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UNIVERSITY OF LOUISVILLE

- I. IMPROVED ICE MANTAGE IN ICE BOXES
- II. TEMPERATURE CONTROL IN TENTS - WINTER PRASE

A Thesis

Submitted to the Faculty
of the Graduate School
of the University of Louisville
in Partial Fulfillment
of the Requirements
for the Degree of

MASTER OF MECHANICAL ENGINEERING

Department of Mechanical Engineering

Robert Beaman Euggan

June, 1949

- I. IMPROVED ICE MELTAGE IN ICE BOXES
- II. TEMPERATURE CONTROL IN BENTS - WINTER PEASE

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ABSTRACT

This thesis consists of two parts: Improved Ice Meltage in Ice Boxes; Temperature Control in Tents - Winter Phase

Improved ice meltage in the Progress refrigerator is discussed in the first part of the thesis. This portion of the work shows that satisfactory refrigerator temperatures and flat ice meltage could not be obtained simultaneously when using V-shaped ice racks, although they could be obtained separately; however, the basket-type chamber offers greater possibilities of obtaining proper temperatures and meltage characteristics. Series of tests are discussed and conclusions drawn from the results.

In the second part of the thesis a method of treating data obtained in the transfer of heat into and from military tents was developed and applied to the experimental data. A study was made of large wall tents, using as insulators the following types of liners: 3.6 oz. cotton, 2 layers of cloth; 4 oz. cotton, 2 layers of cloth, inner fabric - dark green, vinyl-coated; 4 oz. cotton, white, 3 layers of cloth. Measurements were made of tent air and fabric temperatures under constant heat input conditions, and overall heat transfer coefficients were calculated for a comparison of the insulating values of the liners.

PART L. IMPROVED ICE MELTAGE IN ICE BOXES

INTRODUCTION

The ideal ice refrigerator maintains temperatures at a level required for proper food preservation, while offering economy and convenience of operation. Each of these factors is of sufficient importance to warrant serious consideration in the design of ice refrigeration equipment.

Temperatures in the refrigerator not only must be at a sufficiently low level, in the neighborhood of 40° F., but must show little variation within the food compartment. A difference of more than ten degrees between the maximum and minimum temperatures in the ice chamber is not desirable. Furthermore, temperatures in the food compartment must remain substantially constant after the ice block has melted to approximately 20% of its original size. Undue variations of temperature resulting from reductions in the size of the ice block can cause food spoilage.

Ease of cleaning, absence of dripping, and means for convenient arrangement of food within the chamber are the features required by the owner. These considerations dictate a simple arrangement, particularly of the ice chamber, to permit easy assembly and disassembly for cleaning. The design must be such as to channel all dripping from melted ice or from condensed moisture into a drain pipe. Any dripping into the food compartment is a serious failure of the refrigerator. In designing for convenient arrangement of food, it is most important that provision be made for storage of milk bottles in the coldest part of the food compartment.

Milk bottles are bulky objects and their arrangement on standard shelving in a cold spot often presents difficult problems.

Equally important are the features required for the convenience of the "ice-man" who services the refrigerator. A refrigerator is needed into which a standard size block of ice can be placed at periodic intervals without requiring any other attention. This objective has not been attained in any commercial refrigerator now on the market. Usually, the ice-man is required to make room for new ice by chipping away the remains of old blocks. Because of this or other undesirable features, he may be forced to set his ice load on the floor of a kitchen while he works on the refrigerator, a procedure that housewives find objectionable. Suitable provision for easy reicing was the major problem encountered in the work covered by this problem of the thesis.

Also, economic considerations require the refrigerator to be low in first cost and ice consumption. These considerations must obviously be met if the refrigerator is to sell competitively.

THEORETICAL

Under conditions of thermal equilibrium the temperatures that are maintained in the refrigerator are such as to balance the flow of heat through the insulated walls of the refrigerator against the heat transfer between air and ice in the refrigerator. Anything that tends to increase the heat flow through the walls correspondingly increases the temperatures in the refrigerator. Similarly, anything that decreases the rate of heat transfer from air to ice increases the temperatures.

During the investigation covered by this thesis, nothing was done to effect the rate of heat flow into the box. This rate is determined almost entirely by the ambient temperature outside the box, the thermal conductivity of the refrigerator walls, and the equilibrium temperature inside the box. None of these could be considered an independent variable for the purpose of this investigation. In all tests of a quantitative nature, the outside ambient was held constant at a specified temperature of 85° F. Also, no change was made in the insulating properties of the walls. Therefore, any change in the rate of heat flow into the box was caused by a corresponding change of conditions inside the refrigerator.

Thus, the rate of heat transfer between ice and air inside the refrigerator is the greatest single factor determining the temperature level at equilibrium and depends on four variable quantities, all of which are important:

1. The difference in temperature between the air and the ice with which it is in contact. Inasmuch as the ice temperature may be assumed to remain fixed at 32° F., only variations in air temperature need be considered. The rate of heat transfer will vary proportionally with this temperature difference.

2. The area of contact between the ice surface and the air. The larger this area, the higher is the rate of heat transfer obtained.

3. The resistance to heat flow between the ice and the air. This resistance is dependent in a large measure on the velocity of the air flowing past the ice. At low air velocities, a layer of air tends to cling to the ice surface, effectively insulating it from the bulk of the adjacent warmer air. As the velocity of the air past the ice is increased, the thickness of the insulating air film is decreased, and heat transfer is improved.

4. The amount of extended heat transfer surface present. Some heat transfer can take place between air and metal fins, the fins transferring heat to the ice. Within reasonable limits, this extended surface may be regarded as equivalent to additional ice area.

Because of the importance of each of the above factors in determining the temperature level within the refrigerator, each must receive consideration in design.

The question of temperature difference, while subject to only very limited control, merits considerable attention. One of the most difficult problems encountered in this work was objectionably slow ice melting at the center of the ice chamber where the temperature difference is lowest.

The questions of ice area and extended surface area cannot be neglected. The conventional design of ice racks provides a number of horizontal fins on which the ice is supported. In operation, this type of rack causes the bottom surface of the ice to melt into a number of V-shaped grooves, exposing much more area to the air stream than would be the case if the ice surface were flat. Also, the fins act as extended surface giving additional heat transfer area.

Of the factors mentioned, resistance to heat transfer is the most critical because it is subject to the greatest relative variations as a result of conditions within the refrigerator. As has been mentioned, resistance is determined almost entirely by the velocity of the circulating air flowing past the ice surface. Because of the practical importance of thermal circulation in determining rates of heat transfer, detailed consideration will be given to this phenomenon and to the factors that control the rate of thermal circulation.

In the refrigerator of conventional design, air tends to rise at the walls because it is warmed by heat conducted into the refrigerator. After passing over the ice block

and being cooled, air descends at the center of the refrigerator. The following two variables control the rate at which this thermal circulation takes place.

1. Thermal head. This is the pressure differential that would exist if a column of relatively warm air adjacent to the walls of the refrigerator could be held stationary in conjunction with a cold column of air at the center. The magnitude of the thermal head depends on the height of the two columns and the difference in temperature between them.

2. Flow resistance. Flow resistance results from obstructions placed in the path of the circulating air. It is caused by sharp bends in the flow path, restricted flow passages, and sudden changes in cross-section of the flow passage. Flow resistance causes a loss of pressure in the flowing liquid, the exact magnitude of the pressure drop depending also on the velocity of flow. An increase in either resistance or velocity causes an increase in pressure drop. When temperature differentials set up a thermal head and air starts to circulate, the rate of circulation increases until the pressure losses caused by flow resistance balance the pressure differential due to the thermal head.

High rates of heat transfer are associated with high rates of thermal circulation. Any change that decreases the flow resistance in the circulation path, or increases the height of the circulation path, will improve circulation and heat transfer and result in lower refrigerator temperatures.

EXPERIMENTAL

A sketch of the original Progress refrigerator is shown in Figure 1. It has a six cubic foot food compartment with an ice chamber designed for a 100-pound block of ice. This refrigerator has an insulated double wall construction, fabricated from sheet steel mounted on a wooden frame, the space between the walls being filled with blanket glass-wool. The inside and outside surfaces are coated with a white baked enamel finish.

The ice chamber, shown in Figure 2, is large enough to accommodate a standard 100-pound block of ice with enough additional space for approximately thirty pounds of ice that may remain from previous icings. Constructed of sheet steel this chamber consists of air passages designed to channel air-flow past the ice block, racks to support the ice in its proper position, and trays to prevent condensed moisture and ice meltage from dripping into the food compartment.

Two different types of tests were carried out to investigate the performance of various ice-chamber designs. One was both quantitative and qualitative in nature, giving quantitative information on the temperatures attained in the refrigerator and on the rate of ice meltage, while also giving qualitative information on the air circulation paths through the refrigerator and on the characteristics of ice meltage as revealed by the shape of the ice block. The second test was entirely qualitative, being an investigation of the circulation paths through the refrigerator as revealed by smoke

injected into the interior. Hereafter, these tests will be referred to as "temperature tests" and "smoke tests" respectively.

TEMPERATURE TESTS

An external ambient close to 85° F. was maintained during all temperature tests. To maintain this ambient, the refrigerator was kept in a heated room, the temperature being maintained by electric resistance heaters controlled by an on-off type thermostatic switch. All temperatures were measured by copper-constantan thermocouples used in conjunction with a potentiometer calibrated directly in degrees Fahrenheit and having an automatic cold junction compensation. The only other equipment needed was a means of collecting and weighing melted ice from the refrigerator. For this purpose a container was placed under the drain, and platform scales were used.

In all temperature tests a run was started by placing a 100-pound block of ice in the chamber. Standard blocks as perfectly shaped as possible were used, i.e., blocks with an extreme taper or without square corners were avoided. After the box had reached equilibrium, the temperatures were recorded at regular intervals. Water from melted ice was removed periodically, its weight and collection time recorded, and rate of meltage computed.

FIGURE 1 DIAGRAMMATIC SKETCH OF ICE BOX

FIGURE 2 SKETCH OF ORIGINAL ICE CHAMBER

SMOKE TESTS

The principal problem in the smoke tests was making the air currents in the refrigerator sufficiently visible for observation. To attain this objective, a dense, persistent smoke was required which was non-corrosive and could be produced at a low temperature. This latter requirement ruled out all smokes produced by combustion processes. After a period of trial and error, ammonium chloride smoke was adapted for the experiments because, in addition to meeting all the above-mentioned requirements, it was easy to generate. The smoke was generated outside the refrigerator and injected into the air stream through a hole in the wall of the refrigerator. To make the smoke readily visible, a water-soluble black paint was used to coat the interior of the refrigerator.

For convenience in observing the air stream, the insulated door of the refrigerator and the hopper front were replaced by sheets of plate glass, sealed at the edges with masking tape. Because insulating properties of this glass panel were quite different from those of the original door, the ambient temperature was maintained at a higher value outside of the insulated refrigerator walls than outside of the glass. To insure approximately uniform heat conduction through all panels, the external ambients were adjusted to give a surface temperature on the inside of the glass substantially the same as that on the inside of the insulated walls.

A 100-pound standard block of ice was placed in the ice chamber, the glass door and hopper front were placed in position and sealed, and the refrigerator wall allowed to reach equilibrium. The external ambient on the insulated walls was then adjusted, and the refrigerator was given time to reach thenew equilibrium conditions. After the refrigerator attained an equilibrium condition, the smoke was injected into the box. Sketches were made showing the air currents observed. See Figures 4 and 5.

For the sake of convenience the results of individual tests are grouped in several series; the data obtained in each series and their interpretation are presented separately.

First Series

The purpose of this series was to check the performance of the production model of the Progress refrigerator. A temperature test was carried out and the data given in Table I were obtained. Temperatures were measured at the points indicated in Figure 3.

As can be seen from Table I, the temperatures attained in the refrigerator were good, indicating adequate thermal circulation and heat transfer. No water was observed to drop into the food compartment. However, the characteristics of the ice meltage were completely unsatisfactory. The

FIGURE 3 THERMOCOUPLE LOCATIONS
FOR DATA OF TABLE I

TABLE I

TEMPERATURE TRENDS IN FIRST SERIES OF TESTS

Thermocouple Number as Shown in Figure 3	12/23/47 12:45 P.M.	12/23/47 1:28 P.M.	12/23/47 4:43 P.M.	12/24/47 3:52 P.M.
1	41° F.	42° F.	43° F.	42° F.
2	40	41	42	41
3	49	50	52	52
4	49	50	52	52
5	52	55	57	55
6	52	54	56	55
7	58	90	99	89

ice tended to melt into a rounded mound, which was objectionable because it did not present a surface at the time of re-icing on which a new block of ice could be stably supported. Furthermore, the block showed severe undercutting at the back and sides, sufficient in itself to cause static instability. After reicing, the original block remained unmelted for a long period of time at the center of the ice chamber, indicating a tendency for ice to accumulate at that point and add to the inconvenience of reicing.

In Figure 2 a dead air space enclosed between the top of the umbrella and the sweat pan located immediately below it is shown. The purpose of this sweat pan was to prevent the dripping of condensed moisture into the food compartment. It was thought that the insulating properties of this dead-air space had much to do with the objectionable meltage characteristics observed. In the second series of tests efforts were made to correct this condition.

Second Series

The sweat pan was removed, and a second temperature test was carried out. The temperature data recorded during the test were unchanged. Meltage characteristics were only slightly improved, if at all. The ice continued to melt into a rounded mound and to accumulate at the center of the ice

chamber above the umbrella. The explanation for this negligible improvement was that as the descending air streams left the underside of the ice and turned downward into the down draft, there was not sufficient contact with the bottom of the umbrella.

A third series of tests was undertaken to understand more fully the air currents in the refrigerator.

Third Series

All the tests in this series were smoke tests and as such yielded only qualitative information. Because the purpose of the tests was to investigate air currents in the standard production model ice chamber, the regular ice chamber was used with standard 100-pound blocks of ice.

The major air currents observed in the refrigerator were as shown in Figure 4 and Figure 5. An up-draft was observed at the sides of the ice chamber, the major part of which ascended to the top of the refrigerator, with a comparatively small amount entering the ice chamber below the ice level. A much stronger up-draft was found at the front of the refrigerator between the door and ice chamber front, the velocities in this stream being several times those at the side. At the top of the refrigerator the three streams from the front and sides merge and descent slowly over the ice block. The flow over the surface of the block was a thin,

FIGURE 4 SKETCH OF OBSERVED AIR
CURRENTS — FRONT VIEW

FIGURE 5 SKETCH OF OBSERVED AIR
CURRENTS ~ SIDE VIEW

clinging film on the ice surface, with the greatest part of the flow occurring at the back of the block.

While only a small percentage of the total circulating air entered the ice chamber at the sides below the ice rack, it was found that this stream caused the undercutting observed at the sides of the ice block. As the stream entered the ice chamber, it tended to rise in the space between the side of the ice block and the side wall of the ice chamber. In rising, it came in contact with the ice, was cooled, and immediately descended, causing an undercut of characteristic shape.

The undercutting at the back of the ice block was found to be caused by the air stream rising at the front of the refrigerator. As indicated in Figure 5, the kinetic energy of this stream is sufficient to carry it to the rear of the refrigerator after it reaches the top of the ice chamber and turns toward the center. The highest velocities observed in the refrigerator were at the down-draft in the center of the ice chamber where all the air streams merged.

From these smoke studies it was apparent that a flat ice surface could be obtained if heat transfer were restricted to the bottom surface of the block. To obtain this condition the following modifications in the original design were made:

1. A gasket was placed between the door of the refrigerator and the ice chamber. The purpose of the gasket

was to prevent circulations of air from the food compartment to the ice chamber at the front of the refrigerator.

2. Guide vanes were placed in the air stream at the side of the ice chamber to direct this stream under the ice and further prevent circulation to the top of the refrigerator.

3. Plates were fastened to the ice racks at the sides to prevent undercutting. These plates were folded so that they might serve as guides for the ice block.

4. The umbrella was entirely removed, and the ice racks supported at each end.

5. A drip catch was installed beneath the down-draft opening.

Figure 6 shows the ice chamber incorporating these changes. A fourth series of tests was made using this ice chamber.

Fourth Series

Using this modified ice chamber, a smoke test was made. All circulation in the upward direction was through the ice chamber inlets at the sides below the ice racks. Downward circulation was still confined to the down-draft opening at the center of the refrigerator. The air velocity at this point was found to have been decreased substantially.

FIGURE 6 SKETCH OF REVISED ICE CHAMBER
AFTER SMOKE STUDIES

A temperature test on the modified ice chamber was carried out next. Unsatisfactorily high temperatures were attained. Table II gives the temperature data recorded for the test. The thermocouple positions for this table are shown in Figure 7. The ice meltage characteristics observed during the test were satisfactory, however. The initial block of ice maintained a flat upper surface while melting. No difficulty was encountered in reicing the refrigerator.

The next step was to lower the refrigerator temperatures to the desired level without sacrificing these meltage characteristics. It was decided to increase the effective thermal head in the refrigerator by increasing the slope of the ice racks. This change raised the air inlet at the side of the ice chamber to a position several inches above its original location. As a result, the height of the effective air column in the refrigerator was increased.

Figure 8 is a sketch of the ice chamber built on this principle. The ice chamber was installed in the refrigerator, and a fifth series of tests was carried out to check its performance.

Fifth Series

This series consisted entirely of temperature tests intended to provide performance data for the modified ice chamber with steep ice racks. The data recorded during

FIGURE 7 THERMOCOUPLE LOCATIONS FOR
DATA OF TABLES II AND III

FIGURE 8 SKETCH OF ICE CHAMBER WITH STEEP RACKS

TABLE II

TEMPERATURES OF ICE BOX IN FOURTH SERIES OF TESTS

<u>Thermocouple Number as Shown in Figure 7</u>	<u>Ice Chamber Full</u>	<u>Ice Chamber Low</u>
0	86° F.	85° F.
1	43	49
2	52	54
3	52	52
4	50	52
5	38	40
6	48	50
7	86	86
8	50	50

the first test of this series are given in Table III, indicating that once again the temperatures in the refrigerator had been reduced to a satisfactory level. The ice melted flat and, in general, offered no difficulty in reicing.

As the tests were continued through several reicings, a new problem arose. The ice showed a pronounced tendency to shift its position toward either one side of the chamber or the other. Because of the slope of the racks, this shifting resulted in the upper surface of the block assuming an inclined position and counteracted all the advantages of flat meltage that had been gained.

The explanation for this shifting is that the rectangular ice block is inherently in a statically unstable position when it is lying horizontally on the inclined ice racks. It can be shown that this case satisfies the mathematical criterion of static instability. Theoretically, this unstable case also existed when the less steep racks of the original ice chamber were used. Probably, the instability was not observed in the latter case because the time required for the block to tilt and reach equilibrium was much longer. With a longer equilibrium time available the block was able to melt and conform to the V-shape of the rack before appreciable tilting was noticed.

Several attempts were made to force a condition of equilibrium by restraining the block. These restraints took the form of guide vanes placed on the racks in such a

TABLE III

TEMPERATURES OF ICE BOX IN FIFTH SERIES OF TESTS

<u>Thermocouple Number as Shown in Figure 7</u>	<u>Time</u>	
	<u>9:00 A.M.</u>	<u>10:00 A.M.</u>
0	84° F.	84° F.
1	46	46
2	54	54
3	52	52
4	--	--
5	42	42
6	50	49
7	85	84
8	50	50
9	50	50

position as to stop sideward motions. They proved uniformly ineffective because the ice tended to "flow" around this sort of restraint.

While further attempts to stabilize the ice block might have proved successful, it was generally agreed that use of the ice chamber with steep racks in the household might aggravate the stability problem. Consequently, a decision was reached to stop work on the modified ice chamber at this point and to experiment with the "basket-type" chamber in which no stability problem would be encountered.

Basket-type Ice Chamber

The conventional ice chamber was designed in such a manner that when reicing became necessary, a new block of ice was placed on top of the older block. This arrangement necessitated a flat ice surface at the time of reicing. The purpose of the basket-type ice chamber is to provide a flat metal surface for reicing. The basket as originally developed by the Louisville Tin and Stove Company is shown in Figure 9. It consisted of a number of parallel, vertical metal plates running across the width of the ice chamber, the plates being spaced about $1\frac{1}{2}$ inches apart. In the ideal case when reicing is necessary, older blocks of ice will be completely below the metal surface, permitting the new block to be placed on the metal surface.

FIGURE 9 SKETCH OF ORIGINAL
BASKET-TYPE ICE CHAMBER

The advantage of this type of ice chamber is that flat meltage of the ice is not necessary, and air circulation through the ice chamber is permissible.

Sixth Series

In this series the original basket was tested for performance. The results are given in Table IV. It can be seen that the temperatures recorded were completely satisfactory.

The main difficulty encountered in this test was failure of the ice to melt over the down-draft. This was objectionable because ice tended to accumulate at this point, making reicing difficult. The small air-to-ice temperature differential existing at this region was the cause of the difficulty.

To obtain a higher temperature differential a duct as shown in Figure 10 was installed in the ice chamber. The duct was designed to cause a flow of comparatively warm air through the region where the rate of ice meltage was too low.

Initial smoke tests carried out with the duct in position indicated a steady flow of warm air through the duct from the front to the back of the refrigerator. While no method was available for measuring this rate of flow, visual inspection indicated that it would be sufficient to give the desired heat transfer effect.

FIGURE 10 SKETCH OF BASKET-TYPE ICE CHAMBER
WITH DUCT

TABLE IV

TEMPERATURES OF ICE BOX WITH ORIGINAL BASKET CHAMBER

Thermocouple Location	Time	
	6/2/48 8:00 A.M.	6/3/48 8:00 A.M.
First shelf under down-draft	42° F.	44° F.
Four inches above bottom	45	47
Side up-draft	49	50
Ambient	85	84

Seventh Series

Following the preliminary smoke tests, a temperature test was carried out using the modified basket chamber with a duct. Temperatures recorded during this test are given in Table V. Two significant facts were revealed during this test: (1) It was learned that the ice was extremely sensitive to changes in the level of the ice chamber. A slight slope of the racks was capable of causing the ice to shift its position several inches. (2) The metal area on which the ice rested would have to be reduced to a minimum. The rate at which the ice descended through the grids was dependent largely on this area.

To reduce this grid area to a minimum, every other rib in the grid was removed, runners were removed, and the ice chamber retested. The results recorded showed acceptable performance characteristics in all respects. At this point, the tests were ended.

TABLE V

TEMPERATURES OF ICE BOX WITH BASKET-DUCT CHAMBER

Thermocouple Location	6/12/48 10:00 A.M.	6/13/48 12:18 P.M.	6/14/48 8:00 A.M.	Re- iced	6/14/48 4:30 P.M.	6/15/48 8:00 A.M.
First shelf under down- draft	41° F.	42° F.	44° F.		42° F.	42° F
Four inches above bottom	43	45	46		44	44
Side up-draft	46	49	50		48	48
Ambient	85	84	85		86	86

CONCLUSIONS

A number of valuable conclusions can be drawn from the results obtained in testing the two types of ice chambers. Tests of the original ice chamber showed:

1. Satisfactory temperatures can be maintained in the refrigerator provided the circulation is adequate. To obtain adequate circulation, the thermal head must be as great as possible, and flow resistance must be reduced to a minimum.

2. Flat meltage can be obtained provided the ice chamber is accurately leveled before use and no appreciable circulation is permitted in the upper part of the ice chamber. Gaskets and seals must be used to prevent this circulation, and plates must be fastened to ice racks to prevent undercutting or upward circulation of air.

3. No dead air spaces should be permitted to exist in contact with the bottom of the ice. These air spaces insulate the ice from the circulating air and prevent satisfactory melting.

4. Static instability of the ice block may prove troublesome if V-shaped ice racks are used, resulting in tipping of the ice block. Some instability is inherent in this type of rack, although it may not be serious at a very slight slope. Ice guides used to prevent unstable shifting of the ice have only limited effect.

5. The basket-type ice chamber gives lower refrigerator temperatures than does the V-shaped rack because circulation over the ice need not be prevented to insure flat meltage.

6. The time required for the ice to sink below the level of the basket surface and permit reicing depends on the number of ribs and runners in the basket.

7. Rapid melting of ice in the center of the basket, where temperature differentials are ordinarily too low, can be obtained by installing a duct at that point to bring warm air in contact with the ice.

8. If basket-type ice chambers are to operate properly, they must be carefully leveled before use.

PART II. TEMPERATURE CONTROL IN TENTS - WINTER PHASE
INTRODUCTION

The maintenance of comfortable conditions in tentage in moderate or cold climate winter operations requires a supply of heat to the air within the tents. Even with a sizeable heat input to the tent, it is frequently impossible to maintain comfortable conditions; fabric layers, in addition to that of the tent structure itself, are needed to offer resistance to heat losses.

In the winter of 1946-47⁽¹⁾ tests in erected military tents equipped with liners were conducted to establish the relative effectiveness of liners of different materials and construction. In this work conventional and low profile liners of 4 oz. white cotton sheeting, 7.9 oz. cotton, O.D. 7, and a one-half inch thickness of fiberglas between two layers of 4 oz. cotton were examined. All the liners were effective in reducing heat losses, and relatively little difference was detected between liners made of fabrics of different weight and color as long as the fabric construction was essentially the same. The liner containing the fiberglas between the two layers of cotton sheeting proved to be the best overall insulator. However, the use of this latter liner was limited by its bulk and to some extent by its total weight.

In view of the results obtained with the liner containing the fiberglas insulation, an attempt was made to design a liner having an insulating value comparable to that of the two layers of sheeting and the one-half inch thickness of fiberglas without the disadvantageous weight and bulk.

A consideration of the heat transfer mechanism through the liner and tent and the results of previous tests with tents equipped with liners indicated that the concept of placing the major resistance to heat flow in the air films adjacent to the fabric surfaces was applicable in the case of the erected tents. This approach ignored the resistance offered by the fabric itself and was consistent with previous experimental results. These views then permitted the design of liners composed of several layers of lightweight cloth, providing additional air films for heat loss reduction without unduly increasing the weight and bulk of the liner.

Accordingly, two- and three-layer liners were manufactured by the Jeffersonville Quartermaster Depot and erected in large wall tents. The results of tests with these assemblies were then compared with previous results of experiments with single-layer liners and the liner containing the one-half inch thickness of fiberglass.

In this same series of tests the quantity of heat lost through the ground, as well as the variation of the heat loss with humidity change in the tent, was investigated.

THEORETICAL

Experimentations in winter phase temperature control in tentage have basically the object of securing reliable data permitting the best overall design of tents for cold climate usage. For this purpose calculations and correlations of temperature data can be carried out leading to overall coefficients of heat transfer, overall conductivity terms, or overall thermal resistance terms. Then a comparison of the numerical values of these terms for given tent and liner assemblies shows the superior items as far as heat loss and comfortable temperature maintenance is concerned.

From the data obtained in these winter phase studies, overall coefficients of heat transfer are calculated. However, since the data were obtained in tents erected outside, where the maintenance of constant conditions was impossible, certain calculations are necessary to eliminate variations in test conditions that would prevent good comparisons.

GROUNL LOSSES

The total heat input to the tents must leave either through the fabrics, seams and closures, or through the ground. Even though the heat loss through the ground is a variable quantity, the establishment of a reasonable value

for this loss allows a more accurate calculation of the heat loss through the tent and liner assembly. Two techniques were devised for the evaluation of the ground losses. The first method was the location of thermocouples in the ground at various depths down to one foot as shown in Figure 13. Since the amount of heat flowing into the ground is, among other factors, a function of the temperature of the air immediately above the ground and the ground temperature, both variables in these tests, it is possible to calculate, within limits, the heat flow into the ground by using the following equation:

$$Q = Wc_p\Delta t \quad (1)$$

where Q = Quantity of heat, BTU

W = weight of ground, lb.

c_p = specific heat of ground, BTU/(lb.)(°F)

Δt = temperature rise of the soil, °F.

The second method - quite similar to the method using the Nichols heat flow meter⁽²⁾ - gave more satisfactory results. Two thermocouples imbedded in the opposite faces of a white pine board with one face in contact with the ground and the upper face at ground level exposed to the air within the tent permitted the measurement of a temperature gradient. A knowledge of this temperature difference, the distance between the two thermocouples, and the thermal conductivity of the board allows the use of the equation for heat flow by conduction,

$$q = \frac{kA\Delta t}{x} \quad (2)$$

where k = thermal conductivity, BTU/(hr.)(sq.ft.)(°F/ft.)
 A = area at right angles to the direction of heat flow, sq. ft.
 Δt = the temperature difference between any two points, ° F.
 x = the distance between the two points, ft.
 q = quantity of heat flowing per unit time, BTU/hr.

SOLAR RADIATION

The temperatures realized within a tent of a given total heat input, as previously described, are greatly influenced by solar or sky radiation. During the day, radiation from the sun and/or sky adds heat to the tent; at night the tent loses heat by radiation to the cold sky and surroundings.

For good comparisons of test results obtained on different days and nights, it was necessary to account for this radiation. For this purpose, a reference tent identical to the test tent was used and equipped with thermocouples so arranged that readings could be made in the test tent. This reference tent was not internally heated but allowed to reach thermal equilibrium at all times. Then, the air temperature within this tent was the net result of radiation and

convection, i.e., in the daytime its temperature tends to be increased by radiation and lowered by convection, while at night the reverse is true.

Since the test tent and the reference tent were identical, it could be assumed with reasonable accuracy that the overall heat transfer coefficients were the same. Therefore, the total heat input into the test tent was the sum of the measured heat inputs and the net effect of the heat from solar radiation.

The heat lost by the test tent can be represented by the equation,

$$q_t = UA(t_{ia} - t_{oa}) \quad (3)$$

where q_t = heat lost, BTU/hr.

U = overall coefficient of heat transfer from the inside air to the outside air, BTU/(hr.)(sq.ft.)(°F)

A = area of the tent fabric, sq. ft.

t_{ia} = average inside air temperature, ° F.

t_{oa} = ambient outside air temperature, ° F.

The heat received by the reference tent from solar radiation may be assumed with small error to be lost by convection, so that at equilibrium

$$q_r = U_r A(t_{mr} - t_{ia}) = q_c = U_c A(t_{ia} - t_{oa}) \quad (4)$$

where q_r = heat received by radiation, BTU/hr.

q_c = heat lost by convection, BTU/hr.

U_r = overall radiation coefficient, BTU/(hr.)(sq.ft.)(°F)

U'_0 = overall convection coefficient, BTU/(hr.)
(sq. ft.)(°F)

A = fabric area, sq. ft.

t_{mr} = mean radiant temperature, ° F.

t'_{ia} = average inside air temperature of the
reference tent, ° F.

t_{oa} = ambient outside air temperature, ° F.

Then, the total heat lost from the test tent through the fabric becomes

$$q_T = q_h + q_r \quad (5)$$

where q_T = the total heat lost through the fabric,
BTU/hr.

q_h = the net heat supplied by the heaters, BTU/hr.

q_r = the heat gained by radiation, BTU/hr.

or $q_h = q_T - q_r$

Since the reference and test tents are identical, a value of q_r can be substituted from the reference tent giving the expression,

$$q_h = UA(t_{ia} - t_{oa}) - U'_0 A(t'_{ia} - t_{oa}) \quad (6)$$

and since the tents are identical, U and U'_0 can be taken equal with little error, or

$$q_h = UA(t_{ia} - t'_{ia}). \quad (7)$$

In other words, the coefficient of heat transfer from the inside air to the outside air can be calculated from the net heat supplied by the heaters, the fabric area, and the air temperature difference between the test and reference tents.

When the sky was heavily overcast, day or night, the reference tent temperatures approached the ambient temperature and calculations could be made in the usual manner without the use of the reference tent.

Although the method was satisfactory for calculating the data obtained in these tests, the technique would not apply if the surface temperatures of reference and test tents differed appreciably.

Figure 11 shows a comparison of these surface temperatures for a typical day.

INDIVIDUAL COEFFICIENTS

The insulating value of the tent or tent and liner assembly depends, among other things, upon the quantity of heat transferred per unit time per unit area of tent fabric under a given temperature differential driving force. This value is fundamentally a property of the fabric and the environment in which it exists and can be represented by a resistance term, a series of resistance terms, or by conductivities. In these studies the conductivity concept has been used, and overall coefficients of heat transfer have been calculated. High values of these coefficients indicate high heat losses, and low values indicate lesser heat losses or a better insulating system.

FIGURE 11 TIME-TEMPERATURE RELATIONS FOR LARGE
WALL TENTS WITHOUT LINERS

Just as the total thermal resistance of a heat transfer system can be calculated from individual resistances, so can the overall coefficients of heat transfer be calculated from the individual coefficients. In this work the entire resistance to the heat losses has been considered to lie in the air films with negligible resistance in the fabric itself. Accordingly, the overall coefficients of heat transfer can be calculated from the individual film coefficients.

In this case, where the overall coefficient of heat transfer corresponds to heat transfer from air inside the tent to air outside the tent, the film coefficient corresponds to the heat transfer, for example, from the air within the tent to the tent fabric. However, within the test tent the process is complicated by heat transfer to the fabric by convection, radiation, and conduction; and for a simplification of the concept, all heat transfer is considered to be by convection and radiation. Therefore, the heat transferred by convection is characterized by a convection film coefficient and that by radiation, by a radiation coefficient.

The heat radiated from the shield of the heating unit can be calculated by the equation,

$$q_r = 0.172AF_sF_a \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_f}{100} \right)^4 \right] \quad (8)$$

where

q_r = heat radiated from heater shield, BTU/hr.

T_s = temperature of surface of heating unit, ° R.

T_f = temperature of the fabric receiving the radiation, ° R.

A = area of the heating unit, sq.ft.

P_e = emissivity factor (in this case, 0.276 for grey galvanized sheet iron)⁽³⁾

P_a = area factor (in this case, 1.0)

With the knowledge of q_r from the equation above a radiation coefficient can be calculated.

$$q_r = h_r A (t_{ia} - t_f) \quad (9)$$

where h_r = inside radiation coefficient, BTU/(hr.) (sq.ft.)(°F)

A = area of the tent in sight of the heater, sq. ft.

t_{ia} = inside air temperature, ° F.

t_f = fabric temperature, ° F.

The convection coefficient can be calculated using the conventional equation,

$$q_c = h_c A (t_{ia} - t_f) \quad (10)$$

where q_c = heat transferred by convection from inside air to the fabric, BTU/hr.

h_c = inside convection coefficient, BTU/(hr.) (sq.ft.)(°F)

A = area of tent fabric, sq. ft.

t_{ia} = inside air temperature, ° F.

t_f = fabric temperature, ° F.

The heat loss from the outside of the tent by radiation was obtained by considering the tent fabric as radiating to the surroundings at an average temperature approximately

equal to the ground temperature outside the tent. Using the radiation equation,

$$q_r = 0.172AF_e F_a \left[\left(\frac{T_f}{100} \right)^4 - \left(\frac{T_g}{100} \right)^4 \right] \quad (8-a)$$

where q_r = heat lost from tent surface to surroundings,
BTU/hr.

A = area of tent fabric, sq. ft.

F_e = emissivity factor

F_a = area factor

T_f = fabric temperature, ° R.

T_g = temperature of surroundings, taken equal to
outside ground temperature, ° R.

a value of q_r was obtained. Then, using this radiation heat loss, a value of h_r was obtained from the equation,

$$q_r = h_r A (t_f - t_{oa}) \quad (9-a)$$

where h_r = radiation coefficient for outside of the
fabric, BTU/(hr.)(sq.ft.)(°F)

A = area of the fabric, sq. ft.

t_f = fabric temperature, ° F.

t_{oa} = outside ambient temperature, ° F.

Subtracting the outside radiation loss from the total loss through the fabric, a value for external convection losses is obtained. Using this value as a convection loss, a convection film coefficient can be calculated for the external surface with the equation,

$$q_c = h_c A (t_f - t_{oa}) \quad (10-a)$$

giving a value for h_c

A grouping of the four coefficients as follows for the establishment of an overall coefficient

$$\frac{1}{U} = \frac{1}{h'_c + h'_r} + \frac{1}{h_c + h_r} \quad (11)$$

gives a value of U , the overall coefficient.

This method of calculating individual coefficients and their use in establishing a value of the overall coefficient, agreeing well with those values obtained by other means, was considered a good substantiation of the validity of the procedure.

EXPERIMENTAL

In these winter phase temperature control studies, two large wall tents were the basic units used in the tests. These tents were equipped with ground cloths of 12.29 oz. duck with J.C.P. 242 finish, liners, and electric heaters. Three different types of liners were used in the investigation: (1) 3.6 oz. cotton, 2 layers of cloth; (2) 4 oz. cotton, 2 layers of cloth, inner fabric, dark green, vinyl coated; and (3) 4 oz. cotton, white, three layers of cloth.

The liners were suspended within the tents so that an air space of approximately six inches was formed between the tent and liner. The suspension of the liners was by means of tapes over the ridge pole and tapes through metal rings on the fabric of the tent. The liners were of sufficient length to close the air space between the tent and the liner at the ground. The ends of the tents and liners (or fronts and backs) were closed by snap fasteners or hand sewing leaving only enough clearance for a very low entrance to the tent.

The two-layer experimental liners were constructed with lines of stitching eight inches apart running from the ground to the ridge. The fabric toward the interior of the tent measured ten inches between the stitching so that there was a separation of the two layers of cloth over most of the liner. In the case of the three layer liners the same fabrication techniques were employed. However, the stitching did not penetrate the three layers, i.e., the lines of stitching

for the outer and intermediate layers were midway between the lines of stitching for the intermediate and inner layers. A sketch of the arrangements of the stitching and fabrics in the tents with two- and three-layer liners is shown in Figures 12 and 13.

The heating units for these test tents consisted of six 550-watt cone heaters with socket bases mounted on transite and shielded with galvanized iron to prevent direct radiation to the fabric. A 220-110 volt outlet switch box permitted connections to the heaters and a light source. Occasionally, two of these six unit heater assemblies were used in parallel for increased heat input to the tent. The actual heat input to the tent by the heaters was obtained from voltmeter and ammeter readings.

In addition to this heat input, a light bulb supplied the necessary lighting, and two occupants remained in the test tent during the tests. Since the occupants were lightly clothed and more or less at rest, their individual heat output was about 400 BTU per hour⁽⁴⁾. Therefore, the wattage of the light bulb and the bodily heat of the occupants were added to the heat supplied by the heaters to obtain the total heat input to the test tents.

Previous winter phase studies showed that the average inside air temperature of large wall tents coincided with the air temperature at 3.5 feet above the floor⁽⁵⁾. In these tests four thermocouples, each 4 feet from a corner of

FIGURE 12 DIAGRAMMATIC SKETCH OF LARGE WALL
TENT WITH TWO-LAYER LINER

FIGURE 13 DIAGRAMMATIC SKETCH OF LARGE WALL
TENT WITH THREE-LAYER LINER

the tent at the 3.5 feet level and connected in parallel to the potentiometer, were used to indicate the average air temperature within the tent. Likewise, in the case of fabric temperatures, the thermocouples were sewn to the fabric of the tent or liner so that each thermocouple measured the temperature of an equal area of fabric. These thermocouples, connected in parallel to the potentiometer, gave the average fabric temperature in a single reading. All thermocouples were made of 30 gage copper-constantan wire. Using circuits of this type, greater speed was possible and in all probability greater accuracy, since in the two minutes required for making all readings, changing weather could hardly have produced measurable temperature changes.

Air velocity measurements in the vicinity of the test tents were obtained with a three-cup anemometer seven feet from the ground.

Heat losses through the ground cloth were estimated from temperature measurements with thermocouples imbedded on both sides of a pine board (8" x 4" x 9/16") beneath the ground cloth. This heat flow meter was placed midway between the center and edge of the tent so that readings would indicate an average heat flow into the ground. Thermocouples imbedded in the ground at intervals down to one foot gave a means of checking the heat flow meter.

In carrying out the winter phase tests, the tents were erected with ridge poles North-South and equipped with

the apparatus described above. Thermocouples and thermometers were located as shown in the sketch of Figure 14 and as described in Table VI.

In order to evaluate the influence of solar radiation properly, a tent identical to the test tent was erected seven feet from the test tent. This tent, referred to as the reference tent, was completely closed and all thermocouple wires were connected to a selector switch within the test tent.

All data were obtained after the temperatures at the various measuring points indicated equilibrium conditions and were recorded on the data sheets as shown in Figure 15. Wind velocities were recorded before and after each series of runs. However, in most cases the wind velocities were low enough to be negligible.

Whenever possible, the tests were conducted in clear weather with very little wind. In order to minimize solar radiation effects, morning and evening runs were made, as well as a number of night runs. A total of twenty-two tests, each consisting of several groups of equilibrium runs, were made at approximately one hour intervals. The summarized data for these tests are shown in Tables VII, VIII and IX.

Ground loss values were found to vary from 300 to 2100 BTU per hour for the various test conditions with a relative value of about 10 per cent of the total heat input to the tent. There was actually some variation in these

FIGURE 14 LOCATION OF TEMPERATURE MEASUREMENTS

TENT TEMPERATURE DATA - WINTER

Large Wall Tent-12.29 oz. Duck

Equilibrium Conditions

Test No. _____ Date _____ Weather _____

Type Liner _____

TEMPERATURE (°F.)

No.	Location (Fig.5)	TIME											
0	A												
1	B												
2	C												
3	D												
4	E												
5	F												
7	D'												
8	C'												
9	B'												
10	A'												
0	I												
1	J												
2	K												
3	L												
4	M												
5	N												
6	O												
7	P												
1	H												
2	H-1"												
3	H-3"												
4	H-6"												
5	H-12"												
6	G												
7	G-1"												
8	G-3"												
9	G-6"												
10	G-12"												
Voltage													
Current													
Wind (MPH)													

FIGURE 15 DATA SHEET FOR EQUILIBRIUM CONDITIONS

TABLE VI
DESCRIPTION OF TEMPERATURE MEASUREMENTS

- A - Inside air $3\frac{1}{2}$ ft. above floor, 4 thermocouples in parallel, 1 in each corner 4 ft. from the walls.
 - B - Tent fabric, inside, 10 thermocouples in parallel, 2 on side walls, 4 on deck, 2 on each end.
 - C - Ground cloth, 2 thermocouples in parallel in alternate corners of tent, 4 ft. from the wall.
 - D - Inside liner, 8 thermocouples in parallel, 2 on side walls, 2 on deck, 2 on each end.
 - E - Outside ground, 1 thermocouple, 3 ft. from northeast corner of test tent, 1 inch below surface.
 - F - Outside air, 1 thermocouple, 4 ft. from ground, 6 ft. from northwest corner of test tent.
 - G - Ground, 5 separate thermocouples, 1 ft. from heater on the surface of ground, 1", 3", 6" and 12" below the surface.
 - H - Same as G only located at the center of the east side of the tent.
 - I - Liner, outside, 1 thermocouple, not on seam, center of east deck.
 - J - Liner, outside, 1 thermocouple, on seam, center of east deck.
 - K - Liner, inside, 1 thermocouple, not on seam, center of east deck.
 - L - Liner, inside, 1 thermocouple, on seam, center of east deck.
 - M - Tent fabric, outside, 1 thermocouple, center of east deck.
 - N - Tent fabric, inside, 1 thermocouple, center of east deck.
 - O - Top of flow meter white pine wood $\frac{9}{16}$ " thick
 $k = 0.1$ BTU
 - P - Bottom of flow meter $\frac{(hr.) (sq.ft.) (°F/ft.)}{k}$
- 1, 2, 3, 4, 5, 6, 7 - Inside air thermometers, 1", 1', $2\frac{1}{2}'$, $3\frac{1}{2}'$, 5', $6\frac{1}{2}'$, 8' above the floor level, respectively.

Note: All letters with primes are the same thermocouple arrangements in the reference tent.

TABLE VII SUMMARY OF TEMPERATURE DATA

LARGE HALL TENT - NO LINER

Test No.	Date	Time	Wind	Weather	Total Heat Input	Heat Loss Thru Ground	Heat Loss Thru Fabric	Test Tent Tempera- tures of		Reference Tent Tempera- tures, of		Ambient Air Temp. of
					BTU Per Hour			t_{ia}	t_f	t_{ia}	t_f	t_{oa}
1	1/27	9:45 A.M.	negl.	Fair	12,200	10%	11,000	47	34	27	27	9
2	1/27	10:15 A.M.	"	"	"	"	"	47	33	28	30	8
3	1/27	10:30 A.M.	"	"	"	"	"	54	44	34	35	12
4	1/28	2:15 P.M.	"	"	12,000	"	10,800	68	53	45	49	20
5	1/28	4:00 P.M.	"	"	"	"	"	48	35	26	26	20
6	2/2	10:30 A.M.	"	Sunny	"	"	"	57	48	40	40	26
7	2/2	12:30 P.M.	"	"	"	"	"	78	61	51	54	28
8	2/2	1:45 P.M.	"	"	"	"	"	76	60	57	61	35
9	2/2	2:35 P.M.	"	"	"	"	"	72	58	52	58	34
10	2/2	4:50 P.M.	"	"	"	"	"	52	43	32	32	36
11	2/6	12:00 M	"	Night	"	"	"	40	29	21	19	18
12	2/11	3:30 P.M.	"	Cloudy	18,000	"	16,200	78	64	48	48	44
13	2/11	12:00 M	"	Rain Night	22,500	2100	20,400	68	49	35	35	35

 t_{ia} - mean air temperature within tent t_f - mean tent fabric temperature t_{oa} - ambient outside air temperature

TABLE VIII SUMMARY OF TEMPERATURE DATA

LARGE WALL TENT - TWO LAYER LINER

Test No.	Date	Time	Wind	Weather	Total Heat Input	Heat Loss Thru Ground	Heat Loss Thru Fabric	Test Tent Temperatures of			Reference Tent Temperatures, of		Ambient Air Temp. of
					BTU Per Hour			t _{ia}	t _l	t _f	t' _{ia}	t' _f	t _{oa}
14	3/9	12:00 M	Negl.	Overcast	11,900	1800	10,100	80	70	46	40	40	40
15	3/11	5:45 A.M.	4mph	Night Cloudy	17,800	2100	15,700	80	68	33	32	28	24
16	3/11	1:30 P.M.	Negl.	Clear	14,800	1800	13,000	88	76	47	37	36	25
17	3/11	7:30 P.M.	"	"	"	1200	12,800	80	67	34	28	23	23
18	3/12	5:30 A.M.	"	"	"	300	14,500	72	59	26	21	15	15
19	3/12	8:50 A.M.	"	"	"	1500	13,300	89	77	53	35	30	26
20	3/18	5:45 A.M.	"	"	11,000	600	10,400	77	70	45	39	33	40
21	3/18	6:35 A.M.	"	"	"	"	"	75	69	48	40	38	42
22	3/18	7:00 A.M.	"	"	"	1200	9,800	77	72	52	43	43	43
23	3/18	9:30 A.M.	"	"	10,700	3300	7,400	99	86	80	67	80	60
24	3/25	1:00 A.M.	"	"	11,100	600	10,500	85	77	53	46	46	48
25	3/25	2:00 A.M.	"	"	10,650	"	10,050	80	73	51	46	47	49
26	3/25	3:00 A.M.	"	"	10,550	900	9,650	78	72	50	46	46	50
27	3/25	4:00 A.M.	"	"	10,500	"	9,600	77	70	49	45	44	46
28	3/25	6:00 A.M.	"	"	10,600	600	10,000	76	70	49	44	44	44
29	3/30	1:30 A.M.	"	"	11,050	"	10,450	79	75	51	46	46	49
30	3/30	2:30 A.M.	"	"	10,450	"	10,850	74	70	50	46	47	48
31	3/30	3:30 A.M.	"	"	10,550	"	9,950	72	69	49	44	48	46
32	3/30	4:30 A.M.	"	"	13,700	"	13,100	80	75	50	43	--	45
33	3/30	5:30 A.M.	"	"	13,450	900	12,550	78	75	50	43	--	45
34	3/30	6:15 A.M.	"	"	13,700	1200	12,500	82	80	52	47	--	48

TABLE VIII (continued)

Test No.	Date	Time	Wind	Weather	Total Heat Input	Heat Loss Thru Ground	Heat Loss Thru Fabric	Test Tent Tempera- tures OF			Reference Tent Tempera- tures, OF		Ambient Air Temp. OF
					BTU Per Hour			t_{ia}	t_l	t_f	t'_{ia}	t'_f	t_{oa}
35	4/1	12:00 N	Negl.	Overcast	11,000	2100	8,900	86	84	80	59	--	51
				rain									
36	4/1	4:30 P.M.	"	"	"	900	10,100	82	80	64	56	--	51
37	4/3	1:45 A.M.	"	Clear	"	300	10,700	65	62	38	39	37	38
38	4/3	3:00 A.M.	"	"	16,600	"	16,300	74	72	45	37	36	36
39	4/3	4:00 A.M.	"	"	"	600	16,000	78	74	42	36	35	34
40	4/3	5:00 A.M.	"	"	"	"	"	77	74	42	37	35	35

t_{ia} - mean air temperature within tent;

t_l - mean liner temperature

t_f - mean tent fabric temperature

t_{oa} - ambient outside air temperature

Tests 14-28 - 3.6 oz. cotton

28-40 - 4.0 oz. cotton, inner layer, vinyl-coated

TABLE IX SUMMARY OF TEMPERATURE DATA

LARGE HALL TEST - THREE LAYER LINER

Test No.	Date	Time	Wind	Weather	Total Heat Input	Heat Loss Thru Ground	Heat Loss Thru Fabric	Test Tent Tempera- tures OF			Reference Tent Tempera- tures, OF		Ambient Air Temp. OF
					BTU Per Hour			t _{ia}	t _l	t _f	t _{ia}	t _f	t _{oa}
41	2/21	1:06 A.M.	3.5mph	Cloudy	15,500	1500	14,000	85	76	38	31	25	26
42	2/21	3:00 P.M.	Negl.	Snow	14,900	"	13,400	82	72	36	31	26	24
43	3/1	11:50 A.M.	"	Cloudy Rain	8,650	2100	6,550	82	77	66	58	70	45
44	3/1	4:30 P.M.	"	"	"	1200	7,450	76	74	56	53	53	40
45	3/1	10:50 P.M.	"	"	"	1500	7,150	76	70	47	44	42	42
46	3/1	12:00 M	"	"	"	1200	7,450	75	70	45	43	41	41
47	3/3	10:30 A.M.	"	Overcast	8,350	"	7,150	73	67	73	44	47	40
48	3/3	1:20 P.M.	"	"	"	1800	6,550	84	78	84	56	58	46
49	3/3	4:30 P.M.	"	"	"	600	7,750	72	66	72	46	42	42
50	3/3	9:30 P.M.	"	"	"	900	7,450	70	65	70	39	36	36
51	3/5	11:00 A.M.	"	Clear	"	1800	6,550	80	74	58	54	67	42
52	3/5	9:00 P.M.	"	"	"	1200	7,150	70	65	40	38	34	37
53	3/7	12:15 A.M.	1.8mph	"	11,800	"	10,600	73	66	34	32	24	31

t_{ia} - mean air temperature within tentt_l - mean liner temperaturet_f - mean tent fabric temperaturet_{oa} - ambient outside air temperature

ground losses, values above 10 per cent with low heat inputs and lesser values as the ground became warmer with long heating periods or when the outside ground temperature increased. Figure 16 shows the change in ground temperature with time.

The heat flow meter gave a satisfactory measure of the ground losses; however, it was possible to check the results within limits using the ground temperature measurements. During a period of 14.6 hours, the heat content of the ground down to the 12 inch level was increased by 18,490 BTU, and in this same period the total heat supplied to the tent was 126,000 BTU, indicating 14.7 per cent of the total heat to the tent was lost to the ground. Over the same period a ground loss of 16.2 per cent was calculated from measurements with the heat flow meter.

In these runs in which the flow meter was used, the heat loss to the ground was calculated. In other tests the ground loss was assumed to be 10 per cent of the total heat loss. These losses were subtracted from the total heat input to the tent to obtain the heat transferred through the fabric.

Using the techniques of calculating the total heat input, ground loss, and radiation effects as previously described, the overall coefficients of heat transfer from the air inside the tent to the outside air were obtained for all test runs. Average values of these coefficients for the different tent assemblies are shown in Table X.

FIGURE 10 GROUND TEMPERATURE VARIATION AT THE CENTER OF TENT

TABLE X
OVERALL HEAT TRANSFER COEFFICIENTS FOR LARGE WALL TENTS WITH
MULTILAYER LINERS

Assembly	Overall Coefficient* BTU/(hr.)(sq.ft.)(°F)
Large Wall Tent, No Liner	0.92
Large Wall Tent, 2 Layers, 3.6 oz. Balloon Cloth Liner	0.45
Large Wall Tent, 2 Layers 4 oz. Cotton, Vinyl-coated, Liner	0.54
Large Wall Tent, 3 Layers, 4 oz. Cotton, Liner	0.41

* Wind Velocity Negligible

In the tests of the Winter of 1946-47 coefficients for the large wall tents with single layer liners were obtained as shown in the following table⁽⁶⁾:

TABLE XI
OVERALL HEAT TRANSFER COEFFICIENTS FOR LARGE WALL TENTS WITH
SINGLE LAYER LINERS

Assembly	Overall Coefficient* BTU/(hr.)(sq.ft.)(°F)
Large Wall Tent, No Liner	1.26
Large Wall Tent, 4 oz. White Cotton, Liner	0.58
Large Wall Tent, 8 oz. Cotton, O.D., 7, Liner	0.58
Large Wall Tent, $\frac{1}{2}$ in. Thickness of Fiberglas Fine Fiber between 4 oz. Cotton, Liner	0.39

* Wind Velocity Negligible

An inspection of the values in these tables, Table X and Table XI, reveals a lower value of the coefficient for the tent with no liner for the winter of 1947-48 than that obtained in 1946-47. This lesser value was attributed to the lower air velocities, essentially zero, encountered in working with the tent with no liner during the winter of 1947-48. Since the tent with no liner depends upon only two air films for resistance to heat flow, the value of the overall coefficient is seriously influenced by slight changes in external conditions. However, in the case of tents with liners slight changes in external conditions are less noticeable in the value of the overall coefficient.

A consideration of the conductances of air spaces and individual coefficients as reported for structurally stable materials, such as, concrete, metal, glass, etc. shows the ultimate insulation that can be expected of the fabric system in which the resistance offered by the fabric is negligible.

For the type of surface similar to a cotton liner an inside surface coefficient of 1.85 is obtained. Since the wind velocity is negligible in most of these runs, an outside coefficient of 2.0 should apply. For the conductances of the air spaces at a mean temperature of 40° F. a value of 1.1 can be used. (These values agree with those reported in the American Society of Heating and Ventilating Engineers Guide.)(7)

If the tent structure were stable and the air spaces were equivalent to still air, the values of the overall coefficients as calculated for Table XII would be obtained.

TABLE XII
EXPERIMENTAL AND CALCULATED OVERALL HEAT TRANSFER COEFFICIENTS

Assembly	Overall Coefficients BTU/(hr.)(sq.ft.)(°F)	
	Experimental	Calculated
Large Wall Tent, No Liner	0.92	0.96
Large Wall Tent, Single Layer Liner	0.58	0.51
Large Wall Tent, Double Layer Liner	0.45	0.35
Large Wall Tent, Three Layer Liner	0.41	0.27

Since a tent is not structurally stable (as contrasted with a concrete wall) but moves with even the slightest air movement, true still air spaces would hardly be expected. In addition, there is undoubtedly considerable air infiltration through seams, grommet holes, door openings, etc., and in the case of liners of more than one layer, air spaces are destroyed at the lines of stitching holding the two fabric layers together. In many places the liners would not hang in such a way that all possible separation of the fabric layers was realized. All these considerations would tend to increase the experimental values of the coefficients above the calculated values.

From the comparisons of Table XII it appears that the double layer liner is the most efficient in reducing the heat losses for the weight and cost involved. The addition of insulation realized with this three-layer liner is not sufficient to justify the additional weight and cost. Although the liner incorporating the fiberglass (previously examined) gave a coefficient of 0.39, whereas the three-layer liner gave a value of 0.41, its cost, weight, and bulk did not warrant its recommendation for portable tentage use. In addition, it is felt that some changes in the fabricating technique for the multi-layer liners to give maximum utilization of the possible air spaces would increase their value for tentage insulation.

HIGH HUMIDITY WITHIN THE TENT

In most of the runs the partial pressure difference of the water vapor in the inside and outside air was too small for a consideration of diffusion or condensation. To study the effects of a significant vapor pressure difference between the inside and outside air, several runs were made while water was being boiled within the tents.

Two 110-volt heaters were connected in series and paralleled with the heating units. One quart containers were placed on these heaters and water was boiled continuously during the equilibrium runs. Therefore, a portion of the

total electrical power supplied to the tent was expended in vaporizing the water. This vapor, at approximately 212° F., transferred sensible heat to the air vapor mixture.

A calculation of the overall heat transfer coefficients for these runs in the manner previously described gave high values, 0.60 (0.54 with no evaporation) for the two-layer vinyl-coated liner and 0.50 (0.45 with no evaporation) for the two layer 3.6 oz. cotton liner. However, when the heat supplied for the evaporation of the water was deducted from the total heat input and coefficients calculated on this basis, abnormally low values were obtained indicating that some heat was gained by condensation of the moisture. By using the two different values of the coefficients to calculate the heat obtained in the condensation with the value of the latent heat, the amount of condensation was calculated.

For a partial pressure difference of 8.6 mm Hg, it was found by the calculation procedure described above that 50 per cent of the evaporated water condensed when the 3.6 oz. cotton liner was used, although no condensation was observed on the surface of the liner exposed to the interior of the tent. Therefore, it was concluded that the condensation took place on the outer fabric layers.

When the vinyl-coated liner was used with a partial pressure difference of 14.6 mm Hg, 41 per cent of the evaporated water condensed. In this case, condensation was observed on the liner surface to the inside of the tent, indicating that little water condensed in the outer fabric layers.

CONCLUSIONS

From the results of this study liners were found to be an effective means of reducing heat loss from tents. The comparative values of the heat loss coefficients gave good indications of the value of liner-tent combinations over the range of wind velocities encountered in these tests. A tent with no liner was found to have a coefficient of heat loss of $0.92 \text{ BTU}/(\text{hr.})(\text{sq.ft.})(^{\circ}\text{F})$, whereas the coefficients for the single layer liner, double layer liner and three layer liner were found to be 0.58, 0.45, and $0.41 \text{ BTU}/(\text{hr.})(\text{sq.ft.})(^{\circ}\text{F})$, respectively.

All of the experimental coefficients for the liner-tent assemblies were found to be higher than calculated coefficients for the same assemblies. Since all calculated coefficients were based on data for stable structures, the reason for this difference is readily seen. Air infiltration was neglected; true still air spaces would hardly be expected and air spaces were not always uniform. All of these considerations would tend to increase the experimental values of the coefficients.

From the results obtained, multi-layer liners were found to be sufficiently effective for reducing heat losses from tentage to justify their use. Although the three layer liner gave an average overall coefficient slightly below that for the two layer liner, the double layer liner was considered the most satisfactory from the viewpoint of heat loss, bulk, weight and cost.

LITERATURE CITED

- (1) Barnes, T.H., Harvin, E.L., Williams, G.W., and Gerhard, E.F., "Improved Design of Tents and Tentage Materials July, 1946 - June, 1947" University of Louisville Institute of Industrial Research, Louisville, Ky., 1947 pp. 22-85
- (2) Nicholls, P., "Measuring Heat Transmission in Building Structures and a Heat Transmission Meter", ASHVE Transactions, 1924, Vol. 30, pp. 356-369
- (3) McAdams, W.H., "Heat Transmission", 2nd edition, McGraw-Hill Book Company, New York, 1942 pp. 394
- (4) Haber and Hutchinson, "Panel Heating and Cooling Analysis", 1st edition, Vol. I, John Wiley and Sons, New York, 1947 pp. 61-62
- (5) Barnes, T.H., op. cit., p. 33
- (6) Ibid., p. 43
- (7) Severns, Wm. H., "Heating, Ventilating and Air Conditioning Fundamentals", John Wiley and Sons, New York, 1946 pp. 47-48

APPENDIX

LIST OF SYMBOLS

Name	Symbols	Units
Area	A	sq.ft.
Coefficient of Heat Transfer, Convection	h_c	BTU/(hr.)(sq.ft.)(°F)
Coefficient of Heat Transfer, Convection, Overall	U_c	"
Coefficient of Heat Transfer, Overall	U	"
Coefficient of Heat Transfer, Radiation	h_r	"
Coefficient of Heat Transfer, Radiation, Overall	U_r	"
Distance	x	ft.
Emissivity Factor, Radiation	F_e	
Heat Flow, Quantity	Q	BTU
Heat Flow, Rate	q	BTU/hr.
Heat Flow, Rate, Lost by Convection	q_c	"
Heat Flow, Rate, Lost by Test Tent	q_t	"
Heat Flow, Rate, Received by Radiation	q_r	"
Heat Flow, Rate, Supplied by Heater	q_h	"
Heat Flow, Rate, Total Heat Lost	q_T	"
Specific Heat	c_p	BTU/(lb.)(°F)
Temperature, Average, Inside Air	t_{ia}	°F
Temperature, Average, Outside Air	t_{oa}	"
Temperature, Difference	Δt	"
Temperature, Fabric	t_f	"
Temperature, Fabric, Absolute	T_f	°R

LIST OF SYMBOLS (continued)

Name	Symbols	Units
Temperature, Ground, Absolute	T_g	OR
Temperature, Mean Radiant	t_{mr}	°F
Temperature, Surface, Absolute	T_s	OR
Thermal Conductivity	k	BTU/(hr.)(sq.ft.)(°F/ft.)
Weight	W	lb.

NOTE: All primes (') refer to reference tent.

CALCULATION OF OVERALL HEAT TRANSFER COEFFICIENT

Using data for run number 50, the total heat input was 8,350 BTU/hr. The heat loss to the ground was 900 BTU/hr., as calculated using the heat flow meter.

From equation (7),

$$q_h = UA(t_{ia} - t'_{ia})$$

where $q_h = (8,350 - 900) = 7,450$ BTU/hr.

$$A = 600 \text{ ft.}^2$$

$$t_{ia} = 70^\circ \text{ F.}$$

$$t'_{ia} = 39^\circ \text{ F.}$$

a value for U of 0.40 BTU/(hr.)(sq.ft.)(°F) is obtained.

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VITA

The author of this thesis, Robert Beaman Luggan, was born April 22, 1926, in Denver, Colorado, the son of Orton L. Luggan and Frances Clemens Luggan. He attended grade school in Dallas, Texas and Knoxville, Tennessee and Central High School in Knoxville, from which he was graduated in 1944. He then enlisted in the Navy V-12 Program and spent one year in training at Milligan College in Milligan, Tennessee, and one year at Speed Scientific School in Louisville, Kentucky. Upon being discharged, he continued his studies at Speed Scientific School, from which he received the degree of Bachelor of Mechanical Engineering in September, 1947.

Subsequent to his graduation, he enrolled in the graduate school of the University of Louisville and received his Master of Mechanical Engineering degree in June, 1949. His master's work included a fellowship in the Institute of Industrial Research, University of Louisville, where he worked on the project, "Improved Design of Tents and Tentage Materials," sponsored by the Research and Development branch of the Office of Quartermaster General and "Improved Ice Meltage Characteristics of the Progress Ice Box," sponsored by the Louisville Tin and Stove Company.