

Case studies of fluid transients in subcooled pipe flow

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CASE STUDIES OF FLUID TRANSIENTS IN SUBCOOLED PIPE FLOW

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Key words: subcooled pipe flow, water hammer, column separation, liquid plug, air pocket, fluid-structure interaction

Abstract: Transient regimes may cause excessive water hammer, and possible column separation, slug flow, plug flow and fluid-structure interaction (FSI) in the system. Fluid transients may severely disturb operation of the piping system and damage the system components. This paper presents authors' case studies of typical transient regimes in industrial and laboratory piping systems. The first case study deals with the sudden load rejection of a Francis turbine in Plužna HPP (Slovenia). Then the results from pressure vessel blowdown experiments at UMSICHT's (Germany) pilot plant pipework are shown. Pump start with an air pocket in the system has been investigated by WL | Delft Hydraulics (The Netherlands). The last case study concerns impact tests in a structurally unrestrained pipe at the University of Dundee (UK). The underlying theory for these cases is briefly introduced and the measurements are compared with computed results.

1 Introduction

Feasibility and design studies of new and ageing piping systems carrying subcooled¹ fluid flow should include fluid transient analysis in order to ensure safe and economic operation of the system. The main objective of this paper is to present authors' case studies of typical transient regimes in industrial and laboratory piping systems. Transient regimes may cause excessive water hammer, and possible column separation, slug flow, plug flow and fluid-structure interaction in the system. Fluid transients may severely disturb the operation of the piping system and damage the system components, since the forces induced to the pipe supports exceed the design criteria.

In hydroelectric power plants, fluid transients are normally initiated by a turbine load acceptance, load reduction or sudden load rejection, turbine runaway, shut-off valve closure, and a combined operation of the turbine and valve. In water supply pumping systems, transients may be induced by a pump start-up, pump run-down, and opening and closing of valves. A case of the sudden load rejection of a horizontal-shaft Francis turbine in Plužna hydro power plant, Slovenia is presented in this paper. Water hammer during load rejection was controlled by appropriate two-speed wicket gates closure and increased turbine unit inertia. Water hammer is described by a set of hyperbolic partial differential equations, the continuity equation and the equation of motion. The method of characteristics is used for solving the water hammer equations. Cavitating flow occurs when the pressure in the pipe drops to the liquid vapour pressure.

¹ The hydrodynamic effects (water hammer, column separation) are driven by inertia and not influenced by heat transfer.

In chemical/petrochemical industry, fluid transients may occur due to formation of gas pockets in a liquid flow or due to accelerating liquid plugs in gas / blowdown pipes. Liquid plugs may be formed e.g. in discharge pipes (due to the condensed humidity), in vapour and gas (due to heat losses), in pipelines after cleaning with vapour or liquids, or due to valve leakage. Even the simplest pipe configurations may experience severe accidents or plant down-time, if the presence of gas pockets is not anticipated in the design or in operation procedures. Computational and measured results from the pressure vessel blowdown case performed at Fraunhofer UMSICHT, Germany, are presented. A simple theoretical model of gas-induced liquid flow in pipes with gas or vapour is briefly introduced.

The danger of unanticipated air pockets is demonstrated in a case study by WL | Delft Hydraulics, The Netherlands.

The influence of axial pipe vibration on water hammer, and vice versa, has been extensively studied by the last two authors. Herein, the FSI four-equation model is presented and new exact solutions developed at TU Eindhoven, The Netherlands, are compared with unpublished data from impact tests in a laboratory apparatus in the University of Dundee, UK.

2 Theoretical Models

This section deals with theoretical models for the description of pressure wave phenomena in pipelines including water hammer (liquid pipe flow), multiphase transient flow (gas-induced liquid flow in pipes with gas or vapour) and axial pipe vibration.

2.1 Modelling of Water Hammer

Water hammer is the transmission of pressure waves along the pipeline resulting from a change in liquid flow velocity. Unsteady liquid flow in closed conduits is described by two one-dimensional equations; the continuity equation and the equation of motion (Wylie and Streeter, 1993):

$$\frac{\partial H}{\partial t} + v \frac{\partial H}{\partial x} - v \sin \theta_p + \frac{a^2}{g} \frac{\partial v}{\partial x} = 0 \quad (1)$$

and

$$g \frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} + v \frac{\partial v}{\partial x} + \frac{\lambda v |v|}{2D} = 0 \quad (2)$$

in which H = (piezometric) head, t = time, v = flow velocity, x = distance, θ_p = pipe slope, a = wave speed, g = gravitational acceleration, λ = Darcy-Weisbach friction factor, and D = pipe diameter. For most engineering applications, the convective terms in Eqs. (1) and (2) are small and neglected. Usually, the discharge $Q = vA$ replaces the flow velocity v , where A = cross-sectional area. Equations (1) and (2) are a set of quasi-linear hyperbolic differential equations. A transformation by the method of characteristics (MOC) gives the water hammer compatibility equations which are valid along the characteristic curves. The water hammer compatibility equations, written in a finite-difference form, are (small terms are neglected) (Wylie and Streeter, 1993):

- along the C^+ characteristic line ($\Delta x/\Delta t = a$):

$$H_{j,t} - H_{j-1,t-\Delta t} + \frac{a}{gA} \left[(Q_u)_{j,t} - (Q_d)_{j-1,t-\Delta t} \right] + \frac{\lambda \Delta x}{2gDA^2} (Q_u)_{j,t} \left| (Q_d)_{j-1,t-\Delta t} \right| = 0 \quad (3)$$

- along the C^- characteristic line ($\Delta x/\Delta t = -a$):

$$H_{j,t} - H_{j+1,t-\Delta t} - \frac{a}{gA} \left[(Q_d)_{j,t} - (Q_u)_{j+1,t-\Delta t} \right] - \frac{\lambda \Delta x}{2gDA^2} (Q_d)_{j,t} \left| (Q_u)_{j+1,t-\Delta t} \right| = 0 \quad (4)$$

in which j = computational section index, Q_u = discharge at the upstream side of the computational section, Q_d = discharge at the downstream side of the computational section, Δx = reach length, and Δt = time step. A constant value of the Darcy-Weisbach friction factor λ is used in the equations (3) and (4). This assumption may be corrected for the case of rapid transients by introducing an unsteady friction term in the above equations (Anderson et al., 1991; Bergant et al., 2001). Transient cavitating flow at a section occurs when the pressure drops to the liquid vapour pressure. The standard water hammer solution, where $Q_d = Q_u$, is no longer valid and one of the column separation models should be used (Wylie and Streeter, 1993; Bergant and Simpson, 1999). At a boundary (reservoir, valve, pump-turbine, ...), the boundary condition replaces one of the water hammer compatibility equations.

2.1.1 Francis Turbine Boundary Condition

The dynamic behaviour of a governed Francis turbine is described by the turbine, governor and pipeline equations (Wylie and Streeter, 1993). The governor equations are omitted in the analysis for the case of Francis turbine sudden load rejection in which the unit speed change is controlled by the net torque only. The Francis turbine in-line boundary condition for the case of sudden load rejection, which is incorporated into the staggered grid of the method of characteristics, is described by the following equations:

- water hammer compatibility equations (3) and (4)
- head balance equation:

$$H_u - H_r \left(\left(\frac{n}{n_r} \right)^2 + \left(\frac{Q}{Q_r} \right)^2 \right) W_H(y(t), x) - H_d = 0 \quad (5)$$

- dynamic equation of the turbine unit rotating masses:

$$\left(\left(\frac{n}{n_r} \right)^2 + \left(\frac{Q}{Q_r} \right)^2 \right) W_T(y(t), x) + \left(\frac{T}{T_r} \right)_{t-2\Delta t} - I \frac{\pi n_r}{30 T_r} \frac{1}{\Delta t} \left(\left(\frac{n}{n_r} \right) - \left(\frac{n}{n_r} \right)_{t-2\Delta t} \right) = 0 \quad (6)$$

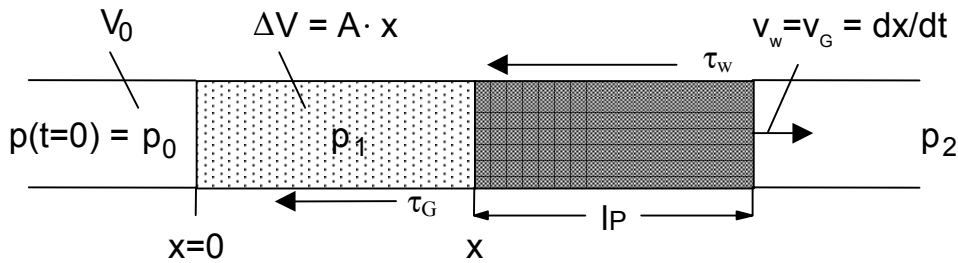
in which H_u = head at the upstream side of the turbine, H_r = rated turbine head, n = turbine rotational speed, subscript r = rated conditions, $W_H(y(t), x)$ = dimensionless turbine head characteristic, $y(t)$ = dimensionless wicket gates position, $x = \arctan((Q/Q_r)/(n/n_r))$ = angular position of the turbine characteristic curve, H_d = head at the downstream side of the turbine,

$W_T(y(t), x)$ = dimensionless turbine torque characteristic, T = torque, and I = polar moment of inertia of rotating parts.

2.2 Modelling of Gas Induced Liquid Flow in Pipes with Gas or Vapour

When, due to the response of the safety valve on a pressure vessel, a gas blow down into the relief pipe has occurred, the pressure loss in the partly filled pipeline is expected to be higher than in the empty pipe. In case of blowdown of gases or vapour, the accumulation of liquid in discharge pipes results into accelerating liquid plugs in the pipe system. A simplified theoretical model describing the decreasing pressure time history in a blow off vessel has been developed according to Figure 1. The air-filled pressure vessel will be emptied in the partly water-filled pipeline. Since there is no exact knowledge of the flow regime available, the application of two-phase pressure loss models has been waived.

Figure 1. Blow off vessel model.



The following conditions are assumed in the model:

- During the process the plug fills the whole pipe cross section and keeps constant length L_P .
- The change of state in the air-filled space is adiabatic with $p_1 V^{\gamma} = p_0 V_0^{\gamma} = \text{constant}$.
- The opposing pressure p_2 during the process remains constant (no friction of the displaced gas).
- For the pipe friction of the plug, Darcy's resistance law is assumed. The skin-friction coefficient for gas in the investigated pressure, temperature and velocity range is constant.
- Liquid plug flow: rigid-column theory.

The plug flow is calculated from the resulting forces on the liquid plug as shown in Fig. 1:

$$F = (p_1 - p_2) \cdot A - (\tau_w \cdot \pi \cdot D \cdot L_P) \quad (7)$$

The flow rules for the passed distance and time of the plug in the pipeline can be derived as (Dudlik, 1999):

$$\frac{dv}{dx} = \frac{(p_0 \cdot \alpha - p_2)}{v \cdot \rho_w \cdot L_P} - \frac{v}{2 \cdot D} \left[\lambda_w + (\lambda_G \cdot x) \cdot \left(\frac{V_0}{V_0 + A \cdot x} \cdot \frac{\rho_{G0}}{\rho_w \cdot L_P} \right) \right] \quad (8)$$

$$\frac{dv}{dt} = \frac{(p_0 \cdot \alpha - p_2)}{v \cdot \rho_w \cdot L_p} - \frac{v^2}{2 \cdot D} \left[\lambda_w + (\lambda_G \cdot x) \cdot \left(\frac{V_0}{V_0 + A \cdot x} \cdot \frac{\rho_{G0}}{\rho_w \cdot L_p} \right) \right] \quad (9)$$

with the ratio $\alpha \equiv \frac{p_1}{p_0} = \left(1 + \frac{A \cdot x}{V_0} \right)^{-\chi}$

in which F = force, p = pressure, τ = shear stress, L = length, ρ = density, V = volume, λ_w and λ_G = skin friction coefficient of water and gas, respectively, and χ = adiabatic exponent.

2.3 Modelling of Water Hammer and Axial Pipe Vibration

If in a pipe system one or more pipes have the possibility to vibrate in axial direction, fluid-structure interaction (FSI) may be important in the prediction of pressures, stresses and anchor forces. This will particularly be the case for flexibly supported pipes subjected to pressure waves with steep wave fronts. The basic model describing the coupled propagation of pressure waves in the fluid and axial-stress waves in the pipe wall consists of the four equations:

$$\frac{\partial H}{\partial t} + \frac{a^2}{g} \frac{\partial v}{\partial x} = \frac{2\nu}{E} \frac{a^2}{g} \frac{\partial \sigma}{\partial t} \quad (10)$$

$$g \frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} = 0 \quad (11)$$

$$\frac{\partial \sigma}{\partial t} - \rho_s a_s^2 \frac{\partial \dot{u}}{\partial x} = \nu \rho g \frac{D}{2e} \frac{\partial H}{\partial t} \quad (12)$$

$$\frac{\partial \sigma}{\partial x} - \rho_s \frac{\partial \dot{u}}{\partial t} = 0 \quad (13)$$

in which σ = axial pipe stress, \dot{u} = axial pipe velocity, ν = Poisson ratio, E = Young modulus, e = wall thickness, and the subscript s denotes structure (i.e. pipe wall). The equations (10) and (11) are the water hammer equations (1) and (2) with an additional term due to FSI, but without terms for convection and friction. The equations (12) and (13) are the stress wave equations for the pipe wall with an additional term due to FSI. If the Poisson ratio ν is taken equal to zero, the wave equations for pressure (10-11) and stress (12-13) are decoupled, in which case the speeds of propagation are:

$$a = \sqrt{\frac{K}{\rho}} \sqrt{\frac{1}{1 + \frac{DK}{eE}}} \quad \text{and} \quad a_s = \sqrt{\frac{E}{\rho_s}} \quad (14)$$

where K = bulk modulus.

The strong FSI mechanisms that may occur at unrestrained pipe ends are modelled through coupled boundary conditions. The four-equation model (10-13) can be solved exactly (Tijsseling, 2002). More information on the subject of FSI can be found in the review paper by Wiggert and Tijsseling (2001) and on the FSI web site.

3 Case Study: Water Hammer in Plužna HPP, Slovenia

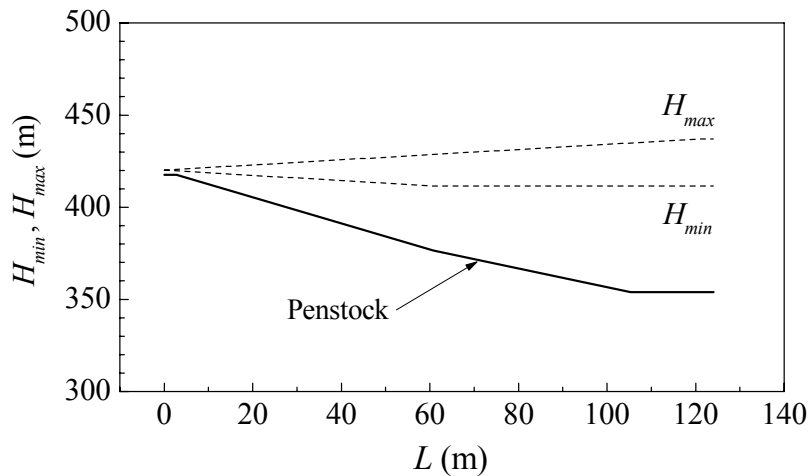
Plužna hydro powerplant (Bergant and Sijamhodžić, 1997) comprises an upstream reservoir, penstock of diameter $D = 0.85$ m and length $L = 121.2$ m (see Fig. 2), turbine inlet valve of diameter $D = 0.8$ m, horizontal-shaft 1.87 MW Francis turbine of rated head $H_r = 63$ m and discharge $Q_r = 3.3 \text{ m}^3\text{s}^{-1}$, and a downstream reservoir. The water level z_u in the upstream reservoir is in the range of 421.5 m to 420.0 m; the level z_d in the downstream reservoir is 352.1 m. The rated speed of the turbine is $n_r = 750.0 \text{ min}^{-1}$ and the polar moment of inertia of the unit rotating parts $I = 8.86 \times 10^3 \text{ kg.m}^2$ (after adding a flywheel of $7.70 \times 10^3 \text{ kg.m}^2$).

Various operating regimes were examined during commissioning tests, including turbine start-up, load acceptance, load reduction and sudden load rejection. The resulting water hammer was controlled by appropriate adjustment of wicket gates closing and opening manoeuvres, and increased unit inertia. The MOC numerical model was used for computational analysis.

3.1 Sudden Load Rejection of the Turbine

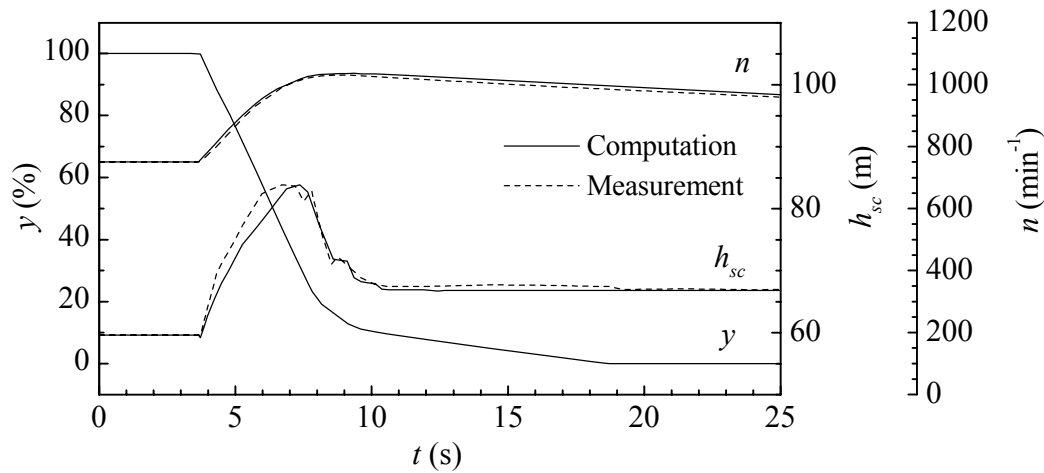
Sudden load rejection of the turbine is the most severe transient regime occurring under normal operating conditions (Chaudhry 1987). The turbine was disconnected from the electrical grid followed by the full closure of the wicket gates.

Figure 2. Computed envelopes of maximum and minimum heads along the penstock after sudden load rejection of the turbine (H_{max} = maximum head, H_{min} = minimum head, L = penstock length).



Computed envelopes of the maximum and minimum piezometric heads along the penstock profile are shown in Figure 2. This diagram is important for design engineers to construct a safe and economic pipeline system. The envelope of the minimum head (H_{min}) indicates the danger of liquid column separation when the pressure drops below the penstock profile. In this case, the computed minimum head is well above the penstock profile.

Figure 3. Comparison of computational and experimental results after sudden load rejection of the turbine (y = dimensionless wicket gates position, h_{sc} = scroll-case pressure head, n = turbine rotational speed, t = time).



Maximum pressure in the scroll case and maximum turbine rotational speed are two important parameters in turbine design. The pressure head in the scroll case at the turbine inlet h_{sc} (datum level $z = 353.95$ m) and the turbine rotational speed n are depicted in Figure 3. There is a good match between the results of computation and measurement. The computed maximum head $(h_{sc})_{max,c} = 83.9$ m equals the measured maximum head. The computed maximum turbine rotational speed $n_{max,c} = 1036 \text{ min}^{-1}$ is marginally higher than the measured speed $n_{max,m} = 1030 \text{ min}^{-1}$.

4 Case Study: Liquid Plugs in Fraunhofer UMSICHT PPP, Germany

For the investigation of transient flow in the case of blow-off pressure vessels, a pilot plant pipework (PPP) of DN 100 has been designed and constructed at Fraunhofer UMSICHT. The experimental set-up is shown in Figure 4. The pressurised vessel B1 (5-30 bar) may expand into a pipe system (114 mm diameter) that is partly filled with water (20-65 litre) at the low points 1 and 2. The pressure time history (P24-P05), the induced forces at the fixed points 1 and 2 as well as the air fraction in the pipeline cross-section were measured in a frequency range between 1 and 10 kHz. The flow regime was qualitatively acquired by wire-mesh sensors positioned downstream the closing valve. More information on PPP can be found on the web site of Fraunhofer UMSICHT (see References)

4.1 Pressure Vessel Blowdown Runs

Figure 5 shows the calculated pressure characteristics of a blowdown vessel compared with the measured pressure characteristics. As may be seen from Figure 5 the calculated values are in good agreement with the measured values. At higher initial pressures the calculated pressure decrease is faster than the measured one. For a vessel that has to be protected this means a non-conservative calculation. For the design of the pipe supports this characteristic describes a conservative construction, because a fast pressure decrease leads to high liquid plug velocity in the pipe. The reasons for the deviation of calculation and measurement are as follows:

- The friction acting on the displaced gas in the pipeline has not been considered.
- The liquid plug flanked by two-phase flow has a higher pressure loss than the one-phase flow.

Figure 4. Experimental set-up for blowdown experiments at PPP.

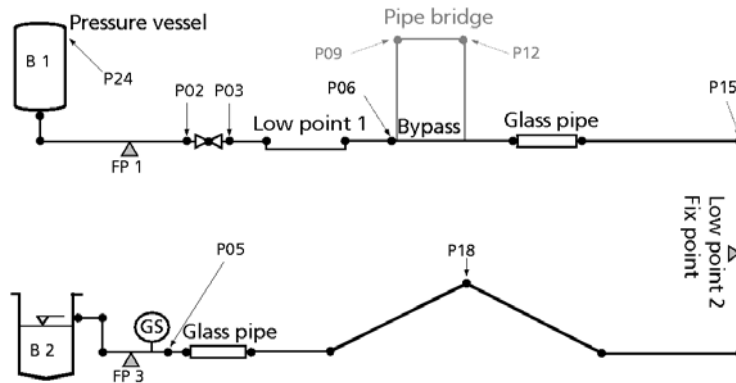
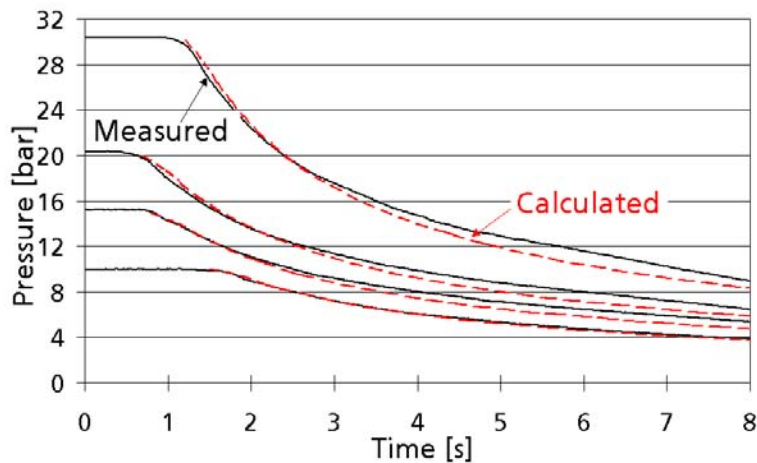


Figure 5. Pressure Vessel Blowdown Runs.



5 Case Study: Filling Line

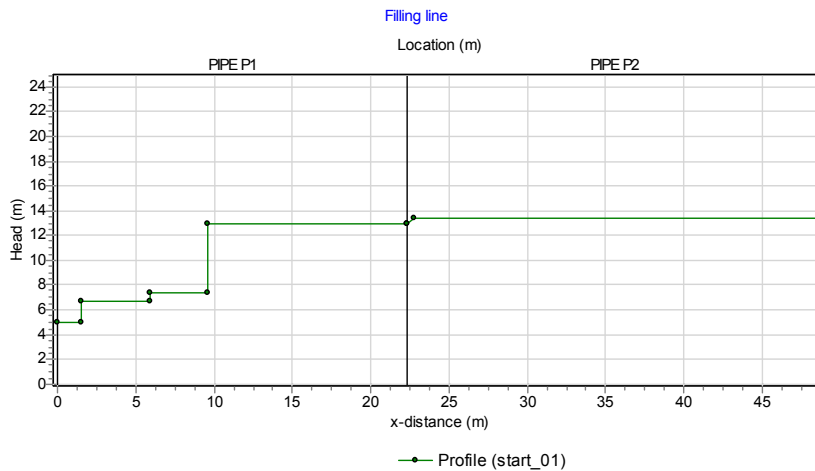
WL | Delft Hydraulics has investigated an incident in a filling line of a filter installation. This case study illustrates that severe transients may occur in any pipe system, irrespective of the length or complexity of the system. The filling line is operated in a batch process. Hence start and stop procedures are triggered regularly: up to 5 times per hour.

5.1 System Description

The DN200 filling line is only 60 m long with an elevation of approximately 10 metres (see Figure 6). The filter back pressure is 2 bar. The design flow rate is 280 m³/h at 34 m head. The shut-off head of the pump is 45 m. The pipe material is steel, wall thickness is 3.8 mm, resulting in a wave speed of 1200 m/s and characteristic $2L/a$ time of 0.1 s.

The shut-off valve is located at the downstream side of the filling line just upstream of the filter installation. The normal start procedure is to start the pump against the closed shut-off valve (ball type). When the pressure at the valve has increased above 4 bar, the valve opens in 6 s. Such a start procedure is generally recommended from a transient control point of view.

Figure 6. Filling line pipe profile; pump is located at 0 m, shut-off valve at 48 m horizontal distance.



5.2 Case Study

The following event was reported by the system operators. Immediately after every pump start a tremendous bang was heard. The pipe vibrated severely during several seconds.

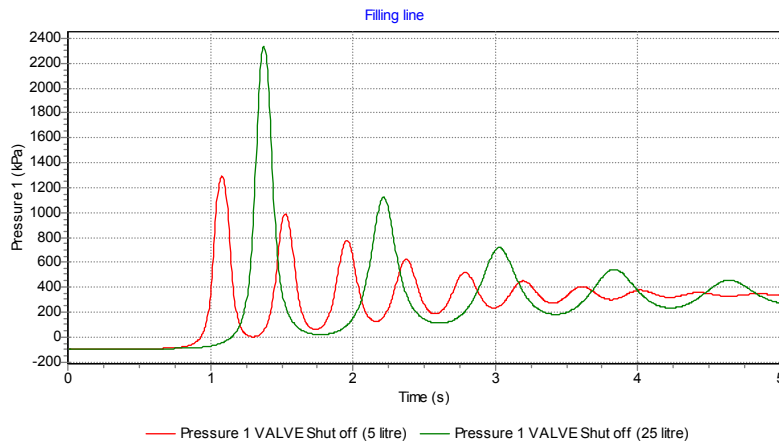
A simulation confirmed that the normal start procedure should not result in extreme system pressures.

However, pump start with an air pocket upstream of the ball valve can create pressures far exceeding the design pressure (see Figure 7). An air pocket can easily develop at this location if the pump is idle. A small leak at a joint or in the check valve causes the pressure to drop and dissolved air to be released and collected upstream of the shut-off valve.

During this start scenario the water column accelerates and compresses the air pocket. The non-linear elastic behaviour of the air pocket causes the pressure peak. The magnitude of the pressure peak depends on the size of the initial air pocket. Another consequence of this scenario is that the water rapidly decelerates (20 m/s^2) to negative velocities. This deceleration cannot be followed by the swing-type check valve, which closes in return flow; this effect has not been included in the model results shown in Figure 7.

The solution to the problem was to fill the line before starting the pump. Filling could be easily carried out under atmospheric conditions by opening the shut-off valve temporarily such that filter water returns to the filling line. The valve is then closed again before the normal start procedure is executed.

Figure 7. Computed pressure evolution upstream of the shut-off valve with initial air pockets of 5 and 25 litre.



5.3 Concluding Remarks

This case study illustrates that even simple filling lines may cause downtime in a process plant if the surge analysis includes the default scenarios only; i.e. valve closure, pump trip, normal start and stop procedures. It is recommended to include scenarios that are triggered by “generally accepted malfunctioning” of components. The term “generally accepted malfunctioning” highly depends on the application and may require expert judgement from experienced operators. Small leak rates are accepted in many pipe systems. If such a pipe system has a regular start-stop cycle, then the normal start procedure with air pockets in the system should be evaluated.

6 Case Study: Water Hammer in a Vibrating Pipe

Water hammer in a vibrating pipe has been investigated experimentally in the apparatus shown in Figure 8. The apparatus - designed and constructed by Vardy and Fan (1989) at the University of Dundee (UK) - consists of a 4.5 m long, 52 mm diameter, 4 mm thick, closed pipe, filled with pressurised water and suspended on two steel wires. Transient vibration is generated by the axial impact of a 5 m long solid steel rod at one of its ends.

Figure 8. Sketch of single pipe impact test.

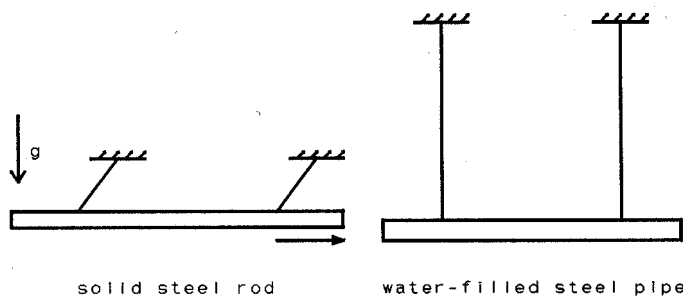
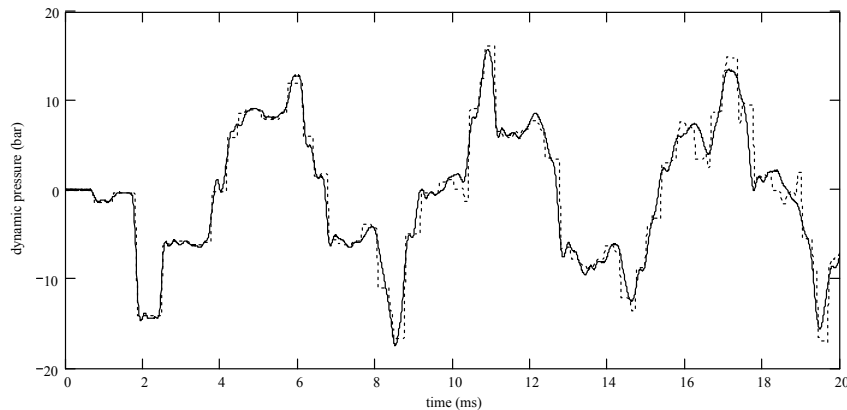


Figure 9. Dynamic pressure in Dundee single pipe experiment. Solid line: measurement. Broken line: calculation.



The pressure measured 3.376 m away from the impact end is shown in Figure 9, together with the prediction by the four-equation model presented in Section 2.3. The agreement between theory and experiment is excellent in view of the complexity of the FSI phenomenon and the imperfections in both the mathematical model and the experimental apparatus. The observed discrepancies are primarily due to the masses (and lengths) of the end caps that close the pipe: these have not been included in the mathematical model. Other things excluded from the present model are: finite rise times, 2D effects, thick-wall theory, damping and friction. Inaccuracy in the input parameters (material and geometrical properties), imperfections in the pipe (additional screw thread, wire suspension points, attached measuring equipment, lateral vibration, plastic deformation of impact point) and limited performance of instrumentation (measurement error, frequency response of amplifiers too low to record nearly-instantaneous rise times) may also contribute to the small discrepancies.

7 Conclusions

Four case studies of typical transient regimes in industrial and laboratory piping systems have been presented. The first case study dealt with the sudden load rejection of a Francis turbine in Plužna HPP (Slovenia). The second case study showed the results of pressure vessel blowdown experiments at UMSICHT's (Germany) pilot plant pipework. The third study reported an incident involving an unanticipated air pocket. The fourth study focussed on FSI in the axial vibration of a water-filled pipe. In all the case studies the power and usefulness of mathematical models was demonstrated. The developed numerical models produced reliable results.

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8 References

- Anderson, A., Arfaie, M., Sandoval-Pena, R., Suwan, K., 1991. *Pipe-friction in water hammer*, XXIV IAHR Congress, Madrid, Spain, D23 - D30.
- Bergant, A., Sijamhodžić, E., 1997. *Water hammer problems related to refurbishment and upgrading of hydraulic machinery*, Hydropower into the Next Century, Portorož, Slovenia, pp. 611 - 622.
- Bergant, A., Simpson, A.R., 1999. *Pipeline column separation flow regimes*, ASCE Journal of Hydraulic Engineering, 125(8), pp. 835 - 848.
- Bergant, A., Simpson, A.R., Vitkovský, J., 2001. *Developments in unsteady pipe flow friction modeling*, IAHR Journal of Hydraulic Research, 39(3), pp. 249 – 257.
- Chaudhry, M.H., 1987. *Applied Hydraulic Transients*, Van Nostrand Reinhold Company, New York, USA.
- Dudlik, A., 1999. *Vergleichende Untersuchungen zur Beschreibung von transienten Strömungsvorgängen in Rohrleitungen*, Dissertation, UMSICHT-Schriftenreihe Band 20, Fraunhofer IRB Verlag, Dortmund, Germany.
- Tijsseling, A.S., 2002. *Exact solution of linear hyperbolic four-equation system in axial liquid-pipe vibration*, In Proceedings of the 5th International Symposium on Fluid-Structure Interactions, Aeroelasticity, Flow-Induced Vibration and Noise (Editor M.P. Paidoussis), New Orleans, USA, November 2002, ASME - AMD, Vol. 253, Paper IMECE2002-32209, pp. 1-12 (CD-ROM, Vol. 3) or ISBN 0-7918-3659-2 (printed book).
- Vardy, A.E., Fan, D., 1989. *Flexural waves in a closed tube*, In Proceedings of the 6th International Conference on Pressure Surges, BHRA, Cambridge, UK, pp. 43-57.
- Wiggert, D.C., Tijsseling, A.S., 2001. *Fluid transients and fluid-structure interaction in flexible liquid-filled piping*, ASME Applied Mechanics Reviews, 54, pp. 455-481.
- Wylie, E.B., Streeter, V.L., 1993. *Fluid Transients in Systems*, Prentice Hall, Englewood Cliffs, USA.
- http://www.umsicht.fhg.de/WWW/UMSICHT/Produkte/SP/pdf/Publikation/pilot_plant_pipework.pdf
- <http://www.win.tue.nl/fsi>
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