Dynamic water behaviour due to one trapped air pocket in a laboratory pipeline apparatus

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Dynamic water behaviour due to one trapped air pocket in a laboratory pipeline apparatus

A Bergant¹, U Karadžić², A Tijsseling³
¹Litostroj Power d.o.o., 1000 Ljubljana, Slovenia
²University of Montenegro, 81000 Podgorica, Montenegro
³Eindhoven University of Technology, 5600 MB Eindhoven, The Netherlands

E-mail: anton.bergant@litostrojpower.eu

Abstract. Trapped air pockets may cause severe operational problems in hydropower and water supply systems. A locally isolated air pocket creates distinct amplitude, shape and timing of pressure pulses. This paper investigates dynamic behaviour of a single trapped air pocket. The air pocket is incorporated as a boundary condition into the discrete gas cavity model (DGCM). DGCM allows small gas cavities to form at computational sections in the method of characteristics (MOC). The growth of the pocket and gas cavities is described by the water hammer compatibility equation(s), the continuity equation for the cavity volume, and the equation of state of an ideal gas. Isentropic behaviour is assumed for the trapped gas pocket and an isothermal bath for small gas cavities. Experimental investigations have been performed in a laboratory pipeline apparatus. The apparatus consists of an upstream end high-pressure tank, a horizontal steel pipeline (total length 55.37 m, inner diameter 18 mm), four valve units positioned along the pipeline including the end points, and a downstream end tank. A trapped air pocket is captured between two ball valves at the downstream end of the pipeline. The transient event is initiated by rapid opening of the upstream end valve; the downstream end valve stays closed during the event. Predicted and measured results for a few typical cases are compared and discussed.

1. Introduction

The control of air (gas) pockets is one of the major operational problems in hydropower plants and pumping systems. Air may be found in water pipelines mainly as stationary pockets or moving bubbles of various sizes [1]. Air pockets can develop in a pipeline by bubble entrainment at inflow locations (headrace tunnel intake structure, pump sump) and by gas release as the water pressure falls (steady or unsteady flow conditions) or the temperature rises. In addition, air valves may contain residual air if the air discharge is not properly controlled [2]. Transport of large pockets of air can occur during filling and emptying of pipelines. Air movement along the pipeline can be slow during filling and the air column can become trapped adjacent to a closed valve or at a high point thus separating two water columns. Homogeneously distributed air bubbles or trapped air pockets in a liquid pipeline system can significantly reduce pressure wave propagation velocity (wave speed) and cause changes in the attenuation, shape and timing of pressure waves. This depends on the amount of the air in bubbles and pockets [3], [4]. The effects of entrapped or entrained air on surge pressures can be either beneficial or detrimental. These effects are dependent on (1) the characteristics of the...
pipeline and its components (pump, turbine, valve), (2) the amount and location of the air, and (3) the nature and cause of the transient events. Water hammer in a pipeline that contains air may create pressure pulses that are either higher than or lower than those that would occur in pure liquid. Burrows and Qiu [5] demonstrated how the presence of air pockets could severely amplify pressure waves during pump shut-down. The most severe pressure rise would occur during the rapid acceleration of a liquid column towards a volume of air that is completely confined [6]. The resulting pressure peak can be several times higher than the Joukowsky pressure if the transient is generated rapidly. On the other hand, a large air cavity may act as an accumulator (air cushion) and thereby suppress the maximum pressure surge because of the increase in storage capacity. Numerical and experimental examples of pressure surges associated with the presence of entrapped air in pipelines have been shown in previous research on this topic [6-11], just to quote only a few excellent publications. This paper investigates dynamic behaviour of a single trapped air pocket in a relatively long laboratory pipeline system (total length 55.37 m); at least much longer than the pipelines that are traditionally presented in the literature (length between 5 to 10 m). The air pocket is captured at the downstream end of the pipeline. It is incorporated as a boundary condition into the discrete gas cavity model (DGCM). An isentropic relation is used for the trapped gas pocket and an isothermal one for small gas cavities distributed along the pipeline. Computed and measured results for two distinct runs are compared and discussed.

2. Theoretical modelling

The classical discrete gas cavity model (DGCM) is based on simplified water hammer equations (the convective transport terms are neglected). The dependent variables pressure $p$ and flow velocity $V$ (cross-sectionally averaged velocity) are traditionally replaced by piezometric head $H$ and discharge $Q$. The DGCM allows gas cavities to form at all computational sections in the MOC (internal and at boundaries). A liquid phase with a constant wave speed $a$ is assumed to occupy the reaches connecting the computational sections. A discrete gas cavity is described by the water hammer compatibility equations, the continuity equation for the gas cavity volume, and the ideal gas equation. Their standard finite-difference form within the staggered grid of the method of characteristics is [12]:

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

$$
H_{i,t} - H_{i-1,t-2\Delta t} + \frac{a}{gA} ((Q_u)_{i,t} - (Q_u)_{i-1,t-2\Delta t}) + \frac{f\Delta x}{2gDA^2} (Q_u)_{i,t} \bigg|_{Q_{i-1,t-2\Delta t}} = 0
$$

(1)

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

$$
H_{i,t} - H_{i+1,t-2\Delta t} - \frac{a}{gA} ((Q_u)_{i,t} - (Q_u)_{i+1,t-2\Delta t}) - \frac{f\Delta x}{2gDA^2} Q_i \bigg|_{Q_{i+1,t-2\Delta t}} = 0
$$

(2)

- continuity equation for the gas cavity volume:

$$(\forall g)_{i,t} = (\forall g)_{i,t-2\Delta t} + (1-\psi) (Q_{i,t-2\Delta t} - (Q_u)_{i,t-2\Delta t}) + \psi (Q_{i,t} - (Q_u)_{i,t}) 2\Delta t
$$

(3)

- ideal gas equation (assuming isothermal conditions):

$$(\forall g)_{i,t} (H_{i,t} - z_i - h_i) = (H_0 - z_0 - h_i) a_g A \Delta x$$

(4)

Section 7 explains the symbols. At a boundary (reservoir, valve) a device-specific equation replaces one of the compatibility equations. The DGCM model can be successfully used for the simulation of
unsteady pure liquid pipe flow by utilizing a low gas void fraction \( \alpha_{g0} \leq 10^{-7} \) and for transient cavitating flow [4], [12]. It should be noted that in the MOC-based DGCM the Courant number is held equal to unity. This condition is difficult to fulfil in complex pipe networks without applying interpolations or without modification of wave speeds and/or pipe lengths.

2.1 Trapped air pocket at closed downstream end valve

In our experimental runs presented in this paper a relatively small trapped air pocket is captured at the downstream end of the pipeline apparatus (at the valve V3/3E in Fig. 1). The selected pipe reach volume \( A_i \Delta x \) is assured to be at least 10 times larger than the effective volume of the trapped air pocket (see Section 3.2 for details). For this case the pipe reach length is assumed to be constant (lumped air pocket model) and the air pressure is equal to the liquid pressure at the computational section (boundary). The transient event is initiated by instantaneous opening of the upstream end valve; the downstream end valve is closed during the event (confined system). The assumption of instantaneous valve opening is valid for the cases with valve opening time less than the occurrence time of the first peak pressure pulse [13]. The corresponding downstream end trapped air pocket boundary condition written in a standard finite-difference form within the staggered grid of the method of characteristics is [3]:

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

\[
H_{i,t} - H_{i-1,i-2\Delta t} + \frac{a}{gA} ((Q_{a,i,t} - Q_{i-1,i-2\Delta t}) + \frac{f\Delta x}{2gDA^2} (Q_{a,i,t} |Q_{i-1,i-2\Delta t}) = 0 \tag{5}
\]

- continuity equation for the trapped air pocket volume:

\[
(\forall_p)_{i,t} = (\forall_p)_{i,t-2\Delta t} - ((Q_{a,i,t})_{i-2\Delta t} + (Q_{a,i,t})_{i,t}) \Delta t \tag{6}
\]

- ideal gas equation (assuming polytropic conditions):

\[
(H_i - z_i - h_i)(\forall_p)_{i,t}^n = (H_0 - z_0 - h_i)(\alpha_{g0}A_i \Delta x)^n \tag{7}
\]

The polytropic exponent \( n \) can take values between 1 (isothermal, traditionally used for small gas cavities in DGCM) and 1.4 (isentropic, used for our simulations of the trapped gas pocket). It is assumed that \( (Q_{a,i,t}) = Q_{a} \) (i.e., \( N+1 \), \( N = \) number of computational reaches). The nonlinear system of equations (5), (6) and (7) is solved numerically by Newton’s method.

2.2 Unsteady friction

Traditionally the steady (or quasi-steady) friction (skin friction) terms are incorporated in the MOC algorithm. This assumption is satisfactory for slow transients where the wall shear stress has a quasi-steady behaviour. Previous investigations using the quasi-steady friction approximation for rapid transients have shown significant discrepancies in attenuation, shape and timing of pressure traces when computational results were compared with measurements [14]. A number of unsteady friction models have been proposed in the literature including one-dimensional (1D) and two-dimensional (2D) models. The MOC based DGCM algorithm in this paper incorporates a convolution-based (with quasi-2-D weighting function) unsteady friction model [15]. The unsteady frictional head loss per unit length \( h_f \) is formulated by the following term:
where "∗" represents convolution. The subscript '0' denotes parameter values based on the steady-state condition preceding the transient event. The Darcy-Weisbach relation (first term) defines the steady-state component and the unsteady component (second term) is defined by the convolution of a weighting function with past accelerations (∂Q/∂t). The weighting function W₀ is based on an assumed steady-state viscosity distribution preceding the transient event, which is kept constant during the event (the “frozen viscosity” assumption). In our case studies, the liquid is at rest initially; therefore, Zielke’s [15] weighting function for transient laminar flow is used in the simulations. An accurate and efficient Vítkovský et al. approximate scheme [16] that does not require convolution with the complete history of velocities required at each time step is used in this paper.

3. Laboratory pipeline apparatus

A small-scale multi-purpose pipeline apparatus has been designed and constructed at the Faculty of Mechanical Engineering, University of Montenegro, for investigating rapid water hammer events including column separation and fluid-structure interaction. Recently the apparatus has been modified for performing pipeline filling and emptying events that are characterized both by rapid and gradual pressure changes [17]. The apparatus is comprised of a horizontal pipeline that connects the upstream end high-pressurized tank to the outflow tank (steel pipe of total length $L = 55.37$ m; internal diameter $D = 18$ mm; pipe wall thickness $e = 2.0$ mm; maximum allowable pressure in the pipeline $p_{\text{max, all}} = 25$ MPa) – see Fig. 1. The dynamic response of the liquid column due to a trapped air pocket in this apparatus should be similar for both small and large pipelines with similar scalings.

Four valve units are positioned along the pipeline including the end points. The valve units at the upstream end tank (position 0/3) and at the two equidistant positions along the pipeline (positions 1/3 and 2/3) consist of two hand-operated ball valves (valves $Vi/3U$ and $Vi/3D$; $i = 0, 1, 2$) that are connected to the intermediate pressure transducer block. A T-section with an on/off air inlet valve (V0/3A) is installed between the upstream end valve unit (position 0/3) and the high-pressurized tank to facilitate pipeline emptying tests. The horizontal pipe upstream end service valve (V0/3SH) is installed between the T-section and the high-pressurized tank in order to isolate upstream end tank during emptying tests. There are four 90° bends along the pipeline with radius $R = 3D$. The pipeline is anchored against axial movement at 37 points (as close as possible to the valve units and bends). The anchors are loosed for fluid-structure interaction tests. The air pressure in the upstream end tank (total volume $V_{\text{HPT}} = 2$ m³; maximum allowable pressure in the tank $p_{\text{HPT max, all}} = 2.2$ MPa) can be adjusted up to 800 kPa. Pressure in the tank is kept constant during each experimental run by using a high precision fast-acting air pressure regulator (precision class: 0.2%) in the compressed air supply line.

3.1 Instrumentation

Four dynamic high-frequency pressure transducers are positioned within the valve units along the pipeline including the end points (see Fig. 1). Pressures $p_{0/3}, p_{1/3}, p_{2/3}$ and $p_{3/3}$ are measured by Dytran 2300V4 high-frequency piezoelectric absolute pressure transducers (pressure range: from 0 to 6.9 MPa; resonant frequency: 500 kHz; acceleration compensated; discharge time constant: 10 seconds (fixed)). All four piezoelectric transducers were flush mounted to the inner pipe wall. These transducers perform accurately for rapid water hammer events including column separation [18]. In these events the water hammer pressure pulse in the apparatus (Fig. 1) lasts about or less than the wave reflection time $2L/a = 0.08$ seconds, which is less than 10/100 of a second event during which the sensor will discharge 1% of its voltage. The Endress+Hauser PMP131 strain-gauge pressure
A pressure transducer has been installed at the control valve V3/3C (pressure \( p_{3/3C} \); pressure range: from 0 to 1 MPa) to measure (1) initial pressure in the confined space between the valves V3/3E and V3/3H (or V3/3P), and (2) transient response of a trapped air pocket after rapid opening of either V3/3H or V3/3P (see Section 3.2 for details). In addition, this transducer does not discharge its voltage (analogue output); consequently, it is used to control ‘slow’ events (low frequency) measured by piezoelectric transducers. The datum level for all pressures measured in the pipeline and at the tank is at the top of the horizontal steel pipe (elevation 0.0 m in Fig. 1). The water temperature is continuously monitored by the thermometer installed in the outflow tank. The water hammer wave speed was determined as \( a = 1340 \pm 10 \) m/s.

3.2 Test procedure for dynamic response of trapped air pocket

The test procedure for measuring the dynamic response of the trapped air pocket at the downstream end valve (V3/3E – see Fig. 1) is as follows. The pressure in the upstream end high-pressurized tank is adjusted to a desired value using a high precision air pressure regulator. The control needle valve (V3/3C) is fully opened. The upstream end valve (V0/3U) at the pressurized tank (position 0/3 in Fig. 1) is closed. All other valves of the four valve units are fully opened. The air inlet valve (V0/3A) is closed (isolation of the compressed air supply into the pipeline), and the downstream end emptying valve V3/3E is opened. The filling of the initially empty pipeline is initiated by quickly opening valve V0/3U. When a steady state is achieved, the downstream end valve (either V3/3P or V3/3H) is closed as fast as possible. After complete valve closure, a large amount of water is flushed into the outflow tank. The pressure downstream the closing valve drops to the atmospheric pressure and upstream the valve to the reservoir pressure. Then the emptying valve V3/3E is closed and the system
is ready for experiment. The trapped air pocket run is initiated by rapid opening of the hand operated valve V3/3H (or V3/3P). It should be noted that the space between the valves V3/3E and V3/3H (or V3/3P) is occupied by air and residual water, both at atmospheric pressure. The initial effective trapped gas volume is predicted by a trial and error method based on comparison between the measured and computed first pressure peak at the valve.

4. Numerical and experimental results

The following two distinct experimental runs in the test apparatus (Fig. 1) are presented and analyzed theoretically in this section: (1) the first case study investigates the influence of a trapped air pocket at the downstream end valve on pressure wave traces (amplitude, shape, timing) in a low-head system and (2) the second case study investigates this influence in a medium-head system. Numerical results from the DGCM (see Section 2) are compared with results of laboratory measurements at the downstream end valve (pressures \( p_{3/3} \) and \( p_{3/3-1/2} \) in Fig. 1) and along the pipeline (pressures \( p_{2/3} \) and \( p_{1/3} \) in Fig. 1). The effect of unsteady friction is included in the simulations (Zielke’s model using weighting function for transient laminar flow [15]). Computational and measured results are presented for the case with initial static pressure head in the upstream end pressurized tank \( H_{\text{HTP}} = 21 \text{ m (low-head test case)} \) and \( H_{\text{HTP}} = 52 \text{ m (medium-head test case)}. \) The estimated initial trapped air pocket volume at atmospheric conditions for the two cases is about \( \sqrt[3]{\alpha_0} = 15 \text{ cm}^3 \left(1.5 \times 10^{-5} \text{ m}^3\right). \) This air volume is very small in comparison to the total water volume of 13.6 litres (0.0136 m\(^3\)). The number of pipe reaches for all computational runs is \( N = 12 \) and the time step is \( \Delta t = 0.0033 \text{ s}. \) The corresponding DGCM void fraction at the downstream end closed valve (trapped air pocket) is in the order of \( \alpha_{\text{void}} = 10^{-3} \) (\( \alpha_{\text{void}} = \sqrt[3]{\alpha_0}/\sqrt[3]{\alpha_{\text{reach}}} \) and much larger than the void fractions of \( \alpha_{\text{void}} = 10^{-7} \) at the other 11 computational sections (except \( 0.5 \times 10^{-7} \) at the upstream end reservoir). Simulations using the isentropic relation for the trapped gas pocket and the isothermal one for small gas cavities produce the best fit with measured results for the two considered case studies. In addition, simulations with larger numbers of pipe reaches (24, 48) produce practically the same results (robustness of the DGCM). Water and surrounding temperatures were about 25 and 30 °C, respectively.

4.1 Low-head test case

First the computational and measured results are presented for the case with an initial static head in the upstream end pressurized tank of \( H_{\text{HTP}} = 21 \text{ m (low-head test case)} \) and an estimated initial trapped air pocket volume at atmospheric conditions of \( \sqrt[3]{\alpha_0} = 16 \text{ cm}^3 \left(1.6 \times 10^{-5} \text{ m}^3\right). \) Figure 2 shows: (1) variation of air volume \( \sqrt[3]{\alpha} \) (VAIR in Fig. 2a), (2) liquid flow velocity at the water/lumped air pocket interface \( \sqrt[3]{\alpha} \) (Fig. 2b), (3) absolute heads in the space of air pocket position and upstream end of the valve V3/3P \( H^*_{\alpha} \) and \( H^*_{3/3}; \) \( H^* \) = absolute head herein; \( H^* = H - z - h_i \) in Eqs. (4) in (7)), respectively (Fig. 2c), and (4) computed and measured absolute heads at the valve V3/3P \( H^*_{3/3} \) (Fig. 2d). The effective valve opening time of \( t_{\text{off}} = 0.028 \text{ s} \) is significantly shorter than the water hammer wave reflection time of \( 2L/a = 0.056 \text{ s} \) in the classical water hammer sense. The effective valve opening time is also much shorter than the time of occurrence of the maximum (peak) head at the air pocket interface \( t = 0.25 \text{ s} \) (Fig. 2d). The assumption of instantaneous valve opening used in DGCM simulations is justified [13]. The computed and measured heads at V3/3P (Fig. 4d) agree reasonably well. There is a good match between maximum head peaks and pressure wave timing in the early phase of the transient event. There are some discrepancies in timing and attenuation at later times. The discrepancies may be contributed to additional head losses at the control needle valve (not accounted in the simulations) and possible air pocket separation and transfer of some air bubbles with reverse flow into the initial pure liquid zone. The same degree of matching between measured and computed results is seen for heads along the pipelines (\( H^*_{2/3} \) and \( H^*_{3/3} \) in Figs. 3a and 3b, respectively). The pressure stays above the liquid vapour pressure at all times. The pressure traces exhibit reservoir wave reflections off the trapped air pocket and explain the occurrence of ‘high-frequency’ velocity peaks in Fig. 2b in contrast to ‘low-frequency’ bulk velocity oscillations. As a conclusion, air pressure and liquid-pressure pulsations at the closed end \( \left(H^*_{\alpha} \right) \) and \( H^*_{3/3} \) in Fig. 2c) conform classical pulsations found in the literature [6-11].
∀g (VAIR), liquid flow velocity (V3/3), absolute air and liquid heads at the
downstream dead end (H*air and H*3/3 (measured and calculated)): H_{HTP} = 21 m; ∀g0 = 16 cm³.

Fig. 3 Comparison of absolute heads along the pipeline (H*2/3 and H*1/3): H_{HTP} = 21 m; ∀g0 = 16 cm³.

4.2 Medium-head test case
This section deals with the more striking medium-head test case that in its nature departs from the classical low-head test case presented in Section 4.1. The computed and measured results are presented for the case with initial static head in the upstream end pressurized tank H_{HTP} = 52 m and an estimated initial trapped air pocket volume at atmospheric conditions ∀g0 = 13 cm³ (1.3 × 10⁻⁵ m³). Figure 4 shows: (1) variation of air volume ∀g (VAIR in Fig. 4a), (2) liquid flow velocity at the water/lumped air pocket interface V3/3 (Fig. 4b), (3) absolute heads in the space of air pocket position and upstream end of the valve V3/3P (H*air and H*3/3, respectively (Fig. 4c), and (4) computed and measured absolute heads at the valve V3/3P (H*3/3 in Fig. 4d). In this case, the effective valve opening time of t_{ef} = 0.015 s is again significantly shorter than the water hammer wave reflection time of 2L/a = 0.056 s in the classical water hammer sense. The effective valve opening time is also much shorter than the time of occurrence of the maximum (peak) head at the air pocket interface t = 0.15 s in the classical sense as in Fig. 2d. However, the maximum (peak) head for the medium-head test case occurs as a kind of ‘short duration’ pressure
pulse at time $t = 0.0175$ s (Fig. 4d). This peak pressure pulse is due to the superposition of the bulk head wave with the reservoir-reflected wave. Again, the assumption of instantaneous valve opening used in DGCM simulations is justified. The computed and measured heads at V3/3P (Fig. 4d) agree reasonably well. There is a good match between maximum head peaks and pressure wave timing in the early phase of the transient event. There are some discrepancies in timing and attenuation at later times. The magnitude of the discrepancies is not the same as for the low-head test case in Section 4.1 due to the different operating conditions. There is the same degree of matching between measured and computed results for heads along the pipelines ($H^{*\,2/3}$ and $H^{*\,1/3}$ in Figs. 5a and 5b, respectively). The pressure traces exhibit reservoir wave reflections off the trapped air pocket and explain the occurrence of ‘high-frequency’ velocity peaks in Fig. 2b in contrast to ‘low-frequency’ bulk velocity oscillations. Moreover, careful investigation of computed head traces along the pipeline between the positions 3/3 and 2/3 (Fig. 1) and the measured head at position 2/3 (absolute head $H^{*\,2/3}$ in Fig. 5a) reveals a distributed vaporous cavitation zone that is condensed back to the liquid phase by the reservoir-reflected wave. The whole phenomenon occurs within the time period between $t = 0.22$ and 0.28 seconds.

Fig.4 Variation of air volume $\forall_{g}(VAIR)$, liquid flow velocity ($V_{3/3}$), absolute air and liquid heads at the downstream dead end ($H^{*\,\text{air}}$ and $H^{*\,3/3}$ (measured and calculated)): $H_{HTP} = 52$ m; $\forall_{g0} = 13$ cm$^3$.

Fig.5 Comparison of absolute heads along the pipeline ($H^{*\,2/3}$ and $H^{*\,1/3}$): $H_{HTP} = 52$ m; $\forall_{g0} = 13$ cm$^3$. 
5. Conclusions
This paper investigates the dynamic behaviour of a single air pocket trapped at the downstream dead end in a relatively long laboratory pipeline system; five to ten times longer than the pipelines that are traditionally presented in the literature. The relatively small air pocket is incorporated as a lumped boundary condition in the discrete gas cavity model (DGCM). An isentropic relation is used for the trapped gas pocket and an isothermal one for small gas cavities distributed along the pipeline. Two distinct experimental runs in the test apparatus are presented and analyzed theoretically: (1) the first case study investigates the influence of the trapped air pocket at the downstream end valve on pressure wave traces (amplitude, shape, timing) in a low-head system and (2) the second case study investigates this influence in a medium-head system. The medium-head test case in its nature departs from the classical low-head test case that is reported in a number of published research papers. Apart from a trapped air pocket at the dead end, there appears a distributed vaporous cavitation zone along the pipeline that is induced by the negative pressure wave that travels towards the upstream end reservoir. The cavitation zone is condensed back to the liquid phase by the wave coming from the reservoir. Computed and measured results for two distinct runs agree reasonably well in the early phase of transient event; but there are some discrepancies in timing and attenuation at later times. The discrepancies may be contributed to additional head losses at the control needle valve and possible air pocket separation and transfer of some air bubbles with reverse flow into the initial pure liquid zone.

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7. List of symbols

- **A**: pipe area
- **a**: wave speed
- **C**<sup>+</sup>, **C**<sup>-</sup>: label of characteristic equations
- **D**: pipe diameter
- **e**: pipe-wall thickness
- **f**: friction factor
- **g**: gravitational acceleration
- **H**: piezometric head (head)
- **H**<sup>*</sup>: absolute head
- **h**<sub>f</sub>: head loss per unit length
- **h**<sub>v</sub>: vapour pressure head
- **L**: length
- **N**: number of reaches
- **n**: polytropic coefficient
- **P**: pressure
- **Q**: discharge
- **Q**<sub>u</sub>: upstream-end discharge
- **R**: pipe radius
- **t**: time
- **t**<sub>eff</sub>: effective valve closure time
- **V**: average flow velocity
- **x**: distance along pipe
- **z**: elevation
- **Δt**: time step
- **Δx**: space step
- **ν**: kinematic viscosity
- **ψ**: weighting factor in DGCM
- **∀**: volume
- **max**: maximum
- **reach**: reach
- **0**: initial conditions
- **i**: node number

**Abbreviations:**
- **DGCM**: discrete gas cavity model
- **MOC**: method of characteristics

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