A RATING METHOD FOR

REFRIGERATED TRAILER BODIES

HAULING PERISHABLE FOODS

U. S. DEPARTMENT OF AGRICULTURE AGRICULTURAL MARKETING SERVICE
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PREFACE

This study is part of a broad program of research to improve the design and performance of equipment used to transport agricultural products, as a means of improving marketing procedures and of expanding outlets of farm products. The research work reported herein covers the development of a practical method of rating the performance, or effectiveness, of refrigerated trailer bodies used in the transportation of perishables.

This was a joint government-industry project with equipment, facilities, and funds contributed by the following: Agricultural Marketing Service; National Bureau of Standards; Quartermaster Research and Engineering Command, Department of the Army; and the Truck-Trailer Manufacturers Association coordinating the support of the American Trucking Associations and a number of manufacturers of refrigerating units and trailer components. The drawings and illustrations in this report were prepared by the National Bureau of Standards.

Washington, D. C. September 1960

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SUMMARY

Heretofore there has been no standard method to measure the heat transfer rate, or cooling load, of a refrigerated trailer body. Consequently, it has not been possible to determine accurately the performance of an insulated trailer body or to select the proper refrigerating unit to maintain a desired inside temperature. While most trailers probably maintain proper inside temperature, many do not, especially in the 0° F. range for frozen foods.

This report covers road tests and laboratory tests at the National Bureau of Standards on several typical refrigerated trailer bodies. The object was to develop a reliable laboratory rating method which would simulate conditions in over-the-road movement. The road tests were conducted over the Ohio Turnpike.

In addition to development of a rating method, several facts of interest to trailer manufacturers and operators were studied. For instance, it was found that leakage into the trailer walls during road movement at 50 miles per hour caused an increase of 10 to 27 percent in the cooling load, compared to stationary tests in the laboratory. The air leakage was computed to be from 660 to 1,475 cubic feet per hour for the four trailers tested. These figures indicate the improvements in trailer performance which are possible by elimination of air leakage. Also, the underside of the trailer was sometimes heated as much as 15 degrees above ambient temperature during road operation, principally because of waste heat from the tractor engine. Frost accumulation in the trailer walls and on the cooling coil caused an increase of 0.52 to 0.98 pound per hour in trailer weight.

Based upon the results of this study, a standard method of testing refrigerated trailers is recommended. The method utilizes the heat sink principle with test conditions of 0° F. inside the trailer and 100° F. and 50 percent relative humidity in the test room. A brine, methylene chloride,

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is circulated through cooling coils inside the trailer and also through a brine heater located outside the trailer. Since the brine flow rate is the same in all parts of the circuit, the heat gain of the trailer can be calculated from the measured quantity of heat which has been added to the brine heater as electrical energy. This figure is found by multiplying the heat added to the brine heater by the ratio of the rise in brine temperature in the trailer to the rise in temperature of the brine in the brine heater, and subtracting from this quantity the heat introduced into the trailer by auxiliary equipment (blower, fans, space heater).

The report contains a detailed description of the facility, instruments, equipment, and test procedure to be used for testing. Adoption of this standard test method by the trailer manufacturing industry should result eventually in improved design and performance of trailers as well as better quality of perishable foods transported.

BACKGROUND

To date there have been no standard method or standard conditions for testing refrigerated food trailers to determine the cooling load imposed by heat transferred from outside to inside the insulated body. The cooling load can be defined as the refrigeration capacity necessary to hold the inside of the trailer at a given temperature under specified outside temperature and humidity conditions. The reverse heat flow method, with the cargo space heater above the ambient temperature, has been used occasionally to determine the heat transmission rate per unit of temperature difference. However, such tests have been made on stationary trailers, and the temperature and humidity conditions during the tests were such that no opportunity existed for moisture condensation in the insulated spaces.

Published material shows that refrigerated trailers and other vehicles for similar service accumulate considerable water or frost in the air cavities or interstitial spaces in the insulation when the interiors are refrigerated continuously. 2/ Some of these publications point out the importance of air leakage through the outside skin of the vehicle as a method of moisture entry. Field tests of seven commercial 35-foot refrigerated trailers, conducted jointly by the National Bureau of Standards and the U. S. Department of Agriculture in 1956, showed variations in total heat transfer rate of nearly 1½ to 1, in temperature pull-down time or more than 3 to 1, and in refrigerating unit capacity of more than 2 to 1 for systems designed for similar duty.

2/ Eby, S. W., and Collister, R. L. Insulation in Refrigerated Transport Body Design. Refrig. Engin. 63 (7). July 1955.

Palmieri, D. Measurement of K Transfer Coefficient, Methods and Comparison. Ninth Internatl. Cong. Refrig. Proc. Vol. 2, Sec. 7, p. 106. 1955.

Darlot, A., and Perrin, J. Measurement of the K Factor of a Refrigerated Vehicle by Two Methods, Influence of the Water Content of the Insulation. Ninth Internatl. Cong. Refrig. Proc. Vol. 2, Sec 7, p.122. 1955.

Lentz, C. P., and Rooke, E. A. Moisture Condensation in the Insulation of Canadian Railway Cars. Tenth Internatl. Cong. Refrig. Proc. 1959.

More recently a heat sink method has been developed by the National Bureau of Standards for measuring the cooling loads of refrigerated structures under conditions of temperature and vapor pressure in and around the structure that are typical of normal service. 3/ In the heat sink method the interior of the vehicle is refrigerated to the temperature at which the vehicle is designed to operate, and thus the mean temperature of the insulation is comparable to that in actual service. This method provides for a movement of air and water vapor into or through the walls of the vehicle that is typical of actual usage. The use of this method on a representative trailer has shown that the latent heat transfer represented by the continuous formation of frost in the insulation caused an increase of about 7 percent (%) in the heat transfer rate, and that the frost deposit in the insulation during the 70-day test increased the heat transfer rate an additional 7 percent.

In July 1957, a project was initiated at the Washington laboratories of the National Bureau of Standards, to develop a method for measuring the cooling load of refrigerated trailers.

The objectives of the laboratory study were as follows:

(1) To adapt the heat sink method for rating refrigerated trailers so that a laboratory test could be used to measure the heat transfer that would occur on the road under operating conditions,

(2) To compare the cooling loads of several trailers as measured in the laboratory with the cooling loads of the same trailers measured during road operation using the heat sink method in each case,

(3) To correlate the laboratory and road heat transfer data and to revise the laboratory test method to account adequately for the effects of wind, air pressure, and solar radiation,

(4) To provide information on air leakage and moisture transfer processes in refrigerated trailers that might suggest improved insulating and vapor-sealing methods and,

(5) To draft a proposed standard testing and rating method for refrigerated trailers with regard to cooling load, weight gain, and air leakage.

This report summarizes the results and conclusions from laboratory and road tests during the 19 months of the study and includes a draft of a proposed standard testing and rating method.

3/ Achenbach, P. R., and Phillips, C. W. Heat Sink Method For Measuring the Cooling Loads of Refrigerated Structures. Tenth Internatl. Cong. Refrig. Proc. 1959.

DESCRIPTION OF THE LABORATORY FACILITIES

The structure used for the laboratory tests of 35-foot refrigerated semitrailers was located on the grounds of the National Bureau of Standards in Washington, D. C. The test room was approximately 60 feet long, 14 feet wide, and 16 feet high. The floor was concrete; the walls and roof were made of corrugated metal sheathing and were insulated. A sliding overhead door, 12 feet wide and 14 feet high, was installed at one end of the test room to permit movement of semitrailers into the space. Figure 1 is an outside view of the structure and some of the vehicles used for the tests.

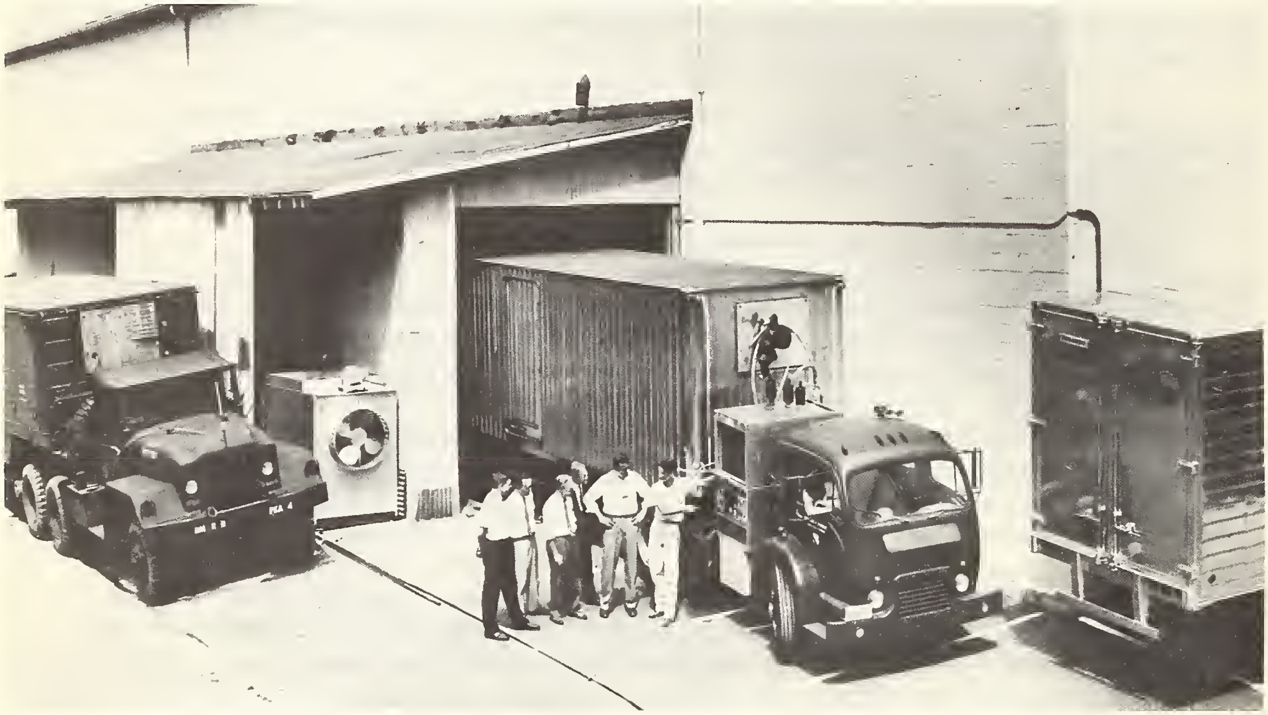

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Figure 1.-- Laboratory test structure and some of the vehicles used in the tests.

A room, adjoining the test chamber, was used to house the instruments necessary to control the test conditions and to observe and record temperature, humidity, electrical energy consumption, and other variables during the tests.

The heat sink apparatus, used to measure cooling loads, consisted of three electrically driven two-speed compressors with water-cooled condensers, a water cooling tower,two brine chillers, a brine pump, and brine heater installed adjacent to the test structure and the cooling coil inside the trailer. Figure 2 is a diagram of the refrigerating circuits of the heat sink apparatus. Refrigerants No. 12 and No. 22 were used as the primary refrigerants in the brine chillers. Methylene chloride was used as the brine or secondary refrigerant because of its favorable viscosity, its stability of density and specific heat in the working temperature range, and its reasonable characteristics of toxicity and flammability. It does affect natural and synthetic rubbers to some extent, so these materials should not be used.

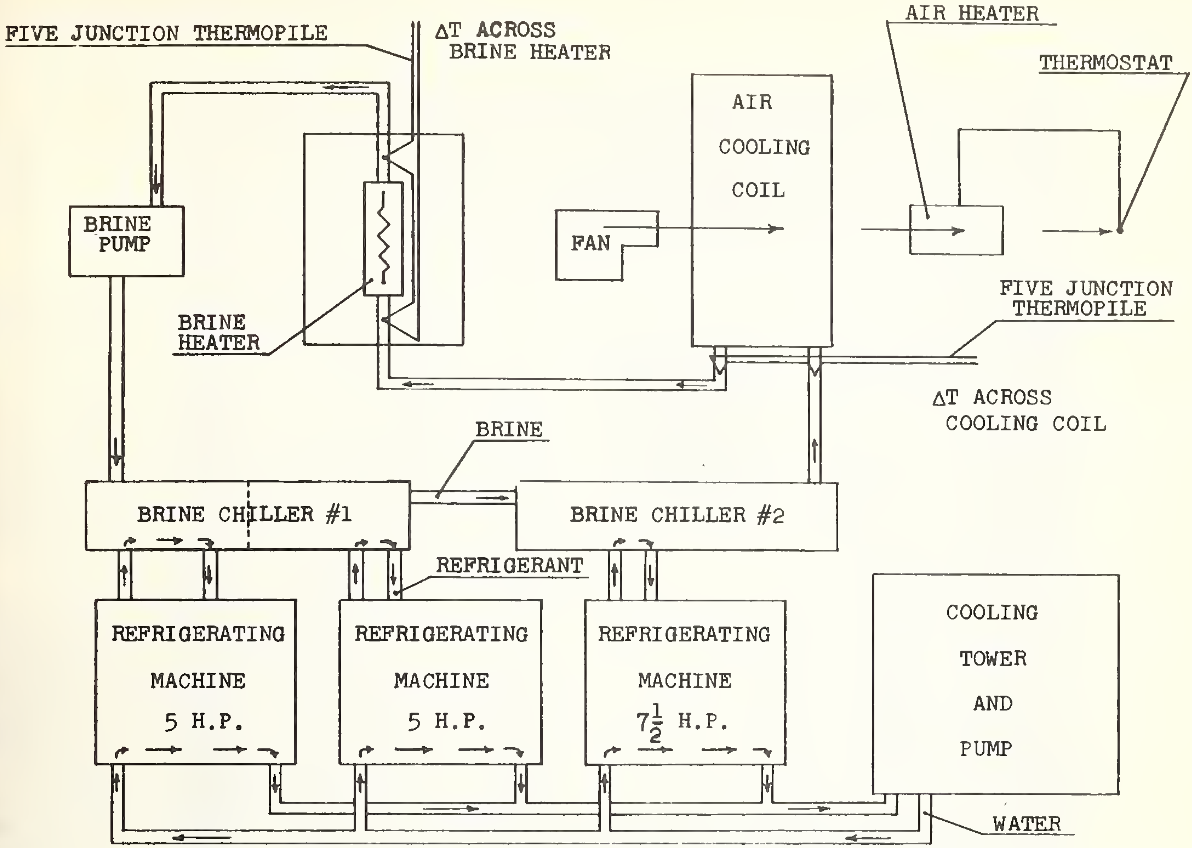

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Figure 2.-- Refrigerating circuits in the heat sink apparatus used for the laboratory tests.

The brine heater 4/ consisted of a vapor-proof insulated box containing an electric immersion heater inserted in the brine line between the pump and the cooling coil which was located inside the trailer, as shown in figure 2. The electric energy consumed by the heater was measured with a calibrated watt-hour meter. The insulation of the box and the design of the incoming and outgoing lines were such that the overall heat leakage into the box was no greater than ½ percent of the normal capacity of the brine heater.

To cool the trailer, a refrigeration coil with an integral blower (fig. 3) was placed inside, with the blower at approximately the same height above the floor as the blower of typical refrigerating units. Oscillating

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4/ Johnson, H. D., Winter, J. C., Phillips, C. W., and others. Heat Transfer Measurements on Refrigerated-Food Trailers. U. S. Dept. Agr. AMS-250, 12 pp., Illus.

fans were placed on the trailer floor to improve air distribution. The interior temperature was controlled by using a shielded electric space heater with built-in fan. This unit was placed on the housing of the cooling coil near the air outlet and was thermostatically controlled to maintain the desired temperature within the trailer by offsetting the slight excess of refrigeration provided.

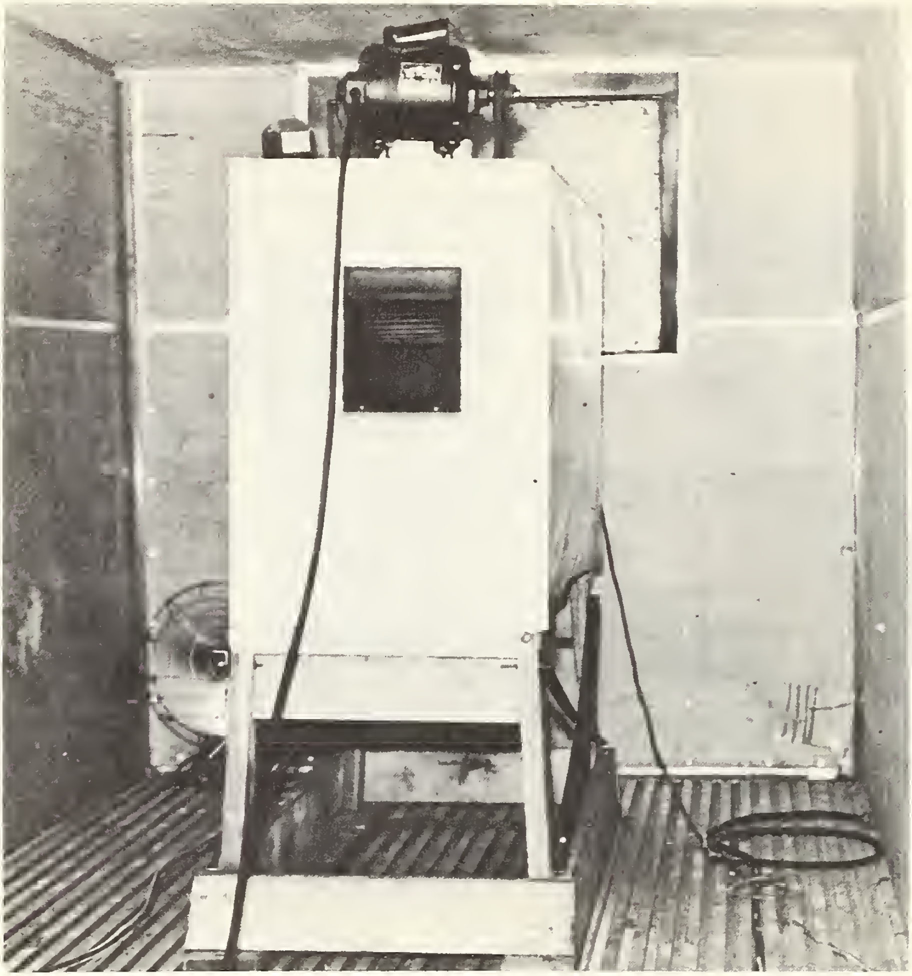

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Figure 3.-- Cooling coil and integral blower located inside a trailer as used for the laboratory tests.

Copper-constantan thermocouples made of 30-gage wire were used to measure brine and air temperatures. Those used to measure brine temperatures were placed in wells immersed in the brine. Each well was constructed and installed in such a way 5/ that the fluid was thoroughly mixed by tees before coming in contact with the end containing the thermocouple element. The heat conduction along the well was minimized by having it immersed in brine for a

5/ See footnote 4.

length at least 25 times its own diameter. Temperature differences in the brine circuit were measured with five-junction thermopiles to increase the accuracy of these readings. The air temperature inside the trailer was determined by averaging the readings of 12 thermocouples placed as follows: One in each corner of the front and rear of the trailer and one in each corner of a section midlength of the trailer. All of these thermocouples were located in air about 6 inches from each adjacent surface. The temperature of the air in the test room was taken by averaging readings of eight thermocouples, one at each exterior corner of the trailer, installed in air not less than 6 inches from any adjacent surface. Numerous other thermocouples were installed as needed for various test observations.

All electric energy used to operate fans, lights, and the heater used to obtain final temperature control in the trailer, was measured by watt-hour meters.

The brine lines between the comparison brine heater and the cooling coil inside the trailer were well insulated. The brine lines, the electric circuits to the fans and heaters, and the thermocouple extensions entered the trailer through a specially designed, insulated wooden plug installed in the refrigeration unit opening in the front of each trailer.

Two thermostatically controlled unit heaters were used to warm the ambient air, and live steam injection controlled by a humidistat was used to maintain the desired humidity in the test room. Controls and equipment were set to maintain the desired test conditions of 0° F. temperature in the trailer, and ambient conditions of 100° F. dry-bulb temperature and 50 percent relative humidity in the test room. Six pedestal-type, 30-inch electric fans were used to circulate the ambient air around the trailer to promote uniformity of temperature and humidity during the tests. All temperatures were determined by thermocouples using either galvanometer or electronic potentiometers. The weight gain caused by condensation and freezing of infiltrated moisture was determined by placing the trailer on three calibrated, heavy-duty, platform-type scales, one under each set of back wheels as shown in figure 4, and one under the front at the king-pin position.

DESCRIPTION OF THE ROAD TEST EQUIPMENT

To equip the tractor used for the road tests, it was necessary to mount various instruments within the cab and to install a 3 kilovolt-ampere (kv.-a) generator, driven by a gasoline engine, and a two-piece, two-stage refrigerating unit, also driven by gasoline engines, immediately behind the cab. Figure 5 shows a view of the right side of a tractor-trailer combination connected for operation.

The refrigerating system used for the road tests was a modification of the heat sink apparatus used for the laboratory tests. For the road tests, the flow rate of the secondary refrigerant was measured with an electronic flowmeter. The specific heat and flow rate of the methylene chloride brine and its temperature change in the trailer cooling coil were used to calculate the cooling loads of the trailers. A brine heater, of smaller capacity than that used in the laboratory tests, was used at intervals to monitor the performance of the flowmeter. Figure 6 is a schematic drawing of the refrigerating circuit built for the mobile test equipment.

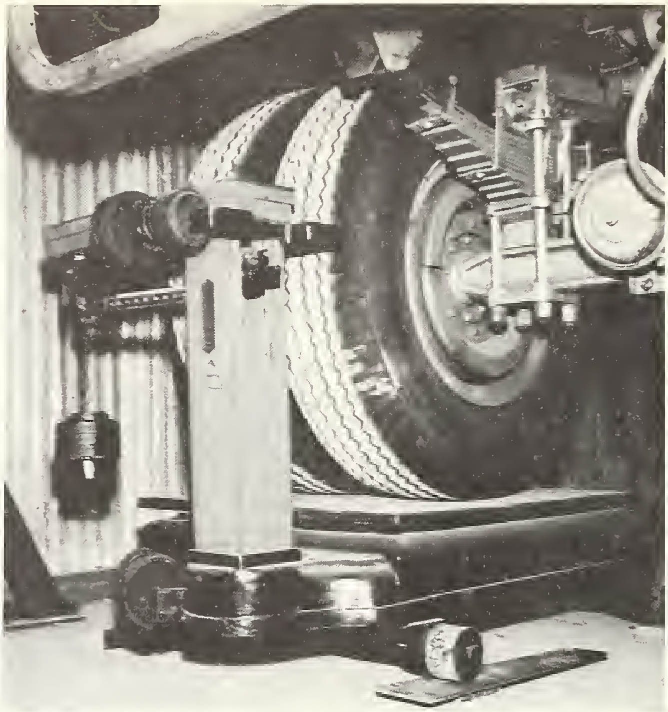

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Figure 4.-- Method of using platform scales to determine the weight gain of the trailers during the laboratory tests.

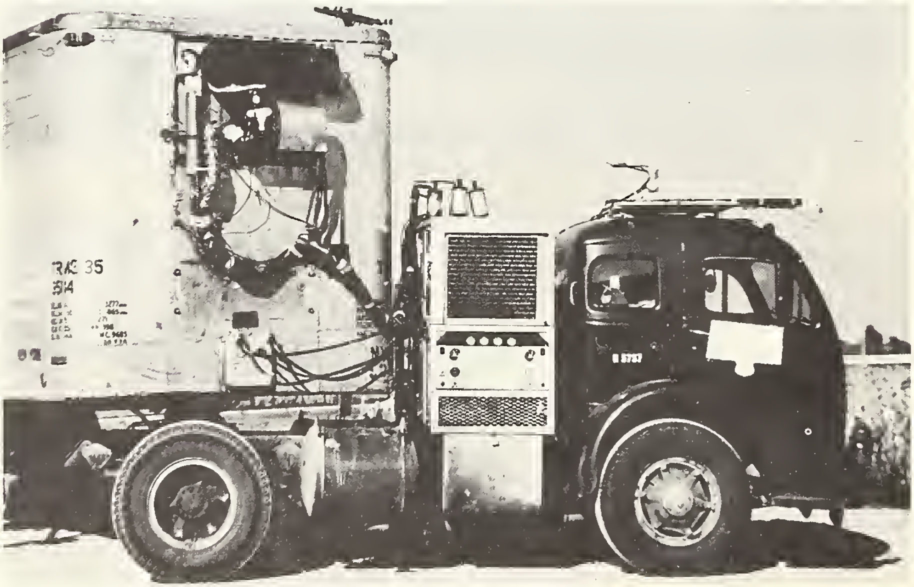

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Figure 5.-- Tractor equipment with a 2-piece refrigerating unit as used for the road tests.

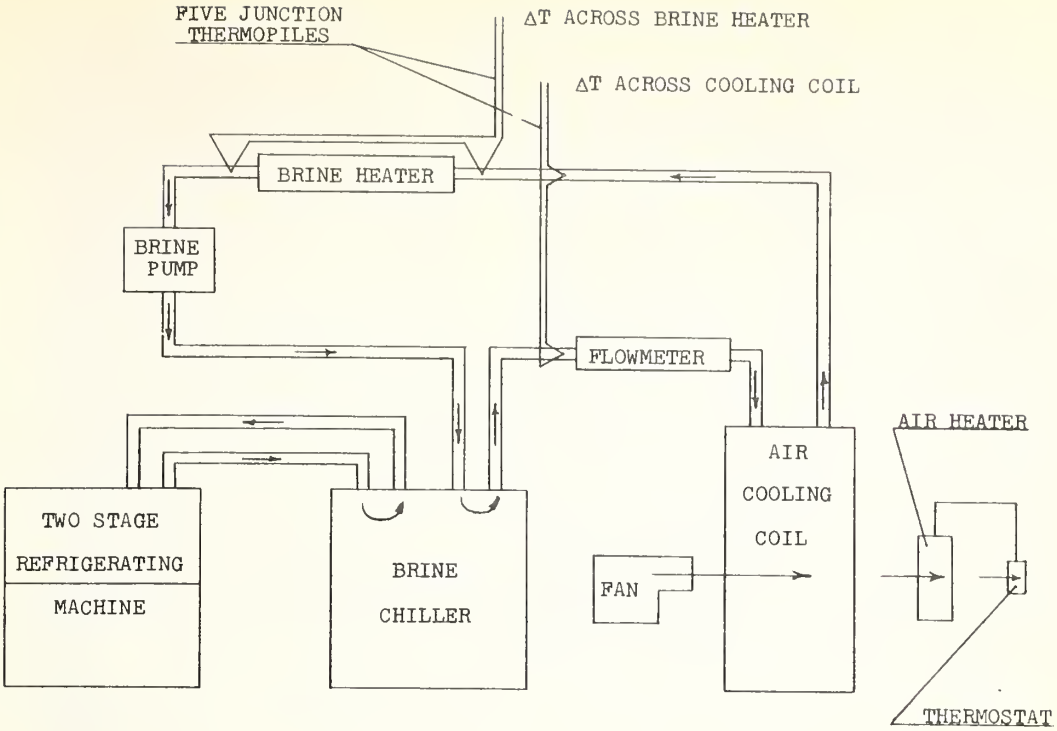

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Figure 6.-- Refrigerating circuits in the heat sink apparatus used for the road tests.

The refrigerating unit was specially designed for the purpose. It consisted of a refrigerant-12 high-stage and a refrigerant-22 low-stage unit, each powered by an air-cooled gasoline engine. This system was used to cool the brine in a well-insulated chiller which was mounted inside the trailer. The air cooling coil was mounted on the box enclosing the brine chiller and was fixed at an elevation which would place the discharge air stream at about the same height above the floor as for typical trailer refrigerating units. An electric heater of about 600 watts capacity with an integral fan was installed in the discharge air stream from the cooling coil as a final means of temperature control. Figure 7 shows the chiller, air cooling coil, and electric heater after installation in a trailer.

An aluminum-faced, insulated plug approximately 5 inches thick was placed in the refrigerating unit opening provided in the front of each trailer. This plug was reinforced near its outer edges with angle iron, and a steel shelf was built on its exterior surface. An electrically driven brine pump, connected to brine lines extending a short distance beyond the inner face of the plug, was mounted on the shelf. A 10-gallon brine storage tank, for make-up and expansion, was also attached to the front face of this plug, above the pump and motor, and was connected into the brine circuit at the pump inlet. The circulating pump, the storage tank, and the brine lines were insulated. A primary refrigerant heat exchanger was mounted on the right-hand side of the plug, and the flexible primary refrigerant liquid and suction lines to the tractor-mounted refrigerating unit were connected to the heat exchanger.

Provisions were also made in the plug for bringing thermocouple extensions, electric power and control circuits, and tubes for measuring air pressure differences inside the trailer.

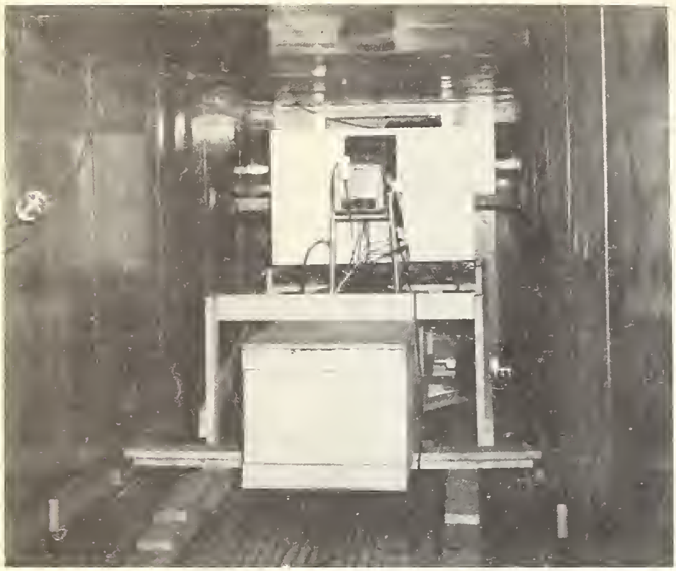

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Figure 7.-- The brine chiller, cooling coil, and electric heater installed in a trailer for the road tests.

Instruments for indicating and recording the ambient conditions and the performance of the test apparatus were mounted inside the cab back of and between the seats as shown in figure 8. Figure 9 identifies the location of each of the instruments in the cab of the tractor. In addition to the instruments normally provided on the dashboard for operation of the tractor, the following instruments were installed in the cab:

1. Electronic indicating potentiometer, for readings of temperature, temperature difference, relative humidity, relative incident radiation, and brine flow rate.

2. Electronic recording potentiometer, for permanent record of each of the variables listed in item 1.

3. Recording speedometer, for permanent record of rate of speed and time of operation.

4. Clock for time.

5. Electric clock frequency meter, for observing performance of the generator.

6. Ammeter and voltmeter, for observing electric generator output.

7. Watt-hour meter, for integration of power input to trailer.

8. Brine flow totalizer and frequency converter, for integrated flow and flow rate observation.

9. Electric hygrometer, for indication of ambient relative humidity.

10. Pressure gages, for measurement of impact and static air pressures.

11. Various switches, constant voltage transformer, hot and cold reference baths for checking temperature instruments, etc.

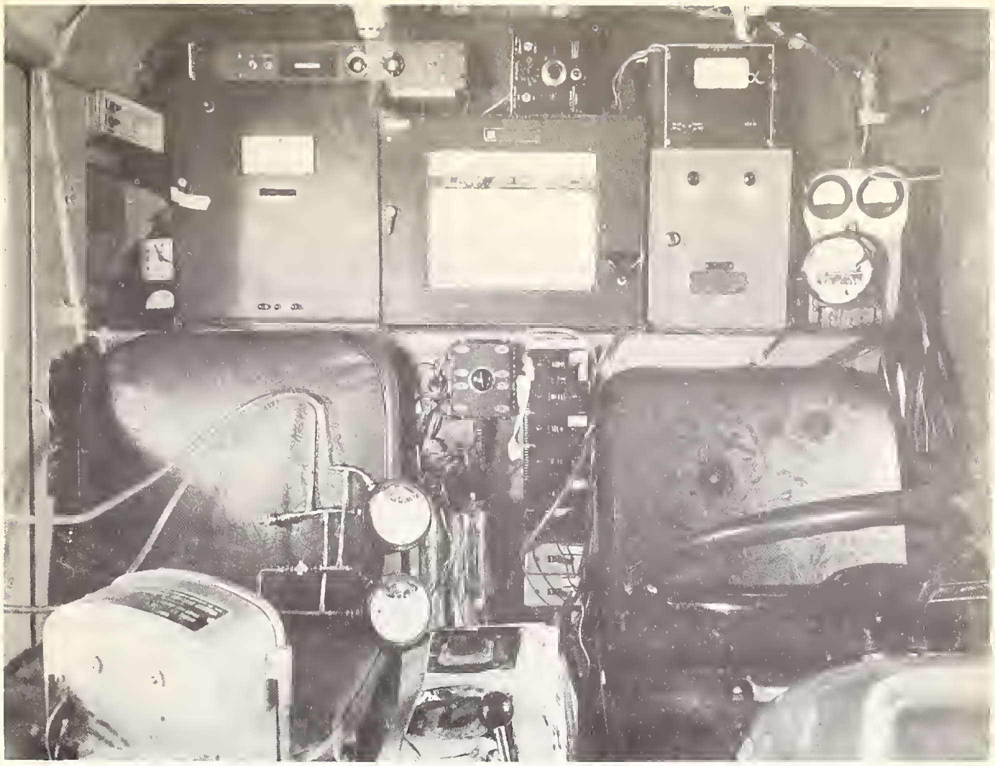

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Figure 8.-- Instruments and controls mounted inside the tractor cab for the road tests.

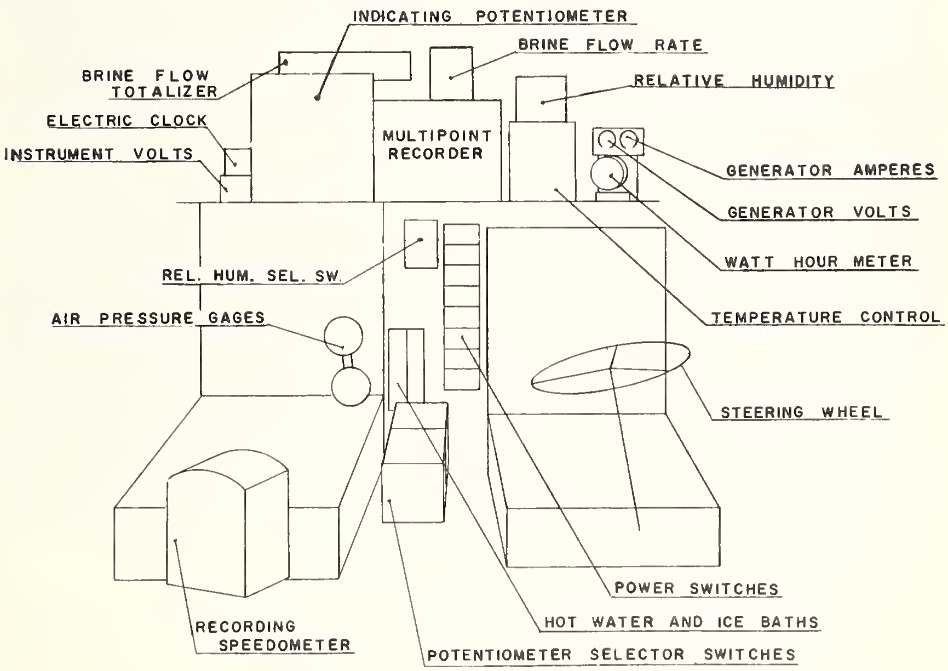

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Figure 9.-- Identification of the instruments and controls mounted inside the tractor cab as shown in figure 8.

In addition to those instruments mounted in the cab, a pyrheliometer for measurement of relative incident radiation, a Pitot tube for measurement of impact and static air pressures during motion, and a protective tube for the electric hygrometer elements were mounted on the cab roof (fig. 10).

Figure 11 shows, diagrammatically, the physical layout of all the components of the mobile test equipment.

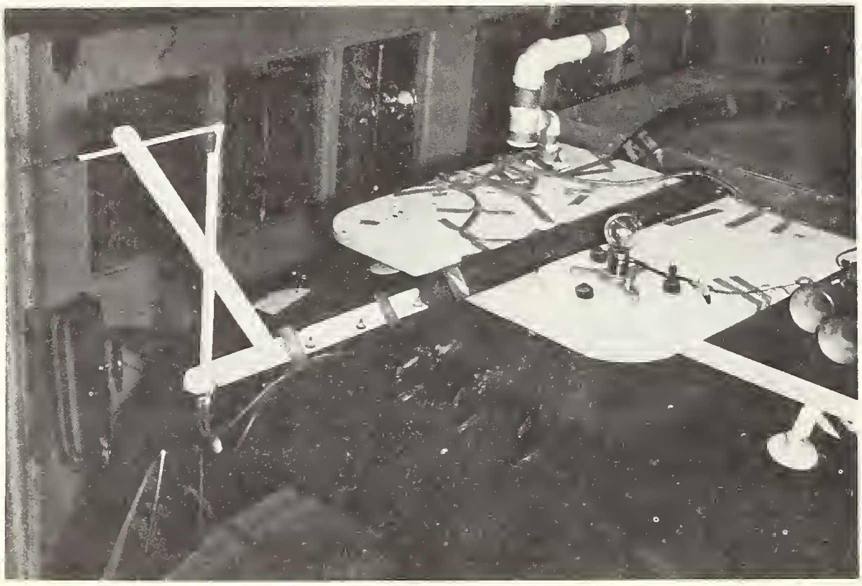

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Figure 10.-- Pyrheliometer, Pitot tube, and protective tube for the electric hygrometer elements mounted on the roof of the tractor cab.

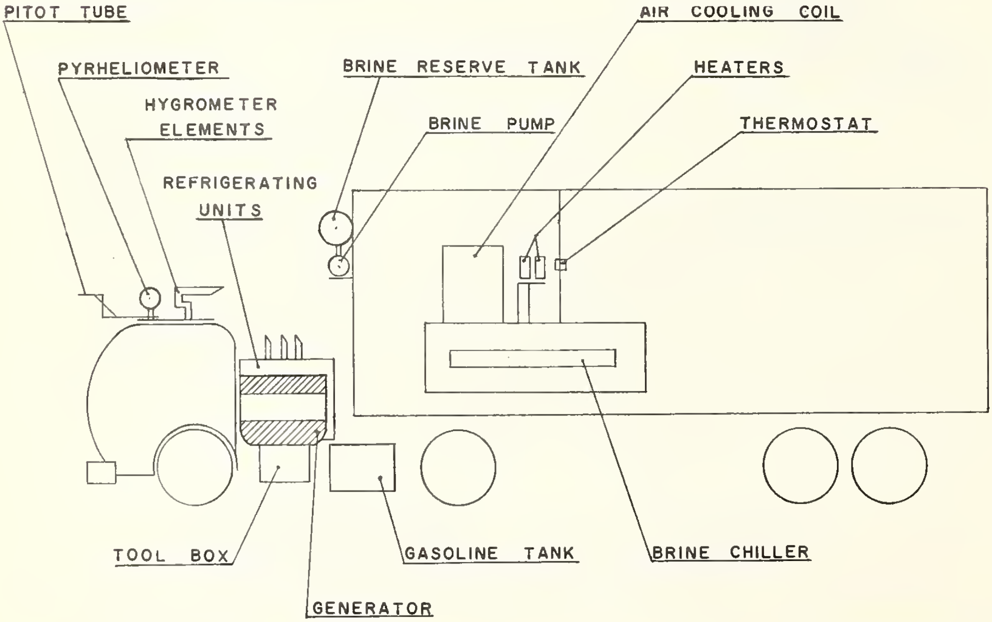

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Figure 11.-- Physical layout of all components used for the road tests.

DESCRIPTION OF THE TEST VEHICLES

Seven trailers were involved in the tests described in this report. Of these, four were commercial trailers and three were military vehicles.

Military Trailers

The three military vehicles were $7\frac{1}{2}$ ton, 21-foot, single-axle trailers manufactured to U. S. Army specifications. These trailers were studied for heat transfer and moisture transfer characteristics by the National Bureau of Standards for the Quartermaster Research and Engineering Command. Although the three vehicles differed slightly, figures 12 and 13 show, respectively, the front and rear of one which is typical. One of these vehicles was used in the study of air pressures on the surfaces and in the insulation and cargo spaces during operation on the road at 50 miles per hour. The military vehicles were also used for infiltration measurements. None was used for road tests to determine cooling load. The military vehicles had aluminum exterior and interior surfaces and had various forms of extruded or formed aluminum floors. Glass fiber insulation was used in the walls and roof, and expanded polystyrene was used in the floors, with insulation approximately 6 inches thick. The empty weights were about 7,300 pounds.

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Figure 12.-- Front and left side of one of the military vehicles used for the tests.

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Figure 13.-- Rear and right side of one of the military vehicles used for the tests.

Commercial Trailer A

This refrigerated semitrailer had been in commercial use before the tests (fig. 14). The hubodometer mileage was about 60,000 miles. The exterior of the trailer was 35 feet long, 8 feet wide, and approximately 8 feet 2 inches high. The outer skin was of riveted aluminum. The inside was approximately 34 feet long, 7 feet 3 inches wide, and 6 feet 11 inches high, above the drainage level of the floor. The empty weight was 13,510 pounds.

According to specifications furnished with the trailer, the subfloor of the vehicle consisted of 3/8-inch marine plywood, coated on both sides with a water-emulsion undercoating. Over the subfloor, 4 inches of rigid expanded polystyrene was placed with all joints sealed with the same type of emulsion, and over this insulation there was an aluminum-alloy interlocking-channel flooring. This aluminum flooring was supported on wood fillers over each frame cross member. The depth of the channel grooves in the flooring was 1 1/8 inches. The sidewalls contained glass fiber insulation 4 inches thick having a density of 3/4 pounds per cubic foot (lb./cu.ft.) with a vapor barrier of paper between the 2 layers of the insulation, except for the lower 10 inches of the walls which were insulated with rigid expanded polystyrene. The inner wall surface of the trailer consisted of 5/16-inch plasticized plywood except for the aluminum flashing at the floor. The roof was insulated with 6 inches of glass fiber insulation identical to that used in the walls with a vapor barrier of paper between the 2 layers of the insulation. The ceiling was 5/16-inch plasticized plywood. The construction of the rear door was the same as that of the sidewalls except that the flashing on the lower portion of the door was of stainless steel. The trailer was equipped with seven 2-inch pipe meat rails, each supported by 13 hangers attached to wooden cross beams. The three center rails were shorter than the four outside rails leaving a 45-inch space at the front end for the refrigeration unit.

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Figure 14.-- Commercial trailer A.

Commercial Trailer B

This refrigerated semitrailer was new. It was a single-axle trailer (fig. 15). The body length of the trailer was 33 feet 4 inches; the width, 7 feet 10 inches; and the height, 8 feet, all exterior dimensions. The inside length was 32 feet 4 inches; inside width, 6 feet 6 inches; and the inside height, 6 feet 7 inches. The empty weight was 9,722 pounds.

The exterior skin of the trailer was stainless steel of spotwelded construction, and the sidewalls were corrugated in the form of vertical exterior channels with a trapezoidal cross section. These channels were 1 inch wide across the outer face, 1 3/4 inches across the base, and the spaces between the bases of adjacent channels were 4½ inches wide. The sides and ceiling of the vehicles were insulated with 6 inches of glass fiber insulation, and the floor with 5 inches of rigid expanded polystyrene. The interior lining was plywood, and the floor was extruded, interlocking channel aluminum alloy with integral flashing. There did not appear to be any vapor barrier in the walls or ceiling other than the exterior skin. There was a curbside door 4 feet 4 inches wide at a distance of 12 feet 9 inches from the front of the body. There were no meat rails or bows in this trailer.

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Figure 15.-- Commercial trailer B.

Commercial Trailer C

This refrigerated semitrailer had travelled about 55,000 miles before being submitted for test. It was a two-axle trailer (fig. 16). This trailer was approximately 35 feet 10 inches long; 8 feet wide; and 8 feet 2 inches high, in overall exterior body dimensions. The trailer interior was approximately 33 feet 8 inches long, measured along the center lines; 7 feet wide; and 7 feet high. The outside skin of the trailer, including the doors, consisted of aluminum or aluminum alloy sheets. The front end was curved with two ventilation hatches near the top. The weight of the trailer was 13,410 pounds.

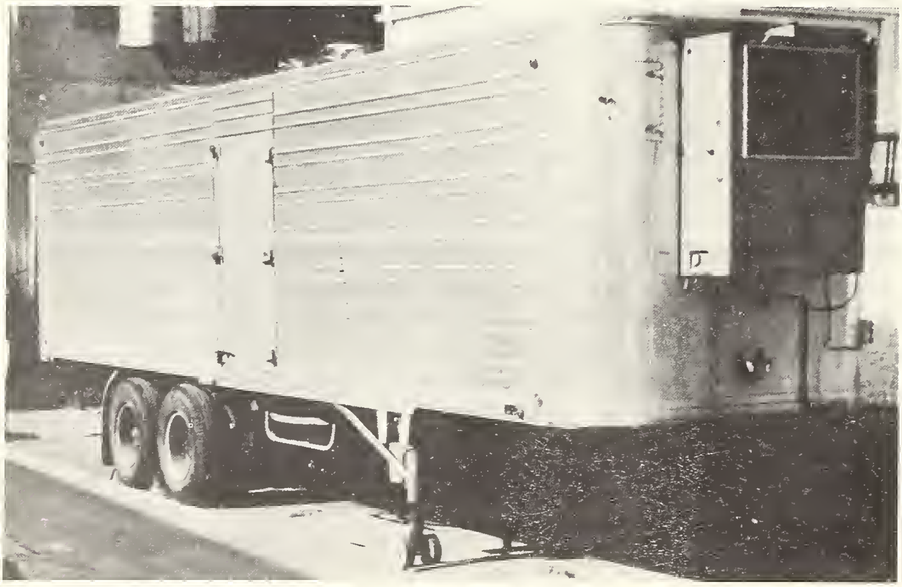

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Figure 16.-- Commercial trailer C.

The rectangular area of the floor was made up of extruded, interlocking channel sections of aluminum alloy with integral flashing on the outside sections. The curved area in the front was apparently made of aluminum alloy. The inner sides of sidewalls, doors, and hatch doors were lined with corrugated plastic sheets. The curved front end of the trailer had a treated plywood liner from the floor to the plug opening; the remainder was lined with aluminum. The ceiling was lined with treated plywood and had seven meat rails attached to the roof structure.

The insulation in the sidewalls and roof consisted of glass fiber batts, 6 inches thick. There was no evidence of a vapor barrier in the sidewalls other than the exterior skin. The type of insulation used in the floor was not determined. The inner liner was attached to the vertical members with blind rivets.

Commercial Trailer D

This refrigerated semitrailer was apparently new when furnished for laboratory and road tests. It was a two-axle trailer (fig. 17). This trailer was approximately 35 feet long, 7 feet 11 inches wide, and 8 feet 5 inches high in overall exterior dimensions. The inside was 34 feet long, 6 feet 11 inches wide, and 7 feet 1 inch high. The empty weight was 12,563 pounds.

NBS-10.3-6

Figure 17.-- Commercial trailer D.

The exterior skin was of riveted aluminum sheets and the interior lining was treated plywood. The walls and roof were insulated with 6 inches of glass fiber insulation; the floor with 6 inches of rigid expanded polystyrene. There was no evidence of a vapor barrier other than the exterior skin, but the specifications required that all interior joints of the shell were to be sprayed with an undercoat. There were 66 holes, ¼-inch in diameter, in the upper flange of the side channels at the base of the walls. These holes were provided to drain water from the insulation space in the walls. The floor was extruded, interlocking channel sections of aluminum alloy with integral flashing. There were cross members, or bows, for supporting meat rails, but no meat rails were provided.

Laboratory tests of each trailer were made to determine the cooling load and simultaneous gain in weight due to accumulation of water or ice under specified laboratory conditions for comparison with the cooling load under actual road conditions. The cooling load was determined with the heat sink apparatus, and the gain in weight during the test was measured with the three platform scales on which the trailer was supported.

For the cooling-load test, ambient conditions of 100° F. dry-bulb temperature and 50 percent relative humidity in the space around the trailer and a temperature of 0° F. in the cargo space were chosen as typifying moderately severe operating conditions. These test conditions already had a considerable measure of acceptance among trailer manufacturers and shippers as indicated by prior discussions with them.

In the heat sink apparatus, illustrated schematically in figure 18, chilled brine was pumped continuously through the closed loop incorporating the air cooling coil inside the trailer, the comparison brine heater outside

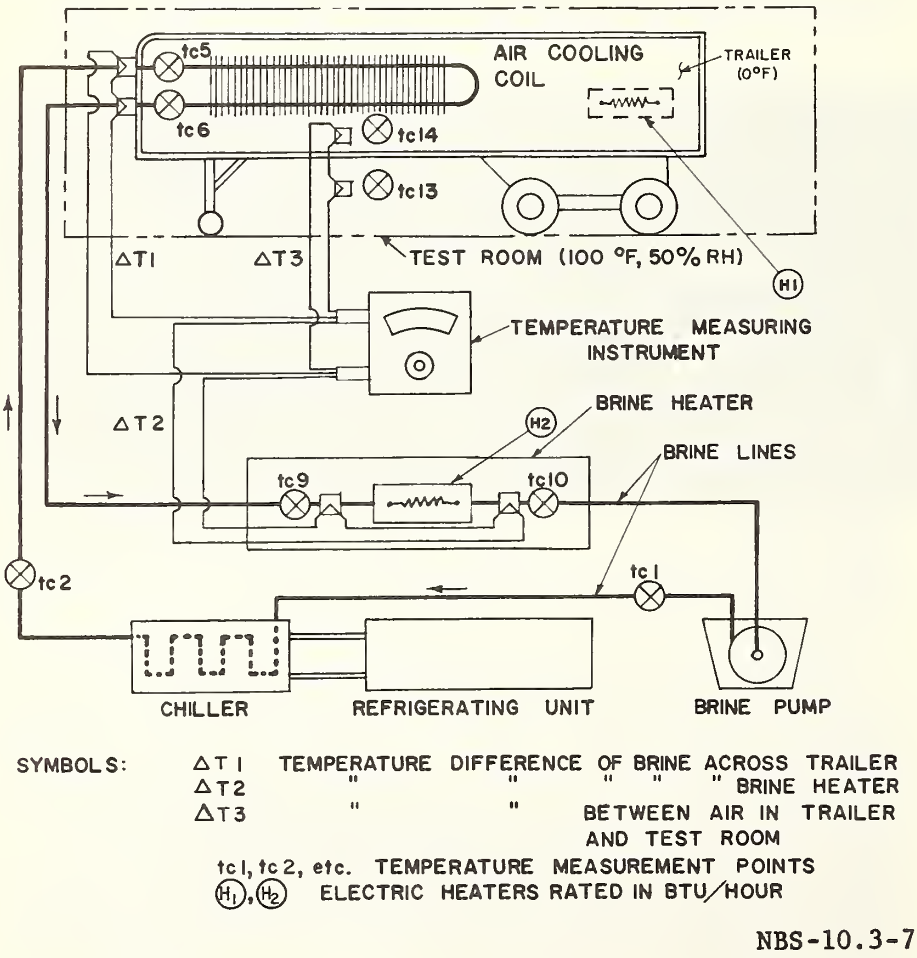

Figure 18.-- Schematic diagram illustrating the principle of operation of the heat sink apparatus.

the trailer, the brine chiller, and the circulating pump. The test was continued and all readings were taken at regular time intervals for a sufficient period to establish weight gain and heat transfer rates. This period ranged from 50 hours to 130 hours on the several trailers.

The following variables were controlled or measured during the laboratory tests:

1. Temperature and relative humidity of the ambient air around the trailer.

2. Temperature inside the trailer.

3. Temperatures at various points in the brine circuit:

   a. Inlet to the cooling coil in the trailer.
   b. Outlet from the cooling coil in the trailer.
   c. Difference across cooling coil in the trailer.
   d. Inlet to comparison brine heater.
   e. Outlet from comparison brine heater.
   f. Difference across comparison brine heater.

4. Electric energy input to comparison brine heater.

5. Electric energy input to trailer for thermostatted heater, fans, etc.

6. Weight of the trailer.

7. Time.

The change in specific heat of the brine between the coil in the trailer and the comparison heater was less than 0.1 percent, and the computed heat leakage through the insulation of the brine heater was less than 0.5 percent of the heat supplied electrically. These factors were neglected in these tests. Therefore, since the brine flow rate was the same in all parts of the circuit, the ratio of the temperature rise of the brine in the air cooling coil in the trailer to that in the brine heater was equal to the ratio of the heat absorbed in the coil to that absorbed in the heater. The equality of these two ratios can be expressed as follows:

$$\frac{\Delta T_1}{\Delta T_2} = \frac{\text{Heat Gain of Trailer} + H_1}{H_2} \qquad (1)$$

where H_1 is the sum of all of the items of auxiliary heat added to the trailer interior for purpose of test, such as fans, blowers, controlling heat, etc., British thermal units per hour (B.t.u./hr.).

H_2 = heat absorbed in the comparison brine heater, (B.t.u./hr.)

ΔT_1 = temperature rise of the brine in the air cooling coil inside the trailer, °F.

ΔT_2 = temperature rise of the brine from inlet to outlet of the comparison brine heater, °F.

The heat gain or cooling load of the trailer through its walls, floor, ceiling, air leakage, etc., would then be:

$$\text{Heat Gain (B.t.u./hr.)} = \left(\frac{\Delta T_1}{\Delta T_2} \times H_2 \right) - H_1 \qquad (2)$$

To determine the heat transmission of the trailer per unit of temperature difference between interior and exterior, the total heat gain determined from equation (2) can be divided by the temperature difference, ΔT_3, between the ambient air and the cargo space of the trailer.

$$\text{Heat Transfer Coefficient, (B.t.u./hr.) (°F.)} = \frac{\left(\frac{\Delta T_1}{\Delta T_2} \times H_2 \right) - H_1}{\Delta T_3} \qquad (3)$$

The total weight gain of the trailer was determined by periodic observation of the platform scales on which it was supported. The portion of the weight gain due to accumulation of frost or ice on the cooling coil was determined by weighing the defrost water at the end of each test. The scales used to weigh the trailer were read to the nearest one-half pound; they had previously been calibrated through the range of weights involved.

The air movement through the trailer walls and the heat transfer to the cooling coil as a result of air leakage was calculated from the weight gain observed during the test. For this computation, it was assumed that the weight gain was due entirely to accumulation of water or ice from the moist air entering at ambient conditions and leaving saturated at 0° F. Achenbach and Phillips 6/ have shown in another experiment that diffusion could account for only a very small fraction of the total transfer of moisture through leaks in the exterior skin of a refrigerated trailer. Based on these assumptions, humidity ratios taken from the ASHAE Psychrometric 7/ chart and the thermodynamic properties of water show that each pound of air that moved through the trailer body would surrender 140.5 grains or 0.02007 pounds of water. It can be seen that these assumptions would establish the minimum movement of air that could produce the observed rate of weight gain for the existing ambient and cargo space conditions.

6/ See footnote 3.

7/ Goff, J. A., and Gratch, S. Thermodynamic Properties of Moist Air. Table 2, pp. 18-21. (In Heating, Ventilating, Air Conditioning Guide, published by American Society of Heating and Air Conditioning Engineers.) 1959.

It is possible that the leakage air leaving the trailer was not cooled to 0° F. and that it was not saturated. In the event that the amount of water vapor surrendered per pound of leakage air was less than that established by assuming that it was saturated at 0° F. as it left the trailer body, the corresponding air leakage would have been greater than the minimum value. For example, if the leakage air were assumed to leave the trailer saturated at 32° F. instead of 0° F., the amount of moisture surrendered would be 119.5 grains per pound of air instead of 140.5 grains. In this case, the computed air leakage would be about 18 percent higher than was calculated for 0° F., and the total cooling load attributable to air leakage about 90 percent of that corresponding to the minimum air leakage value.

The portion of the total cooling load of the trailer due to air leakage was calculated. The cooling load due to air leakage was divided into two components: That due to the dry air, and that due to moisture. The dry air component was computed from the change in enthalpy per pound of dry air for the assumed change in dry-bulb temperature.

The component due to moisture was calculated from the weight gain of the trailer during the test, the difference in enthalpy of the water vapor as it entered the trailer and the enthalpy of frost at 0° F., plus the change in enthalpy of the small amount of water vapor that passed through the trailer body without condensation.

The portion of the total cooling load attributable to transmission, that is, convection, conduction, and radiation through walls, floors, and ceiling, was determined by subtracting the total cooling load caused by the computed air leakage from the total measured cooling load.

LABORATORY TEST RESULTS

The results of the laboratory tests on the four commercial trailers are shown in table 1. In reporting the test results the four trailers have been designated numerically as 1 to 4, whereas in their description they were designated alphabetically from A to D, in a different sequence.

The observed rates of weight gain, and cooling loads for the four trailers are summarized in table 1. The values reported include the frost accumulated both in the insulation spaces and on the cooling coil. The gain in weight was approximately constant during the few days comprising each test. The observed weight gain shown in table 1 for trailer 4 was corrected by computation to compensate for excessive air leakage caused by a hole on the under side of the trailer that was exposed to a frontal air pressure of 0.015 inches Water Gage (in. W.G.) during the test. This correction was made on the basis of the relative weight gain observed during the two tests of the same trailer at a higher level of frontal air pressure. One test was made with the hole on the under side of the trailer exposed to the air pressure and the other with this hole outside the pressurized zone.

Table 1.-- Weight gain rate and cooling loads of four commercial trailers under laboratory conditions 1/

Trailer	Average weight gain rate	Observed total cooling load	Computed air leakage rate	Computed heat transfer rate due to air leakage			Computed heat transfer rate due to transmission
				Total	Dry air	Moisture	
	Lb. per hr.	B.t.u. per hr.	Cu. ft. per hr.	B.t.u. per hr.	B.t.u. per hr.	B.t.u. per hr.	B.t.u. per hr.
1........	0.98	9,520	710	2,410	1,170	1,240	7,110
2........	0.32	10,300	230	790	380	410	9,510
3........	0.54	7,500	390	1,330	650	680	6,170
4........	0.52	9,500	380	1,280	620	660	8,220

1/ Ambient 100° F., 50 percent RH, interior 0° F.

The observed cooling loads of the four commercial trailers tested under laboratory conditions with a cargo space temperature of 0° F. and ambient conditions of 100° F. dry-bulb temperature and 50 percent relative humidity are shown in figure 19. Trailers 1, 2, 3, and 4 are shown to have total cooling loads of 9,520, 10,300, 7,500, and 9,500 B.t.u./hr., respectively. The portion of the total cooling load due to air leakage effects was computed to be 2,410, 790, 1,330, and 1,280 B.t.u./hr., or 25.3, 7.7, 17.8, and 13.5 percent of the total, respectively. Slightly over half of these latter amounts were due to the moisture removed from the air and deposited as frost. The minimum air-leakage rate that could have deposited the observed moisture is shown in table 1.

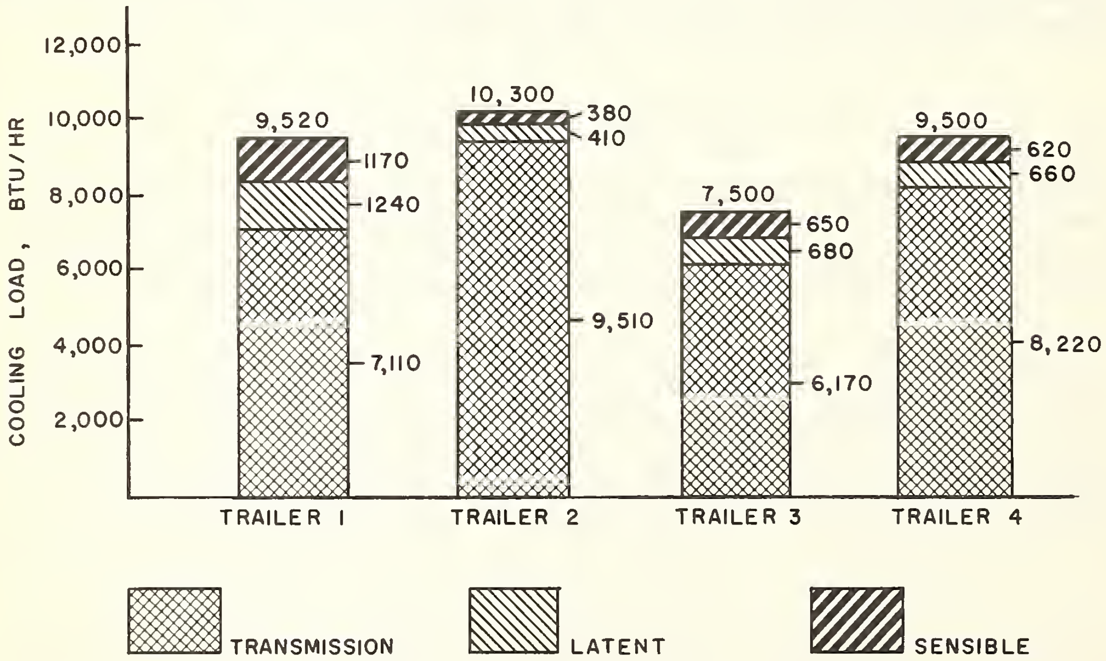

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Figure 19.-- Total incremental cooling loads of four commercial trailers under laboratory conditions. Ambient temperature 100° F., ambient relative humidity 50 percent, cargo space temperature, 0° F.

If the 18 percent higher air-leakage rate, calculated on the basis of air leaving the trailer saturated at a temperature of 32° F., had been used, the heat transfer from the dry air would have been about 20 percent less than that shown in table 1. The latent heat gain would be essentially the same in either case since it is directly related to the average gain in weight of the trailer. It is probable that the leakage air actually left the trailer at a dew point somewhere between 0° F. and 32° F. since most of the moisture was deposited as frost in the walls at places where the temperatures were probably several degrees above 0° F.

The ratio of the computed heat transmission rate shown in the last column of table 1 to the computed heat transmission rate based on handbook thermal conductance factors of the materials of construction ranged from 1.29 to 1.62 for the four trailers. In making the latter computation of the heat transmission rate, no increase in heat transfer was assumed for the framing members of the trailers because the details of construction were unknown.

ROAD TEST PROCEDURES

Road tests of the four commercial trailers were made primarily to determine the heat transfer rate of each trailer while being operated at 50 miles per hour (m.p.h.) for comparison with the laboratory results when the vehicles were stationary. The trailers were empty except for test equipment.

Road tests were made on the Ohio Turnpike during September and October 1957 and October 1958. This location was selected because it was the nearest to Washington, D. C., that met the desired conditions for continuous operation at 50 miles per hour. The tests were conducted from Milan, Ohio, which is located at about the center of the Turnpike.

A typical test run was of approximately 5 hours and 30 minutes duration for each driver and observer team, including turn-around and refueling. The distance traveled was 119 miles each way from Milan to the western terminus of the Turnpike and return, a total of 238 miles, and the 50-mile-per-hour road speed was maintained except for the time required to turn around at the far end of the run and to refuel at Milan.

The time interval between runs was determined by the time required to change drivers, refuel and check oil, brakes, lights, etc., for the tractor-trailer and the three gasoline engines used to drive the generator and refrigeration unit. A retail gasoline station used as a change point was located less than ¼ mile from the Milan entrance to the Turnpike and, usually, the time off the highway for refueling did not exceed 20 minutes. On the basis of 10 minutes for the turn-around and a total of 30 minutes for leaving and reentering the highway, and refueling, about 87 percent of the elapsed time for each test run of 238 miles represented operation at 50 miles per hour. For a majority of the tests, the drivers were able to control the speed to within ± 2 miles per hour during the 50-mile-per-hour portions of the run. As far as possible, the test runs were continuous and deviated from this schedule only when repairs and maintenance were required between runs.

All test runs were made with the trailer interior temperature controlled as near as possible to 0° F., regardless of variation in ambient temperature, relative humidity, and incident radiation. As outlined in "Description of the Road Test Equipment," provisions were made for observing a number of variables, such as skin temperatures, static and impact air pressures on various parts of the trailer, relative incident solar or sky radiation, temperature and relative humidity of the ambient air, road speeds, brine flow, etc. The observer read and recorded measurements of all variables at 30-minute intervals; a complete set of readings required about 20 minutes. The tractor-trailer combination was weighed at the beginning and end of the road tests of each trailer with a full tank of gasoline in every instance. The total distance involved in the road tests of the four vehicles was about 20,000 miles.

The refrigerating effect at the coil in the trailer was determined by means of a flowmeter in the brine circuit and the temperature difference of the brine entering and leaving the coil. The heat transfer rate (cooling load) per unit of temperature difference across the walls of the trailer during each test run was computed using the following equation:

$$\text{Heat Transfer Rate, B.t.u./hr.(°F.)} = \frac{(M \times Cp \times \Delta T_1) - H_1}{\Delta T_2} \qquad (4)$$

where M = flow of methylene chloride brine, pounds per hour

Cp = specific heat of brine, B.t.u. per pound (°F.)

ΔT_1 = temperature rise of brine through the cooling coil in trailer, °F.

ΔT_2 = average temperature difference between air in trailer and ambient air, °F.

H_1 = heat equivalent of electric energy released in trailer for fans and heater, B.t.u. per hour

The heat transferred to the chiller shell from the air in the trailer would reduce the measured cooling load of the vehicle. The chiller was surrounded by sufficient insulation to reduce this heat transfer to less than 1 percent of the measured cooling load, so it was omitted in equation (4).

For a comparison of laboratory and road test results, cooling loads for the road tests were extrapolated by calculation to an ambient temperature of 100° F. and 50 percent relative humidity instead of the actual ambient conditions under which the tests were made. These computed values were obtained by the following steps:

1. The heat-transmission rate, exclusive of air leakage, was assumed to be the same for road operation as for laboratory operation at the same temperature difference and was calculated for the temperature difference observed during the road tests.

2. The remainder of the cooling load observed on the road was assumed to be caused by air leakage at 50 m.p.h. road speed.

3. The air leakage rates required under the ambient conditions observed on the road to cause the increment of heat transfer in excess of heat transmission (item 2 above) were computed assuming that the leakage air left the trailer saturated at 0° F.

4. The heat transfer rate due to air leakage at ambient conditions of 100° F. and 50 percent relative humidity and 50 m.p.h. road speed was computed on the assumption that the leakage rate at a given road speed (50 m.p.h.) was independent of ambient temperature and humidity, that is, that the leakage rate in cubic feet per hour was constant if the road speed was constant.

5. This computed heat transfer rate due to air leakage was added to the heat transmission rate (exclusive of leakage) determined in the laboratory for 100 degrees temperature difference between inside and outside the trailer to determine the total anticipated cooling load under road operation at 50 m.p.h. in ambient conditions of 100° F. and 50 percent relative humidity.

It is recognized that the heat transmission rate for a given air temperature difference was probably greater during the road tests than under laboratory conditions because of solar radiation, heating of the under surface of the floor by rejected engine heat, and an improved heat-transfer coefficient on all surfaces due to the higher air velocity caused by the vehicle motion. But computation shows that these effects are small compared to the increase due to air leakage.

ROAD TEST RESULTS

Most of the heat transfer values reported are the averages for a period of continuous test operation, not less than 24 hours in length, for which the initial and final ambient temperatures and incident solar radiation were not far different. It must be understood that ambient temperature and humidity and solar radiation varied during each test and from one test run to another, so that steady heat and moisture transfer were not attained, even though the interior temperature was held essentially constant.

Figure 20 shows the heat transfer rate of the four test vehicles. Because the ambient conditions were not identical, the heat transfer rates, to be comparable, are expressed in B.t.u. per hour per degree F. (B.t.u./hr. (°F.)) air temperature difference between interior and exterior of the trailer. From figure 20 it can be seen that trailers 1, 2, 3, and 4 had heat transfer rates of 112.9, 116.6, 88.3, and 103.6 B.t.u./hr. (°F.), respectively.

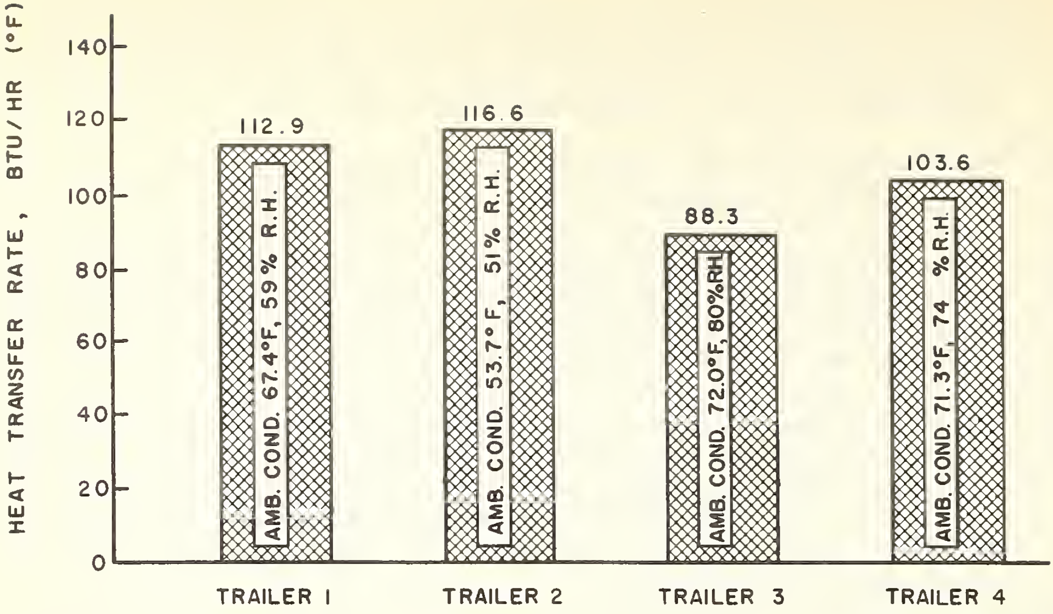

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Figure 20.-- Average heat transfer rate of four commercial trailers at a road speed of 50 m.p.h. under prevailing ambient conditions. Cargo space temperature 0° F.

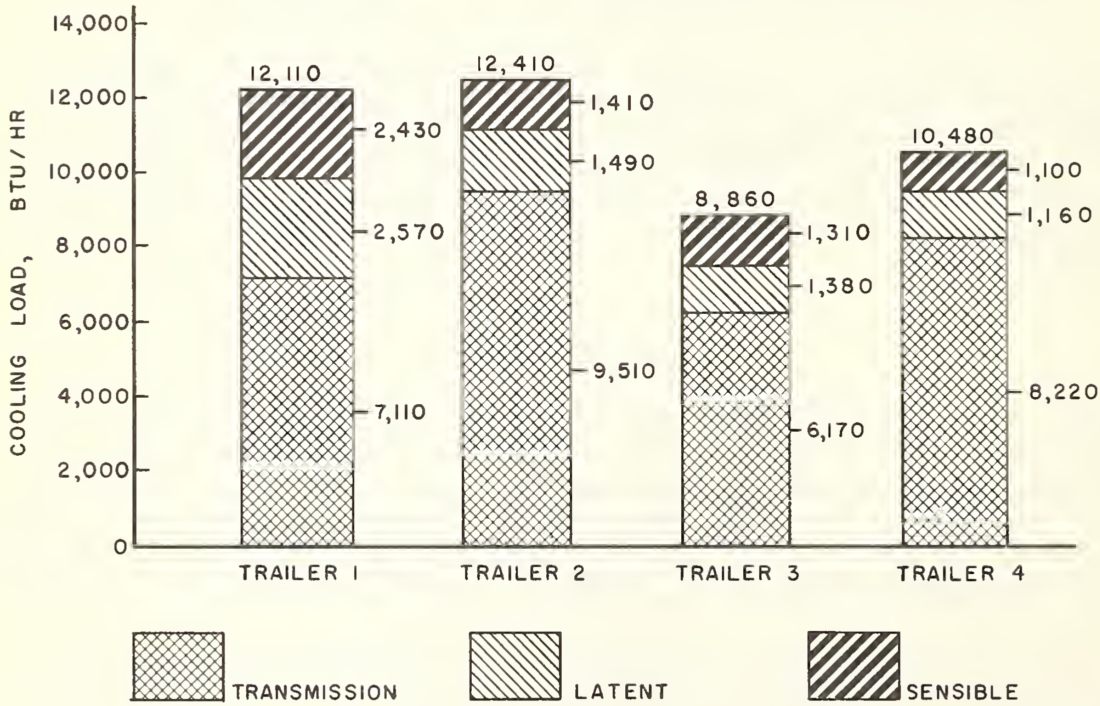

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Figure 21.-- Computed total and incremental cooling loads of four commercial trailers at a road speed of 50 m.p.h. extrapolated to ambient temperature of 100° F. and relative humidity of 50 percent. Cargo space temperature 0° F.

Figure 21 shows the computed cooling loads of the four test trailers when operated at a road speed of 50 miles per hour for ambient and cargo space conditions equal to those used for the laboratory tests; that is, 100° F. ambient temperature at 50 percent relative humidity, and an interior temperature of 0° F., based on the assumptions cited in "Road Test Procedures." The calculated air leakage rates required under the ambient conditions observed on the road to produce the computed additional heat transfer in excess of heat transmission were 109.5, 65.7, 57.8, and 48.7 pounds per hour (lb./hr.), equivalent to about 1,475, 860, 790, and 660 cubic feet per hour (cu.ft./hr.) for trailers 1, 2, 3, and 4, respectively. The corresponding increments of cooling load produced by this leakage at an ambient temperature of 100° F. and an ambient relative humidity of 50 percent would be 5,000, 2,900, 2,690, and 2,260 B.t.u./hr. The total computed cooling loads for road operation under these same ambient conditions, therefore, would be the sums of these quantities and the values for heat-transfer rate due to transmission in the last column of table 1, resulting in totals of 12,110, 12,410, 8,860, and 10,480 B.t.u./hr., respectively, for trailers 1, 2, 3, and 4, (fig. 21).

Figures 22 and 23 show the corresponding ambient humidities and heat transfer rates for the several test runs of trailers 2 and 3. In figure 22, the ambient absolute humidity was of the same order of magnitude for all test runs, and there was no trend toward a rising or falling heat transfer rate for the series of runs. The heat transfer rate for run 12 on trailer 2 was low because the interior temperature of the trailer rose about 4 degrees during the run; therefore, this value was not plotted in figure 22. In figure 23, the ambient absolute humidity decreased significantly during the 19 test runs, and the measured heat transfer rate also showed a significant corresponding decrease.

Each of the four test vehicles gained weight during its week of road testing; the increase ranging from 130 to 230 pounds. Since operating and ambient conditions varied considerably for the four trailers and because exterior accumulations of dirt and moisture would affect the precision of the results, rates of moisture gain cannot be established from the weight values. The results confirm, however, that a significant weight gain, primarily because of accumulation of frost and moisture, does occur. For two of the test trailers, which were refrigerated below 20° F. for the entire period between initial and final weighing in Washington, D. C.--an average time of eight days--the cooling coil in the trailer was not defrosted until after the final weighing. The amount of ice on the coil did not exceed 15 pounds in either case. The doors on these two trailers were not opened between weighings.

Although the tests were not intended to include detailed studies of skin temperatures, the temperature at one point on each of the exterior surfaces of the two sides, floor, and roof was observed. Figure 24 shows observations made during one of the return trips from the Ohio Turnpike when it was convenient to stop just before noon with a clear sky and bright sun. Although the sky clouded over somewhat between noon and 2:00 p.m., certain comparisons can be made from figure 24.

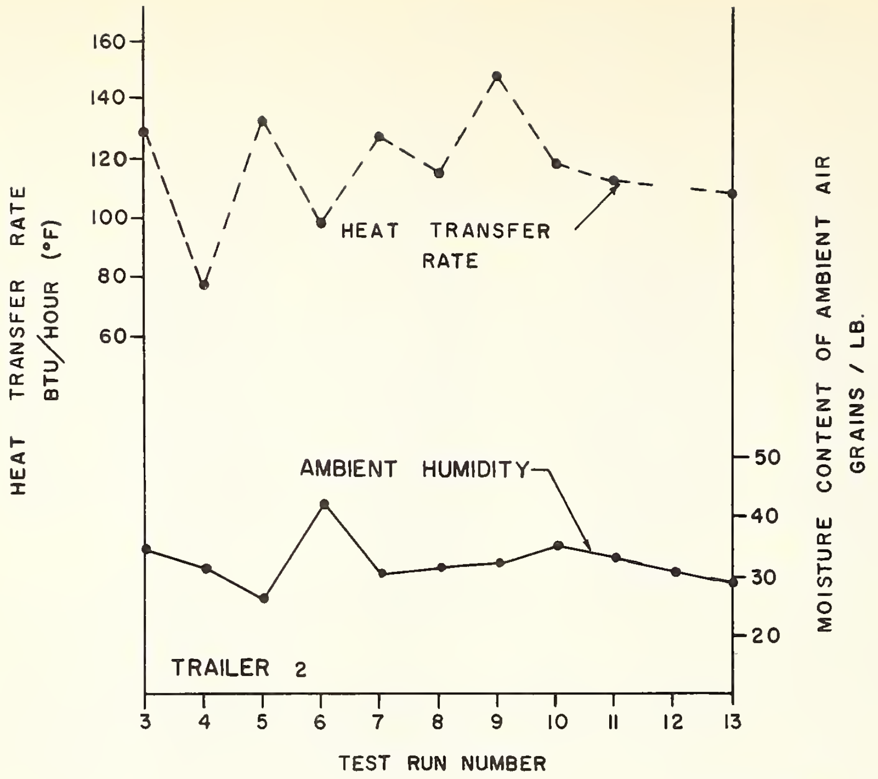

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Figure 22.-- Relation of heat transfer rate to ambient absolute humidity during road tests for trailer 2.

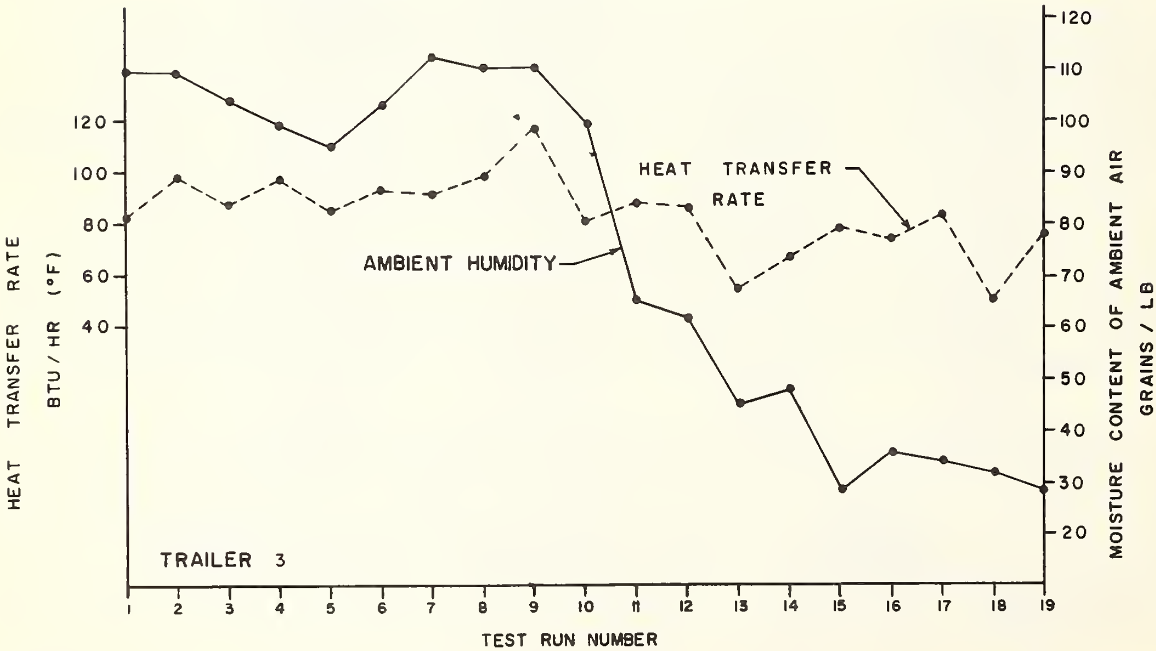

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Figure 23.-- Relation of heat transfer rate to ambient absolute humidity during the road tests for trailer 3.

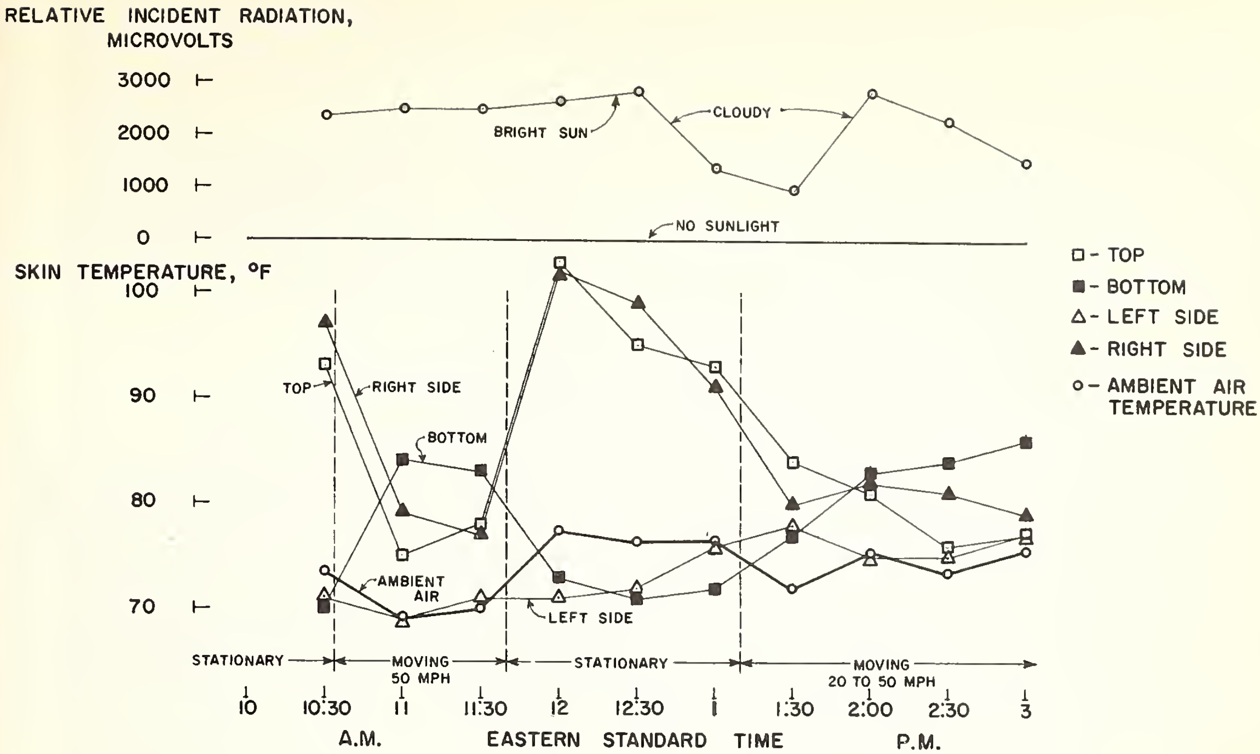

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Figure 24.-- Effect of solar radiation and road speed on exterior surface temperatures of one of the commercial trailers.

The top graph shows relative incident radiation measured with a pyrheliometer, and is expressed in microvolts. Bright sunlight gave a reading of nearly 3,000 microvolts; no sunlight gave a reading near zero. The temperature of the top and right side surfaces decreased and the temperature of the bottom surface increased simultaneously, when travel at 50 miles per hour commenced about 10:35 a.m. The reverse occurred when the vehicle was stationary during the times before and after this period. The trends during the period when the vehicle was again in motion after 1:10 p.m. are not as conclusive because of intermittent cloudiness and change of speed between 20 and 50 miles per hour, including the two stops involved in leaving the Ohio Turnpike and entering the Pennsylvania Turnpike between 2:30 and 2:40 p.m. While figure 24 shows data from only one of the test vehicles, it was noted that all of the vehicles showed similar characteristics. The bottom surface temperature of the trailer was from 5° to 15° F. warmer than the ambient air temperature whenever the vehicle was in motion, day or night, in rainy or dry weather.

During the road tests, the velocity pressure of the air above and slightly ahead of the leading edge of the tractor was measured with a Pitot tube. In addition, the differences between the static pressures in the trailer interior and inside the trailer wall and that at the Pitot tube were measured. The velocity pressure of dry air at a temperature of 70° F. and a pressure of 29.92 inches of mercury (in. Hg.) moving at 50 miles per hour is very nearly 1.21 in. W.G. During the test runs at 50 miles per hour, the observed velocity pressures ranged from 0.6 to 2.0 in. W.G. depending primarily on the velocity and direction of the wind. When the wind was quiet, the observed velocity pressures at 50 miles per hour ranged from 1.1 to 1.25 in. W.G.

The observed static pressure in the cargo space of all the test vehicles at 50 miles per hour ranged from 0.2 to 0.4 in. W.G. lower than the static pressure at the Pitot tube in front of the tractor. The difference observed between the static pressure at the Pitot tube and the static pressure at the one point of observation in the trailer wall was essentially the same as that for the trailer cargo space.

COMPARISON OF LABORATORY AND ROAD TESTS RESULTS

As described previously different forms of the heat sink method were used to measure the total heat transfer of the trailers during laboratory and road tests. A test was made to determine the agreement between these different and completely independent measuring systems: The stationary laboratory equipment using the comparison principle, and the mobile equipment using the flowmeter principle. This comparison was accomplished by measuring the cooling load with the laboratory equipment and again with the mobile equipment, using the same trailer in each case, exposed to identical ambient and interior conditions in the laboratory.

To test the trailer in the laboratory with the mobile equipment, the trailer was placed in the test chamber with the front end near the overhead sliding door. The tractor was uncoupled and moved forward as far as the electrical and refrigeration lines would permit, and the overhead door lowered as far as possible. The open spaces below and around the door were closed off so the desired ambient conditions could be maintained in the test chamber. By placing the tractor outside the test area, the heat produced by the gasoline engines driving the electric generator and the two refrigeration machines could be dissipated to the outside atmosphere without affecting the conditions in the test chamber.

All recorded test data were taken from measurements and observations made with the road test instruments in the tractor cab. All of the electrical energy and refrigeration required for the test was produced by the equipment mounted on the tractor, just as had been done during the road tests. Only the temperature and humidity conditons of the test chamber were controlled by laboratory equipment. The test room ambient temperature was controlled at 100° F. and the relative humidity at 50 percent. The temperature inside the trailer was controlled at 0° F.

The heat transfer rate of the trailer, as determined by this test, was within 1 percent of the rate determined by the laboratory test. This small deviation, which is less than the expected testing error, indicated that the two measuring systems, laboratory and mobile, were suitable for comparison of laboratory and road results.

The observed heat transfer rates of trailers 1, 2, 3, and 4, were 95.2, 103.0, 75.0, and 95.0 B.t.u./hr. (°F.), respectively, at ambient laboratory conditions of 100° F. temperature and 50 percent relative humidity and a trailer temperature of 0° F. The observed heat transfer rates of the four trailers, when operated on the road at a speed of 50 m.p.h., were 112.9, 116.6, 88.3, and 103.6 B.t.u./hr. (°F.) (see p. 15, fig. 14). The road and laboratory results are compared in table 2.

Table 2.-- Average observed cooling loads and heat transfer rates of four commercial trailers during laboratory and road tests 1/

Trailer	Laboratory tests 2/		Road tests			
	Cooling load	Heat transfer rate	Ambient conditions: Temperature	Ambient conditions: Relative humidity	Cooling load	Heat transfer rate
	B.t.u. per hr.	B.t.u. per hr. (°F.)	°F.	Percent	B.t.u. per hr.	B.t.u. per hr. (°F.)
1	9,520	95.2	67.4	59	7,610	112.9
2	10,300	103.0	53.7	51	6,260	116.6
3	7,500	75.0	72.0	80	6,360	88.3
4	9,500	95.0	71.3	74	7,390	103.6

1/ Temperature inside trailer 0° F.
2/ Ambient conditions: Temperature 100° F., relative humidity 50 percent.

Differences in ambient humidity, temperature and solar radiation, etc., during the several road tests, undoubtedly affected the precision of the heat transfer rates determined by dividing the observed cooling loads by the difference in temperature inside and outside the test vehicles. The heat transfer rates so determined, however, are probably the most suitable way to compare the overall performance of vehicles when ambient conditions vary between tests. From table 2, it can be calculated that the increases in heat transfer rates during road tests for trailers 1, 2, 3, and 4 were, respectively, 18.6 percent, 13.2 percent, 17.7 percent, and 9.1 percent of those observed during the laboratory tests.

The increase in cooling load of these trailers during road operation was due primarily to greater air leakage when the vehicles were moving. But some of the increase was caused by (1) solar radiation, (2) heating of the under surface of the vehicle, primarily by the waste heat from the tractor engine but partly from radiation from the road surface, and (3) an improved heat transfer coefficient at all exterior surfaces because of the higher air velocity. The increase in heat transfer rate due to these factors other than air leakage will probably not average more than 5 or 6 percent for a 24-hour period with bright sunlight during the day, when the trailer is in motion. When the trailer is in motion, the under side of the vehicle exhibits the greatest rise in temperature because of the location of the engine exhaust at the tractor rear axle and the waste heat from the tractor, generator, and refrigerating unit engines (fig. 24).

Figure 24 shows that the top and one side can be heated by solar radiation to a temperature at least 25° F. higher than that of the ambient air when a trailer is stationary in bright sunlight. The top and one side of a typical 35-foot trailer is about 45 percent of the total surface. The effect of increasing the temperature of these surfaces 25° F. is equivalent to increasing the temperature around the entire vehicle by about 12° F., and the temperature difference by 12 percent at conditions of 100° F. ambient temperature and 0° F. trailer temperature, but intense solar radiation would not exist for more than about 6 to 8 hours per day. This increase is offset in part by the decrease in temperature of the under surface when the vehicle is stationary.

The method used to extrapolate the cooling loads of the four test vehicles to conditions of 100° F. ambient temperature, 50 percent relative humidity, 0° F. trailer temperature and a road speed of 50 miles per hour was outlined on page 25 and the extrapolated values are shown in figure 21. Figure 25 is a comparison of the observed laboratory and extrapolated road cooling loads at these conditions, and is based on the assumption that air leaving the trailer was saturated and at a temperature of 0° F. From figure 25, it can be determined that the expected increase in cooling loads between similar laboratory and road conditions would be 27.2, 20.5, 18.1, and 10.4 percent for trailers 1, 2, 3, and 4, respectively.

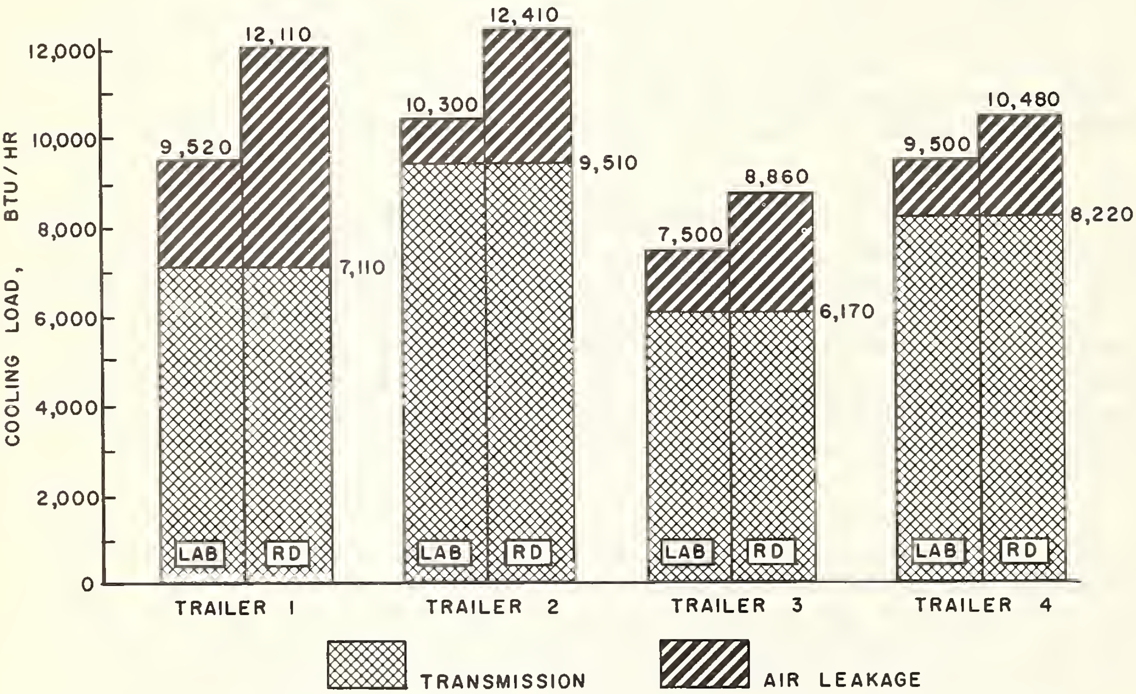

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Figure 25.-- Comparison of observed laboratory cooling loads and extrapolated road cooling loads at 50 m.p.h. for ambient temperature of 100° F., ambient relative humidity of 50 percent, and cargo space temperature of 0° F.

AIR LEAKAGE

Comparison of the cooling loads observed during the laboratory and road tests indicated that the leakage of air into the insulation spaces was a significant factor in increasing the heat transfer rate on the road. Earlier laboratory tests of one of the three military trailers had shown that the air leakage also was responsible for a steady deposition of frost in the insulation and a steady flow of sensible and latent heat into the trailer.

A number of studies were made to determine the location of the leaks in a trailer body, to measure the amount of air leakage, to study the pressure pattern around a moving trailer, and to find a suitable way to simulate in the laboratory the impact air pressure on the front end of a trailer when it was towed on the highway.

Smoke Studies

Smoke tests were made of several trailers to locate in a qualitative way where there were openings in the external envelope of the vehicle. For this purpose, a smoke bomb that produced copious quantities of white smoke was lighted inside the trailer. After closing the door the interior of the trailer was pressurized to about 1 inch W.G. by a centrifugal blower whose discharge penetrated the plug at the front of the trailer. Under these conditions, leaks were indicated by wisps or streams of smoke emanating from the skin of the vehicles. These smoke tests indicated the following results:

(1) The roofs revealed no leaks.

(2) The greatest leakage occurred underneath the trailers where the walls joined the floor structure.

(3) Leakage usually occurred where lights or reflector buttons were attached, or where electrical conduits or other lines penetrated the outside skin.

(4) Some leakage occurred at door gaskets, but usually in isolated sections.

(5) Some leaks occurred at spot-welded lap joints in the sidewalls and where fluted or corrugated side panels joined other panels at right angles.

The smoke test does not indicate the magnitude of the air leakage, but it is a simple and inexpensive way to determine what parts of the construction are responsible for air leakage.

Tracer Gas Infiltration Studies

An infiltration meter based on the tracer gas principle was designed and constructed for measuring the air leakage into trailer bodies. However, this apparatus could only be used to measure the leakage into the cargo space of the trailer; it was not suitable for measuring air that moved through the insulated cavities of the vehicle without entering the cargo space. The results of the tracer gas studies indicated that the air exchange between the cargo space and the outside was a small fraction of the air exchange through the insulation spaces as computed from the laboratory and road tests.

Air Pressure on Moving Vehicles

One of the military trailers was used to determine the air pressure patterns around a moving trailer. A Pitot tube was mounted in front of a military tractor and 44 small holes were drilled at selected places on the outside and inside skin of the trailer. The holes were connected to a manifold so that the pressure at each could be compared with the inside pressure of the trailer and also with the ambient outside pressure as measured by the Pitot tube. Stations for pressure observations were located on the trailer body as follows:

1 on the inside
2 on the front end
6 on one side and 2 on the other
2 on the rear
7 on the roof
2 on the floor.

Except for the station inside the trailer, openings were provided at each station which permitted measuring the total pressures on the exterior skin of the trailer and in the insulation space. Pressures were measured at each station while the trailer was pulled down the highway at a nearly constant road speed of 50 m.p.h. The following conclusions were derived from the data:

(1) The pressure on the front end of the trailer was nearly equal to the impact pressure on the Pitot tube except where the front of the trailer was shielded by the tractor. This pressure approximated 1 in. W.G. at 50 m.p.h.

(2) The interior trailer pressure was approximately ¼ in. W.G. lower than the static pressure at the Pitot tube in front of the tractor.

(3) The insulation space acted essentially as a constant-pressure plenum, ranging from 0 to 0.06 in. W.G. below interior trailer pressure.

(4) The pressure difference between exterior skin and insulation space was such as to continuously drive air into the insulation space on the front end and rear of the trailer and on the rear five stations on the roof. At the other stations on the sides, floor, and roof, the exterior skin pressure ranged from 0.07 in. W.G. below to 0.05 in. W.G. above that in the insulation space. The pressure difference at some of these latter groups of stations fluctuated from positive to negative during the test.

This study revealed that openings in the exterior skin at the front end of the trailer would cause more air leakage than at other locations during road operation because the greatest pressure difference existed across the front of the trailer. The pressure difference between the exterior surface and the insulation space was approximately 1.5 in. W.G. on the front end when the trailer was pulled down the highway at 50 m.p.h. on a day without wind. At the same time the pressure difference between the exterior surface and the insulation space for most of the rest of the trailer was less than 0.1 in. W.G. and frequently changed direction.

Air Leakage Simulation with Exhaust Blower

An attempt was made to simulate in the laboratory the effects of air leakage during road tests by exhausting air from the interior of the trailer during the cooling load test. The suction port of a small centrifugal blower was connected to the cargo space of the trailer for this purpose, and the discharge duct was fitted with a calibrated orifice meter for measuring the air flow rate. Manometers were used to indicate the static pressure in the cargo space and the pressure drop across the calibrated orifice. With this apparatus it was found that the trailer leaked more air with a negative pressure of 0.06 in. W.G. in the cargo space than was observed during any of the road tests. Furthermore, in ambient conditions of 100° F. dry-bulb temperature and 50 percent relative humidity, the cargo space temperature could no longer be maintained at 0° F. with this amount of air leakage.

Figure 26 shows the air leakage rates caused in two of the commercial trailers when negative pressures in the range from 0 to 0.10 in. W.G. were created in the cargo space by the exhaust blower. The computed air leakage of trailers 3 and 4 during the road tests at 50 miles per hour correspond to that shown in figure 26 for a pressure difference of about 0.02 in. W.G. in each case. This comparison indicates the low order of magnitude of the forces causing air leakage over most of the exposed area of the trailer exterior.

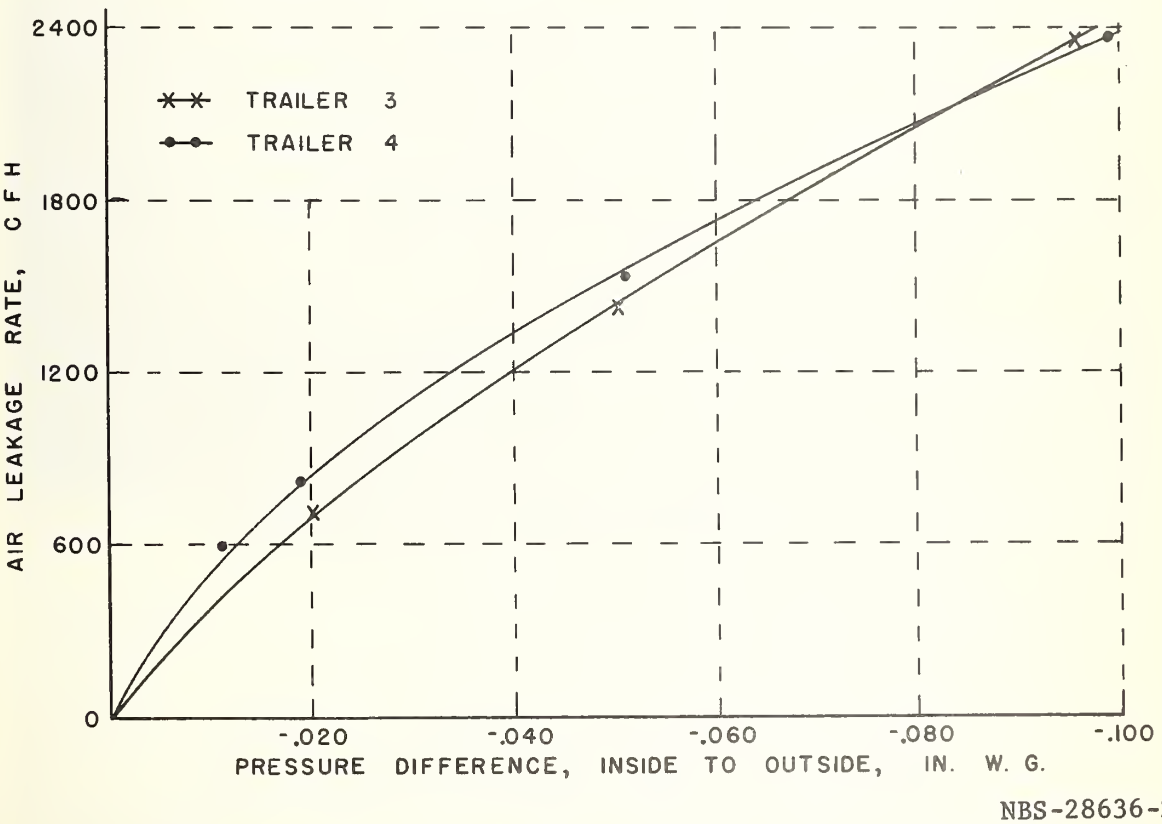

Figure 26.-- Measured air leakage rates of trailers 3 and 4 for a range of negative pressure in the cargo space.

It was concluded that this exhaust method was not the best method for simulating road air leakage in the laboratory for the following reasons: (1) It would be difficult to control a pressure difference as small as 0.02 in. W.G.; (2) this method utilized all of the holes and cracks in the exterior skin for inward air flow whereas some of the holes would be used for outward air flow in actual use of the trailer; (3) the holes and leaks on all exposures would be equally effective in proportion to their area for the exhaust method just described whereas openings in the front end of the trailer would have a much greater adverse effect during road operation; and (4) the chimney effect caused by an 8-foot column of air 100 degrees colder than the ambient air is 0.03 in. W.G., which is the same order of magnitude as the average static pressure difference required to produce the observed road air leakage.

Air leakage of trailer 4 was measured by the exhaust method with the weep holes at the bottom of the trailer walls open and a second time with them sealed. There were 66 such holes, each of about ¼-inch diameter, aggregating an area of about 3 square inches. Figure 27 shows the air leakage for both conditions for the range of pressure difference from 0 to 0.10 in. W.G. The relation of the two curves in figure 27 indicates that the weep holes constituted less than half of the leakage area in this trailer. It was observed that water never ran from these holes when frost was melting in the insulation, but always dripped from other openings at the bottom of the walls.

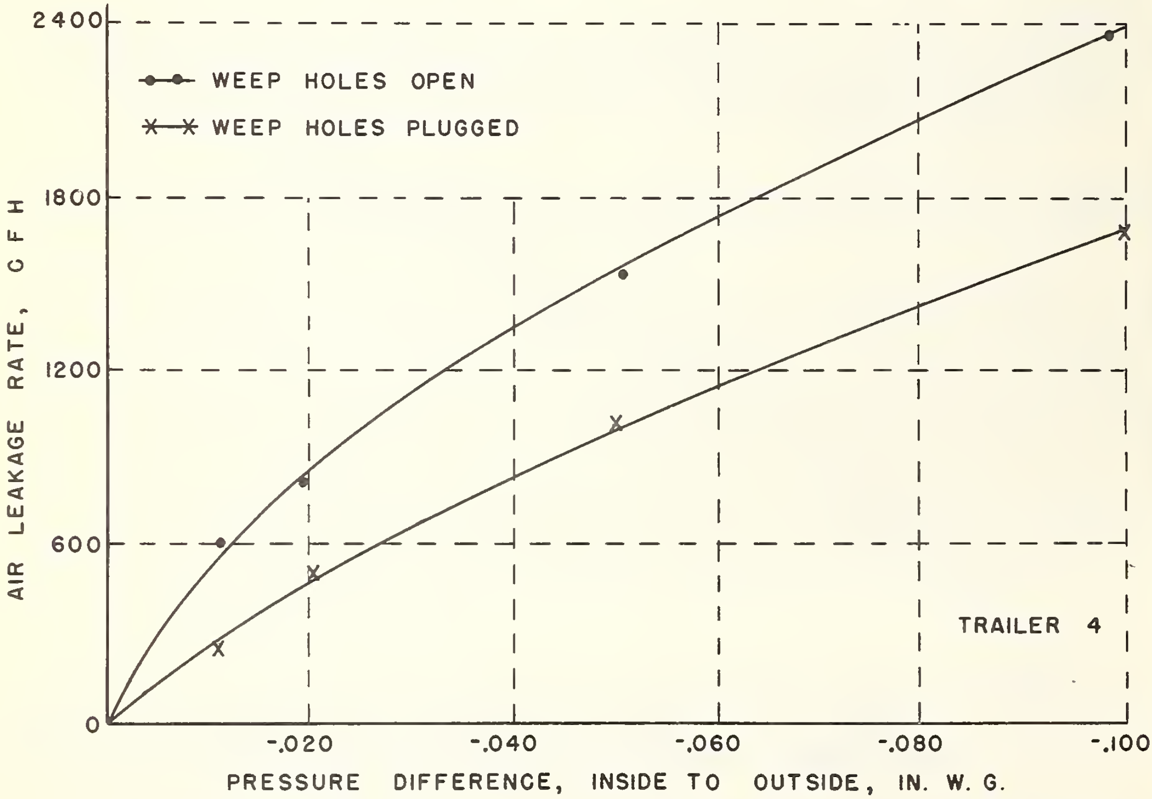

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Figure 27.-- Effect of weep holes at the base of the walls on measured air leakage rates in trailer 4.

Air Leakage Simulation with Front End Plenum

Tests were made on two trailers with a static pressure applied to the front end of the trailer to simulate the air pressure caused by motion over the road. The static pressure was first applied by fastening a plastic bag over the front end and applying the air pressure to the space between the plastic membrane and the skin of the trailer with a centrifugal blower. This method of pressurizing the front end of the trailer worked satisfactorily, but the plastic membrane had to be backed up with canvas to prevent stretching and rupture after a period of use at a temperature of 100° F. A more permanent plenum was built of plywood and wood framing with adjustable sponge rubber strips to restrict the air leakage at the exterior surface of the trailer body. This plenum is shown mounted on one of the commercial trailers in figure 28.

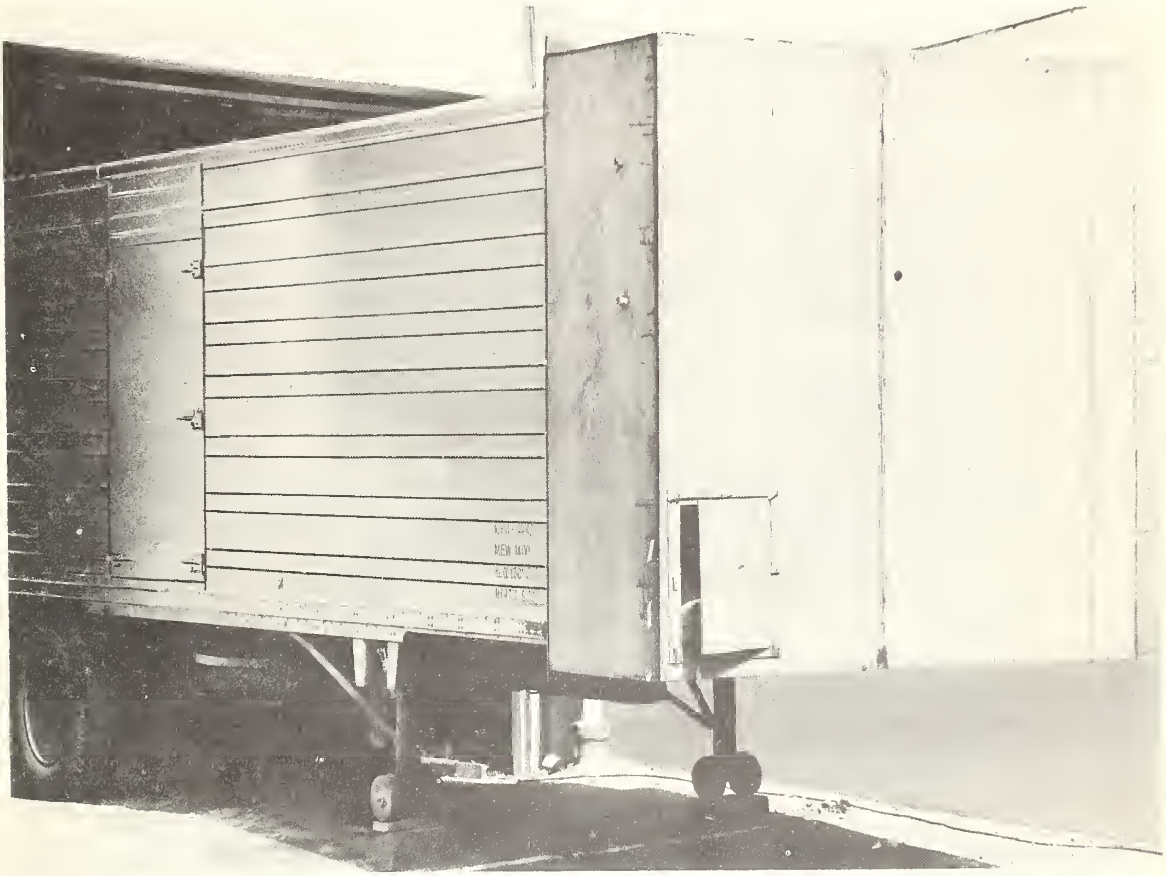

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Figure 28.-- Front end air plenum showing the exit provided for air circulated through the plenum.

The cooling loads of trailers 3 and 4 were measured with the pressure on the front end of the vehicles maintained at various levels in the range from 0.015 to 2.0 in. W.G. The ambient conditions were controlled at 100° F. dry-bulb temperature and 50 percent relative humidity, and the cargo space temperature was maintained at 0° F. Ambient air at the same temperature and humidity was blown into the plenum chamber, and enough leakage was permitted at the joint between the plenum and the trailer body to keep the conditions essentially the same inside and outside the plenum.

Figure 29 shows the effect of the static pressure on the front of the trailer on the cooling load of trailers 3 and 4 as measured in the laboratory. The cooling loads computed for the same temperature conditions during road operation at 50 miles per hour are shown as horizontal lines on the same graph. Figure 29 indicates that a frontal pressure somewhat less than 0.5 in. W.G. on trailer 4 and a frontal pressure of about 2.0 in. W.G. on trailer 3 were required to duplicate the cooling load determined for road operation.

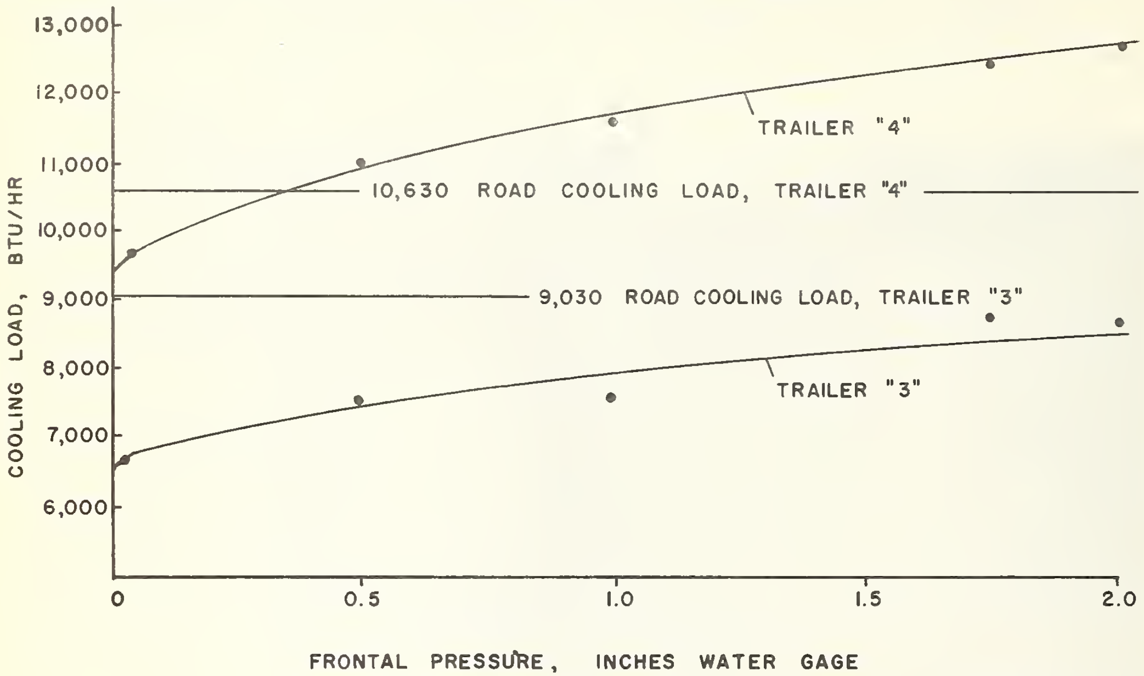

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Figure 29.-- Comparison of cooling load measured in the laboratory with various frontal pressures and extrapolated road cooling load at 50 m.p.h. Ambient temperature 100° F., ambient relative humidity 50 percent, cargo space temperature 0° F.

Static pressures required on the front end of these two trailers, 0.5 and 2.0 in. W.G., to duplicate road cooling loads are below and above the theoretical ram pressure, respectively, that would be expected at a road speed of 50 miles per hour on a still day. Although a definite explanation for this disparity was not found, certain operations performed on the trailers may account for the observed results. A hole around the electrical conduit in the front end of trailer 3 was sealed after road testing and before making the laboratory tests with the pressurized plenum. This source of leakage was discovered when a large ice deposit formed in the front panel and bulged the inner liner. Thus, there was probably less front end air leakage during the laboratory tests of trailer 3 than during the road tests for equivalent frontal pressures. Trailer 4 had a hole in the floor communicating with the space above the fifth wheel plate. Initially, this opening was covered by the front end plenum during the laboratory simulation of road conditions. However, the refrigerating system was unable to produce a cargo space temperature of 0° F. under these conditions. The plenum was then shifted to uncover this opening, and the tests reported in figure 29 were made. Since the static pressure created at this location during road operation at 50 miles per hour is unknown, a conclusion that it was responsible for the disparity of results between road operation and simulated laboratory operation would be speculative.

If a trailer did not leak air into the insulated space or cargo space, there would be but little increase in heat gain at 50 miles per hour on the road compared with stationary performance under similar ambient conditions. Since commercial vehicles appear to have considerable leakage, figure 30 was prepared to show the additional cooling load above the transmission cooling load that would be caused by air leakage at various ambient conditions, when the trailer is at 0° F. For this graph, it was assumed that all air entering the insulated spaces of the trailer would leave saturated at a temperature of 0° F.

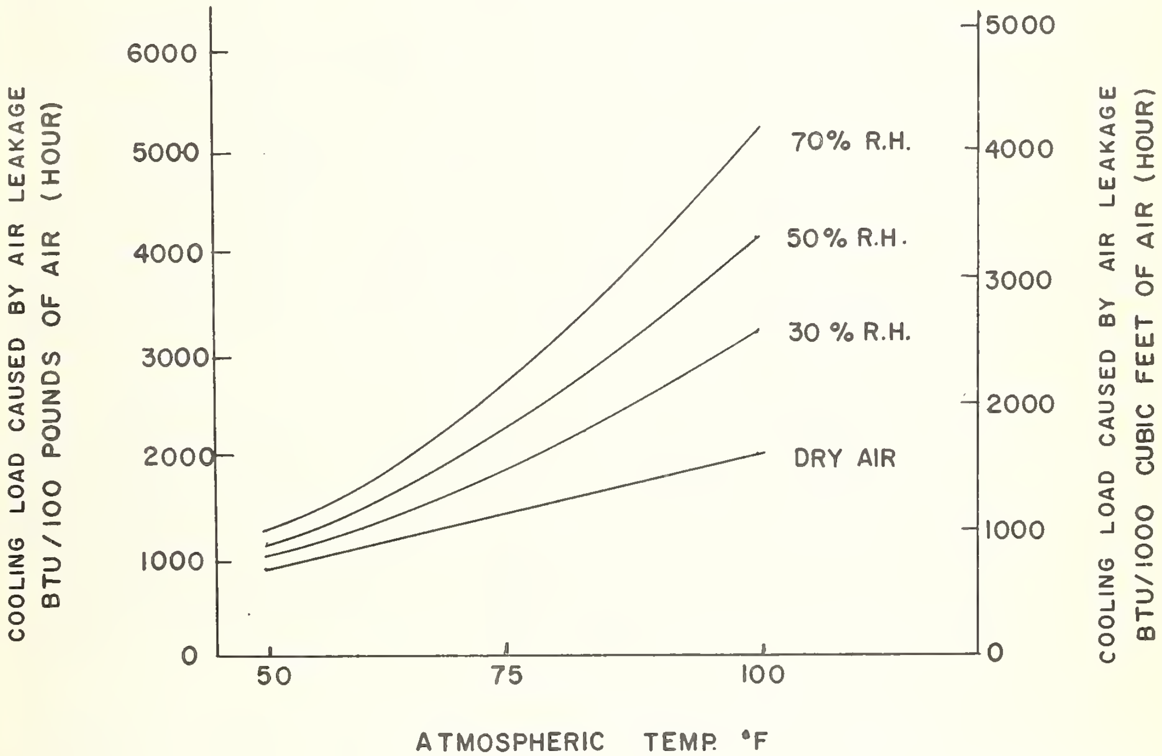

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Figure 30.-- Additional cooling load, above transmission cooling load, caused by air leakage at various rates and ambient conditions for a cargo space temperature of 0° F.

DISCUSSION AND CONCLUSIONS

The tests made of four commercial trailers in the laboratory and on the road and the air leakage and pressure distribution studies made on three military trailers indicated the following conclusions.

1. The cooling loads of the refrigerated trailers tested ranged from 10 to 27 percent greater during road operation at 50 m.p.h. than during

stationary laboratory tests at ambient conditions of 100° F. dry-bulb temperature and 50 percent relative humidity, and with a cargo space temperature of 0° F. The calculated cooling loads for these conditions ranged from 8,900 B.t.u./hr. to 12,400 B.t.u./hr. for a road speed of 50 m.p.h.

2. The increase in cooling load on the road was due principally to air leakage into the trailer structure under the impact pressure of the air although small increases were caused by solar radiation and the movement of engine heat under the floor of the trailer.

3. The minimum computed air leakage rates for the four commercial trailers at a road speed of 50 m.p.h. were 1,475, 860, 790, and 660 cu.ft./hr. and may have been somewhat greater than these amounts. The corresponding computed heat transfer rates due to air leakage at ambient conditions of 100° F. and 50 percent relative humidity would be 5,000, 2,900, 2,690, and 2,260 B.t.u./hr., respectively, assuming that the leakage air left the trailer saturated at 0° F. This latter assumption should be further studied as a part of a large analysis of air and moisture movement in insulated structures. On this basis, the cooling load due to air leakage ranged from 27 percent to 70 percent of the heat transmission component of the total cooling load. These percentages indicate the significant reductions in heat transfer possible by eliminating air leakage in trailer bodies. The air leakage and ice accumulation in the trailers were significant even under stationary conditions. The air leakage amounted to 710, 230, 390, and 380 cu.ft./hr., and the ice accumulation rates averaged 0.98, 0.32, 0.54, and 0.52 lb./hr., respectively, during the laboratory tests.

4. Effects of solar radiation on the trailers were largely nullified by the rapid air motion over the vehicle at a road speed of 50 m.p.h. In a typical test in bright sunshine, incident solar radiation raised the surface temperature of the roof and one side of the trailer about 7.5° F. above ambient air temperature at this road speed. On a weighted average basis this rise in temperature corresponded to approximately three degrees rise in temperature for the entire exterior surface. Under stationary conditions the roof temperatures were increased 25 degrees or more by solar radiation, during this same test.

5. The under side of the trailer was heated as much as 15 degrees above ambient temperature in some tests during road operation, principally by waste heat from the tractor engine. On a weighted average basis the increase may not represent more than three degrees rise in temperature for the entire trailer surface. This rise would not cause a very large increase in overall heat gain of the trailer, but could significantly affect the preservation of food on the interior floor of the trailer, if chilled air were not circulated around and under the load.

6. Without wind effects, ram air pressures up to 1.25 in. W.G. will occur on the front end of a trailer at a road speed of 50 m.p.h. on the portions unshielded by the tractor, although the average ram pressure will probably be below this value. The static pressures in the cargo space, in the insulation space and over most of the exterior skin surface

(excluding the front end and part of the roof) were about equal and ranged from 0.2 to 0.4 in. W.G. below atmospheric pressure for road speeds of 50 m.p.h. The leakage air probably entered the trailer body primarily at the front end of the trailer and left the body over the remainder of the surface, during still weather, leaving most of the moisture in the ambient air deposited as ice in the insulation space. Wind could alter the air movement pattern through a moving trailer and change the pressure pattern on the exterior considerably.

7. Exterior skins of the commercial trailers tested leaked considerably. Under laboratory test conditions the pressure difference tending to cause leakage was only about 0.030 in. W.G., but the air leakage rates ranged from 230 to 710 cu.ft./hr. as indicated by the rate of weight gain of the trailers. There was probably several times as much air exchange between the insulation space and the outdoors as between the cargo space and outdoors.

8. Smaller refrigerating units could be used if air leakage could be eliminated, or alternately, less insulation might be required if air leakage were significantly reduced. It is also probable that the deterioration of trailer bodies would proceed more slowly if moisture could be kept out of the insulation spaces.

9. A laboratory rating method for refrigerated trailers should incorporate some means for simulating the air leakage that will occur on the road with present types of body construction. Pressurization of the front end of a trailer by an attached plenum is considered to be the most practical and representative simulation of the effects of the ram air pressure for use in a stationary laboratory test. This method applies a static pressure on the parts of the body that would normally be pressurized on the road and applies approximately the proper weighting to the leaks on the various sides of the body. It does not simulate the effect of cross winds or wind from the rear of the trailer. A frontal pressure of 2 in. W.G. is considered to be an appropriate value for a simulated laboratory test, since it includes an allowance for at least a minimum wind force on the trailer body.

10. An exhaust blower can be used to measure the air leakage into the cargo space of a trailer under a small negative pressure. It is an economical and effective way to evaluate overall airtightness of a trailer on a production line, to compare different trailers in a laboratory, or to evaluate the change in airtightness of a trailer in regular service.

11. The smoke test is a simple method for finding the parts of a trailer body that have not been made sufficiently airtight during manufacture or to locate the leaks at any time during the useful life of a trailer.

12. A practical laboratory testing and rating method for refrigerated trailers can be established on the heat sink method for evaluation of the cooling load, modified to simulate the air leakage occurring during normal operation of the vehicle on the road.

13. If trailer construction is improved so that air leakage is a minor factor, some modification of the testing and rating method may be desirable to simulate the effects of solar radiation and of engine heat discharge beneath or around the trailer.

14. Based upon the results of this study a recommended standard testing and rating method for refrigerated trailer bodies is set forth in the following pages of this report.

RECOMMENDED STANDARD RATING METHOD TO DETERMINE COOLING LOAD FOR REFRIGERATED TRAILER BODIES

1.0 Purpose

The purpose of this standard is to describe methods of testing and rating refrigerated trailers with respect to cooling load under selected standard ambient conditions, and to measure air and moisture leakage rates under specified conditions.

1.1 Scope:

This standard applies to refrigerated trailers of any length used for transporting frozen food or other materials at cargo space temperatures well below 32° F. It describes a laboratory method for measuring the cooling load of a trailer that will satisfactorily represent the cooling load under typical road speeds as a basis for rating the trailer. The effect of ram air pressure on the front of a moving trailer is simulated during the rating test by a pressurized plenum on the front end of the trailer. The standard does not apply to the rating of trailers under stationary conditions with or without solar radiation on the vehicle or to trailers primarily used for short trips.

The test method described can also be used to measure the cooling load of a trailer at any time during its useful life to evaluate changes in performance.

2.0 Basis for Rating

2.1 Ratings:

Results to be determined from the rating test shall consist of the cooling load in B.t.u.per hour, air leakage in cubic feet per minute, and weight gain 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.

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 trailers shall be measured under the following standard rating conditions:

2.2.1 Cooling Load and Weight Gain Tests:

Test room ambient air temperature

Dry-bulb	100° F.
Wet-bulb (standard barometer)*	83.5° F.

Trailer interior air temperature

Dry-bulb	0° F.
Air pressure on trailer front,	2.0 in. W.G.

2.2.2 Air Leakage Tests:

Trailer interior and test room ambient air temperature--

Dry-bulb	70°-90° F.
Trailer interior air pressure	+0.1 in. W.G.

2.2.3 Deviations:

Deviations allowed in test conditions from Standard Rating

Conditions:

Reading	Deviation of arithmetical average of all readings from standard conditions	Maximum deviation of individual readings
Cooling Load and Weight Gain Test		
Test room air temperature		
Dry-bulb	±1.0° F.	±2.0° F.
Wet-bulb	±1.0° F.	±2.0° F.
Trailer Interior Air Temperature		
Dry-Bulb	±0.5° F.	±1.0° F.
Air pressure on trailer front	±0.05 in.W.G.	±0.10 in.W.G.

* 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 inch Hg. decrease in barometric pressure.

Air Leakage Test

Trailer interior air pressure	±0.005 in. W.G.	±0.010 in. W.G.

2.3 Application Ratings

Although this standard is intended for use in determining Standard Ratings, ratings at other conditions, known as Application Ratings, may be determined by using the techniques and procedures of this standard, provided the test conditions are clearly stated in expressing the application ratings.

3.0 Instruments

3.1 Temperatures shall be measured by either 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 trailer and across external brine heater	±0.3° 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 should be kept clean.

Relative humidity measurements, if used, shall be made with sufficient accuracy to obtain compliance with the accuracy requirements to 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 into the circuit. Instruments or systems used to measure the temperature differences of the brine across the cooling coil in the trailer and across the brine heater shall be compared with each other before they are installed and in the range used they shall agree within 0.2° F. when immersed in the same bath.

3.2 Brine flow shall be measured with an integrating liquid flowmeter having an accuracy of ± 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 resistance loads, an ammeter and volt-meter may be used.

Accuracy of instruments used to measure the electrical input to heaters in the trailer or in the external brine 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 trailer under test shall be capable of detecting and measuring a change of 0.5 pounds and shall have an accuracy of ± 0.5 percent of the quantity measured.

3.5 Instruments used for measuring the air pressure on the front of the trailer during the heat leakage test and the air pressure in the trailer during the air leakage test shall be accurate within ± 2.0 percent of the pressure or pressure difference measured.

3.6 Air flow measurements shall be made with indicating or integrating instruments having an accuracy within ± 2.0 percent of the flow measured.

4.0 Test Room

An insulated test room approximately 16 feet wide and 55 feet long and 18 feet high is required for testing trailers. A door at least 9 feet wide and 14 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 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. A suggested arrangement of the facilities is shown in figure 31, but others equally good may be found.

Distributed heating and humidity sources are desirable in the test room to promote 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 trailer at a velocity in excess of 400 ft./min. High air velocities around the trailer 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.

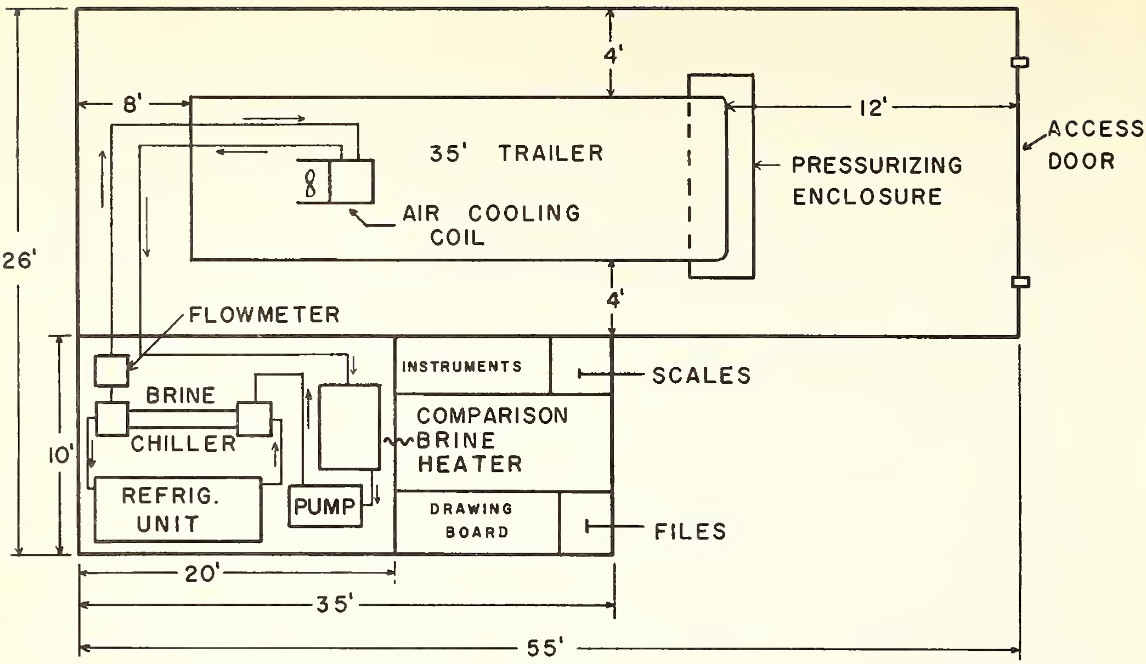

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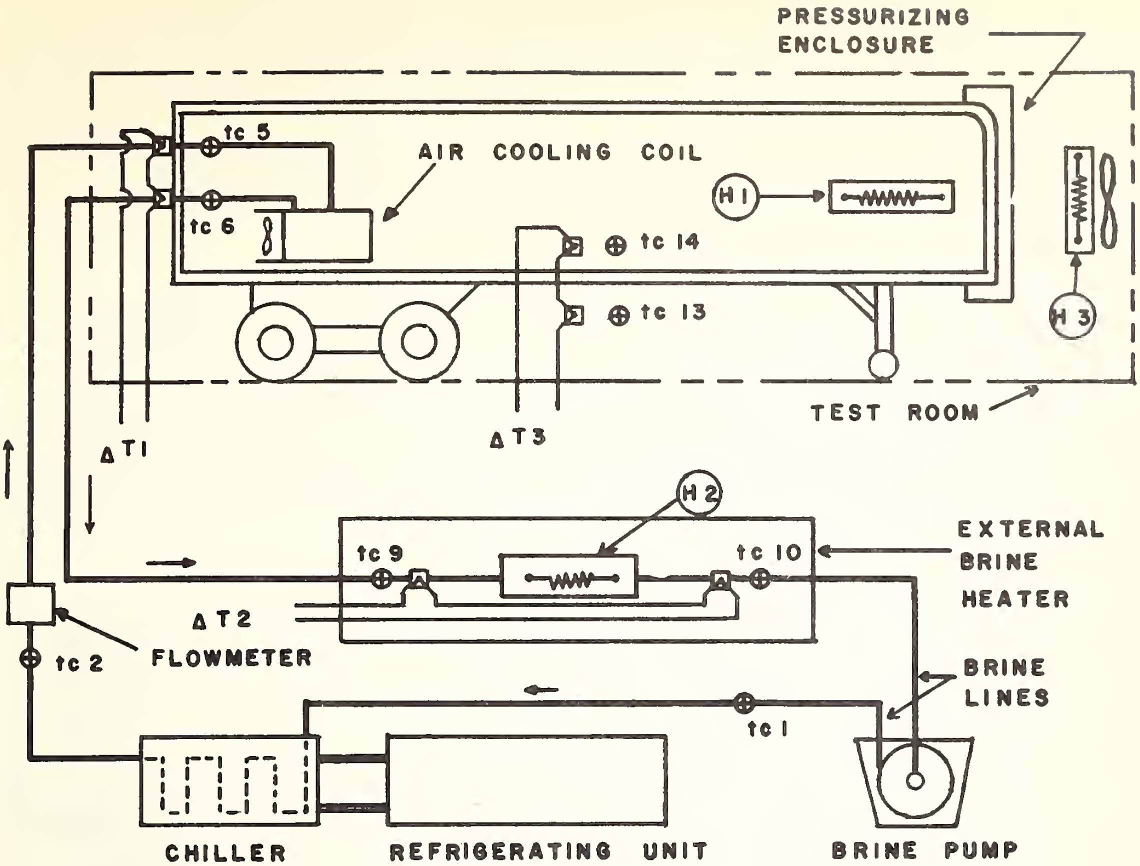
Figure 31.-- Suggested arrangement of laboratory facilities for testing refrigerated trailers.

4.1 Cooling Load Test Apparatus

The refrigerating equipment and temperature measurements required to determine the cooling load are shown diagrammatically in figure 32. As indicated in the figure the equipment consists of a refrigerating unit and brine chiller, a brine pump, an air cooling coil inside the trailer, and an external brine heater and flowmeter in the brine circuit outside the trailer.

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 desirable 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 trailer cooling coil or comparison brine heater should not exceed 0.2° F.

The cooling coil inside the trailer 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 largest trailer to be tested plus fan loads and a limited amount of controlling heat with a mean temperature difference between coil surface and trailer air temperature of 20° F. The blower in the cooling unit shall deliver sufficient air to produce a temperature difference of 10° F. or less between the air entering and leaving the coil.



SYMBOLS:

SYMBOLS:	ΔT1	TEMPERATURE DIFFERENCE OF BRINE ACROSS TRAILER
	ΔT2	" " " " " BRINE HEATER
	ΔT3	" " BETWEEN AIR IN TRAILER AND TEST ROOM
	tc 1, tc 2, tc 3, etc.	TEMPERATURE MEASUREMENT POINTS
	H1, H2, etc.	ELECTRIC HEATERS RATED IN BTU / HOUR

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Figure 32.-- Schematic diagram of heat sink apparatus required for testing and rating refrigerated trailers.

The brine heater outside the trailer shall be insulated sufficiently to reduce the heat transmission from the surroundings to the brine to ½ percent or less of the heating capacity of the brine heater. The heating capacity of the brine heater shall be approximately equal to the maximum total heat absorption in the trailer and the heater capacity should be adjustable. Voltage regulation shall be provided for the power supply to the brine heater and to the heaters inside the trailer 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 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 trailer and the brine heater shall be about 10° 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, 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 trailer temperature. All electric power to fans, motors, heaters, etc., in the trailer shall be measured.

A pressurizing enclosure covering the entire front of the trailer shall be attached for the cooling load test. A blower shall be used to maintain a positive air pressure in the enclosure in accordance with the rating conditions. The under side of the trailer within the enclosure shall be sealed so the applied pressure will not cause air to leak into the insulated floor from underneath, but no other surface shall be sealed. The enclosure shall be attached to the trailer not more than 6 inches to the rear of the termination of the front end curvature. An exhaust opening with a damper shall be provided in the enclosure at a point opposite the air inlet to permit sufficient movement of air through the enclosure to maintain proper temperature and humidity conditions. The enclosure may be of either rigid or flexible construction. Except where attached to the trailer, a minimum clearance of 1 foot shall be maintained between the front end of the trailer and the enclosure. A pressure-tight seal at the points of attachment to the trailer is not necessary.

Brine lines, power cables, instrument leads, etc., may be brought into the trailer at any convenient point. Where no opening is available, it is recommended that a suitable sleeve be installed in one of the rear 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 trailer to be tested shall be placed on the scale mechanism and the test equipment and measuring devices installed. After the trailer 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 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.

The temperature difference of the brine across the cooling coil in the trailer and across the external brine heater shall each be held at a constant value between 8° F. and 12° 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 other than the front of the trailer, and one geometrically centered in the pressurizing enclosure on the front of the trailer. The temperature difference between any two of these stations at a given time shall not exceed 3° F.

The ambient wet-bulb temperature shall be the average of not less than two points, one at the rear of the trailer and the other in the pressurizing enclosure at the front of the trailer. The difference in wet-bulb temperature between any two points of measurement at a given time shall not exceed 2° F. Air for the pressurizing enclosure shall be taken from the test room, and sufficient air shall be moved through the enclosure to maintain the desired temperature and humidity conditions within the enclosure.

The air temperature inside the trailer shall be the average of the observations at 12 stations located as follows: 4 at the front, one in each corner suspended 6 inches from each adjacent surface; 4 similarly located at the rear; and 4 at the middle of the trailer, one at each corner 6 inches from each adjacent surface. If desired, each group of 4 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 may differ by more than 3° F.; if the groups of four are used, no two readings may differ by more than 2° F. at a given time.

During the portion of the test used to determine the cooling load rating, all observations shall be made at 30-minute intervals.

Trailers equipped with removable plug 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 trailer. Care must be taken that air discharged from any fan does not blow directly on cracks or seams.

The brine lines within the trailer 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 external brine heater	watts
5.	" " " heater in trailer	watts
6.	" " " fan motors, etc. in trailer	watts
7.	Applied voltage to external brine heater	volts
8.	" " " heater in trailer	volts
9.	" " " fan motors, etc. in trailer	volts
10.	Electric current to external brine heater	amps
11.	" " " heater in trailer	amps
12.	" " " fan motors in trailer	amps
13.	Dry-bulb temperatures of air inside trailer	°F.
14.	" " " " " 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.	" " " " outlet " " "	°F.
18.	Temperature difference of brine in and out of trailer	°F.
19.	" of brine at inlet of external brine heater	°F.
20.	" " " " outlet " " " "	°F.
21.	" difference of brine in and out of external brine heater	°F.
22.	Temperature of brine entering flowmeter	°F.
23.	Brine flow rate	lb./hr.
24.	Weight of trailer, or change in weight	lb.
25.	Air pressure in pressurizing enclosure	in.W.G.

7.0 Calculations of 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 trailer and the temperature rise in the external brine heater; the other method uses the temperature rise of the brine in the trailer and the mass flow of the brine as measured by the flowmeter to determine the cooling load. 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.

Because both methods rely on the temperature rise of the brine in the trailer, two separate measuring elements shall be used to measure this brine temperature difference and must agree within 0.2° F.

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

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

where: $\Delta T1$ = Temperature rise of brine in the trailer, °F.

$\Delta T2$ = Temperature rise of brine between inlet and outlet of brine heater, °F.

$H2$ = Heat input to external brine heater, B.t.u. per hour

$H1$ = Heat input to trailer heater, fan motors, etc., B.t.u. per hour

$\Delta T3$ = Temperature difference between air in trailer and air in test room, °F.

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

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

where: $\Delta T1$ = Temperature rise of brine in the trailer, °F.

M = Brine flow rate, lb. per hour

C = Specific heat of brine at mean temperature in the cooling coil, B.t.u. per lb. (°F.)

$H1$ = Heat input to heater, fan motors, etc., inside the trailer, B.t.u. per hour

The rated cooling load shall be the average of the values determined by the two methods, expressed to the nearest even 500 B.t.u. per hour.

The rated weight gain shall be 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. per hour.

8.0 Air Leakage Rating Test Procedure

This test is intended to provide a rating on the overall tightness of the vehicle against air leakage.

The test may be made either before or after the cooling load rating test. The trailer must be at about the same temperature as the ambient air and in the range from 70° to 90° F.

For the test, air shall be forced into the trailer,and the amount of air required to maintain a positive pressure of 0.1 in. W.G. in the trailer shall be measured. The rated air leakage shall be expressed to the nearest 0.5 cubic feet per minute. All floor drains shall be plugged for this test.

Air for this test can be provided by a blower or from a house air supply and can be metered with an orifice, volumetric displacement gas meter, or other flowmeter having the necessary accuracy. The capacity of the air supply should be at least 50 cubic feet per minute, and the measuring device must be capable of metering any flow at or below 50 c.f.m.

9.0 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), B.t.u./hr. based on tests conducted in accordance with (Sponsor's designation) Standard Method of Testing and Rating the Cooling Load of Refrigerated Trailers." 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.

It is probable that most published ratings would include at least the Standard Cooling Load Rating. Standard Weight Gain Rating and Standard Air Leakage Rating should be determined for each trailer tested but need not be published in order to publish the Standard Cooling Load Rating.

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