1989

Heat recovery from flue gasses using finned tubes /

Hart K. Lichtenwalner
Lehigh University

Heat Recovery from Flue Gases Using Finned Tubes

By

HART K. LICHTENWALNER

Heat Recovery from Flue Gases

Using Finned Tubes

by Hart K. Lichtenwalner

Lehigh University
Bethlehem, Pennsylvania
October, 1949

This thesis is presented to the Faculty of Lehigh University in partial fulfillment of the requirements for the degree of Master of Science in Chemical Engineering.

Hart K. Lichtenwalner

Hart K. Lichtenwalner

Acknowledgement

This project was sponsored by David F. Kennedy, Incorporated, manufacturers of Tilco-Fin tubing, and was undertaken under the direction of Dr. Darrel E. Mack. The author extends his thanks to Dr. Mack for his valuable suggestions and his assistance in overcoming technical difficulties.

Thanks are due also to Messrs. Lynn and Clarke for their contributions to the construction of the apparatus, and to Messrs. Foust and Boyer for their aid in assembling the apparatus and in obtaining the data.

Index

Summary

In order to investigate the characteristics of heat transfer from gases at high temperature to water-cooled tubes with extended external surfaces, a combustion chamber embodying a single-row bank of three-inch tubes with one-inch serrated helical fins was constructed. Tests using gas temperatures up to 700° F. and Reynolds numbers from 1700 to 2800 indicated values of gas film coefficients in good agreement with the expression:

$$\left(\frac{h_g}{cG}\right)\left(\frac{c\mu_f}{k_f}\right)^{2/3} = 0.935\left(\frac{DG_m}{\mu_f}\right)^{-0.525}$$

This relation has been obtained (7) from studies at lower gas temperatures using similar tubes of smaller diameter.

At the gas temperatures involved, no appreciable radiative heat transmission was observed.

Direct determination of gas film coefficients from measured gas-wall temperature differences was found to be generally the simplest and most reliable of several methods investigated.

Meaning of symbols used

The symbols used in this report, largely in the legends of the figures and in the sample computations, have the following significance:

A	area in square feet; sub f, of the fins, neglecting edge area; sub b, of the base; sub T, total; sub i, internal
a	the square root of the quantity $2h_g/kT$
c	heat capacity at constant pressure, Btu per pound per degree Fahrenheit
D	external base diameter of the finned tubes, feet
G	mass velocity, pounds per square foot per hour; sub m, at the minimum cross-section
h	film coefficient, Btu per square foot per hour per degree Fahrenheit; sub g, of the gas film; sub w, of the water film
k	thermal conductivity, Btu per square foot per hour per unit temperature gradient, degrees Fahrenheit per foot; sub f, at the film temperature
Q	rate of heat transfer, Btu per hour; sub f, in the finned tube bank; sub s, in the shielding bank
T	fin thickness, feet
t	temperature, degrees Fahrenheit; sub f, of the film
Δt	temperature difference, degrees Fahrenheit; sub m, logarithmic mean temperature difference
U	overall heat transfer coefficient based on gas-side area, Btu per square foot per hour per degree Fahrenheit; sub f, of the finned bank; sub s, of the shielding bank
w'	length of fin from base to tip plus one-half the fin thickness, feet
x	distance from fin base to any point on the fin, feet

ϵ' fin effectiveness, the ratio of the amount of heat transferred from fin to gas to the amount transferred by an equal area of surface all at the same temperature of the fin base

ϵ overall effectiveness of the finned tube including both fin and base areas

μ viscosity, pounds per foot per hour; sub f, at the film temperature

θ gas-metal temperature difference, degrees Fahrenheit; sub b, at the base wall; sub x, at some point on the fin

Nu Nusselt number, hD/k

Pr Prandtl number, $C\mu/k$

Re Reynolds number, $Du\rho/\mu$

St Stanton number, h/cG

Introduction

The heat transfer characteristics of Tilco-fin tubing in air conditioning service have been the subject of a series of investigations carried out at Lehigh University. The results have been recently summarized by McDonnell (6). This tubing, manufactured by David E. Kennedy, Inc., is of the transverse serrated-fin type, suitable for gas-liquid or gas-steam heat transfer.

The present phase of the work constitutes an extension of these studies to higher gas temperatures, up to about 750° F. The tubing used is also considerably larger in size, having a three-inch base diameter with one-inch fins. Tubing of this type would be suitable for industrial use in economizers, pipe stills and high-temperature heat recovery apparatus.

The apparatus and experimental techniques described in this report are intended principally for investigations at still higher gas temperatures, such as might be encountered in the industrial applications mentioned above. The data presented here, however, will provide a link between high and low temperature service and should permit prediction of performance at moderately high gas temperatures.

Theoretical Background

The convective transfer of heat to finned surfaces in contact with fluids has been treated from a theoretical standpoint by a number of authors (1, 2, 3), the earliest, and probably simplest, treatment being that of Harper and Brown (2). The analysis involves setting up the differential equation for heat flow in a fin on the basis of certain assumptions, and determining the amount of heat transferred to the fin in terms of the amount which would be transferred to the same area of base tube. The ratio so obtained is called the fin effectiveness. In the Harper and Brown development, the assumption open to the greatest objection is that the film coefficient is uniform over the fin surface. (In applying the method to finned tubes, the same film coefficient is assumed as applying to the exposed base tube surface). Since the velocity of the fluid relative to the surface in transverse flow will not be uniform, the assumption is invalid. The final Harper and Brown relationship is:

$$\epsilon' = \tanh aw'/aw'$$

where ϵ' = fin effectiveness, the ratio of the heat transferred to or from the fin to the heat transferred to or from an equal area of surface all at the temperature of the fin base

w' = corrected fin width, the distance from base to tip of fin plus one-half the fin thickness

a = $\sqrt{2h/kt}$, with the dimension length $^{-1}$

h = film coefficient of convective transfer to or from the surface

k = thermal conductivity of material of fin

T = fin thickness

After obtaining the fin effectiveness, the overall effectiveness of the finned tube assembly can be computed from:

$$\epsilon = \frac{A_{base}\ (1) + \epsilon' \ A_{fin}}{A_{total}}$$

where the A's represent areas.

There are some ambiguities in the application of these equations to analysis of heat transfer data for finned tubes. The equation for fin effectiveness was developed for infinitely long fins, in which the contribution to heat transfer of the lateral edges (but not the tip edge) would be negligible. In serrated finned tubes, this area may be considerable. The arbitrary distinction is made that in computing heat transfer coefficients directly from experimental data, this edge area is included in the total, but in the calculation of fin effectiveness, the fin and total area do not include the lateral fin edges.

A more important factor is the choice of the proper value to use for the metal (fin and wall) resistance to the flow of heat when correcting overall coefficients for resistance of the wall and the fluid film inside the tubes. It will be shown in another paper that the metal resistance depends in this case on the outside film coefficient and is hence variable. In this paper, the necessity of computing the metal wall resistance has been overcome by measuring directly the gas-metal temperature difference on the finned side.

Description of Apparatus

In order to measure heat-transfer characteristics at high gas temperatures, the finned tubing was incorporated as a tube bank in a combustion chamber. The combustion chamber consisted essentially of a shell of sheet steel about eight feet high and having a cross section twenty-five inches square. For versatility and ease of assembly, the chamber was built up out of shorter box-like sections, in which the tube banks and other apparatus were installed. Schematic diagrams of the combustion chamber assembly are shown in Figures 1 and 2. For the tests described in this paper, the chamber was not insulated.

The flue gases were supplied by a bank of six gas burners, situated in the lowest section of the chamber. This section was equipped with eight sliding dampers, four above and four below the burners, for the admission of secondary air. After generation, the hot gases passed upward through the chamber, eventually reaching a reducing section leading to an eight-inch stack. From this stack, the gases passed to a second stack equipped with a blower, so that a certain amount of forced draft could be applied to the chamber by aspiration.

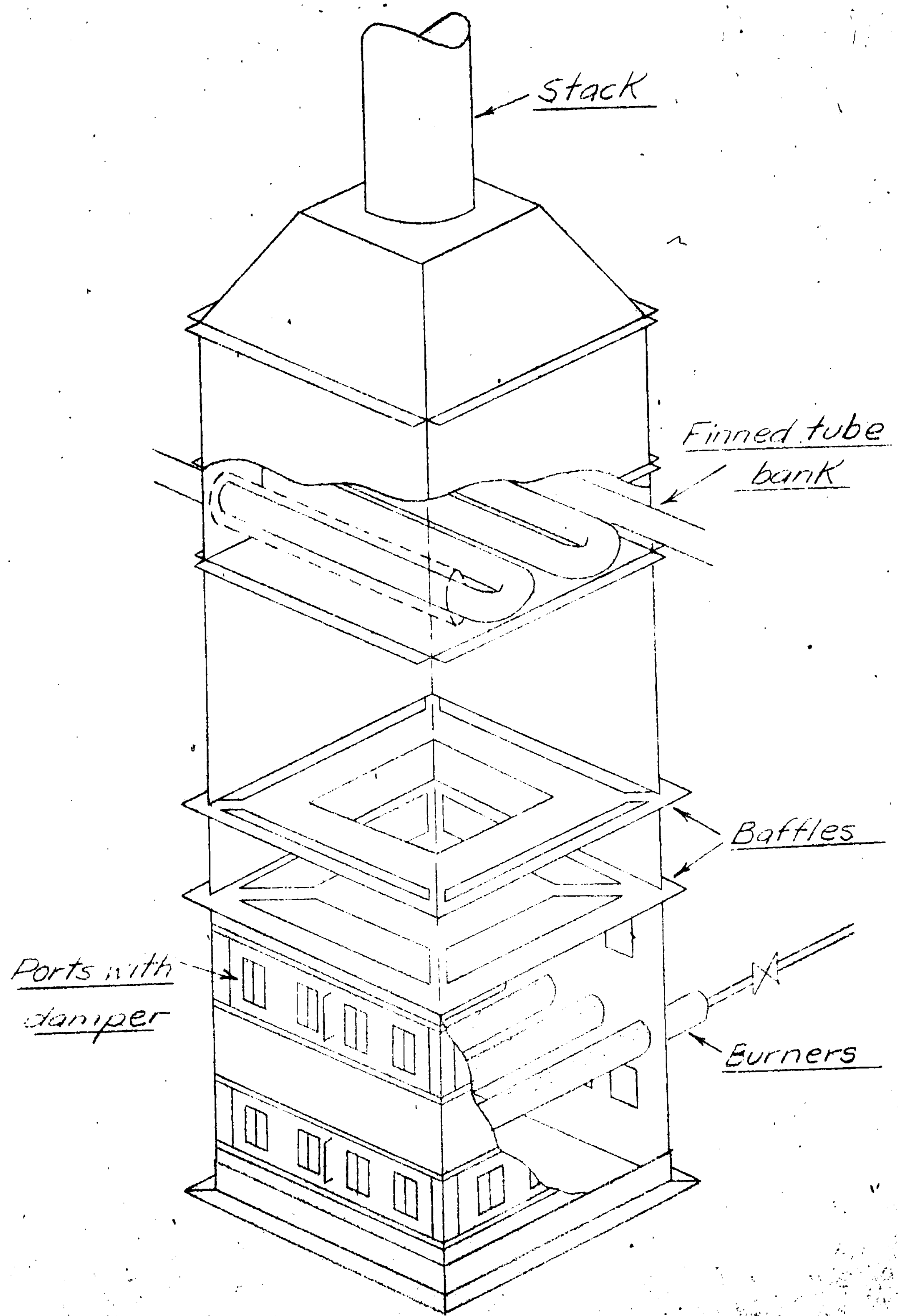
The tube bank consisted of a single row of five finned tubes connected in series by means of return bands. The chamber and tube sections were so made that the fins reached all the way to the walls and to the fins of the adjacent tube. There was no free space and no overlapping of fins. The physical characteristics of the finned tubing are presented in Table I.

Water from the city mains was passed through the bank, the rate of flow being regulated by means of a valve in the outlet line. The line carrying the effluent to the drain was raised above the level of the tube bank, and a vent cock was installed on the top side of one of the finned tubes, so that the tube bank was always sure to be running full, any trapped air being removed through the vent cock. The finned tube bank assembly was installed near the top of the chamber, just below the section leading to the stack.

For the preliminary performance tests, two stainless steel baffles were provided which could be installed between the flame and the finned tubes. These were simply cut-out sheets, one the complement of the other, intended to act as a shield against heat transfer to the tubes by direct radiation. For later runs, a three-row shielding tube bank was installed in order to conform more nearly with actual industrial installation practice. The individual tubes, of one-inch pipe, were connected in series, and water was passed through the lowest, middle and top bank in that order. The physical measurements of the shielding bank are given in Table II. The water rate through this bank was independently controlled by means of a valve in the outlet line.

Figure 1

Preliminary form of combustion chamber



Ch. E. Lab., Lehigh University

Figure 2

Revised form of combustion chamber

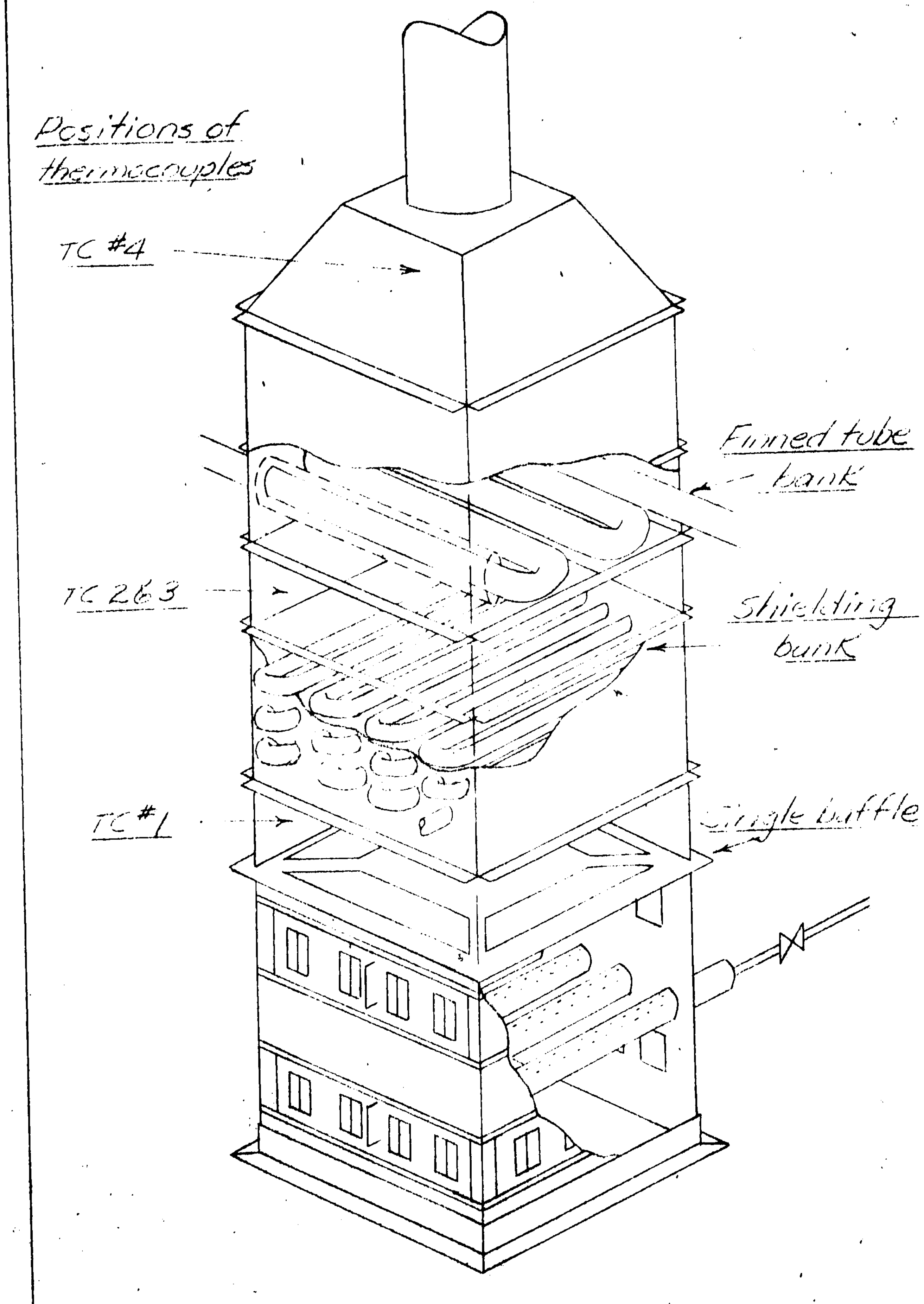


Table I

Physical characteristics of finned tube bank

Base tube:	
Outside diameter	3.00 inches
Inside diameter	2.74 inches
Material	Steel
Fins:	
Type	Serrated transverse helical
Length from base to tip	0.963 inch
Thickness	0.0375 inch
Width	0.344 inch
Pitch	45 rows per foot
Material	Steel
Method of attaching	Base bent and welded
Bank:	
Number of rows	1
Number of tubes per row	5
Finned length per tube	25 inches
Center-to-center distance	5 inches
Clearance of fin tips at wall and between tubes	0
Heat transfer areas:	
Total	75.5 sq. ft.
Base	7.2 sq. ft.
Fins	68.3 sq. ft.
Fins neglecting edges	61.7 sq. ft.

Table II

Physical characteristics of shielding bank

Components	1-inch standard pipe
Number of rows	3
Number of tubes:	
Lowest row	10
Middle	10
Top	9
Center-to-center distance in each row	$2\frac{1}{2}$ inches
Row arrangement	Top and middle rows in line, bottom row staggered
Center-to-center distance between rows:	
Top-middle	1 7/8 inches
Middle-bottom	2 1/8 inches
Heat transfer area	20.8 sq. ft.

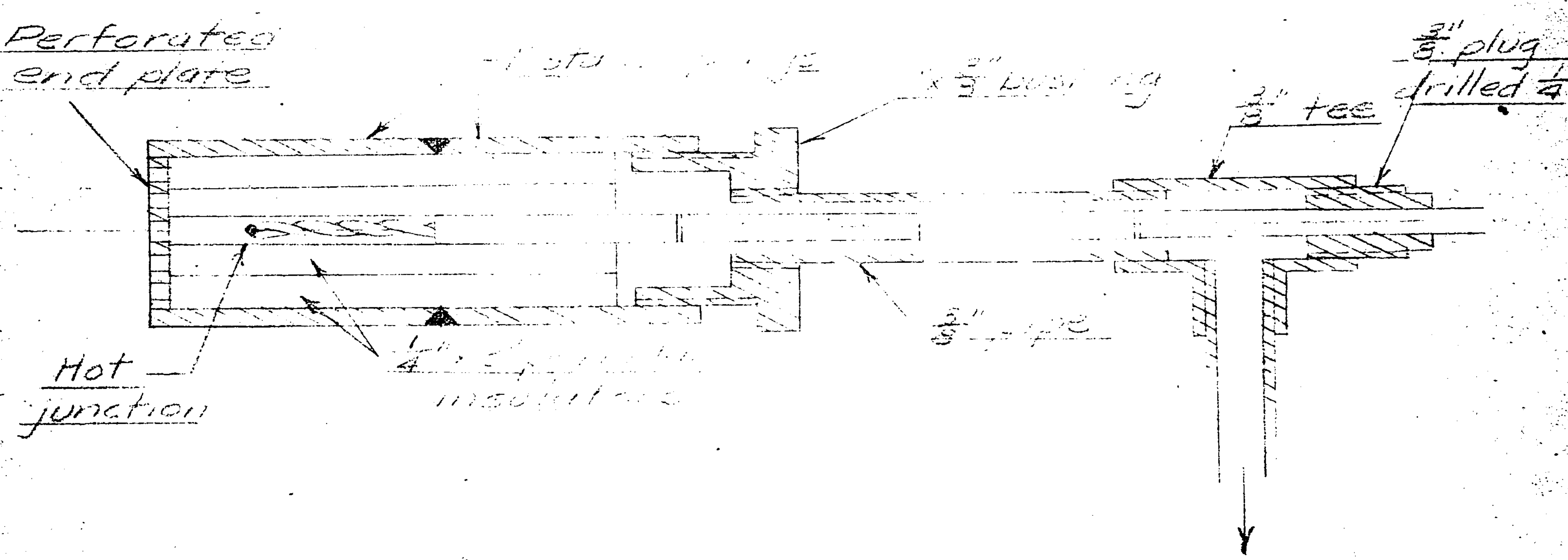
Instrumentation

For the preliminary tests, gas temperatures were read by means of a portable bare-wire thermocouple and an Alnor "Pyrocon". This instrument is a multirange millivoltmeter calibrated to read directly in degrees Fahrenheit to the nearest ten degrees. Gas velocities were read in the stack by means of an Alnor "Velometer", a vane-deflection type of instrument reading linear velocity in feet per minute in twenty-feet-per-minute divisions. A suitable attachment for obtaining velocities in a duct was used with this instrument. The inlet and outlet water temperatures in the finned tube bank were read by means of thermometers with two-degree divisions, readable to one-half degree. For measuring water rate in this bank, a calibrated orifice in the inlet line was utilized.

In the improved form of the apparatus, gas temperatures were measured by means of high-velocity thermocouples. These were of the type shown in Figure 3, and were permanently installed at four points in the combustion chamber: one below the shielding bank, two between the shielding bank and the finned tube bank, and one above the finned tube bank at the entrance to the stack. The location of these thermocouples, notably those between the two tube banks, were studied carefully in order to insure reliable temperature indications. The shielding bank, which was originally placed only a few inches below the finned tube bank, had to be moved downward in order to give the gases rising from this bank an opportunity to mix before entering the high-velocity thermocouples. Temperature traverses immediately above the shielding bank showed marked temperature variation with position, depending on whether the thermocouple was placed directly over or between tubes in the top row of the shielding bank. The gas drawn through the high-velocity thermocouples passed to a common manifold and thence to a small motor-driven fan and the atmosphere. Thermocouple voltages were read by means of a potentiometer with a 60-millivolt range. In addition, fine iron-constantan thermocouples were silver-soldered to the tube wall and across adjacent fin tips on both the lower and upper side of the middle tube of the finned tube bank. As before, the gas velocities were read by means of a Velometer, but the flow rates so determined were used only as a check on the rates determined from heat balances. The water thermometers were replaced by precision thermometers graduated in 0.2° F. divisions, in order to insure precise measurement of heat flow. When the shielding bank was in use, its water outlet temperature was read by means of a thermometer with two-degree divisions. Pressure in this bank was indicated by means of a Bourdon-type gauge, and the flow rate was measured by collecting the effluent over a measured period of time.

Figure 3

Shielded high-velocity thermocouple construction



Elements: No. 8 iron-constantan

Experimental Procedure

In starting up the combustion chamber, water was first introduced into the tube banks, the burners were lighted, stack and thermocouple blowers were started (if used), and gas and water flow rates were adjusted. A period of twenty minutes to one-half hour was usually required to reach a steady state. During this period air was vented from the finned tube bank and the orifice manometer connections, and the sliding dampers were adjusted. In runs with the revised instrumentation, the dampers were regulated so that the two thermocouples below the finned tube bank had nearly identical readings. No attempt was made to regulate the gas flow rate, burners being either turned on full or off.

After reaching equilibrium, instrument readings were taken at ten-minute intervals for forty minutes, making a total of five sets of readings for each run. Two or three determinations of gas velocity with the Velometer, and of water rate through the shielding bank, were made during the run. In the preliminary runs, it was necessary to insert the bare-wire thermocouple through ports at various points in the combustion chamber and allow time for the Pyrocon to reach a steady temperature. Since the thermocouple was not fixed, there was a possibility of measuring the temperature at different points on successive readings. Care was taken to minimize any such variation. In all cases, the various instrument readings were taken in the same order so as to maintain equal increments of time between recorded values.

Precision of Data as Read

In the improved form of the apparatus, water temperatures in the finned tube bank could be read to the nearest 0.1° F. The outlet temperature fluctuated continuously over a range of 0.3 to 0.5°, so that a "mental integration" process based on observations over half a minute or more was necessary to ascertain the true value. The duplicability of these readings is felt to offer support for their reliability. Water flow rates could be obtained from manometer readings with a precision of one per cent.

Gas temperatures showed some variability, but no serious fluctuations. At times there occurred a rise in temperature as indicated by one of the thermocouples situated below the finned tube bank, while at the same time there was a corresponding decrease in the reading of the other. These may have been due to disturbances in the gas flow pattern at the low velocities used. In any case the variations were always below twenty degrees and usually much smaller.

Some variability was noted in the readings of thermocouples attached directly to the finned tubes, especially that indicating the tube wall temperature of the under side. This variation was equivalent to only a few degrees, and an attempt was made to obtain an average reading in recording these values. The potentiometer galvanometer showed a definite deflection for a difference of 0.05 millivolts in slide-wire setting from the balance points. This difference corresponds to less than 2° F., and it is felt that most of the readings were less in error than this.

Auxiliary readings, such as Velometer readings, were subject to greater uncertainty. Velometer readings, for example, depended on the position in the stack of the duct tube; considerable variation, from reading to reading, was noted in stack velocities. Water flow rate in the shielding bank, found by collecting the effluent over a period of time and weighing it, is probably valid only to plus or minus five per cent because the quantity was relatively small.

The overall operation of the equipment was influenced to a certain extent by environmental conditions. Fluctuations in water pressure, gas pressure, and room temperature and air movement affected steady state conditions and caused the discarding of a considerable amount of data. No run was considered acceptable unless marked variations in conditions were absent, and no continuous trends in any readings were observed.

Calibration of Instruments

All instrumentation used was checked or calibrated. The thermometers in the tube bank were compared with a Bureau of Standing calorimeter thermometer at the same immersion depth as was used in installation. The thermometer in the shielding bank was also compared with a precision Centigrade thermometer. All thermocouples were tested by comparison with a mercury thermometer immersed in a glycerol bath. The potentiometer was equipped with a standard cell and cold junction corrections were applied. The Velometer had been checked by the manufacturer and was certified accurate within three per cent of full scale (30 feet per minute). These data are included in the Appendix, Tables X, XI, and XII. Figure 11 shows the calibration curve for the orifice in the finned tube circuit.

The most critical measurements from the standpoint of technique are the high gas temperatures. The multiple-shield high-velocity thermocouples will register a lower reading than true gas temperature if the velocity of the gas past the thermocouple is insufficient (9, 10). Although the most suitable blower available was used for drawing air over these thermocouples, it was felt advisable to attach the blower to each thermocouple separately under operating conditions to determine whether any increase in indicated temperature would be observed. This was done a number of times, with variable results, summarized in Table IV. It appears definite that thermocouples No. 3 and 4 require no correction for insufficient gas velocity. The maximum correction for thermocouples No. 1 and 2 would be 15 and 20° F. respectively. In using these readings, no corrections were applied for insufficient gas velocity, for reasons given in the Discussion of Results.

Discussion of Results

Treatment of Data

From averaged values of observed data, presented in Table VII the overall heat transfer coefficient was calculated for each run. This was then corrected for water film and metal wall (but not fin) resistance, to provide an observed gas-side film coefficient. The fin effectiveness for the situation was then determined from a nomograph of the Harper and Brown relationship, due to Mack and Pitcher (8). This nomograph does not take into account the correction for curvature in helical fins given by Harper and Brown (2). The correction, however, is small and uniform in the range of film coefficients occurring here, and was neglected. The fin effectiveness so obtained was then applied to determine the actual fin-side film coefficient. Averaged observed data and calculated results for the preliminary runs are presented in Table VI in the Appendix.

For the runs with the refined instrumentation, several improvements in treating the data were possible. Because of the uncertainty in the values for water film and metal resistances, the fin-side coefficient was also computed directly from the measured gas-base wall temperature difference. This value was then expressed as the Nusselt number or as the Stanton number multiplied by the Prandtl number raised to the two-thirds power, using the appropriate values for air for the gas film properties. In order to determine the Reynolds number, a heat balance across the finned tube bank was made, and the gas flow rate was computed. Plots of Nusselt and Stanton-Prandtl numbers versus Reynolds number were made to correlate the data. Computed values for the second set of runs are given in Table III. A sample computation is presented in the Appendix.

Results of Correlation

The results of the preliminary tests when plotted as Stanton-Prandtl numbers versus Reynolds number showed a rather diffuse point field in the general vicinity of McDonnell's correlation, as shown in Figure 4. These results showed the desirability of increasing the range of available Reynolds numbers, but gave qualitative support to the possibility of extending McDonnell's correlation to higher temperatures. No radiation effects were evident; however, it was possible that the metal baffles used were ineffective as radiation shields, being themselves at high temperature. A striking observation was that the temperature of the gas stream was reduced as much as 400° F. in a single passage over a five-inch depth of fins.

The tests on the revised apparatus led to excellent agreement with previous work over the entire range of temperature and Reynolds number available, as shown in Figures 5 and 6. These results make use of a fin-side film coefficient determined directly from gas-base wall temperature difference. The gas film coefficients computed by correcting the observed overall coefficient for water film and tube wall resistance were somewhat lower. The relation between these computed film coefficients and the directly determined values is shown in Table V. By taking into account the resistance of the fins as well as the wall, better agreement between the two sets of values is obtained (Table V).

Again there was no observable increase over the expected value of film coefficient when the shielding was partially removed by eliminating the flow of water in the shielding bank. As the maximum gas temperatures attained was only 720° F., radiation effects might be expected to make relatively small contributions to heat transfer.

The effect of a possible error in measuring the gas temperature approaching the finned tube bank was investigated for two cases. In order to ascertain the maximum effect of such a change, the calculations were repeated using the observed approach temperature plus 20° F. This was the maximum possible correction to one of the thermocouples indicating this temperature, the other requiring no correction. (Table IV) No correction was required in the temperature of the gas leaving the finned bank. The results of these calculations is shown in Figure 7. Since the gas flow rate was determined from a heat balance over the finned bank, a change in both Reynolds number and Stanton-Prandtl number occurred, the net effect being a shift toward lower Reynolds number, but still in reasonable agreement with McDonnell's correlation. The low-temperature run used as one example would represent an extreme case, since the correction would be expected to be considerably smaller than twenty degrees as lower temperatures. The use of the properties of air for the gas stream is felt to be a reasonable simplification for the dilute flue gases involved. Furthermore, it is indicated by the lack of reliable information on gas properties in general. In order to have a reliable, consistent set of values for air, the tables given by Keenan and Kaye (4) were used, although these yield slightly lower Prandtl number values than were used by McDonnell, starting at 0.72 and decreasing somewhat with increasing temperature. Correction for the presence of carbon dioxide and water vapor would increase this value slightly (5).

By the end of the preliminary series of runs, the entire exterior surface of the finned tubes had become covered with a thin, flaky rust layer. Since no accurate measurements had been made before its formation, the effect of this layer on the heat transfer could not be evaluated. Its resistance was, however, undoubtedly very small in comparison with that of the gas film, and this effect may have been partially offset by the increased turbulence provided by the roughened surface.

A possible point of difference between the data obtained here and McDonnell's correlation is that the latter was based largely on banks of several rows of tubes; these might be expected to show improved turbulence and hence improved heat transfer, by analogy with bare tubes. The data used by McDonnell do, nevertheless, include some single-row results which are not distinguishable from the multiple-row figures. The turbulence does not therefore appear to be improved in multiple-row banks when using finned tubes.

The basic assumption of uniform fin-side film coefficient used by Harper and Brown could be readily tested in these runs by means of their expression relating gas-fin temperature difference at any point to the gas-base wall temperature difference. From measurement of fin-tip and wall temperatures above and below, it was found that the lower side, facing the approaching hot gas, had film coefficients on the order of 3 to

5 Btu per square foot per hour per degree Fahrenheit, while the top, facing the receding cooled gas had coefficients of one or less. These values were difficult to determine precisely because of the small change in hyperbolic cosine of x with x in this region. Neither the arithmetic or logarithmic mean of these values showed good general agreement with the experimental value of the film coefficient, as indicated in Table V.

In view of these considerations, it might be held that the agreement between the present data and McDonnell's correlation is to some extent fortuitous. Nevertheless, it appears that the application of the fin effectiveness concept to an arrangement of the type studied here does yield consistent results, which, in the light of the support lent by determinations at lower gas temperatures, make possible the application of this correlation to design problems involving gases of moderately high temperatures.

Table III

Calculated values of quantities for runs on revised apparatus

Run	Conditions	Q_f	Q_s	Air mass rate computed from: Velometer reading Upper	Lower	Q_f as basis for air rate, lb/hr	Q_s	G_m	U_s	U_f	Water-wall Δt, °F.	t_f water °F.
1	6	42100	49200	560	610	607	580	422	3.87	1.64	51	96°
2	4	32800	34600	540	600	591	520	410	3.41	1.55	41	91
3	2	18200	17800	430	490	498	540	346	3.00	1.40	22	79
4	6 Bl.	42400	42000	830	950	860	1070	597	4.48	1.88	51	97
5	6 Bl. ports open	41500	38800	940	1060	876	870	609	4.10	1.89	50	96
6	4 Bl.	31700	28600	850	930	820	950	570	3.96	1.83	33	86
7	4 Bl. ports open	30600	27600	910	1040	788	860	547	3.78	1.79	37	89
8	2 Bl.	16400	12600	840	900	674	710	468	2.94	1.68	16	76
9	6 Bl. no shld.	58000		790	900	815		565		1.94	61	103
10	6 No shld.	60800		580	680	678		470		1.83	61	104
11	6 Bl.	47100	42800	870	940	781	810	542	3.99	1.96	39	88
1	recalculated	42100	49200			566		393	3.79	1.61	51°	96°
3	recalculated (Figure 7)	18200	17800			439		305	2.88	1.35	22°	79

Bl. -- draft blower on
No shld. -- no water flowing in shielding bank

Table III, continued

Run	Water velocity	125% h water	h_g obs.	h_g corr.	Gas-wall ΔT.	Gas t_f °F.	Re	h_ϵ from Δt	h_g from wall temp.	$\frac{hd}{k}$	$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{\frac{2}{3}}$
1	0.442	180	1.82	2.04	277	308	1860	2.02	2.28	28.3	0.0171
2	0.435	175	1.72	1.91	232	264	1890	1.87	2.08	27.2	0.0164
3	0.451	170	1.54	1.68	143	186	1720	1.69	1.87	26.8	0.0175
4	0.435	180	2.12	2.40	243	275	2720	2.31	2.64	34.0	0.0142
5	1.435	180	2.13	2.41	232	271	2790	2.37	2.73	35.3	0.0144
6	0.431	170	2.08	2.35	191	224	2730	2.20	2.50	34.0	0.0142
7	0.451	190	1.99	2.24	182	225	2620	2.22	2.53	34.4	0.0149
8	0.458	180	1.87	2.09	109	156	2430	1.99	2.24	33.4	0.0155
9	0.484	205	2.17	2.46	318	347	2400	2.42	2.80	33.5	0.0158
10	0.490	205	2.03	2.28	363	379	1950	2.22	2.52	29.1	0.0169
11	0.856	295	2.12	2.40	269	286	2440	2.32	2.66	33.9	0.0158
1 recalculated	0.442	180	1.78	1.99	285	314	1720	1.96	2.20	27.0	0.0177
3 recalculated	0.451	170	1.48	1.69	151	192	1520	1.59	1.75	25.0	0.0186

Table IV

Thermocouple corrections for blower capacity

Date	Correction to thermocouple Number			
	1	2	3	4
10-18-49		18°		
10-25-49	16°	23°	0	0
10-31-49			0	0
11-1-40	0	0	0	0

Table V

Comparison of gas film coefficients calculated by various methods

Run No.	A	B	C	D
1	2.88	2.04	2.39	
2	2.08	1.91	2.07	
3	1.87	1.68	1.48	
4	2.64	2.40	2.14	
5	2.73	2.41	2.28	
6	2.50	2.35	2.29	2.52
7	2.53	2.24	1.96	
8	2.24	2.09	1.45	2.23
9	2.80	2.46	2.64	
10	2.52	2.28	2.42	
11	2.66	2.40	2.12	

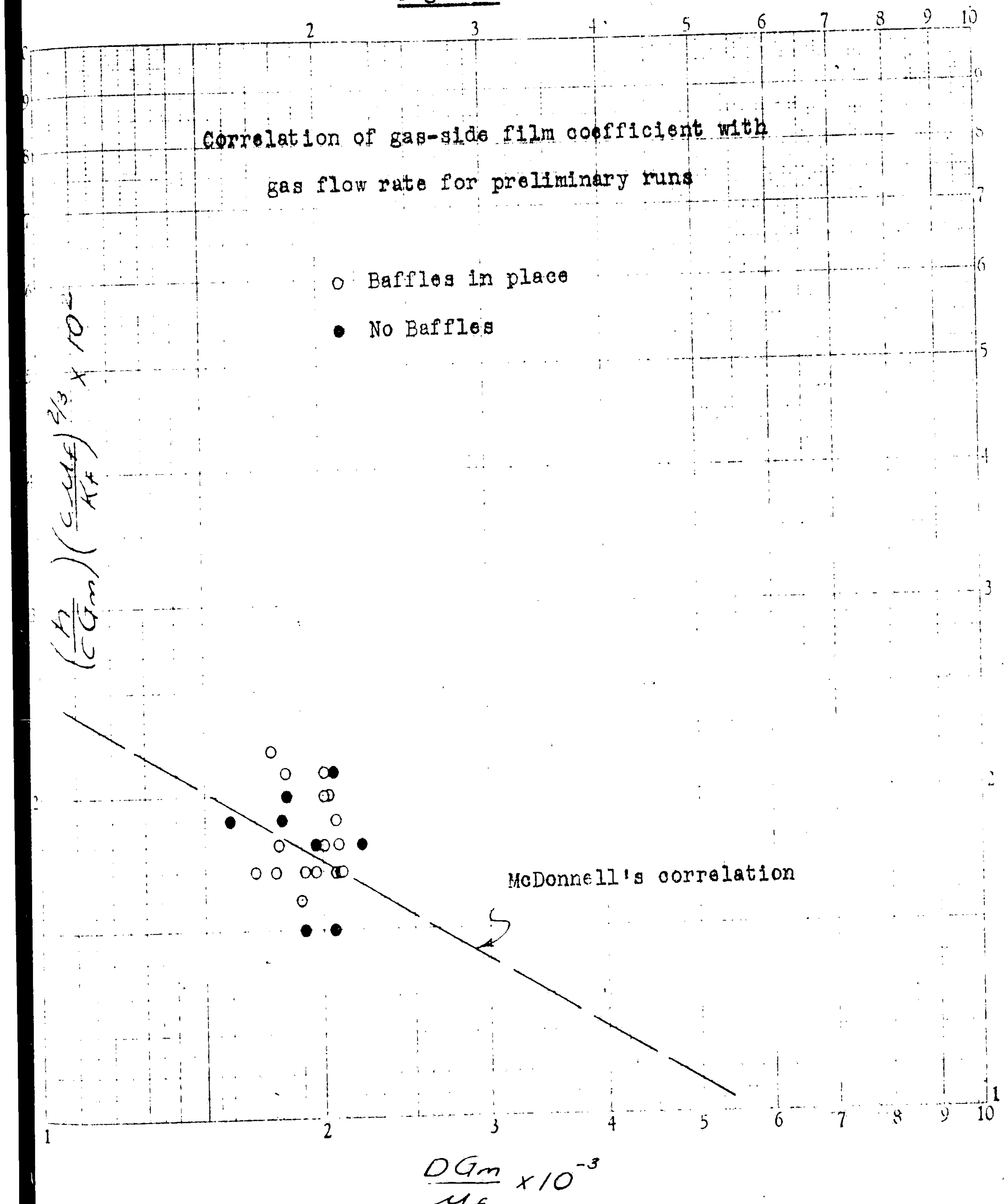
A - calculated from wall temperature

B - calculated from overall coefficient neglecting fin resistance

C - logarithmic mean of upper and lower values calculated from fin-tip and base wall temperature

D - calculated from overall coefficient including fin resistance

Figure 4



Correlation of gas-side film coefficient with
gas flow rate for preliminary runs
○ Baffles in place
● No Baffles
McDonnell's correlation
$\left(\frac{h}{C G_m}\right)\left(\frac{C \mu_f}{K_f}\right)^{2/3} \times 10^{2}$
$\frac{D G_m}{\mu_f} \times 10^{-3}$

Figure 5

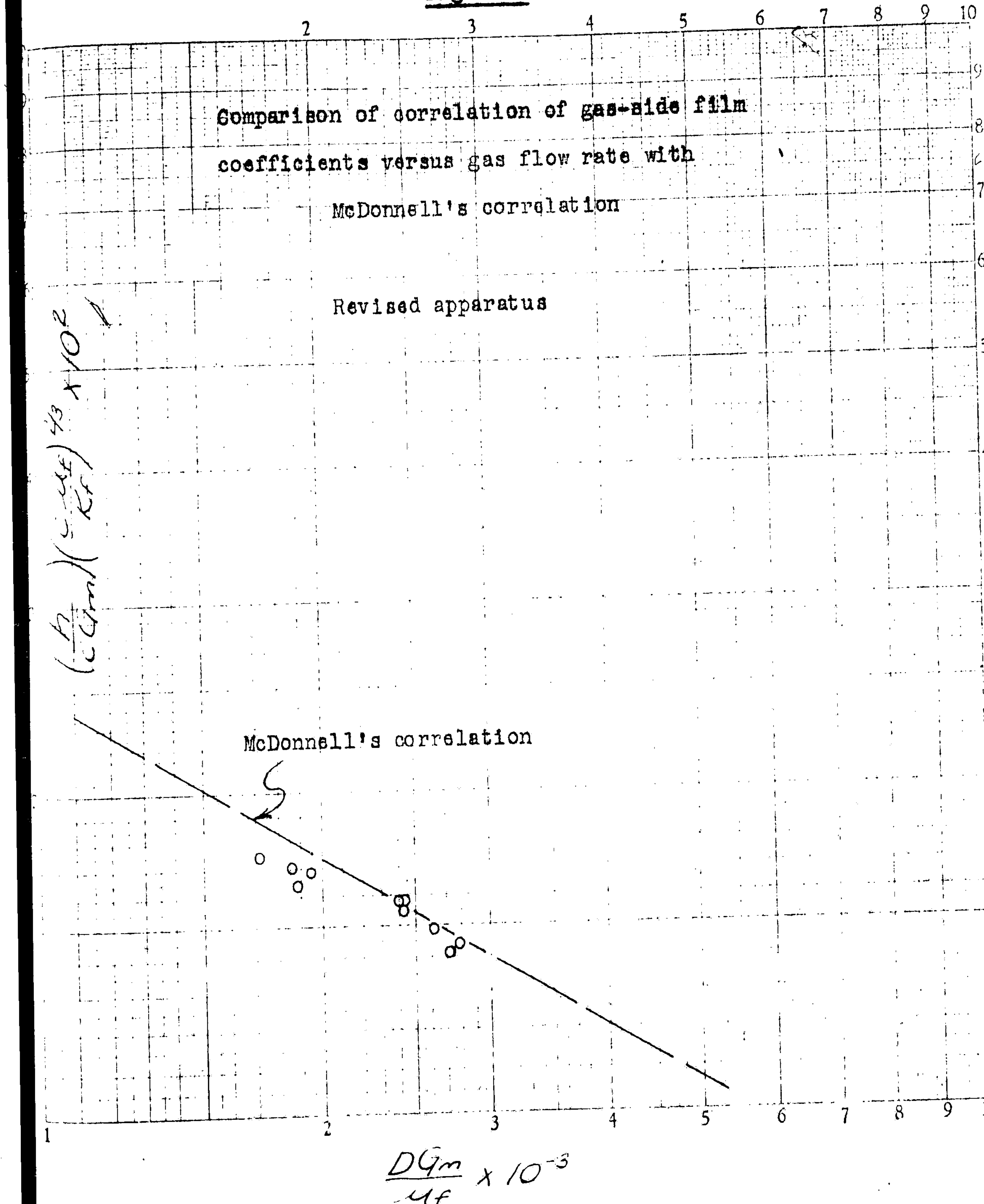

Comparison of correlation of gas-side film
coefficients versus gas flow rate with
McDonnell's correlation
Revised apparatus
McDonnell's correlation
$\left(\frac{h}{c G_m}\right)\left(\frac{c \mu_f}{k_f}\right)^{2/3} \times 10^2$
$\frac{D G_m}{\mu_f} \times 10^{-3}$

Figure 6

Correlation of gas-side film coefficients with gas flow rate: Nusselt number versus Reynolds number

Revised apparatus

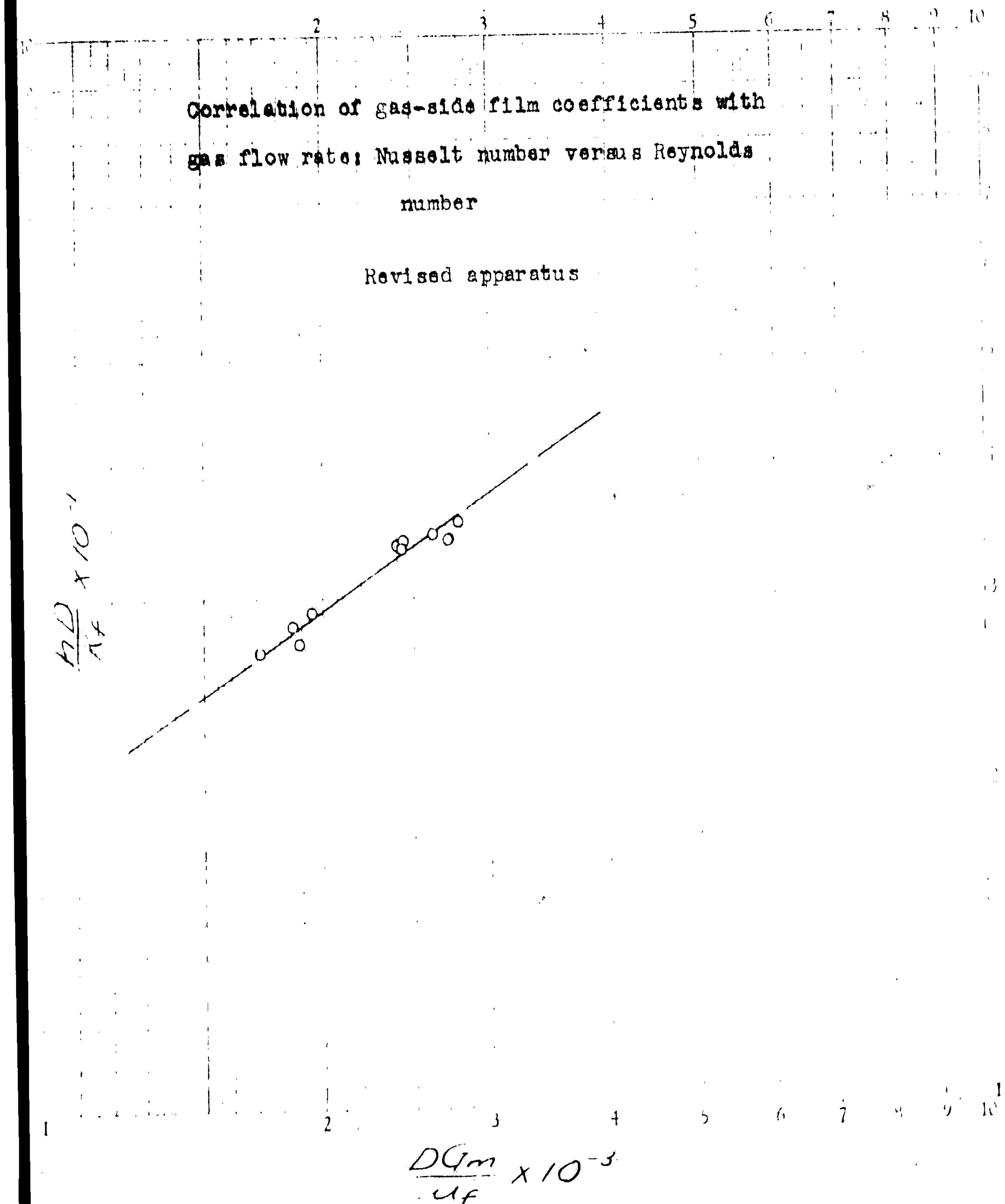


Figure 4

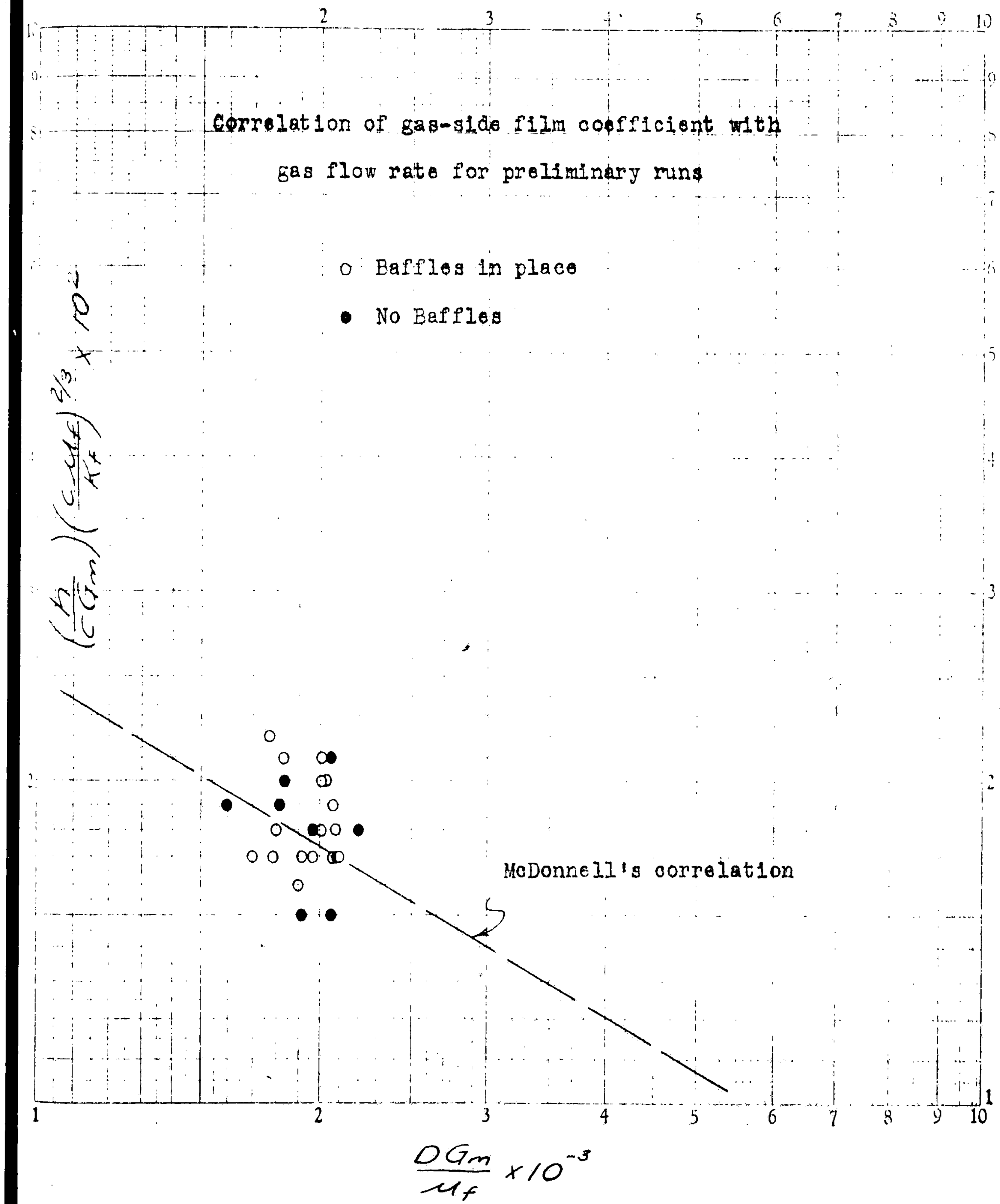
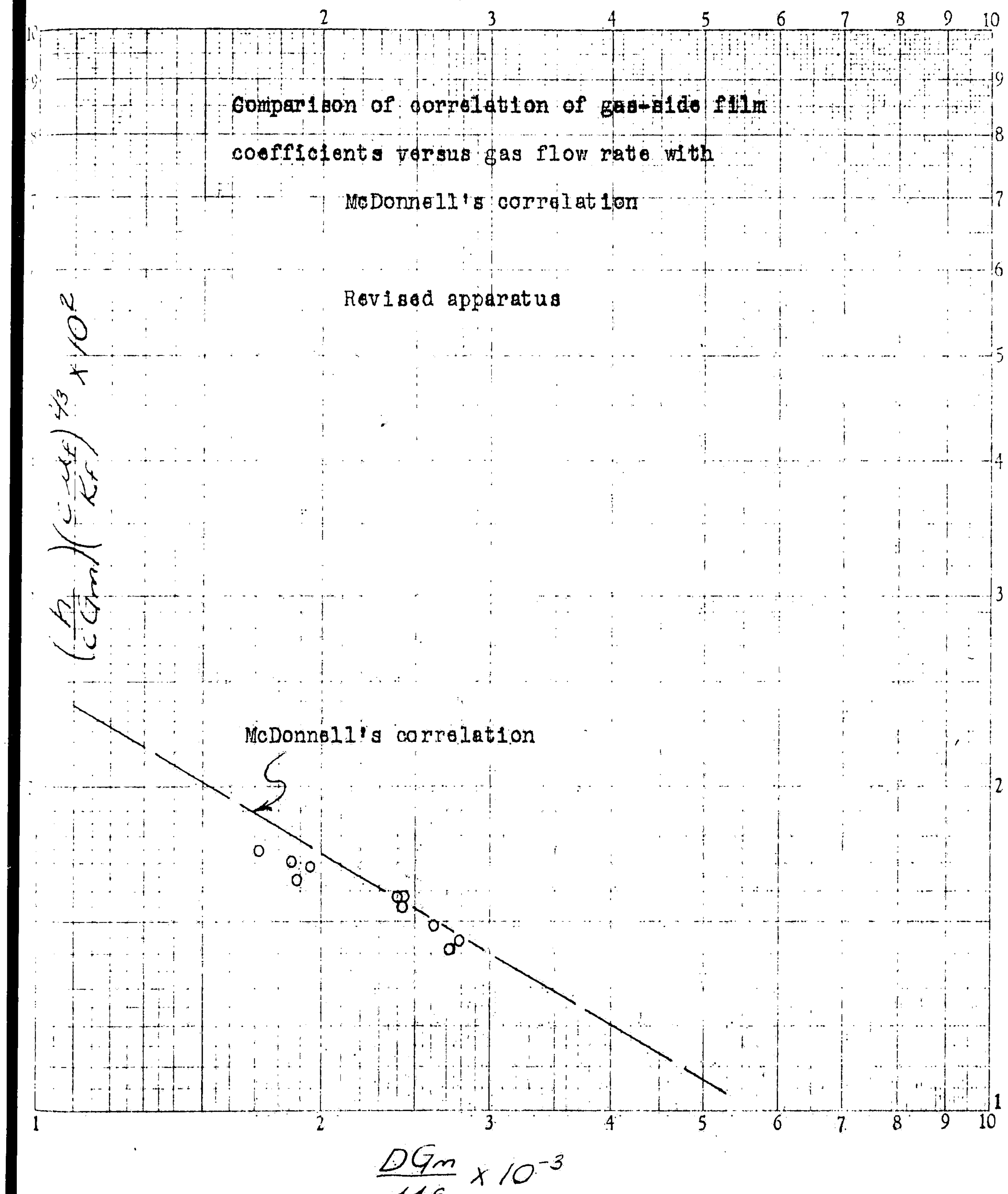

Correlation of gas-side film coefficient with
gas flow rate for preliminary runs
○ Baffles in place
● No Baffles
McDonnell's correlation
$\left(\frac{h}{C G_m}\right)\left(\frac{C \mu_f}{k_f}\right)^{2/3} \times 10^2$
$\frac{D G_m}{\mu_f} \times 10^{-3}$

Figure 5



Comparison of correlation of gas-side film
coefficients versus gas flow rate with
McDonnell's correlation
Revised apparatus
McDonnell's correlation
$\left(\frac{h}{c\,G_m}\right)\left(\frac{c\,\mu_f}{k_f}\right)^{2/3} \times 10^2$
$\frac{DG_m}{\mu_f} \times 10^{-3}$

Figure 6

Correlation of gas-side film coefficients with gas flow rate: Nusselt number versus Reynolds number

Revised apparatus

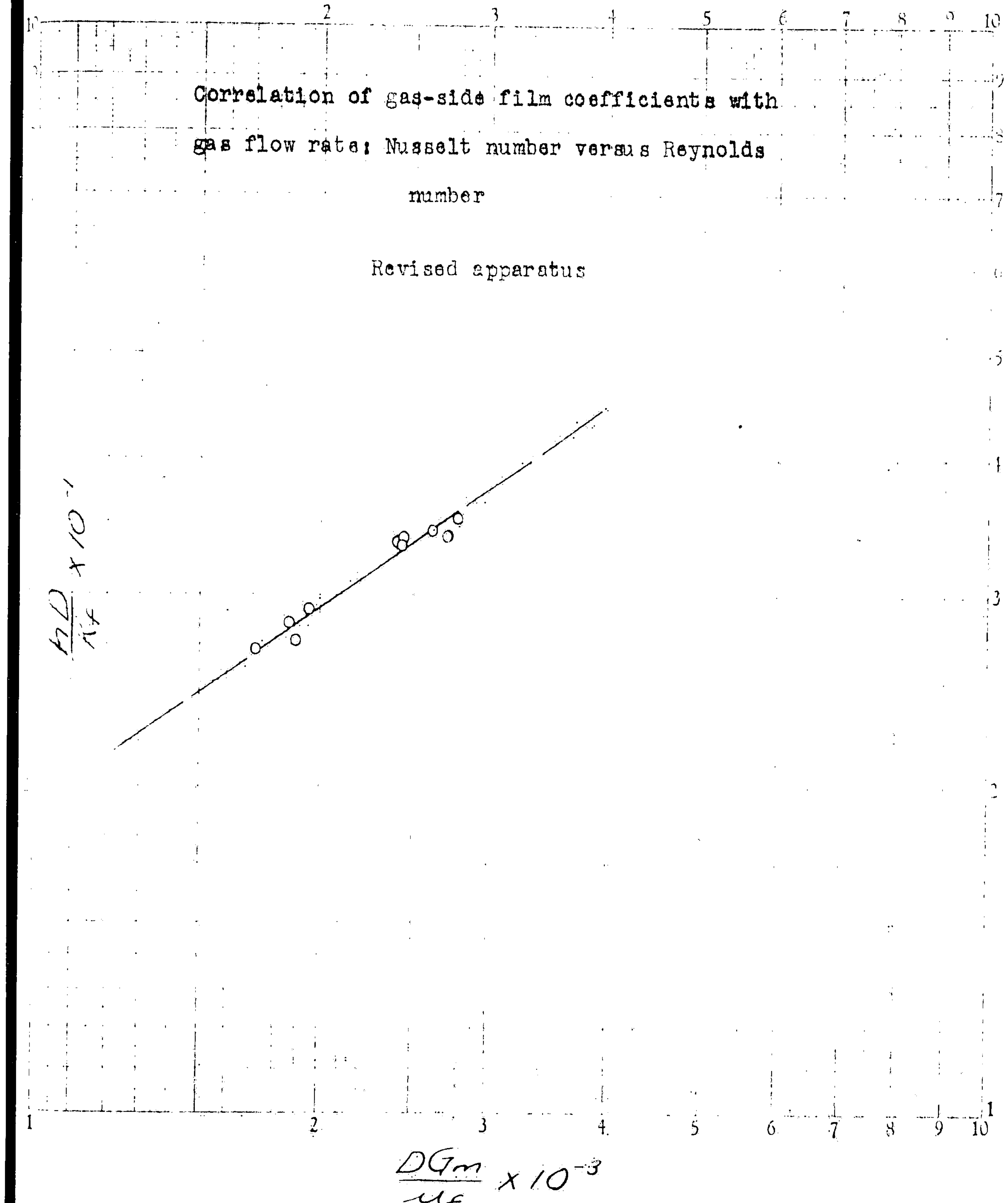
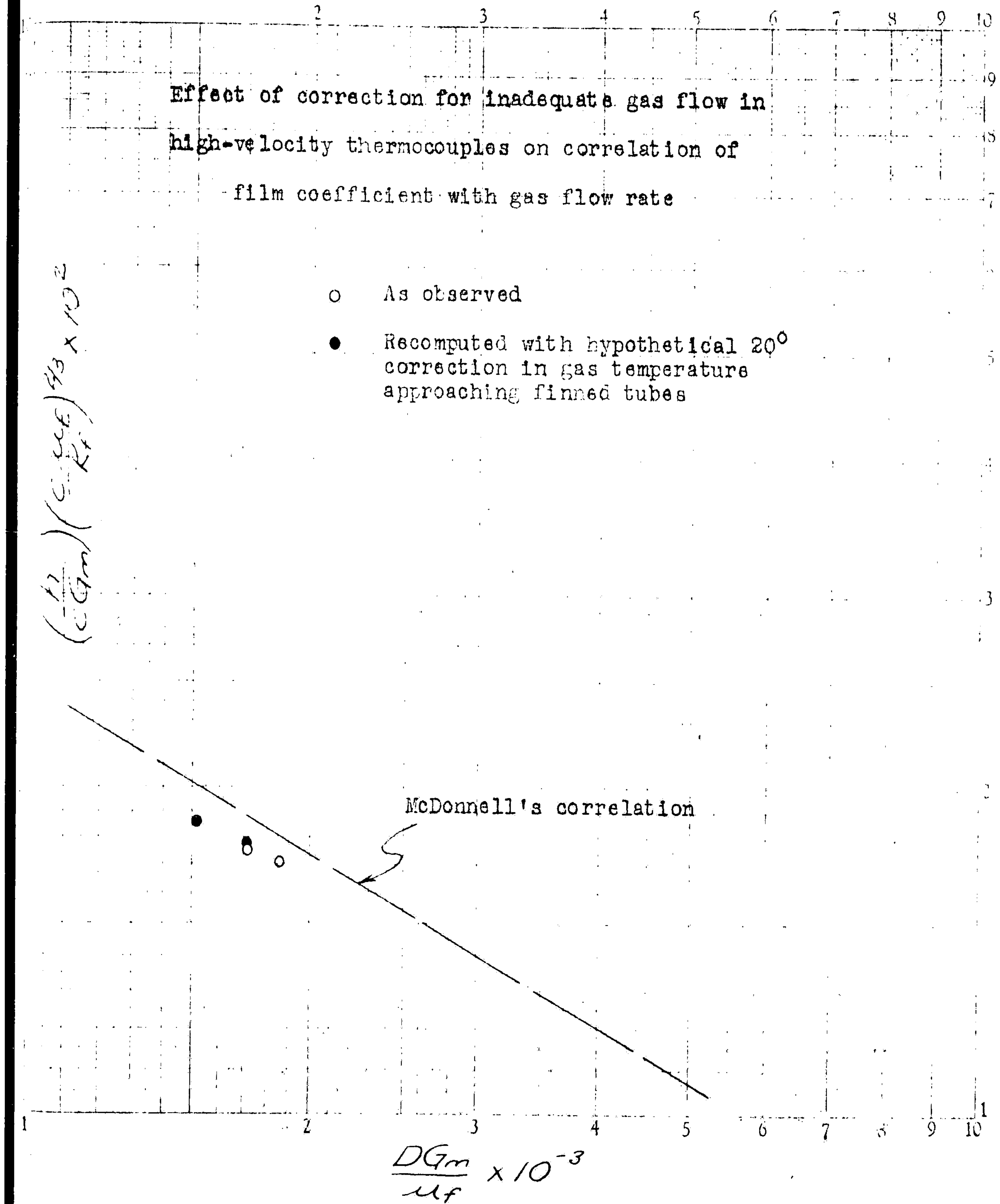


Figure 7



Effect of correction for inadequate gas flow in high-velocity thermocouples on correlation of film coefficient with gas flow rate

Conclusion.

The McDonnell correlation,

$$\left(\frac{h_g}{cG}\right)\left(\frac{c\mu_f}{k_f}\right)^{2/3} = 0.935\ Re^{-0.525}$$

has been shown to be applicable to heat transfer characteristics of finned tubes at gas temperatures up to about 700° F. Previous data on which this correlation was based covered gas temperatures up to only about 200° F. No appreciable radiative transfer was observed at the temperature studied.

The agreement of values obtained here, to which the Harper-Brown concept of fin effectiveness was applied, with the expression given above, lends qualitative support to the reliability of this concept at high values of fin effectiveness.

A critical comparison of various methods of arriving at the gas film coefficient has shown that (1) great precision would be necessary to obtain acceptable film coefficient values from measurements of fin-tip and base wall temperatures; (2) usual methods of calculating gas film coefficient from overall coefficient are not precise in that they neglect the largest resistance (except for the gas film), namely the resistance of the metal in the fins; but (3) values of gas film coefficient can be calculated which are in good agreement with directly observed values by taking into account the fin resistance; (4) the simplest method of obtaining reliable gas film coefficients is by direct measurement of base wall temperatures to obtain the gas-wall mean temperature difference.

The apparatus and techniques developed during the investigation described here should prove suitable for investigation of still higher gas temperatures and of the roles of convection and radiation in heat transmission to finned tubes.

Appendix

Table VI

Observed and calculated values for preliminary runs

Run No.	(1	2	3	4)	(5	6	7	8)
Conditions No. of burners and port position	6 - closed				6 - top open			
Water rate, lb./min.	63	86	111	128	57	92	115	126
Δt, °F.	16.0	13.0	9.0	9.5	20.0	12.0	9.5	8.0
Q, Btu./hr.	60500	67100	60000	73000	68500	66300	65600	60500
Ave. water temp., °F.	70	70	70	70	70	70	70	70
Gas in, °F.	780	730	750	770	700	690	700	690
Gas out, °F.	330	330	330	340	350	340	330	320
$(\Delta t)_m$	450	430	440	450	430	420	420	410
U, Btu./ft.^{2}-hr.-°F./ft.	1.8	2.1	1.8	2.2	2.1	2.1	2.1	2.0
Stack temp., °F.	330	300	300	330	340	326	310	300
Gas vell, fpm.	470	460	450	460	520	510	510	510
Corr. gas vel., fpm.	570	550	540	560	640	620	620	610
G_m	420	420	410	410	460	460	470	470
Gas film temp.	330	320	320	330	[illegible]	310	310	300
μ_f, lb./ft.-hr.	0.0592	0.0586	0.0586	0.0592	0.0586	0.0580	0.0580	0.0574
DG/μ	1800	1830	1790	1770	2010	2030	2080	2100
Water film temp., °F.	100	100	100	100	100	100	100	100
h for water	140	180	225	250	130	190	230	245
1/U	0.55	0.48	0.55	0.45	0.48	0.48	0.48	0.50
$1/h_w$	0.0071	0.0056	0.0044	0.0040	0.0077	0.0053	0.0044	0.0041
Water and wall resistance	0.076	0.061	0.048	0.044	0.082	0.058	0.048	0.045
h_g apparent	2.1	2.4	2.0	2.4	2.5	2.4	2.3	2.2
h_g corrected	2.4	2.8	2.2	2.8	2.0	2.8	2.7	2.5
$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{\frac{2}{3}}$	0.018	0.021	0.017	0.022	0.020	0.020	0.018	0.017

Table VI, continued

Run No.	(9	10)	(11	12)	13	14	15)
Conditions No. of burners and port position	6 - open		6 - open		6	Closed 4	2
Water rate, lb./min.	65	95	113	130	63	64	63
Δt, °F.	13.0	95	8.0	7.0	16.0	10.5	7.0
Q, Btu./hr.	50700	54100	54300	54600	60500	40400	26500
Ave. water temp., °F.	70	70	70	70	70	70	70
Gas in, °F.	610	620	600	600	780	570	440
Gas out, °F.	310	300	290	290	330	270	220
$(\Delta t)_m$	370	370	350	350	450	330	240
U, Btu./ft.^{2}-hr.-°F./ft.	1.8	1.9	2.1	2.1	1.8	1.6	1.5
Stack temp., °F.	290	280	280	290	330	250	220
Gas vel., fpm.	500	500	490	490	470	430	360
Corr. gas vel., fpm.	600	590	580	580	570	500	410
G_m	460	460	450	450	420	410	350
Gas film temp.	280	280	270	270	330	260	210
μ_f, lb./ft.-hr.	0.0564	0.0564	0.0558	0.0558	0.0592	0.0554	0.0526
DG/μ	2090	2090	2070	2070	1800	1900	1700
Water film temp., °F.	100	100	100	100	100	90	90
h for water	145	195	225	250	140	135	135
1/U	0.55	0.53	0.48	0.48	0.55	0.63	0.67
$1/h_w$	0.0069	0.0051	0.0044	0.0040	0.0071	0.0074	0.0074
Water and wall resistance	0.074	0.056	0.048	0.044	0.076	0.079	0.079
h_g apparent	2.1	2.1	2.3	2.3	2.1	1.8	1.7
h_g corrected	2.4	2.4	2.7	2.7	2.4	2.0	1.9
$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{\frac{2}{3}}$	0.017	0.017	0.019	0.019	0.018	0.016	0.017

Table VI, continued

Run No.	(16	17	18)	(19	20	21)	(22	23	24)
Conditions	Closed, no baffles			Top open			Top open - no baffles		
No. of burners and port position	6	4	2	6	4	2	6	4	2
Water rate, lb./min.	65	63	66	57	57	57	57	57	56
Δ t, ^{o}F.	18.0	13.0	6.0	20.0	14.0	7.5	22.0	14.0	7.5
Q, Btu./hr.	70200	49200	23800	68500	47900	25700	75300	47900	25200
Ave. water temp., ^{o}F.	70	70	70	70	70	70	80	70	70
Gas in, ^{o}F.	770	610	400	700	550	390	750	590	370
Gas out, ^{o}F.	360	310	200	350	270	210	370	290	200
$(\Delta t)_m$	470	370	210	430	320	220	450	350	200
U, Btu./ft.^{2}-hr.-^{o}F./ft.	2.0	1.8	1.5	2.1	2.0	1.6	2.2	1.8	1.7
Stack temp., ^{o}F.	340	300	200	340	270	200	380	280	200
Gas vel. fpm.	480	430	320	520	460	410	540	480	400
Corr. gas vel., fpm.	590	520	360	640	540	460	630	570	450
G_m	430	400	320	460	430	400	480	450	390
Gas film temp.	340	280	190	320	250	200	340	270	180
μ_f, lb./ft.-hr	0.0598	0.0564	0.0514	0.0586	0.0548	0.0520	0.0598	0.0558	0.0506
DG/μ	1840	1820	1600	2010	2010	1970	2060	2070	1970
Water film temp., ^{o}F.	100	100	90	100	90	90	110	100	80
h for water	145	140	140	130	120	120	135	130	115
1/U	0.50	0.55	0.67	0.48	0.50	0.63	0.45	0.55	0.59
$1/h_w$	0.0069	0.0071	0.0071	0.0077	0.0083	0.0083	0.0074	0.0077	0.0087
Water and wall resistance	0.074	0.075	0.076	0.082	0.088	0.088	0.079	0.082	0.092
h_g apparent	2.3	2.1	1.7	2.5	2.4	1.9	2.7	2.1	2.0
h_g corrected	2.7	2.4	1.9	2.9	2.8	2.1	3.2	2.4	2.2
$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{2/3}$	0.020	0.019	0.019	0.020	0.021	0.017	0.021	0.017	0.018

Table VI, continued

Run No.	(25	26	27)	(28	29	30)
Conditions		Open			Open - no baffles	
No. of burners and port position	6	4	2	6	4	2
Water rate, lb./min.	63	65	65	65	66	65
Δ t, °F.	13.0	10.5	6.0	14.5	10.0	5.0
Q, Btu./hr.	50700	41000	23400	56600	39600	19500
Ave. water temp., °F.	70	70	70	70	70	70
Gas in, °F.	610	530	380	630	540	350
Gas out, °F.	310	270	190	310	280	190
$(\Delta t)_m$	370	310	200	380	320	190
U, Btu./ft.^{2}-hr.-°F./ft.	1.8	1.8	1.6	2.0	1.6	1.4
Stack temp., °F.	290	250	180	310	280	190
Gas vel., fpm.	500	460	380	540	470	390
Corr. gas vel., fpm.	600	530	420	650	560	430
G	460	430	380	490	440	380
Gas film temp.	280	250	180	290	250	180
μ_f, lb./ft.-hr.	0.0564	0.0548	0.0506	0.0570	0.0548	0.0506
DG/μ	2090	2010	1920	2200	2060	1920
Water film temp., °F.	100	90	80	100	90	80
h for water	145	140	130	145	140	130
1/U	[illegible]	0.55	0.63	0.50	0.63	0.71
$1/h_w$	0.0069	0.0071	0.0077	0.0069	0.0071	0.0077
Water and wall resistance	0.074	0.076	0.082	0.074	0.076	0.082
h_g apparent	2.1	2.1	1.8	2.3	1.8	1.6
h_g corrected	2.4	2.4	2.0	2.7	2.0	1.8
$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{\frac{2}{3}}$	0.017	0.018	0.017	0.018	0.015	0.015

Table VII

Averaged values of data for second set of runs

Run	Conditions	Temp. below shield: TC 1, °F.	Temp. below finned bank TC 2	Temp. below finned bank TC 3	Temp. above finned bank TC 4	Wall below TC 16	Fin-tip below TC 17	Wall above TC 19	Fin-tip above TC 18
1	6 burners	905	575	569	289	210	321	82	98
2	4 burners	747	482	482	251	176	258	80	92
3	2 burners	461	321	330	175	134	179	72	76
4	6, blower on	639	481	482	280	180	267	89	102
5	6, blower on, ports open	648	467	471	275	186	266	87	102
6	4, blower on	509	389	385	228	151	216	78	91
7	4, blower on, ports open	515	384	385	225	159	213	81	92
8	2, blower on	328	256	254	154	115	144	71	75
9	6, blower on, no shielding	719	633	624	341	235	368	86	106
10	6, no shielding	878	721	719	357	244	401	86	105
11	6, blower on	736	523	528	280	183	276	75	90

Table VII, continued

Run	Finned bank Water temperature In	Finned bank Water temperature Out	Rate, lb./min.	Shield bank Water temperature In	Shield bank Water temperature Out	Rate, lb./min.	Velometer reading fpm. Upper	Velometer reading fpm. Lower
1	65.3	75.7	67.5	65	143	10.5	420	460
2	65.7	73.9	66.5	66	146	7.2	400	440
3	66.0	70.4	69	66	142	3.9	300	340
4	66.0	76.6	66.5	66	136	10	630	710
5	65.6	76.0	66.5	66	134	9.5	700	790
6	65.4	73.4	66.0	65	121	8.5	600	660
7	65.9	73.3	69.0	66	120	8.5	660	750
8	65.6	69.5	70.0	66	96	7.0	570	610
9	65.8	78.9	74.0				620	700
10	65.8	79.3	75.0				460	520
11	65.2	71.2	131	65	140	9.5	650	700

Table VIII

Sample data from Run 3, revised apparatus

Time		Fin Bank Water temperature, °F. In	Fin Bank Water temperature, °F. Out	Manometer reading ΔH, in.	Shielding bank Water temperature, °F. In	Shielding bank Water temperature, °F. Out	Water rate	Velometer reading, fpm Upper	Velometer reading, fpm Lower
4:20	2 burners	65.9	70.3	2.05	66	145	3.9 lb/min.	300	340
4:30	2 burners	65.9	70.3	2.1	66	140			
4:40	2 burners	66.0	70.4	2.1	66	140		310	340
4:50	2 burners	66.0	70.4	2.1	66	141			
5:00		66.0	70.4	2.1	66	142		300	350
		t = 4.4°		2.1 in. = 69 lb/min		t = 76°			

Thermocouple readings, mv.

Time	Gas temperatures TC 1 Below shield	Gas temperatures TC 2 Below finned bank	Gas temperatures TC 3 Below finned bank	TC 4 Above bank	TC 16 Wall Below	TC 17 Fin tip below	TC 18 Fin tip above	TC 19 Wall above	Cold jct.
4:20	11.24	6.98	7.19	2.55	1.29	2.70	-0.41	-0.54	
4:30	11.18	7.00	7.19	2.55	1.30	2.62	-0.39	-0.51	90
4:40	11.14	6.75	7.23	2.52	1.39	2.67	-0.40	-0.52	
4:50	11.19	7.00	7.15	2.52	1.28	2.60	-0.39	-0.51	
5:00	11.18	6.80	7.29	2.52	1.24	2.59	-0.41	-0.51	90
Average	11.19	6.91	7.21	[illegible]	1.30	2.64	-0.40	-0.52	90
Corrected mv.	12.85	8.57	8.87	4.19	2.96	4.30	1.26	1.14	1.66
Temperature, °F.	461°	321°	330°	175°	134°	179°	76°	72°	

Sample Computations

Tube areas based on data in Tables I and II:

Total finned tube length $= \frac{5 \times 25 \text{ in.}}{12} = 10.4$ ft.

End area of fins $= \frac{0.0375 \text{ in.} \times 3.075 \text{ in.} \times \pi}{144}$

$\times$ 45 turns /ft. $\times$ 10.4 ft. $=$ 1.18 sq. ft.

(Correction of end length for pitch is negligible)

Base area $= \frac{3.075 \text{ in.}}{12} \times \pi \times 10.4$ ft. - end area

$= 8.39 - 1.18 = 7.21$ sq. ft.

Number of fins /ft. $=$ 45 turns /ft. $\times \frac{3.075\,\pi}{11/32}$

$=$ 1265 /ft.

Side area of fins $= \frac{2\,(11/32 \times 0.963) \times 1265/\text{ft.}}{144}$

$\times$ 10.4 ft. $=$ 60.6 sq. ft.

Edge area of fins $= \frac{2\,(0.963 \times 0.0375)}{144} \times 1265$/ft.

$\times$ 10.4 ft. $=$ 6.60 sq. ft.

Total area $= 1.2 + 7.2 + 60.6 + 6.6$

$=$ 75.6 sq. ft.

Total fin area neglecting edges $=$ 61.8 sq. ft.

Inside area $= \frac{125 \text{ in.}}{12} \times \pi \times \frac{2.74 \text{ in.}}{12}$

$=$ 7.47 sq. ft.

Outside area of shielding bank $=$ 29 tubes $\times \frac{25 \text{ in.}}{12}$

$\times \frac{1}{2.904 \text{ ft/ sq. ft.}}$ $=$ 20.8 sq. ft.

Averaged values of data are taken from Table VII for Run 5:

Q_f, heat taken up by finned tube bank = 60 min./hr. x 10.4° x 66.5 lb./min. = 41500 Btu./hr.

Q_s, heat taken up by shielding bank = 60 x 9.5° x 68 = 38800 Btu./hr.

U_f, overall heat transfer coefficient in finned bank

$$= \frac{41500}{75.6\ (\Delta t)m} = 1.89 \text{ Btu./hr. - sq.ft.-}^\circ\text{F.}$$

$$(\Delta t)m = \frac{(469-71) - (275-71)}{\ln \frac{398}{204}} = 290^\circ$$

where 71° is the mean water temperature.

$$U_s = \frac{38800}{20.8\ (\Delta t)m} = 4.10 \text{ Btu./hr. - sq.ft.-}^\circ\text{F.}$$

$$(\Delta t)m = \frac{(648-65) - (469-121)}{\ln \frac{583}{348}} = 455^\circ$$

Here the general direction of water flow is concurrent with gas flow.

$$\text{Gas rate, from } Q_f = \frac{41500}{(469^\circ - 275^\circ)\ (0.244)} = 876 \text{ lb./hr.}$$

where 0.244 Btu./lb. - °F. is the average specific heat of air in the range of temperature covered.

$$G_m, \text{ maximum mass velocity} = \frac{876 \text{ lb./hr.}}{1.44 \text{ sq. ft.}} = 609 \text{ lb./sq.ft.-hr.}$$

The minimum area between finned tubes = 5 tubes x $\frac{25 \text{ in.}}{12}$ x $\frac{2 \times 0.963 \text{ in.}}{12}$ - 45 turns /ft. x $\frac{2 \times 0.963 \text{ in.} \times 0.0375 \text{ in.}}{144}$ x 10.4 ft. = 1.67 - 0.23

= 1.44 sq. ft.

Mean water-wall temperature difference $= \dfrac{(186 - 71) - (87-71)}{\ln \dfrac{115}{16}}$

$= 50^{\circ}$

Water film temperature $= 71^{\circ} + \dfrac{50^{\circ}}{2} = 96^{\circ}$ F.

Water velocity $= \dfrac{66.5 \text{ lb./min.}}{60 \text{ sec./min.}} \times \dfrac{144}{(1.37 \text{ in.})^2 \pi}$

$\times \dfrac{1}{62.3 \text{ lb./cu.ft.}} = 0.44$ ft./sec.

From Figure 8, the film coefficient is found to be 145 Btu./hr. - sq. ft. - °F. To this is added 25% of its value because of the high turbulence caused by the return bands, making h = 180.

Adding resistances:

$$\frac{1}{U_f} = \frac{1}{h_g} + \frac{LA_f}{K\ A_1} + \frac{A_f}{h_w\ A_1}$$

where the symools have the significance given in under Nomenclature,

$$\frac{1}{1.89} = \frac{1}{h_g} + \frac{(0.131 \text{ in.})\ 75.6 \text{ sq. ft.}}{26 \text{ Btu./hr.} - \text{ft.} - {}^{\circ}\text{F.})7.47 \text{ sq. ft.}}$$

$$+ \frac{75.6 \text{ sq. ft.}}{(180)\ 7.47 \text{ sq. ft.}}$$

$h_g = 2.13$ Btu./hr. - sq. ft. - °F., observed

From Figure 10, the actual value of h_g is 2.41 Btu./hr.-sq.ft.-°F.

Second computation of h_g:

Mean gas-wall temperature difference =

$$\frac{(469 - 186) - (275 - 87)}{\ln \dfrac{283}{188}} = 232^{\circ}$$

$$h\epsilon = \frac{Q_f}{A_f\,(\Delta t)_m} = \frac{41500 \text{ Btu./hr}}{75.6 \text{ sq. ft. } (232^\circ)}$$

$$= 2.37 \text{ Btu./hr. - sq.ft.-}^\circ\text{F.}$$

From Figure 10, h_g $= 2.73$ Btu./hr. - sq.ft.-°F.

$$\epsilon = \frac{2.37}{2.73} = 0.868$$

The mean fin-gas temperature difference $= \epsilon$ (gas-wall Δt)

$$= 0.868 \times 232^\circ = 201^\circ$$

Gas film temperature $=$ mean gas temperature $- \frac{1}{2}$ (gas-fin Δt) $= 372^\circ - 101^\circ = 271^\circ$

By interpolation in Table IX, the properties of air at this temperature are:

Viscosity	=	0.0559 lb./ft.-hr.
Conductivity	=	0.0198 Btu./ft.-hr.-°F.
Heat capacity	=	0.242 Btu./lb.-°F
Prandtl number	=	0.68

From these values,

$$Re = \frac{DG}{\mu} = \frac{(3.07/12) \text{ ft. } (609 \text{ lb./sq.ft.-hr.})}{0.0559 \text{ lb./ft.-hr.}}$$

$$= 2790$$

$$Nu = \frac{hD}{K} = \frac{(2.73)\ (0.256 \text{ ft.})}{0.0198 \text{ Btu./hr.-ft.-}^\circ\text{F.}}$$

$$= 35.3$$

$$(Pr)^{2/3}(St) = \left(\frac{c\mu}{K}\right)^{2/3}\left(\frac{h}{cG}\right)$$

$$= (0.68)^{2/3} \frac{2.73}{(0.242 \text{ Btu./lb.-}^{\circ}\text{F.}) (609 \text{ lb./sq.ft.-hr.}}$$

$$= 0.0144$$

Figure 8:

Values were found for h at $V = 1$ ft./sec.
and various temperatures.

For $t_f = 80^{\circ}$,

$$h = \frac{160\ (1+0.012\ t_f)\ (V_s)^{0.8}}{(D')^{0.2}} = \frac{160\ (1+0.96)\ (1)}{1.223}$$

$$= 256$$

Lines were then drawn through these points with slope $= 0.8$

Figure 10:

A value for h was assumed, and the nomograph, Figure 9, was used to find ϵ'. For $h = 1$, $\epsilon' = 0.94$

$$\text{Then } \epsilon = \frac{1\ (A_b) + \epsilon'\ (A_f)}{A_t} = \frac{7.2+0.94\ (61.8)}{69.0}$$

$$= 0.945$$

The value of h observed $= h\,\epsilon$

$= 1 \times 0.945 = 0.945$ Btu./hr.-sq.ft.-°F.

Gas-side film coefficient computed from fin-tip and base wall temperature difference:

In the derivation of the expression for fin effectiveness, Harper and Brown (2) arrive at the following intermediate equation:

$$\theta_x = \theta_b \frac{\cosh a(x - w')}{\cosh aw'}$$

where θ = gas-metal temperature difference; sub b, at the base wall; sub x, at distance x from the base

a = the square root of the quantity $2h/kT$

h = gas side film coefficient, Btu per sq. ft. per ^{o}F. per hr.

k = thermal conductivity of fin material, Btu per sq. ft./per hr. per ^{o}F. per ft.

= 26 Btu per sq. ft. per ^{o}F. per ft. per hr.

T = fin thickness, ft. = 0.0031 ft.

x = distance from base wall to point where θ_x is measured = w' for the fin tip

w' = distance from base wall to end of fin, corrected for effect of fin thickness, taken as 1 inch = 0.083 ft.

Substituting these values for conditions at the fin tip, where x = w':

$$\theta_b/\theta_x = \cosh 0.414\ h^{\frac{1}{2}}$$

For Run 5, on the top side of the finned bank:

$$\theta_b = 275^{o} - 87^{o} = 188^{o}F.$$

$$\theta_x = 275^{o} - 102^{o} = 173^{o}F.$$

$$\theta_b/\theta_x = 1.088$$

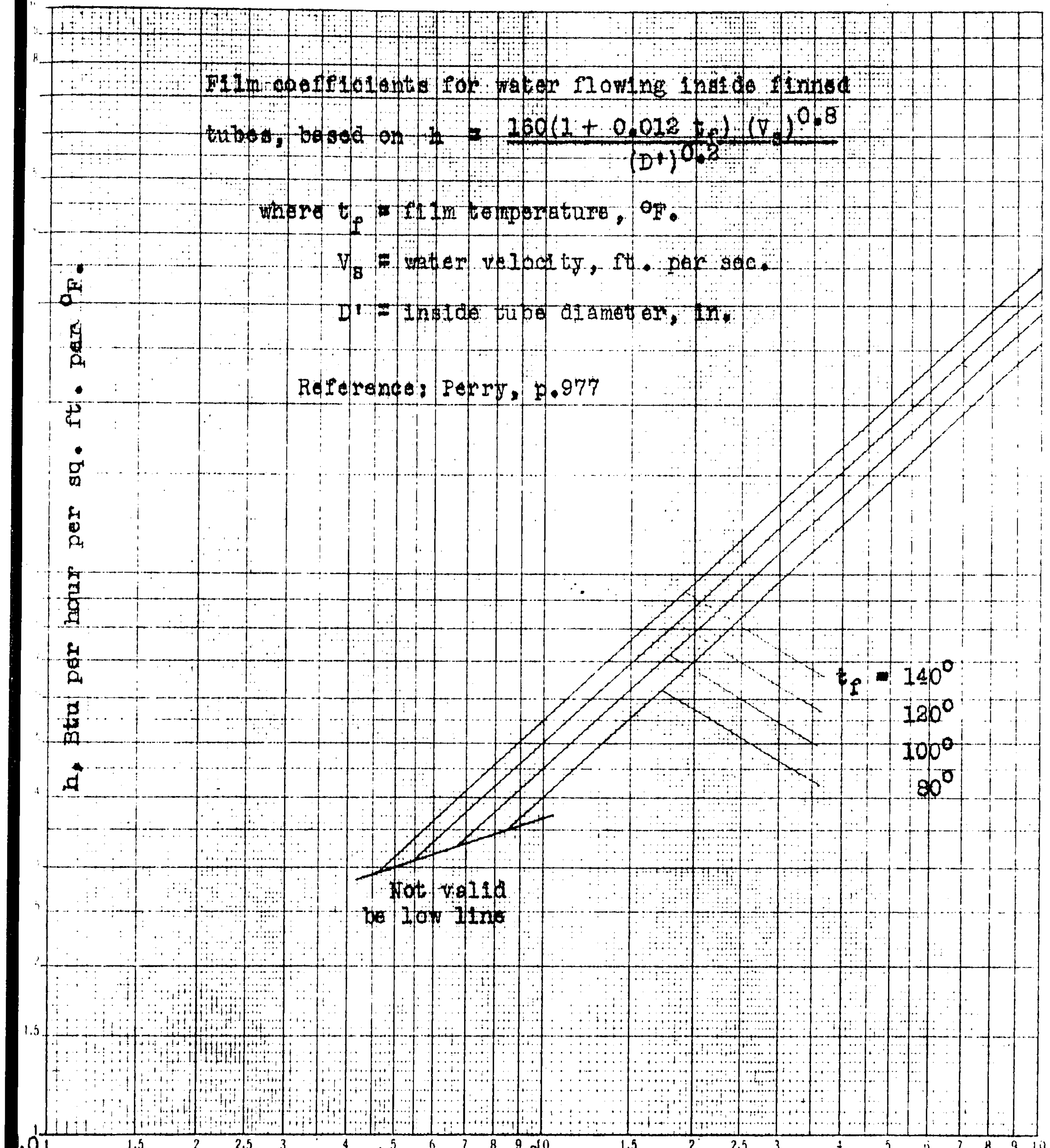
arccosh 1.088 = 0.430 from standard tables of hyperbolic functions

Therefore $h_{top} = (0.430/0.414)^2 = 1.08$

Similarly $h_{bottom} = 4.32$

The logarithmic mean of these two quantities is 2.28 Btu per sq. ft. per $^{o}F.$ per hr.

Figure 8



V_s, water velocity, feet per second

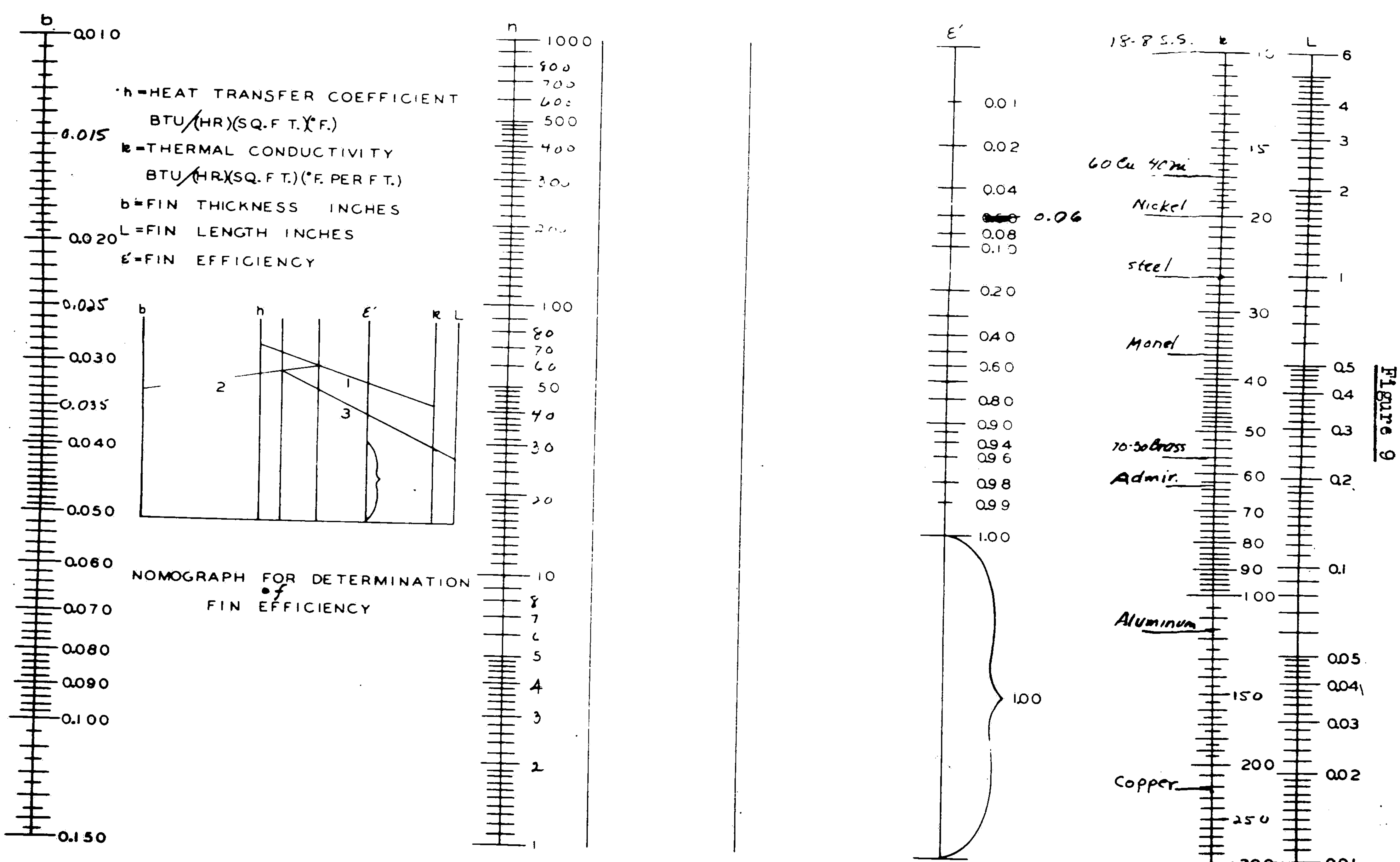

h = HEAT TRANSFER COEFFICIENT
BTU/(HR)(SQ. FT.)(°F.)
k = THERMAL CONDUCTIVITY
BTU/(HR)(SQ. FT.)(°F. PER FT.)
b = FIN THICKNESS INCHES
L = FIN LENGTH INCHES
ε' = FIN EFFICIENCY
NOMOGRAPH FOR DETERMINATION of FIN EFFICIENCY
b
n
ε'
k
L
18-8 S.S.
60 Cu 40 Ni
Nickel
steel
Monel
70-30 Brass
Admir.
Aluminum
Copper
0.06

Figure 9

Figure 10

Correction of film coefficient for fin effectiveness for steel fins, one inch long and 0.037 inch thick

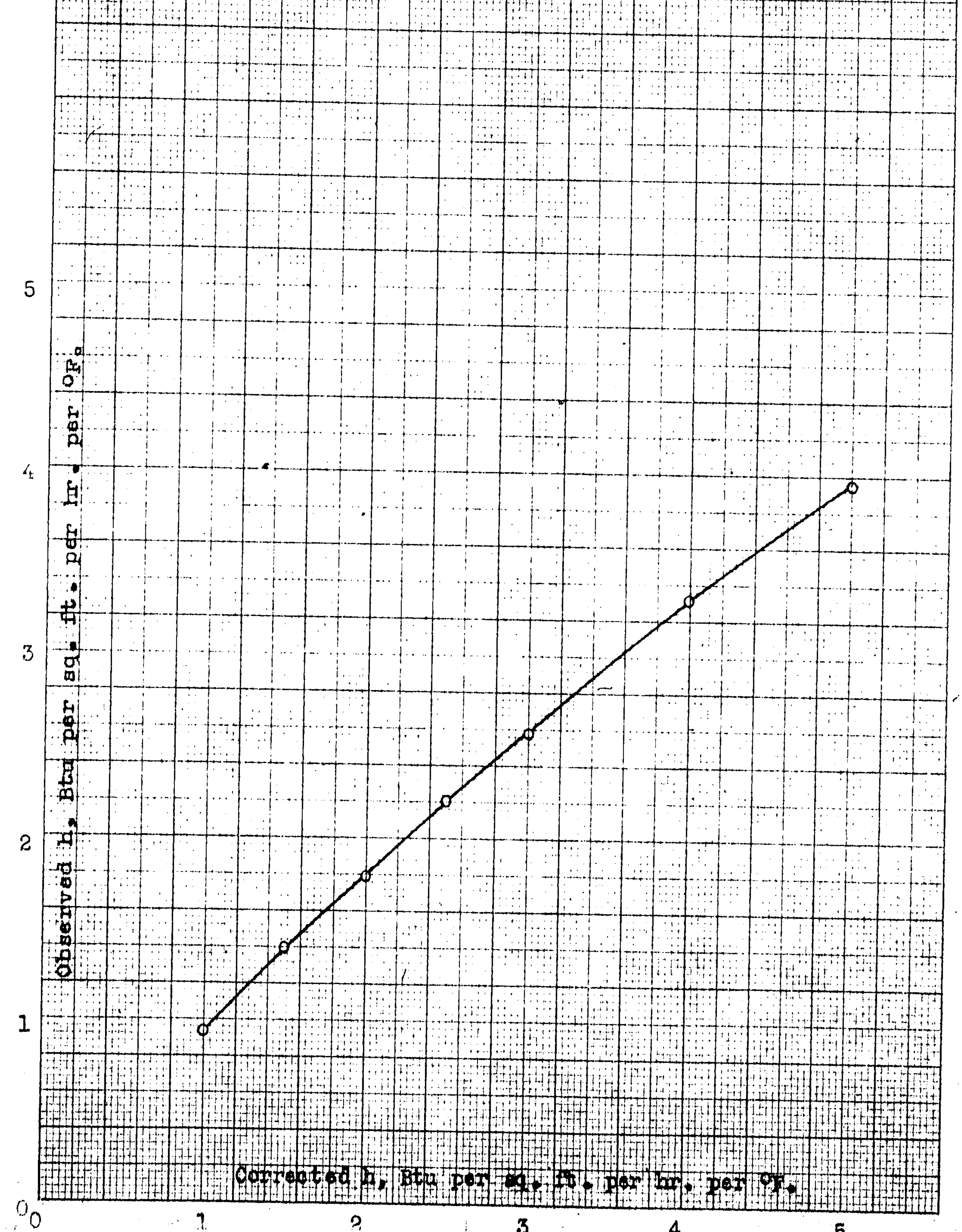


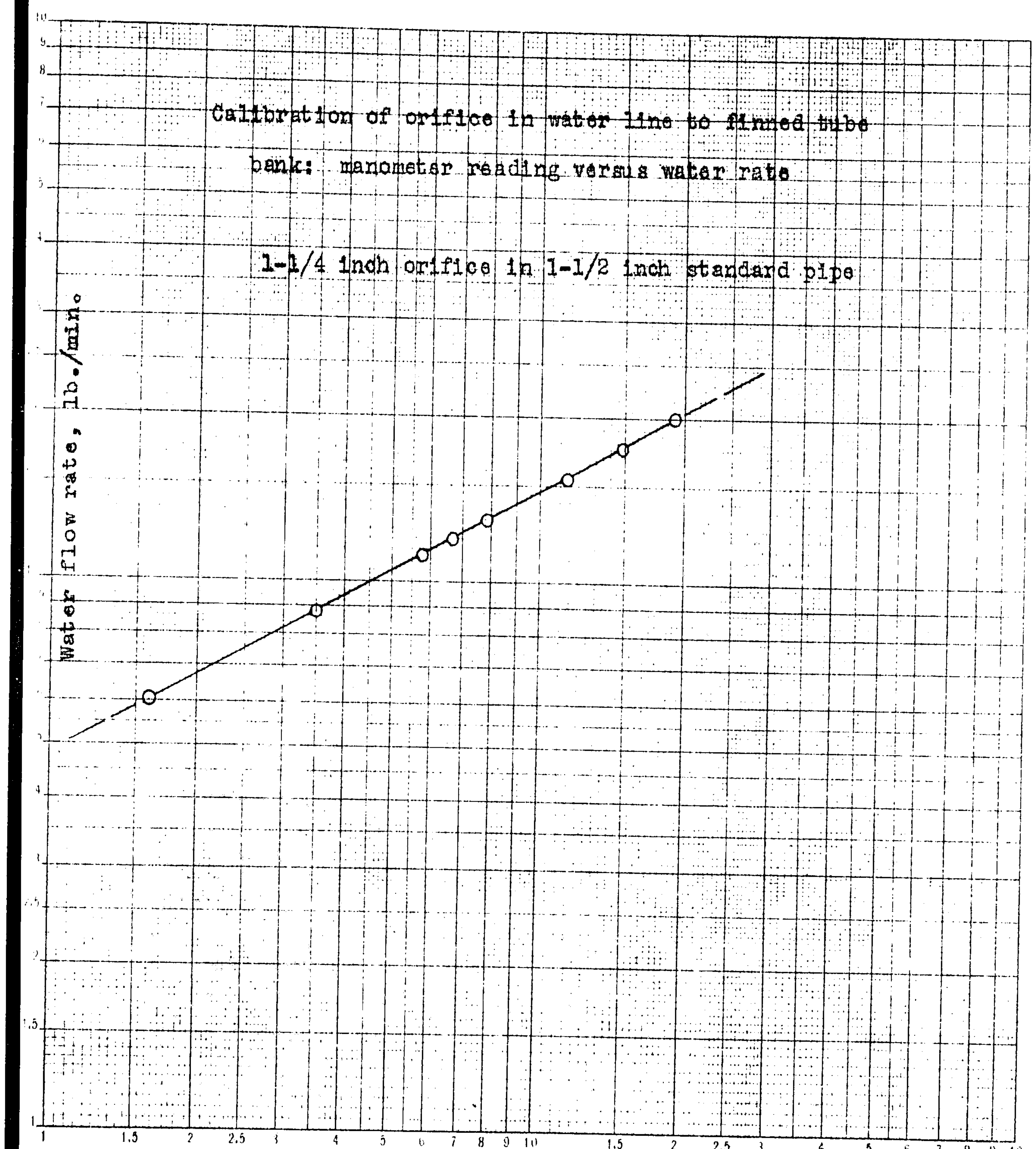
Table IX

Properties of air from Keenan and Kaye: "Gas Tables"

Table 2, Page 34

Temp. °F.	C_p Btu./(lb. - °F.)	μ lb./(ft. - hr.)	k Btu./(hr.-ft.-°F.)	Prandtl No. $\frac{C_p \mu}{k}$	$\left(\frac{c\mu}{k}\right)^{\frac{2}{3}}$
9.7	0.2394	0.0392	0.0130	0.72	0.81
40.3	.2396	.0425	.0143	.71	.80
90.3	.2399	.0454	.0156	.70	.79
140.3	.2403	.0486	.0168	.70	.79
190.3	.2409	.0515	.0180	.69	.78
240.3	.2416	.0544	.0191	.68	.78
290.3	.2424	.0569	.0202	.68	.78
340.3	.2434	.0598	.0213	.68	.78
440.3	.2458	.0645	.0237	.67	.77
540.3	.2486	.0691	.026	.66	.76
640.3	.2516	.0738	.028	.66	.76
740.3	.2547	.0785	.030	.66	.76
840.3	.2579	.0829	.032	.66	.76
940.3	.2611	.0871	.035	.65	.75
1040.3	.2642	.0911	.037	.65	.75
					.75
				" to	1940°F.

Figure 11



Manometer reading, inches of water over carbon tetrachloride

Table X and XI

Calibration of Thermometers

Water inlet and outlet thermometers in finned tube bank:

Inlet: Cenco N6360, 0-150°F. in 0.2° divisions

Outlet: Cenco N6367, 0-150°F. in 0.2° divisions

Standard: Parr 410422, N. B. S. certificate 73822, 65-105° F. in 0.05° divisions

All immersed 4 inches in water

Standard	N6360	N6367
75.65	75.6	75.6
80.45	80.4	80.4
87.85	87.8	87.8
91.45	91.4	91.4
98.45	98.3	98.3
104.30	104.1	104.1

Water outlet thermometer in shielding bank:

Outlet: Eimer & Amend 40-300° F. in 2° divisions

Standard: Parr 410422, described above, up to 100° F.

Brooklyn No. 93924, 0-100° C. in 0.1° divisions

Standard °C.	Standard °F.	Eimer and Amend
	72.7	72
	76.2	76
	88.2	88
	100.2	100
44.1	111.4	111

Table X, continued
and Table XI

Standard °C.	Standard °F.	Eimer and Amend
45.7	114.3	114
50.7	123.3	123
56.5	133.7	133
62.9	145.2	145
69.3	156.7	156
74.7	166.5	166
79.2	174.6	174
84.4	183.9	184
90.2	194.4	194

Table XII

Thermocouple check versus mercury thermometer

Thermocouples: Eleven No. 8 iron constantan elements

Comparison standard: A. H. Thomas Co. 600°F. in 2° divisions

In glycerol bath

Mean thermocouple reading, millivolts *	Cold junction		Indicated temperature		Standard °F.
	°F	Mv.	Mv.	°F.	
1.87	77	1.30	3.17	141	141
3.56	77	1.30	4.86	198	197
7.10	78	1.33	8.43	316	316
10.49	78	1.33	11.82	426	426
11.39	78	1.33	12.72	455	456

* Deviation of individual readings from this mean did not exceed 0.03 mv., with a mean deviation of 0.02 mv. or less.

Bibliography

1. Gardner, K. A.: Trans. Am. Soc. Mech. Engrs. 67, 621-31 (1945)

2. Harper, D. R., and W. B. Brown: Report No. 158, Natl. Advisory Comm. for Aeronautics Annual Report 8, 679-708 (1922)

3. Kayan, C. F.: Ind. and Eng. Chem. 40, 1044-9 (1948)

4. Keenan, J. H., and J. Kaye: "Gas Tables", Table 2, p. 34, New York, Wiley, 1948

5. McAdams, W. H.: "Heat Transmission", second edition, p. 415, New York, McGraw-Hill Book Company, 1942

6. McDonnell, R. W.: Thesis, Lehigh University, Bethlehem, Pennsylvania, 1949

7. McDonnell, R. W.: Personal communication

8. Mack, D. E., and A. E. Pitcher: Personal communication, to be published

9. Mullikin, H. F.: "Gas temperature measurement and the high-velocity thermocouple", in "Temperature, Its Measurement and Control", American Institute of Physics, pp. 775-804, New York, Reinhold Publishing Corporation, 1941.

10. Mullikin, W. H., and W. J. Osborn: "Accuracy tests of the high-velocity thermocouple", ibid., pp. 805-29