ON THE SUPPRESSION OF COUPLED LIQUID/PIPE VIBRATIONS

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ABSTRACT

The vibration of liquid-filled pipe systems can be caused by internal (pressure surges) or external (rotating machinery) sources, or a combination of both. Liquid pulsations can be diminished by devices like an air chamber, whereas pipe vibrations can be reduced by suitable supports. The performance of vibration suppression devices in the system under consideration can be investigated by numerical simulation. This is usually done with conventional waterhammer or pipe-stress computer codes. Present-day software also permits fully-coupled liquid-pipe analyses. The analysis of suppression devices and their interaction with the entire system is an important tool in design and trouble shooting.

The dynamic behaviour of a simple system consisting of one pipe, one liquid column and one suppression device is studied herein by numerical simulation and physical experiment. Liquid-pipe interaction mechanisms are taken into account. A closed water-filled steel pipe is excited by external impact. The suppression device is a short piece of plastic pipe. The closed system prevents rigid-column behaviour of the liquid relative to the pipe, so that the local device does not decrease *amplitudes* of vibration. It does, however, shift the natural frequencies of the system, which is an important means of vibration control. This simple and cheap device may find its application in reducing fatigue problems and in noise reduction.

Keywords: Fluid-structure interaction (FSI), waterhammer, pipe, vibration, suppression.

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1. Introduction

Vibrations of liquid-transporting pipe systems can be caused by internal (waterhammer, turbulence) or external (earthquake, shaking machinery) sources. In both cases, pressure pulsations in the liquid interact with mechanical vibrations of the pipe walls. Fluid-structure interaction (FSI) phenomena in liquid-filled pipes have been extensively studied since the 1960s (Tijsseling 1996). In all studies, the emphasis has been on measuring and analysing the phenomenon rather than on assessing methods of reducing its consequences.

This paper presents initial results of an investigation into alternative methods of suppressing waterhammer-induced pipe vibrations. The vibrational behaviour of steel pipe systems can be changed greatly by a plastic pipe insert at the right location. The location and dimensions of the plastic insert should be determined from a coupled hydraulic and structural analysis. The aim is to move natural system frequencies away from exciting source frequencies and hence to diminish amplitudes of vibration.

A simple system, namely a single water-filled steel pipe, is investigated herein by physical experiment and numerical simulation. A plastic (ABS) pipe extension is used as a simple and cheap device to change the system's dynamic behaviour. In the experiment, the closed pipe is excited by external impact. In the simulation, a state-of-the-art FSI analysis is utilised.

2. Theory

Full details of the theory underlying the numerical simulations can be found elsewhere (Wiggert *et al.* 1985, 1987; Lavooij & Tijsseling 1991; Kruisbrink & Heinsbroek 1992; Tijsseling *et al.* 1996). A short description is given for guidance.

When a liquid-filled pipe is excited at some location, pressure waves in the liquid and stress waves in the pipe wall are generated and propagate away from the source of disturbance. The acoustic speeds of propagation are approximately

$$c_f = \left(\frac{K}{\rho_f}\right)^{\frac{1}{2}} / \left(1 + \frac{2RK}{eE}\right)^{\frac{1}{2}} \quad \text{and} \quad c_t = \left(\frac{E}{\rho_t}\right)^{\frac{1}{2}} \qquad (1, 2)$$

for pressure and *axial* stress waves, respectively. Section 8 (Nomenclature) gives the meaning of the symbols. Due to the FSI *Poisson coupling* mechanism, disturbances in the liquid pressure/velocity are accompanied by disturbances in the pipe axial stress/velocity and vice versa. In the method-of-characteristics (MOC) approach, this is expressed by the compatibility equations

Liquid

$$\frac{dP}{dt} \pm \rho_f c_f \frac{dV}{dt} - 2\nu \frac{\rho_f}{\rho_t} \left\{ \left(\frac{c_t}{c_f}\right)^2 - 1 \right\}^{-1} \left\{ \frac{d\sigma}{dt} \mp \rho_t c_f \frac{d\dot{u}}{dt} \right\} = 0 \qquad (3)$$

Pipe

$$\frac{d\sigma}{dt} \mp \rho_t c_t \frac{d\dot{u}}{dt} + \nu \frac{R}{e} \left\{ \left(\frac{c_t}{c_f}\right)^2 - 1 \right\}^{-1} \left\{ \frac{dP}{dt} \pm \rho_f c_t \frac{dV}{dt} \right\} = 0 \qquad (4)$$

which are valid along paths with directions $dz/dt = \pm c_f$, for Eq. (3), and $dz/dt = \pm c_t$, for Eq. (4), in the distance-time (z-t) plane. The equations (3) and (4) are simplified equations without gravity and friction terms. Water-hammer, Eq. (3), and steel-hammer, Eq. (4), equations remain for the hypothetical case where the Poisson ratio ν is zero.

When an acoustic wave reaches a pipe end or junction, it is (partly) reflected. A strong FSI *junction coupling* may then occur. For example, at an unrestrained, massless, closed pipe end, liquid and pipe variables are proportional to each other:

$$A_f P = A_t \sigma \quad \text{and} \quad V = \dot{u} \qquad (5, 6)$$

and at an unrestrained, massless, axial junction of two pipes:

$$\{ A_f (V - \dot{u}) \}_1 = \{ A_f (V - \dot{u}) \}_2 \quad \text{and} \quad \{ P \}_1 = \{ P \}_2 \qquad (7, 8)$$

$$\{ \dot{u} \}_1 = \{ \dot{u} \}_2 \quad \text{and} \quad \{ A_f P - A_t \sigma \}_1 = \{ A_f P - A_t \sigma \}_2 \qquad (9, 10)$$

3. Experiment

The FSI test rig at Dundee University has been extensively used in experiments with: a single pipe (Vardy & Fan 1986, 1989; Tijsseling & Vardy 1996), multi-pipe systems (Fan & Vardy 1994; Vardy *et al.* 1996) and cavitation (Fan & Tijsseling 1992; Tijsseling *et al.* 1996).

The single pipe system has been extended with a short plastic (ABS) pipe (Fig. 1). The maximum allowable pressure for the ABS pipe (class T) is 12 bar. A static pressure of about 6 bar, combined with dynamic pressures of ± 3 bar, is sufficient to prevent cavitation and over-pressures. The 0.25 m long ABS extension is screwed onto one end of the 4.50 m long steel pipe, which is struck axially at the other end by a solid steel rod moving at velocity 0.118 m/s. The impact is considered instantaneous and the rod exerts a constant force during the contact time of about 2 ms. Thereafter, the rod and pipe are separated (because of the arrival of stress waves from their remote ends).

Pipe and liquid vibrations are sensed with strain-gauges, piezo-electric pressure-transducers and a laser-Doppler vibrometer, at a sampling rate of 125 kHz. Table 1 gives the essential data on the apparatus.

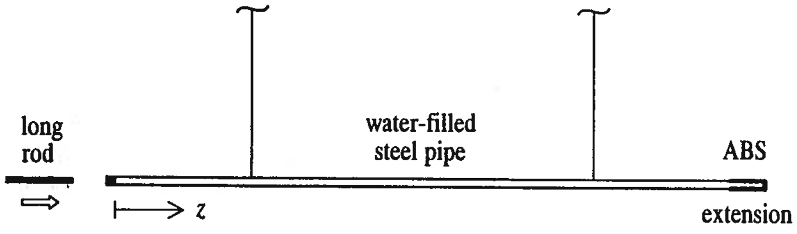

Figure 1. Schematic diagram of experimental apparatus.

Steel pipe	ABS pipe	Water
L = 4502 mm	L = 250 mm	K = 2.14 GPa
R = 26.01 mm	R = 22.8 mm	ρ_f = 999 kg/m^3
e = 3.945 mm	e = 7.4 mm	**Solid steel rod**
E = 168 GPa	E = 2.4 GPa	L_r = 5006 mm
ρ_t = 7985 kg/m^3	ρ_t = 1055 kg/m^3	R_r = 25.37 mm
ν = 0.29	ν = 0.42	E_r = 200 GPa
m_1 = 1.312 kg	m_2 = 0.3234 kg	ρ_r = 7848 kg/m^3

Table 1. Input data for simulations.

3.1. STATIC TESTS

The modulus of elasticity (2.4 $\pm$ 0.2 GPa) and the Poisson ratio (0.42 $\pm$ 0.02) of the ABS material were assessed at 22 °C from static compression/decompression measurements on an Instron 1196 test bench using a 200 mm long ABS pipe instrumented with three-way strain gauges at four circumferential positions midlength. During loading and unloading up to 0.4% strain, the test specimen showed almost linearly elastic behaviour (Fig. 2). It is noted that static tests cannot reveal any frequency-dependent behaviour.

The mass density (1055$\pm$2 kg/m^3) and the average cross-sectional wall-area (1232$\pm$5 mm^2) of the ABS pipe were obtained from immersion tests on an electronic weighing scale. The measured (with a vernier slide gauge) diameter (up to 0.1 mm differences) and wall-thickness (up to 0.2 mm differences axially and up to 0.4 mm differences circumferentially) of the ABS pipe were non-uniform. The relatively large circumferential variation in wall-thickness is assumed to partly explain the differences from average in Fig. 2. The mass of the ABS extension, including threaded thickening (at junction) but excluding end cap, was measured 0.5335 kg. In the numerical simulation, based on Table 1, a value of 0.325 kg was used (concentrated mass at junction neglected).

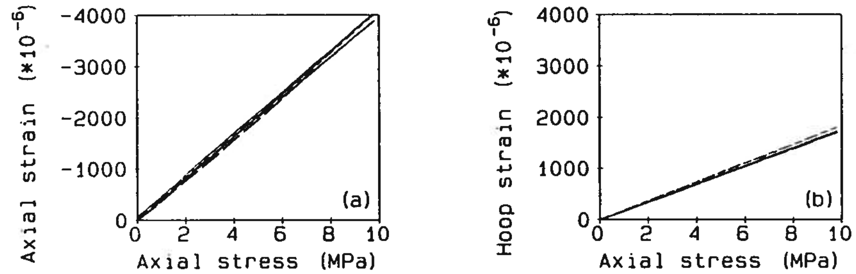

Fig. 2. ABS strain-stress relations obtained from static compression/decompression tests. Circumferential average and individual strains: (a) axial strains, (b) hoop strains.

4. Results

4.1. VALIDATION OF NUMERICAL MODEL

The numerical model outlined in Section 2 has been validated for water-filled steel pipes with closed ends and 90° junctions (see Section 3 for references). Typical results obtained in a 4.5 m long pipe are shown in Fig. 3. Two new elements in the present study are the *axial junction*, represented by the Eqs. (7-10), and the *plastic material* (ABS) of the pipe extension. The masses of the ABS pipe and its steel end cap (m_2) are of the same order of magnitude. The end cap, modelled as a lumped mass, slows down the axial ABS vibration significantly.

The complete system (Fig. 1) consists of two connected pipes, each with its own natural frequency and impedance. It is the liquid pulsations that make the problem interesting. They interact strongly with the vibrating pipes (FSI).

The theoretical wavespeeds in the system, according to the classical formulae (1-2), are c_f = 1354 m/s and c_t = 4587 m/s for the steel pipe and c_f = 574 m/s and c_t = 1508 m/s for the ABS extension. Due to Poisson-coupling-induced axial inertia forces in the pipe wall, the pressure wavespeeds, c_f, are slightly lower: 0.06% in the steel pipe and ~~0.1~~ 1.2% in the ABS pipe (Stuckenbruck *et al.* 1985). Due to Poisson-coupling-

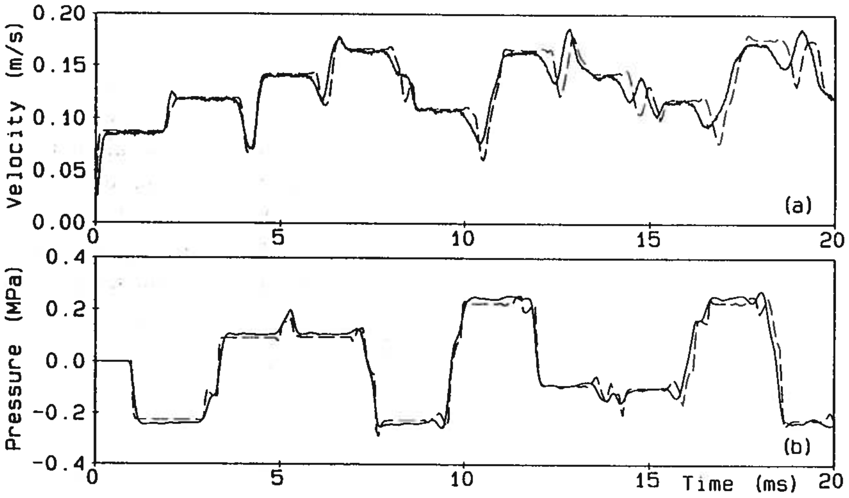

Fig. 3. Comparison calculation (- - -) / measurement (——) in 4.5 m long water-filled steel pipe: (a) axial pipe velocity at z = 0.047 m, (b) liquid pressure at z = 4.5 m.

induced pressure changes, the stress wavespeeds, c_t, are higher: 0.7% in the steel pipe and ~~8.9~~ 9.7% in the ABS extension.

Calculated dynamic pressures, axial strains and axial velocities agree well with the measured transients (Fig. 4). The amplitudes and main frequencies of vibration of both the steel and ABS pipe are predicted accurately. The main frequency of the liquid pulsation is predicted slightly too high. This could result from a small mismatch in wavespeeds or from errors in the assumed effective mass in the junction and boundary conditions.

4.2. PERFORMANCE AS VIBRATION SUPPRESSION DEVICE

A practical and convenient way to assess the influence of plastic inserts on the dynamic behaviour of steel pipe systems is by numerical simulations with a validated computer code. In Fig. 5, numerical results for the pipe with ABS extension (Figs. 1 & 4) are compared with numerical results for a 4.75 m long entirely steel pipe (comparable to Fig. 3). It is seen that the ABS piece shifts natural frequencies and makes wave fronts less steep, both factors being most evident from the pressures in Figs. 5d & 5e, where the main frequency is significantly lower. The amplitudes of fluctuations are roughly the same in both systems, except in Fig. 5c, where the strains in steel are compared with the much larger strains in ABS.

The similarity of amplitudes with and without an ABS piece is a consequence of the short length of the ABS piece relative to the steel pipe. Simulations with a 2.25 m long

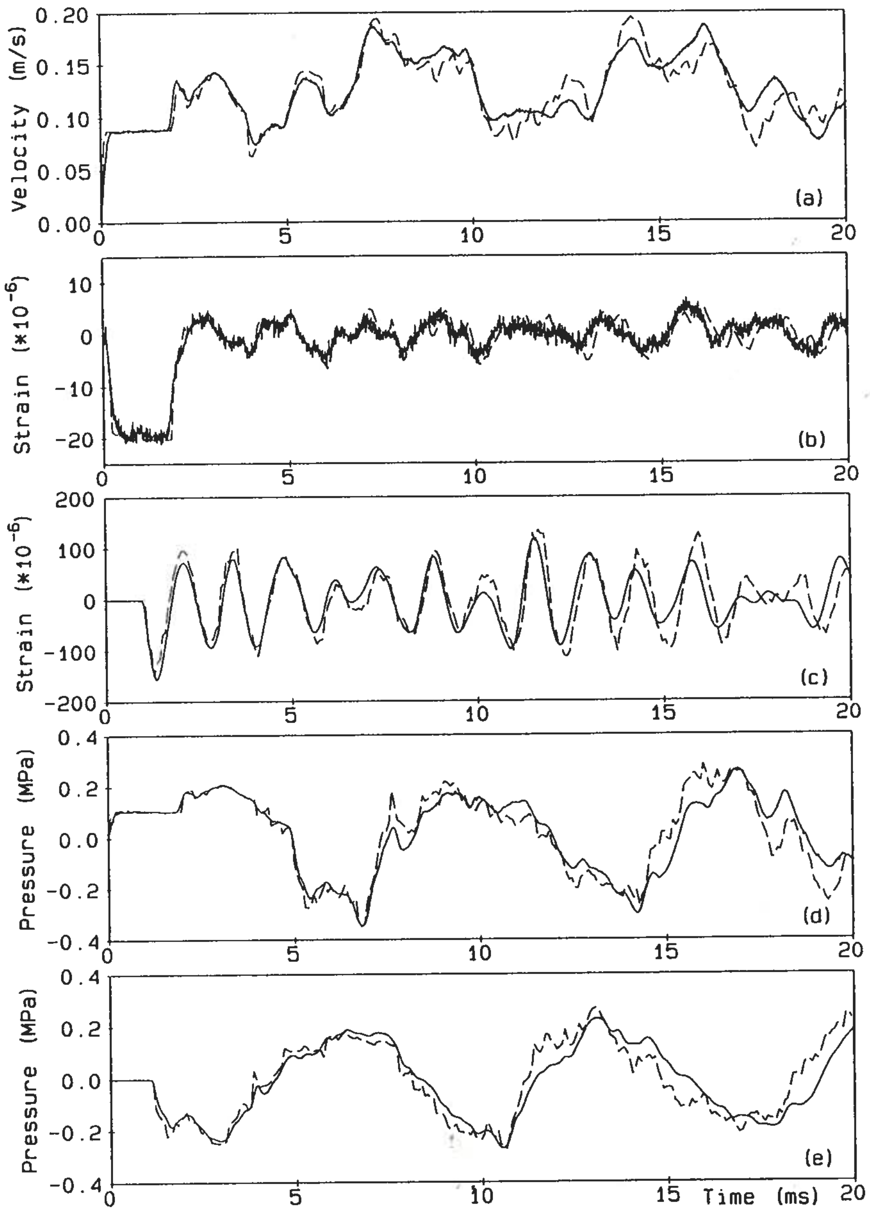

Fig. 4. Comparison calculation (- - -) / measurement (——) in 4.5 m long water-filled steel pipe with 0.25 m ABS extension: (a) axial (steel) pipe velocity at z = 0.047 m, (b) axial (steel) strain at z = 0.574 m, (c) axial (ABS) strain at z = 4.625 m, (d) pressure at z = 0.020 m, (e) pressure at z = 4.750 m.

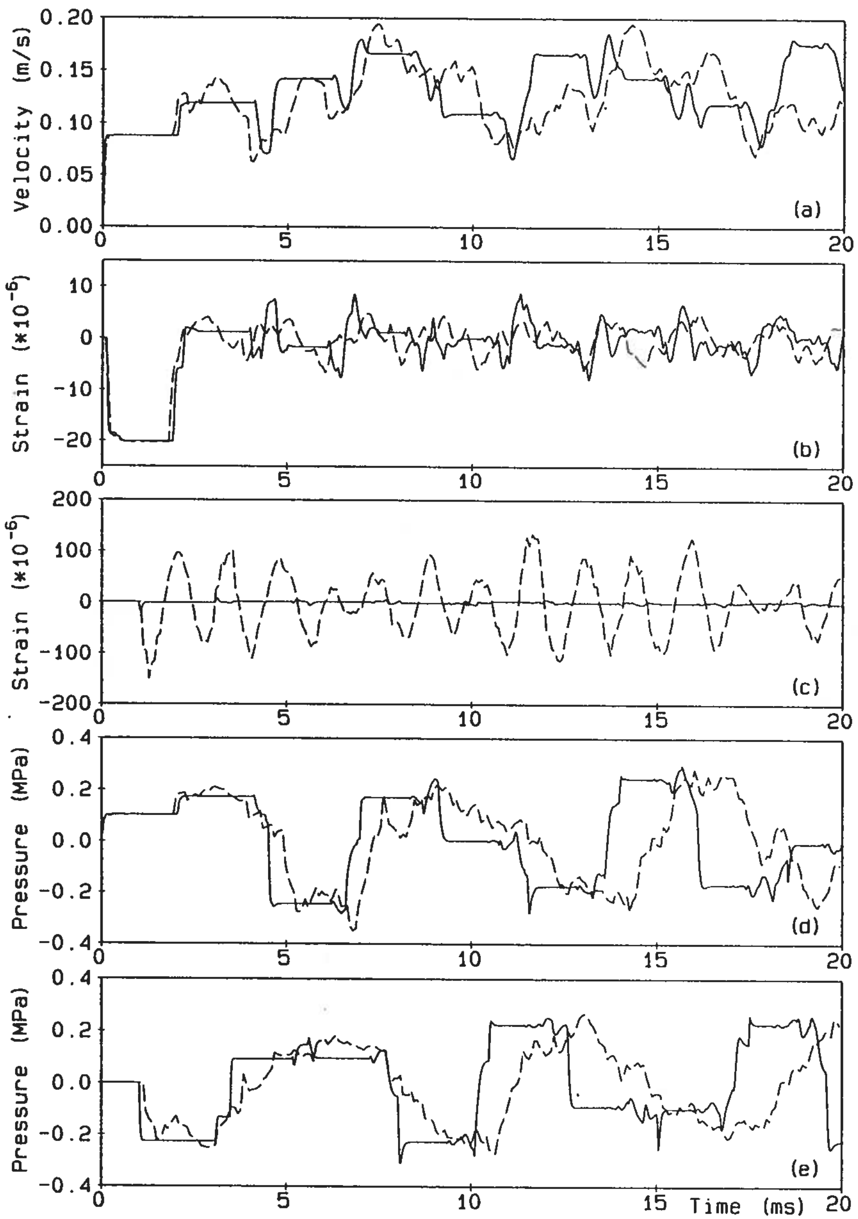

Fig. 5. Comparison 4.50 m steel pipe + 0.25 m ABS pipe (- - -) / 4.75 m steel pipe (——), both systems water-filled, calculations: (a) axial (steel) pipe velocity at z = 0.047 m, (b) axial (steel) strain at z = 0.574 m, (c) axial (ABS/steel) strain at z = 4.625 m, (d) pressure at z = 0.020 m, (e) pressure at z = 4.750 m.

ABS extension (not shown here) predict a 40% reduction in pressure amplitudes, relative to those predicted for a 6.75 m long entirely steel pipe. It is also expected that the ABS pipe would have greater influence if it were located close to the source of excitation.

5. Concluding Remarks

It has been shown that a short plastic extension significantly influences the axial vibration of a water-filled steel pipe. The natural frequencies of the system change and wave fronts become less steep. The amplitudes of vibration are not greatly reduced, when the plastic section is short relative to the steel pipe. Significant amplitude reduction can be obtained with long plastic sections however.

The system herein (Fig. 1) has been successfully simulated with a fully-coupled FSI mathematical model, so that, for the first time, the *junction coupling* (Eqs. 7-10) of two pipes of different materials and with different cross-sectional areas has been validated against experimental data.

Due to FSI *Poisson coupling*, the theoretical axial stress wavespeed in the water-filled ABS pipe is about 9% larger than the value given by classical theory. Other FSI wavespeeds were within 1% of the classical values.

The elastic behaviour of the short plastic (ABS) pipe can be satisfactorily represented by constant values for the Young modulus and the Poisson ratio. Reasonable agreement has been obtained between measurements and theoretical predictions even though visco-elastic, temperature and frequency-dispersion effects have been disregarded.

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8. Nomenclature

A	cross-sectional area	V	fluid velocity (cross-sectional average)
ABS	acrylonitrile butadiene styrene	z	axial coordinate (distance along pipe)
c	wave propagation speed	ε	axial strain ($\{ \sigma - \nu (R/e) P \} / E$)
E	Young modulus of elasticity	ν	Poisson ratio of pipe wall material
e	pipe wall thickness	ρ	mass density
FSI	fluid-structure interaction	σ	axial stress
K	liquid bulk modulus		
L	length	*Subscripts*	
m	mass of end cap		
MOC	method of characteristics	f	fluid
P	pressure (cross-sectional average)	r	rod
R	(inner pipe) radius	t	tube, pipe
t	time	1	1st pipe, junction side
$\dot{u}$	axial pipe velocity	2	2nd pipe, junction side

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