Int. J. Heat Mass Transfer. Vol. 22, pp. 547–556
© Pergamon Press Ltd. 1979. Printed in Great Britain

0017-9310/79/0401-0547 $02.00/0

A GENERAL CORRELATION FOR HEAT TRANSFER DURING FILM CONDENSATION INSIDE PIPES

M. M. SHAH
15 Rush Street, Port Jefferson Station, NY 11776, U.S.A.

(*Received* 27 *March* 1978 *and in revised form* 7 *August* 1978)

Abstract—A simple dimensionless correlation for predicting heat-transfer coefficients during film condensation inside pipes is presented. It has been verified by comparison with a wide variety of experimental data. These include fluids water, R-11, R-12, R-22, R-113, methanol, ethanol, benzene, toluene, and trichloroethylene condensing in horizontal, vertical, and inclined pipes of diameters ranging from 7 to 40 mm. One data set for condensation inside an annulus has also been analyzed. The range of parameters covered includes reduced pressures from 0.002 to 0.44, saturation temperatures from 21 to 310°C, vapor velocities from 3 to 300 m/s, vapor qualities from 0 to 100%, mass flux 39 000–758 000 kg/m^2 h, heat flux from 158 to 1 893 000 W/m^2, all liquid Reynolds numbers from 100 to 63 000, and liquid Prandtl numbers from 1 to 13. The mean deviation for the 474 data points analyzed was found to be 15.4%.

NOMENCLATURE

A,	cross-sectional area of pipe;
C_{pl},	specific heat of liquid;
D,	inside diameter of pipe;
G,	total mass flux $= W/A$;
g,	acceleration due to gravity;
h_{TP},	two phase boiling or condensing heat-transfer coefficient;
h_{TPL},	local two phase heat-transfer coefficient;
h_L,	heat-transfer coefficient assuming all mass to be flowing as liquid;
h_l,	superficial heat-transfer coefficient of liquid phase i.e. assuming the liquid phase to be flowing alone in the pipe;
k_l,	thermal conductivity of liquid;
L_c,	total length of pipe in which condensation occurred;
L,	length of condenser pipe under consideration;
p,	actual pressure;
p_c,	critical pressure;
p_r,	reduced pressure $= p/p_c$;
Pr_l,	Prandtl number of liquid;
q,	local heat flux based on ID of pipe;
Re_L,	Reynolds number assuming all mass flowing as liquid, GD/μ_l;
Re_l,	superficial Reynolds number of liquid phase $= Re_L(1-x)$;
T_s,	saturation temperature of fluid;
W,	total mass flow rate;
V,	nominal vapor velocity $= W/(A\rho_g)$;
x,	thermodynamic vapor quality;
y,	axial distance from the point where condensation started;
Z,	correlating parameter for condensation heat transfer, defined by equation (6).

Greek symbols

Ψ,	h_{TP}/h_l, h_l defined by equations (4) and (5);
ρ_l,	density of liquid;
ρ_g,	density of vapor.

INTRODUCTION

A VERY large number of techniques for predicting the heat-transfer coefficients during condensation inside pipes have been proposed over the last 50 years or so. These range from very arbitrary correlations to highly sophisticated treatments of the mechanics of the flow. While many of these have been valuable as practical design tools and have added to our understanding of the phenomena involved, there does not appear to be any general predictive technique which has been verified over a wide range of parameters.

The author's efforts have resulted in the development of a very simple dimensionless correlation which has been verified by comparison with a wide variety of experimental data. Data analyzed include Freons, water, methanol, benzene, toluene, trichloroethylene, and ethanol, condensing in horizontal, vertical, and inclined pipes of diameters ranging from about 7 to 40 mm. Some data for condensation inside an annulus have been examined. A very wide range of heat flux, mass flux, vapor velocities and pressures has been covered. The 473 data points from 21 independent experimental studies are correlated with a mean deviation of about 15%. Judging by the fair agreement with experimental data, it appears that this correlation can be used with confidence for all Newtonian nonmetallic fluids over much of the range of practical interest.

The objective of this paper is to present this correlation and demonstrate its agreement with experimental evidence. So that the correlation may be viewed in the proper perspective, brief discussions on some other predictive techniques are also included.

DEVELOPMENT OF THE CORRELATION

This correlation was developed from the author's correlation for saturated boiling heat transfer [6] by noting the similarity between the mechanisms of heat transfer during film condensation and boiling without bubble nucleation (evaporation). That correlation was developed by analyzing a vast amount of varied data from horizontal, vertical upflow, and vertical downflow orientations. It was found that in the absence of bubble nucleation and as long as the entire pipe surface remains wetted by the liquid, the following equation applied to all flow orientations:

$$\Psi = 1.8/Co^{0.8}. \qquad (1)$$

The parameters Ψ and Co are defined as follows:

$$\Psi = h_{TP}/h_1 \qquad (2)$$

$$Co = \left(\frac{1}{x} - 1\right)^{0.8} (\rho_g/\rho_1)^{0.5}. \qquad (3)$$

The superficial heat-transfer coefficient of the liquid phase h_1, was calculated as:

$$h_1 = h_L(1-x)^{0.8}. \qquad (4)$$

Where h_L is the heat-transfer coefficient assuming all the mass flowing as liquid and is calculated by the Dittus–Boelter equation as:

$$h_L = 0.023 Re_L^{0.8} Pr_l^{0.4} k_l/D. \qquad (5)$$

In the process of condensation, liquid will be formed whenever vapor contacts the pipe surface and thus the pipe circumference will always be wetted at all flow rates and in all flow orientations. Hence equation (1) may be expected to apply to condensation in all flow orientations with minimum modification.

Analysis of experimental data in terms of Y and Ψ confirmed this belief but the scatter was about ±50%. The trend of deviations indicated that replacement of (ρ_g/ρ_l) by the reduced pressure p_r would improve the correlation. This led to a new parameter Z defined as:

$$Z = \left(\frac{1}{x} - 1\right)^{0.8} p_r^{0.4}. \qquad (6)$$

THE CORRELATION

By analyzing a large amount of varied data in terms of Y and Ψ, the following equation was found to give the best fit:

$$\Psi = 1 + 3.8/Z^{0.95}. \qquad (7)$$

Using equations (2), (4) and (6), equation (7) yields:

$$h_{TP} = h_L\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{p_r^{0.38}}\right]. \qquad (8)$$

VARIATION OF VAPOR QUALITY WITH LENGTH

For calculating mean heat-transfer coefficients or for analyzing the experimental data reported in terms of mean heat-transfer coefficients, the knowledge of vapor quality distribution with length is necessary. Figure 1 shows the local vapor quality x plotted against the dimensionless length y/L_c from some experimental studies. It is noted that variation of vapor quality with length of condenser is not always linear. The integrated mean quality for the total condensing length during complete condensation (from $x = 1.0$ to 0.00) can vary from 50% to as low as 30%. Study of a large amount of experimental data showed that linear quality distribution occurs in comparatively few cases. The value of integrated mean quality for complete condensation is generally closer to 40% than 50% as is the case for linear distribution. Hence the assumption of linear distribution often made in analyses can be in considerable error.

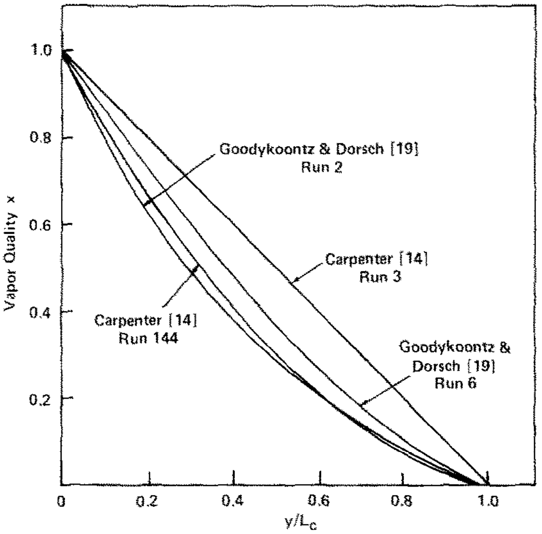

FIG. 1. Variation of vapor quality with dimensionless condensing length.

However, study of Fig. 1 also reveals that if the change of vapor quality is less than 20%, the error involved in the assumption of linear quality variation is always negligible. Even in the case when vapor quality change is as high as 40%, the error involved in assuming a linear quality change is acceptably small.

MEAN HEAT-TRANSFER COEFFICIENTS FROM EQUATION (8)

As an approximation, it can be assumed that:

$$h_{TPM} = \frac{1}{L}\int_0^L h_{TPL}\,\mathrm{d}L. \qquad (9)$$

Assuming negligible change in transport properties of liquid phase and pressure along the length of condenser, and that equation (9) provides a sufficiently accurate definition of mean heat-transfer

Table 1. Summary of data analyzed for the development and verification of the proposed correlation

Source	Fluid	Orientation	Pipe ID mm	T_s °C	p_r	x	$q \times 10^{-3}$ W/m^2	$G \times 10^{-3}$ kg/h m^2	Re_L	Re_1	V m/s
Cavallini [16]	R-11	V	20.0	22	0.021	0.30	9	298	3207	510	15
				37	0.036	1.00	35	1108	11504	2309	22
Powell [10]	R-11	H	12.9	36	0.035	0.25	16	928	9333	7000	30
Traviss [21]	R-12	H	8.0	21	0.14	0.07	6	547	6000	300	5
				59	0.36	0.95	57	5517	62900	58500	18
Traviss [21]	R-22	H	8.0	24	0.20	0.20	9	683	6600	330	4
				47	0.36	0.95	85	3613	43750	35000	12
Bae [9]	R-12	H	12.5	26	0.16	0.04	9	752	13300	800	5
				48	0.28	0.94	62	2715	69000	47000	11
Bae [8]	R-22	H	12.5	26	0.22	0.17	19	1172	20000	800	7
				39	0.30	0.96	38	2368	42000	35000	10
Altman [1]	R-22	H	8.7	34	0.27	0.06		1094	14400	800	5
				56	0.44	0.96		3091	41000	25000	8
Azer [7]	R-12	H	12.7	37	0.29	0.30		757	14700	130	3
				50	0.39	0.99		1611	31200	20400	6
Goodykoontz [18]	R-113	V	7.4	37	0.02	0.16	35	2993	11000	800	61
				85	0.09	0.95	268	5273	19000	18000	79
Goodykoontz [20]	Water	V	7.4	77	0.0019	0.06	173	318	2500	125	91
				127	0.011	0.95	1893	1640	13000	12000	305
Goodykoontz [19]	Water	V	15.9	101	0.005	0.00	63	78	1320	660	20
				143	0.017	1.00	568	268	4560	2300	71
Jakob [4]	Water	V	40.0	100	0.0046	0.70	47	88	3670	180	40
						1.00	233	171	7370	960	80
Ananiev [3]	Water	H	8.0	164	0.031	0.00	0.2	1382			
				310	0.44	1.00	6	5760			
Carpenter [14]	Water	V	11.6	100	0.005	0.00	16	59	445	200	27
				104	0.005	1.00	199	503	5110	2500	120
Tepe [5]	Methanol	Inclined	18.9	64	0.013	0.06	28	59	900	420	14
				65	0.014	1.00	31	107	1600	490	26
Carpenter [14]	Methanol	V	11.6	66	0.013	0.00	9	78	386	200	19
				71	0.014	1.00	82	527	3460	1700	93
Tepe [5]	Benzene	V, H, Inclined	18.9	80	0.020	0.02	13	88	1500	600	9
				81	0.021	1.00	47	317	5000	2100	33
Carpenter [14]	Trichloroethylene	V	11.6	86	0.020	0.00	6	127	1010	500	8
				98	0.025	1.00	72	1367	10300	5000	64
Carpenter [14]	Toluene	V	11.6	83	0.024	0.00	9	117	1290	600	11
				111	0.026	1.00	41	552	6220	3000	47
Carpenter [14]	Ethanol	V	11.6	78	0.016	0.00	13	39	104	50	3
				82	0.017	1.00	50	513	2430	600	74
Borchmann [12]	R-11	H	6.8*	39	0.037	0.62	9	131	670	50	4
						1.00	14	1317	6700	1670	40

*Equivalent diameter of annulus 31.2 mm ID, 38 mm OD.
V, Vertical;
H, Horizontal.

coefficient, h_{TPM} over the length from L_1 to L_2 is obtained by integrating equation (8) as:

$$h_{TPM} = \frac{h_L}{(L_2-L_1)} \int_{L_1}^{L_2} \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{p_r^{0.38}} \right] \mathrm{d}L. \qquad (10)$$

For the case of non-linear quality variation, equation (10) can generally be solved only by numerical methods. Assuming that vapor quality variation is linear with length, expanding the second term in brackets by the binomial theorem, and neglecting very small terms, we have:

$$h_{TPM} = \frac{h_L}{(x_2-x_1)} \left[-\frac{(1-x)^{1.8}}{1.8} + \frac{3.8}{p_r^{0.38}} \times \left(\frac{x^{1.76}}{1.76} - \frac{0.04x^{2.76}}{2.76} \right) \right]_{x_1}^{x_2}. \qquad (11)$$

For the case of $x = 1$ to $x = 0$, equation (11) yields:

$$h_{TPM} = h_L(0.55 + 2.09/p_r^{0.38}). \qquad (12)$$

An important point to be noted now is that if $x = 0.5$ is substituted in equation (8), the heat-transfer coefficient thus calculated differs from that given by equation (12) by only 5%. For smaller variations of vapor quality the results obtained by substituting arithmetic mean quality in equation (8) are in even closer agreement with equation (11). Thus is cases where vapor quality variation is known to be linear, the mean heat-transfer coefficient can be calculated by simply substituting the arithmetic mean quality in equation (8). This result was found to be very useful in analyzing mean heat-transfer data assuming linear quality distribution.

DATA ANALYSIS

The experimental data analyzed are listed in Table 1 along with the range of some dimensional and nondimensional parameters covered by each data set. The complete range of parameters covered by all the data analyzed is given in Table 2.

The data of Azer *et al.* [7], Jakob *et al.* [4], and Ananiev *et al.* [3] have been extracted from the graphical representations in those papers. The other data are from tabulations. Where the references provided very extensive tabulations, only samples representative of the range of parameters covered were taken at random. Such economy in data analysis was necessitated by the fact that all calculations were done by hand without the use of computers.

In analyzing the mean heat-transfer data, if the actual quality distribution was unknown, linear variation was assumed. Cavallini and Zecchin [16], Powell [10], Altman *et al.* [1], and Jakob *et al.* [4] have reported only mean heat-transfer coefficients and the variations of quality with length are not known. However, the vapor quality changes were always less than 40% and generally less than 20%. Hence the assumption of linear quality distribution does not involve any significant error as was pointed out earlier in this paper.

Tepe and Mueller [5] have also tabulated only mean heat-transfer coefficients and provided no information on vapor quality variation. The assumption of linear distribution could be in error here as the quality changes ranged from 40 to 95%.

Goodykoontz and Dorsch [19] have tabulated both local and mean heat-transfer coefficients. However, their wall temperature measurements showed considerable scatter. For this reason, comparison was made only with their mean heat-transfer data. Local vapor qualities were calculated, then local h_{TP} calculated with equation [8] which were then numerically integrated to yield the predicted mean heat-transfer coefficients.

Carpenter [14] did not measure the wall temperatures and, in fact, measured only the mean heat-transfer coefficients. However, he calculated the local heat-transfer coefficients from fluid temperature profiles for a few of the test runs. In order to make use of data from all test runs, comparison has been made only with the mean heat-transfer coefficients re-

Table 2. The complete range of parameters in which the correlation has been tested and the range in which its use is recommended

Parameter	Tested range	Recommended range
Flow channel	Pipes, annulus	Pipes, annuli
Flow direction	Horizontal, vertical, 15° inclined to horiz.	All directions
Pipe ID, mm	7.4–40	7–40
T_s, °C	21–310	21–310
x, %	0–100	0–100
q, W/m^2	158–1 893 000	All values
G, kg/m^2 h	30 000–5 758 400	39 000–5 758 400
p, 10^6 N/m^2	0.07–9.8	0.07–9.8
p_r	0.0019–0.44	0.002–0.44
Re_L		
pipes	104–62 900	350 and higher
annulus	670–6,700	3000 and higher
V, m/s	3–300	3–300
Pr_l	1–13	> 0.5
Flow patterns	All	All

Table 3. Summary of comparison of experimental data with proposed correlation

Data of	Fluid	No. of data points analyzed	Mean deviation %
Cavallini and Zecchin [16]	R-11	31	7.1
Powell [10]	R-11	1	3.0
Traviss *et al.* [21]	R-12	48	19.7
Bae *et al.* [9]	R-12	46	21.8
Azer *et al.* [7]	R-12	39	25.1
Traviss *et al.* [21]	R-22	36	17.3
Bae *et al.* [8]	R-22	46	8.4
Altman *et al.* [1]	R-22	16	11.4
Goodykoontz and Brown [18]	R-113	23	13.5
Jakob *et al.* [4]	Water	29	15.5
Annaniev *et al.* [3]	Water	67	22.4
Goodykoontz and Dorsch [19]	Water	14	13.0
Goodykoontz and Dorsch [20]	Water	25	13.1
Carpenter [14]	Water	2	18.0
Carpenter [14]	Methanol	2	7.0
Tepe and Mueller [5]	Methanol	4	26.7
Tepe and Mueller [5]	Benzene	20	11.3
Carpenter [14]	Trichloro-ethylene	5	19.0
Carpenter [14]	Toluene	2	24.5
Carpenter [14]	Ethanol	2	6.5
Borchmann [12]	R-11	14	20.0

Total number of data points = 474
Mean deviation, giving equal weight to each data point = 16.8%
Mean deviation, giving equal weight to each data set = 15.4%

ported by Carpenter. For the tests in which vapor quality distribution was known, mean heat-transfer coefficients were calculated by numerically solving equation [10]. For other tests, equation (11) was used. In table 1, only those tests of Carpenter are included for which quality distribution was known. However, all data are shown in Fig. 5.

For analyzing the data of Borchmann [12] for condensation in an annulus, the equivalent diameter was defined as four times the flow area divided by the wetted perimeter and h_L then calculated by equation (6).

The properties of Freon (halogenated hydrocarbons) refrigerants and water were taken from [17]. The properties of chemicals were taken from several sources as no single reference listing all the properties could be found. All properties have throughout been calculated at the saturation temperature of the fluid.

RESULTS OF DATA ANALYSIS

Figures 2–6 show comparison of experimental data with the proposed correlation. Table 3 summarizes the results of all data analysis. The deviation for each data point is calculated by the following:

$$\text{Deviation} = \frac{\text{Predicted } \Psi - \text{Measured } \Psi}{\text{Measured } \Psi}. \quad (13)$$

The mean deviation for each data set is calculated as the arithmetic mean of the individual deviations, ignoring their signs positive or negative. It is noted that the mean deviation for all 474 data points is 17% if equal weight is given to each data point. The mean deviation is 15% if equal weight is given to each data set.

DISCUSSION OF RESULTS

Considering the very wide range of parameters covered, the results shown in Figs. 2–6 and Table 1 can be viewed with much satisfaction. Four hundred and seventy-four data points, representative of a much greater number, from 21 independent studies, have been correlated with a mean deviation of about 15%. The correlation is seen to be equally effective for horizontal, vertical, and inclined orientations. Pipe diameters from 7 to 40 mm are included. Nominal vapor velocities (velocity calculated assuming all mass flowing as vapor) from 3 to 300 m/s are included. It will be remembered that vapor velocity is generally regarded to be one of the most significant parameters affecting condensing heat transfer.

Some results of the data analysis are now discussed in more detail and possible limitations of this correlation are exposed.

High vapor qualities and entrainment

Quite a few of the measurements at vapor qualities between 85 and 100% were found to be substantially higher than predictions. One explanation could be the entrance effects as such high qualities generally prevail near the entrance. Another possibility could be that the vapor shear may be so high as to cause high entrainment or even breakdown of a continuous liquid film. On the other hand, the point at

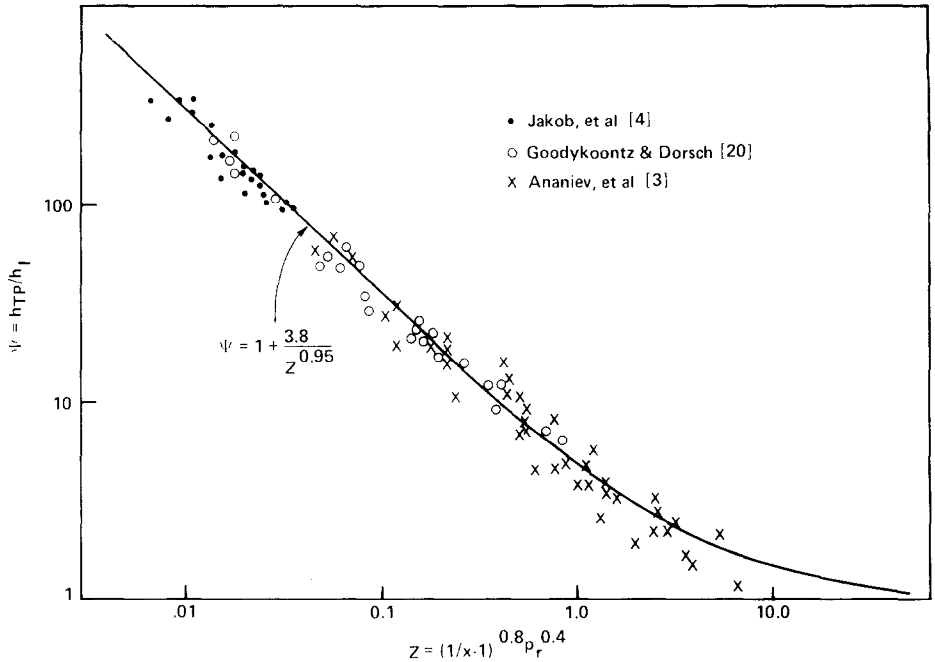

FIG. 2. Comparison of some condensing water data with the proposed correlation.

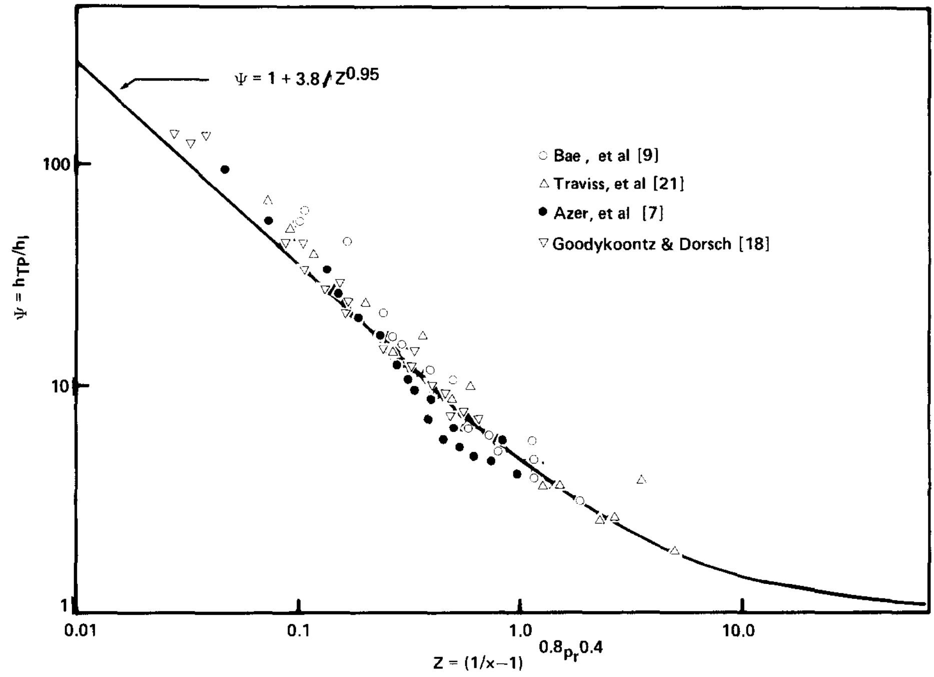

FIG. 3. Comparison of the proposed correlation with some data for condensation of R-12 and R-113.

which condensation starts is often difficult to locate and the estimated vapor qualities and heat-transfer coefficients at high vapor qualities can be in substantial error.

Effect of Reynolds number

As mentioned earlier, h_L has been calculated throughout by the Dittus–Boelter equation, equation (5), irrespective of the value of Re_L or Re_l. For pipes, measurements down to a Re_L of 386 and Re_l of 200 are satisfactorily correlated. The data at a very low Reynolds number are too few to allow any confident recommendation. Based on the evidence in hand, it is the author's suggestion that the use of this correlation for circular pipes be restricted to $Re_L > 350$ until more data at lower Re_L are analyzed.

Only one set of data for condensation in annulus, that of Borchmann [12], has been analyzed. In Fig. 6, the deviations of these data from equation (8) are plotted as a function of Re_L. The accuracy of correlation seems to decrease with decreasing Reynolds numbers.

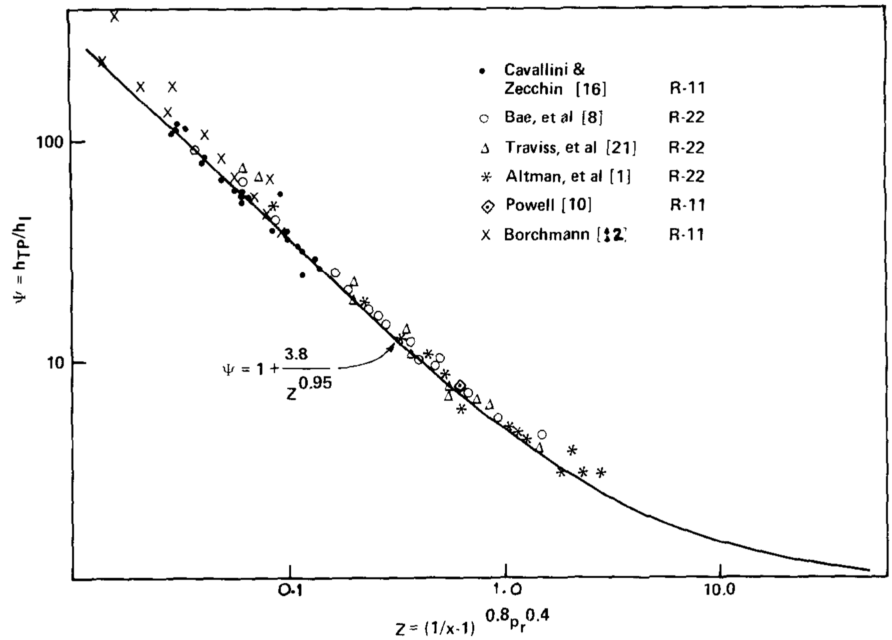

FIG. 4. Comparison of the proposed correlation with condensing R-11 and R-22 data.

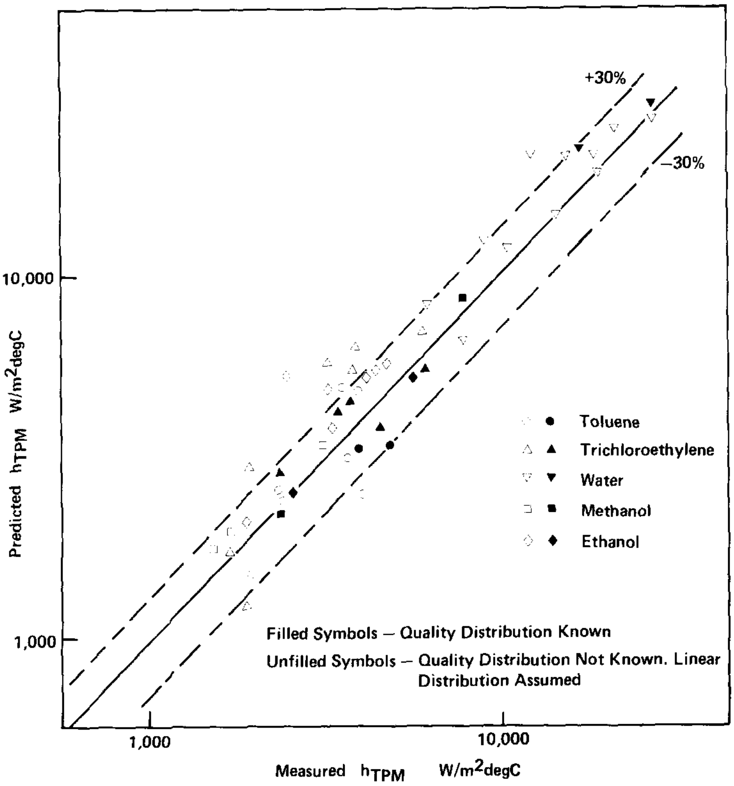

FIG. 5. Comparison of mean heat-transfer coefficients reported by Carpenter [14] with predictions of the proposed correlation.

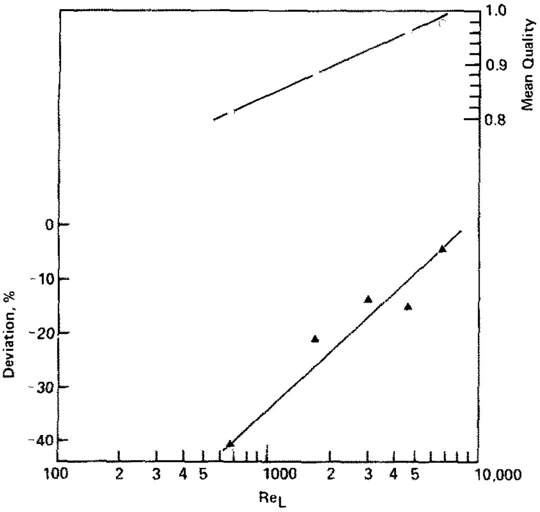

FIG. 6. Deviations of mean heat-transfer coefficients reported by Borchmann [12] from proposed correlation as function of Reynolds number. R-11 condensing in horizontal annulus.

However, the data for Re_L greater than 3000 are quite well correlated. Again, the number of data points are too few to make reliable generalizations possible. The author feels that, for the present, the application of this correlation for annuli should be restricted to $Re_L > 3000$.

SOURCES OF ERROR

While all deviations from the correlation can be conveniently explained by assuming that the correlation is defective, the possibility of errors in measurement, data reduction, etc., also offers a plausible explanation. Some common sources of error are briefly discussed here.

Impurities can greatly increase or decrease the heat-transfer rates. The transport properties of refrigerants listed in various sources often differ by 20 to 40%. The property data of other chemicals are scarce and considerable doubts regarding their accuracy can be entertained. The point where condensation is completed is often difficult to find. Where the rise in coolant temperature is small, estimates of local heat flux and heat-transfer coefficients can be in considerable error. Estimated local vapor quality can easily be in error to the extent of the heat balance error which is often 5–10%. In the very low quality range, estimated quality may be in very serious error due to this reason. In addition, one must consider the error due to inadequate instrumentation, faulty instruments and human errors.

Entrance effects can cause the heat-transfer coefficients to be abnormally high. Some of the data analyzed here appear to show such effects. Very little is known about the phenomena involved and no further discussion on this topic is undertaken.

In view of the many sources of error discussed in the foregoing, it is felt that the agreement between measurements and the proposed correlation is quite good.

OTHER PREDICTIVE TECHNIQUES

In view of the practical importance of the problem, it is not surprising that a very large number of predictive techniques have been proposed. Indeed, almost all of the references to this paper contain a new correlation. Only a few better known solutions are discussed here.

By far the best known solution is that by Nusselt, assuming zero vapor shear at interface and laminar flow of liquid. It has long been recognized that this equation almost always predicts too low. In another analysis, Nusselt included interfacial vapor shear but retained the assumption of laminar liquid flow. Jakob *et al.* [4] found reasonable agreement of their data with this latter analysis. However, calculations by this method are very cumbersome and no other report of comparison of data with this method could be found.

The correlation of Carpenter and Colburn [15] was based on analogy between momentum and heat transfer as indeed most better known correlations are in some way or other. However, its agreement with Carpenter's data is only marginal, the scatter being about +100 to −50%. Soliman *et al.* [13] presented an improved version of the Carpenter–Colburn correlation. They compared their correlation with the data from [1, 14, 18, 20] and found reasonable agreement. This correlation is also tedious to use and no other report on its evaluation came to this author's notice.

The widely known correlation of Akers *et al.* [2] has been found to predict too low by several researchers. Soliman *et al.* [13] comparing with data from [1, 14, 18, 20]; Bae *et al.* [9] comparing with their own data; and Cavallini and Zecchin [16] comparing with their R-11 data reached the same conclusion.

The correlation proposed by Ananiev *et al.* [3] is very simple and correlates their own water data well. Bae *et al.* [9] compared the Ananiev correlation with data of Altman *et al.* [1] and their own data. The former are correlated fairly well, the maximum deviation being −37%. However, the agreement with the data of Bae *et al.* is not good, most of the predictions being between 0 and −60% of the measurements.

A number of authors have carried out elaborate mathematical analyses assuming a uniform liquid layer, zero entrainment, and applicability of single phase velocity profiles to the liquid film. Quite a few other assumptions and simplifications have also to be made. In most cases, the solutions thus developed are very cumbersome to apply. Only two of such analyses are mentioned here. One of the early ones was by Dukler [11] who compared his solution with the data of Carpenter and found good agreement. A recent one is by Traviss *et al.* [21] who compared their own data as well as those of Bae *et al.* [8] with their own solution. While the latter are in agreement with predictions, their own data shows large deviations at vapor qualities higher than 50%.

Comparison with one or two data sets does not provide a conclusive test of a correlation. For proper evaluation, a correlation must be tested against data from many experimental studies covering as wide a range of parameters as possible. Regrettably, most of the proposed predictive techniques have not been evaluated adequately. However, the evidence quoted in the foregoing indicates that the correlation of Akers *et al.* is unsatisfactory. The correlation of Ananiev *et al.* requires further evaluation. Among the other available correlations, only that of Soliman *et al.* has been tested against a fairly wide variety of data and one may feel some confidence in its use. As for the others, there is not sufficient evidence to warrant either their acceptance or rejection.

CONCLUDING REMARKS

In the foregoing, the correlation has been compared with a large amount of varied data from many sources. While the agreement cannot be considered excellent, in all cases it is close enough to be considered satisfactory for practical designs. The accuracy of prediction is by no means inferior to that obtained with highly complicated analyses. It could be that among the many solutions presented by other researchers, some may be equally satisfactory or better. However, none of them have been tested against a sufficient variety of data to substantiate such a belief. It is therefore suggested that this correlation be given consideration for practical designs.

Some areas for further research are suggested here. The correlation needs to be tested at reduced pressures outside the range of 0.002–0.44. Failure close to the critical pressure will not be very surprising. The correlation could perhaps be improved in the quality range 0.85–1.0 by incorporating a parameter to take into account entrainment effects. Much more data for annuli need to be analyzed, especially at lower Reynolds numbers. It could be that at low Reynolds numbers, h_L in annuli may have to be calculated by some other equation. Finally, it is suggested that data on condensation of metallic vapors be analyzed in terms of Z and Ψ but calculating h_1 by an equation suitable for turbulent flow of metallic fluids.

The original manuscript of this paper was submitted while the author was with Gilbert-Commonwealth.

REFERENCES

1. M. Altman, F. W. Staub and R. H. Norris, Local heat transfer and pressure drop for refrigerant 22 condensing in horizontal tubes, ASME-AIChE, Storrs, Conn. (August 1959).
2. W. W. Akers, H. A. Deans and O. K. Crosser, Condensing heat transfer within horizontal tubes, *Chem. Engng Prog. Symp. Ser.* **59**(29), 171–176 (1958).
3. E. P. Ananiev, L. D. Boyko and G. N. Kruzhilin, Heat transfer in the presence of steam condensation in a horizontal tube, *Int. Dev. Heat Transfer*, Part 2, 290–295 (1961).
4. M. Jakob, S. Erk and H. Eck, *ForschHft. Geb. IngWes.* **3**, 290–295 (1932).
5. J. B. Tepe and A. C. Mueller, Condensation and subcooling inside an inclined tube, *Chem. Engng Prog.* **43**, 267–278 (1947).
6. M. M. Shah, A new correlation for heat transfer during boiling flow through pipes, *ASHRAE Transactions* **82**(2), 66–86 (1976).
7. N. Z. Azer, L. V. Abis and H. M. Soliman, Local heat transfer coefficients during annular flow condensation, *ASHRAE Transactions* **78**, 135–143 (1972).
8. S. Bae, J. L. Maulbetsch and W. M. Rohsenow, Refrigerant forced convection condensation inside horizontal tubes, M.I.T. Report DSR-79760-64 (1969).
9. S. Bae, J. L. Maulbetsch and W. M. Rohsenow, Refrigerant forced convection condensation inside horizontal tubes, M.I.T. Report DSR-79760-59 (1968).
10. C. K. Powell, Condensation inside a horizontal tube with high vapor velocity, MS thesis, Purdue University (1961).
11. A. E. Dukler, Fluid mechanics and heat transfer in vertical falling film system, ASME–AIChE, Heat Transfer, Storrs, pp. 1–10 (1960).
12. J. Borchman, Heat transfer of high velocity vapors condensing in annuli, *ASHRAE Transactions* **73**, VI. 2.1–VI. 2.13 (1967).
13. M. Soliman, J. R. Schuster and P. J. Berenson, A general heat transfer correlation for annular flow condensation, *J. Heat Transfer* **90**, 267–276 (1968).
14. F. G. Carpenter, Heat transfer and pressure drop for condensing pure vapors inside vertical tubes at high vapor velocities, Ph.D. Thesis, University of Delaware (1948).
15. F. G. Carpenter and A. P. Colburn, The effect of vapor velocity on condensation inside tubes, in General Discussion on Heat Transfer, The Institute of Mechanical Engineers and ASME, pp. 20–26 (July 1951).
16. A. Cavallini and R. Zecchin, High velocity condensation of R-11 vapors inside vertical tubes, in *Heat Transfer in Refrigeration*, pp. 385–396, International Institute of Refrigeration (1971).
17. *Handbook of Fundamentals*, ASHRAE, New York (1972).
18. J. H. Goodykoontz and R. G. Dorsch, Local heat transfer and pressure distributions for Freon-113 condensing in downward flow in a vertical tube, NASA TN D-3952 (1967).
19. J. H. Goodykoontz and R. G. Dorsch, Local heat transfer coefficients for condensation of steam in vertical downflow within a 5/8 inch diameter tube, NASA TN D-3326 (1966).
20. J. H. Goodykoontz and R. G. Dorsch, Local heat transfer coefficients and static pressures for condensation of high velocity steam within a tube, NASA TN D-3953 (1967).
21. D. P. Traviss, A. B. Baron and W. M. Rohsenow, Forced convection condensation inside tubes, M.I.T. Report DSR-72591 (1971).
22. D. P. Traviss, W. M. Rohsenow and A. B. Baron, Forced convection condensation inside tubes: A heat transfer equation for condenser design, *ASHRAE Transactions* **79**(1) (1973).

UNE FORMULATION GENERALE POUR LE TRANSFERT THERMIQUE PENDANT LA CONDENSATION EN FILM DANS LE TUBES

Résumé—On présente une formule simple, sans dimension, pour prévoir les coefficients de transfert thermique lors de la condensation en film dans les tubes. Elle a été vérifiée par comparaison avec un grand nombre de données expérimentales. Celles-ci concernent l'eau, R11, R12, R22, R113, méthanol, éthanol, benzène, toluène et trichloréthylène en condensation sur des tubes horizontaux, verticaux ou inclinés, de diamètre variant entre 7 et 40 mm. On analyse aussi une série d'expériences sur la condensation à l'intérieur d'un espace annulaire. Le domaine des paramètres comprend des pressions réduite allant de 0,002 à 0,44, des températures saturantes entre 21 et 310°C, des vitesses de vapeur entre 3 et 300 m/s, des qualités de vapeur entre 0 et 100%, des flux massiques spécifiques entre 39 000 et 758 000 kg/m^2, des flux thermiques entre 158 et 1 893 000 W/m^2, des nombres de Reynolds pour les liquides entre 1 et 13. La déviation moyenne, pour les 474 données considérées, est égale à 15,4%.

EINE ALLGEMEINE GLEICHUNG FÜR DEN WÄRMEÜBERGANG BEI FILMKONDENSATION IN ROHREN

Zusammenfassung—Es wird eine einfache dimensionslose Gleichung für die Berechnung des Wärmeübergangskoeffizienten bei Filmkondensation in Rohren angegeben. Sie wurde durch Vergleich mit einer Vielzahl experimenteller Daten bestätigt. Diese betreffen die Fluide Wasser, R11, R12, R22, R113, Methanol, Äthanol, Benzol, Toluol und Trichloräthylen, die in horizontalen, vertikalen und geneigten Rohren von 7 bis 40 mm Durchmesser kondensiert wurden. Ein Daten-Satz für Kondensation in einem Ringraum wurde auch ausgewertet. Der Bereich der Parameter umfaßt reduzierte Drücke von 0,002 bis 0,44; Sättigungstemperaturen von 21 bis 310°C, Dampfgeschwindigkeiten von 3 bis 300 m/s, Dampfgehalte von 0 bis 100%. Massenströme von 39 000 bis 758 000 kg/m^2 h, Wärmeströme von 158 bis 1 893 000 W/m^2, auf die Flüssigkeit bezogene Reynolds-Zahlen von 100 bis 63 000 und auf die Flüssigkeit bezogene Prandtl-Zahlen von 1 bis 13. Die durchschnittliche Abweichung für die 474 verwendeten Meßpunkte ergab sich zu 15,4%.

ОБОБЩЕННАЯ ЗАВИСИМОСТЬ ДЛЯ ТЕПЛОПЕРЕНОСА ПРИ ПЛЁНОЧНОЙ КОНДЕНСАЦИИ ВНУТРИ ТРУБ

Аннотация — Предложена простая безразмерная зависимость для расчёта коэффициентов переноса тепла при плёночной конденсации внутри труб. Она подтверждена на большом объёме экспериментальных данных, полученных на таких жидкостях, как вода, R-11, R-12, R-22, R-113, метанол, этанол, бензол, толуол и трихлорэтилен при их конденсации в горизонтальных, вертикальных и наклонных трубах диаметром от 7 до 40 мм. Проведен анализ экспериментальных данных по конденсации внутри кольцевого канала. Исследовались следующие диапазоны изменения параметров: приведенное давление от 0,002 до 0,44; температура насыщения от 21 до 310°C; скорость пара от 3 до 300 м/сек; паросодержание от 0 до 100%; поток массы от 39 000 до 758 000 кг/м^{2}час; тепловой поток от 158 до 1 893 000 Вт/м^2; значения числа Рейнольдса для жидкости от 100 до 63 000 и числа Прандтля для жидкости от 1 до 13. Среднее отклонение для 474 точек анализируемых данных составило 15,4%.