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Environmental Tests of A 450 Cubic Foot Air Transportable Field Refrigerator

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
National Engineering Laboratory
Center for Building Technology
Building Equipment Division
Washington, DC 20234

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NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

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ABSTRACT

The National Bureau of Standards performed environmental tests on a newly developed 450 cubic foot air transportable field refrigerator for the U.S. Army Natick Research and Development Laboratories. The refrigerator, in assemblage with a supplied refrigeration unit, was found to be more than capable of pulling down to and holding an interior temperature of −17.8°C (0°F) when surrounded by an ambient temperature of 50.4°C (123°F). Additional tests indicated that the performance of this assemblage would exceed that required to meet any likely combinations of sensible and radiant (solar) heat gain.

Key words: field refrigerator; frozen food storage; military refrigerators; refrigerator assemblage testing; refrigerators; walk-in refrigerators

1. INTRODUCTION

This report covers work done by the National Bureau of Standards (NBS) in response to a request by the United States Army Natick Research and Development Laboratory under Natick Project Order 83-186, dated April 8, 1983, to perform environmental testing on a newly developed 450 cubic foot air transportable field refrigerator.

Environmental tests conducted included:

1. Reverse heating leakage without refrigeration unit installed in refrigerator. This test was a measurement of the refrigerator thermal transmittance (UA) by the internal heating method. This thermal transmittance value is used in calculating refrigeration unit capacity, refrigeration load at various ambient temperatures, and as a measure of the insulating quality of the refrigerator.

2. Capacity test at -17.8°C (0°F) when ambient temperature was 37.8°C (100°F). This test was performed to determine whether the refrigeration unit was performing at its rated capacity. The actual rating conditions for the refrigeration unit were -17.8°C (0°F) average air temperature entering the evaporator and 43.3°C (110°F) average air temperature entering the condenser.

3. Pull down test at a 51.7°C (125°F) ambient condition. For this test the unit was started and allowed to pull the refrigerator temperature down to -17.8°C (0°F) in a 51.7°C (125°F) ambient after an extended

thermal stabilization period in this ambient. This is an operational test that measures the unit's ability to operate with high load on the compressor as a result of high suction and discharge pressures.

4. Capacity test at -17.8°C (0°F) with a 51.7°C (125°F) ambient. This test was used with the rating point test (2) to estimate the unit performance, by interpolation, at other high ambient temperatures.

5. Determination of the ability of the assemblage to maintain -17.8°C (0°F) under solar conditions by calculating from the above test data. These calculations were performed based on the "sol-air" concept as presented in Chapter 26 of the American Society of Heating, Refrigerating and Air-Conditioning Engineers Handbook of Fundamentals [1] and the environmental conditions of Military Standard 810c.

2. DESCRIPTION OF TEST SPECIMEN

The walk-in refrigerator was shipped directly from its manufacturer, R.S.P. Industries Inc.,* 329 Herzl Street, Brooklyn, New York 11212, to the National Bureau of Standards. Nameplate data were, in addition to the manufacturer's identity, Model No. AA-450-WPR; Contract No. DAAK60-82-M-4117; and Serial No. 83-3-0757.

It was measured to be 2.36 m (7 ft. 9 in.) wide, 3.66 m (12 ft.) long, 2.29 m (7 ft. 6 in.) high to the extremes of the insulated shell exterior, and 2.43 m (7 ft. 11 1/2 in.) high to the bottom of its integral skid. The doorway was 0.91 m (3 ft.) wide and 1.83 m (6 ft. high). The unit hole was 0.72 m (28 1/2 in.) square. The wall thickness was 102 mm (4 in.).

The nameplate data of the refrigeration unit were:

Refrigeration Unit, Mechanical Type II

Capacity: 5,000 Btu, 0°F Refrig.
7,500 Btu, 35°F Refrig.

Refrigerant: 12

Charge: 136 oz.

NSN: 4110-01-092-3913

Model No.: REMD-G/11-VI-S

Serial No.: TAG-116

Power Supply: 208 v., 3 phase, 60 cycle

Manufacturer: A.R.E. Mfg. Co., Inc.*
Frederick, Maryland

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As received the unit had 96 hours on its running time meter. During initial checkout it was observed that the unit had lost pressure and, upon pressurizing the system, the vibration eliminator in the compressor suction line was found to be leaking. This defective part was replaced and the unit evacuated and recharged prior to performance of the test series described in this report.

3. TEST SET UP AND INSTRUMENTATION

The refrigerator was placed for testing in the largest of the NBS environmental chambers of nominal dimensions 14.9 m long x 12.8 m wide x 9.4 m high (49 ft. long x 42 ft. wide x 31 ft. high). This chamber had an earthen floor which was covered with 19 mm (3/4 inch) plywood to a distance of approximately 0.6 m (2 ft.) beyond the refrigerator outside dimensions. The refrigerator was supported on concrete blocks to provide a 0.3 meter (1 ft.) air space beneath it to insure a uniform surrounding ambient temperature. There was an additional 122 mm (4 13/16 in.) air space incorporated in the refrigerator between the bottom of its insulated floor and the bottom of its integral skid.

Individual Type T thermocouples were installed at each interior and exterior corner of the refrigerator at a distance of 150 mm (6 in.) from the surface. Additional averaging thermocouples were installed in the refrigerator interior to assist in test set up and for connection to the heater controller.

Electrical resistance heat to the refrigerator interior was provided by a single 1.5 kW, 208 v heater placed at the center of its floor. A separate 110 v circulating fan was placed at the unit end of the refrigerator pointed directly at the heater. The power lines were passed out of the unit through the floor drain hole which was located at the unit end of the refrigerator. The controller for the heater power and the watt hour meters for the heater and the fan were located in the test chamber control room.

For the reverse direction heat loss test, a plug constructed of 100 mm (4 in.) of extruded polystyene insulation (two 50 mm pieces) was placed in the unit hole and sealed on the inside with duct tape and sheathed on the outside with 19 mm (3/4 in.) plywood.

The refrigeration unit was instrumented with a 4-junction averaging thermocouple measuring the temperature of air entering the evaporator. Another 4-junction averaging thermocouple measured the temperature of air entering the condenser. For later tests, thermocouples were added to the vapor line at the expansion valve bulb and downstream from the liquid line to the suction line heat exchanger to allow evaluation of the expansion valve setting. For all tests the unit power consumption was read and recorded from a three-phase watt-hour meter. The unit suction and discharge gauges were read and recorded during all tests. The unit suction gauge was checked for its zero setting and found to be reading 24 kPa (3.5 psig) high, which required corrections to be made to the reported data.

4. REFRIGERATOR THERMAL TRANSMITTANCE TEST

The refrigerator thermal transmittance was measured using the reverse-direction (heated interior) heat loss method in which the ambient temperature surrounding the refrigerator was maintained at approximately -17.8°C (0°F) and the refrigerator interior temperature was maintained at approximately 37.8°F (100°F) by an automatically-controlled, metered electrical resistance heater and circulating fan.

For this test only, the insulated plug described in Section 3 was substituted for the refrigeration unit. The heat loss through the plug was calculated following the procedures in Chapter 23 of the ASHRAE Handbook [1] at 0.13 W/°C (0.25 Btu/h°F). This value was subtracted when calculating the refrigerator thermal transmittance.

Prior to taking test data, thermal conditions were allowed to stabilize for 25 hours after the nominal interior and ambient test temperatures had been achieved. Test data were taken for a period of 24 hours with the following summarized results.

Average ambient air temperature	-17.9°C (-0.3°F)
Average refrigerator temperature	37.5°C (99.5°F)
Average temperature difference	55.4°C (99.8°F)
Mean wall temperature	9.8°C (49.6°F)
Power to resistance heater	482.8 W
Circulating fan power	102.9 W
Refrigerator thermal transmittance	10.44 W/°C (19.8 Btu/h°F)

The above listed mean wall temperature is the average of the listed ambient and refrigerator temperatures calculated as specified in Section 8.18.1 of American National Standard MHS.1.2M-1980 [2].

5. CAPACITY TESTS WITH 43.3°C (110°F) CONDENSER ENTERING AIR TEMPERATURE

Following the reverse direction thermal transmittance test described in Section 4, a measurement was made of the capacity of the refrigeration unit with a condenser entering air temperature of 43.3°C (110°F). For this test the unit was installed in the refrigerator in place of the insulated plug used in the thermal transmittance tests. The unit was then operated to maintain an average temperature of approximately -17.8°C (0°F) within the refrigerator and the ambient temperature surrounding the refrigerator raised to the level necessary to provide a condenser entering air temperature of 43.3°C (110°F). After air temperatures had been attained, an additional 24 hours were allowed to elapse in order to obtain thermal equilibrium before data were taken. The unit was not allowed to cycle for this test, the thermostat being turned down to its lowest setting and the automatic controller for the electrical resistance heaters which had been installed in the refrigerator for the test series being set to control at -17.8°C (0°F). After thermal equilibrium had been achieved, the unit was defrosted, yielding approximately 600 ml (20.5 fl. oz) of water. It was then returned to steady-state operating conditions and the following data were observed over a six-hour period:

Average air temperature entering condenser	43.3°C (110.0°F)
Average discharge pressure	1,270 kPa (169 psig)
Average air temperature entering evaporator	-17.9°C (-0.2°F)
Average suction pressure	86 kPa (-2.2 psig)
Average ambient temperature	42.0°C (107.7°F)
Average refrigerator temperature	-17.8°C (-0.1°F)

Refrigerator heat gain	625 W (2133 Btu/h)
Electrical resistance heat	405 W (1383 Btu/h)
Unit capacity	1030 W (3516 Btu/h)
Average unit power draw	2.5 kW
Unit efficiency, COP	0.41

The refrigerant liquid line sight glass was observed to be clear during the above data period.

This test was followed by a 51.7°C (125°F) ambient temperature pulldown and capacity test as described in Section 6. At the completion of the 51.7°C (125°F) ambient test it was decided to rerun the above reported 43.3°C (110°F) capacity test since the measured capacity of 1030 W (3516 Btu/h) was substantially below the rated capacity for the unit of 1465 W (5000 Btu/h). Prior to this rerun, thermocouples were attached to the suction line at the thermal expansion valve bulb and immediately after the heat exchanger to provide a measure of the adequacy of the unit charge and expansion valve setting. Also prior to this rerun, the unit-to-refrigerator seal was taped to insure against air infiltration at this point. When thermal conditions had been reestablished following a procedure similar to that for the first test, the unit capacity was found to have unaccountably increased. The defrost water recovered prior to this test was approximately 300 ml (10.1 fl. oz) after approximately 12 hours of running. This frost collection rate of 25 ml/h versus the previous comparable test's 27 ml/h is clearly not responsible for the change in unit capacity.

The thermocouple on the suction line indicated a cycle with occasional flooding through the heat exchanger indicating that neither increased charge nor expansion valve superheat adjustment would be likely to increase capacity. The summarized observed performance for this test was:

Average air temperature entering condenser	43.5°C (110.3°F)
Average discharge pressure	1,290 kPa (172.3 psig)
Average air temperature entering evaporator	-17.8°C (-0.2°F)
Average suction pressure	101 kPa (-0.1 psig)
Average ambient temperature	42.4°C (108.3°F)
Average refrigerator temperature	-17.7°C (0.2°F)
Refrigerator heat gain	626.8 W (2139.2 Btu/h)
Electrical resistance heat	561.6 W (1916.6 Btu/h)
Unit capacity	1188.3 W (4055.8 Btu/h)
Average unit power draw	2.55 kW
Unit efficiency, COP	0.47

For this test the stabilization period was 16 hours and the data period was 9 hours. As before, the refrigerant sight glass was clear throughout the data period. The increase in suction pressure from the previous test without change in evaporator entering air temperature would indicate that a change in unit performance did occur as a result of some change internal to the refrigeration unit such as a particle of dirt partially blocking the expansion valve orifice.

The observed suction pressure which was equivalent to -24.8°C (-12.6°F) saturated suction temperature at the compressor and the cyclic flooding into the heat exchanger would indicate that the unit was operating properly during this repeat test.

The ambient dew point for these tests was maintained at approximately 15.6°C (60°F) which, at a dry bulb temperature of 43.3°C (110°F), corresponds to a relative humidity of approximately 20%.

Immediately following this test the 51.7°C (125°F) pulldown and capacity test was repeated.

6. PULLDOWN AND CAPACITY TESTS WITH 51.7°C (125°F) AMBIENT AIR TEMPERATURE

Each of the 43.3°C (110°F) condenser entering air temperature capacity tests discussed in Section 5 was followed by a 51.7°C (125°F) ambient temperature pulldown and capacity test.

For the first of the indicated tests the refrigerator, with its door open, was allowed to come to thermal equilibrium for a period of 16 hours after its exterior and interior air temperatures had reached the value of approximately 51.7°C (125°F). At the end of this period the door was closed and the unit started. After 1 hour and 20 minutes of operation the refrigerator interior temperature had reached an average of 1.7°C (35°F). After 6 hours and 30 minutes from unit start-up, the refrigerator interior temperature had reached an average value of -17.8°C (0°F). The metered electrical resistance heaters which had been installed in the container were then energized with their automatic controller set to maintain the refrigerator interior temperature at -17.8°C (0°F) and the unit was run for an additional 24 hours to allow thermal conditions to stabilize before steady-state data were taken. The unit was defrosted yielding approximately 500 ml (16.9 fl. oz.) of water five hours before the steady-state data were taken.

The summarized data for the steady-state period are as follows:

Average air temperature entering condenser	52.6°C (126.6°F)
Average discharge pressure	1,515 kPa (205 psig)
Average air temperature entering evaporator	-17.8°C (-0.1°F)
Average suction pressure	98 kpa (-0.5 psig)

Average ambient temperature	50.4°C (122.7°F)
Average refrigerator temperature	-17.7°C (0.1°F)
Refrigerator heat gain	711 W (2425 Btu/h)
Electrical resistance heat	101 W (346 Btu/h)
Unit capacity	812 W (2771 Btu/h)
Average unit power draw	2.56 kw
Unit efficiency, COP	0.32

As discussed in Section 5, this test was followed by a repeat 43.3°C (110°F) condenser entering air air unit capacity test and then was itself repeated.

For the repeat 51.7°C (125°F) pulldown and capacity tests, the refrigerator was allowed to come to thermal equilibrium for a period of 10 hours with its door open after its exterior and interior air temperature had reached a value of approximately 51.7°C (125°F). At the end of this period the door was closed and the unit started. After 1 hour and 18 minutes of operation, the refrigerator interior temperature had reached an average value of 1.7°C (35°F). After 5 hours from unit start-up the refrigerator interior temperature had reached an average value of -17.8°C (0°F). As before, the metered electrical resistance heaters were then energized with the automatic controller set to maintain the refrigerator interior temperature at -17.8°C (0°F) and the unit ran for a short additional period (two hours) before the following steady-state data were taken:

Average air temperature entering condenser	52.6°C (126.7°F)
Average discharge pressure	1510 kPa (204.5 psig)

Average air temperature entering evaporator	-17.7°C (0.1°F)
Average suction pressure	92 kPa (-1.4 psig)
Average ambient temperature	50.4°C (122.8°F)
Average refrigerator temperature	-17.4°C (0.6°F)
Refrigerator heat gain	708 W (2417 Btu/h)
Electric resistance heat	108 W (369 Btu/h)
Unit capacity	816 W (2786 Btu/h)

The results of this test are not substantially different from those of the previous 51.7°C (125°F) test.

The ambient dew point for these tests was maintained at approximately 15.6°C (60°F) which, at a dry bulb temperature of 51.7°C (125°F), corresponds to a relative humidity of 15 percent.

7. EFFECT OF SOLAR RADIATION

A popular way of handling the effect of solar radiation on cooling load is the sol-air concept as described in the air-conditioning cooling load chapter of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers Handbook of Fundamentals [1]. The sol-air temperature, t_e, is calculated from the equation:

$$t_e = t_o + \alpha I_t/h_o - \varepsilon \Delta R/h_o$$

where

α = absorptance of the surface for solar radiation.

I_t = total solar radiation incident on the surface, W/m^2 ($Btu/(h \cdot ft^2)$)

h_o = coefficient of heat transfer by long wave radiation and convection at the outer surface, $W/(m^2 \cdot °C)$ ($Btu/(h \cdot ft^2 \cdot F)$).

t_o = outdoor air temperature, °C (F).

ε = hemispherical emittance of the surface

ΔR = the difference between the longwave radiation incident on the surface from the sky and surroundings, and the radiation emitted by a blackbody at outdoor air temperature, W/m^2 ($Btu/(h \cdot ft^2)$).

For horizontal surfaces that receive longwave radiation from the sky only, such as the roof of this refrigerator, Chapter 26 of the ASHRAE Handbook [1] recommends a value of -3.9°C (-7°F) for the longwave reradiation correction term, $\varepsilon \Delta R/h_o$. The ASHRAE recommended maximum value (i.e., for dark-colored surfaces) for α/h_o is 0.052 (0.30). The maximum value for direct, normal I_t given in the ASHRAE tables of solar intensity in Chapter 27 of reference [1], 1058 w/m^2 (335 $Btu/(h \cdot ft^2)$), corresponds to the maximum value specified by MIL-STD-810c [3]. Substituting these numbers gives:

$$t_e = t_o + (0.052)(1058) - 3.9 = t_o + 51.1°C\ (92°F)$$

Since this sol-air temperature should only be applied to the roof of the container, a new sol-air temperature, $t_{e,amb}$ can be defined by reducing the term to be added to the ambient by multiplying by the mean roof area divided by the total mean insulated surface area. Mean areas are the mean of the inside and outside areas of each surface. The roof area of this container is 8.04 m^2 (87 ft^2) and the total mean area is 41.49 m^2 (447 ft^2), hence:

$$t_{e,amb} = t_o + (51.1)(8.04)/41.49) = t_o + 9.9°C\ (17.8°F)$$

A similar calculation can be made for the other high ambient test point of MIL-STD-810c [3] which, when used with the refrigerator thermal transmittance measured in Section 4, gives the unit capacity requirements in Table 1.

Table 1. Unit Capacity Requirements

Ambient Temperature °C (°F)	Solar Radiation W/m^2 (Btu/(h · ft^2))	$t_{e,amb}$ °C(°F)	Unit Capacity Requirement W (Btu/h)
47.8°C (118)	1058 (335)	59.2 (138.6)	801 (2744)
48.9°C (120)	739 (234)	55.6 (132.1)	764 (2616)

Since the unit capacity measured in a 50.4°C (122.8°F) ambient was 816 W (2786 Btu/h) and increased at lower ambient temperatures, these requirements are clearly met under the assumption that for calculating refrigeration load the direct solar normal incident radiation criteria should only be applied to the surface area of the refrigerator roof. The question of whether a higher total radiation load would result from assuming angular incident radiation covering the roof and one side and the effects of mass storage in producing a coincident load peak were not investigated.

8. CONCLUSIONS

Although this particular refrigeration unit was operating somewhat below its rated capacity, it was able to pull down the refrigerator interior temperature from 51.7°C (125°F) to -17.8°C (0°F) in a 51.7°C (125°F) ambient and to hold the refrigerator interior temperature at -17.8°C (0°F) in that same ambient temperature with sufficient extra capacity to meet a likely solar load. It can therefore be concluded that the assemblage of this refrigerator with a 1460 W (5000 Btu/h) refrigeration unit would be expected to perform adequately in the field with respect to maintaining required refrigeration temperatures, even with some degradation of unit and/or refrigerator performance.

9. REFERENCES

1. "ASHRAE 1981 Handbook of Fundamentals", American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, Georgia 30329.

2. "American National Standard MH5.1.2M-1980, Requirements for Thermal Van Containers (Refrigerated, Heated, and Insulated)", American Society of Mechanical Engineers, New York, New York 10017.

3. "Environmental Test Methods", Military Standard 810c, 1975, Aernoautical Systems Division, Wright-Patterson Air Force Base, Ohio 45433.

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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 environmental tests on a newly developed 450 cubic foot air transportable field refrigerator for the U.S. Army Natick Research and Development Laboratories. The refrigerator, in assemblage with a supplied refrigeration unit, was found to be more than capable of pulling down to and holding an interior temperature of -17.8°C (0°F) when surrounded by an ambient temperature of 50.4°C (123°F). Additional tests indicated that the performance of this assemblage would exceed that required to meet any likely combinations of sensible and radiant (solar) heat gain.

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

Field refrigerators; frozen food storage; military refrigerators; refrigerator assemblage testing; refrigerators; walk-in refrigerators.

13. AVAILABILITY

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