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Design for a Gas-Cooled Reactor System

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~~Information Limited to
Research and Development~~
Abstract

References to significant design information for a steam generator in a gas-cooled reactor system are presented. Papers concerning heat transfer and pressure drop correlations for gas flow across banks of bare and circular finned tubes and for subcooled water, boiling water and superheated steam flow inside pipes are listed and briefly discussed.

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NOMENCLATURE

A	area
$C_{1,2,3}$ etc	constants
C_p	specific heat
D	characteristic dimension or diameter noted by subscript
$f_{1,2,3}$ etc	friction factors
g_c	gravitation constant
G	mass velocity
h	film coefficient
j	$St(Pr)^{2/3}$
k	thermal conductivity
L	fluid flow length
n	number of fins per unit length
N	number of major restrictions in bank of tubes
Nu	hD/k Nusselt Number - dimensionless
P	pressure
Pr	$C_p \mu / k$ Prandtl Number - dimensionless
Q	heat transfer rate
Re	DG_{max} / μ Reynolds Number - dimensionless
St	$h / C_p G$ Stanton Number - dimensionless
T	temperature
U	overall heat transfer coefficient
v	specific volume
x	weight fraction vapor
Y	fin thickness
α	thermal diffusivity
λ	latent heat of vaporization
μ	viscosity
ρ	density
σ	surface tension
S	Center-to-center distance for tubes

NOMENCLATURE (Continued)

Subscripts

b	bulk
boil	boiling conditions
f	indicates physical properties should be evaluated at convection film temperature
fin	fin property
F	final
i	inside
init	initial conditions
J	Jameson (reference 29)
K	Katz, et al. (reference 33)
l	liquid
m	mean
max	maximum
L	longitudinal
o	outside
s	wall surface
sat	saturated conditions
t	tube
T	transverse
TP	two phase
v	vapor
V	volumetric ($D_V = \frac{4(\text{free volume})}{\text{tube surface area}}$)
W	Ward and Young (reference 25)

INTRODUCTION

The general configuration of a steam generator, as conceived by ORNL, for use with a gas-cooled reactor system consists of the reactor cooling gas flowing past banks of bare or finned tubes in which subcooled water, boiling water and superheated steam are being heated. The design of such a device requires heat transfer and pressure drop information for both the gas and water systems.

It is the purpose of this memorandum to present the results of a rather comprehensive literature search to obtain correlations for the predictions of heat transfer film coefficients and frictional characteristics of banks of bare and finned tubes. The reported results also include heat transfer and pressure drop correlations for subcooled and boiling water and superheated steam flowing in pipes. The references which, in the author's opinion, represent the most significant contribution in their respective fields, in general, will be emphasized. The work will be presented under the general groups of Gas Flow and Water Flow and will be subdivided according to the following outline.

Gas Flow (Shell Side)

1. Bare tubes
 - a. heat transfer
 - b. pressure drop
2. Circular finned tubes
 - a. heat transfer
 - b. pressure drop

Water Flow (Tube Side)

1. Subcooled liquid
 - a. heat transfer
 - b. pressure drop
2. Boiling liquid
 - a. heat transfer
 - b. pressure drop
3. Superheated steam
 - a. heat transfer
 - b. pressure drop

Gas Flow (Shell Side)

1. Bare tubes
 - a. Heat transfer

Colburn's^{(1)*} early analysis for the heat transfer to or from gases flowing normal to a single row of bare tubes represents the first attempt to establish a performance criterion for this type of exchanger configuration. His suggested equation for a Reynolds Number range $2000 < Re < 32,000$ is as follows:

$$Nu_F = C_1 (Re_F)^{0.6} (Pr_F)^{1/3} \quad (1)$$

where $C_1 \begin{cases} = 0.33 \text{ for triangular assay} \\ = 0.26 \text{ for rectangular assay} \end{cases}$

Note: The Reynolds number is given by $Re_F = \frac{D_t G_{\max}}{\mu_F}$

The papers of Pierson⁽²⁾, Huger⁽³⁾ and Grimison⁽⁴⁾ appearing in 1937 were the first thorough experimental and analytic studies of the

*Numbers following author's names represent bibliography references.

heat transfer and frictional characteristics of bare tube banks. Pierson and Huge carried out experiments and presented their results in the form of plots of Nusselt number, Nu , versus Reynolds number, Re , (using D_t). Grimson developed similarity correlations using the data of Pierson and Huge and compared his work with the results published at that time in foreign literature. His correlations are presented in convenient graphical form.

In 1949, Omohundro, Bergelin and Colburn⁽⁵⁾, started a study of fluid friction and heat transfer during viscous flow across banks of tubes. By testing idealized models they initiated a long term study to investigate the operating characteristics of commercial shell and tube heat exchangers. Their results indicated that the Colburn equation (Eq. 1) adjusted by the Sieder-Tate⁽⁶⁾ viscosity-ratio correction $\left(\left[\frac{\mu_s}{\mu_b} \right]^{0.14} \right)$, gave a fair prediction of the test data.

Further work with these flow models in the viscous range by Bergelin, Davis and Hull⁽⁷⁾ and Bergelin, Brown, Hull and Sullivan⁽⁸⁾ demonstrated the effects in changes of tube pitch to tube diameter ratio. A heating and cooling viscosity correction was also developed and a somewhat better correlation of the heat transfer data [expressed as $j' = (St)(Pr)^{2/3} \left(\frac{\mu_s}{\mu_b} \right)^{0.14}$] was obtained by plotting j' versus Reynolds number where:

$$Re = \frac{D_v G_{max}}{\mu} \quad (2)$$

and D_v = volumetric hydraulic diameter
 $= \frac{4(\text{free volume})}{(\text{exposed area of tubes})}$

A comprehensive presentation of these studies in the viscous flow regions can be found in reference (9).

A paper by Bergelin, Brown and Doberstein⁽¹⁰⁾ presents experimental data for the Reynolds number range $100 < Re < 10,000$ where Re is calculated with D_t . This work demonstrated the effect of triangular versus rectangular tube pitch upon heat transfer in the transition zone between viscous and turbulent flow. Kays, London and Lo⁽¹¹⁾, using a fine transient test technique, also studied transition zone flow ($500 < Re$ [using D_t] $< 20,000$) of gas normal to tube banks and verified Bergelin's⁽¹⁰⁾ observations on the heat transfer of in-line tube arrangements. Kays, et al. chooses to correlate $j = (St)(Pr)^{2/3}$ versus Re where the Reynolds number is calculated with a hydraulic diameter similar to that of Eq. 2. A plot (j versus Re [using D_t]) is presented comparing their test data with the results of Grimson⁽⁴⁾, Colburn⁽¹⁾ and Bergelin⁽¹⁰⁾, and indicates the accuracy of Colburn's early equation (Eq. 1). For a triangular tube array in a range $500 < Re_e$ (see below) $< 10,000$ their heat transfer data correlated according to the equation

$$j \equiv St(Pr)^{2/3} = C_h (Re_e)^{-0.4} \quad (3)$$

where Re is calculated with a flow hydraulic diameter $D_e = 4A_c L/A$,

A_c = minimum free-flow area

A = heat transfer area

L = flow length path

Kays determines the constant C_h empirically for various tube configurations and presents his results graphically. Ward and Young⁽²⁵⁾ obtained a verification of Kays correlation for several tube

configurations. A thorough presentation of Kays and London's experimental work on banks of bare tubes is presented in reference (12).

De Bortoli, Grimble and Zerbe⁽¹³⁾, also using the transient-test technique, have determined heat transfer coefficients for gross and local conditions in the range $35,000 < Re \left[\text{using } D_t \right] < 80,000$. Their results are compared to Colburn's equation and indicate that an extrapolation of the equation to these high Reynolds numbers would not be unreasonable.

The work of Dwyer, Sheehan, Weisman, Horn and Schoner⁽¹⁴⁾, using water, is the only data available for the range $105 < Re \left[\text{using } D_t \right] < 10^6$. They obtain the following heat transfer correlation:

$$Nu = 0.0331(Re)^{0.8}(Pr)^{1/3} \quad (4)$$

In closing this section it might be worthwhile to mention the analysis of Weisman⁽¹⁵⁾. This work consists of a highly empirical correlation of heat transfer data of some of the above mentioned experimenters utilizing void volumes and tube pitch ratios in an attempt to develop a general correlation for porous solid heat transfer. It should be noted that, although the author does not indicate it as such, the only tube bank data used is that for bare tubes.

b. Pressure drop.

The first significant work in this area was presented by Chilton and Genereaux⁽¹⁶⁾ in 1933 as a critical comparison of proposed methods of correlation in the light of the then existing data. They proposed separate correlation parameters for friction factor in the viscous and turbulent region and a distinction was made between

triangular and rectangular arrays for turbulent flow. These correlations were based on very meager experimental data.

Grimison⁽⁴⁾ has correlated the data of Pierson⁽²⁾ and Hoge⁽³⁾ by expressing, on convenient carpet plots, a friction factor, f , defined by;

$$f_1 = \frac{C_2 \rho \Delta P}{NG_{\max}^2} \quad (5)$$

versus Reynolds number with parameters of transverse and longitudinal pitch to tube diameter ratios. The range of this data is $2000 < Re \text{ (using } D_t) < 40,000$.

Jakob⁽¹⁷⁾ has developed analytical expressions for the friction factor given by Grimison's graphical presentation. These are given as follows:

(a) Triangular tube array

$$f_1 = (Re)^{-0.16} \left(0.25 + \frac{0.1175}{(a-1)^{1.08}} \right) \quad (6)$$

(b) Rectangular tube array

$$f_1 = (Re)^{-0.15} \left(0.044 + \frac{0.086}{(a-1)^{0.43} + (1.13/b)} \right) \quad (7)$$

where $Re = D_t G_{\max} / \mu$

$a = S_T / D_t$

$b = S_L / D_t$

Gunter-Shaw⁽¹⁸⁾ suggest plotting a special friction factor,

f_2 , defined by,

$$f_2 = \frac{2g_c \rho D_V \Delta P}{G_{\max}^2 L} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} \left(\frac{D_V}{S_T} \right)^{-0.4} \left(\frac{S_L}{S_T} \right)^{-0.6} \quad (8a)$$

against the volumetric Reynolds number (Eq. 2). By this method of plotting, they claim that a single curve is obtained covering the results for both triangular and rectangular arrays, viscous and turbulent flow and tube banks comprised of either plain or extended surfaces. Their correlation is as follows:

(a) Viscous region ($Re_V < \sim 200$)

$$f_2 = \frac{180}{Re_V} \quad (8b)$$

(b) Turbulent region ($Re_V > \sim 200$)

$$f_2 = \frac{1.92}{(Re_V)^{0.145}} \quad (8c)$$

The most comprehensive and critical analysis of all available pressure drop data (at that time) was made by Boucher and Lapple⁽¹⁹⁾. Their conclusions can be summarized as follows:

1. Incorporation of allowances for tube-bank configuration in a single friction factor-Reynolds number relationship is fundamentally unsound.
2. Correlations should be based on isothermal and isobaric data.
3. No correlation can be expected to represent the data of all investigators to better than $\pm 30\%$.
4. Data for single rows can be related to the drag coefficient for single cylinders.
5. Available data for turbulent flow, when compared by the Chilton-Genereaux⁽¹⁶⁾ and/or Gunter-Shaw⁽¹⁸⁾ methods of representation, show a three-to-sixfold spread and in many cases are unreliable in predicting experimental results. (Note: Omohundro, et al.,

came to a similar conclusion).

6. The graphical correlation of Grimson⁽⁴⁾ and the resulting empirical expressions of Jakob⁽¹⁷⁾ appear to be the best available method for a general representation of data for both triangular and rectangular array over a Reynolds number range of 2000 to 40,000.

Very little viscous flow data were available at the time of the Boucher-Lapelle report. Since then Bergelin, et al.,⁽⁷⁾⁽⁸⁾⁽⁹⁾⁽¹⁰⁾ have reported extensive data for this and the transition region. They use a modification of the Chilton-Genereaux method utilizing the volumetric equivalent diameter (Eq. 2) to obtain the best correlation. They also find that a variable exponent on the ratio μ_b/μ_s is suitable to correlate the nonisothermal pressure drop data.

Kays, et al.,⁽¹¹⁾⁽¹²⁾ define the friction factor f_3 as:

$$f_3 = \frac{C \Delta P D_e \rho}{L G_{\max}^2} \quad (9)$$

$$\text{where } D_e = \frac{4 A_c L}{A} = \quad (10)$$

A_c = minimum free flow area

A = heat transfer area

L = flow length of the bank.

They obtain a correlation for this friction factor for $500 < Re$ (using D_t) $< 20,000$ and triangular tube pitch as follows;

$$f_3 = C_f Re^{-0.18} \quad (11)$$

where $Re = \frac{D_e G}{\mu}$ (see Eq. 10). C_f is found from Fig. 13 of reference (11).

It should be noted that all of the above references for bare

tubes apply to only pure crossflow and cannot be used directly in the design of standard baffled shell and tube heat exchangers. Information of this type can be found in references (20), (21), (22), (23), (24), (27) and (34).

2. Circular finned tubes

This literature search was specifically concerned with the heat transfer and frictional characteristics of only circular finned tubes. For this reason, all literature referred to in this section is primarily concerned with this type of configuration. For those interested in other fin configurations references (12), (26), (28) and (31) might be useful.

a. Heat transfer

The first important paper on the heat transfer characteristics of circular finned tubes was published in 1945 by Jameson⁽²⁹⁾. He correlates his data for several tube and fin configurations by plotting the heat transfer factor, j , (Eq. 3) versus a Reynolds number calculated with a hydraulic diameter defined as;

$$d = \frac{2A}{\pi B}$$

where A is the air-side tube and fin surface per linear ft and B is the projected tube perimeter in ft.

Katz, Beatty and Foust⁽³⁰⁾ present data for a group of tubes with integral spiral fins of various diameters tested with several fluids. Since they correlate their data utilizing the overall heat transfer coefficient, U , the information presented is not useful for general design purposes.

Tate and Cartinhour⁽³¹⁾ report heat transfer data for circular fins of several cross sections. Unfortunately the data is meager and very scattered (j versus Re [using D_t]) and, therefore, of limited utility.

The numerous publications of Kays and London have been republished in their excellent text on extended surface heat transfer and pressure drop⁽¹²⁾. They choose to correlate their heat transfer data by plotting j versus Re (Eq. 3) for the individual tube and fin arrangements. Although no general geometric correlation is attempted the data reported is extensive and is useful for a design where the configuration of their test can be duplicated.

A paper by Schmidt⁽³²⁾ presents an equation of the outside heat-transfer coefficient for banks of finned tubes in terms of the coefficient for bare tubes, the fin height and the number of fins per in. This approach seems excessively involved since it requires the solution of two separate empirical equations each of which introduces some uncertainty.

The first attempt to develop a general correlation of available heat-transfer and pressure drop data for gas flow normal to banks of circular finned tubes was presented by Katz, et al.⁽³³⁾. An extensive amount of work by the group led to the following empirical correlation:

$$Nu = 3.34(Re_K)^{0.551}(Pr)^{1/3} \quad (12)$$

$$\text{where } Re_K = \frac{D_K G}{\mu} \quad (13)$$

$$\text{and } D_K = C_4 n D_{fin} \left(\frac{S_L + S_T}{2D_t} \right)^2$$

The most recent work in this field is that of Ward and Young⁽²⁵⁾ on seven fin banks of triangular array. They also attempted to develop a

general correlation for both low and high fins and for various tube pitches and diameters. The following correlation was obtained by means of a regression analysis program on an IBM 650 digital computer.

$$Nu = 0.364(Re_W)^{0.68}(Pr)^{1/3} \left(\frac{D_{fin}}{D_t} \right)^{0.45} \left(\frac{Y}{D_{fin}} \right)^{0.30} \quad (14)$$

where Re and Nu are calculated with D_t and Y = fin thickness.

While this equation is shown to correlate not only Ward and Young's data (+20% - 10%), but also Jameson's⁽²⁹⁾ and Katz, et al.⁽³⁾, one cannot help noticing that it contains no tube pitch parameters and thus infers that the heat transfer is independent of tube arrangement. This seems unlikely.

b. Pressure drop

Jameson⁽²⁹⁾ correlates pressure drop by plotting friction factor, f_1 (Eq. 5) versus a Reynolds number calculated with a highly complex empirical hydraulic diameter, D_J . His results are:

$$f_1 = 1.53 \left(\frac{D_J G}{\mu} \right)^{-0.25} \quad (15)$$

The Gunter-Shaw⁽¹⁸⁾ equation (Eq. 8), according to its developers, will also correlate finned tube data. This has been challenged, and rightly so, I believe, by Boucher and Lapple⁽¹⁹⁾. Ward and Young⁽²⁵⁾ have confirmed Boucher and Lapple's comments by demonstrating that Eq. 8c does not correlate their data satisfactorily. These authors suggest the following expression as the pressure drop across banks of circular finned tubes:

$$\Delta P = \frac{C G_{max}^2 L}{D_t \rho} \left(\frac{D_{t max} G}{\mu} \right)^{-0.264} \left(\frac{Y}{D_{fin}} \right)^{-0.0377} \left(\frac{S}{D_t} \right)^{0.396} \quad (16)$$

Kays and London⁽¹²⁾ present their frictional data in plots of f_3 (Eq. 9) versus

Re calculated with D_e of Eq. 10. As with their heat transfer data no general correlation is developed but extensive data is available for design of similar tube bank configurations.

Schmidt⁽³²⁾ defines his friction factor as;

$$f_4 = C_6 \Delta P V^2 \rho \quad (17)$$

The value of f is found by an expression involving five constants which vary with tube bank geometry. The correlation seems quite involved and is probably no more accurate than the graphical method of Kays and London⁽¹²⁾.

A friction factor, f_5 , is defined by Katz, et al.⁽³²⁾ as follows:

$$f_5 = \frac{C_7 \Delta P D_K \rho}{NG_{max}^2} \quad (18)$$

where D_K is given by Eq. 13. These authors correlated f_5 in terms of Re_K (Eq. 13) and a parameter K defined as;

$$K = \frac{n D_{fin}}{D_V^{0.2}}$$

The result is a plot of f_5 versus Re_K with parameters of K .

Water Flow (Tube Side)

1. Subcooled water

a. Heat transfer

Numerous authors have verified the Colburn equation is applicable to high temperature-high pressure systems^{(35),(36),(37)}. This equation is:

$$Nu_f = 0.023 (Re)_f^{0.8} (Pr)_f^{1/3} \quad (19)$$

Buchberg, et al.,⁽³⁸⁾ have established that Eq. 19 can be applied over the ranges

$$50,000 < Re < 600,000$$

$$200 < T_m < 650^\circ\text{F}$$

$$100 < P < 2500 \text{ psi}$$

Other experimenters^{(39),(40),(41)} prefer the Seider-Tate⁽⁶⁾ expression which uses a viscosity correction ratio. This equation;

$$Nu = 0.027 Re^{0.8} Pr^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14}, \quad (20)$$

is not significantly different than the Colburn equation (Eq. 19) and the choice between the two is somewhat arbitrary.

b. Pressure drop

The pressure drop of subcooled water flowing in pipes at high temperature and pressure is calculated by the standard methods that have been developed for the determination of pipe friction for liquids at ordinary temperatures and pressures^{(42),(43)}.

2. Boiling Liquid

a. Heat transfer

It is fortunate that the interested engineer or scientist can find a single comprehensive source of information in a field obscured by years of experimentation and hundreds of diverging opinions. In the field of boiling heat transfer such a presentation is given by Westwater^{(44),(45)}. This treatise is a clearly written, complete review of the development of mechanism and theory of boiling. The only major area not fully discussed by Westwater is that of boiling in conduits under simultaneously-forced convection.

For the reasons mentioned above, the only references discussed in this section will be those of boiling with forced convection, since this is

of particular interest to the steam generator designer, and also those articles of general interest published since Westwater's work.

In particular the work of Ellion⁽⁴⁶⁾ should also be referred to by those interested in the basic mechanisms of boiling heat transfer.

This author chooses to divide the studies of forced circulation boiling into two groups: Boiling in which there is no net production of vapor and boiling in which there is net production of vapor. The division point between these types of boiling is, admittedly, nebulous and the classification is used only since the available literature on forced circulation boiling appears to fall into either one or the other of the groupings. One excellent paper by Boarts, et al.⁽⁴⁷⁾ does cover the region which could be described as being between the above groups.

Boiling, during forced convection, in which there is no net production of vapor, occurs when a vapor bubble forms and grows at a solid wall where the temperature is above saturated conditions and either collapses when it grows into an area where the bulk fluid temperature is below saturated conditions (subcooled) or collapses when it is swept from the wall into a region of lower temperature. This phenomena is called "surface" or "local" boiling, and it is discussed briefly by Westwater⁽⁴⁵⁾. It affects very high heat transfer rates and has been the subject of much experimental investigation.

Kreith and Summerfield⁽³⁷⁾ have made a study of surface boiling of water with forced convection and have attempted to correlate the heat transfer and pressure drop by use of an empirical viscosity correction factor.

In an investigation of surface boiling of water in annuli of different materials, Guibert and Romie⁽⁴⁸⁾ determined that the effects of the

different metals, if any, were marked by other influences and that the boiling heat transfer was a function of flow velocity.

Clark and Rohsenow⁽⁴⁹⁾, studying the surface boiling of degassed, distilled water flowing in a vertical 3/16 in. tube, state that forced-convection boiling data in the range of pressures and velocities investigated is correlated by the following equation:

$$\frac{C_{pl}(T_s - T_{sat})}{\lambda} = 0.006 \left[\frac{(Q/A)_{boil}}{\mu\lambda} \sqrt{\frac{g_o \sigma}{g(\rho_l - \rho_v)}} \right]^{0.33} (Pr)_l^{1.7} \quad (21)$$

A very recent paper by Bernath and Begell⁽⁵⁰⁾, resulting from extensive investigations of surface boiling in narrow aluminum annuli, states that the mechanisms involved are much more complicated than previously postulated. Formulae are presented, based on an empirical fit of the data, that permit prediction of $(T_w - T_{sat})$ under all conditions in the range of the variables studied.

The heat transfer mechanism for boiling in which there is a net production of vapor is not the same as in the case of surface or local boiling. This is understandable in light of the obvious differences in the velocity and temperature distributions between the two conditions. Despite the fact that considerably less work has been done in this area, (particularly for high vapor fractions), than in the study of boiling in general, there are a few experimental and analytical papers available for both nucleate and film boiling* under forced convection.

Mumm⁽³⁶⁾, studying the heat transfer to boiling water in forced convection in a uniformly heated stainless steel tube (1/2 in. I.D.) established

* See Westwater⁽⁴⁴⁾ for a discussion of nucleate and film boiling.

the following empirical equation by using the techniques of dimensional analysis.

$$Nu = \left[4.3 + 5.0 \times 10^{-4} \left(\frac{v_{\ell v}}{N \ell} \right)^{1.64} (x) \right] \left[\frac{(Q/A)_{\text{boil}}}{G \lambda} \right]^{0.464} \left[Re_f \right]^{0.808} \quad (22)$$

Mumm gives the range of validity of this equation as follows:

$$70 < G < 280 \text{ lb/sec-ft}^2$$

$$45 < P < 200 \text{ psia}$$

$$50,000 < Q/A < 250,000 \text{ Btu/hr-ft}^2$$

$$0 < x < 0.6$$

This equation can only be used for design purposes through an iteration process since the calculation of the film coefficient requires knowledge of the heat flux Q/A .

The Russians have done some interesting experimental and theoretical studies of forced convection boiling^{(51),(52),(53)}. Although the analytical and experimental techniques seem good the reports and/or translation are not organized well and this with the unusual nomenclature, results in developments which are difficult to follow.

Dengler and Addoms⁽⁵⁴⁾ have studied the heat transfer mechanism for vaporization of water in a vertical tube. Their experimental measurements give T_w , T_b , x and Q/A as functions of vertical distance along the evaporator. After subtracting out the natural convection effects, they correlate these heat transfer data as:

$$\frac{h}{h_L} = F \frac{3.5}{x_\mu^{0.5}} \quad (23)$$

where $\frac{h}{h_L}$ is the ratio of the observed two-phase heat transfer coefficient to the predicted all-liquid coefficient.

$$\frac{1}{x_{\mu}} = \left(\frac{x}{1-x} \right)^{0.9} \left(\frac{N_g}{N_l} \right)^{0.5} \left(\frac{\mu_g}{\mu_l} \right)^{0.1} \quad (24)$$

$$F = 0.67 \left[\left\{ (T_s - T_b) - (T_s - T_b)_{init} \right\} \left(\frac{\partial P}{\partial T_{sat}} \times \frac{D}{\sigma} \right)_{T_s} \right]^{0.1} \quad (25)$$

These authors also present some data in the high vapor content range which indicate a sudden decrease in the film coefficient. Clearly, Eq. 23 cannot apply there since $1/x_{\mu} \rightarrow \infty$ as $x \rightarrow 1.0$.

Rankin and Gerster⁽⁵⁵⁾ have reported heat transfer coefficients for forced convection film boiling under conditions of high temperature difference (from 80 to 320°F). These studies involved flow of Freon-113 or methanol in 1 in. (I.D.) vertical pipe. A theoretical equation for the prediction of heat transfer coefficients was derived assuming the presence of a continuous smooth core of liquid surrounded by an annular film of vapor flowing upward in laminar motion under the influence of buoyant force. The experimental results were twice as large as those predicted by the equation. The stability of the vapor film was also investigated.

Probably the best discussion of boiling heat transfer under forced convection occurs in a recent paper by Forster and Greif⁽⁴⁰⁾. The existing theories of heat transfer mechanism for boiling with convection are presented and discussed. With simple yet convincing argument, Forster and Greif dismiss, in turn, these theories in favor of a "vapor-liquid exchange mechanism". The postulated theory is defended by demonstrating that its application can explain the experimental observations of boiling heat transfer.

By the application of the techniques of dimensional analysis coupled with experimental data, Forster and Greif develop correlations for

nucleate boiling which give the correct dependence of heat flux on superheat and system pressure. One of these correlations is as follows:

$$Q/A = \frac{C_8 \alpha C_p \rho_l T_{sat}}{\sigma^{1/2} (\lambda \rho_v)^{3/2}} (C_p T_{sat} \sqrt{\alpha})^{1/4} \left(\frac{\rho_l}{\mu}\right)^{5/8} (Pr)^{1/3} \Delta P^{2*} \quad (26)$$

where ΔP^* = pressure difference corresponding to the superheat, $(T_s - T_{sat})$.

The following three references are to recent publications in the fields of boiling or bubble formation which may be of interest to the reader.

In their paper Zuber and Tribus⁽⁵⁶⁾ present a clear review of nucleate boiling and an analytical expression is developed for the prediction of the maximum nucleate heat flux in pool boiling of saturated or subcooled liquids. A scholarly analysis of stability and burnout problems is also presented and the extension of this analysis to flow systems is briefly discussed.

A generalized correlation to describe pool surface boiling of liquids is derived in a recent paper of Levy⁽⁵⁷⁾. The expression which "correlates all fluid independent of pressure and heating surface fluid" is:

$$Q/A = \frac{k_l C_p \rho_l^2}{\sigma^{T_{sat}} (\rho_l - \rho_v) B_L} (T_s - T_{sat})^3 \quad (27)$$

where B_L is an empirically determined coefficient determined as a function of $(\rho_v \lambda)$.

Two excellent bibliographies on boiling heat transfer, bubble formation and other related subjects have been prepared by Hawkins⁽⁵⁹⁾ and Bloomfield, McElroy and Skinner⁽⁵⁸⁾. With these sources and with the bibliographies of Westwater^{(44), (45)}, and Zuber and Tribus⁽⁵⁶⁾ the reader will have available references to an extensive number of unclassified publications in this field.

The question as to which of the methods outlined above should be used by the designer to obtain the heat transfer characteristics in a boiling, forced convection system is still unanswered. Several approaches have been taken; each leading to a different expression which, as yet, is relatively unverified. These conditions indicate the need for continued investigation of forced convection boiling heat transfer. For the present, the designer is advised to determine from his particular problem the type of boiling that will probably be involved (nucleate, film, etc.,) and with the system pressure select the correlation which resulted from the most similar experimental conditions.

b. Pressure drop

The simultaneous flow of vapor and liquid in tubes has been widely studied under the title of two-phase flow in gas-liquid systems. A recent comprehensive literature survey on this subject has been prepared by Bennett⁽⁶⁰⁾. This work classifies the available publications into the groups; two phase-two component (eg., air - water) and two phase-one component (eg., steam-water).

For those interested in two phase-two component flow a recent paper by Petrick⁽⁶¹⁾ presents some new data on the subject.

In general though the designer will require information on two phase-one component flow with vaporization. Martinelli and Nelson⁽⁶²⁾ present an analysis for the prediction of pressure drop during forced-circulation boiling of water. By extrapolation of past work on co-current, isothermal flow of air-water and air-oil mixtures, these authors obtained curves of local ratio of the pressure gradient due to two-phase friction to the pressure gradient due only to liquid friction for various qualities of mixture. A graphical integration then permitted them to predict the pressure drop from entrance to

exit of a horizontal uniformly heated tube for any calculated exit quality and system pressure. Their analysis was checked against experimental data and found to be accurate to about $\pm 30\%$.

Jakob, Leppert and Reynolds⁽⁶³⁾ have developed an empirical correlation from experimental results for pressure drop during forced-circulation boiling of distilled water in an electrically-heated horizontal tube. The experimental variables had the following approximate ranges.

$$\begin{aligned} 110 < G < 360 \text{ lb/sec-ft}^2 \\ 30 < P < 200 \text{ psia} \\ 0 < Q/A < 350,000 \text{ Btu/hr-ft}^2 \\ 0 < x < 0.4 \end{aligned}$$

Their final equation is

$$\frac{\Delta P_{TP}}{\Delta P_L} = \frac{0.795(10)^5 x_F^{1.93 - 0.37 \log_{10} P}}{P^{1.04} (2.93 - 0.37 \log_{10} P)} + 6 \quad (28)$$

where P = static pressure psf. The authors compare their empirical equation and experimental data with the analysis of Martinelli and Nelson⁽⁶²⁾ and find a reasonable agreement between the two particularly at the higher pressures.

A recent English paper by Lester⁽⁶⁴⁾ gives more data for two-phase, one component flow without vaporization. The writers correlate their pressure drop data for water-steam flow in large diameter pipes (4 and 6 in.) by several methods proposed by previous experimenters. Comments on the best correlation are included.

3. Superheated Steam

a. Heat transfer

To this author's knowledge, only one major paper is available on the heat transfer to superheated steam at high pressure. This is the work of McAdams, Kennel and Addoms⁽⁶⁵⁾. An empirical correlation of experimental data leads to an expression for the prediction of heat transfer coefficients at pressures of 115 to 3500 psi and temperature from 430 to 1000°F. Their final expression is:

$$Nu = 0.0214(1 + \frac{2.3}{L/D})(Re)_f^{0.8}(Pr)_f^{1/3} \quad (29)$$

b. Pressure drop

For many years it has been standard practice among boiler manufacturers to calculate the pressure drop of steam in superheaters by the method of incompressible fluid flow in pipes^{(42),(43)}. For standard boiler design this is surely adequate in any system where the linear velocity is less than ~300 ft/sec and/or the superheater pressure drop is less than approximately 20% of the absolute pressure.

If, under unusual circumstances of a particular design, the above limits are significantly surpassed compressible effects must be taken into account. Herbert and Knowles⁽⁶⁶⁾ present a theoretical and experimental study for the friction coefficients for the compressible flow of steam in which the interested reader may find design information. References for other publications in this area are also presented.

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