FLUID-STRUCTURE INTERACTION IN NON-RIGID PIPELINE SYSTEMS
- LARGE SCALE VALIDATION TESTS -
(EUREKA PROJECT 274)

A.C.H. KRUISBRINK and A.G.T.J. HEINSBROEK
Industrial Technology Division
Delft Hydraulics
P.O. Box 177, 2600 MH Delft, The Netherlands

ABSTRACT

To validate the fluid-structure interaction computer code FLUSTRIN, developed by DELFT HYDRAULICS, experiments are performed in a large scale 3D test facility. The test facility consists of a flexible pipeline system which is suspended by wires. Pressure surges, which excite the system, are generated by a fast acting shut off valve. Dynamic pressures, structural displacements and strains (in total 70 signals) are measured under well known initial and boundary conditions.

The experiments are simulated with FLUSTRIN, which solves the acoustic equations using the method of characteristics (fluid) and the finite element method (structure).

The agreement between experiments and simulations is shown to be good.

The FLUSTRIN computer code enables the user to determine dynamic fluid pressures, structural stresses and displacements in a liquid filled pipeline system under transient conditions. As such, the code may be a useful tool to process and mechanical engineers in the design and operation of pipeline systems.

NOMENCLATURE

A_f	= cross-sectional discharge area	u_x	= vertical lateral displacement
A_t	= cross-sectional pipe wall area	u_y	= horizontal lateral displacement
c_f	= pressure wave speed	u_z	= axial displacement
D	= internal diameter	V	= fluid velocity
E	= Young's modulus of pipe material	V_r	= relative fluid velocity
e	= pipe wall thickness	z	= distance along pipe axis
f	= Darcy-Weisbach friction factor	γ	= pipe elevation angle
G	= shear modulus	$\varepsilon_{1,2,3}$	= strain

g = gravitational acceleration
H = fluid pressure head
I_t = moment of inertia
J = polar moment of inertia
K_f = fluid bulk modulus
L = pipe length
P = fluid pressure
t = time
θ_z = torsional rotation
ν = Poisson's ratio
ρ_f = fluid mass density
ρ_t = pipe mass density
$\sigma_{1,3}$ = normal stresses
σ_z = axial stress
τ_{13} = shear stress
ξ = loss coefficient

INTRODUCTION

In general the transient behaviour of fluid-filled pipelines is determined by both the hydraulic and structural conditions of the system. Hydraulic conditions like flow and pressure changes, caused by e.g. valve closure, will change structural conditions like pipe strains and displacements. On the other hand structural conditions like pipe expansion or contraction and bend motion will change the hydraulic conditions. This interaction between hydraulic and structural conditions is referred to as fluid-structure interaction (FSI).

For a long time, the transient flow in pipeline systems has been described by the classical water hammer theory, in which the structure is assumed to be rigidly supported. Structural conditions like the pipe elasticity, the wall thickness and the way the pipeline is supported can be found only in the propagation speed of pressure waves [27]. Structural dynamics are not considered in this approach.

In the second half of this century, the water hammer theory is extended by taking into account 1) pipe wall inertia effects related to the expansion and contraction of the pipe under pressure changes (the so-called Poisson effect), 2) the pipe motion caused by friction (friction effect) and 3) the motion caused at pipe ends, elbows or tees (junction effect). These structural motions influence the hydraulic conditions. The interactions between hydraulic and structural conditions are referred to as Poisson, friction and junction coupling respectively. In most cases the junction coupling is dominant.

Up to now the phenomenon of fluid-structure interaction has been studied by many researchers [1-3, 5-14, 16-26]. However, the number of experimental studies is restricted to a few tests at a rather small scale [7, 8, 18, 19, 21, 24, 25, 26]. The experiments described in [18, 19, 21, 24, 26] have already been simulated with previous versions of the fluid-structure interaction computer code FLUSTRIN [10, 11, 16, 17].

In the present paper experiments are described which are performed in a large scale test facility at DELFT HYDRAULICS. The 3D test loop is especially designed to obtain significant FSI effects. The relatively flexible pipeline system allows axial, lateral and torsional motion. The results of the experiments are used to validate the FLUSTRIN computer code.

FLUID-STRUCTURE INTERACTION MODEL

The transient behaviour of fluid-filled pipeline systems is governed by acoustic waves propagating in both fluid and pipe. Pressure waves in the fluid, referred to as water hammer, coexist with axial, lateral and torsional stress waves in the pipe wall. Due to radial expansion and contraction of the pipe wall, fluid and pipe interact at the fronts of axial waves. Interactions between all four types of waves may take place at pipe junctions. The behaviour of the fluid is governed by extended water hammer equations [10, 13, 21], whereas the dynamics of the pipe is modelled by standard beam theory [4].

The theory is applied under the assumption that the pipe is thin-walled and linearly elastic. The radial inertia of the pipe wall is neglected. The basic equations are:

Fluid Equations
The fluid behaviour is described by extended equations of momentum and mass conservation:

$$\frac{\partial V}{\partial t} + g\frac{\partial H}{\partial z} + \frac{f\, V_r \,|\, V_r \,|}{2D} = 0 \tag{1}$$

$$\frac{1}{c_f^2}\frac{\partial H}{\partial t} + \frac{1}{g}\frac{\partial V}{\partial z} - \frac{2\nu}{gE}\frac{\partial \sigma_z}{\partial t} = 0 \tag{2}$$

The extended equation of momentum conservation is equal to its classical equivalent, with the exception of the friction term, in which the fluid velocity V is replaced by the relative fluid velocity $V_r = V - \dot{u}_z$ ($\dot{u}_z$ = axial velocity of the pipe wall). In the extended equation of mass conservation an extra term is added to account for the Poisson effect.

The pressure wave speed c_f is defined as:

$$c_f = \left[\frac{\rho_f}{K_f} \left[1 + \frac{DK_f}{Ee} \right] \right]^{-1/2} \tag{3}$$

Structural Equations
The structural behaviour is described by the equations of motion, applied in axial, lateral and torsional direction. The local coordinate system used is given in figure 1.

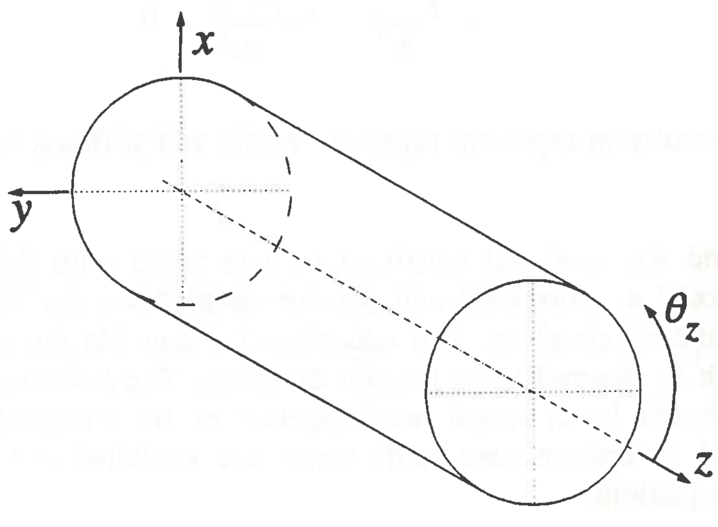

Figure 1. Local coordinate system.

Axial motion: The equation of motion is given by:

$$\rho_t A_t \frac{\partial^2 u_z}{\partial t^2} - EA_t \frac{\partial^2 u_z}{\partial z^2} = \frac{\nu D A_t}{2e} \frac{\partial P}{\partial z} + \frac{\rho_f A_f f V_r \mid V_r \mid}{2D} + \rho_t A_t g \sin\gamma \tag{4}$$

The first term at the left hand side represents the axial inertia of the pipe. Note that the fluid mass density is not included. The second term accounts for the axial stiffness of the pipe. At the right hand side the various external loads on the pipe in axial direction are given. The first term represents the force due to the fluid pressure. In this term the Poisson's ratio is present. The second term represents the force due to fluid friction. Again the relative fluid velocity V_r is used. The third term describes the axial component of the gravitational force on the pipe. Note that also here the fluid mass density is not present.

Lateral motion: The Bernoulli-Euler beam theory has been applied. The equations of motion in the x- and y-direction (see figure 1) are respectively:

$$(\rho_t A_t + \rho_f A_f) \frac{\partial^2 u_x}{\partial t^2} + EI_t \frac{\partial^4 u_x}{\partial z^4} = -(\rho_t A_t + \rho_f A_f) g \cos\gamma \tag{5}$$

$$(\rho_t A_t + \rho_f A_f) \frac{\partial^2 u_y}{\partial t^2} + EI_t \frac{\partial^4 u_y}{\partial z^4} = 0 \tag{6}$$

The first term at the left hand side represents the lateral inertia of the pipe. The fluid inertia is included. The second term describes the bending stiffness to which the fluid does not contribute. At the right hand side, the lateral component of the gravitational force is given, only for the non-horizontal x-direction.

Torsional motion: The equation of motion is given by:

$$\rho_t J \frac{\partial^2 \theta_z}{\partial t^2} - GJ \frac{\partial^2 \theta_z}{\partial z^2} = 0 \tag{7}$$

The first and second term represent torsional inertia and stiffness respectively.

Interaction

Both the fluid and the structural equations contain terms with fluid as well as structural quantities. The axial equations (2) and (4) are coupled via the Poisson's ratio, which is referred to as Poisson coupling. The equations (1) and (4) are coupled via the friction coefficient, which is referred to as friction coupling. The Poisson and friction effects are caused by distributed loads which are modelled in the differential equations. Junction effects are caused by concentrated loads which are modelled in the boundary conditions. They couple all equations.

Strains

Internal forces and moments are derived from structural displacements, according to the standard FEM. Normal and shear stresses are derived from the internal forces and moments and the fluid pressure under the assumptions: 1) plane stress conditions, 2) cross sections remain plane and perpendicular to the neutral axes, 3) effective shear area can be applied for shear forces. From the axial stress σ_1, hoop stress σ_3 and shear stress τ_{13} (see figure 2) the strains in the directions 1 to 3 are computed:

$$\varepsilon_1 = \frac{1}{E}(\sigma_1 - \nu\sigma_3) \tag{8}$$

$$\varepsilon_2 = \frac{2\tau_{13}(1+\nu) + (\sigma_1 + \sigma_3)(1-\nu)}{2E} \tag{9}$$

$$\varepsilon_3 = \frac{1}{E}(\sigma_3 - \nu\sigma_1) \tag{10}$$

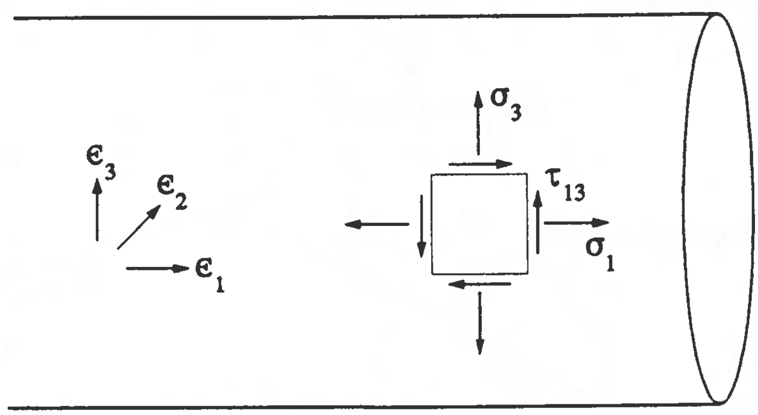

Figure 2. Stresses

FLUSTRIN COMPUTER CODE

The FLUSTRIN computer code is suitable to solve FSI problems in serial pipeline systems in a convenient way. The hydraulic components presently incorporated in the code are: Pumps, control and check valves, air vessels, surge towers, reservoirs, resistances and pipe rupture. The structural components are: Rigid supports, springs, hangers and dampers. Fluid-structure interaction takes place along the pipes (Poisson and friction coupling) and at elbows, dead ends and axially moving components (junction coupling).

The numerical procedures in FLUSTRIN, used to solve the basic equations together with their initial and boundary conditions, are described in detail in [10]. The fluid equations (1) and (2) are solved by the method of characteristics (MOC). The structural equations (4) to (7) are treated by the finite element method (FEM). For the time integration the Newmark $\beta = 1/4$ method is applied.

An iteration process takes care of the FSI coupling mechanisms.

TEST FACILITY

The test facility is a water-filled closed loop consisting of a variable speed pump, an air vessel, a welded pipe with six elbows (square bends), a fast acting shut off valve and a control valve (see figure 3). A flexible hose closes the loop between the control valve and the pump. The structural boundary conditions of the system are:

- Rigid supports at the locations A and H. These supports allow neither translation nor rotation.
- Bend supports at the locations B and G. These supports only allow translation and rotation around the X_1-direction.
- Suspension wires located at about every 6 m along the pipe, which allow translation in the horizontal plane (bouncefree) and rotation in all directions.
- An adjustable spring at location E. The spring can be mounted in the X_1- or X_3-direction.

The stiffness of the "rigid" supports was measured under static conditions. The relationship between force and displacement appeared to be linear, indicating that the supports can be considered as springs with a constant stiffness. The axial stiffness of the supports A and H is 316778 kN/m and 214307 kN/m repectively. The stiffness of the suspension wires is specified by the manufacturer. The stiffness of the adjustable spring varies from 30 to 100 kN/m.

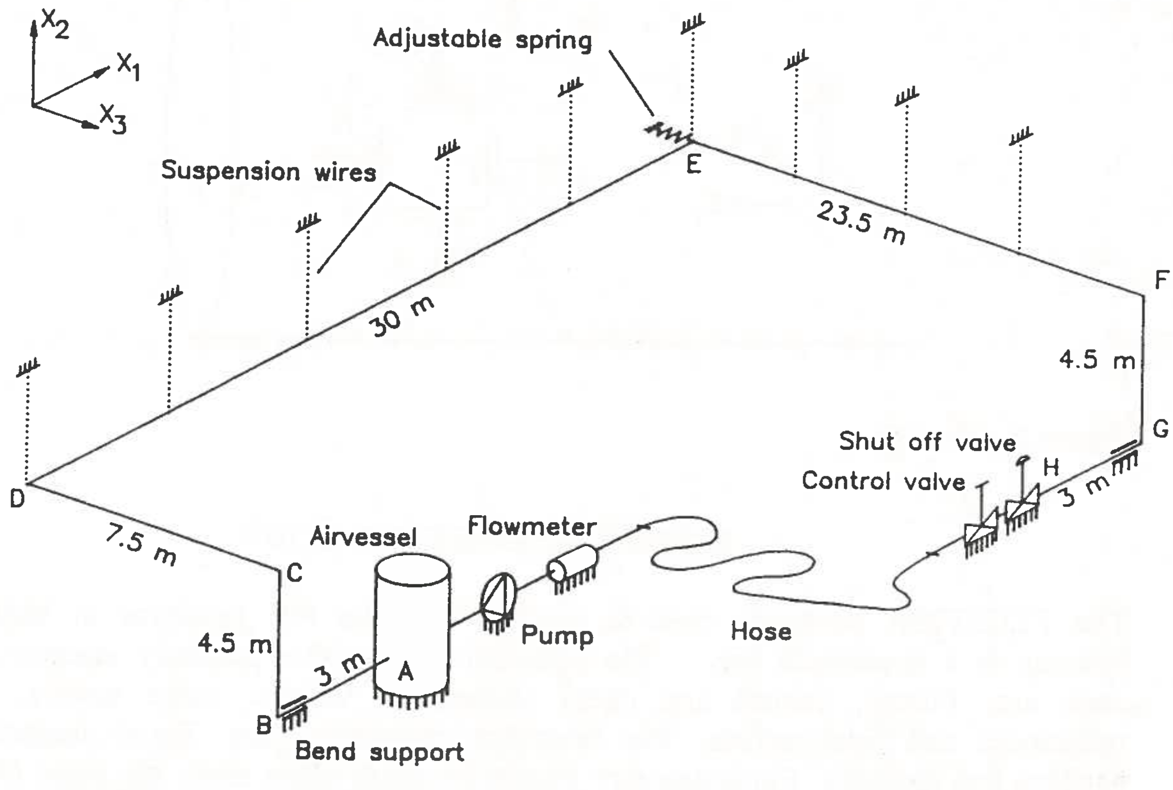

Figure 3. The tested and simulated pipeline system.

EXPERIMENTS

The experiments were carried out starting from steady state conditions. Transients were generated by closing the fast acting shut off valve at the downstream end. Different initial and boundary conditions were obtained by varying the initial flow rate, the closure time of the shut off valve and the stiffness and direction of the adjustable spring. During an experiment, the following signals were measured:

- The steady state flow rate using an electromagnetic flowmeter.
- 2 steady state fluid pressures using static pressure transducers.
- 6 dynamic fluid pressures using piezo-electric transducers.
- 9 structural displacements using inductive transducers. Eight displacements of elbows were measured in different directions. The displacement of the shut off valve was measured in axial direction.
- 3 forces in suspension wires using load cells.
- 48 pipe wall strains at four locations along the pipe. At each location four three-way strain gauges (rosettes) were used (see figure 4).
- The valve disc position using an inductive transducer.

The signals were recorded simultaneously with a sample rate of 800 Hz during 5 seconds.

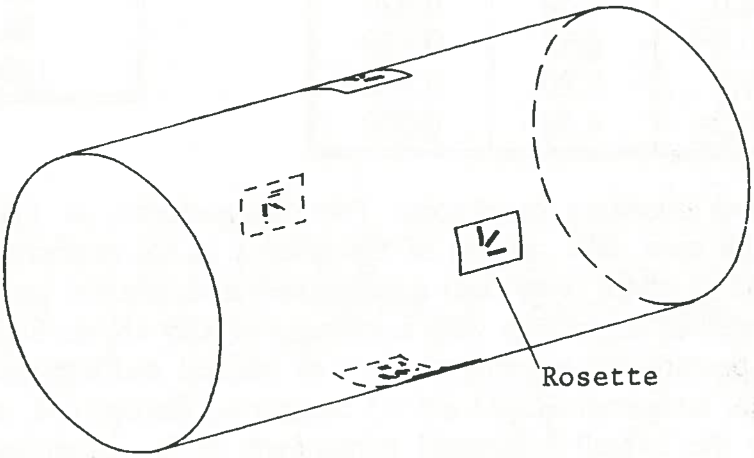

Figure 4. Strain gauge arrangement.

COMPUTER SIMULATIONS

The test facility as shown in figure 3 is simulated with the FLUSTRIN computer code.

The pipeline between locations A and H is divided into 7 straight pipe sections, connected by 6 elbows (B to G). The pipe length between the rigid supports A and H is 76 m with an additional length of 1.5 m between the airvessel and rigid support A. This leads to a total length of L is 77.5 m, corresponding to 51 elements of 1.5 m. The internal pipe diameter D is 108.7 mm, the wall thickness e is 3.07 mm, the effective friction factor f (including bend losses) is 0.031, the mass density ρ_f is 8000 kg/m^3, the Young's modulus E is $2.00 * 10^{11}$ N/m^2 and the Poisson's ratio ν is 0.3.

The fluid is water with a density ρ_f of 998.23 kg/m^3 and bulk modulus K_f of $2.19 * 10^9$ N/m^2 (temperature: 20 °C). The pressure wave speed c_f according to equation (3) is 1257 m/s.

Hydraulic boundary conditions: The upstream boundary consists of a pump and an air vessel. The pump is modelled by its head and efficiency characteristics (see table 1) with constant suction head. The area of the air vessel is 1.109 m^2 and the height is 2.865 m. Air expansion is assumed to be adiabatic. The downstream boundary consists of a shut off valve and a control valve. The shut off valve is modelled using its pressure loss characteristic (see table 2) with constant downstream head. The control valve was not used during the experiments, and is not modelled.

Table 1
Pump characteristics
(speed of 960 rpm)

discharge [m³/h]	head [m]	efficiency [-]
0.0	8.66	0.000
10.0	8.65	0.250
20.0	8.60	0.370
30.0	8.50	0.465
40.0	8.30	0.570
50.0	7.90	0.630
60.0	7.40	0.670
70.0	6.75	0.680
80.0	5.80	0.660
90.0	4.50	0.560

Table 2
Pressure loss characteristic
Shut off valve
(diameter D = 107 mm)

$(\Delta P = \xi \ \frac{1}{2}\rho V^2)$

valve position [% open]	ξ [-]
20.0	340.17
40.5	45.03
59.1	22.19
80.6	11.82
89.5	10.03
100.0	8.80

Structural boundary conditions: The displacements of the supports A and H are assumed to be zero. The motion of the elbows is not restrained, with the exception of elbows B and G where only axial displacement and rotation are allowed. The suspension wires are modelled as springs with a stiffness of 285 kN/m. Since the wires may assume non-vertical positions, the springs have both vertical and horizontal stiffness components. The horizontal component (.243 kN/m) can not be disregarded, since it is relatively large compared to the overall horizontal component of the structural stiffness. Concentrated masses like rigid and bend supports, and attached equipment (e.g. strain gauge cable boxes and suspension clamps) are neglected.

Initial conditions: The measured steady state flow rate and air vessel pressure prescribe the hydraulic initial conditions, whereas the structural initial conditions consist of computed pipe displacements due to hydraulic and gravitational loads.

RESULTS

Results are presented of a representative (cavitation free) experiment and corresponding simulation. The initial flow rate is 0.3 m/s (pump speed 169 rpm). The initial pressure in the air vessel is 7 bar (pump suction head 59.70 m and downstream head 59.79 m) and its initial water level is 1.732 m. The closure time of the shut off valve is 10 ms. The adjustable spring is not used here.

Pressures

Figures 5 and 6 show measured and computed dynamic pressures near the locations E and H respectively. The agreement in amplitude and frequency is very good, although the measured extreme values are slightly exceeded by the simulation.

The pressure signals show a more or less triangular shape where the classical theory would predict a block shape. The maximum pressure at the shut off valve is about 5 bar where the classical theory would predict 3.76 bar (Joukowsky: $\Delta P = \rho_f \, c_f \, \Delta V$ = $998.2 \times 1257 \times 0.3 = 3.76 \times 10^5$ N/m²). The basic period of pressure waves is about 0.20 s, corresponding to a virtual wave speed of 1550 m/s, where the classical theory would predict 0.247 s ($4L/c_f$ = 4 × 77.5 / 1257). Since Poisson coupling only slightly changes wave speeds [15], this virtual increase of the pressure wave speed is attributed to junction coupling [9].

A wave with a period of approximately 0.04 s is superimposed on the basic wave.

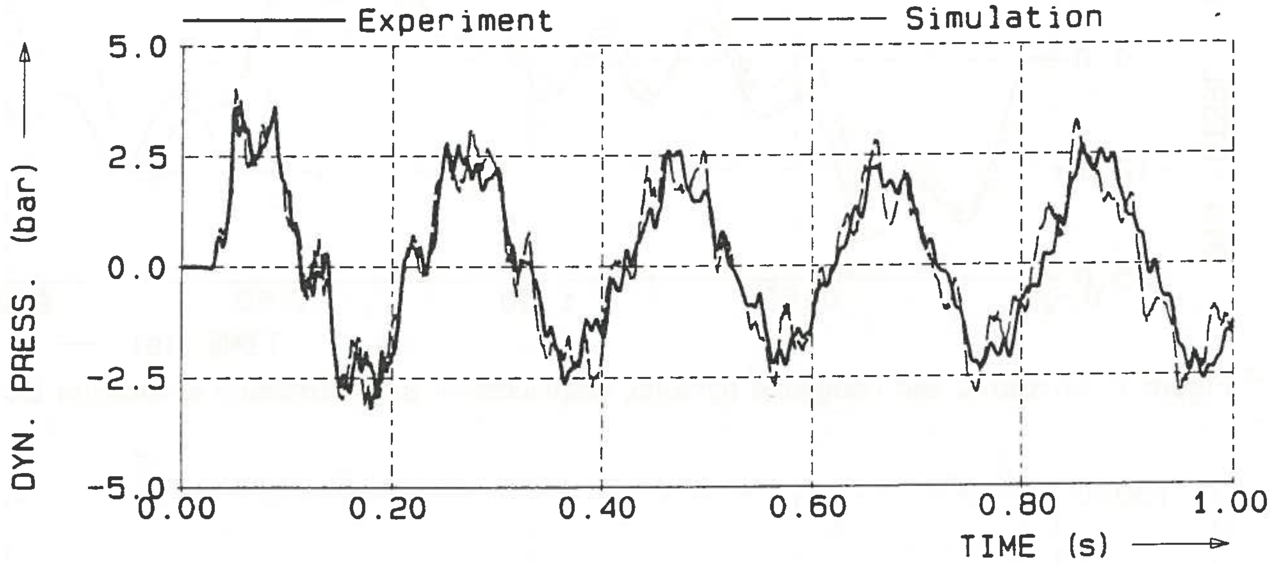

Figure 5. Measured and computed dynamic pressure 3 m upstream of location E.

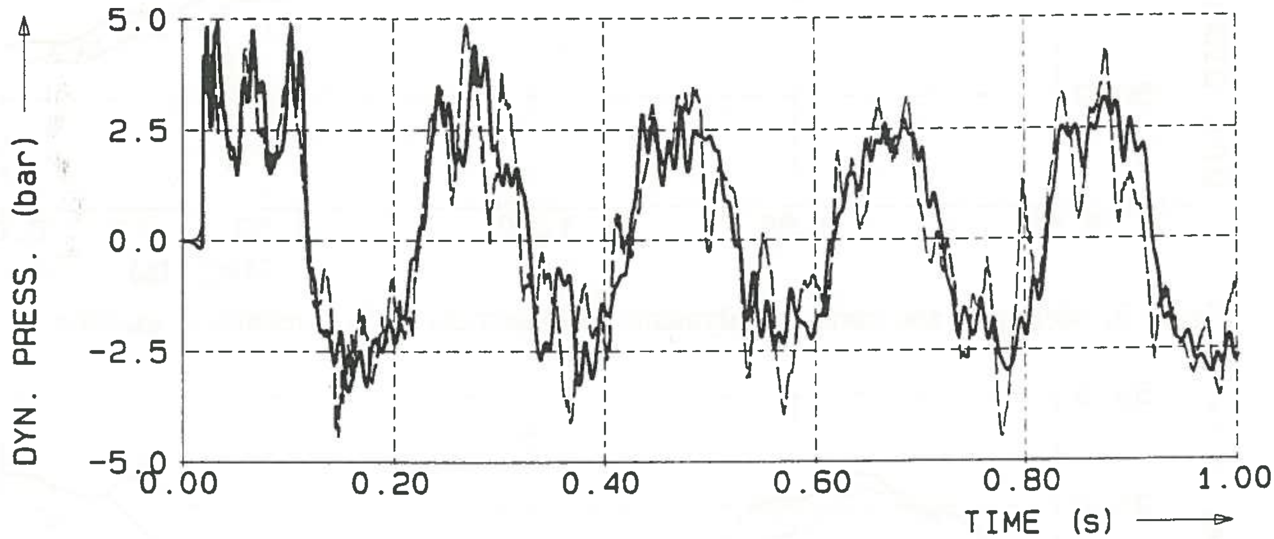

Figure 6. Measured and computed dynamic pressure at the shut off valve.

Displacements

In figures 7, 8 and 9 dynamic displacements are presented at elbows D and E.

Amplitudes are predicted quite well by the simulation, whereas the basic periods tend to be too small. This is attributed to ignoring, among others, the motion of "rigid" supports and the inertia of concentrated masses.

The three signals differ in basic period, since the ratio of effective mass and stiffness varies with direction and location. However, the period of 0.20 s, effected by pressure waves, is equal in all three cases.

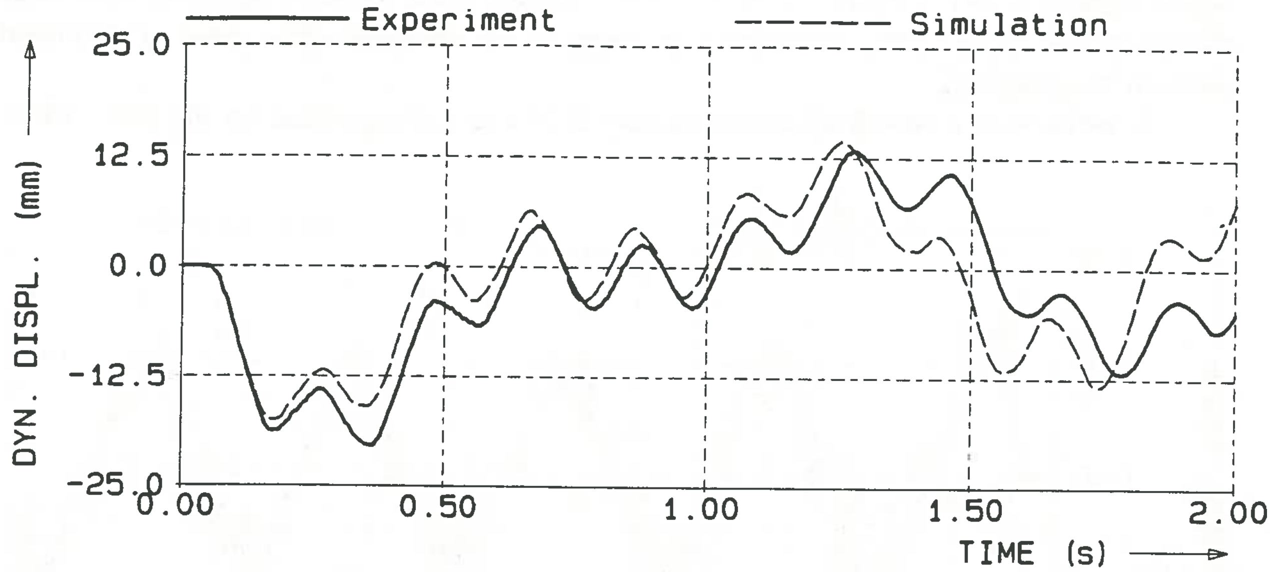

Figure 7. Measured and computed dynamic displacement in X_3-direction at location D.

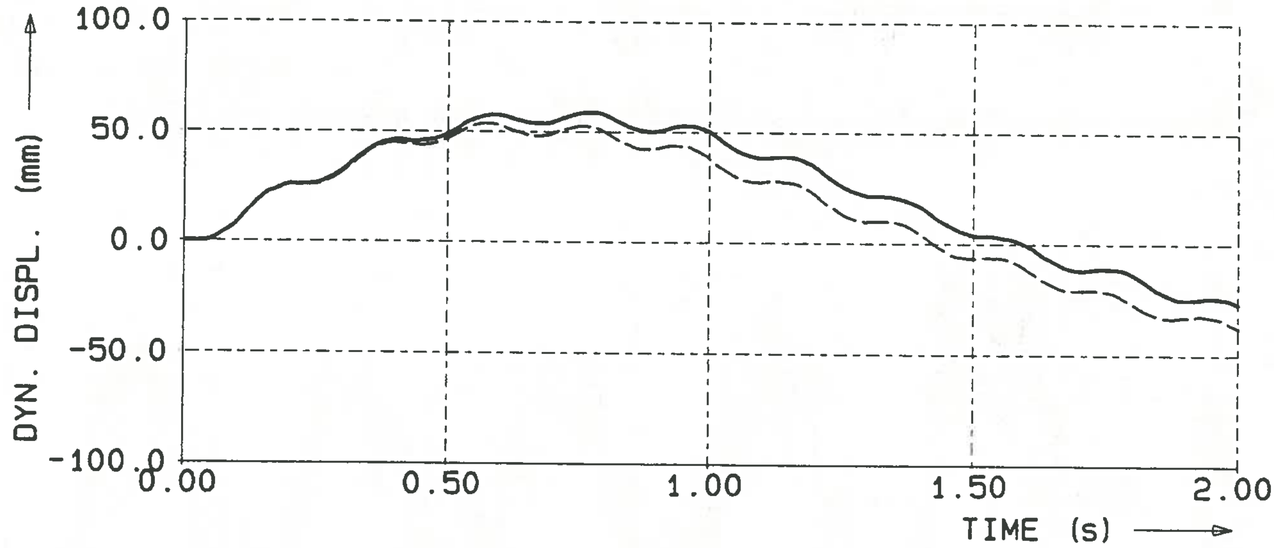

Figure 8. Measured and computed dynamic displacement in X_1-direction at location E.

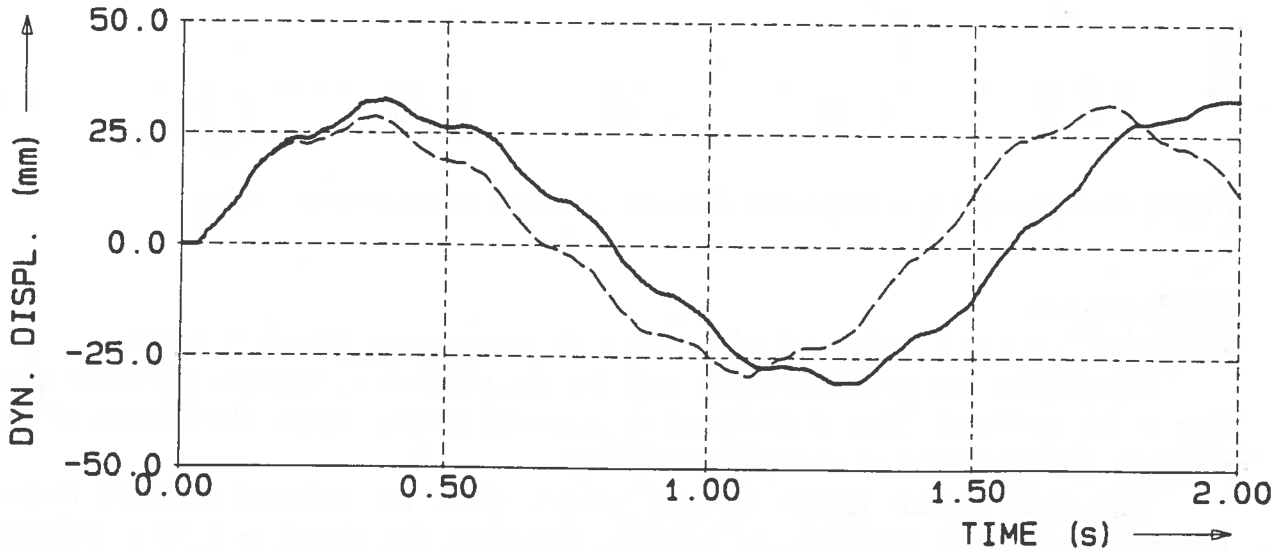

Figure 9. Measured and computed dynamic displacement in X_3-direction at location E.

Strains

Figures 10, 11, 12 and 13 show dynamic strains near the locations E and H.

In figure 10 an axial strain 1.5 m upstream of location E is presented. The overall tendency in amplitudes agrees, although the predicted frequencies are too high.

The strains shown in figures 11, 12 and 13 are measured 0.3 m upstream of the valve, whereas they are computed at the valve. The axial strain is shown in figure 11. Compared to the other strains, this strain is relatively small in magnitude and of less importance. The measured and computed amplitudes differ significantly, due to the fact that the axial motion of the valve (measured displacements in the order of 0.1 mm) is not simulated. The pressure wave period of 0.20 s can be recognized. The shear strain is given in figure 12. The agreement in amplitudes and frequencies is good, apart from some drift. The hoop strain is shown in figure 13. The agreement between the measured and computed values is excellent, which was expected because the hoop strain is dominated by the (well predicted) pressures.

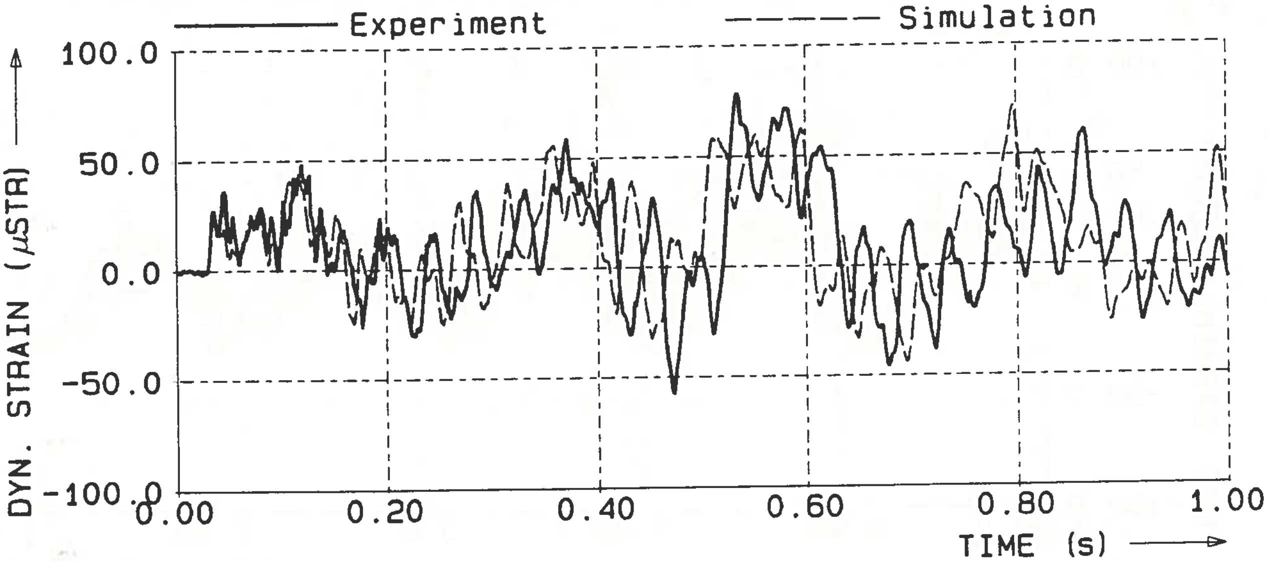

Figure 10. Measured and computed dynamic axial strain 1.5 m upstream of location E.

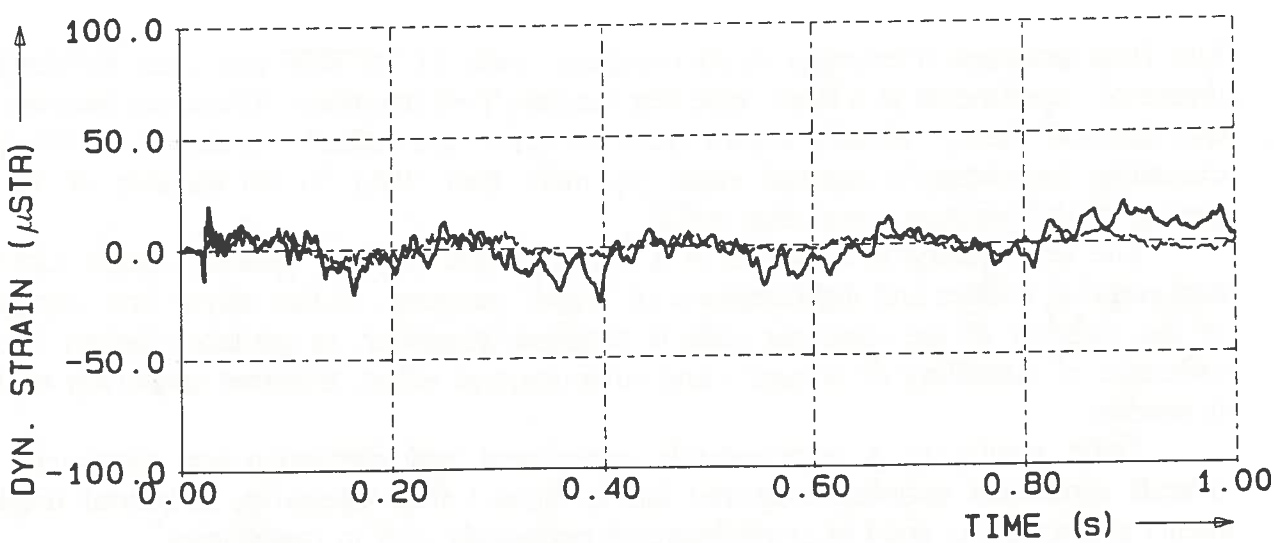

Figure 11. Measured and computed dynamic axial strain at the shut off valve.

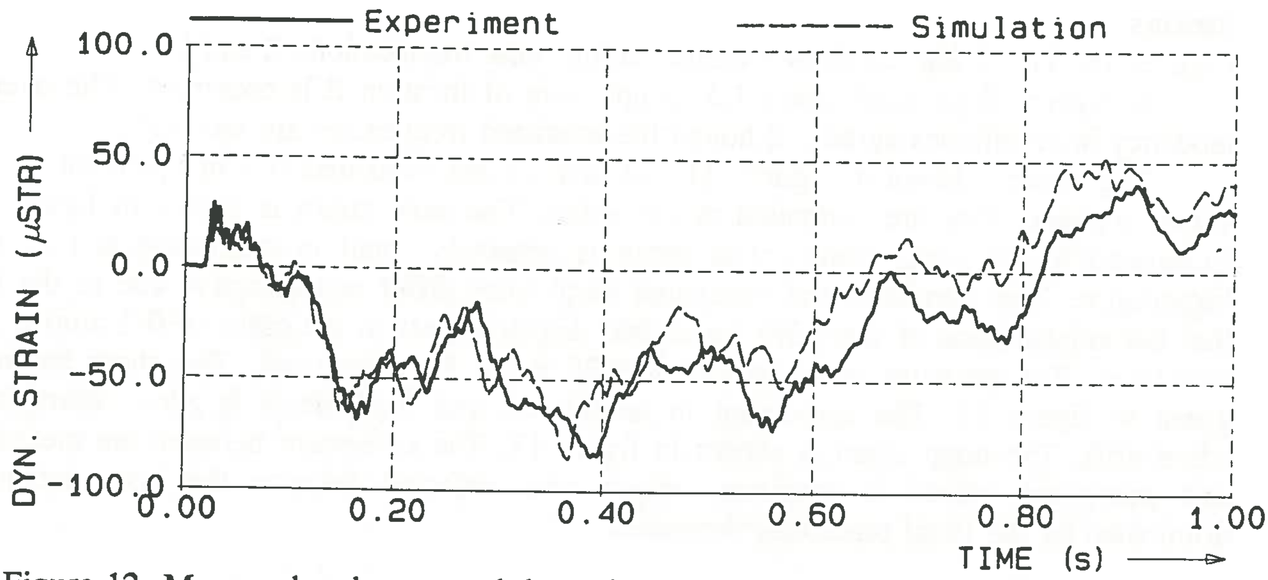

Figure 12. Measured and computed dynamic shear strain at the shut off valve.

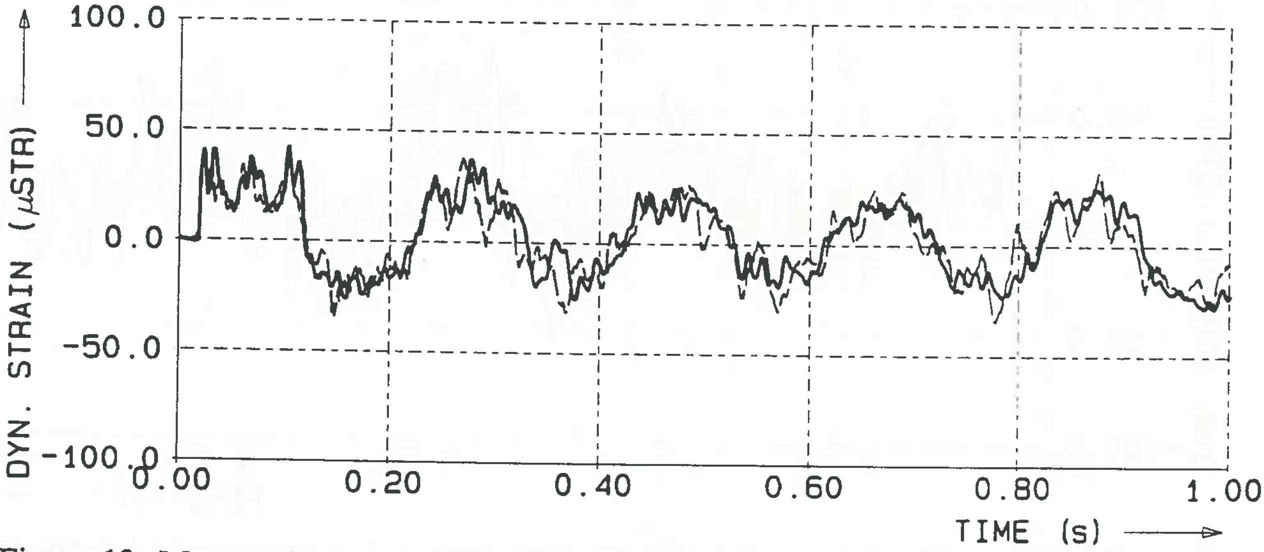

Figure 13. Measured and computed dynamic hoop strain at the shut off valve.

CONCLUSIONS

The fluid-structure interaction (FSI) computer code FLUSTRIN has been validated by means of experiments in a large scale test facility. Two important differences between FSI and classical theory, already known from literature, are actually measured: 1) Pressures exceeding Joukowsky's classical value (by more than 30%), 2) An increase of the frequency of the pressure waves due to FSI.

The test facility is modelled in a rather simple way by ignoring details like e.g. concentrated masses and displacements of "rigid" supports. In this way a first impression of the usability of the computer code is obtained. However, to get more insight into the influence of modelling on accuracy and computational effort, a further sensitivity analysis is needed.

Some results of a representative experiment and simulation are compared. The overall agreement between measured and computed fluid pressures, structural displacements and strains is good in amplitudes and reasonably well in frequencies.

It is demonstrated that the classical theory is inadequate to describe the transient behaviour of the flexible pipeline system considered here. In this case and for a number of practical pipeline systems the more sophisticated FSI approach is necessary. To judge in which cases FSI is of importance, guidelines as proposed in [10] are needed. The validated computer code FLUSTRIN offers possibilities to develop these guidelines.

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