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Fluid-structure interaction in straight pipelines: Friction coupling mechanisms

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ABSTRACT

The present paper approaches fluid-structure interaction by means of a 4-equation model. Experimental data collected from a straight copper pipe-rig lying directly on the lab floor is used for the model validation in terms of wave shape, timing and damping. The main focus lies on the friction coupling modelling considering skin and dry friction. For skin friction three approaches are analysed: quasi-steady, Brunone's and Trikha's unsteady friction. For dry friction Coulomb's model is added in the beam momentum conservation equation. Results present a good fitting between experimental and numerical data, showing the dissipative effect of dry friction phenomenon which complement that of skin friction, specially in the short term simulation.

1. Introduction

Fluid-structure interaction (FSI) in pressurized hydraulic transients analyses is frequently approached by considering the first two pipe vibration modes (*i.e.*, pressure wave propagation in the fluid and axial stress wave propagation in the pipe-wall). For the description of pressure waves in pipe systems, one-mode or two-mode solutions are sufficient [26]. Two-mode models can be implemented either by using MOC-FEM procedure (*i.e.*, the method of characteristics for the fluid and finite element method for the structure) [40] or MOC procedure (*i.e.*, the method of characteristics for both the fluid and the structure) [41]. Lavooij and Tijsseling [20] applied the two approaches to solve the four basic conservation equations in the time domain, concluding that for straight pipe problems the MOC procedure is more accurate. Thus, a 4-equation model represents a suited tool to describe the ideal reservoir-pipe-valve system in its basic FSI configurations, namely either considering an anchored or non-anchored downstream valve.

Several authors combined FSI with other wave dissipating phenomena, such as: FSI and pipe-wall viscoelasticity [39,37,25]; FSI and cavitation [30,29,26]; and the most complete including FSI, column separation and unsteady friction (UF) in a viscoelastic pipe [19]. However, the effects of unsteady friction and pipe-wall viscoelasticity are hard to distinguish [9] and, to the knowledge of the authors, unsteady friction effect has never been separately assessed in a two-mode FSI model. Due to FSI, the pipe-wall vibrates axially at a different rate than the fluid, hence, the relative velocity between both (V_r) must be considered for skin shear stress assessment. The higher the Mach number (V_r/a_f) is, the greater the wall shear stress effects are [16]. Therefore, unsteady friction effects may be increased when fluid-structure interaction is important.

Besides, in the implementation of a 4-equation model a major question may arise: Is there movement in the pipe supports? Anchorages of pipelines aim to avoid the pipe-wall movement essentially by means of dry friction [11]. However, from Newton principles, when a system is loaded, null deformation/displacement by means of only resistance is not possible. Pipe supports are never entirely stiff or entirely inert when loaded by impacts [27]. Thus, movement occurs. Dry friction is proportional to the normal force, hence, for a high normal force, important energy might be dissipated from the structure to its supports/surroundings. Furthermore, in this context, it is crucial to define with good criteria the stick-slip transitions.

Tijsseling and Vardy [28] included Coulomb's dry friction in a 4-equation model with the goal to describe the behaviour of pipe racks, proposing a quantitative guideline equation aiming at assessing when dry friction forces may be relevant during hydraulic transients. In the present paper, dry friction is approached differently not at a single point but distributed all throughout the

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Notation

A_f	fluid cross-sectional area (m^2)	N	normal force (N)
a_f	pressure wave speed ($m\ s^{-1}$)	n_k	exponential sum coefficients (–)
A_s	pipe-wall cross-sectional area (m^2)	p	fluid pressure (Pa)
a_s	axial stress wave speed ($m\ s^{-1}$)	Re	Reynolds number (–)
$C^\star$	Vardy decay coefficient (–)	r	inner radius of the pipe-wall (m)
D	pipe inner diameter (m)	t	time (s)
E	pipe-wall Young's modulus (Pa)	t_v	valve closure time (s)
e	pipe-wall thickness (m)	U	pipe-wall velocity ($m\ s^{-1}$)
f	Darcy skin friction coefficient (–)	V	fluid mean velocity ($m\ s^{-1}$)
f_s	steady Darcy coefficient (–)	V_r	relative velocity ($m\ s^{-1}$)
f_u	unsteady Darcy coefficient (–)	W	Zielke weighting function (–)
F	force acting in the system (N)	x	distance along the pipe axis (m)
F_{df}	dry friction force (N)	Y_k	Trikha function (–)
g	gravity acceleration ($m\ s^{-2}$)	ν	Poisson's ratio (–)
H	hydraulic head (m)	μ	Coulomb dry friction coefficient (–)
H_{Jk}	Joukowsky hydraulic head (m)	μ_s	static Coulomb coefficient (–)
h_f	skin friction losses (m)	μ_k	kinetic Coulomb coefficient (–)
h_{f_s}	steady skin friction losses (m)	ρ_s	pipe density ($kg\ m^{-3}$)
h_{f_u}	unsteady skin friction losses (m)	ρ_f	fluid density ($kg\ m^{-3}$)
K	bulk modulus of compressibility (Pa)	σ	pipe axial stress (Pa)
k	Brunone coefficient (–)	σ_{RK}	Rankine pipe axial stress (Pa)
L	pipe length (m)	τ	valve closure degree (–)
M_v	valve mass (kg)	τ_t	dimensionless time-step (–)
m_k	exponential sum coefficients (–)	λ	eigenvector ($m\ s^{-1}$)

pipeline. For this purpose, a new right-hand-side term in the momentum equation of the pipe-wall axial movement was incorporated.

This research aims at assessing firstly the effect of different skin friction models during hydraulic transients in a FSI 4-equation (two-mode) solver. For this purpose, three skin friction models are assessed: (i) quasi-steady friction; (ii) Brunone's unsteady friction formulation, which is based on instantaneous local and convective accelerations; and (iii) Trikha's unsteady friction model, which is based on weights of past velocity changes. Secondly, dry friction is implemented, nesting its computation into the friction coupling mechanism, and its dissipation effect over the transient wave is assessed.

Experimental tests were carried out in a straight copper pipe-rig for different initial conditions and structural scenarios: (a) anchored downstream pipe-end, and (b) non-anchored downstream pipe-end. These tests allowed to corroborate and validate the modelling assumptions.

The aim of the paper is the assessment of different friction dissipation assumptions in a FSI two-mode model. A 4-equation solver is implemented including the three basic coupling mechanisms: Poisson, junction and friction coupling; and the last one nests the skin friction models (*i.e.* quasi-steady, Brunone's and Trikha's) and the dry friction model (*i.e.* Coulomb's friction). The innovation of this research is the incorporation of dry friction computation in the fundamental equations of the two-mode (four-equation) waterhammer model. This implies a modification of the pipe-wall momentum equation in the axial direction. The effect of dry friction is compared with skin friction and results are assessed by means of experimental data in a straight copper pipe rig.

2. Experimental data collection

A straight copper pipe rig was assembled at the Laboratory of Hydraulics and Environment of Instituto Superior Técnico (LHE/IST). The system is composed of a 15.49 m long pipe, with an inner diameter $D = 0.020$ m and pipe-wall thickness $e = 0.0010$ m. Young's modulus of elasticity and Poisson's ratio of the copper material were experimentally determined by measuring stress-strain states over a pipe sample for the experimental range of pressures. The pipe segment, with closed ends, was pressurized and strains measured using strain gauges disposed in the circumferential and axial directions. By means of stress-strain relations the Young's modulus and the Poison's ratio were determined. The obtained experimental values were the Young's modulus of elasticity $E = 105$ GPa and the Poisson's ratio $\nu = 0.33$; these values are in agreement with theoretical values from literature. At the upstream end, there is a storage tank followed by a pump and an air vessel, and at the downstream end, there is a ball valve pneumatically operated that allows the generation of fast transient events, with an effective closing time of $t_v = 0.003$ s. The ball valve together with the actuator mechanisms and the supporting system have a mass of $M_v = 6$ kg and it is anchored by glue-in-bolts on the floor of the laboratory. Downstream the valve there is a hose conveying the water to the water-tank, thus closing the pipe system circuit.

Three pressure transducers (WIKA S-10) were installed at the upstream, midstream and downstream positions of the pipe (PT1, PT2 and PT3). Strain gauges (TML FLA-2-11) disposed in the axial (SG1 and SG3) and circumferential (SG2 and SG4) directions were glued at the midstream and the downstream end of the pipe. The initial discharge was measured for steady state conditions by a rotameter located downstream of the valve. The sampling frequency was set to 1200 Hz after preliminary tests in order to measure with accuracy the FSI response of the pipe system during the waterhammer events. The wave celerity in the fluid was estimated from pressure measurements obtaining a value of $a_f = 1239$ m/s [14]. The axial stress wave celerity in the pipe-wall was theoretically determined $a_s = \sqrt{\frac{E}{\rho_s}} = 3848.4$ m/s. Fig. 1 shows an illustration of the experimental set-up, with the location of the pressure transducers (PT) and strain gauges (SG).

Two supporting configurations have been analysed: (a) the conduit anchored against longitudinal movement at both downstream and upstream ends; and (b) the conduit only anchored against lon-