A STUDY OF UNSTEADY STATE NATURAL CONVECTION FOR A VERTICAL PLATE

by

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Professor L.F. Hamilton
Secretary of the Faculty
Massachusetts Institute of Technology
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Dear Sir:

In partial fulfillment of the requirements for the Degree of Bachelor of Science in Chemical Engineering, I respectfully submit this thesis entitled "A Study of Unsteady State Natural Convection for a Vertical Plate."

Respectfully yours,

Herbert E. Klei

ACKNOWLEDGEMENT

The author wishes to sincerely thank Professor G.C. Williams, under whom this study was made, for his invaluable guidance and unselfish aid offered during the term of this investigation. A word of thanks is given also to Mr. Chen for his help in operating the Sanborn recorder, and his many suggestions in construction of the apparatus.

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FIGURES AND ILLUSTRATION

I. SUMMARY

This paper deals with the unsteady state convection coefficient for a vertical plate in air under natural convection. The specific objective of this work is to obtain qualitative and quantative values for the unsteady state convection coefficient h.

Heat losses were studied from a piece of vertical platinum foil. Heat was put into the foil by passing an electric current through it. This heat was recognizable as loss by sensible heat and convection. The transient temperature was measured by thermocouples, and recorded on a Sanborn chart recorder.

A theoretical study depends upon the formation of a film of warm air next to the surface. Once the film has formed heat transfer becomes one of conduction through the film and one of convection from the surface of the film to the bulk of the air.

It was found that,

1. The heat transfer coefficient, h, rises to infinity as the time of heating approaches zero.
2. At the transition from a stagnant air film next to the surface to laminar flow, h dips below its unsteady state value.
3. After three seconds of heating, h is nearly at its steady state value, although the temperature of

the foil does not reach steady state until seven seconds.

4. The values of h obtained at steady state, agree very favorably with the equation given by McAdams

$$h = .29\left(\frac{\Delta T}{L}\right)^{0.25}$$

5. On cooling of the foil, h decreases almost linearly, after remaining a few seconds at the steady state value.

It is recommended that,

1. Interference photographs should be taken to determine when the transition from a stagnant layer to laminar flow takes place.
2. The temperature of the foil may be determined by resistance thermometry.
3. Measure h with the foil inclined at various angles rather than just vertically as done here.

II. INTRODUCTION

A. Background

The term heat might be considered as the basis around which most engineering operations revolve. Heat is so important to an engineer because either it is present when least desired, or it is absent when most needed. In order to obtain some control over the amount of heat present at any time, the engineer relies on its effective transfer from one region to another.

Heat is transferred by three methods: conduction, radiation, and convection, the latter being the broad area of this investigation. Heat loss by convection is expressed by the relation,

$$q = h\,A(T_s - T_r) \qquad (1)$$

For definition of symbols see nomenclature in the appendix. In order to apply eq. (1) to any operation, the heat transfer coefficient, h, must be known with some degree of accuracy. This convective coefficient, however, is a function of many variables; consequently it is the focus of most work in convective studies. Walker, Lewis, McAdams, and Gilliland (7) give (h) as a function of several operating conditions and fluid properties,

$$h = \frac{f}{2} Cp\ G(Pr)^{-2/3}. \tag{2}$$

Equation (2) may be used for most operations involving flow of fluids in pipes, and may be modified to cover many specialized cases. However, it is limited only to steady state operations. Unsteady state conditions are those where the rate of heat flow into a system is not equal to the rate of heat flow out of the system. Generally unsteady state conditions, such as heating of an ingot in a furnace, explosion in a gun barrel, and lighting of an incandescent bulb, are only of extremely short duration before steady state conditions are obtained. Consequently any measurement of unsteady state transfer coefficients must be made using apparatus having a quick response. To obtain this quick response, considerable instrumentation is required.

Since the heat transfer coefficient is a function of the geometry of the heating surface, any study of it must define the surface considered. This investigation will cover the unsteady state heat transfer between a flat vertical plate and air in natural convection. The air will be allowed to circulate around the plate in natural convection because this eliminates the variable, mass velocity, and is applicable to many engineering operations. The specific objective of this work is then to study the heat transfer coefficient, (h), both qualitatively and quantitatively under unsteady state conditions.

Very little work has been done on unsteady state convection coefficients between a plate and air. Some work was reported concerning the heat transfer between metals and air in furnaces. Sinnott and Siebert (5) found that the convection coefficient was almost constant at 3.45 BTU/Hr. sq. fT °F throughout the heating of a steel plate in a recirculating furnace from 70°F to 800°F in from 34 minutes to 110 minutes depending upon the test sample. They also showed that the convection coefficient is independent of the temperature difference between the metal being heated and the furnace. Huebler (3) concluded from theoretically comparing convective and radiative heat transfer rates that approximately twenty percent of the total heat input to a sample is done by convection with rapid heating in high temperature furnaces. It was also emphasized in his paper the need to know more about the convection coefficient.

A relatively recent tool for studying heat flow problems is the Mach-Zehnder interferometer. Its use in transient heat transfer studies is emphasized by Coulbert (1). As one of his references, Coulbert gives a M.S. thesis by G.S. Wong at the University of California titled "Determination of Local Heat Transfer Coefficients of Convection for a Cylinder in Air with the Mach-Zehnder Interferometer." At present a copy of this thesis is not available; when it is available, it should give much relevant information.

B. THEORETICAL CONSIDERATIONS

In attempting to give some qualitative picture of expected results, the formation of a film next to the heating surface must be kept in mind. As soon as the surface becomes warmer than the molecules of air next to the surface a thin film of warm air begins to form. During the heating period, the thickness of the film, l , goes from zero at the beginning to a steady state value. Thus once the film has formed, heat transfer becomes one of conduction through the film, and one of convection from the surface of the film to the bulk of the surrounding air.

If the air next to the surface is considered as a semi-infinite solid, Hildebrand (2) gives on p. 457 the expression for the temperature, t, in the air as a function of distance from the surface, x, and time Θ, after the surface of the air next to the solid was subjected to a sudden temperature use of ΔT. The relation is,

$$t = \Delta T \operatorname{erf}\left[\frac{x}{2\alpha \sqrt{\Theta}} \right] \tag{3}$$

where α^2 is the thermal diffusivity of air and is defined as

$$\alpha^2 = \frac{k}{Cp\,\rho} \tag{4}$$

Considering only conduction of heat through this gas film next to the surface, the heat loss is

$$q/A = -k\ dt/dx \tag{5}$$

If equation (3) is differentiated with respect to x, evaluated at x = 0, and substituted into equation (5), the result is equal to the heat lost by convection from this air layer to the bulk of air. These results,

$$q/A = k\left(\frac{\Delta T}{\alpha \sqrt{\pi\Theta}} \right) = h\ \Delta T \tag{6}$$

Therefore from equation (6) there results,

$$h = \frac{k}{\alpha \sqrt{\pi \Theta}} \quad . \tag{7}$$

Thus as time Θ approaches zero plus, h should approach infinity. This analogy is only good as Θ approaches zero, since the assumption that a stagnant layer of air exists near the surface is only good near Θ equal to zero, due to the chimney effect of the air as it becomes heated. The sensible heat of the gas film is only considered for a stagnant gas layer. Once the chimney effect developes, mass transfer has to be also considered.

This problem may be also approached from a less sophisticated point of view. If it is assumed that the whole temperature drop between the surface and the bulk of air occurs across this gas film, then equation (5) gives,

$$q/A = \frac{k(T_s - T_r)}{l} \tag{8}$$

If equation (8) is set equal to equation (1), it is seen that,

$$h = \frac{k}{l} \tag{9}$$

It is further see that as l approaches zero, h should approach infinity. During the formation of the film, h should come from infinity to a steady state value corresponding to the value of the steady state film thickness.

III. PROCEDURE

The system under study is a portion of a piece of platinum foil, mounted vertically in air (see Illus. 1.). Heat will be put into this foil by passing an electric current through it, from a constant current source composed of cone type heaters (see Illus. 2). The heat into the foil, will be recognizable as sensible heat and by convection losses from the surface. It is assumed that the temperatures involved would be low enough to cause minor radiation effect. In order to avoid fringe effects only a center portion of the foil was considered. Writing a heat balance around the section gives,

$$hA(T_s - T_r) = \frac{E\ I}{4.18} - w\ Cp\,\frac{d\ T}{d\theta}\ . \tag{10}$$

In equation (10) all of the terms except h are known or are experimentally obtainable. The temperature of the surface is to be measured by two copper-constantan thermocouples placed a short horizontal distance apart. The two thermocouples were used to insure on isothermal portion of the surface under consideration. An advantage of the copper-constantan thermocouple is the high thermal voltage developed (millivolts) in comparison to platinum-10 percent rhodium, platinum (microvolts). This higher

ILLUS. 1
Mounted Platinum Foil

Illus. 2
Complete Apparatus

voltage helps to supress noise pickup by the thermocouple. The voltage drop, E, across the section in consideration is measured by determining the drop across two platinum wires, the distance between them determining the length of the portion under consideration. Two platinum wires were used to eliminate the possibility of thermal voltages developing at the foil surfaces due to unequal temperatures between the two points of contact on the foil. The current is found by measuring the voltage drop across a standardized resistor. Vines (6) gives the heat capacity, Cp , for platinum by the following equation,

$$C_p \text{ (Cal/gm)} = .031678 + 6.305 \times 10^{-6} T - 1.624 \times 10^{-10} T^2 \quad (11)$$

where T = temperature of platinum in $^{\circ}$C. The value of $dT/d\theta$ may be obtained from a plot of T vs θ. $dT/d\theta$ is given in $^{\circ}$C/sec to correspond to the units of Cp given in equation (10)

Since the time of operation was very short these four voltages (two thermocouple, one current, and one for drop across section) were fed into a four channel Sanborn chart recorder. In addition to giving instantaneous readings, it also gives a record of values which may be referred to at a later time.

The only variable between runs was the current passed through the foil. Three values of current were used: 8, 11, and 13 amperes. Measurements were taken from θ equal zero minus to steady state conditions, and continued from steady state conditions with the current flowing to steady state conditions without the current flowing.

Having a plot of T vs. θ, $dT/d\theta$ was determined at .25 sec. intervals to steady state conditions. At each time interval, E, I, and ΔT were measured and a h determined from equation (10).

A similar procedure was followed during the cooling of the foil.

IV RESULTS

The results are best shown in graphical form, although they are listed in tabular form in the Appendix. Figure 1 shows the ΔT of the foil for the three different currents: 8, 11 and 13 amperes. Steady state conditions for all runs were reached within nine seconds. As the current was increased, the temperature approaches the steady state value much more rapidly; for the 8 ampere run the temperature reaches steady state almost asymptotically where as in the 13 ampere run the temperature appears to be almost a linear function of Θ for most of the distance to steady state. The steady state ΔT for the 8, 11, and 13 ampere runs were 91, 174, and 199 $^{\circ}$F respectively.

Figure 2 shows h plotted vs Θ for each run. The transfer coefficients, h, decreases very rapidly during the first 1.5 seconds, nearly reaching steady state after three seconds. In the 11 and 13 ampere runs, h dips below the steady state value at 1.5 - 1.75 seconds. In the 8 ampere run the dip below steady state is somewhat noticable around 2.5 seconds. Although the h rises very slowly to steady state after three seconds, steady values are almost reached at the end of three seconds.

If the heat loss by convection, q_c, is plotted vs Θ, Figure 3 is obtained. For the 11 and 13 ampere runs, q rises almost linearly for 1.5 seconds, and then rises very considerably during the next two seconds. The marked rise in q_c occurs at the same value of Θ that the dip occurs in the plot of h vs Θ, Figure 2. For the 8 ampere run, q_c rises almost linearly to steach state.

During the cooling of the foil the temperature of the foil, T, falls rapidly from the steady state value with the current on, to nearly room temperature within seven seconds. Figure 4 shows ΔT plotted vs Θ for the cooling of the foil. The cooling curve for the 11 and 13 ampere runs are almost co-linear after Θ equal to three seconds. For the 8 ampere runs, ΔT vs Θ curve is distinctly lower than that for the 11 and 13 ampere runs. All three curves converge as ΔT approaches zero.

If h is plotted vs Θ for the cooling of the foil, a scattering of points is obtained as shown in Figure 5. The coefficient, h, appeared to remain essentially constant at the steady state value for the first 1.5 seconds of cooling. After the 1.5 seconds, it decreased gradually. Due to the scattered distribution of points, it was difficult to draw with any degree of certainty a curve through the points. However a straight line seemed the best after a Θ of 1.5 seconds. Although a straight line correlation is very arbitrary, the points seem to indicate that such a relationship is not too unreasonable.

The important results of this study are the increase of h as Θ approaches zero, and the dip in h below steady state value at Θ equal to about 1.5 seconds.

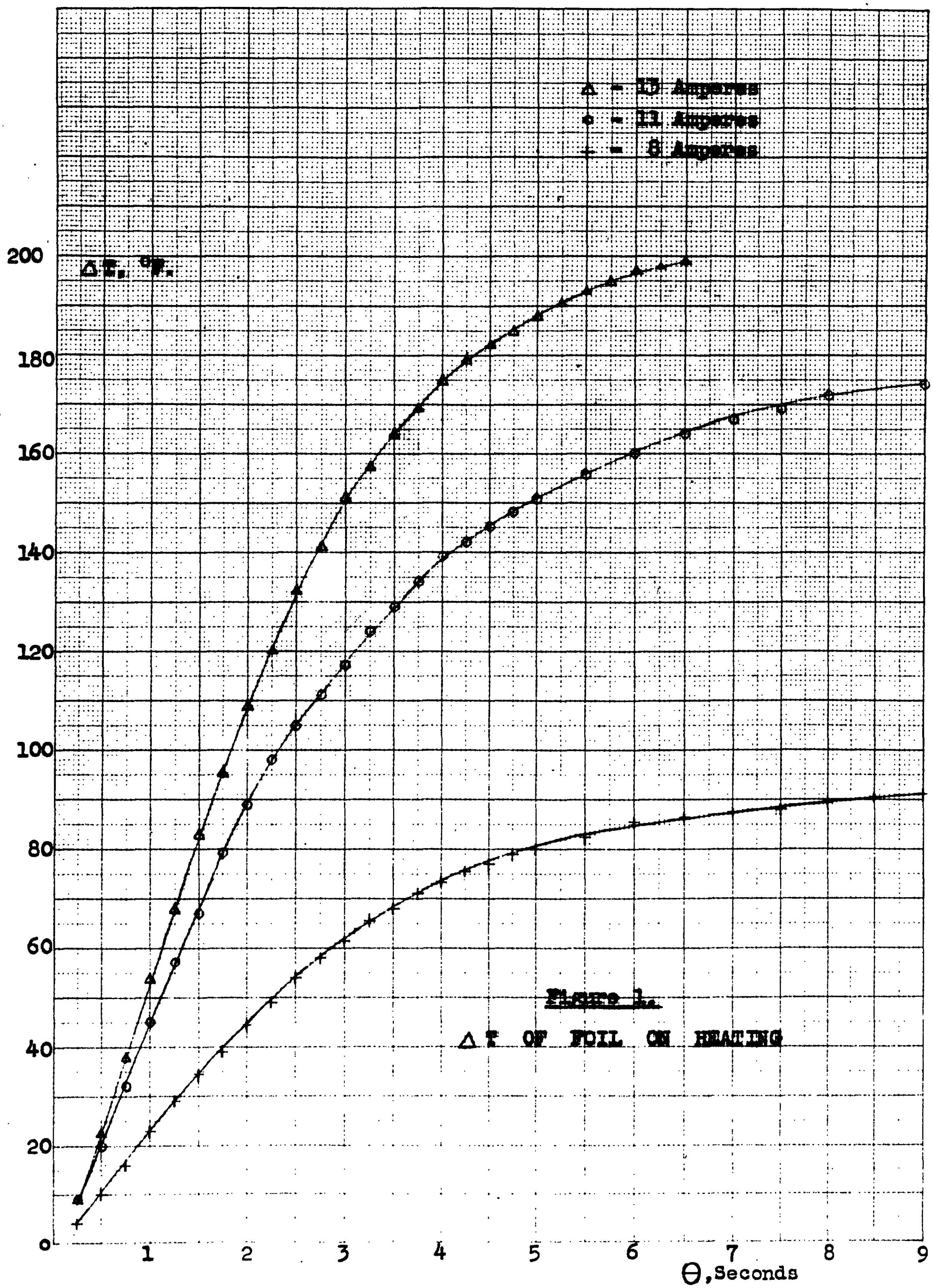


Figure 1.

ΔT OF FOIL ON HEATING

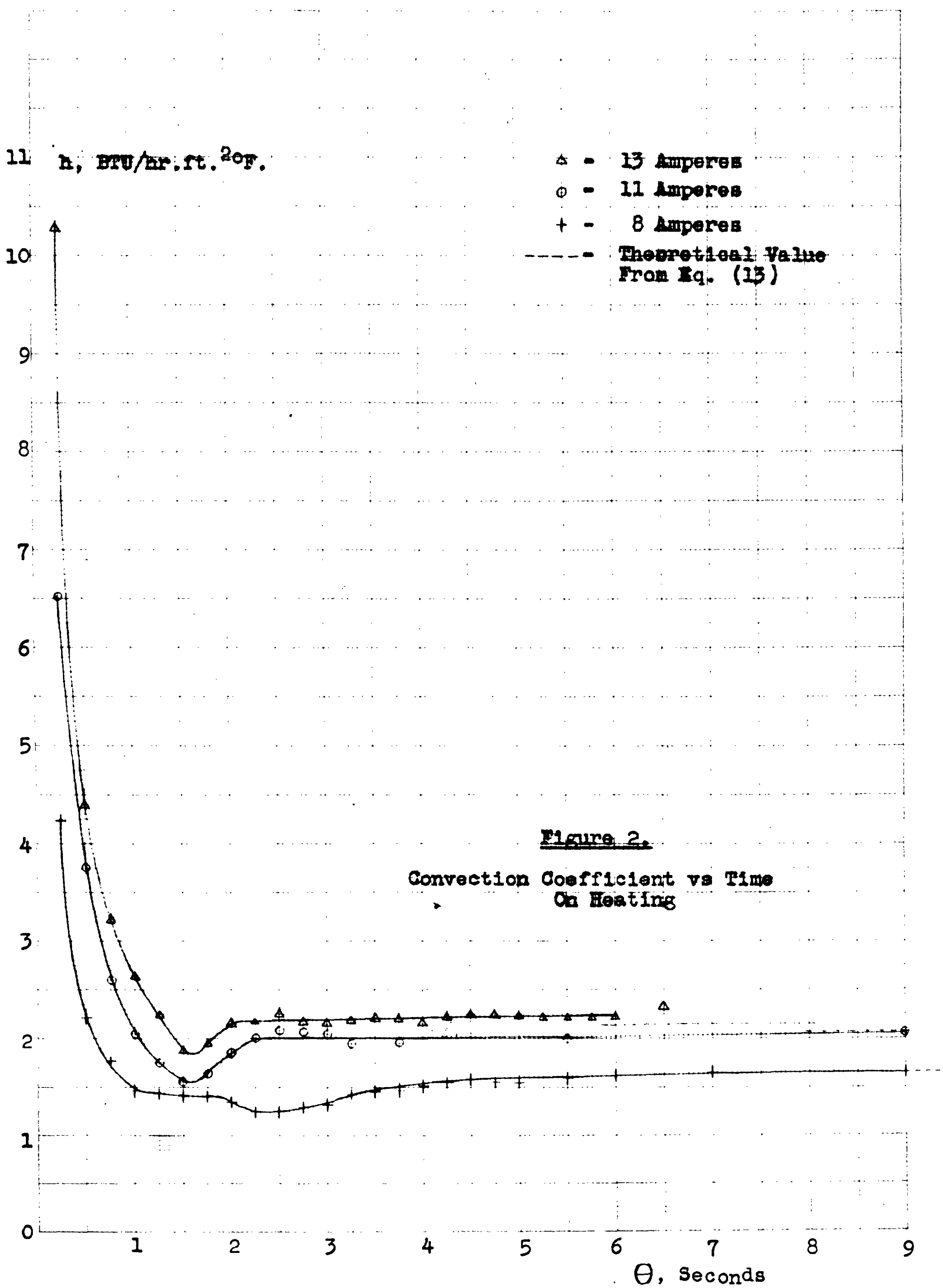


Figure 2.

Convection Coefficient vs Time
On Heating

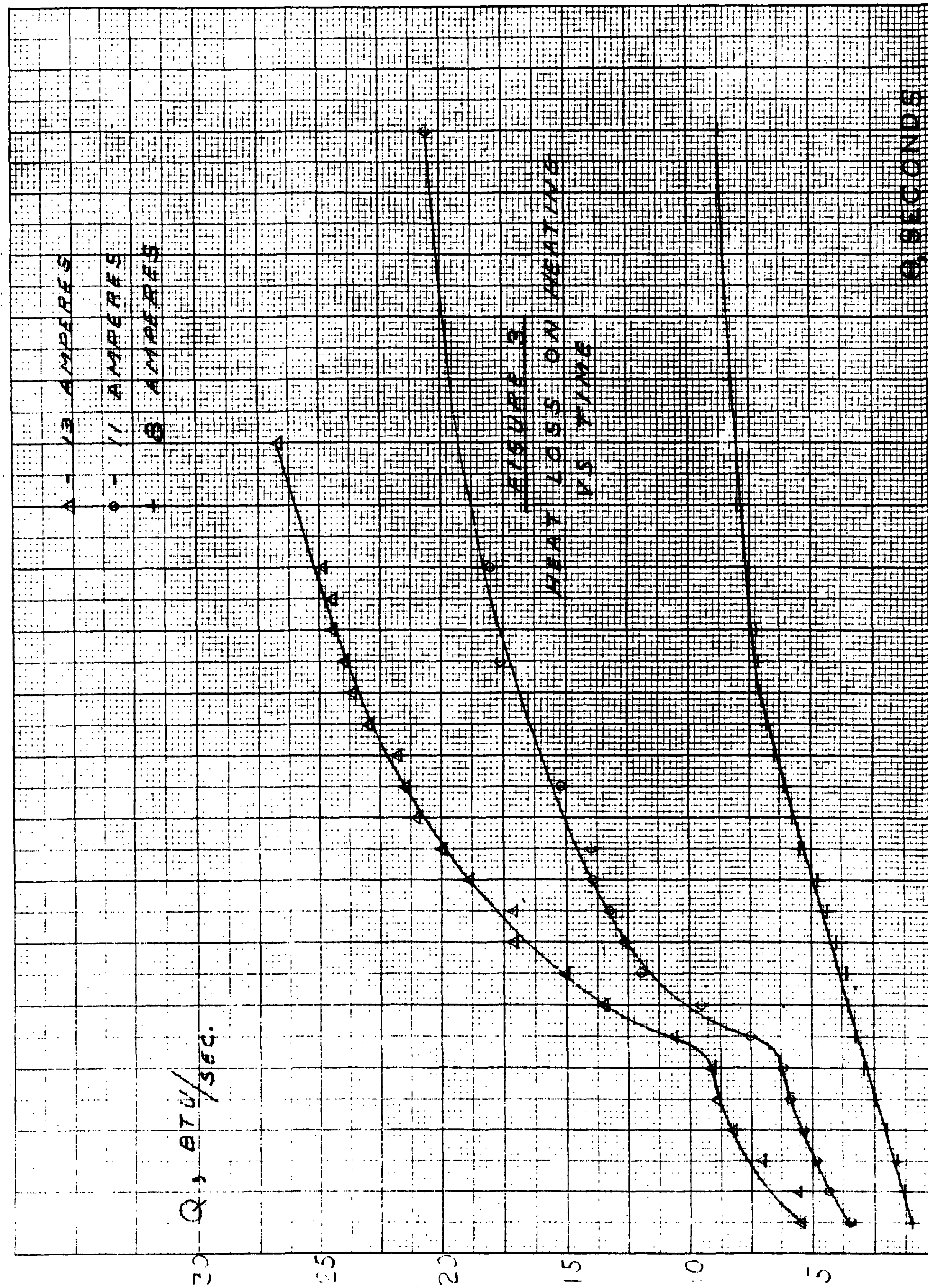
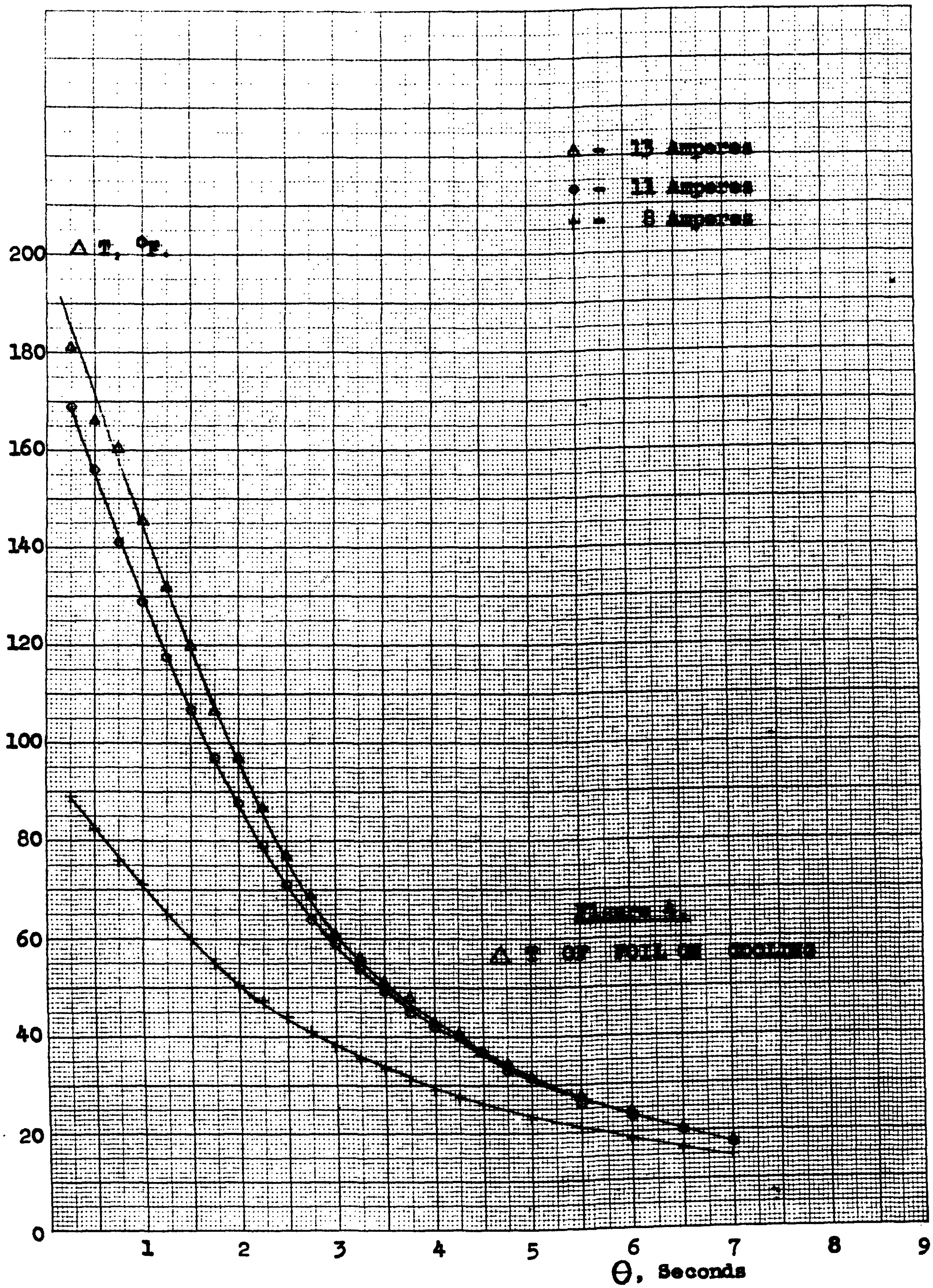


FIGURE 3

HEAT LOSS ON HEATING VS TIME



13 Amperes
11 Amperes
8 Amperes
ΔT, °F.
200
180
160
140
120
100
80
60
40
20
0
1
2
3
4
5
6
7
8
9
θ, Seconds
ΔT OF FOIL ON COOLING

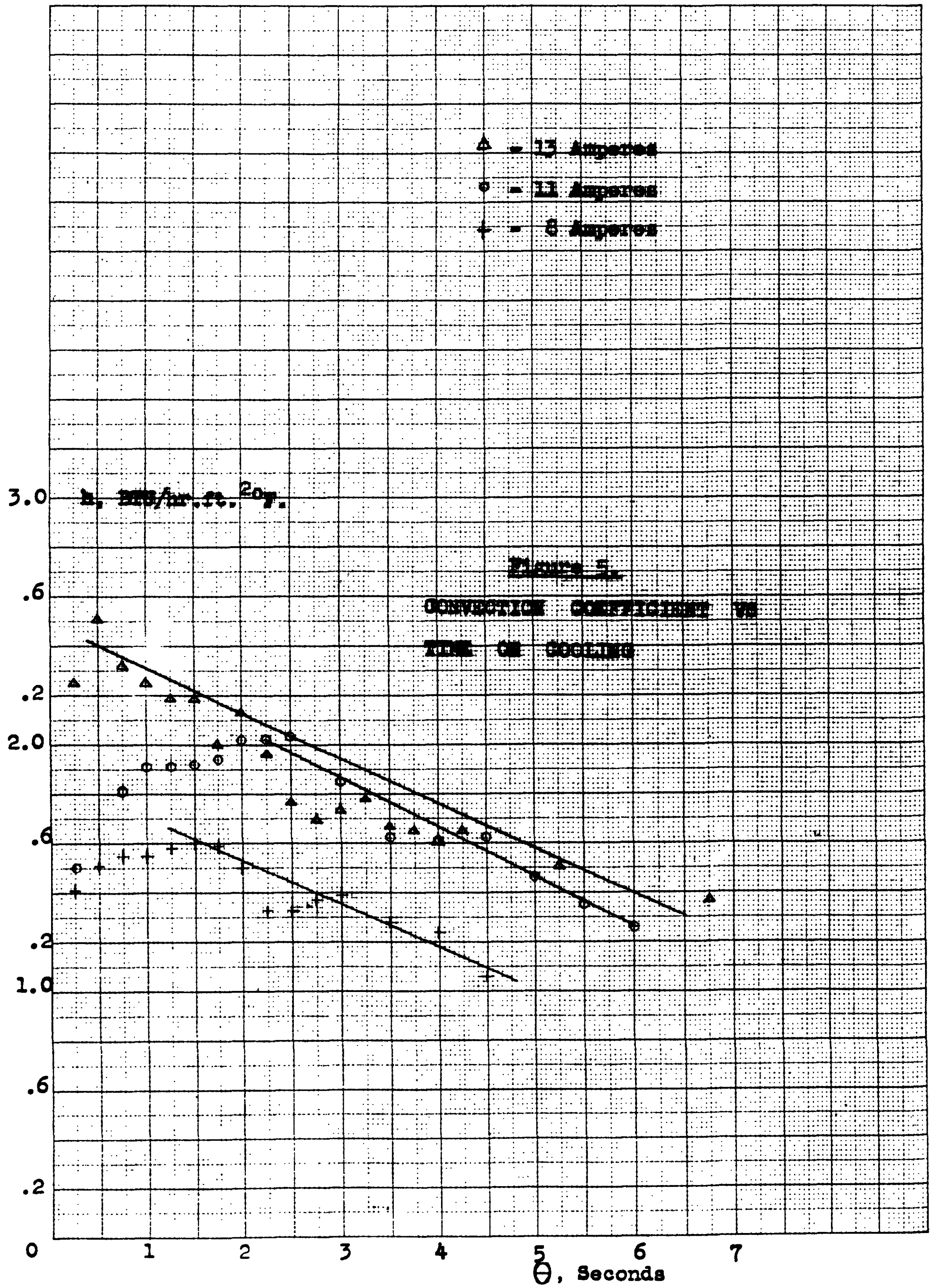


Figure 5.

CONVECTION COEFFICIENT VS TIME OF COOLING

V DISCUSSION OF RESULTS

The results agree very well qualitatively with that predicted from film theory. Equations (7) and (9) both predict that as Θ and l approach zero, h should approach infinity. This appears to be the case as Figure 2 illustrates. The slight dip in h below the steady state value at 1.5 seconds is particularly noteworthy. This can qualitatively be explained by assuming the formation of a film of warm air next to the surface of the platinum foil. As this film developes the heat transfer becomes one of conduction through a stagnant layer of air. After a certain time, however, the buoyancy of this warm layer of air becomes significant to cause it to rise leading to a chimney effect next to the surface. McAdams (4) on p. 168 gives interference photographs which show this chimney flow to be laminar. This chimney effect increases the mass velocity, G, next to the surface. If the value of h at the bottom of the dip is assumed to be some pseudo-steady state value resulting from conduction then as G increases equation (2) would predict h to also increase to the true steady state value with convection. This is observed to be the case.

Just when after the start of heating, the air film changes from a stagnant layer to laminar flow is not completely certain. It would be of considerable value to take interference photographs of the air film during the first few seconds of heating to determine at which time this transition takes place. Should this change occur at about 1.5 seconds, then the given explanation for the dip in the h vs θ curve would be substantiated.

If the foil was not held vertically there would be a different value of h for each side of the platinum foil. This would be caused by different patterns of air flow next to the foil for the top and bottom surfaces. The effect on h by varying the angle of the foil would be suitable for future study.

The gradual rise of h to steady state in the 11 and 13 ampere runs after the dip is due to the beginning of radiation effects. The h observed is composed of a convection term, h_c, and a radiation term, h_r. At the beginning of the 11 and 13 ampere runs and during the 8 ampere run, temperatures are low enough so that h_r may be neglected.

McAdams (4) recommends equation (13) for heat transfer by natural convection for a vertical plate in air if,

$$X = \frac{L^3 \rho_f^2 g \beta_f \Delta T}{\mu_f^2} \left(\frac{C_p \mu}{k}\right)_f = 10^4 \text{ to } 10^9 \tag{12}$$

$$h_c = .29 \left(\frac{\Delta T}{L}\right)^{.25} \tag{13}$$

In equation (13), L is the height of the foil, β is the coefficient of volumetric expansion, and μ is the viscosity of air at T_f, the temperature of the gas film.

Assuming the temperature of the film T_f, is the arithmetic average between the plate temperature and room, $\rho^2 g\beta Cp/\mu_f k$ can be read off from Figure 7-8 in McAdams where $\rho^2 g\beta Cp/\mu_f k$ is plotted vs T_f .

Equation (13) gives $h_c = 1.665$ BTU/Hrft² °F for the 8 ampere run. This compares favorably with the experimental value of 1.655 BTU/Hr ft² °F.

Similarly for the 11 and 13 ampere runs, h_c equals 1.96 and 2.025 BTU/Hr ft² °F respectively.

The h for the 11 and 13 ampere runs should be corrected for radiation effects. Walker,Lewis, McAdams, and Gilliland (7) gave the following relationship for h_r the coefficients of radiant-heat transfer,

$$h_r = \frac{.173\ p\ [(Ts/100)^4 - (Tr/100)^4]}{T_s - T_r} \tag{14}$$

McAdams (4) gives a value of p, the emissivity, equal to .054 - .104 for pure polished platinum plate. Using an average of .70, hr is equal to .13 and .117 for the 11 and 13 ampere runs respectively. The h_r for the 11 ampere run is higher than that for the 13 ampere run because the surface temperature of the 13 ampere run was a little lower due to a 40 F lower room temperature. Therefore the total h for the 11 and 13 ampere runs, which is the sum of hc and hr equals 2.09 and 2.15 BTU/Hr. ft.^{2} $^{\circ}$F respectively.

The value of h = 2.09 for the 11 ampere run agrees favorably with that obtained experimentally of 2.04 BTU/Hr. ft^2 $^{\circ}$F.

The value of h = 2.15 for the 13 ampere run is lower than that of 2.32 which was obtained experimentally. An explanation of this discrepancy can not be readily offered.

The change in h on cooling the foil cannot be readily correlated to any analytical expression. The coefficient, h, remains almost at the steady state value for about 1.5 seconds and then decreases. This might be explained again by film theory. Once this chimney effect has been established, there is a certain momentum to overcome in order to slow down this flow of air. As soon as the current is shut off, the temperature of the foil immediately starts to drop. However, the momentum of the air past the foil does not allow the mass velocity of the air to immediately decrease giving a lower h. After 1.5 seconds the mass velocity of the gas does start to decrease, giving a lower h as observed. The scattering of the data cannot be accounted for, except by assuming stray air currents to give an oscillatory mass velocity.

The source for the largest amount of error in this calculation of h, is in evaluating the slope $dT/d\theta$ from the plot of T vs θ. This uncertainty is not as critical as might be the case if T did not vary almost linearly for the first 1.5 seconds to yield a constant $dT/d\theta$ during that time when h is changing the most rapidly. After the first 1.5 seconds $dT/d\theta$ begins to change about 10 percent for each .25 seconds. During this time, h varies little from steady state.

The voltages from the thermocouples could be measured to $\pm$ .02 millivolts or $\pm$ .5°F. This error is not significant.

An alternate method of measuring the temperature would be by resistance thermometry. Since the resistivity of platinum has been very accurately measured as a function of temperature, resistance thermometry could be readily adapted to this sytem. This method would give some average temperature over the portion of foil under study. It would also eliminate the possibility of measuring hot spots with the thermocouple.

VI CONCLUSIONS

1. The heat transfer coefficient, h, rises to infinity as time θ approaches zero.
2. At the transition from a stagnant air film next to the surface to laminar flow, h dips below its steady state value.
3. After three seconds of heating, h is nearly at its steady state value, although the temperature of the foil does not reach steady state until seven seconds.
4. The values of h obtained at steady state,agree very favorably with equation (13) as given by McAdams.
5. On cooling of the foil, h decreases almost linearly after remaining a few seconds at the steady state value.

VII RECOMMENDATIONS

1. Interference photographs should be taken to determine when the transition from a stagnant layer to laminar flow takes place.
2. The temperature of the foil may be determined by resistance thermometry.
3. Measure h with the foil inclined at various angles rather than just vertically as done.

VIII APPENDIX

A. SUPPLEMENTARY DETAILS

The size of platinum foil used was 18 inches long, 1.0inch high and 0.0005 inches thick. The total weight of this piece of platinum foil was 4.709 gms. The distance between the platinum voltage taps was 38 mm. or 1.5 inches. This makes $w = .392$ gms. and $A = 3.00$ square inches.

The size of the copper and constantan wire was 0.002 inches in diameter. The size of the platinum wire was .010 inches in diameter. The thermocouples were soldered to the platinum. The soldering operation was a very delicate operation as a minimum of solder had to be used in order to keep the contact from acting as a heat sink.

The contact at the end of the platinum foil was two pieces of copper sheet 1.5 inches high and 0.5 inches wide. The copper wire from the current source was soldered to the two copper strips. The platinum foil was sandwiched between these two copper strips. The two copper strips were held together by two pieces of fiber board held by two screw clamps (see Illus. 1).

The current was determined by measuring the voltage across a standardized 0.001 ohm oil bath resistor.

The current through the foil was noticed to decrease slightly as steady state conditions were approached. This was due to the increased resistance in the cone heaters as they become warm. Direct current was used to help eliminate the possibility of 60 cycle pickup by the thermocouples. However, it was found necessary to shield the thermocouple wires by grounded aluminum foil. This eliminated most of the noise which was largely composed of 60 cycle line interference.

B. SUMMARY OF DATA AND CALCULATED VALUES.

TABLE B-1 HEATING AT 8 AMPERES

TIME (Sec)	T (^oF)	ΔT (^oF)	E (VOLTS)	I (AMPS)	POWER (WATTS)	C_p Cal/gm^oC	$dT/d\theta$ (^oC/Sec	h $BTU/H_r ft^2\ ^oF$	q_c BTU/Sec
0	91	0	0	0	0	-	-	-	-
1/4	95	.4	.109	7.90	.862	.0319	14.35	4.23	.98
1/2	101	10	.1115	7.94	.885	"	14.10	2.21	1.28
3/4	107	16	.1135	"	.902	"	13.75	1.77	1.64
1.0	114	23	.1150	"	.915	.0320	13.30	1.465	1.95
1/4	120	29	.1162	"	.923	"	12.52	1.445	2.43
1/2	125.5	34.5	.1175	7.90	.928	.0321	11.80	1.415	2.83
3/4	130	39	.1188	"	.938	"	11.30	1.415	3.19
2.0	135.5	44.5	.120	7.88	.947	.0322	10.90	1.345	3.47
1/4	140	49	.1205	7.86	.947	.0323	10.65	1.255	3.56
1/2	145	54	.1210	7.84	.950	"	9.93	1.255	3.94
3/4	149	58	"	"	"	"	9.15	1.29	4.32
3.0	152.5	61.5	"	7.82	.947	"	8.28	1.325	4.72
1/4	156.5	65.5	"	"	.950	"	7.02	1.43	5.42
1/2	159	68	.1215	7.80	.948	"	6.32	1.46	5.75
3/4	162	71	.1217	"	.950	.0324	5.77	1.47	6.06
4.0	164.5	73.5	.1220	"	.952	"	5.03	1.52	6.46
1/4	166.5	75.5	.1223	"	.953	"	4.37	1.55	6.80
1/2	168	77	.1225	"	.955	"	3.94	1.58	7.05
3/4	170	79	"	"	.955	"	3.79	1.55	7.12
5.0	171	80	"	7.78	.953	"	3.57	1.55	7.20
1/2	173.5	82.5	"	"	"	"	2.78	1.595	7.62
6.0	176.5	85.5	"	7.76	.950	"	1.85	1.625	8.05
7.0	178.5	87.5	.1200	7.70	.924	"	.885	1.64	8.32
9.0	182	91	"	7.64	.917	.0325	0	1.655	8.70

TABLE B-2 HEATING AT 11 AMPERES

TIME (SEC)	T (^oF)	ΔT (^oF)	E (VOLTS)	I (AMP)	POWER (WATTS)	C Cal/gm C	$dT/d\theta$ C/sec	h BTU/Hr ft^2 ^oF	q_c BTU/sec
0	95	0	0	0	0	-	-	-	-
1/4	104	9	.160	11.10	1.775	.0319	26.7	6.52	3.41
1/2	115	20	.169	"	1.878	"	"	3.76	4.35
3/4	127	32	.174	"	1.93	"	"	2.61	4.85
1.0	140	45	.179	"	1.982	.0320	"	2.05	5.35
1/4	152	57	.183	"	2.03	"	"	1.76	5.82
1/2	162	67	.185	"	2.055	.0321	26.4	1.565	6.17
3/4	174	79	.189	"	2.10	"	24.2	1.64	7.52
2.0	184	89	.191	"	2.12	.0322	20.7	1.84	9.50
1/4	193	98	.192	"	2.13	"	17.3	2.01	11.9
1/2	200	105	.194	11.05	2.15	.0323	15.15	2.08	12.65
3/4	206	111	.195	"	2.155	"	14.10	2.06	13.25
3.0	212	117	.196	11.00	"	"	12.85	2.05	13.90
1/4	219	124	.197	"	2.165	"	12.70	1.95	14.0
3/4	229	134	.199	10.95	2.18	.0324	12.05	1.96	15.2
4 3/4	243	148	.200	10.90	"	"	5.95	1.96	17.7
5 1/2	251	156	"	"	"	.0325	5.36	2.01	18.1
9.0	269	174	"	10.85	2.17	"	0	2.04	20.6

TABLE B-3 HEATING AT 13 AMPS

TIME (SEC)	T (^{o}F)	ΔT (^{o}F)	E (VOLTS)	I (AMPS)	POWER (WATTS)	dT/dθ ^{o}C/sec	C_g Cal/gm^{o}C	h BTU/Hrft^2 ^{o}F	q_c BTU/sec
0	59	0	0	0	-	-	-	-	
1/4	68	9	.1825	12.75	2.33	33.3	.0319	10.28	5.36
1/2	81.5	22.5	.1910	12.95	2.435	"	"	4.40	5.60
3/4	97	38.0	"	"	2.55	"	"	3.21	7.07
1.0	113	54.0	.2060	"	2.665	"	.0320	2.63	8.23
1/4	127	68	.2105	"	2.725	"	.0321	2.25	8.83
1/2	142	83	.2108	12.75	2.74	"	.0322	1.88	9.05
3/4	154.5	95.5	.2215	12.65	2.76	29.65	.0323	1.94	10.7
2.0	168	109	.2235	"	2.80	25.0	"	2.15	13.55
1/4	179.5	120.5	.2260	12.60	2.815	23.0	"	2.17	15.10
1/2	191.5	132.5	.2275	"	2.845	19.8	.0324	2.24	17.15
3/4	200	141	.2290	12.55	2.86	18.9	"	2.09	17.1
3.0	210	151	.2300	12.50	"	16.85	"	2.16	18.9
1/4	216.5	151.5	.2310	"	2.875	14.40	"	2.19	20.0
1/2	223	164	.2318	"	2.89	12.8	.0325	2.21	21.0
3/4	228	169	.2325	12.45	2.88	11.4	"	2.20	21.5
4.0	234	175	"	12.40	"	10.5	"	2.16	21.9
1/4	238	179	"	"	"	8.33	"	2.22	23.1
1/2	241	182	"	"	"	7.07	"	2.25	23.7
3/4	244	185	"	12.35	2.87	6.57	"	2.24	24.0
5.0	247	188	"	12.30	2.86	6.15	"	2.25	24.5
1/4	249.5	190.5	"	12.25	2.85	5.55	"	2.22	24.5
1/2	252	193	"	"	"	5.20	"	2.22	24.8
3/4	254	195	"	"	"	4.45	"	2.22	
6.0	256	197	"	"	"	3.45	"	2.23	
6. 1/2	258	199	"	"	"	0	"	2.32	26.7

TABLE B-4 COOLING OF FOIL I = 8A.

Time	T °F	ΔT °F	Cp cal/gm C	dT/dθ C/sec.	h
0	183.5	-	-	-	-
1/4	180	89	.0324	14.15	1.41
1/2	173.5	82.5	"	14.10	1.51
3/4	167	76	"	13.35	1.55
1.0	162	71	"	12.50	1.55
1/4	156	65	.0323	11.08	1.58
1/2	151	60	"	10.92	1.60
3/4	146	55	"	9.93	1.59
2.0	141.5	50.5	"	8.61	1.50
1/4	138.5	47.5	.0322	7.22	1.33
1/2	135	44	"	6.72	1.34
3/4	132	41	"	6.39	1.37
3.0	129	38	"	6.02	1.39
1/2	124.5	33.5	"	4.38	1.28
4.0	120	29	"	4.13	1.24
1/2	117	26	"	3.17	1.06
5.0	114	23	.0320	2.82	1.07
1/2	112	21	"	2.62	1.09
6.0	109.5	18.5	"	2.36	1.09

TABLE B-5 COOLING OF FOIL I = 11A.

TIME (SEC.)	T (^oF)	ΔT (^oF)	Cp Cal/gm C	dT/dθ ^oC/sec	h
0	269	-	-	-	-
1/4	264	169	.0325	28.6	1.50
1/2	251	156	"	"	1.70
3/4	236	141	.0324	"	1.81
1.0	224	129	"	27.8	1.91
1/4	212.5	117.5	.0323	25.5	1.91
1/2	202	107	"	23.3	1.92
3/4	192	97	.0322	21.3	1.94
2.0	182.5	87.5	"	20.2	2.02
1/4	174	79	.0321	18.3	"
1/2	166	71	"	16.6	2.03
3.0	154	59	.0320	10.8	1.85
1/2	144.5	49.5	"	9.13	1.62
4.0	137	42	"	7.78	1.61
1/2	131.5	36.5	.0319	6.78	1.62
5.0	126.5	31.5	"	5.30	1.46
1/2	121	26	"	4.05	1.35
6.0	118	23	"	3.33	1.26

TABLE B-6 COOLING OF FOIL I = 13A.

TIME (SEC)	T °F	ΔT °F	Cp Cal/gm C	dT/dΘ °C/sec	h
0	295	-	-	-	-
1/4	286	181	0325	4.5	2.20
1/2	271	166	"	"	2.39
3/4	255.5	160.5	"	36.5	2.32
1.0	240.5	145.5	"	31.5	2.25
1/4	227	132	.0324	27.8	2.19
1/2	215	120	"	23.6	2.19
3/4	201.5	106.5	0323	20.5	2.09
2.0	192	97	"	18.35	2.13
1/4	182	87	0322	15.15	1.97
1/2	172	77	"	12.65	1.79
3/4	164	69	"	11.0	1.70
3.0	156	61	.0321	10.3	1.74
1/4	151	56	"	9.5	1.78
1/2	146	51	.0320	8.5	1.67
3/4	143	48	"	7.41	1.65
4.0	138	43	.0319	6.67	1.60
1/4	135.5	40.5	"	6.38	1.65
5 1/4	124	29	"	4.12	1.51
5 3/4	120	25	"	3.23	1.37

C. SAMPLE CALCULATIONS

1. Experimental Heat Transfer Coefficient

Using equation (10), and using appropriate conversion factors:

$$hA(Ts-Tr) = EI(9.48 \times 10^{-4}) - 3.97 \times 10^{-3}\ wC_p \frac{dT_s}{d\theta} \qquad (15)$$

where $A = 3.00$ square inches

$w = .392$ gms.

Example: Heating of foil, $I = 13$ amps; $\theta = .25$ sec.

From Table B-3,

$$h(3.00)(68-59)=(.1825)(12.75)(9.48 \times 10^{-4})-3.97 \times 10^{-3}(.392)(.0319)(33.3)$$

$$h(27) = 22.05-16.5$$

$$h = .205 \text{ BTU/Hr. ft}^2\ {}^\circ\text{F}$$

$$= 10.28 \text{ BTU/Hr ft}^2\ {}^\circ\text{F}$$

2. Theoretical Heat Transfer Coefficient

Example: 13 ampere run,

$$T_f = 159^\circ F = 619^\circ R$$

$$T_s = 258^\circ F = 718^\circ R$$

$$T_r = 59^\circ F = 519^\circ R$$

From Figure 7-8 in McAdams,

$$X = .81 \times 10^6(199)(\frac{1}{12})^3 = 9.32 \times 10^4$$

Substituting into equation (13),

$$h_c = .29(199 \times 12)^{0.25} = 2.025 \text{ BTU/Hr,ft}^2, {}^\circ F$$

In order to obtain h_r, equation (14) will be used. Substituting into equation (14),

$$h_r = \frac{.173(.07)[(7.18)^4-(5.19)^4]}{718-519}$$

$$= .117 \text{ BTU/Hr ft}^2 \ {}^\circ F$$

Therefore $h = h_c + h_r = 2.025 + .117 = 2.15 \dfrac{\text{BTU}}{\text{Hr ft}^2 \ {}^\circ F}$

D. NOMENCLATURE

A	surface area of section, cm.^{2}
Cp	specific heat at constant pressure, BTU/lb. ^{o}F
E	voltage across section, volts
f	friction factor in Fanning equation
g	acceleration due to gravity, 4.17×10^8 ft/hr^2
G	mass velocity, lb./hr. ft^2 of cross-section of fluid.
h	overall heat transfer coefficient, $h = h_c + h_r$, BTU/hr. ft^{2}^{o}F
I	current through foil, amperes
k	thermal conductivity, BTU/hr. ft. ^{o}F
l	thickness of film, ft.
L	height of foil, ft., L = 1/12 ft.
p	emissivity of surface for radiation
q_c	heat lost by convection per unit time, BTU/sec.
T_f	temperature of gas film, $T_f = \frac{T_s + T_r}{2}$, ^{o}F
T_r	temperature of room, ^{o}F
T_s	temperature of platinum surface ^{o}F
ΔT	$T_s - T_r$
$\frac{dT_s}{d\theta}$	rate of change of surface temperature, ^{o}C/sec
(Pr)	Prandtl number, equals $\frac{C_p \mu}{k}$.
w	weight of section, gms.

x	perpendicular distnace in air from platinum foil surface, cm
α^2	thermal diffusivity, $ft^2/hr.$
β	coefficient of volumetric expansion, $(^oF)^{-1}$
θ	time, seconds
μ_f	viscosity of air at T_f, ob./hr. ft.
ρ	density, lb/ft^3

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