Pergamon

Int. J. Multiphase Flow Vol. 20, No. 3, pp. 659–666, 1994

0301-9322(93)E0016-X

0301-9322/94 $7.00 + 0.00

THE PROPULSION OF AN ISOLATED SLUG THROUGH A PIPE AND THE FORCES PRODUCED AS IT IMPACTS UPON AN ORIFICE PLATE

I. Owen and I. B. Hussein

Department of Mechanical Engineering, University of Liverpool, P.O. Box 147, Liverpool L69 3BX, England

(Received 15 December 1990; in revised form 1 November 1993)

Abstract—This paper is concerned with the propulsion of an isolated slug of liquid along a pipeline by an expanding gas. The particular case in mind is when a body of liquid accumulates behind a closed valve in a piping system carrying a wet or condensing gas, and that valve is suddenly opened to expose the slug of liquid to a high-pressure gas. The slug will be driven along the pipeline by an expansion wave which weakens as the slug accelerates, whilst its motion will be resisted by friction and by the compression wave ahead of it. This situation has been analysed and compared with slug velocities measured in a 50 mm pipe. Also presented in this paper are measurements of the considerable forces generated by the slug as it impinges onto an orifice plate installed in the pipe.

Key Words: pipeline transients, fluid transients, slugs, steam hammer, water-hammer, two-phase flow

1. INTRODUCTION

Fluid transients in pipelines can generate significant pressure surges and the resulting forces are capable of causing considerable damage. Water-hammer is probably the best known and most widely reported phenomenon in this respect (e.g. Sharp 1981). Rapid valve closure in gas systems also leads to pressure waves being generated, as reported by Coccio (1966) and Katze & Ernst (1983).

In two-phase gas–liquid flow, the nature of the flow is such that pressure surges are unavoidable when it encounters obstructions such as valves. A particularly violent two-phase flow is slug flow, and this has been investigated in detail by, amongst others, Dukler and various co-authors (e.g. Dukler & Hubbard, 1975; Taitel & Dukler 1976; Dukler *et al.*, 1985). The force generated by a transient slug as it emerged from the end of a pipeline to impact upon a target was measured by Sakaguchi *et al.* (1987).

Much of the work published on two-phase flow is related to the nuclear power and oil recovery industries. In these studies the flow has been continuous and, over a reasonable time-scale, steady. The work reported in this paper is different from other studies related to fluid transients, in that it concerns a single slug of liquid being propelled along a pipeline by a high-pressure gas. The scenario is one where water has collected behind a closed value in a steam main, and the valve is suddenly opened so that the water is forced along the pipe like a bullet in a gun. In practice, steam valves are opened slowly to avoid this problem. Nevertheless, damage to steam valves and flowmeters is quite common, which indicates that the scenario is not improbable. It is also of value to consider this worst case to evaluate the dynamics of the slug.

Also of interest in this paper is the force generated when the slug impacts upon an obstruction. The obstruction encountered in practice could be a sharp bend, a partially opened gate valve, a globe valve, a non-return valve, a flowmeter etc. Practical experience has shown that such impacts can cause considerable damage. In the present study an orifice plate was used to demonstrate the level of forces that can be generated. Again it is of value to consider the worst case of rapid valve opening to evaluate the damage that can potentially occur.

2. THEORETICAL ANALYSIS OF SLUG DYNAMICS

Consider an idealized slug in a pipe, as shown in figure 1. Behind the slug is a high-pressure expanding gas which is forcing the slug along the pipe, thereby compressing the low-pressure gas in front. Also resisting the motion of the slug will be the friction between it and the pipe wall. It is appreciated that as the slug moves through the pipe, it will shed liquid behind it. However, the area of the slug upon which the gas is acting will still be that of the pipe cross-section, and the total mass of liquid being accelerated is that of the whole slug. For the sake of the analysis it is assumed that the slug remains intact, although the practical aspects of this will be discussed later in the paper in the light of the experimental data.

Returning to figure 1, the pressure, P_e, of the expansion wave upstream of the slug is given by Shapiro (1954) as

$$\frac{P_e}{P_{e0}} = \left(1 - \frac{\gamma - 1}{2}\frac{U_e}{a_e}\right)^{\frac{2\gamma}{\gamma-1}}, \quad [1]$$

where P_{e0} refers to the stagnation pressure upstream of the slug, γ is the ratio of specific heats, U_e is the speed of the pressure wave and a_e is the initial speed of sound upstream of the slug.

Downstream of the slug the pressure, P_c, of the compression wave is given by

$$\frac{P_c}{P_{c0}} = \left(1 + \frac{\gamma - 1}{2}\frac{U_c}{a_c}\right)^{\frac{2\gamma}{\gamma-1}}, \quad [2]$$

where P_{c0} refers to the downstream stagnation pressure, U_c is the speed of the compression wave and a_c is the initial downstream speed of sound.

The wall friction force, F_w, is given in terms of the wall shear stress, τ, as

$$F_w = \tau l \pi D, \quad [3]$$

where l is the length of the slug and D is the pipe diameter. The wall shear stress is, in turn, related to the slug velocity, U_s, by

$$\tau = \tfrac{1}{2}\rho U_s^2 C_f, \quad [4]$$

where ρ is the liquid density and C_f is the skin friction coefficient calculated using the Blasius equation:

$$C_f = 0.079\mathrm{Re}^{-1/4}, \quad [5]$$

where the Reynolds number, Re, is based on the slug velocity and the pipe diameter.

The expansion wave, the slug and the compression wave will all be moving with the same velocity, say U, and therefore the equation of motion for the slug becomes

$$\rho l \frac{dU}{dt} = \left[P_{e0}\left(1 - \frac{\gamma - 1}{2}\frac{U}{a_e}\right)^{\frac{2\gamma}{\gamma-1}} - P_{c0}\left(1 + \frac{\gamma - 1}{2}\frac{U}{a_c}\right)^{\frac{2\gamma}{\gamma-1}} - \frac{2\rho U^2 C_f l}{D}\right]. \quad [6]$$

Equation [6] can be solved numerically to find the distance the slug has travelled as a function of time.

Consider now the impact of the slug with an orifice plate, as shown in figure 2. If the slug was assumed to behave as a solid and to remain intact during the impact, the impact pressure would

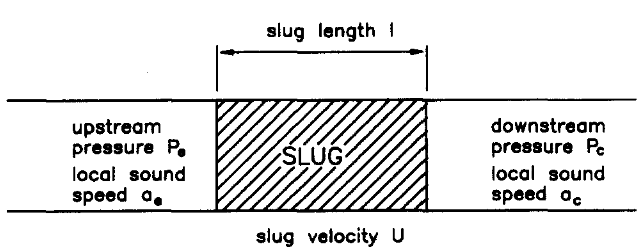

Figure 1. Isolated slug in a pipe.

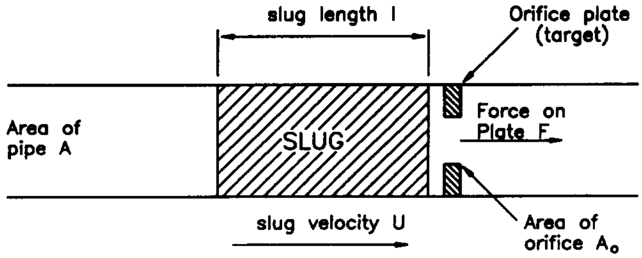

Figure 2. Slug impacting on an orifice plate.

be the same as that calculated for water-hammer. If, on the other hand, it is assumed that the flow passes easily through the orifice, as it would through an orifice plate flowmeter, then the pressure drop across the orifice might be expected to follow the typical orifice-plate flowmeter equation. In practice, the situation is somewhere between these two. The slug does not remain as a solid on impact, but extrudes through the orifice so that not all the inlet momentum flux is given up. Neither does the slug pass easily through the orifice—it slows down significantly on impact, so that the actual velocity with which it approaches the orifice after the initial impulse and the velocity with which it then flows through the orifice are indeterminate.

In this case, therefore, it is not possible to produce an analytical expression for the impact pressure or force and if a practical expression is desired, then an empirical approach using dimensional analysis must be used. Collecting the variables shown in figure 2, the following relationship suggests itself:

$$\frac{F}{\rho U^2 A} = \text{function}\left[\left(\frac{A_0}{A}\right),\left(\frac{l}{D}\right)\right]. \qquad [7]$$

This expression will be considered later in the paper in the light of the experimental data.

3. EXPERIMENTAL INVESTIGATION

The situation being considered is one where a steam valve has been opened rapidly. Upstream of the valve is a pressure source of infinite volume; downstream is an isolated slug of water being propelled into a empty pipe. How this was modelled experimentally is shown in figure 3.

Air, with pressures up to 11 bar (absolute) and of ambient temperature, was used as the driving gas. The volume of the air reservoir was sufficiently large that the drop in pressure as the air was expelled was $<3\%$. For the majority of the tests the air and the water slug were expelled through a 50 mm diameter steel pipe. To enable the rapid expulsion of the air, a quarter-turn butterfly valve was located between the reservoir and the pipe. The water slug was held between the closed butterfly valve and a thin polythene sheet sandwiched between flanges downstream of the valve. The water was poured into the reservoir so formed through a filling port. By using two different lengths of pipe in which to hold the slug (0.99 and 2.16 m), it was possible to obtain three different lengths of slug (0.99, 2.16 and 3.15 m) whilst maintaining the overall length of the rig constant at 13.0 m. Since the end of the pipe was open to the atmosphere, the initial air temperature and pressure in the pipe were atmospheric; also the initial temperature of the air in the reservoir was close to ambient.

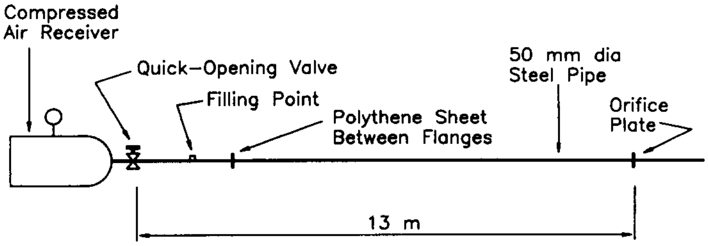

Figure 3. Slug impact rig.

To propel the slug down the pipe, the valve was opened quickly by hand so that the sudden rise in pressure behind the water forced it through the polythene sheet. To confirm the consistency of the procedure, a number of valve openings were carried out and no discernible difference in the slug velocity was detected. To measure the slug velocity at the end of the pipe, two conductivity probes 0.215 m apart were used, as shown in figure 4. The leading edge of the slug was detected as it passed the probes and this was recorded on a transient recorder to give the time interval between the slug passing the two positions.

To measure the impact at the orifice plate, a piezo-electric pressure transducer was located close to its upstream face, figure 4. The output from the charge amplifier was also connected to the transient recorder.

For each slug length, different air reservoir pressures were used to propel the slug down the pipeline. The velocity measurements were made with the orifice plate in position, and with it removed so the water discharged freely from the end of the pipe. It was observed that for the tests with the 2.16 and 3.15 m slugs, the trace recorded from the conductivity probes showed a step-change as the front of the slug passed by. For the 0.99 m slug, however, the trace recorded from the conductivity probe was extremely erratic, indicating that the slug had broken up. That this had happened was confirmed by the trace from the pressure transducer, which showed a much lower impact pressure.

These observations led to a supplementary experiment being carried out where a 1.24 m slug was propelled along a 28 mm bore pipe of much longer length relative to the 50 mm pipe. With the initial pipe length of 12.5 m (length-to-diameter ratio of 446), the slug was propelled along the pipe at four different air pressures and the velocity probes were used to indicate whether the slug had disintegrated. The tests were repeated another six times with the pipe length being shortened on each occasion until, at a length of 3.55 m, the slug was seen to remain intact.

4. RESULTS AND DISCUSSION

4.1. Slug velocities

Figure 5 shows the calculated velocity of the 2.16 and 3.15 m slugs as they travel along the pipe. The initial pressure upstream of the slug corresponds to that of the air receiver whilst the initial downstream pressure was taken to be atmospheric. After the initial rapid acceleration, the rate of increase of velocity reduces as the pressure of the expansion wave reduces and that of the compression wave increases. It can be seen how the heavier slug accelerates more slowly, as do both slugs for lower driving pressures. How the calculated velocities compare with the experimental data is shown in figure 6. For each of the slug lengths in figure 6, there is also a different length between the starting position of the leading edge of the slug and the position where the velocity is measured; this is because the overall length of the pipe was constant. It can be seen from figure 6 that there is no distinguishable difference between the slug velocities measured with and without the orifice

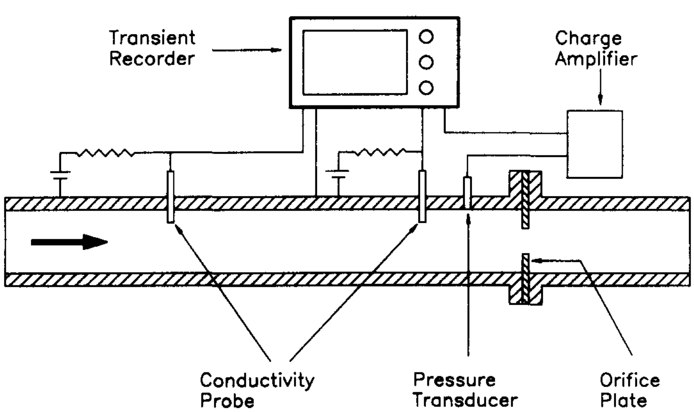

Figure 4. Velocity and pressure measurement.

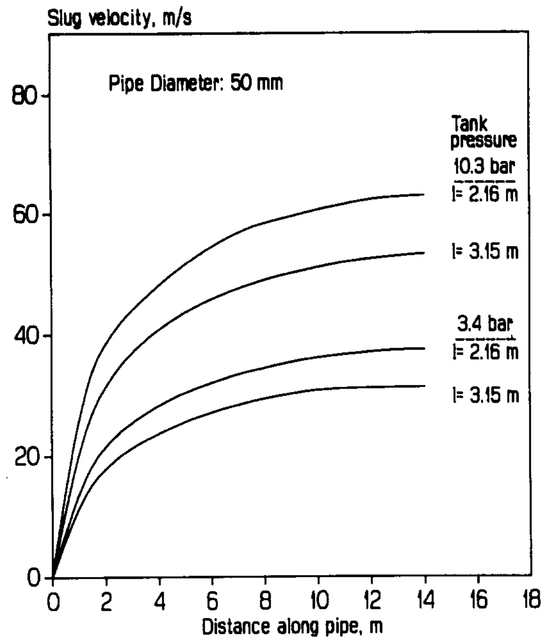

Figure 5. Slug velocity as it travels along the pipe.

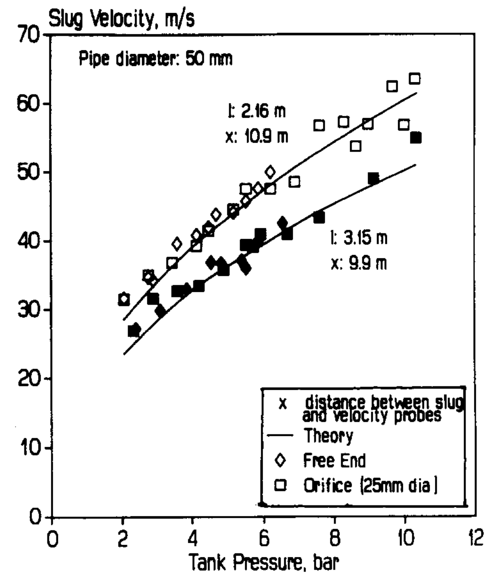

Figure 6. Comparison between calculated velocities and experimental data.

plate in place. This suggests that the air being compressed between the orifice plate and the slug has little effect on the velocity of the slug as it approaches the plate.

Results are presented for only two of the three slug lengths because, as stated earlier, it was observed that the shorter slug disintegrated before it reached the end of the pipe. To investigate this further, a supplementary experiment was carried out using a smaller diameter (28 mm) pipe, the results of which are shown in figure 7. In figure 7 the approximate distance that the slug travelled before disintegrating is shown for different air pressures. How these distances relate to the overall velocity–time histories calculated for the slug is shown in figure 8. The curves show that the slug approaches a terminal velocity and that the slug will, in fact, break up before it reaches

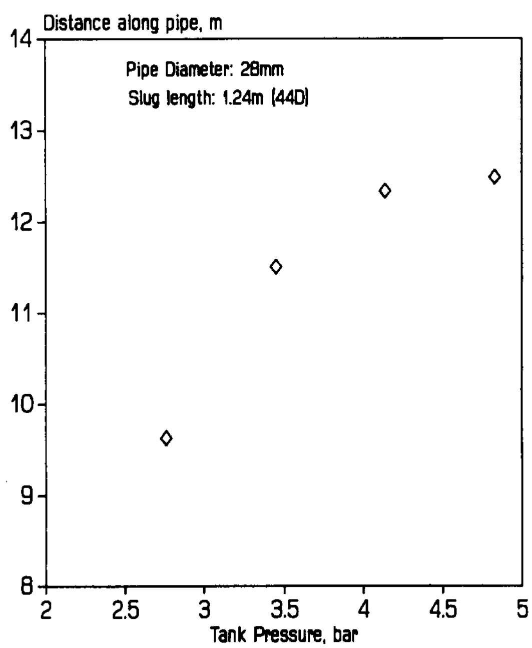

Figure 7. Approximate distance at which the slug breaks up.

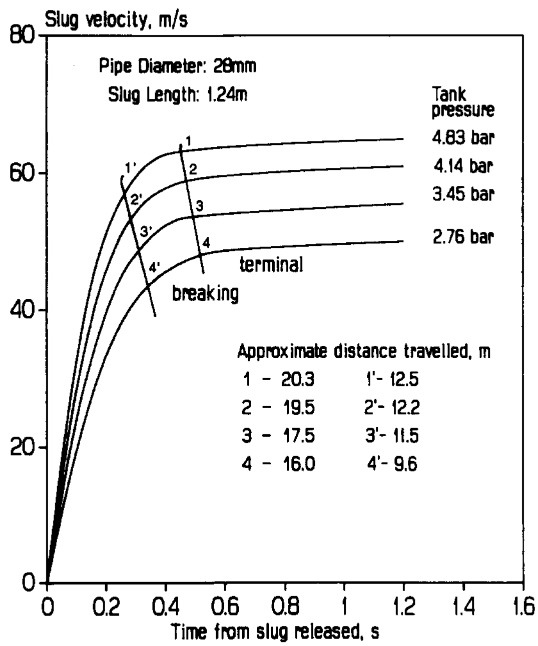

Figure 8. Slug velocity in the 28 mm diameter pipe.

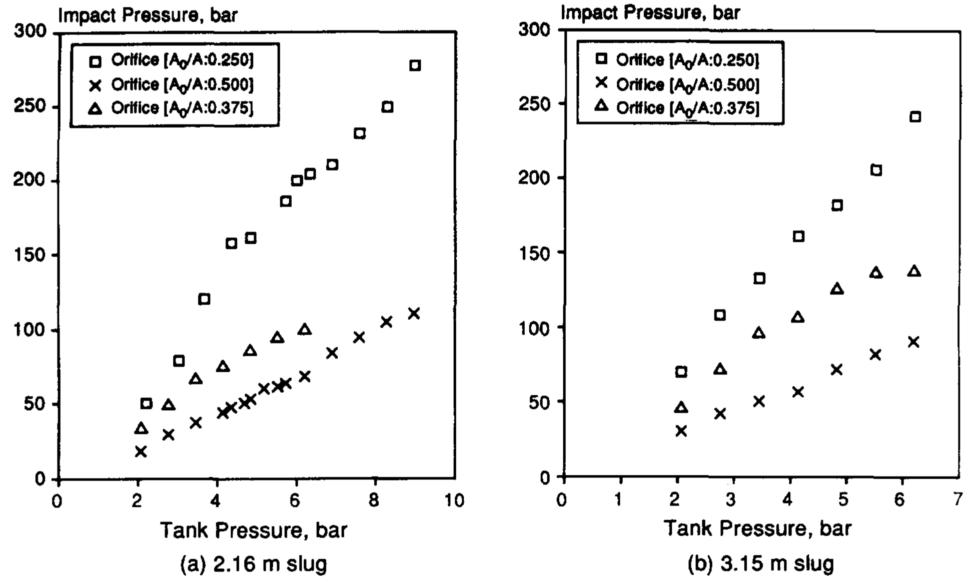

Figure 9. Maximum impact pressure on the orifices.

this velocity. This data is for one slug length and the observation just made cannot be applied universally. However, it is useful to have insight into the practical aspects of the slug dynamics, since one might worry needlessly about the impact damage from a slug which has already disintegrated in the pipeline.

The main difference between the slug being considered in the present work and more normal two-phase slug flow is that in the latter the slug is "riding" over a layer of liquid; picking up liquid at the front and shedding it behind. In the present case the slug is only shedding from the rear, and even then the driving gas is likely to sweep the shed water along. Taitel (1987)† proposed that the rate of shedding from the rear of the slug, X, is approximated by:

$$X = 0.2A\rho UR, \qquad [8]$$

where A is the pipe cross-section, ρ is the liquid density, U is the slug velocity and R is the liquid volume fraction in the slug. In the present case the slug will initially be all water and R will be unity. However, Barnea & Brauner (1985) suggest that R can quickly fall towards 0.5 in a moving slug. The original mass of a slug of length l is ρAl and, therefore, for an average slug velocity of $\bar{U}$, the time taken for the slug to disintegrate is $l/0.1\bar{U}$. If an average velocity of 30 m/s for the slug between its release and its breakup is taken from figure 8 for the slug length of 1.24 m, then according to this relation the time for the slug to disintegrate is about 0.4 s; which is comparable with the time shown in figure 8. Regardless of the simplistic nature of this calculation, it nevertheless demonstrates that a single slug being propelled along an empty pipeline has only a limited time before it disintegrates.

4.2. *Impact of slugs on the orifice plate*

Figure 9 shows the impact pressures generated by the two different slug lengths as they impacted upon orifice plates of different area ratios and with different velocities. Comparing figure 9(a) and (b) it can be seen that the shorter slug has higher impact pressures for a given air pressure; this is because, as shown earlier in figure 6, the shorter slug has the higher velocity, because of its lower inertia, reduced frictional surface area and because it has travelled a little further. The pressures generated are considerable; figure 10 shows how the impact pressure consistently amplifies the original tank pressure for a given orifice diameter and slug length. Orifice plate flowmeters are required to have sharp edges for them to comply to the relevant standard (e.g. ISO 5167-1: 1991). The orifice plate used most frequently in the present tests, and for setting up and developing

†If this reference is consulted, a subsequent Letter to the Editor from Bendiksen *et al.* (1988) should also be consulted.

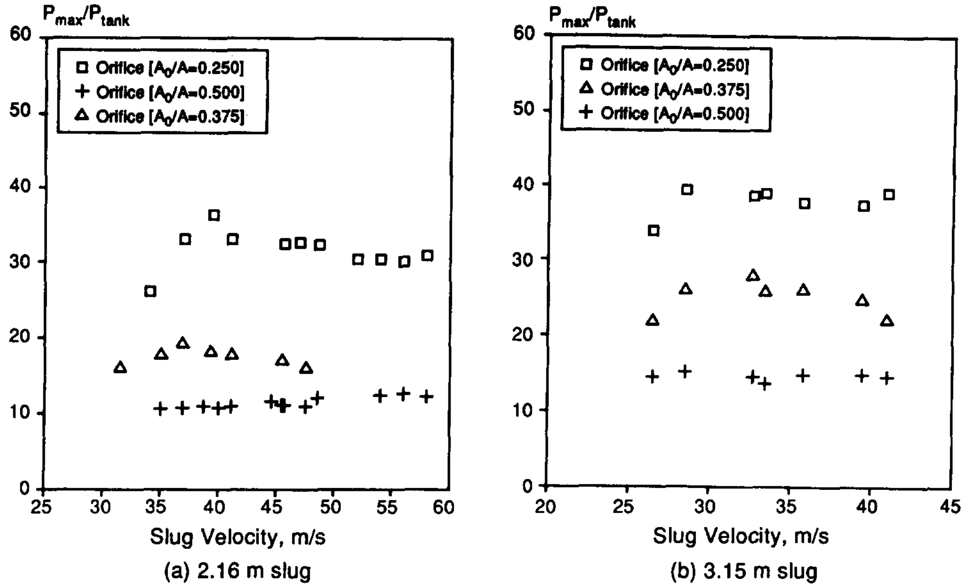

Figure 10. Maximum impact pressure of a slug on the orifice as a ratio of the initial tank pressure.

procedures, was made from mild steel, had a 25 mm orifice and was 3 mm thick. After about 20 impacts, the plate had deformed substantially and, therefore, it was frequently replaced thereafter. The potential for damage to orifice-plate flowmeters is therefore significant, as is the sudden pressure rise for the differential pressure measurement device.

Recalling [7], the data has been collected together in figure 11 according to the relation:

$$F = \text{const} \times \rho A U^2 \left(\frac{A_0}{A}\right)^2 \cdot \left(\frac{l}{D}\right). \qquad [9]$$

Because only two slug lengths and just the one pipe diameter have been used in the impact tests, [9] cannot be applied universally. However, it can be seen from figure 11 that the present data correlates well with this expression and further experiments would confirm this or yield the correct relationship. This, however, was beyond the scope and resources of the present work.

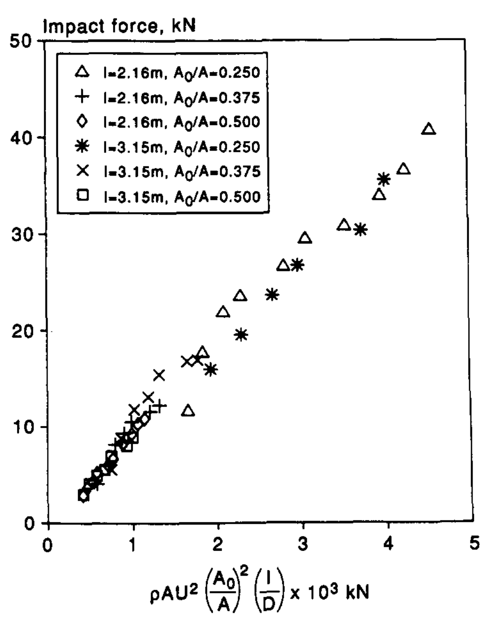

Figure 11. Correlated impact force.

5. CONCLUSIONS

This work has been concerned with the acceleration of an isolated liquid slug along an empty pipeline by an expanding high-pressure gas. The velocity of the slug has been successfully predicted and compared with the experimental data. It has been observed, however, that because of liquid being shed from the rear of the slug, it will disintegrate after a certain time and its damage potential will diminish accordingly.

The pressure generated by the slug impacting on an orifice plate has been measured and has been shown to be sufficiently large to have significant consequences for the maintenance of steam distribution systems or for any piping systems containing condensing or wet gas. Whilst this is well-known from practice, the present study has quantified the problem and, for the conditions investigated, the impact forces generated have been correlated.

Acknowledgements—The authors wish to acknowledge the financial support of the Science and Engineering Research Council and Spirax Sarco Ltd. The contribution through helpful discussions held with Dr J. W. Cleaver is also gratefully acknowledged.

REFERENCES

Barnea, D. & Brauner, N. 1985 Holdup of the liquid slug in two-phase intermittent flow. *Int. J. Multiphase Flow* **11**, 43–49.

Bendiksen, K., Hoyer, N. & Malnes D. 1988 Effect of gas expansion on slug length. *Int. J. Multiphase Flow* **14**, 653–654.

Coccio, C. L. 1967 Steam hammer in turbine piping systems. ASME paper 66-WA/FE-32.

Dukler, A. E. & Hubbard, M. G. 1975 A model for gas-liquid slug flow in horizontal and near horizontal tubes. *Ind. Engng Chem. Fundam.* **14**, 337–347.

Dukler, A. E., Maron, D. M. & Brauner, N. 1985 A physical model for predicting the minimum stable slug length. *Chem. Engng Sci.* **80**, 1379–1385.

ISO 1991 ISO 5167-1 1991 Measurement of fluid flow by means of pressure differential devices. International Standards Organization.

Katze, D. & Ernst, G. (1983) Utilizing a real gas formulation for the analytic solution of steam hammer. *Advances in Fluid Structure Interaction Dynamics*, Vol. 75, pp. 43–55. ASME publication PVP, New York.

Sakaguchi, T., Ozawa, M., Hamaguchi, H., Nishiwaki, F. & Fujii, E. 1987 Analysis of the impact force by a transient slug flowing out of a horizontal pipe. *Nucl. Engng. Des.* **99**, 63–71.

Shapiro, A. H. 1954 *The Dynamics and Thermodynamics of Compressible Flow*, Vol. 2. Ronald Press, New York.

Sharp, B. B. 1981 *Water Hammer: Problems and Solutions*. Arnold, London.

Taitel, Y. 1987 Effect of gas expansion on slug length in long pipelines. *Int. J. Multiphase Flow* **13**, 629–637.

Taitel, Y. & Dukler, A. E. 1976 A model for predicting flow regime transition in horizontal and near horizontal gas–lqiuid flow. *AIChE Jl* **22**, 47–55.