DETERMINATION OF HEAT TRANSFER COEFFICIENT IN A GAS FLOWING RADIALLY BETWEEN PARALLEL PLATES

A THESIS

Presented to

the Faculty of the Graduate Division

Georgia Institute of Technology

In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Chemical Engineering

By

Harris Burns, Jr.

December 1957

52
12R

DETERMINATION OF HEAT TRANSFER COEFFICIENT IN A

GAS FLOWING RADIALLY BETWEEN PARALLEL PLATES

Approved: ____________________

____

____

____

Date Approved by Chairman: May 27, 1957

ACKNOWLEDGEMENTS

The author wishes to acknowledge his indebtedness to Dr. J. M. DallaValle for suggesting this problem and for his patient guidance and assistance in performing the investigation. The author further wishes to express his appreciation to the Atlanta Gas Light Company for the use of a calibrated gas meter in the experimental work; and finally to Miss Louise Hitch for typing the thesis and to Miss Eve Rogers for her work on the illustrations.

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

LIST OF FIGURES

ABSTRACT

The problem of heat transfer to a fluid flowing radially between heated, parallel plane surfaces is one which has been practically ignored, although the problem itself seems simple and obvious. The present investigation was undertaken in an effort to determine the variation of heat transfer coefficients with the velocity of fluid and the distance between the surfaces in such a system.

Air was chosen as the most convenient fluid to use. A heat exchanger consisting of two steam heated plates, one of which had a central inlet tube, was constructed. Compressed air was introduced through the inlet tube and allowed to flow out to the atmosphere. The temperature distribution on the plates was determined by means of thermocouples imbedded in the plates. The air temperature was determined by means of a thermocouple probe.

Plate separations were effected by means of separators cut from brass rod. Separations of 0.10, 0.20, 0.30, 0.40, and 0.50 inches were used. Air rates, measured by means of a calibrated gas meter supplied by the Atlanta Gas Light Company, were varied between about one and six cubic feet per minute.

Overall film coefficients were determined on the basis of a logarithmic mean temperature difference. Parameters for an empirical relationship between the film coefficient and the two independent variables, the air rate and the separation, was obtained from dimensional analysis. It was found that this relation was not followed exactly, but that the

effect of separation was less than predicted. An explanation was proposed assuming that the path followed by the air does not entirely fill the air space, but leaves a substantial region of still air next to the bottom plate.

Certain qualitative observations were made regarding the flow characteristics of the system. Further investigations of the system, including detailed mathematical analysis of the system and extensive experimental work on the flow problem were recommended.

CHAPTER I

INTRODUCTION

General:--An experimental investigation was made to determine the characteristics of a heat exchanger consisting of two parallel disks, one of which has a circular inlet tube at the center. Air or other fluid is introduced through the inlet tube and allowed to flow radially through the space between the disks. An attempt was made to determine coefficients for the transfer of heat in such a system from the heated surfaces to the cold fluid.

Certain characteristics of the system are immediately evident. Neglecting the effects of pressure and thermal expansion, the velocity of the fluid will vary inversely as the distance from the center of the disk. This large change in velocity means that a transition from turbulent to viscous flow is to be expected within the system. The velocity of the fluid will be smallest where the temperature difference between the fluid and plates is smallest; this suggests that a lower exit temperature difference might be practical with an exchanger of this type than with a shell-and-tube exchanger. (An interesting variation in this system would be the case in which the fluid flows toward the center; then the velocity would be smallest where the temperature difference is greatest.)

Literature Survey:--Although this problem seems a simple one, very little information on such a system is available in the literature. Schenk and co-workers (1, 2, and 3) have investigated the problem of a fluid flowing from a hot region to a cold region (Graetz' Problem) for a viscous liquid flowing between infinite parallel plates. Wagner (4) derived an expression for the heat transfer coefficient between a heated, rotating disk and the ambient air. Goldstein (5) presents a number of problems involving heat transfer between a moving fluid and a plane surface. However, none of these involve radial flow.

The flow problem itself has been investigated for some cases. Benedikt (6) derives an equation for the pressure at any point in a system in which a perfect, viscous liquid flows radially between parallel planes. Comolet (7, 8) derives the equation

$$w = \frac{d^3}{6\mu} \cdot \frac{P_1}{RT} (P_0 - P_1) \tag{1}$$

where

w = flow rate (lb./hr.),

d = distance between plates (ft.),

P_0 = pressure at inlet (atm.),

P_1 = pressure at outlet (atm.),

μ = viscosity (lb./ft.hr.),

T = absolute temperature (°R.),

and R = gas constant (cu.ft.atm./°R.),

for a compressible, viscous fluid flowing radially between parallel plates.

Unfortunately, no information on pressure drops or flow rate in turbulent or transitional flow was found, nor was any means of predicting the transition from turbulent to viscous flow available.

Theoretical Considerations:--The fundamental equation in heat transmission by convection is (9)

$$\frac{dq}{U\Delta T} = dA \tag{2}$$

where

q = rate of heat transfer (Btu./hr.),

U = heat transfer coefficient (Btu./hr.ft.^{2}°F.),

ΔT = temperature difference between fluid and surface (°F.),

and A = area of heat transfer surface (ft.^{2}).

If ΔT is the temperature difference across the film only, the heat transfer coefficient is equivalent to the so-called "film coefficient," h. Equation (2) can be integrated, assuming U is constant, to give

$$q = U\Delta T_{av.} A. \tag{3}$$

If the heat capacities of the fluid and the surface are constant, it can be shown (9) that the average temperature difference is

$$\Delta T_{av.} = \frac{\Delta T_{II} - \Delta T_{I}}{\ln\,(\Delta T_{II}/\Delta_{I})}\,, \tag{4}$$

where the subscripts I and II refer to the initial and final temperature differences respectively. This is the logarithmic mean temperature difference.

The problem may be considered from the point of view of dimensional analysis, and a functional relation written in the form

$$F(h,d,k,w,C_p,\rho,\mu) = 0 \tag{5}$$

where

h = film coefficient (Btu./hr.ft.^{2}°F.),

d = distance between plates (ft.),

k = thermal conductivity of fluid (Btu./hr.ft.°F.),

w = mass rate of flow (lb./hr.),

C_p = heat capacity of fluid (Btu./lb.°F.),

ρ = density of fluid (lb./cu.ft.),

and μ = viscosity of fluid (lb./ft.hr.).

If, in place of w and d, we substitute the linear velocity u and the equivalent diameter D_e, the relation assumes the form

$$f(h, D_e, k, u, C_p, \rho, \mu) = 0. \tag{6}$$

Since the equation contains seven variables in four dimensions, according to the π theorem, the equation contains at least three dimensionless groups.

One possible equation is the Dittus-Boelter relation

$$\frac{hD_e}{k} = K\left(\frac{D_e u\rho}{\mu}\right)^p \left(\frac{C_p\mu}{k}\right)^q \tag{7}$$

where K, p, and q are constant. This equation must be rewritten in terms of the geometry of the system.

The equivalent diameter is defined as four times the hydraulic radius, which, in turn, is the ratio of the cross-sectional area of the fluid to the "wetted perimeter." For the parallel-plate heat exchanger this takes the form

$$r_h = \frac{2\pi rd}{4\pi r}$$

or

$$r_h = d/2.$$

The equivalent diameter is therefore equal to 2d.

The product $u\rho$ is equal to the mass rate of flow per unit area or

$$u\rho = w/2\pi rd.$$

If these relations are substituted into Equation (7), the equation can be written

$$\frac{2hd}{k} = K\left(\frac{w}{\pi r\mu}\right)^{p}\left(\frac{C_p\mu}{k}\right)^{q}. \tag{8}$$

A somewhat different relation results if the Stanton number is used in place of the Nusselt number in Equation (8). The transformation can be made by multiplying Equation (8) by $k/C_pD_eu\rho$. If the right hand side is then multiplied by μ/μ, the equation becomes

$$\frac{h}{C_pu\rho} = \frac{kK}{D_eC_pu\rho}\left(\frac{D_eu\rho}{\mu}\right)^{p}\left(\frac{C_p\mu}{k}\right)^{q}\cdot\frac{\mu}{\mu} \tag{9}$$

which reduces to

$$\frac{h}{C_p\mu\rho} = K\left(\frac{\mu}{D_eu\rho}\right)^{1-p}\left(\frac{k}{C_p\mu}\right)^{1-q}. \tag{10}$$

Substituting for D_e and $u\rho$, the equation becomes

$$\frac{2\pi r dh}{C_p w} = K\left(\frac{\pi r\mu}{w}\right)^{1-p}\left(\frac{k}{C_p\mu}\right)^{1-q} . \qquad (11)$$

In view of the complexity of the problem, no effort has been made to evaluate the constants in equations (8) and (11), nor has the data been correlated in terms of these equations. Since this work was a preliminary investigation, it is hoped that further research will be undertaken in the near future which will enable the constants to be evaluated.

Implicit in the derivation of equation (9) is the assumption that the heat transfer coefficient may be expressed by a single relationship throughout the system. Inasmuch as the flow pattern is expected to change from turbulent to streamline flow somewhere in the apparatus, this probably is not the case; and the constants in the equation may change in the transition region.

Overall coefficients of the form

$$U = q/A\Delta T_{av.} \qquad (12)$$

are less significant in this system than in the case of a shell-and-tube exchanger, since it would be difficult to apply them to larger or smaller equipment without a knowledge of the local coefficients. However, since the calculation of local coefficients requires more precise measurements of gas temperatures than could be made with the equipment available, as well as more rigid control of flow rates, this program was limited to the determination of overall coefficients. The overall coefficients represent an average of the local coefficients and should vary in a similar manner.

CHAPTER II

APPARATUS

General:--Basically, the experimental apparatus consisted of two steam-heated, parallel cast-iron plates, with a tube through the center of one of them, through which air was introduced. A schematic diagram of the apparatus is shown in Figure 1. Thermocouples were imbedded in the surfaces of the plates, and a thermocouple probe was used to measure the air temperature. The air rate was controlled by means of a needle valve, and measured by a commercial gas meter. Thermocouple e.m.f.'s were measured by means of a Leeds and Northrup Portable Precision Potentiometer.

Steam System:--Details of the heat exchanger are shown in Figure 2. The heat-transfer surfaces were faces of cylindrical steam chests. These were made by brazing 3/8-inch cast-iron plates to three-inch sections of eight-inch pipe. The inside of the bottom face of each of the steam chests was 1/8 inch higher at the center than at the rim to facilitate drainage of condensate.

Steam pressure was reduced from line pressure--approximately 75 pounds per square inch--to atmospheric pressure by means of a pressure regulator. The steam was admitted to the steam chests through 3/8-inch copper pipes, which were made fairly long to provide flexibility in changing the pipe separation.

The steam chests were nickel plated to prevent corrosion.

Condensate was removed from the chests through a 1/4-inch pipe, set at a 30° angle to the horizontal, which led to a steam trap. The steam traps were placed about four inches below their respective steam chests in order to prevent the condensate's standing in the bottom of the chests.

Air System:--Air was admitted through a 1/4-inch copper tube through the center of the bottom plate. In order to reduce heat transfer to the air inside the inlet tube, a piece of 3/4-inch pipe was placed in the center of the lower steam chest and the inlet tube run through it (Figure 2). The annular space between the inlet tube and the pipe was filled with asbestos insulation.

Compressed air from the air line in the Unit Operations Laboratory was used. The air pressure fluctuated considerably, and consequently the air rate was not perfectly constant, especially at high rates of flow. The flow rate was controlled by means of a needle valve. A commercial gas meter, supplied by the Atlanta Gas Light Company, was used to measure the rate of gas flow.

Thermocouples:-Chromel-alumel thermocouples were imbedded in the heat-transfer surfaces to give the temperature gradients. Slots were milled in the plates to a depth of 1/8 inch and to the desired length (see Figure 3), the thermocouple wires were put in the slots, and brass strips were soldered over the wires. The twisted junctions, approximately 1/8 inch in length, were bent up so that they reached just to the surface of the plate. The measured temperatures are taken as surface temperatures, but probably represent temperatures slightly below the surface.

Thermocouple e.m.f.'s were measured by means of a Leeds and Northrup Portable Precision Potentiometer. The thermocouple leads were connected to the potentiometer by means of a ten-point switch. The reference junction was immersed in an ice bath at 32° F.

Probe.--A probe was used to measure air temperature. This was made by imbedding a pair of thermocouple wires in a strip of 3/32-inch balsa wood. The junction was wrapped with aluminum foil to reduce radiation heating. The distance from the junction to the edge of the steam chests was indicated by marks on the probe.

Condensate Tube.--For purposes of a heat balance, condensate was collected through a copper tube bent to an S-shape. The long end of the tube was connected to the exit pipe of the steam trap by means of a polyethylene tube. The loop was dipped in an ice bath to cool the condensate and minimize evaporation losses. Cooled condensate was discharged into a weighed beaker to determine the heat transfer rate.

Insulation.--The steam pipes and steam chests were insulated with one inch of asbestos. In addition, the outside of the insulation was wrapped with aluminum foil to reduce losses due to radiation.

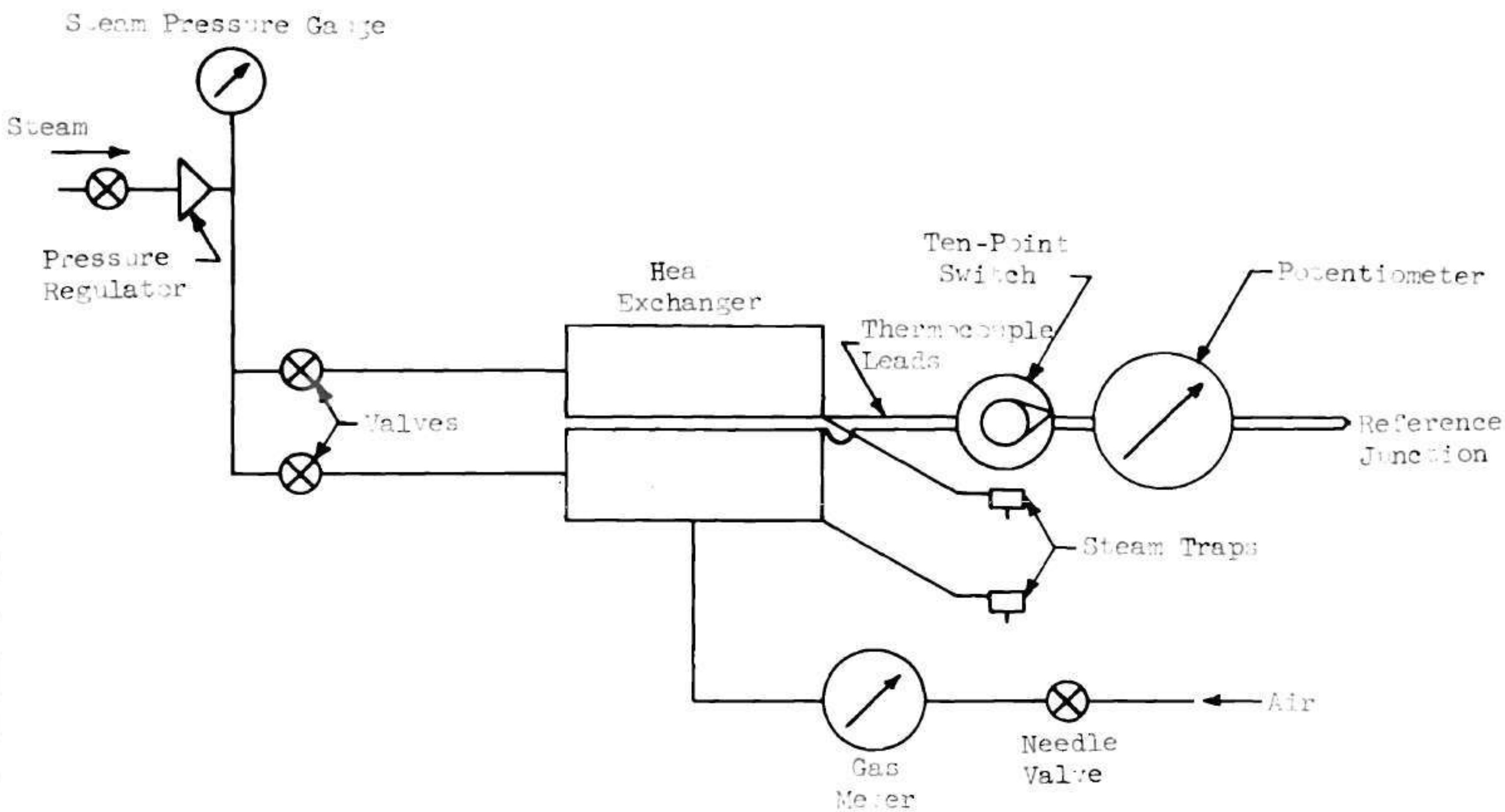

FIGURE 1

Schematic Diagram of Experimental Apparatus

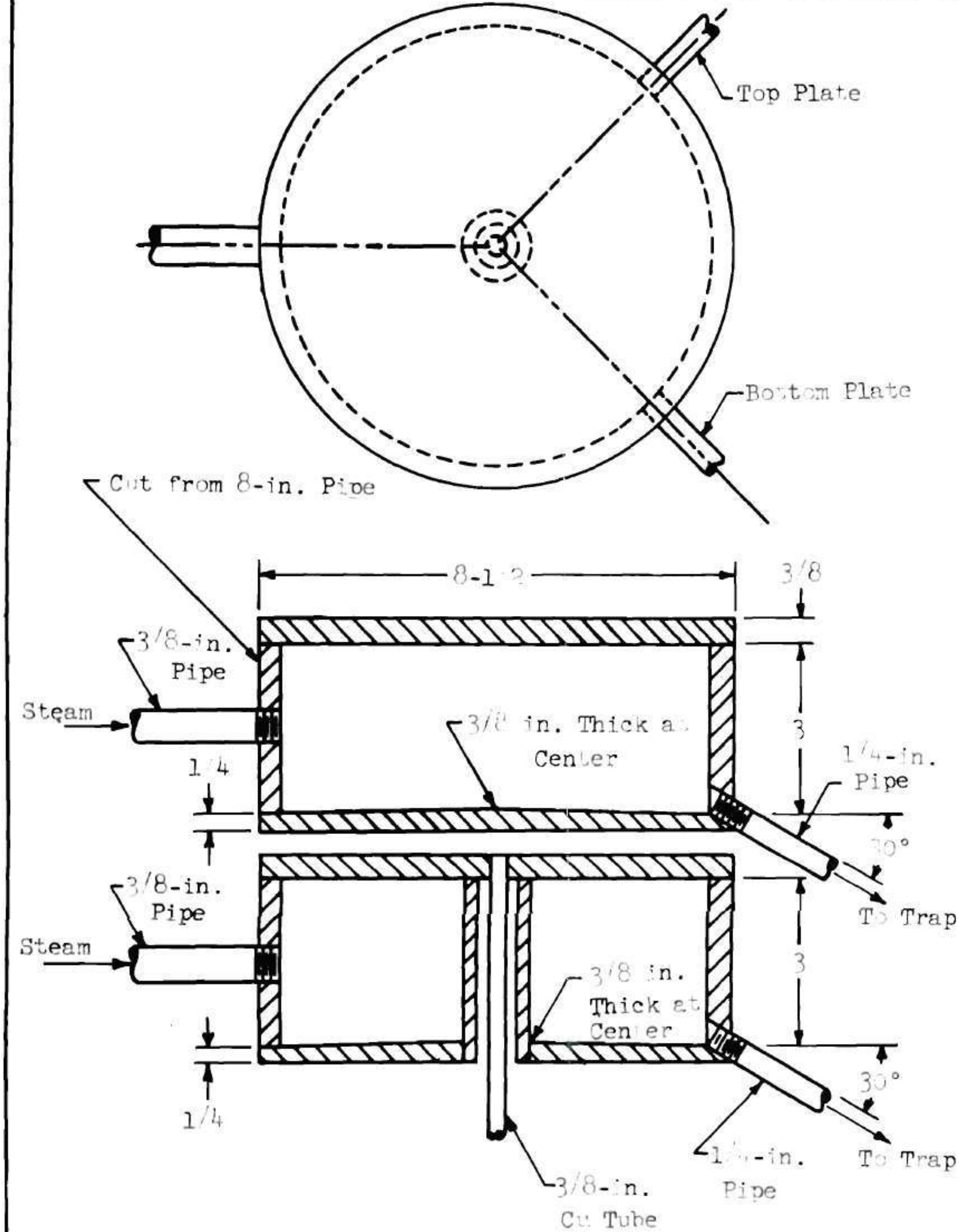

FIGURE 2

Details of Heat Exchanger

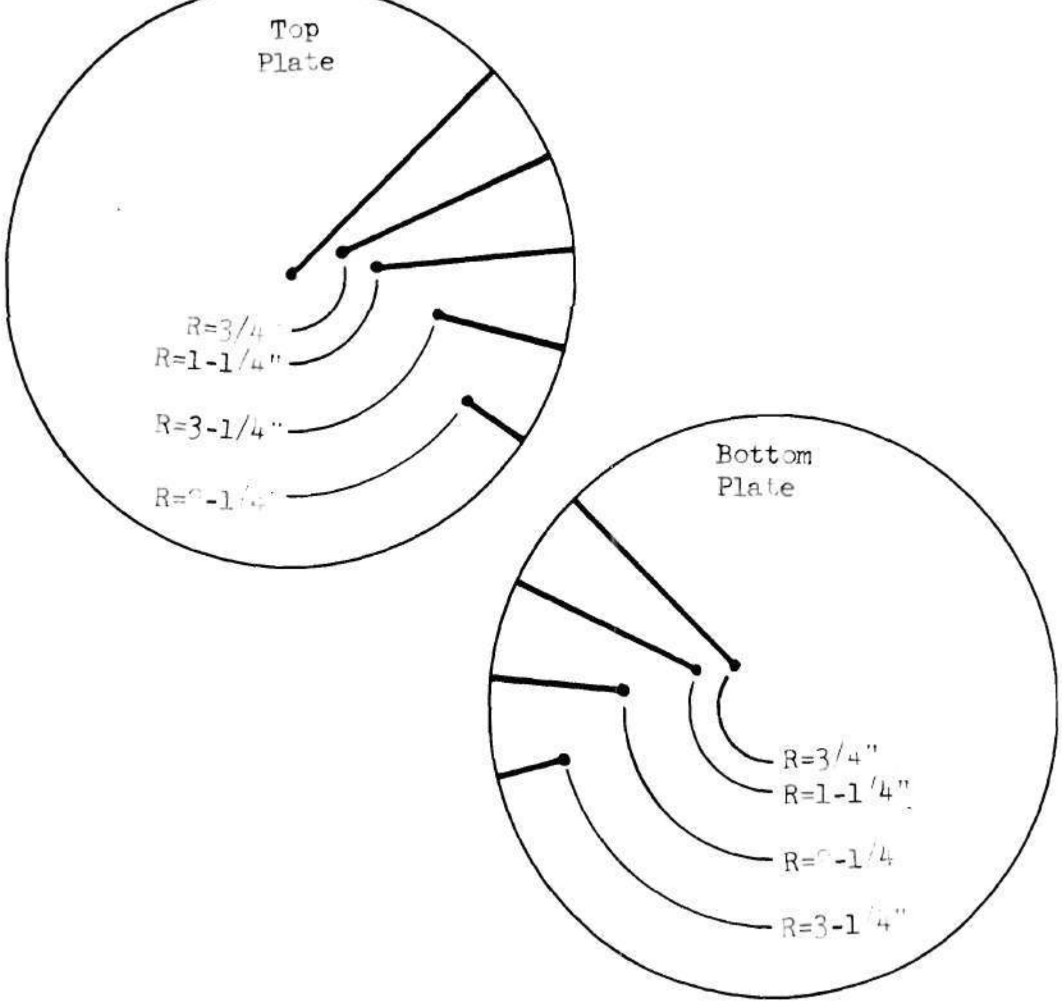

FIGURE 3

Arrangement of Thermocouples

FIGURE 4

Thermocouple Probe

CHAPTER III

EXPERIMENTAL PROCEDURE

General:--Experimental variables were the air rate through the exchanger and the distance of separation between the plates. The air rate was varied from about one cubic foot per minute to five cubic feet per minute. These rates were determined by measuring the time required for five cubic feet of air to pass the gas meter. A Kodak Timer was used for timing.

Distance between the plates was determined by separators cut from brass to the desired thickness. Separation of 0.10, 0.20, 0.30, 0.40, and 0.50 inches were used, separators being cut from one-half inch brass rod to tolerances of less than 0.001 inch. Three separators were arranged between the plates in approximately the form of an equilateral triangle.

Temperature distributions at the surface of the plates and in the air stream were determined for each run. From the inlet and outlet air temperatures and the temperatures of the plates, overall film coefficients were calculated using a logarithmic mean temperature difference.

In addition, steam rates from the top and bottom plates were determined in runs made at 0.10 and 0.30 inches. The condensate was collected from the steam traps and run through a length of polyethylene tubing to the condensate tube; this tube was then immersed in an ice bath in order to cool the condensate and reduce evaporation losses. Condensate was collected over a period of 45 minutes and weighed. A sample of condensate was collected in the same manner with no air flowing, and the weight of this sample was subtracted from the weight at the rates used to determine the heat transfer rate to the air.

Air Rate.--The air compressor maintains pressure in the lines between 80 and 100 pounds per square inch. Since the pressure variation between the time the compressor stopped and the time it started again was considerable, fairly wide fluctuations in air rate occurred during the course of a run. Because the flow rate was measured over a fairly long period of time, it is to be expected that the figures given for flow rate represent reasonable averages. This was borne out by occasional checks on the flow rate. However, the temperatures could not always be measured at precisely the average flow rates. Since these represent instantaneous values, some error is to be expected in them. An effort was made to avoid temperature measurements at very high and very low rates during runs at high overall flow rates, where fluctuations were greatest.

The measured flow rates were corrected for pressure, indicated by a pressure gauge at the meter, in order to determine the mass rate of flow. Temperature of the air in the line was measured and found to be reasonably constant at 73° F.

Steam.--The steam rate was determined for two flow rates at a separation of 0.10 inch and one rate at a separation of 0.30 inch. The steam rate at zero flow was determined first by collecting the condensate from each plate for a period of 45 minutes. The same procedures was followed in each of the three runs with air flowing. The results of the steam balances were not altogether consistent, and there is considerable doubt as to their accuracy. They do, however, give some indication of the distribution of the heat load between the two plates.

Air Temperature.--The measurement of the air temperature presented difficulties because of the danger of heating of the measuring device by radiation from the hot plates. An effort was made to prevent or at least minimize radiation heating by means of a radiation shield of aluminum foil wrapped around the thermocouple junction of the probe. It is likely, nevertheless, that the values given for air temperature are slightly high because of the radiation effect.

Considerable difficulty was experienced in obtaining reasonable values for air temperatures at the edge of the plates, since air currents in the room disrupted the flow. Where exit air temperatures seem much too low, more reasonable temperatures have been found for use in calculation by extrapolating the curve of air temperature versus radius. These values are shown in parenthesis in Appendix 1.

In some cases, a fairly rapid fluctuation of air temperature was noted. The cause of this phenomenon was the subject of some conjecture, but it is believed to be the existence of a region of transition from turbulent to viscous flow, attended by a local unevenness in flow rate. Mean values for the temperatures were estimated in such cases. These values are marked with an asterisk (*) in Appendix 1.

Some unevenness of flow was noted around the periphery of the plates. Because of the pattern, it was believed to have been caused by the presence of the separators in the air stream. If this explanation is correct, the flow within the apparatus should be substantially uniform around the circumference; and any effect of the unevenness of flow has been neglected in calculations.

CHAPTER IV

CALCULATIONS

General:--Overall film coefficients were calculated from the equation

$$h = \frac{q}{A\Delta T_{lm}} .$$

The heat rate was determined by the equation

$$q = wC_p(T_{II} - T_I).$$

The heat capacity was assumed to have a constant value of 0.24 Btu. per pound per degree Fahrenheit.

The mass rate of flow of air was determined from the volume rate of flow by correcting the metered volume for pressure shown by the gauge on the meter, and for temperature, assumed to be constant at 73° F. The air was assumed to be an ideal gas, and its density at standard temperature and pressure was calculated.

A slide rule was used for all calculations.

Sample Calculation:--Data for run number three were as follows:

Volume rate of flow . . . . 3.57 cu. ft./min.

Meter pressure. . . . . 0.7 psig

Rise in air temperature . . . . 117° F.

Log mean temperature difference . . . . . 35.0° F.

Density was calculated on the basis of the ideal gas law:

$$\rho = \frac{29}{359} = 0.081 \text{ lb./cu.ft.}$$

The volume rate of flow was corrected for pressure and temperature and changed to an hourly basis:

$$3.57 \times (15.4/14.7) \times (492/533) \times 60 = 207 \text{ cu. ft./hr.}$$

The corrected volume was then multiplied by the density at standard conditions to get the mass rate of flow:

$$207 \times 0.081 = 16.8 \text{ lb./hr.}$$

The heat rate was obtained by multiplying the mass rate of flow by the heat capacity and the temperature rise of the air:

$$16.8 \times 0.24 \times 117 = 473 \text{ Btu./hr.}$$

The heat transfer coefficient was then determined by dividing the heat rate by the product of the total area of the heat exchanger and the logarithmic mean temperature difference:

$$\frac{473}{0.789 \times 35.0} = 17.1 \text{ Btu./hr.ft}^{2}{}^{\circ}\text{F.}$$

Steam Balance.--The steam balance was made primarily for the purpose of getting some indication of the heating load carried by each of the two plates. Since the data obtained were quite inaccurate, the only calculations made were to determine the percentage of the total heat load that each plate carried.

The steam rate at zero flow was determined by collecting a sample of condensate for 45 minutes with no air flowing. Condensate samples were then taken with air flowing. A sample calculation of the heat load distribution is as follows:

Flow rate. . . 3.00 c.f.m. Separation . . . 0.10 inch

	Top plate	Bottom Plate
Beaker plus condensate	520.86	591.03
Beaker	171.20	171.20
Condensate	349.66	419.83
Zero Flow	271.34	354.51
Net	78.32	65.32

Total flow 143.64

Per cent to top 78.32/143.64 = 54.5 per cent

Per cent to bottom 65.32/143.64 = 45.5 per cent

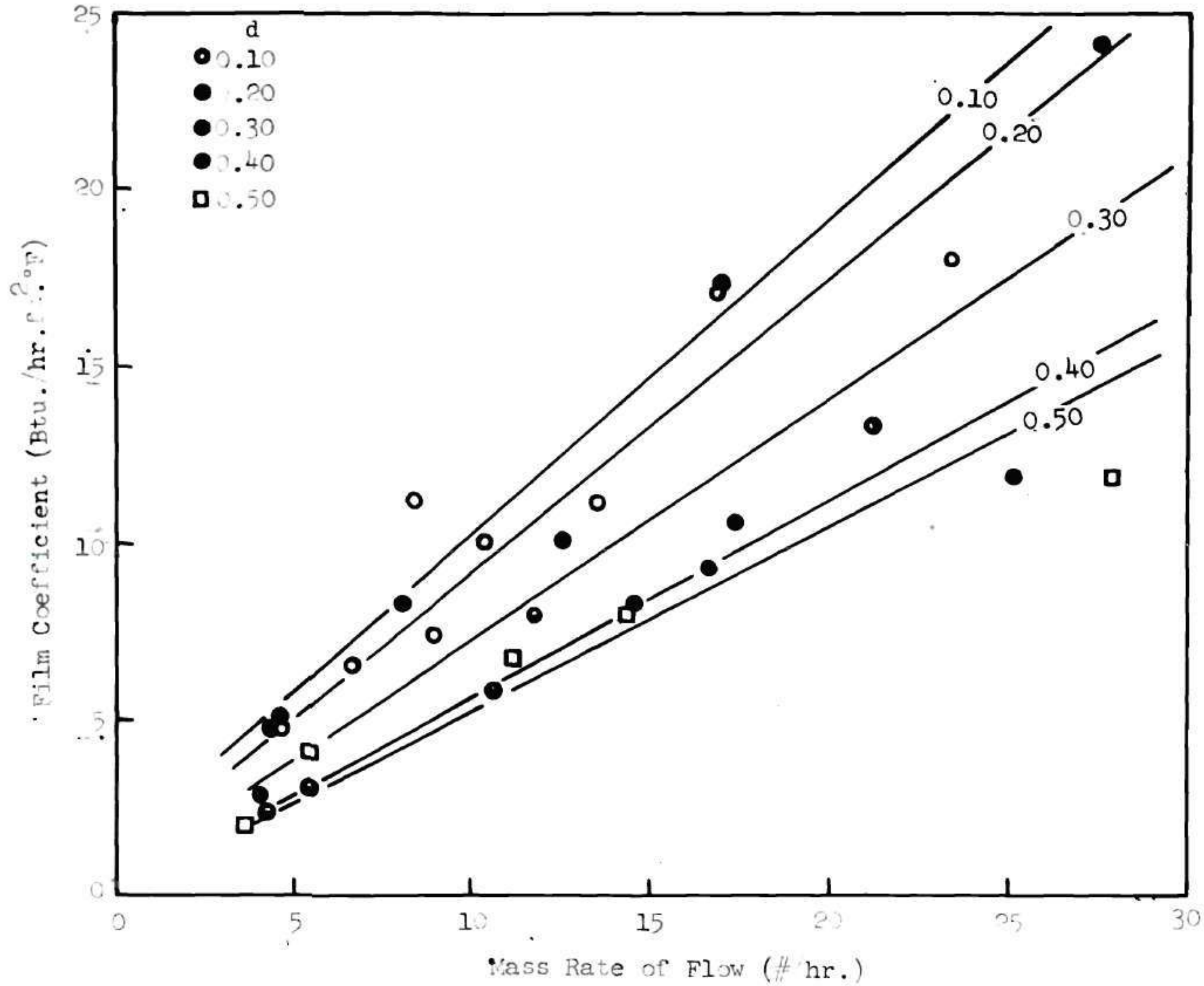

FIGURE 5

Plot of Film Coefficient vs. Mass Rate of Flow.

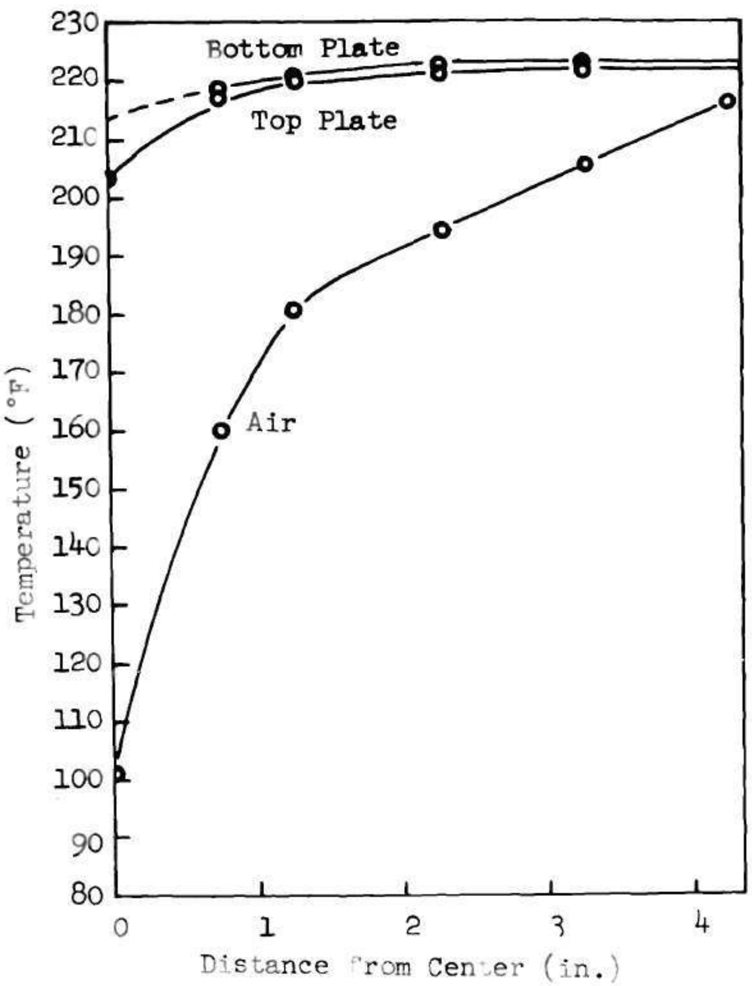

FIGURE 6

Temperature Variation Through Exchanger
Run No. 8

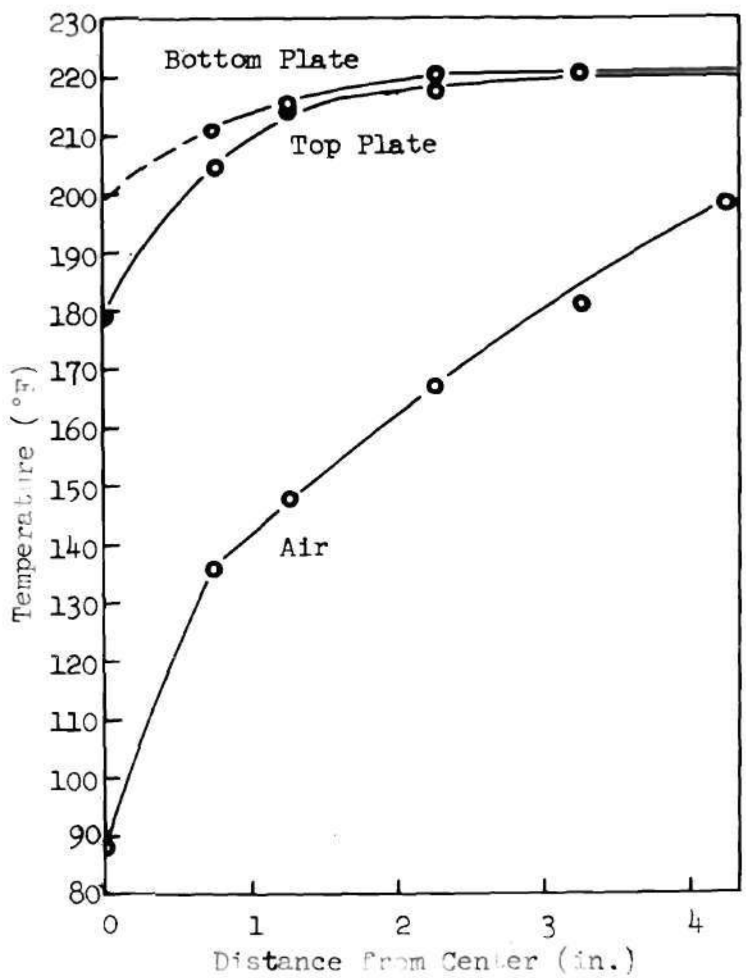

FIGURE 7

Temperature Variation Through Exchanger
Run No. 11

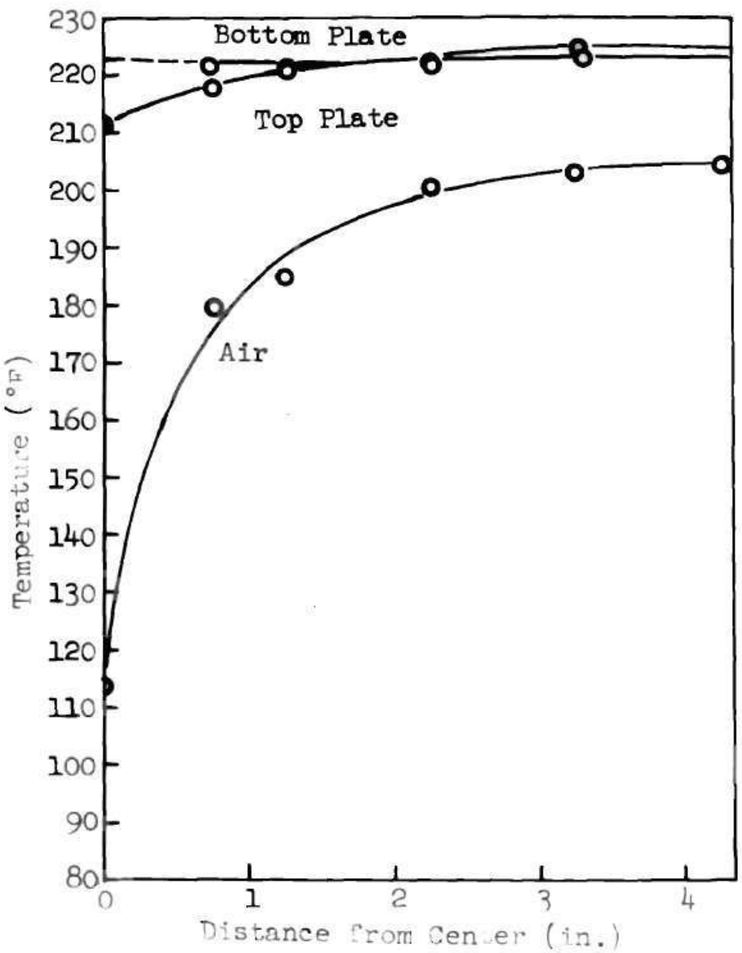

FIGURE 8

Temperature Variation Through Exchanger
Run No. 17

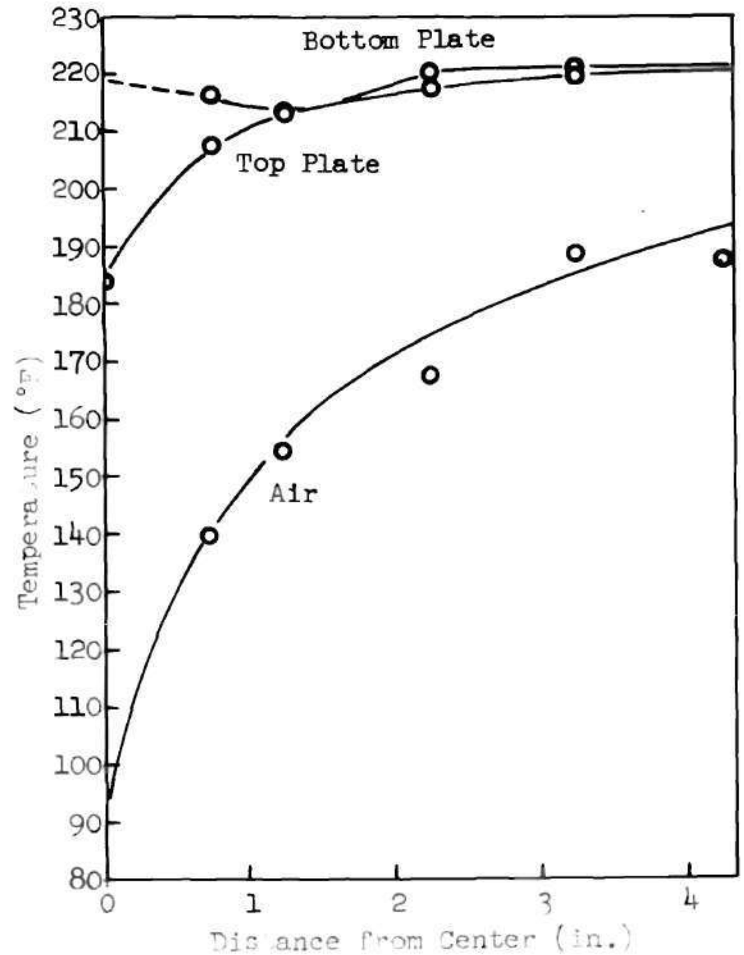

FIGURE 9

Temperature Variation Through Exchanger
Run No. 16

CHAPTER V

DISCUSSION AND CONCLUSIONS

General:--The data compiled in this investigation are inconclusive, in that they give no insight into such important aspects of the problem as the flow characteristics and the manner of variation of local heat transfer coefficients, without which no fundamental analysis of the problem is possible. The primary value of this work has been to suggest a number of problems to be solved by further research, and to indicate methods of attack that had not been considered heretofore. However, a number of generalizations can be made from the data available.

Overall Coefficients:--Although the local film coefficients are of primary interest in this problem in view of the fact that a wide variation in these coefficients is indicated, the data were not sufficiently precise to permit their accurate determination. It was felt that overall coefficients would provide an average value to give some indication of the manner in which the local coefficients vary.

The variation of overall coefficients with mass velocity for each of the five separations is shown in Figure 5. The scattering of points on these curves is considerable. However, it is apparent that, although the overall coefficients are somewhat smaller at wider separation, the variation is not as a power of the separation, as predicted by Equation (8).

An explanation for this discrepancy is suggested by the temperature distribution curves in Figures 7 and 8. According to these curves, the temperature of the central portion of the bottom plate is actually higher than the temperature half way between the center and the edge.

This can be accounted for by assuming that the air does not flow out with a velocity that is evenly distributed throughout the height of the air space, but rather that it impinges upon the top plate and flows outward in a widening channel along the top of the air space, leaving a layer of comparatively still air near the bottom plate. The suggested flow pattern is shown in Figure 10. The effect of such channelling of the air would be to reduce the cross-sectional area of flow, thereby counteracting the effect of increasing the distance between plates.

It would be of interest to make a further investigation using some sort of diffusion device at the inlet tube to eliminate this effect. Also, at very high air velocities, the air stream might tend to fill the air space more completely. Under such conditions, the relation between film coefficient and distance between plates might be more nearly that predicted by Equation (8).

Steam Balance:--The comparatively uniform temperature distribution across the bottom plate, contrasted with the wide variation in temperature between the center and edge of the top plate, led to the conclusion that the top plate was carrying most of the heat load. The hypothesis that a still layer of air existed next to the bottom plate would lend strong support to this conclusion. A steam balance was run to check the assumption of greater heat load on the top plate.

The steam balance confirmed the conclusion in a general way, although it indicated that the difference was less than had been originally believed. In one case, it actually showed a greater heat load to the bottom plate. Some of the condensate from the bottom plate can be accounted for because of heating of the air in the inlet tube. However, it would be a mistake to attribute too much precision to these measurements, since they showed considerable variation from one time to another and are not considered very reliable. They can be construed to be little more than an indication that more heat was transferred from the top plate than from the bottom . No quantitative conclusions have been drawn from them.

Flow Characteristics:--A detailed analysis of the flow characteristics of this system was outside the scope of this investigation. However, as the work progressed, it became evident that the flow characteristics had a very strong bearing on the problem and could not be separated from it. Therefore, certain qualitative observations were made based on the data available.

Figures 6 and 7 show a rapid change in slope in the air temperature distribution curve about half way between the center and edge of the plate. Since there is no reason to expect such an effect on the assumption that the local heat transfer coefficient is constant or varies continuously over the radius of the plate, it may be supposed that the manner of variation of the coefficient changes at that point. Such a change would be explained by a transition from turbulent to viscous flow at that point.

In some cases, the air temperature at a point fluctuated considerably, making the measurement difficult and imprecise. This effect is discussed in Chapter III, above. It is suggested that such fluctuation of temperature might occur in a region of transition from turbulent to streamline flow.

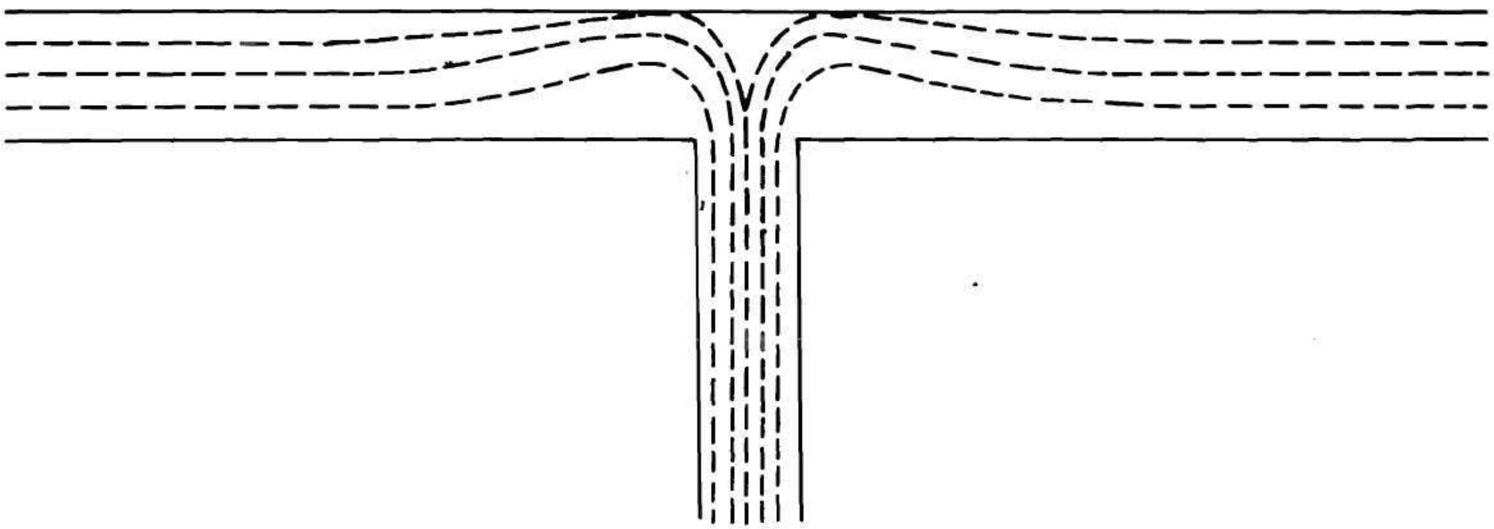

FIGURE 1

Suggested Flow Pattern of Air Through Heat Exchanger

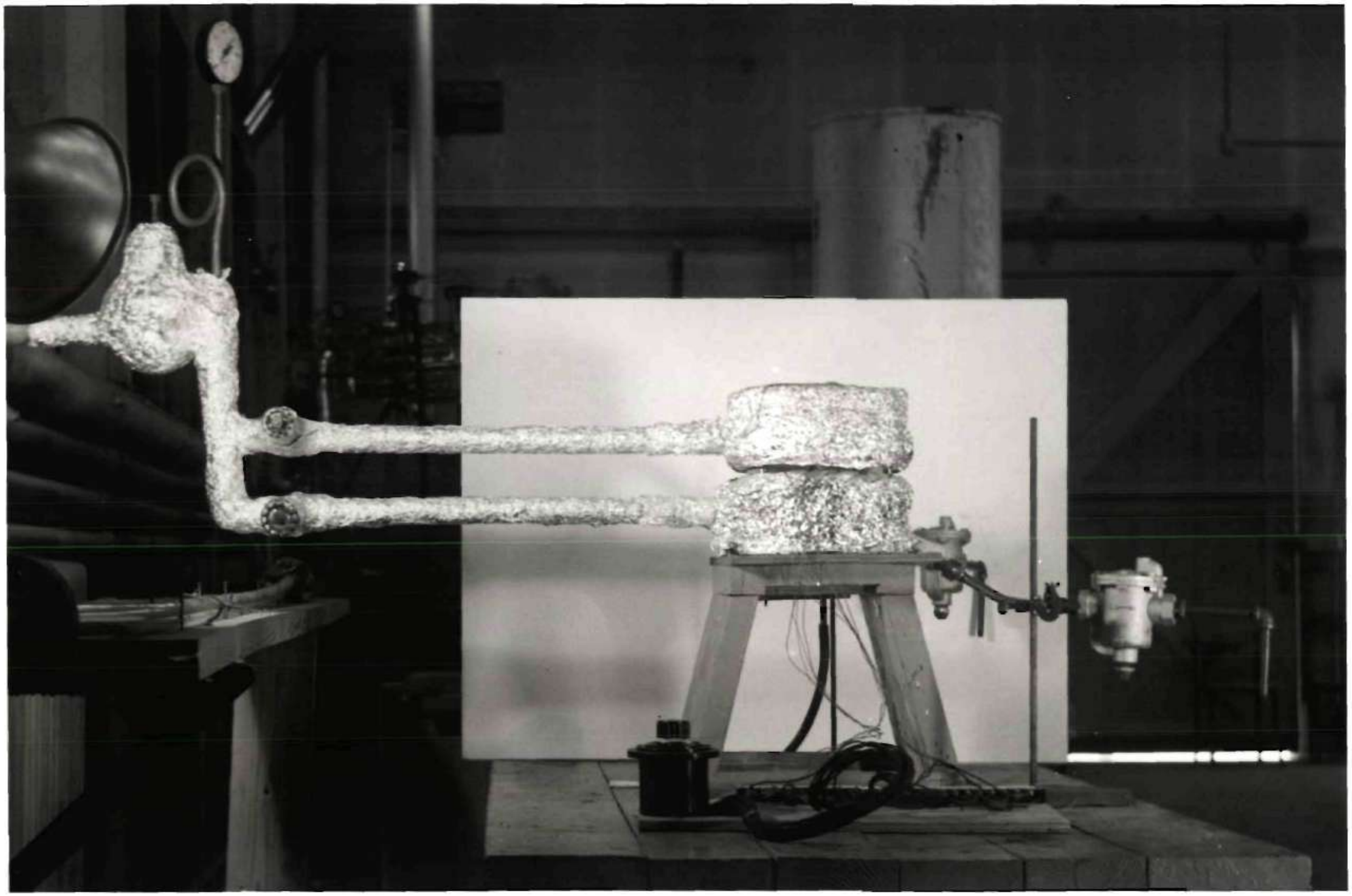

FIGURE 11

Experimental Apparatus

These explanations are admittedly speculative. Possibly other, equally valid, explanations could be proposed for the phenomena in question. Certainly the effects were not sufficiently consistent to warrant rigorous interpretation in any way. However, in view of the large variation in air velocity, a transition from turbulent to viscous flow within the apparatus seems probable. Further investigation of the flow characteristics of the system would be of immense value in interpreting results.

Recommendations:--It is hoped that further experiments of greater precision can be undertaken in the near future. A separate investigation of the flow problem would be very useful, as would a detailed mathematical analysis of the problem, which the present investigator did not have the background to undertake.

APPENDICES

Appendix I

Original Data

Run Number	1		2		3	
Separation (in.)	0.10		0.10		0.10	
Air Rate ($ft.^3$/min.)	1.11		1.87		3.57	
Meter Press. (psig)	0		0		0.7	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number						
1	4.343	222.8	4.330	222.2	4.312	221.3
2	4.335	222.3	4.323	221.9	4.305	221.1
3	4.314	221.5	4.290	220.6	4.268	219.4
4	4.277	219.9	4.223	217.5	4.165	215.0
5	4.314	221.5	4.301	220.9	4.290	220.5
6	4.297	220.8	4.276	219.8	4.254	219.0
7	4.279	220.0	4.254	219.0	4.229	217.9
8	4.236	218.1	4.178	215.5	4.108	212.6
9	3.947	206.6	3.771	198.0	3.585	190.0
Air Temperature						
Radius (in.)						
0	1.976	120.0	1.722	108.9	1.432	96.0
0.75	3.068	167.6	2.955	162.6	2.551	145.2
1.25	3.788	198.8	3.541	188.0	3.026	165.8
2.25	4.059	210.4	3.893	203.3	3.429	183.1
3.25	4.205	216.6	4.106	212.5	3.812	199.9
4.25	4.229	217.9	4.231	217.9	4.121	213.1
ΔT_I		86		89		94
ΔT_{II}		7		3		8
ΔT_{lm}		28.3		25.4		35.0
Increase in air temperature		98		109		117

Appendix I (Continued)

Original Data

Run Number	4		5		6	
Separation (in.)	0.10		0.10		0.10	
Air Rate (ft.^{3}/min.)	4.55		1.48		3.00	
Meter Press. (psig)	2.2		0		0.4	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number						
1	4.293	220.5	4.380	224.3	4.253	218.9
2	4.291	220.4	4.366	223.7	4.252	218.9
3	4.234	218.0	4.349	223.0	4.211	217.1
4	4.046	209.9	4.294	220.6	4.103	212.4
5	4.277	219.9	4.371	224.0	4.240	218.3
6	4.220	217.4	4.351	223.1	4.203	216.8
7	4.192	216.2	4.323	221.9	4.160	214.9
8	3.980	207.0	4.249	219.8	3.984	207.3
9	3.465	184.8	3.809	199.8	3.346	179.5
Air Temperature						
Radius (in.)						
0	1.246	87.8	1.400	94.6	0.982	76.3
0.75	2.542	144.9	---	---	---	---
1.25	2.931	161.6	---	---	---	---
2.25	3.734*	196.3	---	---	---	---
3.25	3.806	199.6	---	---	---	---
4.25	3.886	203.0	4.215	217.3	3.891	203.2
ΔT_I		96		105		103
ΔT_{II}		17		8		15
ΔT_{lm}		45.6		37.7		45.5
Increase in air temperature		115		122		127

Appendix I (Continued)

Original Data

Run Number	7		8		9	
Separation (in.)	0.20		0.20		0.20	
Air rate (ft.^{3}/min.)	0.99		1.79		3.01	
Meter Press. (psig)	0		0		0.2	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number						
1	4.350	223.0	4.334	222.3	4.322	221.9
2	4.344	222.7	4.330	222.1	4.317	221.6
3	4.332	222.2	4.303	221.0	4.270	219.6
4	4.299	220.9	4.243	219.4	4.190	216.0
5	4.321	221.8	4.309	221.3	4.297	220.8
6	4.307	221.1	4.296	220.7	4.274	219.9
7	4.287	220.4	4.267	219.5	4.237	218.2
8	4.251	219.0	4.224	217.6	4.158	214.9
9	3.979	207.0	3.893	203.2	3.743	196.9
Air Temperature						
Radius (in.)						
0	1.792	111.9	1.547	101.2	1.400	94.6
0.75	3.042	166.6	2.890	159.8	2.626	148.4
1.25	3.480	185.4	3.365	180.4	3.056	167.1
2.25	3.965	206.4	3.680	194.2	3.477	185.3
3.25	4.140	213.9	3.927	204.9	3.809	199.7
4.25	4.138	(218)	4.189	216.1	4.017	208.7
ΔT_I		95		102		102
ΔT_{II}		5		6		13
ΔT_{lm}		30.6		33.8		43.1
Increase in air temperature		106		115		114

Appendix I (Continued)

Original Data

Run Number	10		11		12	
Separation (in.)	0.20		0.20		0.30	
Air Rate (ft.^{3}/min.)	3.65		5.27		0.96	
Meter Press. (psig)	0.6		2.4		0	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv	Temp. °F.
Thermocouple Number						
1	4.305	221.1	4.288	220.3	4.346	222.8
2	4.299	220.9	4.281	220.1	4.340	222.5
3	4.220	217.5	4.169	215.2	4.318	221.8
4	4.136	213.7	4.084	211.6	4.288	220.4
5	4.281	220.1	4.271	219.6	4.319	221.8
6	4.246	218.6	4.222	217.7	4.303	221.1
7	4.200	216.7	4.149	214.3	4.283	220.2
8	4.077	211.2	3.922	204.6	4.246	218.6
9	3.610	191.1	3.317	178.4	3.984	207.4
Air Temperature						
Radius (in.)						
0	1.334	91.8	1.253	88.2	1.868	115.2
0.75	2.508	143.3	2.335	135.9	3.209	173.7
1.25	2.702	151.8	2.603	147.4	3.613	191.4
2.25	3.180	172.5	3.040	166.4	3.767	197.9
3.25	3.566	189.2	3.372	180.9	4.001	207.9
4.75	4.110	212.7	3.793	199.0	3.968	(213)
ΔT_I		99		90		92
ΔT_{II}		8		21		19
ΔT_{lm}		36.2		47.5		46.1
Increase in air temperature		121		111		98

Appendix I (Continued)

Original Data

Run Number	13		14		15	
Separation (in.)	0.30		0.30		0.30	
Air Rate (ft.^{3}/min.)	1.04		2.27		3.67	
Meter Press. (psig)	0		0		0.8	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number						
1	4.335	222.4	4.334	222.4	4.317	221.6
2	4.327	222.1	4.326	222.1	4.306	221.1
3	4.301	221.0	4.270	219.6	4.210	217.1
4	4.271	219.7	4.263	219.2	4.231	218.1
5	4.307	221.2	4.312	221.4	4.294	220.6
6	4.293	220.6	4.291	220.5	4.266	219.5
7	4.273	219.9	4.265	219.4	4.201	216.7
8	4.236	218.2	4.193	216.3	4.093	212.0
9	3.927	204.8	3.864	202.1	3.708	195.3
Air Temperature						
Radius (in.)						
0	1.680	106.9	1.604	103.7	1.490	98.6
0.75	3.095	168.8	3.071	167.8	2.540	144.7
1.25	3.375	180.9	3.250	175.5	2.846	157.9
2.25	3.755	197.3	3.599	191.6	3.510	186.8
3.25	4.102*	212.4	3.990	207.5	3.708	195.4
4.25	4.081	(217)	3.899	(215)	3.601	(199)
ΔT_I		98		98		97
ΔT_{II}		5		7		22
ΔT_{lm}		31.5		34.4		49.9
Increase in air temperature		110		111		100

Appendix I (Continued)

Original Data

Run Number	16		17		18	
Separation (in.)	0.30		0.40		0.40	
Air Rate (ft.^{3}/min.)	4.55		0.894		1.18	
Meter Press. (psig)	1.6		0		0	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number						
1	4.300	220.9	4.341	222.7	4.340	222.6
2	4.278	220.0	4.336	222.4	4.338	222.5
3	4.142	214.1	4.317	221.6	4.297	220.8
4	4.178	215.7	4.301	221.0	4.301	220.9
5	4.275	219.9	4.325	222.0	4.336	222.4
6	4.222	217.3	4.313	221.5	4.330	222.2
7	4.126	213.5	4.289	220.4	4.286	220.3
8	3.986	207.4	4.230	218.0	4.190	216.1
9	3.439	183.7	4.079	211.2	4.021	208.9
Air Temperature						
Radius (in.)						
0	1.343	92.2	1.819	113.1	1.551	101.3
0.75	2.416	139.5	3.331	179.0	3.376	181.0
1.25	2.763	154.8	3.451	184.1	3.379	181.6
2.25	3.076	167.9	3.814	200.1	3.752	197.3
3.25	3.556	188.7	3.845	201.3	4.020*	208.4
4.25	3.550	(197)	3.915	204.2	3.951	(216)
ΔT_I		92		98		108
ΔT_{II}		24		18		7
ΔT_{lm}		50.7		47.2		37.0
Increase in air temperature		105		91		115

Appendix I (Continued)

Original Data

Run Number	19		20		21	
Separation (in.)	0.40		0.40		0.40	
Air Rate ($ft.^3/min.$)	2.00		2.38		2.63	
Meter Press. (psig)	0		0		0	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number						
1	4.301	220.9	4.319	221.8	4.303	221.0
2	4.298	220.8	4.306	221.1	4.291	220.5
3	4.245	218.6	4.256	219.1	4.240	218.2
4	4.264	219.3	4.277	219.9	4.260	219.2
5	4.292	220.7	4.299	220.8	4.288	220.3
6	4.279	220.0	4.281	220.1	4.251	218.9
7	4.204	216.9	4.221	217.6	4.149	214.3
8	4.081	211.4	4.129	213.6	3.971	206.7
9	3.897	203.5	3.821	200.2	3.722	196.0
Air Temperature						
Radius (in.)						
0	1.406	94.9	1.580	102.6	1.270	88.9
0.75	3.312	178.1	3.166*	171.9	3.253	175.7
1.25	3.221	164.1	3.188*	172.8	3.169	172.1
2.25	3.570	189.3	3.651*	193.0	3.397	181.8
3.25	3.870*	202.2	3.671	193.9	3.706	195.2
4.25	3.841	(210)	3.774	198.3	3.706	(202)
ΔT_I		108		98		107
ΔT_{II}		11		23		19
ΔT_{lm}		42.3		52.8		50.9
Increase in air temperature		115		95		113

Appendix I (Continued)

Original Data

Run Number	22		23		24	
Separation (in.)	0.40		0.40		0.40	
Air Rate (ft.^{3}/min.)	3.12		3.57		5.00	
Meter Press. (psig)	0.6		0.6		1.8	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number						
1	4.310	221.3	4.307	221.2	4.292	220.6
2	4.305	221.0	4.285	220.2	4.251	218.8
3	4.240	218.4	4.236	218.1	4.207	216.8
4	4.265	219.4	4.260	219.2	4.111	212.9
5	4.300	220.9	4.289	220.3	4.289	220.4
6	4.251	218.9	4.267	219.5	4.212	217.1
7	4.149	214.3	4.200	216.6	4.068	210.9
8	3.960	206.2	4.110	212.8	3.760	197.6
9	3.681	194.1	3.649	192.9	3.597	190.6
Air Temperature						
Radius (in.)						
0	1.235	87.4	1.295	90.0	1.122	82.5
0.75	3.309	168.0	3.014	165.2	3.168	172.0
1.25	3.093	168.7	3.059*	167.1	2.930	161.6
2.25	3.460	184.0	3.541	188.1	3.023	165.8
3.25	3.689	194.5	3.605	190.9	3.274	176.5
4.25	3.713	195.4	3.679	194.1	3.460	184.5
ΔT_I		107		103		108
ΔT_{II}		26		27		36
ΔT_{lm}		57.4		56.7		65.5
Increase in air temperature		107		104		102

Appendix I (Continued)

Original Data

Run Number	25		26		27	
Separation (in.)	0.50		0.50		0.50	
Air Rate (ft.^{3}/min.)	0.808		1.22		2.50	
Meter Press. (psig)	0		0		0	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number						
1	4.391	224.8	4.302	221.0	4.323	221.9
2	4.388	224.6	4.301	220.9	4.311	221.3
3	4.362	223.6	4.278	220.0	4.289	220.4
4	4.369	223.9	4.293	220.6	4.296	221.7
5	4.386	224.5	4.316	221.5	4.315	221.6
6	4.382	224.4	4.315	221.5	4.300	220.9
7	4.358	223.3	4.279	220.0	4.282	220.1
8	4.318	221.7	4.221	217.7	4.112	212.9
9	4.110	212.7	3.981	207.1	3.771	198.1
Air Temperature						
Radius (in.)						
0	2.382	138.0	1.626	104.6	1.656	105.7
0.75	3.680	194.2	3.515	187.0	3.590	190.1
1.25	3.608	191.0	3.473	185.9	3.476	185.3
2.25	3.792	198.9	3.789	198.9	3.631	192.0
3.25	3.988	207.5	3.949	205.8	3.746	196.9
4.25	4.034	209.4	3.926	(209)	3.757	(200)
ΔT_I		74		102		92
ΔT_{II}		16		12		22
ΔT_{lm}		37.9		42.0		49.0
Increase in air temperature		71		104		94

Appendix I (Continued)

Original Data

Run Number	28		29	
Separation (in.)	0.50		0.50	
Air Rate ($ft.^3/min.$)	3.12		5.00	
Meter Press. (psig)	0.4		3.5	
Surface Temperature	emf mv.	Temp. °F.	emf mv.	Temp. °F.
Thermocouple Number				
1	4.299	220.8	4.244	218.6
2	4.281	220.1	4.202	216.7
3	4.262	219.2	4.147	214.5
4	4.274	219.9	4.184	216.0
5	4.288	220.3	4.230	217.9
6	4.243	218.5	4.123	213.3
7	4.170	215.2	4.016	208.6
8	4.050	210.1	3.850	201.4
9	3.669	193.8	3.405	182.2
Air Temperature				
Radius (in.)				
0	1.268	88.8	1.114	82.1
0.75	3.609	191.0	2.726*	152.8
1.25	3.389	181.5	2.744*	153.5
2.25	3.449*	184.1	2.785*	155.2
3.25	3.595*	190.5	2.772*	154.9
4.25	3.680	194.1	3.274	176.4
ΔT_I		105		100
ΔT_{II}		27		43
ΔT_{lm}		57.2		66.5
Increase in air temperature		105		94

*Temperature showed wide fluctuation.

Appendix II

Calculations

Run No.	Volume Metered	Pressure Correction	Temperature Correction	Volume Corrected to STP	Density at STP	Mass Rate of Flow
1	1.11 x 60	1	0.924	61.5	0.081	4.59
2	1.87 x 60	1	0.924	103.6	0.081	8.41
3	3.57 x 60	15.4/14.7	0.924	207	0.081	16.8
4	4.55 x 60	16.9/14.7	0.924	290	0.081	23.4
5	1.48 x 60	1	0.924	82.1	0.081	6.65
6	3.00 x 60	15.1/14.7	0.924	170.5	0.081	13.8
7	0.99 x 60	1	0.924	55.4	0.081	9.48
8	1.79 x 60	1	0.924	99.0	0.081	8.04
9	3.01 x 60	14.9/14.7	0.924	169	0.081	12.66
10	3.65 x 60	15.3/14.7	0.924	210	0.081	17.05
11	5.27 x 60	17.1/14.7	0.924	347	0.081	27.5
12	0.96 x 60	1	0.924	53.2	0.081	4.30
13	1.04 x 60	1	0.924	57.6	0.081	4.66
14	2.27 x 60	1	0.924	125.6	0.081	10.18
15	3.67 x 60	15.5/14.7	0.924	214	0.081	17.4
16	4.55 x 60	16.3/14.7	0.924	262	0.081	21.2
17	0.894 x 60	1	0.924	49.4	0.081	4.01
18	1.18 x 60	1	0.924	65.4	0.081	5.30
19	2.00 x 60	1	0.924	111	0.081	8.99
20	2.38 x 60	1	0.924	132	0.081	10.69
21	2.63 x 60	1	0.924	146	0.081	11.80
22	3.12 x 60	15.3/14.7	0.924	180	0.081	14.58
23	3.57 x 60	15.3/14.7	0.924	206	0.081	16.67
24	5.00 x 60	16.5/14.7	0.924	311	0.081	25.1
25	0.808 x 60	1	0.924	44.7	0.081	3.62
26	1.22 x 60	1	0.924	67.5	0.081	5.47
27	2.50 x 60	1	0.924	138	0.081	11.21
28	3.12 x 60	15.1/14.7	0.924	177	0.081	14.37
29	5.00 x 60	18.2/14.7	0.924	343	0.081	27.8

Appendix II (Continued)

Calculations

Run No.	Heat Capacity	Temperature Rise	Heat Rate	Area	Average Temperature Difference	Film Coefficient
1	0.24	98	108	0.789	28.3	4.75
2	0.24	109	220	0.789	25.4	10.95
3	0.24	117	473	0.789	35.0	17.10
4	0.24	115	648	0.789	45.6	18.00
5	0.24	122	194	0.789	37.7	6.55
6	0.24	127	422	0.789	45.5	10.91
7	0.24	106	114	0.789	30.6	4.73
8	0.24	115	222	0.789	33.8	8.32
9	0.24	114	347	0.789	43.1	10.20
10	0.24	121	495	0.789	36.2	17.39
11	0.24	111	906	0.789	47.5	24.20
12	0.24	98	102	0.789	46.1	2.79
13	0.24	110	123	0.789	31.2	5.01
14	0.24	111	271	0.789	34.4	10.01
15	0.24	100	418	0.789	49.9	10.60
16	0.24	105	535	0.789	50.7	13.36
17	0.24	91	87.5	0.789	47.2	2.35
18	0.24	115	147	0.789	37.0	3.12
19	0.24	115	248	0.789	42.3	7.44
20	0.24	95	244	0.789	52.8	5.86
21	0.24	113	320	0.789	50.9	7.98
22	0.24	107	375	0.789	57.4	8.31
23	0.24	104	417	0.789	56.7	9.32
24	0.24	102	618	0.789	65.5	11.93
25	0.24	71	61.8	0.789	37.9	2.06
26	0.24	104	137	0.789	42.0	4.13
27	0.24	94	253	0.789	49.0	6.55
28	0.24	105	362	0.789	57.2	8.05
29	0.24	94	627	0.789	66.5	11.96

Appendix III

Steam Balance

Zero Flow

	Top plate	Bottom plate
Beaker plus condensate	442.54	525.71
Beaker	171.20	171.20
Condensate	271.34	354.51

Flow rate...3.00 c.f.m. Separation...0.10 inch

Beaker plus condensate	520.86	591.03
Beaker	171.20	171.20
Condensate	349.66	419.83
Zero flow	271.34	354.51
Net	78.32	65.32

Total Flow 143.64

Per Cent to Top 78.32/143.64 = 54.5 per cent

Per Cent to Bottom 65.32/143.64 = 45.5 per cent

Air rate...6.68 c.f.m. Separation...0.30 inch

Beaker plus condensate	589.0	654.3
Beaker	171.2	171.2
Condensate	411.8	483.1
Zero flow	271.3	354.5
Net	140.5	128.6

Total Flow 269.1

Per Cent to Top 140.5/269.1 = 52.2 per cent

Per Cent to Bottom 128.6/269.1 = 47.8 per cent

Appendix III (Continued)

Steam Balance

Air rate...2.68 c.f.m. Separation...0.30 inch

	Top plate	Bottom plate
Beaker plus condensate	478.66	564.62
Beaker	171.20	171.20
Condensate	307.46	393.42
Zero flow	271.34	354.51
Net	36.12	38.91

Total Flow 75.03

Per Cent to Top 36.12/75.03 = 48.2 per cent

Per Cent to Bottom 38.91/75.03 = 51.8 per cent

BIBLIOGRAPHY

1. Prins, J. A., Muller, J., and Schenk, J., "Heat Transfer in Laminary Flow Between Parallel Plates." Applied Scientific Research 2A, 431-38 (1951).

2. Van der Does de Bye, J. A. W. and Schenk, J., "Heat Transfer in Laminary Flow Between Parallel Plates." Applied Scientific Research 3A, 308-16 (1952).

3. Schenk, J. and Beckers, H. L., "Heat Transfer in Laminar Flow Between Parallel Plates." Applied Scientific Research, 405-13 (1954).

4. Wagner, Carl, "Heat Transfer from a Rotating Disk to Ambient Air." Journal of Applied Physics 19, 837-39 (1948).

5. Goldstein, Sydney, Modern Developments in Fluid Dynamics, Oxford, The Clarendon Press, 1938.

6. Benedikt, E. T., "On the Flow of a Perfect Fluid Between Two Plates from a Circular Inlet Channel." Journal of Applied Physics 19, 1094-95 (1948).

7. Comolet, Raymond, "Mecanique des Fluides." Comptes Rendus, Academie des Sciences, Paris 235, 1190-93 (1952).

8. Comolet, Raymond, "Mecanique des Fluides." Comptes Rendus, Academie des Sciences, Paris 235, 1366-69 (1952).

9. McAdams, William H., Heat Transmission, New York, McGraw-Hill Book Company, Inc., 1954.