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STEADY STATE FLOW IN SINGLE-PHASE AND TWO-PHASE NATURAL CIRCULATION LOOPS

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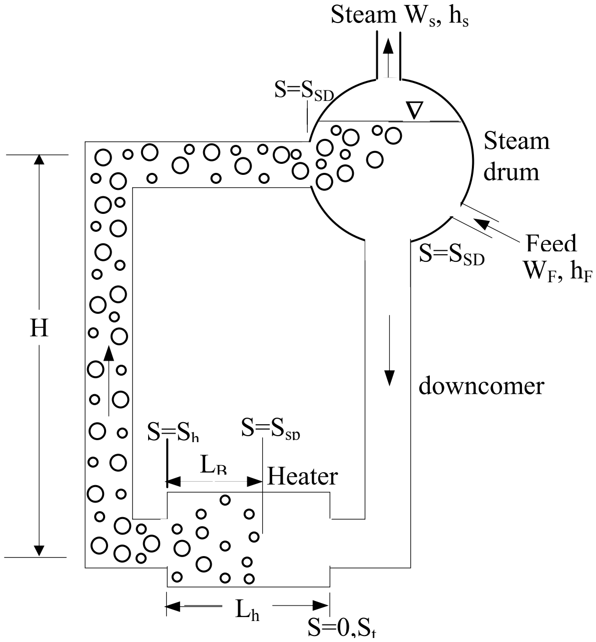
Abstract

Derivation of a steady state flow correlation valid for both single- and two-phase natural circulation systems has been presented. The correlation has been extensively tested with data from both uniform and nonuniform diameter single-phase loops showing good agreement. Experiments have been conducted in four uniform diameter two-phase loops differing in diameter. The data from all these loops are also found to be in good agreement with the theoretical correlations.

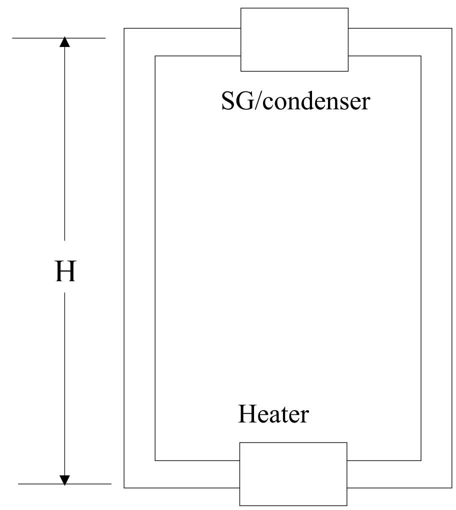
1. Introduction

Natural circulation loops are extensively used in several industries. Both single-phase and two-phase natural circulation systems are industrially important. Single-phase systems are used in the solar water heaters, transformer cooling and nuclear reactor 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.

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 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.



(a) Loop relevant to BWRs



(b) Loop relevant to PWRs

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

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 assumed to be low 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 around 5 ^{0}C.

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:

$$\frac{\partial W}{\partial s}=0 \qquad \text{Conservation of mass} \qquad (1)$$

$$\frac{W}{A_i}\frac{\partial h}{\partial s}=\begin{cases}\dfrac{4q_h''}{D_h} & \textit{heater}\\ 0 & \textit{pipes}\end{cases} \qquad \text{Conservation of energy} \qquad (2)$$

$$\frac{W^2}{A_i^2}\frac{\partial}{\partial s}\left(\frac{1}{\rho}\right)=-\frac{\partial p}{\partial s}-\rho g\sin\theta-\frac{fW^2}{2D_i\rho A_i^2}-\frac{KW^2}{2\rho A_i^2 L_t} \quad \text{Conservation of momentum} \quad (3)$$

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

$$W^2\sum_{i=1}^{N_t}\frac{1}{A_i^2}\oint\partial v=-\oint\partial p-g\oint\rho\,dz-\frac{W^2}{2\rho}\sum_{i=1}^{N_t}\left(\frac{fL}{DA^2}+\frac{K}{A^2}\right)_i \qquad (4)$$

Noting that $\oint\partial v=\oint\partial p=0$ for a closed loop, we can write

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

The density is assumed to vary as $\rho=\rho_r\left[1-\beta_h(h-h_r)\right]$ in the buoyancy force term where $\beta_h=(1/\mathrm{v})(\partial\mathrm{v}/\partial\mathrm{h})_p$. 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

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

where L_B is the boiling length in the heated section and N_s, N_B and N_{tp} are the number of pipe segments in the single-phase, two-phase heated section and the unheated riser section respectively. In Eq. (6), the local loss coefficients have been replaced by an equivalent length such that $\left(L_{eq}\right)_i = K_i D_i / f_i$ and $\left(L_{eff}\right)_i = L_i + \left(L_{eq}\right)_i$. While estimating $\left(L_{eq}\right)_i$, 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 coefficient and local loss coefficient. 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 $f_i = p/\mathrm{Re}_i^b$ then Eq. (6) can be rewritten as

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

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 $\left(\phi_{LO}^2\right)_i$ for the i[th] segment instead of numerically integrating $\int \phi_{LO}^2(s)ds$ over the heated section as in Eq. (6).

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

If we can approximate β_h and ϕ_{LO}^2 by their mean values for the heated section, then we obtain

$$g\rho_r \overline{\beta}_h \oint h\,dz = \frac{p\mu_l^b W^{2-b}}{2\rho_l}\left\{\sum_{i=1}^{N_s}\left(\frac{L_{eff}}{D^{1+b}A^{2-b}}\right)_i + \overline{\phi}_{LO}^2\sum_{i=N_s}^{N_B}\left(\frac{L_{eff}}{D^{1+b}A^{2-b}}\right)_i + \phi_{LO}^2\sum_{i=N_B}^{N_t}\left(\frac{L_{eff}}{D^{1+b}A^{2-b}}\right)_i\right\} \tag{8b}$$

The above equations can be nondimensionalized using the following substitutions:

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

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

$$A_r = \frac{1}{L_t}\sum_{i=1}^{N_t} A_i L_i = \frac{V_t}{L_t}; D_r = \frac{1}{L_t}\sum_{i=1}^{N_t} D_i L_i \;; \rho_r = \rho_l \;;\; and\; h_r = h_F \tag{9b}$$

At steady state $\omega_{ss} = 1, \quad Q_{SD} = Q_h$ and the non-dimensional equations will become

$$\frac{d\,\Theta}{dS} = \begin{cases} \phi_h \dfrac{V_t}{V_h} & heater\ (0 < S \le S_h) \\ 0 & pipes\ (S_h < S \le S_{SD}\ and\ S_{SD} < S \le S_t) \end{cases} \tag{10}$$

$$g\rho_r \Delta h_{ss} \overline{\beta}_h H \oint \Theta\, dz = \frac{pL_t \mu_l^b W_{ss}^{2-b}}{2\rho_l D_r^{1+b} A_r^{2-b}} \left\{ \begin{array}{l} \displaystyle\sum_{i=1}^{N_s}\left(\frac{l_{eff}}{d^{1+b}a^{2-b}}\right)_i + \overline{\phi}_{LO}^2 \sum_{i=N_s}^{N_B}\left(\frac{l_{eff}}{d^{1+b}a^{2-b}}\right)_i + \\ \displaystyle\phi_{LO}^2 \sum_{i=N_B}^{N_t}\left(\frac{l_{eff}}{d^{1+b}a^{2-b}}\right)_i \end{array} \right\} \tag{11}$$

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

$$\Theta = \Theta_{in} + \phi_h \frac{V_t}{V_h} S \qquad \text{for the heated section } 0<S \le S_h \tag{12a}$$

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

$$\Theta_l = \Theta_{in} + \phi_h \frac{V_t}{V_h} S_{sp} \quad or \quad S_{sp} = \left(\frac{\Theta_l - \Theta_{in}}{\phi_h}\right)\frac{V_h}{V_t} \tag{12b}$$

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. (12a) as

$$\Theta_e = \Theta_{in} + 1 \qquad \text{for } S_h < S < S_{SD} \tag{12c}$$

The steam drum, where complete separation of the steam-water mixture is assumed to take place without any carryover or carry-under 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

$$\Theta_{in} = \Theta_l - x_e(\Theta_l - \Theta_F) \tag{12d}$$

Using the above in Eq. (12a), we obtain

$$\Theta_e = \Theta_l - x_e(\Theta_l - \Theta_F) + 1 \tag{12e}$$

If we choose h_r=h_F, then (12e) can be rewritten as

$$\Theta_e = \Theta_l (1 - x_e) + 1 \tag{12f}$$

Replacing $(\Delta h)_{ss} = Q_h / W_{ss}$ in Eq. (11) and noting that $\oint \Theta \, dZ = 1$ we obtain

$$W_{ss} = \left[\frac{2}{p} \frac{g \rho_l^2 \overline{\beta}_h H Q D_r^b A_r^{2-b}}{\mu_r^b N_G} \right]^{\frac{1}{3-b}} \tag{13}$$

which can be expressed in dimensionless form as

$$\mathrm{Re}_{ss} = 0.1768 \left(\frac{Gr_m}{N_G} \right)^{0.5} \quad \text{for laminar flow loop} \tag{14}$$

and $$\mathrm{Re}_{ss} = 1.9561 \left(\frac{Gr_m}{N_G} \right)^{0.36364} \quad \text{for turbulent flow loop} \tag{15}$$

Where $$\mathrm{Re}_{ss} = \frac{D_r W_{ss}}{A_r \mu}; \quad Gr_m = \frac{D_r^3 \rho_l^2 \overline{\beta}_h g Q_h \mathrm{H}}{A_r \mu_r^3}; \quad and \tag{16}$$

$$N_G = \frac{L_t}{D_r} \left[\sum_{i=1}^{N_s} \left(\frac{l_{eff}}{d^{1+b} a^{2-b}} \right)_i + \sum_{i=N_s}^{N_B} \left(\frac{\overline{\phi}_{LO}^2 l_{eff}}{d^{1+b} a^{2-b}} \right)_i + \phi_{LO}^2 \sum_{i=N_B}^{N_t} \left(\frac{l_{eff}}{d^{1+b} a^{2-b}} \right)_i \right] \tag{17a}$$

where an average value of $\overline{\phi}_{LO}^2$ is used over each segment. On the other hand, if a mean value could be used over the entire heated length, then we get

$$N_G = \frac{L_t}{D_r} \left[\sum_{i=1}^{N_s} \left(\frac{l_{eff}}{d^{1+b} a^{2-b}} \right)_i + \overline{\phi}_{LO}^2 \sum_{i=N_s}^{N_B} \left(\frac{l_{eff}}{d^{1+b} a^{2-b}} \right)_i + \phi_{LO}^2 \sum_{i=N_B}^{N_t} \left(\frac{l_{eff}}{d^{1+b} a^{2-b}} \right)_i \right] \tag{17b}$$

It may be noted that for uniform diameter loop (UDL), N_G reduces to the following equation

$$N_G = \frac{L_t}{D_r} \left[(l_{eff})_{sp} + \overline{\phi}_{LO}^2 (l_{eff})_B + \phi_{LO}^2 (l_{eff})_{tp} \right] \tag{18}$$

3. Evaluation of $\overline{\beta}_h, \overline{\phi}_{LO}^2$ and ϕ_{LO}^2

We had used mean values of the parameters β_h and 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, $\overline{\phi}_{LO}^2$ 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 $\overline{\phi}_{LO}^2$.

$$\phi_{LO}^2 = \frac{\rho_l}{\rho_e}\left[\frac{1}{1+x_e\left(\frac{\mu_l}{\mu_g}-1\right)}\right]^b \text{ and } \overline{\phi}_{LO}^2 = \frac{\rho_l}{\overline{\rho}_m}\left[\frac{1}{1+\frac{x_e}{2}\left(\frac{\mu_l}{\mu_g}-1\right)}\right]^b \tag{19}$$

where $\rho_e = \dfrac{\rho_g \rho_l}{x_e(\rho_l - \rho_g) + \rho_g}$ and $\overline{\rho}_m = \dfrac{\rho_g\,\rho_l}{0.5x_e\left(\rho_l - \rho_g\right) + \rho_g}$

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).

We have proposed a new parameter, β_h, which is the volumetric thermal expansion coefficient defined as $\beta_h = (1/v)(\partial v/\partial h)_p$. While deriving Eq. (14), 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.

$$\overline{\beta}_h = \frac{1}{v}\left(\frac{\partial v}{\partial h}\right)_p = \frac{1}{\mathrm{v_m}}\frac{\Delta \mathrm{v}}{\Delta \mathrm{h}} = \frac{1}{0.5(v_{in}+v_e)}\left(\frac{v_e - v_{in}}{h_e - h_{in}}\right) = \frac{\rho_{in}-\rho_e}{0.5(\rho_e+\rho_{in})\Delta h} = \frac{\Delta\rho}{\rho_m \Delta h} \tag{20}$$

It may be noted that the averaging procedure described above has made it possible to calculate the flow rates using the simple equation (13). At this stage, it becomes essential to validate the averaging procedure described by Eq. (19) and (20). For this, comparison, the flow rates were calculated using Eq. (13) with the N_G calculated by Eq. (17) (see Fig.3). It can be seen that the averaging leads to close results above a quality of approximately 3% which is adequate for engineering calculations.

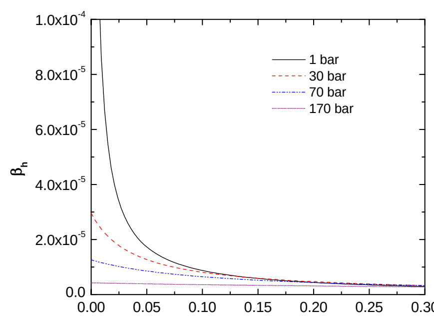


Fig.2: Variation of β_h with pressure

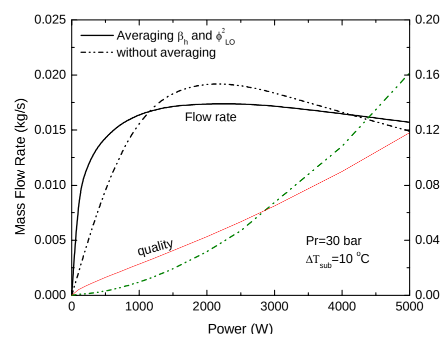


Fig.3: Effect of averaging of β_h and ϕ_{LO}^2

4. SPECIAL CASES

Equations (13) to (15) were derived for loops relevant to BWRs. However, it can be easily extended to other loop geometries.

4.1 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

$$\frac{d\Theta}{dS} = \begin{cases} \phi_h \dfrac{V_t}{V_h} & heater\ (0 < S \le S_h) \\ 0 & pipes\ (S_h < S \le S_{hl}\ and\ S_c < S \le S_t) \\ -\phi_c \dfrac{V_t}{V_c} & condenser\ (S_{hl} < S \le S_c) \end{cases} \tag{21}$$

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 are the same as given before leading to the same equations for the flow rate. However, the equation for N_G is obtained as

$$N_G = \frac{L_t}{D_r}\left[(l_{eff})_{sp} + \overline{\phi}_{LO}^2 (l_{eff})_B + \phi_{LO}^2 (l_{eff})_{tp} + \overline{\phi}_{LO}^2 (l_{eff})_C\right] \tag{22}$$

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

3.2 Single-phase natural circulation

For single-phase natural circulation $x_e = 0$ and by definition $\phi_{LO}^2 = 1$. Using this value in Eq. (17), we get

$$N_G = \frac{L_t}{D_r}\sum_{i=1}^{N_s}\left(\frac{l_{eff}}{d^{1+b}a^{2-b}}\right)_i \tag{23}$$

which is same as that given for single-phase flow by Vijayan (2002). Similarly, noting that h=CpT with Cp assumed to be a constant we get

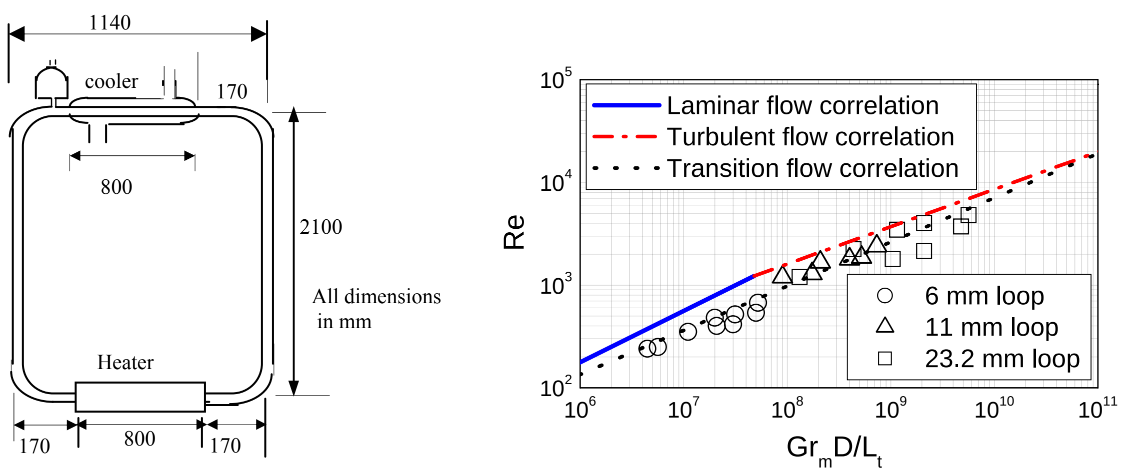
$$\beta_h = \frac{1}{v}\left(\frac{\partial v}{\partial h}\right)_p = \frac{1}{vCp}\left(\frac{\partial v}{\partial T}\right)_p. \text{ Noting that } \beta_T = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_p \text{ we obtain } \beta_h = \frac{\beta_T}{Cp}. \tag{24}$$

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 modified Grashof number for single-phase flow becomes

$$Gr_m = \frac{D_r^3 \rho_r^2 \beta_T g \Delta T_r}{\mu_r^2} \quad where\ \Delta T_r = \frac{Q_h H}{A_r \mu_r Cp} \qquad (25)$$

which is same as that reported in Vijayan et al. (2004). With this substitution, equations (14) and (15) are applicable to single-phase natural circulation systems. For the special case of uniform diameter loops (UDL), Eq. (18) reduces to

$$N_G = \frac{\left(L_{eff}\right)_t}{D} \text{ and in case of UDLs with negligible local pressure losses } N_G = \frac{L_t}{D}. \qquad (26)$$



(a) The experimental loop (b) Comparison of data with correlations

Fig.4: Effect of loop diameter on steady state NC (Vijayan et al. (1992)

4. TESTING OF THE STEADY STATE CORRELATION

The steady state data from both single-phase and two-phase loops are used to test the correlations derived above.

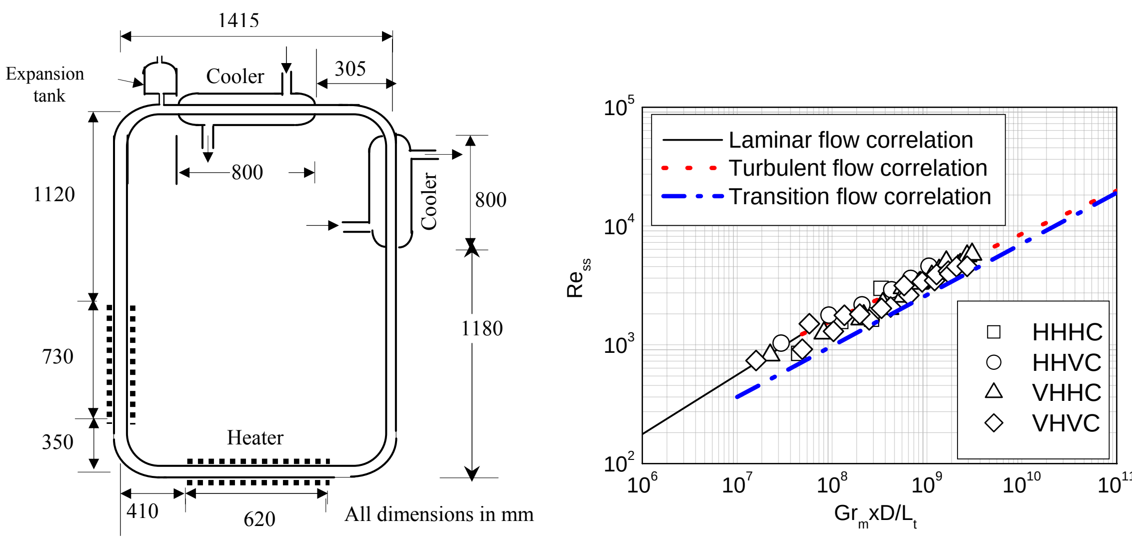
4.1 Single-phase NCLs

4.1.1 Database for Uniform Diameter Loops (UDL)

The correlations (14) and (15) together with Eqs. (25) and (26) were extensively tested with data from simple uniform diameter loops. Among the simple 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), Hoe 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.

It was found that uniform diameter rectangular loop data from loops differing in diameter could be reasonably represented by the above correlation (Fig. 4). Even better comparison is obtained with data generated for different orientations of heater and cooler (Fig. 5). All the four orientations of heater and cooler were studied, i.e. horizontal heater horizontal cooler (HHHC), horizontal heater vertical cooler (HHVC), vertical heater horizontal cooler (VHHC) and vertical heater vertical cooler (VHVC). The data from all uniform diameter loops neglecting the effect of local losses are plotted in Fig. 6a. Effect of local losses was found to improve the agreement with the data in the turbulent regime (Fig. 6b). However, it has no significant influence in the laminar regime data since fL_t/D is much greater than sum of all loss coefficients.

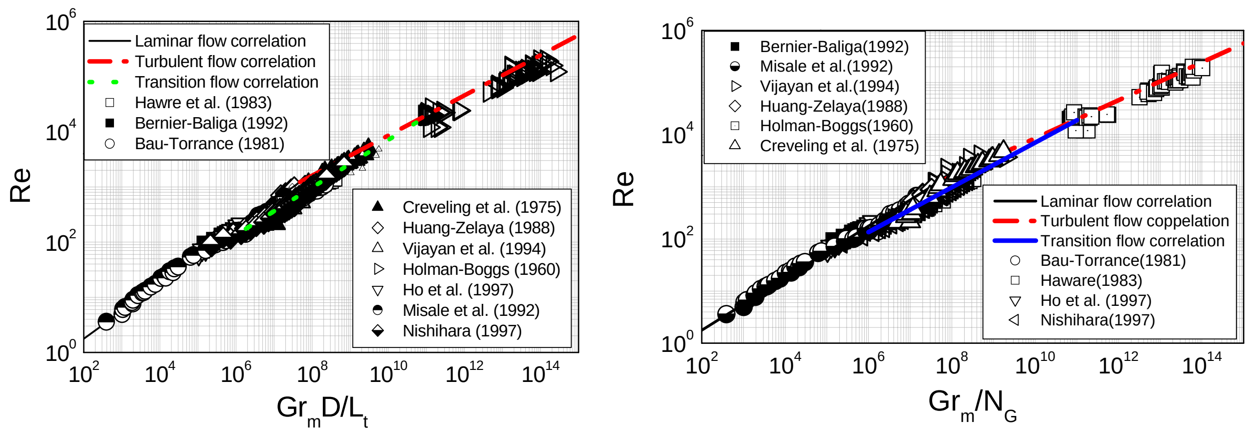


(a) Experimental loop (b) Effect of heater and cooler orientation

Fig. 5: Effect of heater and cooler orientation on steady state NC (Vijayan et al. (2001))

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.



(a) UDL data neglecting local losses (b) UDL data considering local losses

Fig. 6 Data from all uniform diameter loops

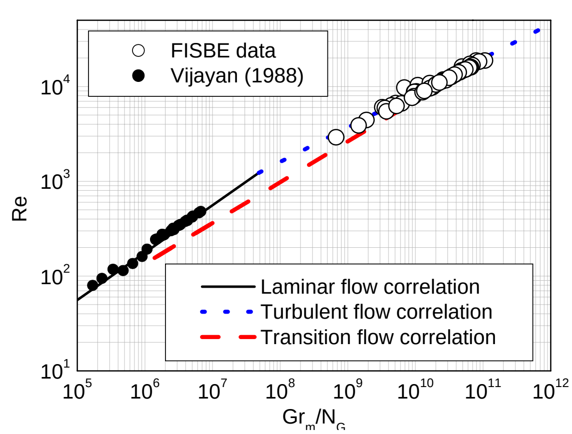


Fig.7: In-house data on figure-of-eight loops

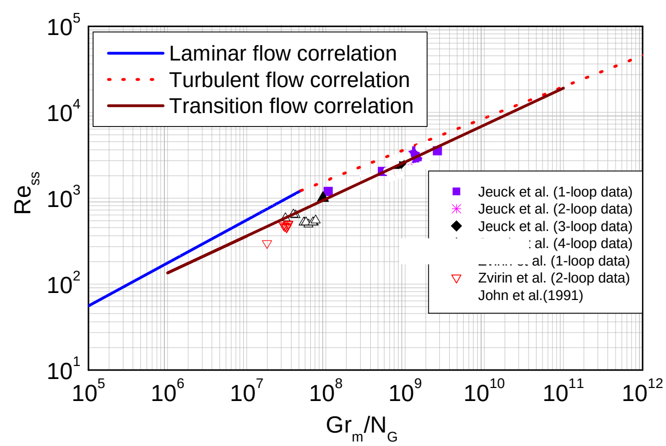


Fig.8: Parallel loops data neglecting local losses

In-house data on nonuniform diameter figure-of-eight loops are tested with the correlations in Fig. 7. The parallel loop data of Jeuck et al. (1981) and Zvirin et al. (1981) are also found to be in reasonable agreement with the theoretical correlation (Fig.8). The data from all NDLs are plotted in Fig. 9 neglecting the local losses. Both uniform diameter and nonuniform diameter loop data neglecting the local losses are plotted together in Fig. 10. 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 ($5x10^6 < Gr_m/N_G < 10^{11}$) significant deviation is observed where the flow is neither fully laminar nor fully turbulent. Thus, it becomes clear that normal fully developed flow friction factor correlations are valid if the entire loop is either in laminar or turbulent flow irrespective of whether it is a UDL or NDL.

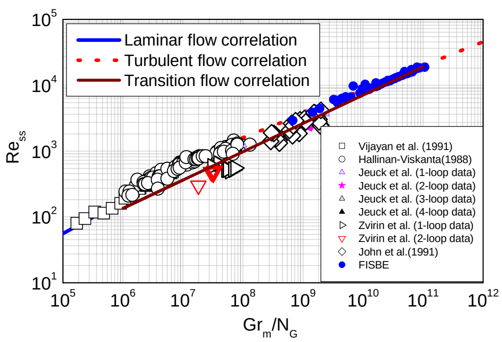


Fig. 9: NDL data neglecting local losses

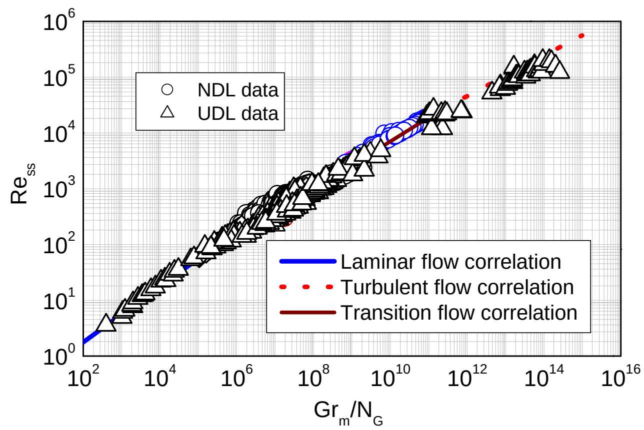


Fig. 10: All single-phase NC data neglecting local losses

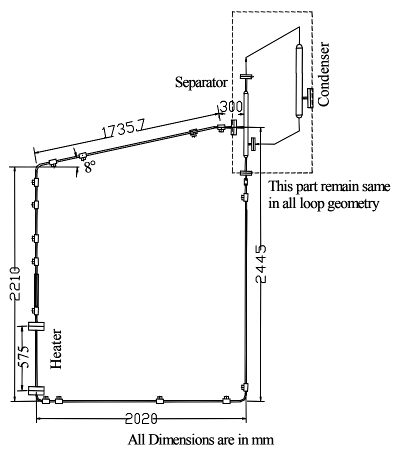


Fig. 11: Schematic of the experimental loop

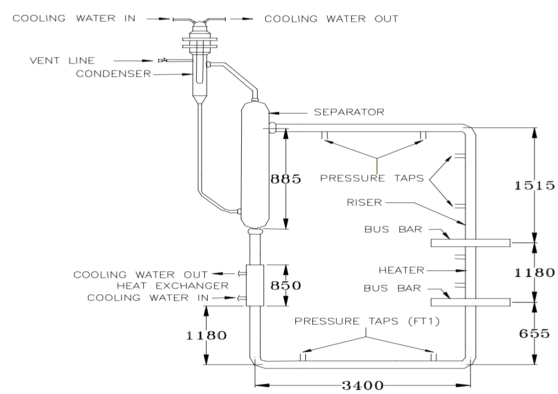


Fig. 12: Large diameter loop

4.2 Two-phase NCLs

4.2.1 Database for uniform diameter two-phase loops

Experiments were conducted in three loops of inside diameter 10.21 mm, 15.74 mm and 19.86 mm respectively in a facility having the geometry shown in Fig. 11. The steam separator, the condenser and the associated piping (the portion inside the rectangular box in Fig. 11) were the same for all the loops. The steam separator was made up of 59 mm inside diameter (2.5” NB Sch 80) pipe. The vertical heater section was direct electrically 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 was returned to the steam drum. The loop was extensively instrumented to measure temperature, pressure, differential pressure, level, flow rate, 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. Further details of the loop are available in the report by Dubey et al. (2004). In addition, experimental data were generated in a 49.3 mm inside diameter loop shown in Fig. 12. Further details of the 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 calculated reference diameter is 8.47 mm and the data were available for 55 < pressure ≤ 138 bar, 8.2 < quality ≤ 69.3% and 8.3 < power ≤ 65 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 correlations (Fig. 13). The experimental data is observed to be very close to the theoretical correlation (within an error bound of +/- 40%) for all the two-phase natural circulation loops confirming the validity of the proposed correlations.

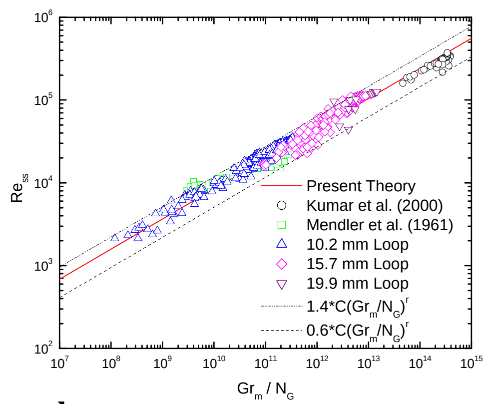


Fig. 13: Comparison of theoretical and experimental results

5. Concluding Remarks

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 for single-phase loops included uniform and nonuniform diameter loops which were found to be in close agreement with the proposed correlation. The effect of local pressure loss coefficient was found to be negligible for laminar flows. 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.

NOMENCLATURE

A : flow area, m^2
a : dimensionless flow area, A/A_r
b : constant in equation (7)
D : hydraulic diameter, m
d : dimensionless hydraulic diameter, D/D_r
f : Darcy-Weisbach friction factor
g : gravitational acceleration, m/s^2
Gr_m : modified Grashof number defined by Eq. (16)
h : enthalpy, J/kg
H : loop height, m
Θ : dimensionless enthalpy, $\Theta = (h - h_r)/(\Delta h)_{ss}$
K : local pressure loss coefficient
l : dimensionless length, L_i/L_t
L : length, m
N : total number of pipe segments
N_G : dimensionless parameter defined by Eq. (17)
p : constant in equation (7)
q'' : heat flux, W/m^2
Q : total heat input rate, W
Re : Reynolds number, $D_r W/A_r \mu$
s : co-ordinate around the loop, m
S : dimensionless co-ordinate around the loop, s/H
T : temperature, K
v : specific volume, m^3/kg
V_t : total loop volume, m^3
W : mass flow rate, kg/s
x : quality
z : elevation, m
Δz : Centre line elevation difference
Z : dimensionless elevation, z/H

Greek Symbols

α : void fraction
β_h : enthalpic thermal expansion coefficient, kg/J
β_T : Thermal expansion coefficient based on temperatue, K^{-1}
μ : dynamic viscosity, $N\,s/m^2$

ϕ_h	: $a_h H / L_t$
ϕ_{LO}^2	: two-phase friction multiplier
$\bar{\phi}_{LO}^2$	: average two-phase friction multiplier
ρ	: density, kg/m^3
ρ_r	: reference density, kg/m^3
ω	: dimensionless mass flow rate

Subscripts

B	: boiling length
C	: condensing section
c	: cooler
eq	: equivalent
eff	: effective
g	: vapor
h	: heater
he	: heater exit
i	: ith segment
in	: inlet
l	: liquid
LO	: liquid only
m	: mean
p	: pipe
r	: reference value
sp	: single phase
ss	: steady state
t	: total
tp	: two-phase

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