

STEADY STATE BEHAVIOUR OF SINGLE-PHASE AND TWO-PHASE NATURAL CIRCULATION LOOPS

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Abstract

Extensive experimental investigations have been carried out on the steady state performance of both single-phase and two-phase loops. The steady state flow rate in single- and two-phase loops can be expressed by a generalized correlation in terms of a single dimensionless parameter. The experimental data from both single-phase and two-phase loops are found to be in reasonable agreement with this correlation. The generalized steady flow correlation has also been compared with code predictions.

1. Introduction

Natural circulation loops are extensively used in several industries. Both single-phase and two-phase natural circulation systems are important for nuclear industry. Single-phase NC is used for decay heat removal in PWRs, VVERs and PHWRs during upset conditions like pumping power failure. It is also used in district heating reactors and certain low power reactors like CAREM as the normal mode of core cooling. Compared to single-phase systems, two-phase systems are capable of generating larger buoyancy force and hence larger flow rates. Typical industrial applications of two-phase systems are Natural Circulation Boiling Water Reactors (NCBWRs), Natural Circulation Boilers (NCBs) in fossil fuelled power plants, Natural Circulation Steam Generators (NCSG) in PWRs & PHWRs and thermosyphon reboilers in chemical process industries. Use of natural circulation has been demonstrated in small scale (maximum power about 65 MWe) boiling water reactors like Humbholt Bay 3, Dodewaard and VK-50. Currently several natural circulation BWRs are under development such as ESBWR, VK-300 and AHWR.

The primary function of a natural circulation loop is to transport heat from a source to a sink. The heat transport capability of natural circulation loops is directly proportional to the flow rate it can generate. Hence reliable prediction of the flow rate is essential for the design and performance evaluation of natural circulation loops. Generally the prediction methods are available in dimensional form. For two-phase natural circulation loops, however, even explicit dimensional correlations for steady state flow are not readily available. For comparison of the steady state performance of different loops, and to extrapolate the results from small scale to prototype plants, dimensionless correlations are preferable. Generally accepted dimensionless groups are not readily available for two-phase natural circulation loops. With many of the reported nondimensionalization procedures, the resulting dimensionless groups are too many and it is not possible to explicitly obtain the flow rate as a function of the different dimensionless groups. The authors were successful in explicitly expressing the flow rate in single-phase loops as a function of a single dimensionless group. However, for two-phase flow, no such nondimensional groups were reported.

In the present paper, we present an exact analytical expression for the two-phase flow rate based on the same methodology followed for single-phase loops. Then the nondimensionalization procedure followed for single-phase loops is extended to two-phase loops to obtain an explicit correlation for the flow rate as the function of a single dimensionless parameter. Subsequently, experiments were conducted in both single-phase and two-phase loops. In addition experimental data was also compiled from literature for single-phase and two-phase loops. The experimental data was used to compare the theoretical correlations for flow rate. Both in-house and literature data of single-phase and two-phase natural circulation systems were found to match well with the correlations. The paper presents the details of the nondimensional correlations derived, the experimental data used for the assessment of the correlations and the results obtained.

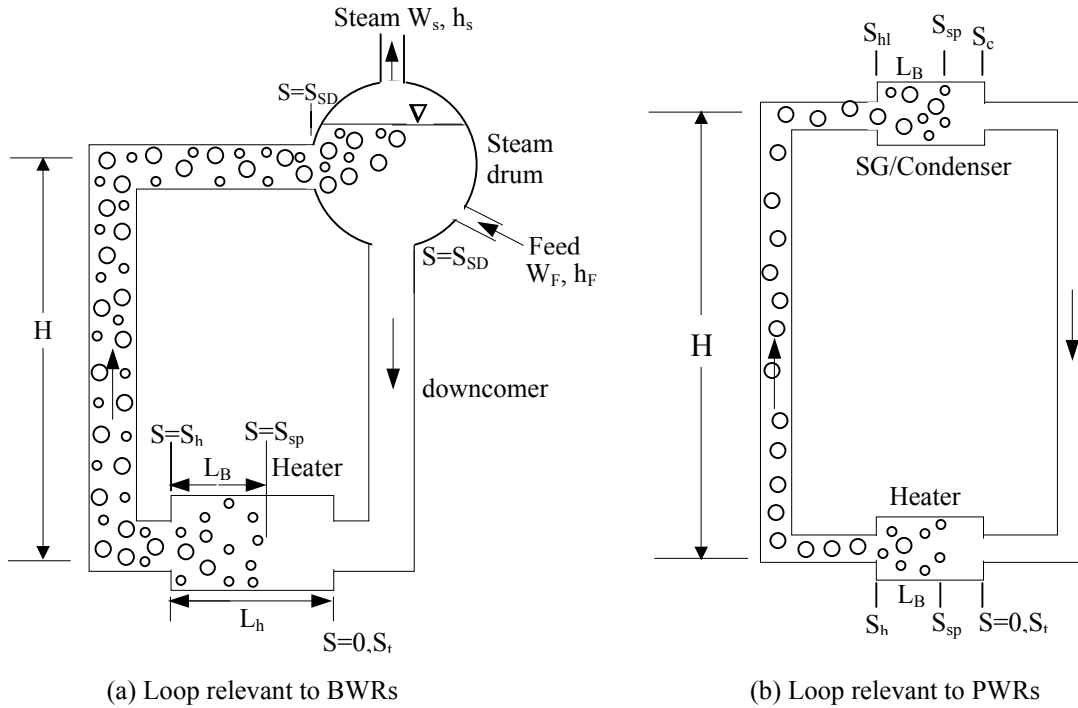


Fig. 1: Two-phase NCLs relevant to nuclear reactors

2. Theoretical Development

The theoretical development described below is applicable for both single-phase and two-phase natural circulation systems and is based on the homogeneous equilibrium model. The geometry and coordinate system considered for the theoretical analysis is described in Fig. 1a and it is shown later that the theory can be extended to the geometry and coordinate system shown in Fig. 1b. In order to account for phase change, it is convenient to work with the fluid enthalpy rather than fluid temperature in the governing equations. In addition, the following assumptions are made in the theoretical development:

- 1) Heat losses in the piping are negligible
- 2) The pressure losses in the system are negligible compared to the static pressure so that the fluid property variation with pressure is negligible. For most natural circulation systems, this is a reasonable assumption considering that the loop driving head is only a few meters of water column.
- 3) The inlet subcooling is negligible so that the density variation in the single-phase heated section can be neglected. This assumption is reasonable for the normal operating condition of BWRs where the inlet subcooling is low. With these assumptions all fluid properties can be assumed constant at the saturation value.

- 4) Complete separation of steam and water is assumed to occur in the steam drum so that there is no liquid carryover with the steam and no vapor carry-under with water (Fig. 1a).
- 5) A constant level is maintained in the SD, so that the single-phase lines always run full (Fig. 1a).
- 6) The heater is supplied with a uniform heat flux and the SD can be approximated to a point heat sink.

With the above assumptions, the steady state one-dimensional governing equations for the two-phase NCS of Fig. 1a can be written as follows:

$$\frac{dW}{ds} = 0 \quad \text{Conservation of mass} \quad (1)$$

$$\frac{W}{A_i} \frac{dh}{ds} = \begin{cases} \frac{4q_h''}{D_h} & \text{heater} \\ 0 & \text{pipes} \end{cases} \quad \text{Conservation of energy} \quad (2)$$

$$\frac{W^2}{A_i^2} \frac{d}{ds} \left(\frac{1}{\rho} \right) = -\frac{dP}{ds} - \rho g \sin \theta - \frac{f W^2}{2D_i \rho A_i^2} - \frac{K W^2}{2\rho A_i^2 L_i} \quad \text{Conservation of momentum} \quad (3)$$

Noting that $v = 1/\rho$ and integrating the momentum equation around the circulation loop

$$W^2 \sum_{i=1}^{N_t} \frac{1}{A_i^2} \oint dv = -\oint dP - g \oint \rho dz - \frac{W^2}{2\rho} \sum_{i=1}^{N_t} \left(\frac{fL}{DA^2} + \frac{K}{A^2} \right)_i \quad (4)$$

Noting that $\oint dv = \oint dP = 0$ for a closed loop, we can write

$$-g \oint \rho dz = \frac{W^2}{2\rho} \sum_{i=1}^{N_t} \left(\frac{fL}{DA^2} + \frac{K}{A^2} \right)_i \quad (5)$$

The density is assumed to vary as $\rho = \rho_r [1 - \beta_h (h - h_r)]$ in the buoyancy force term where $\beta_h = (1/v)(\partial v / \partial h)_p$. It may be noted that this is equivalent to the Bossinesq approximation in single-phase systems. For the estimation of frictional pressure loss, ρ_l is used in the single-phase regions. For the calculation of the frictional pressure loss in the heated two-phase and the adiabatic two-phase sections the two-phase friction factor multiplier, ϕ_{LO}^2 , is used. Hence equation (5) can be rewritten as

$$g \rho_r \oint \beta_h (h - h_r) dz = \frac{W^2}{2\rho_l} \left\{ \sum_{i=1}^{N_s} \left(\frac{fL_{eff}}{DA^2} \right)_i + \frac{1}{L_B} \sum_{i=N_s+1}^{N_B} \left(\frac{fL_{eff}}{DA^2} \right) \int_{L_s}^{L_B} \phi_{LO}^2(s) ds + \phi_{LO}^2 \sum_{i=N_B+1}^{N_{tp}} \left(\frac{fL_{eff}}{DA^2} \right)_i \right\} \quad (6)$$

where L_B is the boiling length of the heated section and N_s , N_B and N_{tp} are the number of pipe segments in the single-phase, boiling and the unheated two-phase sections respectively. In Eq. (6),

the local loss coefficients have been replaced by an equivalent length such that $(L_{eq})_i = K_i D_i / f_i$ and $(L_{eff})_i = L_i + (L_{eq})_i$. While estimating $(L_{eq})_i$, it is recognized that appropriate multiplier has to be used to account for the enhancement of K_i in case of two-phase flow. In the development presented here, the same multiplier model has been used for friction and local loss coefficients. Further, a closed loop may contain different geometric sections obeying different friction law. If the friction coefficient over the entire loop can be expressed by a correlation of the form $f_i = p / \text{Re}_i^b$ then Eq. (6) can be rewritten as

$$g \rho_r \oint \beta_h (h - h_r) dz = \frac{p \mu_l^b W^{2-b}}{2 \rho_l} \left\{ \sum_{i=1}^{N_s} \left(\frac{L_{eff}}{D^{1+b} A^{2-b}} \right)_i + \frac{1}{L_B} \int_{L_s}^{L_B} \phi_{LO}^2(s) ds \sum_{i=N_s+1}^{N_B} \left(\frac{L_{eff}}{D^{1+b} A^{2-b}} \right)_i \right. \\ \left. + \phi_{LO}^2 \sum_{i=N_B+1}^{N_p} \left(\frac{L_{eff}}{D^{1+b} A^{2-b}} \right)_i \right\} \quad (7)$$

Where μ_l and ρ_l have been used as reference values for the calculation of pressure losses both in single-phase and two-phase regions of the loop. In the absence of appropriate relationships for β_h and ϕ_{LO}^2 , the integrals in the above equation can be obtained numerically. It is customary to use an average value of $(\phi_{LO}^2)_i$ for the i^{th} segment instead of numerically integrating $\int \phi_{LO}^2(s) ds$ over the heated section as in Eq. (6).

$$g \rho_r \oint \beta_h (h - h_r) dz = \frac{p \mu_l^b W^{2-b}}{2 \rho_l} \left\{ \sum_{i=1}^{N_s} \left(\frac{L_{eff}}{D^{1+b} A^{2-b}} \right)_i + \sum_{i=N_s+1}^{N_B} \left(\frac{\bar{\phi}_{LO}^2 L_{eff}}{D^{1+b} A^{2-b}} \right)_i + \phi_{LO}^2 \sum_{i=N_B+1}^{N_p} \left(\frac{L_{eff}}{D^{1+b} A^{2-b}} \right)_i \right\} \quad (8)$$

The above equations can be nondimensionalized using the following substitutions:

$$\omega = \frac{W}{W_{ss}}; \mathcal{H} = \frac{h - h_r}{(\Delta h)_{ss}}; Z = \frac{z}{H}; S = \frac{s}{H}; a_i = \frac{A_i}{A_r}; d_i = \frac{D_i}{D_r}; l_i = \frac{L_i}{L_t} \text{ and } (l_{eff})_i = \frac{(L_{eff})_i}{L_t} \quad (9a)$$

The reference values of flow area, hydraulic diameter, density and enthalpy used are respectively given by

$$A_r = \frac{1}{L_t} \sum_{i=1}^{N_t} A_i L_i = \frac{V_t}{L_t}; D_r = \frac{1}{L_t} \sum_{i=1}^{N_t} D_i L_i; \rho_r = \rho_{in}; \text{ and } h_r = h_{in} \quad (9b)$$

At steady state $\omega_{ss} = 1$, $Q_{SD} = Q_h$ (i.e. heat carried away by steam from the steam drum is equal to the power supplied to the heater) and the non-dimensional equations will become

$$\frac{d \mathcal{H}}{dS} = \begin{cases} \phi_h \frac{V_t}{V_h} & \text{heater } (0 < S \leq S_h) \\ 0 & \text{pipes } (S_h < S \leq S_{SD} \text{ and } S_{SD} < S \leq S_t) \end{cases} \quad (10)$$

$$g\rho_r\Delta h_{ss}H\oint\beta_h\mathcal{H}dZ = \frac{pL_t\mu_l^bW_{ss}^{2-b}}{2\rho_lD_r^{1+b}A_r^{2-b}} \left\{ \sum_{i=1}^{N_s} \left(\frac{l_{eff}}{d^{1+b}a^{2-b}} \right)_i + \bar{\phi}_{LO}^{-2} \sum_{i=N_s+1}^{N_B} \left(\frac{l_{eff}}{d^{1+b}a^{2-b}} \right)_i + \phi_{LO}^2 \sum_{i=N_B+1}^{N_p} \left(\frac{l_{eff}}{d^{1+b}a^{2-b}} \right)_i \right\} \quad (11)$$

$$\text{which can be written as } g\rho_r\Delta h_{ss}\beta_I H = \frac{p\mu_l^bW_{ss}^{2-b}N_G}{2\rho_lD_r^bA_r^{2-b}} \quad (12)$$

$$\text{where } \beta_I = \oint\beta_h\mathcal{H}dZ \text{ and} \quad (13)$$

$$N_G = \frac{L_t}{D_r} \left\{ \sum_{i=1}^{N_s} \left(\frac{l_{eff}}{d^{1+b}a^{2-b}} \right)_i + \bar{\phi}_{LO}^{-2} \sum_{i=N_s+1}^{N_B} \left(\frac{l_{eff}}{d^{1+b}a^{2-b}} \right)_i + \phi_{LO}^2 \sum_{i=N_B+1}^{N_p} \left(\frac{l_{eff}}{d^{1+b}a^{2-b}} \right)_i \right\} \quad (14)$$

Noting that $\Delta h_{ss} = Q_h / W_{ss}$ and $\rho_r = \rho_l$ for the saturated inlet condition, Eq. (12) can be rearranged to get the following explicit equation for the mass flow rate

$$W_{ss} = \left[\frac{2g\rho_l^2\beta_I H Q_h D_r^b A_r^{2-b}}{p\mu_l^b N_G} \right]^{\frac{1}{3-b}} \quad (15)$$

Equation (15) can be expressed in the dimensionless form as

$$\text{Re}_{ss} = C \left(\frac{Gr_m}{N_G} \right)^r \quad (16)$$

$$\text{Where } \text{Re}_{ss} = \frac{D_r W_{ss}}{A_r \mu_l}; Gr_m = \frac{D_r^3 \rho_l^2 \beta_I g Q_h H}{A_r \mu_l^3}; C = \left(\frac{2}{p} \right)^r \text{ and } r = \frac{1}{3-b} \quad (17)$$

For laminar flow with $p = 64$ and $b = 1$, the values of the constants $C = 0.1768$ and $r = 0.5$. Similarly, for turbulent flow with $p = 0.316$ and $b = 0.25$, the corresponding values are $C = 1.96$ and $b = 1/2.75$.

2.1 Evaluation of $\bar{\phi}_{LO}^{-2}$ and ϕ_{LO}^2

We had used mean value of the ϕ_{LO}^2 over the heated section without actually providing an expression for their estimation. Since quality variation is linear for the uniformly heated test section, $\bar{\phi}_{LO}^2$ can be evaluated at half the value of the exit quality. From the basic definition of ϕ_{LO}^2 and McAdam’s model for two-phase viscosity, we can obtain the following equations for ϕ_{LO}^2 and $\bar{\phi}_{LO}^{-2}$.

$$\phi_{LO}^2 = \frac{\rho_l}{\rho_e} \left[\frac{1}{1 + x_e \left(\frac{\mu_l}{\mu_g} - 1 \right)} \right]^b \quad \text{and} \quad \bar{\phi}_{LO}^2 = \frac{\rho_l}{\bar{\rho}_m} \left[\frac{1}{1 + \frac{x_e}{2} \left(\frac{\mu_l}{\mu_g} - 1 \right)} \right]^b \quad (18)$$

$$\text{where } \rho_e = \frac{\rho_g \rho_l}{x_e (\rho_l - \rho_g) + \rho_g} \quad \text{and} \quad \bar{\rho}_m = \frac{\rho_g \rho_l}{0.5 x_e (\rho_l - \rho_g) + \rho_g} \quad (19)$$

It may be recognized that there are several other two-phase friction multiplier models in the literature and one could choose any one of this (IAEA-Tecdoc-1203).

3. SPECIAL CASES

The following special cases will illustrate the utility of the generalized flow correlations developed.

3.1 Constant β_h

If we can approximate β_h by a mean value over the whole loop, then

$$\beta_l = \bar{\beta}_h \oint \mathcal{H} dZ = \bar{\beta}_h \mathcal{H}_l \quad \text{where} \quad \mathcal{H}_l = \oint \mathcal{H} dZ \quad (20)$$

Integrating Eq. (10) with appropriate boundary conditions we obtain the following equations for the distribution of enthalpy in the various segments of the loop

$$\mathcal{H} = \mathcal{H}_{in} + \phi_h \frac{V_t}{V_h} S \quad \text{for the heated section } 0 < S \leq S_h \quad (21a)$$

The dimensionless length of the single-phase heated section, S_{sp} , can be obtained as

$$\mathcal{H}_l = \mathcal{H}_{in} + \phi_h \frac{V_t}{V_h} S_{sp} \quad \text{or} \quad S_{sp} = \left(\frac{\mathcal{H}_l - \mathcal{H}_{in}}{\phi_h} \right) \frac{V_h}{V_t} \quad (21b)$$

The boiling length, S_B , is then calculated as $S_B = S_h - S_{sp}$. The enthalpy in the two-phase unheated portion (riser) is obtained by setting $S = S_h$ in Eq. (21a) as

$$\mathcal{H}_e = \mathcal{H}_{in} + 1 \quad \text{for } S_h < S \leq S_{SD} \quad (21c)$$

The steam drum, where complete separation of the steam-water mixture is assumed to take place without any carryover or carryunder can be approximated to a point heat sink where the entire enthalpy of the steam is lost and some amount of inlet subcooling is obtained due to the mixing caused by the subcooled feed water. By a heat balance, the enthalpy at the downcomer inlet can be calculated as

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$$\mathcal{H}_{in} = \mathcal{H}_l - x_e(\mathcal{H}_l - \mathcal{H}_F) \quad \text{for } S_{SD} < S \leq S_r \quad (21d)$$

Using the above in Eq. (21c), we obtain

$$\mathcal{H}_e = \mathcal{H}_l - x_e(\mathcal{H}_l - \mathcal{H}_F) + 1 \quad (21e)$$

If we choose $h_r = h_F$, then (21e) can be rewritten as

$$\mathcal{H}_e = \mathcal{H}_l(1 - x_e) + 1 \quad (21f)$$

Using Eq. (21) in Eq. (20) we obtain $\oint \mathcal{H} dZ = 1$ so that

$$\beta_l = \bar{\beta}_h \quad (22)$$

3.1.1 Evaluation of $\bar{\beta}_h$

While deriving Eq. (20), it was assumed that β_h is a constant. In reality β_h varies as shown in Fig.2 for water. In fact, it is a constant only above a critical quality, which depends on the system pressure. Again, one could numerically integrate Eq. (6) to obtain a more accurate prediction or use an average value calculated from the following equation.

$$\bar{\beta}_h = \frac{1}{v} \left(\frac{\partial v}{\partial h} \right)_p = \frac{1}{v_m} \frac{\Delta v}{\Delta h} = \frac{1}{0.5(v_{in} + v_e)} \left(\frac{v_e - v_{in}}{h_e - h_{in}} \right) = \frac{\rho_{in} - \rho_e}{0.5(\rho_e + \rho_{in})\Delta h} = \frac{\Delta \rho}{\rho_m \Delta h} \quad (23)$$

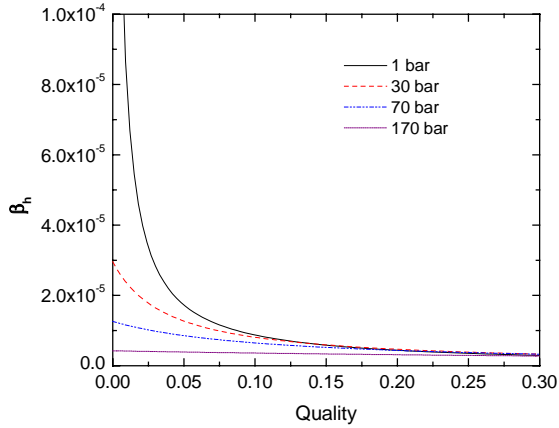


Fig.2: Variation of β_h with pressure

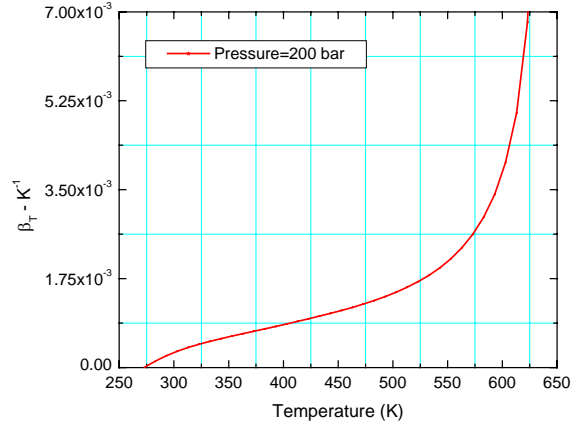


Fig.3: Variation of β_T with temperature

3.2 Single-phase natural circulation systems

For single-phase natural circulation $x_e = 0$ and by definition $\phi_{LO}^2 = 1$. Using this value in Eq. (14), we get

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$$N_G = \frac{L_t}{D_r} \sum_{i=1}^{N_i} \left(\frac{l_{eff}}{d^{1+b} a^{2-b}} \right)_i \quad (24)$$

which is the same as that given for single-phase loops by Vijayan (2002). Similarly, noting that $h=CpT$ and if Cp is a constant, then we get

$$\beta_h = \frac{1}{v} \left(\frac{\partial v}{\partial h} \right)_p = \frac{1}{v Cp} \left(\frac{\partial v}{\partial T} \right)_p. \text{ Noting that } \beta_T = \frac{1}{v} \left(\frac{\partial v}{\partial T} \right)_p \text{ we obtain } \beta_h = \frac{\beta_T}{Cp}. \quad (25)$$

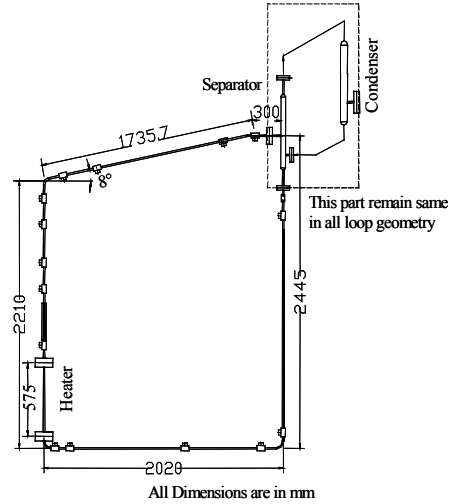
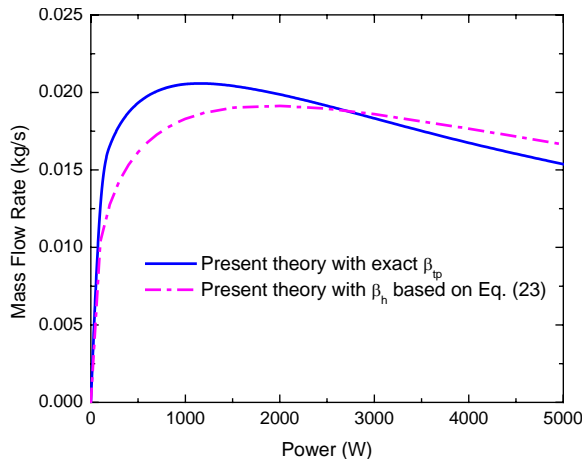
Variation of β_T with temperature for water is shown in Fig. 3. It may be mentioned that both β_h and β_T are volumetric expansion coefficients. Hence to differentiate between the two we denote β_h as enthalpic expansion coefficient and β_T as temperature coefficient of expansion. Using this, the steady state flow is obtained as

$$W_{ss} = \left[\frac{2 g \rho_0^2 \beta_T H Q_h D_r^b A_r^{2-b}}{p \mu_r^b N_G Cp} \right]^{\frac{1}{3-b}} \quad (26)$$

The modified Grashof number for single-phase flow becomes

$$Gr_m = \frac{D_r^3 \rho_r^2 \beta_T g \Delta T_r}{\mu_r^2} \text{ where } \Delta T_r = \frac{Q_h H}{A_r \mu_r Cp} \quad (27)$$

which is same as that reported in Vijayan et al. (2004). With this substitution, equation (16) is applicable to single-phase natural circulation systems.



3.3 Vertical Heater

Equation (13) was derived for two-phase loops with a horizontal heated section as in Fig 1a. However, it can be easily extended to other loop geometries. For vertical heater, the same equation can be used with the loop height replaced by the centre line elevation difference, ΔZ , between the

steam drum and the heater. The error involved in this approximation is not yet evaluated for two-phase loops. However, for single-phase loops, it was shown that the error involved is less than 1% for most practical applications (Vijayan et al. (2004)).

It may be noted that the averaging procedure described by Eq. (23) has made it possible to calculate the flow rates using the simple equation (15). However, this is expected to introduce some error for loops with vertical heater and it becomes essential to validate the averaging procedure for β_h described by Eq. (23). For this, the flow rates calculated using Eq. (15) with β_l estimated by Eq. (22) and Eq. (23) is compared in Fig. 4. In either case, N_G was calculated using Eq. (14). It can be seen that the averaging leads to close results adequate for engineering calculations. The calculations were done for the loop geometry shown in Fig.5.

3.4 Loops Relevant to PWRs

Equation (6) is applicable for the loop shown in Fig. 1a, where the point heat sink assumption is reasonable. In case we have a heat exchanger where condensation (Fig. 1b) is taking place, then the dimensionless energy equation becomes

$$\frac{d\mathcal{H}}{dS} = \begin{cases} \phi_h \frac{V_t}{V_h} & \text{heater } (0 < S \leq S_h) \\ 0 & \text{pipes } (S_h < S \leq S_{hl} \text{ and } S_c < S \leq S_t) \\ -\phi_c \frac{V_t}{V_c} & \text{condenser } (S_{hl} < S \leq S_c) \end{cases} \quad (28)$$

where complete condensation is assumed to take place in the heat exchanger. Since the cooler is also assumed to have uniform heat flux, the solution of the energy equation is the same as given before leading to the same equations for the flow rate. However, the equation for N_G is obtained as

$$N_G = \frac{L_t}{D_r} \left\{ \sum_{i=1}^{N_s} \left(\frac{l_{eff}}{d^{1+b} a^{2-b}} \right)_i + \sum_{i=N_s+1}^{N_B} \left(\frac{\bar{\phi}_{LO}^2 l_{eff}}{d^{1+b} a^{2-b}} \right)_i + \sum_{i=N_B+1}^{N_p} \left(\frac{\phi_{LO}^2 l_{eff}}{d^{1+b} a^{2-b}} \right)_i + \sum_{i=N_p+1}^{N_C} \left(\frac{\bar{\phi}_{LO}^2 l_{eff}}{d^{1+b} a^{2-b}} \right)_i \right\} \quad (29)$$

where the subscripts B and C refer to the boiling and condensing sections respectively.

3.5 Uniform diameter loops

It may be noted that for a uniform diameter loop (UDL), $D_r = D$, $A_r = A$ so that $a_i = d_i = 1$ and N_G reduces to the following equation

$$N_G = \frac{L_t}{D_r} \left[(l_{eff})_{sp} + \bar{\phi}_{LO}^2 (l_{eff})_B + \phi_{LO}^2 (l_{eff})_{ip} + \bar{\phi}_{LO}^2 (l_{eff})_C \right] \quad (30)$$

For the loops without the condensing section, the last term in the above equation can be dropped and if local losses are negligible, $l_{eff} = l$. For single-phase uniform diameter loops $\phi_{LO}^2 = 1$ so that $N_G = (L_{eff}/D)$ and if local losses are negligible then, $N_G = L/D$.

4. TESTING OF THE STEADY STATE CORRELATION

As equation (16) is valid for both single- and two-phase loops, its adequacy is tested against experimental data on steady state natural circulation in the following sections.

4.1 Single-phase NCLs

A large database exists on the steady state behavior of single-phase natural circulation loops.

4.1.1 Database for Uniform Diameter Loops (UDL)

The dimensionless correlation (16) was extensively tested with data from simple uniform diameter loops. Among uniform diameter loops, rectangular loops are experimentally studied most. Typical examples are the investigations by Holman and Boggs (1960), Huang and Zelaya (1988), Misale et al. (1991), Bernier and Baliga (1992), Vijayan et al. (1992), Ho et al. (1997), Nishihara (1997) and Vijayan et al. (2001). Uniform diameter open loops were investigated by Bau-Torrance (1981) and Haware et al. (1983). Creveling et al. (1975) experimented with a uniform diameter toroidal loop. For all the UDL data covered in the present database, the loop diameter was in the range of 6 to 40 mm and the loop height varied from 0.38 to 2.3 m. The total circulation length varied from 1.2 to 7.2 m. The working fluid was mostly water and in one case Freon. The loop pressure was mostly near atmospheric except for the data of Holman and Boggs which was for near critical pressure. Database for all the four orientations of heater and cooler are included.

4.1.2 Database for Nonuniform Diameter Loops (NDLs)

Most practical applications of natural circulation employ non-uniform diameter loops. Common examples are the nuclear reactor loop, solar water heater, etc. Most test facilities simulating nuclear reactor systems also use non-uniform diameter loops. The non-uniform diameter loops experimentally studied can be categorized into two groups depending on the operating pressure as (1) High pressure loops and (2) Low pressure loops. Most studies are conducted in the high-pressure test facilities simulating nuclear reactor loops. Typical examples of such facilities are the SEMISCALE, LOBI, PKL, BETHSY, ROSA, RD-14 and FISBE. Some studies, however, are carried out in low pressure facilities. Examples are the experiments carried out by Zvirin et al. (1981), Jeuck et al. (1981), Hallinan-Viskanta (1986), Vijayan (1988), and John et al. (1991). Most of the available experimental data in a usable form (i.e. full geometrical details are known) are from the low-pressure test facilities. High-pressure test data in a usable form was available only from FISBE. The nonuniform loops considered had pipe segments with the hydraulic diameter varying from 3.6 mm to 97 mm and loop height varying from 1 to 26 m with pressure ranging from near atmospheric to 9 MPa. The total circulation length of the loops considered varied from about 10 to 125 m. All these loops used water as the working fluid.

Both uniform diameter and nonuniform diameter loop data neglecting the local losses are plotted together in Fig. 6. In general, the laminar and turbulent flow data show good agreement with the respective theoretical correlations. For the intermediate values of Gr_m/N_G ($5 \times 10^6 < Gr_m/N_G < 10^{11}$) significant deviation is observed where the flow is neither fully laminar nor fully turbulent. Further, it may be noted that all the reported data have been used without any filtering and a vast majority of data are found to be well within $\pm 40\%$ in spite of discarding the local pressure losses shows that the generalized correlation is reasonable for both UDLs and NDLs.

2nd RCM on the IAEA CRP on ‘Natural circulation phenomena, modeling and reliability of passive systems that utilize natural circulation’ to be held during August 29 to September 02, 2005 at Corvallis, Oregon State University, USA.

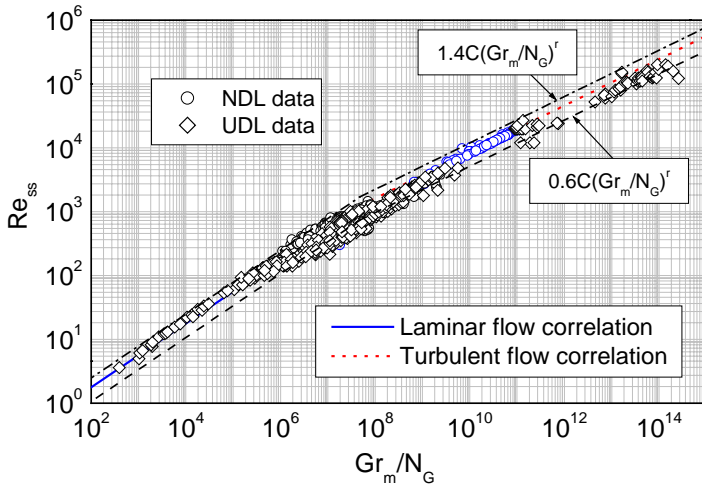


Fig. 6: Single-phase NC data neglecting local losses

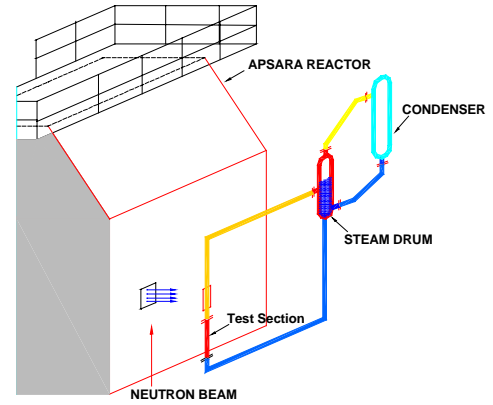


Fig. 7: Test loop as installed in Apsara reactor hall

4.2 Two-phase NCLs

Although, several experimental investigations are reported in literature, in most cases, complete details of the loop geometry were not available. Hence experiments were carried out in different two-phase loops.

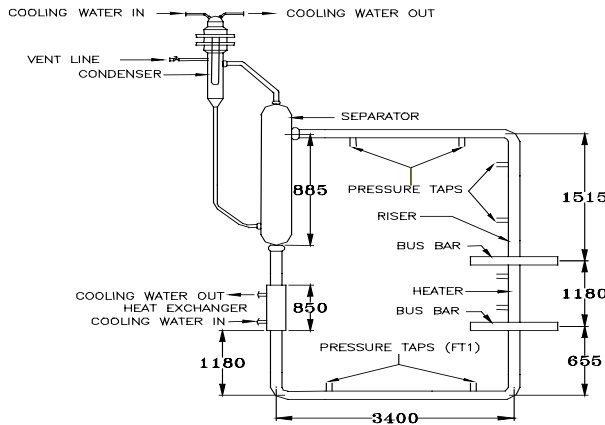


Fig. 8: Two-phase NCL with 49.3 mm diameter

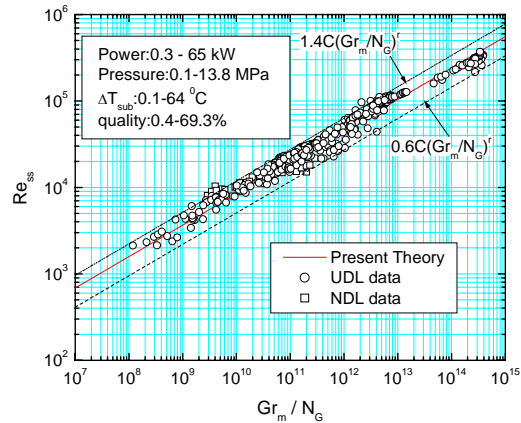


Fig. 9: Testing of correlation with data

4.2.1 Database for uniform diameter two-phase loops

Experiments were conducted in three loops of inside diameter 10.2, 15.74 and 19.86 mm respectively in a facility having the geometry shown in Fig. 4. The steam separator, the condenser and the associated piping (the portion inside the rectangular box in Fig. 4) were the same for all the loops. The steam separator was made of 59 mm inside diameter (2.5” NB Sch 80) pipe. The vertical heater section was directly heated with a high current low voltage power supply. The steam-water mixture produced flows to the separator and the separated steam was condensed and the condensate returns to the steam drum by gravity. The loop was extensively instrumented to measure heater power, fluid temperatures, steam drum pressure, differential pressure at various sections of the single-phase and two-phase pipes, SD level, flow rates (loop, condensate and cooling water flows), void fraction and its distribution. The void fraction was measured using both Neutron Radiography (NRG) and Conductance Probe (CP) techniques. The neutron radiography also helped to visualize the flow patterns. To facilitate neutron radiography, the loop was installed in the Apsara reactor hall in front of the neutron beam hole (Fig.7). During the experiments, the

system pressure was maintained with the help of nitrogen cylinders. Further details of the loop are available in Dubey et al. (2004). In addition, experimental data were generated in a 49.3 mm inside diameter loop shown in Fig. 8. Further details of this facility are available in Kumar et al. (2000). The data generated in these loops fall in the following range of parameter: Loop diameter: 9.6-49.3 mm, Circulation length: 8.7-13.1 m, Pressure: 0.1-7 MPa, quality: 0.4-24% and power: 0.3-40 kW.

4.2.2 Database for nonuniform diameter two-phase loops

Database for nonuniform diameter two-phase loops are reported in several papers. However, complete details of the loop geometry are often not available. In the present case complete details of the loop geometry were available only for the one used by Mendler (1961). For this loop, the data were available for $55 < \text{pressure} \leq 138 \text{ bar}$, $8.2 < \text{quality} \leq 69.3\%$ and $8.3 < \text{power} \leq 65 \text{ kW}$.

4.3 Testing of the Steady State Correlation with Experimental Data

The steady state data from the five different two-phase natural circulation loops are compared with the theoretical correlation for turbulent flow in Fig. 9 as all the data fall in this regime. The experimental data is observed to be close to the theoretical correlation (within an error bound of $\pm 40\%$) for all the loops confirming the validity of the proposed correlations.

4.4 Comparison of the Correlation with Codes

The correlation was compared in Fig. 10 with the predictions of an in-house computer code TINFLO based on homogeneous model over a wide range of pressures (0.1-15 MPa). In addition, the predictions obtained using the homogeneous and the two-fluid models of the RELAP5/MOD3.2 code are also given in Fig. 10. These predictions are for the loop geometry given in Fig. 4 with inside diameter of 10.2 mm. It may be mentioned that the TINFLO code used the same single-phase friction factor model used for developing the correlation and hence the predictions are closer to the generalized correlation. The RELAP5 code uses a different friction factor model and hence the predictions are somewhat different. However, both the homogeneous and the two-fluid models in RELAP5 give practically the same value of flow rate. Typical comparison of the measured flow rates with the predictions of the various codes is given in Fig. 11. RELAP5 code predictions are somewhat lower (5-8%) than the test data whereas predictions of equation (15) and TINFLO are respectively 5-7% and 5-13% higher. The entire steady state data generated for the 10.2 mm loop are compared with the RELAP5/MOD3.2 code in Fig. 12 which shows that the RELAP5 predictions are consistently lower.

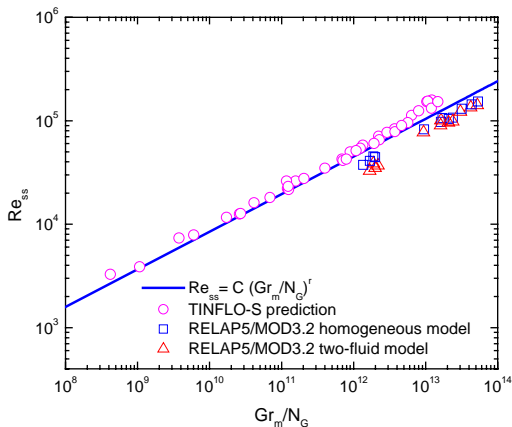


Fig.10: Comparison of code prediction with the correlation

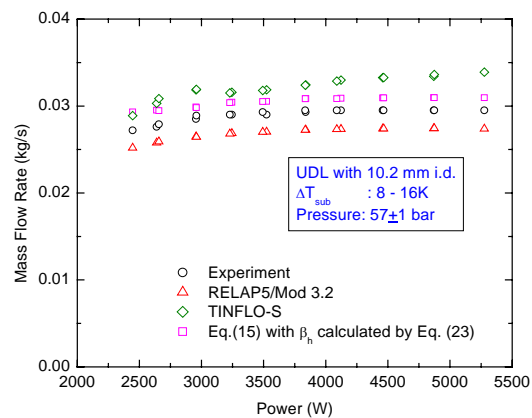


Fig.11: Comparison of code predictions with data

4.5 Flow Regimes in Two-phase loops

One of the drawbacks of dimensionless correlations is that important parametric effects are sometimes masked. The performance of NCLs depends strongly on the pressure and power as shown by Fig.13. In fact three flow regimes can be clearly identified in the figure. These are designated as gravity dominant regime, friction dominant regime and the compensating regime. In a natural circulation loop, the gravitational pressure drop (or the buoyancy pressure differential) is always the largest component of pressure drop and all other pressure drops (friction and local) must balance the buoyancy pressure differential at steady state. However, the natural circulation flow regimes are differentiated based on their change with quality (or power). In the gravity dominant regime, for a small change in quality there is a large change in the void fraction (see Fig. 14) and hence the density and buoyancy force. The increased buoyancy force is to be balanced by a corresponding increase in the frictional force which is possible only at a higher flow rate. As a result, the gravity dominant regime is characterized by an increase in the flow rate with power. At higher qualities and moderate pressures, the increase in void fraction with quality is marginal (Fig.14) leading to almost constant buoyancy force. However, the continued conversion of high density water to low density steam due to increase in power requires that the mixture velocity must increase resulting in an increase in the frictional force and hence a decrease in flow rate. Thus the friction dominant regime is characterized by a decrease in flow rate with increase in power. Between these two, there exists a compensating regime, where the flow rate remains practically unaffected with increase in power. However, the flow regimes depend strongly on the system pressure. In fact, at high pressures only the gravity dominant regime may be observed if the power is low. The friction dominant regime shifts to low pressures with increase in loop diameter (Fig.15).

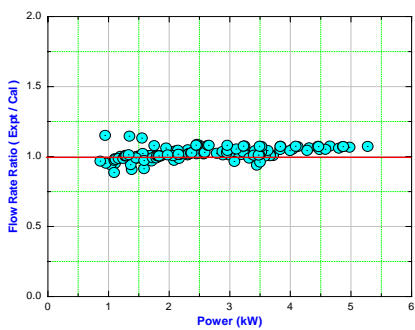


Fig.12: Comparison of RELAP5/ MOD3.2 prediction with data

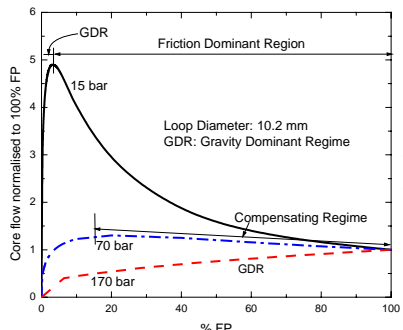


Fig.13: Flow regimes in two-phase loops

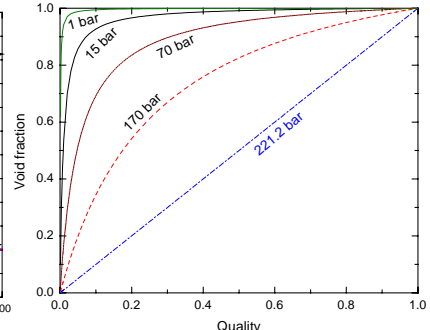


Fig.14: Effect of pressure on void fraction

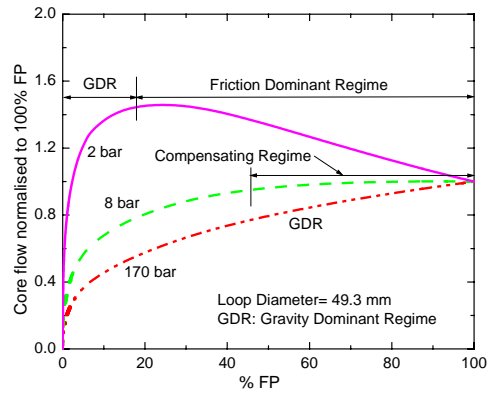
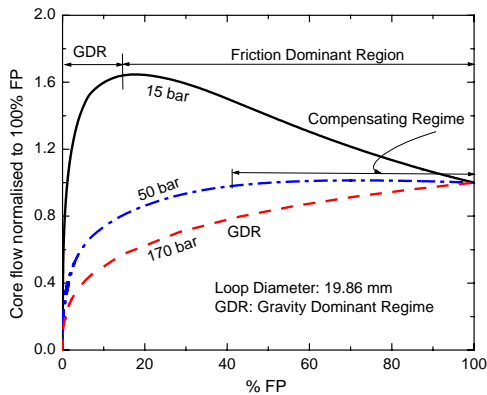


Fig.15: Effect of loop diameter on the flow regimes in two-phase loops

5. Conclusions

A generalized correlation for steady state flow applicable to both single-phase and two-phase natural circulation systems has been presented. For both single-phase and two-phase natural circulation systems, the steady state behavior can be simulated by preserving Gr_m/N_G same in the model and prototype. The given correlation has been tested with data from several single-phase and two-phase natural circulation loops. The database included steady flow data from both uniform and nonuniform diameter loops which were found to be in reasonable agreement with the proposed correlation. The data for two-phase natural circulation included loops relevant to BWRs and PWRs which were found to be in reasonable agreement with the proposed correlation. The given correlations are useful to identify the gravity dominant, friction dominant and the compensating flow regimes in two-phase loops. The flow regimes are found to be strong function of the system pressure and loop hydraulic diameter.

NOMENCLATURE

A	: flow area, m ²	P	: pressure, Pa
a	: dimensionless flow area, A_i/A_r	q''	: heat flux, W/m ²
b	: exponent in the friction factor equation	Q	: total heat input rate, W
D	: hydraulic diameter, m	Re	: Reynolds number, $D_r W_{ss}/A_r \mu_l$
d	: dimensionless hydraulic diameter, D_i/D_r	s	: co-ordinate around the loop, m
f	: Darcy-Weisbach friction factor	S	: dimensionless co-ordinate around the loop, s/H
g	: gravitational acceleration, m/s ²	T	: temperature, K
Gr_m	: modified Grashof number defined by Eq. (17)	v	: specific volume, m ³ /kg
h	: enthalpy, J/kg	V_t	: total loop volume, m ³
H	: loop height, m	W	: mass flow rate, kg/s
\mathcal{H}	: dimensionless enthalpy, $\mathcal{H} = (h - h_r)/(\Delta h)_{ss}$	x	: quality
K	: local pressure loss coefficient	z	: elevation, m
l	: dimensionless length, L_i/L_t	Δz	: Centre line elevation difference, m
L	: length, m	Z	: dimensionless elevation, z/H
N	: number of pipe segments		
N_G	: dimensionless parameter defined by Eq. (14)		
p	: constant in the friction factor equation		

Greek Symbols

α	: void fraction	ϕ	: $a_i H/L_t$
β_h	: enthalpic thermal expansion coefficient, kg/J	ϕ_{LO}^2	: two-phase friction multiplier
β_T	: Thermal expansion coefficient based on temperature, K ⁻¹	$\overline{\phi_{LO}^2}$: mean value of ϕ_{LO}^2
μ	: dynamic viscosity, Ns/m ²	ρ	: density, kg/m ³
		ω	: dimensionless mass flow rate

Subscripts

B	: boiling length	g	: vapor	p	: pipe
C	: condensing section	h	: heater	r	: reference value
c	: cooler/ condenser	i	: i th segment	SD	: steam drum

e	: heater exit	in	: heater inlet	sp	: single phase
eq	: equivalent	l	: liquid	ss	: steady state
eff	: effective	LO	: liquid only	t	: total
F	: feed	m	: mean	tp	: two-phase

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