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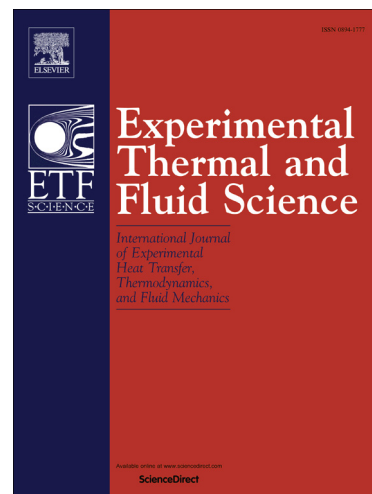
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# Interfacial friction in upward annular gas–liquid two-phase flow in pipes

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## Abstract

Accurate prediction of interfacial friction between the gas and liquid in annular two-phase flow in pipes is essential for the proper modelling of pressure drop and heat transfer coefficient in pipeline systems. Many empirical relationships have been obtained over the last half century. However, they are restricted to limited superficial liquid and gas velocity ranges, essentially apply to atmospheric pressures, and the relationships are only relevant for pipes with inner diameters between 10 and 50 mm. In this study, we carried out experiments in a large diameter flow loop of 101.6 mm internal diameter with the superficial gas and liquid ranges of 11–29 m/s and 0.1–1.0 m/s respectively. An examination of published interfacial friction factor correlations was carried out using a diverse database which was collected from the open literature for vertical annular flow. The database includes measurements in pipes of 16–127 mm inner diameter for the liquid film thickness, interfacial shear stress, and pressure gradient for air-water, air-water/glycerol, and argon-water flows. Eleven studies are represented with experimental pressures of up to 6 bar. Significant discrepancies were found between many of the published correlations and the large pipe data, primarily in the thick film region at low interfacial shear stress. A correlation for the interfacial friction factor was hence derived using the extensive database. The correlation includes dimensionless numbers for the effect of the diameter across pipe scales to be better represented and better fit the wide range of experimental conditions, fluid properties, and operating pressures.

**Keywords:** Co-current annular flow, pressure gradient, interfacial shear stress, liquid film thickness, vertical two-phase flow.

## 25 Nomenclature

**A. Roman**

$A$	[m <sup>2</sup> ]	Cross-sectional area
$D$	[m]	Pipe internal diameter
$e$	[-]	Entrained liquid fraction
$Fr$	[-]	Froude number = $u/\sqrt{gD}$
$f$	[-]	Interfacial friction factor
$g$	[m/s <sup>2</sup> ]	Acceleration due to gravity
$L$	[m]	Pipe length
$\dot{m}$	[kg/s]	Phase mass flow rate
$P$	[Pa]	Local pressure
$\Delta P$	[Pa]	Differential pressure
$\frac{dP}{dz}$	[Pa/m]	Pressure gradient
$Re$	[-]	Reynolds number = $\rho u D / \mu$
$t$	[m]	Film thickness
$u$	[m/s]	Phase superficial velocity
$x$	[-]	Gas quality = $\rho_g u_{sg} / (\rho_g u_{sg} + \rho_l u_{sl})$
$z$	[m]	Axial distance along pipe

**B. Greek**

$\delta$	Unit depends on quantity in question	Error in quantity indicated in bracket
$\varepsilon$	[-]	Void fraction
$\gamma$	[-]	Liquid droplet holdup
$\nu$	m <sup>2</sup> /s	Kinematic viscosity
$\mu$	[kg/s-m]	Dynamic viscosity
$\rho$	[kg/m <sup>3</sup> ]	Density
$\sigma$	[N/m]	Surface tension
$\tau$	[Pa]	Shear stress

**C. Subscripts**

$c$	Core
$d$	Droplet
$e$	Entrainment
$g$	Gas phase
$i$	Interfacial
$l$	Liquid phase
$lf$	Liquid film
$s$	Single phase
$sg$	Superficial gas
$sl$	Superficial liquid
$w$	Wall

## 27 1 Introduction

### 28 1.1 Background

29 Gas–liquid annular two-phase flow frequently occurs in many practical industrial applications.  
30 Common examples are gas condensates in oil and gas pipelines, evaporating and condensing flows in  
31 refrigeration systems and heat exchangers, and steam–water flows during nuclear reactor core cooling.  
32 This flow regime is characterised by the flow of liquid at the pipe inner wall periphery with a gas core  
33 flowing in the central region. At the interface between the two fluids, ripples and/or disturbance  
34 waves can be observed, the latter with high crests that can be sheared off at high gas velocities, which  
35 leads to liquid droplets being entrained in a highly turbulent gas core. These droplets can be re-  
36 deposited on the liquid film.

37 Because of the interactions of mass, momentum, and occasionally heat transfer at the  
38 interface, annular two-phase flows are complex and do not readily lend themselves to analytical  
39 modelling. Therefore, physically based models such as those based on two- or three-field models are  
40 usually solved numerically to obtain estimations of the phase fractions or pressure drop, which  
41 requires closure laws. Relations for closure laws include the fraction of liquid entrained as droplets  
42 and the interfacial friction factor, and they heavily rely on relevant experimental data.

43 The closure for interfacial friction factor deserves particular attention, because drag on the  
44 gas–liquid interface is dominant in the force balance (Belt et al. [1]). As will be shown later, the  
45 interfacial friction factor is an essential parameter in the force balance that couples the wavy film and  
46 the turbulent gas core. It is important to note that the interfacial friction factor differs from the two-  
47 phase friction factor. For the latter, a homogenous mixture is assumed to flow in the pipe and may be  
48 flow regime independent; therefore, mixture properties – density, viscosity and velocity – are hence  
49 defined and used. On the other hand, properties of the flowing gas are used in the case of the  
50 interfacial friction factor, where a momentum balance is done with the liquid film (not pipe wall) as  
51 boundary. Correlations for the interfacial friction factor derived by earlier investigators go back to  
52 Wallis [2] who used the analogy of single phase flow to propose a linear approximation to the

53 turbulent flow relationship in rough pipes. This works only for a small range of film thicknesses in the  
54 low interfacial shear stress region. Since then, modifications based on his theory have been made (see  
55 Belt et al. [1], [3]–[6]) but these correlations also fit limited ranges of data. Moreover, these do not  
56 usually give good estimations when the inner pipe inner diameter goes beyond 100 mm (the observed  
57 boundary between small and large diameter pipes).

58 Therefore many authors (Oliemans et al. [7]–Smith et al. [15]) insist that there is a need to  
59 increase the knowledge of multiphase flow behaviour for large diameter pipe systems. For instance,  
60 Oliemans et al. [7] compared entrainment correlations with large diameter test data and concluded  
61 there is not much confidence in the predicted values of the correlations. Kataoka & Ishii [8] showed  
62 that the use of the conventional drift flux model for pool void fraction prediction to relatively large  
63 pipes was only limited to low gas fluxes, and thus had to develop a new correlation for large systems  
64 where annular flow, for example, occurs at higher gas velocities. Disturbance waves which greatly  
65 contribute to wall shear stress and are a source of entrained droplets were observed by Azzopardi et al.  
66 [16] to be what they called “incoherent” in large diameter pipes. Careful observations revealed that in  
67 large pipes, the waves were not perpendicular to the flow direction but were curved “bow waves” and  
68 was also observed by Van der Meulen [17]. This contrasts to what obtains in smaller tubes where the  
69 waves are continuous around the circumference (see Nedderman and Shearer [18], and Zhao et al.  
70 [19]). The study by Omebere-Iyari and Azzopardi [20] on disturbance wave velocity provided strong  
71 quantitative indication of pipe diameter effect on the gas–liquid interface behaviour. They established  
72 that Pearce’s coefficient, which is proportional the to wave velocity, increases with pipe diameter  
73 until it reaches a constant value of 0.9 for large pipe diameters.

74 The foregoing means that the interaction between the fluids is different between the two pipe  
75 scales and published data on large diameter pipes is scarce. It is therefore important to focus on  
76 understanding the underlying mechanisms and finding more relationships that can apply to large pipes  
77 for more optimal designs of pipeline systems.

## 78 1.2 Previous interfacial friction factor correlations

79 Table 1 summarises several previous correlations for the interfacial friction factor in upward gas–  
80 liquid annular flow. It is pointed out that the interfacial friction factor is different from the two-phase  
81 friction factor which is based on the wall shear stress rather than the interfacial shear stress. The latter  
82 considers the flowing fluids as a homogenous mixture such that a mixture density and mixture  
83 velocity are used to calculate the wall shear stress and hence frictional pressure drop. Therefore, the  
84 prediction methodology is different from what is presented in this paper and correlations for the two-  
85 phase friction factor (e.g. Klausner et al. [21], and Cioncolini et al. [22]) are not included in the table.  
86 One of the earliest and most widely used correlations for the interfacial friction factor in two-phase  
87 flow modelling is that by Wallis [2], which is essentially a linear fit of the friction factor against the  
88 non-dimensional film thickness. Many modifications have been made to Wallis' correlation through  
89 simple adjustments of the coefficients in order to fit experimental data (Belt et al. [1], Moeck [3], Fore  
90 et al. [6], and Fukano and Furukawa [23]).

91

92 **Table 1: Reported interfacial friction factor correlations used for comparison with the current**  
93 **experimental data**

94

95 Various correction methods were applied in order to update Wallis' correlation such as raising the  
96 dimensionless film thickness to a power (Moeck [3]), introducing a function of the gas Reynolds  
97 number (Fore et al. [6]), or introducing a ratio to account for the influence of a change in fluid  
98 viscosity (Fukano and Furukawa [23]). As will be shown later, a plot of the Wallis-type correlations  
99 against the present data and reported data shows that one curve cannot be used to represent all of these  
100 data, as agreements that occur between some data and correlations do not occur with others.

### 101 1.3 Experimental data from the open literature

102 A total of 332 data points were collected for the vertical annular two-phase pressure drop in  
103 smooth pipes. The data were obtained from 12 sources using pipes with internal diameters ranging  
104 from 5 to 127 mm, including the current data. Key information about the sources in the database is  
105 given in Table 2 with a breakdown of selected parameters provided in the histograms in Figure 1. All  
106 the data are for macroscale gas–liquid flows obtained with transparent test sections, which allow for  
107 visual identification of the flow regime. All data are in the annular flow regimes, so it is assumed that  
108 there is little or no contamination from intermittent flow regimes such as slug or churn flows.

109

#### 110 **Table 2: Experimental data for upward interfacial friction factor (all air/water except stated otherwise)**

111

112 Development lengths ( $L/D$  ratios) for the different datasets as reported by their authors were  
113 examined and these are presented in Table 2. This is also shown in the histogram in Figure 1 (b), for  
114 which most of the data were collected between 41 and 100 pipe diameters with the largest being that  
115 of Owen [32] at 600 pipe diameters with 97 data points. Omebere-Iyari et al. [10] examined vertical  
116 two-phase flow with different fluid combinations and concluded that  $L/D = 40$  is sufficient for  
117 reasonably well-developed flow after showing that the probability distribution functions (PDFs) of the  
118 void fraction have very similar shapes at this value and higher. Their experiments were carried out at  
119 46 bar pressure, and this may have facilitated the relatively short development length of 40 pipe  
120 diameters. For the current study, the development length as given in Table 2 is 46 pipe diameters,  
121 slightly higher than that of Omebere-Iyari et al. [10]. As will be shown later (in Section 3.2), we  
122 observed that at  $L/D = 46$  reasonably developed flow was produced. The inlet condition for the  
123 upward section of the serpent pipe may have greatly assisted the relatively short development length.  
124 This is because the momentum generated from the preceding downward flowing section is inherited  
125 and this aids the upward flow development.

126

127 **Figure 1: Histograms showing selected parameters of the experimental database given in Table 2**

128

129 Four sets of data were obtained from experiments large diameter pipes. These sets include the current  
130 study (101.6 mm) and those by Van der Meulen [17], Zangana [30], and Skopich [31] who  
131 investigated flow in pipes with internal diameters of 127, 127, and 101.6 mm, respectively. The  
132 remaining eight data sources are for smaller diameter pipes (less than 100 mm) ranging from 5 to 50.8  
133 mm.

134 A majority of the measurements in the database are for air–water fluid combinations, except  
135 for those of Alia et al. [34], and Fore and Dukler [35], who reported work on fluid combinations of  
136 argon–water, and air/glycerine mixtures respectively. For system pressures, these runs range from just  
137 below atmospheric pressure (0.9 bar) in the air and water study by Skopich et al. [31] to 6 bar by Alia  
138 et al. [34] for argon and water. As shown in Figure 1 (c), the majority lie around atmospheric pressure.  
139 Many data points (108) fall between 2 bar (Owen [32]) and 3 bar (Van der Meulen [17]).

140 The database contains experimental measurements on pressure gradients, and several of the  
141 studies include direct measurements for liquid film thickness using conductance probes (the current  
142 study, Belt et al. [1], Alia et al. [34], and Wongwises and Kongkiatwanitch [29]). Others determine  
143 the liquid film thickness by measuring the cross-sectional void fraction and assuming a geometrically  
144 uniform film along the pipe circumference [11], [17], [30], [31]. Owen [32] measured the entrained  
145 droplet fraction in the gas core using the method of the isokinetic probe and estimated the droplet flow  
146 rate. Others such as Fore et al. [6] measured the entrained fraction by sucking out the film using a  
147 porous wall feature and subtracting from the liquid input. The entrained fraction is important in the  
148 calculation of interfacial fraction factor from the measured pressure gradient, because the entrained  
149 droplets change the properties of the gas core such as its density, and velocity. Belt et al [1] and Fore  
150 and Dukler [36] noted that the entrained droplets can in some cases contribute up to 20% of the total  
151 pressure gradient. Therefore, where entrainment occurs, this must be accounted for in the momentum  
152 balance equation.



153 We performed experiments in a large diameter flow loop in order to collect data on interfacial  
154 friction factor in upwards annular flow. Together with the diverse experimental database collected  
155 from the open literature, we will show that earlier correlations for the interfacial friction factor from  
156 small pipe measurements do not adequately represent data for large pipes and for conditions when the  
157 liquid film is relatively thick. Consequently, a correlation will be proposed that better fits our data as  
158 well as those from the diverse sources that cover a wider range of fluid velocities and pipe diameters.

## 159 **2 Experimental setup and data processing**

### 160 **2.1 Description of flow loop**

161 The experimental setup shown in Figure 2 is the Serpent Rig, an air/water facility at the Oil  
162 and Gas Engineering Laboratory of Cranfield University. It is divided into three main parts: the Fluid  
163 Supply (air and water) and Metering Area, the Test Area, and the Separation Section. The flow rig  
164 receives measured flow rates of water and air from the Flow Metering area to the test rig and finally  
165 into the ventilation tank, where the air and water are separated. The water is then returned to the  
166 storage tank while the air is vented.

167 The Test Area consists of the flow loop, which is a pipeline that is approximately 20 m long  
168 with an internal diameter of 101.6 mm (4 in.). The loop includes four vertical sections made from  
169 ABS (acrylonitrile butadiene styrene) plastic with upward and downward flows and connected by  
170 three Perspex 180-degree bends. The two 6-m vertical pipes in the middle are equipped with various  
171 instruments that collect all the data. While the right arm of the U is the downward flowing section, the  
172 left vertical section is the upward flowing section which is the area of interest of this study and where  
173 all the data were collected.

174

175 **Figure 2: Serpent Rig facility indicating the upward section used for this study**

176 Flow metering equipment includes two air flow meters capable of measuring 0–4250 Sm<sup>3</sup>/h  
177 of air and a water flow meter with a range of 0.06–16 L/s. Six GE Sensing UNIK 5000 pressure

178 transducers (0–1.5 barg  $\pm 0.04\%$ ) are located at positions P2, P5, and P6 for the upward flow section.  
179 Conductance probes are installed in the flow rig to measure the liquid film thickness, and a 32×32  
180 grid capacitance wire mesh sensor (WMS) measures the cross-sectionally averaged gas void fraction,  
181 which was used to identify the flow regime. Their locations are shown by dotted lines in Figure 2. A  
182 DeltaV system and LabVIEW software were used for data acquisition. The DeltaV system records  
183 and controls the air flow rates with a fixed sampling rate of 1 Hz, while LabVIEW acquires and  
184 records the pressures, film thicknesses, and fluid temperatures using a sampling rate of 100 Hz.  
185 Details about the design, calibration, and uncertainty analysis of the instrumentation are given in  
186 previous studies [37]–[40].

## 187 2.2 Instrumentation

### 188 2.2.1 Film thickness probes

189 The conductance probe is one of the most common techniques for determining liquid holdup  
190 in gas – liquid systems. It relies on measurements of electrical impedance between two electrodes in  
191 contact with a conducting fluid and several configurations have been over the years depending on the  
192 application and channel geometry. Its relative simplicity in application when compared to other  
193 methods has been identified as a major advantage as well as the low cost of equipment involved.  
194 Further details on types, principle of operation, design, and electronics can be found in the many  
195 published articles are available in the open literature (e.g. Coney [41], Koskie et al. [42],  
196 Tsocharitzidis et al. [43], and [44]). The film thickness probes used in this study were manufactured  
197 in-house and the design is shown schematically in Figure 3.

198

199 **Figure 3: (a) film thickness sensor spool with four flush-mounted probes (b) details of individual sensor**  
200 **design (all dimensions in mm) (c) blocks of different diameters used for probe calibration (d) sample of**  
201 **film thickness calibration curve (e) Repeatability tests for film thickness probes showing the mean film**  
202 **thickness at  $u_{sl} = 0.1$  m/s. For more conditions, see Almagro [37]**

203

204 The spool consists of four film conductivity sensors which are evenly distributed  
205 circumferentially to measure the circumferential distribution of the liquid film thickness at the  
206 location where the spool is installed. As shown in Figure 3 (b) the sensing part of the conductivity  
207 film thickness probe comprises a 10 mm diameter stainless steel rod and a stainless steel sleeve (18  
208 mm outer diameter by 2 mm wall) arranged concentrically. Between them is a 2 mm thick insulation  
209 layer. The end of the sensor is flush with the inner surface of the spool. Each conductor is electrically  
210 in contact with the liquid film when the liquid film flows over them, so a conductive bridge is formed.  
211 The conductivity and hence voltage between the two conductors is expected to change with the  
212 thickness of the water film. The normalised output, i.e. ratio of output voltage to full scale voltage,  
213 was used for calibration and measurements, as it minimises measurement errors caused by  
214 inconsistent liquid conductivity and environmental factors such as temperature. The calibration was  
215 carried out by using acrylic blocks (which like air, are electrical insulators) of known diameters  
216 inserted into the probe spool concentrically to form a liquid layer with known thickness. Calibration  
217 curves are plotted for the measured film thickness against normalised voltage output obtained offline.  
218 Equations of these curves were used to convert online voltages obtained during experiments to film  
219 thicknesses. Temperature correction coefficients for the sensors are regularly identified and applied  
220 for correcting the temperature drift errors in the film thickness measurement. This is in done in  
221 addition to the use of normalised film sensor outputs. The correction is applied to a temperature range  
222 of 10–35 °C, which covers the two-phase mixture temperature range 15–22 °C in which the tests were  
223 performed. During a test both the water and water/air mixture temperatures were logged so  
224 temperature compensation was able to be implemented in order to obtain film thickness values during  
225 the offline data processing. The repeatability of the liquid film thickness probes were checked by  
226 performing triplicate runs at a constant superficial liquid velocity and the full range of superficial gas  
227 velocities between 12.5 and 28.9 m/s. Figure 3 shows a plot of such a test at 0.1 m/s. A standard  
228 deviation of 0.1 mm in repeated film thickness measurements was established as a result, and this  
229 gives a  $\pm 3.3\%$  full scale error in the film thickness measurement. It is also possible that inlet/outlet  
230 stability could also add to small changes in the repeatability of the test, and this may have also

231 contributed in the  $\pm 3.3\%$  difference between respective measurements. It was considered that the  
232 uncertainty caused by calibration to be much smaller in comparison, and hence negligible, since the  
233 calibration curves of the probes give R-square values of 0.99 and above for fifth-degree polynomials.

### 234 2.2.2 Wire mesh sensor (WMS)

235 The capacitance Wire Mesh Sensor (WMS) used for the cross-sectional void distribution  
236 measurement at the stated locations along the pipe axis has a  $32 \times 32$  wire grid (Figure 2 (c)). The  
237 sensor, associated electronics and data processing software were provided by Helmholtz-Zentrum  
238 Dresden-Rossendorf, Germany. The method of phase fraction distribution measurement in air/water  
239 flows using similar WMS has been validated by a number of studies (Prasser et al. [45] and Da Silva  
240 et al. [46]). In the sensor assembly, wire electrodes perpendicular to each other are across the flow  
241 cross-sectional area.

#### 242 243 **Figure 4: Capacitance wire mesh sensor used in this study**

244 One set of the perpendicular wires acts as signal sender, while the other acts as receiver. The  
245 WMS measures the local permittivity of the fluid in the gaps of each crossing point by continuously  
246 applying an excitation voltage (of 5 MHz) to each sender electrode while keeping others at ground  
247 potential then synchronously measuring the alternating current flow to all receivers. Based on these  
248 measurements, the cross-sectional fluid distribution across the pipe the sensor can be estimated. For  
249 the sensor used in this study, the separation between the sender and receiver planes of wires is 2.5 mm.  
250 The spacing between two wires in parallel is 3.2 mm. A sampling rate of 1000 frames per second is  
251 used for the measurement. Validation studies were carried out on the measurement accuracy of the  
252 present WMS system and it was reported by Almagro et al. [39] that errors of around  $\pm 10\%$  are  
253 obtainable. Further reading on the theory and principles of operation of the WMS can be found in the  
254 works of Prasser et al. [45], Da Silva et al. [47], and Da Silva et al. [48].

## 255 2.3 Data processing

### 256 2.3.1 Film velocity and entrained droplet fraction

257 The entrained droplet fraction was calculated from measurements obtained for the liquid film  
 258 thickness discussed earlier, and the liquid film velocity measurements. Mean liquid film velocity  
 259 measurements were made using the electrolyte injection method. The method involves injecting  
 260 sodium chloride electrolyte into the liquid film to create a surge in conductivity. The transit time of a  
 261 conductivity surge between two identical sensors 100 mm apart in a probe spool is determined using a  
 262 cross-correlation algorithm in MATLAB. The film velocity is then calculated by dividing the distance  
 263 by the signal transit time or time delay between the two sensors. The assumption here is that as the  
 264 liquid films are very thin ( $\sim 1 - 1.5 \text{ mm}$ ), the velocity profile within the film is such that the wave  
 265 (or structure) velocities are close to the bulk film velocity, meaning velocities with higher uncertainty  
 266 for the thicker films. Nevertheless, uncertainties of around  $\pm 8\%$  in the film velocity measurements  
 267 were produced and this has been discussed extensively [40].

268 Using the definitions given in

269 Figure 5, the measured velocities can then be used to calculate the entrained droplet fraction  
 270 using the relationship:

$$e = \frac{\dot{m}_l - \dot{m}_{lf}}{\dot{m}_l} = \frac{\rho_l u_{sl} A - \rho_l u_{lf} A_{lf}}{\rho_l u_{sl} A} = 1 - \frac{4u_{lf}t}{u_{sl}D} \quad (1)$$

271 where  $\dot{m}_l$ , and  $\dot{m}_{lf}$  are the total liquid flow rate and liquid film flow rate in kg/s, while  $e$ ,  $t$ ,  $u_{lf}$ ,  $u_{sl}$ ,  
 272 and  $D$  are the entrained liquid fraction, measured film thickness, measured film velocity, liquid  
 273 superficial velocity and pipe internal diameter respectively. Tabulated and graphical results obtained  
 274 using Equation (1) for different flow conditions are given in Almbrok [37] and Aliyu et al. [40].

275

276

**Figure 5: Representation of annular flow with droplets**

277 Figure 6 (a) and (b) show typical examples of time series data collected for the liquid film  
 278 thickness and pressure gradient. The measurement conditions include a superficial liquid velocity of  
 279 1.0 m/s and superficial gas velocity of 18.56 m/s. Data were recorded for 2 minutes per run, but the  
 280 figures show the behaviour for 10 seconds only. The wavy nature of the liquid film recorded by the  
 281 conductance probe is clearly seen in Figure 6 (a) with wave heights that are one to two times greater  
 282 than the thickness of the base film. This behaviour has widely been reported for annular films (Hall  
 283 Taylor et al. [49], and Sawant et al. [50]).

284 **Figure 6: Examples of time series data for (a) liquid film thickness and (b) pressure gradient with a**  
 285 **superficial liquid velocity of 1.0 m/s and superficial gas velocity of 18.56 m/s**

286 For subsequent steady-state analyses, the time series data were averaged over the whole two-  
 287 minute recording time, as shown in Table 3. For all superficial liquid velocities, the liquid film  
 288 thickness increases with superficial liquid velocity but decreases with increasing gas velocity. This  
 289 decrease with gas velocity is asymptotic for vertical upflow regardless of liquid velocity and viscosity  
 290 (see Fukano and Furukawa [23]). Equation (2) is the steady-state momentum balance for the two-  
 291 phase mixture flowing upwards in the pipe:

$$-\frac{dP}{dz} = \{\rho_c \varepsilon + \rho_l (1 - \varepsilon)\}g + \frac{4}{D}\tau \quad (2)$$

292 Where  $\rho_g$  and  $\rho_l$  are the gas and liquid densities respectively;  $\varepsilon$  is the time-averaged cross-sectional  
 293 void fraction. The first term on the right hand side of the equation refers to the gravitational  
 294 component of the pressure gradient; the second is the frictional component with  $\tau$  being the shear  
 295 stress, which consists of the wall and interfacial shear stresses:

$$\tau = \frac{1}{2}\{f_w \rho_l (1 - \varepsilon) u_{lf}^2 + f_i \rho_c \varepsilon (u_c - u_{lf}) |u_c - u_{lf}|\} \quad (3)$$

296 where  $f_w$  and  $f_i$  are the wall and interfacial friction factors respectively. As shown in Table 3, for all  
297 superficial liquid velocities, the liquid pressure gradient increases with the gas superficial velocity.  
298 This is easily explained by Equations (2) and (3), where the frictional component of the pressure  
299 gradient increases as the superficial fluid velocities increase. Wall friction increases as liquid velocity  
300 increases; while interfacial friction increases with increasing slip at the gas–liquid interface, and this  
301 is more affected by the core velocity since the liquid velocity does not change by as much.

302

303 **Table 3: Averaged measured quantities for the air–water system in the upward section of the Serpent Rig**

304

305 A flow regime map for upward flow was produced using visual observations and this is  
306 presented in Figure 7 (a). Flow regimes that are either bubbly or churn are marked as “other” and  
307 these have been excluded from the data in Table 3. The regimes were identified with the help of  
308 videos made from WMS reconstructed images. Specially designed software was used to stack the  
309 images acquired at 1000 fps and these can be played back at slower speeds than real-time. Example  
310 snapshots of the videos at  $u_{sl} = 0.2$  m/s are shown in Figure 7 (a) and they were used to differentiate  
311 flow at  $u_{sl} = 9.7$ , and 12.09 m/s. It can be seen that for the former condition, the gas core is not  
312 continuous. Large liquid lumps can be seen to be torn off from base film. Such a flow was not  
313 classified as annular since some up and down oscillatory movement was observed. The regime was  
314 ultimately classified as “churn” flow and screened out from those used for the subsequent analyses in  
315 this paper. At critical conditions near flow regime transition regions, identification is difficult to  
316 achieve with the naked eye, and this underscores the importance of using the WMS. Conversely, the  
317 snapshot on the top right-hand corner of Figure 7 (a) was classified as annular where a distinct gas–  
318 liquid interface and a continuous gas phase can be seen.

319 Figure 7 (b) is a plot of measured pressure gradient ( $dP/dz$ ) for the upward flow section. For  
320 all superficial liquid velocities,  $dP/dz$  values decrease with increasing gas velocity. There is a decrease

321 in  $dP/dz$  because increasing gas void fraction results in decreasing cross-sectional liquid holdup and  
 322 the liquid contribution to the gravitational pressure gradient also decreases. However, a minimum is  
 323 reached when increasing the gas velocity results in the frictional component of  $dP/dz$  equalling the  
 324 gravitational component. This results in a competition between inertia and gravity forces and  
 325 oscillatory behaviour is observed with the liquid. The minimum  $dP/dz$  is a critical point corresponding  
 326 to a flow regime transition from churn to annular flow (see Hewitt and Hall-Taylor [51], McQuillan  
 327 and Whalley [52], and Hewitt [53]). Further increasing the gas velocity causes the pressure gradient to  
 328 increase due to the increasing dominance of the frictional component over the gravitational  
 329 component. The flow is now in the annular region and the conditions for this study were chosen from  
 330 this region indicated by the closed markers in Figure 7 (b). This agrees with our visual observations  
 331 as well as the reconstructed WMS images.

332

333 **Figure 7: Effect of fluid superficial velocities on flow regime map produced by visual observations aided**  
 334 **by WMS visualisation**

335

### 336 2.3.2 Determination of interfacial friction factor from measurements

337 From the foregoing discussion, the pressure gradient, liquid film thickness and velocity measurements  
 338 obtained for this study were verified to be in the annular region. These were used to calculate the  
 339 interfacial friction factor, which in fully developed annular two-phase flow is defined as:

$$f_i = \frac{2\tau_i}{\rho_g u_{sg}^2} \quad (4)$$

340 The superficial gas velocity  $u_{sg}$  in this work is calculated based on the whole pipe cross-sectional  
 341 area. In many cases, droplets are torn off from the crests of the liquid film (or atomised) and become  
 342 entrained in the gas core, thereby affecting its density. The interfacial friction factor thus becomes:

$$f_i = \frac{2\tau_i}{\rho_c u_c^2} \quad (5)$$



343 where the subscript  $c$  denotes the core region. The following three steps are used to determine the core  
 344 velocity, core density, and the interfacial friction factor respectively:

### 345 1. Core velocity

346 Estimation of the core density  $\rho_c$  requires knowledge of the liquid film velocity, which was  
 347 determined experimentally for the present dataset. Details are well documented for the measurement  
 348 technique using cross correlation of conductance signals from adjacent axial probes and the  
 349 corresponding uncertainties [37], [38], [40], [54]. The core velocity  $u_c$  is estimated by writing a mass  
 350 balance of the gas input, which yields the relationship for the mean velocity for the gas core (see  
 351 Vieira et al. [55]):

$$u_c = \frac{(u_{sg} + u_{sl}e)D^2}{(D - 2t)^2} \quad (6)$$

352 where  $t$  is the measured mean liquid film thickness assuming it is circumferentially uniform, and  $e$  is  
 353 the entrained droplet fraction determined experimentally. In cases where experimental data for  $e$  was  
 354 not available, it was calculated using the correlation of Cioncolini and Thome [56]. Other empirical  
 355 correlations exist such as those of Ishii and Mishima [57], and Sawant et al. [58].

356

357 **Figure 8: Representation and notation of the various phases (film, gas, and droplets) occupying the total**  
 358 **pipe area. Subscript  $d$  denotes droplets. Where  $\gamma$ ,  $\varepsilon$ , and  $\varepsilon_c$  are the droplet holdup, void fraction and core**  
 359 **void fraction respectively (Cioncolini et al. [22])**

### 360 2. Core density

361 The density  $\rho_c$  of a gas core laden with entrained droplets used in Equation (5) is calculated  
 362 as the arithmetic mean of the pure gas and liquid droplet densities:

$$\rho_c = (1 - \varepsilon_c)\rho_l + \varepsilon_c\rho_g \quad (7)$$

363 where  $\varepsilon_c$  (Figure 8) is the gas core void fraction estimated as:

$$\varepsilon_c = \frac{\varepsilon}{\varepsilon + \gamma(1 - \varepsilon)} \quad (8)$$

364  $\varepsilon$  is the cross-sectionally averaged void fraction, and  $\gamma$  is the droplet holdup. An expression for the  
 365 droplet holdup  $\gamma$  can be obtained using the definition of the entrained droplet fraction  $e$  which is the  
 366 ratio of the entrained liquid droplet mass flow rate  $\dot{m}_d$  to that of the total liquid mass flow rate  $\dot{m}_l$  i.e.:

$$e = \frac{\dot{m}_d}{\dot{m}_l} = \frac{\rho_l u_d A_d}{\rho_l u_{sl} A} = \frac{u_d A_d}{u_{sl} A} \quad (9)$$

367 If the phase slip between the droplets and the carrier gas is ignored, the droplets travel at the velocity  
 368 of the carrier gas, therefore  $u_d = u_g$ . Equation (9) now becomes:

$$e = \frac{u_g A_d}{u_{sl} A} = \frac{u_{sg} A_d}{\varepsilon u_{sl} A} \quad (10)$$

369 Substituting the relation for the cross-sectional area occupied by the droplets  $A_d$  based on the phase  
 370 splits as shown in Figure 8, we can rewrite Equation (10) as follows:

$$e = \frac{u_{sg} \gamma (1 - \varepsilon)}{u_{sl} \varepsilon} \quad (11)$$

371 The gas quality  $x$  is defined as the ratio of the gas mass flux to the total gas mass flux; i.e.  $x =$   
 372  $\rho_g u_{sg} / (\rho_g u_{sg} + \rho_l u_{sl})$ . Rearranging it as  $\frac{u_{sg}}{u_{sl}} = \frac{x \rho_l}{(1-x) \rho_g}$  and substituting in Equation (11) yields:

$$e = \frac{x \rho_l}{(1-x) \rho_g} \frac{\gamma (1 - \varepsilon)}{\varepsilon} \quad (12)$$

373 which can be rearranged to the following to give an expression for the droplet holdup for use in  
 374 Equation (8):

$$\gamma = e \frac{\varepsilon}{1 - \varepsilon} \frac{1 - x \rho_g}{x \rho_l} \quad (13)$$

### 375 **3. Interfacial shear stress**

376 The interfacial shear stress  $\tau_i$  is obtained from a momentum balance around the gas core in the axial  
 377 direction of the flow, as shown in Figure 9. It is assumed that the flow is fully developed, at steady  
 378 state, and at equilibrium where the rate of droplet entrainment equals the rate of droplet deposition.  
 379 Therefore,

$$\tau_i = \frac{D - 2t}{4} \left( -\frac{dP}{dz} - \rho_c g \right) \quad (14)$$

380 where  $-dP/dz$  is the measured pressure gradient. For a detailed derivation of Equation (14), see Fore  
 381 et al. [6], and Wongwises and Kongkiatwanitch [29]. Equations (6)–(14) can then be used to calculate  
 382 the interfacial friction factor as defined in Equation (5).

383

384

385 **Figure 9: Control volume for the momentum balance in Equation (14). Based on the schematic by**  
 386 **Wongwises and Kongkiatwanitch [29]**

### 387 **3 Discussion of results**

#### 388 **3.1 Flow regime maps**

389 The conditions for the current experiments and those obtained from the literature were plotted  
 390 together on the Taitel et al [59] flow regime map, as shown in Figure 10. It can be seen that all the  
 391 data points chosen for this study fall in the annular flow region. Taitel et al's [59] flow regime map  
 392 was constructed using physically based mechanisms which underlie each transition. They also derived  
 393 models for the transitions based on these mechanisms. Along with the Hewitt and Roberts [60] flow  
 394 regime map (Figure 11Figure 11), it is one of the most widely used for gas–liquid flow in vertical  
 395 pipes. The test conditions range from the near annular transition (e.g. Van der Meulen [17], and Alia  
 396 et al. [34]) to the fully annular flow region (e.g. Shearer and Nedderman [61]). Flow regime  
 397 transitions are not sharp and occur within a range of superficial fluid velocities, which could be  
 398 affected by the liquid viscosities and densities of liquid and gas phases. As a result, data points that  
 399 may seem to fall in the churn flow region of the map were actually observed to be annular flow. For  
 400 example, Alia et al. [34] report data that are deeply within the annular flow regime region when  
 401 plotted on the Hewitt and Roberts [60] flow regime map shown in Figure 11Figure 11, but these  
 402 points fall in the annular flow region nearer the transition areas when plotted on the Taitel et al. [59]  
 403 flow regime map. However, in the flow regime map published by Shell [62] and shown in Figure 12,  
 404 the conditions for the entire database can be seen to be in the annular flow region. The flow regime

405 map is based on the gas and liquid Froude numbers on the horizontal and vertical axes respectively.  
 406 Here, the Froude numbers are densimetric and defined as follows:

$$Fr_g = u_{sg} \sqrt{\frac{\rho_g}{(\rho_l - \rho_g)gD}}, Fr_l = u_{sl} \sqrt{\frac{\rho_l}{(\rho_l - \rho_g)gD}} \quad (15)$$

407

408

409 **Figure 10: Experimental data against the Taitel et al. [59] flow regime map at (a) 1 bar (b) 2.5 bar**  
 410 **pressure**

411

412 **Figure 11: Experimental data bank against Hewitt and Roberts [60] flow regime map showing all data**  
 413 **are in the annular flow regime**

414

415

416 **Figure 12: Experimental databank against the Shell [62] flow regime map Film thickness, void fraction,**  
 417 **and flow development**

418 The measurement station within the dotted circle of the upward flowing section of the flow  
 419 rig in Figure 2 is located at 46 pipe diameters from the bottom bend. An extensive study on axial  
 420 variation of measured quantities along the straight sections of the rig was reported in the doctoral  
 421 thesis of Almabrok [37]. It showed that a significant reduction in entrance effects is already achieved  
 422 on reaching the top location. Time series and probability distribution functions of film thickness,  
 423 pressures, and void fraction exhibited similar established flow at the top ( $L/D = 46$  from the U-bend)  
 424 and middle positions ( $L/D = 28$ ). These tendencies substantially differ from those observed at the  
 425 bottom position ( $L/D = 5$ ) which exhibited flow maldistributions due to bend effects. Confirmation of  
 426 flow development was done by observing slow-motion videos made by stacking individual WMS  
 427 frames (achieved using specially designed software).

428

429 **Figure 13: Normalised liquid film thickness axial development in the upwards flowing section of the rig**  
 430 (See Almbrok [37] for more)

431

432 Figure 13 shows normalised mean film thicknesses obtained from the conductance probes at  
 433 the stated axial positions for  $u_{sl} = 0.2, 0.3,$  and  $0.48$  m/s. The normalised film thickness here is  
 434 defined as the ratio of film thickness at other L/D positions to that at L/D = 46 (i.e.  $t_{norm} =$   
 435  $t/t_{L/D=46}$ ). This means that  $t_{norm}$  at 46 pipe diameters is unity. The error bars shown in the graphs  
 436 were calculated as errors propagated from the  $\pm 3.3\%$  uncertainty in the film thickness probes; and  
 437 these are  $\sqrt{3.3^2 + 3.3^2} = \pm 4.7\%$ . It can be seen that there is minimal change in the normalised mean  
 438 film thickness between the middle and bottom positions. Similarly, Figure 14 Figure 14 shows the  
 439 normalised void fraction obtained from the WMS at L/D = 5, 28, and 46 positions from the bottom U-  
 440 bend for  $u_{sl} = 0.2, 0.48,$  and  $1.0$  m/s. The void fractions were normalised with respect to that at L/D  
 441 = 46 such that  $\varepsilon_{norm} = \varepsilon/\varepsilon_{L/D=46}$  and  $\varepsilon_{norm} = 1$  at L/D = 46. The error bars shown in the graphs for  
 442  $\varepsilon_{norm}$  were calculated as errors propagated from the  $\pm 10.0\%$  uncertainty in the WMS; and these are  
 443  $\sqrt{10.0^2 + 10.0^2} = \pm 14.4\%$ . Again, as shown, the ratios at the bottom position reach up to 1.2 in  
 444 contrast to those obtained at the middle which converges very close to 1. The effect of superficial  
 445 liquid velocity can be seen from Figure 13 and Figure 14. As liquid velocity increases, there is  
 446 increasing disparity between the film thickness/void fraction ratios at the bottom positions of the pipe  
 447 for different gas velocities. However, the ratios become similar by converging towards unity for all  
 448 conditions once the flow reaches the middle part of the pipe. While L/D = 46 is one of the least in the  
 449 database collected (see Table 2), flow development is helped by the momentum gathered in the  
 450 downward flowing section and transferred as the flow negotiates the bend to the upward flowing  
 451 section. Therefore, the data used for the analyses in this paper are those at L/D = 46, represent highly  
 452 developed upwards annular flow (fully developed flow is difficult to achieve and requires very long  
 453 pipes). Additional details can be found in Almbrok [37].

454

455 **Figure 14: Normalised gas void fraction axial development in the upwards flowing section of the rig**

456

### 457 3.2 Comparison of experimental friction factors and existing correlations

458 The interfacial friction has an inverse relationship with the gas Reynolds number as expected,  
 459 which is akin to the turbulent flow region of the Moody Chart for single-phase flow. The gas  
 460 Reynolds number used is here is based on the superficial gas velocity and pipe diameter i.e.

$$Re_g = \frac{\rho_g u_{sg} D}{\mu_g} \quad (16)$$

461 Fore et al. [6] note that the at sufficiently high Reynolds numbers, the interfacial friction  
 462 factor reaches an asymptotic value and at such conditions, the friction factor becomes dependent only  
 463 on the relative liquid film thickness. Nevertheless, it is clear that the gas Reynolds number is a  
 464 candidate for interfacial friction factor correlation, and one curve that is a function of the Reynolds  
 465 number only cannot represent all the data. Other parameters representing the different experimental  
 466 conditions, pipe scales, and fluid properties need to be considered. This is perhaps evident from the  
 467 different trends exhibited by the various datasets, because the ranges of liquid flow rates and pipe  
 468 sizes widely differ. Error analysis of the interfacial friction factor is given in the Appendix, and this  
 469 shows relative errors of  $\pm 6$ –14%.

470 A plot of  $f_i$  vs  $t/D$  values for the current experiments, as well as those from previous studies  
 471 is given in

472 Figure 15. It can be seen that there are general under- and over-predictions that are magnified  
 473 as  $t/D$  increases. For instance, the measurements by Shearer and Nedderman [33] and those by Owen  
 474 [32] are within 50% of the values predicted by the correlations of Wallis [2], Moeck [3], and Fore et  
 475 al. [6] at low  $t/D$  values, but they deviate by more than 10 times at higher values. One reason for the  
 476 general agreement at small  $t/D$  values could be that most data and correlations were obtained at  
 477 conditions in the full annular region, where the films are thinner and smoother. These are far from the

478 transition region where thick and rough films are found. With the dimensionless film thickness being  
 479 able to show the behaviour of the interfacial friction factor within the various regions of annular flow,  
 480 it is therefore a relevant parameter in correlating interfacial friction factor data.

481

482

483 **Figure 15: Experimental interfacial friction factors versus superficial gas Reynolds number**

484

485

486 **Figure 16: Comparison of current and other experimental friction factors with Wallis-type correlations**

487 Another group of  $f_i$  correlations have Wallis-type structure but no functional dependence on  
 488  $t/D$ . Examples are those by Ambrosini et al. [26], and Holt et al. [28] who use several combinations  
 489 of the bulk gas Reynolds and Weber numbers. Asali et al. [5] and Ambrosini et al. [26] used  
 490 correlations with the film thickness ( $t^+$ ) non-dimensionlised by the friction velocity. Implementation  
 491 of these equations is iterative since they are implicit functions of the interfacial friction factor. As  
 492 shown in Figure 17 (e) and (h), these correlations provide good predictions in the higher shear regions,  
 493 where the films are thinner and smoother but rapidly deteriorate for thicker films. Asali et al. [5] and  
 494 Ambrosini et al. [26] used similar databases for their correlations, which included data obtained from  
 495 pipes with internal diameter of 10–42 mm and fluid combinations of helium and  
 496 hydrocarbons/glycerine mixtures, in contrast to the air or steam/water combinations mostly  
 497 considered in the present database.

498 Hori et al. [24] opted for a direct power law relationship for the friction factor using the  
 499 Reynolds and Froude numbers of the gas and liquid. The Froude numbers are defined as follows:

$$Fr_g = \frac{u_{sg}}{\sqrt{gD}}, Fr_l = \frac{u_{sl}}{\sqrt{gD}} \quad (17)$$

500 Their correlation produced good estimations at low values of the interfacial friction factor. However,  
 501 as shown in Figure 17 (d), substantial over-estimations occur at high values of  $f_i$  corresponding to low

502 gas flow regions where the liquid films are thick and rough. Again, this is because the experimental  
503 conditions do not cover these regions. Wongwises and Kongkiatwanitch [29] also used a power law  
504 relationship for their friction factors obtained in a 29-mm pipe at atmospheric pressure. They used the  
505 gas Reynolds number and the non-dimensional film thickness  $t/D$  as their correlating dimensionless  
506 numbers and their correlation was less successful over the experimental database.

507 In summary, these interfacial friction factor correlations were developed with data obtained  
508 for small pressure ranges. For example, Fukano and Furukawa [23] carried out measurements at  
509 around 1 bar, Holt et al. [28] used 0.2–1.5 bar, Ambrosini et al. [26] used 0.2–1.9 bar, and Asali et al.  
510 [5] used 1–2 bar. Only Fore et al. [6] had a wider range of test pressures than the current database at  
511 3.4–17 bar for experiments with air–water, air–water/glycerine, and nitrogen–water gas–liquid  
512 combinations. However, their experiments were conducted with a small diameter pipe and in the thin  
513 film region, in contrast to the current database of larger and smaller-diameter pipes containing much  
514 thicker liquid films.

515 It is clear that there is a need to extend the applicable range of prediction methods for the  
516 interfacial friction factor. While a mechanistic or physical approach is desirable, it has not been  
517 achieved due to the difficulties introduced in annular two-phase flow by high turbulence and complex  
518 momentum transfer across the highly deformable gas–liquid interface. Improvements may be  
519 achieved using correlations over broader ranges of flow conditions and tube sizes. We therefore  
520 propose a new correlation using a more diverse database.

521

522

523 **Figure 17: Comparison of 332 interfacial friction factor data points with predictions of various**  
524 **correlations**



### 525 3.3 Correlation of interfacial friction factor

526 Fore et al. [6] noted that the Wallis correlation and other correlations with Wallis-type dependency on  
 527  $t/D$  do not adequately account for changes in Reynolds number in annular flow. They are best  
 528 applied in annular flow where the dependence on the Reynolds number is minimal, such as for thin,  
 529 smooth films. In such a situation, the friction factor depends on only the dimensionless film thickness  
 530 and has been argued to have linear dependence (e.g. Belt et al [1]). However, this conclusion is based  
 531 on only a small number of data points collected in a small-diameter pipe in the annular flow regime.  
 532 For films that are deeply in the annular regime and especially in large pipes, the flow behaviour  
 533 cannot be dictated by the rough film mechanism because these films are smooth and thin.

534 Azzopardi [37] and Jayanti et al. [64] show that the waves on the crests of the film in large  
 535 pipes are circumferentially incoherent, meaning that they cover only a part of the tube circumference.  
 536 They therefore do not solely dictate the flow behaviour, and the flowing gas plays a vital role in  
 537 determining many of the flow characteristics, including interfacial friction. It is thus necessary to fit  
 538 the friction factor data in order to reflect other parameters that dictate the flow behaviour, not only the  
 539 dimensionless film thickness as in the classic correlations. The following form is therefore proposed  
 540 for the interfacial friction factor:

$$f_i = f_s [1 + a(t/D)^b Re_g^j Fr_g^k]^l \quad (18)$$

541 Where  $f_s = 0.046 Re_g^{0.2}$ . Equation (18) includes the gas Reynolds and Froude numbers (defined in  
 542 Equation 17) but essentially retains the form of a Wallis-type correlation. This preserves the  
 543 behaviour of the Wallis correlation, which works quite well at small values of  $t/D$  (Figure 16 and  
 544 Figure 17). The Reynolds number ensures that the correlation captures inertia changes in the gas core.  
 545 The gas Froude number is included because it is a ratio of inertial forces of pressure-driven gas/liquid  
 546 flow to the opposing gravitational force, particularly important for high-velocity gas flows in vertical  
 547 upward annular two-phase flow. The inertial forces dominate gravitation, resulting in  $Fr_g > 1$  and this

548 is termed supercritical flow where the tendency of films to flow downwards is overcome by inertia  
 549 from the gas flow.  $Fr_g \approx 1$  is the critical gas Froude number at the flow reversal point. In fact, authors  
 550 such as Wallis ([2], [65]) and Barbosa et al. [66] have used  $Fr_g > 1$  as a criterion for the transition  
 551 from churn to annular flow and annular flow. If  $Fr_g < 1$ , annular flow is not considered to have  
 552 occurred and the flow is still in the churn regime where film oscillations occur due to the two  
 553 competing forces. Therefore, the inclusion of the gas Froude number is consistent with the  
 554 phenomenon of annular flow. Partial correlational analysis with other dimensionless numbers such as  
 555 the core Reynolds, and Weber numbers as well as the liquid Reynolds number showed that these have  
 556 little effect on the interfacial friction factor.

557 Non-linear least squares regression with the entire experimental database yields the following  
 558 correlation:

$$f_i = f_s [1 + 0.3(t/D)^{0.12} Re_g^{0.54} Fr_g^{-1.20}]^{1.5} \quad (19)$$

559 where  $Re_g$  and  $Fr_g$  are defined in Equations (16) and (17). The correlation is valid for  $D^* > 2$ , where  
 560  $D^* = D/\sqrt{\sigma/g(\rho_l - \rho_g)}$  the dimensionless hydraulic diameter as defined by Kataoka and Ishii [8].  
 561 Pipes with  $D^*$  greater than about 37, corresponding to  $D = 100$  mm, are considered to be large  
 562 diameter. For  $D^* < 2$  corresponding to  $D < 5$  mm, Equation (19) may not apply due to lack of  
 563 representative data (for  $D < 5$  mm) in the databank. For such pipes capillary forces will be prevalent  
 564 and non-dimensional quantities such as the Capillary number (Ca) may be used to correlate data.

565 Figure 18 shows that this correlation fits the data better than any of those in Figure 17 (a)–(m). From  
 566 the figure, it is evident that there is a clear segregation in the magnitudes of the experimental friction  
 567 factors along pipe scales, with the larger pipes having higher interfacial friction factors. This might  
 568 seem surprising at first but is entirely logical since larger pipes produce smaller relative slip between  
 569 the phases compared to small-diameter pipes (i.e., a lesser value of  $(u_c - u_{if})$ ).

570

571 **Figure 18: Predictions of proposed correlation compared with entire experimental database**

572

573 For the gas Reynolds number in the proposed correlation, the positive index of 0.55 may seem  
574 inappropriate at first, given that there is always an inverse relationship between friction factors and  
575 the Reynolds number or fluid velocity. This is also true in the present case, as the superficial gas  
576 velocity inherent in  $Re_g$  and  $Fr_g$  carries a net negative index of  $-0.66$ , which is within the range of  $-$   
577  $0.6$  to  $-0.89$  obtained for the index of  $Re_g$  by previous authors in their correlations for  $f_i$  (Fore et al.  
578 [6], Wongwises and Kongkitatwanitch [29], Hori et al. [24], Ambrosini et al. [26], and Holt et al.  
579 [28]). Furthermore,  $t/D$  in Equation (19) is raised to a power of  $0.1$  which is a rather small number  
580 indicating a lesser than expected dependence of  $f_i$  on film thickness. This was explained by  
581 Cioncolini and Thome [67] and Narcy [68] to be because the velocity profile in the film is  
582 concentrated in the region near the pipe wall. This near wall region is much smaller than the film  
583 thickness, hence the small power on  $t/D$ .

584 The computed statistics (MSE, MAE, etc.) show that the proposed correlation predicts the  
585 current database quite well. It exhibits the lowest values of mean square error (MSE) (see Table 4).  
586 Although other correlations have comparable values (e.g. those of Asali et al. [5], and Hori et al. [24]),  
587 their predictions have much higher mean absolute error and lower number of points within the  $\pm 50\%$   
588 error band. Over half of the predicted points lie within the  $\pm 50\%$  error band for the correlations  
589 reported by Asali et al. [5], Fore et al. [6], and Wongwises and Kongkitatwanitch [29], as shown in  
590 the table. However, these values carry a bias based on the large number of data from small pipes, and  
591 the data are mostly for relatively thin films, as shown in

592

593 Figure 16 and Figure 17. As stated earlier, the shape, size, and localisation of the waves in  
594 large pipes have been shown by many authors in the past to differ from smaller pipes and these  
595 features affect the manner in which droplet entrainment/deposition and momentum transfer occur. The  
596 new correlation can be used as a closure relationship in numerical codes to produce better predictions

597 of interfacial friction factor, especially in the thick film regions where the gas velocity is low (around  
598 15–20 m/s) and the gas–liquid interface is rougher.

599

600 **Table 4: Statistical comparison between predictions of proposed and previous correlations**

601

602 While no correlations are universally acceptable, due to their limited or lack of physical insight,  
603 we suggest a more mechanistic approach for future work. This is not easy since the turbulent gas–  
604 liquid interface presents unique challenges of momentum transfer by droplet deposition and  
605 entrainment. There are currently no mechanistic models for droplet entrainment and deposition, which  
606 adds another layer of difficulty for such a methodology. Another approach towards improvement  
607 could be correlations fitted with more local measurements of local phenomena rather than with bulk  
608 properties of the film and flowing gas. However, this will require the collation of a large pool of more  
609 local measurements such as the disturbance wave heights, wave frequencies for the film, and velocity  
610 profiles within both phases. Complete sets are currently scarce or remain largely unpublished. The  
611 addition of evaporating, boiling, condensing flows, and data from experiments conducted in  
612 microgravity will also expand the applicable ranges of new empirical models. Nevertheless, as shown,  
613 the current correlations can be used with more confidence, given that they have been developed from  
614 a broader range of flow conditions and pipe diameters than what has been reported.

#### 615 **4 Conclusions**

616 In this study experiments were performed using a large diameter flow loop of 101.6 mm internal  
617 diameter within the superficial gas and liquid ranges of 11–29 m/s and 0.1–1.0 m/s respectively. Data  
618 on pressure gradient, as well as film thickness were collected in the annular flow regime and these  
619 were used to calculate the interfacial friction factor. The effectiveness of using existing prediction  
620 methods for the interfacial friction factor was then examined in large-pipe annular flow, as well as in  
621 small-pipe flow from a variety of sources. These sources include data from a wide spectrum of flow  
622 conditions and pipe sizes. For example, the range of pipe diameters is 20–127 mm, and the data are

623 obtained from experiments performed with air–water, argon–water, and air–glycerol combinations as  
624 the test fluids. The interfacial friction factor is one of the important parameters in determining  
625 pressure gradient in the annular flow regime. However, our analyses show that many of the previous  
626 correlations can be inadequate when used outside certain ranges of experimental conditions,  
627 especially for large industrial pipe simulations. Nevertheless, many of these are used as closure laws  
628 in commercial thermal-hydraulic codes based on one-dimensional two- or three-fluid models. The  
629 results indicate the following:

- 630 • Previous correlations for the interfacial friction factor of vertical annular flow are mostly  
631 arithmetic modifications of Wallis correlation ( $f_i = 0.005[1 + 300t/D]$ ), where the intercept  
632 and/or factor are changed to fit ranges of experimental conditions.
- 633 • Wallis-type correlations which are functions of only  $t/D$  give some deviations of  $f_i$   
634 predictions for thicker liquid films especially in large diameter pipes.
- 635 • The proposed correlation ( $f_i = f_s[1 + 0.3(t/D)^{0.12}Re_g^{0.54}Fr^{-1.20}]^{1.5}$ ) gives better  
636 description of the interfacial friction factor in upward annular flow across pipe scales in the  
637 database. It maintains the shape and characteristics of the Wallis-type dependency on the  
638 dimensionless film thickness at low  $t/D$  values. However, introducing simple functions of the  
639 core Reynolds and Froude numbers ensures its relevance for thicker films.

640 As mentioned, mechanistic methodologies are required to provide more generally applicable models.  
641 More studies are needed for optimal design and operation of pipeline systems and other equipment in  
642 novel applications such as microchannels for cooling in microelectronics, as well as many traditional  
643 cases in oil and gas, refrigeration, heat exchange, and other process industries.

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## 655 Appendix A

### 656 Error estimation for the current friction factor calculations

657 Error analysis of the calculated variables is important to provide an estimation of uncertainty  
 658 propagation due to measurements. Here, the quantity of interest is the interfacial friction factor. To  
 659 estimate it, we substitute Equation (14) in Equation (5), which yields a relationship for calculating the  
 660 interfacial friction factor in terms of all the measured experimental quantities of the pressure gradient,  
 661 liquid film thickness, and core density (a function of the measured liquid film velocity):

$$f_i = \frac{D - 2t}{4} \left( -\frac{dP}{dz} - \rho_c g \right) \quad (A1)$$

$$\frac{1}{\frac{1}{2}\rho_c u_c^2}$$

662 The quantities in Equation (A1) that are dominant in the uncertainty determination are  $-\frac{dP}{dz}$ ,  $\rho_c$ ,  $t$ ,  $u_c$ ,  
 663 and  $u_{lf}$ . The uncertainty in  $f_i$  is caused by the random errors in the measurements and can be given as  
 664 a relative error. Thus,

$$\frac{\delta f_i}{f_i} = \sqrt{\frac{(\delta S)^2 + (\delta \rho_c)^2}{(S + \rho_c g)^2} + \left(\frac{\delta t}{D - 2t}\right)^2 + \left(\frac{\delta \rho_c}{\rho_c}\right)^2 + \left(\frac{2\delta u_c}{u_c^2}\right)^2} \quad (A2)$$

665 where  $S = -\frac{dP}{dz}$ , and  $\delta$  represents the uncertainties in the respective quantities. For pressure gradient  
 666 measurements, the uncertainty in  $-\frac{dP}{dz}$  is  $\pm(\sqrt{(0.04)^2 + (0.04)^2}) = \pm 0.057\%$  based on the  
 667 manufacturer quoted error of  $\pm 0.04\%$  for each transducer. These give 0.42–1.58 Pa/m uncertainty in  
 668 pressure gradient for the current experiments (See Table 3 for the actual values of pressure gradient

669 obtained). The specified range of measurement of the transducers is (0–1.5 barg  $\pm 0.04\%$ ). For film  
 670 thickness (measurements were between 0.7–1.3 mm), the uncertainty is  $\pm 3.3\%$  as estimated by three  
 671 repeated film measurements; while for the film velocity, it is  $\delta u_{lf} = \pm 8\%$  of local measurements (see  
 672 Aliyu et al. [40]). The uncertainty in the gas core density is not readily available since they are  
 673 derived quantities. Therefore, the uncertainties in  $\rho_c$  and  $u_c$  are deduced from their definitions in  
 674 Equations (6) and (7), respectively

675 The uncertainty in  $\rho_c$  in Equation (A2) is deduced from the definition of  $\rho_c$  in Equation (7). The  
 676 wetted area  $A_{lf}$  of liquid film and the input liquid mass flow rate in Equations (8) and (13) are  
 677 substituted in Equation (7) and simplified. This yields an expression for  $\rho_c$  in terms of  $u_{lf}$ :

$$\rho_c = \rho_l - \frac{\varepsilon(\rho_l - \rho_g)}{\varepsilon + \left(1 - \frac{\rho_l u_{lf} A_{lf}}{\dot{m}_l}\right) \frac{\varepsilon}{1 - \varepsilon} \frac{1 - x \rho_g}{x} \frac{\rho_l}{\rho_l}} \quad (\text{A3})$$

678 where  $\dot{m}_l$  is the liquid input mass flow rate (in kg/s). Now,

$$\delta(\rho_c) = \delta(u_{lf}) \frac{\partial \rho_c}{\partial u_{lf}} \quad (\text{A4})$$

679 where  $\frac{\partial \rho_c}{\partial u_{lf}}$  is obtained by differentiating Equation (A3). After rearranging,

$$\frac{\partial \rho_c}{\partial u_{lf}} = \frac{\varepsilon(\rho_l - \rho_g) \frac{\varepsilon}{1 - \varepsilon} \frac{1 - x \rho_g}{x} \frac{\rho_l A_{lf}}{\dot{m}_l}}{\left[\varepsilon + \left(1 - \frac{\rho_l A_{lf}}{\dot{m}_l}\right) \frac{\varepsilon}{1 - \varepsilon} \frac{1 - x \rho_g}{x} \frac{\rho_l}{\rho_l}\right]^2} \quad (\text{A5})$$

680 For the uncertainty in the core flow velocity  $u_c$ , its definition in Equation (6) is used to deduce  $\delta(u_c)$ :

$$\delta(u_c) = \sqrt{\left[\delta(e) \frac{\partial u_c}{\partial e}\right]^2 + \left[\delta(t) \frac{\partial u_c}{\partial t}\right]^2} \quad (\text{A6})$$

681 where  $\frac{\partial u_c}{\partial e}$  and  $\frac{\partial u_c}{\partial t}$  are obtained by differentiating Equation (6) with respect to  $e$  and  $t$  respectively.

682 Thus,

$$\frac{\partial u_c}{\partial e} = \frac{u_{sl}}{(D - 2t)^2} \quad (\text{A7})$$

683 and

$$\frac{\partial u_c}{\partial t} = \frac{4t(u_{sg} + u_{sl}e)}{(D - 2t)^3} \quad (\text{A8})$$

684 where the entrained droplet fraction  $e$  is determined experimentally from the measured liquid film  
 685 velocity using Equation (1). Lastly, the percentage error in  $f_i$  for each experimental condition is  
 686 calculated as follows:

$$Err(f_i) = \frac{\delta f_i}{f_i} \times 100\% \quad (A9)$$

687 As similarly shown in our previous publication [40], the errors for determining the interfacial friction  
 688 factor from the experimental measurements using Equations (A1)–(A9) are between 5 and 15%, with  
 689 a majority of errors in the vicinity of 7–13% (Figure A 1).

690

691 **Figure A 1: Experimental uncertainties in  $f_i$**

692

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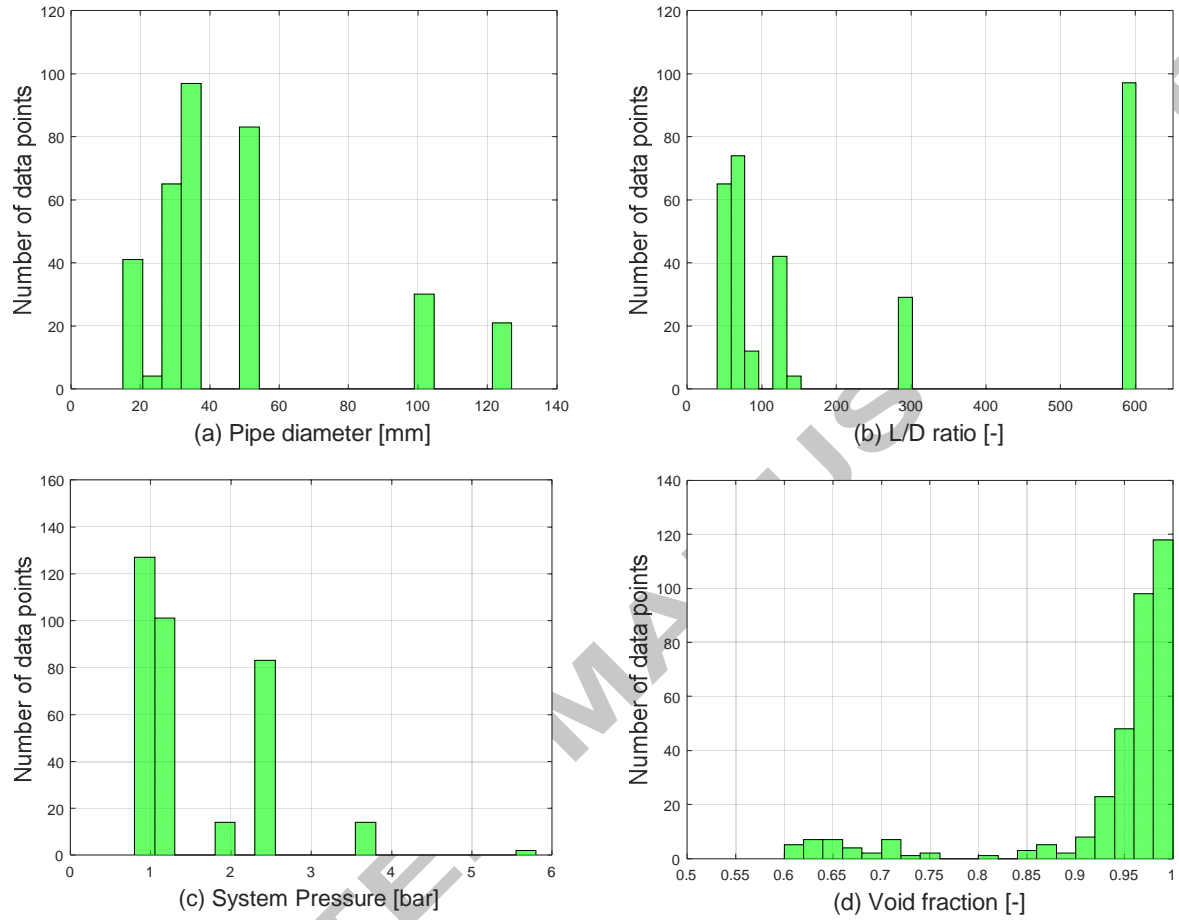
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Figures:



**Figure 1: Histograms showing selected parameters of the experimental database given in Error! Reference source not found.**

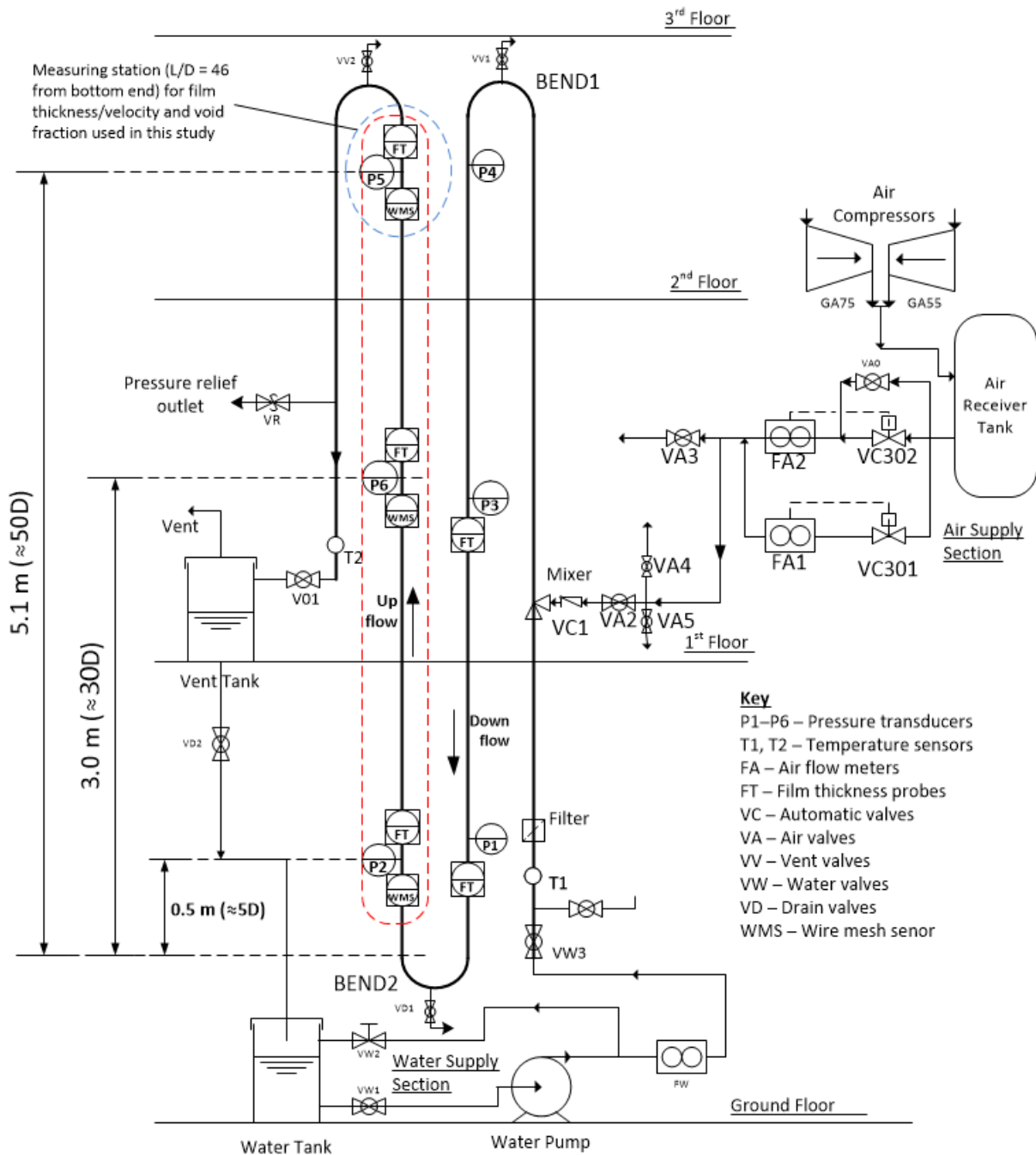
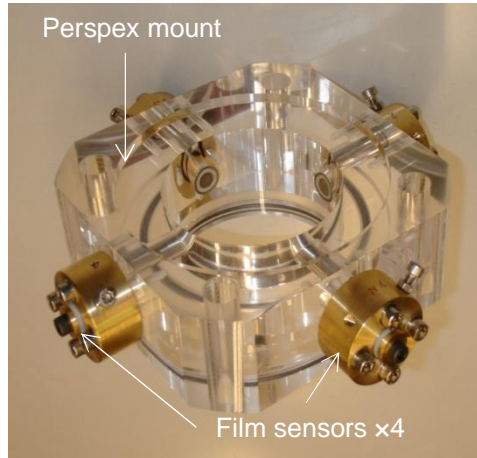
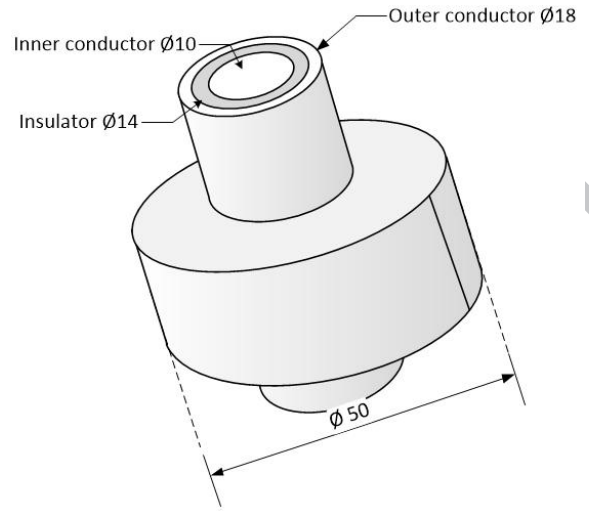


Figure 2: Serpent Rig facility indicating the upward section used for this study





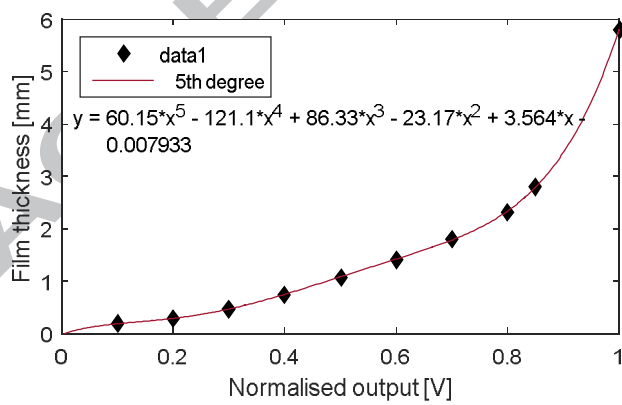
(a)



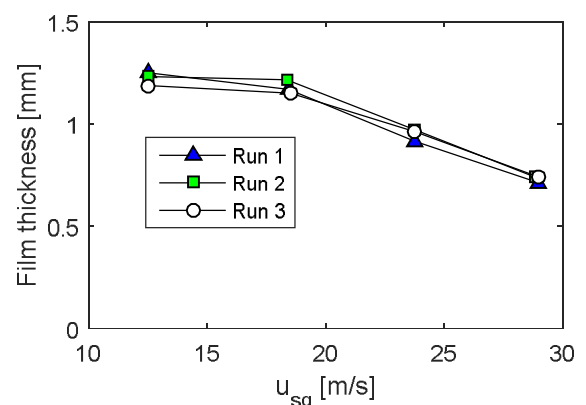
(b)



(c)



(d)



(e)

**Figure 3: (a) film thickness sensor pool with four flush-mounted probes (b) details of individual sensor design (all dimensions in mm) (c) blocks of different diameters used for probe calibration (d) sample of film thickness calibration curve (e) Repeatability tests for film thickness probes showing the mean film thickness at  $u_{sl} = 0.1$  m/s. For more conditions, see Almagro [37]**



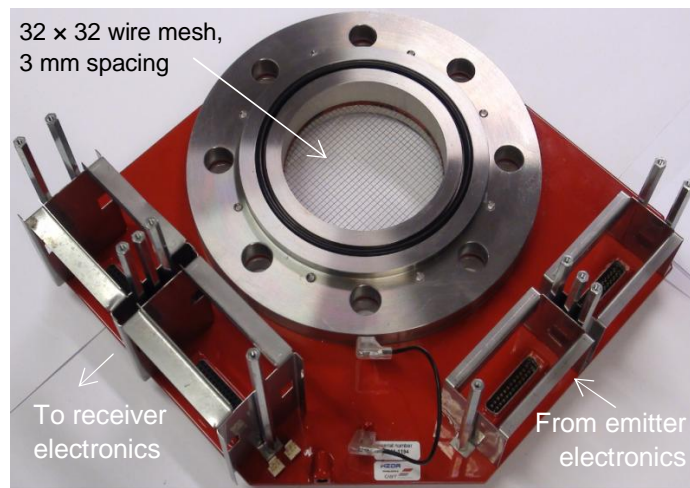


Figure 4: Capacitance wire mesh sensor used in this study

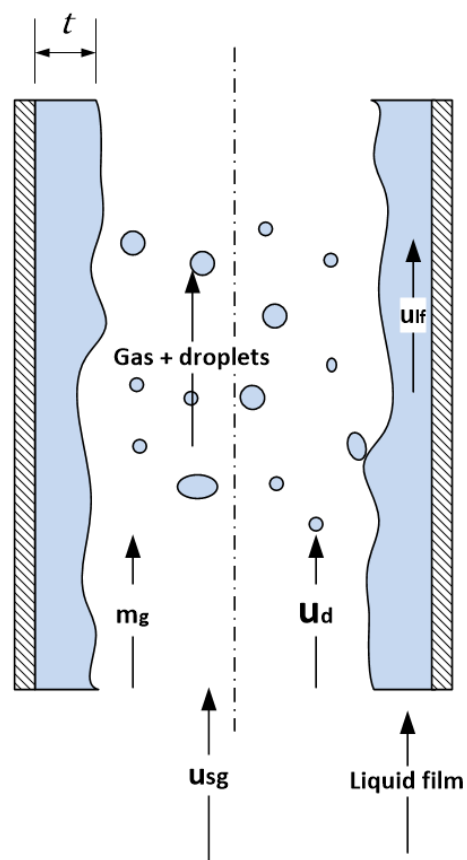
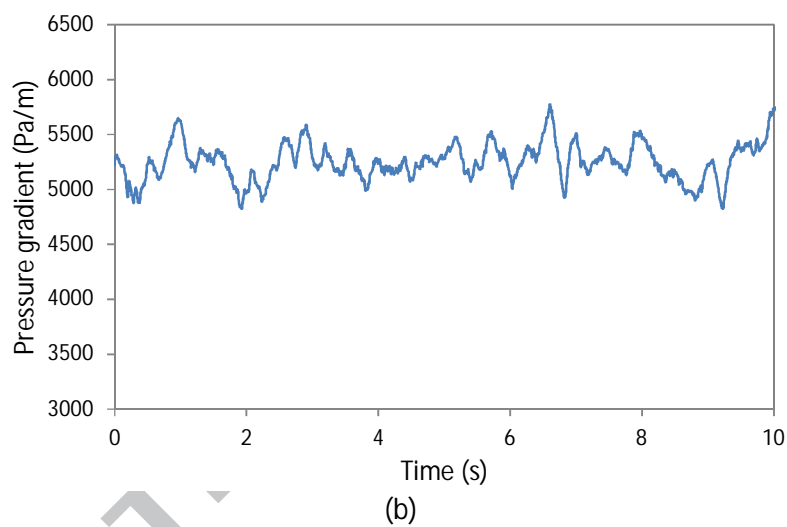
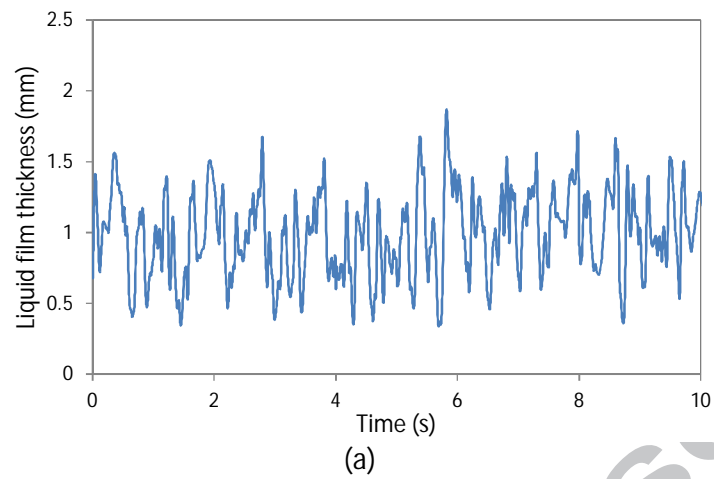


Figure 5: Representation of annular flow with droplets



**Figure 6: Examples of time series data for (a) liquid film thickness and (b) pressure gradient with a superficial liquid velocity of 1.0 m/s and superficial gas velocity of 18.56 m/s**

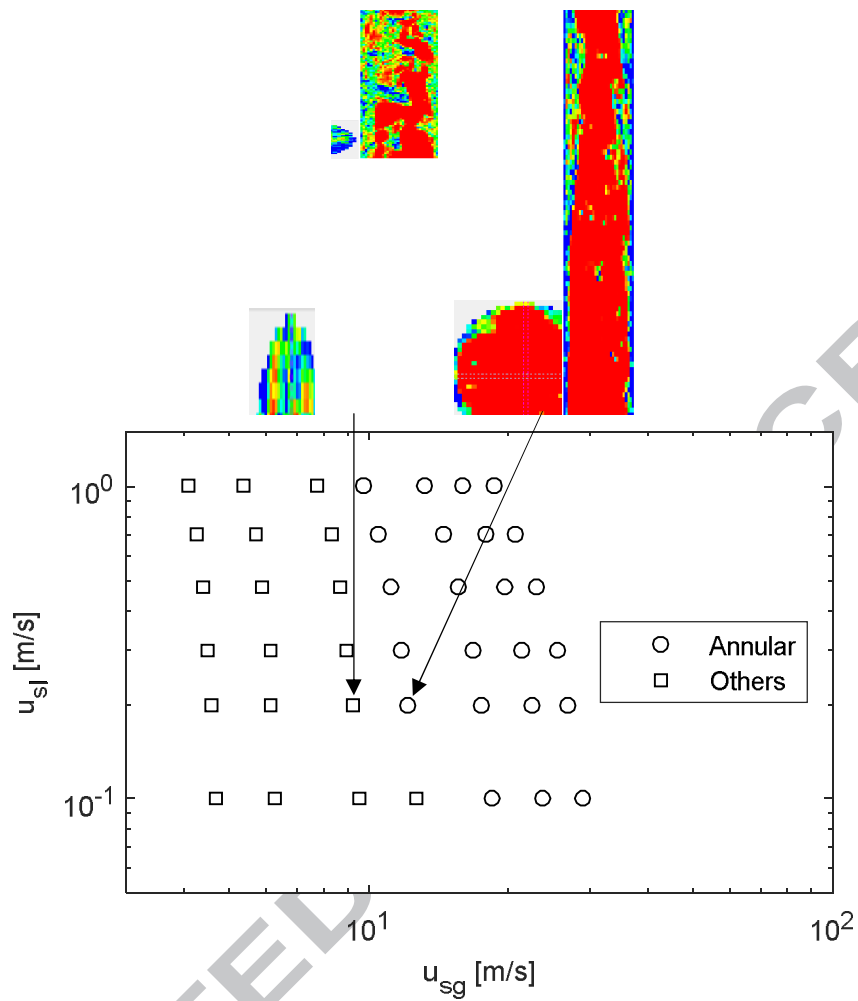
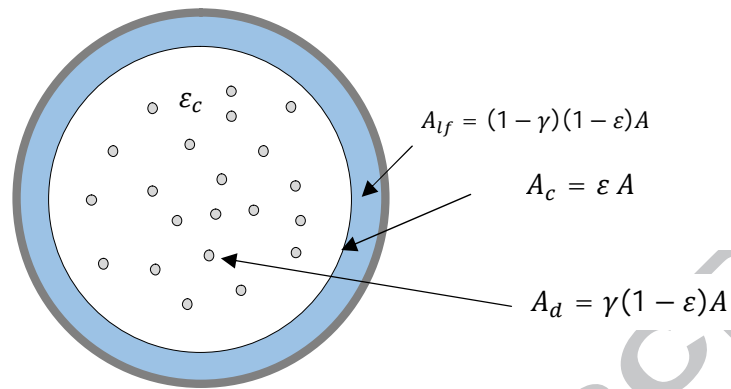


Figure 7: Effect of fluid superficial velocities on flow regime map produced by visual observations aided by WMS visualisation



**Figure 8: Representation and notation of the various phases (film, gas, and droplets) occupying the total pipe area. Subscript  $d$  denotes droplets. Where  $\gamma$ ,  $\epsilon$ , and  $\epsilon_c$  are the droplet holdup, void fraction and core void fraction respectively (Cioncolini et al. [22])**

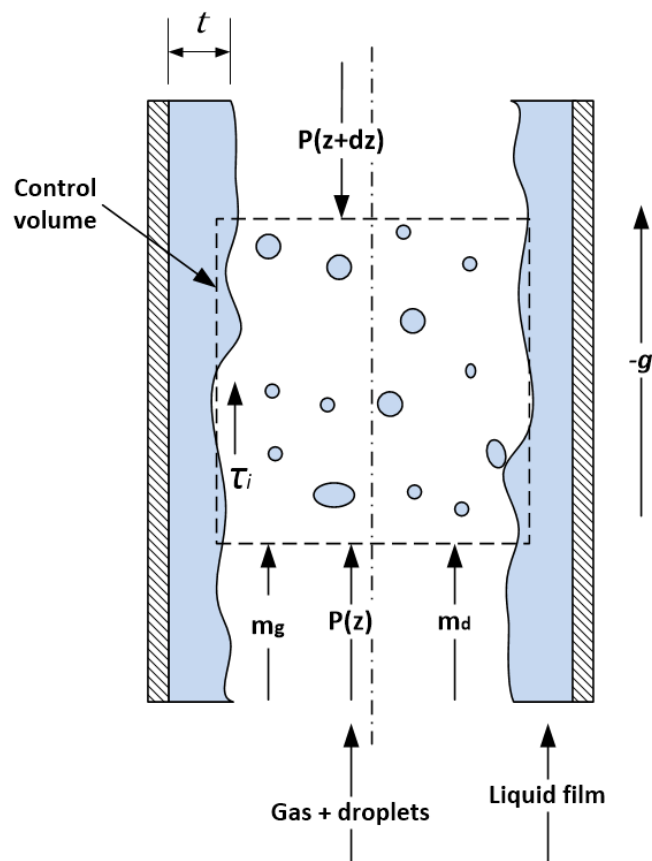
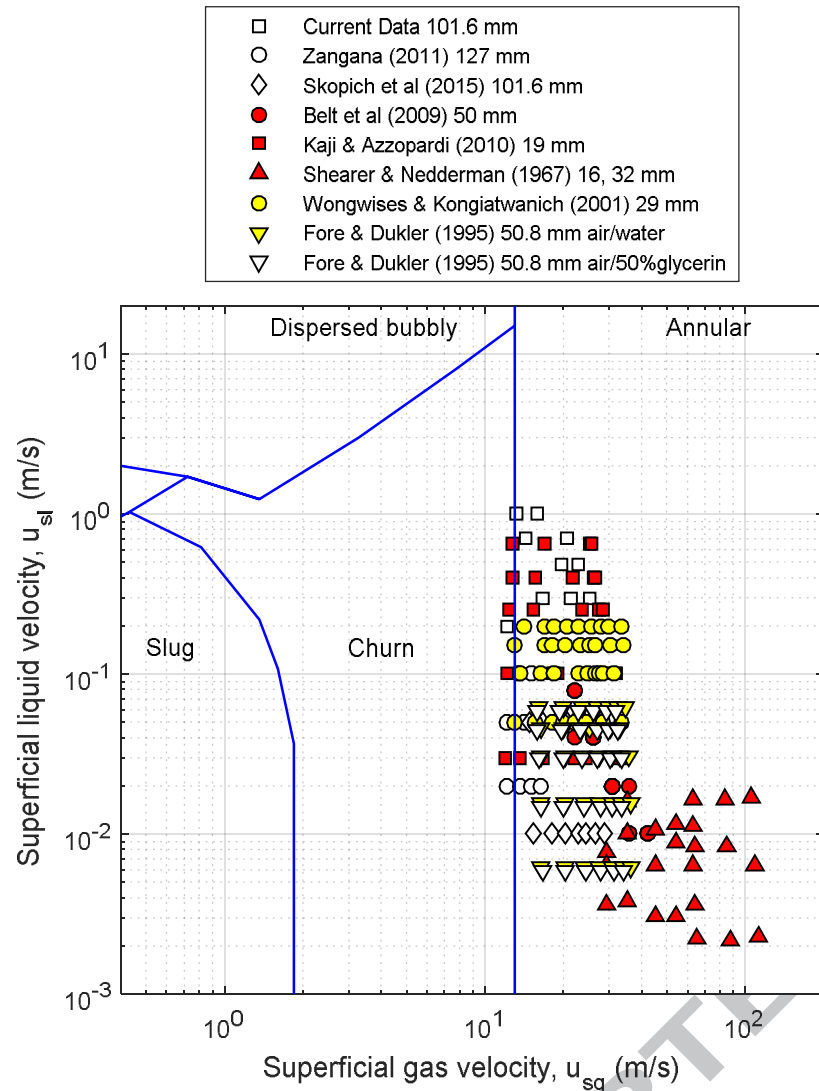
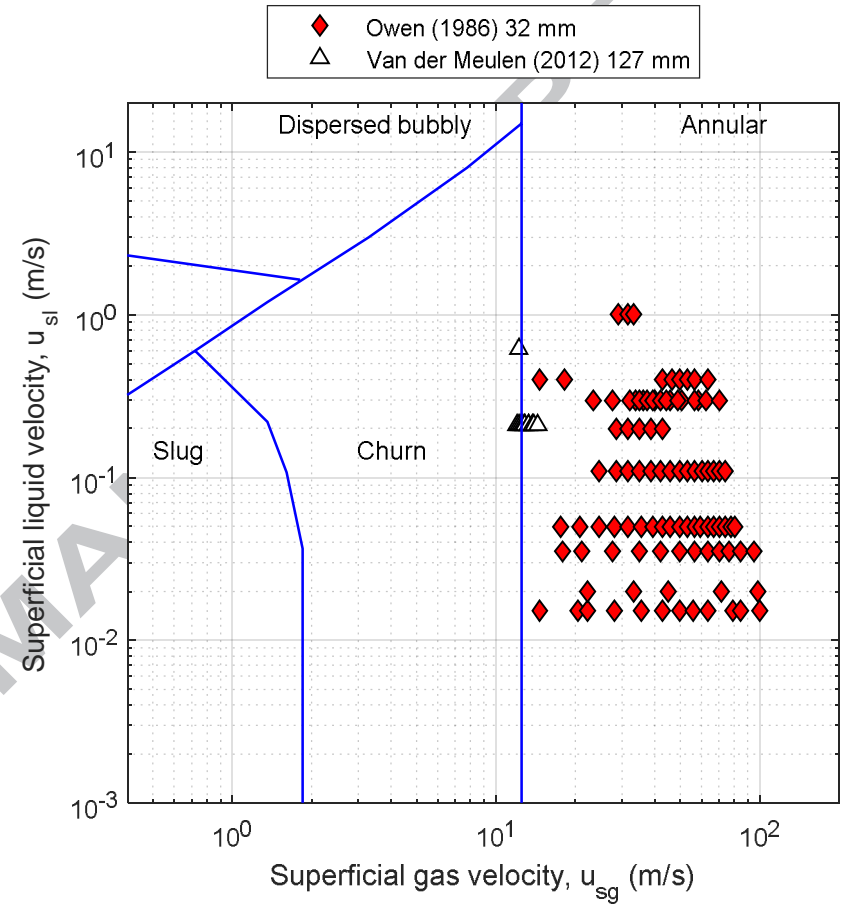


Figure 9: Control volume for the momentum balance in Equation Error! Reference source not found..  
Based on the schematic by Wongwises and Kongkiatwanitch [29]

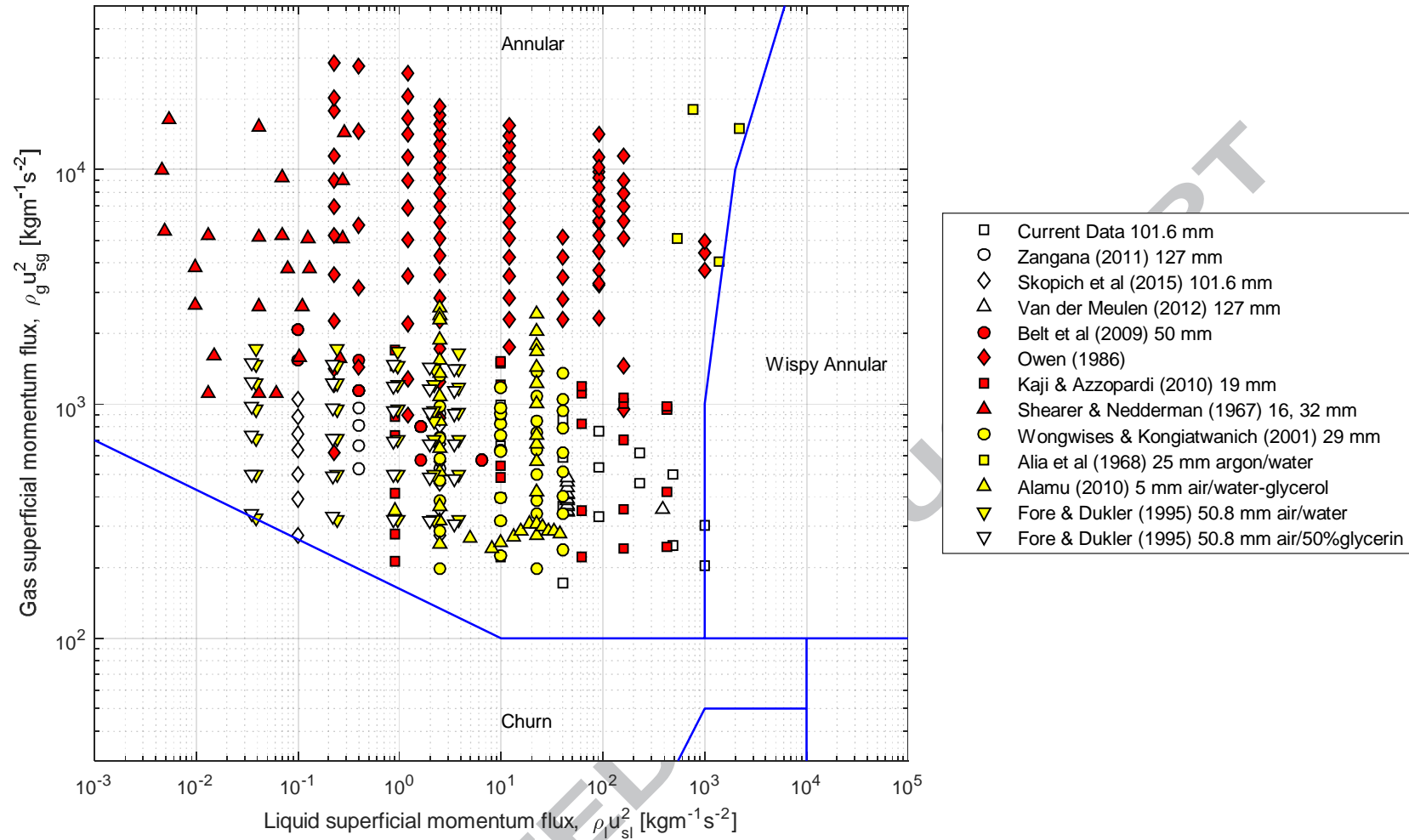


(a)



(b)

Figure 10: Experimental data against the Taitel et al. [59] flow regime map

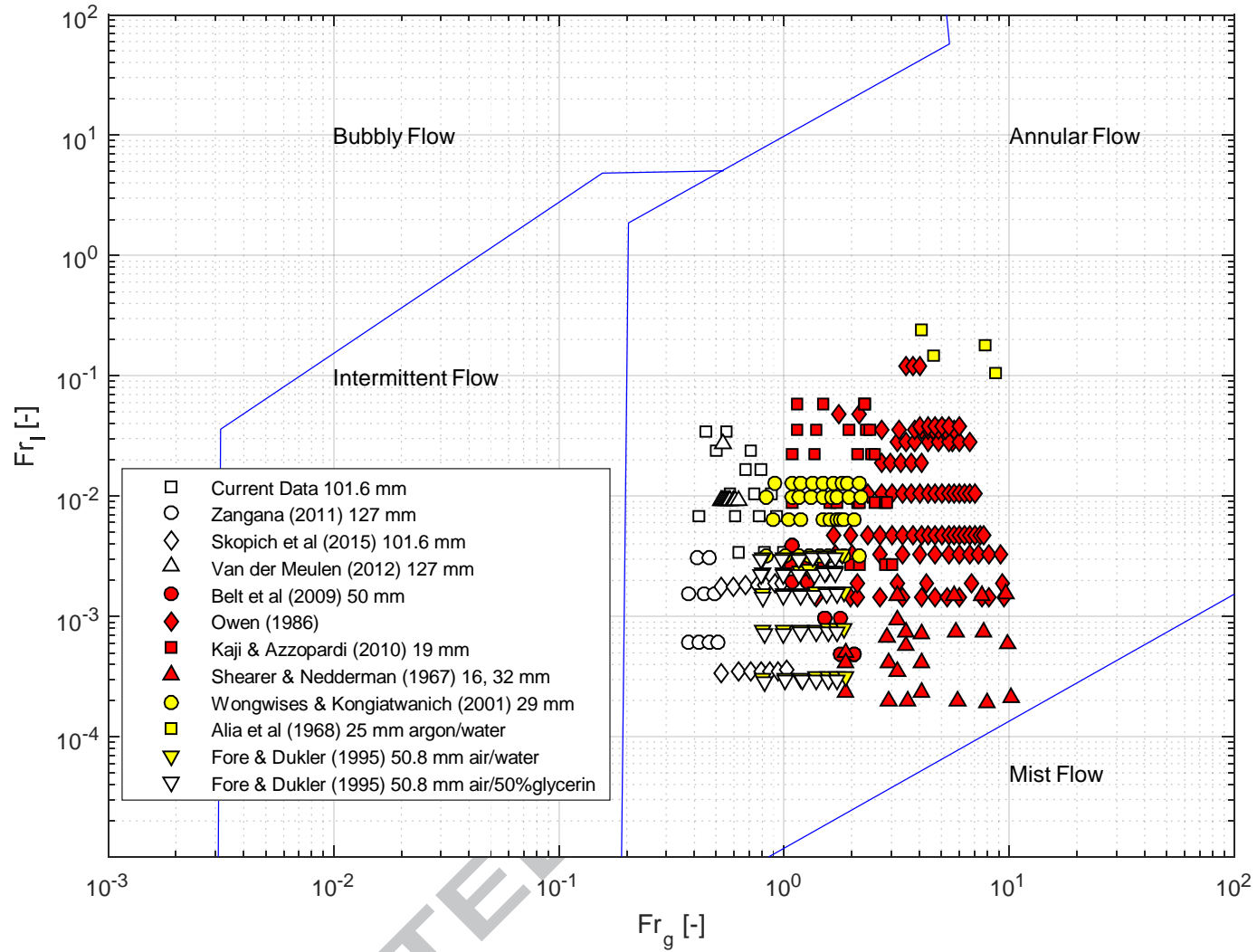


0

1

2

Figure 11: Experimental data bank against Hewitt and Roberts [60] flow regime map showing all data are in the annular flow regime



3

4

Figure 12: experimental databank against the Shell [62] flow regime map



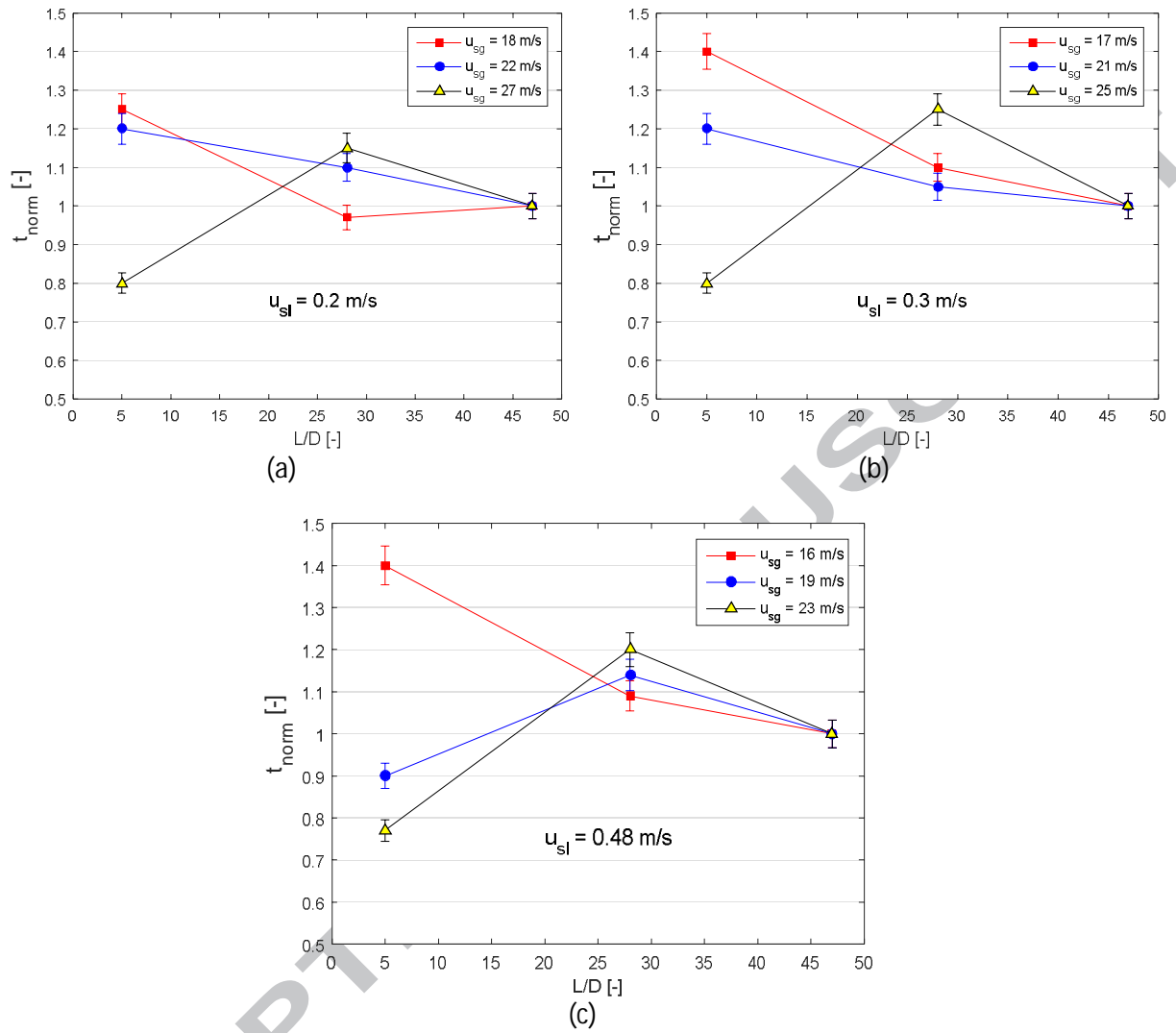


Figure 13: Normalised liquid film thickness axial development in the upwards flowing section of the rig  
(See Almbrok [37] for more)

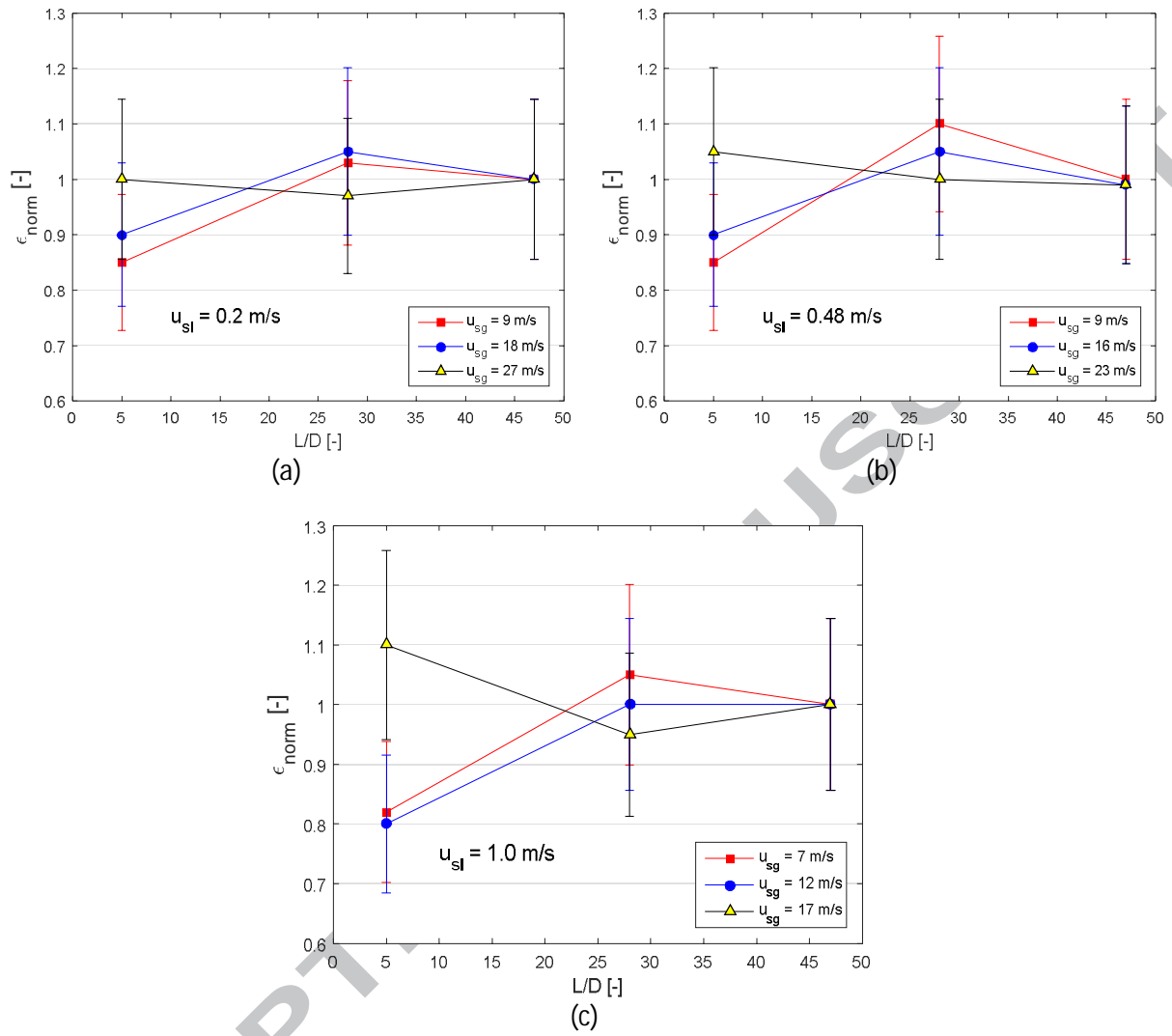
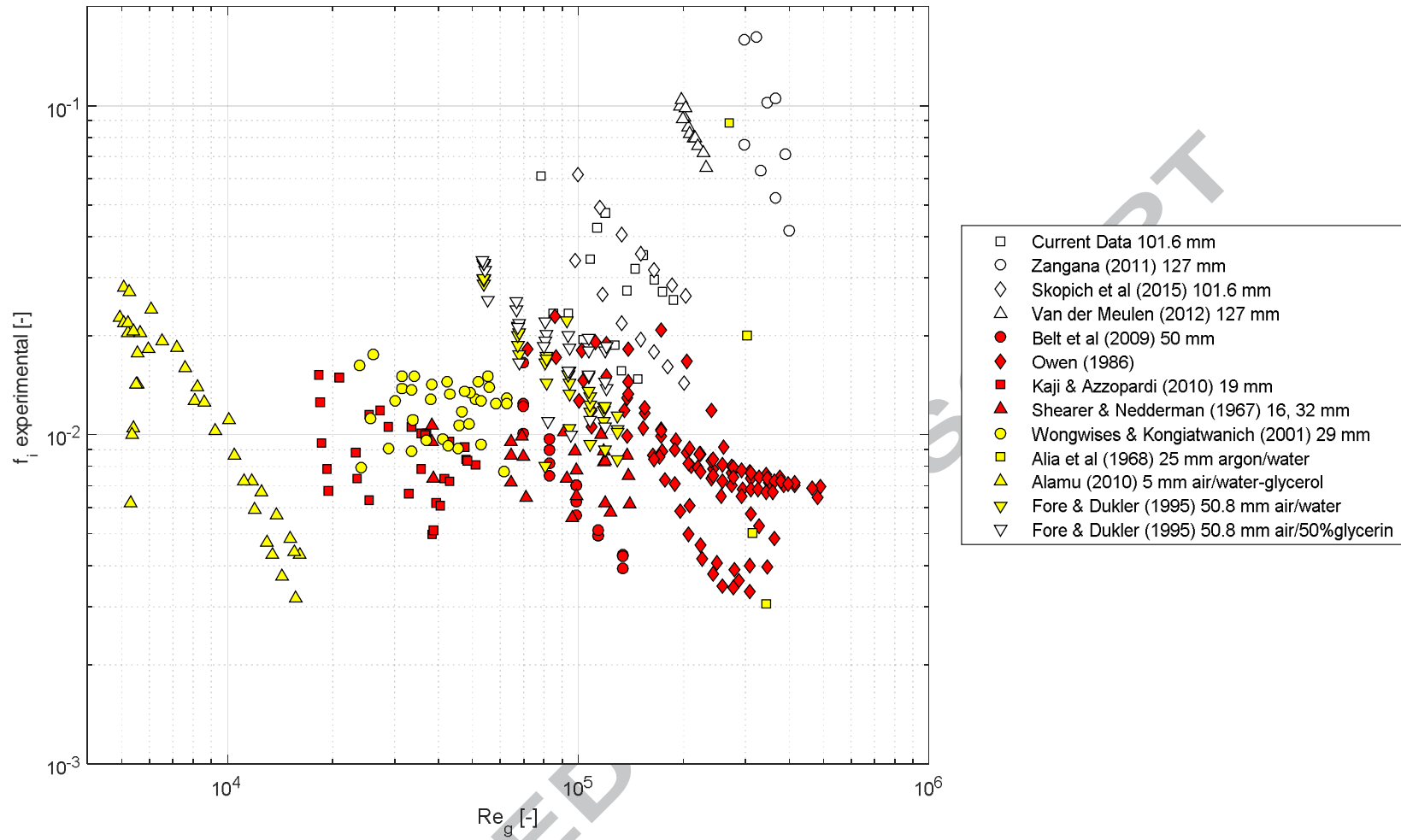


Figure 14: Normalised gas void fraction axial development in the upwards flowing section of the rig



5

6

**Figure 15: Experimental interfacial friction factors versus superficial gas Reynolds number**

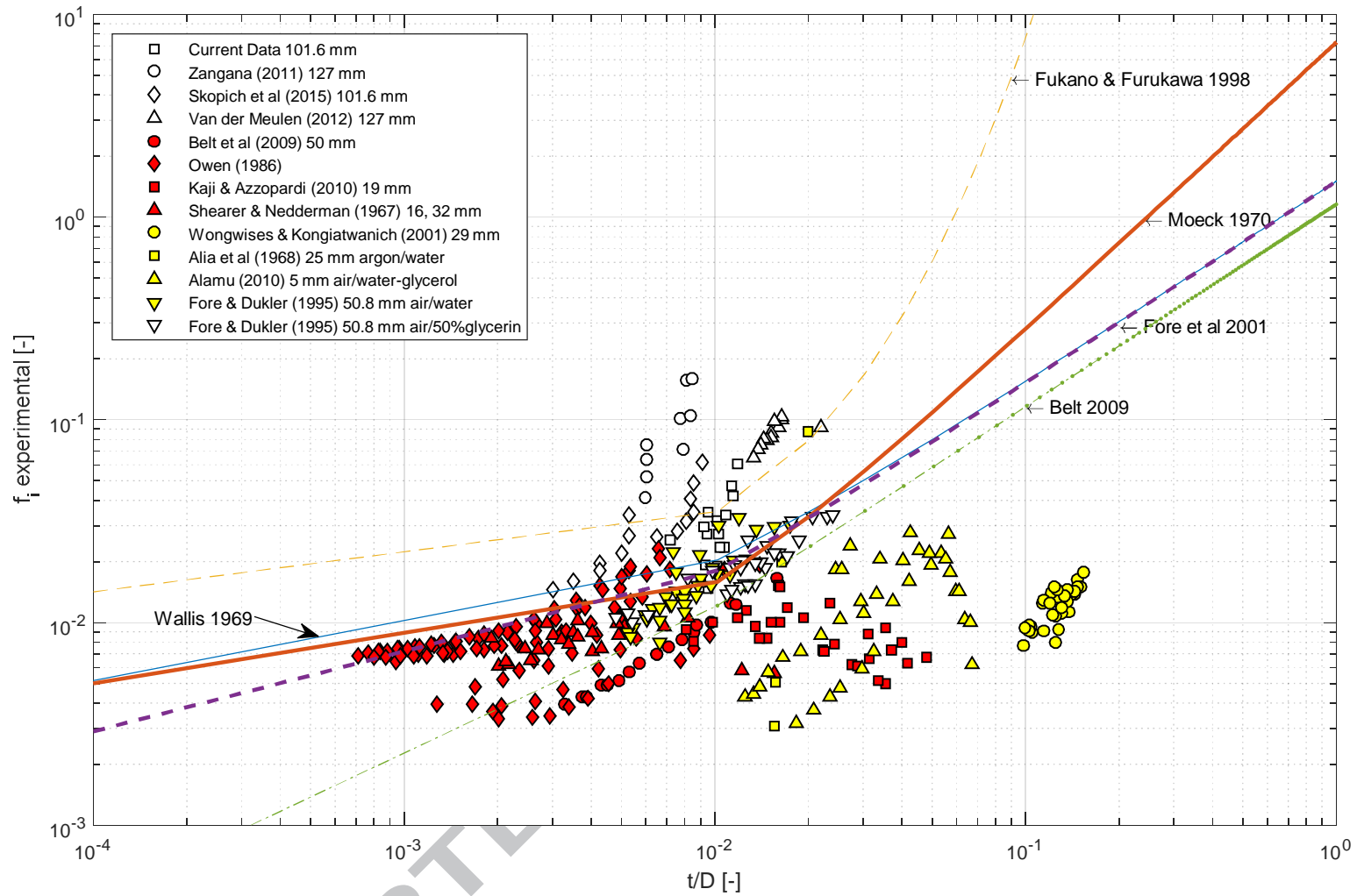
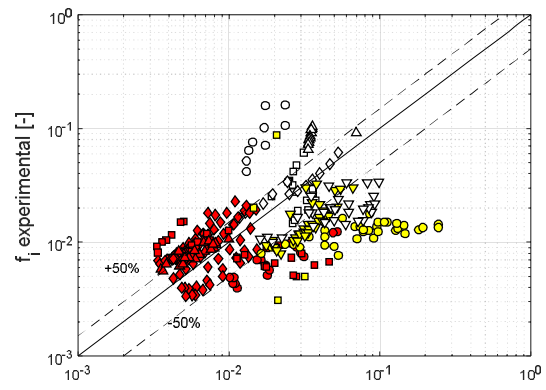
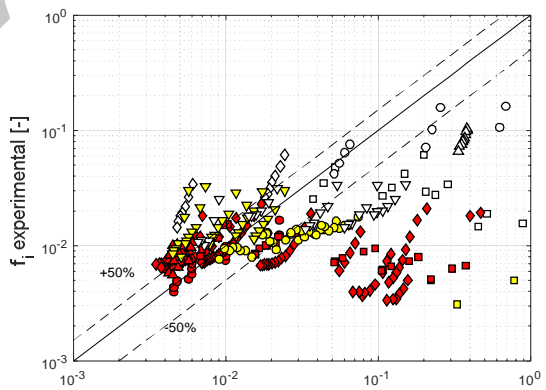
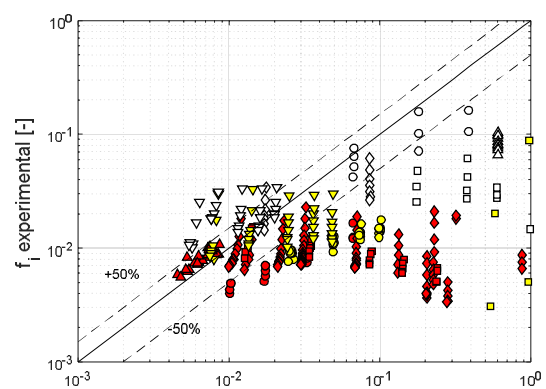
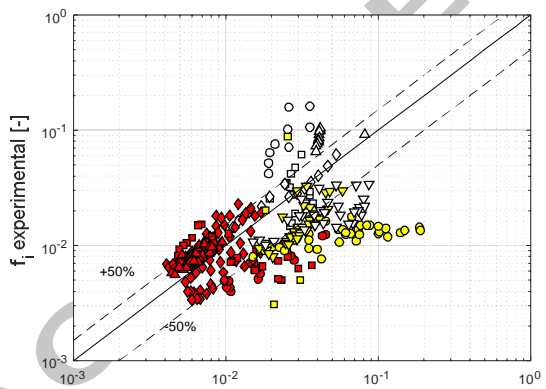
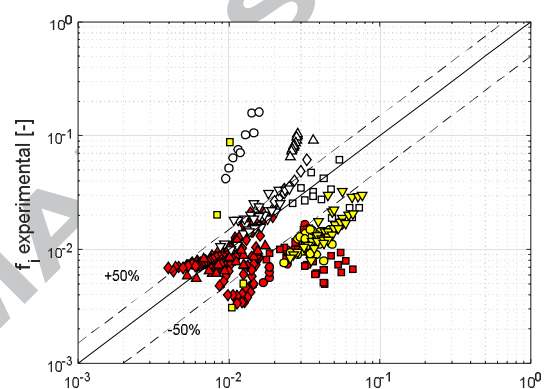
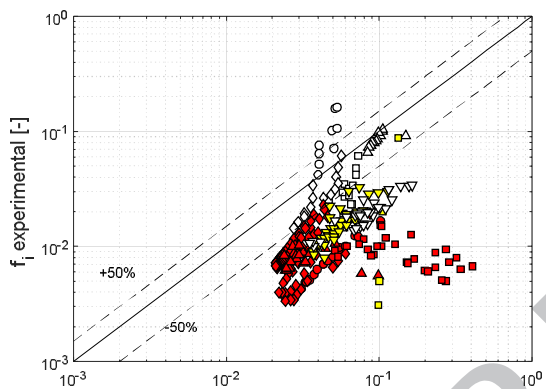
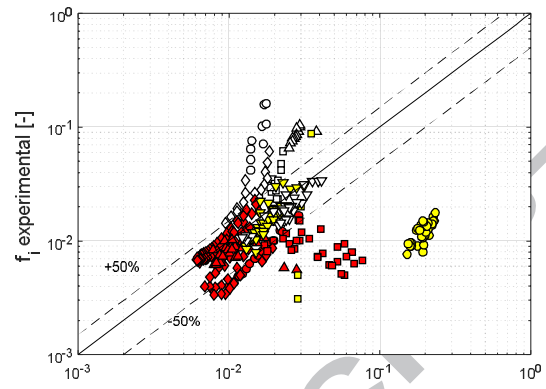
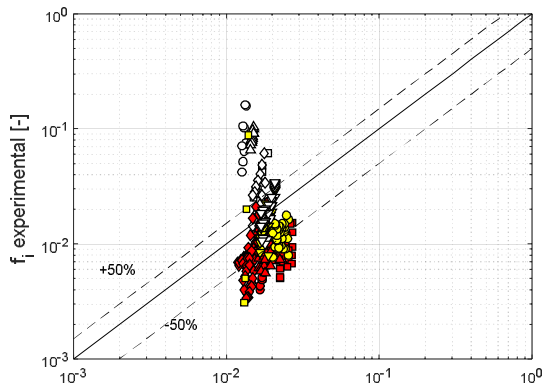


Figure 16: Comparison of current and other experimental friction factors with Wallis-type correlations



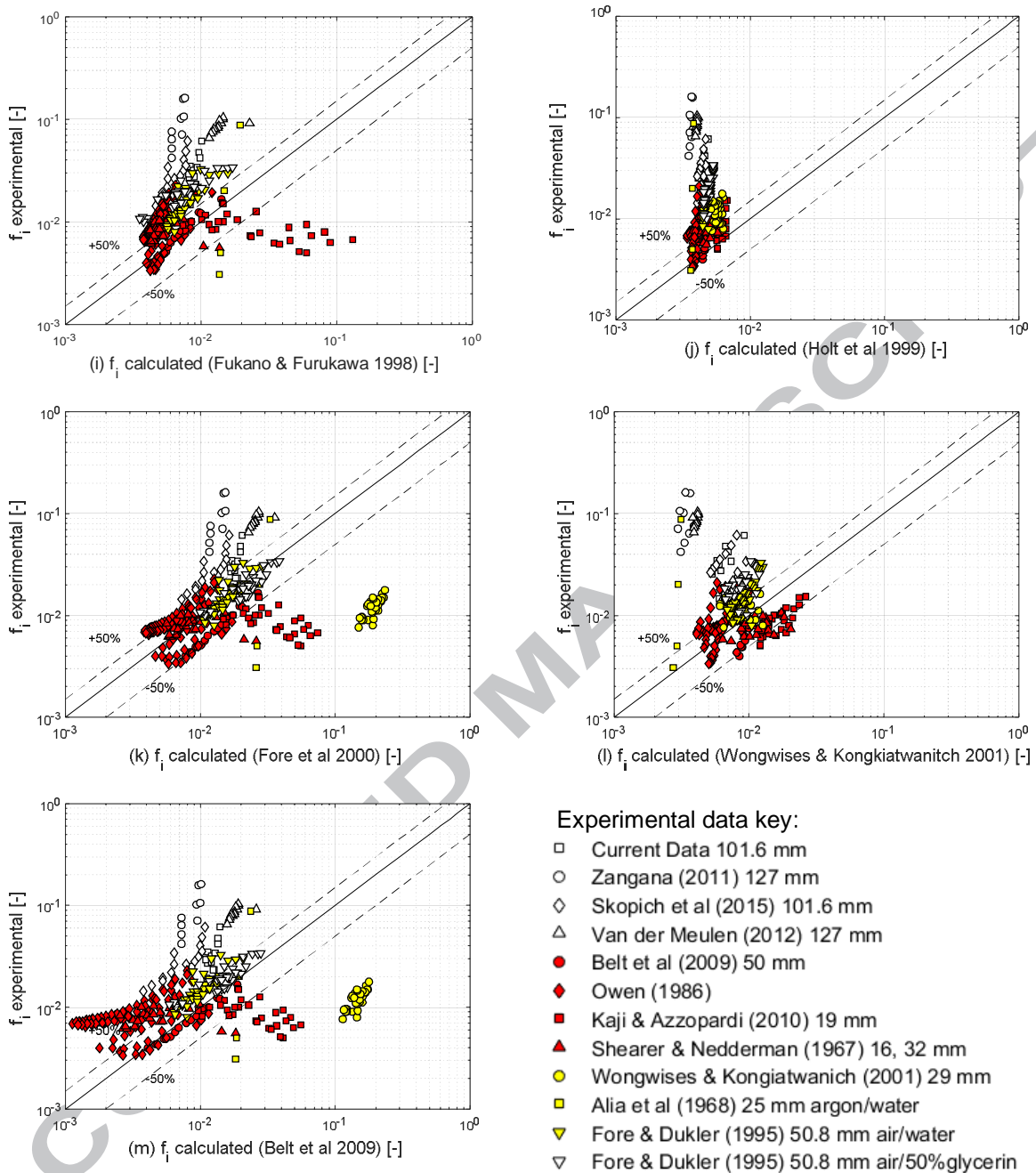


Figure 17: Comparison of 332 interfacial friction factor data points with predictions of various correlations

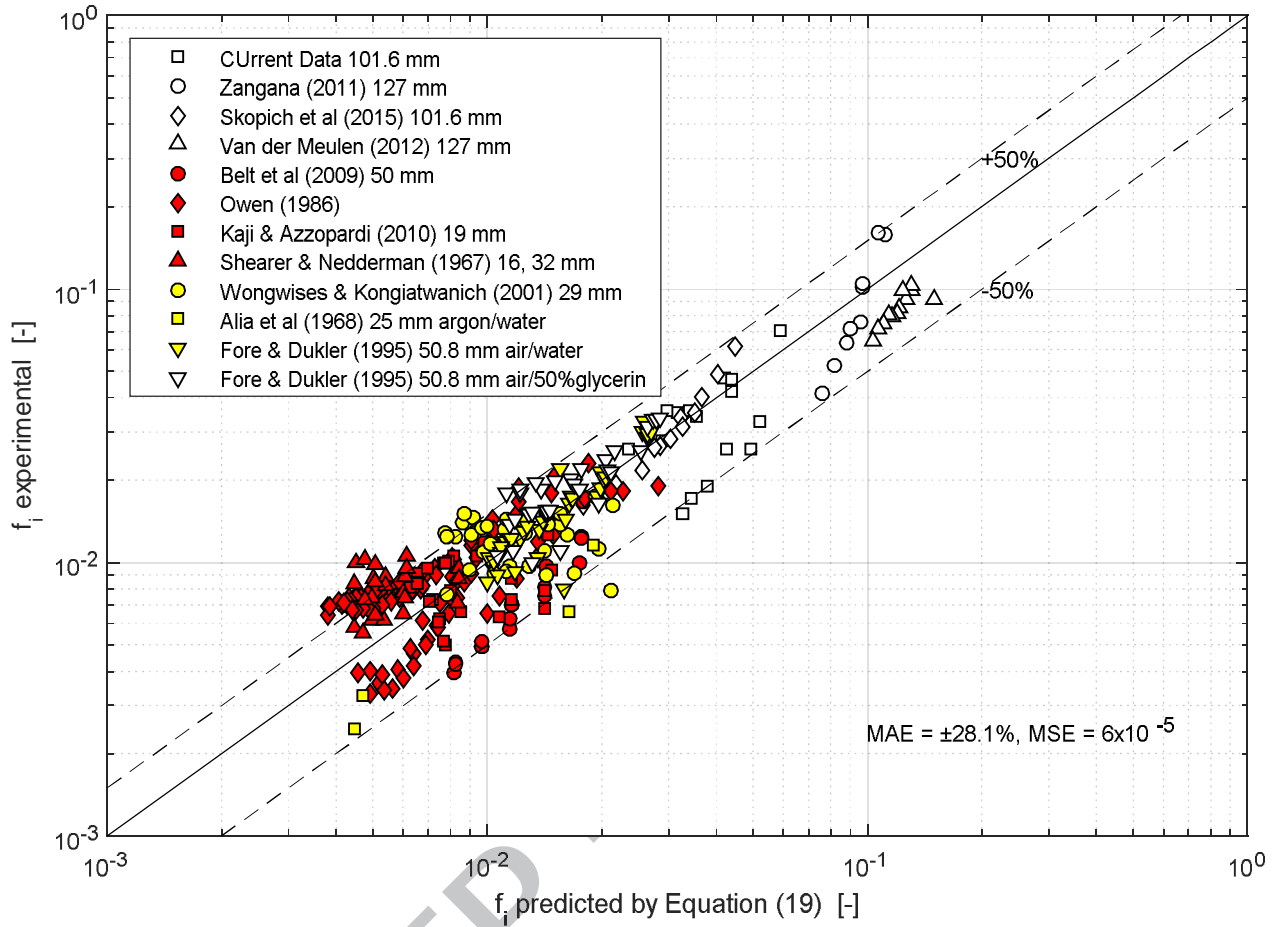


Figure 18: Predictions of proposed correlation compared with entire experimental database

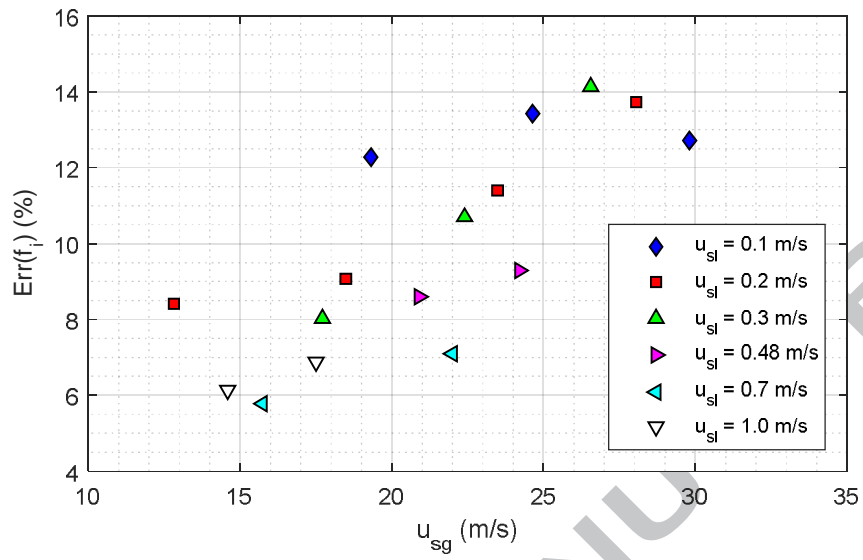


Figure A1: Experimental uncertainties in  $f_i$



## Tables

**Table 1: Reported interfacial friction factor correlations used for comparison with the current experimental data**

Author(s)	Equation	Remarks
Blasius	$f_i = 0.316 Re_g^{-0.25}$	Valid for $Re$ up to $10^5$
Wallis [2]	$f_i = 0.005[1 + 300t/D]$	Theoretical model
Moeck [3]	$f_i = 0.005[1 + 1458(t/D)^{1.42}]$	Pipe 24 mm, steam/water
Hori et al. [24]	$f_i = 1.13 Re_g^{-0.89} Re_l^{0.68} Fr_g^{0.25} Fr_l^{-0.45} (\mu_l/\mu_w)^{0.7}$ , $Fr_g = u_{sg}/\sqrt{gD}$ , $Re_g = \rho_g u_{sg} D/\mu_g$	Pipe 13, 19.8, 26 mm, $u_{sg} = 53$ – 56 m/s, $u_{sl} = 0.006$ – $0.0061$ m/s
Asali et al. [5]	$f_i = f_g [1 + Re_g^{-0.2} (Re_g \sqrt{f_i/2} t/D - 4)]$ , where $f_g = 0.316 Re_g^{-0.25}$	Pipe 22.9, 42 mm, air/water, air/glycerine, 1–2 bar system pressure
Fukano et al. [25]	$f_i = f_g [1 + 1 \times 10^{-5} Re_g^{0.7} Re_l^{0.65} X^{0.52}]$ , $X = \sqrt{\frac{\Delta P_{Lo}/\Delta L}{\Delta P_{Go}/\Delta L}}$ , $\Delta P_{Lo}/\Delta L = \frac{1}{2}(f_L/D)\rho_L u_{sl}^2$ , $\Delta P_{Go}/\Delta L = \frac{1}{2}(f_g/D)\rho_g u_{sg}^2$	Rectangular duct, 80×10 mm, air/water, 0.98–1 bar pressure, $Re_l = 10$ –300, $u_{sg} = 10$ –50 m/s
Ambrosini et al. [26]	$f_i = f_g [1 + 13.8 We^{0.2} Re_g^{-0.6} (t^+ - 200 \sqrt{\rho_g/\rho_l})]$ , for $\dot{m} > 100 \text{ kg m}^{-2} \text{ s}^{-1}$ , where $t^+ = 0.19 Re_{if}^{0.7}$	Pipe 10–42.2 mm, air/water, helium/water, air/various hydrocarbons, 0.2 – 1.9 bar
Fukano et al. [27]	$f_i = f_g [1 + 8.53 \times 10^{-4} X^{2.82} Re_g^2/Re_l]$	Pipe, 10, 16, 26 mm, 1.02 – 1.35 bar, $u_{sg} = 20$ – 60 m/s, $u_{sl} = 0.06$ – 0.1 m/s
Fukano and Furukawa [23]	$f_i = 0.425(12 + v_l/v_w)^{-1.33} (1 + 12t/D)^8$	Pipe 4.6 mm, air/water, air/glycerol, 1–1.2 bar, $u_{sg} = 10$ –50 m/s, $u_{sl} = 0.04$ –0.3 m/s
Holt et al. [28]	$f_i = f_g [1 + 13.8 We^{0.175} Re_g^{-0.7}]$ , for $\dot{m} > 100 \text{ kg m}^{-2} \text{ s}^{-1}$	Pipe 5, 10 mm, square duct 7.7×2.6 mm, trapezoidal duct, 2×7×4.4 mm, air/water, nitrogen/water, helium/water, air/glycerol, 0.2–1.5 bar pressure
Fore et al. [6]	$f_i = 0.005[1 + 300(t/D - 0.0015)]$	Rectangular duct, 101.6×5.08 mm, nitrogen/water, 3.4–17 bar
Wongwises and Kongkiatwanitch [29]	$f_i = 17.172 Re_g^{-0.768} (t/D)^{-0.253}$	Pipe 29 mm, air/water, 1 bar pressure
Belt et al. [1]	$f_i = 1.158t/D + 3.143 \times 10^{-4}$	Pipe 19 mm, air/water, 1 bar pressure

**Table 2: Experimental data for upward interfacial friction factor (all air/water except stated otherwise)**

S/No.	Authors	D [mm]	L/D [-]	Test pressure [bara]	$u_{sl}$ range [m/s]	$u_{sg}$ range [m/s]	Measurements made	Number of data points	Symbol
1	Current	101.6	46	1.0–1.4	0.1–1.0	10–29	$dP/dz, t, u_{lf}$	23	○
2	Zangana [30]	127	66	1.0	0.02–1.00	10–17	$dP/dz, \varepsilon, \tau_w$	9	⊙
3	Skopich et al. [31]	101.6	58 – 92	0.9–1.2	0.01–0.05	14–27	$dP/dz, \varepsilon$	14	◇
4	Van der Meulen [17]	127	87	2.0	0.02–0.70	10–17	$dP/dz, \varepsilon$	12	△
5	Belt et al. [1]	50	120, 140	1.0	0.01–0.08	22–42	$dP/dz, t$	18	•
6	Owen [32]	32	600	2.0–4.0	0.02–1.00	14–100	$dP/dz, e$	97	•
7	Kaji and Azzopardi [11]	19	300	1.2	0.03–0.65	10–34	$dP/dz, \varepsilon$	29	▪
8	Shearer and Nedderman [33]	16, 32	133–267	1.1	0.02–0.10	10–16	$dP/dz, t$	24	▲
9	Wongwises and Kongkiatwanitch [29]	29	41	1.0	0.05–0.2	9–34	$dP/dz, t$	35	●
10	Alia et al. [34]*	25	140	2–6	0.07–1.5	7–150	$dP/dz, t$	6	■
11	Fore and Dukler [35]	50.8	69	1.0	0.006–0.06	16–36	$dP/dz, t, e, \tau_w$	35	▼
12	Fore and Dukler [35]**	50.8	69	1.0	0.006–0.06	16–36	$dP/dz, t, e, \tau_w$	30	▽
Total								332	

\*Argon–Water, \*\* Air–50% glycerin, fluid combination

**Table 3: Averaged measured quantities for the air–water system in the upward section of the Serpent Rig**

Test Pressure (bara)	$u_{sl}$ (m/s)	$u_{sg}$ (m/s)	$t$ (mm)	$u_{lf}$ (m/s)	$dP/dz$ (Pa/m)
1.09	0.10	18.39	1.2	0.86	1039.2
1.13	0.10	23.66	1.0	1.00	1268.6
1.15	0.10	28.87	0.7	1.01	1592.2
1.10	0.20	12.08	1.2	0.78	1680.4
1.14	0.20	17.51	1.2	0.80	1419.6
1.18	0.20	22.41	1.0	1.02	1488.2
1.23	0.20	26.90	1.0	2.02	1576.5
1.13	0.30	11.70	1.3	0.74	1878.4
1.18	0.30	16.74	1.1	0.73	2505.9
1.23	0.30	21.33	1.0	1.03	2172.5
1.29	0.30	25.47	0.9	2.03	1890.2
1.17	0.48	11.16	1.2	0.98	1794.1
1.25	0.48	15.63	1.1	1.08	1839.2
1.32	0.48	19.62	1.0	1.10	2045.1
1.40	0.48	22.87	1.0	1.14	2441.2
1.23	0.70	10.50	1.1	0.89	3068.6
1.33	0.70	14.45	1.1	1.10	2452.9
1.42	0.70	17.91	1.0	1.12	2149.0
1.53	0.70	20.59	0.9	1.19	2145.1
1.32	1.00	9.65	1.2	0.94	2284.3
1.44	1.00	13.15	1.1	1.30	2772.5
1.56	1.00	15.98	1.0	1.37	3260.8
1.67	1.00	18.56	0.9	1.42	3937.3

**Table 4: Statistical comparison between predictions of proposed and previous correlations**

Correlation	Statistic	MSE*	MAE**	Percentage of points within $\pm 50\%$ error band
Proposed		0.00006	28.1	83.85
Blasius		0.00051	123.3	28.57
Wallis [2]		0.00412	259.7	49.69
Moeck [3]		0.27280	1814.1	6.83
Hori et al. [24]		0.00067	139.5	43.48
Asali et al. [5]		0.00099	115.2	49.38
Ambrosini et al. [26]		0.00157	143.0	44.72
Fukano et al. [27]		0.06933	678.6	46.89
Fukano & Furukawa [23]		5.59788	5468.3	45.03
Holt et al. [28]		0.00066	139.5	36.34
Fore et al. [6]		0.00404	231.6	57.76
Wongwises & Kongkiatwanich [29]		0.00062	49.6	54.97
Belt et al. [1]		0.00250	185.3	41.93

\*MSE is the Mean Square Error defined as  $\frac{1}{n} \sum_i^n (f_{i,exp} - f_{i,pred})^2$

\*\*MAE is the percentage Mean Absolute Error defined as  $\frac{1}{n} \sum_i^n \left| \frac{f_{i,exp} - f_{i,pred}}{f_{i,exp}} \right| \times 100$

**Highlights**

- Gas–liquid two-phase flow experiments conducted in a large diameter flow loop
- Data were collected for interfacial friction factor in upwards annular flow regimes
- Data were also gathered from other sources spanning both small and large diameter pipes
- Previous correlations' predictions deviated at high shear regions mainly for large pipes
- Improved correlation is proposed to fit the diverse database of more than 300 data points

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# Interfacial friction in upward annular gas–liquid two-phase flow in pipes

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2017-02-17

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