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Two-phase flow boiling pressure drop in small channels

Madhavi V. Sardeshpande\textsuperscript{a}, Parikshit Shastri\textsuperscript{b} and Vivek V. Ranade\textsuperscript{a*}

\textsuperscript{a}Industrial Flow modeling group, National Chemical Laboratory, Pune – 411 008
\textsuperscript{b}Chemical Engineering Department, Visvesvaraya National Institute of Technology Nagpur – 440 010
Email: mv.sardeshpande@ncl.res.in, vv.ranade@ncl.res.in
Abstract

Two-phase flow boiling in small channels finds a variety of applications in power and process industries. Heat transfer, boiling flow regimes, flow instabilities, pressure drop and dry out are some of the key issues related to two-phase flow boiling in channels. In this work, the focus is on pressure drop in two-phase flow boiling in tubes of 19 mm diameter. These tubes are typically used in steam generators. Relatively limited experimental database is available on 19mm ID tube. Therefore, in the present work, the experimental set-up is designed for studying flow boiling in 19 mm ID tube in such a way that any of the different flow regimes occurring in a steam generator tube (from pre-heating of sub-cooled water to dry-out) can be investigated by varying inlet conditions. The reported results cover a reasonable range of heat and mass flux conditions such as 9 – 27 kW/m² and 2.9 – 5.9 kg/m² s respectively. In this paper, various existing correlations are assessed against experimental data for the pressure drop in a single, vertical channel during flow boiling of water at near-atmospheric pressure. A special feature of these experiments is that time-dependent pressures are measured at four locations along the channel. The steady-state pressure drop is estimated and the identification of boiling flow regimes is done with transient characteristics using time series analysis. Experimental data and corresponding results are compared with the reported correlations. The results will be useful for understanding key aspects of flow boiling in small channels.

Keywords: flow boiling, pressure drop, pressure fluctuations, flow regimes, instability
1. Introduction

Boiling in small channels finds applications in a variety of areas including steam tubes in boilers, compact evaporators, compact heat exchangers etc. Two-phase flow inside tubes provides a very effective way of heat transfer and fluid movement. The design of such steam generator tubes involves an understanding of heat transfer coefficient, flow regimes, pressure drop and the flow instabilities. In the present work, an approach is developed for mimicking the flow, heat transfer regimes and flow instabilities occurring in long steam generator tubes in laboratory scale tubes. In this approach, laboratory scale steam generator tube of 1m length and diameter 19mm is used. Inlet and boundary conditions are varied to simulate different flow regimes occurring in long steam generator tubes. Study of two-phase flow boiling pressure drop, flow instability and identification of flow regimes using pressure fluctuations is the main focus of the present work.

Two-phase pressure drop depends on various parameters like the geometric configuration of the channel, mass and volume fractions of the individual phases, pressure, fluid properties, mass flux, orientation of the channel (i.e. horizontal, vertical or inclined) and flow patterns. Further, in many engineering applications, two-phase flow systems can be adiabatic or diabatic. To cater to the needs of these diverse applications, a large amount of work has been devoted to the study of fluid flow and heat transfer along with pressure drop mechanisms in micro and mini channels. Specifically, researchers have concentrated on the prediction of flow patterns\(^1\), heat transfer characteristics\(^2\)\(^-\)\(^4\) and instability\(^5\)\(^-\)\(^6\) (in two-phase flows in micro channels). Kandlikar\(^7\) has highlighted the effect of small diameter on the boiling phenomena, the nature of heat transfer and pressure drop and the difference between single and multiple channels. The literature survey reveals that most of the work in the area of micro-channels is focused in the range of 100 µm – 4mm using refrigerants and relatively limited studies have been performed with water as the working fluid. In recent years, a notable research growth was observed on two-phase flow and evaporation heat transfer in the micro-scale channels. However, despite a large number of publications (See Table 1), it is rare to find detailed information regarding pressure drop, heat transfer and stability characteristics for the small channels. These flow boiling characteristics are extremely important to evaluate the current status of the research in the mentioned areas.

Predicting the two-phase distribution in the system for given operating conditions is a formidable task for most of the industrial applications. One of the critical unknown parameter involved in predicting the pressure drop/loss and heat transfer in any gas–liquid system is the void fraction (ε) which is the volume of space occupied by the vapors. The accurate prediction of void fraction in vertical channels is of immense importance in determining the two-phase pressure drop. For many practical situations, designers and analysts often require some guidance to choose appropriate correlations. Most of these correlations are flow pattern–specific and are limited in their application in terms of the range of flow variables. In order to choose the right correlation for the desired application from a pool of available correlations in the
literature, it is utmost importance to sort out and identify the correlations applicable to a range of flow variables and channel orientations.

Therefore, in the present work, an attempt has been made to systematically investigate the two-phase flow with phase change in a 19mm channel size. Heat transfer experiments were performed in uniformly heated, single, annular, vertical channel using water as a working fluid. The experiments were performed under adiabatic and diabatic conditions to study flow instabilities occurring because of flow boiling. Efforts were taken to acquire two-phase pressure fluctuations at different axial locations and the results were compared with the experimentally calculated pressure drop over a range of experimental conditions using existing empirical correlations. An attempt has been made to classify different flow regimes based on the analysis of pressure fluctuations. The results will be useful for understanding the key aspects of the flow boiling in small channels.

2. Experimental facility

Vertical concentric double pipe heat exchanger with an inner diameter of 19mm and the heated length of 1.05 meter was used for the present study. This experimental test section was fabricated and designed with SS-110. The schematic of the test section is as shown in Figure 1a.

The main components of the system include the feed vessel, pump, rotameter, heating oil bath, pre-heater and a condenser. As indicated in Figure 1a, water was pumped by the peristaltic pump to the test section through regulated rotameter (calibrated over the range of 1 – 5 l/h flow rate of water). Before starting the two-phase flow experiments; calibration of heating oil bath (manufactured by JULABO: SL_6) was carried out to establish actual mass flow rate at different temperature set points of the oil bath. Data acquisition system was connected to the computer for online measurement via HMI (Human Machine Interface). The test section was designed like a co-current flow tube-in-tube heat exchanger as shown in Figure 1b where water was flowing from inside tube and hot oil was circulated from the outer tube. Water, as a working fluid, was circulated from bottom to top along the inner pipe and hot oil, used as a heating medium, was made to flow co-currently along the annulus. Pressure sensors “WIKA-S10” having a pressure range of 0-0.16 bar, sampling rate of 1Hz and response time <1msec were used for absolute pressure measurement. Four such pressure sensors were mounted horizontally along the axial length of the channel at locations Z1, Z2, Z3, Z4 to measure corresponding fluid pressure P1, P2, P3, P4 respectively inside the annulus as shown in Figure 1b. Special care was taken for pressure sensors to be leak proof. Heating medium was stored and circulated using hot oil bath with varying heat flux. Water flow rate was regulated by a peristaltic pump (RS232; MXI Technologies). Before taking heat transfer measurements, initial system testing was performed and the power as well as water to the facility was activated after ensuring proper functioning of all the components. The variable speed peristaltic pump was then turned on to establish steady water flow. The flow oscillations were dampened using damper and continuous steady flow was sent to test section. Initially single phase flow
experiments (adiabatic) and then two-phase experiments (diabatic) were carried out; continuous cold water supply was maintained externally to the condenser to cool the outlet steam from channel and also to avoid over-heating of feed tank.

Two-phase flow experiments were performed at four different heat flux conditions (9.8, 14.8, 21.4, 27.8 kW/m²) for each of the three mass flux conditions (2.9, 3.9, 5.9 kg/m² s). Each of these experiments were carried out with three different conditions of inlet water temperature (30°C, 60°C, 95°C). Accordingly, absolute pressure fluctuations data was recorded via HMI (Human Machine Interface) during each experiment at every 1 sec up to 60 minutes at points P1, P2, P3, P4 where average pressure fluctuations data for last 10 minutes of each run was taken for detailed steady-state analysis. Fully developed conditions were analyzed for different operating conditions and observed pressure fluctuations pattern was saved on personal computer. Temperature fluctuations and corresponding energy balance has been broadly discussed by Sardeshpande et al., 9.

2.1 Data reduction
2.1.1 Pressure drop in Single Phase flow
Pressure drop in single-phase (adiabatic process) in vertical channels is attributed to gravitational head and frictional head. Acceleration pressure drop ($\Delta P_a$) is zero in single phase since density variation is negligible.

$$\Delta P_T = \Delta P_g + \Delta P_f$$

(1)

Single phase gravitational pressure drop is defined as

$$\Delta P_g = (\rho gz)_i - (\rho gz)_o$$

(2)

where z in each case denote the height of water column above that particular point.

Single phase frictional pressure drop is defined as

$$\Delta P_f = \frac{2f_LG^2z_t}{DhP_L}$$

(3)

where $f_L$ frictional factor i.e. fanning friction factor for single phase in laminar region is given by

$$f_L = \frac{16}{Re} \text{ where Re is Reynolds number}$$

(4)

Single phase pressure drop was estimated using experimental data. Reproducibility of the data was validated by carrying out each experiment three times. Absolute pressure readings were recorded for 4 different flow rates (1.9, 2.9, 3.9 and 5.9 l/h) in laminar flow region (40 < Re <
It was observed that the pressures at all locations were steady within fixed pressure range and were well in agreement with each other in ± 0.05 % range. Absolute pressure readings were recorded for 4 different flow rates (1.9, 2.9, 3.9 and 5.9 l/h) in the range of Reynolds number 40-140 (i.e. laminar flow region); and the related pressure drop was calculated. In case of frictional pressure drop, frictional factor plays an important role and is closely related to Reynolds number (calculated using Equation 3). The experimental and calculated pressure at P1, P2, P3, P4 locations was found to be in agreement with each other. Single phase validation was done by plotting frictional factor versus Reynolds number as per Moody’s chart (Figure 2). Moody’s chart is based on the calculation of fanning friction factor. Fanning friction factor (experimental) for various flow rates was calculated and plotted against the Reynolds number. According to Moody\textsuperscript{10}, this frictional factor should theoretically satisfy correlation with Reynolds number given in Equation 4.

The results of frictional factor versus Reynolds number were in agreement with Equation 4 and the same reflected from the present experimental data.

2.2 Pressure drop in two-phase diabatic flow

2.2.1 Preliminary observations of raw pressure data
Total theoretical pressure drop was calculated using basic gravitational and frictional head in case of single phase i.e. adiabatic flow system. In two-phase flow boiling system, total pressure drop is attributed as per Figure 3a and Figure 3b. As per Figure 3a, in case of diabatic flow system, total two-phase pressure drop consisted the gravitational, acceleration and the frictional pressure drop. Various models such as homogeneous, Lockhart-Martinelli, general Grahams etc. are available for estimating total pressure drop. Most of the researchers have put their efforts to establish two-phase flow boiling frictional component of pressure drop in various ways. Schematic of overall data reduction for the entire pressure drop component is as shown in Figure 3b. In the present case, pressure drop was calculated using recorded experimental pressure data and was compared it with various pressure drop models and co-relations reported in literature.

Experiments were carried out at three inlet water temperature conditions of 30, 60 and 95\degree C respectively. The acquired pressure data at Z1, Z2, Z3, Z4 locations was analyzed at 2.9 kg/m\textsuperscript{2}s and 14.8 kW/m\textsuperscript{2} mass and heat flux conditions respectively. It was observed that once heating started from wall side, nucleate boiling played a significant role and pressure fluctuations were observed along the vertical length of the channel. At 30\degree C inlet condition, flow boiling initiated along the length with minor pressure fluctuations at respective sensor locations. In order to get an idea of pressure fluctuation, a sample raw data of pressure at Z1, Z2, Z3 and Z4 locations was plotted at inlet temperature of 60\degree C and was compared with corresponding single phase pressure data (See Figure 4). It was observed that flow instability initiated inside the channel starting from sub-cooled to saturated flow condition. An attempt was made to study these flow
instabilities and extract information from these fluctuations to categorize various boiling regimes. Detailed description is discussed in results and discussion section.

2.2.2 Role of void fraction and vapor quality

From Figure 3b, it is indicated that various parameters such as mixture density, void fraction, vapor quality play a significant role in two-phase boiling flows i.e. diabatic flow system. While estimating gravitational, acceleration and frictional pressure drop, density variation is observed along the length due to phase change process. Mixture density needs to be calculated using void fraction and vapor quality. These two parameters are interlinked with each other and play a significant role in flow boiling phenomenon. The value of void fraction governs many important parameters such as two-phase mixture density and viscosity. The fact that pressure due to gravity dominates the total pressure in vertical channels and in turn depends upon two-phase mixture density make void fraction a very crucial aspect of present study. Woldesemayat et. al. has done an extensive study on void fraction co-relations for different flow patterns at various design conditions.

Therefore, choice of void fraction correlation is the key to quantify pressure drop along the channel. The void fraction was calculated from homogeneous, Lokhart-Martinelli model and Graham’s co-relations and it was found that the homogeneous correlation is suitable for existing experimental data. Hence the void fraction was calculated from homogeneous correlation and plotted against experimentally calculated vapor quality (using energy balance) as per the following. In order to find out vapor quality using energy balance, heat flux, surface area, mass flux and enthalpy of subcooled and saturated liquid were taken into account. The specific enthalpy of liquid and vapor at saturation was known. Based on the energy balance, the thermodynamic vapor quality $x_{th}$ at a cross section at position $Z$ can be evaluate using following equations:

\[
x_z = \frac{i_{L,z} - i_{L,sat}}{\lambda}, \quad Z < Z_{sat} \tag{5}
\]

\[
x_z = \frac{1}{\lambda} \left( \frac{q S Z}{G A_{c/s}} + i_{L,inlet} - i_{L,sat} \right), \quad Z > Z_{sat} \tag{6}
\]

Where $Z_{sat}$ was the distance between the saturation point and inlet, $G$ was the total mass flux, $q$ was the heat supplied to channel and $\lambda$ was the latent heat and $i_L$ was the specific enthalpy of liquid at inlet and saturated condition. The thermodynamic quality is the vapor quality and it is the ratio of mass of vapor to the total mass of flow that represents the true flow fraction of the vapor phase and can only have values between 0 and 1.

As said earlier, the void fraction and vapor quality are interlinked with each other. The homogeneous co-relation was found to be suitable for the current study (Figure 5). It established that the pattern for $30^\circ C$ and $60^\circ C$ inlet conditions was well in co-ordination with the existing trend available in Thome. However for $95^\circ C$ inlet due to excessive vapor formation
even at pressure sensors P1 and P2, a different pattern was observed showing void fraction of 0.6 and more all throughout the vaporizer. The pressure fluctuation data and its relation with void fraction and vapor quality was explored to extract meaningful conclusions. Further, the experimental data analysis was carried out to study effect of heat flux and mass flux on pressure drop and flow instability. Experimental data variations were “propagated” into the uncertainty of the derived quantity. Therefore, in the present work, uncertainties for the experimental HTCs, pressure drop and vapor quality were determined using “propagation method” (See Table 2)

3. Results and discussions

In two-phase flow boiling system, flow instability is of great importance while designing many industrial systems such as steam generators, boiling water reactors, re-boilers and chemical process systems etc. Flow instabilities are undesirable as it can cause mechanical vibrations as well as disturb the heat transfer characteristics of the system that may turn into burn out/dry out condition. This dry out condition lead to sudden increase in wall temperature and continual cycling on the wall temperature leads to thermal stresses in the system as per reported observations. Therefore, in the present study an attempt has been made to enhance the knowledge on the subject of these flow instabilities and corresponding pressure drop occurring inside the system and categorize flow boiling regimes using this instability data and validate pressure drop with reported models.

3.1 Flow instability

According to literature, it was observed that there are three distinct terms associated with the flow instabilities in two-phase flow boiling system viz. “density-wave oscillations”, “pressure drop oscillation” and “thermal oscillations”. It was pointed out that the pressure-drop oscillations occurred only when the pressure-drop across the test section increased with increasing flow rate and the oscillation period was governed by the volume and compressibility of the vapor in the system. In this study, pressure drop characteristics were studied and estimated experimentally by measuring pressure at four axial locations.

As mentioned earlier, the pressure fluctuation data was recorded during each experiment at P1, P2, P3, P4 for an hour while for analysis purpose this data was averaged for last 10 minutes to get steady-state data. Flow boiling pressure variations along the length at 2.9 kg/m² s and 14.3 kW/m² (mass and heat flux condition respectively) are shown in Figure 6a, Figure 6b, Figure 6c at 30, 60, 95°C inlet conditions respectively. It revealed that increasing the inlet fluid temperature led to more frequent and higher amplitude pressure fluctuations at the axial locations along the tube. Pressure fluctuations were due to the acceleration of liquid slugs by the formation and growth of individual confined bubbles. These fluctuations could be attributed to the phase change from sub cooled to saturation vapor-liquid flow. The high-frequency fluctuations cause local fluctuations in saturation temperature region where bubble nucleation, growth and coalescence drive the fluctuations. At such situations the possibilities of identifying
various flow regimes like bubbly flow, slug flow and annular flow was explored by analyzing the measured pressure fluctuations. Representation of overall instability data from single phase to two-phase (30, 60 and 95°C inlet condition) has been shown in Figure 6d. The figure indicated the subsequent decrease in pressure at different axial locations as a result of increasing nucleate and convective boiling. Dominance of nucleate and convective boiling resulted into higher and higher vapor generation which in turn led to significant flow instability. In the present work, attempt was made to categorize flow boiling regimes using this flow instability data with respect to time series data using Fast Fourier Transform (FFT) analysis.

3.2 Time series analysis

Time series is a collection or sequence of data which is measured at fixed consecutive intervals of time. Time series analysis comprises of methods of analyzing time series data to extract meaningful statistics and other characteristics of the data.

The nonlinear analysis tool was used on the time series to characterize the oscillations at varied inlet fluid conditions. Raw pressure data of all pressure sensors (i.e. for fluid) was used to study variation in flow boiling regimes. MATLAB software was used for analyzing the acquired transient pressure data. The collected time-pressure data was analyzed using the Fast Fourier Transform (FFT) technique. FFT analysis was performed on pressure fluid data to acquire the frequency distribution for different fluid temperatures at different axial locations at 2.9 kg/m² s mass flux and 14.8 kW/m² heat flux conditions.

FFT analysis makes it simpler to carry out analysis of complex signals in frequency domain. Time domain data reflected the variation of signals with respect to time whereas the frequency domain echoed the amount of signals that lie within each given frequency band. The Continuous Time Fourier Transform which transforms time domain to frequency domain is defined as:

\[ X(f) = \mathcal{F}\{x(t)\} = \int_{-\infty}^{\infty} x(t) e^{-j2\pi ft} dt \]

(7)

where \( x(t) \) is the time domain signal, and \( X(f) \) is its Fourier Transform.

Energy spectral density which governs the energy distribution of pressure signal \( x(t) \) was analyzed using following equation

\[ E = \sum_{-\infty}^{\infty} |x(t)|^2 \Delta t \]

(8)

Power spectral density which governs the energy distribution of pressure signal \( x(t) \) was analyzed using following equation

\[ P = \frac{1}{T} \sum_{-\infty}^{\infty} |x(t)|^2 \Delta t \]

(9)
Using the FFT technique on the raw pressure fluctuation data with consideration of 3600 continuous-time pressure signals per hour, the non-periodic spectrum with the sampling rate of 1Hz was obtained. The frequency oscillations in the signals of pressure were characteristic of cycles as seen in earlier figures.

For stable flow conditions (at 30°C as inlet), no significant differences in pressure oscillations was observed and so it was difficult to identify any regime pattern. However when inlet fluid temperature was 60°C and 95°C, repetitive dominant frequencies were observed that indicated the continuous vapor formation. The identification of dominant frequency was made by plotting amplitude data contained in the raw fluctuating pressure signal against its frequency. In order to substantiate the above claims, the spectra obtained for 60°C and 95°C inlet conditions were respectively compared with 30°C inlet (See Figure 7a and 7b). Figure 7a and 7b, are the comparative representation of variation in pressure signals at 30 and 60°C and at 30 and 95°C as inlet fluid temperature respectively. Such as comparative representation explicitly show cased the significant frequency variation in different flow boiling regimes.

An attempt was made to categorize the basic flow boiling regimes (bubbly, slug and annular flow, mist) using pressure signals and analyzing their corresponding amplitude spectra. Bubbly flow frequency domain showed the characteristic frequency maxima; while for slug and annular flow, the frequency peaks were observed in the vicinity of 7, 10, 15 Hz for 60°C inlet and 10, 20 Hz for 95°C inlet temperature condition respectively. In case of mist flow this amplitude decreased with indication of higher vapor quality.

3.3 Effect of heat and mass flux

The effect of heat flux is one of the crucial factors studied for the complete understanding of two-phase flow boiling phenomenon. In present experiments, the hot oil bath was used as an indirect mode of heating the process fluid (water). Water that entered the vaporizer as subcooled liquid was transformed into vapor phase under an influence of heat flux as it proceeded along the vertical channel. Once liquid received heat at initial stage, nucleate boiling dominated the early process where bubble formation, bubble growth and bubble departure closely trailed the heat transfer. Pressure fluctuations in the system were governed by the formation of bubbles on the heated surface and continuation of phase change process.

Plot of variation of pressure versus heat flux revealed that with increasing heat flux the pressure decreased for every pressure sensor. Figures 8a and 8b reflected the analysis of pressure drop and vapor quality along the axial location of the tube. It can be clearly seen that both vapor quality and pressure drop showed increasing trend along the length of the channel. It was attributed to the fact that with increasing heat flux, the produced vapors increased the vapor quality and led to further drop in pressure. As the heat flux increased, the wall superheat became significant enough to evaporate the entire liquid and the dry out conditions were imminent. One can easily make out from the Figure 8b that the 30°C and 60°C inlet conditions at 14.1 kW/m² heat flux cannot lead to dry out as vapor quality was below 0.6. However in case of 95°C inlet, the observed vapor quality (i.e. x >0.6) strongly signify possibility of dry out
or near to dry out conditions at 0.82m along the channel. Even though the pressure data was not recorded at intermediate position; the dry out conditions can be predicted with the present analysis. The minor rise in pressure drop for 95°C inlet was governed by the fact that the superheat generated was adequate enough to drive the vaporizer towards the saturated vapor phase region. In order to take into consideration the effect of heat flux; the above parameter was studied for 21.4 kW/m² heat flux conditions keeping other conditions constant (Figure 8b). It revealed that increased heat flux was adequate for dry or near to dry out conditions to take place for all three inlet conditions. Comparing the behavior of 95°C inlet for both heat flux condition; it was observed that with the increase in heat flux, the dry out regime, which for 14.1 kW/m² occurred at 0.82m, shifted to 0.72m. In order to have a comprehensive view of the effects of heat fluxes, occurrence of dry out phenomenon for all different heat fluxes was studied. Accordingly, the locations along the length of channel where the dry or near to dry out conditions occur were identified and tabulated (See Table 3).

Flow boiling regimes were classified on the groundwork of application of qualitative analysis of Collier and Thome on our experimental data at vapor quality ranging from 0 to 1 (See Figure 9a, 9b and 9c). If vapor quality is zero then flow regime corresponds to sub cooled boiling regime (as per literature). If vapor quality lies between 0 < x < 0.2 bubbly flow and if 0.2 < x < 0.6 then slug or annular flow regimes may exist. For vapor quality values above 0.6 and less than 0.8, dry out or partial dry out regime may occur. Beyond the value of 0.8, flow regime considered as mist flow. In the present case, identification of flow regimes was postulated according to dynamic flow behavior and estimated vapor quality using energy balance along with information and an understanding reported by literature. Comparative study of homogeneous model and Lokhart-Martenilli model was done with pressure drop vs vapor quality where it was found out that these model were very well suitable for slug and annular region at 2.9 kg/m² s (i.e. low mass flux condition) whereas for 5.9 kg/m² s (i.e. high mass flux condition) these model were very well predicting bubbly flow regime only.

A closer look at the Figure 7a and 8a, Figure 7b and 8b reveals a strong interlinking between them and gives more clarity for the results obtained from them. On simultaneously studying Figure 7b and 8b, frequent appearance of high magnitude peaks near P1 and P2 where vapor quality varies between 0.2 - 0.4 strongly indicated the presence of slug flow. The unsaturated conditions (low vapor formation) and low pressure drop observed near P1 and P2 further strengthened the possibility of slug flow thereby ruling out the possibility of annular or mist flow. On the similar grounds; it can be concluded that the peak observed near P3, where the vapor quality varies between 0.4 - 0.6, indicates the possibility of annular flow. Magnitude of the peak being smaller than that of slug flow further clarifies that it belongs to region where there is considerable vapor formation. Accordingly, still lower magnitude peak near P4, where high pressure drop and near saturation conditions exist, indicates the mist flow. Similarly the flow regimes can be predicted for other conditions of heat and mass flux.
3.4 Comparison of pressure drop with existing correlations

In the present work, an attempt has been made to compare acquired pressure drop data with existing conventional correlations such as homogeneous model and the separated flow model like Lokhart - Martinelli model. Comparison of predicted pressure drop with experimental pressure drop is as shown in terms of parity plot (See Figure 10a and 10b). It was found that Lockhart-Martinelli (vapor-laminar, liquid-laminar) model consistently under-predicted the experimental pressure drop data for all the mass flux conditions. At 2.9 kg/m²s, more than 70% experimental data stood under-predicted. The performance even worsened at 5.9 kg/m²s when about 90% of the data was under-predicted. The possible reason for this can be attributed to the different experimental operating conditions of mass and heat flux ranges. The homogeneous model satisfactorily predicted 50% of the experimental data in ±25% range (Figure 10a) when the mass flux was 2.9 kg/m²sec. Moreover it predicted 84% of the data well within ±25% (Figure 10b) at higher flux 5.9 kg/m²sec. It was observed that homogeneous model works best when the phases are intermixed (i.e. at high velocities). In order to substantiate the above observed phenomenon the comparative plots of performance of homogeneous correlation were presented at increasing mass flux from 2.9 kg/m²s to 5.9 kg/m²s (see Figures 10a, 10b).

4. Conclusions

The flow boiling characteristics of water (& steam) in small channel with the diameter of 19mm ID were investigated systematically. Two-phase flow instability and two-phase flow pressure drop along the axial length at different mass and heat flux condition studied in detail. Time series analysis was done using FFT for identification of flow regimes. Key conclusions are as follows:

a) Comparative study of single phase and two-phase flow pressure fluctuation data showed the occurrence of vapor at diabatic flow condition. The pressure uniformly decreased along the every axial location (i.e. from bottom to top). Vapor generation caused flow instability which was evident in flow pattern transition from stable single phase to characteristics repetitive pressure fluctuation pattern in two-phase flow boiling (See Figure 4)

b) The homogeneous void fraction plotted against the vapor quality showed the similar trend for void fraction vs vapor quality as given by Thome[14] at 30° and 60°C inlet condition. Whereas significant vapor formation (in case of 95°C inlet) caused due to high wall superheat led to high and almost constant void fraction even there was increase in vapor quality.

c) With raw pressure fluctuation and spectral analysis data; the comparison of amplitude vs frequency was done for 30° and 60°C and 30° and 95°C. At 30°C inlet conditions; since the entering sub-cooled liquid couldn't generate significant amount of vapors, no major pressure fluctuations pattern were observed along the channel. Consequently the amplitude spectra couldn't identify the dominant frequency.
d) In 30°C and 60°C comparative study, at 60°C inlet conditions the low frequency and high amplitude peaks and correspondingly the vapor quality below 0.2 showed the existences of bubbly flow regime. Similarly the occurrence of high frequency, low amplitude peaks (near 7, 10, 15 Hz) and vapor quality between 0.2 - 0.6 strongly indicated the occurrence of slug regime around sensor P2 and annular regime around P3 (See Figure 7a and Figure 8a). When the inlet water temperature was 95°C, slug (P1, P2), annular (P3) and mist (P4) flow regimes were identified (near 10, 20 Hz) on similar grounds (See Figure 7b and Figure 8b).

e) The pressure drop and vapor quality increased along the axial length of the channel whereas dry out was observed for 0.83m for 14.1 kW/m² to 0.71m for 21.4 kW/m². Thus with increasing heat flux, the shift of dry out location to lower height was observed.

f) Homogeneous correlation predicted 50% the experimental pressure drop data well within ±25% for 2.9 kg/m²sec as mass flux and better performance was observed for higher mass flux of 5.9 kg/m²sec with prediction of 84% of data.

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Nomenclature

z           Height
x           Vapor Quality
v    Velocity
P            Pressure
ΔP          Pressure Drop
T           Temperature
d           Diameter
Re         Reynolds Number
C          Chisholm parameter
f  Single Phase Frictional Factor
q  Heat Flux
A   Flow area
\(A_r\)  Flow area ratio
\(A_0\)  Smaller flow area

Subscripts
    a       Acceleration
    g       Gravitational
    f       frictional
    f       frictional factor
    m       Mixture
    L       Liquid
    G       Gas
    \(G\)  Flow rate
    g       Gravitational constant
    \(t_p\) Two-phase
    h       Hydraulic

Greek Letters
    \(\varepsilon\) Void Fraction
    e       Surface Roughness
    \(\Phi\) Two-Phase Multiplier
    \(\rho\) Density , kg/m\(^3\)
    \(\mu\) Dynamic Viscosity, kg/m.s
References


<table>
<thead>
<tr>
<th>Author</th>
<th>Geometry</th>
<th>Operating Conditions</th>
<th>Parameter Study</th>
<th>Method/approach</th>
<th>Remarks/Understanding</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vijayarangan et al. 10</td>
<td>Vertical TS</td>
<td>Working Fluid: (R-134a) Heat Flux: 35-80 kW/m² Mass Flux: 1200-2000 kg/m²/s Exit Vapor Quality: 0.19-0.81</td>
<td>ΔP decreases as the system pressure is increased</td>
<td>Conventional homogenous and separated flow approach and flow pattern based method for identifying flow regimes was used.</td>
<td>Various flow regimes such as bubbly, slug and annular, three other flow regimes, viz., single phase liquid, post-dryout and single phase vapour were identified using flow pattern based approach.</td>
</tr>
<tr>
<td>Qi et al. 11</td>
<td>Vertical TS</td>
<td>Working fluid: Refrigerants Heat Flux: 5-21 W/m² Mass Flux: 440-3000 kg/m²sec</td>
<td>At ONB, mass flux drops suddenly while ΔP increases. Small mass flux causes vapor generated in the micro-tube to be discharged freely, no vapor patch formation at outlet and so stable flow boiling.</td>
<td>The adiabatic and diabatic two-phase flow pressure drop characteristics in micro-tubes are investigated. The three ΔP components were obtained by the homogeneous, L-M, Chisholm B coefficient and Friedel model, respectively.</td>
<td>Homogenous co-relation generally under predicts the actual pressure drop. While in case of separated flow models like L-M, Chisholm, and Friedel the effects of heat and mass fluxes are not directly considered.</td>
</tr>
<tr>
<td>Bhide et al. 12</td>
<td>Horizontal TS</td>
<td>Heat Flux: 0-1120 kW/m² Mass Flux: 280-1410 kg/m²sec Exit Vapor Quality: 0.6</td>
<td>The onset of two-phase flow increases ΔP which keeps on increasing with further heating. Acceleration ΔP increases with an increase in the exit quality.</td>
<td>Experiments conducted in both smooth and rough micro channels. HM model, Separated flow model like L-M (liquid–laminar, vapour–turbulent), Friedel used along with L-M (liquid–laminar, vapour–laminar used to correctly predict ΔP.</td>
<td>HM used for simplicity. L-M predict the two-phase pressure drop data within 20%. Micro-channels of smaller hydraulic diameter have lesser instabilities. Frictional ΔP is the dominant component.</td>
</tr>
<tr>
<td>Choi 13</td>
<td>Horizontal TS</td>
<td>Liquid superficial velocities of 0.06-1.0 m/s Gas superficial velocities of 0.06-72 m/s Liquid mass flux - 66-1000 kg/m²/sec Vapor mass flux 0.075-80 kg/m²/sec Adiabatic two-phase flow</td>
<td>ΔP decreased as gas superficial velocity decreased. ΔP increased as gas superficial velocity increased.</td>
<td>ΔP measured through embedded ports. Flow pattern visualized using a high-speed camera. Seven two-phase HFM viscosity models and ten SFM correlations used.</td>
<td>The typical trend of ΔP is as follows Region I: A bubble flow regime including bubbly, slug bubble and elongated bubble flow patterns. Region II: Transition flow patterns. Region III: Liquid ring flow. Best two-phase viscosity model: Beattie and Whalley (1982). Best correlation: Lee and Mudawar (2005).</td>
</tr>
<tr>
<td>Thome et al. 14</td>
<td>Horizontal TS</td>
<td>Working fluid R-134a and R-245fa</td>
<td>Higher the saturation temperature, the lower is the two-phase frictional ΔP.</td>
<td>Adiabatic two-phase frictional ΔP calculated.</td>
<td>Frictional ΔP studied for Reynolds range (Laminar, transition and turbulent)</td>
</tr>
<tr>
<td>Author</td>
<td>Geometry</td>
<td>Operating Conditions</td>
<td>Parameter Study</td>
<td>Method/approach</td>
<td>Remarks/Understanding</td>
</tr>
<tr>
<td>-------------------</td>
<td>-----------------------------</td>
<td>--------------------------------</td>
<td>---------------------------------------------------------------------------------</td>
<td>----------------------------------------------------------------------------------</td>
<td>----------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Wongwises et. al. 15</td>
<td>Concentric annular</td>
<td>Air-water system</td>
<td>At low fluids velocity, ΔP increases as inclination increases from 0° to 60°. ε increases with increasing gas velocity and decreases with increasing liquid velocity.</td>
<td>ΔP calculated experimentally using differential pressure transducer and the variation in ΔP explained on the basis of flow patterns observed. Flow phenomena observed by high speed camera.</td>
<td>For inclination =0, the homogeneous flow model (HEM) gave the overall agreement. As inclination angle is increased onset of transition from PF to SF region (horizontal) &amp; from PF to S/BF flow region (inclined channels) shift to a lower value of superficial air velocity.</td>
</tr>
<tr>
<td>Godbole et. al. 8</td>
<td>Vertical TS</td>
<td>Operating pressure: 114–260kPa.</td>
<td>Annular flow is observed when the gas flow rate becomes sufficiently high.</td>
<td>Performance of 52 void fraction correlations and experimental data verified. Flow predicted theoretically using co-relations based on superficial flow velocities and cross checked with observed.</td>
<td>The distinctive major flow patterns observed in the upward vertical two-phase flow are dispersed bubble, slug, churn/froth, and annular.</td>
</tr>
<tr>
<td>Thome et. al. 17</td>
<td>Horizontal TS</td>
<td>R-134a, R-123, R-402A, R-404A and R-502.</td>
<td>Chisholm method significantly over predict the experimental values.</td>
<td>2 phase frictional ΔP calculated using co-relations and vapor quality data: Friedel, L-M, Gronnerud, Chisholm, Bankoff, Chawla Muller-Steinhagen and Heck</td>
<td>Gronnerud predicts these data accurately. The peak in the experimental two-phase frictional ΔP was observed to coincide with the onset of dryout in annular flows at high vapor qualities, similar to the equivalent peak in the flow boiling htc.</td>
</tr>
<tr>
<td>Palm et. al. 18</td>
<td>Circular vertical quartz</td>
<td>Working fluid: R134a</td>
<td>At low mass flux the location of the liquid front fluctuated with waves whereas at high mass flux the location of the liquid front was more stable.</td>
<td>Flow regime visualization using high speed camera.</td>
<td>Heat transfer at vapor qualities above 0.6 is fairly low.</td>
</tr>
</tbody>
</table>

L-M - Lockhart and Martinelli  
TS - Test Section  
HM - Homogeneous  
htc - heat transfer coefficient
Table 2: Sample calculations for uncertainties of measured quantities and calculated parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
<th>% Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surface Area</td>
<td>m²</td>
<td>0.081</td>
<td>3.6 %</td>
</tr>
<tr>
<td>Mass flow rate of water</td>
<td>Kg/s</td>
<td>$8.2 \times 10^{-4}$</td>
<td>1.0 %</td>
</tr>
<tr>
<td>Mass flow rate of Hot oil</td>
<td>Kg/s</td>
<td>$7.0 \times 10^{-2}$</td>
<td>2.3 %</td>
</tr>
<tr>
<td>Heat supplied to test section</td>
<td>W</td>
<td>$1.8 \times 10^{3}$</td>
<td>9.6 %</td>
</tr>
<tr>
<td>Heat loss from the test section</td>
<td>W</td>
<td>43.3</td>
<td>6.8 %</td>
</tr>
<tr>
<td>$T_w - T_{sat}$</td>
<td>°C</td>
<td>53.89</td>
<td>2.62 %</td>
</tr>
<tr>
<td>Heat flux</td>
<td>W/m²</td>
<td>$2.2 \times 10^{4}$</td>
<td>12.4 %</td>
</tr>
<tr>
<td>Local HTC</td>
<td>W/m² K</td>
<td>389.7</td>
<td>12.65 %</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>mbar</td>
<td>45</td>
<td>1.2 %</td>
</tr>
<tr>
<td>Vapor quality</td>
<td></td>
<td>0.51</td>
<td>5.55 %</td>
</tr>
</tbody>
</table>
Table 3: The location of occurrence of dry out along the length of channel for 2.9 kg/m²s

<table>
<thead>
<tr>
<th>q’ (kW/m²)</th>
<th>T_in</th>
<th>30° Inlet</th>
<th>60° Inlet</th>
<th>95° Inlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>9.01</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.97m</td>
</tr>
<tr>
<td>14.1</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.83m</td>
</tr>
<tr>
<td>21.3</td>
<td>0.82m</td>
<td>0.85m</td>
<td>0.71m</td>
<td></td>
</tr>
<tr>
<td>27.8</td>
<td>0.65m</td>
<td>0.72m</td>
<td>0.60m</td>
<td></td>
</tr>
</tbody>
</table>
Figure 1a: Schematic of experimental set up
Figure 1b: Schematic of locations of pressure sensors
Figure 2: Frictional factor $f$ versus Re plot for single phase

Frictional Factor, $f$

Reynolds Number, Re

Single Phase

Experimental

$\cdots f = 16/Re$
Two-phase pressure drop

Frictional pressure drop

ΔP occurs all along the length
Equation is applicable for single phase and homogeneous two-phase flows

\[ \Delta P_f = \frac{2 f_w L V^2 \rho_w}{d g} \]

Acceleration ΔP

Caused by change in flow area or density

\[ \Delta P_a = \frac{(1 - A_r^2) G^2 \phi}{2 A_o^2 \rho_L} \]

\( \phi = 1 \) for single phase

\( \phi = \left( \frac{x^3}{\rho_G^2 \alpha^2} + \frac{(1-x)^3}{\rho_L^2 (1-\alpha)^2} \right) \left( \frac{\rho_G \rho_L}{x \rho_L + (1-x) \rho_G} \right) \) for two phase

Elevation/static ΔP

In vertical test section, elevation ΔP is the largest component

\[ \Delta P = \rho g \Delta z \cos \theta \]

for two phase flows

\( \rho_{TPF} = \rho_L (1-\varepsilon) + \rho_G \varepsilon \)

Accurate prediction of void fraction is necessary which again calls for judicious selection of void fraction correlation

Correlation for void fraction needs to be chosen judiciously

Figure 3a: Total pressure drop in two-phase flow boiling Sardeshpande and Ranade\textsuperscript{20}
Acceleration Pressure Drop
\[ P_{tpa} = G^2 \left( \frac{1}{(\rho_m)_{out}} - \frac{1}{(\rho_m)_{in}} \right) \] or
\[ \Delta P_a = G^2 \Delta z \left( x^2 \frac{\rho_L}{\rho_G} + \frac{(1-x)^2}{1-e} - 1 \right) \]

Frictional Pressure Drop
Homogenous Flow Approach
\[ P_{tpf} = \frac{2 f_{tpf}^2 \varepsilon_f}{d \rho_f} \left[ 1 + \frac{f_{tpf}^2}{2} \left( \frac{\rho_f}{\rho_g} \right) \right] \]
Where \( f_{tpf} = 0.003 \)
Separated Flow Approach
\[ \Delta P_{tpf} = \Delta P_L \Phi_f^2 \]

Gravitational Pressure Drop
\[ \Delta P_{tpg} = (\rho_m g z)_{i} - (\rho_m g z)_{o} \]

Total Pressure Drop
\[ \Delta P_{tp} = \Delta P_{tpg} + \Delta P_{tpf} + \Delta P_{tpa} \]

Mixture Density
\[ \rho_m = (1-\varepsilon) \rho_L + \varepsilon \rho_g \]

Homogeneous
\[ \varepsilon = [1 + (\frac{1-x}{x}) \left( \frac{\rho_g}{\rho_L} \right)^{0.5}]^{-1} \]
Lockhart-Martinelli\textsuperscript{21}
\[ \varepsilon = [1 + 0.28 (\frac{x}{1-x})^{0.64} \left( \frac{\rho_g}{\rho_L} \right)^{0.36} \left( \frac{\mu_L}{\mu_g} \right)^{0.07}]^{-1} \]
Fauske\textsuperscript{22}
\[ \varepsilon = [1 + \left( \frac{1-x}{x} \right) \left( \frac{\rho_g}{\rho_L} \right)^{0.5}]^{-1} \]
Chisholm\textsuperscript{23}
\[ \varepsilon = [1 + \left( \frac{1-x}{x} \right) \left( \frac{\rho_g}{\rho_L} \right)^{0.5}]^{-1} \]
Spedding and Chen\textsuperscript{24}
\[ \varepsilon = [1 + 2.22 (\frac{x}{1-x})^{0.65} \left( \frac{\rho_g}{\rho_L} \right)^{0.65}]^{-1} \]
Graham et. al.\textsuperscript{25}
\[ \varepsilon = [1 + \left( \frac{1}{F_t} \right) + \left( \frac{1}{X_{tt}} \right)]^{-0.321} \]
where \( F_t = \left( \frac{G^2 x}{(1-x) \rho_g \rho_L D} \right)^{0.5} \) and
\[ X_{tt} = \left( \frac{\mu_L}{\mu_g} \right)^{0.1} \left( \frac{x}{1-x} \right)^{0.9} \left( \frac{G^2}{\rho_L} \right)^{0.5} \]

Re (Liq only) = \( \frac{G \times (1-x) D h}{\mu_L} \)
Re (Gas Only) = \( \frac{G \times x \times D h}{\mu_G} \)

Single Phase Frictional Pressure Drop
\[ \Delta P_f (fo) = \frac{2 f L \sigma_f^2 (1-x^2) \Delta z}{d_h \rho_L} \]

\[ f = 64/Re \; \text{for } Re < 2000 \]
\[ \frac{1}{f} = 0.86 \ln \left( \frac{e}{3.7 D} + \frac{2.51}{Re \sqrt{f}} \right) \; \text{for } Re > 3000 \]
\[ f = 0.079/Re^{0.25} \; \text{ (Diabatic study by LM)} \]

Lockhart Martinelli (liquid–laminar, vapour–laminar)
\[ \Phi_L^2 = 1 + \frac{5}{x_{vv}} + \frac{1}{x_{vv}^2} \]

Lockhart Martinelli (liquid–laminar, vapour–turbulent)
\[ \Phi_L^2 = 1 + \frac{12}{x_{vv}} + \frac{1}{x_{vv}^2} \]

Friedel\textsuperscript{26}
\[ \Phi_L^2 = A_1 + \frac{3.34 A_2 A_3}{Fr_H^{0.045} We_H^{0.035}} \]
where
\[ Fr_H = \frac{G^2}{\mu_L h \rho_L} \] and \[ We_H = \frac{G^2 d_h}{\sigma \rho_h} \]

Chisholm B coefficient model\textsuperscript{27}
\[ \Phi_L^2 = 1 + (r^2 - 1) \left[ B x^{0.075} (1 - x^{0.075}) + x^{1.75} \right] \]
where \( r^2 = \frac{\mu_L}{\mu_G} \left( \frac{\rho_L}{\rho_G} \right) \)
B = 55/G\textsuperscript{0.5} for 0 < r < 9.5;
B = 520/G\textsuperscript{0.5} for 9.5 < r < 28;
B = 15000/r\textsuperscript{2} G\textsuperscript{0.5} for r < 28

Figure 3b: Schematic of overall two-phase flow boiling pressure drop
Figure 4: Raw data of two-phase pressure fluctuations at four axial locations (2.9 kg/m²·s and 14.1 kW/m² and 60°C as inlet) compared with that of single phase at 2.9 kg/m²·s.
Figure 5: Variation of void fraction with vapor quality for different inlet temperatures at 2.9 kg/m²sec and 14.1 kW/m².
Figure 6a: Study of pressure fluctuations at 30° inlet.

Flow rate: 2.9 kg/m²sec
Heat Flux: 14.1 kW/m²
Inlet: 30°C
Flow rate: 2.9 kg/m$^2$sec
Heat Flux: 14.1 kW/m$^2$
Inlet: 60$^\circ$C

Figure 6b: Study of pressure fluctuations at 60$^\circ$ inlet
Flow rate: 2.9 kg/m²·sec
Heat Flux: 14.1 kW/m²
Inlet: 95°C

Figure 6c: Study of pressure fluctuations at 95°C inlet
Figure 6d: Single phase and two-phase pressure variation at each pressure sensors for different inlet conditions to study the effect of heat flux.
Figure 7a: Flow regime identification using time series analysis for 30° and 60°C inlet at 2.9 kg/m²·sec and 14.1 kW/m².
Figure 7b: Flow regime identification using time series analysis for $30^\circ$ and $95^\circ$C inlet at 2.9 kg/m$^2$sec and 14.1 kW/m$^2$
Figure 8a: Variation of experimental pressure drop and vapor quality along the vaporizer length for different inlet conditions at 2.9 kg/m²s mass flux and 14.1 kW/m² heat flux.
Figure 8b: Variation of experimental pressure drop and vapor quality along the vaporizer length for different inlet conditions at 2.9 kg/m² s mass flux and 21.3 kW/m² heat flux.
Figure 9a: Regime identification using pressure drop versus vapor quality at various heat flux with 2.9 kg/m²sec flow rate.
Figure 9b: Regime identification using pressure drop versus vapor quality at various heat flux with 3.9 kg/m²sec flow rate.
Figure 9c: Regime identification using pressure drop versus vapor quality at various heat flux with 5.9 kg/m²sec flow rate.
Figure 10a: Performance of homogeneous correlation at flow rate of 2.9 kg/m²s
Figure 10b: Performance of homogeneous correlation at flow rate of 5.9 kg/m²sec