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Physical Modelling of Multi-Phase Flow

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Editors: H.S. Stephens
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PAPER A1

COMPONENT PRESSURE LOSS DURING TWO-PHASE FLOW

C.P. Fairhurst, BSc(Eng), MSc.

BHRA Fluid Engineering

Cranfield, Bedford MK43 OAJ, England

Summary

The reasons for studying two-phase pressure losses are introduced, and the main areas of interest discussed. An analytical approach to two-phase pressure loss is then developed by extending the single-phase method to a separated two-phase flow. This approach is then applied to the prediction of abrupt component pressure change. Theoretical models are then compared with experimental results obtained with air/water mixtures in both horizontal and vertical pipes of approximately 25 mm diameter, and operating at close to atmospheric conditions. A semi-empirical design method is presented and future development of the analytical approach discussed.

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NOMENCLATURE

Notation	Definition	SI Units
A	Flow cross-section area	m^2
A_R	Area ratio	-
C_C	Contraction coefficient	-
D	Internal pipe diameter	m
f	Friction factor	-
G	Mass velocity	kg/sm^2
K	Slip ratio, loss coefficient	-
$\dot{m}$	Mass flow	kg/s
P	Pressure	N/m^2
Re	Reynolds number	-
S	Perimeter	m
U	Velocity	m/s
x	Dryness fraction	-
z	Length	m
α	Void fraction	-
ΔP	Pressure change	N/m^2
ε	Roughness value	-
μ	Viscosity	Ns/m^2
ρ	Density	kg/m
τ_o	Wall shear stress	N/m^2
ϕ^2	Two-phase multiplier	-

Subscript	Definition
g	Gas
go	Gas only
h	Homogeneous or zero slip conditions
l	Liquid
lo	Liquid only
s	Maximum slip conditions
tp	Two-phase

1. INTRODUCTION

Two-phase flows occur by design in many engineering applications, particularly in equipment related to oil, chemical, process and power generation industries. Such flows may also occur following a departure from normal operating conditions in single-phase plant. All flows are essentially multi-phase in nature, however the degree to which this effects plant design and operation may vary. Hence an understanding of multiphase phenomena is essential and consequently much research activity has been directed towards this aim. There are, however, important areas which have received little attention. One of these areas is the study of pressure losses attributable to two-phase flow through components in pipes of varying orientation.

2. TWO-PHASE FLOW IN INDUSTRY

The oil and gas industries experience two-phase and multi-phase problems associated with hydrocarbon recovery. For example, oil, gas, and water mixtures may become emulsified in well and riser sections as a result of the increased turbulence produced by the expanding gas phase. Treatment of the emulsified mixture involves costly separation plant which is required to be of compact design. Considerable economies may be achieved by pumping gas and hydrocarbon condensates ashore in the same pipeline, particularly with increasing construction costs related to longer and deeper pipeline projects. The design of such pipelines and handling equipment requires the accurate prediction of pressure drop and liquid holdup, allied with an indepth understanding of the complex interactions between the phases. Such information has not been readily available in the past, and designers have had to rely heavily on judgement in sizing such facilities. Inevitably this approach has led to expensive over-design in the past which is unacceptable in future designs. Multi-phase flows may be set up when gas alone is transported, since condensation may occur. Such pipelines often incorporate slopes, and slugs develop producing undesirable pressure surge and transient flows which also reduce flowmeter integrity as well as requiring separation plant and condensate handling systems such as line pigging.

Conventional power generation plant often experiences two-phase flow in boilers, condensers and other associated heat exchange equipment. Understanding the behaviour of such flows is essential to the design of this equipment, requiring a knowledge of two-phase pressure drop, heat transfer, flow distribution and vibration phenomena. This knowledge has also progressed in parallel with the nuclear power programme and in particular with the development of Boiling Water and Pressurised Water Reactors, where a knowledge of two-phase thermo-hydraulic behaviour is essential to system design and hazards analysis. The systems usually contain straight pipes, valves, abrupt changes of section and bends, together with associated heat exchangers, condensers and boilers.

Chemical and process plants often incorporate devices and systems which operate under two-phase conditions. Such flows are used to advantage in gas lift reactors and ejectors in order to pump fluids and solids. Heating and refrigeration also produce evaporating and condensing two-phase mixtures, these having a significant effect on the performance of throttling devices, flow meters and distributing components such as tees and headers. It may also be necessary to design systems in order to avoid undesirable flow regimes which could give rise to operational and control problems.

Automatic control following an accident often includes two-phase mixtures which may not be present under normal operation, but require consideration in a hazards analysis where it may be required to estimate flow rates and loss of toxic mixtures under critical conditions.

3. THEORETICAL APPROACH TO TWO-PHASE FLOWS

Although the basic laws of fluid mechanics are applicable to two-phase systems, the calculations are complicated by the interactions between the phases. The flow is no longer wholly surrounded by a rigid pipe wall boundary, but includes the mobile interface between the phases, allowing mixing, evaporation, and condensation to take place. For this reason a semi-empirical approach is often adopted using simplified models to represent the flow in conjunction with correlations based on experimental results.

Three basic models are used, these being the homogeneous, separated and flow pattern models.

3.1 The Homogeneous Model.

The homogeneous model assumes that the two phases are interspersed such that they may be represented by a single-phase fluid having properties derived from those of each component phase. The assumptions inherent in this model are:

(1) Equal flow velocities of the liquid and gas phase.

(2) Thermodynamic equilibrium has been established between the phases.

With the homogeneous model, the single-phase equations of fluid mechanics are written using the modified fluid properties ρ_h, μ_h, u_h.

where $\dfrac{1}{\rho_h} = \dfrac{x}{\rho_g} + \dfrac{(1-x)}{\rho_\ell}$, similarly

$$\frac{1}{\mu_h} = \frac{x}{\mu_g} + \frac{(1-x)}{\mu_\ell}, \text{ and}$$

$$U_1 = U_g = U_h = \frac{\dot{m}}{A\rho_h}$$

which is Isbin's description of homogeneous viscosity, although different definitions have been put forward by Owens and Dukler, for example.

The two-phase friction factor, f_{tp}, is obtained using the single-phase curves and substituting the homogeneous fluid properties into the equations, i.e. for a circular pipe of diameter D, this gives:

$$\text{Pressure gradient due to friction} = \frac{2f_{tp}G^2}{D\rho_h}.$$

3.2 The Separated Flow Model.

The separated flow model assumes that the two phases are separated such that a set of equations may be written for each phase, hence requiring a knowledge of the frictional interaction between the phases and the flow cross-section occupied by each phase.

Empirical information is used to determine the two-phase quantities since they are not easily measured and are usually unknown at the region of interest. The model also assumes the attainment of thermodynamic equilibrium between the phases, and constant, but not necessarily equal, velocities for each phase.

Using the separated flow model the two-phase frictional gradient may be expressed in terms of the single-phase gradient calculated assuming that the total flow is that of the liquid phase, multiplied by a two-phase multiplier $\phi^2_{\ell o}$.

3.3 The Flow Pattern Model.

The flow pattern model assumes that the two-phase flow exists in one of several configurations, the basic equations of fluid mechanics are then solved using assumptions based on each of the idealised flow patterns. Implicit in this method is a knowledge of the particular flow regime which must be obtained by visual observation or some more sophisticated technique.

Of the methods discussed above the homogeneous method is the simplest to use, however its accuracy is limited. The separated flow model may be more accurate but is complicated by the semi-empirical nature of the solution. The flow pattern model is the most recent approach,and development is underway in order to improve its prediction.

3.4 Straight Pipe Development.

The analysis of two-phase experimental data may be simplified by the use of a two-phase multiplier. As an example of this approach, consider the development of the two-phase multiplier for the frictional pressure drop component of a straight pipe of constant area. For the stream tube of single phase liquid in Fig. 1 a force balance for the element gives:-

Pressure force + force due to wall shear stress + gravitational force = change in momentum

i.e. $$\frac{dP}{dZ}\,\delta ZA + \tau_o\,\delta ZS + A\delta Z\rho g\sin\theta = -\frac{d}{dZ}(\dot{m}u)\;\delta Z$$

$$\therefore\quad \frac{-dP}{dZ} = \frac{S}{A}\tau_o + \rho g\sin\theta + \frac{\dot{m}^2}{A}\frac{d}{dZ}\left(\frac{1}{\rho}\right)$$

The frictional component is the result of the wall shear stress, τo, which depends on the friction factor, f. For similar cross-sectional shape

$$f = fn(Re,\varepsilon) \text{ and } Re = fn(\rho,u,d,\mu)$$

also $$\tau o = f\,\frac{1}{2}\,\frac{\dot{m}^2}{\rho}$$

Hence the shear stress depends on the single phase fluid properties of density and viscosity. This relationship is complicated when a second phase is introduced. Consider the separated two-phase model illustrated in Fig. 2.

For a force balance

$$\frac{dP}{dZ}\,\delta ZA + \tau_o\delta ZS + (\rho_g A_g + \rho_l A_l)g\sin\theta\;\delta Z = -\frac{d}{dZ}(\dot{m}_g u_g + \dot{m}_l u_l)\;\delta Z$$

Using continuity relations gives:

$$-\frac{dP}{dZ} = \frac{S\tau_o}{A} + g\sin\theta\,(\alpha\rho_g + (1-\alpha)\rho_l) + \frac{\dot{m}^2}{A}\frac{d}{dZ}\left(\frac{x^2}{\alpha\rho_g} + \frac{(1-x)^2}{(1-\alpha)\rho_l}\right)$$

where dryness fraction $x = \dfrac{\dot{m}_g}{\dot{m}_g + \dot{m}_l}$

void fraction $\alpha = \dfrac{A_g}{A_g + A_l}$

The gravitational and accelerational components of pressure change may be calculated from a knowledge of the void fraction. The friction component requires a different approach.

For a separated flow: $f_{tp} = f(Re,\varepsilon)_{tp}$ and $Re = fn(\rho,u,d,\mu)_{tp}$

also $\tau o = f_{tp} \dfrac{1}{2} \dfrac{\dot{m}^2}{\rho_{tp}}$

Problems arise in defining the two-phase fluid properties ρ_{tp} and μ_{tp}. The usual approach is as follows:

$$\left(\frac{-dP}{dZ}\right)_f = \frac{S\tau o}{A} = \frac{4f_{tp}}{D} \cdot \frac{1}{2} \frac{\dot{m}^2}{\rho_{tp}}$$

$$= \frac{4}{D} f_{lo} \frac{1}{2} \frac{\dot{m}^2}{\rho_l} \times \frac{f_{tp}}{f_{lo}} \cdot \frac{\rho_l}{\rho_{tp}}$$

$$\therefore \quad \left(\frac{-dP}{dZ}\right)_f = \phi^2_{lo} \left(\frac{-dP}{dZ}\right)_{lo}$$

Hence the liquid only two-phase multiplier is given by:

$$\phi^2_{lo} = \frac{f_{tp}\,\rho_l}{f_{lo}\,\rho_{tp}}$$

this form provides simple limits since

for $x = 0$, $\phi^2_{lo} = 1$

for $x = 1$, $\phi^2_{lo} = \dfrac{f_{go}\,\rho_l}{f_{lo}\,\rho_g}$

experiments are undertaken to determine the form of ϕ^2_{lo} enabling the single-phase calculations to be corrected for the two-phase effects. This approach is also used for abrupt component pressure loss analysis.

3.5 Application to Component Pressure Drop.

The following describes the development of a mathematical model for an abrupt component two-phase multiplier, which can be used with simplified models for the flow to predict the two-phase pressure loss across such devices.

Consider the single-phase flow of a liquid through an abrupt component illustrated in Fig. 3. Steady state, turbulent flow conditions are assumed and compressibility effects are ignored.

The static pressure change ΔP_{lo} is the sum of the reversible and irreversible pressure changes, the irreversible pressure change being the frictional dissipation, and the reversible pressure change being theoretical change in kinetic energy. If the upstream and downstream diameter are the same, then the theoretical kinetic energy change is zero, since compressibility effects are ignored. Assuming that the contraction from the upstream cross-section down to the vena contracta occurs with little frictional dissipation and that the pipe wall friction may be ignored, then, the static pressure change is due to the irreversible dissipation during the expansion from the vena contracta volume area, Ac, to the downstream pipe area, A_2. This may be estimated from a momentum balance written for the control volume, abcd, in which the following assumptions are inherent;

1. Uniform velocity at c and 2.
2. Neglect the frictional force on the pipe wall between ac and bd.
3. Pressure Pc acts over efac.

With these assumptions a momentum balance gives:

$$A_2(P_2-Pc) = \dot{m}\,(Uc-U_2)$$

Using continuity relations this can be re-arranged to give:

$$P_2-Pc = \Delta P_{lo} = \frac{G_2{}^2}{\rho l}\,\frac{A_2}{Ac}\left(1 - \frac{Ac}{A_2}\right) \tag{1}$$

which is the expression for the single-phase static pressure change due to an abrupt component.

For a two-phase development consider the separated flow model illustrated in Fig. 4.

If compressibility and mass transfer effects are ignored, the theoretical kinetic energy change is zero and the same assumptions apply.

Writing a simplified momentum balance for the combined flow gives:

$$A_2(P_2-Pc) = \dot{m}_g\,(Ugc-Ug_2) + \dot{m}_l\,(U_l-U_{l2})$$

Using continuity relations this can be re-arranged to give:

$$P_2-Pc = \frac{\dot{m}^2}{A_2}\left(\frac{x^2}{\rho gAc}\left(\frac{1}{\alpha c} - \frac{Ac}{A_2\alpha_2}\right) + \frac{(1-x)^2}{\rho lAc}\left(\frac{1}{(1-\alpha c)} - \frac{Ac}{A_\alpha(1-\alpha_2)}\right)\right) \tag{2}$$

If it is assumed that the void fraction remains unchanged, i.e. $\alpha c = \alpha_2 = \alpha$ then the expression can be reduced to:

$$P_2-Pc = \Delta P_{tp} = \frac{G_2{}^2}{\rho l}\,\frac{A_2}{Ac}\left(1-\frac{Ac}{A_2}\right)\left(\frac{x^2}{\alpha}\,\frac{\rho l}{\rho g} + \frac{(1-x)^2}{(1-\alpha)}\right) \tag{3}$$

The assumption of constant void fraction is only approximately true as discussed by Petrick and Swanson (Ref. 1) and Richardson (Ref. 2).

Dividing equation (3) by equation (1) gives an expression for the two-phase multiplier, i.e.:

$$\phi^2{}_{lo} = \frac{\Delta P_{tp}}{\Delta P_{lo}} = \frac{x^2}{\alpha}\,\frac{\rho l}{\rho g} + \frac{(1-x)^2}{(1-\alpha)} \tag{4}$$

This expression can also be derived for abrupt expansions and contractions and may be used as a general result for abrupt components where the inherent assumptions are applicable.

In practice, the void fraction, α, is unknown and a direct solution for ϕ^2_{lo} is generally unobtainable, however a value may be calculated if simplistic two-phase models are assumed.

3.6 Zero Slip Prediction.

Using continuity relations the void fraction may be expressed in the form:

$$\alpha = \left(1 + \frac{(1-x)}{x}\frac{\rho g}{\rho l} K\right)^{-1}$$

If it is assumed that the two phases flow at the same velocity then there is zero slip between the phases and the slip ratio, k, equals 1.

hence $$\alpha = \left(1 + \frac{(1-x)}{x}\frac{\rho g}{\rho l}\right)^{-1}$$

Substituting for α in (4) gives the zero slip or homogeneous prediction of two-phase multiplier, $\emptyset^2_{loh}$, i.e.

$$\emptyset^2_{loh} = 1 + x\left(\frac{\rho l}{\rho g} - 1\right) \tag{5}$$

3.7 Maximum Slip Prediction.

Using a separated flow assumption enables a maximum slip prediction to be obtained. If it is assumed that the pressure gradient due to friction is the same for the two phases then:

$$K = \frac{Ug}{Ul} = \left(\frac{\rho l}{\rho g}\right)^{\frac{1}{2}}$$

This leads to a value for α of

$$\alpha = \left(1 + \frac{(1-x)}{x}\left(\frac{\rho g}{\rho l}\right)^{\frac{1}{2}}\right)^{-1}$$

Substituting for α in (4) gives a maximum slip prediction of the two-phase multiplier, $\emptyset^2_{los}$, i.e.

$$\emptyset^2_{los} = \left(1 + x\left(\left(\frac{\rho l}{\rho g}\right)^{\frac{1}{2}} - 1\right)\right)^2 \tag{6}$$

3.8 Chisholm's Prediction.

Previous correlation of two-phase component loss data has been attempted by Chisholm (Ref. 3). This method is based on a rearranged form of equation (4).

i.e. $$\emptyset^2_{lo} = 1 + \left(\frac{\rho l}{\rho g} - 1\right)(Bx + (1-B)\,x^2) \tag{7}$$

The coefficient B incorporates slip effects and is chosen to correlate experimental data. A fit to the high pressure steam/water data of Fitzsimmons (Ref. 4) was achieved using a B value of 2.5 for a globe valve and 1.6 for a gate valve.

4. THE EXPERIMENTAL APPARATUS

4.1 General Description.

An experimental rig was constructed at BHRA Fluid Engineering to obtain two-phase pressure loss data for components inserted in approximately 25 mm diameter horizontal and vertical pipe sections, see Fig. 5. The horizontal section was 540 pipe diameters in length (14.5 m) with a flow development length of 124 diameters before the measuring section. The vertical section was 320 pipe diameters long (8.0 m) with a flow development length of 101 diameters. The static pressure was measured at various points using piezoresistive pressure transducers, the output from these being fed via an amplifier box to a computer used for the data acquisition and analysis. Water was supplied to the test section using a centrifugal pump which either filled a header tank or directly pressurised the circuit, the water mass flow into the mxing device being measured using an electromagnetic flow meter. Air supplied via a compressor at 80 psi and regulated to the required pressure before entering the mixing device, the mass flow being obtained using a turbine meter in conjunction with pressure and temperature measurements.

4.2 Two-phase Pressure Loss Measurements.

Two-phase component pressure loss measurements were obtained using high responsive piezoresistive pressure transducers. Eight transducers were used in all, four upstream and four downstream. The outputs from each transducer were sampled at a frequency of 25Hz and stored on computer floppy disk for analysis. The r.m.s. value of each transducer output was calculated and a linear regression fitted separately to the upstream and downstream values, see Fig. 6. These were extrapolated to the plane of the component and the two-phase component pressure drop taken as the difference between the two extrapolated r.m.s. pressure values. Correlation coefficients were also computed as a check on the accuracy of the regression.

5. EXPERIMENTAL RESULTS

Tests were conducted over a wide range of both horizontal and vertical two-phase flow conditions, using a representative variety of commercial 1" valves and several orifice plates manufactured to BS1042. The valves included a gate valve conforming to BS1952, a globe valve conforming to BS2060 and uPVC diaphragm and ball valves conforming to BS5159 and DIN3300. The range of valve openings and symbols used are given at the end of the text. Studies were performed using air/water mixtures at close to atmospheric conditions, covering water superficial velocities from 0 to 2 m/s and air superficial velocities from 0 to 30 m/s.

Water only loss coefficients were measured for each component and compared well with the values presented in Miller (Ref. 5).

Horizontal flow two-phase test conditions were plotted on a Mandhane flow pattern map (Ref. 6), whereas vertical flow two-phase test conditions were plotted on a Hewitt and Roberts flow pattern map (Ref. 7). All flow regimes were covered except horizontal dispersed bubble flow and vertical wispy annular flow. Good agreement was found between the observed flow regimes and those predicted from the maps.

5.1 General Trends.

The pressure loss data are presented in the form of plots of two-phase multiplier, ϕ^2_{lo} or $\Delta P_{tp}/\Delta P_{lo}$, vs. dryness fraction, x, see Figs. 7-15. The plots are all seen to be of a similar form, however, individual tendencies are present. The effects of a change in component area differ, the most marked change being observed in the diaphragm valve results, Fig. 7, whereas the globe valve results of Fig. 11 fall closely on a single curve. There is hence a tendency for the effects to be suppressed with a more tortuous flow path. The largest differences occurred when slug flow conditions were present, the pressure fluctuations probably giving rise to the least accurate measurement, and may also have been the most dependent on the component configuration, compared with results under bubble or annular flow conditions which tended to be far more steady. For some of the gate valve results both the r.m.s. and mean values of the pressure fluctuations were compared, and little difference was seen, except under violent slug conditions where the mean value was less than the r.m.s. It is thought that the effect of the ball valve and globe valve configurations was to homogenise the flow and hence make these components less sensitive to opening and flow regimes, as compared with gate and diaphragm valves, for example, which have a less uniform throttling effect with opening.

5.2 Pipe Orientation Effects.

The pressure loss plots for vertical flow are of the same form as those obtained from the horizontal test results, and also display similar tendencies. Superimposing the horizontal and vertical flow results indicates that although the corresponding curves are similar, the two-phase multipliers tended to be higher for the vertical flow results. These discrepancies were probably due to the fact that the loss coefficients in the vertical pipe were less due to the effective increase in opening, brought about by a reduction in pipe area since the horizontal flow tests used pipe of 26.9 mm I.D., whereas the vertical flow tests used pipe of 24.7 mm I.D. It is seen in the horizontal results and the vertical globe valve results, that the two-phase multipliers tend to increase with opening.

The effects of a change in pipe orientation alone can be observed by comparing the horizontal 15 mm orifice plate results and the vertical 13.78 mm orifice plate results in Fig. 16. It is seen that the results are similar except for some variations under slug flow conditions that may have been produced by the large pressure transients. Hence it may, with caution, be possible to use the horizontal plots when designing for vertical flow.

5.3 Valve Orientation Effects.

The effects of the valve orientation can be studied by comparing the gate valve results of Figs. 8, 9, and 10. It is seen that although the plots are similar some differences exist, for example, the losses tended to be lower for the bubble flow results when the valve was on its side, and the opening was small. This may have been due to the fact that with this configuration both the phases were throttled by similar amounts, which was unlikely to be the case for the other two configurations.

5.4 Effect of Pipe Size.

Although the effects of pipe size were not investigated in the experiments, reasonable agreement has been found between the present results and those obtained at the University of Strathclyde's 127 mm air/water facility, see Ref. 8. Results obtained with a gate valve of openings 0.5625 and 0.250 are compared with the present results in