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HEAT TRANSFER DURING FILM CONDENSATION IN TUBES AND ANNULI: A REVIEW OF THE LITERATURE

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ABSTRACT

This paper reviews the available information for estimation of heat transfer during film condensation in tubes and annuli. Emphasis is on those fluids which are used in air conditioning and refrigeration. These are ammonia, halocarbon refrigerants, and water. Tables are provided which summarize the available experimental data giving the range of important parameters covered in various studies. Among the topics covered are condensation of low velocity and high velocity vapors, effects of superheat and noncondensibles, effect of oil, effect of return bends, and interfacial phase change resistance. Predictive techniques and experimental data are critically reviewed and recommendations for design calculations are made. Information on further verification of the Shah correlation is also given.

INTRODUCTION

The purpose of this paper is to review the available information for the estimation of heat transfer during film condensation in plain tubes and annuli, with emphasis on fluids used in refrigeration and air conditioning systems. The refrigerants commonly used in vapor compression machines are ammonia and halocarbon refrigerants (freons). Carbon dioxide is no longer used in air conditioning but continues to be a refrigerant in the form of dry ice. Water is the refrigerant in absorption refrigeration machines of the lithium bromide-water type. It is also the refrigerant in steam jet ejector system which is still in limited use. Besides, water is extensively used in steam-heating systems. A variety of chemicals such as propane, isobutane, methyl chloride, and natural gas, are occasionally used in vapor compression machines. Thus design information and techniques are needed not only for the common refrigerants, but also for a wide variety of fluids. Therefore general predictive techniques have been given whenever possible.

The following topics are covered in this paper:

1. Condensation of stagnant and low velocity vapors
2. Condensation of high velocity vapors
3. Effect of vapor superheat
4. Effect of non-condensible gases
5. Effect of oil
6. Effect of return bends
7. Interfacial phase-change resistance

The available predictive techniques and experimental data on these topics have been reviewed. Design recommendations have been made wherever possible. Extensive references to experimental studies and data sources have been provided.

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EXPERIMENTAL STUDIES

Numerous studies on condensation inside tubes have been carried out. Some of these are listed in Table 1 along with the range of parameters covered by each study. There have been comparatively few experiments on condensation in annuli. All the experimental studies known to this author are listed in Table 2. Furthermore, all studies on condensation of ammonia in tubes and annuli known to this author are listed in Tables 1 and 2. The information in these tables would be helpful to researchers and design engineers in locating the data sources for the parameters of interest to them.

STAGNANT VAPOR IN VERTICAL TUBES

Laminar Condensate Film

Nusselt developed analytical solutions for condensation on the surface of a vertical tube from stagnant vapor, the liquid film being assumed to be laminar. In terms of the liquid film Reynolds number the Nusselt equation for local heat transfer coefficient may be written as:

$$h_{TP}\left[\frac{\mu_1^2}{k_1^3\,\rho_1(\rho_1-\rho_g)g}\right]^{1/3} = 1.10\ Re_{1f}^{-1/3} \tag{1}$$

The equation for mean heat transfer coefficient is written as:

$$\bar{h}_{TP}\left[\frac{\mu_1^2}{k_1^3\,\rho_1(\rho_1-\rho_g)g}\right]^{1/3} = 1.47\ Re_{1f}^{-1/3} \tag{2}$$

The liquid film Reynolds number is defined as:

$$Re_{1f} = \frac{4w}{\pi D\mu_1} \tag{3}$$

w is the mass flow rate of condensate. For condensation inside a tube, D is the ID of tube. For condensation on the inner tube of an annulus, D is the OD of the inner tube of the annulus. Derivation of these equations and their alternative forms may be found in many books including that by McAdams[25].

Nusselt equations have had extensive theoretical and experimental verification. Koh et al.[38] and Chen[39] carried out rigorous theoretical analyses in which several factors which had been neglected by Nusselt were considered. Among these are the effect of drag by stagnant vapor on moving condensate and inertia effects. The conclusion was that for liquids with Prandtl number of 1 or higher, the effect of these factors is neglegible.

McAdams[25] studied the experimental data of many early researchers and concluded that Nusselt equations may be used up to a film Reynolds number of 1800. However, he recommended increasing by 20% the mean heat transfer coefficients predicted by the Nusselt equation. Walt and Kroger[40] studied condensation of R-12 in a vertical tube and found that the ratio of measured to predicted coefficients increased from 1.03 to 1.15 as Re_{1f} increased from 5 to 28. Cavallini and Zecchin[16] condensing R-11 in a vertical tube at very low vapor velocities found their measurements 20% higher than Nusselt predictions. Borchmann[12,29] condensed R-11, R-12, and methanol in vertical annuli and found that Nusselt equations gave good correlation at u_G less than 3 m/s. Thus it can be concluded that for laminar condensate film in stagnant vapor, Nusselt equations can be used with confidence.

Turbulent Condensate Film

The liquid film Reynolds number increases with distance from entrance. For long tubes, Re_{lf} may exceed the critical value and the condensate film would then become turbulent. For such cases, the Kirkbride correlation[41] may be used to calculate the mean heat transfer coefficient for the entire length. The Kirkbride equation is:

$$\bar{h}_{TP}\left[\frac{\mu_l^2}{k_l^3\,\rho_l^2\,g}\right]^{1/3} = 0.0077\ Re_{lf}^{0.4} \qquad (4)$$

This equation was developed from and verified with data for diphenyl, gasoline, Dowtherm A, and steam. Calculations may also be done using the analytical solutions of Dukler[11] and Rohsenow et al.[42] for zero vapor shear. However, these authors have not provided any comparison of their solutions with experimental data . Hence the author recommends the Kirkbride correlation.

LAMINAR CONDENSATION IN HORIZONTAL AND INCLINED TUBES AND ANNULI

Nusselt derived the following equation for heat transfer during laminar condensation from stagnant vapor on the external surface of a horizontal tube:

$$h_{Nu} = 0.725\left[\frac{\rho_l(\rho_l-\rho_g)g\ i_{fg}\ k_l^3}{D\ \mu_l(T_{SAT}-T_w)}\right]^{0.25} \qquad (5)$$

This equation would also be applicable to condensation inside tubes if the vapor velocity is neglegible and there is no accumulation of liquid at the bottom of the tube. However, liquid collects in the bottom of the tube and there is very little heat transfer from the liquid surface. The greater the depth of liquid, the lower the heat transfer coefficient.

Chaddock[36] and Chato[37] assumed that the liquid depth decreases along the tube length and that flow occurs due to the difference in liquid depth. Chato developed analytical solutions for horizontal and inclined tubes. His solution for horizontal tubes simplifies to the following relation:

$$h_{TP} = 0.77\ h_{Nu} \qquad (6)$$

where h_{Nu} is given by Eq 5.

The Chato solution for inclined tubes requires iterative solution of three equations. Numerical tables have been provided to assist in calculations. Ref 37 should be consulted for details. Chato also carried out experiments on condensation of R-113 and the data showed fair agreement with his equations. He recommended the use of his equations for Re_G less than 35,000 as above this value, liquid film may become turbulent.

Kroger[45] carried out experiments on condensation of R-12 in tubes of 6.1 to 12.5 mm ID inclined to the horizontal from 0° to 10° downwards. Agreement with Chato[37] equation was good. However, Kroger recommends that for horizontal tubes , predictions of Eq 6 be reduced by 10%. Kern[46] recommends a multiplier of 0.8 (instead of 0.77) in Eq 6.

Jaster and Kosky[49] developed the following equation for horizontal tubes:

$$h_{TP} = \alpha^{3/4}\ h_{Nu} \qquad (7)$$

The void fraction is calculated by the correlation of Zivi[50] which is:

$$\frac{1}{\alpha} = 1 + \frac{1-x}{x}\,(\rho_g/\rho_l)^{2/3} \qquad (8)$$

Rufer and Kezios[47] pointed out that the physical model used by Chato[37] is applicable only to open channel flows. In a closed system, flow of condensate occurs due to system pressure and the liquid depth increases towards the tube exit. Based on this model, they developed equations for calculating void fraction.

Nilsson[48] condensed R-22 in horizontal and inclined tubes. Tube inclinations were 1^o and 3^o downwards and 1^o upwards. With upward inclination, heat transfer decreased compared to horizontal tubes. With downward inclination, heat transfer increased. The results of Nilsson for horizontal and downward inclined tubes are given in Table 3. It is seen that the shorter tubes had higher heat transfer. This is understandable as the condensate collection in a shorter tube would be less. However, the Nilsson results are in conflict with those of Kroger[45]. The L/D ratio in Kroger's tests varied from 47 to 100. Yet their recommended multiplier to h_{Nu} is 0.69 compared to 0.88 given by Nilsson.

The present author compared the data of Borchmann[29] for R-11 entering a horizontal annulus with a vapor velocity of 1 m/s. Eq 5 was found about 30% higher than the measurements. The data of Trapp[32] for oil-free ammonia in a horizontal annulus give about the same result. The size of the outer tube in these tests is not known and hence it is not certain that this reduction is due to liquid accumulation.

Fig. 1 shows the comparison of various correlations for horizontal tubes. The predictions of Jaster-Kosky correlation for water at 1, 10, and 48 bar are plotted. These correspond to reduced pressures of 0.005, 0.047, and 0.22. The reduced pressures of most refrigerant and steam condensers would be in this range. It therefore appears that the Jaster-Kosky correlation generally overpredicts the heat transfer coefficients.

The Chato correlation for horizontal tubes is in reasonable agreement with the Kern recommendation, the data of Kroger, the data of Nilsson, as well as his own data, and is hence recommended. It is also recommended for horizontal annuli as it agrees fairly well with the data of Borchmann[29]. For inclined tubes, the corresponding Chato correlation is recommended. The results of Nilsson (Table 3) may be used for preliminary estimates.

THE ONSET OF TURBULENCE

Rohsenow et al.[42] analyzed condensation on a plate. They found that the film Reynolds number at which the liquid film becomes turbulent is a function of interfacial shear. In the absence of interfacial shear, the transition Reynolds number was found to be about 1800. This is in agreement with the conclusion of McAdams[25] based on experimental data. At higher vapor shear, the transition Reynolds number was found to be as low as 50. The results of Dukler analysis[11] are similar. Carpenter and Colburn[15] examined their high velocity data and concluded that the transition film Reynolds number was 240.

As was mentioned earlier, Borchmann[29,12] found good agreement with Nusselt equations for $u_G < 3$ m/s. The author has found that his correlation[23] for turbulent condensation is satisfactory if $u_G > 3$ m/s. Chato[37] recommends that his laminar film condensation analysis can be used for $Re_G < 35000$. Based on these three observations, the author suggests that Nusselt equations(and other based on laminar flow and stagnant vapor) are applicable if all the following three conditions are satisfied:

$$u_G < 3 \text{ m/s}, \quad Re_L < 1800, \quad Re_G < 35000$$

TURBULENT CONDENSATION WITH VAPOR SHEAR

Annular Flow Model Analysis

In most condensers with high velocity vapor, annular flow pattern prevails over much of the length of the tube. This has prompted numerous mechanistic analyses of the annular flow models. Perhaps the first to attempt such an analysis were Carpenter and Colburn[15] but the accuracy of their equations is poor. The Carpenter- Colburn correlation was improved upon by Soliman et al.[13] Calculations with the Soliman et al. correlation are very tedious and no other report of its evaluation has come to the author's notice.

Many other analyses have been performed following one of two approaches. The first approach is that of the Martinelli analogy in which it is assumed that the velocity in the liquid film can be predicted using universal velocity correlations based on single-phase data. The other approach is that of the Deissler analogy in which it is assumed that the eddy diffusivity of heat and momentum can be predicted using correlations based on single-phase data. Dukler[11] and Razavi and Damle[51] followed this latter approach. Altman et al.[1], Kosky and Staub[52], Traviss et al.[22], Cavallini and Zecchin[17], and Azer et al.[7], are among those who used the Martinelli analogy approach. With either approach, the following two equations developed by Prandtl are solved:

$$\frac{q}{C_{pl}\,\rho_l} = -(\alpha + \epsilon_H)\frac{dT}{dy} \tag{9}$$

$$\frac{\tau}{\rho_l} = (\nu + \epsilon_M)\frac{du}{dy} \tag{10}$$

Estimation of shear stress τ requires the use of correlations for void fraction and pressure drop. Those of Zivi[50] and Lockhart-Martinelli[43] are most commonly used. Other assumptions include uniform thickness of liquid layer, zero liquid entrainment, and equality of momentum and heat eddy diffusivities. Closed form solution of the resulting equations has not been possible without further simplifying assumptions. Most researchers therefore proceeded to solve them numerically on computers. Dukler[11] has arranged his numerical solution in the form of graphs which can be used for design calculations. Azer at al.[7] have fitted an equation to their numerical results. Kosky and Staub[52] and Traviss et al.[22] have developed design equations through further simplifying assumptions. The solution of Traviss et al.[22] is simpler and is expressed by the following equations:

$$\frac{h_{TP}\,D}{k_l} = F_1\,\frac{Pr_l\,Re_l^{0.9}}{F_2}\,,\quad 0.15 < F_1 < 15 \tag{11}$$

where

$$F_1 = 0.15\left[X_{tt}^{-1} + 2.85\,X_{tt}^{-0.476}\right] \tag{12}$$

F_2 is given by the following equations:

$$Re_l < 50,\qquad F_2 = 0.707\,Pr_l\,Re_l^{0.5} \tag{13a}$$

$$50 < Re_l < 1125,\qquad F_2 = 5\,Pr_l + 5\ln\left[1 + Pr_l(0.0964\,Re_l^{0.585} - 1)\right] \tag{13b}$$

$$Re_l > 1125,\qquad F_2 = 5\,Pr_l + 5\ln(1 + 5Pr_l) + 2.5\ln(0.00313\,Re_l^{0.812}) \tag{13c}$$

These equations were derived assuming $Pr_l > 3$ and using the Lockhart-Martinelli correlation for two-phase frictional pressure drop. These were found to be in good agreement with the data of Bae et al.[8]. But the data of Traviss et al.[21] were considerably higher at vapor qualities greater than 50%.

Many objections to the annular flow analyses may be raised such as they ignore liquid entrainment and liquid-vapor interface waves, assume $\epsilon_H = \epsilon_M$ while such is not the case, use empirical correlations for pressures drop of limited accuracy, prediction of flow pattern is not reliable, etc. Besides, their reported agreement with experimental data is not impressive. Razavi and Damale[51] compared their own solution as well as several others with 38 data points for water and various chemicals from the tests of Carpenter[14] and Goodykoontz-Dorsch[19,20]. Their own solution came out best with a mean deviation of 48%. The Dukler solution gave a mean deviation of 57%. Various annular flow model analyses have been discussed by Butterworth[28].

The Shah Correlation

The most verified predictive technique presently available is the author's correlation[23]:

$$\Psi = h_{TP}/h_1 = 1 + 3.8/Z^{0.95} \tag{14}$$

The parameter Z is defined as:

$$Z = \left(\frac{1}{x} - 1\right)^{0.8} p_r^{\,0.4} \tag{15}$$

The superficial heat transfer coefficient h_1 is calculated as:

$$h_1 = h_L\,(1 - x)^{0.8} \tag{16}$$

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

$$h_L = 0.023\left(\frac{GD}{\mu_1}\right)^{0.8} Pr_1^{0.4}\,\frac{k_1}{D} \tag{17}$$

For annuli, D is replaced by D_{HYD}, the hydraulic equivalent diameter. All properties are calculated at the saturation temperature.

This correlation was originally developed using data for p_r up to 0.44. The data of Miropolskiy et al.[44] for water have now also been analyzed. These cover reduced pressures up to 0.82 and show good agreement with the Shah correlation.

In Ref 23, only a few data points for annuli from the tests of Borchmann[12] were analyzed. These showed increasing deviation with decreasing Re_1. Additional data for R-11, R-12, and methanol from tests by Borchmann[12,29] have now been analyzed. The range of parameters covered by these tests is included in Table 2. Satisfactory agreement was found down to a Re_L of 358 and Re_1 of 30. The reason for the large deviations at low Re_1 reported in Ref 23 appears to be entrance effects. Fig. 2 shows some data from a run with R-11 in a horizontal annulus. The predictions of the Shah correlation are shown with and without entrance effect correction factor. To correct for entrance effects, h_{TP} from Eq 14 was multiplied by $[1 + (D_{HYD}/L)^{0.7}]$ as recommended by McAdams[25] for single-phase flows. With this correction, excellent agreement with the Shah correlation is seen. Most of the Borchmann data show the same trend.

Table 4 lists the complete range of parameters over which the Shah correlation has been well-verified. Table 5 lists the deviations of individual data sets compared to this correlation. Thus 777 data points for ten fluids covering a very wide range of parameters have been correlated with a mean deviation of 15.7%. The data included horizontal, vertical, and inclined tubes as well as horizontal and vertical annuli. The Shah correlation is therefore recommended for general use provided all the following conditions are fulfilled:

$$u_G > 3\,\text{m/s}, \quad Re_L > 350, \quad Re_G > 35{,}000$$

Other Correlations

A large number of other correlations have been proposed. Very few of them have been adequately tested with data. A few which have had some evaluation are mentioned here.

A well-verified correlation for halocarbon refrigerants is that of Cavallini and Zecchin[17] which is written as:

$$h_{TP} D/k_1 = 0.05 \, Re_{eq}^{0.8} \, Pr_1^{0.33} \tag{18}$$

where

$$Re_{eq} = Re_1 + \left(\frac{\mu_g}{\mu_1}\right) \left(\frac{\rho_1}{\rho_g}\right)^{0.5} Re_g \tag{19}$$

The data used for its verification include fluids R-11, R-12, R-21, R-22, R-113, and R-114. The ratio (ρ_1/ρ_g) varied from 11 to 314, and Re_1 from 7,000 to 53,000. The standard deviation of data sets varied from 8 to 47%. Hence this correlation can be used with confidence for halocarbon refrigerants in the range of dimensionless parameters covered. Its applicability to water cannot be assumed without verification as its properties differ greatly from halocarbon refrigerants.

A well-known correlation is that of Ananiev et al.[3] which is as follows:

$$h_{TP} = h_L \, (\rho_1/\bar{\rho} \,)^{0.5} \tag{20}$$

The mean vapor-liquid mixture density $\bar{\rho}$ is calculated assuming homogeneous flow as:

$$\bar{\rho} = \frac{\rho_1 \, \rho_g}{\rho_g + x \, (\rho_1 - \rho_g)} \tag{21}$$

The Ananiev et al. correlation agrees well with their own water data as well as those of Miropoloskiy et al.[44] for pressures between 4 and 180 bar. However several other researchers have reported unsatisfactory agreement with data[52,1,9]. It is suggested that the use of this correlation be restricted to water between 4 and 180 bar.

The well-known correlation of Akers et al.[2] has been evaluated by several researchers and found always to predict too low[9,24,16,7]. This correlation is clearly unsatisfactory.

EFFECT OF OIL

In all foregoing discussions, condensing fluid has been assumed to be free of oil or any other contaminent. In most vapor-compression refrigeration machines, significant amounts of lubricating oil get mixed with refrigerant and reach the condenser. Hence the effect of oil on heat transfer is a subject of much interest to refrigeration engineers. The vast majority of refrigeration mechines presently in use employ halocarbon refrigerants which are soluble in oil. Regrettably, the author has been unable to locate any experimental study on oil-halocarbon refrigerant condensation. Some data on condensation of oil containing ammonia and carbon dioxide were found and these are included in Table 1. Both of these refrigerants have negligible solubility in oil.

Mazukewitch[34] condensed ammonia on the inner tube of a vertical annulus 16 mm ID, 46 mm OD. Experiments were done with the surface clean, rusty, and oil-smeared rusty. The results are shown in Fig. 3. It is seen that the data for the oil-smeared rusty surface are about 30% lower than those for the oil-free rusty surface.

Abdulmanov and Mirmov[58,30] report that in U.S.S.R., it is generally believed that the presence of oil in ammonia condensers results in oil film thicknesses of 0.05 to 0.08 mm and design calculations are done on this basis. It is interesting that Shah[6] in his experiments with evaporation of ammonia in tubes actually observed oil films. By analyzing his single-phase liquid heat transfer data, he estimated oil film thicknesses of 0.04 to 0.10 mm.

Abdulmanov and Mirmov[58] condensed ammonia on a horizontal tube 25 mm OD enclosed in a large shell. Tests were done with and without oil. The tests indicated that oil increased the heat transfer by 30%. Mirmov and Yemelyanov[30] have expressed the opinion that the low heat transfer coefficients found in practice are caused by presence of air and not because of any oil film.

Thus the available information about effect of oil on ammonia condensers is apparently conflicting. More investigation is needed. In the tests of Kratz et al.[24], ammonia containing oil was condensed. Useful information about the effect of oil could probably be gathered by comparing these data with the Shah correlation. Similar analysis of experimental data of Schmidt[33] for oil containing carbon dioxide may also yield useful information. Some further experimental work with oil-immiscible refrigerants seems desirable.

EFFECT OF NON-CONDENSIBLE GASES

Presence of non-condensible gases, mainly air, decreases heat transfer. A layer of air forms at the liquid interface through which the condensing vapor must diffuse before reaching the liquid surface. A sharp drop in pressure and temperature of condensing vapor occurs across this layer causing reduction in heat transfer. The effect of air is very strong in condensation of stagnant vapors as the the transfer of vapor through the air layer is by moleculer diffusion. In forced convection, the effect of air is much less. Furthermore, the effect is greater at lower pressures. Sparrow et al.[53,54] have carried out a comprehensive analysis for laminar film condensation. Numerical results for steam are provided in graphical form which may be used, where applicable, for design calculations.

Fig. 4 shows the data of Mazukewithch[34] for condensation of ammonia containing air in a vertical annulus.

For further details and methods of calculation, the book by Collier[35] may be consulted.

EFFECT OF VAPOR SUPERHEAT

In all discussion till now, the vapor has been assumed to be at saturation temperature. In virtually all vapor-compression systems, the vapor entering the condenser is superheated. Hence the prediction of heat transfer with superheated vapor is of much practical interest.

Stagnant Vapor

While the bulk of the vapor is superheated, a thermal boundary layer is created adjacent to the liquid surface in which the vapor temperature drops to ~~the liquid surface in which the the vapor temperature drops to~~ the interface temperature. If the effects of free convection in the vapor boundary layer are neglected, the problem may be analyzed by extending the basic Nusselt theory to get the following result:

$$\frac{h}{h_{SAT}} = \left[\frac{i_{fg} + C_{pg}\,(T_g - T_{SAT})}{i_{fg}} \right]^{0.25} \tag{22}$$

h is the heat transfer coefficient of superheated vapor and h_{SAT} is that of saturated vapor. Under most conditions of practical interest, Eq 22 predicts an increase of only a few percent even with high superheats. Virtually all experimental studies agree with this result. For example Walt and Kroger[40] condensed R-12 with superheats up to 128 °C in a vertical tube and found increases up to 8% over the Nusselt equation for saturated vapor. Minkowycz and Sparrow's theoretical analysis[53] showed a maximum increase of 5% for steam with superheat up to 220 °C. Thus for stagnant vapors (and very low velocity vapors), the effect of superheat may be neglected and calculations done assuming saturated vapor.

Moving Vapor

If the tube wall temperature is above the saturation temperature, heat transfer to vapor occurs purely by single-phase convection and can be predicted by single-phase correlations. If the wall temperature is sufficiently lower than the saturation temperature, condensation starts while the vapor core is still superheated.

As yet no thoroughly verified technique for predicting heat transfer under such non-equilibirium conditions is available. The best available appears to be the correlation of Miropolskiy et al.[44]. They state that even if the wall temperature is below the saturation temperature, condensation will not start unless the vapor temperature is below a border temperature T_B and the vapor quality below a border quality x_B defined by the following equations:

$$T_B = T_{SAT} + \frac{q}{h_B} \tag{23}$$

$$x_B = 1 + \frac{q\ \bar{C}_{pg}}{h_B\ i_{fg}} \tag{24}$$

$\bar{C}_{pg}$ is the mean of vapor specific heats over the range of T_{SAT} and T_B. h_B is calculated by an equation suitable for superheated vapor. If $x > x_B$ and $T_g > T_B$, heat transfer to vapor occurs purely by single-phase convection. For $1 < x < x_B$, their correlation has the functional form:

$$\frac{h_{TP} - h_B}{(h_{TP})_{x=1} - h_B} = fn\ \left(\frac{x-1}{x_B-1}\right) \tag{25}$$

The functional relation is expressed graphically in Fig. 5. To interpret this figure correctly, it is to be noted that when the right hand side of Eq 25 is zero, the left hand side is equal to 1, and vice-versa. Fig. 5 also shows good agreement with data for water between 4 and 180 bars. h_{TP} at $x = 1$ may be calculated by substituting $x = 1$ in the Ananiev correlation, Eq 20. The Shah correlation may also be used with $x = 0.99$.

An important point regarding the calculation of h_B is that Miroposkiy et al.[44] found that to correlate their data for superheated steam in terms of bulk fluid properties, the constant multiplier in Eq 17 should be doubled. This is because of the effect of variation of fluid properties across the cross-section of tube. The procedure generally adopted for calculating heat transfer of superheated fluids is to evaluate the fluid properties at the film temperature defined as the arithmetic mean of wall and bulk temperature.

Miropolskiy et al. developed this correlation only for steam, and the correlating lines in Fig. 5 as given by them were in terms of absolute pressure of water. The present author has replaced absolute pressures by reduced pressures in the hope that it may also be applicable to other fluids. Whether it is in fact generally applicable to fluids other than water remains to be verified.

EFFECT OF BENDS

Traviss and Rohsenow[57] carried out experiments on condensation of high velocity R-12 in a tube downstream of a return bend similar to those generally used in air cooled refrigeration condensers. Their conclusion was that the return bends have no perceptible effect on heat transfer.

INTERFACIAL PHASE-CHANGE RESISTANCE

For the transfer of mass across the vapor-liquid interface, a driving potential is needed. Therefore the pressure and temperature at the interface must be lower than that well away. On this point there could be no disagreement. The question is about the magnitude of this interfacial resistance. The interfacial heat transfer coefficient can be calculated by the following equation[35]:

$$h_i = \frac{2\sigma}{2-\sigma} \left[\frac{1}{2\pi R T_{SAT}}\right]^{0.5} \frac{i_{fg}^2 \, P}{R \, T_{SAT}^2} \qquad (26)$$

σ is known as the condensation or accomodation coefficient and represents the fraction of molecules striking the liquid surface which are captured. Very low values of σ are given by many theories and experiments. On examining the data from a number of experimental studies on condensation of metals, Rohsenow[55] concluded that σ is always close to 1. The reported low values are due to the presence of non-condensibles and measurements errors. Berman[56] reached the same conclusion on examining a large amount of data for non-metallic fluids.

The author recommends that σ be taken as 1 under all conditions for all fluids. With $\sigma = 1$, h_i predicted by Eq 26 would be several orders of magnitudes higher than the heat transfer coefficient of the liquid film for the fluids and conditions of interest to refrigeration engineers. Hence the phase change heat transfer resistance may be neglected.

DESIGN RECOMMENDATIONS

1. If $u_G < 3$ m/s, $Re_L < 1{,}800$, and $Re_G < 35{,}000$, the recommendations are as follows:

   a. For vertical tubes and annuli, use the Nusselt solution, Eq 1 and 2.

   b. For horizontal tubes and annuli, use Eq 6, the Chato correlation.

   c. For tubes and annuli inclined downwards by a few degrees, use the Chato correlation as given in Ref 37. The results of Nilsson as given in Table 3 may be used for preliminary estimates.

   d. The effect of vapor superheat may be neglected.

2. If $Re_L > 1{,}800$ but $u_G < 3$ m/s and $Re_G < 35{,}000$, use the Kirkbride correlation, Eq 4.

3. If $u_G > 3$ m/s, $Re_L > 350$, and $Re_G > 35{,}000$, the following recommendations apply to tubes and annuli in any orientation.

   a. For saturated vapors, use the Shah correlation for all fluids.

   b. For superheated vapors, use the Miropolskiy et al. correlation of Fig. 5. It must be remembered that this correlation has been verified only with steam data. For other fluids, it should be used only if no other predictive technique is available.

4. The effect of return bends on heat transfer may be neglected.

5. Evidence regarding effect of oil in ammonia condensers is conflicting. It would be prudent to reduce pure vapor heat transfer coefficients by 30%, as suggested by the data of Fig. 3.

6. The phase-change interfacial heat transfer resistance may be neglected.

SUMMARY AND CONCLUSION

The available information on heat transfer during condensation in tubes and annuli has been reviewed. For saturated pure vapors, design calculations can be done with confidence over the entire range of interest using the equations of Nusselt, Shah, and Chato. For high velocity superheated vapor, no well-verified predictive technique is available. There is no information available for effect of oil in halocarbon refrigerant condensers. The available information on effect of oil in ammonia condensers is very limited and conflicting. Clear design recommendations have been made wherever possible.

Further research is needed on effect of oil in condensers. Much could be learned by analyzing the data for non-miscible refrigerants from previous experiments. Further experimentation is also needed. 35

Extensive references to experimental studies and data sources have been provided. These should help further research efforts and are also likely to be of value to design engineers.

NOMENCLATURE

C_{pl} specific heat of liquid

C_{pg} specific heat of vapor at constant pressure

D diameter of tube

D_i diameter of inner tube of an annulus

D_o diameter of outer tube of an annulus

D_{HYD} equivalent hydraulic diameter = $D_o - D_i$

F_1, F_2 parameters in Traviss et al. correlation

G total mass flow (liquid + vapor), mass per unit flow area per unit time

g accelaration due to gravity

h heat transfer coefficient

h_{TP} local condensing heat transfer coefficient

h_L heat transfer coefficient assuming all mass to be flowing as liquid

h_l heat transfer coefficient assuming the liquid phase to be flowing alone in the tube

h_i interfacial phase-change heat transfer coefficient

h_{Nu} heat transfer coefficient given by Eq 5, the Nusselt equation for condensation on a horizontal tube

i_{fg} latent heat of vaporization

k_l thermal conductivity of liquid

L distance from the point where condensation starts

L_c length of tube required for condensation from x = 1.0 to x = 0.0

Pr_l Prandtl number of liquid

p absolute pressure

p_r reduced pressure = p/p_c where p_c is the critical pressure

q heat flux

Re_L Reynolds number assuming all mass to be flowing as liquid = $G D/\mu_l$

Re_l superficial liquid Reynolds number = Re_L (1-x)

Re_{lf} Reynolds number of liquid film, defined by Eq 3

Re_G Reynolds number assuming all mass to be flowing as saturated vapor = GD/μ_g

Re_g superficial vapor Reynolds number = $G \times D/\mu_g$

R gas contact

T_g temperature of superheated vapor

T_w wall temperature

T_{SAT} saturation temperature

T_B border temperature as defined by Eq 23

u_G velocity assuming all mass to be flowing as vapor = G/ρ_g

w mass flow rate of condensate

x thermodynamic vapor quality. $x > 1$ for superheated vapor

X_{tt} Martinelli parameter = $(\frac{1}{x} - 1)^{0.9} (\rho_g/\rho_l)^{0.5} (\mu_l/\mu_g)^{0.1}$

Z Shah's correlating parameter given by Eq 15

Greek Symbols

ψ h_{TP}/h_l

ρ density

μ dynamic viscosity

θ angle of inclination to horizontal

σ condensation coefficient

τ shear stress

Subscripts

l of liquid

g of vapor

B at border conditions as defined in Miropolskiy et al. correlation

Superscript

— mean or average

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Table 1. Summary of Experimental Studies on Condensation Inside Tubes

Source	Fluid	Orien-tation	Pipe ID mm	p_r	x	q KW/m^2	$Gx10^{-3}$ kg/hm^2	Re_L	Re_1	u_G m/s	Notes
Cavallini[16]	R-11	V	20.0	0.021 0.036	0.30 1.00	9 35	298 1108	3207 11504	510 2309	15 22	
Powell[10]	R-11	H	12.9	0.035	0.25	16	928	9333	7000	30	
Traviss[21]	R-12	H	8.0	0.14 0.36	0.07 0.95	6 57	547 5517	6000 62900	300 58500	5 18	
Traviss[21]	R-22	H	8.0	0.20 0.36	0.20 0.95	9 85	683 3613	6600 43750	330 35000	4 12	
Bae[9]	R-12	H	12.5	0.16 0.28	0.04 0.94	9 62	752 2715	13300 69000	800 47000	5 11	
Bae[8]	R-22	H	12.5	0.22 0.30	0.17 0.96	19 38	1172 2368	20000 42000	800 35000	7 10	
Altman[1]	R-22	H	8.7	0.27 0.44	0.06 0.96		1094 3091	14400 41000	800 25000	5 8	
Azer[7]	R-12	H	12.7	0.29 0.39	0.30 0.99		757 1611	14700 31200	130 20400	3 6	
Goodykoontz[18]	R-113	V	7.4	0.02 0.09	0.16 0.95	35 268	2993 5273	11000 19000	800 18000	61 79	
Goodykoontz[20]	Water	V	7.4	0.0019 0.011	0.06 0.95	173 1893	318 1640	2500 13000	125 12000	91 305	
Goodykoontz[19]	Water	V	15.9	0.005 0.017	0.00 1.00	63 568	78 268	1320 4560	660 2300	20 71	
Jakob[4]	Water	V	40.0	0.0046	0.70 1.00	47 233	88 171	3670 7370	180 960	40 80	
Ananiev[3]	Water	H	8.0	0.031 0.44	0.00 1.00	0.2 6	1381 5760				
Carpenter[14]	Water	V	11.6	0.005 0.005	0.00 1.00	16 199	59 503	445 5110	200 2500	27 120	
Tepe[5]	Metha-nol	Incli-ned	18.9	0.013 0.014	0.06 1.00	28 31	59 107	900 1600	420 490	14 26	
Carpenter[14]	Metha-nol	V	11.6	0.013 0.014	0.00 1.00	9 82	78 527	386 3460	200 1700	19 93	
Tepe[5]	Benzene	V,H, Inclined	18.9	0.020 0.021	0.02 1.00	13 47	88 317	1500 5000	600 2100	9 33	
Carpenter[14]	Trichlo-roethyl-ene	V	11.6	0.020 0.025	0.00 1.00	6 72	127 1367	1010 10300	500 5000	8 64	
Carpenter[14]	Toluene	V	11.6	0.024 0.026	0.00 1.00	9 41	117 552	1290 6220	600 3000	11 47	
Carpenter[14]	Ethanol	V	11.6	0.016 0.017	0.00 1.00	13 50	39 513	104 2430	50 600	3 74	
Schmidt[33]	CO_2	V	22	0.98 0.82		4 12					a,b
Izumi[27]	R-12	H	7.6 21.2	0.16 0.22	0.00 1.00	6 17	113 2205		15 8000		a,b
Akers[2]	R-12	H	15.8	0.42 0.92			29 1575				b
Akers[2]	Propane	H	15.8	0.52 0.98			6 629				b
Gorodinskaya[26]	Ammonia	H	10.0	0.07 0.12						0 10	b
Kratz[24]	Ammonia	H	50.0	0.09							a,b

Notes:

a. Contained oil

b. Data are not easily analyzable

Table 2. Some Experimental Studies on Condensation in Annuli

Source	Fluid	Orien-tation	D_i mm	D_o mm	p_r	x	$q \times 10^{-3}$ W/m^2	$G \times 10^{-3}$ kg/hm^2	Re_L	Re_1	u_G m/s	Notes
Borchmann[12]	R-11	H	31.2	38.0	.037	0.62 1.00	9 14	102 1021	520 5226	50 1600	4 40	
	R-12	V	31.2	38.0	.166	0.00 1.00	9 20	139 1111	1200 9935	150 993	1 8	
	Methanol	V	31.2	38.0	.013	0.00 1.00	38 90	12 50	358 1524	30 460	11 47	
Borchmann[29]	R-11	H	31.2	49.2	.028	0.00 1.00	15 37	26 553	333 7000	82 700	1 21	
Mazukevitch[34]	Ammonia	V	16.0	46.0	.080		6 23					a,b
Trapp[32]	Ammonia	H	31.0	Not known	.068 .080		6 31					b
Kratz[24]	Ammonia	H	50.0	75.0	.089		2 8					a,b

Notes:

a contained oil

b Data are not easily analyzable

TABLE 3

Results of Nilsson[48] for Condensation of R-22 in Horizontal and Downward Sloped Tubes

		$\overline{h}/h_{Nu}$
L/D < 350	Horizontal tube	0.88
	1^o-3^o inclined	1.05
L/D = 1000	Horizontal tube	0.74
	1^o-3^o inclined	0.87

TABLE 4

The Range of Parameters in which the Shah Correlation Has Been Well-Verified

Parameter	Verified Range
Fluids	Water, R-11, R-12, R-22, R-113, methanol, ethanol, toluene, trichlorethylene, benzene.
Flow channel	Tube, annulus
Flow direction	Horizontal, vertical, 15° incline to horizontal
Tube ID, mm	2.8 to 40
T_{SAT}, °C	21 to 355
x, percent	0 to 100
q, W/m²	158 to 16,000,000
G, kg/m²s	11 to 4000
p, bar	0.7 to 180
p_r	0.0019 to 0.82
Pr_l	1 to 13
Re_L	350 to 100,000
u_G, m/s	3 to 300
Flow pattern	All

TABLE 5

Summary of Comparison of Experimental Data with the Shah Correlation[23].

Data of	Fluids	No. of Data Points Analyzed	Mean Dev.%
Cavallini & 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 & Brown[18]	R-113	23	13.5
Jakob et al[4]	Water	29	15.5
Annaniev et al[3]	Water	67	22.4
Goodykoontz & Dorsch[19]	Water	14	13.0
Goodykoontz & Dorsch[20]	Water	25	13.1
Carpenter[14]	Water	2	18.0
Carpenter[14]	Methanol	2	7.0
Tepe & Mueller[5]	Methanol	4	26.7
Tepe & 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	22.0
	R-12	36	10.0
	Methanol	55	15.1
Borchmann[29]	R-11	21	16.5
Miropolski et al[44]	Water	191	14.0
Total number of data points		777	
Mean Deviation			15.7%

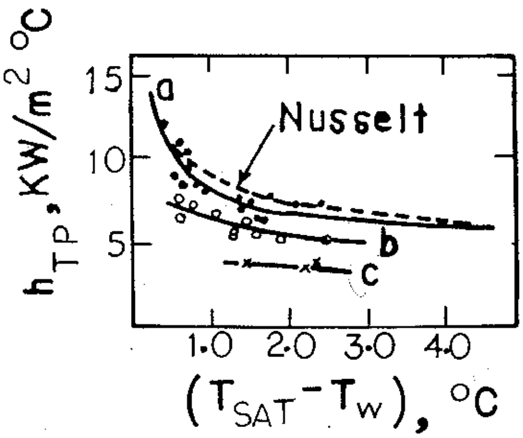

Fig. 3 Effect of oil and rust on condensation of ammonia in a vertical annulus. The various curves are: a = clean tube, no oil; b = rusty tube, no oil; c = rusty oil-smeared tube. Data of Muzukewitch[34]

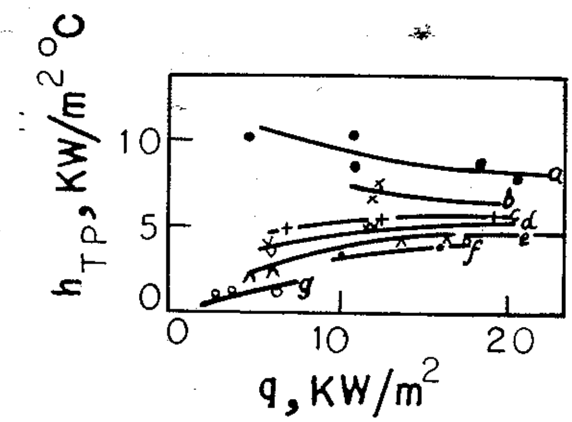

Fig. 4 Effect of air on condensation of oil-free ammonia in a vertical annulus. The lines a, b, c, d, e, f, g, are for volumetric air contents of 0, 1.75, 5.5., 7.2, 11.7, 17.0, and 34.1% respectively. Data of Mazukewitch[34]

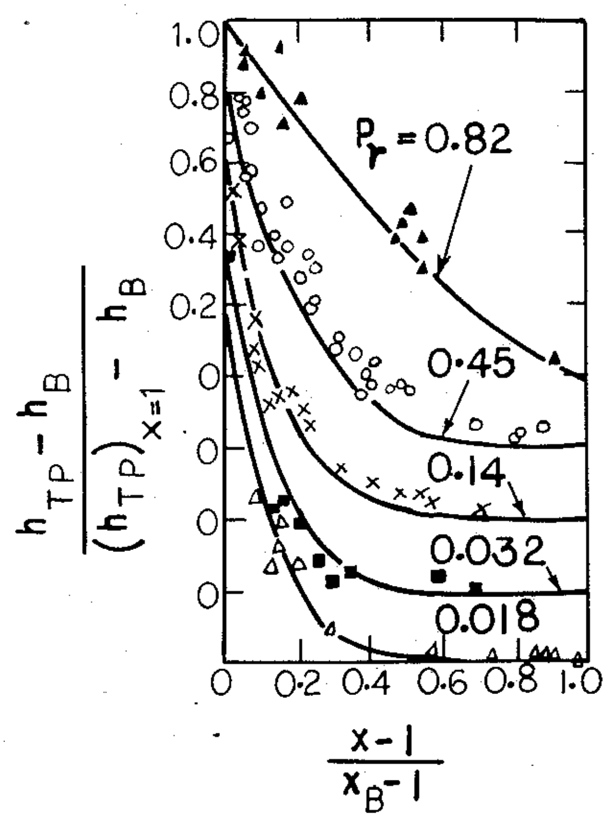

Fig. 5 Correlation of Miropolskiy et al.[44] for condensation of superheated steam in tubes

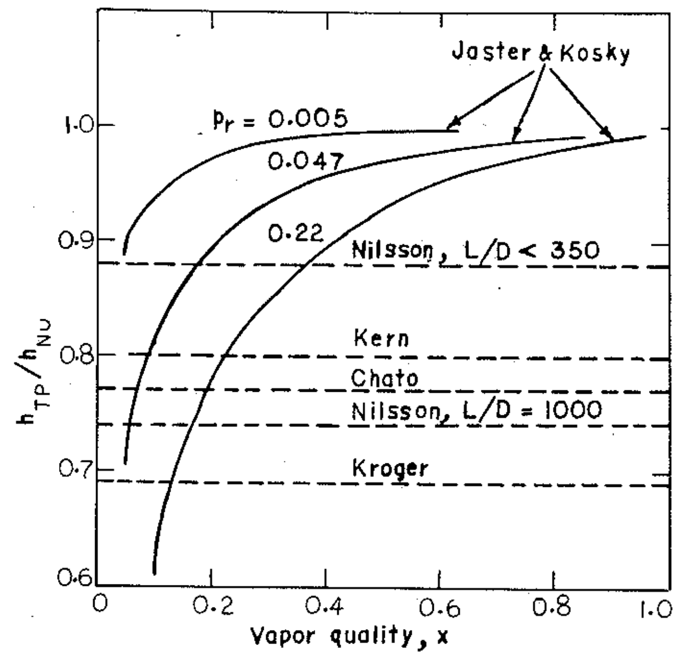

Fig. 1 Comparison of various correlations for laminar condensation in horizontal tube

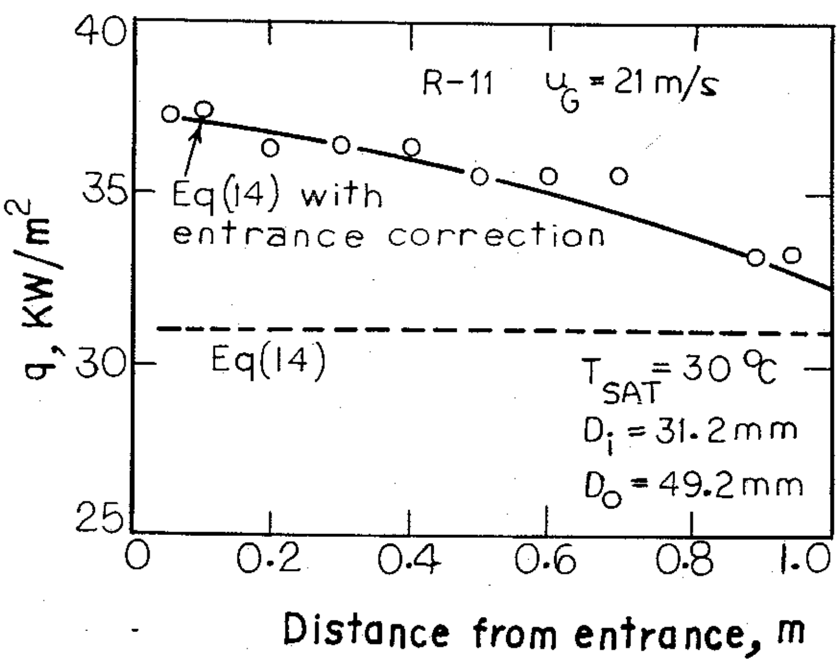

Fig. 2 Comparison of Shah correlation with and without entrance correction factor with the data of Borchmann[29] for condensation in a horizontal annulus cooled by water in counterflow. Water velocity = 4.1 m/s. Water inlet temperature = 15°C. Dry saturated vapor entered the test section.