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REACTORS

TECHNICAL DIVISION
Section V

HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

W.B. HARRISON

[redacted]

OAK RIDGE NATIONAL LABORATORY

OPERATED BY
CARBIDE AND CARBON CHEMICALS CORPORATION
FOR THE
ATOMIC ENERGY COMMISSION
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TECHNICAL DIVISION

Section V

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W. B. Harrison

October 1, 1948

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A HEAT TRANSFER SURVEY

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

FOREWORD

Heat transfer, important as it is in the design and operation of nuclear piles and their accessories, has not been represented adequately in the reports of the Manhattan District. Not only has there been a scarcity of consistent research on this topic but, also, much of the data found by experiment did not reach the stage of being reported formally. In addition, considerable heat transfer material is buried in reports of other titles, where the heat transfer was incidental to the main objective.

Consequently, this critical survey of the Manhattan District and Atomic Energy Commission heat transfer literature by W. B. Harrison both fills a need and points to the continuing need. Although he has done an excellent job of correlating Project data and comparing it with previous work, the scarcity of data leaves much in doubt in some cases of immediate interest.

Progress in this important field can come only if--

(1) Engineers and others are diligent in reporting formally all experimental data, and

(2) An organized effort is made to acquaint all of the laboratories with the status of heat transfer projects so that further correlations can be utilized promptly.

Cooperation is necessary or reactor design will continue to be unnecessarily limited by heat transfer considerations. After all, the generation of energy cannot exceed for long the amount removed.

This report by Mr. Harrison has been accepted by the University of Tennessee as fulfilling the thesis requirements for the degree of Master of Science in Chemical Engineering.

R. M. Boarts

Knoxville, Tennessee
September 20, 1948

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HEAT TRANSFER IN MANHATTAN DISTRICT
AND
ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 1

A HEAT TRANSFER SURVEY

W. B. Harrison
October 1, 1948

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PART 1

A HEAT TRANSFER SURVEY

Introduction

In view of the fact that heat transfer problems are of great interest to many groups concerned with the design and development of processes and operations in the atomic energy field, it was believed that an effort should be made to bring into one volume the data of various observers on cases of interest to the Atomic Energy Commission projects. While attending the Clinton Laboratories Training School of 1946-7, Dr. R. M. Boarts aroused sufficient interest in such a program that administrative approval was granted. Dr. E. J. Murphy, Gale Young, R. N. Lyon and others showed particular interest in the program and they were instrumental in getting the work under way.

The original purposes of the program were to assemble, study, and correlate heat transfer data found in the reports of the Manhattan Project and in the continuing activities of the Atomic Energy Commission for the purpose of making a critical survey of the state of information on this subject. The work was initiated under the direction of Dr. R. M. Boarts in March, 1947, and he has maintained contact with the program since his return to the University of Tennessee in September, 1947. At that time, Dr. E. J. Murphy undertook supervision of the survey, and, in July, 1948, the work was placed under the direction of R. N. Lyon.

The first step in initiating the program was a survey of project literature. The Oak Ridge National Laboratory files and the Atomic Energy Commission files at Oak Ridge have been carefully reviewed for heat transfer references. Some data were obtained through cooperation of K-25 (Carbide and Carbon Chemicals Corporation, Oak Ridge, Tennessee) and NEPA (Fairchild Engine and Airplane Company, Oak Ridge, Tennessee).

The next steps were to eliminate from consideration those data which were not sufficiently complete or accurate to justify correlation, and to assemble the remaining data into related groups based on the heat transfer mechanism involved.

An effort was then made to evaluate on a common basis all heat transfer data of comparable nature. When possible, the data have been compared with correlations recommended in the literature. For a few cases which are unique, it was possible to suggest some correlation, incomplete though it may be, based on present knowledge. The mechanisms which make up the body of the report are combined conduction and natural convection, and combined conduction and forced convection. These mechanisms are discussed for the cases with phase change (specifically, with boiling) and without phase change. Little data were located for the cases of radiation and condensation, but a few notes on these subjects may be found in Part 11 called "Selected Topics Related to Heat Transfer." Although the data included in the body of the report are experimental, a bibliography of analytical solutions to conduction and other problems is included in Part 7.

In general, English units (Btu, lb., ft., hr., °F) have been used throughout the report. A table of equivalents is included in Part 17 for use in changing from one system of units to another. Physical properties data, and graphical calculation aids have also been included in the report to facilitate the making of heat transfer predictions. The meanings of all symbols used in the report may be found in Part 18 entitled "Nomenclature."

The term "project" refers to activities of the Manhattan District, the Atomic Energy Commission, and cooperating laboratories which have reports in the present Atomic Energy Commission files.

An effort has been made to use a change of verb tense in order to indicate which work is extracted from other reports and which is original with this report. It has not been possible to follow this pattern completely, but, in general, the past tense has been used to refer to work of others and present tense indicates discussion by the author of this report.

This paper is not intended to be a beginning course in heat transfer, but it is hoped that it may supplement the information found in existing references[1,2,3,4,5,6,7,8,9].

It is realized that many sets of project heat transfer data have never reached the stage of being given report numbers so that they may be located in the present subject indexes. Such information will never be circulated to other project laboratories unless interested individuals take the initiative to make the information available. It is also possible that some data have been overlooked in the literature search. In view of these considerations, it is expected that additions will be made to this report in the future, and, for that reason, it is being published in several small sections so that additional information may be included with other data on the same case.

The present report is, in general, classified at a higher level than one would expect it should be. The reason for this is that data have been extracted from reports which are classified and the entire part of this report in which they appear has been given the classification of the report from which they were extracted. It is hoped that declassification of certain parts of this report can be initiated soon. It is also hoped that an unclassified edition of the entire report can be made soon.

It is the present objective to make the report very complete enhancing its value to all project laboratories. This cannot be done without the cooperation of other laboratories in the form of forwarding heat transfer data to this location. Any suggestions for improving the present report or any additional data will be greatly appreciated. It is requested that forthcoming comments or information be addressed to the Technical Division, Engineering Research Section, Oak Ridge National Laboratory, Oak Ridge, Tennessee (Attention: R. N. Lyon or W. B. Harrison).

Acknowledgments

Dr. R. M. Boarts has been very valuable as the instigator of this program and a consultant. His many comments and suggestions are reflected throughout the report, and his guidance has been most helpful.

Dr. E. J. Murphy facilitated progress of this work in an administrative capacity. Without his efforts, the program would have been considerably delayed.

R. N. Lyon, W. L. Sibbitt, N. F. Lansing, and J. R. Menke have stimulated the progress and quality of the report with their discussions of various heat transfer cases and by supplying many useful data. R. N. Lyon has been particularly helpful in criticizing the rough draft, and the final draft incorporates many suggestions made by him.

The cooperation of the libraries at Oak Ridge National Laboratory, Atomic Energy Commission, K-25, and Fairchild Engine and Airplane Corporation (NEPA Division) is greatly appreciated.

The figures in this report were drawn by C. C. Hurtt; revisions in the figures were made by R. L. Towns; and the typing of the final report was done by T. K. Sutton and Gladys Darnell.

Many others have stimulated the work by showing an active interest and by supplying information not readily available. Thanks are extended to them collectively, since they are too great in number to mention individually.

W.B. Harrison

W. B. Harrison

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES; A CRITICAL SURVEY

PART 2

HEAT TRANSFER TO LIQUIDS BY COMBINED CONDUCTION AND NATURAL CONVECTION

W. B. Harrison
October 1, 1948

PART 2

HEAT TRANSFER TO LIQUIDS BY COMBINED CONDUCTION AND NATURAL CONVECTION

Heat transfer by combined conduction and natural convection is the natural result of a temperature gradient in a fluid. A system employing this mechanism for cooling is usually simple and economical, but the disadvantages are in the form of low heat transfer and control difficulties. The large majority of studies on heat transfer to liquids by this mechanism have employed water as the coolant, since it is cheap and plentiful.

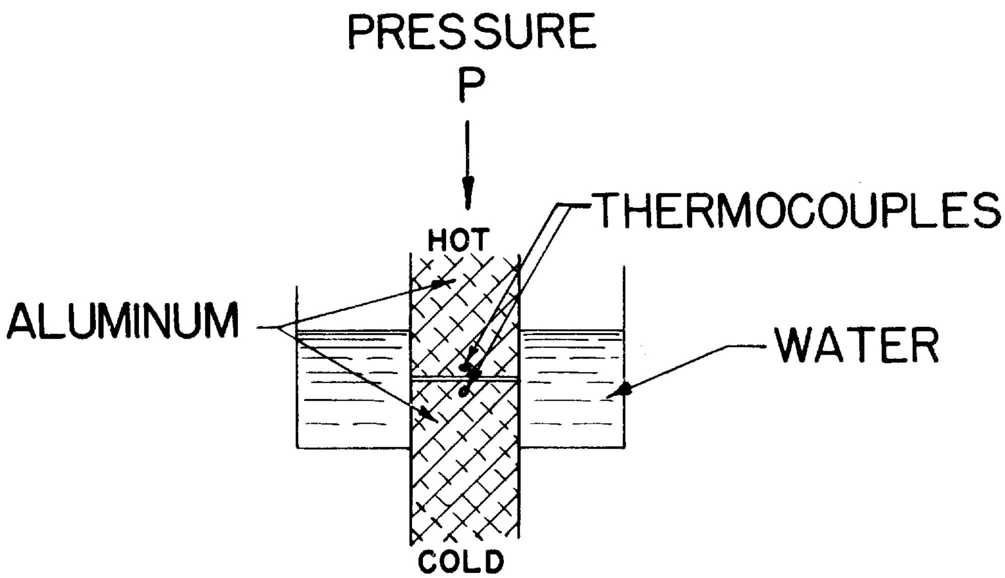
Bauer[1] reported on experiments to determine heat transfer by natural convection. Essentially, the equipment consisted of a vertical $1\frac{1}{2}$ in. aluminum tube surrounded by a $5\frac{1}{2}$ in. steel tube, 7 ft. long. Electric heaters were wound around the outside of the large tube and separate electrical connections were made for each foot of length so as to permit application of heat to the system in various ways. Water between the tubes was circulated by natural convection only and heat was removed by a cold water stream passing through an annulus within the aluminum tube. Temperature of the water was plotted versus height of the apparatus for several different heating conditions. In one case of interest, the heat input approximated a sine law distribution along the heated length. Data furnished were inadequate for calculating heat transfer coefficients but the experiments are of interest in a qualitative way.

Kratz, Schlegel, and Christ[2] reported some work on thermal transfer between two aluminum surfaces under water. Figure 2-1 shows a schematic representation of the apparatus. The heat transfer coefficient across the interface made by two "rough" aluminum surfaces in contact under water was 6000 Btu/hr. ft.^{2} $^{\circ}$F. when the water temperature was about 160 $^{\circ}$F. Changing the pressure which held the surfaces together from 11.3 lb./in.^{2} to 40.4 lb./in.^{2} showed no effect on the heat transfer. When the surfaces were separated by shims to give a definite water thickness (0.0045 in. or 0.0114 cm.), the experimental values of the heat transfer coefficients were 20 - 30% greater than could be accounted for by thermal conductivity alone. This discrepancy was believed to be the result of additional heat transfer by the water at the edges of the interface.

The heat transfer coefficient across the water film between two "smooth" aluminum surfaces was of the order of 21400 Btu/hr. ft.^{2} $^{\circ}$F. The thickness of this water layer was not established but it is of interest that no measurable effect resulted from coating one of the aluminum surfaces with glyptal resin paint.

No definition was given for "rough" and "smooth", so these data must be considered as qualitative.

Drawing # 6680



SCHEMATIC DIAGRAM OF APPARATUS USED TO GET HEAT TRANSFER BETWEEN ALUMINUM SURFACES UNDER WATER.

(CP - 2313)

FIGURE 2 – 1

Stromquist has recently reported on heat transfer to water and air in a rectangular channel. The fluids were circulated by natural convection with two mechanisms: (1) natural convection confined only to the channel, and (2) natural convection in the form of a thermal syphon in the system. No heat transfer coefficients were reported, but the data indicate the flux levels which may be achieved with each mechanism. Utilizing the thermal syphon effect with water, the flux could be maintained at 27400 Btu/hr. ft.^{2} , the limit of the experimental equipment used. Natural convection within the channel permitted fluxes of the order of 18000 Btu/hr. ft.^{2} without vapor binding. Use of air in a thermal syphon limited the flux to less than 50 Btu/hr. ft.^{2} even when the wall temperature was increased to 1220 °F. (660 °C).

In a later memorandum, Stromquist[4] estimated the water velocities developed in the assembly by the thermal syphon to be from 0.2 to 0.6 feet per second.

Use of the thermal syphon in a homogeneous pile has been discussed in two project reports by Quinn[5,6] .

In non-project reports, Touloukian, Hawkins, and Jakob[7] presented data and correlations on free convection heat transfer from vertical surfaces to liquids, and Kern and Othmer[8] discussed the effect of free convection of heat transfer to fluids in viscous flow.

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 3

HEAT TRANSFER TO GASES BY COMBINED CONDUCTION AND NATURAL CONVECTION

W. B. Harrison
October 1, 1948

PART 3

HEAT TRANSFER TO GASES BY COMBINED CONDUCTION AND NATURAL CONVECTION

Heat transfer to gases by combined conduction and natural convection takes place to some degree in many practical heat exchange systems. The following paragraphs are devoted to experiments which employed this mechanism in heating air and other gases or in transferring heat across gas layers.

Data of Kratz[1], and Raeth[2], are in good agreement on the coefficients for heat transfer across an air film between aluminum discs. For discs held together with pressure P, the approximate heat transfer coefficients across an interface are shown on the following table:

Surface condition	P - lb./in.^{2}	h_i -Btu/hr.ft.^{2} °F.
normally air-oxidized	20.4	2050
	41.6	2900
electrolytically oxidized in oxalic acid	20.4	1100
	41.6	1250
	104.4	1450

Kratz and Christ[3] reported heat transfer across a gap between a cylindrical graphite wall and an inner brass tube. Using nitrogen, helium, and argon in the gap, they found that the heat transfer calculated from the thermal conductivity and gap thickness was in close agreement with experimental data as long as the inner tube was centered in the channel.

When the inner tube was permitted to touch the graphite wall, the average heat transfer increased by a factor of 1.67.

CP-214[4] and CP-235[5] also include data concerning heat transfer across a small gap between graphite and copper.

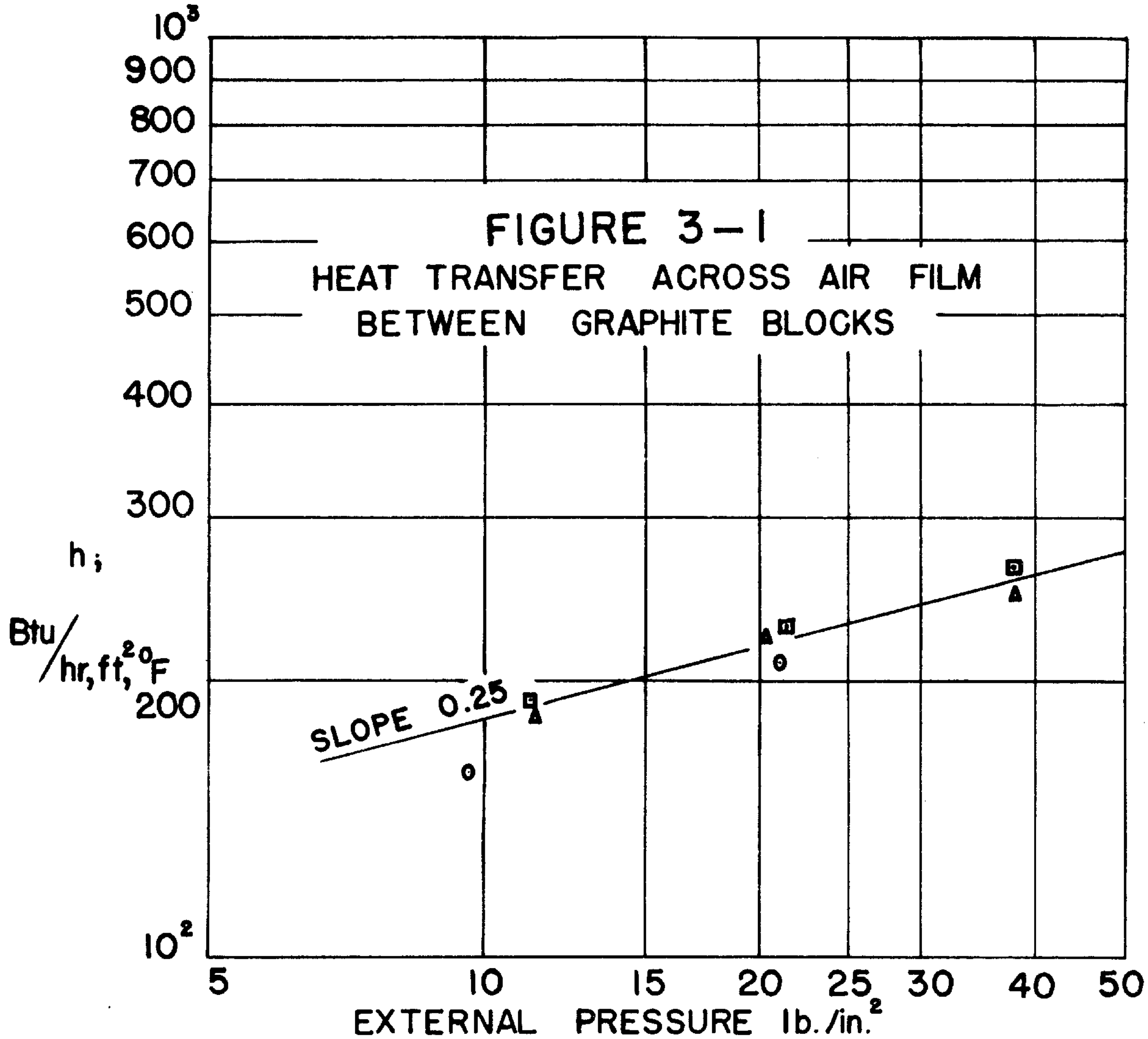
Data of Bankoff and Rosner (summarized in CE-2271[6]) for the case of heat transfer across an air film between graphite blocks are taken from CS-1568[7], CS-1474[8], and CN-1369[9], and are shown in Figure 3-1. Additional data were reported in CP-235[5] for heat transfer between graphite blocks with air, helium, and hydrogen at the interface, and in CE-1217[10] with air at the interface. The data for the cases with air at the interface are not in agreement with those shown in Figure 3-1, but the comparisons with helium and hydrogen are of qualitative value. The discrepancies were believed to result from the fact that data of Figure 3-1 were obtained using more accurately planed graphite surfaces than were used in the other experiments.

Data of CP-235[5] for gas pressure of one atmosphere are shown below:

gas	h_i - Btu/hr.ft.^{2} °F.
air	47
helium	118
hydrogen	155

[redacted]

Drawing # 6681



DATA OF BANKOFF AND ROSNER

REFERENCES

O	CN - 1369
▫	CS -1474
Δ	CS -1368

[redacted]

Smetana and Cabell[11] described an interesting technique for measuring thermal conductivity of compressed magnesium oxide pellets. Compression of the pellets and experimental determination of the conductivity were performed in the same apparatus. Although a complete discussion of the work appears to be beyond the scope of this report, mention of the work is made to facilitate location of the information by those who may be interested. As noted in Part 2 of this report, Stromquist[12] performed experiments to determine heat fluxes to water and air circulated by natural convection in a rectangular channel. Heat fluxes were less than 50 Btu/hr. ft.^{2} for the experiments in which air was circulated by a thermal syphon. These experiments are to be extended in the near future.

Mueller[13] reported data on heat transfer from a vertical wire to air circulated by natural convection. Temperature drop from the wire to surrounding air ranged from about 2 to 212° F. and Nu ranged from about 0.32 to 0.7 while the product $Gr \times Pr$ varied from 3.2×10^{-7} to 10^{-2}.

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ORNL 156-4

HEAT TRANSFER IN MANHATTAN DISTRICT AND TOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 4

HEAT TRANSFER TO LIQUID METALS BY COMBINED CONDUCTION AND FORCED CONVECTION

W. B. Harrison

October 1, 1948

[redacted]

PART 4

HEAT TRANSFER TO LIQUID METALS BY COMBINED CONDUCTION AND FORCED CONVECTION

Use of liquid metals as coolants permits high heat fluxes and high operating temperatures. These characteristic features give liquid metals much promise for use in power producing reactors. Data are confined at present to sodium-potassium alloy and mercury.

Sodium Potassium Alloy

Data on heat transfer to sodium-potassium alloy may be found in NP-10[1], NRL-C-3243[2], and CF-3746[3].

The data of NP-10 were presented by R. C. Werner of the Mine Safety Appliance Company on experiments utilizing sodium-potassium alloy containing 44% potassium by weight. A "figure of eight" type, double-pipe heat exchanger system was used, as shown schematically on Figure 4-1. As indicated in the table below, experimental runs reported in NP-10 covered a rather large range of variables*:

average temperature	100 - 1024 °F.
mass rate	194 - 293 lb./hr.
overall coefficient	148 - 1165 Btu/hr. ft.^{2} °F.
Prandtl number	0.041 - 0.0044
mass velocity - G_o(in annulus)	25,100 - 379,300 lb./hr.ft.^{2}
G_i(in center tube)	76,800 - 1,160,000 lb./hr.ft.^{2}
Reynolds number - Re_o(in annulus)	789 - 70,500
Re_i(in center tube)	2,960 - 228,300

(All values shown are taken from the report.)

*Complete tables of nomenclature are found in Part 18 of this report.

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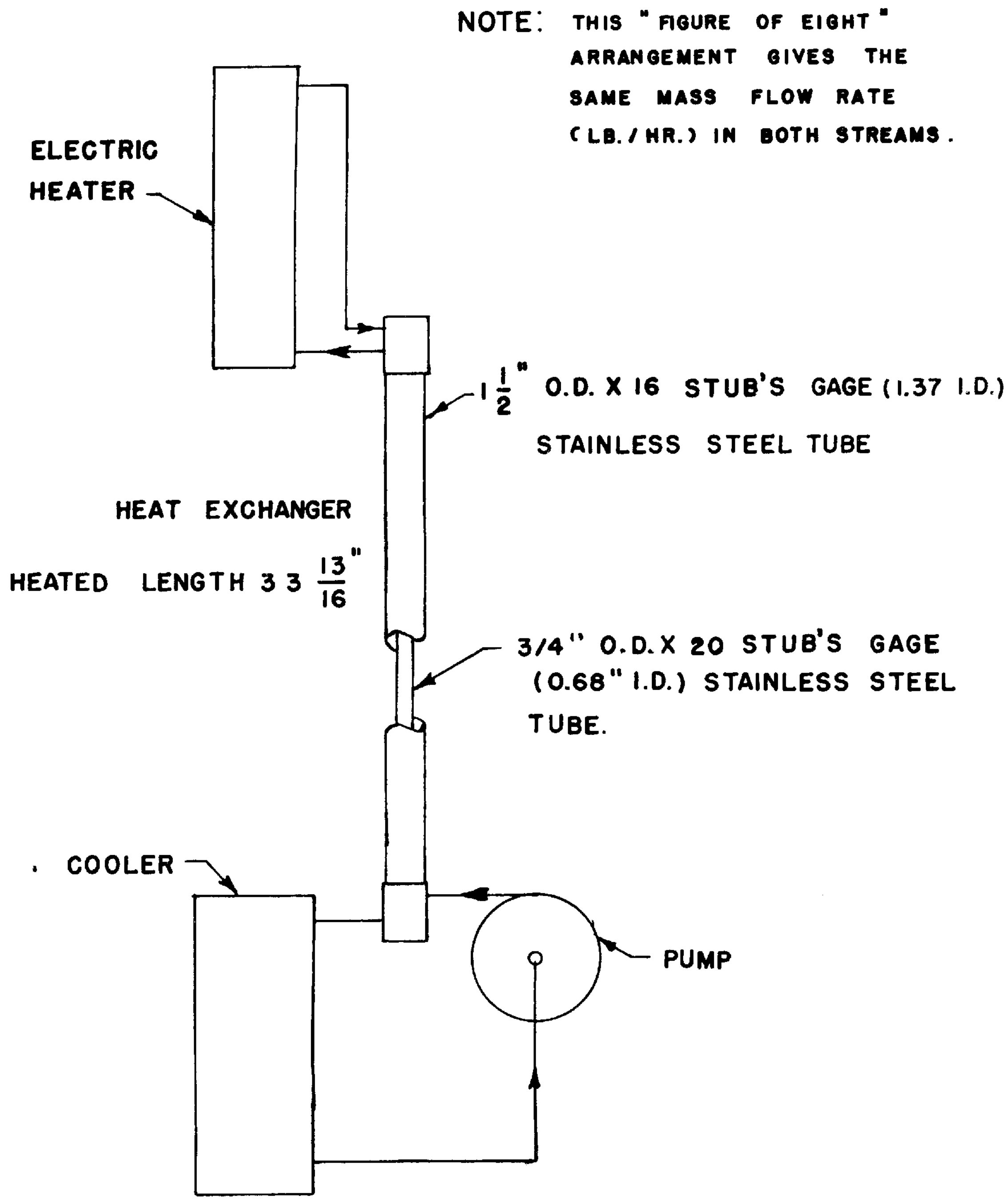


FIGURE 4-1

SCHEMATIC DIAGRAM OF NaK HEAT TRANSFER APPARATUS.
(NP-10)

[redacted]

Data were analyzed in the report by assuming the Colburn equation[4],

$$\frac{h}{cG} = 0.023 \left(\frac{DG}{\mu}\right)^{-0.2} \left(\frac{c\mu}{k}\right)^{-0.67} \qquad \text{(Eq. 4.01)}$$

to apply for the center tube, and the Monrad-Pelton correlation[5],

$$\frac{h}{cG} = \frac{0.020\ (D_2/D_1)^{0.53}}{\left(\frac{D_e\ G}{\mu}\right)^{0.2} \left(\frac{c\mu}{k}\right)^{0.67}}\ , \qquad \text{(Eq. 4.02)}$$

to hold for the annulus. Since both streams had the same mass flow rate, the equations were combined and a modified Wilson line plot[6] was used to evaluate the film coefficients. Recent papers[7,8,9] indicate that there is no theoretical justification for assuming that the equations shown above are valid for use with liquid metal systems. Consequently, it is believed that all values of film coefficients reported in NP-10 are subject to large discrepancies because of the method of analysis employed. This is evidenced by the fact that it was necessary to account for over half of the thermal resistance between the streams with a fouling factor in spite of the careful cleaning operations described in the report and the fact that NaK is generally considered to wet metal surfaces well. In a recent personal communication, the experimenters attributed this to poor heat balances and inaccurate temperature measurements resulting from poor mixing.

Using helpful suggestions made by J. R. Menke** and R. N. Lyon*, the data of NP-10 can be re-evaluated by means of a comparison with predictions from a paper by R. C. Martinelli[7] on the analogy between momentum and heat transfer. The following assumptions are made:

1. No fouling
2. Good wetting of the walls by the NaK
3. Physical property data are reliable
4. Experimental data are reliable
5. No effect of heated length/diameter ratio

The Reynolds number in the annulus (Re_o) was about one-third the Reynolds number in the center stream (Re_i). Since Martinelli's considerations were developed for turbulent flow, all runs having $Re_o < 10^4$ are discarded. The average difference in temperature of the two streams ranged from 10 to 30° F., making the difference in the heat capacity of the alloy in the streams very small. Since the temperature rise of each stream is dependent on mass flow rate and heat capacity, it is seen that temperature changes in both streams should correspond. This consideration leads to the elimination of a few more runs which had a poor heat balance. For purposes of this examination all runs having $\frac{t_H}{t_C} < 0.90$ are discarded. (Subscript H refers to the hot stream; C refers to the cold stream.)

*Oak Ridge National Laboratory, Oak Ridge, Tennessee

**Nuclear Development Associates, Inc., 33 W. 60th St., New York 23, New York

As suggested by Lyon[9], Martinelli's calculations[7] may be approximated by a smooth curve of Nu versus α Pe* as shown on Figure 4-2. The circular tube calculations may be assumed to apply to the center tube of the apparatus but there is some doubt as to the best way to estimate heat transfer in the annulus. Figure 4-2 also shows Martinelli's[7] predictions for a system composed of fluid passing between two parallel plates having the same heat flux (symmetrical flux), and the calculations of Harrison and Menke[10] for a parallel plate system having no flux through one wall (asymmetrical flux). As a first attempt, it is decided to use circular tube predictions for the annulus, using the equivalent hydraulic diameter in place of diameter.

Since no fouling is assumed, overall thermal resistance is equal to the sum of the film resistances and the tube wall resistance[11]. The center tube wall temperature was not measured in the experiments, so the data are evaluated in terms of total film resistance. The overall resistance was measured experimentally and the tube wall resistance was calculated with reasonable accuracy. Thus, the experimental evaluation of total film resistance is the difference between the overall resistance and the tube wall resistance. The theoretical total film resistance is obtained by evaluating the inside and outside film coefficients (h_i and h_o, respectively) by means of Figure 4-2. From these values, the total film resistance is calculated. Overall resistance, tube wall resistance, and inside film resistance are all based on the inside surface area

*Definition and influence of α will be discussed later. For the present, it may be assumed to equal unity.

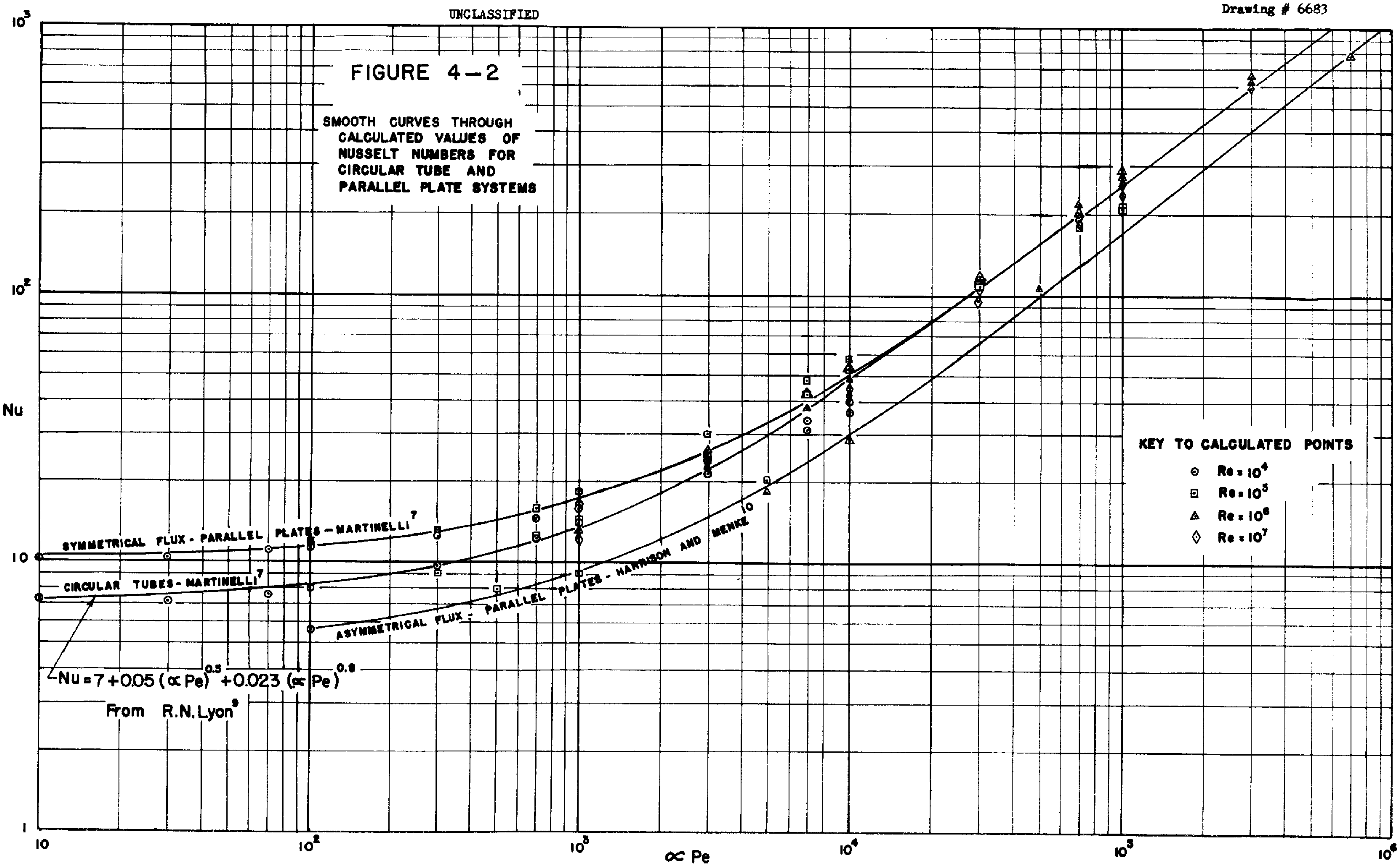


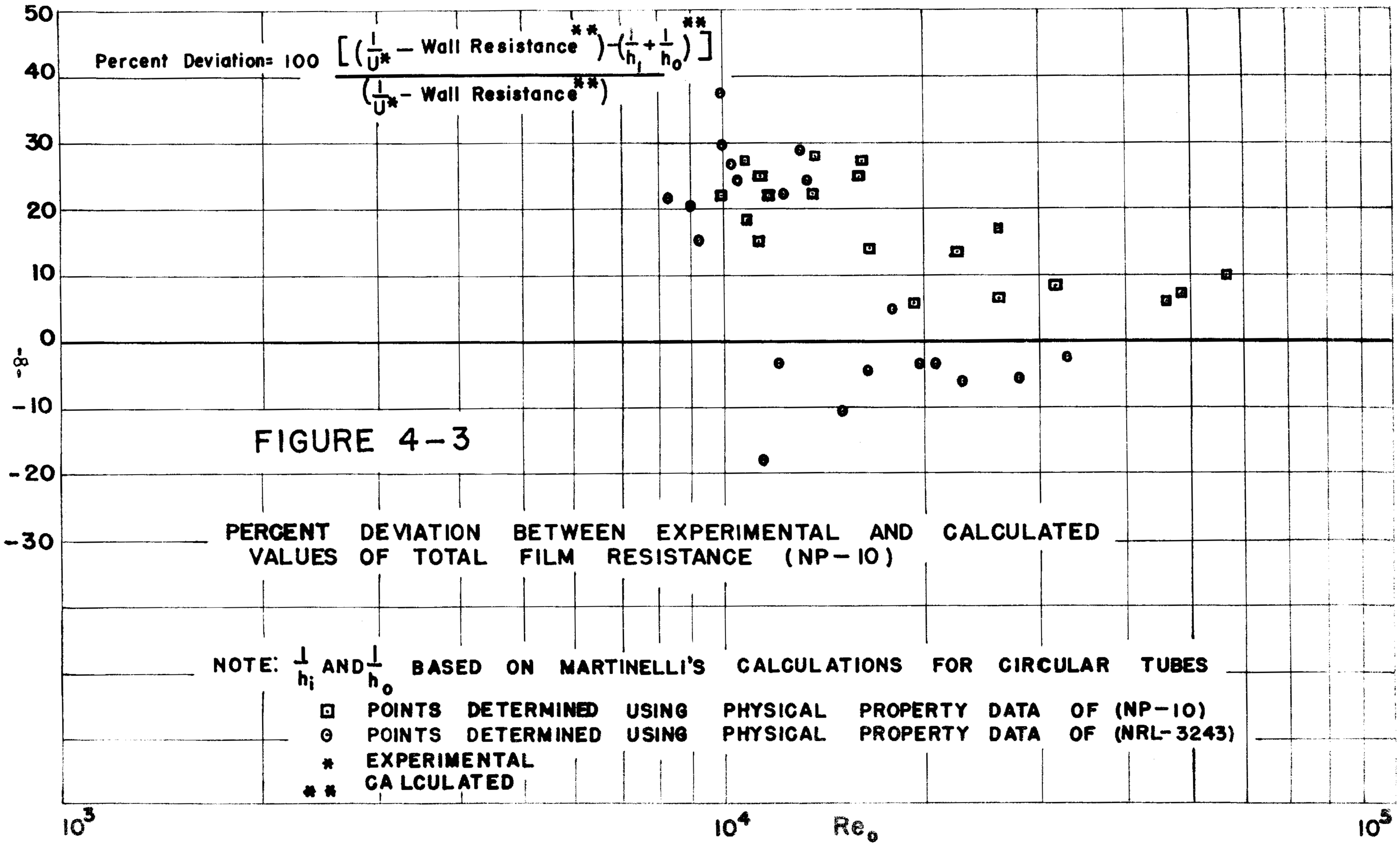
FIGURE 4-2
SMOOTH CURVES THROUGH CALCULATED VALUES OF NUSSELT NUMBERS FOR CIRCULAR TUBE AND PARALLEL PLATE SYSTEMS
SYMMETRICAL FLUX - PARALLEL PLATES - MARTINELLI 7
CIRCULAR TUBES - MARTINELLI 7
ASYMMETRICAL FLUX - PARALLEL PLATES - HARRISON AND MENKE 10
Nu = 7 + 0.05 (∝ Pe)^0.5 + 0.023 (∝ Pe)^0.8
From R.N. Lyon 8
KEY TO CALCULATED POINTS
Re = 10^4
Re = 10^5
Re = 10^6
Re = 10^7
Nu
∝ Pe
10^3
10^2
10
1
10
10^2
10^3
10^4
10^5
10^6

of the center tube. The outside film resistance is not corrected to this basis since the correction factor is small compared with the probable error in applying circular tube calculations to the annulus; considering also the probable error in experimental data, physical properties data, etc. Figure 4-3 shows the results of this analysis presented as percent deviation between experimental and calculated total film resistance versus Re_o, the Re in the annulus. Data are plotted against Re_o because Re_i is about three times Re_o and the system is definitely in turbulent flow for $Re_o > 10{,}000$.

More reliable physical properties data appeared in NRL-C-3243 than were in NP-10, so the calculations mentioned above are repeated using the revised data. These revised calculations are also shown on Figure 4-3. New specific heat values changed the values of overall coefficient, and new viscosity values changed the values of Reynolds numbers, but the magnitude of deviations between experimental and calculated total film resistance are not appreciably affected. The agreement would not be improved by using parallel plate calculations for the annulus, and, in view of the possible error in physical properties and experimental data, it is believed that the analysis given is adequate.

For the experimental runs reported in NRL-C-3243, the equipment as described in NP-10 and shown on Figure 4-1 was slightly modified and used again. The stainless steel heat exchanger unit was replaced with a type "A" nickel unit having essentially the same dimensions except for the center tube which now had an O.D. of 0.75 in. and an I.D. of 0.70 in. compared with the stainless steel tube having the same O.D. but an I.D.

Drawing # 6684



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of 0.68 in. Other slight modifications were introduced in an effort to make the equipment more versatile and the data more accurate. The experimental ranges of variables reported in NRL-C-3243 were as follows:

average temperature		301 - 1206 °F.
mass rate		780 - 6100 lb./hr.
overall coefficient		1117 - 3240 Btu/hr.ft.^{2} °F.
Prandtl number		0.0080 - 0.0212
mass velocity	G_o (in annulus)	108,500 - 848,000 lb./hr.ft.^{2}
	G_i (in center tube)	292,000 - 2,280,000 lb./hr.ft.^{2}
Reynolds number	Re_o (in annulus)	4,720 - 81,200
	Re_i (in center tube)	15,100 - 250,000

In addition to the method of analysis used in NP-10, another method was tried in NRL-C-3243. By assuming that no fouling occurs; that

$$\frac{h_i}{c_i G_i} = a\ (Re_i)^m\ (Pr_i)^n \qquad \text{(Eq. 4.03)}$$

for the center tube, and that

$$\frac{h_o}{c_o G_o} = a\ (0.87)\left(\frac{D_2}{D_1}\right)^{0.53} (Re_o)^m\ (Pr_o)^n \qquad \text{(Eq. 4.04)}$$

for the annulus, it was possible to evaluate a, m, and n by an approximation method. The final equations recommended by Werner[2] as a result of this type of analysis are:

$$\frac{h_i}{c_i G_i} = 0.227\ (Re_i)^{-0.5}\ (Pr_i)^{-0.69} \qquad \text{(Eq. 4.05)}$$

[redacted]

and $$\frac{h_o}{c_oG_o} = 0.1975\left(\frac{D_2}{D_1}\right)^{0.53}(Re_o)^{-0.5}(Pr_o)^{-0.69} \qquad \text{(Eq. 4.06)}$$

The authors of <u>NRL-C-3243</u> pointed out that they were aware of Martinelli's work[7] and expected to use his calculations in correlating data for a forthcoming report. Equations 4.05 and 4.06 fall remarkably close to theoretical predictions[7,9,10] as shown on Figure 4-4 and 4-5.

It is of particular interest to note that in the case of the annulus, the data as calculated by Equation 4.06 fit the predicted values for circular tubes slightly better than either the calculations of Martinelli[7] for the symmetrical flux, parallel plate system or the calculations of Harrison and Menke[10] for the asymmetrical flux, parallel plate system.

It is decided to attempt an examination of total film resistance from the data of <u>NRL-C-3243</u> in the same manner used for the data of <u>NP-10</u>. Using the calculations made for circular tubes as plotted on Figure 4-2, total film resistance is calculated and compared with the difference between overall thermal resistance and calculated wall resistance. Percent deviation is plotted against Re_o on Figure 4-6. In making calculations for Figure 4-6, it was seen that the difference between calculated total film resistance and "experimental" total film resistance is almost constant at $1.0 \times 10^{-4}\ \frac{\text{hr.ft.}^2\ {}^\circ\text{F.}}{\text{Btu}}$, so this constant R was applied and the calculations are repeated. Resulting points are plotted on Figure 4-7, and it is seen that deviations are much smaller than before. If one uses the asymmetrical flux, parallel plate calculations of Harrison and Menke[10] for the annulus, it is seen that a constant R of $2.0 \times 10^{-4}\ \frac{\text{hr. ft.}^2\ {}^\circ\text{F}}{\text{Btu}}$ brings the data

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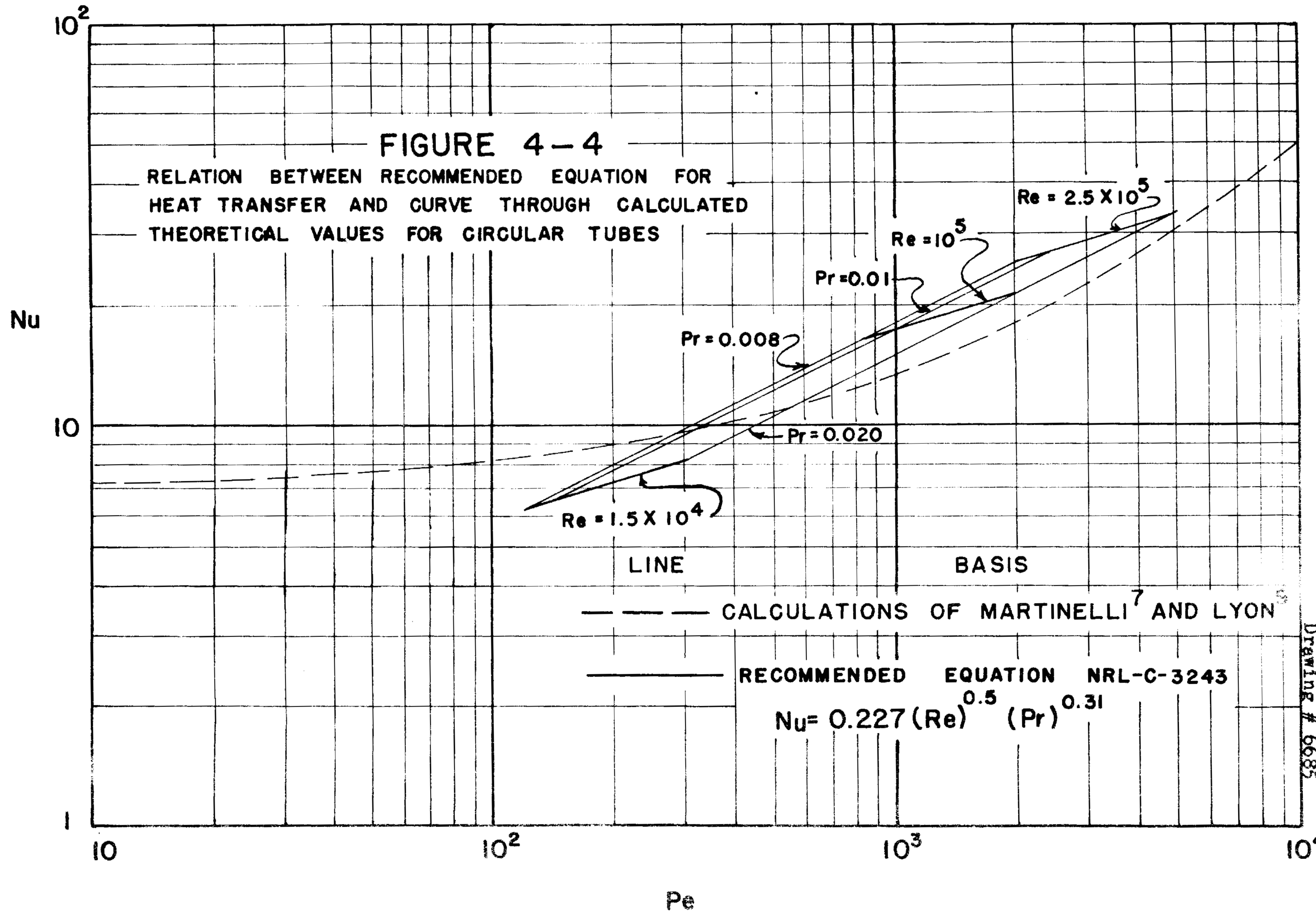

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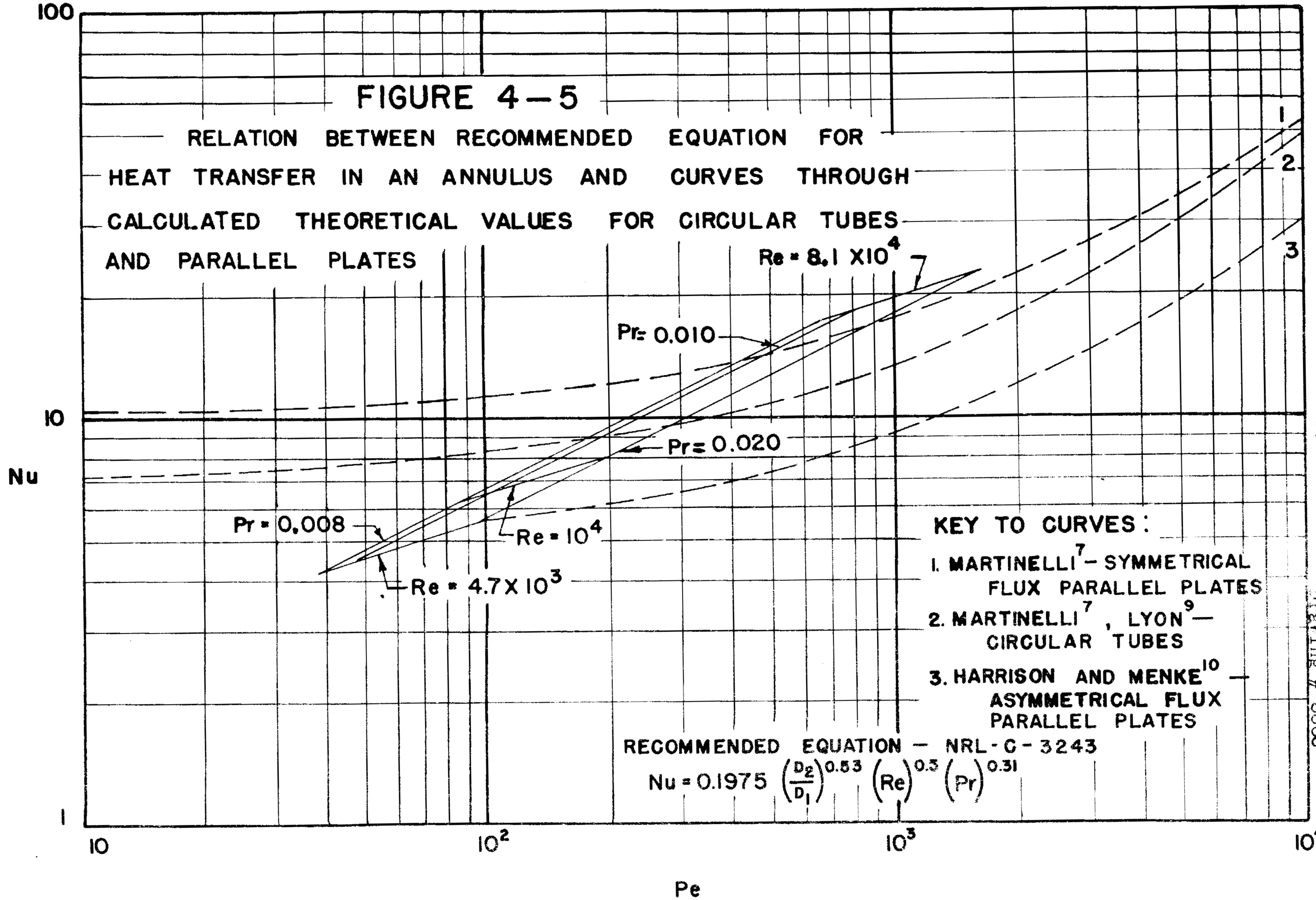

FIGURE 4—5

RELATION BETWEEN RECOMMENDED EQUATION FOR HEAT TRANSFER IN AN ANNULUS AND CURVES THROUGH CALCULATED THEORETICAL VALUES FOR CIRCULAR TUBES AND PARALLEL PLATES

$$Nu = 0.1975 \left(\frac{D_2}{D_1}\right)^{0.53} (Re)^{0.5} (Pr)^{0.31}$$

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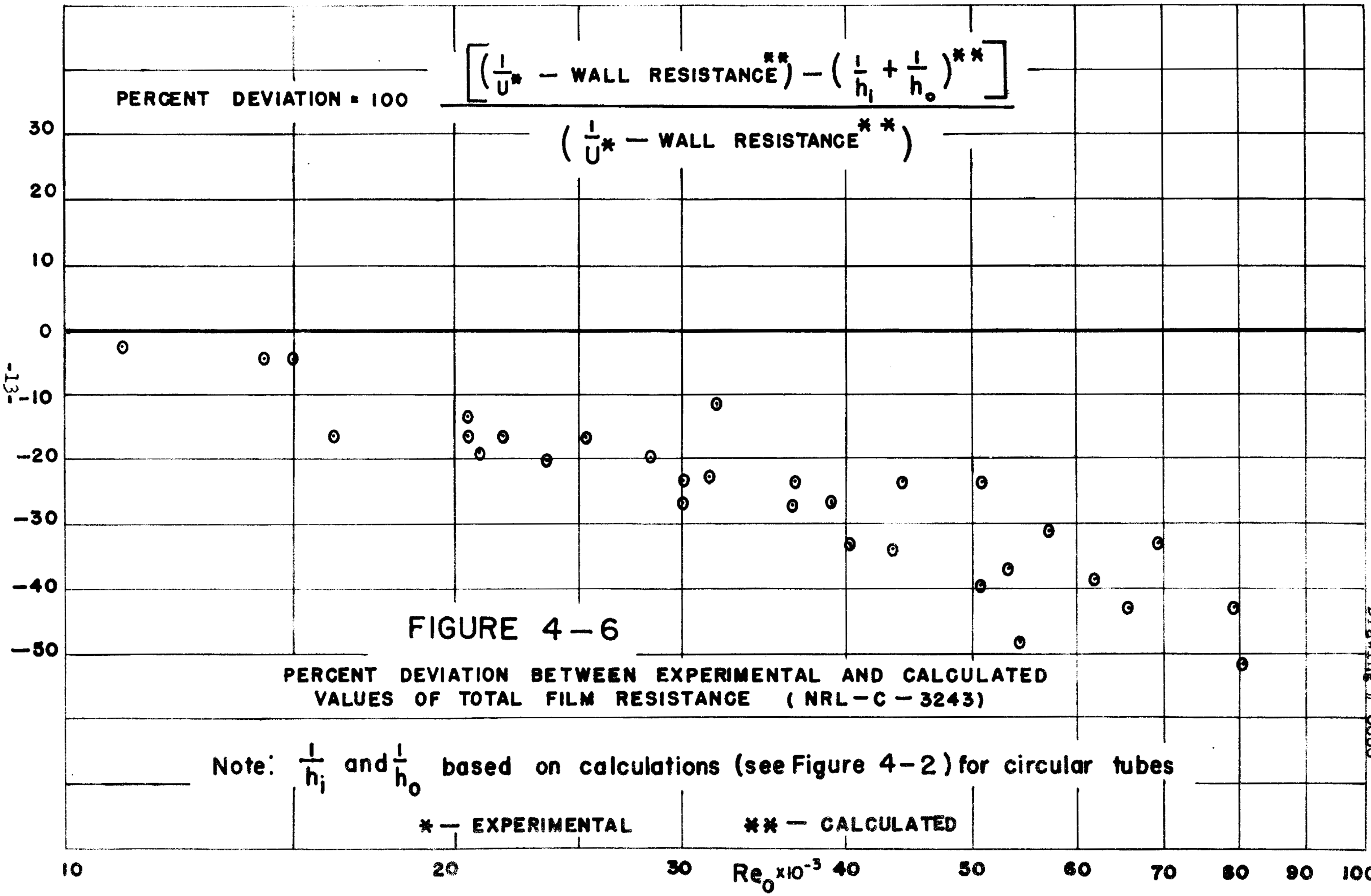

FIGURE 4-6

PERCENT DEVIATION BETWEEN EXPERIMENTAL AND CALCULATED VALUES OF TOTAL FILM RESISTANCE (NRL-C-3243)

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$$\text{PERCENT DEVIATION} = 100\,\frac{\left[\left(\frac{1}{U}{*} - \text{WALL RESISTANCE}^{**}\right) - \left(\frac{1}{h_i} + \frac{1}{h_o}\right)^{**} + \Delta R\right]}{\left(\frac{1}{U}{*} - \text{WALL RESISTANCE}\right)^{**}}$$

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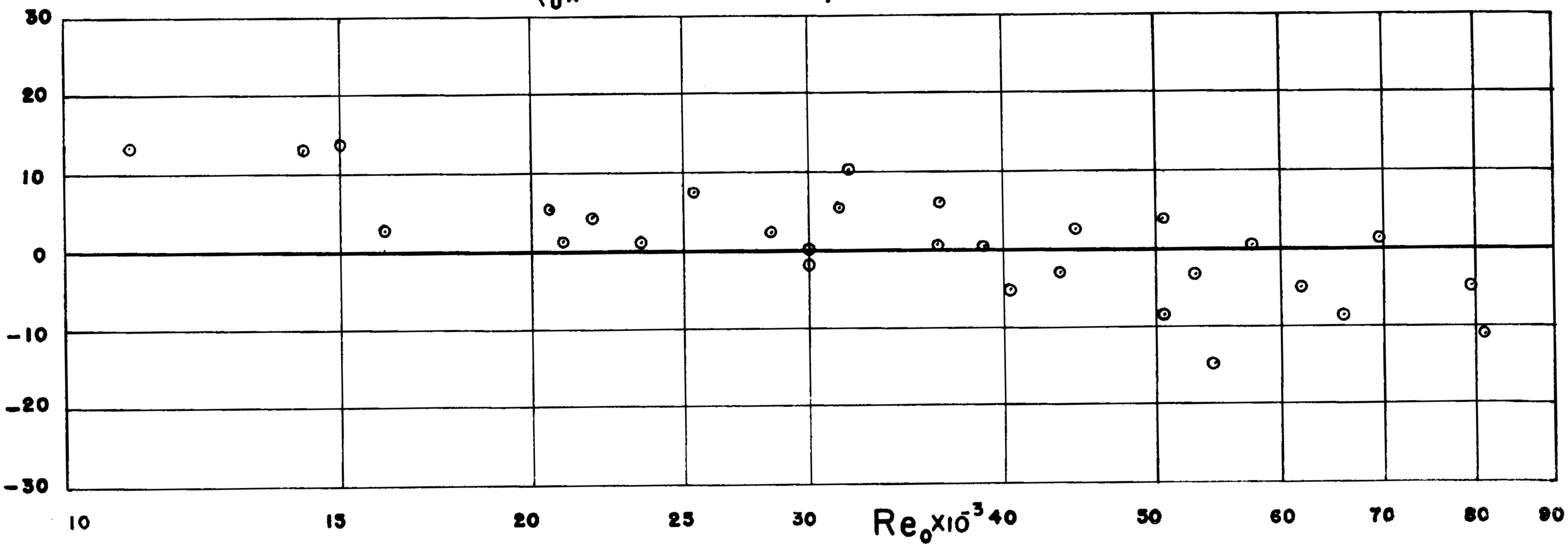

FIGURE 4-7

PERCENT DEVIATION BETWEEN EXPERIMENTAL AND CALCULATED VALUES OF TOTAL FILM RESISTANCE (NRL-C-3243)

NOTE: $\frac{1}{h_i}$ AND $\frac{1}{h_o}$ BASED ON CALCULATIONS (SEE FIG. 4-2) FOR CIRCULAR TUBES; ΔR ASSUMED TO BE 1.0×10^{-4} $\frac{\text{HR. FT.}^2\ {}^\circ\text{F.}}{\text{BTU.}}$

* — EXPERIMENTAL

** — CALCULATED

together as shown on Figure 4-8. No R correction is necessary if Martinelli's[7] symmetrical flux, parallel plate calculations are used for the annulus, but the spread of deviations is rather large as shown on Figure 4-9.

In order to investigate the characteristics of R, experimental and calculated values of total film resistance are plotted vs. Re_o for runs 9-14 on Figure 4-10. Calculated values are obtained by applying theoretical circular tube calculations[7,9] to the inside tube, and applying three sets of calculations to the annulus, as follows:

1. Calculations of Martinelli[7] and Lyon[9] for circular tubes.
2. Calculations of Martinelli[7] for symmetrical flux, parallel plates.
3. Calculations of Harrison and Menke[10] for asymmetrical flux, parallel plates.

In these experiments, typical overall resistance ranged from about 3.3×10^{-4} to about 6.7×10^{-4} $\frac{\text{hr. ft.}^2\ ^\circ\text{F}}{\text{Btu}}$, so R of 1.0×10^{-4} $\frac{\text{hr. ft.}^2\ ^\circ\text{F.}}{\text{Btu}}$ represents 15 to 30% of the total resistance. Calculated wall resistance was of the order of 0.6×10^{-4} $\frac{\text{hr. ft.}^2\ ^\circ\text{F}}{\text{Btu}}$. Since wall resistance is small compared with total resistance, and since R does not appear to fluctuate with any of the operating variables, it seems likely that an error is entering into the calculated total film resistances.

In the theoretical work drawing the analogy between heat and momentum transfer[7,8,9,10], the ratio, $\frac{\epsilon_H\text{, eddy diffusivity of heat}}{\epsilon_M\text{, eddy diffusivity of momentum}}$, has been assumed constant and called α. Martinelli[7] pointed out that

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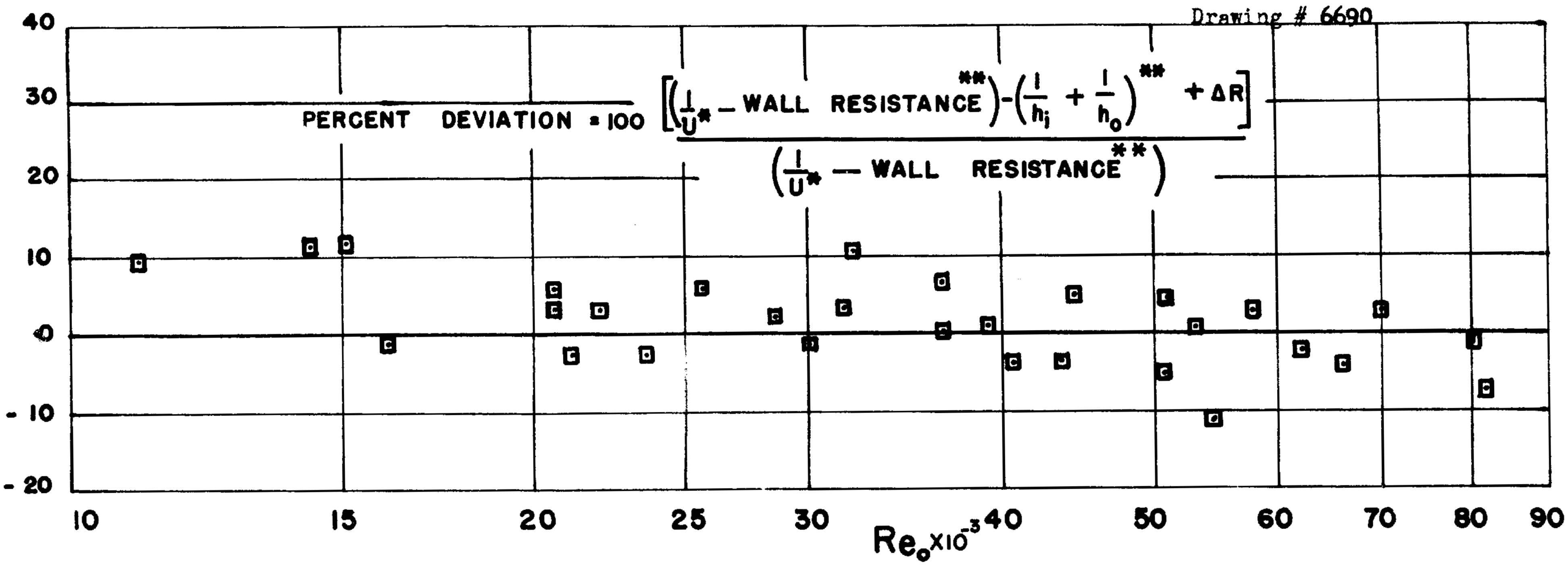

FIGURE 4—8

PERCENT DEVIATION BETWEEN EXPERIMENTAL AND CALCULATED VALUES OF TOTAL FILM RESISTANCE

NOTE: $\frac{1}{h_i}$ BASED ON CALCULATIONS (SEE FIG. 4-2) FOR CIRCULAR TUBES

$\frac{1}{h_o}$ BASED ON CALCULATIONS (SEE FIG. 4-2) FOR ASYMMETRICAL FLUX—PARALLEL PLATES

ΔR ASSUMED TO BE 2.0×10^{-4} $\frac{\text{HR. FT.}^2 \, {}^\circ\text{F}}{\text{BTU.}}$

* - EXPERIMENTAL

** - CALCULATED

[redacted]

$$\text{PERCENT DEVIATION} = 100\ \frac{\left[\left(\frac{1}{U^{*}} - \text{WALL RESISTANCE}^{**}\right) - \left(\frac{1}{h_i} + \frac{1}{h_o}\right)^{**}\right]}{\left(\frac{1}{U^{*}} - \text{WALL RESISTANCE}^{**}\right)}$$

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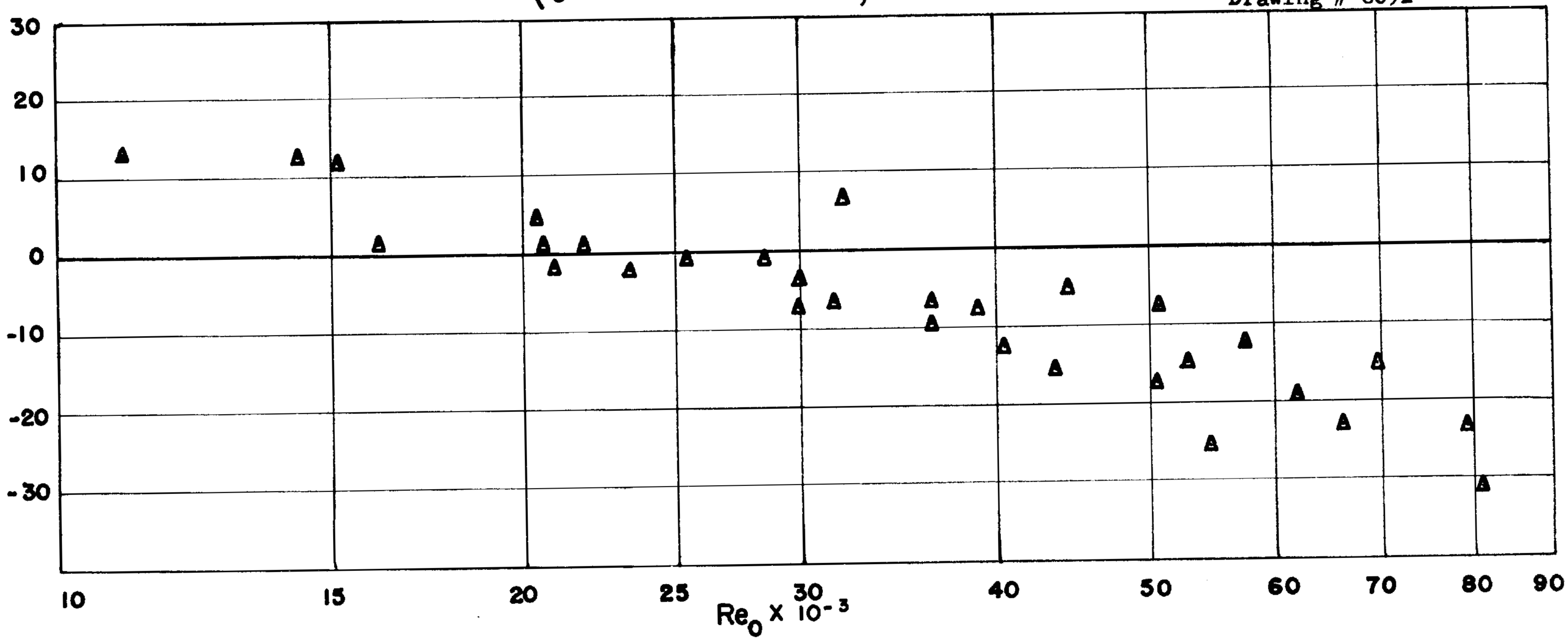

FIGURE 4–9

PERCENT DEVIATION BETWEEN EXPERIMENTAL AND CALCULATED VALUES OF TOTAL FILM RESISTANCE (NRL–C–3243)

NOTE: $\frac{1}{h_i}$ BASED ON CALCULATIONS (SEE FIG. 4 – 2) FOR CIRCULAR TUBES

$\frac{1}{h_o}$ BASED ON CALCULATIONS (SEE FIG 4 – 2) FOR PARALLEL PLATES— SYMMETRICAL FLUX

* – EXPERIMENTAL

** – CALCULATED

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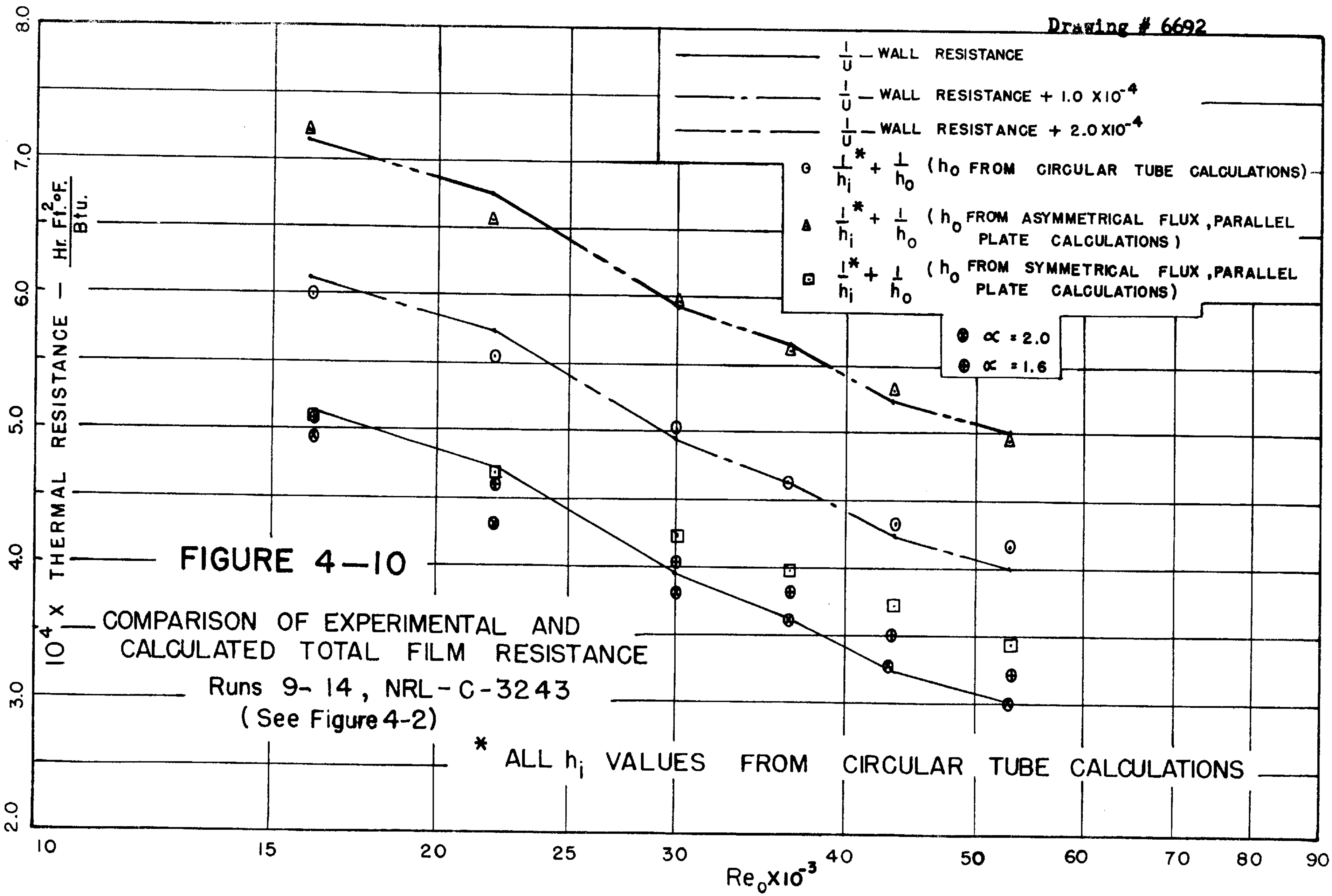

FIGURE 4—10

COMPARISON OF EXPERIMENTAL AND CALCULATED TOTAL FILM RESISTANCE

Runs 9- 14, NRL-C-3243

(See Figure 4-2)

Von Karman[12] postulated that ϵ_H and ϵ_M are equal; Reichardt[13] suggested that $\alpha = 1.4$ to 1.5; Sheppard[14] found α to equal about 1.6. In the paper by Martinelli[7], α was assumed to equal unity, but, in the paper by Lyon[9], α was retained in the final calculations. The equation Lyon has given the smooth curve through values calculated for circular tubes (see Fig. 4-2) is:

$$Nu = 7 + 0.05\ (\alpha\ Pe)^{0.5} + 0.023\ (\alpha\ Pe)^{0.8} \quad \text{(Eq. 4.07)}$$

If one assumes $\alpha = 1.6$ and $\alpha = 2.0$, and uses Equation 4.07 for determining both the center tube and the annulus coefficients, the calculations may be repeated for runs 9-14 and these points are also shown on Figure 4-10. In view of the improved agreement between experimental and calculated total film resistance which results from this approach, all runs reported in <u>NRL-C-3243</u> are evaluated, letting $\alpha = 1.6$. These values, shown on Figure 4-11, should be compared with values determined letting $\alpha = 1.0$, shown on Figure 4-6.

These data indicate that α may actually be a constant at 1.6. However, the only definite conclusion which may be drawn from the analysis is that, for the range of conditions studied, NaK data may be correlated by Equation 4.07 if α is assumed to be 1.6. From the pattern which the deviations made on Figure 4-11, it is indicated that the correlation can be improved by slightly changing Equation 4.07, but it is not considered to be justified at present.

The effect of length/diameter ratio has not been treated. Martinelli[7] mentioned work of Sanders[15] which would indicate that the

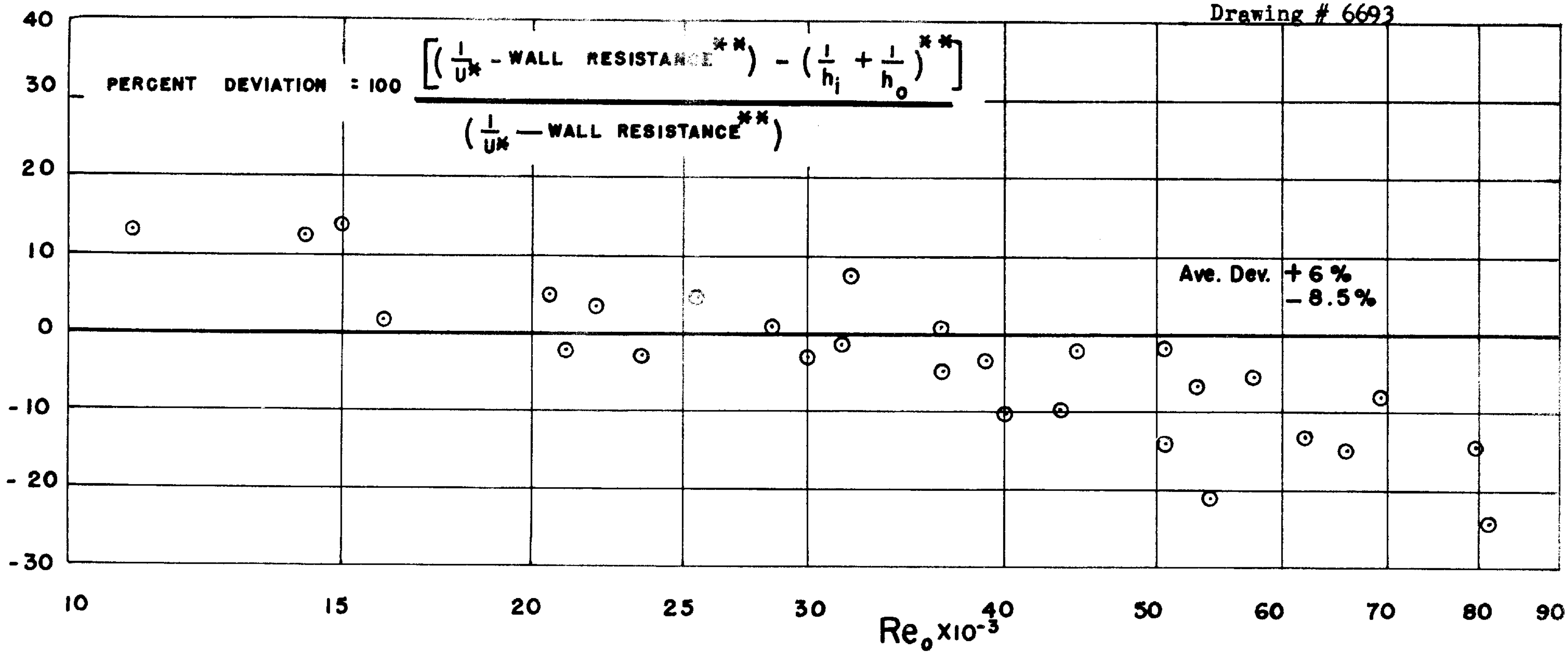

FIGURE 4—11

PERCENT DEVIATION BETWEEN EXPERIMENTAL AND CALCULATED VALUES OF TOTAL FILM RESISTANCE (NRL-C-3243)

NOTE: $\frac{1}{h_i}$ AND $\frac{1}{h_o}$ BASED ON PLOT OF EQUATION RECOMMENDED BY R.N. LYON[9] FOR CIRCULAR TUBES − αC = 1.6

* EXPERIMENTAL ** CALCULATED

effect is often appreciable, even with large L/D ratios. In the apparatus of NP-10 and NRL-C-3243, L/D in the center tube and in the annulus was about 50. It may be said qualitatively that actual coefficients should be somewhat higher than predicted coefficients and that the amount by which they should be higher increases with Re.

Data of NRL-C-3243 show fairly well that temperature level does not affect the correlation, and this, in turn, indicates that wetting characteristics of the NaK are apparently not dependent on temperature.

The agreement between experimental results of NRL-C-3243 and Equation 4.07 (letting $\alpha = 1.6$) lends much support to the underlying theoretical work. It follows that for similar systems using NaK, Equation 4.07 (letting $\alpha = 1.6$) is recommended until such time as additional information indicates a desirable change.

In the work by Novick, McGinnis, and Litchenberger reported in CF-3746[3], data were taken in a copper cast steel tube heat exchanger having tubes 0.500 in. O.D. x 0.049 in. wall x 51 in. long. Some details of the experiment were mentioned by Novick in another paper[16]. The data were analyzed by means of a Wilson line plot[6], assuming heat transfer coefficients to vary with the eight-tenths power of the stream velocity. This analysis led to the relation $h = 1100 \, (V')^{0.8}$, where h is the NaK film coefficient and V' is the linear stream velocity in ft./sec. Experimental conditions included velocities up to 10 ft./sec.; temperature up to 662° F. (350° C); and Reynolds numbers between 5,000 and 50,000. Variation of coefficients with the eight-tenths power of velocity is not substantiated theoretically for the case of heat transfer to liquid metals. If one assumes that overall coefficients were accurately measured, the difficulty lies in proportioning the thermal resistance to the two liquid films and the intervening metal walls. The Wilson line plot facilitates estimating the separate resistances only if the relation between velocity and coefficient is established. It is believed that, since the two streams did not have the same velocity and were composed of different liquids, the influence of the velocity on the coefficients probably is not the same for both streams. It appears that the proposed equations show one method of proportioning the total thermal resistance which was measured, but it should be kept in mind that, even if the equations are approximately correct, they are specific to the equipment and operating variables and cannot be extended to any other use. The type of heat exchanger used complicates the situation by making it very difficult to perform an accurate calculation of the wall resistance, which would be a means for checking the resistance determined by the Wilson line plot. Insufficient data are included in the reports to permit a comparison with predictions of Martinelli[7] and Lyon[9] as done with the data of NP-10 and NRL-C-3243.

The simple relationship reported ignores the effect of change in physical properties with temperature. It is difficult to see how the coefficient can be represented as a constant times the velocity to some power except for one main stream temperature, which must be constant throughout the series of tests, or for the unique case in which effect of temperature change is reflected in some other variable such as velocity. It is not clear from the reports whether or not this limitation of the method was taken into account in the experiments. The Wilson line plots and a plot of Nu vs. Re are included in the report.

For purposes of comparison, a temperature of 482 °F. (250 °C.) is assumed and calculations of Nu are made for velocities from 1 to 10 ft./sec. using the simple relation $h = 1100 (V')^{0.8}$. The values are plotted as Nu vs. Pe on Figure 4-12, along with Equation 4.07, and the Colburn equation[4]. One must assume a temperature and use Pe as abscissa in order to reduce all three equations to a common basis. Physical properties used in making these calculations are taken from NRL-C-3243.

In view of the limitations mentioned above, the comparison shown on Figure 4-12 is believed to have very little significance.

In the near future, R. N. Lyon* is expecting to take data on heat transfer to NaK alloy in a "figure of eight", double pipe heat exchanger system.

R. J. Musser and W. R. Page[17] prepared a thesis on heat transfer to mercury. A schematic diagram of the apparatus which they used is shown on Figure 4-13.

The heating section was double jacketed; the heat was furnished by dry, saturated, 60 psia steam in which dropwise condensation was promoted by addition of oleic acid. The sum of the thermal resistances of the wall and steam film was obtained by using water in the system instead of mercury and applying the Wilson line plot method[6]. This

*Oak Ridge National Laboratory.

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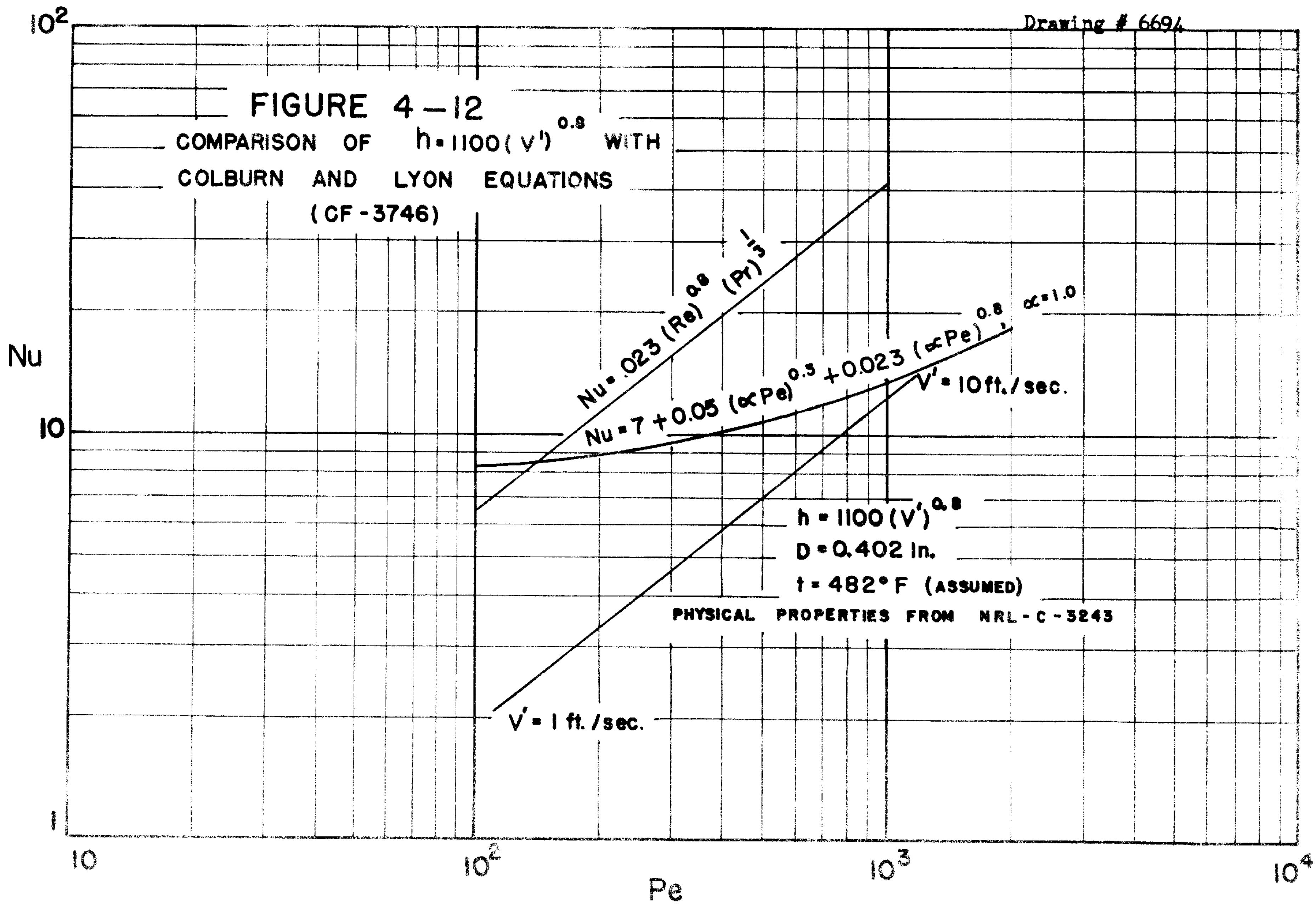

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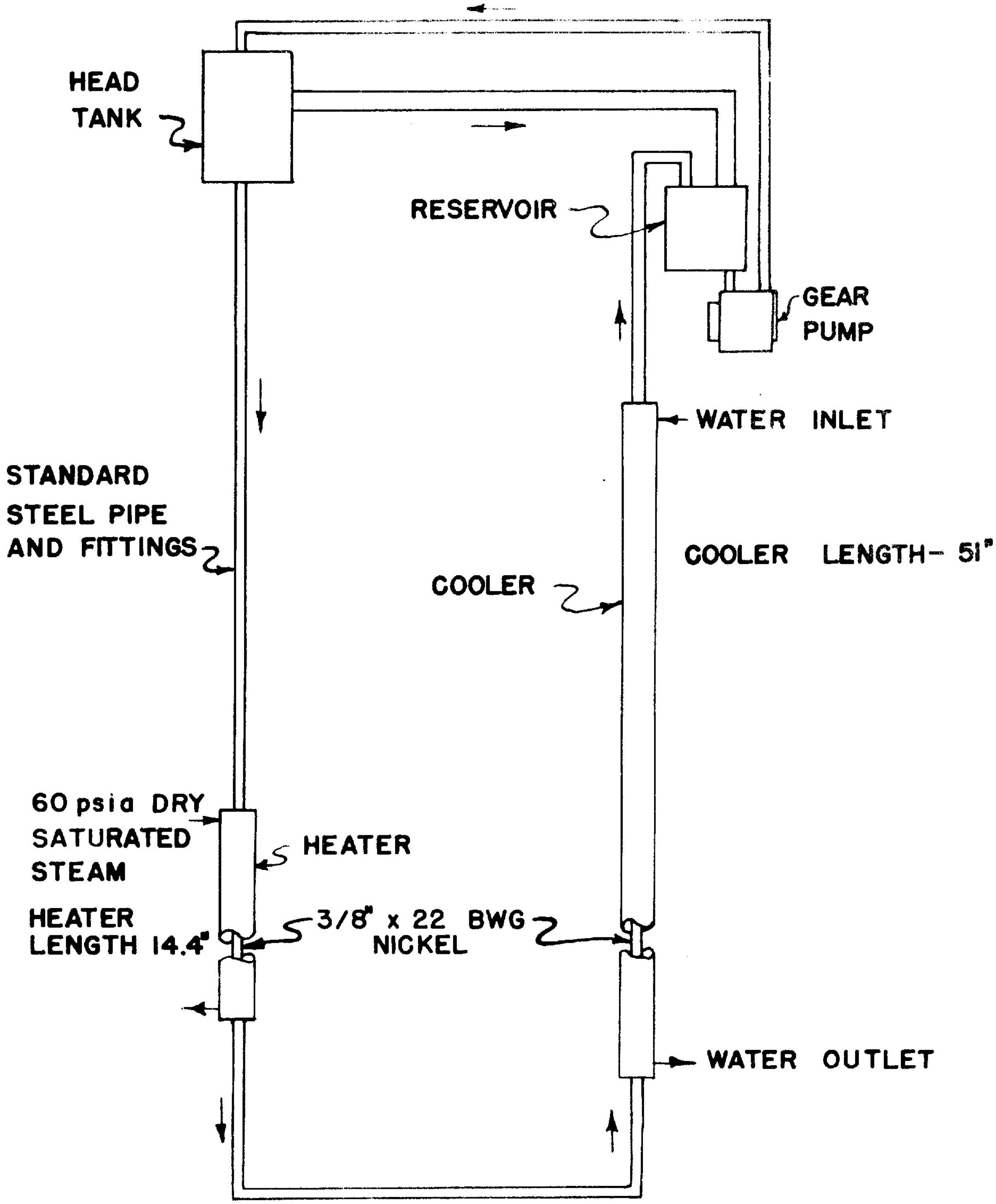

FIGURE 4—13

SCHEMATIC DIAGRAM OF APPARATUS USED BY MUSSER AND PAGE.

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method of analysis appears to be satisfactory, since the variation of film coefficients with the eight-tenths power of velocity is well established for water in turbulent flow. The sum of the wall and steam film resistances found in this way was assumed to remain constant for the mercury runs.

The mercury was cooled by counter-current flow of water in an annulus surrounding the mercury stream. Several cooling water velocities were used for each mercury velocity, making it possible to use the Wilson line plot method to evaluate the sum of the wall and mercury film resistances for each series of runs. Wall resistance was calculated from the wall thickness and thermal conductivity of the tube. It should be kept in mind that mercury did not wet the tube walls in these experimental runs.

It was found that mercury heat transfer coefficients varied as the 0.58 power of the Reynolds number over the range studied - 27,000 to 70,000, corresponding to velocities of 1.10 to 2.72 ft./sec. and flow rates of 1,800 to 4,400 pounds per hour. In this range, film coefficients varied from 1,390 ± 15% to 2,270 ± 15% Btu/hr. ft.^{2} °F. in the heating section, and they were about 10% lower in the cooling section. Prandtl numbers were between 0.0178 and 0.0186 throughout the experiments, corresponding to a small range of average bulk temperatures near 165 °F.

In evaluating the data from these experiments, it is of much interest to note the good agreement with the Von Karman equation[12], as pointed out by Musser and Page -

$$Nu = \frac{0.04\ Re^{3/4}\ Pr}{1 + Re^{-1/8}\left[Pr - 1 + \ln\left(1 + 5/6\left[Pr - 1\right]\right)\right]} \quad \text{(Eq. 4.08)}$$

as shown in Figure 4-14. For comparison, four other equations are also plotted on Figure 4-14:

1. Lyon equation[9] - letting $\alpha = 1.0$ (Eq. 4.07)

   $$Nu = 7 + 0.05(\alpha Pe)^{0.5} + 0.023(\alpha Pe)^{0.8}$$

2. Dittus-Boelter equation[18]

   $$Nu = 0.023\ (Re)^{0.8}\ (Pr)^{0.4} \quad \text{(Eq. 4.09)}$$

3. Modified Colburn equation[19]

   $$St\left(Pr^{2/3} + 0.05\right) = 0.023(Re)^{-0.2} \quad \text{(Eq. 4.10)}$$

4. Equation recommended in NRL-C-3243[2]

   $$Nu = 0.227\ (Re)^{0.5}\ (Pr)^{0.31} \quad \text{(Eq. 4.05)}$$

The fact that these data agree with the Von Karman equation[12] and disagree with the Lyon equation[9] is not considered to be any justification for assuming Von Karman's analogy to be the most accurate for use in predicting heat transfer to liquid metals. It appears that the agreement with Von Karman's analogy is largely fortuitous, since it would be expected that both analogies would be considerably modified if the case of non-wetting of the walls were considered theoretically.

In order to examine more carefully the deviations of the mercury data from the Lyon equation[9], the following assumptions are made:

[redacted]

1. The velocity profile is the same for wetting and non-wetting conditions.

2. The Lyon equation (Eq. 4.07) correctly predicts the mercury film coefficient - letting $\alpha = 1.0$ for first approximation.

3. Additional thermal resistance is caused by the presence of a film of air or mercury vapor.

4. Film resistances reported by Musser and Page are actually the sums of the mercury film and mercury vapor or air film resistances.

Thus, $\frac{1}{h}$ (Musser and Page) $= \frac{1}{h}$ (Lyon) $+ \frac{1}{h_x}$ (air or Hg vapor)

Figure 4-15 shows a plot of $\frac{1}{h_x}$ vs. Re, and it is seen that $h_x = \frac{1}{10}$ Re.

Only four points deviate greatly from this relationship, and they correspond to four points which fall lower than most of the other data plotted on Figure 4-14. If one assumes that an air or mercury vapor film is present, the thickness of the film can be calculated from its thermal conductivity and the thermal resistance reported. These calculations indicate a film thickness of about 2.3 microns at Re = 27000 and about 0.9 micron at Re = 70,000.

Another possible means of explaining the deviations may be that the effect of increasing velocity is to increase the effective rubbing perimeter or the actual total heat transfer area by increasing the contact area between the mercury and the pores in the wall. This idea, however, is considered to be less likely to be valid than the idea of an additional film. An unusual velocity profile may exist for the case of non-wetting and another possible explanation for deviations between theory and experiment could be that as Re is increased, the profile tends toward the profile

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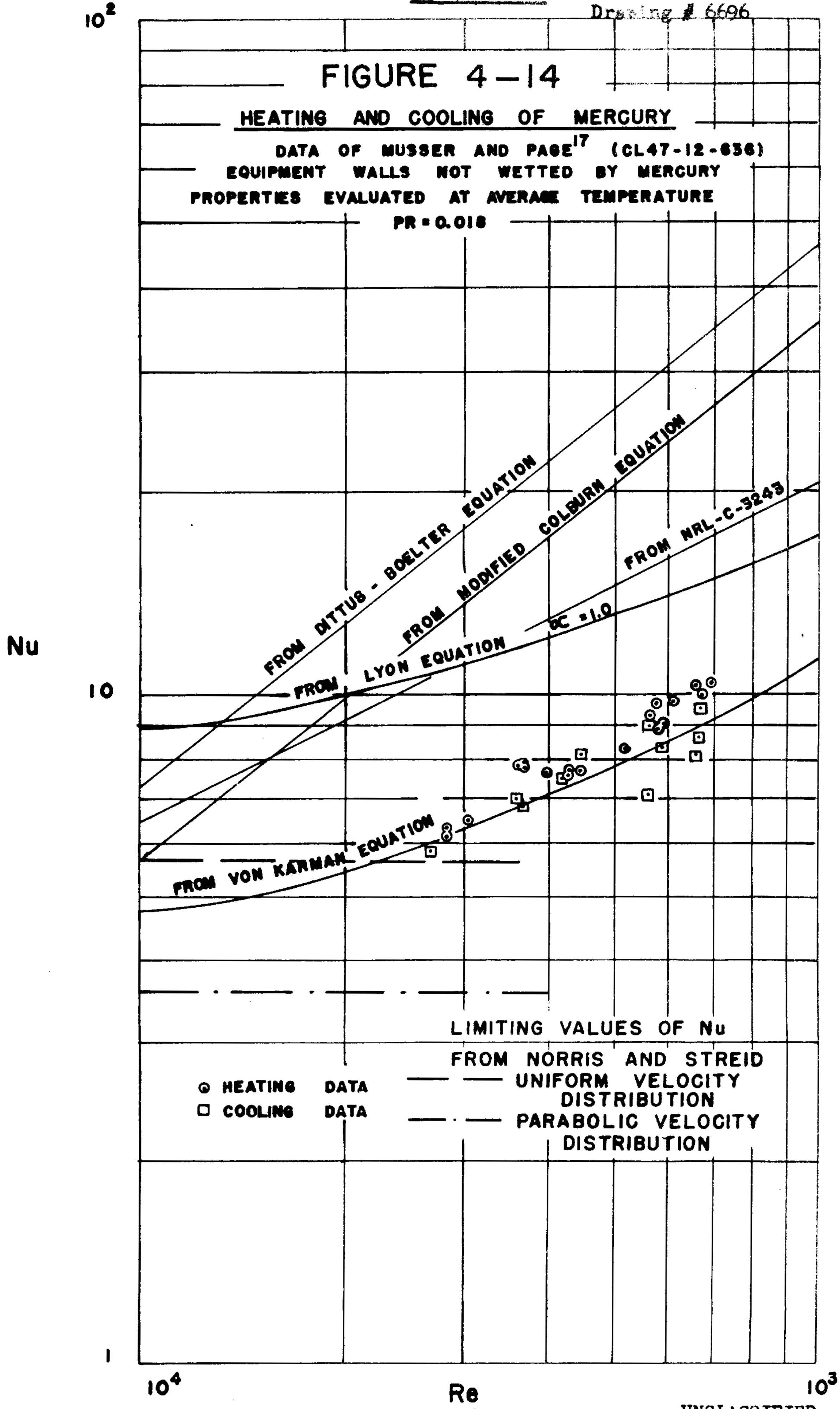

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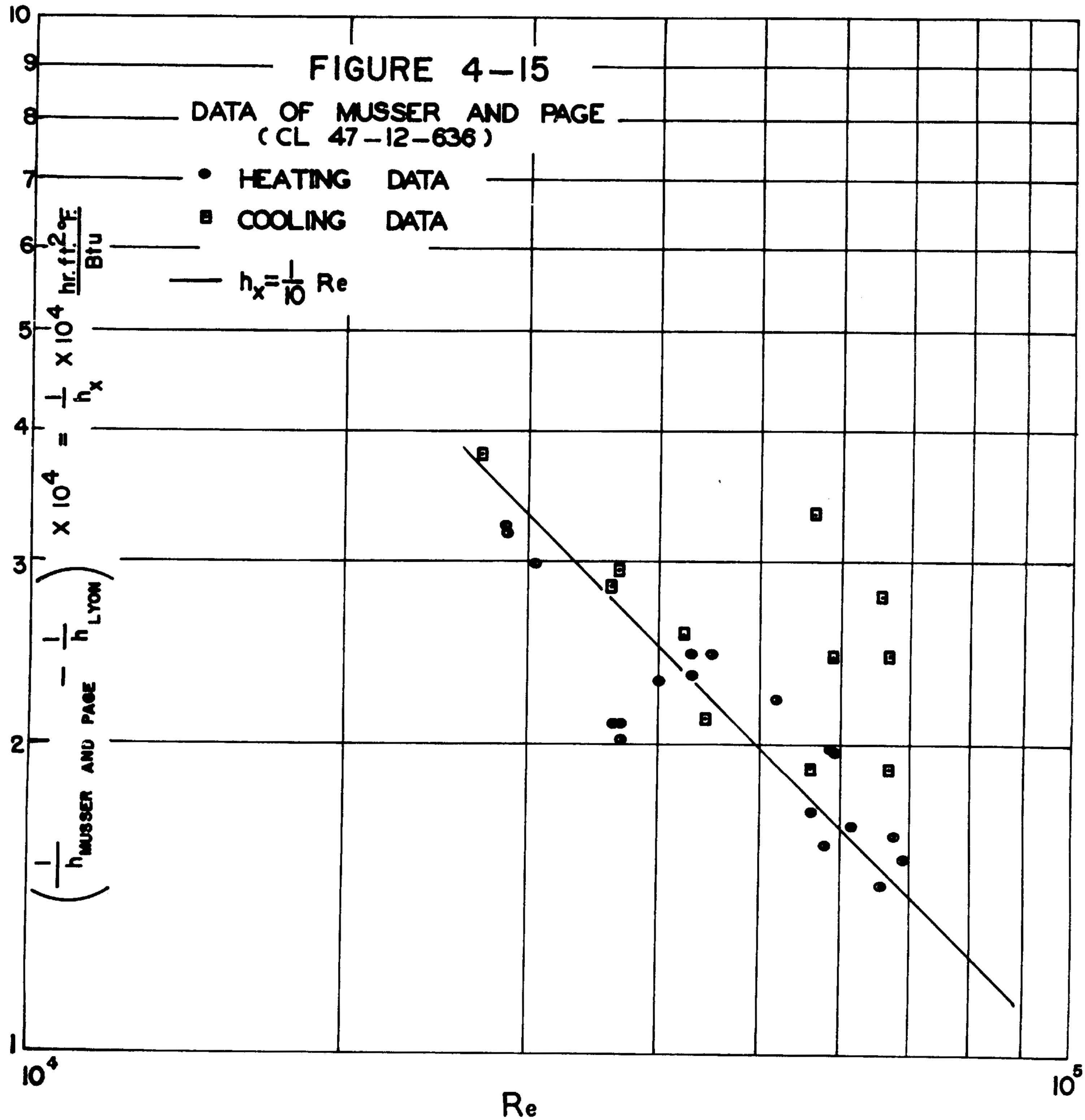

used for Martinelli's[7] and Lyon's[9] calculations. Norris and Streid[20] presented calculations for limiting Nusselt number based on the assumption of heat transfer by molecular conductivity alone. If one considers the case of constant wall temperature, the limiting Nu is 5.8 for constant velocity distribution and 3.66 for parabolic velocity distribution. These values are indicated on Figure 4-14. From the relation of experimental data with respect to the limiting Nu and the Lyon equation (which includes the effect of eddy conductivity), it is suggested that the velocity distribution approaches the parabolic limit and that the effect of eddy conductivity is small.

It is difficult to speculate as to the value of α or the shape of the velocity profile; consequently, little significance can be attached to any of these possible explanations until further experimental work is done on this problem. Musser and Page recommended extending the work to include other temperature ranges and the case of the walls being wetted. Another interesting allied problem might be to compare pressure drop data for wetting and non-wetting conditions. The influence of Reynolds number on friction and pressure drop may be considerably different in the two cases, and the data would be valuable in determining whether or not slip occurs at the wall.

Previous mention has been made of work on NaK reported in CF-3746[3]. In one series of runs, mercury was used as a coolant for the NaK in a copper cast steel tube heat exchanger. The same comments made on the analysis of NaK heat transfer data from this report apply to the data on mercury. A Wilson line plot[6] was used, assuming a change of heat transfer coefficient with the eight-tenths power of the velocity. It is of interest to note that additions of 0.1% magnesium and 0.0016% titanium were considered sufficient to produce satisfactory wetting. The range of Reynolds numbers studied was from 50,000 to 122,000, corresponding to velocities from about 1.3 to 3.2 ft./sec. at 347° F. (175 °C). The Wilson plot resulted in the following expression: $h = 2540\,(V')^{0.8}$, where h is the heat transfer coefficient in Btu/hr. ft.^{2} °F, and V' is velocity in ft./sec. Though little confidence is placed in the form of this expression, it is of interest to compare it with predictions from the Lyon equation[9] (Eq. 4.07) and the equation recommended in NRL-C-3243 (Eq. 4.05). If one uses a velocity from each end of the experimental range, and assumes a temperature of 347 °F. (175 °C), the following comparison may be made: (Physical properties of mercury are taken from Musser and Page[17].)

Velocity	Re	Nu		
ft./sec.		$2540\,(V')^{0.8}\,D/k$	Eq. 4.05,	Eq. 4.07
3.15	1.22×10^5	30.7	20.3	16
1.32	5.1×10^4	15.4	13.1	11.6

Other data on heat transfer to mercury were reported by Styrikovich and Semenovker[21]. Their data fell between the Von Karman and the Lyon equations, and there is some question regarding whether or not the mercury wetted the tube walls in their experiments.

Bakhmeteff[22] and Goldstein[23] are recommended for reference to theories of turbulent flow. Von Karman[12] and Boelter, Martinelli and Jonassen[24] presented developments which serve as a good background for the papers of Martinelli[7], Lyon[9], and Harrison and Menke[10]. From a project laboratory, Karush[25,26] presented two papers on eddy diffusion. An equation presented by Karush[26] for heat transfer to liquids of low Prandtl number gives fair agreement with the equations of Martinelli[7] and Lyon[9].

Brooks and Berggren[27] presented considerations on the variation of eddy diffusivity across a channel. They noted that the velocity profiles reported by Nikuradse[28] were not direct observations but they were based on smoothed gradients. The pattern of smoothing gave reasonable mixing lengths but called for zero eddy diffusivity at the center of a pipe. The interpretation of Brooks and Berggren indicated uniform eddy diffusivity across the central portion of a tube.

In the paper by Harrison and Monke[10] on heat transfer to liquid metals in asymmetrically heated channels, the effects of three different sets of values of eddy diffusivity were compared. One of the assumed curves for eddy diffusivity was essentially the same as the interpretation by Brooks and Berggren noted above.

Preliminary studies reported by Corcoran, Roudebush and Sage[29] indicated that eddy diffusivity reaches a maximum at the center of a channel. Sherwood and Woertz[30] observed that eddy diffusivity was substantially uniform across the main central portion of a large flat duct.

Young[31] presented additional work on the analogy between heat and momentum transfer in a paper on evaporation of a dissolved gas from a liquid.

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES; A CRITICAL SURVEY

PART 5

HEAT TRANSFER TO WATER IN AN ANNULUS BY COMBINED CONDUCTION AND FORCED CONVECTION

W. B. Harrison
October 1, 1948

[redacted]

PART 5

HEAT TRANSFER TO WATER IN AN ANNULUS BY COMBINED CONDUCTION AND FORCED CONVECTION

Double-pipe heat exchangers are very attractive because they are easy to construct and have a simple space-saving geometry. The case of heat transfer from the inner fluid to the inside tube has been correlated by many experimenters, but correlations for the heat transfer in the annulus are not as well established. In this section, heat transfer is discussed for the cases of both turbulent and laminar flow. A few notes on the effect of ribs and the effect of eccentricity in the annulus are also presented.

Turbulent flow

Numerous experimenters have investigated the case of heating water in an annulus. Donovan surveyed the literature for information on heat transfer and pressure drop in an annulus, and the results of this survey were presented in a Clinton Laboratories Memorandum to Briggs[1]. For heat transfer information, Donovan relied rather heavily on a study presented by Wiegand[2] in 1945. Wiegand compared the recommendations of several experimenters[3,4,5,6,7] by using the following general equation as a basis:

$$j = \left(\frac{h}{cG}\right)\lambda\left(\frac{c\mu}{k}\right) = \phi(D_1,\ D_2)\left(\frac{D_e\ G}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.01)}$$

The present status of experimental evidence indicates that the exact influence of each variable is not known. For a given apparatus, an empirical equation can be established which fits the data well, but it is not necessarily applicable to data obtained in other equipment or with widely different operating conditions. Wiegand recommended the following equation as a logical compromise for the case of heat transfer between a turbulent stream and the inner wall of an annulus in which it moves:

$$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{1-n} = 0.023\left(\frac{D_2}{D_1}\right)^{0.45}\left(\frac{D_e\ G}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.02)}$$

For heat transfer to the outer wall, he suggested:

$$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{1-n} = 0.023\left(\frac{D_e\ G}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.03)}$$

If viscosity of the fluid is greater than twice the viscosity of water, Wiegand suggested that properties in these equations should be evaluated at the film temperature[8]. Equation 5.03 can be rewritten as the usual Dittus-Boelter type of equation[9,10], where $n = 0.4$ for heating and 0.3 for cooling. Another survey of transfer processes in annular spaces was presented by Wiegand and Baker[5] in 1942. A brief description of experimental technique was included in the remarks about friction and heat transfer[11,12,13,14] data obtained by various experimenters and, again, a systematic comparison of correlations was included.

A number of additional references on heat transfer[15] and pressure drop in an annulus appeared in the paper which presented a new correlation by Davis[6]. The most recent report on experimental work found in off-the-project literature was presented by Carpenter, Colburn, and Schoenborn in 1946[16].

For purposes of this report, it was decided to note briefly a few of the recommended correlations for heat transfer in an annulus and to choose three representative types of correlations for comparing data of observers on the project.

Monrad-Pelton[3]

$$\frac{h\,D_e}{k} = 0.020\left(\frac{D_2}{D_1}\right)^{0.53}\left(\frac{D_e\,G}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{1/3} \qquad \text{(Eq. 5.04)}$$

or

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{2/3} = 0.020\left(\frac{D_2}{D_1}\right)^{0.53}\left(\frac{D_e\,G}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.04a)}$$

These equations were presented for the case of heat transfer to the inner wall of the annulus, and they correlated experiments having the following characteristic features:

Medium	Range of $\frac{D_e\,G}{\mu}$	D_2/D_1	$D_2' - D_1' = D_e'$ (in.)
Water	19,300-222,000	1.65	0.817 inches
Water	11,300-175,000	2.45	1.818
Air	12,600-31,300	17	0.987

For heat transfer to the outer wall, the Dittus-Boelter equation[9,10] was found to hold for water in an annulus of $D_2/D_1 = 1.85$, ($D_e = 0.46$ in.) and over a range of $\frac{D_e\,G}{\mu}$ from 6300 to 28800.

For heat transfer from a wire to air, Mueller correlated data with the following equation:

$$\frac{hD_1}{k}\left(\frac{D_1}{D_2}\right)^{0.2} = 0.064\left(\frac{D_1G}{\mu}\right)^{0.5} \qquad \text{(Eq. 5.05)}$$

In order to compare this correlation with others which include Prandtl number, Wiegand[2] assumed that $0.064 = a\left(\frac{c\mu}{k}\right)^{n}$. Letting $\frac{c\mu}{k} = 0.74$ and $n = 0.4$ for heating air, Wiegand wrote Equation 5.05 in the following manner:

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{1-n} = 0.077\left(\frac{D_2}{D_1}\right)^{0.2}\left(\frac{D_e}{D_1}\right)^{0.5}\left(\frac{D_eG}{\mu}\right)^{-0.5} \qquad \text{(Eq. 5.05a)}$$

It is believed that the use of the expression $0.064 = a\left(\frac{c\mu}{k}\right)^{0.4}$ should lead to $0.072 \times 0.74^{0.4}$ rather than $0.077 \times 0.74^{0.4}$. This would give

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{1-n} = 0.072\left(\frac{D_2}{D_1}\right)^{0.2}\left(\frac{D_e}{D_1}\right)^{0.5}\left(\frac{D_eG}{\mu}\right)^{-0.5} \qquad \text{(Eq. 5.05b)}$$

Mueller's work is not believed to be applicable to present project needs, however, since the range of diameter ratios $\left(\frac{D_2}{D_1}\right)$ used in the experiment was between 595 and 6850.

Among the experimenters mentioned by Wiegand and Baker[5], it is noted that Foust and Christian[12], and Foust and Thompson[11] presented two more equations for heat transfer from the inner wall to water in an annulus.

Foust and Christian[12]

$$\frac{hD_e}{k} = 0.032\left(\frac{D_2}{D_1}\right)\left(\frac{D_eG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4} \quad \text{or} \qquad \text{(Eq. 5.06)}$$

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{0.6} = 0.032\left(\frac{D_2}{D_1}\right)\left(\frac{D_eG}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.06a)}$$

$\frac{D_2}{D_1}$ ranged from 1.20 to 2.56. Deviations between experimental data and the equation appeared to be increasing as $\frac{D_2}{D_1}$ increased. It is felt that the correlation would not hold for $\frac{D_2}{D_1}$ much greater than 2.5.

Foust and Thompson[11]

$$\frac{hD_e}{k} = 0.027\left(\frac{D_eG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4} \quad \text{or} \qquad \text{(Eq. 5.07)}$$

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{0.6} = 0.027\left(\frac{D_eG}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.07a)}$$

Constant $\frac{D_2}{D_1}$ of 1.58 was used in the experiment correlated by Equation 5.07, so it is impossible to apply a $\frac{D_2}{D_1}$ correction.

Davis[6]

$$\frac{hD_1}{k} = 0.031\left(\frac{D_1G}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}\left(\frac{D_2}{D_1}\right)^{0.15} \qquad \text{(Eq. 5.08)}$$

or

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 0.031\left(\frac{D_2}{D_1}\right)^{0.15}\left(\frac{D_e}{D_1}\right)^{0.2}\left(\frac{D_eG}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.08a)}$$

Equation 5.08 was used by Davis to correlate data with $\frac{D_2}{D_1}$ ranging from 1.18 to 6800. The coefficient 0.031 was printed as 0.038, but it is believed that 0.031 is the correct value[7,16,17]. By analogy, Davis presented an equation for heat transfer to the outer wall of an annulus:

$$\frac{hD_e}{k} = 0.023\left(\frac{D_eG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}\left(\frac{D_e}{D_2}\right)^{q} \qquad \text{(Eq. 5.09)}$$

or

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 0.023\left(\frac{D_eG}{\mu}\right)^{-0.2}\left(\frac{D_e}{D_2}\right) \qquad \text{(Eq. 5.09a)}$$

where q is between 0.05 and 0.15.

McMillen and Larson[7]

Using $\frac{D_2}{D_1}$ from 1.25 to 1.97, McMillen and Larson correlated data two ways:

Davis type equation[6]:

$$\frac{hD_1}{k} = 0.030\left(\frac{D_1G}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}\left(\frac{D_2}{D_1}\right)^{0.15} \qquad \text{(Eq. 5.10)}$$

(This equation is identical with Equation 5.08 except for the coefficient)

Colburn type equation[8]

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)_f^{2/3} = 0.0305\left(\frac{D_1G}{\mu_f}\right)^{-0.2} \qquad \text{(Eq. 5.11)}$$

Carpenter, Colburn, and Schoenborn[16] showed that critical Reynolds number was about 2000 when the Re was based on D_e and μ_a. This gives support to the use of D_e, rather than D_1 or D_2, in Re involved in heat transfer correlations. The apparatus used had a $\frac{D_2}{D_1}$ of 1.334 and Reynolds numbers ranged from 150 to 15,000. As seen from the range of Reynolds numbers, the data were not extended very far into the turbulent region. Data were plotted in terms of:

(1) The Colburn[8] j factor, $\frac{h}{cG}\left(\frac{c\mu}{k}\right)_f^{2/3}$, vs. $\frac{D_eG}{\mu_f}$; and

(2) A j factor based on the Sieder-Tate[18] equation,

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14}, \text{ vs. } \frac{D_eG}{\mu_a}$$

No new equations were suggested, but it appears that the following equation approximates the data for turbulent flow, using the Colburn j factor:

$$j = \frac{h}{cG}\left(\frac{c\mu}{k}\right)_f^{2/3} = 0.0091\left(\frac{D_eG}{\mu_f}\right)^{0.1} \qquad \text{(Eq. 5.12)}$$

Since critical Reynolds number involves μ_a rather than μ_f, the Sieder-Tate type of correlation was thought to be more significant. The data were approximated by the following equation:

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} \cong 0.086\left(\frac{D_eG}{\mu}\right)^{-0.1} \qquad \text{(Eq. 5.13)}$$

The usual Sieder-Tate equation [18] for circular tubes is:

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 0.027\left(\frac{DG}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.14)}$$

For the system involved in these experiments, the Davis equation[6] (Eq. 5.08d) resolves to

$$\frac{h}{cG}\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 0.026\left(\frac{D_eG}{\mu}\right)^{-0.2} \qquad \text{(Eq. 5.08b)}$$

It is of interest to note that Equations 5.14 and 5.08b are practically identical. The data taken fell about 25% lower than the Sieder-Tate equation at Re = 3000, and about 10% lower when extrapolated to Re = 20,000. It is suspected that use of mass flow rate raised to the 0.9 power in a Wilson line plot[19] early in the analysis of the data is reflected in Equations 5.12 and 5.13.

For correlation of project data on heat transfer to the inner wall of an annulus, three equations are selected for comparison:

1. Dittus-Boelter type of equation[9,10]

$$\frac{hD_e}{k} = 0.023 \left(\frac{D_e G}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{n} \qquad \begin{array}{l} n = 0.4 \text{ for heating} \\ n = 0.3 \text{ for cooling} \end{array}$$

(Eq. 5.03a)

In the project experiments which have been studied, $\frac{D_2}{D_1}$ ratios were between 1.07 and 1.35; consequently, Equation 5.03a approximates the Equation 5.02 recommended by Wiegand[2]. Additional support is given to Equation 5.03a by the work of Norris and Sims[20], and the correlation of Foust and Thompson[11] is very similar.

2. Monrad-Pelton Equation[3]

$$\frac{hD_e}{k} = 0.020 \left(\frac{D_2}{D_1}\right)^{0.53} \left(\frac{D_e G}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{1/3} \qquad \text{(Eq. 5.04)}$$

Equation 5.04 is selected for comparison since it was used to correlate data for fairly low $\frac{D_2}{D_1}$ (1.65) over a large range of Re (19,300-222,000).

3. Davis equation[6]

$$\frac{hD_1}{k} = 0.031 \left(\frac{D_1 G}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \left(\frac{D_2}{D_1}\right)^{0.15} \qquad \text{(Eq. 5.08)}$$

Equation 5.08 is supported by the work of McMillen and Larson[7] and it is in fair agreement with the data of Carpenter, Colburn, and Schoenborn[16].

The following paragraphs will be devoted to brief notes referring to experiments performed in project laboratories on heat transfer from the inside wall to water in an annulus. Characteristics $\left(\frac{L}{D_e}\right.$, $\frac{L}{D_1 + D_2}$, $\frac{D_2}{D_1}$ and $\left.D_e'\right)$ of the equipment used in each experiment are shown on Table 5-I. Data of various observers are compared with the Dittus-Boelter type of equation, the Monrad-Pelton equation, and the Davis equation on Figures 5-1, 5-2, and 5-3, respectively.

Of the reports found in the available project literature, one of the most comprehensive reports on heat transfer in an annulus was presented by Gishler and Boyer[21]. Data were taken in a double-tube aluminum heat exchanger with steam passing through the center tube. For the water in the annulus, the range of $\frac{D_eG}{\mu}$ was between 10,800 and 39,000. Excellent agreement was obtained with the original Dittus-Boelter equation[9] for heating water in circular tubes,

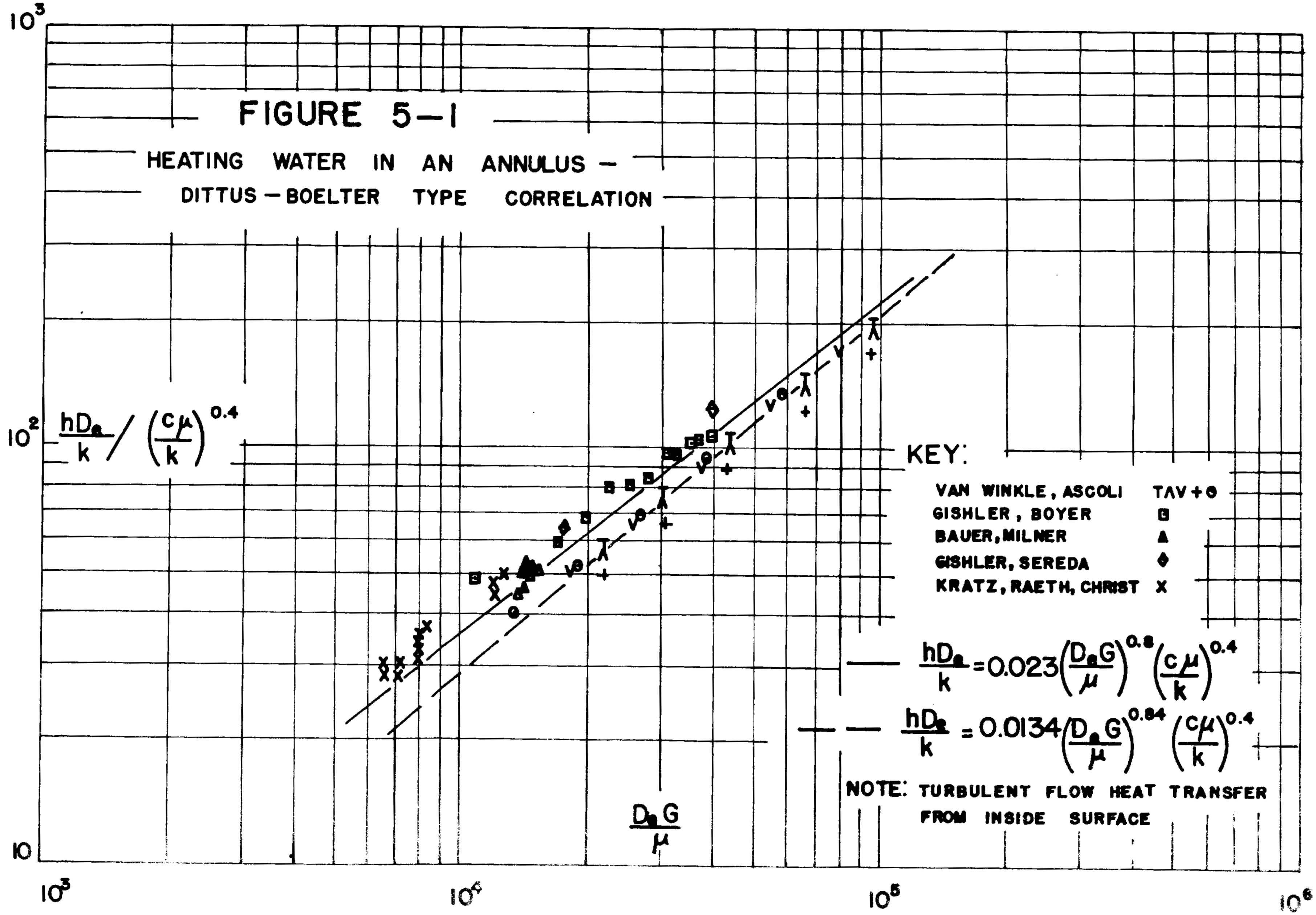
$$\frac{hD}{k} = 0.0243 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{0.4}, \qquad \text{(Eq. 5.15)}$$

with D_e substituted for D.

Gishler and Sereda[22] used two heat exchangers to determine film forming properties of water treated with alum. The initial run in each experiment is used to calculate a point for comparison with the equations mentioned above. Here again, the center tube was steam-heated and water flowed through the annulus. In the heat exchanger having L/D_e of 42.5, the $\frac{D_eG}{\mu}$ in the annulus was 17,450. In the other heat exchanger, $\frac{D_eG}{\mu}$ in the annulus was 39,000.

In determining heat transfer coefficients for polished and anodized aluminum surfaces, Bauer and Milner[23] found good agreement between data for heating water in an annulus and the Colburn equation[8] for heat transfer in circular tubes,

[redacted]



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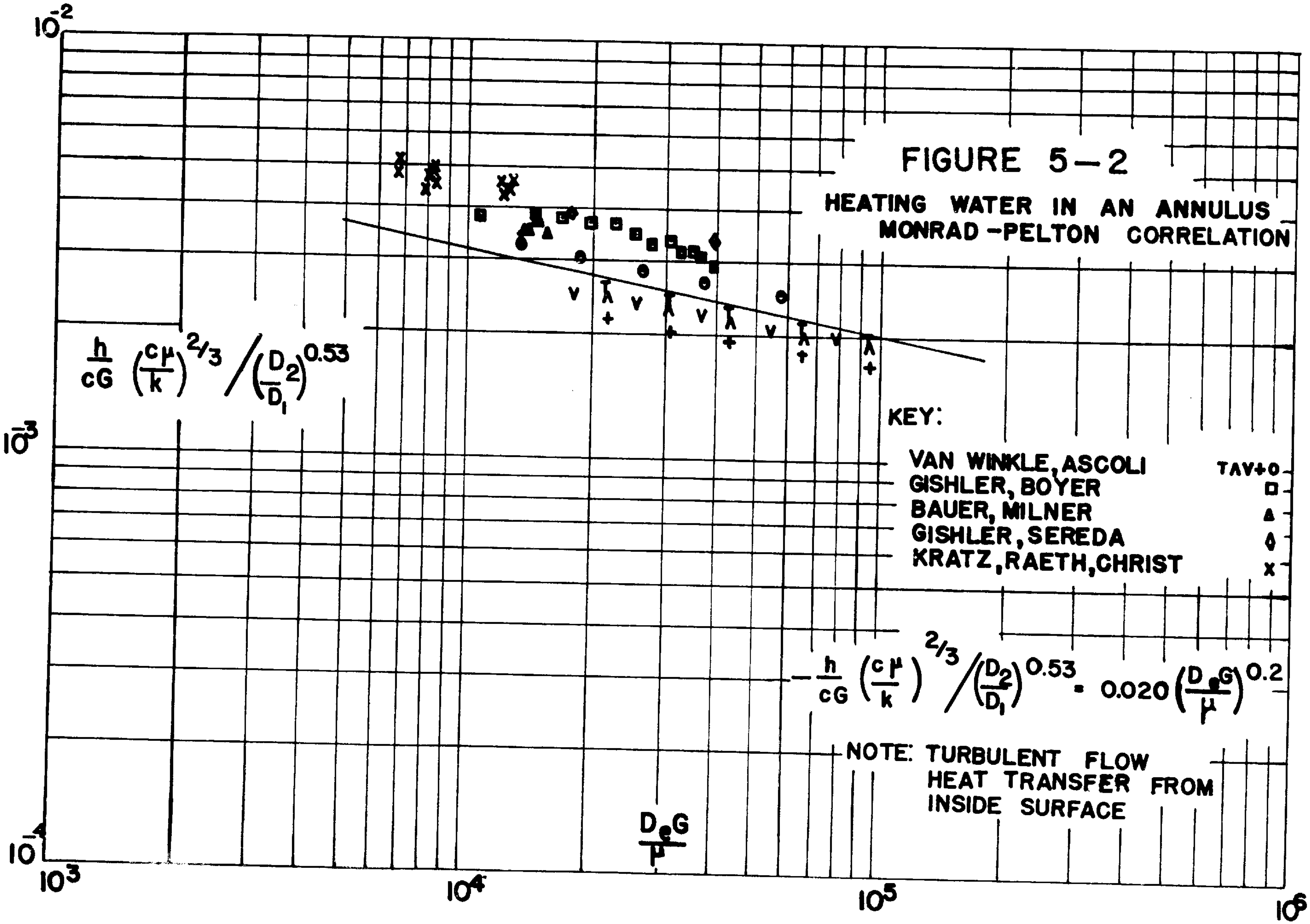

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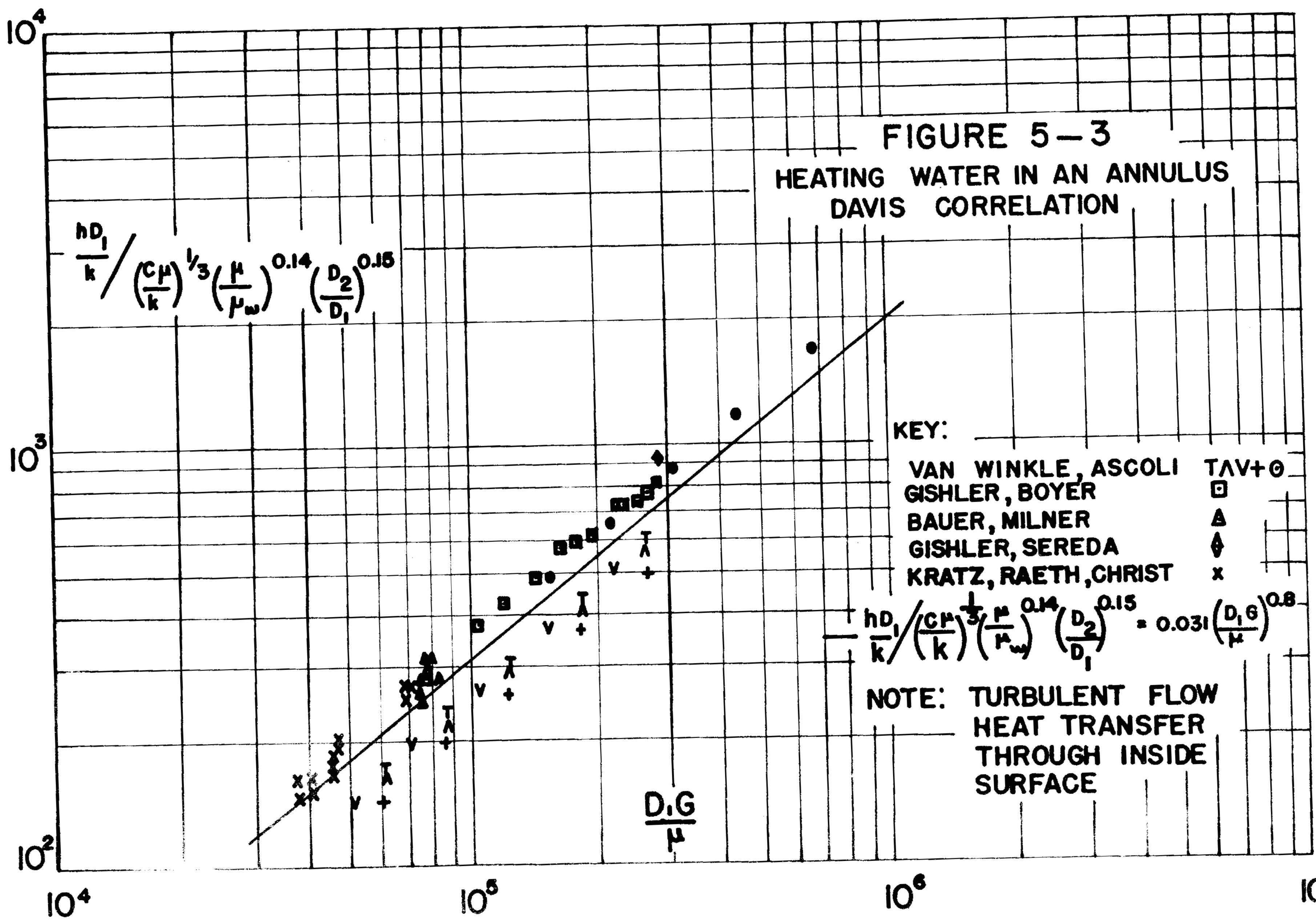

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$$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{2/3} = 0.023\left(\frac{DG}{\mu_f}\right)^{-0.2}, \qquad \text{(Eq. 5.16)}$$

where D_e is used instead of D. The center tube was steam heated, and, for both the polished and the anodized tubes, the $\frac{D_eG}{\mu}$ in the annulus was in a small range near 14,000. The Davis correlation appears to bring the data of Bauer and Milner in line with data of other observers somewhat better than the other two correlations.

Kratz, Raeth, and Christ[24] made a series of heat transfer studies which included a few tests in an electrically heated center tube with water in turbulent flow in the annulus. As shown on the accompanying figures, their data fall in line with the data of Gishler and Boyer, and Gishler and Sereda.

Van Winkle and Ascoli[25] performed a series of experiments which included data on heating water from the inside wall of an annulus, and cooling water through the outside wall of an annulus. The data for heating water were analyzed by using the logarithmic mean temperature difference[26] to calculate the overall coefficient and applying the Wilson line plot type of analysis[19]. The equation recommended by Van Winkle and Ascoli as a result of these experiments is:

$$\frac{hD_e}{k} = 0.0134\left(\frac{D_eG}{\mu}\right)^{0.84}\left(\frac{c\mu}{k}\right)^{0.4} \qquad \text{(Eq. 5.17)}$$

or a simplified version -

$$h = 120(1 + 0.012t)\,\frac{(V')^{0.84}}{(D_e')^{0.16}} \qquad \text{(Eq. 5.18)}$$

where h is in $\frac{\text{Btu}}{\text{hr. ft.}^2\ {}^\circ\text{F}}$; t is average bulk temperature in °F; V' is velocity in ft./sec.; and D_e' is equivalent diameter in inches. For the case having $\frac{D2}{D1} = 1.09$, Reynolds numbers in the annulus ranged from 13,200 to 57,400. In the other case, Reynolds numbers ranged from 21,570 to 93,560.

The data of Van Winkle and Ascoli are seen to fall somewhat lower than the data of other observers as shown on Figures 5-1, 5-2, and 5-3. It is believed that their data exhibit the same effect as shown in the work by Carpenter, Colburn, and Schoenborn[16], and it is further believed that, in both sets of experiments, the deviation from usual equations is a result of the choice of temperature difference and the Wilson line plot method for analysis. Due to a limitation of time, no effort has been made to substantiate this belief by a closer examination of the data, but it appears that use of W, G, or V to an exponent different from 0.8 in the Wilson line plot will be reflected in the final correlation. Good heat balances indicate that heat fluxes used in calculating overall coefficients were very reliable. The logarithmic mean temperature difference[26] is strictly applicable for systems having a constant overall coefficient and it is suspected that deviations from this condition are reflected in the evaluation of the overall coefficients which were used in the Wilson type plots to get individual coefficients.

In comparing Figures 5-1, 5-2 and 5-3, it is seen that the Dittus-Boelter type of correlation gives the best agreement with the data of project observers. This is especially fortunate since the Dittus-Boelter type of equation is the simplest of the three equations. It follows that for heating water in an annulus, with heat being transferred from the inside wall, Equation 5.04a is recommended for systems having characteristics within the range shown on Table 5-I.

TABLE 5-I

Characteristics of Experimental Equipment

Experimenters	L/D_e	$L/D_1 + D_2$	D_2/D_1	D_e' (in.)
Gishler, Boyer[21]	383	24.8	1.14	0.200
Gishler, Serada[22]	383	24.8	1.14	0.200
	42.5	4.28	1.22	0.235
Bauer, Milner[23]	42.9	3.66	1.19	0.28
Kratz, Raeth, Christ[24]	16.9	1.36	1.07	0.241
Van Winkle, Ascoli[25]	762	31.5	1.09	0.151
	650	97.3	1.35	0.178

Van Winkle and Ascoli[25] presented the only data found in project reoorts on cooling water in turbulent flow in an annulus. These experiments differ from the others mentioned previously not only as a result of the fact that the water is being cooled rather than heated, but also as a result of the fact that heat is being transferred to the outside wall. In these experiments, $\frac{D_e G}{\mu}$ ranged from 27,500 to 106,000 in two systems having characteristics as indicated below:

System	L/D_e	$L/D_1 + D_2$	D_2/D_1	D_e' (in.)	D_2' (in.)	D_1' (in.)
1	733	36.4	1.10	0.157	1.662	1.505
2	763	177	1.60	0.151	0.401	0.250

Since the cooling data were taken simultaneously with the heating data mentioned above, the comments made on the method of analysis of the heating data are also applicable to these.

The Dittus-Boelter type of equation and several modifications of the Davis equation are selected to correlate the data of Van Winkle and Ascoli for cooling water through the outside wall of an annulus:

1. Dittus-Boelter type of equation[9]

$$\frac{hD_e}{k} = 0.023 \left(\frac{D_e G}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{n} \quad \begin{array}{l} n = 0.4 \text{ for heating} \\ n = 0.3 \text{ for cooling} \end{array} \qquad \text{(Eq. 5.03a)}$$

This equation was recommended by Wiegand[2], primarily as a result of the findings of Monrad and Pelton[3]. A comparison of the data with Equation 5.03a is shown on Figure 5-4.

2. Davis equation[6], letting $q = 0.1$

$$\frac{hD_e}{k} = 0.023 \left(\frac{D_e G}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \left(\frac{D_e}{D_2}\right)^{q} \qquad \text{(Eq. 5.09)}$$

Equation 5.09 is shown on Figure 5-5.

3. It is seen that if $q = 0.2$, equation 5.09 could be written as follows:

$$\frac{hD_2}{k} = 0.023 \left(\frac{D_2 G}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \text{(Eq. 5.09b)}$$

Although Davis recommended that q should be between 0.05 and 0.15, the data were compared with equation 5.09b on Figure 5-6.

4. It appears from Figure 5-6 that a diameter correction would bring the data in line. Since the Davis equation (Eq. 5.08) for heat transfer to the inside wall involves $\left(\frac{D_2}{D_1}\right)^{0.15}$, it is decided to apply a correction of $\left(\frac{D_1}{D_2}\right)^{0.15}$ to the data for heat transfer to the outer wall. Figure 5-7 shows the data compared with the equation -

$$\frac{hD_2}{k} = 0.031 \left(\frac{D_2 G}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \left(\frac{D_1}{D_2}\right)^{0.15} \qquad \text{(Eq. 5.08b)}$$

[redacted]

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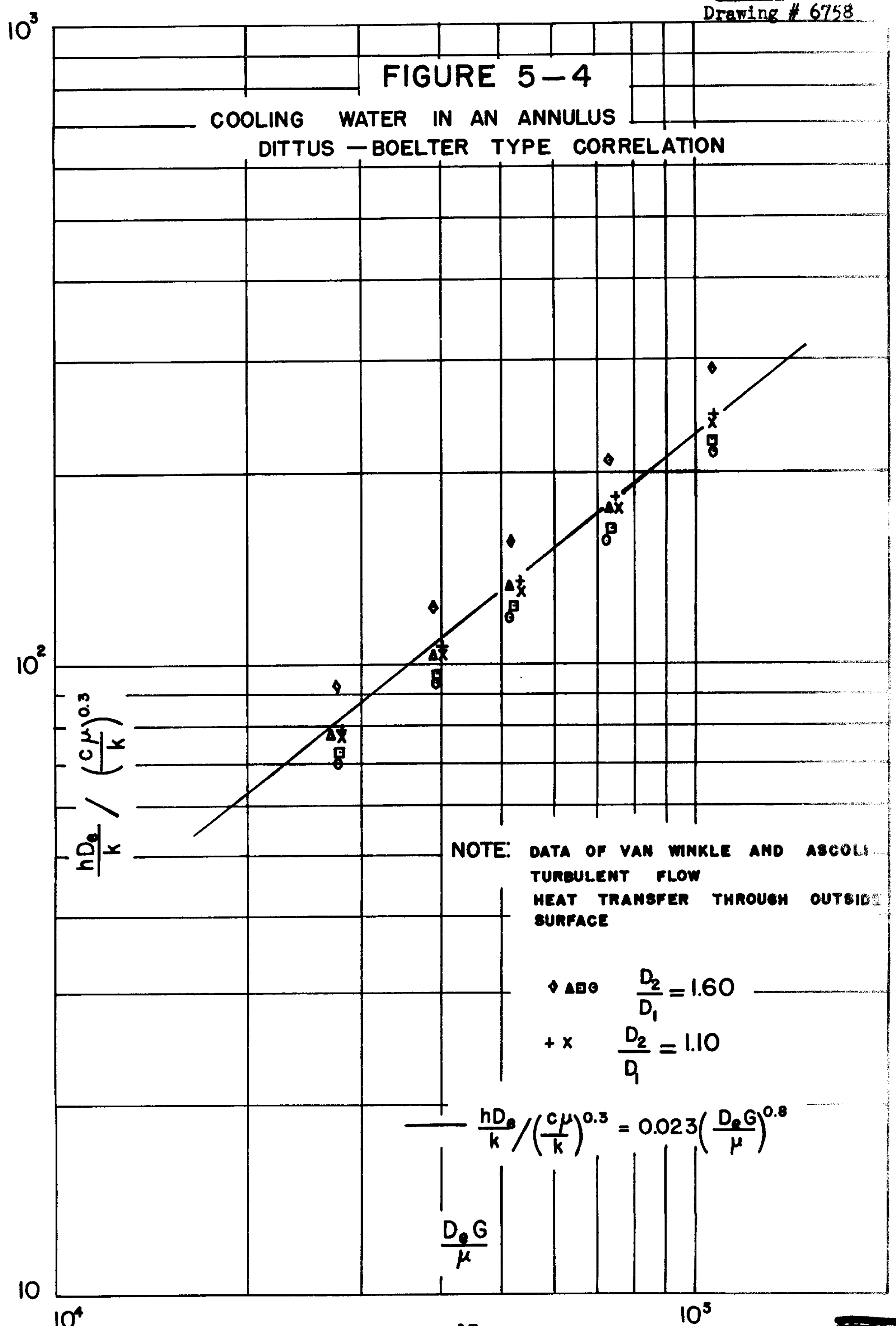
Drawing # 6758



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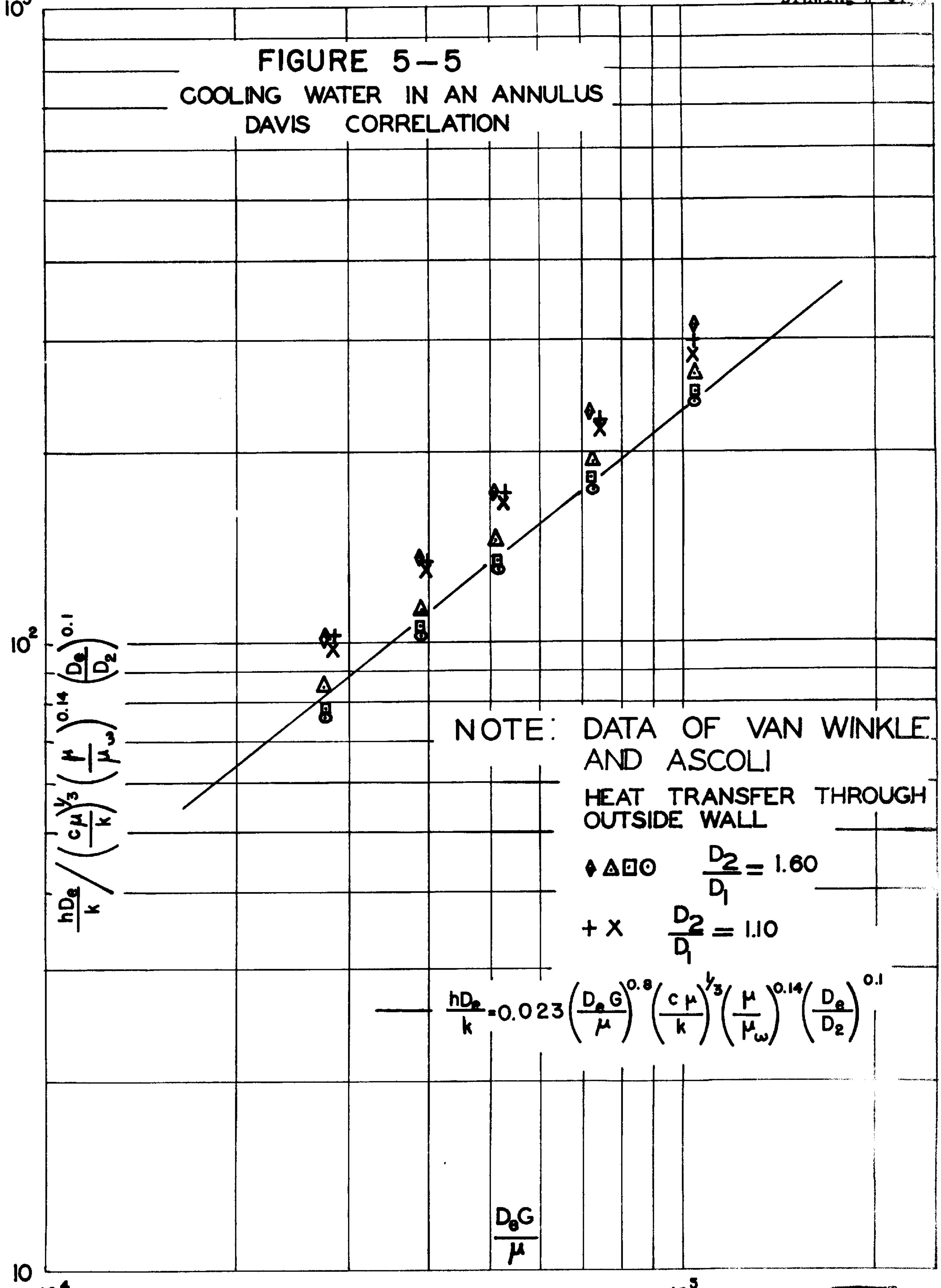
FIGURE 5-5
COOLING WATER IN AN ANNULUS
DAVIS CORRELATION
NOTE: DATA OF VAN WINKLE AND ASCOLI
HEAT TRANSFER THROUGH OUTSIDE WALL
$\Diamond \triangle \boxdot \odot$ $\frac{D_2}{D_1} = 1.60$
$+ \times$ $\frac{D_2}{D_1} = 1.10$
$\frac{hD_e}{k} = 0.023 \left(\frac{D_e G}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_\omega}\right)^{0.14} \left(\frac{D_e}{D_2}\right)^{0.1}$
$\frac{hD_e}{k} \Big/ \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_\omega}\right)^{0.14} \left(\frac{D_e}{D_2}\right)^{0.1}$
$\frac{D_e G}{\mu}$
10^3
10^2
10
10^4
10^5

[redacted]

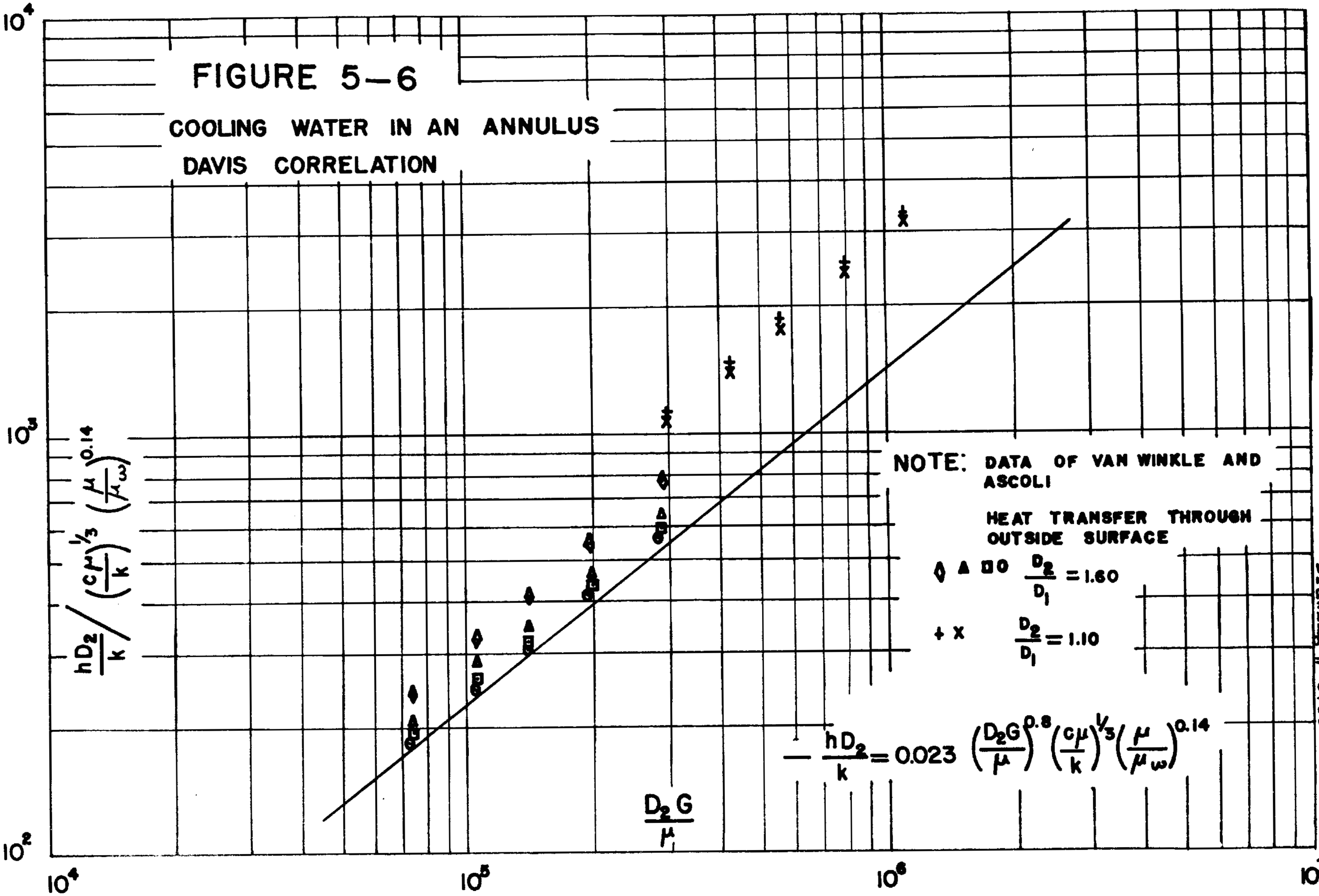

FIGURE 5—6

COOLING WATER IN AN ANNULUS

DAVIS CORRELATION

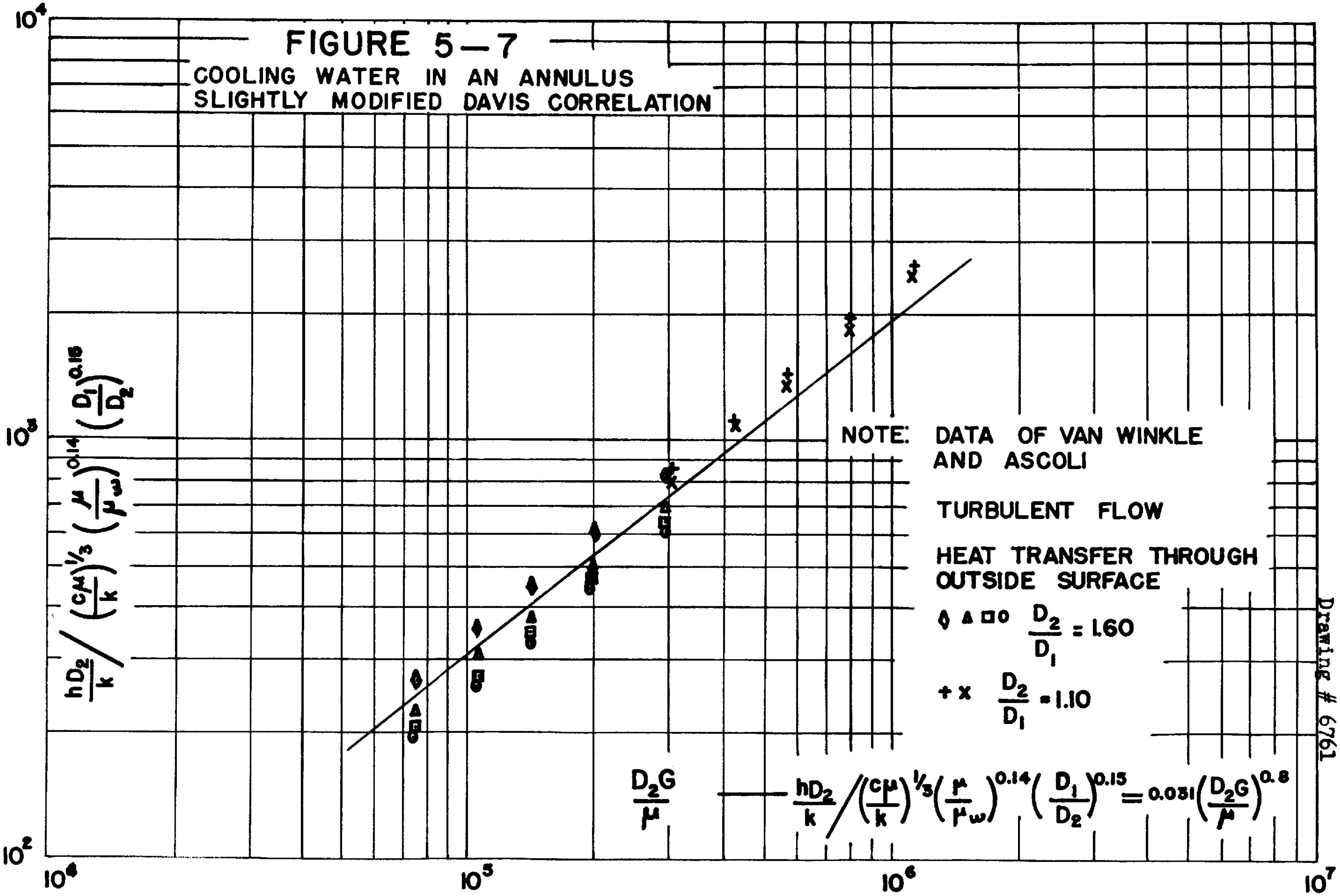

FIGURE 5—7

COOLING WATER IN AN ANNULUS
SLIGHTLY MODIFIED DAVIS CORRELATION

Drawing # 6761

It would appear that, for the case of cooling water through the outside surface of an annulus, Equations 5.03a and 5.08b give the best agreement with the data of Van Winkle and Ascoli. Since the Dittus-Boelter type of equation is easier to use than the Davis type of equation, it is recommended that equation 5.03a be used for this case.

Following is a list of additional reports which contain small amounts of information on the case of heating water in turbulent flow from the inside wall of an annulus:

a) Brown, G.M., Cabell, C.D., CE-2434, January 20, 1945

Recommends Dittus-Boelter type of equation

b) Larson, R.E., Szulinski, M.J., CT-2524, December 23, 1944.

$$h = 150(1 + 0.011t_a)\frac{(V')^{0.8}}{(D_e')^{0.2}}$$

c) Cooper, C.M. (W.K. Woods, R.N. Lyon), CS-1689, April 30, 1944

$$\frac{hD_e}{k} = 0.0196\left(\frac{D_eG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4}$$

d) Cooper, C.M. (C.P. Cabell et al), CS-1339, January 31, 1944

$$h = 130(1 + 0.011t)\frac{(V')^{0.8}}{(D_e')^{0.2}}$$

e) Cooper, C.M. (C.P. Cabell et al), M-CN-1327, February 5, 1944

f) Cooper, C.M. (C.P. Cabell et al, M-CN-1267, January 29, 1944

$$h = 155(1 + 0.011t)\frac{(V')^{0.8}}{(D_e')^{0.2}}$$

[redacted]

g) Cooper, C.M. (C.P. Cabell et al), CE-1184, January 1, 1944

$$h \cong 102(1 + 0.011t) \frac{(V')^{0.8}}{(D_e')^{0.2}}$$

h) Cooper, C.M. (C.P. Cabell et al), CE-1122, November 30, 1943

$$h \cong 173(1 + 0.011t) \frac{(V')^{0.8}}{(D_e')^{0.2}}$$

i) Cooper, C.M. (Cabell et al) CE-898, August 28, 1943

j) Gishler, P.E., Boyer, T.W., CRX-286, August 23, 1946

$$h = 203(1 + 0.00676t) \frac{(V')^{0.8}}{(D_e')^{0.2}}$$

k) Leverett, M.C. (Van Winkle, Ascoli), MonN-201, November 14, 1946

$$h = 120(1 + 0.012t) \frac{(V')^{0.84}}{(D_e')^{0.16}}$$

l) Cooper, C.M. (Cabell), Memo 115, August 14, 1943

m) Cooper, C.M. (Cabell), Memo 114, August 7, 1943

n) Leverett, M.C., Huffman, J.R., MonN-108, May 15, 1946

$$h \cong 150(1 + 0.011t) \frac{(V')^{0.8}}{(D_e')^{0.2}} \quad \text{for } t = 130\text{-}140^\circ \text{ F.}$$

$$h \cong 192(1 + 0.011t) \frac{(V')^{0.8}}{(D_e')^{0.2}} \quad \text{for } t = 70\text{-}75^\circ \text{ F.}$$

o) Cooper, C.M., (Cabell), Memo 108, July 31, 1943

p) Cambron, A., MPX-14, March 22, 1946

$$h = 143(1 + 0.01t_f) \frac{(V')^{0.8}}{(D_e')^{0.2}}$$

In order to show the wide variety of recommendations, all of the equations can be put in the form: $h = a \frac{(V')^{0.8}}{(D_e')^{0.2}}$

The Dittus-Boelter type of equation may be written as follows:

$$h = 26.5\, k\left(\frac{\rho}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4}\frac{(V')^{0.8}}{(D_e')^{0.2}} \qquad \text{(Eq. 5.03b)}$$

The comparison of values of Ω for different temperatures is shown on Figure 5-8. Included in the comparison are recommendations from references b(which corresponds to a recommendation by McAdams for circular tubes[27]) c, d, f, g, h, j, and n. This comparison does not alter the previous recommendation to use the Dittus-Boelter type of equation, but it is included for the purpose of giving a representation of the contribution of other experimenters. No effort can be made to explain the disagreement between their recommendations on account of insufficient data and information on the apparatus and techniques used by the experimenters.

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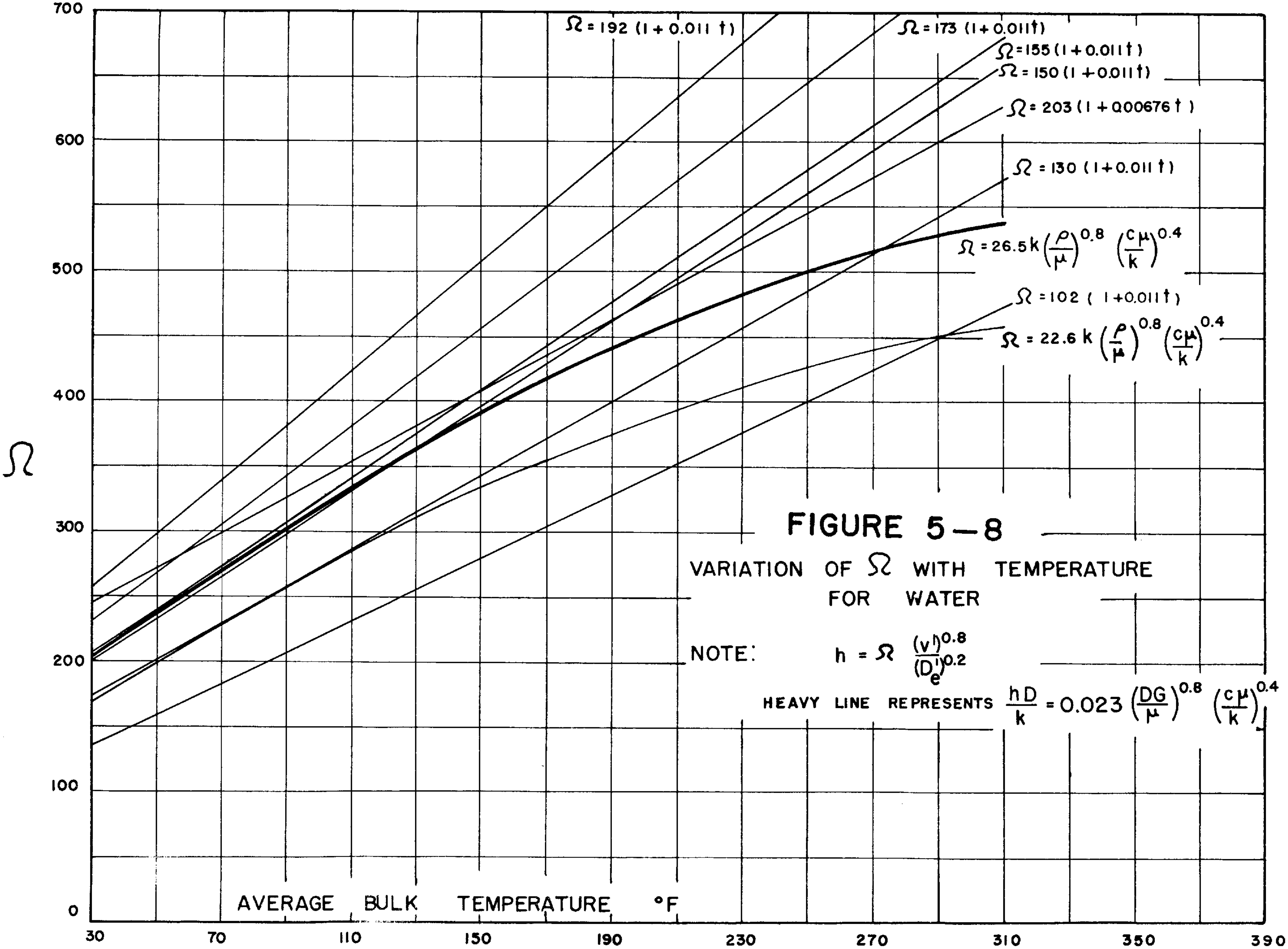


FIGURE 5—8

VARIATION OF Ω WITH TEMPERATURE FOR WATER

NOTE: $h = \Omega \dfrac{(v')^{0.8}}{(D_e')^{0.2}}$

HEAVY LINE REPRESENTS $\dfrac{hD}{k} = 0.023 \left(\dfrac{DG}{\mu}\right)^{0.8} \left(\dfrac{c\mu}{k}\right)^{0.4}$

UNCLASSIFIED

Drawing # 6763

Laminar Flow

Using D_2 of 1 3/8 in. and 2 in. in combination with D_1 of 3/4 in. and 1 in., Chen, Hawkins, and Solberg[28] examined heating and cooling of water in laminar flow in an annulus. The $\frac{L}{D_e}$ ratios employed were 179.8, 108, 67.5, and 54; and $\frac{D_eG}{\mu}$ ranged from 200 to 2,000. Arithmetic mean temerature differences were used to calculate heat transfer coefficients. The following equation was recommended as the best correlation for the conditions studied, with an average deviation of ± 6.6%.

$$\frac{h_a D_e}{k} = 1.02 \left(\frac{D_e G}{\mu_a}\right)^{0.45} \left(\frac{c\mu_a}{k}\right)^{1/3} \left(\frac{\mu_a}{\mu_w}\right)^{0.14} \left(\frac{D_e}{L}\right)^{0.4} \left(\frac{D_2}{D_1}\right)^{0.8} \left(\frac{D_e^3 \rho^2 \beta g \Delta t}{\mu_a^2}\right)^{0.05} \qquad \text{(Eq. 5.20)}$$

Carpenter, Colburn, and Schoenborn[16] investigated heat transfer to water in laminar flow with Reynolds numbers as low as 150 in a system having $\frac{D_2}{D_1} = 1.334$ and $\frac{L}{D_e} = 460$. The best correlation of the laminar flow data was obtained by using film properties and including the effect of free convection[8], as shown in the following equation:

$$\frac{h_L}{cG}\left(\frac{c\mu_f}{k}\right)^{2/3} = 1.615\left(\frac{D_e G}{\mu_f}\right)^{-2/3}\left[\frac{L}{(D_1 + D_2)\Phi}\right]^{-1/3} \qquad \text{(Eq. 5.21)}$$

The symbol Φ represents the free convection effect and is defined as follows:[8]

$$\Phi = \left(\frac{\mu}{\mu_f}\right)\left(1 + 0.015\, Gr_f^{\,1/3}\right)^3$$

$Gr_f = (\beta \Delta T)(D^3 \rho_f\, g/\mu_f^2)$, the Grashof number.

[redacted]

The difficulties involved in using Equation 5.21 limit its practical application; consequently, another equation was examined. It is based on the work of Norris and Streid[29], and the viscosity correction presented by Sieder and Tate[18]. Including modifications made by the experimenters, the equation is as follows:

$$\frac{h_L}{cG}\left(\frac{c\,\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 1.86\left(\frac{D_eG}{\mu}\right)^{-2/3}\left(\frac{L}{D_1 + D_2}\right)^{-1/3} \qquad \text{(Eq. 5.22)}$$

$$\frac{h_LD_e}{k} = 1.86\left(\frac{D_eG}{\mu}\right)^{1/3}\left(\frac{c\,\mu}{k}\right)^{1/3}\left(\frac{D_1 + D_2}{L}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \text{(Eq. 5.22a)}$$

Equation 5.22 can also be written as follows:

$$\frac{h_LD_e}{k}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 1.86\left(\frac{4}{\pi}\frac{wc}{k\,L}\right)^{1/3} \qquad \text{(Eq. 5.22b)}$$

This equation is almost identical with an equation for streamline flow in circular tubes[18,27].

In a discussion[17] comparing the experiments of Chen et al and Carpenter et al, it is pointed out that recommended equations were not based on the same type of temperature difference. (Norris and Streid[29] gave a good discussion of the choice of temperature difference.) It appears that Chen et al used the arithmetic mean temperature difference and Carpenter et al used the logarithmic mean temperature difference. The data of Carpenter et al, when plotted on the same basis as the work of Chen et al, gave good agreement for Re $\gtrsim$ 500 but fell about 40% low at Re $\simeq$ 200. When data of Chen et al were plotted on a basis of the logarithmic mean temperature difference, agreement with Carpenter et al is good for the smaller annuli where natural convection effects are minimized.

Davis[6] recommended an equation of the following form for laminar flow heat transfer:

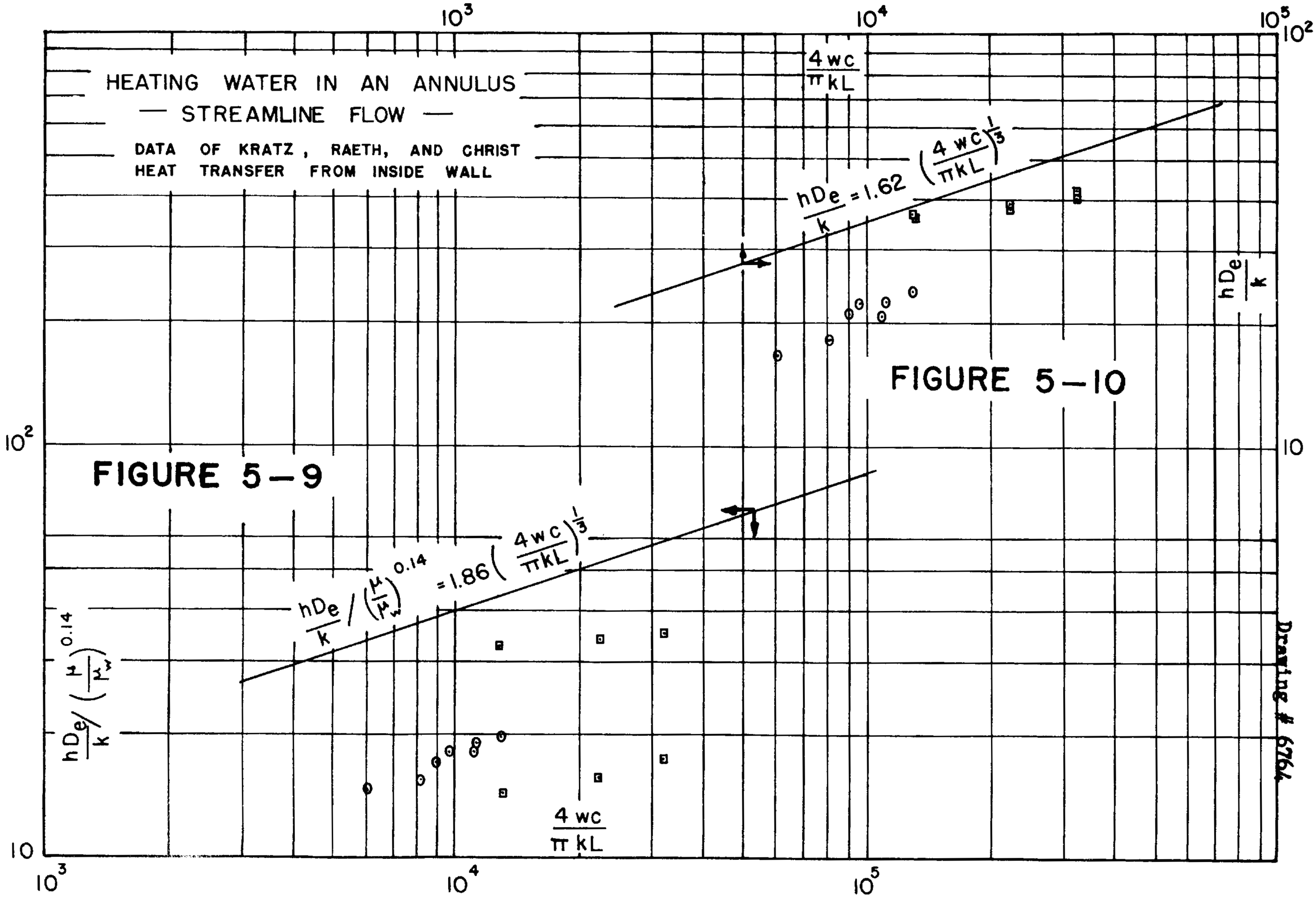
$$\frac{hD_a}{k} = \left(\frac{D_a G}{\mu}\right)^a \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \left(\frac{D_b}{D_a}\right)^e \left(\frac{L}{D_a}\right)^f \qquad \text{(Eq. 5.23)}$$

The exponents have not been established, however, and this equation has little value for the present discussion.

In the survey of work done in project laboratories, the only data on heat transfer to water in streamline flow in an annulus were found in a report of work by Kratz, Raeth, and Christ[24]. Since the data are meager and since the annuli examined are small enough to minimize the effect of natural convection ($D_e' = 0.189$ in. and $D_e' = 1.65$ in.) it is decided to compare the data with Equation 5.22b as shown on Figure 5-9. The viscosity correction appears to spread the data for one set of runs, so it is decided to use another equation recommended for circular tubes[27,30], with D_e substituted for D:

$$\frac{hD_e}{k} = 1.62 \left(\frac{4wc}{kL}\right)^{1/3} \qquad \text{(Eq. 5.24)}$$

Comparison of the data is shown on Figure 5-10 and it is seen that agreement between the data and predictions is slightly better for Equation 5.24 than for Equation 5.22b.



Drawing # 6764

For the case of circular tubes, $\frac{4wc}{\pi kL} = \left(\frac{DG}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{D}{L}\right)$, but it does not follow that $\frac{4wc}{kL} = \left(\frac{D_eG}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{D_e}{L}\right)$. This gives one cause to wonder which group, $\frac{4wc}{kL}$, or $\left(\frac{D_eG}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{D_e}{L}\right)$, is best to correlate data for an annulus. The work of Carpenter, Colburn, and Schoenborn indicated that $\frac{4wc}{kL}$ should be used. Nevertheless, it is decided to examine the data of Kratz, Raeth, and Christ with respect to the two equations which follow:

$$\frac{hD_e}{k} = 1.62 \left[\left(\frac{D_eG}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{D_e}{L}\right)\right]^{1/3} \qquad \text{(Eq. 5.25)}$$

$$\frac{hD_e}{k}\left(\frac{\mu}{\mu_w}\right)^{0.14} = 1.86 \left[\left(\frac{D_eG}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{D_e}{L}\right)\right]^{1/3} \qquad \text{(Eq. 5.26)}$$

The data are shown compared with these equations on Figures 5-11 and 5-12.

It appears that of the correlations attempted, Equation 5.26 gives the best agreement with the data of Kratz, Raeth, and Christ for heat transfer to water in laminar flow in an annulus. Equation 5.26 has not been justified theoretically and it is clear that either of the other three equations tried could be made to fit the data as well as Equation 5.26 by altering the constant coefficient. It is suspected that the spread created in one set of points by applying the viscosity correction may be a result of local boiling, but calculations from the data do not verify this suspicion. In the set of runs which exhibit this spread, $\frac{L}{D_e} = 12.1$, $\frac{D_2}{D_1} = 1.12$, and Reynolds numbers ranged from 950 to 2320. In the other set of runs, $\frac{L}{D_e} = 21.8$, $\frac{D_2}{D_1} = 1.13$, and Reynolds numbers ranged from 835 to 1860.

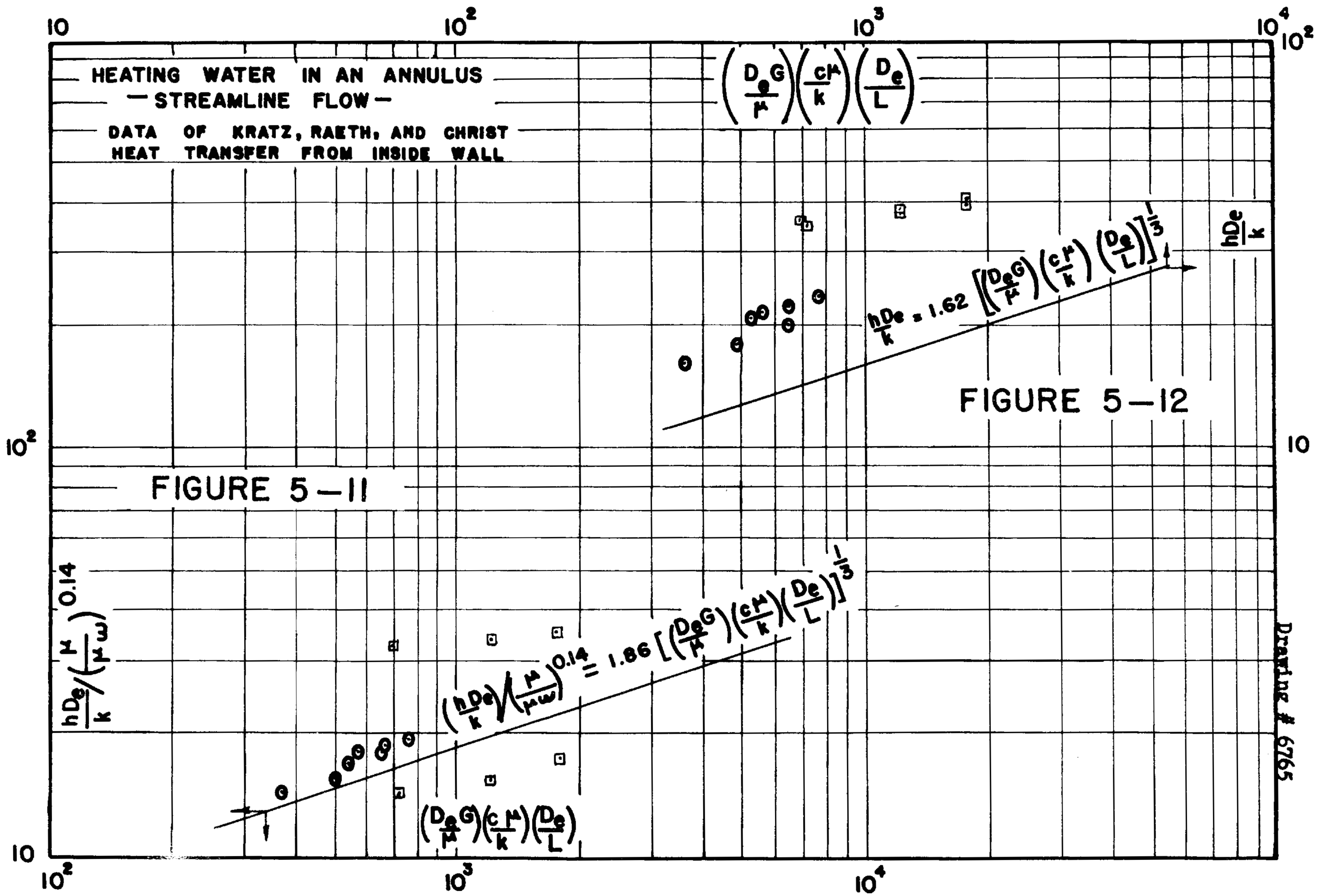


FIGURE 5-11

FIGURE 5-12

Drawing # 6765

In view of the peculiar spread resulting from the viscosity correction and the very short $\frac{L}{D_e}$ ratios which may have produced curious end effects, these data are not considered to be useful in establishing the validity of a correlation for this case.

Effect of Ribs in Annulus

Kratz, Schlegel, and Christ[31] described heat transfer experiments for determining the effect of ribs in an annulus. The apparatus essentially consisted of a brass heater tube surrounded by a four-ribbed aluminum tube. The brass tube was heated by passing an electric current through it, and heat was removed by passing water through the annulus between the brass tube and the ribbed aluminum tube. The ribbed tube and the brass tube were electrically insulated from each other by a coat of glyptal resin paint on the inside of the ribbed tube. The brass tube was filled with bakelite except for four thermocouple lead channels next to the inner surface of the brass tube. The apparatus was built in such a manner that the heater tube could be rotated so as to permit a complete angular traverse by the thermocouples while the ribbed tube remained stationary. Data which were taken included mean water temperature, heat flux, and heater tube wall temperature at various locations with respect to the ribs. Figure 5-13 is an average set of experimental data for the tube wall temperature traverse with water velocity of 25.9 ft./sec., flux of about 126,000 Btu/hr. ft.^{2}, and bulk water temperature of 42.8 °F. Other runs having the same conditions varied about ± 13% from the curve shown. The dotted line shown on Figure 5-13 is obtained by neglecting ribs and assuming that the Dittus-Boelter type equation[9] (Eq. 5.03d) holds. This permitted evaluation of a heat transfer coefficient which could be used, in combination with the flux, to calculate the difference between the average water temperature and the average temperature of the surface. A strict analysis of the problem is complicated by the fact

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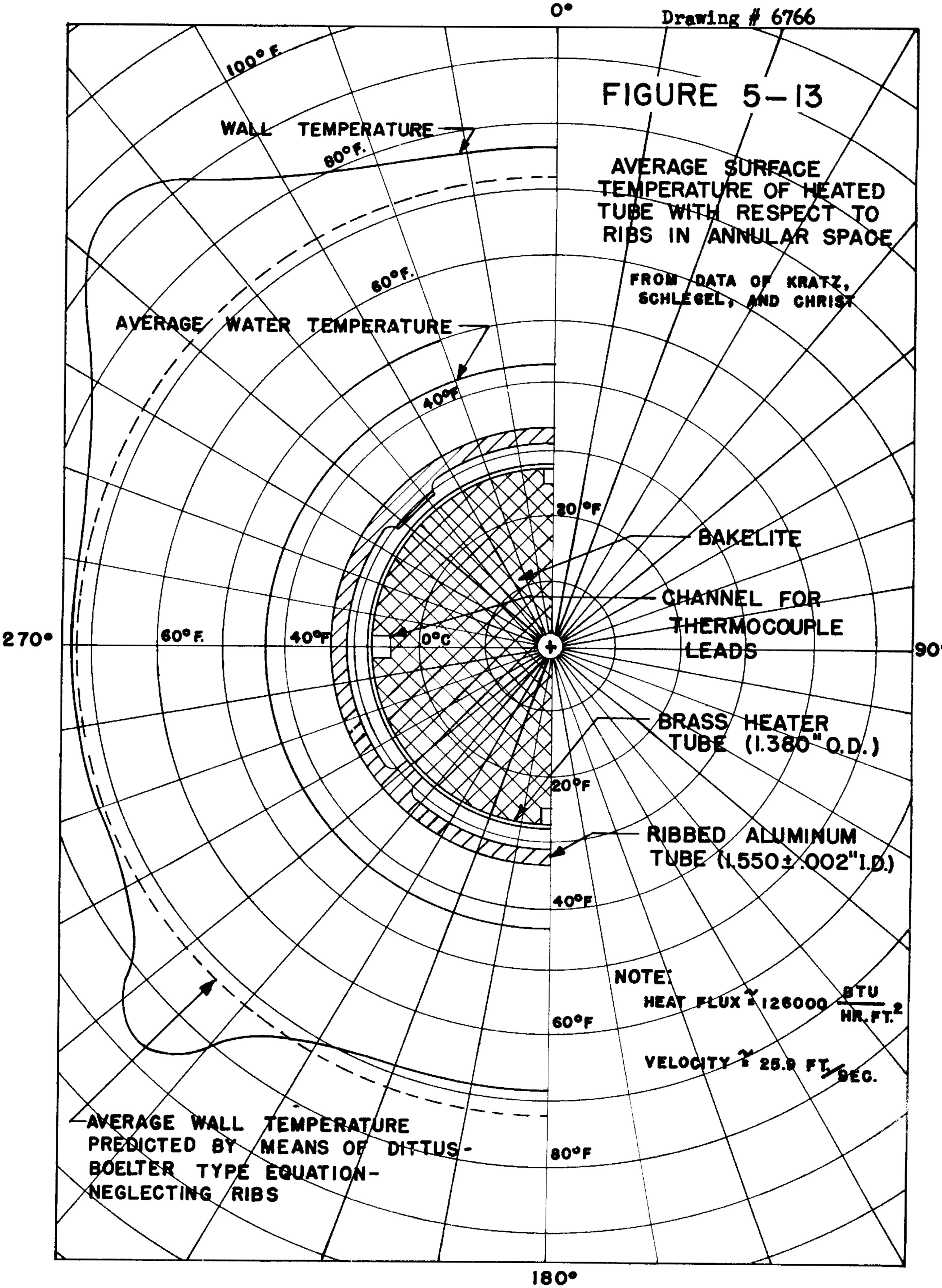

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that the ribs alter the velocity distribution and turbulence level in the stream, and Reynolds number would be strictly a function of location. It also follows that average bulk temperature may depart considerably from temperature in the immediate vicinity of ribs.

A small amount of work on this subject was reported by Cabell, Dunbar and Haaga[32].

Effect of Eccentricity

Wiegand and Baker[5] referred to the work of Caldwell[33] on pressure drop in an off-center annulus, but no references were found on the effect of eccentricity on heat transfer in off-the-project reports.

On the project, however, Kratz, Peterson, and Schegel[34] have reported extensive experiments on slug temperature in an off-center annulus. Two thermocouples were mounted in opposite sides of the slug beneath the slug surface, and the slug was covered with an unbonded aluminum jacket. The thermocouples were oriented in such a way that, as the slug was moved off-center, they fell on the same radius with the points on the slug jacket which were nearest and farthest from the outer wall of the annulus. Heat was supplied by a graphite electrode passing through a hole in the center of the slug, and cooling was accomplished by passing water in the annulus made by the slug jacket (1.440 in. O.D.) and an outer tube (1.625 in. I.D.). Experimental runs were performed using two bulk water temperatures - 129° F. (54° C.) and 158° F. (70° C.); power levels of approximately 12,860 Btu/hr. (900 cal./sec.), 20,000 Btu/hr. (1400 cal./sec.), and 27,200 Btu/hr. (1900 cal./sec.); and average water velocities of 15.7 ft./sec. (480 cm./sec.), 22.0 ft./sec. (670 cm./sec.), and 29.8 ft./sec. (910 cm./sec.). In the original report by Kratz, Peterson, and Schlegel, figures were included to show the variation of measured slug surface temperature with velocity while the slug was in contact position and the power level was constant. Other figures showed the variation of slug surface temperature with degree of eccentricity, while velocity, bulk temperature, and power level were held constant. One particularly interesting result of their experiments is that the slug temperature measured on the side farthest from the outer wall increased as the slug approached contact position. Possible explanations for this effect are offered by the experimenters, and, rather than to repeat them here, the reader is referred to their [illegible] reports[34,35]. In addition to the explanations offered by the experimenters, it is suggested that the effective heat exchange area is reduced in the contact position or that the water in the vicinity of the line of contact may be in laminar flow, increasing the thermal resistance. The case is difficult to analyze since velocity distribution and temperature distribution in the stream have not been explored and the actual slug jacket surface temperature was not measured. These comments are not intended as criticisms of the work, for seemingly, the experimenters did an excellent job of accomplishing the objectives which they had in mind. It is clear, however, that the work does not lend itself to application on heat transfer design problems except for estimating orders of magnitude of the effect of eccentricity.

Murray[36] treated mathematically this case of eccentricity, and estimated the maximum surface temperature of a slug displaced from the center of a tube.

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES; A CRITICAL SURVEY

PART 6

HEAT TRANSFER TO FLUIDS IN TURBULENT FLOW WITHIN CHANNELS HAVING LOW LENGTH TO DIAMETER RATIO

W. B. Harrison
October 1, 1948

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PART 6

HEAT TRANSFER TO FLUIDS IN TURBULENT FLOW WITHIN CHANNELS HAVING LOW LENGTH TO DIAMETER RATIO

In general, the widely used heat transfer correlations for turbulent flow are based on data taken in systems having length to diameter ratios of the order of 50 or greater. Systems employing small values of L/D are known to give higher heat transfer coefficients, but no correlations for the case have been well established. Heat transfer in circular tubes and rectangular channels is considered in the following paragraphs, with special reference to the effect of low L/D ratios.

Work of Kratz, Raeth, and Christ[1,2] included the case of heating water in turbulent flow for circular tubes having L/D ratios as low as 4. McAdams[3] mentioned studies of Lawrence and Sherwood[4] which indicated no effect of the $\frac{L}{D}$, but ratios they employed were between 32 and 224. A recent article by Cholette[5] presented an excellent review of the thinking about the effect of $\frac{L}{D}$ and showed experimental evidence which lends itself to use in establishing new equations for the low $\frac{L}{D}$ range. His experiments were made using air flowing inside steam-heated tubes. The steam jacket was compartmentalized so as to permit determination of local heat transfer coefficients which were average values over a length of about ten diameters. Data of Cholette included the laminar region, the transition region, and the turbulent region, but, since the data of Kratz, Raeth and Christ[1,2] were all in the turbulent region, the present discussion will be limited to the turbulent region alone.

For systems employing Re $>$ 8000 and L/D $<$ 63, the equations recommended by Cholette for heating air are as follows:

$$\frac{h_L}{cG} = 0.040\left(\frac{DG}{\mu}\right)^{-0.2}\left(\frac{L}{D}\right)^{-0.1} \qquad \text{(Eq. 6.01)}$$

or

$$\frac{h_a}{cG} = 0.028\left(\frac{DG}{\mu}\right)^{-0.15}\left(\frac{L}{D}\right)^{-0.15} \qquad \text{(Eq. 6.02)}$$

For $\frac{L}{D} > 63$, Cholette recommended the following equation for heating air:

$$\frac{h_L}{cG} = 0.0265\left(\frac{DG}{\mu}\right)^{-0.2} \qquad \text{(Eq. 6.03)}$$

It is seen that if $\frac{c\mu}{k}$ for air is assumed to be 0.74 and the Nusselt number is assumed to be a function of $\left(\frac{c\mu}{k}\right)^{0.4}$, the equations of Cholette may be written as follows:

$$\frac{h_L D}{k} = 0.0354\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4}\left(\frac{L}{D}\right)^{-0.1} \qquad \text{(Eq. 6.01a)}$$

and

$$\frac{h_L D}{k} = 0.0235\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4} \qquad \text{(Eq. 6.03a)}$$

For all practical purposes, Equation 6.03a is the same as the Dittus-Boelter type of equation described in Part 5 of the report, and the work lends support to the observations of Lawrence and Sherwood for $\frac{L}{D}$ of the order of 50 and larger.

In the experiments by Cholette, Reynolds numbers between 678 and 18,250 were explored for $\frac{L}{D}$ ratios from 10.5 to 63 in increments of 10.5.

[redacted]

For heating water, Christ[2] used an $\frac{L}{D}$ of 8.04 for three different groups of runs as shown below:

Group	Ave. Water Temp. °F.	Range of Re
1	64	58700 - 104200
2	105	93700 - 172600
3	159	156500 - 288500

Kratz, Raeth, and Christ[1] investigated the systems indicated below:

System	L/D	Range of Re	Ave. Water Temp. Range, °F.
1	32.8	9000 - 39800	59-72
2	9.13	15200 - 29100	69-77
3	5.12	4000 - 23200	49-54
4	3.97	7000 - 12400	42-52

As a preliminary attempt to correlate the data of Kratz, Raeth, and Christ[1,2] the Dittus-Boelter type of equation is used as shown on Figure 6-1. All the data are seen to fall above the Dittus-Boelter type of equation, supporting previous evidence that low L/D results in increased heat transfer coefficients.

For the run having $\frac{L}{D} = 32.8$, the great deviation from the Dittus-Boelter type of equation is difficult to explain. This $\frac{L}{D}$ is large enough that the data would be expected to fall much closer to the equation. It also appears that the data for this system indicate an influence of $Re^{1.2}$ rather than $Re^{0.8}$. The report does not contain sufficient information to permit further study of the deviations, so the data of the system having $\frac{L}{D} = 32.8$ cannot be considered as significant without additional data to confirm this unusual effect. It is suspected that

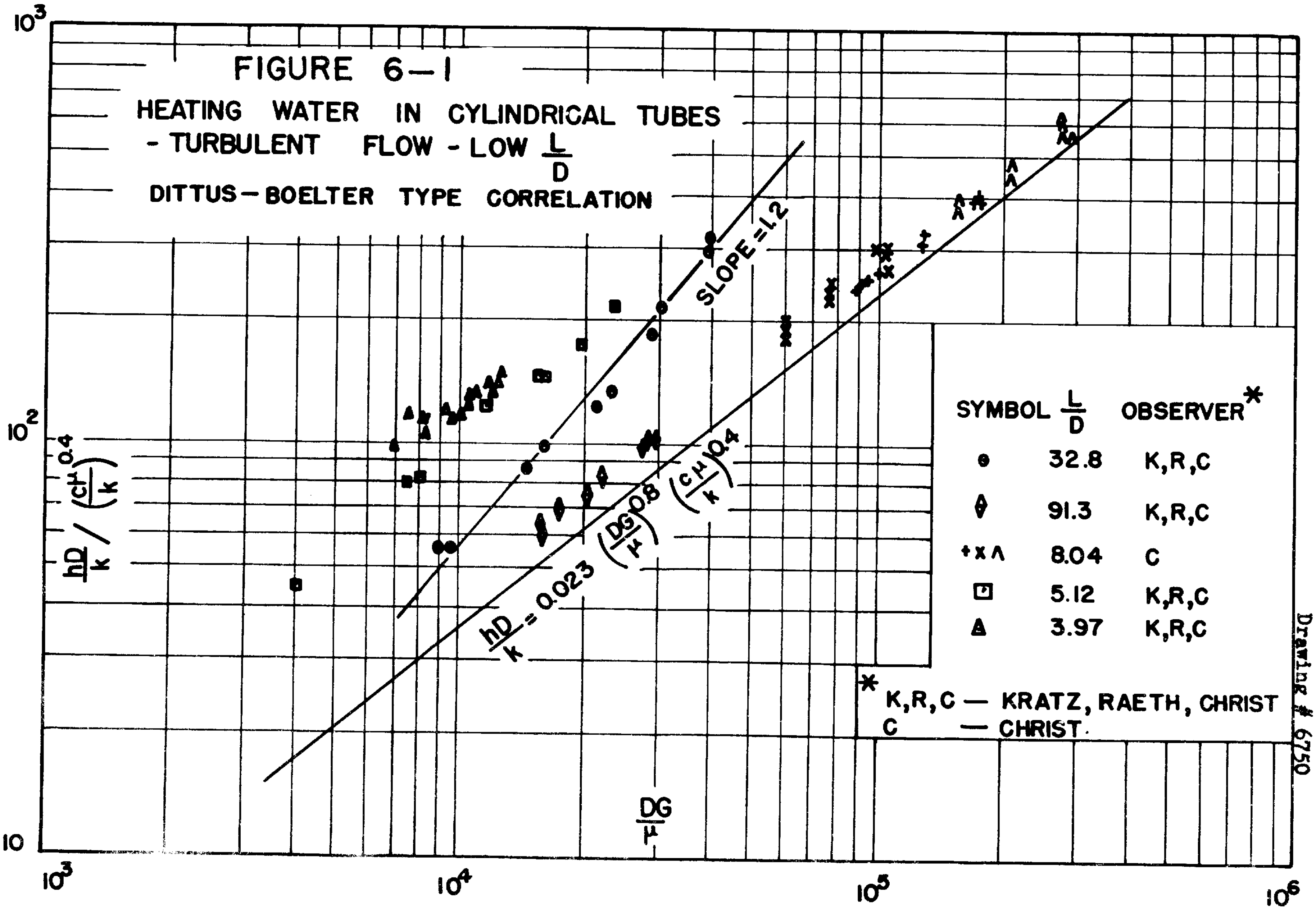


FIGURE 6-1

HEATING WATER IN CYLINDRICAL TUBES - TURBULENT FLOW - LOW $\frac{L}{D}$

DITTUS-BOELTER TYPE CORRELATION

errors occurred in evaluating the temperature change in the water or the temperature difference between the water and the tube wall, since the other data taken by the same observers confirm the influence of $Re^{0.8}$ and since effect of temperature differences would become more pronounced in the longer tube.

In determining the effect of low $\frac{L}{D}$ on the heat transfer, $\chi \equiv \frac{Nu}{Re^{0.8}\,Pr^{0.4}}$ can be plotted against $\frac{L}{D}$. The average value of χ is determined from Figure 6-1 by extending a line of slope 0.8 through each set of points in order to get its intercept with a given ordinate; i.e., $Re = 10^5$. Then the intercept is divided by $Re^{0.8}$ in order to get χ. A plot of χ vs. $\frac{L}{D}$ is shown on Figure 6-2 and this plot indicates that for heating water in systems having an L/D ratio between 4 and 9,

$$\chi = \frac{0.69}{\left(\frac{L}{D}\right)^{1.5}} \quad \text{or} \quad \frac{hD}{k} = 0.69\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c\,\mu}{k}\right)^{0.4}\left(\frac{L}{D}\right)^{-1.5} \qquad \text{(Eq. 6.04)}$$

The data of Kratz, Raeth, and Christ[1,2] are shown on Figure 6-3 as compared with Equation 6.04. The agreement with this equation is interesting but caution must be used in applying it until more data are obtained in this region.

If one assumes that Equation 6.04 is a fair empirical approximation for $\frac{L}{D} < 10$, it is seen that the equation does qualify in one point theoretically - the heat transfer coefficient increases greatly as $\frac{L}{D}$ approaches zero. For the present, the work of Cholette[5] is suggested as a basis for predicting the effect of $\frac{L}{D}$ between 10 and 60.

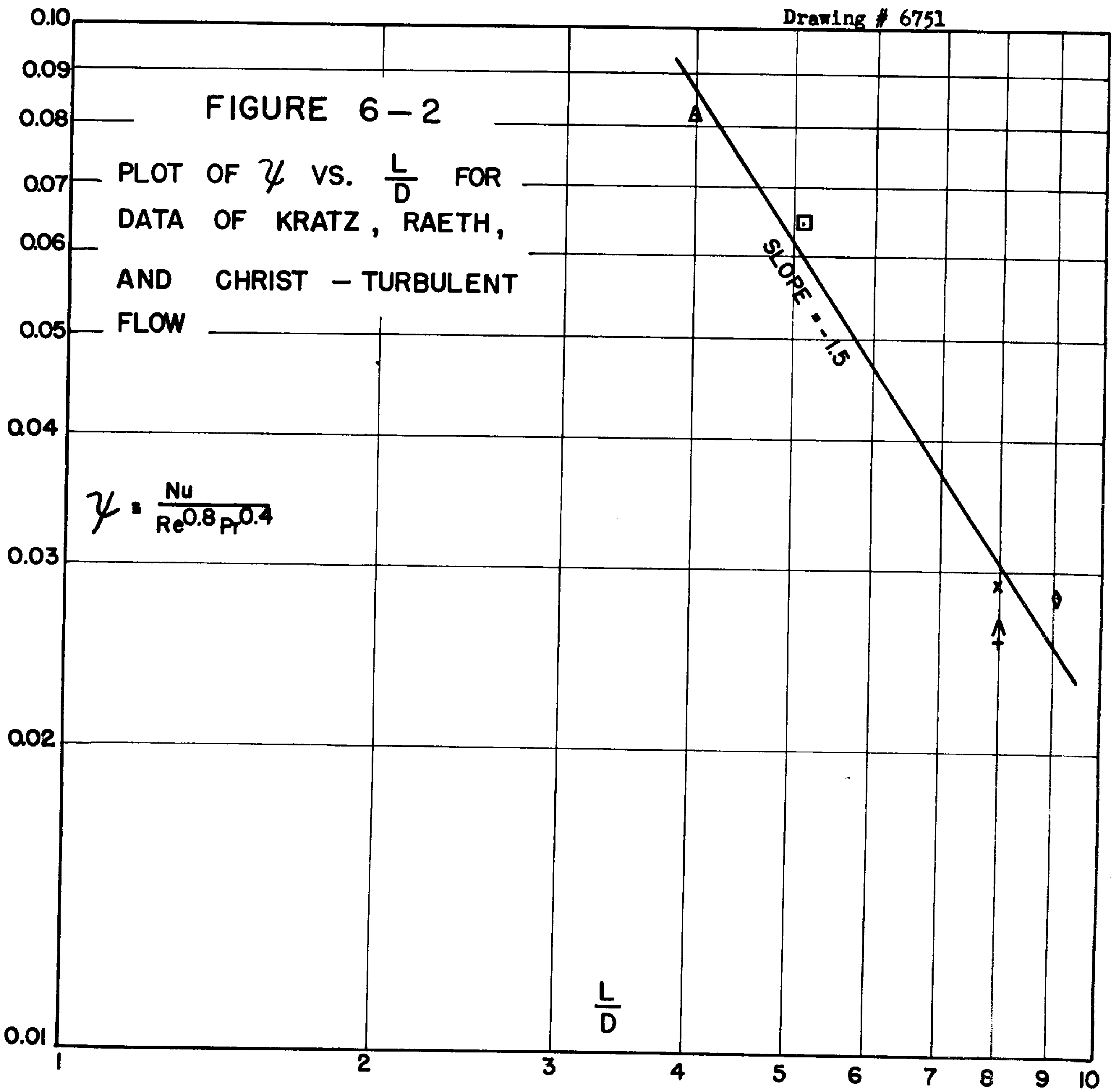
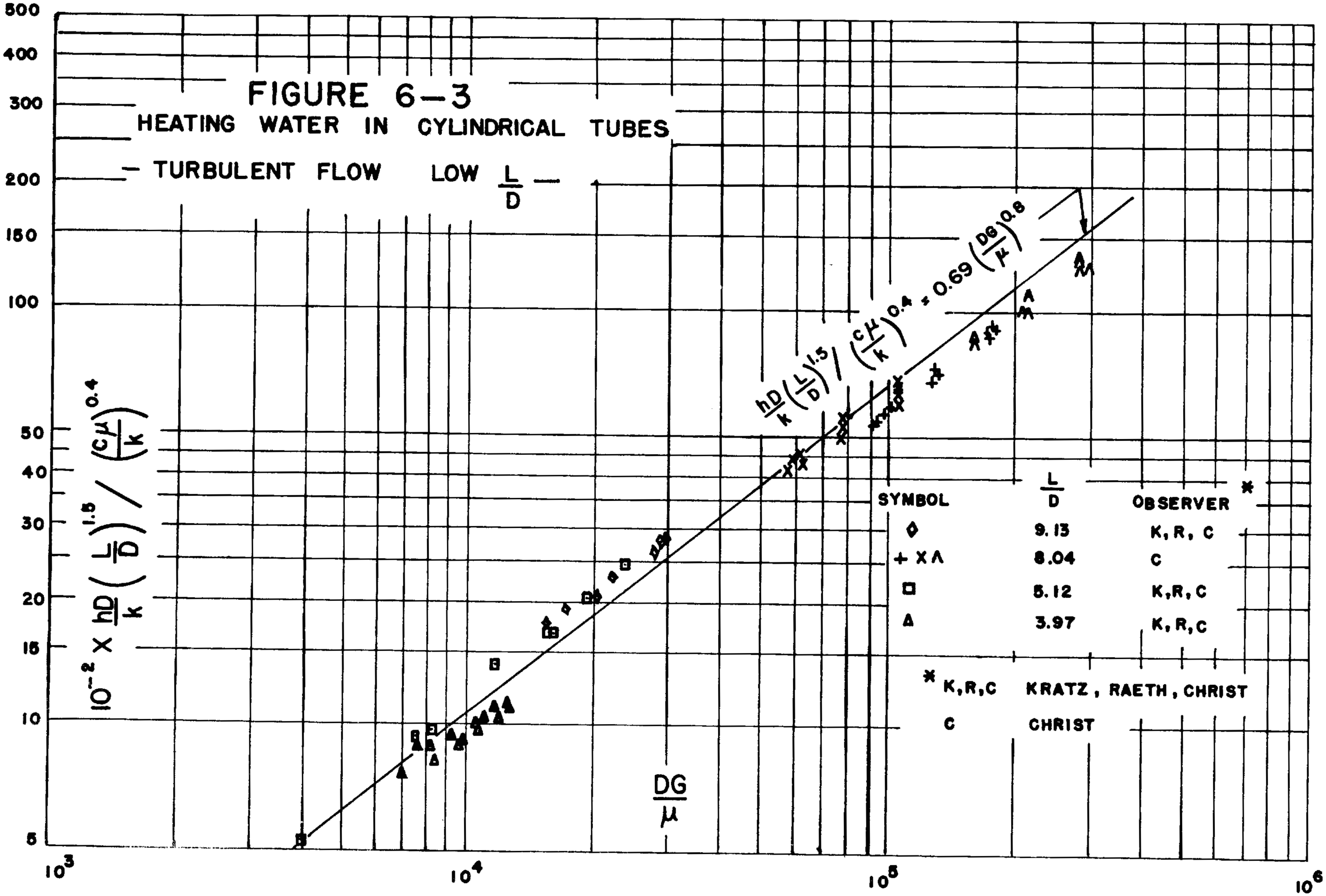


FIGURE 6-2

PLOT OF ψ VS. $\frac{L}{D}$ FOR DATA OF KRATZ, RAETH, AND CHRIST – TURBULENT FLOW

[redacted]



Drawing # 6752

[redacted]

Studies on heat transfer to helium in turbulent flow between parallel plates have been presented by R. N. Lyon[6]. The plates were electrically heated, 1/8 in. thick graphite, 3 5/8 in. long, and separated at the edges by 1/8 in. thick transite spacers. Widths of the plates varied so as to permit the assembly of parallel plates into a 3 5/8 in. diameter hole which necked down to 3 1/4 in. diameter at each end of the assembly. In the first series of runs, the edges of the plates were square and the entrance and exit ducts to the test assembly were given a 30° taper from 3 1/4 in. down to 2 in. diameter. The second series of runs was made with a straight 3 1/4 in. diameter entrance and exit section. The third and fourth series of runs were made by tapering the upstream and downstream edges of each plate with a 30° included angle. Average heat transfer coefficients in the assembly were calculated on the basis of (1) total heat transfer rate in the assembly, (2) total effective area of plates parallel to the direction of gas flow, and (3) the average difference in temperature between the graphite surface temperature in the middle of the assembly and the arithmetic mean of the terminal temperatures of the gas. The results are briefly summarized below, as taken from the original report. The expression $h_{ave.}$ refers to the average heat transfer coefficient as defined above. The Dittus-Boelter type of equation (See Part 5) was used to calculate h_{theory}, with the use of the equivalent hydraulic diameter as D.

Summary of Results

Series	1	2	3	4
Plates tapered	No	No	Yes	Yes
Channel constricted	Yes	No	No	No
h_{ave}/h_{theory}	1.13	1.19	1.03	1.03

The report by Lyon included data plotted as heat transfer coefficient vs. mass velocity for three locations in the assembly. Gas rates ranged from 0.53 to 2.02 lb./sec., corresponding to a maximum velocity of about 735 ft./sec. Heat transfer coefficients of the order of 1300 Btu/hr.ft.^{2} °F. were reported.

Each channel in the assembly had a $\frac{\text{heated length}}{\text{equivalent diameter}} \cong 14.5$. It is of interest to note that the Equation 6.01a indicates h at $\frac{L}{D} = 15$ to be about 15% greater than h at $\frac{L}{D} = 60$. Since the Dittus-Boelter type of equation is assumed to be valid at $\frac{L}{D} \geqq 60$, the data of series 1 and 2 appear to support the approximation for $\frac{L}{D}$ effect. It seems reasonable that tapering the plates should influence the entrance effect and it is noted that for series 3 and 4 the data fall closer to the Dittus-Boelter type of equation than the data of Cholette at the same $\frac{L}{D}$. (Cholette used a cylindrical tube with a bell-shaped entrance having straightening vanes to introduce air to a bundle of parallel tubes.) This effect of tapering the plates is believed to be a result of reduced "turbulence level" as discussed later in this section of the report.

Influence of leading edge shapes on heat transfer to air was explored by Parekh[7] for three edge shapes and several angles of attack. His data confirmed the observations of Lyon in that, at zero angle of attack, blunt edges resulted in higher coefficients than tapered edges.

[redacted]

Parekh also investigated the heat transfer to a single plate and the effect of placing similar plates parallel and close to it. He employed a very interesting experimental technique involving unsteady state heat transfer from a hot body to the coolant, and reference to his paper is recommended for anyone who may be planning to do work of a similar nature.

Joyner[8] presented data which show that coefficients are as much as 100% greater in systems employing short-length staggered surfaces than in systems having the same heat transfer area and hydraulic diameter but greater length.

McAdams[9] reported experimental data taken by various observers in rectangular channels, but no data have been located which appear to be applicable for comparison with the data of Lyon. Washington and Marks[10] explored heat transfer to air in various channels, including one which was 1/8 in. thick and 5 in. wide. The $\frac{L}{D_e}$ ratio was 197, resulting in agreement with the Dittus-Boelter type of equation in which D_e is substituted for D, and the data are in agreement with observations of heat transfer to air in turbulent flow within cylindrical tubes.

Data of Green and King[11] also confirm the observations of Washington and Marks and the Dittus-Boelter type of equation for circular tubes. Their experiments were performed in three types of channels: cylindrical tubes, flattened tubes, and flattened tubes which were dimpled. The data for heating air in definitely turbulent flow came together for the circular and flattened tubes, and the data for the dimpled tube indicated somewhat higher heat transfer coefficients. Here again the $\frac{L}{D}$ was too great to make the data useful in determining the effect of $\frac{L}{D}$, but the fact that the coefficients for the dimpled tube were higher than for the flattened and circular tubes may be significant in developing the "turbulence level" which is discussed later in this section of the report.

Figure 6-4 summarizes the observations made here regarding the effect of L/D on heat transfer. The lines on Figure 6-4 are not considered to be accurate, due to the meager supply of data on which they are based, but they may serve as an approximation until such time as the effect is explored more carefully.

The effect of $\frac{L}{D}$ ratios on heat transfer is difficult to describe mathematically, particularly for turbulent systems. Latzko[12] presented an analytical treatment for four cases of turbulent flow as noted below:

"1. Fully developed hydrodynamic and thermal fields.-

This condition is attained when the fluid has passed through a considerable portion of the tube length. The unit thermal convective conductance is constant---------".

The equation Latzko recommended for this case is

$$\frac{h_{min.}}{cG} = 0.0384\left(\frac{DG}{\mu}\right)^{-1/4} \qquad \text{(Eq. 6.05)}$$

When one compares this equation with the data of Cholette, it is seen that, for $\frac{L}{D} > 10$ at $Re = 10^4$, Equation 6.05 gives a value of $h_{min} = 28.4$ and the data yield a corresponding value of approximately 29.5.

"2. Fully developed hydrodynamic flow field, temperature uniform at entrance section which is maintained at the original fluid temperature by suitable heating. - The unit thermal convective conductance is dependent upon the location in the tube, falls very

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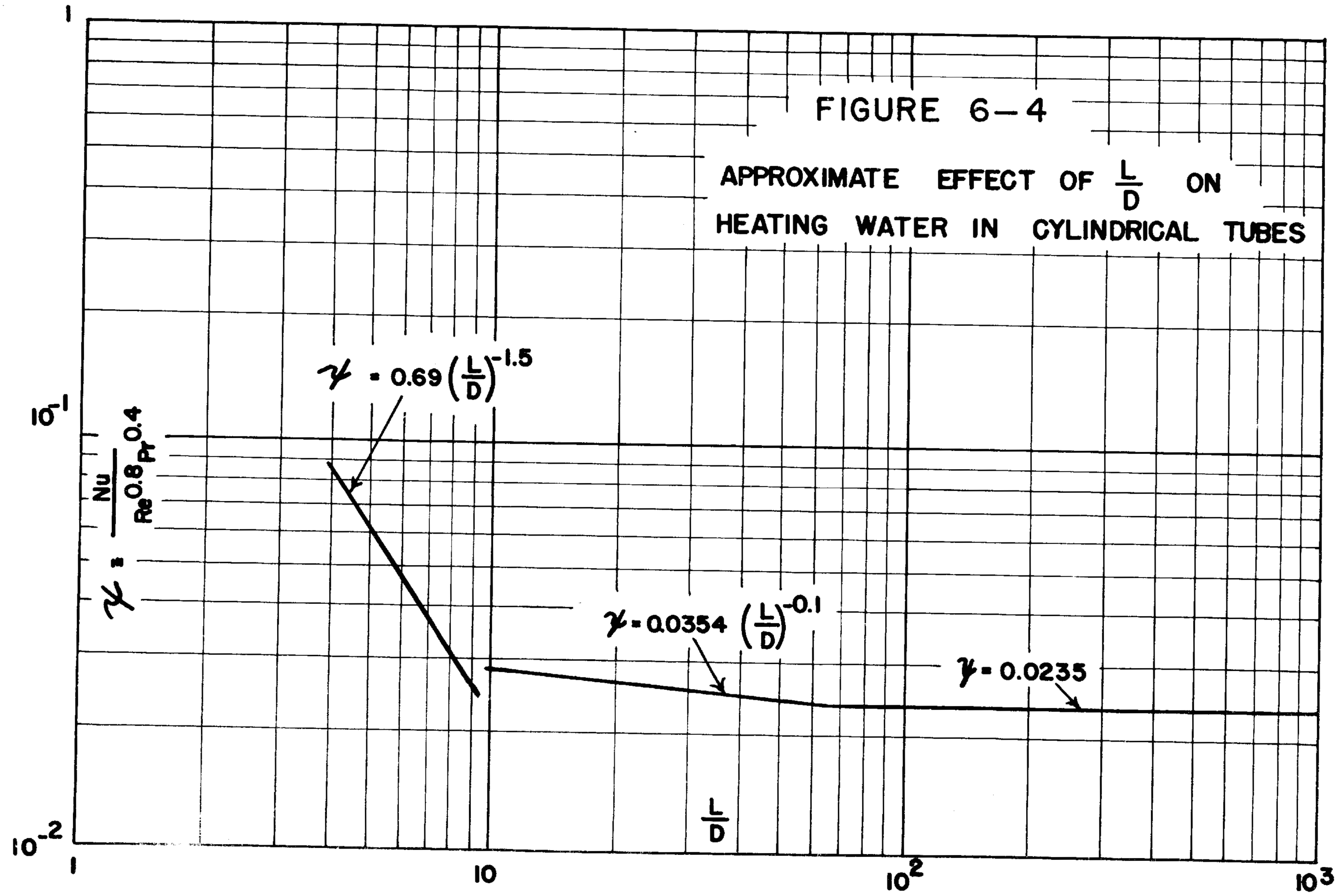

FIGURE 6-4

APPROXIMATE EFFECT OF $\frac{L}{D}$ ON HEATING WATER IN CYLINDRICAL TUBES

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Drawing # 6753

quickly from its maximum value, and asymptotically approaches a constant minimum value."

Latzko presented an equation for this case and it has also been explored by Sanders[13], Iverson[14,16] and Young[15,16].

"3. Uniform velocity and temperature distributions across the section at entrance. - The unit thermal convective conductance is likewise dependent upon the location in the tube, but falls to a minimum value more slowly."

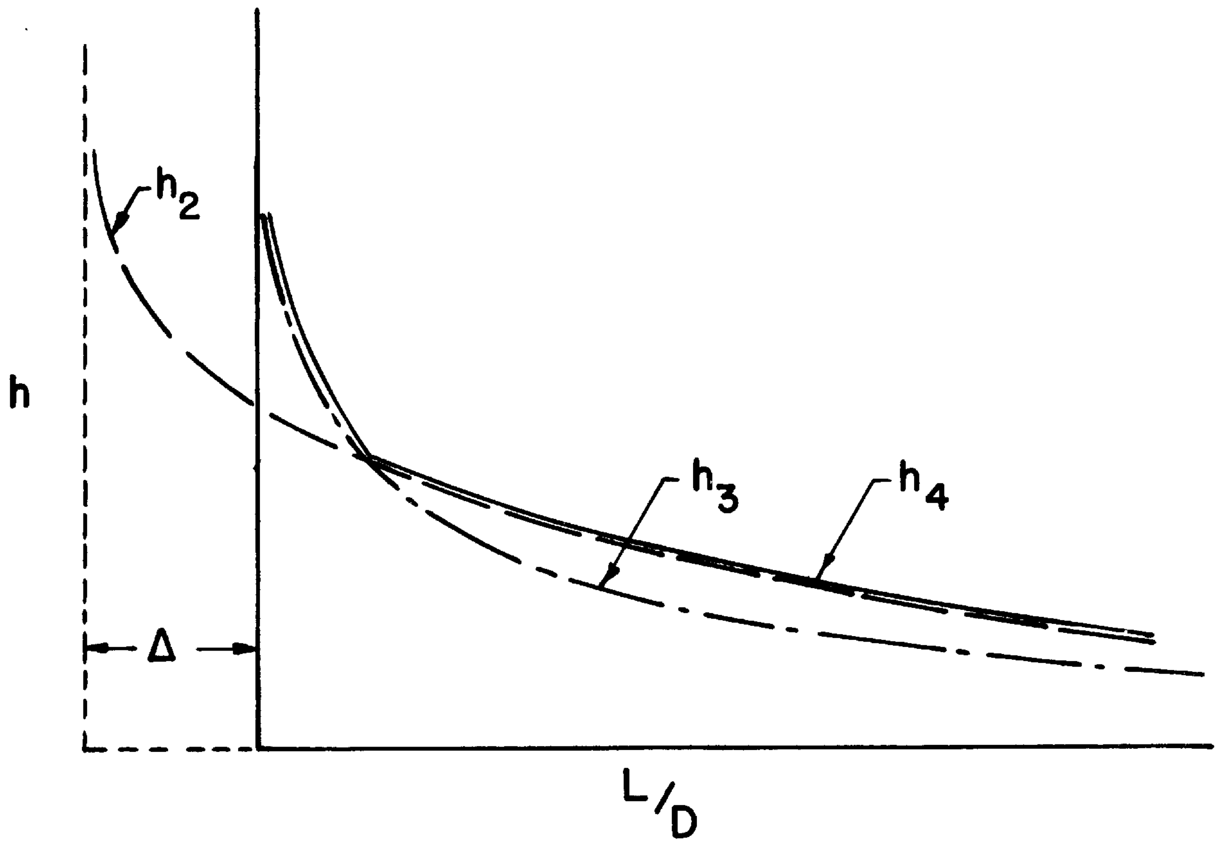
Boelter, Martinelli, et al[17], presented a discussion of the mechanism at an entrance by drawing an analogy between transition from laminar to turbulent boundary layers in ducts and the transition along a single flat plate.

"4. The application of heat begins at a section somewhere in the middle of the calming length." This case presented the need for combining the equations in cases 2 and 3 to form an envelope as shown on Figure 6-5. The heat transfer coefficients for cases 2 and 3 (h_2 and h_3, respectively) are plotted on the same coordinates but h_2 is displaced by Δ, the distance between the entrance and the beginning of heating. It is seen that, at the beginning of the heated length, the coefficient will be very high and will drop rapidly as in case 2. However, the hydrodynamic field is not fully established and the coefficient h_4 will be higher than is indicated by the h_2 curve. The amount by which h_4 is higher than h_2 may be estimated by displacing the h_3 curve by Δ, and indicating the values for coefficient which result if heating begins at the entrance. Then the envelope noted as h_4 represents conservative

Drawing # 6754

FIGURE 6-5

AN EXAMPLE OF CASE 4*



*TAKEN FROM NACA TM NO. 1068

estimates for the coefficients in a system described by Latzko as case 4.

It is believed that another factor of practical significance is the turbulence level at the entrance section. Latzko dealt with the fully developed hydrodynamic flow field and the uniform velocity field but perhaps the fundamental questions are:

1. What is the level of turbulence at the entrance section compared with the turbulence level after the hydrodynamic flow field is established?
2. What conditions affect the change of the turbulence level?
3. What is the quantitative effect of turbulence level on the heat transfer coefficient?

Little information is available on turbulence level, and these questions are raised as possible subjects for thought and experimental work. Comings, Clapp, and Taylor[18] presented data on the effect of turbulence level on heat transfer to air flowing normal to cylinders in a square duct. Among other things, they reported that at constant Reynolds number, Nusselt numbers could be increased 25% by increasing the turbulence alone. The effect of turbulence is also indicated in a study of gas turbine regenerator cores[19] in which hydraulic diameter and heated length were constant, but various core configurations were utilized. The $\frac{L}{D}$ ratios were constant, and the pre-entrance conditions were the same, but considerable difference in heat transfer coefficients resulted from different turbulent conditions. The same effect is indicated in heat transfer systems employing flow of liquids normal to single tubes and tube bundles. Heat transfer coefficients for tube bundles are generally greater than coefficients for single tubes and one would expect the turbulence level to vary the same way.

Young[15] presented information on the effect of entering conditions of the fluid on the heat transfer in the entrance section of a tube.

Data of Lyon[6], Parekh[7], Joyner[8], and Green and King[11] mentioned earlier in this section of the report also serve as additional evidence pointing toward the importance of degree of turbulence.

London[20] and Peebles[21] discussed the effect of L/D on heat transfer and pressure drop, with particular attention being paid to very small diameters and lengths.

Additional experimentation on heat transfer and pressure drop in short tubes is being initiated at Massachusetts Institute of Technology[22,23] It is expected that 1/32 to 1/8 in. diameter tubes will be employed in systems having L/D from 5 to 20. Large temperature drops and small Reynolds numbers will be investigated in tube bundles composed of 25 to 100 tubes in parallel.

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 7

REFERENCES RELATED TO HEAT TRANSFER WITH NO PHASE CHANGE

W. B. Harrison
October 1, 1948

[redacted]

PART 7

REFERENCES RELATED TO HEAT TRANSFER WITH NO PHASE CHANGE

It is beyond the scope of the present report to discuss many of the project reports related to heat transfer.

This part of the report is devoted to noting such references which may be of interest to workers in this field.

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES; A CRITICAL SURVEY

PART 8

HEAT TRANSFER TO BOILING LIQUIDS CIRCULATED BY NATURAL CONVECTION

W. B. Harrison
October 1, 1948

[redacted]

PART 8

HEAT TRANSFER TO BOILING LIQUIDS CIRCULATED BY NATURAL CONVECTION

Very little work has been done in project laboratories on heat transfer to boiling liquids. As a cooling mechanism for a reactor, boiling presents many difficulties from the design and operation standpoint. However, since a mechanism is greatly needed for maintaining high heat fluxes, boiling is now attracting much attention and several laboratories are engaged in attempts to completely define the variables which control the boiling mechanism. This part of the report is devoted to boiling liquids circulated by natural convection.

In 1937, Drew and Mueller[1] presented an excellent review of the status of information on the variables affecting heat transfer rates to boiling liquids. Jakob[2] had previously reported a survey of work done abroad, and, in 1942, McAdams[3] summarized most of the experimental data which were available at that time. A few additional references may be found in a survey by Kreith and Summerfield[4].

The case of boiling a liquid circulated by natural convection is represented qualitatively by Figure 8-1. The curve in this figure is considered to be representative when the bulk of the liquid is at or very near the saturation boiling point for the prevailing pressure, since it takes the general shape of the data summarized by McAdams[3],

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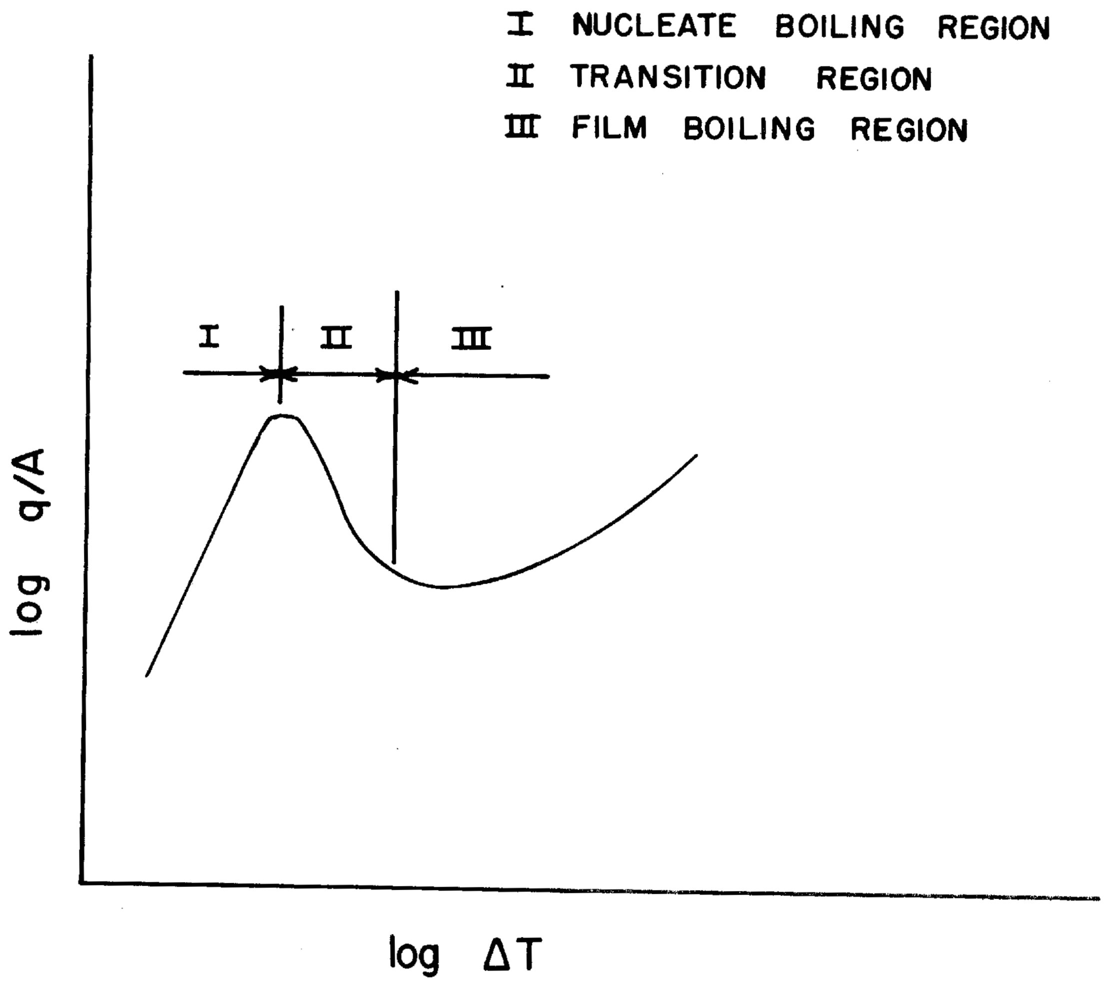


FIGURE 8-1

TYPICAL BOILING CURVE

the data of Farber and Scorah[5], of Boscov[6], and of Nukiyama[7] (as noted by Boscov[6], and Drew and Mueller[1]).

The boiling curve is generally divided into at least three regions - nucleate boiling, transition, and film boiling. Farber and Scorah[5] described six regions in the boiling curve, but the three noted will serve as a basis for the present discussion.

Nucleate boiling is characterized by an increase in heat flux with increase in ΔT, as shown as region I on Figure 8-1. Assuming that the liquid wets the heat transfer surface and that sufficient superheat is present to initiate boiling, bubbles form at the interface and they are released to the liquid bulk, either to condense or to pass to the surface of the liquid.

The film boiling is characterized by very large values of ΔT and, in this case, a stable vapor film which envelops the entire heat transfer surface, increasing the thermal resistance at the interface. This is shown as region III on Figure 8-1 and is limited by the thermal stability of the heat transfer surface.

Region II, the transition region, is the contest between film and nucleate boiling mechanisms for control.

The shape and exact location of the curves appear to be functions of the following variables:

1. pressure
2. nature of surface material
3. condition of surface
4. geometry of system
5. mechanical shock
6. physical properties of the liquid

Three sets of data from Farber and Scorah[5] are shown on Figure 8-2 to illustrate the effects of pressure and the nature of the surface material. Two of the curves are for Chromel C - one at atmospheric pressure and one at 100 psig. If one defines critical temperature difference[3], ΔT_c, as the ΔT at $(q/A)_{max}$, it is seen that the effect of increased pressure on boiling with Chromel C is to increase $(q/A)_{max}$ and reduce ΔT_c. The other curve is for nickel at atmospheric pressure and it was determined in the same apparatus as the curve for Chromel C. Farber and Scorah[5] reported similar curves at one atmosphere for tungsten and Chromel A, and they explored the pressure effect in 25 psig increments up to 100 psig for nickel, Chromel A, and Chromel C. Curves from Boscov[6] and Nukiyama[1,6,7] were obtained with a platinum heater and they are included for comparison. McAdams[6] suggested that the Nukiyama data are probably more accurate than the Boscov data in the film boiling range, since Nukiyama used a longer wire than did Boscov. The differences exhibited by the curves obtained at the same pressure are attributed to the use of different materials and systems of different geometry. One would expect geometry of the system to be important since circulation is by natural convection only, and the effect has been demonstrated experimentally by Nukiyama[7,27].

Larson[8] presented a discussion on factors affecting boiling in a liquid and related wettability or interfacial free adhesion energy to the superheat required for ebullition. In the experimental work he reported, ebullition temperatures were measured for various metal surfaces in contact with water at atmospheric pressure. The bulk temperature attained

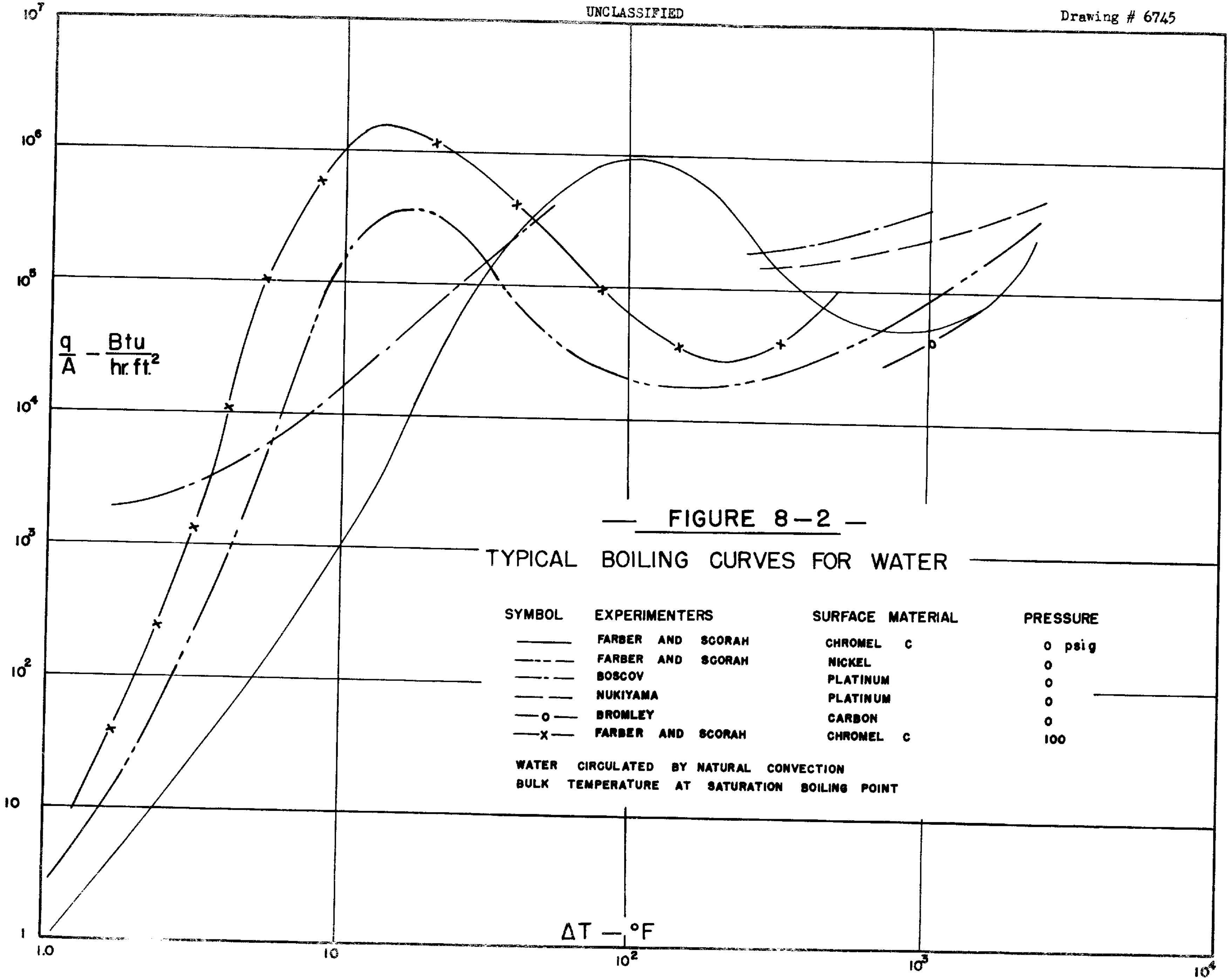


FIGURE 8-2

TYPICAL BOILING CURVES FOR WATER

by the water before ebullition frequently showed superheat, and a rapid drop to saturation boiling temperature occurred upon the beginning of ebullition. Several metal surfaces were examined, verifying the fact that the nature of the surface has considerable influence on the boiling mechanism. The effect of nature of surface material is also shown by data of Castles[9].

Larson[8] noted that mechanical shock would cause ebullition in the superheated water at lower temperatures than were necessary to cause ebullition without shock. Harvey, Barnes, et al[10] point out that, with proper de-gassing treatment, water can be superheated to 392° F (200° C.) without bubbling, but that mechanical shock or high frequency sound could cause bubbling at much lower temperature if the de-gassing treatment was not complete. Austin[11] investigated the effect of mechanical agitation on boiling in the low flux range and Carr[12] examined the effect of vibrating the heat transfer tube. Martinelli and Boelter[13] have also explored the effect of mechanical vibration on heat transfer by natural convection without change of phase.

There may be a close relation between nucleate and film boiling, and drop-wise and film condensation. Some of the considerations presented by Emmons[14] on drop-wise condensation may well be considered in acquiring an understanding of the boiling mechanism.

Reference to McAdams[3] is recommended for extending the background information on effects of pressure, wetting agents, adsorbed air, etc.

[redacted]

Farber and Scorah[5] included a few remarks on the effect of uniform and non-uniform surface roughness. Apparently a specific conditioning process was necessary before the heating wires gave uniform roughness characteristics and yielded reproducible data. Cichelli and Bonilla[15] explored the effect of pressure on boiling for several liquids. They compared their atmospheric pressure data for the nucleate boiling range with several equations from the literature[16,17,18,19,20] to find that the best predictions resulted from an equation recommended by Insinger and Bliss[20] -

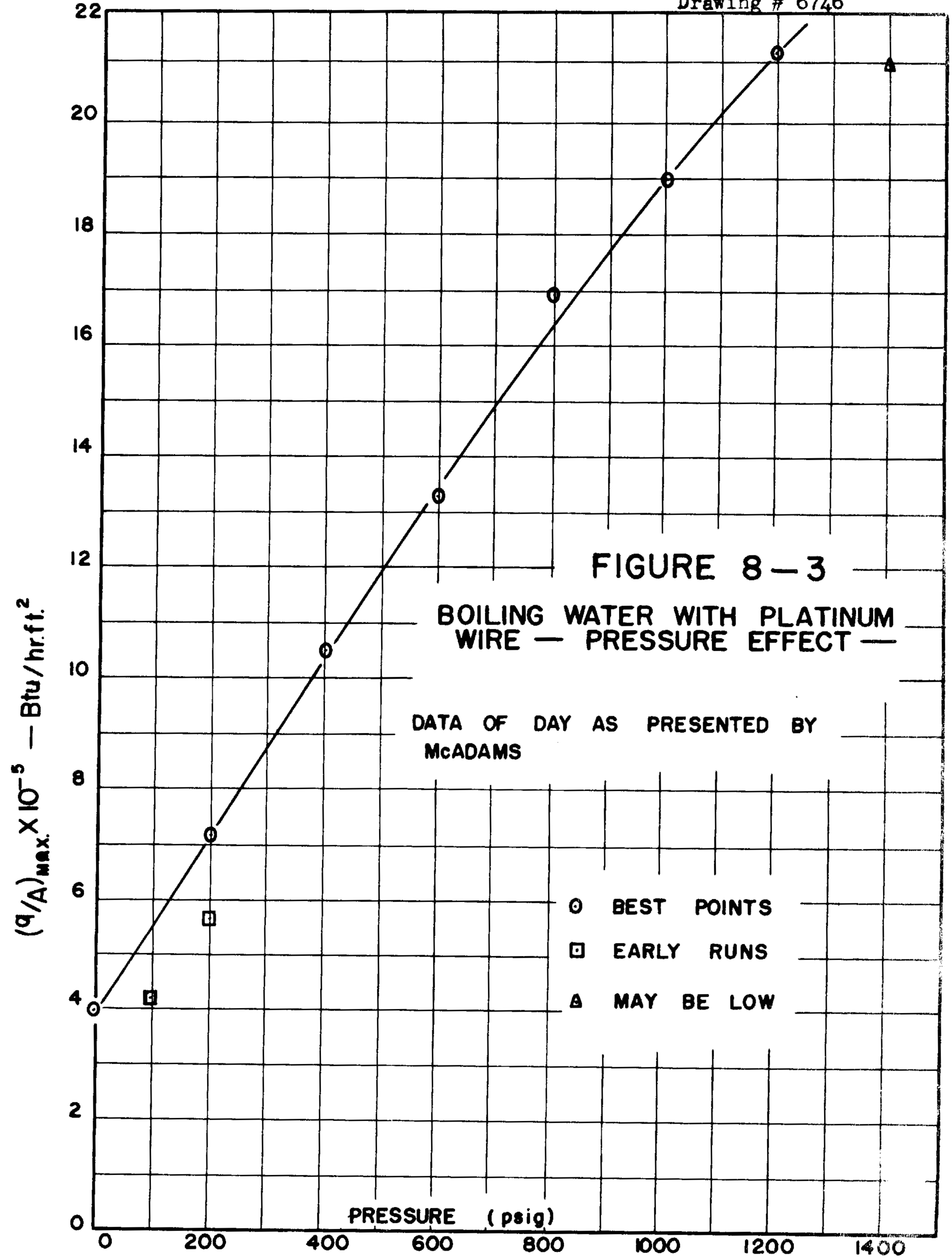
$$\log Y = 0.363 + 0.923 \log X - 0.047 (\log X)^2 \qquad \text{(Eq. 8.01)}$$

$$\text{where } Y = \frac{h(\rho_V \sigma)^{0.5}}{\left(c k \rho_L^3 \sqrt{\lambda^3 J^3 g}\right)^{0.5}} \times 10^{10}$$

$$\text{and } X = \frac{q}{A \rho_L \sqrt{\lambda^3 J g}} \times 10^{10}$$

A very interesting characteristic of this equation is that viscosity does not appear. Cichelli and Bonilla reported a fairly consistent relationship between $\frac{(q/A)_{max}}{P_c}$ and P_r for organic liquids, but it is apparent, upon examining the data of Farber and Scorah[5], that such a treatment will not bring the data for water together without an additional factor for nature of surface material. Day presented data for boiling water with a platinum wire at pressure up to 1400 psig. These data are shown on Figure 8-3 (as presented by McAdams[21]) to illustrate the pressure effect.

Drawing # 6746



Among the boiling data found in project files, the natural convection case was reported primarily by Hoge and Brickwedde[22] (boiling liquid oxygen and liquid hydrogen at atmospheric pressure in the nucleate range) and by Bromley[23] (boiling water and liquid nitrogen in the film boiling range). Due to the difficulty of locating data on the physical properties of liquid oxygen and hydrogen, the small amount of interest generally shown in these particular liquids as coolants, and the difficulties of using Equation 8.01, the data of Hoge and Brickwedde have not been compared with the equation, but they are shown graphically on Figure 8-4.

Bromley[23] developed some approximate equations which correlated his data fairly well for the case of boiling water and nitrogen in the film boiling range. For horizontal tubes,

$$h_c = \epsilon \sqrt[4]{\frac{k^3 \rho_v (\rho_L - \rho_v) g \lambda'}{D \Delta t \mu}} \qquad \text{(Eq. 8.02)}$$

For vertical tubes,

$$h_c = 1.2 \epsilon \sqrt[4]{\frac{k^3 \rho_v (\rho_L - \rho_v) g \lambda'}{D \Delta t \mu}} \qquad \text{(Eq. 8.03)}$$

The radiation coefficient was calculated with the following expression:

$$h_r = \frac{\sigma}{\left(\frac{1}{\epsilon_s} + \frac{1}{\alpha_L} - 1\right)} \frac{(T_S^4 - T_L^4)}{\Delta t} \qquad \text{(Eq. 8.04)}$$

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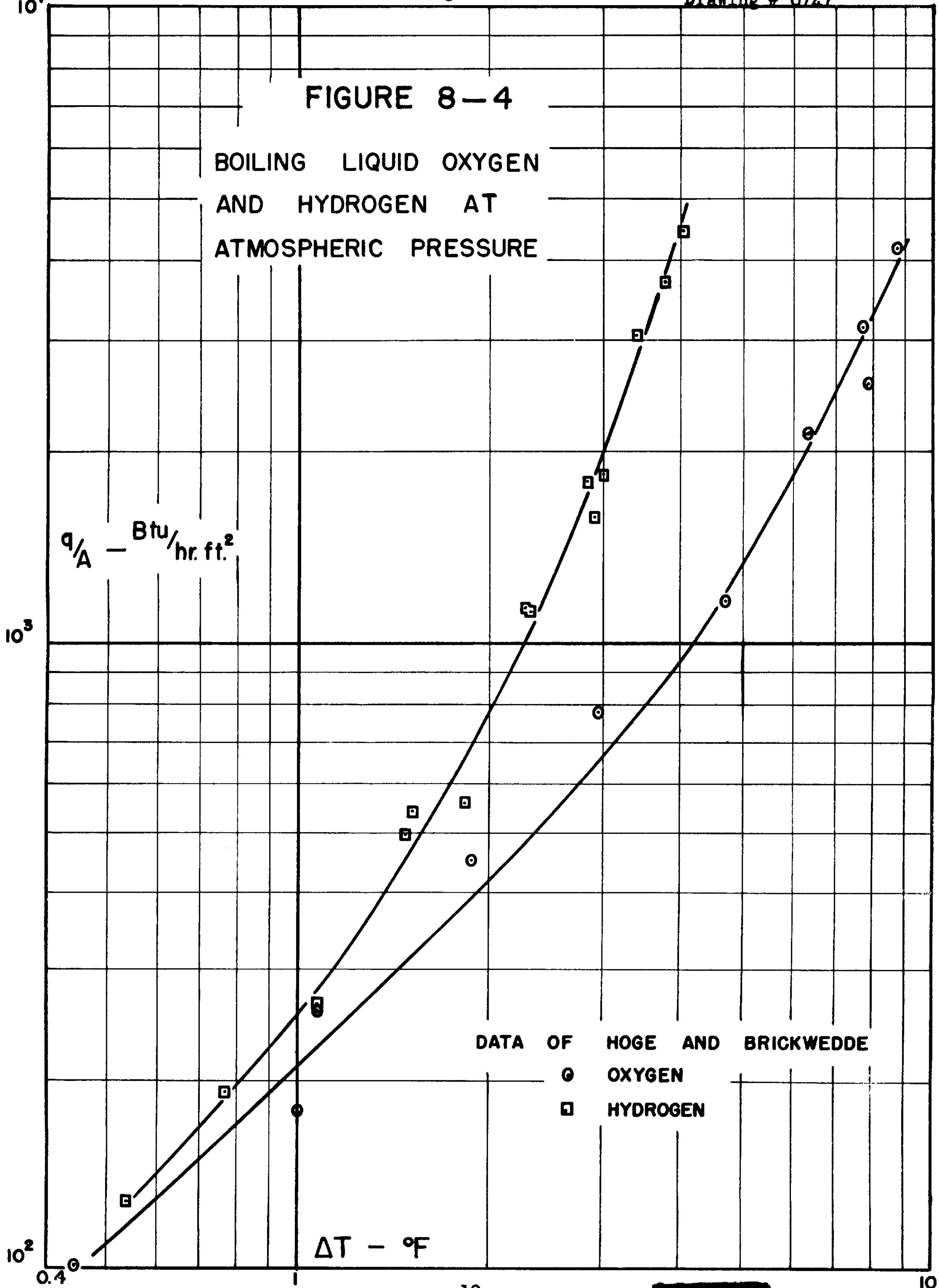


FIGURE 8-4
BOILING LIQUID OXYGEN
AND HYDROGEN AT
ATMOSPHERIC PRESSURE
q/A — Btu/hr. ft.²
ΔT — °F
DATA OF HOGE AND BRICKWEDDE
OXYGEN
HYDROGEN
10⁴
10³
10²
0.4
1
10

And the heat transfer coefficient combining convection and radiation was calculated by the expression -

$$h = h_c + h_r \left[3/4 + 1/4 \frac{h_r}{h_c} \left(\frac{1}{2.62 + \frac{h_r}{h_c}} \right) \right] \qquad \text{(Eq. 8.05)}$$

The value of ε in Equations 8.02 and 8.03 is assumed by Bromley to be of the order of 0.6.

The equations presented above are not recommended for general use, but are included here in order to show the form of correlation Bromley attempted. His approach in applying thermodynamic and physical laws to this case is noteworthy. It may be that ε is a function of system geometry or other variables. The pressure effect has not been included, but possibly it could be introduced in Bromley's type of analysis. From the data of Farber and Scorah[5], it would appear that the nature of the surface is also influential in determing the shape and position of the curve in the film boiling range and must be accounted for in a strict analysis. Bromley's data for boiling water are shown on Figure 8-2, and data for both water and nitrogen are shown on Figure 8-5.

Some additional data on this case of boiling water circulated by natural convection may be found in reports on work done at Massachusetts Institute of Technology under the supervision of McAdams[6,9,21,24,25,28], and it is expected that a summary report on the batch boiler work will be published in the near future.

One very interesting case of boiling water circulated by natural convection is the case in which the bulk temperature of the water is much

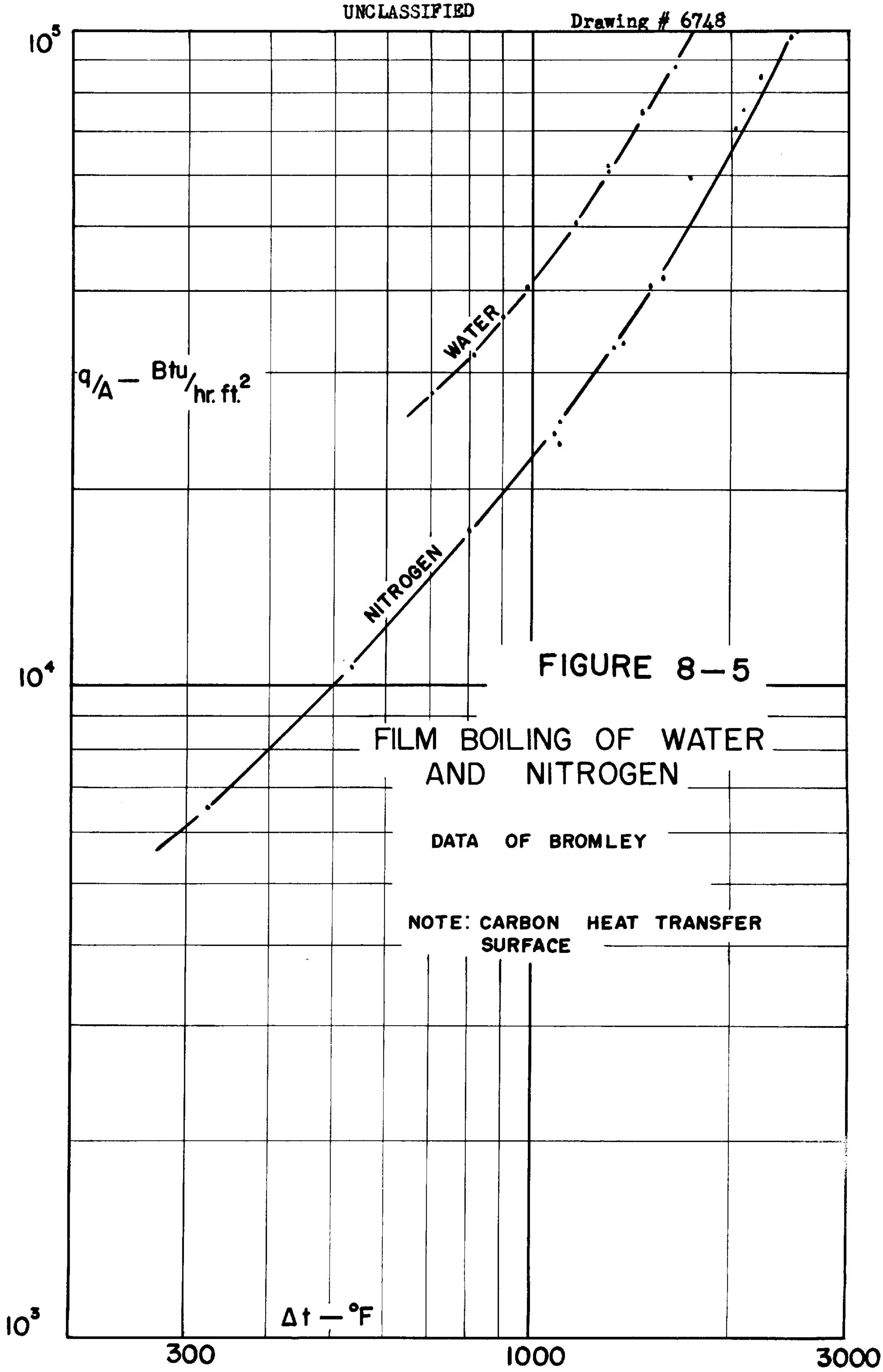
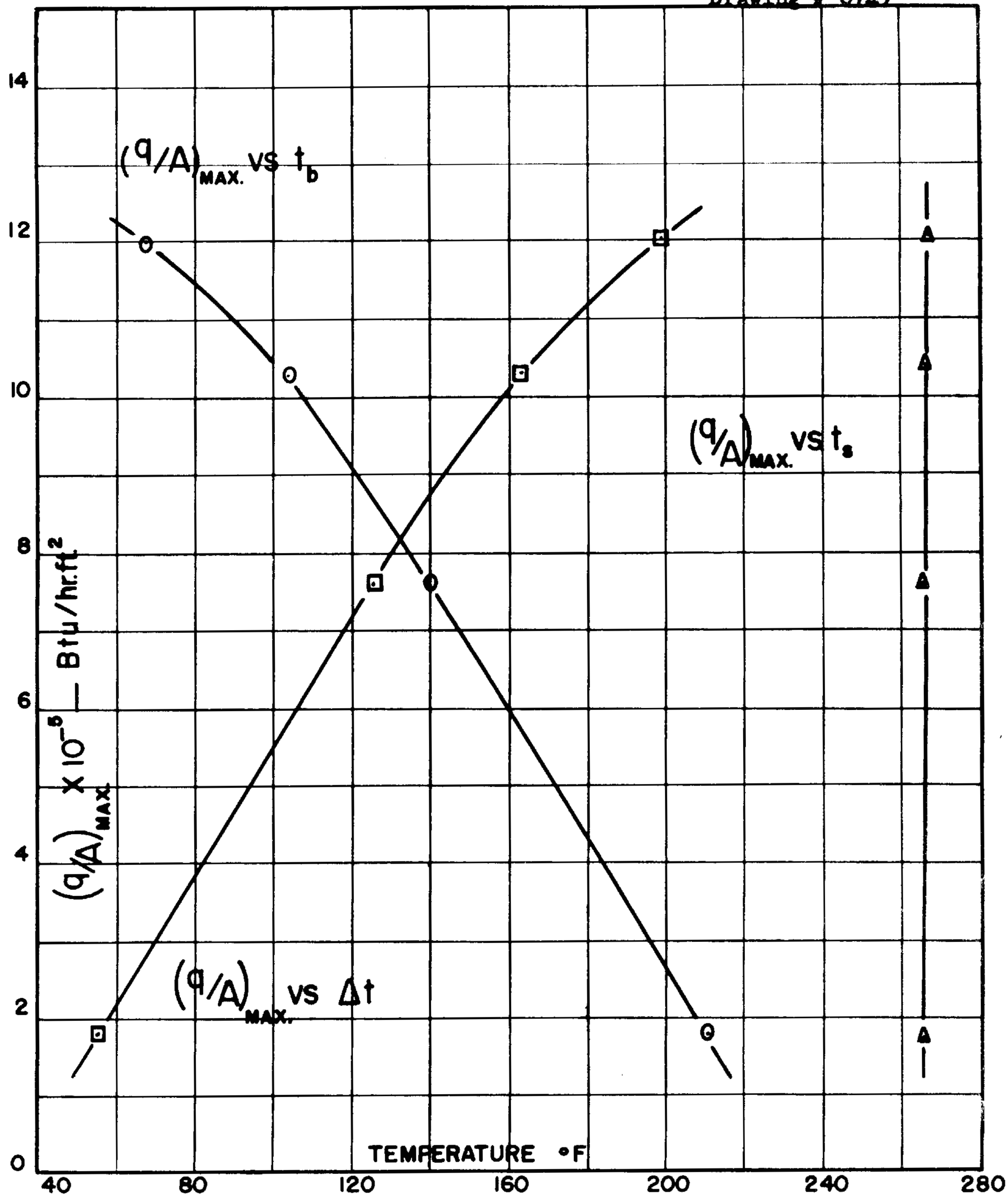


FIGURE 8-5

FILM BOILING OF WATER AND NITROGEN

below the saturation boiling point. This is commonly called local boiling of subcooled water and some data on this case have been presented by Moscicki and Broder[6,26]. They investigated local boiling of subcooled water at one atmosphere, using a platinum wire of approximately 0.004 in. diameter. Figure 8-6 is a plot of their data relating $(q/A)_{max}$ to bulk temperature, surface temperature, and difference between the bulk and surface temperatures. It is interesting to note that the surface temperature remains practically constant for a large change in $(q/A)_{max}$ and varying degrees of subcooling.



t_b = BULK TEMPERATURE

t_s = SURFACE TEMPERATURE

$\Delta t = t_s - t_b$

FIGURE 8-6

LOCAL BOILING OF SUBCOOLED WATER

DATA OF MOSCIKI AND BRODER

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ORNL 156-9

HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 9

HEAT TRANSFER WITH LOCAL BOILING OF SUBCOOLED WATER IN FORCED CONVECTION

W. B. Harrison
October 1, 1948

[redacted]

PART 9

HEAT TRANSFER WITH LOCAL BOILING OF SUBCOOLED WATER IN FORCED CONVECTION

Water flowing normal to cylinders

Apparently the case of local boiling of subcooled water flowing normal to cylinders has not been investigated in any of the project laboratories, but several notes on this case are included in view of the high heat fluxes which can be obtained with this mechanism. The chief sources of data are reports of work done at Massachusetts Institute of Technology[1,2,3]. A few figures will serve to illustrate the effects of amount of subcooling, and the velocity.

Figure 9-1 shows data presented by McAdams[3] for the case of water at 65° F. flowing normal to a 0.048 in. stainless steel tube at three different velocities. Figure 9-2 shows data for three different bulk temperatures, but the same velocity normal to a 0.048 in. stainless steel tube.

The beginning of the local boiling regime is the temperature at which the tube surface exceeds the saturation boiling point, although superheat is usually required before ebullition starts.

FIGURE 9—1
LOCAL BOILING OF SUBCOOLED WATER
— FLOW NORMAL TO 0.048 INCH TUBE —
— VELOCITY EFFECT —

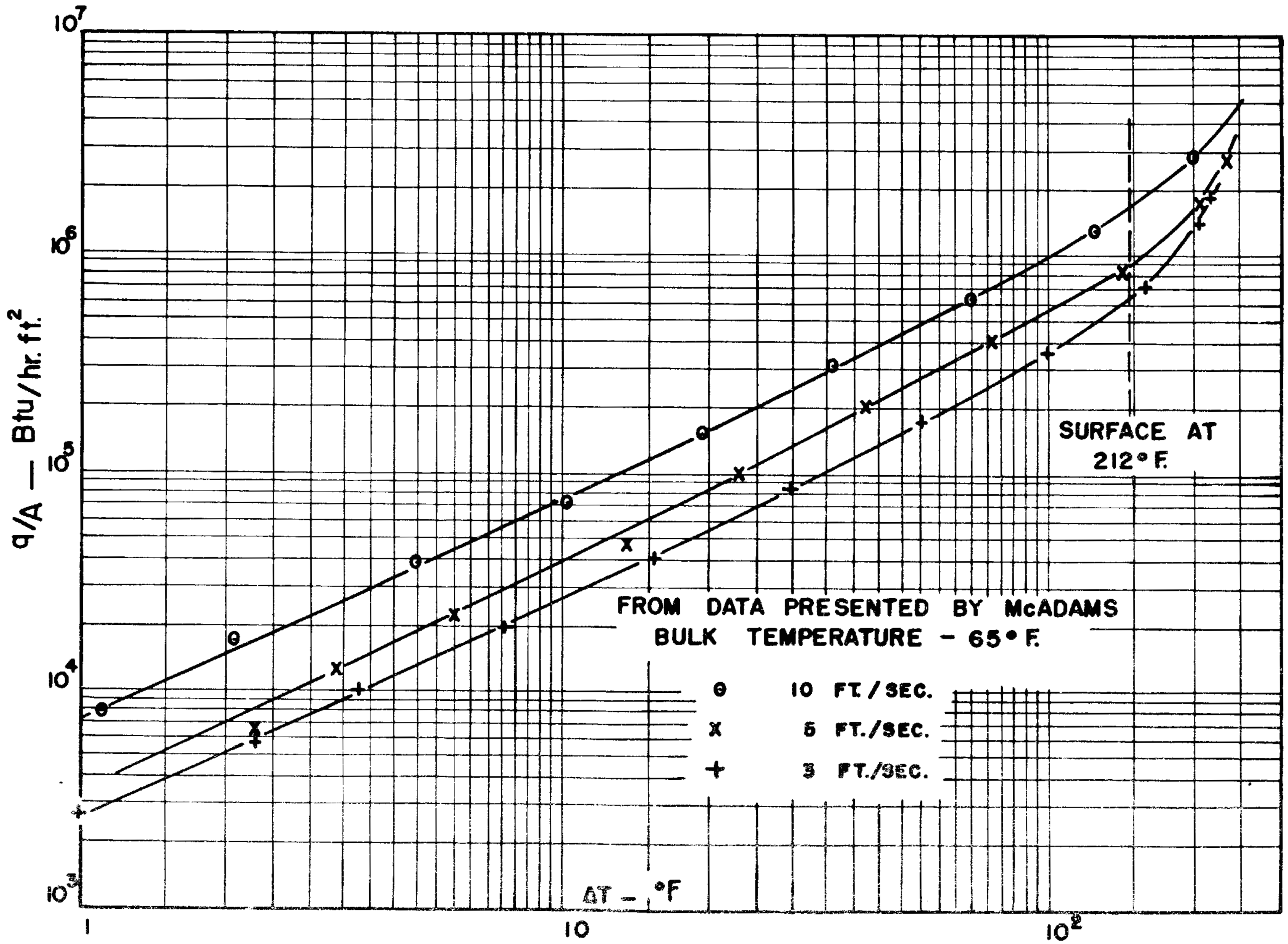

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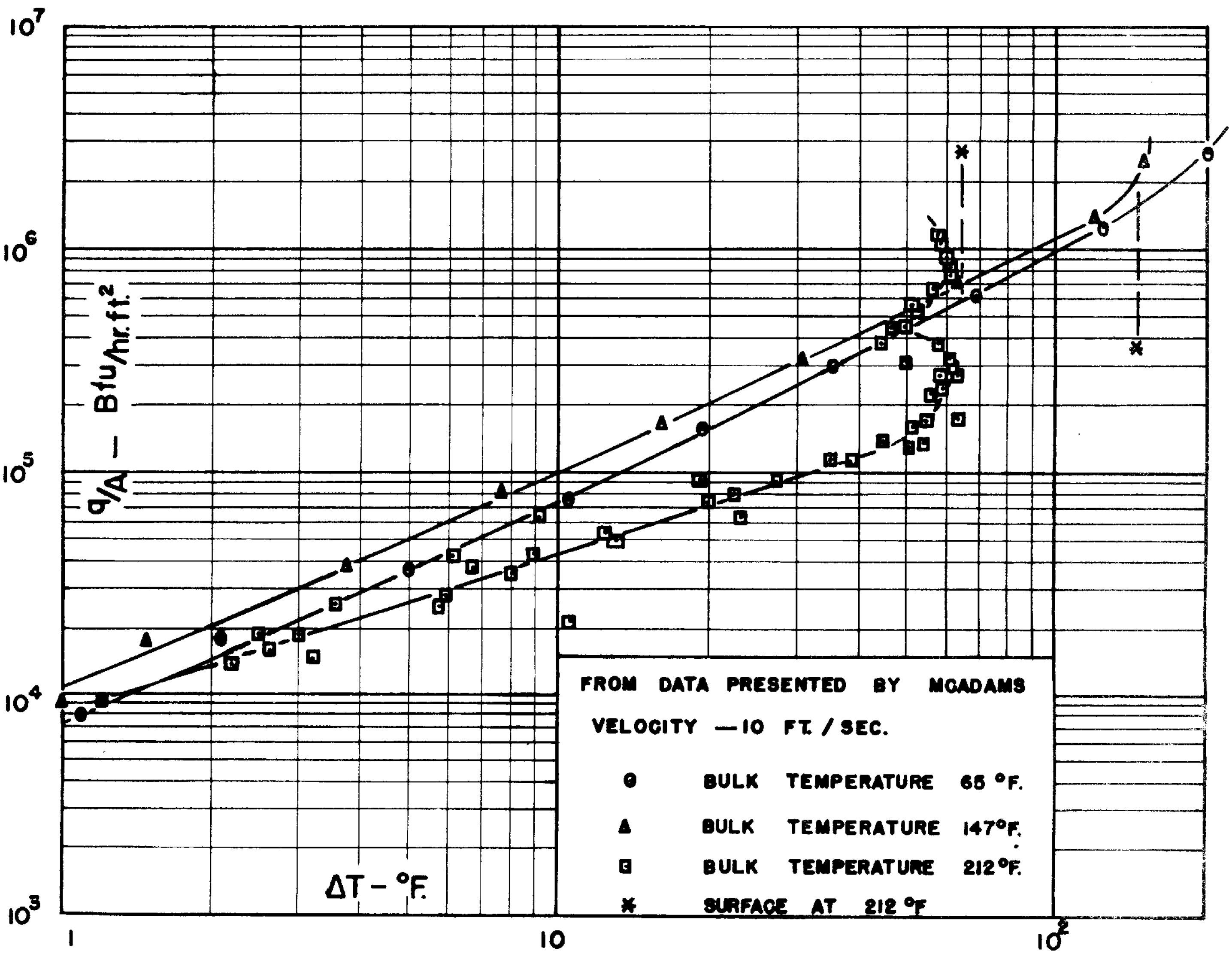
FIGURE 9-2

LOCAL BOILING OF SUBCOOLED WATER
- FLOW NORMAL TO 0.048 INCH TUBE -
- EFFECT OF SUBCOOLING -



It appears that fluxes as high as 1.6×10^6 Btu/hr. ft.^{2} can be attained without boiling for the case of water at 65° F. flowing 10 ft./sec., and fluxes greater than 2.8×10^6 Btu/hr. ft.^{2} can be attained with local boiling for the same case. Referring to Figure 9-1, one may see that increasing the velocity increased the obtainable flux for a given ΔT. It also seems likely that increasing the velocity would increase $(q/A)_{max}$ in the nucleate boiling region.

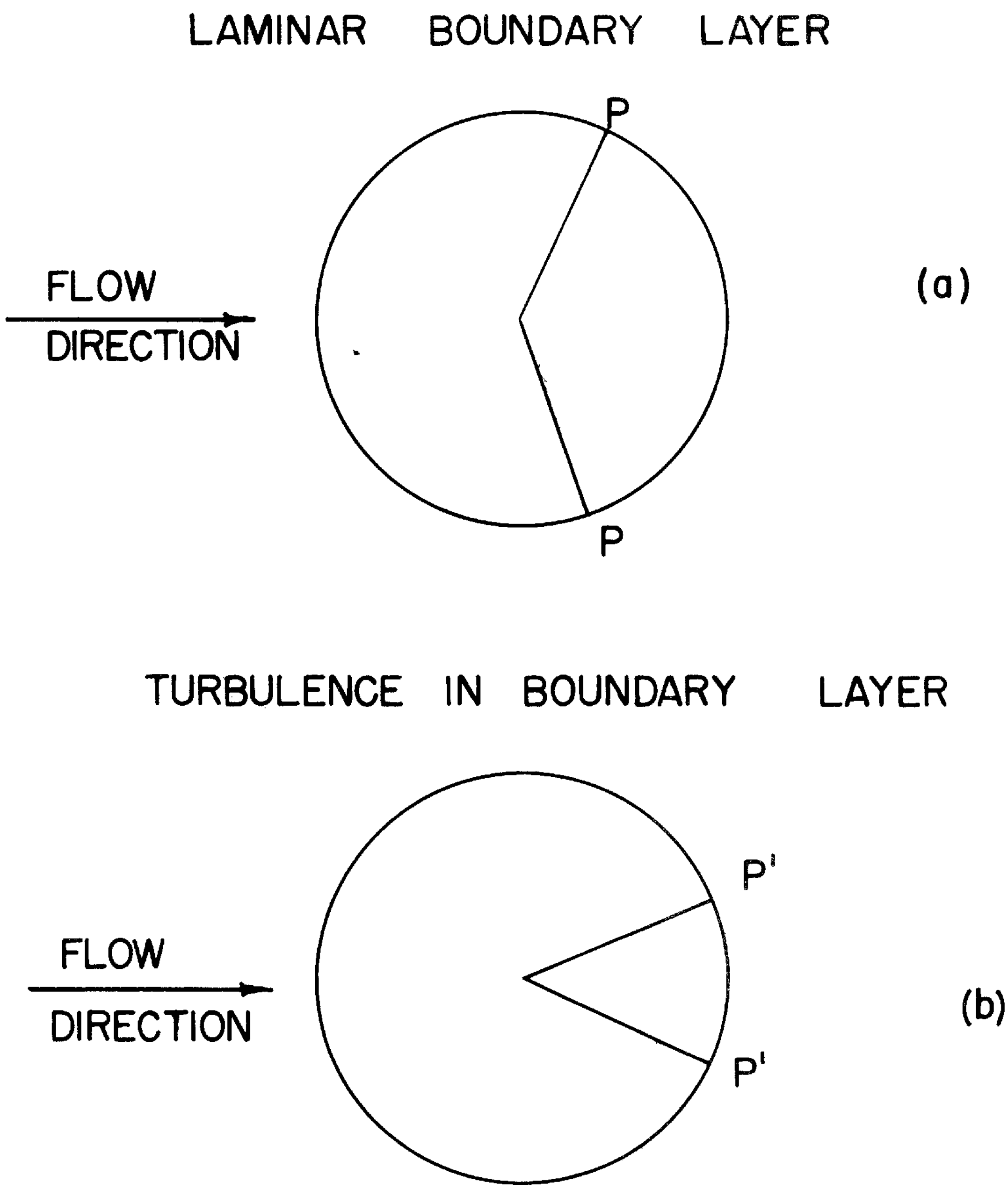
Another variable which has considerable influence is the amount of subcooling, or difference between bulk temperature and saturation boiling point. From Figure 9-2, it is seen that higher fluxes were obtained with water at 147° F. than with water at 65° F., for the same value of ΔT. Probably this difference is due to the change of physical properties with temperature rather than the effect of subcooling. It is believed that lowering the bulk temperature will increase the $(q/A)_{max}$ in the nucleate boiling range. These beliefs cannot be substantiated by the data referred to in this section of the report, but the effects of these variables will be discussed in more detail in the following sections on local boiling of water in forced convection parallel to the heating surface, particularly in an annulus.

For the case of water at 212° F. flowing normal to the heating tube at 10 ft./sec., the data shown on Figure 9-2 are rather bewildering at first glance. McAdams states that the peculiar shape of the curve in the neighborhood of $\Delta T = 55°$ F. is clearly indicated by the data. The case is complicated by the fact that both nucleate and film boiling may be taking place simultaneously but at different locations around the tube.

The following explanation of the peculiar shape of the curve was instigated by N. F. Lansing*, the terminology being consistent with common usage in fluid mechanics.[4] Consider the case of flow normal to a cylinder and assume a laminar boundary layer with separation occurring at the location shown as P on sketch (a), Figure 9-3. As the heat flux increases, bubble formation creates localized turbulence until the point is reached that the turbulent influence spreads in the boundary layer. This is accompanied by a shift in the separation point, increasing the effective heat transfer area. Since the downstream surface is partly blanketed by an increased thermal resistance, and since stainless steel has a rather low thermal conductivity, one may expect the downstream surface to have a higher temperature than the upstream surface. The effect of shifting the separation point from front to rear (at constant flux) would be to lower the measured "average" temperature of the tube.

*Oak Ridge National Laboratory, Oak Ridge, Tennessee

Drawing # 6737



P, P' - SEPARATION POINTS

FIGURE 9—3

SUGGESTED BOUNDARY LAYER TRANSITION

The shifting of the separation point appears to be accompanied by decreasing thermal resistance and increasing flux. Thus, at $q/A \cong 150{,}000$ Btu/hr. ft.^{2} and $\Delta T \cong 50^\circ$ F., the separation point may begin to shift in the direction of P' (sketch b, Figure 9-3). Effective heat transfer area is increased, to give higher flux for the same ΔT; but, at fluxes above 300,000 Btu/hr. ft.^{2}, the average tube temperature begins to drop faster than the flux increases, giving the negative value for $\frac{d(q/A)}{d(\Delta T)}$. Then when the separation point has become essentially stabilized at P', the derivative can again assume a positive value.

This explanation appears to be plausible, but it should be kept in mind that local disturbances caused by boiling at the liquid-solid interface may result in a mechanism which is much different from the usual case without phase change. The ideas outlined above are in accord with considerations presented in Goldstein[4], with the modification that critical Reynolds number for this case is reduced from the order of 10^5 to the order of 10^4. This modification also seems plausible, for critical Reynolds number has been shown to decrease as main stream turbulence increases, and surface irregularities have a very great effect in the same direction. These effects are mentioned by Goldstein, but no data have been located which clarify the flow picture in the case of local boiling. Whether or not the explanation suggested is strictly applicable could possibly be determined by experiment. It may be that form-drag data would be sufficient to show whether or not the explanation is valid, for drag coefficients drop considerably during the change from laminar to turbulent boundary layers accompanied by movement of the separation point to the rear.

Flow within tubes and rectangular channels

Data for local boiling of subcooled water are also meager for case of flow in circular tubes and rectangular channels. Taylor[5] used an electrically heated, 3/8 in. O.D. x 0.30 ± in. I.D. stainless steel tube to heat water which entered the tube at 70° F. with a velocity of 39.5 ft./sec. He measured the outside tube wall temperature at each end and in the middle of the tube, and the water temperature was measured at each end of the tube. Pressure drop through the tube was of the order of 40 lb./in.^{2} The temperature drop through the tube wall could not be estimated very accurately. As shown by Kreith and Summerfield[6], the calculation is rather complex for large temperature gradients and Taylor pointed out that small changes in chromium content make a comparatively large change in the thermal conductivity of stainless steel.

It is believed that the trends indicated are important, and the data of Taylor are shown on Figure 9-4. The values shown cannot be taken for absolute values but they do show orders of magnitude. Actual values of ΔT were smaller than indicated on the figure since the inside surface temperature is assumed equal to the outside wall temperature. For this figure, it also is assumed that total heat dissipated was absorbed by the stream. It is particularly interesting to note that the flux attained was of the order of 3×10^6 Btu/hr. ft.^{2} and, apparently, higher fluxes were possible. The bulk temperature of the water rose to about 105° F. at the outlet end of the tube for the highest fluxes, but it was still well below the saturation boiling point. These data indicate that, with local boiling of subcooled

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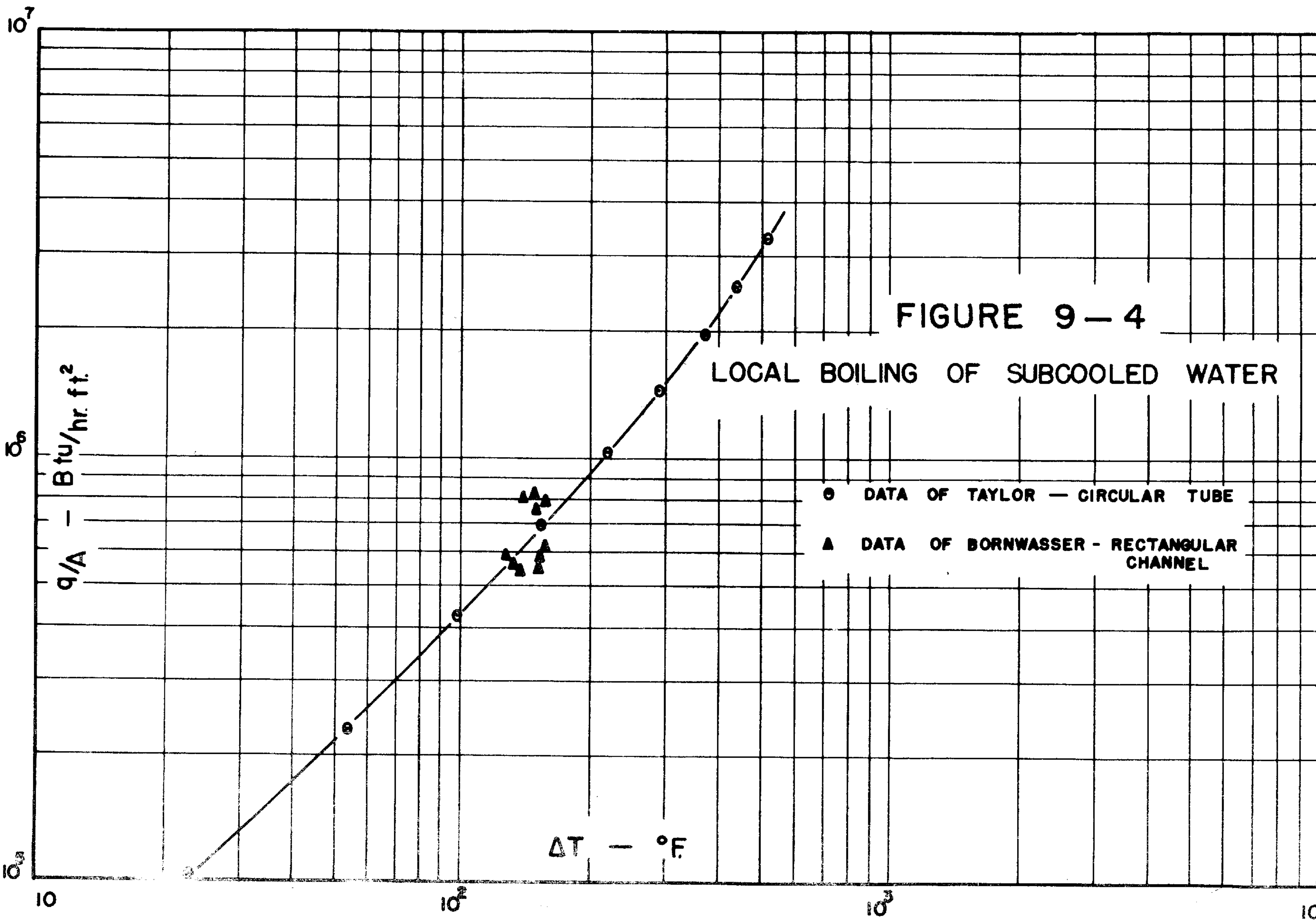

FIGURE 9 — 4

LOCAL BOILING OF SUBCOOLED WATER

Drawing # 6738

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water circulated by forced convection, fluxes can be attained which are roughly eight times greater than those possible in the case of boiling water circulated at its boiling point by natural convection. The limit of flux was not reached in Taylor's system because of a burn-out failure which resulted from an unanticipated fall in water pressure. This indicates that peak flux is a function of pressure and water velocity, and the effects of these variables are discussed in more detail in the section on studies **in an annulus.**

One set of data on local boiling of subcooled water in a rectangular channel was reported by Bornwasser. The channel was about 0.16 in. deep and 1 in. wide. The heat transfer surface was a 1 in. square copper bar heated with a torch on one end. The end of the bar was mounted flush with one of the parallel walls so as to prevent distortion of the flow and give direct contact with the demineralized cooling water which was introduced at about 104° F. and velocities from 8 to 20 ft./sec. The data of Bornwasser included surface temperature measured with thermocouples imbedded in the bar, flow rate, and heat flux as calculated from the temperature rise of the water. In such a system, one might expect some irregularities because of asymmetry of the heat transfer perimeter, temperature discontinuity in the channel, and low length/hydraulic diameter ratio. Consequently, the data are not necessarily quantitative but are shown on Figure 9-4 for comparison.

Flow in an Annulus

The case of local boiling of subcooled water in an annulus is also attractive because of the high heat fluxes attainable. It has been studied more carefully than the forced convection studies reported in previous sections, and it is believed that the trends exhibited in the annulus will also hold for circular tubes and rectangular channels. Knowles[8] made a rather comprehensive study of local boiling in an annulus which he reported early in 1946. In his apparatus, water flowed downward in the annulus made by an electrically heated stainless steel tube within a vertical glass tube. For the determinations made at heat fluxes up to about 9.5×10^5 Btu/hr. ft.^{2} , a 1/4 in. diameter stainless steel tube was used. Three different lengths (30 in., 12 in., and 3 in.) of tubing were used in combination with two inside diameters (0.642 in., 0.394 in.) of glass tube in order to vary the ratio of heated length to hydraulic diameter. (L/D_e ranged from 7.6 to 76).

For high flux studies up to about 2.35×10^6 Btu/hr. ft.^{2} , a 1/8 in. diameter stainless steel tube was used in combination with a 0.642 in. inside diameter glass tube. The heated length was 11.6 in. and the L/D_e ratio was 22.3.

The range of conditions studied is shown in Table 9-I. Pressure at the exit, or downstream, end of the heated length was maintained at 40.6 psia. throughout the investigation.

Figure 9-5 is typical of the data reported by Knowles, and will serve to illustrate the conclusions which may be drawn from his work. A few notes are included below to describe the experimental method and explain the figure.

For a specific hydraulic diameter, rate of flow, and heated length, a certain heat flux, say $(q/A)_1$, was chosen. The bulk temperature of the exit end of the heated length was "A", and the corresponding surface temperature was plotted at "D". As the bulk exit temperature was increased to "B" (by increasing the input water temperature), the surface temperature increased to "E". As the bulk temperature was increased further, the surface temperature remained constant at "E". When the bulk temperature reached "C", the system broke down with a great evolution of steam and a burned-out heater tube.

Then all points lying on the same curve with "E" represent the surface temperature at break down conditions and each point is the critical surface temperature for a specific flux. The region of unstable surface temperatures, leading to breakdown, is to the right of this curve. The curve through "C" confining the stable region of maximum bulk temperatures is called the critical bulk temperature curve. In one case the critical bulk temperature was 104° F. below the saturation boiling point at a flux of 7.0×10^5 Btu/hr. ft.^{2} (Series 3 in Knowles' report). The difference between critical surface temperature and critical bulk temperature ("E" - "C") may be called the critical temperature difference for a given flux. Critical temperature difference increased as flux increased and reached the order of magnitude of 210° F. at fluxes ranging from 4.0 to 9.5×10^5 Btu/hr. ft.^{2}, depending on experimental conditions. The critical surface temperature rises more

Drawing # 6799

FIGURE 9—5

TYPICAL CURVES FOR LOCAL BOILING OF SUBCOOLED WATER IN AN ANNULUS FROM DATA OF KNOWLES

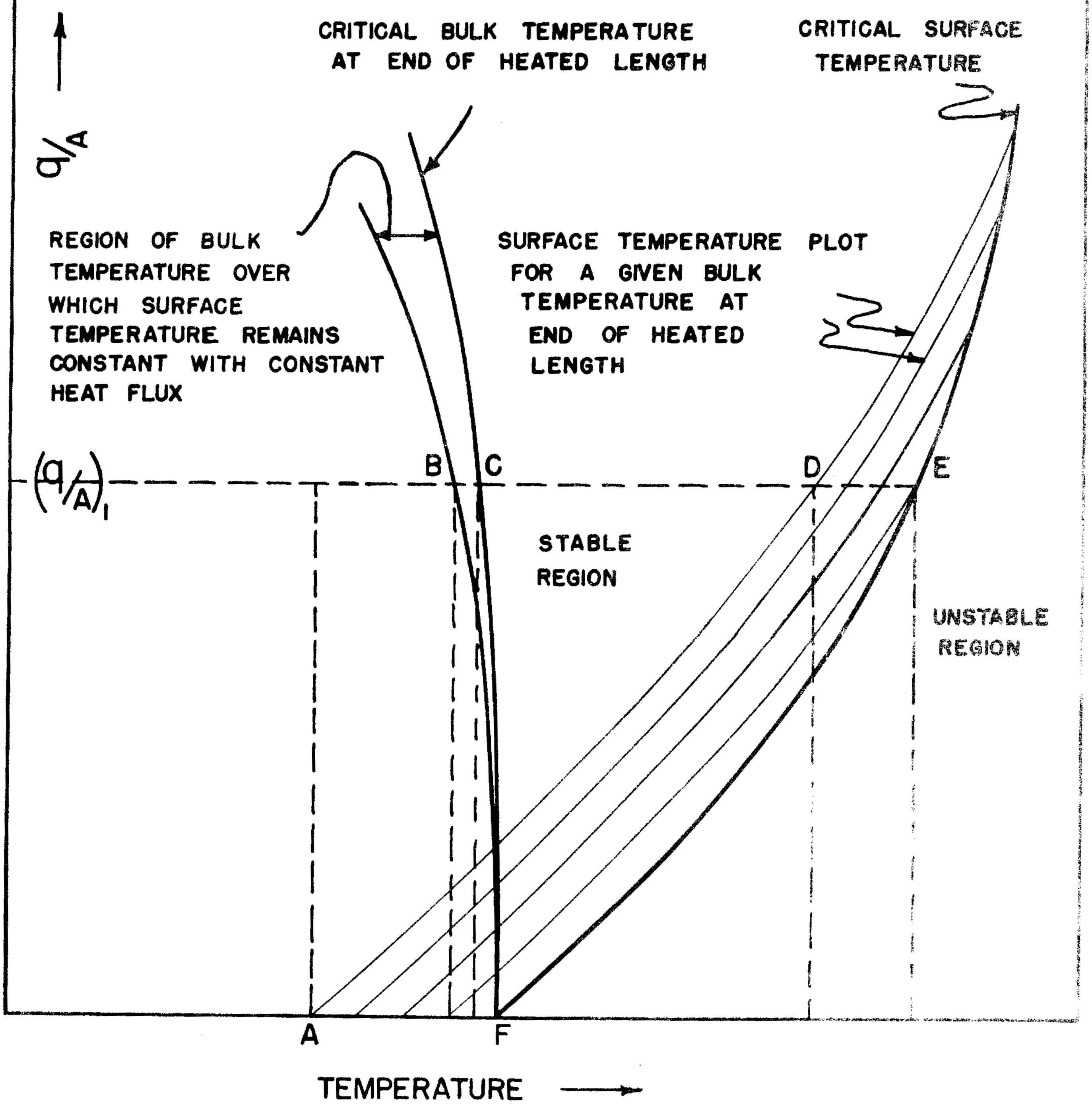

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TABLE 9-I

RANGE OF EXPERIMENTAL CONDITIONS STUDIED
BY KNOWLES

Series No.		2	3	4	5	6	7	8	9	10	11
L' - Length of heater tube - in.		12	12	12	12	30	12	3	3	11.6	11.6
D_e' - Hydraulic diameter - in.		0.185	0.185	0.394	0.394	0.394	0.394	0.394	0.185	0.520	0.520
G Mass flow rate x 10^{-5} lb./hr. ft.^{2}		21.0	7.96	22.3	15.5	7.30	7.30	7.30	5.68	6.35	20.6
V' Velocity (approx.) ft./sec.		9.5	3.6	10.0	6.9	3.3	3.3	3.3	2.5	2.8	9.2
L/D_e		64.3	64.3	30.5	30.5	76.2	30.5	7.6	16.1	22.3	22.3
*Range of q/A x 10^{-5}	high	1.19	0.914	0.869	0.869	2.18	0.847	6.63	7.42	3.24	3.33
Btu/hr.ft.^{2}	low	8.54	6.25	9.07	9.04	3.84	9.04	13.7	7.42	16.4	23.5
*Range of Re x 10^{-4}	high	4.29	0.88	4.30	3.82	4.60	2.08	4.50	2.03	4.24	11.7
	low	7.95	2.21	16.5	5.84	5.95	5.94	5.55	2.12	5.44	20.0
*Range of h x 10^{-3}	high	15.0	5.68	12.2	10.2	4.64	6.27	18.1	18.7	8.41	13.2
Btu/hr. ft.^{2} °F	low	23.2	16.7	19.5	20.9	12.5	28.6	29.4	21.7	23.0	27.2
*Range of Nu$(Re^{-0.3})(Pr^{-0.4})$	high	1.83	1.60	1.54	1.96	1.54	2.32	5.28	6.04	3.60	2.33
x 10^2	low	3.08	4.52	3.16	3.52	3.83	8.84	8.13	6.88	8.90	3.92

*Not corresponding values.

Note: Pressure at exit end of heated length = 40.6 psia [redacted] and more sharply as heat flux increases, and seems to approach an asymptotic value in some cases. On the other hand, critical bulk temperatures rapidly curve over to lower values as flux increases. This means that at high fluxes, bulk exit temperatures must be kept low to avoid breakdown. This, in turn, reduces the temperature rise permitted in the water, and must be compensated for by increasing the flow rate.

Similarly, all points on the curve passing through "D" represent surface temperatures for different fluxes with the bulk exit temperature constant. This curve is essentially a plot of ΔT (surface temperature minus bulk temperature) vs. flux up to fluxes at which the surface is approaching the critical surface temperature. Upward curvature of the surface temperature lines indicates increasing coefficients, since bulk temperature is constant for each curve. All points lying on the curve "B" show the bulk temperature at which the surface temperature reaches the critical value, for a given flux. The difference between the curve through "B" and the critical bulk temperature curve "C" is the amount that the bulk temperature can rise without changing the surface temperature (already critical). The difference between "B" and "C" may reach the order of magnitude of 36° F. (Series 6, Graph 6 in Knowles' report). It is also noted that, with constant flux, the coefficient increases as the bulk temperature increases from "B" to "C".

Since bulk temperature and surface temperature are equal at zero flux, the prevailing bulk temperature for any surface temperature curve (in the plot of flux vs. surface temperature) is the temperature at which the surface temperature curve intersects zero flux.

Conditions were changed by varying hydraulic diameter, heated length, and flow rate independently and several plots similar to Figure 9-5 were made for these different conditions. (See Table 9-I)

Included in the report by Knowles are short discussions on breakdown conditions, visible boiling on the tube, the mechanism of boiling, pressure drop, dirt film formation, and accuracy of results. There are five graphs similar to Figure 9-5, in which experimental data have been plotted for different experimental conditions. In four more graphs, **critical surface** and bulk temperatures for various investigations were superimposed in order to see how different experimental conditions changed the positions of the critical curves. Log (Nu) $(Pr^{-0.4})$ $(Re^{-0.8})$ was plotted vs. **flux** for most of the experimental range. The expression log (Nu) $(Pr^{-0.4})$ $(Re^{-0.8})$ was also plotted vs. L/D_e for the low flow rate determinations. It is believed that a plot which utilizes the Dittus-Boelter type relationship is of little value in an effort to apply the correlation to cases other than those studied. The Dittus-Boelter type relationship cannot be expected to hold, since it implies an exchange of sensible heat only. It may be that variables other than those investigated enter into the considerations as indicated in previous sections of this report and verified in the following section. Definite effects of L/D_e, flow rate, and flux have not been isolated sufficiently to justify any statements of quantitative value.

It would be desirable to make a heat balance showing what percentage of the heat dissipated into the water was used in increasing its bulk temperature in sensible heat, and what portion of the heat is present as latent heat in water vapor. Conceivably, one limiting breakdown consideration could be volume percent of steam or percent heat in latent form.

The data in the report do not permit this type of calculation, however.

Considerable error probably exists in selecting the temperature at which to evaluate the physical properties which entered in the calculation of (Nu) $(Pr^{-0.4})$ $(Re^{-0.8})$, since it was assumed that these properties were to be evaluated at a temperature halfway between the bulk and surface temperatures. This method of selecting temperature has shown promise in correlating data for sensible heat exchange in forced convection systems, but local boiling may require an entirely different basis. Another likely source for error in systems employing electrically heated tubes is the calculation for temperature drop through the tube wall. As noted in previous discussions of boiling data, it is believed that the trends indicated in this work are more valid than any quantitative data reported, although Knowles suggests that heat transfer coefficients reported are accurate within 2%.

Other contributions to the literature on this case have been made recently by Minden[9,10,11], and Carl and Picornell[12] working under the supervision of McAdams at Massachusetts Institute of Technology. One test unit used by Minden[9] consisted of a 1/4 in. O. D., electrically heated, stainless steel tube surrounded by a concentric, glass-lined stainless steel tube to form an annulus of $D_e = 0.42$ inches. The jacket was 14 in. long and it was used in combination with two heated lengths. The heated length of the heat transfer tube was 4 in. for heat fluxes near 2×10^6 Btu/hr. ft.^{2} and an 8 in. tube was used for fluxes less than 1.1×10^6 Btu/hr. ft.^{2} The heater tube was supported in the center of the annulus by a copper tube on one end and a brass rod on the other, both of which served as electrical leads. The principal cases he explored with this apparatus were:

1. Effect of velocity

   a. Velocities of 2, 4, 8, and 16 ft./sec. were used in the equipment, with pressure held constant at 30 psia and bulk temperature at 150° F.

   b. Velocities of 16 and 32 ft./sec. were explored, with pressure at 60 psia and bulk temperature at 193° F.

2. Effect of pressure

   Holding the velocity at 16 ft./sec. and the bulk temperature at 100° F., experimental runs were made for pressures of 30, 60, 90, and 120 psia.

Carl and Picornell[12] held the degree of subcooling (difference between bulk temperature and saturation boiling temperature) constant rather than bulk temperature, and the principal cases they investigated are indicated below:

Pressure, Psia	Subcooling, °F	Velocity, ft./sec.
30	50	1 4 12
60	20	1 4 12
60	50	1 4
60	100	1 4 12 36.6
60	150	4
90	50	1 4

The range of experimental conditions permitted an evaluation of the effect of each of the variables noted in the table above- pressure, subcooling and velocity. Carl and Picornell used a vertical annulus made by a centrally located, 0.254 in. diameter stainless steel heating tube surrounded by either a 0.423 in. or 0.770 in. I. D. glass tube, 18 in. long. The actual heated length (length of stainless steel tube) was 4 inches. The apparatus was essentially the same as used by Minden.

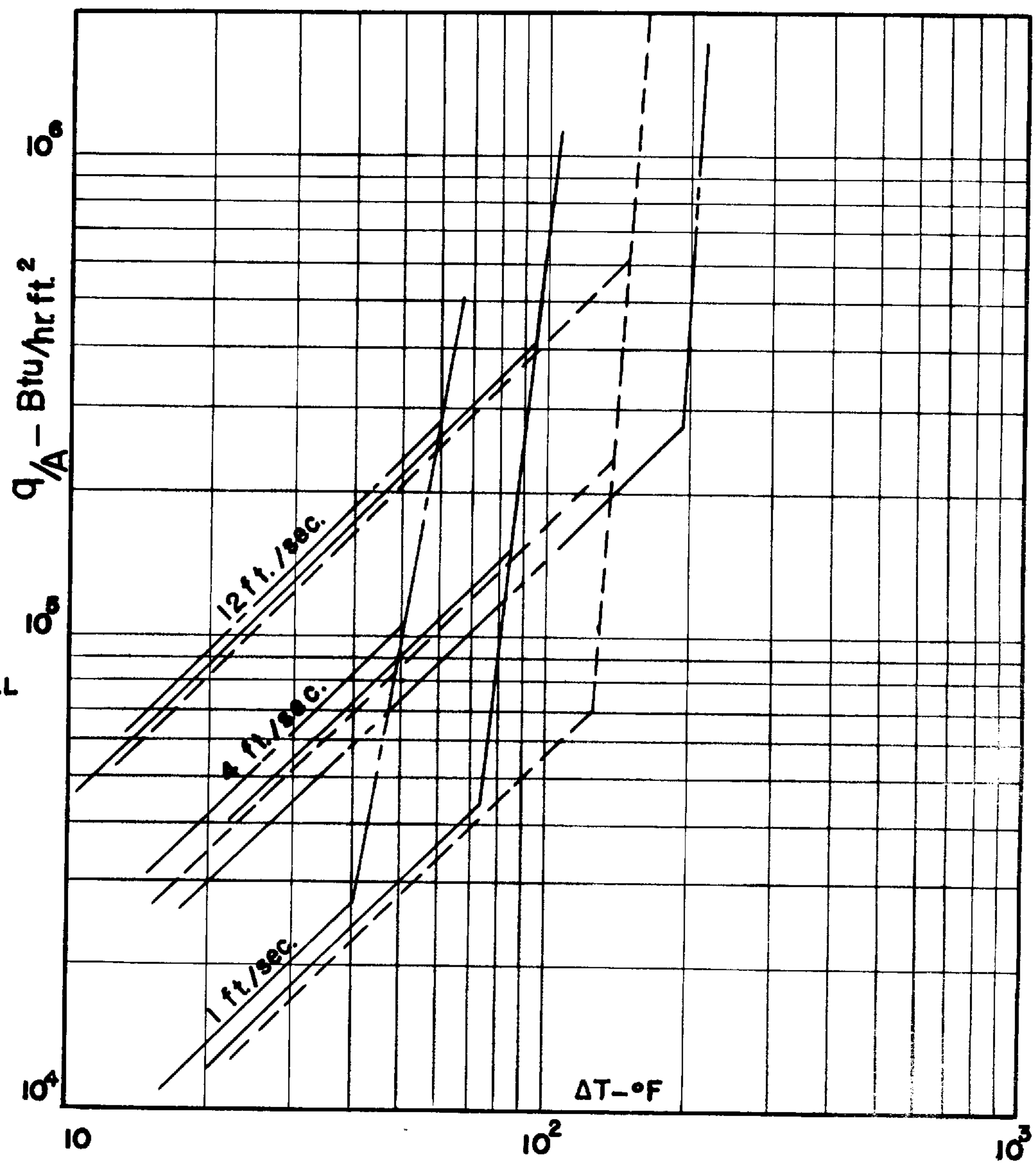
Except for its effect on maximum attainable flux*, pressure change did not have an appreciable effect on the heat transfer to water in the range of conditions investigated. The effects of subcooling and velocity are summarized on Figure 9-6 as copied from the work by Carl and Picornell. The straight line plots represented the data quite well, and the break in the line coincided with the visual observation of the beginning of ebullition. The investigators pointed out that the curves of steepest slope could all be approximated by one curve having as abscissa the difference between surface temperature and saturation boiling point as shown on Figure 9-7. The effects of pressure, velocity, and subcooling on maximum flux are shown on Figure 9-8, and the effects of velocity and subcooling on critical temperature difference are shown on Figure 9-9.

*<u>NOTE</u>: In electrically heated systems, maximum flux is the highest flux which can be attained without thermal instability and burn-out failure.

Drawing # 6740

FIGURE 9—6

SUMMARY CURVE FOR LOCAL
BOILING OF SUBCOOLED WATER IN AN ANNULUS
—— EFFECT OF VELOCITY AND SUBCOOLING ——

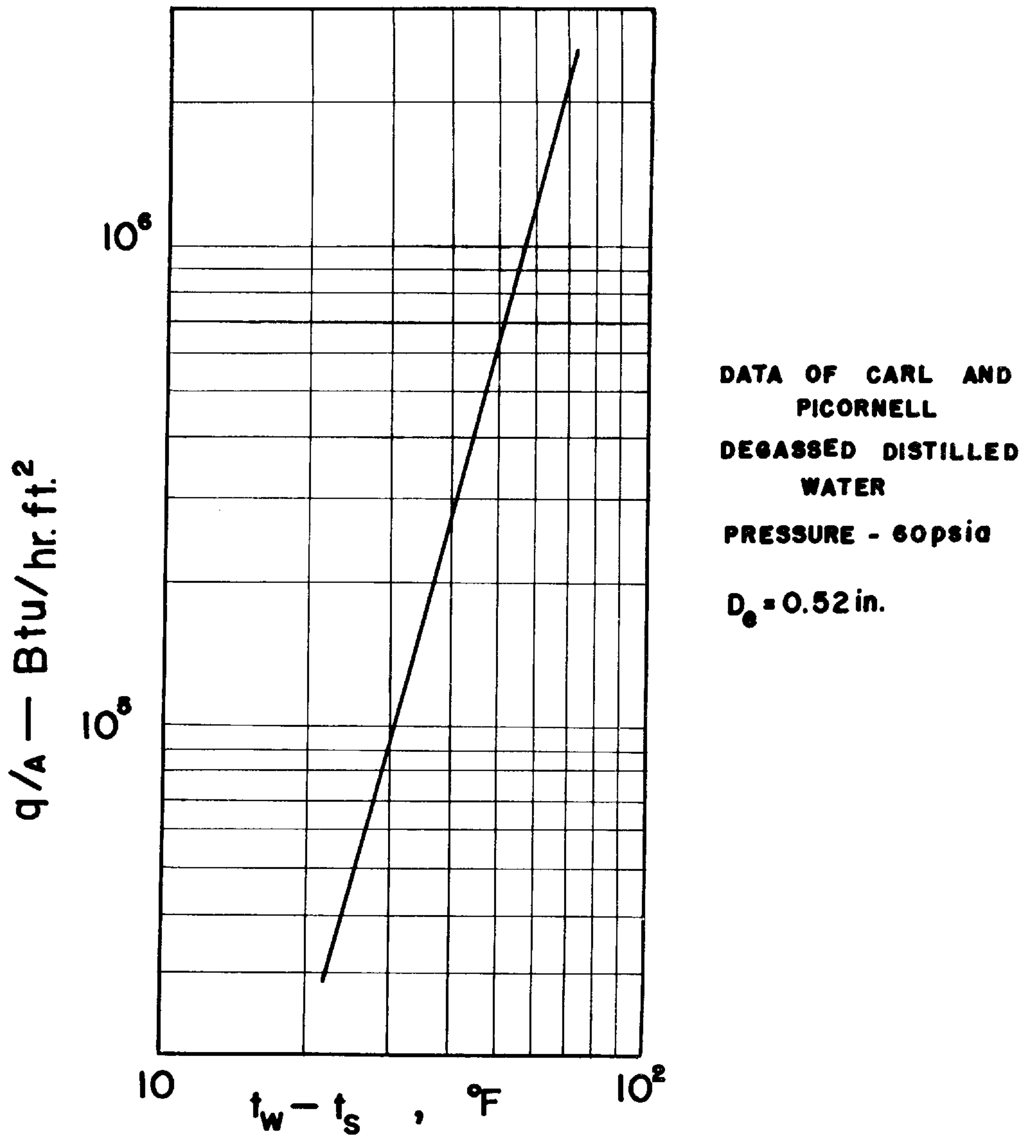


LINE	SUBCOOLING
—-—	20°F.
———	50°F.
— — —	100°F.
—··—	150°F.

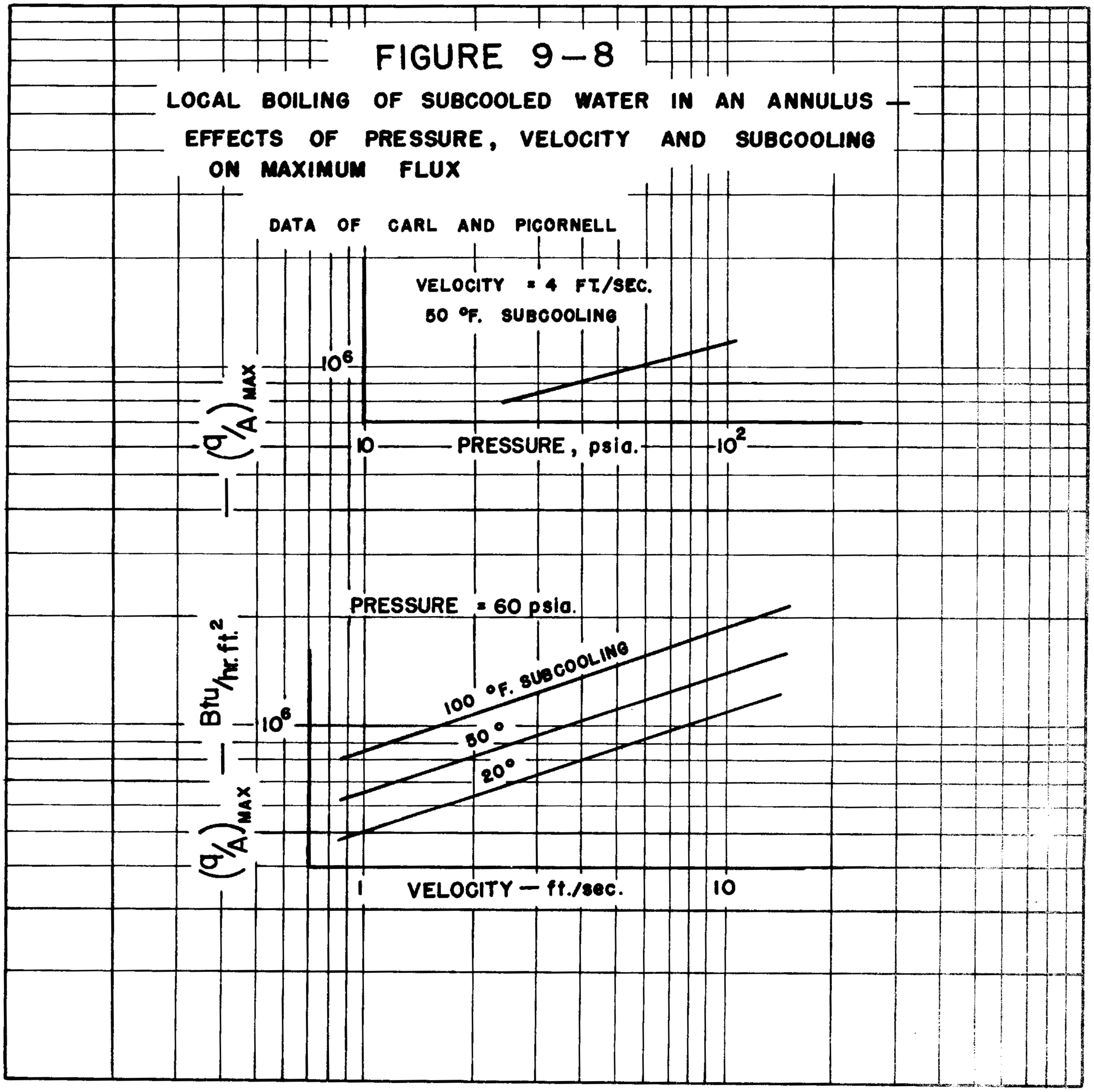
DATA OF CARL AND PICORNELL
PRESSURE = 60 psia
DEGASSED DISTILLED WATER
D_e = 0.52 in.

Drawing # 6741

FIGURE 9—7
APPROXIMATE CURVE FOR LOCAL BOILING OF SUBCOOLED WATER IN AN ANNULUS



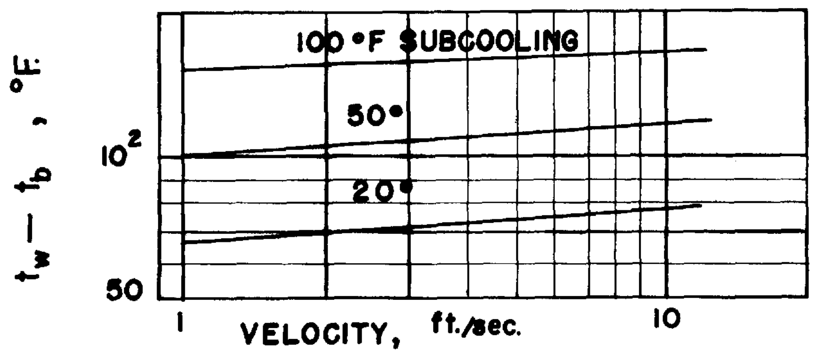
Drawing # 6742

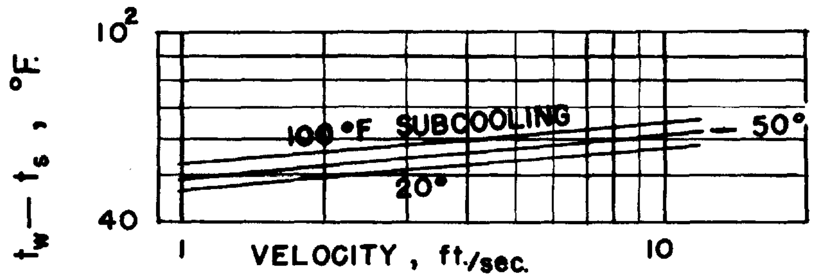


Drawing # 6743

FIGURE 9—9

LOCAL BOILING OF SUBCOOLED WATER IN AN ANNULUS — EFFECTS OF VELOCITY AND SUBCOOLING ON CRITICAL TEMPERATURE DIFFERENCE





DATA OF CARL AND PICORNELL

t_w — WALL TEMPERATURE

t_s — SATURATION BOILING TEMPERATURE

t_b — BULK TEMPERATURE OF WATER

PRESSURE = 60 psia

Most of the data were taken with the 0.770 in. I. D. glass tube but some were taken with the 0.423 in. I. D. tube to determine the effect of annulus size. The effect was not conclusively determined but the data indicated a shift toward lower values of ΔT for the same flux in the smaller annulus. An interesting observation of Carl and Picornell was that there appeared to be a minimum amount by which wall temperature (t_W) exceeded saturation temperature (t_s) before local boiling began. The amount of this temperature difference was influenced by velocity and subcooling as shown by the table below:

TABLE 9-II

Minimum Value of t_W-t_s Required to Initiate Boiling for Various Combinations of Velocity and Subcooling

		Subcooling, °F			
		20	50	100	150
	1	21°F	23 °F	25°F	
Velocity	4	30	34	37	39 °F
ft./sec.	12	39	42	49	

t_W-t_s

The data of Carl and Picornell were reproducible and it is believed that confidence may be placed in the order of magnitude of the values reported.

In two recent memoranda,[13,14] Lansing presented several comments on boiling heat transfer research under way at the University of California at Los Angeles under the direction of Boelter and at Massachusetts Institute of Technology under the direction of McAdams.

PART 9 - BIBLIOGRAPHY

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2. Gamel, C. M., M. S. Thesis in Chem. Eng., Massachusetts Institute of Technology, 1947

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4. Goldstein, S., Modern Developments in Fluid Dynamics, Oxford University Press, 1938, p. 71-74, 424-434

5. Taylor, J. J., Detroit Edison Company, private communication to R. N. Lyon, April 29, 1947

6. Kreith, F., Summerfield, M., Investigation of Heat Transfer at High Heat-Flux Densities. Progress Report No. 4-68 (Appendix), Jet Propulsion Laboratory, California Institute of Technology, Pasadena, California, April 2, 1948

7. Briggs, R. B., Technical Division- Section II Report for Period Ending June 15, 1947 (To M. C. Leverett- Clinton Laboratories) June 16, 1947

8. Knowles, J. W., Heat Transfer with Surface Boiling, MTec-187, National Research Council of Canada, Montreal Laboratory, February 16, 1946

9. McAdams, W. H., (Minden, C. S.) Eleventh Informal Monthly Report of MIT Project DIC 1-6489 for the Period August 18 - September 18, 1947, Massachusetts Institute of Technology, Cambridge, Mass.

10. McAdams, W. H., (Minden, C. S.), Fourteenth Informal Monthly Report of MIT Project DIC-6489 for the Period November 18 - December 18, 1947, Massachusetts Institute of Technology, Cambridge, Mass.

11. Minden, C. S., M. S. Thesis in Chem. Eng., Massachusetts Institute of Technology, 1947

12. Carl, R., Picornell, P., Local Boiling of Water in an Annulus, M.S. Thesis, Massachusetts Institute of Technology, 1948

13. Lansing, N. F., UCLA Heat Transfer Investigation for High Temperature Water-cooled Pile, Oak Ridge National Laboratories, Central Files No. 48-3-459, memo to file, March 24, 1948

14. Lansing, N. F., Visit to Prof. McAdams at MIT and to Code 443 BuShips, Oak Ridge National Laboratories, Central Files No. 48-4-82, memo to H. Etherington, April 2, 1948

[redacted]

ORNL 156-10

HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 10

REFERENCES RELATED TO HEAT TRANSFER WITH CHANGE OF PHASE

W. B. Harrison

October 1, 1948

[redacted]

PART 10

REFERENCES RELATED TO HEAT TRANSFER WITH CHANGE OF PHASE

Boiling and condensation are the cases of primary interest in this section of the report. Parts 8 and 9 are devoted to the boiling mechanisms, and the references noted here are to supplement the background information. One very informative study of boiling has been prepared under the direction of W. H. McAdams at Massachusetts Institute of Technology in the form of slow-motion pictures of the bubble formations on a hot wire.

A few miscellaneous references which may be useful in clearing up the picture of boiling and bubble growth are listed below.

Christ, C., Release of Gas from Supersaturated Solutions, CP-3063, June 23, 1945

Colburn, A. P., et al, Effect of Local Boiling and Air Entrainment on Temperatures of Liquid Cooled Cylinders, NACA Technical Note 1498, March 1948

Feld, B. T., On Cooling by Boiling Water, CE-553, August 1, 1945

Martin, A. V., Heat Flow from a Fin to a Boiling Liquid, CP-2995, May 11, 1945

This paper presents a mathematical investigation of the smallest length of fin which will dissipate 90% as much heat as an infinite length.

Schlegel, R., Evaporation of a Dissolved Gas from Water in a Heat Exchanger, CP-3062, June 23, 1945

Equations are developed for dissolved gas transfer in falling film exchangers.

Smetana, F., Characteristics of Boiling Solutions, CE-3510, March 7, 1945

Solutions were boiled in an open top glass box by passing an electric current through the solution.

Vernon, H. C., P-9 Pile Studies Section Report for Month Ending June 30, 1943, CE-754, June 30, 1943

Approximate formula for bubble formation is given on p.4.

Weills, J. T., Steam Bubble Size and Rate of Rise through Water at Its Normal Boiling Point, CT-910, September 3, 1943

Weills described experimental technique and included small amount of data. Bubble size and rate of rise were obtained photographically.

Wigner, E. P., The Rate of Rise of Bubbles, CP-1691, April 7, 1944

Equations are presented for bubble size and rate of rise.

Young, Gale, Transfer Phenomenon and Bubble Growth, MUC-GY-30, April 4, 1945

Young, Gale, Stability of a Fin in Boiling Liquid, CP-1902, MDDC-745, January 26, 1945

An idealized discussion of heat transfer in boiling is given , and some calculations of fin performance are made.

Young, Gale, Evaporation of Dissolved Gas from a Liquid, MUC-GY-28, March 7, 1945

The analogy between heat and momentum transfer is employed.

The project data on condensation appear to be confined primarily to condensation of solid uranium hexafluoride.

Cooper, G. T., Experimental Cold Trap Test Data, December 29, 1943

(On file at K-25 plant, Carbide and Carbon Chemicals Corporation, Oak Ridge, Tennessee)

Hodgson, M.A,E., Rate of Condensation of Pure VI and Thermal Resistance of VI Deposits, BR-410, April 15, 1944

Johnson, C.A., Study of Cold Trap Test Data, November 13, 1945 (File K-920 at K-25 plant, Carbide and Carbon Chemicals Corporation, Oak Ridge, Tennessee)

Kurtz, J. J., Olson, R. C., Test Results and Calculations on C-616 Condensation in Converters, A-4770, August 22, 1947

Thompson, W. I., Analysis of Single Tube Cold Trap Data, RB-3A, April 24, 1944

Thompson, W. I., Theory of Heat and Mass Transfer in Batch Condensation of Solids, MDDC-66, May 28, 1946

[redacted]

ORNL 156-11

HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 11

SELECTED TOPICS RELATED TO HEAT TRANSFER

W. B. Harrison
October 1, 1948

[redacted]

PART 11

SELECTED TOPICS RELATED TO HEAT TRANSFER

The following paragraphs are devoted to miscellaneous notes on a few topics related to heat transfer. The first notes refer to instruments which may be useful in heat transfer experiments, and the remainder of the section is devoted to allied subjects such as pressure drop, equivalent diameter and heat exchanger design.

This part of the report is essentially a reference section; consequently, little description is included. In order to facilitate locating the source for more information, references are included as footnotes.

(a) Density Measuring Device

In spite of the fact that local boiling permits very high heat fluxes, it cannot be considered as a good prospect for cooling a nuclear reactor until (a) the mechanism is understood and controlled, and (b) the change of density in a cooling channel is known and can be compensated for in the reactor designs. A unique method for density determinations was described by Lansing[1], after a discussion with R. Evans at Massachusetts Institute of Technology. The system is composed of two identical flow channels between a low energy gamma ray source and a counter for each channel. One flow channel is a static control and the other is the heat transfer channel. Evans pointed out that a density measuring instrument which utilizes radioactivity in a similar manner is marketed under the trade name Penetron by Engineering Laboratories, Inc., 610 E. 4th St., Tulsa, Oklahoma.

1. Lansing, N. F., Visit to Prof. McAdams at MIT and to Code 443 Bu Ships, Oak Ridge National Laboratories, Central Files No.48-4-82, Memo to H. Etherington, April 2, 1948

(b) Inductance Thermometer

The inductance thermometer was designed to fill the need for a temperature measuring device which does not interfere with steady flow and heat transfer conditions in a cylindrical tube. It measures the average temperature around the circumference of a tube over a length of one or two inches by making use of the fact that diameter of the tube is a linear function of its temperature. The tube diameter variation with temperature is measured indirectly as a change of electro-magnetic coupling between the tube and a concentric coil. The change of inductance of the coil which results from diameter variation changes the frequency of an oscillator and the frequency chance is made to give a direct meter reading by means of a discriminator circuit. The inductance thermometer was used to measure the average local temperature of a heated tube surrounded by water in forced convection in an annulus made by a Pyrex jacket. A complete description of the circuit, construction, calibration, and use is included in the appendix of a report by Bauer and Milner[2].

2. Bauer, S. G., Milner, P. M., Heat Transfer Experiments with Special Reference to Anodized Layers, MTec 143, National Research Council of Canada, Montreal Laboratory, June 5, 1945.

(c) Thermal Flowmeter

Measurement of non-pulsating mass flow rates of a gas stream may be made with a thermal flowmeter. The apparatus consists essentially of a tube (having low thermal conductivity) with a heating coil in the middle and a means for keeping the temperature constant at each end. The tube temperature difference between the point 1/4 down the tube length and the point 3/4 down the tube length is calibrated against mass flow rate. Accuracy within ± 1% has been attained by such an instrument when calibrated in place, operated under constant conditions and checked at monthly intervals. Swartz[3] has presented a complete study of thermal flowmeters, including construction, associated apparatus, and operating characteristics.

3. Swartz, C. D., Thermal Flowmeters, A-3219, Columbia Serial No. 1C-M470, Columbia University, Division of War Research, SAM Project, January 15, 1945

Other references from Swartz:

Blackett, Henry, and Rideal, Proc. Roy. Soc. A126, 319 (1929)

Blackett, and Henry, Proc. Roy. Soc. A126, 333 (1929)

Haggstrom and Nordsieck, A-87, January, 1942

Booth, Haggstrom, and Callihan, A-88, January, 1942

Callihan, Columbia Serial No. 5B3-L15, April, 1944

(d) Water Rate Determination

For accurate flow rate measurements, Kratz, Raeth and Christ[4] described a method which is particularly useful at low flow rates, where orifice plates and flowmeters were found to be inaccurate. The system involves a circuit which measures automatically the time required for a given weight of water to flow throught the system. The apparatus includes 3 relays and a platform balance with contacts at the top and bottom of the beam swing. Accuracy of timing is reported to be within $\pm$ 0.05 seconds.

(e) Equivalent Diameter

In the case of an annulus, friction data of Carpenter, Colburn, and Schoenborn[5] show that the appropriate choice of diameter for use in the Reynolds number is the equivalent hydraulic diameter, D_e.

$$D_e = \frac{4 \times \text{cross-sectional area}}{\text{perimeter}}$$

4. Kratz, H. R., Raeth, C. H., Christ, C. F., The Thermal Transfer from Slugs to Jackets and from Slugs to Liners, CP-2302, November 7, 1944

5. Carpenter, F. G., Colburn, A. P., Schoenborn, E. M., Tr. A.I.Ch.E. 42, 165-187 (1946)

Use of D_e has also been confirmed by the data of Washington and Marks[6] for heat transfer in rectangular ducts, and the data of Green and King[7] for flattened tubes. As part of a report on gaseous flow considerations, Normand[8] made calculations of average equivalent diameter for channels in which cross-section changes with length.

For the case of parallel plates where one dimension of the flow cross-section is large compared with the other, the expression is often simplified to give twice the distance between plates for the equivalent hydraulic diameter.

In an annulus, the equivalent diameter is the difference between the larger and smaller diameters $D_e = D_2 - D_1$.

Use of equivalent diameter is indicated in the sections of this report on heat transfer to liquids in an annulus (Part 5), heat transfer to fluids between parallel plates (Part 6), and pressure drop (Part 11).

6. Washington, L., Marks, W. M., Ind. Eng. Chem. 29, 337-345 (1937)
7. Green, F. H., King, L. S., Tr. A.S.M.E. 68, 115-122 (1946)
8. Normand, C. E., Gaseous Flow, D-4.440.8, CEW-TEC, August 27, 1946

(f) Critical Reynolds Number

Critical Reynolds number Re_c, has been defined as the value of $\frac{DV\rho}{\mu}$ below which the fluid flow in a system is always laminar or streamline. The Re_c is usually considered to be of the order of 2100 for circular tubes, as pointed out by McAdams[9], but it is of interest to note that it may have much lower values in capillaries. In June 1945, Myerson and Eicher[10] reported Re_c as low as 400, and, in December 1945[11], they reported Re_c as low as 10 in one case, and 100 in another. In their December (1945) report, reference was made to the work of Ruckes[12] who reported Re_c as low as 400 with metal capillaries (having an I.D. of 0.04 cm. and length of 150 cm.) and Re_c of 2000 in glass capillaries of the same size. This may indicate that the surface condition influences the thickness of the laminar boundary sublayer, which becomes a significant thickness with respect to tube diameter in capillaries. This information is also important in considering the case of fluid flow through porous materials.

9. McAdams, W. H., Heat Transmission, 2nd Ed., McGraw-Hill, New York, 1942, p. 100-101.

10. Myerson, A., Eicher, J., Viscosity of Gaseous C-616, M-2518, June 30, 1945

11. Myerson, A., Eicher, J., The Viscosity of Gaseous Uranium Hexafluoride, A-3825, Dec. 26, 1945

12. Ruckes, W., Ann. d. Phys. (4) 25, 983 (1908)

(g) Pressure Drop and Friction Factor

For the case of fluid flow in an annulus, Donovan[13] reviewed the literature in a memorandum to Briggs. Work done on this case in project laboratories is confined primarily to that of Bankoff and Sanford[14,15,16], Sereda[17], and Boyer and Gishler[18], who determined pressure drops over a wide range of velocities and heat fluxes.

13. Donovan, J. R., Heat Transfer and Pressure Drop in Annular Spaces - A Literature Survey, ORNL Central Files No. 48-4-411, Memo to R. B. Briggs, April 27, 1948

14. Bankoff, S. G., Sanford, H. B., Pressure Drop in Ribbed Annuli, CE-952, Sept. 30, 1943

15. Bellas, H. W., Report for Month Ending July 26, 1943, CE-820, Tech. Res. - Eng. Dev.

16. Bellas, H. W., Report for Month Ending June 26, 1943, CE-755, Tech. Res.- Eng. Dev.

17. Sereda, P. J., Flow Characteristics of a Dummy X-Rod Assembly, CRX-263

18. Boyer, T. W., Gishler, P. E., Pressure Measurements in Annuli of Ribbed Tubes, MX-184, Nov. 6, 1945

[redacted]

Bauer and Morrow[19] reported pressure drop data for air flow through diamond-shaped graphite channels, including contraction and enlargement losses and pressure drop with and without slugs in the channel. Air velocities between 50 and 280 ft./sec. were explored. Larson[20] also presented data on flow rate and pressure drop in open and slug-filled channels.

Hebert and Knowles[21] reported determinations of friction coefficients for the compressible flow of steam for Reynolds numbers between 2000 and 13,200.

Examples of pressure drop calculations for project problems may be found as applied to the Daniels experimental power pile[22], high temperature oxide pile[23], and other gas-cooled piles[24], parallel flow helium cooling[25], and sphere beds[26].

19. Bauer, S. G., Morrow, N., The Flow of Air Through Graphite Channels, MTec-200, Jan. 15, 1946

20. Larson, R. E., Air Flow Rate Through Open and Filled "X" Type Channels, CE-2535, December 21, 1944

21. Hebert, S. A., Knowles, J. W., Friction Coefficients for the Compressible Flow of Steam, MTec-209, April 5, 1946

22. Palladino, N. J., Operating Characteristics of Main Circulating System, MonN-232, December 27, 1946

23. Robertson, A. F., Fromm, L. W., Kittredge, H. E., Estimates Regarding Gas Flow and Heat Transfer in a High Temperature Oxide Pile, CT-3460, March 1, 1946

24. Amorosi, A., Heat Transfer in Gas-Cooled Piles, MonN-299, Appendix B, May 29, 1947

25. Feld, B. T., Szilard, L., Examples of Pressure Drop Calculations in Parallel Flow Helium Cooling, CP-308, June 18, 1942

26. Young, G., Jupa, E. C., Fluid Flow and Heat Transfer in Sphere Beds, MUC-GY-31, April 6, 1945

Some data and calculations for the Clinton pile have been presented by Leverett[27].

Martinelli and Nelson[28] presented a method for predicting pressure drop during forced convection boiling of water.

(h) Heat Exchanger Design Calculations

Much work can be done on paper in evaluating heat exchanger considerations, as well as in the development laboratories. Typical of this type of treatment is a report by Matheson[29] on tubular heat exchangers for a homogeneous reactor. Since hold-up volume is an important consideration in this case, Matheson studied the effects on hold-up of the following variables:

a. Dirt film or scale resistance

b. Terminal temperature

c. Tube size

d. Liquid velocity

The report of this work includes tables of calculations and numerous figures which show the influence of the variables.

27. Leverett, M. C., Air Flow and Heat Removal in the Clinton Pile, MonN-157, August 1946

28. Martinelli, R. C., Nelson, D. B., Tr. A.S.M.E. 70, 695-702 (1948)

29. Matheson, J. H. P., Tubular Heat Exchangers for a Homogeneous Unit, ME-39, February 1, 1944

Amorosi[30] presented numerous calculations on design of a gas-cooled reactor. He illustrated the effect of various factors on pile output and gas passage area; the effect of unconventional geometry; the heat transfer conditions which affect high specific thermal output; and the limitations on mechanical design.

Additional information is presented by Leverett[31] on gas lift coolers; Phillips[32], Feder and Ward[33] on tube shell coolers; Phillips and Ward[34] on vertical tube, falling film coolers; Feder and Ward[35], and Stanley[36] on horizontal film, trombone coolers; Monet and Ward[37] on a "Geyser" cooled pile. Kress[38] also presented calculations on the falling film type exchanger, and included some experimental data on maximum flow rates and gas-release.

30. Amorosi, A., Heat Transfer in Gas-Cooled Piles, MonN-299, May 29, 1947 (See also MonN-383)

31. Leverett, M. C., Enriched Pile- Engineering Considerations I, February 6, 1945 (Clinton Laboratories Memorandum)

32. Phillips, B. E., Enriched Pile- Engineering Considerations II, April 12, 1945 (Clinton Laboratories Memorandum)

33. Feder, H. M., Ward, F. R., Enriched Pile-Engineering Considerations II-B, June 12, 1945 (Clinton Laboratories Memorandum)

34. Phillips, B. E., Ward, F. R., Enriched Pile-Engineering Considerations III: Studies of Vertical Tube Falling Film Coolers, June 14, 1945 (Clinton Laboratories Memorandum)

35. Feder, H. M., Ward, F. R., Enriched Pile -Engineering Considerations V: Studies on the Horizontal Film Type Cooler (Trombone), July 13, 1945 (Clinton Laboratories Memorandum)

36. Stanley, W. M., Supplement "A" to Engineering Consideration V: Studies on the Horizontal Film Type Cooler (Trombone), September 26, 1945, (Clinton Laboratories Memorandum)

37. Monet, M., Ward, F. R., Enriched Pile-Engineering Considerations IV- The "Geyser" Pile, June 28, 1945 (Clinton Laboratories Memorandum)

38. Kress, B. A., Falling Film Heat Exchanger, CE-3503, May 30, 1945

Young[39,40] and Schlegel[41] presented theoretical work applicable to falling film exchangers and experimental data are mentioned by Lawrence[42] and Dempster[43].

Considerations involving such problems as the effect of velocity, heat transfer coefficients, piping length, etc., on holdup have been presented by Young[44]. (Young[45] also discusses related pile problems such as effect of temperature level on pile characteristics, pile temperature for greatest output, and decomposition of pile water.) Another paper by Young[46] presented a mathematical examination of an idealized heat exchanger.

Huffman[47], and Ward and Weills[48] published design data furnished by the Andale Company, and included considerations regarding the choice of heat exchanger types and optimum cooling systems.

39. Young, Gale, Some Notes on Falling Films, MUC-GY-30, March 1, 1945

40. Young, Gale, Pile Design Group, CF-2926, April 15, 1945

41. Schlegel, R., Evaporation of Dissolved Gas from Water in a Heat Exchanger, CP-3062, June 23, 1945

42. Lawrence, G. C., Technical Physics Division Progress Report for Month Ending August 1, 1944, MPTec 1, August 1, 1944

43. Dempster, A. J., Report for Month Ending March 15, 1945- Part I, CF-2796, March 15, 1945

44. Young, G., Pile Heat Exchanger System, CP-807, July 22, 1943

45. Young, G., Pile Heat Exchanger System-Part II, CP-807 (II), Aug. 6, 1943

46. Young, G., Pile Heat Exchanger Calculations, CP-2701, Feb. 14, 1945

47. Huffman, J. R., Maximum Utilization of P-9 in Homogeneous Slurry Piles, CE-1143, December 15, 1943

48. Ward, F. R., Weills, J. T., P-9 Utilization in Slurry Pile Cooling System, CT-969, Oct. 5, 1943

Some heat transfer calculations for the Clinton pile have been presented by Leverett[49]. Young[50,51] discussed various cooling proposals for reactors. A paper by Karush and Young[52] is particularly recommended for background material on energy removal from piles.

A few comments were located on cooling with graphite powder[53] but no final report of the project appears to be available.

Additional references which may be useful in the design of heat exchangers are noted in Parts 7 and 10 of this report.

(i) Thermocouples

A few references on thermocouples are included in view of the fact that errors in experimental data are frequently due to an improper thermocouple installation. Krogh[54] has pointed out a few pertinent facts on the selection, application, and limitations of thermocouples; and Roberts and Vogelsang[55] presented a discussion on selection and installation of thermocouple lead wires.

49. Leverett, M. C., Air Flow and Heat Removal in the Clinton Pile, MonN-157, August 1946

50. Young, G., Some Notes on Cooling Streams, CL-GY-7, April 9, 1947

51. Young, G., The Various Cooling Proposals, Memo 8, (No date shown)

52. Karush, W., Young, G., Energy Removal from Piles, MonP-147, July 30, 1946

53. Lewis, W. K., Mooré, T. V., et al, Cooling with Graphite Powder, CS-215, August 4, 1942

54. Krogh, A. E., Instrumentation 1, No. 4, 27-29 (1945)

55. Roberts, C. C., Vogelsang, C. A., Instrumentation 2, No. 5, 27-30 (1946)

Lancaster and Brunot[56] described a technique for calibrating a couple for temperatures between 32 and 1300 °F. At about 850 °F. they reported a deviation of 0.06 °F. as the depth of immersion of a couple in sulphur vapor was changed from 4 to 12 inches.

Cichelli[57] treated several typical cases for design of a reliable temperature measuring element (thermocouple or thermometer installation) for measuring gas temperatures. Boelter and Lockhart[58] presented a report on the conduction error observed in measuring surface temperature.

In a project report, Kratz, Schlegel, and Christ[59] outlined a procedure for soldering thermocouples in a tube wall, so as to get good thermal contact but not to distort the surface on the outside of the tube. Krieth and Summerfield[60] describe a technique for installing tube-wall thermocouples by a method involving peening the wires in place and insuring contact by means of discharging a spot welder through the junction.

56. Lancaster, H. B., Brunot, A. W., Gen. Elec. Review 45, 649 (1942)

57. Cichelli , M. T., Ind. Eng. Chem. 40, 1032-1039 (1948)

58. Boelter, L. M.K., Lockhart, R. W., An Investigation of Aircraft Heaters, "XXXVI. Thermocouple Conduction Error Observed in Measuring Surface Temperature", NACA ARR, February 1946

59. Kratz, H. R., Schlegel, R., Christ, C. F., Thermal Transfer to an Annular Water Stream in the Neighborhood of a Rib, CP-2313, (Fig. 3) Nov. 10, 1944

60. Krieth, F., Summerfield, M., Investigation of Heat Transfer at High Heat Flux Densities: Literature Survey and Experimental Study in Annulus, Appendix A, Progress Report No. 4-65, Jet Propulsion Laboratory, California Institute of Technology, Pasadena, California.

Another wall-temperature measuring device (induction thermometer) is discussed earlier in this section of the report.

McAdams[61] referred to several sources for additional information on thermocouples and made a few comments on thermocouple errors due to radiation.

61. McAdams, W. H., Heat Transmission, 2nd Ed., McGraw-Hill, New York, 1942, p. 149-152, 223-225.

ORNL 156-12

HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES : A CRITICAL SURVEY

PART 12

PHYSICAL PROPERTIES OF WATER AND WATER VAPOR

W. B. Harrison
October 1, 1948

PART 12

PHYSICAL PROPERTIES OF WATER AND WATER VAPOR

Since water is widely used as a coolant, it is frequently convenient to designers to have a compilation of the physical properties of water and water vapor. An attempt has been made to include in this part of the report the physical properties useful in heat transfer calculations. As one would expect, considerable doubt exists regarding absolute values of the properties in the vicinity of the critical point.

For thermal conductivity of liquid water, the data of Barratt and Nettleton[2], Martin and Lang[3], and Schmidt and Sellschopp[4] are shown on Figure 12-1 as taken from Dorsey[1]. The data of Timrot and Vargaftik[5] are in agreement with Schmidt and Sellschopp and they are shown on Figure 12-2 as taken from the translation by K. A. Gardner of the Griscom-Russell Company. The plots of thermal conductivity of water vapor as shown on Figure 12-3 are taken from a translation by Gardner of another paper by Timrot and Vargaftik[6].

Viscosity of water at atmospheric pressure is shown on Figure 12-4 as compiled by Dorsey[1,7]. For temperatures greater than 212° F., data which Dorsey compiled were adapted primarily from the work of Sigwart[8] and Shugayev[9]. Hawkins, Sibbitt, and Solberg[13] recently published a paper which summarizes the viscosity data available for saturated water and steam. Figure 12-5 shows a plot of the data of Shugayev[9], Hawkins et al[10,11], Sigwart[8], and Timrot[12] for saturated water. Values for compressed water are shown on Figure 12-6, utilizing the data of Sigwart[8] as presented by Dorsey[1].

Data for viscosity of water vapor are shown on Figure 12-7. It is apparent from Figure 12-7 and Figure 12-5 that there is a considerable discrepancy in the viscosity data reported near the critical temperature for both liquid water and water vapor. Since the data of Sigwart[8] and Timrot[12] agree for the saturated liquid, confidence is placed in Timort's data[12] for the saturated vapor. The other data shown in Figure 12-7 for the saturated vapor were presented by Hawkins, Solberg, and Potter[14].

Dorsey[1] and Keenan and Keyes[15] present data for heat capacity of liquid water and water vapor. Figure 12-8 is a plot of the heat capacity of liquid water as prepared by N. F. Lansing* from data in the references noted above. Figure 12-9 is a plot of heat capacity of water vapor as taken from Kennan and Keyes.

Prandtl number, $\frac{c\mu}{k}$, was calculated for liquid water as a function of temperature, and the values are plotted on Figure 12-10. For high temperatures, approximate values of Prandtl number are shown on Figure 12-11 . Approximate Prandtl numbers for water vapor are shown on Figure 12-12.

ACKNOWLEDGMENT:

N.F. Lansing* and W. L. Sibbitt** made many contributions to this section of the report. G. Johnstone* performed a part of the calculations for Prandtl number. Other acknowledgments which apply to the entire report may be found in Part I.

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**Department of Mechanical Engineering, Purdue University, Lafayette, Indiana.

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3. Martin, L. H., Lang, K. C., Proc. Phys. Soc. (London) 45, 523-529 (1933)

4. Schmidt, E., Sellschopp, W., Forsch. Gebiete Ingenieurw. 3, 277-286 (1932)

5. Timrot, D. L., Vargaftik, N. B., J. Tech. Phys. (U.S.S. R) 10, 1063 (1940)

6. Timrot, D. L., Vargaftik, N. B., J. Tech. Phys. (U.S.S.R) 9, 63-70 (1939)

7. Dorsey, N. E., Int. Crit. Tables, 5, 10 (1929)

8. Sigwart, K., Forsch. Gebiete Ingenieurw. 7, 125-140 (1936)

9. Shugayev, V., Chem. Abs. 29, 2804 (1935) J. Exp. Theo. Phys.(U.S.S.R.), 4, 760-765 (1934)

10. Hawkins, G.A., Solberg, H. L., Potter, A.A., Tr. A.S.M.E. 57, 395-400 (1935)

11. Hawkins, G.A., Solberg, H. L., Potter, A.A., Tr. A.S.M.E. 58, 258-262 (1936)

12. Timrot, D. L., Journal of Physics (U.S.S.R.) 2, No. 6, 419-435 (1940)

13. Hawkins, G.A., Sibbitt, W. L., Solberg, H.L., Tr. A.S.M.E. 70, 19-24 (1948)

14. Hawkins, G.A., Solberg, H. L., Potter, A.A., Tr. A.S.M.E. 62, 677-688 (1940)

15. Keenan, J. H., Keyes, F. G., Thermodynamic Properties of Steam, John Wiley and Sons, 1936, p. 79-81.

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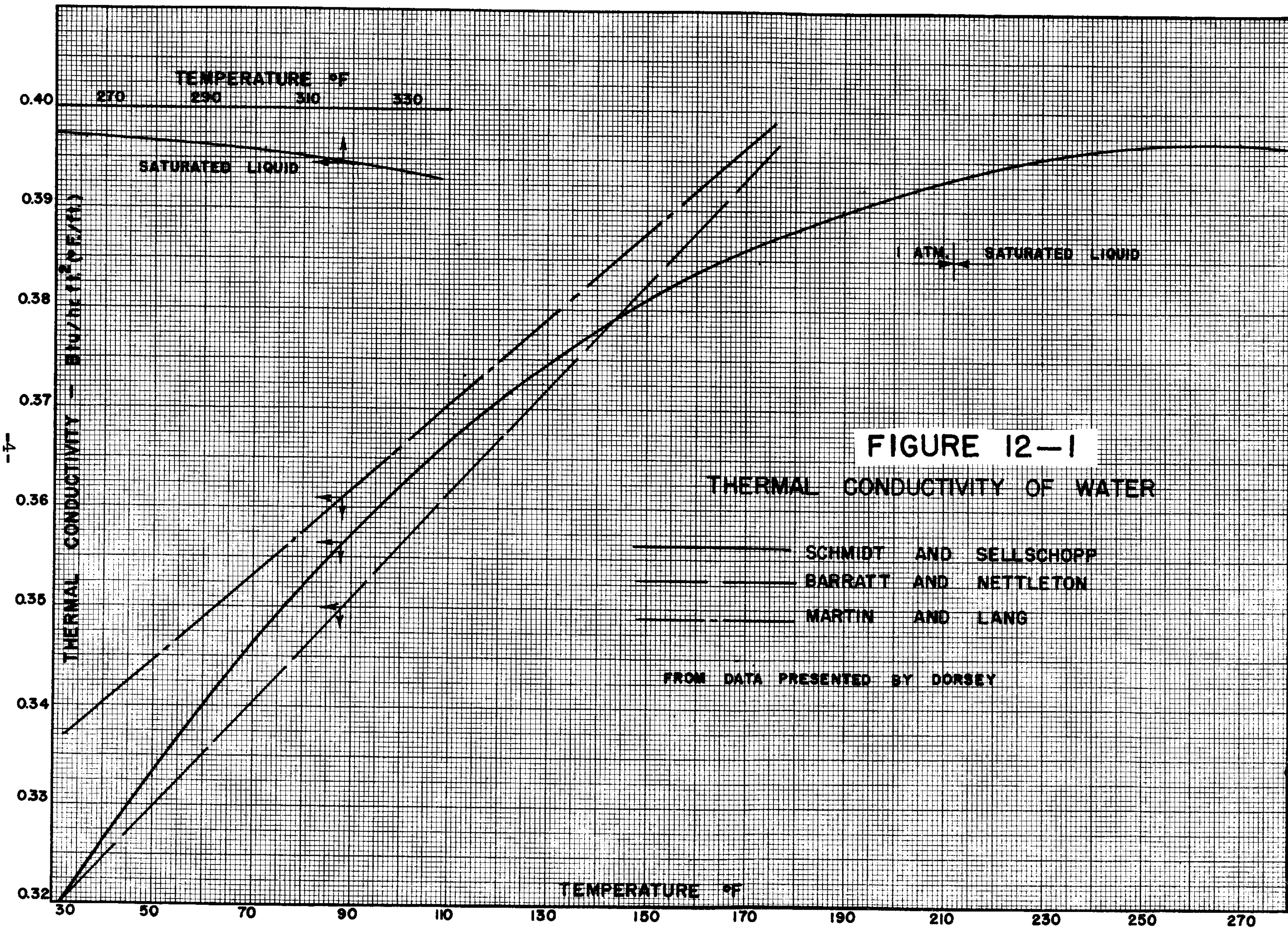

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Drawing # 6702

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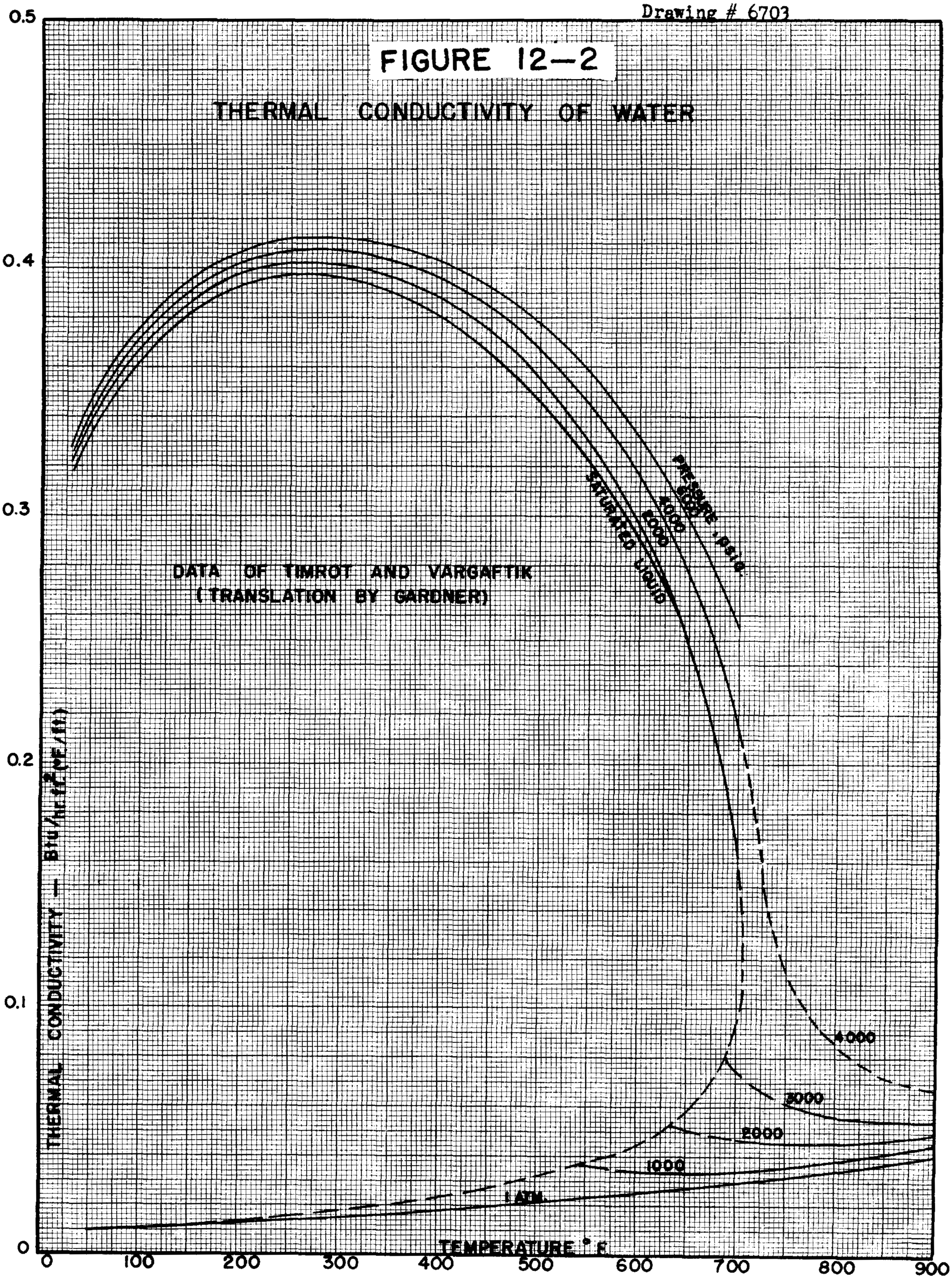

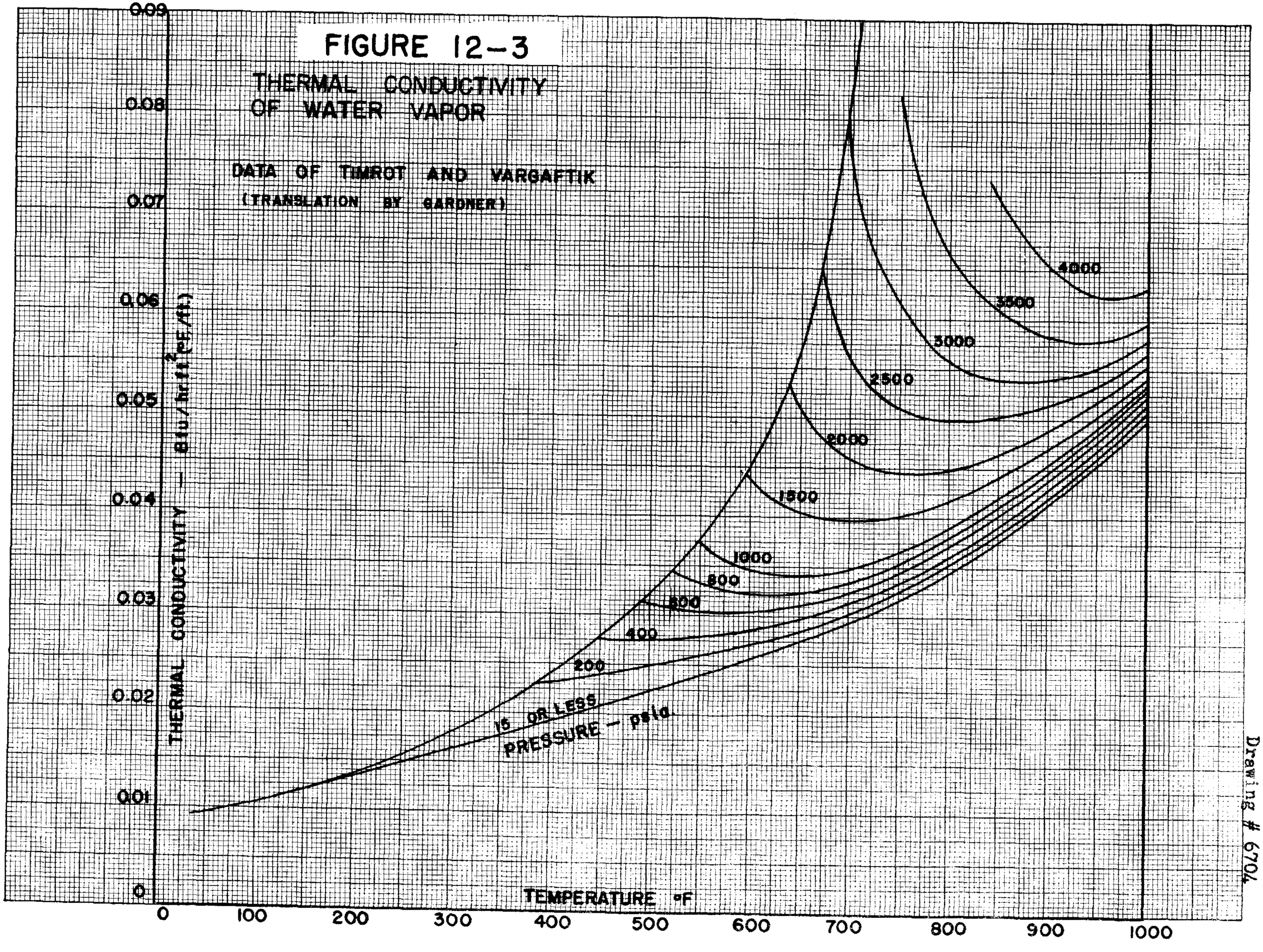

FIGURE 12-3

THERMAL CONDUCTIVITY OF WATER VAPOR

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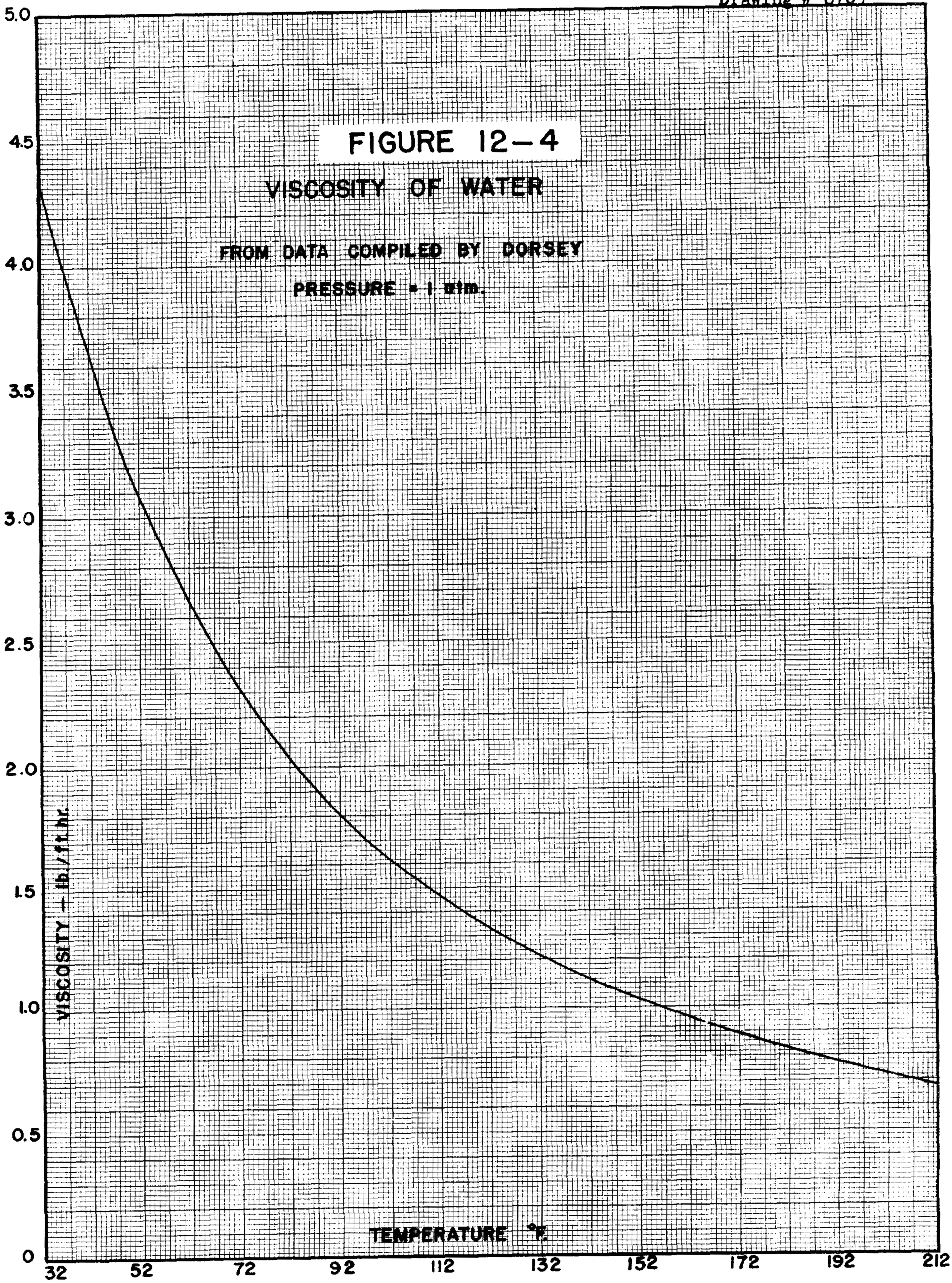
Drawing # 6705



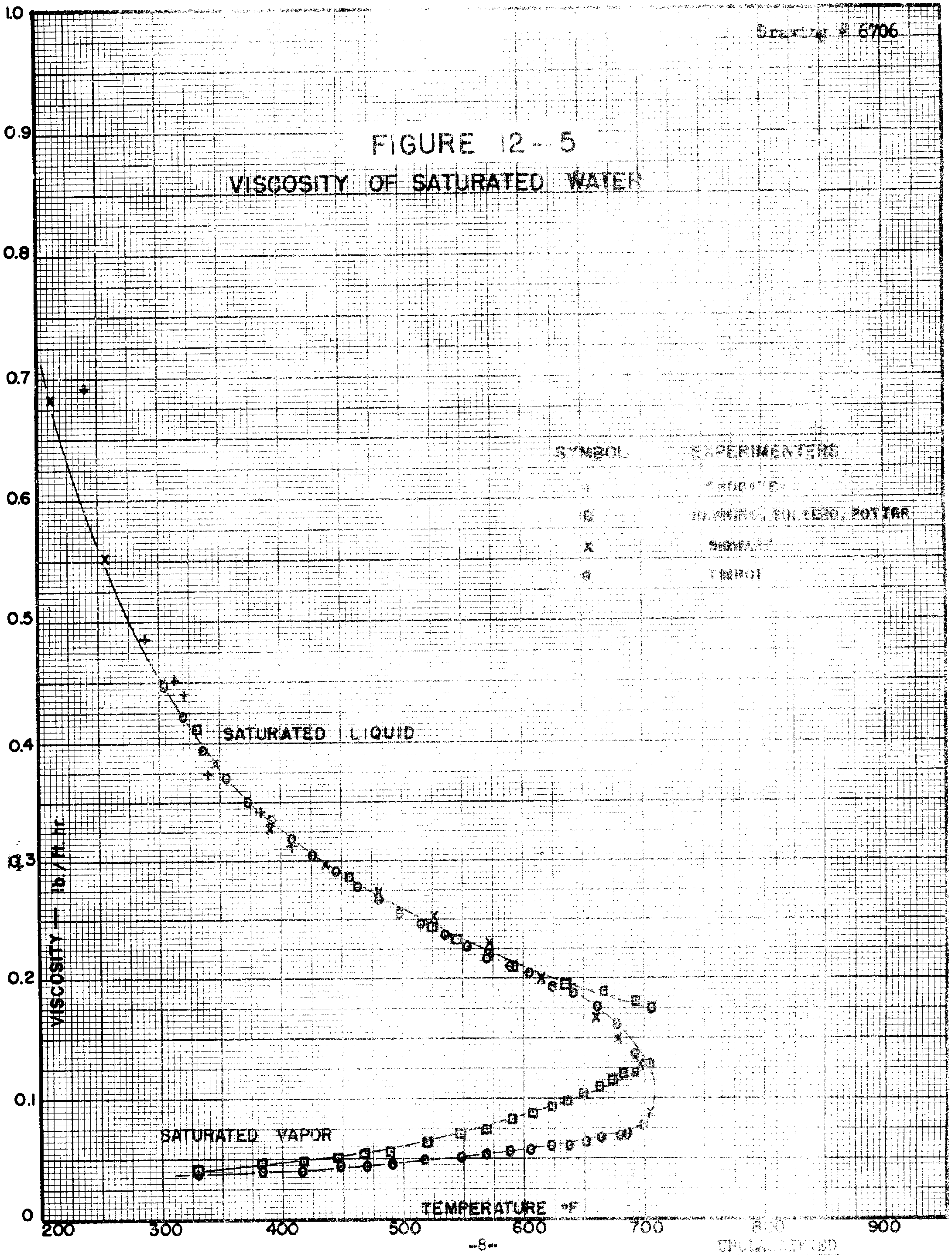
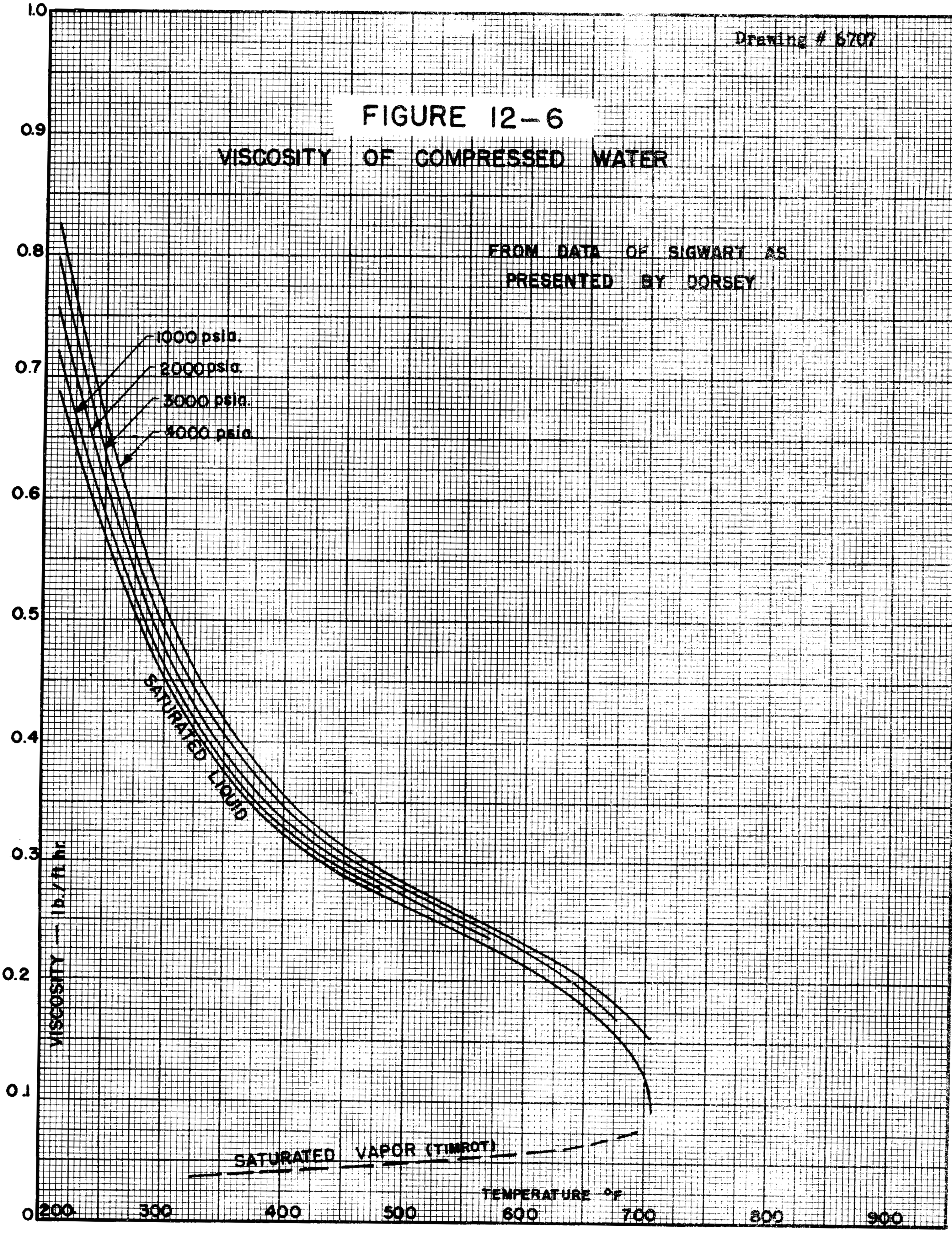
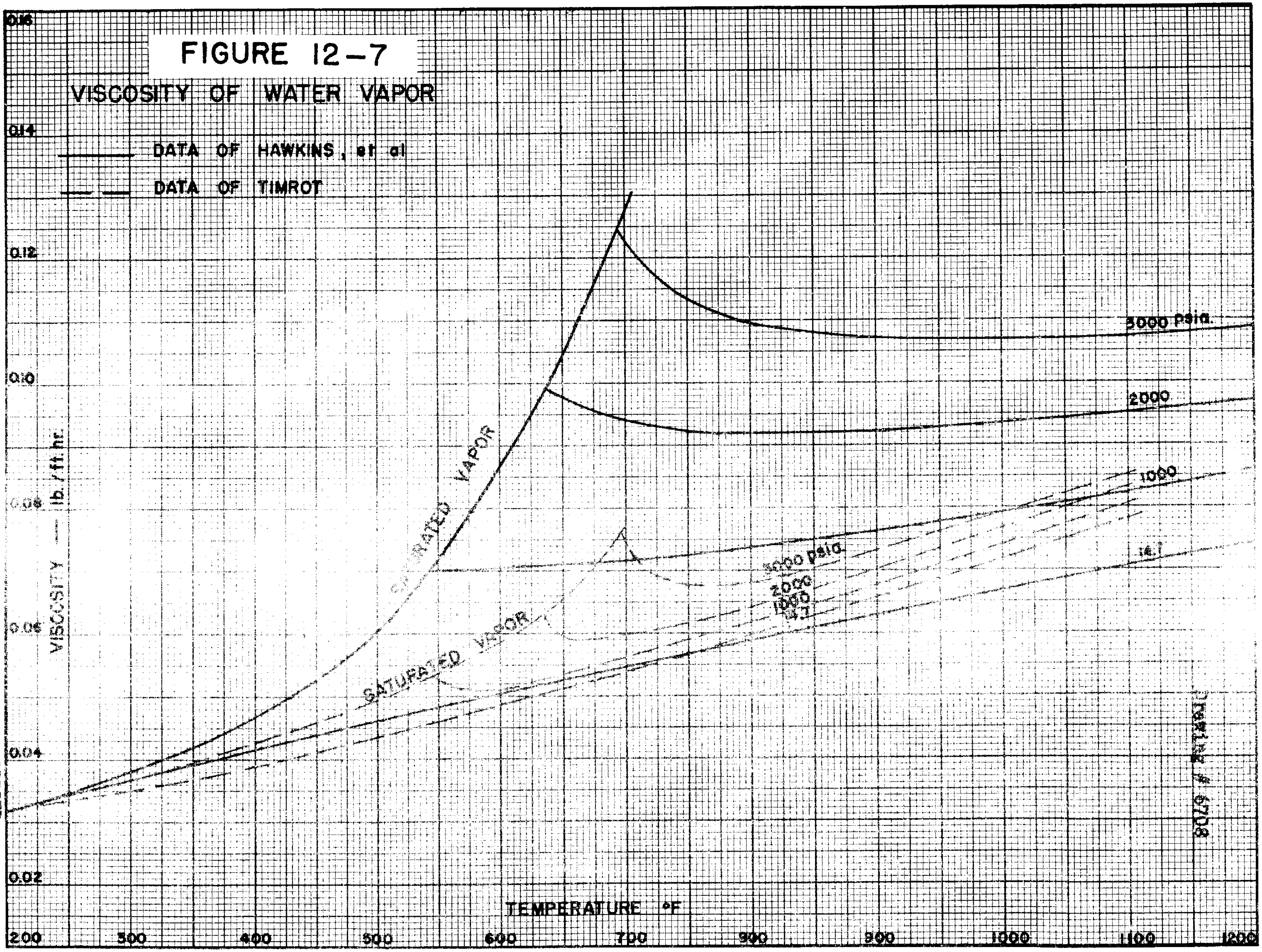


FIGURE 12-5
VISCOSITY OF SATURATED WATER



Drawing # 6707
FIGURE 12-6
VISCOSITY OF COMPRESSED WATER
FROM DATA OF SIGWART AS PRESENTED BY DORSEY
1000 psia.
2000 psia.
3000 psia.
4000 psia.
SATURATED LIQUID
SATURATED VAPOR (TIMROT)
VISCOSITY — lb./ft.hr.
TEMPERATURE °F
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
200
300
400
500
600
700
800
900



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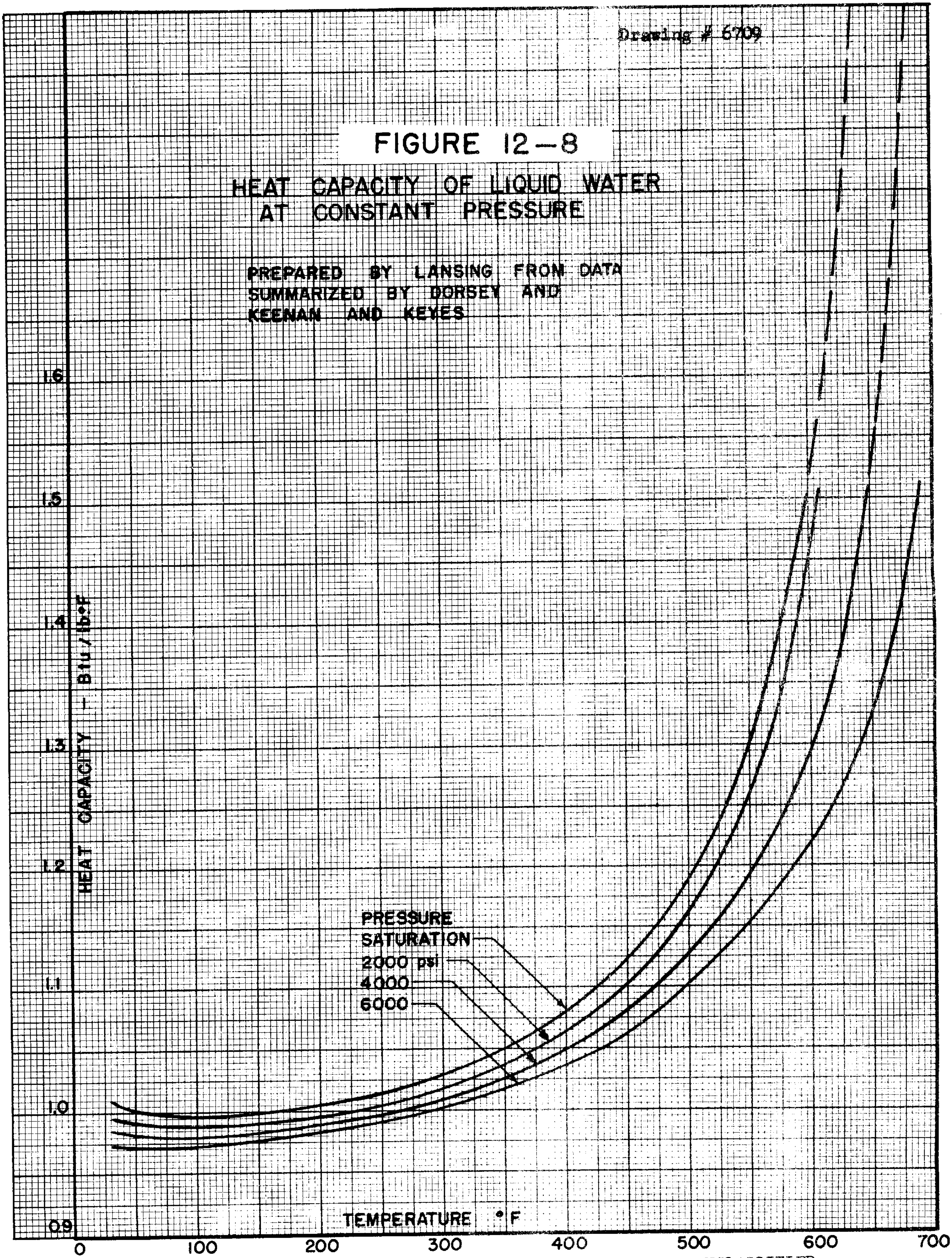
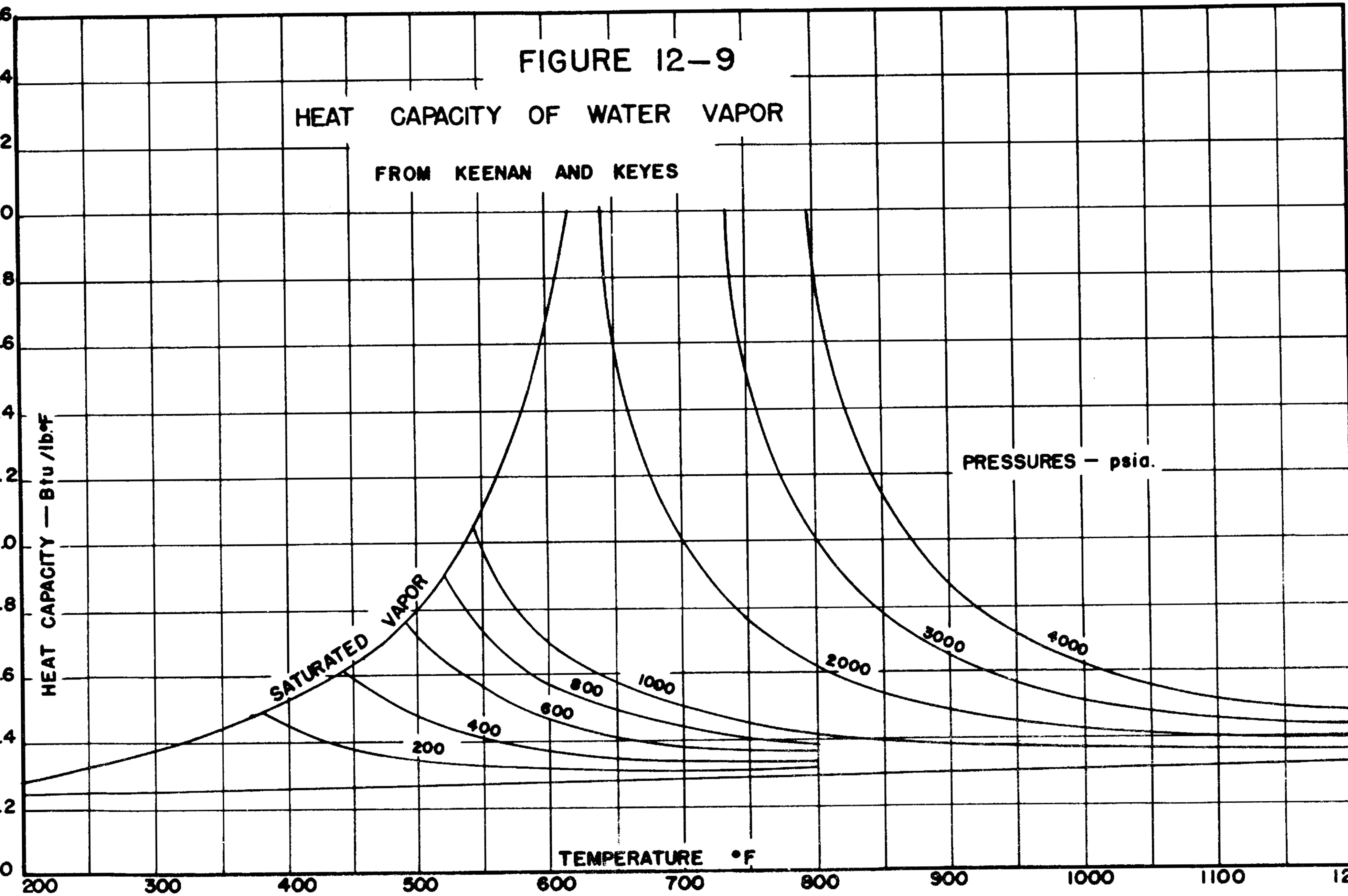


Drawing # 6709
FIGURE 12-8
HEAT CAPACITY OF LIQUID WATER
AT CONSTANT PRESSURE
PREPARED BY LANSING FROM DATA
SUMMARIZED BY DORSEY AND
KEENAN AND KEYES
PRESSURE
SATURATION
2000 psi
4000
6000
HEAT CAPACITY - Btu / lb•F
TEMPERATURE °F
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
0
100
200
300
400
500
600
700



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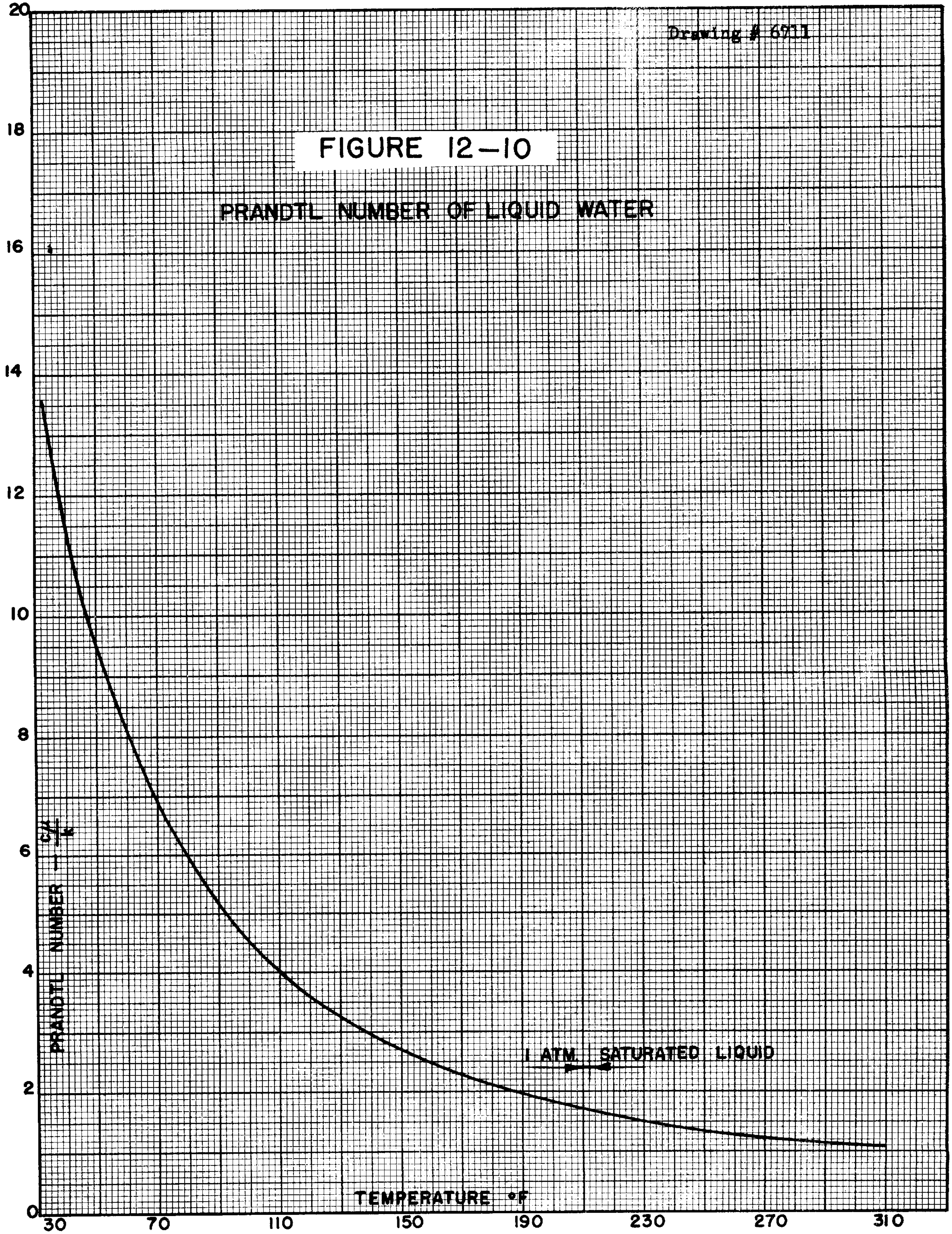
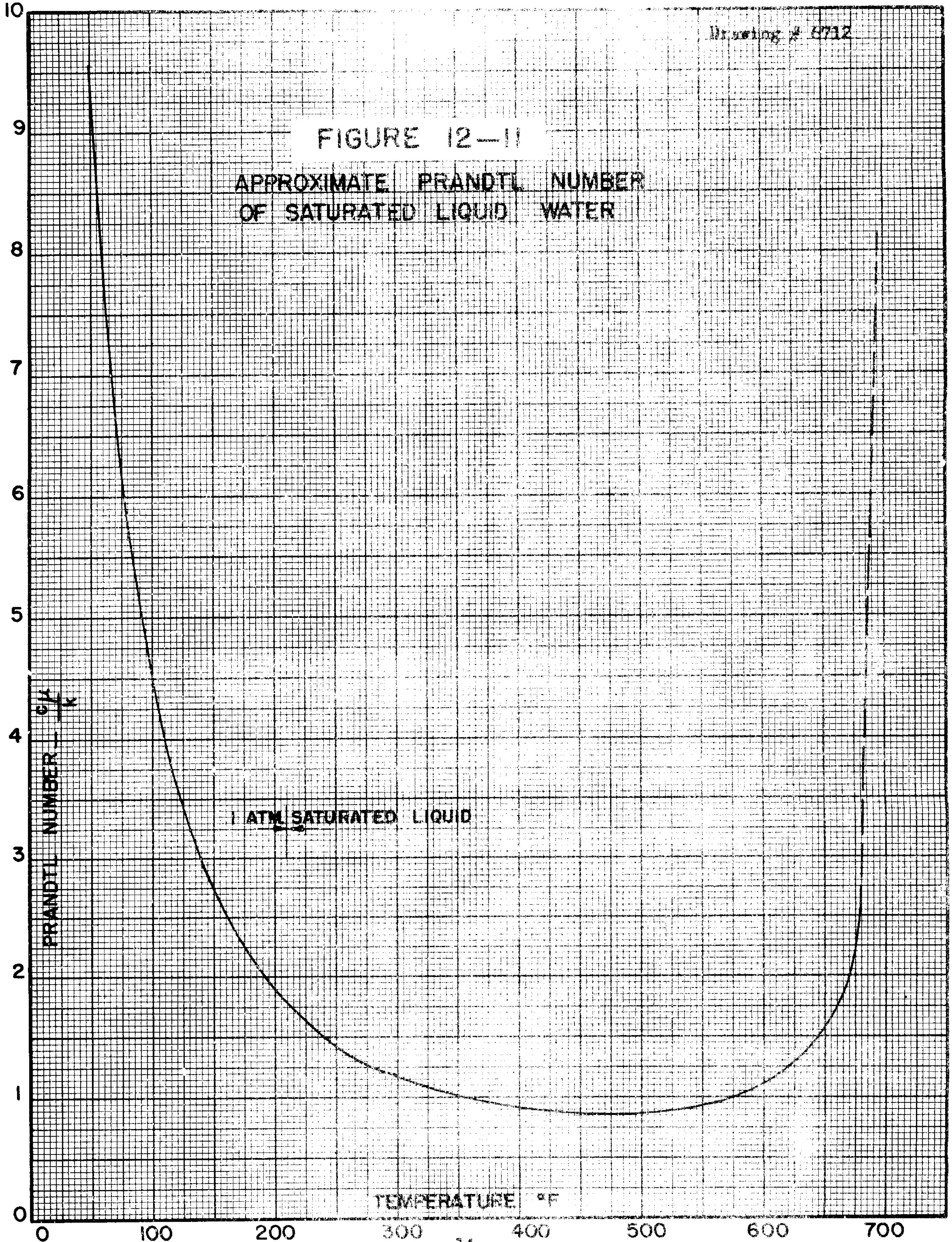
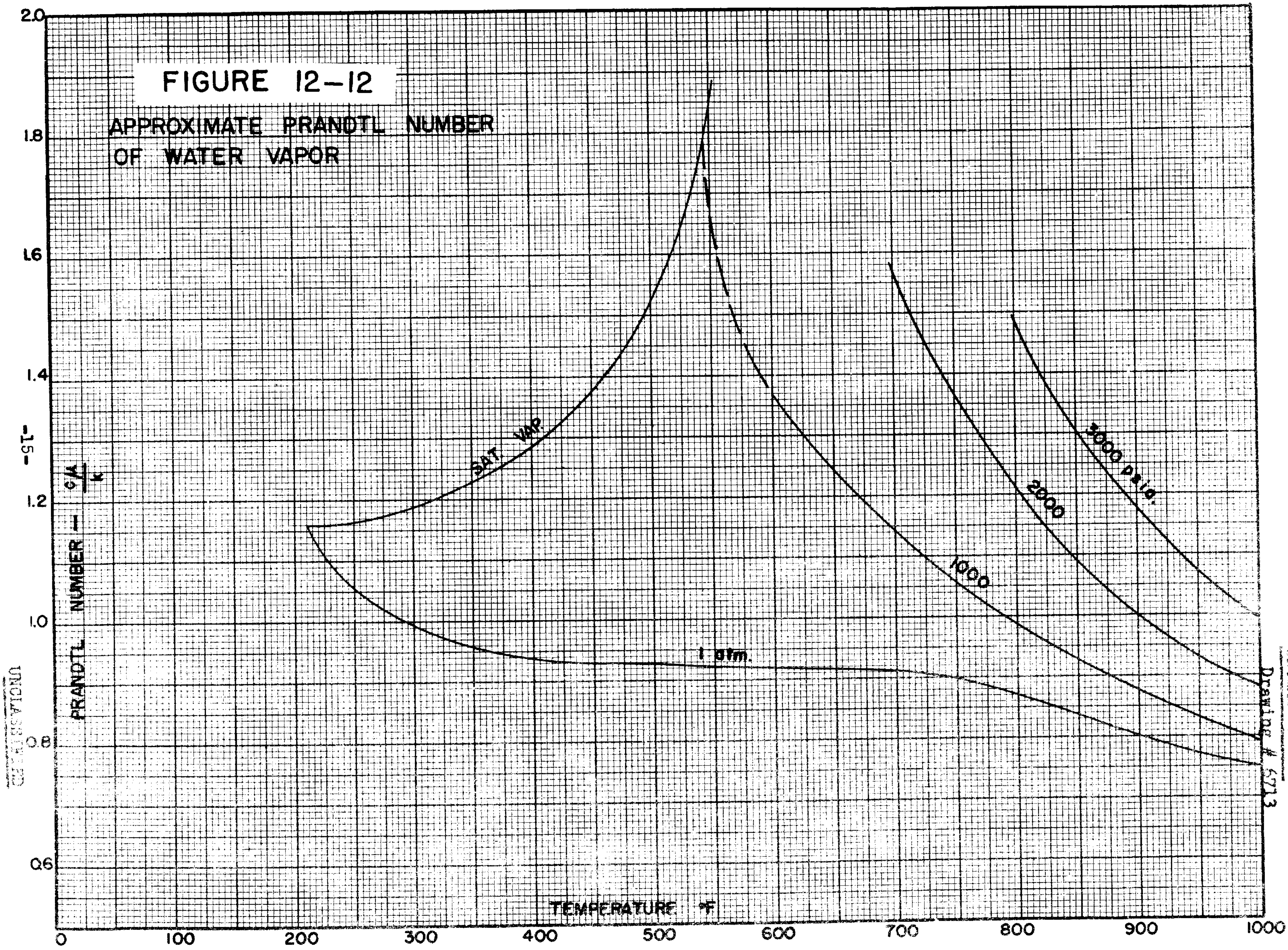


FIGURE 12-10

PRANDTL NUMBER OF LIQUID WATER


Drawing # 6712
FIGURE 12—11
APPROXIMATE PRANDTL NUMBER
OF SATURATED LIQUID WATER
PRANDTL NUMBER — $\frac{c\mu}{k}$
1 ATM SATURATED LIQUID
TEMPERATURE °F
0
1
2
3
4
5
6
7
8
9
10
100
200
300
400
500
600
700



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Drawing # 4713

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ORNL 156-13

HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES : A CRITICAL SURVEY

PART 13

PHYSICAL PROPERTIES OF LIQUID METALS

W. B. Harrison
October 1, 1948

[redacted]L

PART 13

PHYSICAL PROPERTIES OF LIQUID METALS

Heat transfer to liquid metals by combined conduction and forced convection is discussed in Part 4 of this report. Included in this part are plots of the physical properties of sodium-potassium alloy and mercury. The data are believed to be the best available at the present time, but the data for sodium-potassium alloy are likely to be revised in the near future.

Physical Properties of Sodium-Potassium Alloy (44 weight % Na)

Data for the following figures were taken from Ewing, Werner, King and Tidball.[1] Their references are as follows:

Viscosity -	NRL-C-3105 (April 1947)
Density -	NRL-P-3010
Heat Capacity -	NRL-C-3152 (July 1947)
Thermal conductivity-	NRL-C-3152 (July 1947)

1. Ewing, C. T., Werner, R. C., King, E. C., Tidball, R. A., Quarterly Progress Report No. 6 on the Measurement of the Physical and Chemical Properties of Sodium-Potassium Alloy, NRL Report No. C-3243, Naval Research Laboratory, February 25, 1948.

Drawing # 6699

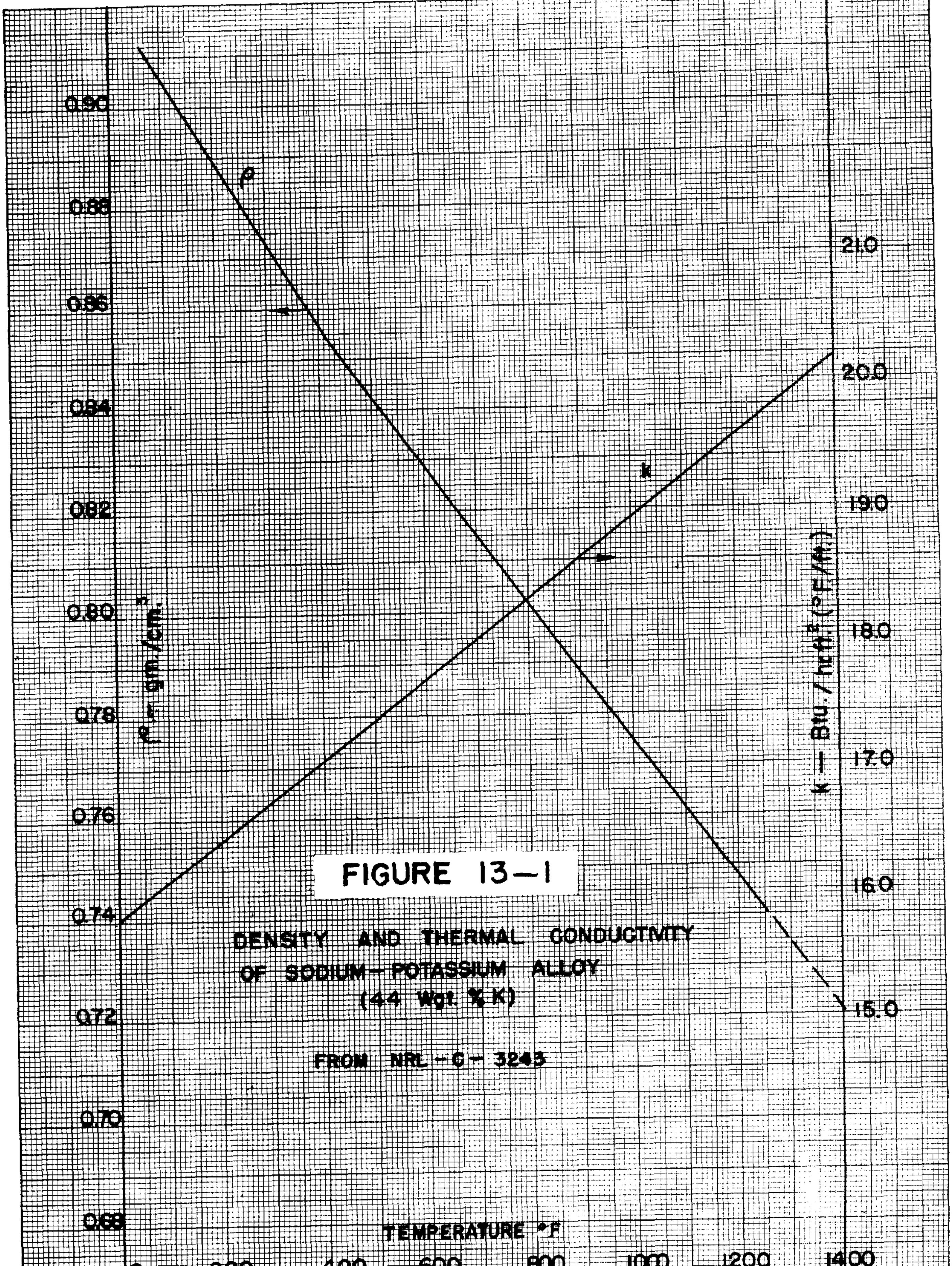


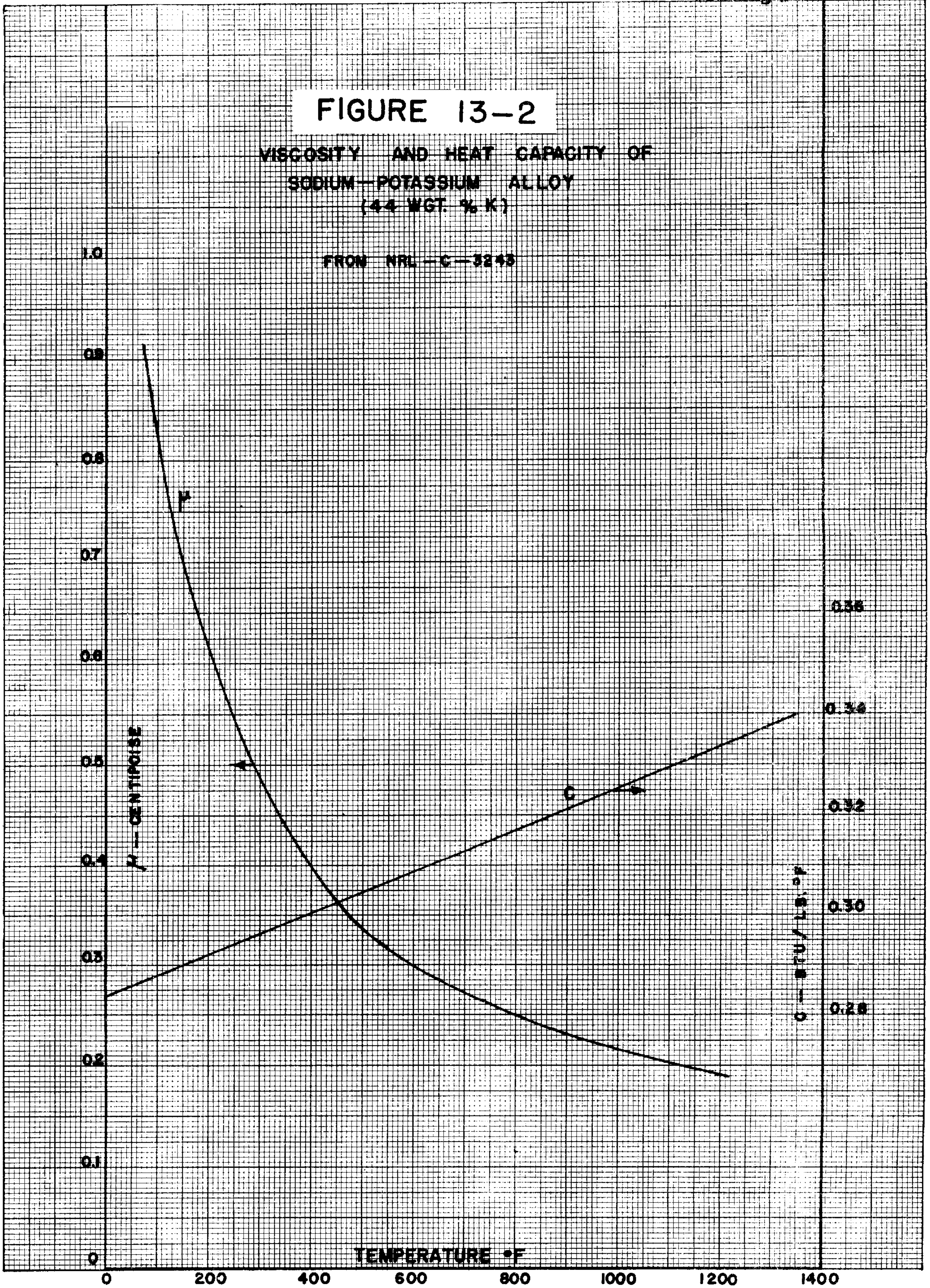
FIGURE 13—1

DENSITY AND THERMAL CONDUCTIVITY OF SODIUM—POTASSIUM ALLOY (44 Wgt. % K)

FROM NRL - C - 3243

[redacted]

Drawing # 6700



[redacted]

Drawing # 6701

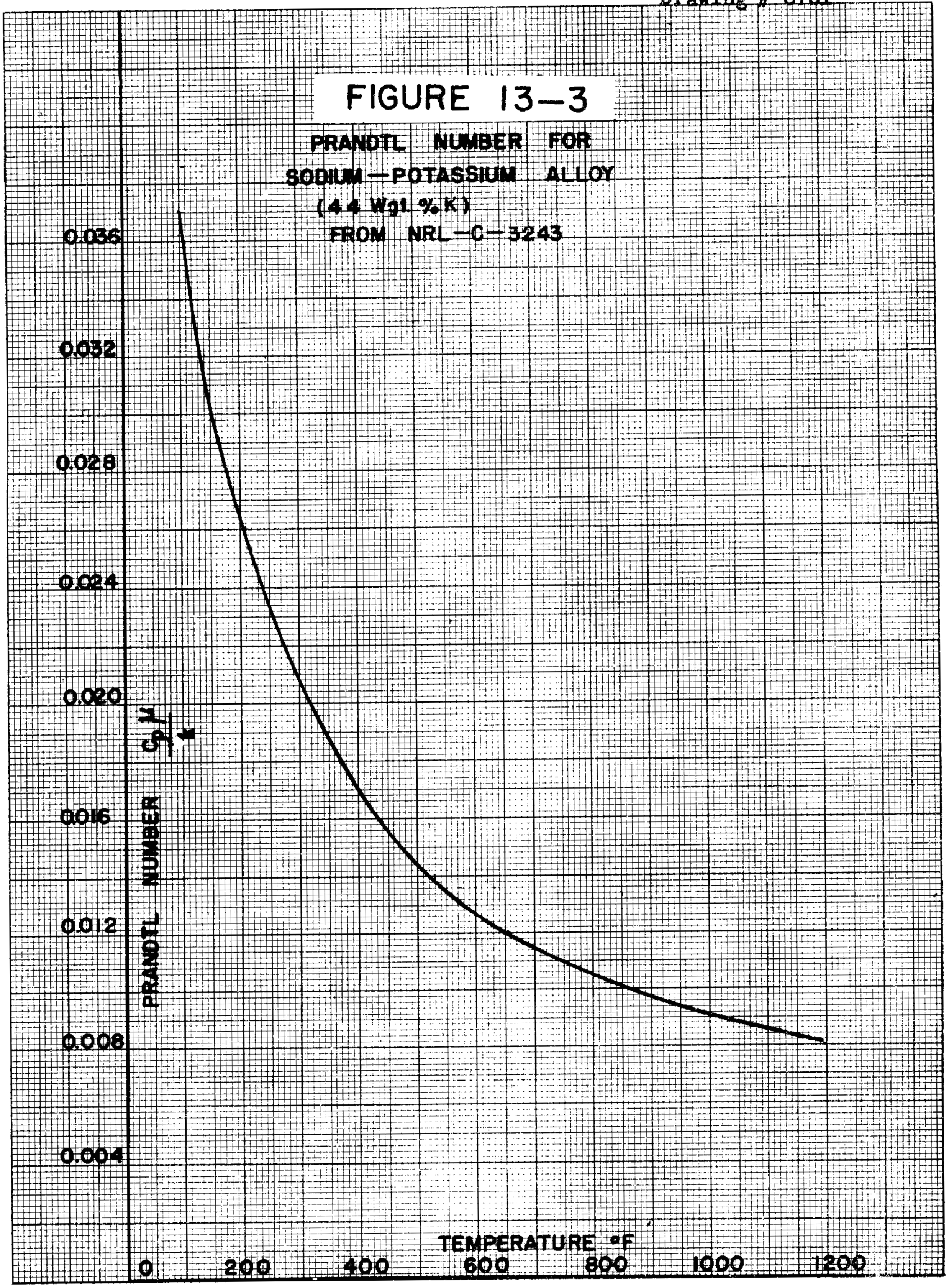


FIGURE 13-3

PRANDTL NUMBER FOR SODIUM—POTASSIUM ALLOY (44 Wgt. % K) FROM NRL—C—3243

[redacted]

PART 13

PHYSICAL PROPERTIES OF LIQUID METALS

The Physical Properties of Mercury

The data shown on the following figure were taken from a thesis by Musser and Page.[2] Their references for the data were a paper by Hall[3] and the I.C.T.[4]

2. Musser, R. J., Page, W. R., Heat Transfer to Mercury, M. S. Thesis, Massachusetts Institute of Technology, 1947.

3. Hall, W. C., Physical Review 53, 1004 (1938)

4. International Critical Tables, McGraw-Hill, New York, 1933.

FIGURE 13-4

VISCOSITY AND HEAT CAPACITY OF MERCURY

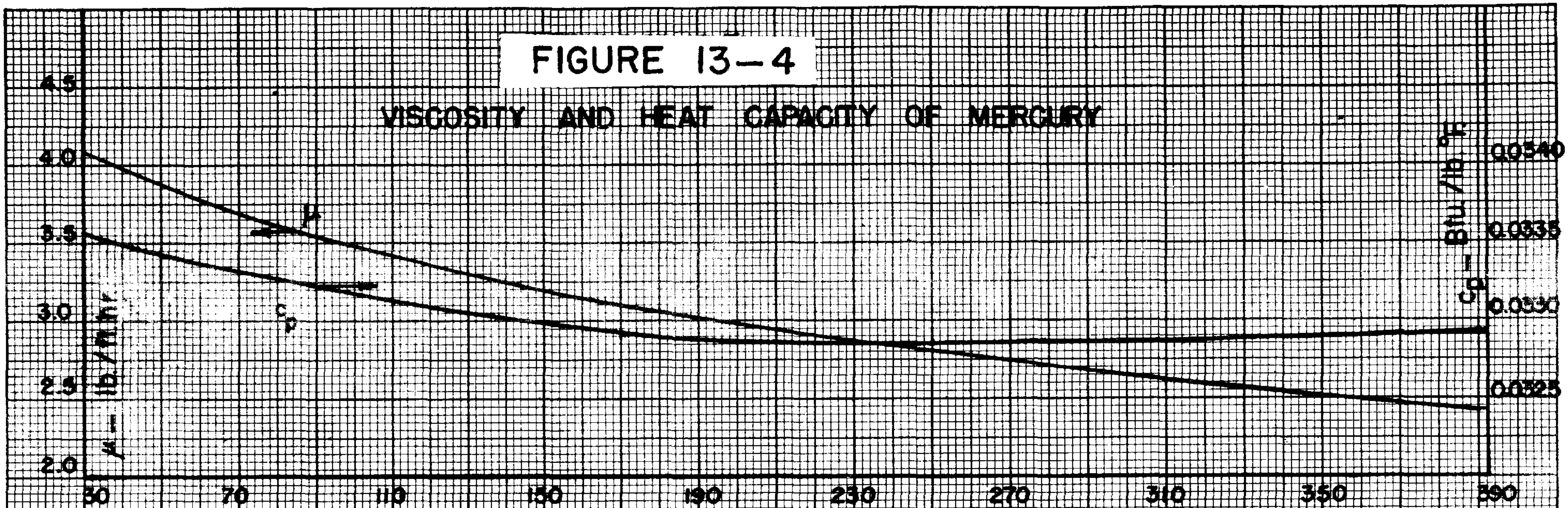
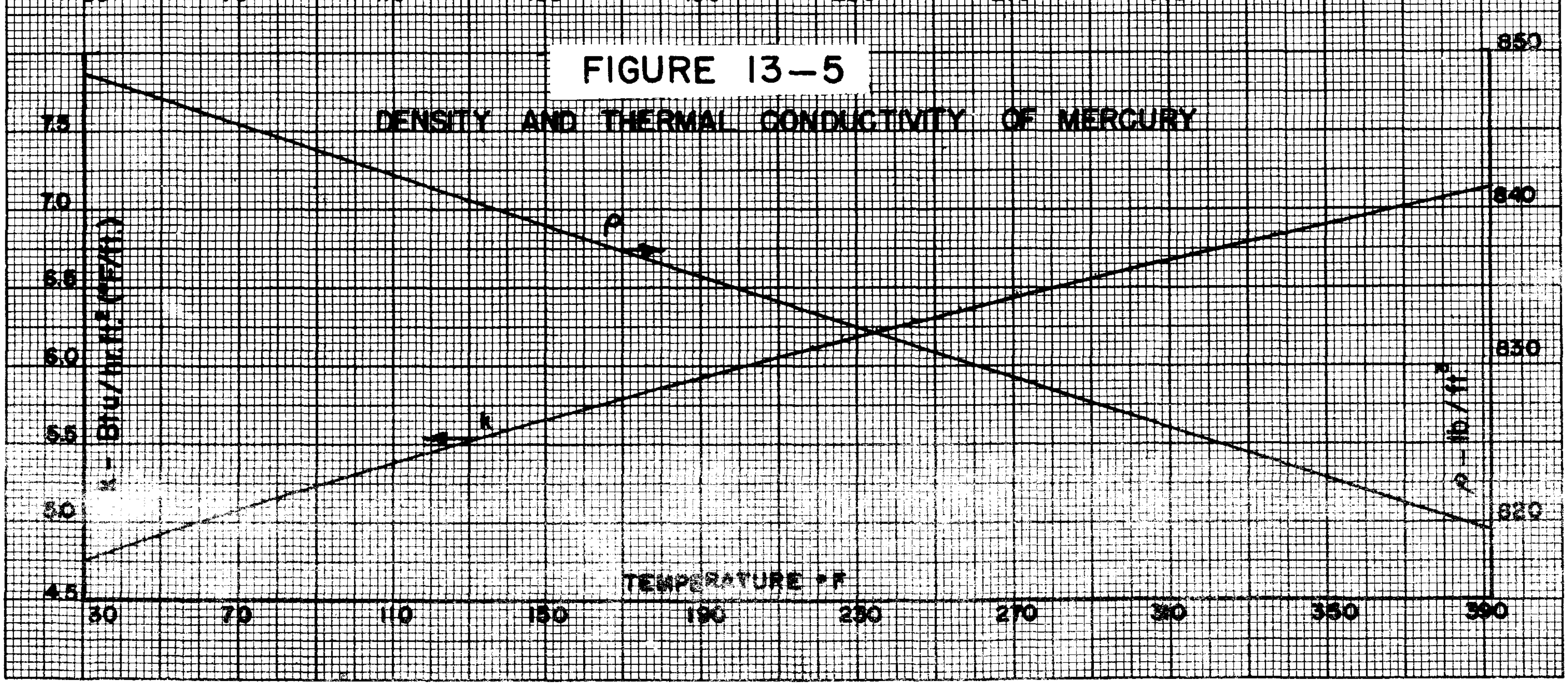


FIGURE 13-5

DENSITY AND THERMAL CONDUCTIVITY OF MERCURY



-9-

Drawing # 6698

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 14

PHYSICAL PROPERTIES OF GASES

W. B. Harrison
October 1, 1948

[redacted]

PART 14

PHYSICAL PROPERTIES OF GASES

The section (a) of this part of the report is devoted to physical properties of gaseous uranium hexafluoride. The properties should be used with little confidence at high temperatures, for large extrapolations have been employed.

Section (b) includes data for carbon dioxide, carbon monoxide, air, oxygen, nitrogen, and hydrogen.

(a) Physical Properties of Gaseous Uranium Hexafluoride

(These Data also appear in the Appendix of Heat Transfer from Hex, by W. B. Harrison.[1])

Remarks on Extrapolation of Physical Properties of Hex

A-753 states that gaseous hex may be considered as a perfect gas between temperature of 18° and 90° C. and pressures of 1.6 and 520 mm. This facilitates calculations for density and lends confidence to the extrapolation of physical properties for higher temperatures. The effect of pressure has been neglected in extrapolating all physical properties except density.

Thermal Conductivity - k

From Figure 14-1, it may be seen that the thermal conductivity data available cover a relatively small range of termperatures, making an accurate extrapolation difficult. In addition to the data plotted, M-466 gives the following equation:

$$k = \frac{1}{4}(9\gamma - 5)\ \frac{\mu\, c_v}{M}\ \text{cal/sec.cm.}\ ^\circ C.$$

$\gamma = c_p/c_v$

$M =$ molecular weight (352.1) in grams

$\mu =$ viscosity in poises (gm./cm.sec.)

$c_v =$ heat capacity at constant volume in cal./gm.-mol.°C.

$c_p = c_v + R$ where $R = 1.99$ cal./gm.-mol. °C.

This equation predicts that $k = 3.7 \times 10^{-3}$ Btu/hr.ft.^{2} (°F./ft.) at 32° F. and $k \cong 5.15 \times 10^{-3}$ Btu/hr.ft.^{2} (°F./ft.) at 212° F. These values are in fair agreement with the data shown but appear to be increasing too rapidly with temperature. Dependence on viscosity and heat capacity evaluations make this equation undesirable since those properties must also be extrapolated.

[redacted]

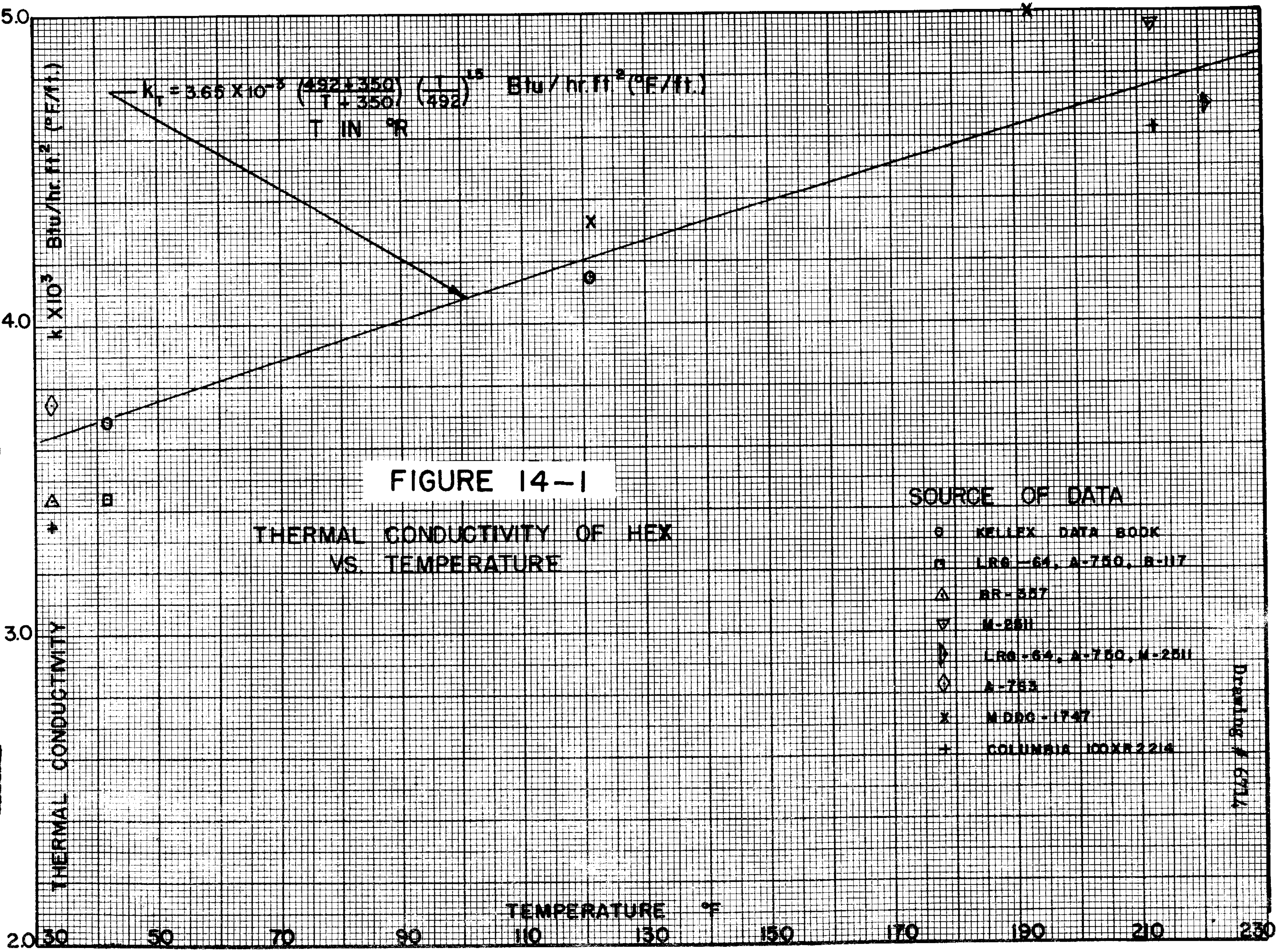


FIGURE 14-1

THERMAL CONDUCTIVITY OF HEX VS. TEMPERATURE

It was decided to use the Sutherland type of equation[2] for extrapolating the thermal conductivity and viscosity. Although LRG-64 suggests a Sutherland constant of 182 for use with Rankine temperatures (101 for use with Kelvin temperatures), a value of 350 was found to give better agreement with the data. If one selects a value of k = 3.65×10^{-3} Btu/hr.ft^2 ($^\circ$F./ft.) at 32°F., Figure 14-3 shows the factor by which this must be multiplied to estimate the thermal conductivity at any temperature. The thermal conductivity equation has the following form:

$$k_T = 3.65 \times 10^{-3} \left(\frac{492 + 350}{T + 350}\right)\left(\frac{T}{492}\right)^{1.5} \text{ Btu/hr.ft}^2 \ (^\circ\text{F./ft.})$$

where T is temperature in $^\circ$R.

Viscosity - μ

As shown in Figure 14-2, data for viscosity are fairly consistent. The following table shows a comparison of several equations for calculating the viscosity of hex:

[redacted]

TABLE 14-I

Source	Equation* - μ in poises	$\mu^{\dagger}(32\ ^\circ F)$	$\mu^{x}(32\ ^\circ F)$	$\mu^{\dagger}(392\ ^\circ F)$	$\mu^{x}(392\ ^\circ F)$
LRG-64	$\mu(0\text{-}200\ ^\circ C) = 1.67(1 + 0.0026t)\times10^{-4}$	1.67×10^{-4}	4.04×10^{-2}	2.54×10^{-4}	6.15×10^{-2}
A-763	$\mu(0\text{-}200\ ^\circ C) = (1.67 + 0.0044t)\times10^{-4}$	1.67×10^{-4}	4.04×10^{-2}	2.55×10^{-4}	6.17×10^{-2}
M-466	$\mu(0\text{-}200\ ^\circ C) = 2.10\times10^{-6}T^{0.779}$	1.67×10^{-4}	4.04×10^{-2}	2.55×10^{-4}	6.17×10^{-2}
	$\mu(85\text{-}165\ ^\circ C) = 2.46\times10^{-6}T^{0.772}$	1.87×10^{-4}	4.52×10^{-2}	2.86×10^{-4}	6.92×10^{-2}
	$\mu(150\text{-}260^\circ C) = 1.78\times10^{-6}T^{0.827}$	1.84×10^{-4}	4.45×10^{-2}	2.90×10^{-4}	7.02×10^{-2}

* $t = {}^\circ C$, $T = {}^\circ K$; $\mu^{\dagger}$ in poises; μ^{x} in lb./ft.hr.

[redacted]

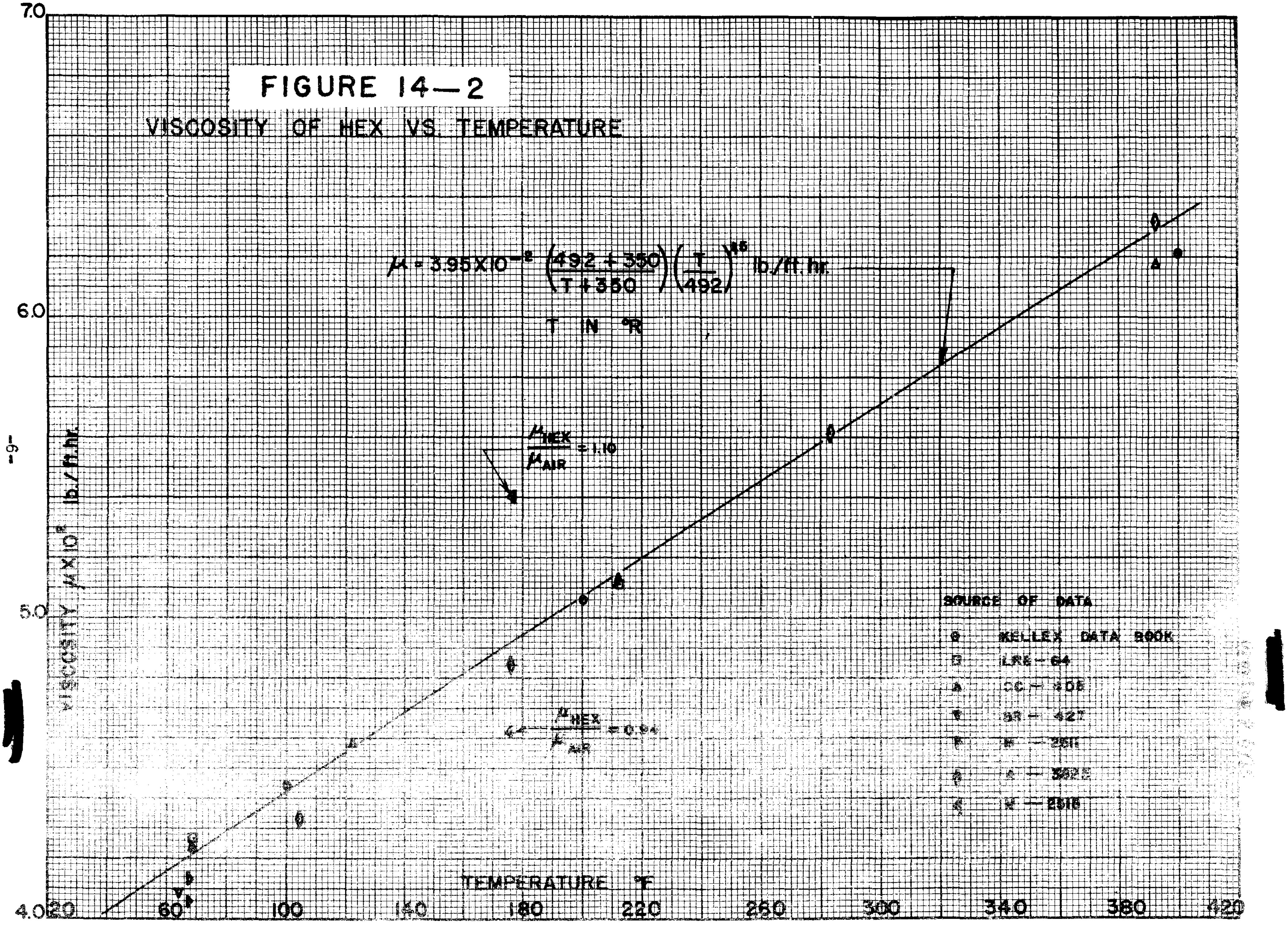
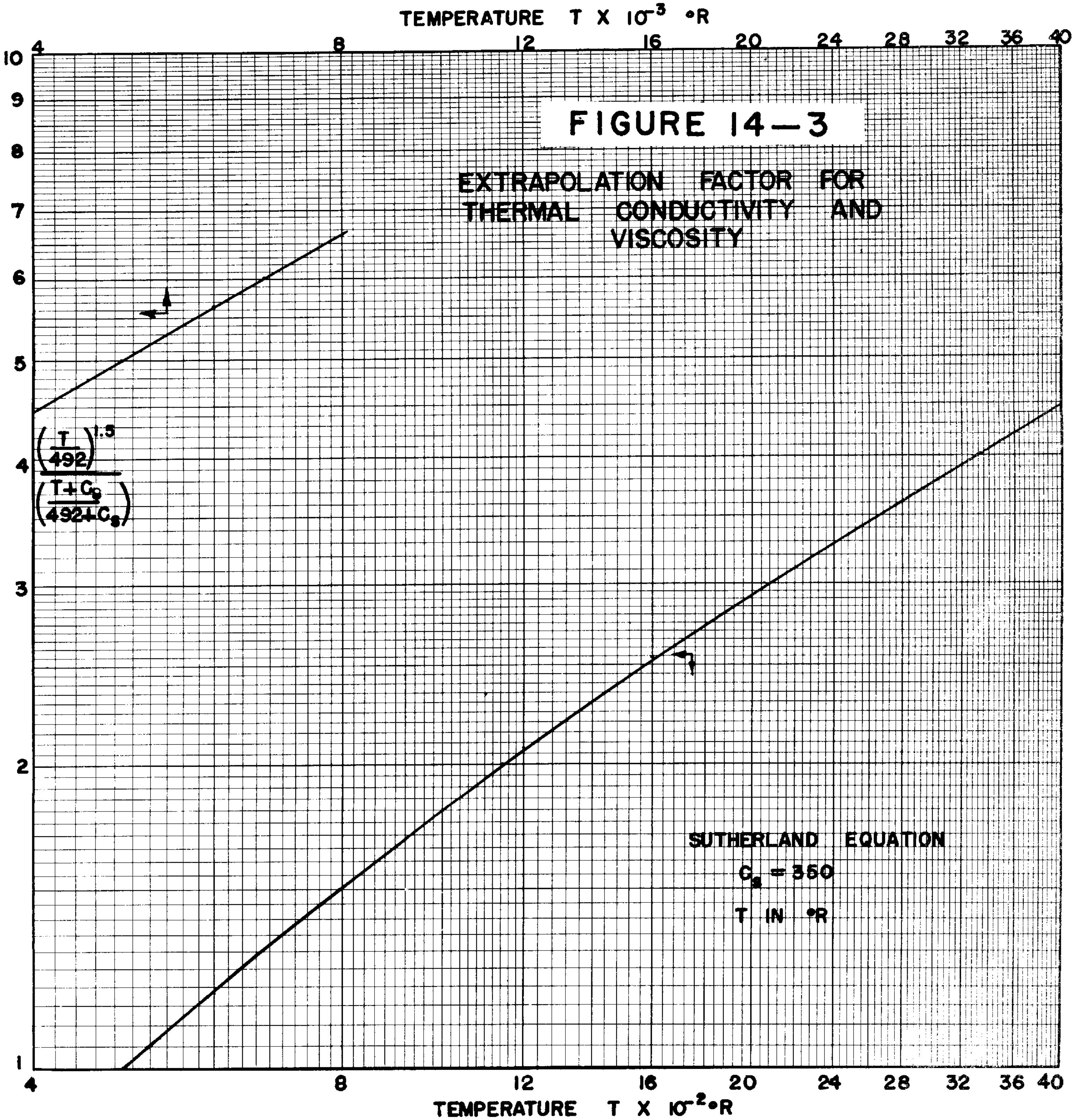


FIGURE 14—2
VISCOSITY OF HEX VS. TEMPERATURE
$\mu = 3.95 \times 10^{-8} \left(\frac{492+350}{T+350}\right)\left(\frac{T}{492}\right)^{1.5}$ lb./ft. hr.
T IN °R
$\frac{\mu_{HEX}}{\mu_{AIR}} = 1.10$
$\frac{\mu_{HEX}}{\mu_{AIR}} = 0.94$
VISCOSITY $\mu \times 10^8$ lb./ft. hr.
TEMPERATURE °F
7.0
6.0
5.0
4.0
20
60
100
140
180
220
260
300
340
380
420
SOURCE OF DATA
KELLEX DATA BOOK
LRE — 64
SR — 427

[redacted]

Drawing # 6716



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If one assumes $\mu = 3.95 \times 10^{-2}$ lb./ft.hr. at 32 °F. and a constant of 350, the Sutherland equation takes the following form:

$$\mu_T = 3.95 \times 10^{-2} \left(\frac{492 + 350}{T + 350} \right) \left(\frac{T}{492} \right)^{1.5} \text{ lb./ft.hr.}$$

Using Figure 14-3 to obtain the factor by which μ(32 °F) must be multiplied to estimate viscosity at any temperature, one finds that μ (392 °F) = 6.25 lb./ft.hr.

Heat Capacity - c_p

Figure 14-4 shows the heat capacity data which were located. For extrapolating these data to higher temperatures, it was decided to use the equation found in M-466.

$$c_p = 32.43 + 0.007936\, T - 3.2068 \times 10^{5}\, T^{-2} \text{ cal./gm.-mol. °C.}$$

T in °K.

or

$$c_p = 0.0921 + 1.252 \times 10^{-5}\, T - \frac{2.95 \times 10^{3}}{T^2} \text{ Btu/lb. °F.}$$

T in °R.

Figure 14-5 is a plot of heat capacity for an extended temperature range.

Density - ρ

In accord with the assumption that hex is an ideal gas (A-753), the density of hex at 10 atm. pressure was calculated and plotted as shown in Figure 14-6.

$$\rho = \frac{\text{mass}}{\text{volume}} = \frac{352.1 \text{ lb.}}{359 \text{ ft.}^3 \times \frac{1 \text{ atm.}}{10 \text{ atm.}} \times \frac{T \text{ °R}}{492 \text{ °R}}} = \frac{4825}{T} \frac{\text{lb.}}{\text{ft.}^3}$$

Prandtl Number - $\frac{c_p \mu}{k}$, Pr

Reynolds Number - $\frac{DV\rho}{\mu}$, Re

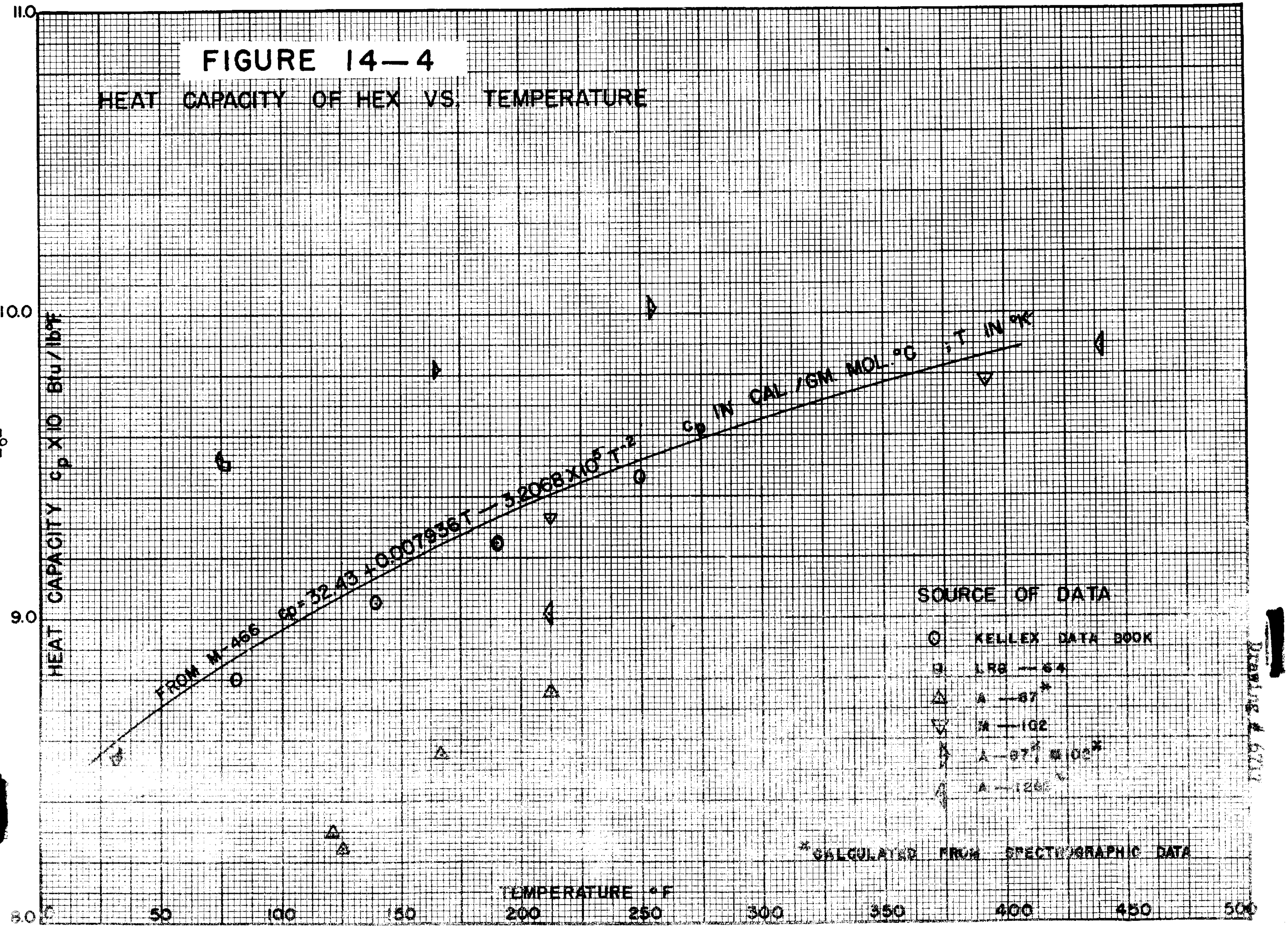
FIGURE 14—4
HEAT CAPACITY OF HEX VS. TEMPERATURE
FROM M-466 cp = 32.43 + 0.007936 T — 3.2068 X 10^5 T^-2 cp IN CAL/GM MOL °C ; T IN °K
HEAT CAPACITY cp X 10^-6 Btu/lb°F
TEMPERATURE °F
11.0
10.0
9.0
8.0
0
50
100
150
200
250
300
350
400
450
500
SOURCE OF DATA
KELLEX DATA BOOK
LRG — 64
A — 87*
M — 102
A — 87*, M 102*
A — 126
* CALCULATED FROM SPECTROGRAPHIC DATA

Drawing #

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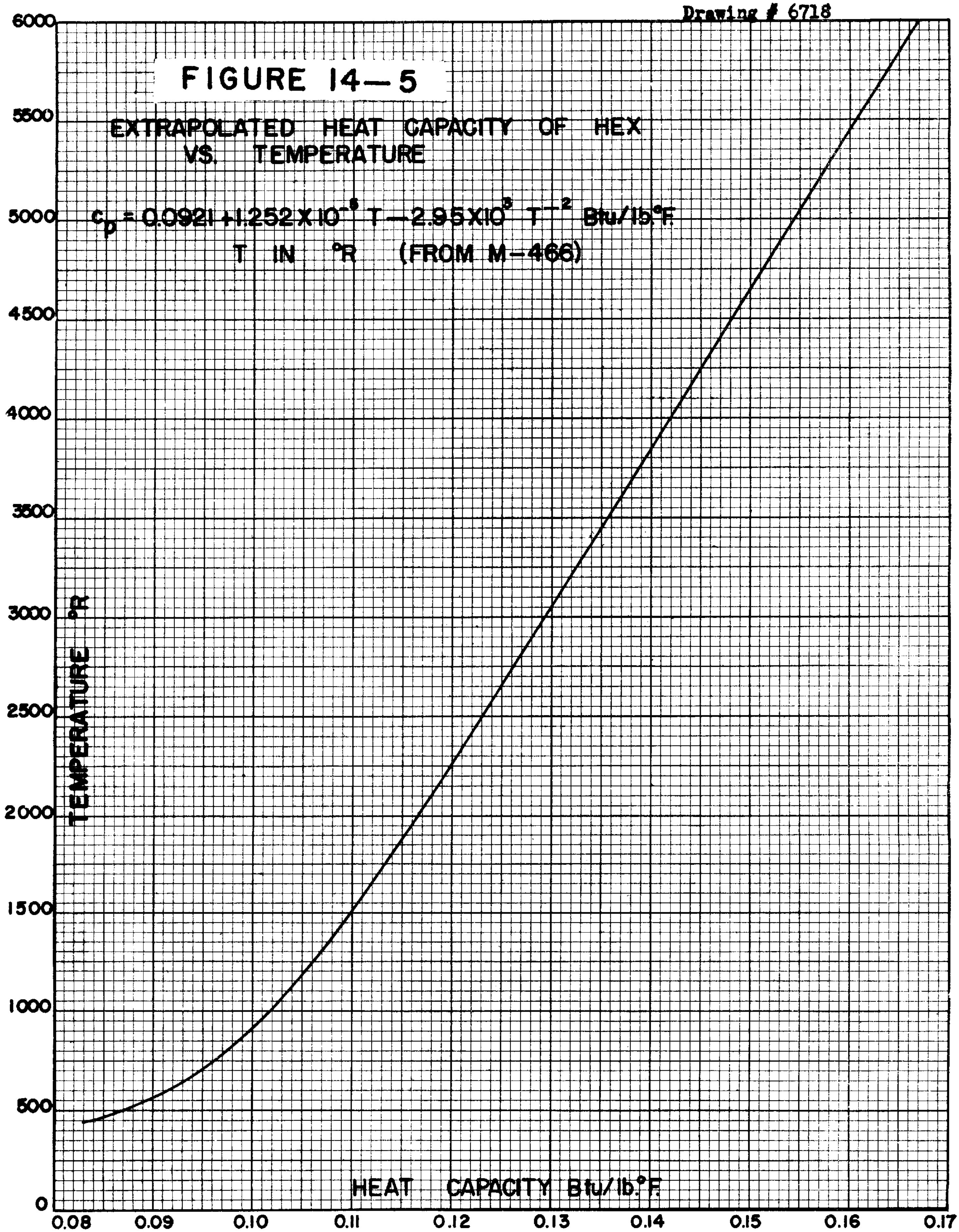

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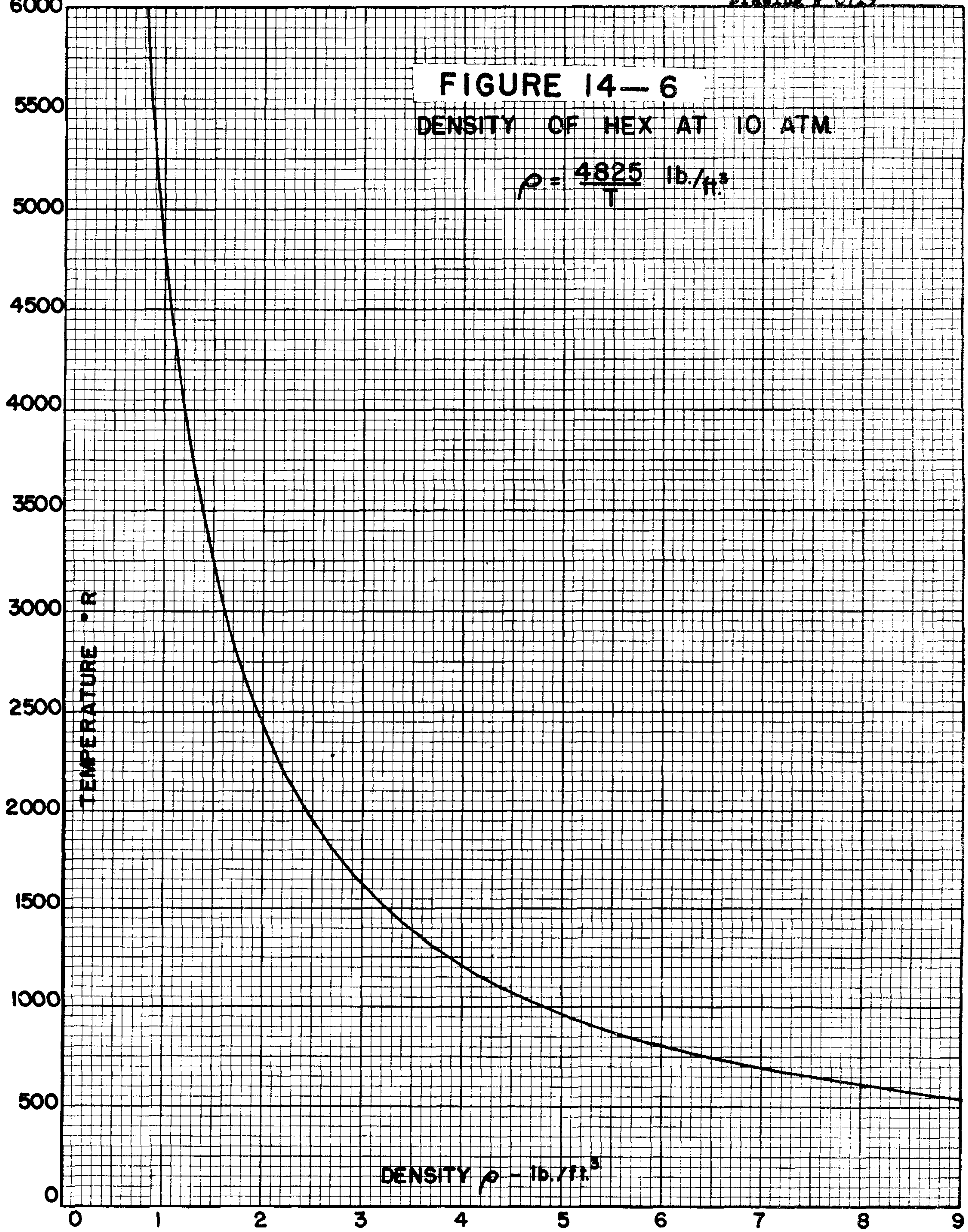

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In order to facilitate estimation of heat transfer by hex in forced convection, a plot of Pr calculated from the extrapolated properties is shown on Figure 14-7. For use in calculations involving Re, ρ/μ is plotted on Figure 14-8.

Emissivity - ϵ

In CNL-39, infrared absorption data were used to estimate emissivity (ϵ) of hex . The values are considered to be very rough, but they may serve as estimates until better data are available. A plot of estimated maximum emissivity is shown on Figure 14-9.

(b) Physical Properties of Air, Hydrogen, Nitrogen, Oxygen, Carbon Dioxide, and Carbon Monoxide.

The data shown on the following figures were taken from a publication by Tribus and Boelter[3]. The properties plotted are heat capacity, thermal conductivity, viscosity and Prandtl number, and they are independent of pressure between 5 mm. of mercury and 2 atmospheres.

Riewe and Rompe[4] presented application of the diffusion theory in taking into account the effect of kinetic, dissociation and ionization energy transport for calculation of thermal conductivity of gases at high temperature.

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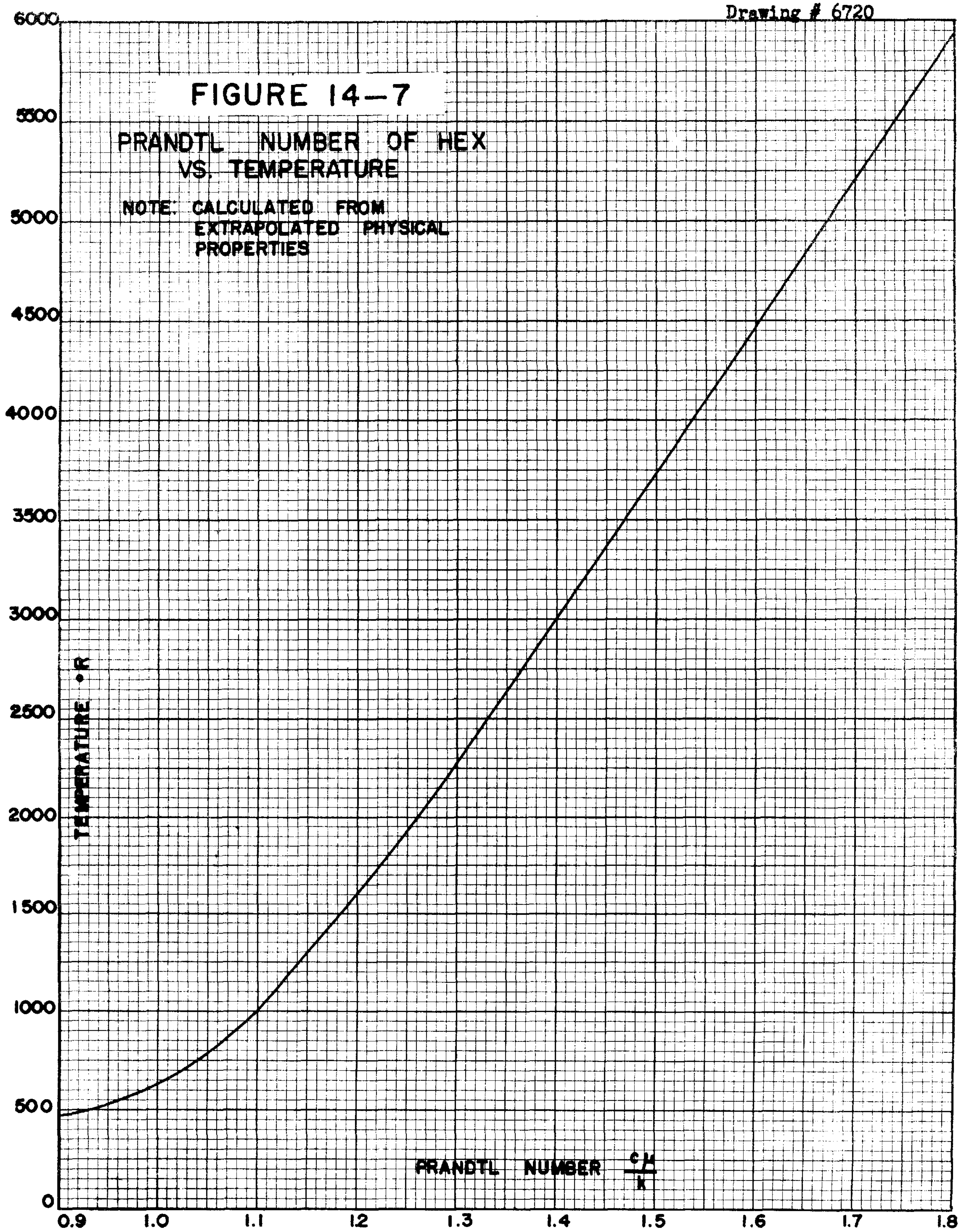
Drawing # 6720



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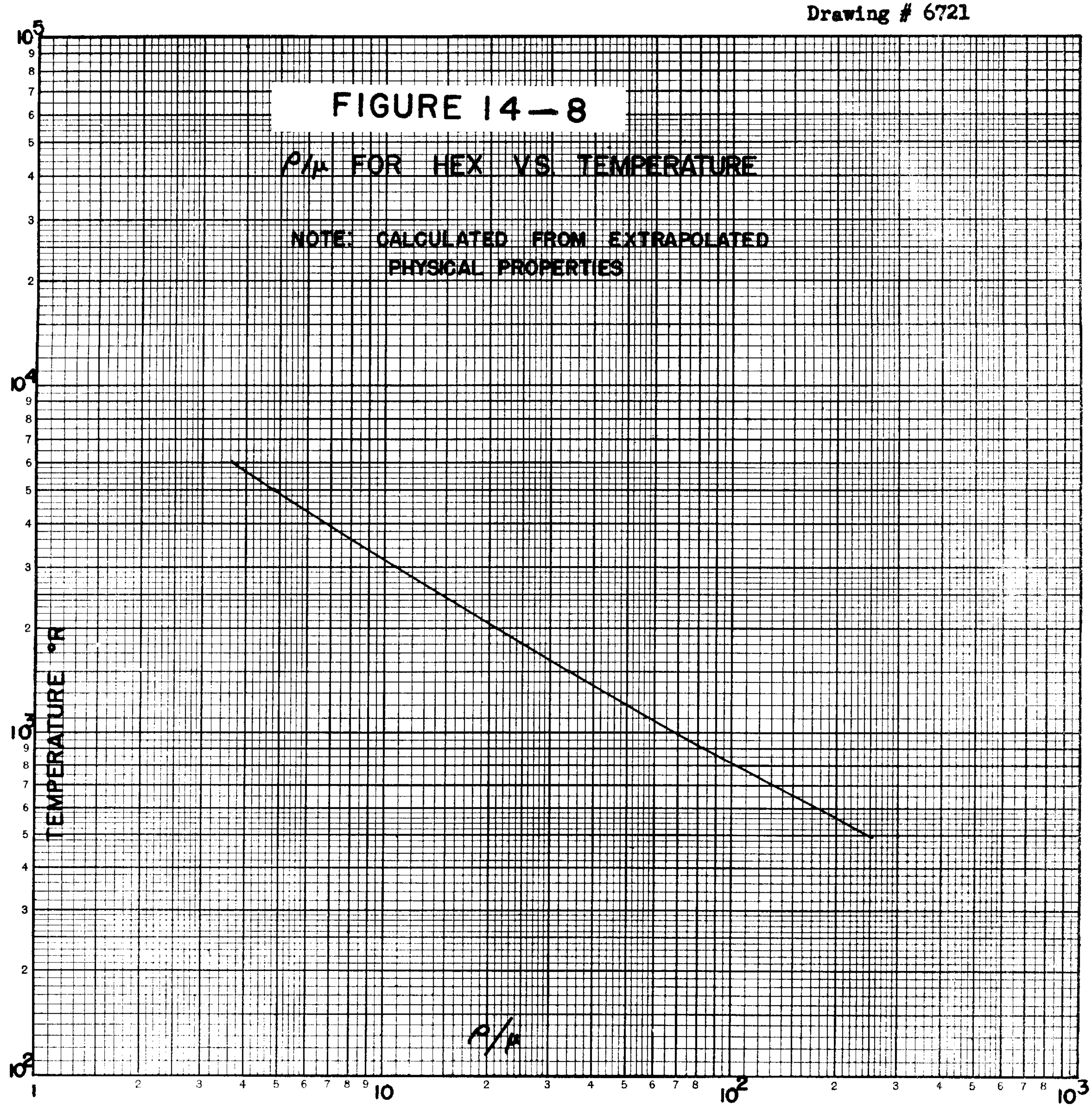

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[redacted]

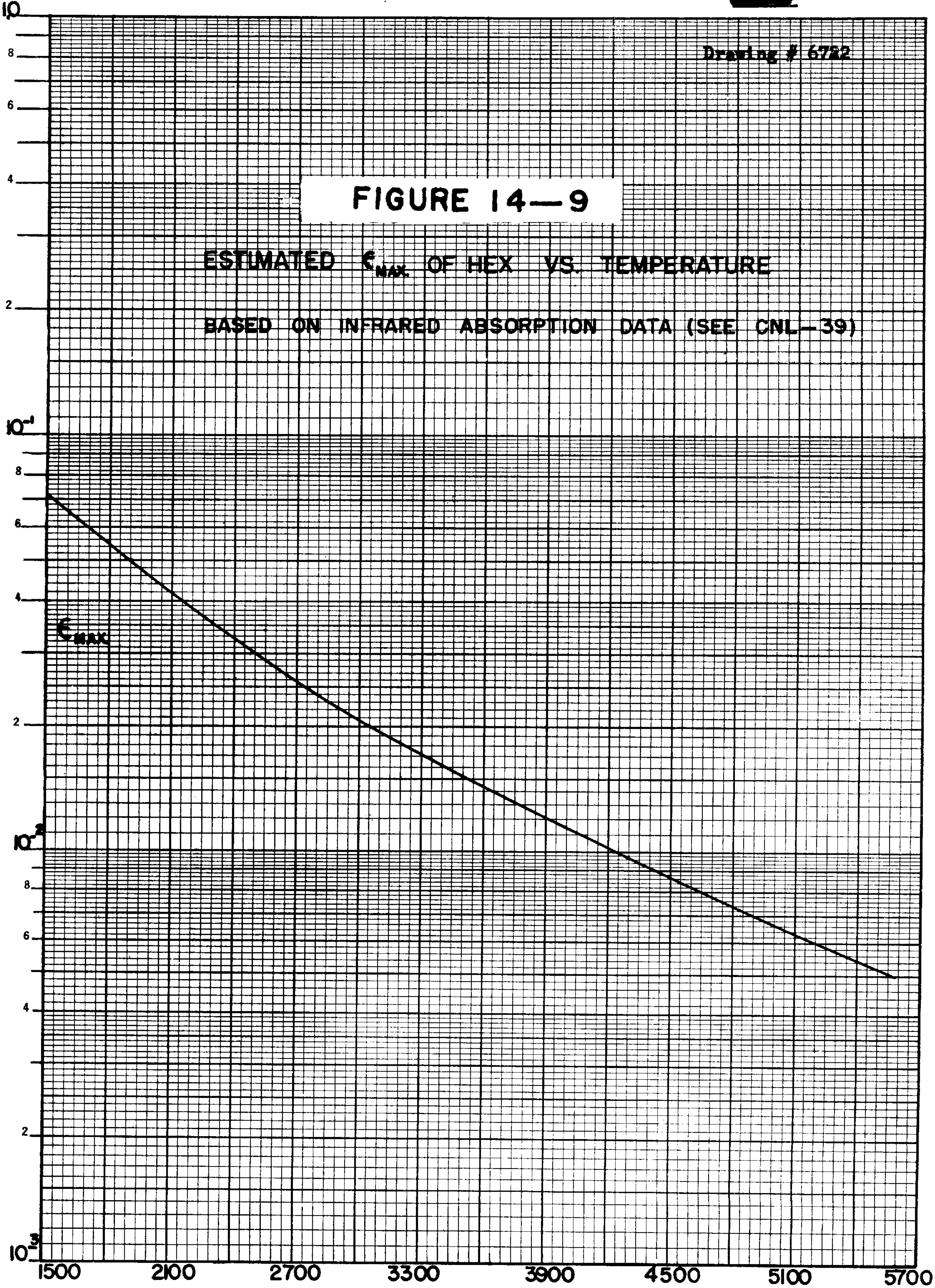

FIGURE 14—9

ESTIMATED $\epsilon_{MAX.}$ OF HEX VS. TEMPERATURE

BASED ON INFRARED ABSORPTION DATA (SEE CNL—39)

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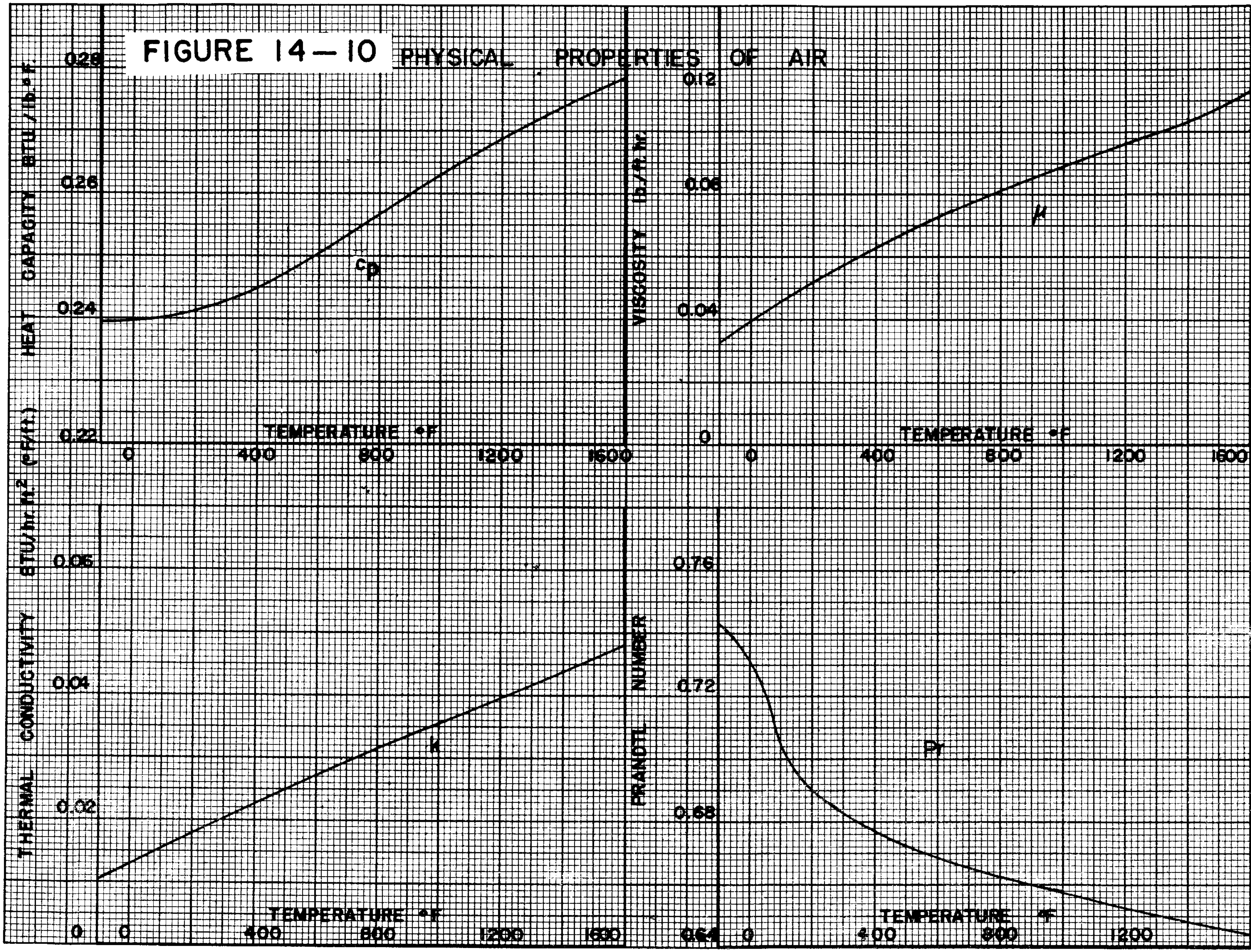

FIGURE 14—10 PHYSICAL PROPERTIES OF AIR

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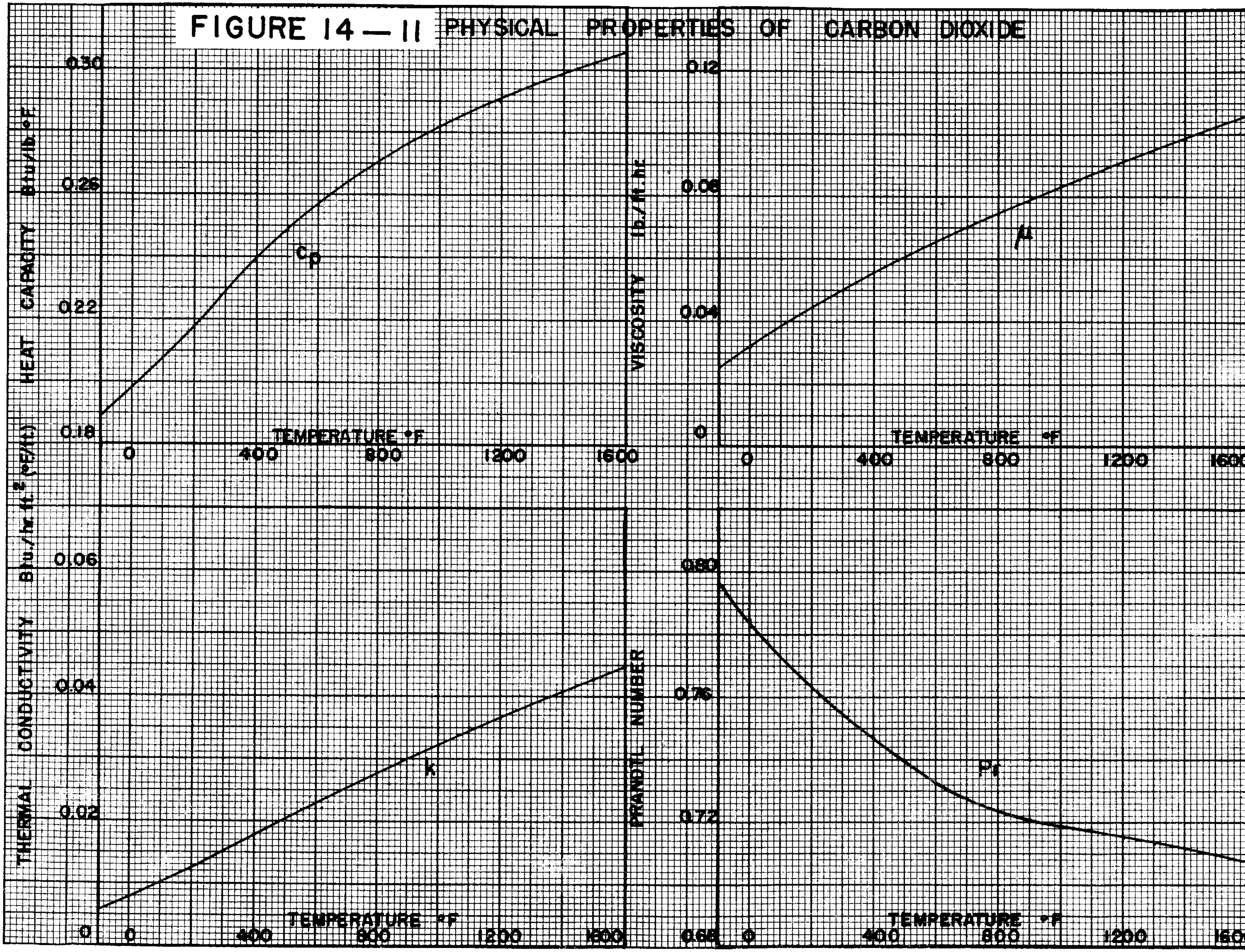
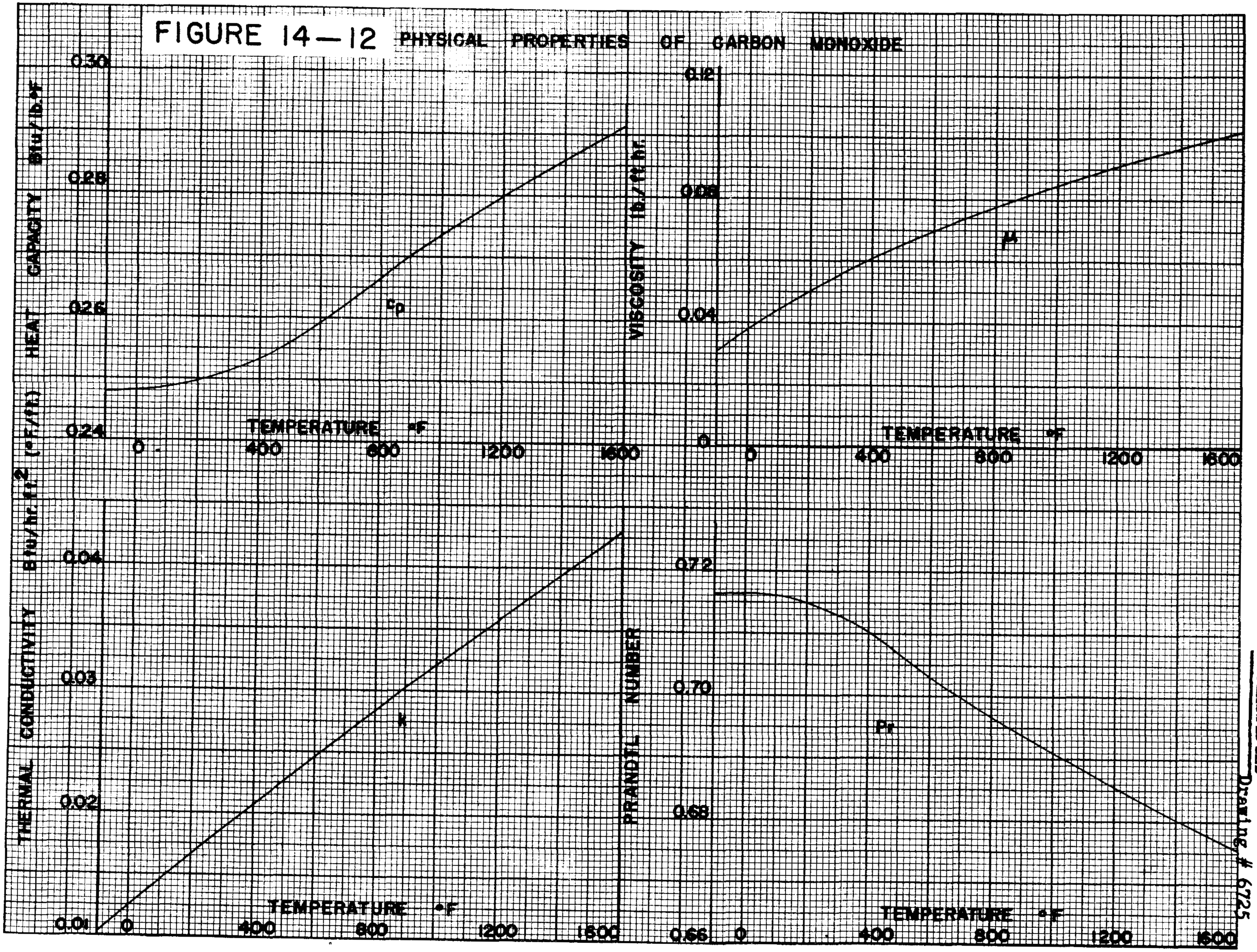


FIGURE 14 — 11 PHYSICAL PROPERTIES OF CARBON DIOXIDE

Drawing # 6724



Drawing # 6725

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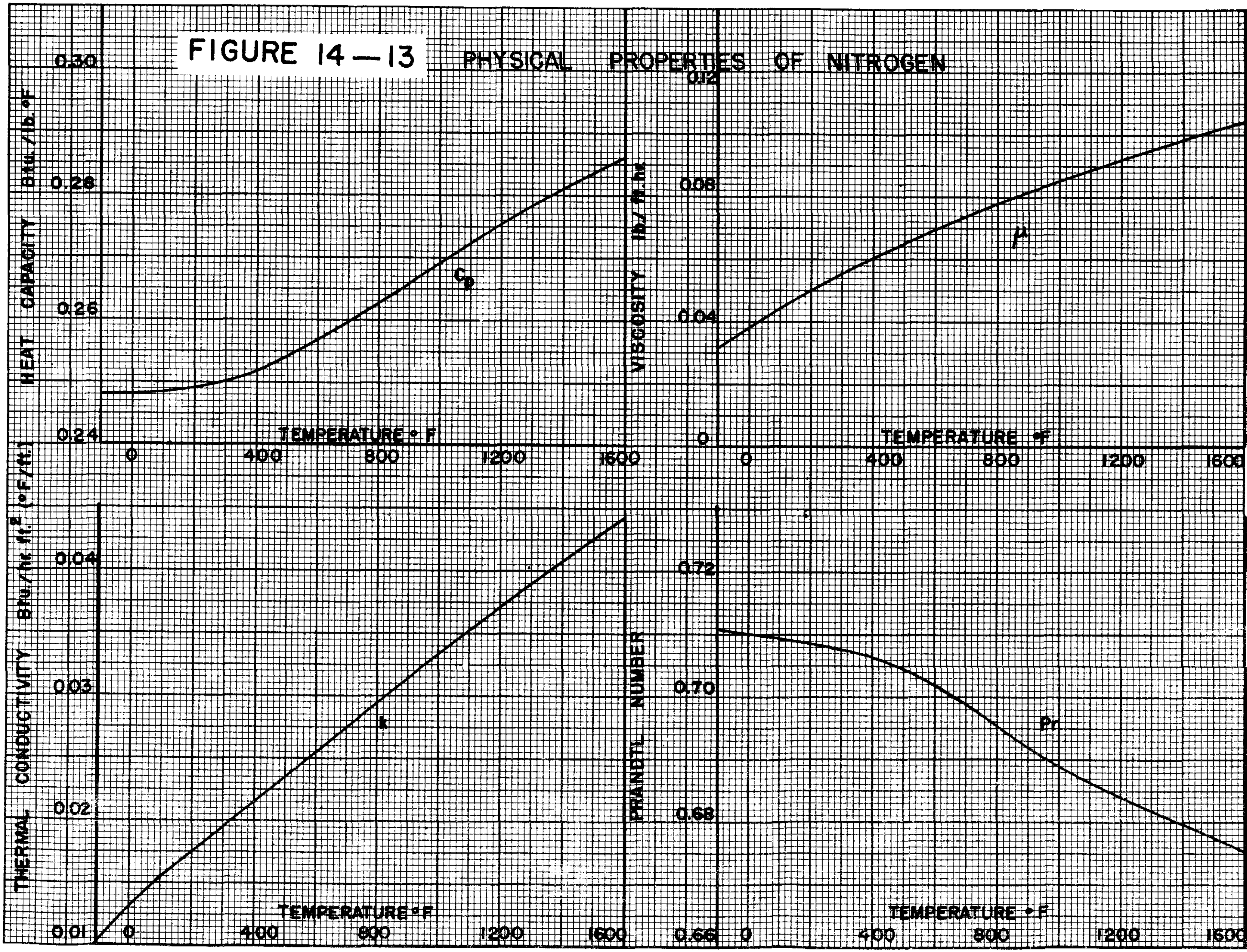

FIGURE 14 — 13 PHYSICAL PROPERTIES OF NITROGEN

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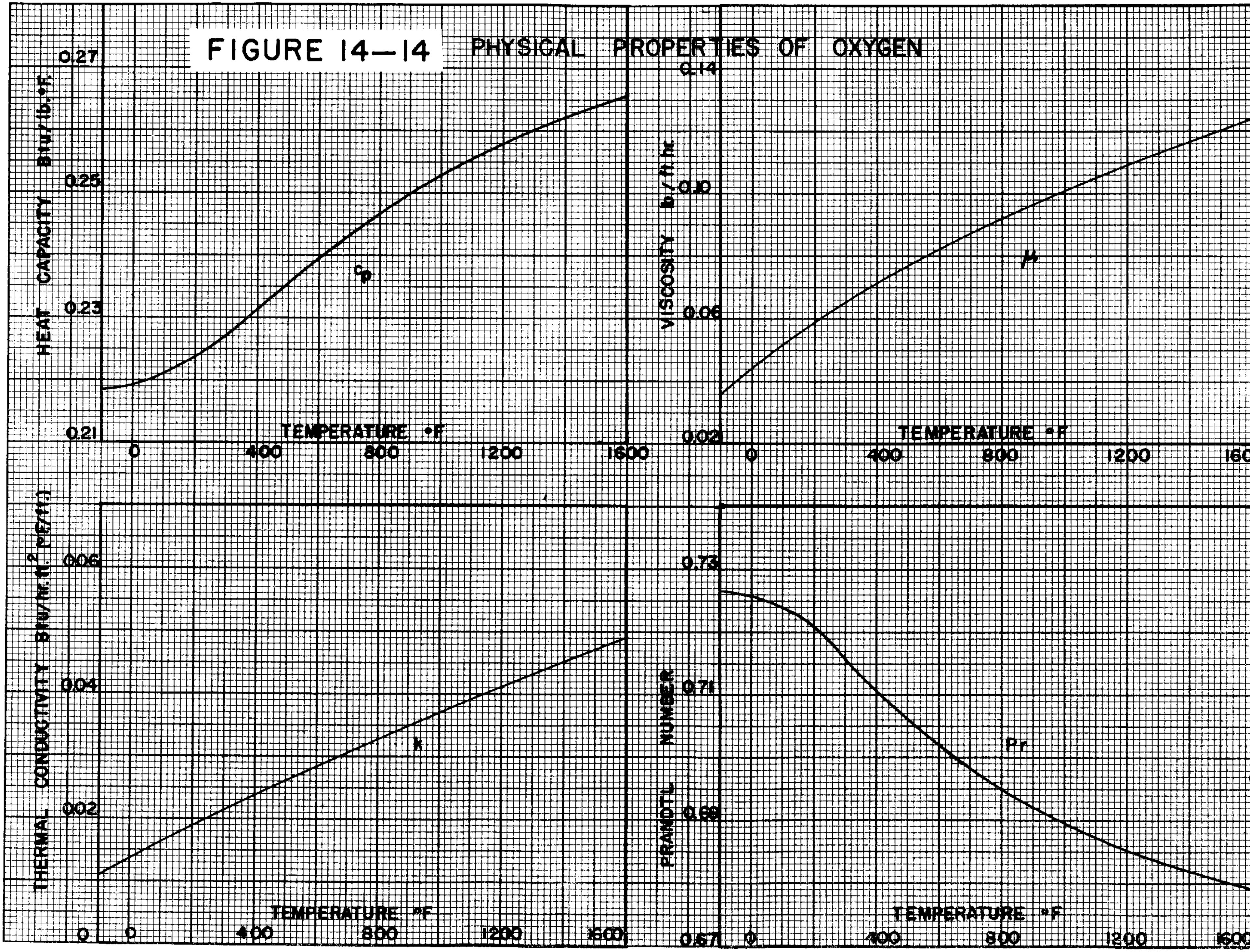
Drawing # 6726



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Drawing # 6727

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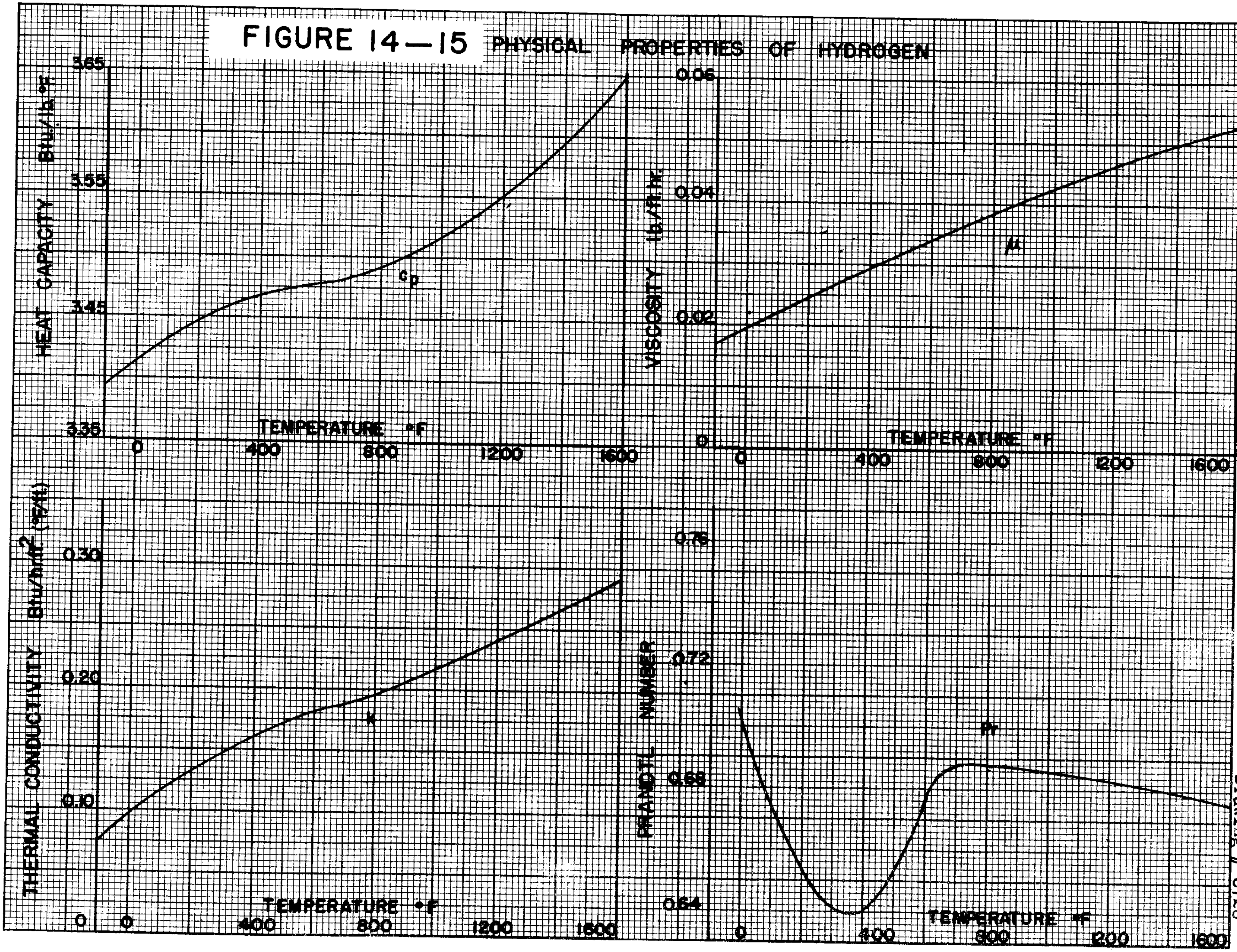

FIGURE 14—15 PHYSICAL PROPERTIES OF HYDROGEN

UNCLASSIFIED

Drawing # 6728

In a recent publication[5], a series of articles was presented on properties of gases as indicated below:

Molecular constant - G. Herzberg

Thermodynamic properties - F. D. Rossini

Vapor pressure, specific volume- PVT data- F. Gratch, F. G. Keyes

Dielectric constant, refractivity- J. G. Miller

Joule-. Thomson effects- H. L. Johnston, David White

Viscosity, Thermal conductivity- G. A. Hawkins

[redacted]

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5. Tr. A.S.M.E. 70, 621-655 (1948)

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ORNL 156-15

HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 15

REFERENCES FOR PHYSICAL PROPERTIES

W. B. Harrison

October 1, 1948

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PART 15

REFERENCES FOR PHYSICAL PROPERTIES

The references noted in this section are in addition to those given in Parts 2 and 3 on graphite blocks and magnesium oxide pellets, and in Parts 12, 13, and 14 on water, sodium-potassium alloy, mercury, gaseous uranium hexafluoride and several other gases. No effort has been made to critically evaluate the data in following reports, but the references are included as a by-product of the search for heat transfer data. The properties which attracted attention to these reports are thermal conductivity and specific heat for solids, and these two properties along with viscosity and density for liquids. One paper on emissivity of uranium is also noted.

<u>Uranium and Uranium compounds</u>

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Gregory, N. W., <u>Thermal Analysis Studies on UI_4 and UBr_3 I, BC-13,</u> August 13, 1946

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Kratz, H. R., Raeth, C. H., The Thermal Conductivity of Uranium, CP-2315, December 18, 1944

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[redacted] ORNL 156-16

HEAT TRANSFER IN MANHATTAN DISTRICT
AND
ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 16

COMPARISON OF COOLANTS

W. B. Harrison
October 1, 1948

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PART 16

COMPARISON OF COOLANTS

It is frequently desirable to compare coolants in order to determine which coolants show the most promise with regard to a given basis.

Off the project, Parsons and Gaffney[1] presented a method for relating heat transfer coefficients and pumping power, and Stoever[2] published a book containing many graphs which facilitate making quick comparisons of coefficients for a number of fluids in each of several specified heat transfer systems.

In two project reports, Burton[3,4] and Davis[3] presented considerations for comparing fluids on the basis of absorption of thermal neutrons per unit of heat capacity.

R. M. Boarts examined several coolants with heat transfer coefficient per unit velocity as a basis, and the results of his comparison are included in section (a) of this part of the report.

Presented in section (b), the method of R. N. Lyon for evaluating pile coolants is based on advantage factors involving pumping power, cross-section for thermal neutrons and permissible temperature range.

(a) HEAT TRANSFER IN TUBES OR SIMILAR CHANNELS

Comparison of Heat Transfer Coefficient for Fluids in Forced Convection Under Similar Conditions

By R. M. Boarts*

Assumptions- Heat is transferred by forced convection without boiling and calculations were made with Dittus and Boelter Equation.

Fluid is flowing in pipes or similar channels.

Values - shown for comparison only- are slide rule computations and should not be used for design calculations. In evaluating a fluid as a heat transfer agent, the specific heat is important because it indicates the quantity of heat that can be removed per unit weight of fluid heated one degree.

FOR TABLE -

Column A - Fluid considered.

Column B - Temperature of fluid, $^{\circ}C$.

Column C - Specific heat, approx., in calories per gram for one degree Centigrade.

Column D - Heat Transfer Coefficient x 10^3 in metric units for unit mass velocity G expressed as unit metric weight of fluid flowing in unit area channel. Shown as H x 10^3 in Column 4, Table I, Report CP-3061[5]. For other flows and channels, multiply by $G^{0.8}/D^{0.2}$.

*Chemical Engineering Department,
University of Tennessee, Knoxville, Tennessee

Column E - Ratio of unit mass velocity coefficient of Column D to the same value for water at 50°C (122° F). This is a comparison with water per unit WEIGHT in the same channel.

Column F - This is Column D corrected for density, $\rho^{0.8}$, to convert to a heat transfer coefficient per unit metric linear VELOCITY in a unit channel. Shown as $H^1 \times 10^2$ in Report CP-3061[5]. Note, however, that gas velocity is normally from 100-600 feet per second whereas liquid water velocity is normally 3- 12 feet per second. Project liquid velocities are much higher. Gas values are shown for one and ten atmospheres pressures.

Column G - Ratio of coefficient for fluid at unit linear VELOCITY (Column F) to value for water at 50° C. (122°F). Note again that gas velocities would normally be some 50 times greater.

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ORNL 156-16

TABLE 16-I

A	B	C	D	E	F	G	Remarks
Bismuth	300°	.0365	4.7	1.24	3.0	7.9	
Bismuth	600°	.0365	5.3	1.40	3.2	8.4	
Lead	350	.0328	3.5	0.92	2.3	6.1	
Lead	700	.0328	3.0	0.79	1.9	5.0	
Mercury	50	.0329	3.3	0.87	2.6	6.8	
Mercury	150	.0329	4.1	1.08	3.2	8.4	
Mercury	350	.0329	5.3	1.40	4.1	10.8	
Sodium	100	0.326	41.0	10.80	3.9	10.2	
Sodium	200	0.326	48.0	12.63	4.5	11.8	
Sodium	350	-	54.0	14.20	4.8	12.6	
Dowtherm	150	0.50	1.0	0.26	0.10	0.26	
Dowtherm	370	0.68	1.5	0.39	0.12	0.32	
Diphenyl	70	0.414	0.7	0.18	0.07	0.18	
Diphenyl	100	0.425	0.8	0.21	0.08	0.21	
Water	30	1.0	3.2	0.85	0.32	0.84	
Water	50	1.0	3.8	1.00	0.38	1.00	Basis
Water	80	1.0	4.7	1.24	0.47	1.24	

[redacted]

[redacted]

TABLE 16-I (Con't)

A	B	C	D	E	F	G	Remarks
Water	100	1.0	5.4	1.42	0.54	1.42	
Water Vapor	100	0.48	1.8	0.47	.0047	0.01	1 Atmos. Pressure
Water Vapor	100	0.48	1.8	0.47	.0296	0.08	10 Atmos. Pressure
Water Vapor	300	0.48	2.1	0.55	.0039	0.01	1 Atmos. Pressure
Water Vapor	300	0.48	2.1	0.55	.0244	0.06	10 Atmos. Pressure
Water Vapor	500	0.49	2.6	0.68	.0038	0.01	1 Atmos. Pressure
Water Vapor	500	0.49	2.6	0.68	.0238	0.06	10 Atmos. Pressure
Air	100	0.240	1.3	0.34	.0040	0.01	1 Atmos. Pressure
Air	100	0.243	1.3	0.34	.0255	0.07	10 Atmos. Pressure
Air	300	0.243	1.4	0.37	.0035	0.01	1 Atmos. Pressure
Air	300	0.243	1.4	0.37	.0221	0.06	10 Atmos. Pressure
Air	540	0.245	1.7	0.45	.0028	0.01	1 Atmos. Pressure
Air	540	0.245	1.7	0.45	.0178	0.05	10 Atmos. Pressure
Helium	0	1.24	6.5	1.70	.0065	0.02	1 Atmos. Pressure
Helium	0	1.24	6.5	1.70	.0408	0.11	10 Atmos. Pressure
Helium	200	1.24	7.0	1.84	.0045	0.01	1 Atmos. Pressure
Helium	200	1.24	7.0	1.84	.0283	0.07	10 Atmos. Pressure

[redacted]

TABLE 16-I (Con't)

A	B	C	D	E	F	G	Remarks
Helium	600	1.24	7.6	2.00	.0030	0.01	1 Atmos. Pressure
Helium	600	1.24	7.6	2.00	.0190	0.05	10 Atmos. Pressure
Helium	1000	1.24	8.0	2.10	.0024	0.01	1 Atmos. Pressure
Helium	1000	1.24	8.0	2.10	.0147	0.04	10 Atmos. Pressure

(b) A Comparison of Coolants: By R. N. Lyon*

(Taken from Clinton Laboratories Pile Technology Lecture 47)

Assume:

(1) A pile of given power, W.

(2) Given overall dimensions (this is permissible with most liquid coolants)

(3) Given equivalent diameter of channels, D

(4) Given reactivity available for coolant.(Neglect such effects as different neutron absorption by different channel walls and neutron blocking or moderating by different materials.)

(5) Constant pressure drop in velocity heads. (Or the pressure drop in psi is proportional to the velocity head. $\frac{V^2}{2g}$)

(6) Constant pump efficiency.

Power = volume x pressure drop

or $$P \alpha \ A V \rho \frac{V^2}{2g}$$

or $$P \alpha \ A \ V^3 \qquad \text{(Eq. 16.01)}$$

using the nomenclature in Part 18 of this report

$$A \alpha \ \frac{M}{\Sigma \sigma \rho}$$

Substituting for A in Equation 16.01,

$$P \alpha \ \frac{M}{\Sigma \sigma} V^3 \qquad \text{(Eq. 16.02)}$$

Two cases may be considered. One case is that in which the temperature rise of the liquid is a determining factor. This is the case in most natural uranium graphite moderated piles, where considerable surface area is available compared with the cross sectional area of the long narrow channels.

*Oak Ridge National Laboratory, Oak Ridge, Tennessee

The other case is that in which the temperature drop from the channel wall to the bulk of the coolant is the determining factor. This will usually be the case in enriched piles with heavy water for a moderator, because of the very high heat fluxes involved in relatively short channels.

In the first case - where bulk temperature rise of coolant is important:

$$\Delta T_L = \frac{W}{V\Lambda \quad c\rho}$$

or substituting for Λ and cancelling out ,

$$\Delta T_L = \frac{W \; \Sigma\sigma}{M \quad c}$$

Solving for V and cubing,

$$V = \frac{W^3 \quad (\Sigma\sigma)^3}{\Delta T_L^3 \quad c^3 \quad M^3}$$

Substituting in Equation 16.02,

$$P \alpha \quad \frac{W^3 \quad (\Sigma\sigma)^2}{\Delta T_L^3 \quad c^3 \; M^2}$$

or, if $L = (\Delta T_L)^3 \quad \frac{c^3 \quad M^2}{(\Sigma\sigma)^2}$ (Eq. 16.03)

$$P \alpha \quad \frac{W^3}{L} \qquad \text{(Eq. 16.04)}$$

Note that in L, c is cubed while M and $\Sigma\sigma$ are squared. This is to be compared with the isocaloric cross section in which they all occur to the same power.

In the second case, when the temperature drop from the wall to the liquid is important,

$$\Delta T_F = \frac{W}{h\ a} \qquad \text{(Eq. 16.05)}$$

$$h \ \alpha \ \frac{k}{D} \left(\frac{DV\rho}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{0.4}$$

$$h \ \alpha \ \frac{k^{0.6}}{D^{0.2}} \ \frac{V^{0.8} \ \rho^{0.8}}{\mu^{0.4}} \ c^{0.4}$$

and

$$a \ \alpha \ \frac{A}{D} \ \alpha \ \frac{M}{D \rho \Sigma \sigma}$$

Substituting for h and a in Equation 16.05,

$$\Delta T_F \ \alpha \ W \ \frac{D^{0.2} \ \mu^{0.4}}{k^{0.6} \ V^{0.8} \ \rho^{0.8} \ c^{0.4}} \cdot \frac{D \rho \Sigma \sigma}{M}$$

$$\Delta T_F \ \alpha \ W \ \frac{D^{1.2} \ \mu^{0.4} \ \rho^{0.2} \ \Sigma \sigma}{k^{0.6} \ v^{0.8} \ c^{0.4} \ M}$$

or, solving for V^3

$$V^3 \alpha \ \frac{W^{3.75} \ D^{4.5} \ \mu^{1.5} \ \rho^{0.75} \ (\Sigma \sigma)^{3.75}}{k^{2.25} \ c^{1.5} \ M^{3.75} \ \Delta T_F^{3.75}}$$

Substituting in Equation 16.02,

$$P \ \alpha \ W^{3.75} \cdot D^{4.5} \ \frac{\rho^{0.75} \ \mu^{1.5} \ (\Sigma \sigma)^{2.75}}{c^{1.5} \ k^{2.25} \ M^{2.75} \ (\Delta T_F)^{3.75}}$$

[redacted]

$$\text{or if } F = \frac{c^{1.5}}{\rho^{0.75}} \; \frac{k^{2.25} \, M^{2.75} (\Delta T_F)^{3.75}}{\mu^{1.5} \, (\Sigma\sigma)^{2.75}} \qquad \text{(Eq. 16.06)}$$

$$P \propto \frac{W^{3.75} \; D^{4.5}}{F} \qquad \text{(Eq. 16.07)}$$

In Table 16-II are listed values for L and F for six liquid coolants. The values for the physical properties are taken principally from Bentley, Brown and Schlegel[5]. Values of certain properties of sodium and sodium-potassium are taken from the quarterly progress reports on sodium-potassium alloy issued by NRL[6,7]. Values for density and viscosity of liquid lithium are guesses based on the density of solid lithium and the viscosities of sodium and potassium.

Cross sections of most of the materials are taken from the Project Handbook. The value for lithium seven is a conservative average of the estimate 0.03 to 0.001 attributed by Young to Way.

ΔT_L and ΔT_F are the difference between the melting point and 600° C except in the case of water where the range found practicable on the project is given.

L and F are advantage factors based only on pumping power, cross section for thermal neutrons and permissible temperature range.

Neutron blocking, neutron moderation, handling difficulties due to chemical or nuclear activity, or due to freezing, and corrosion problems other than those of water have not been included in this evaluation.

TABLE 16-II

Mat'l	M	$\Sigma\sigma$ barnes	k cal./cm. °C. sec.	ρ gm./cm³	c cal./gm. °C.	μ C'poise	$\Delta T_F + \Delta T_L$ °C	L	F
H_2O	18	0.62	0.001	1	1	0.5	50	1.05×10^{8}	3.3×10^{4}
Pb	207	0.18	0.045	10.5	0.033	2.0	300	1.2×10^{9}	1.6×10^{10}
NaK 50 wt%	31	1.74	0.066	0.80	0.25	0.18	600	1.1×10^{9}	3.1×10^{11}
Na	23	0.45	0.19	0.86	0.33	0.25	500	1.2×10^{10}	2.9×10^{13}
Bi	209	0.064	0.038	9.8	0.037	1.25	300	3.2×10^{11}	2.3×10^{14}
Li_7	7	0.01	0.06	0.5	1.7	0.2	400	1.5×10^{14}	2.8×10^{15}

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 17

CALCULATION AIDS

W. B. Harrison
October 1, 1948

PART 17

CALCULATION AIDS

The following section consists of a Table of Equivalents and graphical aids for temperature conversion and heat transfer calculations.

TABLE 17-I

TABLE OF EQUIVALENTS

Complied by W. B. Harrison and R. N. Lyon

TIME - θ (Based on 24 hour day, 30 day month, 12 month year)

second	minute	hour	day	week	month	year
1						
60	1					
3.6×10^{3}	60	1				
8.64×10^{4}	1.44×10^{3}	24	1			
6.05×10^{5}	1.01×10^{4}	168	7	1		
2.59×10^{6}	4.32×10^{4}	720	30	4.29	1	
3.11×10^{7}	5.18×10^{5}	8.64×10^{3}	360	51.4	12	1

POWER - Q/θ or q

Hp.	Kw.*	ft.lb/sec.	Pcu/sec.	Btu/hr.	cal./sec.	mev./sec.
1	0.7457	550	0.393	2547	178.26	4.65×10^{15}
1.341	1	737.6	0.527	3415	239	6.24×10^{15}
1.818×10^{-3}	1.356×10^{-3}	1	7.15×10^{-4}	4.63	0.324	8.46×10^{12}
2.546	1.899	1400.6	1	6480	453.6	1.18×10^{16}
3.93×10^{-4}	2.93×10^{-4}	0.216	1.54×10^{-4}	1	0.070	1.82×10^{12}
5.61×10^{-3}	4.18×10^{-3}	3.088	2.21×10^{-3}	14.29	1	2.61×10^{13}
2.15×10^{-16}	1.60×10^{-16}	1.18×10^{-13}	8.44×10^{-17}	5.47×10^{-13}	3.83×10^{-14}	1

* 1 Kw. = 1000 watts = 1000 joules/sec. = 10^{10} ergs/sec.

Table of Equivalents
(Table 17-1 Con't)

ENERGY - Q

Kw. hr.	Pcu	Btu	ft. lb.	cal.	mev.
1	1896	3415	2.66×10^{6}	8.60×10^{5}	2.24×10^{19}
5.27×10^{-4}	1	1.8	1400.6	453.6	1.18×10^{16}
2.93×10^{-4}	0.5556	1	778.1	252	6.58×10^{15}
3.77×10^{-7}	7.14×10^{-4}	1.29×10^{-3}	1	0.324	8.46×10^{12}
1.16×10^{-6}	2.21×10^{-3}	3.97×10^{-3}	3.088	1	2.61×10^{13}
4.45×10^{-20}	8.44×10^{-17}	1.52×10^{-16}	1.18×10^{-13}	3.83×10^{-14}	1

THERMAL CONDUCTIVITY- k

$\frac{\text{cal.}}{\text{sec.cm.}^2\ (^\circ\text{C/cm.})}$	$\frac{\text{Btu}}{\text{hr. ft.}^2\ (^\circ\text{F/ft.})}$	$\frac{\text{Btu}}{\text{hr. ft.}^2\ (^\circ\text{F/in.})}$	$\frac{\text{watts}}{\text{cm.}^2\ (^\circ\text{C/cm.})}$	$\frac{\text{Pcu}}{\text{hr. ft.}^2\ (^\circ\text{F/ft.})}$
1	241.9	2903	4.183	134.4
4.13×10^{-3}	1	12	0.0173	0.5556
3.45×10^{-4}	0.0833	1	1.440×10^{-3}	0.0464
0.239	57.8	694	1	32.12
7.44×10^{-3}	1.8	21.6	3.12×10^{-3}	1

Table of Equivalents
(Table 17-I Con't)

HEAT FLUX - q/A

watts/in.^{2}	watts/cm.^{2}	cal./sec.cm.^{2}	Btu/hr.ft.^{2}	Pcu/hr.ft^2
1	0.155	0.0371	492	273.2
6.452	1	0.239	3173	1763
26.99	4.183	1	13272	7374
2.03×10^{-3}	3.15×10^{-4}	7.53×10^{-5}	1	0.5556
3.66×10^{-3}	5.67×10^{-4}	1.36×10^{-4}	1.8	1

HEAT TRANSFER COEFFICIENT- h or U

watts/cm.^{2} ^oC	cal./sec. cm.^{2} ^oC	Btu/hr.ft.^{2} ^oF	Pcu/hr.ft.^{2} ^oF
1	0.239	1763	979
4.183	1	7373	4096
5.67×10^{-4}	1.36×10^{-4}	1	0.5556
1.02×10^{-3}	2.44×10^{-4}	1.8	1

Table of Equivalents
(Table 17-I Con't)

VELOCITY - V

cm./sec.	ft./sec.	ft./hr.	miles/hr.
1	0.0328	118.1	0.0224
30.48	1	3600	0.6818
8.47×10^{-3}	2.78×10^{-4}	1	1.89×10^{-4}
44.70	1.467	5280	1

LENGTH - L or D

cm.	in.	ft.
1	0.3937	0.0328
2.54	1	0.0833
30.48	12	1

MASS FLOW RATE - w

gm./sec.	lb./hr.
1	7.937
0.126	1

AREA - A

$cm.^2$	$in.^2$	$ft.^2$
1	0.155	1.08×10^{-3}
6.45	1	6.94×10^{-3}
929	144	1

MASS VELOCITY - G

gm./sec. $cm.^2$	lb./hr.$ft.^2$
1	7373
1.36×10^{-4}	1

MASS - M

lb.	gm.
1	453.59
2.205×10^{-3}	1

DENSITY - ρ

gm./cm.^{3}	lb./ft.^{3}
1	62.43
0.016	1

TEMPERATURE - T*

°F or °R	°C or °K
1.8	1
1	0.5556

HEAT CAPACITY - c

1 cal./gm. °C = 1 Btu/lb. °F

VISCOSITY - μ or η **

poise*** or gm./cm.sec. or dyne sec./cm.^{2}	lb. (mass)/ft.hr. or poundal hr./ft.^{2}	lb. (mass)/ft.sec. or poundal sec./ft.^{2}
1	242	0.0672
4.13×10^{-3}	1	2.78×10^{-4}
14.87	3600	1

*For temperature conversions- (See Figure 17-1)

$T_C = 5/9\ (T_F - 32)$ $\qquad T_K = T_C + 273$

$T_F = 9/5\ (T_C) + 32$ $\qquad T_R = T_F + 460$

** Kinematic viscosity, $\nu = \frac{\mu}{\rho}$ and is measured in L^2/θ, such as ft.^{2}/sec.

*** 1 centipoise = 0.01 poise $\cong$ viscosity of water at 68.6 °F (20.3 °C)

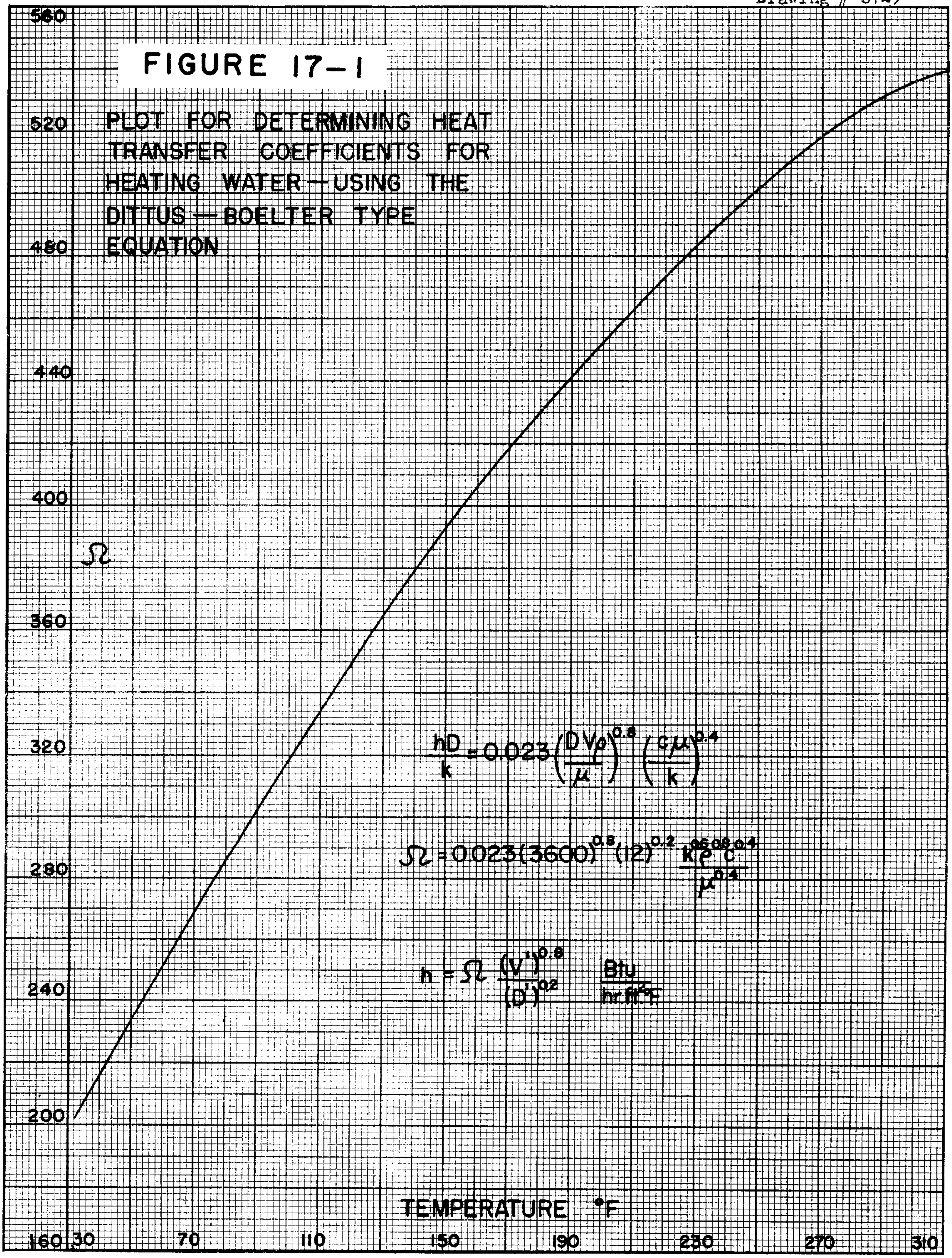


FIGURE 17-1
PLOT FOR DETERMINING HEAT TRANSFER COEFFICIENTS FOR HEATING WATER — USING THE DITTUS — BOELTER TYPE EQUATION
Ω
$\frac{hD}{k} = 0.023\left(\frac{DV\rho}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4}$
$\Omega = 0.023(3600)^{0.8}(12)^{0.2}\frac{k^{0.6}\rho^{0.8}c^{0.4}}{\mu^{0.4}}$
$h = \Omega\frac{(V')^{0.8}}{(D')^{0.2}}$ $\frac{Btu}{hr. ft^2 °F}$
TEMPERATURE °F
560
520
480
440
400
360
320
280
240
200
160
30
70
110
150
190
230
270
310

Simplified Calculation of Heat Transfer Coefficient for Heating Water

Since heating water is a widely used heat transfer operation, it is thought that a method of simplifying the calculation for this case would be useful. The Dittus-Boelter type of equation (see Part 5) is used and written in the following form:

$$h = \Omega \frac{(V')^{0.8}}{(D')^{0.2}} \quad \text{, where}$$

h is in Btu/hr.ft.^{2} $^\circ$F.; V' is velocity in ft./sec.; and, D' is diameter in inches. Figure 17-1 is a plot of Ω versus temperature.

Properties of water are given in Part 12.

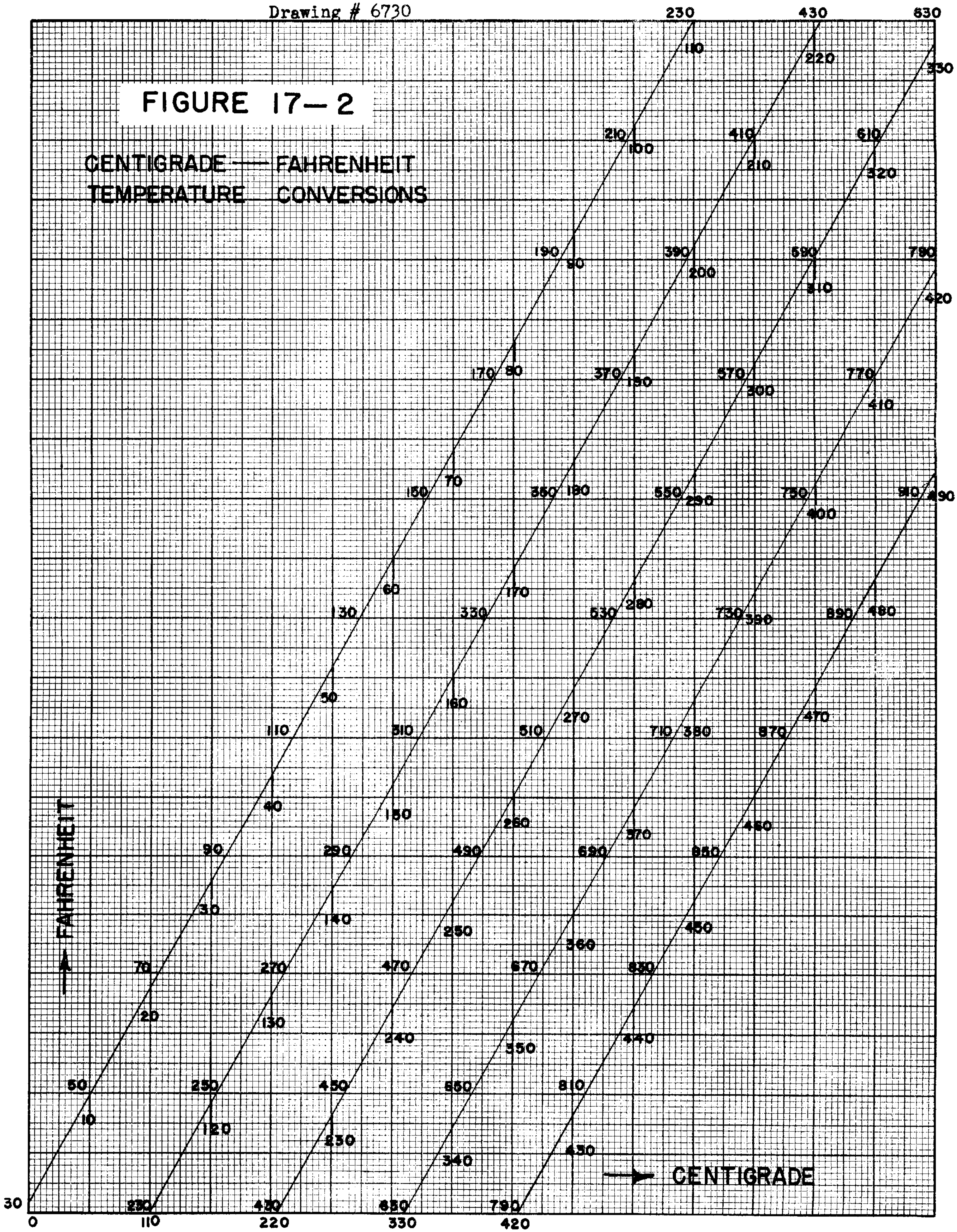
Useful Graphs

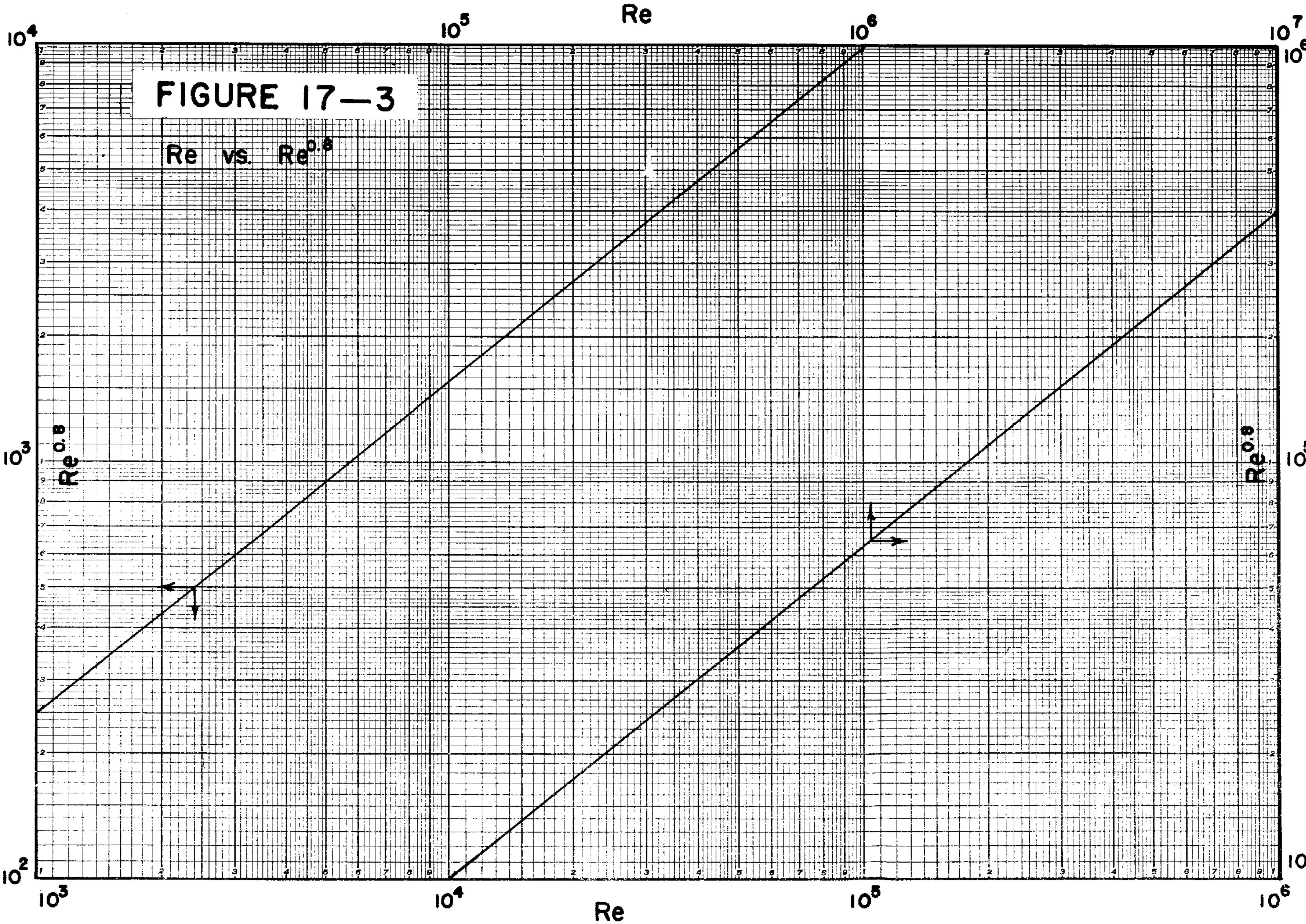
For facilitating temperature conversions and the raising of dimensionless groups to various powers, the following graphs have been included:

Figure	Title
17-2	Centrigrade-Fahrenheit Temperature Conversions.
17-3	Re vs. Re $^{0.8}$
17-4	Pr vs. Pr^n (for $n = 0.3, 1/3, 0.4, 0.6, 2/3, 0.7$)
17-5	V' vs. $(V')^{0.8}$
17-6	D' vs. $(D')^{0.2}$

Prandtl numbers for several coolants may be found in Parts 12, 13, and 14 of this report.

Drawing # 6730





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Drawing # 6731

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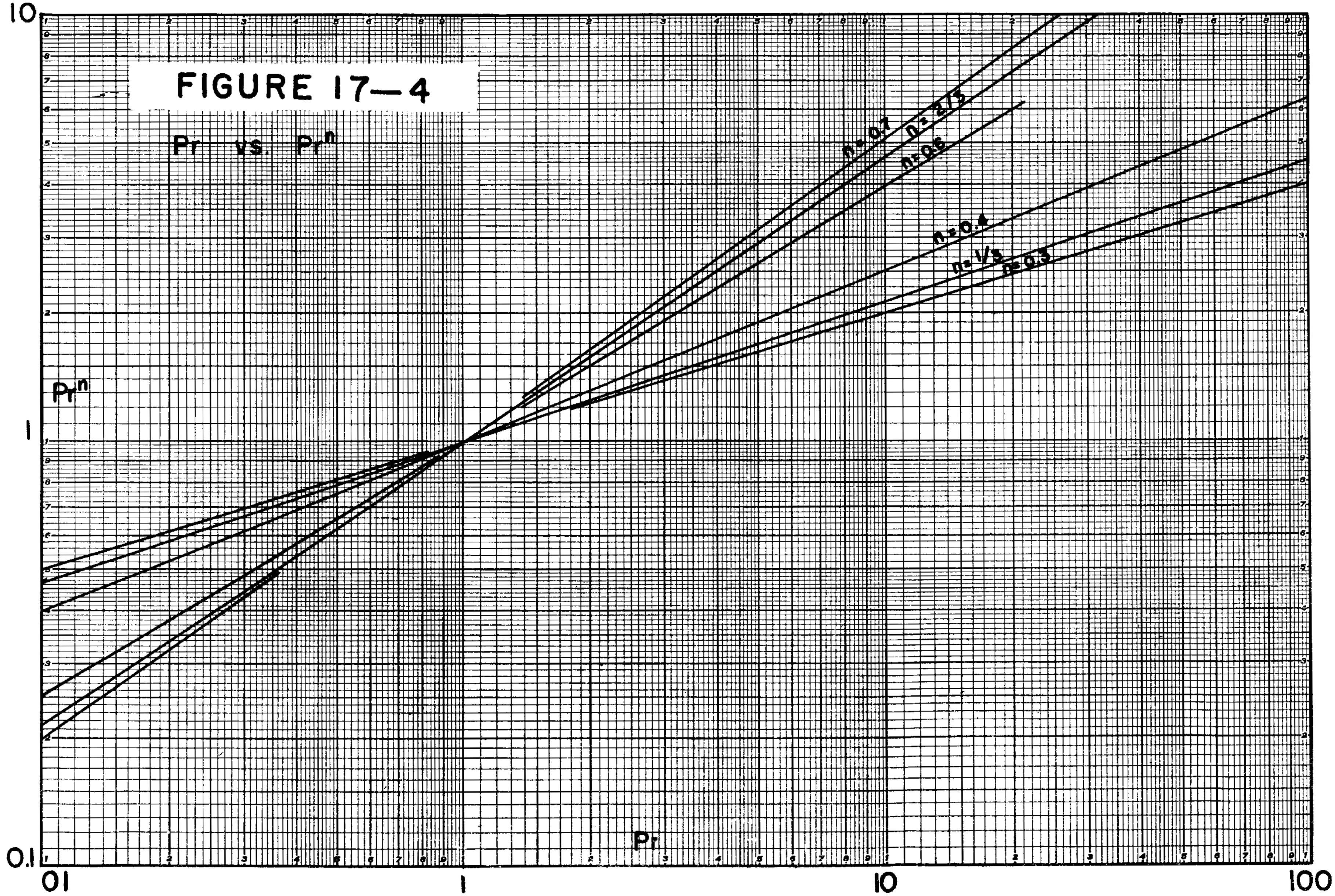

FIGURE 17—4

Pr vs. Pr^n

Drawing # 6732

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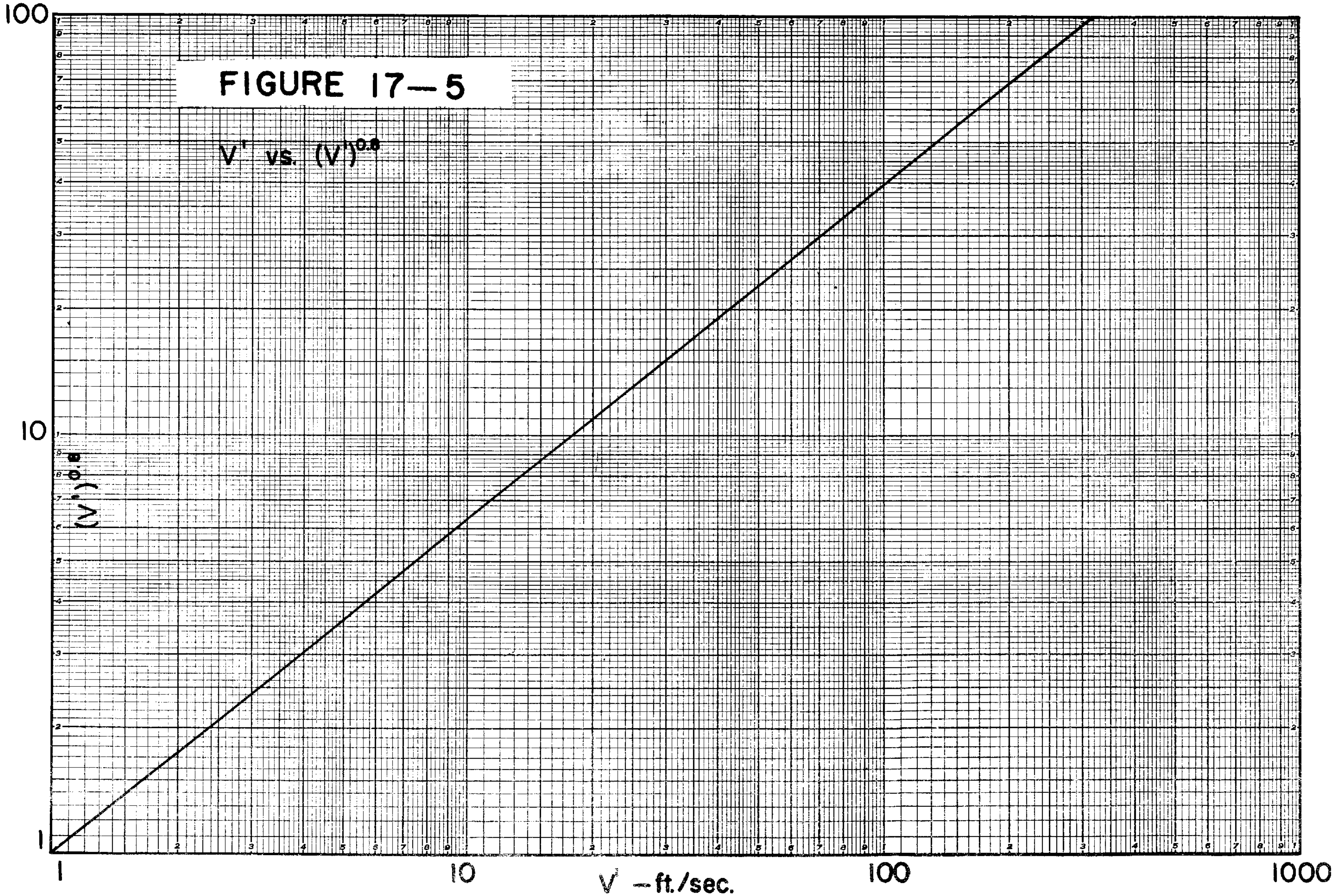

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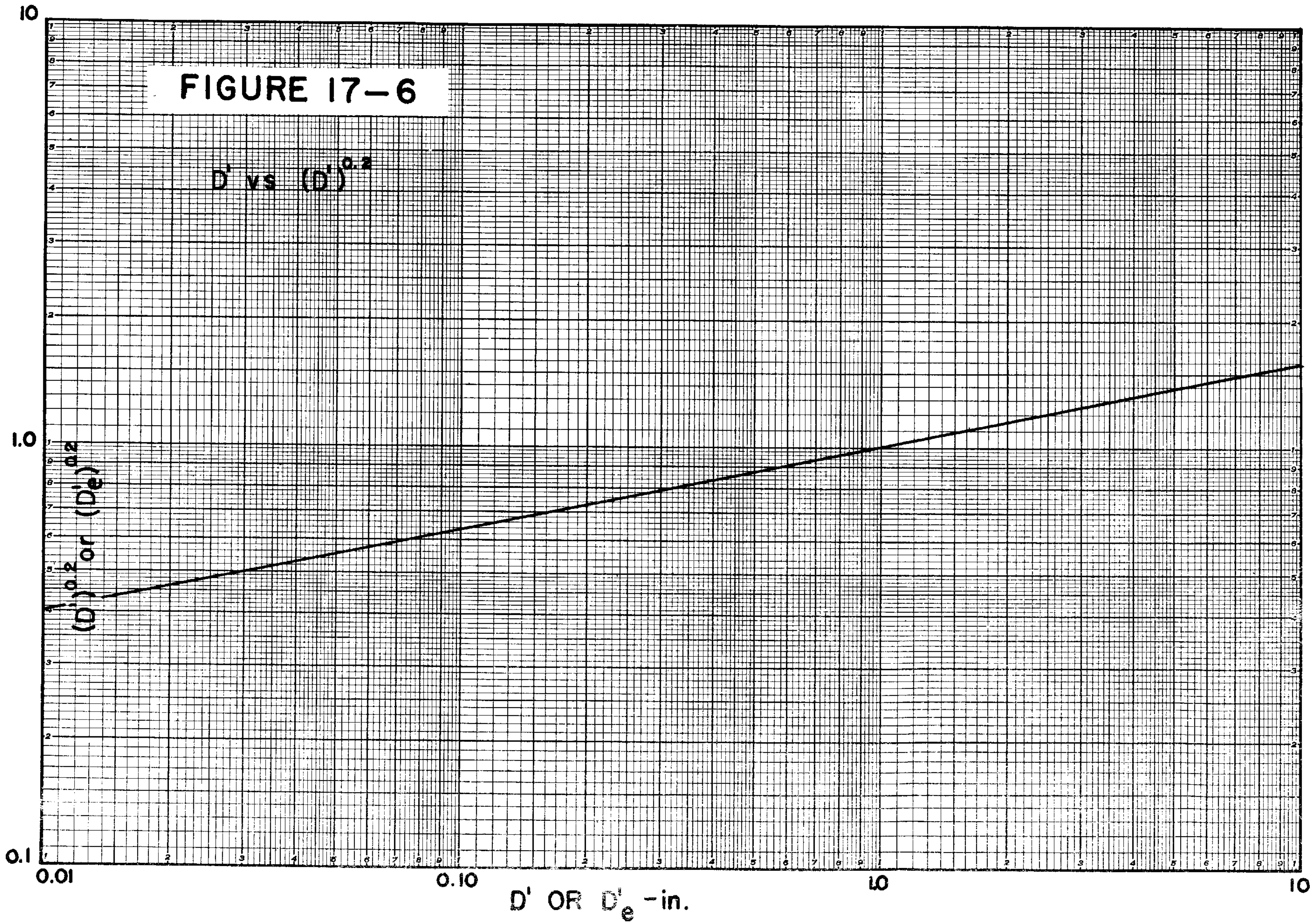

Drawing # 6734

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HEAT TRANSFER IN MANHATTAN DISTRICT AND ATOMIC ENERGY COMMISSION LABORATORIES: A CRITICAL SURVEY

PART 18

NOMENCLATURE

W. B. Harrison
October 1, 1948

[redacted]

PART 18

NOMENCLATURE

Capital letters

- A heat transfer area, $ft.^2$
- A flow area of cooling channels, $ft.^2$ (Part 16)
- D diameter, ft. (D', in.)
- D_e equivalent hydraulic diameter, ft. (D_e', in.)
- F advantage factor when film temperature drop controls
- G mass velocity = $V\rho$, lb./hr. $ft.^2$
- J mechanical equivalent of heat = 778 ft. lb./Btu
- L length, ft.
- L advantage factor when temperature rise of coolant controls
- M molecular weight
- P pressure, lb./$in.^2$
- P pumping power required to circulate coolant (Part 16)
- R thermal resistance, $\frac{hr.\ ft.^2\ {}^oF.}{Btu}$
- T absolute temperature, $^oR.$
- U overall heat transfer coefficient, $\frac{Btu}{hr.\ ft.^2\ {}^oF.}$
- V linear velocity, ft./hr. (V', ft./sec.)
- W power output
- X parameter in correlation of boiling data by Insinger and Bliss $\equiv \dfrac{q}{A \rho_L \sqrt{\lambda^3}\, J_g} \times 10^{10}$

<u>Capital letters (Con't.)</u>

Y parameter in correlation of boiling data by Insinger and Bliss = $\dfrac{h(\rho_V \sigma)^{0.5}}{\left(c k \rho_L^3 \lambda^3 J^3 g\right)^{0.5}} \times 10^{10}$

<u>Lower case letters</u>

a total heat transfer surface area (Part 16)

c heat capacity, Btu/lb. °F.

g gravitational constant, 4.17×10^8 ft./hr.^{2}

h heat transfer coefficient, Btu/hr. ft.^{2} °F.

j dimensionless parameter; usually as defined by Colburn $\dfrac{h}{cG}\left(\dfrac{c\mu}{k}\right)^{2/3}$

k thermal conductivity, $\dfrac{\text{Btu}}{\text{hr. ft.}^2 \ (^\circ\text{F./ft.})}$

q heat rate, Btu/hr.

r radius, ft.

t temperature, °F.

w mass flow rate, lb./hr.

<u>Greek letters</u>

α absorptivity
α denotes proportionality

α $\dfrac{\epsilon_H}{\epsilon_M}$

β volumetric coefficient of expansion

γ c_p/c_v

δ small increment

Δ difference, i.e. Δt

Greek letters (Con't.)

ϵ eddy diffusivity
ϵ emissivity

λ latent heat of vaporization, Btu/lb.
λ denotes function

λ' average difference in heat content of vapor and liquid

μ viscosity, lb./ft. hr.

ρ density, lb./ft.^{3}

σ surface tension, lb./ft.
σ atom capture cross-section for thermal neutrons (Part 16)

$\Sigma\sigma$ sum of atom capture cross-sections in a molecule (Part 16)

Φ free convection effect

χ $Nu/Pr^{0.4}\ Re^{0.8}$

ψ denotes function

Ω $h(D_e')^{0.2}/(V')^{0.8}$

Subscripts

crit critical condition

max maximum

1 small diameter of annulus

2 large diameter of annulus
2 case described by Latzko

3 case described by Latzko

4 case described by Latzko

a arithmetic mean

b bulk property

c critical condition

c convection

Subscripts (Con't.)

f film property

i interface

i inside wall

o outside wall

p constant pressure

r radiation

r reduced (ρ_r)

s solid surface

v constant volume

w wall

x unknown source

C cold stream

F case limited by film temperature drop

H hot stream
H heat

L liquid
L logarithmic mean
L case limited by temperature change in coolant

M momentum

V vapor

Dimensionless groups

Grashof	Gr	$D^3\rho^2 g\beta\Delta t/\mu^2$
Nusselt	Nu	$\frac{hD}{k}$
Peclet	Pe	$Re \cdot Pr = \frac{DV\rho c_p}{k} = \frac{DG\ c_p}{k}$
Prandtl	Pr	$\frac{c_p\mu}{k}$

Dimensionless groups (Con't.)

Reynolds	Re	$\frac{DG}{\mu} = \frac{DV\rho}{\mu}$
Stanton	St	$\frac{h}{cG} = \frac{h}{cV\rho}$