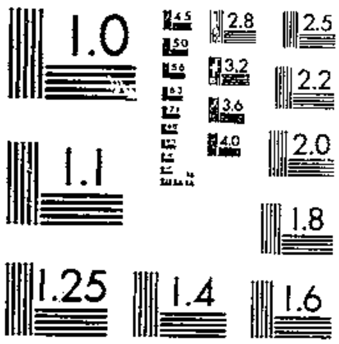
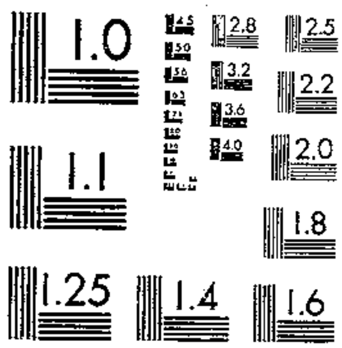
TB 1376 (1967) USDA TECHNICAL BULLETINS UPDATA
DEVELOPMENT OF A METHOD FOR TESTING AND RATING REFRIGERATED TRUCK BODIES
PHILLIPS, C.W., PENNEY, R.W. 1 OF 1

START



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

PREFACE

This bulletin describes a method for measuring the heat and moisture transfer characteristics of refrigerated delivery truck bodies. The Transportation and Facilities Research Division of the U. S. Department of Agriculture, the Truck Body and Equipment Association, and the National Bureau of Standards sponsored a cooperative program to develop a suitable method for rating refrigerated trucks used for transport of perishable and frozen foods. The project was carried out at the National Bureau of Standards under the direction of the Environmental Engineering Section, with technical assistance provided by the U. S. Department of Agriculture. General supervision for the Department of Agriculture was furnished by Harold D. Johnson, Transportation and Facilities Research Division, Agricultural Research Service, now retired. The five vehicles used as test specimens for this study were furnished by the following manufacturers:

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

The drawings and illustrations in this report were prepared by the National Bureau of Standards.

CONTENTS

Development Of A Method For Testing And Rating Refrigerated Truck Bodies

By C. W. PHILLIPS
Building Research Division
Institute of Applied Technology
National Bureau of Standards

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

SUMMARY

A recommended method for testing and rating refrigerated truck bodies in terms of cooling load similar to the method previously developed for trailer bodies is described in this bulletin. The method is based upon the metered heat sink principle with test conditions of 0° F. or 35° F. temperature in the truck body and 100° F. temperature and 50 percent humidity in the test room.

Test facilities needed to rate a truck include an insulated test room of sufficient size to accommodate a truck; a refrigeration system consisting of a brine cooler, closed brine system, brine pump, cooling coil, comparison heater, and associated piping and control instruments; weight measurement equipment; and temperature and humidity control and indicating instruments.

In developing the recommended method, we tested five truck bodies of current design and construction. Cooling loads of these five bodies, including simulated solar load, ranged from 2,150 to 4,600 B.t.u. per hour. Air leakage was found to account for 2 percent to 32 percent of the total steady-state cooling load.

A bank of heaters was positioned on both sides and at the ceiling of the test room to simulate the solar heat load. The heaters consisted of a series of electric resistance-wire coils wound on rods mounted in parabolic reflectors. Voltage to the heaters was programed to duplicate the radiation effect of the sun received by the truck body surfaces in the course of a typical cloudless day. Test results showed that the total cooling load can be suitably approximated by multiplying the total steady-state cooling load by a factor to include solar heat load. Therefore, it is not necessary to have a solar simulation apparatus in a truck rating facility.

INTRODUCTION

Trucks used for local delivery of perishable foods, in addition to performing a transport function, also must provide proper temperature for the product. The capacity of the refrigeration unit must be sufficient to handle the cooling load caused by heat transmission and air and moisture infiltration into the truck body and the cooling load caused by air exchange when the door is opened during unloading at delivery stops. Rating methods for refrigeration units [1] and a technique for determining door-opening cooling loads are available (*6*).[2] This study was conducted to develop a rating procedure for truck bodies similar to the one now used for refrigerated trailer bodies (*7*).

Highway trailers operate day and night over long distances and are opened for loading and unloading only at terminals or major transfer points. The cooling load caused by solar radiation when the trailer is stationary may be less than the cooling load caused by air leakage when the vehicle is traveling at road speeds. Thus, the rating method for trailers contains means for simulating the effects of frontal air pressure on the vehicle but not the effect of solar radiation.

Most delivery trucks are stationary much of the time and operate primarily during daylight hours. Therefore, the rating method for truck bodies described in this bulletin takes into account cooling loads from solar radiation but not the effects of air leakage at road speeds. Because of the wide variety of door-opening schedules in use, it was not appropriate to include effects of opening the door as part of the basic rating method. Information on the effect of door opening on cooling loads is contained in a separate bulletin (*6*). Both the trailer rating method and the truck rating method require determination of weight gain and cooling load caused by air leakage into the vehicle body with doors closed.

METHODOLOGY

Cooling-Load Measurement

Several methods were considered for measuring the cooling load of the truck body. One method used liquid nitrogen as the refrigerant. Several tests were conducted in which cooling loads were measured by the consumption rate of liquid nitrogen. Test results showed that air leakage into the body was reduced because of the slight increase in pressure caused by the expansion of nitrogen gas in the cargo space. This reduction in air leakage, in turn, caused lower cooling-load values than those obtained by using conventional cooling coils, particularly for bodies with high air-leakage rates (*2*).

Other methods considered were transient-state cooling, reverse heat flow, and the metered heat sink. The transient-state cooling method and the reverse heat-flow method were not selected because they do not provide a measurement of the rate of weight gain due to moisture. The metered heat sink was chosen as the basic test method because of its success in rating trailer bodies (*1*). Also, present test facilities for trailers, without extensive

[1] Air-Conditioning and Refrigeration Institute Standard 1110–64, for Speed-Governed Transport Refrigeration Units Employing Forced-Circulation Air-Coolers, and Standard 1120–61 for Variable-Speed Transport Refrigeration Units Employing Forced-Circulation Air-Coolers.

[2] Italic numbers in parentheses refer to Literature Cited, p. 25.

changes in equipment, could be utilized for both trucks and trailers. For consistency with conventional practice, standard test conditions were selected as either 0° F. or 35° F. temperature inside the truck body with an ambient temperature of 100° F. and 50 percent relative humidity.

Metered Heat Sink

Figure 1 is a schematic diagram of the metered heat sink system used to measure cooling loads. The main components of the system are a cooling coil inside the truck body and a comparison heater, brine flowmeter, and refrigeration unit outside the truck body.

The steady state heat flow, sensible and latent, q_t, into the truck from the test chamber as absorbed by the cooling coil in the truck is

$$q_t = c_p M \Delta t_t - h \qquad (1)$$

where

c_p = specific heat of brine, B.t.u. per lb.-deg. F.
M = mass flow rate of brine, lb. per hr.
Δt_t = brine temperature change in the cooling coil, deg. F.
h = total heat input to the truck from fans, heaters, etc., B.t.u. per hr.

The heat input to the brine from the electrical comparison heater is

$$q_c = c_p M \Delta t_c \qquad (2)$$

where

Δt_c = brine temperature change in the comparison heater, deg. F.

In equations (1) and (2) M will be equal at any given time in the closed liquid brine system, and the change in c_p of the methylene chloride brine used between the mean coil temperature and the mean comparison heater temperature is less than 0.5 percent. Solving for $c_p M$ in equations (1) and (2) yields

$$c_p M = \frac{q_t + h}{\Delta t_t} = \frac{q_c}{\Delta t_c} \qquad (3)$$

or

$$q_t = q_c \frac{\Delta t_t}{\Delta t_c} - h \qquad (4)$$

To provide a comparison, q_t was also determined from the basic formula, $q_t = c_p M \Delta t_t - h$, using the mass flow rate of the brine measured by an electronic integrating flowmeter, and a value for c_p taken from the literature.

The principal advantage of the metered heat sink system is the elimination from the heat balance computations of the mass flow rate, M, and the specific heat, c_p.

Test Facilities

The test room was 25 feet long, 17 feet wide, and 12 feet high with double doors at one end to permit entry of trucks. Ambient air was maintained at 100° F. temperature and 50 percent relative humidity during tests. Heating, cooling, humidification, and air circulation in the room were provided by two fan coil units, which discharged air into a plenum above a perforated ceiling

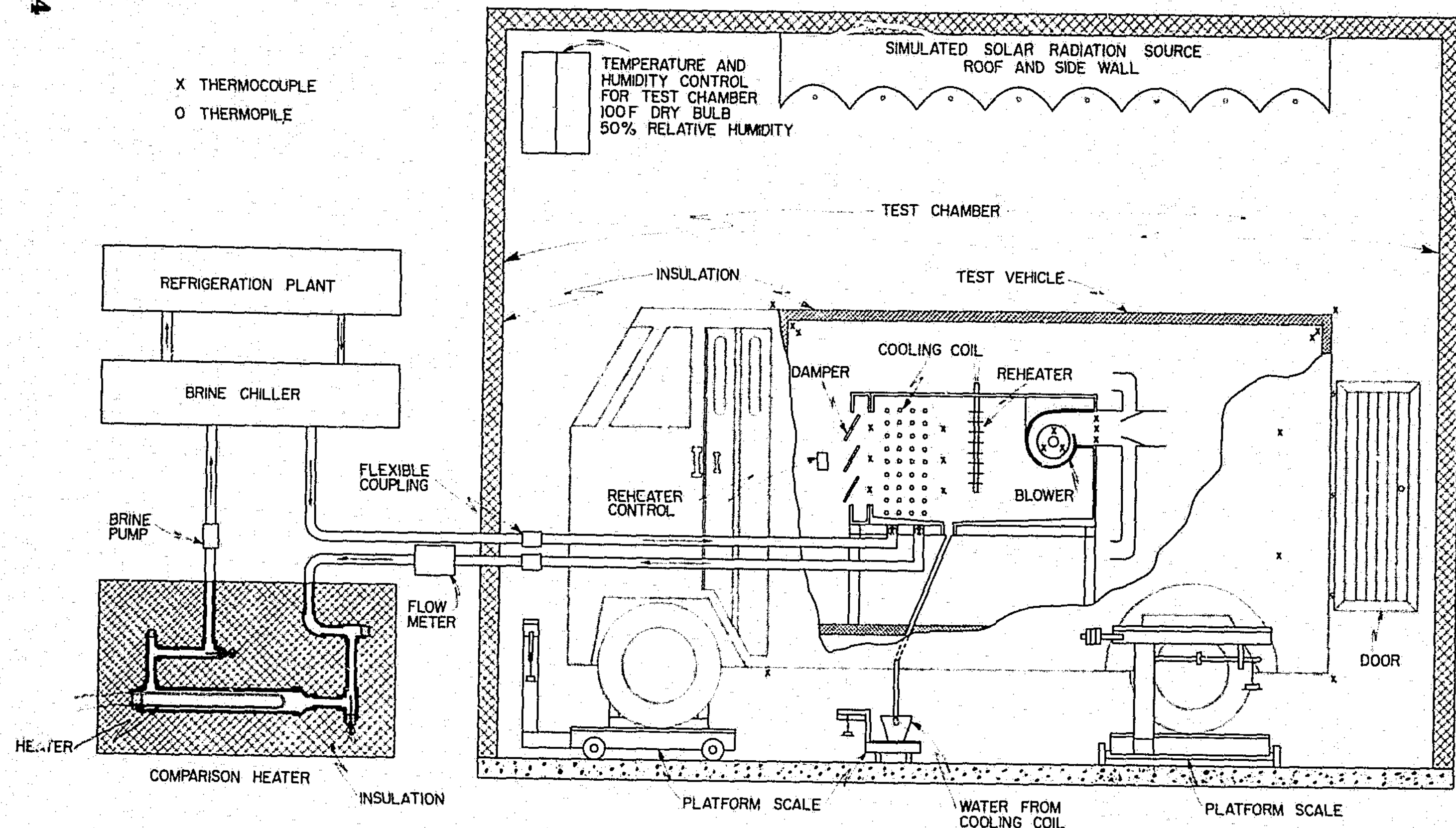


FIGURE 1.—Schematic diagram of metered heat sink system used to measure cooling loads of truck bodies.

with return directly to the fans. Three portable platform scales, each of 6,000 pounds capacity, were used to measure changes in vehicle weight caused by moisture deposited from air leakage into the body. Each rear wheel and the front axle of the truck rested on a scale. Electric heaters with parabolic reflectors at the sides and ceiling of the room were used to simulate solar heat load (fig. 2).

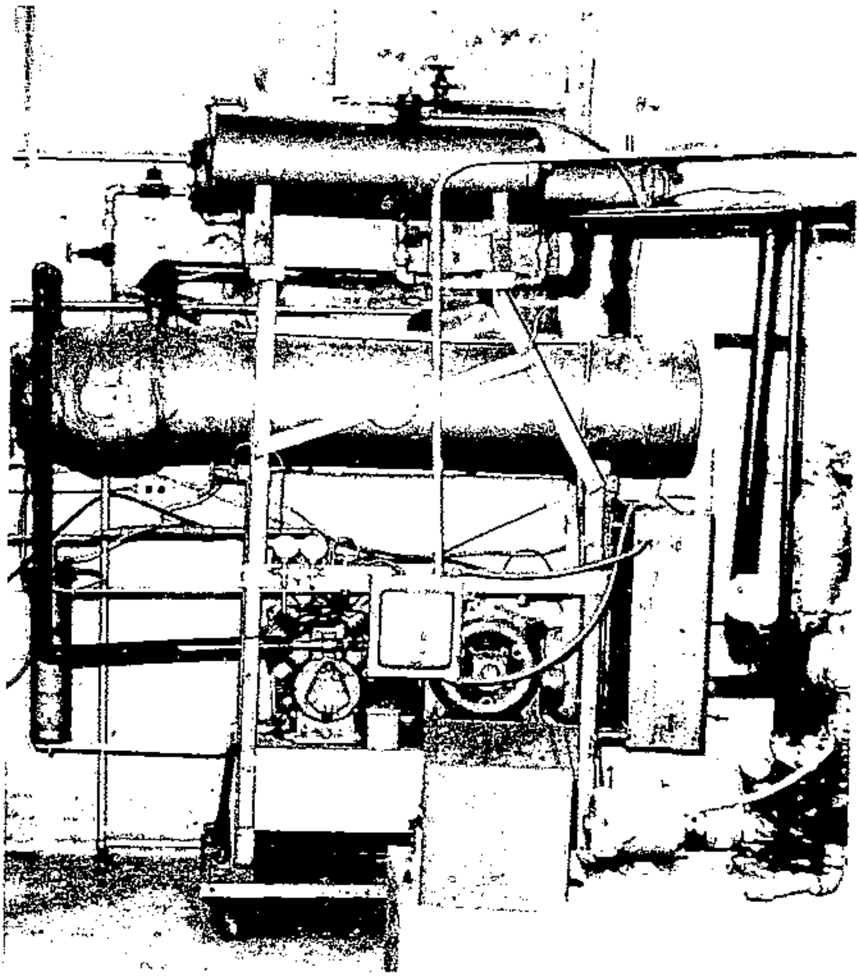
The inside of each test vehicle was maintained at 0° F. temperature (35° F. for one test) by a refrigeration system consisting of a brine cooler, turbine-type brine pump, air-cooling coil, and associated piping and valves. The brine chiller operated at

NBS 30493-1

FIGURE 2.—View of test room showing parabolic heat reflectors and two of the three weighing scales.

two speeds for better control of cooling load over the wide range required. The primary refrigerant was R22 (monochlorodifluoromethane) and the secondary refrigerant, or brine, was R30 (methylene chloride). At the mean cooling-coil temperature used in the tests, the specific heat of the brine was approximately 0.273 B.t.u. per lb.-deg. F. The turbine-type pump, rated at 8 g.p.m. (water) at a 40 p.s.i. head was driven by a 1/2-hp. electric motor, and circulated brine through the secondary refrigerant circuit. The pump was located in the return line just before the brine chiller (figs. 1 and 3).

The brine circuit was heavily insulated and vapor sealed to minimize frost and heat gain. To minimize forces on the truck body exerted by the brine lines, which would interfere with proper weighing of the vehicle, two flexible lines were used where the brine pipes entered the truck body. Thermocouple wells were



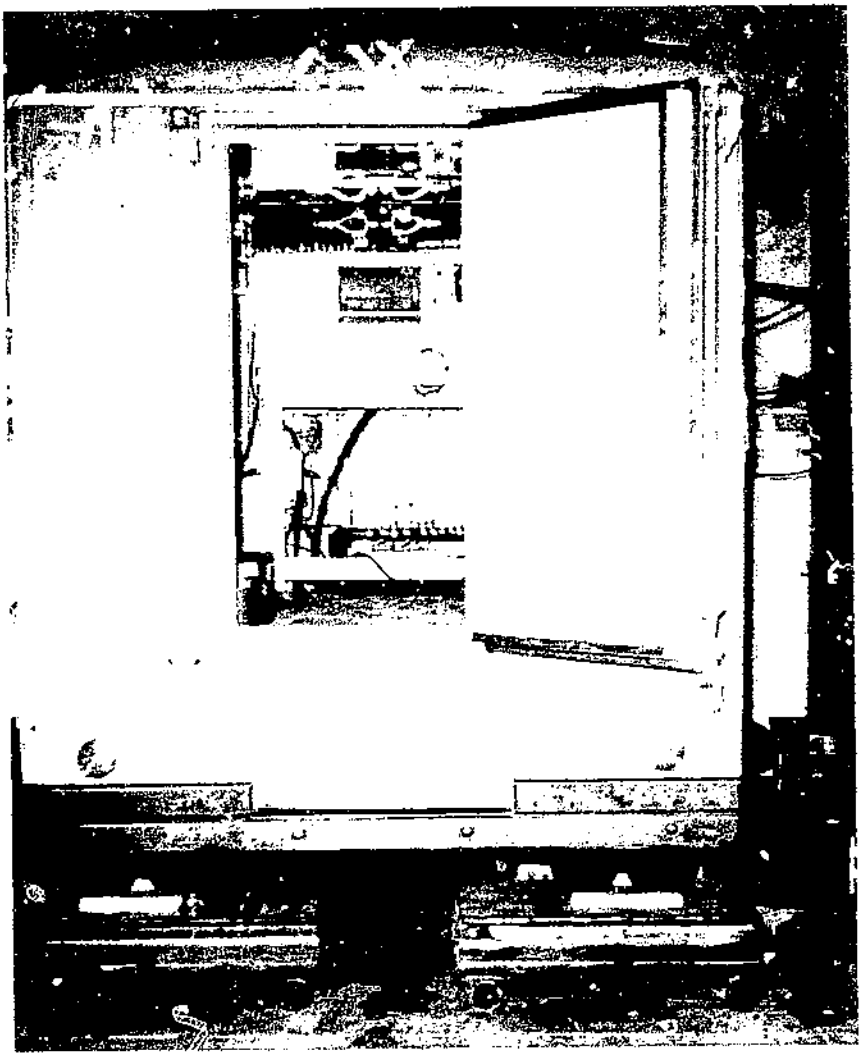
NBS 30364 7

FIGURE 3.—View of the brine chiller, brine pump, and comparison heater used in tests.

located in the brine line to measure the temperature of the brine as it entered and left the truck body.

The air-cooling assembly placed in the truck body contained a coil, blower, damper, and electric heater. The coil was constructed with four fins per inch to permit operation over a long period without defrosting. Three electric resistance heaters were mounted between the coil and blower to control air temperature. At the coil inlet a damper controlled airflow, and at the blower outlet a baffle distributed air throughout the truck body interior (figs. 1 and 4).

The cooling coil was defrosted automatically in less than 10 minutes by a piping system that circulated the brine through the



NBS 30364-1

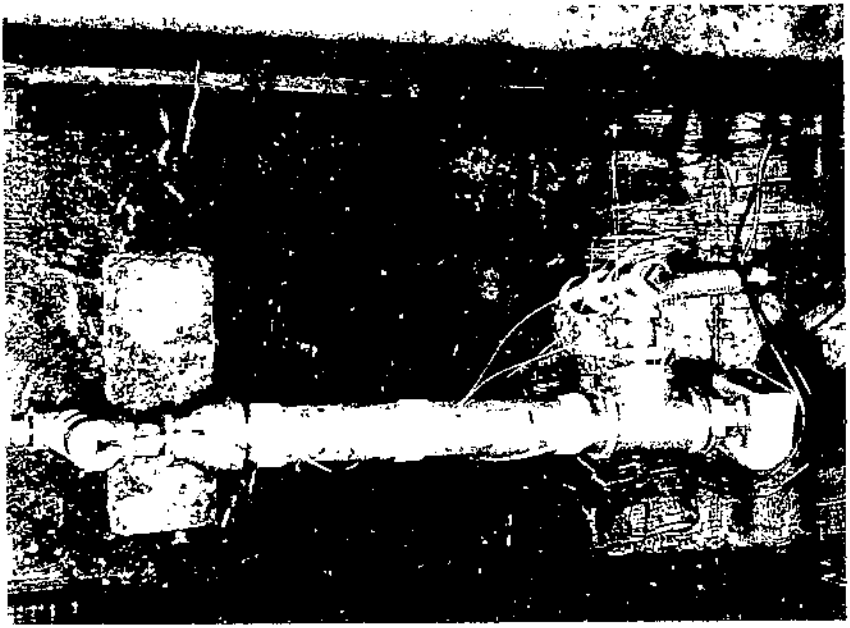
FIGURE 4.—Air-cooling assembly for cooling the truck body interior during tests.

cooling coil and comparison heater while bypassing the chiller. The brine remaining in the brine chiller was subcooled while the chiller operated at low speed. At the end of the defrosting period, the subcooled brine was circulated through the air-cooling coil to quickly restore test temperature in the truck.

It was necessary to collect and weigh the defrost water from the cooling coil so that this weight could be included when computing total moisture gain. To accomplish this, a rubber hose was connected to the defrost pan and emptied the water into a container placed in the test room, as shown in figure 1. A sheathed electric cable heater was placed in the defrost pan and in the rubber hose to prevent freezing. A water trap on the end of the hose prevented air flowing through it.

The comparison heater consisted of special piping enclosed in a plywood box about 40 by 20 by 20 inches, insulated with loose cork fill (fig. 5). The box was vapor sealed so that moisture would not accumulate in the insulation. Thermocouple wells were installed at both ends of a section of piping containing an electric resistance heater and were located to provide thorough mixing of the brine at points of temperature measurement. Piping within the box was installed so that vapor would not collect around points of temperature measurement. A surface thermostat, located on the pipe containing the heater element, protected the heater against damage from overheating. Energy input to the brine by the heater was measured by a calibrated integrating watt-hour meter on the control panel.

Temperatures were measured with calibrated copper-constantan thermocouples, a 16-point electronic recording potentiometer, an electronic indicating potentiometer, and a precision-grade laboratory potentiometer readable to 1 microvolt. 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 manually read on the electronic indicating potentiometer and 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 truck body and entering and leaving the comparison heater. 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 also were placed in the wells to determine the mean brine temperature and to provide a check on the thermopiles. The interior air temperature in the truck body was measured with 12 individual thermocouples: one each suspended 6 inches from each corner at the front and rear of the truck body and one each at the midlength of the truck at the intersection of the walls with the floor and ceiling. Three sets of four thermocouples electrically connected in parallel were also used to read average interior temperatures at the front, middle, and back of the truck body. Test room ambient temperatures were measured with both individual and parallel-connected thermocouples suspended 6 inches from the four exterior corners at each end of the truck body. Test room humidity was measured with electric hygrometer elements at the front and rear of the



NBS 42103 5

FIGURE 5. Comparison heater used in the cooling-load tests.

test vehicle or in the return air circuit to the two room fan coil units. Periodic measurements of the test room humidity were made with a motorized psychrometer as a check on the electric hygrometers. Other thermocouples were placed on the interior and exterior skins, in primary refrigerant lines, and in the air entering and leaving the cooling-coil assembly.

Air-Leakage Measurement

Air leakage into the truck body was evaluated by three methods. In one method, air leakage was calculated from determination of weight gained by the truck body through moisture accumulation under specified test conditions (*1*). This method takes into account (1) the air exchanged directly between the exterior and the cargo space through openings such as door and unit gasket seals, (2) the air exchanged between the exterior and the cargo space through the insulated walls, and (3) the air exchanged between the exterior and spaces within the insulated walls. In calculating the air-leakage rate, it was assumed that air entered the vehicle (either cargo space or walls) at ambient conditions and left saturated at the temperature inside the vehicle. This assumption gives a minimum calculated air-leakage rate with corresponding latent and sensible cooling loads.

In the second method, a helium-trace katharometer (*3*) was used to measure the amount of air exchanged by leakage in the cargo space under standard test conditions.

In the third method, a static pressure of 0.1-inch water gage above ambient pressure was maintained inside the body, with the same temperatures inside and outside the body. The pressure difference was produced by a centrifugal blower connected to the truck interior, with an orifice in the connecting 2-inch pipeline. The pressure drop across the orifice was used to calculate the airflow rate. A manometer was used to measure the pressure difference between the interior and exterior of the truck. The orifice measurements were checked by pitot tube sweeps across the inlet air pipe in the truck.

The static-pressure air-leakage test measures the flow of air between the cargo space and the exterior caused by a small pressure difference between the interior and exterior. The helium-trace air-leakage test, at standard test conditions, measures the rate of exchange of air between the cargo space and the exterior (both direct and through the insulated walls) caused by the difference in air density between interior and exterior. Neither of these two tests indicates the air interchange that takes place between the exterior and the insulation spaces between the cargo space liner and the outer skin of the body. However, the latter air exchange does contribute to the total cooling load of the truck body, directly and by impairing the value of the insulation through accumulation of moisture or frost.

A truck body having either a well-sealed interior liner, or a well-sealed exterior skin, or both, will show a low air-leakage rate by either the static-pressure or helium-trace test. However, if the interior liner is well sealed, but the exterior skin is poorly sealed (contrary to good practice), and the insulation is permeable to air, there may be a significant addition to the cooling load of the vehicle that neither the static-pressure nor helium-trace test would indicate. By making a cooling-load test of the body for a sufficient period of time, however, and observing the weight increase of the body through accumulated moisture (including condensation weight gain on the cooling coil), it is possible to calculate the approximate air-leakage cooling load.

Solar-Load Measurement

Of the radiant energy absorbed by a surface covering insulation, such as the exterior of an insulated truck body, only a small percentage of the energy is conducted through the insulation. Most of the energy is re-radiated to surroundings at lower temperatures or given up to convective air movement at the surface. Bright or polished metallic surfaces absorb less solar energy than dark painted surfaces but, in turn, are less able to re-radiate the energy which they do absorb, thus the amount of energy transmitted through the insulated surface is about equal. As each surface is heated above ambient air temperature, it loses additional heat to convective air currents depending upon the difference in temperature between the surface and the air.

In this study 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 truck bodies in the test series. The low temperature, long wavelength radiation source, used for the simulated solar tests, caused the test vehicles to absorb radiation 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 best for minimizing the heat gain from solar exposure.

Solar-Simulation Apparatus

Previous observations of trucks and trailers exposed to bright summer sunlight indicated roof skin temperatures approaching 70 deg. F. above the ambient temperature; therefore, a value between 65 deg. F. and 70 deg. F. above ambient temperature was selected as an approximate upper limit for roof skin temperature for the simulated solar tests. It was reasoned that the maximum amount of solar energy would impinge on an insulated vehicle when it was parked with its longitudinal axis in a north-south direction so that both 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 patterns over a vehicle was discarded in favor of adjustable stationary banks of electric heating elements and parabolic reflectors facing the two long sides and the top of a test vehicle. To simulate the effect of the changing angle of solar incidence during the day, the voltage applied to the heaters was systematically varied. Each of the three banks was 15 feet long and 9 feet wide. Each consisted of 45 parabolic reflectors, 1 foot by 3 feet at the bank face. 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 176 watts per square foot of projected reflector area. The total heat release of one bank of 45 heaters operating at maximum design voltage was approximately 83,000 B.t.u. per hour.

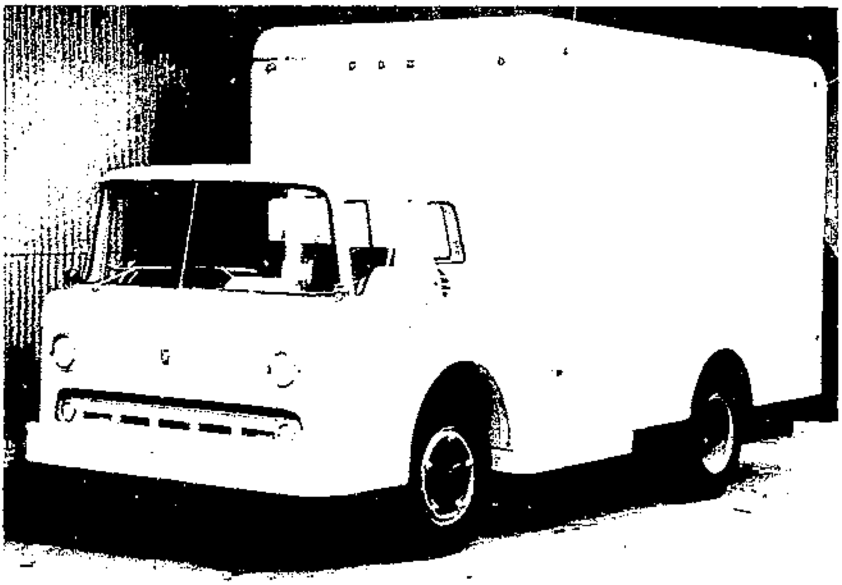
TEST VEHICLES

Five insulated truck bodies were used in this study and are described briefly in this section. The order of listing, such as body 1, 2, etc., does not coincide with the order of listing in the test results section, given as body A, B, etc.

Body 1 was a walk-in ice cream delivery design with exterior dimensions 183 inches long by 91 inches wide by 87 inches high (fig. 6). Internal volume was 544 cubic feet. Access consisted of a single door at the rear and a pass door on the right side forward of the wheel well. The metal exterior, including the roof, was painted a gloss white. The metal interior walls were smooth, except for anchors to which refrigerated plates could be attached on the ceiling and forward wall. Insulation consisted of 7 inches of expanded polystyrene plus 2 inches of glass fiber in the roof, 7 inches of expanded polystyrene in the walls, and 6 inches of expanded polystyrene plus 1 inch of corkboard in the floor. Preformed polystyrene board was used.

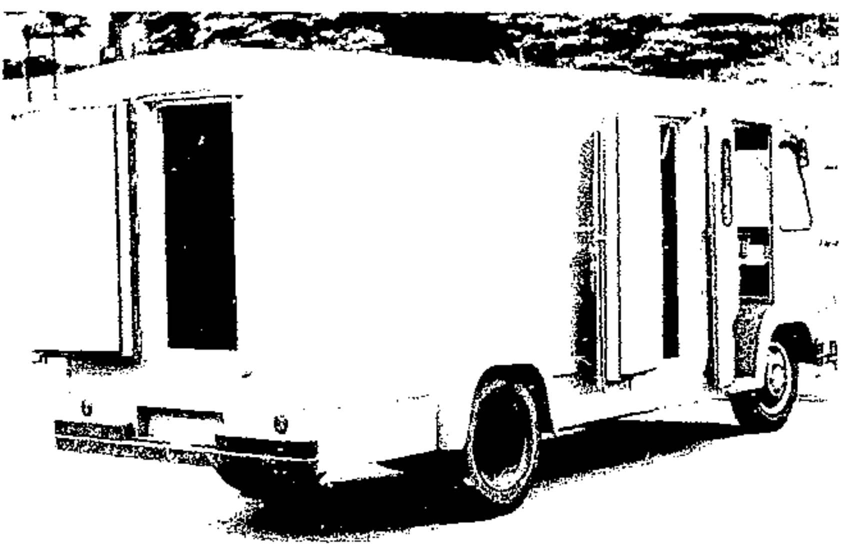
Body 2 was an integral cab design with a door entering from the cab, a door on the curb side, and a single rear door (fig. 7).

The cargo body external dimensions were 176½ inches long by 80½ inches wide by 75½ inches high. Internal volume was about 388 cubic feet. The metal exterior surface was painted a dark color with a nonglare finish. Interior surfaces were corrugated metal on the walls, and a sheet metal floor and ceiling. The insulation consisted of 6 inches of glass fiber in the walls, 8 inches of glass fiber in the roof, and 7 inches of expanded polystyrene board in the floor.



NBS 42103-6

FIGURE 6.—Test vehicle, body 1.



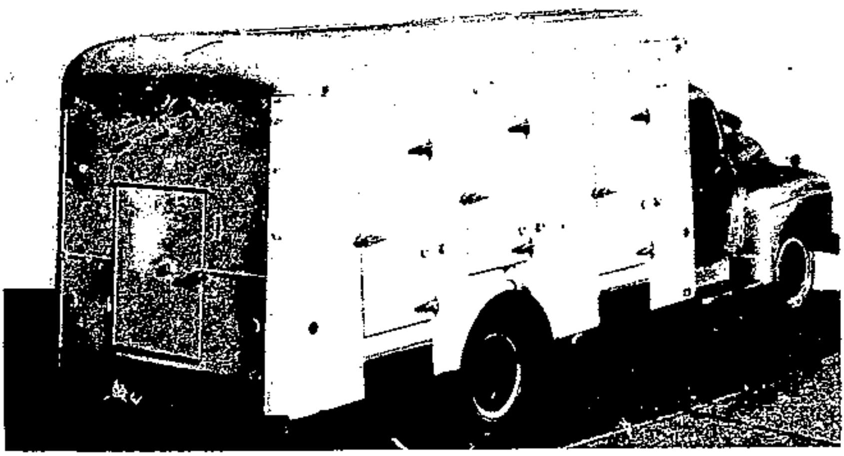
NBS 42103-7

FIGURE 7.—Test vehicle, body 2.

Body 3 was a reach-in design with three doors on each side. A single door at the rear opened to a storage compartment used for supplies, which was not refrigerated and which did not communicate with the larger space (fig. 8). Exterior dimensions were 182 inches long by 90½ inches wide by 88 inches high. The internal volume was 432 cubic feet. The metal exterior surface was painted a gloss white. Insulation consisted of 6 inches of expanded polyurethane in the roof, 4 inches of urethane foam in the walls, and 3 inches of urethane foam plus 1 inch of corkboard in the floor. Spacer strips were provided on the interior walls.

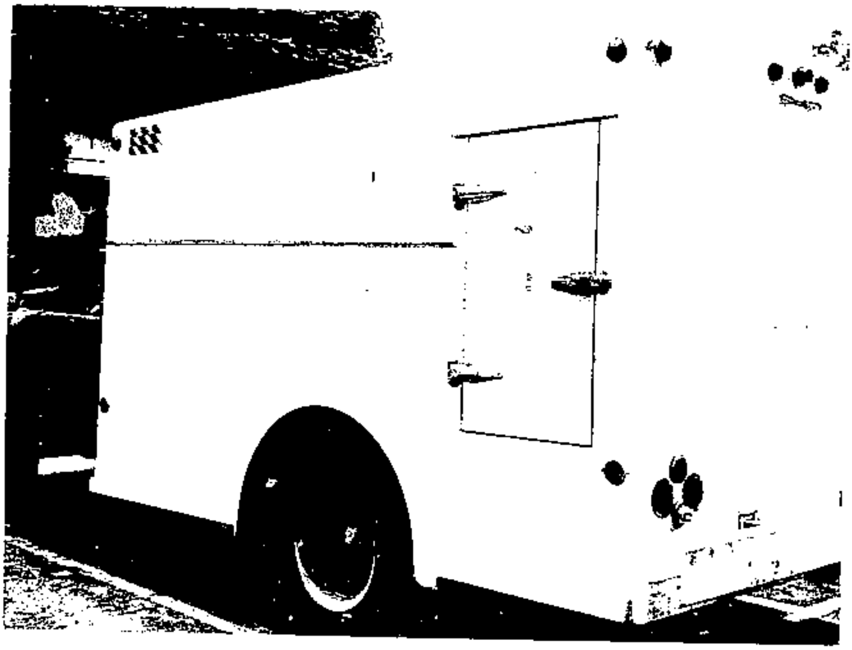
Body 4 was constructed of expanded polyurethane slabs resin bonded to multilayer glass cloth on both interior and exterior surfaces. The bonding agent penetrated the glass cloth, attaching it to the insulation to form a hard, smooth surface after curing in the mold. Sides, ends, and roof were formed in a mold as one subassembly, and the floor formed another. These parts were then assembled and resin bonded with glass cloth overlapping at all joints. Insulation thickness was 2 inches in walls and 3 inches in floor and roof. The color inside and out was white. Overall external dimensions were 140 inches long by 83 inches wide by 68 inches high. Internal volume was 333 cubic feet. A door 22½ inches wide by 44½ inches high was located at the rear of each side (fig. 9).

Body 5 was an integral cab design with double doors at the rear and a sliding door for access from the driver's compartment (fig. 10). The metal exterior surface was painted a flat gray. The metal interior lining had corrugated sheets on the roof, floor, and sidewalls. A combination of glass fiber and preformed urethane board insulations were used as follows: rear wall and roof, 3 inches urethane and 1 inch glass fiber; front wall, 2 inches urethane and 1 inch glass fiber; sidewalls and floor, 3 inches urethane. The glass fiber was used to fill between stiffening and structural members. The internal volume of the body was 183 cubic feet and its exterior dimensions were 92 inches long (not including integral cab) by 79 inches wide by 68 inches high.



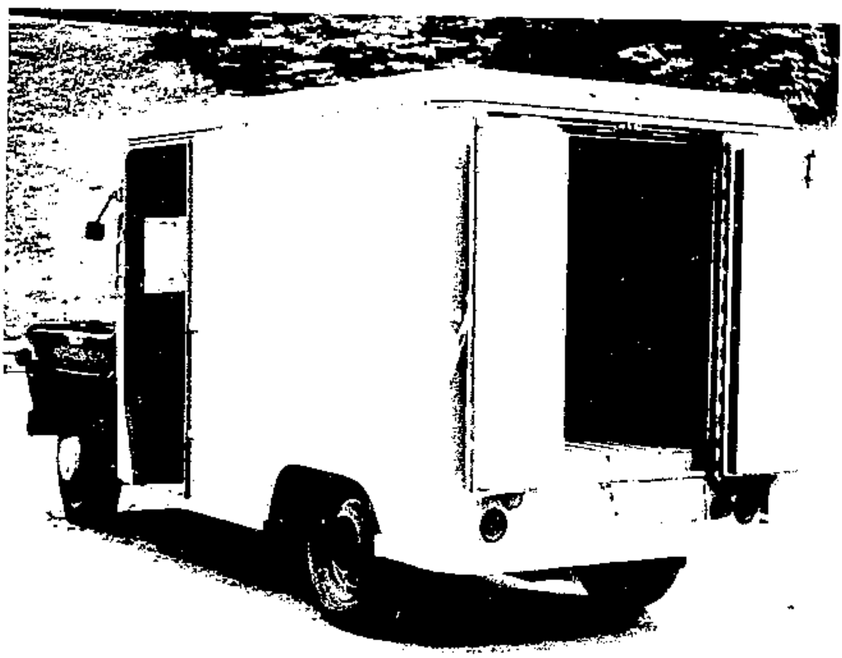
NBS 42103 S

FIGURE 8.—Test vehicle, body 3.



NBS 31518-1

FIGURE 9.—Test vehicle, body 4.



NBS 42103-10

FIGURE 10.—Test vehicle, body 5.

TEST PROCEDURES

Steady-State Cooling Load

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 percent relative humidity. After 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.

For a steady-state cooling load test, the instruments were read at half-hour intervals until a period of not less than 6 hours of uninterrupted and essentially constant heat gain was obtained. Data taken included interior air temperatures, test room temperature and humidity, brine flow rate, brine temperatures, and brine temperature difference in and out of the truck body and in and out of the comparison heater, electric power to the comparison heater, electric power to the fan and control heater in the truck body, and various other temperature measurements. Each test was repeated one or more times, with the vehicle maintained under test conditions for several days.

Air-leakage rates during steady-state cooling load tests were measured by means of the modified helium-trace katharometer. 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, when required to maintain controlled conditions, was done at the end of a period of data observation, allowing nearly 16 hours for recovery of steady-state conditions before the next period of data observations.

Static-Pressure Air Leakage

In static-pressure air-leakage tests, the interior of the truck was pressurized to approximately 0.10 inch of water above atmosphere. The tests were made under isothermal conditions with both interior and exterior of the truck at room temperature. When the airflow necessary to maintain the desired pressure across the truck body walls was established, the pressure drop across the orifice in the 2-inch pipe was observed. One test was made with the doors unsealed and one with the doors sealed with tape.

Solar Load

The test procedure for a vehicle under simulated solar heat load was similar to that of steady state, with the interior air temperature controlled at 0° F. and the ambient air maintained at a temperature of 100° F. and 50 percent relative humidity. 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 was lowered, the temperature of the heater approached the ambient temperature of 100° F., at which temperature the wavelength of maximum emission was about 9.3 microns. The significance of selecting a radiation source at temperatures of 700° F. or lower is related to the ability 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 to establish similar conditions for the simulated solar tests. Metallic surfaces, such as aluminum or stainless steel, or surfaces coated with metallic paint, 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. (*8, ch. I*). Such a solar day extends from 4:40 a.m. to 7:20 p.m., with maximum insolation values of 3.9 B.t.u. per minute per square foot on a vertical east wall at 7:30 a.m. and on a vertical west wall at 4:30 p.m., and 5.3 B.t.u. per minute per square foot on a horizontal surface at noon (fig. 11).

The voltage on the bank of heaters over the roof was adjusted to produce a temperature rise approaching 70 degrees above an ambient temperature of 100° F., thus simulating solar irradiance

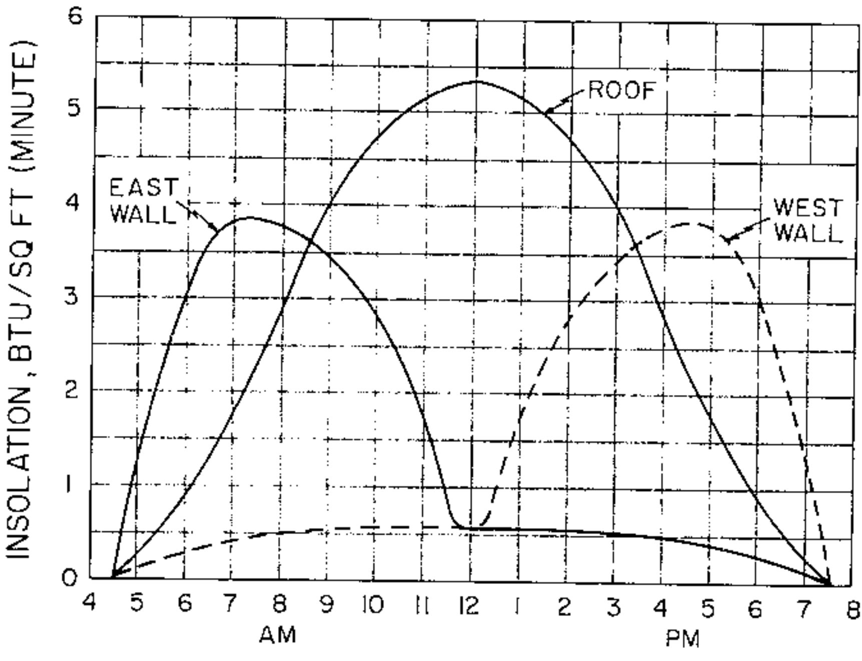


FIGURE 11.—U.S. Weather Bureau hourly values of cloudless day insolation for June 21 at latitude 40° N.

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 11 for the roof. The voltage on the other two banks was adjusted independently to provide power dissipation proportional to the heights of the other two curves in figure 11. The same cycle of power input was used with all vehicles.

The ambient conditions for the seven tests are shown in table 1.

TABLE 1.—*Ambient conditions for the 7 tests of body E under simulated solar loading*

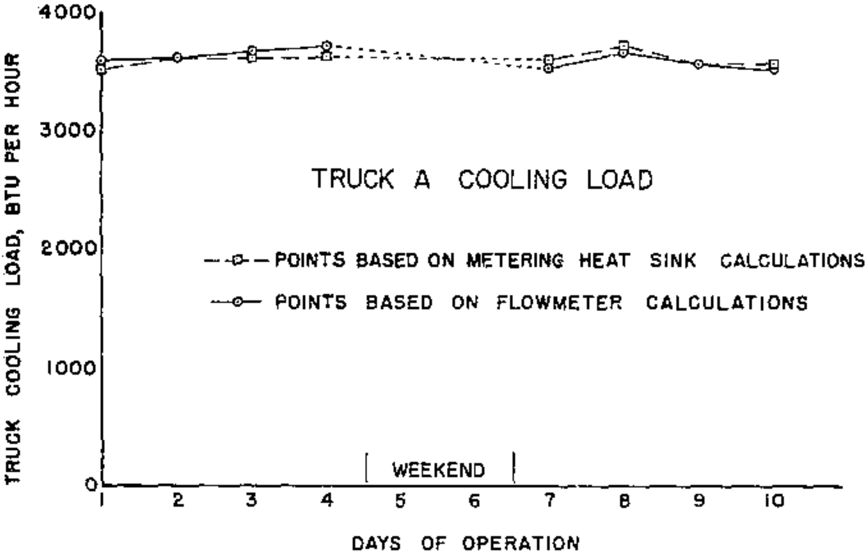
Test No.	Ambient temperature			Solar simulation
	Conditions	Mean	Range	
		° F	° F	
1	Steady	100	——	No
2	Do	100	——	Yes
3	Variable	100	90 to 110	No
4	Do	100	90 to 110	Yes
5	Do	90	80 to 100	Yes
6	Do	90	80 to 100	No
7	Steady	90	——	No

Tests 1 and 2 were run to compare the observed cooling load rates under constant 100° F. ambient temperature with and without a solar cycle. Test 7 was made with the ambient temperature held steady at 90° without a solar cycle for comparison with test 1 at 100° ambient. Tests 4 and 5 were conducted with a solar cycle, and with the ambient temperature varied 20 degrees sinusoidally over the 24-hour period to yield mean temperatures of 100° and 90°, respectively. In tests 3 and 6, the ambient temperature variations of tests 4 and 5 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 percent relative humidity.

TEST RESULTS

Steady-State Cooling Load

The results of steady-state cooling load tests on the five vehicles are shown in table 2. Cooling loads ranged from 1,800 to 3,700 B.t.u. per hour. These values are believed to be accurate within 5 percent. 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. A comparison of the cooling loads simultaneously determined by the metered heat sink method (primary method) and the flowmeter method (secondary, or check method) is shown in figure 12 for truck body A during a series of special test runs. The air-leakage cooling load was calculated from the rate of weight gain of the vehicle through moisture accumulation during the tests. Air-



NBS 42103-12

FIGURE 12.—Cooling load of body A as determined simultaneously by the metered heat sink and flowmeter methods of measurement.

leakage cooling loads ranged from 2 percent to 32 percent of the total cooling load of the truck bodies tested.

TABLE 2.—*Steady-state cooling loads due to air leakage and transmission in 5 truck bodies at specified test conditions* [1]

Truck body	Total cooling load [2]	Transmission [3]	Air leakage
	B.t.u. per hour	*B.t.u. per hour*	*B.t.u. per hour*
A	3,700	2,530	1,170
B (0° F.)	3,200	2,180	1,020
B (35° F.)	2,150	1,510	640
C	2,550	>2,500	<50
D	1,800	1,730	70
E	1,850	1,800	50

[1] Ambient temperature 100° F., relative humidity 50 percent, interior temperature 0° F.; truck body B also was tested with inside temperature 35° F.

[2] Values rounded to nearest 50 B.t.u. per hour.

[3] Transmission cooling load = total cooling load minus calculated air leakage cooling load.

Analysis of the primary test data showed that the smaller the ratio of control heat (the heat required to keep the truck body interior air temperature at 0° F.) to the heat absorbed by the air-cooling coil (and approximately matched by the comparison heater), the more precise the measurement of the cooling load of the truck. Relatively large amounts of control heat unduly affect the precision of the correspondingly smaller truck cooling load portion of the measured heat absorbed by the cooling coil. For this reason, it was concluded that the control heat should be no greater than 50 percent of the heat absorbed by the air-cooling coil.

In developing the test apparatus, it was found necessary to have very steady brine temperatures. Fluctuations in the temperature of the brine leaving the refrigerating unit carried through the entire system and caused variations in the 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 5 percent accuracy of the rating.

Air Leakage

The rate of weight gain in pounds per hour and the air-leakage rate in cubic feet per minute for the five vehicles are given in table 3. Air leakage was computed from the measured weight gain of each truck, on the basis of ambient air entering the body, depositing moisture, and leaving saturated at 0° F. Air-leakage rates for three of the test vehicles (A, B, and E) also were measured with the helium-trace katharometer. For truck bodies A and B, air-leakage rates obtained by the two methods of measurement supported the belief that weight gain is a result of all air exchange in the vehicle body, not just that which occurs in the cargo space. For truck body E, the air-leakage rates were considered too small to permit effective comparison of the two methods of measurement.

A static-pressure air-leakage test was run on four of the five vehicles. Table 4 shows the air-leakage rates for the test vehicles with doors sealed and unsealed. The excessive leakage of truck body A was through the walls, whereas for truck body B, more than half the air leakage was around the doors. The contrast between the observed air-leakage rates for trucks A and B and those for trucks C and D in table 4 indicates a wide range in effectiveness of various techniques for reducing air leakage. Smoke tests of some of the bodies revealed leakage at the doors and doorframe members, body seams, and door gaskets.

The air-leakage rates shown in table 4 for isothermal static-pressure difference tests were higher than those shown in table 3 for refrigerated test conditions because the static-pressure differences chosen for the isothermal test were considerably higher than the static-head-pressure differences developed under the refrigerated test conditions, and because the air leakage was unidirectional (outward through all leaks) in the isothermal test and bidirectional (in through some leaks and out through others) in the refrigerated test.

From these tests it was found that some types of truck body construction are sufficiently tight that air leakage and attendant weight gain may be insignificant. Accordingly, in developing the proposed rating method, it was considered practical to disregard any weight gain (during the 24-hour weight gain portion of the test) less than the "sensitivity requirement" [3] of an appropriate scale or other weighing device.

[3] For definition of "sensitivity requirement," see National Bureau of Standards Handbook No. 44 (*5*).

As shown in table 3, the observed weight gain for truck body B when tested at 0° F. interior temperature was 0.41 lb. per hour and at 35° F. interior temperature, 0.29 lb. per hour. In developing the proposed rating method the observed ratio of $\frac{0.29}{0.41}=0.71$ was used to interpolate the observed weight gain rate in a test at 0° F. interior temperature for a rating at 35° F. interior temperature. The ratio of 0.71 for the observed weight gain rates is in good general agreement (within 6 percent) with a theoretical interpolation which gave 0.65 for the ratio, based on the assumption that air infiltration would be proportional to the square root of the air density difference under the two test conditions and that air entered the test vehicle at ambient conditions (100° F. temperature and 50 percent relative humidity) and left saturated at the respective interior temperatures.

TABLE 3.—*Air-leakage rates for the 5 truck bodies determined from weight gain and measured by the helium katharometer at specified test conditions*[1]

Truck body	Weight gain rate	Air leakage rate determined by—	
		Weight gain rate	Helium trace
	Pounds per hour	*Cubic feet per minute*	*Cubic feet per minute*
A	0.47	5.7	4.7
B (0° F.)	.41	5.0	3.8
B (35° F.)	.29	4.2	——
C	.014	.17	——
D	.028	.34	——
E	.022	.27	.39

[1] Ambient temperature 100° F., relative humidity 50 percent, interior temperature 0° F.; truck body B was also tested with inside temperature 35° F.

TABLE 4.—*Air-leakage rates for 4 truck bodies when subjected to a selected static-pressure difference between cargo space and ambient space*

Truck body	Pressure across walls	Leakage rate	
		Doors unsealed	Doors sealed
	Inches water gage	*Cubic feet per minute*	*Cubic feet per minute*
A	0.07	105	102
B	.11	92	41
C	.10	8	<8
D	.10	2	——

Solar Load

Figure 13 shows the exterior surface temperature curves recorded in a typical solar simulation test; these curves are similar in shape to the insolation curves of figure 11. In table 5 the observed cooling load average for the maximum 4 hours is given and interpreted as the refrigeration requirement under solar exposure. The percentage increases of the cooling loads under solar exposure compared to the steady-state cooling loads are also shown in table 5. For the four trucks tested, the average increase was 22 percent, with individual values ranging from 19 to 25 percent. A 22-percent increase from solar heat load is suggested by these comparative tests. Table 5 also compares the solar load test results with estimated values obtained by multiplying the steady-state cooling load by 1.22.

Seven tests were run on truck body E to compare truck cooling loads with and without simulated solar heating when (1) the ambient temperature was held constant at either 100° or 90° F., and (2) the ambient temperature was varied to follow an approximately sinusoidal daily cycle having a range of 20 degrees with its maximum usually occurring 3.5 hours later than the solar noon and yielding mean ambient temperatures of either 100° or 90° F. The results are given in table 6.

Test pairs a, b, and c in table 6 show that the simulated solar cycle increased the cooling load about 450 B.t.u. per hour, and that the increase was independent of the ambient temperature level and of the steady or sinusoidal cycle of ambient temperature. Test pair d shows that reducing the steady ambient temperature from 100° 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, reduced the cooling

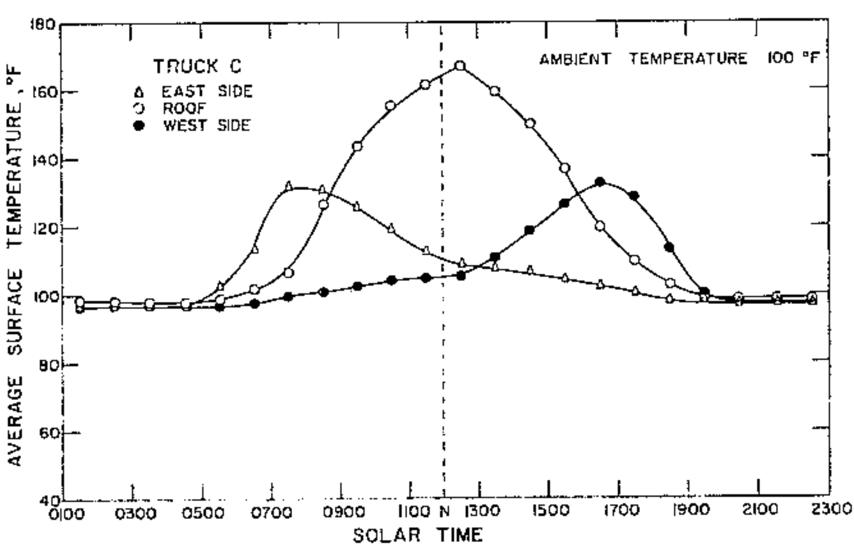


FIGURE 13.—Exterior surface temperatures during simulated solar test.

TABLE 5.—*Steady-state and solar cooling loads for the 5 truck bodies under standard conditions* [1]

Truck body	Steady-state cooling load	Solar load by test [2]	Solar load, increase over steady state	Solar load, estimated [3]	Estimate as percent of test results
	B.t.u. *per hour*	B.t.u. *per hour*	*Percent*	B.t.u. *per hour*	*Percent*
A	3,700	4,600	24	4,510	98
B	3,200	[4]	—	3,900	—
C	2,550	3,070	20	3,110	101
D	1,800	2,150	19	2,200	102
E	1,850	2,320	25	2,260	97

[1] Ambient temperature 100° F., relative humidity 50 percent, interior temperature 0° F.
[2] Average for 4-hour maximum load.
[3] Steady-state cooling load multiplied by 1.22.
[4] Not tested.

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° to 90° F. decreased the cooling load 9 percent. Test pairs g and h show that the effect of the sinusoidally varied ambient temperature at 100° was to increase the cooling load 180 B.t.u. per hour over the steady ambient value, with or without a simulated solar cycle. A smaller increase (120 B.t.u. per hour) is shown by test pair i, at 90° mean ambient temperature.

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 4 hours of 99.5° F. The difference between the ambient temperature 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 4 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 4 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 B.t.u. per hour less than the 4-hour maximum load of test No. 5, conducted with a sinusoidally varied ambient temperature with a 90° F. mean and 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 is probably a good approximation of 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 from door usage, which would depend on the truck service.

Since there was moderately good agreement about the increase of cooling load due to a simulated solar cycle (approximately 22 percent) for all four trucks tested (see table 5), it is suggested that a fair approximation of the maximum cooling load of a truck through 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 (that is, 1 + 410/1,870 for truck E). On this basis, the determination of comparable 4-hour maximum cooling loads for trucks can be effected by tests at

TABLE 6.—*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 [1]	Mean ambient temperature	Solar cycle	Cooling load [2] Measured	Cooling load [2] Difference
			F.		*B.t.u. per hour*	*B.t.u. 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
	3	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.

[2] Cooling loads under variable ambient conditions are the average for the 4-hour period of maximum load.

steady 100° ambient temperature, with no solar cycle, conditions the same as those called for in the procedure previously developed for the rating of refrigerated trailers.

RATING TRUCK BODY FOR 35° F. INSIDE TEMPERATURE

No solar simulation was run on a vehicle whose interior temperature was 35° F. However, the data obtained from the 0° F. interior temperature tests and a theoretical analysis of the effect of a 35° F. interior temperature (see Appendix I) yield multiplication factors for probable solar effects. For a truck tested at 0° F. interior temperature to be rated for 35° F. interior temperature, multiply the steady-state cooling load at 0° F. by 0.87. For a truck tested at and rated for 35° F. interior temperature, multiply the measured cooling load at 35° F. by 1.34. No other cooling load extrapolations are recommended.

To extrapolate the weight gain rate for a truck tested at 0° F. interior temperature and to be rated for 35° F. interior temperature, multiply the observed weight gain rate at 0° F. by 0.71.

CONCLUSIONS

Solar heat gain must be accounted for in any truck body rating method. The test data show that the solar heat gain is consistent enough to permit the use of a 100° F. steady ambient temperature test and a multiplication factor in lieu of either an increased ambient temperature or an actual solar simulation test.

Even though defrosting 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 the determination, the air cooler, particularly the drain pan, should be designed to drain freely during the defrosting operation.

The static-pressure air-leakage test, helium-trace air-leakage test, and smoke bomb test are useful in determining the relative effectiveness of air-sealing techniques. Although these tests are not included in the recommended standard rating technique, they might be useful for in-house tests by a manufacturer to pinpoint major leakage problem areas.

Based upon the results of this study, a recommended standard rating method to determine the cooling load for refrigerated truck bodies is given in Appendix II.

LITERATURE CITED

(1) ACHENBACH, P. R., and PHILLIPS, C. W.
1959. HEAT SINK METHOD FOR MEASURING THE COOLING LOADS OF REFRIGERATED STRUCTURES. Internatl. Inst. Refrig. Xth Internatl. Cong. Proc. Part 2: 272–276, illus.

(2) ——— PHILLIPS, C. W., and PENNEY, R. W.
1965. MEASURING THE COOLING LOAD OF REFRIGERATED VEHICLES BY FREE EVAPORATION OF LIQUID NITROGEN. Supp. 1965-3, Bul. Internatl. Inst. Refrig., Comn. 7: 115–124, illus.

(3) COBLENTZ, C. W., and ACHENBACH, P. R.
1957. DESIGN AND PERFORMANCE OF A PORTABLE INFILTRATION METER. Amer. Soc. Heat., Refrig., and Air-Cond. Engin. ASHRAE Trans. 63: 477–482. illus.

(4) MARVIN, C. F.
1941. PSYCHROMETRIC TABLES FOR OBTAINING THE VAPOR PRESSURE, RELATIVE HUMIDITY, AND TEMPERATURES OF THE DEW POINT FROM READINGS OF THE WET-AND DRY-BULB THERMOMETERS. U.S. Weather Bur. No. 235, 87 pp., illus.

(5) NATIONAL BUREAU OF STANDARDS.
1965. SPECIFICATION, TOLERANCE AND OTHER TECHNICAL REQUIREMENTS FOR COMMERCIAL WEIGHING AND MEASURING DEVICES. Handb. No. 44, Ed. 3, 178 pp., illus.

(6) PENNEY, R. W., and PHILLIPS, C. W.
1967. REFRIGERATION REQUIREMENTS FOR TRUCK BODIES—EFFECTS OF DOOR USAGE. U.S. Dept. Agr. Tech. Bul. 1375, 17 pp., illus.

(7) PHILLIPS, C. W., GODDARD, W. F., ACHENBACH, P. R., and others.
1960. A RATING METHOD FOR REFRIGERATED TRAILER BODIES HAULING PERISHABLE FOODS. U.S. Dept. Agr. Mktg. Res. Rpt. 433, 52 pp., illus.

(8) STROCK, C. and KORAL, R. L.
1965. HANDBOOK OF AIR CONDITIONING, HEATING, AND VENTILATING. Ed. 2. Industrial Press, New York, N.Y.

(9) THRELKELD, J. L.
1962. THERMAL ENVIRONMENTAL ENGINEERING. 514 pp., illus. Prentice-Hall.

APPENDIX I.—MULTIPLICATION FACTORS FOR EXTRAPOLATION OF COOLING LOADS FROM 0° F. TO 35° F. INTERIOR TEMPERATURE

The basic equation for the total heat transfer into a refrigerated vehicle parked in the sun can be expressed as follows (*9, pp. 372–374*):

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

where:

q' = total heat transferred into the vehicle, B.t.u. per hour
U = overall heat transfer coefficient of the vehicle, B.t.u. per hour–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 thermodynamic environment. It is expressed by—

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

where:

Tem = mean sol-air temperature, ° F.
t_o = exterior ambient temperature, ° F.
Φ = insolation to which the vehicle is subject, B.t.u. per hour-ft.^{2}
a = surface absorptivity
h_o = outside heat transfer coefficient, B.t.u. per hour-ft.^{2}–deg. F.

Tem thus represents the energy level of the exterior of the vehicle, is dependent only on 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, as when the increase in insolation increases the heat flow into the vehicle. It is dependent on the transient heat conduction 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

$$q = U(t_o - t_i), \text{ B.t.u. per hour} \qquad (2)$$

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 solar insolation, and the subscript i on q denotes the truck interior temperature.

In the following table 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

$$q'_i/q_i=\frac{122-t'_i}{100-t_i} \tag{4}$$

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

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

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

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

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 *Purpose*

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 *Scope*

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 usage. 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 *Ratings*

2.1 *Standard Ratings*

There shall be three allowable ratings that can be published under this standard. The first two are preferable to the third. The first is the cooling load and weight gain rating for 0° F. interior temperature, 100° F. ambient temperature and 50 percent relative humidity, with the test run at 0° F. interior temperature. The second is the cooling load and weight gain rating for 35° F. interior temperature at the same ambient conditions with the test run at 35° F. interior temperature. The third is a cooling load rating for 35° F. interior temperature at the same ambient conditions (100° F. temperature and 50 percent relative humidity), interpolated as specified (see par. 8.0) from a test run at 0° F., and a weight gain rating for 35° F. interior temperature obtained by multiplying the observed weight gain rate in the 0° F. test by 0.71. The published rating shall show the specified multiplier used to account for solar load (see par. 8.0) and shall state the interior temperature of the rating, and the interior temperature at which the truck was tested.

Results to be determined from the rating test shall consist of the cooling load in B.t.u. per hour, and weight gain rate in pounds per hour, 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. All instruments and readings shall meet the accuracy requirements of this standard.

2.2 *Standard Rating Conditions*

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

2.2.1 *Cooling Load and Weight Gain Test*

	Air temperature	
	Low temperature rating	*moderate temperature rating*
Truck interior	0° F. dry bulb	35° F. dry bulb
Test room ambient [1]	100° F. dry bulb	100° F. dry bulb
	83.5° F. wet bulb	83.5° F. wet bulb

[1] 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 Hg decrease in barometric pressure.

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 one set of readings*
Test room air temperature:		
Dry bulb	±1.0° F.	±2.0° F.
Wet bulb	±1.0° F.	±2.0° F.
Truck interior air temperature:		
Dry bulb	±0.5° F.	±1.0° F.

3.0 *Instruments*

3.1 *Temperature and Humidity*

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 deg. F.
b.	Brine temperatures	±0.4 deg. F.
c.	Brine temperature differences in and out of truck and across external comparison heater	± .05 deg. F.
d.	Other temperatures	±0.5 deg. 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 feet per minute 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 (*4*).

Temperatures of brine in conduits shall be measured by inserting the temperature measuring element at the bottom of a well, which shall be immersed not less than 25 times the outside well diameter at stations where the brine is well mixed. Instruments or systems used to measure the temperature differences of the brine in and out of the truck and across the comparison heater shall be compared with each other before they are installed and in the range used they shall agree within 0.05 deg. F. when immersed in the same baths.

3.2 *Brine Flow*

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

3.3 *Electrical*

Electrical energy usage should be determined preferably with integrating watt-hour meters calibrated for expected conditions of current and voltage. 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 load being measured. Accuracy of instruments used to measure other electrical quantities shall be within ±2.0 percent of the quantity measured.

3.4 *Weight*

Scales or other weighing devices used to measure the change of weight of the truck being tested shall have a "sensitivity requirement" [1] of 0.5 pound maximum under actual test loads.

4.0 *Test Room*

An insulated test room approximately 16 feet wide and 35 feet long and 14 feet high is required for testing trucks. A door at least 9 feet wide and 12 feet 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 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 forced air movement using fans will probably be required to attain the specified uniformity. (See par. 5.0.) 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 feet per minute. 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.

---

[1] See National Bureau of Standards Handbook No. 44 (*5*).

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 14. As indicated in the figure, the equipment consists of an air-cooling coil and fan inside the truck, a refrigerating unit and brine chiller, a brine pump, and a comparison heater and flowmeter in the brine circuit outside the truck.

The refrigerating unit and brine chiller may be single stage or multistage, and the refrigerating unit must have a capacity of not less than twice the cooling load of the largest test specimen, with brine leaving the chiller at a temperature of about −25° F. for tests at 0° 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° F.

The cooling coil inside the truck shall be designed without fins or shall have fin spacing of ¼-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

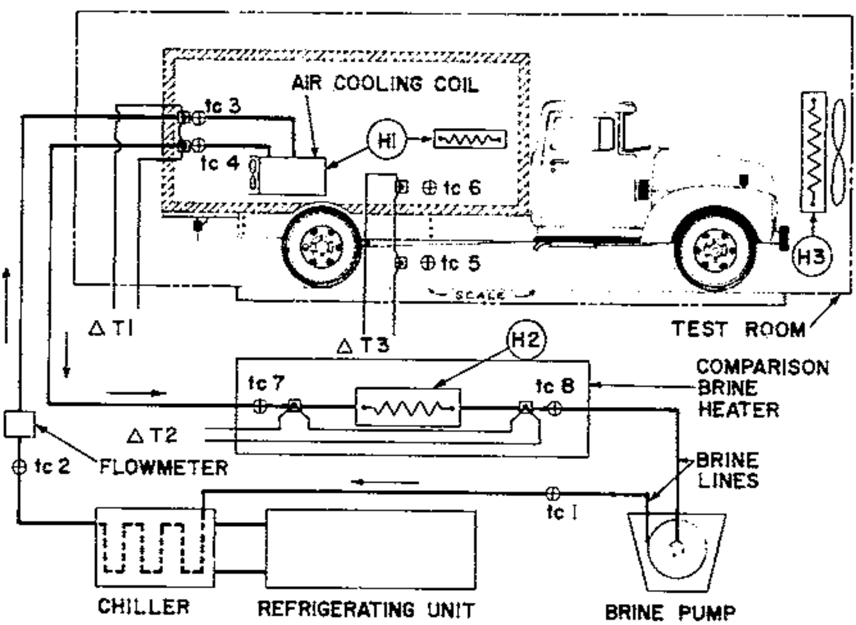


FIGURE 14.—Cooling load test apparatus. H1, H2, and H3 represent heaters; ΔT1, ΔT2, and ΔT3 are differential temperatures, tc 1 to 8 are thermocouples or resistance thermometers.

temperature of 20 deg. F. or less. The blower in the cooling unit shall deliver sufficient air to maintain 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, and 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 ½ percent or less of the electric heat input to the brine heater. The heating capacity of the comparison heater should be adjustable, and for each test shall be made approximately equal to the maximum total heat absorption of the air-cooling coil in the truck. 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 degrees F. Its density shall not vary more than 0.08 percent per deg. F. and its specific heat shall not vary more than 0.02 percent per deg. F. in the range of temperature used in the brine circuit. The specific heat shall be determined within 1 percent. Methylene chloride meets the density and specific heat tolerances specified and has 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 or have flexible connections 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. When 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 differences of the brine in and out of 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 between any two 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 12 elements are read individually, no 2 individual stations may differ by more than 3 deg. F.; if the groups of 4 are used, no 2 groups of 4 may differ by more than 2 deg. F. at a given time.

During the portion of the test used to determine the cooling load rating, all observations shall be made at intervals not exceeding 30 minutes.

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. Av. power input to comparison brine heater	watts
5. Av. power input to heater in truck	watts
6. Av. 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	amperes
11. Electric current to heater in truck	amperes
12. Electric current to fan motors in truck	amperes
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	° 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	° 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 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, in which case the two values are averaged to determine the cooling load rating. If the results of the two methods do not agree within 5 percent, the test must be repeated.

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.

Referring to figure 14, the cooling load measured by the comparison method shall be computed for the standard temperature difference of 100 deg. F. by the following equation:

$$\text{Cooling load, B.tu. per hour} = [H2(\frac{\Delta T1}{\Delta T2}) - H1] \times \frac{100}{\Delta T3}$$

where:

$\Delta T1$ = Temperature rise of brine in the truck, deg. F.
$\Delta T2$ = Temperature rise of brine between inlet and outlet of comparison heater, deg. F.
$H2$ = Heat input to comparison heater, B.t.u. per hour.
$\Delta T3$ = Temperature difference between air in truck and air in test room, deg. F.
$H1$ = Heat input to heater, fan motors, etc., inside the truck, B.t.u. per hour.

The cooling load measured by the flowmeter method shall be computed for the standard temperature difference of 100 deg. F. by the following equations:

$$\text{Cooling load, B.t.u. per hour} = (\Delta T1 \times M \times C_p - H1) \times \frac{100}{\Delta T3}$$

where:

$\Delta T1$ = Temperature rise of brine in the truck, deg. F.
M = Brine flow rate, pounds per hour
C_p = Specific heat of brine at mean temperature in the cooling coil, B.t.u. per pound–deg. F.
$H1$ = Heat input to heater, fan motors, etc., inside the truck, B.t.u. per hour.
$\Delta T3$ = 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 B.t.u. per hour, for example, 1,200, 1,600, 2,200, 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 pounds per hour determined for the final 24 hours of the test, and shall be expressed to the nearest 0.1 pound per hour.

For a rating at 35° F. interior temperature when the test is made at 0° F. interior temperature, the standard weight gain rating is the average weight gain rate in pounds per hour determined for the final 24 hours of the test multiplied by 0.71, and shall be expressed to the nearest 0.1 pound per hour.

If the average weight gain rate is less than 0.05 pound per hour,[5] the published weight gain rating shall be stated as "less than 0.05" or "<0.05" pound per hour. (See par. 10.)

10.0 *Published Ratings*

Published ratings, to conform to this standard, shall be identified as follows: "(sponsor's designation) Standard Cooling Load Rating (test result) B.t.u. per hour, Standard Weight Gain Rating (test result) pound per hour, 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.

---

[5] 0.05 pound per hour = 2.5 times the required maximum sensitivity requirement divided by 24. (See par. 3.4.)

U.S. GOVERNMENT PRINTING OFFICE 1967—O 264-853

END