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Abstract

Wet central heating systems are used throughout the world and the domestic variants are known to account for 16% of the carbon dioxide emissions in the UK. It is an industry wide fact that significant quantities of air leak into the system's closed loop circuit, mostly due to thermal cycling during routine use. The water in the system results in high dissolved gas saturation levels thus resulting in micro bubble nucleation and detachment at the boiler wall. An in depth knowledge into the behaviour of micro bubbles in such systems could result in better deaeration systems thus improving the overall system efficiency. In this study, we will report a measurement of the bubble detachment sizes in this system at different operating conditions, these being, the system flow rate, pressure, heating load and saturation ratio. The Winterton models for zero and finite contact angles for the prediction of the bubble detachment diameters in supersaturated solutions were used as a comparison with the test results. The model for the prediction of bubble detachment diameters at finite contact angles in round pipes has yielded good predictions. A new correlation has been proposed so as to include the effects of pressure and heat flux on the bubble characteristics which were not represented through the Winterton model.

Keywords:

Central heating systems, Micro bubble formation, Dissolved gases in water.

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Measurement of bubble detachment diameters from the surface of the boiler heat exchanger in a domestic central heating system

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Wet central heating systems are used throughout the world and the domestic variants are known to account for 16% of the carbon dioxide emissions in the UK. It is an industry wide fact that significant quantities of air leak into the system's closed loop circuit, mostly due to thermal cycling during routine use. The water in the system results in high dissolved gas saturation levels thus resulting in micro bubble nucleation and detachment at the boiler wall. An in depth knowledge into the behaviour of micro bubbles in such systems could result in better deaeration systems thus improving the overall system efficiency. In this study, we will report a measurement of the bubble detachment sizes in this system at different operating conditions, these being, the system flow rate, pressure, heating load and saturation ratio. The Winterton models for zero and finite contact angles for the prediction of the bubble detachment diameters in supersaturated solutions were used as a comparison with the test results. The model for the prediction of bubble detachment diameters at finite contact angles in round pipes has yielded good predictions. A new correlation has been proposed so as to include the effects of pressure and heat flux on the bubble characteristics which were not represented through the Winterton model.

1. Introduction

The nucleation and subsequent detachment and flow of bubbles in a system is a process that effects a wide spectrum of industries, often with undesirable results. As originally reported by Dean [1], the formation of bubbles finds its origins in the presence of either a supersaturated or a superheated liquid. A number of recent studies have published data on the subject of bubble formation in supersaturated conditions and turbulent bubbly flow in ducts. However, in spite of the inherent importance of this subject in relation to heating systems, the authors of this study are not aware of published data relating to bubble characteristics observed in wet domestic central heating systems.

In a domestic central heating system, micro bubble formation finds its origins in gas supersaturation levels thus resulting in microbubbles nucleating on the boiler wall primary heat exchanger. The use of the latter term for a standard domestic central heating unit may suggest that some form of boiling takes place in the system's primary heat exchanger, consequently leading to the formation of bubbles through superheat. However, under no operating conditions does the phenomenon of flow boiling or sub cooled flow boiling take place in a domestic central heating unit. Therefore, micro bubble nucleation is solely attributed to the presence of gas supersaturation levels in the water flowing in the system's closed loop circuit. Localised high supersaturation levels are present on the boiler wall due to the gas temperature being equal to the boiler wall temperature. Air is mostly absorbed in the system during the cold cycle. At low temperatures, water can absorb the highest quantity of dissolved gasses [2,3]. In most systems this occurs during night time when the system's boiler shuts off. The subsequent morning restart would consequently result in high saturation ratios.

The detachment of microbubbles into the system results in a bubbly flow in the circuit's flow line. Bubbly two-phase flow is characterised by the presence of bubbles of maximum size

much less than the containing vessel or duct. The bubbles are dispersed in a continuous liquid phase[4]. From our test results, no micro bubbles were observed on the return line to the boiler, thus suggesting that micro bubbles dissolve, are deaerated or rise to high points in radiators or vessels while flowing through the system. The analogy between the present study and theories developed for the prediction of bubble diameters in two-phase fully developed turbulent bubbly flow in ducts extended this study to the consideration of the two known models for predicting bubble sizes, these being the Hinze[5] approach as adapted by Winterton and Munaweera[4], and the Winterton and Orby [6] approach. However, these studies are based on experimental data obtained through the artificial insertion of micro bubbles in the flow and consequently were not considered as relevant to the present study.

The present paper is solely concerned in reporting data and proposing a new correlation for the average bubble diameter observed in a domestic central heating system at the exit point of the boiler. From a practical point of view, in the heating industry, such data is important as a good knowledge of bubble sizes is essential for an efficient passive deaeration process. Deaeration is an important feature of such systems as bubbles could accumulate in radiators and result in cold spots, thus reducing the heat transfer area of the radiator. Bubbles are also known to result in unwanted noise, blockages and corrosion issues in such systems.

Although previous studies in bubbly flow may include the use of de-ionised water in order to prevent contamination [4], the use of such water was thought to be irrelevant for the requirements of this study. This approach was adopted as systems are normally operated through the use of standard quality tap water. Consequently, such water was used during all the experiments in this study. Due to the bubble formation through super saturation levels, a categorisation of the dissolved gasses present in the water was done using standard dissolved oxygen and hydrogen sensors. The use of photographic techniques has been made for the work described in this paper.

Although no experimental data and modelling of the predicted nucleation levels in wet central heating systems is included in this paper, such work is planned to be undertaken during future studies at Brunel University. The use of different heat exchanger geometries and surface finish quality was considered for this study. However, this has proved to be very difficult as a result of the domestic boiler architecture and set up. Therefore, this study will make use of a boiler equipped with a heat exchanger design that is widely used in modern condensing boilers and will consider the main four parameters controlling such a system, these being the pressure, system flow rate, heating load and saturation ratio.

2. Calculation of bubble diameters

Bubbles are formed in a solution under super saturation conditions when the gas pressure in a nucleating bubble on the primary heat exchanger surface exceeds the system pressure, thus enabling bubble growth followed by detachment from its nucleating point. Dean [1] defines this process through Eqs. (1) and (2).

$$\Delta p = \frac{2\sigma}{r} \quad (1)$$

$$\Delta p = p_g + p_v - p \quad (2)$$

The theoretical approach that exists for predicting the detachment size of bubbles originating in a supersaturated solution with no knowledge of the nucleating time is based on the resolution of the forces acting on the surface of the nucleating bubble on the boiler heat exchanger wall, parallel to the wall surface. Winterton's [7] approach for the prediction of bubble detachment diameters is based on this theory. More models have been developed for predicting the bubble detachment diameters under stagnant conditions, such as the approach originally suggested by Scriven [8] and later adapted by Jones et al. [9] that are based on the symmetric phase growth controlled by heat and mass transfer. Other models such as that by Akiyama and Tachibana [10] were developed for predicting bubble detachment diameters in boiling conditions. Such models were developed for modelling bubble nucleation and require a comprehensive knowledge of the surface conditions and the bubble nucleating time. The lack of knowledge with respect to the bubble nucleating time in an enclosed shell type heat exchanger tube section made this model unusable in the present study.

As illustrated in Fig. (1), there are three forces acting on a bubble nucleating in a flow system, these being the drag F_d , surface tension F_s that is split into the horizontal and vertical components (F_{sx} , F_{sy}) and the buoyancy forces F_b . Hence, the drag force tends to pull the bubble off the surface into the flow whereas the surface tension force keeps the bubble attached to its surface nucleation point. Winterton [7] reports that the pressure gradient along the channel results in another force acting on the bubble. However, at low flow rates, most of the pressure gradient will be due to the gravitational force. Hence, the pressure gradient force is assumed to be equal to the normal buoyancy force. In vertical pipe flow the buoyancy force could assist or oppose the surface tension force depending on the direction of flow whereas in horizontal flow it does not result in a force component in the flow direction. Winterton [7] does not take into consideration the buoyancy force in his bubble detachment model as this is proportional to the bubble volume and hence is considered to be negligible for small bubbles in high liquid velocities.

Figure 1: Forces acting on a nucleating bubble in vertical and horizontal fluid flow.

Hence, Winterton's [7] bubble detachment model is based on the knowledge that bubbles break away from the surface into the flow when the drag force F_d equals the surface tension force parallel to the tube surface (F_{sx} or F_{sy}). For zero and finite contact angles, a balance of the forces acting parallel to the tube surface results in equations for the prediction of the bubble detachment diameters. For the experimental conditions of the present study, where the bubbles are projected into the transition region of the flow, Winterton [7] derived Eq.(3) for predicting the bubble detachment diameters at a zero contact angle. The transition region was identified through the use of Eq. (4) to calculate the velocity profile in the channel at one bubble radius from the boiler wall through the dimensionless value η . After substituting the calculated bubble radius using Eq.(3) and using the experimental velocity range of 0.3 to 0.8 m/s, all η values resulted to be between 5 and 30, thus suggesting a bubble projection into the transition flow region.

$$\frac{r}{R} = 1.2 \text{Re}^{0.12} \left[\frac{\frac{r'}{R}}{We} \right]^{0.5} \quad (3)$$

$$\eta = \frac{rv_*}{v} \quad (4)$$

The friction velocity v_* was calculated through the equation $v_*^2 = 0.0396u_*^2 Re^{-0.25}$ where the Reynolds number is based on the channel hydraulic diameter. The nucleation cavity radius, r' , was calculated through the use of Eq. (1) which represents the excess pressure required for the growth of a bubble in its nucleating cavity. For a finite contact angle Winterton[7] derived Eq.(5) for the experimental conditions of the present study, where $F(\theta) = \sin\theta_o(\cos\theta_o - \cos\theta_a)$ and $We = (\rho u^2 D_h)/\sigma$. To calculate the net surface tension force holding the bubble to the wall, Winterton assumed that the contact angle is different on each side of the bubble thus introducing the concept of the dynamic and static contact angles as illustrated in Fig. (2). Similarly, the transition flow region was identified through the application of Eq.(4).

$$\frac{r}{R} = 1.4 \left[\frac{F(\theta)}{We} \right] Re^{0.24} \quad (5)$$

Figure 2: Static and dynamic contact angles on a nucleating bubble. (Sectional view through bubble)

Limited consideration has been given in literature to the expected dynamic contact angles with surface and fluid flow conditions. Most literature sources provide details as to the expected static contact angle with surface material. Static contact angles are calculated through the ratio of the measured contact area and bubble diameter for bubbles whose spherical shape has not been distorted by the flow. As reported by Ponter and Yekta-Fard[11] static contact angles are dependent on the bubble diameter and surface conditions. Hence as no data is available to predict the dynamic contact angles on the heat exchanger stainless steel tube surface, the extreme cases were considered, these being $\theta_o = 30^\circ$, $\theta_r = 20^\circ$ and $\theta_a = 40^\circ$ [12].

The equivalent hydraulic diameter for the rectangular boiler tubes, calculated through Eq. (6) was used to calculate the tube radius R in Eqs.(3) and (5). As reported by Hesselgreaves, [13] the hydraulic diameter is calculated through the consideration of the cross sectional area and the wetted perimeter of the rectangular heat exchanger tubes illustrated in Fig. (3). The application of the hydraulic diameter enables the rectangular section to be modelled as a circular tube section.

$$D_h = \frac{4A}{P} \quad (6)$$

Domestic central heating systems make use of tubular primary heat exchangers. Helical structures are commonly found in most modern systems due to the improved space efficiency and the possibility of incorporating the condenser with the primary heat exchanger. [14] The primary heat exchanger design (Giannoni) used in the current study consists of 12 rectangular tubes coiled around the boiler burner in a helical structure. The first 4 tubes at the return end are compartmentalised into a condenser where the flue gases from the gas burner condense on the cold tube surfaces, releasing latent heat, consequently resulting in higher efficiencies. At the inlet, the system mass flow rate is split in half and channelled into two parallel tubes in the helical structure. Therefore, the system mass flow rate is equal to

twice that observed in the boiler tubes.

Figure 3: Primary heat exchanger assembly (frontal view) and rectangular tube sectional diagram

The experimental conditions of the present study did not allow a direct measurement of the bubble diameters at detachment from the boiler wall. However, the positioning of the sight glass at the exit end of the boiler with sufficient insulation to ensure isothermal conditions, allowed the authors of the present study to assume that the observed bubble diameters are reasonably equal to the detachment diameters, thus enabling the comparison of the Winterton [7] models for bubble detachment diameters at zero and finite contact angles. Hence, the effects of bubble coalescence and dissolution were assumed to be negligible due to the limited distance travelled by the bubbles and due to the isothermal conditions.

A number of recent studies have adapted the use of the Winterton[7] model with good results. Amongst these are studies done by Al-Hayes and Winterton[15,16] and Hepworth et al. [17]. Al-Hayes and Winterton adapted the original Winterton[7] model to include the effect of liquid motion on the bubble growth rate due to the change in the mass transfer coefficient for the gas entering the bubble from the bulk liquid. Their final approach is similar to the original Scriven [8] and Jones et al. [18] models and thus requires a comprehensive knowledge of the bubble nucleating time.

3. Experimental set-up and technique used

A test rig has been built to investigate the phenomenon of micro bubble formation in a domestic central heating system. A schematic diagram of the experimental set up is shown in Fig.(4). The test rig consists of a Vaillant eco TEC pro 24 condensing boiler that is connected to 22mm copper tubing that supplies a radiator and a buffer vessel. A condensing boiler is used as this is mandatory equipment for new buildings in most European Union member states[19]. Seven stainless steel sheathed K-type thermocouples are used to measure the fluid temperatures along the circuit. Four K-type thermocouples monitor the boiler wall temperature and three pressure transducers monitor the system pressure. A fourth pressure transducer monitors the dissolved gas partial pressure in combination with a semi permeable silicone membrane. The partial gas pressure monitoring system requires the water to be cooled to a temperature between 20 and 45 °C. Hence, as illustrated in Fig.(4), a tap water cooling heat exchanger was used to cool the system water to a lower temperature.

Figure 4: Schematic diagram of experimental set up.

The system fluid flow rate is monitored through an Electromag 500 Series electromagnetic flow meter. A National Instruments cDAQ-9172 chassis and relevant data modules receive all the signals from the transducers, thermocouples and electromagnetic flow meter. The signals are managed through the use of a block diagram set up on Lab View which transfers saved data to excel files. The thermocouples and pressure transducers and the Lab View system were calibrated using standard calibrating equipment.

The four system parameters are controlled as follows (Table 1):

- i The system flow rate or velocity is varied through the use of a ball valve on the supply line. The velocity in the boiler tubes was stepped between a minimum 0.3 m/s and a maximum of 0.8 m/s. This is equal to a system flow rate ranging from 4.5 to 12.5 L/m.

ii The system pressure is set through the use of a nitrogen gas cylinder connected to a standard cylinder regulator. As shown in Fig.(4) this was done through a one way valve at the top of the radiator. The system pressure was stepped between 2 and 3.75 Bars, the latter being the maximum pressure specification for a standard domestic heating system. Nitrogen was used as it is known to be the predominant dissolved gas in wet central heating systems.

iii The system heating load was varied between a minimum of 7.5 kW and a maximum of 21.5 kW through the step increase in the boiler flame settings. The return temperature was maintained constant through the use of the magnetic tap. As shown in Fig.(4), this tap is connected to the tap water mains supply line. The heating load is equal to a heat flux on the heat exchanger tubes ranging between 17 to 50 kW/m².

iv Low saturation ratios were achieved through the sudden release in pressure followed by a subsequent re-pressurisation and high ratios were achieved through low temperature night cycling. Saturation ratios ranging from 0.85 and 1.20, as defined by Jones et al.[9], were achieved.

Table 1: Experimental parameters.

Standard central heating systems make use of untreated steel radiators and copper piping. The untreated radiators result in a limited amount of oxidation due to the dissolved oxygen present in the tap water. The oxidation process releases iron oxide and some hydrogen gas. The analysis of dissolved gases through the use of an Orbisphere 3655 oxygen sensor resulted in a very low concentration of oxygen present in its dissolved form. This is in line with the findings of studies done in industry through long term experimentation with domestic central heating test rigs. In fact, oxygen, methane and carbon dioxide concentrations were found to make up to 3% of the volume of the gases present at the top of a radiator. Hydrogen was found to make up to 6.5 % of the total volume of gasses. [20] Therefore, Nitrogen is evidently the dominant gas and its properties were used for the dissolved gas properties in this study. The total gas pressure was calculated by subtracting the vapour pressure from the gas transducer reading as defined by Lubetkin and Blackwell [21]. The gas concentration was calculated using nitrogen gas solubility data as provided by Battino[22] and through the use of the Henry's law in Eq.(7) as defined by Gerrard [2].

$$C_{gas} = p_g X^T \quad (7)$$

The saturation ratio was calculated through the use of Eq.(8) as defined by Jones et al.[9] where C_{sat} refers to the maximum gas solubility at the boiler wall temperature.

$$\alpha = \frac{C_{gas}}{C_{sat}} \quad (8)$$

A square sight glass with internal dimensions of 20x20 mm was used for filming micro bubbles at the exit line of the boiler. As illustrated in Fig.(5), a square section was designed to reduce the distortion as a result of viewing bubbles through a curved surface. As discussed by Prodanovic et al.[23], such distortions are due to light refraction. A Vision research Phantom V5 high speed camera connected to a PC was used to film and store the video clips as illustrated in Fig.(6). A monozoom(Navitar) microscope lens was used to develop the desired magnification. A shutter speed of 30 μ s and a frame speed of 100 frames per second were used. Lighting was provided by two high intensity Everest.VIT ELSV 60 W light sources attached to semi rigid fibre optic light guides. The system was calibrated

using a number of standard sized gauges and subsequent scaling. A frame size of 5.62x5.62 mm was used and the depth of field was limited to approximately 1.5 mm.

Figure 5: Bubble size measurement equipment.

Figure 6: Actual experimental set up.

The video films were converted to image frames using the camera software. Typical frames generated in the present study are illustrated in Fig.(7). The calculation of bubble diameters was done manually using the Phantom Version 606 Image Analysis software. This was done after considering the use of a number of macro based automated image analysis software packages. Such software systems did not yield precise results with respect to bubble diameters. Therefore, a manual analysis is considered to be more precise for the analysis of the arithmetic mean of bubble diameters.

Figure 7: Typical camera image.

3.1 Estimate of Errors

The main errors of this study originate from the limitations of the video images due to the computer image pixel size. Illumination shadowing and manual measurement errors are also considered as potential errors. Repeatability tests done using actual experimental results and round steel bearings placed in the test section with water, resulted in a mean absolute error of 3%. The method given by Coleman and Steel [24] was used to calculate the error in the hydraulic diameter and estimated as a mean absolute value of 1.7%. Other errors are due to the flow meter that has an accuracy of 0.5% whereas the pressure transducers have an accuracy of 0.3%. The stainless steel sheathed K-type thermocouples resulted in an accuracy of $\pm 0.1^\circ\text{C}$. The effect of these errors is considered to be minimal for this study's micro bubble measurement requirements. The resultant error for the thermocouples placed on the boiler wall is larger than that of the system's thermocouples. This is due to the difficulty in installing these thermocouples on the inner side of the boiler tube wall. The readings were compared to a theoretical model for predicting the boiler wall temperature and the error is estimated to be at 5%.

4 Experimental results and discussion

The four parameters affecting a central heating system were investigated for effects on the resultant bubble diameter. Therefore an initial assumption was made in the form:

$$D_{bubble} = f(p, \mu_w, \alpha, Q)$$

Where p is the system pressure, $\dot{m}$ is the mass flow rate, α is the saturation ratio as defined by Jones et al. (1999) and Q the heating load.

As done by Prodanovic et al. [23] approximately 100 bubbles per experimental run were analysed and used to calculate the mean bubble diameter. Two Feret diameters were taken and averaged for each bubble along the principal axis of the bubble. In recent studies done for the analysis of bubble growth on static surfaces, the ratio of such diameters was used as a calculation of the bubble elongation with the flow. The analysis done in the present study has shown that bubbles are mostly spherical in shape and thus there is no elongation due to the surrounding bulk fluid flow. This is in agreement with recent studies done on bubbly flows [4] where slip ratios of 1 were observed with similar bulk fluid velocities. The Reynolds number in the boiler coils varied between 5,900 and 16,500. Therefore, the velocity profile for fully developed turbulent flow can be assumed for the present study.

A good representation of the average measured bubble diameters with respect to the experimental parameters as shown in Table 1 and the best model prediction, is given in Figs.(8) - (11). The best prediction is provided by the Winterton[7] model for bubble detachment diameters at finite contact angles. Its predictions are illustrated in Figs.(8) – (11). The zero contact angle model under predicted bubble diameters with an error in excess of 100% from the experimental diameters.

Therefore, the model's predictions are only included in Fig.(8), where a similar trend to the experimental and finite contact angle model predictions was observed.

Figure 8: Experimental and predicted bubble diameters with flow velocity.

Figure 9: Experimental and predicted bubble diameters with system pressure.

Figure 10: Experimental and predicted bubble diameters with heat flux.

Figure 11: Experimental and predicted bubble diameters with saturation ratio.

The experimental results are in reasonable agreement with the theoretical predictions of Eq. (5). Winterton's finite contact angle model predicts the bubble diameters with changing system parameters with a mean absolute error of 20%. When considering the limitations of the present study as highlighted in Section 3.1 and the general limitations inherent to two-phase flow studies [4], the discrepancies between theory and experimental results are considered to be reasonable. In one case, where the system velocity is at its lowest, the predicted detachment diameter is circa 45% larger than the experimental value. This could be attributed to the size limitations of the camera image. Large bubbles could have been ignored due to the fact they were not fully captured in the picture frame and as a result the experimental average could be understated.

The effect of the flow velocity on the bubble detachment diameter as seen in this study is in agreement with the Winterton models [7]. The Winterton model captured the effect of velocity on the predicted bubble detachment diameter through the inclusion of the Weber and Reynolds number. Similar trends were also evident in studies done by Al-Hayes and Winterton[16]. This suggests that the Winterton approach of balancing the drag and surface tension forces at the bubble detachment point is correct. Hepworth et al.[17] observed that the direction of liquid motion relative to the nucleation surface did effect the bubble detachment diameter. In fact, they state that tangential liquid motion produced smaller detachment diameters. As illustrated in Fig. (3) the heat exchanger used in the present study consists of a shell tube type heat exchanger with 12 tubes wound in a helical coil structure around the boiler burner. Consequently, both tangential and normal liquid motions relative to the nucleation surface were present. Therefore, the presence of both types of liquid motion

with the nucleation surface is considered to balance out any differences in the bubble detachment diameters. This was done as the limitations of the present study with respect to the boiler heat exchanger did not allow an experimental analysis of the detachment diameters at the boiler wall. A parallel could also be made with studies in sub cooled flow boiling where similar trends were observed[23]. Al-Hayes and Winterton[15,16]also considered the effect of the flow velocity on the bubble growth rate at its nucleation point, as a result of the liquid motion effecting the mass transfer coefficient for gas entering the bubble from the bulk liquid. A number of recent studies also suggest that the liquid motion could increase the growth rate[15]. However, the limitations of the present study did not enable the growth rate to be analysed.

The Winterton [7] finite contact angle model predicted a bubble diameter, circa 15% less than the experimental results in the pressure tests as illustrated in Fig.(9). Larger bubble diameters, with an average of 0.177mm were observed at the lowest system pressure, this being of 2 bars. This trend was not predicted by the Winterton[7] bubble detachment model for finite contact angles due to the limited change in water properties with the pressure range used in the present study. As most bubble detachment studies in supersaturated solutions are based on the balance of the drag and surface tension forces the authors of this study do not know of similar studies analysing the change in bubble diameter with system pressure. However, a similar trend with an increase in the system pressure was observed in sub cooled flow boiling conditions. Amongst such studies are those done by Prodanovic et al.[23] who conclude that bubble detachment diameters are strongly dependent on the system pressure, thus decreasing with pressure. Prodanovic et al. report that an increase in the pressure resulted in shorter bubble ejection times from their nucleating point. This theory could be extended to the gas diffusion approach, thus enabling the longer ejection times at lower pressures to allow more gas to diffuse into the nucleating bubbles.

For a change in heat flux, as illustrated in Fig.(10), the Winterton[7] finite contact angle model resulted in a predicted bubble diameter of circa 20% less than the experimental results. The increase in boiler wall heat flux from 17 to 50 kW/m², equal to a heating load of 7.5 and 21.5 kW respectively, resulted in a 19% increase in the observed experimental bubble diameters. A negligible increase in diameter was also predicted by the Winterton [7] detachment model for finite contact angles due to a change in the fluid properties, resulting in a reduction in water density, kinematic viscosity and surface tension with temperature. In fact, as the heating load was increased, higher boiler wall temperatures were observed due to the increase in the difference between the return and flow temperatures. Higher temperatures increase the diffusivity of the dissolved nitrogen in gasses. In view of this, the Al-Hayes and Winterton model [15,16] and the Jones et al.[9] model predict a shorter nucleation time with higher diffusion rates but does not reflect any changes on the bubble detachment diameters. Once more, a parallel can be drawn with similar studies in sub cooled flow boiling as done by Prodanovic et al. [23]. However, contrary to this study bubble detachment diameters were observed to decrease with increasing heat flux. Other studies into flow boiling by Abdelmessih et al.[25] observed similar trends to the present study with an increase in bubble diameters with heat flux.

As defined by Jones et al. [9] the low supersaturation ratios achieved in the present study lead to the Type IV non-classical nucleation at the boiler wall conditions. This type of nucleation occurs at pre-existing gas cavities on the surface of the boiler tube wall. An increase in the saturation ratios did not result in a change in the bubble detachment radius. An experimental average bubble size of circa 0.14 mm was observed. As illustrated in Fig.(11), this trend is in agreement with the Winterton's[7] approach for finite contact angles. At supersaturation ratios, the Winterton model for zero contact angles predicted a decrease in bubble detachment diameters with increasing supersaturation ratios. This is a result of the

decrease in the nucleation site radius with increasing pressure difference between the bubble and the bulk fluid as defined by Dean [1].

The Al-Hayes and Winterton[15,16] and the Jones et al.[9] approach for the bubble growth time are based on the diffusion theory and suggest that the gas concentration effects the bubble growth rate but does not result in a direct effect on the bubble detachment radius. This is in agreement with Winterton's detachment radius model which is based on the physical aspects of the bubble, thus governed by the balance of the drag and surface tension forces acting on the nucleating bubble. Recent studies have observed an increase in the bubble detachment diameter with an increase in dissolved gas content [17]. However, relatively high saturation ratios of 3.4 were observed in this study.

The under saturation experimental runs resulted in substantially reduced bubble counts when compared to experimental runs with supersaturated conditions. The authors of this study understand that classical theory for bubble nucleation as defined by Dean [1] suggests that under saturation conditions should not result in bubble nucleation due to the bulk fluid pressure being higher than the gas pressure in a bubble, thus inhibiting growth of existing gas cavities. The release of micro bubbles from the boiler tubes at under saturation conditions in the present study could be attributed to unmonitored high temperatures on the boiler wall that result in localised supersaturation areas at the boiler wall conditions.

Lower contact angles are expected to reduce the surface tension force holding the bubble to the nucleating surface. Therefore, a reduced bubble detachment diameter is predicted as lower drag forces are required to equalise the surface tension force. Hence, Winterton's zero contact angle model under predicted our results considerably. It is worth noting that as the theory of bubble nucleation states that bubbles nucleate in a cavity, some form of contact angle with the surface is always assumed to be present. Hence in his zero contact angle model, Winterton [7] assumed advancing and receding contact angles of 90 and 0 degrees respectively.

Contact angles are predicted to increase with surface roughness. [26] As most modern domestic central heating boilers make use of stainless steel heat exchangers, similar surface conditions and thus contact angles are expected with most contemporary boiler brands. However, it is worth noting that surface scaling as a result of multiple system refilling and prolonged usage could increase the heat exchanger surface roughness and consequently increase the bubble detachment diameters.

The present study does not examine the bubble behaviour following detachment from the nucleation point. Winterton [7] and Hepworth et al. [17] report that as the drag and surface tension forces equalise, the bubble vibrates in a fixed position until finally sliding on the surface before being carried into the main stream flow. They report that the phenomenon of sliding is not considered to have an effect on the bubble detachment diameter as it is the velocity required to detach the bubble from its nucleation point that is expected to determine the detachment diameter. Bubble sliding following detachment and prior to lift off to the bulk fluid flow was also reported in experiments in sub cooled flow boiling. [23]

The wall temperature in the condenser does not result in gas super saturation levels, and therefore no bubbles are released from the condenser wall. As a result, similar bubble characteristics are expected in older type non condensing boilers equipped with a similar primary heat exchanger design. The reasonable prediction given by Winterton's finite contact angle model through the use of the equivalent hydraulic diameter for the rectangular tubes used in the present study suggests that wider rectangular heat exchanger tubes should result in larger bubble diameters. Furthermore, the coil sectional and assembly design should have a minimal effect on the bubble detachment diameters as Winterton's model was originally developed for straight round tubes.

A comprehensive knowledge of the bubble detachment diameters at the primary heat exchanger wall in domestic central heating systems is required for the optimisation of passive deaerators installed with such systems. Passive deaerators are installed to reduce the bubble count at the boiler exit thus reducing the overall saturation ratio present in the system's water. Consequently the overall bubble nucleation rate at the primary heat exchanger wall is also reduced. Such devices consist of a vertical column and a float valve at the upper end of the column. Hence, air bubbles float up the column and accumulate at the top end. The excess air is then exhausted through the action of a float valve.

Due to the increase in the buoyancy force with bubble diameter, larger bubbles are known to facilitate the passive deaeration process used in domestic central heating systems. In fact, larger bubbles tend to float to the upper part of a horizontal pipe and consequently tend to bubble up the deaerator column more easily than their smaller counterparts. Therefore, lower system velocities should lead to improved deaeration rates, thus reducing the system susceptibility to problems related to two-phase flow, namely cold spots in radiators, excessive noise, pipework blockages and cavitation corrosion. Furthermore, lower boiler flow temperatures and higher system pressures result in lower saturation levels at the primary heat exchanger wall conditions, consequently reducing the bubble count in the system.

The present study demonstrates that the largest bubbles are observed at the lower system velocities. As illustrated in Fig. (12), 50% of the bubbles at the highest velocity are smaller than 0.12 mm whereas at the lowest velocity 50% of the bubbles are smaller than 0.17 mm in diameter. Hence, the present study demonstrates that the forces acting on the nucleating bubbles have the greatest influence on the detachment diameter at the heat exchanger wall. The presence of antibacterial and corrosion inhibitors in tap water is not expected to affect the bubble detachment diameter. Such additives could marginally affect the total dissolved gas saturation levels. However, as demonstrated in this study, changes in the saturation ratio are not expected to affect the bubble detachment diameter.

Figure 12: Bubble distribution with velocity.

5. Correlation of experimental data

To correlate the predicted average bubble diameter with respect to the parameters controlling a wet central heating system, the Winterton model [7] for the prediction of bubble detachment diameters in supersaturated solutions with finite contact angles was adopted. As no data is available with respect to the dynamic contact angles on the boiler wall, this term was eliminated and replaced by a constant as in Eq.(9). The effect of the system pressure and heat flux on the predicted bubble size was included through a dimensionless form. This was necessary as the Winterton model for finite contact angles did not capture the effect of heat flux and system pressure on the predicted bubble diameter. As in the original Winterton model, the Reynolds number was included to represent the effect of the bulk fluid velocity on the bubble diameter. The numerical constant and exponents for the three dimensionless numbers were optimised through the iteration method.

$$\frac{r}{R} = \frac{0.002716}{We} \left(\frac{p}{p_{atm}} \right)^{-0.461} \left(\frac{q}{q_{max}} \right)^{0.113} Re^{0.705} \quad (9)$$

The validity range of this correlation is based on the experimental parameter range used in the present study as illustrated in Table 2.

Table 2: Correlation validity range.

The correlation data is compared to the experimental data for all system parameters as illustrated in Fig.(13). Our new correlation predicted the bubble diameter at the exit of the boiler tube with a mean absolute error of 8%. Furthermore, after excluding the experimental data at low velocities, all experimental data points are between $\pm 12\%$ of the new correlation prediction. As discussed in Section 4 of this paper, the experimental error for the velocity tests could be more significant due to the size limitations of the sight glass section, thus resulting in the elimination of the larger bubbles at lower velocities. The authors of this study have considered the limitations of this statistical error analysis as a result of the limited number of data points available. However, a larger set of test runs was not possible due to the narrow parameter range present in such systems. The manual technique used in bubble measurement also limited the number of experimental runs possible. Therefore, further investigation into the validity of our new correlation is considered to be necessary.

Figure 13: Error plot for average experimental and present study correlation results.

6. Conclusions

The data gathered in this study is reasonably consistent with theory for the prediction of bubble detachment diameters in supersaturated solutions. The Winterton approach, based on Eq.(5) predicts the bubble radius on detachment for bubbles that extend into the transition region of the flow before breaking free from their nucleating point in round pipes. This theory is based on the resolution of the forces acting on the bubble at its nucleation point and parallel to the boiler wall surface. Therefore, the present study extends the use of this theoretical model to non circular, rectangular ducts and provides data with respect to the characteristics of micro bubbles present in a domestic central heating system. A good knowledge of micro bubble behaviour in such systems should lead to an optimised deaeration system thereby improving the overall system performance and thus reducing the extensive carbon footprint of such systems.

Particular attention has been drawn to the variation in bubble size with the system flow rate whereby a clear trend is evident between the identified model and the measured data. In fact, as predicted by a number of recent bubble nucleation studies, the bubble diameters decrease with an increase in bulk fluid velocity. Other system parameters such as the heat flux and pressure have shown a marginal effect on the bubble sizes whereas the limited saturation ratios range reached in this test rig did not result in any effect at all.

The present study has suggested a new correlation for wet central heating systems based on the Winterton approach. Our model incorporates the effect of pressure, fluid velocity and heat flux on the predicted bubble diameter at the exit of the boiler and has predicted the bubble diameters at the exit of a central heating boiler with reasonable accuracy.

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Notation

A	Tube cross sectional area, (m ²)
C_{gas}	Gas concentration in system, (standard cm ³ /Litre Water)
C_{sat}	Maximum gas concentration at boiler wall temperature, (standard cm ³ /Litre Water)
D_h	Tube hydraulic diameter, (m)
F_b	Buoyancy force, (N)
F_d	Drag force, (N)
F_{sx}	Surface tension force horizontal component, (N)
F_{sy}	Surface tension force vertical component, (N)
p	System pressure, (Pa)
P	Tube wetted perimeter, (m)
p_g	Partial pressure of dissolved gas, (Pa)
p_v	Vapour pressure, (Pa)
p_{atm}	Standard atmospheric pressure, (Pa)
q	Boiler heat flux, (W/m ²)
q_{max}	Maximum heat flux, (W/m ²)
Q	Heating load, (kW)
r	Bubble detachment radius, (m)
r'	Nucleation site radius, (m)
R	Tube radius, (m)
Re	Reynolds number, (-)
u	Bulk fluid velocity, (m/s)
We	Weber number, (-)
X^T	Gas solubility factor, (standard cm ³ /Litre Water/bar)

Greek Letters

α	Saturation ratio, (-)
ν	Kinematic Viscosity, (m ² /s)
θ_a	Advancing contact angle, (degrees)
θ_0	Static contact angle, (degrees)
θ_r	Receding contact angle, (degrees)
ρ_L	Water density, (kg/m ³)
σ	Surface tension, (N/m)

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Highlights

- We investigate the expected bubble sizes in wet domestic central heating systems.
- Micro bubbles nucleate at the boiler primary heat exchanger wall due to gas supersaturation conditions.
- We have investigated the effect of the fluid velocity, heating load, pressure and saturation ratio on the bubble detachment size.
- The fluid velocity has the greatest effect on the bubble detachment size.
- We correlated the force balance model by Winterton (1972) to fit our results.

Test	Bulk Fluid Velocity (m/s)	System Pressure (Bars)	Heat Flux (kW/m ²)	System Heating Load (kW)	Saturation Ratio at boiler wall conditions	Boiler Exit Temperature (°C)	Boiler Return Temperature (°C)	Duct Hydraulic Diameter (mm)
i	0.3-0.8	2.6	39	11	1.2	85	51-73	7.9
ii	0.8	2-3.75	39	17	0.8-1.2	75	55	7.9
iii	0.8	2.6	17-50	7.5-21.5	1.05	77- 59	50	7.9
iv	0.8	2.6	39	17	0.85-1.2	75	55	7.9

Table 1: Experimental parameters

Bulk Fluid Velocity (m/s)	System Pressure (Bars)	Heat Flux (kW/m ²)	System Heating Load (kW)	Duct Hydraulic Diameter (mm)
0.29 - 0.81	2 - 3.75	17-50	7-22	7.9

Table 2: Correlation validity range

Figure 1: Forces acting on a nucleating bubble in vertical and horizontal flow.

Figure 2: Static (left) and dynamic (right) contact angles on a nucleating bubble (section through bubble)

1. Water return
2. Water outlet

3. Sectional view of rectangular heating tubes - $D_h = 7.9$ mm
4. Location of gas fired burner

Figure 2: Primary heat exchanger assembly and rectangular tube sectional diagram

1. Boiler	10. Radiator
2. Phantom V5 high speed camera	11. Pressure release valve
3. Sight glass	12. Flow meter
4. K-type thermocouple	13. Cooling water tank
5. Bypass with ball valve	14. Electromagnetic tap controlled by Labview
6. Ball valve	15. Total dissolved gasses pressure transducer
7. Passive deaerator	16. Buffer vessel
8. Pressure transducer	17. Heating load heat exchanger
9. Nitrogen Inlet Valve	18. Electromagnetic flow meter

Figure 3: Schematic diagram of experimental set up

1. Light sources
2. Fibre optic light guide
3. Square sight glass section
4. High speed camera

5. Microscope lens
6. PC wired to camera
7. Focal depth of 1.5 mm

Figure 4: Bubble size measurement equipment

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