NATIONAL BUREAU OF STANDARDS REPORT

9378

DEVELOPMENT OF A METHOD FOR TESTING AND RATING

THE COOLING LOAD OF REFRIGERATED TRUCK BODIES

by

C. W. Phillips and R. W. Penney

Report to

Transportation and Facilities Research Division
Agricultural Research Service
U. S. Department of Agriculture

NBS

U. S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS

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C. W. Phillips
Environmental Engineering Section
Building Research Division, IAT

and

R. W. Penney
Transportation and Facilities Research Division
Agricultural Research Service
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1. Acknowledgement

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2. Introduction

Lack of a standard method for determining the cooling load of refrigerated truck bodies has made it difficult, if not impossible, to effectively rate or select these vehicles, so widely used in the transportation and distribution of perishable foods and other commodities.

Following the development, through Government-industry cooperation, of a standard method for measuring the cooling load of refrigerated trailers, persons and organizations, from industry and Government, similarly concerned with the manufacture, selection, application, and utilization of refrigerated trucks, established a project to develop a suitable method for rating these vehicles (see Project Organization).

It is important to recognize that measurement of the cooling load of a refrigerated enclosure (such as a truck, trailer, railway car, or warehouse) does not provide a direct indication of the total cooling capacity required of the refrigerating apparatus for the enclosure. Additional application factors such as loading and unloading techniques, cargo temperatures, door opening schedules, and required temperature pull-down rates all influence service loads and the sizing of the equipment, some perhaps to a greater degree than the rated cooling load of the insulated enclosure.

This report presents the results of cooling load tests, with and without simulated solar heating, and certain other tests of five standard or prototype insulated truck bodies of various types and sizes available for commercial refrigerated service. Based on these test results, a recommended method is presented for measuring the cooling load and weight gain of refrigerated trucks. The method is applicable also to smaller trailers and containers of size and usage comparable to the refrigerated trucks.

3. Purpose

The principal objective of this project was the development of a procedure or technique suitable for use as a standard method for rating the cooling load of insulated truck bodies used for the transportation of perishable and frozen food or other commodities requiring refrigeration. The desired method was to take into account the following:

(1) The cooling load of the vehicle under standard conditions.
(2) The rate of weight gain due to moisture accumulation caused by air leakage under standard conditions.
(3) The increase in cooling load due to solar heating.

4. Background

In an earlier project, a rating method for refrigerated trailers was developed to evaluate the cooling load, air leakage, and moisture gain of those vehicles intended for long-haul highway transportation of frozen foods [1].

Certain differences were noted between usage of highway trailers and local delivery trucks, which indicated that different test procedures should be used in rating the two types of vehicles.

In general, the highway trailers operate day and night, usually at road speeds, and are opened for loading and unloading only at terminals or major transfer points. The additional cooling load caused by solar radiation while the vehicle is stationary is possibly less than that caused by air leakage when the vehicle is in motion. Doors are opened infrequently, if at all, during a trip.

Delivery trucks are stationary for the most part, operate primarily during daylight hours, and are subjected to a wide variety of door opening schedules during the working period. Thus, the rating method for trailers required means for simulating the effect of forward motion, but not solar radiation or door opening. The rating method for trucks, in contrast, requires means for providing the effect of solar radiation, but not necessarily the effect of forward motion. It is recognized that there are exceptions to the general case.

Of the radiant energy absorbed by a surface over insulation such as an insulated truck wall, only a small percentage will actually be transferred through the insulation, while a high percentage will be radiated to surrounding lower temperatures or given up to convective air movement at the surface. Bright metallic surfaces absorb less solar energy than dark painted surfaces, but in turn they are less able to re-radiate the energy absorbed. Thus, differences in the temperatures attained by the surfaces are less than those based solely upon the solar absorbtances of the different surfaces. As each surface is heated above ambient air temperature, it loses additional heat to convective air currents in proportion to the difference in temperature between the surface and the air.

For these tests it was desired to determine the effect of equal solar heating, regardless of the finish of the test vehicle, or the various types or methods of construction of the various truck bodies in the test series. A low temperature wavelength radiation source was used for the simulated solar tests so that all vehicles tested would be heated approximately as if they were painted with dark paint and exposed to solar radiation of equal intensity. It was not an intended purpose of these tests to determine what type of surface treatment was the best for minimizing the heat gain due to solar exposure.

Initially, it was the intent of this project to include the effects of door openings on (a) air and moisture exchange, (b) the increase of cooling load. Much work was done on these subjects; however, it now appears that these effects are related more to the "service load" of the vehicle than to a rating method for the insulated body itself. The effects of air and moisture exchange during door openings, with the attendant increase in cooling load, will be covered in a report to be published at a later date.

5. Project Organization

The Environmental Engineering Section, NBS; Transportation and Facilities Research Division, USDA; and the Truck Body and Equipment Association, beginning July 1, 1960, sponsored a cooperative effort to develop a suitable method for rating refrigerated trucks that would take into account the factors outlined (see Background). The project was carried out at NBS under the direction of the Environmental Engineering Section. A Project Steering Committee was appointed, consisting of representatives from interested industry and Government organizations. The initial Project Steering Committee meeting was held on October 14, 1960. Paul R. Achenbach, of the National Bureau of Standards, was elected and served as chairman. Harry R. McGee, formerly of the Truck Body and Equipment Association, served as secretary until succeeded by Robert F. Guilfoy, of the U. S. Department of Agriculture.

It was a recommendation of the Steering Committee that the proposed rating technique be as consistent as possible with that used for rating trailers [1]. Thus, a test facility could, without major difficulty, rate either trucks or trailers. Based on this, the decision was made to maintain the 0°F interior temperature and the 100°F, 50% RH ambient temperature and humidity conditions used to rate trailers.

Of the five test vehicles used, some were prototype and some production models (see Description of Test Vehicles). They were furnished by the following cooperating manufacturers:

Murphy Body Works, Inc., Wilson, N. C.
Boyertown Auto Body Works, Boyertown, Pa.
Hackney Bros. Body Company, Wilson, N. C.
The Heil Company, Milwaukee, Wisconsin
Divco Truck Division, Divco-Wayne Corporation, Detroit, Mich.

The first of the five vehicles was placed in the test chamber in April 1961.

6. Test Systems

To measure the cooling load of the trucks tested in this project a metered heat sink with comparison heater was used [2]. This apparatus was similar to that used in the rating method for trailers and is similar to that recommended in the proposed rating method for trucks. A measured mass flow rate method was used for the independent simultaneous measurement.

Alternative methods considered for the proposed rating method were: use of liquid nitrogen (or other expendable refrigerant) expanded into the vehicle under test, transient-state cooling load tests, and reverse heat loss tests. None of these was selected for various reasons. Reverse heat loss tests do not account for the effect of moisture deposit in the insulated spaces; transient-state cooling load tests are better suited to homogeneous material construction and might not properly account for the effect of moisture deposit in the insulated cavities. The use of direct discharge of liquid nitrogen into a truck body was tried [3]. Control by use of thermocouple-operated solenoid valves was satisfactory. Direct weighing of the nitrogen tank gave demand and load values. One serious drawback for rating purposes was found; the slight pressure in the truck interior needed to discharge the expanded nitrogen gas to the outside tended to offset the air leakage rate of the vehicle. In other words, it favored a truck with high air leakage rates. If a suitable metered heat exchange method can be found to allow indirect cooling by the liquid and gaseous nitrogen, it can then be evaluated for possible advantages in cost and simplicity for rating purposes. For all alternative methods, an independent simultaneous measurement would be required.

6.1 The Metered Heat Sink Principle

The basic metered heat sink apparatus with the comparison heater is shown in Figure 1. An important feature of this method, outlined in the following derivation, is that measurement of the mass flow rate, M, and the specific heat, c_p, of the refrigerant (brine) is not required to determine the heat flow rate, q_t, into the truck body under test. The heat flow rate into the truck, sensible and latent, is represented by

$$q_t = c_p M \Delta t_t - h$$

where c_p is the specific heat of the brine, Btu/lb deg F
M is the mass flow rate of the brine, lb/hr
Δt_t is the brine temperature change in the air cooling coil, deg F
h is the total electric power input to the truck (fans, heaters, etc.), Btu/hr.

The electrical input to the brine in the comparison heater is represented by

$$q_c = c_p M \Delta t_c$$

where c_p is the specific heat of the brine, Btu/lb deg F
M is the mass flow rate of the brine, lb/hr
Δt_c is the brine temperature change in the comparison heater.

Combining the two equations,

$$c_p M = \frac{q + h}{\Delta t_t} = \frac{q_c}{\Delta t_c}$$

$$q_t = q_c \frac{\Delta t_t}{\Delta t_c} - h$$

Note that both M and c_p are taken to be constant in the two equations combined; M will be constant at any given time with a closed liquid brine system. The change in c_p of methylene chloride brine between the mean coil temperature and the mean comparison heater temperature is less than 0.5 percent for the operating ranges used and recommended.

In the basic metered heat sink method with the comparison heater, a cooling coil refrigerated by an adjustable brine flow is placed in the body under test, and the comparison heater is placed in the brine line from this cooling coil. The refrigerating equipment for cooling the brine is outside of the test chamber. In a variation of this apparatus, based on a design by M. Altieri [4], a complete water-cooled refrigerating unit is placed in the body under test. A comparison heater could be placed in the water line leaving the refrigerating unit to provide an independent simultaneous measurement. The proposed rating method (see Appendix II) recommends the basic comparison heater method using brine and includes requirements for the necessary instrumentation and operating procedure to provide the required accuracy for rating purposes. An independent simultaneous check is provided by the measurement of mass flow rate by flowmeter.

6.2 Air Leakage Determination Methods

In the course of this project, the air leakage rate of the trucks was studied from two aspects. One was the air leakage, under standard test conditions, as calculated from the observed weight gain rate, and the other was the measured air leakage from the cargo space by (1) a helium trace katharometer measurement under standard test conditions [5], and by (2) a static pressure test wherein the cargo space was pressurized under isothermal conditions.

The air leakage as determined from the weight gain is a function of both the air movement between the truck interior and exterior and the air movement into and out of the insulated walls of the vehicle. This method provides a basis for calculating the cooling load due to air leakage. It is determined from the weight gain by assuming that the air enters the vehicle at ambient conditions, and leaves saturated at the interior temperature (see Air Leakage Calculations). This yields a minimum air leakage rate and corresponding latent and sensible cooling loads.

The static pressure air leakage test and the helium trace air leakage test each measure an air flow rate which is principally influenced by the resistance to air movement between the cargo space and the exterior. Neither of these tests provides a basis for determining the cooling load due to air leakage under standard test conditions. In these tests, a truck with a well-sealed interior skin and a poorly sealed exterior skin (contrary to good design practice) might show an air leakage rate no higher than a truck with a well-sealed exterior skin, which would have a much lower latent cooling load.

A degree of correlation does exist between static pressure test results and cooling loads in that a truck with a high air leakage rate will no doubt have a high air leakage cooling load, regardless of the mode of entry and the inability to calculate that load.

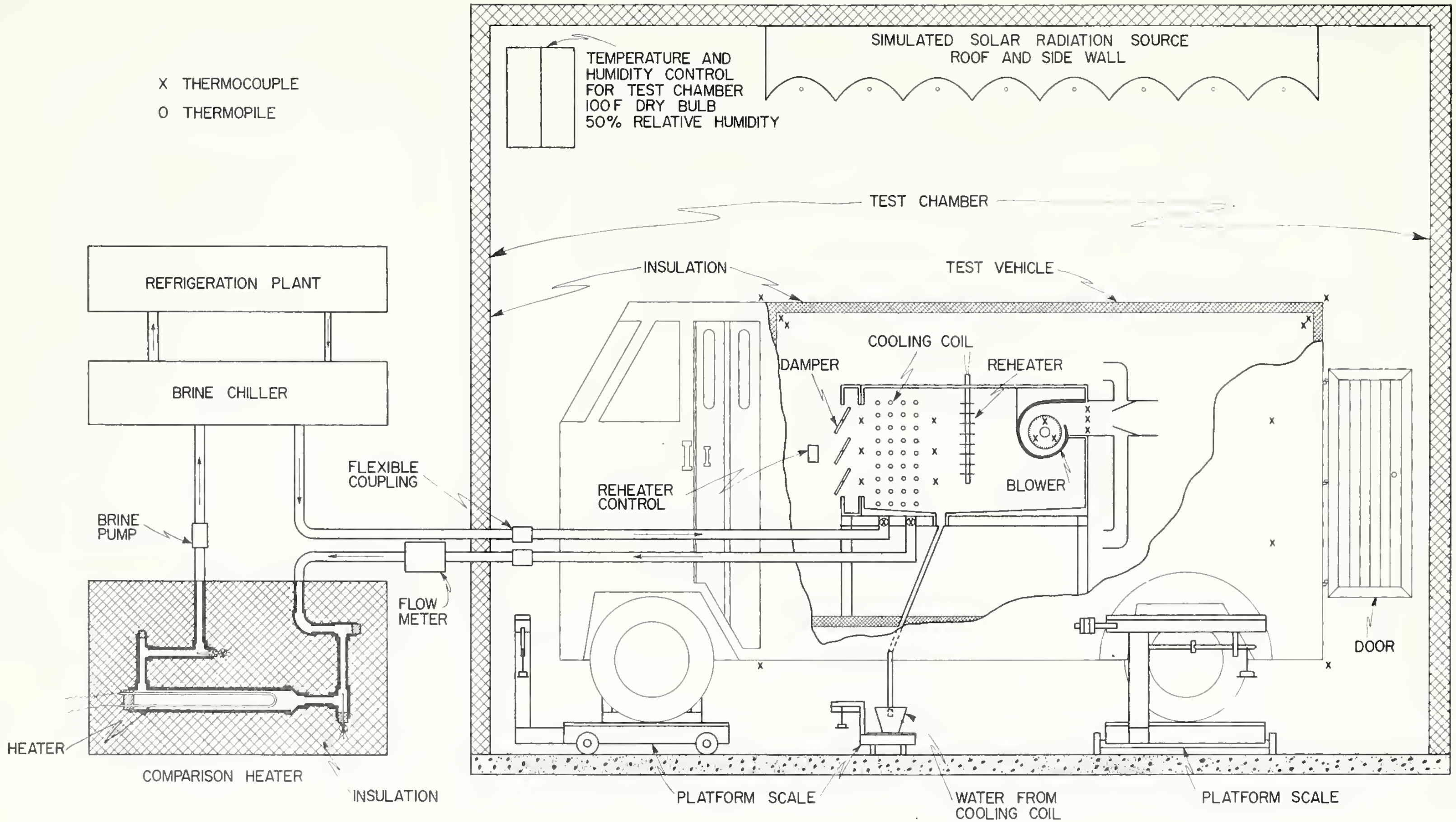
X THERMOCOUPLE
O THERMOPILE
TEMPERATURE AND HUMIDITY CONTROL FOR TEST CHAMBER 100F DRY BULB 50% RELATIVE HUMIDITY
SIMULATED SOLAR RADIATION SOURCE ROOF AND SIDE WALL
TEST CHAMBER
INSULATION
TEST VEHICLE
REFRIGERATION PLANT
BRINE CHILLER
COOLING COIL
DAMPER
REHEATER
FLEXIBLE COUPLING
REHEATER CONTROL
BLOWER
BRINE PUMP
FLOW METER
DOOR
HEATER
COMPARISON HEATER
INSULATION
PLATFORM SCALE
WATER FROM COOLING COIL
PLATFORM SCALE

FIGURE 1

7. Description of Test Facilities

The ambient test chamber used was 25 ft long, 17 ft wide, and 12 ft high. Two large doors at one end permitted entry of the trucks. The ambient test conditions were 100°F temperature and 50% RH, and provisions for conditioning the ambient air were provided by two fan coil units discharging into a common plenum above a perforated ceiling. Air return was direct into each air handler. Refrigeration machinery, steam regulators, and controls were located external to the test room. Figure 2 shows a view of the test chamber.

The interior temperature of each test vehicle was maintained at 0°F (35°F for one test to represent medium temperature applications) by a refrigeration system consisting of a packaged-unit brine cooler, a turbine-type brine pump, an air cooling coil, and the associated piping and valves. The brine cooling unit was capable of two-speed operation for better control of cooling capacity over the wide range required. The primary refrigerant was R22 (monochlorodifluoromethane). The secondary refrigerant, or brine, was R30 (methylene chloride). At the mean coil temperature used in these tests, the specific heat of the brine was approximately 0.273 Btu/lb deg F . The turbine type pump, driven by a 1/2-hp motor, circulated the brine through the secondary refrigerant circuit. The pump was located in the return line, just before the chiller. The rated capacity of the pump was 8 gpm (water) against a 40 psi head.

An air cooling assembly located in the truck body contained a coil, blower, damper, and electric heaters. The coil was selected with only four fins per inch to permit extended operation without defrosting. Three electric resistance heaters were mounted between the coil and the blower. These were for control purposes. In figure 3, the air cooler assembly is opened to show the coil and resistance heaters. At the coil inlet, a damper was installed as a means of controlling air flow. At the blower outlet, a baffle distributed the chilled air to the various parts of the truck body interior. Figure 4 shows the brine chiller, brine pump, and the housing (at bottom right) of the comparison heater used in the metered heat sink apparatus. Figure 1 shows schematically the cooling load calorimeter or metered heat sink system as applied for truck body testing. The cooling coil inside the truck body was defrosted automatically in a short time (<10 min.). This defrost system circulated brine through the cooling coil and the comparison heater (which provided heat for the defrosting operation), while by-passing the chiller. The brine remaining in the chiller was thus sub-cooled by the primary refrigerant system, which continued to operate at low speed. The sub-cooled brine in the chiller was circulated through the cooling coil at the end of the defrost period to restore the test temperature in the truck rapidly.

The brine circuit was heavily insulated and vapor sealed in order to eliminate frost and minimize heat gain. To prevent any vertical forces from being exerted on the truck which would interfere with the proper weighing of the vehicle, two flexible lines were installed horizontally near the point where the brine pipes enter the truck body. The thermocouple wells for the brine lines at the truck were located so as to respond to the temperature of the brine immediately before entering and after leaving the truck. The comparison heater consisted of a special piping configuration enclosed in a plywood box about 40 by 20 by 20 in. (see Fig. 5). The box was insulated with loose cork fill. Thermocouple wells were fitted into the piping near the inlet and outlet. Between the temperature measuring wells, an electric resistance heater was mounted inside the pipe. The thermocouple wells for the comparison heater were located so that the brine would be well mixed at the points of measurement. Piping within the box was installed so that vapor would not collect around points of measurement. A surface thermostat, located on the pipe containing the heater element, protected the heater against damage due to overheating. The insulated box was also vapor sealed so that moisture would not accumulate in the insulation. Energy input to the brine by the comparison heater was measured by a watt-hour meter on the control panel.

An electronic integrating flow meter with a volume-measuring sensing element was located in the brine line between the coil outlet and the comparison heater. The weight rate of flow of the brine is the product of the volumetric rate and the specific gravity of the brine. The weight rate of flow provides one factor for determining a comparative value for the refrigeration load at the coil.

The temperature measuring system used for these tests employed copper constantan thermocouples, a 16-point electronic recording potentiometer, an electronic indicating potentiometer and a precision-grade laboratory potentiometer. The charts from the recording potentiometer were used to determine that steady-state conditions were maintained throughout the measurement period. The temperatures used to calculate cooling loads were read on the electronic indicating potentiometer and/or the laboratory potentiometer. The latter was used to obtain a greater degree of accuracy in determining the critical temperature differences between the brine entering and leaving the comparison heater and the air cooling coil. These temperature differences were sensed by calibrated five-junction copper-constantan thermopiles, which were inserted in the thermocouple wells in the brine lines. Individual thermocouples were also placed in the wells to monitor the thermopiles and to determine mean brine temperatures.

The interior temperature in the truck and the ambient temperatures of the test room were sensed both by parallel-connected (for average temperatures) and individual thermocouples suspended 6 in. from the interior and exterior corners. Other thermocouples were placed to measure various temperatures of interest, such as those of the interior and exterior truck surfaces, primary refrigerant temperature, and those of the air entering and leaving the cooling coil assembly.

It was necessary to weigh the defrost water from the cooling coil to determine total moisture gain and the portion of the total gain which accumulated in the insulation space. To accomplish this, a rubber hose connected to the defrost pan of the cooling coil emptied into a container in the test room outside of the truck. An electric cord heater in the defrost pan and rubber hose prevented freezing during the defrost operation. A water trap on the end of the hose prevented outside air from being drawn through it into the truck interior.

Three separate platform scales were used to weigh the truck to determine the weight gain caused by moisture picked up from air leakage. Each rear wheel rested on a scale. The front axle was supported on a third scale. These scales had a capacity of 6000 lb each, and were sensitive to differences of 1/2 lb at the loading used. Figure 6 shows a truck installed on the scales.

7.1 Solar Simulation Apparatus

Previous observations of unmoving trucks and trailers exposed to bright summer sunlight at NBS had indicated skin temperatures approaching 70°F above ambient, and a value between 65 and 70°F was arbitrarily selected an an approximate upper roof skin temperature limit for the simulated solar tests. It was postulated that the maximum amount of solar energy would enter an insulated vehicle when it was parked with its longitudinal axis in a north-south direction so that both longer sides and the roof would be irradiated in the course of a cloudless day.

The idea of a rotating mechanism to duplicate the effect of the daily sun movement over a vehicle was discarded in favor of adjustable stationary banks of electric heating elements and parabolic reflectors facing each of two sides and the top of a test vehicle [6]. To simulate the effect of sun movement during the day, the voltage applied to the heaters was varied. Each bank was 15 ft long by 9 ft wide and consisted of 45 parabolic reflectors, 1 by 3 ft at the bank face (see Fig. 2). Each reflector was equipped with an 81-ohm helical electric resistance coil wound around a 7-mm O.D. glass tube mounted in the reflector so that the heater axis was at the focus of the parabolic reflector. The heating elements were designed to operate without visible radiation at 208 volts. Maximum heat dissipation of each heater, including both convective losses and radiant transfer, was about 600 Btu/hr ft^2 of projected reflector areas. The total heat release of one bank of 45 heaters operating at maximum design voltage was approximately 83,000 Btu/hr. Figure 7 shows one of the test trucks in the test room for simulated solar load tests.

7.2 Static Pressure Air Leakage Apparatus

The static pressure air leakage apparatus consisted of a centrifugal blower connected to the truck interior, with an orifice in the connecting 2-in. pipe line. The pressure drop across the orifice was used to calculate the air flow rate. A manometer was used to measure the pressure difference between the interior and the exterior of the truck. The orifice measurements were checked by pitot tube sweeps across the inlet air pipe in the truck.

7.3 Helium Trace Katharometer Apparatus

The helium trace katharometer apparatus used to check the air leakage rate was a version of the helium trace apparatus described in reference 5 modified to permit "batch" sampling of the air-helium mixture in the truck. The modified katharometer was used both for the steady-state tests in this report and for the subsequent tests of air exchange through door openings which will be covered in a later report. The details of the modified katharometer will be described fully in that report.

8. Description of Test Vehicles

8.1 Test Vehicle 1

Truck body 1 was a walk-in ice cream delivery design with exterior dimensions of 183 in. long by 91 in. wide by 87 in. high. A single door at the rear was the only access, except for a pass door on the right side forward of the wheel well. Internal volume was 544 cu ft.

The metal exterior, including the roof, was painted a gloss white. the interior metal walls were smooth except for anchors to which refrigerated plates could be attached on the ceiling and forward wall.

Insulation consisted of 7 in. of expanded polystyrene plus 2 in. of glass fiber in the roof, 7 in. of expanded polystyrene in the walls, and 6 in. of expanded polystyrene plus 1 in. of cork in the floor.

Figure 8 shows vehicle 1.

8.2 Test Vehicle 2

Truck body 2 was an integral cab design having a door entering from the cab, a door on the curb side, and a single rear door. Cargo body external dimensions were 176-1/2 in. long by 80-1/2 in. wide by 75-1/2 in. high. Internal volume was about 388 cu ft.

The exterior surface was painted a dark color of a non-glare finish. Interior surfaces had corrugations on the walls, with a sheet metal floor and ceiling.

The insulation in the walls and roof was fiber glass (6 in. and 8 in. thick, respectively) and there was 7 in. of expanded polystyrene in the floor.

Figure 9 shows vehicle 2.

8.3 Test Vehicle 3

Truck body 3 was a reach-in design with three doors on each side. The rear opened to a storage compartment used for supplies, which was not refrigerated and which did not communicate with the larger space. Exterior dimensions were 182 in. long by 90-1/2 in. wide by 88 in. high. The metal exterior surface was painted a gloss white. The internal refrigerated volume was 432 cu ft.

Insulation consisted of 6 in. of urethane in the roof, 4 in. of urethane in the walls, and 3 in. of urethane plus 1 in. of cork in the floor. The urethane was foamed in place. Spacer strips were provided on the metal interior walls.

Figure 10 shows vehicle 3.

8.4 Test Vehicle 4

Truck body 4 was constructed of urethane slabs resin bonded to multi-layer glass laminate for both interior and exterior surfaces. The bonding agent penetrated the glass laminate, attaching it to the insulation and forming a hard smooth surface after curing in the mold. Sides, ends, and roof were formed in a mold as one subassembly and the floor as another. These parts were then assembled and resin-bonded with glass laminate overlapping at all joints. Insulation thickness was 2 in. in walls and 3 in. in floor and roof. The color inside and out was white.

Overall exterior dimensions were 140 in. long by 83 in. wide by 63 in. high. Internal volume was 333 cu ft.

A 22-1/2 in. wide by 44-1/2 in. high door was located at the rear of each side.

Figure 11 shows vehicle 4.

8.5 Test Vehicle 5

Truck body 5 was an integral cab design. Entry was by double doors at the rear or by a sliding door from the driver's compartment.

The metal exterior surface was painted a flat grey. The metal interior lining had corrugated sheets on the roof, floor, and side walls.

A combination of glass fiber and preformed urethane board insulations was used as follows: rear wall, 3 in. urethane and 1 in. glass fiber; front wall, 2 in. urethane and 1 in. glass fiber; side walls and floor, 3 in. urethane; and roof, 3 in. urethane and 1 in. glass fiber. The glass fiber was used to fill between stiffening and structural members.

The internal volume of the body was 183 cu ft and its exterior dimensions were 92 in. long (not including integral cab) by 79 in. wide by 68 in. high.

Figure 12 shows vehicle 5.

9. Laboratory Test Procedures

9.1 Steady-State Cooling Load and Weight Gain

After each truck was placed on the scales in the test chamber, the brine lines, electric cables, and thermocouples were connected. The truck interior was then refrigerated to 0°F temperature (35°F for one medium temperature test), and the ambient temperature and humidity were controlled at 100°F and 50% RH. When controlled conditions were obtained, the scales were balanced. By this time, the change in air density in the truck had taken place, and no longer affected the readings.

Air leakage rates during steady-state cooling load tests were measured by means of the modified helium-trace katharometer.

For a steady-state heat balance, the instruments were read at half-hour intervals until a period of test was obtained having not less than 6 hours of uninterrupted test conditions with essentially constant heat gain. Each test was repeated one or more times, with the vehicle maintained under test conditions for several days.

Weight gain readings were made during the entire test period for each truck with adjustment made for water removed from the cooling coil, if defrosting was required. Defrosting, if required, was done at the end of a period of data observation, allowing nearly 16 hours for recovery of steady-state conditions prior to the next period of data observation.

9.2 Solar Load Test Procedure

The test of a vehicle under simulated solar heat load was similar to that of steady-state, with the interior temperature set at 0°F, and the ambient air temperature and humidity held at 100°F and 50% RH. Simulated solar heating was imposed by the banks of electric heaters. The measured temperature of one of the heater rods operating at the maximum voltage necessary for test was 700°F. At this temperature the wavelength of maximum emission is about 4.5 microns. As the voltage is lowered, the temperature of the heater approaches the ambient temperature of 100°F, at which temperature the wavelength of maximum emission is about 9.3 microns. The significance of selecting a radiation source at temperatures of 700°F or lower is related to the capacity of various surfaces to absorb radiant energy. Surfaces covered with nonmetallic paint, such as most truck bodies, regardless of color, will absorb about the same fraction of radiation at the wavelengths above 4 microns as dark-painted surfaces will absorb at the shorter wavelengths of solar radiation. For this reason it was considered unnecessary to repaint any of the painted vehicles in order to establish similar conditions for the simulated solar tests. Metallic surfaces such as aluminum or stainless steel, or metallic-painted surfaces would have required special consideration. All the vehicles received for the test series were painted with nonmetallic paints.

To establish a pattern for cyclic variation of the simulated solar energy values of hourly insolation on a vertical east and west surface, and on a horizontal (roof) surface were taken from U. S. Weather Bureau curves for June 21 at latitude 40°N. These are shown in Figure 13. Such a solar day extends from 4:40 a.m. to 7:20 p.m., with maximum insolation values of 235 Btu/hr ft^2 on a vertical east wall at 7:30 a.m., and on a vertical west wall at 4:30 p.m.; and 320 Btu/hr ft^3 on a horizontal surface at noon.

The voltage on the bank of heaters over the roof was adjusted to produce a temperature rise approaching 70 degrees above the ambient temperature of 100°F, thus simulating solar irradiance at noon. With this value as a maximum, the voltage on the roof heater at other times of the day was adjusted to produce a power dissipation proportional to the height of the curve in Figure 13 for the roof. The voltage on the other two banks was adjusted independently to provide power dissipations proportional to the heights of the other two curves in Figure 13. The same cycle of power input was used with all vehicles.

Seven tests were run on one vehicle (body E) to determine the effect of solar heat gain on the 4-hour maximum cooling load (average cooling load rate over the 4-hour period of maximum load) when:

(1) The ambient temperature was held constant.
(2) The ambient temperature was varied sinusoidally to follow simulated daily cycle with diurnal range of 20 deg F.

The seven tests run were:

		Ambient		
	Conditions	Mean temp. °F	Temp. range °F	Solar simulation
1.	Steady	100	--	No
2.	Steady	100	--	Yes
3.	Variable	100	90 to 110	No
4.	Variable	100	90 to 110	Yes
5.	Variable	90	80 to 100	Yes
6.	Variable	90	80 to 100	No
7.	Steady	90	--	No

Tests 2 and 1 were run to compare the observed cooling load rates under steady 100°F ambient temperature with and without a solar cycle. Test 7 was made with the ambient temperature held steady at 90°F without a solar cycle for comparison with test 1 at 100°F ambient. Tests 4 and 5 were conducted with a solar cycle, and with the ambient temperature varied 20 deg F sinusoidally over a 24-hour period to yield daily mean temperatures of 100°F and 90°F, respectively. In Tests 3 and 6, the same ambient temperature variations were repeated, but the solar cycle was omitted. For all seven tests, the interior temperature was held at 0°F and the ambient humidity at 50% RH.

HOURLY INSOLATION ON CLOUDLESS DAYS
JUNE 21, LATITUDE 40° NORTH

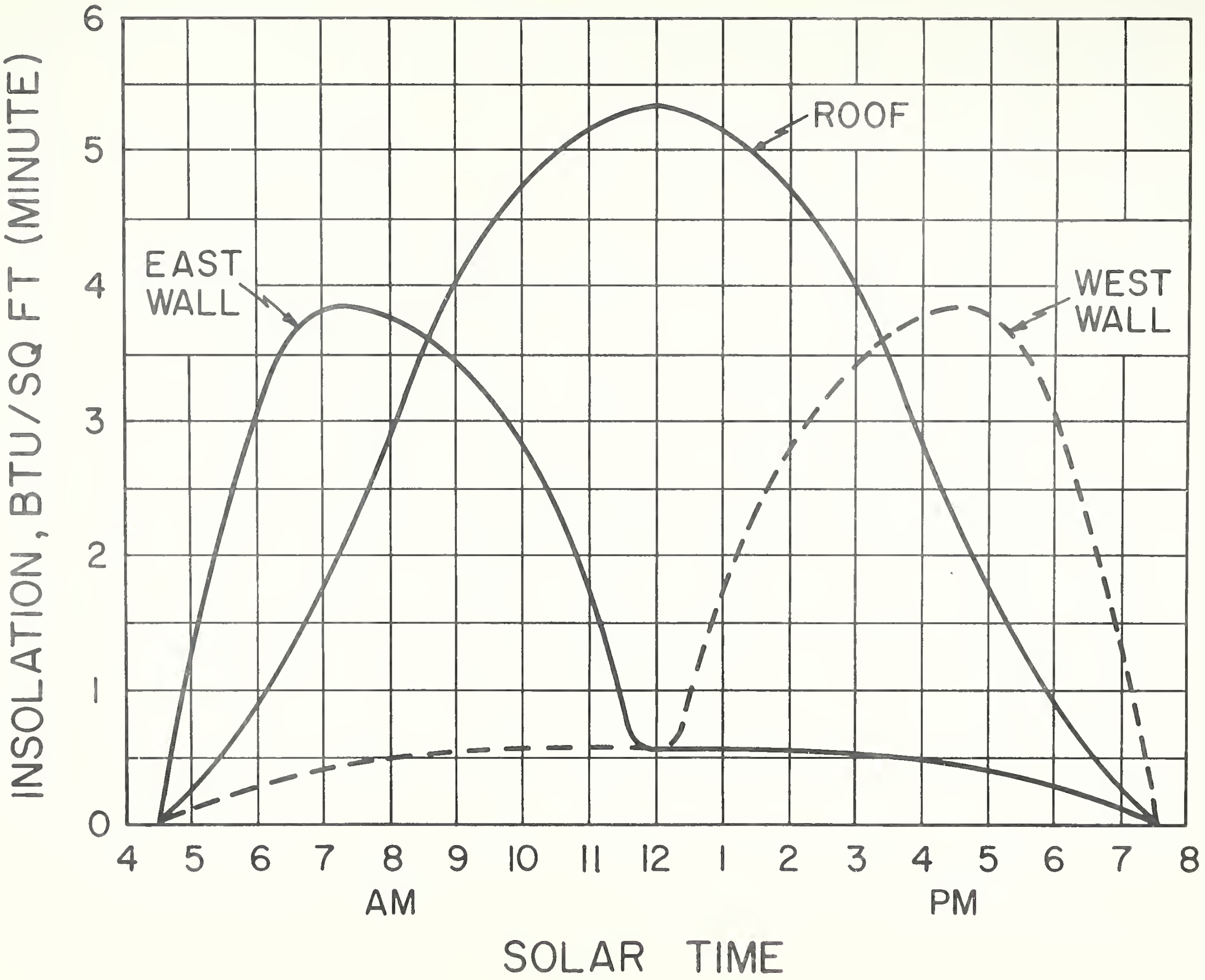

FIGURE 13

9.3 Static Pressure Air Leakage Test

In the static pressure air leakage tests, the interior of the trucks were pressurized to approximately 0.10 in. of water above atmosphere. The tests were made under isothermal conditions with both interior and exterior of the truck at room temperature. When the air flow necessary to maintain the desired pressure across the truck body walls was established, the pressure drop across the orifice in the 2-in. pipe was observed. This flow was verified by use of a pitot tube, as described under test apparatus. A test was made with the doors unsealed, and with the doors sealed with tape.

10. Test Results and Discussion

10.1 Steady-State Tests

The results of steady-state cooling load tests on the five vehicles are shown in Table 1.

Table 1. Cooling load due to air leakage and transmission at 100°F ambient temperature & 50% R.H. and 0°F interior temperature[a]

Truck	Cooling load, Btu/hr		
	Total[b]	Transmission	Air leakage
A	3700	2530	1170
B(0°)	3200	2180	1020
B(35°)	2150	1510	640
C	2550	> 2500	< 50
D	1800	1730	70
E	1850	1800	50

a Note exception of 35°F interior temperature run for truck B.
b Values rounded to nearest 50 Btu/hr.

Cooling loads ranged from 1800 to 3700 Btu/hr. The measured steady state total cooling load includes the transmission cooling load and the air leakage cooling load. No simulated solar heating was used in the steady state tests.

Comparison of the cooling loads determined by the metered heat sink method (primary method) and flow meter method (secondary, or check, method) for one of the test vehicles during a series of special test runs is shown in Figure 14.

The gain in weight of the vehicles during the tests was attributed to moisture deposition and was used to calculate the air leakage cooling load. Air leakage cooling loads ranged from 32 percent to less than 2 percent of the total cooling load of the truck bodies tested.

10.2 Weight Gain and Air Leakage

The rate of weight gain in pounds per hour, and the rate of air leakage in cubic feet per minute, for the five vehicles are given in Table 2.

Table 2. Air leakage rate and weight gain rate at standard conditions

Test vehicle	Weight gain lb/hr	Air leakage[a] cubic ft/min	Air leakage[b] cubic ft/min
A	0.47	5.8	4.7
B(0°F)	0.41	5.0	3.8
B(35°F)	0.29	4.2	-
C	0.014	0.2	-
D	0.028	0.3	-
E	0.022	0.3	0.39

a Calculated from weight gain.
b By helium-trace katharometer.

Air leakage computed from measured weight gain of each truck, on the basis of ambient air entering the body, depositing moisture, and leaving saturated at 0°F temperature is shown for all five test vehicles. Measured air leakage rates for three of the test vehicles using the modified helium-trace katharometer are also shown in Table 2. The deviation of these values from the calculated air leakage rates based on weight gain was approximately 20% and for trucks A and B, was consistent with the concept that the weight gain air leakage accounts for all air exchange, not just that which occurs to and from the cargo space. For truck E, the magnitude of the air leakage rates is believed to be too small to permit effective comparison of the two measurements.

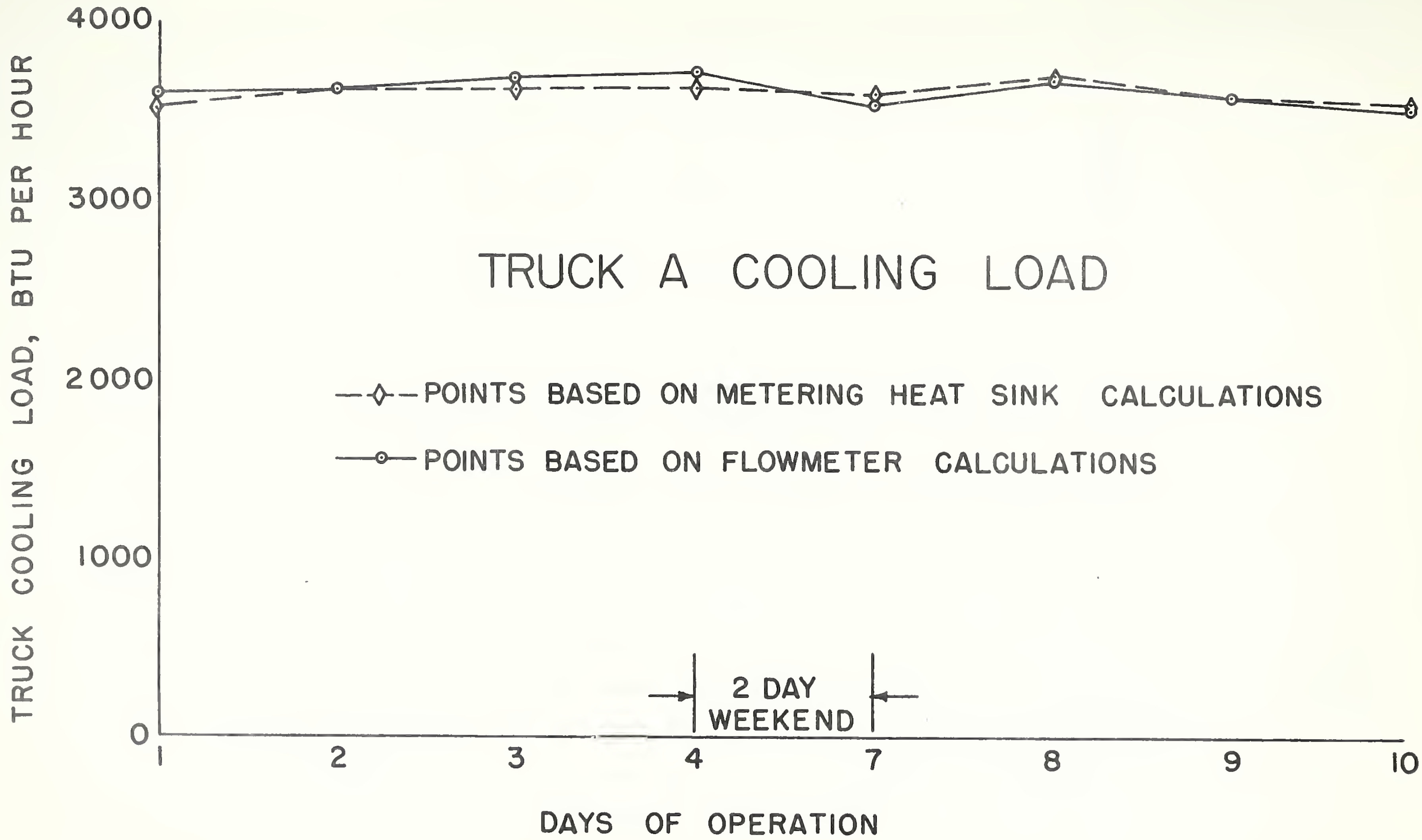
TRUCK A COOLING LOAD
POINTS BASED ON METERING HEAT SINK CALCULATIONS
POINTS BASED ON FLOWMETER CALCULATIONS
2 DAY WEEKEND
TRUCK COOLING LOAD, BTU PER HOUR
4000
3000
2000
1000
0
1
2
3
4
7
8
9
10
DAYS OF OPERATION

FIGURE 14

An air leakage test for selected static pressure difference between the cargo space and the test room was run on four of the five vehicles. Table 3 shows the air leakage rates for the test vehicles with door sealed and unsealed.

Table 3. Air leakage under static pressure

Truck	Pressure across walls	Leakage rate cubic feet per min	
	in. W. G.	Doors unsealed	Doors sealed
A	0.07	105	102
B	0.11	92	41
C	0.10	8	< 8
D	0.10	2	--
E	Not tested	--	--

Comparing the observed leakage rates for Trucks A and B with doors sealed and unsealed shows that the excessive leakage of Truck A was through the body, whereas for Truck B, the door seals accounted for more than half of the leakage. The contrast between the observed air leakage rates for trucks A and B and those for Trucks C and D in Table 3 indicates a wide range of effectiveness of various techniques for reducing air leakage.

Smoke tests of some of the bodies revealed leakage at the doors and door frame members, body seams, and door gaskets.

10.3 Simulated Solar Heating Tests

Figure 15 shows the exterior surface temperatures achieved in a typical solar simulation test. Note the shape similarity to curves in Figure 13, the insolation curve. The maximum 4-hour average observed cooling load was taken as the refrigeration requirement under solar loading. The percentage increase of the maximum 4-hour solar loads over the steady-state cooling loads are shown in Table 4.

Table 4. Percent increase in cooling load with simulated solar heat over steady state

Truck	Percent
A	24
B	Not tested
C	20
D	19
E	25

For the four trucks tested, the average percent increase was 22.4 with individual values ranging from 19.4 to 25.4 percent.

10.4 Comparative Tests at Variable and Steady Conditions

Seven tests were run on truck body E to explore and determine truck cooling loads with and without simulated solar heating when 1) the ambient temperature was held constant at either 100°F or 90°F and 2) the ambient temperature was varied to follow an approximately sinusoidal daily cycle having a range of 20 deg F with its maximum usually occurring 3.5 hours later than the solar noon and yielding mean ambient temperatures of either 100°F or 90°F. The results are given in Table 5, in which the cooling loads when not steady are the average loads over the 4-hour period of maximum load. Also shown in Table 5 are the differences or changes in cooling load made apparent when particular tests are compared.

Test pairs a, b and c in Table 5 show that the simulated solar cycle increased the cooling load about 450 Btu/hr, and that the increase was independent of the ambient temperature level and of the choice of a steady ambient temperature or the selected daily sinusoidal cycle of ambient temperature. Test pair d shows that reducing the steady ambient temperature from 100°F to 90°F (a reduction of 10 percent in the temperature difference) reduced the cooling load by 9 percent. Test pair e shows that the same reduction in average ambient temperature, but with a daily cyclic variation of 20 degrees in each case, reduced the cooling load about 11 percent. Test pair f shows that when the solar cycle was superimposed on the variable ambient temperature cycle, reducing the average ambient temperature from 100°F to 90°F decreased the cooling load 9 percent. Test pairs g and h show that the effect of the sinusoidally-varied ambient temperature was to increase the cooling load to a 4-hour maximum average value 180 Btu/hr greater than the corresponding steady ambient value at 100°F, with or without a simulated solar cycle. A smaller increase (120 Btu/hr) was indicated by test pair i, at 90°F mean ambient temperature.

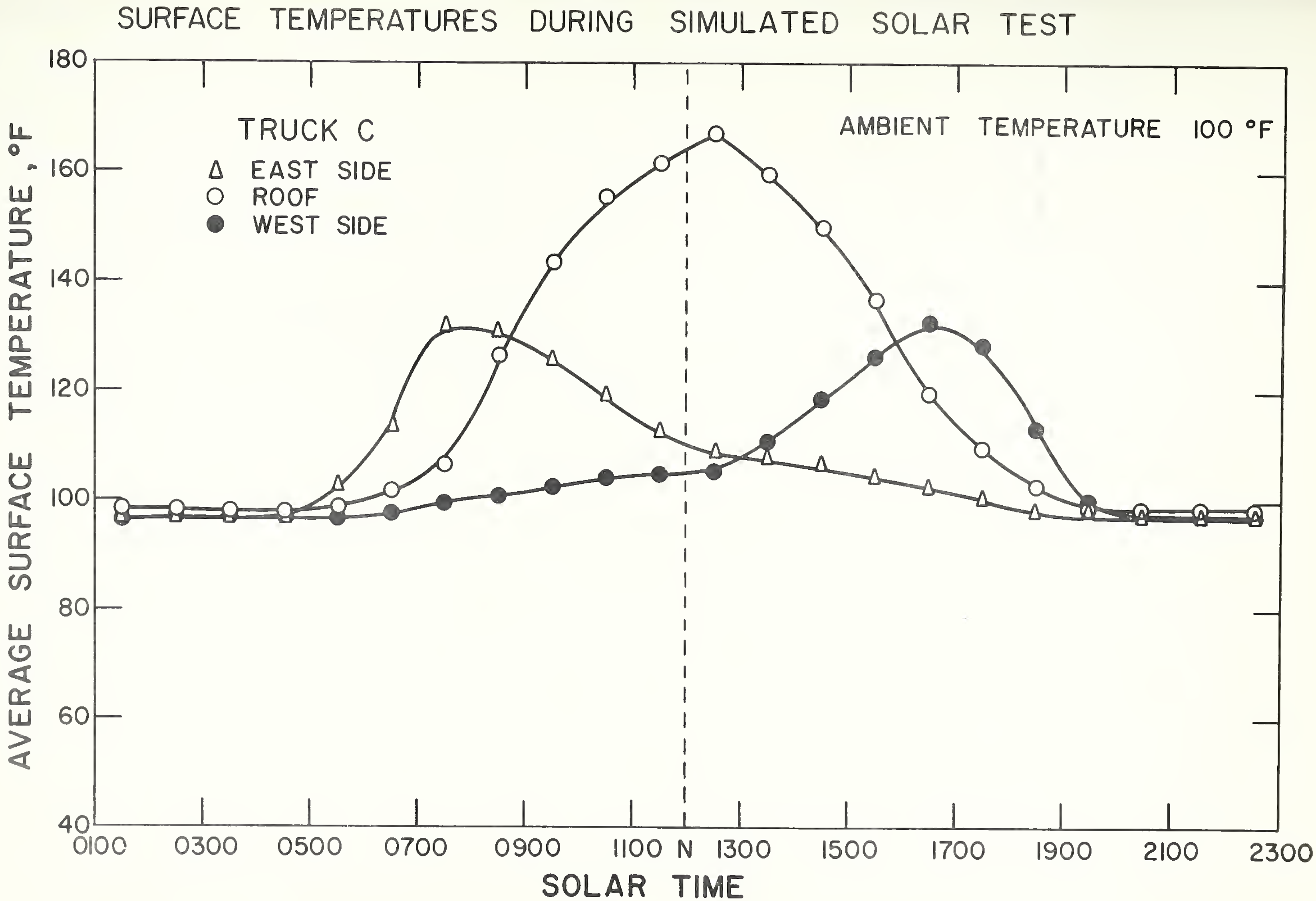
SURFACE TEMPERATURES DURING SIMULATED SOLAR TEST
TRUCK C
Δ EAST SIDE
○ ROOF
● WEST SIDE
AMBIENT TEMPERATURE 100 °F
AVERAGE SURFACE TEMPERATURE, °F
180
160
140
120
100
80
60
40
0100
0300
0500
0700
0900
1100 N
1300
1500
1700
1900
2100
2300
SOLAR TIME

FIGURE 15

Table 5. Cooling loads for truck body E with steady and variable ambient conditions, 0°F temperature inside, and with and without simulated solar heating

Test comparisons	Test no.	Ambient conditions	Mean temperature of ambient	Solar cycle	Cooling load Measured	Cooling load Difference
			Degrees F.		Btu per hour	Btu per hour
a	1	Steady	100	No	1,870	450
	2	Do	100	Yes	2,320	
b	3	Variable	100	No	2,050	450
	4	Do	100	Yes	2,500	
c	5	Variable	90	Yes	2,280	460
	6	Do	90	No	1,820	
d	1	Steady	100	No	1,870	170
	7	Do	90	No	1,700	
e	3	Variable	100	No	2,050	230
	6	Do	90	No	1,820	
f	4	Variable	100	Yes	2,500	220
	5	Do	90	Yes	2,280	
g	1	Steady	100	No	1,870	180
	3	Variable	100	No	2,050	
h	2	Steady	100	Yes	2,320	180
	4	Variable	100	Yes	2,500	
i	7	Steady	90	No	1,700	120
	6	Variable	90	No	1,820	
j	1	Steady	100	No	1,870	410
	5	Variable	90	Yes	2,280	

1/ Steady = ambient temperature same throughout the day. Variable = ambient temperature followed daily temperature cycle.

The cooling load in Test No. 1, with a steady ambient temperature of 100°F, was 102.7 percent of that observed in Test No. 6 with the maximum ambient temperature of 100°F during the daily cycle but with an average ambient temperature during the maximum four hours of 99.5°F. The temperature difference between the ambient and the truck interior for Test No. 1 was 100 degrees, 100.5 percent of the temperature difference of 99.5 degrees for the maximum four hours of Test No. 6. Comparing the ratios of cooling load and temperature differences for these two tests indicates that the heat capacity of the test truck body reduced the cooling load about 2 percent during the maximum four hours of the variable ambient test. This suggests that the steady-state test at 100°F ambient temperature is an acceptable substitute for the more complex variable ambient test procedure.

Test pair j indicates that Test No. 1, conducted with a steady 100°F ambient temperature and no solar cycle had a cooling load 410 Btu/hr less than the 4-hour maximum load of Test No. 5, conducted with a sinusoidally-varied ambient with a 90°F mean temperature and with a solar cycle. The latter condition is thought to be reasonably realistic for the solar and ambient exposure of an operating truck on a typical hot day, and the resulting 4-hour maximum cooling load observed is probably a good first approximation to the maximum cooling load of truck body E due to climatic factors alone. The actual in-use cooling load would, of course, be increased by product loads and by the additional load due to door openings which would depend on the truck service.

Since there was moderately good agreement as to the increase of cooling load due to a simulated solar cycle (approximately 22 percent) for all four trucks tested, as shown in Table 4, it is suggested that a fair approximation to the maximum cooling load of a truck due to climatic exposures alone can be estimated by multiplying the cooling load obtained in a test at a steady 100°F ambient, with no solar cycle, by a factor of 1.22 (i.e., 1 + 410/1870 for truck E). On this basis, the determination of comparable 4-hour maximum cooling loads for trucks can be effected by tests at steady 100°F ambient temperature, with no solar cycle, conditions the same as those called for in the procedure previously developed for the rating of refrigerated trailers.

Table 6 compares the solar test results with estimated values obtained by multiplying the steady-state cooling load by 1.22.

Table 6. Maximum[a] cooling load with solar heating; comparison of estimated vs. test results, Btu/hr

Truck	Estimated[b]	By test	Estimated / test
A	4510	4600	0.98
B	3900	Not tested	--
C	3110	3070	1.01
D	2200	2150	1.02
E	2260	2320	0.97

[a] Four-hour average
[b] 1.22 x steady state cooling load in Table 1

The deviation from unity of the ratio of estimated maximum cooling load to the maximum cooling load measured in test is no greater than 0.03. While no solar simulation test was run on a vehicle whose interior temperature was at 35°F, the data obtained in the 0°F interior temperature tests and a theoretical analysis of the effect of a 35° rise in interior temperature (see Appendix I) yields multiplication factors that can be applied and which include probable solar effects. These factors are: For a truck tested at 0°F interior temperature, and rated for 35°F interior temperature, a multiplication factor of 0.87 times the measured cooling load at 0°F. For a truck tested at and rated for 35°F interior temperature, a multiplying factor of 1.34 times the measured cooling load at 35°F. No other extrapolations are recommended.

11. Conclusions

There is a definite need to account for solar heat gain in any truck rating program. The test data shows that the solar gain is consistent enough to enable the use of a 100°F steady ambient temperature test and a multiplying factor in lieu of either an increased ambient temperature or an actual solar simulation test. The test results and theoretical analysis indicate that for a truck tested at and rated for 0°F interior temperature, a multiplier of 1.22 times the base cooling load can be used to incorporate the solar effect. For a truck tested at 0°F but rated for 35°F interior temperature, a multiplier of 0.87 times the base cooling load can be used, and for a truck tested at and rated for 35°F interior temperature, the multiplier is 1.34 times the basic cooling load at 35°F interior temperature.

It was the recommendation of the Project Steering Committee that the truck rating method be as consistent with that used for trilers [1] as possible. The ram air simulation plenum used on trailers would, of course, be omitted. By using a multiplying factor for solar heat gain, and by having air-cooling coils of different sizes to adapt to the wide range of cooling loads, a test facility used to rate trailers can rate trucks under this recommended technique. The different sized coils are recommended to improve accuracy by limiting the amount of "bucking", or control heat required to hold the desired truck interior air temperature. Because the percentage errors in measurement of the total cooling load absorbed by the air-cooling coil (and matched by the comparison heater) are directly reflected in the determination of the cooling load of the truck itself, the greater the ratio of the bucking heat to the total cooling load absorbed by the air-cooling coil, the less accurate the measurement of the cooling load of the truck. For this reason, it is recommended that in no case should the bucking heat be allowed to exceed 50 percent of the total cooling load absorbed by the air-cooling coil.

Experience gained in the tests at the NBS also pointed up the absolute necessity of very steady brine temperatures. Fluctuations in the temperature of the brine leaving the refrigerating unit carry through the entire system, and cause transients in measured cooling load. For this reason, the maximum cyclic variation of brine temperature leaving the refrigerating unit should not exceed 0.4 deg F during the rating period, for results consistent with the desired accuracy of the rating.

Some types of construction are tight enough so that air leakage and attendant weight gain may be below the accuracy of the weighing facilities and therefore be meaningless. For this reason, it is recommended in the proposed rating method that any observed weight gain less than the sensitivity requirement of an appropriate scale (or equivalent sensitivity of other weighing techniques), or less than the combined sensitivity requirements of the scales (if more than one is used), be shown in the data as being less than that sensitivity requirement.

Even though defrosting of the air cooling coil is not to be done during the final weight gain portion of the proposed rating test, collection and weighing of the condensate provides a means for determining the percentage of the observed total weight gain that is accumulated on the coil. To improve the accuracy of this determination, the air cooler, particularly the drain pan, should be designed to drain freely during the defrost operation.

It was found that static air pressure leakage tests, helium trace air leakage tests, and smoke bomb tests, do not relate directly to cooling loads. For this reason, it is not recommended that they be included in the proposed rating technique. These tests are useful in determining the relative effectiveness of air sealing techniques, and might possibly pinpoint major leakage problem areas, and for this reason might prove useful, as in-house tests for a manufacturer.

It should be emphasized that the proposed rating technique with the solar load multiplier does not give the total cooling load of a refrigerated truck and therefore cannot be used alone for determining the capacity required of the refrigerating unit. The service load caused by door openings and possible cargo cooling loads for some trucks in some types of service may be several times greater than the rated cooling load. Laboratory work has been done on measuring door opening cooling loads, and will be described in a forthcoming NBS-USDA publication. Information on cargo loads, as from a warm cargo or one which respirates, is available in various Department of Agriculture publications, and in the Guide and Data Book of the American Society of Heating, Refrigerating, and Air Conditioning Engineers.

References

[1] Phillips, C. W., W. F. Goddard, P. R. Achenbach, H. D. Johnson, and R. W. Penney, A Rating Method for Refrigerated Trailer Bodies Hauling Perishable Foods. U. S. Dept. of Agriculture, Agricultural Marketing Service, Marketing Research Report No. 433, Washington, D. C., 1960.

[2] Achenbach, P. R. and C. W. Phillips, Heat Sink Method for Measuring the Cooling Loads of Refrigerated Structures. Proceedings Xth International Congress of Refrigeration, International Institute of Refrigeration. 177, Boulevard Malesherbes, Paris (17e) France, Part 2 p 272-276, illus., 1959.

[3] Achenbach, P. R., C. W. Phillips, and R. W. Penney, Measuring the Cooling Load of Refrigerated Vehicles by Free Evaporation of Liquid Nitrogen. Supplement 1965-3 Bulletin of the International Institute of Regrigeration. 177, Boulevard Malesherbes, Paris France, p. 115-124.

[4] Altieri, Mario, A Method of Testing by Internal Cooling the Overall Heat Transfer of Insulated Vehicles, International Institute of Refrigeration. Proceedings of Commission 7 Meeting in Padua, Italy, 1961. 177, Boulevard Malesherbes, Paris France.

[5] Coblentz, Carl W. and P. R. Achenbach, Design and Performance of a Portable Infiltration Meter. American Society of Heating, Refrigerating and Air-Conditioning Engineers, ASHRAE Transactions, Volume 63, 1957.

[6] Achenbach, P. R. and C. W. Phillips, Laboratory Study of the Effect of Solar Radiation on Cooling Load of Stationary Refrigerated Vehicles, International Institute of Refrigeration, Proceedings of Commission 7, Munich, Germany, 1963. 177, Boulevard Malesherbes, Paris France.

Appendix I

Multiplier Factors for Extrapolation from 0°F to 35°F Interior Temperature

An equation for the total heat transfer under a periodic condition into a refrigerated vehicle parked in the sun can be expressed as follows: *

$$q' = U(Tem - t_i + E) \qquad (1)$$

where: q' = total heat transferred into the vehicle, Btu/hr

U = overall heat transfer coefficient of the vehicle, Btu/hr deg F

Tem = mean sol-air temperature (see below), °F

t_i = interior temperature of vehicle, °F

E = heat-sink characteristic factor (see below), °F.

The mean sol-air temperature is an effective temperature that relates the various modes in which the outside skin of the vehicle reacts to its thermal environment. It is expressed by:

$$Tem = t_o + \frac{\Phi a}{h_o}, \ °F$$

where: Tem = mean sol-air temperature, °F

t_o = exterior ambient temperature, °F

Φ = insolation to which the vehicle is subject, Btu/hr ft^2

a = surface absorptance factor

h_o = overall heat transfer coefficient between outside surface and ambient, Btu/hr deg F

Tem thus represents the energy level of the exterior of the vehicle, and is dependent only on the exterior surface and the ambient thermal environment, and is independent of the interior temperature.

The heat sink characteristic factor, E, incorporates the resistance to change in the heat flow through the structure,

* J. L. Threlkeld, Thermal Environmental Engineering, pp. 372-374, Prentice-Hall, 1962.

as when the increase in insolation increases the heat flow into the vehicle. It is dependent on the heat storage and transient heat transmission characteristics of the vehicle, which are in turn dependent on its construction. Thus it can be seen that for any given vehicle subject to a given ambient temperature and insolation for a specified time or time period, tem and E will be constant.

The steady-state heat transfer into a vehicle not subjected to insolation can be expressed by

$$q = U(t_o - t_i), \text{ Btu/hr}, \qquad (2)$$

where U, t_o and t_i are as previously defined.

The results of the comparative tests for any one truck, run at 0°F interior temperature and with and without simulated solar irradiation, can be expressed in the following equation:

$$\frac{q_i'}{q_i} = \frac{U(Tem + E - t_i')}{U(t_o - t_i)} \qquad (3)$$

in which the prime denotes values corresponding to the case of solar insolation, and the subscript i on q denotes the truck interior temperature.

In Table 4, observed values of the ratio q_i'/q_i for the individual trucks are given, for t_o = 100°F and $t_i = t_i'$ = 0°F, from which the value of (Tem + E) for each truck can be calculated, as follows:

Truck	q_o' / q_o	Tem + E, °F
A	1.24	124
B	Not tested	-
C	1.20	120
D	1.19	119
E	1.25	125
	Average	122

Assuming that for t_o = 100°F (Tem + E) is substantially independent of the truck interior temperature, t_i, and using for all of the trucks the average value 122°F, (3) becomes

$$q_i' / q_i = \frac{122 - t_i'}{100 - t_i} \qquad (4)$$

Using (4), it is possible to obtain immediately the multiplier factors for various conditions:

$$q_o' = \frac{122 - 0}{100 - 0} q_o = 1.22 \, q_o \tag{5}$$

$$q_{35}' = \frac{122 - 35}{100 - 0} q_o = 0.87 \, q_o \tag{6}$$

$$q_{35}' = \frac{122 - 35}{100 - 35} q_{35} = 1.34 \, q_{35} \tag{7}$$

Use of the factors given in (5), (6), and (7) will, when used with the standard method of determining q, give values of q' which may be used as ratings for comparing vehicles under conditions of insolation.

Appendix II

Recommended Standard Method for Testing and Rating the Cooling Load of Refrigerated Truck Bodies

1.0 <u>Purpose</u>

The purpose of this standard is to describe methods of testing and rating refrigerated truck bodies with respect to cooling load under selected standard interior and ambient conditions with adjustment for insolation effect, and with respect to weight gain rates.

1.1 <u>Scope</u>

This standard applies to refrigerated truck bodies (or containers) used for transporting frozen food or other materials requiring refrigeration. It describes a laboratory technique for measuring the cooling load under assumed typical ambient operating conditions, and at interior temperatures of 0°F and 35°F. The distinction "truck" is taken to mean a vehicle operating primarily on short haul delivery routes, and thus the standard takes into account the influence of solar loading, but does not assume ram air pressure on the front of the vehicle such as highway use would impose. Also omitted is the effect of door openings. Although significant, these fall under a service load category, and should not be included in a basic truck body rating technique.

The test method described can also be used to measure the cooling load and weight gain rate of a truck body at any time during its operating life to evaluate changes in performance.

2.0 <u>Basis for Rating</u>

2.1 <u>Ratings</u>

Results to be determined from the rating test shall consist of the cooling load in Btu/hr, and weight gain rate in lb/hr, all under specified conditions.

The average of two methods of simultaneously determining the cooling load shall be used for rating. Results of the two methods must agree within 5 percent.

Because the ratings will be in error if measurements are improperly made, or if conditions are not properly maintained, all instruments and readings shall meet the accuracy requirements of this standard.

2.2 Standard Rating Conditions

Tests to determine Standard Ratings of all truck bodies shall be measured under one or both of the following standard rating conditions (low or medium interior temperatures):

2.2.1 Cooling Load and Weight Gain Tests

Test room ambient air temperature		
Dry-bulb		100.0°F
Wet-bulb (at standard barometer)*		83.5°F
Truck interior air temperature		
Dry-bulb:	Low temperature	0.0°F
	Medium temperature	35.0°F

2.2.2 Deviations

Deviations allowed in test conditions from Standard Rating Conditions:

Reading	Maximum deviation of arithmetical average of all readings from standard conditions	Maximum deviation of individual readings
Test Room Air Temperature		
Dry-bulb	±1.0 deg F	±2.0 deg F
Wet-bulb	±1.0 deg F	±2.0 deg F
Truck Interior Air Temperature		
Dry-bulb	±0.5 deg F	±1.0 deg F

2.3 Standard Ratings

There shall be three allowable ratings that can be published under this standard. The first is the basic rating of 0°F interior temperature, 100°F and 50% RH ambient temperature and humidity. The second rating is for the same ambient conditions and a 35°F interior temperature, extrapolated as specified from a test run at 0°F interior temperature. The third rating is

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* For barometric variations from standard (29.92 in. Hg) of 1 in. Hg or more, the standard wet-bulb temperature shall be lowered 1°F for each in. Hg decrease in barometric pressure.

for 35°F interior temperature, at the same ambient conditions, with the test run at 35°F interior temperature. The published rating shall incorporate the specified multiplier to account for solar load (see 8.0) and shall state the interior temperature of the rating, and the interior at which the truck was tested.

3.0 Instruments

3.1 It is suggested that temperatures be measured by one of the following methods:

a. Thermocouple systems
b. Electric resistance thermometer systems

Accuracy of the measurements obtained with the system shall be within the following limits:

a.	Wet- and dry-bulb air temperatures	±0.4°F
b.	Brine temperatures	±0.4°F
c.	Brine temperature differences across coil in truck body and across external comparison heater	±0.05°F
d.	Other temperatures	±0.5°F

The smallest scale division of the temperature measuring instrument shall not exceed twice the specified accuracy.

The temperature measuring system used for measuring temperatures shall be calibrated, or monitored during the test, by comparison with a certified standard temperature measuring instrument calibrated in the appropriate temperature range.

Wet-bulb temperatures shall be read only under conditions which assure an air velocity of 1,000 ±250 ft/min over the wet-bulb, and only after sufficient time has been allowed for evaporative equilibrium to be attained. Care must be exercised in obtaining wet-bulb temperatures to use distilled water on the wick, and to have the wick damp at the time of observation. The wick must be kept clean.

Relative humidity measurements, if used, shall be made with sufficient accuracy to obtain compliance with the accuracy requirements for wet-bulb temperature as stated in paragraph 3.1 of this standard. Relationship of wet-bulb and dry-bulb temperatures to relative humidity shall be based on U. S. Weather Bureau tables.

Temperatures of brine in conduits shall be measured by inserting the temperature measuring element within a well inserted not less than 25 times the outside well diameter into the circuit. Instruments or systems used to measure the temperature differences of the brine across the cooling coil in the truck and across the comparison heater shall be compared with each other before they are installed and they shall agree within 0.05°F when immersed in the same baths at temperatures approximating those of use.

3.2 Brine flow shall be measured with an integrating liquid flowmeter having an accuracy within ±0.5 percent of the volume flow rate measured.

3.3 Electrical energy usage should be determined preferably with watt-hour meters. On steady loads, a wattmeter may be used in lieu of a watt-hour meter; and on steady resistance loads, an ammeter and voltmeter may be used.

Accuracy of instruments used to measure the electrical input to heaters in the truck or in the comparison heater shall be within ±1.0 percent of the quantity measured. Accuracy of instruments used to measure other electrical quantities shall be within ±2.0 percent of the quantity measured.

3.4 Instruments used to measure the change of weight of the truck being tested shall have a sensitivity requirement (see NBS Handbook 44) of 0.5 pound maximum under actual test loads.

4.0 Test Room

An insulated test room approximately 16 ft wide and 35 ft long and 14 ft high is required for testing trucks. A door at least 9 ft wide and 12 ft high is required at one end. The walls and ceiling of the test room should be insulated sufficiently to prevent condensation on the inner wall surface at standard test conditions during cold weather. A low heat loss for the test room makes it easier to keep the temperature uniform throughout the room. A good vapor barrier material should be applied at the inner wall surface or at the inner surface of the insulation. Separate rooms for the refrigerating equipment and instruments are desirable because of the high temperature and humidity maintained in the test room.

Distributed heating and humidity sources are desirable in the test room to provide uniform conditions around the test specimen with the minimum air motion. However, some mixing of the air with fans will probably be required to attain the specified uniformity. If fans are used, they shall be directed so that they do not blow air against the exterior surface of the truck at a velocity in excess of 400 ft/min. High air velocities around the truck affect the air leakage of the vehicle and also make precise weighing more difficult. Scales or other weighing mechanism may be portable or incorporated in the floor construction.

4.1 Cooling Load Test Apparatus

A diagram of the refrigerating equipment and temperature measurements required to determine the cooling load is shown in figure II-1. As indicated in the figure, the equipment consists of a refrigerating unit and brine chiller, a brine pump, an air-cooling coil and fan inside the truck, and a comparison heater and flowmeter in the brine circuit outside the truck or test room.

The refrigerating unit and brine chiller may be single stage or multistage, and it must have a capacity of not less than twice the cooling load of the largest test specimen, with brine leaving the chiller at about -25°F. Capacity control is required to adjust the cooling capacity to the cooling load of particular specimens. The refrigerating unit and its controls should be of a type that will produce a steady cooling effect during the test period. Cyclic variations in brine temperature entering the truck cooling coil or comparison brine heater should not exceed 0.4 deg F.

The cooling coil inside the truck shall be designed without fins or shall have fin spacing of 1/4-inch minimum to prevent rapid stoppage with frost or ice. Provision should be made for rapid defrosting of the coil. The heat transfer surface of the coil should be adequate to absorb the cooling load of the truck to be tested plus fan loads and a limited amount of controlling heat with a mean temperature difference between coil brine and truck air temperature of 20 deg F or less. The blower in the cooling unit shall deliver sufficient air to produce a temperature difference of 10 deg F or less between the air entering and leaving the coil.

The total electric heat input to the truck shall not exceed one-half the measured cooling load absorbed by the air-cooling coil. Because of this, a range of air cooling coil sizes may be necessary, also some means of varying both the capacity of the refrigerating unit and the brine flow rate will be needed.

The comparison heater outside the truck shall be insulated sufficiently to reduce the heat transmission from the surroundings to the brine to 1/2 percent or less of the electric heat input to the brine heater. The heating capacity of the comparison heater shall be approximately equal to the maximum total heat absorption of the air cooling coil in the truck, and the heater capacity should be adjustable. Voltage regulation shall be provided for the power supply to the comparison heater and to the heaters inside the truck that will prevent voltage fluctuations in excess of ±1 percent.

The brine pump shall be of a type that has an essentially flat volume versus pressure performance curve and a pressurized shaft seal or other means to minimize inward leakage of moist air. The capacity of the pump shall be such that the temperature rises of the brine through the cooling coil in the truck and through the comparison heater shall be about 8 deg F (not less than 6 deg F) each for the particular brine used. The brine piping circuit shall be designed to suit the head characteristics of the pump at the selected flow rate of the brine, and shall be insulated to reduce heat gain.

The brine shall have suitable toxicity, viscosity, and vapor pressure characteristics at temperatures ranging from room temperature to -30°F. Its density shall not vary more than 0.08 percent per degree F, and its specific heat shall not vary more than 0.02 percent per degree F, in the range of temperature used in the brine circuit. Methylene chloride meets the density and specific heat tolerances specified, and has most of the other desired characteristics, but other brines may be found that are equally satisfactory.

Electric heaters of a capacity slightly greater than the increments in refrigerating capacity should be installed either in the cooling coil or in the air discharge from the cooling coil, and should be controlled to maintain the required truck temperature. All electric power to fans, motors, heaters, etc., in the truck shall be measured, and the total shall not exceed 50 percent of the cooling load absorbed by the air cooling coil during any test.

Brine lines, power cables, instrument leads, etc., may be brought into the truck at any convenient point. Where no opening is available, it is recommended that a suitable sleeve be installed in one of the doors. These necessary lines must be flexible and must be supported in such a manner that their effect on the measured weight is minimal and constant throughout the test.

5.0 Cooling Load Test Procedure

Test methods incorporated in this standard are intended to produce heat transfer determinations accurate within ±5 percent of the quantity measured. To achieve this overall accuracy the test must be conducted in strict conformance with the limitations and methods outlined in the standard. When improved techniques and instruments are available, their use is encouraged; but they should be approved by the organization sponsoring this standard before being substituted for methods or instruments presently required.

The truck to be tested shall be placed on the weighing mechanism and the test equipment and measuring devices installed. After the truck body temperature and ambient conditions of temperature and humidity required for a rating test have been attained, they shall be maintained for not less than 48 hours. The rated cooling load shall be determined from the average of the data taken during the last 12 hours of the 48-hour test period, and the weight gain rate shall be determined from the measurements taken during the last 24 hours of the 48-hour period. No interruption of steady state conditions, such as defrosting the cooling coil, shall be allowed during the final 24 hours of a test.

The temperature difference of the brine across the cooling coil in the truck and across the comparison heater shall each be held at a constant value between 6 deg F and 10 deg F during the rating test.

The ambient dry-bulb air temperature shall be the average of the observations of not less than six stations, one approximately 1 foot from the center of each surface of the truck. The temperature difference between any two of these stations at a given time shall not exceed 3 deg F during the test period.

The ambient wet-bulb temperature shall be the average of not less than two points, one at the rear and the other at the front of the truck. The difference in wet-bulb temperature at these points of measurement at a given time shall not exceed 2 deg F during the test period.

The air temperature inside the truck shall be the average of the observations at 12 stations located as follows: four at the front, one in each corner suspended 6 inches from each adjacent surface; four similarly located at the rear; and four at the middle of the truck, one at each corner 6 inches from each adjacent surface. If desired each group of four temperature sensing elements may be connected in parallel and read as a single temperature, reducing the number of readings to three. If the twelve elements are read individually, no two readings at a given time may differ by more than 3 deg F; if the groups of four are used, no two readings at a given time may differ by more than 2 deg F.

During the portion of the test used to determine the cooling load rating, all observations shall be made at not more than 30-minute intervals.

Trucks equipped with removable plug-type refrigerating units in the front wall shall be tested for standard rating with the unit removed and the opening carefully closed with an airtight insulated plug.

All floor drains shall be plugged during the cooling load test.

The cooling coil may be mounted at any point in the truck. Care must be taken that air discharged from any fan does not blow directly on joints, cracks or seams of the interior surfaces.

The brine lines within the truck must be well insulated.

6.0 Data to be Recorded

The following items must be recorded:

	Item	Unit
1.	Date and time of test	
2.	Observer	
3.	Barometric pressure	in. Hg.
4.	Power input to comparison brine heater	watts
5.	Power input to heater in truck	watts
6.	Power input to fan motors, etc., in truck	watts
7.	Applied voltage to comparison brine heater	volts
8.	Applied voltage to heater in truck	volts
9.	Applied voltage to fan motors in truck	volts
10.	Electric current to comparison brine heater	amps
11.	Electric current to heater in truck	amps
12.	Electric current to fan motors in truck	amps
13.	Dry-bulb temperatures of air inside truck	°F
14.	Dry-bulb temperatures of air in test room	°F
15.	Wet-bulb temperature of air in test room	°F
16.	Temperature of brine at inlet of cooling coil	°F
17.	Temperature of brine at outlet of cooling coil	°F
18.	Temperature difference of brine in and out of truck	deg F
19.	Temperature of brine at inlet of comparison brine heater	°F
20.	Temperature of brine at outlet of comparison brine heater	°F
21.	Temperature difference of brine in and out of comparison brine heater	deg F
22.	Temperature of brine entering flowmeter	°F
23.	Brine flow rate	lb/hr
24.	Weight of truck, or change in weight	lb

7.0 Calculations of Observed Cooling Load

Two simultaneous methods are used to determine the cooling load. One method uses the comparison between the temperature rise of the brine in the truck and the temperature rise in the comparison brine heater; the other method uses the temperature rise of the brine in the truck and the mass flow rate of the brine as measured by the flowmeter. The results of the two methods must agree within 5 percent for a given test to be acceptable as a rating test.

Because both methods rely on the temperature rise of the brine in the truck, two separate sets of measuring elements shall be used to measure this brine temperature difference and must agree within 0.1 deg. F.

The cooling load measured by the comparison method shall be computed for the standard temperature difference of 100 degrees by the following equation:

$$\text{Cooling load, Btu/hr} = \left[\frac{\Delta T_1 (H_2)}{\Delta T_2} - (H_1)\right] \times \frac{100}{\Delta T_3}$$

where: ΔT_1 = Temperature rise of brine in the truck, deg F

ΔT_2 = Temperature rise of brine between inlet and outlet of brine heater, deg F

H_2 = Heat input to comparison brine heater, Btu/hr

ΔT_3 = Temperature difference between air in truck and air in test room, deg F

H_1 = Heat input to heater, fan motors, etc., inside the truck, Btu/hr.

The cooling load measured by the flowmeter method shall be computed for the standard temperature difference of 100 degrees by the following equation:

$$\text{Cooling load, Btu/hr} = \frac{(\Delta T_1 \times M \times C - H_1)}{\Delta T_3} \times 100$$

where: ΔT_1 = Temperature rise of brine in the truck, deg F

M = Brine flow rate, lb/hr

C = Specific heat of brine at mean temperature in the cooling coil, Btu/lb deg F

H_1 = Heat input to heater, fan motors, etc., inside the truck, Btu/hr

ΔT_3 = Temperature difference between air in truck and air in test room, deg F.

The observed cooling load shall be the average of the values determined by the two methods.

8.0 Standard Cooling Load Rating

The standard cooling load rating shall be the product of the observed cooling load and the appropriate multiplier to account for solar load, expressed to the nearest even 100 Btu/hr, e.g. 1200, 1600, 2200, etc.

Rated interior temperature	Test interior temperature	Multiplier
0°F	0°F	1.22
35°F	0°F	.87
35°F	35°F	1.34

9.0 Standard Weight Gain Rating

The standard weight gain rating is the average weight gain rate in lb/hr determined for the final 24 hours of the test, and shall be expressed to the nearest 0.1 lb/hr.

Weight gain rates less than 1/24 of the sensitivity requirement of the weighing system used shall be simply listed as less than that amount (e.g. weight gain rate = < y lb/hr, where y = sensitivity requirement).

10.0 Published Ratings

Published ratings, in order to conform to this standard, shall be identified as follows: "(sponsor's designation) Standard Cooling Load Rating (test result) Btu/hr, Standard Weight Gain Rating (test result) lb/hr, Rated at (0°F) (35°F) interior temperature, tested at (0°F) (35°F) interior temperature. Tests conducted in accordance with (sponsor's designation) Standard Method of Testing and Rating the Cooling Load of Refrigerated Trucks."

The terms "(sponsor's designation) Standard Method" or "(sponsor's designation) Standard Conditions" shall not be used in connection with published ratings unless such ratings have been determined in accordance with this standard.

DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS