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Convective heat transfer in small enclosed air spaces.

Joseph Charles Kroutil

CONVECTIVE HEAT TRANSFER

IN SMALL ENCLOSED AIR SPACES

BY

JOSEPH CHARLES KROUTIL

A

THESIS

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Approved by

Harry J. Sauer Jr. (Advisor) Robert Gerson

Aaron J. Miles C. R. Remington Jr.

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF ILLUSTRATIONS

NOMENCLATURE

Symbol	Physical Quantity	Units
Fr	Froude number (V^2/Lg)	dimensionless
Gr	Grashof number ($\beta g L^3 \Delta t/\nu^2$)	dimensionless
Gr,m	Modified Grashof number ($\beta g L^3 \Delta t/\nu\alpha$)	dimensionless
Nu	Nusselt number (hL/k)	dimensionless
Pr	Prandtl number ($\mu c_p/k$)	dimensionless
Ra	Rayleigh number (Pr x Gr)	dimensionless
c_p	Specific heat at constant pressure	Btu/(lb mass)°F
D	Diameter	ft
g	Gravitational acceleration	ft/hr^2
h	Mean coefficient of heat transfer by free convection	Btu/hr ft^2 °F
k	Thermal conductivity	Btu/hr ft °F
L	Length or layer thickness	ft
q_a/A	Heat flow per unit area due to the air	Btu/hr ft^2
q_d	Heat flow disc output	microvolts
T_c	Temperature on cold side of air chamber	°F
T_h	Temperature on hot side of air chamber	°F
T_m	Mean temperature in air chamber	°F
U	Overall coefficient of transmission due to the air	Btu/hr ft^2 °F
V	Velocity	ft/hr
α	Thermal diffusivity	ft^2/hr
β	Thermal coefficient of cubical expansion	1/°F

Δt	Temperature difference	^{o}F
μ	Dynamic viscosity	lb m/hr ft
ν	Kinematic viscosity	ft^2/hr

INTRODUCTION

With the growing use of porous materials, insulation, and honeycomb structures in modern aircraft, space vehicles, and many other installations, more information on small enclosed spaces is needed to analyze the heat transfer and thermal stress problems completely.

The contact of any fluid with a hotter surface reduces the density of the fluid and tends to cause it to rise. If the surface is colder than the fluid, the fluid tends to fall. Circulation is set up by the difference in the temperature of the fluid and the surface with which it is in contact. This circulation is called free or natural convection.

The main objective of this investigation was to determine qualitatively the magnitude of natural convective heat transfer in small enclosed air spaces.

SURVEY OF LITERATURE

In surveying literature, the author has found very little information on the subject of natural or free convective heat transfer in small completely enclosed spaces. Investigations have been limited mostly to horizontal air layers heated from below.

The first experimental investigation on convection currents in horizontal layers was made by Thomson (31)* in 1881. He worked with layers of soapy water standing on a leveled metallic plate. The plate was maintained at a uniform temperature, and the soapy water was convectively cooled from the surface. In 1901, Benard (1) found that a thin layer of spermaceti when heated from below produced a cellular motion of a distinct regular pattern that looked like a honeycomb. This pattern shown in Fig. 1 has since come to be known as the "Cells of Benard".

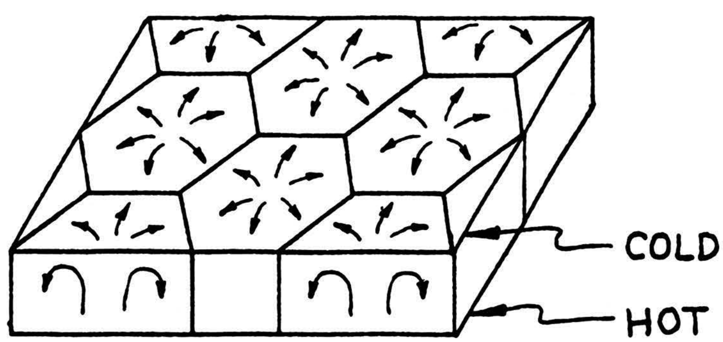

Fig. 1. Cells of Benard

The warmer fluid rose at the centers of the hexagonal prisms and the colder fluid descended at the outer edges.

Experiments show that the direction of circulation changes from fluid to fluid. Liquids and gases usually have opposite directions of circulation.

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*Numbers in parentheses refer to reference listed in the bibliography.

Liquids rise and gases descend in the centers of the cells when heated from below.

Graham (10) suggested that the different circulation may be due to the fact that the kinematic viscosity varies with temperature in opposite manner in liquids and gases. This suggestion was confirmed experimentally by Tippelskirch (32) using liquid sulphur, which has the property that the kinematic viscosity decreases or increases with temperature depending upon whether the temperature is less or greater than 153° C. Later Palm (24) showed mathematically that the direction of circulation within the cell does depend on kinematic viscosity variation with temperature.

There has been some question concerning the final form of the cells. The first contribution to the theory of convection was given by Lord Rayleigh (26) in 1916. However, his theory did not explain any final cell pattern, squares were as likely as hexagons. Some work was done by Malkus and Veronis (19) on the final cell pattern. They tried to find the physically realized solution by examining the stability of various solutions of non-linear equations corresponding to stationary motion. In order to find the stable solution, they added to the basic motion a very special form of disturbance. Conclusions cannot be obtained by this type of method without being disputed.

Starting with an initial motion which was essentially a harmonic wave, Palm showed mathematically that for large values of time hexagons are formed. A mathematical treatment of this cellular motion was extended by Jeffreys (15) and Low (16) in 1928, by Pellow and Southwell (25) in 1940, and by Sutton (30) more recently.

If the heat flow is directed upward through horizontal layers, an unstable stratification is formed, inasmuch as the warmer fluid of lower

density is located below the cooler fluid whose density is higher. Rayleigh recognized that this unstable stratification must break down at a critical value of some parameter which he determined to be the product of Grashof number (Gr) and Prandtl number (Pr). Above this critical value, convective motion is generated.

Further investigations led to the fact that a layer of fluid, when heated from below, should remain at rest until the product of Gr and Pr reaches a certain critical value. When a layer is bounded by two parallel plates, this critical value is about 1706. When this value is exceeded, cellular motion and convection take place. Optical investigations of water by Schmidt and Saunders (28) confirmed the critical value very satisfactorily. When the fluid is at rest, heat is transferred by conduction only.

Probably one of the most elaborate experiments to study heat transfer through horizontal and vertical enclosed air layers was conducted by Mull and Reiher (21) in 1930. However, their thickness to width ratios were small, therefore, the effect of walls was insignificant. They represented their experimental results by plotting k_c/k versus log Gr and obtained a smooth curve in the experimental range of $(Gr)_L$ between 2100 and 8,890,000, where k_c is an equivalent thermal conductivity including the effect of convection.

Additional work was done to bring about a more suitable representation of the above experiments by Held (12) in 1931 and by Jakob (13) in 1946.

Jakob, representing these results in bilogarithmic coordinates, has shown that there exists at least one bend in the curve, in analogy to other cases of free convection.

Held derived the following equations from the measurement of Mull and Reiher for horizontal air layers heated from below, where Nu is the

Nusselt number:

$$Nu = 0.0463\, Gr^{.36} \qquad \text{when } 2.5 \times 10^5 < Gr < 10^7$$

$$Nu = 0.0661\, Gr^{.36} \qquad \text{when } 2.5 \times 10^3 < Gr < 6 \times 10^4$$

$$Nu = 1 \qquad \text{when } Gr < 10^3$$

Jakob derived the following equations from the Mull and Reiher measurements:

$$Nu = 0.068\, Gr^{1/3} \qquad \text{when } Gr > 5 \times 10^5$$

$$Nu = 0.195\, Gr^{1/4} \qquad \text{when } Gr < 5 \times 10^5$$

$$Nu \rightarrow 1 \qquad \text{when } Gr \rightarrow 0$$

The disagreement between the Held equations and the Jakob equation was caused by the fact the Jakob supposed that, for laminar and turbulent flow in air layers, the same exponents of the Gr might be applied as for natural heat convection at single vertical plates.

DeGraaf and Held (9) conducted an investigation of the heat transfer and the convection phenomena in enclosed air layers in horizontal, vertical, and oblique positions. Again, however, the thickness to width ratios were small. Therefore, the walls had little effect on the results. It was concluded that heat transfer through enclosed plane air layers, when the motion of the air is turbulent, depends only on the inclination. Equations have been derived for various angles of inclination and ranges of Gr. Their experiments also confirmed the observations of Chandra (5), and Benard and Avsec (2).

A comparison of the apparatus and technique used by DeGraaf and Van Der Held, Mull and Reiher, and in the present investigation is shown in Table I.

The convection in a fluid between horizontal conducting surfaces is an example of thermal turbulence, because the mean heat transport is

TABLE I

COMPARISON OF APPARATUS AND TECHNIQUE

	De Graaf and Van Der Held	Mull and Reiher	Present Investigation
Shape of fluid chamber	Square Parallelepiped	Rectangular Parallelepiped	Cylinder
Area of fluid chamber, m.^{2}	0.142 (approx.)	0.617	0.0038
Diam. or length/width, mm.	430/430	1010/612	69.85
Spacing, mm.	6.9-22.9	12-196	1.65-139.7
Container wall material	Glass	Wood	Plexiglas
Was it insulated?	No	Yes	Yes
Was there a guard heater?	No	Yes	No
Method of heating	Electrical	Electrical	Electrical
Method of cooling	Water	Water	Ice bath
Max. hot temp., $^{\circ}$C.	146	146	95
Max. temp., difference, $^{\circ}$C.	100	29	74

TABLE I (CONTINUED)

COMPARISON OF APPARATUS AND TECHNIQUE

	De Graaf and Van Der Held	Mull and Reiher	Present Investigation
Method of plate temp. measurement	3 thermocouples in hot plate; mean cooling water temp.	15 thermocouples in hot plate; 6 thermocouples in cold plate	1 thermocouple in each plate
Method of heat transfer measurement	Heat acquired by cooling water	Electrical power consumption	Heat flow meter

independent of position while the distance between these surfaces is the only geometric parameter. A paper by Malkus (17) describes measurements of the heat transport and mean velocity in such convection up to Rayleigh numbers (Ra) of 10^{10}. Six discrete transitions in the slope of the heat-transport curve were observed between Rayleigh numbers of 1700 and 1,000,000.

The apparent lack of uniqueness of the flow patterns associated with a given heat transport suggests a new approach to turbulent phenomena discussed in another paper by Malkus (18). This paper presents a theoretical investigation of various properties of the steady state inhomogeneous turbulent convection of heat in a fluid between horizontal conducting surfaces.

Jakob and Gupta (14) made a study with heat transfer by free convection through a layer of water or carbon tetrachloride bounded by a lower horizontal heating surface and an upper horizontal cooling surface. The results were correlated within the laminar range for (Gr · Pr) from 2×10^3 to 15×10^4 by the following expression:

$$Nu = 0.300\ (Gr \cdot Pr)^{1/4}$$

and within the turbulent range for (Gr · Pr) from 2×10^5 to 3×10^9 by the following form:

$$Nu = 0.1255\ (Gr \cdot Pr)^{1/3}$$

Transition occurred between (Gr · Pr) = 10^5 and 10^6.

No appreciable influence of the ratio of layer thickness L to the horizontal diameter D of the heating surface was observed for the turbulent range. However, there was a noticeable influence of L/D in the laminar range.

The data observed by Jakob and Gupta were found to correlate well according to the following equation:

$$Nu = 0.3615\ (Gr,m)^{1/4}\ (L/D)^{.058}$$

Heat transfer through a horizontal liquid layer bounded on top by a cold surface and on the bottom by a heated surface was measured by Schmidt and Silveston (29), and the results were correlated by dimensionless numbers. Conduction alone occurred up to a critical Rayleigh number of 1700 ± 3%, at which convection appeared.

Three distinct convection regimes exist. The first occurs as convection begins and appears as a honeycomb pattern in the layer. As Ra is increased, the pattern changes to a series of stripes and the heat transfer takes on a laminar charararacteristic of the second regime. At higher values of Ra, the pattern becomes tangled and disordered; this is the turbulent region and the third regime.

O'Toole and Silveston (23) investigated heat transfer through completely confined horizontal layers of gases and Newtonian fluids having a layer width to thickness ratio of more than two, and heated from below. Equations were derived for calculating the heat transfer for the three distinct natural convection regimes, creeping, laminar, and turbulent. The transition from the laminar to the turbulent regime, Ra around 10^5, is the only area of uncertainty.

Chandrasekhar (6) investigated the effect of rotation of solid boundaries on thermal instability. He found that the effect of the Coriolis force is to inhibit the onset of convection. Numerical results of the critical Rayleigh number are derived for one value of the ratio of the kinematic viscosity to the thermal conductivity.

Ostrach and Braun (22) found that the Fronde number (Fr) is the parameter that determines the relative effect of body rotation on natural convection flows. In flows with large Fr (motions induced solely by rotation of a heated cylinder), the action of the Coriolis force prevents appreciable convection of heat.

Although closely related investigations of natural or free convective heat transfer under various conditions have been conducted, it is obvious that little information is available on the effects of heat transfer in small, completely enclosed air spaces. Investigations have been limited to horizontal layers heated from below. The thickness of the layers investigated were small in comparison to the width or diameter, however, in the present investigation, with thickness to diameter ratios near unity, side wall effects can be expected to be a factor of importance.

EXPERIMENTAL METHOD

DESCRIPTION OF APPARATUS

The apparatus used by the author was designed to permit determination of the effect of convective heat transfer in a small enclosed air space. Figure 2 shows a schematic diagram of the equipment. Figure 3 is an overall view of the apparatus located in the Mechanical Engineering Heat Transfer Laboratory. The test section consisted of cylindrical plexiglas air chamber. Heat was supplied through a variable voltage transformer to a heating coil attached to a solid aluminum cylinder. The aluminum cylinder was used to obtain a radial temperature distribution. The heat was carried away from the air chamber to an ice bath by means of another solid aluminum cylinder. The heat source, air chamber, and part of the heat sink cylinder were covered with rock wool insulation. The direction of heat flow through the test section as shown in Fig. 6 is from top to bottom. To obtain heat flow from bottom to top the insulation container and contents were turned up-side-down. Figures 4 and 5 are views of the experimental apparatus with heat supplied from above and from below, respectively.

AIR CHAMBER or TEST SECTION

Several materials were considered from which to make the air chamber, the center section shown in Fig. 6. It was desired to have a minimum amount of heat conducted through the walls of the tube, so that the heat transfer through the air would be a measurable percentage of the total heat flow between the two ends of the test section. Therefore, a material of low thermal conductivity was needed. The material had to have sufficient strength to withstand a high vacuum. Furthermore, the material had to have good machineability.

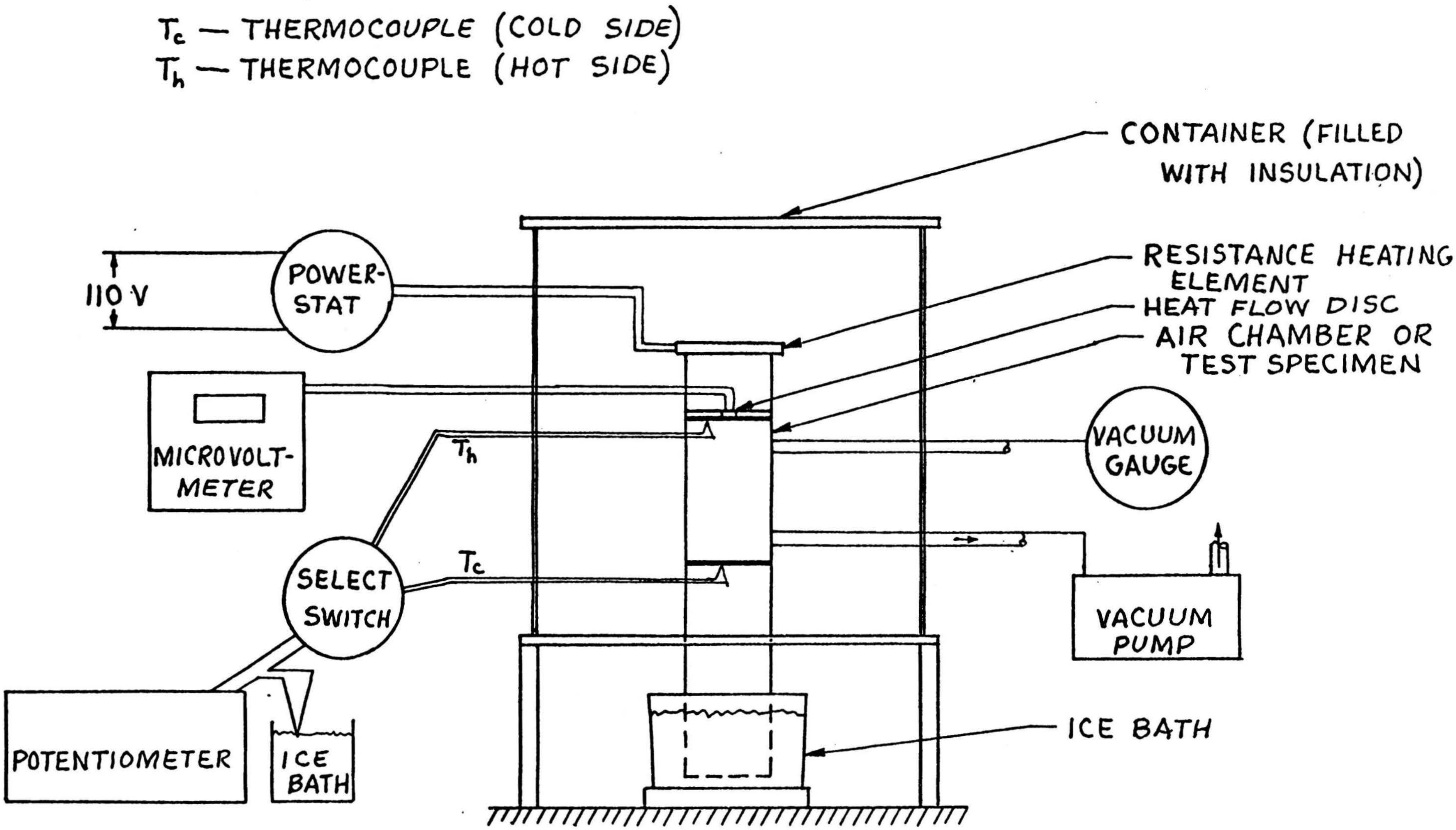

FIG. 2. SCHEMATIC DIAGRAM OF EXPERIMENTAL EQUIPMENT.
(HEAT FROM ABOVE)

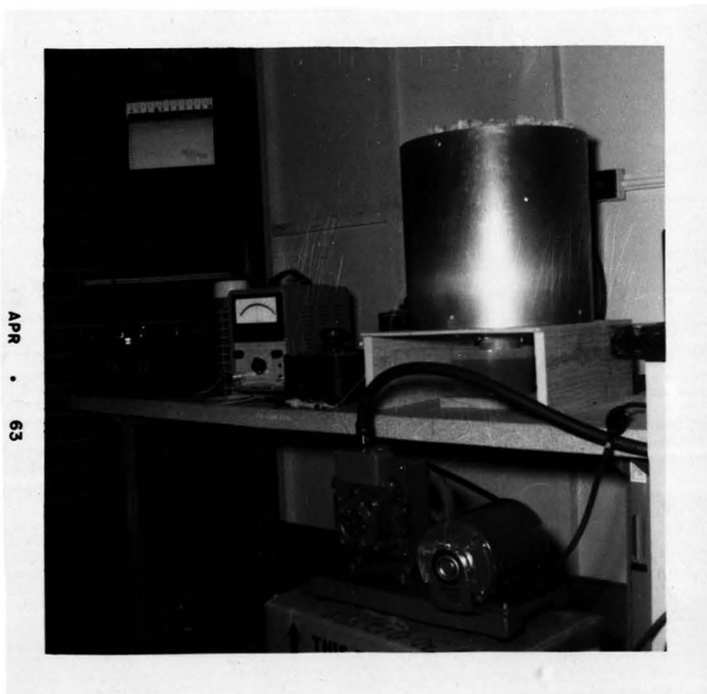

Fig. 3. Overall view of experimental apparatus showing recording and indicating potentiometers, microvoltmeter, container, and vacuum pump.

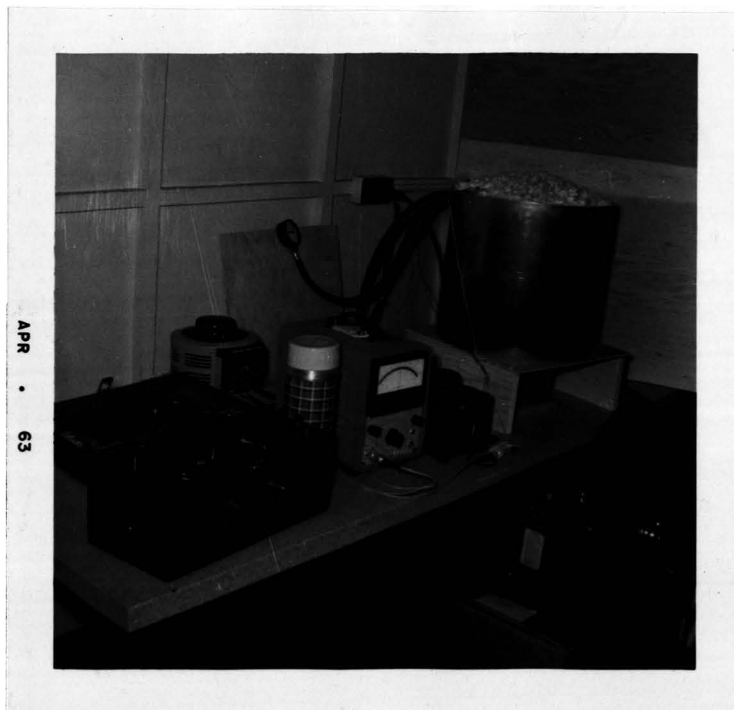

Fig. 4. View of experimental apparatus
with heat supplied from above.

Fig. 5. View of experimental apparatus with heat supplied from below.

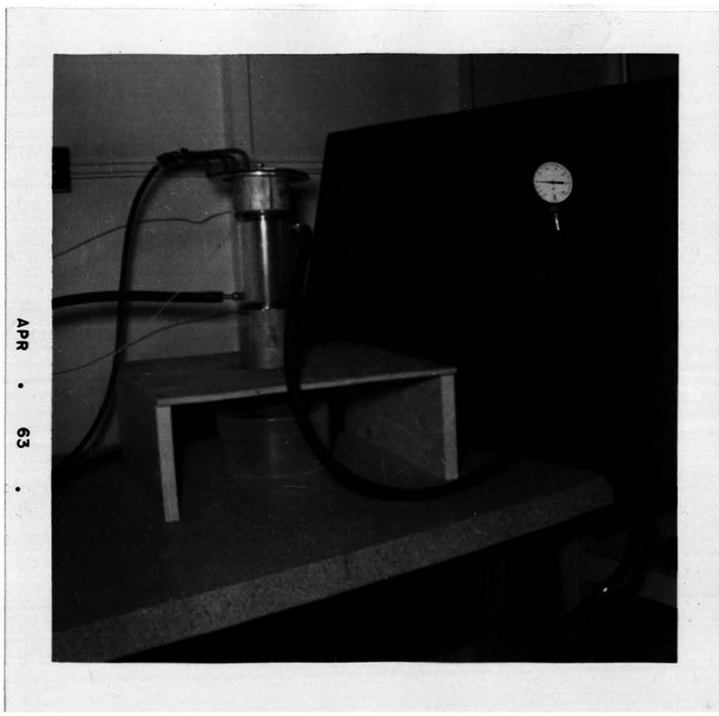

Fig. 6. Close up view of test section, heating element, and ice bath.

Taking into account the above desired characteristics, it was determined to use plexiglas. The air chamber (Fig. 7) was made of a 2.75 inch inside diameter plexiglas tube with 0.25 inch thick walls and 0.08 inch thick aluminum end plates fastened to the tube with Formica contact cement. Two holes were drilled and tapped for vacuum tube fittings. A commercial sealing compound for vacuum work was applied to the outside surface of all joints to provide greater assurance of a leak-proof chamber. Four test sections were used so that investigation could be made in different flow regimes. Test section No. 1 was two diameters or 5.50 inches long. Test section No. 2 was one diameter or 2.75 inches long. Test section No. 3 was 1.03 inches long. Test section No. 4 was 0.65 inches long.

HEAT SOURCE

The heat source, shown as the top section in Fig. 6 consisted of a small heating element attached to a 3.25 inch diameter by 2.0 inches long aluminum cylinder to provide a uniform temperature distribution. The input was controlled with a variable voltage transformer.

HEAT SINK

The heat sink was an ice bath. Connecting the ice bath and test section was a 3.25 inch diameter aluminum cylinder approximately 8.0 inches long. The flat end of the cylinder contacting the test section had a groove cut to accommodate a thermocouple for measuring the temperature of the cold side of the test section.

When the heat flow was from top to bottom, an open plastic container was used to hold the ice and water. When the heat flow was from bottom to top, a hole was cut in the bottom of a plastic container to fit over the aluminum cylinder as shown in Fig. 5. Tape was then applied to provide a

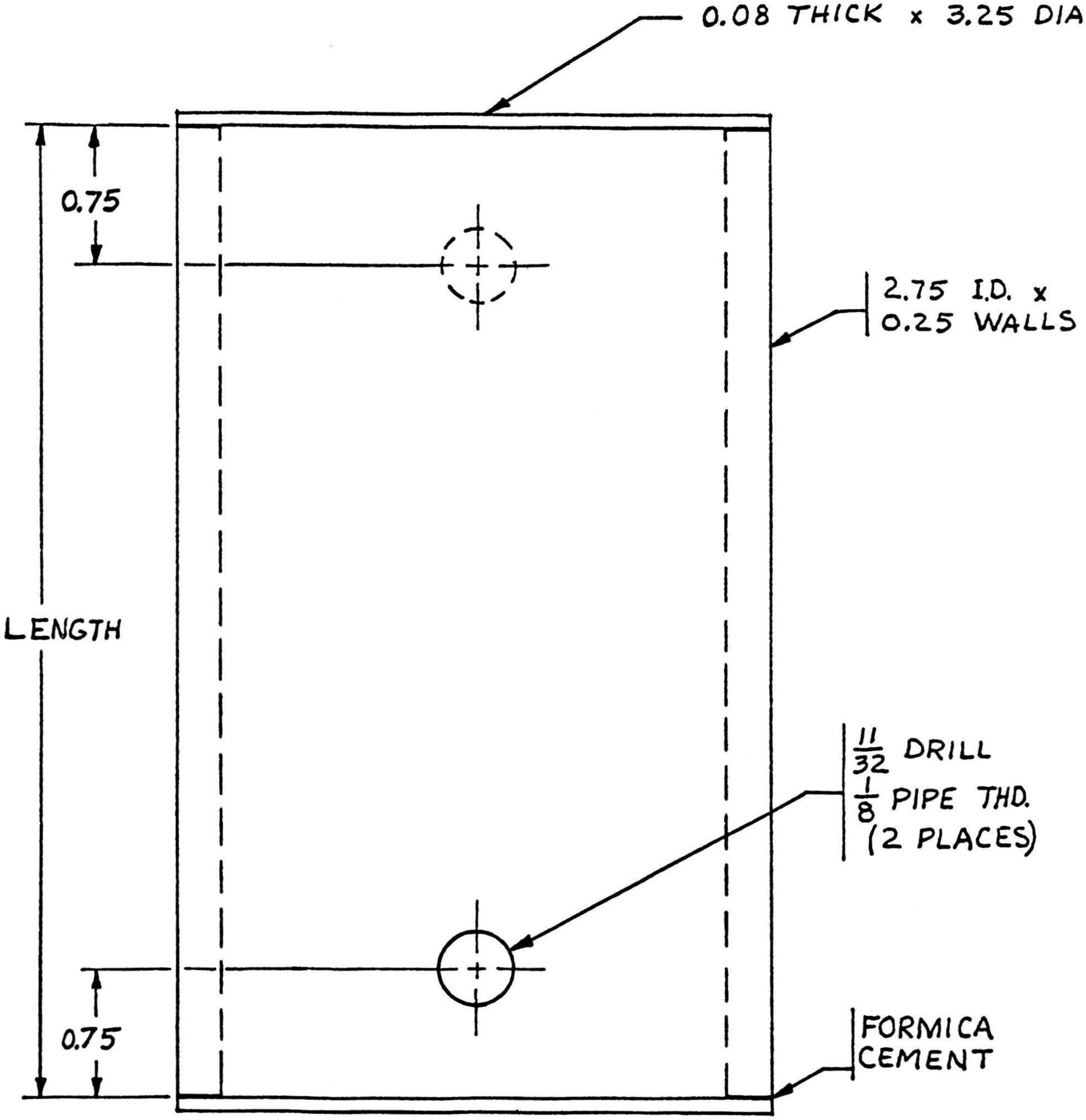

FIG. 7. AIR CHAMBER OR TEST SECTION.

water tight seal.

HEAT FLOW MEASUREMENT

The measurement of heat flow posed an extremely complex problem due to the relatively low rates of flow.

The initial consideration was to construct a heat meter from a phenolic laminate cylinder one inch thick and 3.25 inches in diameter with thermocouples located at a known distance. This material has a low thermal conductivity, therefore, a reasonably large and easily measurable temperature difference could be expected. However, preliminary calibration tests indicated considerable side losses, poor response, and poor reproducibility of results.

In an attempt to reduce losses and increase response, the above heat meter was removed and an attempt was made to use the aluminum cylinders themselves by paralleling several thermocouples at each plane and then connecting the two sets in series opposing one another. This results in the direct measurement of the average temperature difference between the two planes. Due to the high thermal conductivity of aluminum, the output was very small and unstable.

It was finally deemed necessary to obtain calibrated heat meters from the National Instrument Laboratories, Inc. The small (0.4 inch diameter) N. I. L. heat flow disc was placed between the heat source and the test section (Fig. 2). The N. I. L. heat flow transducer is a thin (0.06 inches) disc of material containing a thermocouple assembly which produces a high electrical output proportional to the flow of heat through the transducer. Since the disc is thin, the thermal resistance is low and side losses are negligible. The response is excellent. The output was measured with a precision electronic microvoltmeter. The microvoltmeter used has an

accuracy of plus or minus 3% in the range used with a maximum drift of four microvolts per day.

The heat flow meter was installed in a thin fiberglas spacer disc as shown in Fig. 8. A groove was cut in the spacer for a thermocouple for measuring the temperature on the hot side of the test section.

The output from these iron-constantan thermocouples was measured with a portable precision millivolt potentiometer. In some of the preliminary investigations which included determination of the temperature distribution in the insulation, a recording potentiometer was also used. A reference junction was used and kept at 32° F in an ice bath.

VACUUM SYSTEM

The vacuum system consisted of a portable vacuum pump connected to the air chamber with a rubber vacuum tube and of a vacuum gauge to monitor the pressure in the test chamber. The vacuum pump was capable of pumping the pressure down to 0.001 mm of Mercury.

INSULATION CONTAINER

The insulation container was constructed from 0.025 inch thick aluminum sheet in the form of a cylinder 15 inches in diameter and length. The ends of the cylinder were attached to 3/8 inch plyboard, 16 inches square, with bolts and angle clips for easy removal. To one end were fastened two 1 x 6 boards which were used as legs to support the container so that an ice bath could be placed underneath when the heat flow was from top to bottom. The plyboard to which the legs were attached had a hole to fit the heat sink cylinder. Three clips, equally spaced around the hole, and with bolts directed toward the center were used to hold the cylinder in a fixed position.

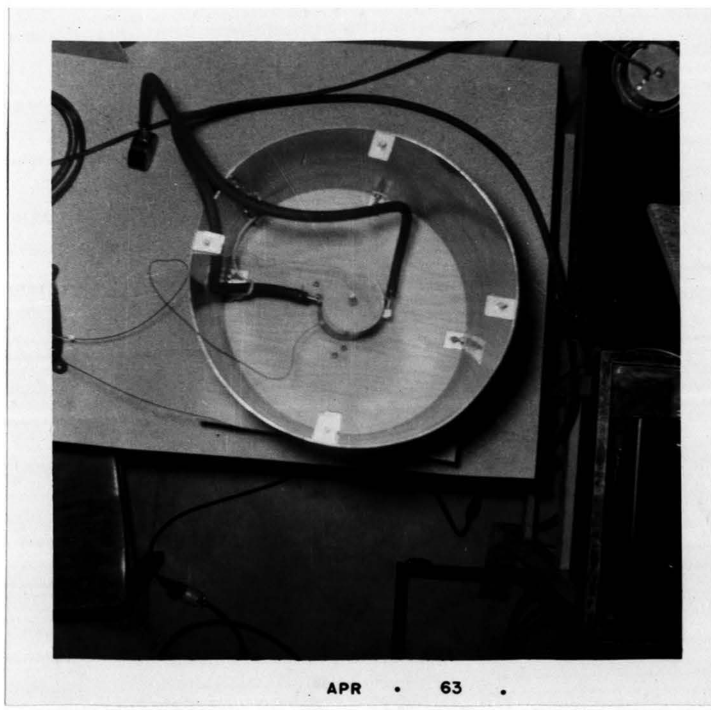

Fig. 8. Installation of heat flow disc.

TEST PROCEDURE

After the equipment was arranged and connected together, ice was put into the ice bath of the heat sink. Ice and water were put into a vacuum bottle for the reference junction of the thermocouples.

The input power to the heating element was controlled by the variable voltage transformer. The power was adjusted as determined from preliminary runs for various temperature levels to about three times steady-state power requirement until the temperature on the hot side of the air chamber reached approximately the desired temperature level. Then the power was reduced and all the components within the insulation container were allowed to reach a steady-state condition. Power was adjusted as needed to obtain the desired temperature.

As a steady-state condition was being approached, outputs of the thermocouples and the heat flow disc were read and recorded every five minutes to determine whether steady-state had been reached. It required several hours after final setting of the power level to reach a steady-state.

The ice and water were stirred periodically in the heat sink to maintain a constant low temperature.

After the steady-state measurements were recorded, the vacuum pump was started and the system again allowed to reach thermal equilibrium before readings were taken. At each condition, the following data were taken: temperature (T_c), temperature (T_h), and the microvolt output (qd) of the heat flow disc.

The same test procedure was followed with the insulation container turned up-side-down so that the heat flow was from top to bottom instead of from bottom to top.

The same procedure also was followed for the other three test chamber lengths.

TABLE II

EXPERIMENTAL DATA

TEST SECTION NO. 1: Length = 5.5 inches; Diameter = 2.75 inches.

Heat flow direction	Down						Up					
Run	1		2		3		1		2		3	
Pressure	atm	vac	atm	vac	atm	vac	atm	vac	atm	vac	atm	vac
T_h (°F)	145.5	145.5	169.6	169.6	202.6	203.0	131.0	139.0	170.0	181.0	203.0	219.0
T_c (°F)	38.0	38.0	38.0	38.0	38.0	38.0	38.0	38.0	38.0	37.3	38.0	38.0
q_d (μv)	205	181	280	247	400	358	340	180	570	306	799	420

TABLE III

EXPERIMENTAL DATA

TEST SECTION NO. 2: Length = 2.75 inches; Diameter = 2.75 inches.

Heat flow direction	Down						Up					
Run	1		2		3		1		2		3	
Pressure	atm	vac	atm	vac	atm	vac	atm	vac	atm	vac	atm	vac
T_h (°F)	139.0	140.0	169.6	169.6	197.5	198.0	141.0	145.3	170.0	178.0	202.5	211.0
T_c (°F)	38.0	38.0	38.0	38.0	38.0	38.0	38.0	38.0	38.0	38.0	38.0	38.0
q_d (μv)	268	208	385	299	520	400	445	230	700	360	1160	685

TABLE IV

EXPERIMENTAL DATA

TEST SECTION NO. 3: Length = 1.03 inches; Diameter = 2.75 inches.

Heat flow direction	Down						Up					
Run	1		2		3		1		2		3	
Pressure	atm	vac	atm	vac	atm	vac	atm	vac	atm	vac	atm	vac
T_h (°F)	143.0	144.0	172.0	173.0	200.0	201.0	135.0	141.0	170.0	175.5	200.0	209.0
T_c (°F)	38.0	38.0	38.0	38.0	38.0	39.0	38.0	38.0	38.0	38.0	38.0	38.0
q_d (μv)	480	395	680	547	865	698	680	405	1000	565	1290	770

TABLE V

EXPERIMENTAL DATA

TEST SECTION NO. 4: Length = 0.65 inches; Diameter = 2.75 inches.

Heat flow Direction	Down						Up					
Run	1		2		3		1		2		3	
Pressure	atm	vac	atm	vac	atm	vac	atm	vac	atm	vac	atm	vac
T_h (°F)	138.0	139.0	169.6	170.6	201.0	202.0	134.3	140.0	171.0	177.0	201.0	210.0
T_c (°F)	38.3	39.0	38.0	38.0	38.5	38.6	38.0	38.0	38.0	38.0	38.6	38.6
q_d (μv)	660	505	962	738	1290	980	842	565	1240	840	1560	1080

METHOD OF ANALYSIS

To determine the effects of natural convection in small enclosed air spaces, the heat transferred by the air was determined as the difference in heat flow with air in the air chamber at atmospheric pressure and with the air essentially removed by pulling a high vacuum.

With the air chamber evacuated, the heat flow due to conduction and convection through the air was virtually eliminated. The change of heat flow due to radiation was relatively small since the temperatures encountered were low and did not vary appreciably during a given test.

The temperature drop across the air chamber is $T_h - T_c$.

The mean temperature is given by

$$T_m = \frac{T_h + T_c}{2}.$$

The rate of heat flow was determined by the measured microvolt output of a N. I. L. heat flow disc. For the heat flow disc used, the calibrated output was 2.16 microvolt per Btu/hr ft^2.

Since the temperature difference changed slightly when the vacuum was pulled, the heat flow during vacuum conditions was corrected to obtain comparative results as follows:

$$q_{d_{vac}} = \left[1 - \frac{(T_h - T_c)_{vac} - (T_h - T_c)_{atm}}{(T_h - T_c)_{vac}} \right] (q_d) \text{ vac measured.}$$

The change in heat flow measured by the heat flow disc in microvolts is

$$q_d = (q_d)_{atm} - (q_d) \text{ vac}$$

or the rate of heat flow due to the air in Btu/hr per sq. ft. is then given by

$$q_a/A = q_d/2.16.$$

An overall coefficient of transmission (U) was determined by

$$U = \frac{q_a}{A\ (T_h - T_c)\ atm}.$$

The values of U were plotted against the mean temperature and the length.

To make a comparison of the results obtained in this investigation, the values of U for the laminar range, the coefficient of conductance (h) in horizontal air spaces as determined by Held and k/L at a mean temperature of 100 degrees F. were plotted versus the distance across the air space. For the turbulent range, the Nusselt number Nu = UL/k was calculated and compared with that obtained by Held.

Since the present system has not been previously investigated, it is not known where the transition is going to occur. Therefore, the Gr was evaluated for each run to compare previous investigations. The Grashof number is given as

$$Gr = g\beta \Delta t L^3 \rho^2/\mu^2.$$

However, this may be rewritten as

$$Gr = aL^3\ \Delta t/Pr$$

where $a = g\beta\rho^2 c/\mu k$ and $Pr = C\mu/k$. Both of these values were found in Table A-2 of Brown and Marco (4) for various temperatures.

After the Gr had been determined, the Nusselt number (Nu) was calculated from the appropriate Held relation as given below:

$$Nu = 0.0463\ Gr^{.36} \qquad \text{when } 2.5 \times 10^5 < Gr < 10^7$$

$$Nu = 0.0661\ Gr^{.36} \qquad \text{when } 2.5 \times 10^3 < Gr < 6 \times 10^4$$

Then the conductance was determined from

$$Nu = hL/k$$

as

$$h = k\ Nu/L.$$

In the turbulent range the Nu was plotted against the Gr as shown in Fig. 10.

TABLE VI

TABULATED RESULTS

TEST SECTION NO. 1: Length = 5.5 inches; Diameter = 2.75 inches.

Heat flow direction	Down			Up		
Run	1	2	3	1	2	3
$T_h - T_c$ (°F)	107.5	131.6	164.6	93.0	132.0	165.0
T_m (°F)	91.8	103.8	120.3	84.5	104.0	120.5
q_a/A (Btu/hr ft^2)	11.11	15.28	18.98	80.73	133.8	192.2
U (Btu/hr ft^2 °F)	.1033	.1160	.1152	.8680	1.013	1.165
UL/k	3.040	3.340	3.267	25.60	29.20	33.00
Gr x 10^{-7}	1.907	2.138	2.494	1.736	2.140	2.497

TABLE VII

TABULATED RESULTS

TEST SECTION NO. 2: Length = 2.75 inches; Diameter = 2.75 inches.

Heat flow direction	Down			Up		
Run	1	2	3	1	2	3
$T_h - T_c$ (°F)	101.0	131.6	159.5	103.0	132.0	164.5
T_m (°F)	88.5	103.8	117.8	89.5	104.0	120.3
qa/A (Btu/hr ft^2)	28.70	39.80	56.00	103.8	167.0	235.6
U (Btu/hr ft^2 °F)	.2841	.3023	.3509	1.008	1.265	1.432
UL/k	4.200	4.360	4.974	14.87	18.25	20.27
Gr x 10^{-6}	2.267	2.660	3.047	2.292	2.668	3.100

TABLE VIII

TABULATED RESULTS

TEST SECTION NO. 3: Length = 1.03 inches; Diameter = 2.75 inches.

Heat flow direction	Down			Up		
Run	1	2	3	1	2	3
$T_h - T_c$ (°F)	105.0	134.0	162.0	97.0	132.0	162.0
T_m (°F)	90.5	105.0	119.0	86.5	104.0	119.0
qa/A (Btu/hr ft^2)	41.10	63.41	77.30	138.3	212.0	259.5
U (Btu/hr ft^2 °F)	.3912	.4725	.4770	1.428	1.606	1.603
UL/k	2.160	2.550	2.530	7.890	8.690	8.510
Gr x 10^{-5}	1.238	1.426	1.618	1.170	1.407	1.617

TABLE IX

TABULATED RESULTS

TEST SECTION NO. 4: Length = 0.65 inches; Diameter = 2.75 inches.

Heat flow direction	Down			Up		
Run	1	2	3	1	2	3
$T_h - T_c$ (°F)	99.7	131.6	162.5	96.3	133.0	162.3
T_m (°F)	88.2	103.8	119.8	86.2	104.5	119.8
qa/A (Btu/hr ft^2)	72.48	106.2	145.8	142.9	202.0	248.5
U (Btu/hr ft^2 °F)	.7270	.8075	.8972	1.487	1.519	1.532
UL/k	2.535	2.755	3.000	5.210	5.175	5.127
Gr x 10^{-4}	2.988	3.520	4.055	2.932	3.557	4.055

DISCUSSION OF RESULTS

Results of this investigation indicate that in completely confined air layers, more heat is transferred through the air than for horizontal layers. Horizontal layers throughout this discussion will refer to small thickness to width ratios in comparison to the present investigation. Also, more heat was transferred by the air when heated from above than by pure conduction.

In Fig. 9, the values of U for the laminar range, the coefficient of conductance (h) in horizontal air spaces as determined by Held and k/L at a mean temperature of 100 degrees F. were plotted versus the distance across the air space. It can be seen clearly that more heat was transferred than that due to conduction alone. Also, there was more heat transferred than can be accounted for by Held conductance coefficients of heat transfer in horizontal layers heated from below. Therefore, the heat transfer by convection was more significant in this investigation than in horizontal layers.

The heat transferred with the heat supplied from below as compared to that with the heat supplied from above was from two to ten times greater depending on the length of the test section. The greater differences were at the longer lengths.

Larger values of U were obtained for shorter lengths which was consistent with previous results for heat transfer. U also increased with the mean temperature as shown in Figures 12, 13, 14, and 15. However, more points were needed to make quantitative correlations.

The results of test section No. 3 were particularly difficult to correlate with other investigations since they were in the transition range. More data is needed to determine the effects properly.

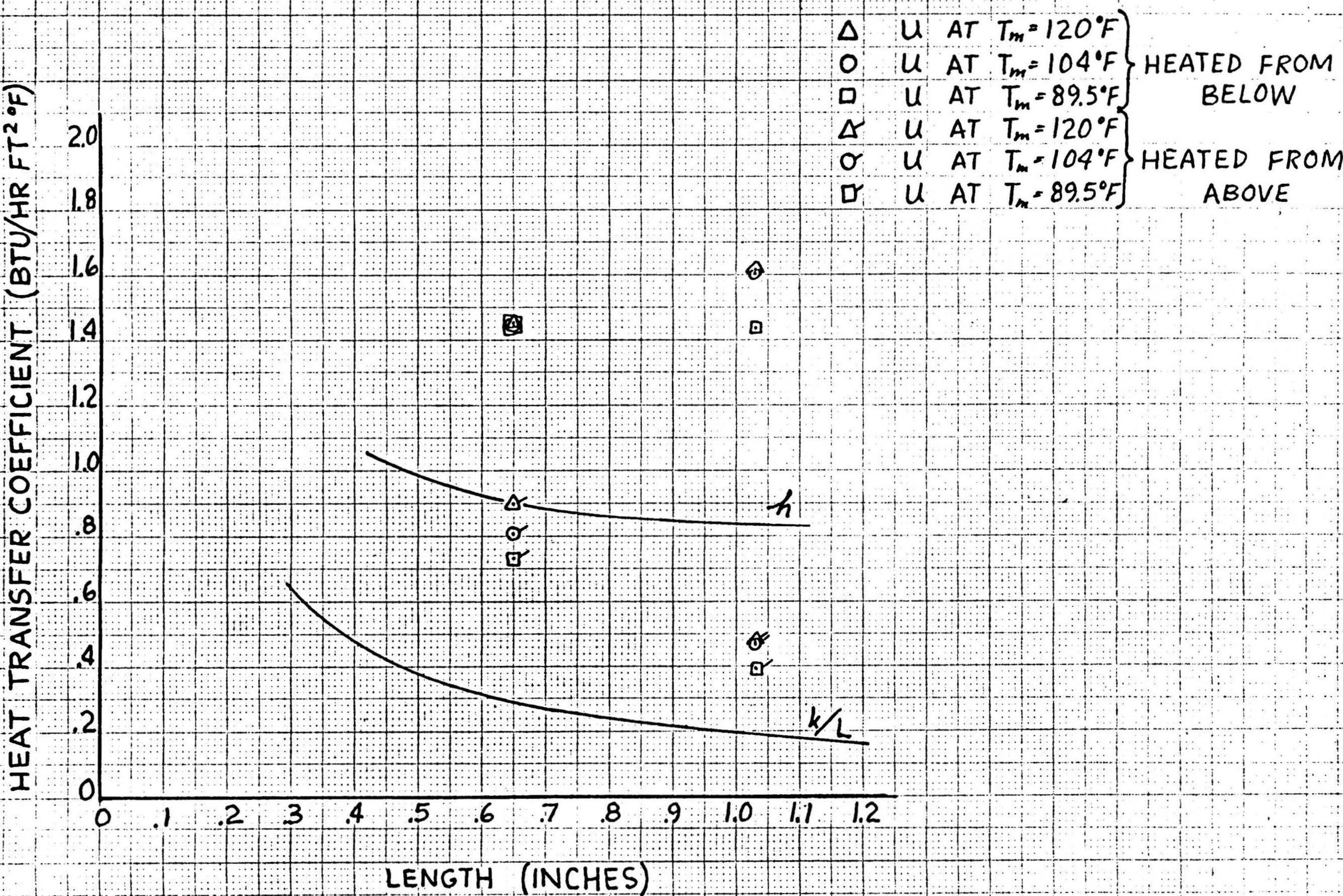

FIG. 9. COMPARISON OF EXPERIMENTAL VALUES OF U WITH PURE CONDUCTION AND HELD CONDUCTANCE. (LAMINAR RANGE)

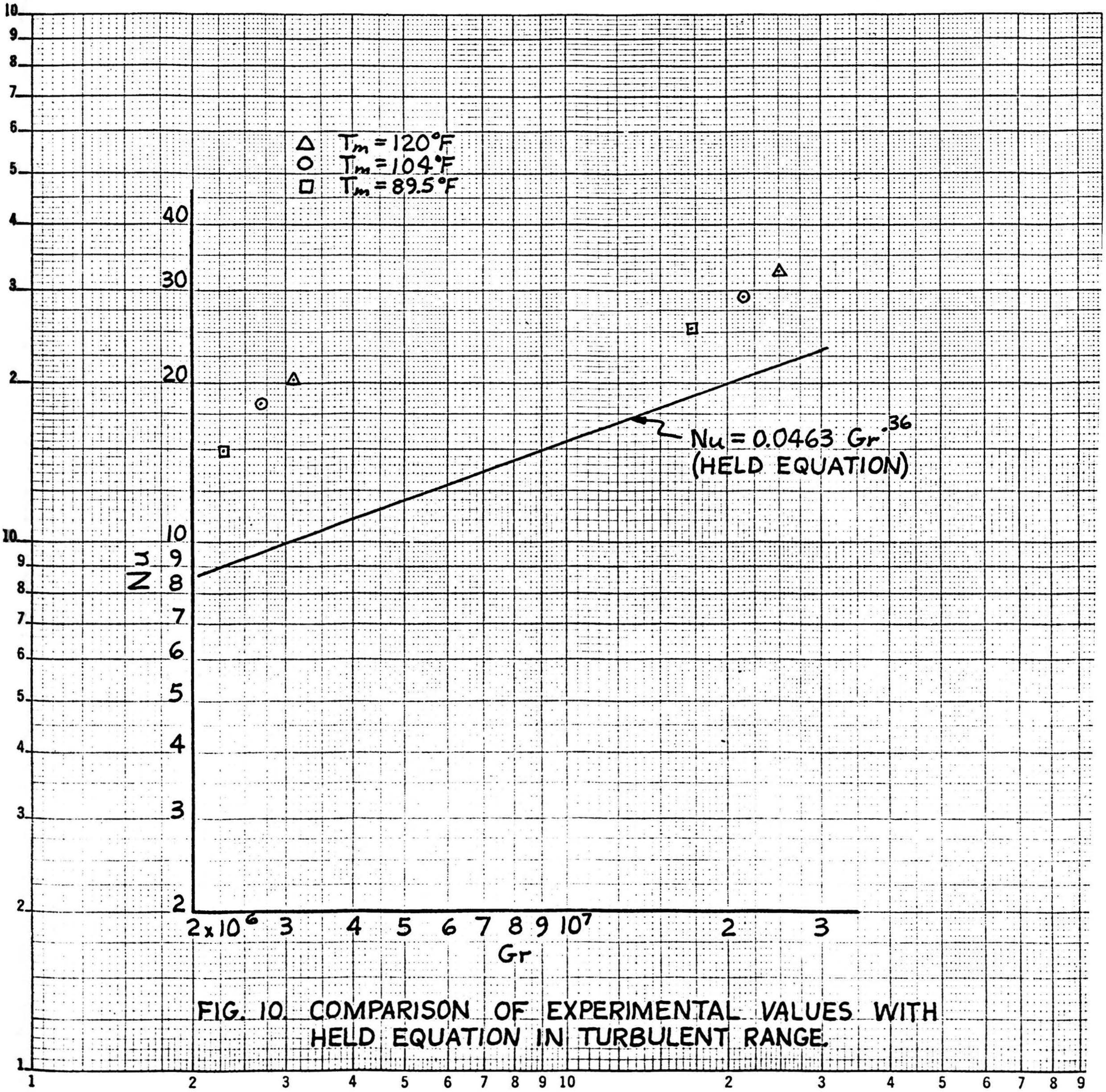

FIG. 10. COMPARISON OF EXPERIMENTAL VALUES WITH HELD EQUATION IN TURBULENT RANGE.

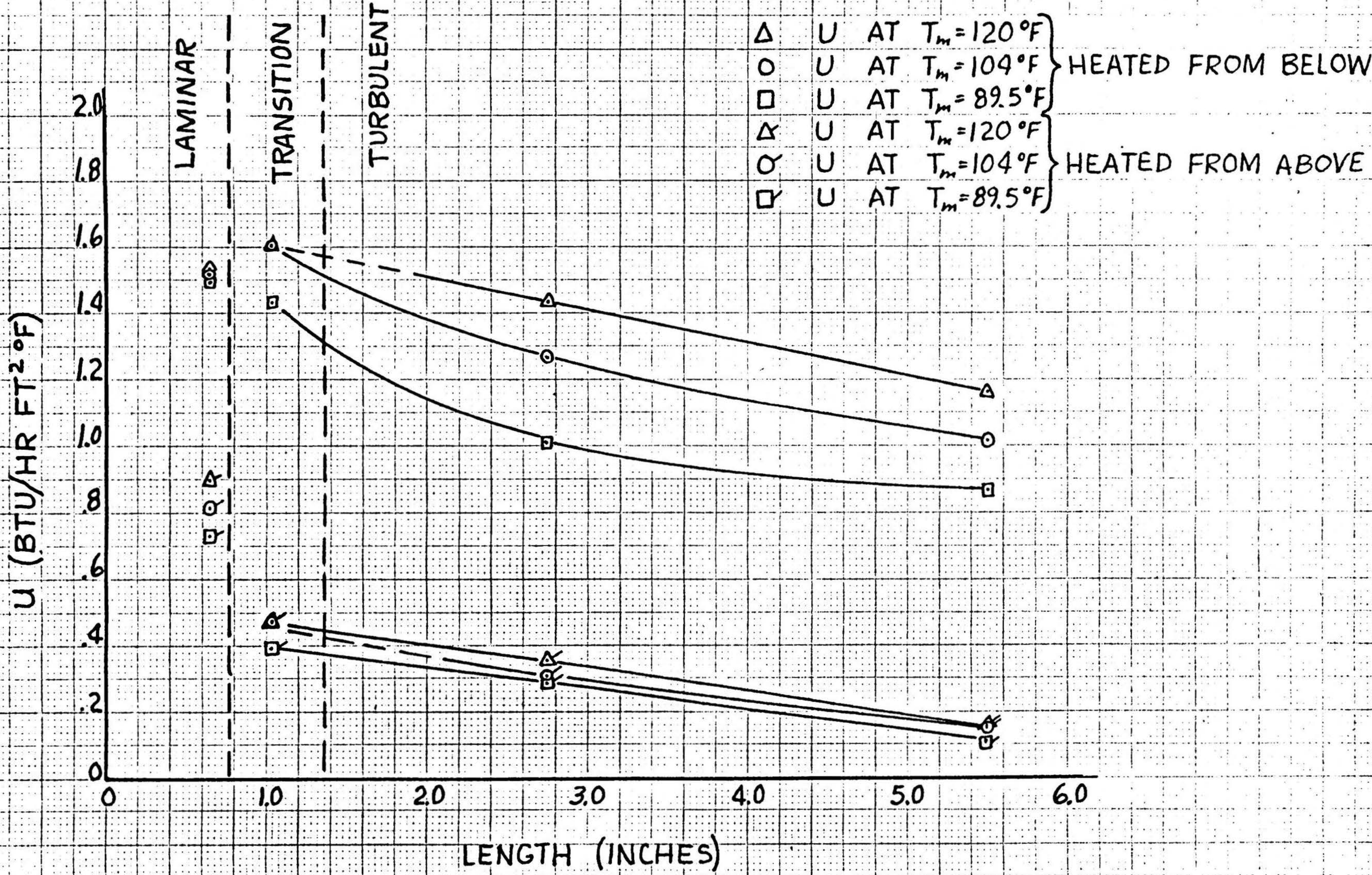

FIG. 11. EXPERIMENTAL VALUES OF U VS. LENGTH.

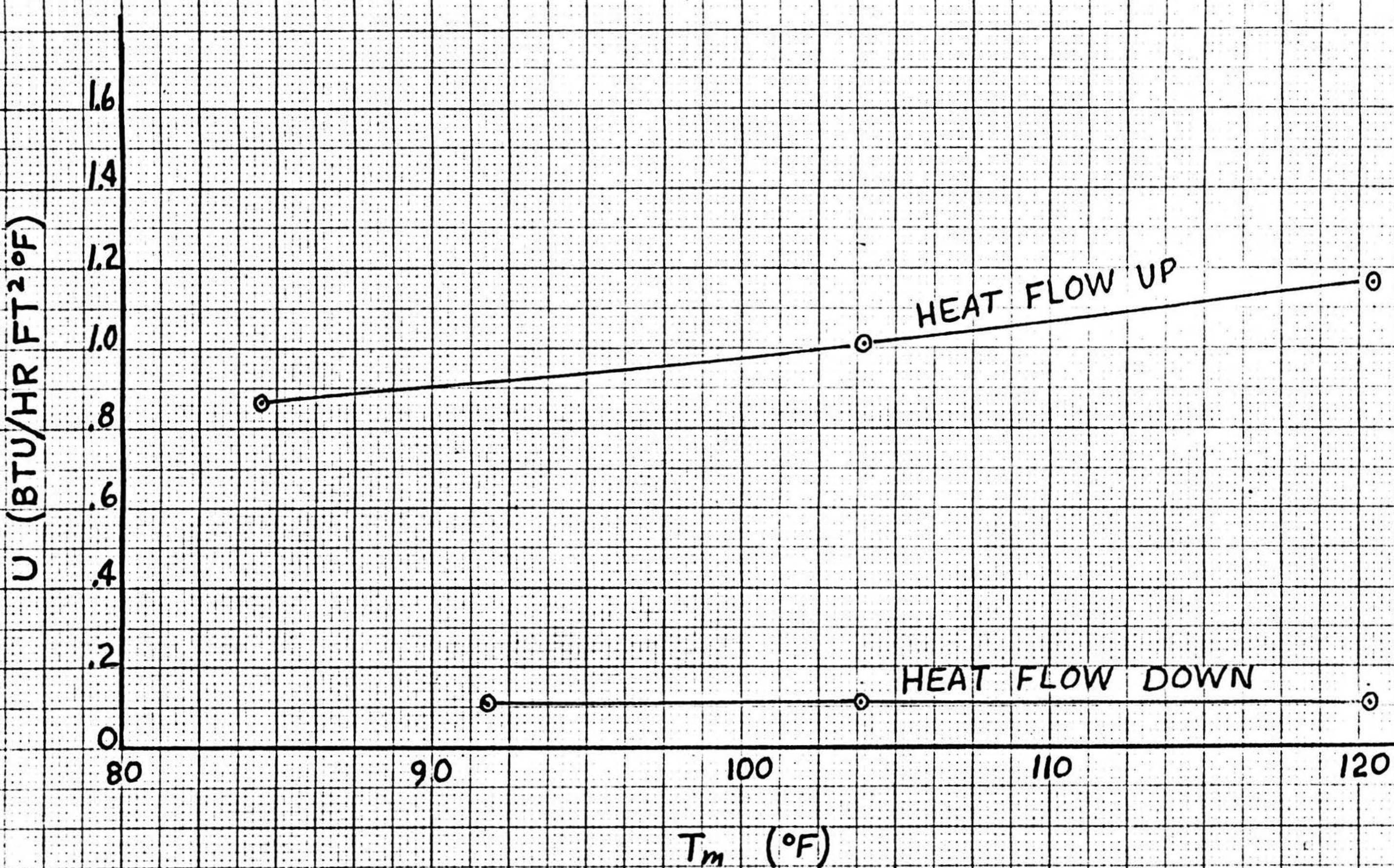

FIG. 12. OVERALL HEAT TRANSFER COEFFICIENT VS. MEAN TEMPERATURE FOR TEST SECTION NO. 1.

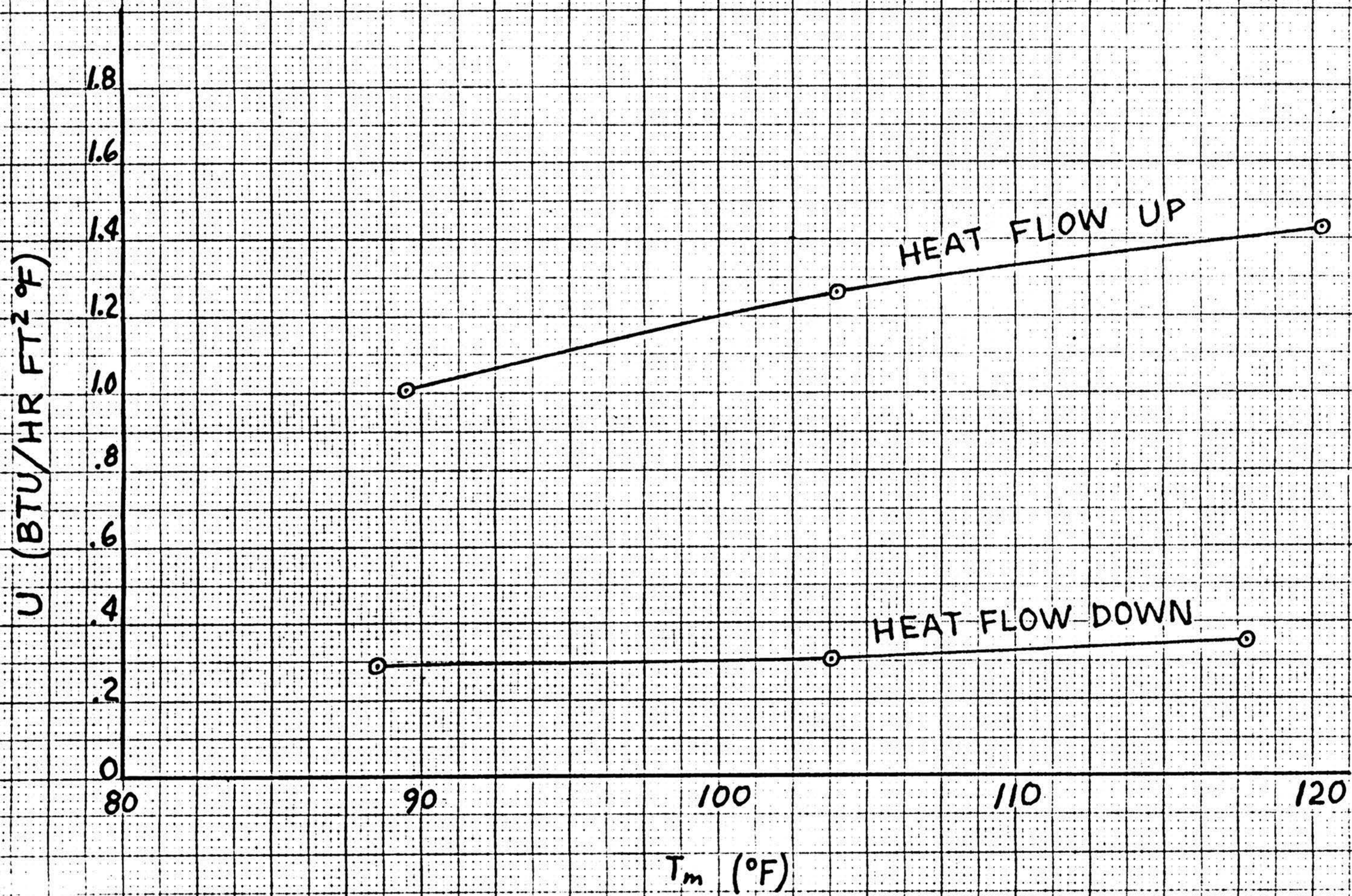

FIG. 13. OVERALL HEAT TRANSFER COEFFICIENT VS. MEAN TEMPERATURE FOR TEST SECTION NO. 2.

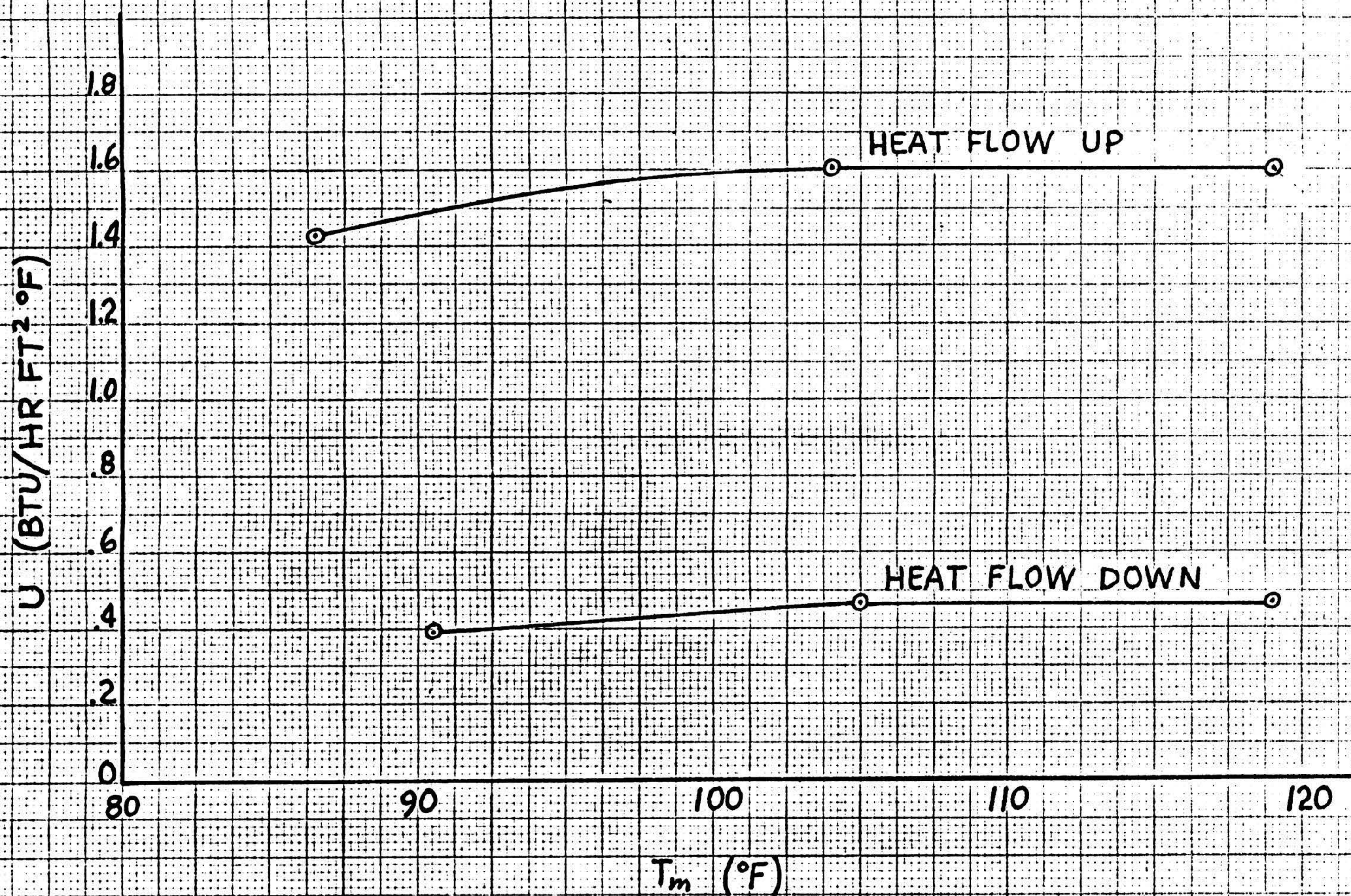

FIG. 14. OVERALL HEAT TRANSFER COEFFICIENT VS. MEAN TEMPERATURE FOR TEST SECTION NO. 3.

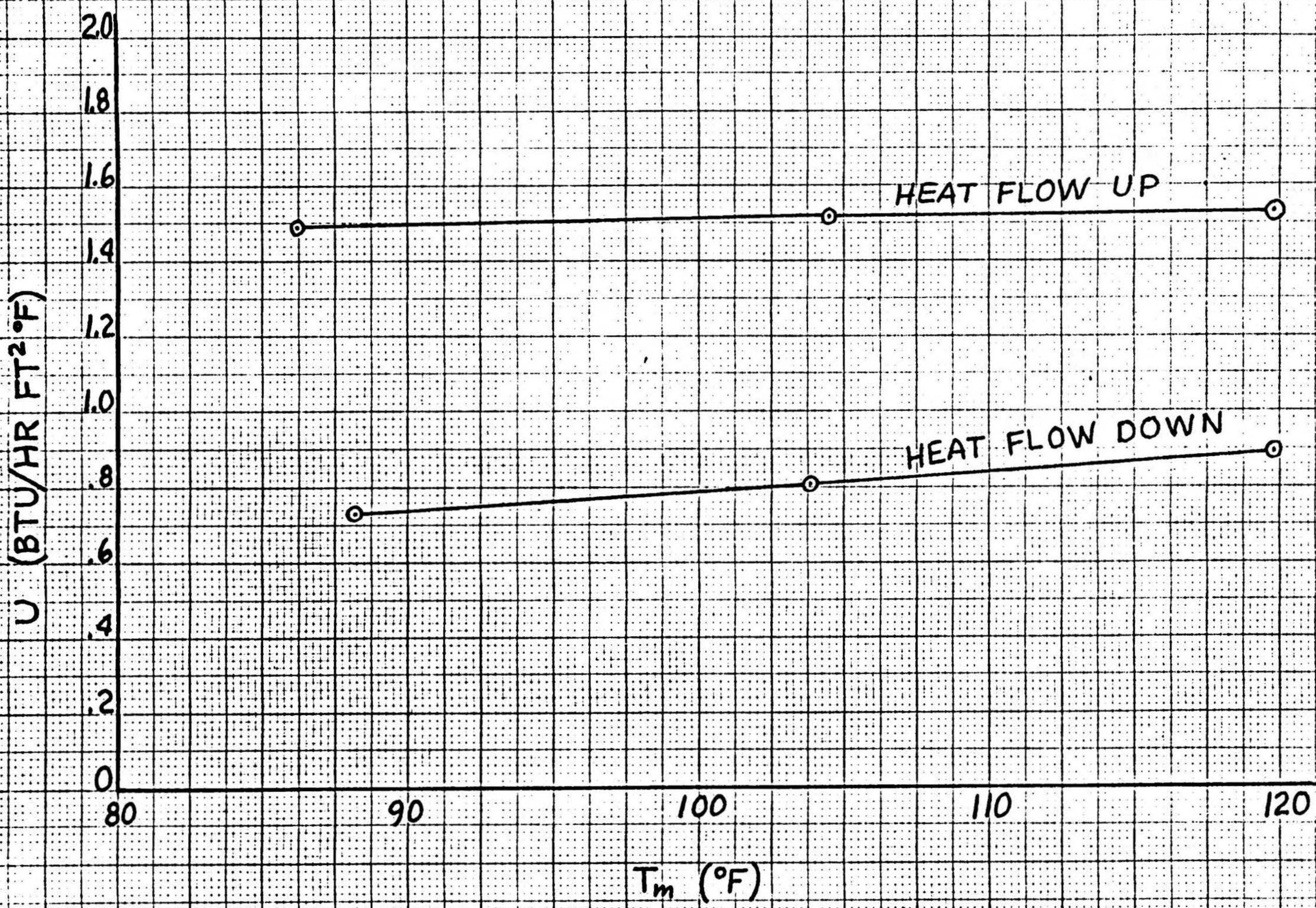

FIG. 15. OVERALL HEAT TRANSFER COEFFICIENT VS. MEAN TEMPERATURE FOR TEST SECTION NO. 4.

With the heat supplied from below in the turbulent range the value of UL/k is higher and at a smaller slope than that given by the Held equation for horizontal layers as is shown in Fig. 10.

The measurement of heat flow posed an extremely complex problem due to the low rates of heat flow. The high resistance of the air space to heat flow caused considerable side losses. However, since the measurement of heat flow through the air was determined as the difference of heat flow with and without air in the air chamber, the effect of side losses was not considered to be a major factor.

The greatest source of error was probably due to the contact pressure of the heat flow disc. This could possibly be corrected by milling a recess in the aluminum heat distribution cylinder to accommodate the heat flow disc and using aluminum foil at the interfaces.

Due to the limitations imposed by the apparatus and the measuring devices only qualitative information can be obtained. However, there was a significant amount of convective heat transfer above that predicted for horizontal layers when heated from below.

CONCLUSIONS AND RECOMMENDATIONS

The heat flow by convection was investigated with the heat supplied from the top and from the bottom. Although definite conclusions cannot be made as to the amount of heat transferred by convection in small enclosed air spaces from this investigation, there definitely was more heat transferred than in horizontal layers where the thickness to width ratios were considerably smaller.

The following general conclusions can be drawn from the test results:

1. With the heat supplied from above more heat was transferred than could be accounted for by conduction alone.
2. With the heat supplied from below more heat was transferred than could be accounted for by convection as determined by the Held equations for convective heat transfer in horizontal air layers heated from below.
3. It was found that the convective heat transfer with the heat supplied from the bottom was several times greater than it was when supplied from the top.
4. With some modification the apparatus can be used to make quantitative studies of convective heat transfer in small enclosed spaces.

Due to the limitations imposed by the apparatus and measuring devices plus the fact that this was the first investigation on small enclosed air spaces, the results cannot be expected to check with investigations where the thickness to width ratios were much smaller.

Modification of some of the apparatus is recommended to improve the results. A guard heater should be used to reduce side losses. The side losses could be determined with the use of two heat flow meters, one on either end of the test section.

The surfaces with which the heat flow disc is in contact must be maintained at a fixed distance. Care must be exercised so that the heat does not flow around the heat flow disc or that a greater amount does not go through than should.

It is further recommended that more data be obtained.

Although this is a qualitative investigation, it is believed that the knowledge can be applied to future investigations conducted to determine quantitatively the effect of convective heat transfer in small enclosed spaces. Then the results can be applied to the solution of heat transfer and thermal stress problems in such items as porous materials, insulation, and honeycomb structures used in modern aircraft, space vehicles, and many other structures.

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VITA

Joseph Charles Kroutil was born near Pleasant Hope in Polk County, Missouri, on January 10, 1935, to Joe W. and Mary A. Kroutil. He received his elementary and high school education from public schools in Polk County, Missouri. After graduating from Marion C. Early High School in May 1953, he spent one year farming and then decided to further his education. While still farming part time, he received two years of pre-engineering from Southwest Baptist College in Bolivar, Missouri. He then transferred to the Missouri School of Mines and Metallurgy and received a Bachelor of Science Degree in Mechanical Engineering on August 2, 1958.

On August 29, 1957, the author married Evelyn M. Francka of Bolivar, Missouri.

After being employed with The Boeing Company in Wichita, Kansas, four years, the author returned to the Missouri School of Mines and Metallurgy in September, 1962. From then until the present, he has been working toward the degree of Master of Science in Mechanical Engineering.