measured performance.

LIST OF FIGURES AT-84-04[illegible]

Fig. 1. The TECH complex on the University of Tennessee Campus. From left to right, TECH houses III, II, and I. (DOE photo 80-648)

Fig. 2. Desuperheater water heater unit installed in TECH house III. (ORNL photo 7985-81)

Fig. 3. TECH house III heating load and calculated steady-state heating capacity of the house heat pump with and without desuperheater as a function of outdoor dry bulb temperature.

Fig. 4. Weekly average COPs for space conditioning and water heating vs. weekly average outdoor temperature during heat pump operation.

Fig. 5. Weekly average COPs for space conditioning and water heating vs. weekly average outdoor temperature.

ORNL PHOTO 1797-81 Top

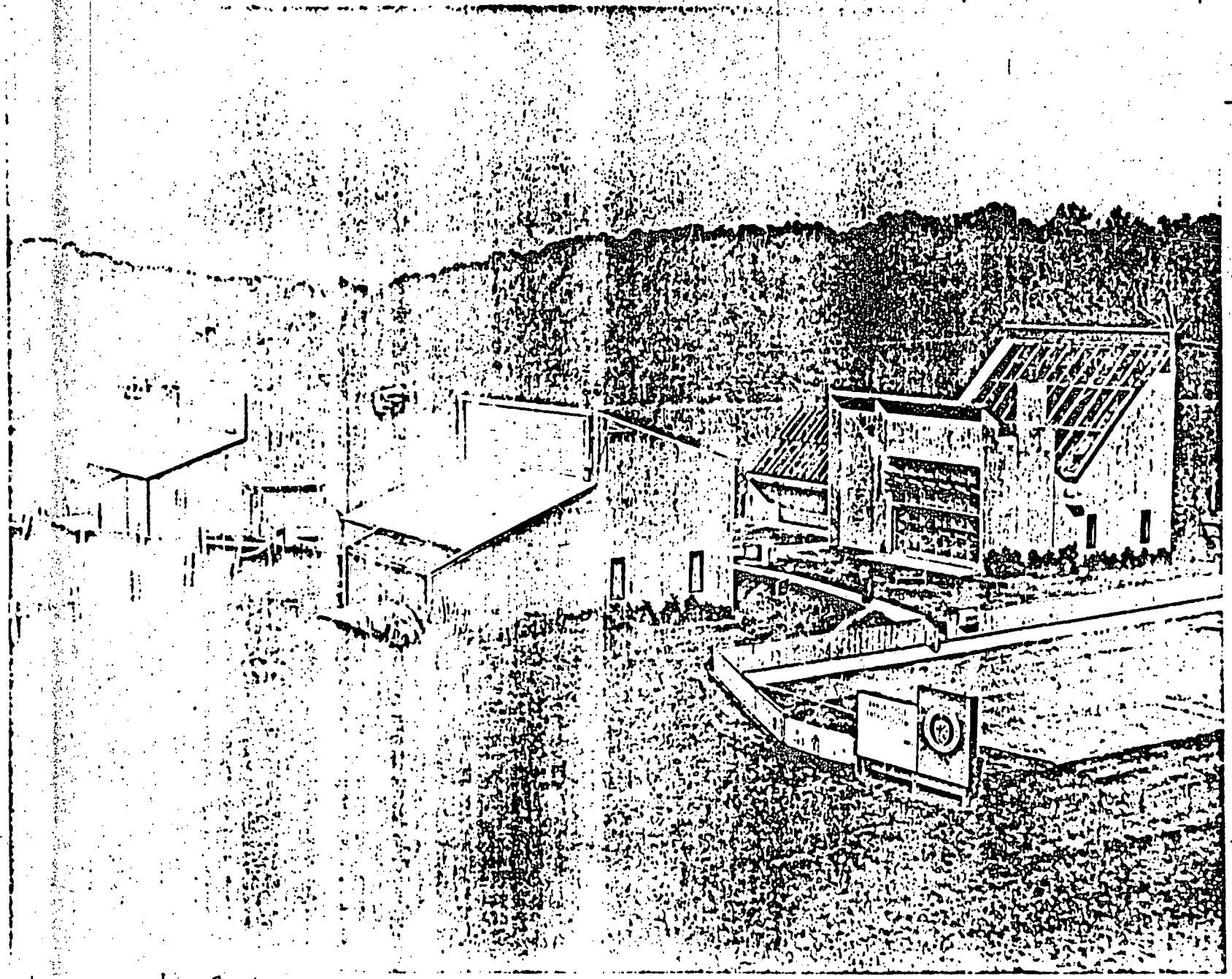


Fig. 2. 1

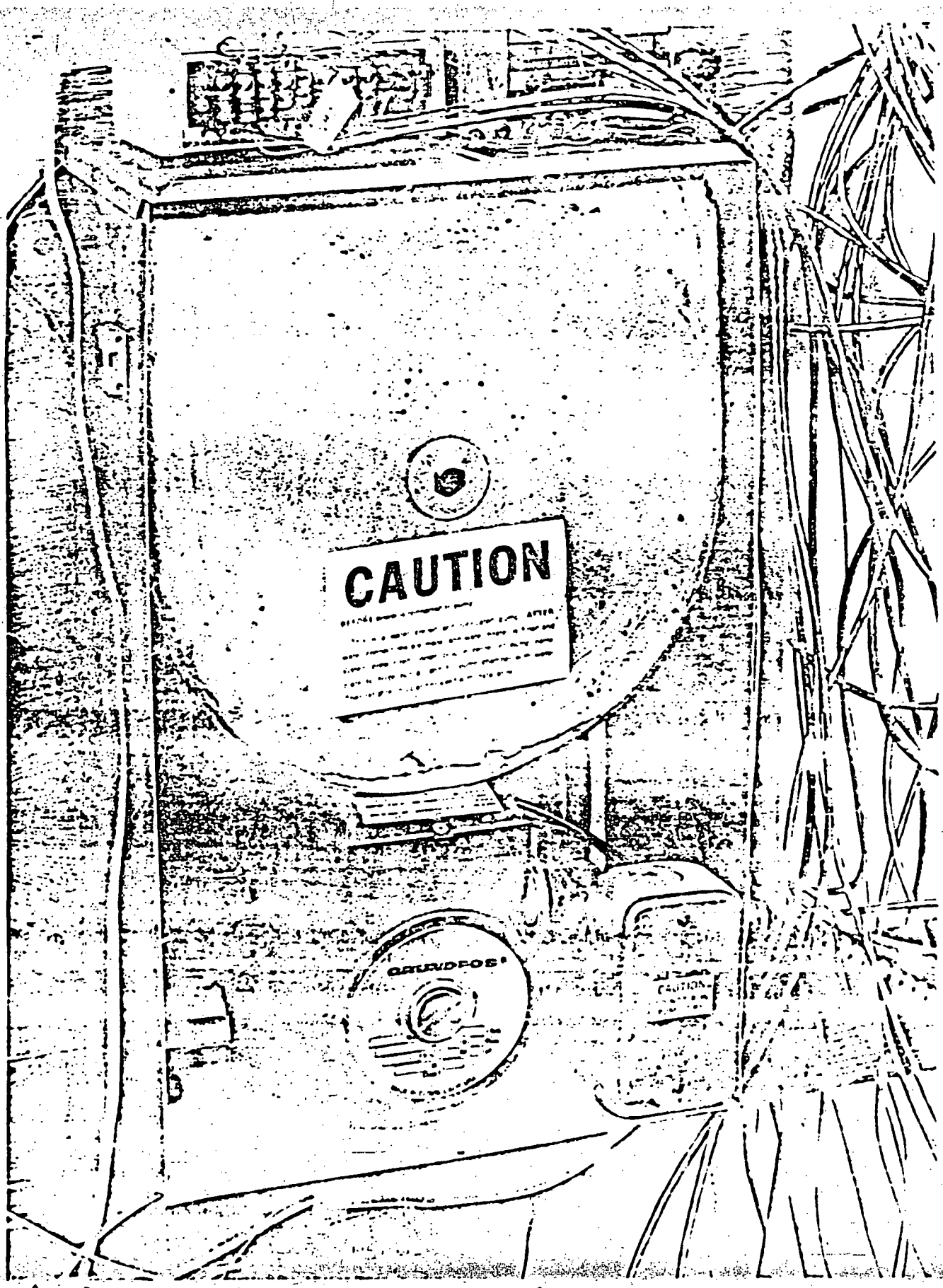
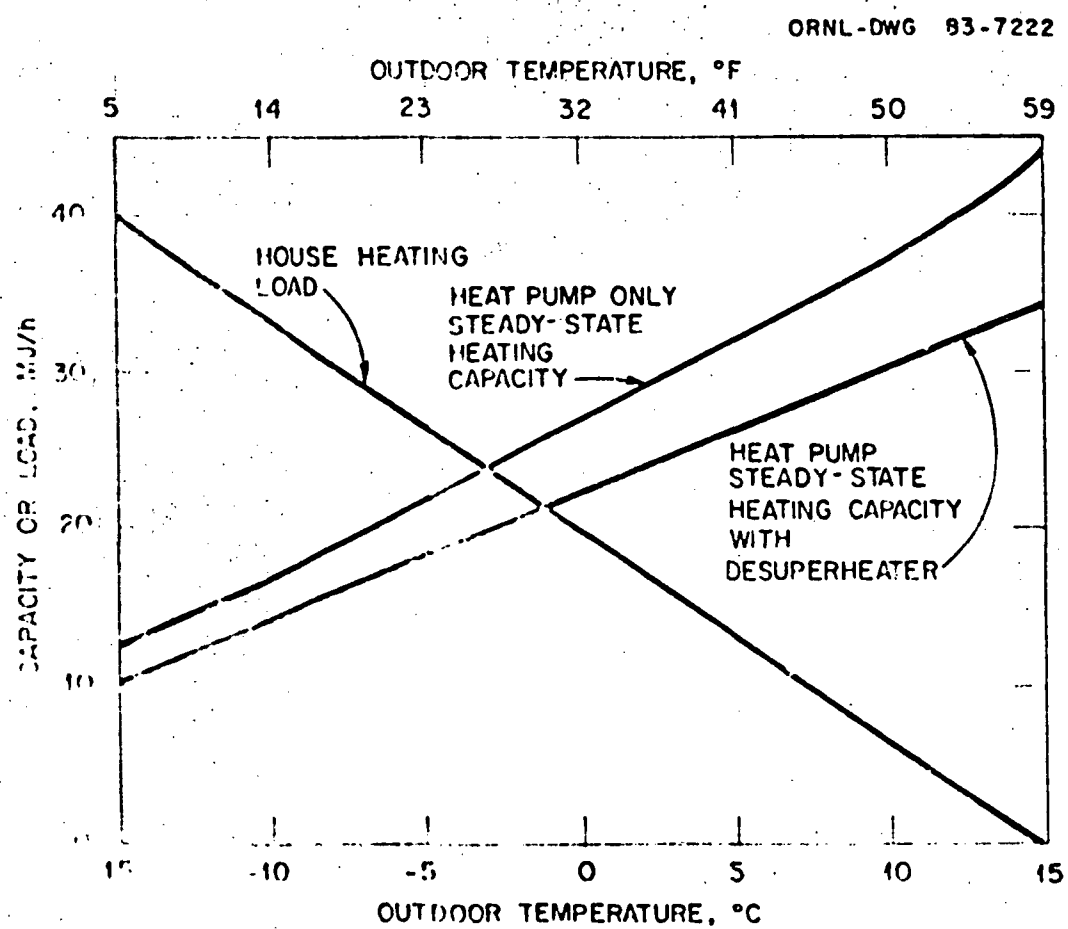



Fig. 2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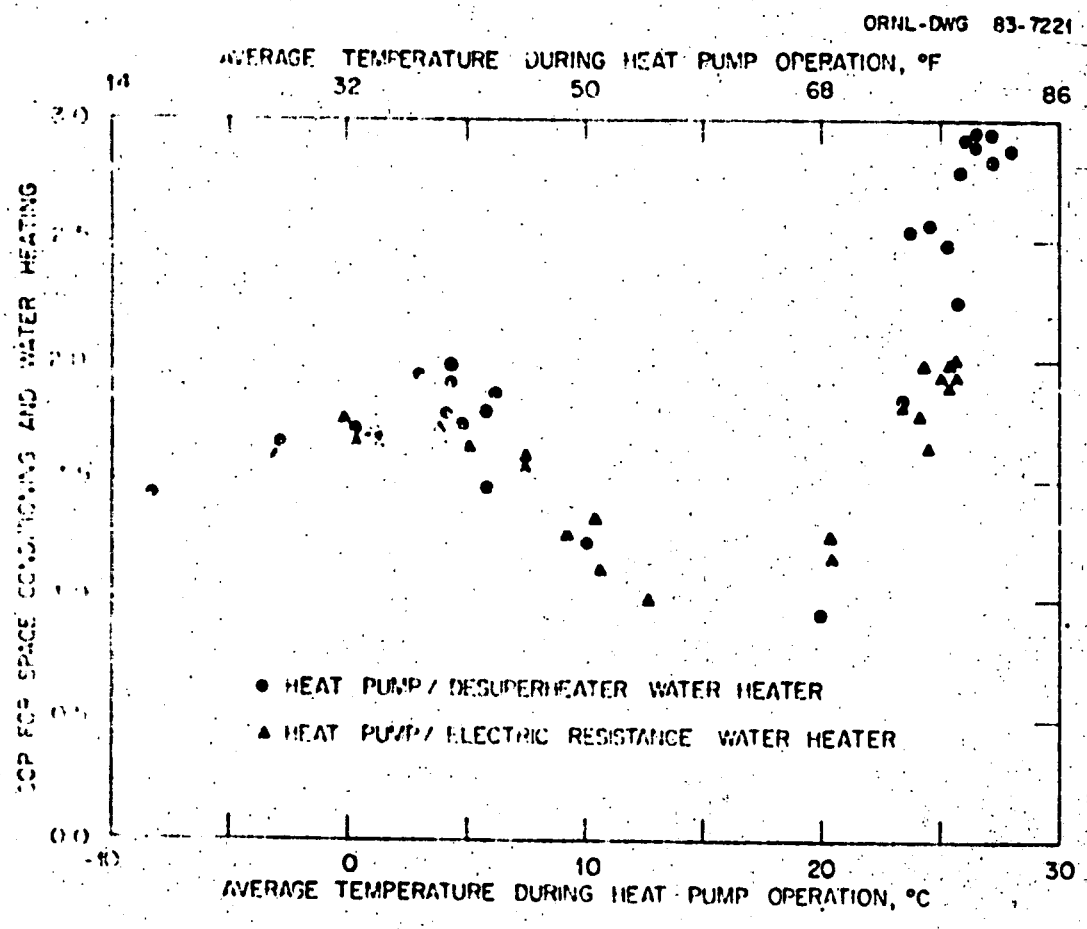
OUTDOOR TEMPERATURE, °F
5
14
23
32
41
50
59
40
30
20
10
CAPACITY OR LOAD, MJ/h
HOUSE HEATING LOAD
HEAT PUMP ONLY STEADY-STATE HEATING CAPACITY
HEAT PUMP STEADY-STATE HEATING CAPACITY WITH DESUPERHEATER
-10
-5
0
5
10
15
OUTDOOR TEMPERATURE, °C

Fig 3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AVERAGE TEMPERATURE DURING HEAT PUMP OPERATION, °F
14
32
50
68
86
COP FOR SPACE CONDITIONING AND WATER HEATING
HEAT PUMP / DESUPERHEATER WATER HEATER
HEAT PUMP / ELECTRIC RESISTANCE WATER HEATER
-10
0
10
20
30
AVERAGE TEMPERATURE DURING HEAT PUMP OPERATION, °C

Fig. 4

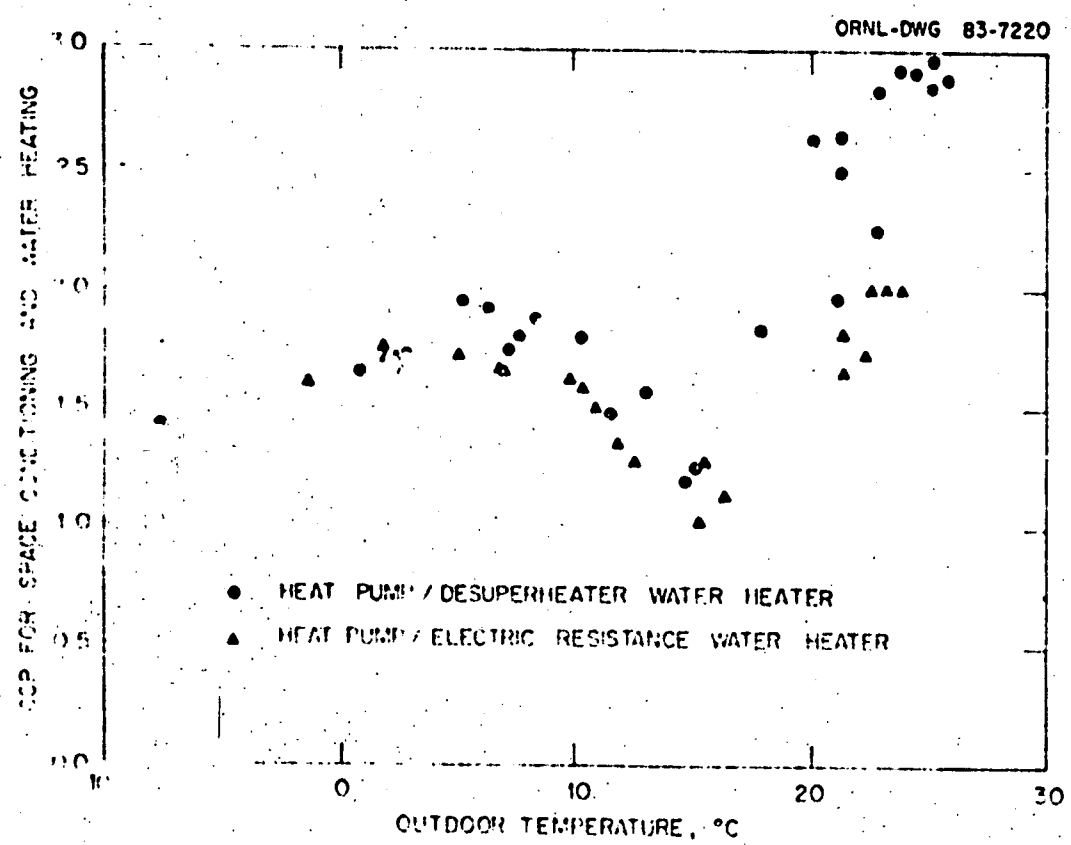
ORNL-DWG 83-7220
HEAT PUMP / DESUPERHEATER WATER HEATER
HEAT PUMP / ELECTRIC RESISTANCE WATER HEATER
0
10
20
30
OUTDOOR TEMPERATURE, °C

Fig 5

END

DATE FILMED

03/07/84