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Capacity and Overload Tests of a 7 1/2 Ton Package Unit Air Conditioner

William Mulroy
David Ward

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
Gaithersburg, MD 20899

January 1988

Final Report

Stimulating America's Progress
1913-1988

Prepared for:

David Taylor Research Center
Annapolis, MD 21402
Code 2722

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U.S.DEPARTMENT OF COMMERCE, C. William Verity, *Secretary*
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

Abstract

The National Bureau of Standards performed capacity and overload tests of a 7 1/2 ton package unit air conditioner for the David Taylor Research Center. The unit capacity was measured to be 83,200 Btu/hr (6.9 tons), the Energy Efficiency Ratio (EER) to be 8.58 Btu/Whr and the cooling effect ratio to be 62.7% at the requested capacity test condition. The unit failed to operate at the requested overload capacity test condition. Several additional tests were performed to more completely describe the performance of the unit.

Key Words: Air conditioner; Air Conditioner Testing; Capacity Testing; Overload Testing.

Acknowledgement

The authors wish to acknowledge the contributions of David Aaron and Tony Stobb in installation and instrumentation of the tested air conditioner.

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1. Introduction

This report covers work done by the National Bureau of Standards (NBS) in response to a request by the David Taylor Research Center (DTRC) under DTRC Delivery Order Number N6153387F1973 dated April 9, 1987 to perform capacity and overload tests on an A.R.E. Manufacturing Company, Inc., 7 1/2 ton package unit air conditioner provided by DTRC. All tests were to be performed in accordance with ASHRAE Standard 37 [1].

The requested test conditions are listed in Table 1.

	Capacity Test	Overload Test
Evaporator inlet air (db/wb)	80°F/67°F	100°F/85°F
Condenser inlet water	95°F	100°F
Air flow @ 1/4" H_2O static @ unit exit	2250 cfm	

Table 1: Requested Test Conditions

For each test the cooling capacity power consumption and cooling effect ratio (sensible heat to total capacity ratio) were to be determined for comparability to the first article tests [3] of this unit.

Additional capacity tests were performed at evaporator inlet air conditions of 85°F dry bulb and 67°F wet bulb to more completely describe the performance of the unit.

2. Description of Test Specimen

The tested air conditioning unit, manufactured by A.R.E. Manufacturing Company Inc.,* Frederick, Maryland, 21701, was shipped directly to NBS from DTRC. Its nameplate data were:

Naval Sea Systems Command
Cont. No. N00-81C-K464
Year 1983
ARE Manufacturing Co., Inc.
Serial No. 83030857-920 Model NAC-920
Std. Com. Class No. 4120-00-933-3497
Navy CID No. 34043004
Nominal Capacity 90,000 Btu
Voltage 440 Phase 3 Cycle 60

The unit was not equipped with a running time meter. No history of previous use was provided.

The unit had a water cooled condenser.

As received at NBS, the unit was fitted with a free air discharge plenum attached to the top of the unit. At the direction of DTRC this was removed and replaced with a duct instrumented according to ASHRAE Standard 37[1].

A technical manual [2] prepared by the manufacturer was provided for guidance in setup and operation. Later a copy of the first article

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inspection report [3] was made available for comparison to NBS results.

3. Test Set Up and Instrumentation

The unit was tested in the environmental chambers of the NBS air conditioner and heat pump laboratory. This facility consists of back-to-back environmental chambers which are normally controlled at different indoor and outdoor conditions for testing air-to-air units. This single package, water-cooled unit was installed in the "outdoor" chamber and a duct was run to the NBS air flow measurement tunnel in the "indoor" chamber. The doors connecting the two chambers were opened and the environmental systems for both chambers were operated to provide the desired return air conditions to the test unit.

The unit is shown set up for testing in Figures 1 and 2. The NBS air flow measurement tunnel is shown in Figure 3.

A 14 inch by 12 inch by 41 inch high plenum was mounted directly on top of the unit. Static pressure taps were located on this plenum 26 inches from the blower outlet. After this plenum, the ducting was reduced to 10 inch diameter round duct for an additional height of 20 inches followed by an elbow and an additional 26 inches of 10 inch round duct at which location the downstream thermocouple grid was located. Additional 14 inch round duct was used to continue to the air flow measurement tunnel. The ducting was insulated with 2 inch fiberglass batt between the unit discharge and the downstream thermocouple grid.

The principal method used to measure the unit sensible and latent cooling capacities was the air enthalpy method, indoor side [1]. The air

Figure 1:

Unit as Tested

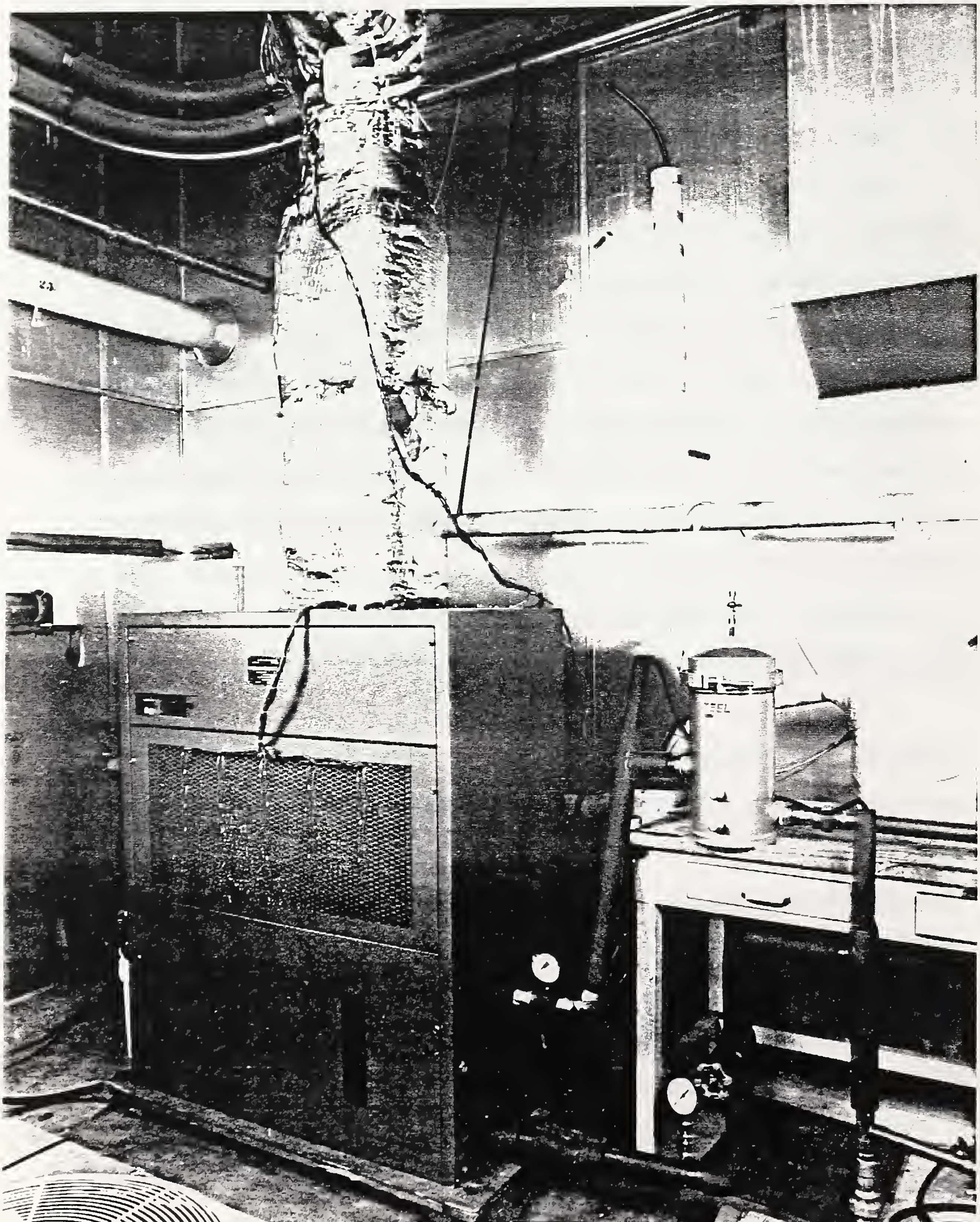


Figure 2:
Unit as Tested

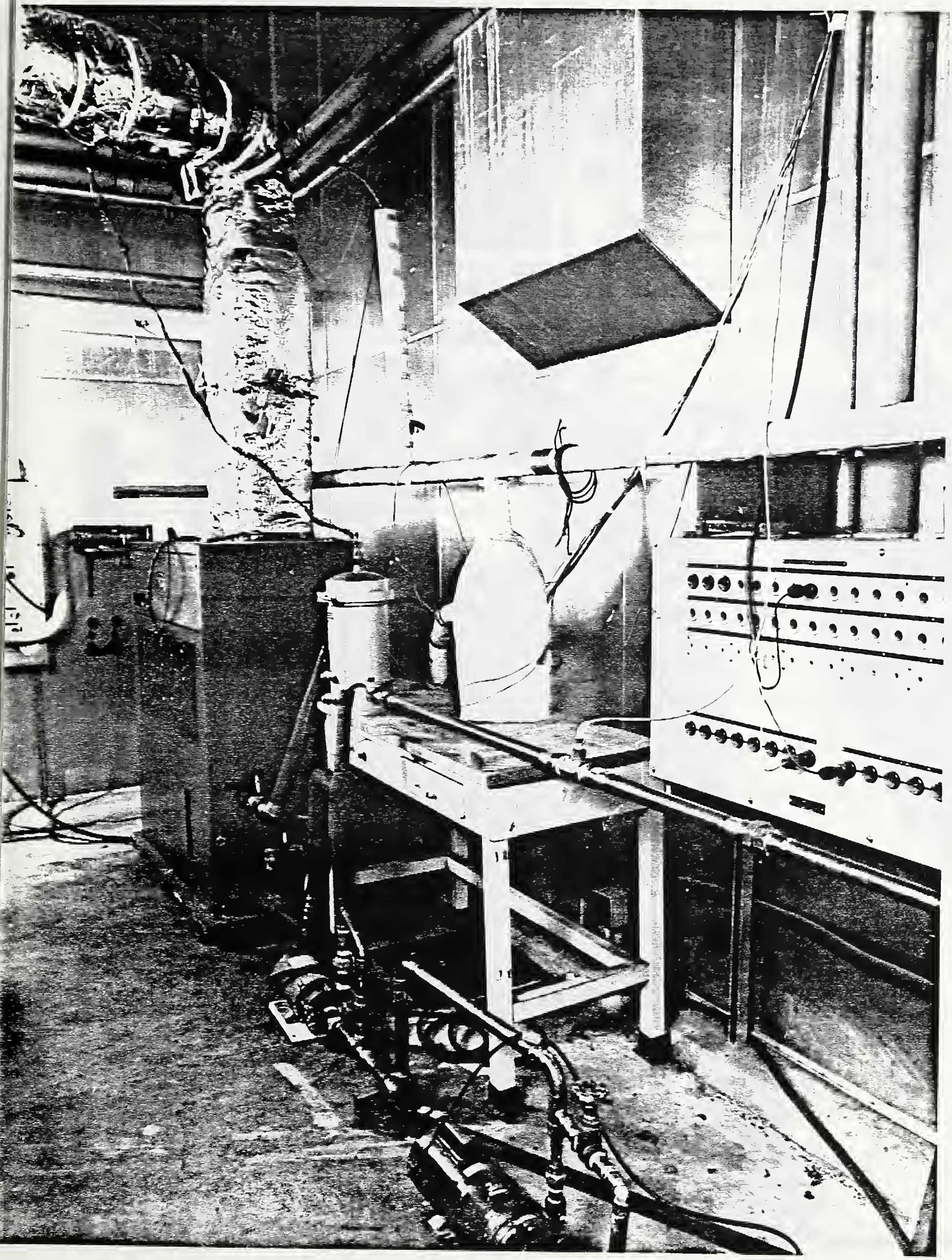
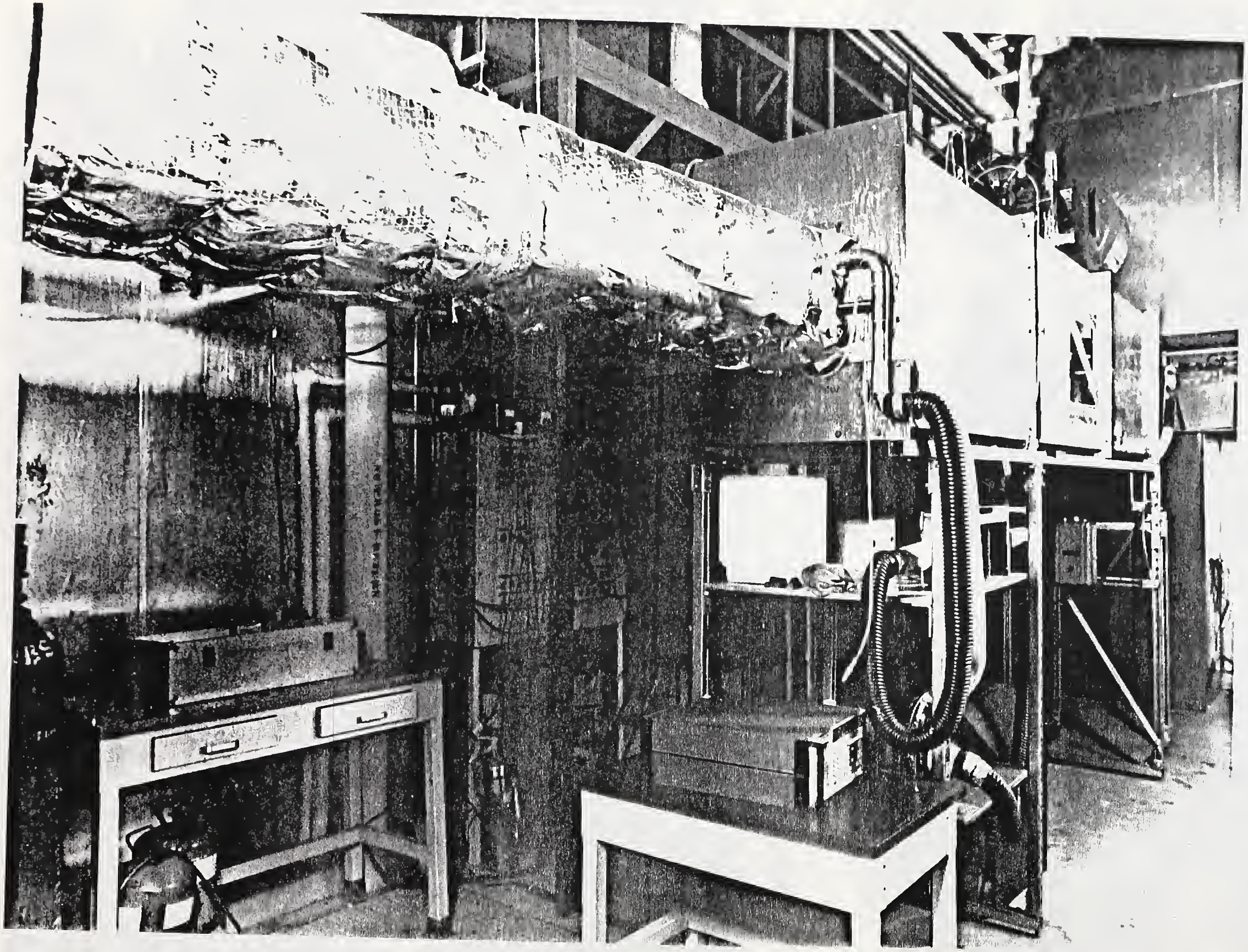


Figure 3:
NBS Air Flow Measurement Tunnel



flow rate was measured in a previously constructed tunnel (Figure 3) consisting of a receiving chamber and a discharge chamber separated by a partition containing a 10 inch nozzle. An exhaust fan was attached to the duct leaving the discharge chamber so that the static pressure of the air leaving the indoor section of the air conditioner could be adjusted. Achieving the required 2250 cfm at 1/4 inch static pressure for this unit required adjustment of the damper on this exhaust fan at the outlet of the tunnel and also the variable speed pulley on the blower integral with the test unit.

Pressure transducers accurate to within 1% of the reading were used to measure the static pressure across the nozzle and at the unit discharge. A thermocouple was installed to determine the air temperature at the nozzle inlet to allow calculation of the air density at the nozzle.

A fifteen junction thermopile was used to measure the air temperature difference entering and leaving the unit. One side of the thermopile was evenly spaced across the unit inlet grill while the other was placed at the previously described location in the insulated discharge duct. Fifteen junction averaging thermocouple grids were also placed at these two locations to read the supply and return air temperatures.

The moisture content of the air entering and leaving the unit was measured via wet bulb and dry bulb temperature measurements taken with aspirated psychrometers fed from sampling rakes, one in front of the unit inlet grill and the other in front of the nozzle. As a check on unit entering air conditions, the dry bulb and dew point sensors of the environmental chamber control system were observed and found to be in close agreement with the aspirated psychrometer. As a check on latent capacity measurement condensate was collected and weighed.

Condenser cooling was provided by a pressurized loop consisting of a pump, filter, and flowmeter with cold water makeup and loop water dump to an open trap drain. Makeup water was controlled by an adjustable water pressure regulating valve to obtain the desired temperature for the entering water.

The outdoor water coil method [1] was used as the secondary test method. A fifteen junction thermopile installed in thermocouple wells constructed in accordance with ASHRAE Standard 41 [4] was used to measure the water temperature difference entering and leaving the unit. Single thermocouples were also installed in these wells to measure the entering and leaving water temperatures. The water flow rate was measured with a turbine flowmeter specifically calibrated for this project by the NBS Metrology Division.

Total unit power draw was measured with a 3-phase digital power analyzer uncertain to within 0.6%. Compressor and blower power were not measured separately.

Barometric pressure was measured with a transducer designed for that purpose located in the control room adjacent to the environmental chambers.

Additional pressure transducers and thermocouples were used to measure the compressor suction and discharge pressures and temperatures, crankcase temperature and the compressor compartment temperature.

Data were collected by an automatic data acquisition system at 5 minute intervals with the exception of unit power and condensate weight which were manually recorded. All tests data periods were a minimum of 40 minutes except where otherwise noted.

4. <u>Capacity Tests</u>

Capacity test data and results are summarized in Table 2. All tests met the ASHRAE Standard 37 [1] requirment of agreement between the capacity measured by the enthalpy method, air side, and the secondary test method to be less than 6 per cent for test validation. The listed average total capacity is the average at the capacity measured by the two methods (air side and water side) and is considered to be the reportable capacity. The air side sensible and latent capacity are the measured values multiplied by the ratio of average total capacity to the air side total capacity as required by section 12.1.3 of [1] (see Appendix A). The condensate latent capacity is presented for comparison only. The cooling effect ratio (sensible cooling ratio) is calculated from the air side sensible capacity and the average total capacity. The COP (coefficient of performance) is calculated from the average total capacity and the total power. The listed air and water temperatures entering and leaving the unit are those measured respectively either by the averaging thermocouple grids or single thermocouples in wells, not the thermopiles used in the capacity calculations. Although fan power was not routinely measured separately for these tests, it was observed to be approximately 780W during unit setup and check out.

The first two capacity tests were conducted at entering air conditions of 85^{o}F dry bulb/67^{o}F wet bulb, 95^{o}F condenser entering water, and at 2250 cfm air flow rate. A water flow rate is not listed in Table 1. The unit technical manual [2] lists capacity test data of 96,000 Btu/hr total capacity, 63,000 Btu/hr sensible capacity, and 8 kW power draw at 19.5 gpm without specifying entering air temperature or air flow rate. After discussion with DTRC, it was decided to perform these capacity tests at a water flow rate of 19.5 gpm for comparability to the unit technical manual.

It is felt that the test of 8/25 is better than that of 9/4 because of the wet bulb temperature being slightly low on 9/4. The tests performed on 9/4/87 were witnessed by personnel from DTRC, the Naval Sea Systems Command, and the Ships Parts Control Center.

On 9/4/87, a copy of the first article inspection report [3] was made available to NBS for review. The first article capacity and overload tests were conducted by WEDJ, Inc., York, PA. The first article capacity tests were performed at inlet air conditions of 80^{o}F dry bulb and 67^{o}F wet bulb with a 2365 cfm air flow rate and a 22.5 gpm water flow rate. For comparison, NBS performed capacity tests on 9/15 at an 80^{o}F dry bulb entering air temperature instead of the 85^{o}F dry bulb previously used and at water flow rates of 19.5 gpm and 21 gpm (the highest flow rate possible with the pump that was being used).

WEDJ reported on air enthalpy method, indoor side, capacity of 98,068 Btu/hr and on outdoor water coil method capacity of 90,430 Btu/hr. They neglected to subtract fan power from the outdoor water coil method capacity which, with this correction, becomes 87,975 Btu/hr. The report states that the outdoor water coil method is "considered more nearly valid, and the most conservative" because of "heat radiation from compressor and condenser surfaces to room air and to the limitation of the degree of accuracy attainable in sensing and reading wet bulb temperatures".

The WEDJ water coil method capacity of 87,975 Btu/hr is 5.7 per cent higher than the average total capacity of 83,200 Btu/hr measured by NBS on 9/15. This is reasonable agreement between different laboratories, particularly if the NBS tested unit had seen substantial use. The NBS measured power of 9.70 kW is unaccountably different from the WEDJ measured power of 7.52 kW.

5. Overload Tests

Overload test data and results are summarized in Table 3. An explanation of some headings for this table is given in the first paragraph of Section 4.

The unit was found to be incapable of operation at the overload test condition of Table 1 as a result of tripping out of the high head pressure control. The head pressure control was observed to trip within the technical manual specified limit of 295 psig $\pm$ 3 psi.

On the first overload test day, 8/26, after the unit had tripped out, it was restarted and data taken at the maximum water flow rate possible with the NBS loop (21.1 gpm), with the inlet air conditions at 100^{O}F dry bulb/85^{O}F wet bulb and the entering water temperature reduced to 98^{O}F from the required 100^{O}F to allow continuous operation.

In discussion with DTRC, it was decided to examine the unit to see if either the water regulator valve was failing to open completely or if the condenser water passages were blocked. The water regulator valve was removed and found to be manually jacked open sufficiently to present its minimal resistance to flow. It was opened slightly further, reinstalled, and the water flow rate observed to be unchanged.

One of the four condenser heads was removed. The tubes appeared clean and free of any sediment or corrosion. In removing this head, two of the captive round head nuts used to hold it on became loose in their holes and had to be held with pliers to allow disassembley and reassembley. Since these nuts were inaccessible on the other heads and their coming loose with their inserted bolt partially loosened would result in neither being able to remove nor to tighten the bolt, the other heads were not removed to allow examination for physical blockage. The pressure drop through the unit was observed to be quite high. Inlet and outlet pressures of 66.5 psig and 23 psig (43.5 psid) were observed at 19.5 gpm and of 70 psig and 20 psig (50 psid) at 21 gpm.

These examinations were completed and the unit reassembled prior to repeat testing on 9/4. The tests on 9/4/87 were witnessed by personnel from DTRC, the Naval Sea Systems Command, and the Ships Parts Control Center. On this day a capacity test was first performed, followed by overload testing. For the recorded overload test data on this day the unit was first run for a full test period with the air inlet conditions unchanged from the capacity test and with the inlet water temperature raised to 100°F. The inlet air dry bulb temperature was then raised to 100°F. Finally the wet bulb temperature was slowly raised while data scans were taken at the normal 5 minute interval. The unit tripped out on high head (at a head pressure of 293 psig) when the wet bulb temperature reached 79.0°F.

The first article overload test performed by WEDJ, Inc., [3] was conducted with entering air conditions of 110°F dry bulb, 85°F wet bulb and with 100°F water entering the condenser. At this condition, correcting the calculated value for fan power as described in Section 4, the water coil method capacity was 84666 Btu/hr. This is 3309 Btu/hr less than measured for the capacity test. Conversely, the NBS measured overload capacity (on 8/26) was 33 per cent higher than the capacity test capacity (on 9/15). Normally unit total capacity would be expected to increase with increasing return air enthalpy (unless limited by some equipment feature such as a crankcase pressure limiting valve) resulting in substantially higher capacity at the overload condition as measured by NBS. This higher capacity results in increased heat rejection to the cooling water and, consequently, a higher likelihood that a unit would trip out on high pressure.

6. Conclusions

At the nominal capacity test conditions of 80°F dry bulb and 67°F wet bulb entering air conditions and a condenser entering water temperature of 95°F, the unit capacity was 83,200 Btu/hr, the power consumption was 9.70 kW, the EER was 8.58 Btu/Whr and the cooling effect ratio (sensible heat ratio) was 62.7%.

The unit failed to operate at the requested overload test conditions.

Test results at several conditions in addition to those requested are presented in Section 4, Capacity Tests, and Section 5, Overload Tests.

7. References

1. "Methods of Testing for Rating Unitary Air Conditioning and Heat Pump Equipment," ASHRAE Standard 37-69, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, Georgia, 1969.

2. "Technical Manual, Air Conditioning Unit, Water Cooled, 7 1/2 Ton Cooling Capacity, Model No. NAC-920, 440V-3ph-60H," NAVSEA-S9514-BA-MMO-010, A.R.E. Manufacturing Co., Inc., Frederick, Maryland, 1982.

3. "Test and Demonstration Report #R0001/F.A., First Article Inspection Report for Contract No. N00104-81-C-K464, Data Item A005," A.R.E. Manufacturing Co., Inc., Frederick, Maryland, 1981.

4. "Standard Measurements Guide: Section on Temperature Measurements," ASHRAE Standard 41.1-74, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, Georgia, 1974.

5. "1985 ASHRAE Fundamentals Handbook", Chapter 6, "Psychrometrics", American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, Georgia, 1974.

Date	8/25	9/4	9/15	9/15
Return Air Dry Bulb, ^{o}F	85.0	85.0	80.4	80.2
Return Air Wet Bulb, ^{o}F	67.0	66.3	68.0	67.7
Return Air Dew Point, ^{o}F	56.6	55.8	61.6	61.2
Supply Air Dry Bulb, ^{o}F	58.7	58.3	58.7	58.6
Air Flow Rate, cfm	2256	2240	2252	2259
Air Static at Unit Exit, In. H_2O	0.23	0.20	0.17	0.17
Inlet Water, ^{o}F	95.2	95.4	95.2	95.1
Leaving Water, ^{o}F	106.6	106.8	106.8	105.8
Water Flow Rate, gpm	19.54	19.50	19.50	20.99
Compressor Suction, psig	65.3	65.1	66.8	66.6
Superheat at Suction, ^{o}F	14.4	13.3	14.8	15.1
Compressor Discharge, psig	257	257	258	256
Compressor Discharge, ^{o}F	197.7	197.8	197.3	196.8
Condensate, lbs/hr	12.6	12.0	26.0	26.0
Air Side Total Capacity, Btu/hr	83740	82240	85160	84120
Water Side Total Capacity, Btu/hr	79900	80590	81230	82120
Total Capacity Difference, %	4.7	2.0	4.7	2.4
Avg. Total Capacity, Btu/hr	81820	81420	83200	83120
Air Side Sensible Capacity/ Btu/hr	63760	65380	52190	52960
Air Side Latent Capacity, Btu/hr	18060	16040	31010	30160
Condensate Latent Capacity, Btu/hr	13390	12770	27550	27580
Cooling Effect Ratio %	77.9	80.3	62.7	63.9
Total Power, kW	9.57	9.51	9.70	9.64
EER, Btu/Whr	8.55	8.56	8.58	8.62

Table 2: Summarized Capacity Test Conditions and Results

Date	8/26	9/4	9/4*	9/4*
Return Air Dry Bulb, ^{o}F	99.8	85.0	100.8	99.1
Return Air Wet Bulb, ^{o}F	85.4	66.7	74.6	79.0
Return Air Dew Point, ^{o}F	81.1	56.5	63.0	71.6
Supply Air Dry Bulb, ^{o}F	76.8	58.9	68.0	71.3
Air Flow Rate, cfm	2255	2244	2250	2250
Air Static at Unit Exit, In. H_2O	0.18	0.19	0.20	0.20
Inlet Water, ^{o}F	98.1	99.7	100.7	100.4
Leaving Water, ^{o}F	111.7	110.9	113.4	114.1
Water Flow Rate, gpm	21.11	19.51	19.51	19.52
Compressor Suction, psig	89.3	66.2	76.7	83.4
Superheat at Suction, ^{o}F	31.4	13.5	17.9	20.6
Compressor Discharge, psig	289	272	288	293
Compressor Discharge, ^{o}F	215.8	202.4	206.7	208.6
Condensate, lbs/hr	50.7	-	-	-
Air Side Total Capacity, Btu/hr	113470	81250	79840	101350
Water Side Total Capacity, Btu/hr	107010	-	-	96050
Total Capacity Difference, %	5.9	-	-	5.4
Avg. Total Capacity, Btu/hr	110240	-	-	98700
Air Side Sensible Capacity/ Btu/hr	54410	64510	79840	66580
Air Side Latent Capacity, Btu/hr	55830	16740	0	32120
Condensate Latent Capacity, Btu/hr	53810	-	-	-
Cooling Effect Ratio %	49.4	79.4	100.0	67.5
Total Power, kW	10.96		-	10.90
EER, Btu/Whr	10.06	-	-	9.06

*Single Reading. All other columns reflect a minimum of 40 minutes of data.

Table 3: Summarized Overload Test Conditions and Results

Appendix A - Sample Calculation

The following sample calculation is for the capacity test performed on September 15, 1987, at a condenser water flow rate of 19.5 gpm. The data used in this example are the average of 10 readings taken at 5 minute intervals during this test. The equations used are referenced to ASHRAE Standard 37-69 [1]. Psychrometric calculations are not given, but were performed using the recommended procedures for numerical calculation of moist air properties given in Chapter 6, Psychrometrics, of the 1985 ASHRAE Fundamentals Handbook [5].

Air Flow Rate, Q_{mi}:

$Q_{mi} = 1096\ C\ A_n\ (P_v\ v'_n)^{0.5}$ (7.4.1 of [1])

C = nozzle coefficient = 0.99 (7.3.2 of [1])

A_n = nozzle area = 0.5454 sq. ft. (for 10" diameter nozzle)

P_v = nozzle static pressure difference = 1.0889 in. H_2O

v'_n = specific volume of air at nozzle = 13.297 cu. ft. per lb. (calculated using a humidity ratio of 0.009011 calculated from supply air psychrometer chamber dry bulb and wet bulb readings of 81.19°F and 68.19°F respectively, a nozzle temperature of 59.84°F, and a barometric pressure of 29.649 in. Hg. by the methods recommended in [5]).

$Q_{mi} = 1096 * 0.99 * 0.5454 * (1.0889 * 13.297)^{0.5} = 2251.8$ cfm

Sensible Cooling Capacity, q_{si}:

$$q_{si} = -\frac{60\ Q_{mi}\ c_{pa}\ (t_{a1}-t_{a2})}{} \quad \text{(3.7.1 of [1])}$$

c_{pa} = specific heat of air = 0.2447 Btu per (lb)(^{o}F)(calculated from c_{pa} = 0.24 + 0.444 W_n (3.7.1 of [1]) using average of supply (0.009011) and return (0.011984) W_n values).

$t_{a1} - t_{a2}$ = supply minus return air temperature difference = 21.679^{o}F (calculated from 15 junction copper-constantan thermopile as 7.2938mv/(15 * 0.02243 mv/^{o}F); as a check, averaging thermocouple grids give 80.41^{o}F - 58.71^{o}F = 21.70^{o}F)

W_n = humidity ratio at nozzle = 0.009011 (see above comments on calculation of v'_n).

$$q_{si} = \frac{60*2251.8*0.2447*21.679}{13.297*(1+0.009011)} = 53{,}420 \quad \text{Btu/hr}$$

Latent Cooling Capacity, q_{lci}:

$$q_{lci} = \frac{63600 Q_{mi}\ (W_{i1}-W_{i2})}{v'_n\ (1 + W_n)} \qquad \text{(3.7.1 of [1])}$$

W_{i1} = return air humidity ratio = 0.0119842 (calculated from aspirated psychrometer readings of 81.19^{o}F dry bulb and 57.65^{o}F wet bulb using the recommended methods of [5]).

W_{i2} = supply air humidity ratio = W_n = 0.009011

$$q_{lci} = \frac{63600*2251.8*(0.0119842-0.009011)}{13.297*(1+0.009011)} = 31{,}740 \quad \text{Btu/hr}$$

Indoor Side Total Cooling Capacity, q_{tci}:

$$q_{tci} = q_{si} + q_{lci} = 53{,}420 + 31{,}740 = 85{,}160 \quad \text{Btu/hr}$$

Latent Cooling Capacity by Condensate Collection, q_{lcc}:

$q_{lcc} = 1060W_c$ (9.2.1 of [1])

W_c = indoor coil condensate = 25.97lbs/hr

$q_{lcc} = 1060 * 25.97 = 27,530$ Btu/hr

Outdoor Side Total Cooling Capacity, q_{tco}:

$q_{tco} = W_w\, c_{pw}\, (t_{w4}-t_{w3}) - 3.41\, E_t$ (6.4.1 of [1])

W_w = water flow rate = 9682.9 lbs/hr (calculated from a turbine meter measured flow rate of 19.509 gpm.and a water density of 61.88 lbs/ft^3 at 107°F.

c_{pw} = specific heat of water = 0.9975 Btu per (lb)($^{\circ}$F) (between 107°F (h=74.95 Btu/lb) and 95°F (h = 62.98 Btu/ lb)).

$t_{w4}-t_{w3}$ = leaving minus entering water temperatures = 11.835°F (calculated from 15 junction copper-constantan thermopile as 4.1133 mv/(15*0.02317 mv/$^{\circ}$F); as a check single thermocouples in the same wells give 106.78°F - 95.22°F = 11.56°F).

E_t = total electric power = 9700kW

q_{tco} = 9682.9*0.9975*11.835-3.41*9700 = 81,230 Btu/hr

Total Capacity:

$$\text{Total Capacity} = \frac{qtci + qtco}{2} \quad \text{(12.1.2 of [1])}$$

$$\text{Total Capacity} = \frac{85,160 + 81,240}{2} = 83,200 \quad \text{Btu/hr}$$

Capacity Difference (required to be less than 6% by 12.1.2 of [1]):

$$\text{Capacity Difference} = \frac{q_{tci} - q_{tco}}{\text{Total Capacity}} * 100$$

$$\text{Capacity Difference} = \frac{85{,}160 - 81{,}230}{83{,}200} * 100 = 4.7\%$$

Sensible Capacity:

$$\text{Sensible Capacity} = \frac{q_{si} * \text{Total Capacity}}{q_{tci}} \qquad \text{(12.1.3 of [1])}$$

$$\text{Sensible Capacity} = \frac{53{,}420 * 83{,}200}{85{,}160} = 52{,}190 \text{ Btu/hr}$$

Latent Capacity:

$$\text{Latent Capacity} = \frac{q_{lci} * \text{Total Capacity}}{q_{tci}} \qquad \text{(12.1.3 of [1])}$$

$$\text{Latent Capacity} = \frac{31{,}740 * 83{,}200}{85{,}160} = 31{,}010 \text{ Btu/hr}$$

Cooling Effect Ratio:

$$\text{Cooling Effect Ratio} = \frac{\text{Sensible Capacity}}{\text{Total Capacity}} * 100$$

$$\text{Cooling Effect Ratio} = \frac{52{,}190}{83{,}200} * 100 = 62.7\%$$

Energy Efficiency Ratio, EER:

$$\text{EER} = \frac{\text{Total Capacity}}{\text{Total Power}} = \frac{83{,}200}{9700} = 8.58 \text{ Btu/Whr}$$

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Ms. Deana Hammer
David Taylor Research Center
Code 2722
Annapolis, MD 21402

10. SUPPLEMENTARY NOTES

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

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

The National Bureau of Standards performed capacity and overload tests of a 7 1/2 ton package unit air conditioner for the David Taylor Research Center. The unit capacity was measured to be 83,200 btu/hr (6.9 tons), the EER to be 8.58 Btu/Whr and the cooling effect ratio to be 62.7% at the requested capacity test condition. The unit failed to operate at the requested overload capacity test condition. Several additional tests were performed to more completely describe the performance of the unit.

12. KEY WORDS (Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)

air conditioner; air conditioner testing; capacity testing; overload testing

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