

A CORRELATION OF RECENT DATA ON SHELL SIDE HEAT TRANSFER

IN

SEGMENTALLY BAFFLED SHELL-AND-TUBE HEAT EXCHANGERS

**Subsidiary to the main
thesis submitted by**

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ABSTRACT.

Several empirical correlations of heat transfer in segmentally baffled heat exchangers have been published, but no systematic attempt has been made to compare the results of different workers, or to correlate them on a common basis. The thesis describes a new method by means of which all detailed experimental data available have been fairly satisfactorily correlated, as shown in Fig.1. The points are fairly well represented by a single line having the equation

$$\frac{hd}{k} = 1.9 \left(\frac{G_{av} d}{\mu} \right)^{0.6} \left(\frac{e\mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14} \left[\frac{P-d \cdot d}{P \cdot D} \right]^{0.4} \left[\frac{Ab}{A+A_c} \right]^2 B \quad (1)$$

The correlation is discussed and the need is shown for (1) Greater accuracy of measurement.
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NOMENCLATURE.

A	Total heat transfer area of outside of tubes	sq.ft.
A _{av}	Average shell side flow area	sq.ft.
A _b	Net flow area through baffle window	sq.ft.
A _c	Clearance area between curved edge of baffle, and shell	sq.ft.
A _m	Average maximum cross flow area between baffles	sq.ft.
A _p	Average minimum cross flow area between baffles	sq.ft.
B	End space factor = $\frac{L_2 + (L_1 - L_2) \left(\frac{2L_3}{L_1 - L_2} \right)^{0.6}}{L_1}$	dimensionless
c	Specific heat per unit mass at mean bulk temperature	B.Th.U./lb ^o F.
C	Ac constant	dimensionless
d	Outside diameter of tube	ft.
D ₁	Inside diameter of shell	ft.
D ₂	Enveloping diameter of tube bundle	ft.
D ₃	Diameter of baffle	ft.
f	A mathematical function	---
G	Mass Velocity	lb./ft. ² hr.
G _{av}	Average mass velocity = W/A _{av}	lb./ft. ² hr.
h	Heat transfer coefficient	B.Th.U./ft. ² hr. ^o F.
k	Thermal conductivity at mean bulk temperature	B.Th.U./ft.hr. ^o F.

NOMENCLATURE (Contd.)

L	Characteristic linear dimension	ft.
L ₁	Total length between tube sheets	ft.
L ₂	Length between end baffles	ft.
L ₃	Free space between adjacent baffles	ft.
Nu	Nusselt number = hd/k	dimensionless
P	Tube pitch	ft.
Pr	Prandtl number = $c_p \mu / k$	dimensionless
Re	Reynolds number = $G_{av} d / \mu$	dimensionless
Re _{eff}	Effective Reynolds number used by Tinker(3)	dimensionless
S	Height of baffle above centre line	ft.
W	Total shell side mass flow rate	lb./hr.
X	Pitch of adjacent rows in tube tank in direction of flow	ft.
Y	Pitch of adjacent tubes in a row perpendicular to flow	ft.
θ	Temperature difference	°F.
μ	Absolute viscosity at mean bulk temperature	lb./ft.Hr.
μ_s	Absolute viscosity at tube surface temperature	lb./ft.hr.

INTRODUCTION.

The shell and tube heat exchanger, of which a typical example is shown in Fig.2, consists of a number of parallel tubes passing through an enveloping shell, which is almost invariably of circular cross section to withstand the pressure of the fluid in the shell, and also for ease of construction. Cross-baffles are frequently used in this type of exchanger to increase the velocity and turbulence of the shell-side fluid; to ensure that the whole heat exchanger surface is effective; and also to support the tubes. The most common type of baffle is the segmental, or half-moon baffle, shown in Figs 2 and 3, which forces the shell-side fluid to flow back and forth across the tubes.

Heat is transferred between one fluid flowing inside the tubes (the "tube-side") and another fluid flowing through the shell outside the tubes (the "shell side"). The present work is concerned with heat transfer by forced convection on the shell side.

APPLICATION OF DIMENSIONAL ANALYSIS.

It can be shown by dimensional analysis (1) that

the heat transfer by forced convection in a system of any given geometrical shape is given by a dimensionless equation

$$\frac{hl}{k} = f\left(\frac{Gl}{\mu}, \frac{c\mu}{k}\right) \quad (2)$$

l being a characteristic linear dimension of the system, and G the mass velocity of fluid measured at any suitable cross section. The form of this equation may be determined by experiment, and it is well established that for fluids flowing outside tubes,

$$\left(\frac{hl}{k}\right)\left(\frac{c\mu}{k}\right)^{-1/3} = f\left(\frac{Gl}{\mu}\right) \quad (3)$$

That is, by plotting

$$\left(\frac{hl}{k}\right)\left(\frac{c\mu}{k}\right)^{-1/3} \quad \text{against} \quad \frac{Gl}{\mu}, \quad \text{a single curve is obtained.}$$

which over a limited range can be expressed approximately by

$$\left(\frac{hl}{k}\right)\left(\frac{c\mu}{k}\right)^{-1/3} = C\left(\frac{Gl}{\mu}\right)^n \quad (4)$$

where C and n are constants. The relationship obtained applies to all systems of the given geometrical shape, regardless of their size. The choice of dimension l , and the cross section at which G is measured, are both arbitrary since all the systems being considered are

geometrically similar; for example, if the system consists of a fluid flowing normal to a bank of tubes, l may be taken as either tube diameter or tube pitch, and G may be measured between adjacent tubes or in the free stream, the resulting relationships being in a constant ratio to each other.

The heat transfer in systems which are not geometrically similar cannot in general be represented by a single relation of the type (2), (3) or (4). Suppose for example that the heat transfer in a bank of tubes could be represented by equation of the type (4); then that in a number of different banks of tubes having several ratios of tube pitch P to tube diameter d would be represented by several corresponding equations of the type (4), possibly with each having the same n but different C . The choice of l and G now becomes significant since alteration in either will alter C , but by judicious choice of l and G , it may be possible to express C as an empirical function of P and d , so that all the geometrical arrangements can be represented by a single empirical relation

$$\left(\frac{hl}{k}\right)\left(\frac{c\mu}{k}\right)^{-1/3} = \left(\frac{Gl}{\mu}\right)^n \times f(P,d) \quad (5)$$

which applies to any size of system, but only to shapes within the range of $f(P,d)$ investigated. This method has

been applied satisfactorily to cross flow over tube banks.

In a baffled heat exchanger, the mass velocity of fluid varies continuously. At the baffle "windows" the flow is parallel to the tubes. Between the baffles, the flow is across the tubes, the mass velocity fluctuating as the fluid flows alternately in the restricted spaces between the tubes in a row, and in the free space between the rows of tubes, and also varying as the width of the cross flow area increases from the baffle window to the shell axis and decreases again to the next baffle window. The choice of a suitable cross-section on which to base G is one of the major problems in obtaining a satisfactory correlation.

Several geometrical variables affect the flow pattern, and therefore the heat transfer, in a baffled heat exchanger, those considered to be most significant being shown in Fig.3; they are:-

Tube diameter, pitch, and arrangement.

Baffle dimensions and spacing.

Shell diameter.

Spaces between end baffles, and tube plates.

Clearance between baffles and shell.

" " tubes, and tube holes in baffles.

" " the outside of the tube bundle,
and the shell.

A correlation, to be generally applicable, should therefore be based on tests in which all these quantities have been varied.

PREVIOUS CORRELATIONS.

McAdams (2) quoting Chilton & Drew, recommends that h be calculated from the ordinary equation for cross flow, reduced by a constant factor of 0.6 "to allow for the effect of leakage that ordinarily occurs in cross-baffled heat exchangers." In the equation for cross-flow, G is based on the minimum flow area between the tubes in a row; in this case, an average value for all rows of tubes between the baffle edges would presumably be used. The method clearly gives only a rough approximation, and implies that the geometrical arrangement affects the heat transfer only in so far as it affects the cross flow mass velocity. Tinker (3) has shown in his Fig. 10 that the factor 0.6 may be in error by several hundred per cent.

Bowman (4) correlated his own data for a number of commercial exchangers. Mass velocity G was calculated on the minimum cross flow area between baffles, at the shell diameter, modified by a leakage factor chosen empirically to bring the results onto Colburn's curve for flow across an un-baffled tube bank. The leakage factor is assumed to

be constant "for all the exchangers in a given line."

This, like the previous method approximates to a given set of results but is obviously unsatisfactory as a general correlation.

Short (6) made a systematic series of measurements of heat transfer in a laboratory heat exchanger, in which the baffle spacing, tube diameter, and tube pitch were varied, baffle height, shell diameter and tube length being fixed. A complicated expression is given for G which is in effect an average of mass velocities through various sections in the shell, empirically weighted. A dimensionless tube arrangement factor is included, and the results all fall remarkably well on a single curve. The expression for G contains the shell diameter, although this was not varied in Short's experiments; it is stated, however, that the correlation also fairly well represents Bowman's data which cover a 4 to 1 range of shell diameters.

Short (5) later re-correlated the same data, using a simpler expression for the mass velocity, namely the arithmetic mean of mass velocities through three cross sections of the shell. There was a slight loss of accuracy even omitting the widest baffle spacing. A new "tube arrangement factor" $\left(\frac{P-d}{P} \cdot d\right)^{0.4}$ was introduced which has dimensions $l^{0.4}$.

Thus the "constant" 1.28 obtained, which should be dimensionless, has dimensions $l^{-0.4}$. It will be shown later in the present paper that a dimensionless term $\left(\frac{P-d}{F} \cdot \frac{d}{D_1}\right)^{0.4}$ gives a good correlation of Tinker's data.

The shell diameter D_1 was not varied in Short's experiments, but Bowman's data for shell diameters up to four times that used by Short's correlate fairly well on Short's curve.

Short's work is of particular value because it shows the effect of varying one dimension while the rest were unchanged; thus the effects of baffle spacing, tube size, and tube pitch were found. The effects of shell diameter, baffle height, and baffle-to-shell clearance were not investigated.

Gardner & Siller (7) carried out experiments similar to Short's, using two arrangements, differing only in the pitch and number of baffles. Heating and cooling runs were made with water, and three oils, and a viscosity ratio similar to that proposed by Sieder & Tate (8) was introduced to bring heating and cooling data onto a single curve. In other respects their method of correlation was similar to Short's original method (6). Their work is useful in establishing the viscosity ratio term which had hitherto been neglected.

Short (9) recently extended his earlier work by investigating the effects of baffle height, and baffle to shell clearance. The shell was accurately bored, and baffles having clearances from 0.06 in. to 0.255 in. on diameter were used. Short's original method (6) of correlation was used, with the addition of a baffle for shell clearance ratio $(A_b/A_c)^{0.17}$. This is not a rational factor, since it tends to infinity as A_c tends to zero; a more rational factor is proposed later in the present paper. All the tests were for the cooling of water on the shell side, and no viscosity ratio was included. Again, only one shell size was investigated.

Tinker(3) has made an entirely new approach to the problem which is perhaps fundamentally more sound than any previous one. The distribution of fluid among the various flow channels in the shell is analyzed from fluid flow considerations, only that part of the fluid which actually flows across the tube nest being considered effective. It is stated that in conventional designs this may be only from 12% to 60% of the total shell side fluid flow, the remainder leaking round the baffles, or by-passing the tubes through the annular space between the tube nest and the shell.

The so-called "effective" mass velocity was obtained by dividing this calculated mass flow across the tubes by the mean cross flow area of the tube nest. A semi-theoretical factor was introduced to allow for the smaller mass flow rate in the sections between the end baffles and the tube sheets. Experimental data are given for eleven different shell and baffle arrangements, but for only two tube diameters and two pitch/diameter ratios. These correlate fairly well by Tinker's method, especially at the higher Reynolds numbers, as shown in Fig. 20. The method, however, is involved, and requires a fairly complete knowledge of the flow distribution. The same data correlate somewhat better by the method proposed later, shown in Fig. 16, the heat transfer being calculated directly from the dimensions of the exchanger, and the fluid properties.

Tinker's paper contains an interesting analysis of the estimated effect on the heat transfer, of various clearances. Special emphasis is put on the detrimental effect of tube bundle to shell clearance. Complete elimination of this, it is claimed, will frequently increase h by over 40%, while sealing strips between the baffle windows, shown in Fig. 3, will increase h by about 20%. Floating heads (Fig. 2) should therefore be designed to

require the minimum clearance between the tube nest and the shell.

Donohue (10) has taken the various experimental data mentioned above, and plotted the results of each observer separately, expressing them in a simple form similar to that recommended by McAdams. Short's results required a different method of correlation from the rest, and have been ignored altogether in arriving at the equations finally recommended. Tinker's data appear to have been incorrectly plotted, the scatter being thereby reduced; furthermore, as already pointed out, Tinker's data cover only a small range of tube arrangements.

Despite its shortcomings, however, this type of correlation might be useful in designing a fairly standardised type of exchanger.

DEVELOPMENT OF THE PROPOSED NEW CORRELATION.

As shown in the previous section, different methods of correlation have been fairly successful for different limited ranges of conditions but no really satisfactory general correlation has emerged. Owing to the highly empirical nature of this type of correlation, the first steps are more or less arbitrary, and subsequent steps

involve some trial and error.

SCOPE AND PROCEDURE.

The correlation was restricted to tests for which detailed data, including the values of all the dimensions considered likely to affect the heat transfer, were available, namely the tests of Tinker (3), Short 1947(9) and Gardner & Siller(7). The range of variables covered is shown in the Tables in Appendix I.

Tinker's results which covered the widest range of variables were first plotted on the basis of Short's 1943 correlation(5). This method was then modified step-by-step until a satisfactory correlation was obtained. This was found to apply well to Short's 1947, and Gardner & Siller's results so finally all the results were plotted on a single graph (Fig.1).

TEMPERATURE AT WHICH TO MEASURE THE FLUID PROPERTIES.

As in previous correlations, fluid properties have been taken at the mean "bulk" temperature of shell side fluid, that is, the arithmetic mean of the inlet and outlet temperatures. This is the simplest procedure to follow, and results in a satisfactory correlation.

CHARACTERISTIC LINEAR DIMENSION.

Again, following previous correlations, the outside

tube diameter is used as the characteristic dimension in the dimensionless groups hd/k , and $G_{av} d/\mu$

MASS VELOCITY.

Short (5) has shown that apart from exchangers with very wide baffle spacing (21 in. in a 6 in. dia. shell), the method of assessing the mass velocity G_{av} used by him in 1943 results in little loss of accuracy compared with the more complicated expression he used in 1942, subsequently used by Gardner & Siller in 1947, and again by Short in 1947. This method was therefore tried, and proved satisfactory.

The method of calculating G_{av} is given below.

CORRELATION OF TINKER'S DATA (3).

As a first comparison, these results were plotted according to Short's 1943 method which consists of plotting

$$\left(\frac{hd}{k}\right)\left(\frac{c\mu}{k}\right)^{-1/3} \left(\frac{P-d}{P} \cdot d\right)^{-0.4} \quad \text{against} \quad \frac{G_{av} d}{\mu} .$$

The resulting plot is shown in Fig.9 and compared with Short's line which has the equation

$$\frac{hd}{k} = 1.28 \left(\frac{G_{av} d}{\mu}\right)^{0.6} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{P-d}{P} \cdot d\right)^{0.4}$$

The points are widely scattered, being in the main above Short's line.

Viscosity ratio μ/μ_s . Sieder & Tate (8) first introduced a factor $(\mu/\mu_s)^{0.14}$ to bring together heating and cooling data for viscous oils inside tubes. In recent experiments at the University of Delaware, Bergelin et alia (11) found that for oil flowing across a cooled tube bank, the same factor applied. For Tinker's exchanger No. 11 heating and cooling data are given which have been plotted both with and without the viscosity ratio, in Fig. 10, showing that the factor brings the two sets of results into a single line. This ratio is discussed further in the section below on Gardner and Siller's data.

End Space factor. The heat transfer coefficient is based on the total heat transfer area A of the exchanger, while mass velocity is based on the space between adjacent baffles. In the end spaces between the end baffles and the tube sheets, however, the mass velocity for cross flow is less than that between baffles, in the ratio $2L_3/(L_1 - L_2)$ where L_1 is the total length between tube sheets, L_2 the length between the end baffles, and L_3 the space between adjacent baffles. Assuming the heat transfer rate to be proportional to $(\text{mass flow})^{0.6}$, the heat transfer rate in the end spaces will be $\left(\frac{2L_3}{L_1 - L_2}\right)^{0.6}$ times that between baffles. Since

the tube surface area is proportional to length, it follows that

$$\frac{\text{Actual heat transfer}}{\text{Heat transfer calculated on mass velocity between baffles}} = B$$

where $B = \frac{L_2 + (L_1 - L_2) \left(\frac{2L_3}{L_1 - L_2} \right)^{0.6}}{L_1}$

This factor was first introduced by Tinker.

Tinker's data have been re-plotted, with reduced scatter, in Fig.11, which includes both the viscosity ratio $(\mu/\mu_s)^{0.14}$ and the end space factor B.

Effect of tube pitch P, tube diameter d, and shell diameter D₁
 The factor $\left(\frac{P-d}{P} \cdot \frac{d}{D_1} \right)^{0.4}$ used in plotting Figs 9, 10, and 11

was used first by Short in his 1943 correlation covering various tube diameters and pitches, but only one shell size and one baffle height. This factor, as already mentioned, is open to objection because it is not dimensionless. As it was found that the ordinates of Fig.11 tended to be higher for the smaller shell sizes than for the larger, the factor was modified to $\left(\frac{P-d}{P} \cdot \frac{d}{D_1} \right)^{0.4}$ with the result shown in Fig.12, of reducing the scatter still further.

Clearance Ratio. The ordinates in Fig.12 are in general

higher, the smaller the baffle to shell clearance. The clearance is included in a dimensionless ratio $A_b/(A_b + A_c)$, where A_b is the net flow area through a baffle window, and A_c the clearance area between the curved edge of a baffle, and the shell. Expressed in this way, the ratio tends to 1 as the clearance is reduced to zero. The ordinates in Fig.12 for each separate exchanger at $Re = 10^2$ has been plotted logarithmically against $A_b/(A_b + A_c)$ in Fig.14. The points are rather scattered, but a line of slope 2 is fairly representative, and the introduction of a factor $\left(\frac{A_b}{A_b + A_c}\right)^2$

reduces the scatter of the correlation as shown in Fig.16 which is the final correlation of Tinker's data, and has the equation

$$\frac{hd}{k} = 2.22 \left(\frac{G_{av} d}{\mu}\right)^{0.6} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_s}\right)^{0.14} \left(\frac{F-d}{F} \cdot \frac{d}{D_1}\right)^{0.4} \left(\frac{A_b}{A_b + A_c}\right)^2 B \quad (6)$$

CORRELATION OF SHORT'S 1947 DATA (9)

These measurements were made with a single tube bundle in a single shell, but with a range of baffle spacings, heights and clearances. The data have been plotted first in Fig.13, on the same basis as Fig.12, that is without a

clearance ratio. A cross-plot was then made (Fig 15) of the ordinate in Fig 13 for each separate baffle arrangement, at $Re = 10^4$, against $A_b/(A_b + A_c)$. Again a line of slope 2 is fairly representative of exchangers having clearance ratios of more than about 0.7. Exchangers having clearance ratios less than this would be unlikely to occur in practice, and have therefore been omitted from the final correlation of Short's data which is given in Fig 17. The line in Fig 17 has equation

$$\frac{hd}{k} = 1.55 \left(\frac{G_{avd}}{\mu} \right)^{0.6} \left(\frac{cm}{k} \right)^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14} \left(\frac{F-d}{F} \cdot \frac{d}{D_1} \right)^{0.4} \left(\frac{A_b}{A_b + A_c} \right)^2 B \quad (7)$$

CORRELATION OF GARDNER & SILLER'S DATA(7).

The particular feature of these experiments was the use of four different fluids in both heating and cooling runs. Owing to the wide scatter of data for water, Gardner & Siller omitted these from their final correlation; they have been included here however, and the scatter is discussed later.

In order to further investigate the viscosity ratio exponent, taken above as 0.14, the plot in Fig 18 has been made, the scattered results for water being enclosed in a ring. Since the viscosity ratio correction

for water is, in any case, invariably small the points for water have been disregarded. It should be pointed out that the actual values obtained for μ/μ_s differ considerably from those plotted by Gardner & Siller, although the slope of the line is about the same. Two lines have been drawn through the points, one having the usual slope of 0.14, and the other the slope 0.19 which was adopted by Gardner & Siller. There appears to be little to choose between these two lines, and in fact a line having a slope of 0.16 or 0.17 would be better than either. In view of the previous evidence (8) (11), in support of the 0.14 exponent, this value has been retained here, and Gardner & Siller's data have been plotted in Fig 19 on the same basis as Figs 16 and 17. The points for the higher Reynolds numbers are scattered, but a mean line has been drawn having equation

$$\frac{hd}{k} = 1.9 \left(\frac{G_{av} d}{\mu} \right)^{0.6} \left(\frac{c \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14} \left(\frac{P-d}{P} + \frac{d}{D_1} \right)^{0.4} \left(\frac{A_b}{A_b + A_0} \right)^2 \quad (8)$$

FINAL CORRELATION.

The results of Tinker, Short (1947) and Gardner & Siller which were plotted separately in Figs 16, 17, and 19, have been combined in a single graph in Fig 1. The nearest

line having slope 0.6, i.e. the same slope as Figs 16, 17, and 19, has equation

$$\frac{hd}{k} = 1.9 \left(\frac{G_{av} d}{\mu} \right)^{0.6} \left(\frac{c\mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14} \left(\frac{P-d}{P} \cdot \frac{d}{D_1} \right)^{0.4} \left(\frac{A_b}{A_b + A_c} \right)^2 B \quad (1)$$

fluid properties except μ_s being taken at the mean bulk temperature

h = shell side heat transfer coefficient, given by

$$\frac{\text{Total heat transfer per hour}}{A \times \log. \text{ mean } \theta}$$

$$\frac{\text{B.Th.U.}}{\text{ft.}^2 \text{ hr. } ^\circ\text{F.}}$$

d = outside tube diameter

k = thermal conductivity of fluid

$$\text{B.Th.U./ft.hr.}^\circ\text{F.}$$

$$G_{av} = \frac{\text{Total shell side mass flow rate}}{A_{av}}$$

$$\text{lb./ft.}^2 \text{ hr.}$$

μ = Absolute viscosity of fluid

$$\text{lb./ft.hr.}$$

μ_s = " " " " at tube surface temperature

$$\text{" " "}$$

c = specific heat of fluid

$$\text{B.Th.U./lb.}^\circ\text{F.}$$

P = tube pitch

$$\text{ft.}$$

D_1 = shell diameter

$$\text{ft.}$$

B = end space factor

$$= \frac{L_2 + (L_1 - L_2) \left(\frac{2L_3}{L_1 - L_2} \right)^{0.6}}{L_1}$$

L_1 = total length between tube sheets

$$\text{ft.}$$

L_2 = length between end baffles

$$\text{ft.}$$

L_3 = space between adjacent baffles ft.

* A_b = net flow area through baffle window ft.²

* A_c = clearance area between curved edge of baffle,
and shell ft.²

A_{av} is obtained as follows:-

$$G_{av} = \frac{1}{3} \left(\frac{W}{A_b} + \frac{W}{A_p} + \frac{W}{A_m} \right) = \frac{W}{A_{av}}$$

where

$$\frac{1}{A_{av}} = \frac{1}{3} \left(\frac{1}{A_b} + \frac{1}{A_p} + \frac{1}{A_m} \right)$$

* A_p = average minimum cross flow area
between baffles ft.²

* A_m = " maximum cross flow area
between baffles ft.²

* Methods of calculating areas for four tube arrangements are given in Figs 4 to 7; methods of other tube arrangements will be apparent. Curves for calculating the area and length of curved perimeter of a circular segment are plotted in Fig 8 to facilitate the calculation of A_b and A_c .

DISCUSSION.

The mean line drawn in Fig 1 coincides with the mean line through Gardner & Siller's results. It is 18% above the line through Short's data and is 17% below the line through Tinker's data. A somewhat closer approximation to the points in Fig 1 would have been given by a line

having a slope of less than 0.6, but it is preferable to have exponents of only one digit where possible. This exponent is further justified by the fact that the Reynolds number exponent in the ordinary cross-flow equation is also 0.6.

It should be noted, however, that the slope of the line for each separate exchanger is unaffected by the method of correlation, and there is a tendency to impress upon the collected points for a group of exchangers, that slope which is considered at the outset to be nearest to the slopes of the lines for the separate exchangers of the group. This effect is shown in the Appendix dealing with the Wilson method of plotting, but also occurs when a cross-plot such as Fig 14 is made from the points in Fig 12, which represent the results for Tinker's group of exchangers, the points for each exchanger lying on a different line (although the lines have not been drawn), all the lines having a slope of approximately 0.6. Each point in Fig 14 represents one exchanger, its ordinate being the ordinate in Fig 12 at which the nearest line of slope 0.6 passing through the points for that exchanger, intersects the line $Re = 10^2$. As a result, the slope^{of the}/nearest line through the collected points in the final correlation in Fig 16 is made to be 0.6. The slope would be known with greater certainty

if there were less scatter of the points for each individual exchanger.

One of the main difficulties in dealing with the results is that in all cases it was U , the overall coefficient, that was measured, while it was the shell side coefficient that was required. Tinker obtained the shell side coefficient from U by calculating the tube side coefficient from measured tube-side data; Short, and Gardner & Siller have used the Wilson method of plotting. Neither method is entirely satisfactory, and it is worth noting that when, during the present work, an attempt was made to obtain Tinker's tube-side coefficients by the Wilson method, the resulting correlation was chaotic. In either case, the experimental errors were magnified when the shell-side resistance was small compared with the tube-side, the errors in determining both the large tube-side resistance, and the large overall resistance, being transferred to the small shell-side resistance. This undoubtedly accounts for the wide scatter of Gardner & Siller's data for water, a fact that has been entirely overlooked in their paper and the ensuing discussion. For some of their experiments, appreciably different results are reported for almost

identical runs, showing either that measurements were inaccurate, or that fouling had occurred.

The value of the correlation is restricted by the narrow range of d/D_1 , namely from $1/9.4$ to $1/27.3$. Extrapolation of equation (1) to cases far outside this range can only be regarded as speculative, and data are required for larger exchangers having smaller d/D_1 ratios.

Tube hole clearance has not been investigated. Fluid flowing through this clearance is certainly not, as Tinker suggests, lost as far as heat transfer is concerned. Effect of tube arrangement. The relative merits of different tube arrangements will not be considered, but only the effect of tube arrangement on the shell-side coefficient.

Tube arrangement has not been considered as a variable, in the correlation, although, while both Short, and Gardner & Siller used an equilateral triangular arrangement, the flow directions relative to the tube patterns in the two cases were at 90° to each other.

Since tube arrangement will affect only the cross flow heat transfer, three typical arrangements shown in Fig 21, will be compared by applying Grimson's correlation for cross-flow heat transfer as given by Fishenden & Saunders (1).

In each case, the tube pitch is taken at $1.5d$. The Reynolds number in Case 1 (measured on the minimum flow area) is taken as 8,000, while in the other two cases a Reynolds number is taken which will give the same total mass flow rate. The relative shell side coefficients are then as follows.

Case	1	2	3
Shell-side coefficient	1	0.98	0.88
" " " " in Case 1			

For normal tube arrangements therefore, tube arrangement has some, but not a major effect on the shell side coefficient, bearing in mind that not all the heat transfer takes place in cross flow, the proportion which does depending on the geometry of the baffles.

.

Summarising, the data from three different sets of experiments have been correlated as accurately as the observations allow. In order to determine the Reynolds number exponent with more certainty, there is need for greater accuracy of individual results, and especially in the determination of the tube-side coefficient. Data are required for larger exchangers. Tube arrangement has only a small effect on the heat transfer coefficient.

APPENDIX IRange of variables included in the correlation.

1. Range of dimensionless groups in equation (1).

<u>Group</u>	<u>Range</u>		
$\frac{G_{avd}}{\mu}$	1.67	-	44,000
$\frac{G\mu}{K}$	2.3	-	2,000
$\frac{\mu}{\mu_s}$	0.052	-	6.45
$\frac{P - d}{P}$	0.167	-	0.29
$\frac{D_1}{d}$	9.4	-	27.3
$\frac{A_b}{A_b + A_s}$	0.76	-	0.98
B	0.67	-	0.95

2. Details of exchangers and fluids used by Tinker.

Exchanger No.	1	2	3	4	5	6	7	8	9	10	11
Shell inside dia.	in.6.000	8.219	8.031	8.070	10.523	3.527	5.981	8.007	7.959	7.959	8.02
Tube outside dia.	in.0.375	0.375	0.375	0.375	0.375	0.375	0.375	0.625	0.375	0.375	0.626
Tube pitch	in.0.531	0.531	0.531	0.531	0.531	0.453	0.453	0.750	0.453	0.453	0.75
No. of tubes	92	170	170	170	262	40	124	68	243	243	68
Length of tube	in.36	24	24	24	24	15	24	48	80	80	72
Baffled length of tube	in.24.5	11.69	11.69	18.06	12.5	10.41	16.25	34.75	67.12	67.12	56.25
Tube hole clearance	in.0.020	0.020	0.020	0.020	0.020	0.010	0.010	0.015	0.011	0.011	0.012
Enveloping dia. of tube bundle	in.5.50	7.50	7.50	7.50	9.125	3.25	5.50	7.25	7.844	7.844	7.1
Number of baffles	9	12	12	18	9	10	9	23	11	11	19
Baffle pitch	in.3.06	1.062	1.062	1.062	1.562	1.156	2.031	1.562	5.5	5.5	3.125
Baffle height from centre	in.0.840	2.12	2.12	2.12	2.31	0.90	1.25	1.955	1.625	0	1.965
Baffle clearance on dia.	in.0.094	0.250	0.062	0.101	0.070	0.017	0.023	0.067	0.037	0.037	0.044

Shell side fluids used:- Water, and Navac 2190 oil.

Properties of Navac 2190 oil

Temp. °F	Specific heat, B.Th.U./lb. °F.	Specific Gravity	Viscosity lb/ft.hr.	$k \left(\frac{\text{Btu}}{\text{ft}^2 \cdot \text{hr} \cdot ^\circ\text{F}} \right)^{1/3}$
50	0.429	0.918	2420	1.87
100	0.457	0.900	242	0.83
150	0.485	0.882	60	0.54
180	0.502	0.871	34	0.44

3. Details of exchangers and fluid used by Short (1947).
and included in correlation.

For all exchangers:-

Shell inside dia.	6.185 in.
Tube outside dia.	0.625 in.
Tube pitch	0.875 in.
No. of tubes	30
Length of tube	60 in.
Baffled length of tube	43 in.
Tube hole clearance	0.005 in.
Enveloping dia. of tube bundle	5.73 in.

Exchanger No.	1	2	3	4	6	7	8	9	11	12	13	20	21	22	23
Number of baffles	19	15	11	7	19	11	19	11	19	15	11	19	19	11	19
Baffle pitch in.	2.33	3.01	4.24	7.11	2.33	4.24	2.33	4.24	2.33	3.01	4.24	2.33	2.33	4.24	2.33
Baffle height from centre in.	2.5	2.5	2.5	2.5	2.00	2.00	1.21	1.21	0	0	0	0	1.21	1.21	1.21
Baffle clearance on dia. in.	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.255	0.125	0.125	0.255

Shell side fluid used :- Water.

4. Details of exchangers and fluids used by Gardner and Biller.

For both exchangers:-

Shell inside dia.	5.94 in.
Tube outside dia.	0.375 in.
Tube pitch	0.5 in.
No. of tubes	80
Length of tube	36 in.
Baffled length of tube (approx.)	30 in.
Tube hole clearance	0.010 in.
Enveloping dia. of tube bundle	5.28 in.
Baffle height from centre	0.75 in.
Baffle clearance on dia.	0.030 in.

Exchanger No.	1	2
Number of baffles	28	15
Baffle pitch	1 in.	2 in.

Shell side fluids used :- Water, Secony oil No.990, Secony oil No. 969, and Velocite S oil.

Properties of the oils used.

Oil	Temp ^{°F}	Viscosity lb./ft.hr.	Specific Gravity	Spec. heat B.Th.U/lb. ^{°F}	Thermal cond. B.Th.U/ft.hr ^{°F}
990	135	204	0.903	0.465	0.0711
"	160	102	0.895	0.477	0.0705
"	200	44	0.882	0.469	0.0697
969	135	53	0.884	0.468	0.0719
"	160	34	0.875	0.480	0.0713
"	200	17.4	0.862	0.499	0.0704
Vel.S	135	18.2	0.838	0.482	0.0761
"	160	13.1	0.829	0.495	0.0754
"	200	8.5	0.815	0.515	0.0745

APPENDIX II.

Note on the Wilson method of plotting.

This method was used by Shert, and by Gardner & Siller, to find the shell-side heat transfer coefficient h , from measurements of overall heat transfer, and shell side mass flow rate W . It avoids making difficult tube surface temperature measurements, or calculating tube side coefficients.

The method is as follows. For a given heat exchanger, a series of tests is made, keeping the tube-side flow rate constant, while the shell-side flow rate W is varied, so that the combined coefficient h_1 for the tube wall, scale, and tube-side fluid remains substantially constant, while the shell-side coefficient h varies. It is then assumed that h is proportional to W^n , a value for n also being assumed. (More strictly h is proportional to W^n/μ , but this refinement is not relevant here.) Now

since
$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h} \quad (1)$$

it follows that

$$\frac{1}{U} = \frac{1}{h_1} + \frac{C}{W^n} \quad (2)$$

U and W being known, and h_1 constant. The Wilson method is to plot $1/U$ against $1/W^n$, and provided the correct law has been assumed for h , the plot gives a straight line which, at $1/W^n = 0$, intersects the $1/U$ axis at the ordinate $1/h_1$. Knowing $1/h_1$, h may then be determined from equation (1). Finally by plotting h logarithmically against W , a straight line of slope n will be obtained.

It is found, however, that by assuming a value for n which is slightly in error, considerable errors in h can result, while the final logarithmic plot will have a slope equal to the wrongly assumed value of n , thus apparently confirming it. This will be illustrated by two hypothetical examples, in which the proper relation between h and W is given by $h = CW^{0.67}$, while the relation is erroneously assumed to be $h = CW^{0.6}$.

Example 1: h large compared with h_1 .

Data		Correct Analysis				Incorrect Analysis		
W	U	$\frac{1}{W^{0.67}}$	$\frac{1}{U}$	$\frac{1}{h}$	h	$\frac{1}{W^{0.6}}$	$\frac{1}{h}$	h
87	222	0.05	0.0045	0.0025	400	0.0675	0.00263	380
122	250	0.04	0.0040	0.0020	500	0.0552	0.00213	470
189	286	0.03	0.0035	0.0015	667	0.0426	0.00163	614
345	333	0.02	0.0030	0.0010	1000	0.0296	0.00113	885
960	400	0.01	0.0025	0.0005	2000	0.0158	0.00063	1,590
from Fig 22, $1/h_1 = 0.002$						From Fig 22, $1/h_1 = 0.00187$		

It will be seen from Fig 22 that although the graph of $1/U$ against $1/w^{0.6}$ is practically linear, it gives the wrong value of h_1 . Furthermore, when the corresponding values of h are plotted logarithmically against W in Fig 23 a (nearly) straight line having the wrong slope 0.6 is obtained, which is low by from 5 to 20%.

Example 2: h small compared with h_1 .

Data		Correct Analysis				Incorrect Analysis		
W	U	$\frac{1}{w^{0.67}}$	$\frac{1}{U}$	$\frac{1}{h}$	h	$\frac{1}{w^{0.6}}$	$\frac{1}{h}$	h
53.0	167	0.070	0.0060	0.0056	179	0.0913	0.0062	161
59.5	179	0.065	0.0056	0.0052	192	0.0854	0.0058	172
67.0	192	0.060	0.0052	0.0048	208	0.0795	0.0054	185
76.5	208	0.055	0.0048	0.0044	227	0.0735	0.0050	200
87.5	227	0.050	0.0044	0.0040	250	0.0675	0.0046	217
From Fig 22, $1/h_1 = 0.0004$					From Fig 22, $1/h_1 = -0.0002$			

In this case, the erroneous Wilson plot gives a negative h_1 which in itself is absurd. Again the logarithmic plot has the slope 0.6, and is low by from 10 to 13%, even although h is small compared with h_1 . Had a wider range of W been considered, from $W = 10$, to $W = 87.5$, the erroneous Wilson plot would still have been almost linear, and the values found for h would have been from 6 to 22% low.

When preparing the present correlation, Wilson plots were made for the results given by Tinker for each of his exchangers assuming $n = 0.6$. In some cases (exchanger No.3 for example) experimental scatter was so great that no straight line could be drawn through the points; in others, an approximate straight line could be drawn, but with such uncertainty as to be useless. In one case (exchanger No.11, runs 116 to 126) a straight line was obtained, which gave a negative h_1 . Eventually, Tinker's calculated values for h_1 were used.

APPENDIX III.Sample Calculations.Short's (1947) exchanger No.1, Run No.1.

Data:- Exchanger dimensions as shown in Fig 24.

Tube-side data.

Fluid	water
Mass flow rate	25,720 lb./hr.
Inlet temperature	68.63°F.
Outlet temperature	75.18°F.

Shell-side data.

Fluid	water
Mass flow rate W	2,790 lb./hr.
Inlet temperature	137.53°F.
Outlet temperature	78.39°F.
Heat transfer coefficient h	344 B.Th.U/ft ² .hr.°F.
(Determined by Short, using Wilson plot).	

Overall data.

Log mean temperature difference	28.36°F.
Overall heat transfer coefficient	242.0 B.Th.U/ft ² .hr.°F.

Calculation of areas.

$\frac{S}{D_1} = 0.404.$ C_1 from Fig 8 = 0.05.

Gross sectional area of shell = 30.04 sq. in.
 Gross area of baffle opening = 0.05 x 30.04 = 1.502 sq.in.
 Gross sectional area of tubes passing through baffle opening = 0.
 Net area of baffle opening = A_b = 1.502 sq.in.

$$A_m = \frac{\sum y}{n_y} \times \text{baffle spacing} = \frac{28.76}{5} \times 2.33 = 13.4 \text{ sq.in.}$$

$$A_p = \frac{\sum x}{n_x} \times \text{baffle spacing} = \frac{14.48}{6} \times 2.33 = 5.62 \text{ sq.in.}$$

$$\begin{aligned} A_c &= \text{Curved length of baffle edge} \times \frac{1}{2} (D_1 - D_3) \\ &= C_2 \text{ (from Fig 8)} \times D_1 \times \frac{1}{2} (D_1 - D_3) \\ &= 0.8 \times 6.185 \times 0.03 = 0.466 \text{ sq.in.} \end{aligned}$$

$$\frac{1}{A_{av}} = 48 \left(\frac{1}{A_b} + \frac{1}{A_m} + \frac{1}{A_p} \right) = 48(0.665 + 0.075 + 0.1779) = 44.1$$

where A_{av} is in sq. ft.

$$\frac{A_b}{A_b + A_c} = \frac{1.502}{1.968} = 0.764$$

Calculation of remaining dimensionless groups.

$$\text{Mean shell-side temperature} = \frac{137.53 + 78.39}{2} = 108^\circ\text{F.} = t_m$$

At this temperature, $\mu = 1.50 \text{ lb./ft.hr.}$, $k = 0.365 \text{ B.Th.U./ft.hr.}^\circ\text{F.}$,
and $Pr^{1/3} = 1.60$.

$$hd/k = 344 \times 0.625/12 \times 0.365 = 49.1$$

$$G_{av}d/\mu = wd/A_{av}\mu = 2790 \times 0.625/44.1 \times 12 \times 1.5 = 4.280$$

To find μ_s/μ , first find t_s , the tube surface temperature.

$$\text{Since } UA\theta_m = hA(t_m - t_s)$$

$$t_s = t_m - UA\theta_m/h = 108 - 242 \times 28.4/344 = 71.6^\circ\text{F.}$$

$$\text{Therefore } \mu_s = 2.33, \text{ and } (\mu_s/\mu) = (2.33/1.50) = 1.55$$

$$\frac{P-d}{P} = \frac{7-5}{7} = \frac{2}{7} = 0.286$$

$$\frac{D_1}{d} = \frac{6.185}{0.625} = 9.90$$

$$B = \frac{43 + (60 - 43) \cdot \left(\frac{2 \times 2.33}{60 - 43} \right)^{0.6}}{60} = 0.859.$$

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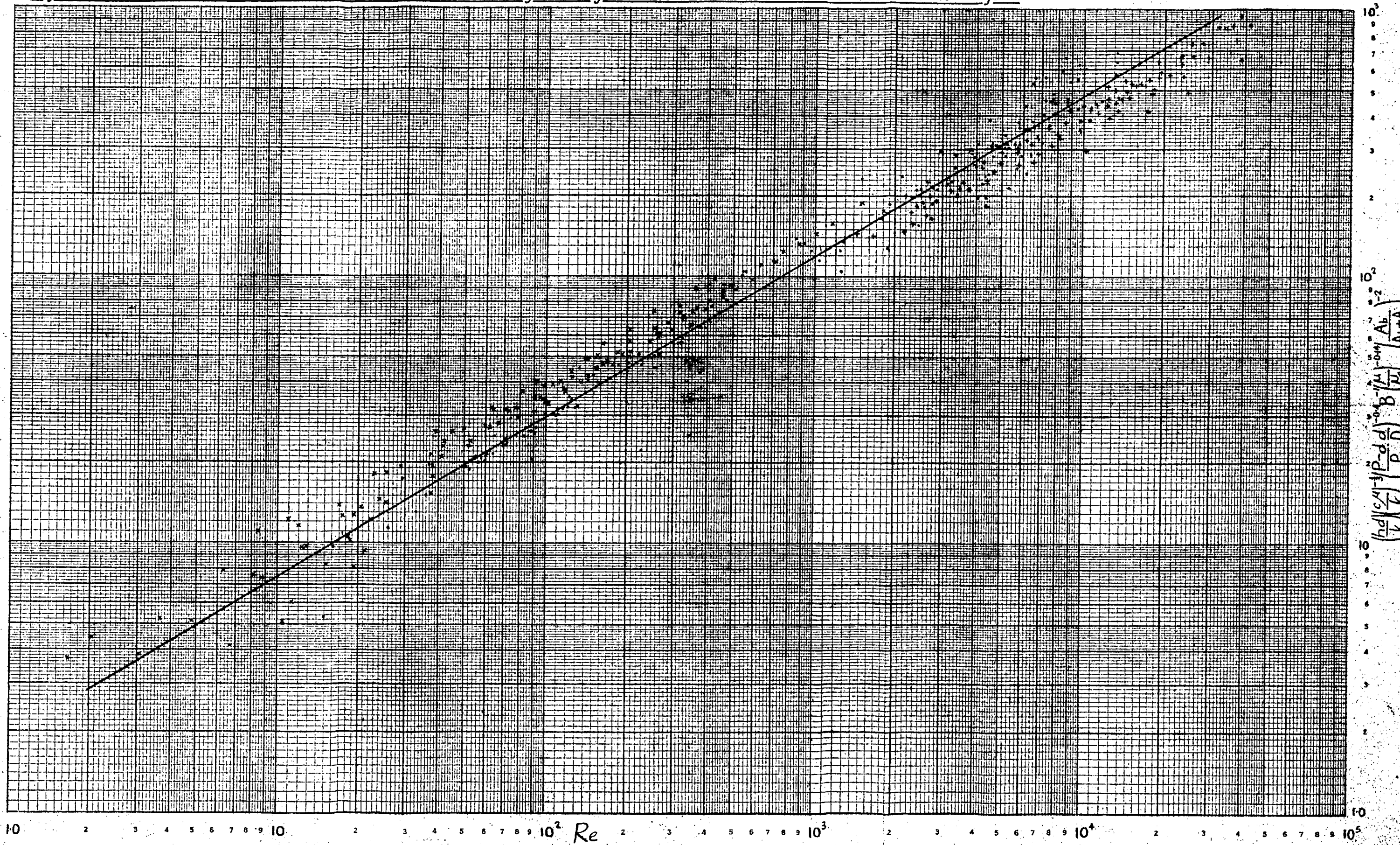
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Fig. 1. Correlation of Shell-side Heat Transfer in Segmentally baffled Shell and Tube Heat Exchangers.

+ Short.
 x Tinker.
 • Gardner
 and
 Siller.

LOG 3 CYCLES X 5 CYCLES

GRAPH SHEET No. 5835



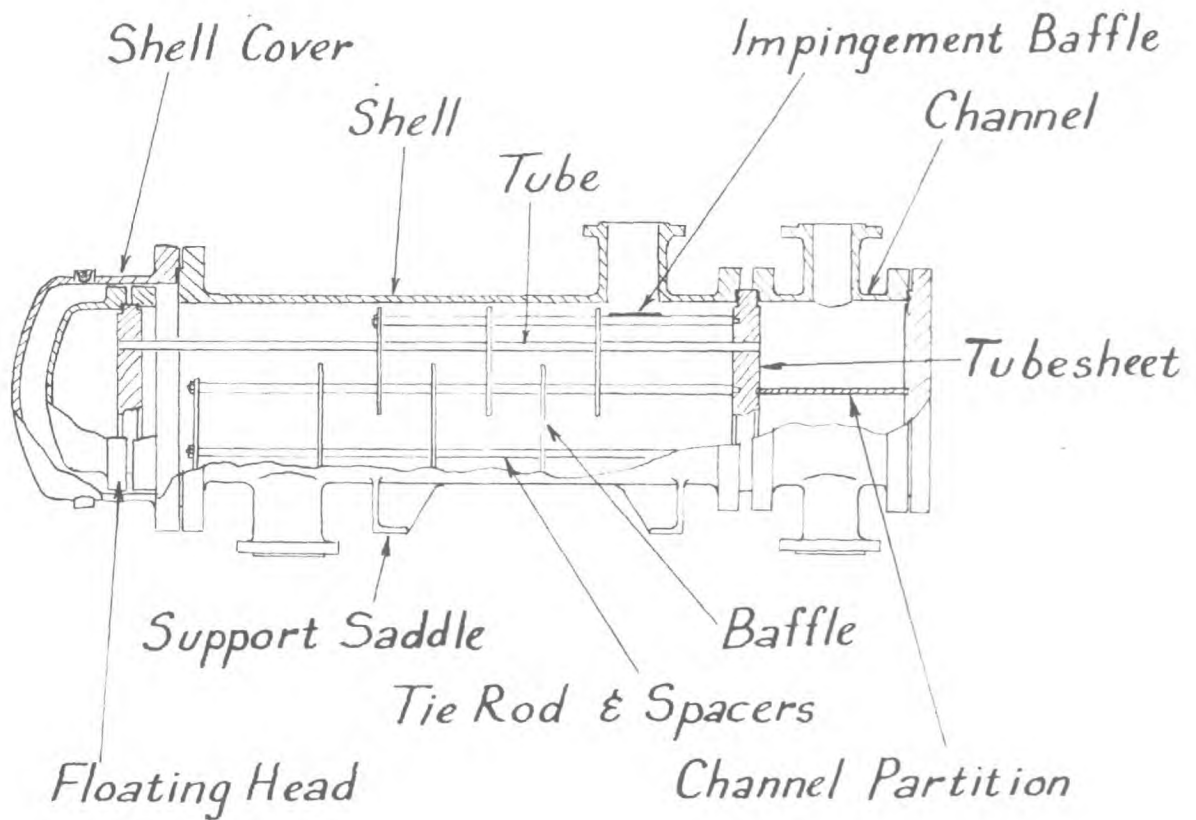


Fig.2. Typical cross baffled Shell & Tube
Heat Exchanger, having double pass
Tubes & single pass Shell.

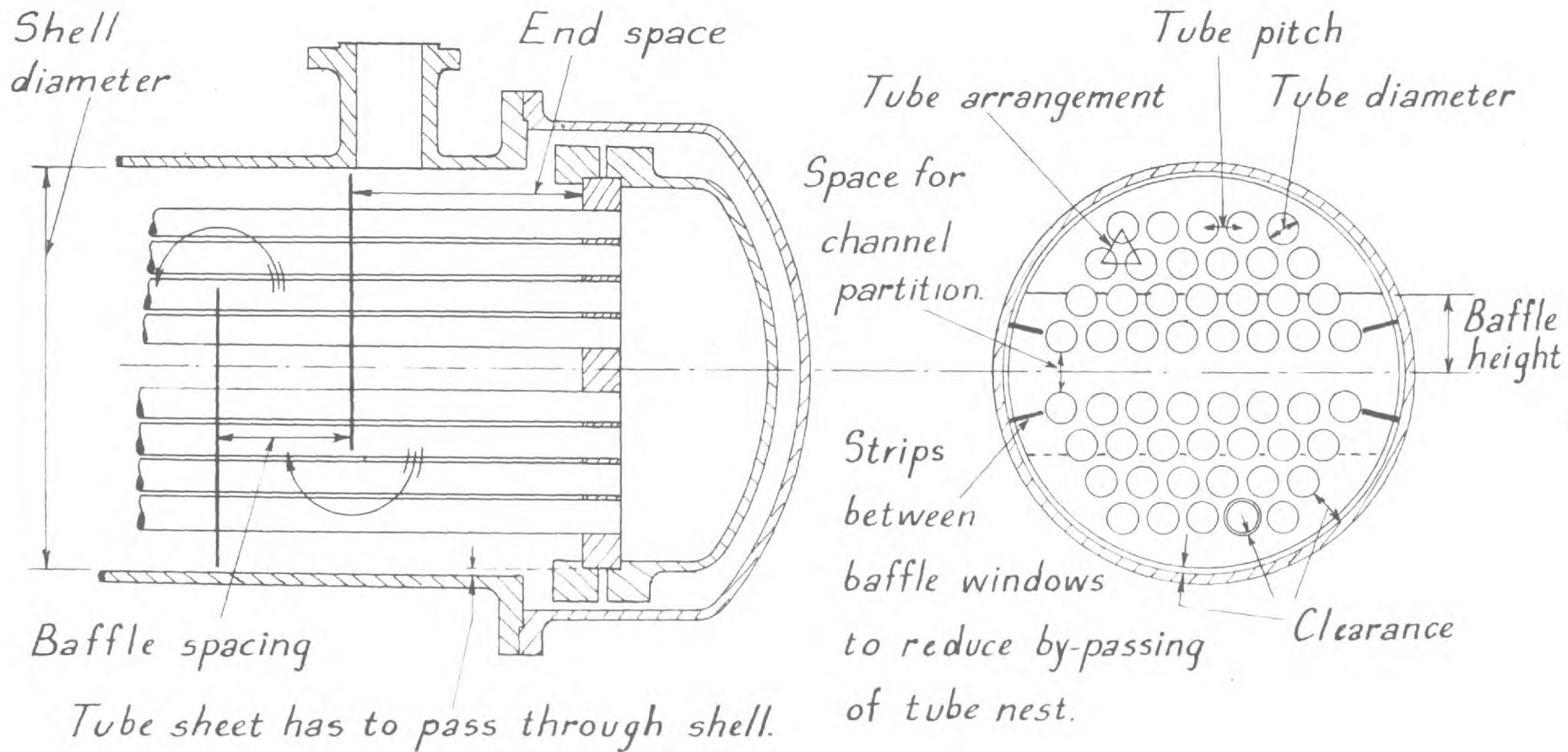


Fig.3. Factors which affect the Flow Pattern.

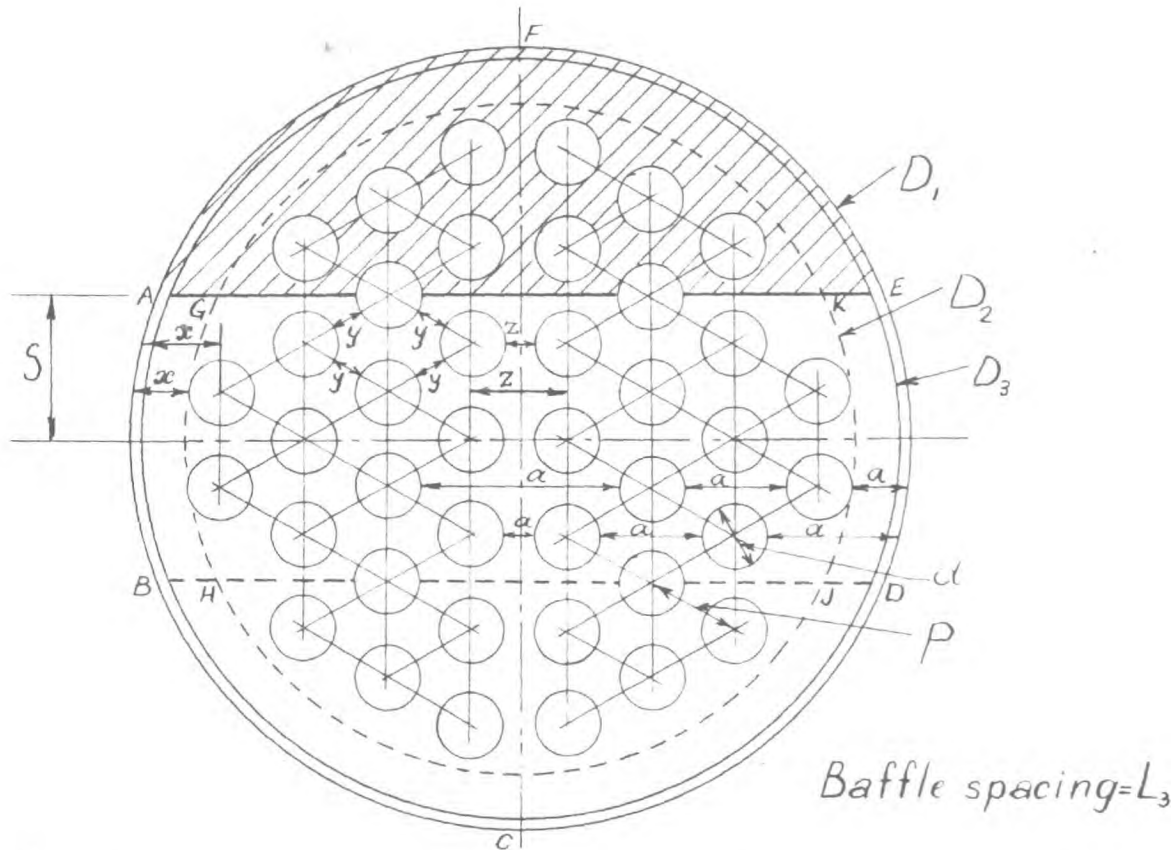


Fig.4. Calculation of flow areas. Equilateral triangular arrangement, with side of triangle parallel to flow.

A_b = shaded area = area of baffle window – cross sectional area of tubes passing through it.

$$A_m = L_3 (\text{mean width of shell between baffle edges.}) \\ = L_3 \times \text{area } ABDE / 2S.$$

$$A_c = \text{Length of curved baffle edge } ABCDE \times (D_1 - D_3) / 2.$$

$$A_p, \text{ when } P < 3.73d = L_3 (\sum x/n_x + \sum y/n_y + \sum z/n_z)$$

$$A_p, \text{ when } P > 3.73d = L_3 (\sum a/n_a)$$

where:-

n_x = number of rows of x 's between baffle edges (=4 in diagram,

n_y = " " " " y 's " " " (=6 " "

n_a = " " " " tubes " " " (=5 " "

n_z = " " " " z 's " " " (=5 " "

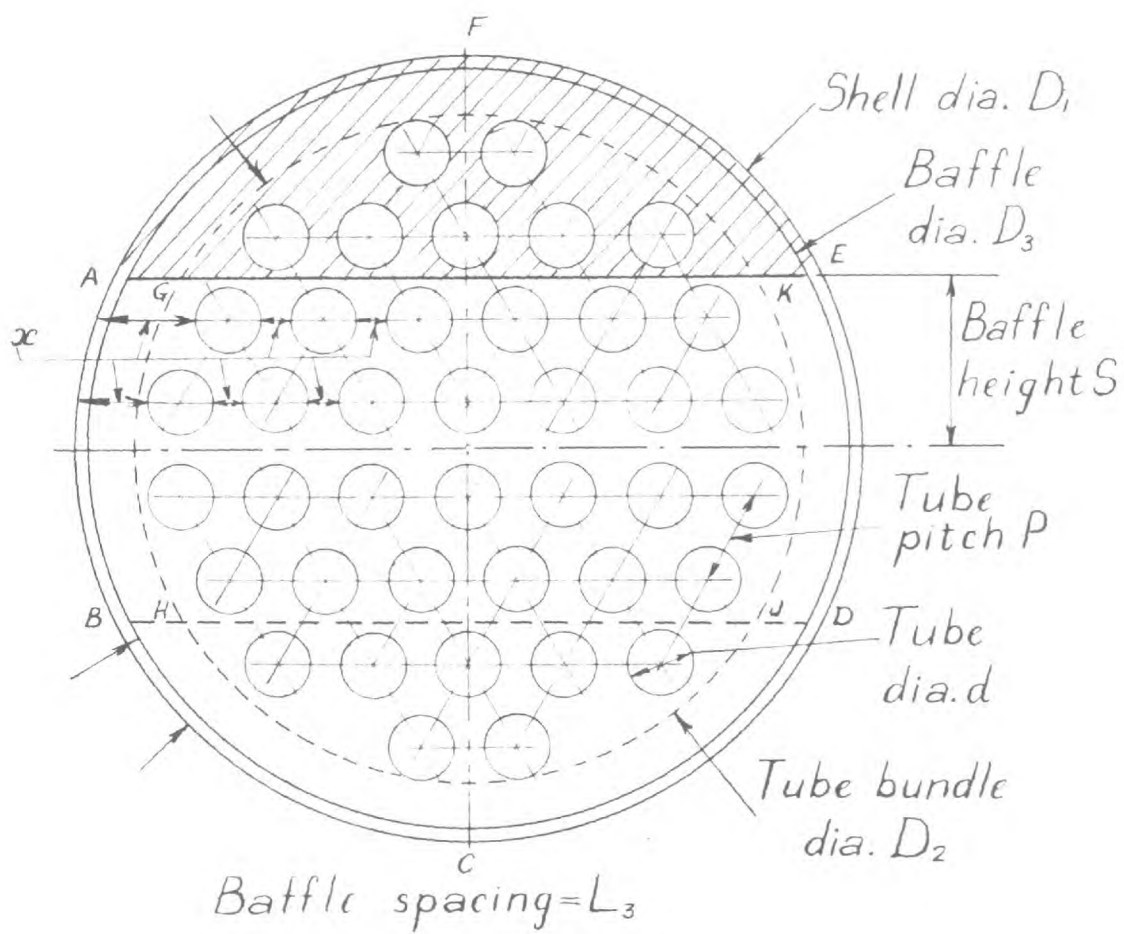


Fig.5. Calculation of flow areas. Equilateral triangular arrangement, with side of triangle perpendicular to flow.

A_b = shaded area = area of baffle window - cross sectional area of tubes passing through it.

$$A_m = L_3 (\text{mean width of shell between baffle edges}) \\ = L_3 \times \text{area } ABDE / 2S.$$

$$A_c = \text{Length of curved baffle edge } ABCDE \times (D_1 - D_3) / 2.$$

$$A_p = L_3 (\sum x / n_x)$$

where:-

n_x = number of rows of tubes between baffle edges (=4 in diagram)

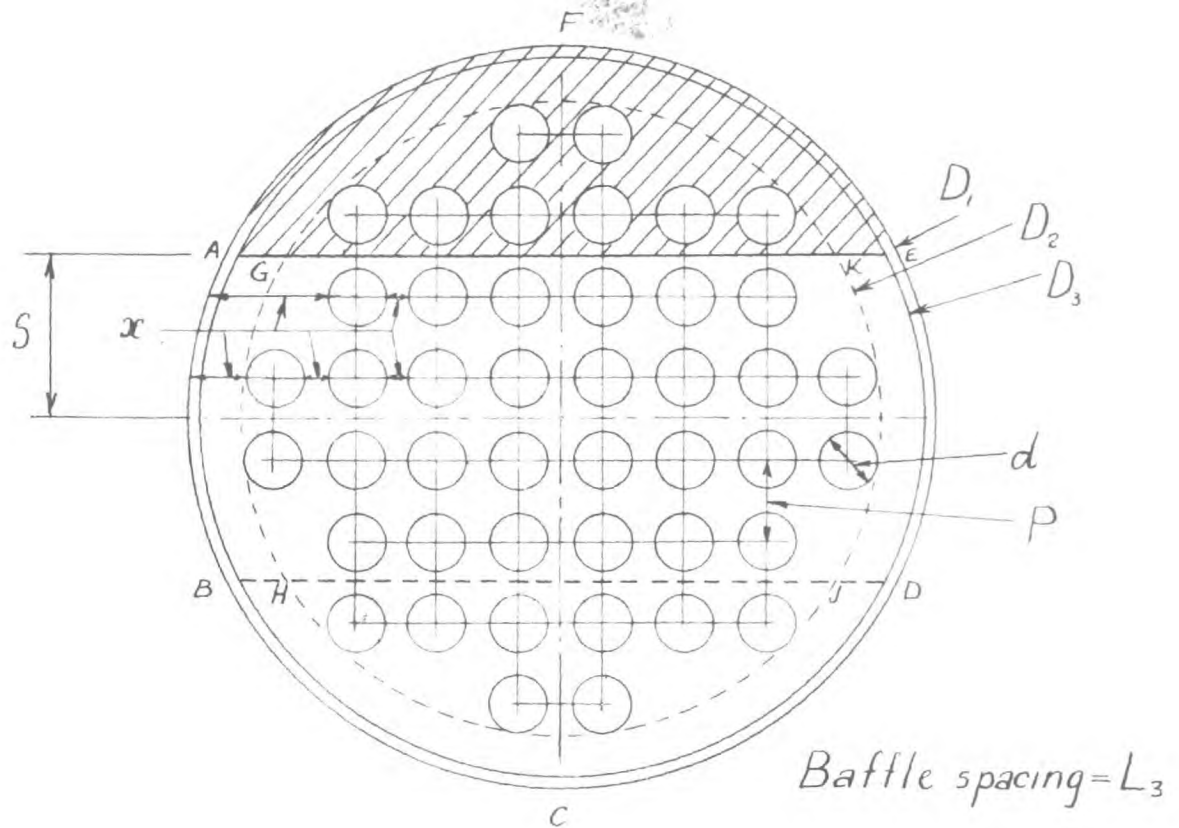


Fig. 6. Calculation of flow areas. In-line square arrangement.

A_b = shaded area = area of baffle window. — cross sectional area of tubes passing through it.

$$A_m = L_3 (\text{mean width of shell between baffle edges.}) \\ = L_3 \times \text{area ABDE} / 2S.$$

$$A_c = \text{Length of curved baffle edge ABCDE} \times (D_1 - D_3) / 2$$

$$A_p = L_3 (\sum x / n_x)$$

where:-

n_x = number of rows of tubes between baffle edges. (=4 in diagram)

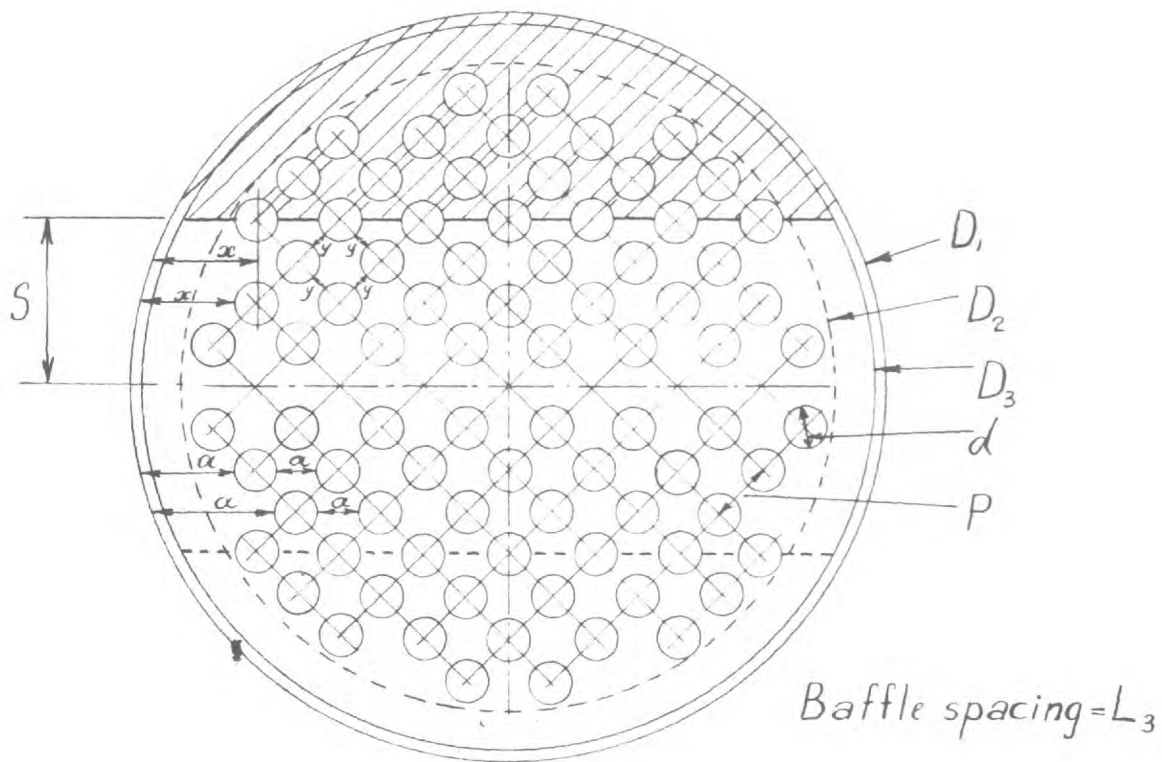


Fig. 7. Calculation of flow areas. Staggered square arrangement.

A_b = shaded area = area of baffle window - cross sectional area of tubes passing through it.

$$A_m = L_3 (\text{mean width of shell between baffle edges.})$$

$$= L_3 \times \text{area ABDE} / 2S.$$

$$A_c = \text{Length of curved baffle edge ABCDE} \times (D_1 - D_3) / 2.$$

$$A_p, \text{ when } P < 1.7d, = L_3 (\sum x/n_x + \sum y/n_y)$$

$$A_p, \text{ when } P > 1.7d, = L_3 (\sum a/n_a)$$

where:-

n_x = number of rows of x 's between baffle edges (= 6 in diagram)

n_y = " " " " y 's " " " (= 6 " ")

n_a = " " " " tubes " " " (= 6 " ")

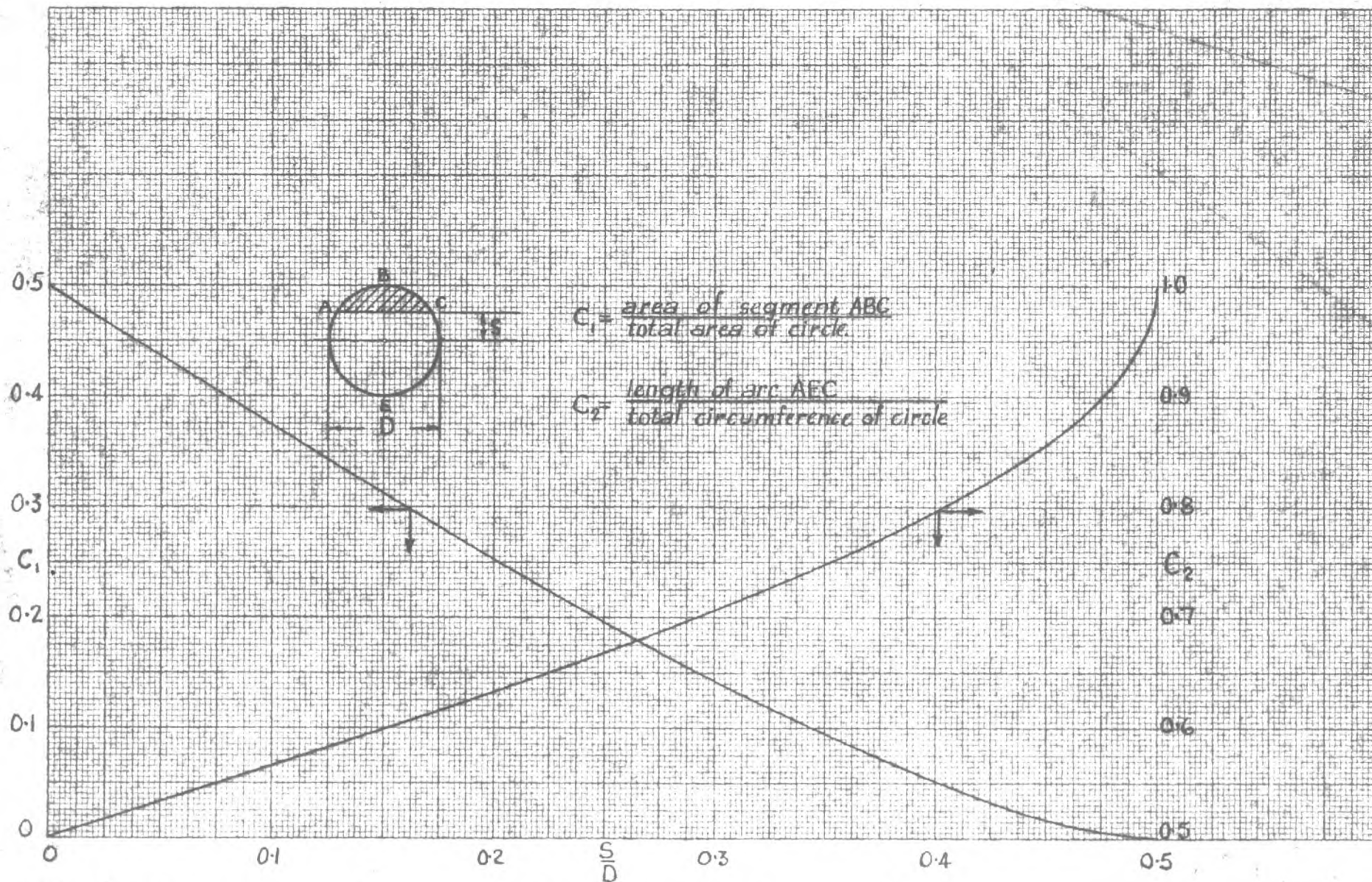


Fig. 8. Curves to facilitate calculation of areas A_b and A_c .

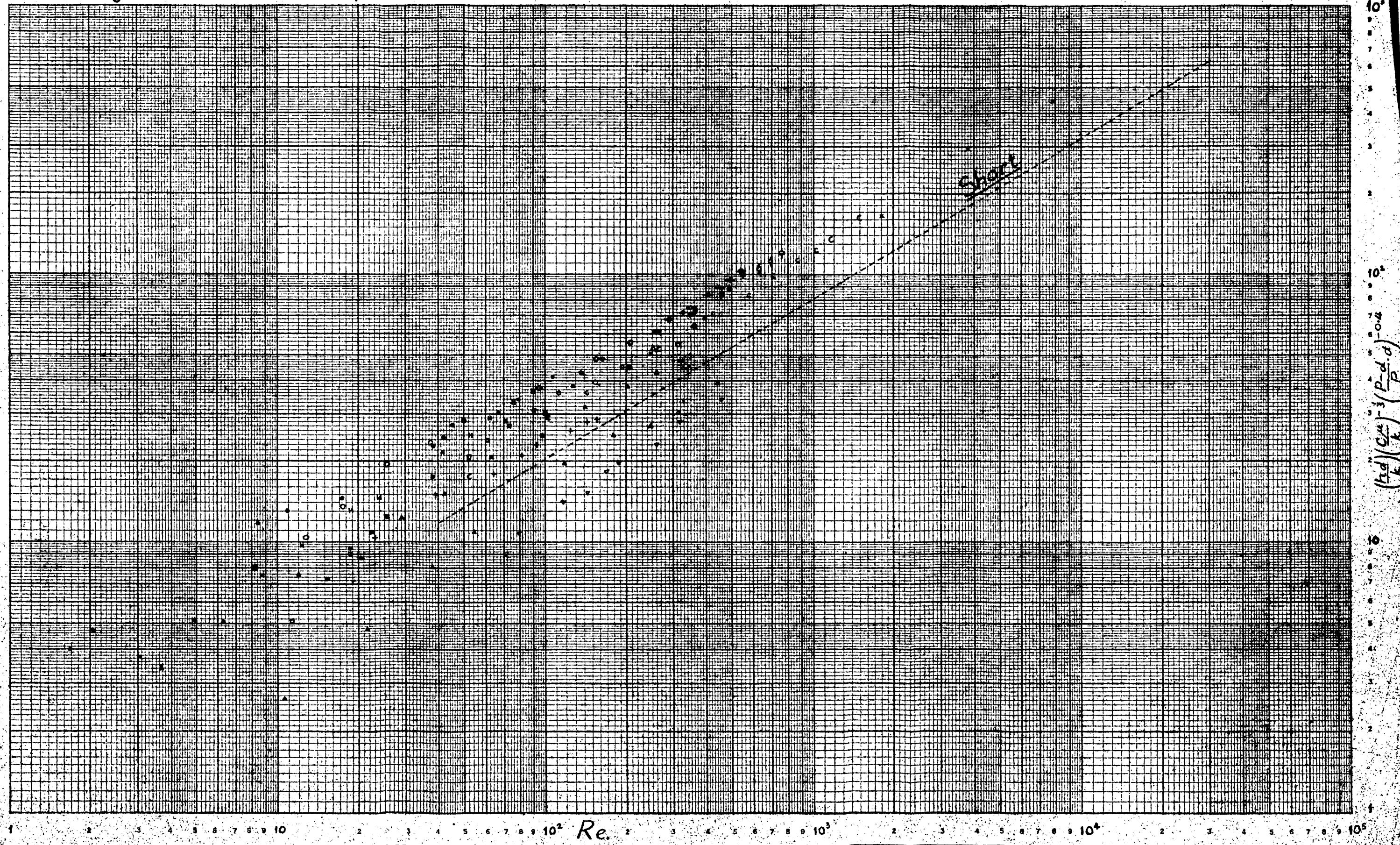
Fig. 9. Tinker's data plotted on Short (1943) basis.

Symbol Unit

- | | |
|---|----|
| x | 1 |
| ▽ | 2 |
| △ | 3 |
| ▲ | 4 |
| + | 5 |
| ○ | 6 |
| ● | 6 |
| □ | 7 |
| ■ | 7 |
| ◇ | 8 |
| ◊ | 8 |
| ◐ | 9 |
| ◑ | 10 |
| ⊖ | 11 |
| ⊕ | 11 |

LOG 3 CYCLES X 3 CYCLES

GRAPH SHEET No. 335



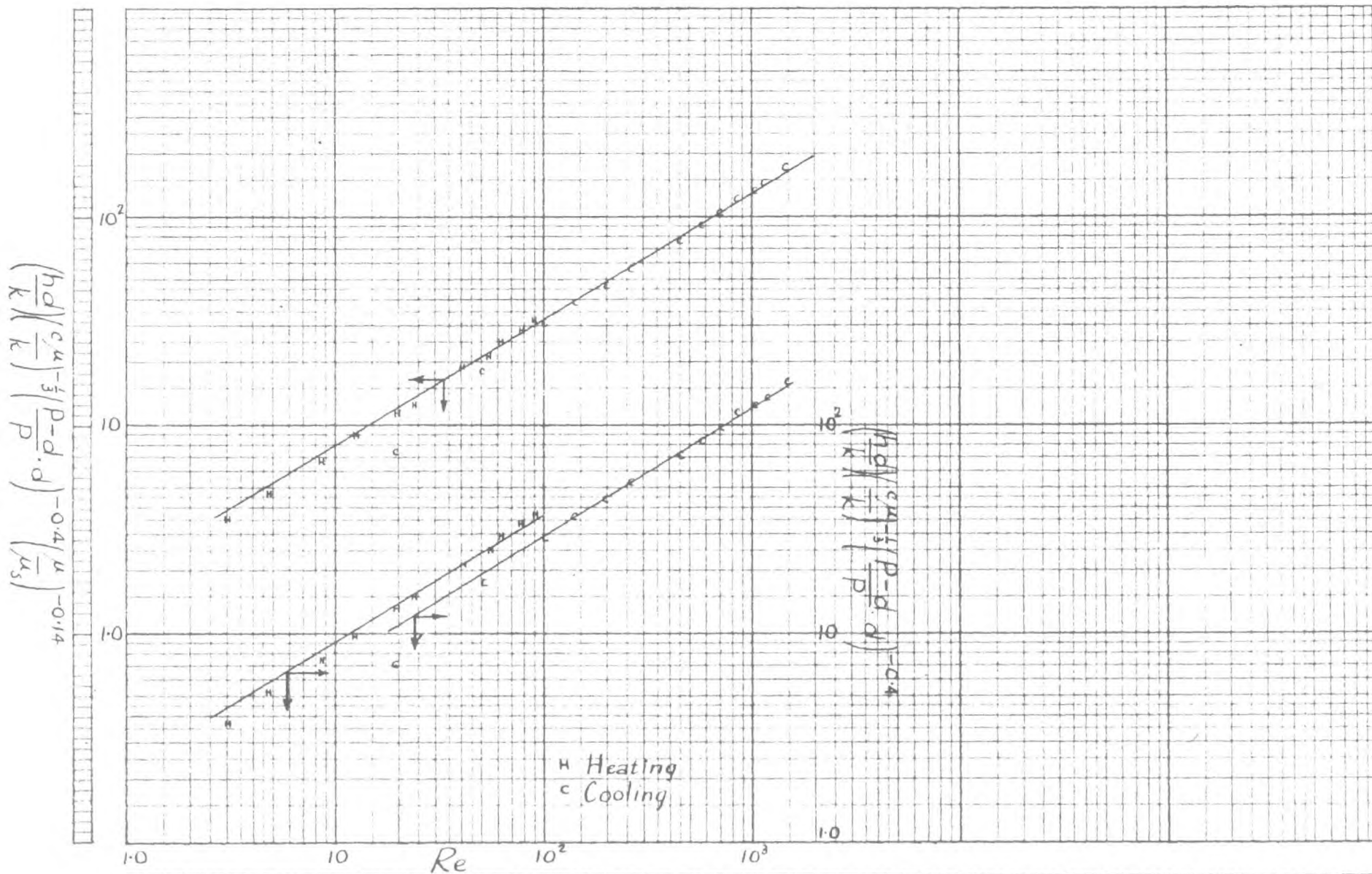
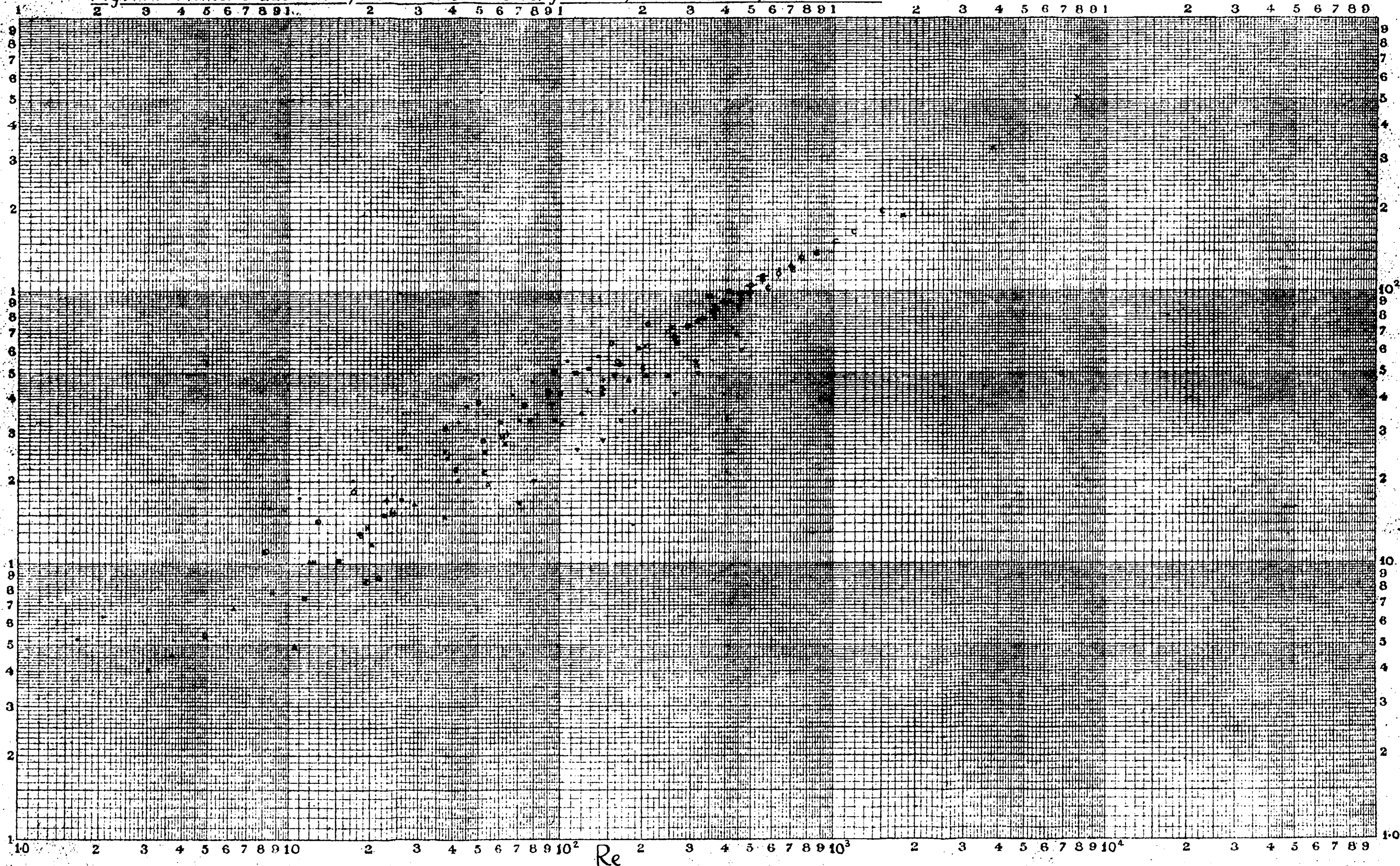


Fig.10. Trial of Exponent 0.14 for Viscosity Ratio, for Tinker's Unit 11.

Fig. 11. Tinker's data re-plotted with Viscosity Ratio, and End Space Factor.

Symbol Unit
 x 1
 ▽ 2
 △ 3
 ▲ 4
 + 5
 o 6
 ● 6
 □ 7
 ■ 7
 ○ 8
 ◊ 9
 ◊ 10
 C 11
 H 11



$$\left(\frac{\mu}{\mu_0}\right)^{0.14} B \left(\frac{\rho - \rho_0}{\rho_0}\right)^{-0.4} \left(\frac{k}{k_0}\right)^{-0.14}$$

DATA SHEET NO. 15 X - SOURCE: A. J. TINKER, J. CHEM. PHYS. 17, 104 (1949)

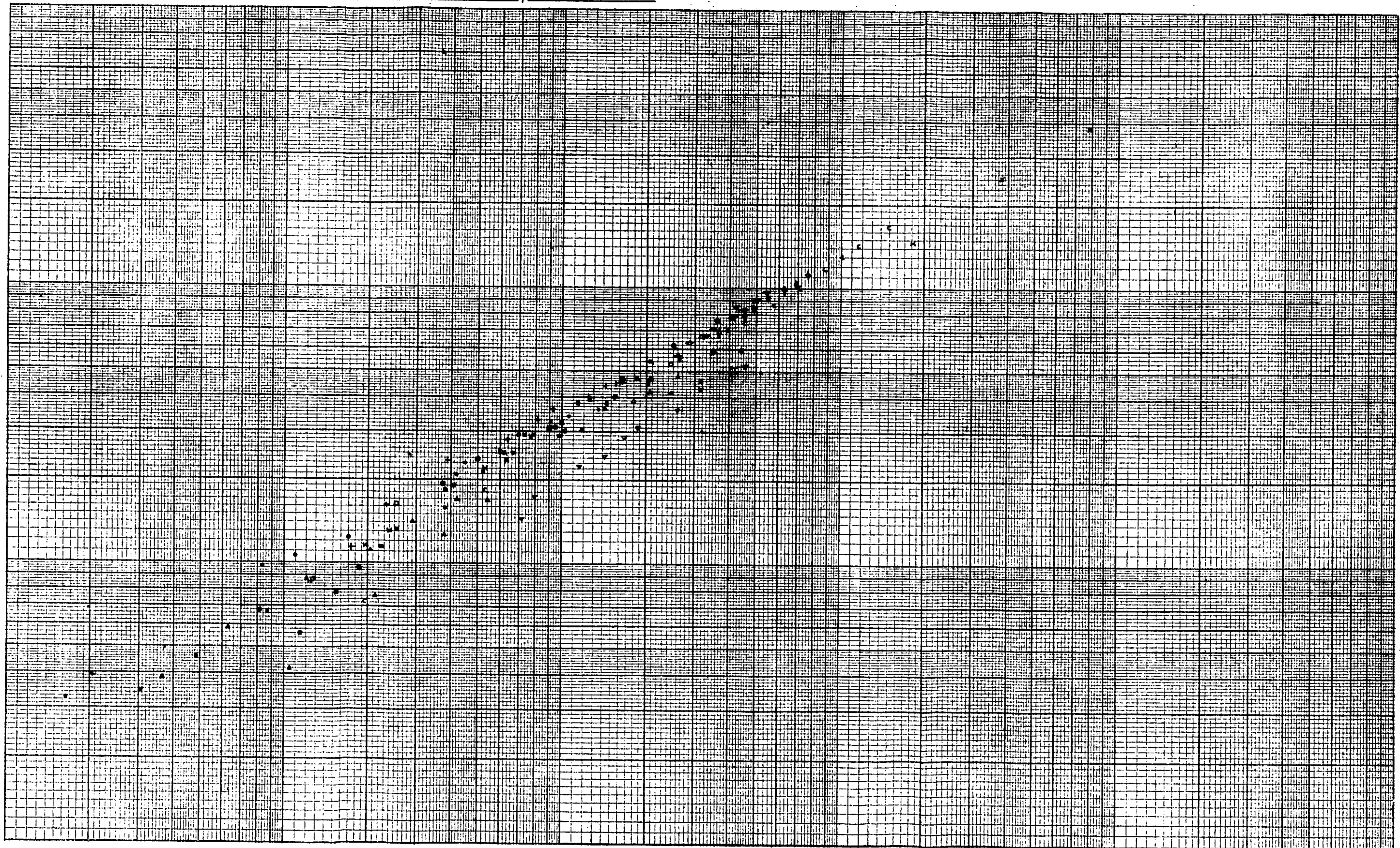
Fig. 12. Tinker's data re-plotted with $\frac{P-d \cdot d}{P \cdot d}$ in place of $\frac{P-d \cdot d}{P \cdot d}$

Symbol Unit

- x 1
- v 2
- Δ 3
- ▲ 4
- + 5
- o 6
- 6
- 7
- 7
- ρ 8
- φ 9
- φ 10
- 11
- ⊖ 11

LOG 3 CYCLES X 5 CYCLES

GRAPH SHEET No. 5835



10³
10²
10¹
10⁰
10⁻¹
10⁻²
10⁻³
10⁻⁴
10⁻⁵
10⁻⁶
10⁻⁷
10⁻⁸
10⁻⁹
10⁻¹⁰
10⁻¹¹
10⁻¹²
10⁻¹³
10⁻¹⁴
10⁻¹⁵
10⁻¹⁶
10⁻¹⁷
10⁻¹⁸
10⁻¹⁹
10⁻²⁰
10⁻²¹
10⁻²²
10⁻²³
10⁻²⁴
10⁻²⁵
10⁻²⁶
10⁻²⁷
10⁻²⁸
10⁻²⁹
10⁻³⁰
10⁻³¹
10⁻³²
10⁻³³
10⁻³⁴
10⁻³⁵
10⁻³⁶
10⁻³⁷
10⁻³⁸
10⁻³⁹
10⁻⁴⁰
10⁻⁴¹
10⁻⁴²
10⁻⁴³
10⁻⁴⁴
10⁻⁴⁵
10⁻⁴⁶
10⁻⁴⁷
10⁻⁴⁸
10⁻⁴⁹
10⁻⁵⁰
10⁻⁵¹
10⁻⁵²
10⁻⁵³
10⁻⁵⁴
10⁻⁵⁵
10⁻⁵⁶
10⁻⁵⁷
10⁻⁵⁸
10⁻⁵⁹
10⁻⁶⁰
10⁻⁶¹
10⁻⁶²
10⁻⁶³
10⁻⁶⁴
10⁻⁶⁵
10⁻⁶⁶
10⁻⁶⁷
10⁻⁶⁸
10⁻⁶⁹
10⁻⁷⁰
10⁻⁷¹
10⁻⁷²
10⁻⁷³
10⁻⁷⁴
10⁻⁷⁵
10⁻⁷⁶
10⁻⁷⁷
10⁻⁷⁸
10⁻⁷⁹
10⁻⁸⁰
10⁻⁸¹
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10⁻⁸⁵
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10⁻⁸⁷
10⁻⁸⁸
10⁻⁸⁹
10⁻⁹⁰
10⁻⁹¹
10⁻⁹²
10⁻⁹³
10⁻⁹⁴
10⁻⁹⁵
10⁻⁹⁶
10⁻⁹⁷
10⁻⁹⁸
10⁻⁹⁹
10⁻¹⁰⁰

10 2 3 4 5 6 7 8 9 10 10² Re 2 3 4 5 6 7 8 9 10³ 2 3 4 5 6 7 8 9 10⁴ 2 3 4 5 6 7 8 9 10⁵

$$\left(\frac{hd}{k} \right) \left(\frac{c\mu}{k} \right)^{-1/3} \left(\frac{P-d \cdot d}{P \cdot d} \right)^{-0.4} \left(\frac{\mu}{\mu_s} \right)^{-0.4} B \left(\frac{\mu}{\mu_s} \right)$$

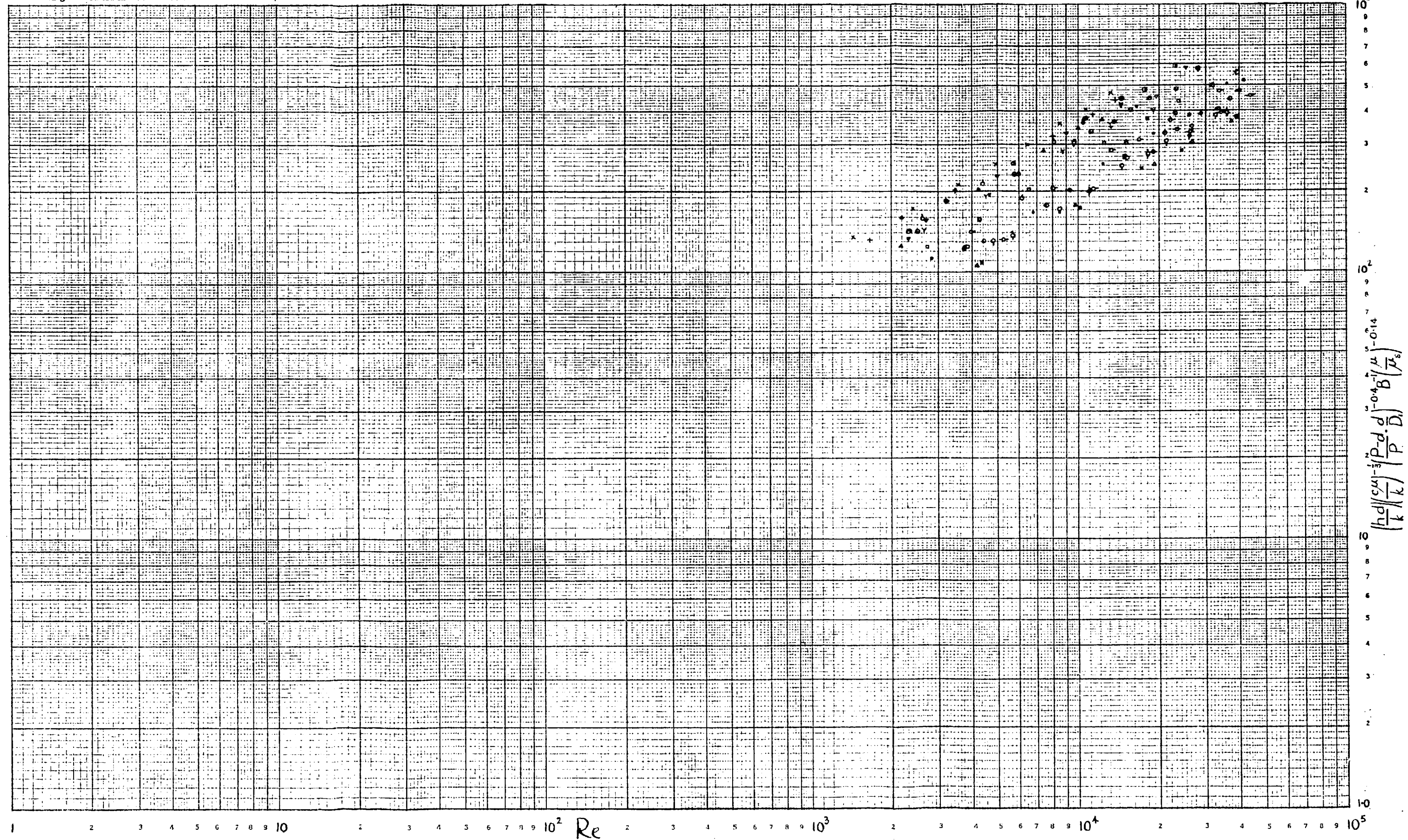
Fig.13. Short's 1947 data plotted on same basis as Fig.12.

Symbol Unit

- 1
- 2
- 3
- ◼ 4
- ◊ 6
- ◻ 7
- ⊙ 8
- ⊖ 9
- × 11
- ⊕ 12
- ⊖ 13
- ⊖ 14
- ⊖ 15
- ⊖ 16
- ⊖ 17
- ⊖ 18
- ⊖ 19
- ⊖ 20
- ⊖ 21
- ⊖ 22
- ⊖ 23

LOG 3 CYCLES X 5 CYCLES

GRAPH SHEET No 5835



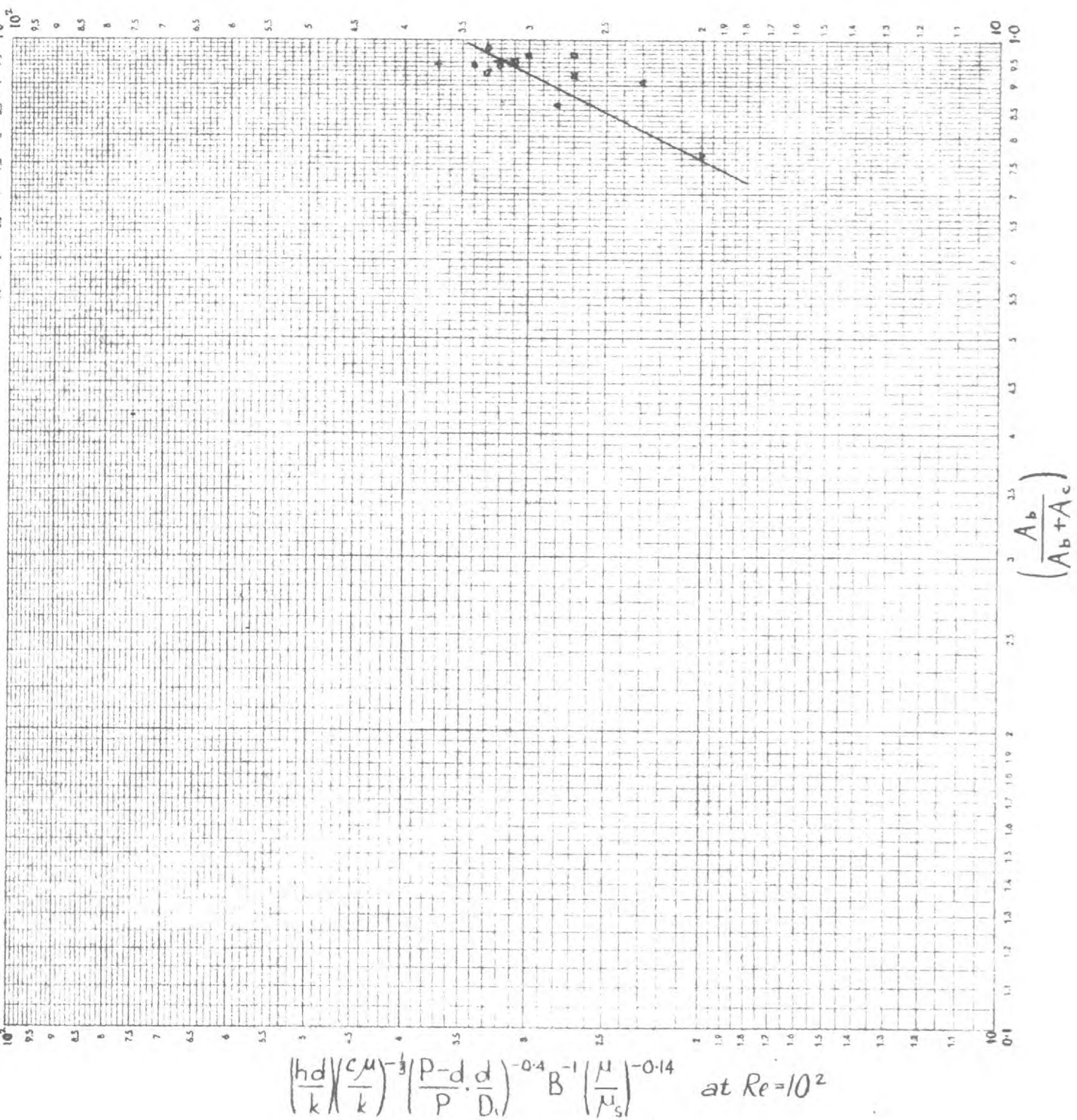


Fig.14. Determination of Clearance Ratio Exponent for Tinker's data.

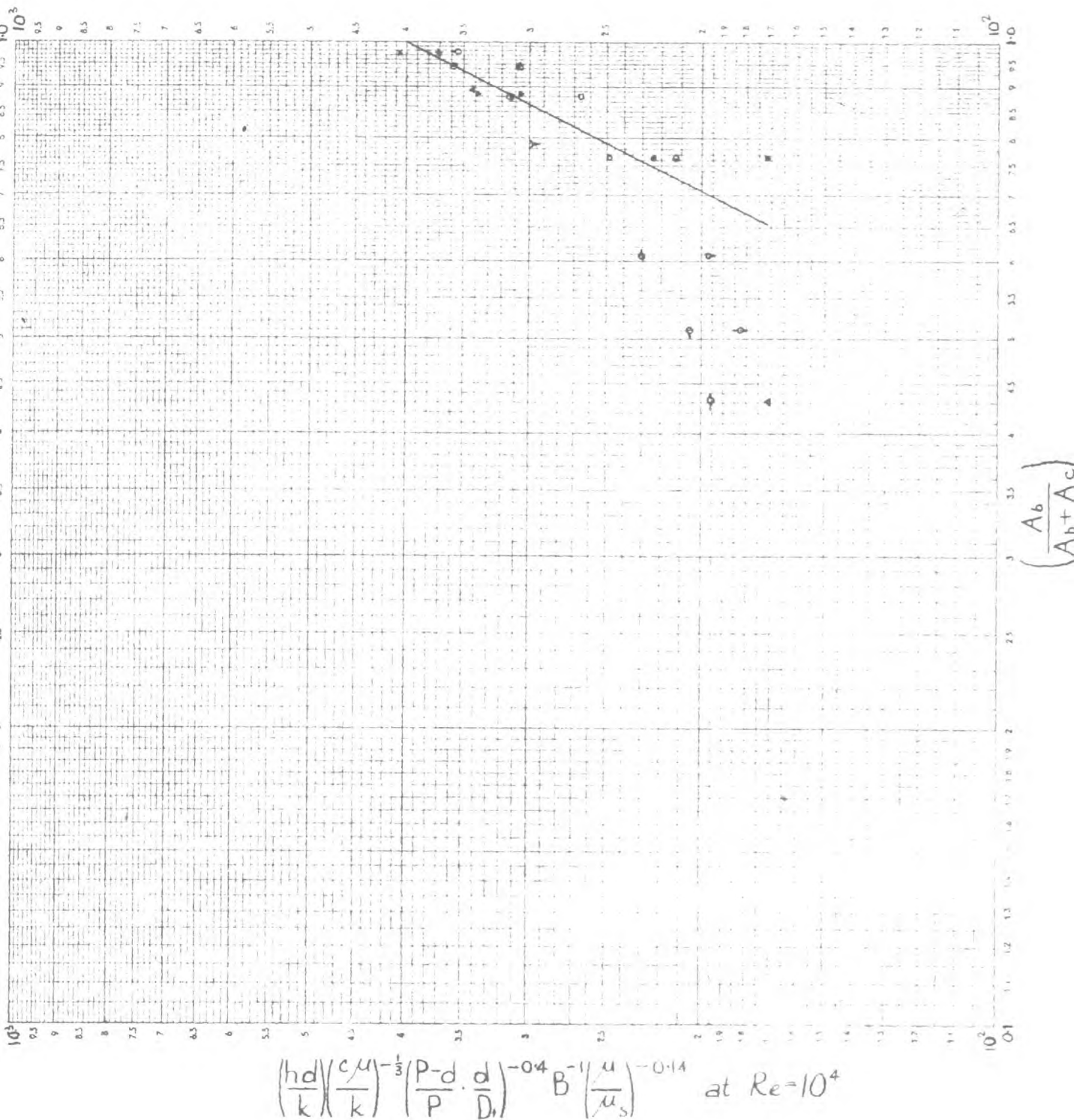


Fig. 15. Determination of Clearance Ratio Exponent
for Short's 1947 data.

Fig.16. Tinkers data re-plotted with Clearance Ratio - Final Correlation.

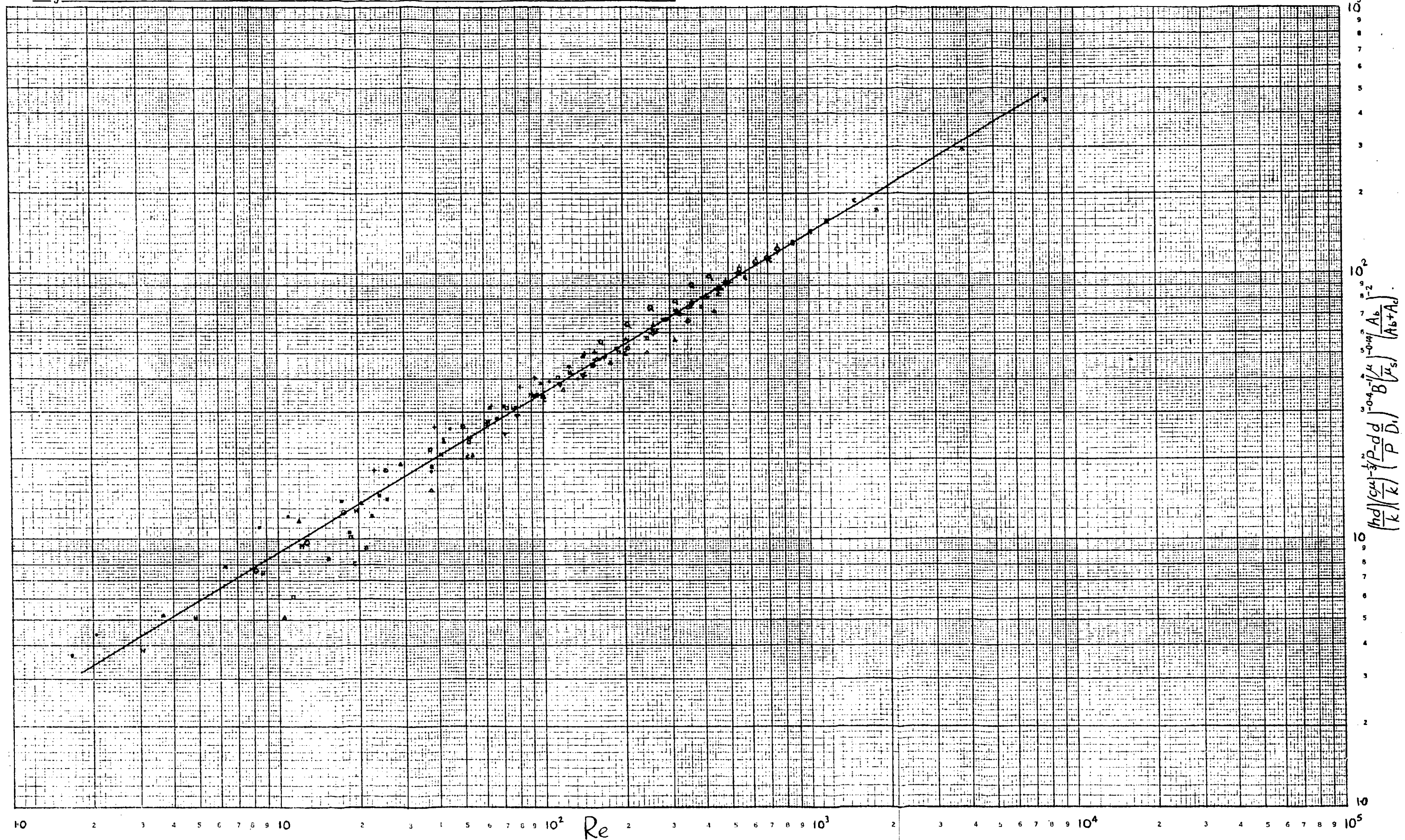


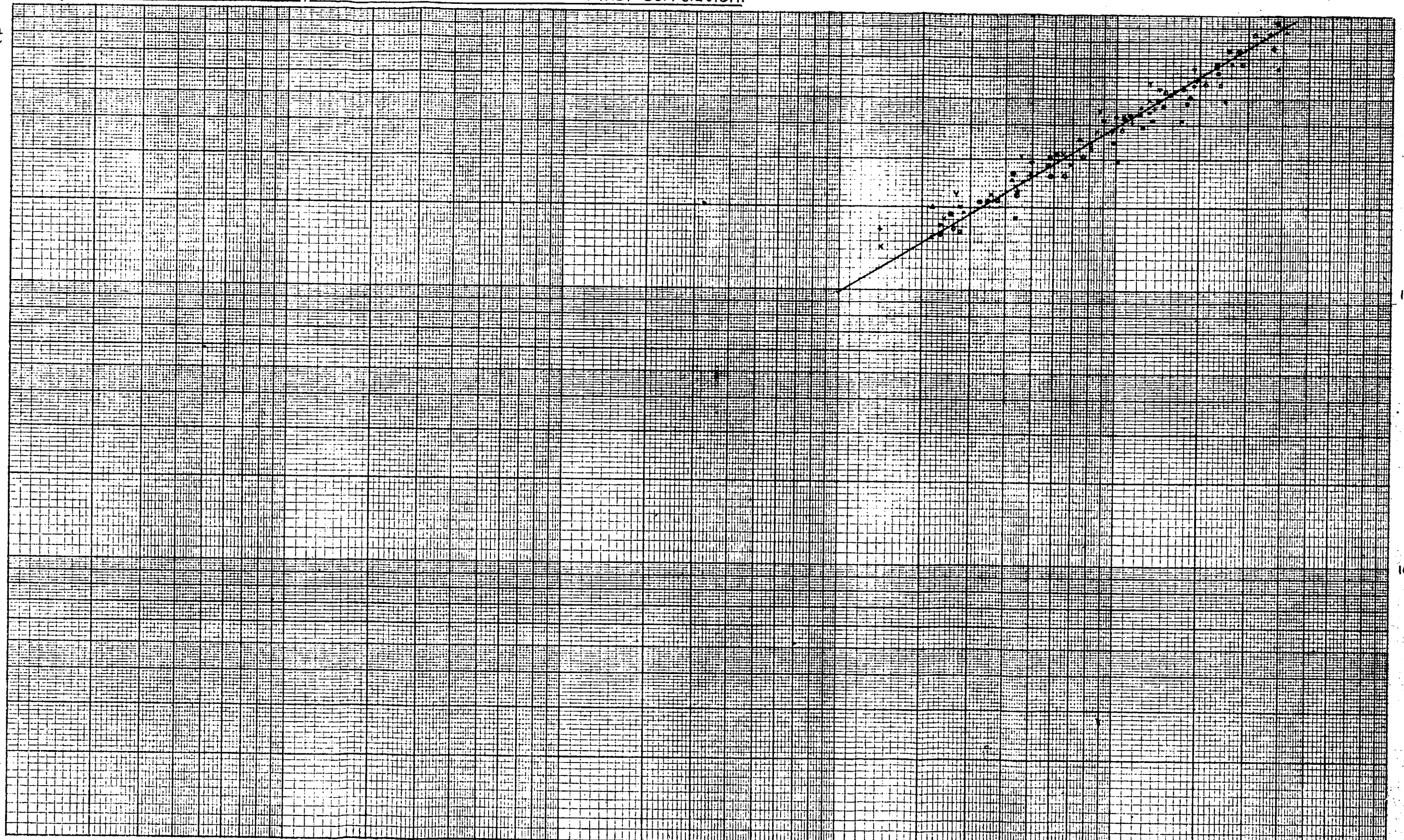
Fig. 17. Short's 1947 data re-plotted with Clearance Ratio - Final Correlation.

Symbol Unit

- 1
- 2
- ◊ 3
- 4
- ◊ 6
- ◊ 7
- ◊ 8
- ◊ 9
- x 11
- + 12
- ◊ 13
- ◊ 20
- ◊ 21
- ◊ 22
- ◊ 23

LOG 3 CYCLES X 5 CYCLES

GRAPH SHEET No. 5835



$$\left(\frac{hd}{k} \right)^{1/3} \left(\frac{P-d}{P} \right)^{1/3} \left(\frac{A_b}{A_b + A_c} \right)^{-0.14} B \left(\frac{A_b}{A_b + A_c} \right)^{-0.14}$$

1.0 2 3 4 5 6 7 8 9 10 2 3 4 5 6 7 8 9 10² Re 2 3 4 5 6 7 8 9 10³ 2 3 4 5 6 7 8 9 10⁴ 2 3 4 5 6 7 8 9 10⁵

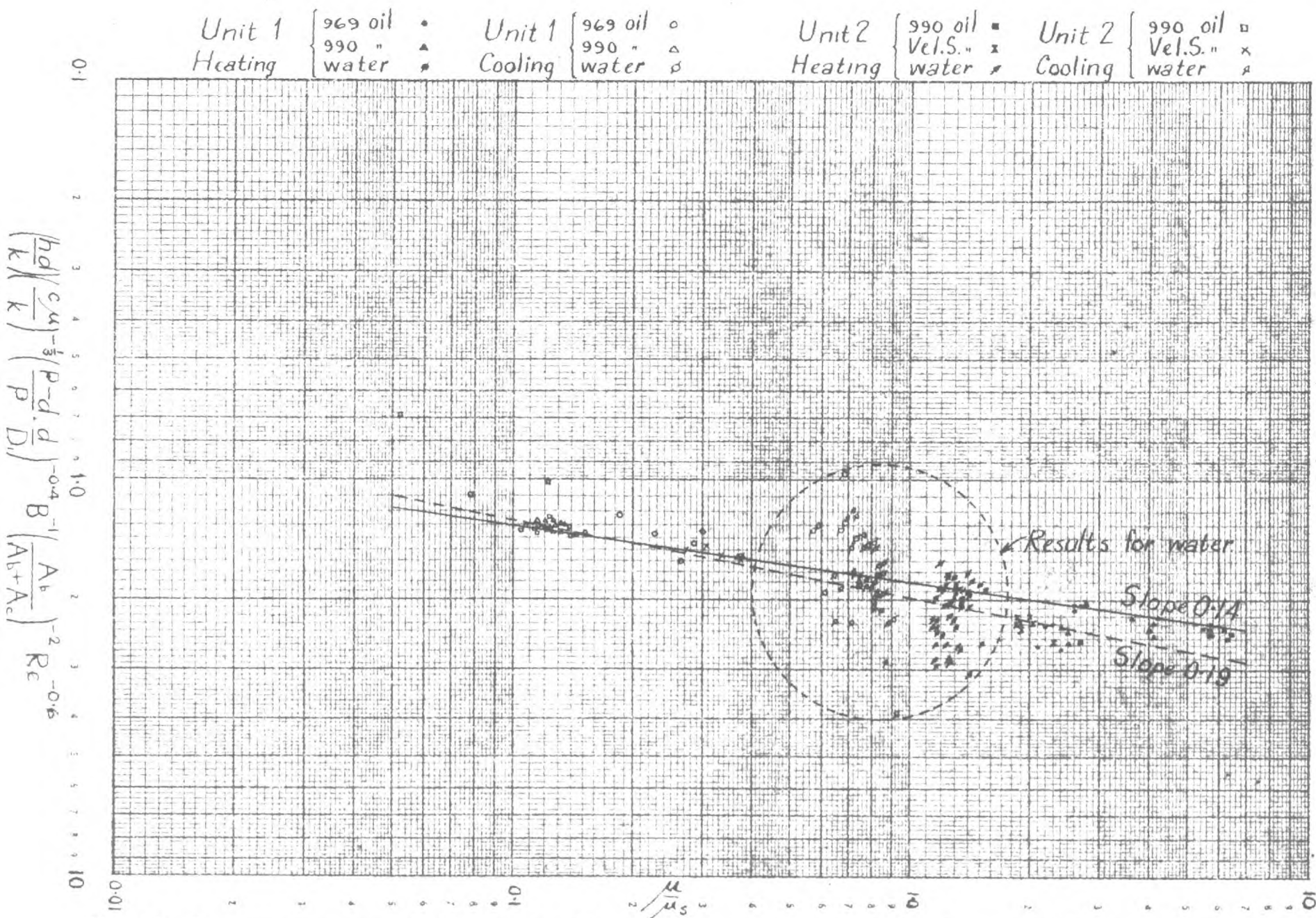


Fig.18. Determination of Viscosity Ratio Exponent for Gardner and Siller's data.

Fig. 19. Correlation of Gardner and Siller's data.

100 CYCLES X 3 CYCLE.

CHARTWELL 100 CYCLES X 3 CYCLE.

Unit 1

Heating

969 oil •

990 " ▲

water ♣

Cooling

969 oil ○

990 " △

water ♂

Unit 2

Heating

990 oil ■

Vel.S. " x

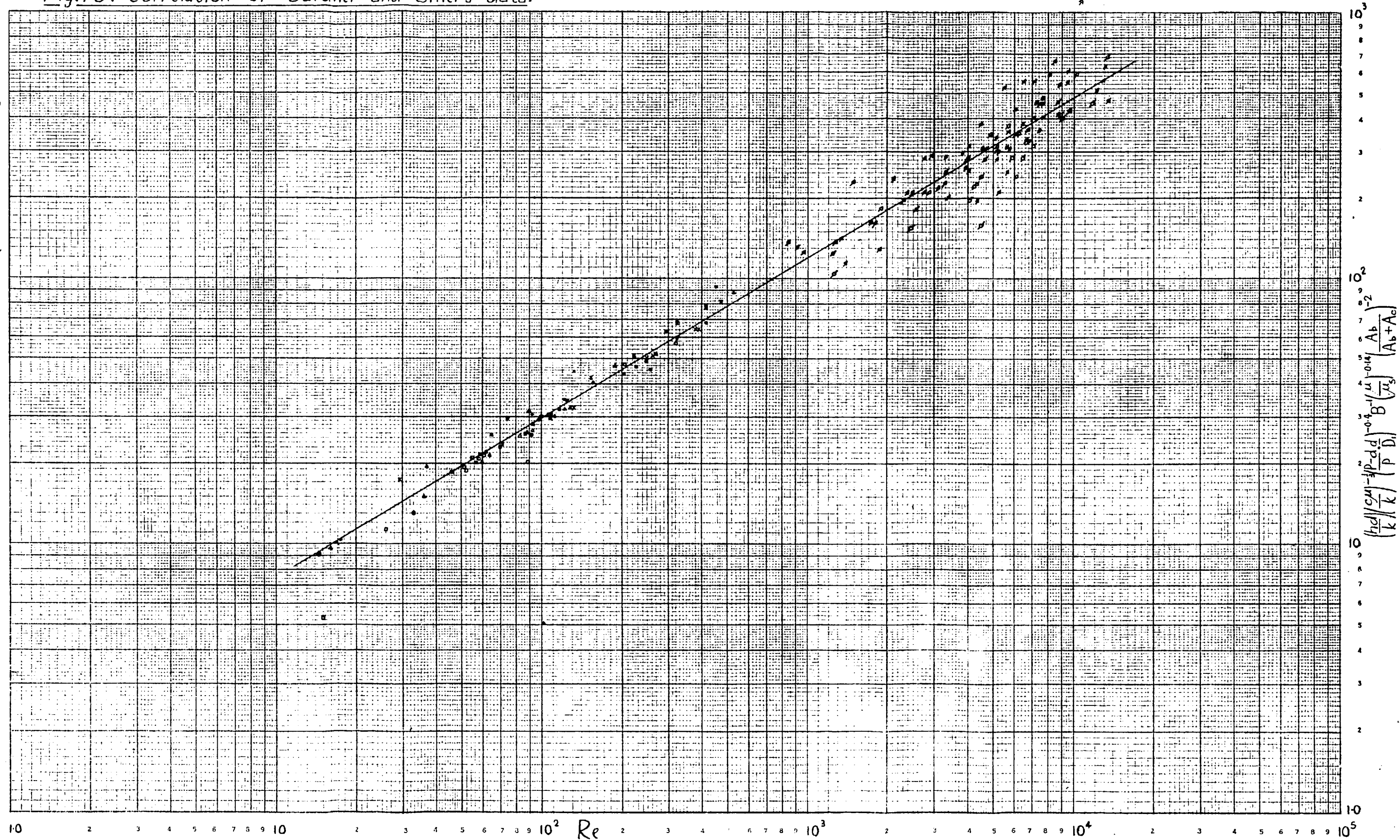
water ♣

Cooling

990 oil □

Vel.S. " x

water ♣



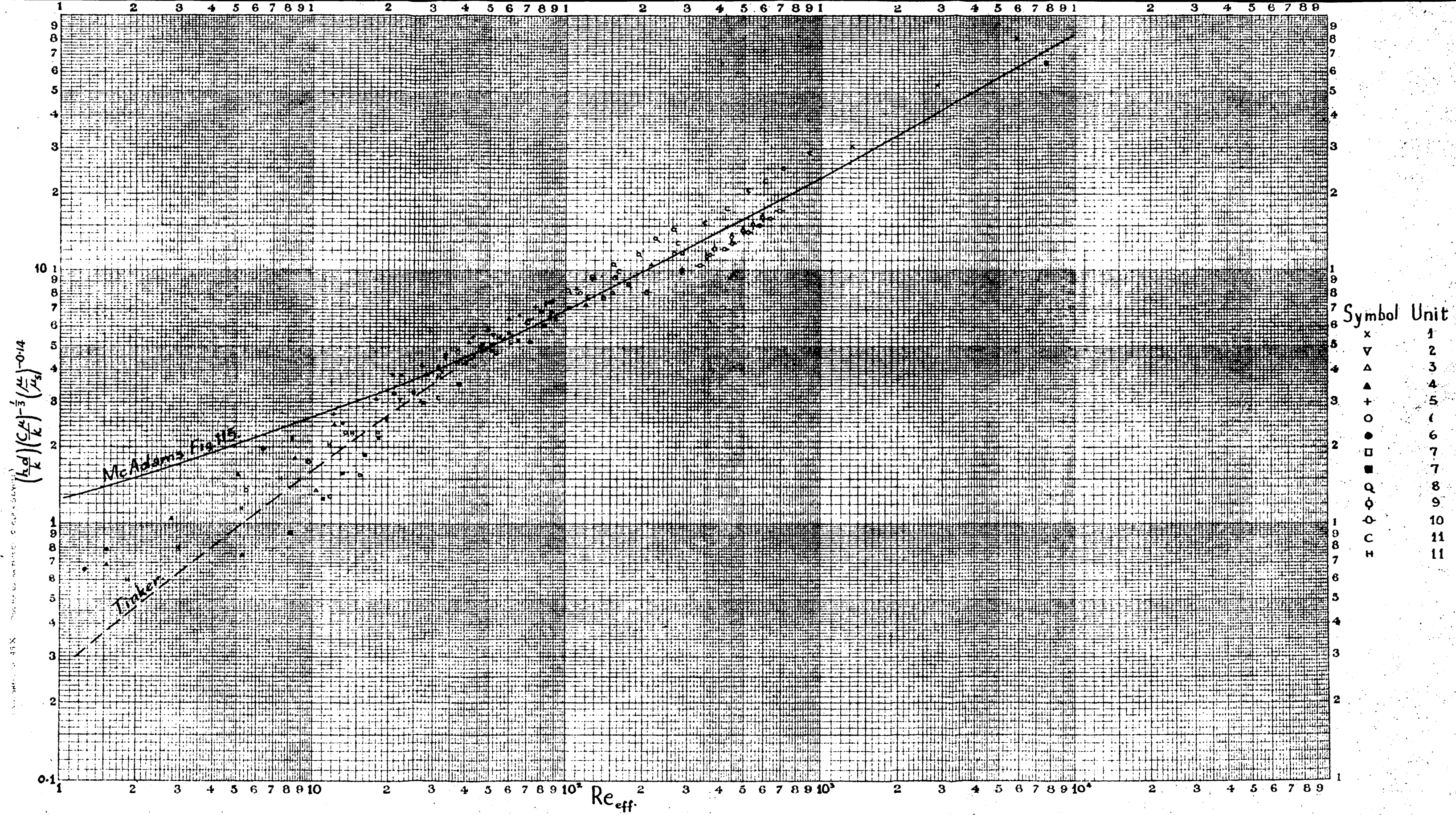
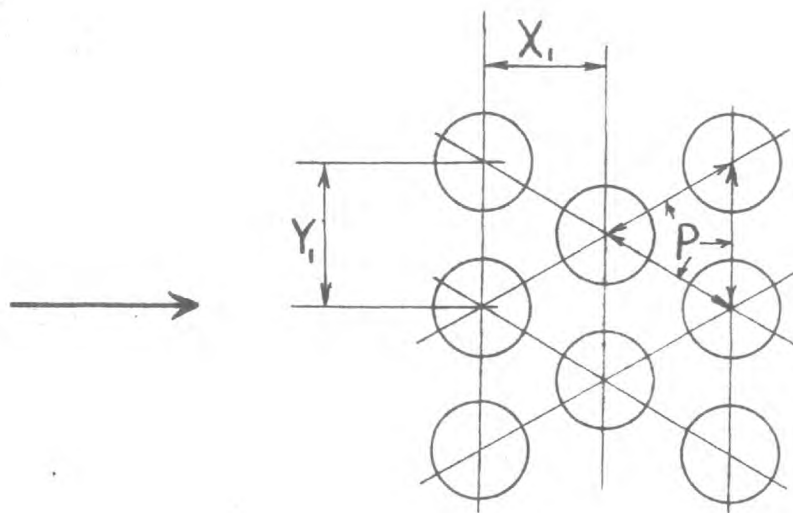


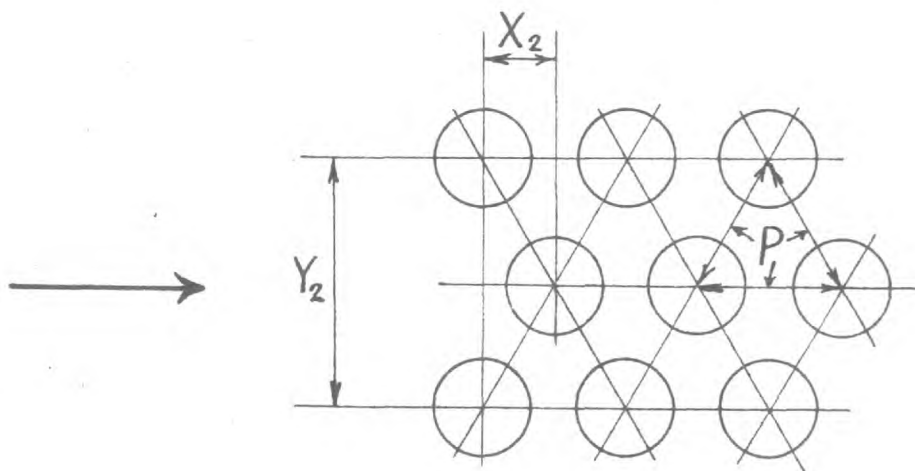
Fig. 20. Tinker's Correlation.



Case 1.

$$X_1 = 0.87P$$

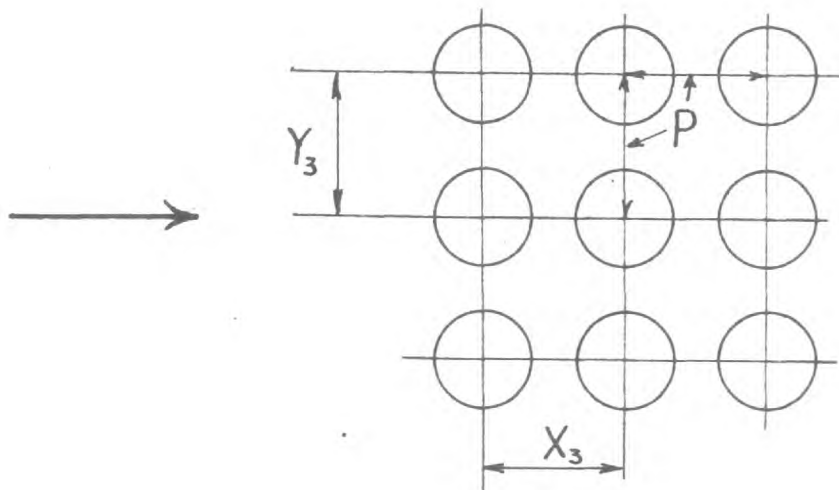
$$Y_1 = P$$



Case 2.

$$X_2 = P/2$$

$$Y_2 = 1.73P$$



Case 3.

$$X_3 = P$$

$$Y_3 = P$$

Fig.21. Typical tube arrangements.

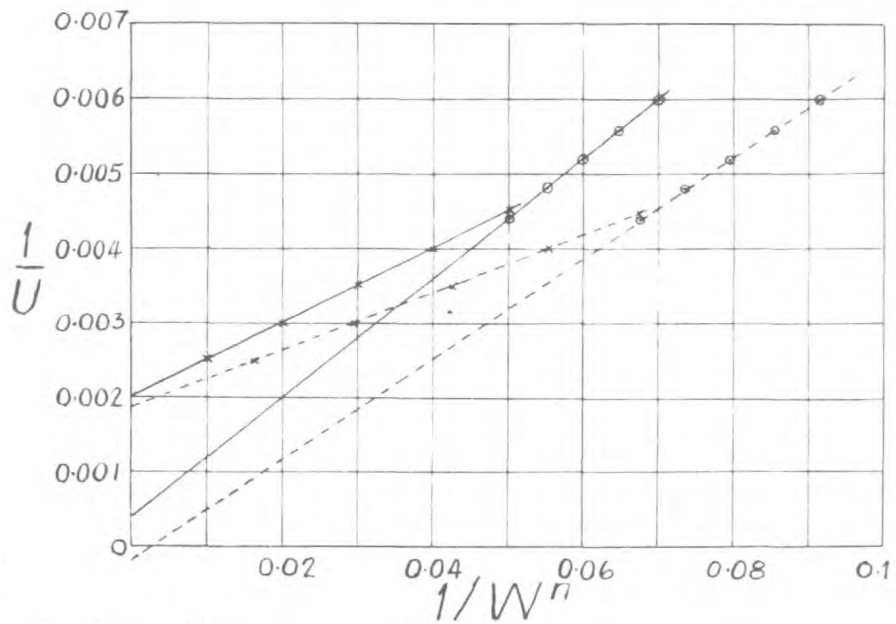


Fig.22. Wilson plot.— $n=0.67$ (correct);--- $n=0.6$
 $\times$ Example 1; $\circ$ Example 2.

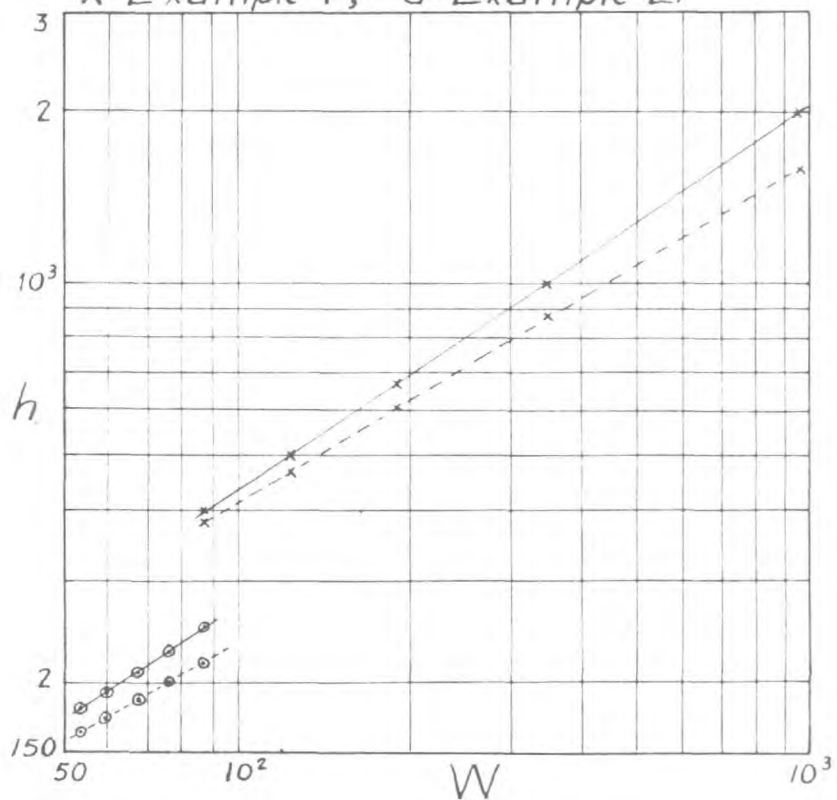


Fig.23. Corresponding logarithmic plots of h against W .

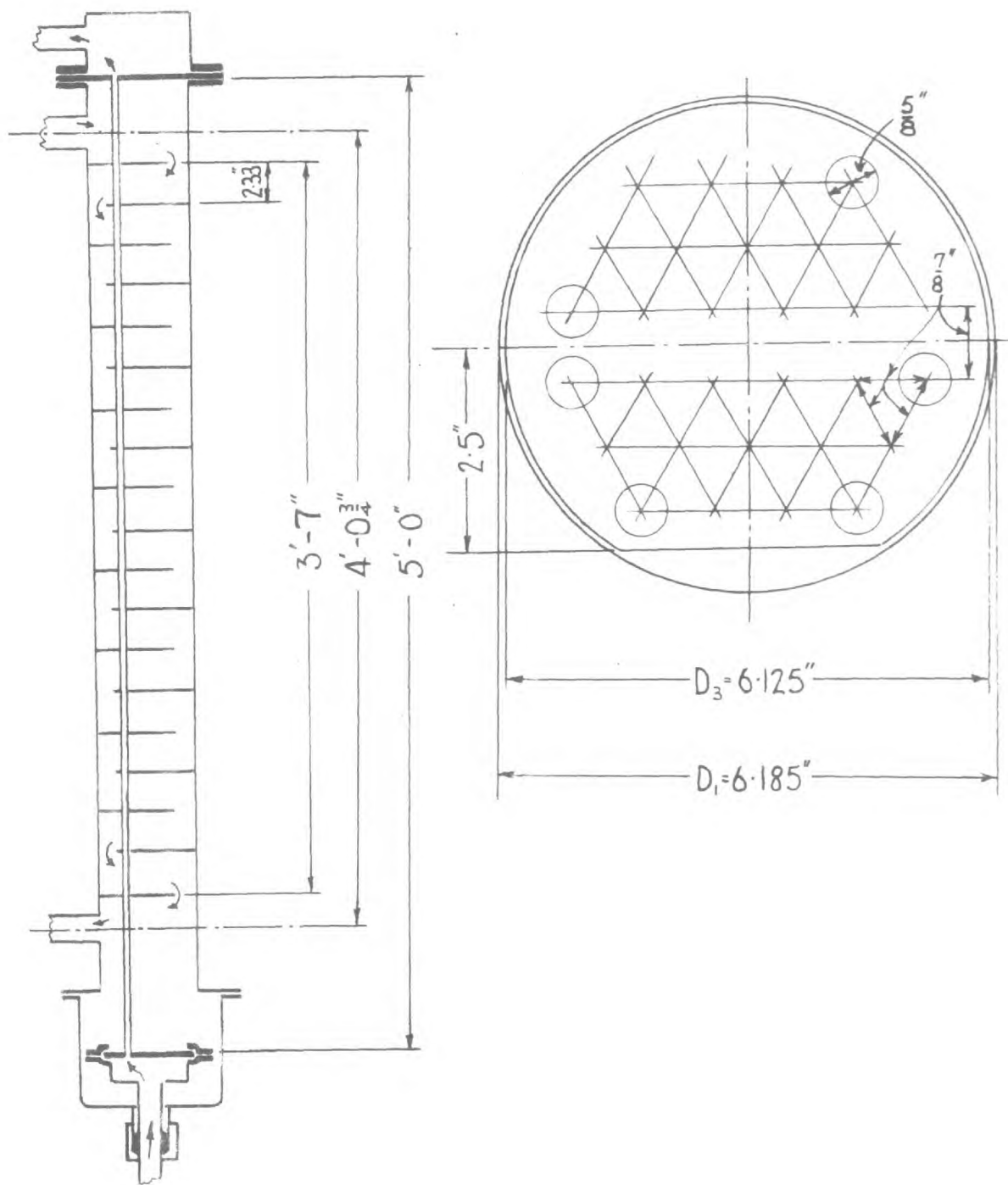


Fig.24. Dimensions of Short's 1947 Exchanger No.1.