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WET STEAM FLOWS IN INDUSTRIAL LARGE-DIAMETER PIPES: FLOWRATE, MOISTURE AND PRESSURE DROP MEASUREMENTS

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Abstract—The flowrate, moisture and pressure drop were measured in a water–steam mixture flowing at a pressure of 1 MPa in pipes with 0.3 and 1.2 m dia. The results are presented here, along with an original method for assessing pressure drops in a water–steam mixture flowing in such industrial pipes.

Key Words: steam water flows, industrial piping, pressure drop measurements

INTRODUCTION

Electricité de France has designed and developed (in cooperation with Stein-Industrie) a new type of steam–water separator, the high velocity separator (H.V.S.).

Promising results were obtained during preliminary tests on a wet steam test rig. Therefore it was decided to design two industrial prototypes. These two H.V.S.s were installed at the high-pressure (H.P.) exhaust of a 900 MW turbine; they have been operating since August 1980. Their design, operating conditions and performances have been described by Cerdan & Talleu (1984).

In order to understand precisely their operating conditions, extensive measurements were made upstream of and downstream from these prototypes. Several parameters were measured: steam velocity, moisture and pressure drop.

The experimental results were obtained at a pressure of approx. 1 MPa in pipes 0.3 or 1.2 m dia and for mass fluxes of some 200 $kg/m^2/s$ and steam qualities ranging from 0.7 to 1. This paper will discuss the main measurement results obtained.

Based on the results obtained, an easily applicable method is proposed for designers to assess the pressure drop in water–steam mixtures flowing in large-diameter pipes incorporating straight runs and bent sections.

MEASUREMENTS

The measurements were carried out at the exhaust of the H.P. cylinder of the Unit 2 turbine generator in the Bugey power plant near Lyon. This unit is a 900 MW-PWR unit. The piping sections studied are presented in figures 1 and 2. The cross-sections and the tapping holes are also indicated in these figures.

Steam velocity measurements

Steam velocities are measured with a 5-hole probe. The probe head is fastened to a mobile bidirectional (translation, rotation) boom. The probe was pre-calibrated on a test bench under actual pressure and moisture conditions. Steel flexible cords filled with water connect the probe holes to pressure transducers. These absolute and differential pressure transducers are Rosemount type 1151 gauges.

At each measuring point, the three components of the velocity vector (axial, radial and tangential) are known. The velocity can be measured over the entire piping diameter. The dry steam flowrate in the piping can then be computed by integrating the axial velocity profile.

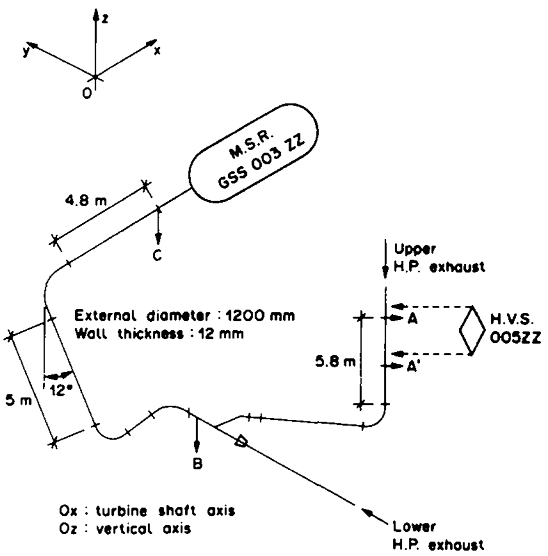

Figure 1. Isometric drawing of the Bugey power plant GSS 003 ZZ M.S.R. supply.

Figure 3 shows an axial velocity profile measured at cross-section C, illustrated in figure 1. The corresponding pressure is 1.02 MPa, steam moisture = 0.3% and outer piping dia = 1.2 m. Another axial velocity profile obtained for cross-section A, represented in figure 2, is shown in figure 4. The corresponding pressure is 0.896 MPa, steam moisture = 8.9% and outer piping dia = 0.324 m.

Steam mass-quality measurement

The water flowrates in the pipes are measured using a chemical tracer method. The method consists of the following procedure (see figure 5). A chemical, which is completely soluble in water and non-volatile under the above-mentioned temperature and pressure conditions, is injected into the pipe at a constant flowrate. (Tracer content in the injected solution, c_i; injection flowrate, W_i.) After travelling over a certain distance,† which will vary according to the pipe geometry, the tracer and liquid phase will be regarded as homogeneously mixed. This means that all liquid (both droplets and wall flow) will be at a concentration c_e.

Samples are taken through a tapping hole drilled into the lower pipe generatrix. The sampling flowrate, W_s, is adjusted‡ so that only a fraction of the water film on the wall is collected through the tapping hole.

The injection concentration, c_i, and the tracer content, c_e, in the samples are measured with a Perkin–Elmer, model 560, spectrophotometer.

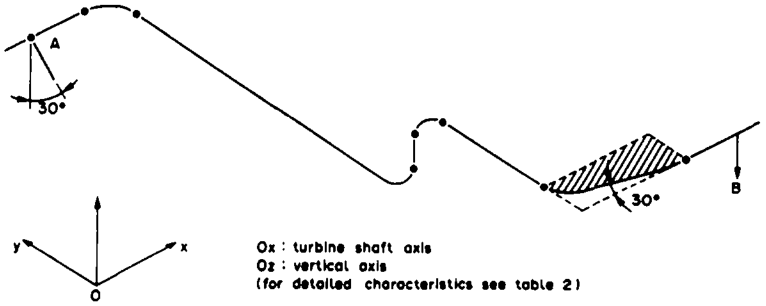

Figure 2. Isometric drawing of the studied piping.

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†In a straight pipe, this distance is about 40 times the diameter under the test conditions. When the pipe incorporates fittings, the equivalent straight run length—in terms of pressure drops—should be considered.

‡The sampling flowrate W_s must not exceed a maximum value $W_{s,L}$. Beyond this value, some steam would be collected and then condensed, thereby diluting the sample (see figure 6).

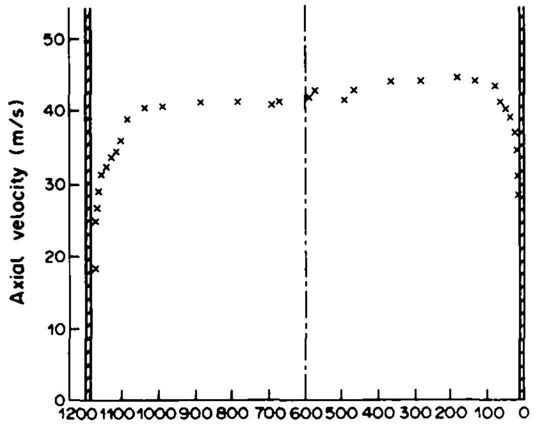

Figure 3. Axial velocity profile, section C in figure 1.

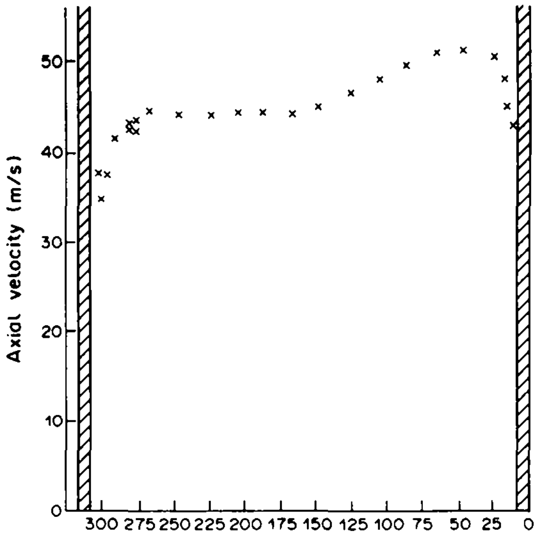

Figure 4. Axial velocity profile, section A in figure 2.

Since the tracer mass remains constant in the piping, we can write

$$W_i \cdot c_i = (W_L + W_i') \cdot c_e.$$

If W_i' denotes the extra liquid flowrate in the pipe due to the tracer injection. Hence,

$$W_L = W_i \frac{c_i}{c_e} - W_i'.$$

Generally,

$$W_i' \ll W_i \cdot \frac{c_i}{c_e}.$$

Hence,

$$W_L = W_i \cdot \frac{c_i}{c_e}.$$

Therefore, if W_G denotes the steam mass flowrate, the steam mass quality x is given by

$$x = \frac{W_G}{W_G + W_L}.$$

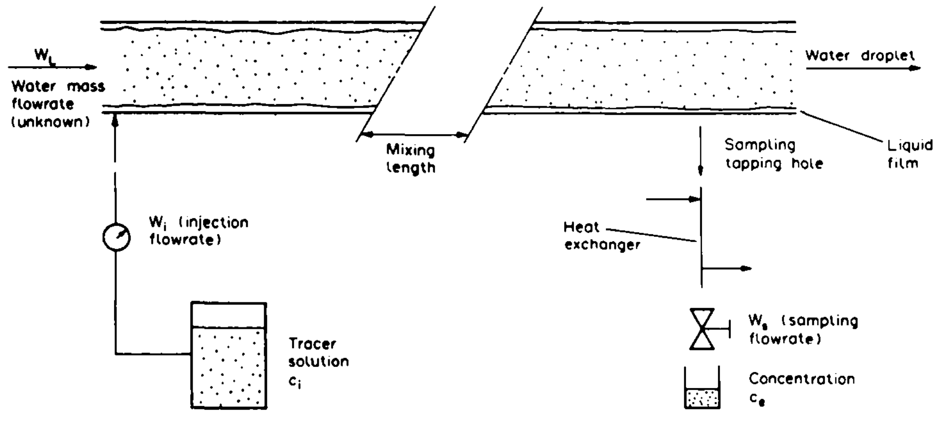

Figure 5. Total water flowrate measurement.

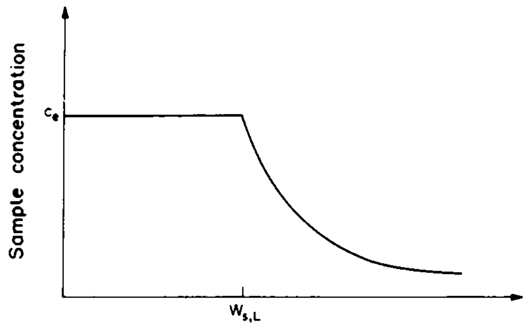

Figure 6. Maximum sampling flowrate.

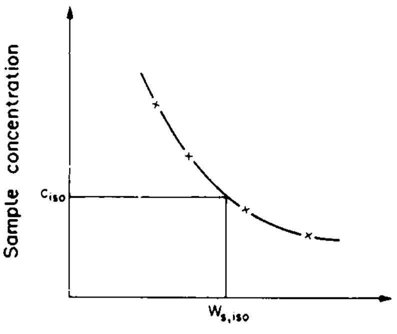

Figure 7. Moisture measurement.

The steam moisture h is given by

$$h = 1 - x.$$

The chemicals used are lithium (in the hydroxide form LiOH) and caesium (in the carbonate form Cs_2CO_3).

Local steam moisture measurement

As for the steam quality, a tracer method is used for measuring the local steam moisture. A sampler with its streamlined head mounted on a mobile bidirectional boom (translation, rotation) is inserted into the pipe at a point where the velocity vector has been previously measured. The sampling mass flowrate, $W_{s,iso}$, must be adjusted so as to be isokinetic to the steam flowrate. Samples are taken after the collected water–steam mixture has passed through a cooler–condenser.

To have a homogeneous mixture, the chemical tracer is injected sufficiently upstream of the measurement cross-section (see the previous subsection). The tracer content c_F of the water film on the wall, as well as the tracer content c_{iso} of the samples collected by the sampler, are measured. By adjusting the various sampling flowrates,† a curve of the type shown in figure 7 can be plotted.

The local steam humidity h_L at the studied point can be computed iteratively as follows. A first estimate of the local moisture is given by

$$h_L = \frac{c_{iso}}{c_F}.$$

Assuming the liquid phase inertial forces are well above those of the gaseous phase, so that the sampled water flowrate is independent of the sampling rate adjustment, the sampling flowrate $W'_{s,iso}$ that should have been adjusted to actually meet the steam isokinetic charateristic is given by

$$W'_{s,iso} = \frac{W_{s,iso}}{1 - h_L}.$$

From the curve in figure 7, it appears that the sample content of interest is c'_{iso}. The local moisture is then assessed again:

$$h'_L = \frac{c'_{iso}}{c_F}.$$

This process is repeated until agreement is obtained. Some local moisture profiles thus obtained are presented in figures 8 and 9.

Figure 8 shows a profile measured at cross-section C (in figure 1) at a pressure of 1.02 MPa for a total steam moisture of 0.3% (outer piping dia = 1.2 m). Figure 9 shows another profile measured at cross-section A (in figure 2) at a pressure of 0.896 MPa for a total steam moisture of 17.3% (outer piping dia = 0.324 m).

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†Higher and lower than $W_{s,iso}$.

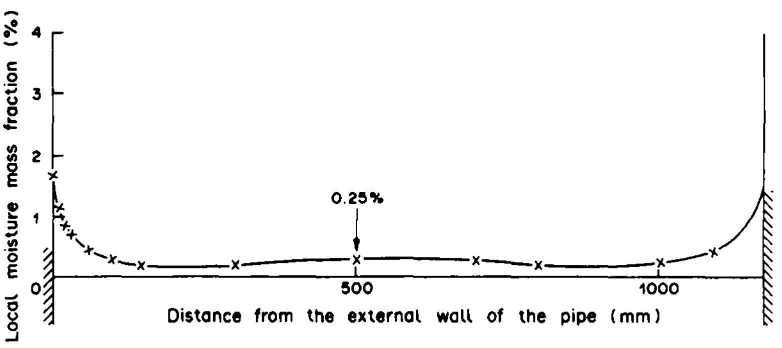

Figure 8. Moisture profile, section C in figure 1.

Pressure drop measurements

Pressure differences are measured using Rosemount sensors of the 1151 type. The tapping holes are located on the lower generatrix of the horizontal pipes. Pipe sections B–C of figure 1 and A–B of figure 2 in particular are studied.

The raw measurement results are corrected to account for gravity, $\bar{\rho} \cdot g \cdot \Delta H$, where $\bar{\rho}$ denotes the mixture density, g is the acceleration due to gravity and ΔH is the difference in height between the two measurement cross-sections. It was ensured that the steam-quality variations produced by heat losses and pressure drops were quite negligible in the cases studied here.

The mixture density $\bar{\rho}$ is described by the relationship

$$\bar{\rho} = \varepsilon \cdot \rho_G + (1 - \varepsilon) \cdot \rho_L,$$

where the void fraction ε is given by Bryce's (1977) correlation.

The frictional pressure drops in both pipe sections are then calculated for different steam moisture levels. As a matter of fact, because the H.V.S.s are prototypes, a number of valves have been incorporated in the associated pipes in order to be able to adjust the steam mass quality in particular.

The pressure drop measurement results can be presented as the curve shown in figure 10, which illustrates the variation of a coefficient ϕ_G^2 with steam moisture. The non-dimensional coefficient ϕ_G^2 is a two-phase frictional multiplier defined by analogy with an expression given by Collier (1972). It gives the frictional pressure drop for the two-phase flow (Δp_F) in terms of the single-phase frictional pressure drop (for gas flow only in the pipe, $\Delta p_{F.G}$):

$$\phi_G^2 = \frac{\Delta p_F}{\Delta p_{F.G}}.$$

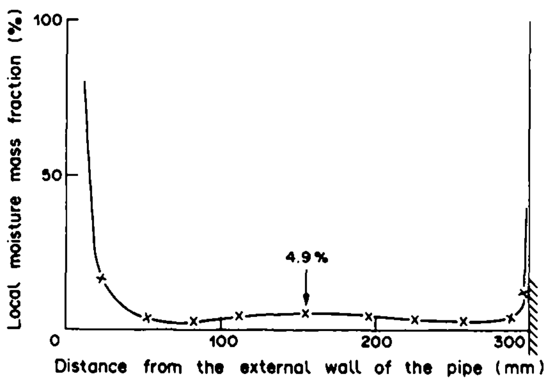

Figure 9. Moisture profile, section A in figure 2.

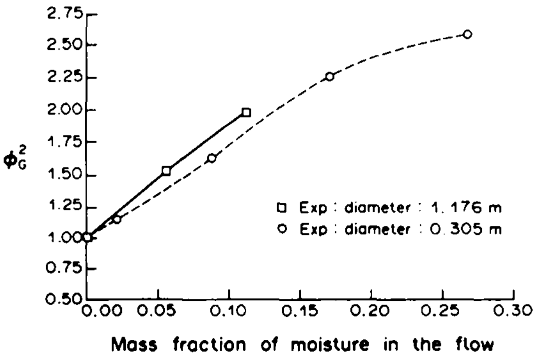

Figure 10. Two-phase friction multiplier ϕ_G^2: experimental results.

Table 1. Experimental values of the coefficient ϕ_G^2

(a) B–C pipe section in figure 1. Test conditions: $p = 1.0$ MPa, $250 < \dot{m} < 280$ kg/m²/s

Moisture	Two-phase friction multiplier, ϕ_G^2	Accuracy (%)
0.003	1.00	—
0.057	1.53	15
0.114	1.98	16

(b) A–B pipe section in figure 2. Test conditions: $p \simeq 0.9$ MPa, $235 < \dot{m} < 250$ kg/m²/s

Moisture	Two-phase friction multiplier, ϕ_G^2	Accuracy (%)
0.0	1.00	—
0.0215	1.15	3
0.089	1.63	8
0.173	2.26	9.5
0.270	2.68	8

$\Delta p_{F.G}$ can be computed using the following relationship:

$$\Delta p_{F.G} = \frac{1}{2\rho_G}\left(\lambda\frac{L}{D} + \sum_{k=1}^{n} \xi_k\right)\dot{m}^2 x^2.$$

The non-dimensional pressure drop coefficients λ and ξ_k are functions of the relative roughness of the pipe and are given, for instance, by Idel'cik (1962).

Table 1 lists the values of ϕ_G^2 for the two pipes studied.

PROPOSED METHOD FOR COMPUTING FRICTIONAL PRESSURE DROPS IN A WATER–STEAM FLOW INSIDE LARGE-DIAMETER PIPES INCLUDING STRAIGHT AND BENT RUNS

Very few publications in the vast literature devoted to two-phase flows deal with the assessment of pressure drops in a water–steam flow inside pipes of large diameter. Moreover, the proposed correlations usually concern the air–water system, pipes with very small diameters and void fractions well below 1.†

Thus, to compute the frictional pressure drop in a water–steam flow inside an industrial circular pipe with a diameter D, incorporating straight runs with a total length L and n bent sections, the following procedure is proposed:

—Assuming the total mass flow $\dot{m}$, the steam mass quality x (or moisture h, $h = 1 - x$), the dimensionless pressure drop coefficients λ and ξ_k and the gas density ρ_G are known, the frictional pressure drops in the sole steam flow throughout the whole pipe section, $\Delta p_{F.G}$, on the one hand, and throughout the bent sections only $\Delta p_{F.B.G}$, on the other, are determined:

$$\Delta p_{F.B.G} = \frac{1}{2\rho_G}\sum_{k=1}^{n} \xi_k \dot{m}^2 x^2$$

and

$$\Delta p_{F.G} = \frac{1}{2\rho_G}\left(\lambda\frac{L}{D} + \sum_{k=1}^{n} \xi_k\right)\dot{m}^2 x^2.$$

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†In the measurements presented here, the void fraction ranges from 0.97 (moisture around 0.30) to 1.0 (no moisture).

—The ratio β is computed:

$$\beta = \frac{\Delta p_{F,B,G}}{\Delta p_{F,G}}.$$

—The correction coefficient ϕ_G^2 is determined from the curve in figure 11, or by using the relationship

$$\phi_G^2 = 1 + 10 \cdot h - 23 \cdot h^2 + 23.46 \cdot h^3 + 16.67 \cdot h^4.$$

—If the ratio β differs from 0.6, the value ϕ_G^2 is adjusted using figure 12.

—The pressure drop in the two-phase flow inside the pipe section studied, Δp_F, is computed using the relationship

$$\Delta p_F = \Delta p_{F,G} \cdot \phi_G^2.$$

Comparison of the present method with published methods; justification and validity range of the present method

The piping in figure 2, whose detailed characteristics are given in table 2, has been particularly studied. As a matter of fact, pressure drops in the constituent units can be considered to add up in these pipe runs.

Calculations have been performed using published methods; their results can be compared with the test results.

First method: homogeneous flow model. The frictional pressure drop for the two-phase flow, Δp_F, is expressed in terms of the single-phase pressure drop for the total flow considered as liquid, $\Delta p_{F,L,0}$. The following relationship is used, as in Collier (1972):

$$\Delta p_F = \Delta p_{F,L,0} \cdot \left[1 + \frac{x(\rho_L - \rho_G)}{\rho_G}\right] \cdot \left[1 + \frac{x(\mu_L - \mu_G)}{\mu_G}\right]^{-1/4}.$$

Second method: homogeneous model for the straight runs; model of Chisholm & Sutherland (1969) for the bent runs. The frictional pressure drop for the two-phase flow, Δp_F, is the sum of two terms $\Delta p_{F,S}$ and $\Delta p_{F,B}$. $\Delta p_{F,S}$ is the frictional pressure drop for the two-phase flow through the straight runs; $\Delta p_{F,S}$ is calculated using the homogeneous flow model as above. $\Delta p_{F,B}$ is the frictional pressure drop for the two-phase flow through the bends; $\Delta p_{F,B}$ is calculated using the model of Chisholm & Sutherland (1969) as suggested by Collier (1972).

Third method: homogeneous model for the straight runs; model of Chisholm (1980) for the bent runs. As in the second method, Δp_F is decomposed in $\Delta p_{F,S}$ and $\Delta p_{F,B}$. $\Delta p_{F,S}$ is calculated as above; $\Delta p_{F,B}$ is calculated using the model given by Chisholm (1980).

Figure 13 shows the comparison of these three calculations with the test results.

The homogeneous model results are well below the test results. The results obtained with the third method are not very different from the test results, especially if the steam moisture is above

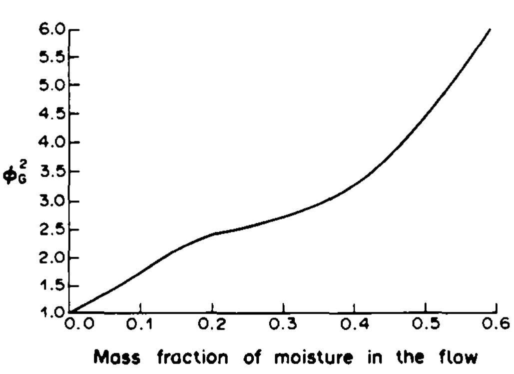

Figure 11. Proposed two-phase multipler ϕ_G^2.

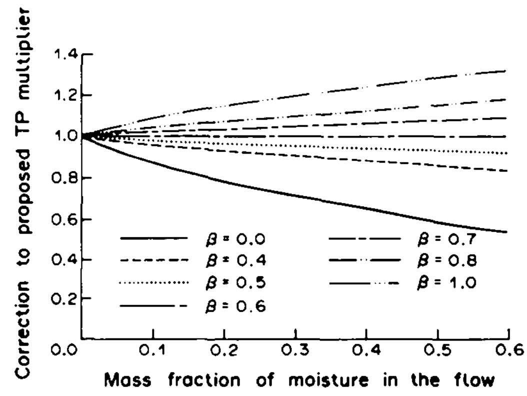

Figure 12. Correction due to the β ratio.

Table 2. Characteristics of pipe section A–B in figure 2

General characteristics	
Material: stainless steel (Z2CN.18.10)	
Piping i.d.	0.305 m
Piping thickness	0.0095 m
Piping relative roughness	$1.2 \cdot 10^{-4}$
Piping slope	0.2°
Description of the circuit between A and B	
Straight run	1.0 m long
90° Elbow = radius of curvature	1.50 *D*
Straight run	6.066 m long
90° Elbow = radius of curvature	1.50 *D*
Upward vertical straight run	0.485 m long
60° Elbow = radius of curvature	1.73 *D*
Straight run	4.929 m long
60° Elbow = radius of curvature	1.73 *D*
Straight run	2.674 m long
30° Elbow = radius of curvatuve	1.55 *D*
Straight run	0.15 m long

0.25 (the relative variation of ϕ_G^2 is 8% when the steam moisture is 0.27). Therefore it was decided to perform a parametric study using this third method.

Apparently, the values of the coefficient ϕ_G^2 depend very little on pressure p,† on superficial steam velocity J_G‡ or on the relative pipe roughness $\bar{\delta}$.§

Finally, the coefficient ϕ_G^2 is only sensitive (see figure 12) to the relative contribution of bent pipe sections to the total pressure drop (the ratio β defined above).

The curve in figure 11 whose first part (moisture <30%) corresponds to test results and whose second part is plotted on the basis of the third method is reliable for assessing the coefficient ϕ_G^2 before a possible correction to take the ratio β into account.

The proposed method is valid for:

—pressures ranging from 0.3 to 6 MPa;
—steam qualities ranging from 0.4 to 1;
—adiabatic, fully-turbulent, steady-state flows;
—industrial large-diameter systems incorporating straight runs and bent sections.

The accuracy of the predictions obtained with this method is 15% for a steam quality >0.7 or 25% for steam qualities ranging from 0.4 to 0.7.

Worked example

The results obtained with the method described in the previous paragraph can be compared with the test results obtained by Manzano-Ruiz *et al.* (1987).

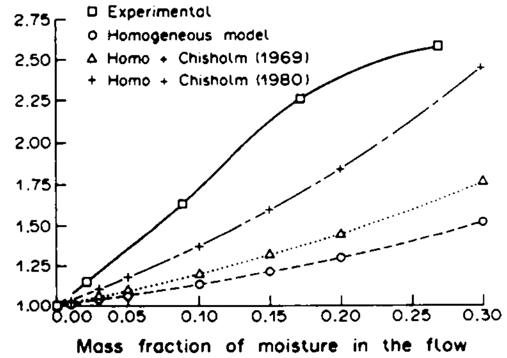

Figure 13. Comparison of test results with calculations.

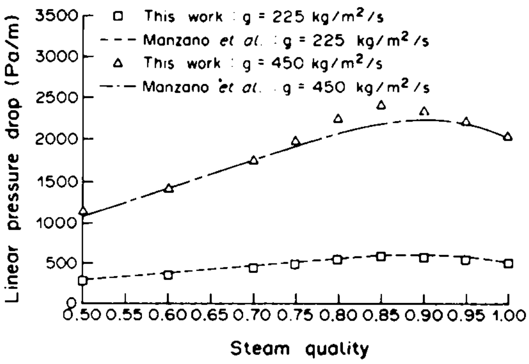

Figure 14. Worked example.

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†From 0.3 to 6 MPa, the maximum relative variation of ϕ_G^2 is <6%.
‡From 25 to 65 m/s, the maximum relative variation of ϕ_G^2 is <2%.
§From $0.5 \cdot 10^{-4}$ to $9 \cdot 10^{-4}$, the maximum relative variation of ϕ_G^2 is <8%.

The pipe section studied is horizontal, straight and 0.097 m dia. The pressure is 2.0 MPa, the mass flux is 225 or 450 kg/m^2/s. Computed and recorded results can be compared in figure 14: the agreement between the results is quite satisfactory (>90% agreement). However, it should be underlined that the maxima in the curve thus computed do not exactly correspond: the steam quality corresponding to the maximum pressure drop is somewhat lower in this study than in Manzano-Ruiz *et al.*'s (1987) test results.

CONCLUSION

The measurement results given here have provided a very precise means of assessing the behaviour in industrial plants of a new model of water–steam separators, i.e. high velocity separators.

We have also proposed a straightforward method for determining friction pressure drops in a water–steam flow inside large-diameter industrial pipes consisting of straight runs and bent sections.

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