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THE REDUCTION OF PRESSURE WAVESPEEDS BY INTERNAL RECTANGULAR TUBES

A.S. Tijsseling
Dept. of Civil Engineering
University of Dundee
Dundee DD1 4HN
United Kingdom
a.s.tijsseling@dundee.ac.uk

A.C.H. Kruisbrink
Industrial Techn. Division
WL| Delft Hydraulics
P.O.Box 177, 2600 MH Delft
The Netherlands
arno.kruisbrink@wldelft.nl

A. Pereira da Silva
Dept. of Civil Engineering
Techn. University of Lisbon
1096 Lisbon
Portugal
coba@mail.telepac.pt

ABSTRACT

The main results of several series of waterhammer measurements in a steel pipeline fitted with an internal air-filled tube of rectangular cross-section are presented. The goal of the research, a significant and predictable reduction of the pressure wavespeed, has been achieved. The reduction is accompanied with much damping when PVC internal tubes are used at low line pressures, whilst at higher line pressures the damping rate is small. The practical use of internal tubes for wavespeed reduction is limited by their strength. At the end of the test series some of the PVC tubes were totally collapsed. Aluminium internal tubes are stronger and give wavespeed reduction without significant additional damping.

1 INTRODUCTION

1.1 Background

It is well known to hydraulic engineers that waterhammer can be controlled by reducing the pressure wavespeeds in pipe systems (Chaudhry 1987, p. 310). Wylie and Streeter (1993, p. 247) surveyed the methods available to achieve wavespeed reduction. One of these methods is to lay a small flexible hose length-wise inside the pipe under consideration. The hose lowers the effective bulk modulus of the system, which results in a lower wavespeed, while pressurising the trapped air allows some degree of controlling the wavespeed.

In the present study wavespeed reduction was *not* required to diminish the effects of waterhammer, but to obtain a "longer" laboratory apparatus. Larger wave-reflection-times ($2L/c$-values) were needed in the experimental validation of scale-laws describing the dynamic behaviour of damped check valves (Kruisbrink and Thorley 1994; Kruisbrink 1996). As it was not practical to use larger pipe lengths (L), the wavespeed (c) had to be reduced in a predictable and controllable manner, preferably without the introduction of extra damping in the system. Flexible hoses inside the pipes gave unsatisfactory results and therefore they were replaced by rectangular tubes made of either PVC or aluminium. The results of several series of surge tests with internal rectangular tubes (Tijsseling 1998; Pereira da Silva and Evin 1998) are summarised in this paper.

A short review of literature is given in Subsection 1.2 and the objectives of the research are stated in Subsection 1.3. Section 2 describes the laboratory apparatus and the tests performed. The equation used to predict pressure wavespeeds is formulated in Section 3, where furthermore the influence of termination impedance on system frequencies is investigated. Typical test results are shown in Section 4 and discussed in Section 5. Section 6 gives the conclusions of the study.

1.2 Previous work

Flexible hoses. Although most hydraulic engineers know about reducing waterhammer by means of a flexible hose, there is not much literature on the subject.

Réméniéras (1952) was the first to experiment with a rubber hose to reduce waterhammer in pipelines and his company (Electricité de France) even obtained a French patent on this novel technique. He derived a theoretical expression for the reduced pressure wavespeed.

Kottmann and Mink (1987) performed and interpreted tests with a pressurised garden hose laid within a steel tube. They

observed wavespeeds much higher than predicted.

Kruisbrink (1996, pp. 211-214) used rubber and PVC hoses and also found wavespeeds higher than anticipated. The hoses appeared to be sensitive to collapsing. The collapse was accompanied with strong dampening effects, which were difficult to predict. Moreover, the coiling up of the pre-pressurised hoses formed a serious limitation in a convenient use of the device. As such, it was not suited for a proper validation of the scale laws, as planned in Kruisbrink's check-valve project.

Coaxial pipes. Instead of a flexible hose, an air-filled plastic pipe can be placed inside a liquid-filled steel pipe in order to lower the pressure wavespeed. For other reasons, Sagomonyan (1973), in the frequency domain, and Bürmann (1975), in the time domain, analysed the interesting case of coaxial pipes both filled with liquid. They derived general expressions for the *two* coupled liquid wavespeeds that occur in such systems.

Rectangular pipes. Jenkner (1971) showed by calculation and by measurement that the pressure wavespeed in rectangular pipes is much smaller than in circular pipes of the same hydraulic radius. See Fig. 1. Thorley and Guymer (1976) extended Jenkner's theoretical work on thin-walled pipes to thick-walled pipes by including the effects of shear deformation.

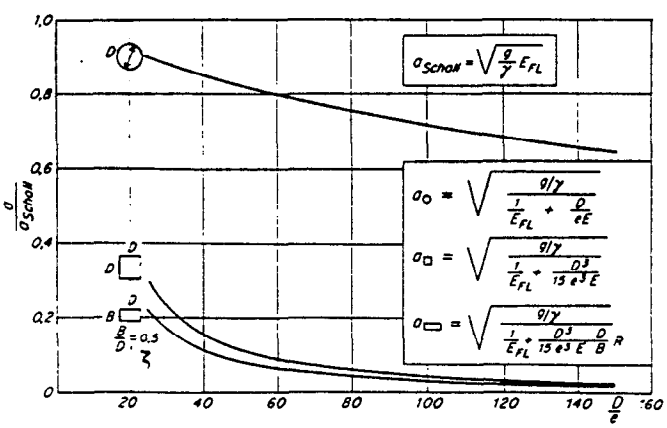


Figure 1. Ratio of pressure wavespeed in steel pipes to speed of sound in unconfined water as a function of ratio of pipe diameter to wall-thickness (Jenkner 1971, p. 102).

1.3 Objectives

The main objective of the present work is to combine the ideas introduced in Subsection 1.2, by placing an air-filled rectangular tube inside a water-filled circular steel pipe, in order to: (1) reduce the natural system frequencies [$c/(4L)$ or $c/(2L)$ values] in a predictable and controlled manner, (2) reduce the system vibrations in a cheap and efficient way.

2 EXPERIMENT

2.1 Test rig

The test rig at WL| Delft Hydraulics consisted of a 24 m long, 0.5 m diameter, pipeline connecting a large high-pressure tank and a smaller air-vessel (containing about 3 m^3 of air); see Fig. 2. Sections of rectangular tube made out of either PVC or aluminium were placed length-wise inside the steel pipeline. The 3 m long sections were sealed at the ends, filled with air at atmospheric pressure and rigidly connected to the top of the pipe (Photograph 1). The pipeline contained one bend, which was also fitted with an internal tube. The flowmeter (0.88 m length), a dummy section (0.5 m length) replacing the check valve, an expansion joint (1.89 m length) and a short section of large-diameter pipe (1.4 m length, 0.8 m diameter), connecting the pipeline to the high-pressure tank, were not fitted with an internal tube. Table 1 gives the geometrical and material properties of the rig.

The pressure waves in the pipeline were captured by four dynamic (piezo-electric) pressure transducers along the pipeline. Pressure transducers were also located at the high-pressure tank and at the air-vessel. A static pressure transducer to record the line pressure, an electro-magnetic flowmeter and a thermometer were available. Kruisbrink (1996, Chapter 13) described the test set up in detail; see Fig. 2.

The manufacturer was not able to provide data on the strength of the rectangular profiles under external loads. Therefore simple tests with dead weight loads have been carried out. A 200 mm long PVC section loaded at its outer breadth of 40 mm collapsed under a weight of 500 kg, which corresponds to a pressure of 6 bar. A 200 mm long aluminium section could sustain a weight of 2000 kg at its outer breadth of 80 mm, which corresponds to a pressure of about 12 bar. These tests gave an indication of the strength of the profiles. However, the profiles will deform differently when loaded by a surrounding fluid.

2.2 Test runs

Experiments have been performed to find the first natural frequency (quantitatively) and damping rate (qualitatively) of the hydraulic system. The stagnant liquid column in the pipeline was subjected to a step load by means of a sudden

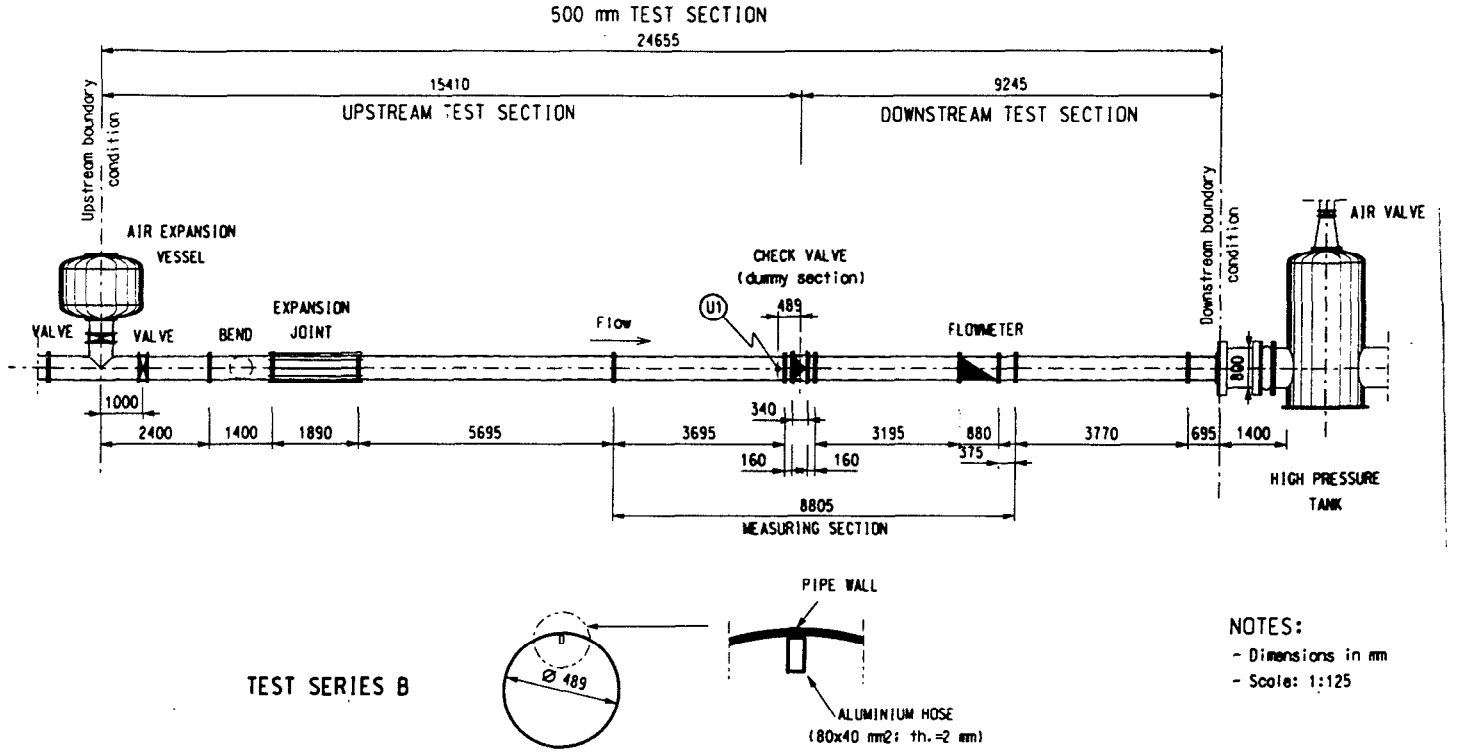
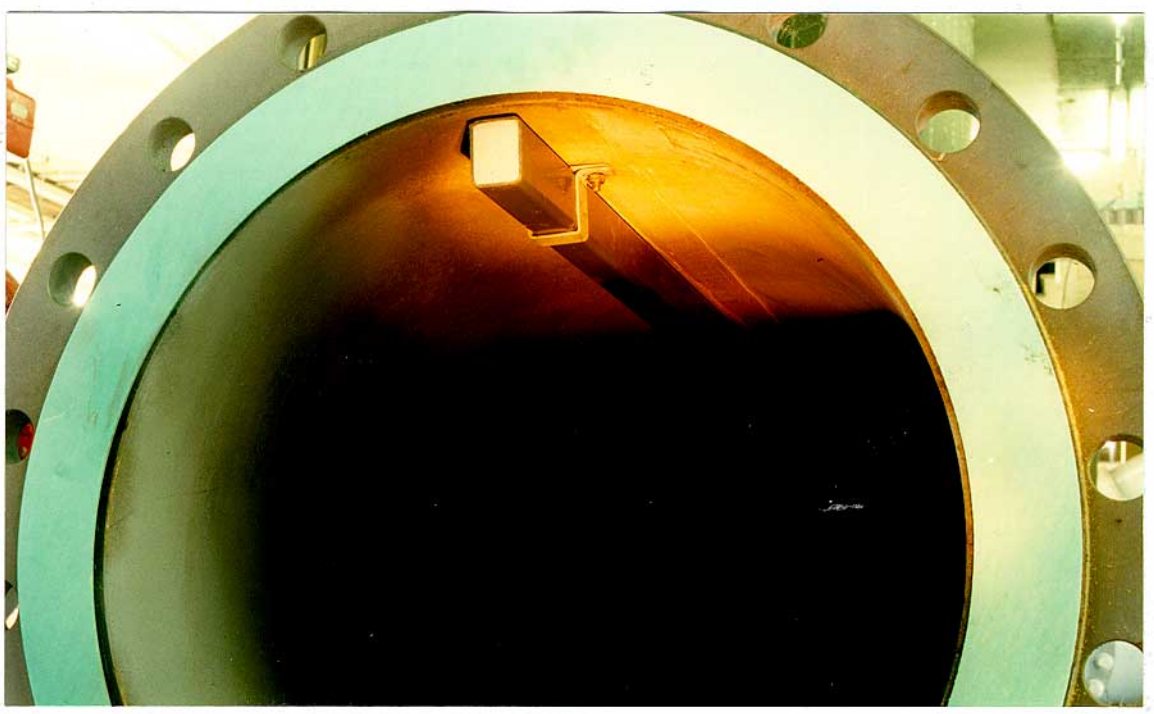


Figure 2. WL| Delft Hydraulics test rig; a dummy section replaces the check-valve in the present study.



Photograph 1. Test pipe with fitted internal rectangular tube.

Water			
bulk modulus	K = 2.19 GPa	mass density	ρ_f = 998.2 kg/m^3
External circular steel pipe (16 bar pressure class)			
length in series A inner diameter wall thickness inner area	L_A = 23.7 m D = 489 mm e = 9.5 mm A = 0.188 m^2	length in series B Young's modulus mass density Poisson's ratio	L_B = 24.7 m E = 210 GPa ρ_t = 8000 kg/m^3 ν = 0.3
Internal rectangular PVC tube		Internal rectangular aluminium tube	
length in series A inner breadth inner height wall thickness Young's modulus mass density Poisson's ratio outer area	L_{PVC} = 20.4 m B = 36 mm H = 26 mm e = 2 mm E = 2.943 GPa ρ_t = 1050 kg/m^3 ν = 0.4 A = 0.0012 m^2	length in series B inner breadth inner height wall thickness Young's modulus mass density Poisson's ratio outer area	L_{Al} = 19.4 m B = 76 mm H = 36 mm e = 2 mm E = 70 GPa ρ_t = 2700 kg/m^3 ν = 0.3 A = 0.0032 m^2

Table 1. Geometrical and material properties of the test rig.

(de)pressurisation of the high-pressure tank. Within a test series the magnitude of the step load (ΔP) and the initial pressure (P_0) of the liquid were varied. The magnitude of the excitation (ΔP) was taken as the difference of the final pressure (P_{final}) and the initial pressure (P_0), which were the static line pressures recorded shortly after and shortly before each test run, respectively.

Test *series A* were with the PVC internal tube and a closed valve (1 m ahead of the disconnected air-vessel) as system termination and test *series B* were with the aluminium tube and the air-vessel as system termination. See Fig. 2.

3 THEORY

3.1 Wavespeed

The pressure wavespeed (c) in the liquid inside pipe 1 (circular, inner diameter D_1^i, internal area A_1^i) and outside the pipes 2 (circular, outer diameter D_2^o, external area A_2^o) and 3 (rectangular, mean breadth B_3^m, mean height H_3^m, $B_3^m > H_3^m$, external area A_3^o), see Fig. 3, assuming constant (atmospheric) pressure inside the internal tubes (2) and (3) and outside pipe (1), is estimated from:

$$c = \left\{ \frac{1}{c_0^2} + \frac{A_1^i}{A_{flow}} \frac{1}{c_1^2} + \frac{A_2^o}{A_{flow}} \frac{1}{c_2^2} + \frac{A_3^o}{A_{flow}} \frac{1}{c_3^2} \right\}^{-\frac{1}{2}} \quad (1)$$

with

$$c_0^2 = \frac{K}{\rho_f} \quad (1a)$$

$$c_1^2 = \frac{E_1 e_1}{\rho_f D_1^i} \quad (1b)$$

$$c_2^2 = \frac{E_2 e_2}{\rho_f D_2^o} \quad (1c)$$

$$c_3^2 = \frac{A_3^o E_3 e_3^3}{\rho_f \phi(B_3^m, H_3^m)} \quad (1d)$$

where Thorley and Guymer's (1976) function ϕ is defined as

$$\phi(a,b) = \frac{(a^3+b^3)}{2(a+b)} \left[\frac{a^3}{6} + \frac{a^2 b}{2} - \frac{b^3}{3} \right] +$$

$$- \frac{a^5}{20} - \frac{a^2 b^3}{4} + \frac{b^5}{5} + \frac{(1+\nu_3)\, e_3^2}{2}(a^3 + b^3) + \frac{a b e_3^2}{2}(a+b) \tag{1e}$$

and the flow area (A_{flow}) is derived from the pipe areas through

$$A_{flow} = A_1^i - A_2^o - A_3^o \tag{1f}$$

Substitution of the data from Table 1 into Eq. 1 (with $A_2^o = 0$ m^2) gives theoretical wavespeeds of 750 m/s and 624 m/s for the PVC and aluminium internal tubes, respectively. Without internal tube the theoretical wavespeed would be 1195 m/s (Eq. 1 with $A_2^o = A_3^o = 0$ m^2).

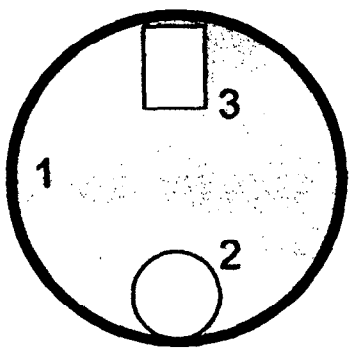


Figure 3. Circular pipe (1) with internal circular (2) and rectangular (3) tubes.

3.2 Termination impedance

In test *series A* the air-vessel in Fig. 2 was disconnected from the pipeline by a closed valve. In test *series B* the valve was open and the air-vessel the system's termination, thereby changing the pipeline from an *open-closed* ($4L/c$) to an *open-open* ($2L/c$) system. It was questioned whether the air-vessel's impedance had an influence on the system's first natural frequency.

The impedance, Z, which is the pressure/velocity ratio P/V, at the air-vessel was estimated by (Wylie and Streeter 1993, p. 323)

$$Z = \frac{n P_a A_{flow}}{\underline{V}_a} \frac{\mathrm{i}}{\omega} = C_{imp} \frac{\mathrm{i}}{\omega} \tag{2}$$

where $\underline{V}_a$ (= 3 m^3) and P_a (= 3 bar) are the vessel's mean air-volume and mean absolute pressure, respectively, and n = 1.4 is the polytropic exponent. The circular frequency ω and the imaginary unit i are basic components in harmonic analyses. The transfer matrix approach of Tijsseling *et al.* (1997) has been used to produce Table 2. From this table it can be seen that the air-vessel's small impedance constant, C_{imp}, of 0.26 bar/m can be neglected.

Termination impedance C_{imp} (bar/m)	Natural frequency f (Hz)
open = 0	$c/(2L)$ = 10.70
1	10.70
10	10.60
100	9.70
1000	6.05
10000	5.40
100000	5.35
closed = ∞	$c/(4L)$ = 5.35

Table 2. Natural frequency as function of termination impedance (example with c = 529 m/s and L = 24.7 m).

4 TEST RESULTS

4.1 Internal rectangular PVC tube and closed-end termination (*series A*)

The results of three typical test runs are shown in the Figs. 4-6. Each figure gives the pressure variation in the high-pressure tank (broken line) and the dynamic pressure at position U1 in the pipeline (solid line). See Fig. 2.

The results in Fig. 4 were obtained with an initial pressure (P_0) of 5.2 barg. A pressure drop $\Delta P = -0.85$ bar initiated the waterhammer event which reveals itself as a harmonic oscillation (solid line) around the tank pressure (broken line). There is not much damping in the system.

With the closed valve at the non-excited end, the elastic liquid column is a quarter-length system of which the first natural frequency (f) is

$$f = \frac{c}{4L_A} \tag{3a}$$

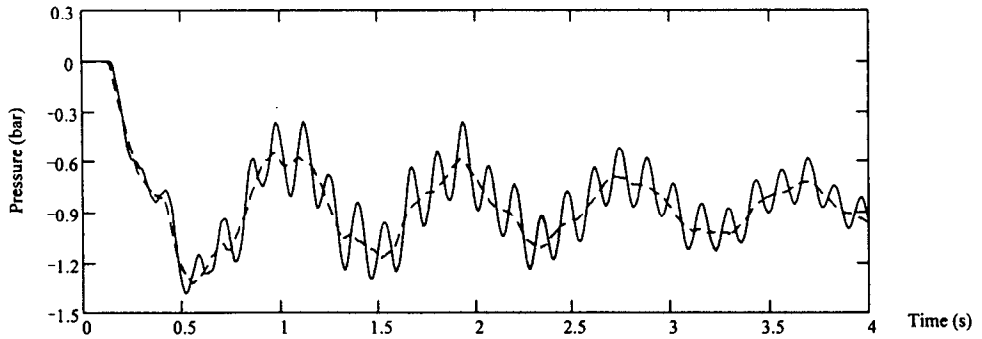


Figure 4. Measured dynamic pressure histories in pipe (solid line) and in tank (broken line) with initial pressure P_0 = 5.2 barg and pressure drop ΔP = −0.85 bar; *series A*.

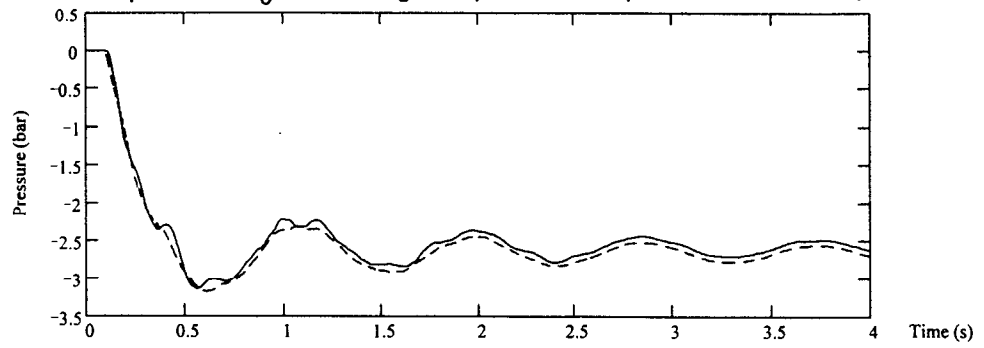


Figure 5. Measured dynamic pressure histories in pipe (solid line) and in tank (broken line) with initial pressure P_0 = 4.2 barg and pressure drop ΔP = −2.5 bar; *series A*.

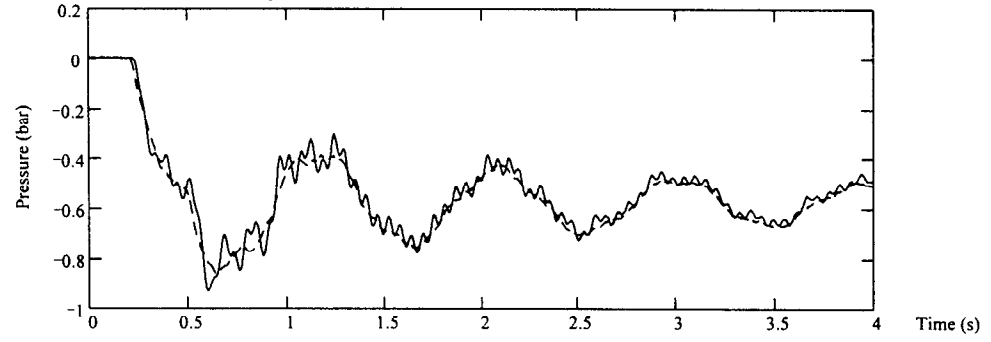


Figure 6. Measured dynamic pressure histories in pipe (solid line) and in tank (broken line) with initial pressure P_0 = 2.7 barg and pressure drop ΔP = −0.55 bar; *series A*.

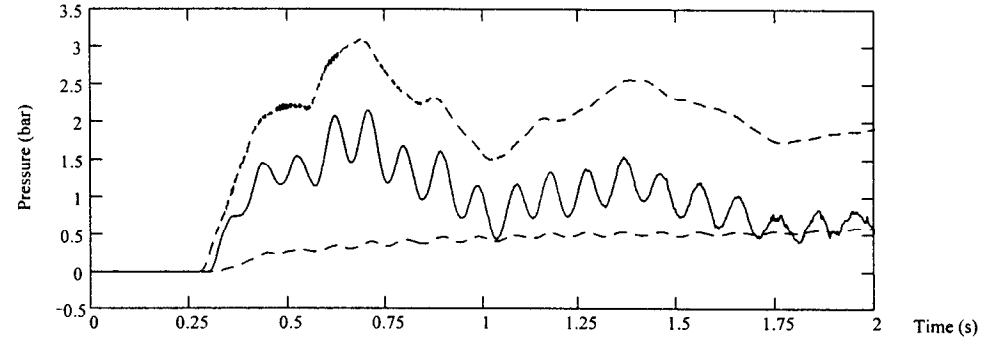


Figure 7. Measured dynamic pressure histories in pipe (solid line), in tank (upper broken line) and at air-vessel (lower broken line); *series B*.

The pressure wavespeed (c) was assessed from measured frequencies through Eq. 3a. Seven measurements with different P_0 and ΔP revealed a first natural system frequency corresponding to a pressure wavespeed of 699 m/s with standard deviation 6 m/s.

The results in Fig. 5 were obtained with a lower initial pressure, P_0 = 4.2 barg, and a larger pressure drop, ΔP = −2.5 bar. There is so much damping in the system now that the waterhammer effect is small and the pressure in the pipe follows quasi-statically the pressure variation in the tank.

The results in Fig. 6, obtained with an even lower initial pressure, P_0 = 2.7 barg, and a pressure drop, ΔP = −0.55 bar, show an unexpected and interesting phenomenon: a higher mode of system vibration has been excited, possibly caused by vibrations of the PVC tubes. Four measurements with different P_0 and ΔP revealed such a higher mode and, using Eq. 3a, its frequency corresponded to wavespeeds of 1683 m/s with standard deviation 2.5 m/s.

There was no evidence that the pressure wavespeed depended on the initial pressure P_0 and on the magnitude of the step load ΔP in the test range considered (1 barg < P_0 < 6 barg; −2 bar < ΔP < 3 bar).

After the test series the pipeline was opened and it was discovered that the PVC tube in the bend (which had been heated before bending) and at a few other places had collapsed or were cracked. It is most likely that the damage occurred during one test run, with initial pressure P_0 = 4.5 barg and pressure *rise* ΔP = 1.8 bar, when a big bang was heard and anomalous pressures were recorded. The results presented in Figs. 4, 5 and 6 were obtained before the "fatal" test run.

4.2 Internal rectangular aluminium tube and open-end termination (*series B*)

Figure 7 shows a typical result obtained with the aluminium tube and with the air-vessel at the non-excited end. The elastic liquid column is then a half-length system of which the first natural frequency (f) is

$$f = \frac{c}{2L_B} \tag{3b}$$

Three measurements with different P_0 and ΔP revealed a frequency of oscillation corresponding to a wavespeed (c) of 524 m/s with standard deviation 9 m/s. The aluminium tube did not introduce much additional damping.

After several tests with relatively large ΔP magnitudes, the aluminium profiles were found slightly deformed. After this observation smaller ΔP magnitudes were used and slightly higher wavespeeds (555 m/s) were then observed.

5 DISCUSSION

A significant wavespeed reduction without additional damping has been obtained with the aluminium internal tube. The same holds for the PVC tube if ΔP is positive and P_0 is larger than about 4 barg.

The *theoretical* pressure wavespeed (Eq. 1) in the steel pipe (Table 1) without internal tube is 1195 m/s. With the internal PVC tube this theoretical speed reduces to 750 m/s and with the aluminium one to 624 m/s.

The *measured* wavespeed in the pipeline without internal tubes was 1067 m/s. With the PVC tube the measured speed reduced to 699 m/s (simply derived from the measured frequency, f, using Eq. 3a with L_A = 23.7 m) and with the aluminium one to 524 m/s (simply derived from the measured frequency, f, using Eq. 3b with L_B = 24.7 m).

Using the measured value of 1067 m/s for the theoretical wavespeed in the external pipe without internal tube (by taking K = 1.575 GPa in Eq. 1a), the theoretical speeds with PVC and aluminium tubes reduce to 715 m/s and 602 m/s, respectively.

Sources of error in the *theoretical* calculation of the wavespeed (c) are: (1) the values of the material properties provided by the manufacturer were not accurate, (2) the visco-elastic behaviour of PVC has not been taken into account, (3) the temperature and strain-rate dependence of system parameters has not been taken into account, (4) the tube deformation mode was different from that assumed by Thorley and Guymer (1976), and (5) the effects of free gas are ignored if K is taken equal to the value given in Table 1.

Sources of error in the *experimental* determination of the wavespeed (c) are: (1) the short section of large diameter pipe (1.4 m length, 0.8 m diameter), connecting the pipeline to the high-pressure tank, has not been taken into account in L/c (inclusion would lead to increased measured wavespeeds: c_{PVC} ≈ 740 m/s; simply derived from the measured frequency, f, using Eq. 3a with L_A = 25.1 m; c_{Al} ≈ 554 m/s; simply derived from the measured frequency, f, using Eq. 3b with L_B = 26.1 m), (2) the fact that there are no internal tubes in the expansion joint (1.89 m length), the flowmeter (0.88 m length) and the dummy section (0.5 m length) has not been taken into account in L/c (account of this would lead to decreased measured wavespeeds: c_{PVC} ≈ 667 m/s and c_{Al} ≈ 489 m/s with both L_A = 23.7 m and L_B = 24.7 m including a section of length 3.27 m which has an assumed effective wavespeed of 1000 m/s) and (3) measuring errors (lengths, times, etc). The combined opposite effects of (1) and (2) give corrected measured wavespeeds c_{PVC} ≈ 699 m/s and c_{Al} ≈ 505 m/s. The corrections are based on one-dimensional plane wave propagation in the entire pipeline, an assumption which might not hold for the short sections without internal tube, and on quarter-length and half-length standing wave patterns.

Note that, for a given external pipe, the wavespeed can be reduced by (1) decreasing the flow area (A_{flow}) and (2) by increasing the storage capacity (elasticity) of the internal tubes.

Note that wavespeed reduction has two effects: (1) it reduces pressure surges, which are proportional to the wavespeed (c) according to Joukowsky's law, and (2) it increases surge reflection times ($2L/c$ values), which lowers the chance of pressure reduction from wave reflections. The net outcome of these opposite effects is difficult to predict without a proper waterhammer analysis.

6 CONCLUSIONS

A reduction of system frequency (pressure wavespeed) can be obtained through application of internal rectangular tubes. However, the allowed pressure range is limited.

The experiments performed with internal rectangular tubes gave a significant improvement with respect to earlier experiments with internal flexible hoses. A larger and more predictable wavespeed reduction with less damping could be obtained.

The predicted and measured wavespeeds were within acceptable agreement in view of the test system, which was not ideal for an accurate measurement of wavespeeds.

The measured wavespeeds were lower than the theoretical predictions.

A higher mode of vibration could be excited with the internal PVC tube at low line pressures.

Care must be taken with the strength of the internal tubes. Some of the PVC sections collapsed and some aluminium sections exhibited small plastic deformation. Regular visual checks are therefore recommended.

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