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THE INFLUENCE OF BEND MOTION ON WATERHAMMER PRESSURES AND PIPE STRESSES

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ABSTRACT

The influence of bend motion on the transient behaviour of liquid-filled pipe systems is studied numerically. The test problem is a reservoir-pipeline-valve system subjected to rapid valve closure. The three-dimensional pipeline has two restrained and four unrestrained bends. Pressure waves in the liquid excite the pipeline at the unrestrained bends (junction coupling), causing in-plane and out-of-plane bending, and axial and torsional motion of the pipes. As a result of contraction effects (Poisson coupling), axial motion is also caused by the pressure-driven radial motion of the pipe walls. On its turn, the vibrating pipeline generates pressure waves in the liquid.

In the numerical analysis, all relevant fluid-structure interaction (FSI) mechanisms are taken into account. The influence of each individual bend, and of each possible combination of bends, is examined. Main frequencies, extreme pressures and stresses, and anchor forces are calculated.

A detailed analysis of a pipeline's dynamic behaviour is presented. The knowledge obtained will be used in the development of practical guidelines and design rules for flexibly suspended pipe systems.

INTRODUCTION

The rupture and collapse of pipes due to waterhammer-related events is a well known and reasonably understood problem. Broken anchors and pipelines coming off their supports are less known, yet more common, issues in industry (Bürmann and Thielen 1988; de Almeida 1991; Hamilton and Taylor 1996). The main cause of the latter problems is the subject of this paper: waterhammer-induced pipe motion.

Steep pressure wavefronts passing unrestrained pipe bends will make the bends move. The bend movements provide elastic storage capability for the contained liquid (Wylie and Streeter 1993, pp. 356-357), which will affect the passing pressure waves. It is long recognised that this effect can be substantial (e.g. Blade *et al.* 1962; Wood and Chao 1971; Wiggert *et al.* 1985; Jezequel *et al.* 1994; Svingen 1996). The "breathing" of the pipes provides a secondary elastic storage capability, which also affects the pressure waves (Skalak 1956; Thorley 1969; Kojima and Shinada 1988; Vardy and Fan 1989).

This paper shows by means of 16 numerical FSI simulations of an existing reservoir-pipeline-valve test system that the motion of free bends has a profound effect on pressures, stresses and anchor forces when the system is excited by nearly instantaneous valve closures. The paper complements a previous paper (Heinsbroek and Tijsseling 1994).

TEST PROBLEM

Figure 1 shows the pipe system analysed. It is the FSI test rig of WL| Delft Hydraulics which has been described in detail in previous publications (Kruisbrink and Heinsbroek 1992; Heinsbroek and Tijsseling 1994; Heinsbroek 1997). The 77.5 m long, 0.11 m diameter, steel pipeline carries water from the air-vessel 1.5 m upstream of the fixed point A to the valve at fixed point H. The pipeline contains 6 mitre bends (B-G) and is suspended in wires to give a highly flexible system. In the present paper the bends B and G are fixed and the bends C-F are either free (unrestrained) or fixed (restrained). Table 1 gives the pipeline's geometrical and material properties.

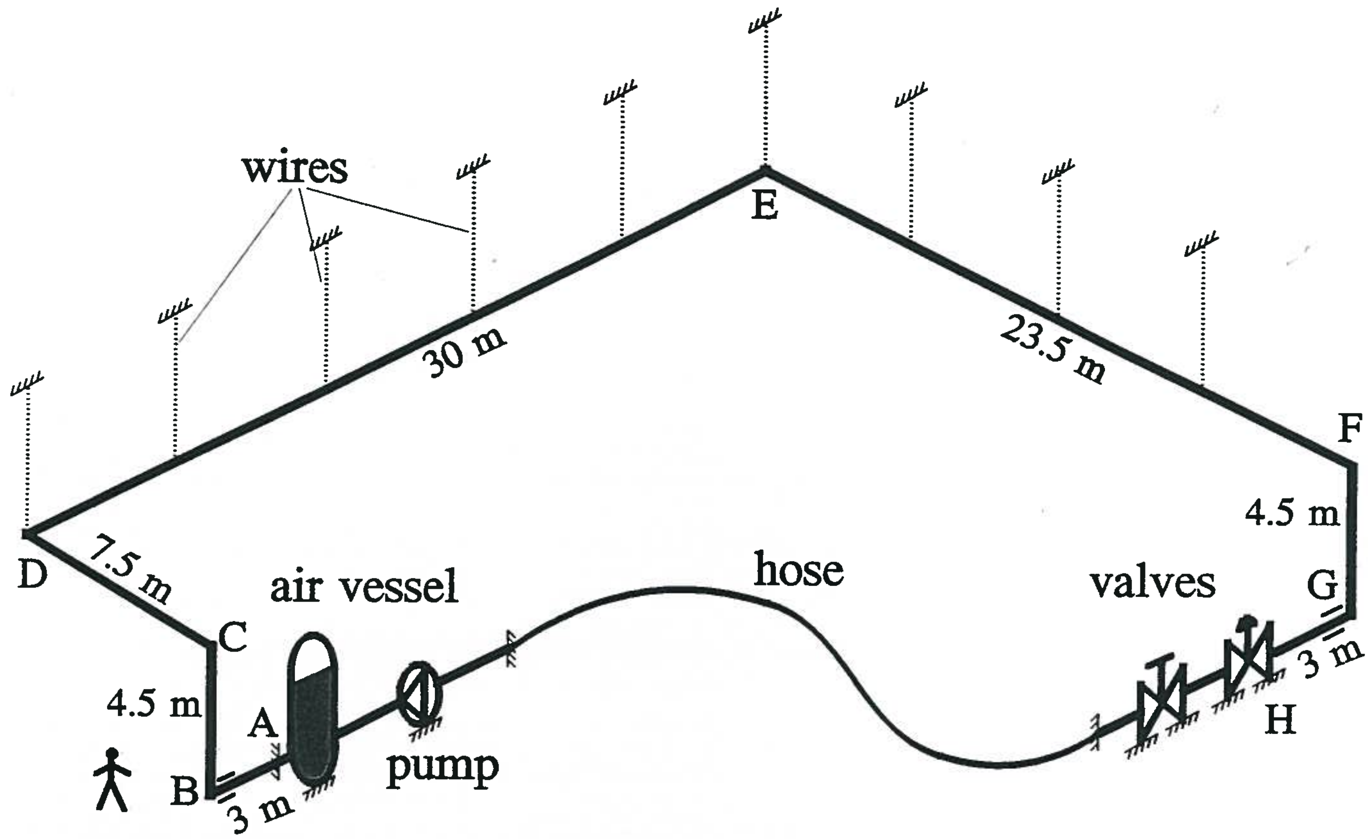

Figure 1. Schematic representation of pipe system analysed.

Steel pipeline	
length inner diameter wall thickness Young's modulus mass density Poisson's ratio stiffness of 1 m pipe	L = 77.5 m D = 109 mm e = 3 mm E = 200 GPa ρ_t = 8000 kg/m^3 ν = 0.3 $EA_t(1\text{m})^{-1}$ = 216 kN/mm
Support stiffness	
anchor A, axial anchor H, axial wire, vertical wire, horizontal (gravity)	k_A = 317 kN/mm (∞ herein) k_H = 214 kN/mm (∞ herein) k_{wv} = 0.3 kN/mm k_{wh} = 0.2 N/mm
Water	
bulk modulus mass density friction factor (American) initial pressure (at valve) initial velocity valve closure time	K = 2.19 GPa ρ_f = 998.2 kg/m^3 f = 0.031 P_0 = 6.0 barg V_0 = 0.3 m/s T_c = 0.01 s

Table 1. Geometrical and material properties of the pipeline.

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
C	r	u	r	r	r	u	u	u	r	r	r	u	u	u	r	u
D	r	r	u	r	r	u	r	r	u	u	r	u	u	r	u	u
E	r	r	r	u	r	r	u	r	u	r	u	u	r	u	u	u
F	r	r	r	r	u	r	r	u	r	u	u	r	u	u	u	u

Table 2. Definition of 16 test problems based on degrees of freedom of the bends C, D, E and F; r = restrained (fixed) bend, u = unrestrained (free) bend.

The influence of a single free bend (problems 2-5), of two free bends (problems 6-11), of three free bends (problems 12-15) and of four free bends (problem 16) on the pipeline's dynamic behaviour is investigated. Problem 1, in which all bends are fixed, serves as a reference. Table 2 lists the 16 defined test problems.

Test prob-lem [#]	Pressure [barg]	Tresca stress (N/mm^2)	Support force (N)	Frequency (Hz)
1	12.9	35	6314	4.06
2	12.2	41	6153	4.06
3	13.2	38	6282	4.07
4	12.0	34	6482	4.05
5	11.6	37	7333	4.02
6	12.0	55	4937	4.18
7	12.4	40	6487	4.06
8	11.8	40	7438	4.01
9	12.5	47	6028	4.54
10	12.7	40	7333	4.04
11	12.8	42	6336	4.10
12	12.2	59	4964	4.84
13	12.6	53	7334	4.14
14	12.4	41	6340	4.09
15	11.6	46	3882	4.67
16	10.9	59	0	5.01

Table 3. Calculated maximum pressures, maximum Tresca stresses, maximum support (in bends C, D, E and F) forces, all including steady-state (static) values, and main waterhammer frequencies.

NUMERICAL SIMULATION

The 16 test problems have been solved with the WL| Delft Hydraulics computer code FLUSTRIN (Lavooij and Tijsseling 1991; Heinsbroek 1997). The code is based on extended waterhammer and beam theories for liquid and pipes, respectively. The pipes are allowed to move in longitudinal and lateral directions and to twist around their axes. The radial pipe motion is quasi-statically related to the fluid pressure. All FSI coupling mechanisms (Tijsseling 1996) are taken into account and pipe flexure is modelled through Bernoulli-Euler beam theory. The bends are pipe junctions with local conservation of mass and momentum. The Method of Characteristics and the Finite Element Method solve the fluid and structural equations, respectively. Correct FSI coupling is attained through a fluid-structure iteration process.

CALCULATED RESULTS

The 16 pressure-head histories calculated at the valve for each individual problem are displayed in Fig. 2. The maximum pressures, maximum Tresca stresses, maximum anchor forces (largest of force components in x_1, x_2 and x_3 directions) and main waterhammer frequencies in the first second after valve closure are given in Table 3.

No free bend

Graph 1 (non) in Fig. 2 is obtained when all bends are fixed. The signal initially resembles the classical waterhammer square wave (with Joukowsky $\Delta P = \rho_f c_f \Delta V$ = 3.8 bar derived from ρ_f = 998.2 kg/m^3, c_f = 1257 m/s and ΔV = 0.3 m/s), but deviates from it when time proceeds. The deviation is due to the "breathing" of the pipes (FSI Poisson coupling). Because of the long lengths of longitudinally unrestrained pipes which are not allowed to move at the (b)ends, a cumulative effect of pressure disturbances develops (as a result of coupled radial/longitudinal pipe vibrations). The increasing high pressure peaks correspond to a beat phenomenon predicted by

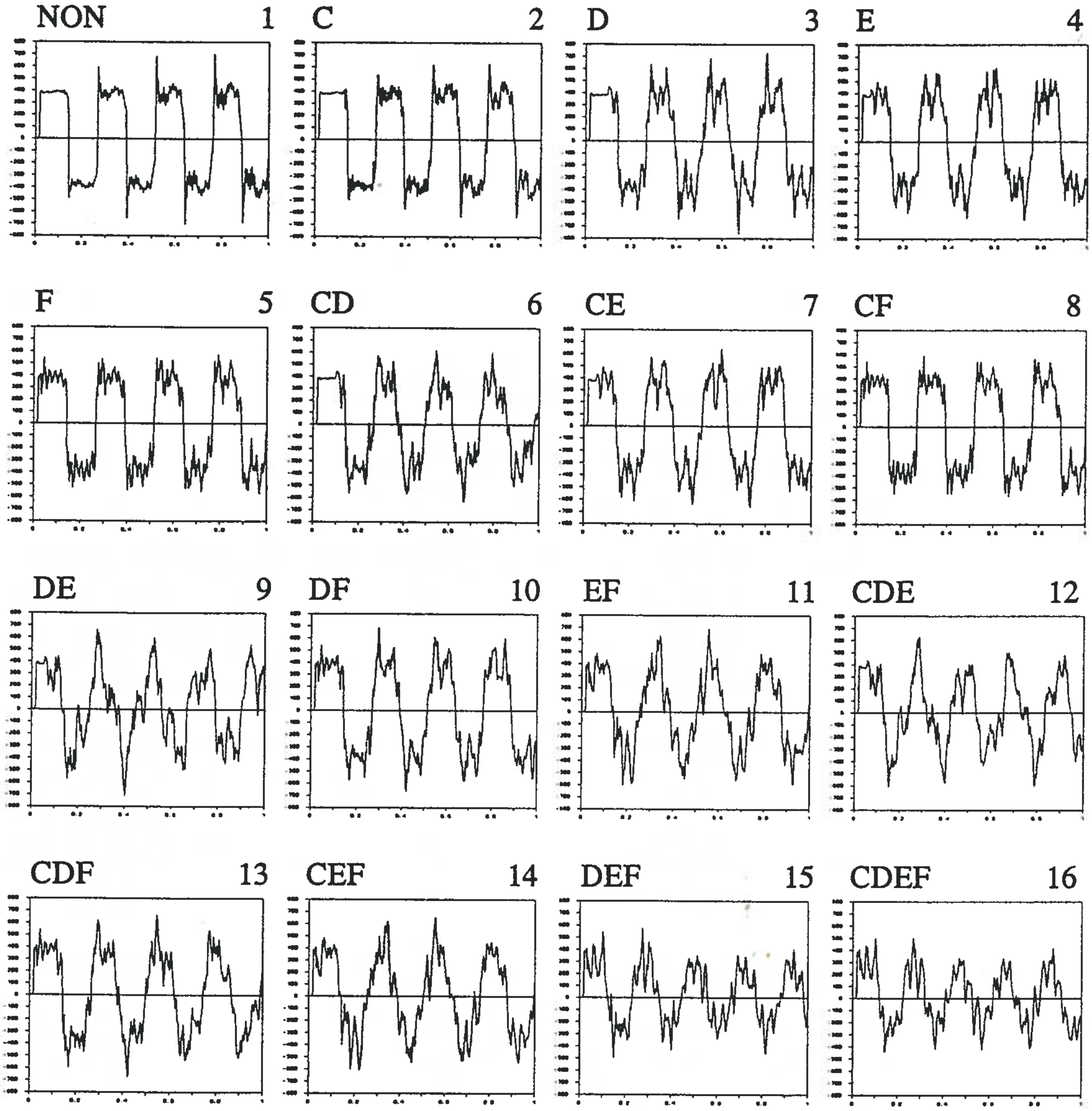

Figure 2. Calculated dynamic pressure-head histories at the valve (location H). Horizontal axes: 0 s < time < 1 s; vertical axes: −800 m < pressure head < +800 m.

Wiggert *et al.* (1986), measured by Budny *et al.* (1991, Fig. 4a) and discussed in more detail by Tijsseling (1997). The fixed bends prohibit waterhammer-induced lateral and torsional pipe vibrations, and they prevent longitudinal rigid-body pipe motion.

One free bend

Graph 2 (C) shows that the remote bend C has a relatively small influence on the pressure history at the valve; graph 2 (C) is very similar to graph 1 (non). Graph 3 (D), where the remote bend D is free, shows a much larger influence; pressure peaks about 90% (!) larger than "Joukowsky" occur. In graph 4 (E), bend E which connects the two longest pipes in the system, is free to move with a notable effect on the pressure waves. In graph 5 (F), the vibrating near bend F introduces high-frequency oscillations around the main waterhammer wave.

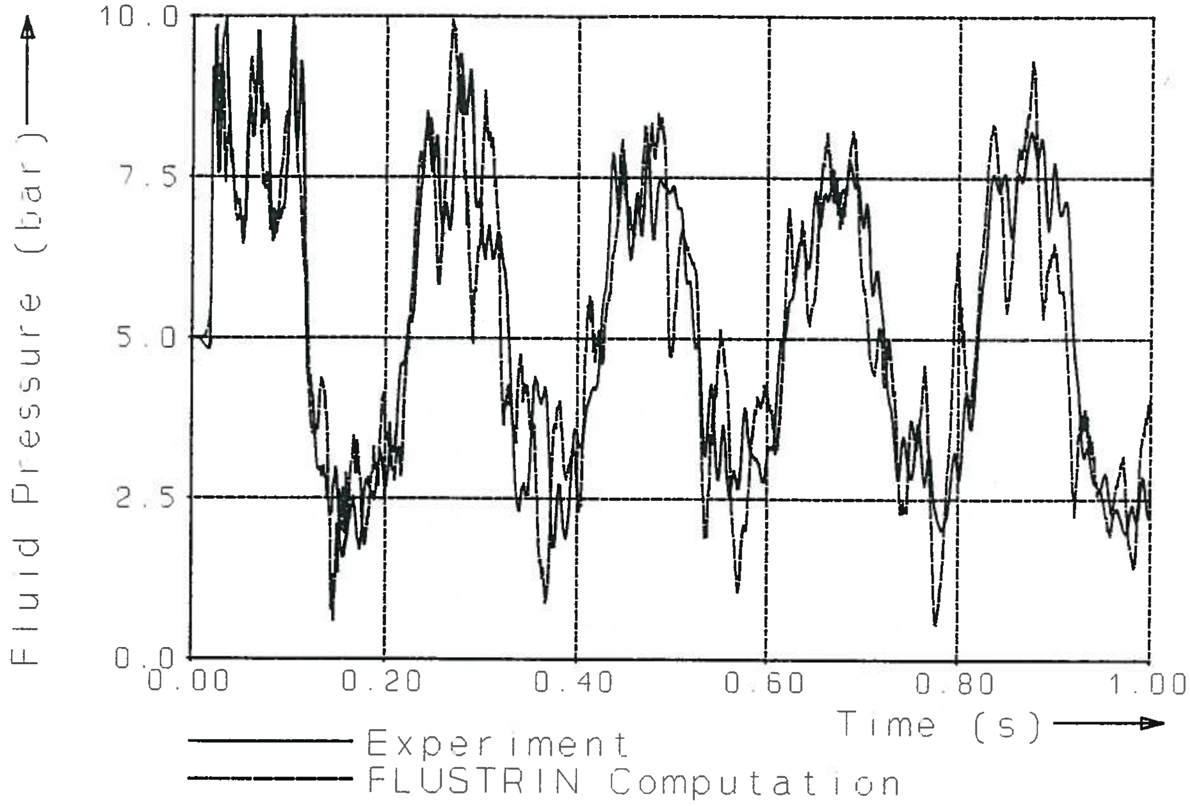

Figure 3. Measured and calculated dynamic pressure histories at the valve (location H) with partly unrestrained bends B and G (Kruisbrink and Heinsbroek 1992). Horizontal axis: 0 s < time < 1 s; vertical axis: −5 bar < dynamic pressure < +5 bar.

Two free bends

The graphs 6 (CD), 7 (CE) and 8 (CF), when compared to the corresponding graphs 3 (D), 4 (E) and 5 (F), show again that bend C has a relatively small influence on the dynamic pressures. Graph 10 (DF) can, with some imagination, be seen as a combination of the graphs 3 (D) and 5 (F), but the graphs 9 (DE) and 11 (EF), which have adjacent free bends, cannot directly be related to the corresponding graph combinations 3 (D) - 4 (E) and 4 (E) - 5 (F), respectively. This is understandable because adjacent free bends allow torsional motion and associated new vibrational modes.

Three free bends

The graphs 12 (CDE) and 13 (CDF), when compared to the corresponding graphs 9 (DE) and 10 (DF), show that the influence of bend C becomes more pronounced when it is combined with a free adjacent bend D, but, from the graphs 14 (CEF) and 11 (EF), that its influence is still small when combined with the non-adjacent bends E and F.

Four free bends

The four free bends in graph 16 (CDEF) give a similar result to the three free bends in graph 15 (DEF), confirming the small influence of bend C. The trident shape just after valve closure, which is typical for the graphs 15 (CDE) and 16 (CDEF), has been validated by experimental data (see Fig. 3). The main waterhammer frequency derived from graph 16 (CDEF) is 5 Hz which would relate to a (physically unrealistic) classical waterhammer wavespeed of 1550 m/s (!) when $c_f/(4L)$ is taken as the system's first fundamental frequency. Axial stress waves, travelling at a speed of $\sqrt{E/\rho_t}$ = 5000 m/s, provoking pressure changes (precursors) and exciting the "pumping" free bends, allow a faster than sonic propagation of disturbances in the water. From Table 3, test problems 9, 12, 15 and 16, it is seen that the free bend combination DE is responsible for a significant increase in the main waterhammer frequency compared to its classical value of 4 Hz (which is $c_f/(4L)$ with c_f = 1257 m/s and L = 77.5 m).

Pressures, stresses and anchor forces

The maximum pressure at the valve predicted by "Joukowsky" ($P_{max} = P_0 + \Delta P = 6.0$ barg + 3.8 bar = 9.8 barg) is exceeded in all 16 test problems, but this maximum pressure is of shorter duration than the classical $2L/c_f$ because of the higher frequency of structure-induced pressure fluctuations.

The maximum stress in the pipeline is the lowest when all, or all but one, bends are fixed (problems 1 to 5). The relatively high stress in problem 2 (C) is attributed to the aforementioned (Poisson-coupling) beat phenomenon. The highest stresses occur when at least the bends C and D are free (problems 6, 12, 13 and 16).

The smallest anchor forces are found in the problems 6, 12, 15 and 16. Trivially, the last case lacks bend anchor forces in C, D, E and F; the adjacent pipes carry the entire waterhammer load. In all other problems the waterhammer load is carried both by anchors and pipes.

Trends

The pipe and bend vibrations introduce higher-frequency components in the classical waterhammer square wave with pressure peaks exceeding "Joukowsky". The square waterhammer wave becomes "triangular" when *either* of the bends D and E is free to move. The main waterhammer frequency increases substantially when *both* the bends D and E are free and the highest stresses occur when both the bends C and D are free.

CONCLUSIONS

The classical theory of waterhammer predicts a square-wave pressure history (at the valve, friction neglected) in a reservoir-pipeline-valve system subjected to sudden valve closure (Chaudhry 1987, p. 14; Wylie and Streeter 1993, p. 50, p. 65). It has been shown herein that this square wave is distorted in flexibly supported pipelines. Moving pipe (b)ends may (1) introduce higher-frequency pressure oscillations, (2) make square waves "triangular" and consequently wavefronts less steep, (3) increase the system's main frequency and (4) invalidate application of Joukowsky's formula.

Fully fluid-structure coupled models are required to analyse flexible pipelines subjected to rapid excitation, in particular to accurately assess the anchor forces.

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