THE UNIVERSITY OF MICHIGAN
INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

HEAT EXCHANGER DESIGN MANUAL

(Shell and Tube Types with Plain Tubes)

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HEAT EXCHANGER DESIGN MANUAL

INTRODUCTION

A wide variety of devices are used for transferring heat from one substance to another. The engineer is interested in those types which transfer heat from a liquid to a liquid, from a gas to a liquid, from a gas to a gas, and from a liquid to a gas. Each case represents a particular type of service, which in turn dictates the type of heat exchanger that must be used. Many of these types of heat transfer service can be carried out in similar units, while others require variations.

The selection of an optimum design at minimum cost poses many process variables from calculation to initial operations. It is the engineer's problem to determine the type of heat transfer unit that will effect cheaply and efficiently the required heat transfer. He must be familiar with the various types of heat exchangers that are available; and from his calculations and knowledge of the problem, he must choose a unit that will satisfy best the requirements.

In many fields, the engineer must often search through a great many sources to obtain specific data bearing upon a given problem. This situation is true in the design of heat exchangers, where the correlations of heat transfer and pressure drop data and the specifications of mechanical aspects are distributed throughout many sources. The purpose of this manual, therefore, is to provide, in a single volume, all pertinent data on the shell and tube type of heat exchangers. The manual covers the following material:

1. General description of shell and tube type heat exchangers, their internal arrangements, nomenclature and terminology.

2. A resume of formulas on heat transfer rates and pressure drop calculations.

3. Sample heat exchanger calculations to illustrate the use of various curves and tables based on the formulas.

The manual adopts the industrial standards for heat exchanger design. Insofar as possible the Standards of TEMA are suggested. The data incorporated in the manual have been applied to actual design work. These data will permit the engineer to select and design a heat exchanger which meets specific requirements of a given problem. It is common experience of engineers in actual industrial design problems to be confronted with data based on scant observations and hence of questionable usefulness or with data of widely varying nature. Quite often, sufficient data is not available from which to design heat exchangers. It is the aim of this manual to assure economic and workable designs.

The two general types of heat exchangers of prime interest are:

1. Heat exchangers in which only one of the fluids is confined.

2. Heat exchangers in which both fluids are confined.

An example of the first type is an air heater. Steam, hot air, or water is circulated through tubes or tube bundles, while air to be heated is passed over the outside of these tubes. Heat is transferred from the hot fluid confined in the tubes, through the tube wall, to the air surrounding the tubes. Other examples of this type are automobile radiators and condenser units in refrigeration systems.

The most widely used unit of the second type is the shell and tube type of heat exchanger. Variations of this unit can be used for any heat exchanger service where both fluids are confined. Shell and tube heat exchangers, for example, are used to condense vapors or to heat and cool liquids and vapors.

This manual stresses the design of shell and tube heat exchangers using plain tubes. Description of this type of exchanger, bringing out the outstanding features and modifications to suit various duties given in the manual will enable the engineer to delineate the process variable, to distinguish "knowns from unknowns" and to proceed with confidence in selection of proper equipment.

While summaries of the formulas correlating various design variables are given, the development of these correlations is excluded from the manual, as theoretical considerations of this nature do not fall within the scope of the manual. An exhaustive list of references at the end of the manual includes relevant references of this kind.

To illustrate the versatile nature of the data incorporated in the manual, sample heat exchanger calculations are included. Such calculations illustrate, by example, the use of curves and tables based on the formulas summarized elsewhere in the manual. In addition, the information presented in the manual may be very helpful in the design of other types of heat exchangers.

I. TYPES AND DESCRIPTION OF HEAT EXCHANGERS

A. Shell and Tube Heat Exchangers

1. Fixed Tube Sheet Type Heat Exchanger

The fixed tube heat exchanger is the simplest and the least expensive of the shell and tube type heat exchangers. In this type, both tube sheets are fixed rigidly to the shell. If the exchanger is short, or if the temperature difference between the shell-side fluid and the tube-side fluid is low, no provision need be made for tube expansion or contraction. If the exchanger is long, or if there is a large difference in temperature between the shell-side fluid and the tube-side fluid, an expansion joint can be fabricated into the shell. This type of unit can be used for many services in which clean shell-side fluids are assured. A drawing of a fixed tube sheet type heat exchanger is shown in Figure I-1.

2. U-Tube Type Heat Exchanger

Another type of heat exchanger involving simple construction and low cost is the U-tube heat exchanger. This type of exchanger is made by fixing bundles of U-shaped tubes into a single tube sheet. The tube sheet is channeled so that fluid can enter one leg of the U-shaped bundle and return through the other. Since straight through-cleaning of the tubes is not possible, only clean fluids are used in the tube-side of the exchanger. In this type of heat exchanger, it is possible to heat and cool over a wide temperature range, as no stresses are set up between the tube and the tube sheet. A sketch of this type of exchanger is shown in Figure I-2.

3. Internal Floating Head, Removable Tube Bundle Type Heat Exchanger

A widely used heat exchanger is the internal floating-head, removable tube bundle type. A differential expansion between the tubes and the shell is provided. This is accomplished by fixing the tubes in a fixed tube sheet at one end of the exchanger, while the tube sheet at the other end is allowed to "float" in the shell. The floating tube sheet has a head cover to separate the two fluids in the exchanger. The tube bundle can easily be removed from the shell for cleaning and repairing. This exchanger is shown in Figure I-3.

4. Packed Floating Head, Removable Tube Bundle Type Heat Exchanger

The packed floating head, removable bundle type heat exchanger illustrated in Figure I-4, is similar to the floating head type, except that no floating head cover is used. Instead, a packing material is put between the floating tube sheet and the shell to separate the two fluids. This type is confined to service where the pressure does not exceed 75 psig, as larger packing glands do not give satisfactory service and need frequent replacements causing greater maintenance cost.

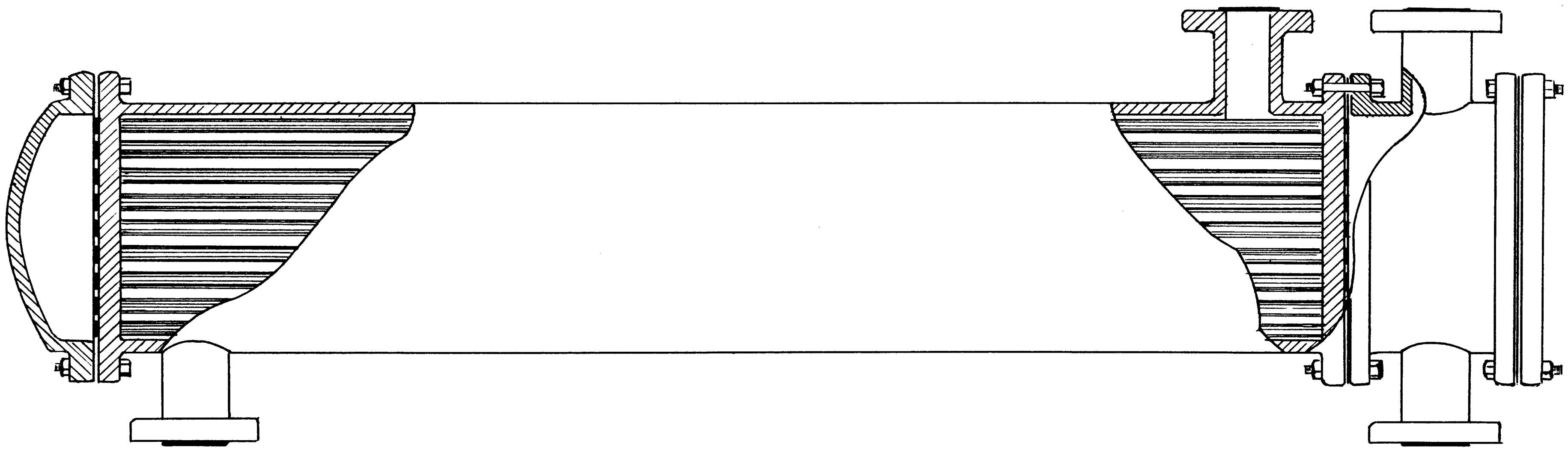


FIGURE I-1

FIXED TUBE-SHEET HEAT EXCHANGER

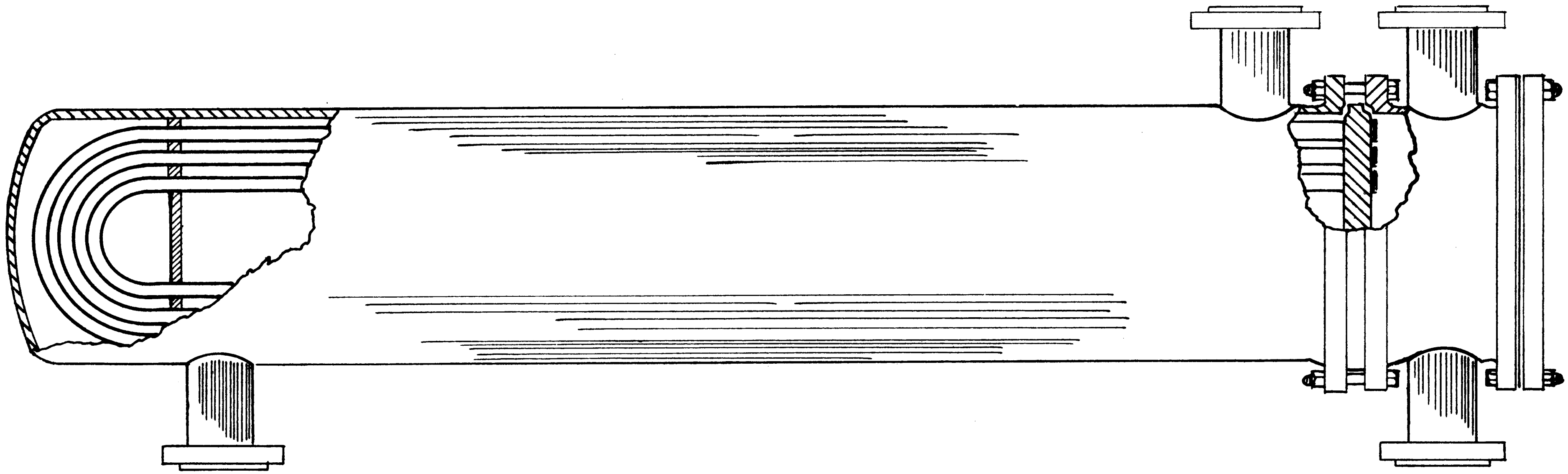


FIGURE I-2

U TUBE HEAT EXCHANGER

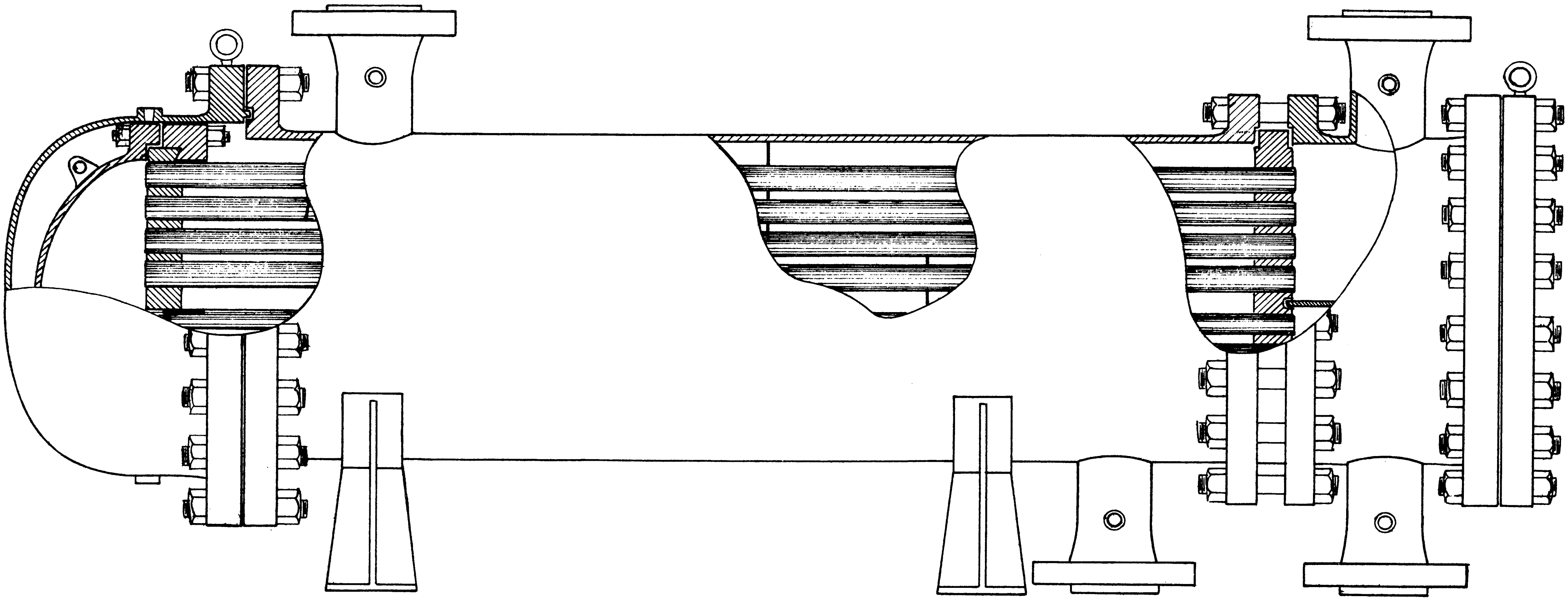


FIGURE I-3

INTERAL FLOATING-HEAD HEAT EXCHANGER

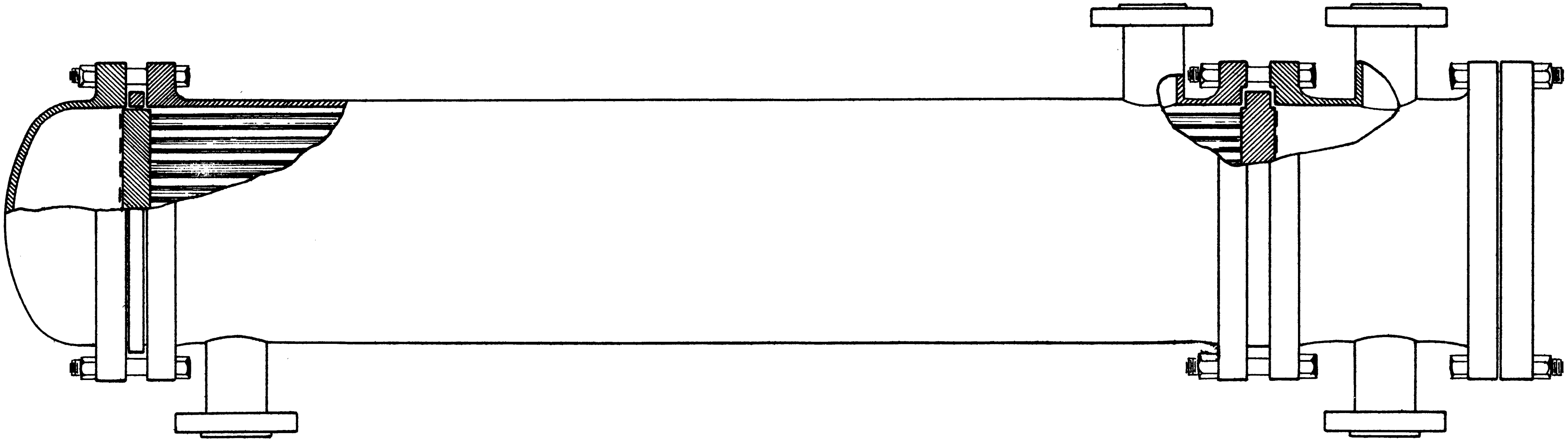


FIGURE I-4

PACKED FLOATING-HEAD HEAT EXCHANGER

5. Reboilers

The most common type of reboiler is shown in Figure I-5. It consists of a tube bundle placed in a shell such that a large vapor space is available. Some reboilers are of the vertical shell and tube heat exchanger type in which boiling takes place in either the tube or the shell side. Reboilers are used extensively in connection with distillation equipment and evaporators.

6. Concentric Tube, or Double Pipe Heat Exchanger

When small quantities of heat are to be transferred, an exchanger known as the double pipe or concentric tube type heat exchanger is used. These exchangers are simple to build, small, and relatively inexpensive. This type is made by jacketing a single tube or pipe within another slightly larger tube or pipe. One fluid flows through the inner tube, while the other fluid flows through the annular space between the two tubes.

B. Internal Arrangements

1. Single Pass Shell - Single Pass Tubes

A typical single pass shell - single pass tube heat exchanger is illustrated in Figure I-6. It is known as 1-1 exchanger. In this arrangement the fluid in the shell side enters one end and leaves the other; while the fluid in the tube side makes one pass through the exchanger counter flow to the liquid on the shell side.

2. Single Pass Shell - Multipass Tubes

Figure I-7 shows a typical single pass shell - multipass tube heat exchanger. The fluid on the shell side enters one end of the exchanger, and leaves at the other; the fluid on the tube side enters one end, passes through a portion of the tubes to the opposite end, where its direction is reversed by a channel, and returns through the remaining tubes to its starting end. Each traverse of the exchanger is termed a pass. The exchanger illustrated is an example of a two-pass exchanger, or a 1-2 exchanger.

3. U-Tube, Single Pass Shell - Multipass Tubes

A U-tube, single pass shell - multipass tube-heat exchanger is illustrated in Figure I-8. The fluid in the shell side of the exchanger enters one end and leaves the other. The fluid in the tube side may make any number of transverses, but in this case the fluid changes direction in the U bends at one end of the exchanger and by means of channels at the other end.

4. Two Pass Shell - Multipass Tubes

When high heat recoveries are required, a two-pass shell - multipass tube heat exchanger is used. An example of such an exchanger is illustrated in Figure I-9. This is an example of a 2-2 exchanger. A longitudinal baffle is placed in the shell of this exchanger so that the fluid on the shell side must enter one end, pass through the shell on one side of the baffle, and return on the other side. The fluid on the tube side can make as many passes as are required.

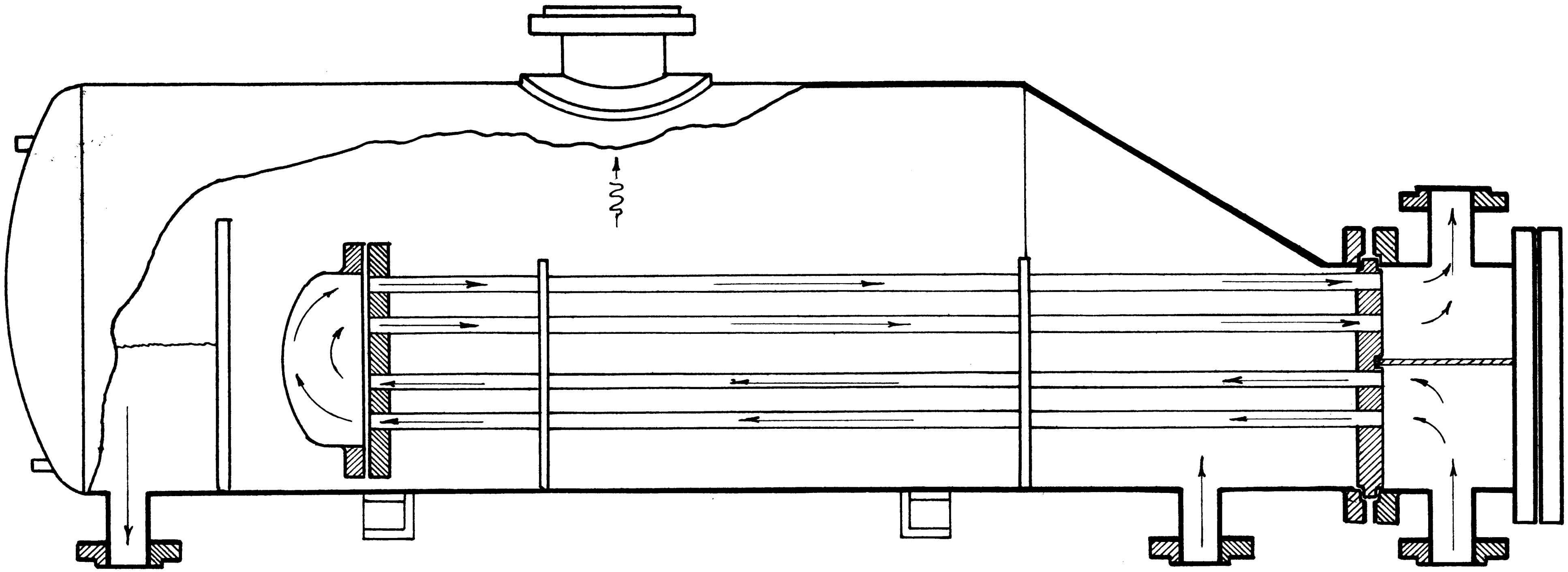


FIGURE I-5

KETTLE TYPE REBOILER

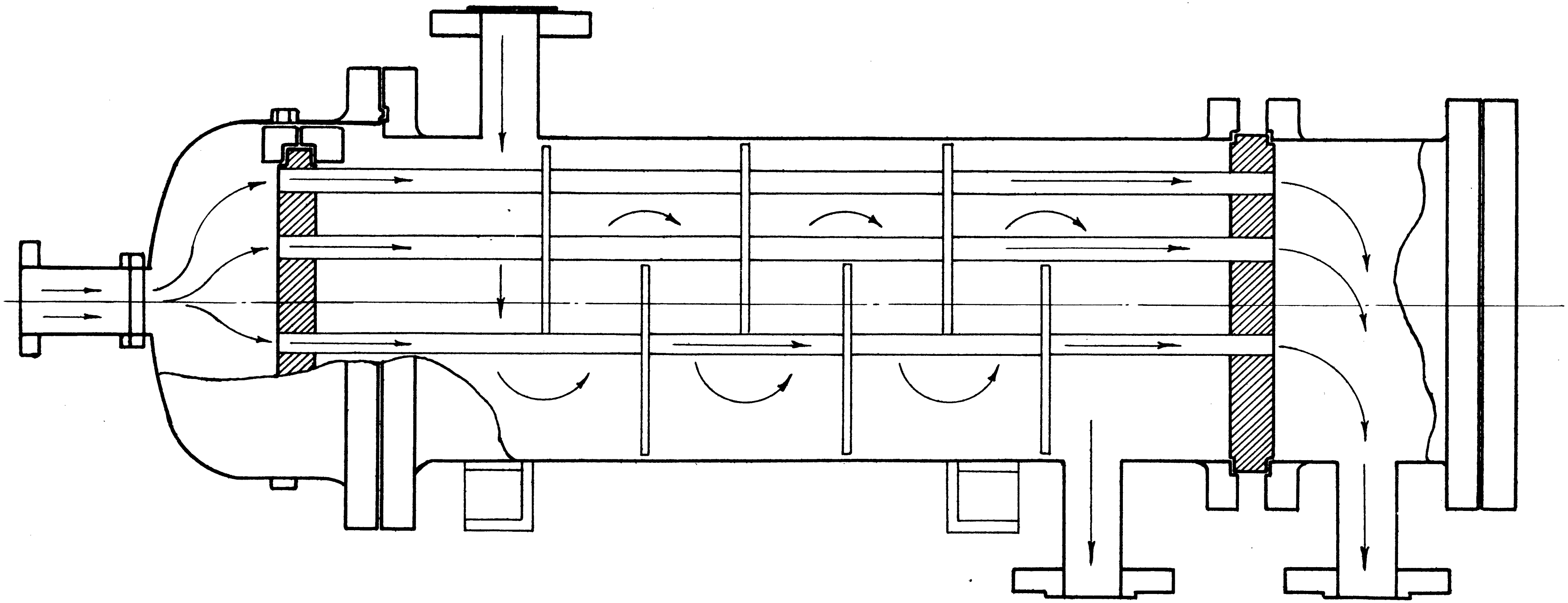


FIGURE I-6

SINGLE PASS SHELL – SINGLE PASS TUBES

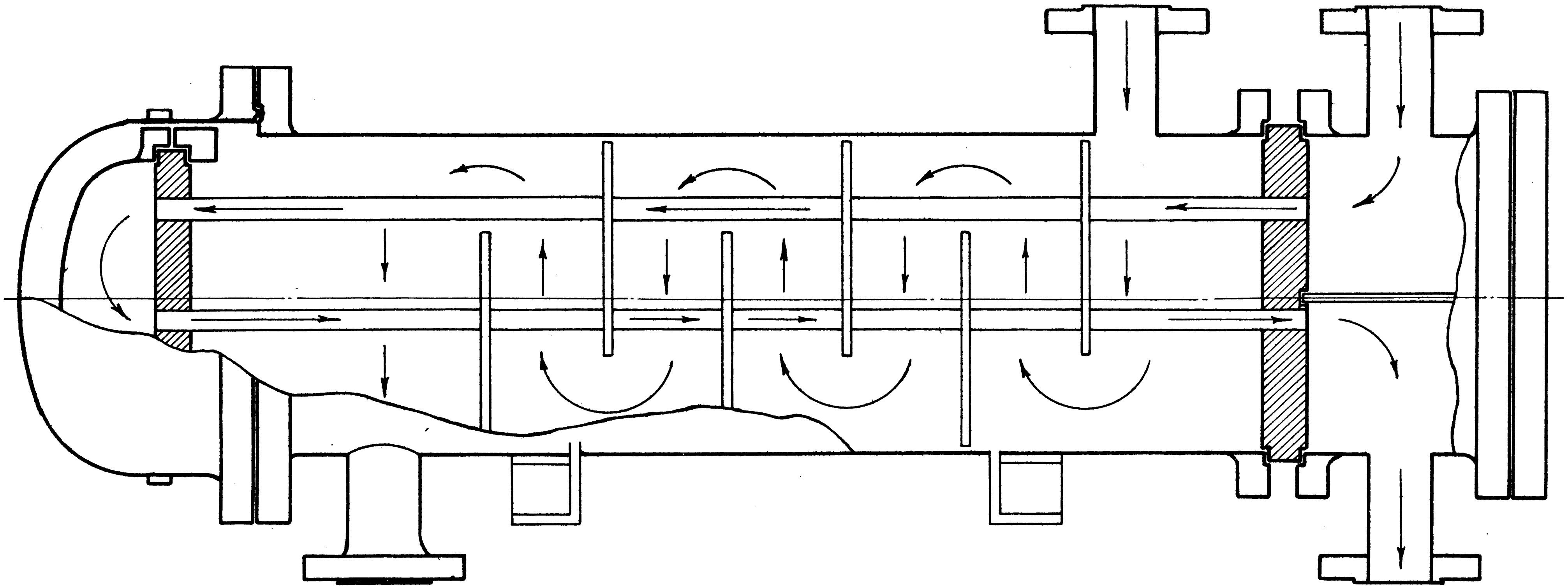


FIGURE I-7

SINGLE PASS SHELL — MULTIPASS TUBES

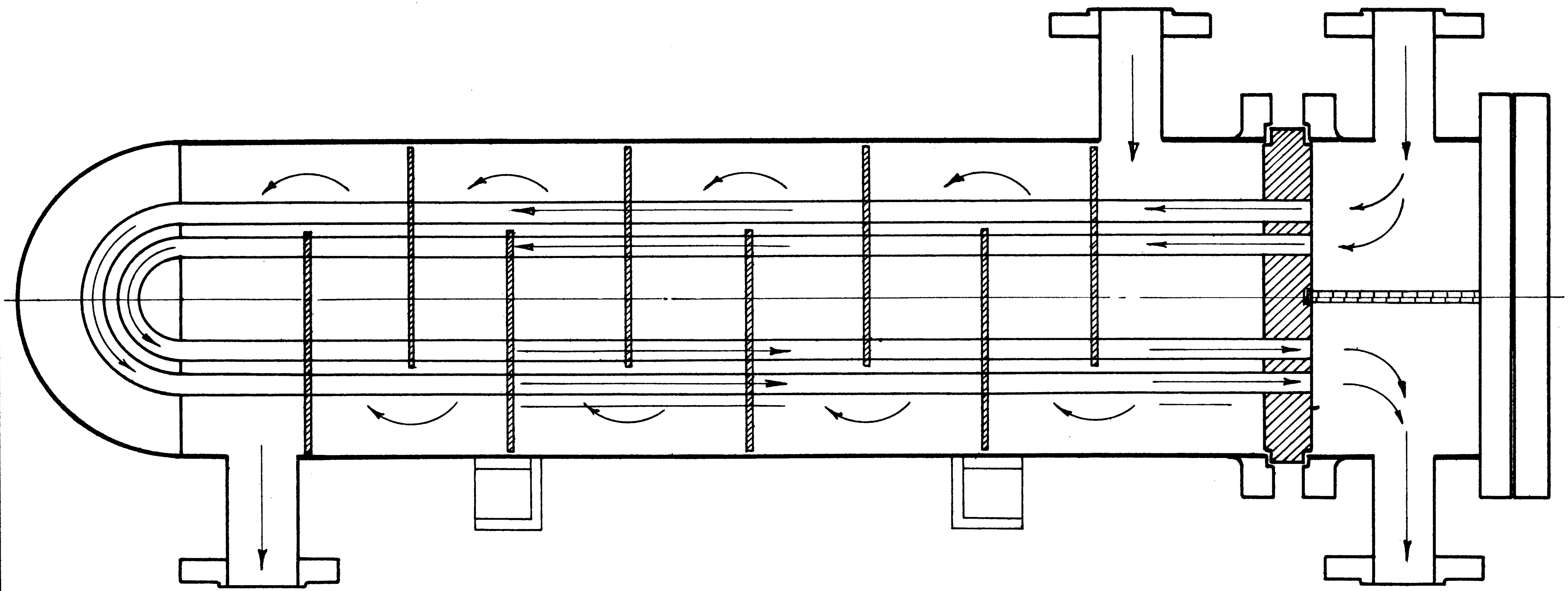


FIGURE I-8

U-TUBE, SINGLE PASS SHELL — MULTIPASS TUBES

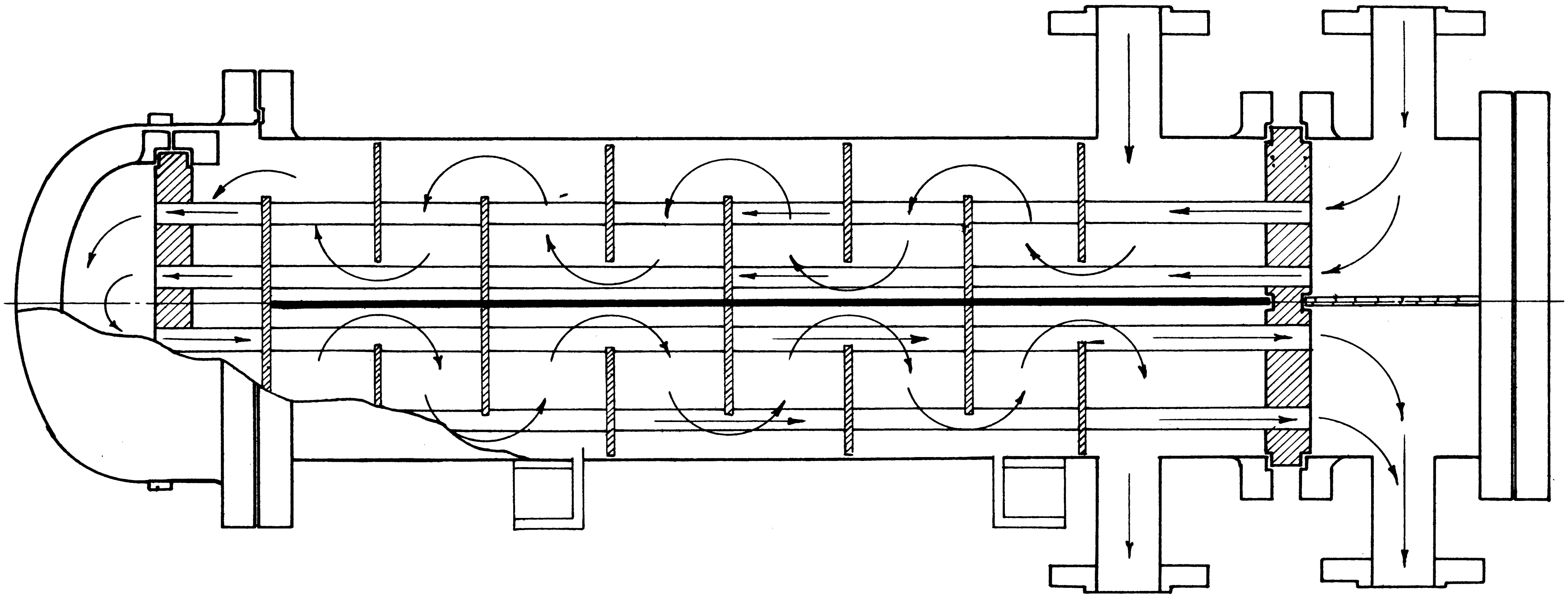


FIGURE I - 9

TWO PASS SHELL — MULTIPASS TUBES

5. Divided Flow Shell - Multipass Tubes

A divided flow shell - multipass tube heat exchanger is shown in Figure I-10. The baffles are arranged so that the fluid enters the shell side, halfway between the two ends of the exchanger. From this point, the flow of fluid splits. A portion of the fluid flows to one end of the exchanger and the balance of it leaves the other end. This baffle arrangement is used when the flow of shell fluid is high and pressure drop through the exchanger becomes an important design consideration. The fluid in the tube side of the exchanger can make as many passes as necessary.

6. Divided Flow Shell - Multipass Tubes with Longitudinal Baffles

Figure I-11 shows a typical divided flow shell - multipass tube exchanger. This exchanger has two shell side inlet nozzles and two shell side outlet nozzles. A longitudinal baffle runs the length of the exchanger and divides the shell into halves. Shell fluid can enter either side of the shell, on either end of the exchanger. The heat transfer problem at hand dictates the best hookup for this exchanger. The fluid in the tube side can make as many passes as required.

7. Transverse Baffles

Since physical considerations limit the number of tubes that can be placed in a given shell, the velocity of the fluid on the outside of these tubes is limited for a given mass flow rate. Generally, an increase in the heat transfer coefficient on the shell side can be achieved if the velocity of the shell side fluid can be increased. This is accomplished by using baffles arranged transversely to the axis of the tubes.

There are three types of transverse baffles used in heat exchanger design, namely:

1. Orifice type,

2. Disc and doughnut type,

3. Segmental type.

Figure I-12 shows an illustration of an orifice-type baffle. These baffles are designed so that a free space, or area, exists between each tube and baffle. The free area at each baffle and between adjacent baffles is shown in the illustration as nonshaded areas. The clearance between the baffles and the shell wall must be as small as possible to minimize channeling and leakage. Tube supports are required if this type of baffle is used. When the shell-side fluid is fouling or corrosive in nature, the orifice-type baffle offers difficulties in maintenance.

A disc and doughnut baffle is shown in Figure I-13. This baffle arrangement consists of disc-shaped baffles and doughnut-shaped baffles placed alternately in the shell. The disc consists of a circular baffle placed transversely to the axis of the tubes. It is smaller in diameter than the shell, and therefore an annular space exists between the edge of the baffle and the shell wall. The doughnut baffle is a circular disc with a hole in the center--much like a doughnut. Its outside diameter allows it to fit snugly in the exchanger shell. The hole is large enough to give the required longitudinal flow velocity across the tubes.

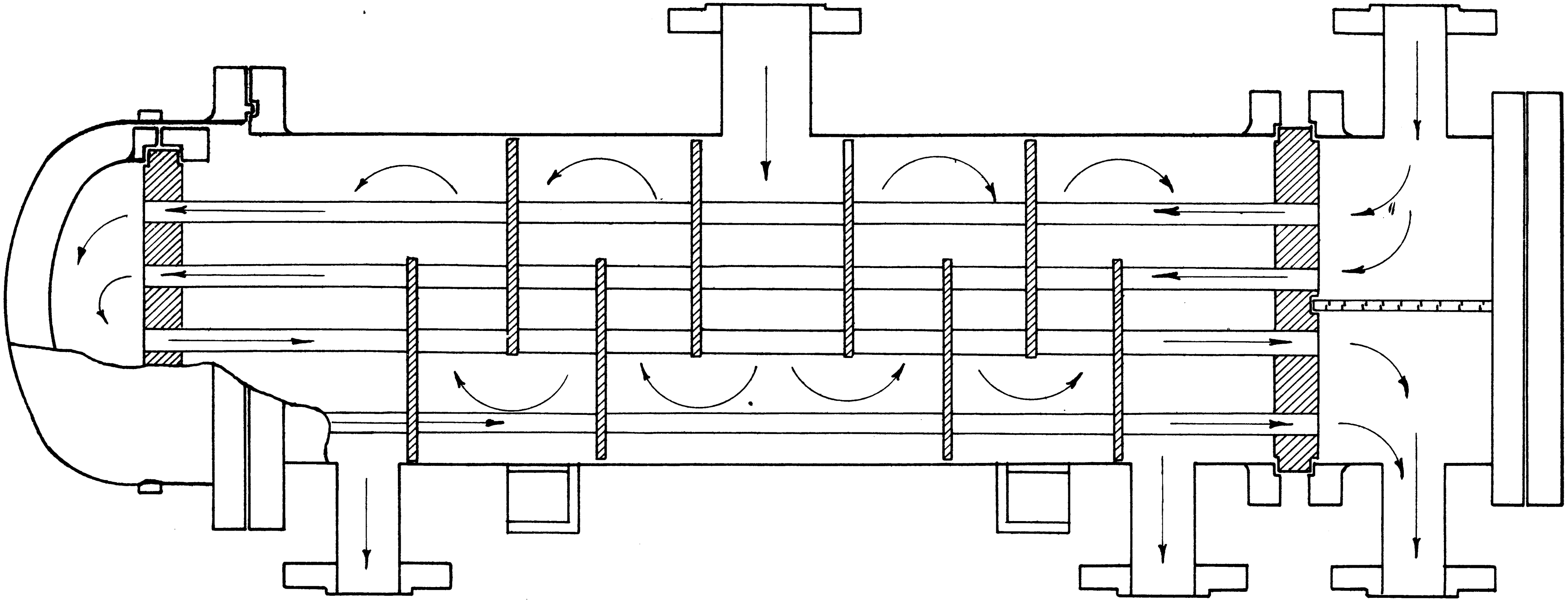


FIGURE I - 10

DIVIDED FLOW SHELL — MULTIPASS TUBES

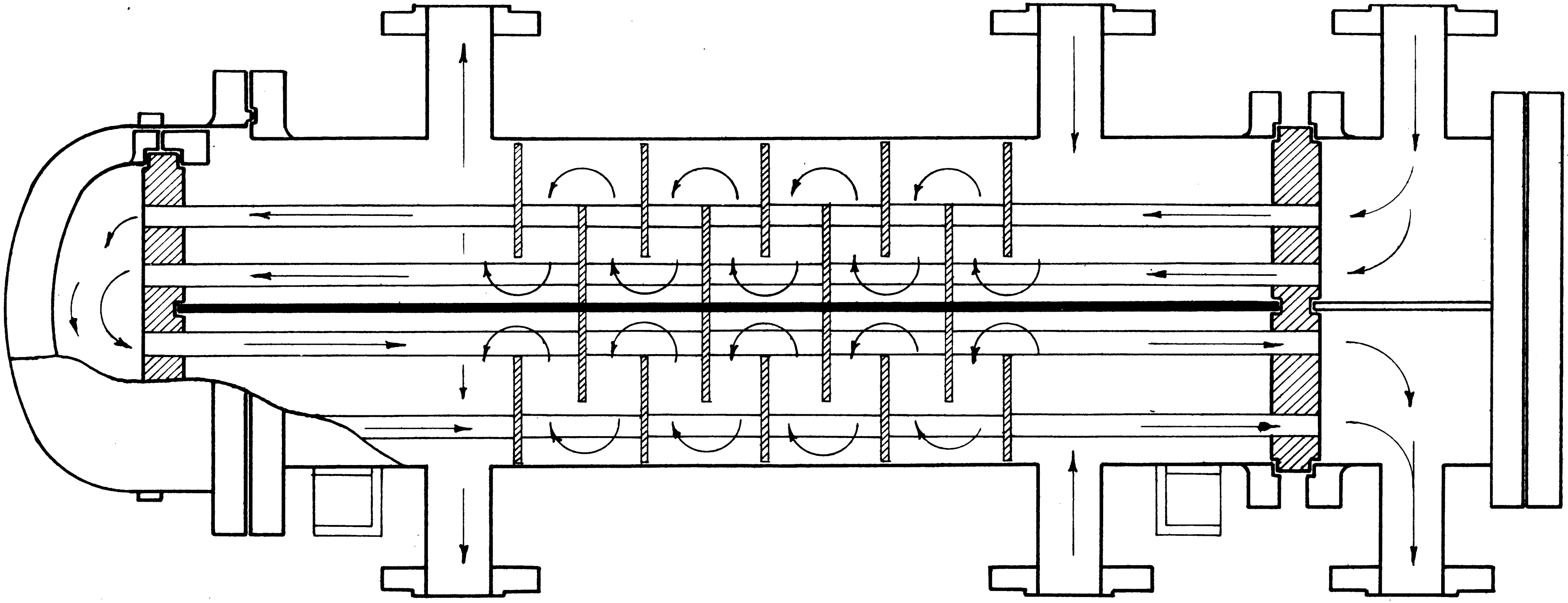
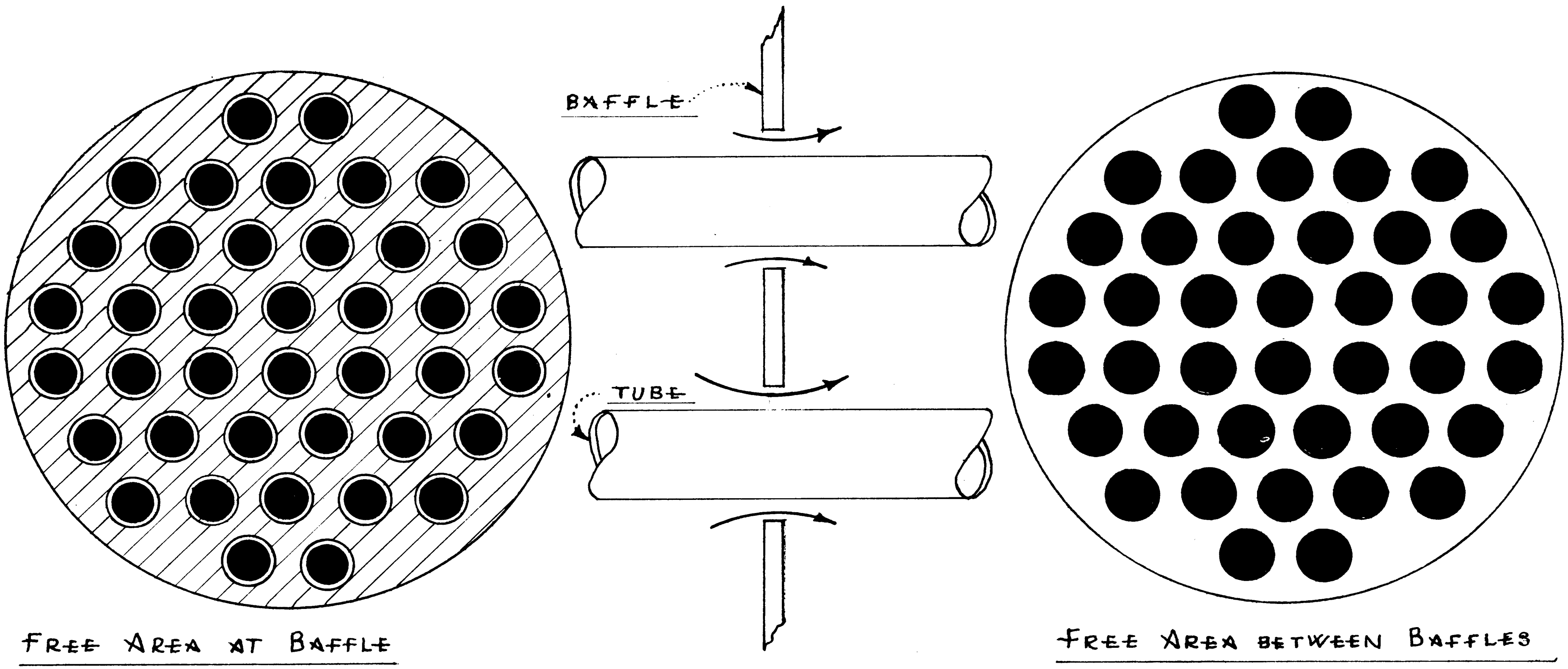


FIGURE I - 11

DIVIDED FLOW SHELL - MULTIPASS TUBES WITH LONGITUDINAL BAFFLE



ORIFICE BAFFLE

Figure I-12

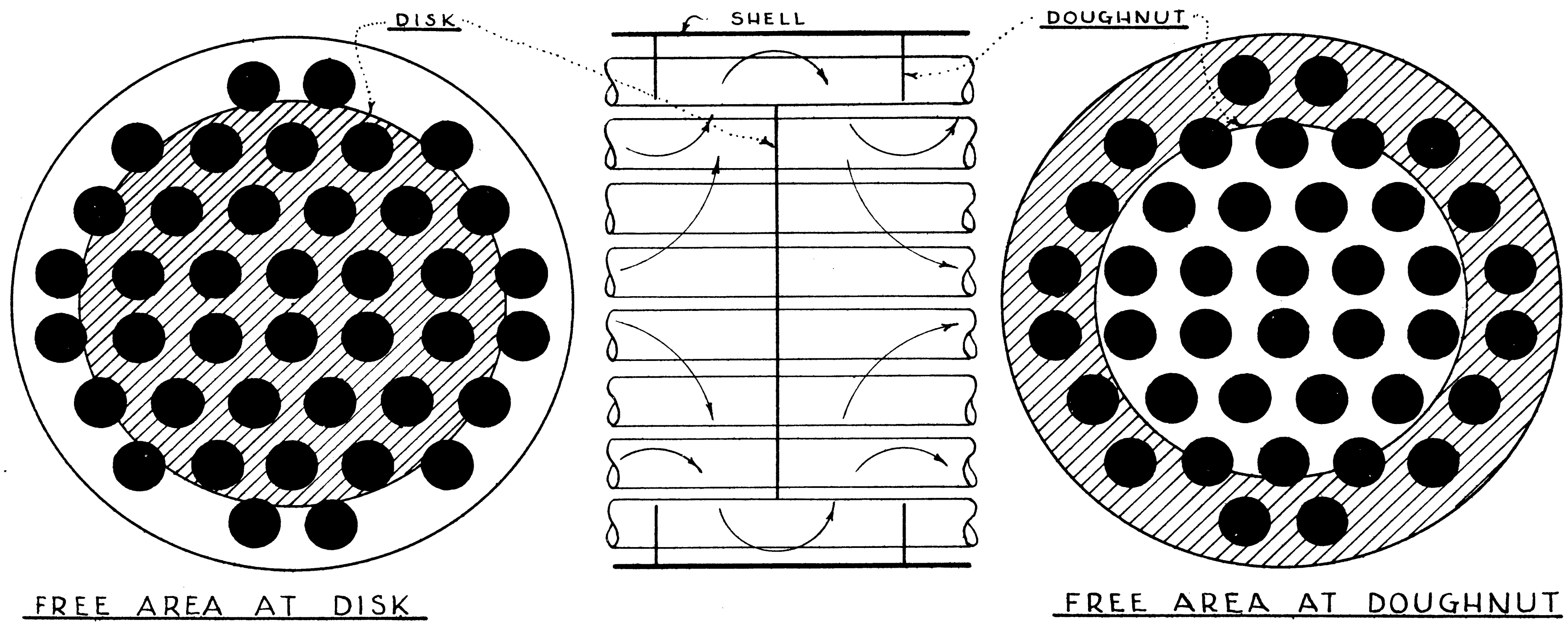

DISK
SHELL
DOUGHNUT
FREE AREA AT DISK
FREE AREA AT DOUGHNUT
DISK-AND-DOUGHNUT BAFFLE

Figure I-13

The most common and generally used transverse baffle is the segmental type. A diagram of this baffle is shown in Figure I-14. This arrangement is made by installing circular discs in the shell, transversely to the tube axis. A segmented area is cut from each disc. These baffles are generally cut with a segment height which is 25 percent cut baffles. Other baffle cuts are made, however, and are designated as being cut a number of tube rows past the centerline of the exchanger.

The baffles are oriented in the exchanger so that the shell fluid flows through the segmented openings longitudinally to the axis of the tubes. The segmented opening on the next baffle is on the opposite side of the exchanger; therefore, the shell fluid must cross the tubes between the baffles before it can pass through the opening of the next baffle. This alternating cross and longitudinal flow around the tubes continues the length of the exchanger. The number of times the shell fluid crosses the tubes depends upon the number of baffles in the shell, which, in turn, is determined by the spacing needed to provide the velocity required.

8. Impingement Baffles

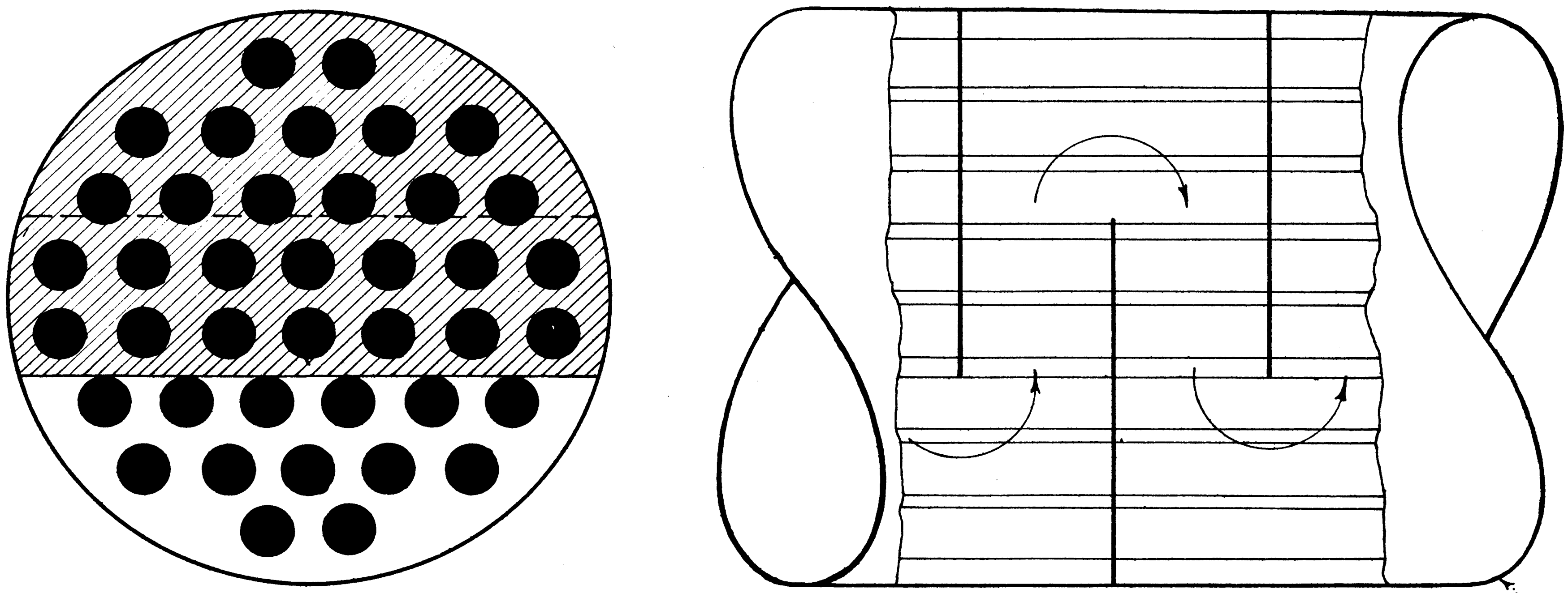
Near the point of entry of the shell fluid, its velocity can be high. Provisions can be made to protect the tubes at this point; otherwise, serious erosion is likely to occur. This can be done by placing an impingement baffle between the tubes and the point of entry of the shell fluid. In some cases, the nozzles are flared so that the velocity of the entering fluid is reduced.

9. Vents and Drains

When designing heat exchange equipment, adequate vents and drains must be provided. The nature of service for which the exchanger is designed will dictate the type and location of vents, drains, steam traps, etc. If the exchanger is designed for high pressure service, relief valves or rupture discs are required.

10. Routing of Fluids

The routing of fluids is determined by two factors: (1) to what degree a fluid fouls the transfer surface, and (2) the corrosiveness of the fluids. In consideration of the first factor, that fluid which fouls the transfer surface more rapidly should be routed through the tube side of the exchanger, since the inside of exchanger tubes are easier to clean. However, if both fluids foul the transfer surface to the same extent, the fluid under the highest pressure should be routed through the inside of the tubes, thus obviating the necessity for designing a costly, high pressure shell. In consideration of the second factor, that fluid with the greater corrosiveness should be routed through the tube side of the exchanger again making it unnecessary to use a shell of costly material.



FREE AREA AT BAFFLE

SHELL

SEGMENTAL BAFFLE

Figure I - 14

II. NOMENCLATURE AND HEAT EXCHANGER TERMINOLOGY

A. Table of Nomenclature

A, A_i, A_o = Heat transfer surface, general, inside, outside respectively, ft^2,

A_L = free area of longitudinal flow of fluid, ft^2,

A_m = mean heat transfer surface, ft^2,

A_x = free area of cross flow of fluid across tubes, ft^2,

a_s = flow area, ft^2,

B_o = box loss, psi,

B = baffle spacing, in.,

C = wall correction factor (Figure 56, Part V),

C_L = clearance between tubes (ligament), in.,

C_o = discharge coefficient for an orifice,

C_p = specific heat of hot fluid, Btu/lb(^oF),

c_p = specific heat of cold fluid, Btu/lb(^oF),

D = inside diameter of tubes, ft,

D_o = outside diameter of tubes, ft,

D_e = equivalent diameter for heat transfer and pressure drop, ft,

D_s = inside diameter of shell, ft.

ID, d = inside diameter of tubes, in.,

d_e = equivalent diameter for heat transfer and pressure drop, in.,

d_o, OD = outside diameter of tubes, in.,

d_s = inside diameter of shell, in.,

F_c = caloric fraction,

F, F_k, F_o, F_p = heat transfer or pressure drop correction factors--as specified in respective curves,

F_t = temperature difference factor, $\Delta t = F_t \times$ LMTD, dimensionless,

F_x = LMTD correction factor,

f = friction factor $ft^2/in.^2$,

G = mass velocity lbs/(hr)(ft^2),

G' = mass velocity defined as lbs/(sec)(ft^2),

G_r = Grashof number,

G_L = mass velocity in shell, longitudinal flow defined as lb/(hr)$(ft)^2$,

G_m = tubeside mass flow rate defined as lbs/tube x hr,

G_x = mass velocity in shell, cross tube flow defined as lb/(hr)$(ft)^2$,

g = acceleration due to gravity ft/sec^2,

h, h_i, h_o = heat transfer coefficient, general, for fluids inside tubes, and for fluids outside tubes respectively, Btu/(hr)(ft^2)(^oF),

h_io = value of h_i when referred to the outside diameter of the tube, Btu/(hr)(ft^2)(^oF),

J_H = factor for heat transfer, dimensionless,

K_c = caloric constant, dimensionless,

k = thermal conductivity, Btu/(hr)(ft^2)(^oF/ft),

L = tube length, also length in direction of heat flow, ft,

L_H = tube length-horizontal, ft L_v = tube length - vertical, ft,

LMTD = log mean temperature difference, ^oF,

M = mass in lbs/hr,

m = 10^3 lbs/hr,

n = number of tube passes,

N = number of shell-side baffles,

N_b = number of tube rows past center line,

N_D = number of tubes on horizontal diameter,

N_t = number of tubes,

N_v = number of tube rows in a vertical tier,

P = pressure in atmospheres,

ΔP = pressure drop, psi,

$\Delta P_T, \Delta P_t, \Delta P_p$ = total, tube and return pressure drop, psi,

P_T = tube pitch, in.,

p = tube loss defined as lbs/10 linear feet,

Q = heat flow, Btu/hr,

q = heat flow per unit area, Btu/hr x sq ft,

r = radius of exchanger shell, in.,

R = resistance to heat flow,

R_t = temperature group $(T_1 - T_2)/(t_2 - t_1)$ dimensionless,

R_d = combined dirt or fouling factor, $(hr)(ft^2)^{o}F/Btu$,

Re = Reynolds' number, dimensionless,

S_2 = temperature group $(t_2 - t_1)/(T_1 - T_2)$ dimensionless,

s = liquid specific gravity (referred to water = 1.0),

T, T_1, T_2 = temperature in general, inlet and outlet of hot fluid, ^{o}F,

T_a = average temperature of the hot fluid, ^{o}F,

T_c = caloric temperature of hot fluid, ^{o}F,

T_x = temperature of shell fluid between first and second passes, ^{o}F,

t_1, t_2 = inlet and outlet temperature of cold fluid, ^{o}F,

t_a = average temperature of cold fluid, ^{o}F,

t_c = caloric temperature of cold fluid, ^{o}F,

t_i = temperature at the end of first pass, ^{o}F,

t_w = tube wall temperature, ^{o}F,

t_y = temperature of tube fluid between second and third passes, ^{o}F,

Δt = the temperature difference in $Q = U_c A \Delta t$, ^{o}F,

$\Delta t_c, \Delta t_h$ = temperature difference at the cold and hot terminals, ^{o}F,

U_c, U_f = clean and design or fouled overall coefficients of heat transfer, $Btu/(hr)(ft^2)(^{o}F)$,

V = velocity, ft/sec,

W = weight flow of hot fluid, lbs/hr,

W_t = modified rate of flow lbs/hr/tube,

Z = height in ft,

w = weight flow of cold fluid, lbs/hr,

μ = viscosity, centipoises x 2.42 lb/(ft)(hr),

μ_w = viscosity at tube wall temperature, centipoises x 2.42 lbs/(ft x hr),

ρ = density, lb/ft^3,

ϕ = $(\mu/\mu_w)^{0.14}$,

$\digamma$ = correction factor for convection--flow in tubes at low Re, dimensionless,

β = coefficient of thermal expansion 1/$^{\circ}$F,

λ = latent heat of condensation, Btu/lb,

Γ = mass velocity of condensing vapor, lbs/ft of total circumferential periphery per hour.

Subscripts (except as noted above):

S = shell side,

t = tube side,

v = vapor,

a,b,c,etc. = designated material, a, b, c, etc.,

c = condensate,

l = liquid,

x = cross tube flow (shell side),

L = longitudinal tube flow (shell side) or liquid,

m = mean value,

f = film, film temperature, or fouled condition.

B. Heat Exchange Terminology

1. General

The terminology for heat exchanger parts is indicated in Figure II-1 and II-2 illustrating typical single and two shell-pass exchangers respectively. The ASME and API-ASME have established a safety code for the design specifications of various exchanger parts and these specifications are presented in the Standards of TEMA, New York.

2. Heat Exchanger Shell

The two major parts of this type of heat exchanger are (i) the shell and (ii) the tube-bundle or tube-bank. The shell size is goverened by the size of the tube-bank and the nature of the performance of the equipment; for example, a larger modified shell is required if the fluid vaporizes as in reboilers. Shell diameter and shell thickness are specified according to the duty and the material of construction. Shell-nozzle connections and materials of constructions are specified by TEMA. A nomograph for the size of nozzles is given on Figure 1 of Part V*for steam, with steam gauge pressure as parameter. Knowing the steam flow, the size of nozzle can be obtained for predetermined velocity. Following is a table (Table-II-1) which gives nozzle capacities for various fluids. Maximum steam velocities are recommended as given in Table II-2.

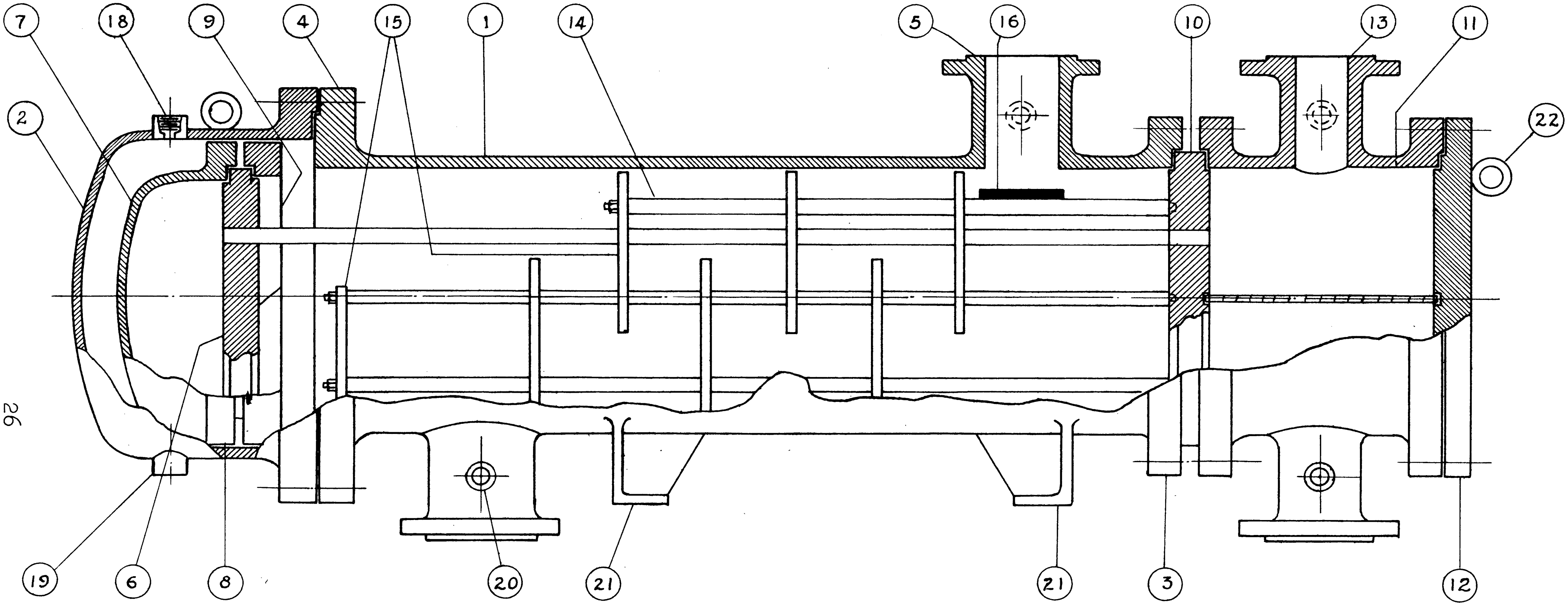
TABLE II-1
NOZZLE CAPACITIES lbs/hr

Size	Area Sq In.	Oil >150 S.S.U. (at 2'/sec)	Oil <150 S.S.U. (at 4'/sec)	Water (ΔP = 11.2'/100')	Condensate (at 1'/sec)
1/2"	0.304	858	1,715	630	475
3/4"	0.533	1,500	3,000	1,500	833
1 "	0.864	2,400	6,880	2,900	1,350
1-1/2"	2.036	5,740	11,480	9,000	3,181
2 "	3.355	9,460	18,920	19,000	5,242
2-1/2"	4.788	13,500	27,000	30,000	7,481
3 "	7.393	20,900	41,800	50,000	11,550
4 "	12.730	35,900	71,800	110,000	19,889
5 "	20.000	56,400	112,800	210,000	31,248
6 "	28.890	81,500	163,000	325,000	45,138
8 "	50.030	141,000	282,000	700,000	78,167
10 "	78.850	222,000	444,000	1,500,000	123,194
12 "	113.090	319,000	638,000	1,900,000	176,691

TABLE II-2
MAXIMUM STEAM VELOCITIES

^{o}F, Super Heat	0	50	100	150
ft/sec	133.5	162.0	150.0	158.4

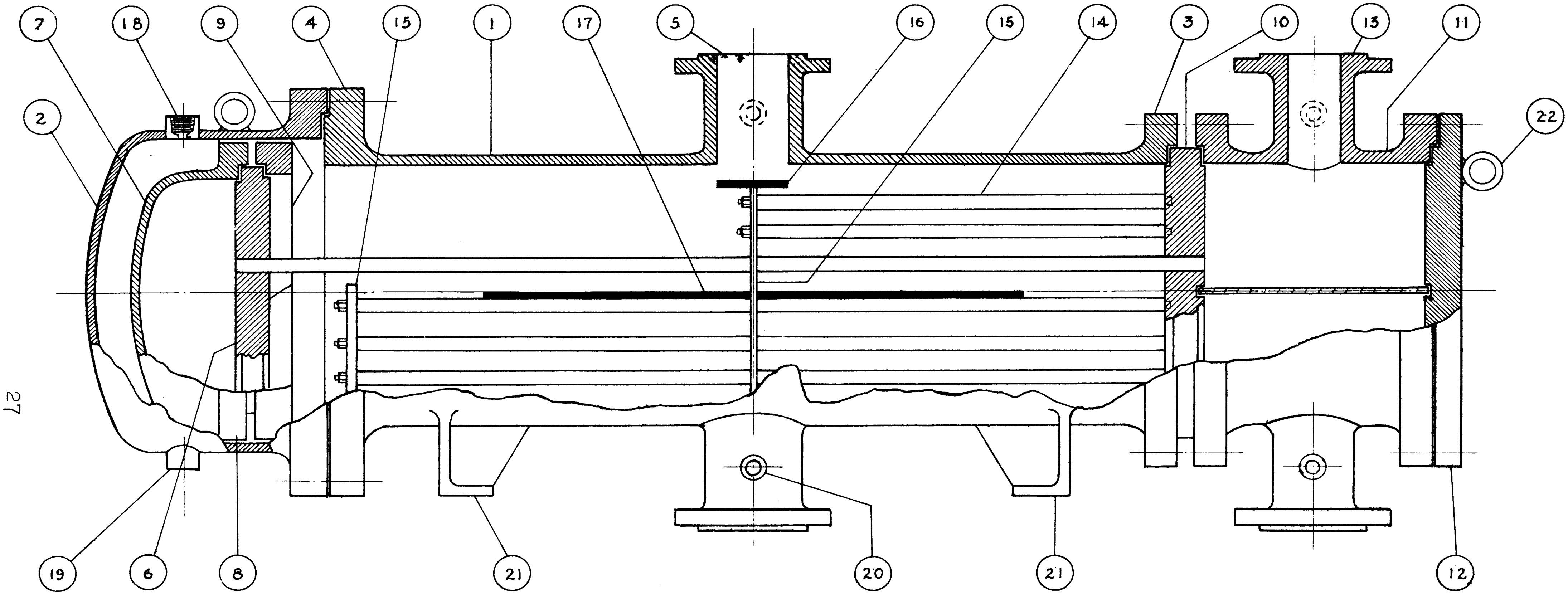
*Figures designated by Arabic numerals appear only in Part V.



NOMENCLATURE OF TYPICAL HEAT EXCHANGER PARTS

1. SHELL
2. SHELL COVER
3. SHELL CHANNEL END FLANGE
4. SHELL COVER END FLANGE
5. SHELL NOZZLE
6. FLOATING TUBESHEET
7. FLOATING HEAD
8. FLOATING HEAD FLANGE
9. FLOATING HEAD BACKING DEVICE
10. STATIONARY TUBESHEET
11. CHANNEL
12. CHANNEL COVER
13. CHANNEL NOZZLE
14. TIE RODS AND SPACERS
15. TRANSVERSE BAFFLES OR SUPPORT PLATES
16. IMPINGEMENT BAFFLE
17. LONGITUDINAL BAFFLE
18. VENT CONNECTION
19. DRAIN CONNECTION
20. TEST CONNECTION
21. SUPPORT SADDLES
22. LIFTING RING

Figure II-1



NOMENCLATURE OF TYPICAL HEAT EXCHANGER PARTS

1. SHELL
2. SHELL COVER
3. SHELL CHANNEL END FLANGE
4. SHELL COVER END FLANGE
5. SHELL NOZZLE
6. FLOATING TUBESHEET
7. FLOATING HEAD
8. FLOATING HEAD FLANGE
9. FLOATING HEAD BACKING DEVICE
10. STATIONARY TUBESHEET
11. CHANNEL
12. CHANNEL COVER
13. CHANNEL NOZZLE
14. TIE RODS AND SPACERS
15. TRANSVERSE BAFFLES OR SUPPORT PLATES
16. IMPINGEMENT BAFFLE
17. LONGITUDINAL BAFFLE
18. VENT CONNECTION
19. DRAIN CONNECTION
20. TEST CONNECTION
21. SUPPORT SADDLES
22. LIFTING RING

Figure II-2

3. Heat Exchanger Tube-Bundles

a. Heat Exchanger Tubes. Heat transfer surface required for a given duty, the tube diameter and tube length fix the number of tubes required for given duty. Standard overall tube lengths are 8, 12 and 16 feet. Heat exchanger tubes or condenser tubes differ from iron pipes in that the outside diameter of heat exchanger tubes is the same as the specified tube diameter in inches. The wall thickness is based on Birmingham wire gauge, that is, the BWG of the tube. The most common sizes of tubes used in heat exchangers are 3/4 in. and 1 in. OD tubes. Other tube sizes available are listed in Table II of Part V. Tubes of 12, 14 and 16 BWG are readily obtainable.

b. Tube Pitch. In laying out tube arrangement, the word pitch is used to designate the shortest center-to-center distance between adjacent tubes. This is the case for square or triangular tube layouts. The shortest distance between two adjacent holes in the tube sheet is termed clearance or ligament. The table below (Table II-3) gives common pitches for various tube arrangements when tubes are rolled into tube sheets. When tubes are to be back-welded, sufficient allowances must be made for the weld passes. The square and triangular pitch dimensions are shown in Figure II-3.

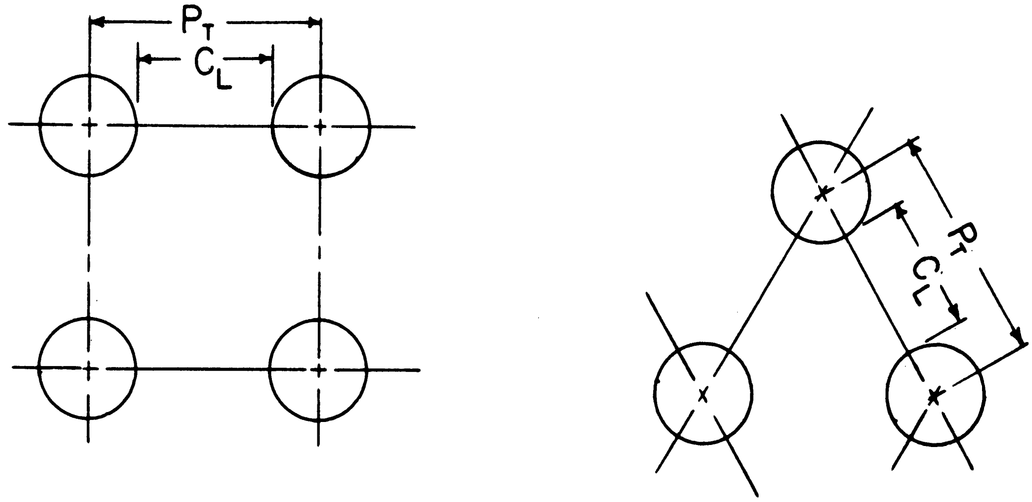


Figure II-3. Pitch and Ligament.

TABLE II-3
COMMON PITCHES FOR VARIOUS TUBE ARRANGEMENTS WHEN TUBES ARE ROLLED INTO TUBE SHEETS

No.	Tube Layout	Tube Size OD	Pitch
1	Square	3/4"	1 "
2	Square	1 "	1-1/4"
3	Triangular	3/4"	15/16"
4	Triangular	3/4"	1 "
5	Triangular	1 "	1-1/4"

c. Tube-Bundle Layout. A bundle is formed by either expanding or welding tubes in tube sheets or by packing the tubes in the tube sheets by means of ferrules. Tubes are arranged in bundles in several ways. Four of the most common arrangements are shown in Figures II-4, 5, 6, and 7. Figure II-4 shows an in-line square tube pitch. The direction of the fluid flow across the tubes is shown by the arrow. Staggered square tube pitch arrangement is shown in Figure II-5. Figures II-6 and II-7 show staggered arrangements with triangular pitches.

Exchangers with a square tube pitch arrangement are most easily cleaned on the shell-side than those which have a triangular pitch. However, the heat transfer coefficient is lower with the square pitch arrangement. TEMA has specified that the clearance between tubes should be at least one-fourth the outside diameter of the tubes and in no case less than one-fourth of an inch in order to facilitate cleaning.

d. Tube-Sheet Layout and Tube-Count. Except in single pass heat exchangers the tube sheet layout is not usually symmetric. For multipass construction, space must be provided in the layout for partitions in the channels and the head covers. Outer tube limit is the diameter free of obstruction. Tubes are laid out within outer tube limit with minimum allowance of space between the partition and adjoining tubes. The number of tubes in such a layout is called the tube count. Tube counts for various tube-sheet layouts and pitch arrangements are given in Table II-V of Part V. For a given shell diameter and a given tube pitch more tubes can be put in a single tube pass exchanger than in a multi-tube pass exchanger.

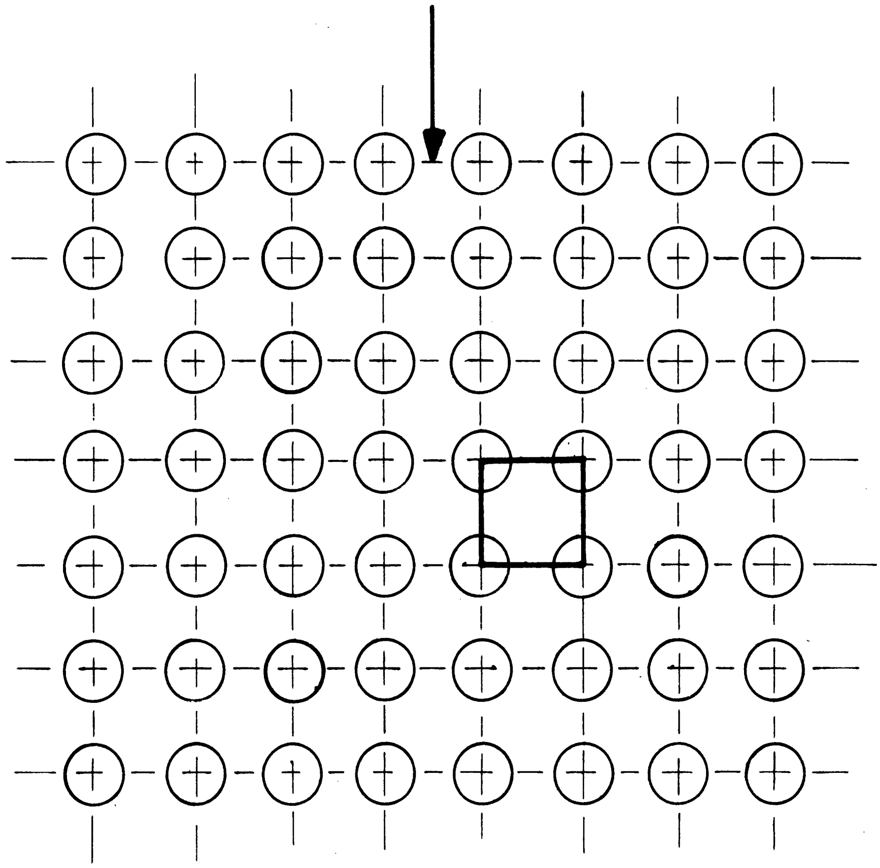


FIG. II-4
IN-LINE SQUARE TUBE-PITCH

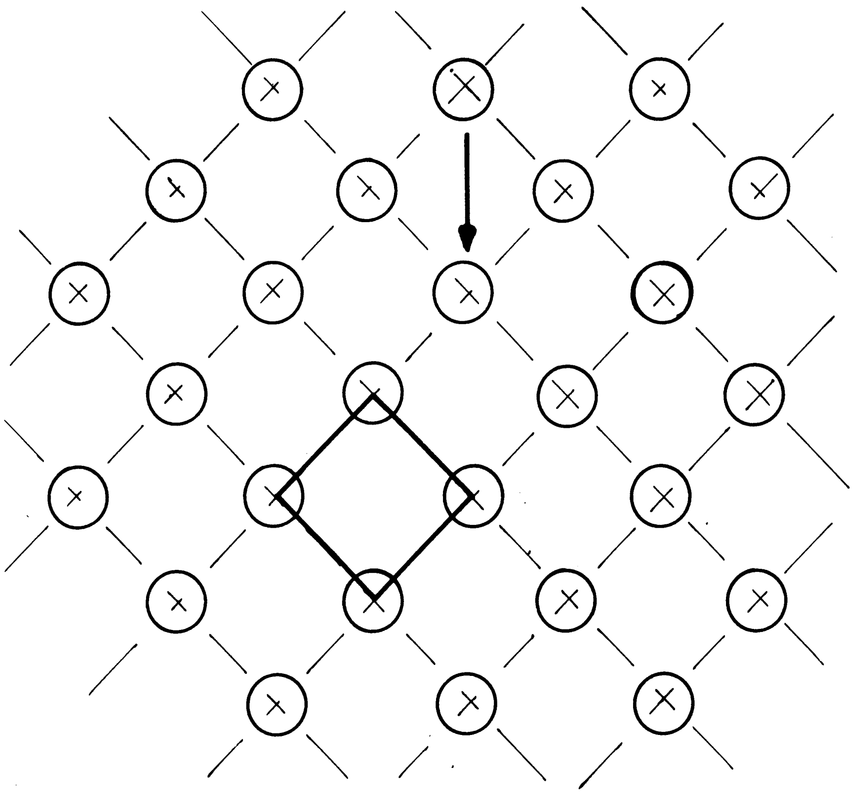


FIG. II-5

STAGGERED SQUARE TUBE-PITCH

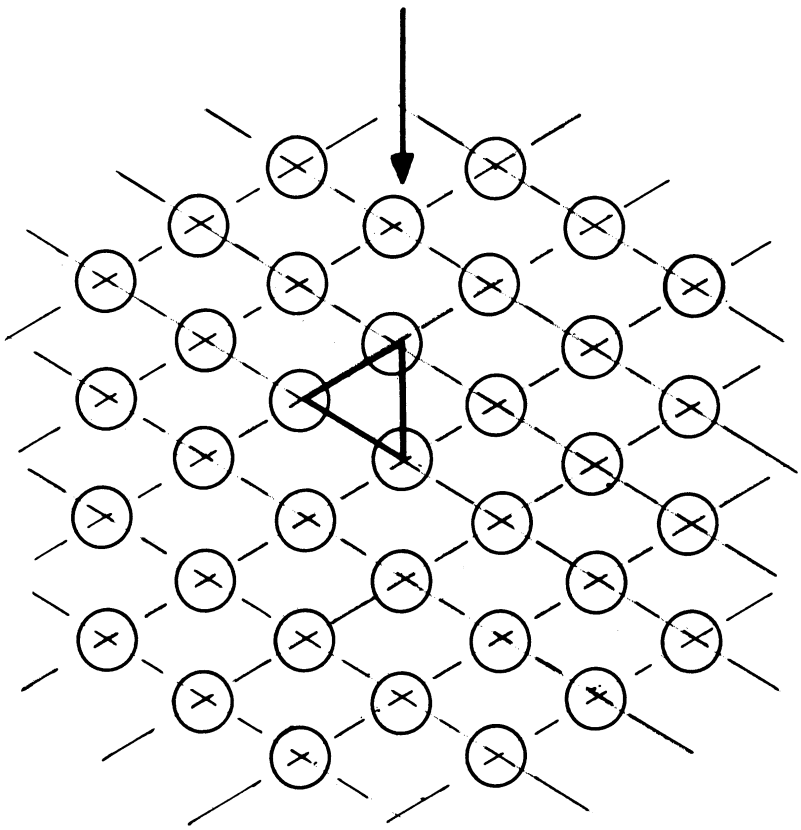


FIG. II-6

TRIANGULAR TUBE-PITCH

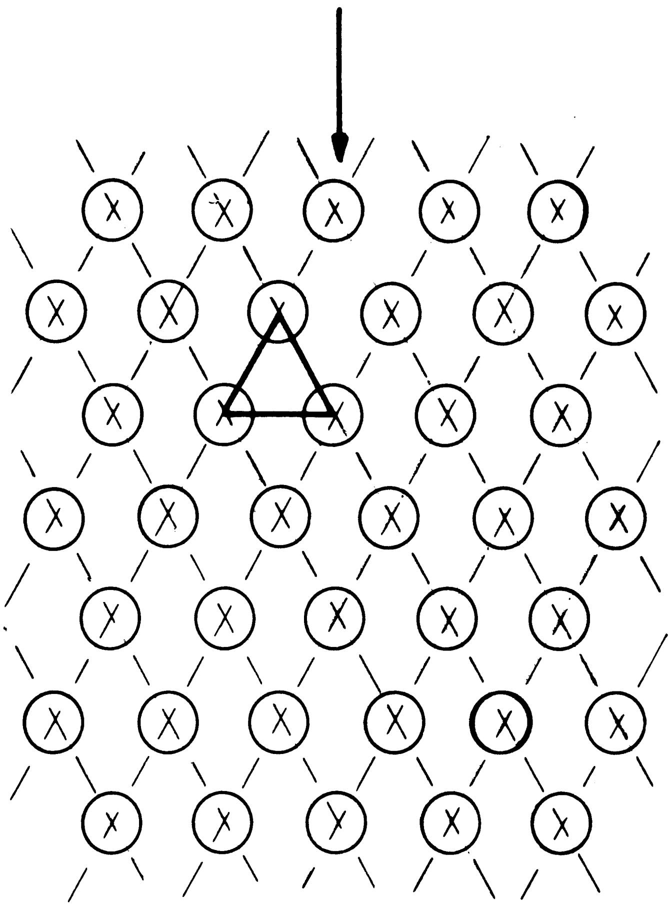


FIG. II-7

TRIANGULAR TUBE-PITCH

III. CORRELATIONS OF HEAT TRANSFER AND PRESSURE DROP DATA

A. General Considerations

1. Heat Transfers through Solid Conductors

a. Resistance Concept. Heat transfer through a solid wall is illustrated in Figure III-1. Mathematically, it can be shown, by the integrated form of the steady state heat conductance equation, as

$$Q = \frac{kA\Delta t}{L} . \tag{1}$$

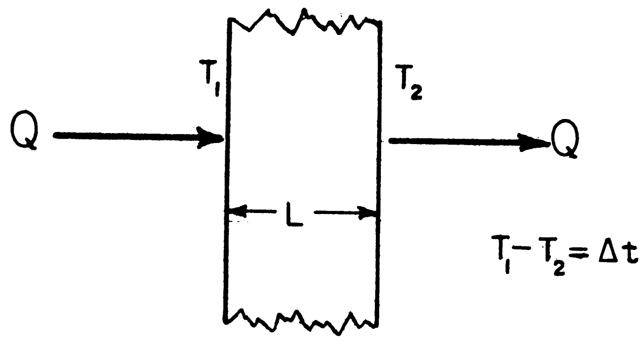


Figure III-1. Thermal Resistance.

This equation is similar to the equation for the conductance of electricity through a solid, if the reciprocal of $\frac{kA}{L}$ is called the resistance R. The equation then becomes,

$$Q = \frac{\Delta t}{R} \tag{2}$$

where Δt is the driving force and R is resistance. If heat flows through a wall consisting of different thicknesses as shown in Figure III-2, the rate of heat conductance through each material will be proportional to the resistance of each material. That is, the ratio of the total temperature across each material to its resistance must be the ratio of the total temperature difference to the total resistance or

$$Q = \frac{\Delta t}{\sum R} = \frac{\Delta t_a}{Ra} = \frac{\Delta t_b}{Rb} = \frac{\Delta t_c}{Rc} . \tag{3}$$

For a system, using actual temperatures at interface and surface,

$$Q = \frac{\Delta T}{\sum R} = \frac{T_1 - t_1}{R_a} = \frac{t_1 - t_2}{R_b} = \frac{t_2 - T_2}{R_c} \tag{4}$$

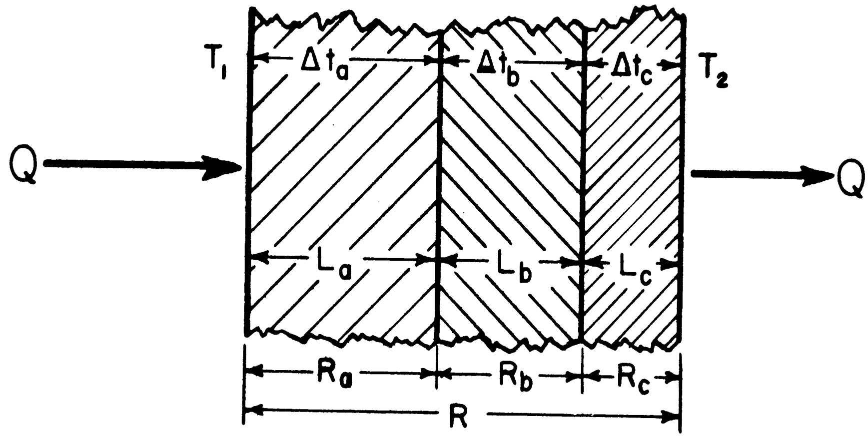


Figure III-2. Resistances in Series

where t_1 and t_2 are temperatures at the interface of a-b and b-c, respectively. Rearranging and substituting R = L/KA into equation (4),

$$Q = \frac{\Delta T}{\sum R} = \frac{T_1 - T_2}{\frac{L_a}{R_a A} + \frac{L_b}{R_b A} + \frac{L_c}{R_c A}} . \qquad (5)$$

b. Area of the Heating Surface. When heat flows through the walls of a cylinder, such as a pipe, as illustrated in Figure III-3, the area at right angles to the flow of heat decreases. It can be shown mathematically that that the logarithmic mean area, A_m, is the correct average value to use. The logarithmic mean area is

$$A_m = \frac{A_2 - A_1}{2.3 \log (A_2/A_1)} \qquad (6)$$

where A_1 and A_2 are areas for surface having radius r_1 and r_2, respectively. In terms of the radii of the two areas, the log mean area is

$$Am = \frac{2\pi r_2 - 2\pi r_1}{2.3 \log (2\pi r_2/2\pi r_1)} \qquad (7)$$

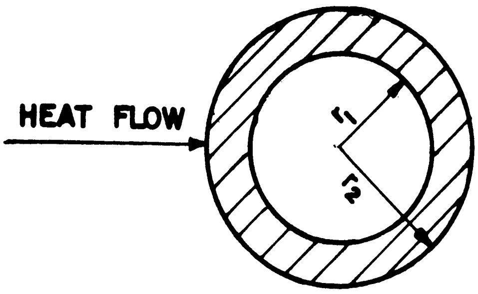


Figure III-3. Cylindrical Thermal Resistance.

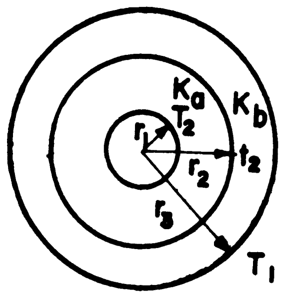


Figure III-4. Cylindrical Resistances in Series.

When heat is transferred through the walls of concentric cylindrical surfaces of varying composition and thickness, as shown in Figure III-4, the amount of heat transferred per unit length of cylinder is

$$Q = \frac{\Delta t}{\sum R} = \frac{T_1 - T_2}{(1/k_a A_{am}) + (1/k_b A_{bm})} \quad . \tag{8}$$

2. Mean Temperature Difference

The temperature difference between a heat source and a heat receiver is the driving force by which heat is transferred. The rate of heat transfer is directly proportional to this driving force; therefore, the determination of the correct temperature difference existing for a given set of heat transfer conditions is important, when calculating the required heat transfer surface.

In the case of counter or parallel flow of both liquids, the correct mean temperature difference is the logarithmic mean temperature difference. This term is commonly abbreviated as the LMTD, and is defined as

$$\text{LMTD} = \frac{\Delta t_2 - \Delta t_1}{2.3 \log (\Delta t_2/\Delta t_1)} \tag{9}$$

where t_1 and t_2 are defined as

$$T_1 \ \underrightarrow{\text{hot fluid}}\ T_2$$

$$t_2 \ \underleftarrow{\text{cold fluid}}\ t_1$$

$$\Delta t_1 = T_2 - t_1 \quad ; \quad \Delta t_2 = T_1 - t_2 \ .$$

The true temperature difference in a 1-2 exchanger, however, is not given by equation (9). This situation arises because the flow in such an exchanger is a combination of counter current and parallel flows. It becomes even more complicated when more tube-passes and multiple shell-passes are used. Curves on Figures 2-7 in Part V give factors for correcting the LMTD for various types of tube-side and shell-side flows.

3. Film and Overall Transfer Coefficients

The summation of individual resistances is called the overall resistance. In an exchanger heat is transferred from hot fluid to the cold fluid through the tube-wall. The coefficient of heat transfer from tube-surface to a fluid or the film coefficient is defined as h in the equation

$$Q = hA\,(t_w - t_a) \ . \tag{10}$$

Taking into consideration the film coefficients of fluids inside and outside the tube the total resistance can be obtained as

$$\sum R = \frac{1}{h_i A_i} + \frac{Lm}{K_m A_m} + \frac{1}{h_o A_o} \quad . \tag{11}$$

The clean overall coefficient of heat transfer, U_c, is defined as

$$\frac{1}{U_c A_o} = \sum R = \frac{1}{h_i A_i} + \frac{Lm}{K_m A_m} + \frac{1}{h_o A_o} \tag{12}$$

or approximately,

$$\frac{1}{U_c} = \frac{1}{h_i} + \frac{Lm}{K_m} + \frac{1}{h_o} \quad . \tag{13}$$

For steady state heat transfer equation (3) gives

$$Q = \frac{\Delta t}{\sum R} \quad .$$

Equation (13) can be substituted in equation (3) so that

$$q = U_c \Delta t \tag{14}$$

where q is heat transferred per unit time per unit area, or

$$Q = U_c A \Delta t \quad . \tag{14a}$$

In most heat transfer work the resistance due to pipe wall is negligible and equation (12) reduces to

$$\frac{1}{U_c} = \frac{1}{h_i \; (A_i/A_o)} + \frac{1}{h_o} \quad . \tag{15}$$

Inasmuch as the areas of the inside and the outside of an exchanger tube are different, h_i and h_o must be referred to the same heat flow area. If the outside area A_o of the pipe or tube is used, h_i must be multiplied by (A_i/A_o) to give the value that h_i would have, if it were calculated originally on the basis of the larger area, A_o.

4. Heat Balance

The total amount of heat transferred through the surface of a heat exchanger is

$$Q = WC_p \; (T_1 - T_2) = wc_p \; (t_2 - t_1), \text{ Btu/hr} \tag{16}$$

or combining with equation (14a)

$$Q = WC_p \; (T_1 - T_2) = wc_p \; (t_2 - t_1) = U_c A \Delta t \; . \tag{16a}$$

5. Shell-Side Mass Velocity

In shell and tube-type heat exchangers, both the fluids are confined. One flows through the tubes and the other flows outside the tube through the shell of the exchanger. Most of the film coefficient and pressure drop correlations are based on the mass velocity of the fluids on the tube side or on the shell side. While only one mass velocity is encountered on the tube-side, two different mass velocities exist on the shell-side of an exchanger. One mass velocity is based on the flow of fluid across the tubes between the baffles and the other is based on the flow longitudinal to the tubes through the segmented baffle openings. These flows are called cross and longitudinal flows respectively. On this basis the cross area, A_x, is given by

$$A_x = \frac{d_s\ (P_T - d_o)\ B}{144 P_T}\ \text{sq ft}\ . \tag{17}$$

Therefore, the mass velocity for cross flow is

$$G_x = \frac{W}{A_x}\ . \tag{18}$$

The curves plotted with shell diameter as parameter on Figures 37-41 of Part V, give values of A_x per inch of baffle pitch. These curves, based on equations (19, 20, 22, 23) adopted from private communications, make slightly different assumptions than equation (17); however, they appear to give comparable values. Figures 37-39 are based on equation

$$A_x = \frac{K_1/2\ [(\theta - \pi/2) - 1/2\ \text{sine}\ 2\theta] - K_2 \cos\theta}{N_b} \tag{19}$$

where

A_x = free area per inch of baffle pitch - sq in.,

θ = arc cos $[0.866\ (P_T - d_o)/\ r]$,

P_T = tube-pitch - in.,

N_b = number of tube rows past condenser C_L,

K_1 = $[-1.732r^2\ (P_T - d_o)]/(0.866P_T)^2$; $K_2 = rd_o/0.866P_T$

r = radius of exchanger shell, in.,

d_o = outer tube diameter, in.

Figures 40 and 41 are based on equation

$$A_x = \frac{K_1}{4N_b}\ (\pi - 2\theta + \sin 2\theta) \tag{20}$$

where

θ = arc cos $[0.866\ (P_T N_b)]/r$,

K_1 = $[4r^2\ (P_T - d_o)]/P_T^2$.

A_x, N_b, P_T, d_o and r have the same significance as above. The longitudinal free area A_L is the total cross-sectional area of the tubes in the segmented baffle cut, subtracted from the area of the segmented baffle cut. The curves on Figures 42 to 55 of Part V give approximate values for these areas for different baffle cuts and shell diameters.

Therefore, the mass velocity of longitudinal flow is

$$G_L = \frac{W}{A_L} . \tag{21}$$

Figures 42-52 are based on the geometry of the tube arrangements, for horizontal baffle cuts and vertical flow and vice versa for an exchanger having full floating head. Figure 53 has curves for approximate free area for longitudinal flow for triangular pitch based on equation

$$A_L = r^2\left(\theta_1 - \frac{\sin\ 2\theta_1}{2}\right) - \frac{\pi d_o^2}{4}\left[\frac{r^2}{0.866P_T^2}\left\{\left(\theta_1 - \frac{\sin 2\theta_1}{2}\right) - \left(\theta_2 - \frac{\sin 2\theta_2}{2}\right)\right\}\left(\frac{r}{0.866P_T - N_b}\right)\right] \tag{22}$$

where

A_L = net free area for longitudinal flow - in.^{2},

θ_1 = arc cos $(0.866\ P_T N_b)/r$,

θ_2 = arc cos $(1 - \cos\ \theta_1)/N_b$,

d_o, r, P_T and N_b have same significance as in Part II-A.

Figures 54 and 55 are for square pitch and are based on equation

$$A_L = r^2\left[\left(\theta_1 - \frac{\sin\ \theta_1}{2}\right) - \frac{\pi d_o^2}{4P_T^2}\left\{\left(\theta_1 - \frac{\sin\ \theta_1}{2}\right) - \left(\theta_2 - \frac{\sin\ 2\theta_2}{2}\right)\right\}\right] \tag{23}$$

where

$$\theta_1 = \text{arc cos}\left(\frac{0.707\ P_T N_b}{r}\right)$$

$$\theta_2 = \text{arc cos}\left(1 - \frac{\cos\ \theta}{N_b}\right) .$$

A_L, r, d_o, P_T, and N_b have the same significance as in Part II. Curves on Figures 53-55 have shell diameter as parameter.

6. Shell Side Equivalent Diameter

Where the range of baffle pitch is restricted between the inside diameter and one-fifth the inside diameter of the shell, excellent correlation of flow in an exchanger is obtained if the hydraulic radius is calculated parallel with instead of at right angles to the long axis of the tubes. The equivalent diameter of the shell is then taken as four times the hydraulic radius.

Figure III-5 shows a diagram of a square tube pitch. The cross-hatched area represents the cross-sectional flow area and the heavy lines on the tubes represents the wetted perimeter.

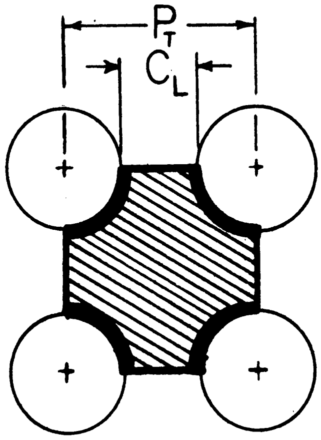


Figure III-5. Wetted Perimeter for Square Tube Pitch.

Inspection of this sketch shows that

$$D_e = \frac{4X \text{ free area}}{\text{wetted perimeter}}$$

or

$$D_e = \frac{4\left(P_T^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o}. \qquad (24)$$

A 60^o triangular V-pitch tube arrangement is shown in Figure III-6. The cross-hatch area represents the cross-sectional flow, while the heavy lines on the tube perimeter represents the wetted perimeter.

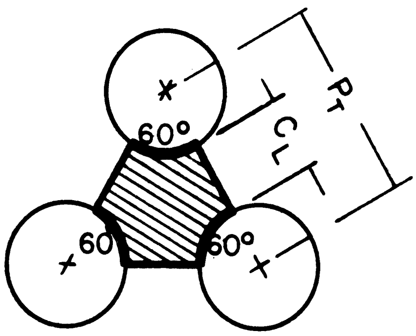


Figure III-6. Wetted Perimeter for 60^o Triangular V-Pitch.

As before,

$$D_e = \frac{4X \text{ free area}}{\text{wetted perimeter}}$$

or

$$D_e = \frac{8\ (0.43\ P_T^2 - \pi d_o^2)}{\pi d_o} \qquad (25)$$

B. Correlations for Film Coefficients

1. Shell-Side Film Coefficients

Kern,[35]* has given an equation for calculating shell side film coefficients based on the cross-flow only, using the maximum area corresponding to the center of the shell. This equation is based on the correlation of industrial data and gives satisfactory results for hydrocarbons, organic compounds, water, aqueous solutions and gases when baffles, with clearances between baffles and tubes and baffles and shells as specified in the Standards of TEMA, are employed. For Reynolds' number from 2,000 to 1,000,000, the data are closely represented by the equation

$$\frac{h_o D_e}{k} = 0.36 \frac{(D_e G_x)^{0.55}}{\mu} \left(\frac{C_p \mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} . \qquad (26)$$

This correlation is represented in Figure 8 of Part V. Figures 9-18 except Figure 12 in Part V are plots of correlation of industrial data on heat transfer coefficients. Following Table III-1 indicates the use of these plots in respective cases. Figure 12 is a nomograph for heat transfer coefficients for gases across banks of tubes based on the equation by Colburn.[27]

$$h = (\text{constants}) \left(\frac{Cp^{1/3}\, k^{2/3}\, G^{0.8}}{\mu^{0.27}\, (do)^{0.4}}\right) . \qquad (27)$$

2. Tube-Side Film Coefficients

The recommended equations for calculating tube film coefficients for both heating and cooling against liquids, aqueous solutions and gases are given below.

a. Stream-Line Flow (Re < 2100).

$$\frac{LD}{k} = 0.186 \left(\frac{\mu}{\mu_w}\right)^{0.14} \left[\left(\frac{DG}{\mu}\right)\left(\frac{Cp\mu}{R}\right)\left(\frac{D}{L}\right)\right]^{1/3} . \qquad (28)$$

b. Turbulent Flow (Re > 10,000).

$$\frac{LD}{k} = 0.027 \left(\frac{\mu}{\mu_w}\right)^{0.16} \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{Cp\mu}{R}\right)^{1/3} \qquad (29)$$

These equations are represented graphically on a single pair of co-ordinates by plotting

$$J_\mu = \left(\frac{h}{C_p G}\right)\left(\frac{Cp\mu}{R}\right)^{2/3} \left(\frac{\mu_w}{\mu}\right)^{0.14}$$

against (DG/μ). Figure 19, Part V is such a plot with L/D ratio as parameter for streamline zone. Curves in transition zone are based on observed data.

*Superscribed numerals refer to serial numbers in the Bibliography.

TABLE III-1

TABLE OF PLOTS ON SHELL-SIDE FILM COEFFICIENTS
BASED ON INDUSTRIAL DATA ADOPTED FROM PRIVATE COMMUNICATIONS TO AUTHORS

Fig. Nos.	Title	Nature of Plot and Parameters	Correction Factors	Equation for Actual Rate	Remarks
9	Heat transfer rate for liquids across banks of tubes.	G_x' against h, with μ in C.P. as parameter.	For specific heat, F_{C_p} For conductivity, F_k For tube diameter, F_D For staggered tube arrangement, F_p For viscosity, $(\mu/\mu_w)^{0.14}$ Correction factors from Figure 9a.	$h_x = h \times F_{C_p} \times F_k \times F_D \times F_p \times (\mu/\mu_w)^{0.14}$	$F_p = 0.780$ for staggered tube arrangement. μ at average fluid temperature.
10	Heat transfer rate for gases across banks of tubes.	G_x' against h, for triangular and rectangular pitch.	F_{C_p}, F_k, F_μ given on Figure 10-a.	$h_x = h \times F_{C_p} \times F_k \times F_\mu$	
11	Heat transfer rate for gases and vapors in cross flow.	G_x' against h, with μ in C.P. as parameter.	F_{C_p}, F_k, F_D given on Figure 11-a.	$h_x = h \times F_{C_p} \times F_k \times F_D$	
13	Heat transfer rates for water in transverse flow in banks of tubes.	G_x' against h, with average fluid temperature as parameter.	Tube diameter and tube pitch as tabulated on the Figure 13-a.	$h_x = h \times F_D \times F_p$	
14	Heat transfer rates for water in longitudinal flow on the shell side.	G_L' against h, with mean bulk temperature as parameter.	F_{D_e} as plotted in the inset on Figure 14.	$h_x = h \times F_{D_e}$.	

TABLE III-1 (cont.)

Fig. Nos.	Title	Nature of Plot and Parameters	Correction Factors	Equation for Actual Rate	Remarks
15	Heat transfer coefficients for water in shell with orifice type of baffles.	V_o (velocity ft/sec through orifice) against h with mean bulk temperature as parameter.	For baffle spacing as tabulated on Figure 15.	$h_x = h \times F_B$	
16	Heat transfer rates for oils in cross flow.	G'_x against h, with μ in C.P. as parameter.	F_{C_p}, F_k, F_D, F_p, $(\mu/\mu_w)^{0.14}$ Correction factors are plotted on Figure 16a.	$h_x = h \times F_{C_p} \times F_k \times F_D \times F_p \times (\mu/\mu_w)^{0.14}$	$F_p = 0.780$ for staggered arrangement. μ at average fluid temperature.
17	Heat transfer rates for oils in longitudinal flow.	G'_x against h, with μ in C.P. as parameter.	F_{C_p}, F_k, F_D, $(\mu/\mu_w)^{0.14}$ Correction factors are plotted on Figure 17a.	$h_L = h \times F_{C_p} \times F_K \times F_D \times (\mu/\mu_w)^{0.14}$	μ at average fluid temperature.
18	Heat transfer rates for oils in shells with orifice baffles.	G against h with μ in C.P. as parameter.	F_B, tabulated on the Figure 18.	$h_o = h \times F_B$	

Figures 20-30 except Figure 23 are curves for tube side heat transfer rates based on industrial data. Following, Table III-2 indicates the uses of these curves in respective cases. Chilton, et al.,[26] suggests that for gases in tubes a dimensional equation gives the heat transfer rates for forced convection.

$$h_t = 13.3\, C_p \mu^{0.2} \frac{60.8}{d^{0.2}} . \tag{30}$$

Figure 23 is a nomograph based on this equation. Following example illustrates the use of this nomograph.

Example: 16,300 lbs/sq ft x hr. SO_2 at 100°C flowing through 1.06 in. ID tubes.

Align 1.06 on D-line with 16.3 on G-line to intersect reference line. The crossing on M-line gives weight flow per hour.

Along 100°C on T-line with $C_p\mu^{0.2}$-line; connect this with the point of intersection of first line with reference line. The intersection on h'line will give the film coefficient h = 9.4 (pound-calorie unit)/sq ft x hr.

3. Free Convection Heat Transfer Rates

At very low Reynolds' numbers the effects of convection become important when calculating film coefficient for flow inside tubes. A correction for free convection can be applied to equation (28) by multiplying with

$$\psi = \frac{2.25\,(1 + 0.01\,Gr^{1/3})}{\log Re} \tag{31}$$

for horizontal flow, only where $Gr = (D^3 p^2 g \beta \Delta t)/\mu^2$, the Grashof number. Grashof number is calculated from properties taken at the average fluid temperature. Rice[44] suggested that the data on heat transfer rates for free convection can be represented by the correlation,

$$h_c = \frac{k}{b} \left(\frac{C_p \mu}{k}\right)^{0.25} \left(\frac{D^3 p^2 g \beta \Delta t}{\mu^2}\right)^{0.25} . \tag{32}$$

It reputedly holds for all fluids, for both vertical and horizontal tubes, inside and outside. Figure 31 is a plot of h against Δt based on Rice's equation with viscosity as parameter. h is corrected for thermal conductivity, specific heat, density, tube diameter and thermal expansion. The actual rate is then obtained as

$$h_c = h \times F_k \times F_{C_p} \times F_p \times F_D \times F_B . \tag{33}$$

The correction factors are plotted on Figures 31a and 31b.

Figure 32 represents free convection heat transfer rates for water at various temperatures and oil of different viscosities.

TABLE III-2

TABLE OF PLOTS ON TUBE-SIDE FILM COEFFICIENTS
BASED ON INDUSTRIAL DATA ADOPTED FROM PRIVATE COMMUNICATIONS
FROM PROFESSOR H. A. OHLGREN

Fig. Nos.	Title	Nature of Plot and Parameter	Correction Factors	Equation for Actual Rate	Remarks
20	Heat transfer rate for liquids in tubes.	Re against h, L/D as parameter.	For Reynold's number, F_{Re} For Grashof number, F_{Gr} A factor $(R/D)(C_p\mu/R)^{1/3}$ $(\mu/\mu_w)^{0.14}$ Correction factors are plotted on Figure 20a.	$h_t = h \times F_{Re} \times F_{Gr} \times (C_p\mu/R)^{1/3} (\mu/\mu_w)^{0.14}$	
21	Overall transfer rate for methyl chloride-water and Freon-water coolers.	Water velocity V_t, ft/sec against U_c.	For water inlet temperature, F_{t_1} as plotted on Figure 21a.	$h_t = h \times F_{t_1}$	
22	Heat transfer coefficients for gases inside pipe or annular spaces.	G'_t against h_v/C_p with mean film temperature as parameter.	For equivalent diameter, F_{De} as plotted in the inset on Figure 22.	$h_t = (h_v/C_p) \times F_{De} (d_i/d_o)$	Curves can be extrapolated.
24 25 26	Heat transfer rates for water in tubes of different sizes and "BWG's"	Velocity, ft/sec against h, with mean water temperature as parameter.	For BWG - F and $(\mu/\mu_w)^{0.14}$ F is tabulated on each figure.	$h_t = h \times F \times (\mu/\mu_w)^{0.14}$	

TABLE III-2 (cont.)

Fig. Nos.	Title	Nature of Plot and Parameter	Correction Factors	Equation for Actual Rate	Remarks
27	Heat transfer rates for water in 3/8 in. OD and 18 BWG tubes.	V_t against h with number of tube pass as parameter.	For mean bulk temperature, F_t	$h_t = h \times F_t$	Correction factors plotted on Figure 26.
28 29 30	Heat transfer rates for oil in tubes of various sizes and "BWG's"	G against h with μ in C.P. as parameter	F_k, F_{C_p}, $(\mu/\mu_w)^{0.14}$ and for different BWG's, correction factors, F_D, F_D for "BWG" is given on respective figures. The rest are plotted on Figure 28a.	$h_t = h \times F_k \times F_{C_p} \times F_D \times (\mu/\mu_w)^{0.14}$	μ in oil velocity = 10 ft/sec. Practical value = 8 ft/sec. Recommended value = 6 ft/sec.

For free convection to air outside of tubes and pipes as well as other surfaces, McAdams[38] has summarized a number of simplified dimensionless equations

Horizontal pipes: $$h = 0.42\left(\frac{\Delta t}{D}\right)^{0.25} \tag{34}$$

Long vertical pipes: $$h = 0.4\left(\frac{\Delta t}{D}\right)^{0.25} \tag{35}$$

Vertical plates less than 2 ft high: $$h = 0.28\left(\frac{\Delta t}{Z}\right)^{0.25} \tag{36}$$

Vertical plates more than 2 ft high: $$h = 0.3\ (\Delta t)^{0.25} \tag{37}$$

Horizontal plates facing upward: $$h = 0.38\ (\Delta t)^{0.25} \tag{38}$$

Facing downward: $$h = 0.2\ (\Delta t)^{0.25}\ . \tag{39}$$

The data for free convection design are somewhat inaccurate and an ample safety factor should be included when designing equipment of this type. Chilton, et al., has developed an equation which gives conservative coefficients for a single pipe. Small error is encountered if it is used to calculate coefficients for convection outside horizontal banks of tubes. The use of this equation places restrictions on pipe spacing in relation to one another, and to the vessel bottom. In the former case, no less than one tube diameter must be used, and in the latter no less than several tube diameters must be used. This equation is as follows:

$$h = 116\left[\left(\frac{k^3\rho^2C_p\beta}{\mu}\right)\left(\frac{\Delta t}{D_o}\right)\right]^{0.25}\ . \tag{40}$$

Figure 32a is a nomograph based on this equation.

4. Film Coefficients for Gravity Flow of Liquids in Layer Form (Falling Films)

The coefficient for water in gravity flow in layer form inside tube is given by the equation

$$h = 120\Gamma^{1/3} = 120\left(\frac{W}{\pi D}\right)^{1/3}\ . \tag{41}$$

Data for other liquids are not available, but the following dimensionless equation may be used for estimating h:

$$h = 0.01\left(\frac{C_p\mu}{R}\right)^{1/3}\left(\frac{4\Gamma}{\mu}\right)^{1/3}\left(\frac{k^3\rho^2 g}{\mu^2}\right)^{1/3}\ . \tag{42}$$

5. Film Coefficients for Condensing Vapors

a. Horizontal Tubes. For film-type condensation of a pure saturated vapor outside of a vertical tier of N_t horizontal tubes

$$h = 0.725\left(\frac{k^3\rho^2 g\lambda}{N_v D\mu\Delta t}\right)^{1/4} = 0.95\left(\frac{k^3\rho^2 g\pi D_o}{\mu_W}\right)^{1/3}. \quad (43)$$

b. Vertical Tubes. For vertical tubes, where $4\Gamma/\mu$ is less than 2100 the following equation is recommended.

$$h = 1.13\left(\frac{k^3\rho^2 g\lambda}{L_v\mu\Delta t}\right)^{1/4} = 1.11\left(\frac{k^3\rho^2 g\pi D_o}{\mu_W}\right)^{1/3}. \quad (44)$$

Figures 33, 34, and 35 give heat transfer rates for film-type single phase condensation. Figures 33 and 34 are based on Nusselt's equations mentioned above. Figure 34a gives correction factors for these plots. Figure 33 is a plot of W_t, a rate defined on the graph, for various tube arrangements against h for horizontal tubes. The curve reading has to be multiplied by correction factors for conductivity, specific heat, tube length and number of tubes in horizontal row as given in Figure 34a.

Figure 34 gives heat transfer rates for condensation inside or outside vertical tubes. The actual rate is obtained after applying correction factors for conductivity, specific heat, and stream line or turbulent flow factors for tube diameter as given on Figure 34a.

Kirkbride and Colburn[37] have correlated data on condensation of vapors on vertical tubes respectively. Figure 35 is a plot of h against $4\Gamma/\mu$ based on their data, so that $h = h_c\,[\mu^2/k^3\rho^2 g]^{1/3}$ where h_c is condensing coefficient.

6. The Caloric Fluid Temperature

The derivation of the equation for calculating the LMTD is based on a number of assumptions. One of these assumptions is that the overall heat transfer coefficient U_c, is constant over the length of the transfer surface. This may be in error, since significant changes of fluid properties with temperature must be considered. Such a change causes the values of h_o and $(h_i\ A_i/A_o)$ to vary over the length of pipe to produce a larger U_c at the hot terminal than at the cold terminal.

If the temperature drop for each fluid going through the exchanger is sufficiently low, h_i and h_o can be calculated on the properties at the arithmetic average fluid temperature. The resulting U_c can then be used with the LMTD to determine A, or Q. If the temperature drop is too great, the fluid properties change enough from entrance to exit so that h_i and h_o must be calculated from fluid properties based on a true mean temperature. The resulting U_c can then be used with the LMTD.

Colburn[28] has derived the following equations for giving a true mean temperature, or the caloric fluid temperature:

$$F_c = \frac{\dfrac{1}{k_c} + \left[\dfrac{1}{1 - R}\right]}{1 + \dfrac{\ln\,(k_c + 1)}{\ln 1/R}} - \frac{1}{k_c} \quad (45)$$

where

$$\frac{1}{R} = \frac{\Delta t_1}{\Delta t_2} = \frac{\Delta t_c}{\Delta t_h} ; \quad R = \frac{\Delta t_h}{\Delta t_c} \tag{46}$$

and

$$k_c = \frac{U_2 - U_1}{U_1} \tag{47}$$

where

U_2 = overall heat transfer coefficient with respect to outer surface,

U_1 = overall heat transfer coefficient with respect to inner surface.

The caloric value for the hot fluid, T_c, is

$$T_c = T_2 + F_c (T_1 - T_2) \tag{48}$$

and for the cold fluid, t_c, is

$$t_c = t_1 + F_c (t_2 - t_1) . \tag{49}$$

Factor F_c can be obtained from the curves on Figure 36. The pipe wall temperature on the outside, t_w, can be calculated if calorific temperatures t_c and T_c are known. For hot fluid on the outside of the tube

$$t_w = t_c + \frac{h_o}{h_{io} + h_o} (T_c - t_c) \tag{50}$$

and

$$t_w = T_c - \frac{h_{io}}{h_{io} + h_o} (T_c - t_c) . \tag{51}$$

For hot fluid on the inside of the tube

$$t_w = t_c + \frac{h_{io}}{h_{io} + h_o} (T_c - t_c) \tag{52}$$

and

$$t_w = T_c - \frac{h_o}{h_{io} + h_o} (T_c - t_c) . \tag{53}$$

7. Fouling Factors

The change in surface condition due to deposition of dirt or scale on either side of pipe offer more resistance to the heat flux. Hence the value of the overall heat transfer coëfficient calculated on the basis of clean surface is reduced considerably and the performance of the exchanger does not come up to expectations. In order to avoid this discrepancy, a resistance called the fouling or dirt factor is introduced in anticipation of the fouling of the heat-transfer surface. The design or fouled overall heat transfer coefficient U_f on which the calculations for heating surface should be based is obtained by the relation

$$\frac{1}{U_f} = \frac{1}{U_c} + R_d . \tag{54}$$

Standards of Tubular Exchanger Manufacturers' Association, New York, (2nd Edition, 1949) include a table of fouling factors for cases commonly encountered in industry. They are given in Table I of Part V. For cases not included in the tables it is recommended that these tables may be made a basis of comparison. Quite often, a designer would set up his own tables of fouling factors based on experience.

C. Correlations of Pressure Drop Data

1. Tube-Side Pressure Drop

The frictional pressure drop due to flow through tubes can be calculated by an equation similar to Fanning's equation. The equation is

$$\Delta P_t = \frac{f \times G_F^2 \times L \times n}{2 \times g_c \times \rho \times d \times \phi t} = \frac{f \times G_F^2 \times L \times n}{5.22 \times 10^{10} \times D \times s \times \phi t} \qquad (55)$$

where f is a dimensional friction factor based on the Fanning friction factor, and

$$\phi t = \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \text{for Re} = 2100$$

$$\phi t = \left(\frac{\mu}{\mu_w}\right)^{0.25} \qquad \text{for Re} < 2100 \ .$$

The friction factors are dimensional, sq ft/sq in. to give ΔP_t in psi directly. A plot of f versus Re_t is given on Figure 56, Part V. To obtain dimensionless friction factors ordinate should be multiplied by 144.

When the fluid changes direction in an exchanger head at the end of each tube pass, an additional pressure drop ΔP_r is encountered. This pressure drop is called the return pressure drop, and is accounted for by allowing four velocity heads per tube pass. Therefore, the return losses for any fluid is

$$\Delta P_r = \frac{4 \times 62.4 \times n \times v^2}{144 \times s \times 2g} \text{ psi} \ . \qquad (56)$$

Figure 57 is a plot of one velocity head $(V^2/2g)(62.4/144)$ psi for S = 1 (i.e., water) against mass velocity. ΔP_r then can be obtained by multiplying the curve reading by a factor (4n/S). Figure 58 is a plot of a friction factor against Reynolds' number based on an equation which accounts for box-loss per pass:

$$\Delta P_t = \frac{f \times s \times V^2 \times L}{d \times \phi t} + \frac{0.0202 \times n \times V^2 \times s}{\phi t} \qquad (57)$$

where

$$\phi t = \left(\frac{\mu}{\mu_w}\right)^{0.25} \qquad \text{for Re} < 17$$

$$\phi t = \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \text{for Re} > 17 \ .$$

A correlation for pressure drops for liquids in tubes under turbulent flow based on Fanning equation for S.H. and C.I. pipes is given by

$$\Delta P_t = \left\{\frac{p \times L}{10} + B_o\right\} \frac{C}{S \times \phi_t} \qquad (58)$$

where

$$\phi_t = \left(\frac{\mu}{\mu_w}\right)^{0.14} .$$

The Figures 59 and 60 give curves of mass velocity per tube against p, tube loss per ten linear feet for two different tube sizes with viscosity in centipoises as a parameter and also of B_o against the mass velocities. The correction factor C for wall thickness are tabulated on the graph. The p-curves contain 20% safety factor. Figures 61, 62, and 63 are curves of pressure drops through tubes for gases and water respectively. Correction factors for Figure 61 are on Figure 61a, while those for Figure 62 are on the graph itself. Figure 63 has curves based on the equation

$$\Delta P_t = \frac{(L + 55d)\ np}{d^{1.24}} \qquad (59)$$

where p is the curve reading. p is plotted in Figure 63 against velocity in ft/sec. The curves are useful for tubes of diameter 5/8 in. to 1-1/2 in. The curves are given for average temperature of water given by the relation:

$$\text{Average Temperature } {}^{\circ}F = [(\text{Steam Sat. Temp.})--(\text{M.T.D.})]\ {}^{\circ}F \qquad (60)$$

changes in velocity cause pressure drop.

1. Enlargement $$\Delta P_e = \frac{(V_1 - V_2)^2}{2g} \qquad (61)$$

2. Contraction $$\Delta P_c = \frac{K_1 V_1^2}{2g} \qquad (62)$$

3. Bends $$\Delta P = \frac{K_2 V^2}{2g} . \qquad (63)$$

Values of K_1 and K_2 can be obtained from Figure 64. Subscripts 1 and 2 refer to smaller and larger sections.

2. Shell-Side Pressure Drop

A correlation has been obtained for determining the pressure drop in the shell side of a heat exchanger, by using the product of the inside diameter of the shell in Feet D_s and the number of times the tube bundle is crossed (N + 1). The equivalent diameter is the same as that used in calculating heat transfer coefficients on the shell-side. The pressure drop of a fluid being heated or cooled including entrance and exit losses is

$$\Delta P_s = \frac{f G^2 D_s (N + 1)}{2g \times p \times D_e \times \phi} = \frac{f G^2 D_s (N + 1)}{5.22 \times 10^{10} D_e \times s \times \phi} \text{ psi} \qquad (64)$$

f is a dimensional friction factor. It is plotted on Figure 65, Part V. To obtain consistent friction factors multiply the ordinate by 144. Figures 66, 67, and 68 are plots of pressure drop for fluids in cross flow on the shell side. Figure 66 is a plot of pressure drop per ten rows of tubes per ten cross bank passes against cross-flow mass velocity. Actual pressure drop is obtained by applying necessary correction factors. The pressure drop is obtained by the relation:

$$\Delta P_s = \frac{\text{curve reading}}{\text{sp gr}} \times \frac{\text{rows of tubes}}{10} \times \frac{\text{No. of passes}}{10} \times \begin{array}{c}\text{Correction}\\ \text{Factors}\end{array} . \tag{65}$$

The correction factors are tabulated in the figure. Figure 67 is a curve of pressure drop for liquids in cross-flow on the shell-side. It also is a plot of pressure drop per ten rows per ten passes. The actual pressure drop is obtained by applying necessary correction factors. The pressure drop is then given by the relation:

$$\Delta P_s = \frac{\text{curve reading}}{\text{sp gr}} \times \frac{\text{rows}}{10} \times \frac{\text{No. of passes}}{10} \times \left(\frac{\mu}{\mu_w}\right)^{-x} \times \text{correction factors} \tag{66}$$

Correction factors are tabulated on the figure itself. The curves on Figure 68 are based on

$$\Delta P_s = \frac{1/2\ f A G_x^2\ NB}{p\,\frac{(d_s)}{(d_p)}} \tag{67}$$

with d_s/d_e as parameter. Pressure drop for liquids in longitudinal flow across a tube bundle baffle in the form of a segment of a circle can be obtained by equation

$$\Delta P = \frac{(G_L')^2 \times N}{640 \times \rho} . \tag{68}$$

Figure 69 gives a curve of G_L' against a factor F based on the equation (68) such that

$$\Delta P = \frac{F \times N}{s} . \tag{69}$$

Professor O. P. Bergelin et al.[21,22,23] at the University of Delaware are conducting investigations on heat transfer and pressure drop data for flow across tube banks. It is hoped that these investigations will lead to general correlations covering all variables involved.

3. Pressure Drop for Condensing Vapors

a. Tube Side Condensation.

$$\Delta P_t = \frac{1/2\ f \times G_t^2 \times L \times n}{522 \times 10^{10} \times D_e \times S} . \tag{70}$$

b. Shell-Side Condensation.

$$\Delta P_s = \frac{1/2 \; f \times G_s^2 \times D_s \times (N + 1)}{522 \times 10^{10} \times D_e \times S} . \qquad (71)$$

Figure 70 gives correction factors for tube side condensation in horizontal tubes. The pressure drop is first calculated considering only the uncondensed vapors. The actual pressure drop is obtained by applying the correction factors.

IV. ILLUSTRATIVE HEAT EXCHANGER CALCULATIONS

A. Procedure

A convenient way of setting up heat exchanger calculations is outlined below.

1. Services

The services for each heat exchanger should be mentioned. All the process variables should be listed.

2. Data

All the available data on the process variables are listed and pertinent calculations to adapt the data to the design requirements should be made.

3. Determination of Inlet and Outlet Temperature

Frequently it is necessary to determine the inlet and outlet temperatures of the hot and cold fluids. This may involve a complete heat balance in cases of condensers. Also, dew-point and bubble-point calculations must be made and a condensing curve plotted. When water is used as one of the fluids a rise in temperature must be assumed. Usually, 110^{o}F is the maximum water outlet temperature used.

4. Duty

This includes the total heat load on the exchanger. It may be the sensible heat of the liquid or gas and the latent heat of phase change. All three may be involved in a single piece of equipment. The total heat duty should be summarized here.

5. Log Mean Temperature Difference (LMTD)

Knowing the inlet and outlet temperatures of the fluids involved theoretical LMTD is determined by equation (9).

A correction factor for multitube passes - multishell passes should be applied using Figures 2-7 in Part V.

6.i. Estimated Surface

Calculate the required heat transfer surface based on the calculated duty, the LMTD, and an assumed overall heat transfer coefficient.

To begin with, assumption of overall transfer coefficient can be based on the tables of miscellaneous overall coefficients given in Table X, Part V. They are adopted from Perry's Chemical Engineer's Handbook (3rd edition), pp. 480-482.

After choosing a unit which appears to be satisfactory, a check on transfer rate should be made considering tube and shell side film coefficients. Tables II to IX in Part V are used to select a standard satisfactory unit.

6.ii. Check on Transfer Rate

a. Tube-Side Film Coefficient. The mass velocity, G_t, as lbs per hour per square foot of flow area for the tube side of the exchanger is calculated. On referring to the appropriate curves in Part V, the tube-side clean heat transfer rate can be determined after applying correction factors wherever necessary as mentioned in Part III.

To account for fouling, a fouling factor given in Table I, Part V is used to convert the clean rate to the fouled rate. However, a factor of 0.8 is normally used.

b. Shell-Side Transfer Rate. The transfer rate in the shell is based on cross tube flow and longitudinal or "long flow". A coefficient is calculated for both, the cross flow and "long flow". An average coefficient, based on cross flow and long flow is used for design. The mass velocities for cross flow and long flow are determined from equations (18) and (21) using appropriate curves for determining flow area.

Again a fouling factor, obtained from Table I, Part V, is used to convert the clean rate to fouled rate. A normal value of the factor used is 0.8.

c. The Overall Transfer Rate. The overall transfer rate is calculated using equation (12) or (13).

7. The Optimum Overall Transfer Rate

The procedure described in the previous paragraph involves the assumption of overall heat transfer coefficient and involves a trial and error solution. Another approach would be to make a guess based on experience regarding the tube side and shell side velocities. Maximum velocity on the tube side is, generally, taken as 10 ft/sec. The shell side mass velocity should not exceed about 180 lbs/sq ft/hr. If the pressure drop on the shell side is negligible, say less than 0.5 psi and the tube side film coefficient is controlling the heat transfer the unit should be optimized on the basis of the tube side velocity and the film coefficients calculated. If shell side film coefficient is controlling, the unit should be optimized for shell side pressure drop. As far as possible a pressure drop of 10 psi should not be exceeded. If the film coefficients on both sides are controlling however, an assumption of overall transfer rate will be in order for calculating heat transfer surface and a check made as outlined in previous paragraphs.

8. Pressure Drop Calculations

At this point the pressure drops for both sides of the exchanger should be calculated. The allowable pressure drops for liquids flowing through an exchanger should be consistent with the available head, and a balance between the cost of increasing this head and the additional heat transfer should be obtained. When a heat exchanger is used in a system operating under vacuum, the pressure drop through the exchanger must be carefully analyzed. Under a vacuum, the pressure drop for a given mass velocity becomes greater as the pressure is reduced.

However, when a gas is operated at a high pressure, a large mass velocity can be used without obtaining impossible pressure drops.

If the resulting pressure drops are too high the design should be revised to get adequate heat transfer surface as well as allowable pressure drops.

9. Summary of Design

A complete summary of the exchanger design is made as in Table IV-1 and a specifications sheet prepared as given in Figure 71, Part V.

B. Illustrative Problems

Five illustrative problems are presented. Most of the problems are based on specific cases and the exchangers were actually built as detailed in this problem. These problems are presented to illustrate the method of designing heat exchangers for some of the more typical heat transfer problems in industry. In addition, methods are shown for using the curves, charts and tables in Part V.

The types of heat transfer problems presented are:

1. liquid to liquid,

2. condensing vapor to a liquid,

3. condensing a mixture of vapors containing a noncondensable gas,

4. falling film evaporator,

5. thermosyphon evaporator.

There are many other special cases which are not covered by these illustrative problems. It is believed, however, that the illustrative problems will help indicate what variables are involved in any design proposition and will point to required data. Once pertinent data is obtained, the actual design can be worked out in the manner illustrated hereafter.

Problem No. 1 - Liquid to Liquid Heat Transfer

1. Statement of the Problem

It is required to heat 508,496.9 lbs/hr of rich absorber oil from an inlet temperature of 85°F to an outlet temperature of 228°F, by using lean absorber oil available at the rate of 1190 gal/min and the temperature of 330°F. Design a heat exchanger which will give the necessary performance.

2. Data

a. Lean Absorber Oil (Hot).

Inlet temperature, T_1	330°F
Sp gr at 85°F	0.85 gm/cc

TABLE IV-1

SUMMARY OF EXCHANGER DESIGN

Specifications	Hot Fluid	Cold Fluid
Material		
Inlet Temperature		
Outlet Temperature		
Duty		
LMTD		
U-Clean		
Fouling Factor		
U-Fouled		
U-Required		
No. of Passes per Unit		
No. of Units		
Surface per Unit		
Total Surface		
Shell Size		
Shell Type		
No. of Tubes per Unit		
Total No. of Tubes		
Tube Length		
Tube Size		
Tube Pitch		
Baffles		
Special Modifications		

b. Rich Absorber Oil (Cold). 85°F

Inlet temperature, t_1 228°F
Outlet temperature, t_2 156.5°F
Average temperature 0.77 gm/cc
Sp gr at average temperature 0.52 Btu/lb°F
Sp ht at average temperature
Viscosity at average temperature 1.75 cp
Thermal conductivity at average temperature 0.81 Btu/(hr)(ft²) (°F/ft)

3. Calculated Data

a. Lean Absorber Oil (Hot).

$$W = 1190 \times 8.34 \times 60 \times 0.82 = 487{,}895.8 \text{ lbs/hr.}$$

Calculations for outlet temperature

$$[W\ Cp\ (T_1 - T_2)]_{hot} = [w\ cp\ (t_2 - t_1)]_{cold}$$

$$\therefore\ T_1 - T_2 = \frac{508{,}496{,}9 \times 0.52 \times 143}{487{,}895 \times 0.555} = 139°F$$

$$\therefore\ \text{Outlet temperature } T_2 = 330 - 139 = 191°F$$

$$\therefore\ \text{Average temperature} = \frac{330 + 191}{2} = 260.5°F$$

$$\text{sp gr at } 260.5°F = 0.75 \text{ gm/cc}$$

$$\text{sp ht at } 260.5°F = 0.555 \text{ Btu/lb°F}$$

$$\text{Viscosity at } 260.5°F = 0.77 \text{ cp}$$

$$\text{Thermal conductivity at } 260.5 = 0.0765 \frac{\text{Btu}}{(\text{hr})(\text{ft}^2)(°\text{F/ft})}$$

$$\text{gal/min at average temperature} = \frac{487{,}895.8}{500 \times 0.77} = 1300 \text{ gpm.}$$

b. Rich Absorber Oil (Cold).

$$\text{gal/min at average temperature} = \frac{508496.2}{500 \times 0.77}$$

$$= 1320 \text{ gpm .}$$

4. Solution

a. Duty (Heat Load).

$$Q = w\ c_p\ (t_1 - t_2) = (508{,}496.9)(0.52)(228 - 85)$$
$$= 37{,}800{,}000 \text{ Btu/hr.}$$

b. Calculated LMTD.

$$\begin{array}{ccc} 330 & \longrightarrow & 191 \\ \underline{228} & \longleftarrow & \underline{85} \\ 102 & & 106 \end{array}$$

$$\text{LMTD} = \frac{\Delta t_2 - \Delta t_1}{\ln \frac{\Delta t_2}{\Delta t_1}}$$

$$= \frac{106 - 102}{2.3 \log \frac{106}{102}} = 104^{\circ}\text{F} .$$

For this case let us choose a two-shell pass exchanger. The LMTD calculated above is then corrected by an LMTD correction factor as obtained from Figure 3, Part V.

$$S = \frac{t_2 - t_1}{T_1 - t_1}, \qquad R_t = \frac{T_1 - T_2}{t_2 - t_1}$$

$$S = \frac{228 - 85}{330 - 85} \qquad R_t = \frac{330 - 191}{228 - 85}$$

$$= 0.583 \qquad = 0.973 .$$

From Figure 3,

$$F_x = 0.918.$$

$$\therefore \text{ Corrected LMTD} = 104 \times 0.918 = 95.5^{\circ}\text{F}.$$

c. Assumed Exchanger. Assume U = 70, Table X, Part V.

$$\text{Required transfer surface} = \frac{Q}{U\Delta t_m}$$

$$= \frac{37{,}800{,}000}{70 \times 95.5} = 5660 \text{ sq ft.}$$

For a duty involving such large heating surface, use two units of equal capacity in series. Taking the normal specifications of tubes as 3/4-inch OD, 16 BWG and 16 ft in length, we have the surface area per foot of tube = 0.1963 sq ft/ft (Table II, Part V).

$$\therefore \text{ Number of tubes per unit} = \frac{5660}{2 \times (0.1963) \times (16)} = 900 .$$

From tube sheet layouts, Section 4 of Table IV, Part V for 3/4-inch OD tube on 1 in. sq pitch for four tube passes a 36-inch OD shell has 860 tubes. Choose two such units.

$$\text{Surface per unit} = (860)(0.1963)(16) = 2700 \text{ ft}^2$$

$$\text{Total surface} = 2 \times 2700 = 5400 \text{ ft}^2 .$$

Therefore, overall transfer coefficient required = 37,800,000/(5400)(95.5) = 73.3 Btu/hr.

d. Mass Velocity. i. Shell Side Hot Oil:

$$\text{Cross flow:} \quad G_x' = \frac{\text{lbs/hr} \times 0.04}{\text{baffle pitch} \times \text{free distance}} .$$

For segmental baffles a pitch of 11 inches is normal. Cut 11 rows pass center-line. Free distance is obtained from Table IXa, Part V

$$G_x' = \frac{(487{,}895.8)(0.04)}{(11)(10.5)} = 169 \text{ lbs/ft}^2 \times \text{sec}$$

$$\text{Longitudinal flow:} \quad G_L' = \frac{\text{lbs/hr} \times 0.04}{\text{net free area in sq in.}^2} .$$

Net free area is determined from Figure 45, Part V

$$G_L' = \frac{487{,}895.8 \times 0.04}{104} = 188 \text{ lbs/ft}^2 \times \text{sec.}$$

ii. Tube Side:

$$G_t' = \frac{\text{lbs/hr} \times 0.04}{\text{No. of tubes} \times \text{tube area in sq in.}} .$$

Since curves for determining tube side pressure drop and heat transfer coefficients are based on the tube side mass flow rate as lbs/tube/hour is used in place of G_t'.

$$G_m = \frac{(508{,}496.9)(4)}{860} = 2370 \text{ lbs/tube} \times \text{hr.}$$

A factor of four is used because each unit has four tube passes.

e. Calculated Transfer Rate. i. Tube Side: From Figure 29, $h_t = h \times F_{cp} \times F_k - F_D \times (\mu/\mu_w)^{0.14}$. Knowing t_w, a value of μ_w is obtained. However, in this particular case $(\mu/\mu_w) = 1$; therefore, this factor is neglected in this problem

$$\therefore \; h_t = (195)(1.01)(1.01)(1) = 199 \frac{\text{Btu}}{\text{hr} \times \text{sq ft} \times {}^\circ\text{F}} .$$

ii. Shell Side: From Figure 16,

$$h_x = h \times F_{cp} \times F_k \times F_D \times F_p \times (\mu/\mu_w)^{0.14}$$

$$= 365 \times 1.00 \times 0.97 \times 1.00 \times 0.78 = 276 \frac{\text{Btu}}{\text{hr} \times \text{sq ft} \times {}^\circ\text{F}} .$$

From Figure 17

$$h_L = h \times F_{c_p} \times F_D \times F_k \times (\mu/\mu_w)^{0.14}$$

$$h_L = 215 \times 1.00 \times 0.96 \times 1.01 = 208.5 \frac{\text{Btu}}{\text{hr x sq ft x }^{\circ}\text{F}}$$

$$h_S = \frac{h_X + h_L}{2} = \frac{276 + 208.5}{2} = 242 \frac{\text{Btu}}{\text{hr x sq ft x }^{\circ}\text{F}} .$$

iii. Overall Coefficient:

$$U_c = \frac{(199)(262)}{441} = 109 \frac{\text{Btu}}{\text{hr x sq ft x }^{\circ}\text{F}} .$$

The fouling factors from Table I, Part V are

For lean oil	0.002
For rich oil	0.001
Overall factor	0.003

$$U_f = \frac{109 \times 333}{442} = 82 \frac{\text{Btu}}{\text{hr x sq ft x }^{\circ}\text{F}} .$$

Since the required U = 73.3, the assumed units have sufficient surface.

f. Pressure Drop. i. Shell Side:

$$\Delta P_s = \Delta P_x + \Delta P_L .$$

From equation (66) and Figure 67

$$\Delta P_x = \frac{\Delta P}{\text{sp gr}} \times \frac{\text{rows cross flow}}{10} \times \frac{\text{cross bank passes}}{10} \times F_r \times F_D \times F_p \text{ psi} .$$

Since there are two units in series

$$\Delta P_x = \frac{2\ (2.6)(2.8)(1.7)(1.0)(1.0)(0.44)}{0.75} = 14.5 \text{ psi} .$$

From equation (67a) and Figure 69

$$\Delta P_L = \frac{\Delta P \times \text{No. of baffles}}{s} \text{ psi} .$$

Since there are two units in series

$$\Delta P_L = \frac{2 \times 0.1 \times 17}{0.82} = 4.2 \text{ psi} .$$

HEAT EXCHANGER SPECIFICATIONS

ITEM NO.	NO. REQ'D.	B. M. NO.	DESCRIPTION
			Heat recovery unit for absorption section

TYPE Four tube-pass, two shell-pass, floating head, shell and tube type

SIZE 36 in. OD SECTIONS PER BANK ---- CONNECTED IN

BANKS PER UNIT four CONNECTED IN series

SURFACE PER SECTION 2700 SQ. FT. TOTAL SURFACE 5400 SQ. FT.

PERFORMANCE

SHELL SIDE		TUBE SIDE	SHELL SIDE		TUBE SIDE
lean oil	FLUID	rich oil		GRAVITY-LIQUID	
			0.85 gm/cc	@ INLET TEMP.	
	TOTAL FLUID IN		0.75 gm/cc	~~@ 60°F~~ avg. temp.	0.77 gm/cc
--------	VAPOR		0.77 cp	VISCOSITY @ 260.5 °F	
1300 gpm	LIQUID	1320 rpm		VISCOSITY @ 456 °F	1.75 cp
--------	STEAM	----	----	MOL. WT. - VAPOR	----
--------	NON CONDENSABLES	----	0.555 Btu/lb x °F	SP. HEAT - LIQUID	0.52 Btu/lb x °F
--------	FLUID VAPORED OR CONDENSED		----	LATENT HEAT - VAPOR	----
--------	STEAM CONDENSED		Two	NO. OF PASSES	Four
330 °F	INLET TEMPERATURE	85 °F	88 lb/sq ft	VELOCITY	2370 lb/tube·h
191 °F	OUTLET TEMPERATURE	228 °F	20 psi	PRESS. DROP	27 psi
	OPERATING PRESSURE		0.002	FOULING FACTOR	0.001

HEAT EXCHANGED - BTU/HR 37,800,000 L.M.T.D. (CORRECTED) 95.5°F

"U" CLEAN 109 Btu/hr x sq ft x °F "U" REQUIRED 73.3 Btu/hr x sq ft x °F

CONSTRUCTION

SHELL SIDE		TUBE SIDE
380 °F	DESIGN TEMPERATURE	300 °F
380	DESIGN PRESSURE	300
450	TEST PRESSURE	450 psi
1/4 in.	CORROSION ALLOWANCE	2 BWG's Less

CODE ASME, Paragraph U-69

MATERIALS OF CONSTRUCTION

SHELL	rolled steel plates
TUBES	carbon steel
TUBE SUPPORTS	rolled steel plate
BAFFLES	rolled steel plate
TUBE SHEETS	carbon steel
CHANNEL	carbon steel
CHANNEL COVER	carbon steel
SHELL COVER	carbon steel
FLOATING COVER	carbon steel
GASKETS	soft iron

TUBES

NUMBER 860 SIZE 3/4 in. O.D.

PITCH 1 in. sq THICKNESS 16 BWG.

LENGTH 16 ft

REMARKS

SHELL 36 in. DIA. THICKNESS 7/16 IN.

TUBE SUPPORTS-SPACING 48 in. THICK. 3/4 IN.

TRANSVERSE BAFFLES

TYPE cross flow SPACING 11 in.

CUT 11 rows pass C_L

THICKNESS 5/16 IN.

LONGITUDINAL BAFFLE

TYPE THICKNESS 1/4 IN.

WEIGHTS

EA. SHELL 2582 LB.: BUNDLE 8000 LB.

FULL OF WATER 18,173 LB.

CONNECTIONS: SEE FLOW SHEET OR DRAWING NO - attached hereto.

FUNCTION OF ITEM AND REMARKS:

This unit is designed to recover heat from the lean oil leaving absorption column of a petroleum refinery.

CHK'D	MADE	DATE	NO.

REVISIONS

APPR.			PROJECT NO.
CHK'D.			
MADE			

TYPED

PAGE OF

$$\therefore \Delta P_s = \Delta P_x + \Delta P_L = 14.5 + 4.2 = 18.7 \text{ psi} \text{ ; say, } 20 \text{ psi}$$

ii. Tube Side: From equation (58) and Figure 59

$$\Delta P_t = \frac{P \times L}{10} + Bo \frac{c}{s \times \phi t} .$$

Since there are four tube passes, tube length L should be multiplied by four.

$$\Delta P_t = 2 \frac{(1.34)(16)(4)}{10} + 1.7 \frac{1.0}{0.77} = 26.7 \text{ psi, say } 27 \text{ psi} .$$

Problem No. 2 - Heat Transfer from a Condensing Vapor to a Liquid

1. Statement of the Problem

It is required to preheat 15,000 lbs/hr of air-alcohol-oleic acid mixture containing 2000 lbs of acid from an alcohol recovery system by means of alcohol vapors at the rate of 16,000 lbs/hr from a distillation column. Design a preheater which will give the required performance.

2. Data

a. Hot Fluid - Shell Side. 16,000 lbs/hr alcohol vapors:

Inlet temperature	165	^{o}F
Condensing temperature	165	^{o}F
Latent heat of vaporization	522	Btu/lb
Density at $165^{o}F$	48.6	lb/ft^3 (approx.)
Specific heat at $165^{o}F$	0.76	Btu/lb x ^{o}F
Viscosity at $165^{o}F$	0.45	C.P.
	1.09	lb/ft x hr
Thermal conductivity	0.093	Btu/hr x ft^2 x ^{o}F/ft.

b. Cold Fluid - Tube Side. 15,000 lbs/hr of vapor with 13,000 lb/hr alcohol and 2,000 lb/hr oleic acid:

Inlet temperature	78.4	^{o}F
Outlet temperature	150	^{o}F
Average temperature	114	^{o}F
Specific heat at $114^{o}F$	0.64	Btu/lb x ^{o}F
Density at $114^{o}F$	52.4	lb/ft^3
Viscosity at $114^{o}F$	0.87	C.P.
	2.11	lb/ft x hr
Thermal conductivity at $114^{o}F$ (assumed)	0.11	Btu/hr x sq ft x ^{o}F/ft.

3. Solution

a. Duty.

$$Q = w\, c_p\, \Delta t = (15{,}000)(0.64)(71.6) = 687{,}500 \text{ Btu/hr}.$$

Latent heat of condensation of alcohol = 522 Btu/lb

$$\therefore \text{ Alcohol vapor condensed} = \frac{687{,}500}{522} = 1315 \text{ lbs/hr}.$$

b. LMTD.

$$\begin{array}{ccc} 165^{o}F & \longrightarrow & 165\ ^{o}F \\ \underline{150^{o}F} & \longleftarrow & \underline{78.4^{o}F} \\ 15\ ^{o}F & & 86.6^{o}F\ . \end{array}$$

Therefore,

$$\text{LMTD} = \frac{86.6 - 15}{2.3 \log \frac{86.6}{15}} = \frac{71.6}{(2.3)(0.761)} = 41^{o}F\ .$$

For LMTD correction, we need,

$$R_t = \frac{T_1 - T_2}{t_2 - t_1} = \frac{165 - 165}{150 - 78.4} = 0\ .$$

Therefore, for $R_t = 0$, correction factor approaches unity as observed from Figure 2, Part V. Therefore, corrected LMTD is the same as calculated LMTD.

c. Assumed Unit. Assume an overall coefficient of 100 Btu/hr x ft^2 x ^{o}F. Then, heat transfer surface

$$A = \frac{Q}{U\, \Delta t_m} = \frac{687{,}500}{41 \times 100} = 168 \text{ ft}^2\ .$$

Try a unit with 3/4 in. OD, 16 BWG tubes of 16 ft length and square pitch employing six tube passes and a full floating head. Surface area per foot of tube = 0.1963 sq ft/ft (Table II, Part V).

$$\therefore \text{ No. of tubes} = \frac{168}{0.1963 \times 16} = 54\ .$$

From Tables IV, Section 4 of Part V, for six tube passes 52 tubes are required for 10 in. diameter shell and 76 tubes are required for 12 in. diameter shell. Therefore, shell diameter will be 12 in. Since six tube passes have been chosen, the same number of tubes must be used in each pass; if 12 tubes are used in each pass, the required number of tubes will be 72.

$$\therefore \text{ Total transfer surface} = 72 \times 16 \times 0.1963 = 226 \text{ ft}^2\ .$$

d. Mass Rates. i. Tube Side: Since six tube passes are used all of the fluid passes through 12 tubes. The curves for determining the tube side pressure drop and heat transfer coefficients are based on the tube side mass flow rate as lbs per tube per hour, the rate is:

$$G_m = \frac{15,000}{12} = 1250 \text{ lb/tube x hr} .$$

ii. Shell Side: Only 1350 lbs/hr of alcohol vapor is condensed. Therefore, an average mass flow rate is calculated.

$$\text{Average flow rate} = W_{av} = \frac{\text{Entrance flow rate + Exit flow rate}}{2}$$

$$= \frac{16,000 + (16,000 - 1,350)}{2} = 15,325 \text{ lb/hr.}$$

Shell side mass velocity in cross flow is:

$$G_x = \frac{W_{av}}{\text{free area of cross section}}$$

where free area of cross section is the difference between shell and total outside tubular cross sections.

$$\therefore \; G_x = \frac{15,325}{\left[\frac{\pi x\,(1)^2}{4} - \frac{(0.441)(72)}{144}\right]} = 27,200 \frac{\text{lb}}{\text{hr x ft}^2}$$

$$\therefore \; G_x' = \frac{27,000}{3,600} = 7.55 \frac{\text{lb}}{\text{ft}^2 \text{ x sec}} .$$

e. <u>Calculated Transfer Rates.</u> i. Shell Side: For heat transfer rate 'h_c' of condensing vapors, Figure 35, Part V is a plot of h against $4\Gamma/\mu$, such that

$$h = h_c \left(\mu^2/k^3\rho^2 g\right)^{1/3} .$$

$$\text{Avg wt of vapor condensing per tube} = \frac{1315}{72} = 18.3 \frac{\text{lbs}}{\text{hr x tube}} .$$

Then,

$$\Gamma = \frac{18.3}{\pi D} = \frac{18.3}{R\;(0.0625)} = 93 \frac{\text{lb}}{\text{ft x hr}} .$$

From Figure 35, h = 0.247,

$$\therefore \; h_c = \frac{0.247}{\left\{\frac{(1.09)^2}{(0.093)^3\;(48.6)^2\;(4.18 \text{ x } 10\;)^8}\right\}^{1/3}} = 217 \frac{\text{Btu}}{\text{hr x ft}^2 \text{ x }^\circ\text{F}} .$$

ii. Tube Side: Using properties evaluated at an average temperature of 114°F and Figure 19, Part V, the tube side film coefficient is calculated. Assume a wall temperature of 135°F, a little less than mean between average hot and cold fluid temperatures.

$$\mu_{w_{135^oF}} = 0.70\,\text{C.P.} = 1.69\,\frac{\text{lb}}{\text{ft x hr}}\ .$$

$$\text{Reynold's number, Re} = \frac{DG_t}{\mu} = \frac{(0.0516)(596{,}000)}{2.11} = 14{,}570\ .$$

From Figure 19, Part V,

$$J_\mu = \left(\frac{h_t}{C_pG}\right)\left(\frac{C_p\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 0.00393\ .$$

$$\therefore\ h_t = \frac{(0.00393)(0.64)(596{,}000)}{\left[\frac{(0.64)(2.11)}{0.11}\right]^{2/3}\left[\frac{1.69}{2.11}\right]^{0.14}} = 291\,\frac{\text{Btu}}{\text{hr x ft}^2\text{ x }^oF}\ .$$

Check on wall temperature.

From the resistance concept, for any given flow of heat, the temperature drop through an object is directly proportional to the thermal resistance, or

$$Q = \frac{\Sigma\Delta t}{R}\quad \frac{\Delta t_1 + \Delta t_2 + \Delta t_3 + \ldots}{R_1 + R_2 + R_3 + \ldots}$$

$$R = 1/h \text{ for films and } R = L/k \text{ for solids}\ ,$$

$$R_{condensing} = \frac{1}{217} = 0.00460\ ,$$

$$R_{tube\ wall} = \frac{0.0054}{26} = 0.00021\ ,$$

$$R_{fluid} = \frac{1}{291} = 0.03344\ ,$$

$$R_{total} = 0.00825\ ,$$

$$\Delta t_m = 41^oF\ .$$

The temperature at the inside tube wall is due to the temperature drop caused by the combined resistances of the condensate film and tube wall, or

$$R_c + R_w = 0.0046 + 0.00021 = 0.00481\ .$$

This resistance caused a temperature drop = $(165 - t_w)$

$$\therefore\ \frac{41}{0.00825} = \frac{165 - t_w}{0.00481}$$

or

$$t_w = 165 - \frac{41 \times 0.00481}{0.00825} = 141^{\circ}F .$$

Because the viscosity of alcohol-oleic acid mixture at 141°F approaches that at 135°F a small change in h at the tube side would result from a new calculation using a new value for μ_w. Further, the value of $(\mu/\mu_w)^{0.14}$ is 0.99 and it is obvious no significant difference in the value of h_t is noted by using correct wall temperature.

$$\therefore \text{ clean } h_t = 291 \text{ Btu/hr x ft}^2 \text{ x }^{\circ}F .$$

Using a fouling factor = 0.001, we get,

$$h_{t_f} = \frac{(1000)(291)}{1291} = 225 \text{ Btu/hr x ft}^2 \text{ x }^{\circ}F .$$

f. Overall Transfer Coefficient (Fouled).

$$U_f = \frac{(225)(217)}{442} = 110 \text{ Btu/hr x ft}^2 \text{ x }^{\circ}F .$$

The proposed exchanger has a transfer surface of 226 sq ft. At the required heat load the overall transfer coefficient must be at least

$$= \frac{Q}{A\,\Delta t_m} = \frac{687,500}{(226)(41)} = 74.2 \text{ Btu/hr x ft}^2 \text{ x }^{\circ}F .$$

We note that by employing $U_f = 110$ we have a safety factor of 110/74.2 = 1.48 which is reasonable.

g. Pressure Drop. i. Shell Side: The pressure drop caused by uncondensed alcohol vapors flowing through the shell-side of the condenser may be treated the same as flow through an annular space.

1. Wetted Perimeter

$$\text{Perimeter of shell} = \pi D_s = \pi(1) = 3.141 \text{ ft ,}$$

$$\text{Perimeter of tubes} = \frac{(72)\pi(0.75)}{144} = 1.697 \text{ ft ,}$$

$$\text{Total wetted perimeter} = 3141 + 1.697 = 4.838 \text{ ft .}$$

2. Free Area of Cross Section

= cross section of shell - total tubular cross section

$$= 0.7854 - \frac{(0.441)(72)}{144} = 0.5650 \text{ ft}^2 .$$

$$\therefore \text{ Hydraulic radius} = \frac{0.5650}{4.8380} = 0.1168 \text{ ft .}$$

Equivalent diameter $= D_e = (4)(0.1168) = 0.4675$ ft .

3. Properties of Alcohol Vapor at 165°F

Viscosity	0.0105 C.P.
Density	0.105 lb/ft³.

$$Re = \frac{D_e G}{\mu} = \frac{(0.4675)(27200)}{0.0254} = 501{,}000 \ .$$

From Figure 65

$$f = 0.001$$

$$\therefore \ \Delta P_s = \frac{f \times G^2 \times D_s \times (N+1)}{d \times g \times \rho \times D_e \times \phi}$$

$$= \frac{(0.001)(27{,}200)(1)(1)}{(2)(4.17 \times 10^8)(0.105)(0.4675)} = 0.018 \text{ lb/in.}^2$$

assuming $\phi = 1$.

ii. Tube Side:

$$\Delta P_t = \frac{p \times L}{10} + B_o \ \frac{C}{s \times \phi_t} \ .$$

From Figure 59 we get, P, B_o and C

$$\therefore \ \Delta P_t = \frac{(0.35)(16)(6)}{10} + 0.7 \left(\frac{1}{84}\right)(1.03) = 4.96 \text{ lbs/in.}^2 \text{ or } 5 \text{ psi.}$$

4. Summary of Exchanger Design

Specifications	Hot Fluid (Shell Side)	Cold Fluid (Tube Side)
Material	Alcohol Vapor	Alcohol-Oleic Acid Mixture
Inlet Temperature	165°F	78.4°F
Outlet Temperature	165°F	150 °F
Duty	687,500 Btu/hr	
LMTD	41°F	
U-Clean		
Fouling Factor		0.001

Specifications	Hot Fluid (Shell Side)	Cold Fluid (Tube Side)
U-Fouled	110 Btu/(hr)(ft^2)(^oF)	
U-Required	74.2 Btu/(hr)(ft^2)(^oF)	
No. of passes/unit	1	6
No. of Units	1	
Surface Unit	226 ft^2	
Total Surface	226 ft^2	
Shell Size	12" OD	
Shell Type	Expansion Joint, Fixed Head	
No. of tubes/unit		72
Total No. of Tubes		72
Tube Length		16 ft
Tube Size		3/4" OD 16 BWG
Tube Pitch		1" square
Baffles	1	
Pressure Drop	negligible	5 psi
Special Modifications		

Problem No. 3 - Condensing a Vapor Mixture Containing a Noncondensable Gas--An Example of Liquid to Vapor, Liquid to Condensing Vapor, and Liquid to Liquid Heat Transfer

1. Statement of the Problem

In the design of an oil extraction plant it is required to condense the vapors vented from various equipment throughout the plant. These vapors consist of a mixture of air, hexane, and water. Cooling water is available at 60°F.

The following information is available as to the quantities of material entering the condenser:

Water

from stripper condenser at 130°F	2.7 lb/hr
from deoderizer condenser at 132°F	60.0 lb/hr
from extractor at 115°F	34.2 lb/hr
Total H_2O	96.9 lb/hr

Hexane

from stripper condenser at 130°F	57.9 lb/hr
from deoderizer condenser at 132°F	914.0 lb/hr
from extractor at 115°F	1252.0 lb/hr
Total Hexane	2223.9 lb/hr

Air

from stripper condenser at 130°F	5.0 lb/hr
from deoderizer condenser at 132°F	208.0 lb/hr
from extractor at 115°F	342.0 lb/hr
Total Air	555.0 lb/hr .

a. Given Data.

Cooling Water

temperature rise allowed	10°F
inlet temperature	60°F
outlet temperature	70°F
average temperature	65°F
viscosity at 65°F	2.5 C.P.
thermal conductivity at 65°F	0.347 Btu/(hr)(ft^2)(°F/ft)
specific heat	1.0 Btu/(lb)(°F) .

b. Calculated Data. i. Temperature of Vapor to Condenser: By inspecting the quantities of vapor entering the condenser it is evident that the temperature of the mixture will be at some value t in between 115°F and 130°F. It is assumed that the vapors are mixed completely before they enter the condenser. Thus, the vapors warmer than the temperature of the mixture will give up heat to the vapors cooler than the mixture.

1. Heat Lost in Cooling Vapors to t

$$\begin{aligned}
(\ 2.7)(130 - t)(.46) &= 161.5 - (\ 1.24)t \\
(\ 60.0)(132 - t)(.46) &= 3{,}643.0 - (\ 27.60)t \\
(\ 57.9)(130 - t)(.40) &= 3{,}010.0 - (\ 23.16)t \\
(914.0)(132 - t)(.40) &= 48{,}259.0 - (365.6\ )t \\
(\ 5.0)(130 - t)(.24) &= 156.0 - (\ 1.2\ )t \\
(208.0)(132 - t)(.24) &= 6{,}859.0 - (\ 49.9\ )t \\
\hline
&\ 61{,}818.5 - (468.7\ )t \quad . \qquad \text{(i)}
\end{aligned}$$

2. Heat Gained in Heating Vapors to t

$$\begin{aligned}
(\ 34.2)(t - 115)(.46) &= (\ 15.73)t - 1{,}808 \\
(252.0)(t - 115)(.40) &= (100.8\ )t - 11{,}592 \\
(342.0)(t - 115)(.24) &= (\ 82.1\ )t - 9{,}441 \\
\hline
&\ (198.6\ )t - 22{,}841 \quad . \qquad \text{(ii)}
\end{aligned}$$

Since the amount of heat lost by the cooled vapors equals the amount of heat gained by the warmed vapors on equating (i) and (ii) above,

$$\therefore\ 61{,}818 - 469t = 199t - 22{,}841 ,$$

$$\therefore \quad 668t = 84{,}659,$$

$$\therefore \quad t = 127^{o}F \; .$$

ii. Composition of Vapors Entering at 127°F, 760 mm:

Component	lb/hr	mols/hr	mol fraction
water vapor	96.9	5.38	0.1069
hexane vapor	2223.9	25.8	0.5128
air	555.0	19.13	0.3803
Total	2875.8	50.31	1.0000 .

iii. Calculation of Dew Point: Partial pressure of water vapor = (0.1069)(760) = 81.2 mm Hg. The dew point for water vapor therefore is the temperature at which water has a vapor pressure of 81.2 mm Hg. This value is 114°F from Figure IV-1.

Partial pressure hexane vapor at 114°F = (0.5128)(760) = 390 mm Hg. The dew point for hexane vapor, therefore, is the temperature at which hexane has a vapor pressure of 390 mm Hg. This value is also 114°F. This mixture, therefore, is azeotropic and has a dew point of 114°F (Figure IV-1).

iv. Composition of Streams Leaving Condenser at 65°F: Partial pressure of azeotrope at 65°F = 140 mm Hg. Therefore, the partial pressure of air-- 760 - 140 = 620 mm Hg, and since there are 19.13 mols of air, the total mols of vapor leaving condenser will be

$$\frac{(19.13)(760)}{620} = 23.44 \text{ mols hr} \; .$$

Therefore, the quantity of azeotrope in the vapor leaving condenser is

$$23.44 - 19.13 = 4.31 \text{ mols/hr} \; .$$

1. Composition of Azeotrope in Vapor

Partial pressure of water vapor at 65°F = 16.3 mm Hg. Therefore, mols of water vapor in azeotrope = (16.3)(4.31)/140 = 0.5018 mols/hr of water vapor or (0.5018)(18) = 9.03 lb/hr water vapor.

$$\therefore \text{ mols of hexane vapor} = 4.31 - 0.5018 = 3.808 \text{ mols hexane vapor}$$

or

$$(3.808)(86.17) = 328 \text{ lb/hr hexane vapor.}$$

2. Quantity of Hexane and Water Leaving Condenser as Condensate Water

$$96.9 - 9.03 = 87.87 \text{ lb/hr} \; .$$

Hexane:

$$223.9 - 328 = 1896 \text{ lb/hr} \; .$$

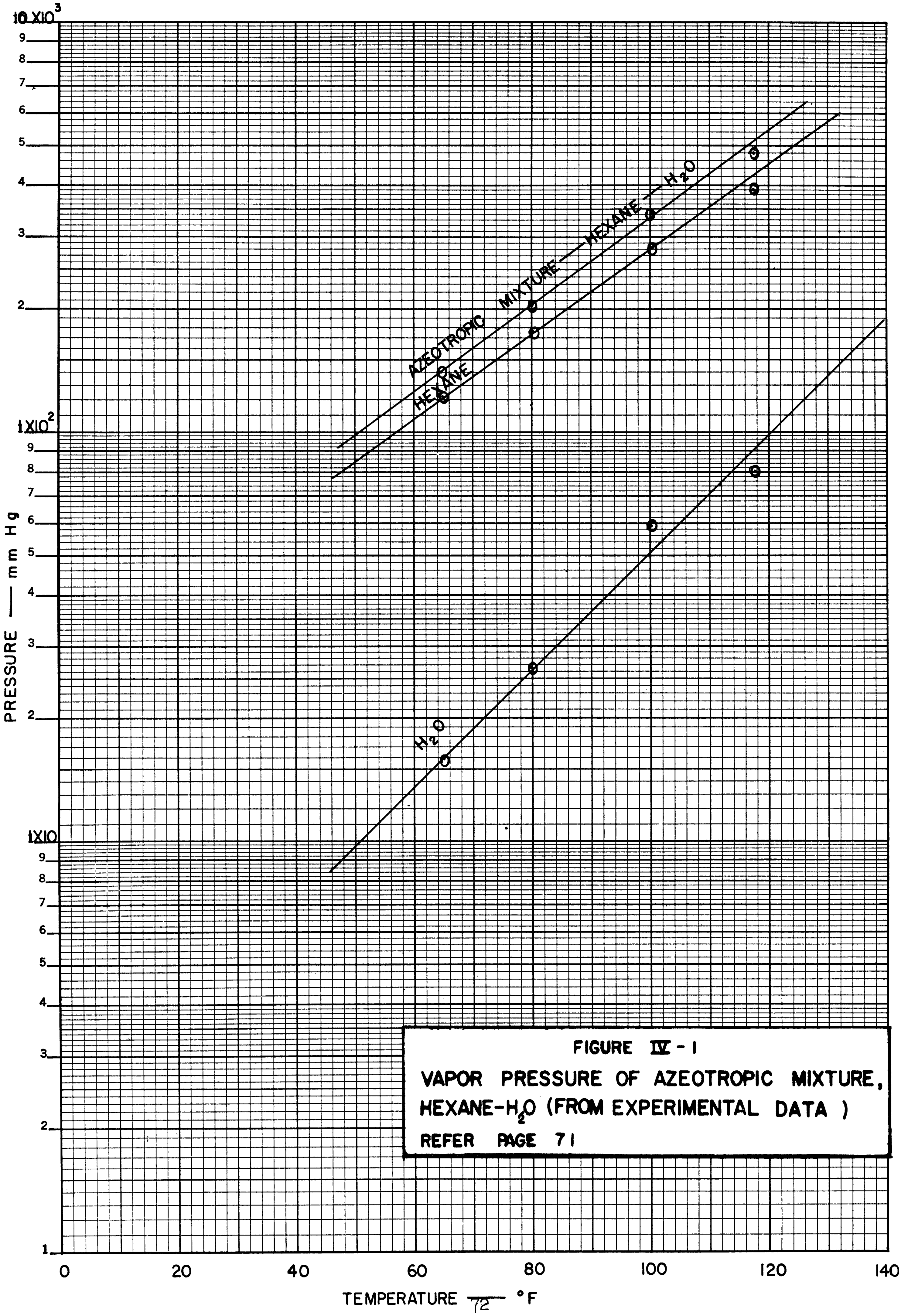


FIGURE IV - 1

VAPOR PRESSURE OF AZEOTROPIC MIXTURE, HEXANE-H_2O (FROM EXPERIMENTAL DATA)

REFER PAGE 71

v. Summary of Calculated Data, Hot Fluid (Tube Side):

Material In		lb/hr
H_2O		96.9
Hexane		2223.9
Air		555
	Total	2875.8

1. Condensing Curve

For heat transfer calculations of condensing vapors it is always necessary to divide the heat exchanger into zones of different degrees of condensation, as condensation will not be uniform throughout. In this particular case the heat exchanger is divided in four zones. For better results and where expediency needs it, it is advisable to take smaller zones.

The first zone is for cooling the incoming vapors from their entrance temperature to that of the dew point (114°F). The second zone is for cooling vapors, condensing and cooling the condensate, as are the third and fourth zones. In the zones where vapor cooling, condensing and condensate cooling takes place simultaneously, it is assumed that half of the total quantity of vapor that condenses is cooled in liquid state and half in the vapor state. Temperature drops of each zone are assumed to be as under

Zone No.	Inlet Temperature	Outlet Temperature	Service
1	127°F	114°F	Vapor Cooling
2	114°F	100°F	Cooling and Condensing
3	100°F	80°F	Cooling and Condensing
	80°F	60°F	Cooling and Condensing.

For plotting a condensing curve the percent condensation in zones 2 and 3 have to be calculated.

a. at 100°F

Vapor pressure of azeotrope = 340 mm Hg. Therefore, the mol fraction of azeotrope in vapor will be 340/760 = 0.447 and mols of azeotrope in vapor will be (0.447)(19.13)/(0.554) = 15.43 mols/hr.

Vapor pressure of hexane = 281 mm Hg.

Therefore, mols of hexane in vapor will be (281)(15.43)/(340) = 12.75 mols/hr and mols of water in vapor will be 15.43 - 12.75 = 2.68 mols/hr.

∴ Pounds of water in vapor = (2.68)(18) = 48.24 lb/hr and pounds of hexane

in vapor = (12.75)(86.17) = 1099 lb/hr.

b. at 80°F

Vapor pressure of azeotrope = 204 mm Hg.

$$\text{Mol fraction of azeotrope in vapor} = \frac{204}{760} = 0.2684\ .$$

$$\text{Mols per hour of azeotrope} = \frac{(0.2684)(19.13)}{0.7316} = 7.02\ \text{mol/hr}\ .$$

$$\text{Vapor pressure of hexane} = 177\ \text{mm Hg}\ .$$

$$\text{Mol fraction of hexane in azeotrope} = \frac{177}{204} = 0.8676\ .$$

$$\text{Mols per hour of hexane in vapor} = (.8676)(7.02) = 6.09\ \text{mols/hr}\ .$$

$$\text{Mols per hour of water in vapor} = 7.02 - 6.09 = 0.93\ \text{mols/hr}\ .$$

$$\text{Pounds water in vapor} = (0.93)(18) = 16.74\ \text{lb/hr}\ .$$

$$\text{Pounds hexane in vapor} = (6.09)(86.17) = 525\ \text{lb/hr}\ .$$

A condensing curve is shown in Figure IV-2, page 75.

2. Duty: Duty is calculated for each zone

a. Zone 1 - Vapor cooling from 127°F to 114°F, $\Delta t = 13°F$

H_2O - (96.9)(0.5)(13)	=	640 Btu/hr
Hexane - (2223.9)(0.4)(13)	=	11,564 Btu/hr
Air - (555)(0.25)(13)	=	1,803 Btu/hr
Total		14,007 Btu/hr .

b. Zone 2 - Vapor cooling from 114°F to 100°F, $\Delta t = 14°F$

Amount of water condensed = 96.9 - 48.24
= 48.66 lb/hr

Amount of hexane condensed = 2223.9 - 1099
= 1124.9 lb/hr .

(i) Condensing

H_2O	(48.66)(1033)	=	50,265 Btu/hr
Hexane	(1124.9)(153.2)	=	172,334 Btu/hr
			222,599 Btu/hr .

(ii) Vapor cooling

H_2O	(72.6)(0.5)(14)	=	508 Btu/hr
Hexane	(1661.4)(0.4)(14)	=	9,303 Btu/hr
Air	(555)(0.25)(14)	=	1,942.5 Btu/hr
			11,753.5 Btu/hr .

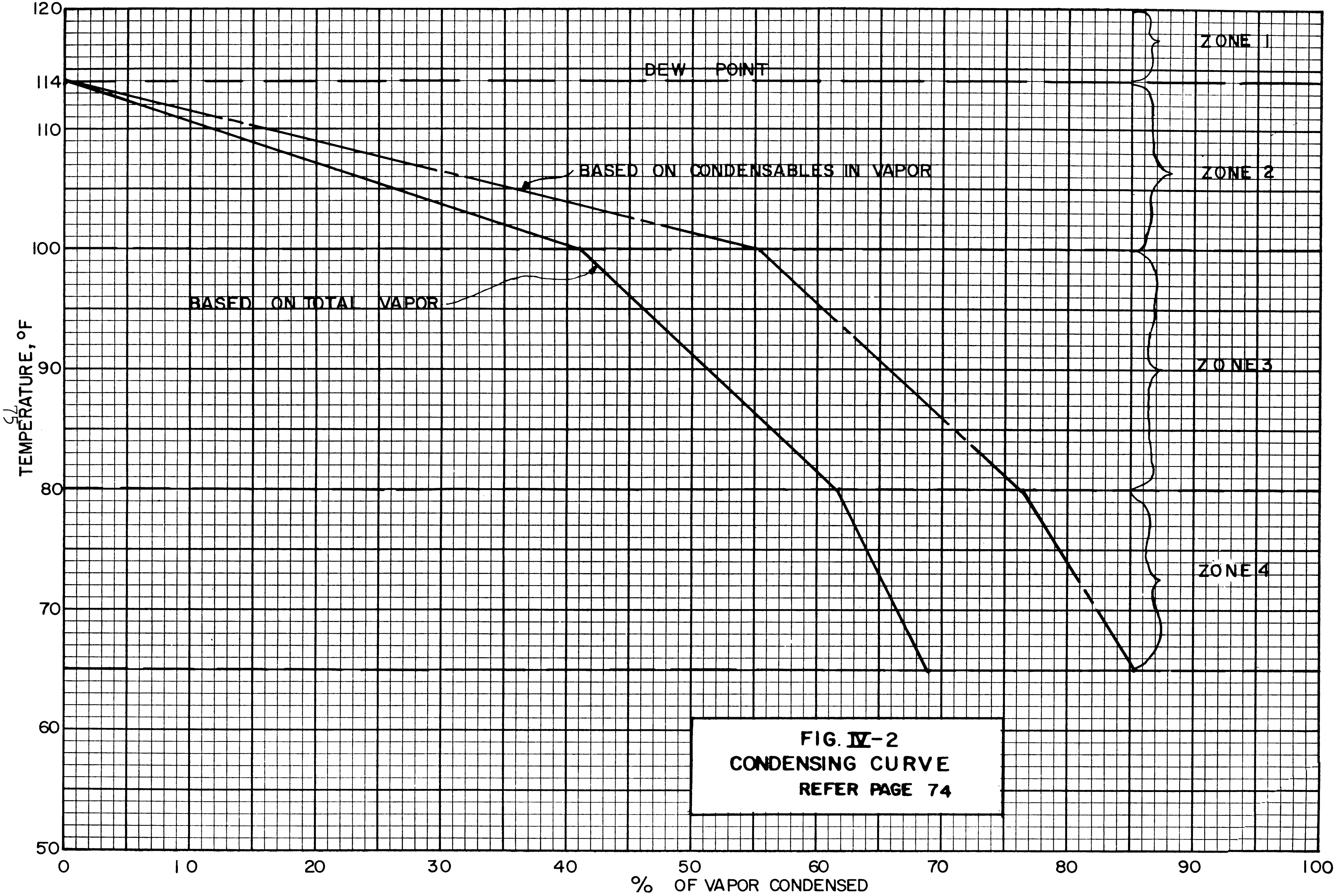


FIG. IV-2
CONDENSING CURVE
REFER PAGE 74

(iii) Liquid cooling

H_2O	(26.33)(1)(14)		340.0 Btu/hr
Hexane	(562.4)(0.625)(14)	=	4921.0 Btu/hr
			5261.0 Btu/hr .

c. Zone 3 - Vapor cooling from 100°F to 80°F, $\Delta t = 20°F$

Amount of water condensed = 48.24 - 16.74

= 31.50 lb/hr .

Amount of hexane condensed = 1099 - 525

= 574 lb/hr .

(i) Condensing

H_2O	(31.49)(1042.9)	=	32,841 Btu/hr
Hexane	(574)(157.2)	=	90,233 Btu/hr
			123,074 Btu/hr .

(ii) Vapor cooling

H_2O	(32.49)(20)(.5)	=	325 Btu/hr
Hexane	(812)(20)(.4)	=	6496 Btu/hr
Air	(555)(20)(.25)	=	2775 Btu/hr
			9596 Btu/hr .

(iii) Liquid cooling

H_2O	(64.50)(20)(1)	=	1,290 Btu/hr
Hexane	(1411.9)(20)(.625)	=	17,648 Btu/hr
			18,938 Btu/hr .

d. Zone 4 - Vapor cooling from 80°F to 65°F, $\Delta t = 15°F$

(i) Condensing

H_2O	(7.27)(1050)	=	7,633 Btu/hr
Hexane	(197)(159.8)	=	31,480 Btu/hr
			39,113 Btu/hr .

(ii) Vapor cooling

H_2O	(12.89)(15)(.5)	=	97 Btu/hr
Hexane	(426.5)(15)(.4)	=	2559 Btu/hr
Air	(555.0)(15)(.25)	=	2081 Btu/hr
			4737 Btu/hr .

(iii) Liquid cooling

H_2O	(84)(1)(15)	=	1,260 Btu/hr
Hexane	(1797)(.62)(15)	=	16,711 Btu/hr
			17,971 Btu/hr .

e. Summary

(i) Zone 1 14,007 Btu/hr

(ii) Zone 2

Condensing	222,599
Vapor cooling	11,753
Liquid cooling	5,261
Total	239,613 Btu/hr

(iii) Zone 3

Condensing	123,074
Vapor cooling	9,596
Liquid cooling	18,938
Total	151,608 Btu/hr

(iv) Zone 4

Condensing	39,113
Vapor cooling	4,737
Liquid cooling	17,971
Total	61,821 Btu/hr.

Total heat load for 4 zones = 467,049 Btu/hr .

Required cooling water

$$W = \frac{Q}{C_p \Delta t} = \frac{467{,}049}{(10)(1.0)} = 46{,}705 \text{ lb/hr} .$$

3. LMTD

a. Calculation of temperature drop of cooling water in each zone

In order to get the LMTD over each zone, the temperature drop of the cooling water in each zone must be determined. It is assumed that the temperature drop over any given zone is proportional to the heat load of that zone. The total Δt is 10°F.

(i) Zone 1

$$\Delta t\ (1) = \frac{14{,}007\ (10)}{467{,}049} = 0.3^{\circ}\text{F} .$$

(ii) Zone 2

$$\Delta t\ (2) = \frac{239{,}613\ (10)}{467{,}049} = 5.13^{\circ}\text{F} .$$

(iii) Zone 3

$$\Delta t\ (3) = \frac{(151{,}608)(10)}{467{,}049} = 3.24^{\circ}\text{F} .$$

(iv) Zone 4

$$\Delta t\ (4) = \frac{(61{,}821)(10)}{467{,}049} = 1.325^{o}F\ .$$

b. Calculation of LMTD for each zone

← zone 1 →		← zone 2 →		← zone 3 →		← zone 4 →	
124 →	117 →		100 →		80 →		65
70 ←	69.7 ←		64.5 ←		61.3 ←		60
54	47.3		35.5		18.7		5

Zone 1

$$LMTD = \frac{54 - 47.3}{2.3 \log \frac{54}{47.3}} = \frac{6.7}{2.3 \log 1.144} = \frac{6.7}{(2.3)} = 50^{o}F\ .$$

Zone 2

$$LMTD = \frac{47.3 - 35.5}{2.3 \log \frac{47.3}{35.5}} = \frac{11.8}{2.3 \log 1.329} = \frac{11.8}{(2.3)(0.124)} = 41^{o}F\ .$$

Zone 3

$$LMTD = \frac{35.5 - 18.7}{2.3 \log \frac{35.5}{18.7}} = \frac{16.8}{2.3 \log (1.898)} = \frac{16.8}{(2.3)(.278)} = 26.2^{o}F\ .$$

Zone 4

$$LMTD = \frac{18.7 - 5}{2.3 \log \frac{18.7}{5}} = \frac{13.7}{2.3 \log 3.74} = \frac{13.7}{(2.3)(.573)} = 10.4^{o}F\ .$$

Overall LMTD

$$LMTD = \frac{54 - 5}{2.3 \log \frac{54}{5}} = \frac{49}{2.3 \log 10.8} = \frac{49}{(2.3)(1.034)} = 20.6^{o}F\ .$$

4. Trial No. 1

a. Estimated overall transfer coefficient

Because there is a considerable amount of noncondensable gas in the mixture, the overall coefficient may be small. Assume overall U = 20 Btu/(hr)(ft^2)(^{o}F)

b. Estimated surface

$$A_x = \frac{467{,}049}{(20)(20.6)} = 1{,}127 \text{ sq ft}$$

using 3/4 in. OD, 16 BWG tubes of 12 ft length on 15/16 in. Δ pitch the number of tubes required = 1127/(12 x 0.1963) = 480.

c. Assumed unit

Table III, Section (5), Part V for shell size of 22 in. OD gives a single pass unit as under:

Tube size	3/4 in. 16 BWG
Tube number	433
Tube total area	1020 ft^2
Shell size	22 in. OD
Shell pass	1
Shell type	fixed tube sheet with expansion joint.
Baffles	5 in. pitch cut 9 rows past centerline.

d. Mass flow rate

1. Shell side

(i) Cross flow

The free area for this condenser is determined from Figure 39. This curve gives a value of 4.8 in per inch of baffle pitch.

$$G_x = \frac{(46,705)(.04)}{(4.8)(5)} = 77.84 \text{ lb/(ft}^2\text{)(sec)} .$$

(ii) Longitudinal flow

From Figure 53 the free area is 20.0 sq in.

$$G_L = \frac{(46,705)(.04)}{20} = 93.4 \text{ lb/ft}^2\text{)(sec)} .$$

2. Tube side

(i) Condensate mass velocity

$$\Gamma = \frac{(1984)(12)}{\pi\ (0.75)(433)} = 23.33 \text{ lb/(hr)(ft)}$$

or

$$\frac{(1984)}{433} = 4.58 \text{ lb/(tube)(hr)} .$$

(ii) Vapor mass velocity

Zone 1

From the condensation curve it is determined that no material is condensed in this zone.

$$G_t = \frac{(2976)(144)}{(433)(.3019)} = 3168 \text{ lb/(hr)(ft}^2)$$

or

$$\frac{3168}{3600} = 0.88 \text{ lb/(ft}^2\text{)(sec)} .$$

Zone 2

For the gas flow rate in each zone assume the flow rate as being the average of the total amount of vapor entering and the total amount leaving that zone.

$$W = \frac{2876 + 1696}{2} = 2286 \text{ lb/hr} .$$

$$G_{t_2} = \frac{(2286)(.04)}{(433)(.3019)} = 0.699 \text{ lb/(ft}^2\text{)(sec)} .$$

Zone 3

$$W = \frac{1696 + 1099}{2} = 1396 \text{ lb/hr} .$$

$$G_{t_3} = \frac{(1396)(.04)}{(433)(.3019)} = 0.427 \text{ lb/(ft}^2\text{)(sec)} .$$

Zone 4

$$W = \frac{1099 + 898}{2} = 998 \text{ lb/hr} .$$

$$G_{t_4} = \frac{(998)(.04)}{(443)(.3019)} = 0.305 \text{ lb/(ft}^2\text{)(sec)}.$$

e. Calculation of transfer coefficients

1. Shell side

(i) Cross flow

From Figure 9

$$h_x = (175)(1.2)(2.6)(1) = 546 \text{ Btu/(hr)(ft}^2\text{)(}^oF).$$

Using a fouling factor of 0.8

$$h_{xf} = (546)(.8) = 437 \text{ Btu/(hr)(ft}^2\text{)(}^oF).$$

(ii) Longitudinal flow

From Figure 14

$$h_L = (360)(1) = \text{Btu/(hr)(ft}^2\text{)(}^oF).$$

Using a fouling factor of 0.8

$$h_{Lf} = (360)(0.8) = 288 \text{ Btu/(hr)(ft}^2\text{)(}^{o}\text{F)}$$

(iii) Average of cross and longitudinal flow

$$h_{sf} = \frac{437 + 288}{2} = 362 \text{ Btu/(hr)(ft}^2\text{)(}^{o}\text{F)} .$$

2. Tube side

<u>Zone 1</u>

(i) Vapor cooling

From Figure 22

$$h_v = (14.7)(.3646)(.826)(1.1) = 4.9.$$

Using a fouling factor of .8

$$h_{vf} = (4.9)(.8) = 3.9 \text{ Btu/(hr)(ft}^2\text{)(}^{o}\text{F)}.$$

(ii) Vapor condensation

From Figure 34

$$h_c = (280)(1)(1.0)(.9) = 252 \text{ Btu/(hr)(ft}^2\text{)(}^{o}\text{F)}.$$

Using a fouling factor of .8

$$h_{cf} = (252)(.8) = 202 \text{ Btu/(hr)(ft}^2\text{)(}^{o}\text{F)} .$$

(iii) Liquid cooling

From Equation 41, Part III, a value can be estimated.

$$h_L = 120 \Gamma^{1/3} = (120)(2.86) = 343 \text{ Btu/(hr)(ft}^2\text{)(}^{o}\text{F)} .$$

Using a fouling factor of .8

$$h_{Lf} (343)(.8) = 274 \text{ Btu/(hr)(ft}^2\text{)(}^{o}\text{F)}.$$

<u>Zone 2</u>

(i) Vapor cooling

From Figure 22

$$h_v = (12)(.3646)(.826)(1.1) = 3.97 \text{ Btu/(hr)(ft}^2\text{)(}^{o}\text{F)} .$$

Use a fouling factor of .8

$$h_{vf} = (3.97)(.8) = 3.17 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

(ii) Condensing and liquid cooling

Because these values are estimated from average conditions over the exchanger, they will be the same for each zone.

$$h_{cf} = 202 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

$$h_{Lf} = 274 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

Zone 3

(i) Vapor cooling

$$h_v = (8)(.364)(.826)(1.1) = 2.65 .$$

Use a fouling factor of .8

$$h_{vf} = (2.65)(.8) = 2.12 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

(ii) Condensing and liquid cooling

$$h_{cf} = 202 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

$$h_{Lf} = 274 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}.$$

Zone 4

(i) Vapor cooling

$$h_v = (6)(.3646)(.826)(1.1) = 1.98 .$$

Use a fouling factor of .8

$$h_{vf} = (1.98)(.8) = 1.58 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

(ii) Condensing and liquid cooling

$$h_{cf} = 202 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

$$h_{Lf} = 274 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

(iii) Overall coefficients

Zone 1

(a) Vapor cooling

$$(U_v)_1 = \frac{(362)(3.9)}{365.9} = 3.85 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

(b) Condensing

$$(U_c)_1 = \frac{(362)(202)}{564} = 130 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

(c) Liquid cooling

$$(U_L)_1 = \frac{(362)(274)}{636} = 156 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

Zone 2

(a) Vapor cooling

$$(U_V)_2 = \frac{(362)(3.17)}{365.17} = 3.14 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

(b) Condensing and liquid cooling

These values are the same for each zone.

$$(U_c)_2 = 130 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

$$(U_L)_2 = 156 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

Zone 3

(a) Vapor cooling

$$(U_V)_3 = \frac{(362)(2.12)}{364.12} = 2.10 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

(b) Condensing and liquid cooling

$$(U_c)_3 = 130 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

$$(U_L)_3 = 156 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}.$$

Zone 4

(a) Vapor cooling

$$(U_V)_4 = \frac{(362)(1.58)}{363.58} = 1.57 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

(b) Condensing and liquid cooling

$$(U_c)_4 = 130 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

$$(U_L)_4 = 156 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}.$$

f. Calculation of required tube surface area

1. Zone 1

(i) Vapor cooling

$$(A_V)_1 = \frac{7356}{(50)(3.85)} = 38.2 \text{ ft}^2 .$$

2. Zone 2

(i) Vapor cooling

$$(A_V)_2 = \frac{14,302}{(41)(3.14)} = 111.1 \text{ ft}^2 .$$

(ii) Condensing

$$(A_C)_2 = \frac{222,599}{(41)(130)} = 41.8 \text{ ft}^2 .$$

(iii) Liquid cooling

$$(A_L)_2 = \frac{6388}{(41)(156)} = 1.0 \text{ ft}^2 .$$

3. Zone 3

(i) Vapor cooling

$$(A_V)_3 = \frac{9.596}{(26.2)(2.1)} = 174.4 \text{ ft}^2 .$$

(ii) Condensing

$$(A_C)_3 = \frac{123,074}{(26.2)(130)} = 36.1 \text{ ft}^2 .$$

(iii) Liquid cooling

$$(A_L)_3 = \frac{18.938}{(26.2)(156)} = 4.6 \text{ ft}^2 .$$

4. Zone 4

(i) Vapor cooling

$$(A_V)_4 = \frac{4737}{(10.4)(1.57)} = 290.0 \text{ ft}^2 .$$

(ii) Condensing

$$(A_C)_4 = \frac{39,113}{(10.4)(130)} = 28.92 \text{ ft}^2 .$$

(iii) Liquid cooling

$$(A_L)_4 = \frac{17{,}971}{(10.4)(156)} = 11.1 \text{ ft}^2 .$$

5. Total tube area required

Zone 1

Vapor cooling	38.2 ft^2

Zone 2

Vapor cooling	111.1 ft^2
Condensing	41.8 ft^2
Liquid cooling	1.0 ft^2

Zone 3

Vapor cooling	174.4 ft^2
Condensing	36.1 ft^2
Liquid cooling	4.6 ft^2

Zone 4

Vapor cooling	290.9 ft^2
Condensing	28.9 ft^2
Liquid cooling	11.1 ft^2
Total area	738.2 ft^2.

6. Safety factor

$$\text{S.F.} = \frac{1020}{738.2} = 1.38 .$$

This condenser has a safety factor of 1.38, but since this unit is a vent condenser the excess tube area is desirable.

g. Pressure drops

1. tube side

$$W_{ave} = \frac{2876 + 898}{2} = 1887 \text{ lb/hr}$$

or

$$G_{ave} = \frac{(1887)(.04)}{(433)(.3019)} = 0.577 \text{ lb/(ft}^2\text{)(hr)} .$$

From Figure 61

$$\Delta P_t = (0.0033)(12)(1.51)(1.58) = 0.094 \text{ lb/in.}^2$$

Call pressure drop 0.2 psi to include box loss.

2. Shell side

(a) From Figure 67 cross flow--use a safety factor 2.2

$$\Delta P_x = \frac{(0.82)(2.8)(2.2)}{1} = 5.043 \text{ lb/in.}^2$$

(b) Longitudinal flow

From Figure 69

$$\Delta P_L = \frac{(.019)(28)}{1} = 0.53 \text{ lb/in.}^2$$

(c) Total shell side pressure drop

Longitudinal	5.043
Cross	.53
Total	5.573 lb/in.^{2}

Summary of Exchanger Design

Specifications	Hot Fluid (Tube Side)	Cold Fluid (Shell Side)
Material	hexane vapor, water vapor, air	cooling water
Inlet temperature	127°F	60°F
Outlet temperature	65°F	70°F
Duty	467,049 Btu/hr	
LMTD	20.6°F	
U-Clean	30.5 Btu/hr x ft^2 x °F	
Fouling Factor	0.8	0.8
U-Fouled		
U-Required	22 Btu/ft^2 x hr x °F	
No. of passes/unit	1	1
No. of units	1	
Surface/unit	1029 sq ft	
Total surface	1029 sq ft	
Shell size	22 in. OD	
Shell type	Expansion joint, fixed tube sheet	
No. of tubes/unit		433
Total number of tubes		433
Tube length		12 ft
Tube size		3/4 in. OD, 16 BWG
Tube pitch		15/16 in triangular V
Baffles	5 in. pitch, cut 9 rows past C_1	
Pressure drop	0.2 psi	
Special modifications		

Problem No. 4 - Design of a Falling Film Evaporator

1. Statement of the Problem

It is required to evaporate 1800 lbs/hr of 22.5% $Al(NO_3)_3$ solution to a concentration of 30%. This evaporation is to be carried out in a falling film type evaporator. Steam is available at pressures up to 40 psig. The feed stream is to enter the evaporator at 75°F.

Information on the evaporation of $Al(NO_3)_3$ in a falling film evaporator was determined by running solutions of $Al(NO_3)_3$ through a small test unit. This evaporator contained three one-inch, OD, 16 BWG condenser tubes, 8 feet long.

These tests indicate that the best overall results are obtained when the steam pressure in the jacket is 10 psig. At this steam pressure, and at feed rates of 30 lb/hr/tube of 22.5% $Al(NO_3)_3$, overall heat transfer coefficients of 200 Btu/hr/ft²/°F are obtained. The concentrated product leaves the evaporator tubes at a temperature of 225°F.

2. Given Data

a. Cold Fluid - Tube Side.

$Al(NO_3)_3$ feed rate	1800 lb/hr
Inlet temperature	75°F
$Al(NO_3)_3$ composition (inlet)	22.5%
$Al(NO_3)_3$ composition (outlet)	30.0%
Outlet temperature	225°F
Boiling point at feed composition	218°F
Specific heat	0.75 Btu/(lb)(°F)
Density	75 lbs/ft³.

b. Hot Fluid (Steam) - Shell Side.

Shell Side Pressure	10 psig
Temperature	240°F
Latent heat of vaporization	950 Btu/lb .

3. Solution

a. Heat Load. i. Preheating Section: Amount of heat required = $W\ C_p$

$$\Delta t = (1800)(0.75)(218 - 75) = 193{,}050 \text{ Btu/hr.}$$

ii. Evaporating Section: Amount of 30% $Al(NO_3)_3$ produced

$$= \frac{(1800)(.225)}{0.3} = 1350 \text{ lbs/hr .}$$

Therefore,

Amount of water evaporated = 1800 - 1350 = 450 lbs/hr .

Quantity of heat required = (450)(950) = 427,500 Btu/hr .

b. Calculated LMTD. i. Feed Preheating:

$$\begin{array}{ccc} 240^{\circ}F & \longleftrightarrow & 240^{\circ}F \\ \underline{218^{\circ}F} & \longleftrightarrow & \underline{75^{\circ}F} \\ 22^{\circ}F & & 165^{\circ}F \end{array} .$$

Therefore,

$$\text{LMTD} = \frac{165 - 22}{2.3 \log \frac{165}{22}} = \frac{143}{(2.3)(.875)} = 71^{\circ}F .$$

ii. Evaporating Section:

$$\begin{array}{ccc} 240^{\circ}F & \longleftrightarrow & 240^{\circ}F \\ \underline{225^{\circ}F} & \longleftrightarrow & \underline{218^{\circ}F} \\ 15^{\circ}F & & 22^{\circ}F \end{array} .$$

$$\text{LMTD} = \frac{22 - 15}{2.3 \log \frac{22}{15}} = \frac{7}{(2.3)(.164)} = 18^{\circ}F .$$

c. Calculated Unit. Since an overall heat transfer coefficient was obtained from the test run, the coefficient as given is used with the appropriate LMTD for the boiling section and the preheating section in calculating the required transfer surface.

i. Surface Required for Feed Preheating:

$$A = \frac{193{,}050}{(71)(200)} = 13.6 \text{ ft}^2 .$$

ii. Surface Required for Evaporation:

$$A = \frac{427{,}500}{(18)(200)} = 118.7 \text{ ft}^2 .$$

iii. Total Surface Required:

Preheating section	13.6
Evaporation section	118.7
Total	132.3 ft^2 .

iv. Calculation of the Number of Tubes: Use 1 in. OD, BWG 16 condenser tubes. Since the optimum evaporation rate occurred at a feed rate of 30 lb/(hr)(tube), this figure is used as a basis for design.

$$\text{No. of tubes} = \frac{\text{total flow}}{\text{flow rate/tube}} = \frac{1800}{30} = 60 \text{ tubes} .$$

Choose a shell size of 12 inches, with 1 in. OD, tubes on a 1-1/4 in. triangular pitch. The unit is a one-pass, fixed-tube sheet exchanger. The tube sheet layout table, Section (2), Table III, Part V, shows 68 tubes in this size shell. This unit would be satisfactory.

v. Calculation of Tube Lengths:

Surface area for 1 in. O.D., 16 BWG tube = 0.228 ft^2/ft (Table II, Part V).

$$\text{Length of tubes required} = \frac{132.3}{0.228} = 580 \text{ ft} .$$

$$\text{Length of each tube} = \frac{580}{68} = 8.52 \text{ ft long} .$$

Choose a tube length of 12 feet, since this is the closest standard tube length available.

vi. Calculation of Vapor Velocity in Tubes: Since 7.5 lb per hour of water are evaporated per tube, the maximum vapor velocity obtained per tube is

$$\frac{(26)(7.5)(144)}{(3600)(0.5945)} = 13.12 \text{ ft/sec} .$$

This velocity is much less than the allowable velocity of 30 feet per second.

vii. Feed Distribution: The feed distribution system is to be made as follows:

1. A distributor plate is to be mounted parallel to, and 3/4 inches above the tube sheet.

2. Holes are to be drilled in the distributor plate so that their centers will be directly above the corners of the hexagonal pattern around each tube. This will allow the feed to be distributed to each tube from six points.

3. Depth of feed on distributor plate is to be greater than 1/4 inch.

viii. Calculation of Hole Size for Holes on Distributor Plate: Hole pattern shows that there are six distributor holes for each tube. Therefore, the number of distributor holes is

$$(6)(68) = 408 \text{ holes.}$$

Since the amount of feed to the distributor is 1800 lb per hour, each hole on the distributor is fed at the rate

$$W = \frac{1800}{(408)(3600)} = 1.2 \times 10^{-3} \text{ lb/sec} .$$

Dodge,[61] suggests that the discharge coefficient for a sharp-edged orifice discharging liquid into a gas space should be approximately 0.6. Since it is imperative that the liquid level on the distributor plate remain at least 1/4 inch, a coefficient of 0.8 is used. This is a conservative assumption.

The equation for an orifice is

$$W = C_o \, a_s \sqrt{\frac{2 \, \Delta H \rho^2}{1 - \left(\frac{D_o}{D_t}\right)^4}} \; .$$

Since the value of $(D_o/D_t)^4$ in this case is very small, the equation becomes,

$$W = C_o \, a_s \, \rho \sqrt{2g \, \Delta H}$$

which can be rearranged to give

$$a_s = \frac{W}{C_o \, \rho \sqrt{2g \, \Delta H}} \; .$$

The area required in order to discharge the correct amount of feed is now calculated.

$$a_s = \frac{1.2 \times 10^{-3}}{(0.8)(75)\sqrt{(64)(0.0208)}} = 1.73 \times 10^{-5} \text{ ft}^2 \; .$$

$$\therefore \quad d_h^2 = \frac{4 a_s}{\pi} = \frac{(1.73 \times 10^{-5})(4)}{\pi} = 22.06 \times 10^{-6} \; .$$

Therefore, the required hole size is

$$d_h = 4.7 \times 10^{-3} \text{ ft} \; .$$

or

$$d_h = (4.7 \times 10^{-3})(12) = 0.047 \text{ inches} \; .$$

Therefore, the distributor plate feed holes are made with a diameter of 0.1 inch.

1. Pressure drop through tubes

From part (vi) the maximum vapor rate through each tube is 7.5 lbs/hr. The average vapor rate is

$$\frac{7.5}{2} = 3.75 \text{ lb/hr} \; .$$

or

$$\frac{(3.75)}{(.0054)} = 685 \text{ lb/(hr)(ft}^2)$$

or

$$\frac{685}{3600} = 0.19 \ \text{lb}/(\text{ft}^2)(\text{sec}) \ .$$

Assume an average vapor temperature of 220°F. At this temperature the average viscosity is 0.025 C.P. Figure 61, Part V gives a pressure drop of 0.002 lb/(in.²)(ft). Assume that vapor flows through the entire length of tube, and that the thickness of the liquid film on the tube wall is negligible.

$$\text{Pressure drop } \Delta P_t = (12)(0.002) = 0.024 \ \text{lb/in.}^2 \ .$$

Summary of Exchanger Design

Specifications	Hot Fluid (Shell-Side)	Cold Fluid
Material	Steam (10 psig)	$Al(NO_3)_3$ solution
Inlet temperature	240°F	75°F at 22.5% conc.
Outlet temperature	240°F	218°F at 30.0% conc.
Duty	427,500 Btu/hr	
LMTD	Preheating zone 71°F Evaporating zone 18°F	
U-Clean	200 Btu/(hr)(ft²)(°F)	
Fouling factor		
U-Fouled		
U-Required		
No. of passes/unit		
No. of units		
Surface/unit	132.3 sq ft	
Total surface	132.3 sq ft	
Shell size	12 sq ft	
Shell type	Fixed tube sheet Expansion joint	
No. of tubes/unit		68
Total no. of tubes		68
Tube length		12 ft
Tube size		1 in. OD, 16 BWG
Tube pitch		1-1/4 in. triangular
Baffles		
Pressure drop		0.024 psi

Special modifications: tubes are to be groud flush with top tube sheet.

Feed distribution: Feed is to be distributed to the tubes by putting a distributor plate parallel to and 3/4 in. from the tube sheet. Distributor plate is to be made from 1/8 in. thick plate drilled with 408 holes of 0.1 in. diameter. These holes are to be positioned such that their centers will be directly above the corners of the hexagonal pattern around each tube.

Problem No. 5 - Design of a Thermosyphon Evaporator

1. Statement of Problem

In a pilot plant recovery process, a dilute solution of a mixture of salts is produced which has a specific gravity of 1.01. It is required to concentrate this solution to a specific gravity of 1.350. This solution is produced at the rate of 235 liters per day. It is required further that a thermosyphon evaporator be used.

2. Summary of Given Data

a. Cold Fluid - Tube Side.

Salt solution feed rate	235 liters/day
Inlet temperature	75°F
Outlet temperature (boiling point)	220°F
Average specific heat (approximate)	1.00 Btu/(lb)(°F)
Other properties are assumed to be the same as those of water at 212°F	
Concentrated solution outlet rate	6.5 liters/day

b. Hot Fluid (Steam) - Shell Side.

Shell side pressure	35 psig
Temperature	281°F .

3. Solution

a. Heat Load. i. Quantity Evaporated:

$$\text{Feed rate} = \frac{(235)(1010)}{(24)(454)} = 21.78 \text{ lb/hr} .$$

$$\text{Product rate} = \frac{(6.5)(1350)}{(24)(454)} = 0.805 \text{ lb/hr} .$$

Amount of water evaporated = 21.78 - 0.805 = 20.97 lb/hr .

ii. Required Sensible Heat:

$$= (\text{feed})\ C_p\ \Delta t$$

$$= (21.78)(1.00)(220 - 75) = 3158 \text{ Btu/hr} .$$

iii. Required Latent Heat:

$$= (\text{water evaporated})(\text{latent heat/lb})$$

$$= (20.97)(970) = 20{,}350 \text{ Btu/hr} .$$

iv. Total Heat Load:

Sensible heat	3,158
Latent heat	20,350
Total	23,508 Btu/hr .

b. Required Heat Transfer Surface. Since this unit is to be used in a pilot plant to insure flexibility of operation, the evaporator is designed to handle twice the above heat load, i.e., 47,000 Btu/hr.

i. Calculation of Δt: The holdup in this type of evaporator is large enough so that the temperature of the solution entering the bottom of the tubes is practically at the boiling point. Therefore, the temperature of the solution throughout the tube length is assumed constant.

$$\Delta t = 281 - 220 = 61^{\circ}F .$$

ii. Assumed Unit - Trial (1): Assume an overall coefficient of 300 Btu/(hr)(ft^2)($^{\circ}$F). The required surface will be

$$A = \frac{47{,}000}{(300)(61)} = 2.56 \text{ ft}^2 .$$

Use 3/4 in., 10 BWG tubes 4 feet long. The area of 1 foot of tube is 0.1963 square feet. The number of tubes required is

$$\frac{2.56}{(.1963)(4)} = 3.2 \text{ tubes} .$$

Call 5 tubes.

iii. Calculation of Condensing Steam Coefficient: The quantity of steam required is

$$\frac{\text{Load}}{\text{Btu/lb of steam}} = \frac{47{,}000}{925} = 50.8 \text{ lb/hr} .$$

Since there are five tubes, the quantity of steam condensing on each tube is

$$\frac{50.8}{5} = 10.16 \text{ lb/(hr)(tube)} .$$

Properties of water at 281°F:

Viscosity	0.13 centiposes
Sp Gr	0.927 g/cc
Thermal Conductivity	0.444 Btu/(hr)(ft^2)($^{\circ}$F/ft) .

Use Figure 34, Part V, for finding the coefficient. From this figure

$$h = 250$$
$$F_k = 5.6$$
$$F_s = 1.44$$
$$F_{ds} = 0.90 .$$

and

$$h_o = (250)(5.6)(1.44)(.90) = 1814 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

Using a fouling factor of 0.8

$$h_{of} = (0.8)(1814)$$

$$= 1451 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}.$$

iv. Tube Wall Coefficient:

$$h_t = \frac{kA_w}{L} = \frac{26 \times 0.54}{0.0111} = 1265 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}.$$

v. Boiling Coefficient: From Figure 171, of McAdam's Heat Transmission,[38] a boiling coefficient of 2,000 is chosen. This is a conservative value, and is valid for a low recirculation rate in the evaporator.

Using a factor of 0.8 to allow for fouling

$$h_{if} = (2,000)(0.8) = 1,600 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)}.$$

vi. Overall Transfer Coefficient:

$$U_f = \frac{1}{\frac{1}{1451} + \frac{1}{1265} + \frac{1}{1600}} = 485 \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} .$$

This value is much larger than the assumed coefficient; therefore the size of the unit is adequate.

c. Design of De-entrainment Section. A de-entrainment factor of 10^{-6} is required. Robinson and Gilliland,[63] give a number of plots which show the amount of entrainment in bubble plate columns for various plate spacings and types of liquids and vapors. Since the top of this evaporator is similar to the top plate of a distillation column, these data should be applicable.

Using a 4-inch layer of crinkle type mesh, 99.9% of any entrained liquid is removed, or 0.1% is carried out with the vapor. In order to insure only one part of liquid to be entrained for every 1,000,000 parts of vapor, an entrainment of 0.001 lb liquid/lb of vapor must be realized in the vapor de-entrainment section above the liquid-vapor deflector. From the above data $V\rho_v^{0.5}$ equals 0.45, based on an 18-in. plate spacing for steam and water.

From steam tables, ρ_v at atmospheric pressure = 0.0373 lb/ft^3, or

$$V = \frac{0.45}{(0.0373)(5)} = 2.33 \text{ ft/sec} .$$

Since 20.97 pounds of water are evaporated per hour, the volume of vapor produced is

$$(20.97)(26.80) = 562 \text{ ft}^3\text{/hr} .$$

The required diameter for the de-entrainment section of the evaporator is

$$d_E^2 = \frac{4\ (562)}{\pi\ (3600)(2.33)} = 0.0852\ .$$

$$d_E = 0.292 \text{ ft}, \quad \text{or} \quad (1.292)(12) = 3.5 \text{ in.}$$

To insure an adequate margin of safety, use a 5-inch diameter de-entrainment section. A liquid-vapor deflector is mounted three inches above the open ends of the evaporator tubes. The metal crinkle mesh is mounted in the top of the de-entrainment section, and an 18-inch space is provided between it and the liquid-vapor deflector.

The de-entrainment liquid runs from the de-entrainment section into a standpipe. The bottom of the standpipe is connected to the bottom of the evaporator tubes, and thus the liquid to be evaporated circulates from the standpipe to the less dense boiling column in the evaporator tubes.

Specifications	Hot Fluid (Shell Side)	Cold Fluid (Tube Side)
Material	Steam at 35 psig	Dilute soln. (spigs 1.01)
Inlet temperature	281°F	75°F
Outlet temperature	281°F	220°F
Duty	23,508 Btu/hr	
LMTD	59°F	
U-Clean		
Fouling factor	0.8	0.8
U-Fouled	397 Btu/(hr)(ft2)(°F)	
U-Required		
No. of passes/unit	1	1
No. of units	1	1
Surface/unit	3.93 sq ft	
Required surface	2.00 sq ft	
Shell size	5 in. (?)	
Shell type		
No. of tubes/unit		5
Total number of tubes		5
Tube length		4 ft
Tube size		3/4 in. OD, 10 BWG
Tube pitch		1 in. sq
Baffles		

Special modifications:

Diameter of de-entrainment section	5 in.
Thickness of metal crinkle mesh	4 in.
Distance between crinkle mess and vapor-liquid deflector	18 in.
Distance between vapor-liquid deflector above top of tubes	3 in.
Entrainment	1 lb liquid/ 10^6 lb of vapor .

V. TABLES AND CURVES

1. Heat Transfer Curves.
2. Tables.
3. "Net Free Area" Curves.
4. "Pressure" Drop Curves.
5. Proforma Heat Exchanger Specifications Sheet.

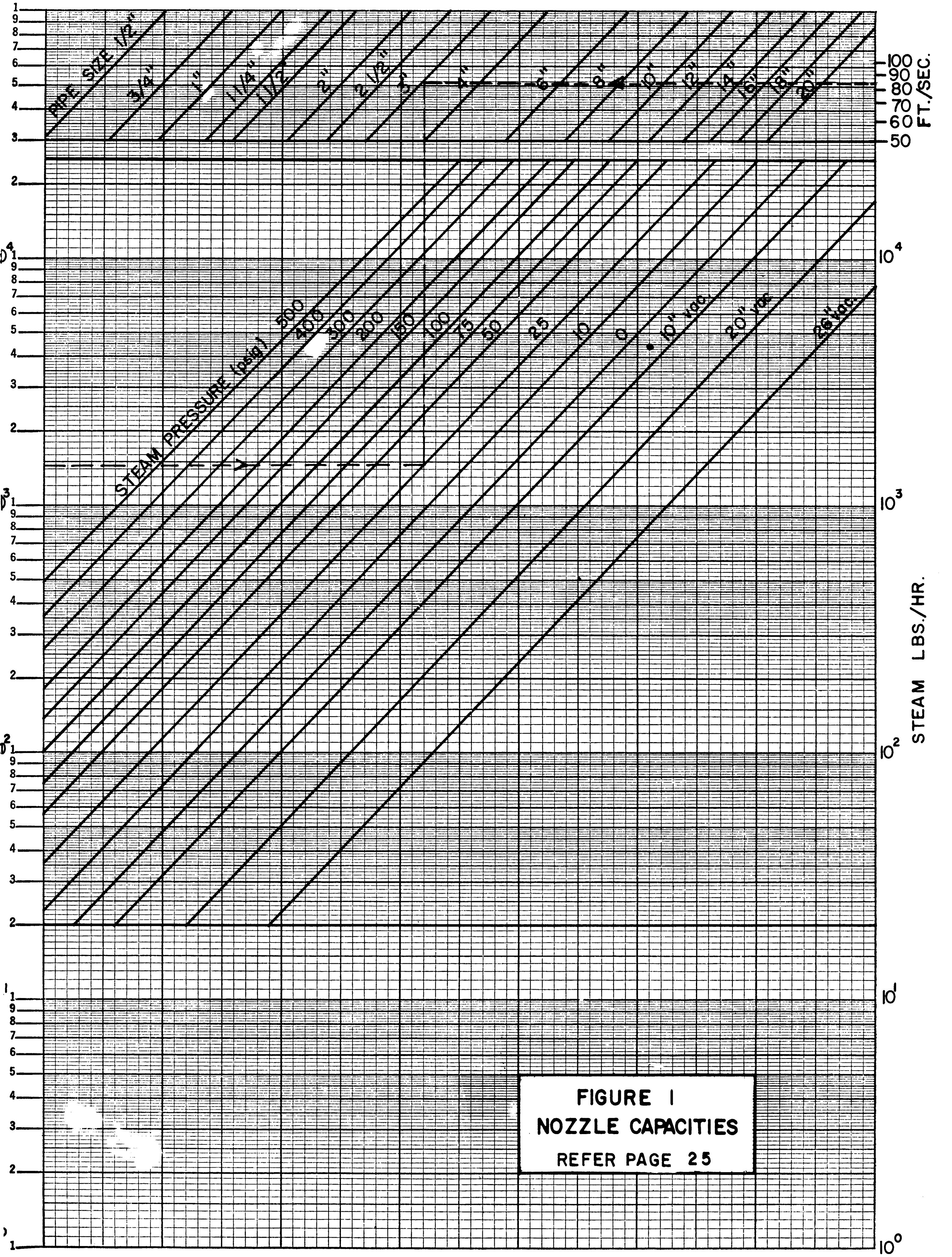
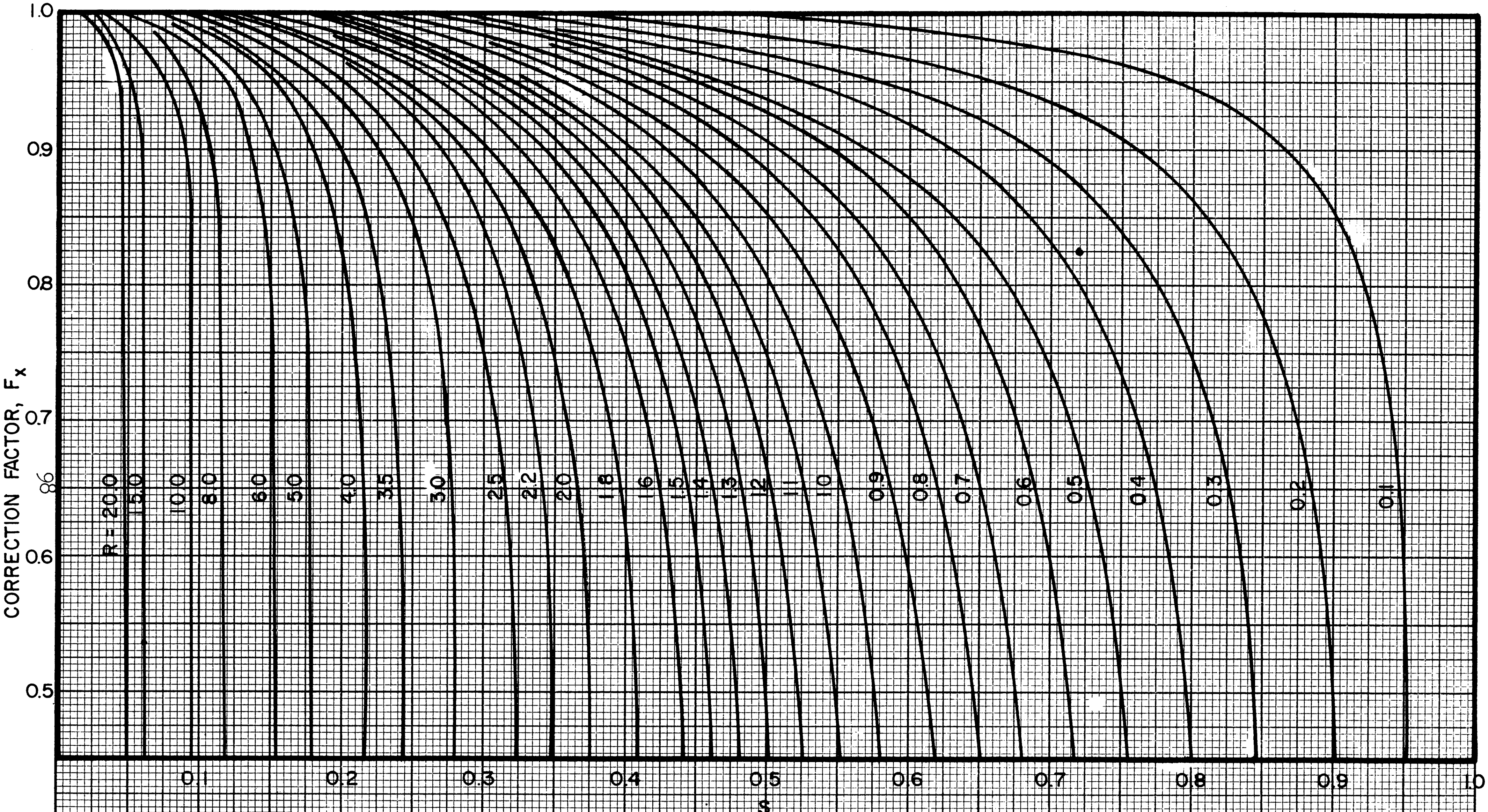


FIGURE I
NOZZLE CAPACITIES
REFER PAGE 25



$$S = \frac{t_2 - t_1}{T_1 - t_1}$$

$$R_t = \frac{T_1 - T_2}{t_2 - t_1}$$

FIGURE 2

M.T.D. CORRECTION FACTORS

1 SHELL PASS
2 OR MORE TUBE PASSES

REFER PAGE 36

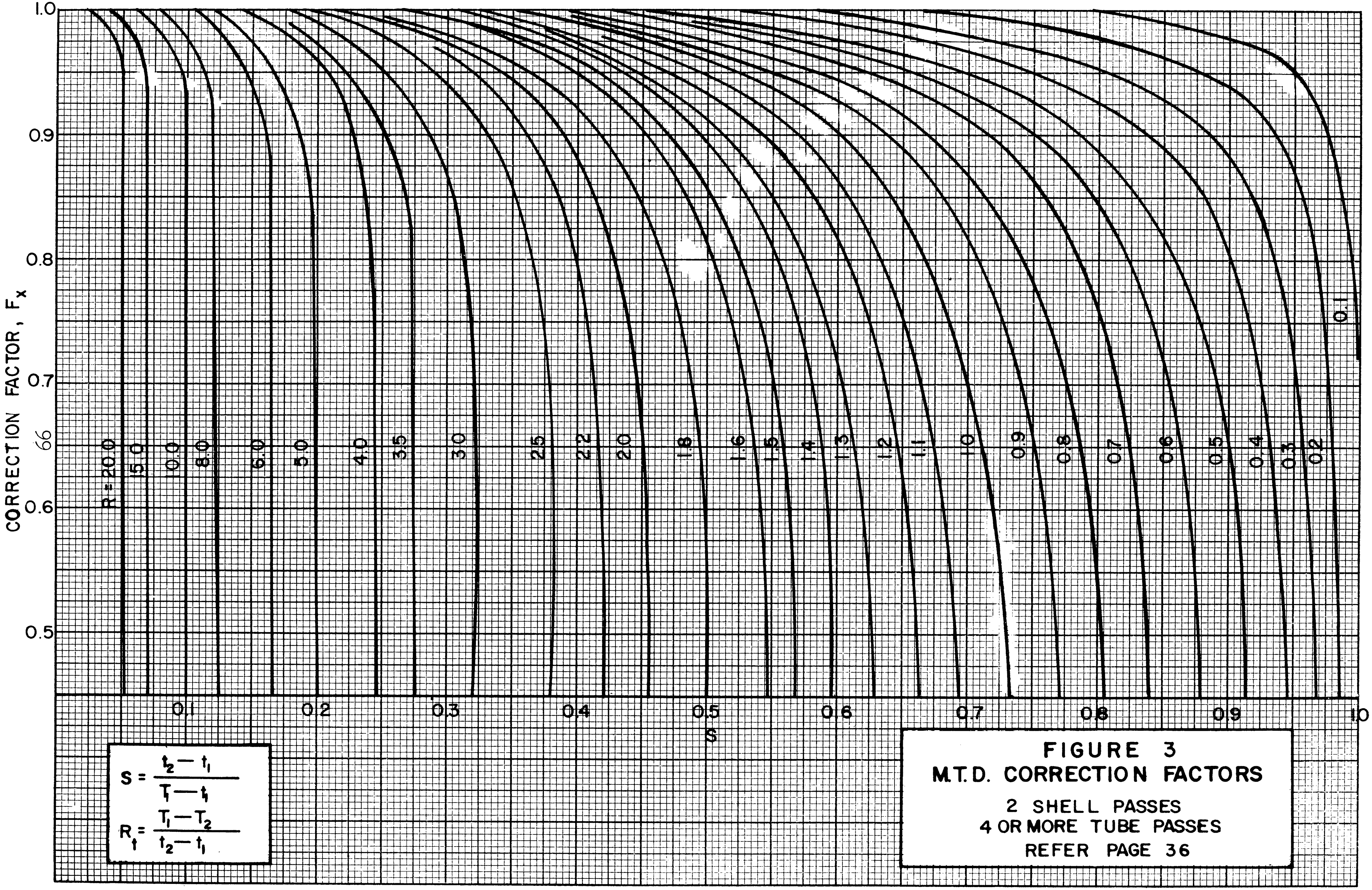
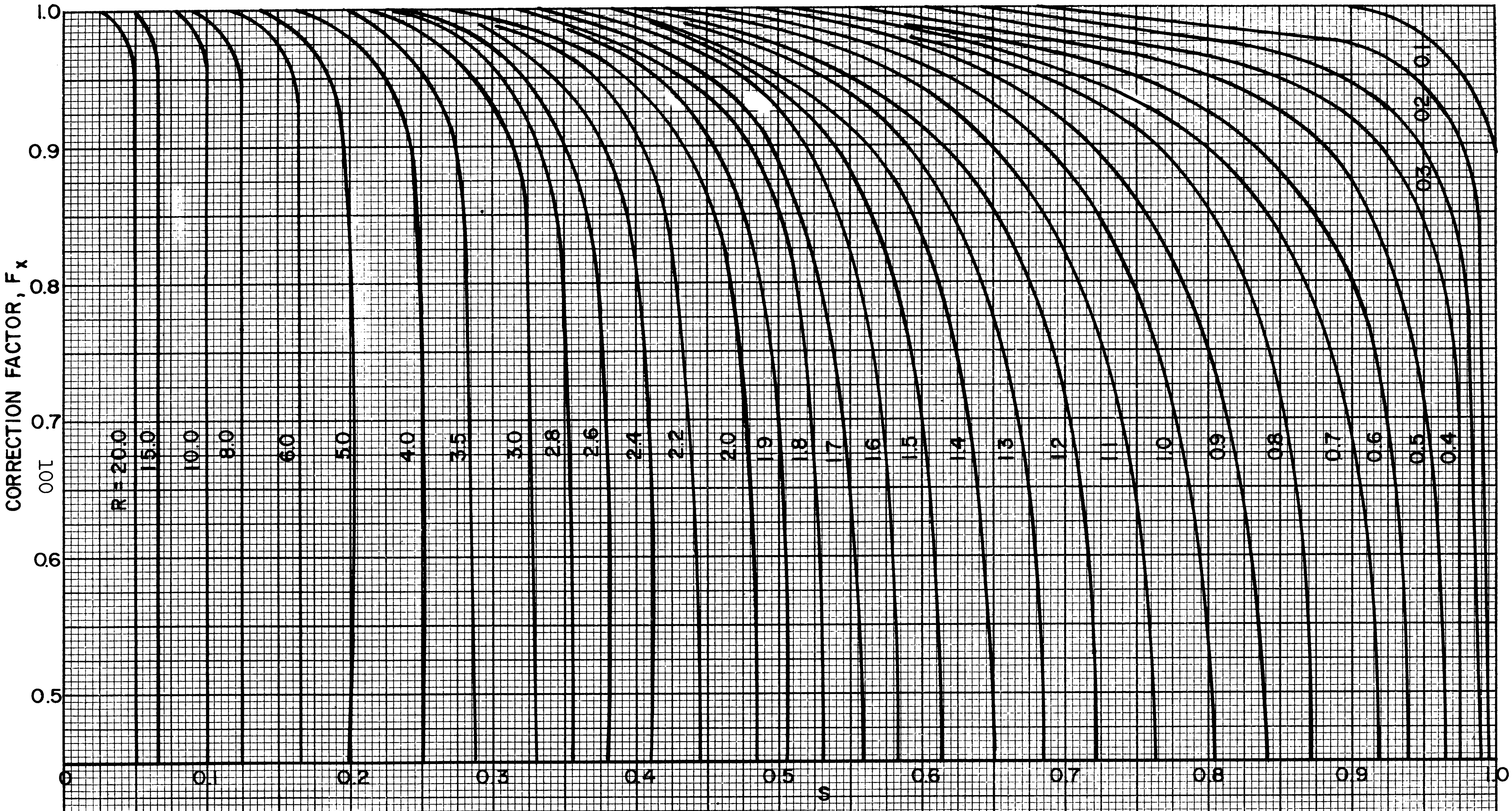


FIGURE 3
M.T.D. CORRECTION FACTORS
2 SHELL PASSES
4 OR MORE TUBE PASSES
REFER PAGE 36



$$S = \frac{t_2 - t_1}{T_1 - t_1}$$

$$R_t = \frac{T_1 - T_2}{t_2 - t_1}$$

FIGURE 4
M.T.D. CORRECTION FACTORS
3 SHELL PASSES
6 OR MORE TUBE PASSES
REFER PAGE 36

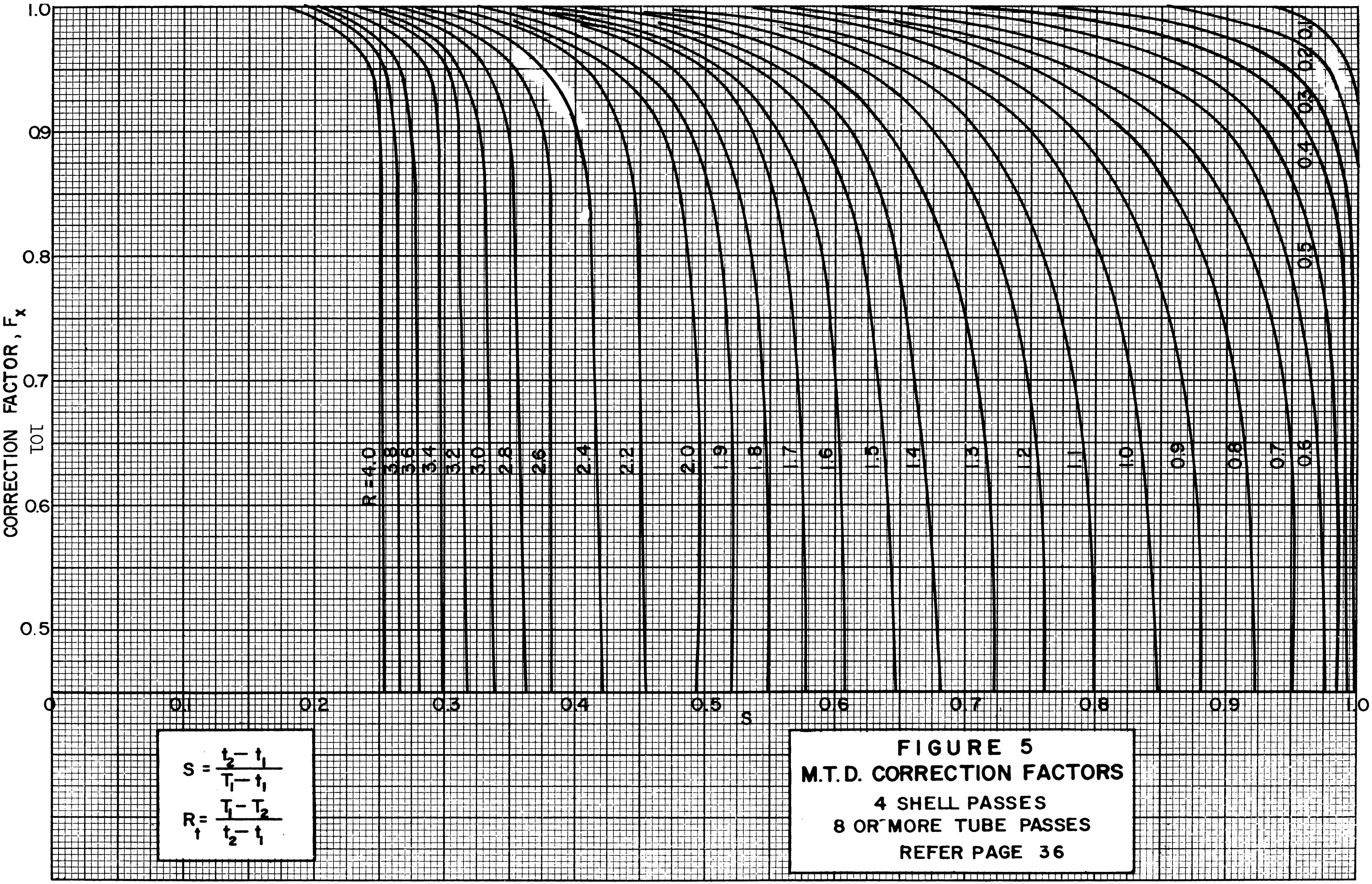
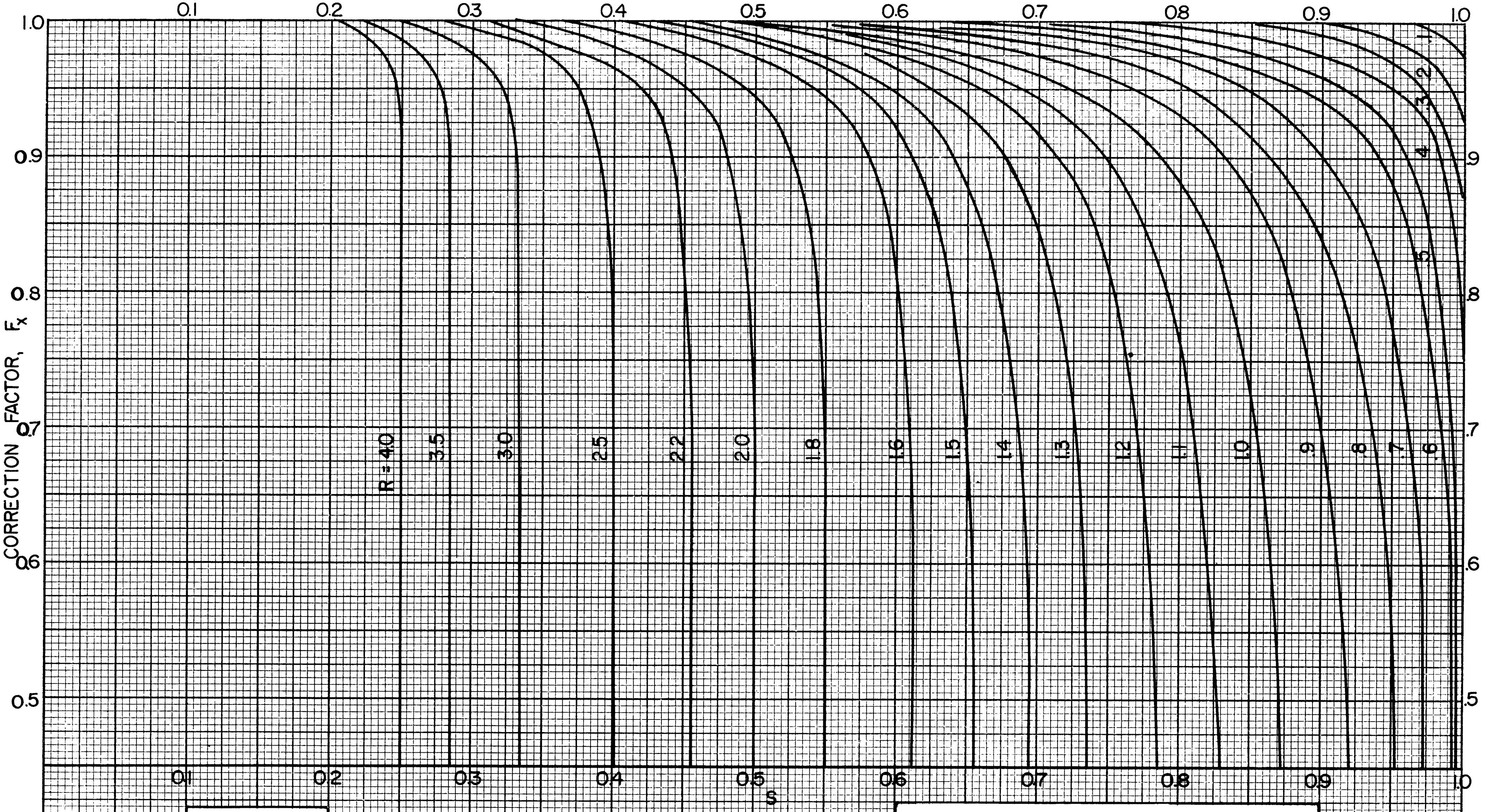


FIGURE 5
M.T.D. CORRECTION FACTORS
4 SHELL PASSES
8 OR MORE TUBE PASSES
REFER PAGE 36



$$S = \frac{t_2 - t_1}{T_1 - t_1}$$

$$R = \frac{T_1 - T_2}{t_1 - t_2}$$

FIGURE 6

M.T.D. CORRECTION FACTORS

5 SHELL PASSES

10 OR MORE TUBE PASSES

REFER PAGE 36

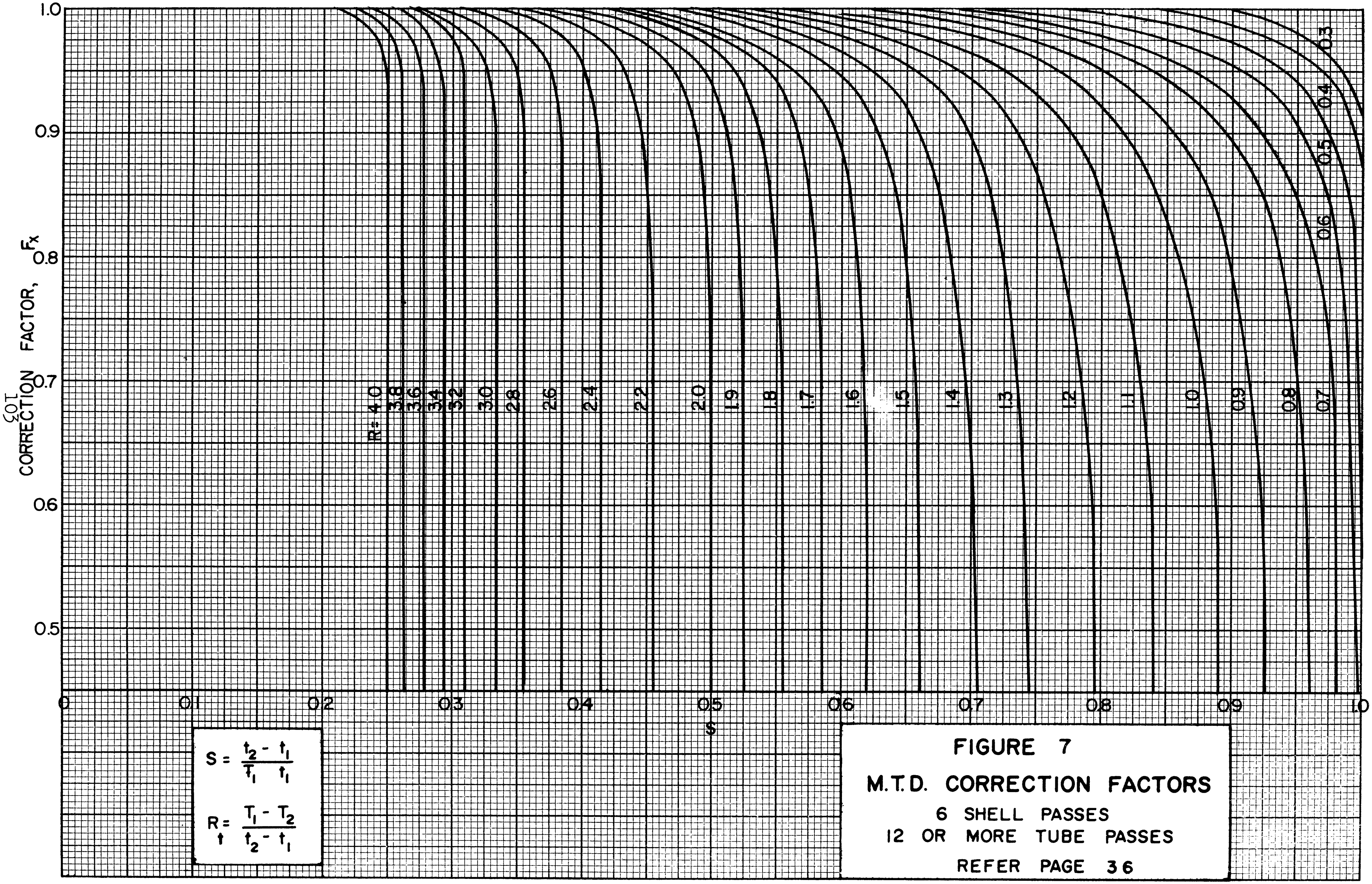
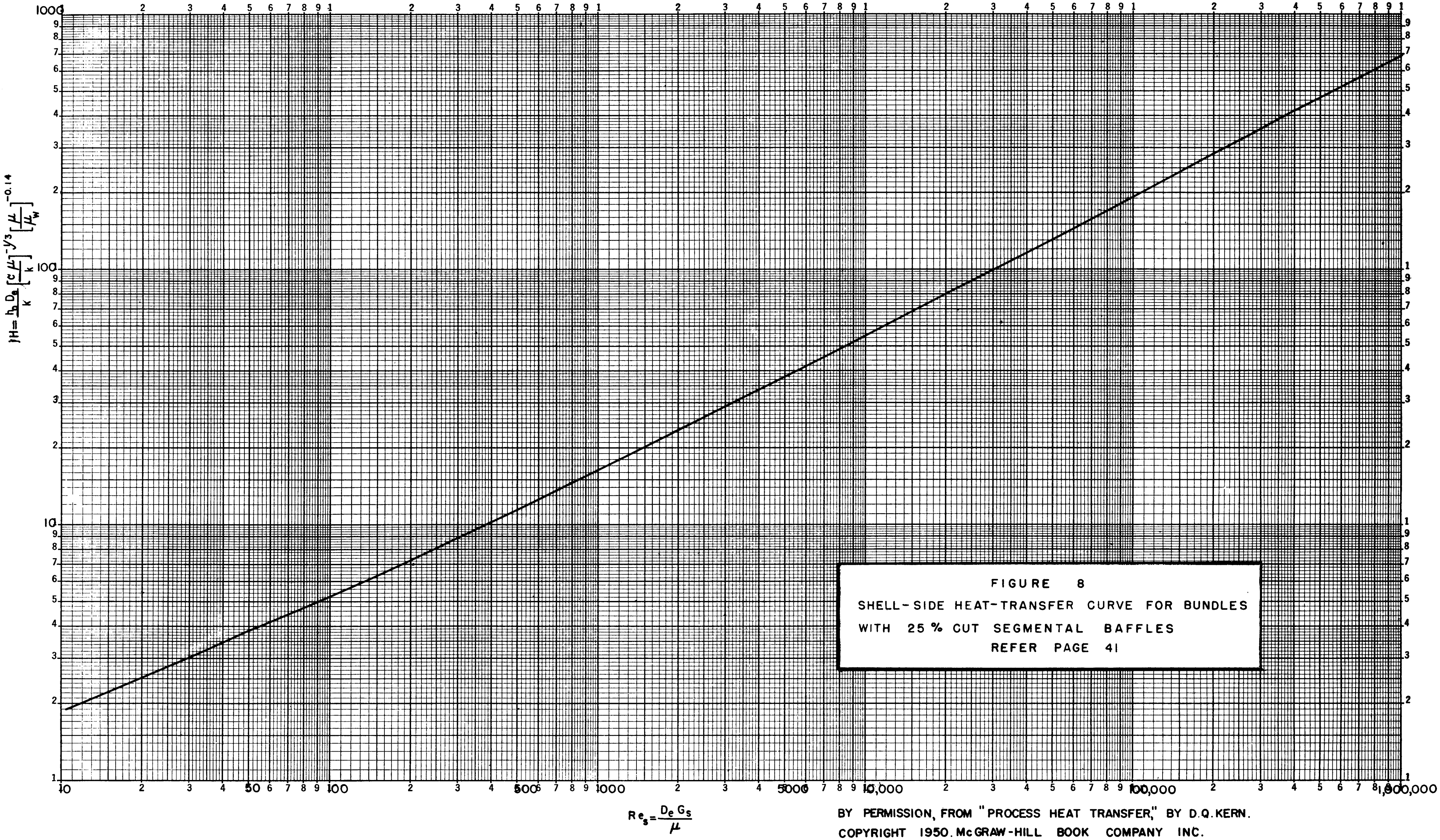


FIGURE 7

M.T.D. CORRECTION FACTORS

6 SHELL PASSES
12 OR MORE TUBE PASSES

REFER PAGE 36



$jH = \frac{h_o D_e}{k}\left[\frac{c\mu}{k}\right]^{-1/3}\left[\frac{\mu}{\mu_w}\right]^{-0.14}$
$Re_s = \frac{D_e G_s}{\mu}$
FIGURE 8
SHELL-SIDE HEAT-TRANSFER CURVE FOR BUNDLES
WITH 25 % CUT SEGMENTAL BAFFLES
REFER PAGE 41

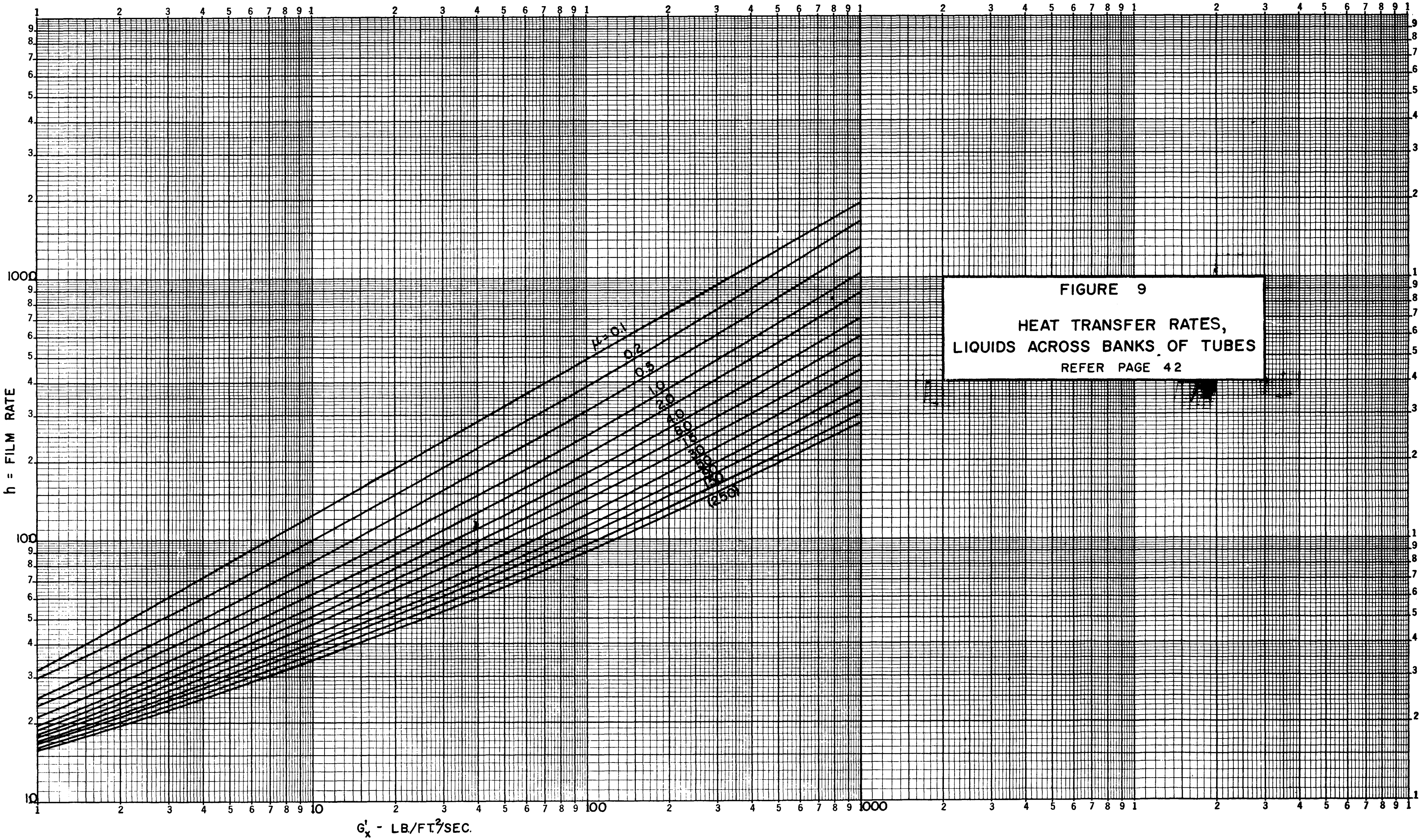


FIGURE 9
HEAT TRANSFER RATES,
LIQUIDS ACROSS BANKS OF TUBES
REFER PAGE 42
h = FILM RATE
G'x - LB./FT.²/SEC.
μ = 0.1
0.2
0.5
1.0
2.0
4.0
6.0
10
100
1000

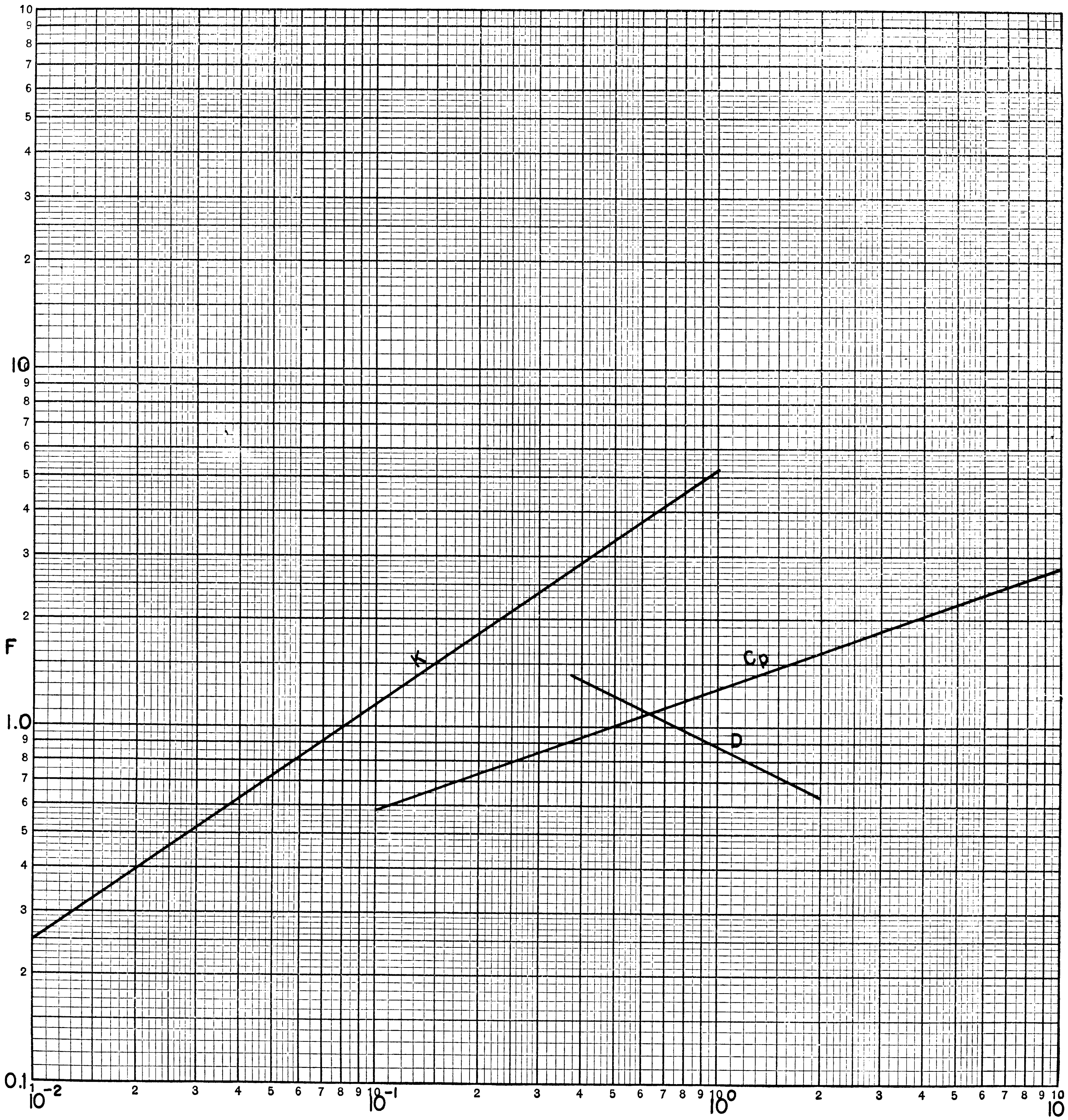


FIGURE 9 - a

CORRECTION FACTORS FOR HEAT TRANSFER RATES
FOR LIQUIDS IN CROSS-FLOW ON SHELL SIDE

REFER FIG. 9

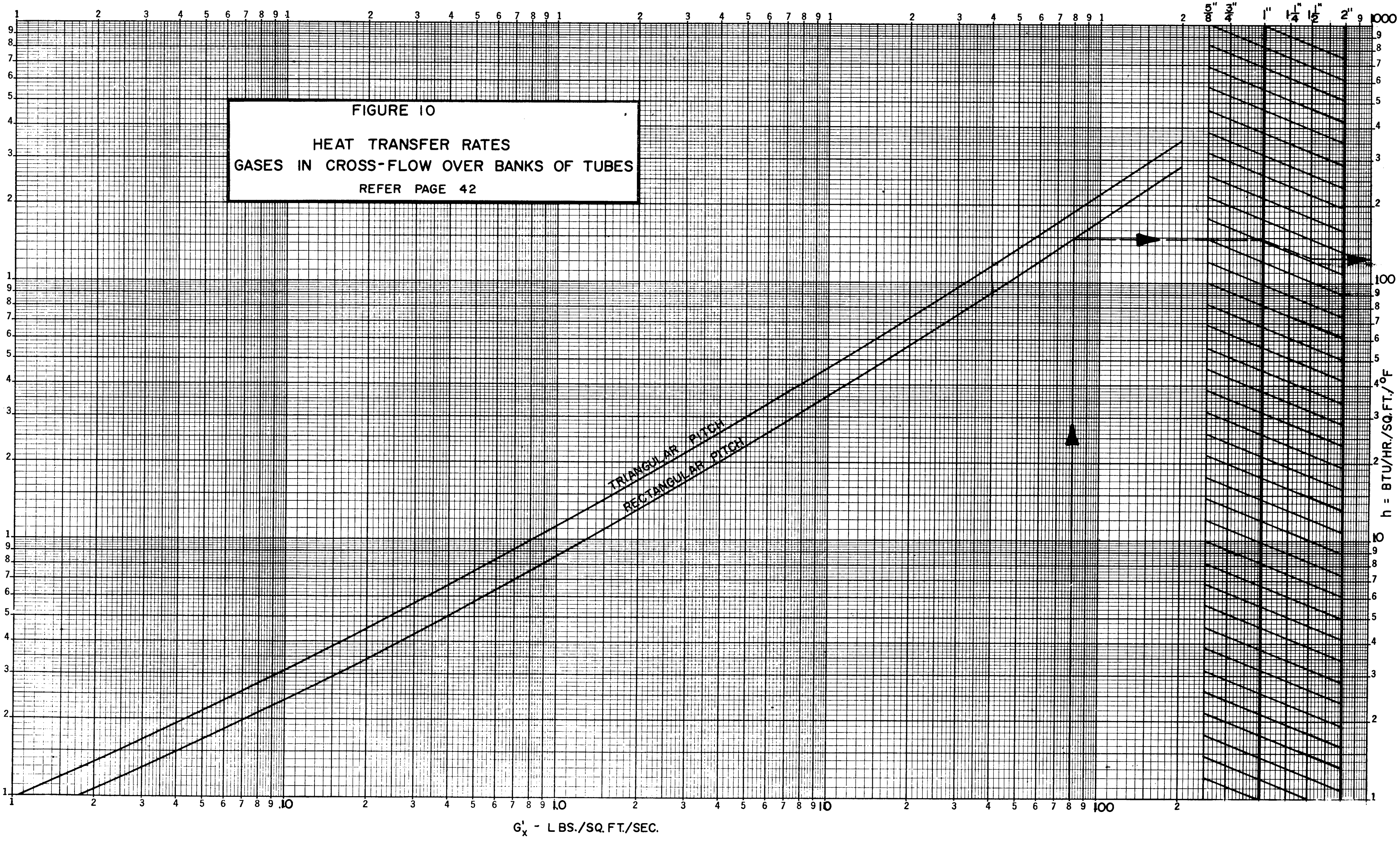


FIGURE 10
HEAT TRANSFER RATES
GASES IN CROSS-FLOW OVER BANKS OF TUBES
REFER PAGE 42
TRIANGULAR PITCH
RECTANGULAR PITCH
5/8" 3/4" 1" 1 1/4" 1 1/2" 2"
h = BTU/HR./SQ.FT./°F
G'x - LBS./SQ.FT./SEC.

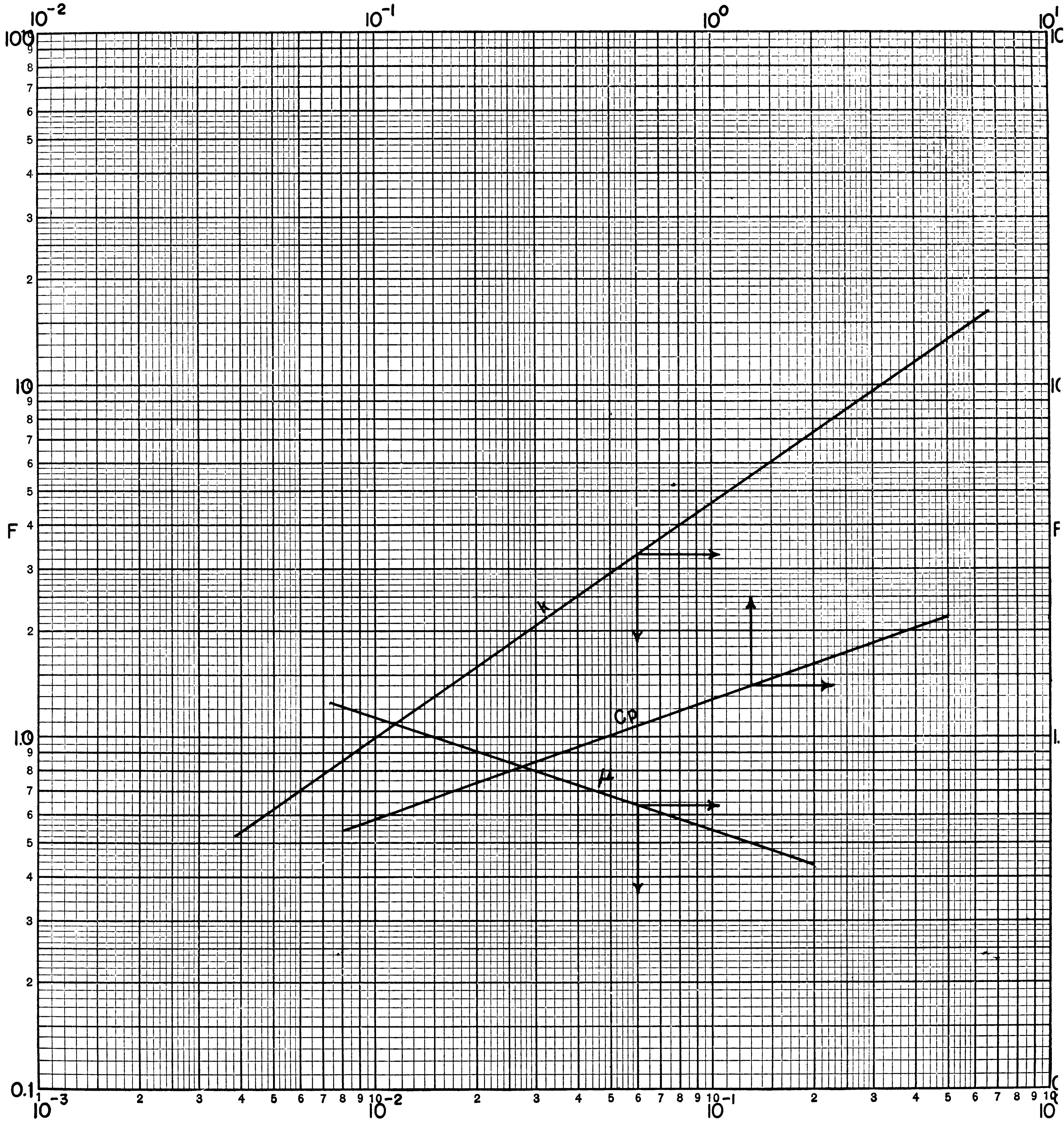


FIGURE 10 – a

CORRECTION FACTORS FOR HEAT TRANSFER RATES FOR GASES IN SHELL-SIDE CROSS FLOW

REFER FIG. 10

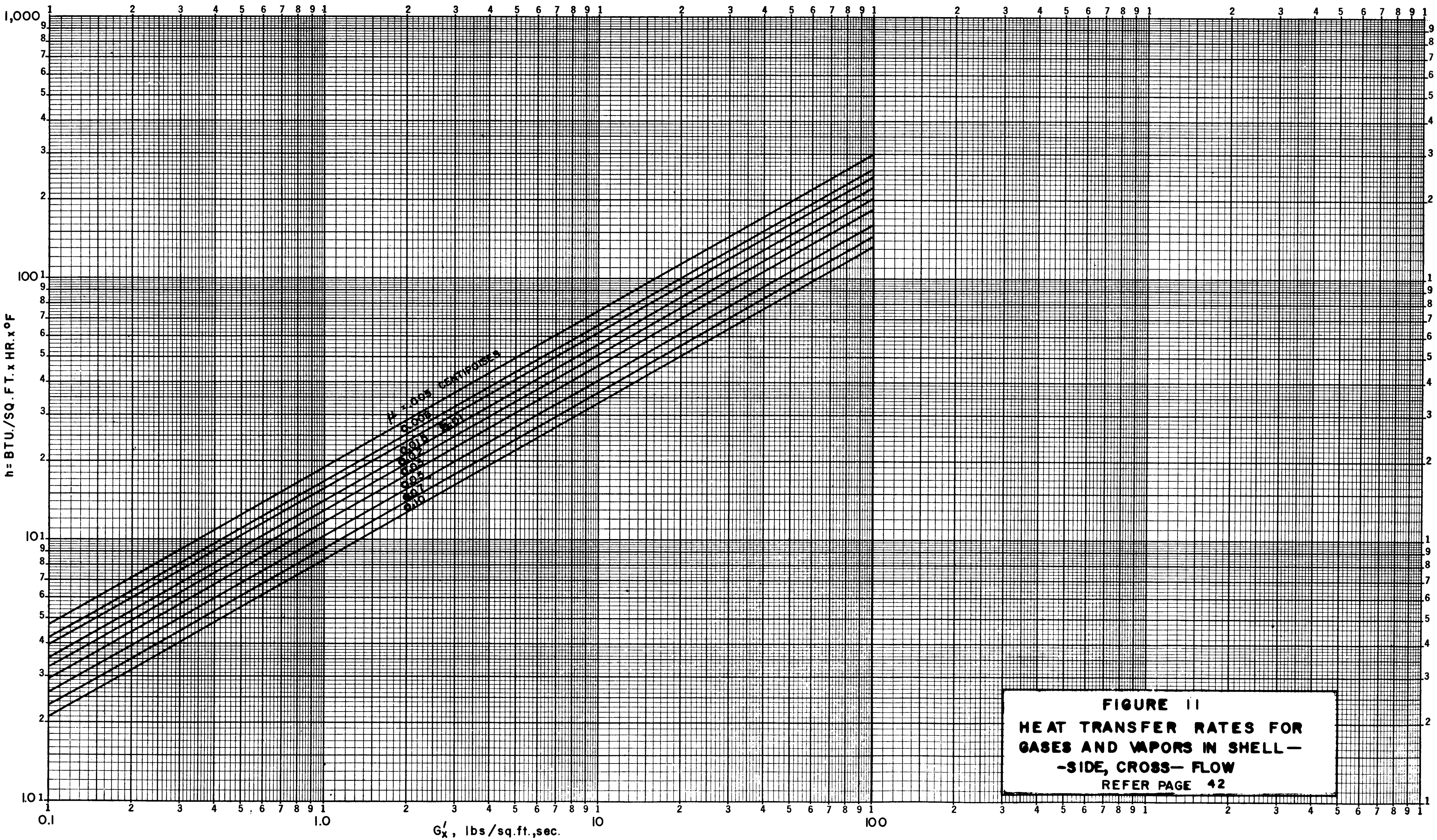


FIGURE 11
HEAT TRANSFER RATES FOR GASES AND VAPORS IN SHELL-SIDE, CROSS-FLOW
REFER PAGE 42

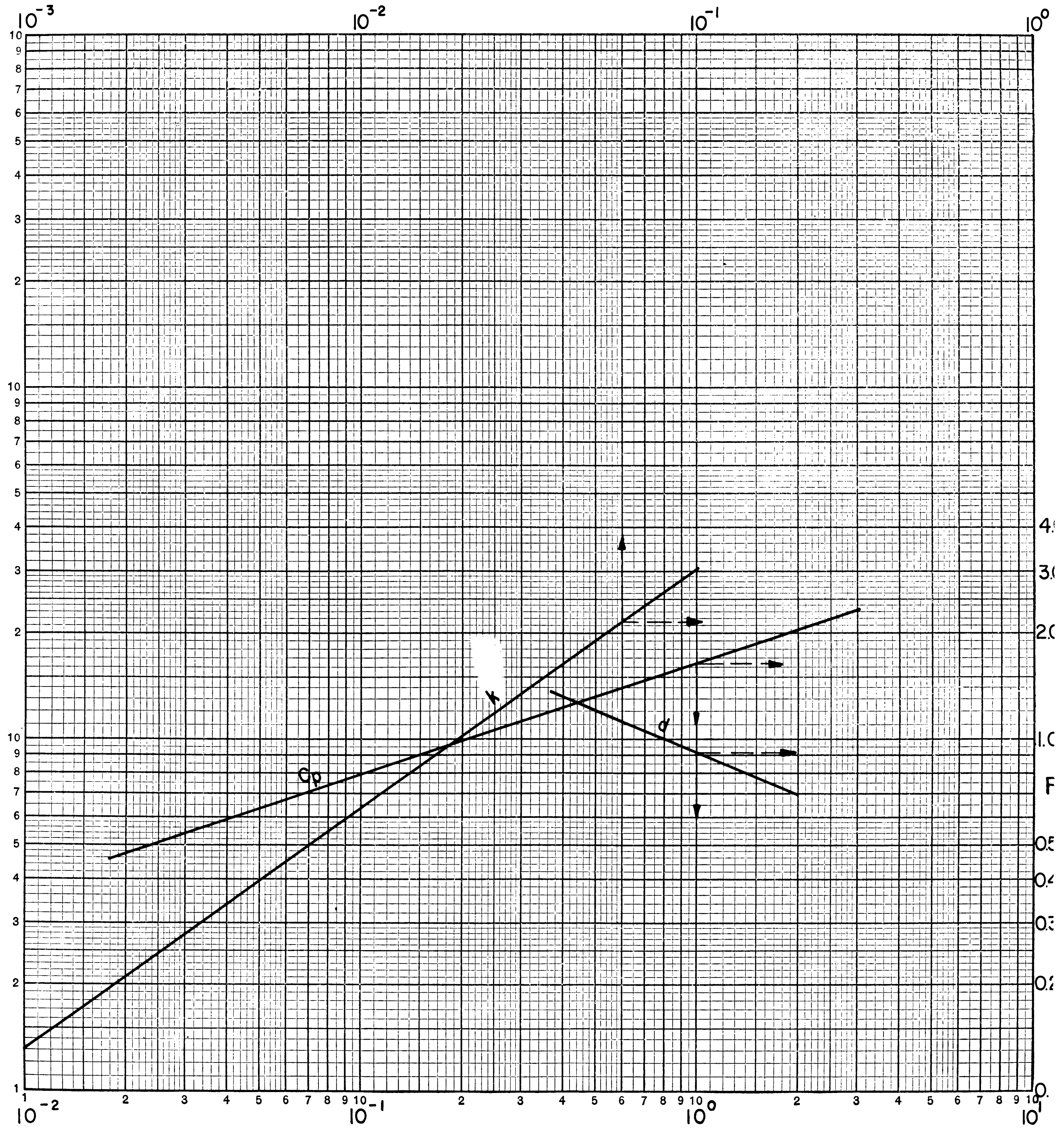


FIGURE 11 – a

CORRECTION FACTORS FOR HEAT TRANSFER RATES FOR GASES & VAPORS IN SHELL — — SIDE CROSS-FLOW

REFER FIG. 11

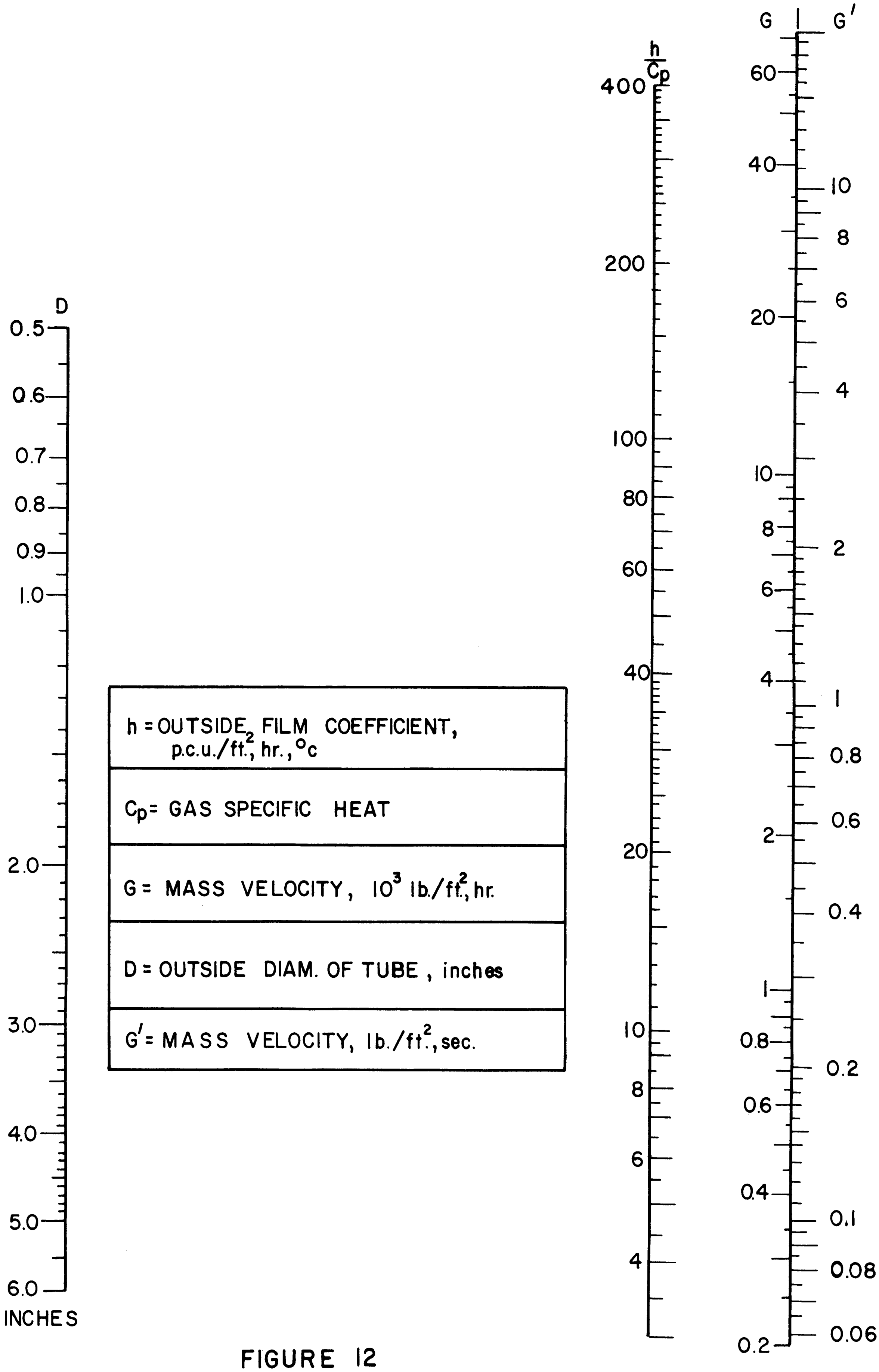


FIGURE 12
HEAT TRANSFER RATES
FOR GASES ACROSS TUBES

REFER PAGE 42

BY PERMISSION OF THE A.S.M.E.

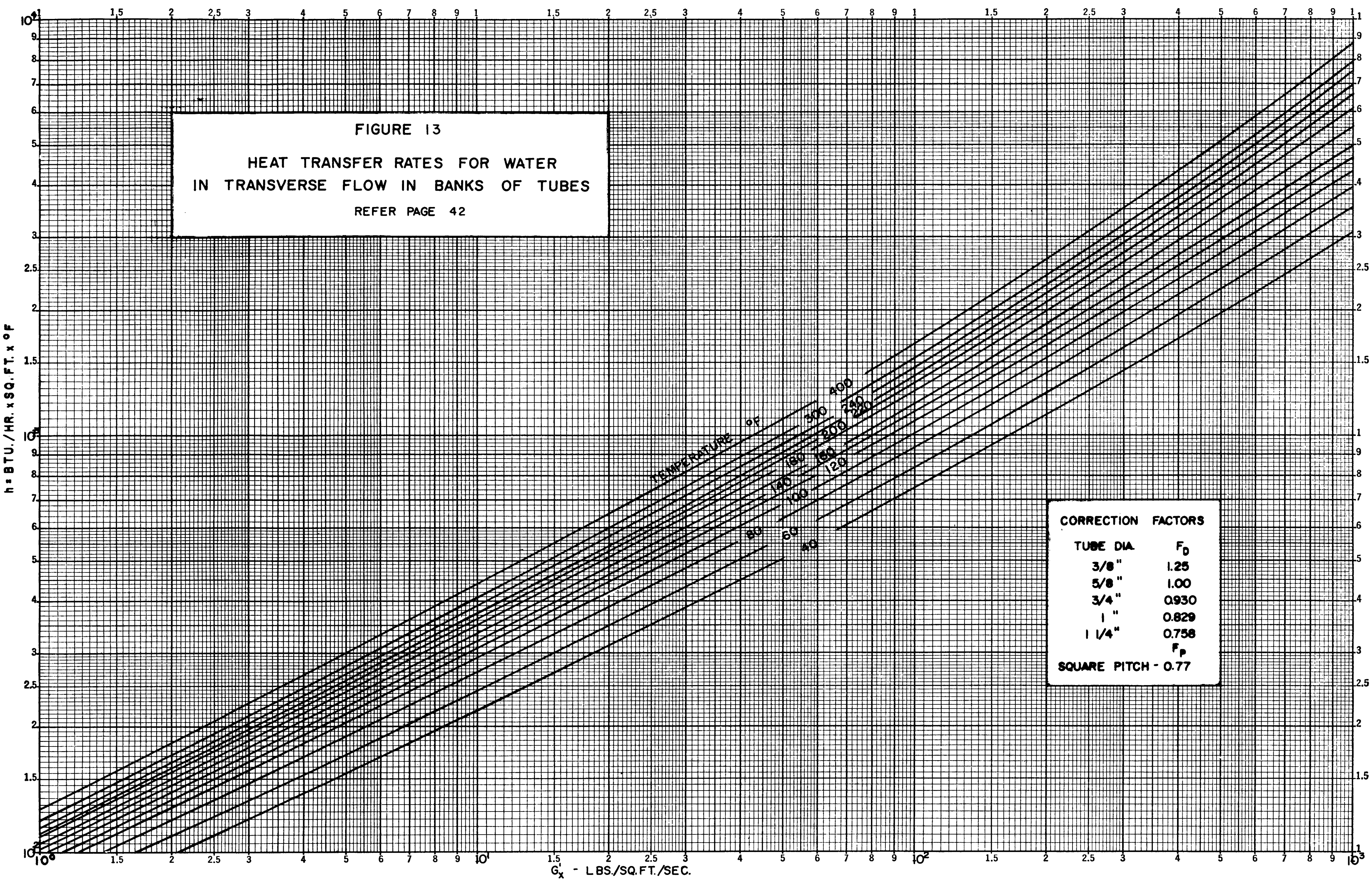


FIGURE 13

HEAT TRANSFER RATES FOR WATER IN TRANSVERSE FLOW IN BANKS OF TUBES

REFER PAGE 42

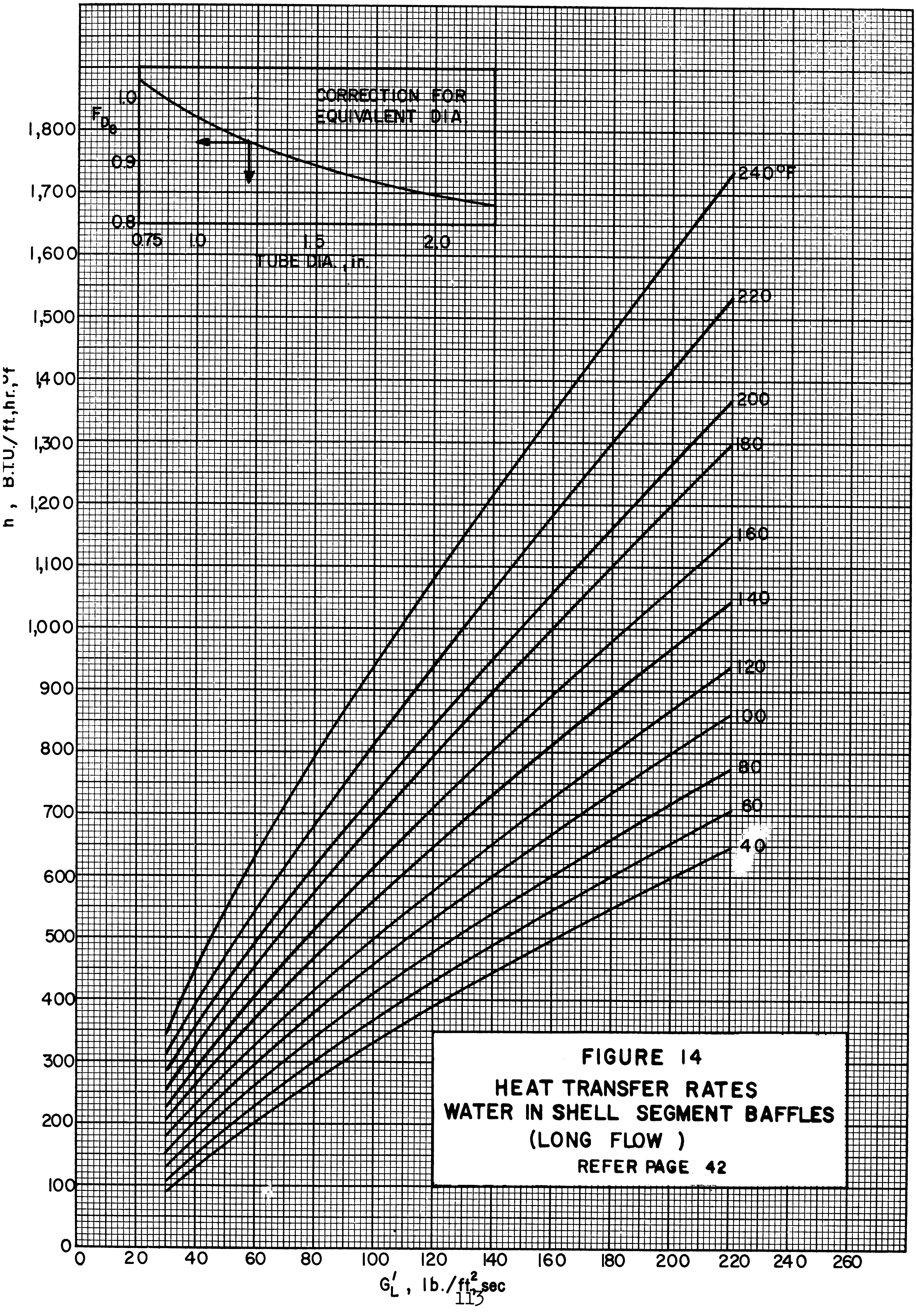
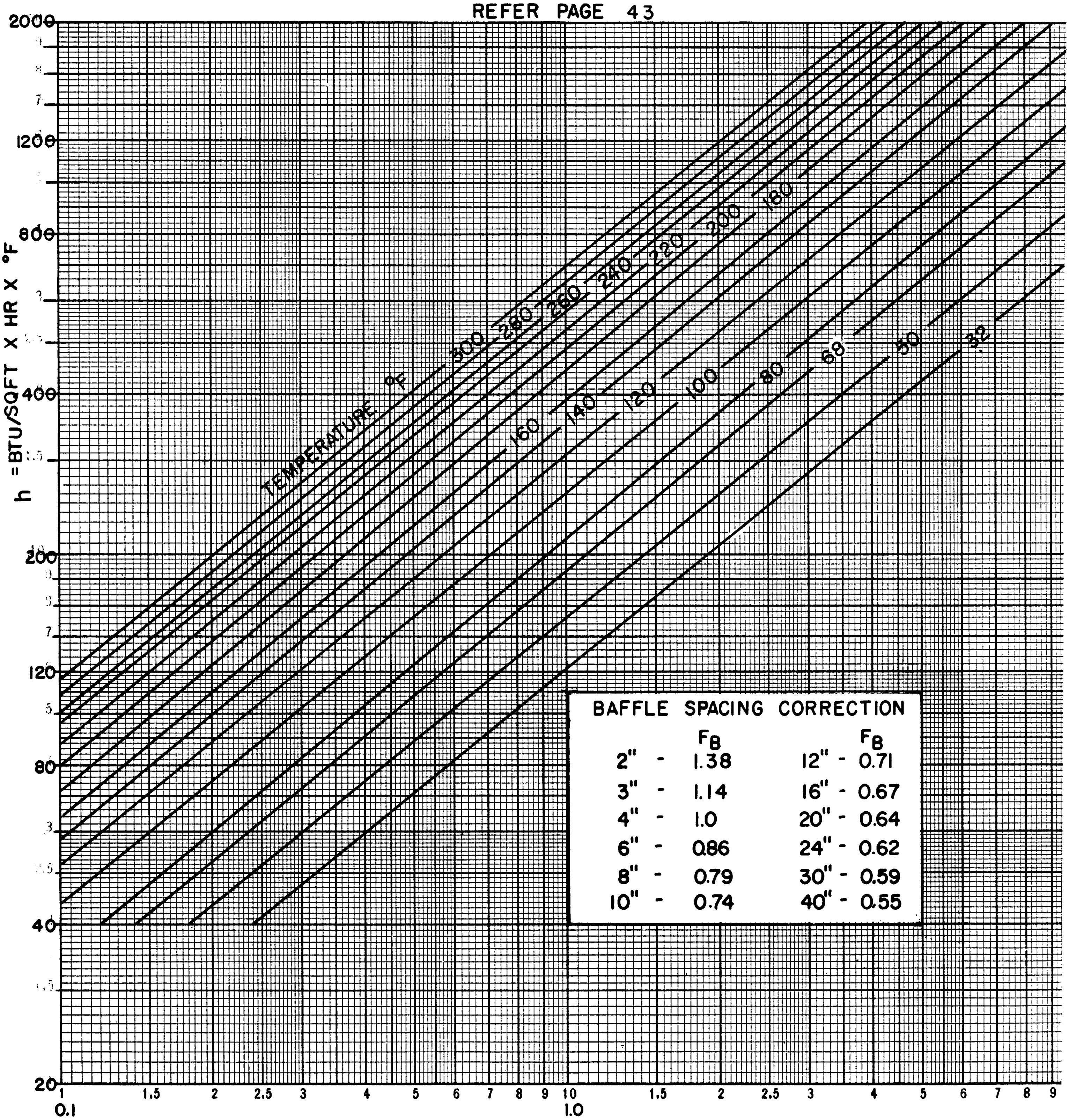


FIGURE 14
HEAT TRANSFER RATES
WATER IN SHELL SEGMENT BAFFLES
(LONG FLOW)
REFER PAGE 42

FIGURE 15

HEAT TRANSFER COEFFICIENTS
WATER IN SHELL,- ORIFICE BAFFLES

REFER PAGE 43



BAFFLE SPACING CORRECTION

	F_B		F_B
2" -	1.38	12" -	0.71
3" -	1.14	16" -	0.67
4" -	1.0	20" -	0.64
6" -	0.86	24" -	0.62
8" -	0.79	30" -	0.59
10" -	0.74	40" -	0.55

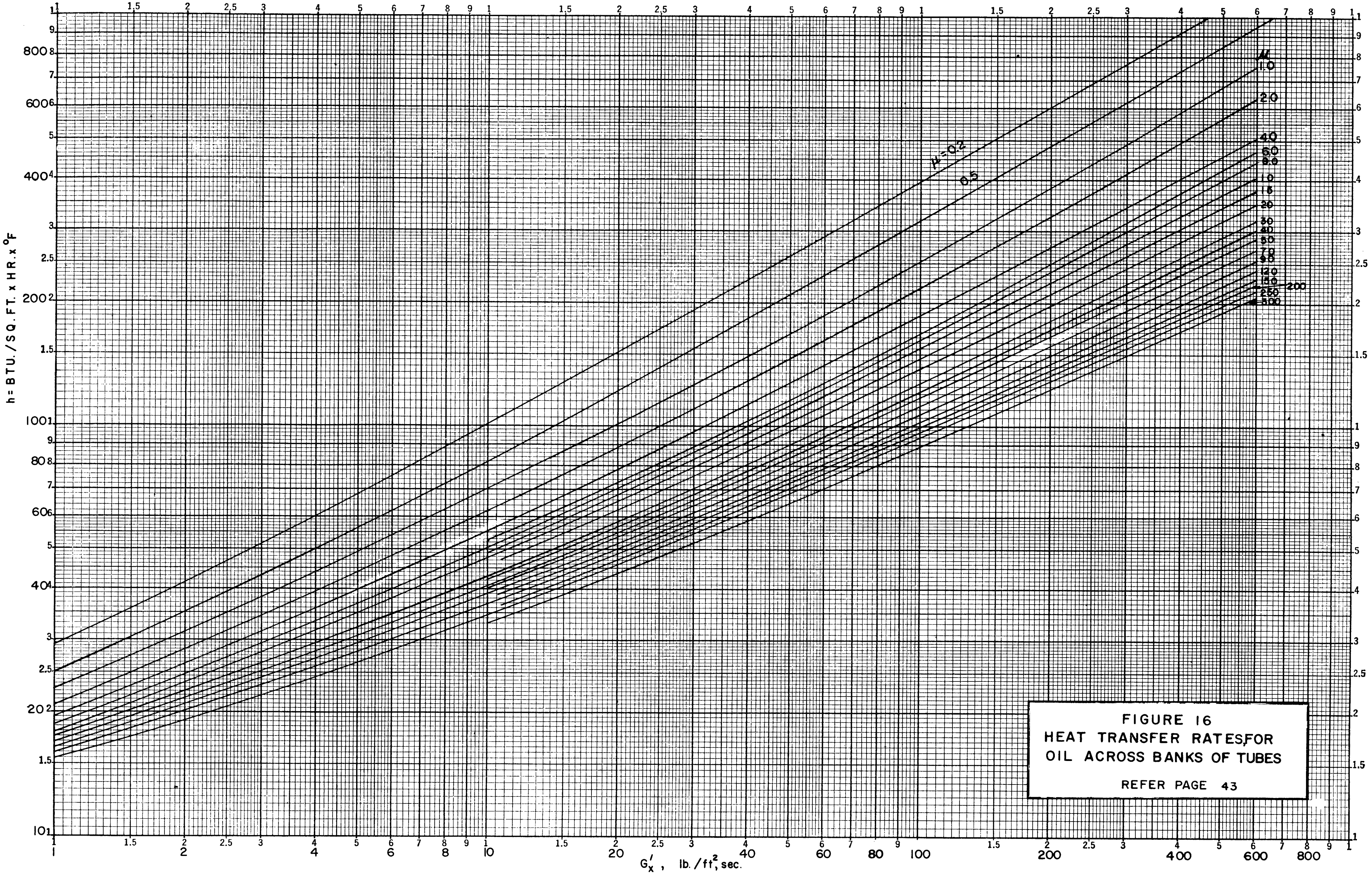


FIGURE 16
HEAT TRANSFER RATES FOR OIL ACROSS BANKS OF TUBES

REFER PAGE 43

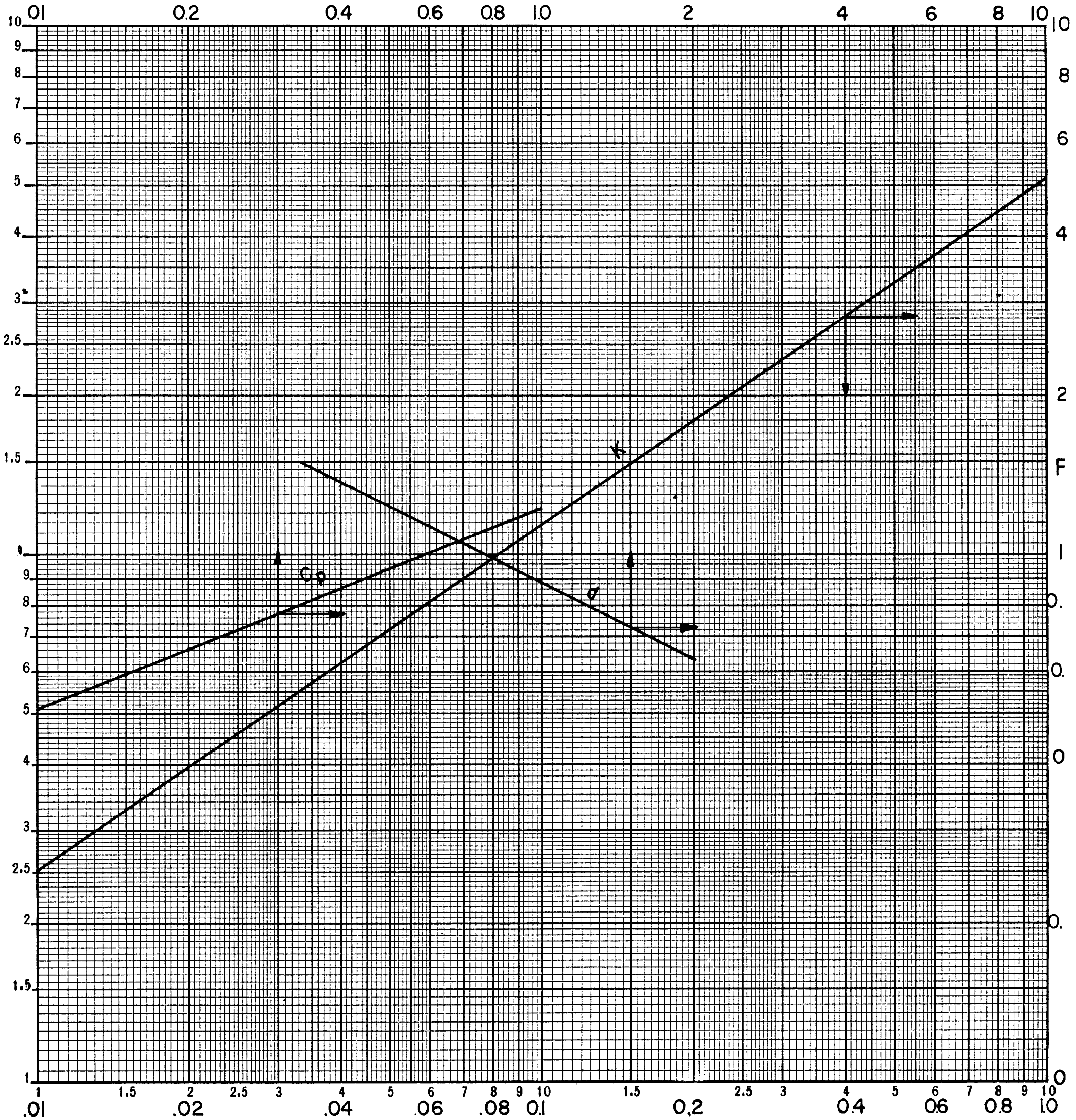


FIGURE 16-a

CORRECTIONS FOR HEAT TRANSFER RATES FOR OIL IN CROSS-FLOW OVER TUBE BANKS

REFER FIG. 16

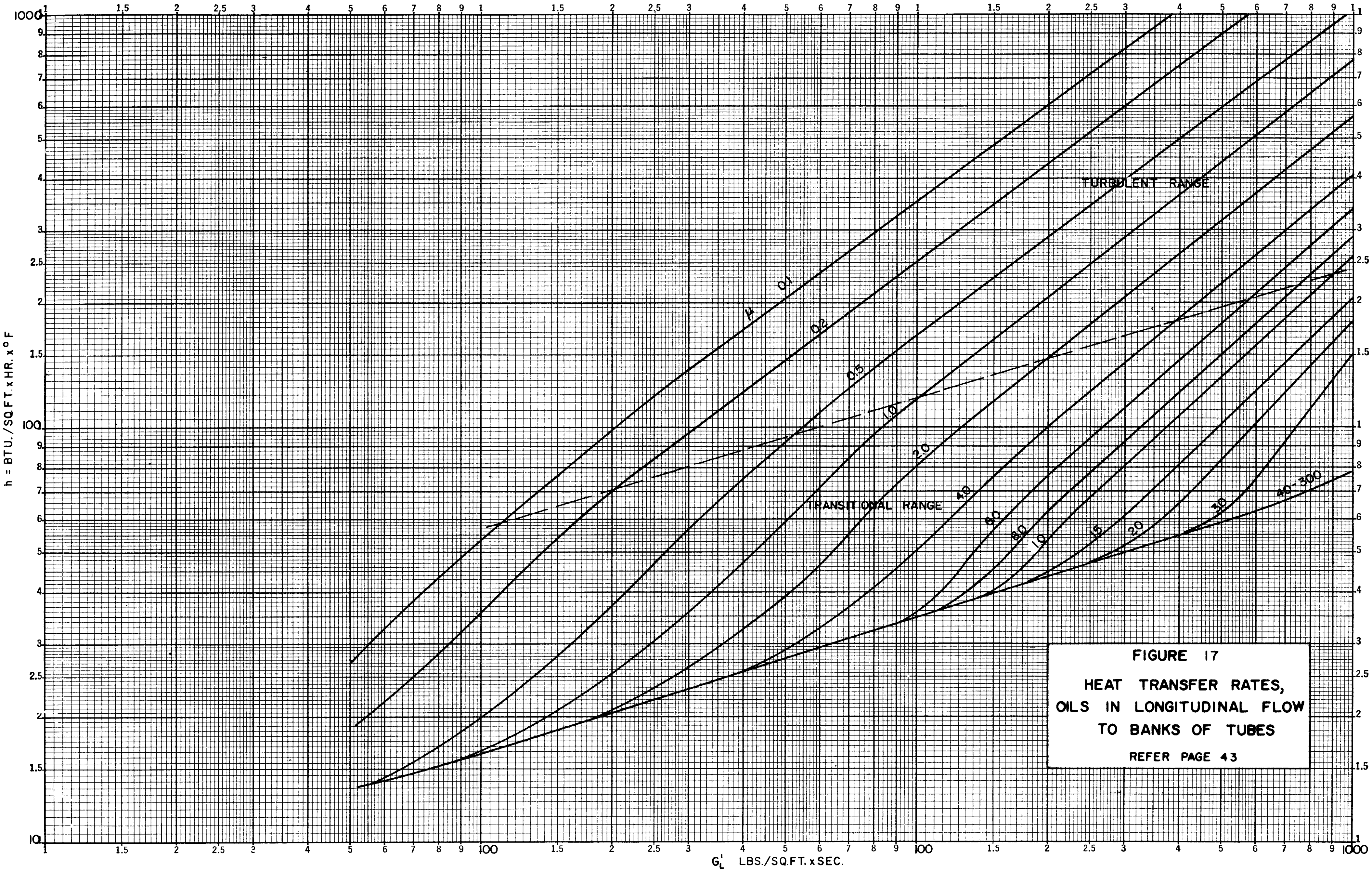


FIGURE 17

HEAT TRANSFER RATES, OILS IN LONGITUDINAL FLOW TO BANKS OF TUBES

REFER PAGE 43

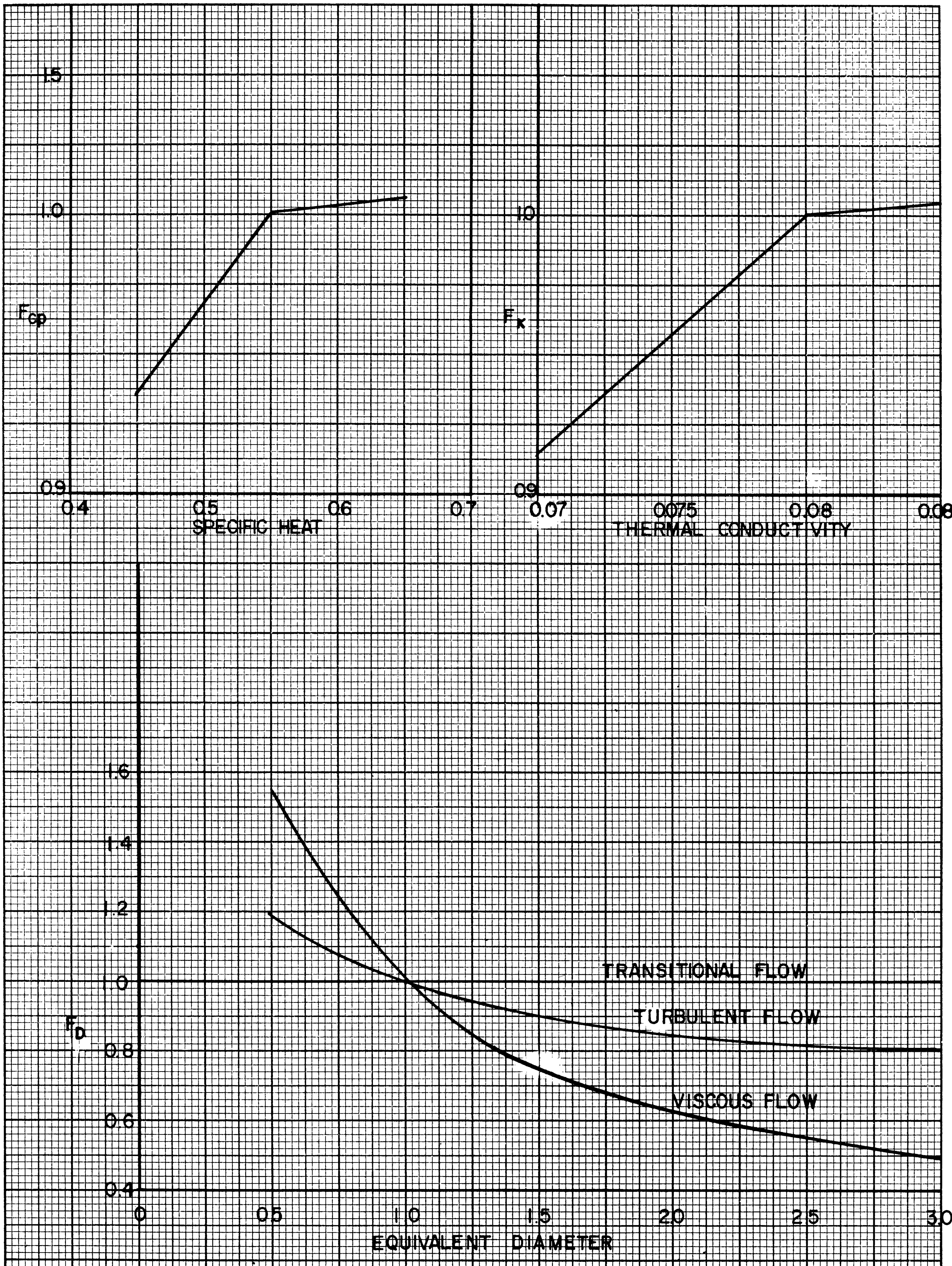
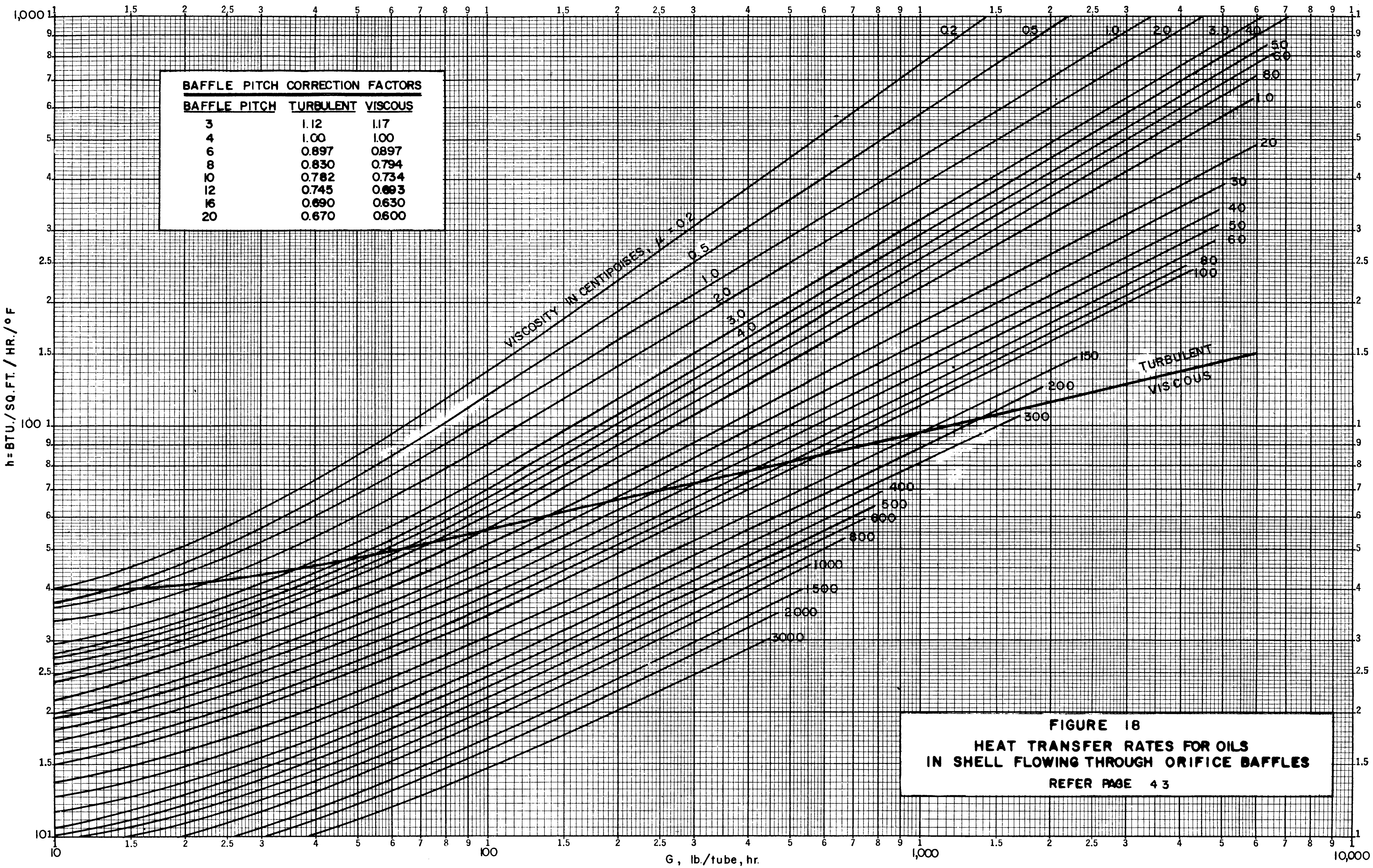


FIGURE 17-a

CORRECTION FACTORS FOR TRANSFER RATES FOR OIL IN LONGITUDINAL FLOW TO BANKS OF TUBES

REFER FIG. 17



BAFFLE PITCH CORRECTION FACTORS		
BAFFLE PITCH	TURBULENT	VISCOUS
3	1.12	1.17
4	1.00	1.00
6	0.897	0.897
8	0.830	0.794
10	0.782	0.734
12	0.745	0.693
16	0.690	0.630
20	0.670	0.600

FIGURE 18
HEAT TRANSFER RATES FOR OILS
IN SHELL FLOWING THROUGH ORIFICE BAFFLES
REFER PAGE 43

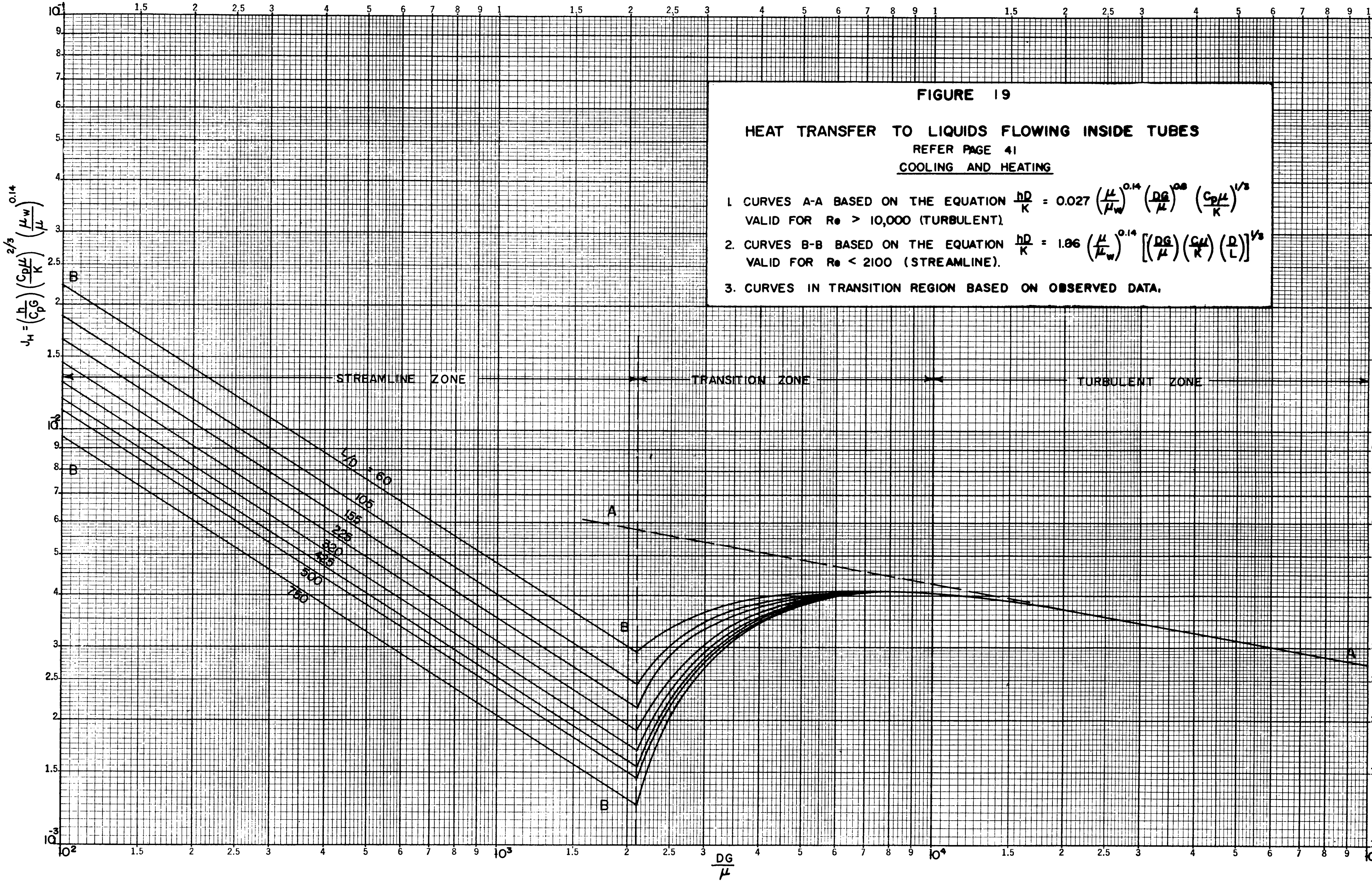


FIGURE 19

HEAT TRANSFER TO LIQUIDS FLOWING INSIDE TUBES

REFER PAGE 41

COOLING AND HEATING

1. CURVES A-A BASED ON THE EQUATION $\frac{hD}{K} = 0.027\left(\frac{\mu}{\mu_w}\right)^{0.14}\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{C_p\mu}{K}\right)^{1/3}$ VALID FOR Re > 10,000 (TURBULENT).
2. CURVES B-B BASED ON THE EQUATION $\frac{hD}{K} = 1.86\left(\frac{\mu}{\mu_w}\right)^{0.14}\left[\left(\frac{DG}{\mu}\right)\left(\frac{C\mu}{K}\right)\left(\frac{D}{L}\right)\right]^{1/3}$ VALID FOR Re < 2100 (STREAMLINE).
3. CURVES IN TRANSITION REGION BASED ON OBSERVED DATA.

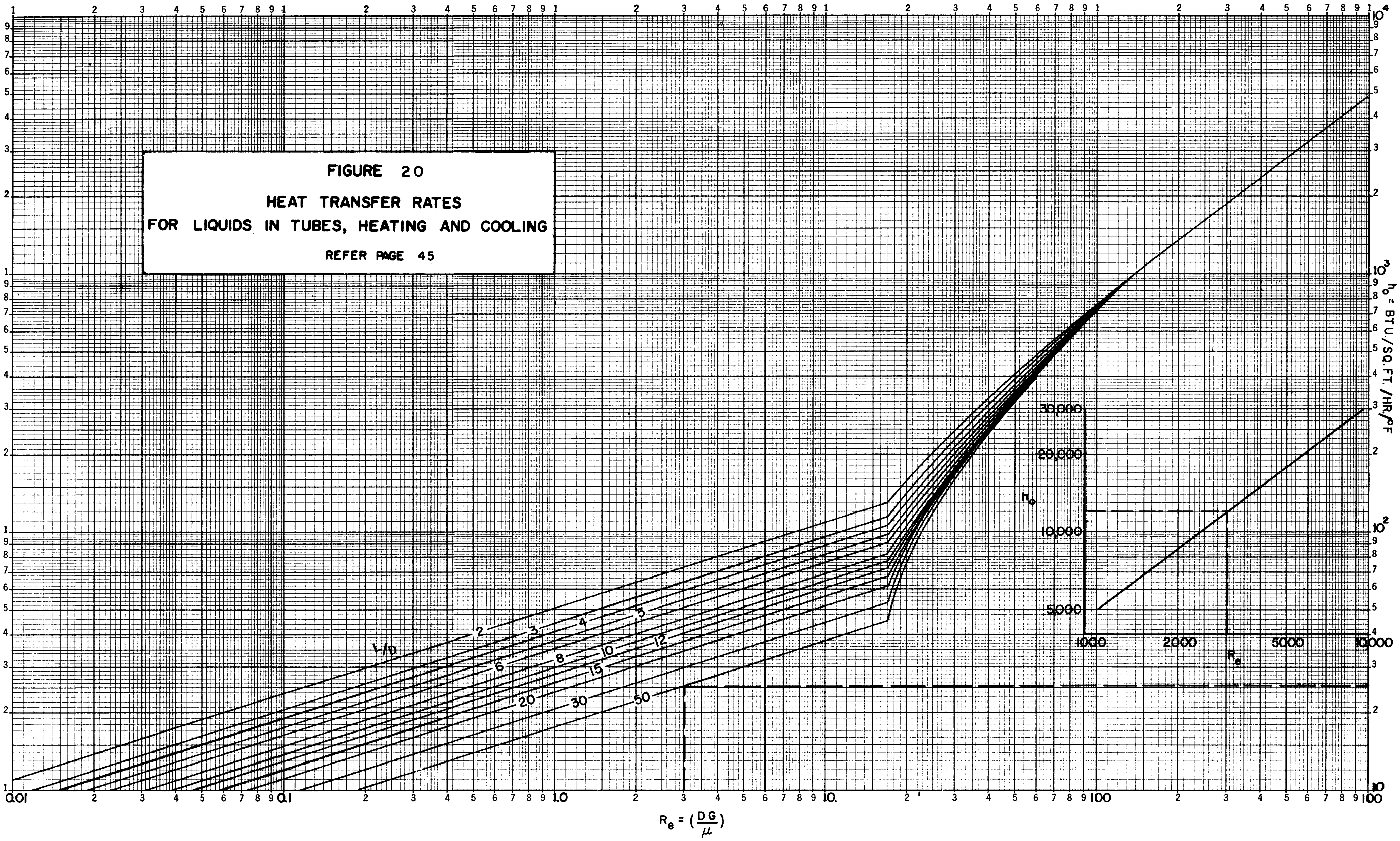


FIGURE 20
HEAT TRANSFER RATES
FOR LIQUIDS IN TUBES, HEATING AND COOLING
REFER PAGE 45
h_o = BTU./SQ.FT./HR./°F
$R_e = \left(\frac{DG}{\mu}\right)$
L/D
2
3
4
5
6
8
10
12
15
20
30
50
h_o
30,000
20,000
10,000
5,000
1000
2000
5000
10,000
R_e

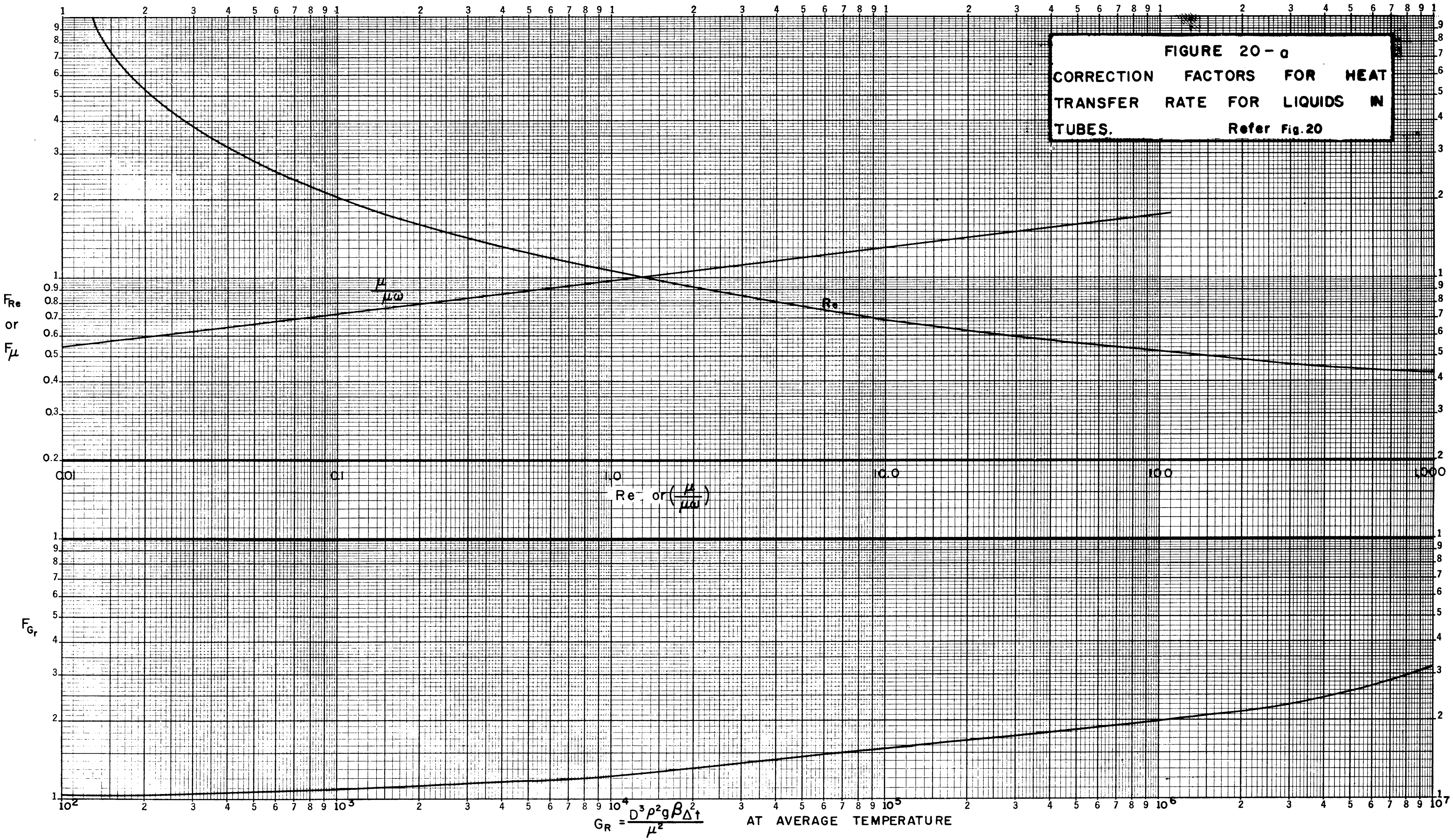


FIGURE 20 - a
CORRECTION FACTORS FOR HEAT TRANSFER RATE FOR LIQUIDS IN TUBES.
Refer Fig.20
F_{Re} or F_{μ}
$\frac{\mu}{\mu\omega}$
Re
Re or $\left(\frac{\mu}{\mu\omega}\right)$
F_{G_r}
$G_R = \frac{D^3 \rho^2 g \beta \Delta t}{\mu^2}$ AT AVERAGE TEMPERATURE

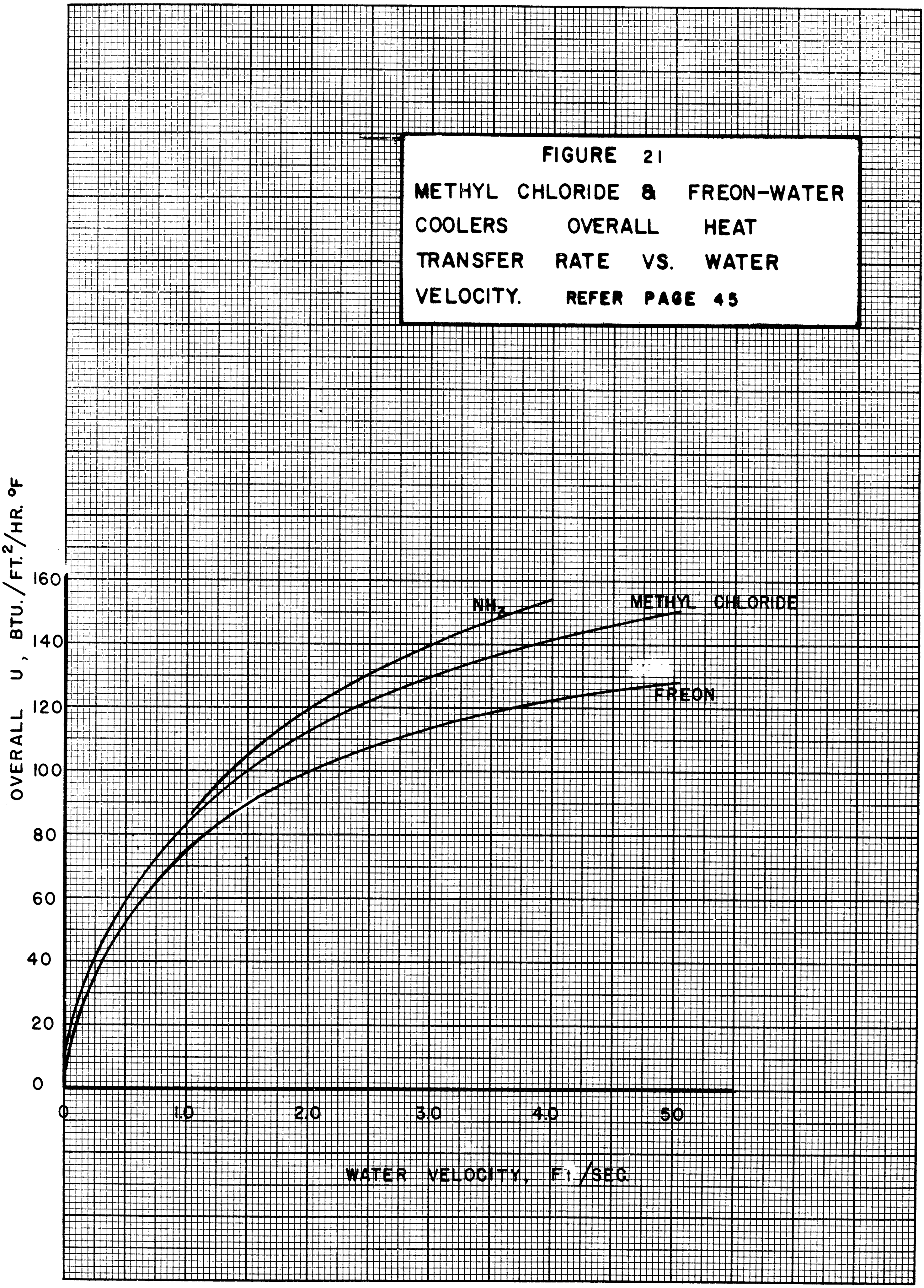


FIGURE 21
METHYL CHLORIDE & FREON-WATER COOLERS OVERALL HEAT TRANSFER RATE VS. WATER VELOCITY. REFER PAGE 45
NH_3
METHYL CHLORIDE
FREON
OVERALL U, BTU./FT.^{2}/HR. °F
160
140
120
100
80
60
40
20
0
0
1.0
2.0
3.0
4.0
5.0
WATER VELOCITY, FT./SEC.

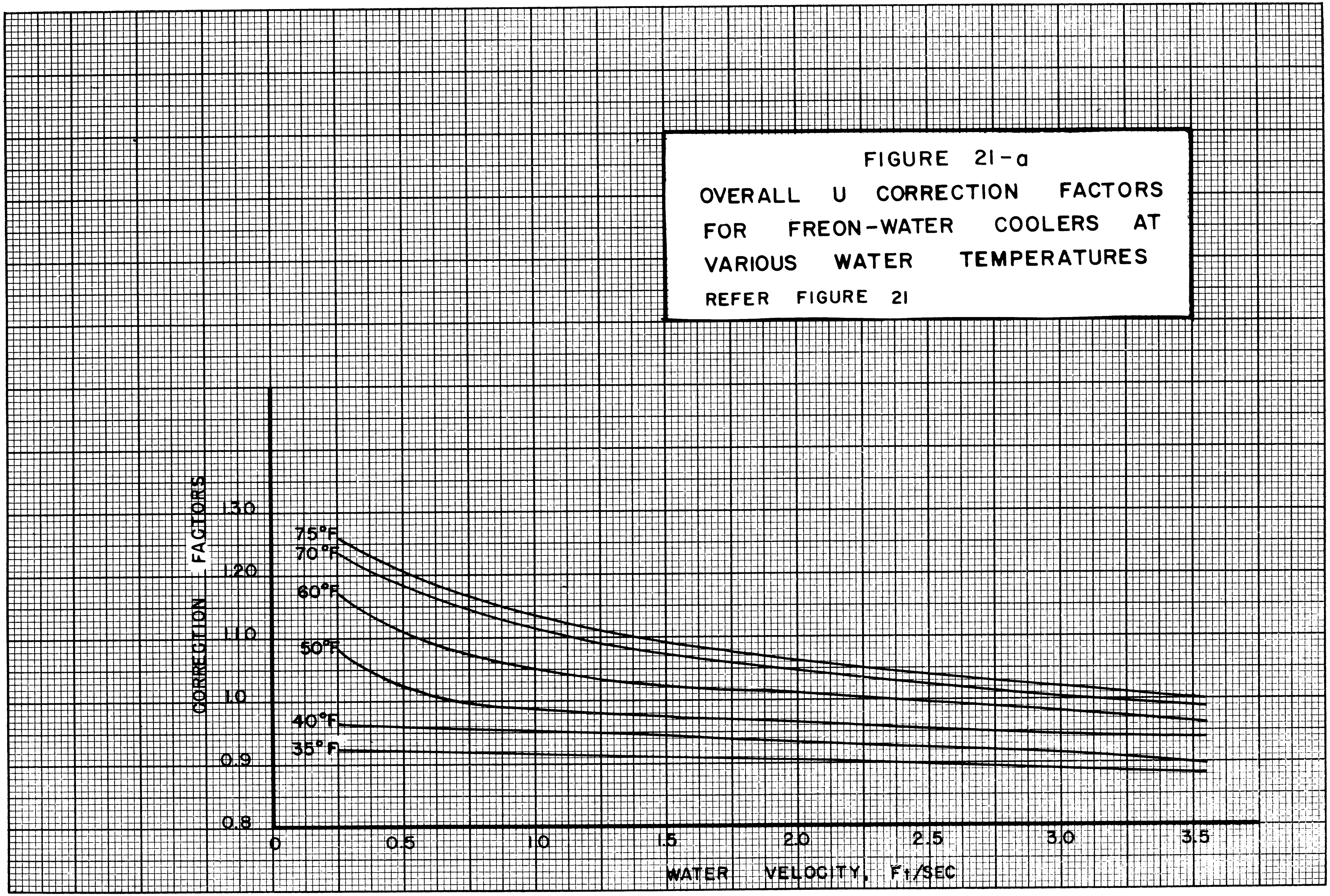
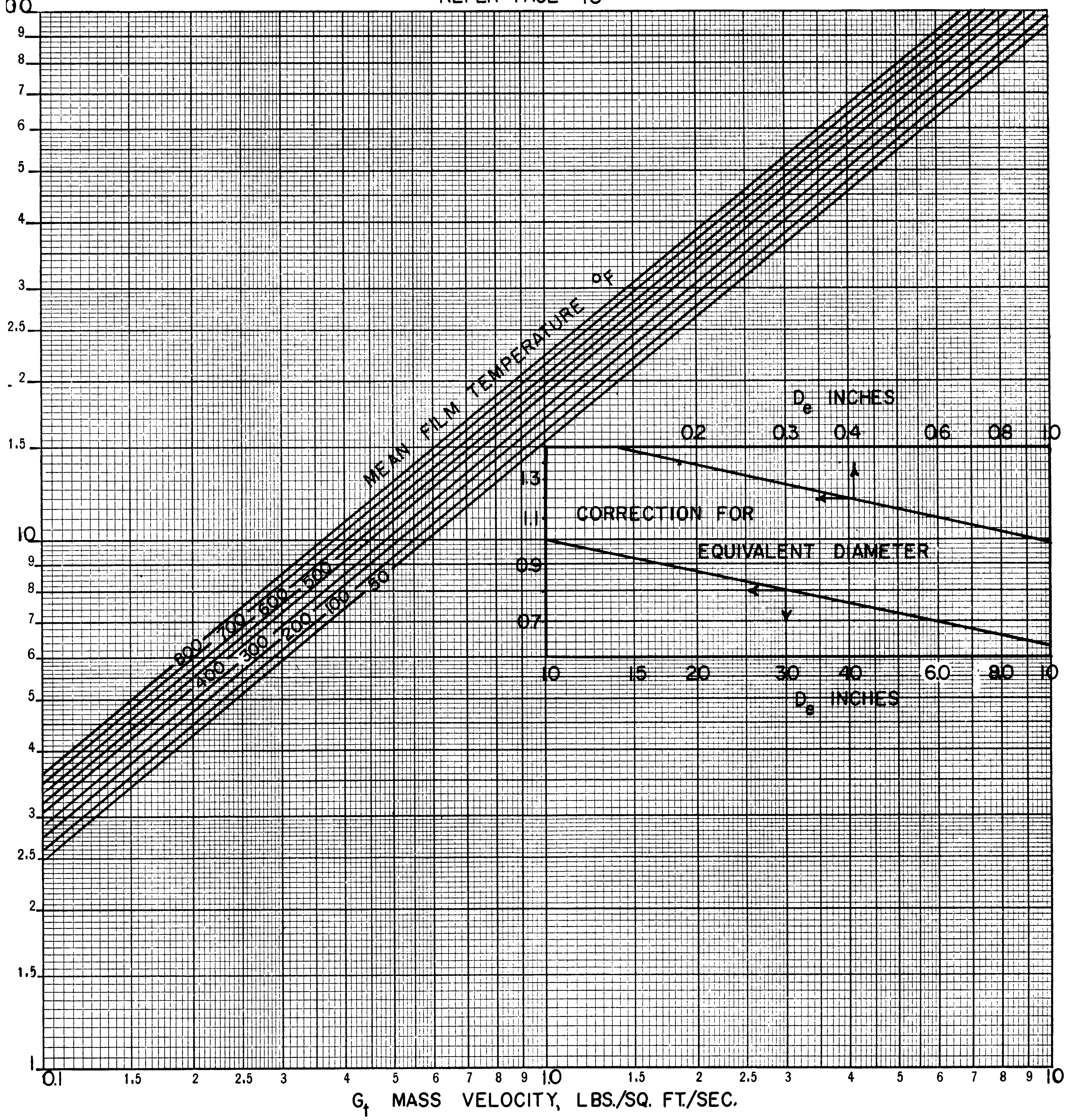


FIGURE 21-a
OVERALL U CORRECTION FACTORS FOR FREON-WATER COOLERS AT VARIOUS WATER TEMPERATURES
REFER FIGURE 21
CORRECTION FACTORS
1.30
1.20
1.10
1.0
0.9
0.8
75°F
70°F
60°F
50°F
40°F
35°F
0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
WATER VELOCITY, Ft/SEC

FIGURE 22

HEAT TRANSFER COEFFICIENTS FOR GAS INSIDE PIPES OR ANNULAR SPACES

REFER PAGE 45



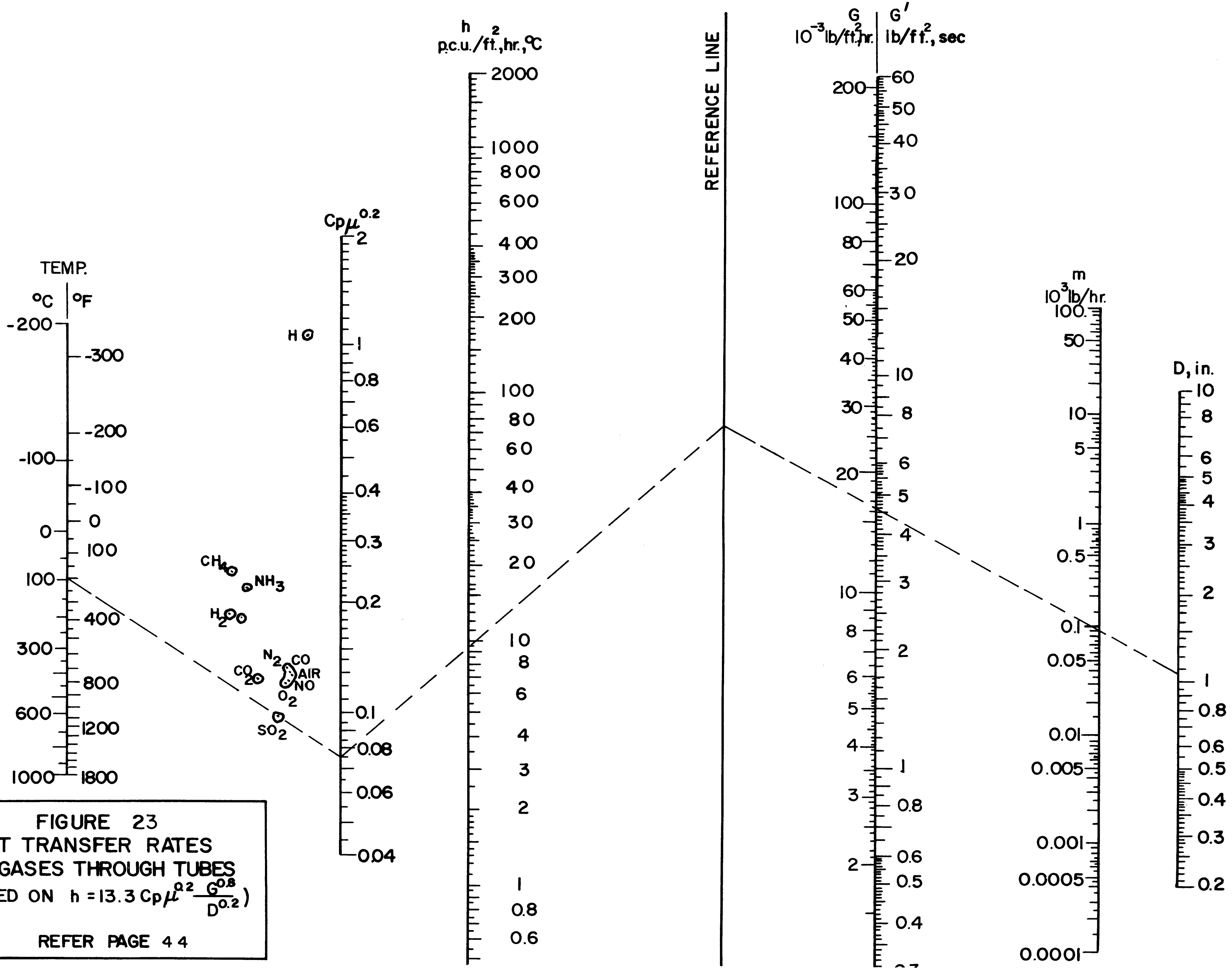


FIGURE 23
HEAT TRANSFER RATES
FOR GASES THROUGH TUBES
(BASED ON $h = 13.3\, Cp\mu^{0.2} \frac{G^{0.8}}{D^{0.2}}$)

REFER PAGE 44

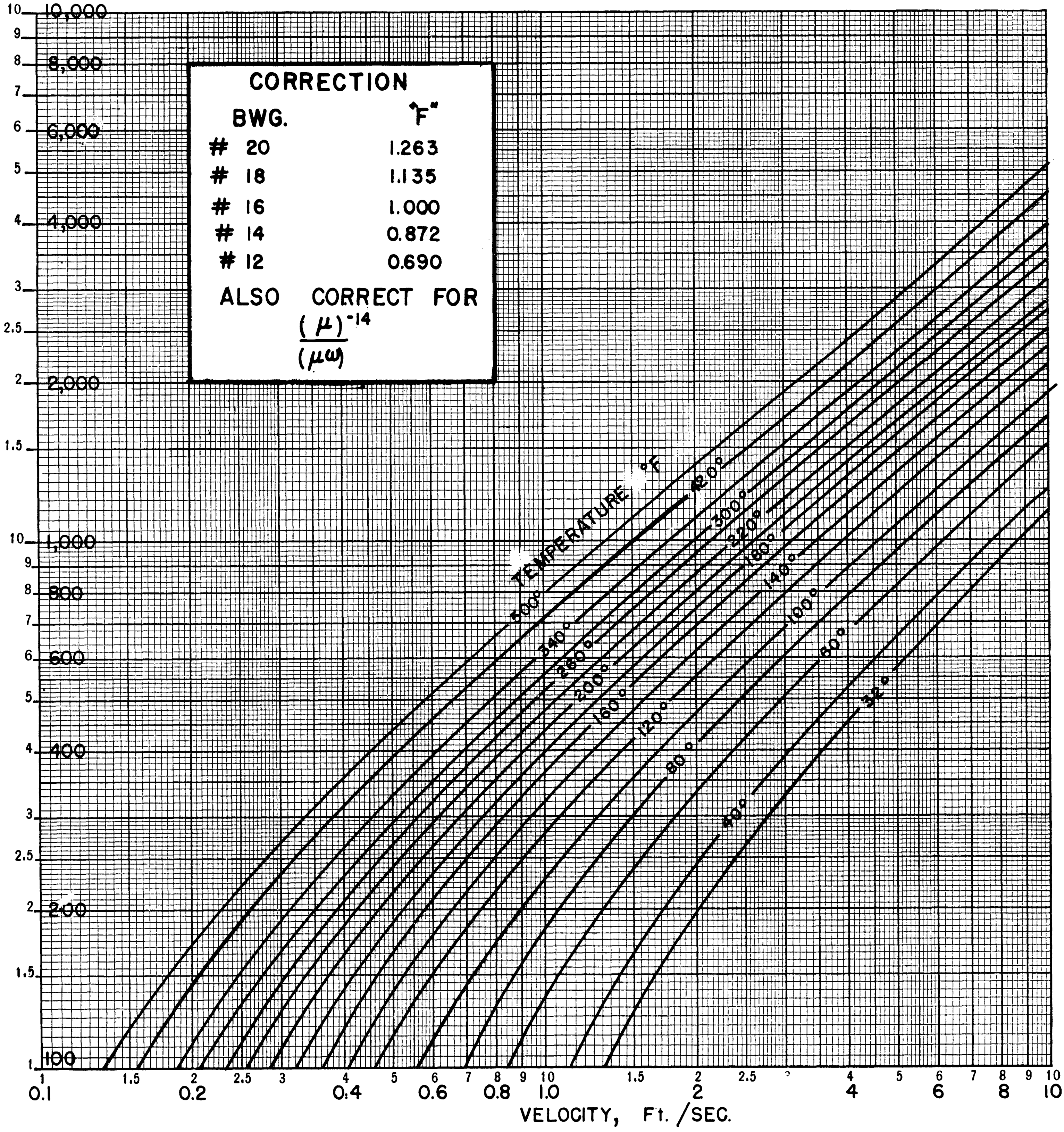


FIGURE 24
HEAT TRANSFER RATES FOR WATER
IN $\frac{5''}{8}$ DIAM., 16 BWG. TUBES.

REFER PAGE 45

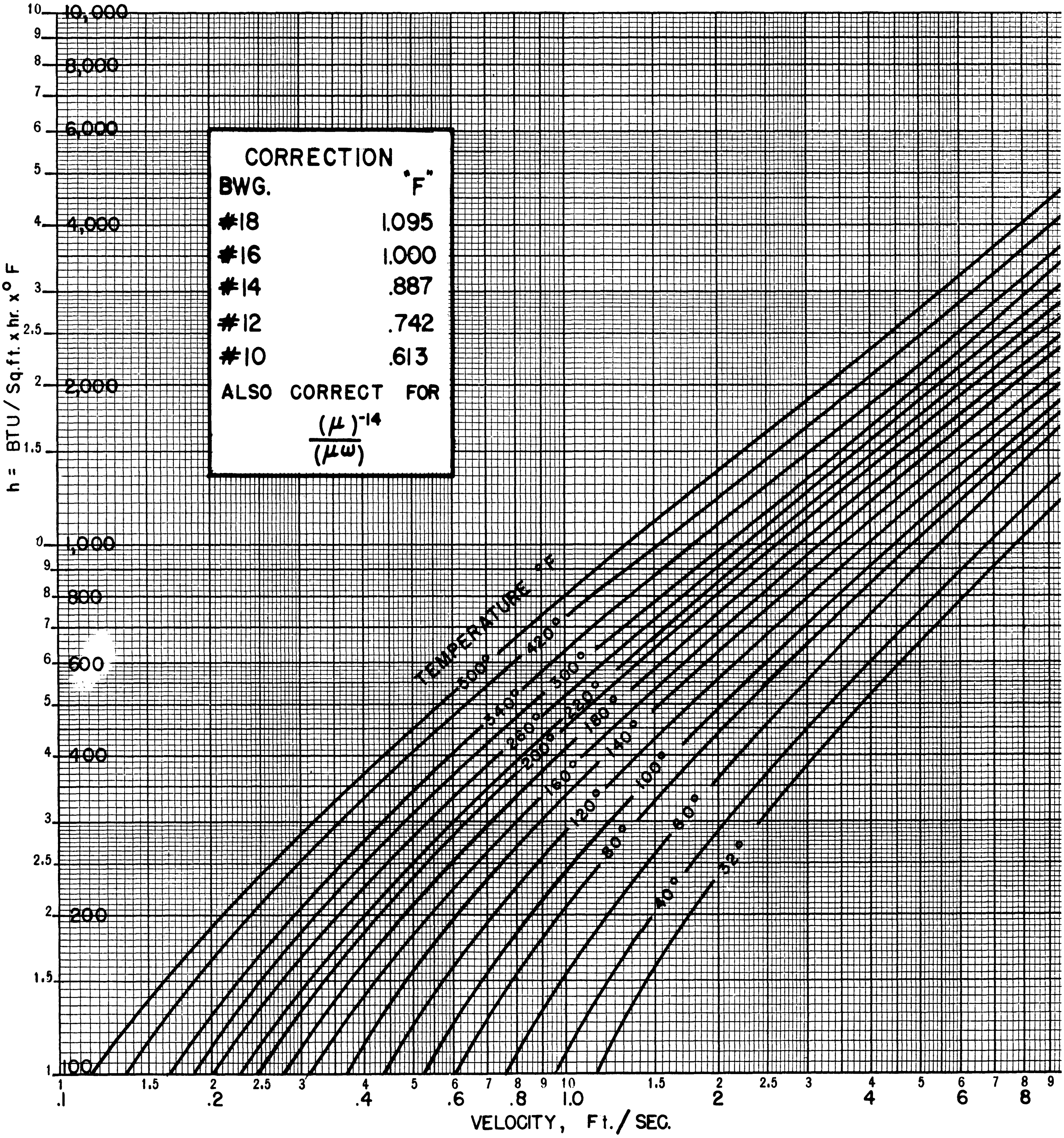


FIGURE 25
HEAT TRANSFER RATES FOR WATER
IN $\frac{3''}{4}$ DIAM., 16 BWG. TUBES.

REFER PAGE 45

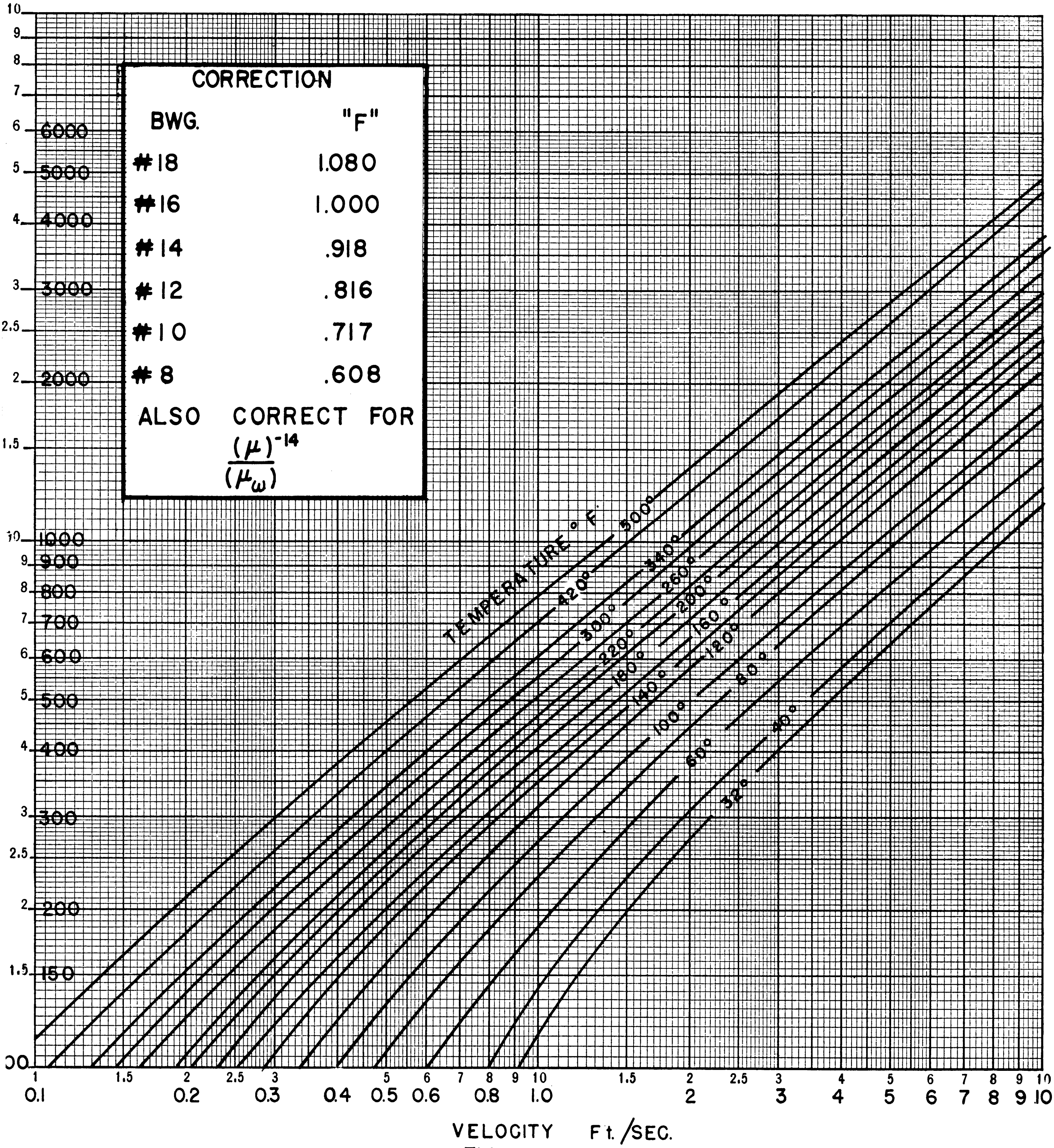


FIGURE 26
HEAT TRANSFER RATES FOR WATER
IN 1" DIAM., 16 BWG. TUBES.

REFER PAGE 45

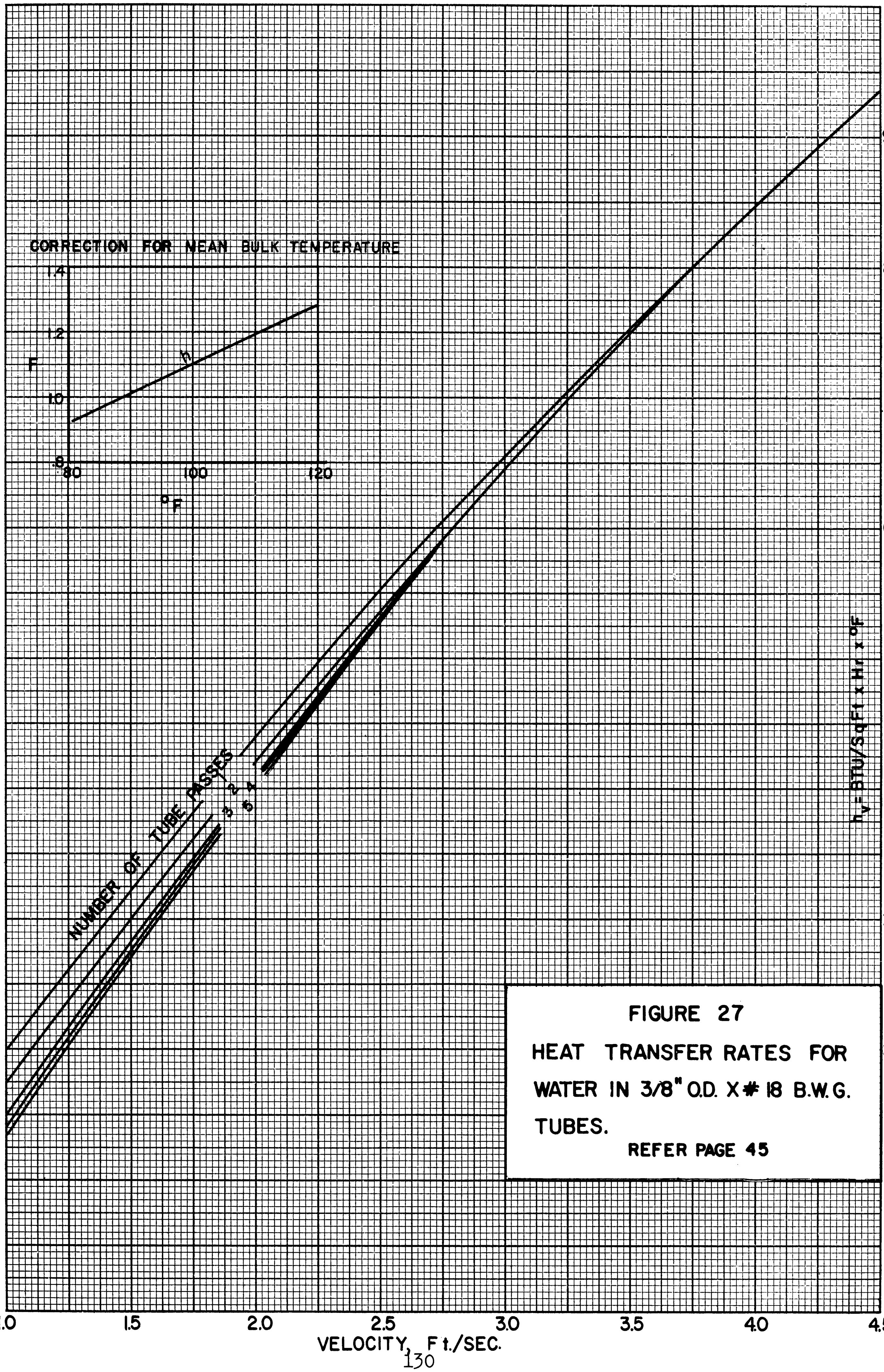
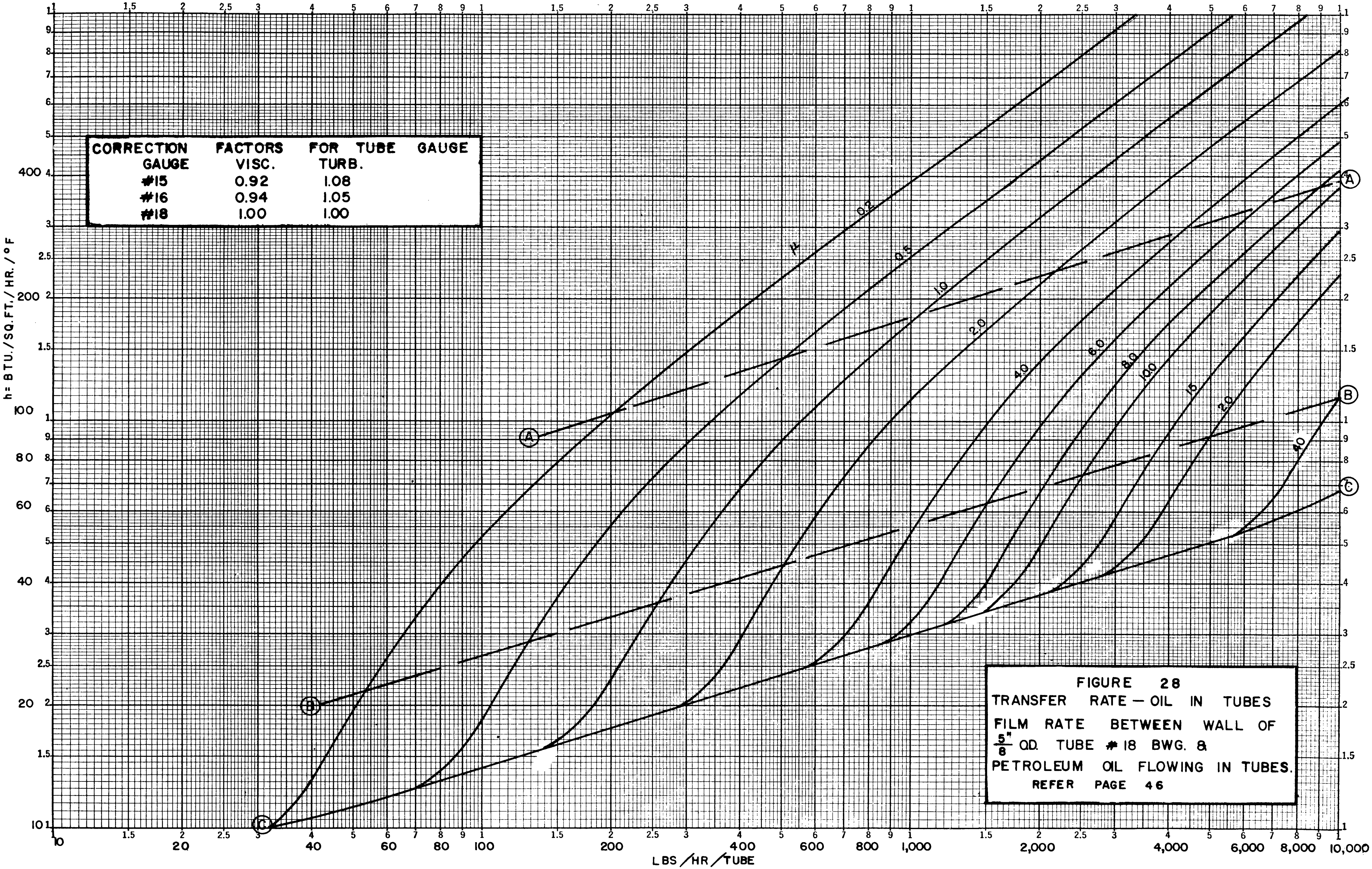


FIGURE 27

HEAT TRANSFER RATES FOR WATER IN 3/8" O.D. X # 18 B.W.G. TUBES.

REFER PAGE 45



CORRECTION GAUGE	FACTORS VISC.	FOR TUBE TURB.	GAUGE
#15	0.92	1.08	
#16	0.94	1.05	
#18	1.00	1.00	

FIGURE 28
TRANSFER RATE — OIL IN TUBES
FILM RATE BETWEEN WALL OF $\frac{5}{8}$" O.D. TUBE # 18 BWG. & PETROLEUM OIL FLOWING IN TUBES.

REFER PAGE 46

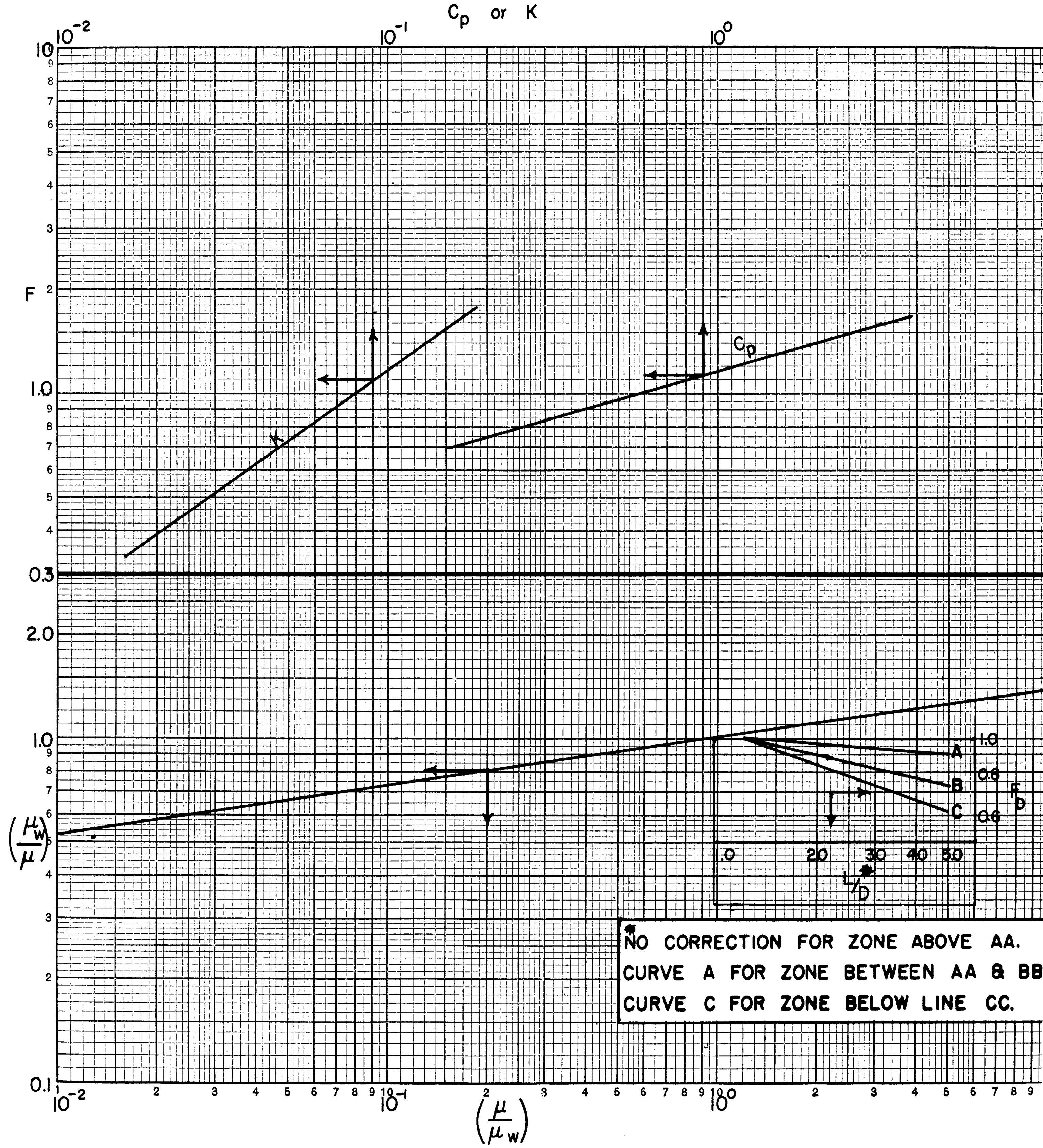
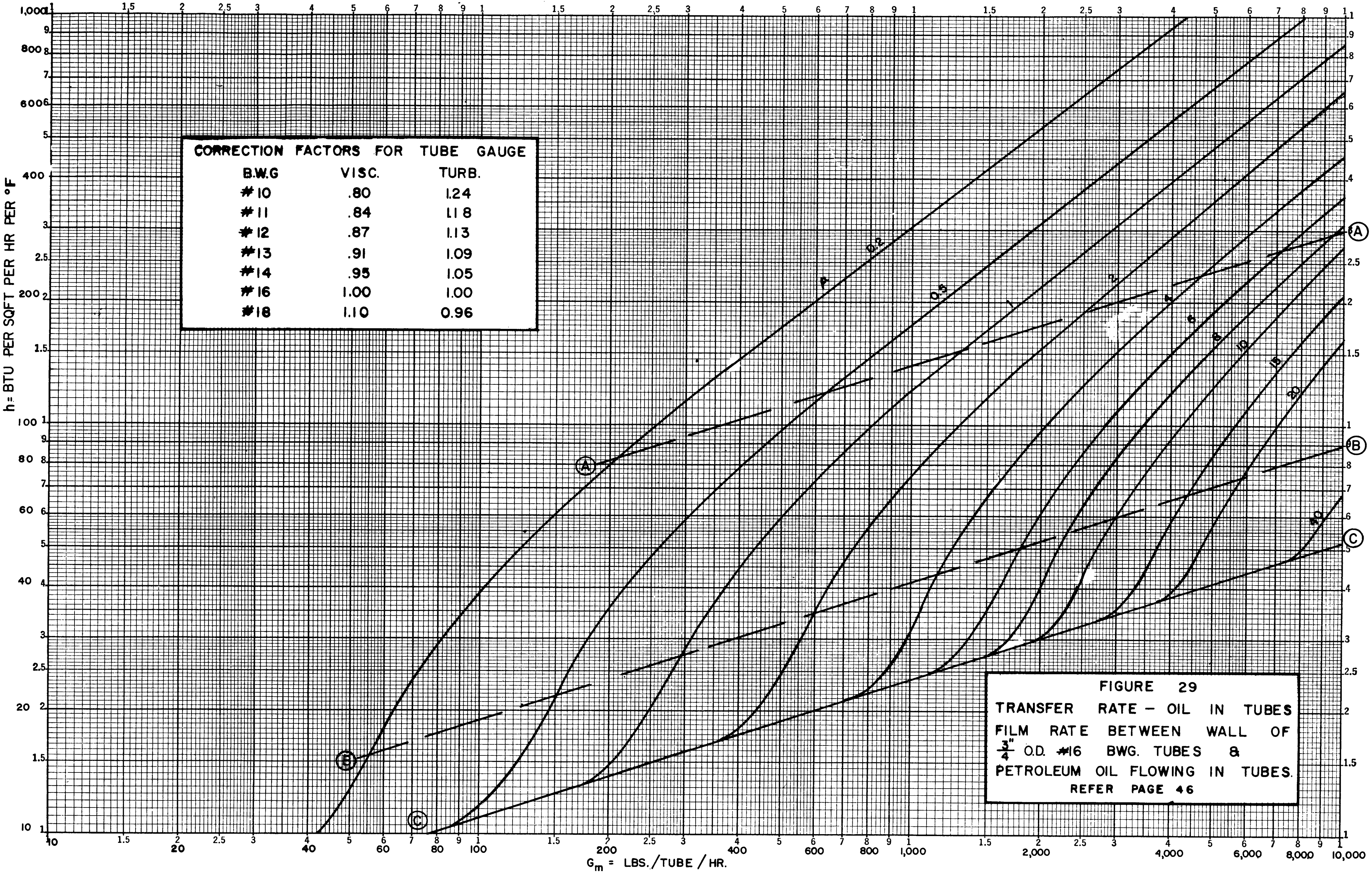


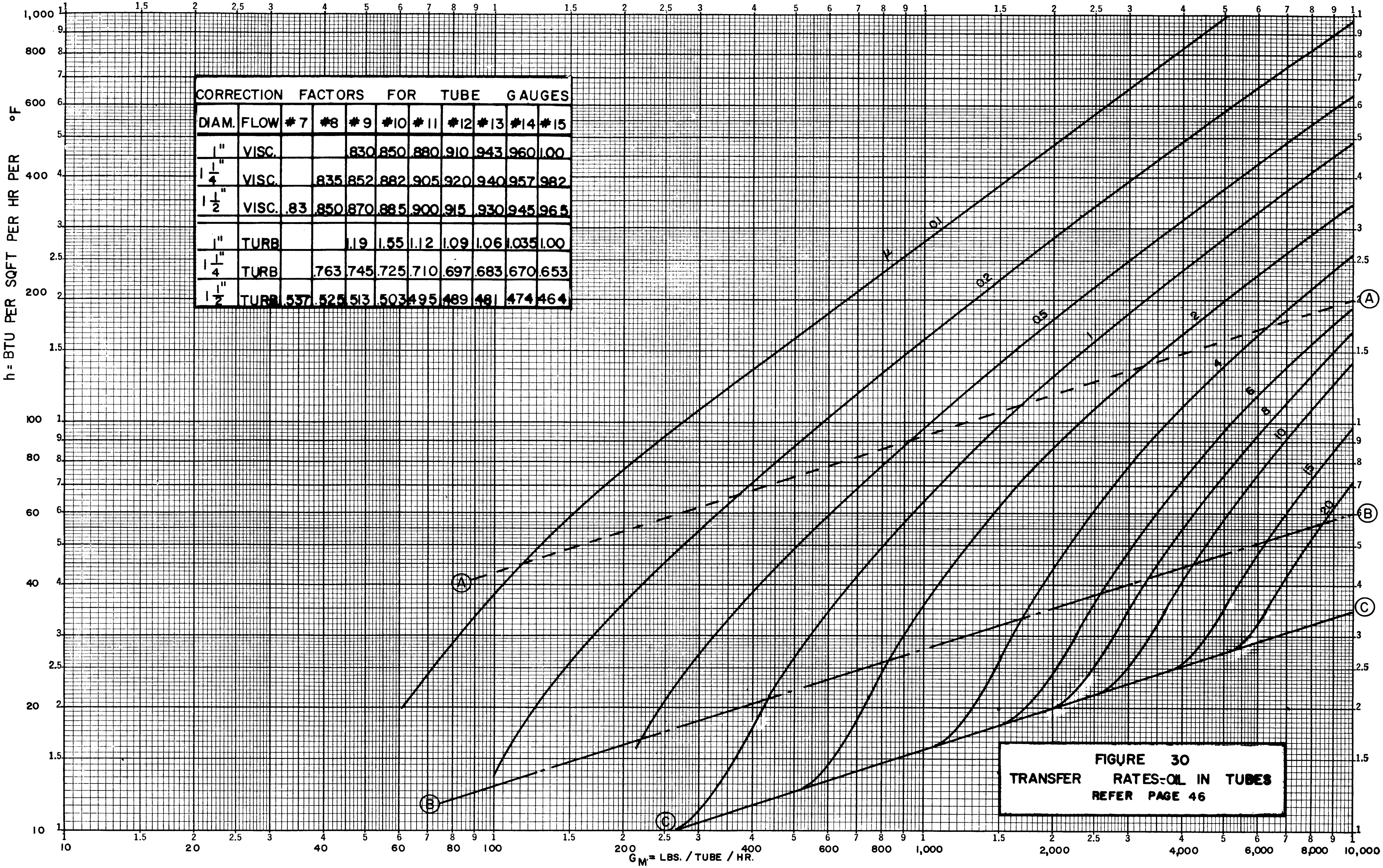
FIGURE 28—a

CORRECTION FACTORS FOR HEAT TRANSFER RATES FOR OILS IN $\frac{5}{8}$" DIAMETER 18 BWG TUBES

REFER FIG. 28



B.W.G	VISC.	TURB.
#10	.80	1.24
#11	.84	1.18
#12	.87	1.13
#13	.91	1.09
#14	.95	1.05
#16	1.00	1.00
#18	1.10	0.96



CORRECTION FACTORS FOR TUBE GAUGES										
DIAM.	FLOW	#7	#8	#9	#10	#11	#12	#13	#14	#15
1"	VISC.			830	850	880	.910	.943	.960	1.00
1 1/4"	VISC.		.835	852	882	.905	.920	.940	.957	.982
1 1/2"	VISC.	.83	.850	.870	.885	.900	.915	.930	.945	.965
1"	TURB			1.19	1.55	1.12	1.09	1.06	1.035	1.00
1 1/4"	TURB		.763	.745	.725	.710	.697	.683	.670	.653
1 1/2"	TURB	.537	.525	.513	.503	.495	.489	.481	.474	.464

FIGURE 30
TRANSFER RATES-OIL IN TUBES
REFER PAGE 46

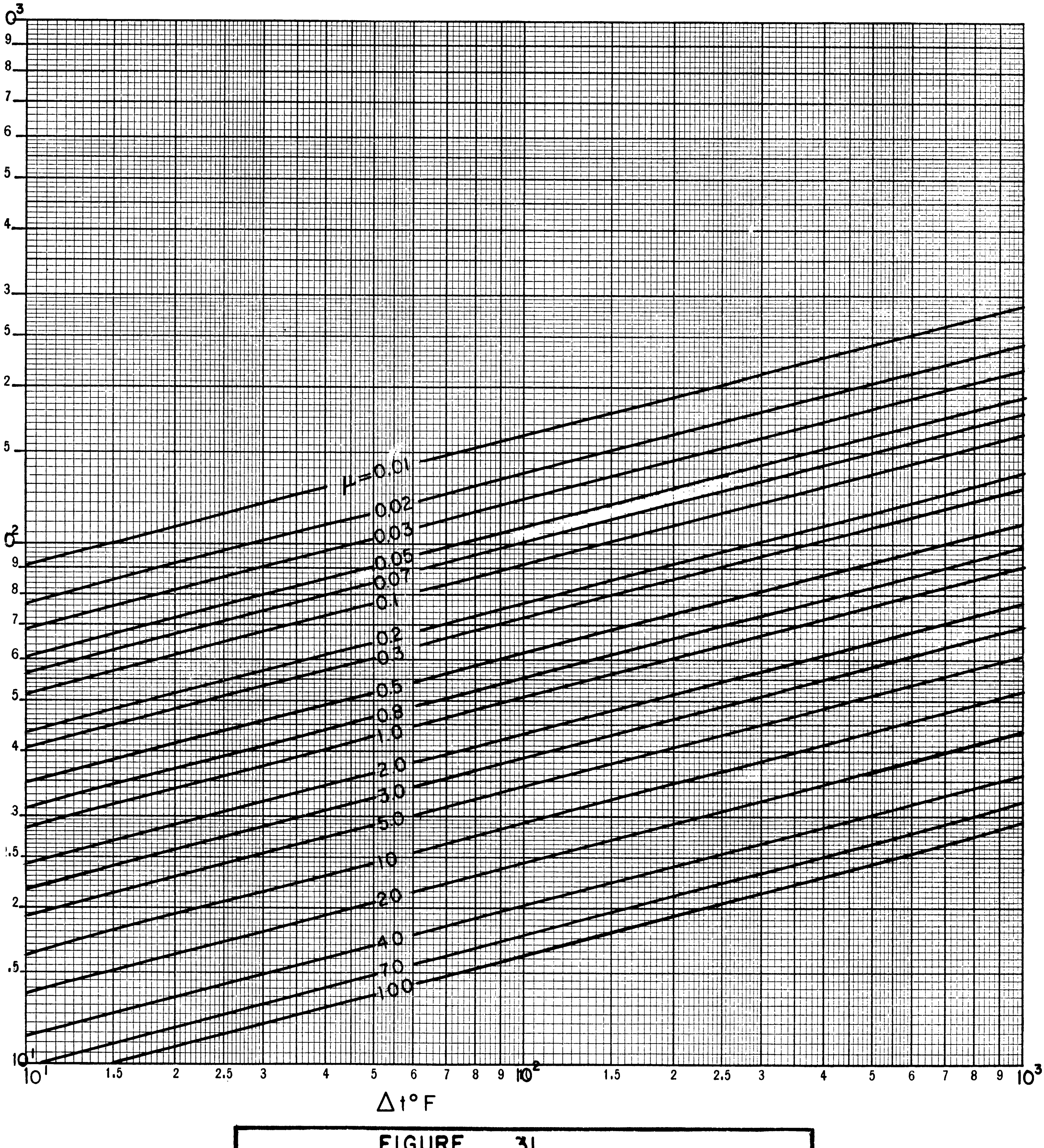


FIGURE 31

HEAT TRANSFER RATES - FREE CONVECTION

(BASED ON RICE'S EQUATION)

REFER PAGE 44

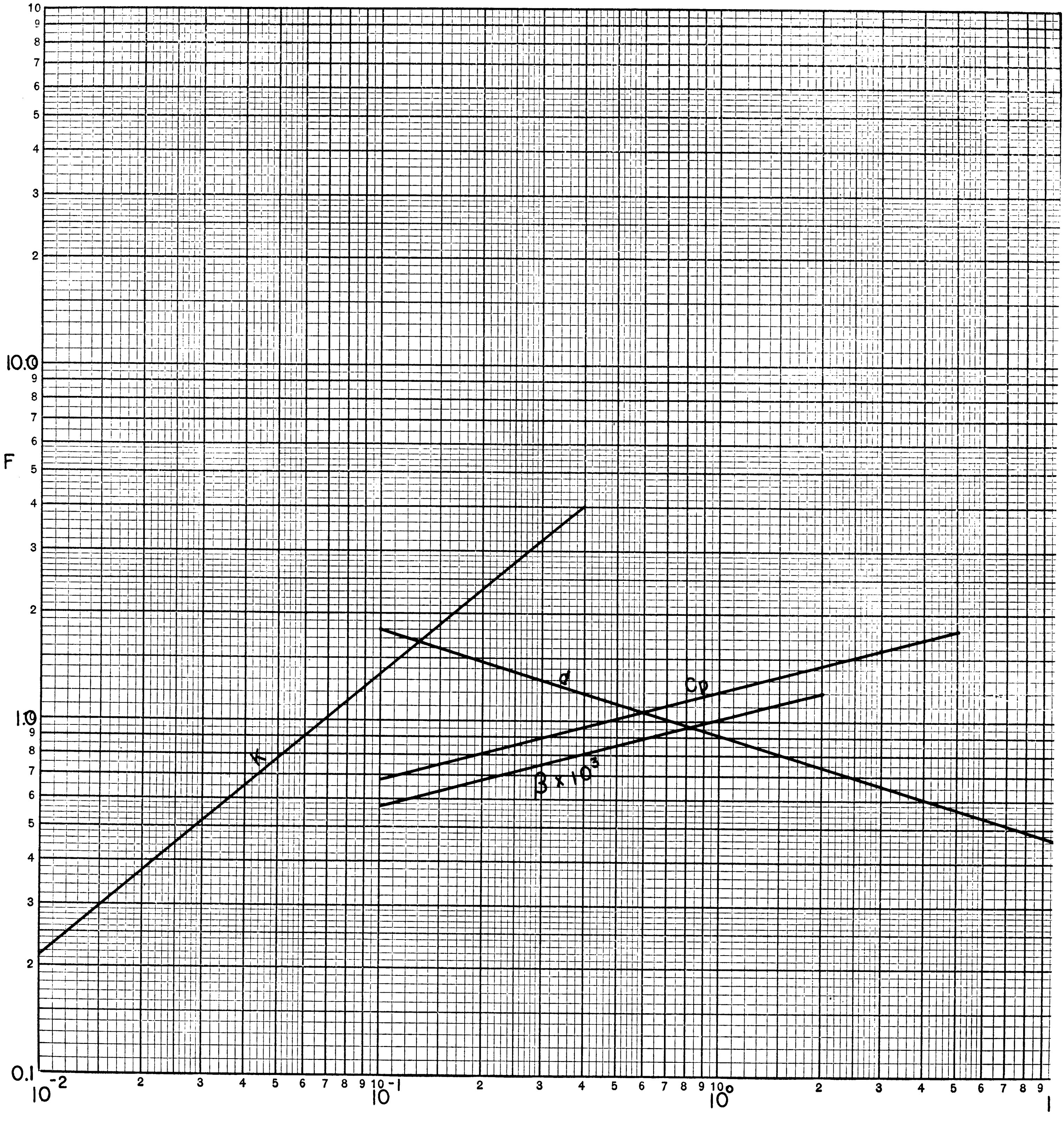


FIGURE 31-a

CORRECTION FOR FREE CONVECTION HEAT TRANSFER RATES

REFER FIG. 31

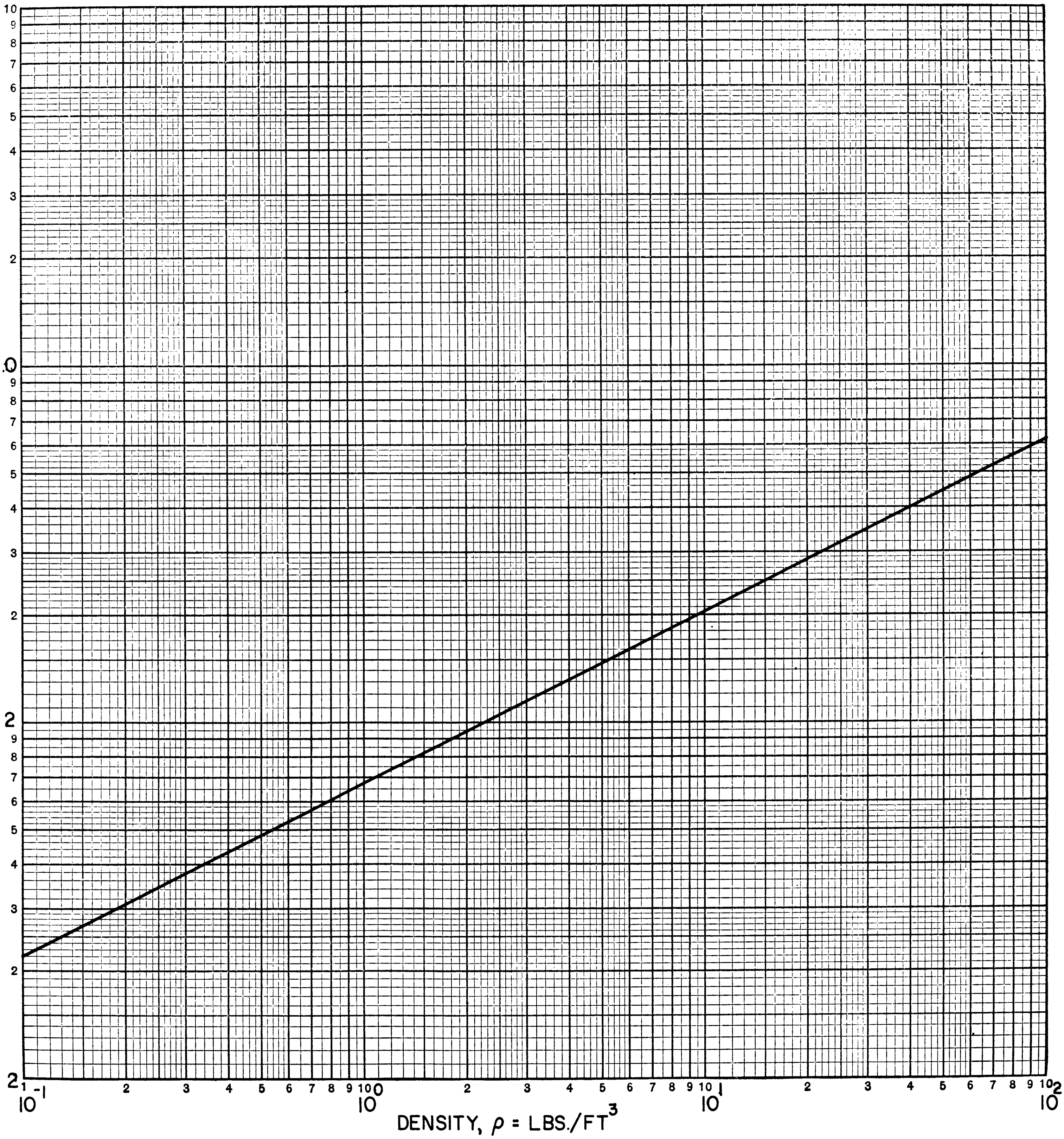


FIGURE 31 - b

CORRECTION FOR FREE CONVECTION HEAT TRANSFER RATES

REFER FIG. 31

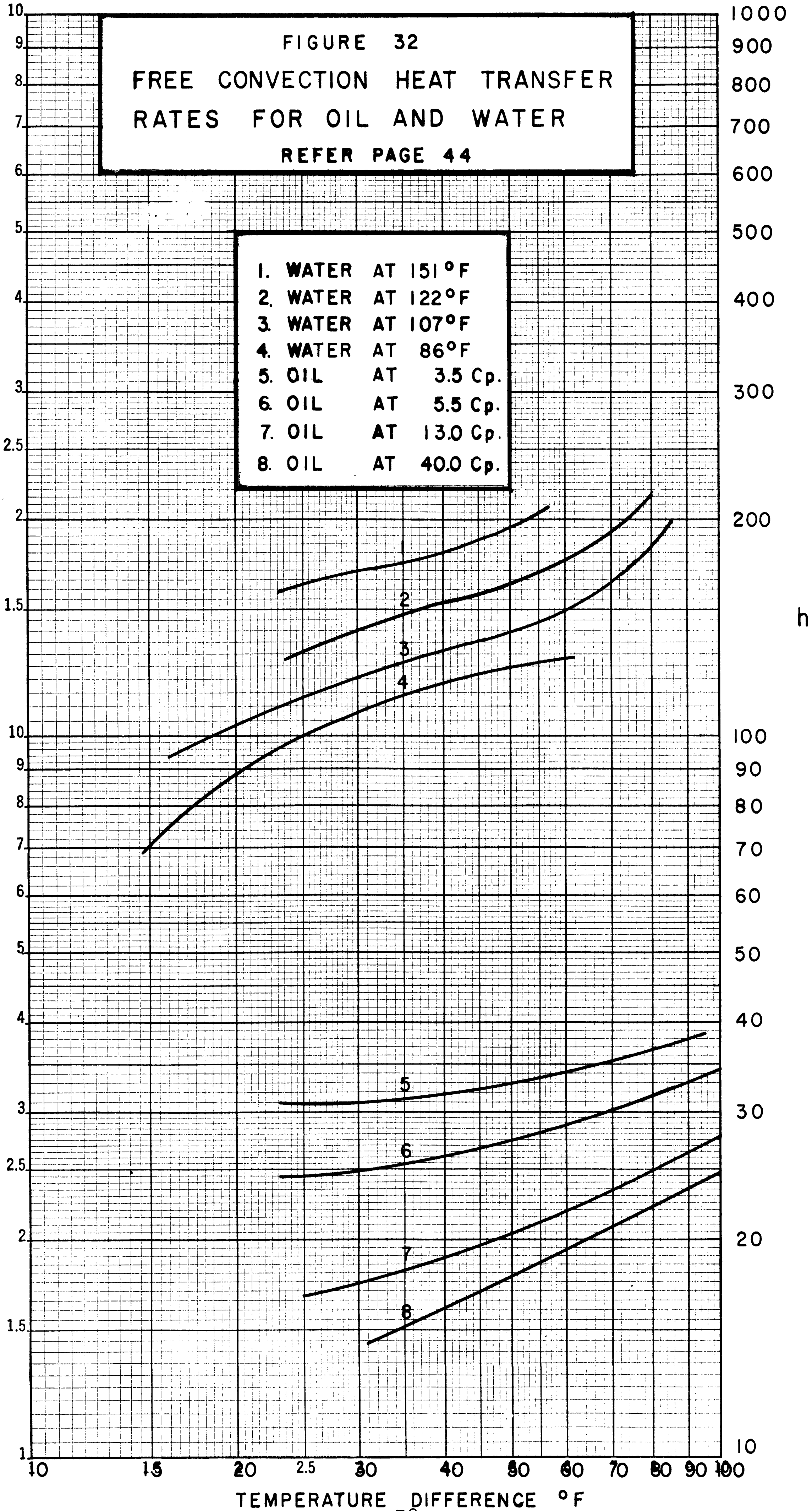


FIGURE 32
FREE CONVECTION HEAT TRANSFER RATES FOR OIL AND WATER
REFER PAGE 44

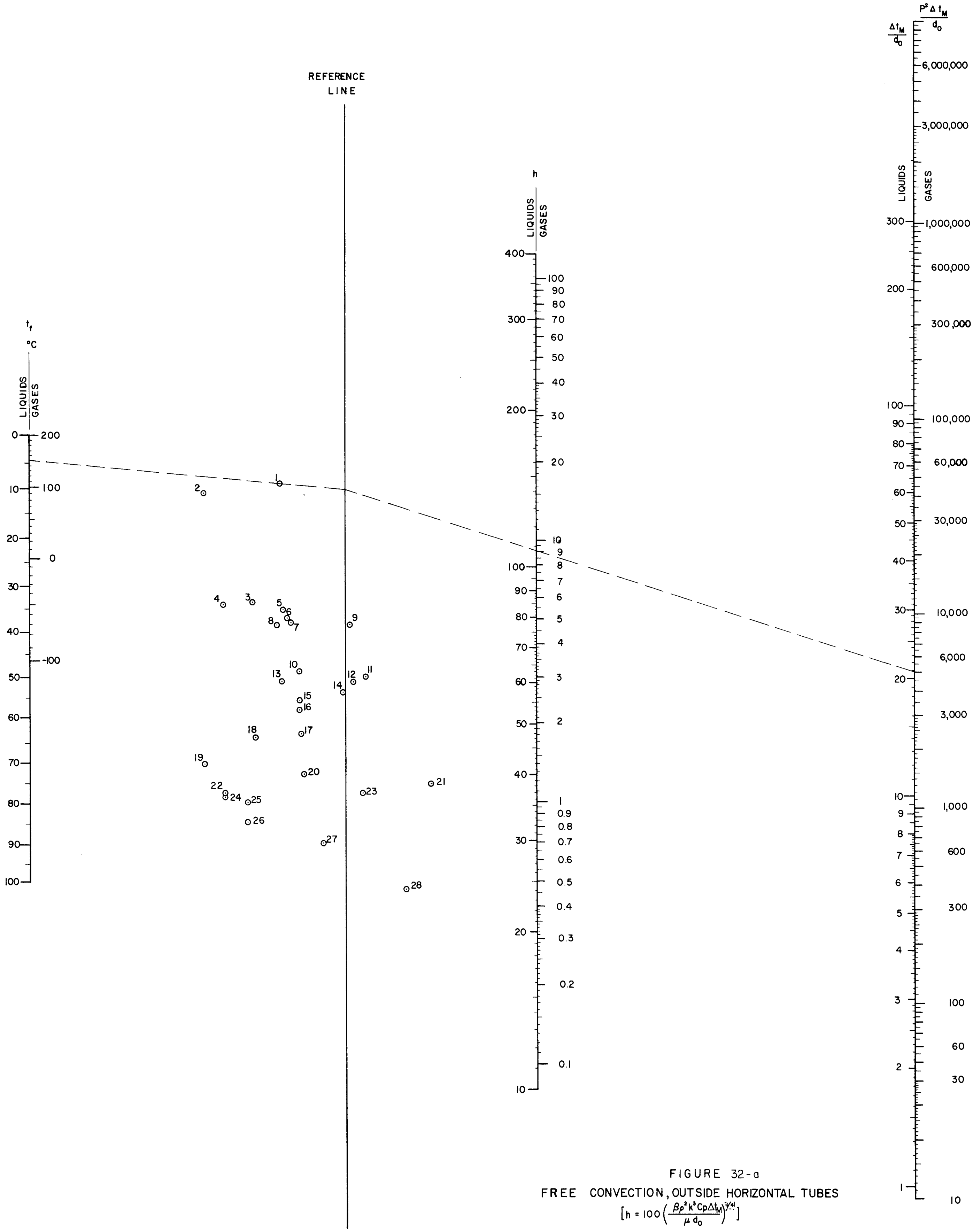


FIGURE 32-a

FREE CONVECTION, OUTSIDE HORIZONTAL TUBES

$$\left[h = 100\left(\frac{\beta\rho^2 k^3 Cp\Delta t_M}{\mu\, d_o}\right)^{1/4}\right]$$

REFER PAGE 47

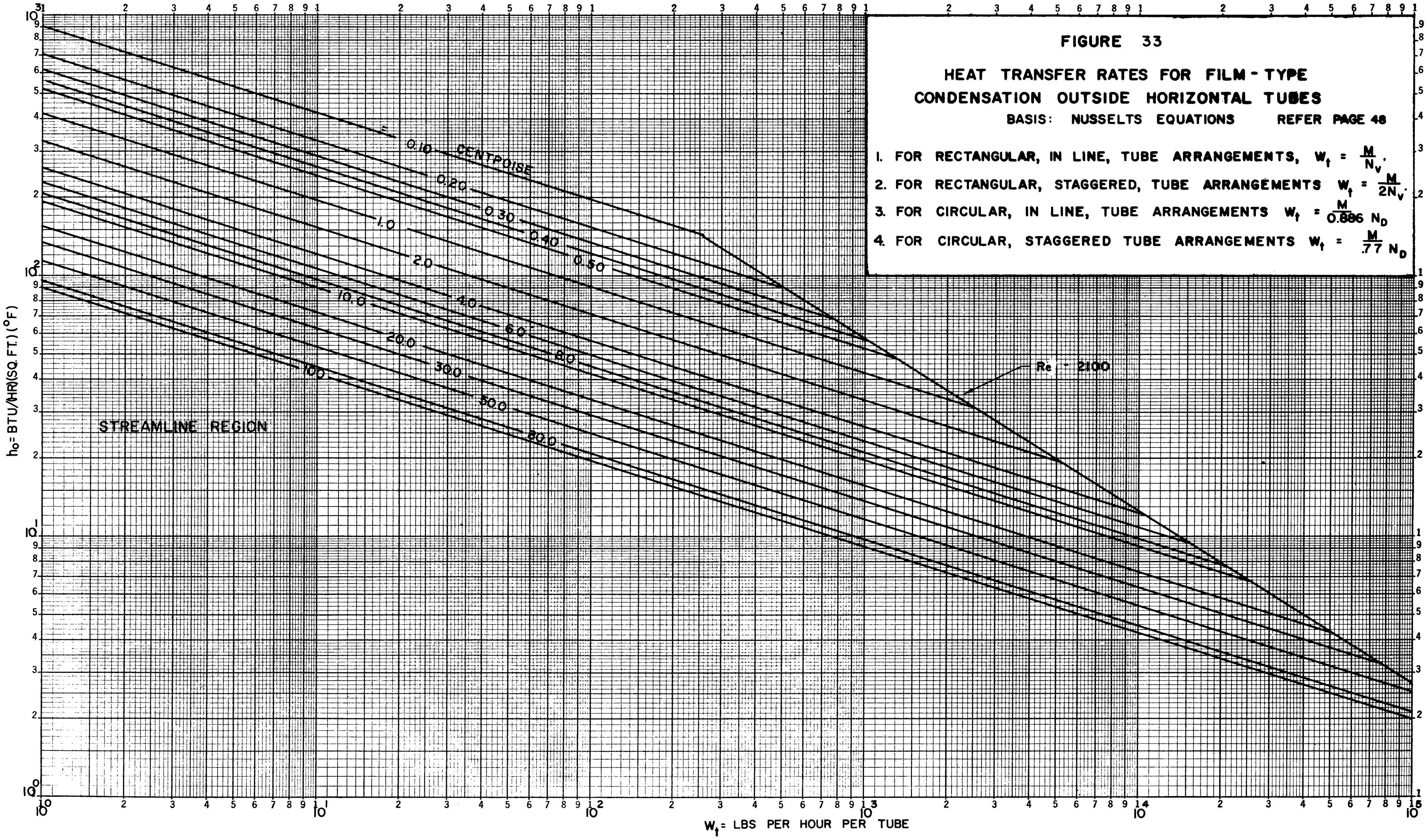


FIGURE 33
HEAT TRANSFER RATES FOR FILM-TYPE CONDENSATION OUTSIDE HORIZONTAL TUBES
BASIS: NUSSELTS EQUATIONS REFER PAGE 48
1. FOR RECTANGULAR, IN LINE, TUBE ARRANGEMENTS, $W_t = \frac{M}{N_V}$.
2. FOR RECTANGULAR, STAGGERED, TUBE ARRANGEMENTS $W_t = \frac{M}{2N_V}$.
3. FOR CIRCULAR, IN LINE, TUBE ARRANGEMENTS $W_t = \frac{M}{0.886\ N_D}$
4. FOR CIRCULAR, STAGGERED TUBE ARRANGEMENTS $W_t = \frac{M}{.77\ N_D}$
μ = 0.10 CENTPOISE
0.20
0.30
0.40
0.50
1.0
2.0
4.0
6.0
8.0
10.0
20.0
30.0
50.0
80.0
100
Re = 2100
STREAMLINE REGION
h_o = BTU/HR(SQ. FT.)(°F)
W_t = LBS PER HOUR PER TUBE

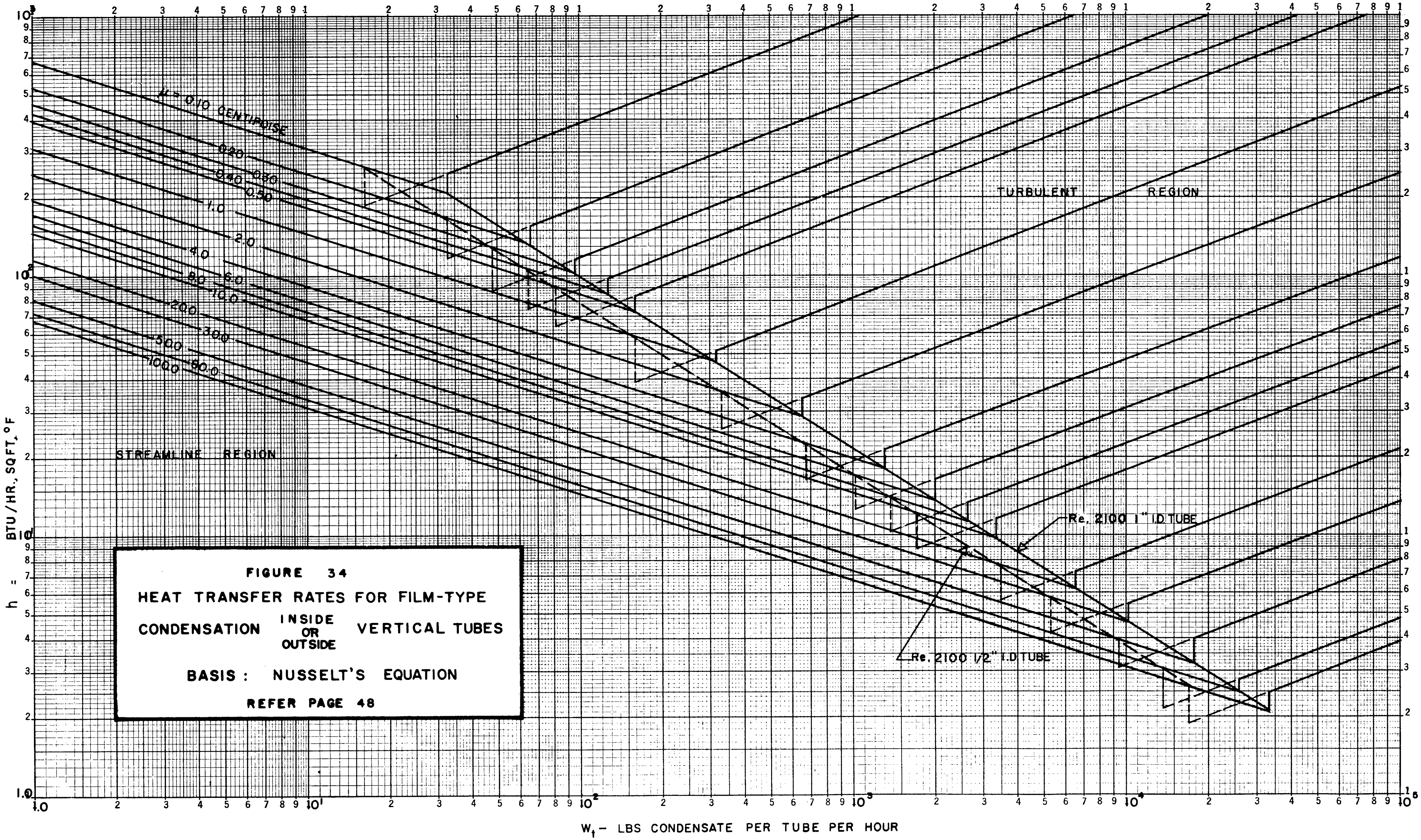


FIGURE 34

HEAT TRANSFER RATES FOR FILM-TYPE CONDENSATION INSIDE OR OUTSIDE VERTICAL TUBES

BASIS : NUSSELT'S EQUATION

REFER PAGE 48

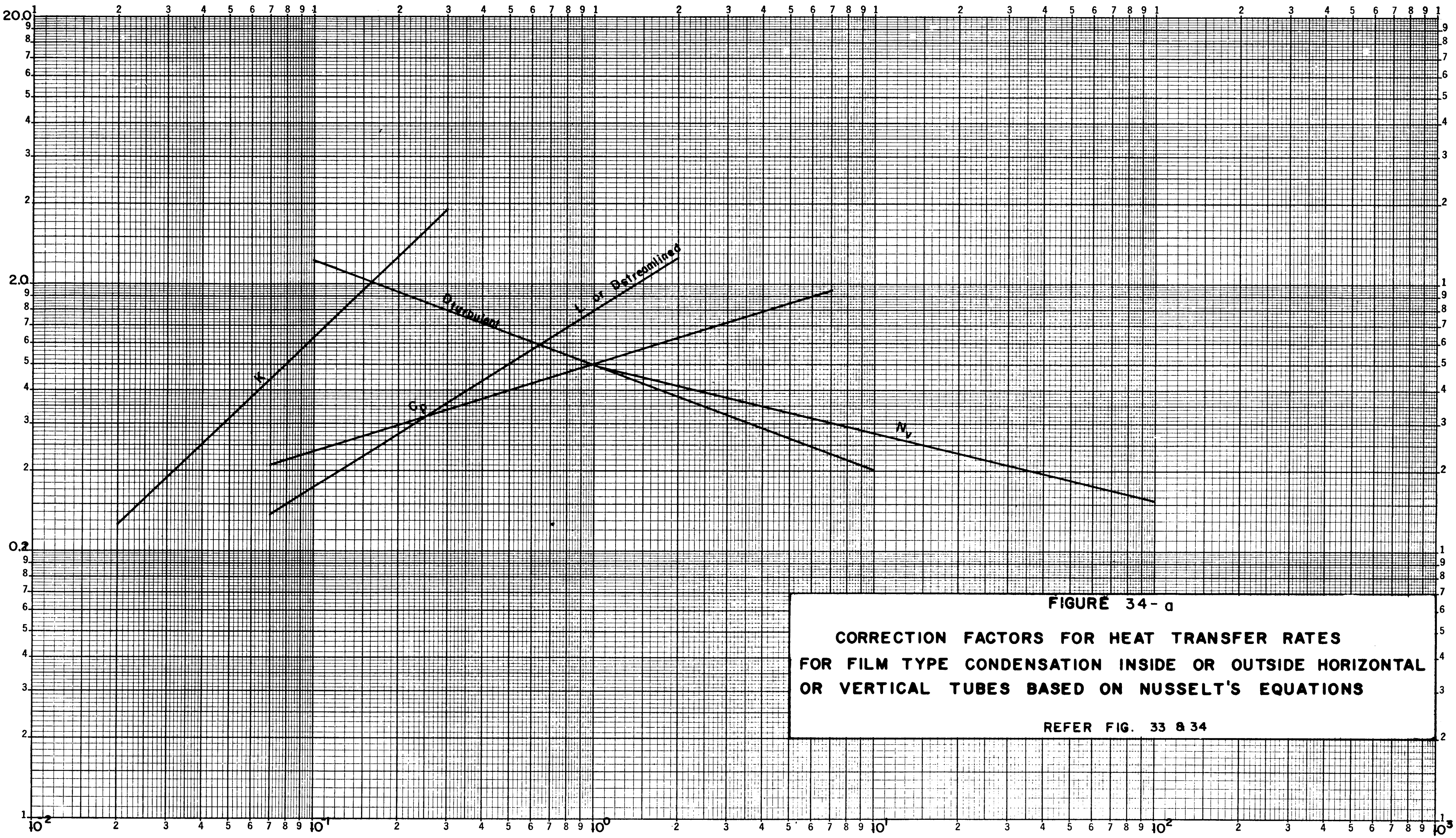


FIGURE 34- a

CORRECTION FACTORS FOR HEAT TRANSFER RATES FOR FILM TYPE CONDENSATION INSIDE OR OUTSIDE HORIZONTAL OR VERTICAL TUBES BASED ON NUSSELT'S EQUATIONS

REFER FIG. 33 & 34

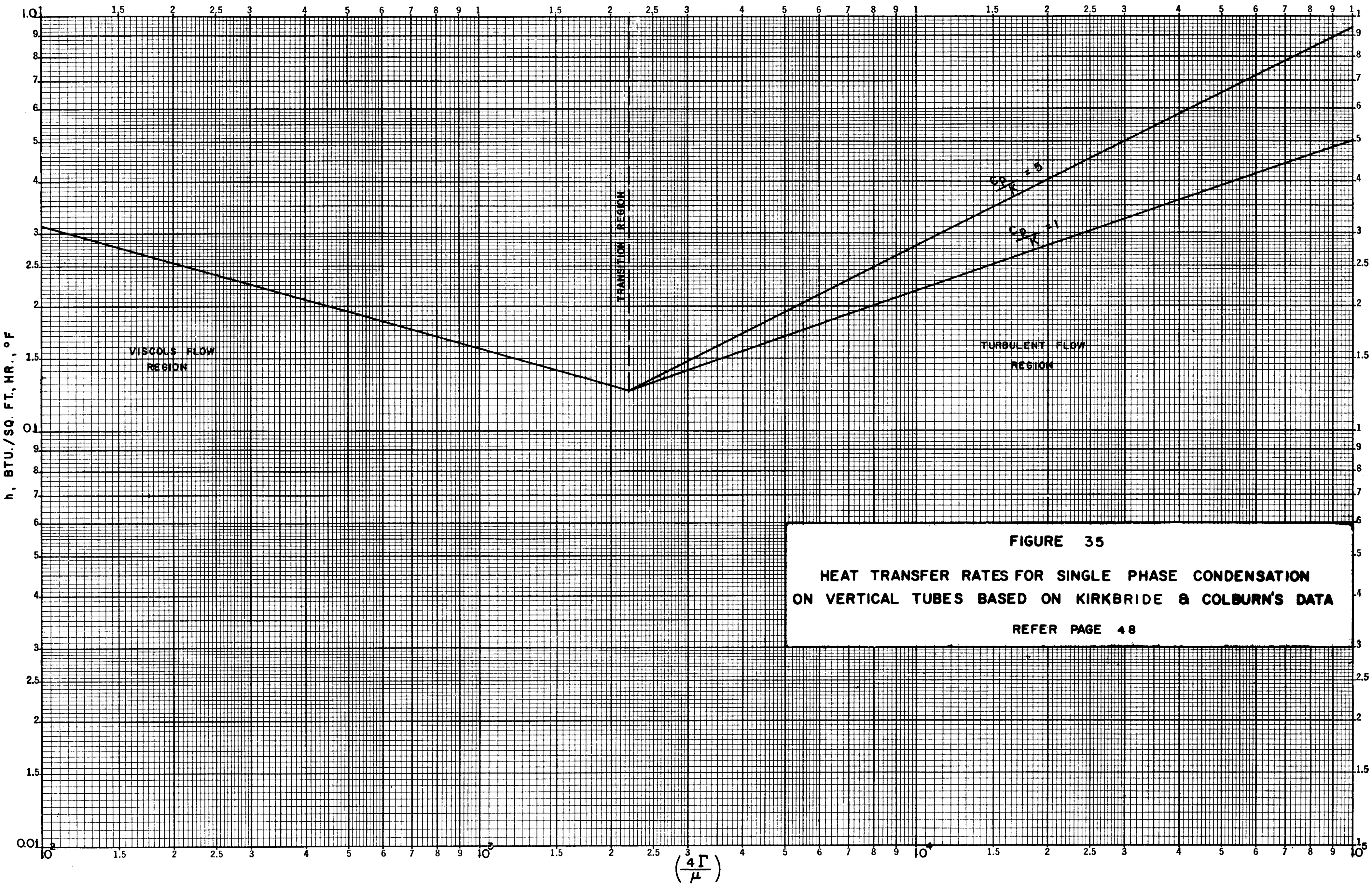


FIGURE 35

HEAT TRANSFER RATES FOR SINGLE PHASE CONDENSATION ON VERTICAL TUBES BASED ON KIRKBRIDE & COLBURN'S DATA

REFER PAGE 48

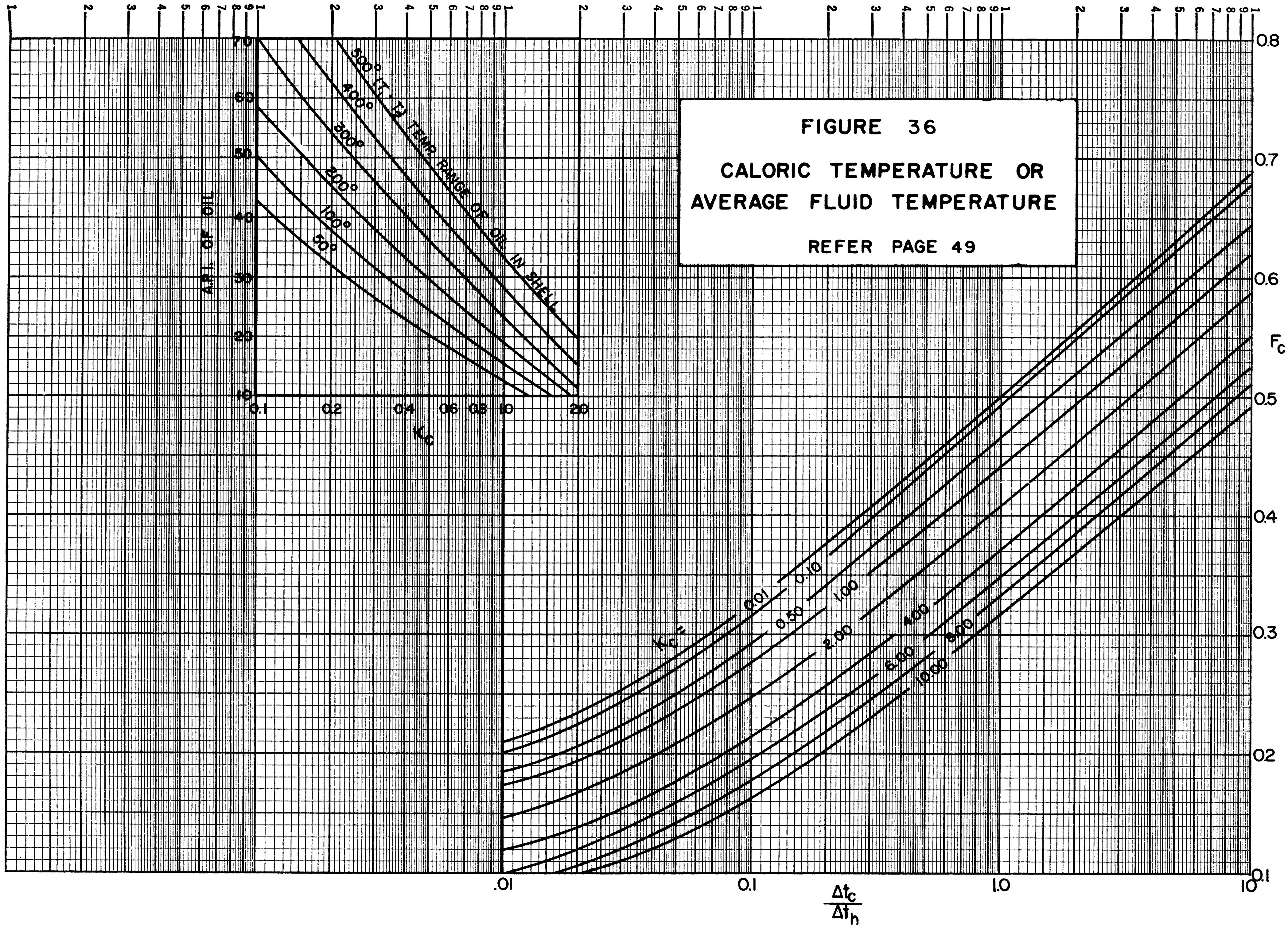


FIGURE 36
CALORIC TEMPERATURE OR
AVERAGE FLUID TEMPERATURE
REFER PAGE 49
F_c
$\frac{\Delta t_c}{\Delta t_h}$
K_c
$K_c =$
0.01
0.10
0.50
1.00
2.00
4.00
6.00
8.00
10.00
500° $(T_1 - T_2)$ TEMP. RANGE OF OIL IN SHELL
400°
300°
200°
100°
50°
A.P.I. OF OIL

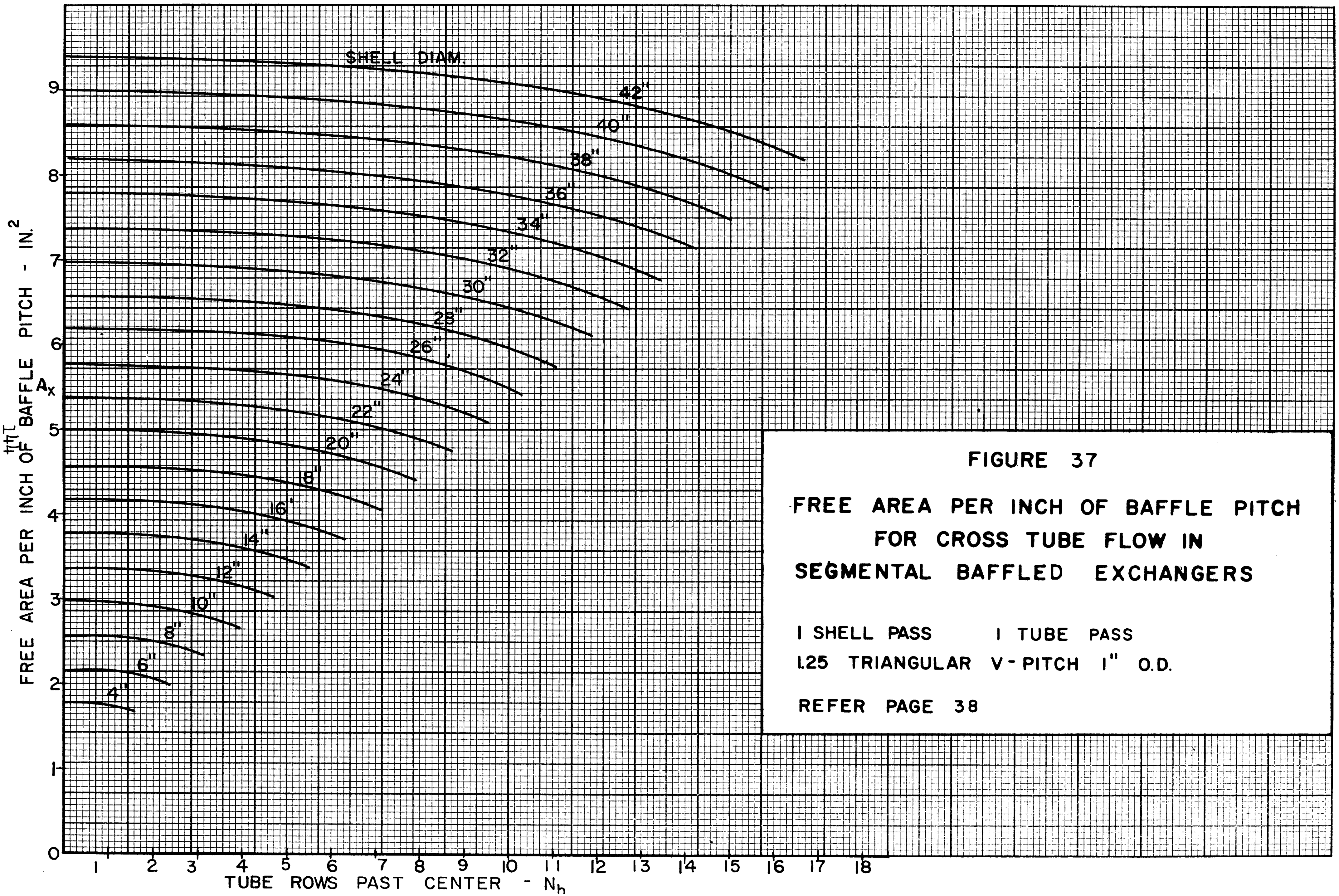


FIGURE 37

FREE AREA PER INCH OF BAFFLE PITCH FOR CROSS TUBE FLOW IN SEGMENTAL BAFFLED EXCHANGERS

1 SHELL PASS 1 TUBE PASS

1.25 TRIANGULAR V-PITCH 1" O.D.

REFER PAGE 38

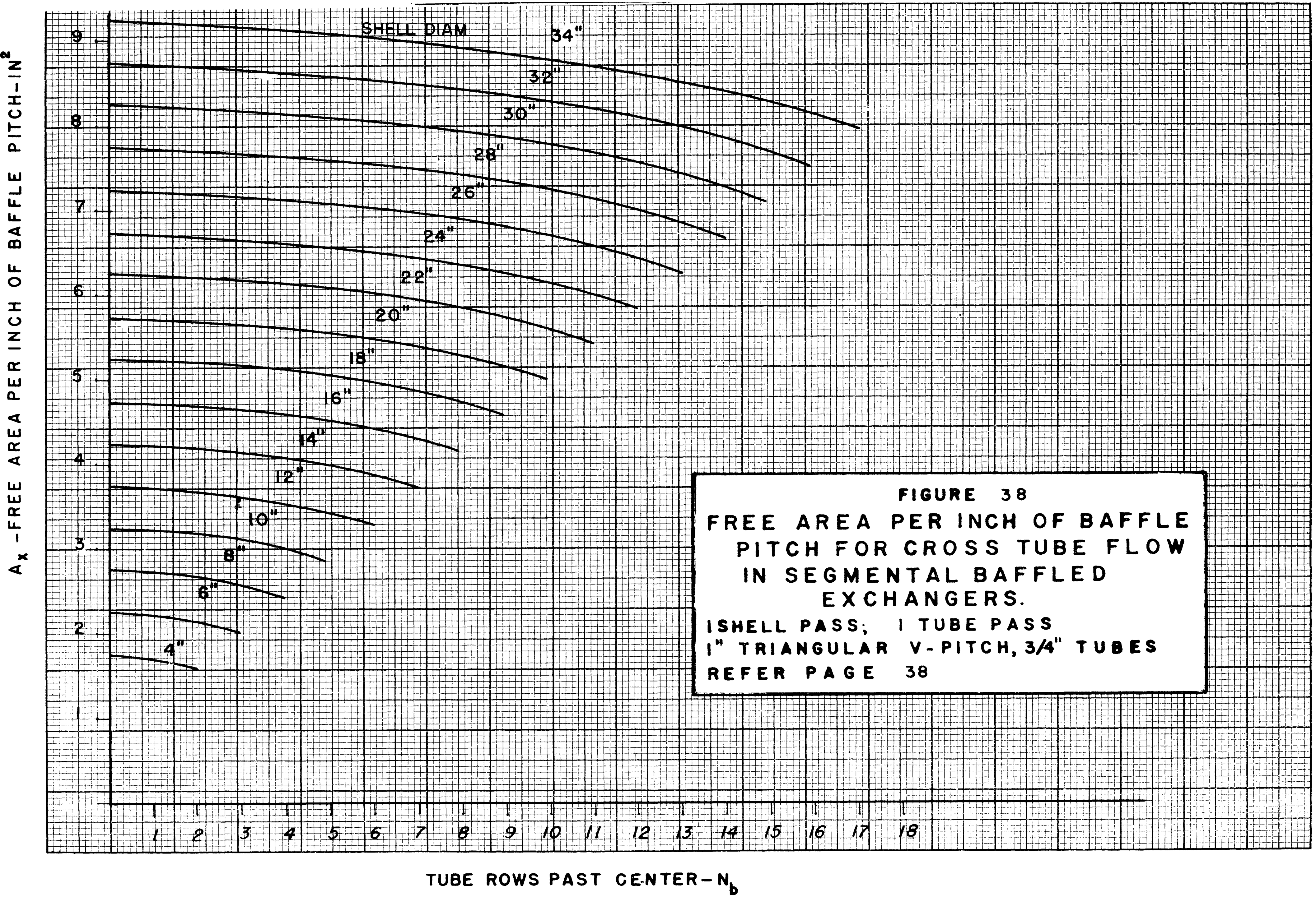


FIGURE 38

FREE AREA PER INCH OF BAFFLE PITCH FOR CROSS TUBE FLOW IN SEGMENTAL BAFFLED EXCHANGERS.

1 SHELL PASS; 1 TUBE PASS

1" TRIANGULAR V-PITCH, 3/4" TUBES

REFER PAGE 38

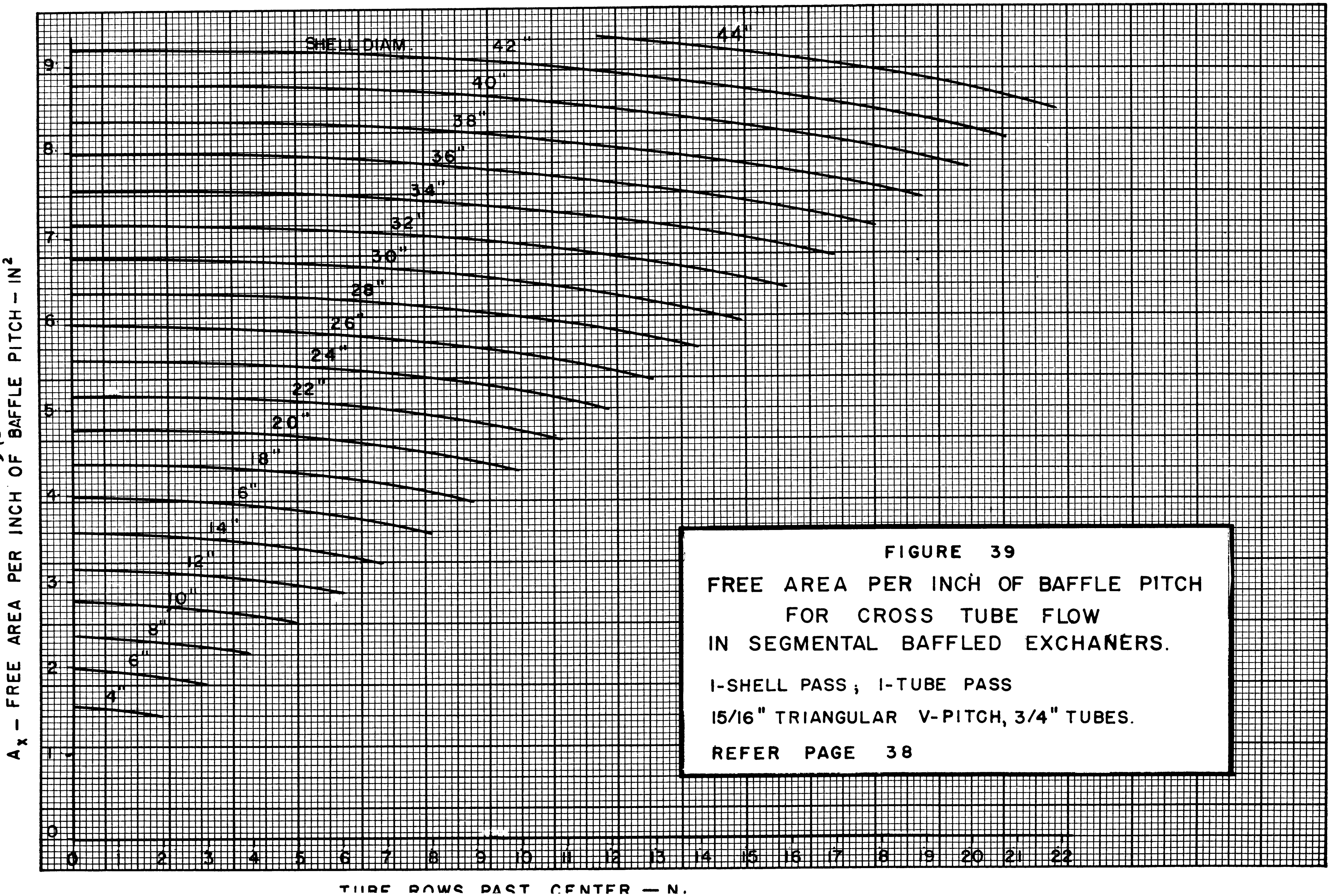


FIGURE 39
FREE AREA PER INCH OF BAFFLE PITCH
FOR CROSS TUBE FLOW
IN SEGMENTAL BAFFLED EXCHANERS.
1-SHELL PASS; 1-TUBE PASS
15/16" TRIANGULAR V-PITCH, 3/4" TUBES.
REFER PAGE 38

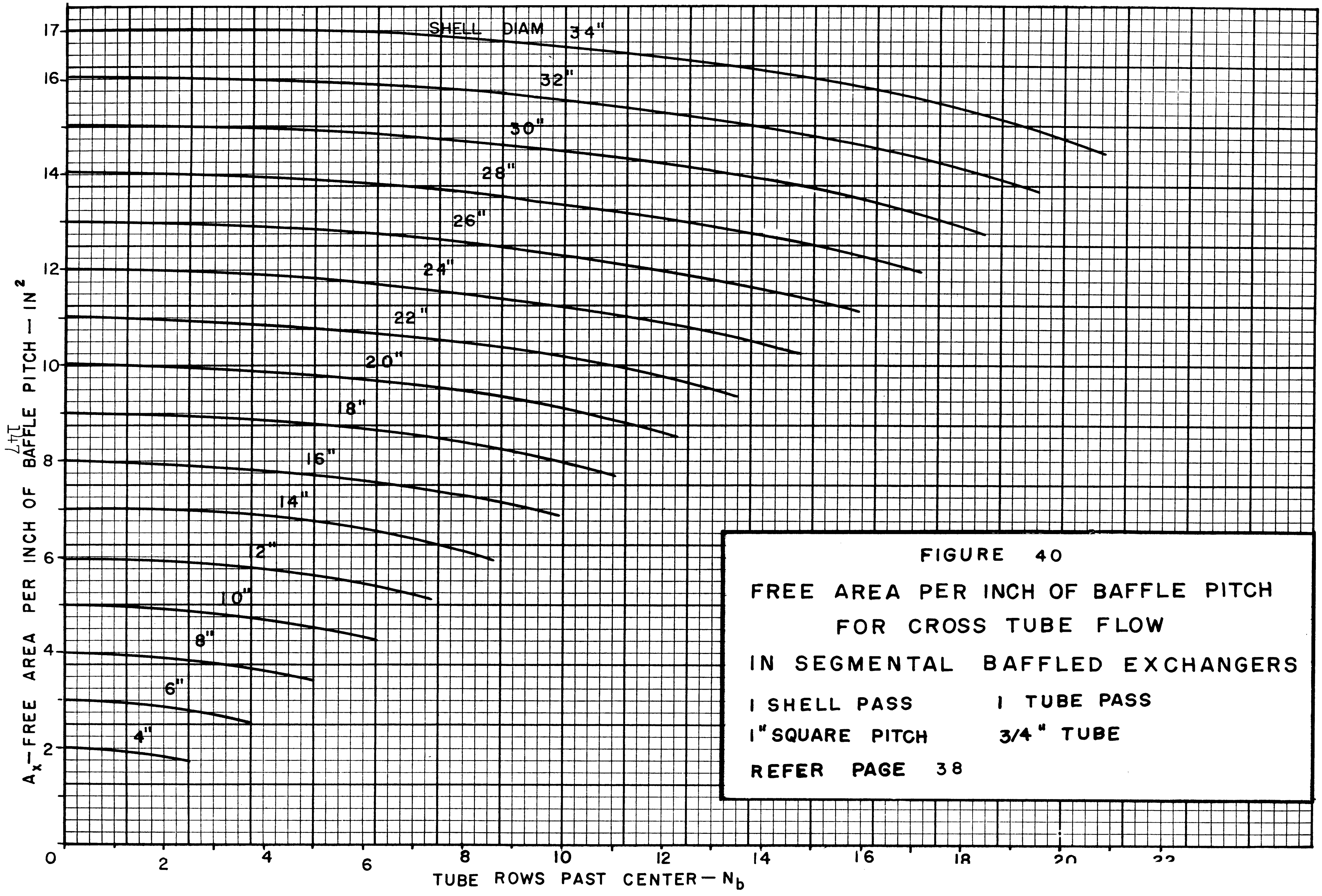


FIGURE 40

FREE AREA PER INCH OF BAFFLE PITCH FOR CROSS TUBE FLOW IN SEGMENTAL BAFFLED EXCHANGERS

1 SHELL PASS 1 TUBE PASS

1" SQUARE PITCH 3/4" TUBE

REFER PAGE 38

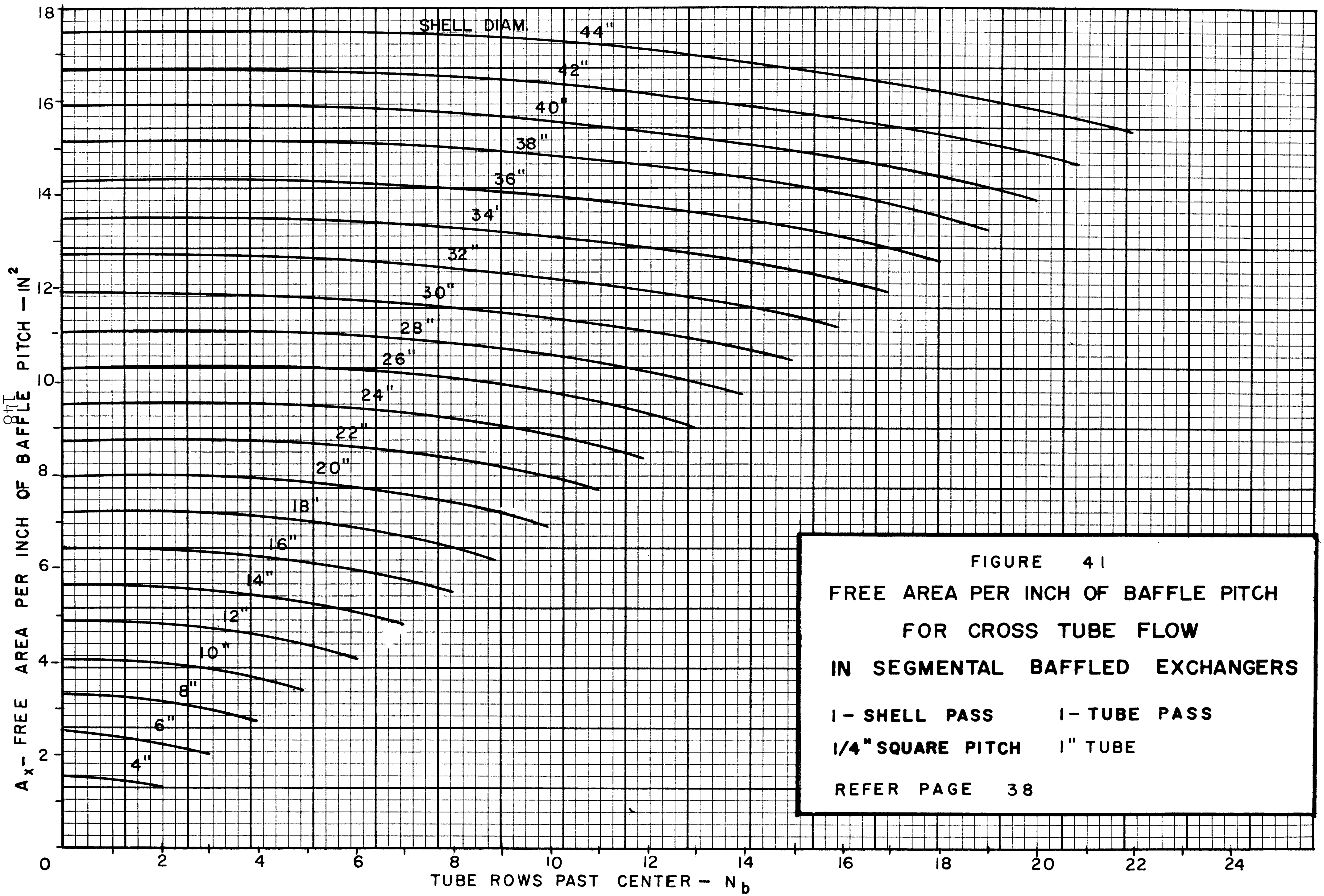


FIGURE 41

FREE AREA PER INCH OF BAFFLE PITCH
FOR CROSS TUBE FLOW
IN SEGMENTAL BAFFLED EXCHANGERS

1 - SHELL PASS 1 - TUBE PASS
1/4" SQUARE PITCH 1" TUBE

REFER PAGE 38

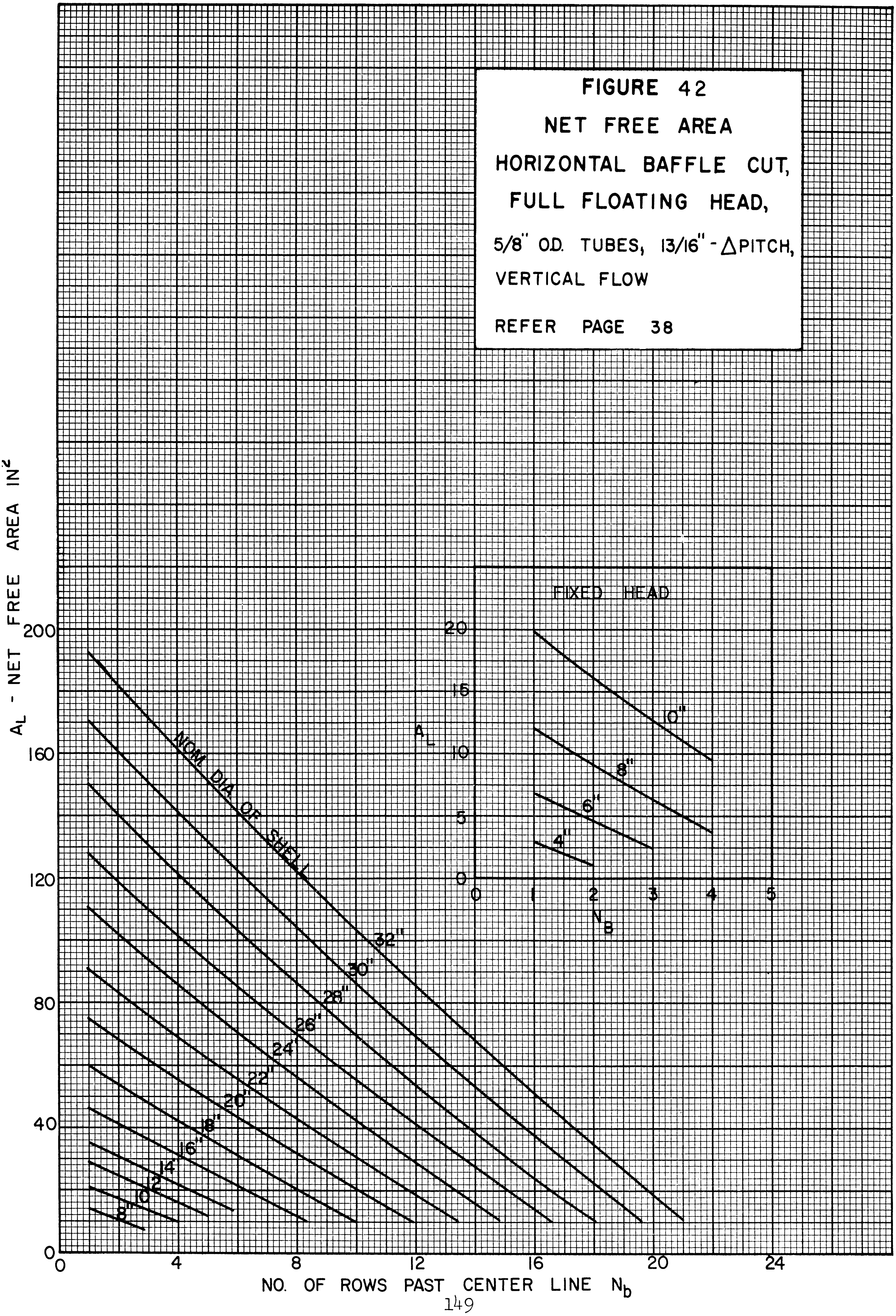


FIGURE 42
NET FREE AREA
HORIZONTAL BAFFLE CUT,
FULL FLOATING HEAD,
5/8" O.D. TUBES, 13/16" - ΔPITCH,
VERTICAL FLOW
REFER PAGE 38
FIXED HEAD
A_L
N_B
10"
8"
6"
4"
NOM. DIA. OF SHELL
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12"
10"
8"
A_L - NET FREE AREA IN²
NO. OF ROWS PAST CENTER LINE N_b

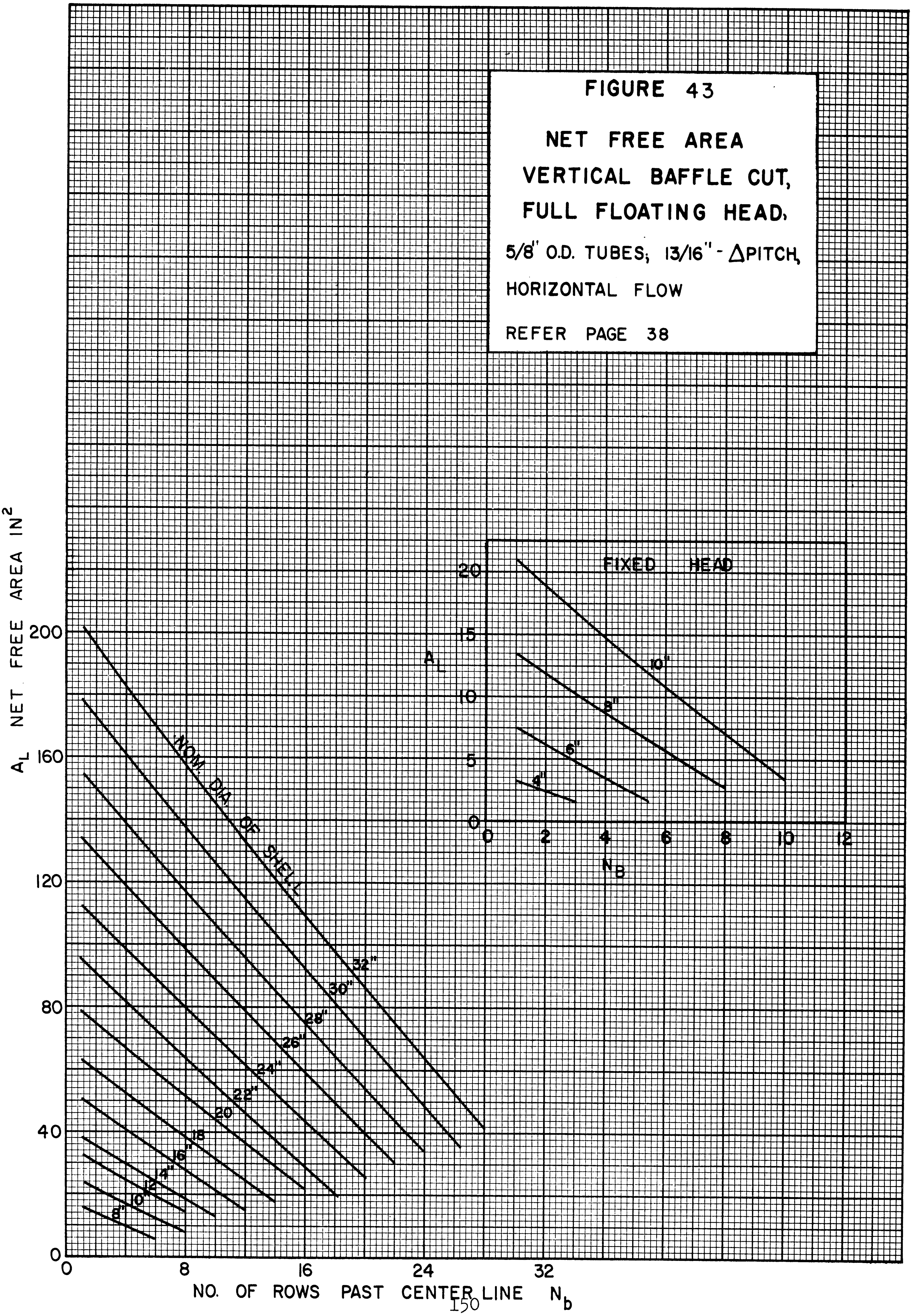


FIGURE 43
NET FREE AREA
VERTICAL BAFFLE CUT,
FULL FLOATING HEAD,
5/8" O.D. TUBES; 13/16" - ΔPITCH,
HORIZONTAL FLOW
REFER PAGE 38
FIXED HEAD
A_L
N_B
10"
8"
6"
4"
0 2 4 6 8 10 12
0 5 10 15 20
A_L NET FREE AREA IN^2
NOM. DIA. OF SHELL
32"
30"
28"
26"
24"
22"
20
18
16"
14"
12
10"
8"
0 40 80 120 160 200
0 8 16 24 32
NO. OF ROWS PAST CENTER LINE N_b

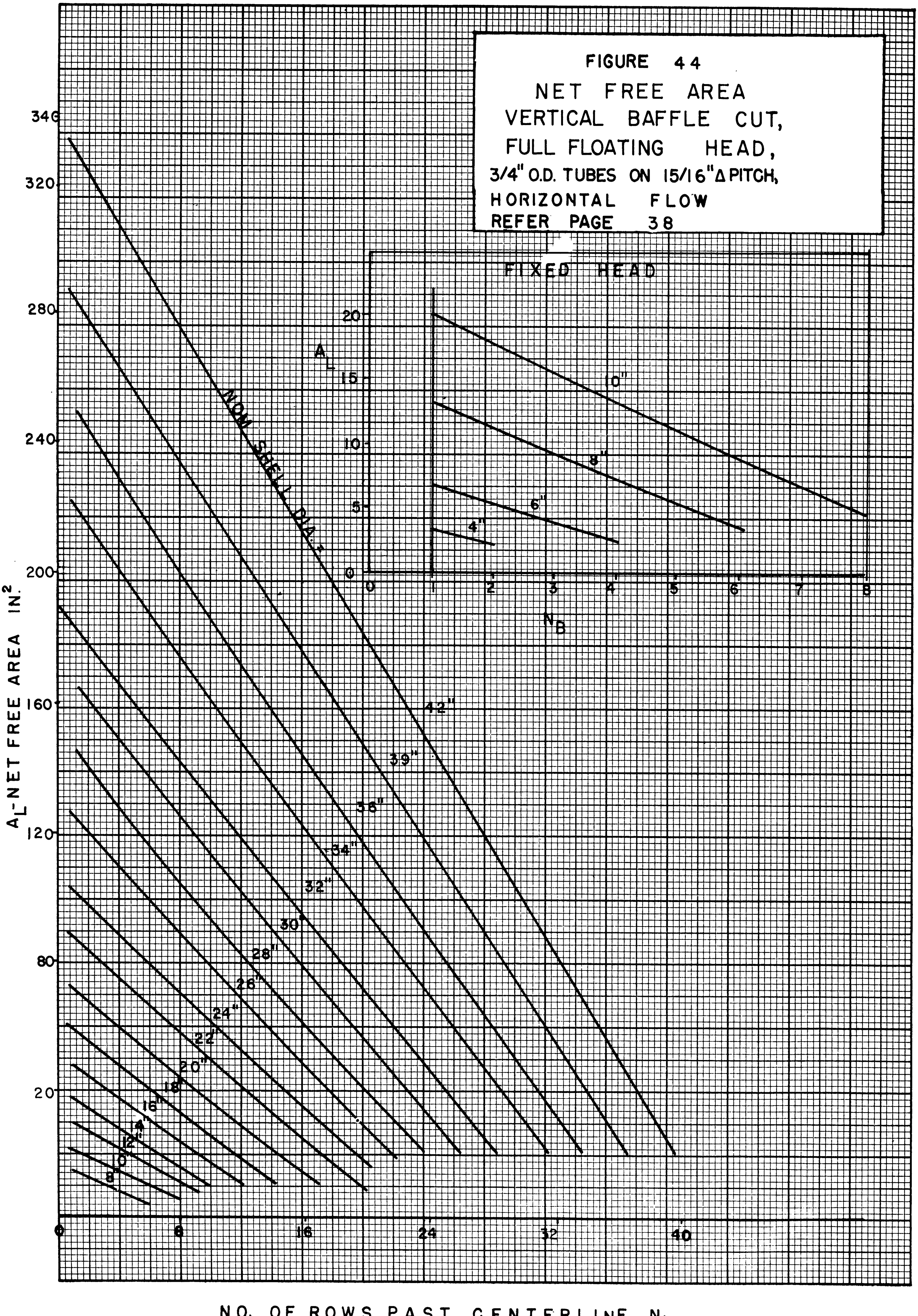


FIGURE 44
NET FREE AREA
VERTICAL BAFFLE CUT,
FULL FLOATING HEAD,
3/4" O.D. TUBES ON 15/16" Δ PITCH,
HORIZONTAL FLOW
REFER PAGE 38
FIXED HEAD
A_L
N_B
10"
8"
6"
4"
NOM. SHELL DIA.
42"
39"
36"
34"
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12"
10"
8"
A_L-NET FREE AREA IN.^{2}
NO. OF ROWS PAST CENTERLINE N_b

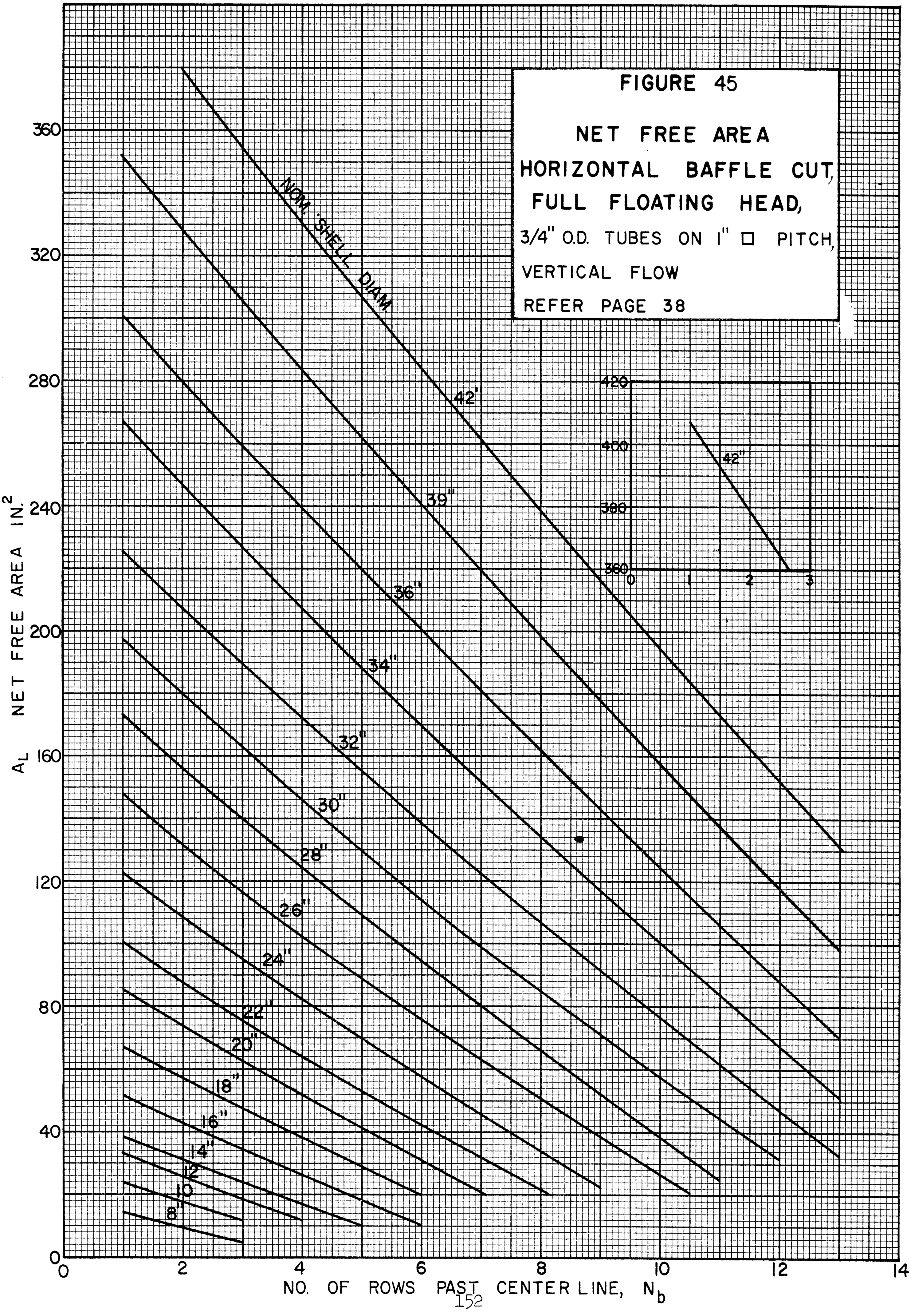


FIGURE 45
NET FREE AREA
HORIZONTAL BAFFLE CUT,
FULL FLOATING HEAD,
3/4" O.D. TUBES ON 1" □ PITCH,
VERTICAL FLOW
REFER PAGE 38
NOM. SHELL DIAM
42"
39"
36"
34"
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12
10
8
A_L NET FREE AREA IN.^{2}
NO. OF ROWS PAST CENTER LINE, N_b
0
40
80
120
160
200
240
280
320
360
2
4
6
8
10
12
14
420
400
380
360
0
1
2
3

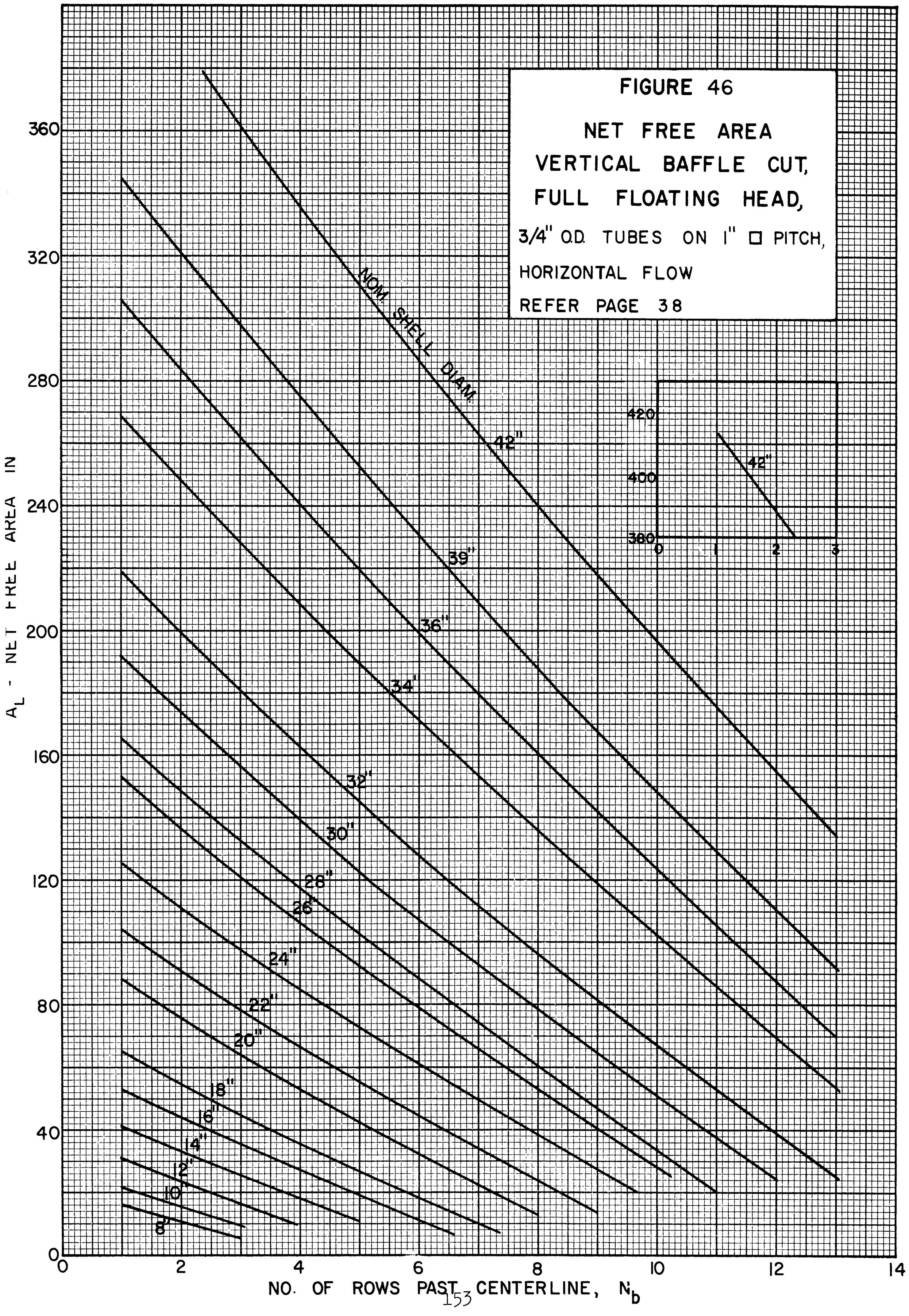


FIGURE 46
NET FREE AREA
VERTICAL BAFFLE CUT,
FULL FLOATING HEAD,
3/4" O.D. TUBES ON 1" □ PITCH,
HORIZONTAL FLOW
REFER PAGE 38
NOM. SHELL DIAM.
42"
39"
36"
34"
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12"
10"
8"
420
400
380
0
1
2
3
360
320
280
240
200
160
120
80
40
0
2
4
6
8
10
12
14
A_L - NET FREE AREA IN
NO. OF ROWS PAST CENTERLINE, N_b

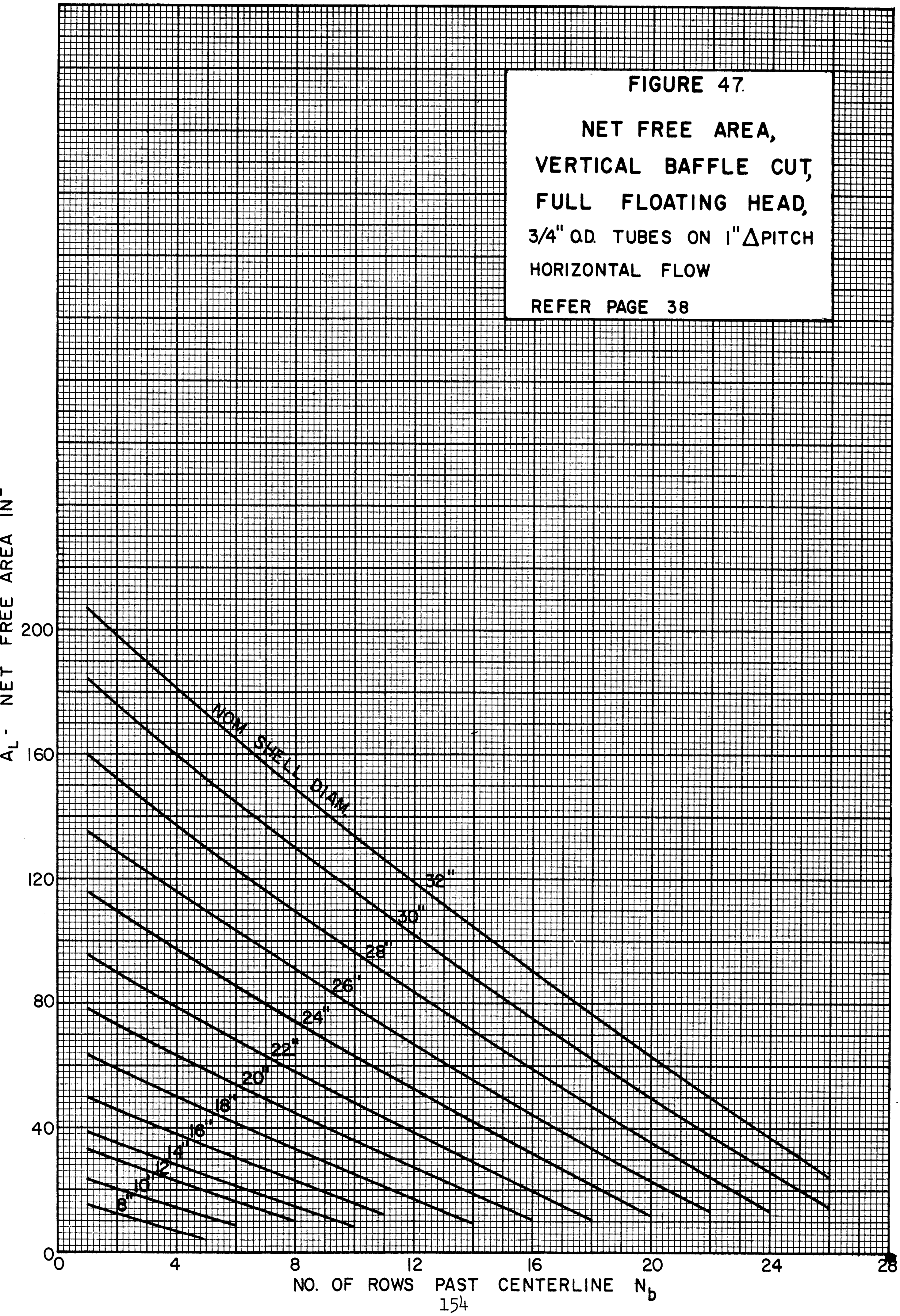


FIGURE 47.
NET FREE AREA,
VERTICAL BAFFLE CUT,
FULL FLOATING HEAD,
3/4" O.D. TUBES ON 1" Δ PITCH
HORIZONTAL FLOW
REFER PAGE 38
NOM. SHELL DIAM.
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12"
10"
8"
A_L - NET FREE AREA IN²
200
160
120
80
40
0
0
4
8
12
16
20
24
28
NO. OF ROWS PAST CENTERLINE N_b

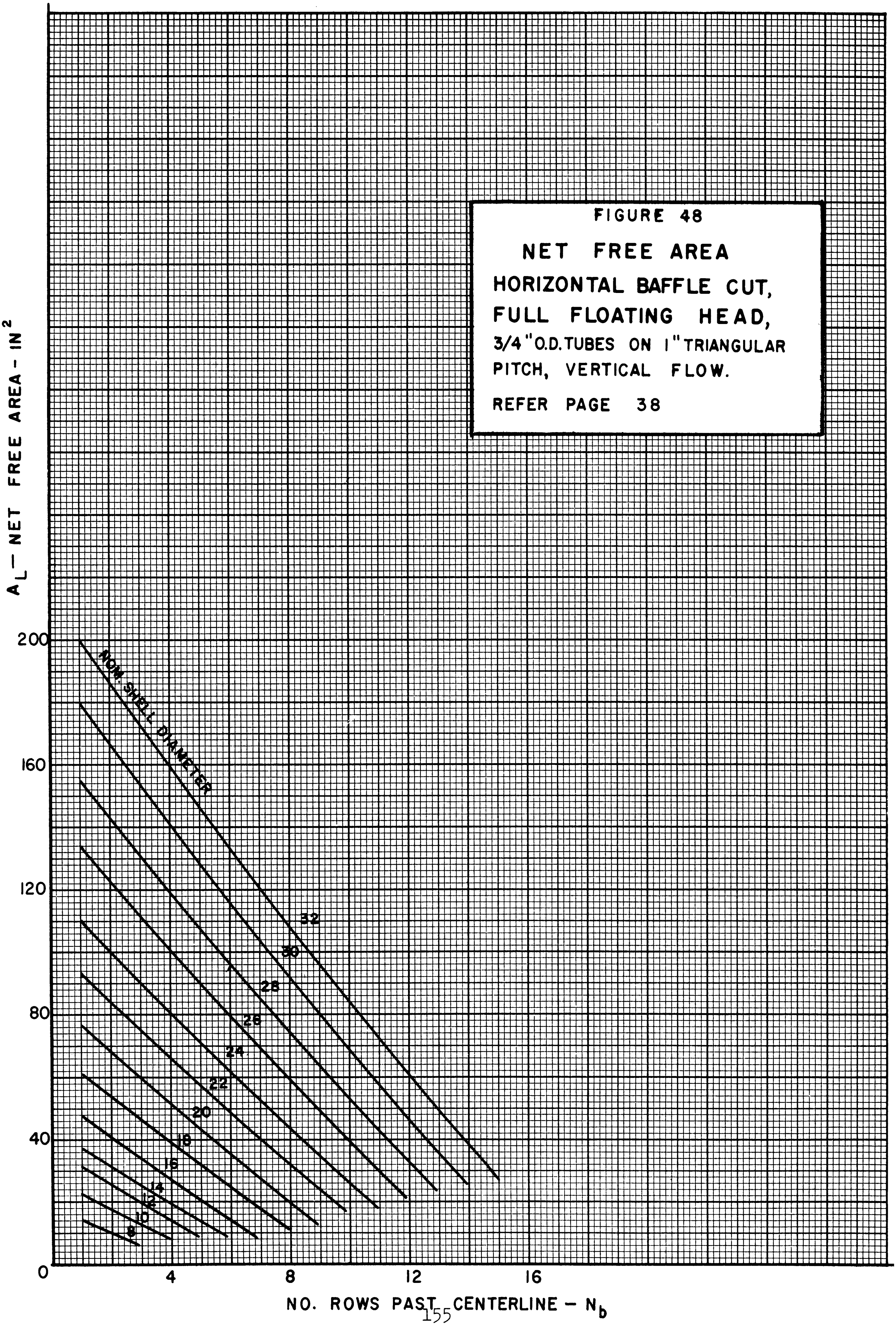


FIGURE 48
NET FREE AREA
HORIZONTAL BAFFLE CUT,
FULL FLOATING HEAD,
3/4" O.D. TUBES ON 1" TRIANGULAR
PITCH, VERTICAL FLOW.
REFER PAGE 38
A_L – NET FREE AREA - IN^2
NO. ROWS PAST CENTERLINE – N_b
NOM. SHELL DIAMETER
0
40
80
120
160
200
4
8
12
16
8
10
12
14
16
18
20
22
24
26
28
30
32

TABLE I

TABLES OF NORMAL FOULING FACTORS

A. Fouling Factors for Water

Temperature of Heating Medium	Up to 240°F		240°F - 600°F	
Temperature of Water	125°F or less		Over 125°F	
Types of Water	Water Velocity ft/sec		Water Velocity ft/sec	
	3 ft and less	over 3 ft	3 ft and less	over 3 ft
Sea water	0.0005	0.0005	0.001	0.001
Brackish water	0.002	0.001	0.003	0.002
Cooling tower and artificial spray pond				
treated make up	0.001	0.001	0.002	0.002
untreated	0.003	0.003	0.005	0.004
City and well water (e.g., Great Lakes)	0.001	0.001	0.002	0.002
Great Lakes	0.001	0.001	0.002	0.002
River water				
Minimum	0.002	0.001	0.003	0.002
Mississippi	0.003	0.002	0.004	0.003
Delaware; Schuykill	0.003	0.002	0.004	0.003
East River; New York Bay	0.003	0.002	0.004	0.003
Chicago Sanitary Canal	0.008	0.006	0.010	0.008
Muddy or silty	0.003	0.002	0.004	0.003
Hard (over 15 grains/gal)	0.003	0.003	0.005	0.005
Engine jacket	0.001	0.001	0.001	0.001
Distilled	0.0005	0.0005	0.0005	0.0005
Treater boiler feed water	0.001	0.0005	0.001	0.001
Boiler blowdown	0.002	0.002	0.002	0.002

If the heating medium temperature is over 400°F, the ratings in the last two columns should be modified accordingly.

B. Fouling Factors for Industrial Oils

Fuel oil	0.005
Clean recirculating oil	0.001
Machinery and transformer oils	0.001
Quenching oils	0.004
Vegetable oils	0.003

C. Fouling Factors for Industrial Gases and Vapors

Coke oven gas and other manufactured gas	0.01
Diesel engine exhaust gas	0.01
Organic vapors	0.0005
Steam (non-oil bearing)	0.0005
Alcohol vapors	0.0005
Steam exhaust (oil bearings from reciprocating engines)	0.001
Refrigerating vapors (condensing from reciprocating compressors)	0.002
Air	0.002

D. Fouling Factors for Industrial Liquids

Organic	0.001
Refrigerating liquids, heating, cooling or evaporating	0.001
Brine (cooling)	0.001

E. Fouling Factors for Atmospheric Distillation Units

Overhead untreated vapors	0.0013
Overhead treated vapors	0.003
Side stream cuts	0.0013

F. Fouling Factors for Vacuum Distillation Units

Overhead vapors to oil	
From bubble tower (partial condenser)	0.001
From flash pot (no appreciable reflux)	0.003
Overhead vapors in water cooler condensers	
From bubble tower (final condenser)	0.001
From flash pot	0.004
Side stream	
To oil	0.001
To water	0.002
Residual bottoms, less than 20° API	0.005
Distillak bottoms, over 20° API	0.002

G. Fouling Factors for Cracking Units

Gas oil feeds	
Under 500°F	0.002
500°F and over	0.003
Naphtha feed	
Under 500°F	0.002
500°F and over	0.004

Separator vapors (vapors from separator) Flash pot and revaporiser	0.006
Bubble tower vapors	0.002
Residuum	0.010

H. Fouling Factors for Crude Oil Stream

	0-199°F Velocity ft/sec			200-299°F Velocity ft/sec		
	Under 2 ft	2-4 ft	6 ft & over	Under 2 ft	2-4 ft	4 ft & over
Dry	0.003	0.002	0.002	0.003	0.002	0.002
Salt*	0.003	0.002	0.002	0.005	0.006	0.004

	300-699°F Velocity ft/sec			500° and over Velocity ft/sec		
	Under 2 ft	2-6 ft	4 ft & over	Under 2 ft	2-6 ft	4 ft & over
Dry	0.004	0.003	0.002	0.005	0.004	0.003
Salt*	0.006	0.005	0.004	0.007	0.006	0.005

*Refers to a wet-crude--any crude that has not been dehydrated.

I. Fouling Factors for Absorption Units

	Oil Field Natural Gasoline Plants	Refinery Vapor Recovery Plants
Gas	0.002	0.002
Fat oil	0.001	0.002
Lean oil	0.002	0.002
O.H. vapors	0.0005	0.001
Gasoline	0.0005	0.0005

J. Fouling Factors for Natural Gasoline Stabilizer Units

Feed	0.0005
O.H. Vapors	0.0005
Product coolers and exchangers	0.0005
Product reboilers	0.001

K. Fouling Factors for Debutanisers, Depropanisers, Depentanisers and Alkylation Units

Feed	0.001
O.H. Vapors	0.001
Product Coolers	0.001
Product Reboilers	0.002
Reactorsfeed	0.002

L. Fouling Factors for Tube Treating Units

Solvent oil mixed feed	0.002
O.H. Vapors	0.001
Refined oil	0.001
*Refined oil heat exchangers water	
Cooled gums and tars	0.003
Oil cooled and steam generators	0.005
Water cooled	0.003
Solvent	0.001

M. Fouling Factors for Deasphalting Units

Feed oil	0.002
Solvent	0.001
Asphalt and resin	
Oil cooled and steam generators	0.005
Water cooled	0.003
Solvent vapors	0.001
Refined oil	0.001
Refined oil and water cooled	0.003

N. Fouling Factors for Dewaxing Units

Tube oil	0.001
Solvent	0.001
Oil wax mix heating	0.001
*Oil wax mix cooling	0.003

O. Fouling Factors for Hydrogen Sulfide Removal Units

For overhead vapors	0.001
Solution exchanger coolers	0.0016
Reboilers	0.0016

*Precautions must be taken against deposition of wax.

TABLE II

PIPE DIMENSIONS AND DATA TO CALCULATE FLOW RATES

Outer Diam. in.	BWG	ID, in.	Surface Area, sq ft/ft	Sq in., Outside	Sq in., Inside	F*	P*
3/8	22	.319	.09815	.1104	.080	4.02	.50
	20	.305			.072	4.46	.555
	18	.277			.060	5.33	.664
1/2	20	.430	.1309	.1964	.1452	2.21	.275
	18	.402			.1269	2.53	.315
	16	.370			.1075	2.99	.372
5/8	20	.555	.1636	.3068	.2419	1.33	.165
	18	.527			.2181	1.47	.183
	16	.495			.1924	1.67	.208
	14	.459			.1655	1.94	.242
3/4	18	.652	.1963	.4418	.3338	1.96	.120
	17	.644			.314	1.02	.127
	16	.620			.3019	1.06	.133
	15	.606			.284	1.13	.141
	14	.584			.2679	1.20	.149
	13	.560			.247	1.30	.162
	12	.532			.223	1.44	.179
1/2 IPS	12	.62	.220	.550	.3019	1.06	.133
1	18	.902	.2618	.7854	.6390	.60	.0626
	17	.884			.6138	.524	.0652
	16	.870			.5945	.540	.0673
	15	.856			.570	.563	.0702
	14	.834			.5463	.587	.0732
	13	.810			.515	.630	.0776
	12	.782			.477	.67	.0839
	11	.760			.455	.705	.088
	10	.732			.421	.77	.095
1-1/4	18	1.152	.32725	1.227	1.0423	.3075	.0383
	17	1.134			1.012	.317	.0395
	16	1.120			.985	.326	.0406
	15	1.108			.967	.332	.0414
	14	1.084			.923	.350	.0434
	13	1.060			.882	.370	.0454
	12	1.032			.838	.390	.0477
	11	1.010			.801	.40	.0499
	10	.982			.757	.430	.0528

TABLE II (cont.)

Outer Diam. in.	BWG	ID, in.	Surface Area, sq ft/ft	Sq in., Outside	Sq in., Inside	F*	P*
1-1/2	16	1.370	.3927	1.767	1.474	.2175	.0271
	15	1.358			1.453	.221	.0275
	14	1.334			1.398	.23	.0286
	13	1.310			1.348	.24	.0297
	12	1.282			1.291	.25	.0310
	11	1.260			1.247	.258	.0321
	10	1.232			1.192	.27	.0336
1-3/4	16	1.620	.4581	2.405	2.061	.156	.0194
	15	1.608			2.036	.158	.0197
	14	1.584			1.971	.163	.0202
	13	1.560			1.911	.168	.0209
	12	1.532			1.843	.174	.0217
	11	1.510			1.791	.179	.0223
	10	1.482			1.725	.186	.0232
2	14	1.834	.5236	3.142	2.642	.1215	.0151
	12	1.782			2.494	.129	.0160
	10	1.732			2.356	.136	.0170

Wall Thickness, inches,	.028	.035	.049	.058	.065	.072	.083	.095
BWG	22	20	18	17	16	15	14	13

Wall Thickness, inches	.109	.120	.134	.148	.165	.180	.203
BWG	12	11	10	9	8	7	6

Use of factors F and P

$$\text{Velocity, ft/sec,} \quad V = \frac{\text{G.P.M.}}{\text{No. tubes/pass}} \times F = \frac{0.321 \times \text{G.P.M.}}{\text{sq in.}}$$

$$\text{Mass Velocity, sq ft/sec,} \quad G_t' = \frac{\text{lbs/hour}}{\text{No. of tubes/pass}} \times P = \frac{0.04 \times \text{lbs/hour}}{\text{sq in.}}\ .$$

TABLE III

TUBE-SHEET LAYOUTS (TUBE-COUNTS) FOR FIXED TUBE SHEETS

A. 1/2 in. IPS on 1-1/32 in. Triangular Pitch

Shell Size, in.	Tube Line in.	Number of Passes 1	2	4	6	8	10	12	Number of Rows Across	Net Free Distance
6	5-13/16	22	20	16						
8	7-13/16	42	40	32						
10	9-15/16	73	68	56						
12	11-13/16	109	98	88						
14	13-1/4	130	126	114						
16	15-1/4	178	172	154						
18	17-1/4	235	224	204						
20	19-1/8	288	282	260						
22	21-1/8	352	346	330						
24	23-1/8	421	412	400						
26	25-1/8	506	498	468						
28	27-1/8	588	578	558						
30	29-1/8	684	672	648						
32	31-1/8	782	774	744						

B. 1 in. OD Tubes on 1-1/4 in. Triangular Pitch

Shell Size, in.	Tube Line in.	1	2	4	6	8	10	12	Number of Rows Across	Net Free Distance
6	5-13/16	14	14							
8	7-13/16	31	26	26	26	20				
10	9-15/16	48	48	42	34	32	34	28		
12	11-13/16	68	66	56	54	52	50	44		
14	13-1/4	88	82	78	74	72	70	68		
16	15-1/4	121	114	110	108	104	92	92		
18	17-1/4	151	148	140	136	132	130	124		
20	19-1/8	199	184	176	168	164	160	156		
22	21-1/8	241	232	220	216	208	204	200		
24	23-1/8	284	282	270	262	256	252	244		
26	25-1/8	345	334	322	316	304	296	288		
28	27-1/8	397	392	372	366	360	346	340		
30	29-1/8	454	454	436	430	416	410	396		
32	31-1/8	530	522	506	494	484	474	468		

C. 3/8 in. OD Tubes on 1/2 in. Triangular Pitch

Shell Size, in.	Tube Line, in.	Number of Passes 1	2	4	6	8	10	12	Number of Rows Across	Net Free Distance
	5-13/16	109	98	78	70	64	62			
6	7-13/16	202	188	162	154	144	138	124		
	9-15/16	333	312	278	256	256	242	232		
8	11-13/16	475	452	410	402	380	366	356		
14	13-1/4	604	578	524	510	496	486	464		
16	15-1/4	805	776	716	704	672	664	644		
18	17-1/4	1039	1006	940	924	888	876	852		
20	19-1/8	1285	1236	1172	1152	1120	1094	1072		
22	21-1/8	1555	1514	1440	1418	1388	1360	1332		
24	23-1/8	1887	1846	1754	1716	1692	1660	1628		
26	25-1/8	2221	2172	2082	2056	2008	1982	1952		
28	27-1/8	2592	2538	2440	2410	2356	2330	2296		
30	29-1/8	3001	2944	2836	2806	2748	2720	2676		
32	31-1/8	3427	3366	3254	3218	3160	3130	3080		

D. 5/8 in. OD Tubes on 13/16 in. Triangular Pitch

6	5-13/16	38	36	28	26	24		
8	7-13/16	73	68	56	54	52	46	
10	9-15/16	121	114	102	96	92	90	84
12	11-13/16	174	168	152	148	140	134	124
14	13-1/4	220	214	196	186	180	172	168
16	15-1/4	301	286	264	258	244	240	232
18	17-1/4	380	372	348	334	324	324	308
20	19-1/8	475	464	432	420	416	394	396
22	21-1/8	583	566	532	522	512	502	484
24	23-1/8	696	682	652	640	624	612	600
26	25-1/8	824	816	770	764	740	728	712
28	27-1/8	970	952	914	896	880	864	856
30	29-1/8	1123	1104	1060	1040	1024	1012	992
32	31-1/8	1285	1258	1216	1202	1188	1170	1148

E. 3/4 in. OD Tubes on 15/16 in. Triangular Pitch

Shell Size, in.	Tube Line, in.	Number of Passes							Number of Rows Across	Free Distance
		1	2	4	6	8	10	12		
6	5-13/16	31	26	20	16	16				
8	7-13/16	55	48	42	42	36				
10	9-15/16	88	82	76	74	72	68	64		
12	11-13/16	130	120	110	108	104	102	92		
14	13-1/4	163	152	146	140	132	130	120		
16	15-1/4	216	214	196	186	188	180	172		
18	17-1/4	284	274	256	252	240	234	224		
20	19-1/8	349	342	322	314	300	292	284		
22	21-1/8	433	420	404	394	380	374	364		
24	23-1/8	518	498	484	474	468	458	440		
26	25-1/8	614	606	574	562	552	542	528		
28	27-1/8	721	708	678	666	644	636	620		
30	29-1/8	843	818	792	776	764	752	736		
32	31-1/8	955	938	910	892	880	868	852		

F. 1-1/4 in. OD Tubes on 1-9/16 in. - 60° Pitch

6	5-13/16	8	6							
8	7-13/16	19	18	16	14					
10	9-15/16	31	30	26	26	24	24			
12	11-13/16	42	40	36	34	32	30	28		
14	13-1/4	55	52	48	46	44	42	40		
16	15-1/4	74	72	66	66	60	58	60		
18	17-1/4	96	98	86	82	80	76	72		
20	19-1/8	121	118	110	108	104	102	100		
22	21-1/8	151	146	138	136	132	128	124		
24	23-1/8	183	180	172	168	160	152	148		
26	25-1/8	212	214	200	192	192	184	180		
28	27-1/8	253	242	236	232	228	226	216		
30	29-1/8	295	286	278	272	268	260	252		
32	31-1/8	337	330	322	310	304	300	292		

G. 5/8 in. OD Tubes on 7/8 in. Triangular Pitch

Shell Size, in.	Tube Line, in.	Number of Passes							Number of Rows Across	Net Free Distance
		1	2	4	6	8	10	12		
6	5-13/16	31	30	26						
8	7-13/16	64	62	56						
10	9-15/16	109	98	88						
12	11-13/16	151	146	134						
14	13-1/4	199	184	176						
16	15-1/4	258	246	236						
18	17-1/4	330	318	302						
20	19-1/8	409	400	378						
22	21-1/8	499	486	472						
24	23-1/8	604	588	566						
26	25-1/8	715	704	670						
28	27-1/8	847	818	796						
30	29-1/8	966	952	924						
32	31-1/8	1106	1088	1052						

TABLE IV

TUBE SHEET LAYOUTS FOR FULL FLOATING HEAD HEAT EXCHANGERS

A. 5/8 in. OD Tubes on 13/16 in. Triangular Pitch

Shell Size, in	Tube Line, in.	Number of Passes 1	2	4	6	8	10	12	Number of Rows Across	Net Free Distance
6	5-1/16		30							
8	7-1/16	56	52	48	46	40	38		15	3-7/8
10	9-3/16	100	98	86	82	76	68	68	21	4-1/2
12	11-1/16	151	146	126	124	120	114	104	25	4-3/8
14	12-3/8	199	184	168	160	152	148	136	29	5-7/8
16	14-3/8	262	250	236	228	220	212	208	33	5-3/4
18	16-3/8	349	334	310	300	292	284	276	39	6-3/4
20	18-1/4	433	416	400	390	368	358	348	45	8-3/4
22	20-1/4	530	514	494	478	468	460	448	49	7-1/2
24	22-1/4	649	626	598	586	572	562	548	53	8-1/16
26	23-7/8	747	732	696	686	672	664	652	57	8-1/2
28	25-7/8	874	858	824	818	804	792	772	63	9-3/8
30	27-7/8	1027	1006	966	952	936	918	900	67	10-3/8
32	29-7/8	1176	1152	1116	1102	1088	1070	1052	71	10-1/4

B. 3/4 in. OD Tubes on 15/16 in. Triangular Pitch

6										
8	7-1/16	12	40	32		24			13	4-5/8
10	9-3/16	76	72	60	58	52	46		17	4-5/8
12	11-1/16	110	106	94	94	84			21	4-3/16
14	12-3/8	140	138	122	118	112	106	104	25	5-1/2
16	14-3/8	199	184	176	164	160	156	148	29	5
18	16-3/8	254	242	228	224	220	212	204	33	5-1/4
20	18-1/4	316	308	294	284	276	268	260	37	6-3/8
22	20-1/4	392	384	364	354	344	342	324	41	7-1/8
24	22-1/4	480	468	442	434	420	414	400	45	7-7/8
26	23-7/8	559	538	514	506	492	484	472	49	8-7/8
28	25-7/8	649	642	618	602	592	580	564	53	8-1/8
30	27-7/8	764	744	720	712	692	684	668	57	8-7/8
32	29-7/8	874	866	832	820	804	796	776	63	10-3/8
34	32-7/16	1039	1024	992	970	960	950	928		
36	34-3/16	1159	1140	1104	1086	1063	1064	1040		
39	37-3/16	1369	1354	1318	1302	1280	1270	1240		
42	40-3/16	1607	1576	1540	1510	1500	1504	1480		

C. 1 in. OD Tubes on 1-1/4 in. Triangular Pitch

Shell Size, in.	Tube Line, in.	Number of Passes							Number of Rows Across	Net Free Distance
		1	2	4	6	8	10	12		
8	7-1/16	22	20	18					9	4-1/8
10	9-3/16	40	40	32	32				13	4-1/2
12	11-1/16	61	52	48	50	44	42		15	4-5/8
14	12-3/8	76	76	70	66	60	58	60	19	4-1/4
16	14-3/8	109	102	94	94	84	84	80	21	4-1/2
18	16-3/8	140	136	126	120	116	110	104	25	6-1/2
20	18-1/4	174	168	156	152	148	144	136	27	6-3/4
22	20-1/4	216	214	200	192	192	184	180	31	7
24	22-1/4	264	254	246	240	236	230	224	33	7-1/4
26	23-7/8	304	302	286	278	272	268	260	37	7-1/2
28	25-7/8	361	350	342	336	328	320	312	39	7-3/4
30	27-7/8	421	412	400	394	388	382	370	43	8
32	29-7/8	499	480	464	454	440	434	428	47	8-1/4
34	32-7/16	571	558	546	536	528	520	508		
36	34-3/16	645	622	614	604	600	584	576		
39	37-3/16	756	740	726	724	712	696	688		
42	40-3/16	883	878	854	846	824	812	812		

D. 4 in. OD Tubes on 1 in. Square Pitch

8	7-1/6	32	32	30	24				7	3-5/8
10	9-3/16	57	56	52	52				9	4-1/14
12	11-1/16	89	82	78	76	76			11	4-5/8
14	12-3/8	108	108	98	92	92			11	6
16	14-3/8	148	148	136	128	128	120	120	14	6-1/2
18	16-3/8	192	188	180	180	180	168	168	16	7
20	18-1/4	242	236	224	220	220	212	216	17	7-3/8
22	20-1/4	304	300	284	280	280	268	268	20	7-7/8
24	22-1/4	362	360	352	344	340	332	332	21	8-3/8
26	23-7/8	421	412	402	392	392	392	392	23	8-7/8
28	25-7/8	502	488	480	476	472	460	460	25	9-3/8
30	27-7/8	580	566	566	548	548	540	540	27	9-7/8
32	29-7/8	673	652	644	636	636	620	624	29	10-3/8
34	32-7/16	793	788	766	756	756	732	736		
36	34-3/16	882	876	860	852	848	828	820		
39	37-3/16	1052	1040	1024	1016	1008	992	992		
42	40-3/16	1224	1216	1196	1188	1184	1164	1168		

E. 1 in. Tubes on 1-1/4 in. Square Pitch

Shell Size, in.	Tube Line, in.	Number of Passes							Number of Rows Across	Net Free Distance
		1	2	4	6	8	10	12		
8	7-1/16	21	16	16					5	4-1/8
10	9-3/16	37	32	32	32				6	4-1/4
12	11-1/16	52	52	48	44	44			8	4-1/4
14	12-3/8	69	64	64	52	52	52	52	10	5-1/2
16	14-3/8	90	88	86	84	80	76	76	11	5-1/2
18	16-3/8	121	120	112	112	112	112	112	13	5-3/8
20	18-1/4	150	150	144	144	144	132	136	14	5-3/8
22	20-1/4	188	182	180	176	176	168	172	16	7-3/8
24	22-1/4	230	224	224	212	212	208	208	17	7-3/8
26	23-7/8	266	266	258	252	252	240	244	19	7-3/8
28	25-7/8	308	308	304	304	304	296	300	20	7-3/8
30	27-7/8	366	360	352	344	344	332	332	21	9-3/8
32	29-7/8	421	412	406	392	392	392	392	23	9-3/8
34	32-7/16	498	488	480	472	472	468	464		
36	34-3/16	558	554	540	536	536	524	524		
39	37-3/16	661	648	648	636	636	620	624		
42	40-3/16	777	772	754	744	732	732	732		

F. 3/4 in. OD Tubes on 1 in. 60^{o} Pitch

8	7-1/16	38	36	32	28	24			11	3-7/8
10	9-3/16	66	66	56	54	52	46		17	4-3/4
12	11-1/16	96	94	86	78	76	76	72	21	5-3/8
14	12-3/8	126	118	110	108	104	100	96	23	5-5/8
16	14-3/8	170	164	152	150	144	140	136	27	6-5/8
18	16-3/8	223	220	200	196	192	184	180	31	7-1/8
20	18-1/4	283	274	256	252	244	238	228	35	7-3/4
22	20-1/4	349	338	326	320	304	296	288	39	7-3/4
24	22-1/4	421	412	396	394	380	370	364	43	8
26	23-7/8	499	480	460	450	436	430	420	47	8-7/8
28	25-7/8	583	562	546	538	524	514	508	51	9-5/8
30	27-7/8	673	654	640	628	612	604	592	55	10-3/8
32	29-7/8	770	756	738	726	712	700	688	59	11-1/8

G. 3/4 in. Std. IPS Tubes 1-5/16 in. 60° Pitch Floating Head Type Exchangers

Shell Size in.	Tube Line, in.	Number of Passes							Number of Rows Across	Net Free Distance
		1	2	4	6	8	10	12		
8	7-1/16	19	18	16	14					
10	9-3/16	37	36	32	28	28				
12	11-1/16	55	52	48	46	44	42	36		
14	12-3/8	68	66	60	58	52	50	44		
16	12-3/8	94	94	86	78	76	76	72		
18	16-3/8	126	120	114	110	104	102	100		
20	18-1/4	158	152	146	140	132	128	124		
22	20-1/4	199	188	184	180	180	168	160		
24	22-1/4	241	232	224	220	212	208	200		
26	23-7/8	276	270	256	248	240	238	232		
28	25-7/8	324	318	306	300	296	288	280		
30	27-7/8	380	372	360	356	344	338	324		
32	29-7/8	442	428	416	406	404	394	384		

TABLE V

U-TUBE LAYOUT

A. 1 in. OD Tubes, 1-1/4 in. Triangular Pitch

Shell Size, in.			6 in.	8 in.	10 in.	12 in.	14 in.*	16 in.*	18 in.	20 in.	22 in.	24 in.	26 in.	28 in.	30 in.	32 in.
Tube Circle, in.			5-13/16	7-13/16	9-13/16	11-13/16	13-1/4	15-1/4	17-1/4	19-1/8	21-1/8	23-1/8	25-1/8	27-1/8	29-1/8	
Tot sq ft/12'nom			25	49.8	121.4	165.7	214	291	392	495	598	736	887	1036	1196	
Sq ft/ft st			2.1	4.2	9.4	14.1	18.3	25.1	34	43.3	52.8	65.3	79.5	93.5	109	
Sq ft in Bends			.6	1.2	3.4	5.5	7.8	12.2	18.6	25.5	34	46	60.1	76.1	94	
No. of U-Tubes			4	8	18	27	35	48	65	83	101	125	152	179	208	
Row No.	Radius in.	Bend in.	Number of U-Tubes													
Cntr.	2	6.3	1	1	2	3	3	4	4	5	5	6	7	7	8	
1	2	6.3	3	5	7	9	10	10	13	15	15	17	19	21	23	
2	3.1	9.7		2	6	8	9	11	12	14	16	16	18	20	22	
3	4.2	13.1			3	5	8	10	11	13	15	17	19	19	21	
4	5.2	16.5				2	5	7	10	12	14	16	18	20	20	
5	6.3	19.9						6	9	11	13	15	17	19	21	
6	7.4	23.3							6	8	10	14	16	18	20	
7	8.5	26.7								5	9	11	13	15	17	
8	9.6	30.1									4	8	12	14	16	
9	10.7	33.5										5	9	13	15	
10	11.7	36.9											4	10	12	
11	12.8	40.3												3	9	
12	13.9	43.7													4	

*Denotes off center pitch.

TABLE V (cont.)

B. 3/4 in. U Tubes on 15/16 in. Triangular Pitch 1-3/4 in. Minimum Radius

Shell Size, in.			8 in.	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	22 in.	24 in.	26 in.	28 in.	30 in.	32 in.
Tube Circle, in.			7-13/16	9-15/16	11-13/16	13-1/4	15-1/4	17-1/4	19-1/8	21-1/8	23-1/8	25-1/8	27-1/8	29-1/8	
Total sq ft/12'nom			87.9	143	215.3	301	404	541	694	856	1019	1218	1445	1653	
Sq ft/ft st			7.5	12.2	18.5	25.9	34.9	47.5	61.2	75.7	90.6	108.5	131	150.5	
Sq ft in Bends			2.2	4.2	7.3	11.4	16.9	25.5	36.2	48.7	63.4	81.0	106.6	127.6	
No. U-tubes, Total			19	31	47	66	89	121	156	193	231	276	333	383	
Row No.	Radius in.	Bend in.	Number of U-Tubes												
Cntr.	1-3/4	5-1/2	3	3	4	5	7	8	9	11	11	13	15	15	
1	1-3/4	5-1/2	7	9	11	13	15	17	19	21	23	25	29	31	
2	2-9/16	8-1/16	6	8	10	12	14	16	20	22	24	26	28	30	
3	3-3/8	10-5/8	3	7	9	11	13	17	19	21	23	25	27	29	
4	4-3/16	13-3/16		4	8	10	12	16	18	20	22	24	26	30	
5	5	15-3/4			5	9	11	15	17	19	21	23	27	29	
6	5-13/16	18-3/8				6	10	12	16	18	20	24	26	28	
7	6-5/8	20-13/16					7	11	13	17	19	23	25	27	
8	7-7/16	23-3/8						8	12	16	18	20	24	26	
9	8-1/4	25-15/16						1	9	13	17	19	23	25	
10	9-1/16	28-1/2							4	10	14	18	20	24	
11	9-7/8	31								5	11	15	19	21	
12	10-11/16	33-1/2									8	12	16	20	
13	11-1/2	36-1/8										9	15	17	
14	12-5/16	38-3/4											10	16	
15	13-1/8	41-1/4											5	11	
16	13-15/16	43-3/16												4	

TABLE V (cont.)

C. 1/2 in. IPS Tubes on 1-1/32 in. Triangular Pitch, 1-3/4 in. Minimum Radius

Shell Size, in.			8 in.*	10 in.	12 in.	14 in.*	16 in.	18 in.	20 in.	22 in.	24 in.	26 in.	28 in.	30 in.	32 in.
Tube Circle, in.			7-13/16	9-15/16	11-13/16	13-1/4	15-1/4	17-1/4	19-1/8	21-1/8	23-1/8	25-1/8	27-1/8	29-1/8	31-1/8
Tot sq ft/12'nom.			68.1	139	214.8	266.5	389.1	501.0	626	786	947	1111	1305	1508	1727
Sq ft/lin ft			5.7	11.9	18.5	22.9	33.9	43.5	55	69.5	84.5	100	117	138	158
Sq ft in Bends			1.6	4.0	7.3	9.5	16.6	23.5	32.7	44.9	59.0	75.7	94.4	116.4	142.3
No. of U-Tubes			13	27	42	52	77	99	125	158	192	228	267	311	358
Row No.	Radius in.	Bend in.	Number of U-Tubes												
Cntr.	1-3/4	5.5	2	3	3	4	5	5	6	7	8	8	9	10	11
1	1-3/4	5.5	6	9	11	12	13	15	17	19	21	23	26	28	30
2	2-21/32	8.4	5	8	10	11	14	16	16	20	22	24	25	27	29
3	3-9/16	11.2		5	9	10	13	15	17	19	21	23	24	26	28
4	4-7/16	14.0		2	6	9	12	14	16	18	20	22	23	27	29
5	5-11/32	16.8			3	6	9	13	15	17	19	21	24	26	28
6	6-7/32	19.6					8	10	14	16	18	20	23	25	27
7	7-1/8	22.4					3	7	11	15	17	19	22	24	26
8	8	25.1						4	8	12	16	18	21	23	25
9	8-29/32	28.0							5	9	13	17	18	22	24
10	9-13/16	30.9								6	10	14	17	19	23
11	10-11/16	33.6									7	11	14	18	20
12	11-19/32	36.5										8	13	15	19
13	12-15/32	39.2											8	12	16
14	13-3/8	42.1												9	13
15	14-1/4	44.8													10

*Denotes off center pitch.

TABLE V (cont.)

D. 3/4 in. OD Tubes on 1 in. Triangular Pitch, Center Tubes on Pitch with 1-3/4 in. Radius, 1-3/4 in. Minimum Radius

Shell Size, in.			8 in.*	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	22 in.	24 in.*	26 in.	28 in.	30 in.	32 in.
Tube Circle, in.			7-13/16	9-15/16	11-13/16	13-1/4	15-1/4	17-1/4	19-1/8	21-1/8	23-1/8	25-1/8	27-1/8	29-1/8	31-1/8
Tot sq ft/12'nom.			69.4	120.6	196.5	249.2	375	486	609	744	893	1071	1252	1454	1667
Sq ft/ft st			5.9	10.6	16.9	21.6	32.6	42.4	53.5	65.5	79.4	97.5	113.5	132	153
Sq ft in Bends			1.8	3.5	6.5	9.2	15.9	22.9	31.6	43.1	53.7	73.2	92.2	114.3	141.6
No. of U-Tubes			15	27	43	55	83	108	136	167	202	246	289	337	390
Row No.	Radius in.	Bend in.	Number of U-Tubes												
Cntr.	1.8	5.5	2	3	4	4	5	6	7	8	9	9	10	11	12
1	1.8	5.5	6	9	11	11	15	17	19	21	22	25	27	29	31
2	2.6	8.2	5	8	10	12	14	16	18	20	21	24	26	28	30
3	3.5	10.9	2	5	9	11	13	15	17	19	22	23	25	27	29
4	4.4	13.7		2	6	8	12	14	16	18	21	22	24	28	30
5	5.2	16.4			3	7	11	13	15	17	20	23	25	27	29
6	6.1	19.1				2	8	12	14	16	19	22	24	26	28
7	7.0	21.8					5	9	13	15	18	21	23	25	27
8	7.8	24.6						6	10	14	17	18	22	24	26
9	8.7	27.3							7	11	14	17	19	23	25
10	9.5	30.0								8	11	16	18	20	24
11	10.4	32.8									8	13	17	19	23
12	11.4	35.8										10	14	18	20
13	12.2	38.5										3	11	15	19
14	13.1	41.3											4	12	16
15	14.0	43.9												5	13
16	14.8	46.7													8

*Denotes off center pitch.

TABLE V (cont.)

E. 5/8 in. OD Tubes on 13/16 in. Triangular Pitch, 1-1/4 in. Minimum Radius
Center Tubes on Pitch 1-3/8 in. Radius Tube Circles 8 in.-32 in./HS-127

Shell Size, in.			8 in.	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	22 in.	24 in.	26 in.	28 in.	30 in.	32 in.
Tube Circle, in.			7-13/16	9-15/16	11-13/16	13-1/4	15-1/4	17-1/4	19-1/8	21-1/8	23-1/8	25-1/8	27-1/8	29-1/8	31-1/8
Tot sq ft/12'nom.			111.6	203.1	288.1	373.9	496.7	647.1	811.6	978	1191	1393	1647	1889	2167
Sq ft/lin ft			9.5	17.4	24.9	32.4	43.2	56.9	71.6	87	106.7	125.2	150	172.5	108.5
Sq ft in Bends			2.5	5.3	9.2	12.9	19.7	29.1	40.6	54.4	71.4	90.7	117	143	177
Tot No. U-Tubes			29	53*	76*	99*	132	174*	219	266	326*	383*	458*	527	607*
Row No.	Radius in.	Bend in.	Number of U-Tubes												
Cntr.	1-3/8	4.3	3	5	5	7	7	9	9	11	13	13	15	15	17
1	1-1/4	3.9	9	12	14	16	17	20	23	25	28	30	32	35	38
2	1-15/16	6.1	8	11	13	15	18	19	22	24	27	29	33	34	37
3	2-5/8	8.3	5	10	12	14	17	20	21	25	26	30	32	35	36
4	3-11/32	10.5	4	9	11	13	16	19	22	24	27	29	31	34	37
5	4-1/16	12.8		6	10	12	15	18	21	23	26	28	30	33	36
6	4-3/4	14.9			7	11	14	17	20	22	25	27	31	32	35
7	5-1/2	17.3			4	8	11	16	19	21	24	26	30	33	34
8	6-1/8	19.3				3	10	13	16	20	23	27	29	32	35
9	6-1/8	21.6					7	12	17	19	22	24	28	31	34
10	7-9/16	23.8						9	12	16	21	23	27	30	33
11	8-1/4	25.9						2	11	15	18	22	26	29	32
12	9	28.3							6	12	17	21	25	28	31
13	9-11/16	30.4								9	14	18	22	25	28
14	10-13/32	32.7									11	15	21	24	27
15	11-1/6	34.7									4	12	18	21	26
16	11-25/32	37.0										9	15	20	23
17	12-1/2	39.3											10	17	22
18	13-3/16	41.4											3	12	19
19	13-7/8	43.6												7	16
20	14-9/16	45.7													11

*Denotes off center pitch.

TABLE V (cont.)

F. 3/4 in. Tubes on 1 in. Square Pitch, 1-3/4 in. Minimum Radius

Shell Size, in.			6 in.*	8 in.	10 in.*	12 in.	14 in.	16 in.	18 in.*	20 in.	22 in.	24 in.	26 in.	28 in.*	30 in.	32 in.
Tube Circle, in.			5-13/16	7-13/16	9-15/16	11-13/16	13-1/4	15-1/4	17-1/4	19-1/8	21-1/8	23-1/8	25-1/8	27-1/8	29-1/8	
Tot sq ft/12'nom.			18.8	55.9	102	156.4	223.5	299.3	396.3	492	618	750	899	1055	1258	
Sq ft/ft st			1.6	4.7	8.6	13.3	19.4	25.9	34.6	43.2	54.6	66.8	80.5	95	110	
Sq ft in Bends			.4	1.3	3.0	5.2	8.5	12.8	19.3	25.9	35.8	47.8	62.2	78.6	97.5	
Tot No. U-Tubes			4	12	22	34	49	66	88	110	139	170	205	242	281	
Row No.	Radius in.	Bend in.	Number of U-Tubes													
1	1-3/4	5.5	4.4	7	8	11	13	15	16	19	21	23	25	26	29	
2	2-1/4	8.7		5	8	9	11	13	16	17	19	21	23	26	27	
3	3-3/4	11.8			6	9	11	13	14	17	19	21	23	26	27	
4	4-3/4	14.9				5	9	11	14	15	19	21	23	24	27	
5	5-3/4	18.1					5	9	12	15	17	19	21	24	25	
6	6-3/4	21.2						5	10	13	15	17	21	22	25	
7	7-3/4	24.4							6	9	13	17	19	22	23	
8	8-3/4	27.5								5	11	13	17	20	23	
9	9-3/4	30.7									5	11	15	18	21	
10	10-3/4	33.8										7	11	16	19	
11	11-3/4	37.0											7	12	15	
12	12-3/4	40.1												6	13	
13	13-3/4	43.2													7	
14	14-3/4	46.4														

*Denotes off center pitch.

TABLE V (cont.)

G. 1 in. U-Tubes on 1-1/4 in. Square Pitch, 2 in. Minimum Radius

Shell Size, in.			8 in.	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	22 in.	24 in.	26 in.	28 in.	30 in.	32 in.
Tube Circle,in.			7-13/16	9-15/16	11-13/16	13-1/4	15-1/4	17-1/4	19-1/8	21-1/8	23-1/8	25-1/8	27-1/8	29-1/8	
Total sq ft/12'nom.			49.6	74.3	129.3	171.3	247	321.5	408	517	636	764	886	1045	
Sq ft/ft st			4.2	6.3	11.0	14.7	21.2	27.8	35.6	45.6	56.6	68.6	79	94.2	
Sq ft in Bends			1.3	2	4.3	6.5	10.9	16	21.5	30.3	41.3	53.6	66.2	83.8	
Tot No. U-Tubes			8	12	21	28	41	53	68	87	108	131	151	180	
Row No.	Radius in.	Bend in.	Number of U-Tubes												
1	2	6.3	5	7	9	9	11	13	15	17	17	19	21	23	
2	3-1/4	10.2	3	5	7	9	11	13	13	15	17	19	21	23	
3	4-1/2	14.1			5	7	9	11	13	15	17	19	19	21	
4	5-3/4	18.1				3	7	9	11	13	15	17	19	21	
5	7	22.0					3	7	9	11	15	17	17	19	
6	8-1/4	25.9							7	9	13	15	17	19	
7	9-1/2	29.9								7	9	13	15	17	
8	10-3/4	33.8									5	9	13	15	
9	12	37.7										3	9	13	
10	13-1/4	41.6												9	

TABLE V (cont.)

H. 5/8 in. Tubes on 7/8 in. Triangular Pitch, Minimum Radius 1-1/4 in.
Center Tubes on Pitch 1-3/8 in. Radius, Tube Circle 8-32 in./HS-127

Shell Size, in.			8 in.*	10 in.	12 in.	14 in.	16 in.*	18 in.	20 in.	22 in.	24 in.*	26 in.	28 in.	30 in.	32 in.
Tube Circle, in.			7-13/16	9-15/16	11-13/16	13-1/4	15-1/4	17-1/4	19-1/8	21-1/8	23-1/8	25-1/8	27-1/8	29-1/8	31-1/8
Tot sq ft/12'nom.			92.3	164.4	244.1	311	422.4	549	689	861	1019	1215	1425	1636	1869
Sq ft/lin ft			7.9	14.1	20.9	26.8	36.4	48.5	60.7	75.5	91.3	109.2	128.5	149.5	171.5
Sq ft in Bends			2.0	4.4	7.6	10.5	16.4	24.6	33.2	46.8	61.4	80	101.7	125.6	153.6
Total No. U-Tubes			24	43	64	82	112	148	186	233	279	334	393	457	524
Row No.	Radius in.	Bend in.	Number of U-Tubes												
Cntr.	1-3/8	4.3	2	2	3	4	4	5	6	7	8	8	9	10	10
1	1-1/4	3.9	8	11	13	15	16	19	21	24	26	27	30	32	35
2	2	6.3	7	10	12	14	17	18	20	23	25	28	29	33	34
3	2-3/4	8.7	6	9	11	13	16	17	21	22	24	27	30	32	35
4	3-15/32	10.9	1	6	10	12	15	18	20	23	25	26	29	31	34
5	4-1/4	13.4		5	9	11	14	17	19	22	24	27	28	32	33
6	5	15.7			6	8	13	16	18	21	23	26	29	31	32
7	5-3/4	18.1				5	10	13	17	20	22	25	28	30	33
8	6-17/32	20.5					7	12	14	19	21	24	27	29	32
9	7-9/32	22.9						9	13	16	20	23	26	28	31
10	8-1/6	25.3						4	10	15	17	22	25	27	30
11	8-25/32	27.6							7	12	16	19	24	26	29
12	9-19/32	30.2								9	13	18	21	25	28
13	10-5/16	32.6									10	15	18	20	25
14	11-1/16	34.8									5	12	17	21	24
15	11-7/8	37.3										7	14	18	21
16	12-19/32	39.6											9	15	20
17	13-3/8	42												12	17
18	14-1/8	44.4												5	14
19	14-7/8	46.7													7
20															

*Denotes off center pitch.

TABLE VI

A. HORIZONTAL FREE DISTANCE

2 Tube Passes (Fixed Head)

Tube OD △ Pitch	3/8 in. 1/2 in.	5/8 in. 13/16 in.	3/4 in. 15/16 in.	1/2 in. IPS 1-1/32 in.	1 in. 1-1/4 in.	1-1/4 in. 19/16 in.
Shell Diam. in.	Horizontal Free Distances (Inches)					
6	2	1-1/2	1-1/2	2	2	2-1/2
8	2-1/2	2-1/2	2	2	2	2
10	3	3	2-1/2	2-1/2	2	2-1/2
12	3-1/2	3-1/2	3	3	3	3-1/2
14	4	3-1/2	4	3-1/2	3-1/2	3-1/2
16	4-1/2	4-1/2	3-1/2	4	3-1/2	4-1/2
18	5	4-1/2	4	4	4-1/2	4
20	5	5	4-1/2	4-1/2	4-1/2	4-1/2
22	5-1/2	5	5	4-1/2	5-1/2	5
24	6	6	6	5	5-1/2	6
26	6-1/2	6	6	5	5-1/2	6-1/2
28	7-1/2	6-1/2	6-1/2	5-1/2	6-1/2	6
30	7-1/2	7	6	6	6-1/2	7
32	8	7-1/2	6-1/2	6	6-1/2	6-1/2
34						
36						
39						
42						

TABLE VI (cont.)

B. ROWS OF TUBES IN HORIZONTAL FLOW

2 Pass (Fixed Head)

Tube OD Δ Pitch	3/8 in. 1/2 in.	5/8 in. 13/16 in.	3/4 in. 15/16 in.	1/2 in. IPS 1-1/32 in.	1 in. 1-1/4 in.	1-1/4 in. 1-9/16 in.
Shell Diam. in.	Rows of Tubes in Horizontal Flow					
6	21	13	11	9	7	3
8	29	17	15	13	11	9
10	37	23	19	17	15	11
12	45	27	23	21	17	13
14	51	31	25	23	19	15
16	59	35	31	27	23	17
18	67	41	35	31	25	21
20	75	45	39	35	29	23
22	83	51	43	39	31	25
24	91	55	47	43	35	27
26	99	61	51	47	39	29
28	107	65	55	51	41	33
30	115	71	61	55	45	35
32	123	75	65	59	49	39
34						
36						
39						
42						

TABLE VII

A. VERTICAL FREE DISTANCE

2 Tube Passes (Fixed Head)

Tube OD Δ Pitch	3/8 in. 1/2 in.	5/8 in. 13/16 in.	3/4 in. 15/16 in.	1/2 in. IPS 1-1/32 in.	1 in. 1-1/4 in.	1-1/4 in. 1-9/16 in.
Shell Size in.	Vertical Free Distance (Inches)					
6	2	3	2	1-1/2	2-1/2	3-1/2
8	3	2-1/2	2	2	2-1/2	3-1/2
10	3-1/2	4	3-1/2	3	3	3-1/2
12	4	3-1/2	3	3-1/2	3-1/2	3-1/2
14	5	4	4-1/2	3-1/2	5	5
16	4-1/2	5	4	4	5	5
18	5-1/2	5	4-1/2	4-1/2	5-1/2	4-1/2
20	6-1/2	6	5	5	6	6-1/2
22	6-1/2	6-1/2	6	5-1/2	6	6-1/2
24	7-1/2	6-1/2	6-1/2	6	6-1/2	6-1/2
26	8	7-1/2	6	6-1/2	6-1/2	6
28	8-1/2	7-1/2	7	7	7	8
30	9	8-1/2	7-1/2	6-1/2	7-1/2	8
32	9-1/2	9-1/2	7-1/2	7	7-1/2	8
34						
36						
39						
42						

TABLE VII (cont.)

B. ROWS OF TUBES IN VERTICAL FLOW

2 Tube Passes (Fixed Head)

Tube OD Δ Pitch	3/8 in. 1/2 in.	5/8 in. 13/16 in.	3/4 in. 15/16 in.	1/2 in. IPS 1-1/32 in.	1 in. 1-1/4 in.	1-1/4 in. 1-9/16 in.
Shell Size in.	Rows of Tubes in Vertical Flow					
6	12	6	6	6	4	2
8	16	10	8	8	6	4
10	22	12	10	10	8	6
12	26	16	14	12	10	8
14	28	18	14	14	10	8
16	32	20	18	16	12	10
18	38	24	20	18	14	12
20	42	26	22	20	16	12
22	46	28	24	22	18	14
24	52	32	26	24	20	16
26	56	34	30	26	22	18
28	60	38	32	28	24	18
30	66	40	34	32	26	20
32	70	42	36	34	28	22
34						
36						
39						
42						

TABLE VIII

A. VERTICAL FREE DISTANCE

2 Pass Tubes (Full Floating Head)

Tube OD Pitch	5/8 in. 13/16 in. △	5/8 in. 7/8 in. □	3/4 in. 15/16 in. △	3/4 in. 1 in. △	3/4 in. 1 in. □	3/4 in. 1 in. ◇	1 in. 1-1/4 in. △	1 in. 1-1/4 in. □	1 in. 1-1/4 in. ◇
Shell Size, in.	Vertical Free Distance (Inches)								
6	2-1/4								
8	4		2-1/2	4	3-1/2	4	3	4	3
10	4		3-1/2	4	4	4-1/2	3	4	4
12	4-1/2		4	4-1/2	4-1/2	5-1/2	5	4	4-1/2
14	5		4	4-1/2	4-1/2	6	4-1/2	5-1/2	4-1/2
16	6		5	5	5	6	5	5-1/2	5
18	6		5-1/2	6	5-1/2	7	5-1/2	5-1/2	6
20	7		5-1/2	6-1/2	6	7-1/2	5-1/2	5-1/2	6-1/2
22	6-1/2		6	7-1/2	6-1/2	8-1/2	6	5-1/2	7
24	7-1/2		6-1/2	7-1/2	7	8-1/2	6-1/2	7-1/2	8
26	8-1/2		7-1/2	9	9	11	8	7-1/2	8-1/2
28	9		8	10	9-1/2	10-1/2	8-1/2	7-1/2	10
30	9		8-1/2	9-1/2	10	11-1/2	9	7-1/2	9
32	10		8-1/2	10-1/2	10-1/2	13	9-1/2	9-1/2	10
34			10		10	13	10	10	11
36			11		10-1/2	13-1/2	10-1/2	10	11-1/2
39			11-1/2		12	15	11-1/2	11	13
42			12		12	16	11	10	13
45									

TABLE VIII (cont.)

B. ROW OF TUBES IN VERTICAL FLOW

2 Pass (Full Floating Head)

Tube OD Pitch	5/8 in. 13/16 in. △	3/4 in. 15/16 in. △	3/4 in. 1 in. △	3/4 in. 1 in. □	3/4 in. 1 in. ◇	1 in. 1-1/4 in. △	1 in. 1-1/4 in. □	1 in. 1-1/4 in. ◇
Shell Size, in.	Rows of Tubes in Vertical Flow							
8	8	6	6	6	8	6	4	6
10	12	10	10	8	11	8	6	8
12	14	12	12	10	13	8	8	10
14	16	14	14	12	16	10	8	12
16	18	16	16	14	18	12	10	14
18	22	18	18	16	21	14	12	16
20	24	20	20	18	24	16	14	19
22	28	24	22	20	26	18	16	21
24	30	26	24	22	30	20	16	23
26	32	28	26	22	32	20	18	25
28	36	30	28	24	35	22	20	26
30	38	32	30	26	37	24	22	30
32	40	36	32	28	39	26	22	32
34		38		32	44	28	24	35
36		40		34	46	30	26	37
39		44		36	51	32	28	39
42		48		40	54	36	32	44

TABLE IX

A. HORIZONTAL FREE DISTANCE

2 Pass Tubes (Full Floating Head)

Tube OD Pitch	5/8 in. 13/16 in. △	5/8 in. 7/8 in. □	3/4 in. 15/16 in. △	3/4 in. 1 in. △	3/4 in. 1 in. □	3/4 in. 1 in. ◇	1 in. 1-1/4 in. △	1 in. 1-1/4 in. □	1 in. 1-1/4 in. ◇
Shell Size, in.	Horizontal Free Distance (inches)								
8	3-1/2		3	3	3-1/2	4	3	4	3
10	3-1/2		3-1/2	3-1/2	3-1/2	4-1/2	3	4	4
12	4		4	4	4	5-1/2	3	4	4-1/2
14	4		3-1/2	4-1/2	4-1/2	6	3-1/2	3-1/2	4-1/2
16	4-1/2		4	5	5	6	4-1/2	5-1/2	5
18	5		5	5-1/2	6	7	4-1/2	5-1/2	6
20	5-1/2		5	6	6	7-1/2	5-1/2	6-1/2	6-1/2
22	5-1/2		5-1/2	6-1/2	6-1/2	8-1/2	5-1/2	6-1/2	7
24	6		6	7	7	8-1/2	6-1/2	6-1/2	8
26	7		6-1/2	7-1/2	7-1/2	11	6-1/2	6-1/2	8-1/2
28	7		7	8	8-1/2	10-1/2	7-1/2	7-1/2	10
30	8		7-1/2	8-1/2	9	11-1/2	7-1/2	7-1/2	9
32	9		7-1/2	9-1/2	9-1/2	13	7-1/2	7-1/2	10
34			8-1/2		10	13	9	9	11
36			9		10-1/2	13-1/2	9	9	11-1/2
39			10		11-1/2	15	10	10	13
42			10-1/2		12	16	10	10	13
45									

TABLE IX (cont.)

B. FULL FLOATING HEAD UNITS

Rows of Tubes in Horizontal Flow

Tube OD Pitch	5/8 in. 13/16 in. △	3/4 in. 15/16 in. △	3/4 in. 1 in. △	3/4 in. 1 in. □	3/4 in. 1 in. ◇	1 in. 1-1/4 in. △	1 in. 1-1/4 in. □	1 in. 1-1/4 in. △
Shell Size in.	Rows of Tube in Horizontal Flow							
6	11							
8	15	13	13	6	8	9	4	6
10	21	17	17	9	11	13	6	8
12	25	19	21	11	13	17	8	10
14	29	23	23	12	16	19	10	12
16	33	29	27	14	18	21	10	14
18	39	33	31	15	21	25	12	16
20	43	37	35	18	24	27	13	19
22	49	41	39	20	26	31	15	21
24	53	45	43	22	30	33	17	23
26	57	53	47	24	32	37	19	25
28	63	53	51	25	35	39	20	26
30	67	57	55	27	37	43	22	30
32	71	61	59	29	39	47	24	32
34		67		32	44	49	25	35
36		71		34	46	53	27	37
39		77		37	51	57	29	39
42		83		40	54	63	32	42

TABLE X*

A. OVER-ALL COEFFICIENTS FOR HEAT EXCHANGERS IN PETROLEUM SERVICE

Service of Exchanger	Fluids: Tubes	Fluids: Shell	Velocity in Tubes, ft/sec	Δt_m, °F	Over-all Coef. U, Btu/hr (sq ft)(°F)
Stabilizer reflux condensers	Water	Condensing Vapors + Residual Gas	3.0	13.5	94
			5.0	22	145
		108°-118°A.P.I.	0.3-0.6		55-67**
			0.7		98-125**
Partial condensers	39°A.P.I. crude	58°A.P.I. gasoline	2.4	147	24
	55°A.P.I. crude	62°A.P.I. naptha	4.4	80	37
	55°A.P.I. crude	62°A.P.I. naptha	6.8	90	48
Stabilizer reboiler	Steam	58°A.P.I. naptha		33.5	42
Absorber reboiler	Steam	37°A.P.I. oil		41.4	45
Stabilizer reboiler	Steam	67-74°A.P.I.			43-183**
Oil preheater	42°A.P.I. oil	Steam	1.4	32	108
Exchangers	60°A.P.I.	58°A.P.I.	1.4	65	74
	63°A.P.I	57°A.P.I	4.6	69	139
	70-82°A.P.I.	67-74°A.P.I.	0.3-0.7		18-37**
	70-82°A.P.I.	67-74°A.P.I	0.3-0.7		35-45**
	43°A.P.I.	37°A.P.I	1.6	59	33
	39°A.P.I. crude	13°A.P.I. residue	3.9	262	19
Coolers	Water	57°A.P.I		40	52
	Water	44°A.P.I		97	40
	Water	67-74°A.P.I	0.2		20**
	Water	67-74°A.P.I	0.4-0.7		51-53**

*Adapted from Chemical Engineers' Handbook, edited by J. H. Perry, with the permission of McGraw-Hill Book Company, Inc., The A.I. Ch. E and the A.S.M.E.

**McGiffin Trans. Am. Inst. Chem. Engrs., 38, 761 (1942). All other data in Table X are from Higgins, "Heat Transfer," P. 56, a special publication of the A.S.M.E. (New York, 1936).

TABLE X (cont.)

B. A RANGE OF VALUES OF MISCELLANEOUS OVER-ALL COEFFICIENTS*

U, Expressed in Btu/(hr)(sq ft)(^{o}F) as Found in Practice
Under Special Conditions Higher or Lower Values May Be Realized

Type of Heat Exchanger	State of Controlling Resistance: Free Convection, U	State of Controlling Resistance: Forced-Convection, U	Typical Fluid	Typical Apparatus
Liquid to liquid	25-60	150-300	Water	Liquid-to-liquid heat exchangers
Liquid to liquid	5-10	20-50	Oil	
Liquid to gas (atm. pressure)	1-3	2-10		Hot-water radiators
Liquid to boiling liquid	20-60	50-150	Water	Brine coolers
Liquid to boiling liquid	5-20	25-60	Oil	
Gas (atm. pressure) to liquid	1-3	2-10		Air coolers, economizers
Gas (atm. pressure) to gas	0.6-2	2-6		Steam superheaters
Gas (atm. pressure) to boiling liquid	1-3	2-10		Steam boilers
Condensing vapor to liquid	50-200	150-800	Steam-water	Liquid heaters and condensers
Condensing vapor to liquid	10-30	20-60	Steam-oil	
Condensing vapor to liquid	40-80	60-300	Organic vapor-water	
Condensing vapor to liquid		15-300	Steam-gas mixture	
Condensing vapor to gas (atm. pres.)	1-3	6-16		Steam pipes in air Air heaters
Condensing vapor to boiling liquid	40-100			Scale-forming evaporators
Condensing vapor to boiling liquid	300-800		Steam-water	
Condensing vapor to boiling liquid	50-150		Steam-oil	
Condensing vapor to boiling liquid		50-400	Steam-organic liquid	Steam-jacketed tubes

*Modified from Lucke, "Engineering Thermodynamics," p.550, McGraw-Hill, New York 1912.

TABLE X (cont.)

C. COILS IMMERSED IN LIQUIDS. OVER-ALL COEFFICIENTS

U, Expressed as Btu/(hr)(sq ft)(^{o}F)

Substance Inside Coil	Substance Outside Coil	Coil Material	Agitation	U	Reference
Steam	Water	Lead	Agitated	70	1
Steam	Sugar and molasses solution	Copper	None	50-240	2
Steam	Boiling aqueous soln.			600	3
Cold water	Dilute organic dye intermediate	Lead	Turbo-agitator at 95 rpm	300	3
Cold water	Warm water	Wrought iron	Air bubbled into water surrounding coil	150-300	4
Cold water	Hot water	Lead	0.40 rpm, paddle stirrer	90-360	5
Brine	Amino acids		30 rpm	100	3
Cold water	25% oleum at 60^{o}C	Wrought iron	Agitated	20	6
Water	Aqueous solution	Lead	500 rpm, sleeve peopeller	250	1
Water	8% NaOH		22 rpm	155	3
Steam	Fatty Acid	Copper (pancake)	None	96-100	7
Milk	Water		Agitation	300	8
Cold water	Hot water	Copper	None	105-180	9
60^{o}F water	50% aqueous sugar soln.	Lead	Mild	50-60	10
Steam and hydrogen at 1500 lb/sq in.	60^{o}F water	Steel		100-165	10

Note: Chilton, Drew, and Jebens, Ind. Eng. Chem., 36, 510 (1944) give film coefficients for heating and cooling agitated fluids using a coil in a jacketed vessel.

References:
1. Read, private communication.
2. Stose and Wittemore, Thesis, Mass. Inst. Tech., 1922.
3. Chambers and Steves, private communication.
4. Chilton and Colburn, private communication.
5. Pierce and Terry, Chem., and Met. Eng., 30, 872 (1924).
6. Boertlein, private communication.
7. Mills and Daniels, Ind. Eng. Chem., 26, 248-250 (1934).
8. Feldmeier, Adv. paper, Am. Soc. Mech. Engrs. Meeting Dec. 4, 1934; published in "Heat Transfer," 69-74, ASME, New York, 1936.
9. Storrow, J. Soc. Chem. Ind., 64, 322 (1945).
10. Private communication.

TABLE X (cont.)

D. MISCELLANEOUS: SPECIAL EQUIPMENT AND MATERIALS

Type of Equipment	Hot Material	Cold Material	U	Remarks	Reference
High pressure boiler	Molten salt	Boiling water	100-150		1
Tubular exchanger	Molten salt	Oil	52-80		1
Steam superheater	Molten salt	Steam	70		1
Air heater	Molten salt	Air	6		1
Catalyst case	Gas	Molten salt	6	Fins on outside of tube	1
Double-pipe Karbate exchanger	Water	Water	300-500		2
Karbate trombone cooler	20°Be HCl	Water	300	Water Γ_1 = 1750	3
Karbate tube reboiler	Steam	20% HCl	136	Vertical thermosiphon reboiler	10
	Steam	35% HCl	472-575		10
Double-pipe pyrex glass ex. using heat exchanger tubing	Air-water vapor	Water	25-75	Cooling water in annulus	4
	Water	Water	80-110		4
	Condensing steam	Water	100-125		4
Glass trombone cooler	50% sugar soln.	60°F water	50-60	Sugar solution inside pipe	9
Glass pipe in trough	20°Be HCl	Water	25		3
Rotator	Water	Water	520-1120	Rotor velocity = 300-1900 rpm	5
Pebble heater	Solid pebbles	Air	4	Heating gases to 1900°F using 1/2-in. pebbles	6
		Methane	9		6
		Hydrogen	22		6
Long-tube vertical evaporator	Condensing steam	Water	300-1200		7
Falling-film condenser	Condensing steam	Water	574-2300	Water Γ_2 = 400-21,000 inside tubes	8
Stainless-steel conveyor belt	Molten TNT	50°F air	5-7	Air blowing under and over belt	9
Partial-condenser	Hydrocarbons and chlorinated hydrocarbons	Boiling propane	55-76	Refrigerated condenser	10
Shell and tube reboiler	Hot water	Hydrocarbons	42-88	Hot water in tubes	10
Reboiler	Steam	Chlorinated hydrocarbons	67	Clean reboiler, Δt = 12°F	10
			20	Same reboiler after several months service, Δt = 96°F	

U = Btu/(hr)(sq ft)(°F).
Γ_1 = lb/(hr)(ft) of pipe length for each side of pipe.
Γ_2 = lb/(hr)(ft) of periphery.

1 Newton and Shimp, Trans. Am. Inst. Chem. Engrs., 41, 197 (1945).
2 Werking, Trans. Am. Inst. Chem. Engrs., 35, 489 (1939).
3 Lippman, Chem. and Met. Eng., 52, No. 3, 112 (1945).
4 Thompson and Foust, Chem. and Met. Eng., 47, 410 (1940).
5 Houlton, Ind. Eng. Chem., 36, 522-528 (1944).
6 Norton, Chem. and Met. Eng. 53, No. 7, 116 (1946).
7 Cessna, Lientz, and Badger, Trans. Am. Inst. Chem. Engrs., 36, 759 (1940).
8 McAdams, Drew, and Bays, Trans. Am. Soc. Mech. Engrs., 62, 627 (1940).
9 Private communication.
10 Breidenbach and O'Connell, Trans. Am. Inst. Chem. Engrs., 42, 761 (1946).

TABLE X (cont.)

E. JACKETED VESSELS. OVER-ALL COEFFICIENTS
U, Expresses in Btu/(hr)(sq ft)(°F)

Fluid Inside Jacket	Fluid in Vessel	Wall Material	Agitation	U	Reference
Steam	Water	Enameled C.I.*	0-400 rpm	96-120	1
Steam	Milk	Enameled C.I.	None	200	2
Steam	Milk	Enameled C.I.	Stirring	300	2
Steam	Milk boiling	Enameled C.I.	None	500	2
Steam	Milk	Enameled C.I.	200 rpm	86	1
Steam	Fruit slurry	Enameled C.I.	None	33-90	1
Steam	Fruit slurry	Enameled C.I.	Stirring	154	1
Steam	Water	C.I. and loose lead lining	Agitated	4-9	3
Steam	Water	C.I. and loose lead lining	None	3	3
Steam	Boiling SO_2	Steel	None	60	3
Steam	Boiling water	Steel	None	187	3
Hot water	Warm water	Enameled C.I.	None	70	1
Cold water	Cold water	Enameled C.I.	None	43	1
Ice water	Cold water	Stoneware	Agitated	7	3
Ice water	Cold water	Stoneware	None	5	3
Brine, low velocity	Nitration slurry		35-58 rpm	32-60	4
Water	Sodium, alcoholate solution	"Frederking" (cast-in-coil)	Agitated, baffled	80	4
Steam	Evaporating water	Copper		381	5
Steam	Evaporating water	Enamelware		36.7	5
Steam	Water	Copper	None	148	6
Steam	Water	Copper	Simple stirring	244	6
Steam	Boiling water	Copper	None	250	6
Steam	Paraffin wax	Copper	None	27.4	7
Steam	Paraffin wax	Cast iron	Scraper	107	7
Water	Paraffin wax	Copper	None	24.4	7
Water	Paraffin wax	Cast iron	Scraper	72.3	7
Steam	Solution	Cast iron	Double scrapers	175-210	8
Steam	Slurry	Cast iron	Double scrapers	160-175	8
Steam	Paste	Cast iron	Double scrapers	125-140	8
Steam	Lumpy mass	Cast iron	Double scrapers	75-96	8
Steam	Powder (5% moisture)	Cast iron	Double scrapers	41-51	8

*C.I. = cast iron.

References:

1 Poste, Ind. Eng. Chem., 16 469 (1924).
2 Bowen, Agr. Eng., 11, 27 (1930).
3 Read, private communication.
4 Chambers and Steves, private communication.
5 Robson, Australian Chem. Inst. J. and Proc., 3, 47-54 (1936).
6 Chemical Engineering Charts No. 4, Ind. Chemist, 82, 374 (1931).
7 Huggins, Ind. Eng. Chem., 23, 749-753 (1931).
8 Laughlin, Trans. Am. Inst. Chem. Engrs., 36, 345 (1940).

TABLE X (cont.)

F. VALUES OF U FOR AMMONIA CONDENSERS

Type of Condenser	Water Rate	Btu/(hr)(sq ft)(°F over-all Δt)			
		Δt_m = 1.5°F	Δt_m = 3.5°F	Δt_m = 7°F	Δt_m = 2-6°F
Vertical tube and shell	Γ_v = 400	220	170	150	
	800	275	225	215	
	1200	310	270	260	
	1600	350	315	300	
	2000		390	340	
	2400		430	370	
Horizontal drip	Γ_h = 400				250
	800				330
	1200				400
Double pipe	V = 4	350	270	230	
	6	410	320	280	
	8	470	390	350	

Γ_v = lb of water/hr/ft of periphery.

Γ_h = lb of water/hr/ft of tube length for each side of tube.

V = ft/sec.

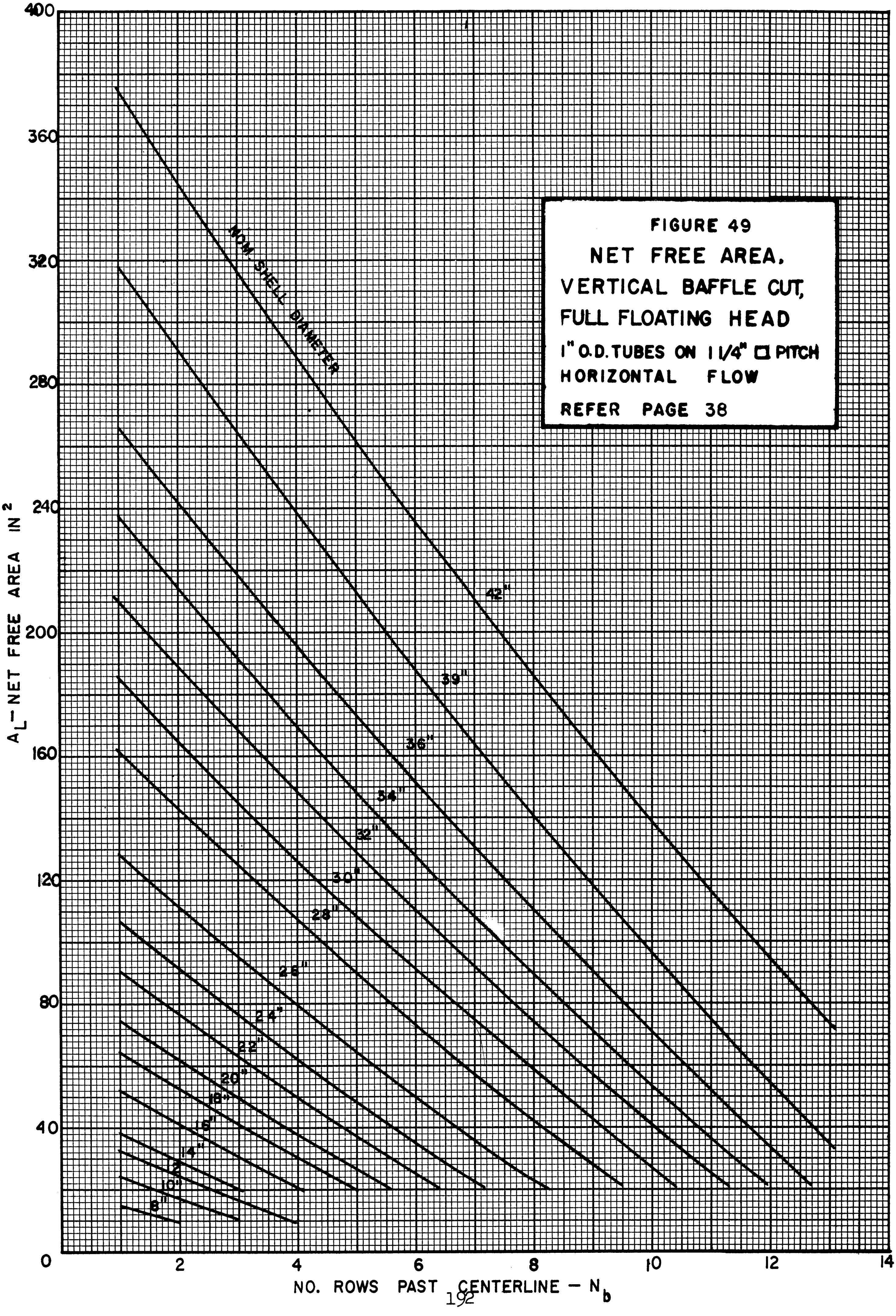


FIGURE 49
NET FREE AREA,
VERTICAL BAFFLE CUT,
FULL FLOATING HEAD
1" O.D. TUBES ON 1 1/4" □ PITCH
HORIZONTAL FLOW
REFER PAGE 38
NOM. SHELL DIAMETER
42"
39"
36"
34"
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12"
10"
8"
A_L - NET FREE AREA IN^2
NO. ROWS PAST CENTERLINE - N_b
0
40
80
120
160
200
240
280
320
360
400
2
4
6
8
10
12
14

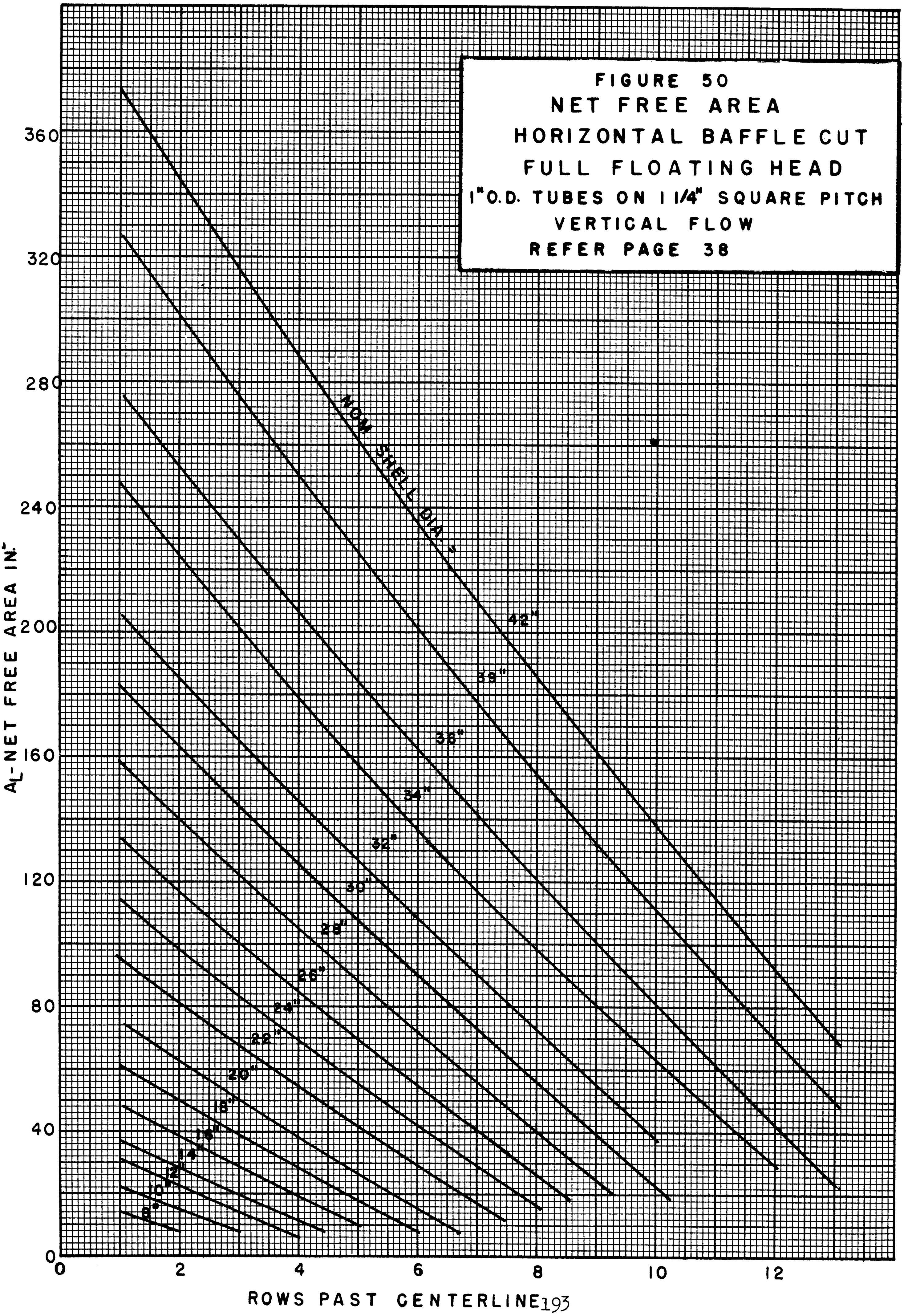


FIGURE 50
NET FREE AREA
HORIZONTAL BAFFLE CUT
FULL FLOATING HEAD
1" O.D. TUBES ON 1 1/4" SQUARE PITCH
VERTICAL FLOW
REFER PAGE 38
NOM SHELL DIA."
42"
39"
36"
34"
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12"
10"
8"
A_L - NET FREE AREA IN.^{2}
360
320
280
240
200
160
120
80
40
0
0
2
4
6
8
10
12
ROWS PAST CENTERLINE

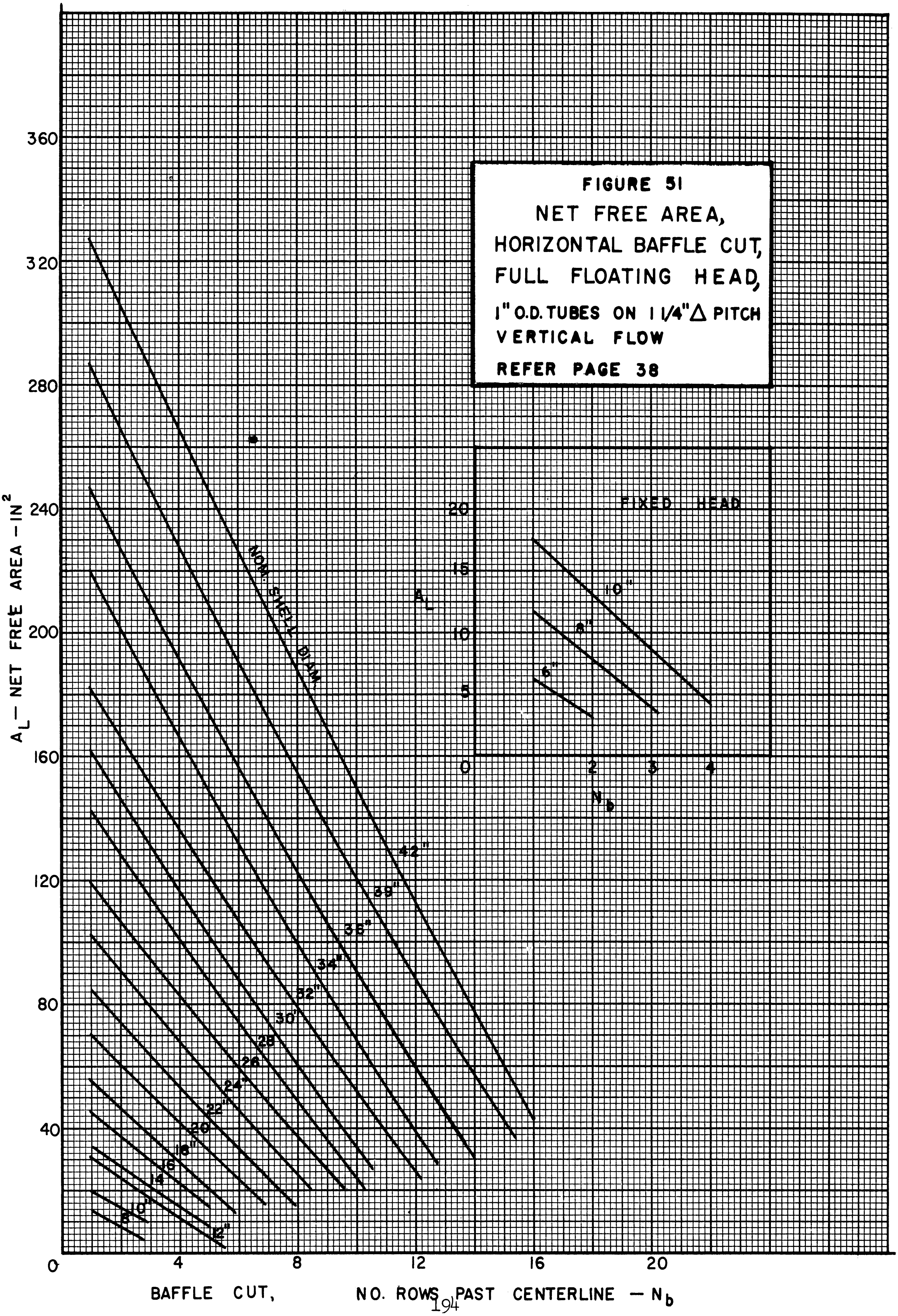


FIGURE 51
NET FREE AREA,
HORIZONTAL BAFFLE CUT,
FULL FLOATING HEAD,
1" O.D. TUBES ON 1 1/4" Δ PITCH
VERTICAL FLOW
REFER PAGE 38
NOM. SHELL DIAM.
42"
39"
36"
34"
32"
30"
28
26
24"
22"
20
18"
16
14
12"
10"
8"
FIXED HEAD
10"
8"
6"
A_L
N_b
A_L – NET FREE AREA – IN^2
BAFFLE CUT, NO. ROWS PAST CENTERLINE – N_b
0
4
8
12
16
20
40
80
120
160
200
240
280
320
360
0
2
3
4
5
10
15
20

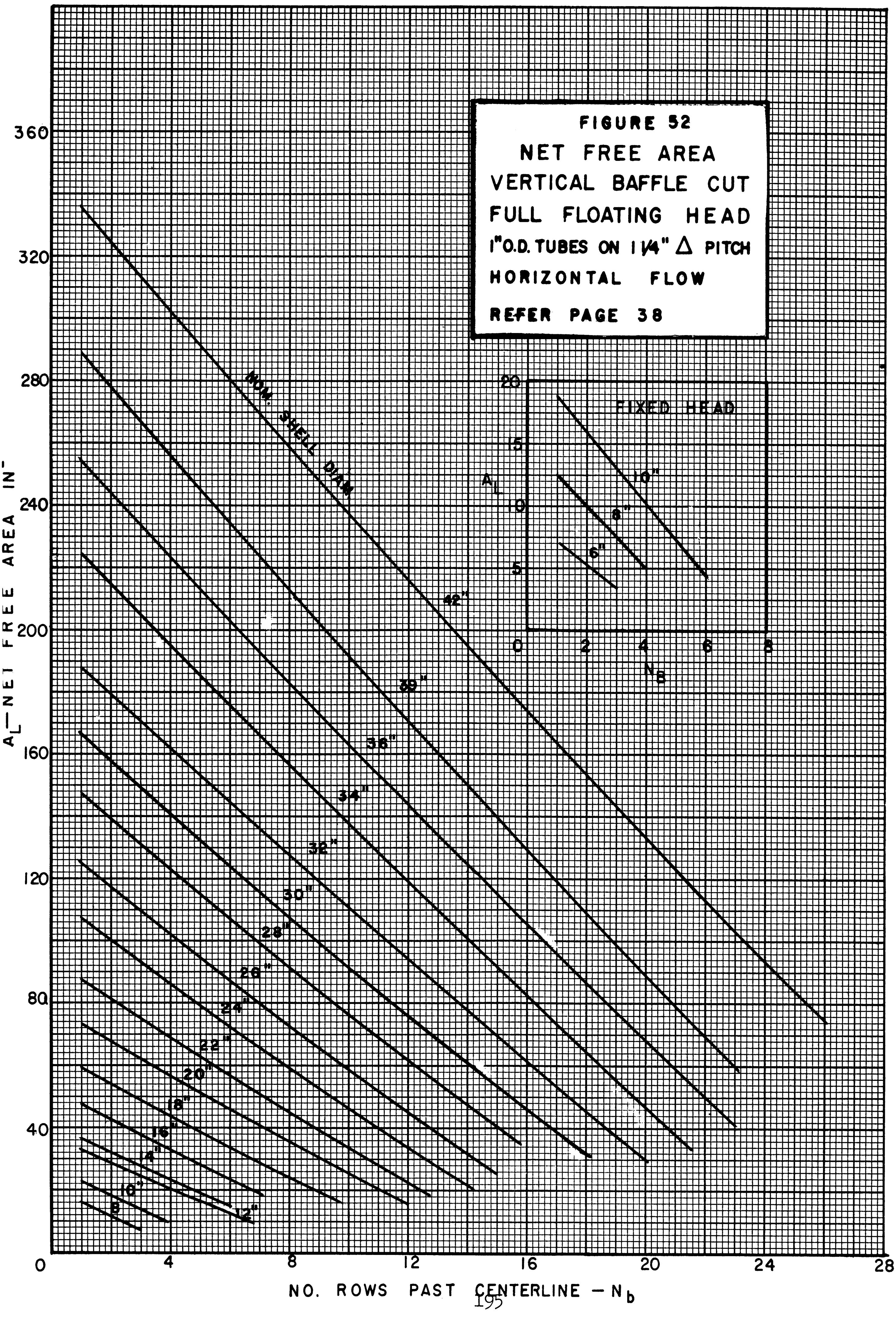


FIGURE 52
NET FREE AREA
VERTICAL BAFFLE CUT
FULL FLOATING HEAD
1" O.D. TUBES ON 1 1/4" Δ PITCH
HORIZONTAL FLOW
REFER PAGE 38
NOM. SHELL DIAM.
42"
39"
36"
34"
32"
30"
28"
26"
24"
22"
20"
18"
16"
14"
12"
10"
8
FIXED HEAD
10"
8"
6"
A_L
N_B
0
2
4
6
8
5
10
15
20
A_L — NET FREE AREA IN²
360
320
280
240
200
160
120
80
40
0
4
8
12
16
20
24
28
NO. ROWS PAST CENTERLINE — N_b

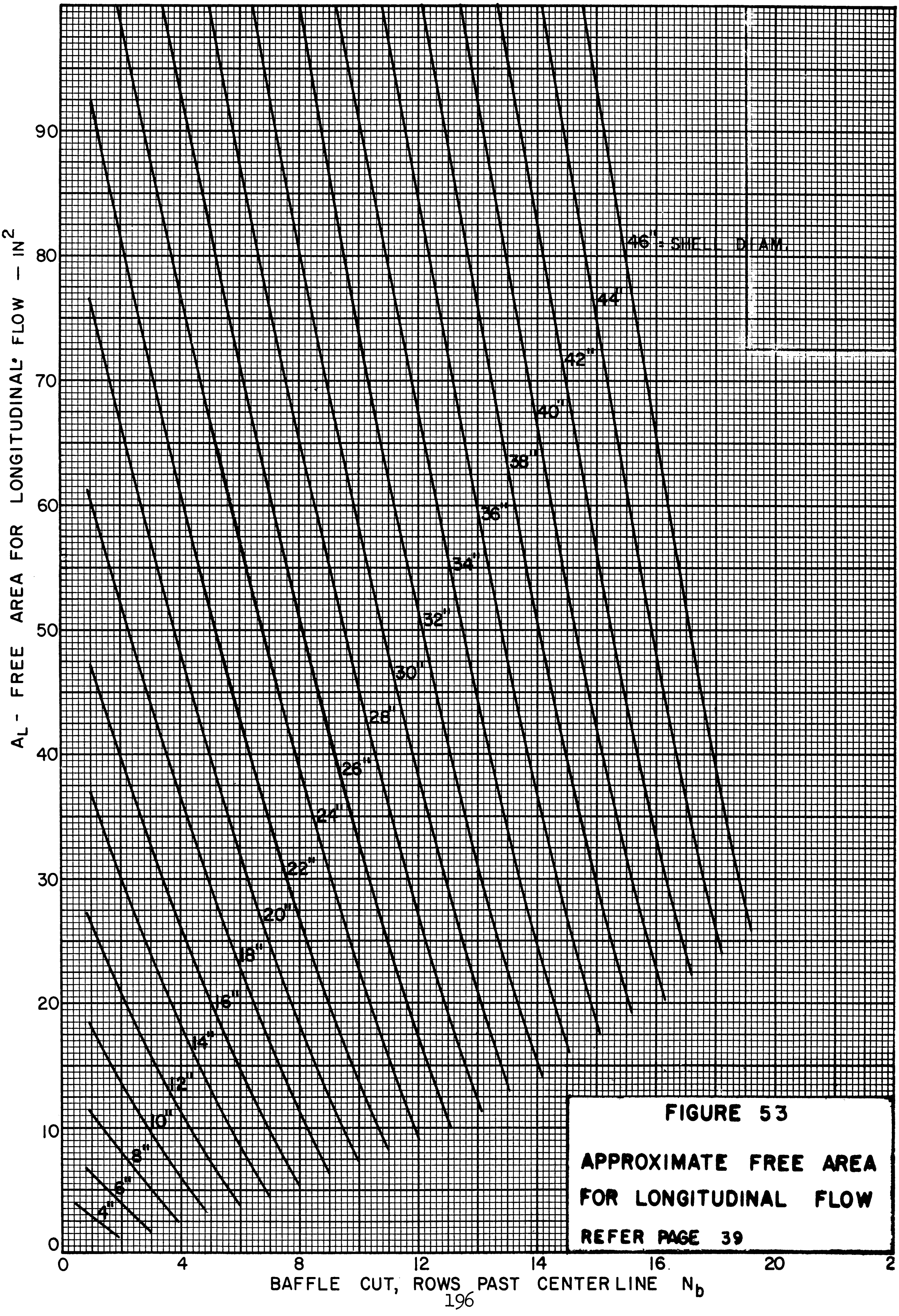


FIGURE 53

APPROXIMATE FREE AREA FOR LONGITUDINAL FLOW

REFER PAGE 39

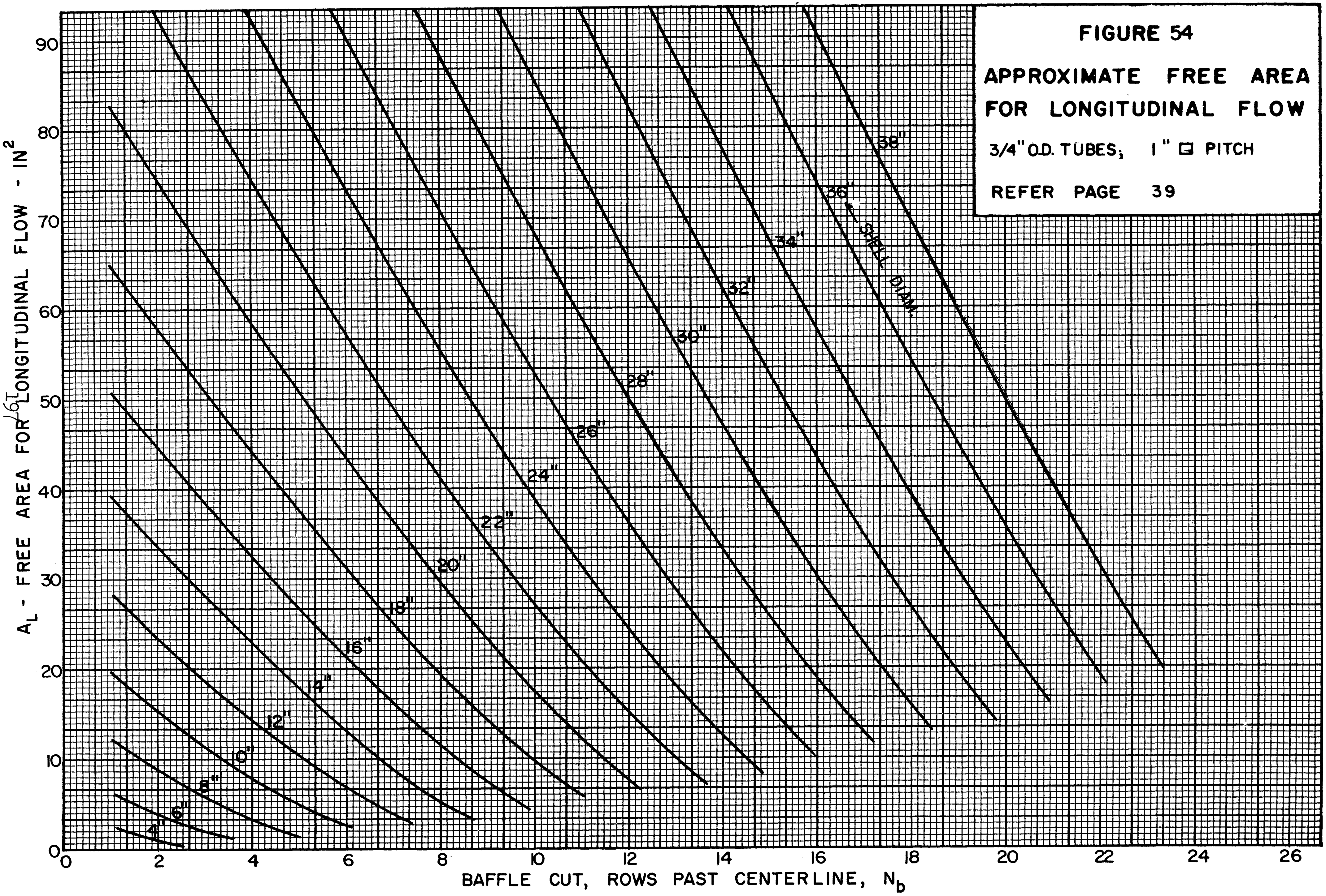


FIGURE 54

APPROXIMATE FREE AREA FOR LONGITUDINAL FLOW

3/4" O.D. TUBES; 1" ◻ PITCH

REFER PAGE 39

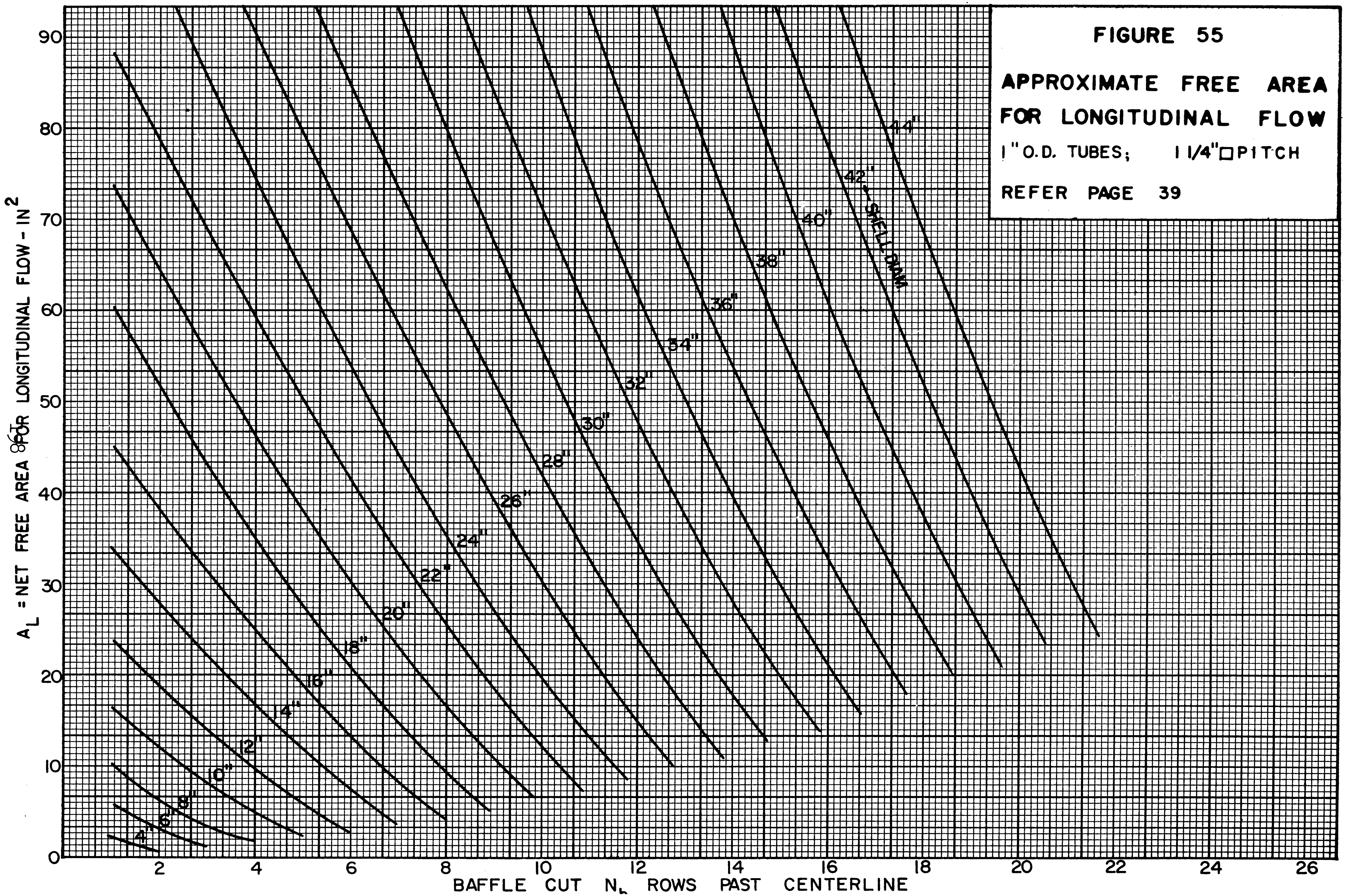


FIGURE 55

APPROXIMATE FREE AREA FOR LONGITUDINAL FLOW

1" O.D. TUBES; 1 1/4" □ PITCH

REFER PAGE 39

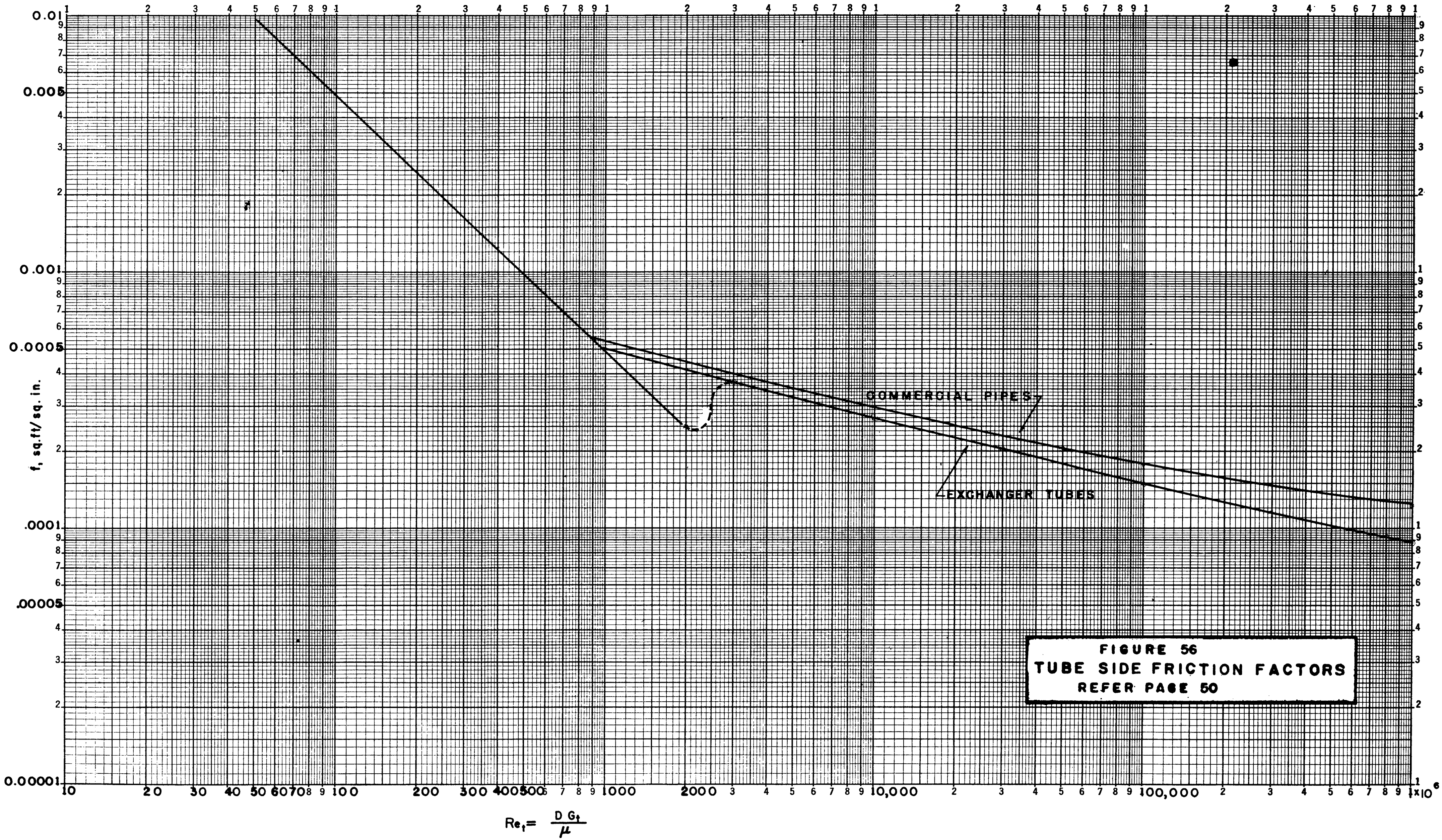
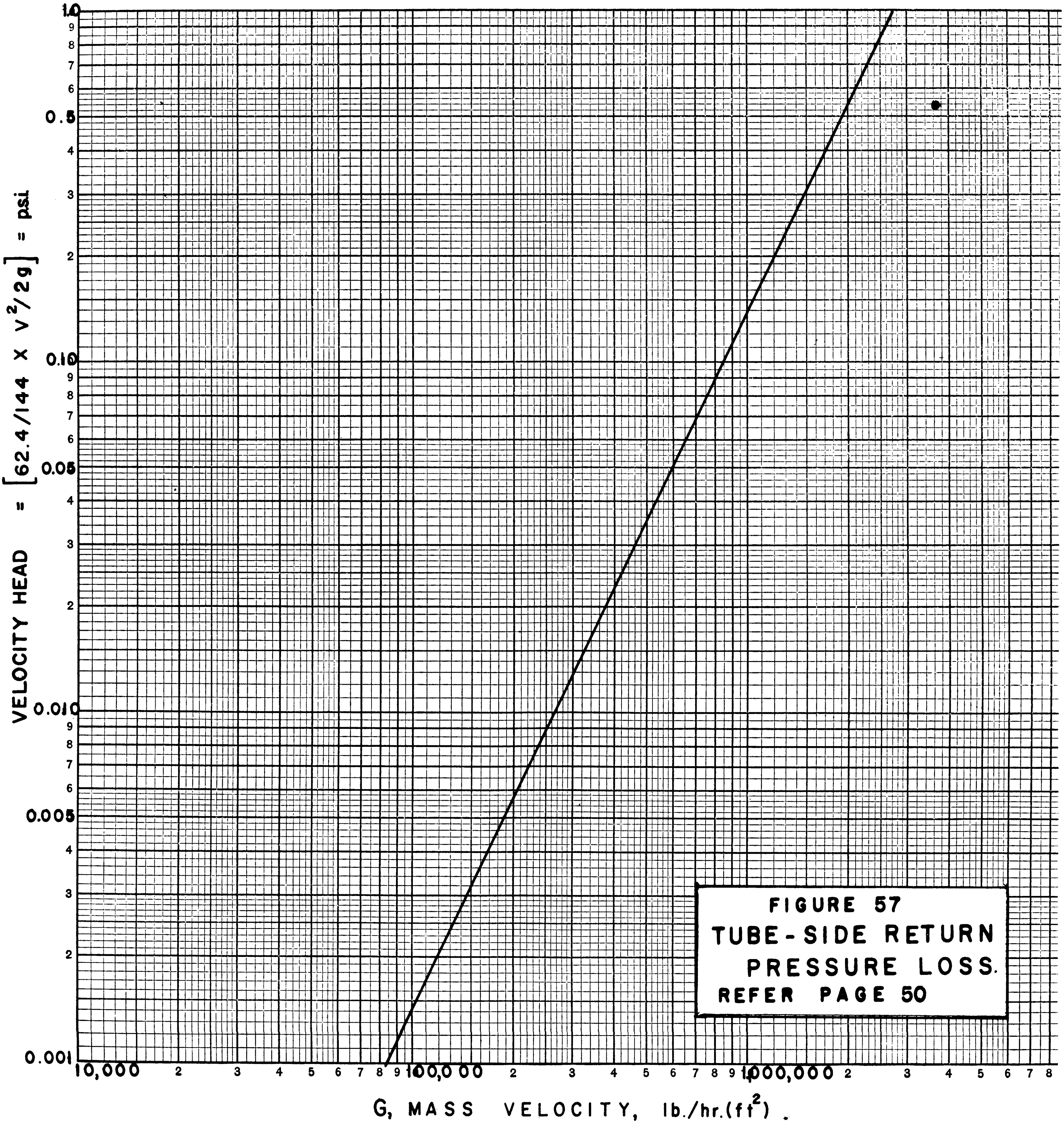


FIGURE 56
TUBE SIDE FRICTION FACTORS
REFER PAGE 50


VELOCITY HEAD = [62.4/144 X V²/2g] = p.s.i.
1.0
0.5
0.10
0.05
0.010
0.005
0.001
10,000
100,000
1,000,000
G, MASS VELOCITY, lb./hr.(ft²)
FIGURE 57
TUBE-SIDE RETURN PRESSURE LOSS.
REFER PAGE 50

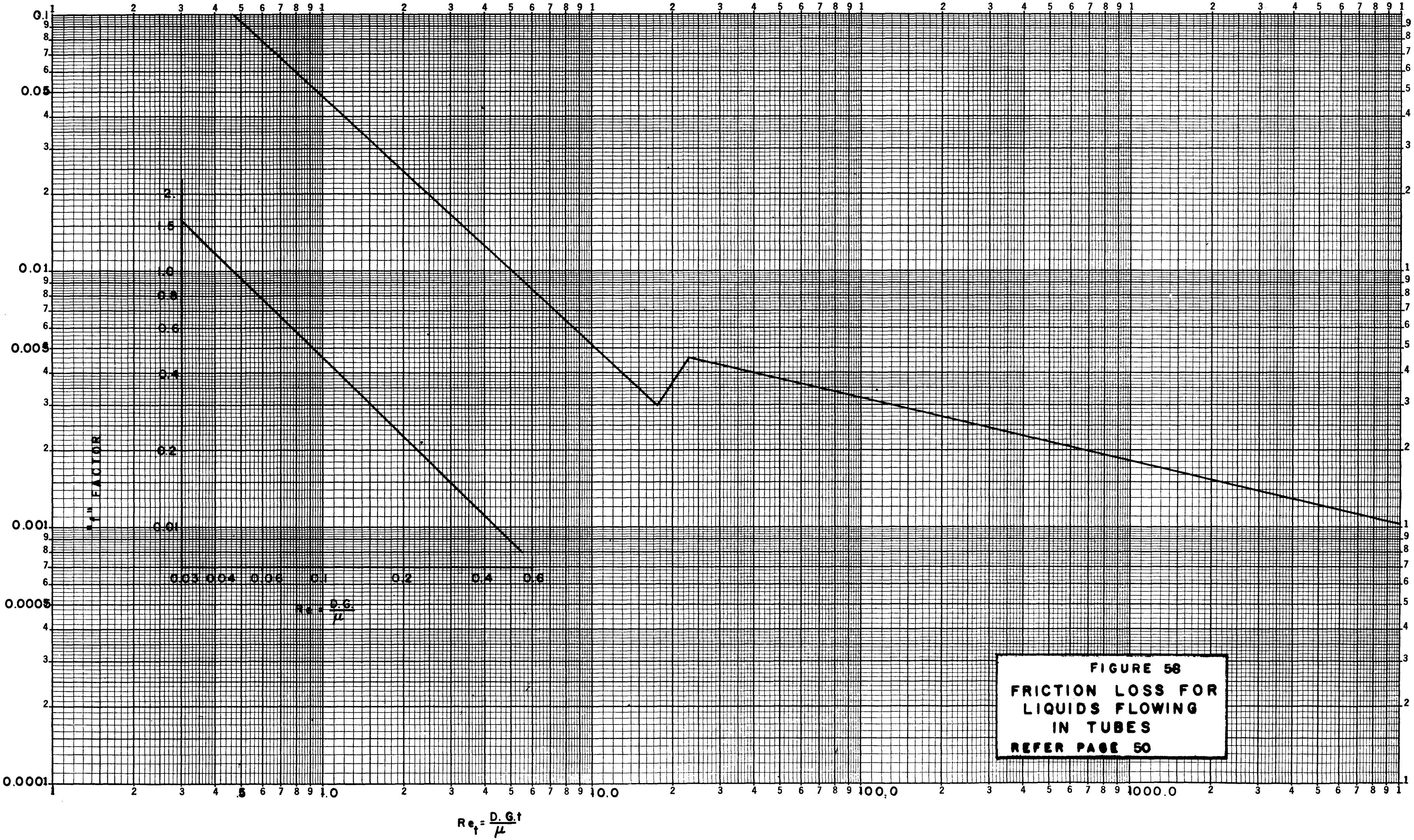
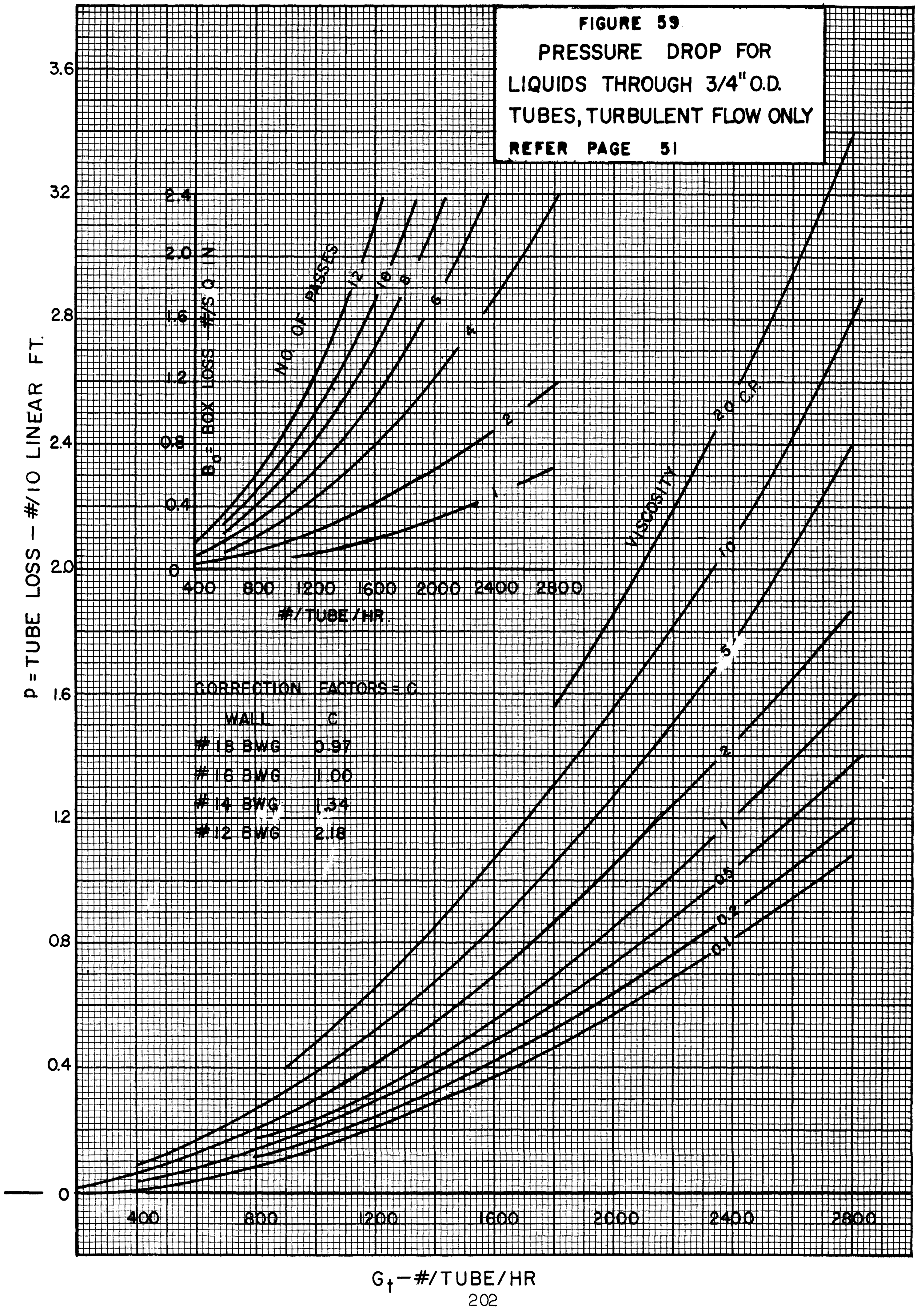


FIGURE 58
FRICTION LOSS FOR LIQUIDS FLOWING IN TUBES
REFER PAGE 50

FIGURE 59
PRESSURE DROP FOR LIQUIDS THROUGH 3/4" O.D. TUBES, TURBULENT FLOW ONLY
REFER PAGE 51



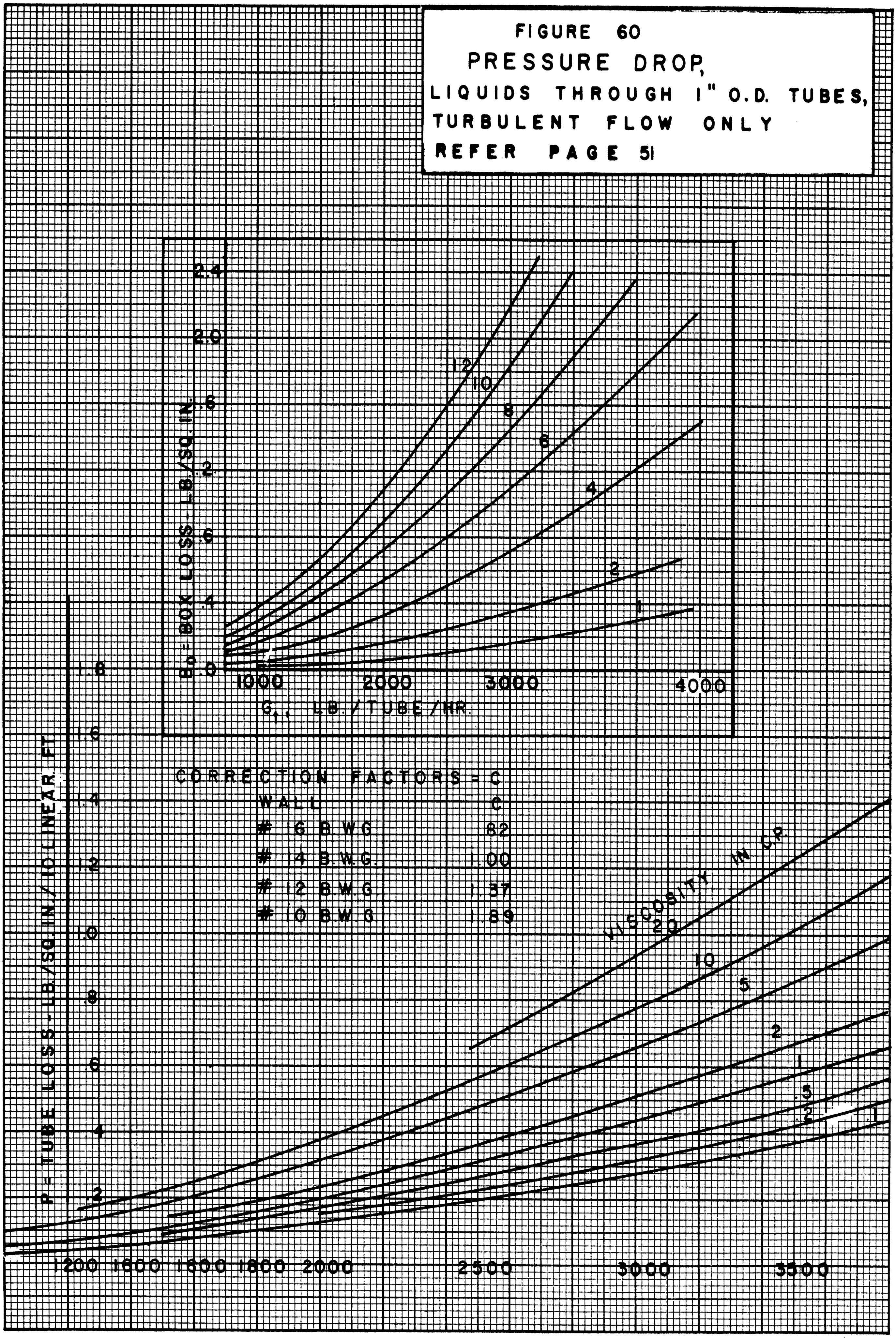


FIGURE 60
PRESSURE DROP,
LIQUIDS THROUGH 1" O.D. TUBES,
TURBULENT FLOW ONLY
REFER PAGE 51

G_t = LB./TUBE HR.

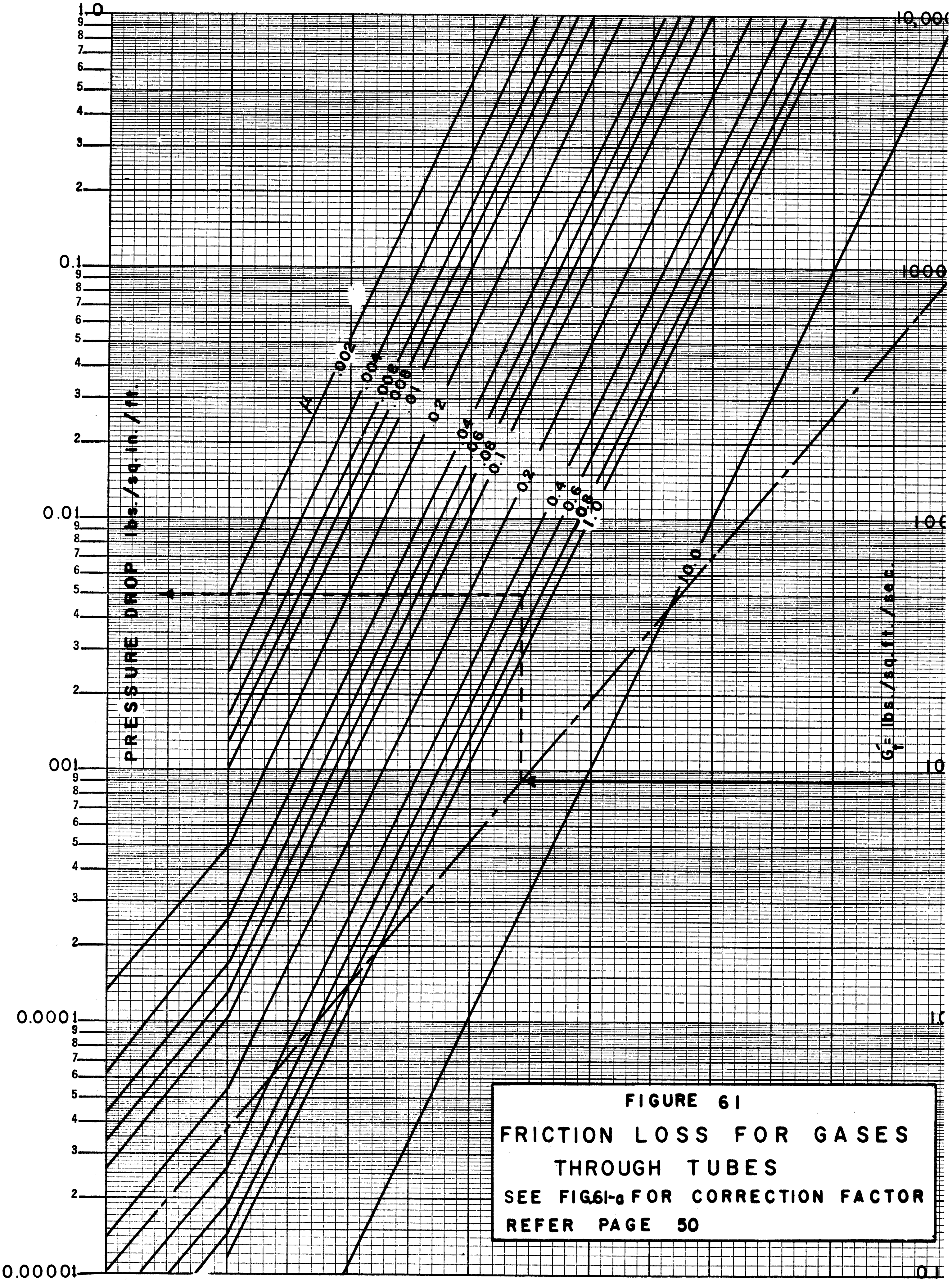


FIGURE 61

FRICTION LOSS FOR GASES THROUGH TUBES

SEE FIG.61-a FOR CORRECTION FACTOR

REFER PAGE 50

I CORRECTION FACTORS FOR OUTSIDE TUBE DIAMETER

BWG	5/8"	3/4"	1"
10			1.24
11			1.17
12			1.14
14		1.68	1.05
16	2.00	1.51	1.00
18	1.85	1.43	0.96

II CORRECTION FACTORS FOR INSIDE TUBE DIAMETER

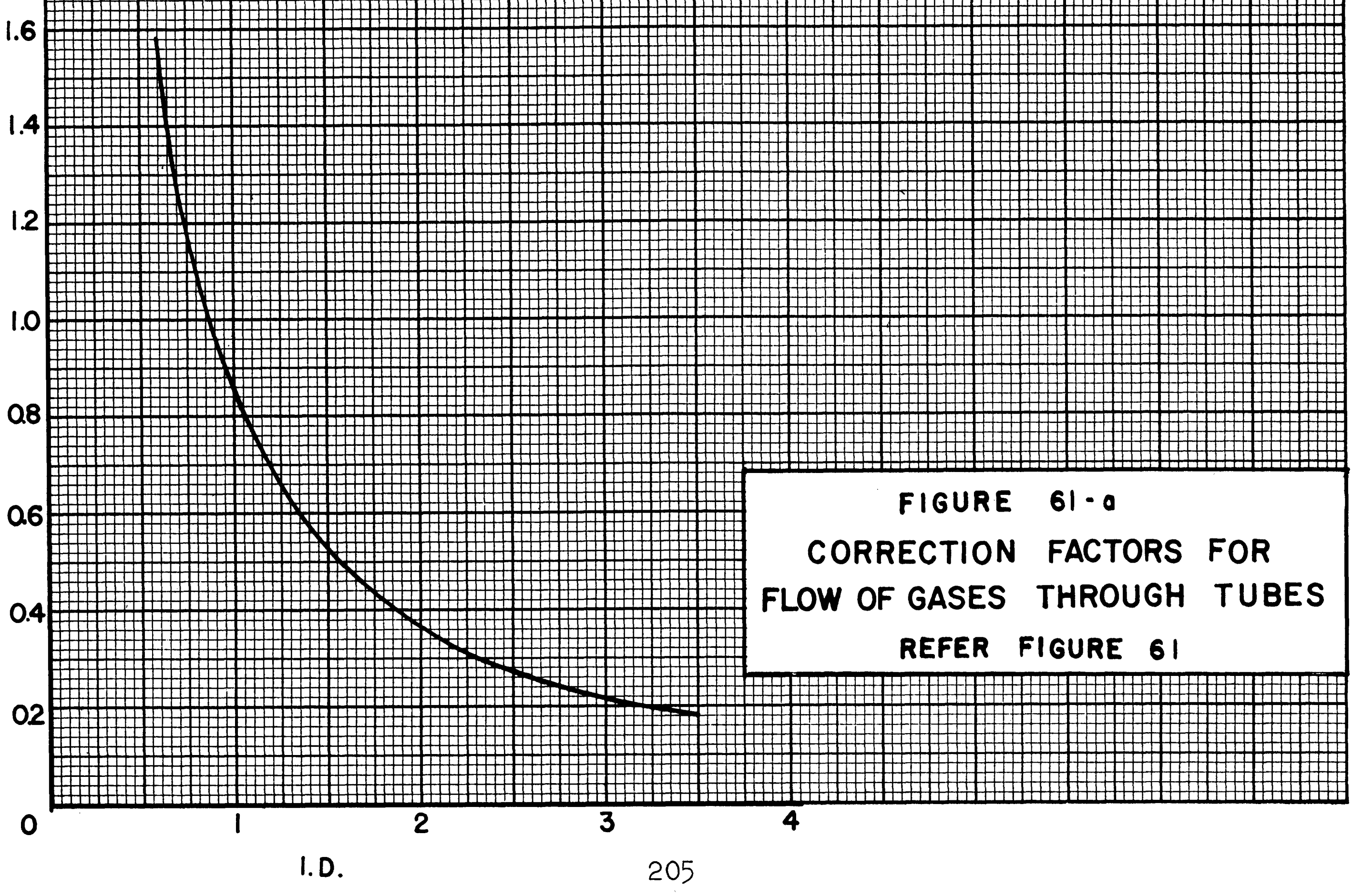


FIGURE 61-a
CORRECTION FACTORS FOR
FLOW OF GASES THROUGH TUBES
REFER FIGURE 61

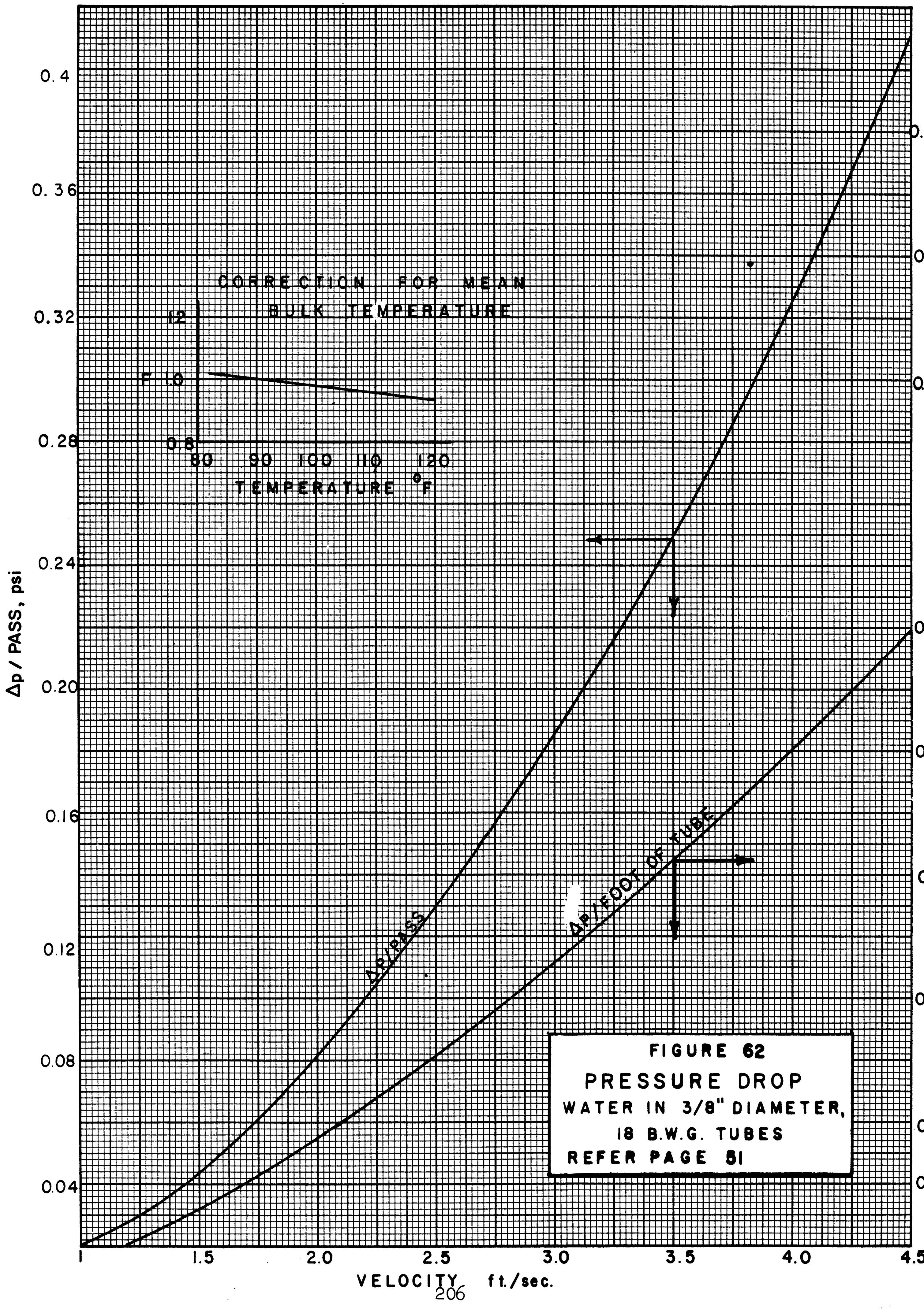


FIGURE 62
PRESSURE DROP
WATER IN 3/8" DIAMETER,
18 B.W.G. TUBES
REFER PAGE 51

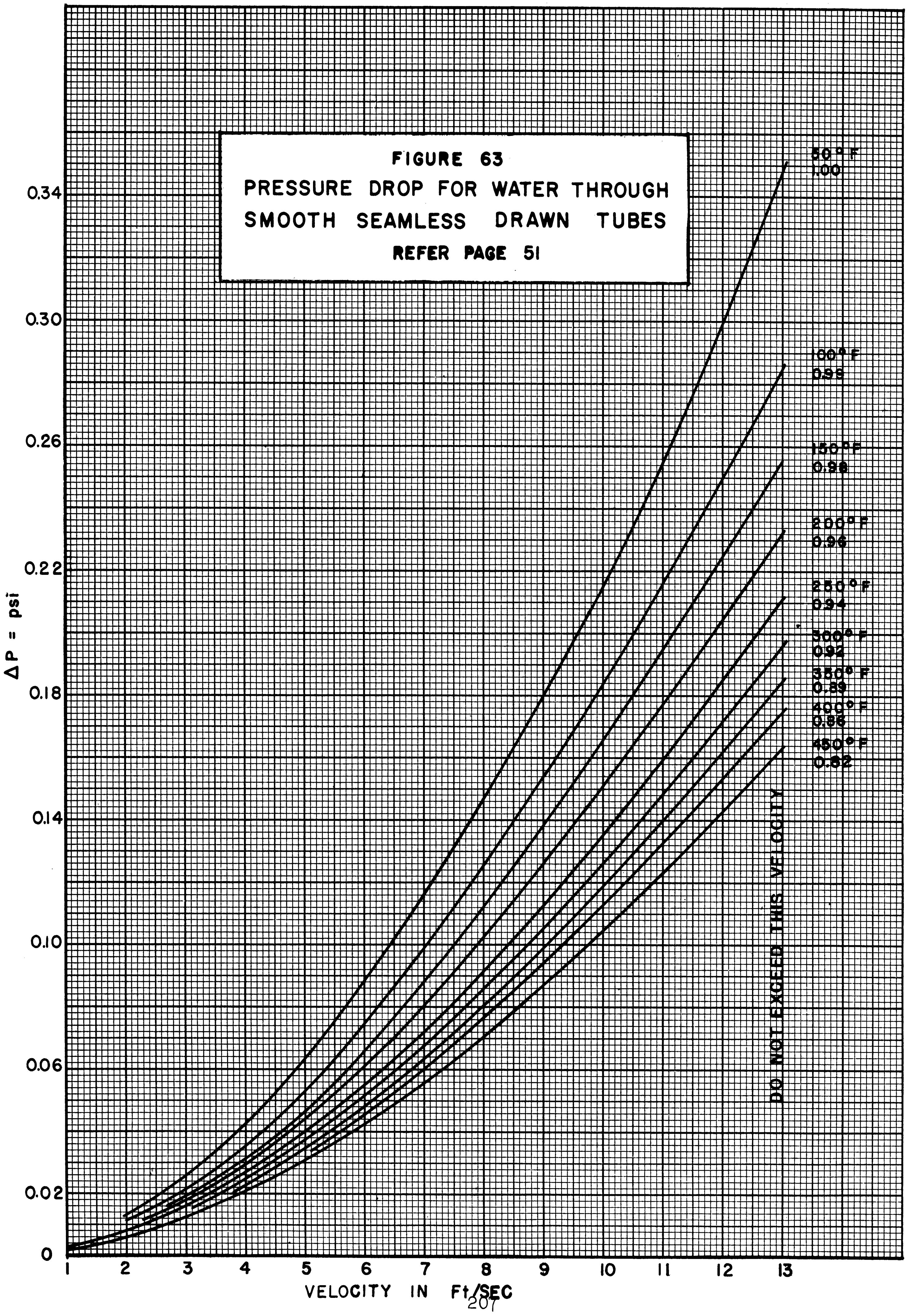


FIGURE 63
PRESSURE DROP FOR WATER THROUGH
SMOOTH SEAMLESS DRAWN TUBES
REFER PAGE 51
ΔP = psi
0.34
0.30
0.26
0.22
0.18
0.14
0.10
0.06
0.02
0
1
2
3
4
5
6
7
8
9
10
11
12
13
VELOCITY IN Ft/SEC
50° F
1.00
100° F
0.99
150° F
0.98
200° F
0.96
250° F
0.94
300° F
0.92
350° F
0.89
400° F
0.86
450° F
0.82
DO NOT EXCEED THIS VELOCITY

TABLE — VALUES OF K_2 FOR BENDS

VEL. IN FT/SEC	RADIUS OF BENDS IN FT 0.5	1.0	2.0
1	.26	.18	.14
2	.31	.21	.17
3	.34	.24	.19
4	.36	.26	.20
5	.38	.27	.21
6	.40	.28	.22
8	.43	.30	.24
10	.46	.32	.25
15	.50	.36	.28
20	.54	.38	.30

CURVE — VALUES OF K FOR CONTRACTION

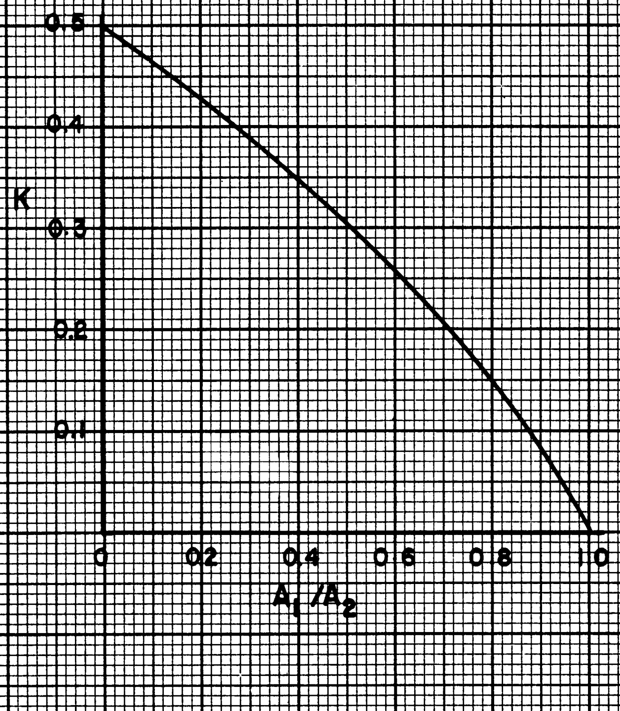
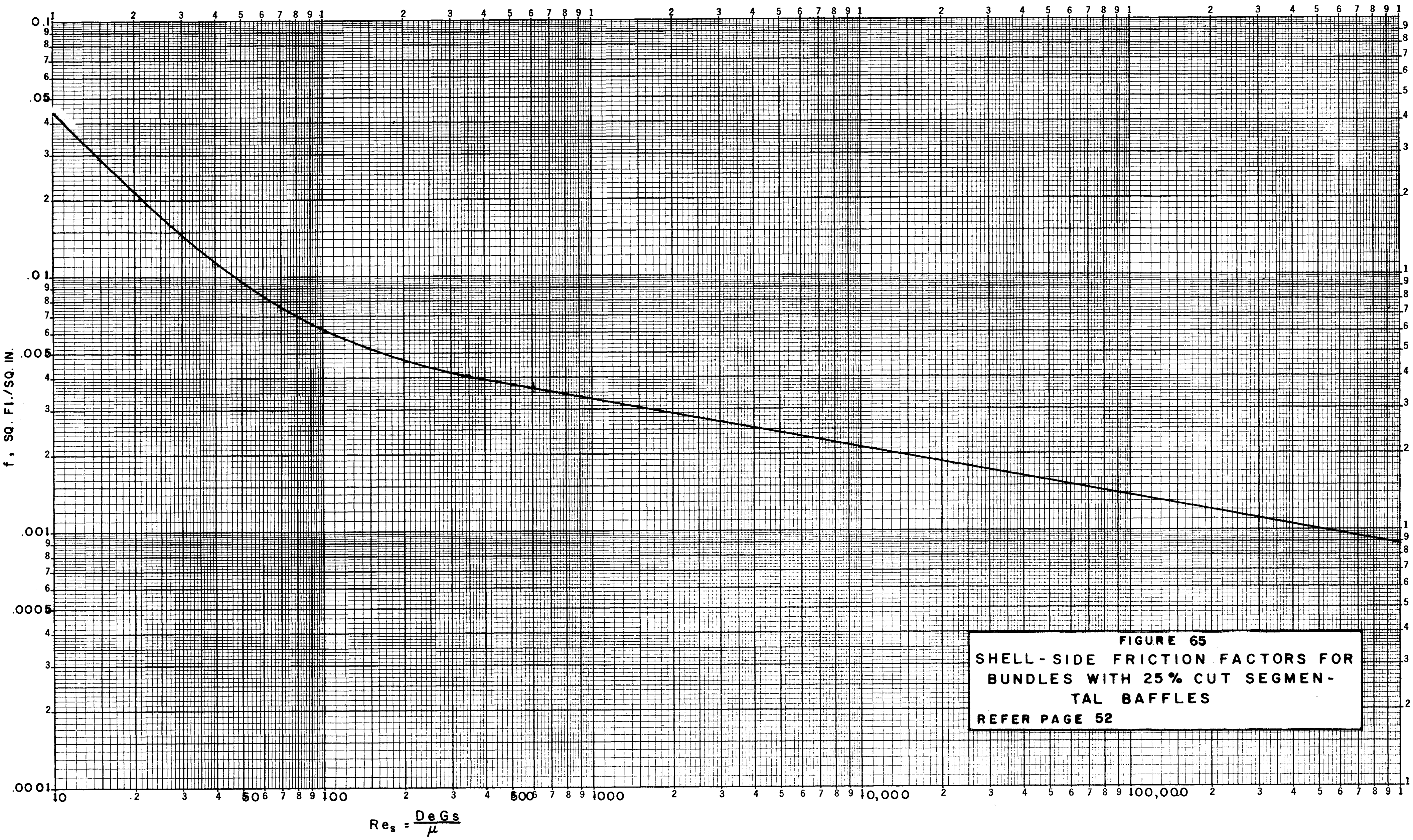


FIGURE 64
FLUID PRESSURE DROP DUE TO BENDS AND CHANGE IN VELOCITY
REFER PAGE 51



f, SQ. FT./SQ. IN.
0.1
.05
.01
.005
.001
.0005
.0001
10
100
1000
10,000
100,000
$Re_s = \frac{DeGs}{\mu}$
FIGURE 65
SHELL-SIDE FRICTION FACTORS FOR BUNDLES WITH 25% CUT SEGMENTAL BAFFLES
REFER PAGE 52

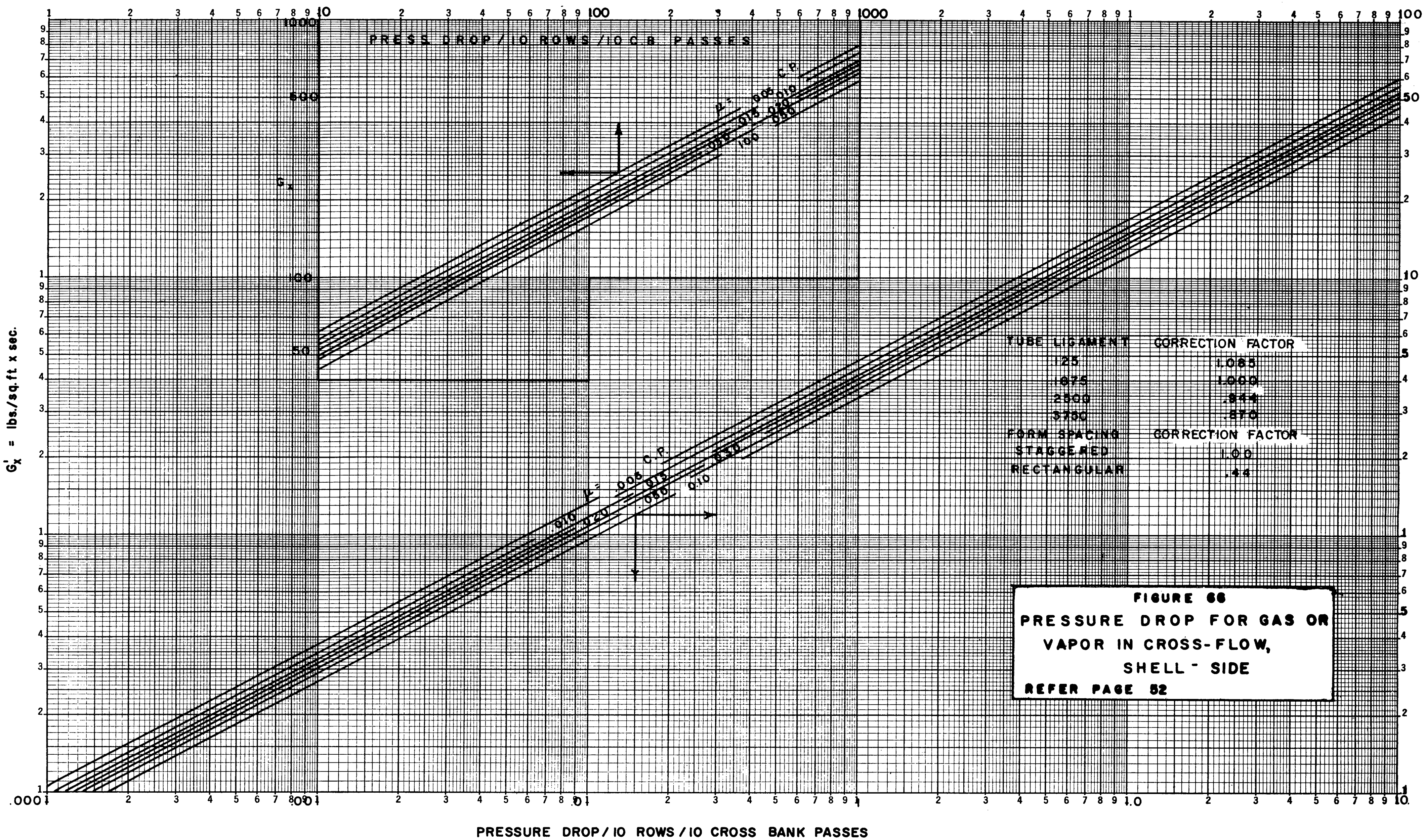


FIGURE 66
PRESSURE DROP FOR GAS OR VAPOR IN CROSS-FLOW, SHELL - SIDE
REFER PAGE 52

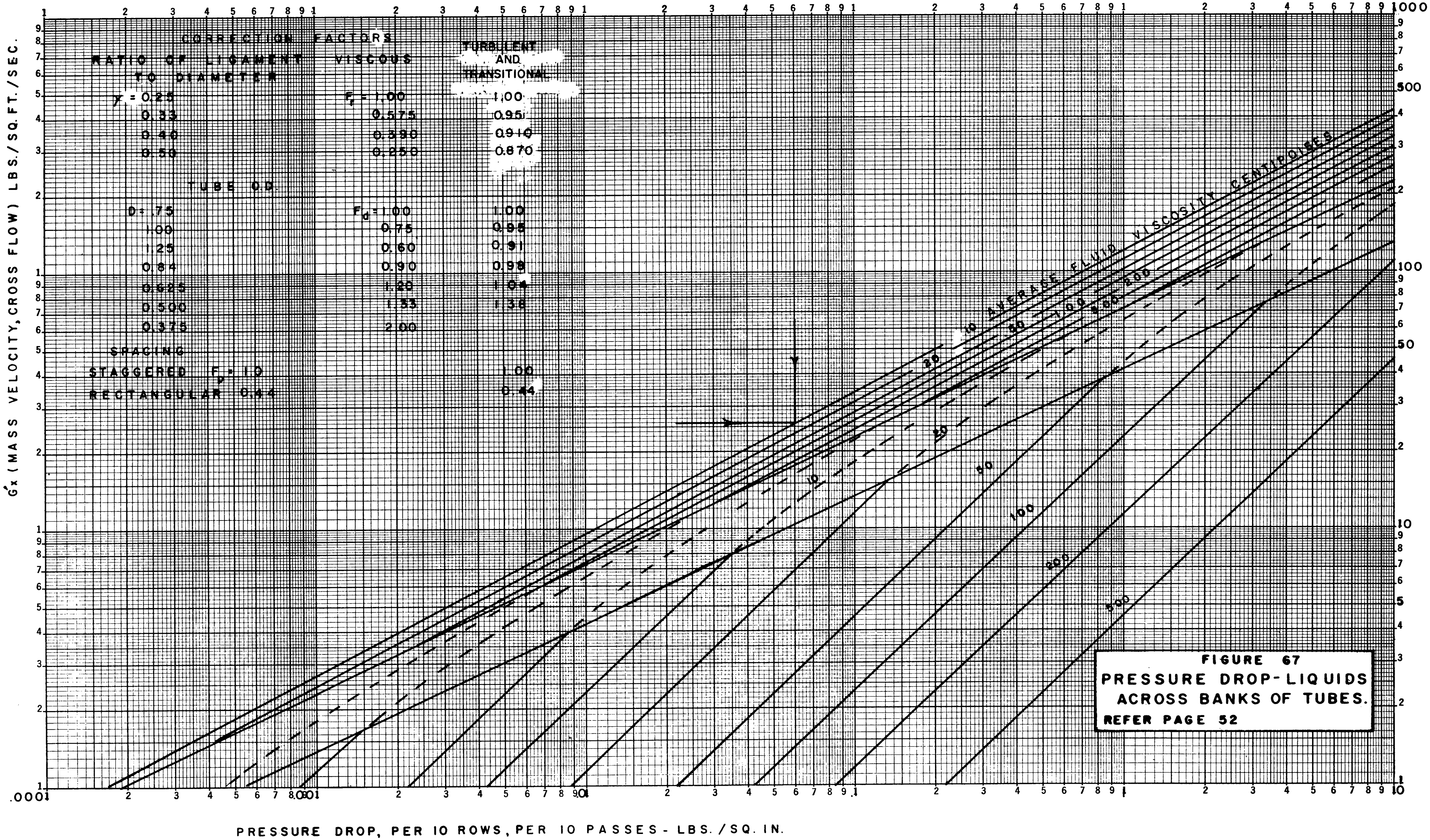


FIGURE 67
PRESSURE DROP-LIQUIDS ACROSS BANKS OF TUBES.
REFER PAGE 52

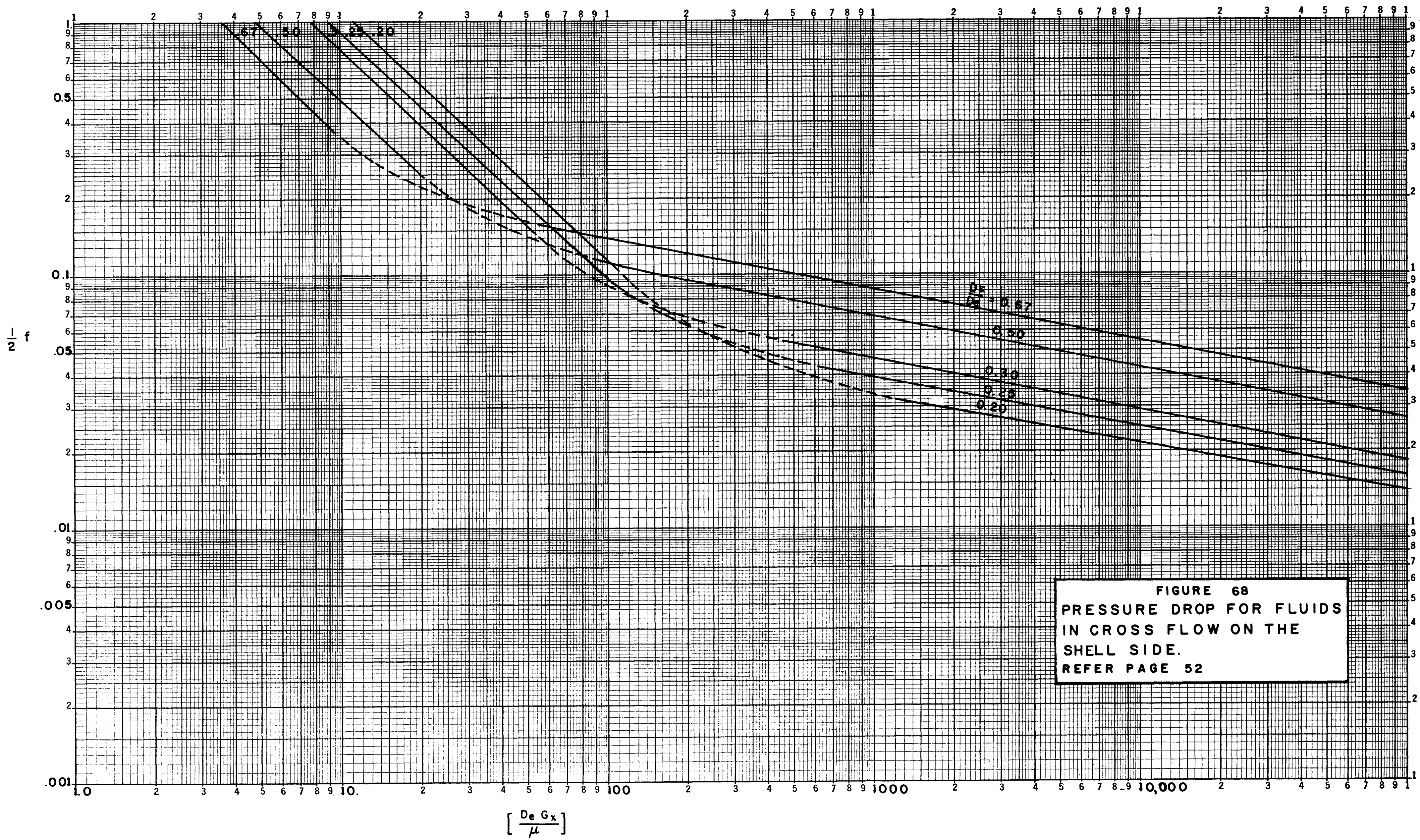
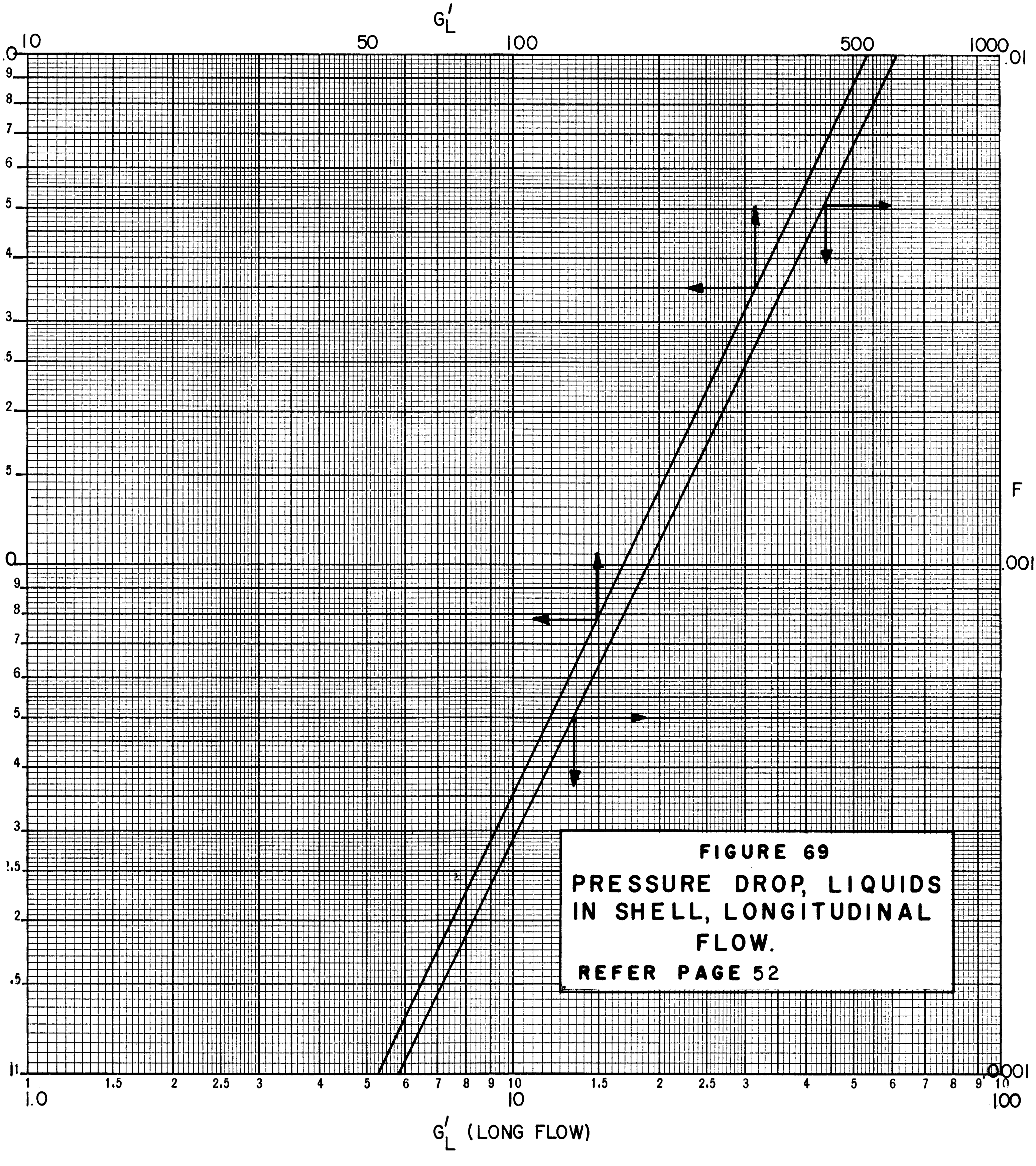


FIGURE 68
PRESSURE DROP FOR FLUIDS IN CROSS FLOW ON THE SHELL SIDE.
REFER PAGE 52


G'L
10
50
100
500
1000
.01
F
.001
.0001
1.0
10
100
G'L (LONG FLOW)
FIGURE 69
PRESSURE DROP, LIQUIDS IN SHELL, LONGITUDINAL FLOW.
REFER PAGE 52

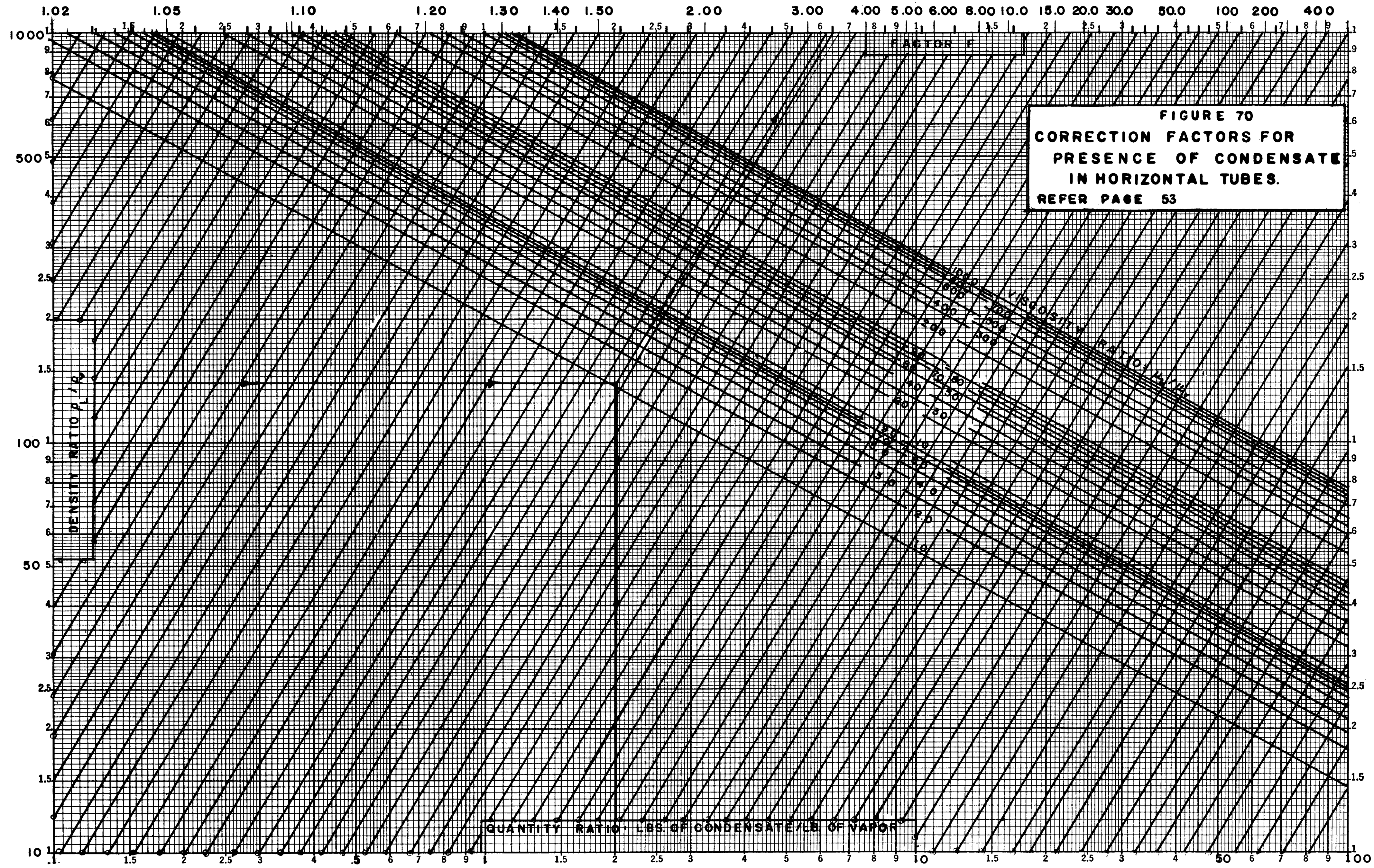


FIGURE 70
CORRECTION FACTORS FOR
PRESENCE OF CONDENSATE
IN HORIZONTAL TUBES.
REFER PAGE 53
FACTOR F
DENSITY RATIO ρ_L/ρ_v
QUANTITY RATIO LBS OF CONDENSATE/LB OF VAPOR
VISCOSITY RATIO μ_L/μ_v

FIG 71 HEAT EXCHANGER SPECIFICATIONS

REFER PAGE 56

ITEM NO.	NO. REQ'D.	B. M. NO.	DESCRIPTION

TYPE		
SIZE	SECTIONS PER BANK	CONNECTED IN
	BANKS PER UNIT	CONNECTED IN
SURFACE PER SECTION SQ. FT.	TOTAL SURFACE	SQ. FT.

PERFORMANCE

SHELL SIDE		TUBE SIDE	SHELL SIDE		TUBE SIDE
	FLUID			GRAVITY-LIQUID	
				@ INLET TEMP.	
	TOTAL FLUID IN			@ 60°F	
	VAPOR			VISCOSITY @ °F	
	LIQUID			VISCOSITY @ °F	
	STEAM			MOL. WT. - VAPOR	
	NON CONDENSABLES			SP. HEAT - LIQUID	
	FLUID VAPORED OR CONDENSED			LATENT HEAT - VAPOR	
	STEAM CONDENSED			NO. OF PASSES	
°F	INLET TEMPERATURE	°F		VELOCITY	
°F	OUTLET TEMPERATURE	°F		PRESS. DROP	
	OPERATING PRESSURE			FOULING FACTOR	

HEAT EXCHANGED - BTU/HR	L.M.T.D. (CORRECTED)
"U" CLEAN	"U" REQUIRED

CONSTRUCTION

SHELL SIDE		TUBE SIDE	CODE
°F	DESIGN TEMPERATURE	°F	MATERIALS OF CONSTRUCTION
	DESIGN PRESSURE		SHELL
	TEST PRESSURE		TUBES
	CORROSION ALLOWANCE		TUBE SUPPORTS
TUBES			BAFFLES
NUMBER	SIZE	O.D.	TUBE SHEETS
PITCH	THICKNESS	BWG.	CHANNEL
LENGTH			CHANNEL COVER
REMARKS			SHELL COVER
SHELL	DIA. THICKNESS	IN.	FLOATING COVER
TUBE SUPPORTS-SPACING	THICK.	IN.	GASKETS
TRANSVERSE BAFFLES			
TYPE	SPACING		
CUT			
THICKNESS	IN.		WEIGHTS
LONGITUDINAL BAFFLE			EA. SHELL LB.: BUNDLE LB.
TYPE	THICKNESS	IN.	FULL OF WATER LB.

CONNECTIONS: SEE FLOW SHEET OR DRAWING

FUNCTION OF ITEM AND REMARKS:

CHK'D	MADE	DATE	NO.

REVISIONS

			PROJECT NO.	TYPED
APPR.				
CHK'D.				
MADE				PAGE OF

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