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A NEW CORRELATION FOR HEAT TRANSFER DURING BOILING FLOW THROUGH PIPES

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The design of boilers and evaporators requires a means for calculating the heat transfer coefficients of boiling fluids. The practical importance of this problem is evident from the fact that hundreds of correlations for such calculations have been proposed, including one by the author himself(25). However, most of them are not reliable beyond the range of data on which they were based. Only the correlation of Chen(6) can at all be considered as general. However, its application is limited to vertical flows.

The author's efforts have resulted in the development of the CHART correlation, a graphical method of solution, which appears to be considerably superior to any other available predictive technique in range of applicability and ease of computation. It has been compared with about 800 data points from 18 independent experimental studies. These data include most of the common refrigerants in their entire range of application. Also included are data for boiling water between 15 to 2500 psia pressure, which covers practically the entire range of boiler operation. Almost all common pipe materials, horizontal and vertical orientations, circular and annular flow channels, upward and downward flow, as well as a very wide range of heat and mass flux are covered. Hence from the evidence in hand, the CHART appears to be applicable to saturated boiling inside pipes of all Newtonian fluids (except metallic fluids) over the entire practical range.

The objective of this paper is to present the CHART correlation, explain its use, and demonstrate the agreement of its predictions with experimental data. Some suggestions regarding the design of refrigeration evaporators using the CHART as well as other correlations are also made.

It is to be noted that subcooled boiling and stable film boiling are completely excluded from consideration. The CHART has primarily been developed for saturated boiling at subcritical heat fluxes. However, it is also applicable in the liquid-deficiency regime (super critical heat flux) provided that the dryout starts at vapor qualities 80% or higher.

SIMILARITY OR CORRELATING PARAMETERS

The CHART correlation employs four dimensionless parameters defined by Eq 1 to 4:

$$\psi = h_{TP}/h_1 \tag{1}$$

$$Co = (1/x-1)^{0.8} (\rho_g/\rho_1)^{0.5} \tag{2}$$

$$Bo = q/Gh_{fg} \tag{3}$$

$$Fr_L = G^2/\rho_1^2 g D \tag{4}$$

Co is the Convection number, Bo the Boiling number, and Fr_L is the Froude number assuming all the mass to be flowing in liquid form. Co is the only new parameter introduced here. The other three parameters have been used in many other correlations. Their physical significance will become clear in the following section.

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In Eq 1, h_1 is always calculated by the Dittus-Boelter equation, Eq 5, irrespective of the value of superficial Reynolds numbers of the two phases:

$$h_1 = 0.023 \left[\frac{G\ D(1-x)}{\mu_1}\right]^{0.8} Pr_1^{0.4}\ k_1/D \qquad (5)$$

For annulii D is replaced by D_e, the effective diameter of the annulus.

DEVELOPMENT OF THE CHART

A few steps in developing the CHART are briefly described here in order to clarify the significance of the correlating parameters used.

In the nucleate boiling regime, it may be expected that the heat transfer enhancement factor ψ will be a function of Bo which may be interpreted as the mass flux of vapor generated at the pipe surface to the total mass flux parallel to the pipe surface. This expectation is justified by Fig. 1 which shows the data of Mumm(19) for saturated water boiling at zero vapor quality in a horizontal pipe. As the vapor quality in these data was zero, the enhancement in heat transfer coefficient is entirely due to bubble nucleation. Fig. 1 shows that all data, from 15 to 200 psia pressure, are well correlated by the simple equation:

$$\psi = 230\ Bo^{0.5} \qquad (6)$$

This equation may be expected to apply where nucleate boiling effects are dominant, i.e., at low vapor qualities and high Bo.

Fig. 2 shows the analysis of the data of Mumm(19), Guerrieri and Talti(12) and Gouse and Coumou(13). It is noted that at high values of Co, the data from all three sources are in good agreement with Eq 6. At low values of Co, corresponding to high vapor qualities, all data appear to merge into a line of constant slope which may be represented by the following equation:

$$\psi = 1.8 / Co^{0.8} \qquad (7)$$

Eq 7 represents the condition of pure convective boiling in which bubble nucleation has been completely suppressed. In between the limits of Eq 6 and 7 is the region of bubble suppression in which both Bo and Co are significant.

A tentative correlation was formed by smoothing out the curves in Fig. 2 and extending its range by using Eq 6 and 7 to define its extreme limits. This tentative correlation was essentially that shown in Fig. 4 but without the Fr_L lines whose need was not apparent at that stage.

When it was compared to the data of Chawla(8) for R-11 boiling in horizontal pipes, it was found that while the data for 6- and 14-mm pipes were well correlated, the data for the 25-mm pipe was mostly far below the predictions. From the tabulations in Chawla's paper, it was noted that the flow patterns in the tests with 25-mm pipe were such that part of the pipe surface remained dry in most of the tests. On the other hand, the flow patterns for the 6- and 14-mm pipes were such that the pipe surface remained fully wetted in most of the tests. Obviously, some means for accounting for partially dry surfaces in horizontal pipes was needed. Some success could be expected by introducing the Froude number as a parameter as it has been successfully used in several correlations for flow pattern prediction. Furthermore, physical considerations suggest that the ratio of gravitational and inertia forces is likely to determine the extent of pipe filling and liquid film climbing. The Froude number is that ratio.

Fig. 3 shows the data of Chawla in convective boiling regime in terms of ψ, Fr_L, and Co. It is apparent that there is a correlation between Fr_L and ψ. It may further be noted that for vertical pipes, the pipe surface will always remain wetted until liquid-deficiency occurs. Hence Fr_L is not required as a similarity parameter for vertical pipes.

Combining Fig. 2 and 3 and several cycles of graphically carried out iterations using a wide range of data, the final CHART correlation shown in Fig. 4 was arrived at. It is noted that for Fr_L greater than 0.04, the correlation is exactly the same for both horizontal and vertical pipes. Reliable data for Bo less than 0.5×10^{-4} were not available. The broken line BC has been tentatively drawn for Bo $\lesseqgtr 0.25 \times 10^{-4}$ to be verified later with more reliable data in this range.

METHOD FOR CALCULATIONS WITH CHART

The method for calculating local heat transfer coefficients is first explained. Calculation of mean heat transfer coefficients is discussed later.

Knowing the mass flow rate, vapor quality, heat flux and pressure, Co, Bo, and Fr_L are easily calculated. For vertical pipes, Fr_L need not be calculated.

For vertical pipes and horizontal pipes in which Fr_L is greater than 0.04, the method of reading the CHART for ψ is completely analogous to the method for reading the Moody chart for friction factors in pipe. For such conditions the Fr_L lines are completely ignored.

The procedure is somewhat more complex for horizontal pipes in which Fr_L is less than 0.04. The use of CHART for various conditions is explained through the following four examples whose solution is shown graphically in Fig. 5.

Example 1

Vertical pipe

$Co = 0.10$, $Bo = 20 \times 10^{-4}$, $Fr_L = .002$

As the pipe surface is completely wetted for vertical orientation, ignore Fr_L.

Draw a vertical line at Co = 0.1 to intersect the curve for $Bo = 20 \times 10^{-4}$. From this point of intersection, draw a horizontal line to the left and read ψ at the ordinate line.

The solution is $\psi = 20$.

Example 2

Same as Example 1, but $Bo = 2 \times 10^{-4}$.

Draw a vertical line at Co = 0.1 as before till it intersects line AB. As $Bo \times 10^4$ at this point of intersection is 4.5, the actually prevalent Boiling number of 2×10^{-4} has no influence. Draw a horizontal line from point of intersection to the ordinate line.

The solution is $\psi = 11$.

Example 3

Same as Example 1, but the pipe is horizontal.

Draw a vertical line from Co = 0.1. From the point it intersects the line for $Fr_L = 2 \times 10^{-3}$ draw a horizontal line to the right to intersect the line AB. From this intersection point, draw a vertical line to intersect the curve for $Bo = 20 \times 10^{-4}$. Then draw a horizontal line from this last point of intersection to the left to intersect the ordinate.

The solution is $\psi = 14$.

Example 4

Same as Example 2 but the pipe is horizontal.

Proceed as in Example 3 till the line AB is intersected. As this point of intersection is above the point where the curve for $Bo = 2 \times 10^{-4}$ intersects the line AB, Bo has no influence. Draw a line to the left to read ψ.

The solution is $\psi = 5.4$.

CALCULATION OF MEAN HEAT TRANSFER COEFFICIENTS

The exact method for calculation of the mean heat transfer coefficient h_{TPM} for the whole evaporator is to calculate local heat transfer coefficients for short element of length, calculate the temperature distribution along the length, and then numerically solve the equation:

$$h_{TPM} = \frac{\int_0^L q \, dL}{\int_0^L (T_w - T_f) \, dL} \qquad (8)$$

A simpler and reasonably accurate method is to calculate the arithmetic mean vapor quality and then proceed with the estimation of ψ using this mean vapor quality just as for local heat transfer calculations explained earlier. Thus if the inlet vapor quality is 0.2 and exit vapor quality is 0.8, the mean vapor quality is 0.5. Co is calculated with Eq 2 using $x = 0.5$ and ψ then estimated from Fig. 4. Again using $x = 0.5$ in Eq 5, h_l is calculated and then Eq 1 yields the value of the mean heat transfer coefficient for the whole evaporator. A formal proof to demonstrate the accuracy of the calculation method outlined in the previous paragraph is difficult as the necessary derivations lead to integrals which cannot be solved exactly. For this reason, this method was verified by doing several calculations both ways. It was found that the accuracy of the approximate method depends mainly on the change in vapor quality across the evaporator. However, even with a vapor quality change of 80%, the results of calculations by the two methods differ by no more than 5%. Because of the tremendous savings in calculation labor, this small error was considered acceptable and all mean heat transfer data was analyzed by this approximate method. It may be mentioned that Chaddock and Noerager(7) as well as Anderson et al.(2) used essentially the same method for comparing their experimental mean heat transfer data with various correlations.

In the foregoing explanation of calculation procedures, it has been tacitly assumed that the heat flux is known. However, in many cases including air conditioning coils, it is not the case. Iterative calculations with assumed values of heat flux are then required. Such calculation rechniques are well known to all concerned with heat exchanger design and need not be elaborated upon here.

APPLICATION OF CHART TO LIQUID-DEFICIENT REGIME

Many evaporators, including all DX type refrigerant evaporators, operate with part of their length in the liquid-deficient regime or in other words at supercritical heat flux. While the CHART has been primarily designed for use upstream of the dryout point, it can be applied to the liquid-deficient region under certain conditions discussed in the following.

Consider an evaporator such that $Fr_L > 0.04$, $h_L = 100$ and $(\rho_g/\rho_l)^{0.5} = 0.10$. Vapor quality varies from 0 to 100%. For these conditions, the CHART predictions for $Bo \times 10^4 = 1$ and 20 are shown in Fig. 6. It is noted that at the higher heat flux or Boiling number, heat transfer coefficient does not change much till about 80% quality. Then it drops down rather sharply. At the lower Boiling number, h_{TP} increases rapidly till a quality of about 70%, is fairly constant till about 90% quality, and then diminishes sharply. We may then expect that in evaporators which show similar heat transfer characteristics, the CHART will also be applicable in the liquid-deficiency regime.

Most refrigerant evaporators show a similar behavior. As examples, one may study the data of Anderson et al.(2) with R-22, Chawla(8) with R-11, and Shah(26) with ammonia. In all these experiments, dryout occurred in the vicinity of 80% vapor quality. Hence one way expect that the CHART can be applied for predictions in the liquid deficiency region for most refrigerant evaporators. It may, however, be mentioned here that the situation regarding Freon refrigerants containing appreciable amounts of oil is not very clear. Detailed discussions on effects of oil on heat transfer are beyond the scope of this paper, though recommendations regarding the design of evaporators containing oil are made later.

Fig. 7 shows the comparison of some of Chawla's local heat transfer data with the CHART predictions. The two are seen to be in good agreement right up to 95% vapor quality. However, Chawla's measurements at 98% quality are much lower than predictions. Hence the limit of CHART applicability is somewhere between 95 and 98% quality. Thus if the total change in quality in the evaporator is large, the assumption that the CHART is applicable up to 100% quality does not involve any significant error. Mean heat transfer data from refrigerant evaporators with complete evaporation have been alalyzed on this assumption and their agreement with the correlation also indicate that the CHART is applicable to the liquid deficient portion of such evaporators.

DATA ANALYZED

The salient features of the experimental studies from which data for developing and verifying the CHART have been taken are described in Table 1 along with the data range. Table 2 lists the range of dimensional and dimensionless parameters in these data. It is noted that almost the entire range of practical application is covered for water and Freon refrigerants. All common pipe materials (steel, copper, brass, glass) are included. Circular and annular flow sections as well as horizontal, vertical up and vertical down flows are included. All common heating methods are explored, electrical heating, heating by condensing fluids, and heating by fluids on the outside of the evaporator section.

It may be mentioned that all boiling water data as well as the data of Chawla(8), Gouse and Coumou(13), and Guerrieri and Talti(12) are in terms of local heat transfer coefficients. All other data analyzed are in terms of mean heat transfer coefficients.

Some explanation regarding the selection of experimental data are desirable. No date have been deleted simply because they do not fit the correlation. All data tabulated or shown graphically in the references have been included except in four cases. Mumm and Chawla have provided very extensive tabulations and to analyze all their data would have required much labor without yielding any new information. Hence only part of their total data, representative of the entire experimental ranges, were used. Wright(31) has tabulated about 2,000 data points all of which show strong entrance effects. The data for the station at entrance were not considered. Furthermore, his data for Bo less than 0.5×10^{-4} were not considered as their reliability is doubtful, Wright himself estimating that the reported heat transfer coefficients may be in error by as much as 100% for small temperature difference between pipe wall and fluid. This elimination left about 1500 data points from which data representative of the range were selected. It may be noted that if all the data tabulated by Chawla and Mumm as well as all of Wright's data at Bo $>0.5 \times 10^{-4}$ were used, the mean deviation for all data shown in Table 3 would be much improved. Finally, in the data tabulated by Yodder and Dodge (32), those for exit vapor quality greater than 90% were not considered. This is because the dryout occurred in these experiments at vapor qualities well below 80%, probably in the range of 40 to 60%. Hence the application of CHART to qualities higher than 90% would not be appropriate.

For calculations, all fluid properties were taken from Ref. 14 with the following exceptions. The properties of water for the analysis of the high pressure data of Naitoh et al. (21) were taken from Ref 23. Properties of cyclohexane were taken from the book by Maxwell (20). The properties of ammonia for the calculation on which Fig. 15 is based, are from Ref 11.

RESULTS OF DATA ANALYSIS

Fig. 8, 9, 10, 11 show the comparison of measured heat transfer coefficients with CHART predictions for the fluids water, R-11, R-12, and R-22 respectively. Fig. 12 shows the comparison of measured and predicted ψ for R-113 and cyclohexane data. In Fig. 13, the data of Naitoh et al.(21) for water boiling at a pressure of 171 atmospheres are shown along with the CHART predictions.

Table 3 summarizes the results of all data analysis. It is noted that the mean deviation for all data is 14% if equal weight is given to each data point. If equal weight is given to each data set, the mean deviation is 17%. For each data point, deviation is defined as:

$$\text{Deviation} = \frac{\text{Predicted } \psi - \text{Measured } \psi}{\text{Measured } \psi} \tag{9}$$

The mean deviation for each data set is calculated as the arithmetic mean of the individual deviations ignoring their signs, positive or negative.

DISCUSSION OF RESULTS

A striking result of the data analysis is that all data for boiling water from all five sources are very well correlated, very few points varying beyond ±30% of measured values. All the larger deviations are confined to refrigerant evaporator data. The explanation for this may be that in boiling water experiments, impurities were carefully excluded and the transport properties of water are well established. Refrigerant evaporators, on the other hand, frequently contain impurities like moisture, air, oil, and other unidentified impurities as were noted in the tests by Ashley(1) and Chaddock and Noerager(7). Besides, there is a great deal of uncertainty regarding the properties of refrigerants. The value given in different sources often differ by 20 to 30%.

The range of boiling water data examined is very wide and virtually covers the entire range of practical interest. Included are two pipe materials (stainless steel and copper), three flow orientations (horizontal, vertical up and vertical down), three heating modes (electric, steam, liquid sodium), and three modes of inlet vapor quality generation (boiling in test section, mixing of liquid and vapor before entrance, flashing by throttling of liquid prior to entrance). The range of heat and mass flux explored is very wide and both circular and annular flow geometries are included. The most encouraging feature is that pressures from 15 to 2500 psia (reduced pressure 0.0045 to 0.8) are included. Hence it appears reasonable to recommend the CHART for practical boiler-design at least up to 2500 psia pressure. However, it must be pointed out that no data have been analyzed in between the limits of 200 and 2500 psia and only one set of data at the higher pressure was available. Furthermore, for correlating the data of Bennet et al. for boiling in an annulus, the effective diameter had to be based on the heated perimeter rather than on wetted perimeter as is more usual. Whether this is in fact the proper definition of effective diameter for boiling flows remains to be confirmed through analysis of more data from other sources. The data of Yodder and Dodge(32) for R-12 boiling in an annulus, are not helpful in this matter as the pipe sizes were such that values of ψ calculated using either definition differ by only 5%.

Data for all common refrigerants except ammonia and R-502 have been examined almost throughout the entire range of their use. The absence of R-502 data, however, is no serious handicap as its properties and behavior are not too different from those of R-12 and R-22. The question of ammonia evaporators is briefly discussed later.

The refrigerant data which show wider deviations are those of Bryan and Siegel(4), Anderson et al.(2), and Staub and Zuber(24). In all these sets, some of the data points tend to be higher than the measurements. This may, of course, indicate shortcomings of the correlation. On the other hand, the discrepancies may be due to experimental errors or impurities. However, the most likely cause appears to be entrance effects. As an example to support this belief, Fig. 14 shows the measured local heat transfer coefficients from one of the tests by Bryan and Siegel, along with CHART predicitons. It is evident that the heat transfer coefficients at and near the entrance are abnormally high. The entrance effects appear to persist till about 120 diameters downstream of the entrance. After that, the measurements and predictions are in reasonable agreement.

An encouraging feature of the data analysis is that except for the entrance effects noted above, the deviations appear to be random. The deviations can not be correlated to the basic parameters like heat and mass flux, pressure, pipe material or size, etc., or to dimensionless parameters like Co, Bo, Re_l, Fr_L, etc. This suggests that the deviations are not due to shortcomings of the correlation but are due to other causes such as experimental and data reduction errors, presence of impurities, etc.

It will be noted that the concept of dissipative flow regimes (turbulent-turbulent, turbulent-viscous, etc.) advanced by Martinelli and coworkers does not appear to have any significance for boiling heat transfer. As can be seen in Table 2, the superficial liquid phase Reynolds numbers varied from less than 100 to more than 100,000 in the data analyzed. This result is not surprising in view of physical considerations. There is no evidence that if, for example, Re_l is less than 1000, the liquid phase flows laminarly. In fact, visual observations indicate turbulence in liquid films even at very low Reynolds numbers. As an example, one may study Fig. 30-15 in Jakob's text(16).

OTHER CORRELATIONS

Among the correlations for which general applicability has been claimed, the best known are those of Dengler and Addoms(9), Shrock and Grossman(28), Bennet et al.(3), Guerrieri and Talti(12) and Chen(6). In his paper, Chen showed that the other four are in fact not generally applicable. On the other hand, the Chen correlation was compared to about 650 data points from several experiments including those of Dengler and Addoms, Shrock and Grossman, Guerrieri and Talti, Bennet et al, and Wright(31) and was shown to be in excellent agreement. All the data analyzed by Chen were for vertical flow at subcritical heat fluxes.

Anderson et al.(2) and Chaddock and Noerager(7) compared their own mean heat transfer data for boiling in horizontal pipes with several correlations. They found that the correlations of Guerrieri and Talti, Lavin and Young(18), Shrock and Grossman, and Sachs and Long(29), are very unsatisfactory. The Dengler and Addoms correlation is satisfactory if its nucleate boiling correction factor is omitted which means that the complete correlation is unsatisfactory. The Pierre correlation(22) showed good agreement. With Chen correlation, the predictions at low mass flow rates were much higher than the measurement.

Johnston and Chaddock(15) compared their measurements on low temperature R-12 and R-22 boiling in a horizontal pipe with the Pierre correlation. They found that the measurements were up to 60% lower than predictions, the mean through data being about 30% below predictions.

After considering the foregoing reports and several others, it is this author's conclusion that among the presently available correlations, only the one by Chen can be considered as satisfactory. It can be recommended for boiling in vertical pipes at subcritical heat fluxes. Its applicability to the liquid deficiency regime and high pressures remains to be explored. Furthermore, it is not satisfactory for horizontal pipes. Finally, it is to be noted that calculations with the Chen correlation require cumbersome iterations as it does not give an explicit solution for the heat transfer coefficient.

The Pierre correlation appears satisfactory for R-12 and R-22 evaporators in the range of parameters covered in Pierre's experiments. Outside that range, its applicability is doubtful as is evident from the data of Johnston and Chaddock mentioned earlier which were at lower temperatures.

It will be noted that the CHART has been shown to satisfy most of the data on which the Chen correlation was based as well as those from Pierre's experiments. Hence availability of the CHART correlation makes the Pierre and Chen correlations superfluous.

RECOMMENDATIONS FOR USE OF CHART CORRELATION

The empirical evidence supporting the correlation has been presented in the foregoing along with that small portion of data which does not show good agreement. The reader can therefore make his own judgment regarding its validity and utility. The author's own recommendations for the use of the CHART are given in the following:

General Recommendations

The CHART is recommended for all pure Newtonian fluids except metallic fluids in the reduced pressure range 0.004 to 0.8. Application beyond this pressure range should be made with caution, specially at higher pressures. As the single-phase heat transfer correlations developed from data on non-metallic fluids do not apply to metallic fluids, applicability of the CHART to boiling metallic fluids is very doubtful. If the CHART is at all used to calculate the heat transfer coefficients for boiling metals, it is tentatively suggested that h_1 be calculated with a single-phase heat transfer correlation known to be applicable to metallic fluids. Calculations of h_1 with the well known Martinelli or Lyon analogies may be appropriate.

For Bo $\lesseqgtr 0.5 \times 10^{-4}$, use of CHART should be made with caution, as reliable data in this region were not available. There is no evidence of any other restrictions on the correlation.

Freon Evaporators Containing Oil

For Freon refrigerant evaporators containing oil, it is suggested that calculations be made with the CHART under the assumption that oil is absent. This recommendation is based on the fact that Pierre(22) found that if the heat transfer coefficients were calculated on the basis of the saturation temperature of pure refrigerant, up to 18% oil circulating with R-12 had no perceptible effect. It must be stressed that there is no general agreement on the effects of oil and it will be appropriate to also consult other literature on this topic.

Ammonia Evaporators

While no data for boiling ammonia completely free of oil was available, it appears fairly certain that the CHART is applicable to oil-free ammonia evaporators, ammonia being a Newtonian fluid with properties in between those of R-12 and water. For ammonia evaporators containing oil, the experiments of Shah(25 and 26) show that insulating oil films are formed which drastically reduce the heat transfer rates. Hence the CHART predictions have to be corrected for the resistance of insulating oil films. No method for finding the thickness of these oil films in the boiling region is available. However, for the nonboiling region, Shah (26) proposed the following equation:

$$\delta/D = 0.028/Re_L^{0.23} \tag{10}$$

where δ is the thickness of oil film, D is the pipe diameter and Re_L is the Reynolds number of ammonia liquid. The oil film thicknesses in the boiling region are likely to be somewhat less than those given by Eq 10.

The suggested method for calculating boiling heat transfer coefficients is as follows:

1. Calculate the heat transfer coefficient for pure ammonia using the CHART.
2. Calculate oil film thickness using Eq 10.
3. Correct the prediction for pure ammonia for the resistance of oil film just as is done for the resistance of scale in tubes.

An alternative procedure is to use the $Y - \psi$ correlation proposed by Shah(25). If that correlation is used, it is recommended that the mean curve through data shown in Fig. 11 of Ref 25 be used rather than the equation given in the appendix to that paper. The reason is that the mean curve gives a better fit to the data of Shah as well as that of Cosijn and Van Maale(30).

Fig. 15 shows the comparison of some data from Shah's experiments with the standard CHART predictions, CHART predictions corrected for oil film thickness from Eq 10, as well as the estimates from the Shah correlation(25). The agreement between measurements and the two methods of prediction is reasonably good. As the resistance of oil film is the controlling factor, and the thickness of oil film can vary considerably depending on many factors, the agreement is as good as could reasonably be expected.

Calculations in Entrance Region

As yet there is very little understanding of the entrance phenomena for boiling flows. The author is unaware of any proposed method to account for these effects. As a first approximation, it is suggested that the ψ calculated from the CHART be multiplied by the enhancement factor obtained from single-phase correlations for entrance regions. Information on single-phase enhancement factors for entrance regions may be found in several texts, including that by Knudson and Katz(17).

Application to Liquid Deficiency Regime

The CHART should be applied to the liquid-deficiency region only if dryout occurs at vapor qualities in the neighborhood of 80% or higher. In such cases, the CHART correlation may be used for calculating local heat transfer coefficients up to 95% vapor quality. For most refrigerant evaporators, dryout occurs at vapor qualities around 80%, and hence the CHART can be used for estimation of mean heat transfer coefficients for the case of complete evaporation in DX coils by the simplified method explained earlier in this paper. However, the superheated vapor region should be estimated separately by a single-phase correlation. At what vapor quality the liquid deficiency will occur has to be determined by some independent method.

CONCLUSION

In the foregoing, it has been demonstrated that the CHART correlation is in reasonable agreement with a wide variety of experimental data. It may therefore be stated that the choice of Co, Bo, Fr_L and ψ as correlating parameters has been adequately justified and that the CHART can be recommended as a reliable tool for the prediction of boiling heat transfer coefficients in both horizontal and vertical pipes. No other correlation has been verified over such a wide range of parameters.

Several areas for further research and development may be suggested. Among these are, analysis of data at reduced pressures higher than 0.8, analysis of data for Bo less than 0.5×10^{-4}, and expressing the CHART in terms of mathematical equations. It is further suggested that the data for boiling metals be analyzed in terms of the correlating parameters used in CHART. It is possible that the CHART may apply to boiling metals if h_l is calculated by a correlation suitable for liquid metal heat transfer.

All data referred to in Table 1 which have been used to develop and verify the CHART are tabulated in Ref 27 which also provides more detailed discussions on several topics which had to be dealt with very briefly in this paper due to limitations of space. Among such topics are entrance effects, effect of oil on refrigerant evaporators, various other calculation techniques, occurrence of dryout, various boiling regimes, etc.

NOMENCLATURE

As the CHART correlation is dimensionless, any consistent system of units may be used.

A	Cross-sectional area of pipe or annulus
Bo	Boiling number, defined by Eq 3
Co	Convection number, defined by Eq 2
D	Inside diameter of pipe
D_e	Equivalent diameter of annulus
Fr_L	Froude number assuming all mass flowing as liquid, defined by Eq 4
G	Total mass flux, W/A
g	Accelaration due to gravity
h_{TP}	Two-phase heat transfer coefficient
h_1	Superficial heat transfer coefficient of liquid phase, i.e. assuming the liquid phase to be flowing alone in the pipe
hg	Superficial heat transfer coefficient of gas phase
h_{fg}	Latent heat of vaporization
k_1	Thermal conductivity of liquid
L	Length of the evaporator
Pr	Reduced pressure, i.e. actual pressure/critical pressure
Pr_1	Prandtl number of liquid
q	Heat flux based on ID of pipe
Re_1	Superficial Reynolds number of liquid phase
Re_L	Reynolds number assuming all the mass flowing in the form of liquid
h_L	Heat transfer coefficient assuming all mass flowing as liquid
T_{SAT}	Saturation temperature of fluid
T_B	Bulk temperature of fluid, which is the temperature measured by a thermal sensor
T_f	Fluid temperature, bulk or saturation
T_w	Temperature of inside surface of wall
W	Total mass flow rate
x	Thermodynamic vapor quality
Y	Shah's correlating parameter, $= h_1/h_g$
μ_1	Dynamic viscosity of liquid phase
ψ	h_{TP}/h_1, h_1 defined by Eq 5
ρ_1	Density of liquid phase
ρ_g	Density of gas phase
δ	Thickness of oil film in ammonia evaporators

ABBREVIATIONS

SS	Stainless steel
H	Horizontal
V	Vertical
VU	Vertical up
VD	Vertical down

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TABLE 1

SUMMARY OF BOILING HEAT TRANSFER DATA ANALYZED FOR THE DEVELOPMENT AND VERIFICATION OF THE CHART CORRELATION

Source	Fluid	Flow Direction	Test Section Geometry	Data Range: T_{SAT} F	$G \times 10^{-6}$ lb/hr ft^2	$q \times 10^{-6}$ Btu/hr ft^2	x %	h_{TP} Based On
Mumm(19)	Water	H	Electrically heated SS pipe o.465 in. ID, 7 ft long	275 382	0.25 1.02	0.05 0.25	0 54	T_{SAT}
Wright(31)	Water	VD	Electrically heated SS pipes 0.7194 in. ID, 5.67 ft long and 0.4716 in. ID, 4.69 ft long	216 301	0.4 0.25	0.014 0.088	0 19	T_{SAT}
Bennet et al.(3)	Water	VU	Annulus, inner pipe SS 0.623 in. OD, outer pipe glass 0.866 in. ID. Length 29 in. Inner pipe electrically heated	213 258	0.05 0.21	0.032 0.158	0 55	T_{SAT}
Dengler & Addoms(9)	Water	VU	Steam heated vertical copper pipe 1 in. ID 20 ft. long	245 262	0.053	0.03 0.14	1.7 13.5	T_{SAT}
Chawla(8)	R-11	H	Electrically heated copper pipes 0.236, 0.551 & 0.984 in. ID 4.7, 13.78 & 13.78 in. long respectively, Substantial calming lengths preceded entrance	32 68	0.009 0.184	0.0004 0.022	10 98	T_{SAT}
Bryan & Siegel(4)	R-11	H	Electrically heated copper pipe 0.55 in. ID, heated length 127 in.	87 109	0.05 0.07	0.0016 0.0029	0.3 100	$\frac{T_{SAT} + T_B}{2}$
Bryan & Quaint(5)	R-11	H	Electrically heated copper pipe, 0.314 in. ID, heated length 120 in.	84 103	0.08 0.39	0.0005 0.0058	3 99	T_{SAT}
Chaddock & Noerager(7)	R-12	H	Electrically heated SS pipe, ID 0.46 in. heated length 76 in.	53	0.09 0.43	0.002 0.011	19 88	T_B
Johnston & Chaddock(15)	R-12 R-22	H	Electrically heated copper pipe 0.46 in. ID, heated length 72 in.	-61 5	0.01 0.07	0.0005 0.0068	2 71	T_B
Ashley(1)	R-12	H	Water heated copper U-tube 0.577 in. ID, 32.77	40 60	0.065 0.25	0.0009 0.004	11 100	T_B
Yodder & Dodge(32)*	R-12	VU	Annulus made from copper pipes. Outer pipe 0.954 in. ID, inner pipe 0.25 in. OD. Outer pipe electrically heated	-106 -73	0.007 0.023	0.001 0.003	25 90	T_B
Naitoh et al(21)	Water	Inclined up	16.5 mm ID SS pipe formed in helix with vertical axis. Heated by liquid sodium in annular jackets around the pipe.	670	0.92	0.033 0.166	0 70	unknown
Donavan(10)	R-12	H	Water heated 0.569 in. ID copper pipe; 30 ft. long	40	0.093 0.174	0.002 0.004	14 100	unknown
Pierre(22)	R-22	H	Brine heated copper pipes. 0.709 & 0.472 in. ID, 28.6 & 15.7 ft. long respectively	-4 14	0.039 0.164	0.001 0.007	15 90	T_{SAT}
Anderson et al.(2)	R-22	H	Water heated copper pipe 0.665 in. ID, 7.5 ft long	40	0.023 0.255	0.002 0.028	19 100	T_B
Staub & Zuber(24)	R-12	VU	Electrically heated SS pipe 0.3985 in. ID, 60.6 in. long	43	0.11 0.64	0.004 0.024	4 18	T_{SAT}
Gouse & Coumou(13)	R-113	H	Electrically heated glass pipe. ID 0.43 in. length 150 in.	117 130	0.38 0.51	0.004 0.007	2 37	T_B
Guerrieri & Talti(12)	Cyclo-hexane	VU	Oil heated machined brass pipes 0.75 in. ID, 6 ft long and 1 in. ID, 6.5 long	180	0.29 0.36	0.0025 0.0098	0.65 6	T_B

*Approximately 5% oil by weight in all tests. In experiments of all other researchers, oil content was zero or negligibly small.

TABLE 2

Range of Dimensional and Dimensionless Parameters

Explored in Boiling Heat Transfer Data Analyzed

Fluid	Flow Direction	Flow Section Geometry	ID In.	x %	G x 10^{-6} lb/hr ft^2	q x 10^{-6} Btu/Hr ft^2	Bo x 10^4	Co	Fr_L*	Re_1 x 10^{-3}	P_r
Water	H, VU, VD inclined	Circular & Annular	0.465 1.00	0 70	0.053 1.02	0.033 0.25	0.5 11	0.04 7	1 19	11 287	0.0045 0.77
R-11	H	Circular	0.236 0.984	0 100	0.009 0.390	0.0004 0.022	0.5 30	0.0056 0.45	0.0003 0.9	0.09 11	0.011 0.055
R-12	H, VU	Circular & Annular	0.46 0.704#	2 100	0.01 0.43	0.0005 0.011	1 17	0.016 0.34	0.0008 1.6	0.2 17	0.0015 0.15
R-22	H, VU	Circular	0.3985 0.709	4 100	0.01 0.64	0.0005 0.028	1 16	0.022 1.8	0.001 0.45	0.24 38	0.012 0.12
R-113	H	Circular	0.43	2 37	0.38 0.51	0.004 0.007	1.3 3	0.11 1.44	1 2	6 15	0.026 0.04
Cyclo-hexane	VU	Circular	0.75 1.0	0 6	0.29 0.36	0.0025 0.0098	0.5 1.9	0.5 3.5	---	7 12	0.026

*Only horizontal pipes were considered for this column.

#Effective diameter for annular section.

TABLE 3

SUMMARY OF COMPARISON OF BOILING HEAT

TRANSFER DATA WITH PREDICITIONS OF CHART

No.	Source	Fluid	Test Section	No. of Data Points Analyzed	No. of Data Points Deviating More Than 30%	Mean Deviation For All Data Points %
1	Mumm(19)	Water	0.465 In. ID pipe	189	6	12.8
2	Wright(31)	Water	0.7194 In. ID pipe	71	3	10.7
			0.4716 in. ID pipe	36	2	17.1
3	Bennet et al. (3)	Water	Annulus 0.623 in. ID, 0.866 in. OD	7	1	24.4
4	Dengler & Addoms (9)	Water	1 in. ID pipe	5	1	16.8
5	Chawla(8)	R-11	Pipes, ID			
			6 mm	48	11	28.5
			14 mm	31	0	10.3
			25 mm	83	7	16.0
6	Bryan & Siegel(4)	R-11	0.55 in. ID pipe	17	4	22.0
7	Bryan & Quaint(5)	R-11	0.314 in. ID pipe	31	1	7.7
8	Chaddock & Noerager(7)	R-12	0.46 in. ID pipe	20	1	11.9
9	Johnston & Chaddock(15)	R-12	0.46 in. ID pipe	18	5	27.4
10	Ashley(1)	R-12	0.575 in. ID pipe	21	6	24.0
11	Yodder & Dodge(32)	R-12	Annulus 0.25 in. ID 0.954 in. OD	41*	6	17.7
12	Donavan(10)	R-12	0.569 in. ID pipe	19	0	12.9
13	Johnston & Chaddock(15)	R-22	0.46 in. ID pipe	23	2	15.4
14	Pierre(22)	R-22	Pipes, ID 18 & 22 mm	15	0	4.1
15	Anderson et al(2)	R-22	0.665 in. ID pipe	42	14	25.3
16	Staub & Zuber(24)	R-22	0.3985 in. ID pipe	8	6	42.0
17	Gouse & Co-umou(13)	R-113	0.43 in. ID pipe	10	0	10.7
18	Guerrieri & Talti(12)	Cyclo-hexane	Pipes, ID 0.75 & 1 in.	23	0	16.3
19	Naitoh et al(21)	Water	Pipe, ID 16.5 mm	22	0	15.0

*Only data with exit vapor quality less than 90% considered.

Total number of data points = 780
Number of data points showing deviation greater than ±30% = 76
Mean deviation for all data:

giving equal weight to each set of data = 17%
giving equal weight to each data point = 14%

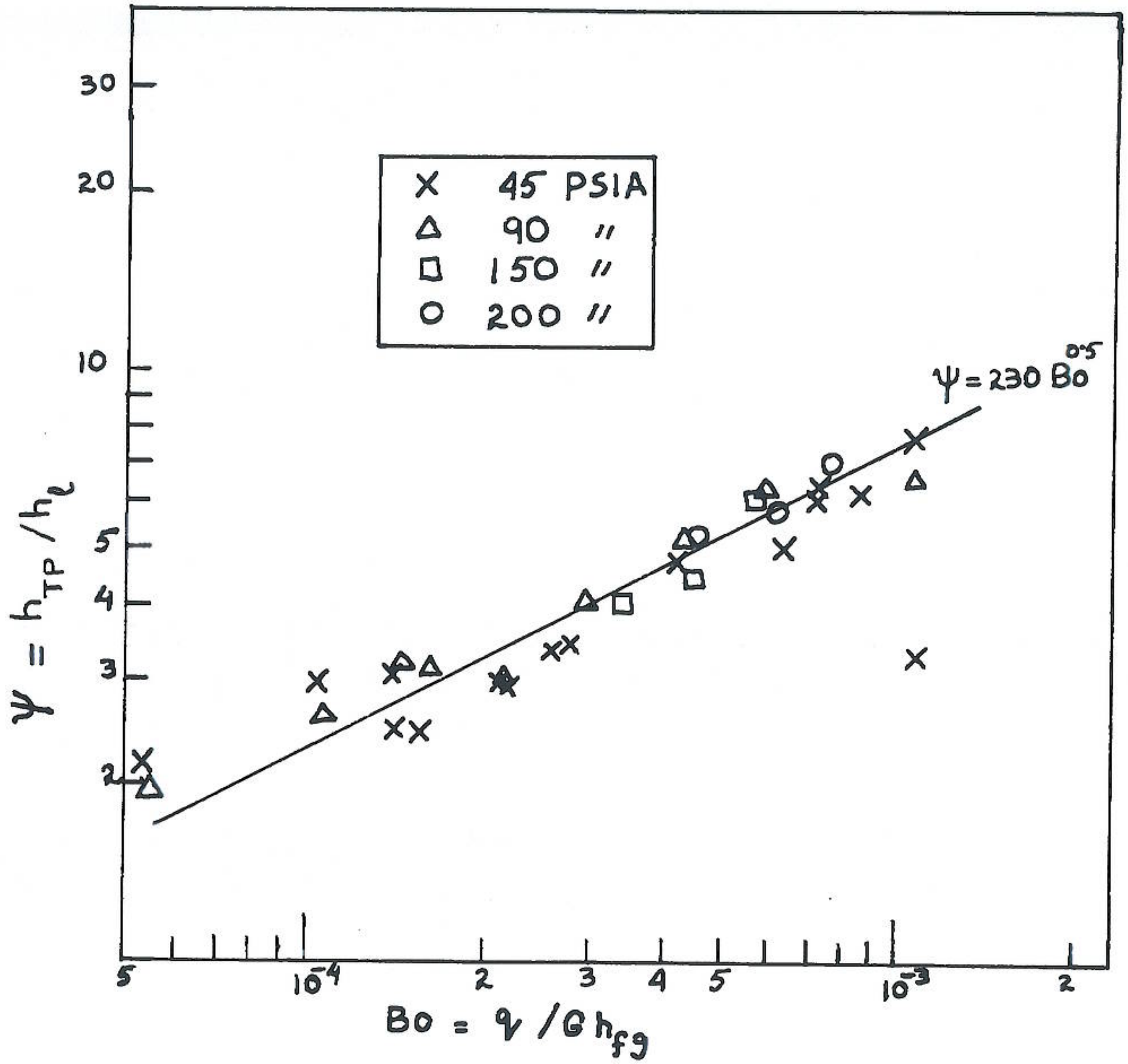

Fig. 1 Relation between boiling number and ψ for saturated boiling of water at zero vapor quality. Data of Mumm[19].

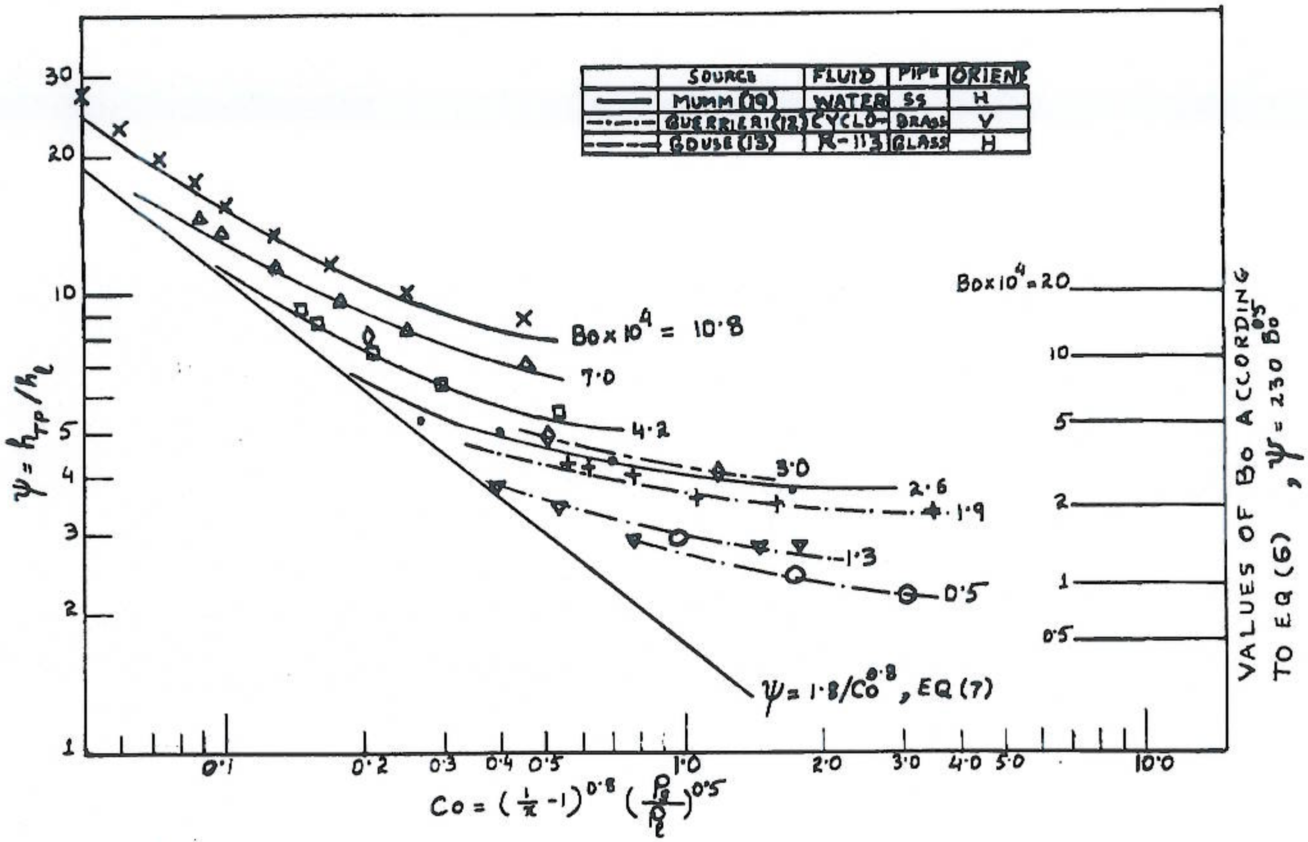

Fig. 2 Preliminary analysis of some boiling heat transfer data in terms of Co and Bo

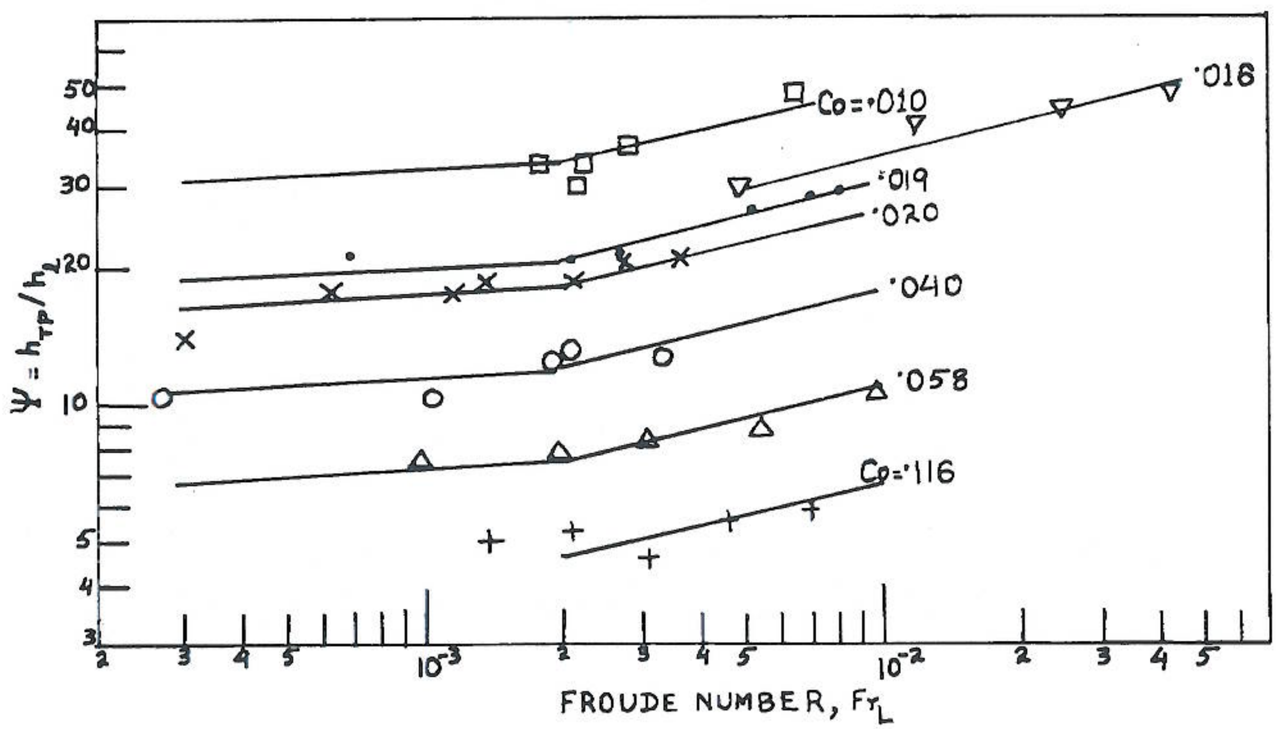

Fig. 3 Analysis of the data of Chawla (8) to investigate the suitability of Fr_L to account for the effect of partially dry pipe circumference

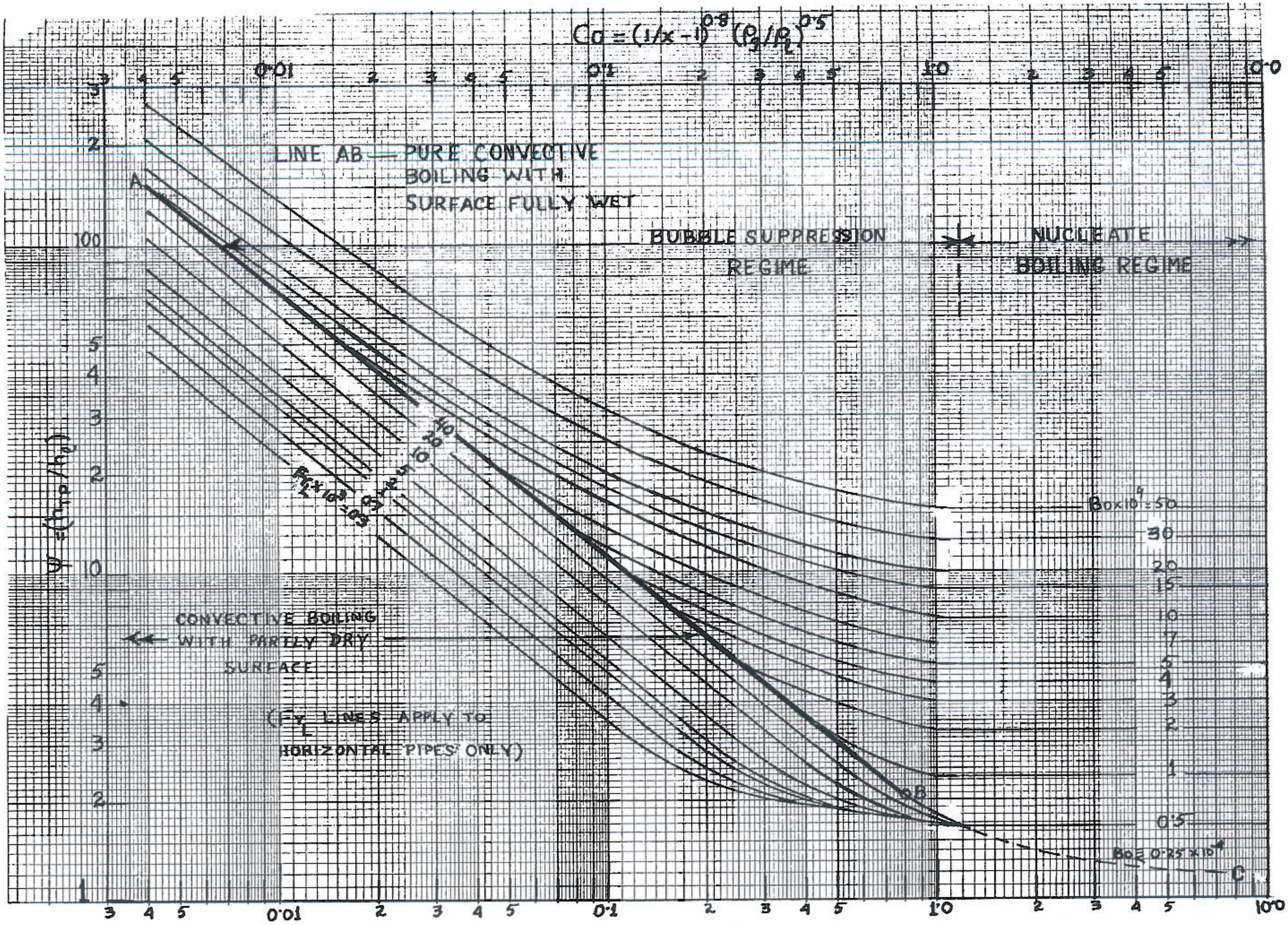

Fig. 4 The CHART correlation

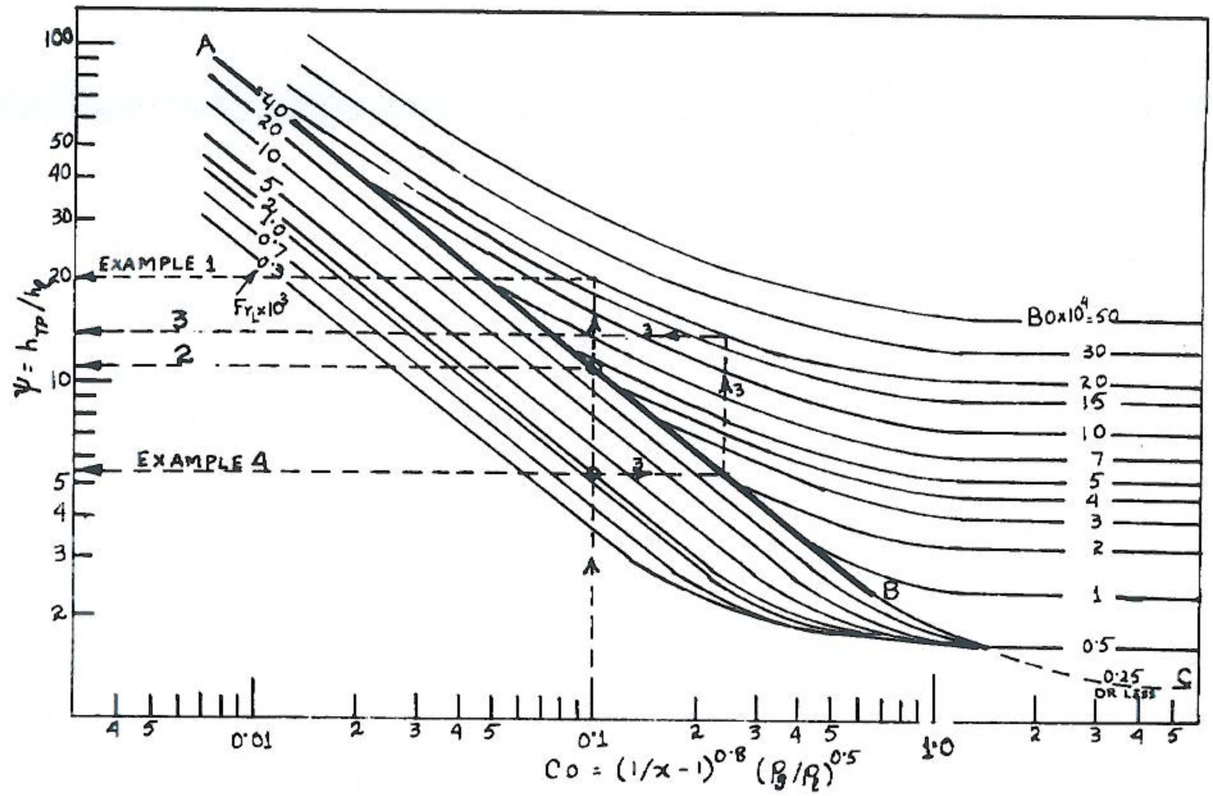

Fig. 5

Solution of Examples 1 to 4 to explain the use of the CHART

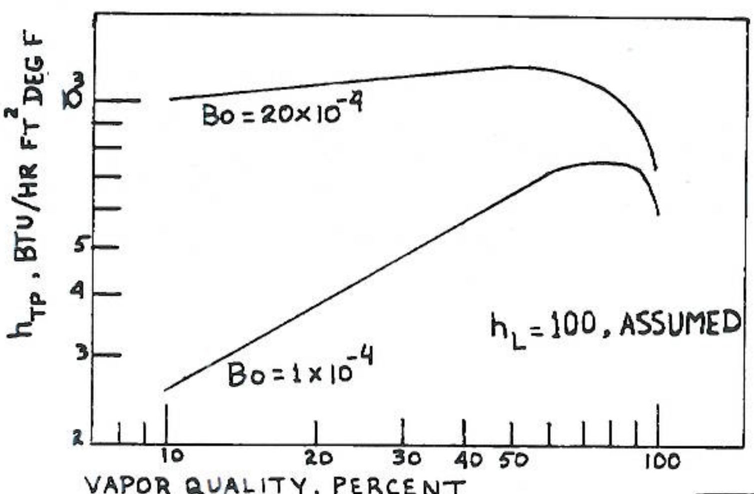

Fig. 6

Predictions of the CHART for the hypothetical case to show some properties of the correlation

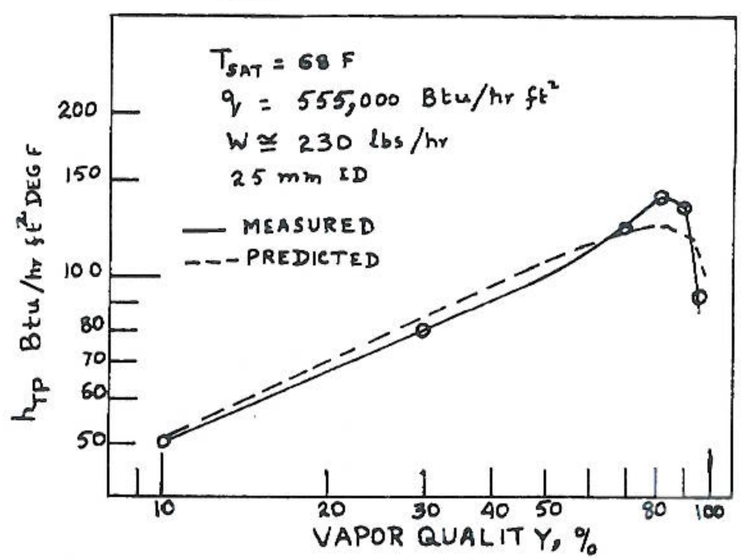

Fig. 7

Comparison of boiling R-11 data of Chawla[8] with CHART predictions

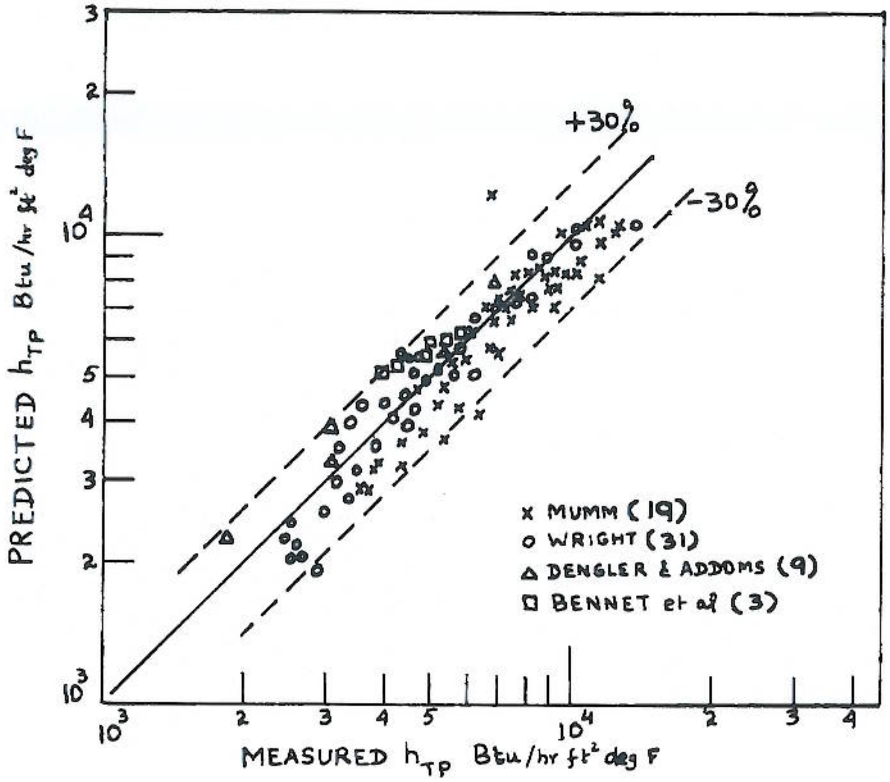

Fig. 8 Comparison of CHART predictions with boiling water data from various sources

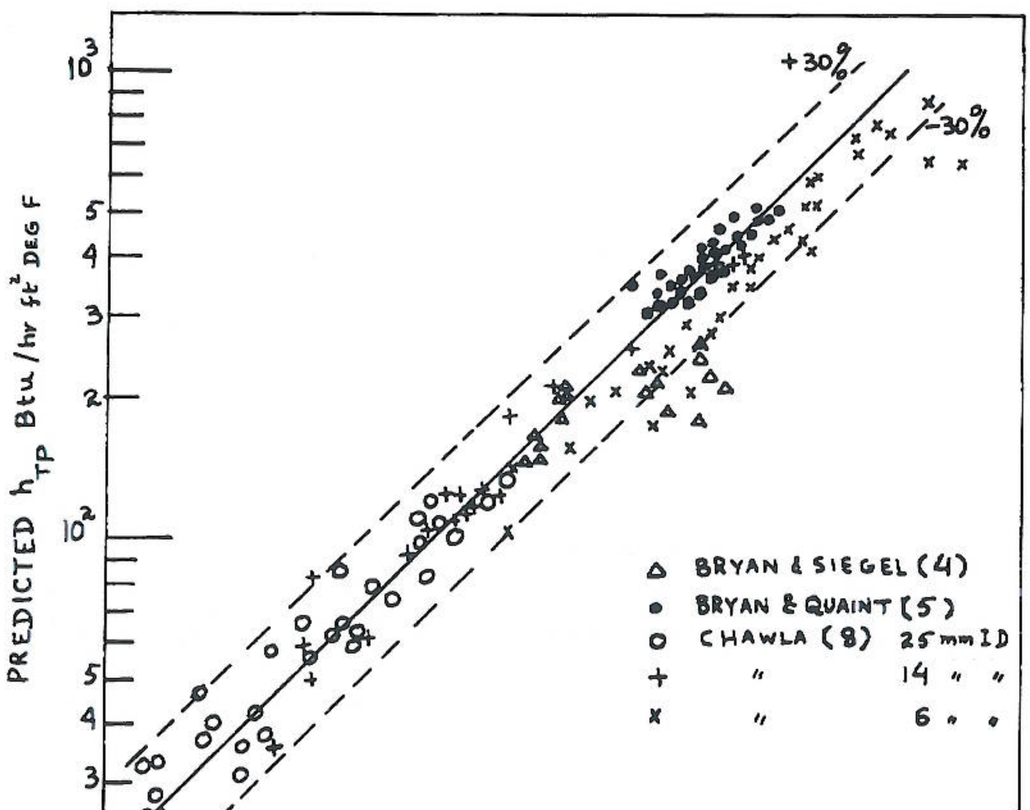

Fig. 9 Comparison of boiling R-11 data from several sources with CHART predictions

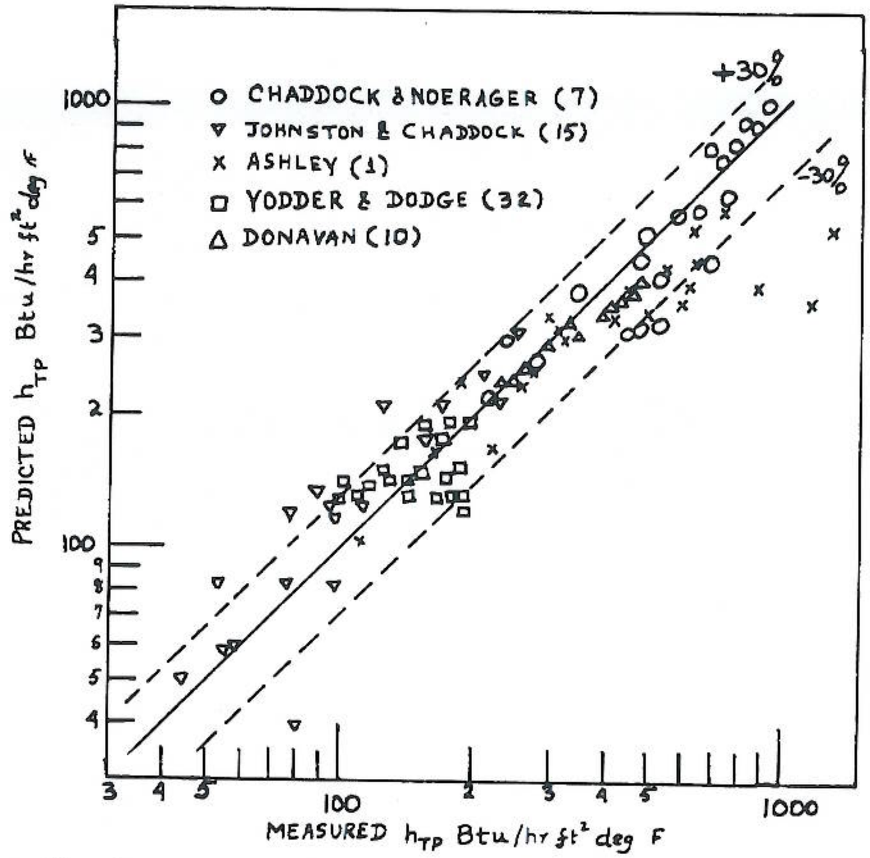

Fig. 10 Comparison of mean heat transfer data for boiling R-12 from several sources with CHART predictions

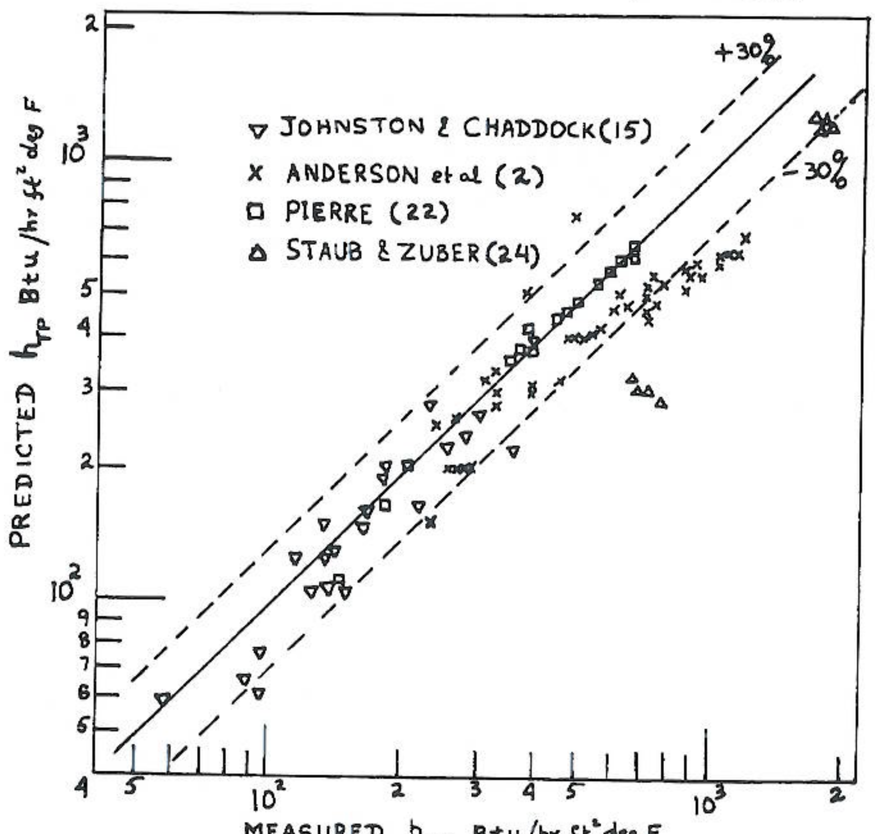

Fig. 11 Comparison of boiling R-22 data from several sources with CHART predictions

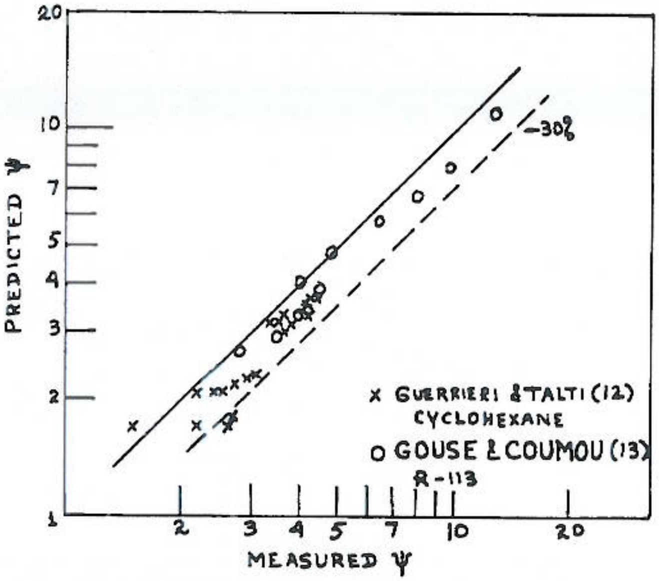

Fig. 12 Comparison of boiling R-113 and cyclohexane data with CHART preditions

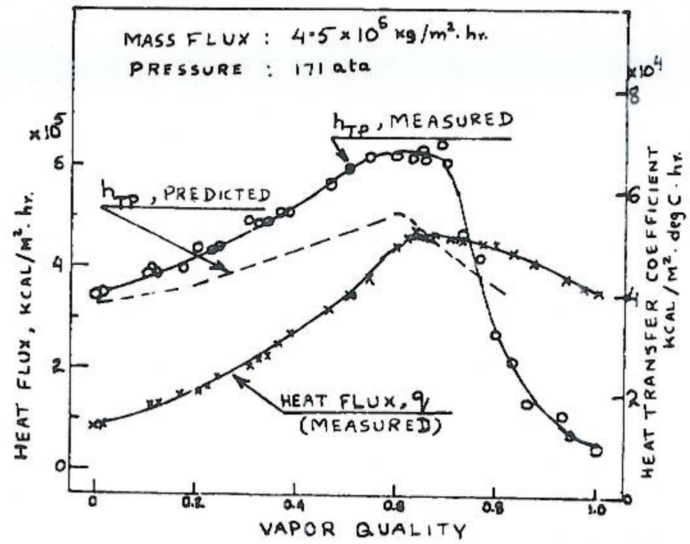

Fig. 13 Data of Naitoh et al.[21] for boiling water compared with CHART preditions

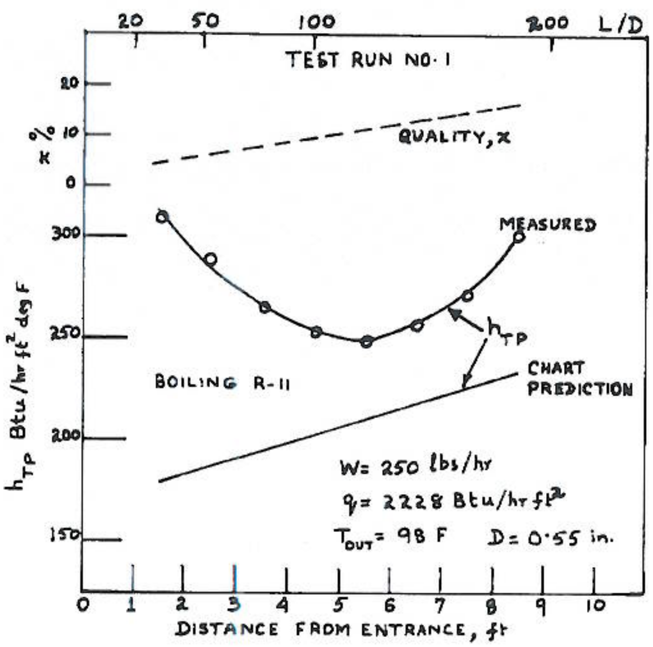

Fig. 14 Comparison of the CHART predictions with the data of Bryan and Siegal[4]. Strong entrance effects are evident

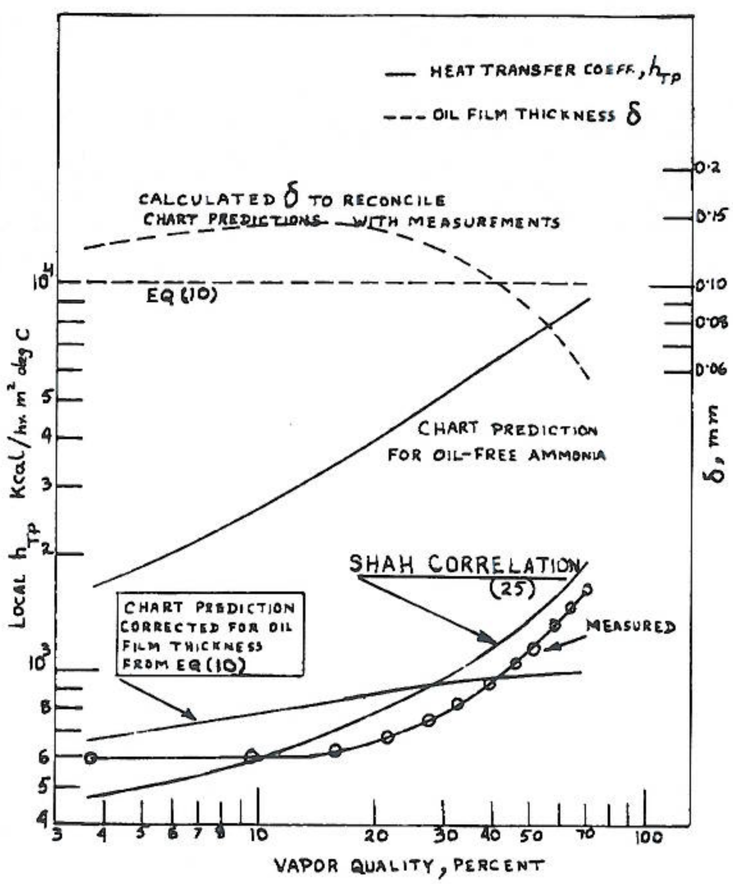

Fig. 15 Data of Shah[25] from his Test No. 55 compared with the Shah correlation and the CHART. Ammonia containing oil evaporating in 26.2-mm ID pipe

APPENDIX

1. Heat Transfer Through Annuli

It was noted in the paper that for correlating the data of Bennet et al.[3], the effective diameter of the annulus had to be based on the heated perimeter rather than the wetted perimeter as is more usual. Since this paper was written, several other data sets for saturated and subcooled boiling in annuli have been examined. These data suggest the following empirical rule:

$$\text{Clearance} < 4 \text{ mm} \quad D_e = \frac{4(\text{flow area})}{\text{heated perimeter}} \tag{11}$$

$$\text{Clearance} > 4 \text{ mm} \quad D_e = \frac{4(\text{flow area})}{\text{wetted perimeter}} \tag{12}$$

where clearance = $(D_o - D_i)/2$, D_o and D_i being the outside and inside diameters of the annulus respectively.

While a considerable amount of data have been examined, more data from several sources need to be examined before complete confidence can be expressed in this rule. It is hoped to deal with this question in a future paper. Till then, Eq 11 and 12 represent the author's best recommendation.

2. Heat Transfer in Rod Bundles

Heat transfer during flow parallel to heated rod bundles is of considerable importance, notably in boiling water nuclear reactors. Hence the applicability of the CHART to this case was investigated. The only data that could be found are those of Matzner.[33] e used a triangular array of 12 vertical electrically heated rods. The rods were 0.44 in. OD and 17 in. long. The pitch of the array was 0.4625 in., the pitch by diameter ratio being 1.05. The coolant was water at a pressure of 1200 psia. The vapor quality varied from 4.4 to 10.3%. D_e was calcualted with Eq 12. The measured heat transfer coefficients were on the average 70% of those predicted by the CHART.

Fig. 118 of Ref 34 shows that for a triangular array with a pitch/diameter ratio of 1.05, the heat transfer coefficients in single-phase flow are 70% of those for circular pipes. Hence the data of Matzner suggest that the CHART can be applied to heated rod bundles if h_l is calculated by a correlation suitable for that particular geometrical configuration.

It is hazardous to make a generalization based on only one data set. However, the results are quite encouragining and further data analysis may confirm the applicability of CHART to rod bundle heat transfer.

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