PRESSURE LOSS IN VALVES DURING HORIZONTAL TWO-PHASE FLOW

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ABSTRACT Theoretical pressure loss correlations for valves in two-phase flow differ by more than 400% compared to the experiment. In this paper, experimental data for pressure loss and flow pattern transition in fittings are presented and compared to established correlations. For this reason, a tansparent 50mm diameter ball valve has been tested in horizontal two-phase flows of oil, water and air. The emphasis was on stratified flow, slug flow and transitions to other flow patterns. at superficial velocities from 1 to 10 m/s for gas and 0.07 to 2 m/s for liquid. Pressure loss, gauge pressure, temperature and void fraction measurement was triggered by a video system. The results clearly show the strong influence of the fittings on phase distribution. Close to transition lines on flow pattern maps, this can lead to a change of the flow pattern and thus decrease the accuracy of the applied pressure loss correlations.

I. INTRODUCTION

Multi-phase pipelines become more and more important for oil production systems in technical and economic fields. Especially for offshore and badland applications it is a cheap alternative to connect a new well to an existing production manifold by a multi-phase pipeline, even for distances larger 20 kilometres. Mostly such a pipeline contains oil and gas with a wide range of the void fraction. The common flow patterns are transient like slug flow. Such long distance pipelines contain a large number of valves and fittings.

Two-phase flow through pipes is well known through experience in oil industry, beginning with Lockhart-Martinelli [1] and Baker [2], and in power plant design. However few publications are found that address two-phase flow through valves and orifices, which introduce more than non-transient flow patterns.

In this study two things should be evaluated. On the one hand several known pressure losses should be tested against the measured values for each liquid phase. On the other hand the influence of another fluid with higher viscosity should be evaluated, because many pressure loss correlations neglect the influence of viscosity.

II. THEORY OF TWO-PHASE FLOW

Calculation of two-phase pressure loss is normally done by applying a factor Φ^2 to the single-phase pressure loss

$$\Delta p_{TP} = \phi^2 \Delta p_{SP} \tag{1}$$

Single phase pressure loss is calculated for pipes by the well known

$$\Delta p = \frac{1}{2} \frac{l}{d} \lambda \varrho_f u^2 \tag{2}$$

where the friction factor λ is related to the Reynold's number. For laminar flow, in which the Reynold's number in a pipe is normally below 2300, λ is determined by

$$\lambda = \frac{64}{Re} \tag{3}$$

For turbulent flow, λ is calculated as

$$\lambda = \frac{0.3164}{Re^{1/4}} \tag{4}$$

These correlations assume, that velocities are far from the sonic velocity with λ not depending on the fluid properties.

In calculation of pressure loss through a pipe fitting the following correlation is typically employed.

$$\Delta p = \frac{1}{2} \xi \varrho_f u^2 \tag{5}$$

Where the pressure loss coefficient ξ is empirical and should be obtained by measurements in the same range of Reynold's number so that ξ remains nearly constant.

Two–phase flow models

There are two different models used in two–phase flow. One of them is the so called homogeneous model, that treats the two–phase flow as a single phase represented by mean properties of the two phases. This model assumes that the two phases have the same velocity. The advantage is that many flow conditions can be described by this model in a sufficient way, but sometimes it is difficult to calculate the properties with respect to physics. Normally they are based on the mass fractions.

The other one is called separated flow model. For some flow patterns like stratified flow it is obvious that a separate calculation of the pressure loss in gas and in liquid phase makes more sense. In this model each phase is treated alone connected only with frictional interaction between the phases. This model includes also thermodynamic equilibrium and constant, but not necessary equal, velocities.

III. EXISTING CORRELATIONS FOR COMPONENT PRESSURE LOSS

Fitzsimmons [3] conducted the first tests of pressure loss in water/steam flow through pipe components in 1964. Later investigations were performed by Heckle [4] in water/air systems. Heckle assumed that every two–phase flow through a valve can be described by a homogeneous flow, with fluid properties calculated from the conservation law that is the most important for this geometry. The difficulty is, that the correlations are only checked against his own measurements and local void fraction calculation is not practical.

Beattie [5] started from a separated flow model, which means gas and liquid phase are flowing separated from each other. The slip between the phases is implicit in the correlation.

$$\Delta p_{TP,B} = \Delta p_{lo}\left[1+\dot{x}\left(\frac{\varrho_l}{\varrho_g}-1\right)\right]^{0.8}\left[1+\dot{x}\left(\frac{\varrho_l\mu_g}{\varrho_g\mu_l}-1\right)\right]^{0.2} \quad (6)$$

Simpson [6] developed his correlation also from separated flow model, but neglecting the influence of viscosity.

$$\Delta p_{TP,S} = \Delta p_{lo}\left\{1+\dot{x}\left[\left(\frac{\varrho_l}{\varrho_g}\right)^{1/6}-1\right]\right\}^{0.8}\left\{1+\dot{x}\left[\left(\frac{\varrho_l}{\varrho_g}\right)^{5/6}-1\right]\right\} \quad (7)$$

Both equations are appropriate to stratified and slug flow according to these authors.

Fairhurst [7] based his correlation on the homogeneous flow model and a semiempirical separated flow model. He proposed a critical quality $\dot{x}_0$, which divides the calculation of the two–phase multiplier into the two models.

$$\dot{x}_0 = \left[\frac{\lambda_g/\lambda_l}{1-\varrho_g/\varrho_l}\right] \quad (8)$$

If $\dot{x}_0 > \dot{x}$

$$\Delta p_{TP,F} = \Delta p_{fo}\left[1-\frac{\lambda_g}{\lambda_l}\frac{\varrho_l}{\varrho_g}\left(\frac{\dot{x}}{2-\dot{x}}\right)\right] \quad (9)$$

and if $\dot{x}_0 < \dot{x}$

$$\Delta p_{TP,F} = \Delta p_{fo}\left[1+\dot{x}\left(\frac{\varrho_l}{\varrho_g}-1\right)\right] \quad (10)$$

Chisholm [8] calculates pressure loss in nozzles and valves under the assumption of constant slip s. His two–phase multiplier is defined as

$$\phi^2_{TP,C} = 1+\left(\frac{\varrho_l}{\varrho_g}\right)\left[\frac{1}{s}\dot{x}(1-\dot{x})+\dot{x}^2\right] \quad (11)$$

if $\rho_l >> \rho_g$.

IV. TEST SETUP

General Description

The experimental rig was built to study multi–phase pressure loss and flow pattern transition in pipe components with horizontal flow. A 2 inches small scale test loop was designed to study the behaviour of simple multi–phase systems under near atmospheric pressure. For visualisation of the flow in the test section all essential parts were made up of acrylic plastics.

In this study a cylindrical valve, simulating a ball valve, was installed in the test section. At an angle of zero degree the opening area of the valve is identical to that of the pipe. The test section is 2.06 m long, with an upstream and downstream length of 19 times the tube diameter of the valve. Absolute pressures were measured by wire strain based transducers, pressure loss by inductive transducers having a natural frequency of 1200 Hz, which is much higher than the possible slug frequency. The volume flow rate of water is measured by an inductive flow meter. However, due to the significant lower conductivity the mass flow rate of oil had to be measured by the pressure loss across a quarter circle nozzle. Air flow rate was measured integral by a calibrated gas counter. The liquid is forced through a loop by a centrifugal pump. The air is supplied by a compressor from 5 bars level reduced to the test pressure level. Air and liquid are mixed together in a jet pump and then flow first vertically for 3m (producing slug flow), then through a smooth bend joining a horizontal pipe to ensure the flow pattern is developed. Flow pattern and void fraction are described best with equations of [10], who has upstream more than 150 diameters to allow flow development. The measurements are then performed in the test section. The two-phase mixture is then separated in the storage vessel. The liquids used were plain water and Shell Ondina 15, a white oil with a viscosity of 40 mm²/s at 20°C.

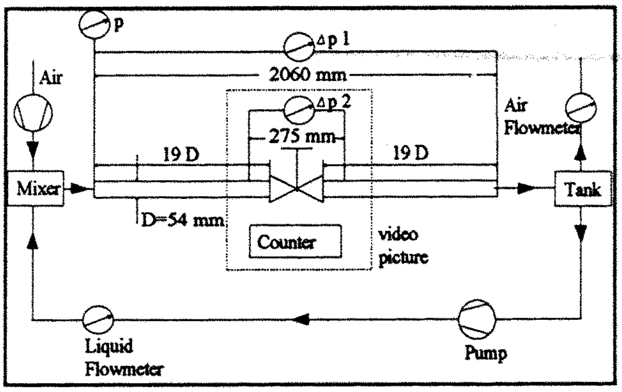

Fig.1 Test section and measurement setup

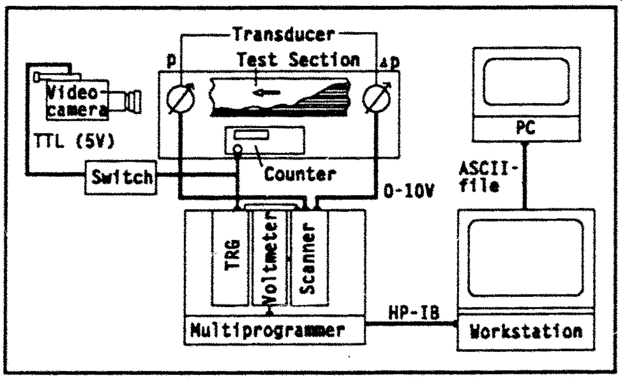

Fig.2 Plan of instrumentation including trigger path

Figure 1 shows the test section and the instrumentation. During a test run of 100 seconds the video camera takes 50 frames per second triggering each time of the data acquisition from all of the pressure taps (Fig.2). The absolute pressure tap was mounted at the beginning of the test section. The first differential transducer took its values across the complete section. The second one measures directly across the valve. The analog to digital converter has a sample rate of 100 kHz, which is much higher than the trigger frequency of 50 Hz, so that each set of values is taken at nearly the same time as the picture. To enumerate the pictures a counter is located right below the valve, triggered in the same way as the data acquisition. The measurement starts by switching the 5V-TTL 'picture ready' signal of the camera to the counter, where the 5000 sets of values will be automatically taken. The pictures show the valve, parts of the pipe up and downstream and the counter. On the pictures the flow pattern and the local void fraction can be determined. Video is much cheaper than film cameras and the results in difficult light environments are much better due to the sensitivity of the chip in the video camera. Exposure periods of 1/4000 second are no problem. This is important for filming opac mixtures of oil and gas at high velocities. By comparing the plots of pressure and pressure drop to the film pictures, an analysis and explanation of events can be obtained. This layout has the advantage of distinguishing 'heavy' wavy stratified flow from high frequency slug flow, because the pattern looks similar in the pressure history. The video pictures show clearly the slugs traveling at nearly the high velocity of the gas and the slow waves of the wavy stratified flow. Additionally to separate a very high wave, called pseudo slug, from a real slug in a flow regime where gas and liquid velocities have low differences is very hard with the video alone. The pressure loss shows exactly the high peak of the slug and the lower peak of the pseudo slug.

Void fraction in the videos was checked for single events with manually controlled digital image processing using a professional frame grabber card and the program 'IMAGE PRO'. Single pictures could be catched into the video memory of the grabber card. In the picture a rectangle with known length of each side is used as reference giving a number of pixels corresponding to the known dimensions. The height of the liquid is integrated over the whole picture assuming the surface of the liquid is horizontal in the direction of the camera view. The other assumption is that the velocities are low enough that all important changes of void fraction are captured by the video pictures. Fraction effects have been neglected for these particular evaluations, because often the border between the phases was near the axis of the pipe or the pipe was completely filled with liquid. The closing angle φ of the valve was fixed at 30 degrees corresponding to an opening ratio of 0.46 (Fig.3).

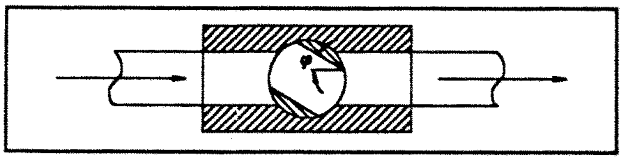

Fig.3 Cut through the ball valve with a closing angle φ of 30 degree

V. CONDUCTED MEASUREMENTS

The measurements were conducted over the region of slug flow and its lower borders on the superficial velocities as Fig. 4 shows, covering superficial velocities

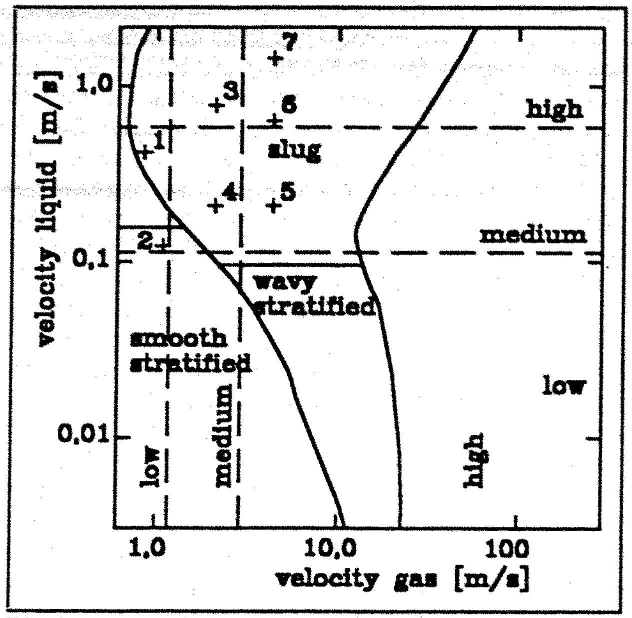

Fig.4 Measured points plotted on flow pattern map of Mandhane and Gregory [9] for horizontal water/air flow

up to 10 m per second for air and 4 meters per second for liquid. On three levels of air flow rate two water flow rates each were chosen on two lower levels and three on the higher with respect to the borders of slug flow area. These seven points characterised by the superficial velocities of gas and liquid phases were selected again, ensuring a constant void fraction for the oil–air two phase flow. Of course there would be other possibilities in selecting points with respect to the boundaries, describing flow pattern transition, which are changed by viscosity, but the uncertainties would be much higher in this case.

VI. RESULTS AND DISCUSSION

In this study two things should be evaluated. On the one hand several known pressure losses should be tested against the measured values for each liquid phase. On the other hand the influence of another fluid with higher viscosity should be evaluated, because many pressure loss correlations neglect the influence of viscosity.

Comparing gauge pressure plots slug detection is only possible in the water–air system, because the 40 times higher viscosity of the oil at a temperature of 40°C buffers the pressure waves. This buffering behaviour also moves the border of occuring slug flow to higher liquid and gas superficial velocities. If the introduced energy is high enough, e.g. gas superficial velocity 10 meters per second or more, the differences in the flow patterns and also differences in pressure loss are minimized.

At the low level of gas flow rate, Fig. 5 shows absolute pressure and Fig. 6 the pressure loss across the whole test section. This test has been performed with water and air. The corresponding test (gas–liquid ratio constant) with the oil and air system is shown in Fig. 7 and 8. The mean values are designated in the plot by a ▵ marker and are noted below them. The peak values are signed by a ▿.

It is easy to see that the water–air system forms orderly slugs marked by regular peaks in a characteristic shape. The plot of pressure loss shows only frequent bubbles moving through the test section. The pressure loss across the whole test section in the oil/air system is nearly twice as high as in the water/air system. The pressure loss across the valve is not so important, showing only an impression of how big is the influence of the valve on the local flow.

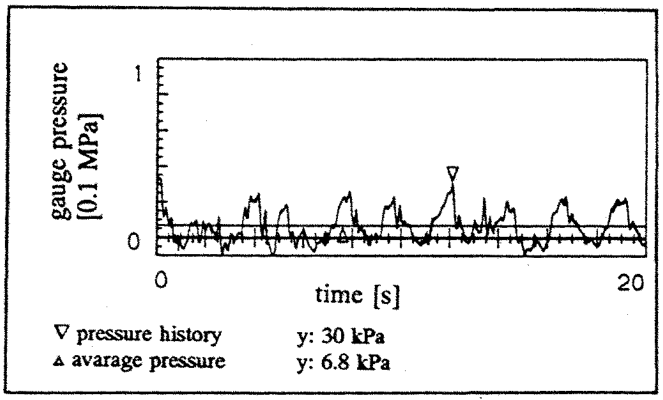

Fig.5 Plot of gauge pressure in the test section with water and air at low level gas volume rate (Point 1) and 30 degrees valve opening

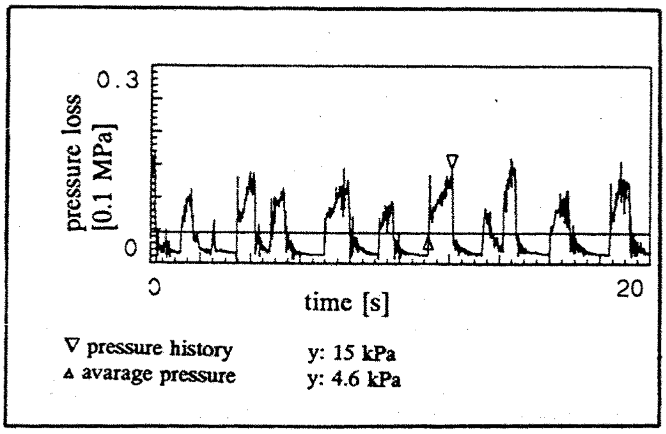

Fig.6 Plot of pressure loss across the test section with water and air at low level gas volume rate (Point 1) and 30 degrees valve opening

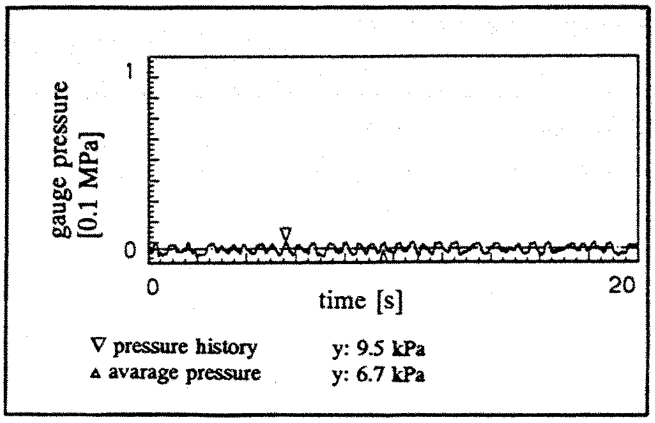

Fig.7 Plot of gauge pressure in the test section with oil and air at low level gas volume rate (Point 1) and 30 degrees valve opening

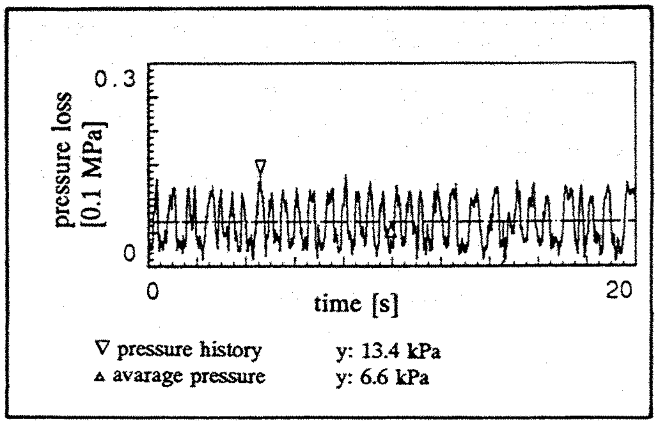

Fig.8 Plot of pressure loss across the test section with oil and air at low level gas volume rate (Point 1) and 30 degrees valve opening

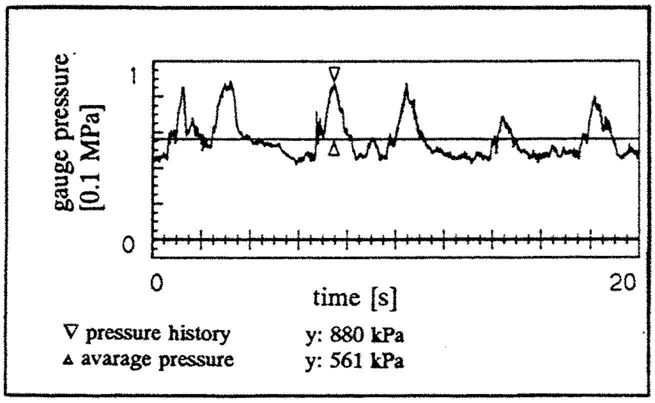

Fig.9 Plot of gauge pressure in the test section with water and air at medium level gas volume rate (Point 5) and 30 degrees valve opening

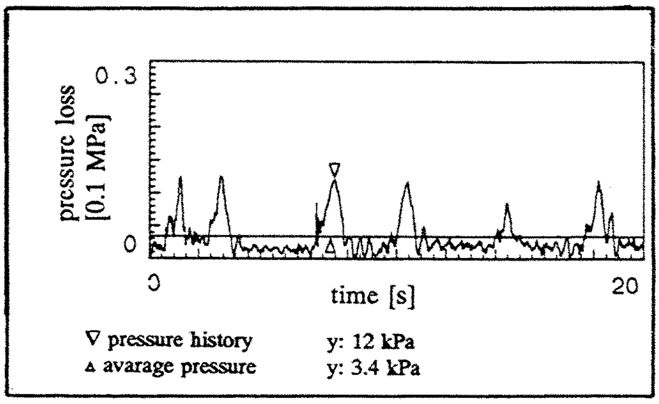

Fig.10 Plot of pressure loss across the test section with water and air at medium level gas volume rate (Point 5) and 30 degrees valve opening

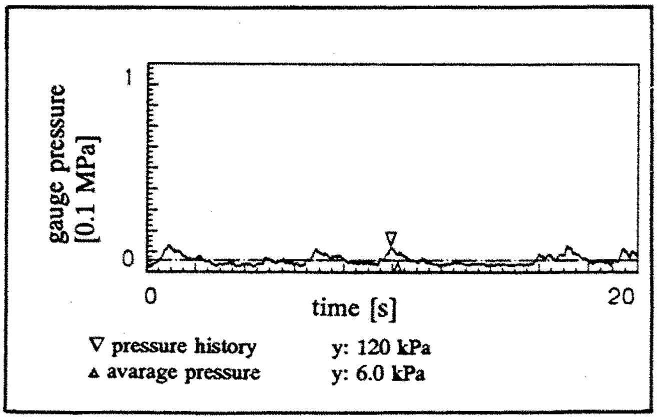

Fig.11 Plot of gauge pressure in the test section with oil and air at medium level gas volume rate (Point 5) and 30 degrees valve opening

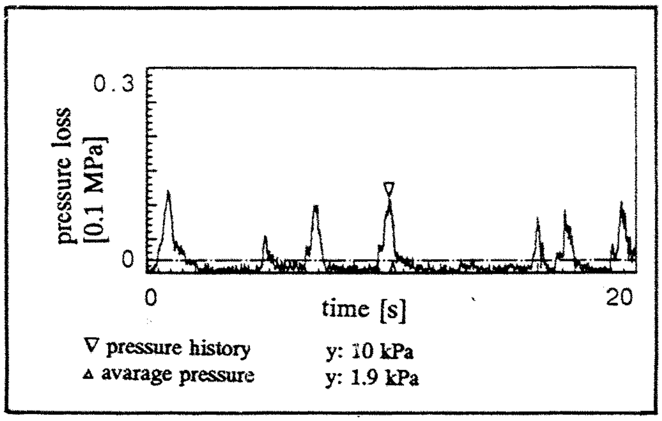

Fig.12 Plot of pressure loss across the test section with oil and air at medium level gas volume rate (Point 5) and 30 degrees valve opening

The figures 9 to 12 show both two-phase systems, but at a higher liquid flow rate. Additionally the measured pressure loss across the valve is shown. The gas flow rate is kept constant, whilst the liquid flow rate is twice as high. In principle both set of plots are similar. The only difference is that the oil peaks aren't as sharp as the water peaks. However, the slug frequency in the water-air system is a little higher, but the over all pressure loss of differs. This shows clearly, that at comparable flow pattern the pressure loss depends not on the viscosity, but is sensitive to density ratio. If e.g. the gauge pressure is 50% higher, pressure loss also raises in the same range. Although if the flow patterns are different for two systems as shown in Fig. 5 to 8 some assumptions based on either flow pattern lead to unreliable predictions in the system with the other flow pattern.

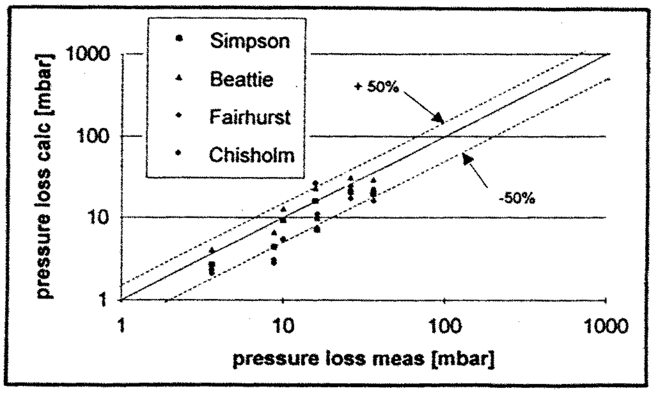

Fig.13 Comparison between measured and calculated pressure drops by the shown correlations at an opening angle of 0 degrees valve opening for the oil/air-system

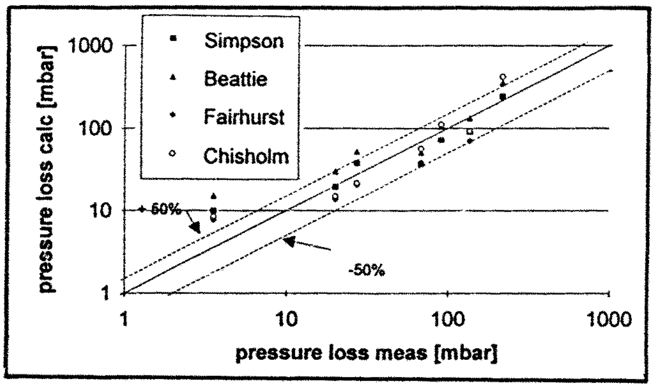

Fig.14 Comparison between measured and calculated pressure drops by the shown correlations at an opening angle of 30 degrees valve opening for the oil/air-system

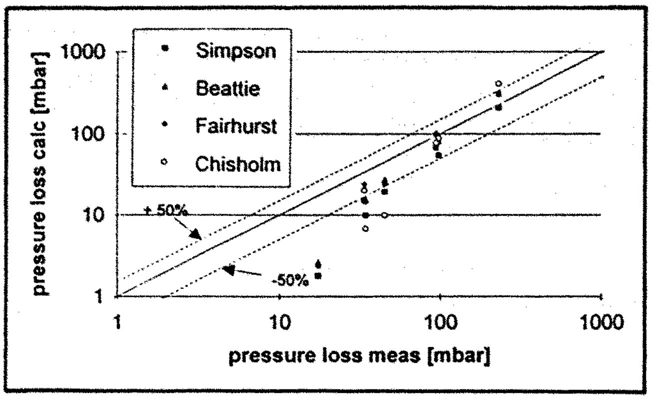

Fig.15 Comparison between measured and calculated pressure drops by the shown correlations at an opening angle of 30 degrees valve opening for the water/air-system

Figures 13, 14 and 15 compare prediction accuracy of the correlations discussed above. Fairhurst's prediction is consistantly too high. It is remarkable that values calculated by the Simpson correlation are in most cases close to those of Beattie, but on the higher side. Beginning with the water-air system, Simpson's predictions are closer to the measured values. This continues in the oil-air system with slightly more advantages on the Beattie correlation. Also Chisholm's results seems to fit the measured data best for 30°, but only if Chisholm own slip Finally the correlations of Beattie and Simpson could be both recommended in this region of flow pattern.

VII. SUMMARY

Pressure loss measurements at different ranges of two-phase slug flow through a modelled 2 inches ball valve have been performed. Mixtures used were water-air and oil-air. Results were tested against several pressure loss correlations and the influence of the change of viscosity studied. Although there exists instability in the transient two-phase flow results of recalculation the measured values are still adequate for the equation of Simpson in higher flow rates. If the flow patterns from water-air and oil-air are comparable, the influence of viscosity is low. However if the flow patterns are different the model based on either flow pattern fails to reformulate.

VIII. LITERATURE

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IX. NOMENCLATURE

d	diameter	m
l	length	m
$\dot{m}$	mass flow rate	kg/s
Δp	pressure loss	Pa
Re	Reynold's number	–
s	slip [8]	–
$\dot{V}$	volumetric flow rate	m^3/s
u	velocity	m/s
ε	void fraction	m^3/m^3
$\dot{x}$	quality	kg/kg
λ	friction factor	–
μ	viscosity	Pa s
φ	closing angle of ball valve	degrees
ρ	density	kg/m^3
ξ	pressure loss coefficients	–
Φ^2	two-phase multiplier	–

Subscripts

l	liquid phase
g	gas phase
lo	liquid only
h	homogenious
SP	single phase
TP	two-phase
B	Beattie
S	Simpson
F	Fairhurst
C	Chisholm